



**International Conference:
Mechanics, Earthquake Engineering,
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**Международная конференция:
Механика, Сейсмостойкость,
Машиностроение**

***«Ensuring seismic safety and
seismic stability of buildings and structures,
applied problems of mechanics»***

dedicated to the 90th anniversary of Academician T.R. Rashidov

***«Обеспечение сейсмической безопасности и
сейсмостойкости зданий и сооружений,
прикладные задачи механики»***

посвященная 90-летию академика АН РУз Т.Р. Рашидова

III-ТОМ

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**СБОРНИК МАТЕРИАЛОВ МЕЖДУНАРОДНОЙ КОНФЕРЕНЦИИ:
МЕХАНИКА, СЕЙСМОСТОЙКОСТЬ, МАШИНОСТРОЕНИЕ**

ПО ПРОБЛЕМАМ

**ОБЕСПЕЧЕНИЯ СЕЙСМИЧЕСКОЙ БЕЗОПАСНОСТИ И
СЕЙСМОСТОЙКОСТИ ЗДАНИЙ И СООРУЖЕНИЙ,
ПРИКЛАДНЫЕ ЗАДАЧИ МЕХАНИКИ**

**посвященная 90-летию академика АН РУз Т.Р. Рашидова
27-29 мая 2024 года**

III – ТОМ

Ташкент

Сборник материалов международной научной конференции: Механика, Сейсмостойкость, Машиностроение по проблемам «Обеспечения сейсмической безопасности и сейсмостойкости зданий и сооружений, прикладные задачи механики», посвященной 90-летию академика АН РУз Т.Р. Рашидова.

Сборник материалов состоит из 3х томов:

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The collection of materials consists of 3 volumes:

- Volume I contains materials of sections 1 "Seismic resistance of buildings and structures" and
2 "Engineering seismology and seismic risk";
Volume II contains materials of section 3 "Mechanics of deformable rigid bodies, geotechnics";
Volume III contains materials of sections 4 "Mechanics of liquids, gases, and multiphase media" and
5 "Dynamics and strength of machines."

The proceedings of the International Conference of present scientific reports from research institutions and universities of Uzbekistan, Kazakhstan, Kyrgyzstan, the USA, Russia, and Tajikistan.

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Закиров Батир Иркинович

Министр строительства и жилищно-коммунального хозяйства Республики Узбекистан

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Министр по чрезвычайным ситуациям Республики Узбекистан

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Султанов Карим Султанович (*сопредседатель*)

доктор физико-математических наук, профессор, директор Института механики и сейсмостойкости сооружений АН РУз. Ташкент, Узбекистан

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доктор технических наук, профессор, Санкт-Петербургский политехнический университет Петра Великого. Санкт-Петербург, Россия

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Бахадиров Г.А. – профессор, Институт механики и сейсмостойкости сооружений АН РУз. Ташкент, Узбекистан.

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ПРОГРАММНЫЙ КОМИТЕТ КОНФЕРЕНЦИИ

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СЕКРЕТАРИАТ

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Умархонов С.И.	к.т.н.	Ташкент
Хамдамов М.	PhD.	Ташкент

Section D - Mechanics of fluids, gas, and multiphase media

Секция D – Механика жидкости, газа и многофазных сред

SIMULATION OF A DAMPER CONNECTED TO THE END OF A PIPELINE WITHIN A REAL FLUID AT A GIVEN INLET FLUID FLOW

Bakhtiyorov B.¹

¹Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan

E-mail: bozorov.bobur1992@gmail.com

Abstract. *The problem of the state of an elementary section of a pipeline with a variable slope is considered. At the entrance to the section, the mass flow rate of the liquid is set, and at the exit there is an air cap with a certain amount of air. Modeling of the process of propagation of compaction and rarefaction waves in an elementary section of a pipeline was carried out according to the quasi-one-dimensional nonlinear model of N.E. Zhukovsky. A new version of the boundary condition with an air cap is proposed, which takes into account the real properties of liquid and air. The equation of state of the liquid is compiled taking into account the volumetric compressibility of the liquid, and the equation of state of air is compiled taking into account the compressibility coefficient, which for air is 1 in the range of 0.1-15.0 MPa with an accuracy of 99%. An effective algorithm for implementing a new nonlinear boundary condition has been developed, taking into account the constant volume of air in the air cap and its compressibility.*

Keywords. *quasi-one-dimensional equations N.E. Zhukovsky, real fluid, air compressibility, counter traveling waves, air cap, finite difference method, Newton's tangent method.*

1. Introduction

Pipeline transport of various media is an economically and environmentally feasible mode of transport. It is especially developed in post-Soviet countries, as it is distinguished by its reliability and controllability in terms of intended use. Pipeline networks for the transportation of gases and liquids continue to develop and expand. There is a demand in the global market for pipeline networks, accordingly, research should continue to reduce energy intensity, increase the productivity and reliability of networks, taking into account the technological requirements in application areas.

From the hydrodynamics course it is known that the values of momentum, total pressure and energy of the medium are conserved when passing through a shock wave [1,2]. In this case, the density, speed, temperature and pressure of the medium undergo abrupt changes. Accordingly, the concept of «compression wave» can be used in the case of a decrease in the density of the medium when passing through a shock wave in the direction of its propagation. From this point of view, in «discharge waves» the density of the unperturbed will be greater than after the shock wave. By instantly blocking the cross-section of a functioning pipeline, for example, it is possible to organize: compaction waves before the overlap section and rarefaction waves after the overlap cross-section. This is an artificial way of generating a shock wave. The formation of a shock wave during the propagation of a sinusoidal disturbance of the density of a medium is described in the literature [3-4].

In technology, technological processes and in nature, a shock wave can be a positive or negative factor. In particular, in the processes of delivering liquid or gas, as well as transporting a thermal agent to the consumer surges in impulse and pressure are undesirable factors. Both the consumer and the installations themselves suffer from them [5]. If the pipeline is used for the purpose of transmitting mechanical energy [6], then surges are likely to serve the purpose.

The features of shock waves are widely studied in the scientific literature [1-6]. But it has its own distinctive features during pipeline transportation of low- and super-compressible media, which are connected by the presence of two speed scales (hydrodynamic speed and the speed of propagation of small disturbances), dissipation of kinetic energy due to friction, deformation of both the transposed medium and the pipeline wall, and others factors.

The equations used in this case are nonlinear, the solutions of which were obtained with significant simplifications [7-11]. In rare cases, the variability of the temperature of the transported medium and the influence of this factor on the stress-strain state of the medium and the pipeline under conditions of their direct contact are taken into account. Often, equations are linearized using auxiliary functions [3-4,12], introducing mass flow [7,11,13], as well as averaging individual factors of nonlinear terms [11,14].

In this paper, we propose a mathematical model of a pipeline, at the inlet of which the law of change in the mass flow rate of liquid over time is specified, and at the outlet an air cap is installed. Unlike well-known works, the work implements a liquid model taking into account the volumetric compression coefficient under the influence of variable pressure, and air is represented within the framework of a real gas taking into account the compressibility coefficient.

With the introduction of dimensionless variables and an auxiliary function, the equations are presented in the form of nonlinear equations of analogues of counterpropagating traveling waves, which are solved numerically. Below we propose numerical methods for solving traveling wave equations and implementing nonlinear boundary conditions at the inlet and outlet of an elementary pipeline section in the form of specifying mass flow and an air cap.

2. Formulation of the problem.

The state of an elementary section of a pipeline with length l , diameter D and resistance coefficient is studied λ , to the end of which an air cap (AC) is connected, and at the inlet the law of change in the mass flow rate of liquid over time is specified.

The isothermal state of the liquid in the pipeline is described by quasi-one-dimensional equations of conservation of momentum and mass [15,16]:

$$\begin{cases} \rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} + \frac{\partial p}{\partial x} = -\frac{\lambda}{2D} \rho u |u| - \rho g \sin \alpha, \\ -\frac{\partial \rho}{\partial t} = \frac{\partial \rho u}{\partial x}. \end{cases} \quad (1)$$

Here and below $\rho(x,t)$, $u(x,t)$, $p(x,t)$ are the average values of density, velocity and pressure of the fluid in the section x at time t . Otherwise, the usual designations for pipeline transportation are used [15,16].

The equation of state of the liquid is adopted taking into account the coefficient of volumetric compression k of the liquid:

$$p - p_* = \frac{k}{\rho_*} (\rho - \rho_*) \quad (2)$$

where $p_* = 101235 \text{ Pa}$ and $\rho_* = 998.15 \text{ kg/m}^3$ at $T_0 = 293.15 \text{ K}$; the propagation speed of small pressure disturbances in a liquid is defined as $c = \sqrt{k / \rho_*}$ [15].

The distributions of pressure and mass flow of liquid in a stationary mode were taken as the initial conditions

$$M(x, 0) = f \rho(x, 0) u(x, 0) = M^0 = \text{const}. \quad (3)$$

(Hereinafter $f = \pi D^2 / 4$ – the cross-sectional area of the pipeline.)

A compressor (air blower) is usually connected to the air caps, which are installed on top of the water pipes of pumping stations. After a certain period (with a period of, for example, three months), the compressor is started to pump air into the air cap to the required mass. Those. In our problem, the mass of air m_g in the air cap can be considered constant. Taking z , R_g , T_0 constant values, we determine the mass of air in the air cap

$$m_g = V_g(t) \rho_g(t) = \text{const}. \quad (4)$$

From here we find

$$V_g(t) = \frac{m_g}{\rho_g(t)} = \frac{c_g^2 m_g}{p(l,t)} = \frac{c_g^2 m_g}{p_* - k + c^2 \rho(l,t)}, \quad (5)$$

Where $c_g^2 = zR_g T_0$. Here it was taken into account that the pressure value p relates both to the cross-section $x = l$ of the pipeline and to the volume of the air cap. This is justified by the fact that, firstly, the distance between the pipeline and the air cap is quite small, and secondly, the propagation speeds of small pressure disturbances in the liquid and air are quite large.

Let us express the density of the gas in terms of the density of the liquid at a given pressure:

$$\rho_g(t) = \frac{p(l,t)}{zR_g T_0} = \frac{p_* - k + c^2 \rho(l,t)}{zR_g T_0}. \quad (6)$$

In this regard, the derivative of air density with respect to time is defined as

$$\frac{d\rho_g(t)}{dt} = \frac{c^2}{zR_g T_0} \frac{\partial \rho(l,t)}{\partial t}. \quad (7)$$

Substituting the obtained dependencies into the boundary condition gives

$$f \rho(l,t) u(l,t) = Q_l(t) + \frac{c_g^2 m_g}{p_* - k + c^2 \rho(l,t)} \frac{c^2}{zR_g T_0} \frac{\partial \rho(l,t)}{\partial t}. \quad (8)$$

When taking into account the dependence, $c_g^2 = zR_g T_0$ this condition is simplified, and the boundary condition at $x = l$ takes the final form:

$$f \rho(l,t) u(l,t) = Q_l(t) + \frac{m_g}{\frac{p_* - k}{c^2} + \rho(l,t)} \frac{\partial \rho(l,t)}{\partial t}. \quad (9)$$

The question remains about determining the value of the z air compressibility coefficient at known values p , ρ_g and T_0 .

Graphs of the gas compressibility coefficient at fixed values T_0 for a certain pressure range are complex: first it decreases and then increases [16,17,18]. In engineering practice, the method for determining the value of the compressibility coefficient has evolved from tabular and (or) nomogram (Brown- Katz) representations to analytical methods [17]. Various approximation formulas have been proposed in the form of polynomials, piecewise variables and exponential functions. More commonly used for this purpose today are the Redlich – Kwong and Peng –Robinson dependencies, which clarify the original source, the van der Waals equation, and form a cubic equation regarding the compressibility coefficient [16,19]. Unfortunately, the first of them (Redlich – Kwong equation) refers only to the gas state of a substance (it cannot be used when the aggregate state of a substance changes), and the second requires the value of the acentric factor, i.e. additional, hard-to-get information is needed. In this regard, we turned to the results of experimental studies [20].

Table 1.

Experimentally determined values of the compressibility coefficient air z at 300 K depending on pressure (atm) [20]

1	5	10	20	40	60	80	100	150	200	250	300	400	500
0.9999	0.9987	0.9974	0.9950	0.9917	0.9901	0.9903	0.9930	1.0074	1.0326	1.0669	1.1089	1.2073	1.3163

The data presented above and other data from this and other sources showed that at pressure values from 1 atm to 150 atm and temperatures close to normal, the value of the air compressibility coefficient

with a maximum deviation of 1% can be taken equal to 1. This representation z greatly simplifies the solution process air cap problems.

For comparison, we present a version of the condition that was obtained by I.A. Charny for an incompressible liquid and using the state of an ideal gas [8]:

$$w(l, t) + \frac{\rho c^2 V_0}{p_0 f} \frac{\partial w(l, t)}{\partial x} = \frac{Q_l(t)}{f}, \quad (10)$$

where V_0 and p_0 are constants – the values of air volume and pressure in the undisturbed (initial) state; a $Q_l(t)$ is the output volumetric flow rate.

At initial pressure $p(l, 0)$ and temperature, T_0 the mass 1 $\mathcal{M}^3$ of air is

$$m_{gm^3} = \frac{p(l, 0)}{R_g T_0} (\kappa z). \quad (11)$$

In accordance with this, with the volume of air in the air chamber $V_g(0)$ and the pressure value $p(l, 0)$ in the cross section $x = l$ at the initial moment we find the air masses in the VC

$$m_g = \frac{p(l, 0) V_g(0)}{z R_g T_0}. \quad (12)$$

Approximation of the input condition at a given mass flow. To solve the problem, the initial equations are reduced to dimensionless form and presented as analogues of the equations of counterpropagating waves. When solving the problem numerically, the algorithm presented in [12] was used. In the new version, the numerical algorithm takes into account: the real properties of liquid and gas and upper relaxation in the numerical scheme. Below we present methods for solving nonlinear boundary conditions specified according to the problem statement.

We write the condition at the entrance to the section in dimensionless form:

$$\bar{u}(0, \bar{t}) \bar{\rho}(0, \bar{t}) = \frac{Q_0(\bar{t})}{f c \rho_*} = \bar{Q}_0(\bar{t}). \quad (13)$$

In this case, the value is considered known f_{20}^{n+1} .

If $\bar{Q}_0(\bar{t}) = 0$, then from condition (13) it is uniquely determined $\bar{u}_0(\bar{t}) = 0$ and then calculate $\varphi_0^{n+1} = -f_{20}^{n+1}$ and $f_{10}^{n+1} = \varphi_0^{n+1}$. Otherwise, the nonlinear equation – representation of the condition – is solved numerically.

Because $\bar{\rho} = e^{\frac{f_1 - f_2}{2}}$, then $\bar{u}_0 e^{\frac{f_1 - f_2}{2}} = \bar{Q}_0(\bar{t})$. Here $f_1 = 2\bar{u}_0 - f_2$ and $\frac{f_1 - f_2}{2} = \bar{u}_0 - f_2$. Therefore,

when notated, $\bar{u} = \bar{u}_0^{n+1}$ the condition takes the form:

$$\bar{u} e^{\bar{u}} = \bar{Q}_0^{n+1} e^{f_{20}^{n+1}}. \quad (14)$$

Let's create an auxiliary function

$$F(\bar{u}) = \bar{Q}_0^{n+1}(\bar{t}) e^{f_{20}^{n+1}} - \bar{u} e^{\bar{u}}. \quad (15)$$

Since $F'(\bar{u}) = -(\bar{u} + 1)e^{\bar{u}} < 0$, the function $F(\bar{u})$ in the region of interest to us $0 \leq \bar{u} \leq 1$ is monotonically decreasing. It is not difficult to verify that when $\bar{u} = 0$ the function has a positive value, and at $\bar{u} = 1$ – a negative value. Then, according to Cauchy's theorem on continuous functions, in the interval $0 \leq \bar{u} \leq 1$ the function has a single zero value.

To solve the equation, a successive approximation based on Newton's tangent method is used. Given the convexity of the function graph (the second derivative has a negative value $-F''(\bar{u}) = -(\bar{u} + 2)e^{\bar{u}} < 0$), it should start from the zero approximation $\bar{u}^0 = 1$.

In k the th iteration step through the point $(\bar{u}_0^k, F(\bar{u}_0^k))$ let's make a tangent

$$\frac{F(\bar{u}^{k+1}) - F(\bar{u}^k)}{\bar{u}^{k+1} - \bar{u}^k} = F'(\bar{u}^k). \quad (16)$$

At a point, $(\bar{u}^{k+1}, 0)$ this tangent intersects the axes $\bar{u}$ (abscissa). Accordingly $F(\bar{u}^{k+1}) = 0$, the abscissa of the intersection point is

$$\bar{u}^{k+1} = \bar{u}^k - \frac{F(\bar{u}^k)}{F'(\bar{u}^k)}. \quad (17)$$

From here the recurrent formula is formed

$$\bar{u}^{k+1} = \bar{u}^k + \frac{\bar{Q}_0^{n+1} e^{f_{20}^{n+1} - \bar{u}^k} - \bar{u}^k}{1 + \bar{u}^k}. \quad (18)$$

Since $|\bar{u}| \ll 1$, then in the calculations there is no problem associated with division by zero, and at an approximate value of the input dimensionless speed $|F(\bar{u}^{k+1})| < 10^{-12}$ is obtained.

Since the meaning f_{20}^{n+1} and the end result $\bar{u}_0^{n+1} = \bar{u}^{k+1}$ are known, then we calculate

$$f_{10}^{n+1} = 2\bar{u}^{k+1} - f_{20}^{n+1}, \quad (19)$$

$$\varphi_0^{n+1} = \ln(\bar{Q}_0^{n+1} / \bar{u}^{k+1}). \quad (20)$$

Let us present typical approximation results for the input speed when moving from 3 to 4 m/s ($n = 100$).

Table 2.

Results of the iterative process during implementation input boundary condition for mass flow

$\bar{u}^k$	$F(\bar{u}^k)$	$\bar{u}^{k+1}$
1	-0.998991697632788	0.500504151183606
0.500504151183606	-0.498842579518003	0.168054168206543
0.168054168206543	-0.165737305650419	0.0261623705813558
0.0261623705813558	-0.0234922978251417	0.00326901715247048
0.00326901715247048	0.000537112406689999	0.00273365485431593
0.00273365485431593	-2.87158163117876e-7	0.00273336847900407
0.00273336847900407	-8.212267671448e-14	0.00273336847892218

Approximation of the output condition when installing an air cap

Let us present the condition in dimensionless form:

$$\bar{\rho}(1, \bar{t}) \bar{u}(1, \bar{t}) = \frac{Q_l(\bar{t})}{f \rho_* c} + \frac{m_g}{f l \left(\frac{p_* - k}{c^2} + \rho_* \bar{\rho}(1, \bar{t}) \right)} \frac{\partial \bar{\rho}(1, \bar{t})}{\partial \bar{t}} \quad (21)$$

and in discrete form:

$$\bar{\rho}_{N_x}^{n+1} \bar{u}_{N_x}^{n+1} = \frac{Q_l^{n+1}}{f \rho_* c} + \frac{m_g}{f l \left(\frac{p_* - k}{c^2} + \rho_* \bar{\rho}_{N_x}^n \right)} \frac{\bar{\rho}_{N_x}^{n+1} - \bar{\rho}_{N_x}^n}{\tau}. \quad (22)$$

When introducing the notation

$$s = \frac{m_g}{\tau f l \left(\frac{p_* - k}{c^2} + \rho_* \bar{\rho}_{N_x}^n \right)} (> 0), \quad Q_r = \frac{Q_l^{n+1}}{f \rho_* c} - s \bar{\rho}_{N_x}^n \quad (23)$$

we arrive at the nonlinear equation

$$\bar{\rho}_{N_x}^{n+1} \bar{u}_{N_x}^{n+1} - s \bar{\rho}_{N_x}^{n+1} = Q_r. \quad (24)$$

It was believed that when counting down, x the value of $f_{1N_x}^{n+1}$. Therefore, we choose the condition as the unknown quantity $\bar{u} = \bar{u}_{N_x}^{n+1}$, and express the density in terms of the remaining quantities:

$$\bar{\rho}_{N_x}^{n+1} = e^{\frac{f_{1N_x}^{n+1} - f_{2N_x}^{n+1}}{2}} = e^{\frac{f_{1N_x}^{n+1} - (2\bar{u} - f_{1N_x}^{n+1})}{2}} = e^{f_{1N_x}^{n+1} - \bar{u}}. \quad (25)$$

Substituting this expression and transforming the resulting equality gives:

$$e^{f_{1N_x}^{n+1} - \bar{u}} (\bar{u} - s) = Q_r. \quad (26)$$

When solving the transcendental equation

$$F(u) = e^{-f_{1N_x}^{n+1}} Q_r - e^{-\bar{u}} (\bar{u} - s) = 0 \quad (27)$$

we proceed from the fact that the desired value $\bar{u}$, according to the introduced judgments, has the greatest value of 1, and the smallest value of -1 . When substituting, -1 the left side of the equation will be less than the right side and vice versa. These judgments are represented by inequalities $F(-1) > 0$ and $F(+1) < 0$.

The graph of the function $F(\bar{u})$ is decreasing, as evidenced by the inequality:

$$F'(\bar{u}) = -e^{-\bar{u}} (1 + s - \bar{u}) < 0. \quad (28)$$

Let's calculate the second derivative

$$F''(\bar{u}) = e^{-\bar{u}} (2 + s - \bar{u}) > 0. \quad (29)$$

She is positive. This means that the graph of the function $F(\bar{u})$ is concave and the tangent method must be applied starting from the left border.

Let us assume that the tangent is drawn through the point $(\bar{u}^k, F(\bar{u}^k))$:

$$F'(\bar{u}^k) = \frac{F(\bar{u}^{k+1}) - F(\bar{u}^k)}{\bar{u}^{k+1} - \bar{u}^k}. \quad (30)$$

This straight line intersects the x-axis at point $(\bar{u}^{k+1}, F(\bar{u}^{k+1}))$. Those. $F(\bar{u}^{k+1}) = 0$. This is where the recurrent relationship comes from [20-24]:

$$\bar{u}^{k+1} = \bar{u}^k - \frac{F(\bar{u}^k)}{F'(\bar{u}^k)} = \bar{u}^k + \frac{e^{\bar{u} - f_{1N_x}^{n+1}} Q_r - \bar{u} + s}{1 - \bar{u} + s}. \quad (31)$$

Recurrent calculations continue until one of the conditions $|\bar{u}^{k+1} - \bar{u}^k| < \varepsilon$ and is satisfied $|F(\bar{u}^{k+1})| < \varepsilon$.

In this case, we find the corresponding density as

$$\bar{\rho}_{N_x}^{n+1} = \frac{Q_r}{\bar{u}_{N_x}^{n+1} - s} \cdot n \quad (32)$$

Let us present the approximate results of the iterative process, demonstrating the rapid convergence of the proposed method.

Table 3.

Results of the iterative process during implementation boundary condition with damper

k	$\bar{u}^k$	$F(\bar{u}^k)$	$\bar{u}^{k+1}$
1	0	1.64896320192167	0.00205039726156074
2	0.00205250482257954	0.00169493091436834	0.00205250482257954
3	0.00205250482257954	1.78829395736102 e- 09	0.0020525048248032

Of practical interest is the change in air volume in the air cap over time, which is inversely proportional to the pressure value at the end of the pipeline section:

$$V_g(t) = \frac{R_g m_g T_0}{p(l, t)} (M^3). \quad (33)$$

Based on the largest and smallest values $V_g(t)$ under permissible disturbances in the boundary conditions, the volume of the air cap is selected.

The process of approximating speed, density and mass flow over time for a specific case, when the dimensionless step in space is 0.001 and in time 0.0002, is presented in the table, which demonstrates a fairly good convergence of the computational process.

Table 4.

Approximate indicators of approximation of the computational process

n	$\max_i \bar{u}_i^{n+1} - \bar{u}_i^n $	$\max_i \bar{\varphi}_i^{n+1} - \bar{\varphi}_i^n $	$\max_i \bar{\rho}_i^{n+1} \bar{u}_i^{n+1} - \bar{\rho}_i^n \bar{u}_i^{n+1} $
10000	1.0376845471392 e-06	1.07492229226101 e-06	5.28143274487053 e-05
2 0000	3.03293628038659 e-07	3.31117707919916 e-07	2.25176282669428 e-05
3 0000	1.37120205582347 e-07	1.61202822140417 e-07	1.01426850976891e-05
40000	7.12793549097895 e-08	4.55207889698411 e-08	3.7457464189108 e-06
50000	2.52709727128694 e-08	3.31614250660001 e-08	1.68952977890354 e-06
60000	1.69238583303671 e-08	1.5730307330146 e-08	5.22595196366917 e-07
70000	1.08441096701231e-08	3.57471952733279 e-09	1.1239282606172 e-07
80000	3.81917755797392 e-09	5.04548891902545 e-09	1.67753939818685 e-07
90000	1.51291195630426 e-09	2.15806607913732 e-09	1.14845524823407 e-07
100000	8.4617548601687 e-10	1.17133186223334 e-09	6.77142840777516 e-08
110000	4.31182796065133 e-10	3.78053192917616 e-10	4.17664739386887 e-08
120000	1.01124774352923 e-10	2.69250580164915 e-10	2.31690014679947 e-08
130000	1.2152736802995 e-10	9.79716099913652 e-11	8.29142876640411 e-09

3. Conclusion

In the framework of a real fluid, a quasi-one-dimensional model of fluid movement in a relief pipeline is proposed, when the law of change in the mass flow rate of the fluid is specified at the inlet of the elementary section, and an air cap is installed at the outlet. When simulating a connected air cap, the air is assumed to be a real gas, the compressibility coefficient of which is almost equal to 1. The initial distributions of the density and velocity of the liquid along the length of the section are specified as the initial conditions.

To solve the problem in the equations and conditions, we switched to dimensionless variables and, with the introduction of an auxiliary function, the equations were presented in the form of analogues of counter-propagating waves. To implement nonlinear boundary conditions, algorithms based on I. Newton's tangent method are proposed.

The results of a computational experiment are presented, which prove the effectiveness of the proposed algorithm and software product for solving quasi-one-dimensional equations N.E. Zhukovsky in the framework of real liquids.

The developed software product will be used to select air cap indicators when studying various non-stationary cases in pipelines.

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INVESTIGATION OF FLUID FLOW IN A HETEROGENEOUS REGION BASED ON THE RAKHMATULIN MODEL

Dalabaev U.¹

¹University of World Economy and Diplomacy, Tashkent, Uzbekistan

E-mail: udalabaev@mail.ru

Abstract. *This paper considers a model for the joint calculation of fluid flow in complex heterogeneous areas. For flow in complex areas, Rakhmatulin's two-speed model is used, which describes the flow of joint motion of systems of particles and fluid. Based on this approach, it is possible to study flows in complex areas using a single equation. The equations are based on the generalized Navier-Stokes equations, obtained by averaging over the characteristic volume of a two-phase medium. The transition from the free subregion to the porous one is carried out through the porosity of the medium, while in the free zone we obtain the Navier-Stokes equation, and in the porous zone we have an equation generalizing the Brinkman equation. SIMPLE has developed a similar algorithm for calculating this model. The problem of motion in a flat channel with various types of porous structure for a stationary flow is considered.*

Keywords: *Navier-Stokes, Rakhmatulin model, numerical solution, modeling, porous media, Darcy's law, two-phase continuum.*

1. Introduction

In many processes, viscous fluid flows in a heterogeneous region occur. A heterogeneous medium means the flow of liquid in a free zone and porous inclusions. Here is a partial list of examples: the flow of gas and oil in and above porous media, underground hydrology, blood flow in stenoses, processes associated with movement through a rough surface, flow in rivers with vegetation and over dunes, etc. In such environments, flow occurs in two subregions: in the free zone and in the subregion with porous media.

There are two approaches to modeling the flow of a liquid or gas in a heterogeneous region. In the first approach, the flow is modeled by its own equations in each subdomain. Simulation of flow in a free zone is described using systems of Navier-Stokes/Stokes equations. There are several options for describing the filtration flow through a saturated porous layer [1-9].

To model flows through a saturated porous medium, there are various equations that depend on the properties of the porous medium. For example, Darcy's law [2], in the case of high permeabilities, the nonlinear Forchheimer equation is used [8]. The following generalizations of the filtration model are associated with the name of Brinkman [4]. When Darcy or Forchheimer type laws are used, the order of partial differential equations in the free fluid region and in the porous medium region is of a different type. The order of the Brinkman differential equations for describing flow in a porous region and the Navier-Stokes equations coincide.

In the case of using a two-regional model, the main issue is the choice of interregional boundaries [1].

Another approach to describe the flow of liquid gas is associated with the use of a single differential equation for the entire combined region [10-13]. For the free region, the equation turns to the Navier-Stokes equation, and in the porous region, terms reflecting the resistance of the porous medium are added to the equations.

In this article we use the second approach, where the model is derived from the two-velocity continuum first outlined by H.A. Rakhmatulin [14]. The two-speed theory of motion of two-phase media was developed in the works of Faizullaev D.F. [15], Nigmatulina R.I. [16], Latipova K.Sh. [17].

In the one-area approach, the difficulty associated with the presence of an interface boundary disappears, and the calculation is carried out for the weight of the area using a single equation. Based on the theory of a two-velocity continuum, a single equation can be derived that allows the study of flow in the combined region. In works [12-13] exactly this approach was used. Here this approach is applied to new problems.

2. Methods

As already stated in the introduction, there are various methods and approaches for describing combined flows in free and porous media. Here we briefly summarize Rakhmatulin's filtration equation and the numerical method used to solve it

3. Mathematical model

Let a porous medium be located in the region of flow of a viscous medium. To simulate such flows, we proceed from an interpenetrating model of two-phase media.

Let's consider an interpenetrating model that describes the flows of two-phase media in a Cartesian coordinate system (two-dimensional problem) [15-16]:

$$\rho_i \frac{\partial u_j^i}{\partial t} + \sum_{k=1}^2 \rho_i u_k^i \frac{\partial u_j^i}{\partial x_k} = -f_i \frac{\partial p}{\partial x_j} + \mu_i \sum_{k=1}^2 \left(\frac{1}{3} \delta_j^k + 1 \right) \frac{\partial}{\partial x_k} \left(f_i \frac{\partial u_j^i}{\partial x_k} \right) +$$

$$+ \mu_i \sum_{k=1}^2 \left(1 - \frac{5}{3} \delta_j^k \right) \frac{\partial}{\partial x_k} \left(f_j \frac{\partial u_{3-j}^i}{\partial x_{3-k}} \right) + K(u_j^{3-i} - u_j^i) + \rho_i g_j^i, \quad i, j = 1, 2$$

$$\frac{\partial \rho_i}{\partial t} + \sum_{k=1}^2 \frac{\partial (\rho_i u_k^i)}{\partial x_k} = 0,$$

$$f_1 + f_2 = 1.$$

Here u_j^i – is the j-th component of the i-th phase velocity; p – pressure, f_i – volume concentration of i-phase ($f_i = \rho_i / \rho_{0i}$); δ_i^k – Kronecker symbol, μ_i – viscosity of the i-th phase; K – phase interaction coefficient; ρ_i, ρ_{0i} – reduced and true densities of the i-th phase ($\rho_{0i} = \text{const}$), g_j^i – j-th component of the gravitational force of the i-th phase.

System of equations (1 – 3) describe the flow of a heterogeneous mixture (liquid and particles).

Assuming in the two-speed model: that 1) there is no motion of the discrete phase ($u_j^2 \equiv 0$), 2) the deformation of the discrete phase can be neglected, the equations of motion can be written in Cartesian coordinates in the following form:

$$\rho \frac{\partial u_j}{\partial t} + \sum_{k=1}^2 \rho u_k \frac{\partial u_j}{\partial x_k} = -f \frac{\partial p}{\partial x_j} + \mu \sum_{k=1}^2 \left(\frac{1}{3} \delta_j^k + 1 \right) \frac{\partial}{\partial x_k} \left(f \frac{\partial u_j}{\partial x_k} \right) +$$

$$+ \mu \sum_{k=1}^2 \left(1 - \frac{5}{3} \delta_j^k \right) \frac{\partial}{\partial x_k} \left(f \frac{\partial u_{3-j}}{\partial x_{3-k}} \right) - K u_j + \rho g_j,$$

$$\sum_{k=1}^2 \frac{\partial f u_k}{\partial x_k} = 0.$$

Here, $\rho = \rho_1$ – liquid density, $f = f_1$ – volumetric concentration, $u_j \equiv u_j^1$ – liquid velocity.

Based on equation (4)–(5), it is possible to study flows both inside and outside porous media, since when $f = 1$ we obtain the Navier-Stokes equations for an incompressible fluid. This circumstance allows the use of the end-to-end counting algorithm.

4. Numerical method

The SIMPLE method [18], which is used to solve the Navier-Stoke equation, uses a staggered (staggered) grid. Equations in (4) and (5) include three unknowns: u_1 , u_2 and p . For each velocity and pressure component, its own mesh is constructed.

The advantage of a checkerboard grid is described in detail in [18].

For clarity, let's write an equation for each velocity component. For longitudinal velocity, the equation of motion has the form:

$$\begin{aligned} \rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = -f \frac{\partial p}{\partial x} + \frac{4\mu}{3} \frac{\partial}{\partial x} \left(f \frac{\partial u}{\partial x} \right) + \\ + \mu \frac{\partial}{\partial y} \left(f \frac{\partial u}{\partial y} \right) - \frac{2\mu}{3} \frac{\partial}{\partial x} \left(f \frac{\partial v}{\partial y} \right) + \mu \frac{\partial}{\partial y} \left(f \frac{\partial v}{\partial x} \right) - Ku + \rho g_1, \end{aligned} \quad (6)$$

For transverse speed, the equation of motion has the form:

$$\begin{aligned} \rho \frac{\partial v}{\partial t} + \rho u \frac{\partial v}{\partial x} + \rho v \frac{\partial v}{\partial y} = -f \frac{\partial p}{\partial y} + \mu \frac{\partial}{\partial x} \left(f \frac{\partial v}{\partial x} \right) + \\ + \frac{4\mu}{3} \frac{\partial}{\partial y} \left(f \frac{\partial u}{\partial y} \right) + \mu \frac{\partial}{\partial x} \left(f \frac{\partial u}{\partial y} \right) - \frac{2\mu}{3} \frac{\partial}{\partial y} \left(f \frac{\partial u}{\partial x} \right) - Kv + \rho g_2, \end{aligned} \quad (7)$$

The continuity equation takes the form:

$$\frac{\partial(fu)}{\partial x} + \frac{\partial(fv)}{\partial y} = 0. \quad (8)$$

In (6.7), we use the Kozeny-Karman relation [21] as the interaction coefficient K, since it is accepted in porous media:

$$K = \frac{\mu \cdot f}{k}, \quad k = \frac{d^2 f^3}{\alpha(1-f)^2},$$

where k is permeability, d is the characteristic size of the porous medium.

For numerical calculation, we write equations (6) and (7) in dimensionless form:

$$\begin{aligned} Sh f \frac{\partial u}{\partial t} + fu \frac{\partial u}{\partial x} + fv \frac{\partial u}{\partial x} = -\frac{f}{Re} \frac{\partial p}{\partial x} + \frac{4}{3Re} \frac{\partial}{\partial x} \left(f \frac{\partial u}{\partial x} \right) + \\ + \frac{1}{Re} \frac{\partial}{\partial y} \left(f \frac{\partial u}{\partial y} \right) - \frac{2}{3Re} \frac{\partial}{\partial x} \left(f \frac{\partial v}{\partial y} \right) + \frac{1}{Re} \frac{\partial}{\partial y} \left(f \frac{\partial v}{\partial x} \right) - Cu, \end{aligned} \quad (9)$$

$$\begin{aligned} Sh f \frac{\partial v}{\partial t} + fu \frac{\partial v}{\partial x} + fv \frac{\partial v}{\partial x} = -\frac{f}{Re} \frac{\partial p}{\partial y} + \frac{1}{Re} \frac{\partial}{\partial x} \left(f \frac{\partial v}{\partial x} \right) + \\ + \frac{4}{3Re} \frac{\partial}{\partial y} \left(f \frac{\partial u}{\partial y} \right) + \frac{1}{Re} \frac{\partial}{\partial x} \left(f \frac{\partial v}{\partial y} \right) - \frac{2}{3Re} \frac{\partial}{\partial y} \left(f \frac{\partial v}{\partial x} \right) - Cv \end{aligned} \quad (10)$$

where u, v – longitudinal and transverse components of velocity; x, y – Cartesian coordinates; f – layer porosity; $Sh = h/UT$ – Strouhal number; $Re = hU\rho_0/\mu$ – Reynolds number; U – characteristic speed; h – characteristic size; ρ_0, μ – density and viscosity of the medium; $D = \sqrt{\alpha}h/d$ – dimensionless number; d is the characteristic size of the porous medium; α – coefficient of proportionality. Dimensionless variables are related to dimensional ones as follows:

$$x = \frac{x'}{h}, \quad y = \frac{y'}{h}, \quad u = \frac{u'}{U}, \quad v = \frac{v'}{U}, \quad p = \frac{Re p'}{\rho_0 U^2}$$

(here the variables with strokes are dimensional).

The dimensionless form of equation (8) is preserved.

The system of equations (8)-(10) describes the flow of liquid and gas both inside and outside the porous layer, while the entry of a point into the porous or free zone is controlled through the porosity. For equations (8)-(10) the control volume method with corresponding generalizations.

5. Results and discussions

Here we present the results of calculating flow in a flat pipe with various configurations near an impermeable wall of a stationary flow. Note that in work [12] the flow of liquid in a partially filled and horizontally located porous medium was studied using the proposed model and a comparison was given with experiment [22], which showed good agreement. In work [11], a single-regional approach to modeling such flows was used. It is based on the generalized Navier-Stokes equations. In this case, the equations were obtained by averaging over the characteristic volume of the porous medium for the entire combined region. For the numerical calculation of this model, a calculation algorithm based on the finite element method has been developed. One of the problems considered in this work relates to the completely filled part of the pipe. This problem was also solved in [23] using the Rakhmatulin model; a comparison of these results with [11] shows the identity of the velocity profiles.

6. Flow with symmetrically located porous inclusions

The flow area and the direction of the coordinate axis are presented in Fig. 1. Equation (8)-(10) is considered in the region: $0 \leq x \leq L$, $0 \leq y \leq 1$. ($L=3$) without taking into account external forces. A porous medium is located near the wall, the boundary is represented by a dashed line.

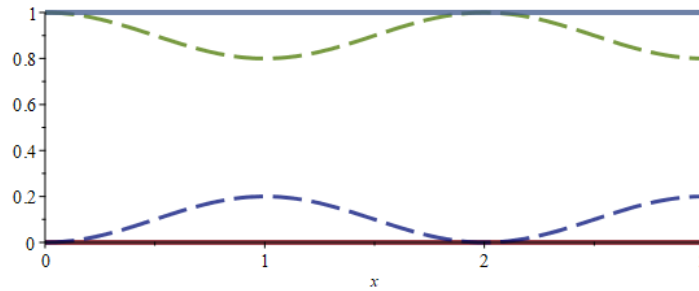
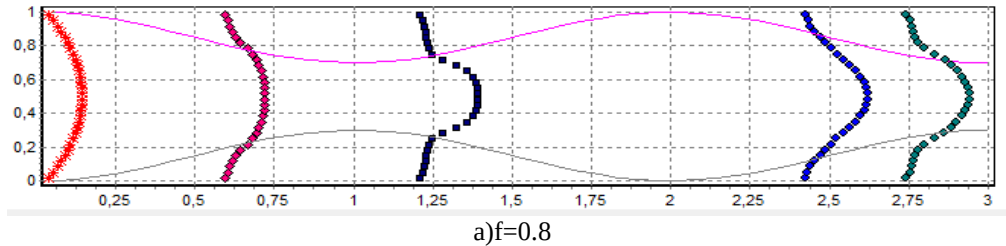


Figure 1. - Flow region and direction of axes.

At the boundaries of the computational domain, a no-slip condition is specified on the solid wall; at the entrance to the channel, constant pressure and Poiseuille flow are specified; and at the output - a soft boundary condition L

$$\frac{\partial u}{\partial x} = 0, v = 0.$$

Figure 2 shows the longitudinal velocity distribution.



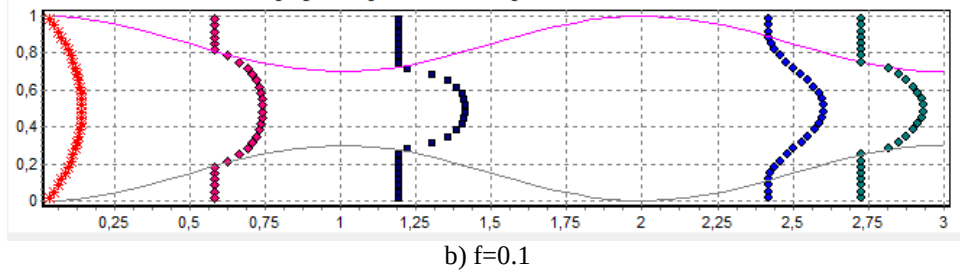


Figure 2. - Longitudinal velocity profiles for different sections. Flow parameters $Re=500$, $D=100$

Comparison of Fig. 2 a) and b) shows a strong influence of porosity, showing the presence of a boundary layer near the interregional boundary, which has been studied by many authors [7,24].

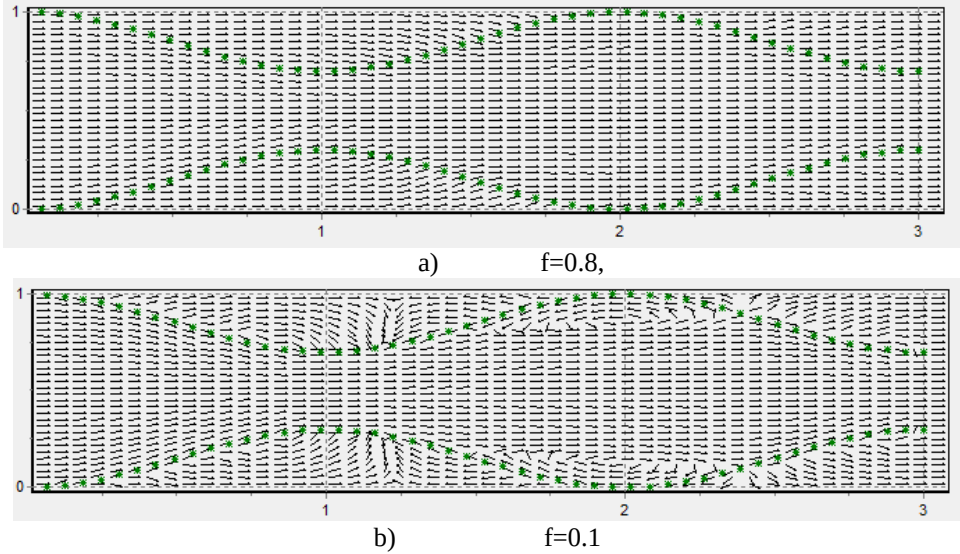


Figure 3. - Current velocity field. $Re=500$, $D=100$.

Figure 3 shows the flow field with flow parameters: $Re=500$, $D=100$. A comparison of Fig. 3 shows a strong dependence of the flow pattern upon changes in porosity: at large porosities, the direction of the flow field in the porous medium and the main flow are identical; at small porosities of the medium, a circulation flow is observed in both subregions.

7. Flow with asymmetrically located porous inclusions

Let us now consider the flow in a channel with asymmetrically located porous media near the boundary. Figure 4 shows the calculation area and axis direction.

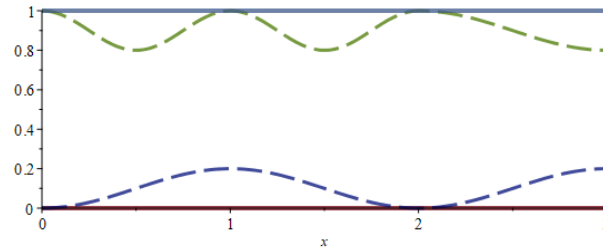


Figure 4. - Flow area and axis direction

The boundary conditions are the same as in the previous problem.

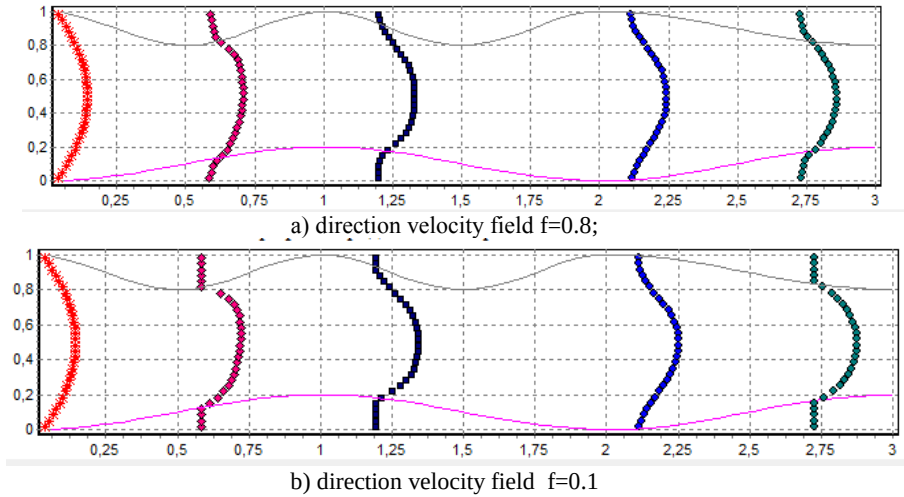


Figure 5. - **Longitudinal velocity profiles.** Calculation parameters: $D=100$; $Re=500$.

Fig. 5 shows that, regardless of the symmetry of porous structures, at high porosities there is a boundary layer near the inter-boundary interfaces. In the case of small porosity, a uniform velocity profile takes place inside the porous media.

Fig. 6 shows the direction of the velocity field at $f=0.1$; a small circulation zone is observed in the main flow. In the case of $f=0.8$, the tendency of the direction of the velocity field is similar to the symmetric case

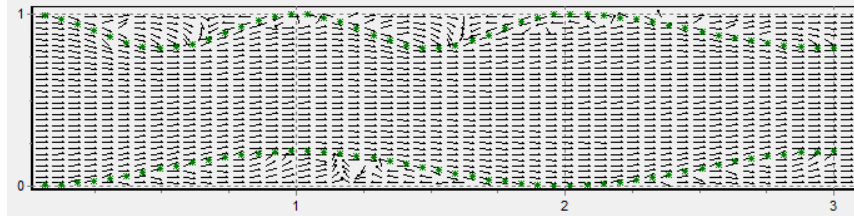


Figure 6. - **Direction of the velocity field ($f=0.1$; $Re=500$)**

An increase in the amplitude of the porous layer will lead to severe deformation of the flow (Fig. 7)

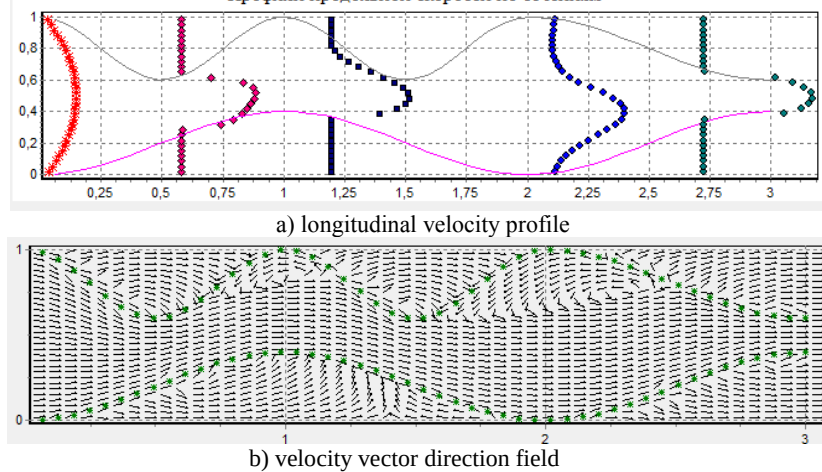


Figure7. - **Calculation with the amplitude of the porous layer equal to 0.4. ($Re=500$, $f=0.1$)**

Fig. 7 shows the influence of the vibration amplitude number of the porous layer on the flow structure.

8. Conclusion.

The Rakhmatulin filtration model, which is obtained on the basis of a two-speed interpenetrating model, qualitatively describes the behavior of the liquid in the combined region. A convenient tool will be created in terms of numerical implementation for comparison of the two-region model. Using the proposed model, it is possible to obtain characteristics of the hydrodynamic fields of joint flow with porous inclusions. The extended Navier-Stokes model can be used to study hydrodynamic structures for flow in complex regions using a single equation without explicitly identifying the interregional boundary.

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MATHEMATICAL MODEL OF COMBUSTION OF METHANE-AIR MIXTURE

Ergashev D.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after. M.T. Urazbaeva, Tashkent, Uzbekistan
E-mail: dilshod77m@gmail.com*

Abstract. *This study proposes a numerical method to solve the finite rate chemical reaction propagation and combustion problem using a satellite airflow oxidizer in a turbulent flow. Methane is considered as a fuel. The model used dimensionless equations of a turbulent boundary layer of reacting gases in von Mises coordinates. In addition, a two-layer six-point implicit finite-difference scheme was used for the numerical solution. This scheme provides second-order approximation accuracy in both coordinates. As a result of the nonlinearity of the differential equations for the conservation of substance, an iterative process was organized. The results of a computational experiment are shown. Comparison of the results of calculating changes in the temperature of the axial flow from experimental data and Sekundov turbulence models $v_t - 92$.*

Keywords: *Mathematical model, turbulent jet, chemical reaction rate, total enthalpy, Arrhenius law, finite differences, numerical algorithm, computational experiment.*

1. Introduction

Combustion is a physical and chemical process that produces heat and a bright glow as a result of a chemical oxidation reaction. In the modern world, jet streams of reacting media are used in many areas of technology and production, including everyday life, thermal and electrical energy production, internal combustion chambers, chemical lasers, the production of building materials and mercury, rocket engine propulsion and much more.

Various facility management problems depend on the purpose of the organization and the design of gas combustion devices. In particular, gas makes up approximately 70% of the global energy balance. Gas combustion devices remain inefficient despite continuous improvements in their performance.

The rate of dissipation of turbulent energy is the second independent variable to describe the properties of turbulence. This is due to the fact that, on the one hand, for this quantity it is possible to construct a constitutive equation under reasonable physical premises, and on the other hand, this quantity is included in the constitutive equation for the energy of turbulent pulsations.

The turbulence model $k - \varepsilon$ is used to solve individual problems of isothermal jets and jets reacting to gas mixtures at the present time [1, 2].

Ya.B. Zeldovitsky conducted the most extensive study of the combustion of unmixed gases [3]. Schwab [4] was the first to develop a general scheme of the turbulent plume process, based on the theory of free jets. Diffusion combustion was also studied by the authors of [5] and others.

The idea of a plume is based on the idea that a hot plume and turbulent jets have a common transport mechanism [6]. The results of a computational, theoretical and experimental study of the aerodynamics of an axisymmetric turbulent plume are presented. They make it possible to determine the location of the flame front, the distribution of temperature, concentrations and speed in the cross sections of the torch. Both a quantitative assessment and a physical explanation of experimentally known facts are included, such as the increase in the degree of turbulence at the flame front, as well as the difference between unit values of the Pr and Sc numbers in shear flows.

Work [7] describes the process and provides the results of a numerical solution to the problem of a turbulent diffusion plume with a finite reaction rate. In the Mises-Mager-Libby plane and using an obvious four-point finite-difference scheme, a numerical integration of the system of boundary layer equations is performed. The momentum equation is linearized. Despite these assumptions, the experimental results are fully consistent with the calculation results.

In [8], a mathematical model was developed that allows one to calculate the characteristics of a turbulent diffusion flame that propagates in an isobaric subsonic flow. It is assumed that a one-step irreversible reaction occurs between the oxidizer and the fuel entering the flame front. Unlike well-known models of diffusion combustion, the assumption of an infinitely high rate of chemical reactions is not used. Thus, the equation for the average fuel concentration is considered together with the equations for

the average flow velocity and the concentration of passive impurities. The rate of a chemical reaction is proportional to the product of the fuel concentration C_f , the oxidizer concentration C_0 and the average flow velocity across the plume $\frac{\partial u}{\partial z} \cdot \left(\omega \cdot C_f \cdot C_0 \cdot \frac{\partial u}{\partial z} \right)$, which makes integration of the equations much simpler.

The system of equations is solved as follows: self-similar solutions of the equations describe the field of average velocity and the concentration of passive impurities, each of which is closed. Solving these equations leads to the conclusion that the coefficient of turbulent viscosity remains constant. The developed model determines the position of the flame front, temperature distribution and concentration of reactants in the diffusion plume.

Article [9] discusses recent advances in the study of chemical reactions in turbulent flow. Homogeneous mixture combustion and diffusion combustion are two possible options. It is shown that the assumption of an infinitely high rate of chemical reaction makes the description of the combustion process much simpler. Calculation methods that allow taking into account minor deviations of the medium from the equilibrium thermodynamic state are analyzed. It is emphasized that such calculations are especially important for predicting the content of toxic substances such as nitrogen oxides and soot in the flame. In addition, the influence of heat generation on the flow field, especially on turbulence characteristics, is considered. There is special attention to the effects of density pulsations. The analysis showed that models designed to describe constant-density flows may not be suitable for computing turbulent combustion.

Research on the swirling jet, which is successfully used in the national economy as one of the rational ways to improve mixing and combustion processes, should be considered when considering the efficient combustion of gaseous fuels.

The spatial integration method was used in the study [10] to solve equations describing an axisymmetric incompressible turbulent swirling jet flow.

Isotropic turbulent viscosity $k-\varepsilon$ was used. Turbulence characteristics such as Reynolds stress, turbulent kinetic energy and dissipation rate, in addition to the distribution of averaged velocities, are determined. Good agreement with the experimental results was obtained. This study shows that the hypothesis of isotropic turbulent viscosity is well founded.

As a result of an analysis of the scientific literature [1-10], it was found that methods of homogeneous combustion of gases, in which methane is the main component, have not yet been sufficiently studied. As is known, research on the topic of combustion continues due to the fact that the fundamental principles of chemical exothermic reactions under conditions of turbulent jet flow of a nozzle are not sufficiently developed.

2. Formulation of the problem

We consider a turbulent jet of flammable gas flowing from a round nozzle with a diameter $2a$ and spreads in a cocurrent flow of the oxidizer (in a flooded space).

We will assume that the indicators are given, uniform and homogeneous, and also that the initial (at $x = 0$) distributions of temperature, fuel and oxidizer concentrations will be uniform and have a tangential break in the indicators at the nozzle boundary.

The system of equations for the turbulent motion of a multicomponent gas in the presence of a chemical reaction, that is, combustion between the components, and when the Lewis number for the components is equal to unity ($Le_i = 1$, i.e. $Pr = Sc_i$) can be written in cylindrical coordinates taking into account the Sekundov model $\nu_t - 92$ [13-15]:

$$\left\{ \begin{array}{l} \rho u \frac{\partial u}{\partial x} + \rho g \frac{\partial u}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left(\rho (\nu + \nu_t) r \frac{\partial u}{\partial r} \right), \\ \frac{\partial(\rho u r)}{\partial x} + \frac{\partial(\rho g r)}{\partial r} = 0, \\ \rho u \frac{\partial H}{\partial x} + \rho g \frac{\partial H}{\partial r} = \frac{1}{Pr r} \frac{\partial}{\partial r} \left(\rho (\nu + \nu_t) r \frac{\partial H}{\partial r} \right), \\ \rho u \frac{\partial c_n}{\partial x} + \rho g \frac{\partial c_n}{\partial r} = \frac{1}{Sc r} \frac{\partial}{\partial r} \left[\rho \left(\frac{1}{Re} + \nu_t \right) r \frac{\partial c_n}{\partial r} \right] + \omega_n \quad (n=1..N), \\ \rho u \frac{\partial \nu_t}{\partial x} + \rho g \frac{\partial \nu_t}{\partial r} = P_v - D_v + \frac{1}{r} \frac{\partial}{\partial r} \left(\rho r (\nu + c_0 \nu_t) \frac{\partial \nu_t}{\partial r} \right) + \\ + \frac{1}{r} \frac{\partial}{\partial r} \left(\rho r (-\nu + (c_1 - c_0) \nu_t) \right) \frac{\partial \nu_t}{\partial r}. \end{array} \right. \quad (2)$$

This system is closed by the following dependencies:

$$\begin{aligned} H &= c_p T + c_2 h_2^*, \quad c_p = \sum_{n=1}^N c_{pn} c_n, \quad p = \rho \frac{R_0}{m} T = const, \quad m = \left(\sum_{n=1}^N m_n / c_n \right)^{-1}, \\ P_v &= \rho c_1 F_2 \left(\nu_t G_1 + A_1 \nu_t^{\frac{4}{3}} G_2^{\frac{2}{3}} \right) + \rho c_2 F_2 A_2 N_1 (\nu + \nu_t)^{\frac{1}{2}} G_1^{\frac{1}{2}} + \\ &+ \rho c_3 \nu_t \left(\frac{\partial^2 \nu_t}{\partial r^2} + N_2 \right), \\ D_v &= \frac{\rho c_5 \nu_t^2 G_1^2}{a_{SZ}^2} + c_4 \rho \nu_t \left(\frac{\partial u}{\partial r} + \left| \frac{\partial u}{\partial r} \right| \right) + \frac{(\rho c_6 \nu_t (N_1 d_w + \nu_{t,\omega}) + c_7 \rho F_1 \nu \nu_t)}{d^2}, \\ F_1 &= \frac{N_1 d_w + 0.4 c_8 \nu}{\nu_t + c_8 \nu + \nu_{t,\omega}}, \quad F_2 = \frac{\chi^2 + 1.3 \chi + 0.2}{\chi^2 - 1.3 \chi + 1}, \quad \chi = \frac{\nu_t}{7\nu}, \quad G_1 = \left| \frac{\partial u}{\partial r} \right|, \quad G_2 = \left| \frac{\partial^2 u}{\partial r^2} \right|, \\ N_1 &= \left| \frac{\partial \nu_t}{\partial r} \right|, \quad N_2 = \left| \frac{\partial N_1}{\partial r} \right|, \\ \nu_{t,\omega} &= 0, \quad k_s = 0, \quad A_1 = -0.5, \quad A_2 = 4, \quad c_0 = 0.8, \quad c_1 = 1.6, \quad c_2 = 0.1, \quad c_3 = 4.0, \\ c_4 &= 0.35, \quad c_5 = 3.5, \quad c_6 = 2.9, \quad c_7 = 31.5, \quad c_8 = 0.1, \quad c_s = 0.1, \quad a_{SZ} = 0.2. \end{aligned}$$

Here and further u, ν – averaged longitudinal and radial components of the velocity vector ($m \cdot c^{-1}$); x, r – longitudinal and radial coordinates (m); ρ, T – density ($kg \cdot m^{-3}$) and absolute temperature (K) of the gas mixture; p – static pressure (Pa); Pr, Sc_n – turbulent analogues of the Prandtl and Schmidt numbers; C_n – mass concentration n of the n th gas component in the mixture ($kg \cdot kg^{-1}$); ω_n – mass rate of formation and disappearance n – of the gas component ($kg \cdot m^{-3} \cdot c^{-1}$); $c_p = \sum_{n=1}^N c_{pn} c_n$ And C_{pn} – heat capacity of the gas mixture and n the n th component at constant pressure ($J \cdot kg^{-1} \cdot K^{-1}$); h_n^* – calorific value n of the n th component ($J \cdot kg^{-1}$); ν_t – kinematic turbulence coefficient ($m^2 \cdot c^{-1}$); H – total enthalpy of gas ($J \cdot kg^{-1}$);

The gas mixture is assumed to be perfect, so its state satisfies the Mendeleev-Clapeyron equation [13]:

$$p = \rho R_0 T / m, \quad (3)$$

where $m = \left(\sum_{n=1}^N c_n / m_n \right)^{-1}$ and m_n – molar masses of the gas mixture n and gas component (кг моль^{-1});

R_0 – universal gas constant ($8341.41 \text{ Дж моль}^{-1} \text{ K}^{-1}$).

The boundary conditions of the problem for the input section $x = 0$ are specified as:

$$F(0, y) = \begin{cases} F_2 & \text{на срезе трубы,} \\ F_1 & \text{на спутном потоке.} \end{cases}$$

Here $F = u, T, c_n, v_t$. The total enthalpy value for the inlet section is determined according to the above formula.

When $x > 0$ on the jet axis, the conditions for no flow are set:

$$\frac{\partial F(x, 0)}{\partial y} = 0, \quad v(x, 0) = 0,$$

and at the jet boundary - according to the indicators F_1 of the coflow.

Methane is considered as a combustible gas. The one-stage kinetics of methane combustion in air is given through the stoichiometric equation



Indices 1, 2, 3, 4 and 5 highlight the parameters of the components - oxygen, methane, carbon dioxide, water vapor and nitrogen, respectively.

Then, according to [11], which is borrowed from [12], the reaction rate of methane with oxygen has the form:

$$\omega_2 = -\frac{A}{10^{20} \cdot 4.1868^2} c_1^2 c_2 \rho^3 \exp\left(-\frac{A_{r2}}{T}\right) \left[\frac{\kappa \mathcal{E}}{\mathcal{M}^3 c}\right].$$

At the same time, for the remaining components, the rate of chemical transformations in dimensional form is presented in the form

$$\omega_1 = \frac{\nu_1 m_1}{\nu_2 m_2} \omega_2, \quad \omega_3 = -\frac{\nu_3 m_3}{\nu_2 m_2} \omega_1, \quad \omega_4 = -\frac{\nu_4 m_4}{\nu_2 m_2} \omega_1, \quad \omega_5 = 0 \quad (\nu_5 = 0).$$

The conservation equation for n the th component ($n = 1..5$) is written as $L(c_n) = \omega_n$,

Where
$$L(c_n) = \rho u \frac{\partial c_n}{\partial x} + \rho v \frac{\partial c_n}{\partial y} - \frac{1}{\text{Sc}_n} \frac{\partial}{\partial y} \left(\rho \varepsilon y \frac{\partial c_n}{\partial y} \right).$$

The boundary conditions of the problem for the mass concentration of components for the inlet section $x = 0$ are specified in the following form: at $0 \leq y \leq a$: $c_n(0, y) = (c_n)_2$; at $a \leq y \leq y_\infty$: $c_n(0, y) = (c_n)_1$.

In particular, if the combustible mixture contains 50% methane by weight and 50% oxygen, then it can be taken $(c_2)_2 = 0,500$, $(c_1)_2 = 0,500$. The air composition is taken in the form $(c_1)_1 = 0,232$, $(c_5)_1 = 0,768$ [11]. In this case, the remaining concentrations of components at the input of the computational domain have zero values: $(c_2)_1 = (c_3)_1 = (c_4)_1 = (c_3)_2 = (c_4)_2 = (c_5)_2 = 0$.

When $x > 0$ the symmetry conditions are set on the jet axis: $\frac{\partial c_n(x, 0)}{\partial y} = 0$, and on the jet boundary -

according to the air flow indicators: $c_n(x, y_\infty) = (c_n)_1$.

Convective and diffusive transport of components have the same intensities, taking into account the neglect of thermal and barodiffusion, as well as the same values of the turbulent analogue of the Schmidt number for each component. Thus, we write the conservation equations for the five components in the form

$$L(c_2) = \omega_2, \quad L(c_1) = \frac{\nu_1 m_1}{\nu_2 m_2} \omega_2, \quad L(c_3) = -\frac{\nu_3 m_3}{\nu_2 m_2} \omega_2, \quad L(c_4) = -\frac{\nu_4 m_4}{\nu_2 m_2} \omega_2, \quad L(c_5) = 0.$$

Using the first of these equations, we reduce the next three equations to a homogeneous form:

$$L\left(c_1 - \frac{\nu_1 m_1}{\nu_2 m_2} c_2\right) = 0, \quad L\left(c_3 + \frac{\nu_3 m_3}{\nu_2 m_2} c_2\right) = 0, \quad L\left(c_4 + \frac{\nu_4 m_4}{\nu_2 m_2} c_2\right) = 0.$$

When $n = 1, 3, 4$ we introduce the notation $\tilde{c}_n = c_n \mp \frac{\nu_n m_n}{\nu_2 m_2} c_2$ (the upper sign refers to $n = 1$, and the lower one - $n = 3, 4$), we write the equations in a single form $L(\tilde{c}_n) = 0$ and formulate the boundary conditions for the air flow at the inlet in the form $(\tilde{c}_n)_1 = (c_n)_1 \mp \frac{\nu_n m_n}{\nu_2 m_2} (c_2)_1$, for the fuel flow - $(\tilde{c}_n)_2 = (c_n)_2 \mp \frac{\nu_n m_n}{\nu_2 m_2} (c_2)_2$, as well as at the boundary and axis of the jet.

If we introduce relatively excess functions in the form of dependence $\bar{\bar{C}}_n = \frac{\tilde{c}_n - (\tilde{c}_n)_1}{(\tilde{c}_n)_2 - (\tilde{c}_n)_1}$, then for

the transformed three equations and the equation of $\tilde{c}_5 = C_5$ inert gas concentration the boundary conditions become the same: at the fuel inlet - $(\bar{\bar{C}}_n)_2 = 1$, at the air inlet and at the boundary of the boundary layer - $(\bar{\bar{C}}_n)_1 = 0$, and at the jet axis - $\frac{\partial \bar{\bar{C}}_n(x, 0)}{\partial y} = 0$. Accordingly, these equations become

mutually equivalent and can be replaced by one equation for $\bar{\bar{C}}$. In this regard, instead of the five equations of the diffusion problem, the equation is solved with respect $\bar{\bar{C}}$ to the source term and the equation with respect to the fuel concentration c_2 taking into account the rate of the chemical reaction.

The reverse transition from relatively excess concentration $\bar{\bar{C}}$ to mass concentrations of components C_n at $n = 1, 3, 4$ is carried out according to the formula

$$c_n = [(\tilde{c}_n)_2 - (\tilde{c}_n)_1] \bar{\bar{C}} + (\tilde{c}_n)_1 \pm \frac{\nu_n m_n}{m_2 \nu_2} c_2,$$

and when $n = 5$ - according to the formula $c_5 = [(\tilde{c}_5)_2 - (\tilde{c}_5)_1] \bar{\bar{C}} + (\tilde{c}_5)_1$.

Gas combustion occurs at relatively low speeds relative to the speed of sound. Thus, the kinetic energy of the gas can be ignored. Thus, a similar replacement of the introduced concentration can be applied to the total enthalpy, which will lead to the normalization of the total enthalpy [13].

$$\bar{\bar{H}} = \frac{H - H_1}{H_2 - H_1} = \bar{\bar{C}}.$$

Here $H_1 = (c_p)_1 T_1 + (c_2)_1 h_2^*$, $H_2 = (c_p)_2 T_2 + (c_2)_2 h_2^*$.

In this case, the temperature of the mixture is calculated using the formula

$$T = (H - c_2 h_2^*) / c_p = \left[H_1 + (H_2 - H_1) \bar{\bar{H}} - c_2 h_2^* \right] / \sum_{n=1}^N c_{pn} c_n.$$

The density of the gas mixture required for further calculations is found using the formula

$$\rho = \frac{p}{R_0 T \sum_{n=1}^N c_n / m_n}.$$

Introduction of dimensionless and von Mises variables .

To solve the problem, we switched to dimensionless variables using characteristic quantities

a, u_2, ρ_2 :

$$\bar{x} = \frac{x}{a}, \quad \bar{y} = \frac{y}{a}, \quad \bar{u} = \frac{u}{u_2}, \quad \bar{\rho} = \frac{\rho}{\rho_2}, \quad \text{Re} = \frac{u_2 a}{\nu}, \quad \bar{G}_1 = \frac{G_1 a}{u_2}, \quad \bar{G}_2 = \frac{G_2 a^2}{u_2}, \quad \bar{N}_1 = \frac{N_1}{u_2}, \quad \bar{N}_2 = \frac{N_2 a}{u_2}.$$

Thus, the differential equations took the form:

$$\left\{ \begin{array}{l} \bar{\rho} \bar{u} \frac{\partial \bar{u}}{\partial \bar{x}} + \bar{\rho} \bar{g} \frac{\partial \bar{u}}{\partial \bar{r}} = \frac{\partial}{\partial \bar{r}} \left[\bar{\rho} \left(\frac{1}{\text{Re}} + \bar{v}_t \right) \bar{r} \frac{\partial \bar{u}}{\partial \bar{r}} \right], \\ \frac{\partial (\bar{\rho} \bar{u} \bar{r})}{\partial \bar{x}} + \frac{\partial (\bar{\rho} \bar{g} \bar{r})}{\partial \bar{r}} = 0, \\ \bar{\rho} \bar{u} \frac{\partial \bar{\bar{H}}}{\partial \bar{x}} + \bar{\rho} \bar{g} \frac{\partial \bar{\bar{H}}}{\partial \bar{r}} = \frac{1}{\text{Pr} \bar{r}} \frac{\partial}{\partial \bar{r}} \left[\bar{\rho} \left(\frac{1}{\text{Re}} + \bar{v}_t \right) \bar{r} \frac{\partial \bar{\bar{H}}}{\partial \bar{r}} \right], \\ \bar{\rho} \bar{u} \frac{\partial \bar{c}_2}{\partial \bar{x}} + \bar{\rho} \bar{g} \frac{\partial \bar{c}_2}{\partial \bar{r}} = \frac{1}{\text{Sc} \bar{r}} \frac{\partial}{\partial \bar{r}} \left[\bar{\rho} \left(\frac{1}{\text{Re}} + \bar{v}_t \right) \bar{r} \frac{\partial \bar{c}_2}{\partial \bar{r}} \right] - A_{r1} c_1^2 c_2 \bar{\rho}^3 \exp(-A_{r2} / T), \\ \bar{\rho} \bar{u} \frac{\partial \bar{v}_t}{\partial \bar{x}} + \bar{\rho} \bar{g} \frac{\partial \bar{v}_t}{\partial \bar{r}} = \bar{P}_v - \bar{D}_v + \frac{1}{\bar{r}} \frac{\partial}{\partial \bar{r}} \left(\bar{\rho} \bar{r} \left(\frac{1}{\text{Re}} + c_0 \bar{v}_t \right) \frac{\partial \bar{v}_t}{\partial \bar{r}} \right) + \\ + \frac{1}{\bar{r}} \frac{\partial}{\partial \bar{r}} \left(\bar{r} \bar{\rho} \left(-\frac{1}{\text{Re}} + (c_1 - c_0) \bar{v}_t \right) \right) \frac{\partial \bar{v}_t}{\partial \bar{r}}. \end{array} \right.$$

In closing relations we proceed to dimensionless variables as follows.

We have $p = \rho \frac{R_0}{m} T = \rho_2 \frac{R_0}{(m)_2} T_2$.

From here we find the expression for density: $\bar{\rho} = \frac{m T_2}{(m)_2 T}$.

Let's transform the conditions for the required ones.

For $\bar{x} = 0$ and $0 \leq \bar{r} < 1$ we have: $\bar{u} = 1$, $\bar{\bar{H}} = 1$, $c_n = (c_n)_2$.

At $\bar{x} = 0$, $\bar{r} \geq 1$ and at $\bar{x} > 0$, $\bar{r} \rightarrow \bar{r}_\infty$: $\bar{u} = \frac{u_1}{u_2}$, $\bar{\bar{H}} = 0$, $c_n = (c_n)_1$.

On the jet axis at $\bar{x} > 0$: $\frac{\partial \bar{u}}{\partial \bar{r}} = 0$, $\bar{g} = 0$, $\frac{\partial \bar{\bar{H}}}{\partial \bar{r}} = 0$, $\frac{\partial c_n}{\partial \bar{r}} = 0$.

The peculiarity of this problem is that the flow moves along the x-axis from left to right. Since they do not affect propagation processes, there are no boundary conditions on the right side of the flow.

Thus, the procedure should be described in Mises coordinates and the stream function should be introduced ψ in accordance with the relations [13-16]:

$$\rho u r = \psi \frac{\partial \psi}{\partial r}, \quad \rho \vartheta r = -\psi \frac{\partial \psi}{\partial x}$$

(Here and below we will omit the sign of non-dimensionality in equations and relationships.) The second coordinate, which is officially introduced, is called $\xi = x$.

Derivatives in the equations are replaced according to the dependencies:

$$\begin{aligned} \frac{\partial}{\partial x} &= \frac{\partial}{\partial \xi} \frac{\partial \xi}{\partial x} + \frac{\partial}{\partial \psi} \frac{\partial \psi}{\partial x} = \frac{\partial}{\partial \xi} + \frac{\partial}{\partial \psi} \frac{\partial \psi}{\partial x} = \frac{\partial}{\partial \xi} - \frac{\rho \vartheta r}{\psi} \frac{\partial}{\partial \psi}, \\ \frac{\partial}{\partial r} &= \frac{\partial}{\partial \xi} \frac{\partial \xi}{\partial r} + \frac{\partial}{\partial \psi} \frac{\partial \psi}{\partial r} = \frac{\partial}{\partial \psi} \frac{\partial \psi}{\partial r} = \frac{\rho u r}{\psi} \frac{\partial}{\partial \psi}. \end{aligned}$$

The differential equations for the conservation of substances take the form [15-20]:

$$\begin{cases} \frac{\partial u}{\partial \xi} = \frac{1}{\psi} \frac{\partial u}{\partial \psi} \left[\frac{\rho^2 u r^2 (\nu_l + \nu_t)}{\psi} \frac{\partial u}{\partial \psi} \right], \\ \frac{\partial \bar{H}}{\partial \xi} = \frac{1}{\text{Pr} \psi} \frac{\partial}{\partial \psi} \left[\frac{\rho^2 u r^2 (\nu_l + \nu_t)}{\psi} \frac{\partial \bar{H}}{\partial \psi} \right], \\ \frac{\partial c_2}{\partial \xi} = \frac{1}{\text{Sc} \psi} \frac{\partial}{\partial \psi} \left[\frac{\rho^2 u r^2 (\nu_l + \nu_t)}{\psi} \frac{\partial c_2}{\partial \psi} \right] - A_{r1} \frac{c_1^2 c_2 \bar{\rho}^2}{u} \exp(-A_{r2} / T), \\ \frac{\partial \nu_t}{\partial \xi} = \frac{P_v}{\rho u} - \frac{D_v}{\rho u} + \frac{1}{\psi} \frac{\partial}{\partial \psi} \left[\frac{\rho^2 u r^2}{\psi} \left(\frac{1}{\text{Re}} + c_0 \nu_t \right) \frac{\partial \nu_t}{\partial \psi} \right] + \\ + \frac{1}{\psi} \frac{\partial}{\partial \psi} \left[\frac{\rho r}{\psi} \left(-\frac{1}{\text{Re}} + (c_1 - c_0) \nu_t \right) \right] \left(\frac{\rho u r}{\psi} \frac{\partial \nu_t}{\partial \psi} \right). \end{cases}$$

$$T = \frac{H_1 + (H_2 - H_1) \bar{H} - c_2 h_2^*}{c_p}, \quad \bar{\rho} = \frac{m T_2}{(m)_2 T}.$$

$$G_1 = \left| \frac{\rho u r}{\psi} \frac{\partial u}{\partial \psi} \right|, \quad G_2 = \left| \frac{\rho^2 u^2 r^2}{\psi^2} \frac{\partial^2 u}{\partial \psi^2} \right|, \quad N_1 = \left| \frac{\rho u r}{\psi} \frac{\partial \nu_t}{\partial \psi} \right|, \quad N_2 = \left| \frac{\rho u r}{\psi} \frac{\partial N_1}{\partial \psi} \right|.$$

$$P_v = \rho c_1 F_2 \left(\nu_t G_1 + A_1 \nu_t^{\frac{4}{3}} G_2^{\frac{2}{3}} \right) + \rho c_2 F_2 A_2 N_1 \left(\frac{1}{\text{Re}} + \nu_t \right)^{\frac{1}{2}} G_1^{\frac{1}{2}} + \rho c_3 \nu_t \left(\frac{\rho^2 u^2 r^2}{\psi^2} \frac{\partial^2 \nu_t}{\partial \psi^2} + N_2 \right).$$

$$D_v = \frac{\rho c_5 \nu_t^2}{a_{sz}^2} G_1^2 + \frac{\rho u r c_4 \rho \nu_t}{\psi} \left(\frac{\partial u}{\partial \psi} + \left| \frac{\partial u}{\partial \psi} \right| \right) + \frac{\left(\rho c_6 \nu_t (N_1 d_w + \nu_{t,w}) + c_7 \rho F_1 \frac{1}{\text{Re}} \nu_t \right)}{d^2}.$$

At the entrance $\xi = 0$ to the computational domain, the following conditions are imposed:

at $0 \leq \bar{\psi} < 1$: $\bar{u} = 1$, $\bar{H} = 1$, $c_2 = (c_2)_2$, $\bar{\nu}_t = 0.007$.

at $1 \leq \bar{\psi} < \bar{\psi}_\infty$: $\bar{u} = \bar{u}_1$, $\bar{H} = 0$, $c_2 = (c_2)_1$, $\bar{\nu}_t = 0.005$.

With $\xi > 0$ we have with $\psi = 0$: $\frac{\partial \bar{u}}{\partial \bar{\psi}} = 0$, $\frac{\partial \bar{H}}{\partial \bar{\psi}} = 0$, $\frac{\partial c_2}{\partial \bar{\psi}} = 0$, $\frac{\partial \nu_t}{\partial \bar{\psi}} = 0$.

at $\psi \rightarrow \psi_\infty: \bar{u} = \bar{u}_1, \bar{H} = 0, c_2 = (c_2)_1, \bar{v}_t = 0.005$.

3. Experimental results

Calculations were carried out in accordance with the above. The mass composition of air was specified in the form $(c_1)_1 = 0.232, (c_5)_1 = 0.768$. The flammable mixture included a propane-butane mixture and nitrogen. The speed of the main flow was 61 m/s, and the speed of the associated flow was 18.3 m/s. The air temperature is $T_1 = 300\text{ K}$, and the combustible mixture temperature is $T_2 = 300\text{ K}$. Numerical integration step by x – was equal to 0.001, to ψ – 0.02. When moving to a dimensionless coordinate, r the data were presented with a constant step of 0.02.

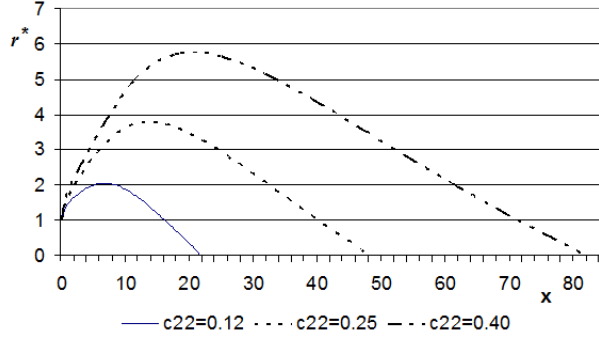


Figure-1. - Flame front shapes at $(c_2)_2 = 0.12, 0.25, 0.4$

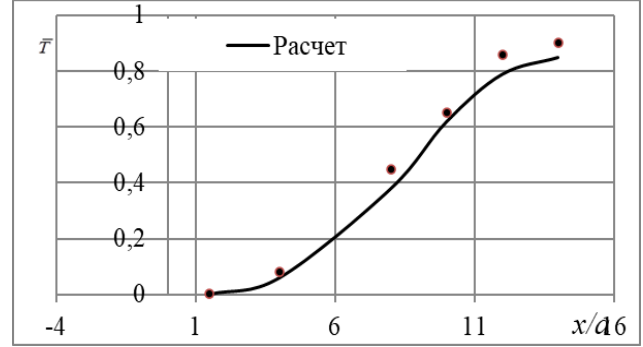


Figure-2. - Comparison of the calculation of axial excess temperature with experiment [5]:

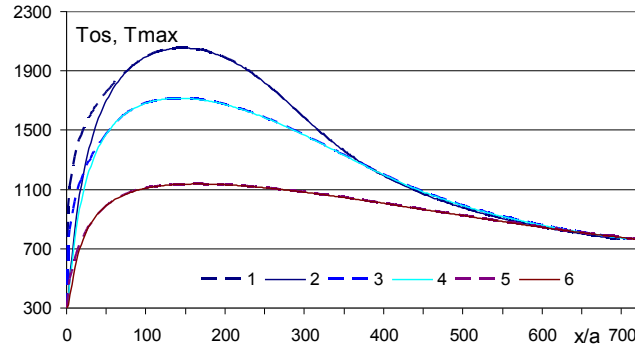


Figure-3. - Curves of the highest (dashed) and axial (solid) temperature values along the dimensionless length of the torch at different radii of the fuel nozzle: 1, 2 – 0.05 m; 3, 4 – 0.01 m; 5, 6 – 0.001 m

At $a = 0.001\text{ m}$ the highest radial coordinate of the highest temperature (690 K) in the section $r/a \approx 2.6$ is observed at $x/a \approx 14$. The combination of the highest and axial temperatures (at 867 K) occurs at $x/a \approx 30$. The highest temperature in the calculation area is 1134 K, and the axial temperature at the end of the conventional plume (at $\tilde{H}_{ocb} = 0.01$) is 756 K.

Calculations were also carried out for the values of the initial fuel concentration $(c_2)_2 = 0.120, 0.25, 0.40$ and 1.00. The shapes of the flame fronts of the first of these three options are shown in Fig.1. For four calculation options, the end of the flame front corresponded to the dimensionless longitudinal coordinates 21.428, 47.418, 80.892 and 281.150.

With an increase in the concentration of fuel in the mixture of the main flow, expansion and elongation of the flame front are observed.

4. Findings and conclusion

In the case of turbulent combustion of methane in a cocurrent air flow with a finite exit velocity from the nozzle, methods of introducing Schwab-Zeldovich functions and relatively excess enthalpy

were used. To solve the problem of the propagation of an axisymmetric fuel jet in a cocurrent air flow in Mises variables, a numerical method based on finite-difference schemes is proposed. For this, the Secundov turbulence model is used $\nu_t - 92$.

An approximation scheme of second order accuracy in both coordinates is used to solve the problem numerically. This gives a more detailed understanding of the object and the physical nature of the combustion process. For longitudinal velocity and other indicators of heat and mass transfer processes, algorithms and a calculation program with internal and external iterations were developed. The computational experiments in this article and field experiments show that the flame front expands and lengthens with increasing fuel concentration in the main flow mixture.

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SIMULATION OF TURBULENT FLOW AROUND A THIN RECTANGLE USING $k - \varepsilon$ TURBULENCE AND ENVIRONMENT MODELS COMSOL MULTIPHYSICS

Hamdamov M.¹, Muzaffarov S.¹.

¹Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev, Tashkent, Uzbekistan

E-mail: mmhamdamov@mail.ru

Annotation. This article presents the results of a study of the aerodynamics of a vertical - axis wind turbine blade by numerically solving the Reynolds-averaged Navier-Stokes equations and using the finite volume method. Some results of calculating the flow around a rectangular airfoil using standard Comsol solvers are presented. Multiphysics with in $k - \varepsilon$ turbulence models at Reynolds numbers $10\,000 < Re < 2\,000\,000$ and angle of attack $\alpha = 0^\circ - 20^\circ$. Speed results were obtained for different angle values.

Keywords. vertical axis wind turbine (VAWT); Navier-Stokes equations; turbulence models; numerical modeling; computational experiment; field of substances .

1. Introduction

Recently, various measures have been taken in the fuel and energy industry of Uzbekistan aimed at further increasing the efficiency of energy use both in industry and at home, as well as introducing more economical technologies for the use of electricity and renewable energy sources.

Solar, hydraulic, wind, geothermal and biomass energy are the main components of Uzbekistan's renewable energy sources. The research results showed that the technical potential of the Republic of Uzbekistan for the use of renewable energy sources is 180 million tons of oil equivalent, which is more than three times its annual energy demand.

Uzbekistan has begun to widely use non-traditional and renewable energy sources. The use of wind energy is one of their main types, which is environmentally friendly and affordable.

By 2030, Uzbekistan plans to produce a quarter of its electricity from renewable energy sources. To attract private capital, the country's government offers a wide range of benefits , including long-term guarantees for the purchase of green energy. In addition, Uzbekistan works with Chinese and other leading companies for the production and installation of solar and wind panels.

The large power industry mainly uses horizontal axis wind turbines. But in populated areas and small industrial facilities remote from the power supply network, vertical axis wind turbines can be used. They have a simpler design than horizontal axis machines, and their lower blade speeds reduce safety and noise concerns. Although vertical axis turbines do offer significant operational advantages, development has been hampered by the difficulty of modeling the aerodynamics involved as well as their rotating geometry [1]. Torque is transmitted by the shaft from the blades to the generator. Like ancient sailing ships, the aerodynamic power of a modern wind turbine is based on lift rather than drag.

Two rectangular orifice wing models were designed and tested in the Duke University wind tunnel to better understand the effects of damage [2]. A rectangular hole is used to simulate damage. A wing with a hole is structurally modeled with a thin elastic plate using the finite element method. The unsteady aerodynamics of a plate wing with an opening is modeled using the doublet lattice method. The equations of aeroelastic motion are derived using the Lagrange equation. The flutter boundary is found using the Vg method. The location of the opening affects the mass, stiffness, aerodynamics and therefore the aeroelastic properties of the wing. It is shown that linear theoretical models are capable of predicting critical flutter speed and frequency, confirmed by wind tunnel tests.

Turbulent separated flows are an important part of modern aerodynamics due to the need to analyze non-standard flight conditions and solve other problems inextricably linked with the phenomenon of flow separation from a streamlined surface [3–5]. It is now known that numerical simulations of separated flows can be performed using various turbulence models and even an ideal fluid model, where numerical diffusion is formed as a result of energy dissipation effects. It has also been noted in the literature that numerical diffusion can be used to model the effects of turbulence, which improves the convergence and stability of the mathematical calculation [5-10].

The principle of summarization of features, proposed in the study [11], is used to determine the flow around profiles when vortices and sources are distributed over the surface of load-bearing elements. Here it is applied to the three-dimensional case.

A prototype of a wind turbine with a vertical axis has been developed in the Laboratory of Mechanics of Fluid, Gas and Hydraulic Drive Systems of the Institute of Mechanics and Seismicity of the Academy of Sciences of the Republic of Uzbekistan. This sample can operate at low wind speeds and can be installed throughout the republic where there is sufficient wind potential.

This paper presents simulation results for a basic vertical axis wind turbine calculated using the commercial finite volume code Star-CCM+. Emphasis was placed on the dynamic characteristics of stall and wake formation, which have the greatest impact on turbine performance. A model is developed to reproduce the blade-wake interaction characteristic of higher tip speeds, which significantly improves the accuracy of the blade-element momentum model.

To numerically solve the system of initial nonstationary equations and the turbulence mode $k - \varepsilon$. The finite element method and standard COMSOL Multiphysics 6.1 solvers were used.

Some results of calculating the flow around a rectangular airfoil at Reynolds numbers $10\,000 < Re < 2\,000\,000$ and angle of attack $\alpha = 0^\circ - 20^\circ$ are presented. Speed results for different angle values showed that the implementation of the Comsol software package Multiphysics provides good convergence, stability and high accuracy of the model.

2. Physic-mathematical formulation of the problem

The flow of a two-dimensional turbulent air flow around a rectangular profile is considered. The physical picture of the flow under study and the configuration of the calculated fields are shown in Fig.1.

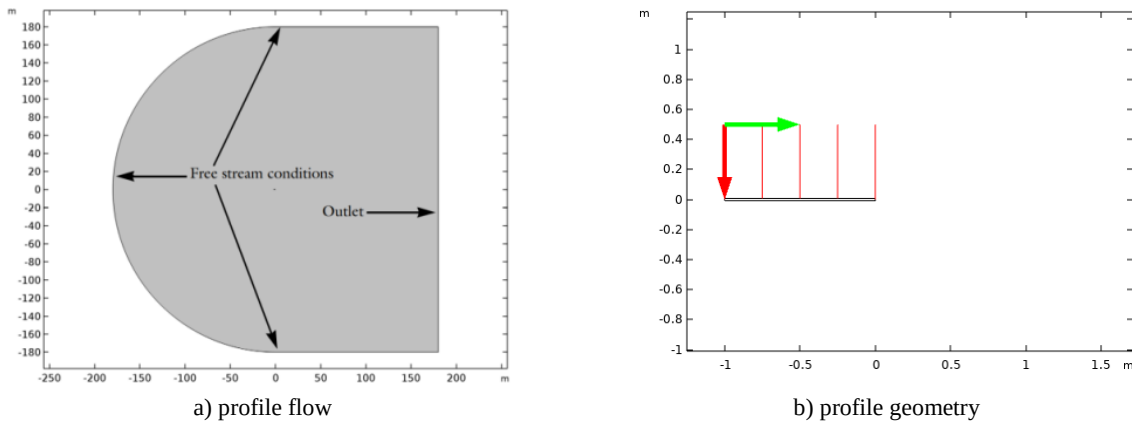


Figure 1. - Scheme of calculation areas

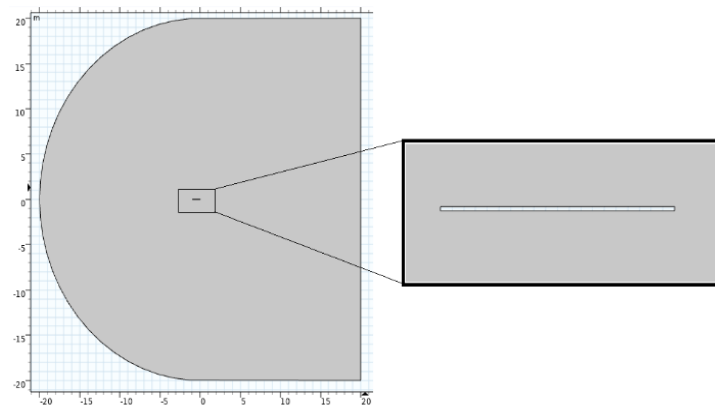


Figure 2. - Flow profile geometry

For the flow problem around a rectangular airfoil, the Reynolds number based on the chord length is $10000 < Re < 2 \times 10^6$. Chord length $c = 1 \text{ m}$. Options for the angle of attack of the profile from $\alpha = 0^\circ$ to $\alpha = 20^\circ$.

Numerical modeling takes into account the following laws of conservation and transfer of substances.

Equation of conservation of mass. This is the continuity equation, also called the general mass conservation equation [10-15]:

$$\frac{d\rho}{dt} + \nabla(\rho\vec{v}) = 0.$$

here ρ is the density of the liquid; $\vec{v}$ – velocity vector.

Equation of conservation of momentum. The general equation can be represented as follows [15-20]:

$$\frac{d}{dt}(\rho\vec{v}) + \nabla(\rho\vec{v}) = -\nabla p + \nabla\vec{T} + \rho\vec{g} + \vec{F},$$

where p – static pressure; $\vec{g}$ and $\vec{F}$ are the gravitational force of the body and the external forces of the body, such as the influence of the dispersed phase, respectively.

The stress tensor $\vec{T}$ is taken in the following form:

$$\vec{T} = \mu \left[\left(\nabla\vec{v} + \nabla\vec{v}^T - \frac{2}{3} \nabla\vec{v}I \right) \right],$$

where μ is molecular viscosity; I is the unit tensor.

Laminar flows are characterized by low Reynolds numbers. The higher the Reynolds number, the more likely the flow is to be turbulent. The Reynolds number gives a measure of the relative importance of inertial and viscous forces: for a wing it is defined as:

$$Re = \frac{\rho ul}{\mu}.$$

To find out whether the flow is laminar or turbulent, we need to calculate the Reynolds number:

$$Re = \frac{\rho ul}{\mu} = \frac{1.2043 \times 20 \times 1}{1.81397 \times 10^{-5}} = 1.327805 \times 10^6.$$

As can be seen from this large Re value, a developed turbulent flow regime is expected. Therefore, we should select a turbulent model that represents this process and use it to formulate and solve this problem.

Effect of flow turbulence. The presence of walls is critical to the specificity of turbulent flows. Obviously, the average velocity field is affected by the non-slip condition, which must be satisfied on the wall.

Viscous damping very close to the wall reduces tangential velocity, and kinematic locking reduces normal vibrations. However, the kinetic energy of turbulence caused by large mean velocity gradients rapidly increases turbulence towards the outer part of the near-surface region [14]. RSM and LES models $k - \varepsilon$ are best valid for turbulent core flows. Thus, it is necessary to think about how to make these models suitable for wall flows. Models Spalart and Almaras and $k - \omega$ were designed for use throughout the entire boundary layer if the grid resolution is sufficient. This simulation experiment used a turbulent model $k - \varepsilon$ with standard wall function.

This choice is due to the possibility of a close-wall grating of the generated mesh. Turbulence model $k - \varepsilon$ is one of the most common options. This is a two equation model; It uses two additional transport equations to represent the turbulent properties of fluid flows. However, the SST model $k - \omega$

was also used in a simulation which did not produce realistic results and was therefore not consistent with the current model.

Modified model $k - \varepsilon$. Unlike well-known works, here it is proposed to use a modified $k - \varepsilon$ model:

$$\begin{cases} \frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - 2\rho \varepsilon M_t^2 + S_k, \\ \frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho \varepsilon u_j) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S \varepsilon - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}} + C_{1\varepsilon} \frac{\varepsilon}{k} C_{3\varepsilon} G_b + S_\varepsilon. \end{cases}$$

which contributes to a more adequate description of heat and mass transfer processes. The notation used here is

$$C_1 = \max \left[0.43, \frac{\eta}{\eta + 5} \right], \quad \eta = S \frac{k}{\varepsilon}, \quad S = \sqrt{2S_{ij}S_{ij}}, \quad \mu_t = \rho C_\mu \frac{k^2}{\varepsilon}, \quad C_\mu = \frac{1}{A_0 + A_S \frac{kU^*}{\varepsilon}},$$

$$U^* \equiv \sqrt{S_{ij}S_{ij} + \tilde{\Omega}_{ij}\tilde{\Omega}_{ij}}, \quad \Omega_{ij} = \bar{\Omega}_{ij} - 2\varepsilon_{ijk}\omega_k, \quad A_S = \sqrt{6} \cos \phi, \quad \phi = \frac{1}{3} \cos^{-1}(\sqrt{6}W),$$

$$W = \frac{S_{ij}S_{jk}S_{ki}}{\tilde{S}^3}, \quad \tilde{S} = \sqrt{S_{ij}S_{ij}}, \quad S_{ij} = \frac{1}{2} \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right), \quad G_k = -\rho \overline{u_i' u_j'} \frac{\partial u_j}{\partial x_i}, \quad S \equiv \sqrt{2S_{ij}S_{ij}},$$

$$G_b = \beta g_i \frac{\mu_t \partial T}{\text{Pr}_t \partial x_i}, \quad a_0 = 1/\text{Pr} = k/\mu c_p, \quad \beta = -\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_p, \quad G_b = -g_i \frac{\mu_t}{\rho \text{Pr}_t} \frac{\partial \rho}{\partial x_i}, \quad a = \sqrt{\gamma RT}.$$

$$\text{Pr}_t = 1/a_t \quad M_t = \sqrt{\frac{k}{a^2}}.$$

The empirical constants $k - \varepsilon$ of the model take standard values: $C_{1\varepsilon} = 1.44$, $C_2 = 1.9$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.2$, $A_0 = 4.04$. $k - \varepsilon$ the model appears to be the best model of first-level closure turbulence. For the turbulent energy mass density k , a system of two nonlinear diffusion equations is used to explain its turbulent quantities and turbulent energy dissipation rates ε . More than 30 years ago, a simpler version of this model appeared [14]. Since then, the model $k - \varepsilon$ has been widely used to calculate a wide range of problems, mainly to describe shear incompressible turbulence.

Calculation grids. In this work, a thickening of the mesh was used near the surface of the profile shown in Fig.3.

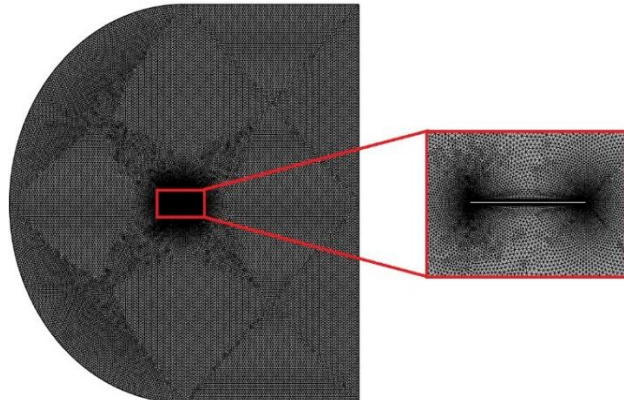


Figure 3. - Calculation mesh of flow around the profile

For the area of flow around the profile, a computational mesh with 60200 elements was used. For the system of equations (1)-(3), obvious boundary conditions for adhesion on solid walls are specified. The output accepts extrapolation conditions for all parameters. At the inlet, uniform profiles of the longitudinal velocity component c were applied $V_x = U_0$, the transverse velocity component and pressure were assumed to be zero $V_y = P = 0$. Also at the input are the values of relative velocities (disturbances): $\mathcal{G}_x = 0.004$, $\mathcal{G}_y = 0$.

The dimensions of the initial data for calculating the dimensional parameters were taken as follows: wind speed – m/s, angular speed – rad/s, air density – kg/m³ (Fig. 4).

Label: Parameters 1

Name	Expression	Value	Description
U_inf	20[m*s^-1]	20 m/s	Free-stream velocity
rho_inf	1.2043[kg*m^-3]	1.2043 kg/m ³	Free-stream density
mu_inf	1.81397e-5[kg*m^-1*s^-1]	1.814E-5 kg/(...)	Free-stream dynamic vis...
L	20[m]	20 m	Domain reference length
c	1[m]	1 m	Chord length
k_inf	0.1*mu_inf*U_inf/(rho_inf*L)	1.5062E-6 m ² /s ²	Free-stream turbulent kin...
om_inf	10*U_inf/L	10 1/s	Free-stream specific dissi...
alpha	0	0	Angle of attack
Re	U_inf*c*rho_inf/mu_inf	1.3278E6	

Figure 4. - Source data window with dimensions and values

3. Solution method and results

Time-averaged and longitudinal velocity (or flow velocity) profiles allow the flow characteristics to be assessed at various points downstream of the profile. Time-averaged and normalized longitudinal velocity profiles are evaluated at five points ($x/c = 0, 0.25, 0.5, 0.75$, and 1) behind the profile.

In Fig. 5 shows graphs of longitudinal velocity in various sections at angle of attack values $\alpha = 2^\circ, 6^\circ, 10^\circ, 14^\circ, 18^\circ, 20^\circ$. The speed of the undisturbed flow was 20 m/s, and the Reynolds number was 1,327,800.

At the angle of attack of the wing, $\alpha = 2^\circ$ the highest longitudinal speed of 22.1 m/s is observed at $x/c = 0.25$, and the lowest speed is at $x/c = 1$. With this installation of the wing, it is clear that high speed is generated on the opposite side of the slope and in the initial part of the plate. As the plate moves upward, the influence of the plate weakens and the flow again acquires its initial speed. The change in speed along the axis y is not that great, i.e. the results did not differ much from the results of flow around a plate at an angle $\alpha = 0^\circ$, since in this case the angle of attack of the wing is not large.

The highest flow rate at $\alpha = 6^\circ$ also has at point at $x/c = 0.25$. When $x/c = 1.00$ he reached a low speed. In this case, the maximum speed was 25.07 m/s. At this angle of attack of the wing, the flow velocity at the point $x/c = 0.5$ equal to 23.8 m/s. After $y = 1$ m, the results of the stream took on the same value as in the original stream.

Maximum speed at a point $x/c = 0.5$ near the wing at $\alpha = 10^\circ$ was 22.9 m/s. At points $x/c = 0.25$ And $x/c = 0.75$ the longitudinal velocity took on almost the same value. There is a noticeable tendency that the highest velocity values in the sections move upward in height and in this case, after $y = 1.5$ m, the results of the longitudinal velocity tend to their undisturbed value. It has been found that this angle of attack value produces the desired torque results to drive the rotational motion of the wing at this distance.

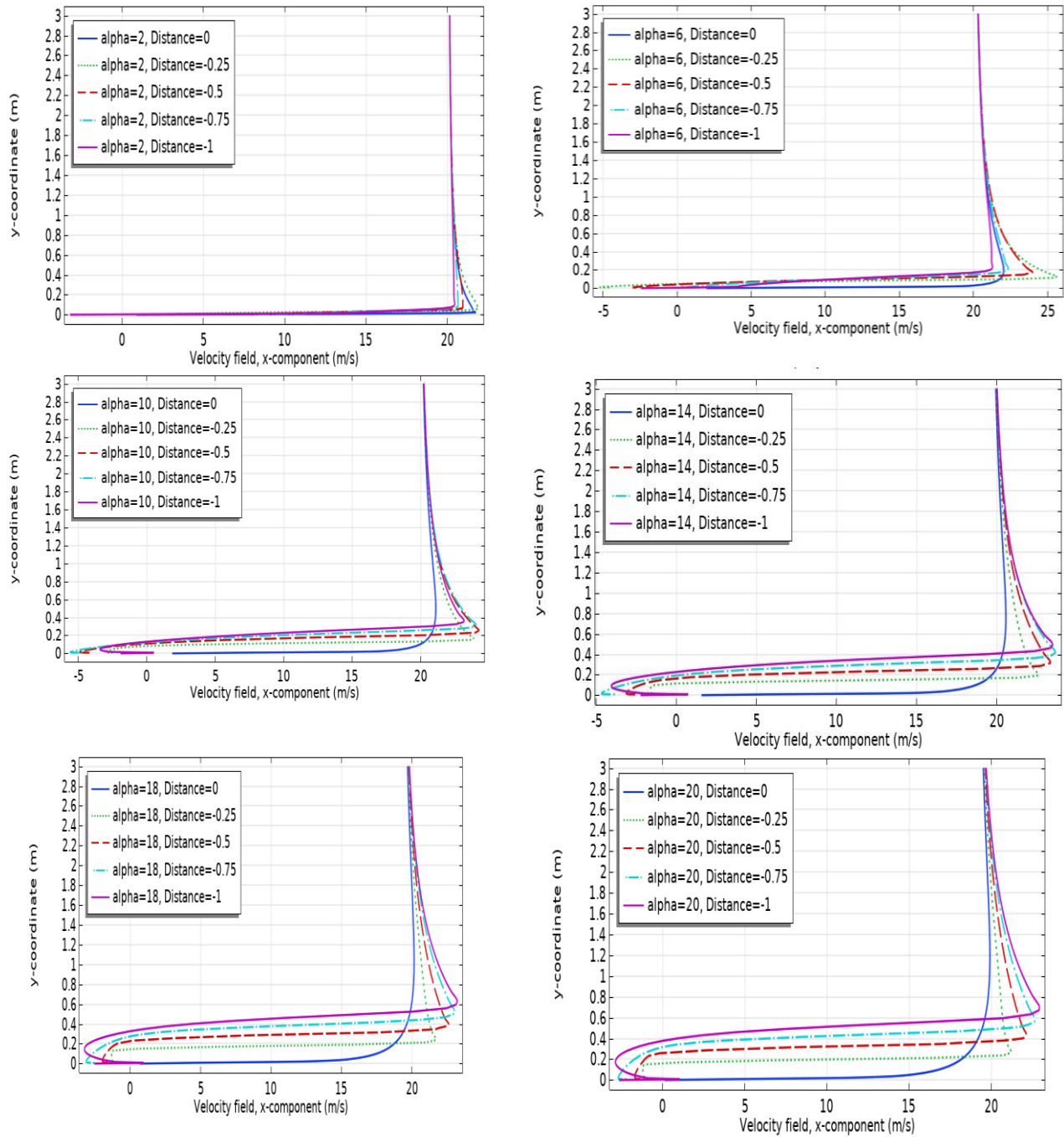


Figure 5. - Longitudinal velocity field at sections $x/c = 0, 0.25, 0.5, 0.75, 1$ at $\alpha = 2^\circ, 6^\circ, 10^\circ, 14^\circ, 18^\circ, 20^\circ$

When the angle of attack of the wing $\alpha = 14^\circ$ is at the point $x/c = 0.75$ the maximum speed observed was 23.2 m/s. At points $x/c = 0.5$ and $x/c = 1.0$ the speed took on almost the same value. At the point $x/c = 0$ it reaches low speed. After height $y = 1.85$ m longitudinal velocity is practically not disturbed. And in this case there is a favorable opportunity for wing rotation.

The tendency for the highest velocity value to shift upward in height at the angle of attack $\alpha = 18^\circ$ reaches its completion: the maximum speed of 23.2 m/s is achieved at the end point $x/c = 1.0$, i.e. at the end of the wing. At the same time, at points $x/c = 0.75$ and $x/c = 0.5$ the speed took on almost equal values. As the angle of attack increases, the size of the generated vortex increases. Calculations up to a height of

3 meters showed that in this case, after $y = 2,2$ m disturbances in the longitudinal flow velocity are not significant.

At the angle of attack of the wing $\alpha = 20^\circ$, the maximum speed equal to 23.8 m/s is observed at the end of the wing (i.e. at $x/c = 1,0$). At the point $x/c = 0,75$ the speed has a value of 23.5 m/s. At a point $x/c = 0$ (at the beginning of the wing) it reached a small value. As the angle of attack increases, you can also see an increase in drag. Results at 3 m downstream of the wing were analyzed as this is the critical span for wing performance.

4. Conclusion

At different angles of attack, longitudinal velocity fields were obtained and analyzed in a two-dimensional setting. Modeling of wall friction stresses can be reliable if the boundary layer characteristics are defined correctly. An erroneous determination of the boundary layer characteristics can lead to a distortion of the pressure coefficient corresponding to the aerodynamic force factors.

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CONSTRUCTION OF A MATHEMATICAL MODEL OF THE PROBLEM OF THE THEORY OF FILTRATION OF ANOMALOUS, ANOMALOUSLY STRUCTURED FLUIDS IN HYDRO-DYNAMICALLY CONNECTED TWO RESERVOIR ENVIRONMENTS

Kayumov Sh.¹, Ziyadullaeva Sh.S.¹, Bekchanov Sh.E.¹, Khusanov E.A.¹

¹Tashkent State Technical University, Tashkent, Uzbekistan

E-mail: kayumovmatemic@gmail.com.

Abstract. The paper studied the use of a multiparametric mathematical model of nonlinear problems of the theory of filtration of abnormal and abnormally structured fluids in hydro-dynamically connected two-layer formations with their individual characteristics. The constructed models and algorithms for their solution are tested on hypothetical data.

Keywords: anomalous, multivariable, connected, structured, laws, multilayer, formations, algorithms, flows, flow, computational, boundaries.

1. Introduction

Work [1-4] is devoted to the study and construction of a mathematical model for the problem of linear and nonlinear filtration of fluids in single-layer and multilayer porous media, where the physical characteristics and mathematical aspects of the movement of Newtonian and non-Newtonian fluids are considered. There is more work [5-8] where the specific problems of the theory of filtration in layered media with the presentation of methods for their solution are studied. The analysis of existing studies on the construction of mathematical models gave impetus to the optimization of the modeling process and led to multivariable models [9,10].

The use of a parametric function made it possible, in one multi-parameter mathematical model, to cover all existing types of mathematical models corresponding to various types of filtration laws. The introduction of special parameters into the boundary conditions made it possible to display the equation one-to-one with its various boundary conditions. The constructed multiparameter model made it possible to develop parameterized multifunctional computational algorithms and the corresponding mathematical software. The multi-parameter model made it possible to classify models and solution algorithms in single-layer systems. As a result, researchers were able to use the available data for a specific existing field, without resorting to the modeling stage, to select from a multi-parameter model and obtain the boundary value problem that they needed and corresponded to the parameters of a given field. Since the selected model is ready, all that remains is to enter data and carry out computational procedures.

On the other hand, a multiparameter mathematical model has become the basic elements of computational procedures and it has made it possible to use all stages of computing technology [10-12]. Let us apply this optimization multiparameter model to a two-layer, connected porous medium.

2. Method.

Let us assume that the filtration area ($\bar{\Omega}$) under study consists of two layers (D_1 and D_2), where D_1 one is highly permeable (where the horizontal characteristics of the formation and fluid prevail over the vertical characteristics and, due to the fluid movement, occurs in horizontal directions), and the other D_2 is poorly permeable (where the vertical characteristics prevail over the horizontal and therefore, fluid movements occur vertically). In addition, the roof of the area D_1 coincides with the bottom of the area D_2 and it is considered permeable.

Let the porous medium of the first layer (D_1) be saturated with anomalously structured fluids, and the second layer (D_2) be saturated with fluids and have viscous-plastic properties.

If fluid is withdrawn from an area D_1 , then with the start of reservoir operation (at $t=t_1$) a disturbance of the filtration area occurs in it, which leads to fluid flow from D_2 along the entire contact boundary to the lower area D_1 , and these flow values depend on the properties and velocity of the fluid,

on the amount of fluid extraction, as well as on the filtration characteristics and heterogeneity of the boundary structures of the formation.

This is a physical problem, mathematically formulated as follows: it is necessary to find continuous unknown functions $U(x, \bar{z}, t)$, $V(\bar{x}, z, t)$ and unknown boundaries $R_i(x, \bar{z}, t)$, $G_j(x, t)$ from the following system:

$$\frac{\partial}{\partial x} \left(x^\alpha \frac{hk(x)}{\mu} A(|\nabla u|, \beta) \right) \frac{\partial u}{\partial x} = M(x, t, u) \frac{\partial u}{\partial t} + B(|\nabla v|, \bar{\beta}) \frac{\partial V}{\partial x} \Big|_{z=h_1} + f(x, t), \quad t > 0, x \in D_1. \quad (1)$$

and

$$\frac{\partial}{\partial z} \left(\frac{K(z)B(|\nabla v|, \bar{\beta})}{v_1} \frac{\partial V}{\partial z} \right) = M_2(\bar{x}, z, t) \frac{\partial V}{\partial t}, \quad t > 0, z \in D_2. \quad (2)$$

with initial conditions

$$u(x, \bar{z}, 0) = V(\bar{x}, z, 0) = u_0(x, z) = \text{const}. \quad (3)$$

boundary conditions

$$a_1 \frac{x^{\tilde{\alpha}} hk(x)}{\mu b} A(|\nabla u|, \beta) \frac{\partial u}{\partial x} \Big|_{x=x_0} + b_1 u(x, \bar{z}, t) \Big|_{x=x_0} = \varphi_0(x_0, t), \quad t > 0, \quad (4)$$

$$\frac{x^{\tilde{\alpha}} hk(x)}{\mu b} A(|\nabla u|, \beta) \frac{\partial u}{\partial x} \Big|_{x=x_L} = \varphi_L(x_L, t), \quad t > 0, \quad (5)$$

and conditions on unknown moving boundaries:

$$C_1 A(|\nabla u|, \beta) \Big|_{x=R_r^\pm} = 0, \quad C_1 (u(x, t) \Big|_{x \in D} - u_0(x)) = 0, \quad (7)$$

$$x = R_r^\pm \pm 0, \quad r = \{0, 1\}; \quad (r = 0, \quad x = R_0^+ + 0, \quad r = n, \quad x = R_n^- - 0),$$

$$C_2 \left(|\nabla u| \Big|_{x=R_r^\mp \mp 0} = |\nabla u| \Big|_{x=R_r^\mp \pm 0} = \beta \right), \quad C_2 \left(u(x, t) \Big|_{x=R_r^\mp \mp 0} - u(x, t) \Big|_{x=R_r^\mp \pm 0} \right) = 0, \quad r = \overline{0, n}.$$

$$C_3 \left(|\nabla u| \Big|_{x=R_r^\pm \mp 0} - \beta \right) = 0, \quad C_3 (u(x, t) \Big|_{x \in D} - u_0) = 0, \quad (8)$$

$$C_4 \left(|\nabla u| \Big|_{x=R_{1\sigma}^\pm - 0} = |\nabla u| \Big|_{x=R_{1\sigma}^\pm + 0} = \beta_1 \right), \quad C_4 \left(|\nabla u| \Big|_{x=R_{2\sigma}^\pm - 0} = |\nabla u| \Big|_{x=R_{2\sigma}^\pm + 0} = \beta_2 \right), \quad (9)$$

$$C_4 \left(u(x, t) \Big|_{x=R_{1\sigma}^\pm - 0} - u(x, t) \Big|_{x=R_{1\sigma}^\pm + 0} \right) = 0, \quad C_4 \left(u(x, t) \Big|_{x=R_{2\sigma}^\pm - 0} - u(x, t) \Big|_{x=R_{2\sigma}^\pm + 0} \right) = 0, \quad (10)$$

$$C_4 \left(|\nabla u| \Big|_{x=R_{1r}^\pm - 0} = |\nabla u| \Big|_{x=R_{1r}^\pm + 0} = \beta_1 \right), \quad C_4 \left(|\nabla u| \Big|_{x=R_{2r}^\pm - 0} = |\nabla u| \Big|_{x=R_{2r}^\pm + 0} = \beta_2 \right), \quad (11)$$

$$r = \overline{1, n-1}, \quad \sigma = \{0, 1\}.$$

$$C_4 \left(u(x, t) \Big|_{x=R_{1r}^\pm - 0} - u(x, t) \Big|_{x=R_{1r}^\pm + 0} \right) = 0, \quad C_4 \left(u(x, t) \Big|_{x=R_{2r}^\pm - 0} - u(x, t) \Big|_{x=R_{2r}^\pm + 0} \right) = 0, \quad (12)$$

as well as boundary conditions

$$B(\Delta v, \beta) \frac{\partial V}{\partial z} \Big|_{z=h_2} = 0, \quad t \geq 0. \quad (13)$$

In problem (1)-(13) the function $A(\eta)$ is a parameterized function and has the form:

$$A(\eta) = \frac{\left[\eta^\lambda - \gamma^\lambda (2 - \lambda) \right]^\theta \cdot \eta^{-1} \cdot \beta^{\theta-1}}{\left\{ 1 + \left[(\lambda - \gamma)^\lambda + \eta^\lambda \right]^{\frac{1}{2}} \right\}^{\lambda-1}}. \quad (14)$$

Where, parameters $\theta, \lambda, \gamma, \eta$ and C_1, C_2, C_3, C_4 takes values in accordance with Table 1 and thereby determines the specific type of function $A(\eta)$ for each filtration law, corresponding to the approximation of fluid velocity and pressure gradient [4,9]

Table 1

laws	$\overline{\Omega}$	θ	λ	γ	η	C_1	C_2	C_3	C_4	Note
I	Ω_r	1	1	1	d_1	1	0	0	0	Law with initial gradient [6]
	Ω	1	1	η	d_2	1	0	0	0	
II	Ω_r	1	1	μ_0	d_1	0	1	0	0	Polygonal law [7]
	Ω	1	1	0	d_2	0	1	0	0	
III	Ω	1	2	1	η	0	0	0	0	Hyperbolic law [7]
IV	Ω	1	2	2	η	0	0	0	0	Haeg's Law [8]
V	Ω	1	1	0	η	0	0	0	0	Darcy's Law [7,13]
VI	Ω_r	1	1	ξ	d_1	0	1	0	0	Curvilinear Law [4]
	Ω	1	2	1	d_2	0	1	0	0	
VII	Ω_r	θ	1	1	d_1	1	0	0	0	Law according to [13,14]
	Ω	1	1	η	d_2	1	0	0	0	
VIII	Ω_r	1	1	0	d_1	0	0	1	0	Law according to [5]
	Ω	1	1	η	d_2	0	0	1	0	
IX	Ω_r	1	1	μ_0	d_1	0	1	0	0	Variant of the curvilinear law [4]
	Ω	1	1	2	d_2	0	1	0	0	
X	Ω	$\alpha + 1$	1	0	η	0	0	0	0	Power law [13]
XI	Ω	1	2	2	η	0	0	0	0	Khristanovich's law [15]
XII	Ω_1	1	2	2	g_1	0	0	0	1	Structured law [16,17]
	Ω_2	1	1	ξ_2	g_2	0	0	0	1	
	Ω_3	1	1	0	g_3	0	0	0	1	
XIII	Ω_1	1	2	1	η	0	0	0	1	Variant of the structured law [9-18]
	Ω_2	1	1	ξ_2	g_2	0	0	0	1	
	Ω_3	1	1	0	g_3	0	0	0	1	

In Table 1 the parameters $d_i (i = 1, 2)$, $g_i (i = 1, 3)$, $\xi_i (i = 1, 3)$ are similar to [10].

The parameters C_1, C_2, C_3, C_4 set the conditions on the moving boundaries of disturbances, as well as on the boundaries between zones of mobility based on the choice of the filtration law, $\xi, \xi_1, \xi_2, \beta_1, \beta_2, \beta_3$ determine the coefficients for critical pressure gradients.

In problems (1)–(14), the coefficients and parameters $k(x), \mu, k(z), M(x, t), M_1(z, t), f(x, t)$ are specified variables and functions that describe the characteristics of the medium and fluid [4,8,14], in addition

$$\Omega = D_1 \cup D_2, D_1 = \overline{\Omega} - \sum_{r=0}^n \Omega_r, \Omega_r = \{x: R_r^- < x < R_r^+\}, \overline{\Omega} = \{x: x_0 \leq x \leq x_L\}$$

$$D_2 = \{z: h_1 \leq z \leq h_2\}, w_1 = \{z: h_1 \leq z \leq C_1\} w_2 = \{z: C_1 \leq z \leq C_2\}$$

$$f(x, t) = a_0 \sum_{r=1}^{n-1} q_r \delta(x - x_r), a_1 + b_1 \neq 0. \quad a_1 = \{0; 1\}, b_1 = \{1; 0\}$$

In the case of laws XII and XIII, the filtering areas will be:

$$\Omega_1 = \left\{ x: x_0 < x < R_{01}^+, \bigcup_r (R_{1r}^- < x < R_{1r}^+) \right\}, \Omega_2 = \left\{ x: \bigcup_r (R_{1r}^+ < x < R_{2r}^+; R_{2r}^- < x < R_{1r}^-) \right\},$$

$\Omega_3 = \left\{ x: \bigcup_r (R_{2r}^+ < x < R_{2(r+1)}^-) \right\}$. $u = \{u_1; u_2; u_3\}$ If $x = \{\Omega_1; \Omega_2; \Omega_3\}$ respectively, for structured fluids, [12,19,20], if problems 1 or 11 are considered, then the filtration area around the source will be Ω_1 and Ω_1 where $\Omega_1 = \{x: x_r < x < R_{1r}^-\}$, $\Omega_2 = \{x: R_{1r}^+ < x < R_{2r}^-\}$. $\varphi_0(x_0, t)$ and $\varphi_L(x_L, t)$ may take values or depending on the physics of the reservoir and development conditions, describing this boundary value problem.

It is obvious that problem (1) – (14) is nonlinear and analytical solutions are practically impossible, therefore, numerical methods are used to solve it: the iteration method, the method of straight lines and the difference version of flow sweeping [11,16-21].

Since the multi-parameter model (1)-(14) includes 13 different laws, the calculation algorithms are developed taking into account belonging to a certain group of the mathematical model, since the computational processes for them are different and have their own specifics.

According to the specifics of filtering, problems I, VII, VIII are combined into the first group and they are characterized by the existence of disturbed and undisturbed regions. Laws II, VI, IX, XII, XIII, included the second group where filtration occurs throughout the entire area with unknown moving boundaries between different filtration zones. The third includes laws: III, IV, V, X, XI where there are no boundaries of disturbances and each of them has an individual characteristic nonlinearity that must be taken into account.

The sequence of calculations for these groups is different and has its own characteristics, and multi-parameter computational algorithms are built taking these phenomena into account. Computational procedures also have a table similar to Table 1, where coefficients and parameters are taken based on membership in classification groups.

3. Results and discussions.

To test the computational algorithms, the following test data was used:

At the point $x_0 = 0$ there is a source with a flow rate.

$$q_0 = 0,0864, \Delta\tau = 0,00675; u_0 = 1, h = 25 \cdot 10^{-3}, \beta = 0,01, k = 0,8, L = 4000m,$$

$$M = 1, \mu_0 = \nu_0 = 1.$$

Table 2, for Law VI, shows the values of the function for different t and x as well as the corresponding positions of the disturbance boundary.

Table 2

x \ t	10^{-2}	10^{-1}	$3 \cdot 10^{-1}$	$7 \cdot 10^{-1}$
0,00	0,98883	0,96660	0,94040	0,90314
0,25	0,99722	0,97650	0,95004	0,91140
0,50	0,99886	0,99200	0,97258	0,94074
0,75	0,99983	0,99712	0,98403	0,95784
1,00	0,99999	0,99742	0,98712	0,96003
$l(t)$	0,2411	0,5852	0,7531	0,8820

In order to study the rate of propagation of disturbance boundaries between small and large disturbance areas, the same problem is considered for $q_0 = 0$ and the well is located at the point

$x_1 = 0,5$; with the flow rate $q_1 = 0,1718$ and the initial disturbance boundary $l_1^{(0)} = 0,4839$ (it is calculated using the formula $l_1^{(0)} = x_1 - l_1^{(0)} \cdot \frac{q_1}{2}$). Some of the results of the movement of the left and right borders are shown in Table 3.

Table 3

t	l_1^-	l_1^+
$1 \cdot 10^{-3}$	0,4211	0,5825
$5 \cdot 10^{-3}$	0,3323	0,6731
$1 \cdot 10^{-2}$	0,2714	0,7340
$5 \cdot 10^{-2}$	0,0917	0,9122
$8 \cdot 10^{-2}$	0,0000	1,0000

Table 4

Laws \ t	$675 \cdot 10^{-5}$	$675 \cdot 10^{-4}$
I	0,9813	0,8133
VI	0,9850	0,8267
II	0,9861	0,8297
III	0,9891	0,8328
V	0,9901	0,8341

The average function values $u(x, t)$ for the selected filtration laws in the integral balance sense are given in Table 4.

Considered when the sources are at points $x_1 = 0.25$, $x_2 = 0.5$ and $x_3 = 0.75$, $q_1 = q_2 = q_3 = 0.29629$, zero approximation of the disturbance boundary $R_1^{(0)} = 0.24168$; $R_2^{(0)} = 0.491680$; $R_3^{(0)} = 0.24168$; $x = 0.8$, $\beta = 0.8$, $\psi_0 = 1$, $H = 4000$; $\Delta\tau = 0.00675$, $\Delta x = 0.00025$.

The calculation results for Law 1 are given in Table 5.

Table 5

$x \backslash t$	$0.675 \cdot 10^{-5}$	$135 \cdot 10^{-4}$	$377 \cdot 10^{-4}$
0,00	1,0000	1,0000	1,0000
0,062	1,0000	1,0000	0,9951
0,125	1,0000	1,0000	0,9389
0,187	0,9829	0,9543	0,8652
0,250	0,9179	0,8811	0,7685
0,3126	0,9829	0,9543	0,8652
0,375	1,0000	1,0000	0,8263
0,437	0,9829	0,9543	0,7989
0,500	0,9179	0,8811	0,7345
0,5625	0,9829	0,9543	0,7989
0,625	1,0000	1,0000	0,8263
0,687	0,9829	0,9543	0,8147
0,750	0,9179	0,8811	0,7685
0,8125	0,9829	0,9543	0,8147
0,9187	1,0000	1,0000	0,9951
1,0000	1,0000	1,0000	1,0000
l_1^-	0,1667	0,1345	0,0565
l_1^+	0,3341	0,3637	0,3655
l_2^-	0,4167	0,3845	0,3637
l_2^+	0,5841	0,6155	0,6363
l_3^-	0,6667	0,6345	0,6214
l_3^+	0,8340	0,8655	0,9445

The Table shows that at $t = 337 \cdot 10^{-5}$ the internal galleries form clear independent regions of disturbance, and at the edges of the region there are undisturbed zones, and the internal boundaries merge.

In this case, the influence of the central source becomes decisive and its boundaries expand, covering a circle with a radius $R = 0.1362$ and center at the point 0.5. The convergence of the iterative process is ensured by 2-3 iterations. When $z > 337 \cdot 10^{-4}$ the boundary of the disturbance reaches the boundary of the region and the second phase of filtering begins.

Table 6

$x \backslash$	0	0,1	0,2	0,3	0,4
I	1,0000	0,9800	0,8931	0,7615	0,6634
III	0,9031	0,8792	0,83912	0,7854	0,7208
IV	0,8813	0,8625	0,8290	0,7840	0,7381
V	0,7982	0,7951	0,7858	0,7709	0,7504
IX	0,9621	0,9213	0,8715	0,8050	0,7611

For various laws table 6 gives the results of calculating the value of the function on the left side of the region when a well is located at the point $x = 0,5$, as well as the time step taken $\tau = 10^{-2}$.

The calculation results allow us to conclude that the curve change of the function for laws III, IV, IX lies between laws V and I, and in this case the minimum pressure will be at the bottom of the source. Table 7 for $m = 0,2$, $\beta = 0,5 \cdot 10^{-5}; 10^{-3}$, $\mu = 0,3$, $m = 0,18$, $x = 0,005$, $\alpha = 1$, $b = 5a$, $h_1 = 80$, $q = 50$, $u_0 = 1$. shows the change in pressure in the region D_1 and D_2 at the point $x = 0$ and the magnitude of the flow from D_2 to D_1 at various values of β . Average pressure values for various values of parameter β and according to t .

Table 7

t	$\beta = 0$	$\beta = 10^{-5}$	$\beta = 10^{-3}$
0,01	0,98150	0,97502	0,92163
0,05	0,95261	0,93121	0,89121
0,2	0,92110	0,90422	0,87125
0,15	0,89216	0,87134	0,84786

The value of the flow between layers from D_2 to D_1 .

t	$\beta = 0$	$\beta = 10^{-5}$	$\beta = 10^{-3}$
0,01	0,12143	0,10346	0,08732
0,05	0,18314	0,16342	0,11425
0,1	0,21210	0,19121	0,16522
0,15	0,25146	0,22751	0,19344

An analysis of this table shows that the values of the flow coefficients depend on the intensity of the selection of the area D_1 they are different for different filtration laws.

4. Conclusion.

Analysis of partial calculation results shows that the proposed multi-parametric model for a two-layer reservoir reflects all existing laws of fluid filtration and is a compact, economical, and also implementable form of representation of mathematical models. The developed model and calculation algorithms can be used in calculating the technical and economic indicators of fluid development in multilayer connected porous media, in particular two-layer media.

It should be noted that when carrying out an iteration to determine the position of the left and right boundaries of disturbances of a point source when considering the law with an initial gradient (law I), you can use a two-way iteration or the method of “shuttle” iterations [4,18] which allows you to quickly determine the position of the boundary, which gives opportunities to clarify the magnitude of fluid flow from area D_2 to area D_1 .

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APPLICATION OF A MODERN TWO-FLUID TURBULENCE MODEL FOR METHANE-AIR SIMPLE JET COMBUSTION

Khaydarov S.I.¹

¹Andijan Maschine-building Institute, Andijan City, Uzbekistan.

E-mail: khaydarovsardor@gmail.com.

Abstract: The work simulates a turbulent compressible multicomponent fluid with non-isothermal chemical reactions based on a two-fluid approach. It is shown that the two-fluid approach leads to a closed system of equations. To substantiate the model, the problem of turbulent non-premixed combustion of methane-air in a jet is considered. The Arrhenius-Magnussen kinetic theory was used to simulate chemical combustion reactions. The obtained numerical results are compared with known experimental data, from which it is clear that the two-fluid approach allows a fairly accurate description of the combustion process.

Keywords: turbulence, two-fluid model, methane-air combustion, concentration, Arrhenius law, diffusion combustion.

1. Introduction

The combustion process of natural gas has many practical applications, including use in internal combustion engines and heat and power generation. Combustion modeling allows one to predict various parameters such as combustion rate, heat release and combustion product formation. One of the main tools for studying flows with chemical reactions is mathematical modeling of turbulent flow of fluids. Despite the fact that turbulence has been studied for a long time, it still remains an unresolved problem in classical physics. Because, as we know, Reynolds' approach leads to an open system of equations and all models aimed at closing this system are called RANS models. Despite the empiricism of these models, they are still an effective tool for research in this area. However, the development of modern science and technology requires more accurate calculation results, which requires the development of more adequate turbulence models.

More recently, in addition to the RANS approach, the so-called two-fluid approach has been developed in [1]. In the works [2,3], based on a new two-fluid approach, authors carried out modeling of turbulent heat transfer of a compressible flow and a multicomponent fluid. The main difference between this approach and RANS is that it leads to a closed system of equations. This paper presents the results of methane combustion in a jet, which are compared with experimental data [4].

The thermodynamics system of gas combustion based on the modern two-fluid approach has the form

$$\begin{aligned}\frac{\partial \rho}{\partial t} + \frac{\partial \rho V_j}{\partial x_j} &= 0, \\ \rho \frac{\partial V_i}{\partial t} + \rho V_j \frac{\partial V_i}{\partial x_j} + \frac{\partial p}{\partial x_i} &= -\frac{\partial \rho g_j g_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\rho v \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) \right], \\ \rho \frac{\partial g_i}{\partial t} + \rho V_j \frac{\partial g_i}{\partial x_j} &= -\rho g_j \frac{\partial V_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\rho v_{ij} \left(\frac{\partial g_i}{\partial x_j} + \frac{\partial g_j}{\partial x_i} \right) \right] + f_i, \\ \rho \frac{\partial H}{\partial t} + \rho V_j \frac{\partial H}{\partial x_j} &= -\frac{\partial \rho h g_j}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\rho k \frac{\partial H}{\partial x_j} \right] + \sum_{m=1}^M M_m Q_m \\ \rho \frac{\partial h}{\partial t} + \rho V_j \frac{\partial h}{\partial x_j} &= -\rho g_j \frac{\partial H}{\partial x_j} + \frac{\partial}{\partial x_j} \left[k_j \frac{\partial h}{\partial x_j} \right] - K_h h, \\ \rho \frac{\partial C_m}{\partial t} + \rho V_j \frac{\partial C_m}{\partial x_j} &= -\frac{\partial \rho c_m g_j}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\rho D \frac{\partial C_m}{\partial x_j} \right] + M_m,\end{aligned}$$

$$\begin{aligned}
\rho \frac{\partial c_m}{\partial t} + \rho V_j \frac{\partial c_m}{\partial x_j} &= -\rho \mathcal{G}_j \frac{\partial C_m}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\rho D_j \frac{\partial c_m}{\partial x_j} \right] - K_c c_m. \\
\mathbf{f} &= \rho C_s \mathbf{rot} \mathbf{V} \times \mathcal{G} - \rho K_f \mathcal{G}, \quad D_j = 3D + 2 \sum_{m=1}^M C_m \frac{|c_m \mathcal{G}_j|}{|\mathbf{grad} C_m|}, \\
\nu_{ij} &= 3\nu + 2 \left| \frac{\mathcal{G}_i \mathcal{G}_j}{\mathbf{def}(\mathbf{V})} \right| \text{ for } i \neq j, \quad \nu'_{ii} = 3\nu + \sum_{k=1}^3 \frac{2I_k}{I} \left| \frac{\mathcal{G}_k \mathcal{G}_k}{\mathbf{def}(\mathbf{V})} \right|, \\
I_k &= \frac{\partial \rho \mathcal{G}_k}{x_k}, \quad I = |I_1| + |I_2| + |I_3|, \quad k_j = 3k + 2 \left| \frac{h \mathcal{G}_j}{\nabla H} \right|.
\end{aligned}$$

here p — is the static pressure, ρ — is the density of the mixture, M_m and Q_m the intensity of mass transfer and heat release as a result of the reaction, ν , D , k — effective coefficients of kinematic viscosity, molecular diffusion and thermal diffusivity, V_i , H , C_m — average speeds, enthalpy and concentration of the m -th component of the liquid, and $\mathcal{G}_i$, h , c_m — relative speeds, enthalpy and concentration of the m -th component of the liquid. In [1] it is shown that the friction and heat transfer coefficients are found as follows

$$K_f = C_{f1} \lambda_{\max} + C_{f2} \frac{|d \cdot \mathbf{v}|}{d^2}, \quad K_h = C_{t1} \lambda_{\max}.$$

Here d is the closest distance to the wall, and $\lambda_{\max}$ is the largest root of the characteristic equation $\det(A - \lambda E) = 0$.

In this expression E is the identity matrix, and A is the next matrix

$$A = \begin{vmatrix} -\frac{\partial V_1}{\partial x_1} & -\frac{\partial V_1}{\partial x_2} - C_s \zeta_3 & -\frac{\partial V_1}{\partial x_3} + C_s \zeta_2 \\ -\frac{\partial V_2}{\partial x_1} + C_s \zeta_3 & -\frac{\partial V_2}{\partial x_2} & -\frac{\partial V_2}{\partial x_3} - C_s \zeta_1 \\ -\frac{\partial V_3}{\partial x_1} - C_s \zeta_2 & -\frac{\partial V_3}{\partial x_2} + C_s \zeta_1 & -\frac{\partial V_3}{\partial x_3} \end{vmatrix}, \quad \zeta = \mathbf{rot} \mathbf{V}.$$

The constants of the two-fluid model are equal

$$C_s = 0.2, \quad C_{t1} = 0.92, \quad C_{t2} = 0.1, \quad C_{f1} = 0.7825, \quad C_{f2} = 0.306.$$

The temperature of the mixture is determined from the relation

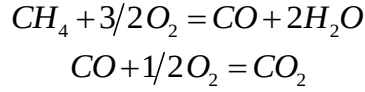
$$\int_{T_0}^T C_p dT = H, \quad C_p = \sum_{m=1}^M C_m C_{pm}.$$

In this expression, C_p is the specific heat capacity of the mixture, C_{pm} is the specific heat capacity of the m -th component.

2. Combustion modelling

There are various methods of methane combustion described in literature. For detailed mechanisms, we can cite [5–8]. Therefore, reduced mechanisms are often used for practical reasons [9–12].

In this work, the study of diffusion combustion is modeled in two steps, where the formation and oxidation of carbon monoxide is taken into account [10].



The reaction rate is determined by the Arrhenius relation.

$$\omega_{mk} = AT^\gamma Y_f^\alpha Y_o^\beta \exp\left(-\frac{E}{RT}\right).$$

In this expression A is the pre-exponential coefficient, α , β are the component exponent of each reaction, E is the activation energy of fuel, Y_f and Y_o are the molar concentrations of the reacting components of the gas, respectively, the fuel and the oxidizer in $kmol/m^3$ unit. The pre-exponential factor, concentration powers, and activation energies are empirical. The parameters of the Arrhenius model required for a two-stage reaction are presented in Table. 1. [13]

Table 1.

Parameters for the Arrhenius model					
Reactions	A	$E, J/kmol$	α	β	γ
first	2.8×10^{12}	2.03×10^8	-0.3	1.3	0
second	2.91×10^{12}	1.67×10^8	1	0.25	0

According to Prandtl's theory, it is known that the rate of turbulent mixing is proportional to the deformation rate $\text{def}(\mathbf{V})$ and, according to Magnussen's theory [14], we present it in the form

$$\omega_{mk}^{turb} = \min\left(AC_f \text{def}(\mathbf{V}), A \frac{n_f \mu_f}{n_o \mu_o} C_o \text{def}(\mathbf{V})\right).$$

The combustion reaction rates were calculated using the combined Arrhenius-Magnussen model [15].

$$\omega_{mk}^{com} = \min(\omega_{mk}^{turb}, \omega_{mk}).$$

The intensity of mass transfer through the reaction rate is determined by the formula

$$M_m = \sum_{m=1}^M n_m \mu_m \omega_{mk}^{com}.$$

Here μ_m is the molecular weight of the component m , n_m is the stoichiometric coefficient.

3. Problem statement

In this work, the axisymmetric turbulent non-premixed methane-air combustion is studied. Methane is injected through a jet nozzle with a diameter of 4 mm, concentric with a 1m long annular burner. The flame configuration is shown in Fig. 1. The mass flow rate of methane is $1.72 \times 10^{-4} kg/s$, the flow rate of air is $118 \times 10^{-4} kg/s$ at the temperature $300K$. At such mass flow rates, the velocities of methane and air are equal to $U_{jet} = 20.3 m/s$ and $U_{cof} = 0.45 m/s$ respectively. The Reynolds number at the nozzle exit is 5000.[16]

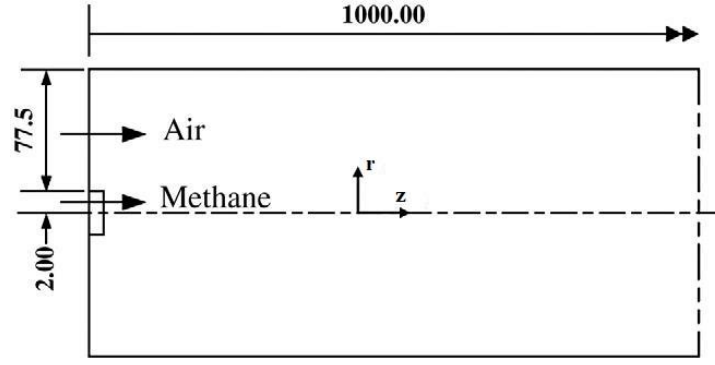


Figure1. **Flame configuration. Dimensions in mm .** [16]

4. Solution method

Since combustion occurs in a free jet at atmospheric pressure, the pressure can be considered constant. Also, in the diffusion terms, derivatives in the axial direction can be neglected. These assumptions are generally accepted and significantly facilitate the solution of the problem. Because the system of equations takes on a parabolic form and can be solved numerically using the marching method [17]. For this, Von Mises variables (ξ, ψ) [18] are introduced:

$$\xi = \tilde{z}, \quad \tilde{\rho}\tilde{V} = \frac{\psi^n \partial \psi}{\tilde{r} \partial \tilde{r}}, \quad \tilde{\rho}\tilde{U} = \frac{\psi^n \partial \psi}{\tilde{r} \partial \tilde{r}}.$$

Here n – is used to regulate the thickening of the computational grid near the jet axis. In this article, the value $n = 1$ was used for calculations [19].

$$\begin{aligned} \frac{\partial \tilde{U}}{\partial \xi} &= -\frac{\partial \tilde{\rho}\tilde{r}\tilde{u}\tilde{g}}{\psi \partial \psi} + \frac{1}{\text{Re}} \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U}}{\psi} \frac{\partial \tilde{U}}{\partial \psi} \right], \\ \frac{\partial \tilde{u}}{\partial \xi} &= -(1 - C_s) \frac{\tilde{\rho}\tilde{r}\tilde{g}}{\psi} \frac{\partial \tilde{U}}{\partial \psi} + \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U} \tilde{v}_{zr}}{\psi} \frac{\partial \tilde{u}}{\partial \psi} \right] - \frac{K_f \tilde{u}}{\tilde{U}}, \\ \frac{\partial \tilde{g}}{\partial \xi} &= -C_s \frac{\tilde{\rho}\tilde{r}\tilde{u}}{\psi} \frac{\partial \tilde{U}}{\partial \psi} + \frac{\partial}{\psi \partial \psi} \left[\frac{2\tilde{\rho}^2 \tilde{r}^2 \tilde{U} \tilde{v}_{rr}}{\psi} \frac{\partial \tilde{g}}{\partial \psi} \right] - \frac{2\tilde{v}_{rr} \tilde{g}}{\tilde{r}^2 \tilde{U}} - \frac{K_f \tilde{g}}{\tilde{U}}, \\ \frac{\partial \tilde{C}_m}{\partial \xi} &= -\frac{\partial \tilde{\rho}\tilde{r}\tilde{g}\tilde{c}_m}{\psi \partial \psi} + \frac{1}{\text{ReSc}} \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U}}{\psi} \frac{\partial \tilde{C}_m}{\partial \psi} \right] + M_m, \\ \frac{\partial \tilde{c}_m}{\partial \xi} &= -\frac{\tilde{\rho}\tilde{r}\tilde{g}}{\psi} \frac{\partial \tilde{C}_m}{\partial \psi} + \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U} \tilde{D}_{mr}}{\psi} \frac{\partial \tilde{c}_m}{\partial \psi} \right] - \frac{K_c \tilde{c}_m}{\tilde{U}}, \\ \frac{\partial \tilde{H}}{\partial \xi} &= -\frac{\partial \tilde{\rho}\tilde{r}\tilde{g}\tilde{h}}{\psi \partial \psi} + \frac{1}{\text{RePr}} \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U}}{\psi} \frac{\partial \tilde{H}}{\partial \psi} \right] + \sum_{m=1}^M M_m Q_m, \\ \frac{\partial \tilde{h}}{\partial \xi} &= -\frac{\tilde{\rho}\tilde{r}\tilde{g}}{\psi} \frac{\partial \tilde{H}}{\partial \psi} + \frac{\partial}{\psi \partial \psi} \left[\frac{\tilde{\rho}^2 \tilde{r}^2 \tilde{U} \tilde{k}_r}{\psi} \frac{\partial \tilde{h}}{\partial \psi} \right] - \frac{K_h \tilde{h}}{\tilde{U}}, \\ \tilde{v}_{zr} &= \frac{3}{\text{Re}} + 2 \frac{|\tilde{u}\tilde{g}|}{\left| \frac{\tilde{\rho}\tilde{r}\tilde{U}}{\psi} \frac{\partial \tilde{U}}{\partial \psi} \right|}, \quad \tilde{v}_{rr} = \frac{3}{\text{Re}} + 2 \frac{|\tilde{g}\tilde{g}|}{\left| \frac{\tilde{\rho}\tilde{r}\tilde{U}}{\psi} \frac{\partial \tilde{U}}{\partial \psi} \right|}, \\ \tilde{D}_{mr} &= \frac{3}{\text{ReSc}} + 2 \frac{|\tilde{g}\tilde{c}_m|}{\left| \frac{\tilde{\rho}\tilde{r}\tilde{U}}{\psi} \frac{\partial \tilde{C}_m}{\partial \psi} \right|}, \quad \tilde{k}_r = 3 \frac{3}{\text{RePr}} + 2 \frac{|\tilde{g}\tilde{h}|}{\left| \frac{\tilde{\rho}\tilde{r}\tilde{U}}{\psi} \frac{\partial \tilde{H}}{\partial \psi} \right|}, \end{aligned}$$

$$\begin{cases} \lambda_{\max} = \sqrt{A}, & A \geq 0 \\ \lambda_{\max} = 0, & A \leq 0 \end{cases}, A = C_s(1 - C_s) \left[\left(\frac{\tilde{\rho} \tilde{r} \tilde{U}}{\psi} \frac{\partial U}{\partial \psi} \right)^2 \right], C_s = 0.2, C_1 = C_{s1} = 0.7825,$$

$$K_c = C_{s1} \lambda_{\max}, K_f = C_1 \lambda_{\max}, K_h = C_1 \lambda_{\max}$$

The numerical calculation of the system of equations was carried out using an implicit finite-difference scheme. In the transverse direction, the central difference is used. To solve the implicit scheme, the Thomas algorithm [20] was used. Its approximation accuracy was $O(\Delta\xi, (\Delta\psi)^2)$. The integration steps were $\Delta\xi = 0,0001$, $\Delta\psi = 0,01$. The number of calculated nodes in the transverse direction was 1000. The standard initial and boundary conditions for axisymmetric flow were set for the problem.

5. Results and discussions

The results of the numerical solution of the system are illustrated below. To compare these results, experimental data from [4] are also presented.

Fig. 2 shows the dependence of the dimensionless axial temperature on the distance to the nozzle.

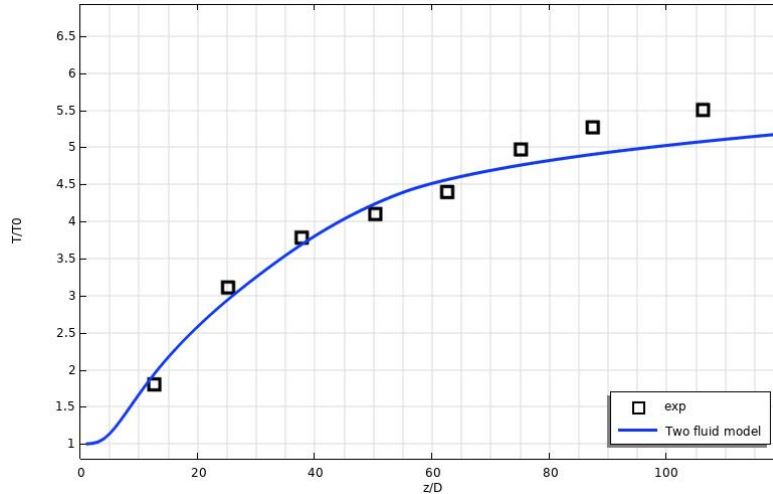


Figure 2. Axial dimensionless temperature

Fig. 3 illustrates the profiles of dimensionless longitudinal temperature at a distance 150 mm from the nozzle.

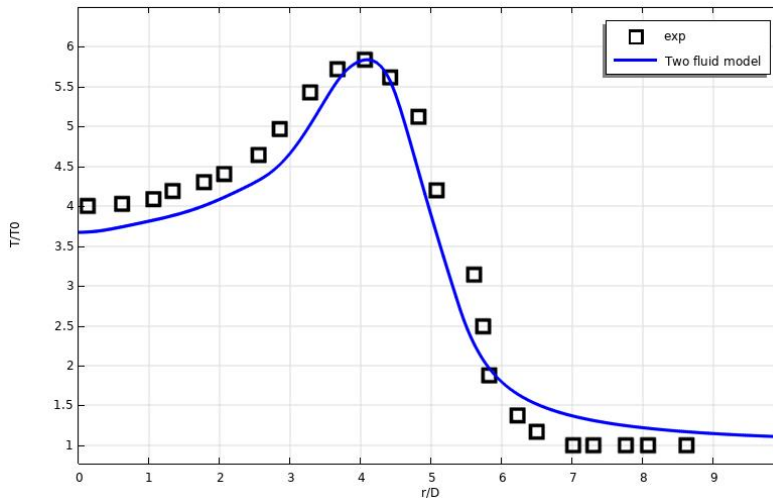


Figure 3. Dimensionless temperature profile at a distance of 150 mm from the nozzle

Fig. 4 illustrates the profiles of dimensionless longitudinal temperature at a distance 200 mm from the nozzle.

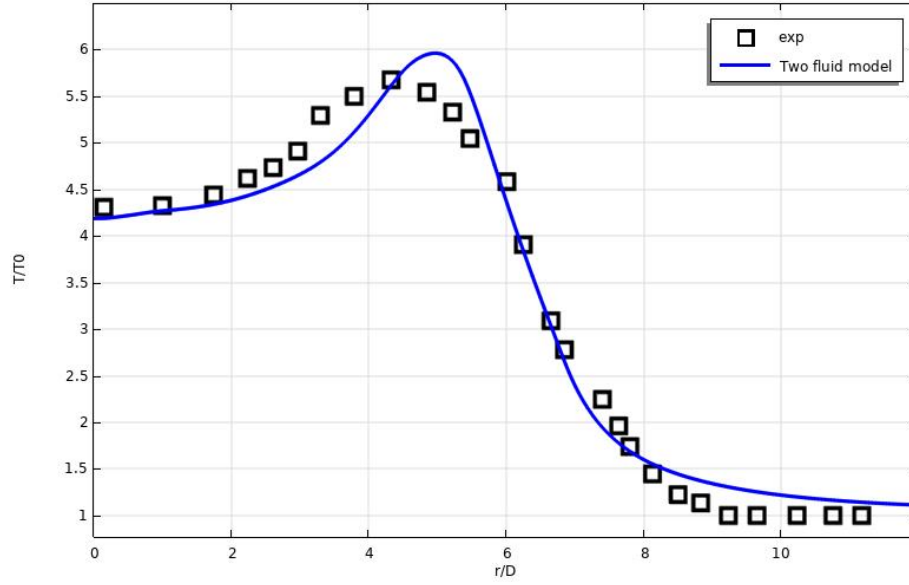


Figure 4. Dimensionless temperature profile at a distance of 200 mm from the nozzle

Fig. 5 illustrates the profiles of dimensionless longitudinal temperature at a distance 300 mm from the nozzle.

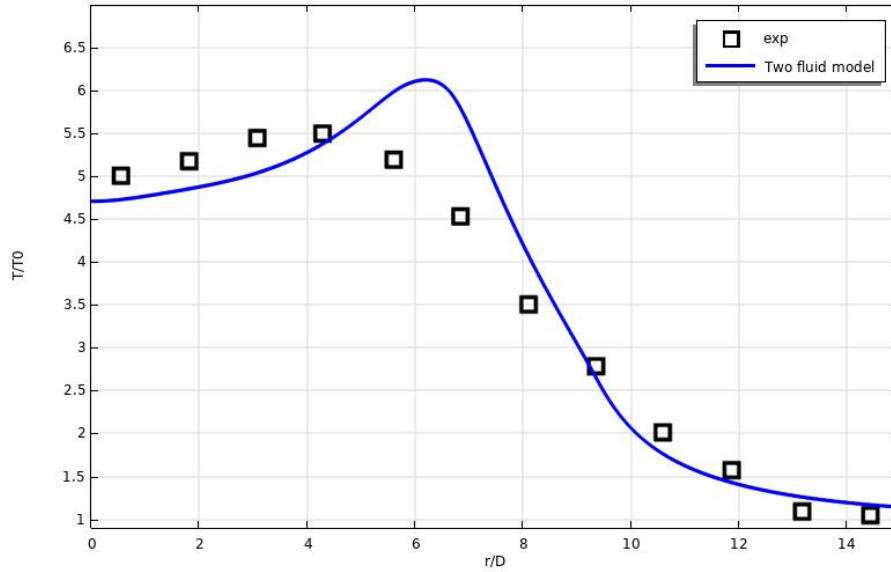


Figure 5. Dimensionless temperature profile at a distance of 300 mm from the nozzle

6. Conclusions

The work simulates the combustion of a multicomponent gas based on a two-fluid approach to turbulence. Numerical results of the proposed model for the turbulent non-premixed combustion of a methane jet are obtained and compared with known experimental data. It is shown that the results of the two-fluid approach model are close to experimental data.

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NUMERICAL METHOD FOR SOLVING PIPELINE TRANSPORT EQUATIONS OF REAL FLUID UNDER GIVEN LAWS OF CHANGE OF PRESSURE AT INLET AND MASS FLOW AT OUTLET OF PIPELINE

Khojikulov Sh.Sh.¹

¹*Institute of Mechanics and seismic stability of structures, Tashkent, Uzbekistan*

E mail: xojikulovshaxzod@mail.ru

Abstract: A numerical method is proposed for solving the problem of the state of an elementary section of a pipeline, when the law of change in pressure is specified at the inlet, and the law of change in mass flow of liquid is specified at the outlet. The equation of state of the liquid is constructed taking into account the coefficient of volumetric compression of the liquid. Nonlinear quasi-one-dimensional equations N.E. Zhukovsky, which additionally take into account the slope of the pipeline route, are dimensionless and, using an auxiliary function, are presented in the form of equations of counter-propagating waves. To implement nonlinear boundary conditions, an algorithm based on Newton's tangent method has been developed. Numerical results were obtained for the problems of jump transition to new values of inlet pressure and outlet mass flow rate of liquid.

Key words: quasi-one-dimensional equations N.E. Zhukovsky, pressure, speed, density, mass flow, traveling wave, finite differences, directional derivative.

1. Introduction

The study of the propagation of compaction and vacuum waves in pipelines has a centuries-old history. The main provisions of the theoretical and experimental study of the propagation of disturbances in pipelines were formulated in the classical works of N.E. Zhukovsky [1-5]. The quasi-one-dimensional model of pipeline transportation of liquids formulated by him in the form of nonlinear equations of conservation of momentum and mass is currently used in studying the features of the functioning of a pipeline network intended for the transport of liquids and gases, various mixtures, for the transmission of mechanical energy in various thermal and pressure conditions.

In the formative stage of this direction of science, the main attention was paid to the solution of simplified (usually linearized) equations under certain boundary conditions and to the accumulation of experimental data. From the point of view of the hydrodynamics of pipeline transportation of various media, Nikuradze made a significant scientific contribution. He was the first to experimentally identify laminar, transitional flow regimes and three subregimes of turbulent flow around a rough surface of a pipe with a circular cross-section [5]. Theoretical studies were mainly carried out within the framework of linearized equations of transfer and conservation of momentum [1,5,6-8]. Solving problems of the movement of liquids and gases, even in a stationary setting, was difficult. In particular, taking into account the full range of force factors, the stationary problem in the isothermal formulation has limited solution options [1,7,9-11]. Nonstationary solutions to the problems were obtained, in particular, in [45] in the form of Bessel functions. The practical application of this solution to the calculation of a hydraulic network using the Laplace transform was carried out in [12]. Analytical solutions to a wide range of problems of pipeline transportation of multiphase media are given in the monograph [13,14].

Of great practical interest is the applicability of one or another version of N.E. Zhukovsky equations. to simulate the operation of pipelines in unsteady cases caused by the formation and propagation of a shock wave. Numerous analytical solutions to shock wave problems have been obtained and studied within the framework of linearized equations of short and long pipeline approaches [1,4-5,15-17].

With the increase in computing resources and the development of accessible software, numerical methods for solving problems based on nonlinear equations for pipeline transportation of liquids and gases began to develop [17-24]. They related both to a separate elementary section of the pipeline and to a complex network of pipelines with a ring and combined structure.

Below we propose a numerical method for solving quasi-one-dimensional nonlinear equations for pipeline transportation of liquid, the equation of state of which takes into account the coefficient of volumetric deformation of the liquid. The equations are reduced to dimensionless equations of counter-

propagating waves. A simple first-order approximation scheme is proposed, which takes into account the directions of propagation of counterpropagating waves and implements the provisions of upper relaxation. Some results of a computational experiment related to discontinuous conditions are presented.

2. Formulation of the problem.

The initial distributions of speed and pressure in the relief section of the pipeline are specified, which is determined by the diameter D , length l , resistance coefficient λ , variable leveling height of the pipeline axis $y(x)$ (the slope is defined as $\sin \alpha = \frac{dy(x)}{dx}$). At $t > 0$ the input, the law of change in pressure versus time is specified, and at the output, the law of change in mass flow versus time is specified.

It is required to develop a numerical method for solving the problem within the framework of the quasi-one-dimensional isothermal model N.E. Zhukovsky.

The isothermal state of the liquid in the pipeline is described by quasi-one-dimensional equations of conservation of momentum and mass [1,2,23]:

$$\begin{cases} \rho \frac{\partial u}{\partial t} + \rho u \frac{\partial u}{\partial x} + \frac{\partial p}{\partial x} = -\frac{\lambda}{2D} \rho u |u| - \rho g \sin \alpha, \\ -\frac{\partial \rho}{\partial t} = \frac{\partial \rho u}{\partial x}. \end{cases} \quad (1)$$

Here and below $\rho(x, t)$, $u(x, t)$, $p(x, t)$ are the average values of density, velocity and pressure of the fluid in the section x at time t .

The equation of state of the liquid has the form:

$$p - p_* = \frac{k}{\rho_*} (\rho - \rho_*),$$

where received $\rho_* = 998.23 \text{ kg/m}^3$, $k = 2120 \text{ MPa}$ at $p_* = 101325 \text{ Pa}$, $T_0 = 293.15 \text{ K}$.

3. Method for solving the problem.

When solving a problem numerically, undesirable cases arise due to different scales of the quantities used. In particular, $c^2 \approx 10^6 \text{ m}^2/\text{sek}^2$, $D \approx 0.01 - 2 \text{ m}$, $u \approx \pm 20 \text{ m/s}$. In this regard, it is necessary to move on to dimensionless variables. As scale values we take: c – for speed; l – for length; ρ_* – for density; l/c – for time. In this regard, dimensional quantities in equations and conditions are taken in the form: $x = l \bar{x}$, $u = c \bar{u}$, $t = \bar{t} l/c$, $\rho = \rho_* \bar{\rho}$.

Let's rewrite the equations in terms of dimensionless quantities:

$$\begin{cases} \frac{c}{l/c} \frac{\partial \bar{u}}{\partial \bar{t}} + \frac{c^2}{l} \bar{u} \frac{\partial \bar{u}}{\partial \bar{x}} + \frac{c^2}{l} \frac{\partial \bar{\varphi}}{\partial \bar{x}} = -\frac{\lambda c^2}{2D} \bar{u} |\bar{u}| - g \sin \alpha, \\ \frac{1}{l/c} \frac{\partial \bar{\varphi}}{\partial \bar{t}} + \frac{c}{l} \frac{\partial \bar{u}}{\partial \bar{x}} + \frac{c}{l} \bar{u} \frac{\partial \bar{\varphi}}{\partial \bar{x}} = 0. \end{cases} \quad (2)$$

We divide the terms of the first equation by c^2/l , and the terms of the second by c/l . As a result, we get:

$$\begin{cases} \frac{\partial \bar{u}}{\partial \bar{t}} + \bar{u} \frac{\partial \bar{u}}{\partial \bar{x}} + \frac{\partial \bar{\varphi}}{\partial \bar{x}} = -\frac{\lambda l}{2D} \bar{u} |\bar{u}| - \frac{lg}{c^2} \sin \alpha, \\ \frac{\partial \bar{\varphi}}{\partial \bar{t}} + \frac{\partial \bar{u}}{\partial \bar{x}} + \bar{u} \frac{\partial \bar{\varphi}}{\partial \bar{x}} = 0. \end{cases} \quad (3)$$

With a similar change in the original equations, the coefficient for the term $\frac{\partial \bar{p}}{\partial \bar{x}}$ will be $\frac{p_*}{c^2 \rho_*}$. To avoid the formation of a new dimensionless criterion, $\bar{p}$ is taken as the pressure scale $p_* = c^2 \rho_*$. In this regard, the question arises of determining the speed of propagation of small disturbances c .

We accept $\rho_0 = \rho_*$, $p_0 = 101325 \text{ Pa}$, $T_0 = 293.15 \text{ K}$, $\rho_0 = 998.23 \text{ kg / m}^3$ and $k = 2120 \text{ MPa}$. To make it convenient to present and analyze the results, the pressure scale is taken as $p_* = 100000 \text{ Pa}$. Then the speed of sound in a liquid medium is calculated by the formula:

$$c = \sqrt{\frac{p_* - p_0 + k}{\rho_0}}. \quad (4)$$

Taking this data into account, the initial

$$\bar{u}(\bar{x}, 0) = u^0(\bar{x}) / c, \quad \bar{p}(\bar{x}, 0) = p^0(\bar{x}) / p_* \quad (\bar{\rho}(\bar{x}, 0) = \rho^0(\bar{x}) / \rho_*)$$

and boundary conditions

$$\bar{\rho}(0, \bar{t}) = \frac{\Pi_0(\bar{t}) - p_0 + k}{p_*}, \quad (5)$$

$$\bar{\rho}(1, \bar{t}) \bar{u}(1, \bar{t}) = \frac{Q_l(t)}{f \rho_* c} = \bar{Q}_l(\bar{t}). \quad (6)$$

From the last system we pass to the system of equations from analogues of traveling waves [16]:

$$f_1 = \bar{u} + \varphi, \quad f_2 = \bar{u} - \varphi. \quad (7)$$

Term-by-term summation of the terms of the system equations leads to the first equation:

$$\frac{\partial f_1}{\partial \bar{t}} + (1 + \bar{u}) \frac{\partial f_1}{\partial \bar{x}} = -\frac{\lambda l}{2D} \bar{u} |\bar{u}| - \frac{lg}{c^2} \sin \alpha, \quad (8)$$

and term-by-term subtraction of the terms of the second equation from the terms of the first equation gives the second equation:

$$\frac{\partial f_2}{\partial \bar{t}} + (\bar{u} - 1) \frac{\partial f_2}{\partial \bar{x}} = -\frac{\lambda l}{2D} \bar{u} |\bar{u}| - \frac{lg}{c^2} \sin \alpha. \quad (9)$$

According to the introduced modifications, we transform the boundary conditions.

For the initial distribution, φ_i^0 you can use the formula:

$$\varphi_i^0 = \ln \bar{\rho}(\bar{x}, 0). \quad (10)$$

The new initial conditions have the form:

$$f_{1,2}(\bar{x}, 0) = \bar{u}(\bar{x}, 0) \pm \ln \bar{\rho}(\bar{x}, 0). \quad (11)$$

Let us transform the condition at $\bar{x} = 0$:

$$\varphi(0, \bar{t}) = \ln \bar{\rho}(0, \bar{t}) = \ln \frac{P_0(\bar{t}) - p_0 + k}{p_*}, \quad (12)$$

and the condition at $\bar{x} = 1$ remains in force.

Methods for implementing these boundary conditions are described in the process of compiling an algorithm for solving the problem regarding analogues of counter-propagating waves f_1 and f_2 .

Discrete coordinates are entered $i = x / h$ in the interval $i \in [0; N_x]$ with a constant step $h = l / N_x$ ($N_x = 1000$). The time step was $\tau = l / (5000c)$.

The algorithm for solving the equations of analogues of traveling waves under the given boundary conditions is similar to the algorithm from [25]. In contrast to this cited algorithm: (a) the dimensionless

time step is taken small enough so as not to organize an iterative process for the new $n+1$ th time step; (b) the time scheme is taken to be implicit, i.e. A kind of upper relaxation is used.

Due to the use of upper relaxation, a certain order of the sequence of operations is imposed. First, we find $n+1$ the values for the i th time step f_{1i}^{n+1} with an increasing index $i = 0..N_x$, taking $f_{2i}^{n+1} = f_{2i}^n$. Next, they were found f_{2i}^{n+1} by descending index $i = N_x..0$ using already known values f_{1i}^{n+1} .

When $i = 0$ the boundary condition was specified in the form φ_0^{n+1} and the value was considered known $f_{20}^{n+1} = f_{20}^n$. Then we find $f_{10}^{n+1} = f_{20}^{n+1} + 2\varphi_0^{n+1}$ and $\bar{u}_0^{n+1} = f_{10}^{n+1} - \varphi_0^{n+1}$.

When $i = 1..N_x$ the first equation was approximated as:

$$\frac{f_{1i}^{n+1} - f_{1i}^n}{\tau} + (1 + u_i^n) \frac{f_{1i}^{n+1} - f_{1i-1}^{n+1}}{h} = \Phi(\bar{u}_i^n, \alpha_i) \quad (13)$$

Where $\Phi(\bar{u}_i^n, \alpha_i) = -\frac{\lambda l}{2D} \bar{u}_i^n |\bar{u}_i^n| - \frac{l g}{c^2} \sin \alpha_i$.

From here, at $\sigma = \tau / h$, we found the value f_{1i} in a new approximation:

$$f_{1i}^{n+1} = \frac{f_{1i}^n + \sigma(1 + \bar{u}_i^n) f_{1i-1}^{n+1} + \tau \Phi(\bar{u}_i^n, \alpha_i)}{1 + \sigma(1 + \bar{u}_i^n)}. \quad (14)$$

When $i = N_x$ this formula is also used, because disturbance f_1 is transferred from left to right.

The newly found values f_{1i}^{n+1} for calculations f_{2i}^{n+1} with a descending index $i = N_x..0$. And let's start the calculations from the boundary node $i = N_x$, for which the condition is specified:

$$\bar{\rho}(1, \bar{t}) \bar{u}(1, \bar{t}) = \frac{Q_l(t)}{f \rho_* c} = \bar{Q}_l(\bar{t}). \quad (15)$$

The condition is approximated in the form:

$$\bar{\rho}_{N_x}^{n+1} \bar{u}_{N_x}^{n+1} = \bar{Q}_l^{n+1}. \quad (16)$$

If so $\bar{Q}_l^{n+1} = 0$, then it should be accepted $\bar{u}_{N_x}^{n+1} = 0$. Accordingly, we find $\varphi_{N_x}^{n+1} = f_{1N_x}^{n+1}$ and $f_{2N_x}^{n+1} = -\varphi_{N_x}^{n+1}$.

When $\bar{Q}_l^{n+1} \neq 0$ the nonlinear condition is solved with respect to $\bar{u} = \bar{u}_{N_x}^{n+1}$.

Given the known, $f_{1N_x}^{n+1}$ we find

$$\bar{\rho}_{N_x}^{n+1} = e^{\varphi_{N_x}^{n+1}} = e^{f_{1N_x}^{n+1} - \bar{u}}. \quad (17)$$

Therefore the condition takes the form:

$$e^{f_{1N_x}^{n+1} - \bar{u}} \bar{u} = \bar{Q}_l^{n+1}. \quad (18)$$

Let's create an auxiliary function

$$E(\bar{u}) = e^{-f_{1N_x}^{n+1}} \bar{Q}_l^{n+1} - e^{-\bar{u}} \bar{u}, \quad (19)$$

whose zero is the desired solution.

Liquid is taken from the end of the pipeline, i.e. $\bar{Q}_l(t) > 0$. Respectively, $0 < \bar{u} < 1$. At the largest value, $\bar{u} = 1$ inequality occurs $E(1) < 0$ since the subtrahend increases. At the smallest value, $\bar{u} = 0$ inequality occurs $E(0) > 0$. These judgments are confirmed by the fact that

$$E'(\bar{u}) = -e^{-\bar{u}} (1 - \bar{u}) < 0. \quad (20)$$

Let's calculate the second derivative of the function $E(\bar{u})$:

$$E''(\bar{u}) = e^{-\bar{u}} (2 - \bar{u}). \quad (21)$$

It has a positive meaning, i.e. The graph of the function $E(\bar{u})$ is concave in the interval of interest to us. Under such conditions, when solving a given transcendental equation, one can use Nyuton's tangent method, starting from the zero approximation $\bar{u}^0 = 0$.

Let $(\bar{u}^k, E(\bar{u}^k))$ a tangent be drawn through a point in the curve of a function $E(\bar{u})$ with an angular coefficient [26]:

$$E'(\bar{u}^k) = \frac{E(\bar{u}^{k+1}) - E(\bar{u}^k)}{\bar{u}^{k+1} - \bar{u}^k}. \quad (22)$$

At the point $(\bar{u}^{k+1}, E(\bar{u}^{k+1}))$ this line intersects the x-axis: $E(\bar{u}^{k+1}) = 0$. Then the last equality implies a recurrent dependence for the successive approximation:

$$\bar{u}^{k+1} = \bar{u}^k - \frac{E(\bar{u}^k)}{E'(\bar{u}^k)} = \bar{u}^k - \frac{e^{-f_{1N_x}^{n+1} \bar{Q}_l^{n+1}} - e^{-\bar{u}} \bar{u}}{-e^{-\bar{u}} (1 - \bar{u})} = \bar{u}^k + \frac{e^{\bar{u} - f_{1N_x}^{n+1} \bar{Q}_l^{n+1}} - \bar{u}}{1 - \bar{u}}. \quad (23)$$

Successive approximation continues until one of the strict conditions is reached $|\bar{u}^{k+1} - \bar{u}^k| < 10^{-10}$ and $|E(\bar{u}^{k+1})| < 10^{-10}$.

According to the obtained value, $\bar{u}^{k+1} = \bar{u}_{N_x}^{n+1}$ we find $f_{2N_x}^{n+1} = 2\bar{u}_{N_x}^{n+1} - f_{1N_x}^{n+1}$ and $\varphi_{N_x}^{n+1} = f_{1N_x}^{n+1} - \bar{u}_{N_x}^{n+1}$ which are used when finding $f_{2N_x-1}^{n+1}$. In general, for a decreasing index, $i = N_x - 1..0$ the second equation is approximated taking into account the direction of propagation f_2 (against the direction of the axis x):

$$\frac{f_{2i}^{n+1} - f_{2i}^n}{\tau} + (\bar{u}_i^n - 1) \frac{f_{2i+1}^{n+1} - f_{2i}^{n+1}}{h} = \Phi(\bar{u}_i^n, \alpha_i). \quad (24)$$

From here we get the recurrent formula for the descending index:

$$f_{2i}^{n+1} = \frac{f_{2i}^n + \sigma(1 - \bar{u}_i^n) f_{2i+1}^{n+1} + \tau \Phi(\bar{u}_i^n, \alpha_i)}{1 + \sigma(1 - \bar{u}_i^n)}. \quad (25)$$

After calculation f_{1i}^{n+1} and f_{2i}^{n+1} in all calculation nodes, we find the values

$$\bar{u}_i^{n+1} = \frac{f_{1i}^{n+1} + f_{2i}^{n+1}}{2}, \quad (26)$$

$$\varphi_i^{n+1} = \frac{f_{1i}^{n+1} - f_{2i}^{n+1}}{2}, \quad (27)$$

which are necessary for calculations in $n+1$ the time step. We accept the values f_{1i}^{n+1} , f_{2i}^{n+1} , u_i^{n+1} and φ_i^{n+1} as data for n the time step and, if necessary, move on to the next time step.

4. Results

We have compiled a program for calculating mass flow, velocity and pressure in transient processes in a horizontal section of a pipeline, when the initial and boundary conditions are expressed by constant values. In the calculations, the following values were used: length of the section, $l = 5000 \text{ m}$, diameter of the section, $D = 0.2 \text{ m}$, drag coefficient, $\lambda = 0.01$, speed of propagation of small disturbances. $c = 1457.31 \text{ m/s}$. A series of calculations were carried out with and without taking into account the friction force and in time steps $l/(5000c)$.

A computational experiment was carried out based on the proposed calculation algorithm.

Let's imagine the results that are obtained when the input pressure $p_{00} = 7.5 \text{ MPa}$ instantly changes to a new value $p_{00} = 7.5 \text{ MPa}$. The initial and new output speeds were $w_{00} = w_l = 3 \text{ m/s}$.

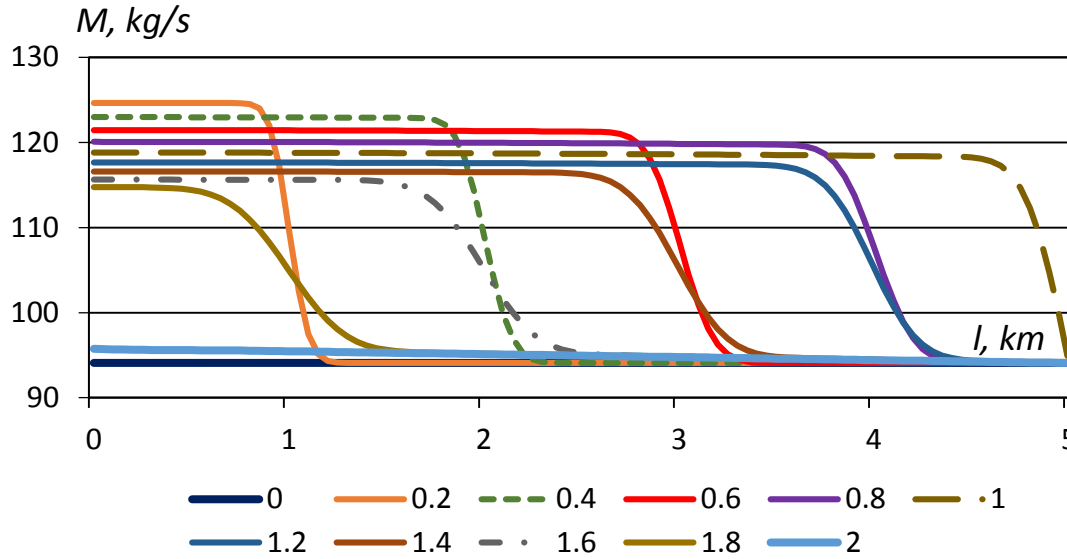


Figure 1. - Graphs of mass flow along the length of the section, obtained in the first ten time steps at $\tau = l / (5c)$. $D = 0.2 \text{ m}$, $\lambda = 0.01$, $l = 5000 \text{ m}$, $w_{00} = w_l = 3 \text{ m/s}$, $p_{00} = 7.5 \text{ MPa}$, $p_{0t} = 9 \text{ MPa}$

In figure 1 shows the mass flow curves for this case. The initial moment of mass flow is constant (lower line). An increase in inlet pressure produces a mass flow shock that propagates downstream. During the movement, the maximum value of the mass flow rate of the liquid decreases. The jump in the mass flow graphs appears only over a short period of time. With the passage of time, a smooth transition through the conditional shock front is established. This is especially noticeable in the right transition parts of the graphs. It follows from the graphs that the highest value of mass flow rate decreases rapidly from 124.647 to 114.744 kg/s, and the lowest value slowly increases from 94.081 to 95.734 kg/s.

In the first half of the conventional half – l/c time period, the envelope of the mass flow curves is represented by the initial mass flow profile. When the wave returns to the beginning of the section, a new upper envelope is formed, which passes through the points (5 km; 94.081 kg/s) and (0 km; 95.734 kg/s).

In the interval $(0, l/c)$ there was a compaction wave, and $(l/c, 2l/c)$ a rarefaction wave. Because at the beginning of the section the pressure remains constant, then at the outlet there remains a constant mass flow of liquid.

Formed in the time interval $(l/c, 2l/c)$ remains an envelope, but an upper one (fig. 2).

The fact is that upon reaching the inlet cross section, the flow slows down and then recedes back. Those. a rarefaction wave moves back to the end of the section, and a compaction wave moves in the opposite direction over a period of time $(3l/c; 4l/c)$. A new upper envelope is formed, passing through the points (5 km; 94.081 kg/s) and (0 km; 95.734 kg/s). The graphs in the intervals $(2l/c, 4l/c)$ are smoother than in the intervals $(0, 2l/c)$.

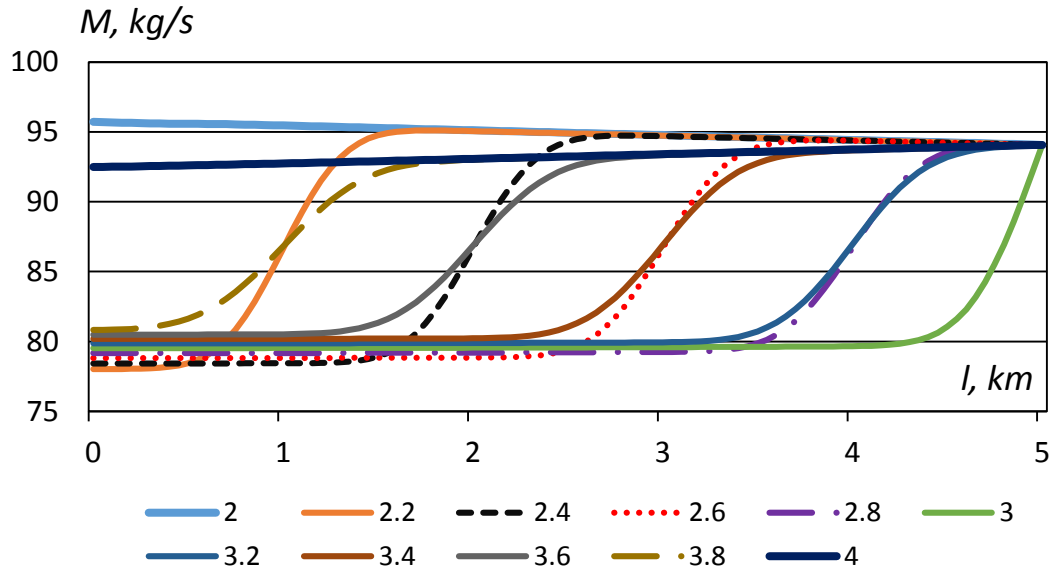


Figure-2. - Graphs of mass flow along the length of the section, obtained in the second conditional period at a time step $\tau = l / (5c)$
Data see figure-1

Forming new and new envelopes with such a sequence, the mass flow gradually moves to its original profile.

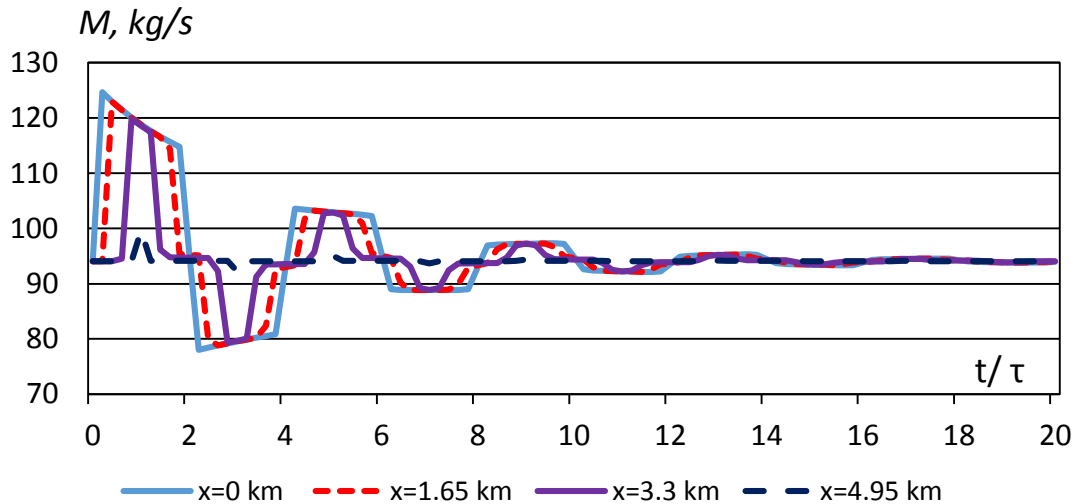


Figure-3. - Change in mass flow over time in different sections of the pipeline. Time step l/c . Data see fig. 1

The curves of mass flow versus time in sections $x = 0.1, 1.65, 3.3$ are 4.95 km presented in fig. 3. As can be seen from the figure, with distance from the end of the section where the mass flow rate is fixed, the deviation of the mass flow rate from the original distribution is noticeable. But as time passes, these deviations tend to zero, i.e. the stream is established.

The same patterns were revealed in the pressure features during the propagation of a pressure surge wave. The pressure curves in the first conventional time period for this problem are presented in the following figure.

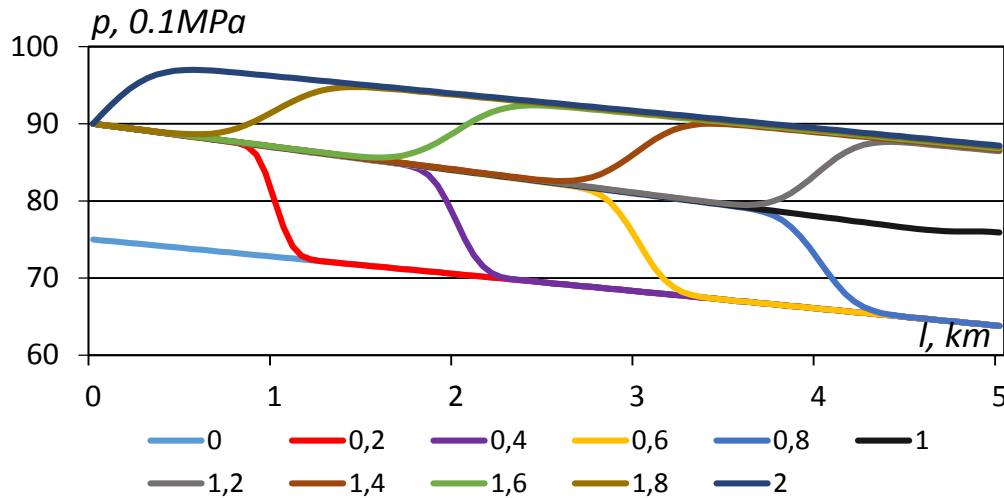


Figure 4. - Pressure graphs along the length of the section, obtained in the first conditional period through time steps $\tau = l / (5c)$.
Data see figure-1

5. Conclusion

The presented algorithm can be used to solve non-stationary problems, including problems of transition from one mode to another mode in terms of pressure and mass flow. In the general case, the calculation can be continued until a certain time step value or until the conditions $\min_{i \in [0..N_x]} |\bar{u}_i^{n+1} - \bar{u}_i^n| < 10^{-8}$ and are met simultaneously $\min_{i \in [0..N_x]} |\varphi_i^{n+1} - \varphi_i^n| < 10^{-8}$. If the problem of establishment is being solved (transition from one mode to another), then the results of the dimensionless mass flow rate of the last two time steps are additionally compared $\min_{i \in [0..N_x]} |\bar{M}_i^{n+1} - \bar{M}_i^n| < 10^{-6}$.

Numerous existing works focus on abrupt changes in mass flow or velocity at the ends of an elementary section. Boundary pressure jumps are considered, in particular, in [1]. The developed program allows solving problems with an abrupt increase and decrease in pressure, which is the success of the proposed algorithm. The computational experiment carried out showed that with a smooth change in the boundary functions, the program also provides the necessary convergence. In contrast to the results of solving a similar problem for real gas [8,15], pressure and flow rate jumps are less pronounced.

The above-discussed and expected new results make it possible to study the features of the functioning of an elementary section in a non-stationary setting. This is especially useful when selecting air cap geometric and mechanical parameters for various hydraulic systems.

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FLUID DYNAMICS SIMULATION OF TWIN SQUARE CYLINDERS INTERACTION USING A TWO FLUID TURBULENCE MODEL MALIKOV

Madaliev M.E.¹, Mamadaliev Kh.A.², Hamdamov M.M.^{1,2}

¹Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan,

²Tashkent University of Information Technologies named after Muhammad al-Khwarizmi, Tashkent, Uzbekistan

E-mail: mmhamdamov@mail.ru

Abstract: This study presents a two-fluid Malikov turbulence model. In this article, the SIMPLE algorithm with a semi-implicit anti-flow scheme of second order accuracy was used to solve the problem. The numerical code is tested and compared using a two-dimensional benchmark problem of flow around an isolated square cylinder at $Re=47000$, then applied to the transverse flow around two square cylinders side by side at the same Reynolds number. The ratio of the distance between two cylinders T/d ranges from 2.4 to 5 and has a significant impact on the flow patterns. Using the Malikov turbulence model, two types of wake, slot flow interference and synchronized antiphase are obtained. In this case, the characteristics of the aerodynamic pressure coefficients for the cylinders and the time-averaged velocity field are studied. The results of the study are also compared with experimental data.

Keywords: Navier–Stokes equations, separated flow, control volume method, two-fluid model, SIMPLE algorithm.

1. Introduction

Flow around an isolated stationary circular cylinder has been carefully focused over the past few decades, and rich fundamental fluid mechanics have been revealed [1–4]. Unlike a round cylinder, the separation points when flowing around a square cylinder are always fixed on sharp edges, which leads to different development of the shear layer and the formation of a wake. Currently, the flow around a square cylinder at $Re=22000$ with an angle of incidence $\alpha=0^\circ$ has always been taken as a reference example to test the implementation of the turbulence model [5-6].

In addition, in the presence of two square cylinders immersed in the flow region and located side by side, the flow dynamics are affected not only by Reynolds numbers, but also by the distance between the cylinders [7-9]. This topic is expected to be significantly more complex, but has received significantly less attention than the single-cylinder configuration. [8] carried out a two-dimensional numerical simulation of an unsteady compressible fluid using the Navier-Stokes equations for two side-by-side square cylinders in a laminar flow regime using the finite difference method, identifying six different wake models at different interval ratios. [9] conducted cross-flow experiments around two side-by-side square cylinders in an open-loop wind tunnel with a Reynolds number ranging from 2262 to 28000 and a relative distance between 1 and 13. The flow patterns were divided into three modes: single mode, slot flow mode and paired vortex formation mode. They also presented how parameters such as drag coefficient and vortex shedding frequency change for different modes.

Experimental and numerical studies are the two main approaches to studying turbulent flow problems. When conducting experimental tests in water and wind tunnel, based on the complexity of the simulation, numerical simulation may have the advantage of being cheaper and faster. Among the numerical methods, direct numerical simulation (DNS) is able to provide accurate solutions, but the grid resolution requirement is on the order of $Re^{9/4}$, which results in huge computational costs. The numerical effort of the large eddy simulation (LES) model is considered to be less than that of DNS, and various kinds of LES models have been successfully applied to some unsteady problems, such as turbulent flow around various obstacle shapes [10–16]. One limitation of LES is that it is still computationally intensive, especially at high Reynolds numbers. This is primarily due to the high resolution in the near-wall regions and in turbulent wakes.

Another approach is the Reynolds Average Navier-Stokes (RANS) method, which allows 2D or 3D calculations even at higher Reynolds numbers. A number of related studies have been conducted using two-equation RANS models such as k- ϵ models [1]. Recently, [17] successfully carried out a study using the Spalart-Allmaras (SA) model [18, 19] to the problem of turbulent flow around prismatic cylinders. As a typical single-equation RANS turbulence model, the SA model has the advantage of solving fewer equations compared to other two-equation turbulence models such as the k- ϵ and SST k-

w models. Since this model requires fewer mesh numbers, the overall computational cost is less than that of LES. However, despite the numerous RANS models, there is still no universal model. This means that each model has its own area of application in solving turbulence problems. Therefore, engineers need to select the most appropriate turbulence model depending on the problem being solved. The work [20–21] concludes that the most accurate results for the longitudinal velocity profile are predicted by the Reynolds stress method RSM. However, this method also predicts turbulent stress profiles quite roughly.

Another approach to mathematical modeling of turbulence is Spalding's two-fluid approach [22]. This model is based on the idea of dividing a turbulent flow into two fluids according to some chemical or physical properties. Then, for each fluid, its own equations of motion are written by analogy with a two-phase flow. In [23–25], a two-fluid model was used to determine intermittency in turbulent flows. The flow was divided into "turbulent" and "non-turbulent" fluids. In [26], when modeling the combustion process, the flow was considered as a mixture of "burnt" and "unburnt" gases. The main disadvantage of this approach was that it was unclear how to divide an ordinary turbulent flow into two fluids. Therefore, Spaulding later proposed another version of the two-fluid model, where the turbulent flow is divided into "faster" and "slower" fluids. In this model, he introduced a symmetrical law of mass transfer between two fluids. Based on Prandtl's hypothesis, he derived a formula for eddy viscosity and pointed out the existence of a transverse force arising in a shear velocity field. This revised two-fluid model has been successfully applied to a wide range of confined and free turbulent shear flow. However, in Spalding's two-fluid model, additional equations were used to find the Prandtl path length. As a result, the number of equations doubled compared to linear RANS models, which increased computational resources. Most likely for this reason, Spalding's two-fluid model has not been widely used for solving engineering problems. However, recent publications [27–29] have developed a new two-fluid model. In [27], based on Reynolds' first hypothesis (the velocity in a turbulent flow consists of an averaged and a fluctuating velocity), it is shown that a turbulent flow can be represented as a heterogeneous mixture of two fluids that perform relative movements. In [28], based on the two-fluid approach, modeling of heat transfer is shown, and in [29], this approach is used to simulate a multicomponent compressible flow. To verify this model, a number of turbulence problems from the NASA database were solved. Comparisons with reliable experimental data showed high accuracy, stability and ease of implementation of the new two-fluid model. The advantage of the proposed model is that it is capable of describing complex anisotropic turbulence. For example, in [30], a two-fluid model was successfully used to calculate an air centrifugal separator.

The purpose of this work is to use a two-fluid turbulence model to solve the problem of separated flow around a square block and compare it with the experimental results [7] at Reynolds number $Re = 47000$.

2. Basic equations for turbulent flow.

The main equations for studying the problem posed are the hydrodynamic equations of the two-fluid model [27] for an incompressible medium

$$\begin{cases} \frac{\partial V_j}{\partial x_j} = 0, \\ \frac{\partial V_i}{\partial t} + \frac{\partial V_j V_i}{\partial x_j} + \frac{\partial p}{\rho \partial x_i} = \frac{\partial}{\partial x_j} \left[\nu \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) - g_j g_i \right], \\ \frac{\partial g_i}{\partial t} + \frac{\partial V_j g_i}{\partial x_j} = -g_j \frac{\partial V_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\nu_{ji} \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) \right] + F_{si} + F_{fi}, \\ \nu_{ji} = 3\nu + 2 \frac{g_i g_j}{\text{def}(\vec{V})} \quad i \neq j, \quad \nu_{ii} = 3\nu + \frac{g_k g_k \left| \frac{\partial g_k}{\partial x_k} \right|}{\text{def}(\vec{V}) \left| \frac{\partial g_j}{\partial x_j} \right|}, \quad \vec{F}_f = -K_f \vec{g}, \quad \vec{F}_s = C_s \text{rot} \vec{V} \times \vec{g}. \end{cases} \quad (1)$$

In the given system of equations, V_i is the component of the average flow velocity, g_i is the component of the relative velocity, ν_{ji} are the effective molar viscosities, p is the pressure, ρ is the density of the medium, ν is the molecular viscosity, K_f is the coefficient of friction, C_s is the coefficient at the Saffman force, $\text{def}(\vec{V})$ is the strain rate, determined as follows:

$$\text{def}(\vec{V}) = \sqrt{2S_{i,j}S_{i,j}}, \quad S_{i,j} = \frac{1}{2} \left(\frac{\partial V_i}{\partial x_j} + \frac{\partial V_j}{\partial x_i} \right) \quad (2)$$

The coefficient of friction is determined from the following relation

$$K_f = C_1 \lambda_{\max} + C_2 \frac{|d \cdot g|}{d^2} \quad (3)$$

In this expression, d is the nearest distance to the solid wall, $\lambda_{\max}$ is the real part of the largest root of characteristic equation $\det(A - \lambda E) = 0$, where matrix A has the following form

$$A = \begin{pmatrix} -\frac{\partial V_1}{\partial x_1} & -\frac{\partial V_1}{\partial x_2} - C_s \zeta_3 & -\frac{\partial V_1}{\partial x_3} + C_s \zeta_2 \\ -\frac{\partial V_2}{\partial x_1} + C_s \zeta_3 & -\frac{\partial V_2}{\partial x_2} & -\frac{\partial V_2}{\partial x_3} - C_s \zeta_1 \\ -\frac{\partial V_3}{\partial x_1} + C_s \zeta_2 & -\frac{\partial V_3}{\partial x_2} + C_s \zeta_1 & -\frac{\partial V_3}{\partial x_3} \end{pmatrix}, \quad \zeta = \text{rot} \vec{V}. \quad (4)$$

Constants of model are

$$C_s = 0.2, C_1 = 0.7825, C_2 = 0.306. \quad (5)$$

3. Solution domain and boundary conditions.

In the current section, the Malikov model is used to further investigate the case of one and two square cylinders side-by-side at $Re = 47000$. The single-cylinder and two-cylinder casings are chosen both because of their full application in engineering projects such as towers, high-rise buildings, and chimneys. and because it reveals important fundamental hydrodynamic characteristics.

Fig. 1(a, b), a rectangular computational domain $10d \times 24d$ is shown, where $d=1$ is the length of the side of the square, which is also equal to the characteristic length. The square cylinder is located with the center at the point $(0, 0)$. The boundary conditions are shown in fig. 1 (a, b).

As regards the effects of different T/d interval ratios, a total of seven cases were identified for the Malikov model. In these cases, T/d is separately set to 2.4, 3.5 and 5. The Reynolds number $Re = \rho U_0 d / \mu = 47000$, and the time step is fixed at $\Delta t = 0.001$. The grid resolution details are given in Table 1. The node numbers are 40,000. For the case of $T/d=0$ and $T/d=2.4$, the calculated grid is shown as an example in fig. 1(s, e).

Table 1

Mesh details in different cases for flow around two square cylinders located next to each other at Re=47000.

Model	Interval ratios T/d	Amount of elements	Δt
Malikov	2.4	40 000	0.001
	3.5		
	5		

In Fig. 1(a, b) shows a diagram of a mesh consisting of quadrangular elements for the flow region. In the case of two cylinders, the rectangular computational domain expands to $10d \times 24d$ (Fig. 1e). The centers of two square cylinders, "1" and "2", are equal to $(0, \pm T/2)$, so the ratio of the distance between the centers of two cylinders can be defined as T/d , where $d = 1$ has the same definition as in case above.

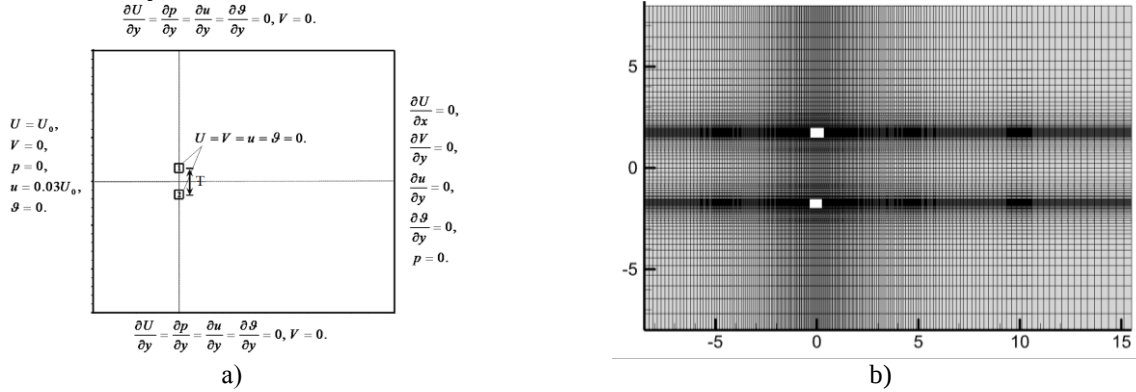


Figure 1. - Two-dimensional computational domain and boundary conditions for flow around two adjacent square cylinders.

4. Solution method

In this work, for the numerical implementation of the system of equations (6), the finite volume method and the upstream scheme of the second order of accuracy are used. To match the pressure with the velocities, the computational grid was placed in a checkerboard pattern. Velocity correction through pressure was carried out using the SIMPLE procedure [31–35]. In the article, a semi-implicit scheme against the flow of the second order of accuracy was used to solve the problem.

5. Calculation results and their discussion

In Fig. 2 (a), where the separation ratio reaches $T/d = 2.2-3.0$, pictures of the interference trace of the slot flow are presented. In this design, the distance between the cylinders increases, which increases the gap flow energy generated in the area between the two cylinders. The slot flow is strong enough to break away from the cylinders, interact with the vortices on the top side of cylinder 1 and the bottom side of cylinder 2 and regenerate an irregular and very complex wake flow (mode A).

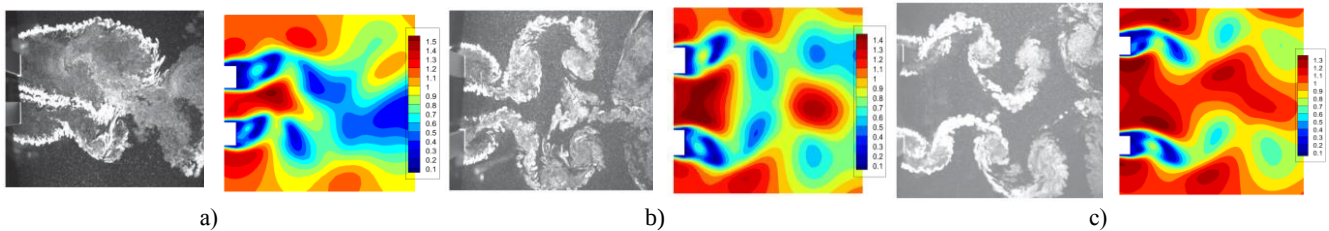


Figure 2. - Instantaneous vorticity distributions of the flow around two square cylinders with different distances T/d at $Re=47000$ study results (a-g-right) in comparison with the cited experimental work (a-g-left) at $Re = 47000$ according to [7], (a) $T/d=2.4$, (b) $T/d=3.5$, (c) $T/d=5$.

In Fig. 2 (a), where the separation ratio reaches $T/d = 2.2-3.0$, pictures of the interference trace of the slot flow are presented. In this design, the distance between the cylinders increases, which increases the gap flow energy generated in the area between the two cylinders. The slot flow is strong enough to

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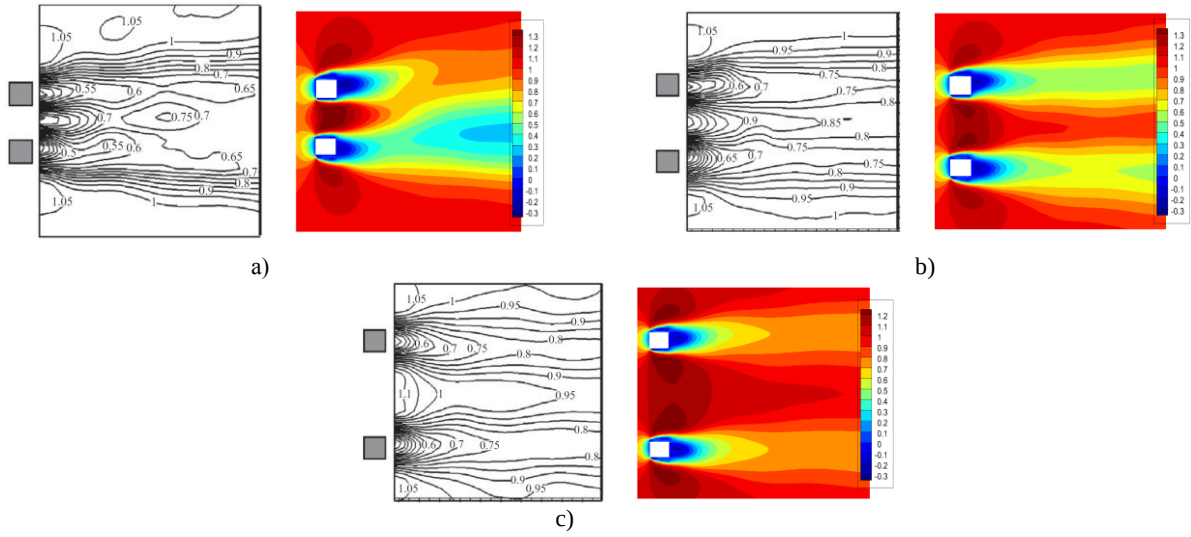


Figure 3. - Isocontours of the averaged longitudinal velocity U in flows around two square cylinders with different distances T/d at $Re=47000$ study results (a-g-right) in comparison with the cited experimental work (a-g-left) at $Re = 47000$ [7], (a) $T/d=2.4$, (b) $T/d=3.5$, (c) $T/d=5$.

In Fig. 4(a)-(d) show the time-averaged transverse velocity fields at different interval ratios.

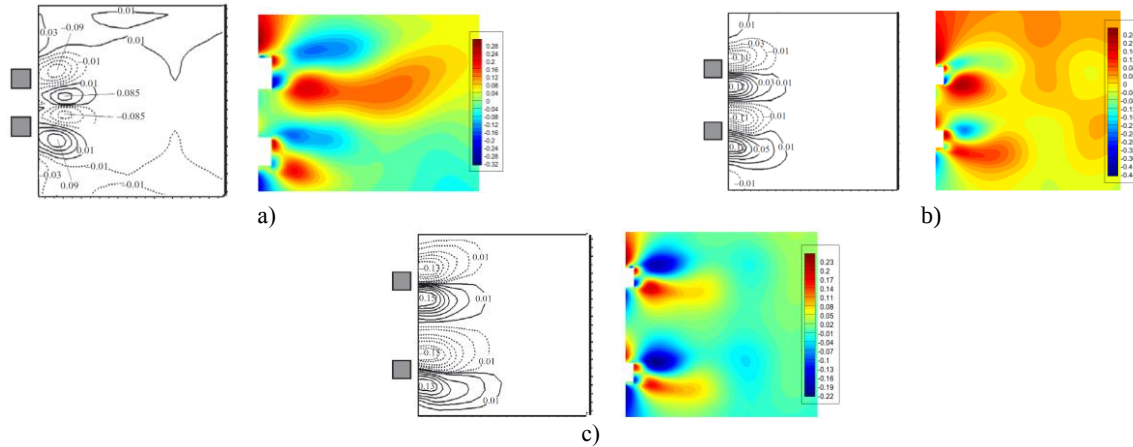
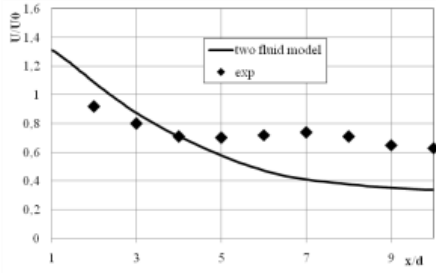


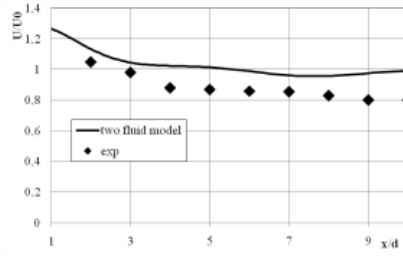
Figure 4. - Isocontours of the average transverse velocity V in the flows around two square cylinders with different distances T/d at $Re=47000$ study results (a-g-right) in comparison with the cited experimental work (a-g-left) at $Re = 47000$ [7], (a) $T/d=2.4$, (b) $T/d=3.5$, (c) $T/d=5$.

An idea of the involvement of the surrounding fluid in the wake, as well as of vortex formation and forces acting on the cylinders, can be obtained by studying the transverse velocity V in various modes (Fig. 4).

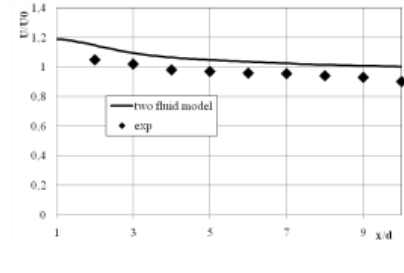
In Fig. Figure 5 shows the time-averaged longitudinal velocity on the center line of the channel for different interval ratios $T/d = 2.4$ to 5 and $Re = 47000$. The averaged velocity was compared with experimental results [7].



T/d = 2.4



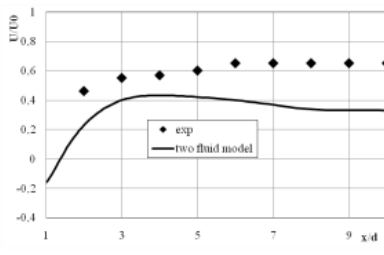
T/d = 3.5



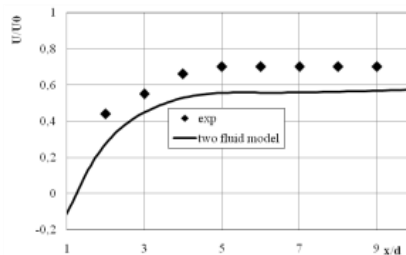
T/d = 5

Figure 5. - Time-averaged longitudinal velocity on the center line for different interval ratios.

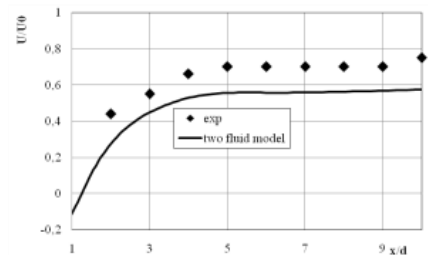
On fig. 6 shows the time-averaged longitudinal velocities on the center line of the first and second cylinders for various interval ratios.



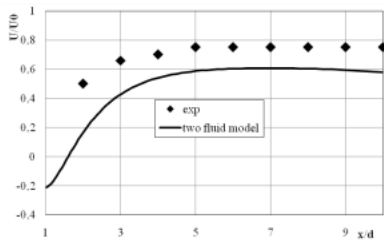
T/d = 2.4, Cylinder 1



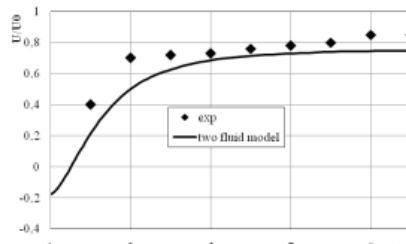
T/d = 2.4, Cylinder 2



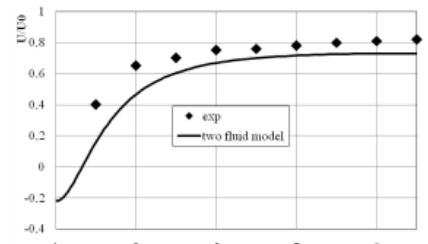
T/d = 3.5, Cylinder 1



T/d = 3.5, Cylinder 2



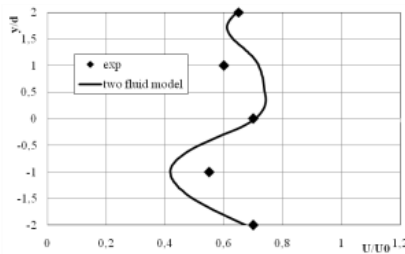
T/d = 5, Cylinder 1



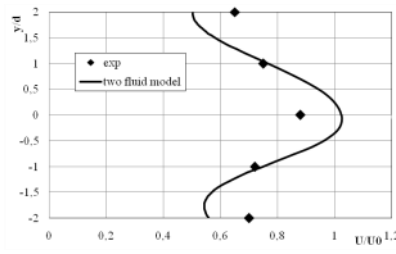
T/d = 5, Cylinder 2

Figure 6. - Time-averaged longitudinal velocity on the center line of the first and second cylinders for different interval ratios.

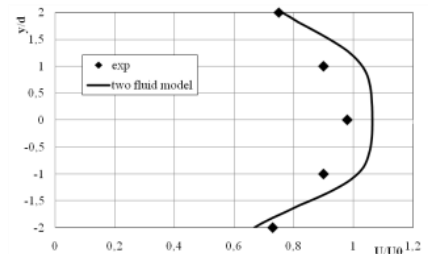
On fig. 7 shows the time-averaged longitudinal velocity per cross section $x/d=4$ for various interval ratios.



T/d = 2.4



T/d = 3.5



T/d = 5

Figure 7. - Time-averaged longitudinal velocity per section $x/d=4$ for different interval ratios.

On fig. 8 shows the time-averaged lateral velocity on the $x/d=1$ and $x/d=2$ sections for different interval ratios.

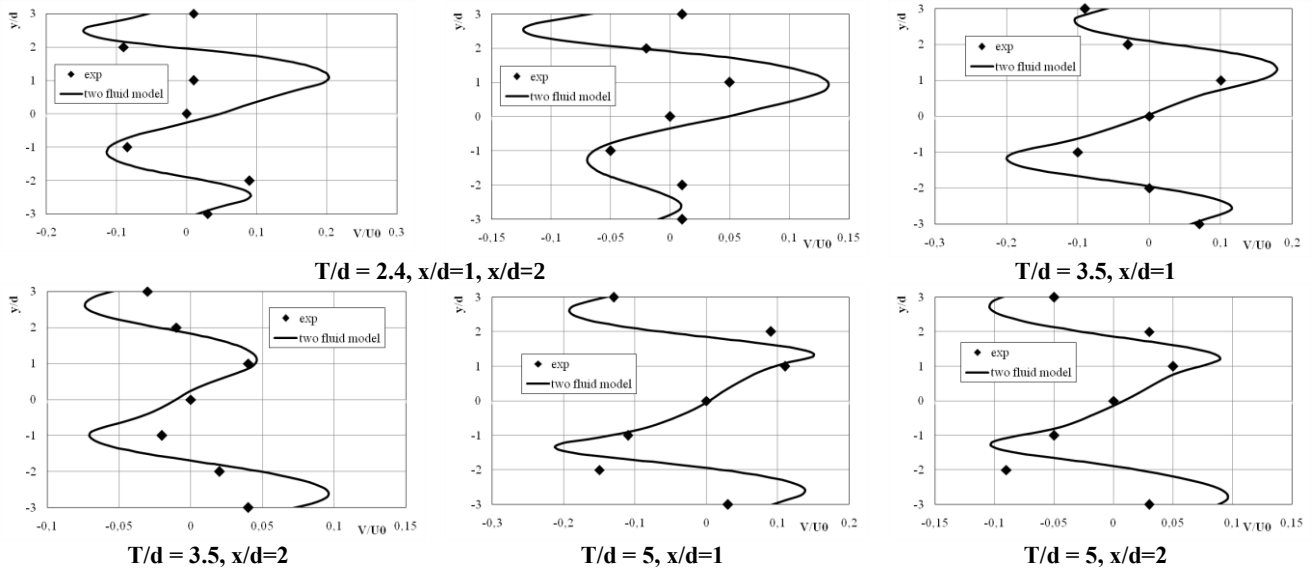


Figure 8. - Time-averaged lateral velocity for different interval ratios.

From Fig. 5-8 it can be seen that Malikov's model very well describes the time-averaged velocity for the interval ratio $T/d = 3.5-5$. In mode C $T/d \geq 3.0$, vortex formation from one cylinder is constantly coupled with vortex formation from another with the same frequencies. In both of these examples, the distance between the cylinders is large enough that a vortex will form and flow down the top and bottom sides of each cylinder, as yours said. And in modes B $1.02 < T/d < 3.5$ the flow in the gap between the cylinders forms one narrow and one wide street, this leads to instability of the flow.

6. Conclusion

The article demonstrates the capabilities of the recently developed "two-fluid" Malikov model for the problem of turbulent flow around a square cylinder. Computational models are compared with experiment using a two-dimensional benchmark problem of flow around a square cylinder at Reynolds number 47000. Flow around two square cylinders placed side by side at $Re=47000$. The interval ratios vary in seven T/d cases: 2.4, 3.5 and 5. Some of the main conclusions on this issue are as follows. For the case of one and two square cylinders ($T/d \geq 3.5$), the Malikov model can predict acceptable results for the distribution of the average pressure coefficient, time-averaged flow velocities in different sections of the channel. For the flow around two square cylinders located side by side, the Malikov model obtained three wake patterns, a body pattern with one flow around, an interference pattern of a slot flow and an antiphase synchronous pattern, which is consistent with the conclusions from experimental data and other publications.

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NUMERICAL STUDY OF FLOW FLOW IN FLAT PLASTIC WITH ZERO PRESSURE GRADIENT BASED ON A TWO-FLUID MODEL.

Malikov Z.M.¹, Nazarov F.Kh.^{1,2}

¹Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan

²National University of Uzbekistan, Tashkent, Uzbekistan

¹malikov.z62@mail.ru, ²farruxnazar@mail.ru

Abstract: This paper presents a numerical study of flow around a plate with zero pressure gradient. To create a mathematical model of the problem, the Navier-Stokes system of equations was used. To close this system of equations, a two-fluid turbulence model was used. For the numerical implementation of the system of equations, von Mises variables were used. a) $Max=2$, $T_w/T_\infty=1.712$; b) $Max=5$, $T_w/T_\infty=1.09$; c) $Max=5$, $T_w/T_\infty=2.725$; d) $Max=5$, $T_w/T_\infty=5.45$ results were obtained in several cases. The results obtained were compared with different values of the Reynolds number.

Key words: Navier-Stokes equations Two-fluid turbulence model, Reynolds number, Mach number, Prandtl number, implicit finite-difference scheme, sweep method.

1. Introduction.

Flow around a flat plate at high Mach numbers is a complex problem in aerodynamics. Such flows are characterized by high dynamics and important physical processes, such as the formation of a shock wave, heating of the gas and changes in its viscosity and density. Understanding flow behavior under these conditions is not only essential for technology development in the aviation and space industries, but also has important implications for other fields such as energy and defense. The selection and development of a suitable turbulence model plays a key role in the study of flow around a flat plate at high Mach numbers. Given the complexity of the physical processes in such flows, it is necessary to use models that can account for their features, including compressibility effects, heat transfer and chemical reactions, if applicable. The development of specialized turbulence models that take into account transition phenomena from laminar to turbulent flow has become an increasingly important task in this area of research.

Turbulence models can be divided into two main categories [1,2]. Empirical Models: These models are based on experimental data and empirical laws that describe the behavior of turbulent flow. They are usually used in cases where it is not sufficient to accurately physically model every aspect of a turbulent flow, and they are assumed to introduce specific parameters that are fitted to the data. However, these models may not be accurate enough for complex flows such as high Mach number flows.

Physically based models: These models are built based on the fundamental equations of turbulence, such as the Navier-Stokes equations, and physical principles. They typically involve a more detailed description of the flow and may take into account compressibility, heat transfer, and other important physical effects. However, their application requires a higher level of computing resources and expert understanding of turbulent processes. The choice between these categories depends on the specific requirements and characteristics of the stream being studied.

Semi-empirical RANS (Reynolds-Averaged Navier-Stokes) models are still not suitable for engineering calculations in aerodynamics and fluid dynamics. They provide the optimal balance between accuracy and computational complexity for a wide range of problems. However, the lack of a universal model for all types of turbulent flows means that a suitable model must be selected depending on the specific conditions of the problem. The SST (shear stress transport) model is a good example of this approach, combining features of the k - ϵ and k - ω models to improve the accuracy of the description of different types of flows. This approach allows the SST model to be successfully applied to a wide range of hydrodynamics and aerodynamics problems, including near-surface flows and jet flows, which makes it one of the most common and widely used models in engineering practice [3–7].

A new two-fluid turbulence model has recently been published [8–10]. In this paper, a new model is used to study turbulent flow around a plate with zero pressure gradient. The system of dimensionless equations in the Cartesian coordinate system of the new two-fluid model has the form [11].

$$\left\{ \begin{array}{l} \frac{\partial \rho U}{\partial x} + \frac{\partial \rho V}{\partial y} = 0, \\ U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} = \frac{\partial}{\partial y} \left[\rho \nu \frac{\partial U}{\partial y} - \rho u g \right], \\ U \frac{\partial u}{\partial x} + V \frac{\partial u}{\partial y} = -(1 - C_s) \frac{\partial U}{\partial y} g + \frac{\partial}{\partial y} \left[\rho \nu_{xy} \frac{\partial u}{\partial y} \right] - K_f u, \\ U \frac{\partial g}{\partial x} + V \frac{\partial g}{\partial y} = -C_s \frac{\partial U}{\partial y} u + \frac{\partial}{\partial y} \left[2 \rho \nu_{yy} \frac{\partial g}{\partial y} \right] - K_f g, \\ U \frac{\partial T}{\partial x} + V \frac{\partial T}{\partial y} = \frac{\partial}{\partial y} \left[\rho k \frac{\partial T}{\partial y} - \rho t g \right] + \frac{\nu(\gamma - 1)}{\gamma R} \left[\left(\frac{\partial U}{\partial y} \right)^2 + \left(\frac{\partial u}{\partial y} \right)^2 - \frac{2}{3} \left(\frac{\partial g}{\partial y} \right)^2 \right], \\ U \frac{\partial t}{\partial x} + V \frac{\partial t}{\partial y} = -\frac{\partial T}{\partial y} g + \frac{\partial}{\partial y} \left[\rho k_y \frac{\partial t}{\partial y} \right] + \frac{2\nu(\gamma - 1)}{\gamma R} \frac{\partial U}{\partial y} \frac{\partial u}{\partial y} - K_t t. \end{array} \right. \quad (1)$$

K_f - friction coefficient and K_t - heat transfer coefficient, which are equal

$$K_f = C_1 \sqrt{C_s(1 - C_s)} \left| \frac{\partial U}{\partial y} \right| + C_2 \frac{|g|}{y} \quad (2)$$

$$K_t = C_{t1} \sqrt{C_s(1 - C_s)} \left| \frac{\partial U}{\partial y} \right| + C_{t2} \Pr \left(\frac{T_t}{T_w} \right)^{0.5} \frac{|g|}{y} \quad (3)$$

First members in K_f, K_t mean the contribution of the vortex flow, and the second the influence of the wall on the coefficients. In expressions (2.3) the empirical constants $C_1 = 0.7825$, $C_2 = 0.306$, $C_{t1} = 0.92$, $C_{t2} = 0.3$, $C_s = 0.3$, l - closest distance to the wall, τ_w - wall friction stress, T_w - wall temperature, T_t - flow stagnation temperature and $\Pr$ - molecular Prandtl number. The flow stagnation temperature is determined from the relation

$$T_t = T \left(1 + \frac{\gamma - 1}{2} M^2 \right) \quad (4)$$

Here M is the Mach number.

λ_{max} the largest root of the characteristic equation (1) is equal to

$$\lambda_{max} = \sqrt{C_s(1 - C_s)} \left| \frac{\partial U}{\partial y} \right| \quad (5)$$

Molar viscosities are determined by the relations:

$$\nu_{xy} = 3\nu + 2 \left(\frac{u g}{\frac{\partial U}{\partial y}} \right), \nu_{yy} = 3\nu + 2 \left(\frac{g g}{\frac{\partial U}{\partial y}} \right), k_y = 3k + 2 \left(\frac{t g}{\frac{\partial T}{\partial y}} \right) \quad (6)$$

2. Formulation of the problem.

In Figure 1 shows the arrangement of simple flat plate arrays and typical boundary conditions for supersonic flow.

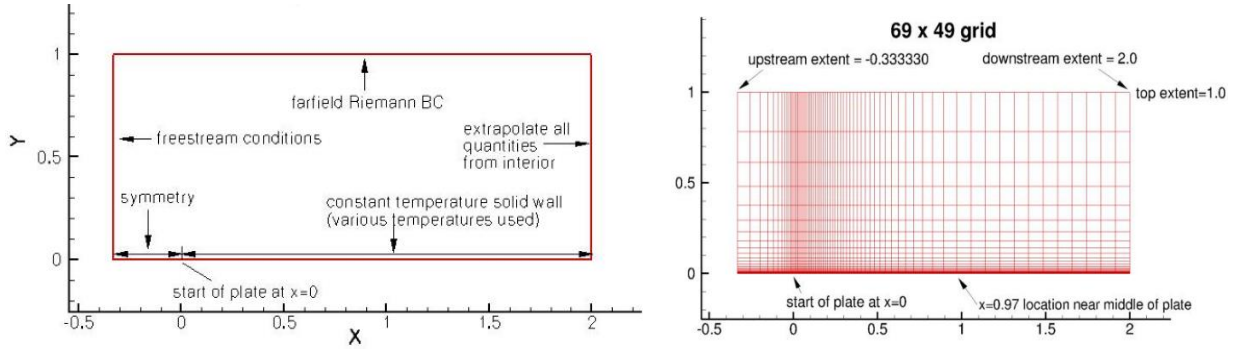


Figure 1. - Flat plate with zero pressure gradient a) boundary conditions and b) computational mesh and domain

3. Solution method.

For the numerical implementation of system (1), Von Mises variables were used $-(\xi, \psi)$ [12]:

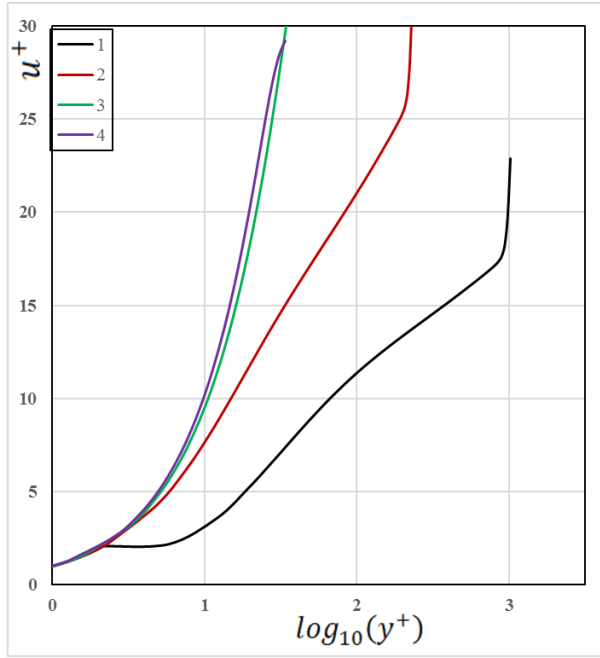
$$\begin{aligned}\xi &= x, \quad \rho U = \frac{\psi \partial \psi}{\partial y}, \quad \rho V = -\frac{\psi \partial \psi}{\partial x}, \\ \frac{\partial}{\partial x} &= \frac{\partial}{\partial \xi} - \frac{\rho V}{\psi} \frac{\partial}{\partial \psi}, \quad \frac{\partial}{\partial y} = \frac{\rho U}{\psi} \frac{\partial}{\partial \psi}\end{aligned}\tag{7}$$

Substituting (7) into (1), we get the following:

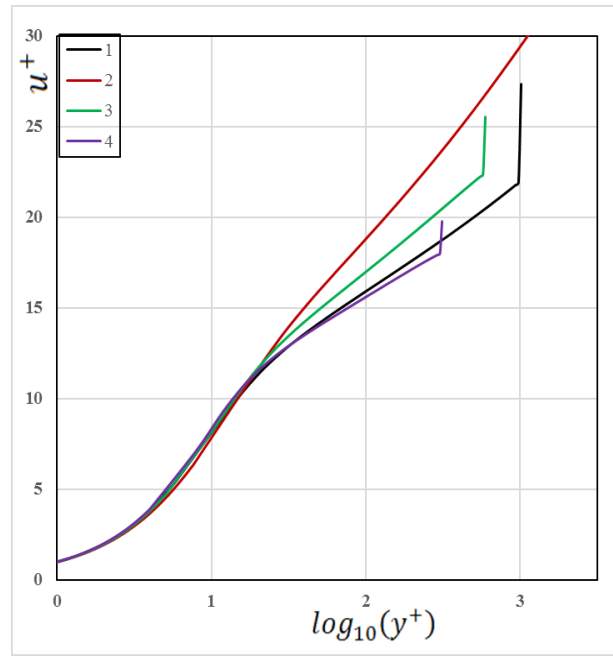
$$\begin{cases} \frac{\partial U}{\partial \xi} = \frac{\partial}{\psi \partial \psi} \left[\frac{\rho v U}{\psi} \frac{\partial U}{\partial \psi} - \rho u g \right], \\ \frac{\partial u}{\partial \xi} = -\rho(1-C_s) \frac{\partial U}{\psi \partial \psi} g + \frac{\partial}{\psi \partial \psi} \left[\frac{\rho^2 v_{xy} U}{\psi} \frac{\partial u}{\partial \psi} \right] - K_f \frac{u}{U}, \\ \frac{\partial g}{\partial \xi} = -\rho C_s \frac{\partial U}{\psi \partial \psi} u + \frac{\partial}{\psi \partial \psi} \left[\frac{2\rho^2 v_{yy}}{\psi} \frac{\partial g}{\partial \psi} \right] - K_f \frac{g}{U}, \\ \frac{\partial T}{\partial \xi} = \frac{\partial}{\psi \partial \psi} \left[\frac{\rho k U}{\psi} \frac{\partial T}{\partial \psi} - \rho t g \right] + \frac{\rho^2 U v (\gamma - 1)}{\psi^2 \gamma R} \left[\left(\frac{\partial U}{\partial \psi} \right)^2 + \left(\frac{\partial u}{\partial \psi} \right)^2 - \frac{2}{3} \left(\frac{\partial g}{\partial \psi} \right)^2 \right], \\ \frac{\partial t}{\partial \xi} = -\rho \frac{\partial T}{\psi \partial \psi} g + \frac{\partial}{\psi \partial \psi} \left[\frac{U \rho^2 k_y}{\psi} \frac{\partial t}{\partial \psi} \right] + \frac{2\rho^2 v U (\gamma - 1)}{\psi^2 \gamma R} \frac{\partial U}{\partial \psi} \frac{\partial u}{\partial \psi} - K_t \frac{t}{U}. \end{cases}\tag{8}$$

Numerical calculations of equations in new variables were carried out using an implicit finite-difference scheme. In the transverse direction, the central difference is used. To solve the implicit scheme, the sweep method was used [13–15].

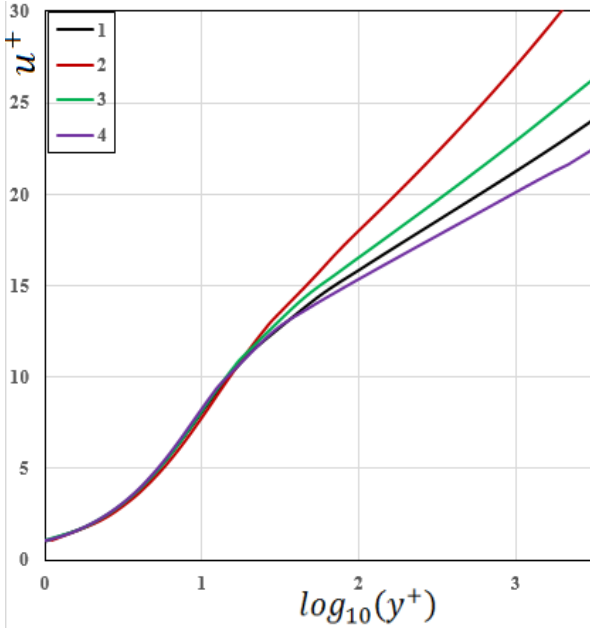
Figure 2 shows the dimensionless velocity profiles at various temperatures and the Reynolds number of the plate.



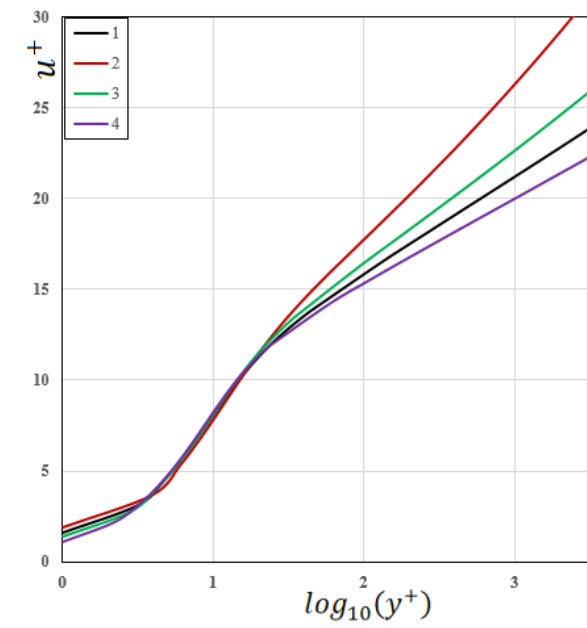
Re=50 000



Re=500 000



Re=5000 000



Re=15000 000

Figure 2. -. Dimensionless velocity profiles at different Reynolds numbers of the plate. 1- Max=2, $T_w/T_\infty=1.712$; 2- Max=5, $T_w/T_\infty=1.09$; 3- Max=5, $T_w/T_\infty=2.725$; 4- Max=5, $T_w/T_\infty=5.45$.

The temperature profile of the flow velocity at different Reynolds numbers is also of great interest. Therefore, Figure 3 shows the profile of the value T/T_∞ .

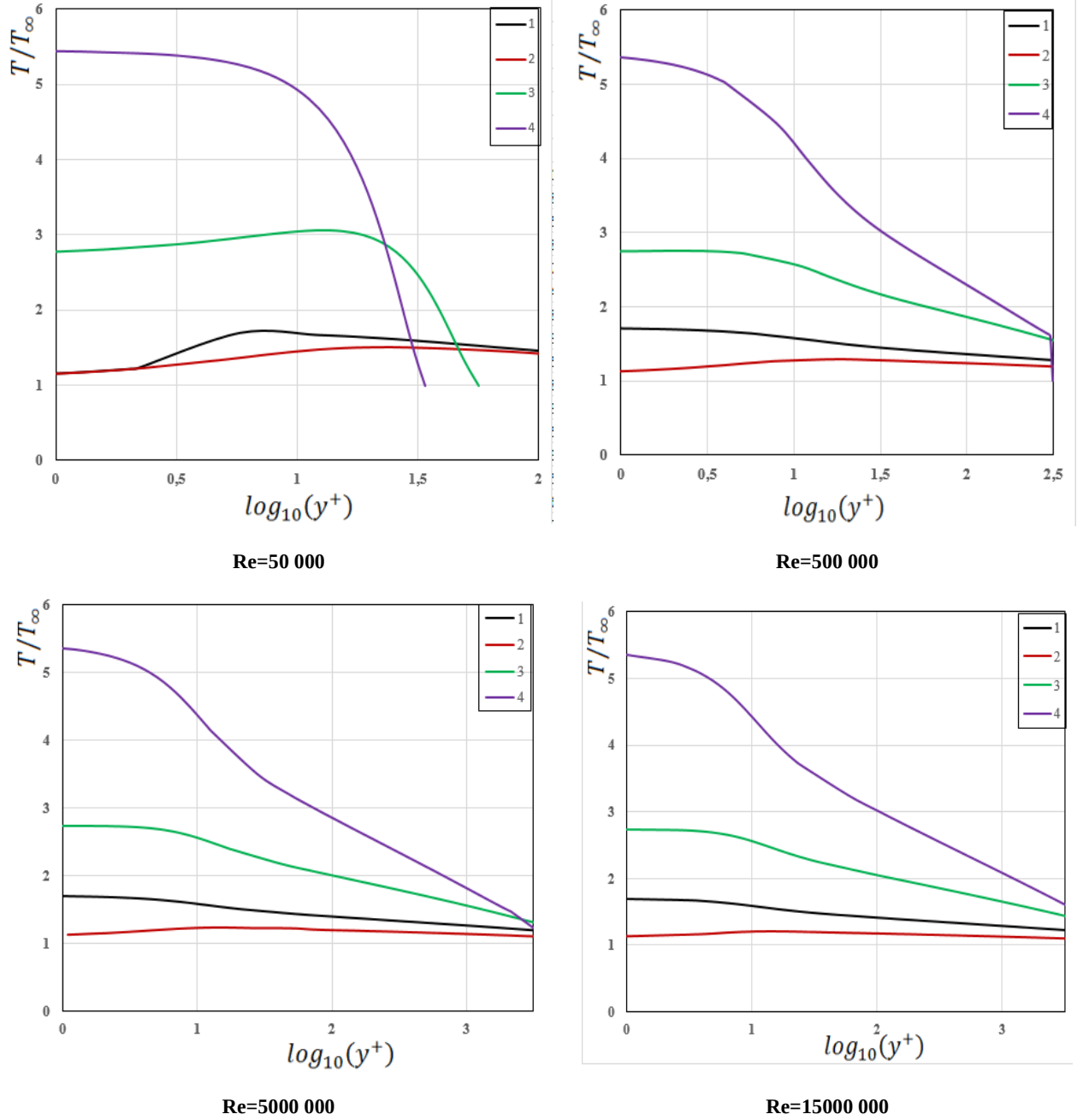


Figure 3. -. **Dimensionless temperature profile.** 1- Max=2, $T_w/T_\infty=1.712$; 2- Max=5, $T_w/T_\infty=1.09$; 3- Max=5, $T_w/T_\infty=2.725$; 4- Max=5, $T_w/T_\infty=5.45$.

To determine the Prandtl number, we represent the turbulent stress and heat flow in the form

$$\text{Pr}_t = \frac{\nu_t}{k_t} = \left| \frac{u \partial T / \partial y}{t \partial U / \partial y} \right| \quad (9)$$

Here ν_t , k_t turbulent coefficients of viscosity and thermal diffusivity. From these expressions we find that

$$\nu_t = -\frac{u \vartheta}{\partial U / \partial y}, \quad k_t = -\frac{t \vartheta}{\partial T / \partial y}. \quad (10)$$

Figure 3 shows the profile of the turbulent Prandtl number.

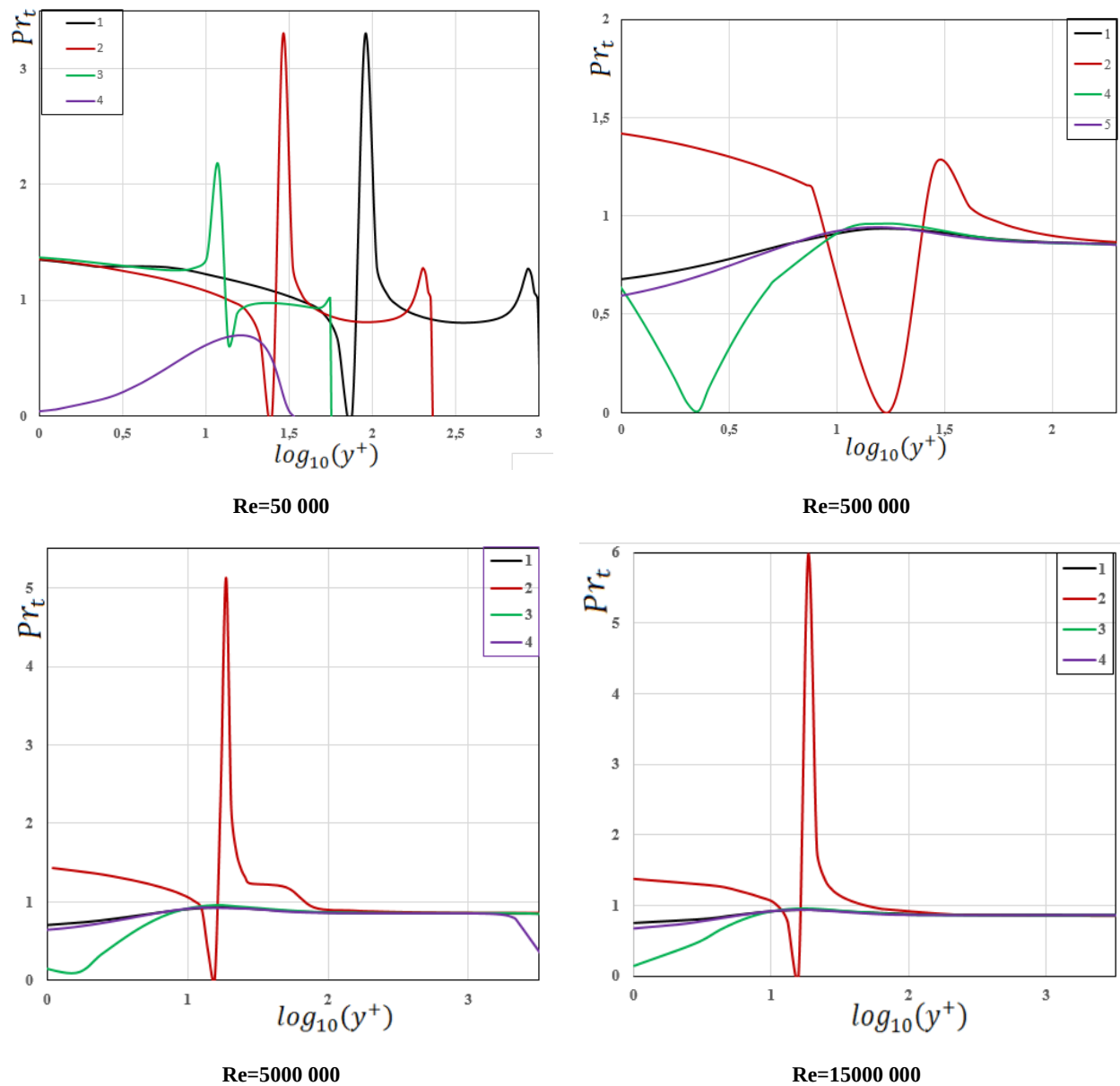
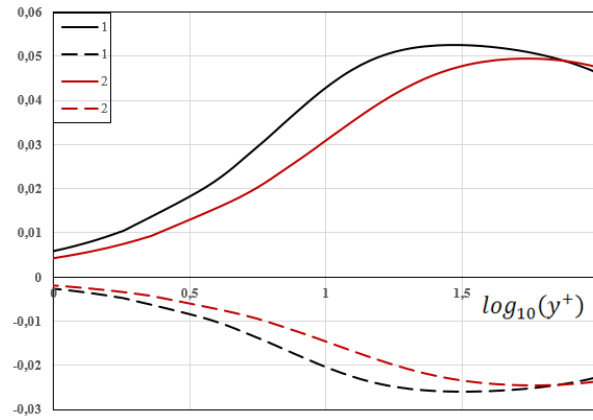


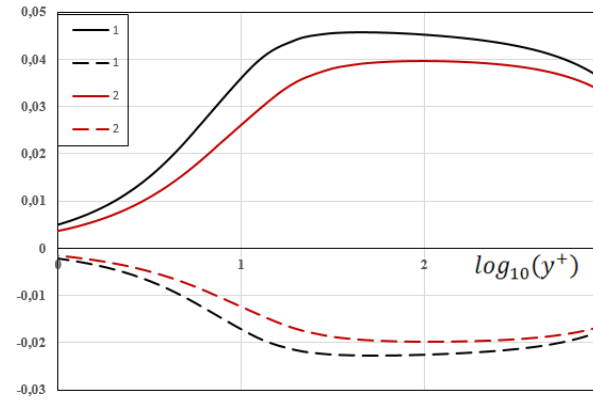
Figure 4. - Turbulent Prandtl number profile. 1- Max=2, $T_w/T_\infty=1.712$; 2- Max=5, $T_w/T_\infty=1.09$; 3- Max=5, $T_w/T_\infty=2.725$; 4- Max=5, $T_w/T_\infty=5.45$.

From Figure 4 it can be seen that the turbulent Prandtl number in the viscous and transition layers has different values at different plate temperatures. However, far from the plate, the turbulent Prandtl number is the same and equals 0.9 for all the given variants of Reynolds numbers 500000, 5000000 and 15000000.

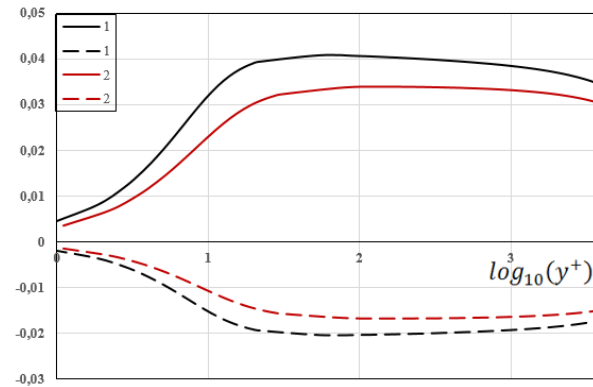
In Figure 5 shows dimensionless profiles of relative longitudinal and transverse velocities for different Reynolds numbers. From this figure it can be seen that in the turbulent zone the value of the longitudinal velocity is 2 times higher than the value of the transverse velocity. Here the relative longitudinal velocity is a straight line, the relative transverse velocity is a dotted line



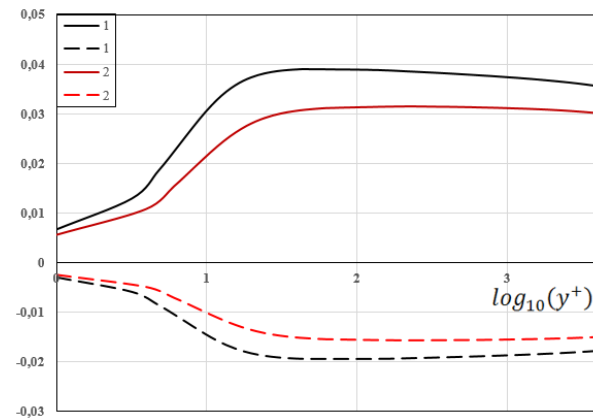
Re=50 000



Re=500 000



Re=5000 000



Re=15000 000

Figure 5. - Longitudinal and transverse velocity profiles 1- Max=2, $T_w/T_\infty=1.712$; 2- Max=5, $T_w/T_\infty=1.09$

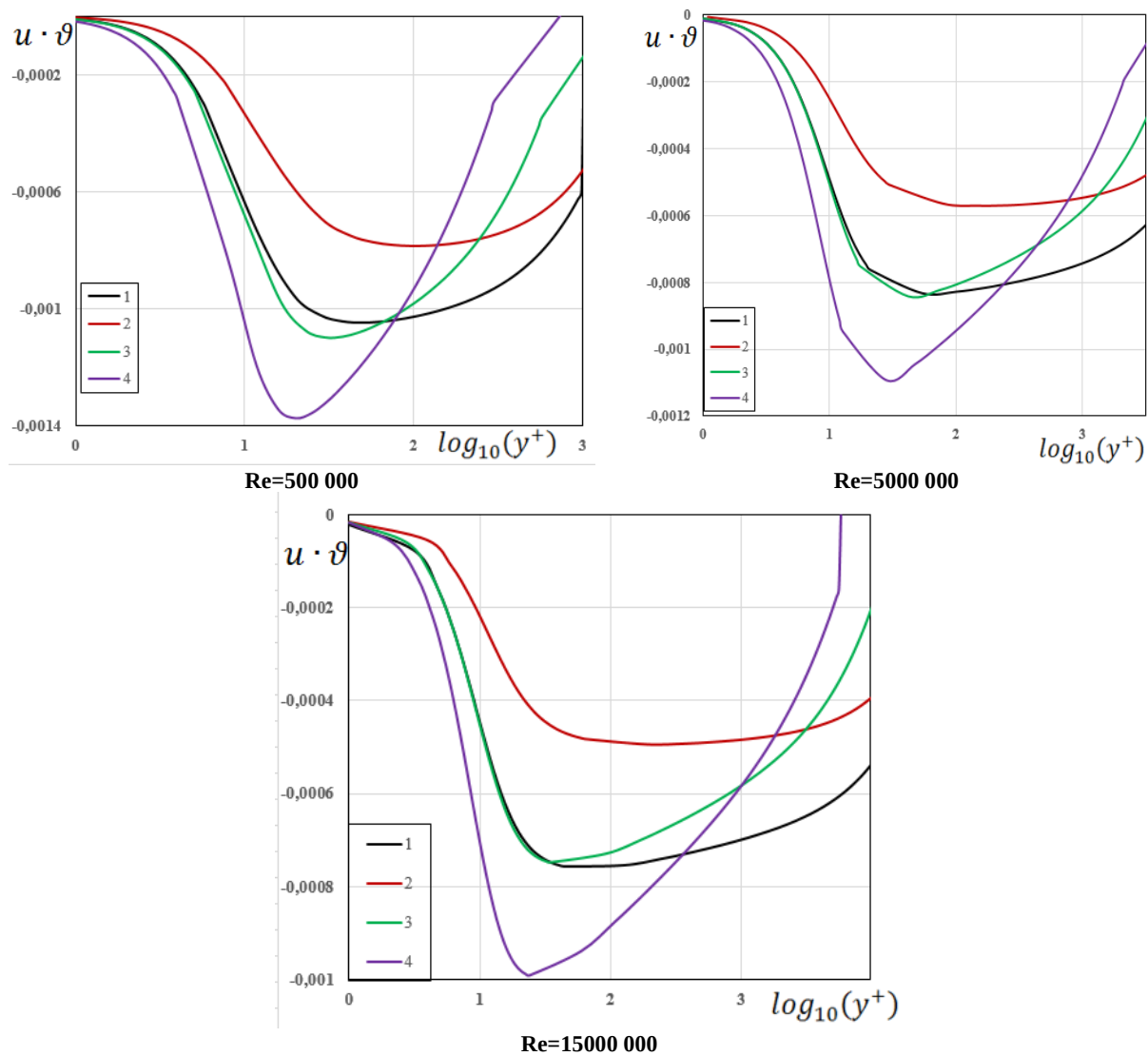


Figure 6. - Profiles of turbulent stresses at different values of Reynolds numbers 1- Max=2, $T_w/T_\infty=1.712$; 2- Max=5, $T_w/T_\infty=1.09$; 3- Max=5, $T_w/T_\infty=2.725$; 4- Max=5, $T_w/T_\infty=5.45$.

4. Conclusion.

As can be seen from the numerical results obtained, the temperature of the flow distribution has almost the same values at different values of the Reynolds number. Moreover, when calculating the Prantl number after the transition from the laminar to the turbulent region, the same value of 0.9 was obtained. The two-fluid model is also capable of describing the transition zone from laminar to turbulent flow. Thus, the two-fluid turbulence model has much higher accuracy.

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COMPARISON OF HIGH EFFICIENCY CENTRIFUGAL HYDROCYCLONES

Malikov Z.M.¹, Yuldashev A.T.¹, Nuritdinov A.A.², Shoev M.A.³

¹*Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan*

²*JSK « Almalyk MMK» (AMMK), Tashkent, Uzbekistan*

³*Fergana Polytechnic Institute, Fergana, Uzbekistan*

E-mail: shoyevmardonbek1@gmail.com

Abstract: The behavior of flow in a hydrocyclone is a rather complex phenomenon. This complexity has led developers to rely on empirical equations to predict hardware performance. There are many more publications on empirical hydrocyclone models than there are attempts to model fluid flow. Empirical models relate a classification parameter such as fraction size to device dimensions and suspension properties. However, they can only be used within the limits of the experimental data from which the model parameters were determined. On the other hand, models based on computational fluid dynamics (CFD) techniques have proven useful in simulating fluid flow in hydrocyclones and in predicting separator solids separation efficiency for a wide range of operating and design conditions. The shape and size of the hydrocyclone separator have a direct effect on the structure of the internal continuous phase flow and thus the separation of the solid phase. The work compares highly efficient centrifugal hydrocyclones, the original hydrocyclone and the new highly efficient hydrocyclone. The numerical study was carried out using the Comsol Multiphysics 6.1 software package. The v^2 -f model was chosen as the turbulence model. Based on the obtained numerical data, it is shown that the efficiency of the new hydrocyclone is significantly greater than that of the original hydrocyclone.

Keywords: cyclone, mathematical modeling, turbulence models, Reynolds-averaged Navier-Stokes equations, model v^2 -f.

1. Introduction.

Since the middle of the last century, the world has seen a rapid increase in the use of hydrocyclones in various industries, such as chemical, mining, coal, powder, etc. The reasons for this popularity lie in the simplicity of design and operation, high performance, low maintenance and operating costs, and the small physical size of the device.

A typical hydrocyclone consists of a cylindrical section with a central pipe connected to a conical section with an outlet pipe. The inlet pipe is attached to the top of the cylinder. Liquid injected tangentially into the hydrocyclone causes vortex and thus creates centrifugal force within the device. This centrifugal force field results in the rapid sorting of solid particles from the environment in which

they are found [1-4].

The flow behavior in a hydrocyclone is quite complex. This flow complexity has led designers to rely on empirical equations to predict hardware performance. These empirical relationships are derived from the analysis of experimental data and include the influence of operational and geometric variables. Different sets of experimental data lead to different equations for the same basic parameters [5-8]. Empirical models relate a classification parameter, such as slice size, to device dimensions and multiphase mixture properties (Dahlstrom, 1949; Yoshiota and Hotta, 1955; Fahlstrom, 1963; Agar and Herbst, 1966; Lynch and Rao, 1975; Plitt, 1976). However, these models, like any other empirical models, suffer from the inherent disadvantage that the model can only be used within the limits of the experimental data from which the model parameters were determined. Due to this shortcoming, mathematical models based on fluid mechanics are highly desirable.

Computational fluid dynamics (CFD) is a versatile tool for predicting velocity profiles over a wide range of design and operating conditions. Numerical treatment of the Navier–Stokes equations is the basis of any CFD method, gradually making its way into hydrocyclone analysis in the early 1980s. This occurred as a result of rapid improvements in computers and a better understanding of the numerical processing of turbulence [9-11].

In many cases, to improve the efficiency of the device, it is necessary to develop a fundamentally new hydrocyclone. Therefore, in this work, a new hydrocyclone is investigated. The study is mainly comparative in nature, i.e. The effectiveness of a classic hydrocyclone is compared with a new one. Figure 1- shows the schematic diagrams of a classic and new centrifugal hydrocyclone.

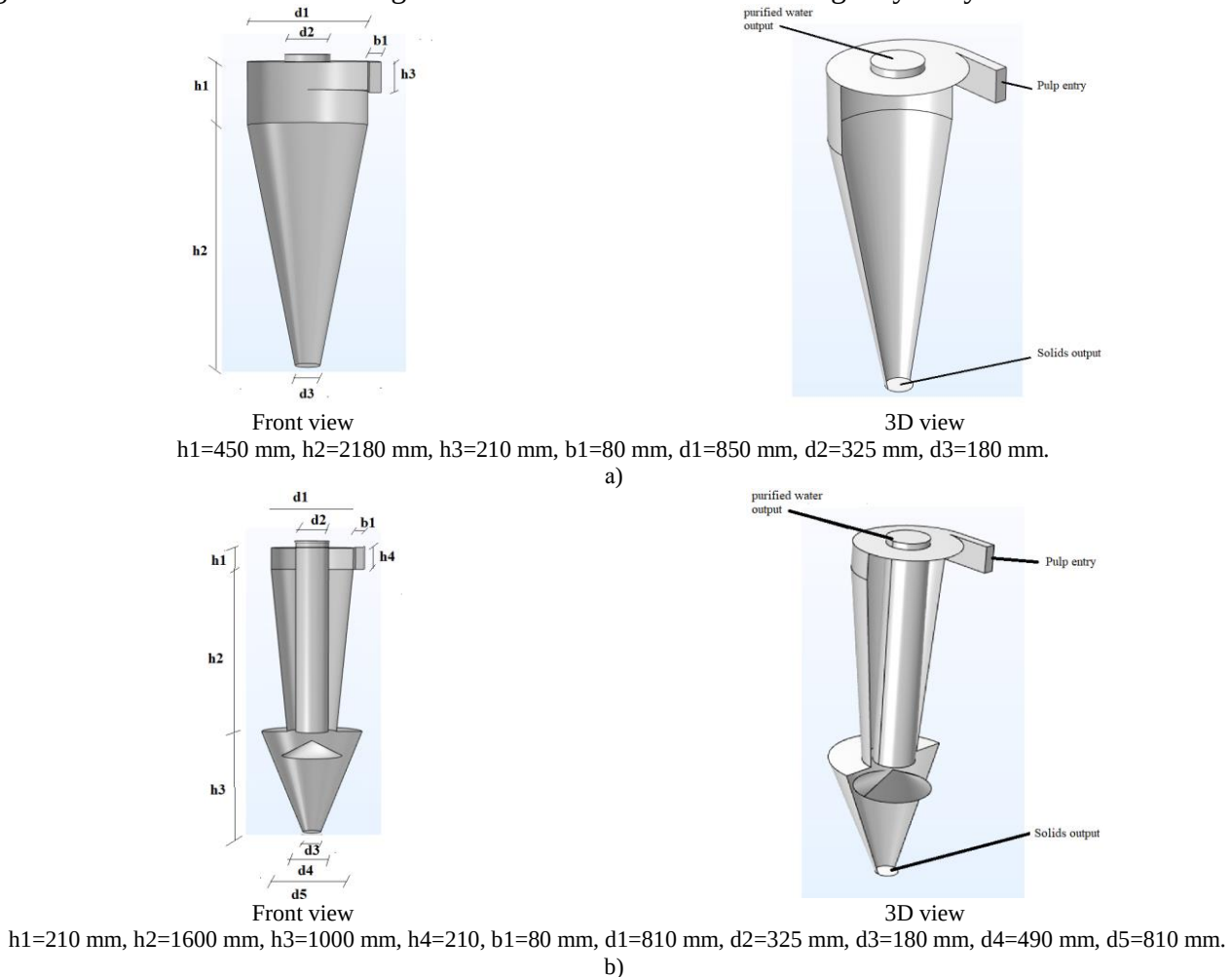


Figure 1- Centrifugal hydrocyclones. a) original hydrocyclone, b) new centrifugal hydrocyclone.

2. Physical and mathematical formulation of the problem.

The present study uses numerical simulation to predict the flow field variables, pressure drop, and collection efficiency. The Navier–Stokes equations are discretized using the finite element method (FEM). The choice of a particular turbulence model depends on the nature of the flow, the available hardware resources, and its ability to capture the complete physics of fluid flow. To model Reynolds stresses in RANS equations, Comsol Multiphysics 6.1 provides various closure models; The v^2 - f model was used in this work. The v^2 - f models are very robust turbulence models suitable for most engineering and industrial flows.

To study the flow dynamics, consider a system of hydrodynamic equations. In this case, we will make the assumption that the volume density of particles in the main flow zone is negligible compared to the gas flow density. Indeed, in real cyclones the density of dust does not exceed 10 g/m^3 , and the density of water does not exceed 1000 kg/m^3 . However, the density of dust near the wall due to accumulation can greatly exceed this indicator. But this happens in a rather narrow layer, the size of which can be neglected in comparison with the characteristic parameters of the hydrocyclone.

To numerically study the problem posed, a system of Reynolds-averaged Navier-Stokes equations is used [12,13].

The system of equations does not take into account the forces due to the effects of turbulent migration, Sefman, Magnus (lift) and Coriolis. Because they are significantly less than centrifugal force. Thus, for mathematical modeling of the processes of transfer of dust particles and aerosols in dust collectors, it is sufficient to take into account centrifugal, gravitational and Stokes forces.

$$\begin{cases} \frac{\partial \bar{U}_i}{\partial t} + \bar{U}_j \frac{\partial \bar{U}_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + F + \frac{\partial}{\partial x_j} \left[(\nu + \nu_t) \left(\frac{\partial \bar{U}_i}{\partial x_j} + \frac{\partial \bar{U}_j}{\partial x_i} \right) \right], \\ \frac{\partial \bar{U}_{pi}}{\partial t} + \bar{U}_j \frac{\partial \bar{U}_{pi}}{\partial x_j} = k_m (\bar{U}_i - \bar{U}_{pi}), \\ \frac{\partial \bar{\rho}_m}{\partial t} + \bar{U}_{pi} \frac{\partial \bar{\rho}_m}{\partial x_j} = D \frac{\partial}{\partial x_j} \left[\frac{\partial \bar{\rho}_m}{\partial x_j} + \frac{\partial \bar{\rho}_m}{\partial x_i} \right], \\ \frac{\partial \bar{U}_j}{\partial x_j} = 0. \end{cases} \quad (1)$$

Here $\bar{U}_i$ – respectively, axial, radial and tangential components of air flow velocity;

$\bar{U}_{pi}$ – similar velocity components for the m th dust fraction; $\bar{p}$ – hydrostatic pressure; ρ – gas density;

ν , ν_t – molecular and turbulent viscosities; ρ_m – dust mass density; k_m – coefficient of interaction between

air and m -th fraction of dust; N – number of dust fractions; $D = \frac{\rho}{S_c(\rho + \rho_p)}(\nu + \nu_t)$ – solid diffusion

coefficient, $S_c = 0.8$ – Schmidt coefficient.

The interaction coefficient between phases was determined through the Stokes parameter:

$$k_m = \frac{18\rho\nu}{\rho_p \delta_m^2}.$$

In this expression ρ_p – material density of dust particles, δ_m – effective particle diameter.

v^2 - f turbulence model:

Near solid walls, the intensity of velocity fluctuations in the direction tangential to the wall is usually much greater than the intensity of fluctuations in the direction normal to the wall. In other words, velocity fluctuations are characterized by anisotropy. As you move away from the wall, the intensity of fluctuations in all directions becomes the same. Velocity fluctuations become uniform or isotropic. The

anisotropy of turbulent fluctuations in the boundary layer is described by the v^2 - f turbulence model by introducing two additional equations, solved together with the equations for the kinetic energy of turbulence (k) and the rate of dissipation of kinetic energy (ε).

$$\begin{cases} (\mathbf{U} \cdot \nabla)k = \nabla \left[\left(\nu + \frac{\nu_t}{\sigma_k} \right) \nabla k \right] + P - \varepsilon, \\ (\mathbf{U} \cdot \nabla)\varepsilon = \nabla \left[\left(\nu + \frac{\nu_t}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + \frac{1}{\tau} (C_{\varepsilon 1}'(\zeta, \alpha)P_k - C_{\varepsilon 2}'(k, \varepsilon, \alpha)\varepsilon), \\ (\mathbf{U} \cdot \nabla)\zeta = \nabla \left[\left(\nu + \frac{\nu_t}{\sigma_\zeta} \right) \nabla \zeta \right] + \frac{2}{k} \left[\alpha^3 \nu + \frac{\nu_t}{\sigma_\zeta} \right] \nabla k \nabla \zeta + (1 - \alpha^3)f_w + \alpha^3 f_h - \frac{\zeta}{k} P_k, \end{cases} \quad (2)$$

Turbulent eddy viscosity is calculated using: $\nu_t = C_\mu k \zeta \tau$. The remaining coefficients and functions were presented in article [14]. There is no single representation for modeling the kinematics of particle motion in a turbulent two-phase flow that would allow a correct description of the object [15]. The model based on the concept of “trajectory particles” is considered incorrect due to the lack of consideration of the factor of interaction between Reynolds stresses and particles. On the other hand, the advantages of the Lagrangian approach are undeniable, which is closer to real processes and allows one to obtain the necessary information about particle trajectories, the residence time of particles in the apparatus, and the minimum size of captured particles [16–20]. In this regard, in this work, the Lagrangian approach is used to model the efficiency of a centrifugal cyclone.

For discrete-phase modeling, a gil with a density of 1560 kg/m^3 is used. Particle sizes are taken in the range from 10 to 80 microns. Solid particles are injected using the normal surface condition at a speed equal to the speed of the fluid entering the cyclone. Particles reaching the walls of the cyclone collide with it and are reflected back. For the condition of low dust load, it is quite reasonable to assume that the collision of discrete phase particles with the walls of the cyclone is ideally elastic, therefore the recovery coefficients in both the tangential and normal directions of the wall are taken as unity. The total density of gils at the inlet was equal $\rho_m = 10 \text{ g/m}^3$ and distributed uniformly over the cross section.

Calculation grids. In this study, a non-structural type of calculation grid was used; the number of elements for the original hydrocyclone was 172922, and for the new centrifugal hydrocyclone it was 288663 (Figure 2).

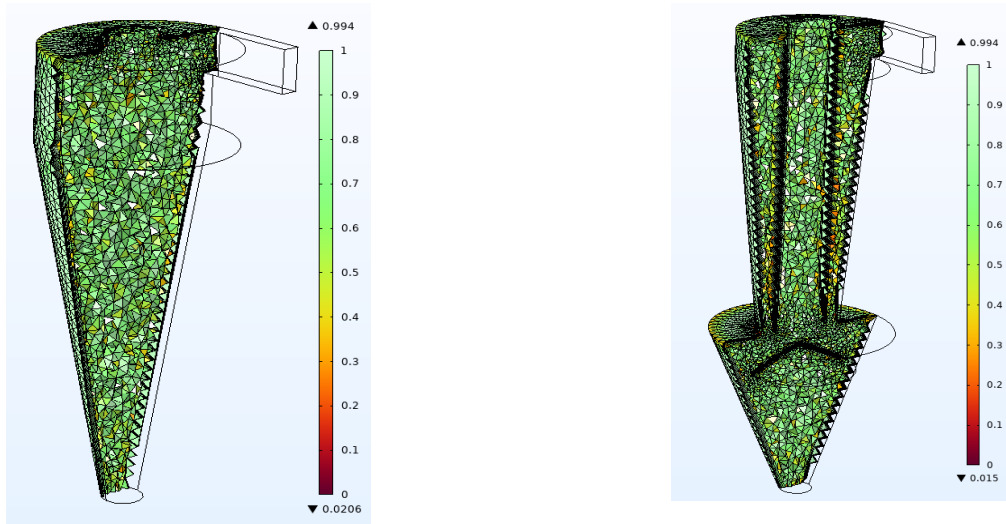


Figure 2-Calculation grids for hydrocyclones.

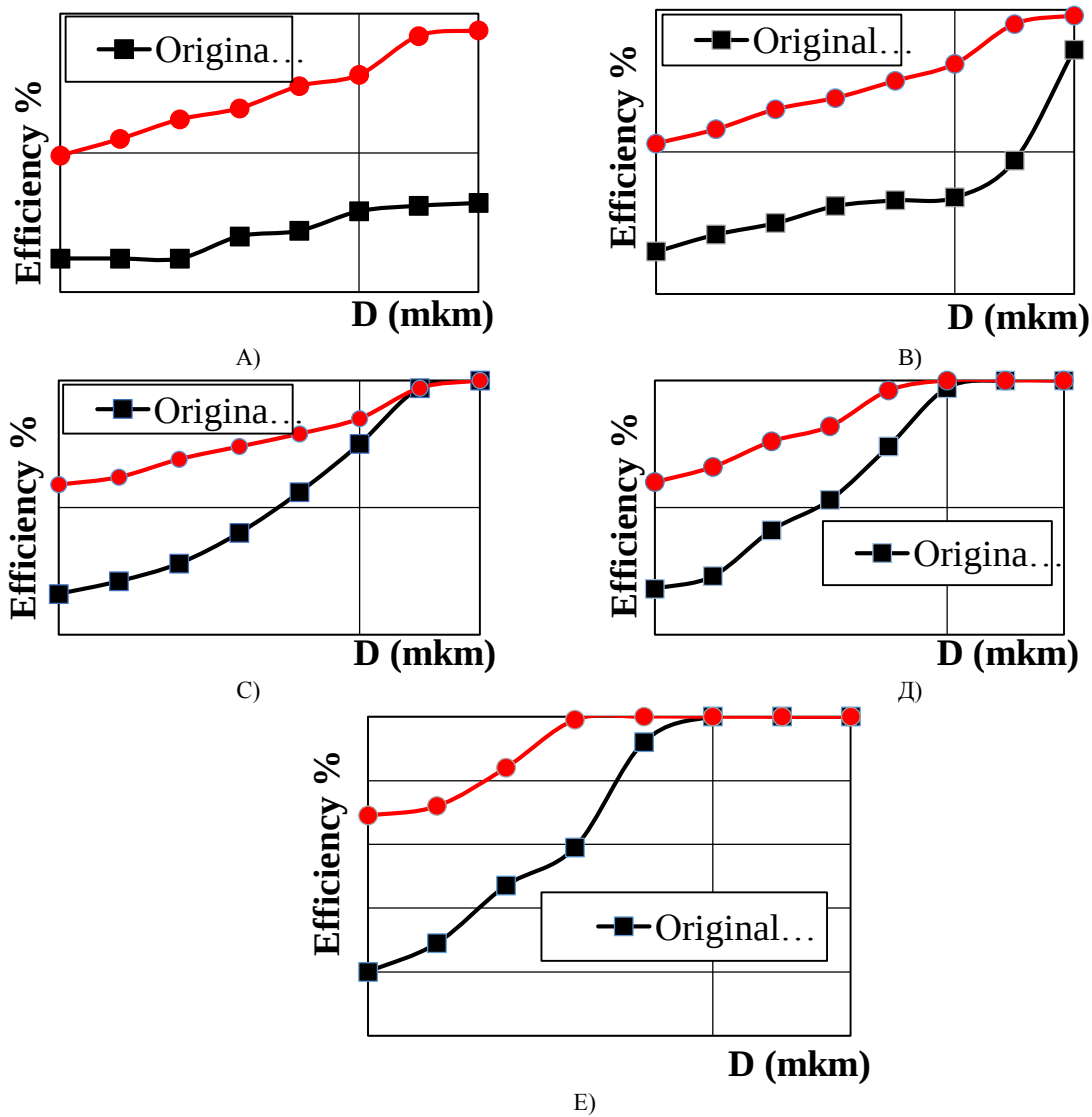


Figure 3- Efficiency of hydrocyclones in particle capture ORH-Original hydrocyclone, RH- Recommended hydrocyclone.

3. Solution method

To numerically solve the system of initial nonstationary equations (1), the finite element method was used. For the solution, standard COMSOL Multiphysics 6.1 solvers were used.

4. Calculation results and their discussion.

To determine the efficiency of dust collectors in Fig. Figure 3 shows the percentage values of outgoing dust particles of a certain fraction at different inlet speeds.

From Fig. 3 shows that at a speed of 1 m/s the recommended hydrocyclone represents 71%, at a speed of 2 m/s 53%, at a speed of 3 m/s 35%, at a speed of 4 m/s 30% and at a speed of 5 m/s 27 % better capture than the original hydrocyclone.

In Figure 4-shows the aerodynamic resistance of hydrocyclones.

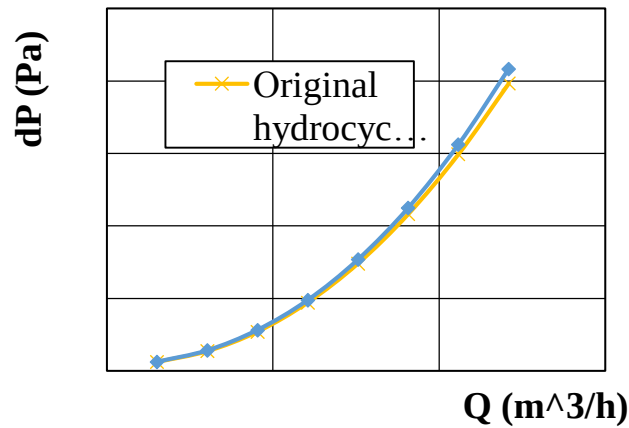


Figure 4- Resistance of hydrocyclones.

From Figure 4 - shows that the resistance of the developed centrifugal hydrocyclone is 3% greater than the original hydrocyclone. This does not affect the efficiency of the developed centrifugal hydrocyclone.

The main factors influencing the efficiency of a hydrocyclone are the gas flow rate in the inlet pipe, the size and density of the cylinders. Therefore, comparisons were made in the work at inlet speeds

In Figure 5-shows contour graphs of the average axial velocity and pressure of hydrocyclones in the middle plane.

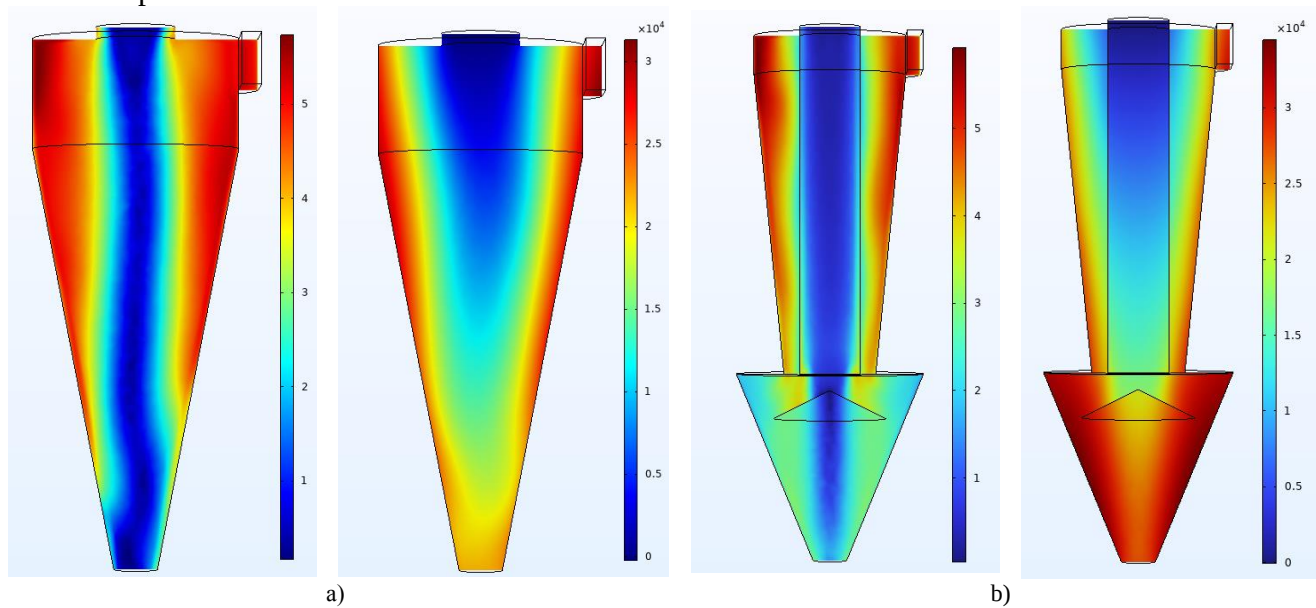


Figure 5- Isolines of the velocity and pressure fields in hydrocyclones a) original hydrocyclone, b) recommended hydrocyclone.

In Figure 6- shows the current line.



Figure 6- Flow line a) original hydrocyclone, b) recommended hydrocyclone.

5. Conclusion.

The work used the Comsol Multiphysics 6.1 software package to simulate the hydrodynamic processes of a two-phase flow inside hydrocyclones. In this case, the v^2-f turbulence model was used. The results obtained confirm that the new hydrocyclone is more efficient than the original hydrocyclone. Thus, it can be argued that modeling hydrodynamic processes using modern software packages is an effective tool for the design of various hydrocyclones and separators.

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NUMERICAL SIMULATION OF HYDROTRANSPORT OF DISPERSED PARTICLES IN A PIPELINE

Mirzoev A.A.¹, Khodjaev Ya D.², Ibrokhimov A.R.³, Matkarimov S.Y.⁴

¹*Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan*

²*Republican air-geodesic Centre, Tashkent,, Uzbekistan.*

³*Fergana Polytechnic Institute, Fergana, Uzbekistan,*

⁴*Tashkent university of architecture and civil engineering, Tashkent, Uzbekistan*

E-mail: akmal_mirzo@mail.ru

Abstract. In this paper, a turbulent flow with a ridge is numerically modeled using a standard *k-e* model based on the RANS approach. The Reynolds number was calculated to be 21400. Flow calculation grid, flow lines, flow isoline were made using Comsol Multiphysics software. Dispersed particles were considered as two phases. Gravity and Archimedes forces acting on particles were taken into account. The Lagrangian approach was used to determine the trajectory of the particles. The movement of particles with a diameter of 0.01-0.1 mm was calculated numerically. The trajectories of the particle were constructed when the initial speed of 0.35 m/s, 0.55 m/s, 0.75 m/s, 1 m/s was given for each particle.

Keywords: Multiphase flows, RANS, Standart *k-e* model, dispersed phase, Lagrangian approach, Comsol Multiphysics, numerical simulation.

1. Introduction

The flow of dispersed systems is often found in the environment and numerous industrial processes. For example, gas-liquid transportation, distribution of pollutants, spraying dryers for the food industry, proppant transportation in fractures, and these are just a few of them. A large group of industrial materials values are emulsions and suspensions [1]. In suspensions, the dispersed phase consists of small insoluble solid particles, whereas in emulsions the dispersed phase consists of small immiscible drops of liquid. Because of interactions between different phases are difficult for such flows theoretical description. To design and optimize such two-phase systems requires a detailed understanding of interfacial transport phenomena.

A current example of suspension flow is transportation proppant in a hydraulic fracturing fracture, which is carried out on hard-to-recover oil fields. Hydraulic fracturing (HF) is today one of the key technologies for increasing oil recovery. Injection of a suspension consisting of a low-viscosity liquid and proppant, with high flow rate under high pressure leads to the formation of a network complex cracks. The placement of proppant in these fractures is closely related to its gravitational settling, interaction

between particles, leakage liquids into the reservoir. The disperse system is a multiphase heterogeneous system consisting of many small particles (dispersed phase), distributed in continuous liquid, gaseous or solid dispersion media, such as water droplets in the air or gas bubbles in a liquid, as well as solid particles carried by liquid. In emulsions, the dispersed phase consists of small immiscible drops of liquid [2]. In most cases dispersed droplets are stabilized from coalescence due to the presence surfactant at the interface [3]. Subsidence and stratification, in which particles and drops fall or rise in a liquid, in which they are suspended under the influence of gravity, is common in suspensions and emulsions [4].

Behavior of dispersed systems involving the movement of small particles relative to the surrounding fluid, covers a wide range of phenomena, of interest to researchers [5]. Particles can move together in the mass through the liquid during sedimentation or may remain motionless. The mechanics of multiphase media deals with the study of the dynamics of dispersed systems [6]. Multiphase flows occur in various forms in manufacturing practice and technology depending on the corresponding phase distribution [7]. And depending on the phases involved, they are divided into transitional, detachable and dispersed two-phase flows, in which one phase is present in the form of particles, drops or bubbles dispersed in a continuous carrier phase have the importance of interfacial transport mechanisms [8].

Current state of research in the field of particle, droplet and bubbles in non-Newtonian liquids is given in monograph [9] and work [10]. Research in a relatively new field of physical chemistry – physical chemical dynamics of disperse systems are presented in [13]. Because of them technological significance, the rheological behavior of suspensions has long been is the subject of research, reviews of which in the literature can be see works [11, 14].

The flow of dispersed systems is determined by the so-called creeping motion or Stokes equations. Dimensionless criterion that determines the relative importance of inertial and viscous effects is the number Reynolds [12].

Simulation of flows of two-phase systems and development of numerical tools are one of the important tasks of modern computing hydrodynamics. Various modeling approaches and numerical methods have been developed and applied in this field over the past decades. This process leads to the conclusion that the applicability of a particular approach to simulation is highly dependent on the type of two-phase configuration used flow.

One of the widely used methods of numerical modeling dispersed systems is the solution of the equations of conservation of mass, momentum and energy along with the governing phase equations and conditions at the boundary between continuous and dispersed phases. This approach offers a completely solved simulation of any two-phase dispersed flow problem. The main task Such a direct method is the resolution of phase boundaries. Numerical methods associated with this problem are extremely computationally expensive. Relation, so they can only be applied to problems in which a relatively small number of dispersed objects are involved. For this reason, direct methods are mainly used in fundamental research. In industrially relevant complex simulations it is common to very large numbers of particles or bubbles are involved and these methods not applicable. The most widely used methods in this field are methods for control volume of liquid [15] methods for setting level [16] and edge tracking methods [17].

2. Most used mathematical models

In addition to direct methods, which by their nature are the most accurate, there are simpler and therefore more applicable models:

Euler-Euler models,

Euler-Lagrange models.

Euler-Euler models.

A widely used method is to create transfer laws on based on the volume fractions of two phases in each computational control volume, resulting in a continuous representation of both phases. This approach is called two-fluid model or also called Euler-Euler model. In progress [18] one of the first to mention such a concept as interpenetrating continuums.

The key idea behind building a two-phase model is that multiphase flow is represented as a set of interacting continuums. Behavior of each component phase of a multiphase model described by the laws of conservation of mass, momentum and energy. Phase interaction relative to each other is taken into account by adding special (algebraic or differential) terms.

Euler-Lagrange models.

The Lagrangian approach, also known as the "dilute phase approach», used when the amount of dispersed phase is small and does not interfere with movement continuous phase [19]. In the Lagrangian approach, three main ones can be distinguished modeling method:

- Direct Numerical Simulation (DNS),
- Large Eddy Simulation (LES),
- Reynolds averaging by the Navier-Stokes method (RANS) [2]

DNS modeling approach requires particles to be smaller on the scale of Kolmogorov. This requirement greatly limits the use of DNS low Reynolds numbers or very small particles. To overcome this is a limitation, LES modeling can be used. Application as DNS, Likewise, LES is limited to dilute systems, where collisions and hydrodynamic interactions can be neglected and are assume done-way connection between the dispersed and carrier phases [20]. In recent studies [3] one-way coupled Eulerian-Lagrangian models have been used to study dilute solid-liquid flows weigh it. Reviews of Lagrangian-Eulerian methods for multiphase flows have been given in works [19, 20]

This can only be achieved with a very fine mesh, which therefore leads to unrealistically high computational time and memory costs. Therefore, a subgrid drift velocity model was developed to allows coarser meshes [10, 12]. All phenomena on a scale smaller than numerical grid, require first of all modeling of flow turbulence fluids using RANS or LES methods. However, such models need extensions to account for two-way communication.

COMSOL Multiphysics is a powerful simulation software package that allows you to model and simulate a variety of physical phenomena in a variety of disciplines, including fluid dynamics, heat transfer, structural mechanics, electromagnetism, and chemical reactions. COMSOL Multiphysics uses the finite element method to solve fluid dynamics equations.

In this article, a mathematical model of the movement of dispersed particles in a flow is created and numerical results are obtained. For this purpose, the speed of water movement was determined using the Navier-Stokes equation [7]. The trajectories of dispersed particles were determined using the Lagrangian approach.

3. Physical and mathematical formulation of the problem

In Figure 1 shows a schematic representation of the flow of dispersed particles

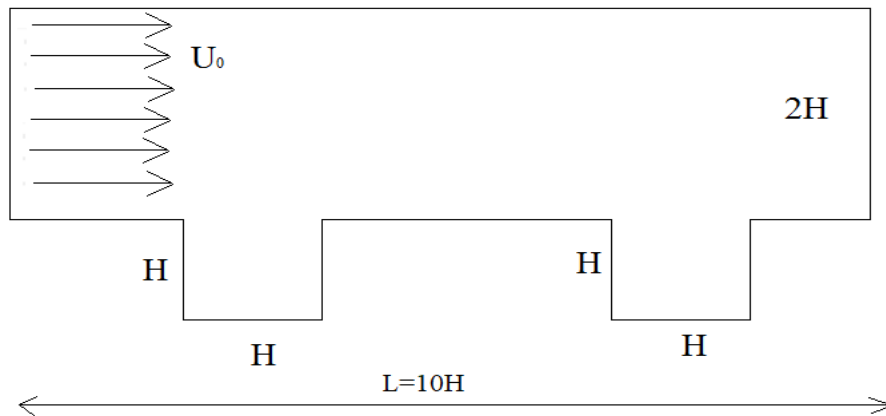


Figure 1. - Schematic view of the flow

For the numerical study of the problem posed, the system of Navier-Stokes equations averaged over Reynolds is used [12, 13]. The system of equations does not take into account the forces due to the effects of turbulent migration, Sefman, Magnus (lift) and Coriolis. Because they are significantly less than the centrifugal force.

Thus, for mathematical modeling of the processes of transfer of dust particles, it is sufficient to take into account the centrifugal, gravitational $\bar{G}$, Archimandial $\bar{F}_A$ and Stokes force $\bar{F}$.

$$\begin{cases} \frac{\partial \bar{U}_i}{\partial t} + \bar{U}_j \frac{\partial \bar{U}_i}{\partial x_j} = -\frac{1}{\rho} \frac{\partial p}{\partial x_i} + \bar{G} + \frac{\partial}{\partial x_j} \left[(\nu + \nu_t) \left(\frac{\partial \bar{U}_i}{\partial x_j} + \frac{\partial \bar{U}_j}{\partial x_i} \right) \right], \\ \frac{\partial \bar{U}_{pi}}{\partial t} + \bar{U}_j \frac{\partial \bar{U}_{pi}}{\partial x_j} = \sum_{k=1}^n F_k = \bar{G} + \bar{F}_A + \bar{F}, \\ \frac{\partial \rho_m}{\partial t} + \bar{U}_{pi} \frac{\partial \rho_m}{\partial x_j} = D \frac{\partial}{\partial x_j} \left[\frac{\partial \rho_m}{\partial x_j} + \frac{\partial \rho_m}{\partial x_i} \right], \quad \frac{\partial \bar{U}_j}{\partial x_j} = 0. \end{cases} \quad (1)$$

Here $\bar{U}_i$ – longitudinal and transverse of the fluid flow velocity; $\bar{U}_{pi}$ – similar velocity components for m -th particle fractions; $\bar{p}$ – hydrostatic pressure; ρ – density of fluid; ν, ν_t – molecular and turbulent viscosities; ρ_m – particle mass density; m – number of particle fractions; $D = \frac{\rho}{Sc(\rho + \rho_p)}(\nu + \nu_t)$ – diffusion

coefficient for the solid phase, $Sc = 0.8$ – Schmidt number.

The force of gravity is defined as follows.

$$\bar{G} = m_p g = \frac{1}{6} \pi d_e^3 \rho_p g \quad (2)$$

where m_p – particle mass, g – acceleration of gravity, d_e – equivalent particle diameter, ρ_p – particle density.

$$\bar{F}_A = V_p \rho g = \frac{1}{6} \pi d_e^3 \rho g. \quad (3)$$

Hydrodynamic force $\bar{F}$ was determined through the Stokes parameter for turbulent flow:

$$\bar{F} = \frac{1}{2} C_D S \rho (\bar{U} - \bar{U}_{pi}) |\bar{U} - \bar{U}_{pi}| \quad (4)$$

where C_D – drag coefficient determined through the Reynolds criterion, those. $C_D = C_D(Re)$; S – particle midsection area, $S = \frac{\pi d_e^2}{4}$

If the particle lags behind the flow ($\bar{U} < \bar{U}_{pi}$), the aerodynamic force will accelerate the particle and help equalize the velocities $\bar{U}$ and $\bar{U}_{pi}$. Otherwise – force $\bar{F}$ slows down the particle
In [14], the k- ϵ turbulence model was described in detail. The turbulence model k- ϵ in tensor form is as follows:

$$\begin{cases} \frac{\partial \rho k}{\partial t} + \bar{U}_j \frac{\partial \rho k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \varepsilon \rho, \\ \frac{\partial \rho \varepsilon}{\partial t} + \bar{U}_j \frac{\partial \rho \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \frac{\varepsilon}{k} P_k - C_2 \rho \frac{\varepsilon^2}{k} \end{cases} \quad (5)$$

where P_k — energy generation of turbulent pulsations, the turbulent viscosity is calculated as

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon}, \quad \nu_t = C_\mu \frac{k^2}{\varepsilon} \quad (6)$$

According to the modeling of the kinematics of particle motion in a turbulent two-phase flow, there is no single representation that would allow one to correctly describe an object [15]. The model based on the concept of "trajectory particles" is considered incorrect due to the lack of consideration of the factor of interaction between Reynolds stresses and particles. On the other hand, the advantages of the Lagrangian approach, which is closer to real processes and makes it possible to obtain the necessary information about the particle trajectories, the residence time of the particles in the apparatus, and the minimum size of particles to be captured, are indisputable [16–19]. In this regard, in this work, the Lagrangian approach was used to model the efficiency of particle.

The numerical solution of the presented systems of equations was carried out in the physical variables velocity–pressure by physically splitting the velocity and pressure fields [20]. The numerical solution of the transport equation is carried out on a hybrid, checkerboard difference grid by the control volume method. When solving systems of equations (1-4) numerically, the same method was used.

4. Solution method and boundary conditions

COMSOL Multiphysics offers a range of solvers to solve different types of physics problems. The choice of solver depends on the type of physics being modeled, the complexity of the problem, the desired accuracy, and the available computing resources. To solve the equations of the standart k-ε model, a fully coupled approach (Fully Coupled) using a direct solver algorithm (PARDISO) was used. Newton's iterative method of 0.1 was used. The iteration process for the problem of flow past a flat plate with zero pressure gradient continued up to 350 iterations, and for the remaining problems the iterative process continued up to 450 iterations. The tolerance factor is 1, the residual factor is 1500.

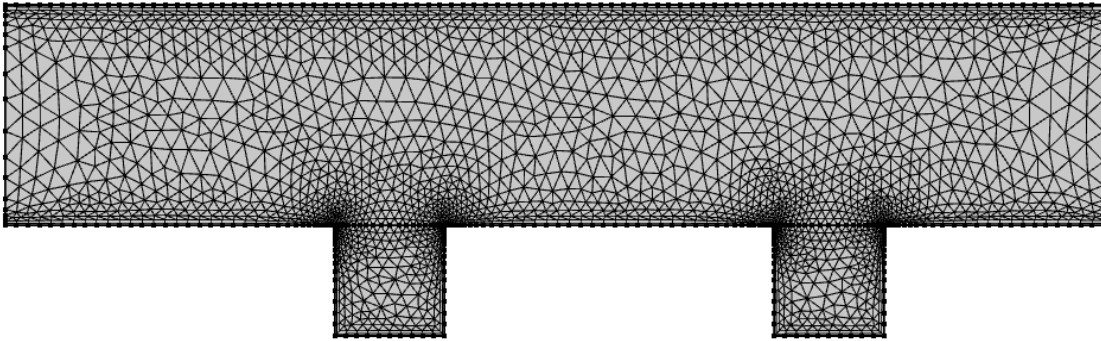


Figure 2. - **Mesh of flow**

In Figure 2. the computational mesh is built in the Comsol multiphysics program. A calculation grid consisting of 1812 points was built.

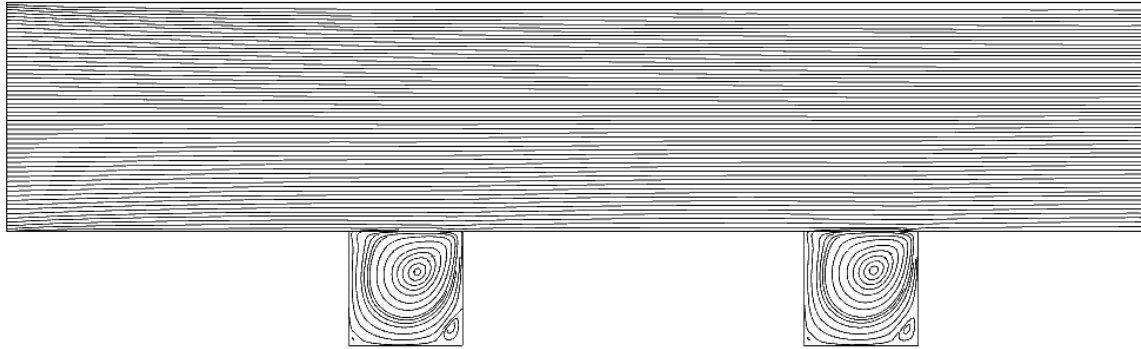


Figure 3. - Streamline

In Figure 3 shows streamlines. we can see where the vortices appear in the flow and their length.

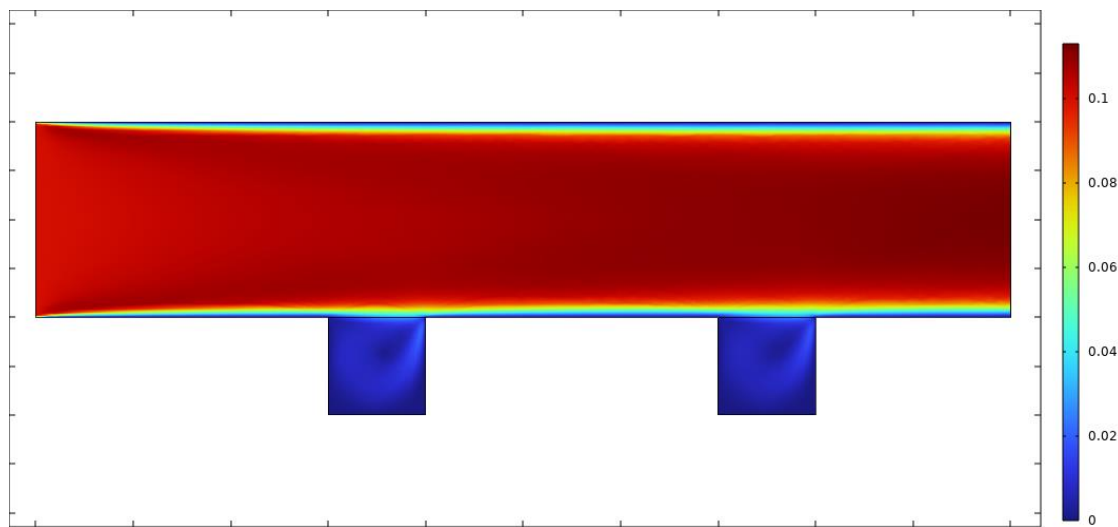
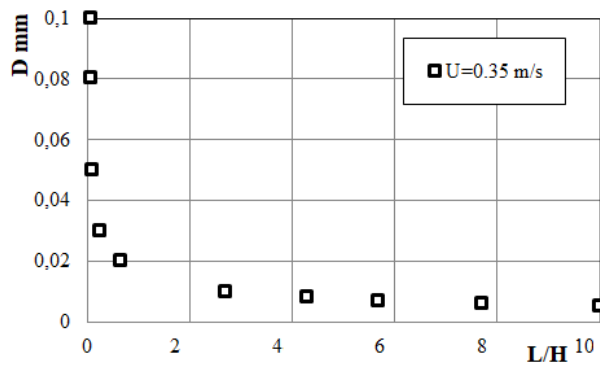
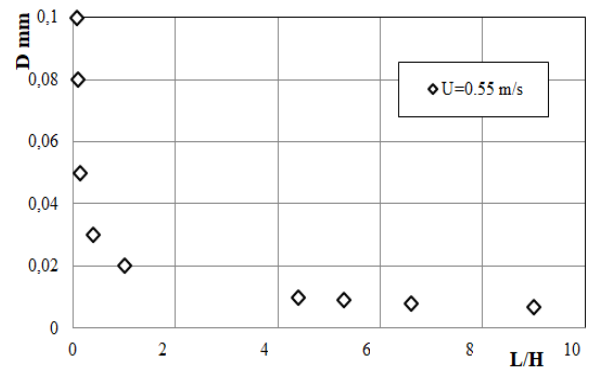


Figure 4. - Flow contour

Isolines of longitudinal velocity of fluid flow are lines connecting points with the same value of the corresponding parameters. Longitudinal velocity contours are usually depicted as lines connecting points of equal velocity along a profile. They show how the flow speed changes along the profile and which areas have higher or lower speeds.



a)



b)

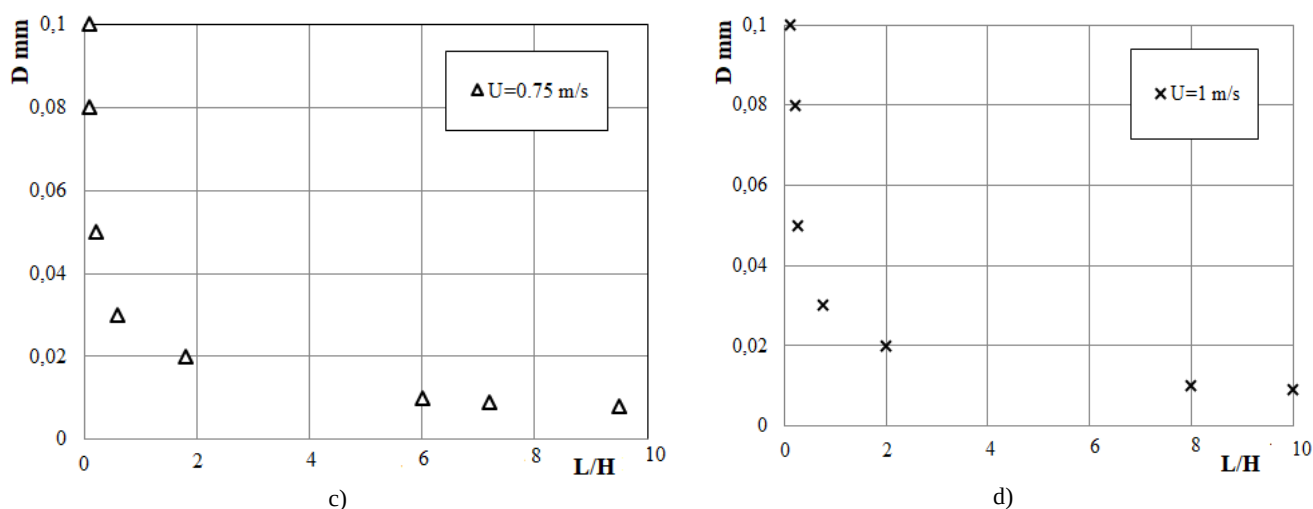


Figure 5.- Particle settling distance as a function of flow velocity

In Figure 5 shows particles of different diameters at settling distances. The forces acting on the particle and the velocities in the flow are 0.35 m/s, 0.55 m/s, 0.75 m/s, 1 m/s we can see the particles settling.

5. Conclusion

In this article, the movement of solid particles in open channels was studied. The RANS method and the standard k- ϵ turbulence model were used to determine the flow velocities. A Lagrangian approach was used to determine the trajectory of the particles. In numerical calculations, it was determined that if the size and size of the particle is large, it will sink very quickly, if the particle is light, it will travel a long distance, and it is shown in the graphs.

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SIMULATION OF SWIRLING JET FLOWS BASED ON TWO FLUID TURBULENCE MODEL

Navruzov D.P.¹, Madaliev M.E.², Fayziyev A.A.¹, Hamroyev N.Sh.¹

¹Bukhara Institute of Natural Resources Management, National Research University "TIAME", Bukhara, Uzbekistan

²Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan

E-mail: navruzov.d@mail.ru,

Abstract. *This article presents the results of a two-fluid model and a turbulence model of the RSM (Reynolds stress model) family for a swirling jet. The two-fluid model used in this study has been developed recently. Pioneering works show that the basis for constructing this model is the possibility of representing a turbulent flow in the form of a non-homogeneous mixture of two fluids. Therefore, this model is derived on the basis of the dynamics of two fluids. These studies also show that the developed two-fluid model is able to adequately describe complex anisotropic turbulences. The fundamental difference of this model is that the two-fluid model is presented by the equations of dynamics. The two-fluid model reduces computational costs and makes the methodology practical from an "engineering" point of view for modern computing resources. In the numerical implementation in convective terms, the control volume method was used, and the correction for the velocity was conducted according to the SIMPLEC method. The numerical results obtained are compared with experimental data from the ERCOFTAC database. It is shown that the results of the two-fluid model are consistent with the experimental data.*

Key words: *swirling jet, two-fluid model, upwind scheme, swirl, QUICK scheme.*

1. Introduction.

Swirling flows, in connection with their numerous practical applications, represent an important branch of modern hydrodynamics. The complexity of these flows stimulates the use of the most modern research methods in their study. The methods of numerical simulation and various fine measurement methods, including non-contact ones, which are no longer the property of individual research laboratories and are becoming more widespread, allow us to hope for significant progress in studying and predicting the properties and structure of complex hydrodynamic processes, including swirling flows [1, 2]. Swirling flows are observed in many natural phenomena and are often used in technology. As an example (in the case without combustion), one can point to cyclones and tornadoes in the atmosphere, to whirlpools in rivers and reservoirs, to the vortex trail left by the wing of an aircraft, to cyclone-type separators and heat exchangers, to sprinkler and fertilizer-sprinkling agricultural machines, to jet pumps and vortex flow chemical reactors. Elements of the theory of swirling flows are used even when analyzing the flight of a boomerang and a bee. In technical devices with combustion, the significant and beneficial effect of the swirling of the supplied air and fuel flows on the stabilization and intensification of the combustion process is intensively used to ensure efficient and environmentally friendly combustion of fuel in a wide variety of practical areas: in gasoline and diesel engines, in gas turbines and gas turbine engines, in industrial furnaces, boilers, as well as in a number of other heating systems. As experimental studies show, swirling has a radical effect on the flow field - its important properties and characteristics, such as the propagation and attenuation of jets and the ejection

of matter by a jet in inert flows, the size and shape of the flame, and the intensity of combustion in reacting flows, significantly depend on the degree of swirl, imparted to the flow [3, 4].

Experiments (being quite expensive) are of great interest to designers and a preliminary calculation of the flow field using mathematical models and numerical methods would greatly contribute to cheaper development and lower operating costs. Combining experimental and theoretical data on the aerodynamics of combustion with detailed numerical gas-dynamic calculations and applying and improving appropriate programs, it is possible to significantly reduce the duration and cost of developments.

The effect of the initial swirl of the flow on the flowfield increases sharply with an increase in the degree of swirl (expressed by the swirl parameter S or the blade angle φ). The effect of a weak swirl ($S < 0.4$) is reduced to an increase in the width of a free or confined jet: an increase in the width of the jet, flow mixing, and a decrease in the jet velocity occur more intensely with increasing the degree of swirl. Although there may be significant pressure gradients in the transverse (or radial) direction in any section of the jet, the non-zero gradient is a consequence of the effect of swirling on the pressure distribution [5-8]. For applied problems with combustion, one of the most significant and useful phenomena in swirling jet flows is the occurrence of a recirculation zone formed in the central section at supercritical values of the swirl parameter ($S=0.6$ for swirling devices with direct outlet). Numerical studies of swirling flows with standard models of turbulence with two k - ε equations showed that these models could significantly underestimate mixing rates. Modifications to two-equation models to improve mixing predictions in swirling flows have varied with the change in the swirl viscosity [9-12]. The direct numerical simulation (DNS) method and large eddy simulation (LES) method are time-consuming ones and their use to solve complex aerodynamic problems requires high-speed computers. Therefore, their widespread practical application is associated with the development of computer technology and, according to experts, can begin only at the end of this century. Therefore, in the near future, semi-empirical methods will remain the main working tool for solving applied problems of aerodynamics. A new mathematical model has been tested for a number of turbulence problems [13-15]. The results obtained were compared with experimental data presented in the ERCOFTAC database [16].

2. Mathematical model.

RSM Turbulence Model. To simulate the problem posed, the system of Reynolds averaged Navier-Stokes equations in a cylindrical coordinate frame was used [14].

$$\begin{cases} \frac{\partial U}{\partial z} + \frac{\partial V}{r \partial r} = 0, \\ \frac{dU}{dt} + \frac{1}{\rho} \frac{\partial p}{\partial z} = \frac{1}{\text{Re}} \left(\frac{\partial^2 U}{\partial r^2} + \frac{\partial U}{r \partial r} + \frac{\partial^2 U}{\partial z^2} \right) + \frac{\partial}{\partial r} (-rv'u') + \frac{\partial}{\partial z} (-u'u'), \\ \frac{dV}{dt} + \frac{1}{\rho} \frac{\partial p}{\partial r} = \frac{1}{\text{Re}} \left(\frac{\partial^2 V}{\partial r^2} + \frac{\partial V}{r \partial r} + \frac{\partial^2 V}{\partial z^2} - \frac{V}{r^2} \right) + \frac{\partial}{\partial r} (-rv'v') + \frac{\partial}{\partial z} (-u'v') - \frac{1}{r} (-w'v'), \\ \frac{dW}{dt} = \frac{1}{\text{Re}} \left(\frac{\partial^2 W}{\partial r^2} + \frac{\partial W}{r \partial r} + \frac{\partial^2 W}{\partial z^2} - \frac{W}{r^2} \right) + \frac{\partial}{\partial r} (-v'w') + \frac{\partial}{\partial z} (-w'u') + \frac{2}{r} (-v'w'). \end{cases} \quad (1)$$

here U, V, W are the axial, radial, and tangential components of the vector of air flow velocity, respectively; p is the hydrostatic pressure; ρ is the gas density; V is the gas molecular viscosity; $\rho u'v', \rho u'u', \rho u'w', \rho v'w', \rho v'v', \rho w'w'$ are the components of the Reynolds stress tensor.

The Reynolds stress model SSG/LRR-RSM is described in [17, 18], so here we will only give equations for stresses and the scale length:

$$\begin{aligned} \frac{\partial R_{ij}}{\partial t} + \frac{\partial U_k R_{ij}}{\partial x_k} &= P_{ij} + \Pi_{ij} - \varepsilon_{ij} + D_{ij}, \\ \frac{\partial \omega}{\partial t} + \frac{\partial U_k \omega}{\partial x_k} &= \frac{a_\omega \omega}{k} \frac{P_{kk}}{2} - B_\omega \omega^2 + \frac{\partial}{\partial x_k} \left(\left(\mu + \sigma_\omega \frac{k}{\omega} \right) \frac{\partial \omega}{\partial x_k} \right) + \sigma_d \frac{1}{\omega} \frac{\partial k}{\partial x_j} \frac{\partial \omega}{\partial x_j}. \end{aligned} \quad (2)$$

here $\rho R_{ij} = -\tau_{ij} = \rho u_i' u_j'$, P_{ij} is the generation of Reynolds stresses, D_{ij} is the diffusion.

Two-fluid turbulence model.

The two-fluid model is described in detail in [19]. A summary of the essence of the two-fluid model is also presented in [21]. In [20], this approach was successfully used to simulate heat transfer. In these publications the possibility of representing a turbulent flow as a non-homogeneous mixture of two fluids with relative motion was shown [21, 22]. The system of equations for an incompressible turbulent jet in the new two-fluid model has the following

$$\begin{cases} \frac{\partial \bar{V}_i}{\partial t} + \bar{V}_j \frac{\partial \bar{V}_i}{\partial x_j} + \frac{\partial \bar{p}}{\rho \partial x_i} = \frac{\partial}{\partial x_j} \left[\nu \left(\frac{\partial \bar{V}_i}{\partial x_j} + \frac{\partial \bar{V}_j}{\partial x_i} \right) - v_j v_i \right], \\ \frac{\partial v_i}{\partial t} + \bar{V}_j \frac{\partial v_i}{\partial x_j} = -\rho v_j \frac{\partial \bar{V}_i}{\partial x_j} + \frac{\partial}{\partial x_j} \left[\nu_{ji} \left(\frac{\partial \bar{V}_i}{\partial x_j} + \frac{\partial \bar{V}_j}{\partial x_i} \right) \right] + \frac{F_{si}}{\rho} + \frac{F_{fi}}{\rho}, \\ \nu_{ji} = 3\nu + 2 \left| \frac{v_i v_j}{\text{def}(\bar{V})} \right| \quad \text{for } i \neq j, \quad \nu_{ii} = 3\nu + \frac{1}{\text{div} \bar{V}} \left| \frac{v_k v_k}{\text{def}(\bar{V})} \right| \frac{\partial v_k}{\partial x_k}, \\ \bar{F}_f = -\rho K_f \bar{V}, \quad \bar{F}_s = \rho C_s \text{rot} \bar{V} \times \bar{V}, \quad \frac{\partial \rho \bar{V}_j}{\partial x_j} = 0, \quad K_f = C_1 \lambda_{\max} + C_2 \frac{|\bar{d} \cdot \bar{V}|}{d^2}. \end{cases} \quad (3)$$

In the above equations, $\bar{V}_i$ is the vector of the averaged flow velocity, v_i are the axial and radial relative velocities, v_i – axial and radial velocities, respectively, ν_{ji} are the effective molar viscosities, K_f is the friction coefficient, d is the nearest distance to the solid wall, $\lambda_{\max}$ is the largest root of the characteristic equation $\det(A - \lambda E) = 0$, where matrix A has the following form:

$$A = \begin{vmatrix} -\frac{\partial \bar{V}_1}{\partial x_1} & -\frac{\partial \bar{V}_1}{\partial x_2} - C_s \zeta_3 & -\frac{\partial \bar{V}_1}{\partial x_3} + C_s \zeta_2 \\ -\frac{\partial \bar{V}_2}{\partial x_1} + C_s \zeta_3 & -\frac{\partial \bar{V}_2}{\partial x_2} & -\frac{\partial \bar{V}_2}{\partial x_3} - C_s \zeta_1 \\ -\frac{\partial \bar{V}_3}{\partial x_1} - C_s \zeta_2 & -\frac{\partial \bar{V}_3}{\partial x_2} + C_s \zeta_1 & -\frac{\partial \bar{V}_3}{\partial x_3} \end{vmatrix} \quad (4)$$

and $\vec{\zeta} = \text{rot} \bar{V}$.

We study the problem in cylindrical coordinates.

$$Re = \frac{DU_{in}}{\nu} = 20000, \quad S = \frac{\int_0^R V_{0\varphi} V_{0z} r^2 dr}{R \int_0^R V_z^2 r dr}$$

$$v_\varphi = 0.2U_{in}, \quad v_z = 0.2U_{in}, \quad v_r = 0.05U_{in}$$

Here Re is the Reynolds number, S is the parameter that determines the degree of swirling of the incoming flow, U_{in} is the maximum axial velocity at the inlet. A grid of 300x400 was used for calculations. Equations in a dimensionless form were given by correlating all velocities to U_{in} , and spatial scales - by correlating to the pipe diameter. Time integration was conducted with dimensionless step $\Delta t = 0.02$.

To ensure stability in solving the equations of system (5) in convective terms, an upstream differencing scheme of second-order accuracy was used according to the finite volume method (FVM). As for diffusion terms, the implicit central difference was used. The explicit Leonardo QUICK scheme [23] was used for the convective terms, and the viscous terms were approximated by the central difference. The viscosity terms in the longitudinal direction were approximated in an explicit form, and in the transverse direction - in an implicit form. The Leonardo QUICK scheme has second-order accuracy. Consequently, the calculation scheme had the accuracy of the first order in time and the second order in space. The sweep method was used to implement the semi-implicit scheme. The connection between pressure and velocities was realized by the SIMPLEC procedure [24, 25], where the iterative method of over-relaxation was used [26-32].

3. Numerical results.

Figure- 1 shows the results of a two-fluid and RSM model. It also presents the experimental data given in [16] for dimensionless axial velocity profiles at swirl $S=0.23$.

Figures- 2 a), b), and c) show the results of the radial axial velocity for various sections $x/D=0$, $x/D=1$, $x/D=2$, $x/D=3$, $x/D=4$, and $x/D=5$ at swirl $S=0.23$. The solid line represents the results obtained using the two-fluid and RSM models, and the dotted lines represent experimental data.

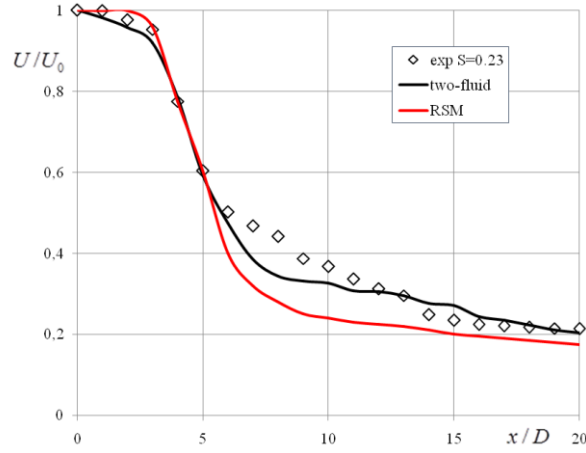


Figure 1. - Axial flow rate

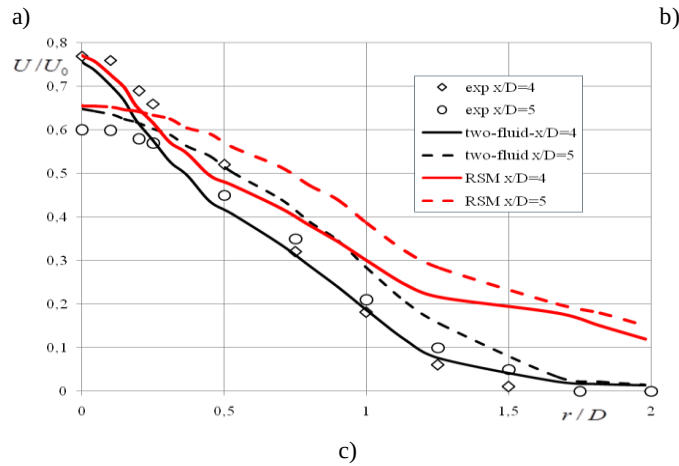
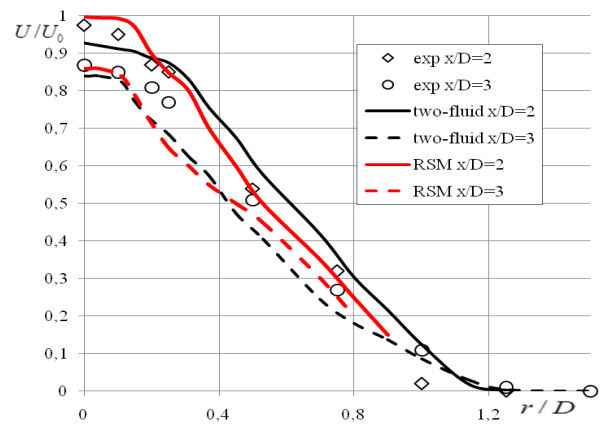
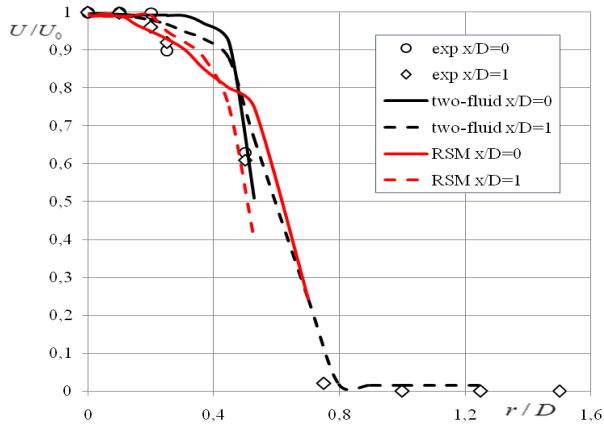


Figure 2. - Comparison of longitudinal flow velocity

Figures- 3 a), b) and c) show the results of the radial axial swirl velocity for various sections $x/D=0$, $x/D=1$, $x/D=2$, $x/D=3$, $x/D=4$, and $x/D=5$. Solid lines represents the results obtained using the two-fluid model and dotted lines show the experimental data. The results were obtained at swirl $S=0.23$.

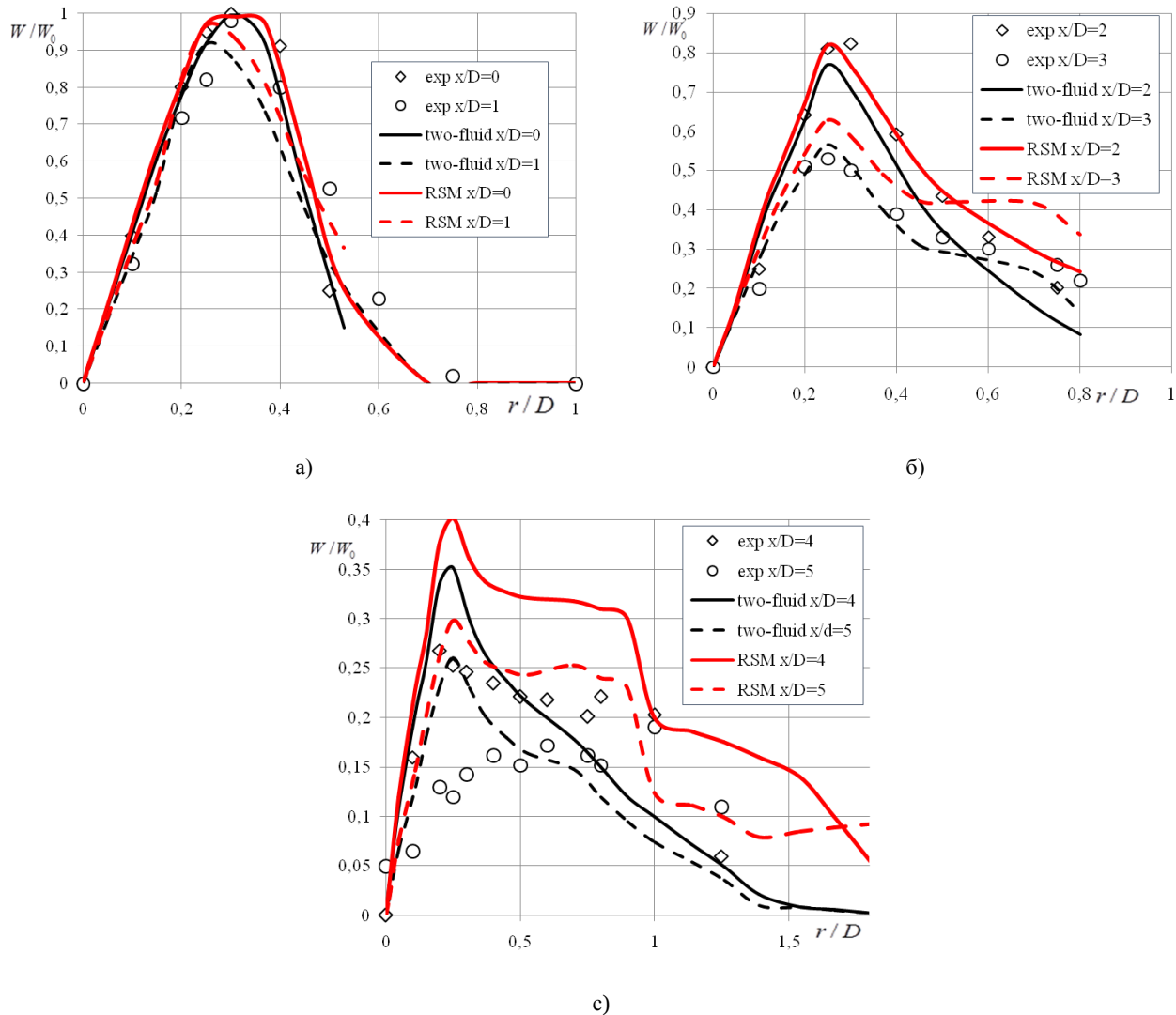


Figure 3. - Radial flow rate comparison

The results obtained regarding the verification of the effectiveness of the two-fluid model in relation to the simulation of turbulent eddy flows can be summarized as follows. For the case of weak swirl, $S = 0.23$, the axial velocity and its profile at different turbulence sections predicted by the two-fluid model are in good agreement with the experimental data.

4. Conclusion.

Comparative testing of a two-fluid turbulence model with the results of experiments for swirling jet problems was performed in the article. The study showed that the model describes well the process with the swirl. The two-fluid model has the advantage that it is able to describe complex anisotropic flows and is easy for numerical implementation. Therefore, the new two-fluid model can be successfully used in other, more complex problems with swirling flows.

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LAMINAR UNSTATIONARY FLOW OF A VISCOELASTIC FLUID IN A FLAT CHANNEL

Navruzov Q.¹, Mirzoev A.A.², Fayziev R. A.³, Akhmedov N.O.⁴

¹ Urgench State University, Tashkent, Uzbekistan

²Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan

³Tashkent State University of Economics, Tashkent, Uzbekistan

⁴Tashkent State Transport University, Tashkent, Uzbekistan

E-mail: akmal_mirzo@mail.ru

Abstract. The problems of unsteady flow of a viscoelastic fluid in a flat channel under the influence of a constant pressure gradient are solved based on the generalized Maxwell model. By solving the problem, formulas for velocity distribution, fluid flow and other hydrodynamic quantities were determined. Based on the formulas found, transient processes during unsteady flow of a viscoelastic fluid in a flat channel are analyzed. Based on the results of the analysis, it was shown that the processes of transition of the characteristics of a viscoelastic fluid from an unsteady state to a stationary state at small values of the Deborah number practically do not differ from the transition processes of a Newtonian fluid. Exceeding the Deborah number relatively unity, it has been established that the process of transition of a viscoelastic fluid from an unsteady state to a stationary state is of a wave nature, in contrast to the transition process of a Newtonian fluid, and the transition time is several times longer than that of a Newtonian fluid. It was also discovered that perturbed processes can arise during the transition. These disturbances occurring in unsteady flows of a viscoelastic fluid can be stabilized by stirring the Newtonian fluid. The implementation of this property is important in technical and technological processes, in preventing technical failures or malfunctions.

Keywords: viscoelastic fluid, unsteady flow, longitudinal velocity, fluid flow, steady flow.

1. Introduction

The study of joint unsteady flow of viscous and viscoelastic fluid in a flat channel under the influence of a constant pressure gradient is widely used in technical processes. As is known, the viscoelastic properties of non-Newtonian fluids significantly affect the hydrodynamic characteristics of the flow. For example, when transporting highly viscous and heavy oil and petroleum products over long distances, one of the important tasks is to develop an effective method for reducing the hydrodynamic resistance of flows [1,2]. Especially, during the starting and stopping mode of the working bodies (mechanisms), an unsteady flow is observed. In such cases, the change in fluid flow rate and other hydrodynamic characteristics of the flow differ significantly from the characteristics of conventional Newtonian fluid flows. However, such flows, in addition to the above-mentioned industries, are often used in various technological processes, in chemical technology, in biological mechanics and in acoustics [3,4].

Many studies [5-12] are devoted to the study of unsteady flows of Newtonian and non-Newtonian, in particular viscoelastic fluids in pipes and channels. For the first time, the unsteady flow of a viscous incompressible fluid in a cylindrical pipe was studied in the works of I.S. Gromek [5,6]. In them, he determined the change in speed, fluid flow and shear stress on the wall. Using these formulas, it is possible to determine the time of establishment of hydrodynamic quantities during the flow of a viscous fluid in a cylindrical pipe. Then this problem was given in many textbooks as the problem of I.S. Gromek [7-12].

In [13], unsteady viscoelastic flows of an Oldroyd-B fluid through an infinite pipe of circular cross-section were analyzed. A liquid moves under the influence of a time-dependent pressure gradient in the following three cases: 1) the pressure gradient changes with time in accordance with the exponential law; 2) the pressure gradient pulsates; 3) the pressure gradient is constant. Formulas are obtained for the distribution of fluid velocity, fluid flow rate and other hydrodynamic characteristics of the flow.

Unsteady pulsating flows of a viscous fluid in a round cylindrical pipe of infinite length under the influence of a harmonic changing pressure gradient were studied in [8]. By solving the problem, calculation formulas for the distribution of fluid velocity and flow rate were obtained. Numerical calculations have shown that in a pulsating flow at lower values of the dimensionless oscillation frequency, the speed, flow rate and other hydrodynamic parameters from the zero initial state are established slowly, comparatively at high oscillation frequencies and are close to the parameters of a

non-pulsating flow. In an oscillating flow at high oscillation frequencies, these parameters are established almost instantly.

Based on the Maxwell model, the problem of unsteady oscillatory flow of a viscoelastic fluid in a flat channel is considered [14,15]. Formulas for determining dynamic and frequency characteristics are obtained. Using numerical experiments, the influence of vibration frequency and relaxation properties of the liquid on the shear stress on the wall was studied. It is shown that the viscoelastic properties of the fluid, as well as its acceleration, are the limiting factors for the use of the quasi-stationary approach. In [16,17], the problems of unsteady flow of viscoelastic fluid in long pipes of annular cross-section were solved. Constant rheological quantities of a viscoelastic fluid were considered to be independent of the rate of deformation; with this condition, the problem is reduced to a solution by a linear differential equation and its analytical solution is obtained using the Laplace transform. The calculation results were carried out for a fluid with constant properties corresponding to a Newtonian fluid, where the development of velocity profiles is of a diffusion nature. The velocity and shear stresses increase monotonically to their stationary values. The elasticity of a liquid gives a wave character to the development of its flow.

Laminar oscillatory flows of Maxwell and Oldroyd-B viscoelastic fluids were studied in [18]. Where many interesting features are demonstrated that are absent in the flows of Newtonian fluids. In [19-22], the electro-kinetic flow of viscoelastic liquids in a flat channel under the influence of an oscillatory pressure gradient was studied. It is assumed that the fluid movement occurs laminar and unidirectional; therefore, the fluid movement is in a linear mode. The surface potentials are assumed to be small, so the Poisson-Boltzmann equation is linearized. During flow, resonant behavior appears when the elastic properties of the Maxwell fluid predominate. The resonance phenomenon enhances electro-kinetic effects, and at the same time the efficiency of conversion of electro-kinetic energy is enhanced.

In the works listed above, the oscillatory and unsteady flow of a Newtonian fluid is mainly studied and the fluid velocity field is determined under various modes of change in pressure gradient. Changes in the maximum longitudinal velocity, fluid flow rate and tangential shear stress on the wall that arise during the movement of viscous and viscoelastic fluid in an unsteady flow have been studied relatively little. In most cases, in hydrodynamic models of unsteady flows, fluids were replaced by a sequence of flows with a quasi-stationary distribution of hydrodynamic quantities. At present, the question of the legitimacy of studying quasi-stationary characteristics to determine the field of tangential stresses in unsteady flows of viscous and viscoelastic fluids has practically not been resolved. Naturally, under such conditions there is a need to use hydrodynamic models of unsteady processes that take into account changes in the hydrodynamic characteristics of the flow depending on time.

In this paper, a generalized two-fluid Maxwell model is proposed for the combined flow of viscous and viscoelastic fluids. Based on the proposed model, the unsteady combined flow of a viscous and viscoelastic fluid in a flat channel under the influence of a constant pressure gradient is studied. Calculation formulas are determined for the distribution of longitudinal velocity and fluid flow. Transient processes during the starting mode of viscous and viscoelastic fluid flow are analyzed.

2. Statement of the problem and solution method

Based on Maxwell's two-fluid model, we will consider the solution to the problem of unsteady flow of a viscous and viscoelastic incompressible fluid between two stationary parallel planes extending in both directions to infinity. Let us denote the distance between the walls by $2h$. Axis Ox runs horizontally in the middle of the channel along the flow. Axis Oy directed perpendicular to the axis Ox . The differential equation of motion of a viscous and viscoelastic incompressible fluid has the following form [7-12,20]

$$\begin{cases} \rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} - \frac{\partial \tau}{\partial y}, & \tau = \alpha_1 \tau_1^0 + \alpha_2 \tau_2^0, \\ \tau_1^0 = -\eta_1^0 \frac{\partial u}{\partial y}, & \lambda \frac{\partial \tau_2^0}{\partial t} + \tau_2^0 = -\eta_2^0 \frac{\partial u}{\partial y} \end{cases} \quad (1)$$

Here u - longitudinal velocity of the mixture; p - mixture pressure; ρ - density of the mixture; τ - mixture tangential stress; t - time; τ_1^0 - true shear stress of the Newtonian fluid; τ_2^0 - is the true shear stress of the Maxwellian fluid. The coefficients of dynamic and kinematic viscosity of the mixture can be similarly expressed: $\eta = \alpha_1 \eta_1^0 + \alpha_2 \eta_2^0$, $\nu = \alpha_1 \nu_1^0 + \alpha_2 \nu_2^0$. Here η_1^0 , ν_1^0 - true dynamic and kinematic viscosities of Newtonian fluid; η_2^0 , ν_2^0 - true dynamic and kinematic viscosities of the Maxwellian fluid.

Substituting the second, third and fourth equations into the first equation of system (4) to determine the fluid velocity, we obtain the equation

$$\rho \frac{\partial u}{\partial t} = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial y} \left(\alpha_1 \eta_1^0 \frac{\partial u}{\partial y} + \alpha_2 \eta_2^0 \frac{1}{1 + \lambda D} \frac{\partial u}{\partial y} \right) \quad (2)$$

To solve equations (2), it is necessary to formulate the initial and boundary conditions. We believe that until the moment $t = 0$ the fluid is at rest. From the moment $t = 0$ fluid moves due to a positive constant pressure gradient

$\frac{\partial p}{\partial x} = \text{const}$ In this case, the initial condition and the no-slip condition on the wall will have the form:

$$u = 0 \quad t = 0, \quad u = 0 \quad y = h, \quad u = 0 \quad y = -h \quad (3)$$

Carrying out the Laplace-Carson transformation, i.e. passing from the original to the image in equation (2) using the initial and boundary conditions (3), we obtain:

$$\frac{d^2 \bar{u}(y, s)}{dy^2} + \frac{i^2 \rho s}{\eta \eta(s)} \bar{u}(y, s) = \frac{1}{\eta \eta(s)} \frac{d\bar{p}(x)}{dx} \quad (4)$$

$$\text{Here } \eta(s) = \left(\frac{\alpha_1 \eta_1^0}{\eta} + \frac{\alpha_2 \eta_2^0}{\eta} \frac{1}{1 + s\lambda} \right) = (X + Z \frac{1}{1 + s\lambda})$$

Using boundary conditions (3), we obtain a solution to equation (4) to determine the speed in the image:

$$\bar{u}(y, s) = \frac{1}{\rho s} \left(-\frac{d\bar{p}(x)}{dx} \right) \left(1 - \frac{\cos \left(i \sqrt{\frac{\rho s}{\eta \eta(s)}} y \right)}{\cos \left(i \sqrt{\frac{\rho s}{\eta \eta(s)}} h \right)} \right) \quad (5)$$

Using the inversion formula for the Laplace-Carson transform to determine the fluid velocity, we obtain the following integral expression:

$$u(y, t) = \frac{1}{2\pi i} \int_{\sigma - i\infty}^{\sigma + i\infty} e^{st} \frac{1}{\rho s} \left(-\frac{dp(x)}{dx} \right) \left(1 - \frac{\cos \left(i \sqrt{\frac{\rho s}{\eta \eta(s)}} y \right)}{\cos \left(i \sqrt{\frac{\rho s}{\eta \eta(s)}} h \right)} \right) \frac{ds}{s} \quad (6)$$

To calculate the integral (6) over a complex variable, it is necessary to establish residues for the integral expression. Equating the denominator to zero, and taking into account that the roots of the cosine are real numbers, we find:

$$s = 0 \quad \text{and} \quad s = -\nu \frac{s_{1,2,n}}{h^2} \quad (7)$$

Here $S_{1,2,n}$ there are solutions to the transcendental equation

$$\cos\left(i\sqrt{\frac{\rho s}{\eta\eta(s)}}h\right)=0 \quad (8)$$

All poles of equation (8) will be simple, so we can use the decomposition of a meromorphic function into simple fractions in the form

$$\frac{F_1(s)}{F_2(s)} = \frac{\left(-\frac{dp}{dx}\right)\left(\cos\left(i\sqrt{\frac{\rho s}{\eta\eta(s)}}h\right) - \cos\left(i\sqrt{\frac{\rho s}{\eta\eta(s)}}y\right)\right)e^{st}}{\rho s^2 \cos\left(i\sqrt{\frac{\rho s}{\eta\eta(s)}}h\right)} = \frac{C_0}{s} + \sum_{i=1}^2 \frac{C_{i,n}}{s - S_{i,n}} \quad (9)$$

To determine the deduction C_0 we must multiply both sides of equality (9) by S and then rush S to zero, and for $C_{i,n}$ you need to multiply (9) by the difference $S - S_{in}$ and direct S to value S_{in} . In this case, to determine the parameter S_{in} the result is a quadratic equation with two roots, which can be real or complex conjugate.

$$De\bar{s}^2 - \bar{s}(1 + XDea_0^2) + a_0^2 = 0 \quad (10)$$

Here $a_0 = \frac{2n+1}{2}\pi$, $De = \frac{\lambda\nu}{h^2}$ -The Deborah ah number characterizes the relaxation property of a viscoelastic fluid. Thus, we find a solution to equation (2) in the following form:

$$u(y,t) = \frac{h^2}{2\eta} \left(-\frac{dp(x)}{dx}\right) \left[\left(1 - \frac{y^2}{h^2}\right) + 32 \sum_{n=1}^{\infty} \sum_{i=1}^2 \frac{(-1)^n \cos\left(\left(\frac{2n+1}{2}\right)\pi \frac{y}{h}\right) e^{-\frac{\nu}{h^2} \bar{s}_{in} t}}{(2n+1)^3 \pi^3 \frac{1 - 2De\bar{s}_{in} + De^2 \bar{s}_{in}^2 X}{(1 - s_{in} De)^2}} \right] \quad (11)$$

$$\frac{u(0,t)}{u_0} = 1 + 32 \sum_{n=1}^{\infty} \sum_{i=1}^2 \frac{(-1)^n e^{-\frac{\nu}{h^2} \bar{s}_{in} t}}{(2n+1)^3 \pi^3 \frac{1 - 2De\bar{s}_{in} + De^2 \bar{s}_{in}^2 X}{(1 - \bar{s}_{in} De)^2}} \quad (12)$$

Here $u_0 = \frac{h^2}{2\eta} \left(-\frac{dp(x)}{dx}\right)$ maximum speed for stationary flow of Newtonian fluid. Expression

(11) indicates that when tending t towards infinity the velocity distribution becomes parabolic. Thus, the solution to the problem of stationary fluid motion between parallel walls is obtained from the solution to the problem of unsteady motion during circulation t to infinity.

3. Numerical calculations and discussion

Using the obtained formulas (11) and (12), numerical calculations were carried out to establish unsteady processes of hydrodynamic characteristics during unsteady flow of a viscoelastic fluid. For the convenience of comparing the established Newtonian fluid with the established viscoelastic fluid, we will first conduct a study of the Newtonian fluid. Formulas for calculating Newtonian fluid are obtained from formula (8) with $\lambda = 0$, $\eta(s) = 0$, $X = 1$, $Z = 0$

In this case, the solution to the transcendental equation has the form:

$$\cos\left(i\sqrt{\frac{\rho s}{\eta}}h\right)=0, \quad \left(i\sqrt{\frac{\rho s}{\eta}}h\right)=\frac{2n+1}{2}\pi, \quad s=-\frac{\nu}{h^2}\bar{s}, \quad \bar{s}=\frac{(2n+1)^2}{4}\pi^2 \quad (13)$$

Using these expressions from formula (12), one can find calculation formulas for studying Newtonian fluid

$$\frac{u(0,t)}{u_{0\max}}=1+32\sum_{n=1}^{\infty}\frac{(-1)^n}{(2n+1)^3\pi^3}e^{-\frac{\nu(2n+1)^2}{h^2}\pi^2 t} \quad (14)$$

Based on formula (14), numerical calculations were made and Figure. 1 shows graphs of the change in the ratio of the maximum speed of an unsteady flow to the maximum speed of a stationary flow depending on time. It can be seen that the relative maximum speed during an unsteady flow of a Newtonian fluid, depending on time, monotonically increases to a value corresponding to a stationary flow.

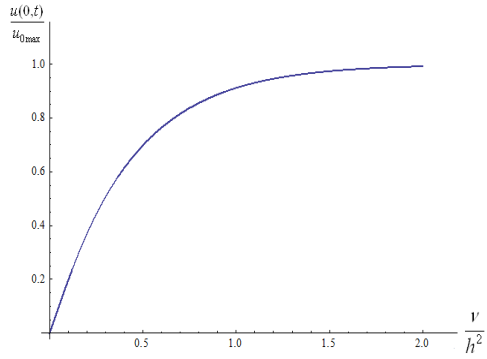


Figure 1- Change over time in the ratio of the maximum fluid velocity to the maximum velocity of a stationary profile during unsteady flow of a Newtonian fluid

Information about the process of transition of a Newtonian fluid from an unsteady flow regime to a stationary one is given in many literary sources [7-12]. But the process of transition of a viscoelastic fluid from an unsteady flow to a stationary one has not been sufficiently studied. Existing works [15,16] are also devoted to unsteady motions of viscoelastic fluid in some annular pipes, the solution of which is based on the finite difference method. Below we present an analysis of the problem solved on the basis of the generalized two-fluid Maxwell model. For this we use formulas (11) and (12). Based on the obtained formulas, the process of transition from a non-stationary state to a stationary state is analyzed.

In this case, the solution of the quadratic equation (7) determines the roots

$s_{1n,2n}$ included in formulas (11) and (12). It is known that the quadratic equation (10) has two roots: real different, real equal, and complex concomitant

$$s_{1n} = \frac{1 + XDea_0^2 + \sqrt{1 - 2Dea_0^2(X-2) + X^2De^2a_0^4}}{2De} \quad (15)$$

$$s_{2n} = \frac{1 + XDea_0^2 - \sqrt{1 - 2Dea_0^2(X-2) + X^2De^2a_0^4}}{2De}$$

In order for the roots of a quadratic equation to be real, it is necessary that the discriminant $1 - 2Dea_0^2(X-2) + X^2De^2a_0^4$ in formulas (15) must be equal to or greater than zero. Under such conditions, solution (15) does not change. Analysis of formula (12) is provided in Figure 2

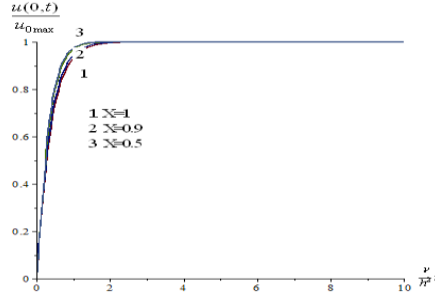


Figure 2- Change over time in the ratio of the maximum speed of the mixture to the maximum speed of the stationary profile during unsteady flow of the mixture (when the Deborah number $De = 0.1$, and at different concentrations of the Newtonian fluid)

From Figure 2, it is clear that in this case the processes of non-stationarity in the flow of a viscous and viscoelastic fluid are practically no different from the processes of non-stationarity in a Newtonian fluid. In this case, instead of the unsteady flow of a viscoelastic fluid, the process of establishing an unsteady flow of a Newtonian fluid can be adopted. Now consider the case when the roots of the quadratic equation (13) consist of complex conjugate roots. This case is observed when the discriminant equation (13) is less than zero. In this case, the solution to the quadratic equation is determined as follows.

$$s_{1n} = \frac{1 + XDea_0^2 + i\sqrt{2Dea_0^2(2-X) - X^2De^2a_0^4 - 1}}{2De} = a + bi, \quad (16)$$

$$s_{2n} = \frac{1 + XDea_0^2 - i\sqrt{2Dea_0^2(2-X) - X^2De^2a_0^4 - 1}}{2De} = a - bi$$

Here $a = \frac{1 - XDea_0^2}{2De}$, $b = \frac{\sqrt{2Dea_0^2(2-X) - X^2De^2a_0^4 - 1}}{2De}$ In this case, solution (16) has the following form;

$$\frac{u(0,t)}{u_0} = 1 + 32 \sum_{n=0}^{\infty} \frac{2(-1)^{n+1} e^{-\frac{\nu}{h^2} a_n t}}{(2n+1)^3 \pi^3 (M_2^2 + N_2^2)} (M_3 \cos b_n \frac{\nu}{h^2} t + N_3 \sin b_n \frac{\nu}{h^2} t) \quad (17)$$

Here

$$M_1 = 1 - 2a_n De + a_n^2 De^2 - b_n^2, \quad N_1 = 2a_n b_n - 2b_n De$$

$$M_2 = 1 - 2a_n De + a_n^2 De^2 X - De^2 b_n^2 X, \quad N_2 = 2a_n b_n De^2 X - 2b_n De$$

$$M_3 = M_1 M_2 + N_1 N_2, \quad N_3 = -(M_1 N_2 - M_2 N_1)$$

Using formula (17), we analyze the results of a numerical calculation of the unsteady flow of a viscous and viscoelastic fluid when the solutions to equation (16) consist of complex conjugate roots. Figure 3 shows the change over time in the ratio of the maximum speed to the maximum speed of a stationary profile during unsteady flow of the mixture (when the Deborah number has the values $De = 3$ and at different values of the concentration of the Newtonian liquid) From Figure 3 it is clear that the transition from an unsteady state to a stationary state during the flow of the mixture manifests itself as waves, in contrast to the Newtonian liquid and the transition time of the viscoelastic liquid is several times longer than the transition time of the Newtonian liquid. The main reason for the sharp increase in hydrodynamic quantities during the transition of a viscoelastic fluid can be called the inertial force taken into account in the Maxwell model. It is easy to see that if we remove the action of the inertial force, then it is no different from a Newtonian fluid. The effect of inertial force is explained by the presence of the Deborah number, that is, relaxation.

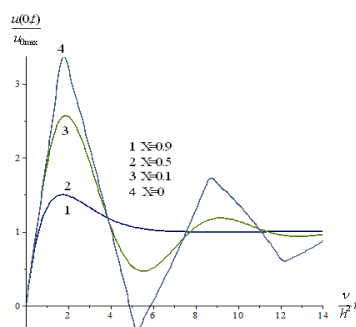


Figure 3 - Change over time in the ratio of the maximum speed of the mixture to the maximum speed of the stationary profile during unsteady flow of the mixture (when the Deborah number $De = 3$ and at different concentrations of Newtonian fluid)

The fact that the increase in the Deborah number exceeds the flow rate of the viscoelastic fluid is 4.5-5 times compared to the flow rate of the Newtonian fluid in the transient process. These changes are presented in Figure 3. From this it can be seen that at higher Deborah numbers, the process of transition of a viscoelastic fluid from an unsteady state to a stationary state has a wave form, in contrast to the transition process of a Newtonian fluid, and the transition time is several times longer than the transition time of a Newtonian fluid. It was also discovered that perturbed processes can arise in the transition, which will be stabilized by mixing the Newtonian fluid in it. The implementation of this property is important in technical and technological processes, in preventing technical failures or malfunctions.

4. Conclusion

Based on the proposed two-fluid Maxwell model, the problem of unsteady flow of a viscoelastic fluid under the influence of a constant pressure gradient in a flat channel is solved. To solve the problem, the Laplace-Carson transformations were applied, formulas were determined for the distribution of the velocity profile and fluid flow in an unsteady flow of viscoelastic fluid. Based on the established formulas, the processes of transition of the characteristics of a viscoelastic fluid in a flat channel from an unsteady state to a stationary state are analyzed. Based on the results of the analysis, it was shown that the transient processes of the unsteady flow of a viscoelastic fluid, at small values of the Deborah number, are not fundamentally different from the transient process in a Newtonian fluid. Exceeding the Deborah number relatively unity, it has been established that the process of transition of a viscoelastic fluid from an unsteady state to a stationary state has a wave form, in contrast to the transition process of a Newtonian fluid, and the transition time is several times longer than the transition time of a Newtonian fluid. It was also discovered that perturbed processes can arise in the transition, which will be stabilized by mixing the Newtonian fluid in it. The implementation of this property is important in technical and technological processes, in preventing technical failures or malfunctions.

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COMPARISON OF NUMERICAL SCHEMES FOR THE PROBLEM OF LAMINAR FLOW IN A CHANNEL WITH A SMALL OBSTACLE

Nazarov F.Kh.¹, Abdukhamidov S.K.²

¹*Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan*

E-mail: Sardor.abdukhamidov@mail.ru

Abstract: *This paper presents a numerical study of laminar flow in a channel with a small obstacle using various computational schemes. The Navier-Stokes equation was used to numerically solve the laminar flow in the channel. To calculate the Navier-Stokes equation, the third-order explicit QUICK scheme, the second-order Peaceman-Rahford scheme, and the second-order implicit upstream scheme were used. For the difference approximation of the original equations, the finite difference method was used, and the relationship between velocities and pressure was found using the SIMPLE procedure. The results of longitudinal velocity in different sections of the channel at different values of Reynolds numbers were compared for three different schemes. In addition, numerical results were obtained for current contours and friction coefficients at various Reynolds numbers.*

Keywords: *Navier-Stokes equations, laminar flow, QUICK scheme, Peaceman-Ratchford scheme, implicit scheme, obstacle.*

1. Introduction

Laminar flow in a channel with a small obstacle occurs in many hydrodynamic devices. In this case, the fluid speed increases in a narrow section and then decreases at the exit from it. As a result, an area of high pressure is formed. This results in a more laminar flow, meaning the fluid moves more smoothly and orderly. One example of laminar flow in a channel with an obstacle is the flow in blood vessels. Venous bleeding can cause slow blood flow and the formation of blood clots. On the other hand, narrow capillaries create increased resistance for blood, which promotes more laminar flow and more efficient delivery of oxygen and nutrients to tissues [1].

It is well known that the numerical solution of problems on the motion of a viscous incompressible fluid based on the Navier-Stokes equations is complicated not only by its nonlinearity, but also by the absence of an explicit equation for determining pressure. It should be noted that many previous experimental studies were carried out for turbulent fluid motion, and laminar flow in sudden expansion channels was usually considered at low Reynolds numbers. However, modern progress in computational fluid mechanics makes it possible to improve the quality of modeling physical processes of vortex formation and obtain stable numerical solutions at moderate Reynolds numbers. In [2], an effective method for numerical integration of the complete system of nonstationary Navier-Stokes equations in the physical variables velocity-pressure for an incompressible fluid was proposed.

This work is aimed at solving the internal hydrodynamics problem of separated fluid flow in a straight channel with a small obstacle - a step. In this case, in addition to studying the structure of the flow behind the step depending on the Reynolds number, the goal is to test the efficiency of calculating complex flows characterized by the presence of swirling flow regions.

Numerical schemes are methods for approximate solution of mathematical problems when the exact solution cannot be found analytically. They are the basis of computational mathematics and are used in many fields of science and technology, including physics, economics, engineering, biology, etc. Numerical schemes are divided into several classes depending on how they use information about the original problem and how they approximate its solution. One of the classes of numerical schemes is finite difference methods. These methods are based on approximating the derivatives of the original function by the differences between the values of this function at neighboring points on the grid. Based on these differences, system equations are constructed, which are then solved numerically.

Another class of numerical schemes is finite element methods. They are based on partitioning the region where a solution is required into a finite number of simpler subdomains called finite elements. In each element, the solution is approximated by a function that is specified on this element using a small set of parameters (coefficients). The resulting equations are solved numerically, first in each element, and then the system of equations is combined into a single system, which is also solved numerically.

There are other classes of numerical schemes, for example, spectral decomposition methods, finite volume methods, and others. Each class of circuits is used depending on the specific task and requirements for accuracy and efficiency of calculations [3,4].

The purpose of the work is to study various numerical schemes for solving laminar flow in a channel with a small obstacle and test their effectiveness. To solve the complete Navier-Stokes equations, three schemes with different orders of accuracy were used:

- 1) The scheme is implicit [5];
- 2) QUICK scheme (Quadratic Upwind Interpolation for Convective Kinematics) [6];
- 3) Peaceman-Rachford scheme [7].

2. Physical and mathematical formulation of the problem.

A two-dimensional laminar flow in a flat channel with a small obstacle is considered. The physical picture of the analyzed flow and the configuration of the computational domain are shown in Fig. 1. The width of the two-dimensional channel is H , the length is $30H$, the height of a small obstacle is $H/2$, the distance to it is $10H$.

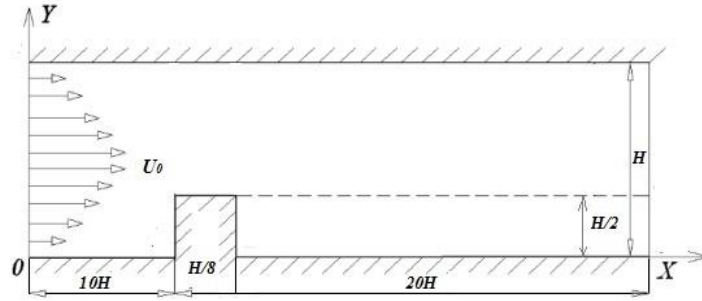


Figure 1. - *Channel with minor obstruction.*

At the entrance to the channel, a parabolic Poiseuille flow profile was specified for the longitudinal velocity $U(Y)=6*Y(1-Y)$ and the vertical velocity V equal to zero. The unsteady Navier-Stokes equations are used to describe fluid motion without any simplifying assumptions. When introducing dimensionless quantities, the length scale is taken to be the channel width H , and the velocity scale is the average flow rate U_0 at the channel inlet.

The simulation is performed by solving the two-dimensional Navier-Stokes equation for a flow with constant density $\rho = const$ and molecular kinematic viscosity ν . The dimensionless continuity and momentum equations for a stationary two-dimensional flow with constant density are given as follows [8]:

$$\begin{cases} \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0; \\ \frac{\partial U}{\partial t} + U \frac{\partial U}{\partial x} + V \frac{\partial U}{\partial y} + \frac{\partial P}{\partial x} = \frac{1}{Re} \left(\frac{\partial^2 U}{\partial x^2} + \frac{\partial^2 U}{\partial y^2} \right); \\ \frac{\partial V}{\partial t} + U \frac{\partial V}{\partial x} + V \frac{\partial V}{\partial y} + \frac{\partial P}{\partial y} = \frac{1}{Re} \left(\frac{\partial^2 V}{\partial x^2} + \frac{\partial^2 V}{\partial y^2} \right). \end{cases} \quad (1)$$

Here U, V is the dimensionless horizontal and vertical components of the flow velocity, P is the dimensionless hydrostatic pressure and $Re = \frac{HU_0}{\nu}$ is the Reynolds number.

Obvious boundary conditions of adhesion $U|_{\Gamma} = 0$ and $V|_{\Gamma} = 0$ are set on all motionless solid walls, where Γ is the solid boundary.

3. Solution method

To numerically solve the system of initial nonstationary equations (1), the finite difference method was used. Due to the complexity of matching the velocity and pressure fields, to discretize the equations of motion in Y X, directions and continuity equations, a grid with a staggered structure of the location of grid nodes for the dependent variables was used. This means that the velocity and pressure components are determined at different nodes. This approach is similar to SIMPLE methods and provides certain advantages when calculating the pressure field [9].

For the finite-difference approximation of the original equations of motion, the two-step implicit Peaceman-Rachford scheme was used [7], and for the continuity equations, an implicit scheme was used. The Peaceman-Rachford scheme for approximating derivatives with respect to time and space of second order accuracy has the form $O(\Delta t^2, \Delta x^2, \Delta y^2)$. The diffusion and convective terms are approximated using a central difference scheme. The Peaceman-Rachford scheme for the longitudinal velocity equation is as follows:

Step 1: Use the implicit method in direction i

$$\begin{aligned} \bar{U}_{i,j}^{n+1} - U_{i,j}^n + \Delta t UU \frac{\bar{U}_{i+1,j}^{n+1} - \bar{U}_{i-1,j}^{n+1}}{2\Delta x} + \Delta t VV \frac{U_{i,j+1}^n - U_{i,j-1}^n}{2\Delta y} &= \Delta t \frac{U_{i,j+1}^n - 2U_{i,j}^n + U_{i,j-1}^n}{\text{Re } \Delta y^2} + \Delta t \frac{\bar{U}_{i+1,j}^{n+1} - 2\bar{U}_{i,j}^{n+1} + \bar{U}_{i-1,j}^{n+1}}{\text{Re } \Delta x^2} + F; \\ F &= -\Delta t \frac{p_{i+1,j}^{n+1} - p_{i-1,j}^{n+1}}{2\Delta x}, A = U_{i,j}^n + \Delta t \frac{U_{i,j+1}^n - 2U_{i,j}^n + U_{i,j-1}^n}{\text{Re } \Delta y^2} - \Delta t VV \frac{U_{i,j+1}^n - U_{i,j-1}^n}{2\Delta y} + F, \\ \bar{U}_{i,j}^{n+1} \left(1 + \Delta t \frac{2}{\text{Re } \Delta x^2} \right) - \bar{U}_{i+1,j}^{n+1} \left(\frac{\Delta t}{\text{Re } \Delta x^2} - \Delta t \frac{UU}{2\Delta x} \right) - \bar{U}_{i-1,j}^{n+1} \left(\frac{\Delta t}{\text{Re } \Delta x^2} + \Delta t \frac{UU}{2\Delta x} \right) &= A, \\ B = \left(1 + \Delta t \frac{2}{\text{Re } \Delta x^2} \right); C = \left(\frac{\Delta t}{\text{Re } \Delta x^2} - \Delta t \frac{UU}{2\Delta x} \right); D = \left(\frac{\Delta t}{\text{Re } \Delta x^2} + \Delta t \frac{UU}{2\Delta x} \right). \\ \bar{U}_{i,j}^{n+1} B - \bar{U}_{i+1,j}^{n+1} C - \bar{U}_{i-1,j}^{n+1} D &= A \end{aligned} \quad (2)$$

Step 2: Use the implicit method in the j direction

$$\begin{aligned} U_{i,j}^{n+1} - \bar{U}_{i,j}^{n+1} + \Delta t UU \frac{\bar{U}_{i+1,j}^{n+1} - \bar{U}_{i-1,j}^{n+1}}{2\Delta x} + \Delta t VV \frac{U_{i,j+1}^{n+1} - U_{i,j-1}^{n+1}}{2\Delta y} &= \Delta t \frac{U_{i,j+1}^{n+1} - 2U_{i,j}^{n+1} + U_{i,j-1}^{n+1}}{\text{Re } \Delta y^2} + \Delta t \frac{\bar{U}_{i+1,j}^{n+1} - 2\bar{U}_{i,j}^{n+1} + \bar{U}_{i-1,j}^{n+1}}{\text{Re } \Delta x^2} + F; \\ F &= -\Delta t \frac{p_{i+1,j}^{n+1} - p_{i-1,j}^{n+1}}{2\Delta x}, A = \bar{U}_{i,j}^{n+1} + \Delta t \frac{\bar{U}_{i+1,j}^{n+1} - 2\bar{U}_{i,j}^{n+1} + \bar{U}_{i-1,j}^{n+1}}{\text{Re } \Delta x^2} - \Delta t UU \frac{\bar{U}_{i+1,j}^{n+1} - \bar{U}_{i-1,j}^{n+1}}{2\Delta x} + F, \\ U_{i,j}^{n+1} \left(1 + \Delta t \frac{2}{\text{Re } \Delta y^2} \right) - U_{i,j+1}^{n+1} \left(\frac{\Delta t}{\text{Re } \Delta y^2} - \Delta t \frac{VV}{2\Delta y} \right) - U_{i,j-1}^{n+1} \left(\frac{\Delta t}{\text{Re } \Delta y^2} + \Delta t \frac{VV}{2\Delta y} \right) &= A, \\ B = \left(1 + \Delta t \frac{2}{\text{Re } \Delta y^2} \right); C = \left(\frac{\Delta t}{\text{Re } \Delta y^2} - \Delta t \frac{VV}{2\Delta y} \right); D = \left(\frac{\Delta t}{\text{Re } \Delta y^2} + \Delta t \frac{VV}{2\Delta y} \right). \\ U_{i,j}^{n+1} B - U_{i,j+1}^{n+1} C - U_{i,j-1}^{n+1} D &= A \end{aligned} \quad (3)$$

The QUICK scheme [10] has third order accuracy in space and first order in time. QUICK scheme on a regular grid:

$$\frac{\partial F}{\partial t} + U \frac{\partial F}{\partial x} + V \frac{\partial F}{\partial y} = \frac{1}{\text{Re}} \left(\frac{\partial^2 F}{\partial x^2} + \frac{\partial^2 F}{\partial y^2} \right) + P^F \quad (4)$$

$$\text{Here } F = \begin{pmatrix} U \\ V \end{pmatrix}, \quad P^F = \begin{pmatrix} -\frac{\partial P}{\rho \partial x} \\ -\frac{\partial P}{\rho \partial y} \end{pmatrix}$$

Let's write it down

$$U_E = 0.5(U_{i,j}^n + U_{i+1,j}^n), U_W = 0.5(U_{i,j}^n + U_{i-1,j}^n), V_N = 0.5(V_{i,j}^n + V_{i+1,j}^n), V_S = 0.5(V_{i,j-1}^n + V_{i+1,j-1}^n).$$

$$\begin{aligned} U_E > 0 &\Rightarrow F_E = \frac{3}{8}\Phi_E + \frac{3}{4}\Phi_P - \frac{1}{8}\Phi_W; U_E < 0 \Rightarrow F_E = \frac{3}{8}\Phi_P + \frac{3}{4}\Phi_E - \frac{1}{8}\Phi_{EE}; \\ U_W > 0 &\Rightarrow F_W = \frac{3}{8}\Phi_P + \frac{3}{4}\Phi_W - \frac{1}{8}\Phi_{WW}; U_W < 0 \Rightarrow F_W = \frac{3}{8}\Phi_W + \frac{3}{4}\Phi_P - \frac{1}{8}\Phi_E; \\ V_N > 0 &\Rightarrow F_N = \frac{3}{8}\Phi_N + \frac{3}{4}\Phi_P - \frac{1}{8}\Phi_S; V_N < 0 \Rightarrow F_N = \frac{3}{8}\Phi_P + \frac{3}{4}\Phi_N - \frac{1}{8}\Phi_{NN}; \\ V_S > 0 &\Rightarrow F_S = \frac{3}{8}\Phi_P + \frac{3}{4}\Phi_S - \frac{1}{8}\Phi_{SS}; V_S < 0 \Rightarrow F_S = \frac{3}{8}\Phi_S + \frac{3}{4}\Phi_P - \frac{1}{8}\Phi_N; \end{aligned}$$

Here,

$$\begin{aligned} \Phi_E &= 0.5(U_{i,j}^n + U_{i+1,j}^n), \Phi_P = 0.5(U_{i-1,j}^n + U_{i,j}^n), \Phi_W = 0.5(U_{i-2,j}^n + U_{i-1,j}^n), \Phi_N = 0.5(U_{i-1,j+1}^n + U_{i,j+1}^n), \\ \Phi_S &= 0.5(U_{i-1,j-1}^n + U_{i,j-1}^n), \Phi_{EE} = 0.5(U_{i+2,j}^n + U_{i+1,j}^n), \Phi_{WW} = 0.5(U_{i-2,j}^n + U_{i-3,j}^n), \\ \Phi_{NN} &= 0.5(U_{i-1,j+2}^n + U_{i,j+2}^n), \Phi_{SS} = 0.5(U_{i-1,j-2}^n + U_{i,j-2}^n). \end{aligned}$$

Let us write (4) as follows:

$$\begin{aligned} \frac{F_{i,j}^{n+1} - F_{i,j}^{n+1}}{\Delta t} + \frac{U_E F_E - U_W F_W}{\Delta x} + \frac{U_N F_N - U_S F_S}{\Delta y} &= \frac{F_{i+1,j}^n - 2F_{i,j}^{n+1} + F_{i-1,j}^n}{\text{Re} \Delta x^2} + \frac{F_{i,j+1}^{n+1} - 2F_{i,j}^{n+1} + F_{i,j-1}^{n+1}}{\text{Re} \Delta y^2} - P^F. \\ A &= \left(1 + \frac{2\Delta t}{\text{Re} \Delta y^2} + \frac{2\Delta t}{\text{Re} \Delta x^2}\right), B = \left(\frac{\Delta t}{\text{Re} \Delta y^2}\right), C = \Delta t \frac{F_{i+1,j}^n + F_{i-1,j}^n}{\text{Re} \Delta x^2} \\ D &= F_{i,j}^{n+1} - \Delta t \left(\frac{U_E F_E - U_W F_W}{\Delta x} + \frac{U_N F_N - U_S F_S}{\Delta y}\right) - P^F + C \\ F_{i,j}^{n+1} A - F_{i,j+1}^{n+1} B - F_{i,j-1}^{n+1} B &= D \end{aligned} \quad (5)$$

The result is a finite difference equation for the internal computational nodes. To solve equations (2), (3) and (5), the sweep method was used [10-11].

A similar scheme is used for lateral velocity. A special feature of discretization is that the finite-difference approximation is centered on the selected template. In this case, the grid indices for the dependent variables are shifted [12]. The velocities obtained from schemes (2), (3) and (5) do not satisfy the continuity equation. Therefore, following the SIMPLE procedure, we introduce a correction for pressure δp , satisfying the following condition.

$$\begin{cases} U^{n+1} = \tilde{U} - \Delta t \frac{\partial \delta p}{\partial x} \\ V^{n+1} = \tilde{V} - \Delta t \frac{\partial \delta p}{\partial y} \end{cases} \quad (6)$$

Equations (6) superscript “ $\tilde{U}$ ” denotes the intermediate grid function for the velocity vector; $\delta p = p^{n+1} - p^n$ pressure correction. Multiplying equation (6) by the gradient and taking into account the solenoidality of the velocity vector at the $(n+1)$ th time layer, we obtain the Poisson equation for determining the correction to pressure:

$$\Delta t \left(\frac{\partial^2 \delta p}{\partial x^2} + \frac{\partial^2 \delta p}{\partial y^2} \right) = \frac{\partial \tilde{U}}{\partial x} + \frac{\partial \tilde{V}}{\partial y}. \quad (7)$$

The solution to equation (7) was carried out using the iteration method, for which equation (7) was reduced to parabolic form

$$\frac{\partial \delta p}{\partial t_0} - \Delta t \left(\frac{\partial^2 \delta p}{\partial x^2} + \frac{\partial^2 \delta p}{\partial y^2} \right) = \frac{\partial \tilde{U}}{\partial x} + \frac{\partial \tilde{V}}{\partial y} \quad (8)$$

where the fictitious t_0 time is the iterative parameter. When solving equation (6) for the time step, you can write $\Delta t_0 = a_1 \Delta t$, while the value a_1 is constant, as a rule, less than unity and is chosen from the condition of rapid convergence of the numerical process. The Neumann condition is used as a boundary condition for the pressure correction, which is satisfied if the exact value $\tilde{U}$ is used for U^{n+1} at the boundary. To numerically solve the transport equation of system (8), a finite-difference scheme against the flow for convective terms was used. And the diffusion terms were implicitly approximated by the central difference.

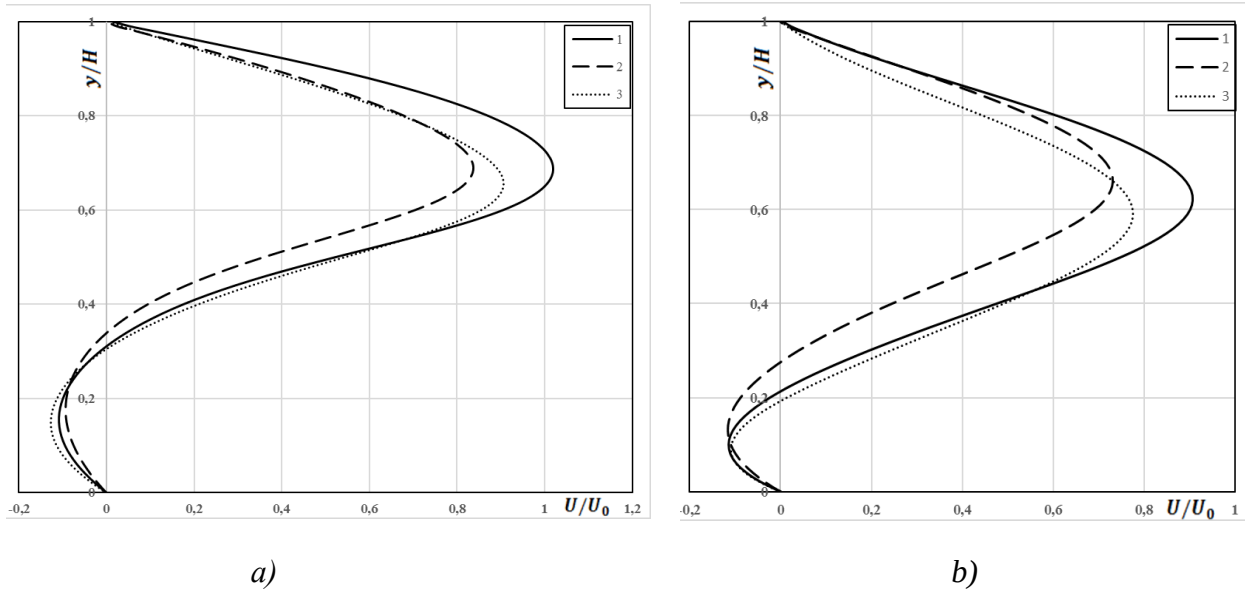
To numerically solve equation (6), a semi-implicit scheme was used, which is effectively implemented by the sweep method

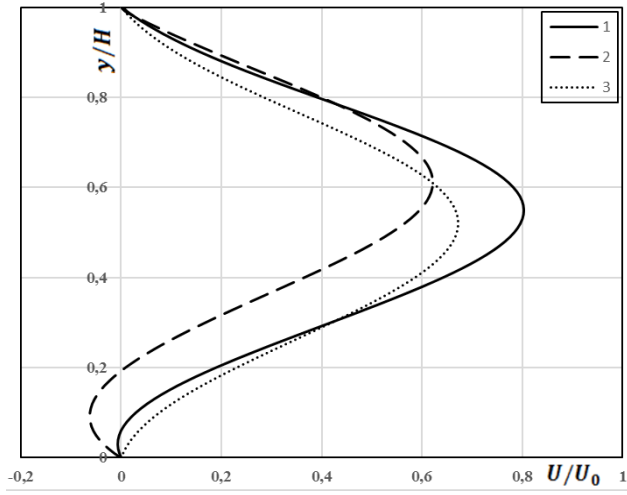
$$\frac{\delta p_{i,j}^{n+1} - \delta p_{i,j}^n}{\Delta t_0} - \left(\frac{\delta p_{i+1,j}^n - 2\delta p_{i,j}^n + \delta p_{i-1,j}^n}{\Delta x^2} \right) - \left(\frac{\delta p_{i,j+1}^n - 2\delta p_{i,j}^n + \delta p_{i,j-1}^n}{\Delta y^2} \right) = \frac{1}{\Delta t} \left(\frac{\tilde{U}_{i,j} - \tilde{U}_{i-1,j}}{\Delta x} + \frac{\tilde{V}_{i,j} - \tilde{V}_{i,j-1}}{\Delta y} \right). \quad (9)$$

Thus, first the system of equations (2), (3), (5) and (6) is solved by the establishment method, then equation (9) and in accordance with (7) the velocity vector at the $(n+1)$ time layer and the pressure $p^{n+1} = p^n + \delta p$ are determined. It is easy to show that the calculation scheme has an order of accuracy of $O(\Delta t, \Delta x, \Delta y^2)$ [13-14].

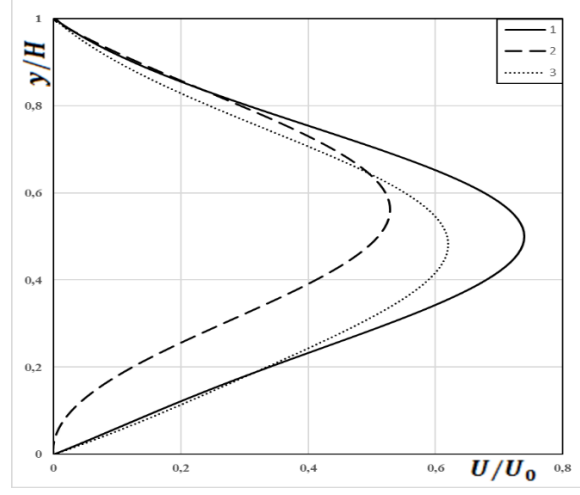
Results were also obtained for the implicit, QUICK and Peaceman-Rachford schemes. To solve the problem, a rectangular computational grid of size 300×100 with a step of $\Delta x = 0.05$, $\Delta y = 0.01$, $\Delta t < 0.001$. was used.

Calculation results and their discussion. Figures 2, 3 and 4 show the results for three schemes at different sections of the channel at Reynolds numbers of 200, 600 and 1000.



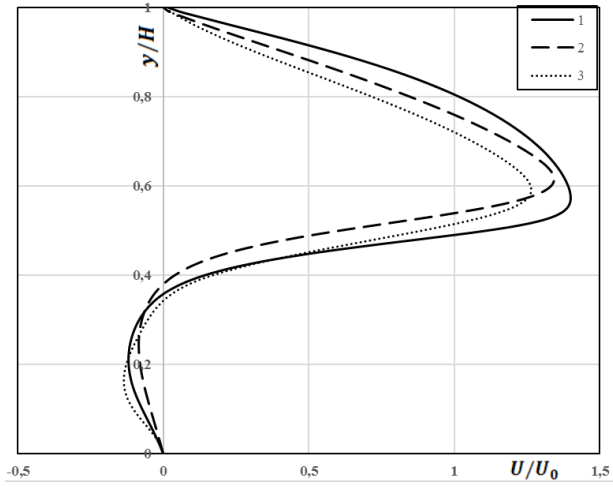


v)

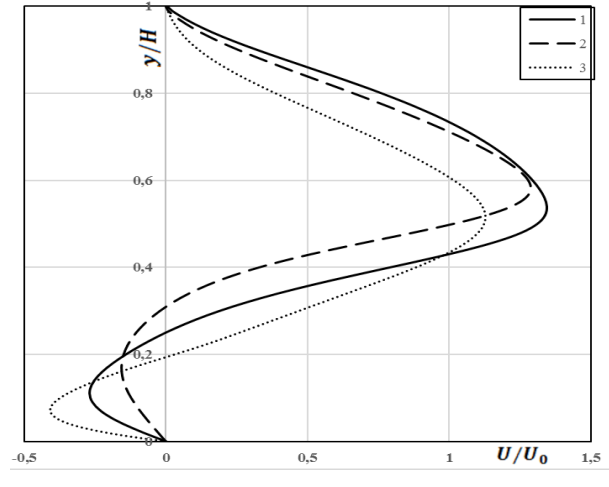


g)

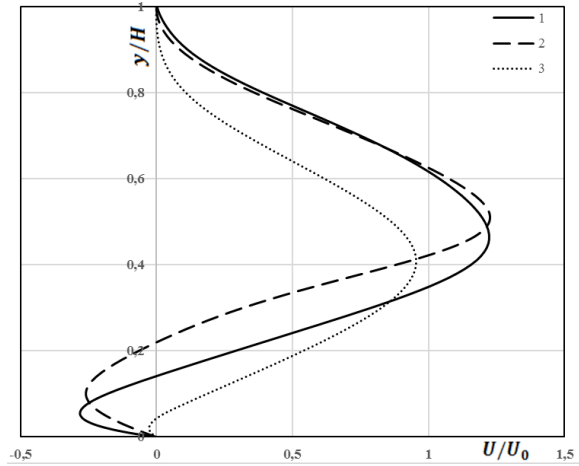
Figure 2. - Results of calculating the longitudinal velocity at $Re = 200$ and $x/H = 11$ (a), 11.5 (b), 12 (c), 12.5 (d) for the implicit (1) QUICK (2) and Peaceman-Rachford schemes (3).



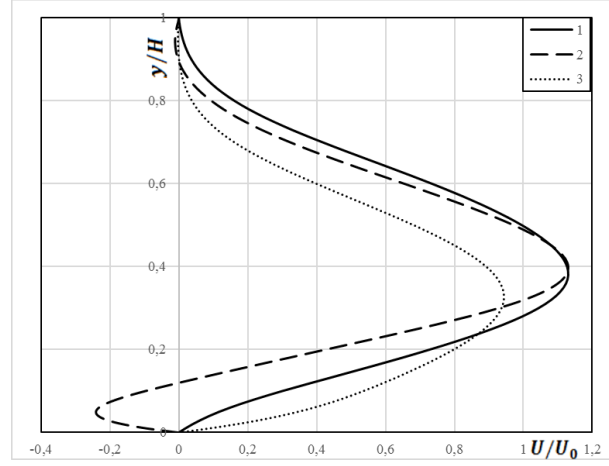
a)



b)



c)



d)

Figure 3. - Results of calculating the longitudinal velocity at $Re=600$ and $x/H = 11$ (a), 11.5 (b), 12 (c), 12.5 (d) for the implicit (1) QUICK (2) and Peaceman-Rachford schemes (3).

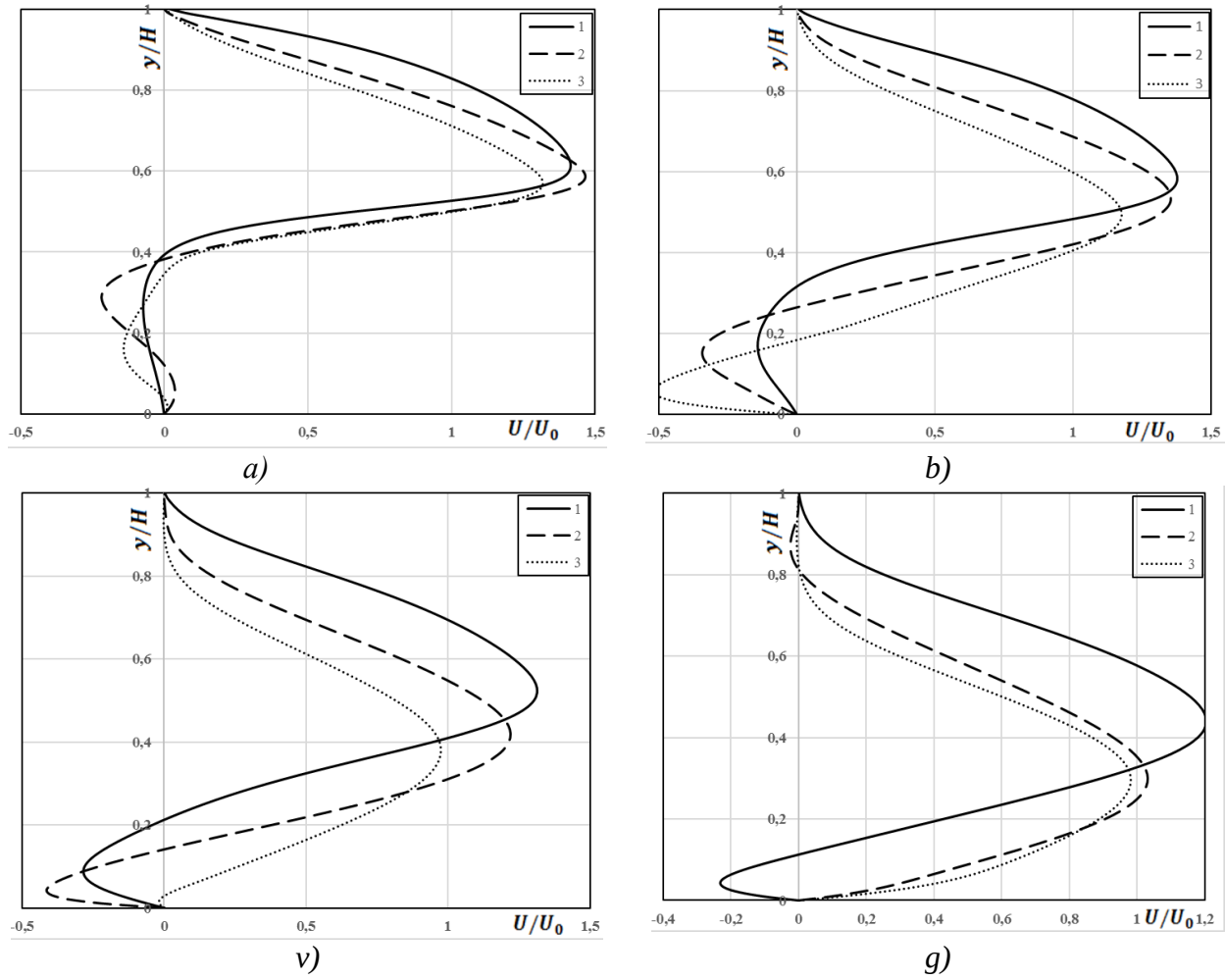
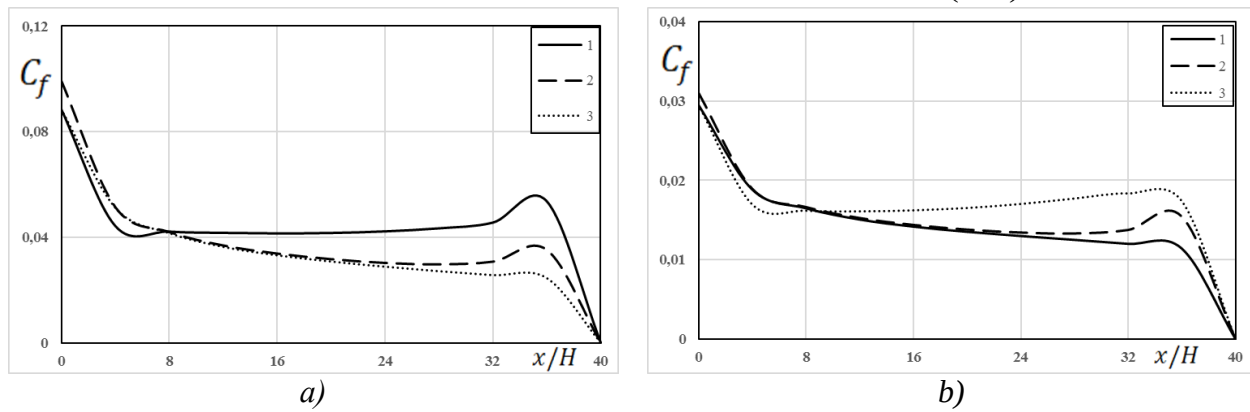


Figure 4. - Results of calculating the longitudinal velocity at $Re=1000$ and $x/H = 11$ (a), 11.5 (b), 12 (c), 12.5 (d) for the implicit (1) QUICK (2) and Peaceman-Rachford schemes (3).

In Fig. Figure 5 shows the distribution of the friction coefficient on the lower wall of the channel along its length for different values of the Reynolds number according to the results obtained using three schemes. The friction coefficient is calculated using the formula $C_f = \frac{2}{Re} \left(\frac{\partial U}{\partial y} \right)$.



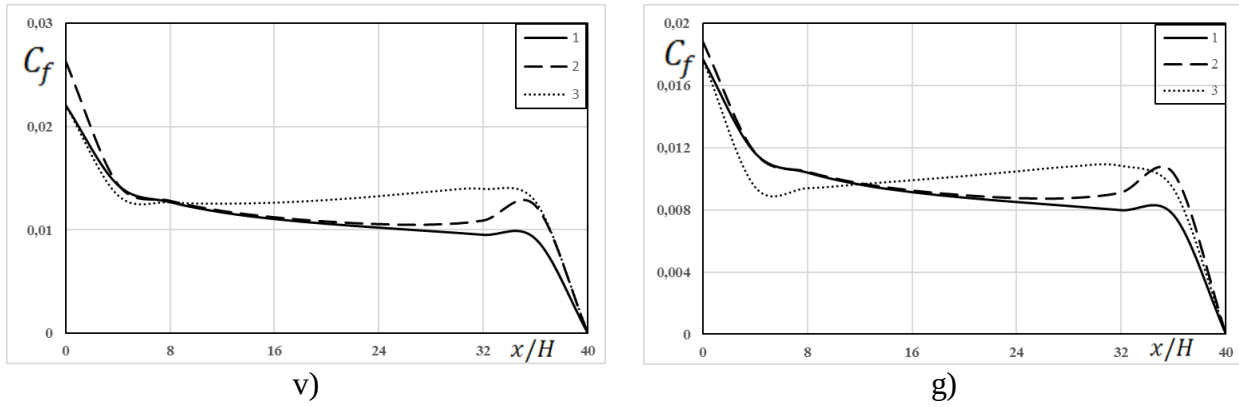


Figure 5. - Distributions of the friction coefficient along the channel length at $Re = 200$ (a), 600 (b), 800 (c), 1000 (d) for the implicit (1), QUICK (2), and Peaceman-Rachford schemes - (3)

In Fig. 6, 7 and 8 show longitudinal velocity isolines for three schemes at Reynolds numbers $Re = 200, 600, 800$ and 1000.

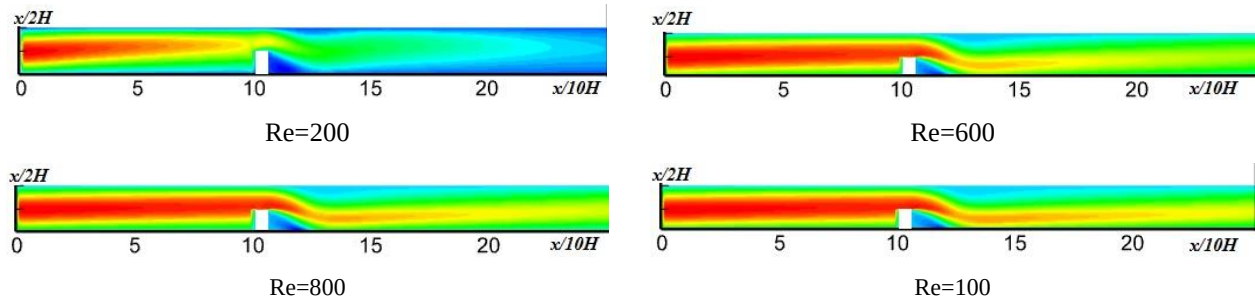


Figure 6. - The longitudinal velocity isolines for the implicit scheme are shown.

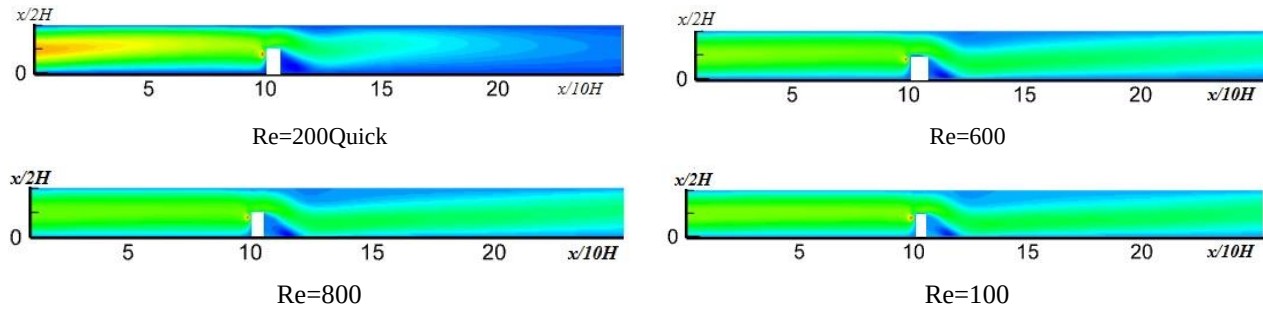


Figure 7. - Longitudinal velocity contours for the QUICK scheme are shown.

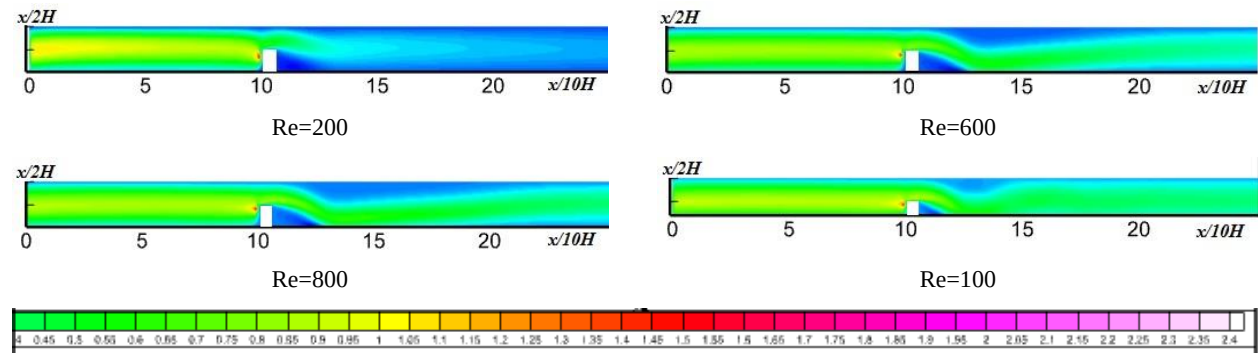


Figure 8. - Longitudinal velocity contours for the Peaceman-Rachford scheme are shown.

Table 1

The values of time (s) spent on performing calculations according to various schemes are given below:

Implicit		1260
QUICK		1440
Peaceman-Rachford		2160

Calculation using the implicit and QUICK schemes requires approximately almost the same time, while the Peaceman-Rachford scheme requires 2 times longer.

In figure 9 shows the calculated dependences of the length of the circulation zone behind the step x/H depending on the Reynolds number with different normalizations at the value of the step height $H/4$.

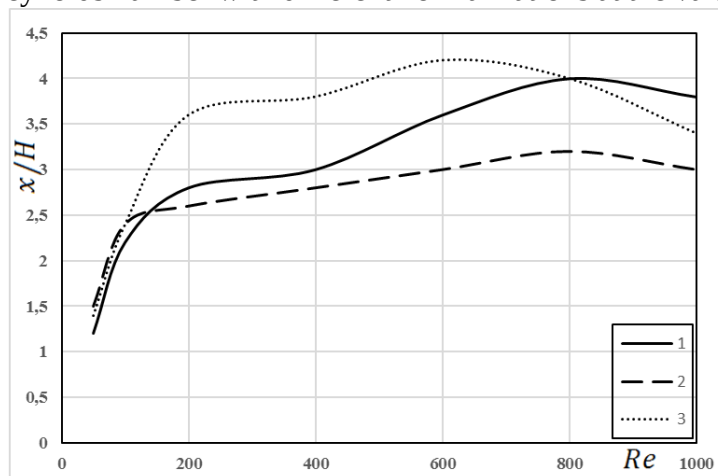


Figure 9. - The lengths of the circulation zone behind the step x/H are shown as a function of the Reynolds number in a flat channel with a small obstacle for the implicit (1), QUICK (2) and Peaceman-Rachford (3) schemes.

4. Conclusion

Calculation results are presented for the finite-difference implicit, QUICK, and Peaceman-Rachford schemes when studying laminar flow in a channel with a small obstacle. As a result of a detailed numerical study of the fields of velocity, pressure and friction, the patterns of formation of a vortex flow structure in a flat channel with a small obstacle, including secondary vortex formations and unsteady separated flow regimes, are shown. It has been established that for flows in which separation and circulation zones arise, more accurate schemes should be used. For separated flows, the Peaceman-Rachford scheme does not give good results, but it is more stable than other schemes. The implicit and QUICK schemes give approximately the same results for the longitudinal velocity profile in different sections depending on the Reynolds number and friction coefficients, but the dependence on the length of the primary vortex does not change they give the same results.

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SIMULATION OF FLOW DYNAMICS IN A GAS NETWORK PIPE

Ravshanov Sh.A.¹, Aminov Kh.², Rahmonova N.³

¹*Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan*

²*Oriental university, Tashkent, Uzbekistan*

³*Tashkent University of Information Technologies named after Muhammad al-Khwarizmi, Tashkent, Uzbekistan*

E-mail: shohjaxonravshanov22@gmail.com

Abstract: The work simulates some applied aerodynamics problems, such as mixing two gases in a pipe, throttle calculations, and selecting compressor characteristics. Mathematical models were compiled and approximate calculations were carried out for these problems. The results showed that Simulink performs well in aerodynamic calculations of heat and mass transfer processes.

Keywords: Pneumatic drive chain, Prandtl number, Simscape Simulation

1. Introduction

Two thousand years ago, people discovered the laws of communicating vessels, the basic laws of hydrostatics (known as Toricelli's law) and much more. They built hydraulic presses, wells, aqueducts, water mills, siphons and many other things that were useful for the intellect of people. Recent advances now and in the future will enrich this information. The technology in this area is becoming more precise and the scale of its application is becoming noticeable.

Pipelines are widely used both in everyday life and in technology. They are designed to transport liquids, gases and their mixtures in combination with solid components, as well as to transmit thermal and mechanical energy. They can create and propagate compaction and discharge waves, and also operate continuously.

With the growth of technology, technology and living conditions, new problems associated with pipelines arise. The successful solution of these problems depends on the development of theoretical and applied research. A huge amount of mathematical software has been created for pipeline transport problems in various environments. This is used in engineering calculations. However, network topologies are becoming more complex due to new consumers and new sources. That is, engineering communications systems are constantly evolving. However, the main results concern stationary flow regimes. As already mentioned, the non-stationary mode of network operation is associated with the formation of differences in individual indicators.

The graphical extension MATLAB- Simulink is a popular tool in knowledge-intensive fields. Thanks to this extension, you can create a knowledge base on an object, the design of an object, and obtain comprehensive information, provided that the initial information is specified by a certain vector.

The work simulates some applied aerodynamics problems, such as mixing two gases in a pipe, throttle calculations, and selecting compressor characteristics. Mathematical models were compiled and approximate calculations were carried out for these problems. The results showed that Simulink performs well in aerodynamic calculations of heat and mass transfer processes.

The structure of a linear gas pipeline network usually includes a telescopic pipeline with several consumer connection points. The telescopic nature of the pipeline can be illustrated by the different diameters of the network with a linear structure. To calculate telescopic piping, a method called flow coefficient was developed. This method transforms the formulas used for calculation into standard formulas [7]. The work generalizes the flow coefficient method for five flow regimes [4].

The path selection method was developed to determine the number of consumer connection points [8]. Work [4] generalizes the path selection method for five hydrodynamic flow regimes.

Looped gas collection networks from fields [9] or distribution networks are used to increase the reliability of pipeline networks and reduce gas pressure loss [1,2,3]. In fact, the failure of any individual element of a linear structure network disables the entire network. Gas will move along operable arcs of the network when one section of the looped network burns out [11].

To calculate a looped pipeline network, parallel threads with jumpers are used that can transport liquid and gaseous media [2,3]. In this case, the principles of graph theory are used, as well as analogues of Kirchhoff's first and second laws. However, various formulas are used to transport liquid and gas in a looped pipeline network [2,3, 14].

Cities and engineering communications with them are gradually developing. As a result, in some cities a combined gas supply structure with ring and linear substructures is formed. The calculation of the gas supply network of many cities and states around the world has been the subject of research in the literature [10,11,12].

In addition, pipeline networks continue to develop. The transition to high operating pressures and increasing pipeline diameters is typical for main gas pipelines [5]. Analysis of a simple formula for calculating a horizontal gas pipeline [7] shows that the pressure curve presented in the coordinate plane (x, p) has the shape of a parabola, symmetrical to the x axis. The vertex of the parabola is on the x axis, and its branches continue to $x \rightarrow -\infty$. The derivative of pressure p with respect to distance x has a negative value (i.e., pressure decreases with distance). The modulus derivative of pressure decreases with distance from the vertex of the parabola. Thus, the pressure loss decreases as the inlet pressure increases. This explains the trend towards high operating pressures in the gas pipeline network.

However, the balancing term, which takes into account the resistance forces, includes the fifth power of the pipe diameter in the denominator - this is for a developed turbulence regime [7, 8, 17]. Thus, when organizing pipeline gas transportation, increasing the pipe diameter significantly reduces operating costs.

2. Formulation of The Problem

The paper examines the dynamics of gas flow in the gas network G, represented by a block (Fig. 1). By transferring heat through solid walls and overcoming resistance forces, Simulating can account for pressure and heat losses. When gas flows in a block, both stationary and non-stationary problems are considered. In this case, the compressibility of the gas and the heat capacity of the gas are taken into account. A mathematical model was compiled, formulas were formulated and numerical results were obtained to solve the problem.

The dynamics of pipe flow in a gas network are represented by a pipe block (G). The block takes into account convective heat transfer from the pipe wall and viscous friction losses. Gas is constantly in the pipe. The compressibility and heat content of the gas volume determine the pressure and temperature. When the flow speed reaches the speed of sound, the flow becomes blocked.

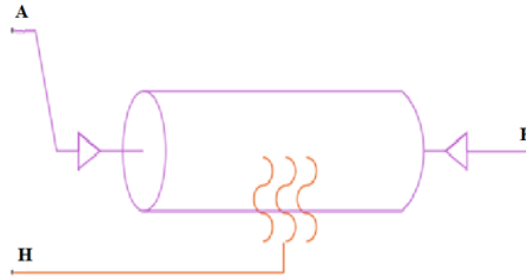


Figure1. - Dynamics of pipe flow in a gas network (G)

This block can completely inhibit the gas flow. This occurs when a G block mass flow source or a G block controlled mass flow source connected to a block (G) pipe generates a mass flow rate that exceeds the network capacity. This leads to a modeling error. Let's create formulas for modeling.

Mass Balance

Conservation of mass relates mass flow to changes in pressure and temperature inside the node representing the volume of gas [13, 15]:

$$\frac{\partial M}{\partial p} \frac{dp_I}{dt} + \frac{\partial M}{\partial T} \frac{dT_I}{dt} = m_A + m_B, \quad (1)$$

where $\frac{\partial M}{\partial p}$ is the partial derivative of the mass of the gas volume with respect to pressure at constant temperature and volume; $\frac{\partial M}{\partial T}$ – partial derivative of the mass of the gas volume with respect to temperature at constant pressure and volume; p_I – gas volume pressure; T_I – volume gas temperature; t – time; m_A and m_B are the mass flow rates at ports A and B, respectively.

The flow velocity associated with a port is positive as it flows into the block.

Perfect Energy Balance. Energy saving connects the consumption of energy and heat with the dynamics of pressure and temperature inside the unit representing the volume of gas. [13,15]:

$$\frac{\partial U}{\partial p} \frac{dp_I}{dt} + \frac{\partial U}{\partial T} \frac{dT_I}{dt} = \Phi_A + \Phi_B + Q_H, \quad (2)$$

where $\frac{\partial U}{\partial p}$ is the partial derivative of the internal energy of the gas volume with respect to pressure at constant temperature and volume; $\frac{\partial U}{\partial T}$ – partial derivative of the internal energy of the gas volume with respect to temperature at constant pressure and volume; ϕ_A and ϕ_B – energy consumption in ports A and B respectively; Q_H – heat consumption at port H.

Partial Derivatives for Ideal and Real Gas Models

The gas properties model determines the partial derivatives of the mass M and internal energy U of the gas volume with respect to pressure and temperature at constant volume. The equations for real and ideal gas are as follows [13,15]:

$$\frac{\partial M}{\partial p} = V \frac{\rho_l}{p_l}, \quad \frac{\partial M}{\partial T} = -V \frac{\rho_l}{T_l}, \quad \frac{\partial U}{\partial p} = V \left(\frac{h_l}{ZRT_l} - 1 \right), \quad \frac{\partial U}{\partial T} = V \rho_l \left(c_{pl} - \frac{h_l}{T_l} \right). \quad (3)$$

Here ρ_l – gas volume density; V – volume of gas; h_l – specific enthalpy of gas volume; Z – compressibility coefficient; R – specific gas constant; c_{pl} – specific heat capacity at constant pressure of the gas volume.

Partial Derivatives for A Real Gas Model

For a real gas model, the partial derivatives of mass M and internal energy U volume of gas in relation to pressure and temperature at constant volume are equal [13,15,18,19,21]:

$$\frac{\partial M}{\partial p} = V \frac{\rho_l}{\beta_l}, \quad \frac{\partial M}{\partial T} = -V \rho_l \alpha_l, \quad \frac{\partial U}{\partial p} = V \left(\frac{\rho_l h_l}{\beta_l} - T_l \alpha_l \right), \quad \frac{\partial U}{\partial T} = V \rho_l (c_{pl} - h_l \alpha_l); \quad (4)$$

where β is the isothermal volumetric modulus of gas volumetric compression; α is the isobaric coefficient of thermal expansion of the gas volume.

Pulse Balance

The balance of momentum in each part of the pipe shows the decrease in pressure caused by viscous friction and momentum transfer [1]:

$$p_A - p_I = \left(\frac{m_A}{S} \right)^2 \left(\frac{1}{\rho_I} - \frac{1}{\rho_A} \right) + \Delta p_{AI},$$

$$p_B - p_I = \left(\frac{m_B}{S} \right)^2 \left(\frac{1}{\rho_I} - \frac{1}{\rho_B} \right) + \Delta p_{BI}, \quad (5)$$

Where p – gas pressure in the port A, port B or internal node I, as indicated by the subscript; ρ – density at the port A, port B or internal node I, as indicated by the subscript; S is the cross-sectional area of the pipe; Δp_{AI} and Δp_{BI} – pressure loss due to viscous friction.

The conservation of energy equation allows the heat exchange occurring with the pipe wall through port H to add energy to the volume of gas represented by the internal assembly. Thus, the balance of impulses for each part of the pipe between the internal node and port A, as well as between the internal node and port B, is considered an adiabatic process. Adiabatic relations are written in the form [15,18,19]:

$$h_l + \frac{1}{2} \left(\frac{m_A}{\rho_A S} \right)^2 = h_I + \frac{1}{2} \left(\frac{m_I}{\rho_I S} \right)^2,$$

$$h_B + \frac{1}{2} \left(\frac{m_B}{\rho_B S} \right)^2 = h_I + \frac{1}{2} \left(\frac{m_I}{\rho_I S} \right)^2. \quad (6)$$

Here h – specific enthalpy in the port A port B or internal node I, as indicated by the subscript.

Pressure loss due to viscous friction Δp_{AI} and Δp_{BI} depend on the flow regime. The Reynolds numbers for each part of the pipe are defined as:

$$\begin{aligned} \text{Re}_A &= \frac{|m_A| D_h}{S \mu_I}, \\ \text{Re}_B &= \frac{|m_B| D_h}{S \mu_I}, \end{aligned} \quad (7)$$

here D_h is the hydraulic diameter of the pipe; μ_I – dynamic viscosity in the internal unit;

If the Reynolds number is less than the value **of the upper limit of the Reynolds number for laminar flow**, then the flow occurs in laminar mode. If the Reynolds number is greater than **the limit value of the lower Reynolds number for turbulent flow**, then the flow occurs in a turbulent regime.

In the laminar flow regime, the pressure loss due to viscous friction is [18,19]:

$$\begin{aligned} \Delta p_{AI_{lam}} &= f_{shape} \frac{m_A \mu_I}{2 \rho_I D_h^2 S} \frac{L + L_{eqv}}{2}, \\ \Delta p_{BI_{lam}} &= f_{shape} \frac{m_B \mu_I}{2 \rho_I D_h^2 S} \frac{L + L_{eqv}}{2} \end{aligned} \quad (8)$$

here f_{shape} is the shape factor for parameter values **viscous friction of laminar flow**; L_{eqv} – parameter value **"Cumulative equivalent length of local resistances"**.

In a turbulent flow regime, the pressure loss due to viscous friction is:

$$\begin{aligned} \Delta p_{AI_{tur}} &= f_{Darcy_A} \frac{m_A |m_A|}{2 \rho_I D_h S^2} \frac{L + L_{eqv}}{2}, \\ \Delta p_{BI_{tur}} &= f_{Darcy_B} \frac{m_B |m_B|}{2 \rho_I D_h S^2} \frac{L + L_{eqv}}{2}; \end{aligned} \quad (9)$$

Where f_{Darcy} – Darcy friction coefficient in the port **A** or **B**, as indicated by the subscript.

Darcy friction coefficients are calculated from the Haaland correlation [15,16,19]:

$$\begin{aligned} f_{Darcy_A} &= \left[-1.8 \log \left(\frac{6.9}{\text{Re}_A} + \left(\frac{\varepsilon_{rough}}{3.7 D_h} \right)^{1.11} \right) \right]^{-2}, \\ f_{Darcy_B} &= \left[-1.8 \log \left(\frac{6.9}{\text{Re}_B} + \left(\frac{\varepsilon_{rough}}{3.7 D_h} \right)^{1.11} \right) \right]^{-2}; \end{aligned} \quad (10)$$

Where ε_{rough} – parameter value **absolute roughness of the inner surface**.

A flow is in transition between laminar and turbulent regimes when the Reynolds number is between the upper and lower limits of the Reynolds number for laminar flow. Viscous friction in the transition region causes a smooth coupling between losses in laminar and turbulent flow regimes.

Convective Heat Transfer. Equation of convective heat exchange between the pipe wall and the internal volume of gas [15,19]:

$$Q_H = Q_{conv} + \frac{k_I S_{surf}}{D_h} (T_H - T_{in}); \quad (11)$$

here S_{surf} is the surface area of the pipe,

$$S_{surf} = \frac{4SL}{D_h}.$$

Assuming an exponential temperature distribution along the pipe, convective heat transfer is equal to

$$Q_{conv} = |m_{avg}| c_{p_{avg}} (T_H - T_{in}) \left(1 - \exp \left(- \frac{h_{coeff} S_{surf}}{|m_{avg}| c_{p_{avg}}} \right) \right), \quad (12)$$

here T_{in} is the inlet temperature, depending on the flow direction, $m_{avg} = (m_A - m_B) / 2$ the average mass flow rate from the port A V port IN; $c_{p_{avg}}$ – specific heat capacity calculated at average temperature.

Heat transfer coefficient h_{coeff} depends on the Nusselt number:

$$h_{coeff} = Nu \frac{k_{avg}}{D_h}, \quad (13)$$

Where k_{avg} – thermal conductivity coefficient calculated at average temperature . The Nusselt number depends on the flow regime.

The Nusselt number in laminar flow mode is constant and equal to the value of the **Nusselt number for the heat transfer parameter of laminar flow** . The Nusselt number in a turbulent flow regime is calculated from the Gnielinsky relation [6,16]:

$$Nu_{tur} = \frac{\frac{f_{Darcy}}{8} (Re_{avg} - 1000) Pr_{avg}}{1 + 12.7 \sqrt{\frac{f_{Darcy}}{8} (Pr_{avg}^{2/3} - 1)}} \quad (14)$$

Where Pr_{avg} – Prandtl number calculated at average temperature.

Reynolds number is

$$Re_{avg} = \frac{|m_{avg}| D_h}{S \mu_{avg}}, \quad (15)$$

Where μ_{avg} – dynamic viscosity , rated at medium temperature.

The Nusselt number is determined by smoothly transitioning between laminar and turbulent flow values when the average Reynolds number is between the upper and lower limit parameter values.

Blocking The Flow. Mass flow when throttling from pipe at ports A And B is (1):

$$m_{A_{choked}} = \rho_A a_A S, \quad m_{B_{choked}} = \rho_B a_B S; \quad (16)$$

Where a_A And a_B – speed of sound in ports A And B respectively.

Port open pressure A or B is the value of the corresponding Across variable in this port:

$$p_{A_{unchoked}} = Ap, \quad p_{B_{unchoked}} = Bp. \quad (17)$$

Pressures with throttling at ports A And B are obtained by substituting mass flow rates with throttling into the momentum balance equations for the pipe (1, 7):

$$\begin{aligned} p_{A_{choked}} &= p_1 + \left(\frac{m_{A_{choked}}}{S} \right)^2 \left(\frac{1}{\rho_I} - \frac{1}{\rho_A} \right) + \Delta p_{AI_{choked}}, \\ p_{B_{choked}} &= p_1 + \left(\frac{m_{B_{choked}}}{S} \right)^2 \left(\frac{1}{\rho_I} - \frac{1}{\rho_B} \right) + \Delta p_{BI_{choked}}. \end{aligned} \quad (18)$$

Here $\Delta p_{AI_{choked}}$ they $\Delta p_{BI_{choked}}$ represent the pressure loss due to viscous friction, assuming that throttling has occurred. They are calculated in a similar way Δp_{AI} and Δp_{BI} with the replacement of the mass flow values at the ports A And B values of mass flow in the fittings.

Depending on whether closure has occurred, the block assigns the closed or open pressure value as the actual port pressure. Blockage can occur at the outlet of a pipe, but not at the entrance to the pipe. Therefore, if $p_{A_{unchoked}} \geq p_I$, then the port A is the input and $p_{A_{unchoked}} = p_I$.

If $p_{A_{unchoked}} < p_I$, then the port **A** is the exit and

$$p_A = \begin{cases} p_{A_{unchoked}}, p_{A_{unchoked}} \geq p_{A_{choked}} \\ p_{A_{choked}}, p_{A_{unchoked}} < p_{A_{choked}} \end{cases} \quad (19)$$

In the same way, if $p_{B_{unchoked}} \geq p_I$, then the port **B** is the input and $p_B = p_{B_{unchoked}}$. If $p_{B_{unchoked}} < p_I$, then the port **B** is the exit and

$$p_B = \begin{cases} p_{B_{unchoked}}, p_{B_{unchoked}} \geq p_{B_{choked}} \\ p_{B_{choked}}, p_{B_{unchoked}} < p_{B_{choked}} \end{cases} \quad (20)$$

Model Assumptions and Limitations.

The pipe wall is absolutely rigid.

The flow is fully developed.

Friction and heat transfer losses do not include input effects. The influence of gravity is negligible.

The inertia of the liquid is negligible.

This block does not model supersonic flow.

Pneumatic Drive Chain

This example shows how gas components from the Foundation library can be used to model a controlled pneumatic actuator. Variable local restriction (G) blocks are used in the masked directional valve subsystem to simulate the opening and closing of flow paths. The dual-acting subsystem consists of translational mechanical converter units (G) designed to simulate the interface between the gas network and the mechanical transmission network (Fig.2).

3. Results

During the simulation, the valve spool moves to maximum positive displacement, which results in maximum actuator travel. The valve spool is then centered, holding the weight. Subsequently, as shown in Fig.3, the valve spool moves to the maximum negative displacement. As a result, the actuator is retracted to the minimum stroke.

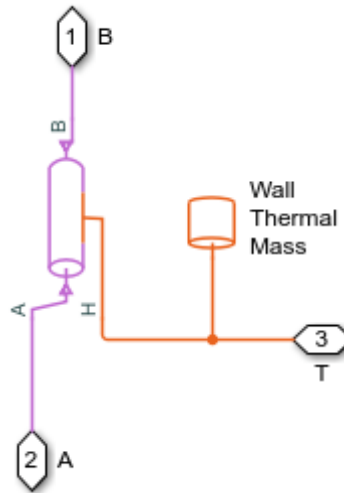


Figure-2. Feed pipe subsystem

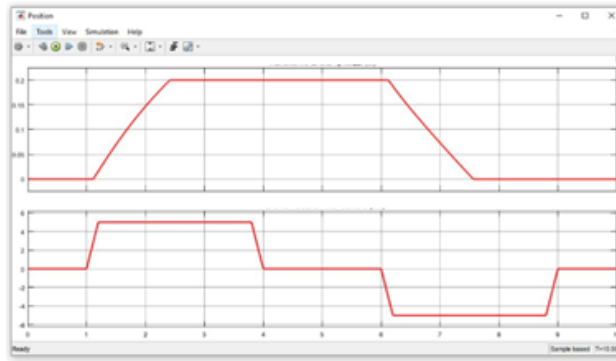


Figure-3. - Results of modeling the position of the actuator rod and valve spool using oscilloscopes

The load is then held by the centering of the valve spool. In Figure 4 shows how thermal convection occurs between the pipes and the environment, allowing the circuit to dissipate heat from the operation of the pressure source.

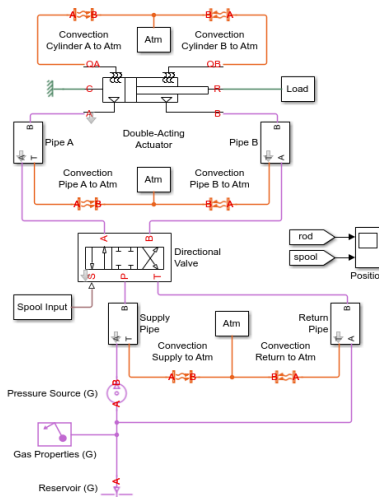


Figure 4. - Model built by Simulink for the presented object

4. Simscape Simulation Results Logging.

When the valve spool position is positive, port PA and port TB open, allowing gas to flow from port P to port A and from port B to port T. When the valve spool position is negative, port PB and port TA open, allowing gas to flow from port P to port B and from port A to port T [13,15].

The length of the pipe in the direction of flow is 5 m, the internal area of the pipe is perpendicular to the direction of flow 0.01 m^2 ; the diameter of an equivalent cylindrical pipe with the same cross-sectional area is 0.1 m.

Total length of all local resistances in a pipe Local resistances include anchors, fittings, bends and pipe inlets and sockets. Local resistances increase the effective length of the pipe section. For friction calculations only, this length is added to the geometric pipe length parameter. Only the geometric pipe length of 0.1 m determines the gas volume.

The average depth of all surface defects on the inner surface of the pipe (equivalent roughness), which affects the pressure loss in a turbulent flow mode, is $15 \cdot 10^{-6} \text{ m}$.

Within the Reynolds number, the flow begins to change from laminar to turbulent. This number is equal to the maximum Reynolds number in 2000, which corresponded to fully developed laminar flow.

Below the Reynolds number, the flow begins to transition from a turbulent regime to a transient regime. This amounts to the minimum number of Reynolds corresponding to fully developed turbulent flow, 4000.

A dimensionless factor that takes into account the influence of pipe geometry on viscous friction losses in laminar flow conditions. Standard values include 64 for a circular cross-section, 57 for a square cross-section, 62 for an aspect ratio 2 rectangular cross-section, and 96 for a thin ring cross-section.

Dynamics of convective and conductive heat transfer in laminar flow regime The magnitude depends on the geometry of the pipe cross-section and the thermal boundary conditions of the wall, such as constant temperature and constant heat flow. At a constant wall temperature, the average value for a circular cross section is 3.66 [13,15,20]. In Fig. Figure 5 shows the numerical results of the problem, which show changes in gas temperature and pressure in blocks A and B.

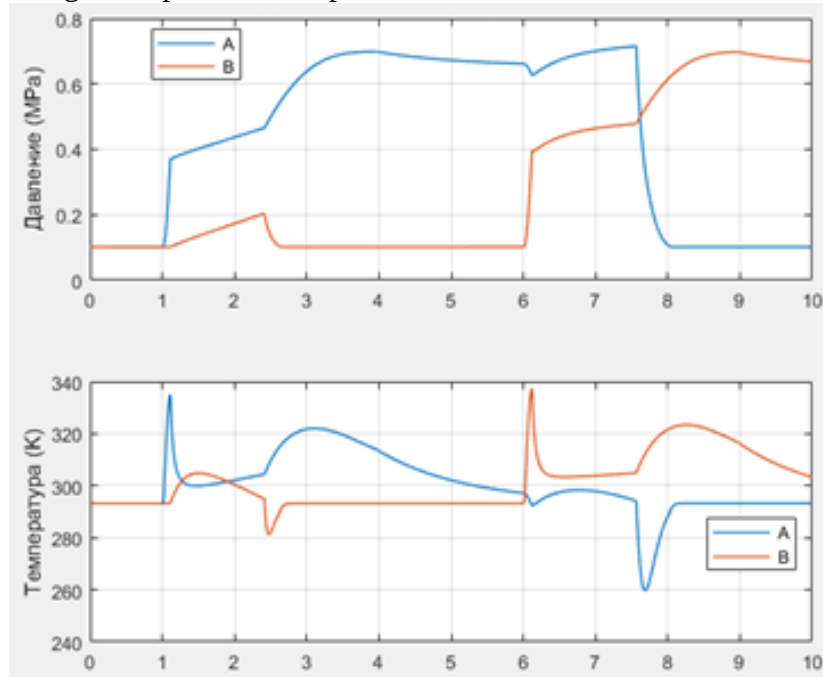


Figure-5. - Simulation results

Shifts in temperature and pressure are associated with the transition from one section of the network to another. Chapter Conclusions

5. Conclusion

Simulink, a graphical extension of MATLAB, is well suited for the calculation of gas and pipeline networks because it offers the designer, library, and calculation algorithms needed for a wide range of problems.

The structure and functioning components of Simulink have been studied. Three practice problems were considered and solved to determine the capabilities of the software system.

The problem of heat and mass transfer in a gas network is solved taking into account the input of various gases and the output of the mixture relative to the pressure and temperature of the real gas under different flow regimes (turbulent and laminar), as well as convective heat transfer.

By highlighting options for blocking the flow, the throttling problem is solved within the capabilities of Simulink. The results of this problem can be used to select control pressure, gas flow and tap.

Pumps and compressors used in hydraulic power plants provide a certain output pressure, operating within a certain range of input pressure and mass flow.

For engineering practice, the choice of a compressor that provides a continuous supply of air under the conditions of the law of change in input flow rate is of interest. A model was created for this problem and smooth results were obtained when specifying a sinusoidal input flow rate.

The results showed that the use of Simulink significantly simplifies the work of design engineers and gas network calculations.

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THERMAL CALCULATION OF A MODERN MULTI-SECTION HEAT EXCHANGER

Shirinov Z.Z.¹

¹*Institute of Mechanics and Seismic Stability of Structures, Tashkent, Uzbekistan*
E-mail: ziyomatshirinov@umail.uz

Abstract: A mathematical model and algorithm for calculating a heat exchanger consisting of similar sections and connected to a two-line heating network are proposed. The upper and lower, as well as vertical heat exchange pipes are presented with their own lengths and internal diameters. The outer diameters of the nozzles are presented taking into account the ribs. For known values of arc flow rates, input temperature and ambient temperature, the path heat loss is described by Shukhov's formula. When flows merge in the lower nodes, the coolant temperature is defined as the average flow rate. The results of calculations for various numbers of sections in the heat exchanger are given.

Keywords: multi-section heat exchanger, coolant flow, nodal temperature, Shukhov's formula, vertical, horizontal pipes, total heat loss.

1. Introduction

The growth of industrial and housing construction, improvement of amenities and improvement of sanitary and living conditions of the population lead to a significant increase in the volume of consumption of water, gas and electricity for economic and productive needs. These needs are met through the construction of utility networks, as well as through the modernization of existing systems [1-9]. The latter option contributes to a significant reduction in capital investment and requires a search for system reserves that make it possible to provide the population and infrastructure of the economy with the necessary product [10].

One of these reserves is to improve the management of technological processes for the supply and distribution of the target product (water, gas and electricity) based on the use of modern methods of mathematical and computer modeling. At the same time, the main attention must be paid to the reliability and controllability of utility networks in order to guarantee a reserve of their capacity and the possibility of their prompt reconstruction. Computer modeling and automation must accompany the entire life cycle of engineering networks: the design stage, work monitoring (operational management) and reconstruction [11-15].

Engineering networks belong to the class of continuously evolving systems, the development of which occurs in time and space. When operational management capabilities are exhausted, the reconstruction mechanism is activated. Here, the main role is played by capacity capabilities and consumption volume, as well as controlled and uncontrollable internal and external factors.

In terms of their structure, utility networks can be linear, looped or combined. The widespread use of a looped or multi-ring network is explained by the fact that the flow in it, observing the laws of nature, self-generates with minimal energy consumption. Another advantage of multi-circuit networks is to ensure the reliability of functions in the event of failure of certain arcs of the utility network [13-16]. These failures are corrected by increasing the consumption of target products along other routes (arcs). And in networks with a linear structure, the failure of a certain arc leads to the failure of the entire system.

As an example of the positive features of the ring organization, let us cite the presence of parallel threads of main gas pipelines. They communicate with each other through jumpers, which are installed at certain distances [17-19].

If the parallel threads have different diameters, the bridges contribute to the same pressure loss along the parallel threads. If a certain section of the mainline fails, then disconnecting this section leads to a redistribution of the flow along the existing individual parallel threads. Those capital investments in installing jumpers are justified by ensuring the reliability of the utility network.

Let's introduce the main consumers of the theory of flow distribution. In addition to the gas transportation system, this can include water supply networks, oil and petroleum product pipelines, pneumatic drives and hydraulic drives [20].

The provisions of the theory of flow distribution are also successfully implemented when modeling space heating, heat supply for a microdistrict or an entire city, as well as traffic flow on a road network.

Below we propose an algorithm for thermal calculation of a separate heat exchanger with a limited number of sections, which is connected to a two-strand heat supply network.

The basics of the theory of heat transfer and its types are presented in works [1-6]. Variants of the heat transfer equation are proposed for normal conditions, when the heat flow is proportional to the temperature gradient and the equation is of parabolic type, and for extreme conditions that occur under the influence of powerful sources: chemical laser, electric or gas welding and are described by hyperbolic type equations taking into account the relaxation properties of the process. Analytical solutions to heat transfer problems have been obtained for various variants of boundary conditions in various orthogonal spatial coordinates.

A special place in thermal calculations is occupied by the problems of the heat transfer process from flowing coolants, which can represent free or forced convection [21-25]. Unfortunately, taking into account convective heat transfer requires turning to a system of nonlinear Navier-Stokes equations, which are practically not amenable to analytical solution, and existing solutions differ significantly from the phenomenon being studied, at least in that in nature the flow is turbulent. In this regard, an engineering version of the calculation has been developed - a quasi-one-dimensional approach to describing the hydrodynamics of the flow, accompanied by the heat transfer process [20-23].

A frequently used result of this engineering approach is the Shukhov formula [26]. It was obtained taking into account the average value of the heat transfer coefficient in the coolant-pipeline-thermal insulation-environment system and the average value of the outer diameter of the pipeline. In literary sources one can also find modifications of the Shukhov formula [26].

Along with Shukhov's formula, when developing an algorithm for calculating the heat exchanger, a formula was used that determines the average flow temperature of the currents combining into one flow.

Some numerical results are presented that demonstrate the features of the proposed mathematical model and calculation algorithm.

2. Statement and mathematical model of the problem.

In Fig. 1 schematically shows a heat exchanger with $M + 1$ identical sections, which is connected to the upper and lower lines through jumpers. The upper jumper is equipped with a valve-regulator. Considering the input temperature, ambient temperature, geometric parameters and arc flow rates of the coolant by volume as given, it is necessary to determine the outlet temperature of the coolant.

The object is represented as a system consisting of finned tubes connected in series and parallel (Fig. 1). For the calculation, we need geometric quantities (mainly arc flow rates) and thermophysical indicators of the system.

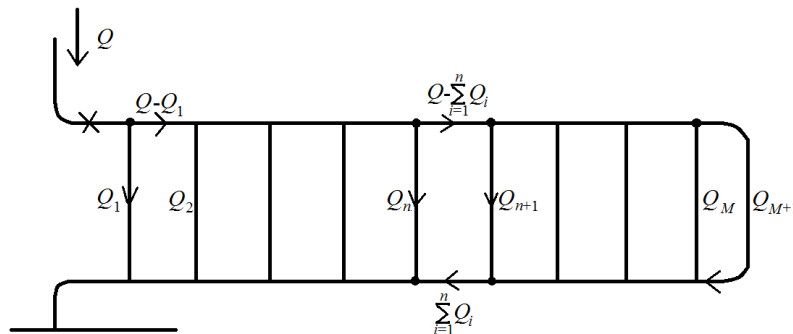


Figure 1. - Schematic representation of the heat exchanger, connected to a two-pipe network on one side

The upper and lower left parts of the figure attach the input and output jumpers connecting the heat exchanger with the upper (coolant distributing) and lower (coolant collecting) lines.

The input jumper, taking into account local resistance, has a length of $\dot{l}_p$, and the output jumper has a length of l_p . The leveling height difference in them is $\dot{h}_p$ and h_p . The internal diameters are $\dot{D}_p$ and D_p , and the cross-sectional areas are $\dot{f}_p$ and f_p . Because the thickness of their walls, taking into account the ribs, is $\dot{\delta}_p$ and δ_p , then the outer diameters are $\dot{D}_p + 2\dot{\delta}_p$ and $D_p + 2\delta_p$.

New sections can be added to the heat exchanger or the number of sections can be reduced. In general, we believe that the number of sections is $M+1$. The height of the sections is h_c , length l_c , internal diameter of vertical pipes D_v , and horizontal ones D_g , averaging of wall thickness taking into account the ribs - δ_v and δ_g . Depending on these data, the cross-sectional areas f_v and are determined f_g , as well as the outer diameters of the vertical and horizontal pipes of the heat exchanger.

From the hydrodynamic indicators we need: input (and output) flow, Q arc (or section) flow of the upper q_i and lower $q'_i = q_i$, as well as vertical Q_i pipes. We consider that they are defined in $\mathcal{M}^3 / c$.

The thermophysical indicators of the system consist of the heat capacity c_p of the working fluid, the average heat transfer value k_{cp} of the coolant elements, the temperature at the inlet to the heat exchanger is T_H , and at the outlet T_K , ρ , Q is the mass flow rate of the coolant. The coolant temperature changes smoothly along the length of the arcs, which is described by Shukhov's formula [26]:

$$T(x) = T_{oc} + (T_H - T_{oc}) \exp\left(-\frac{kD_{op}}{c_p \rho Q} x\right).$$

An abrupt change in temperature is observed at the points of confluence of two flows with flow rates Q_1 and Q_2 and temperature T_1 and T_2 . Then the average flow temperature of the coolant after merging is determined by the formula

$$T_\kappa = \frac{T_1 Q_1 + T_2 Q_2}{Q_1 + Q_2}.$$

The total loss of thermal energy in the heat exchanger is calculated by the formula:

$$Q_{n.m} = \rho Q c_p (T_H - T_\kappa).$$

3. Methods

Let us number the nodes of the upper arc through 1, 2, 3, ..., M , $M+1$, and the nodes of the lower arc through 1', 2', 3' ..., M' , $(M+1)'$. First, we determine the nodal temperatures in the upper arc ($T_1, T_2, \dots, T_M$), then - in the lower nodes of the vertical pipes (for them, the input data are the nodal temperatures of the upper arc), then - the temperatures in the lower arcs and at the end - the output temperature T_K .

Node 1. Calculation of the temperature of the working agent at the end of the input jumper. In it, the water flow rate is Q , the average diameter of the outer surface $\dot{D}_p + 2\dot{\delta}_p$, and the inlet temperature T_H .

Let us determine the value of the Shukhov coefficient for the input jumper:

$$Sh_p = \frac{k_{cp} \pi (\dot{D}_p + 2\dot{\delta}_p)}{\rho Q c_B}.$$

Then in node 1 the temperature of the heated water is

$$T_1 = T_{oc} + (T_H - T_{oc}) \exp(-Sh_p \dot{l}_p),$$

where $\dot{l}_p$ is the total length of the input jumper without taking into account local resistances.

Node 2 . The arc consumption 12 is q_1 Due to this

$$T_2 = T_{oc} + (T_1 - T_{oc}) \exp(-Sh_g l_c),$$

where l_c is the length of one section; here and further

$$Sh_g = \frac{k_{cp} \pi (D_g + 2\delta_g)}{\rho q_1 c_B}.$$

Nodes 3..M +1 . At these we use a similar recurrent formula for temperature:

$$T_m = T_{oc} + (T_{m-1} - T_{oc}) \exp(-Sh_g l_g).$$

To calculate **the temperature** of heated water **at the lower ends of vertical pipes** 1, 2, ..., m, ..., (M +1) a single formula is used

$$\bar{T}_{m'} = T_{oc} + (T_m - T_{oc}) \exp(-Sh_{vm} h_c),$$

where h_c is the height of the section (the distance between the axes of the upper and lower nozzles);

$$Sh_{vm} = \frac{k_{cp} \pi (D_v + 2\delta_v)}{\rho Q_m c_B}.$$

But here different flow rates Q_m through the vertical pipes are used.

Let's move on to calculating the nodal temperatures in the lower arcs. Note that in links $M(M +1)$

, $(M +1)(M +1)'$, $(M +1)'M'$ the coolant flow rate is the same: $Q_{M+1} = q_{(M+1)'} = q_{M+1}$.

Heated water flows Q_{M+1} to the node on the right with flow rate M' and temperature

$$\dot{T}_{M'} = T_{oc} + \left(T_{(M+1)'} - T_{oc} \right) \exp(-Sh_{v(M+1)'} h_g).$$

Here and below we used the notation

$$Sh_{vm'} = \frac{k_{cp} \pi (D_g + 2\delta_g)}{\rho q_{m'} c_B}.$$

Due to the merging of flows with flow rates Q_M , Q_{M+1} the average temperature of the heated agent at the beginning of the arc $M'(M -1)'$ is

$$\dot{T}_{M'} = \frac{Q_M \bar{T}_{M'} + Q_{M+1} \dot{T}_{M'}}{Q_M + Q_{M+1}} = \frac{Q_M \bar{T}_{M'} + Q_{M+1} \dot{T}_{M'}}{q_{M'}}.$$

A similar formula is obtained for the nodal temperatures of water at the nodes $m = (M -1)', ..., 1'$:

$$T_{m'} = \frac{Q_m \bar{T}_{m'} + q_{m'} \dot{T}_{m'}}{q_{m-1}},$$

to use which you first need to find $\bar{T}_{m'}$.

In particular, at the outlet of the node $1'$ the average water temperature is

$$T_{1'} = \frac{Q_1 \bar{T}_{1'} + q_{1'} \dot{T}_{1'}}{Q}.$$

It remains to determine the temperature of the water that enters the “return” - the lower line:

$$T_K = T_{oc} + (T_{1'} - T_{oc}) \exp(-Sh_p l_p),$$

where l_p is the length of the output jumper;

$$Sh_p = \frac{k_{cp} \pi (D_p + 2\delta_p)}{\rho Q c_B}.$$

The presented algorithm is easily implemented in the form of a calculation program where known local costs are used.

The main interest is the nodal temperatures of the water and, especially, the temperature at the outlet of the heat exchanger.

As an example, below we present the values of nodal temperatures for a heat exchanger with ten sections (Table 1), obtained at $T_H = 343.15 \text{ K}$ and $T_{oc} = 340.76 \text{ K}$.

The nature of the temperature change of the working agent along the upper arc is decreasing. But here a feature of Shukhov's formula manifests itself: with a decrease in liquid flow, a change in the temperature of the heated water becomes noticeable. Also, noticeable changes in water temperature are observed along the vertical pipes.

Despite the fact that the fluid velocity is higher than in horizontal pipes, large values of the outer surface per linear meter and their length led to this result. As a result, in the lower extreme node of the heat exchanger, the coolant temperature takes on its lowest value. In the lower arc, in the direction of flow, the temperature increases almost uniformly, which is explained by the merging of flows from the vertical pipes to the flow of the lower arc.

Table 1.

Values of nodal temperatures (K) in a heat exchanger with ten sections

Nodal temperature (K)	Arc numbers									
	1	2	3	4	5	6	7	8	9	10
Upper arc	343.12	343.08	343.03	342.97	342.9	342.82	342.72	342.58	342.38	341.97
At the lower end of the vertical pipe	341.33	341.23	341.13	341.04	340.94	340.84	340.72	340.58	340.38	339.99
Lower arcs	340.76	340.7	340.64	340.58	340.51	340.44	340.36	340.27	340.15	339.99

Below we present some of the comparison results in the form of graphs. Here and further in the graphs, numbers 1, 2, 5, 6 correspond to the speed of the total flow in the heat exchanger $w = 0.3 \text{ m s}^{-1}$, numbers 3, 4, 7, 8 – speed 0.4 m s^{-1} ; numbers 1, 3, 5, 7 – ambient temperature $T_{oc} = 293.15 \text{ K}$, and numbers 2, 4, 6, 8 – 298.15 K ; numbers 1-4 heat transfer coefficient $k_{cp} = 7.5 \text{ Bm m}^{-2} \text{ s}^{-1}$, and numbers 5-8 – $17.5 \text{ Bm m}^{-2} \text{ s}^{-1}$.

The following three figures provide a summary of the eight calculation options.

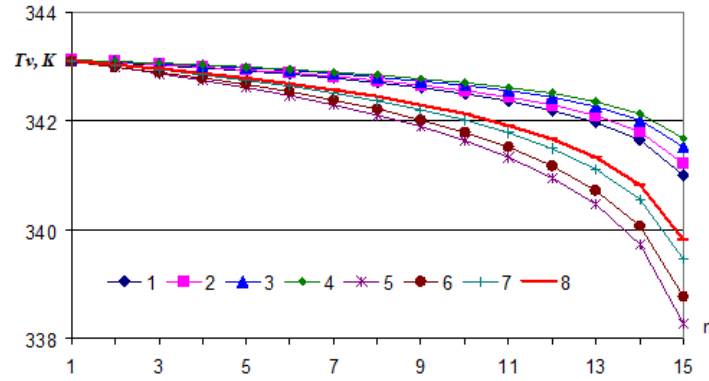


Figure 2. - Coolant temperature values in the nodes of the upper arc of a heat exchanger with 15 sections

The temperature value at the nodes of the upper arc (Fig. 2) serves as input (boundary) conditions in calculating the temperature difference of the coolant through the vertical pipes. The coolant temperature values at the lower ends of the vertical pipes depending on the number of heat exchanger sections are presented in Fig. 3. The more intense temperature drop at the end of the heat exchanger is also explained by the lower coolant flow rate in the last sections.

A more uniform increase in coolant temperature in the direction of flow is observed in the nodes of the lower arc (Fig. 4). The temperature values $n=15$ for the considered options repeat the results in Fig. 10. $n=14$ The coolant enters the unit both from the left and from the top. At the outlet of the unit, $n=14$ the coolant temperature represents the average flow rate of these flows. Etc. The temperature value at $n=1$ represents the final temperature of the coolant in a heat exchanger with 15 sections.

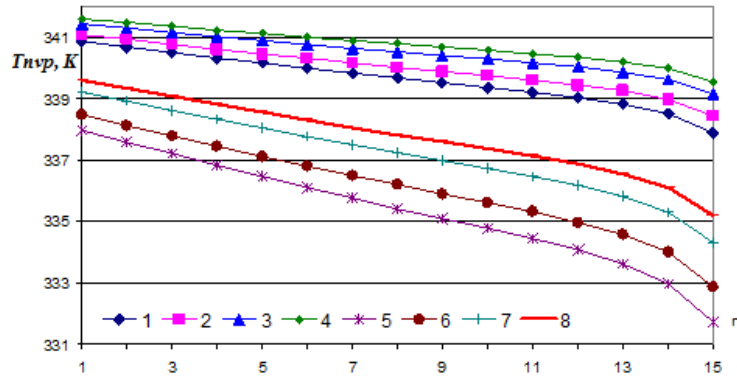


Figure 3. - Values of the coolant temperature at the approach to the lower end of the vertical pipes of the heat exchanger with 15 sections (for designations, see the text and Fig. 2)

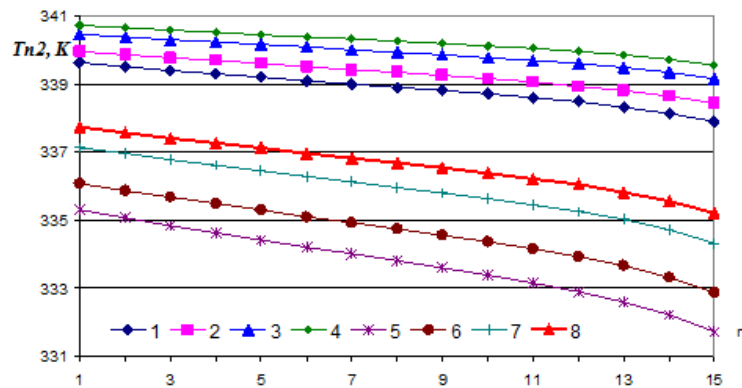


Figure 4. - Values of nodal temperatures in the lower arc of the heat exchanger

In Fig. 3.5 shows changes in the coolant outlet temperature depending on the number of heat exchanger sections for options 1-8. They indicate that:

a) with an increase in the value of the average heat transfer coefficient, the heat removal process intensifies (this can also be achieved by increasing the area of contact with the coolant, heat exchanger and air);

b) at lower flow rates, an increase in heat removal is also observed;

c) the heating process is self-regulating: as the ambient temperature increases, the heat transfer of the coolant decreases.

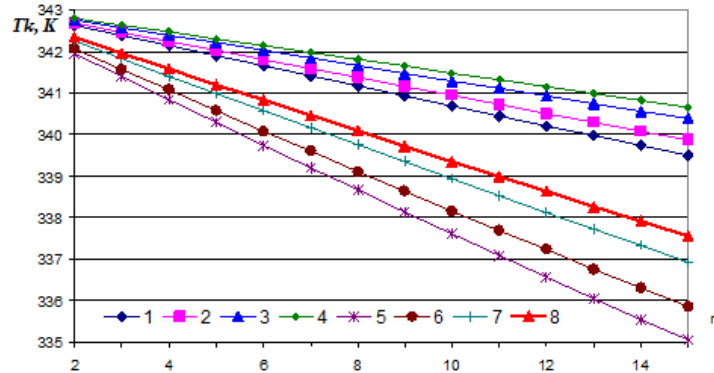


Figure 5. - Temperature values of the coolant at the outlet of the heat exchanger depending on the number of sections in the heat exchanger

These judgments are also confirmed by the graphs in Fig. 6, which presents the results of a computational experiment on the total heat loss through a heat exchanger with n sections for the eight options of initial data listed above.

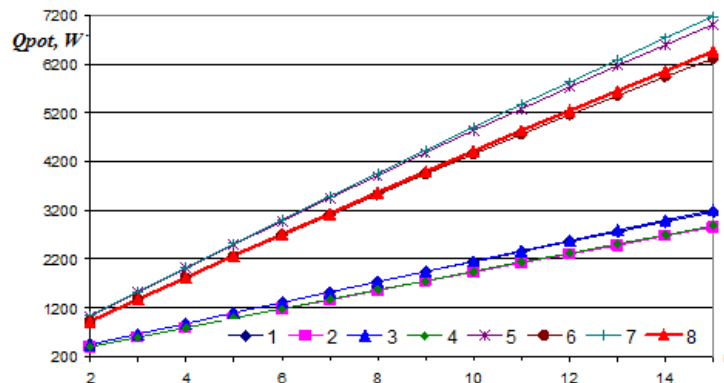


Figure 6. - Total heat loss for eight variants of a computational experiment depending on the number of sections in the heat exchanger

4. Conclusion

Summarizing the results of the work, we note that an algorithm and program for thermal calculation of a multi-section heat exchanger have been developed. Using the developed program, a computational experiment was carried out for the quadratic resistance mode for various values of the number of heat exchanger sections and arc flow rates, nodal temperatures and pressures were determined depending on numerous initial parameters of the object.

It was revealed that the costs through the vertical radiator pipes are almost uniform: with an increase in the number of sections, as well as with distance from the entrance to the heat exchanger, the costs through the vertical pipes decrease.

The lowest temperature in the heat exchanger is formed in the lower part of the far section.

The developed algorithm for calculating a separate heat exchanger can be used when developing an algorithm for calculating a two-line heating network.

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CONVENIENCE OF A GENERALIZED MODEL FOR STUDYING UNSTATIONARY FLOW OF ELASTIC-VISCOUS FLUID IN A FLAT CHANNEL

Shukurov Z.K.¹, Navruzov K.², Kodirova M.A.¹

¹Kattakurgan branch of Samarkand State University, Kattakurgan, Uzbekistan

²Urgench State University, Urgench, Uzbekistan

E-mail: shukurovzoxid742@gmail.com

Abstract. This paper examines the development of analytical methods and the solution of practical problems associated with determining the unsteady flow of elastic-viscous liquids in a flat channel using their rheological models. It is known that in most cases, the classical Maxwell model in one-dimensional space is used for the flow of an elastic viscous fluid. Using the Shulman-Husid topological model, problems solved by the Maxwell model, which determines the processes of elastic-rheological viscous fluids, are studied. Using the Shulman-Husid model to study the behavior of unsteady flows of elastic viscous fluids in a flat channel, we show that this model is a general case of the Newton and Maxwell models.

Keywords: Flat channel, elastic-viscous fluid, stationary fluid, non-stationary fluid, one-dimensional model, generalized model, spectrum, velocity.

1. Introduction.

The mathematical model of viscoelastic liquids is mainly based on taking into account changes in the elasticity of the liquid and is generalized in the form of the Maxwell model [1, 2, 7, 17, 18, 20]. In it, cases of relaxation and integral types are determined based on the conditions of mathematical equivalence of nonlinear rheological equations.

The flow of elastic viscous fluid in channels with flat and round sections was studied in the work of Z.P. Shulman, B.M. Khusid [2].

The problems of laminar unsteady and stationary flows of viscous Newtonian fluids as a result of changes in the pressure gradient over time were considered in studies [3, 4, 5, 6, 8], and flows in a turbulent regime - in works [13, 15, 18].

Consideration of the motion of a Newtonian fluid as a multiphase medium allows one to approximate the experimental results of theoretical studies in this area. Research [5, 6, 12] is presented based on perfect and practical applications.

However, despite many studies carried out on the behavior of non-Newtonian fluids, problems using the Shulman-Husid model are currently not sufficiently studied and applied. In particular, hydrodynamic changes that occur during the flow of viscoelastic fluids, during unsteady flow of a sharp

increase in the pressure gradient and a sharp decrease to zero, have not been studied in detail [9, 10, 11, 14, 16, 19, 20].

In this paper, we study the problem of unsteady flow of a viscoelastic fluid in a long flat channel using the Shulman-Husid model.

2. Materials, methods and object of study.

Let us present the Shulman-Husid model, which generalizes the rheological models of all elastic-viscous liquids in the form of the Maxwell model in the form of a topological model:

$$\begin{aligned} T &= \sum_{k=1}^{\infty} \left(1 + \frac{\varepsilon}{2} \right)' T_k^{(1)} + \frac{\varepsilon}{2} T_k^{(2)}, \quad T_k^{(1)} + \frac{g_k}{\lambda_k} T_k^{(1)} = 2p_k \Omega, \\ T_k^{(2)} + \frac{g_k}{\lambda_k} T_k^{(2)} &= -2p_k \Omega, \quad \frac{Dp_k}{Dt} + \frac{g_k}{\lambda_k} p_k = \frac{\eta_k}{\lambda_k^2} f_k. \end{aligned} \quad (1)$$

Here the upper and lower convective derivatives are defined as:

$$T_k^{(1)} = \frac{DT_k^{(1)}}{Dt} - T_k^{(1)} \nabla V^T - \nabla V \cdot T_k^{(1)}, \quad T_k^{(2)} = \frac{DT_k^{(2)}}{Dt} + T_k^{(2)} \nabla V + \nabla V^T \cdot T_k^{(2)}.$$

And the Jaumann derivative is given in the following form: $\frac{DA}{Dt} = \frac{\partial A}{\partial t} + V \nabla A + W A - A W$, here

$\nabla V = \Omega + W$, $\Omega = \frac{1}{2}(\nabla V^T + \nabla V)$, $W = \frac{1}{2}(\nabla V - \nabla V^T)$ existence, Ω – strain rate tensor; ∇V –

deformation rate gradient; ∇V^T – transposed strain rate gradient; is entered as the parameter ε , representing the difference of non-zero second normal stresses, which is determined by the following

formula $\frac{\varepsilon}{2} = \frac{\Psi_2}{\Psi_1}$. Here $\Psi_1 = (\sigma_{11} - \sigma_{22}) / \gamma^2$, $\Psi_2 = (\sigma_{22} - \sigma_{33}) / \gamma^2$ are functions, respectively, of the

difference between the first and second, second and third voltages; p_k is a parameter that can be determined from the equation itself; $T_k^{(1)}$, $T_k^{(2)}$ – deformation tensors, consisting of nine elements for each;

$T_k^{(1)}$, $T_k^{(2)}$ are stress tensors specified in an arbitrarily chosen coordinate system, and the derivatives defined for them denote the upper and lower convective derivatives, respectively [18-20].

Let us present the Shulman-Husid model as a modified model in this form [10, 11]:

$$\begin{aligned} T &= \sum_{k=1}^{\infty} T_k, \quad \lambda_k \frac{\partial T_k}{\partial t} + T_k = 2\eta_k \Omega, \\ \Omega &= \frac{1}{2} \frac{\partial u}{\partial y}; \end{aligned} \quad (2)$$

From this model we give origin to the models of Newton and Maxwell. In the case when the relaxation λ coefficient tends to zero, the Newtonian model is derived:

$$T = \eta \frac{\partial u}{\partial y} \quad (3)$$

When the number α , characterizing the distribution of the relaxation time spectrum, tends to infinity, the Maxwell model is derived:

$$\tau + \lambda \dot{\tau} = \eta \frac{\partial u}{\partial y} \quad (4)$$

Using these three models, we study the process of transition of an elastic-viscous fluid from an unsteady state to a stationary state.

3. Statement of the problem.

Denoting the distance between the flat walls of the channel as $2h$, and assuming that the channel is sufficiently long, we choose a characteristic relative length for the flow of a viscoelastic viscous fluid. In this case, the ratio of the channel width to the length along the longitudinal axis is considered an infinitesimal value. Therefore, transverse velocities do not appear in currents, and the x axis is taken in the horizontal direction along the length of the channel, and the y axis is taken in the vertical direction, perpendicular to the x axis. In such cases, the movement of liquid in straight channels, taking into account its rheological properties, is expressed by a system of simplified equations in the following form:

$$\begin{cases} \frac{\partial u}{\partial t} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{1}{\rho} \frac{\partial \tau}{\partial y}, & \frac{\partial p}{\partial y} = 0, \\ \tau = \sum_{k=1}^{\infty} \tau_k, & \lambda_k \frac{\partial \tau_k}{\partial t} + \tau_k = \eta_k \frac{\partial u}{\partial y}. \end{cases} \quad (5)$$

Here $\lambda_k = \frac{\lambda}{k^\alpha}$, $\eta_k = \frac{\eta}{\xi(\alpha)k^\alpha}$.

The initial and boundary conditions are introduced as follows:

$$\text{if } t=0 \text{ is equal to } u=0, \quad \frac{\partial p}{\partial x} = 0. \quad (6)$$

$$\text{if } t \geq 0 \text{ and } y=0 \text{ are equal to } \frac{\partial u}{\partial y} = 0; \quad (7)$$

$$\text{if } t \geq 0 \text{ and } y=h \text{ are equal to } u=0; \quad \frac{\partial p}{\partial x} \neq 0$$

Taking into account the initial conditions (6) and applying the Laplace-Carson substitution,

$$\bar{u} = s \int_0^\infty e^{-st} u dt, \quad \bar{\tau} = s \int_0^\infty e^{-st} \tau_{21} dt \quad (8)$$

where s is the substitution parameter to the system of linearized equations (5).

$$\begin{aligned} \frac{\partial u}{\partial t} &\rightarrow s\bar{u}, & \frac{\partial u}{\partial y} &\rightarrow \frac{d\bar{u}}{dy} \\ \frac{\partial \tau}{\partial t} &\rightarrow s\bar{\tau}, & \frac{\partial p}{\partial x} &\rightarrow \frac{d\bar{p}}{dx} \end{aligned}$$

As a result, we get the following system of equations:

$$\begin{cases} \rho s \bar{u} = -\frac{d\bar{p}}{dx} + \frac{d}{dy}(\bar{\tau}), \\ \bar{\tau} = \sum_{k=1}^{\infty} \bar{\tau}_k, & s\bar{\tau}_k + \frac{1}{\lambda_k} \bar{\tau}_k = \frac{\eta_k}{\lambda_k} \frac{d\bar{u}}{dy}, \end{cases} \quad (9).$$

In this case, the boundary conditions take the following form:

$$\text{if } y=0 \text{ is equal to } \frac{d\bar{u}}{dy} = 0; \text{ if } y=h \text{ is equal to } \bar{u} = 0 \quad (10)$$

After some calculations, let's create this equation

$$\frac{d^2 \bar{u}}{dy^2} - \frac{\rho s}{\eta_k(s)} \bar{u} = \frac{1}{\bar{\eta}_k(s)} \frac{d\bar{p}}{dx}. \quad (11)$$

Taking into account the generated solutions, we determine the exact solution of equation (11) as follows

$$\bar{u}(y, s) = \frac{1}{\rho s} \left(-\frac{dp}{dx} \right) \left(1 - \frac{\cos \left(i \sqrt{\frac{\rho s}{\bar{\eta}_k(s)}} y \right)}{\cos \left(i \sqrt{\frac{\rho s}{\bar{\eta}_k(s)}} h \right)} \right) \quad (12)$$

Substituting this solution into the Laplace-Carson originalization formula, we form the following integral:

$$u(y, t) = \frac{1}{2\pi i} \int_{\sigma-i\infty}^{\sigma+i\infty} e^{st} \frac{1}{\rho s} \left(-\frac{dp}{dx} \right) \left(1 - \frac{\cos \left(i \sqrt{\frac{\rho s}{\bar{\eta}_k(s)}} y \right)}{\cos \left(i \sqrt{\frac{\rho s}{\bar{\eta}_k(s)}} h \right)} \right) \frac{ds}{s}. \quad (13).$$

After several calculations, we express the solution to the integral equation:

$$u(y, t) = \frac{h^2}{2\eta} \left(-\frac{dp}{dx} \right) \left[\left(1 - \frac{y^2}{h^2} \right) + 32 \sum_{n=1}^{\infty} \sum_{k=1}^{\infty} \frac{(-1)^n \xi(\alpha)}{(2n+1)^3 \pi^3 \sum_{k=1}^{\infty} \frac{(k^\alpha - 2EL\bar{s}_{k,1n})}{(k^\alpha - EL\bar{s}_{k,1n})^2}} \times \right. \\ \left. \times \cos \left(\left(\frac{2n+1}{2} \right) \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} \bar{s}_{k,1n} t} + \frac{(-1)^n \xi(\alpha)}{(2n+1)^3 \pi^3 \sum_{k=1}^{\infty} \frac{(k^\alpha - 2EL\bar{s}_{k,2n})}{(k^\alpha - EL\bar{s}_{k,2n})^2}} \times \cos \left(\left(\frac{2n+1}{2} \right) \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} \bar{s}_{k,2n} t} \right], \quad (14).$$

$$\frac{u(0, t)}{u_{0\max}} = 1 - 32 \sum_{n=0}^{\infty} (-1)^n \frac{\sqrt[3]{\xi^2(2)EL}}{3\sqrt[3]{(2n+1)^7} \pi^3} \left(\cos A\sqrt{3}t - \sqrt{3} \sin A\sqrt{3}t \right) e^{-\frac{\nu}{h^2} At}. \quad (15)$$

where $u_{0\max} = \frac{h^2}{2\eta} \left(-\frac{dp}{dx} \right)$ is the maximum stationary speed and $A = \frac{\sqrt[3]{(2n+1)^3} \pi^2}{8\sqrt[3]{\xi^2(2)EL}}$.

The found solution (15) is a solution to the Shulman-Husid model, and through it the process of transition of a viscoelastic fluid from an unsteady state to a stationary state in a flat channel is studied.

In the special case, if solution (14) has $\lambda_k = \frac{\lambda}{k^\alpha} = 0$, $\bar{\eta}_k = \eta$, $EL = \frac{\nu}{h^2} \lambda = 0$, $\eta_k^* = 1$, then we obtain the following solution

$$u(y, t) = \frac{h^2}{2\eta} \left(-\frac{dp}{dx} \right) \left[\left(1 - \frac{y^2}{h^2} \right) + 32 \sum_{n=1}^{\infty} \frac{(-1)^n}{(2n+1)^3 \pi^3} \cos \left(\left(\frac{2n+1}{2} \right) \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} st} \right], \quad (16)$$

$$\frac{u(0, t)}{u_{0\max}} = 1 + 32 \sum_{n=1}^{\infty} \frac{(-1)^n}{(2n+1)^3 \pi^3} \cos \left(\frac{2n+1}{2} \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} st}. \quad (17)$$

This found solution (17) represents the transition state of a Newtonian fluid from an unsteady state to a stationary one..

Now, if we shift the spectrum α in solution (14) to infinity, then we will find the following solution for the Maxwell model

$$u(y,t) = \frac{h^2}{2\eta} \left(-\frac{dp}{dx} \right) \left[\left(1 - \frac{y^2}{h^2} \right) + 32 \sum_{n=1}^{\infty} \left[\frac{(-1)^n}{(2n+1)^3 \pi^3 \frac{(1-2EL\bar{s}_{1n})}{(1-EL\bar{s}_{1n})^2}} \times \right. \right. \quad (18)$$

$$\left. \times \cos \left(\left(\frac{2n+1}{2} \right) \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} \bar{s}_{1n} t} + \frac{(-1)^n}{(2n+1)^3 \pi^3 \frac{(1-2EL\bar{s}_{2n})}{(1-EL\bar{s}_{2n})^2}} \times \cos \left(\left(\frac{2n+1}{2} \right) \pi \frac{y}{h} \right) e^{-\frac{\nu}{h^2} \bar{s}_{2n} t} \right].$$

$$\frac{u(0,t)}{u_{0\max}} = 1 + 32 \sum_{n=1}^{\infty} \left[\frac{(-1)^n}{(2n+1)^3 \pi^3 \frac{(1-2EL\bar{s}_{1n})}{(1-EL\bar{s}_{1n})^2}} e^{-\frac{\nu}{h^2} \bar{s}_{1n} t} + \frac{(-1)^n}{(2n+1)^3 \pi^3 \frac{(1-2EL\bar{s}_{2n})}{(1-EL\bar{s}_{2n})^2}} e^{-\frac{\nu}{h^2} \bar{s}_{2n} t} \right] \quad (19)$$

This solution (19) represents the process of transition from unsteady flow to steady flow of a viscoelastic fluid through the Maxwell model.

4. Results and discussion.

Let us describe the results of the numerical calculation of the found solutions (15), (17) and (19) in the graphs below.

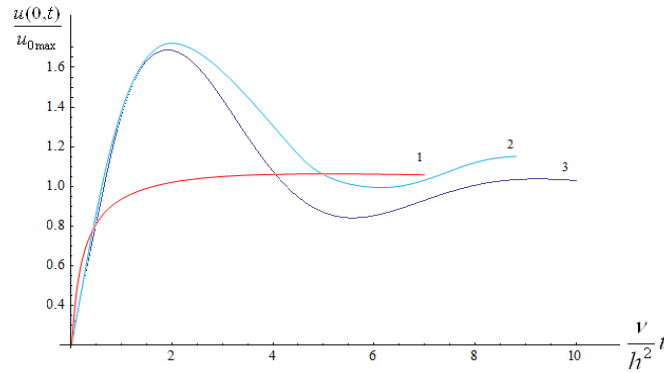


Figure 1. - Time dependence of the ratio of the maximum longitudinal velocity of a viscoelastic fluid in an unsteady flow to the maximum longitudinal velocity in a stationary state (1 – movement of a Newtonian fluid, 2 and 3 – Shulman-Husid model: 2 – at $EL = 2$, $\alpha = 2$, $\xi(2) \approx 1.7$; 3 – when $EL = 1$, $\alpha = 2$, $\xi(2) \approx 1.7$).

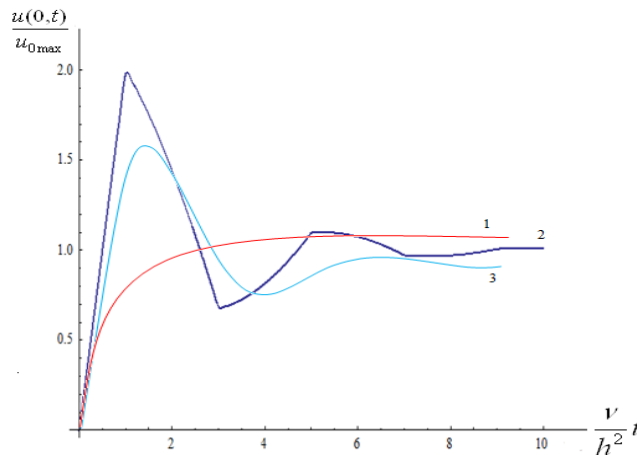


Figure 2. - Time dependence of the ratio of the maximum speed of an unsteady viscoelastic fluid to the maximum speed in a stationary state 1 – Newtonian fluid; 2 – with Maxwell liquid, $EL = 1$; 3 – in the case of Shulman-Husid liquid, when $EL = 1$, $\alpha = 2$, $\xi(2) \approx 1.7$.

From the first and second graphs it is clear that the process of transition of a viscoelastic fluid from an unsteady state to a stationary state differs sharply from the process in a Newtonian fluid. That is, during the transition from an unsteady state to a stationary state, the maximum relative velocity of a viscoelastic fluid increases within one and a half to two times the maximum relative velocity of a Newtonian fluid, subsequently oscillates around the same value of the processes and approaches asymptotically.

5. Conclusion

The movement of elastic-viscous fluids in a flat channel was studied on the basis of simplified mathematical models, and the results obtained were compared with the laws of transient processes in the unsteady flow of a Newtonian fluid, which are privately available, and as a result, new hydrodynamic effects were identified that differ from it.

The figure shows that the transition of an elastic-viscous fluid from an unsteady state to a stationary state based on the Maxwell and Shulman-Husid models manifests itself in a wave form, in contrast to a Newtonian fluid. It is noted that the transit time in the Maxwell fluid is several times greater than the transit time in the Newtonian fluid and the Shulman-Husid fluid. The reason for this is an increase in the number characterizing the distribution spectrum of the elasticity coefficient and relaxation time, which leads to an increase in the amplitude of current oscillations.

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MATHEMATICAL MODEL AND CALCULATION ALGORITHM MODERN HEAT EXCHANGER CONNECTED TO A TWO-PIPE HEATING NETWORK

Khujaev I.K.¹, Shirinov Z.Z.¹, Ravshanov Sh.A.¹

¹*Institute of mechanics and seismic stability of structures named after M.T.Urazbaev, Tashkent, Uzbekistan*

E-mail: ziyomatshirinov@umail.uz

Abstract. Using friction and gravity resistance forces, a model of coolant movement through a multi-section heat exchanger has been developed. The friction drag coefficient is taken in the form of a one-term Leibenzon representation. The heat transfer of elementary arcs is taken into account by the averaged heat transfer coefficient through the finned outer surface of the pipelines. To solve the hydraulic problem, analogues of Kirchhoff's laws and the method of mathematical induction were used, and to solve the thermal problem, Shukhov's formulas and the concept of the average flow temperature of the coolant at the confluence of flows were used. Some results of a computational experiment carried out using a newly developed software product based on a mathematical model and calculation algorithm are presented.

Keywords. multi-section heat exchanger, heat and mass transfer, formula for calculating pressure drop, Shukhov's formula, analogues of Kirchhoff's laws, mathematical induction, computational experiment.

1. Introduction

Large-scale work on green energy is underway around the world. Nuclear power plants and hydroelectric power stations are being built, and energy based on solar, wind and geothermal energy is being developed. Along with this, the development and widespread use of resource-saving technologies in relation to sectors of the economy and in everyday life require great attention. In particular, the transition from incandescent lamps to diode lamps contributed to a significant reduction in the electricity used for consecration. Various household devices with maximum energy savings have been developed and are being developed.

A similar picture is observed in the area of heating residential and industrial premises. The trend of abandoning a centralized integrated heating system in favor of separate, compact, multi-circuit and economical heating systems has made it possible to significantly reduce the volume of domestic gas consumption in production.

Design work on heating systems is carried out in accordance with existing standards, which in an integral form determine the volume of heat loss in the heating system. Depending on the methods of using heat and the needs of production shops, two- and multi-line heating networks are organized [1,2].

In practice, for example, thermal power plants were widely used to heat the working unit, which could provide heat to the city or its individual areas [2-4]. To ensure the reliability of the heating network, a ring structure was used [5, 6].

The methodology for hydraulic and thermal calculation of the heat supply network developed gradually. The first steps in this direction related to the solutions of linearized quasi-one-dimensional equations of conservation of mass, momentum and energy in individual sections of pipelines [3, 5, 7]. Subsequently, the theory of flow distribution was formed [5, 6, 8-10]. This theory is based on information about the hydrodynamic regime of friction and analogues of Kirchhoff's laws. Depending on the law of resistance, which is determined by the flow regime, and the averaging methods used, a system of equations is formed from linear and nonlinear ordinary differential equations or partial differential equations. In addition, the direction of flow in individual arcs of the network is not known in advance. The Lobachev-Cross, Vyhandu methods and their modifications for solved such systems have been developed [8, 9, 11-15].

Below is a mathematical model of a heat exchanger that is connected to a two-strand network. The friction resistance law was adopted according to Leibenzon, which is a one-term representation [11-12]. Using the method of mathematical induction and analogues of Kirchhoff's laws, formulas for hydraulic calculations were obtained taking into account changes in the leveling height of the route.

Thermal calculation of a multi-section heat exchanger was carried out using the Kirchhoff formula and the formula for the average coolant temperature along the flow.

For the quadratic flow regime, numerical results of the exact solution of the problem of the state of the working fluid in the heat exchanger are presented.

2. Statement and mathematical model of the problem.

Heat exchangers are made up of separate sections that have the same shapes and dimensions: height h_c , width l_c , internal diameters: vertical pipes D_v and horizontal pipes D_g . To calculate the heat exchanger, the transverse areas of the pipes f_v and f_g , their resistance coefficients λ_v and λ_g , are also required. In addition, we need the diameter and outer surface area of the sections and jumpers, the average heat transfer value k_{cp} of the heat exchanger and other parameters that are used in the process of calculating the pressure and heat loss of the working fluid when passing through the heat exchanger.

Hot water descends from above - from the upper main line (Fig. 1).

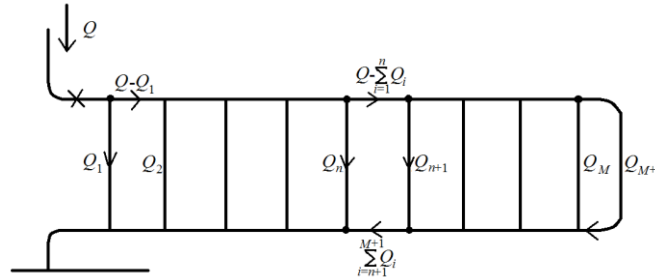


Figure 1. - Schematic representation of the heat exchanger, connected to a two-pipe network on one side

The heat exchanger is connected to the main line through a bend and a valve regulator. Depending on the position of the tap, a different cross-sectional clearance of the jumper is formed, which regulates the water flow Q through the heat exchanger. The total length of the input jumper, taking into account the equivalent Δl_p of local resistance, is l_p , the height difference in it h_p , the average diameter and cross-sectional area determined for the jumper are $\bar{D}_p$ and $\bar{f}_p$. The resistance coefficient of the bend (elbow) and crane is λ_p . The spent working agent, through the final section of the heat exchanger, enters the

lower line through the outlet pipe, which has indicators l_p , h_p , D_p , f_p and Δl_p . The last indicator is responsible for the local bending resistance of the output jumper.

We assume that there is no loss of water in the heat exchanger, i.e. the input and output water flow rates are the same and amount to Q .

It is required to determine the pressure and temperature losses of the liquid in the heat exchanger if the inlet temperature of the heated water is T_H , the inlet pressure is p_H , and the average room temperature is T_{oc} .

First, the hydraulics problem is solved: arc flow rates and node pressures are determined, and then the water temperature values in the design nodes of the heat exchanger are calculated.

3. Methods

Hydraulic calculation of the network is carried out taking into account changes in the leveling height of nodes h and pressure drop according to the Darcy-Weisbach law:

$$\frac{dp}{dx} + \rho g \frac{dh}{dx} + \frac{\lambda}{2D} \rho w^2 = 0,$$

where traditional notations are used [3, 5, 7].

When calculating the pressure drop, we implement a one-term representation of the law of viscous friction resistance, generalizing Nikuradze's experimental results for the resistance coefficient of a round pipe at various flow rates, diameters and equivalent pipe roughness [11-12]:

$$\lambda = \zeta (k/D)^\theta \text{Re}^n.$$

Here, in the laminar flow regime, you can use the Stokes formulas ($\zeta = 64$, $\theta = 0$, $\varphi = -1$), in the transitional regime - the Zaichenko formulas ($\zeta = 0.0025$, $\theta = 0$, $\varphi = 1/3$), in the smooth regime of turbulent flow around roughness - the Blasius formulas ($\zeta = 0.3164$, $\theta = 0$, $\varphi = -1/4$), in the mixed regime - the Leibenzon formulas ($\zeta = 10^{-0.627}$, $\theta = 0.127$, $\varphi = -0.123$), and in the developed regime - the Shifrinson formulas ($\zeta = 0.11$, $\theta = 0.25$, $\varphi = 0$).

The upper units of the heat exchanger with $M+1$ sections as 1, 2, ..., m , ..., M , $M+1$, and the lower ones – through 1', 2', ..., m' , ..., M' , $(M+1)'$. We denote the costs through vertical pipes as Q_1 , Q_2 , ..., Q_m , ..., Q_M . If their values are known, then the fluid flow through the last pipe on the right is $Q_{M+1} = Q - \sum_{i=1}^M Q_i$. The arc flow rates through the upper nozzle are $q_1 = Q - Q_1$, $q_2 = Q - Q_1 - Q_2$, ..., $q_m = Q - Q_1 - Q_2 - \dots - Q_m$, ..., $q_{M-1} = Q - Q_1 - \dots - Q_m - \dots - Q_{M-2} - Q_{M-1}$ and $q_M = Q - Q_1 - Q_2 - \dots - Q_m - \dots - Q_{M-1}$. The arc flow rates along the lower part are also determined.

As you can see, the total water flow through the vertical pipes is Q . By this we ensured the fulfillment of an analogue of Kirchhoff's first law.

We denote the nodal pressures by p_1 , p_2 , ..., p_m , ..., p_M , p_{M+1} , $p_{1'}$, $p_{2'}$, ..., $p_{m'}$, ..., $p_{M'}$, $p_{(M+1)'}$. To determine the values of these quantities, we use an analogue of Kirchhoff's second law.

The calculation begins by determining the pressure drop p and leveling height h at the inlet jumper according to the solution of the above equation.

The input jumper has the shape of a bend (elbow). Its resistance consists of friction resistance along the length of the jumper and local bending and valve-regulator resistances, expressed in total through Δi_p . The total length of the input jumper is l_p . If we denote the inlet pressure by p_H , then we have

$$p_H + \rho g h_p - p_1 = \frac{\lambda_p \rho (i_p + \Delta i_p)}{2 D_p f_p^2} Q^\alpha = K_1 Q^\alpha.$$

The density ρ of the liquid is taken for the average temperature of the coolant.

In the lower jumper, the pressure drop is calculated using the formula

$$p_1 + \rho g h_p - p_K = \frac{\lambda_p \rho l_p}{2 D_p f_p^2} Q = K_K Q^\alpha,$$

where p_K is the value of water pressure at the point of connection of the heat exchanger to the lower line.

Here the values of l_p , λ_p and D_p , as noted above, are taken without a valve-regulator.

The main difficulty of hydraulic calculations is related to the circularity of the network under consideration, which is due to the unknown direction of flow in specific arcs. To solve this problem, we turn to the provisions of mathematical induction, taking the number of rings $M=1$, $M=2$ etc., and determine the values of arc flow rates [13-16].

Number of circuits $M=1$. In this case, a contour 122'1 of two sections is formed (Fig. 2.). The total flow is distributed over the arcs 11' and 122'. Let us denote them by Q_1 and Q_2 , moreover $Q_1 + Q_2 = Q$. The pressure drop along the arcs 122'1 will 11' be the same:

$$\begin{cases} p_1 + \rho g h_c - p_{1'} = K_v Q_1^\alpha, \\ p_1 + \rho g h_c - p_{1'} = (2K_g + K_v) Q_2^\alpha. \end{cases} \quad (1)$$

Here and below we use the following notation:

$$K_g = \frac{\lambda_g \rho l_c}{2 D_g f_g^2}, \quad K_v = \frac{\lambda_v \rho h_c}{2 D_v f_v^2}, \quad (2)$$

which relate to the horizontal and vertical pipes of the heat exchanger sections.

System (1) implies the equality

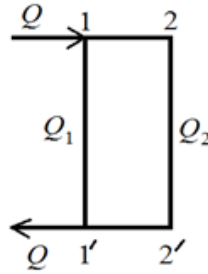


Figure-2. - Diagram of a heat exchanger with two sections

$$K_v Q_1^\alpha = (2K_g + K_v) Q_2^\alpha.$$

The advantages of our mathematical model over well-known models [5, 6, 8, 10] of a multi-circuit network are that the direction of flow in the arcs is determined in advance so that all costs Q_i have positive values. Because coefficients K_g and K_v are also positive quantities, then we can take:

$$\sqrt[\alpha]{K_v} Q_1 = \sqrt[\alpha]{2K_g + K_v} (Q - Q_1). \quad (3)$$

This form of notation in mathematics is accepted for the case $\alpha = 2$, i.e. in quadratic resistance mode. But in other cases ($\alpha \neq 2$) the designation is also accepted $\sqrt[\alpha]{x} = x^{1/\alpha}$. In this case, from equation (3) we find

$$Q_1 = \frac{\sqrt[\alpha]{2K_g + K_v}}{\sqrt[\alpha]{K_v} + \sqrt[\alpha]{2K_g + K_v}} Q = r_1 Q, \quad Q_2 = (1 - r_1) Q. \quad (4)$$

Then from the first equation of system (1) we can find

$$p'_1 = p + \rho g h_c - K_v Q_1^\alpha. \quad (5)$$

Formulas for $M = 2, 3, \dots$

For the occasion M parallel contours problem is solved as follows.

Respecting the generality with the previous options, we proceeded from the equality

$$Q_2 = r_{M-1} (Q - Q_1).$$

Here

$$r_{M-1} = \frac{\sqrt[\alpha]{2K_g + r_{M-2}^\alpha K_v}}{\sqrt[\alpha]{K_v} + \sqrt[\alpha]{2K_g + r_{M-2}^\alpha K_v}}.$$

At $r_0 = 1$ and further the values r_m are calculated recursively.

Let us turn to the contour, where the flows $122'1'$, Q_2 , $Q - Q_1$ and Q_1 , have been established along arcs along the clock hand. $Q - Q_1$

Let's calculate the pressure drop along the arcs $11'$ and $122'1'$:

$$p_1 + \rho g h_c - p'_1 = K_v Q_1^\alpha = K_g (Q - Q_1)^\alpha + K_v Q_2^\alpha + K_g (Q - Q_1)^\alpha.$$

Taking into account the new value, Q_2 we obtain the equation

$$K_v Q_1^\alpha = (2K_g + K_v r_{M-1}^\alpha) (Q - Q_1)^\alpha.$$

From here we find

$$Q_1 = \frac{\sqrt[\alpha]{2K_g + r_{M-1}^\alpha K_v}}{\sqrt[\alpha]{K_v} + \sqrt[\alpha]{2K_g + r_{M-1}^\alpha K_v}} Q = r_M Q.$$

Let us determine the arc flow rates for the remaining vertical pipes: $Q_2 = r_{M-1} (1 - r_M) Q$,

$$Q_3 = r_{M-2} (1 - r_{M-1}) (1 - r_M) Q, \dots, Q_m = r_{M-m+1} (1 - r_{M-m+2}) \dots (1 - r_M) Q,$$

$$Q_M = r_1 (1 - r_2) (1 - r_3) \dots (1 - r_M) Q, \quad Q_{M+1} = (1 - r_1) (1 - r_2) (1 - r_3) \dots (1 - r_M) Q.$$

Let us determine the costs for the upper horizontal arcs: $q_1 = (1 - r_M) Q$, $q_2 = (1 - r_{M-1}) (1 - r_M) Q, \dots$,

$$q_m = (1 - r_{M-m+1}) (1 - r_{M-m+2}) \dots (1 - r_M) Q, \dots, q_M = (1 - r_1) (1 - r_2) \dots (1 - r_M) Q,$$

Similar dependencies for the lower horizontal arcs of the network: $q'_1 = Q_M + Q_{M-1} + \dots + Q_2$,

$$q'_2 = Q_M + Q_{M-1} + \dots + Q_3, \dots, q'_m = Q_M + Q_{M-1} + \dots + Q_m, \dots, q'_{M-1} = Q_M + Q_{M-1}, q'_M = Q_{M+1}.$$

These values could be expressed in terms of r_M , Q , but here is an option that is easily implemented in programming.

The same algorithm was compiled to calculate the degree (square) pressure difference along an arbitrary arc.

The pressure value p_1 at the 1st node of the upper pipe is known. Then for subsequent nodes $i = 2..M+1$ of the upper arc the pressure values are calculated using the recurrent formula:

$$p_i = p_{i-1} - K_g q_{i-1}^\alpha,$$

in the lower nodes of vertical pipes at known values K_v :

$$p'_i = p_i - K_v Q_i^\alpha,$$

at the end of the lower arc, the pressure value at the nodes of the lower arc can also be calculated using another formula:

$$p'_i = p'_{(i+1)'} - K_g q_i^\alpha \cdot$$

We are only interested in the change in pressure along the arc 11'. The pressure value at the outlet of the heat exchanger (to the input of the output jumper) is:

$$p_{1'} = p_1 + \rho g h_c - K_v Q_1^\alpha = p_1 + \rho g h_c - K_v r_M^\alpha Q^\alpha \cdot$$

Thus, the flow rates for the vertical pipes Q_i , the upper pipes q_i , and the lower q'_i pipes, as well as the pressure drop along the arc, are determined 11', which corresponds to the total resistance of the heat exchanger.

Taking into account the upper and lower jumpers, the relationship between the inlet and outlet pressures of the heat exchanger is

$$p_H + \rho g (h_c + \dot{h}_p + h_p) - p_K = (K_1 + K_v r_M^\alpha + K_K) Q^\alpha \cdot$$

4. Materials

Based on the above material, a program was compiled in the Pascal ABC environment. The main indicators were: $\rho = 1000.0 \text{ кг м}^{-3}$, $l = 0.07 \text{ м}$, $h = 0.6 \text{ м}$, $h_p = 0.15 \text{ м}$, $\dot{h}_p = 0.15 \text{ м}$, $D_g = 0.03 \text{ м}$, $D_v = 0.01 \text{ м}$, $D_p = 0.03 \text{ м}$, $D_m = 0.03 \text{ м}$, $\dot{D}_p = 0.03 \text{ м}$, $\lambda_g = 0.002$, $\lambda_v = 0.002$, $\dot{\lambda}_p = 0.2$, $\lambda_p = 0.2$, $\lambda_m = 0.002$, $p_H = 110000.0 \text{ Па}$, $g = 9.80665 \text{ м с}^{-2}$.

The reaction of the final results to changes in a number of initial indicators was checked. Changes in the values of internal and external diameters, length and height of sections showed adequate changes in results according to physical concepts [17-21].

Let us present the results of calculating the coefficient r_i . Since the geometric dimensions of the heat exchanger did not change in the calculations, they applied to all calculation options.

Table 1.

Coefficient values r_i for given geometric parameters heat exchanger sections

i	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
r_i	1	0.50012	0.33381	0.25108	0.2019	0.16962	0.14707	0.13066	0.11839	0.10903	0.10181	0.09618	0.09178	0.08831	0.08558	0.08343	0.08172	0.08038	0.07931	0.07847	0.0778

Below we present some of the comparison results in the form of graphs. Here and further in the graphs, number 1 corresponds to the speed of the total flow in the heat exchanger $w = 0.3 \text{ м с}^{-1}$, number 2 - speed 0.4 м с^{-1} . Calculations were carried out for variants of ambient temperature $T_{oc} = 293.15$.

Let us imagine the values of arc volumetric flow rates (liters per second) at $T_{oc} = 293.15 \text{ K}$, $w = 0.3 \text{ м с}^{-1}$ for radiators with five, ten and fifteen sections.

Here, in three options, the flow rate through the first arc is the total flow rate (0.21206 l/s) of the working agent through the heat exchanger. In the first vertical pipes, the flow rates were 0.04282, 0.02312 and 0.01815 l/s, and in the final vertical pipes - 0.04219, 0.02040 and 0.01254 l/s. It is not difficult to make sure that it is distributed between the vertical pipes in descending order and the sum of the flow rate through all vertical pipes is equal to the total flow rate through the heat exchanger (Fig. 3). In the results for the upper and lower arcs there were differences (± 1) in the last digit.

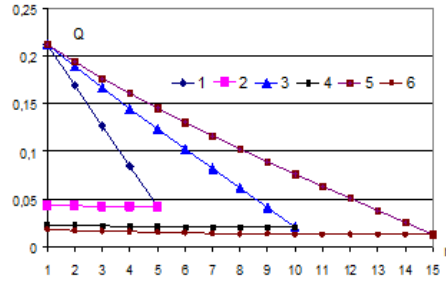


Figure 3. - **Distribution of volumetric flow rates in the upper (1,3,5) and vertical pipes (2,4,6) in heat exchangers with five, ten and fifteen sections**

With an increase in the number of sections, the costs for the vertical pipes decrease, because, as already noted, the sum of the flow rate for all vertical pipes is equal to the total flow rate for the heat exchanger.

Calculations of nodal pressures were carried out according to the quadratic law of resistance. The results, in particular from table. 2 showed that without taking into account changes in leveling height, the main pressure drops correspond to the vertical pipes of the heat exchanger, since the diameters of the vertical pipes ($D_v = 0.01 \text{ m}$) are significantly smaller than the other pipes. In the calculation algorithm, the pressures in the upper arc were first calculated, then the pressure drop along the vertical pipes was calculated, and finally a number of values of the nodes of the lower arc were calculated.

Table 2.

Values of nodal pressures (Pa) in a heat exchanger with ten sections

Nodal pressure (Pa)	Arc numbers									
	1	2	3	4	5	6	7	8	9	10
Upper arc nodes	111470	111469.8	111469.7	111469.6	111469.5	111469.4	111469.4	111469.4	111469.4	111469.4
The lower ends of the vertical pipe branch	111464.8	111464.9	111465	111465.1	111465.2	111465.3	111465.3	111465.3	111465.3	111465.3
Knots of the bottom arcs	111464.8	111464.9	111465.1	111465.2	111465.2	111465.3	111465.3	111465.3	111465.3	111465.3

The results of nodal pressures for a heat exchanger with ten sections are presented. Similar results were obtained for a heat exchanger with five (output 111452.13 Pa) and fifteen (output 111466.75 Pa) sections (output of a heat exchanger with five sections 111464.8 Pa). A comparison of the results showed that with an increase in the number of sections, the output pressure of the heat exchanger increases. This fact is consistent with the laws of hydraulics [7]: an increase in the number of parallel threads contributes to a nonlinear decrease in pressure loss.

The upper curves of Fig. 4 correspond to $w = 0.4 \text{ m} \cdot \text{s}^{-1}$, and the lower ones correspond to $w = 0.3 \text{ m} \cdot \text{s}^{-1}$. In practice, there is a linear change in the flow rate along the upper and lower arcs (Fig. 4), since the flow rate at the beginning of the upper arc is an order of magnitude greater than the flow rate through the vertical pipes (Fig. 5). With the removal of the inlet section, the flow rates through the vertical pipes decrease more slowly and this decrease is close to the exponential law.

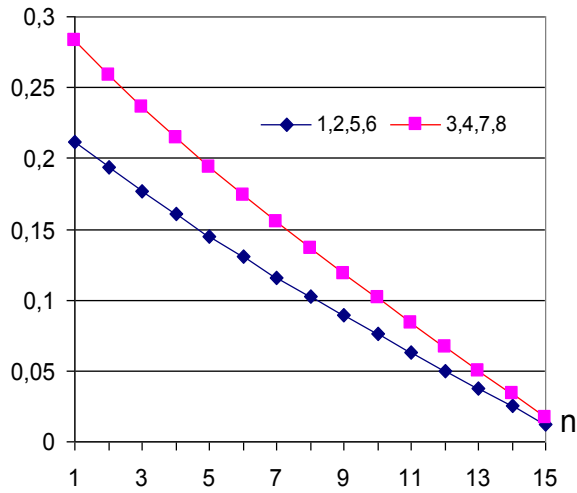


Figure 4. - Comparison of the results of local flow rates of the upper and lower arcs for two calculation options (see text for symbols)

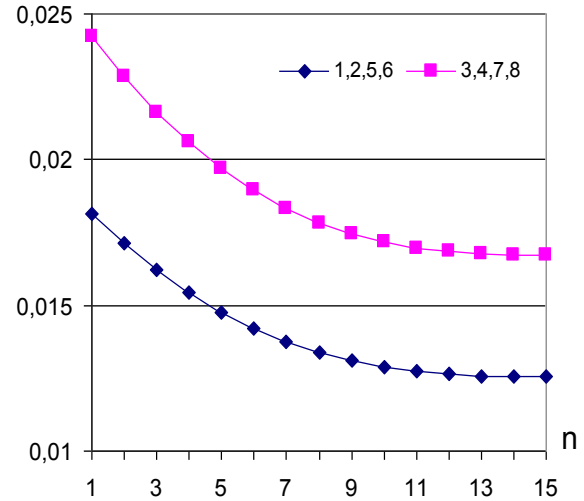


Figure 5. - Comparison of the results of local flow rates for vertical pipes for two calculation options (see text for symbols)

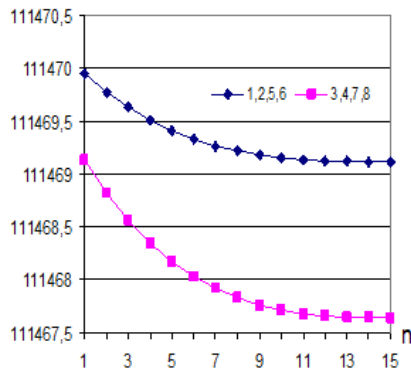


Figure 6. - Comparison of the results of nodal pressures of the upper arc for two calculation options (see text for symbols)

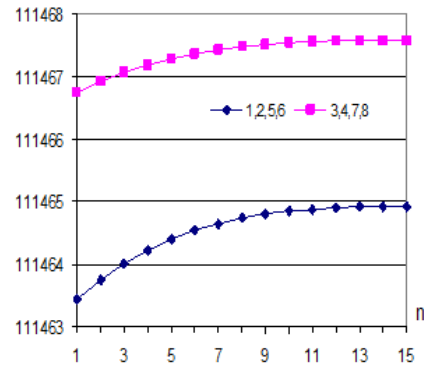


Figure 7. - Comparison of the results of nodal pressures of the lower arc for two calculation options (see text for symbols)

In Fig. 8 shows the pressure values at the outlet of the heat exchanger with 2, 3,... and 15 sections. The upper final pressure curves are related to the speed $w=0.3 \text{ } \mu\text{C}^{-1}$, and the lower ones are related to $w=0.4 \text{ } \mu\text{C}^{-1}$. As the number of sections increases, the pressure loss decreases, which is explained by Kirchhoff's second law: with an increase in the number of parallel threads, the pressure loss decreases.

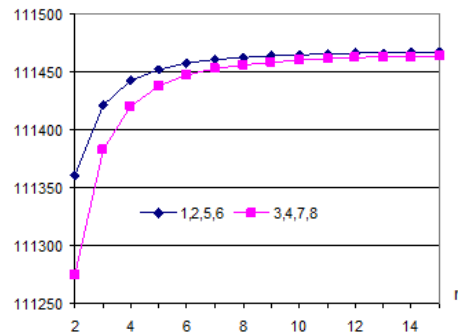


Figure 8. - Pressure values at the outlet of the heat exchanger for 8 variants of the initial data, depending on the number of sections in the heat exchanger (see text for symbols)

5. Conclusion

Within the framework of a quasi-one-dimensional approach, a mathematical model of a modern heat exchanger connected to a two-strand heating network is proposed.

Using analogues of Kirchhoff's two laws and the provisions of mathematical induction, a method for hydraulic calculation of a modern heat exchanger in a limited number of sections has been developed. The law of resistance takes into account all five hydrodynamic flow regimes, since it is used in the formula of its one-term representation. When describing mass transfer, the factor of leveling height is taken into account.

Analytical formulas are obtained for calculating local flow rates of incompressible fluid and pressure values in nodes. With known values of sectional calculations, the values of nodal pressures of the working fluid in a multi-section heat exchanger were determined.

It was revealed that the costs through the vertical radiator pipes are almost uniform: with an increase in the number of sections, as well as with distance from the entrance to the heat exchanger, the costs through the vertical pipes decrease.

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ГАЗОДИНАМИЧЕСКИЙ МЕТОД АЛЬТЕРНАТИВНОГО ПОЛУЧЕНИЯ НОВЫХ МАТЕРИАЛОВ ПОКРЫТИЯ

Москвитин Г.В.¹, Архипов В.Е.¹, Пугачев М.С.¹

¹Федеральное государственное бюджетное учреждение науки Институт машиноведения им. А.А. Благонравова Российской академии наук (ИМАШ РАН), Москва, Россия

E-mail: GVMoskvitin@yandex.ru

Аннотация: в работе показана возможность использования метода «холодного» газодинамического напыления, как альтернативного способа нанесения покрытий типа «латуни» и латуни разного состава. Методом рентгеноструктурного фазового анализа выявлено формирование в медно-цинковом покрытии твёрдых растворов электронного типа на базе CuZn_3 (ϵ -фазы), на базе Cu_5Zn_8 (γ – фаза), присущей сплавам меди и цинка с массовой долей 15,6% и 22,6% соответственно. Рентгенофлуоресцентным анализом выявлено зависимость химического состава покрытия от температуры напыления и возможность получения покрытия латуни марок Л80, Л70, Л60. Испытание медно-цинкового покрытия типа латуни в среде индустриального масла в паре со сталью ШХ15 показали низкую интенсивность изнашивания вплоть до эффекта безызносности без ухудшения качества поверхности пары трения.

Ключевые слова: покрытия; фазовый состав, газодинамическое напыление; структура, испытания.

1. Введение

К новым образцам и видам оборудования, которое используется в таких важных направлениях развития экономики как добыча, транспортировка и переработка углеводородов, скоростной транспорт, военная техника и т.д., предъявляют повышенные требования к ресурсу и безотказности, что требует уделять значительное внимание разработке новых материалов и покрытий, например, для узлов трения скольжения. Для изготовления подшипников трения скольжения используются многие металлические и неметаллические материалы, металлы и сплавы и, в частности медь, латуни и бронзы [1]. Для нанесения покрытий на основе меди и сплавов на её основе используют многочисленные методы, основанные на принципах химического, электрохимического и физического осаждения и нанесения металлов и сплавов на поверхность железо - углеродистых сплавов [2]. Традиционные методы нанесения покрытий, например, газотермические, хотя и используются для нанесения износостойких и коррозионностойких покрытия, но процесс из-за значительного энерго вложения сопровождается значительным разогревом детали (изделия) и, соответственно, изменением механических свойств материала подложки и короблением изделия, что требует для изделий проведения дополнительной термической и механической обработки.

Альтернативным и наиболее перспективным способом получения на железоуглеродистых сплавах покрытий различного функционального назначения можно считать метод «холодного» газодинамического напыления, который позволяет наносить покрытия на локальные участки и значительные площади поверхности деталей и изделий при минимальном разогреве подложки ($\leq 160^\circ\text{C}$) при отсутствии изменения механических свойств и коробления [3,4]. В российских и зарубежных научных центрах проводятся работы по нанесению покрытий триботехнического назначения на основе частиц латуни, меди и механической смеси частиц меди, а также меди и цинка с оксидом алюминия [3-6]. Газодинамическое напыление позволяет наносить покрытия, прочность сцепления которых с подложкой из стали, и когезионная прочность зависит от параметров напыления и от свойств металлов и составляет, например, для меди и цинка до 50

МПа и до 130 МПа соответственно, что позволяет их использовать при изготовлении или восстановлении ответственных узлов [4].

Результаты исследования процесса диффузии и структурно фазовых превращений в наносимом покрытии при напылении механической смеси частиц меди, цинка и корунда (оксида алюминия) показывают наличие диффузии меди в частицу цинка с формированием фаз, присущих латуням. Рентгеноструктурный фазовый анализ выявляет продукты превращения, которые присутствуют в правой части диаграммы состояния медь-цинк, и представляют собой твёрдый раствор меди в цинке (η -фаза) и твёрдые растворы электронного типа на базе CuZn_3 (ε -фаза) и на базе Cu_5Zn_8 (γ – фаза) [7]. Выявлена существенная зависимость массовой доли продуктов превращения от температуры и продолжительности напыления, и, например, массовая доля γ -фазы увеличивается с 17% до 33% при увеличении температуры напыления с 360°C до 450°C [7]. Исходя из наличия модифицированных частиц цинка с твёрдостью до 170 HV, а также цинка и меди с твёрдостью до 50 HV и 110 HV соответственно структуру медно-цинкового покрытия типа «латуни» можно назвать градиентной многокомпонентной и отнести к НОВЫМ МАТЕРИАЛАМ. Такой характер формирования и строения медно-цинкового покрытия типа «латуни» может оказать влияние на повышение триботехнических характеристик поверхности и сопряжённой пары в целом, что и показали его сравнительные испытания в условиях сухого трения [8].

Целью исследования является анализ и оценка влияния температуры и продолжительности напыления и последующей термической обработки на формирование фазового состава медно-цинкового покрытия и трибологические свойства поверхности.

2. Материалы, методы и объект исследования

Покрытие на подложку из стали 40X напыляется с использованием газодинамической установки «ДИМЕТ – 404» и механической смеси частиц меди, цинка и оксида алюминия (Al_2O_3) марки С-01-11 с соотношением ингредиентов по массе $\text{Cu}:\text{Zn}:\text{Al}_2\text{O}_3 = 35\%:35\%:30\%$, поставляемой Обнинским центром порошкового напыления [3]. Напыление проводится при скорости перемещения сменного сопла относительно поверхности образца - 10 мм/с и расстояния от среза сопла до поверхности напыления 10 мм при смещении сопла на $L=3$ мм относительно ранее нанесённого слоя металла [7].

Термическую обработку нанесённого медно-цинкового покрытия типа «латуни» проводили в печи при температуре в интервале 420-430°C и продолжительностью процесса до 180 мин.

Фазовый состав покрытий системы Cu-Zn исследуется на многофункциональном рентгеновском дифрактометре Rigaku Ultima IV с использованием $\text{CuK}\alpha$ -излучения и параллельного пучка. Для формирования параллельного пучка используется рентгеновская оптика в виде многослойного параболического зеркала. Дифрактограммы снимаются в симметричном режиме, диапазон сканирования 2θ от 20 до 120, шаг 0,05°, скорость детектора

Анализ дифрактограмм осуществляется на основе программного продукта PDXL (Rigaku) с использованием базы данных PDF-2. Количественный анализ проводится с использованием метода Ритвельда, реализованного в пакете программ PDXL (Rigaku).

Рентгенофлуоресцентный анализ осуществляется на спектрометре RigakuPrimusII с использованием в качестве источника рентгеновского излучения рентгеновской трубки с Rh-анодом, в вакууме и диапазоне элементов от Ca до U. Количественный анализ проводится после записи и расшифровки экспериментальных спектров от исследуемых образцов с помощью специального программного обеспечения фирмы Rigaku ZSX методом фундаментальных параметров SQX (без использования эталонных образцов).

Триботехнические испытания пар сталь 40X с покрытием–сталь ШХ15 проводили на стенде МТ-8 при возвратно-поступательном скольжении плоских сопрягаемых тел в среде

индустриального масла И-20А со средней скоростью скольжения 0,19 м/сек при ступенчатом повышении нагрузки в диапазоне 30-100 Н. Выбор схемы трения обусловлен тем, что реверсивный вид движения пары трения относится к наиболее тяжёлым и позволяет адекватно выявить преимущества или недостатки того или иного вида обработки конструкционных материалов [9]. Величины интенсивности изнашивания (I_n) определяли по потере массы каждого из образцов пары, оценивали также суммарную интенсивность изнашивания покрытия (I_Σ), отражающую его предельный износ и предельную работоспособность пары в целом.

3. Постановка задачи

Исследуется влияние температуры напыления в интервале от 270°C до 540°C и продолжительности процесса напыления – количества циклов повторения нанесения покрытия (3 и 5) на фазовый и химический состав медно-цинкового покрытия. Определяется продолжительность термической обработки медно-цинкового покрытия при температуре 420-430°C для получения покрытия латуни. Исследуется интенсивность изнашивания медно-цинкового покрытия в исходном состоянии и после термической обработки в течение 10 мин и состояние поверхности сопряжённой пары.

Рентгеноструктурный фазовый анализ медно-цинкового покрытия, нанесённого при температуре потока воздуха 270-540°C и разной продолжительности процесса напыления (3 и 5 циклах напыления), представлены в табл. 1.

Таблица 1.

Результаты рентгеноструктурного фазового анализа медно-цинкового покрытия

Температура, °C/циклы напыления	Массовая доля металлов и фаз, %			
	Cu	Zn	ε	γ
270/3	83,8	1,3	8,5	-
270/5	82,1	5,5	5,2	-
360/3	69,0	12,6	10,4	-
360/5	56,7	26,0	9,0	-
450/3	40,3	27,0	15,1	-
450/5	48,1	30,8	12,9	-
540/3	41,6	28,6	15,6	-
540/5	30,9	24,7	13,7	22,6

Рентгеноструктурный фазовый анализ медно-цинкового покрытия, нанесённого при 450°C и 5 циклах напыления после термической обработки при температуре 420-430°C представлен в табл. 2.

Таблица 2.

Фазовый состав медно-цинкового покрытия после напыления и термической обработки

Фазы и химические соединения	Массовая доля элементов, химических соединений и фаз, %							
	Время выдержки, мин							
	0	3	5	10	20	40	60	180
Cu	31	40	38	28	25	18	13	-
α – тв. p-p	-	-	13	31	34	40	61	70
CuZn (β')	-	-	32	23	21	22	10	12
Cu ₅ Zn ₈ (γ)	47	52	-	-	-	-	-	-
Zn	16	-	-	-	-	-	-	-
ZnO	5	7	8	8	12	14	13	12
Al ₂ O ₃	1	1	9	10	8	6	3	6

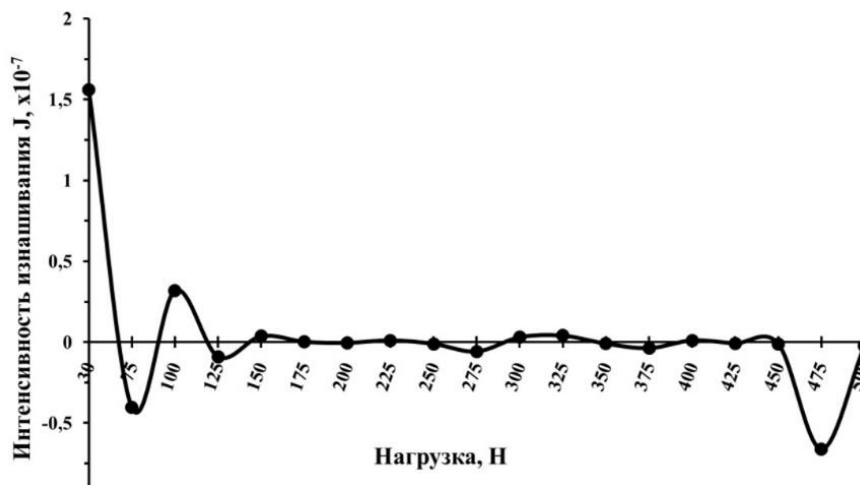
Состав медно-цинкового покрытия, нанесённого при температуре 270-540°C и 5 циклах напыления, полученного методом рентгенофлуоресцентного анализа, представлен в табл. 3.

Таблица 3.

Результаты химического анализа медно-цинкового покрытия, нанесённого при 5 циклах напыления

Температура, °C	Химические элементы, ат%		Фазы
	Cu	Zn	
270	82,2	17,7	«α»
360	69,41	30,1	«α»
450	56,87	41,87	«α+ β'»
540	45,33	53,83	«β' + γ»

Результаты испытания медно-цинкового покрытия, нанесённого при температуре 450°C и 5 циклах напыления в паре с контртелом из стали ШХ15, представлены на рис. 1.



а



б

Рисунок 1. - Зависимость интенсивности изнашивания от нагрузки на образец: а - сталь 40Х с покрытием; б - контртело сталь ШХ15

Результаты исследования микрогеометрии поверхности образцов с медно-цинковым покрытием в исходном состоянии и после термической обработки в течение 10 минут и контртела из стали ШХ15 с твёрдостью 60-62 HRC представлены в табл. 4.

Таблица 4.

Параметры микрогеометрии поверхностей трения

Материал покрытия Cu-Zn-Al ₂ O ₃	Характеристики микрогеометрии поверхности					
	Исходное состояние		Класс чистоты	После трения		Класс чистоты
	R _a	R _z		R _a	R _z	
К о н т р о л ь с т а л ь ШХ15						
Исходное покрытие	0,527	3,555	8 - 9	0,386	2,314	9 - 8
Покрытие после ТО	0,435	2,975	8 - 9	0,680	1,645	8 - 9
О с н о в н о й о б р а з е ц С т а л ь 40Х с покрытием						
Исходное покрытие	0,285	1,955	9	0,566	4,412	8
Покрытие после ТО	0,276	1,668	9	0,276	2,081	9

4. Анализ результатов

Повышение температуры напыления от 270°C до 540°C при трёх циклах напыления сопровождается снижением массовой доли меди с 83,8% до 41,6% и повышением цинка с 1,3% до 28,6%, что связано с энергетическими параметрами процесса газодинамического напыления и изменением химического состава покрытия (табл. 1, табл. 3). Повышение продолжительности напыления не влияет на массовую долю меди при низкой температуре напыления, которая снижается до 30,9% при максимальной температуре напыления. Массовая доля цинка увеличивается при температуре 270°C до 5,5% и снижается до 24,7% при температуре потока воздуха 540°C, что связано с изменением фазового состава медно-цинкового покрытия. Напыление смеси порошков меди, цинка и корунда сопровождается диффузией меди в цинк и формированием твёрдого раствора электронного типа на базе CuZn₃, массовая доля которого увеличивается с 8,5% до 15,6% при с повышением температуры при использовании 3 циклов напыления и с 5,2% до 13,7 при увеличении продолжительности процесса. Снижение массовой доли ε-фазы обусловлено изменением условий диффузии компонентов, а именно: количества вакансий, величины микродеформаций, размера элементов субструктуры, что влияет на межграницную диффузию и диффузию по вакансионному механизму. Однако, при использовании максимальной температуры формируется твёрдый раствор электронного типа на базе Cu₅Zn₈ (γ-фаза). Этот процесс обусловлен большей скоростью деформации металлов частицами корунда и формированием точечных и протяжённых дефектов. Формирование фаз присущих латуням в процессе напыления механической смеси порошком меди, цинка и корунда и наличие их в медно-цинковом покрытие с разной массовой долей даёт основание назвать формируемое покрытие, как покрытие типа «латуни» и отнести к НОВЫМ МАТЕРИАЛАМ.

В покрытии, нанесённом при 450°C рентгеноструктурный фазовый анализ медно-цинкового покрытия показал наличие в нём меди, цинка и твёрдого раствора электронного типа на базе Cu₅Zn₈ (γ - фазы) с массовой долей 31%, 16% и 47% (табл.2). Термическая обработка с максимальным временем выдержки в течение 180 мин приводит к формированию покрытия на основе твёрдого раствора цинка в меди (α – фаза) и твёрдого раствора электронного типа на базе CuZn (β' – фаза) с массовой долей 70% и 12%. Однако, уже после термической обработки в продолжении 60 мин у меди (табл.2) выявляется решётка с параметрами 0,3633 нм, что следует считать твёрдым раствором цинка в меди и соответствует содержанию меди ≈92%. Таким образом, термическая обработка в печи медно-цинкового покрытия в течение 60 минут сопровождается формированием латуни типа Л65 с двумя типами α – фазы и особенными трибологическими свойствами [10].

Повышение температуры напыления с 270°C до 540°C сопровождается последовательным снижением в медно-цинковом покрытии содержания меди с 82,2 ат% до 45,33 ат% и соответственно повышением содержания цинка с 17,7 ат% до 53,83 ат% (табл. 3). Таким образом, напыление медно-цинкового покрытия на изделия из железоуглеродистых сплавов при температуре 270°C и 360°C сопровождается нанесением слоя металлов в соотношении, которое позволит после термической обработки получить латунь типа полутомпак Л80 (медь 79-81 ат%) и типа Л70 (медь 69-72 ат%) (рис.1). Последующая термическая обработка покрытий, нанесённых при температурах 450°C и 540°C приведёт за счёт диффузии (взаимодиффузии) компонентов и превращений в твёрдой фазе к формированию на поверхности изделий из железоуглеродистых сплавов покрытие двухфазных латуней на основе $\alpha+\beta'$ фаз типа Л60 (медь 59-62 ат%) и на основе $\beta'+\gamma$ фаз, аналогов которой в номенклатуре двойных латуней не приводится. Эти латуни значительно отличаются по механическим свойствам, например по пределу прочности 260-310 МПа и относительному удлинению (пластичности) 45-50%, что позволит использовать покрытия при эксплуатации деталей и изделий в разных условиях воздействия внешней среды и взаимодействия сопряжённых пар [11]. Таким образом, за счёт изменения режимов «холодного» газодинамического напыления и последующей термической обработки можно осуществлять управление свойствами и характеристиками поверхности железоуглеродистых сплавов. А метод «холодного» газодинамического напыления следует назвать альтернативным способом получения медно-цинковых покрытия типа «латуни» и покрытий латуни посредством последующей термической обработки.

На рис. 1 приведены экспериментальные данные величины износа стали 40Х с медно-цинковым покрытием и контртела из стали ШХ15 в процессе испытаний пары трения в неактивном жидком смазочном материале (индустриальное масло) при ступенчатом повышении внешнего давления. Видно, что после небольшого пути трения при минимальной нагрузке наблюдается резкое уменьшение интенсивности изнашивания покрытия с $1,10^{-7}$ до нулевой отметки, а дальнейшее увеличение давления сопровождается, частичным переносом медного сплава на сопряженное контртело. Этот процесс на основном образце (сталь 40Х) имеет выраженный волнообразный характер, его особенность состоит во взаимном переносе материала покрытия на контртело и наоборот. При повышенном давлении износ образца с покрытием практически приближается к нулю, а некоторое изменение на отдельных участках пути трения сопровождается частичным переносом покрытия на контртело [12].

Следует заметить, что износ сопряженного образца (контртела) при всех нагрузках не происходит: поверхность трения полностью защищена покрытием типа «латуни», сформировавшимся в результате массопереноса в зоне контакта и «намазывания» медного сплава на него (табл. 4).

Кроме того, полученные экспериментальные исследования структуры покрытия и работоспособности пары трения указывают на то, что деформационный процесс в зоне фрикционного контакта стали 40Х с медно-цинковым покрытием в паре со сталью ШХ15 приводит к результату, близкому к эффекту «классической безызносности». Однако отличительной особенностью этого результата является формирование не сервовитной пленки меди (с особым структурным состоянием), а медного или медно-цинкового третьего тела, которое обладает способностью перехода с образца на контртело и наоборот с контртела на основной образец, обеспечивая защиту от разрушения зоны контактного взаимодействия. Таким образом, высокая износостойкость материалов пары трения достигается благодаря пленкам фрикционного массопереноса [12].

5. Обсуждение

Полученные результаты, безусловно, свидетельствуют о перспективности метода «холодного» газодинамического напыления, как альтернативного способа получения покрытий

«типа латуни» и латуни. Способ позволяет за счёт изменения технологических параметров напыления получать покрытия, имеющих широкий спектр механических, коррозионных и трибологических свойств, что позволяет использовать покрытия для изделий, работающих в различных условиях эксплуатации.

6. Заключение

Представляет значительный научный и практический интерес проведение исследований трибологических характеристик, коррозионных и механических свойств покрытий «типа латуни» и латуни полученных методом газодинамического напыления с последующей термической обработкой для определения областей их использования.

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МАТЕМАТИЧЕСКАЯ МОДЕЛЬ И МЕТОД РАСЧЁТА УЧАСТКА ГАЗОПРОВОДА, ФУНКЦИОНИРУЮЩЕГО В КВАДРАТИЧНОМ РЕЖИМЕ ТЕЧЕНИЯ

Тошов Б.Р.¹, Ходжаев Ш.Т.², Абдукаримов А.²

¹Навоинский Государственный горно-технологический университет, Навои, Узбекистан

²Самаркандский филиал Ташкентского университета информационных технологий

им. Мухаммада ал-Хорезми, Самарканд, Узбекистан

khodzhaev_shukhrat@mail.ru

Аннотация: В данной работе представлена математическая модель и новый метод расчета участка газопровода, функционирующего в квадратичном режиме сопротивления при наличии множества узлов отбора газа. Проектирование и управление газопроводными системами с большим числом таких узлов представляют сложную задачу, требующую точных инструментов анализа и прогнозирования. Предложенная модель и метод расчета способствуют эффективному учету влияния множества узлов отбора на работу участка газопровода при использовании формулы Шифринсона. Представленные результаты полезны при проектировании, эксплуатации и оптимизации газотранспортных систем.

Ключевые слова: газопровод, давление, газ, расход, плотность, скорость, путевой отбор.

1. Введение

В современном мире трубопроводные системы являются важнейшими элементами различных технологий и быта, обеспечивающих транспортировки природного газа, воды, нефти, нефтепродуктов и других флюидов, а также тепловой и механической энергии от места выработки (добычи) до потребителя.

В работах [1-3] представлен математический аппарат расчета трубопроводных систем, которые состоят из линейных и точечных составляющих. Линейные составляющие сети трубопроводов моделируются с привлечением квазиодномерных уравнений, которые представляют законов сохранения массы, импульса и энергии в трубопроводах. Данные уравнения, как наследники обобщенных уравнений Навье-Стокса, являются строго нелинейными и сначала они разрабатывались для элементарных участков газо- и трубопровода, а далее использовались при разработке способов расчета сети трубопроводов с лучевой, закольцованной и комбинированной структурами. Начали учитывать различные внешние и внутренние факторы, которые влияют на работу сети трубопроводов.

В работе [4] предложены модели оледенения и оттаивания внешней поверхности морского газопровода в условиях северных морей. При формировании условия фронта оледенения учтен фактор солёности воды. Нестационарная задача сформулирована в рамках многослойной цилиндрической области. Приведены алгоритм численного решения нестационарной задачи оледенения (оттаивания) многослойной цилиндрической области методом явного выделения фронта и результаты расчета вариантов задач, представляющих практический интерес.

В работе [5] рассмотрен процесс течения жидкости в упругом трубопроводе с закрепленными концами. Показано, что в таком трубопроводе возможно возникновение автоколебаний стенки и потока жидкости. Решение системы нелинейных гидродинамических уравнений получены в элементарных функциях, которые адекватно описывают эти автоколебания. Выявлена причина возникновения стоячих волн в упругом трубопроводе.

В работе [6] исследована оптимальная совместная работа различных гидравлических систем на базе А, Б и С линии узбекского сечения на Трансгазизатском газопроводе. Намечены пути увеличения пропускной способности данной сети с возведением акведуков и параллельных линий.

В работе [7] предложен гибридный подход для решения квазиодномерных газодинамических уравнений. В областях изоэнтропического течения идеального газа вместо уравнения сохранения и переноса энергии предложено использовать уравнение относительно энтропии. Приведены результаты численных расчётов некоторых модельных задач, полученные классическим методом Годунова и алгоритмом на основе гибридного подхода. Они сравнены для

различных комбинациях исходных данных, которые показали успехи и недостатки этих численных методов при решении модельных задач.

Примером привлечения полных уравнений Навье-Стокса для трубопровода с учетом молярного переноса субстанций является работа [8]. В ней получены точные решения уравнения Навье-Стокса, приведенные к уравнению третьего порядка с учетом молекулярного и молярного переноса между слоями потока.

Исследование [9] посвящено безопасной и эффективной эксплуатации и управлению крупномасштабных газопроводных сетей с целью анализа показателей надежности системы, устройств, и структуры управления системой трубопроводной сети. Цель, значение и содержание исследований надежности трубопроводных сетей природного газа подробно обсуждаются на системном уровне с учетом топологии трубопроводов и расположения силовых установок. Обсуждаемые методы перспективны и обеспечат техническую поддержку и поддержку принятия решений для безопасной эксплуатации сетей газопровода.

В статье [10] обсуждается проблема оптимального проектирования наземной трубопроводной системы хранения газа в условиях закачки и отбора газа из газохранилища.

В работе [11] рассматриваются математические модели и многоуровневые методы численного решения задач нестационарного потока газа в газотранспортных системах произвольной конфигурации, содержащие внутренние и внешние замкнутые петли. Рассмотренные методы ориентированы на использование различных вариантов метода конечного объема, вариационного метода и многосетевого приближения обобщенных решений параболических уравнений. Они позволяют использовать в расчетах успехи параллельных вычислений с удобным интерфейсом.

В монографии [12] подробно представлен подход путевого отбора газа, когда в правой части уравнения сохранения массы образуется отражающий линейного изменения массового расхода жидкости член. Работа интересна тем, что широко рассматриваются способы приложения функций Хэвисайда и дельта-функции Дирака к моделям трубопроводной транспортировки малосжимаемых сред.

В работах [13-15] сформулированы основы теории потокораспределения, где широко используются положения квазиодномерного подхода к задачам трубопроводной транспортировки флюидов,

В диссертационной работе А.Р. Акбасова [16] представлен краткий обзор работ по теории потокораспределения и о тепловых сетях ряда городов. Разработаны математическая модель, численный алгоритм и программное средство для расчета закольцованной сети теплоснабжения города с элементами управления. Разработана концепция интеллектуальной системы управления тепловыми сетями города, которая использована в системах теплоснабжения Казахстана.

Как отметили выше, расчет линейных составляющей сети трубопроводов проводится в рамках полуэмпирических законов сопротивления в трубах круглого сечения. В зависимости от гидродинамического режима течения предложены различные расчетные формулы. Более приемлемым для инженерных расчетов является одночленное представление закона сопротивления. В работах [17-18] представлена версия обобщенного коэффициента сопротивления трения по Лейбензону, которые являются замыкающими соотношениями для законов Кирхгофа при расчете простых и сложных сетей трубопроводов для пяти гидродинамических режимов течения флюидов.

В работе [19] сравнены результаты расчетов, полученных на основе подходов путевого и концентрированного отбора транспортируемой среды. Доказано, что использование подхода путевого отбора значительно сокращает объема расчетов при проектировании сети трубопроводов.

Работа [20] посвящена разработке «мягких» методов математического моделирования применительно к системам магистральных газопроводов. Такой вопрос возникает, например,

если известен интервал интенсивности отбора из сети газопроводов, а не точное ее значение. В работе разработан алгоритм решения задачи на основе метода интервалов.

На сегодняшний день разработаны различные способы решения стационарных и нестационарных задач элементарного участка и сети трубопроводов с учетом многочисленных факторов. Но проблемы усовершенствования математической модели трубопровода с учетом переменного диаметра и уточняющих факторов, всестороннего анализа состояния среды в процессе транспортировки, а также в условиях сложной топологии сети, требуют проведения дополнительных исследований. Эффективное управление такими системами требует разработки точных математических моделей и методов расчета, особенно в условиях, когда существует множество узлов подвода и отбора газа.

В этом плане продолжив хорошую традицию по использованию подхода путевого отбора [12-15, 19, 21 и др.], предложен новый метод решения задачи путевого отбора при квадратичном режиме сопротивления и выведены условия его применения.

Данный подход продуктивен и полезен для случаев, когда количество подключения потребителей достаточно большое. На основе приведенных уравнений можно составить алгоритмы для расчета n последовательных участков, которые имеют отличные друг от друга гидравлические показатели (диаметр, уклон, коэффициент сопротивления и др.) и интенсивность путевого отбора.

Для выяснения роли отдельных параметров на формировании давления, плотности и скорости газа по длине участка был проведен вычислительный эксперимент для различных комбинаций исходных параметров.

2. Постановка задачи

В простом варианте использования квадратичного закона сопротивления и в равноудаленном расположения узлов отбора газа по всей длине участка можно ставить и решить задачу о распределении давления следующим образом.

Пусть, на линейном участке с длиной L , диаметром D и коэффициентом сопротивления λ во взаимно равноудаленных n узлах осуществляется локальный отбор газа с одинаковой интенсивностью q ($\text{кг}/\text{с}$). Расход газа на входе в участок составляет M_0 ($\text{кг}/\text{с}$). Рассматриваются случаи $M_0 \geq nq$. Притом, $M_0 = nq$ соответствует тупиковому случаю, а $M_0 > nq$ – случаю транзитного переноса газа с расходом $M_0 - nq$ через участок.

Требуется разработать способ гидравлического расчета данной задачи.

3. Материалы, методы и объект исследования

За основу расчета берем формулу [18, 22]:

$$p_K^2 = p_H^2 - \frac{\lambda}{A^2 D^5} M^2 l. \quad (1)$$

Здесь $A = \frac{\pi T_{cm}}{4 p_{cm}} \sqrt{\frac{R_{гозд}}{Z R \Delta}}$; $\Delta = \rho_{газ} / \rho_{гозд}$; p_H, p_K – значения давления на входе и выходе участка;

$M = f \rho w$ – массовый расход на данном элементарном участке.

Определим значения массового расхода газа на отдельных участках, которые нумеруются индексами 1, 2, ..., n :

$$\begin{aligned} M_1 &= M_0, \quad M_2 = M_0 - q, \quad M_3 = M_0 - 2q, \dots, \\ M_i &= M_0 - (i-1)q, \dots, M_n = M_0 - (n-1)q. \end{aligned}$$

Длина каждого элементарного участка составляет $l = L / n$.

Узлы пронумеруем как $i=1, 2, \dots, n+1$.

Значения давления в узлах составляют последовательность

$$p_H = p_1, p_2, p_3, \dots, p_{n+1} = p_K.$$

Согласно формуле расчета для участков имеем:

$$p_1^2 = p_2^2 - \frac{\lambda}{A^2 D^5} M_0^2 l,$$

$$p_2^2 = p_3^2 - \frac{\lambda}{A^2 D^5} (M_0 - q)^2 l,$$

$$p_3^2 = p_4^2 - \frac{\lambda}{A^2 D^5} (M_0 - 2q)^2 l, \dots$$

$$p_i^2 = p_{i+1}^2 - \frac{\lambda}{A^2 D^5} (M_0 - (i-1)q)^2 l, \dots$$

$$p_n^2 = p_{n+1}^2 - \frac{\lambda}{A^2 D^5} (M_0 - (n-1)q)^2 l.$$

Суммирование этих равенств приводит к равенству

$$p_1^2 = p_{n+1}^2 - \frac{\lambda}{A^2 D^5} \left[M_0^2 + (M_0 - q)^2 + (M_0 - 2q)^2 + \dots + (M_0 - (i-1)q)^2 + \dots + (M_0 - (n-1)q)^2 \right] l.$$

Выражение в средней скобке преобразуем следующим образом:

$$\begin{aligned} & M_0^2 + (M_0 - q)^2 + (M_0 - 2q)^2 + \dots + (M_0 - (i-1)q)^2 + \dots + (M_0 - (n-1)q)^2 = \\ & = nM_0^2 - 2M_0q(1 + 2 + 3 + \dots + i + \dots + (n-1)) + q^2(1^2 + 2^2 + 3^2 + \dots + i^2 + \dots + (n-1)^2) = \\ & = nM_0^2 - 2M_0q \frac{n(n-1)}{2} + q^2 \frac{(n-1)n(2n-1)}{6}. \end{aligned}$$

Здесь использовали формулы для сумм первых $n-1$ натуральных чисел $\sum_{i=1}^{n-1} i = \frac{n(n-1)}{2}$ и

суммы квадратов первых $n-1$ натуральных чисел $\sum_{i=1}^{n-1} i^2 = \frac{(n-1)n(2n-1)}{6}$ [10].

С учетом $l = L/n$, $p_H = p_1$, $p_{n+1} = p_K$ последнее равенство приобретает вид

$$p_K^2 = p_H^2 - \frac{\lambda}{A^2 D^5} \left[M_0^2 - 2M_0q \frac{n-1}{2} + q^2 \frac{(n-1)(2n-1)}{6} \right] L.$$

Величина nq представляет суммарную для данного участка интенсивность отбора. Как и в [12, 15], введем интенсивность путевого отбора газа на участке $m = nq/L$ с единицей измерения $\text{кг с}^{-1} \text{ м}^{-1}$: $mL = nq$. Тогда последнее равенство принимает вид:

$$p_K^2 = p_H^2 - \frac{\lambda}{A^2 D^5} \left(M_0^2 - 2M_0mL + \frac{1}{3}m^2L^2 + \Delta \right) L.$$

Полагаем, что количество узлов отбора увеличивается до бесконечности. Соответственно $n \rightarrow \infty$, $q \rightarrow 0$. Но, при этом соблюдается условие $nq \rightarrow mL = \text{const}$. В результате члены, обозначенные через Δ , превращаются в нуль ($\Delta \rightarrow 0$), и получим формулу

$$p_K^2 = p_H^2 - \frac{\lambda}{A^2 D^5} \left(M_0^2 L - 2M_0mL^2 + \frac{1}{3}m^2L^3 \right).$$

Из этой формулы, поскольку при выборе длины участка L не предпринимали какие-либо ограничения, можно перейти к длине x :

$$p^2(x) = p_H^2 - \frac{\lambda}{A^2 D^5} \left(M_0^2 x - 2M_0 m x^2 + \frac{1}{3} m^2 x^3 \right). \quad (2)$$

Фактически мы получили формулу расчета (2) для изменения давления при равномерном путевом отборе газа с постоянной интенсивностью m ($\text{кг м}^{-1} \text{с}^{-1}$), когда в непрерывных величинах система квазиодномерных уравнений имеет вид [1-3]:

$$\begin{cases} \frac{dp}{dx} + \rho \frac{\lambda w^2}{2D} = 0, \\ M = M_0 - mx = \rho w f, \quad p = z \rho R T. \end{cases} \quad (3)$$

Здесь и далее p , ρ , w , M – статическое давление, плотность, средняя скорость и массовый расход газа являются переменными координаты x ; M_0 – массовый расход газа во входном сечении; D , f , λ , $m = nq/L$ – внутренний диаметр, площадь живого сечения, коэффициент сопротивления и интенсивность отбора газа из 1 м длины участка в течении 1 секунда времени – постоянные величины. Полагаем также, что коэффициент сверхсжимаемости Z , температура T и удельная газовая постоянная R газа имеют постоянные (осредненные) значения.

При известном распределении давления газа по длине сети плотность газа определяется по формуле:

$$\rho(x) = p(x) / (zRT), \quad (4)$$

а скорость газа – по формуле:

$$u(x) = \frac{M_0 - mx}{f \rho(x)}. \quad (5)$$

4. Результаты

Для выяснения роли отдельных параметров на формировании давления, плотности и скорости газа по длине участка был проведен вычислительный эксперимент для различных исходных параметров.

Был проведен вычислительный эксперимент по формуле (1) для короткого горизонтального трубопровода ($L = 2000$ м). В качестве исходных данных принимали $D = 408$ мм, $k = 0.005$ м (эквивалентная шероховатость живого сечения трубопровода), $T = 293.15$ К. В первом варианте входной массовый расход составлял $M_0 = 4, 6, \dots, 20$ кг/с (это соответствует входной среднерасходной скорости $w_0 = 5.3464$ м/с). Значение 0.02058 коэффициента сопротивления, вычисляли по формуле Шифринсона: $\lambda = 0.11(k/D)^{0.25}$.

На рис. 1 представлены результаты расчета, полученные при $p_H = 1.0$ МПа. Как видно из графиков, с увеличением массового расхода давление убывает интенсивнее. Все эти кривые, как уже отметили выше, представляют верхней ветви квадратичной параболы с вершиной в положительной части оси абсцисс.

Кривые плотности (в расчетах значение плотности газа прямо пропорциональна значению давления с коэффициентом $(z\rho R)^{-1}$ расположились в промежутке 6.85–7.16 кг/м³. При больших значениях массового расхода заметна слабая выпуклость графиков давления и скорости (рис. 2). При уменьшении массового расхода скорость газа интенсивно возрастает вниз по потоку. Этим и объясняется возрастание скорости убывания давления по направлению потока. Практика

показала, что нельзя допустить этого, так как при больших скоростях потока газа ($> 20 \text{ м/с}$) образуются высокочастотные шумы и вибрации в трубопроводе [2].

Следующий расчет касался случая различных значений входного давления.

Характер путевого изменения давления в зависимости от входного его значения для массового расхода газа $M_0 = 15 \text{ кг/с}$ представлен на рис. 3.

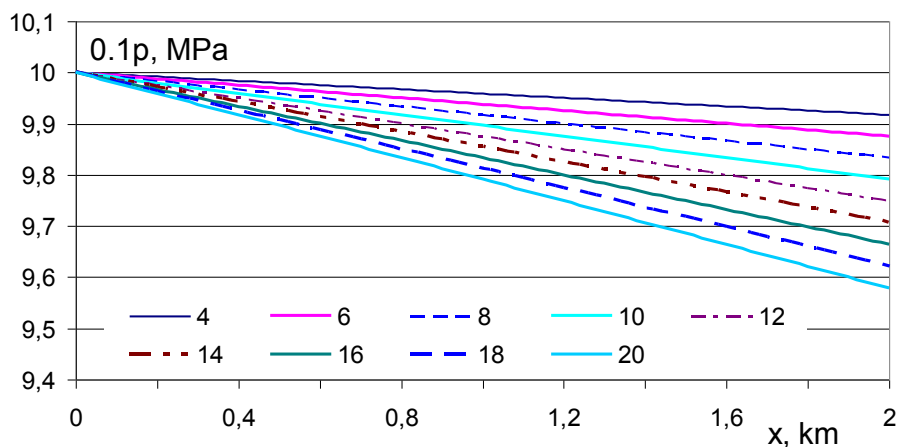


Рисунок 1. Путовое изменение давления при различных значениях массового расхода газа (кг/с). $L = 2000 \text{ м}$, $D = 408 \text{ мм}$, $k = 0.005 \text{ мм}$, $p_H = 1.0 \text{ МПа}$, $\lambda = 0.02058$

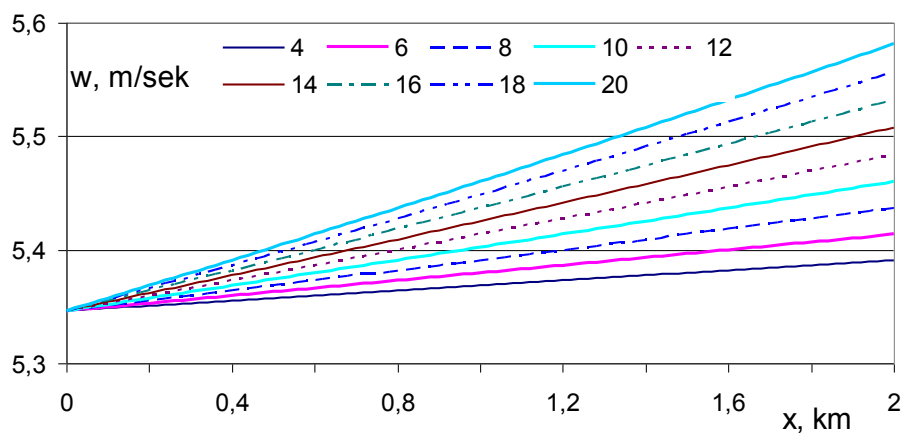


Рисунок 2. Путовое изменение скорости потока газа при различных значениях массового расхода газа (кг/с). $L = 2000 \text{ м}$, $D = 408 \text{ мм}$, $k = 0.005 \text{ мм}$, $p_H = 1.0 \text{ МПа}$, $\lambda = 0.02058$

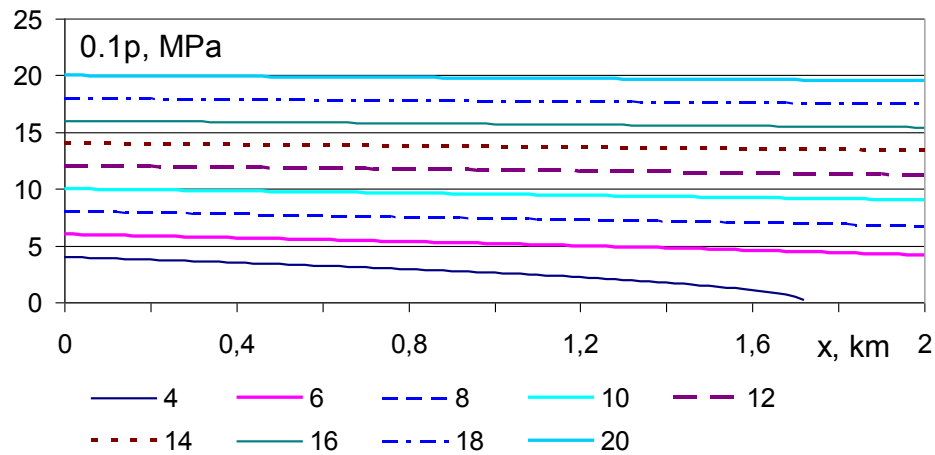


Рисунок 3. Путевое изменение давления при различных значениях входного давления (0.1 МПа). $L = 2000$ м, $D = 408$ мм, $k = 0.005$ мм,

$$M_0 = 15 \text{ кг/с}, \lambda = 0.02058$$

Верхние кривые для $p_H = 2.0 - 0.6$ МПа количественно подтвердили достоверность суждения о том, что с приближением к вершине параболы ускоряется падение давления. А кривая давления при $p_H = 0.4$ МПа урезана, т.к. такое входное давление при $M_0 = 15$ кг/с не обеспечивает транспортировки газа на расстояние $L = 2000$ м.

На рис. 4 приведены соответствующие этим данным распределения скорости потока по длине участка. Характер изменения скорости по длине и по входному давлению объясняется следующим образом. При малых давлениях потенциальная энергия сжатия (давление) быстрее переходит к кинетической энергии газа, т.е. интенсивно увеличивается скорость потока. Это, в свою очередь, усиливает потери потенциальной энергии сжатия и, таким образом, образуется причинно-следственный замкнутый круг процесса.

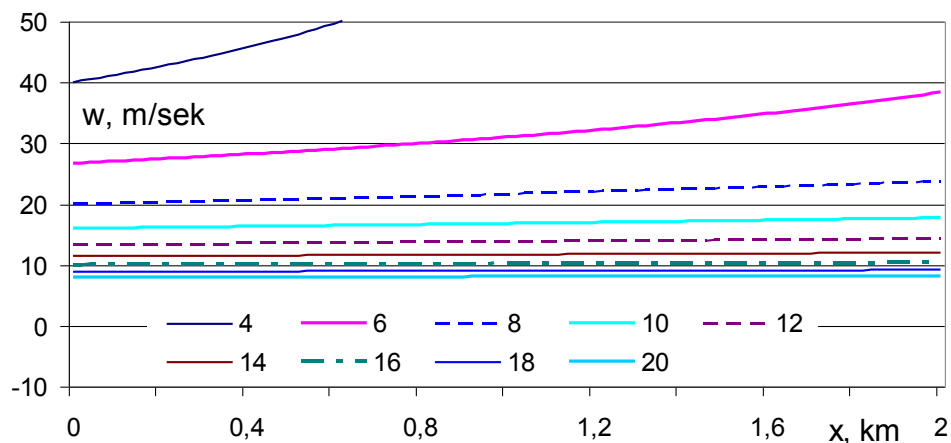


Рисунок 4. Путевое изменение скорости газа при различных значениях входного давления (0.1 МПа). $L = 2000$ м, $D = 408$ мм, $k = 0.005$ мм,

$$M_0 = 15 \text{ кг/с}, \lambda = 0.02058$$

Следующие расчеты проводились с учетом равномерного путевого отбора газа. Значение интенсивности путевого отбора соответствовала тупиковому режиму течения: $m = M_0 / L$. Изменения давления для этого случая представлены на рис. 5.

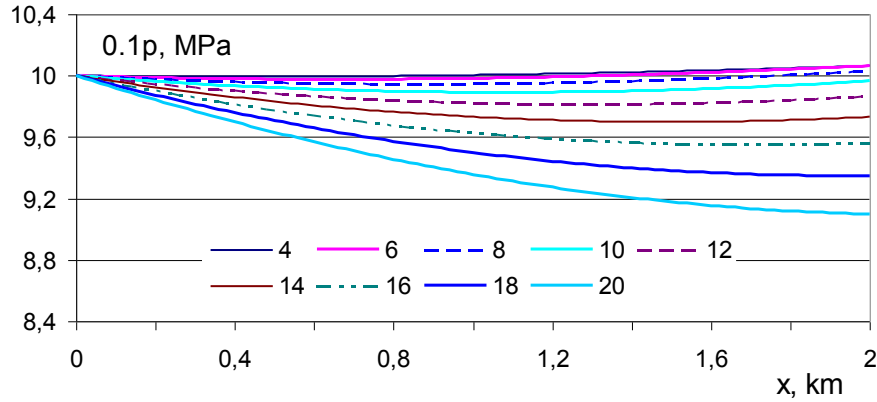


Рисунок 5. Путевое изменение давления при различных значениях входного массового расхода газа (M_0 , кг/с). $L = 2000$ м, $D = 408$ мм, $k = 0.005$ мм, $p_H = 1.0$ МПа, $\lambda = 0.02058$, $m = M_0 / L$

Верхние кривые, которые получены при малых значениях входного массового расхода газа, являются немонотонные. Такой факт известен в аэродинамике, когда производится отбор газа через перфорированную стенку [23]. Если увеличить интенсивность отбора, то сначала падение давления вдоль по потоку уменьшается. А при дальнейшем увеличении интенсивности отбора образуется положительный градиент давления.

При больших значениях массового расхода получены монотонно убывающие кривые давления. Т.е. при малых числах точек подключения потребителей подход путевого отбора не работает. Для того чтобы применить данный подход, необходимо выделить участки применимости подхода с убыванием давления.

Согласно нашему решению, путевое убывание давления происходит при выполнении условия:

$$\frac{dp^2(x)}{dx} = -\frac{\lambda}{A^2 D^5} (M_0^2 - 4M_0 m x + m^2 x^2) < 0,$$

т.е. при обозначении $\beta = m / M_0$ требуется выполнения условия:

$$\beta^2 x^2 - 4\beta x + 1 > 0.$$

Нулевые значения данного выражения наступают при $(\beta x)_{1,2} = 2 \pm \sqrt{3}$. Поэтому подход путевого отбора, с учетом максимального значения координаты $x = L$, можно применять при условии $m < (2 - \sqrt{3}) \frac{M_0}{L}$, либо при условии $(2 + \sqrt{3}) \frac{M_0}{L} < m$. Иначе образуются участки с возрастанием давления, что противоречит физике процесса, как это получено на рис. 5.

Рис. 6 демонстрирует поведение скорости газа при путевом отборе, которые получены для тупикового режима.

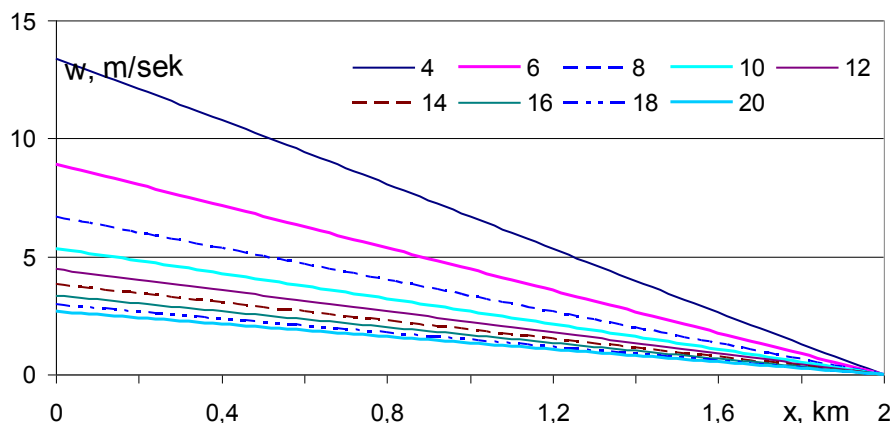


Рисунок 6. Путевое изменение скорости газа при различных значениях входного массового расхода газа давления ($\text{кг} / \text{с}$) • $L = 2000 \text{ м}$,

$$D = 408 \text{ мм}, p_H = 1.0 \text{ МПа}, \lambda = 0.02058$$

Общей чертой их является нулевое значение в конце участка. Наибольшие скорости газа образуются, когда массовый расход газа имеет малое значение.

5. Анализ результатов

В ходе проведенного вычислительного эксперимента для короткого горизонтального трубопровода получены результаты путевого изменения давления и скорости потока газа при различных значениях массового расхода газа и входного давления, а также результаты путевого изменения давления при различных значениях входного массового расхода газа и путевого изменения скорости газа.

6. Заключение

Основными результатами работы являются следующие:

- с привлечением подхода равномерного путевого отбора газа из трубопровода разработана квазиоднородная математическая модель для квадратического закона сопротивления при изотермическом случае и постоянном уклоне трассы от горизонта;
- новым способом получена формула для давления при равномерном путевом отборе газа в квадратичном законе сопротивления;
- выделена область применения равномерного путевого отбора, которая уместна для большого числа подключения потребителей и при больших значениях входного давления;
- проведен вычислительный эксперимент для горизонтального газопровода без путевого отбора, результаты которого подтверждают правильность тенденции развития трубопроводной транспортировки различных сред: по мере возможностей необходимо перейти к высоким рабочим давлениям и большим диаметрам. Этим достигается сокращение эксплуатационных расходов при функционировании газопроводов;
- получены графические зависимости путевого изменения давления и скорости потока газа при различных значениях массового расхода газа и при различных значениях входного давления.

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ПРИМЕНЕНИЕ МЕТОДА ПРЯМЫХ ПРИ ИЗУЧЕНИИ РАСПРОСТРАНЕНИЯ ТЕПЛОВЫХ ВОЛН В ПРЯМОУГОЛЬНОЙ ОБЛАСТИ

Хужаев Ж.И.¹, Шаймов К.М.², Эшмуродов М.Х.²

¹НИИ развития цифровых технологий и искусственного интеллекта, Ташкент, Узбекистан

²Самаркандский архитектурно-строительный университет, Самарканд, Узбекистан

E mail: jamolkhojaevnms@gmail.com

Аннотация: Метод прямых модифицирован для решения линейного двухмерного уравнения параболического типа с разрывными граничными условиями в декартовых координатах. Приближенно-аналитический метод применим для задания начального и граничных условий в виде произвольных функций. Демонстрированы особенности процесса распространения тепловых волн в прямоугольной области, где большую температуру имеют одна или две стороны расчетной области. Показаны способы применения метода для решения двухмерных уравнений эллиптического и гиперболического типов.

Ключевые слова: теплопередача, задача Дирихле, метод прямых, собственные числа и векторы, тепловая волна, замена переменных, формулы перехода.

1. Введение

Развитие численных методов решения многомерных задач обусловлено тем, что с одной стороны решаемые задачи выражены адекватными линейными и нелинейными уравнениями и их краевыми условиями, а с другой стороны год за годом увеличиваются ресурсы вычислительной техники и технологий до привлечения кластеров и графических процессоров.

Для решения задач теплопередачи разработаны различные точные и приближенные методы [1-4]. В работе [5] приведены многочисленные аналитические решения задач теплопередачи в стационарной постановке в декартовых, цилиндрических и сферических координатах. За последние годы интенсивно развиваются численные методы – метод конечных разностей, метод конечных элементов, метод Бубнова-Галеркина, метод Рвачева и другие применительно к параболическим или гиперболическим уравнениям теплопередачи [3].

Особый класс задач образуется при реализации метода конечных разностей, которые имеют единую трехдиагональную матрицу при переходе к конечно-разностным уравнениям для произвольного временного шага:

$$A = \begin{pmatrix} -2 & 1 & 0 & 0 & \dots & 0 & 0 & 0 \\ 1 & -2 & 1 & 0 & \dots & 0 & 0 & 0 \\ 0 & 1 & -2 & 1 & \dots & 0 & 0 & 0 \\ \dots & & & & \dots & & & \\ 0 & 0 & 0 & 0 & \dots & 1 & -2 & 1 \\ 0 & 0 & 0 & 0 & \dots & 0 & 1 & -2 \end{pmatrix}_N \quad (1)$$

Такая матрица образуется при решении задач Дирихле относительно уравнений эллиптического, параболического и гиперболического типов, когда оператор лапласиана в уравнении имеет постоянный коэффициент при использовании метода прямых. Система конечно-разностных уравнений с такой основной матрицей решается многократным повторением в зависимости от реального (при решении параболических и гиперболических уравнений) или фиктивного (при решении эллиптического уравнения) времени.

Метод прямых формировался и развивался как способ перехода из многомерных уравнений в частных производных к системе обыкновенных дифференциальных уравнений, для решения которых были разработаны различные методы. В частности, в работе [6] демонстрированы примеры использования метода прямых для треугольной области, области в виде сегмента и других. В области решения задачи проводятся равноудаленные прямые и совместно уравнения в частных производных образует система обыкновенных дифференциальных уравнений, количество которых равно количеству проведенных прямых.

Различные двух- и трехмерные задачи в декартовых, цилиндрических и сферических ортогональных координатах в работе [7] рассмотрены и решены методом прямых. Демонстрированы особенности решения многомерных задач методом прямых. Сначала решается задача относительно первой координаты, для которой составляется матричное уравнение, определяются вспомогательные фундаментальная и диагональная матрица, осуществляется переход к линейной комбинации искомым функций, формируются соответствующие краевые условия и получается решение по данной координаты. В следующем шаге процедура повторяется для второй координаты и т.д.

Анализ результатов работы [7] показывает, что использование метода прямых при решении двухмерной задачи по одной из координат, когда по второй координате использован простой метод прогонки, также обеспечивает согласования результатов для фиксированного времени без введения промежуточных массивов искомой функции. Аналогичное можно утверждать также для решения трехмерной задачи: применение метода прямых для двух координат и метода простой прогонки по третьей координате также обеспечивает согласования результатов расчета.

В работе [8] предложен новый усовершенствованный метод прямых, который позволяет понижение размерности многомерных задач строительной механики при совместном использовании с методом проекционный метод Бубнова-Галеркина-Петрова. Метод применяется для расчета толстых пластин, пластин переменной толщины, неоднородных и многослойных пластин. Предлагаются конечно-разностные соотношения заменить проекционными, что расширит возможности метода прямых и позволит использовать метод в задачах динамики.

Работа [9] посвящена разработке модификации метода прямых для решения Кортевега-де Фриза.

В работе [10] метод прямых использован при решении уравнения диффузии дробного порядка производной, которые в настоящее время широко применяются при описании широкого класса физических и химических процессов, протекающих во фрактальных средах, при математическом моделировании экономических и социально-биологических процессов. Приведены результаты численного моделирования первой краевой задачи. Способ применения метода прямых в случае дробных производных приведен также в работах [11,12].

В работе [13] предложены варианты метода прямых для решения задач Робина, Неймана и периодических задач в одно-, двух- и трехмерных постановках.

Статья [14] предложена нелинейная модель мелкой воды для изучения распространения волн цунами в различных точках вдоль берега океана. При решении задач использован метод прямых. При моделировании учтены береговые склоны и глубина океана, высота и скорость набега волны цунами. Точность математической модели проверена путем решения классической тестовой задачи с известным аналитическим решением. Вычисленные высота наката и скорость удовлетворительно согласуются с физика волны цунами. Аналогичная задача рассмотрена в работе [15], где предлагается новый метод определения местоположения гибридных линий на основе бегущей волны разлома. Задача прогноза цунами на берегах Бангладеша решена в работе [16] с применением метода дифференциально-разностного метода.

Разработаны варианты метода прямых для смещенных граничных условий первого и третьего родов [17,18].

В целом, метод прямых широко используется во всех научных направлениях, где моделирование объекта приводит к уравнениям в частных производных, начиная от простых задач механики до задач многопроводных линиях передачи энергии, электромагнитных явлений, нелинейных уравнений Шрёдингера, и вплоть до социальных задач.

Применяемые при решении многомерных уравнений математической физики многочисленные способы расщепления, метод предиктор-корректор, метод переменных направлений и другие методы имеют общий недостаток – необходимо согласовать результаты факторизации (прогонки) по разным направлениям (факторам). В связи с этим при изучении

распространения тепловых волн в прямоугольнике обратимся к методу прямых – представителю дифференциально-разностных методов [19-20], который лишен от этого недостатка.

2. Постановка задачи

Применение обобщенной функции ζ в задачах теплопередачи [5] позволяет получить и качественные, и количественные результаты. При изложении материала параграфа в качестве такой функции использована относительно-избыточная температура ζ , а координаты приведены к безразмерному виду с привлечением длины участка l_x . В связи с этим прямоугольник с размерами (l_x, l_y) в расчетах принимается с размерами $(1, l)$ (в нашем случае $l=1$).

Рассмотрим следующую задачу, где коэффициент теплопроводности, плотность и приведенная теплоемкость тела имеют постоянные значения.

Прямоугольное тело имеет размеры l_x, l_y . Начальное значение относительно-избыточной температуры (в дальнейшем сокращенно температуры) – нулевое. Процесс начинается с того, что в одной или двух сторон прямоугольника устанавливается температура 1. Остальные стороны прямоугольника имеют нулевое значение. При такой постановке уравнение теплопередачи имеет вид:

$$\frac{\partial \zeta}{\partial t} = \frac{\partial^2 \zeta}{\partial x^2} + \frac{\partial^2 \zeta}{\partial y^2} + f'(x, y, t),$$

где $\zeta(x, y, t)$ – относительно-избыточная температура; $f'(x, y, t)$ – приведенная по плотности и удельной теплоемкости материала мощность источника тепла в массе, которая может быть непрерывной или разрывной функцией; t – произведение коэффициента температуропроводности на реальное время, имеет размерность m^2 .

Координаты x и y обезразмерены относительно длины l_x , а начальным условием задачи служит:

$$\zeta(x, y, 0) = \zeta^0(x, y),$$

а граничными условиями –

$$\begin{aligned} \zeta(0, y, t) &= \mu_{x0}(y), \quad \zeta(1, y, t) = \mu_{xl}(y), \\ \zeta(x, 0, t) &= \mu_{y0}(x), \quad \zeta(x, l, t) = \mu_{yl}(x). \end{aligned}$$

Начальное и граничные условия могут быть и ненулевыми. Но нас интересуют тепловые волны, которые характеризуются высокими тепловыми напряжениями на теле, многократное повторение которых может привести к разрушению целостности объекта.

Требуется разработать алгоритм решения задачи методом прямых.

3. Материалы, методы и объект исследования

Вводится равномерная сетка по координатам и времени с шагами h_x, h_y, τ и индексами $i = 0..N_x + 1, j = 0..N_y + 1, n = 0..N_t$.

Аппроксимируя уравнение по координате x и соответствующие граничные условия, составляется матричное уравнение

$$\frac{\partial Z_j}{\partial t} = \frac{1}{h_x^2} A_j^{(x)} Z_j + \frac{\partial^2 Z_j}{\partial y^2} + F_j, \quad (2)$$

где

$$Z_j = (\zeta_{1,j}^{n+1}, \zeta_{2,j}^{n+1}, \dots, \zeta_{N_x-1,j}^{n+1}, \zeta_{N_x,j}^{n+1})^T,$$

$$F_j = \left(f_{1,j}' + \frac{\mu_{0,j}^{n+1}}{h_x^2}, f_{2,j}', \dots, f_{N_x-1,j}', f_{N_x,j}' + \frac{\mu_{N_x+1,j}^{n+1}}{h_x^2} \right)^T = (f_{1,j}, f_{2,j}, \dots, f_{N_x-1,j}, f_{N_x,j})^T.$$

Здесь $(\cdot)^T$ – операция транспонирования матрицы, т. е. Z_j и F_j являются вектор-столбцами.

Матрица $A_j^{(x)}$ имеет структуру (1) и размерность N_x . В связи с этим только член с $A_j^{(x)}Z_j$ представляет матрицу, а остальные члены уравнения (2) являются вектор-столбцами.

Аппроксимируя производные, при фиксированных значениях j можно задачу решить методом прогонки. Но при этом значения членов $\frac{\partial Z_j}{\partial t}$, $\frac{\partial^2 Z_j}{\partial y^2}$ и F_j берутся из предыдущего временного шага (полушага). Соответственно, процесс придется повторить для второй координаты. В результате образуется классический метод факторизации, в котором результаты первого и второго полушагов будут разные. В связи с этим при решении эллиптического уравнения организуется итерационный процесс, чтобы согласовать результаты первого и второго полушагов. Метод прямых позволяет исключить ошибки решения связанной введением искомых в промежуточном полушаге. Для этого необходимо выделить отдельные уравнения из системы (2).

Для выделения отдельных уравнений из этого матричного уравнения, умножим (2) слева на матрицу B_x^{-1} , которая представляет обратной матрицы фундаментальной матрицы B_x с элементами $b_{s,p}^{(x)} = (-1)^{s+p} \sqrt{\frac{2}{N_x+1}} \sin \frac{\pi s p}{N_x+1}$, являющиеся элементами собственных векторов матрицы $A_j^{(x)}$. С учетом $A_j^{(x)} = B_x \Lambda_x B_x^{-1}$, где Λ_x представляет диагональную матрицу с элементами $\lambda_s^{(x)} = -2 \left(1 + \cos \frac{\pi s}{N_x+1} \right)$ собственных значений матрицы $A_j^{(x)}$ [21]. Тогда матричное уравнение принимает вид:

$$\frac{\partial \tilde{Z}_j}{\partial t} = \frac{1}{h_x^2} \Lambda_x \tilde{Z}_j + \frac{\partial^2 \tilde{Z}_j}{\partial y^2} + \tilde{F}_j,$$

где

$$\begin{aligned} \tilde{Z}_j &= B_x^{-1} Z_j = B_x Z_j = (\tilde{\zeta}_{1,j}^{n+1}, \tilde{\zeta}_{2,j}^{n+1}, \dots, \tilde{\zeta}_{N_x-1,j}^{n+1}, \tilde{\zeta}_{N_x,j}^{n+1})^T, \\ \tilde{F}_j &= B_x^{-1} F_j = B_x F_j = (\tilde{f}_{1,j}^{n+1}, \tilde{f}_{2,j}^{n+1}, \dots, \tilde{f}_{N_x-1,j}^{n+1}, \tilde{f}_{N_x,j}^{n+1})^T. \end{aligned}$$

Из последнего матричного уравнения выделяется уравнение

$$\frac{\partial \tilde{\zeta}_{i,j}}{\partial t} = \frac{1}{h_x^2} \lambda_i^{(x)} \tilde{\zeta}_{i,j} + \frac{\partial^2 \tilde{\zeta}_{i,j}}{\partial y^2} + \tilde{f}_{i,j}^{n+1}.$$

Преобразованными граничными условиями данных уравнений служат

$$\tilde{\mu}_{i,0} = \tilde{\zeta}_{i,0} = \sum_{p=1}^{N_x} b_{i,p}^{(x)} \mu_{p,0}^{n+1}, \quad \tilde{\mu}_{i,N_y+1} = \tilde{\zeta}_{i,N_y+1} = \sum_{p=1}^{N_x} b_{i,p}^{(x)} \mu_{p,N_y+1}^{n+1}.$$

Аналогично преобразуются начальные условия для уравнений $\tilde{\zeta}_{i,j}$.

Повторным применением описанной выше процедуру метода прямых с помощью матриц B_y и Λ_y приходим к отдельным уравнениям

$$\frac{\partial \check{\zeta}_{i,j}}{\partial t} = \frac{1}{h_x^2} \lambda_i^{(x)} \check{\zeta}_{i,j} + \frac{1}{h_y^2} \lambda_j^{(y)} \check{\zeta}_{i,j} + \check{f}_{i,j}.$$

Отсюда, используя схему назад по времени, переходим к дискретному представлению производной по времени и находим значения вновь введенных функций

$$\check{\zeta}_{i,j}^{n+1} = \frac{\check{\zeta}_{i,j}^n + \tau_n \check{f}_{i,j}^{n+1}}{1 - \tau_n \left(\frac{\lambda_i^{(x)}}{h_x^2} + \frac{\lambda_j^{(y)}}{h_y^2} \right)}. \quad (3)$$

Формирование начального условия для вновь введенной функции осуществляется по формуле

$$\check{\zeta}_{i,j}^0 = \sum_{p=1}^{N_x} b_{i,p}^{(x)} \sum_{q=1}^{N_y} b_{j,q}^{(y)} \zeta_{p,q}^0,$$

а обратный переход к относительно-избыточной температуре – по формуле

$$\zeta_{i,j}^{n+1} = \sum_{p=1}^{N_x} b_{i,p}^{(x)} \sum_{q=1}^{N_y} b_{j,q}^{(y)} \check{\zeta}_{p,q}^{n+1}.$$

Таким образом, используя положений метода прямых, исключили невязку, которая образуется при использовании методов факторизации и прогонки.

4. Анализ результатов

По представленному материалу была составлена программа и проведен вычислительный эксперимент.

Отметим, что точность аппроксимации составляет $O(\tau + h_x^2 + h_y^2)\tau = 0.01$. В нашем случае безразмерные шаги имели значения $\tau = 0.01$, $h_x = h_y = 0.05$. Точность аппроксимации можно увеличить, например, используя, например, неявную по времени схему или уменьшив шаги численного интегрирования. Но основным успехом этого метода, как уже отметили, является точное, в рамках машинных округлений, решение конечноразностных уравнений.

Остановимся на некоторые результаты вычислительного эксперимента.

В качестве задача 1 рассмотрим случай с краевыми условиями

$$\zeta^0(x, y) = 0, \mu_{x0}(y) = 0, \mu_{xl}(y) = 0, \mu_{y0}(x) = 0, \mu_{yl}(x) = 1.$$

Т.е. начиная от $t=0$ на границе $y=1$ устанавливается температура 1. Начальная температура и на остальных границах температура нулевая.

На рис. 1 приведены изотермы, полученные при $t = 50 (m^2)$. Нулевые граничные условия на боковых стенках постепенно сужают фронта тепловой волны, образуя своеобразные «языки» высокой температуры.

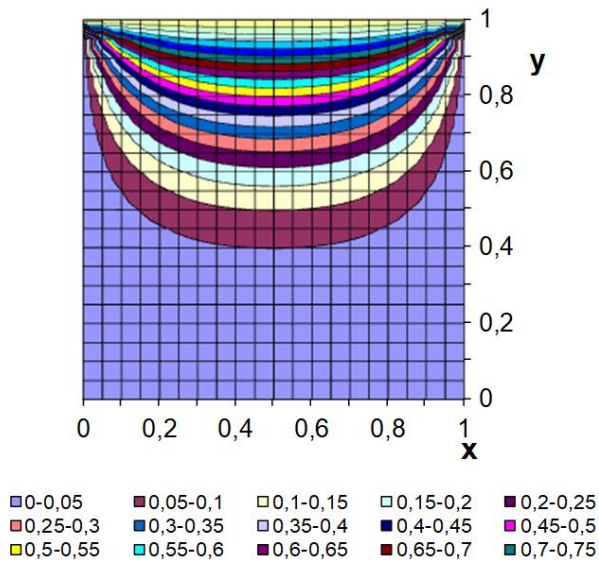


Рисунок. 1. - Изотермы в квадратичной области для случая распространения тепловой волны из верхней границы расчетной области при $t = 50 \text{ (м}^2\text{)}$

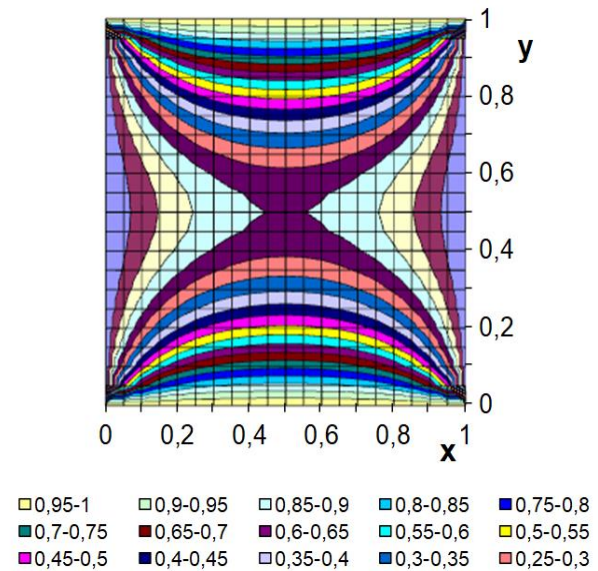


Рисунок. 2. - Изотермы в квадратичной области для случая распространения тепловой волны из верхней и нижней границ расчетной области при $t = 140 \text{ (м}^2\text{)}$

Данный фактор более ярко выражается при распространении встречных волн, когда $\mu_{y0}(x)=1$, $\mu_{yl}(x)=1$. На рис. 2 приведены изотермы, полученные для этого случая при $t = 140 \text{ (м}^2\text{)}$.

Аналогичные расчеты проводились для случаев, когда граничные значения температуры 1 относились границам $x=0$ и $y=1$, и когда на границах $y=0$ температура имеет значение 1, а в остальных сторонах квадрата – нулевое значение. Представление результатов расчета в виде изотерм в расчетной области позволяет выделить зон высокого температурного напряжения. Они характеризуются сгущением изотерм. Таковыми являются точки границы области расчета, где происходит скачкообразный переход к другому значению температуры.

Эти результаты представляют динамику тепловой волны в области формы квадрата. В литературе по теории теплопередачи зачастую рассматриваются и решаются задачи в стационарной постановке при $\frac{\partial \zeta}{\partial t} \rightarrow 0$. Т.е. решается эллиптическое уравнение при независимых от времени граничных условиях. Численное решение таких задач производится с введением фиктивного времени, который известен как метод установления решения. С этой точки зрения в рамках условия точности

$$\max_{i,j} |\zeta_{i,j}^{n+1} - \zeta_{i,j}^n| < 0.001$$

решение эллиптического уравнения получается при $n=1400$ шагов по времени при распространении тепла только с одной стороны.

Для решения эллиптического уравнения для квадратичной области также составлены алгоритм и программа расчета, где формула (3) принимается в виде $\check{\zeta}_{i,j} = -\frac{\check{f}_{i,j}^{n+1}}{\lambda_i^{(x)} / h_x^2 + \lambda_j^{(y)} / h_y^2}$,

а алгоритм с соответствующими поправками. Результаты, которые получены при граничном значении температуры 1 для $x=0$ и $y=1$ и когда на границах $y=0$ температура имеет значение 1, а в остальных сторонах квадрата – нулевое значение, приведены на следующих двух рисунках

в форме изолиний. Заметим, что в отличие от известных методов в данном случае приведенный алгоритм использован только один раз.

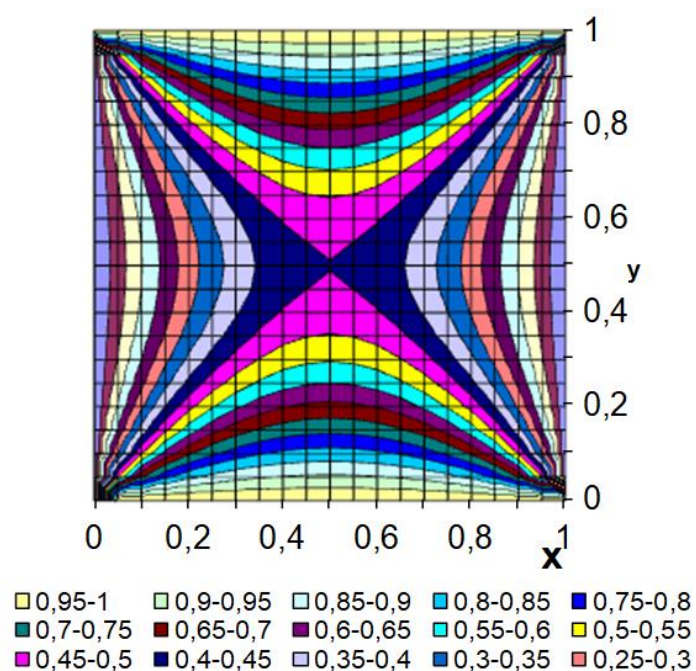


Рисунок 3. - Изотермы для случая распространения встречных тепловых волн, полученные при решении эллиптического уравнения методом прямых

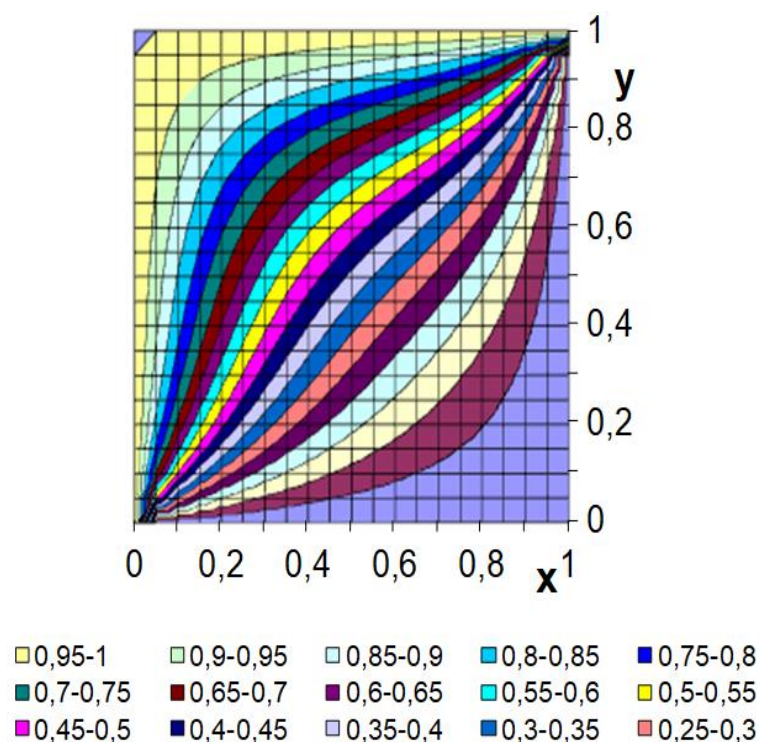


Рисунок 4. - Изотермы для случая распространения тепловых волн из левой и верхней границ при решении эллиптического уравнения методом прямых

5. Заключение

На основе метода прямых разработаны алгоритмы решения параболических и эллиптических уравнений теплопроводности для области в виде единичного квадрата. Точность аппроксимации метода составляет $O(\tau + h_x^2 + h_y^2)$. Алгоритм можно применять также для решения уравнения гиперболического типа – для этого случая формула (3) принимается в виде

$$\tilde{\zeta}_{i,j}^{n+1} = \frac{2\tilde{\zeta}_{i,j}^n - \tilde{\zeta}_{i,j}^{n-1} + \tau_n \tilde{f}_{i,j}^{n+1}}{1 - \tau_n (\lambda_i^{(x)} / h_x^2 + \lambda_j^{(y)} / h_y^2)}$$
 и проводятся соответствующие видоизменения последующих формул.

Алгоритм использовали для изучения особенностей динамики распространения одно- и двусторонних тепловых волн в области единичного квадрата. Выявлено, что в рассмотренных задачах большие градиенты температуры ожидаются в точках границы области, где происходит разрыв температуры.

Доказано, что применение метода прямых при решении эллиптического уравнения позволяет многократно сокращает объем вычислений, так как не привлекается метод итераций. Это полезно, в частности, для решения задач гидродинамики в представлении вихрь-функция тока.

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Section E - Dynamics and strength of machines Секция Е – Динамика и прочность машин

KINEMATICS OF A COMBINED GEAR-LEVER DIFFERENTIAL TRANSMISSION MECHANISM

Abdulkarimov A.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan
E-mail: aabdusalam54@gmail.com*

Abstract. *In the past, researchers in the theory of mechanisms and machines have studied lever, cam, simple gear, gear planetary, gear differential and gear-lever differential mechanisms. Today, researchers around the world are showing great interest in combined, especially gear-lever combined and gear-lever combined with non-circular gears mechanisms. Compared with other mechanisms, gear-lever combination and gear-lever combination with non-circular gears mechanisms can realize more diverse movements and meet more special kinematic and dynamic requirements of different machines. Therefore, the creation and research and development of methods for calculating combined gear-lever mechanisms and their modification is an urgent problem in the field of the theory of mechanisms and machines.*

Keywords: *kinematics, mechanism, combined, gear-lever, differential, gear circuit, lever circuit.*

1. Introduction.

The development of methods for structural, kinematic, dynamic research and synthesis of newly developed mechanisms is the primary objective of the course on the theory of mechanisms and machines. Structural, kinematic, dynamic analysis and synthesis of the mechanism using the developed research methods are the main necessary sections in the analysis of mechanisms. An in-depth analysis of the existing mechanisms used in certain machines from the point of view of the completeness of the response to the requirements presented to these machines allows us to determine the advantages and disadvantages of these mechanisms. Creation of new mechanisms, taking into account the shortcomings of existing ones, and their use in machines makes it possible to increase the durability of the machine, improve the quality of the processed material and the progress of the technological process, and increase labor productivity.

We have developed a new combined gear-lever differential transmission mechanism (CGLDTM) and received a patent for the invention[1]. This CGLDTM can be used in two-roll technological machines, where the theory is currently being rapidly developed [2-10]. Early on, we created a number of gear-lever differential mechanisms and developed methods for their study [11-13].

Many scientists around the world are engaged in the creation and development of methods for studying combined gear-lever mechanisms. Han et al. and Sui et al. [14,15] proposed methods for synthesizing six and eight link linkages with four precise constraints and applied them to simulate the grasping movements of human fingers. These objects of research for the synthesis of solid-state guidance are flat four-bar, six-bar, eight-bar purely lever mechanisms, and combined gear-lever mechanisms consisting of gears, levers and a chain have been little studied.

Compared to flat linkage mechanisms, combination mechanisms of gears, cam arms or chains can realize a greater variety of movements and meet more complex special kinematic and dynamic requirements. Shao et al. [16] developed a method for synthesizing the trajectory of a cam seven-link mechanism with two degrees of freedom and applied it to a human lower limb gait rehabilitation device to simulate the normal trajectory of human gait. Sun et al [17] designed a human knee prosthesis based on a five-link gear-lever mechanism. This human knee prosthesis can precisely adjust the relative instantaneous center of rotation and accurately follow the trajectory of the ankle joint, and its bionic performance is much better than that of a four-link prosthetic knee. Modeler et al. [18] considered all types of five-link gear-lever mechanisms and proposed a synthesis method for calculating a given transfer function, which is often highly nonlinear. Mundo et al. in [19] proposed a method for accurately synthesizing the paths of a non-circular five-bar gear-lever mechanism with

non-circular gears. They also gave an example of the exact implementation of the synthesis of the path of this mechanism along two parallel lines.

After a large review of the literature, it can be seen that most of the existing research is done on linkage and gear-linkage mechanisms, and there are few research works on combined gear-linkage differential transmission mechanisms.

The article describes the devices, operating principles and methods of kinematic research of a combined gear-lever differential transmission mechanism. The developed technique was carried out using the centroid method using the instantaneous center of rotation of the mechanism links. The purpose of the kinematic analysis of the transmission mechanism is to determine the speeds and accelerations of characteristic points of the mechanism links, the gear ratio of the mechanism, and also to determine the degree of synchronism of rotation of the driving and driven links of the mechanism (working shafts of the roller technological machine where the transmission mechanism in question is used). Kinematic analysis was carried out when changing the interaxial distance of the driving and driven links of the mechanism. First, the lever contour of the mechanism is considered. In the lever contour of the mechanism, the geometric parameters of the levers, the position of the lever contour of the mechanism, as well as the magnitude and direction of the angular velocity of the driving link are known. The kinematic parameters of the lever circuit of the mechanism were determined analytically. Next, using the results of the lever contour of this mechanism, the gear contour of the mechanism was analytically studied. Mathematical models are derived to determine the kinematic parameters of the mechanism. It is shown that in an analytical study of the kinematics of a gear-lever mechanism, the derived mathematical models for determining the kinematic parameters make it possible to draw the necessary conclusions about the properties of the mechanism. The kinematic parameters of the entire mechanism are determined. It is shown that mathematical models for calculating the kinematic parameters of the mechanism are too cumbersome. Therefore, in order to clarify the issue only about the gear ratio, whether it changes with a change in the interaxial distance of the shafts or not, it is advisable to consider a combined planetary gear-lever mechanism, which is obtained on the basis of a gear-lever differential mechanism. It has been proven that in a combined gear-lever differential transmission mechanism, with a change in the interaxial distance of the driving and driven links, the gear ratio changes. For a mechanism with variable gear ratios, a picture is shown of an increase or decrease in the gear ratio depending on changes in the center distance and the direction of rotation of the drive gear. The useful application of variability and constancy of gear ratios of the considered mechanisms in two-roll modules is shown.

This section should review the literature and the relevance of the study. Next, the purpose of the study, the hypotheses and assumptions used are formulated, and the problem to be solved is determined.

2. Design and principle of operation of a combined gear-lever differential transmission mechanism.

CGLDTM belongs to the mechanical engineering industry, and can be used in technological machines with variable center distance of working shafts with large diameters.

In industries such as leather, agriculture, mining, metallurgy, construction and other industries, various transmission mechanisms between the working shafts are used in the roller machines with variable center distance of the working shafts. These mechanisms are used to transmit torque from one working shaft to another working shaft.

The VOPM-1800-K squeezing machine uses a transmission mechanism consisting of two gears, which are fixed at the output ends of the working shafts [20].

The disadvantages of such a transmission mechanism are the low quality of material extraction and low reliability of the machine, as well as the difficulty of using monchons of different thicknesses associated with the imperfection of this transmission mechanism.

In this squeezing machine, depending on the thickness of the material being processed, the interaxial distance of the working shafts changes, as a result of which the gears often disengage. This disengagement of gears leads to geometric sliding between the working shafts and the processed

material, which reduces the quality of the processed material and the reliability of the technological machine.

Gear-lever differential transmission mechanisms consisting of four sequentially contacting gears between each other are used in the TMPH 1800-K drawing-softening machine [20].

To use such a gear-lever differential transmission mechanism in process roll machines with large diameters of working shafts, it will be necessary to use gears with large diameters. This will lead to an increase in metal consumption and a sharp increase in the inertial force in these differential transmission mechanisms.

The task of a combined gear-lever differential transmission mechanism is to transmit torque from one working shaft to another working shaft in technological machines with large diameters of working shafts and with a large change in the center distance of these working shafts.

This problem is solved in the following way: in technological machines with large diameters of working shafts and with a large change in the interaxial distance of these working shafts, a combined gear-lever differential transmission mechanism is used, consisting from gears and sprockets with small diameters, as well as chains. Thus, this roller technological machine is provided with a rational combined gear-lever differential transmission mechanism. This differential transmission mechanism is designed to transmit torque from one working shaft to another working shaft of a technological machine with large diameter working shafts. It can also be used in technological machines with variable center distance of working shafts with small diameters of working shafts.

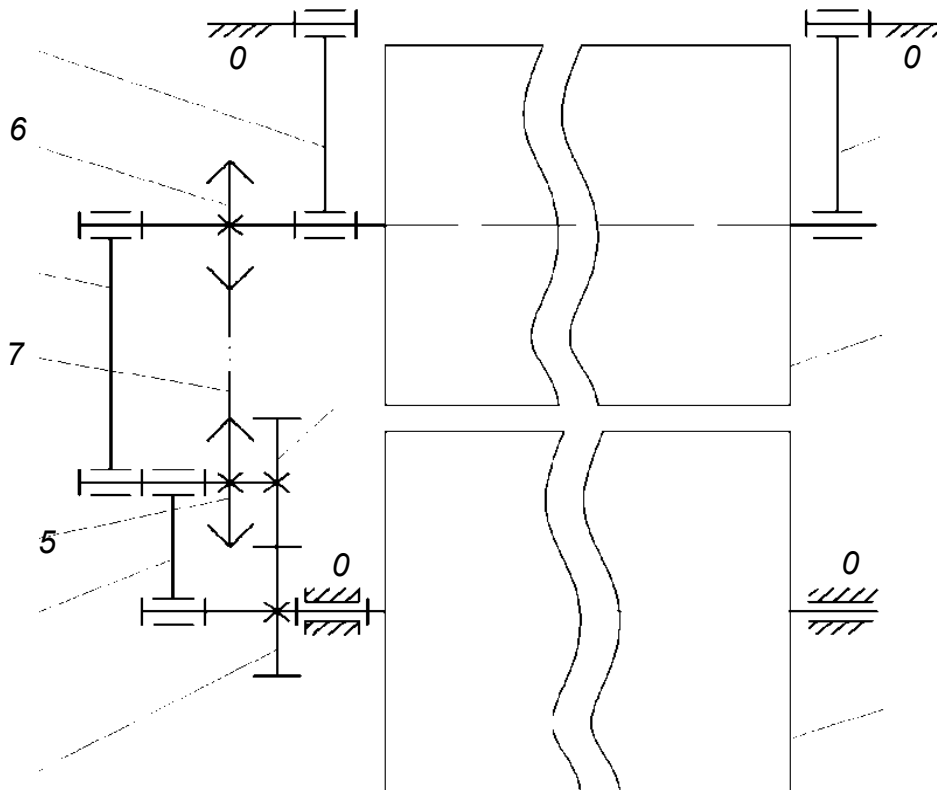


Figure 1. – Structural-kinematic diagram of a combined gear-lever differential transmission mechanism, side view. 1-drive working shaft, 2-driven working shaft, 3-drive gear, 4-drive gear, 5-drive sprocket, 6-driven sprocket, 7 chain, 8,9,10,11 levers.

Schemes of a combined gear-lever differential transmission mechanism designed to transmit torque from one working shaft to another working shaft of a technological machine with large diameter working shafts are shown in Figure 1 on the side, in Figure 2 on the front.

The combined gear-lever differential transmission mechanism consists of the following links: bed (fixed link) 0; driving shaft 1; driven working shaft 2; drive gear 3; intermediate gear 4; intermediate sprocket 5; driven sprocket 6; endless chain 7; levers 8, 9, 10 and 11.

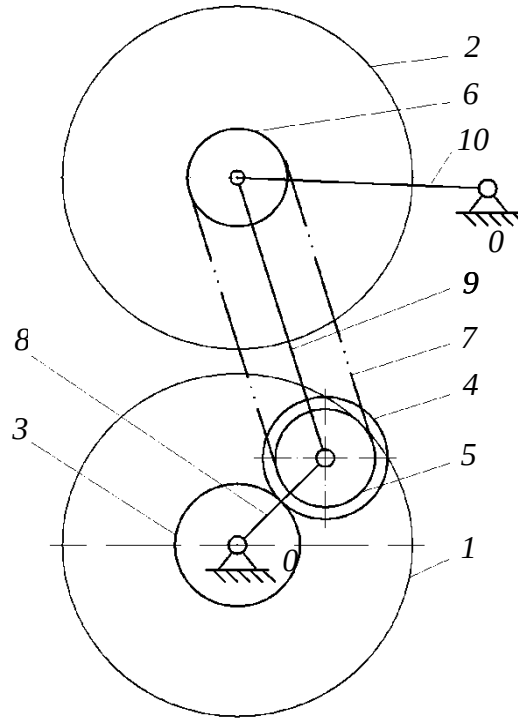


Figure 2. – **Structural-kinematic diagram of a combined gear-lever differential transmission mechanism, front view**

The combined gear-lever differential transmission mechanism works as follows. The torque from the driving shaft 1 to the driven shaft 2 is transmitted through the driving gear 3; intermediate gear 4; intermediate sprocket 5; endless chain 7; driven sprocket 6. When the processed material with variable thickness gets between the driving working shaft 1 and the driven working shaft 2, the interaxial distance $O_1 O_3$ (Figure 2). In this case, the O_2 and O_3 change their positions, the O_2 axis moves around the O_1 axis and the O_3 axis moves around the O_4 . Lever 8 ensures the constancy of the interaxial distance $O_1 O_2$ of the drive gear 3 and the intermediate gear 4, pivotally connecting the axes of these gears. Lever 9 ensures the constancy of the center distance $O_2 O_3$ of the intermediate sprocket 5 and the driven sprocket 6 by pivotally connecting the axes of these sprockets. An endless chain 7 transmits torque from the intermediate sprocket 5 to the driven sprocket 6. The driven sprocket 6, rigidly attached to the output end of the driven working shaft 2, transmits torque to this working shaft. Levers 10 and 11 pivotally connect the axes of the driven working shaft 2 and the driven sprocket 6 O_3 with the frame 0 via the O_4 .

3. Kinematics of a combined gear-lever differential transmission mechanism.

Kinematic analysis of the mechanism was carried out to determine the angular and linear velocities and accelerations of the mechanism links depending on the angular speed of the driving link (gear 1) and the linear speed and acceleration of the driving lever 7 (the center of rotation of the free working shaft (2). And also to determine the gear ratio of the mechanism at the moment of changing the interaxial distance of the working shafts and the linear speeds of the surfaces of the working shafts at the point of contact of these shafts with the material being processed (Figure 3 and 4). Kinematic analysis was performed using the centroid theory using the instantaneous center of rotation of the links.

Let gear 1 rotate with angular velocity, ω_1 and lever 7 move with speed $\bar{V}_{O_3}$. The position of the mechanism is determined by the angle φ .

From the geometric parameters of the mechanism, the length of the link 5 is known

$$O_2 O_3 = d_1 + d_2. \quad (1)$$

Link about 5 with a vertical line and it makes an angle α

$$\alpha = 90 - \varphi. \quad (2)$$

We determine the kinematic parameters of the lever circuit of the combined gear-lever differential transmission mechanism (Figure 3).

The distance between the axes of links 1 and 4 is determined by the expression

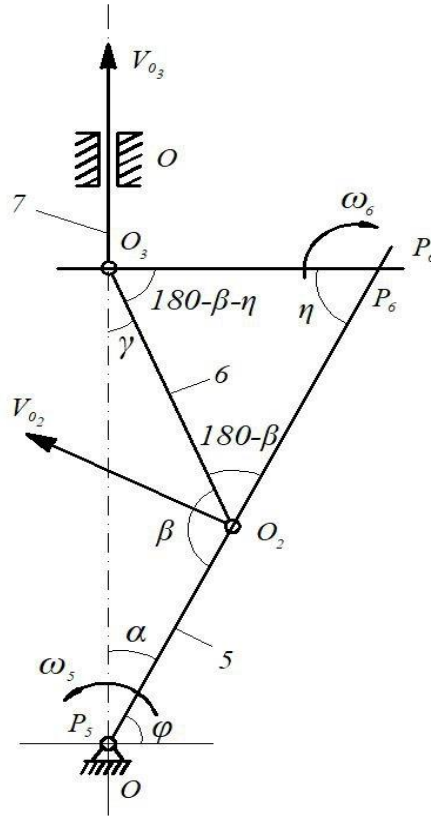


Figure 3. – Calculation diagram of the kinematics of the lever circuit of a combined gear-lever differential transmission mechanism

$$OO_3 = O_3O_2 \cdot \cos \gamma + OO_2 \cdot \cos \alpha. \quad (3)$$

Solving triangles OO_3P_6 and OO_2O_3 , we determine angles $\beta, \gamma, \alpha, \eta$.

$$\frac{\sin \alpha}{O_2O_3} = \frac{\sin \beta}{OO_3} = \frac{\sin \gamma}{OO_2}, \quad (4)$$

$$\frac{\sin \alpha}{O_2O_3} = \frac{\sin \gamma}{OO_2}, \quad (5)$$

$$\sin \gamma = \frac{OO_2 \cdot \sin \alpha}{O_2O_3}, \quad (6)$$

$$\gamma = \arcsin \frac{OO_2 \sin \alpha}{O_2O_3}, \quad (7)$$

$$\frac{\sin \beta}{OO_3} = \frac{\sin \gamma}{OO_2}, \quad (8)$$

$$\sin \beta = \frac{OO_3 \sin \gamma}{OO_2}, \quad (9)$$

$$\eta = 90 - \alpha, \quad (10)$$

$$\operatorname{tg} \alpha = \frac{O_3P_6}{OO_3}, \quad (11)$$

We determine the instantaneous centers of rotation of links 5 and 6 of the lever circuit of the combined gear-lever differential transmission mechanism. The instantaneous center of rotation of link 5 (P_5) coincides with the axis of the stationary working shaft (O).

We determine the lengths of the segments (O_3P_6) and (O_2P_6) showing the instantaneous centers of rotation of link 6 (P_6).

$$O_3P_6 = OO_3 \operatorname{tg} \alpha, \quad (12)$$

$$\sin \alpha = \frac{O_3P_6}{P_5P_6}, \quad (13)$$

$$P_5P_6 = \frac{O_3P_6}{\sin \alpha}, \quad (14)$$

$$O_2P_6 = P_5P_6 - OO_2. \quad (15)$$

We determine the instantaneous angular velocities of link 6 and 5 (ω_6) and (ω_5) as well as the instantaneous linear velocities of the centers of rotation of gear 2 and sprocket 3.

$$\omega_6 = \frac{V_{O_3}}{O_3P_6}, \quad (16)$$

$$\omega_6 = \frac{V_{O_2}}{O_2P_6}, \quad (17)$$

$$\frac{V_{O_2}}{O_2P_6} = \frac{V_{O_3}}{O_3P_6}, \quad (18)$$

$$V_{O_2} = \frac{V_{O_3} \cdot O_2P_6}{O_3P_6}, \quad (19)$$

$$V_{O_2} = \omega_5 \cdot OO_2, \quad (20)$$

$$\omega_5 = \frac{V_{O_2}}{OO_2}, \quad (21)$$

$$\omega_5 = \frac{V_{O_3} \cdot O_2P_6}{O_3P_6 \cdot OO_2}. \quad (22)$$

We will transfer certain angular and linear velocities of the lever contour of the combined gear-lever differential transmission mechanism to the gear contour of this mechanism (Figure 4). We determine the linear speed (V) of the contact point (n) of the gear wheels with 1 and 2.

$$V_n = \omega_1 \cdot \frac{d_1}{2}. \quad (23)$$

We determine the instantaneous centers of rotation of gear 2 and sprocket 3 (P_2 and P_3). Next, the instantaneous angular velocities (ω_2), (ω_3) of gear 2 and sprocket 3 are determined by the expressions.

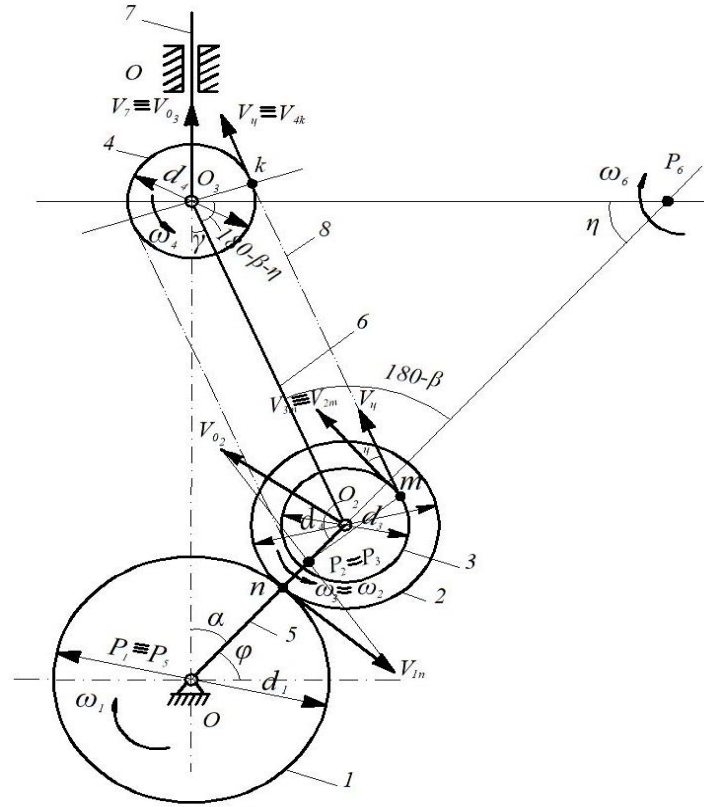


Figure 4. – Calculation diagram of the kinematics of a combined gear-lever differential transmission mechanism

$$\omega_2 = \frac{V_{O_2}}{P_2 O_2} = \frac{V_n}{n \cdot P_2} \Rightarrow \omega_2 = \omega_3, \quad (24)$$

$$\omega_3 = \frac{V_{O_2}}{P_2 O_2}, \quad (25)$$

$$\omega_2 = \omega_3. \quad (26)$$

The linear speeds of characteristic points of gear 2 and sprocket 3 are determined by the formulas

$$V_3 = \omega_3 \cdot m P_3, \quad (27)$$

$$P_2 O_2 = P_3 O_2, \quad (28)$$

$$V_3 = \frac{V_{O_2}}{P_2 O_2} \cdot \left(P_3 O_2 + \frac{d_3}{2} \right), \quad (29)$$

$$V_3 = V_{O_3} \cdot \frac{O_2 P_6}{O_3 P_6} \cdot \frac{P_3 O_2 + \frac{d_3}{2}}{P_2 O_2}, \quad (30)$$

The linear speed of the chain is determined by the formula

$$V_u = V_3 \cdot \cos \xi, \quad (31)$$

$$V_u = V_{4m}, \quad (32)$$

$$V_{2m} = V_{3m} = \omega_3 \cdot m P_2 \quad (33)$$

$m P_2$ determined by the solution of the triangle $\Delta m P_2 O_2$

$$V_y = V_{2m} \cdot \cos \xi \quad (34)$$

$$V_{4k} = \omega_4 \cdot k \cdot P_4 \quad (35)$$

The angular speeds of the driven sprocket are determined by the formulas

$$\omega_4 = \frac{V_{4k}}{kP_4}, \quad (36)$$

$$\omega_4 = \frac{V_{O_3}}{O_3P_4}, \quad (37)$$

$$\omega_4 = \frac{V_y}{kP_4}. \quad (38)$$

Using formulas 1-38 we will finally write

$$\omega_4 = V_{O_3} \cdot \frac{2 \cdot kP_4 \cdot nP_2}{O_3P_4 \cdot \cos \xi \cdot mP_2 \cdot \frac{d_1}{2}}, \quad (39)$$

$$\omega_4 = V_{O_3} \cdot \frac{2 \cdot kP_4 \cdot nP_2}{d \cdot O_3P_4 \cdot \cos \xi \cdot mP_2}. \quad (40)$$

4. Conclusions.

From formula 40 it is clear that the instantaneous angular speed of the driven sprocket is variable and depends on the angular speed of the drive gear 1, the instantaneous linear speed of its center of rotation and the position of the mechanism. Consequently, the gear ratio of the mechanism, when the center distance of the drive gear and driven sprocket changes, will be variable, and with a constant center distance of the drive gear and driven sprocket, it will be constant.

The degree of variability of the gear ratio can be changed depending on changes in the position angle and geometric parameters of the mechanism.

The variability of the angular and linear velocities of all parts of the mechanism at a constant angular velocity of the drive gear makes it possible to obtain more diverse movements and meet more complex special kinematic and dynamic requirements.

The results of the analysis and the derived formulas allow us to study the dynamics, as well as carry out geometric and dynamic synthesis of this transmission mechanism.

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FORCE ANALYSIS OF GEAR-LEVER DIFFERENTIAL TRANSMISSION MECHANISM OF A MACHINE WITH SHAFT

Abdullajonov A.A.¹, Akbarov A.N.¹, Saydokulov I.Kh.¹, Rakhmonov Kh.N.²

¹Namangan Engineering-Construction Institute

²Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan

E-mail: asrorabdullajanov@gmail.com

Abstract. In the article, the gear-lever differential mechanism designed for the working shafts of the machine with shafts of different diameters is theoretically researched. The researched gear-lever differential mechanism ensures the equality of the linear speed of rotation of the working technological machine with different diameters, for example, the upper shaft is small and the lower shaft is large. It ensures uniform movement of the processed product between two working shafts and high accuracy of impact. The main forces of this gear-lever differential mechanism, i.e., the forces acting on the working shafts, the levers to which they are attached, and the gear wheels have been determined and analyzed. As a result, the force calculation of the forces affecting the gear-lever differential mechanism was performed and the analysis was presented.

Keywords: shaft machine, transmission mechanism, levers, gears, rotational speed, rotational acceleration, force calculation.

1. Introduction.

Dynamics problems can be divided into two main types. These problems are as follows for a free material point:

In the first fundamental problem of dynamics, given the mass of a material point and its law of motion, it is asked to find the driving force.

The second main problem of dynamics is to determine the kinematic elements formed by the effect of this force when the mass of a material point and the force acting on it are known [1].

In the technique, it is necessary to solve many problems related to the verification of the motion of a material point involuntarily (in contact). In such cases, a bond placed on a point forces it to move on a fixed surface or line [2].

Taking into account the properties of various products, the work of scientists is dedicated to researching methods of processing them [3, 4].

In [5–11], the authors developed designs and studied the parameters of roller technological machines. The rational parameters of working bodies and actuators of machines were theoretically determined. The technological parameters of mechanical processing of fibrous materials, using raw hides as an example, were experimentally determined and scientifically substantiated.

We can see the scientific work carried out on the transmission mechanisms developed for machines with variable shaft interaxial distance of the working shafts. This article deals with the synthesis of a ten-link tooth-lever differential transmission mechanism. The article contains an analytical review of modern scientific research on the synthesis of tooth-lever differential transmission mechanisms of roller machines with a variable center distance of the working shafts; a method for the synthesis of tooth lever differential transmission mechanisms of roller machines with a variable center distance of the working shafts described on the example of a ten-link tooth-lever differential transmission mechanism; the conditions for the synthesis of the mechanism given and substantiated when this mechanism is used in a roller machine; one of its working shafts has the ability to rotate about its own axis, and the second working shaft, in addition to rotation about its own axis, has the ability to move relative to the first working shaft along a line passing through the center of the axes of rotation of both working shafts; the geometric synthesis of the tooth and lever contours of the mechanism, the dynamic synthesis of the mechanism, taking into account the angles of pressure between the lever link of the lever contour of the mechanism, which allows us to determine the optimal working position of the mechanism where the angles of pressure are within acceptable limits; the graphs of changes in the angles of pressure between the links of the lever contour of the mechanism, plotted depending on its position [12, 13, 20].

2. Materials, methods, and object of study.

$$m\bar{a} = \bar{F} + \bar{N} \quad (1)$$

So, in the first basic problem of the dynamics of an involuntary material point, the reaction force is determined when the mass of the material point and its law of motion and the force acting on this point are known; and in the second problem, when the mass of the material point and the force acting on it are known, it is necessary to determine the reaction force of the material point by the law of motion [1].

$$m\bar{a} = \bar{F} \quad (3)$$

here F_x, F_y are projections of the force on the OX, OY coordinate axes.

Figure 1. – Calculation scheme of the balance of the driven gear wheel with the driven working shaft

1 – we take the projections of the forces acting on the scheme on the OX , OY coordinate axes and put them in equations (4) and (5) [17–19]:

$$m\ddot{x} = R_{61} - P_{109} \cos(\varphi_n - \psi) \quad (6)$$

$$m\ddot{y} = -Q + T - P_{109} \sin(\varphi_n - \psi) \quad (7)$$

Integrating equations (6) and (7) once over time, we determine the projections of the speed on the V_x , V_y axes

$$V_x = \dot{x} = \frac{1}{m} \int_0^t (R_{61} - P_{109} \cos(\varphi_n - \psi)) dt$$

$$V_x = \dot{x} = \frac{1}{m} (R_{61} - P_{109} \sin(\varphi_n - \psi)) t \quad (8)$$

$$V_y = \dot{y} = \frac{1}{m} \int_0^t (-Q + T - P_{109} \sin(\varphi_n - \psi)) dt$$

$$V_y = \dot{y} = \frac{1}{m} (-Q + T - P_{109} \cos(\varphi_n - \psi)) t - P_{109} \quad (9)$$

We determine the full speed using the defined expressions (8) and (9).

$$V_{10} = \sqrt{V_x^2 + V_y^2} = \sqrt{\left(\frac{1}{m} (R_{61} - P_{109} \sin(\varphi_n - \psi)) t \right)^2 + \left(\frac{1}{m} (-Q + T - P_{109} \cos(\varphi_n - \psi)) t - P_{109} \right)^2} \quad (10)$$

Using equations (6) and (7), we determine the projections of the acceleration on the coordinate axes a_x , a_y

$$a_x = \ddot{x} = \frac{1}{m} (R_{61} - P_{109} \cos(\varphi_n - \psi)) \quad (11)$$

$$a_y = \ddot{y} = \frac{1}{m} (-Q + T - P_{109} \sin(\varphi_n - \psi)) \quad (12)$$

We determine the total acceleration using expressions (11) and (12).

$$a_{10} = \sqrt{a_x^2 + a_y^2} = \frac{1}{m} \sqrt{(R_{61} - P_{109} \cos(\varphi_n - \psi))^2 + (-Q + T - P_{109} \sin(\varphi_n - \psi))^2} \quad (13)$$

We take the projections of the forces acting on the scheme in Fig. 2 on the coordinate axes OX and OY and put them in equations (4) and (5).

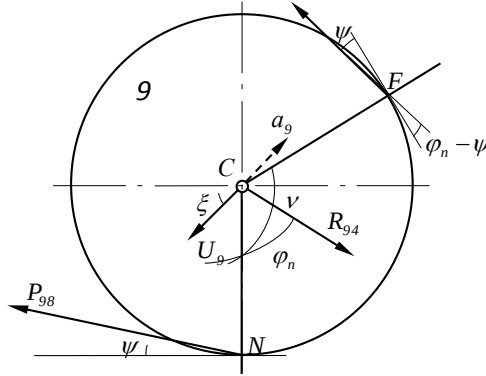


Figure 2. – 9-calculation of the balance of the intermediate gear wheel

$$m\ddot{x} = R_{94} \cos(90 - \varphi_n) - U_9 \cos \xi - P_{910} \cos(\varphi_n - \psi) - P_{98} \cos \psi \quad (14)$$

$$m\ddot{y} = R_{94} \sin(90 - \varphi_n) - U_9 \sin \xi + P_{910} \sin(\varphi_n - \psi) - P_{98} \sin \psi \quad (15)$$

Integrating equations (14) and (15) once over time, we determine the projections of the speed on the V_x , V_y axes

$$V_x = \dot{x} = \frac{1}{m} \int_0^t (R_{94} \cos(90 - \varphi_n) - U_9 \cos \xi - P_{910} \cos(\varphi_n - \psi) - P_{98} \cos \psi) dt$$

$$V_x = \dot{x} = \frac{1}{m} (R_{94} \sin(90 - \varphi_n) - U_9 \sin \xi - P_{910} \sin(\varphi_n - \psi) - P_{98} \sin \psi) t \quad (16)$$

$$\begin{aligned} V_y = \dot{y} &= \frac{1}{m} \int_0^t (R_{94} \sin(90 - \varphi_n) - U_9 \sin \xi + P_{910} \sin(\varphi_n - \psi) - P_{98} \sin \psi) dt \\ V_y = y &= \frac{1}{m} \left(R_{94} \cos(90 - \varphi_n) - U_9 \cos \xi - \right. \\ &\quad \left. - P_{910} \cos(\varphi_n - \psi) - P_{98} \cos \psi \right) t + R_{94} - U_9 + P_{910} - P_{98} \end{aligned} \quad (17)$$

We determine the full speed using the defined expressions (16) and (17).

$$V_9 = \sqrt{V_x^2 + V_y^2} = \sqrt{\left(\frac{1}{m}(R_{94}\sin(90 - \varphi_n) - U_9 \sin \xi - P_{910}\sin(\varphi_n - \psi) - P_{98}\sin \psi)t\right)^2 + \left(\frac{1}{m}(R_{94}\cos(90 - \varphi_n) - U_9 \cos \xi - P_{910}\cos(\varphi_n - \psi) - P_{98}\cos \psi)t + R_{94} - U_9 + P_{910} - P_{98}\right)^2} \quad (18)$$

Using equations (16) and (17), we determine the projections of the acceleration on the coordinate axes OX, OY

$$a_x = x = \frac{1}{m}(R_{94} \cos(90 - \varphi_n) - U_9 \cos \xi - P_{910} \cos(\varphi_n - \psi) - P_{98} \cos \psi) \quad (19)$$

$$a_y = y = \frac{1}{m}(R_{94} \sin(90 - \varphi_n) - U_9 \sin \xi - P_{910} \sin(\varphi_n - \psi) - P_{98} \sin \psi) \quad (20)$$

We determine the total acceleration using expressions (19) and (20).

$$a_9 = \sqrt{a_x^2 + a_y^2} = \sqrt{\left(\frac{1}{m}(R_{94} \cos(90 - \varphi_n) - U_9 \cos \xi - P_{910} \cos(\varphi_n - \psi) - P_{98} \cos \psi)\right)^2 + \left(\frac{1}{m}(R_{94} \sin(90 - \varphi_n) - U_9 \sin \xi - P_{910} \sin(\varphi_n - \psi) - P_{98} \sin \psi)\right)^2} \quad (21)$$

3 - we take the projections of the forces acting on the scheme on the OX , OY coordinate axes and put them in equations (4) and (5)/

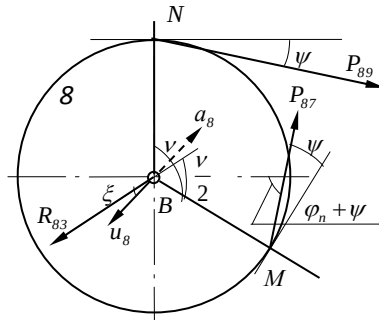


Figure 3. – 8-the calculation scheme of the balance of the intermediate gear wheel

$$m\dot{x} = R_{89} \cos \psi - U_8 \cos \psi - R_{83} \cos(90 - \frac{\nu}{2}) + R_{87} \cos(\varphi_n + \psi) \quad (22)$$

$$my = R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi) \quad (23)$$

(22) and (23) once over time, we determine the projections of the speed on the axes V_x and V_y .

$$\begin{aligned} V_x &= \dot{x} = \frac{1}{m} \int_0^t (R_{89} \cos \psi - U_8 \cos \psi - R_{83} \cos(90 - \frac{\nu}{2}) + R_{87} \cos(\varphi_n + \psi)) dt \\ V_x &= \dot{x} = \frac{1}{m} (R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi)) t \end{aligned} \quad (24)$$

$$V_y = y = \frac{1}{m} \int_0^t (R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi)) dt$$

$$V_y = y = \frac{1}{m} (R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi)) t - R_{89} - U_8 - R_{83} + R_{87} \quad (25)$$

We determine the full speed using the defined expressions (24) and (25).

$$V_8 = \sqrt{V_x^2 + V_y^2} = \sqrt{\left(\frac{1}{m}(R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi))t \right)^2 + \left(\frac{1}{m}(R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi))t - R_{89} - U_8 - R_{83} + R_{87} \right)^2} \quad (26)$$

Using equations (22) and (23), we determine the projections of the acceleration on the coordinate axes OX, OY

$$a_x = \frac{1}{m} \left(R_{89} \cos \psi - U_8 \cos \psi - R_{83} \cos(90 - \frac{\nu}{2}) + R_{87} \cos(\varphi_n + \psi) \right) \quad (27)$$

$$a_y = \frac{1}{m} \left(R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi) \right) \quad (28)$$

We determine the total acceleration using expressions (27) and (28).

$$a_8 = \sqrt{a_x^2 + a_y^2} = \sqrt{\left(\frac{1}{m}(R_{89} \cos \psi - U_8 \cos \psi - R_{83} \cos(90 - \frac{\nu}{2}) + R_{87} \cos(\varphi_n + \psi)) \right)^2 + \left(\frac{1}{m}(R_{89} \sin \psi - U_8 \sin \psi - R_{83} \sin(90 - \frac{\nu}{2}) + R_{87} \sin(\varphi_n + \psi)) \right)^2} \quad (29)$$

4 - we take the projections of the forces acting on the scheme on the OX, OY coordinate axes and put them in equations (4) and (5)

$$mx = R_{71} \cos(\varphi_n + \psi) - P_{78} \cos(\varphi_n + \psi) \quad (30)$$

$$my = R_{71} \sin(\varphi_n + \psi) - P_{78} \sin(\varphi_n + \psi) - G \quad (31)$$

Integrating equations (30) and (31) once over time, we determine the projections of the speed on the V_x and V_y axes

$$V_x = \dot{x} = \frac{1}{m} \int_0^t (R_{71} \cos(\varphi_n + \psi) - P_{78} \cos(\varphi_n + \psi)) dt$$

$$V_x = \dot{x} = \frac{1}{m} (R_{71} \sin(\varphi_n + \psi) - P_{78} \sin(\varphi_n + \psi))t \quad (32)$$

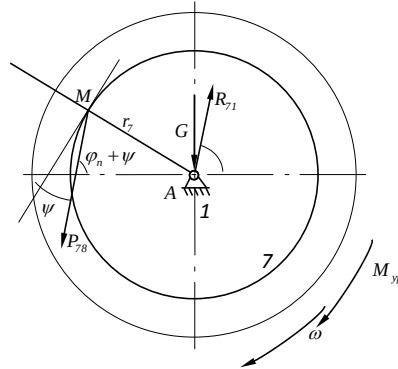


Figure 4. – Scheme of calculating the balance of the driving working shaft and the leading gear wheel

$$V_y = \dot{y} = \frac{1}{m} \int_0^t (R_{71} \sin(\varphi_n + \psi) - P_{78} \sin(\varphi_n + \psi) - G) dt$$

$$V_y = \dot{y} = \frac{1}{m} ((-R_{71} \cos(\varphi_n + \psi) + P_{78} \cos(\varphi_n + \psi) - G)t + R_{71} - P_{78}) \quad (33)$$

3. Results.

We determine the full speed using expressions (32) and (33).

$$V_7 = \sqrt{V_x^2 + V_y^2} = \sqrt{\left(\frac{1}{m}(R_{71}\sin(\varphi_n + \psi) - P_{78}\sin(\varphi_n + \psi))t\right)^2 + \left(\frac{1}{m}((-R_{71}\cos(\varphi_n + \psi) + P_{78}\cos(\varphi_n + \psi) - G)t + R_{71} - P_{78})\right)^2} \quad (34)$$

Using equations (30) and (31), we determine the projections of the acceleration on the coordinate axes OX, OY

$$a_x = \frac{1}{m}(R_{71}\cos(\varphi_n + \psi) - P_{78}\cos(\varphi_n + \psi)) \quad (35)$$

$$a_y = \frac{1}{m}(R_{71}\sin(\varphi_n + \psi) - P_{78}\sin(\varphi_n + \psi) - G) \quad (36)$$

We determine the total acceleration using expressions (35) and (36).

$$a_7 = \sqrt{a_x^2 + a_y^2} = \sqrt{\left(\frac{1}{m}(R_{71}\cos(\varphi_n + \psi) - P_{78}\cos(\varphi_n + \psi))\right)^2 + \left(\frac{1}{m}(R_{71}\sin(\varphi_n + \psi) - P_{78}\sin(\varphi_n + \psi) - G)\right)^2} \quad (37)$$

4. Discussions.

Formulas (10), (18), (26), (34) determine the full speed of the four gear wheels of the mechanism. Formulas (13), (21), (29), (37) determine the full acceleration of the four gears of the mechanism. As a result, the gear wheels of the differential transmission mechanism with different diameters are considered as material points, and the main equation of their dynamics was developed. Based on the second problem of dynamics, the developed equations determined the reaction force with the law of motion of the material point when the mass of the material point and the force acting on it are known.

5. Conclusions.

The kinetostatic analysis of the gear-lever differential transmission mechanism with different diameters of the leading and driven gears intended for the working shafts of the shaft machine consists in determining the reaction forces falling on the joints, and is used in the synthesis and design of this mechanism. is used.

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VIBRATION ANALYSIS OF A RUBBER SHOCK ABSORBER WITH A CURVED SURFACE OF A COTTON SQUEEGEE

Abduvakhobov D.A.¹, Gofurjanov I.I.¹, Madrahimova M.B.¹

¹Namangan Engineering-Construction Institute, Namangan, Uzbekistan

E-mail: gofurjanovi@gmail.com

Abstract. By means of theoretical studies, the laws of oscillations of the curvilinear wavy surface of the reflector are determined in the design scheme of an effective purifier of cotton from fine impurities proposed in the article. The nature of the vibration is determined depending on the performance, cushioning and parameters of the wipers. The recommended parameters of the system are based on the analysis of the constructed graphical links.

Keywords: cleaner, cotton raw material, fine dirt, pile drum, deflector, curved wavy surface, rubber damper, vibration, uniformity, dispersion, amplitude, coverage, speed, friction, force.

1. Introduction.

It is recommended to install the cotton wool on the return device of the small waste cleaner through rubber shock absorbers [1,2]. During operation, the recoil shock absorber is deformed due to the impact of cotton wool pieces on the recoil surface. Depending on its singularity and the mass of the repulsor, oscillations occur. These vibrations occur both vertically and horizontally. Since the oscillations of rolling are not significant, we considered mainly vertical oscillations of the deflector with a curved wavy surface. Figure 1 shows the layout and calculation scheme of the return of the cleaner.

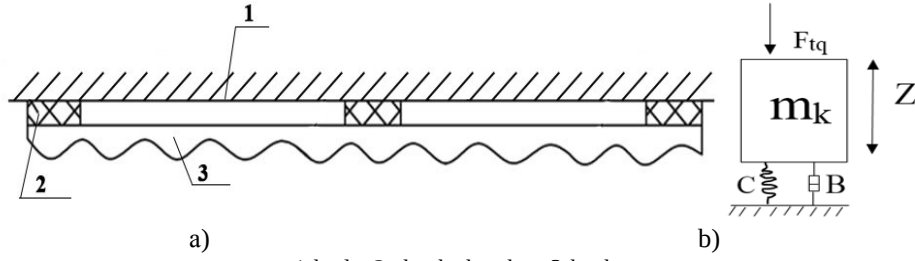


Figure 1. – Wiper return scheme (a) and calculation (b)

2. Materials, methods, and object of study.

The main driving force of the rotor vibration is the technological resistance, that is, from the side of the pile surface. The total value of the force of the thrown pieces of cotton on the return device is calculated [3,4]:

$$F_{t,q} = [F_1 \sin \alpha_1 + F_2 \sin \alpha_2 + \dots + F_i \sin \alpha_i] \frac{1}{\sin \alpha} \quad (1)$$

here, $F_1, F_2, F_3, \dots, F_i$ – in the force of the impact forces on the returning cotton pieces; the $\alpha_1, \alpha_2, \alpha_3, \dots, \alpha_i$ – angles of deflection of the impact forces; the $F_{t,q}, \alpha$ – angle of its deflection with the force of technological resistance.

The technological process assumes that the technological resistance changes randomly;

$$F_q = F_{t,q} \pm \delta F_{t,q} \quad (2)$$

here, $\delta F_{t,q}$ – random organizer of technological resistance. We use the second-order logranche equation [5,6] to build a mathematical model of the law of oscillations of the proposed cotton purifier.

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{Z}} \right) = \frac{\partial T}{\partial Z} - \frac{\partial \Pi}{\partial Z} - \frac{\partial \Phi}{\partial \dot{Z}} + Q(Z) \quad (3)$$

here T, Π, Φ are the kinetic and potential energies of the oscillatory system and the dissipative function of the relay [7,8]; Z is the generalized coordinate (vertical shift of the return); $Q(Z)$ – generalized external forces.

Accordingly, based on [3], the kinetic potential energies and the dissipative function of the reflector are [9,10]:

$$T = \frac{1}{2} m_q \left(\frac{dZ}{dt} \right)^2; \quad \Pi = \frac{1}{2} c_q \xi Z^2; \quad \Phi = \frac{1}{2} b_q \left(\frac{dZ}{dt} \right)^2 \quad (4)$$

here, m_q is the mass of the repulsor; c_q, b_q – Its parameters are determined on the basis of (3) given in the linear uniformity and dissipation coefficient of the return rubber shock absorber:

$$\begin{aligned} \frac{\partial T}{\partial \dot{Z}} &= \frac{\partial}{\partial \dot{Z}} \left[\frac{1}{2} m_q \left(\frac{dZ}{dt} \right)^2 \right] = m_q \left(\frac{dZ}{dt} \right); \quad \frac{d}{dt} \left(\frac{\partial T}{\partial \dot{Z}} \right) = m_q \frac{d^2 Z}{dt^2}; \quad \frac{\partial \Pi}{\partial Z} = \frac{\partial}{\partial Z} \left(\frac{1}{2} c_q Z^2 \right) = c_q Z; \\ \frac{\partial \Phi}{\partial \dot{Z}} &= \frac{\partial}{\partial \dot{Z}} \left[\frac{1}{2} b_q \left(\frac{dZ}{dt} \right)^2 \right] = b_q \dot{Z}; \end{aligned}$$

$$F_q = (F_1 \sin \alpha_1 + F_2 \sin \alpha_2 + \dots + F_i \sin \alpha_i) \frac{1}{\sin \alpha} \pm \delta F_q \quad (5)$$

Substituting the obtained expressions (5) in (3), we obtain a differential equation representing the fluctuations of the cotton cleaner from fine impurities [11]:

$$m_q \frac{d^2 Z}{dt^2} + c_q Z + b_q \frac{dZ}{dt} = (F_1 \sin \alpha_1 + F_2 \sin \alpha_2 + \dots + F_i \sin \alpha_i) \frac{1}{\sin \alpha} \pm \delta F_q \quad (6)$$

The numerical solution of the problem was carried out on a computer using special programs [11,12]. In this case, the initial parameter values are obtained as follows:

$$m_q = (2,5 \div 4,2) \kappa z; c_q = (1,5 \div 2,5) \cdot 10^3 H / M; b_q = (2,0 \div 2,3) H / M;$$

$$F_{t,q} = (2,0 \div 3,5) H; \delta F_q = \pm(0,05 \div 0,11) F_q;$$

$$10_{\sigma} = (46 \div 52) c^{-1}; \alpha_i = (45^0 \div 65^0)$$

As a result of the numerical solution of the problem, the laws of vibration of the cotton cleaner from small impurities were obtained. In particular, Figure 2 shows the displacement and velocity laws of the recommended cotton purifier. The obtained values of the backlash ranges and speed changes increase productivity, i.e. $F_{t,q}$ increases with increasing .

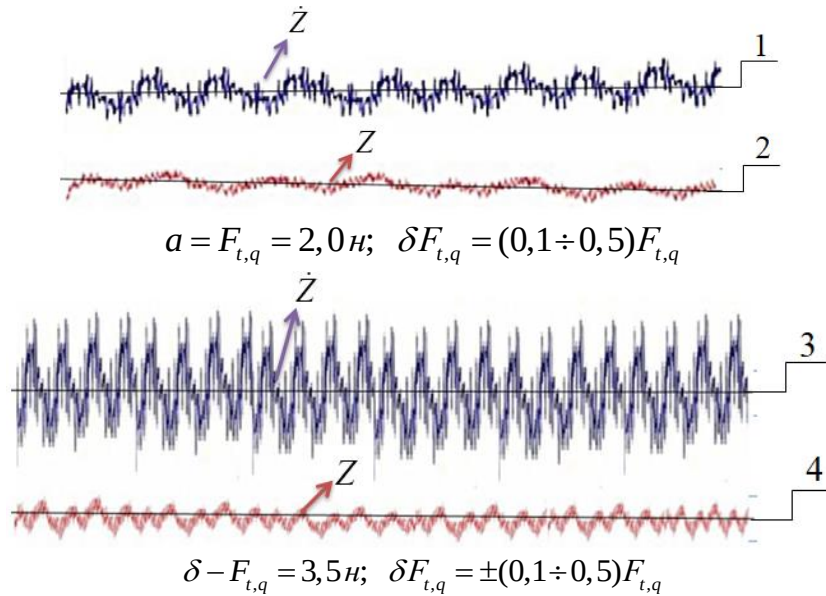


Figure 2. – Recommended cleaning of cotton from small impurities, anti-slip and speed change laws

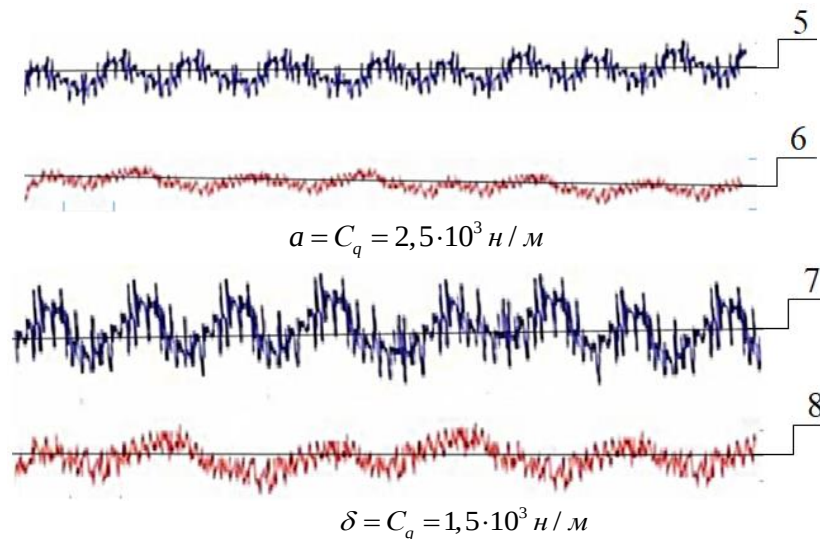


Figure 3. – The regularity of wiper, brake, shift and speed oscillations (depending on the change of base performance)

In this case, random resistance generators from the laws of vibration were used to obtain a numerical solution using a special numerical generator [20].

It is known that increasing the stiffness of the belt element in vibration systems reduces the vibration amplitude. Increasing the uniformity coefficient of the wiper rubber damper reduces the

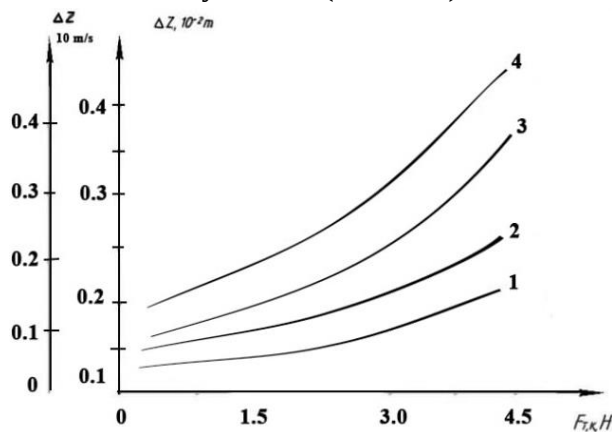
amplitudes of oscillations of displacement and acceleration of the wiper, respectively (Fig. 3, Laws 5.6).

3. Results and discussions.

At fig. 3 shows the patterns of sweep and velocity fluctuations (depending on the change in the uniformity of the base). The regularities of the reverse movement and the speed of the brush movement (depending on the change in the uniformity of the base) have been established.

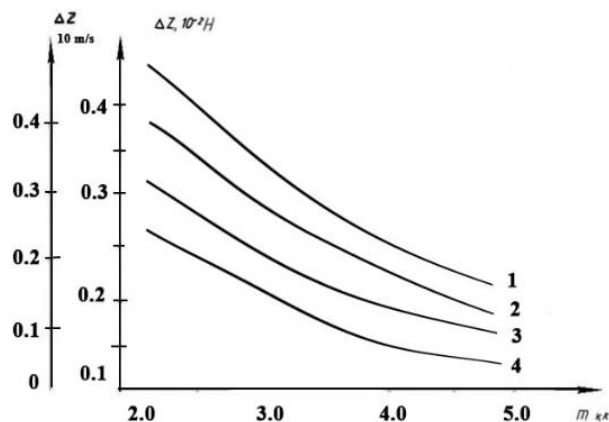
Based on the patterns of oscillations of the Oliingan transducer, graphs of the connection of parameters were constructed. At fig. 4 shows graphs of the dependence of the displacement values and the ranges of fluctuations in the speed of the rubber damper of the cotton wiper on fine impurities on the values of the technological resistance emanating from cotton.

Analyzing the obtained correlation coefficients, the technological resistance of touching the wool of the deflector surface increases from 0.81 n to 4.5n, and the vibration range of the deflector at the mass of the deflector is 2.5 kg. The Δ_Z value $1,41 \cdot 10^{-3} M$ of the non-linear connection $2,61 \cdot 10^{-3} M$ increases to . Accordingly, the range of vibration velocity increases from 1.31 m/s to 4.73 m/s according to a nonlinear pattern (Fig. 4, graphs 2, 4). Vibration coating based on the results of experiments of the Δ_Z value of the $(2,2 \div 2,7) \cdot 10^{-3}$ technological strength of cotton, so that it does not exceed, it $(3,2 \div 3,8) H$ is recommended to be between In this case, the values of the vibration velocity of the damper with a curved wavy surface $(3,0 \div 4,0) M / c$ will be between



$$1, 2 - \Delta Z = f(F_{t,q}); \quad 3, 4 - \Delta \dot{Z} = f(F_{t,q}); \quad 2, 4 - m_q = 2,5 \kappa z; \quad 1, 3 - m_q = 4,1 \kappa z;$$

Figure 4. – Graphs of the dependence of the displacement values and the range of vibration velocities on the values of the technological resistance emanating from cotton

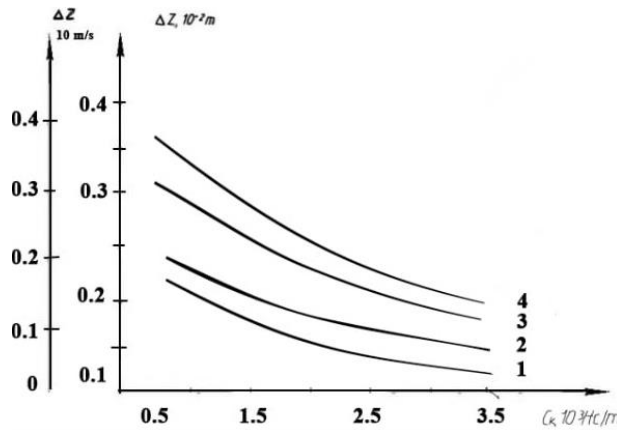


$$1, 2 - \Delta Z = f(m_q); \quad 3, 4 - \Delta \dot{Z} = f(m_q); \quad 1, 3 - c_q = 2,5 \cdot 10^{-3} H / c; \quad 2, 4 - c_q = 1,5 \cdot 10^{-3} H / c;$$

Figure 5. – The graphs of the dependence of the values of displacement and the rate of oscillation of the coating, changes in its mass

It should be noted that with a reflector mass of 4.1 kg, the Δ_z values sharply decreased $1,25 \cdot 10^{-3} M$ from $1,8 \cdot 10^{-3} M$ increases to Accordingly, the $\Delta \dot{Z}$ values increase from 0.76 m/s to 3.7 m/s with non-linear coupling. Therefore, it is desirable to reduce the problem of feedback.

At fig. 5 shows graphs of changes in movements and speeds of a rubber shock absorber of a cotton cleaner with a curved wavy surface, as well as the dependence of its mass on the change. Analyzing the plotted graphs, with an increase in the wiper weight from 2.23 kg to 5.0 kg, the damper stiffness is equal to $2,5 \cdot 10^{-3} M$ from $1,21 \cdot 10^{-3} M$ decreases in a nonlinear relationship to .Also, the $\Delta \dot{Z}$ values increase in mass and decrease in singularity, i.e. $c_q = 1,5 \cdot 10^{-3} H / M$ cover the velocity fluctuations when the decrease $4,32 \cdot 10^{-3} M$ from $2,34 \cdot 10^{-3} M$ is visible to (Fig. 5, graphs 2.5). In general, the weight of the reciprocating mechanism $m_q \leq (4,1 \div 4,6) \kappa z$ is recommended to be taken between.



$$1, 2 - \Delta Z = f(c_q); \quad 3, 4 - \Delta \dot{Z} = f(c_q); \quad 1, 3 - F_{t,q} = 2, 0 H; \quad 2, 4 - F_{t,q} = 3, 5 H;$$

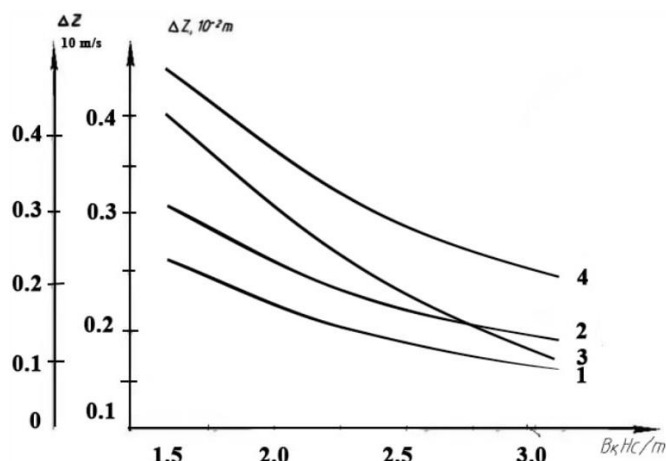
Figure 6. – Graphs of the dependence of the displacement values and speed oscillation ranges of the rubber damper of a cotton cleaner with a curved wavy surface on a given damper uniformity coefficient

It should be noted that the vibration of a large mass reflector is a difficult task from a practical point of view. Because it takes a lot of power to vibrate the return switch. Therefore, it is desirable to prepare the reflector in separate pieces from the mass recommended above. To change the frequency of transitions with an amplitude of oscillations, it is necessary to change the population of the tape element.

At fig. 6 shows graphs of the dependence of the displacement values and speed oscillation ranges of a rubber shock absorber with a curved wavy surface on a given shock absorber uniformity coefficient. Based on the analysis of the plotted graphs, the uniformity coefficient of the rubber supports $0,65 \cdot 10^3 H / M$ of the deflector is from $3,6 \cdot 10^3 H / M$ When you get to 100% of the reverse vibration range. $F_{t,q} = 2, 0 H$ When it was $2,38 \cdot 10^{-3} M$ from $1,23 \cdot 10^{-3} M$ decreases to $\Delta \dot{Z}$ value $0,38 \cdot 10 M / c$ from $0,13 \cdot 10 M / c$ decreases with nonlinear coupling to (Fig. 2.11, schedules 1, 3). Accordingly $F_{t,q} = 3, 5 H$ When it has been $2,65 \cdot 10^{-3} M$ the ΔZ value from $1,61 \cdot 10^{-3} M$ decreases to $0,43 \cdot 10 M / c$ the $\Delta \dot{Z}$ value from $0,17 \cdot 10 M / c$ decreases to $1,13 \cdot 10 M / c$

Reversible vibration protection to $(2,2 \div 2,7) \cdot 10^{-3} M$ ensure that it does not exceed the uniformity coefficient of the rubber band $(1,2 \div 2,0) \cdot 10^3 H / M$ it is recommended to be between

The belt element significantly changes the dissipation coefficient, the amplitude of oscillations and the time of the transition process.



$$1, 2 - \Delta Z = f(\epsilon_q); \quad 3, 4 - \Delta \dot{Z} = f(\epsilon_q); \quad 1, 3 - F_{t,q} = 2, 0 \text{ H}; \quad 2, 4 - F_{t,q} = 3, 5 \text{ H};$$

Figure 7. – Graphs of the dependence of the displacement values and speed oscillation ranges of the rubber damper of a cotton cleaner with a curved wavy surface on a given damper dissipation coefficient

At fig. 7 shows graphs of the dependence of the displacement values and speed oscillation ranges of a rubber damper of a cotton wiper with a curved wavy surface on a given damper dissipation coefficient. The dissipation coefficient according to the analysis $0,64 \text{ Hc/m}$ from $3,0 \text{ Hc/m}$ at increase to $F_{t,q} = 2,0 \text{ H}$; When it was $2,4 \cdot 10^{-3} \text{ m}$ from $1,24 \cdot 10^{-3} \text{ m}$ decreased to $\Delta \dot{Z}$ value $1,41 \cdot 10 \text{ m/c}$ from $1,13 \cdot 10 \text{ m/c}$ decreases to In ϵ_q It $\Delta \dot{Z}$ can be seen that they have a sufficient effect on .

Likewise vibration $F_{t,q} = 3,5 \text{ H}$ coverage when ΔZ values $4,9 \cdot 10^{-3} \text{ m}$ from $1,93 \cdot 10^{-3} \text{ m}$ up to $\Delta \dot{Z}$ value $5,1 \text{ m/c}$ from $2,36 \text{ m/c}$ decreases in nonlinear coupling to . Recommended values of the deflector damping coefficient to ensure the specified vibration coating $\epsilon_q \geq (2,3 \div 2,6) \text{ Hc/m}$ are recommended to be between

4. Conclusions.

The article obtains the vibration laws of a deflector mounted on a rubber shock absorber with a curved wavy surface for cleaning cotton from fine impurities. Connection schedules are built on the basis of speeds, based on the recommended values of the system parameters.

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EQUATION OF MOTION OF A RAW ACCELERATOR ROLLER OF THE SAW GINING ZONE

Akhmedjanov Y.A.¹

¹Tashkent State Technical University, Tashkent, Uzbekistan,
E-mail:yusuf1956@mail.ru,

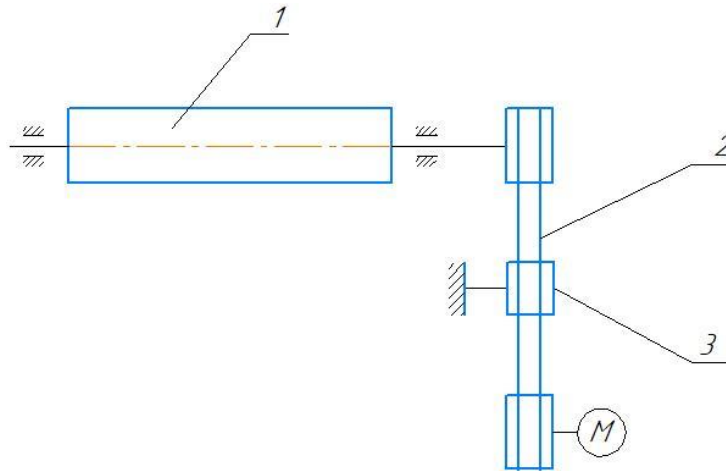
Abstract: New raw chamber accelerator mechanism reviewed and a dynamic model of a two-mass variable-ratio rotation system, the equations of motion of the machine unit were derived in the form of nonlinear differential equations, the results of the digital solution of equations are presented taking into account the variation of technological parameters, as well as the amplitude of oscillations of the gear ratio, the coefficients of rigidity and dissipation of the elastic transmission and the moments of inertia of the wheels of the machine unit. Based on the solution of the problem, the laws of motion of the rotating masses of the machine unit with the accelerator mechanism of the raw chamber of the saw fiber separator were obtained, taking into account the inertial, elastic-dissipative properties of the elements, as well as the variability of the transmission ratio of the elastic transmission and the technological load from raw cotton.

Keywords: machine unit, options, working body, process load, tension roller, roller accelerator, eccentricity, differential equations, not linear equations.

1. Introduction.

The drive mechanism of the gin accelerator consists of an electric motor, V-belt pulleys, an eccentric tension roller and a raw roller accelerator (Fig. 1). The eccentric tension roller periodically changes the sliding of the belt, due to which the accelerator rotates at a variable frequency, imparting

impulsive rotation to the raw roller. As a result of this, firstly, shaking of the raw roller occurs, secondly, a cyclic change in the vacuum in the seed comb area, and thirdly, due to the impulsive rotation of the raw roller, a variable centrifugal force appears. Raw cotton periodically attacks the teeth of the saw blade, and the bare seeds are intensely released [1, 2].



1. Accelerator, 2. Belt drive of the accelerator drive, 3. Belt tensioned

Figure 1. – Kinematic diagram of the saw gin accelerator drive:

Shaking the raw roller causes cracks to appear in it, through which the seeds fall out more easily. In addition, shaking the raw roller contributes to some extent to the loss of debris from the fibrous mass [3].

Cyclic changes in rarefaction lead to a more complete loss of bare seeds from the raw chamber. In the area of the seed comb, raw cotton fibers are captured by the saw teeth. The impulsive force in the raw roller, resulting from the uneven rotation of the accelerator, contributes to better capture of raw cotton slices by the saw teeth and better retention of them on the saw teeth. This leads to a more complete separation of the fiber from the seeds.

Under the influence of variable centrifugal force, bare seeds, the adhesion force of which to the fibrous mass is significantly less than the adhesion force of the flakes, move to its periphery and, when passing through the seed comb zone, under the influence of the same force and changes in rarefaction, they come off the raw roller and fall out of the working chamber, thereby accelerating the removal of bare seeds from the raw roller. By increasing the rate of release of bare seeds from the raw roller, its mass in the working chamber decreases, fibrousness increases, and damage to seeds and fiber decreases. As the fiber content of the raw roll increases, the productivity of the gin also increases (4.5).

2. Statement of the problem.

To describe the ginning process from the point of view of the dynamics of mechanisms, the task was set to derive a formula for the equation of motion of the raw roller accelerator, connecting the constant and variable factors of the accelerator wire, as well as the mechanical characteristics of the engine.

3. Solution method.

Was based on a mathematical model of machine motion unit with an accelerator mechanism and a Lagrange differential equation of the second kind.

The dynamic model of the machine unit of the raw saw roller accelerator is a two-mass linear system (Fig. 2), which has elastic, dissipative and electromagnetic properties [5, 6].

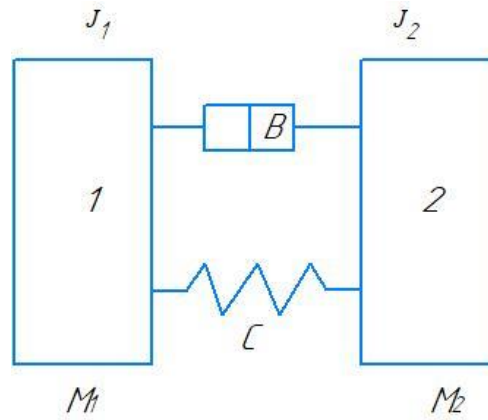


Figure 2. – Design diagram of a 2-mass drive machine unit in an accelerator

The mathematical model of the machine unit is described by a system of differentiated equations.

To compile differentiated equations of motion of the system, we will use the Lagrange equation of the second kind [7, 8].

$$\frac{d(\partial T)}{dt(\partial \dot{q}_j)} - \frac{\partial T}{\partial q_j} + \frac{\partial \Phi}{\partial q_j} + \frac{\partial \Pi}{\partial q_j} = Q_j, \quad (1)$$

here: q_j - generalized coordinate; $\dot{q}$ - generalized speed; T - kinetic energy of the system; Π - potential energy of force; Φ - dissipated energy (Rayleigh function); Q_j - generalized external forces.

For the generalized coordinates q we take the angular displacements of the engine rotor and accelerator shaft ω_1 and ω_2 .

The kinetic energy for this dynamic model is equal to [9]:

$$T = \frac{J_1 \cdot \omega_1^2}{2} + \frac{J_2 \cdot \omega_2^2}{2}, \quad (2)$$

here J_1 and J_2 - moments of inertia of the electric motor rotor with the shaft and drive pulley.

The potential energy for this dynamic model is determined by the formula [9]:

$$\Pi = \frac{1}{2} \sum_{j=1}^n C_j (\omega_j - i_{j,j+1} \omega_{j+1})^2, \quad (3)$$

here n - the number of masses of the system; C_j - rigidity coefficient of elastic transmissions of the system; $i_{j,j+1}$ - transmission ratio of the system.

The Rayleigh dissipative function is determined from the expression [10]:

$$\Phi = \frac{1}{2} \sum_{j=1}^n b_j (\omega_j - i_{j,j+1} \omega_{j+1})^2, \quad (4)$$

here b_j - coefficient of viscous resistance of elastic transmissions of the system.

Let us define the terms of the Lagrangian equations:

$$\begin{aligned} \frac{\partial T}{\partial \omega_j} &= J_j \omega_j; \\ \frac{d(\partial T)}{dt(\partial \dot{\omega})} &= J_j \omega_j; \\ \frac{\partial \Pi}{\partial \omega_1} &= e(\omega_1 - i_{12} \omega_2) \cdot \left(1 - \frac{\partial i_{12}}{\partial \omega_1} \omega_1 \right); \end{aligned} \quad (5)$$

$$\frac{\partial \Pi}{\partial \omega_2} = -c(\omega_j - i_{12}\omega_2) \cdot i_{12}; \quad (6)$$

$$\frac{\partial \Phi}{\partial \omega_1} = b(\omega_1 - i_{12}\omega_2) \cdot \left(1 - \frac{\partial i_{12}}{\partial \omega_1} \omega_1\right); \quad (7)$$

$$\frac{\partial \Phi}{\partial \omega_2} = -b(\omega_j - i_{12}\omega_2) \cdot i_{12}, \quad (8)$$

here $\frac{\partial i_{12}}{\partial \omega_1} = 0$.

The transmission ratio between the drive and driven shafts with an eccentric tension roller is expressed by the following equation [1].

$$i_{12} = \frac{D_2 \cdot E \cdot F}{D_1 \left\{ E \cdot F - \left[S_0 + \left(a \cos \varphi_p + \sqrt{r^2 - a^2 \sin^2 \varphi_p} - r \right) S_k \right] \right\}}, \quad (9)$$

here E - belt modulus of elasticity; F - belt cross-sectional area; S_0 - initial belt tension; α - belt wrap angle equal to the sliding arc; r - radius of the tension roller circle; e - tension roller eccentricity; S_k - belt tension coefficient. The gear ratio depending on the angle of rotation of the drive pulley is as follows:

$$\frac{\partial i_{12}}{\partial \omega_1} = \frac{-\frac{D_2}{D_1} \cdot E \cdot F - (e^{\mu\alpha} - 1) \cdot S_k \cdot \left[-a \sin(\varphi_1 \cdot i_{12}) + \frac{a^2 \sin(\varphi_1 \cdot i_b)}{\sqrt{r^2 - a^2 \sin^2(\varphi_1 \cdot i_b)} - r} \right]}{\left\{ E \cdot F - (e^{\mu\alpha} - 1) \cdot \left[S_0 + a \cos(\varphi_b \cdot i_b) + \sqrt{r^2 - a^2 \sin^2(\varphi_1 \cdot i_b)} - r \right] \cdot S_k \right\}^2}, \quad (10)$$

here i_b - tension roller shaft gear ratio, e - base of natural logarithms; μ - belt friction coefficient; The generalized forces of the dynamic model are equal to

$$\begin{aligned} Q_1 &= M_1 - M_2; \\ Q_2 &= -M_2, \end{aligned}$$

here M_1 - motor shaft torque; M_2 - moment of technological resistance forces of the fibrous material acting on the accelerator of the raw roller.

Substituting the resulting expression into the Lagrange equation of the second kind, we obtain a differential equation describing the motion of a two-mass elastic system:

$$\begin{aligned} J_1 \omega_1 &= M_1 - e(\omega_1 - i_{12}\omega_2) \cdot \left(1 - \frac{\partial i_{12}}{\partial \omega_1}\right) - b(\omega_1 - i_{12}\omega_2); \\ J_2 \omega_2 &= e(\omega_1 - i_{12}\omega_2) \cdot i_{12} + b(\omega_1 - i_{12}\omega_2) \cdot i_{12} - M_2. \end{aligned} \quad (11)$$

To fully assess the dynamic characteristics of a machine unit, it is necessary to take into account the dynamic mechanical characteristics of the steady state of motion, where electromechanical transient processes are also taken into account. They are not taken into account in equation (11).

The equation for the dynamic characteristics of an asynchronous electric motor for a steady state has the form [2]:

$$\frac{\omega_0 - \omega_1}{\omega_0} = \frac{S_k}{2M_k} \cdot M_1 + \frac{1}{2\omega_c \cdot M_k} \cdot M_1, \quad (12)$$

here: M_K - critical (maximum) torque of the electric motor in static (tipping) mode, n.m; S_K - critical slip, i.e. sliding at $M_1 = M_K$; ω_e - angular frequency of the electrical network, c^{-1} ; ω_0 - idle speed c^{-1} .

The critical torque of the electric motor is [11]:

$$M_k = 9550 \frac{N_H}{n_H} \cdot \lambda, \quad (13)$$

here N_H - rated power consumption of the electric motor, kW; n_H - rated speed of the electric motor rotor; λ - coefficient characterizing the value of the ratio of the critical and nominal moments.

The value of the critical slip of the electric motor rotor is determined by the formula:

$$S_k = \lambda \cdot S_H \left(1 + \sqrt{1 - \frac{1}{\lambda}} \right), \quad (14)$$

here S_H - nominal slip value determined from the relation:

$$S_H = \frac{\omega_0 - \omega_H}{\omega_0}.$$

The idle angular speed and the frequency of the electric current f_e are related by the relation:

$$\omega_0 = \frac{\omega_c}{P} = 2 \frac{\Pi f_c}{P},$$

here P - number of alternating poles of the stator winding.

The movement of the machine unit, taking into account the dynamic characteristics of the asynchronous electric motor, variable gear ratio and formula (12), looks like:

$$\frac{\omega_0 - \omega_1}{\omega_0} = \frac{S_k}{2M_k} M_1 + \frac{1}{2\omega_0 \cdot M_k} \cdot M_1. \quad (15)$$

Substituting the values into formula [11], we obtain:

$$I_1 \omega_1 = \frac{M_1 - e(\omega_1 - i_{12} \omega_2) \cdot \left\{ 1 - \frac{D_2}{D_1} EF \left[(e^{\mu \alpha} - 1) \cdot S'_K \cdot \left[a \cdot \sin(\varphi_1 \cdot i_b) + \frac{a^2 \cdot \sin(\varphi_1 \cdot i_b)}{\sqrt{r^2 - a^2 \cdot \sin^2(\varphi_1 \cdot i_b)} - r} \right] \right] \right\}}{\left\{ EF - (e^{\mu \alpha} - 1) \cdot \left[S_0 + a \cdot \cos(i_b \cdot \varphi_1) + \sqrt{r^2 - a^2 \cdot \sin^2(\varphi_1 \cdot i_b)} - r \right] \cdot S_K \right\} - b(\omega_1 - i_{12} \omega_2)};$$

$$I_2 \omega_2 = e(\omega_1 - i_{12} \cdot \omega_2) i_{12} + b(\omega_1 - i_{12} \cdot \omega_2) i_{12} - M_2, \quad (16)$$

here e - eccentricity of the tension roller.

Belt transmission ratio:

$$i_{12} = \frac{D_2 EF}{D_1 \left\{ EF - (e^{\mu \alpha} - 1) \cdot \left[S_0 + a \cdot \cos\left(\varphi_1 - \frac{D_1}{D_2}\right) + \sqrt{\frac{D_6}{2} - a^2 \cdot \sin^2\left(\varphi_1 \cdot \frac{D_1}{D_6}\right) - \frac{D_6}{2}} \right] \right\} \cdot S_K}, \quad (17)$$

here: D_1 - diameter of the pulley on the electric motor shaft; D_b - tension roller diameter.

The resulting system of differential equations of the machine unit is nonlinear.

4. Conclusions.

A calculation diagram and mathematical model of the machine unit with the accelerator mechanism of the raw chamber of the saw fiber separator have been compiled, taking into account the inertial, elastic-dissipative properties of the elements, as well as the variability of the transmission ratio of the elastic transmission and the technological load from raw cotton.

The differential dependence of the gear ratio of a belt drive with an eccentric tension roller was obtained depending on the angle of rotation of the drive pulley.

A formula for the motion of a machine unit of a two-mass elastic system has been derived, which depends on the geometric characteristics of the transmission, the elastic-dissipative properties of the belt and the eccentricity of the tension roller.

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MATHEMATICAL MODELING OF THE OPERATION OF THE SYNCHRONIZER OF THE PLANETARY MECHANISM

Alimov B.M.¹, Kodirova Y.¹

¹Tashkent State Transport University, Tashkent, Uzbekistan
E-mail: lena.kodirova@mail.ru

Abstract. The article is devoted to the results of a theoretical and experimental study of forced oscillation of the epi-hypocyclic planetary mechanism synchronizer used to drive the spindles of the cotton machine apparatus. Spectrum of natural oscillation frequency of synchronizer is determined and mass moment of their inertia is set for stability in operating mode when satellite comes in and out of gear engagement of planetary mechanism. Periodic nature of system oscillations under action of external forces of satellite is considered. The graph of oscillations of actual excitation of the synchronizer in the form of a pulse semi-sinusoidal curve is obtained. We determined experimentally the resonance modes for high-speed rotations of 12 and 15 satellite carriers, with synchronizer arc length $l = 0.054$ m, synchronizer moment of inertia $J = 0.00003$ Nms² and spring stiffness $l = 0.078$ N/m. The rotation speed of both carriers was at different modes in the range $n=100...150$ rpm. The beam diagram is built that allows you to make delaminations from the resonant modes of operation of the synchronizer. The forbidden zones of the natural frequencies of the synchronizer, for 12 satellite carriers, were the frequencies P_1 and P_2 , which varied in the interval $P = 20-30$ Hz, and for 15 satellite carriers they varied in the range $P = 25.6 ... 38$ Hz.

Keywords: oscillation, frequency, satellite, carrier, synchronizer, moment of inertia, planetary mechanism.

1. Introduction.

Planetary mechanisms use tooth gear [1, 2] lantern gear [3], friction wheels [4] and belt gears [5, 6], which are used in the drive of the spindles of the cotton machine apparatus. Also, planetary mechanisms having a compact design have been widely used in the national economy: in the solution mixing mechanisms [7, 8] of the food industry, especially in the design of various mixers [9-12], for treatment facilities and reservoirs [13, 14] in the ventilation of waste water treatment [15]. In operation [16, 17] kinematic scheme of epicyclic planetary mechanism is given for their use in internal combustion engine with creation of equation of kinematic motion of satellites. At work [18], a kinematic scheme of an epi-hypocyclic mechanism for driving spindles is given, in which gears are designed in the form of two toothed segments, one is designed for epicyclic motion of satellites, the other segment is made with hypocyclic toothed engagement, therefore, at one rotation of the carrier, the satellites make two different rotational movements – epicyclic and hypocyclic. Brake devices and synchronizers for smooth engagement of teeth are installed in transition zones.

2. Materials, methods, and object of study.

The article proposes theoretical developments of a planetary mechanism for driving spindles, which were carried out according to the methodology of the basic positions of the theory of oscillation and stability, as well as theoretical mechanics and technological processes of the cotton harvester. Lagrange equation is used in oscillatory motion of synchronizer and results of machine analysis of graphs according to specially set experimental graphs. The processing of the results was carried out using mathematical statistics and computer graphics.

3. Results and discussions.

In the operating mode of direct rotation, under the action of periodically changing forces from the side of the rotating carrier, when the satellite enters and leaves the engagement, the synchronizer performs a forced angular oscillation about the rotation axis D (Fig. 1). The use of well-known theoretical methods of calculation with the experimental studies performed on a bench installation made it possible to establish the most successful design forms of the synchronizer of the planetary mechanism. This synchronizer ensures the normal operation of the mechanism in the absence of resonant modes. The synchronizer of the planetary mechanism (Fig. 1), when disturbed by the tooth gear, moves through the angle φ . For the generalized coordinate we take the angle φ . The considered mechanical system has one degree of freedom.

The kinetic energy E of the synchronizer rotating around its support axis D is equal to

$$E = \frac{1}{2} \cdot J_D \cdot \dot{\varphi}^2 \quad (1)$$

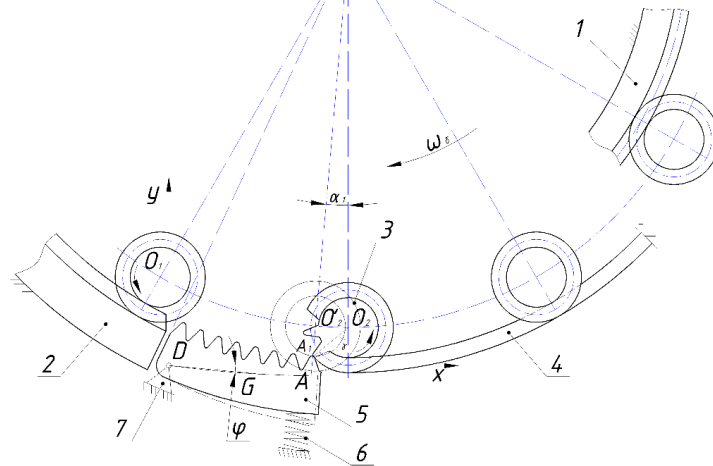


Figure 1. – Kinematic diagram of synchronizer operation

here J_D is the mass moment of inertia of synchronizer relative to rotation axis D , kgm^2 $J_D = J_S + m \cdot l_1^2$; J_S is the mass moment of inertia of the synchronizer relative to the center of gravity $\text{kg} \cdot \text{m}^2$; where m is the mass of the synchronizer, kg ; $m = G/g$, G is the gravity link at the center of the synchronizer, H ; g is the acceleration of gravity, 9.81 m/s^2 ; l_1 is the distance from support point to synchronizer center of gravity, m .

When the synchronizer is rotated by an angle $d\varphi$, the spring is shortened by dS , then

$$dS = l_2 \cdot d\varphi$$

In this case, the potential energy of the system will be

$$\Pi = \frac{CdS}{2} = \frac{C}{2} \cdot l_2^2 (d\varphi)^2 \quad (2)$$

here l_2 is the distance from the center of the spring axis to the hinge D of the synchronizer support axis, m ; C – coefficient of the spring constant.

Elementary operation of perturbing force

$$\delta A_\varphi = M_r \cdot \delta\varphi \quad (3)$$

where M_r – torque of the carrier, $H \cdot m$;

$$M_r = F_r \cdot l$$

F_r is the radial force acting on the first tooth of the synchronizer segment, H ;

l is the distance from the thrust axis D to the top of the first synchronizer tooth, m. The force acting on F_r synchronizer tooth is found according to experimental data. Considering that the periodic nature of the oscillations of the system under the influence of the external forces of the satellite gear, during the rotational movement of the carrier of the planetary mechanism, occurs through the synchronizer, then it is the impulse character of the semi-sinusoidal curve and is a perturbing force F . And this perturbing force can be decomposed into a Fourier series, then

$$F = F_A \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right] \quad (4)$$

where F_A is the axial force acting in gear engagement of cylindrical gears, H ; ω is the satellite angular velocity, $\omega = 130.08$ radian/s; t is time, s.

Substituting numerical values and solving at Math CAD, we obtain a graph of oscillation in the actual forced of the synchronizer $F = 0.048$ s (Fig. 2).

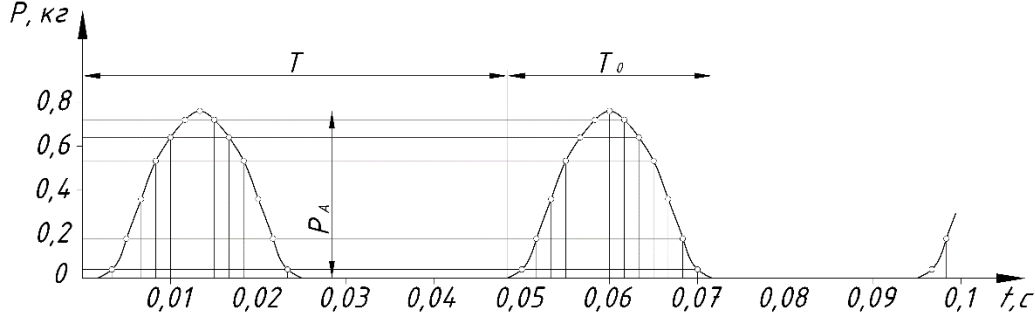


Figure 2. – Satellite tooth hit curve graph by synchronizer tooth

Knowing the time of one circulation of the carrier of the planetary mechanism $t = 60/n$, s we determine one cycle of the period F of the satellite for the carrier having the number of satellites $z = 12$, and the carrier rotation speed $n = 102.5$ cir/min, then we find the cycle time $F = 60/z \cdot n$. Axial reinforcement F_A engageable cylindrical gears

$$F_A = F_t \cdot t g \alpha_w$$

here α_w is the gearbox engagement angle $\alpha_w = 20^\circ$; F_t is circle force, $H F_t = 2M/d_\partial$; M is the torque of the carrier of the planetary mechanism, we obtain from experimental data, $N \cdot m$; d_∂ is the diameter of the carrier along the centers of the satellites, m; then

$$F_A = 2M/d_\partial \cdot t g \alpha_w = 2Mt g \alpha_w / d_\partial$$

using the Lagrange equation [12], we obtain the oscillation equation of the synchronizer

$$J_D \cdot \ddot{\varphi} + cl^2 \cdot \varphi = M_r$$

or

$$J_D \cdot \ddot{\varphi} + cl^2 \cdot \varphi = F_t \cdot l \quad (5)$$

substituting from equation (4) to (5), we obtain

$$J_D \cdot \ddot{\varphi} + cl^2 \cdot \varphi = F_A \cdot l \left[\frac{1}{4} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{4} + \frac{\cos 6\omega t}{35} \right) \right] \quad (6)$$

or

$$\varphi + \frac{cl^2}{J_D} \varphi = \frac{F_A \cdot l}{J_D} \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right] \quad (7)$$

If equation (7) without the right side equals zero

$$\ddot{\varphi} + \frac{cl^2}{J_D} \cdot \varphi = 0,$$

then the characteristic equation will J_D be

$$r^2 + \frac{cl^2}{J_D} = 0,$$

then its roots $r = \pm P_i$,

where P_i is natural frequency system $P_i = l \sqrt{\frac{c}{J_S}}$

from here $\varphi_1 = \cos pt$; $\varphi_2 = \sin pt$.

Let us write down the solution of equation (7) as follows:

$$\varphi = C_1(t) \cos pt + C_2(t) \sin pt \quad (8)$$

Let's compose a system of equations to find $C_1(t)$ and $C_2(t)$

$$\begin{aligned} C_1'(t)\cos pt + C_2'(t)\sin pt &= 0, \text{ then} \\ -pC_1'(t) \cdot \sin pt + pC_2'(t) \cdot \cos pt &= \\ = \frac{F_A \cdot l}{J_D} \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right] \end{aligned} \quad (9)$$

solving this system, we find

$$\begin{aligned} C_1'(t) &= -C_2'(t) \cdot \tan pt ; \\ C_2'(t) \cdot \sin pt \cdot \tan pt + C_2'(t)\cos pt &= \\ = \frac{p_A \cdot l}{I_D \cdot p} \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right], \end{aligned}$$

and from this system of equations, we find the following values

$$\begin{aligned} C_2'(t) &= \frac{\cos pt}{\sin^2 pt + \cos^2 pt} \cdot \\ &\cdot \frac{F_A \cdot l}{J_D \cdot p} \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right] \\ C_1'(t) &= \frac{F_A \cdot l \cdot \sin pt}{J_D \cdot p} \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{\cos 2\omega t}{3} + \frac{\cos 4\omega t}{15} + \frac{\cos 6\omega t}{35} \right) \right] \end{aligned}$$

Integrating these values gives

$$\begin{aligned} C_1(t) &= \frac{F_A \cdot l}{J_D \cdot p} \int \sin pt \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\cos 2\omega t + \frac{1}{15} \cos 4\omega t + \frac{1}{35} \cos 6\omega t \right) \right] dt = \\ &= \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi \cdot p} \cos pt - \frac{1}{4} \left[\frac{1}{p - \omega} \cdot \sin(p - \omega) t - \frac{1}{p + \omega} \sin(p + \omega) t \right] \right. \\ &\quad - \frac{1}{3\pi} \left[\frac{1}{p + 2\omega} \cos(p + 2\omega) t + \frac{\cos}{p + \cos 2\omega} (p - 2\omega) t \right] \\ &\quad - \frac{1}{15\pi} \left[\frac{1}{p + 4\omega} \cos(p - 4\omega) t \right. \\ &\quad \left. \left. - \frac{1}{35} \left[\frac{1}{p - 6\omega} \cos(p - 6\omega) t + \frac{1}{p - 6\omega} \cos(p - 6\omega) t \right] \right] \right\} + k_1 \end{aligned}$$

we also find

$$\begin{aligned} C_2(t) &= \frac{F_A \cdot l}{J_D \cdot p} \int \cos pt \left[\frac{1}{\pi} + \frac{1}{2} \sin \omega t - \frac{2}{\pi} \left(\frac{1}{3} \cos 2\omega t + \frac{1}{15} \cos 4\omega t + \frac{1}{35} \cos 6\omega t \right) \right] dt = \\ &= \frac{1}{35\pi} \cos 6\omega t] dt \\ &= \frac{F_A \cdot l}{J_D \cdot p} \left[\frac{1}{\pi p} \sin pt + \frac{1}{4} \left[-\frac{1}{p + \omega} \cos(p + \omega) t + \frac{1}{p - \omega} \cos(p - \omega) t \right] \right. \\ &\quad - \frac{1}{3\pi} \left[\frac{1}{p - 2\omega} \cdot \sin(p - 2\omega) t + \frac{1}{p + 2\omega} \sin(p - 2\omega) t \right] \\ &\quad + \frac{1}{15} \left[\frac{1}{p - 4\omega} \sin(p - 4\omega) t + \frac{1}{p + 4\omega} \sin(p - 4\omega) t \right. \\ &\quad \left. \left. + \frac{1}{35} \left[\frac{1}{p - 6\omega} \sin(p - 6\omega) t + \frac{1}{p - 6\omega} \sin(p - 6\omega) t \right] \right] \right] + k_2 \end{aligned}$$

where k_1 and k_2 – integration derivatives.

And after substituting the values $C_1(t)$ and $C_2(t)$ into equation (8), we get the rotation angle φ of the synchronizer around the axes of support D .

$$\begin{aligned}\varphi = & \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi \cdot p} \cos pt - \frac{1}{4} \left[\frac{1}{p - \omega} \sin(p - \omega) t - \frac{1}{p + \omega} \sin(p + \omega) t \right] \right. \\ & - \frac{1}{3\pi} \left[\frac{1}{p - 2\omega} \cos(p - 4\omega) t - \frac{1}{p - 2\omega} \cos(p - 2\omega) t \right] \\ & - \frac{1}{15\pi} \left[\frac{1}{p - 4\omega} \cos(p - 4\omega) t + \frac{1}{p - 4\omega} \cos(p - 4\omega) t \right] \\ & \left. - \frac{1}{35\pi} \left[\frac{1}{p - 6\omega} \cos(p - 6\omega) t + \frac{1}{p - 6\omega} \cos(p - 6\omega) t \right] \right\} \\ & + \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi \cdot p} \sin pt + \frac{1}{4} \left[\frac{1}{p - \omega} \cos(p - \omega) t - \frac{1}{p + \omega} \cos(p + \omega) t \right] \right. \\ & - \frac{1}{35} \left[\frac{1}{p - 2\omega} \sin(p - 2\omega) t + \frac{1}{p + 2\omega} \sin(p + 2\omega) t \right] \\ & - \frac{1}{15\pi} \left[\frac{1}{p - 4\omega} \sin(p - 4\omega) t + \frac{1}{p - 4\omega} \sin(p + 4\omega) t \right] \\ & \left. - \frac{1}{35\pi} \left[\frac{1}{p - 6\omega} \sin(p - 6\omega) t + \frac{1}{p + 6\omega} \sin(p + 6\omega) t \right] \right\} \sin pt + k_2 \sin pt\end{aligned}$$

After the corresponding transformations, we get

$$\varphi = \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi \cdot p} + \frac{p \sin \omega t}{2(p^2 - \omega^2)} - \frac{2p \cos 2\omega t}{3\pi(p^2 - 4\omega^2)} - \frac{2p \cos 4\omega t}{15\pi(p^2 - 16\omega^2)} - \frac{2p \cos 6\omega t}{35\pi(p^2 - 36\omega^2)} \right\} + k_1 \cos pt + k_2 \sin pt \quad (10)$$

Speed of synchronizer movement around the axes of support D

$$\begin{aligned}\varphi = & \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{p \omega \cos \omega t}{2(p^2 - \omega^2)} - \frac{2p}{\pi} \left[\frac{-2\omega \sin 2\omega t}{3(p^2 - 4\omega^2)} - \frac{4\omega \sin 4\omega t}{15(p^2 - 16\omega^2)} - \frac{6\omega \sin 6\omega t}{35(p^2 - 36\omega^2)} \right] \right\} - k_1 p \sin pt \\ & + k_2 p \cos pt\end{aligned}$$

with $t = 0$ and $\varphi = 0$, $\dot{\varphi} = 0$ and hence to determine the derivatives of the constants k_1 and k_2 we solve the following system of equations

$$\frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi} - \frac{2p}{\pi} \left[\frac{1}{3(p^2 - 4\omega^2)} + \frac{1}{15(p^2 - 16\omega^2)} + \frac{1}{35(p^2 - 36\omega^2)} + k_1 = 0 \right] \right\}; \frac{F_A \cdot l}{J_D} \cdot \frac{\omega}{2(p^2 - \omega^2)} + k_2 p = 0. \quad (11)$$

From the equation of the system (11) we have

$$\begin{cases} k_1 = \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{1}{\pi p} - \frac{2p}{\pi} \left[\frac{1}{3(p^2 - 4\omega^2)} + \frac{1}{15(p^2 - 16\omega^2)} + \frac{1}{35(p^2 - 36\omega^2)} \right] \right\}; \\ k_2 = \frac{F_A \cdot l \omega}{2J_D p (p^2 - \omega^2)}. \end{cases}$$

And after substituting the constant derivatives of k_1 and k_2 into equation (10), we get the form of oscillations of the angle φ of the synchronizer rotation around the support axis during the operating mode of the harvester spindle drive

$$\varphi = \frac{F_A \cdot l}{J_D \cdot p} \left\{ \frac{(1 - \cos pt)}{\pi p} + \frac{p \sin \omega t}{2(p^2 - \omega^2)} - \frac{2p}{\pi} \left[\frac{\cos 2\omega t - \cos pt}{3(p^2 - 4\omega^2)} - \frac{\cos 4\omega t - \cos pt}{15(p^2 - 16\omega^2)} + \frac{\cos 6\omega t - \cos pt}{35(p^2 - 36\omega^2)} \right] - \frac{\omega \sin pt}{2(p^2 - \omega^2)} \right\} \quad (12)$$

The period of forced oscillations of the synchronizer system is

$$T = \frac{2\pi}{l} \sqrt{\frac{J_D}{c}} \quad (13)$$

Frequency of disturbing forces of ω_{dis} gear at rotation of carrier, which acts at synchronizer of planetary mechanism in operating mode, is found

$$\omega_{dis} = K \frac{Z_C \cdot n}{60} \quad (14)$$

where K is the multiplicity of changes in frequencies of forced oscillations affecting the synchronizer $K = 1, 2$; Z_C is the number of satellites; n is the rotational speed of the carrier.

At rotational speed of the carrier, $n = 102.5$ cir/min and number of satellites $z = 12$, get $\omega_{dis} = 20.5 \cdot K$, and at $z = 15$, $\omega_{dis} = 25.6 \cdot K$. Considering that the synchronizer is affected by one factor – from the side of the pinion gear and when the teeth engage with the synchronizer, the

multiplicity of changes in frequencies, forced vibrations will be equal to $K = 1$, then $\omega_{dis.1} = 20.5 \text{ Hz}$ and $\omega_{dis.2} = 25.6 \text{ Hz}$.

To define the adjustment coefficient $Z_H = \frac{\omega_{dis}}{P}$ find numerical values of natural frequency P of synchronizer. To do this, experimentally setting the long arc of the synchronizer $l = 0.054 \text{ m}$ and its weight $G = 0.19 \text{ kg}$, determine the moment of inertia of the synchronizer $J_D = 0.0000311 \text{ N} \cdot \text{m} \cdot \text{s}^2$ and spring constant $C = 0.078 \text{ H/m}$. By substituting these values, we get, $P_1 = 2.7 \text{ Hz}$ – for a 12 satellite carrier and $P_2 = 3.25 \text{ Hz}$ – for 15 satellite carrier. By matching these values, we see that $\omega_{dis} \neq P$, at the same time $\omega_{dis} > P$, then $Z_{H1} = \frac{\omega_{dis}}{P_1} = 7.58$ and $Z_{H2} = 7.88$. Thus, the mechanism relates to a system of forced high frequency oscillations. Find the dynamic coefficient of the mechanism by the formula

$$\mu = \left| \frac{1}{1 - \frac{\omega_{dis.2}^2}{P^2}} \right| \quad (15)$$

That is, from the graph (Fig. 3) we see that the amplitude of the resonant oscillations is very small and is located on the right side of the graph within $\frac{\omega_{dis}}{P_1} = 7.58$. If the tuning coefficient value $z = 7.58$, the dynamic coefficient μ tends to zero, i.e. $\mu = 0.017$.

After planetary determination of the spectrum of natural frequencies of the system according to the method of E.L. Ayrapetov - M.D. Genkin for gears [19-20], the so-called “beam diagram” is built (Fig. 4), which allows us to conclude that it is possible to detach from resonant modes. Along the abscissa axis of the beam diagram, the rotation speed of the carrier n is deposited, and along the axis the ordinate of the frequency of the disturbing factor $\omega_{dis.1}$ and $\omega_{dis.2}$. Further we apply own frequency of the synchronizer P_1, P_2, P_3, P_4 on the beam chart in the form of the straight lines parallel to abscissa axis. The intersection points of the lines, for example, P_1 and P_2 determine resonant speed modes of rotation of the 12 satellite carrier. Since the operating speed of the satellite carrier $n = 100 \dots 150 \text{ cir/min}$, the forbidden zones of the natural frequencies of the synchronizer are P_1 and P_2 , which vary in the interval $P = 20 - 30 \text{ Hz}$. Therefore, the natural frequency of the synchronizer must be lower than P_2 . For the 15 satellite carrier, the forbidden zones of natural frequencies are synchronizers P_3 and P_4 , which fluctuate in the interval $P = 25.6 \dots 38 \text{ Hz}$. From this beam diagram you can determine the optimal parameters of the synchronizer: moment of inertia J_D , arc length l and spring constant C . By setting the condition $\omega_{dis} \geq P$, we get $20.5 \geq l \sqrt{\frac{C}{J_D}}$ or $l \leq 20.5 \sqrt{\frac{J_D}{C}}$. Selecting different values of moments of inertia J_D synchronizer and spring constant C determined optimal length of arc l of synchronizer. At the selected values of J_D and l , spring constant C was calculated: $C \leq J_D \left(\frac{20.5}{l} \right)^2$.

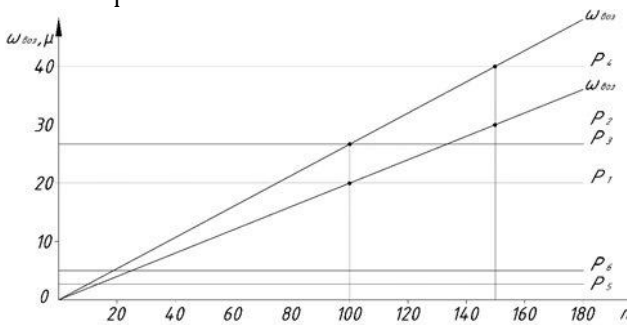


Figure 3. – Amplitude-frequency response graph planetary mechanism

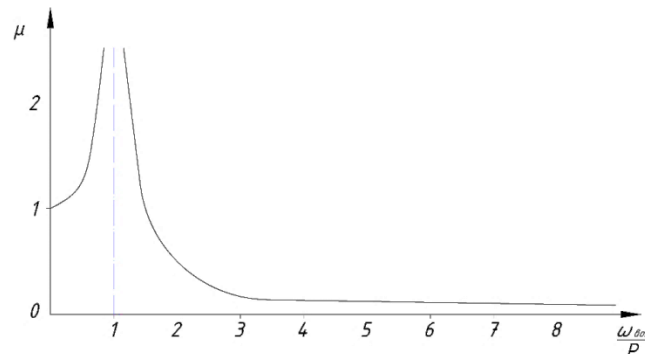


Figure 4. – Beam diagram of the planetary mechanism

Considering that the length l of the synchronizer for the 12 satellite carrier was determined from the geometric synthesis within the range of l equal to $0.05-0.06 \text{ m}$, then the spring constant C is found experimentally, then the response time of the t_{cp} synchronizer at various modes of rotation of the satellite carrier will be considered a criterion for calculating the mechanism. Having established that curve 2 (Fig.4) – the synchronizer actuation time at the selected spring constant C is equal to 0.078

H/m, is located near curve I - the required time for the synchronizer actuation in the theoretical way and quite manages to function with the approach of the subsequent gear, then you can choose the spring constant $C = 0.078 \text{ N/m}$.

Knowing two parameters of the synchronizer – length of an arch of l and $J_D = 5.42 \cdot 10^{-6} \text{ N} \cdot \text{m} \cdot \text{s}^2$ we will establish rigidity of a spring constant C , from a condition defined the lower bound of the moment of inertia of the $J_s = 0.00000542 \text{ N} \cdot \text{m} \cdot \text{s}^2$ synchronizer, in this case own frequency of the synchronizer made $P = 10.5$ and when choosing the upper bound we will receive it $J_s = 0.00000542 \text{ N} \cdot \text{m} \cdot \text{s}^2$.

In this case, the natural frequency P of the synchronizer will be less than the frequency of forced oscillations $P < \omega_{\text{dis}}$. In this case, forced oscillations of low frequency occur on the left side of the graph (Fig. 3). At the same time, the dynamic coefficient C increases from 1 to infinity. This means that the satellite drive mechanism will operate dynamically, which is highly undesirable. Therefore, the natural frequency of the synchronizer must be selected below $P = 20.5 \text{ Hz}$, i.e., the condition must be met:

$$J_D \leq C(l/20.5)^2 \text{-for a 12 satellite carrier and } J_D \leq C(l/25.6)^2 \text{-for a 15 satellite carrier.}$$

4. Conclusions.

Forced oscillations of synchronizer are investigated taking into account external forces of satellite of epi-hypocyclic planetary mechanism when entering gear engagement of internal segment wheel. The graph of oscillations of actual excitation of the synchronizer when entering the crown wheel gear is obtained. Resonant high-speed modes of the carrier of the planetary mechanism were determined and a forbidden zone of natural frequencies of the synchronizer was established, $P = 20.5 \dots 38 \text{ Hz}$, so for a 12-satellite carrier, the natural frequency P_1 should be lower than $P_1 = 20.5 \text{ Hz}$, and for a 15-satellite carrier lower than $P_2 = 25.6 \text{ Hz}$.

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DETERMINATION OF THE ROTATION FREQUENCY OF THE SATELLITES OF THE PLANETARY MECHANISM IN THE EPICYCLIC ZONE OF THE COTTON HARVESTER APPARATUS

Alimov B.M¹., Alimova D.B²., Mirzaeva M.M².

¹Tashkent State Transport University, Tashkent, Uzbekistan

²Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan

E-mail: alimova007@inbox.ru

Abstract. The work is devoted to determining the rotation frequency of satellites of a biplanetary mechanism for driving spindles in the working zone of the harvesting apparatus, in which the kinematic scheme of the epicyclic arrangement of the friction block for rotation with epicycloidal rotation of the spindle trajectory during cotton picking from a cotton bush is considered. The theoretical calculation was carried out on the basis of methods of analytical and descriptive geometry, as well as the kinematics of the movement of points on the surface of the satellite and the carrier of a planetary mechanism performing a complex plane-parallel motion, and an analytical equation was obtained for the number of revolutions of the satellite in the friction block zone, with the stationary position of the apparatus. Also, when turning the drum, the angles for the first and subsequent epicycloid loops are obtained, in which the spindle roller rolls without sliding along the outer circumference of the friction block of reverse rotation. The graph of the dependence of the number of rotations n of the spindle on the rolling radius r of the satellite roller in both zones: picking and removing cotton from the spindle. The graph of the dependence of the driver's rotation angle on the rolling radius r of the satellite roller for epicycloid loops is also constructed. When the harvesting unit is moved mobile, the curvature of the loop of the transcendental curve of the epicycloid is obtained. The rolling radius of the spindle roller during the mobile movement of the cotton harvester and the angular velocity ω_a of the driver are analytically obtained. The graph of the movement of the cotton harvester from the rolling radius r of the satellite roller in both zones of the friction block and belt drive is given.

Keywords: satellite, drive, driver, apparatus, rotation speed, epicycloid.

1. Introduction.

In all mechanisms of the spindle drive of the cotton harvesting machine, a biplanetary mechanism is used in which the spindles periodically change the direction of rotation during one revolution of the carrier, so when picking cotton, the spindles rotate counterclockwise and when removing them rotate clockwise, and in the transition zones the spindle reverses [1]. The study [2] provides a graphical representation of the change in the angle of approach of the spindle in the working zone of the harvester, depending on the diameter of the cotton boll. In [3], a model in the form of a deformed ball-cylinder is proposed and analytical expressions for calculating the working slot, the pitch of neighboring spindles and their number on the drum are given on its basis. In the study [4], with a single cotton harvest during experimental field experiments, it was revealed that with an increase in the working slot from 32mm to 36mm in front of a pair of drums, the clogging of the harvested cotton decreased by 1.3 times. In [5], an analytical equation is given for the required number of spindles on the drum and, in which, according to theoretical calculations, it was possible to recommend installing 14 spindle drums instead of 12 spindle drums. In [6], the kinematics of the satellite of a biplanetary mechanism with flexible and frictional transmission links is presented, in which the coefficient of elastic sliding of the roller roller along the belt and the crown block is revealed, which is equal to $\zeta_1=0.07...0.168$ and $\zeta_2=0.126...0.138$. In studies [7] in the MathCAD 15 programming environment graphs of the patterns of change of the hodograph are constructed the speed, the vector of absolute movement of the spindle tooth, the absolute speed and its active component at the specified speed modes of the machine. The work [8] presents the results of field experiments with sequentially arranged drums of 12, 14 and 15 spindles with the same diameters and

concludes that an increase in the number of spindles contributes to an increase in the completeness of cotton harvesting. In the studies [9], the geometric sliding of the drive rollers along flexible belt friction gears was taken into account due to the small size of their girth angle. In [10], the influence of various parameters on their performance was determined, such as: an increase in the length of the rutting and an increase in the speed of cotton harvesters. Foreign researchers [11-19], designers from various companies are working on the design of new high-performance energy- and resource-saving cotton harvesters that meet the quality and quantity of cotton harvested using machines.

2. Materials, methods, and object of study.

The work was carried out according to the methodology of the main provisions of the technological processes of harvesting raw cotton from a cotton bush, analytical and descriptive geometry, theoretical mechanics, theory of mechanisms and machines. The obtained numerical results were processed by methods of mathematical statistics.

3. Results and discussions.

The kinematic scheme (Fig.1) of an epi-hypocyclic biplanetary mechanism for driving spindles is considered, in which, when the spindle drum rotates clockwise, the rollers 3 of the spindles, in contact with the friction track, also rotate clockwise in the area of picking by the spindle from the cotton bush 5. Further, when the roller 3 exits the friction block, the V-belt zone is approached, the spindle is reversed in the opposite direction and rotates counterclockwise, the cotton is removed from the spindle by the puller 4.

The purpose of the study is to graphoanalytically determine the trajectory of the spindle tooth point and the number of revolutions of the spindle roller with an epicyclic arrangement of the spindle drive in the picking area of the cotton harvester in stationary and mobile movement of the cotton harvester.

Cotton picking by a spindle consists of three speed parameters: with the rotational movement of the drum ω_v ; rotation of the spindle around its axis ω_e and the linear speed V_C of the cotton harvester.

The two rotational speed parameters of the spindle drum ω_v and the spindle ω_e are considered in the horizontal H plane, and the movement of the cotton picker V_C occurs in the vertical V plane.

Therefore, we combine three links of the mechanism into one absolute motion, located in two planes V and H , in which the parametric velocities rotate at an angle of 90° . Therefore, in absolute motion, the spindle points consist of three speeds:

- 1 – during rotational movement ω_r , around its axis of the spindle;
- 2 – with a portable rotational ω_e movement of the spindle around the axis of the spindle drum;
- 3 – with the rectilinear movement V_C of the cotton harvesting machine.

In the first two cases, the movement of the velocity of the links of the planetary mechanism is considered in the horizontal H plane, here the angular velocities ω_s of the spindle in absolute motion are equal to the algebraic sum of the angular velocities in relative ω_r and portable ω_e movements [20]:

$$\omega_s = \omega_r + \omega_e \quad (1)$$

The spindle drive is a planetary mechanism, the outer solar wheel is mounted in blocks with a friction track 1, and the inner fixed track 2 is driven as a belt drive (Figure 1).

When the drum rotates ω_d clockwise, the spindle 3, coming out of the belt drive 2, rotates counterclockwise by inertia ω_3 and after contact with the friction block 1, reverses in the opposite direction and rotates ω_3 clockwise making an epicyclic trajectory of movement, the technological process of raw cotton from a cotton bush 4 takes place.

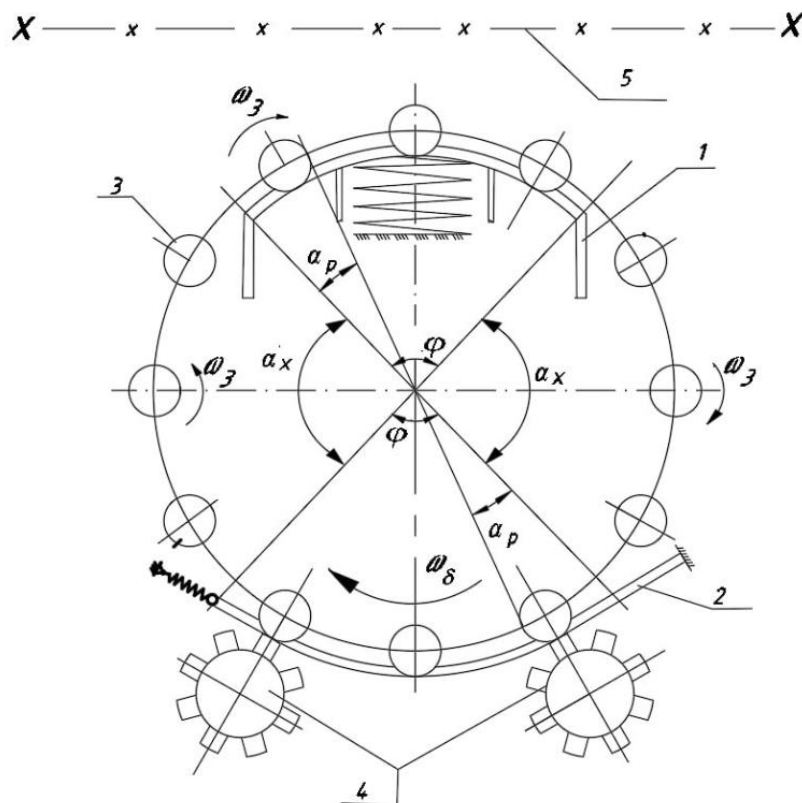


Figure 1. – Kinematic scheme of the epi-hypocyclic biplane mechanism for spindle drive

With the rectilinear movement of the harvesting machine, in the horizontal H plane it looks like the rectilinear speed V_e of the cotton harvester along the axis $X - X$ of the cotton row. The trajectory of the tractor wheel in the vertical V projection plane looks like a cycloid $A''K''B''$ (Figure 2), then the equation of the trajectory of the cycloid curve in parametric form in the vertical V projection plane looks like:

$$\begin{cases} X = R(r - \sin \varphi) \\ Y = R(1 - \cos \varphi) \end{cases}, \quad (2)$$

where R is the radius of the producing circle of the tractor wheel, m.
 φ is the angle of rotation of the producing circle of the wheel, radian.

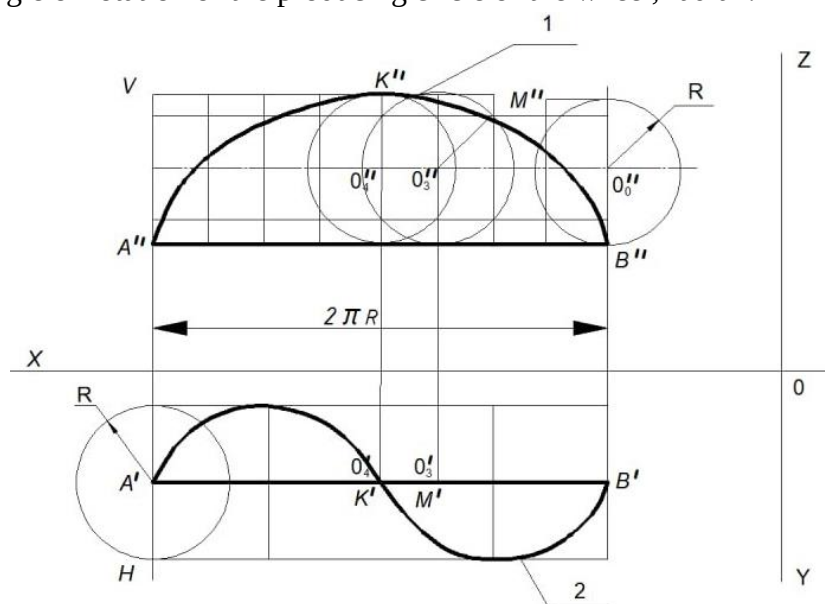


Figure 2. – Graphic representation of cycloid 1 and sinusoids 2 in two projections V and H

On the horizontal H plane of the projection, the cycloid looks like a rectilinear segment $A'B'$. Then, when rotating the drum without spindles, on a horizontal H plane, the trajectory of the circle of the driver looks like a sinusoid 2, and in the vertical V plane, as a vertical segment $A''B''$. With the spindles mounted on the drum, the trajectory of the spindle tooth looks like an epicycloid and a hypocycloid [20].

The parametric equation of the epicyclic trajectory of the spindle will be:

$$\begin{cases} X = R_H \cos \varphi_H - r_3 \cos(\varphi_3 - \varphi_H) \\ Y = R_H \sin \varphi_H - r_3 \sin(\varphi_3 - \varphi_H) \end{cases}, \quad (3)$$

where R_H is on the horizontal plane of the cycloid projection, the radius of the spindle drum along the centers of the spindle axis, mm; φ_H is the angle of rotation of the spindle drum, radian; r_3 is the radius of the spindle roller roller, mm; φ_3 is the spindle rotation angle, degree.

The capture and extraction of raw cotton from the boll by spindles occurs during the angle α_1 of approach of the spindle of the harvester, therefore, depends on the diameter value d_s (24-30mm) of the spindle, the width B (23-34mm) of the working slot of the harvester, the distance L between two spindles of adjacent drums, the number (12-14) pcs of spindles on one spindle drum, as well as the diameter d of the opened boll d_b (40.5 – 51.4) mm of cotton and all this varies within a small interval within the angle of entry of the spindles $\alpha_1 = 29^\circ \div 33^\circ$ and the angle of exit of the spindles within $\alpha_2 = 80^\circ \div 82^\circ$.

At the angle of entry of the spindle, cotton is wound on the surface of the spindle, and at the angle α_2 of exit of the spindle, the trapped raw cotton is retained.

Considering that the spindle drum of the cotton harvesting machine is a multi-satellite biplanetary mechanism, then for the convenience of maintaining calculation data on friction pads and reverse rotation belts, we assume that the roller rollers roll without sliding.

Graphoanalytic construction and calculation are performed in the horizontal H plane and all equations (1), (2) and (3) are combined in the horizontal plane.

Theoretically, the number of spindle rotations n in the working zone is determined by the ratio for the epicyclic movement of the satellite:

$$n = \frac{[(\varphi_w + \varphi_i) - \alpha_r](R + 2r_3)}{(2\pi \cdot r_3)} \quad (4)$$

here φ_w is the angle of girth of the roller rollers of the spindles of the friction block of the satellite drive $\varphi_p = 110^\circ$ (1,88 radian);

φ_i is the idling angle of spindle rotation, $\varphi_i = 60^\circ$ (1,05 radian), α_r is the spindle reverse angle, $\alpha_r = 28^\circ \dots 39^\circ$ accept $\alpha_r = 35^\circ$ (0,61 radian); R is the radius of the drum circumference, drawn along the centers of the spindle arrangement, $R = 146\text{mm}$; r_3 is the rolling radius of the spindle rollers, $r_3 = (11 \div 15)\text{mm}$.

Substituting numerical values into equation (4), we obtain a graph of the change in the number of rotations n of the spindle from the rolling radius r_r of the roller (Figure 3), here: 1 is in the belt drive zone; 2 is in the friction block zone.

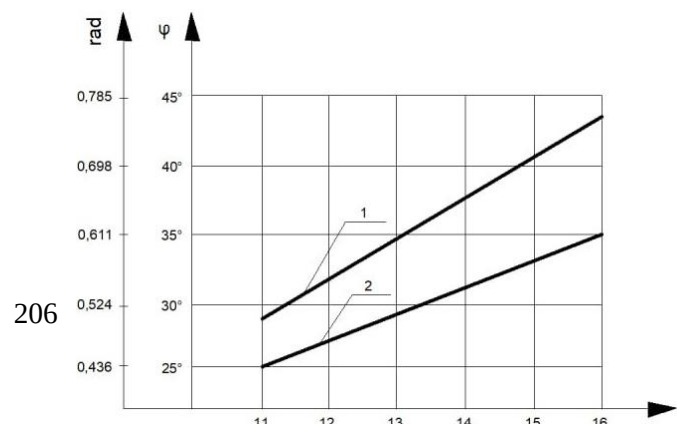
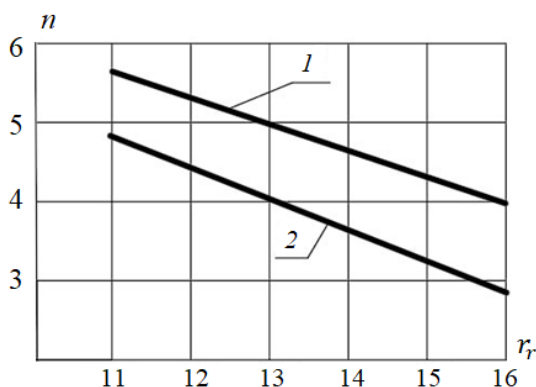


Figure 3. – Graph of the dependence of the number of rotation n of the spindle on the rolling radius of the roller

Figure 4. – Graph of the dependence of the angle of the drum on the rolling radius

It can be seen from the graph that with an increase in the rolling radius r_r of the spindle roller, the number of rotations n of the spindle decreases in both zones, so that $r_r = 16mm$ in the zone of the friction gear block, the number of revolutions of the spindle varies from $n = 4.8$ rotations to $n = 3.2$ rotations, and in the belt drive zone it is similar from $n = 5.7$ to $n = 4$ rotations. Also, from the graph (Fig. 4) it can be seen that with an increase in the radius of the roller roller r_r , the angle φ of rotation of the spindle drum increases, here 1 is in the belt drive zone, 2 is in the friction block zone.

When determining the number of rotations n of the spindle in the cotton picking zone by the spindle, the calculation and graphical construction of the device mechanism scheme are carried out in two stages: the first stage is built when the spindle drum rotates in a stationary position of the device, and the second stage when the cotton harvester X_1 is mobile.

Taking the scale factor of the length $\mu_l = 0.001m/mm$ in the drawing, we determine the dimensions of the radius R_3 of the drum of the images in the mechanism diagram $R_v = R_3 / \mu_l = 0.146 / 0.001 = 146mm$, where R_v is the radius of the drum in mm depicted in the mechanism diagram; R_v – the actual size of the drum radius in m.

We find the angle φ of rotation of the spindle drum for the first loop of the epicycloid, from point A to point B according to the formula:

$$\varphi_1 = \frac{2\pi \cdot r}{R_v - r} \quad \text{or} \quad \varphi_1 = 360^\circ \frac{r}{R_v - r} \quad (5)$$

After substituting the numerical values we get $\varphi_1 = 41^\circ 13'$. We put this angle to the left of the coordinate axis YOX and mark the starting point A and the end point B along the arc O_1O_1 , then we get the trajectory of the curvature of the epicycloid, the arc length AB of which is equal to: $AB = 2\pi \cdot r_3 = 6.28 \cdot 15 = 94.2mm$. We divide this arc length AB into 8 parts and draw an epicycloid loop at eight points (Figure 5).

At the second stage of the graphical construction, we take into account the mobile movement of the cotton harvesting machine and find for the first loop the distance X_1 when moving the end point B of the epicycloid while the machine is moving at a speed according V_c to the following formula:

$$X_1 = \frac{V_c \cdot 2\pi \cdot r}{\omega_v (R_v - r)} \quad (6)$$

here V_c is the speed of the machine MX-1.8 at the first high-speed cotton picking mode, $V_c = 4.8km/h$ or $V_c = 1.147m/s$; ω_v is angular velocity of the drum, $\omega_v = 12.56s^{-1}$; r is the rolling radius of the spindle rollers, $r = 15mm$; R_v is the radius of the drum circumference, drawn along the centers of the spindle arrangement, $R_v = 146mm$.

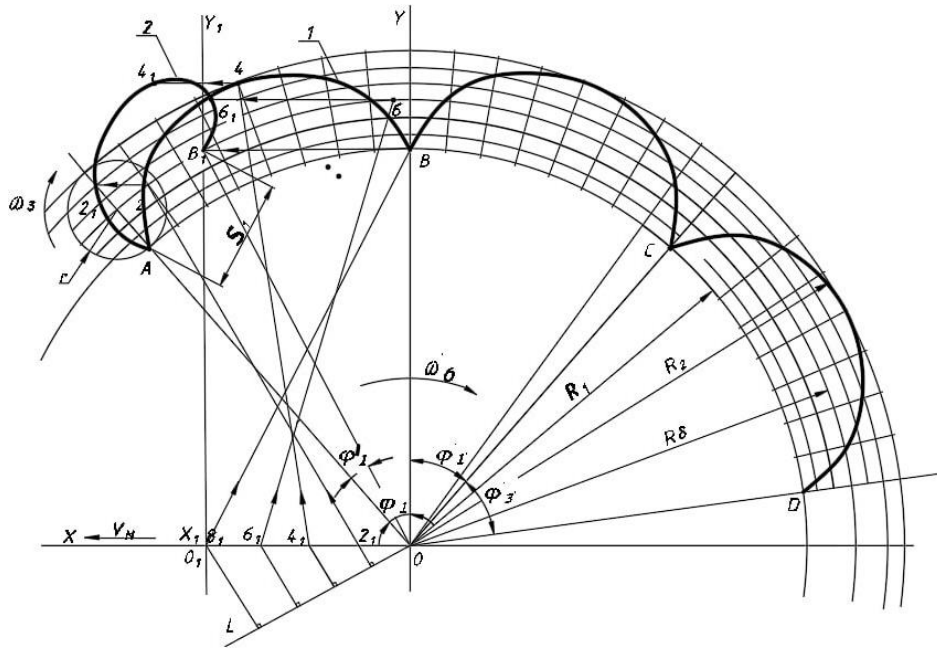


Figure 4. – Graphical construction of an epicycloid in stationary (1) and mobile (2) movement of the harvesting apparatus

Substituting digital values, we obtain $X_1 = 65.66mm$ and select a segment OX_1 along the abscissa axis and postpone the distance $OX_1 \parallel BB_1$ along the course of the machine, then the end point B of the epicycloids will be at a point B_1 when moving, we also carry out these procedures with other points 6_1 , 4_1 and 2_1 , and in this case, the points found will connect the starting point A sequentially with the point B_1 and get the transcendent curvature $\tilde{A}\tilde{B}_1$ of the epicycloids. If the arc length AB is denoted by S_1 then $AB = S$, then the analytical formula for the transcendental curvature of the epicycloids looks like:

$$S = 2\pi \cdot r - V_c \frac{2\pi \cdot r}{\omega_v(R-r)}, \quad (7)$$

here S is the shortest distance of two extreme points A and B_1 for the first loop of the transcendental hypocycloid curve during mobile movement of the device.

Substituting digital data into equation (7), we obtain $S = 40.8mm$.

From the triangle OAB , we determine the mobile angle ϕ'_1 of the first loop of the transcendental epicycloid $\sin \phi'_1 = \frac{S}{R_v - r} = 40.8/131 = 0.371$; $\phi'_1 = 18^\circ 07'$.

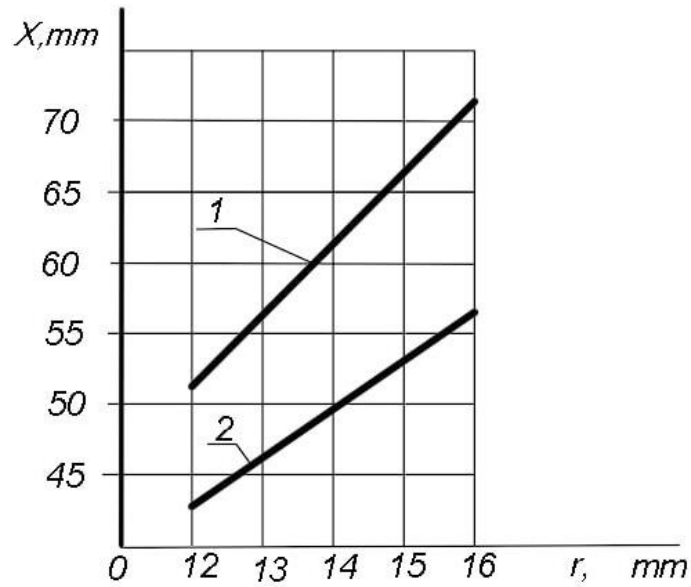


Figure 5. – The graph of the dependence of the movement X of the machine on the radius r of the spindle roller, where 1 is in the zone of the friction block; 2 is in the zone of the belt drive

If we select in the interval $d = 50 \div 60 \text{ mm}$ at the diameter of the open cotton boll, then the obtained value $S = 40.8 \text{ mm}$ at the mobile angle $\phi'_1 = 18^\circ 07'$ confirms that a spindle with a rolling radius of the roller $r = 15 \text{ mm}$ is quite sufficient to capture cotton from the cotton boll with one rotation of the spindle around its axis, and the maximum rotations are necessary to hold cotton on the spindle to the cotton removal zone from the spindle.

From equation (7), we find the required rolling radius r_r of the spindle roller during the mobile movement of the cotton harvester:

$$r = \frac{\sqrt{(2\pi\omega_v R - 2\pi V_C - S\omega_v) - 8\pi\omega_v^2 RS} - (2\pi\omega_v R - 2\pi V_C - S\omega_v)}{4\pi\omega_v} \quad (9)$$

and the angular velocity ω_v of the spindle drum during mobile movement of the device:

$$\omega_v = \frac{2\pi \cdot V_C \cdot r}{(R - r)(2\pi \cdot r - S)} \quad (10)$$

We take the resulting segment S as a criterion and, knowing the speed of movement of the machine V_C , the angular velocity of the drum ω_v , we set the required rolling radius of the spindle roller r . In this case, the radius of the serial spindle drum R_v is taken in all cases as a constant value $R_v = 146 \text{ mm}$. By varying the segment S in the selected interval $S_1 + S_i$, we set the optimal speed of movement of the machine V_C , the angular velocity of the drum ω_v and the rolling radius r of the spindle roller.

Having selected a segment S in the range of the diameter of the opened cotton boll $d = 50 \div 60 \text{ mm}$, we make calculations for the first operating speed of the MX-1.8 cotton harvester, where $V_C = 1.1477 \text{ m/s}$, $\omega_v = 12.56 \text{ s}^{-1}$ and after substituting numerical data into equation (9), we obtain $r = 13 \div 15 \text{ mm}$.

4. Conclusions.

The analytical method has been used to obtain the formula for the equation number of revolutions of the spindle in the cotton picking zone by the spindle with an epicyclic arrangement of the kinematic scheme of the biplanetary mechanism and a graph of the dependence of the spindle rotation on the rolling radius of the roller has been constructed, the angle of approach of the spindle at a stationary position is also established: $\phi = 41^\circ 18'$ and mobile movement of the unit $\phi = 18^\circ 07'$ at

the rolling radius of the cylinder roller $r = 15\text{mm}$. The transcendental curvature of the epicycloids was obtained graph analytically during the mobile movement of the harvesting apparatus.

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STABILITY OF A TWO-STAGE PNEUMO-HYDRAULIC RELAY-CONTROLLED SYSTEM WITH A DEAD ZONE

Annakulova G.K.¹, Astanov B.Zh.¹, Saidov S.A.¹

¹Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev, Tashkent, Uzbekistan

E-mail: annaqulova_g@mail.ru

Abstract. The problem of analyzing the stability of a two-stage pneumo-hydraulic relay-controlled system with a dead zone operating in a sliding mode is discussed in the article. Using Zubov's method, the equations of motion of a pneumo-hydraulic nonlinear system are reduced to a partial differential equation of a special form. A discontinuous Lyapunov function satisfying the Lyapunov theorem and an estimate of the time the regulator reaches the dead zone of the pneumo-hydraulic system are obtained. Diagrams of Lyapunov functions and their level lines were constructed for various parameters of the pneumo-hydraulic drive, and the region of stability of the system was determined. It was established that a) the stability region of a system with a relay dead zone is wider than that of a system with an ideal relay; b) external disturbance helps to reduce the stability of the system, and an increase in the feedback coefficient helps to improve stability. The time to reach the sliding surface of the control system was studied by numerical simulation and using the Lyapunov function. The relationship between the parameters obtained shows that the stability of the system improves when a pulse is set in the first relay by increasing the servomotor time, i.e. reducing the operation speed of the tracking mechanism, which confirms the results of the studies conducted by V.V. Petrov on the influence of servomotor time on the suppression of auto-oscillations of two-stage servomechanisms.

Keywords: Lyapunov function, Zubov method, stability, pneumo-hydraulic system, relay control

1. Introduction.

The development of automation of production processes and the introduction of control systems in various fields of technology pose new complex tasks. The requirements for accuracy and speed of systems and their elements are increasing. Servomechanisms that are included in control systems as components play a special role. Under certain conditions, auto-oscillations and sliding modes arise in the servomechanism; the stability of studying these conditions and determining the parameters under which the characteristic operating modes of the servomechanism change occur is an urgent task.

Among the various control principles that make it possible to best manage the object of regulation, the attention of modern scientists and specialists is attracted by systems in which control actions are discontinuous functions of its coordinates and external influences, in particular, in systems with relay control elements [1,2]. Such control systems are used in many fields of technology. When analyzing relay control systems, one encounters problems associated with the property called finite-time stability [2,3], and with estimating the stabilization time (setting time). These problems are solved by the method of Lyapunov functions [4,5,6]. For specific relay systems, the corresponding Lyapunov functions can be nonsmooth [7,8,9] and discontinuous [10].

In [11,12], definitions of a solution to a discontinuous system were introduced and the main properties of solutions to differential equations with a continuous right-hand side were presented. The problem of synthesis in systems with discontinuous control actions comes down to the choice of surfaces in the phase space on which the control functions undergo discontinuities [13]. The possibilities of sliding movements were identified and used in [14,15,16]. In [17], an experimental setup was studied, which is a prototype of a standard hydraulic logging crane. A design of a regulator with a time-varying amplifier was proposed. Lyapunov analysis is presented to demonstrate the stability and convergence properties of the algorithms. To analyze the asymptotic stability of the origin of coordinates, a continuous positive definite Lyapunov function is determined, such that for any solution to the problem under consideration there is a monotonic decrease. The case when the Lyapunov function is continuously differentiable was studied in [18].

Articles [19] and [20] present results for the case when the Lyapunov function is discontinuous. In [21], a Lyapunov function was developed to analyze the stability of the system for the "Twist" controller to implement the second-order sliding mode. A generalization of Zubov's method for constructing the Lyapunov function, based on the application of the method of characteristics, is presented. Reference [22] is devoted to solving the problem of eliminating discontinuities in nonlinear control systems of variable structure. The solution procedure is based on the application of a control approach that provides a second-order sliding mode. The study in [23] presents a modern Lyapunov method for analyzing stability within the framework of mathematical nonlinear control theory.

Particular attention is paid to the problem of the existence of Lyapunov functions and their regularity. There are sections on the Zubov method and Lyapunov functions for discontinuous and cascade systems.

In [24], frequency domain analysis of second-order sliding mode algorithms, in particular, the twisting algorithm, was considered using the describing function method. It is shown that in the presence of an actuator, the transient process can converge to periodic motion. A comparison is made of periodic solutions in systems with higher-order sliding mode controllers with oscillations arising in classical systems of sliding mode with actuators.

The study in [25] is devoted to recent advances in developing feedback laws for global stabilization of nonlinear systems using state or output feedback, in particular, the analysis of the stability of interconnected nonlinear systems. In [26], second-order sliding modes were used to accurately limit the second relative degree or simply to avoid discontinuity, i.e., in cases where the use of a standard (first-order) sliding mode is possible or impossible. In [27], a control algorithm was developed for the global stabilization of an electric servomotor operating under conditions of uncertainty, in the presence of complex nonlinear phenomena such as backlash, stick-slip motion, etc. Stability analysis conducted in the framework of Lyapunov functions, extending to discontinuous dynamic systems, additionally revealed the finite time convergence of the system's trajectories to the equilibrium point. Reference [28] presents the mathematical apparatus necessary for analyzing the asymptotic, exponential, finite, and fixed stability of discontinuous control systems using discontinuous Lyapunov functions.

2. Materials, methods, and object of study.

Consider the problem of the stability of a tracking mechanism made in the form of a two-stage amplifier, in which the control elements of each amplification stage represent a relay with a dead zone (Fig. 1) [29].

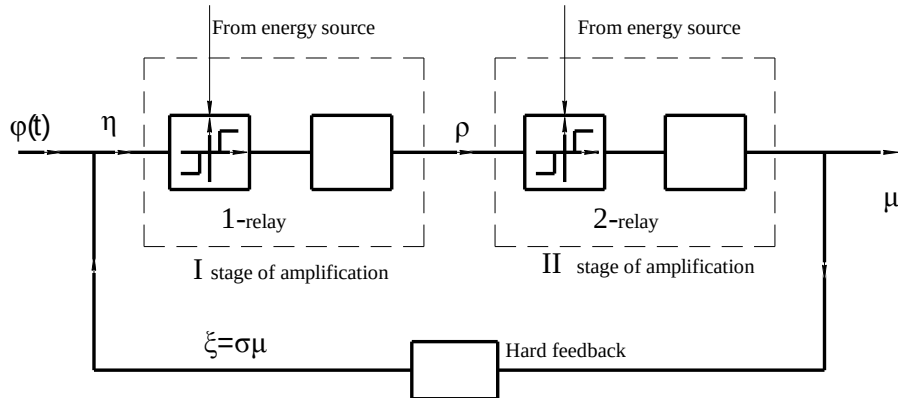


Figure 1. – Block diagram of a tracking relay-controlled mechanism with a dead zone

3. Statement of the problem.

The equations of motion of a two-stage servo mechanism under relay control in the first amplification stage with rigid feedback are written according to work [29] in the following form:

$$T_p \dot{\rho} + \rho = u_1(\eta), \quad (1)$$

$$T_f \dot{\xi} = u_2(\rho), \quad (2)$$

$$\eta = \frac{1}{\delta} \varphi - \xi \quad (3)$$

where φ is the relative value of the input coordinate, η is the relative value of the error, ρ is the relative value of the output coordinate of the first amplification stage, μ is the relative value of the output coordinate of the second amplification stage, ξ is the relative value of the reduced coordinate of the output of the tracking mechanism, δ is the constant coefficient, T_p is the setting

time of the pulse of the first amplification stage, T_f is the reduced time of the servomotor $T_f = T_s / \sigma$, σ is the feedback coefficient, T_s is the constant of the servomotor, $u_1(\eta)$ and $u_2(\rho)$ are the characteristics of the control elements of the first and second amplification stages (Fig. 1).

$$\begin{aligned}
 u_1(\xi) &= -u_1(\eta) \\
 u_1(\xi) &= \begin{cases} -1 & \text{for } \xi < -(\xi_\xi), \\ 0 & \text{for } -(\xi_\xi) < \xi < (\xi_\xi), \text{ for } \dot{\xi} > 0 \\ 1 & \text{for } \xi > (\xi_\xi), \end{cases} \quad (4) \\
 u_1(\xi) &= \begin{cases} 1 & \text{for } \xi > (\xi_\xi), \\ 0 & \text{for } (\xi_\xi) > \xi > -(\xi_\xi), \text{ for } \dot{\xi} < 0 \\ -1 & \text{for } \xi < -(\xi_\xi), \end{cases} \\
 u_2(\rho) &= \begin{cases} 1 & \text{for } \rho > \rho_\xi, \\ 0 & \text{for } -\rho_\xi < \rho < \rho_\xi, \\ -1 & \text{for } \rho < -\rho_\xi, \end{cases} \quad (5)
 \end{aligned}$$

where $2\xi_\xi, 2\rho_\xi$ are the relative values of the relay dead zones of the first and second amplification stages.

To study the stability of the system of equations (1) - (3), considering Eqs. (4) and (5), we reduce them to the following form:

$$\begin{aligned}
 \dot{\xi} &= -c_p b_1(\xi) u_1(\xi) + c_f b_2(\rho) u_2(\rho) - \frac{1}{\delta} \varphi(t), \\
 \dot{\rho} &= -c_p \rho,
 \end{aligned} \quad (6)$$

where the following notations are accepted:

$$\begin{aligned}
 c_f &= \frac{1}{T_f}, \quad c_p = \frac{1}{T_p}, \quad c_\delta = \frac{1}{\delta T_p}, \\
 0 &\leq b_1^{\min} \leq b_1(\xi) \leq b_1^{\max}, \\
 0 &\leq b_2^{\min} \leq b_2(\rho) \leq b_2^{\max},
 \end{aligned} \quad (7)$$

here $(b_1^{\min}, b_1^{\max}) = (-\xi_\xi, \xi_\xi)$, $(b_2^{\min}, b_2^{\max}) = (-\rho_\xi, \rho_\xi)$, $b_1(\xi)$ and $b_2(\rho)$ are unknown but bounded functions, and function $\varphi(t)$ satisfies inequality $\varphi(t) \leq c_i$.

For an arbitrary continuous function $V(\xi, \rho)$, according to [21], one can find the function differentiable in $V(\xi, \rho)$, the derivative of which on trajectories (6) is estimated as:

$$\dot{V} \leq h_1(\xi, \rho, \mu_1, \mu_2, \gamma) \frac{\partial V}{\partial \xi} + h_2(\xi, \rho) \frac{\partial V}{\partial \rho} \quad (9)$$

$$\begin{aligned}
 \text{where} \quad h_1(\xi, \rho, \mu_1, \mu_2, \gamma) &= -\mu_1 u_1(\xi) + \mu_2 u_2(\rho) + \gamma, \\
 h_2(\xi, \rho) &= -c_p \rho,
 \end{aligned} \quad (10)$$

$$\begin{aligned}
 \mu_1 &= \frac{1}{2} b_1^{\min} \left(1 - \text{sign} \left(\frac{\partial V}{\partial \rho} u_1(\xi) \right) \right) + \frac{1}{2} b_1^{\max} \left(1 + \text{sign} \left(\frac{\partial V}{\partial \rho} u_1(\xi) \right) \right) \\
 \mu_2 &= \frac{1}{2} b_2^{\min} \left(1 - \text{sign} \left(\frac{\partial V}{\partial \rho} u_2(\rho) \right) \right) + \frac{1}{2} b_2^{\max} \left(1 + \text{sign} \left(\frac{\partial V}{\partial \rho} u_2(\rho) \right) \right) \\
 \gamma &= c_i \text{sign} \left(\frac{\partial V}{\partial \rho} \right).
 \end{aligned}$$

Lyapunov's function $V(\xi, \rho)$ is sought in the form of a continuous solution to the partial differential equation [21]:

$$h_1(\xi, \rho, \mu_1, \mu_2, \gamma) \frac{\partial V}{\partial \xi} + h_2(\xi, \rho) \frac{\partial V}{\partial \rho} = -kV^{1/2}, \quad (11)$$

where $k > 0$ is a positive parameter.

Parameters γ , μ_1 and μ_2 may take only two values $\gamma \in \{c_1, c_2\}$, and $\mu_{1,2} \in \{b_1^{min}, b_2^{min}; b_1^{max}, b_2^{max}\}$, therefore can be taken as constant. The solution to equation (11) is obtained using the method of characteristics [30]. According to [8], equation (11), with (10), takes the following form:

$$\begin{aligned} -c_p \rho \frac{\partial V}{\partial \rho} - \text{sign}[\rho] \tilde{\gamma} \frac{\partial V}{\partial \xi} &= -kV^{1/2}. \\ \tilde{\gamma} &= c_f \mu_2 u_2 - c_p \mu_1 u_1 - \gamma. \end{aligned} \quad (12)$$

Assuming that $\tilde{\gamma} = \text{const}$, we determine the solution to equation (12). The ODE characteristic corresponding to system (12) has the following form:

$$\frac{d\rho}{-c_p \rho} = \frac{d\xi}{-\text{sign}[\rho] \tilde{\gamma}} = \frac{dV}{k\sqrt{V}} \quad (13)$$

The first integrals of equation (13) are obtained as:

$$\varphi_1(\rho, \xi) = \frac{1}{c_p} |\ln \rho| + \frac{\xi}{\tilde{\gamma}}, \quad \varphi_2(V, \xi) = \frac{\xi \text{sign}[\rho]}{\tilde{\gamma}} + \frac{2\sqrt{V}}{k}, \quad (14)$$

Function Φ is selected in the following form:

$$\Phi(\varphi_1, \varphi_2) = k_0 \sqrt{\varphi_1} + \varphi_2 = 0, \quad (15)$$

where k_0 is the real parameter. Substituting relations (14) into (15), we obtain:

$$\frac{2\sqrt{V}}{k} = k_0 \sqrt{\frac{|\ln \rho|}{c_p} + \frac{\xi}{\tilde{\gamma}}} - \frac{\xi \text{sign}[\rho]}{\tilde{\gamma}}. \quad (16)$$

The right side of expression (16) holds for all $|\rho| > 0$ when $\tilde{\gamma} > 0$ or when $c_f b_2^{max} > c_p b_1^{max} + c_\delta$. Since the left side of (16) is always non-negative, we obtain:

$$k_0 > \frac{\xi \text{sign}[\rho] \sqrt{\tilde{\gamma} c_p}}{\sqrt{\tilde{\gamma} |\ln \rho| + c_p \xi}}. \quad (17)$$

Thus, we determine function $V(\rho, \xi)$ from relation (16) in the following form:

$$V(\rho, \xi) = \frac{k^2}{4} \left(k_0 \sqrt{\frac{|\ln \rho|}{c_p} + \frac{\xi}{\tilde{\gamma}}} - \frac{\xi \text{sign}[\rho]}{\tilde{\gamma}} \right)^2. \quad (18)$$

By choosing parameters $V(\rho, \xi)$, we eliminate discontinuities in function (18). Consider the partial limits of function $V(\rho, \xi)$ (18),

as $\rho \rightarrow 0$

$$V(\rho, \xi) \rightarrow \frac{k^2}{4} \left(\frac{k_0}{\sqrt{\tilde{\gamma}}} - \frac{\text{sign}[\rho] \xi}{\tilde{\gamma}} \right)^2 \xi^2, \quad (19)$$

as $\xi \rightarrow 0$

$$V(\rho, \xi) \rightarrow \frac{k^2 k_0^2}{4} |\ln \rho|. \quad (20)$$

To eliminate discontinuities in lines $\rho = 0$ and $\xi = 0$, it is necessary to resolve the following system of equations:

$$k^2 \left(\frac{k_0}{\sqrt{\tilde{\gamma}}} - \frac{\text{sign}[\rho^\xi]}{\tilde{\gamma}} \right)^2 = \bar{k}^2, \quad (21)$$

$$k^2 k_0^2 = 1.$$

For positive values of $\bar{k}$, we obtain:

$$k = \frac{|\sqrt{\tilde{\gamma}} - \bar{k}|}{\tilde{\gamma}} > 0, \quad k_0 = \frac{\tilde{\gamma} \text{sign}(\rho^\xi)}{|\sqrt{\tilde{\gamma}} - \bar{k}|}. \quad (22)$$

Thus, function $V(\rho, \xi)$ (18) must satisfy the following conditions:

$$V(0, \xi) = \frac{k^2}{4} \xi^2, \quad V(\rho, 0) = \frac{\ln|\rho|}{4}. \quad (23)$$

With relations (17) and (22), we formulate the following lemma.

Lemma. If $c_f b_2^{\max} > c_p b_1^{\max} + c_\delta$, $c_p b_1^{\max} > c_\delta$ and parameter $\bar{k}$ in relation (22) are chosen such that the following inequalities are satisfied

$$\frac{1}{\sqrt{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta}} < \bar{k} < \frac{1}{\sqrt{(c_f b_2^{\max} - c_p b_1^{\max} - c_\delta)}} \quad (24)$$

then $\bar{k} > 0$ and condition (17) are satisfied $\rho^\xi \neq 0$.

Proof.

1. For $\rho^\xi > 0$, we check the validity of the following inequality:

$$\frac{1}{\bar{k} - \sqrt{\tilde{\gamma}_+}} > \frac{1}{\sqrt{1 + \tilde{\gamma}_+ \frac{\ln|\rho|}{c_p \xi}}},$$

considering that

$$\tilde{\gamma}_+ = c_f b_2^{\max} + c_p b_1^{\max} + c_\delta \text{sign} \left[\rho \frac{\partial V}{\partial \xi} \right],$$

which follows from the following inequality

$$\bar{k} < \frac{1}{\sqrt{c_f b_2^{\max} - c_p b_1^{\max} - c_\delta}} \geq \frac{1}{\sqrt{\tilde{\gamma}_+}} \quad (25)$$

2. For $\rho^\xi < 0$, it is necessary to check that

$$\frac{1}{\sqrt{\tilde{\gamma}_-} - \bar{k}} > \frac{1}{\sqrt{1 + \tilde{\gamma}_- \frac{\ln|\rho|}{c_p \xi}}}, \quad \text{assuming that } \tilde{\gamma}_- = c_f b_2^{\max} - c_p b_1^{\max} - c_\delta \text{sign} \left[\rho \frac{\partial V}{\partial \xi} \right], \text{ which follows}$$

$$\text{from } 0 < \bar{k} < \frac{1}{\sqrt{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta}} \leq \frac{1}{\sqrt{\tilde{\gamma}_-}}. \quad (26)$$

3. For $\sqrt{\tilde{\gamma}_+} - \bar{k} > 0$ and $\sqrt{\tilde{\gamma}_-} - \bar{k} < 0$, we have $\text{sign}[\rho^\xi] = \text{sign}[\sqrt{\tilde{\gamma}_-} - \bar{k}]$ and $k_0 > 0$ for all $\rho^\xi \neq 0$

$$V(\rho, \xi) = \begin{cases} \frac{k^2}{4} \left(k_0 \sqrt{\frac{\ln|\rho|}{c_p} + \frac{\xi}{\tilde{\gamma}}} - \frac{\xi \text{sign}[\rho]}{\tilde{\gamma}} \right)^2 & \text{if } \rho\xi \neq 0 \\ \frac{k^2 \xi^2}{4} & \text{if } \rho = 0, \\ \frac{\ln|\rho|}{4} & \text{if } \xi = 0, \end{cases} \quad (27)$$

considering k, k_0 in (22) and $\bar{k}$, satisfying inequality (24).

Theorem. If $c_f b_2^{\max} > c_p b_1^{\max} + c_\delta$, then function $V(\rho, \xi)$ (18) considering

$$\tilde{\gamma} = c_f b_2^{\max} - (c_p b_1^{\max} + c_\delta) \text{sign}[\rho\xi] \quad (28)$$

has the following properties:

a) absolutely continuous in space R^2 and differentiable for $\rho\xi \neq 0$;

b) the time derivative of function $V(\rho, \xi)$ on trajectory (6) satisfies

$$\frac{dV}{dt} = -k_{\min} \frac{c_f b_2^{\max} - c_p b_1^{\max} - c_\delta}{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta} \sqrt{V}, \quad (29)$$

where $k_{\min} = \min(k)$.

b) corresponding guaranteed time of reaching

$$t_{\text{reach}} \leq \frac{2(c_f b_2^{\max} + c_p b_1^{\max} + c_\delta)}{k_{\min}(c_f b_2^{\max} - c_p b_1^{\max} - c_\delta)} \sqrt{V(\rho(0), \xi(0))} \quad (30)$$

Proof. a) Function $V(\rho, \xi)$ can have breaks only on lines $\rho = 0$ and $\xi = 0$. Since parameters k and k_0 (22) are chosen in such a way that breaks should be removed (23), function $V(\rho, \xi)$ is continuous;

c) Partial derivatives of function $V(\rho, \xi)$ are given in the following form:

$$\frac{\partial V}{\partial \rho} = \frac{k k_0 \sqrt{V}}{3 c_p \rho} \sqrt{\left(\frac{\ln \rho}{c_p} + \frac{\xi}{\tilde{\gamma}} \right)^3}, \quad (31)$$

$$\frac{\partial V}{\partial \xi} = k \sqrt{V} \left[\frac{k_0}{3 \tilde{\gamma}} \sqrt{\left(\frac{\ln \rho}{c_p} + \frac{\xi}{\tilde{\gamma}} \right)^3} - \frac{\text{sign}[\rho]}{\tilde{\gamma}} \right] \quad (32)$$

d) The time derivative of function $V(\rho, \xi)$ on the trajectory of system (6) has the following form:

$$\frac{dV}{dt} = k \sqrt{V} \left[- \left(1 - \frac{\bar{\gamma}}{\tilde{\gamma}} \right) \sqrt{\left(\frac{\ln \rho}{c_p} + \frac{\xi}{\tilde{\gamma}} \right)^3} - \frac{\bar{\gamma}}{\tilde{\gamma}} \text{sign}[\rho] \right], \quad (33)$$

$$\text{where } \bar{\gamma} = c_f b_2^{\max} - c_p b_1^{\max} \text{sign}[\rho\xi] - \varphi(t) \text{sign}[\rho\xi], \tilde{\gamma} = c_f b_2^{\max} - c_p b_1^{\max} \text{sign}[\rho\xi] - c_\delta \text{sign}[\rho\xi] \quad (34)$$

From relations (34), for $\rho\xi > 0$, the following inequality holds $1 \geq \frac{\bar{\gamma}}{\tilde{\gamma}}$, and for $\rho\xi < 0$,

$1 \geq \frac{\bar{\gamma}}{\tilde{\gamma}} \geq \frac{c_f b_2^{\max} - c_p b_1^{\max} - c_\delta}{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta}$, then the following inequality holds

$$- \left(\bar{\gamma} - \frac{\bar{\gamma}}{\gamma} \right) \sqrt{\left(\frac{\ln \rho}{c_p} + \frac{\xi}{\tilde{\gamma}} \right)^3} - \frac{\bar{\gamma}}{\gamma} \text{sign}[\rho] \leq - \frac{c_f b_2^{\max} - c_p b_1^{\max} - c_\delta}{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta}. \quad (35)$$

From here we have

$$\frac{dV}{dt} \leq -k_{\min} \frac{c_f b_2^{\max} - c_p b_1^{\max} - c_\delta}{c_f b_2^{\max} + c_p b_1^{\max} + c_\delta} \sqrt{V} \text{ for } \rho \xi \neq 0. \quad (36)$$

Thus, inequality (36) is satisfied along the trajectory of motion of system (6). The authors of [31] considered a pneumo-hydraulic system with an ideal relay, and obtained inequality (36) in the following form:

$$\frac{dV}{dt} \leq -k_{\min} \frac{c_f - c_p - c_\delta}{c_f + c_p + c_\delta} \sqrt{V} \text{ for } \rho \xi \neq 0, \quad (37)$$

and the corresponding guaranteed time to reach

$$t_{\text{reach}} \leq \frac{2(c_f + c_p + c_\delta)}{k_{\min}(c_f - c_p - c_\delta)} \sqrt{V(\rho(0), \xi(0))} \quad (38)$$

4. Results and discussions.

The first column of Table 1 presents Lyapunov functions with lines of levels of the Lyapunov function for a two-stage pneumo-hydraulic system with ideal relay control, the second column presents a relay with a dead zone for various values of parameters $c_f, c_p, c_\delta, b_1, b_2$. As seen in Table 1, the presence of a dead zone expands the stability area.

Table 1.

Lyapunov function and their level lines for different values of parameters c_f, c_p, c_δ

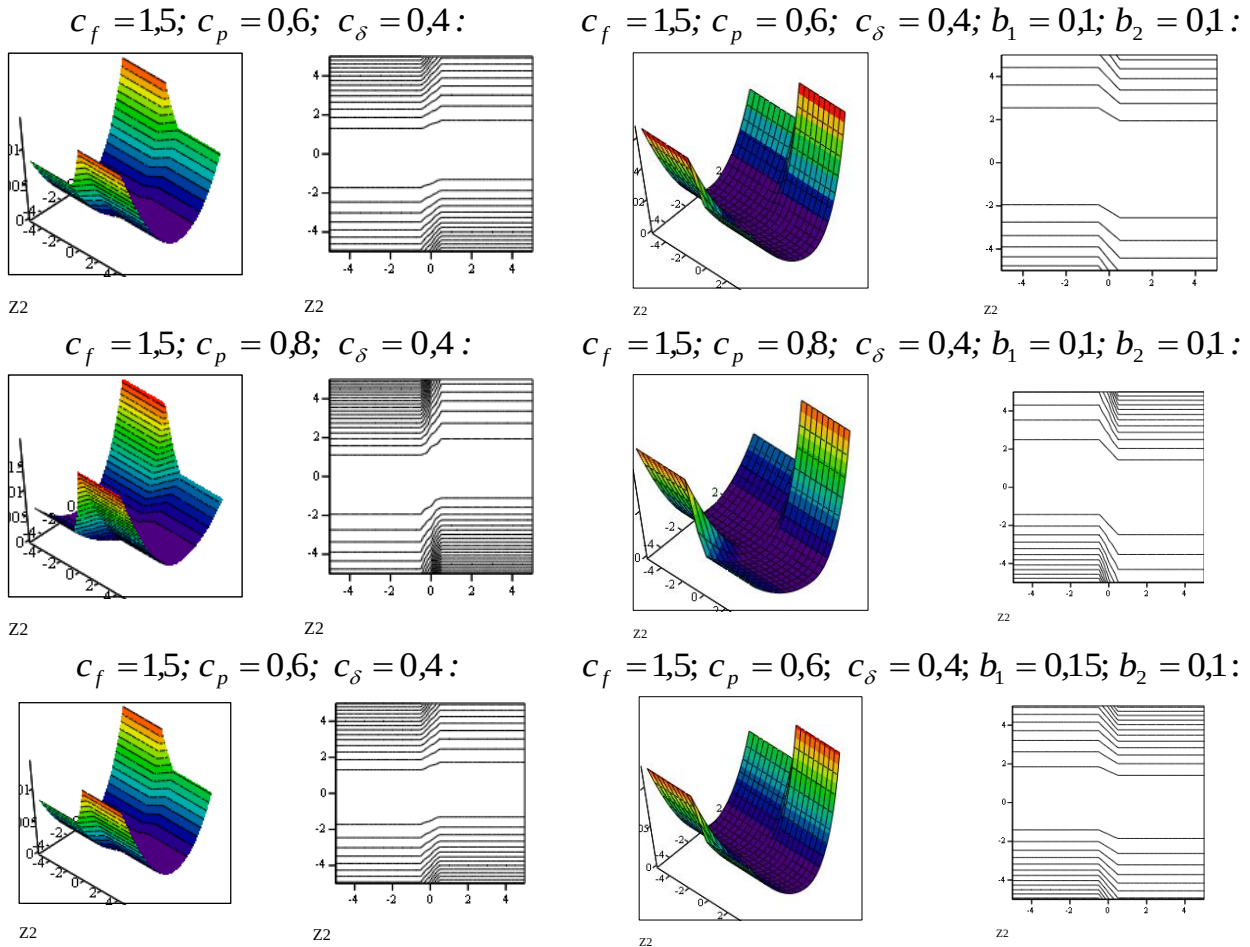
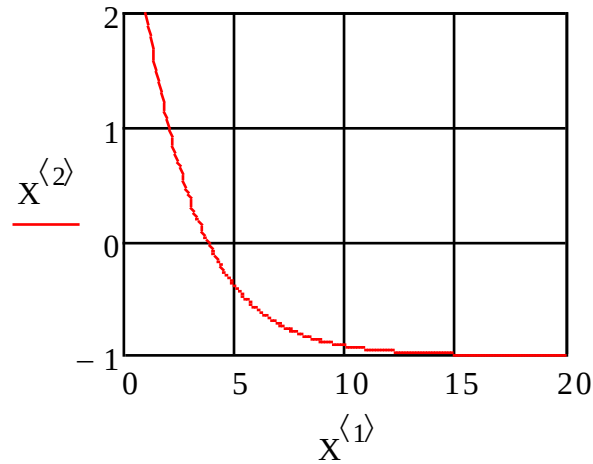


Figure 2 shows the trajectory of system (1) – (2), obtained by numerical modeling of the control process for the initial state (2, 0,3) and parameters $c_f = 1.5, c_p = 0.4, c_\delta = 0.4, k = 0.316$. According

to [31], the estimate of time the system reaches the origin of coordinates $t_r \leq 12.5$, obtained using the Lyapunov function by formula (27) and numerical simulation is $t_r = 10.19$.



$$c_f = 1.5, c_p = 0.4, \xi(0) = 2, \rho(0) = 0.3, X^{(1)} \rightarrow t, X^{(2)} \rightarrow \xi.$$

Figure 2.– Estimation of the time the system reaches the origin of coordinates or the horizontal axis of the control system

Function $V(\rho, \xi)$ is positive definite, and the time derivative is negative if condition (28) is satisfied, i.e.

$$c_f \xi_\xi > c_p \rho_\xi + c_\delta \quad (39)$$

where

$$T_f = \frac{T_s}{\sigma}, c_f = \frac{1}{T_f}, c_p = \frac{1}{T_p}, c_\delta = \frac{1}{T_\delta \delta}, b_1^{max} = \xi_\xi, b_2^{max} = \rho_\xi \quad (40)$$

Relationship (39) is a condition for the stability of system (1) according to Lyapunov's theorem (stability theorem). Following notation (40) through the initial parameters of the system, the stability condition (28) is written in the following form:

$$\frac{\xi_\xi}{T_p} < \sigma \frac{\rho_\xi}{T_s} - \frac{1}{\delta T_\delta}, \quad (41)$$

Figures 3 a) and b) show graphical representations of the stability areas of systems with an ideal relay and a relay with a dead zone.

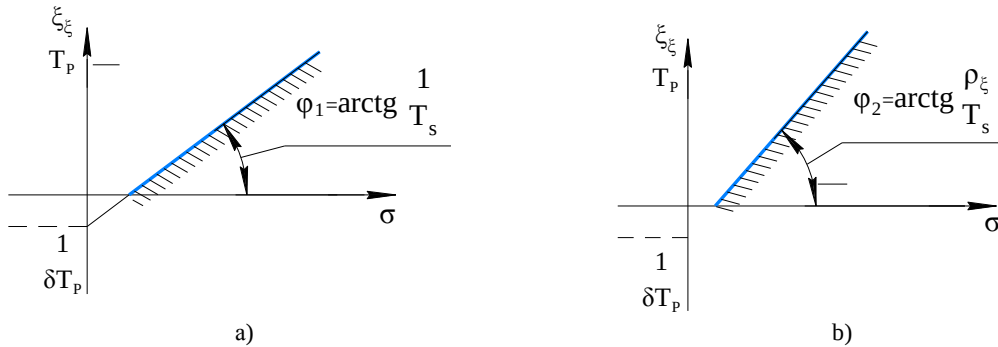


Figure 3. – Graphic representations of stability areas of systems with an ideal relay and with a dead zone

Angles φ_2 and φ_1 according to (39) are $\varphi_1 = \arctg \frac{1}{T_s}, \varphi_2 = \arctg \frac{\rho_\xi}{T_s}$.

As seen in Figs. 3 a) and b), the stability region of a system with a dead zone is wider than that of a system with an ideal relay.

Note that the external disturbance, as seen in Fig. 3, helps to reduce the stability of the system, and an increase in the feedback coefficient improves stability.

5. Conclusions.

Based on Zubov's method, a technique was developed for constructing the Lyapunov function, which ensures the finite convergence of the dynamics (trajectories of movement) of second-order pneumo-hydraulic relay systems. A discontinuous Lyapunov function satisfying the Lyapunov theorem and an estimate of the time to reach the regulator with the dead zone of the pneumo-hydraulic system were obtained. Diagrams of Lyapunov functions and their level lines were constructed for various parameters of the pneumo-hydraulic drive, and the region of stability of the system was determined. It was stated that: a) the stability region of a system with a relay dead zone is wider than that of a system with an ideal relay; b) external disturbance helps to reduce the stability of the system, and an increase in the feedback coefficient improves stability. The time to reach the sliding surface of the control system was studied by numerical simulation and using the Lyapunov function. The obtained relationship between the parameters showed that the stability of the system improves when a pulse is set in the first relay by increasing the time of the servomotor, i.e. reducing the operation speed of the tracking mechanism, which confirms the results obtained by V.V. Petrov on the influence of servomotor time on the suppression of auto-oscillations of two-stage servo mechanisms.

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STUDY OF THE DEVICE FOR CONVEYING MATERIALS INTO THE PROCESSING AREA

Bahadirov G.A.¹, Musirov M.U.², Akhmedov Kh.A.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev, Tashkent, Uzbekistan*

²*Tashkent State Transport University, Tashkent, Uzbekistan.*

Email: musirov.mech.1992@mail.ru

Abstract. String conveying devices are widely used in light industry and agriculture to transport various products to the mechanical processing zone. The correct selection of geometric and kinematic parameters is necessary for the long-term effective operation of conveying devices. The oscillatory motion of a string conveying device located at a certain angle to the horizontal plane was studied in the article on the Euler coordinate system. The equation of motion of the conveying device is derived using D'Alembert's principle. The solution to the parametric equation of motion of the conveying device is expressed by the parametric Mathieu equation using the Galerkin method for small oscillations, expanding in terms of the Taylor vector. Graphical solutions to a parametric equation are expressed using the Maple graphics software.

Keywords: string conveyor; flat material; variable mass; transverse vibrations.

1. Introduction.

A device for levelling flat materials and transferring them to the processing zone was developed [1].

The influence of the technological processing of flat materials on the quality of semi-finished leather products was determined. It is necessary to analyze the condition of the processed material after each technological process. Experimental research in the leather industry aims at solving complex multifaceted problems and determining rational methods of technological processes for treating leather raw materials. It was established that semi-finished leather products' physical and mechanical properties vary depending on the moisture content [2-7].

It is known that the processed product is flat and smooth. The feed into the working area is one of the important operations in various fields. As a result, significant savings in raw materials can be achieved by solving the problem of eliminating shortages and losses [8, 9].

Strain properties of semi-finished leather processed between pressure rollers coated with water-absorbing elastic materials were studied. The influence of the feed rate and the pressure of the compression rollers on the deformation in the topographic section of the semi-finished leather product (shoulder, belly, and butt) was determined [10-12].

Four analytical methods were used to analyze test results to determine string conveyor parameters and elastic modulus. They are the Euler-Bernoulli and finite element methods for determining the bending modulus of unstretched strings, and the Timoshenko and Mindlin-Reisner methods [13].

The slippage of conveyor strings depends on the diameter of the driven roller and the vertical tension at the point of contact; the vibration resistance of the string is very low [14].

In [15], the conveyor gluing process based on the Mooney-Rivlin law was studied.

Conveyor strings play an important role in dynamic behavior. Transverse vibrations of strings were calculated by the Kirchhoff method using differential equations of motion of the system [16, 17].

It was determined that speed control in belt conveyors leads to the high energy efficiency and stable operation of conveyor equipment parts [18].

A virtual energy storage model was proposed to reduce the energy consumption of belt conveyors. This model was studied to reduce energy consumption over time by controlling the speed of conveyor belts, the speed of product feed, and the speed of other parts of equipment [19].

Mechanical damage may occur after a certain period of conveyor operation. This process was studied and a methodology for estimating the probability of damage during operations was described using available statistical models [20].

In [21–26], the authors developed designs and studied the parameters of roller technological machines. The rational parameters of working bodies and machine drives were theoretically determined. The technological parameters of the mechanical processing of fiber materials (in the example of raw hides) were experimentally defined and scientifically substantiated.

2. Materials, methods, and objects of research.

The movement of a string conveyor, which increases the productivity of a technological machine and feeds semi-finished products to the machining zone is examined in this study.

The oscillatory motion of a string conveyor located at a certain angle to the horizontal plane was studied in the article on the Euler coordinate system. The equation of motion of a string conveyor was obtained using D'Alembert's principle. The solution to the equation of motion is expressed by the parametric Mathieu equation using the Galerkin method for small oscillations, expanding in Taylor vector.

3. Statement of the problem.

In this problem, the semi-finished product moves on a string conveying device to the machining zone between rotating working rollers. The conveyor is located at a certain angle to the horizontal axis to make it convenient to throw semi-finished products onto the strings. As a result, the weight component of the semi-finished product will resist the movement of the strings. Below we will consider the issue of the stability of this process (Fig. 1).

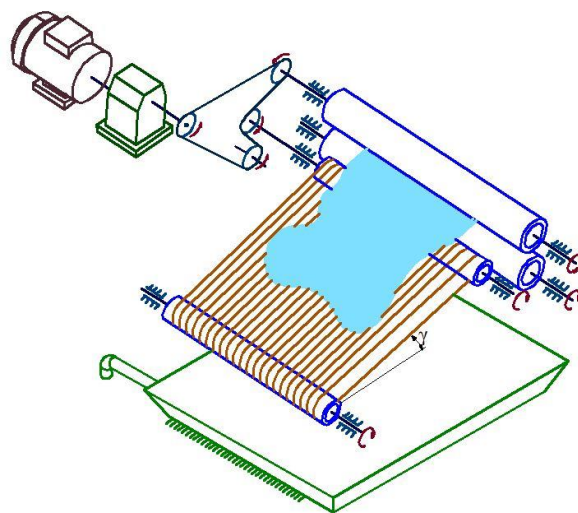


Figure 1. – Schematic diagram of a machine for processing semi-finished products

Consider the problem of transverse vibrations of strings under nonuniform gravity forces.

The strings stretched over the grooves of the rollers are subject to nonuniformly distributed vertical gravity forces of flat material.

Figure 2 shows a side view of a string conveyor feeding semi-finished products into the processing area between rotating working elements.

Let us consider small oscillations of the system.

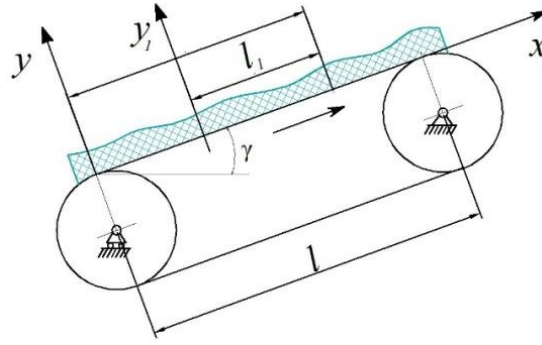


Figure 2. – Pattern of mass distribution of flat material

In the device considered, the strings move at a constant speed.

When solving this problem, it is appropriate to consider it on the Euler coordinate system, so the acceleration equation can be written in the following form:

$$\frac{d^2 y}{dt^2} = \frac{\partial^2 y}{\partial t^2} + 2\omega R \frac{\partial^2 y}{\partial x \partial t} + \omega^2 R^2 \frac{\partial^2 y}{\partial x^2}.$$

Using d'Alembert's principle, we write the following [27, 28]:

$$m \left(\frac{\partial^2 y}{\partial t^2} + 2\omega R \frac{\partial^2 y}{\partial x \partial t} + \omega^2 R^2 \frac{\partial^2 y}{\partial x^2} \right) = E \frac{\partial^2 y}{\partial x^2} - P \sin \gamma,$$

here m is the mass corresponding to a unit of length.

The pattern of distribution of the flat material mass along the strings of the conveyor is written in the following form:

$$m = m_2 + m_0 + m_1 \sin \left(\frac{2\pi}{l_1} (x - \omega R t) \right),$$

where, m_0 is the average value of the mass of the conveyor product per unit length of conveyor strings (kg/cm); m_1 is the variable amplitude of the function of flat material mass (kg/cm); is the mass of conveyor strings per unit length (kg/cm); m_2 is the step of change in the variable mass of the semi-finished product; E is the modulus of elasticity of the working strings of the conveyor.

It is difficult to find a general analytical solution to this problem, so, if we consider the deviation of m_1 (i.e. nonuniformly distributed mass) from m_0 to be small, and the linear part around point $m_1=0$ expanded into a Taylor series, then the equation of transverse motion will have the following form [29]:

$$\frac{\partial^2 y}{\partial t^2} + 2\omega R \frac{\partial^2 y}{\partial x \partial t} + \omega^2 R^2 \frac{\partial^2 y}{\partial x^2} = \frac{1}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \sin \left(\frac{2\pi}{l_1} (x - \omega R t) \right) \right) \left(E \frac{\partial^2 y}{\partial x^2} - P \sin \alpha \right) \quad (1)$$

The initial and boundary conditions for the resulting differential equation are:

$$\begin{cases} y(0, t) = 0 \\ y(l, t) = 0 \end{cases}$$

Consider the issue of the priority of small fluctuations using the Galerkin method.

$$y = f(t) \sin \frac{\pi x}{l} \quad (2)$$

It is easy to see that the approximation of a function of this type satisfies the boundary conditions. Substituting relation (2) into the equation of motion (1), we derive a second-order differential equation for function $f(t)$ [30-33]:

$$\begin{aligned} & \ddot{f} \sin\left(\frac{\pi x}{l}\right) + \frac{2v\pi}{l} \dot{f} \cos\left(\frac{\pi x}{l}\right) + \left(\frac{\pi}{l}\right)^2 \left(\frac{E}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \sin\left(\frac{2\pi}{l_1}(x - \omega Rt)\right) \right) - \omega^2 R^2 \right) \cdot \\ & \cdot f \sin\left(\frac{\pi x}{l}\right) + \frac{P \sin \gamma}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \sin\left(\frac{2\pi}{l_1}(x - \omega Rt)\right) \right) = 0 \end{aligned} \quad (3)$$

to solve the differential equation, we construct the following system of differential equations:

$$\begin{cases} \frac{2v\pi}{l} \dot{f} = 0 \\ \ddot{f} + \left(\frac{\pi}{l}\right)^2 \left(\frac{E}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \sin\left(\frac{2\pi}{l_1}(x - \omega Rt)\right) \right) - \omega^2 R^2 \right) f = 0 \\ \frac{P \sin \gamma}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \sin\left(\frac{2\pi}{l_1}(x - \omega Rt)\right) \right) = 0 \end{cases} \quad (4)$$

The first equation of the system of equations (4) always satisfies equation (1). We solve the second equation of the system.

$$\ddot{f} + \left(\frac{\pi}{l}\right)^2 \left(\frac{E}{m_0 + m_2} \left(1 - \frac{m_1}{m_0 + m_2} \left(a_{11} \cos \frac{2\pi\omega Rt}{l_1} - a_{12} \sin \frac{2\pi\omega Rt}{l_1} \right) \right) - \omega^2 R^2 \right) f = 0 \quad (5)$$

Here

$$\begin{aligned} a_{11} &= \frac{l_1^2}{\pi^2 v^2} \int_0^l \sin \frac{2\pi x}{l_1} \sin^2 \frac{\pi x}{l} dx = -\frac{l_1^2}{\pi^2 v^2} \frac{l_1}{4\pi} \frac{l_1^2}{l^2 - l_1^2} \left(1 - \cos \frac{2\pi l}{l_1} \right) \\ a_{12} &= \frac{l_1^2}{\pi^2 v^2} \int_0^l \cos \frac{2\pi x}{l_1} \sin^2 \frac{\pi x}{l} dx = -\frac{l_1^2}{\pi^2 v^2} \frac{l_1^2}{4\pi(l - l_1)} \sin \frac{2\pi l}{l_1} \end{aligned}$$

Let us write the resulting equation.

$$\ddot{f} + \left(a - b \sin\left(\frac{2\pi\omega Rt}{l_1} + \beta\right) \right) f = 0 \quad (6)$$

here

$$a = \left(\frac{\pi}{l}\right)^2 \left(\frac{T}{m_0 + m_2} - \omega^2 R^2 \right), b = \frac{Tm_1}{(m_0 + m_2)^2} \left(\frac{\pi}{l}\right)^2 \sqrt{a_{11}^2 + a_{12}^2}, \beta = \arctg\left(\frac{a_{11}}{a_{12}}\right).$$

Let us introduce the following substitution:

$$\frac{2\pi\omega Rt}{l_1} = 2\tau - \frac{\pi}{2} - \beta$$

After this substitution, we obtain the Mathieu equation.

$$\ddot{f} + (a + 2q \cos 2\tau) f = 0 \quad (7)$$

alternatively

$$2q = b.$$

The general solution to differential equation (7) has the following form:

$$f(t) = C_1 \text{MathieuC}(a, -q, \tau(t)) + C_2 \text{MathieuS}(a, -q, \tau(t)),$$

where MathieuC is the cosine of Mathieu, MathieuS is the sine of Mathieu, C_1 and C_2 are the Mathieu differentiation constants defined as:

$$C_1 = \frac{\text{MathieuC}(\text{MathieuA}(n, q), q, x)}{\text{MathieuCE}(n, q, x)}, C_2 = \frac{\text{MathieuS}(\text{MathieuB}(n, q), q, x)}{\text{MathieuSE}(n, q, x)}.$$

We are looking for a solution to the third equation of the system of equations (4):

$\sin \gamma \neq 0$, expression (8) is reasonable since $1 - \frac{m_1}{m_0 + m_2} \sin\left(\frac{2\pi}{l_1}(x - \omega R t)\right) = 0$.

For equation (4) to have a solution based on expression (7), the following condition must be met:

$$0 \leq \frac{m_1}{m_0 + m_2} \leq 1. \quad (9)$$

With expression (8), the second equation of the system of equations (4) can be written as follows:

$$\ddot{f} + \left(\frac{\pi}{l}\right)^2 \omega^2 R^2 f = 0, \quad (10)$$

equation (10), satisfying conditions $f(0) = 0$ and $\dot{f}(0) = \omega R$ is:

$$f = \frac{l}{\pi} \sin \frac{\pi}{l} \omega R t. \quad (11)$$

Considering that the trigonometric expression (11) is limited, to determine the solution to equation (1), it is enough to determine the solution to the second equation of the system of equations (4).

4. Results and discussions.

Problems of this type are included in the Mathieu equation in various ways. The resulting Mathieu equation (7) is appropriate for solving, determining, and selecting rational parameters of the working parts of the conveying device of a technological machine.

By solving differential equation (7) with the given parameters $\lambda_1 = 0,3 \text{ m}$, $\lambda = 1 \text{ m}$, $E = 300 \text{ N/cm}^2$, $\omega R = 0,2 \text{ m/s}$, $m_0 = 15 \text{ kg/cm}$, $m_1 = 2 \text{ kg/cm}$, $m_2 = 3 \text{ kg/cm}$, we obtain a graphical solution (Figure 3).

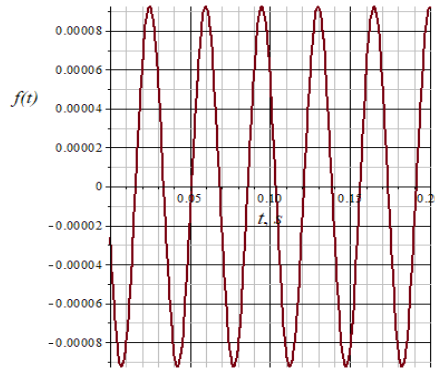


Figure 3. – Graphical solution to the Mathieu equation (6) depending on time

Let us plot a graph of expression (11), which is a partial solution to differential equation (3) (Figure 4).

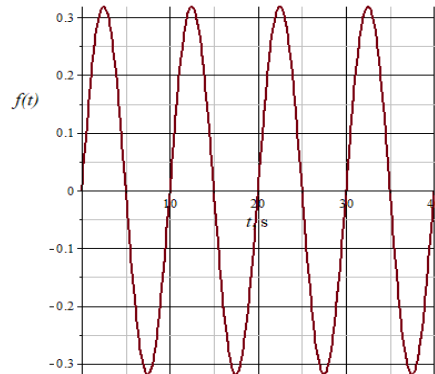


Figure 4. – Graphical solution to the equation of oscillatory motion of a string conveyor

With the parameters selected for the experimental device, the resulting graphical solution (Figure 4) of Mathieu's equation (7) shows that the amplitude of vibrations of the conveyor strings does not change. If the amplitude does not change, the motion is stable. This ensures long-term and efficient operation of the working parts of the technological machine [2].

5. Conclusions.

The equations of motion of a conveyor transporting flat materials of variable mass to the machining zone are reduced to parametric oscillations, the amplitude of oscillations is limited, and the increase in amplitude is determined depending on the values of constants related to the system. An increase in vibration amplitude over time leads to a state of parametric resonance, and this process is considered hazardous for the mechanisms of the conveying device. The movement of conveyor strings stretched on cylindrical rollers is determined by the Mathieu equation. The parameters of the Mathieu equation depend on the speed of the conveyor strings and the geometric dimensions of the system. Kinematic and geometric parameters should be selected so that they fall within the stability region in order to limit the amplitude of the vibration movement of the conveyor strings. These parameters depend on the geometric dimensions and speed of the conveying device of the technological machine.

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EXPERIMENTAL STUDY OF THE PROCESS OF COMPRESSION OF SEMI-FINISHED PRODUCTS THROUGH SHAFT PAIRS OF DIFFERENT DIAMETERS

Bahadirov G.A.¹, Umarov A.A.¹, Rakhimova Z.A.¹, Achilov G.Q.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev,
Uzbekistan Academy of Sciences
E-mail: umarov_aa@bk.ru*

Abstract. This paper presents experimental studies on a roller machine with different diameters of compression rollers proposed for use in the leather industry. In experimental studies, taking into account five input factors, i.e., rotation speed of the shafts, compression pressure force, angles of delivery of processing materials to the compression zone, different diameters of the shafts, and the thickness of the leather raw material, a regression equation was derived in which the output factor depends on the residual moisture. The solutions to the regression equation were obtained and analyzed graphically.

Keywords: shaft rotation speeds, pressing process, pressing pressure force, diameter of pressing shafts, thickness of leather raw material.

1. Introduction.

In the world, leather raw materials prepared by processing different types of animal skins are of great importance in the production of shoes, clothes, and other leather goods, as well as in the decoration of furniture and car salons. In this regard, in the production of quality leather products, it is important to use high-efficiency methods of processing leather raw materials and shaft technological machines. Today, extensive measures are being taken to reduce labor and energy

consumption in the production of leather raw materials, and to develop highly effective technical tools and methods that increase the efficiency of technological processes in processing. In the implementation of these tasks, it is important to create technically and technologically modernized machines that meet technological requirements, along with high-quality processing of leather raw materials for leather products.

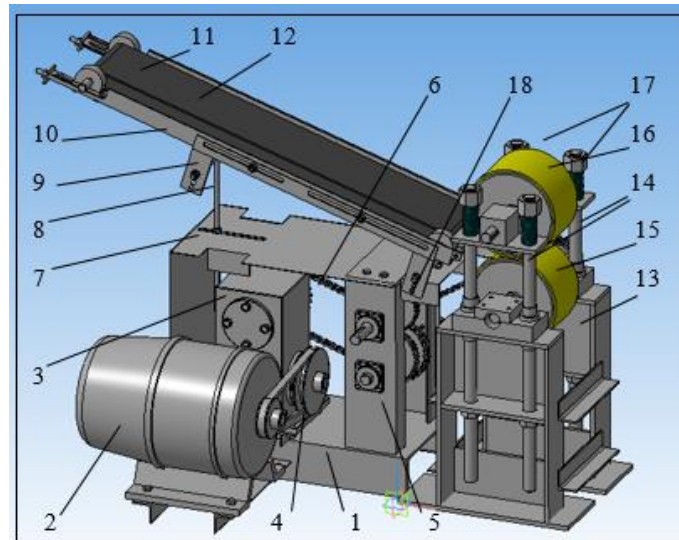
Material drying in tanneries is one of the most energy-intensive processes and the most important step used in wet processing. Optimization of drying is very promising to reduce energy consumption, improve product quality, and improve drying process parameters at the same time. In this study, the sorption properties of cattle skin moisture were determined using the static gravimetric method at temperatures of 30⁰, 40⁰, 50⁰, 60⁰, and 70⁰ C. Both the enthalpy of sorption and the entropy of desorption increase to a maximum with increasing humidity and then decrease sharply with increasing humidity. The adsorption enthalpy decreases with increasing humidity. However, as the moisture content increases by a maximum of 6.29 J/K.mol, the adsorption entropy increases smoothly. Spreading pressure increases with increased water activity. The net integral enthalpy seemed to decrease and then increase to become asymptotic. With increasing humidity, the net integral entropy decreases [1, 3]. Indeed, it is necessary to analyze the heat and mass transfer in the product during drying. The simultaneous transfer of heat, mass, and momentum can clearly explain the complexity of the drying phenomena. Working with porous media, the many changes in the product during this process increase the complexity of the problem. A good knowledge of the thermo physical properties of the material helps to improve their drying process, drying schedules and helps to use them effectively to achieve optimal qualities [2]. In this scientific work [5], the technology of dyeing the leather semi-finished product of chrome tanning was developed based on the study of interaction processes in the system with dermis collagen dye-titanium compounds. In this study [6], the authors identified the factors affecting productivity in leather processing in the Indian leather industry. A dynamic model was developed to study the vibration characteristics of the shaft-disk system. The appropriateness of the obtained model was confirmed by the authors based on the simulation results. The effect of boundary conditions on the frequency parameters and geometric parameters of the val-disk system has been determined [8]. A system for dynamic measurement of the vibration of rotating shafts has been developed. The developed method provides multidimensional measurement of the vibration parameters of rotating shafts. The advantage of the proposed measurement system, which provides accuracy in determining dynamic characteristics, is shown by the analysis of rotating shafts in operating modes [9, 10]. Over the past 20 years, works on the definition and analysis of shaft fracture have been extensively studied and analyzed. The authors propose a system analysis mechanism to detect shaft faults under operating conditions. It was determined that shaft breakage occurs mainly due to wear, corrosion, deformation, and erosion due to periodic loads during their operation [11]. In this work [12], the effect of a pair of toothed vibration shafts used on one and two shafts was studied. The authors obtained the equations of motion that determine the circumferential vibrations of a pair of gears and the torsional vibrations of the shafts and carried out numerical calculations. Regularly treating leather products up to 5 m² across the entire surface to high-quality consumer goods such as shoes, bags, vests, furniture, and car interiors requires a uniform thickness across the entire surface. Precise adjustment of the thickness is achieved through the process of whittle. On an industrial scale, rotary knife shafts are used to remove the semi-finished skins from the bony part, as a result of which the specified thickness is adjusted and a smooth surface is formed. In order not to lose quality, the temperature of the skin should not be allowed to rise above the norm during the extraction of the skin patch. The finite element method was used for thermal simulation of the process. Measured temperatures, blade roll geometry, and process data are taken into account to model the temperature development near the blade edge. The obtained results increase our understanding of the processes of temperature generation during the operation of the machine [4]. In order to carry out the process of whittle qualitatively, it is necessary to achieve uniformity of residual moisture over the entire surface of semi-finished skins. For this, it is necessary to research the parameters of the process before the whittle process, i.e., the compression process. The research on compression process parameters is mostly done by the experimental method. There are also a

number of scientists who conducted research through experimental experiments. For example, the following study [7] identified and analyzed four factors: processing personnel, processing machine, processing method, and processed material. Based on experimental measurements, a system for measuring the forces that appear in areas of the impact zone with the pressure shafts of the conveyor belt has been developed. The authors say that when the pressure of the pressure shafts on the conveyor belt is insufficient, the process of transporting bulk materials is not carried out. This system allows the construction of regression models and their use in the analysis of conveyor failures [7]. The article deals with the problem of determining the motion of a moisture-saturated flat material between a pair of rollers [13]. In this research work, scientific studies on the study of the experimental and mechanical properties of the process of mechanical processing of a multi-layered leather tanned product with a shaft machine were carried out and put into practice [14-18]. The kinematic analysis of the mechanism is carried out in order to determine the angular and linear speeds and accelerations of the links of the mechanism, depending on the angular speed of the driving link (gear) and the linear speed and acceleration of the center of rotation of the free working shaft. The kinematic analysis was carried out according to the centroid theory by the method of the instantaneous center of rotation of the links [19, 20, 21]. The work is devoted to the optimization of the parameters of ether pressing machines. Regression models of the residual moisture content of leather and the intensity of the pressing process are developed. Which depends on the intensity of the load. The radius and the speed of the rollers. A number of patterns were established that allow increasing the efficiency of leather pressing by rollers. An extremum problem was solved to determine the diameter of the rollers that provide the minimum deflection of their working parts [22]. The shapes of the contact curves of the rollers in the two-roll module considered in the article are described for the case when, in the process of interaction, the deformation of the layer of material occurs in the vertical direction. The equations obtained for the contact curves of the rollers are general in the sense that they are applicable for particular cases of contact interaction in two-roll modules [23, 24]. In this research, the process of vertical transmission of semi-finished products to the coverage area by pairs of shafts was studied [25]. Information about the working principle and functions of the improved design of the vertical-type shaft machine is present [26].

In this regard, special attention is being paid to the development and improvement of the method and roller technological machine that ensure high efficiency in the technological processes of processing leather raw materials, as well as to the justification of the technological process, parameters, and operating modes. The significance of the experimental work is great on the basis of theoretical research aimed at the development of new scientific and technical solutions.

2. Materials, methods, and object of study.

As a result of the conducted scientific research, an experimental stand of a machine with different diameters of compression shafts for mechanical processing of leather raw materials was developed. This device was designed, technical documents, drawings were created to provide the opportunity to manage experimental tests and obtain the necessary results by changing the diameters, coatings, rotation speeds, compressive force between the shafts, and the angles of delivery of the processed material to the working zone. The compression device created for conducting experimental work consists of three parts: the driving part (1), the part that supplies the processed material to the interval of the shaft pair, i.e., the conveyor part (2), and the part for compressing and dehumidifying the processed material with pressure, i.e., the working part (3) consists of (Figure 1).



1- base of the driving part, 2- electric motor, 3- reducer, 4- belt transmission, 5- parasitic mechanism, 6- sprocket, 7- conveyor base, 8- threaded rod, 9- lower part of the conveyor, 10- upper part of the conveyor

Figure 1. – The structure of the compression device created for conducting experimental work

Experimental work was carried out on the experimental stand of the mechanical processing machine for the proposed leather raw material. The purpose of the experiment is to determine the residual moisture after pressing the leather raw material with pairs of shafts and to justify the kinematic and geometric parameters of the shaft machine. A method of planning 25 full factorial experiments was chosen for the experiment. All identified main factors are divided into two levels (+1 and -1), and the number of experiments conducted is equal to $25=32$. After determining the main factors and their variation levels, the process of determining the residual moisture of leather raw materials was studied considering five factors, namely, $x_1(V)$ output shaft speed, $x_2(t_0)$ thickness of leather raw material, $x_3(P)$ compressive strength, $x_4(\tau)$ material delivery angle and $x_5(D)$ was studied taking into account the values of the diameter of compression shafts.

In the full factorial design of experiments, the general form of the regression equation is:

$$y = b_0 + b_1X_1 + b_2X_2 + \dots + b_nX_n + \dots + b_{12}X_1X_2 + \dots + b_{n-1}X_{n-1}X_n. \quad (1)$$

here is the amount of residual moisture released in coded form; y – b_0 – the free term of the regression equation; $b_n, b_{12} \dots b_{n-1, n}, b_{11} \dots b_{nn}$ – coefficients of the regression equation; X_n – conditional value of factors.

Table 1.

Levels and intervals of change of experimental factors

Factor			Levels of variation factor a			Interval variations	Calculation factor a
Name	Designation						
	Natural	code	+1	0	-1		
Linear speed of contact point of shafts (m/sec)	V	$x1$	0.28	0.3	0.32	0.02	$x1=(V-0.3)/0.02$
Thickness of processed material (m)	t_0	$x2$	0.001	0.003	0.005	0.002	$x2=(t_0-0.003)/0.002$
Shaft contact point voltage (N/m)	P	$x3$	32000	40000	48000	8000	$x3=(P-40000)/8000$
Upper shaft diameter (m) (lower 0.32 m)	D	$x4$	0.125	0.2225	0.32	0.0975	$x4=(D-0.125)/0.2225$

The accuracy of the experimental results largely depends on the accuracy level of all input and output parameters and their constancy. Therefore, multiple control of the input and output parameters

of the shaft compression machine helps in conducting each experiment. Using Table 1, an experiment planning matrix was constructed.

Table 2.

Experiment planning matrix

№	Matrix				Natural values Variables			
	x1	x2	x3	x4	V	t ₀	P	D
1	-1	-1	-1	-1	0.28	0.001	32000	0.125
2	1	-1	-1	-1	0.32	0.001	32000	0.125
3	-1	1	-1	-1	0.28	0.005	32000	0.125
4	1	1	-1	-1	0.32	0.005	32000	0.125
5	-1	-1	1	-1	0.28	0.001	48000	0.125
6	1	-1	1	-1	0.32	0.001	48000	0.125
7	-1	1	1	-1	0.28	0.005	48000	0.125
8	1	1	1	-1	0.32	0.005	48000	0.125
9	-1	-1	-1	1	0.28	0.001	32000	0.32
10	1	-1	-1	1	0.32	0.001	32000	0.32
11	-1	1	-1	1	0.28	0.005	32000	0.32
12	1	1	-1	1	0.32	0.005	32000	0.32
13	-1	-1	1	1	0.28	0.001	48000	0.32
14	1	-1	1	1	0.32	0.001	48000	0.32
15	-1	1	1	1	0.28	0.005	48000	0.32
16	1	1	1	1	0.32	0.005	48000	0.32

We find the dispersion reproducibility:

Table 2.

Dispersion reproducibility

№	x1	x2	x3	x4	Y ₁₁	Y ₁₂	Y ₁₃	$\bar{Y}$	S ² {Y1}
1	-1	-1	-1	-1	56.7	56.4	56.2	56	0.063
2	1	-1	-1	-1	58.3	58.8	58.6	59	0.063
3	-1	1	-1	-1	58.2	57.9	57.7	57.5	0.063
4	1	1	-1	-1	61.1	62.2	61.9	61.5	0.323
5	-1	-1	1	-1	45.7	45.4	45.2	45	0.063
6	1	-1	1	-1	47.3	47.8	47.6	48	0.063
7	-1	1	1	-1	49.7	49.4	49.2	49	0.063
8	1	1	1	-1	52.8	52.6	53.7	53	0.343
9	-1	-1	-1	1	59.3	59.8	59.6	60	0.063
10	1	-1	-1	1	64.7	64.4	64.2	64	0.063
11	-1	1	-1	1	62.8	62.6	63.7	63	0.343
12	1	1	-1	1	65.7	65.4	65.2	65	0.063
13	-1	-1	1	1	45.8	46.3	46.1	46	0.063
14	1	-1	1	1	50.7	50.4	50.2	50	0.063
15	-1	1	1	1	51.3	51.1	52.2	51.5	0.343
16	1	1	1	1	53.3	53.8	53.6	54	0.063
Σ								1765.0	4.47

The variance values of the output factor are calculated using the following formulas:

$$S_1^2\{Y\} = \frac{\sum_{i=1}^3 (y_i - \bar{Y})^2}{n-1};$$

$$S_2^2\{Y\} = \frac{\sum_{i=1}^3 (y_i - \bar{Y})^2}{n-1}; \quad S_{16}^2\{Y\} = \frac{\sum_{i=1}^3 (y_i - \bar{Y})^2}{n-1}. \quad (2)$$

The homogeneity of the regression values was checked using the Cochran test.

$$G_x = \frac{S_i^2\{Y\}_{\max}}{\sum_{i=1}^N S_i^2\{Y\}} \quad (3)$$

where G_x is the estimated value of the Cochran criterion; $S_i^2\{Y\}$ - the largest dispersion value of the test results; $\sum_{i=1}^N S_i^2\{Y\}$ - the sum of the variance values.

The calculated value of the Cochran criterion is compared with the value selected from the table. In this $G_x < G_{tabl}$ the condition must be fulfilled. The value of the Cochran criterion selected from the table at the 0.5% level is equal to:

$$G_{tabl} = \{f_1 = N = 16, f_2 = m - 1\} \quad (4)$$

So, in the case under study, dispersions are homogeneous. Because the condition has been fulfilled. $0.07 < 0.16$

In the next step, we construct the regression equation and determine the regression coefficients.

The regression equation looks like this:

$$Y = 53.1562 + 1.65x_1 + 1.65x_2 - 5.6x_3 + 1.5x_4 - 0.09x_1x_2 + 0.003x_1x_3 + \dots + 0.03x_1x_2x_3x_4 - 0.01375x_1x_2x_3. \quad (5)$$

Experience №	x1	x2	x3	x4	x1	x2	x3	x4	x1x2	...	x1x2x3x4	Y'
1	-1	-1	-1	-1	-56	-56	-56	-56	56		-56	55.90625
2	1	-1	-1	-1	59	-59	-59	-59	-59		59	59.21875
3	-1	1	-1	-1	-57.5	57.5	-57.5	-57.5	-57.5		57.5	59.21875
4	1	1	-1	-1	61.5	61.5	-61.5	-61.5	61.5		-61.5	62.53125
5	-1	-1	1	-1	-45	-45	45	-45	45		45	44.71875
6	1	-1	1	-1	48	-48	48	-48	-48		-48	48.03125
7	-1	1	1	-1	-49	49	49	-49	-49		-49	48.03125
8	1	1	1	-1	53	53	53	-53	53		53	51.34375
9	-1	-1	-1	1	-60	-60	-60	60	60		60	58.96875
10	1	-1	-1	1	64	-64	-64	64	-64		-64	62.28125
11	-1	1	-1	1	-63	63	-63	63	-63		-63	62.28125
12	1	1	-1	1	65	65	-65	65	65		65	65.59375
13	-1	-1	1	1	-46	-46	46	46	46		-46	47.78125
14	1	-1	1	1	50	-50	50	50	-50		50	51.09375
15	-1	1	1	1	-51.5	51.5	51.5	51.5	-51.5		51.5	51.09375
16	1	1	1	1	54	54	54	54	54		-54	54.40625
Σ	0	0	0	0	53.00	53.00	-179.0	49.00	-3.00		0.00	1765.00

$$b_0 = \frac{1}{N} \sum_{i=1}^N \bar{Y}_i, \quad (6)$$

here N - the number of experiments.

$$b_i = \frac{1}{N} \sum_{i=1}^N x_{ji} x_{ij} \bar{Y}_i, \quad b_{ji} = \frac{1}{N} \sum_{i=j=1}^N x_{ij} \bar{Y}_i, \quad (7)$$

here i - test procedure; j - order of factors.

In the next step, we check the significance of the regression coefficients using the Student's test:

$$t_R = \frac{(b_i)}{S(b_i)}, \quad S^2(b_i) = \frac{S_Y^2}{N \cdot n}, \quad (8)$$

here S_Y^2 - linear dispersion, it is defined by the following expression:

$$S_Y^2 = \frac{1}{N} \sum_{i=1}^N S_i^2 \{Y\} \quad (9)$$

We calculate the numerical value of the serial variance:

$$S_Y^2 = \frac{1}{N} \sum_{i=1}^N S_i^2 \{Y\} = \frac{4.47}{32} = 0.14$$

After that, we calculate the numerical value of the dispersion coefficient:

$$S^2(b_i) = \frac{S_Y^2}{N \cdot n} = \frac{0.14}{32 \cdot 3} = 0.001458(3)$$

$$S(b_i) = \sqrt{0.001458(3)} = 0.03818813079$$

We determine the estimated values of the Student's criterion [107] for the regression coefficients

$$t_{xb_0} = \frac{55.1563}{0.038} = 1446.48, \quad t_{xb_1} = \frac{1.6562}{0.038} = 43.4355, \quad t_{xb_2} = \frac{1.6562}{0.038} = 43.4355,$$

$$t_{xb_3} = \frac{-5.59375}{0.038} = -146.697, \quad t_{xb_{1234}} = \frac{0.003125}{0.038} = 0.81954.$$

Student criteria We select from the table according to the calculated value of. The following condition must be met:

$$t_x > t_{tabl}$$

If the above condition is met, the calculated regression coefficients are considered significant, but if it is not, the same regression coefficient is considered insignificant and excluded from further calculations.

Then we write the regression equation as follows:

$$Y = 55.15 + 1.65x_1 + 1.65x_2 - 5.59x_3 + 1.5x_4 - 0.09x_1x_2 + 0.09x_1x_4 + 0.65x_2x_3 - 0.72x_3x_4 - 0.34x_1x_2x_4$$

After that, (3.12) regression equation was tested for adequacy based on Fisher's test. We express the calculated value of Fisher's criterion as follows

$$F_x = \frac{S_{ad}^2}{S_Y^2}, \quad (10)$$

here $N - M > 0$ when S_{ad}^2 - adequacy difference; M is the number of significant coefficients.

$$S_{ad}^2 = \frac{1}{N - k - 1} \cdot \sum_{i=1}^N (\bar{Y}_i - Y_{Ri})^2. \quad (11)$$

We determine the numerical value of the adequacy difference. We determine the estimated value of Fisher's criterion. Fisher's criterion value in the table F_t with F_x compared to the accounting value $F_t > F_x$ the condition is met and the adequacy of the model is established.

Each regression coefficient in the regression equation is significant in describing the output factor.

If the value of the regression coefficient in front of the input factor participating in the regression model is large, the influence of this factor on the output factor is high, and on the contrary, the influence is small. The signs in front of the coefficients show how they affect them, that is, if (+) is positive, the effect of the outgoing factor is positive, if (-) is negative, it is negative.

Only important coefficients are included in the mathematical model of the process. Experiments were conducted for leather raw materials. The system of equations obtained when processing the data using the Excel computer program has the following form:

$$Y = 55.15 + 1.65x_1 + 1.65x_2 - 5.59x_3 + 1.5x_4 - 0.09x_1x_2 + 0.09x_1x_4 + 0.65x_2x_3 - 0.72x_3x_4 - 0.34x_1x_2x_4 \quad (12)$$

Mathematical calculation of the fit of the obtained equation (12) showed that the model is in good agreement with the experimental results.

The essence of the regression coefficients in the models characterizes the contribution of the relevant factor to the value of the output parameter in the transition of the factor from the main level to the bottom or top. The contribution of the factor to the value of the output parameter during the transition from the lower to the higher level is called the efficiency of the factor. The larger the regression coefficient, the higher the effect of this factor, that is, the stronger the effect of the factor on the output parameter. In this way, factors are sorted in the models according to the value of the regression coefficients due to their influence y_1, y_2, y_3 .

3. Results and discussions.

Based on the regression equation (12) determined in experimental studies, the following graphs were constructed and analyzed. A graph of the speed of the pressing shafts as a function of the residual moisture of the leather raw material was constructed at different numerical values of the diameter of the pressing shafts, i.e., $d=125\text{ mm}$, $d=225\text{ mm}$, $d=320\text{ mm}$ (Fig. 2).

If we analyze the residual moisture of leather raw materials based on the graph, it can be seen from the graph that as the speed of the pressing shafts increases, the residual moisture of leather raw materials also increases. In this case, the speed of the compression shaft is $V=0.28\text{ m/s}$, $P=32\text{ kN}$, $t_0=1\text{ mm}$, $d=125\text{ mm}$, the residual moisture content of the leather raw material was $\Delta W=52.5\%$. $V=0.30\text{ m/s}$, $P=32\text{ kN}$, $t_0=1\text{ mm}$, $d=225\text{ mm}$, the residual moisture content of the leather raw material was $\Delta W=54.5\%$. $V=0.32\text{ m/s}$, $P=32\text{ kN}$, $t_0=1\text{ mm}$, $d=320\text{ mm}$, the residual moisture content of the leather raw material was $\Delta W=56.5\%$. The graph of the speed of the compressing shafts as a function of the residual moisture of the leather raw material was constructed at different numerical values of compressive pressure force $R=32\text{ kN}$, $P=40\text{ kN}$, $P=48\text{ kN}$ (Fig. 3).

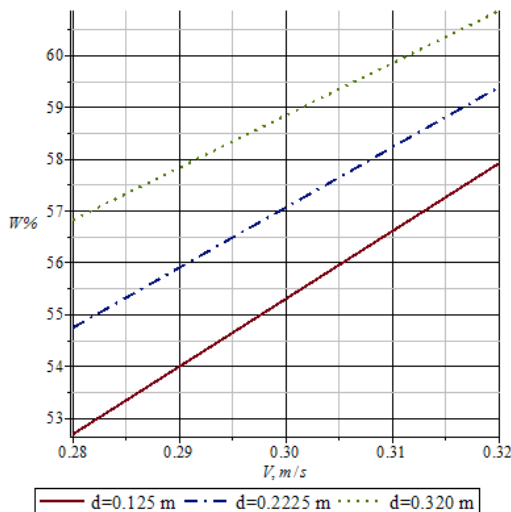


Figure 2.– The graph of the speed of the pressing shafts as a function of the residual moisture of the leather raw material (constructed at different numerical values of the diameter of the pressing shafts)

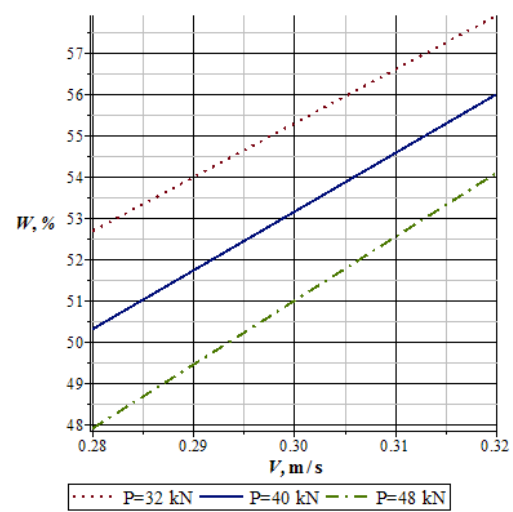


Figure 3.– The graph of the speed of the pressing shafts as a function of the residual moisture of the leather raw material (constructed at different numerical values of the compressive pressure force)

Based on the constructed graph, the following results can be cited: $V=0.28\text{ m/s}$, $P=32\text{ kN}$, $t_0=1\text{ mm}$, $d=125\text{ mm}$, the residual moisture content of leather raw material was $\Delta W=52.5\%$. $V=0.30\text{ m/s}$, $P=40\text{ kN}$, $t_0=1\text{ mm}$, $d=125\text{ mm}$, the residual moisture content of leather raw material was $\Delta W=53\%$. $V=0.32\text{ m/s}$, $P=48\text{ kN}$, $t_0=1\text{ mm}$, $d=125\text{ mm}$, the residual moisture content of leather raw material was $\Delta W=54\%$. The graph of the dependence of the compressive force of the shafts on the residual moisture of the leather raw material was constructed at different numerical values of the speed of the compressing shafts $V=0.28\text{ m/s}$, $V=0.30\text{ m/s}$ and $V=0.32\text{ m/s}$ (Fig. 4).

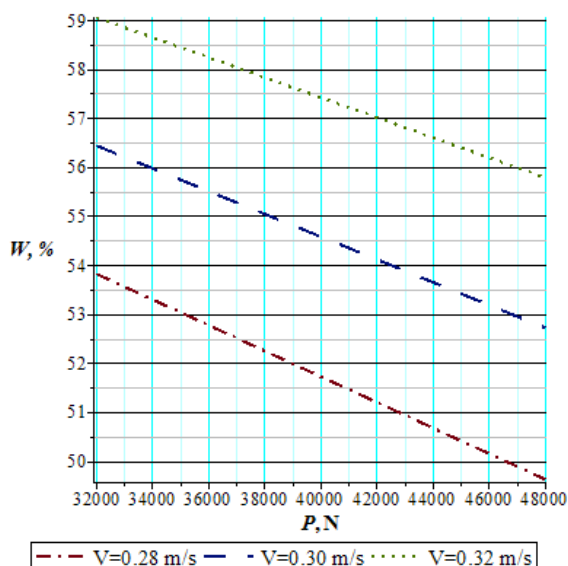


Figure 4. – The graph of the dependence of the compressive force of the shafts on the residual moisture of the leather raw material (constructed at different numerical values of the speed of the compacting shafts)

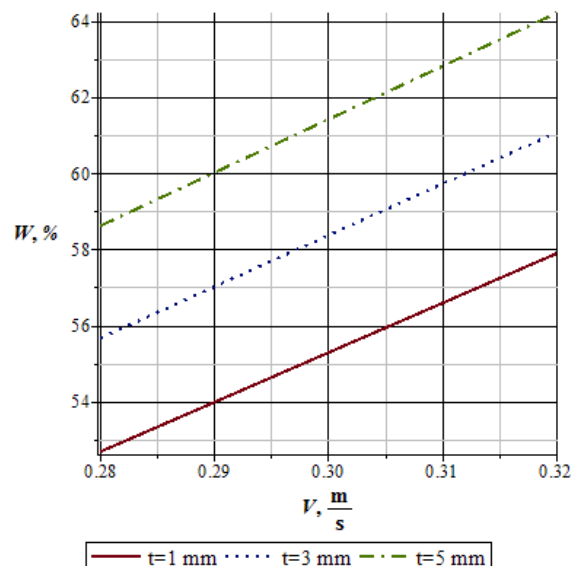


Figure 5.– The graph of the dependence of the compressive strength of the shafts on the residual moisture of the leather raw material (constructed at different numerical values of the leather raw material)

It can be seen from this graph that as the pressing force of the pressing shafts increases, the residual moisture of the leather raw material decreases. The graph of the dependence of the compressive strength of the shafts on the residual moisture of the leather raw material, different numerical values of the leather raw material, i.e. $t_0 = 1\text{ mm}$, $t_0 = 3\text{ mm}$, $t_0 = 5\text{ mm}$ was built when (Fig. 5).

In this graph, as the thickness of the leather raw material increases, the speed of the pressing shafts increases, the residual moisture content of the leather raw material increases.

4. Conclusions.

As a result of the conducted experimental research, we can conclude as follows: when the diameter of the compacting shafts is small $d=125\text{ mm}$, the value of the compressive pressure force is the largest value $P=48\text{ kN}$ and the speed $V=0.28\text{--}0.32\text{ m/s}$, the residual moisture of leather raw materials has the smallest result, The maximum residual moisture of leather raw material is obtained when the diameter of the pressing shafts is the largest $d=320\text{ mm}$, the value of the pressing force is the smallest $P=32\text{ kN}$ and the speed $V=0.28\text{--}0.32\text{ m/s}$. Also, the size of the thickness of the material, the diameter of the shafts, the compressive pressure, and the values of the speed have a great influence on the residual moisture content of the leather raw material.

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APPLICATION OF DIGITAL MODELS AND SATELLITE NAVIGATION MEANS FOR DETERMINING THE MAIN RESISTANCE TO THE MOVEMENT OF FREIGHT WAGONS

Boronenko Yu.P.¹, Kron I.R.¹, Raxmatov X.A.²

¹Emperor Alexander I St. Petersburg State Transport University, St. Petersburg, Russia

²Tashkent State Transport University, Tashkent, Uzbekistan

E-mail: x.raxmatov90@gmail.com

Abstract. The article justifies the application of computer modeling of freight wagons in calculating aerodynamic characteristics and recording motion parameters with a navigation receiver during tests to determine the main resistance to movement using the coasting method. An analysis of the test results has been performed, and a methodology for processing primary data has been defined. Resistance to movement characteristics has been constructed, and a comparison has been made with resistance values obtained through traditional dynamometric tests. A conclusion is drawn about the satisfactory convergence of the results of field tests using the developed coasting method and dynamometric tests of the investigated model wagon.

Keywords: digital wagon models, aerodynamics, navigation receiver, main resistance to movement, coasting method tests.

1. Introduction.

One of the most priority directions in optimizing the movement process on the railways of the Russian Federation is to maintain the volume of transportation while reducing the associated costs [1-3]. It is known that in the operation of rolling stock, the main consumer of energy resources is the process of creating traction force by the locomotive to overcome the resistance to movement [4]. The resistance to movement is conventionally represented by components, with the mechanical and aerodynamic components being the main ones.

Previously, a hypothesis was put forward [5] regarding the possibility of conducting tests to determine the main resistance to movement using the coasting method, similar in essence to the rolling method [6], which is primarily applied to self-propelled rolling stock due to its technological features. For freight wagons, the dynamometric method is currently used to determine the main resistance to movement [7], requiring a minimum of 65 wagons of the same model in an experimental train. This methodological requirement poses difficulties when there is a need to assess the resistance to movement of innovative wagons that have not yet entered mass production, and also entails significant financial costs for conducting such tests.

In this regard, it was proposed to conduct coasting tests of freight wagons for determining the main resistance to movement [8-10]. These tests can only be carried out with specialized equipment designed for this type of research. The equipment includes an electromagnetic valve remotely controlled by an associated electronic block and a control panel, ensuring operation at a distance of at least 100 meters. It also includes a precision manometer, a navigation receiver [11] MNP-M7, and a computer (PC) mounted on the test wagon, along with a power source with an output voltage of 12 V.

2. Results and discussions.

The navigation receiver MNP-M7 calculates the current coordinates of the wagon's position in local geographic coordinates (latitude, longitude, height), time, and the vector of ground speed, receiving up to 24 GLONASS signals in parallel. The absolute accuracy of the device in determining coordinates is 2 meters when using the code differential data logging mode with a confidence probability of 0.95 [12].

To conduct tests for determining the resistance to movement using the coasting method, a series of preparatory works is required. These include measuring the parameters of the test section of the railway track and designing models of the test wagon for subsequent aerodynamic calculations during single wagon movement and its travel within a train.

The developed models of the freight semi-wagon model 12-196-01, subjected to full-scale tests according to the methodology, are presented in Fig. 1.

a

b

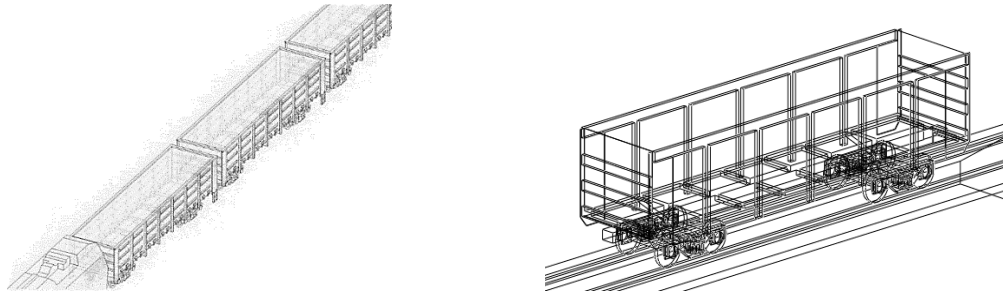


Figure 1. – **Models of the semi-wagon 12-196-01 for aerodynamic calculations: a – as part of a train; b – during single movement**

The task of determining the aerodynamic drag, taking into account the viscosity of the air medium, is reduced to solving a specific case for the Navier-Stokes equations system, as depicted in Figure 2:

$$\begin{cases} \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{v}) = S_m, \\ p \frac{\partial \vec{v}}{\partial t} + p(\vec{v} \cdot \nabla) \vec{v} = -\nabla p + \mu \nabla^2 \vec{v} \end{cases}, \quad ((1))$$

where, ρ – the static pressure, t – time, $\vec{v}$ – the vector of the calculated medium, S_m – the mass source (sink) of the calculated medium, p – the static pressure, and μ – the molecular viscosity.

The measurements of track parameters were carried out using satellite geodetic equipment, including Spectra Precision ProMark700 and Spectra Precision SP80. The accuracy of measurements was verified using the N-3 level, and the difference in measurements was found to be 0.1 mm. The measurement process and its results are depicted in Figure 3.

The essence of the tests lies in subjecting the non-propelled rolling stock sample, moving on the test section in a test consist of two locomotives and a laboratory car equipped with a remotely controlled brake system and a navigation receiver linked to a computer, to acceleration up to the test speed. Subsequently, it is uncoupled from the laboratory car and freely rolls under the influence of inertia forces until it comes to a complete stop or until the brake is applied using the remote control [13].

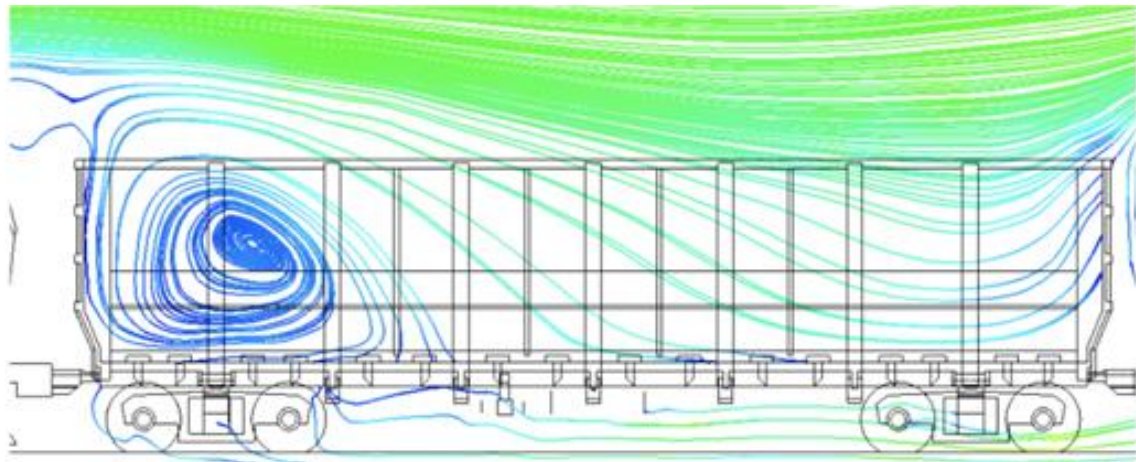


Figure 2. – **Velocity gradient lines in the air when solving the system of equations (1)**

Initial velocities for uncoupling are established in the developed methodology and are set at 30, 60, and 90 km/h. For each velocity, no less than three uncoupling are performed to accumulate statistical data.

The obtained experimental data are processed using a specially developed program for these tests called “Resistance to Motion Handler” (RTM Handler), implemented in the Python programming language [14]. The processing involves transformation, cleaning, and analysis of the data using internal and external libraries.

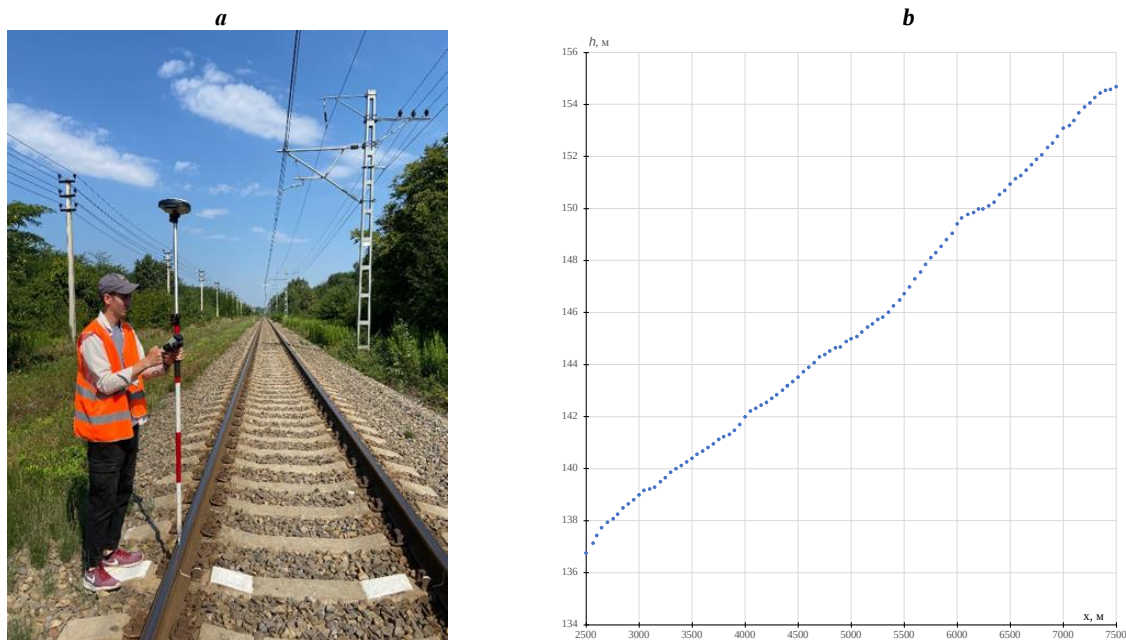


Figure 3. – Measurement of parameters of the test track in Belorechensk, Russia: *a* – measurement process; *b* – measurement results

From the accurately leveled coordinates of the test track with the established elevation above sea level, the time and velocity values of the wagon are extracted from the experimental primary data. The work of the resistance forces during the wagon displacement is estimated based on the change in velocity and elevation. According to Newton's second law and the law of conservation of energy, the acceleration of the wagon is calculated using the formula:

$$a = \frac{1}{S} \cdot \left[\frac{V_2^2 - V_1^2}{2} + g(h_2 - h_1) \right], \quad (2)$$

here, a – acceleration of the wagon; S – distance covered during the change in velocity from V_2 to V_1 ; V_2 and V_1 – respectively, the final and initial velocity values of the wagon during the time interval t ; g – acceleration due to gravity; h_2 and h_1 – respectively, the final and initial elevation coordinates of the wagon during the time interval t .

Knowing the gross mass of the investigated wagon, one can determine the resultant force of motion resistance using Newton's second law formula.

The calculated acceleration values underwent a Kalman filtering process [15], in accordance with which the system's state at any given time is determined by the expression

$$x_n = \psi_{n,k} x_k + \sum_{i=k}^{n-1} \psi_{n,i} B_i u_i, \quad (3)$$

where, $\psi_{n,k}$ – the transition state matrix; x_k – the system state at the initial time k ; B_i – the rectangular matrix of coefficients; u_i – the input sequence describing the system.

The obtained values were subjected to polynomial regression analysis. Its adequacy was assessed based on the coefficient of determination.

Thus, the total resistance to motion of the tested wagon in the train is given by

$$W_K = W_{K0} - W_{aap,0} + W_{aap,n},$$

where, W_K – main resistance to motion of the tested wagon in the train; W_{K0} – main resistance to motion of a single tested wagon obtained in the experiment by processing measurements of the wagon's speed and its position on the track; $W_{aap,0}$ – the aerodynamic resistance of a single tested wagon obtained by calculation on the digital model of a single wagon; $W_{aap,n}$ – is the aerodynamic resistance of the tested wagon in the train obtained by calculation on digital models of the wagon in the train.

To enhance the clarity of the resistance forces variation with respect to the wagon's speed, the data obtained during tests at different speeds were overlaid. The resulting statistical data were approximated and are presented in Figure 4.

The developed methodology for conducting tests to determine the main resistance to motion of freight wagons by the coasting method has been verified and aligned with the Scientific Research Institute of Railway Transport (JSC "VNIIZHT") and the Technical Policy Department of Russian Railways (JSC "RZD") [16, 17].

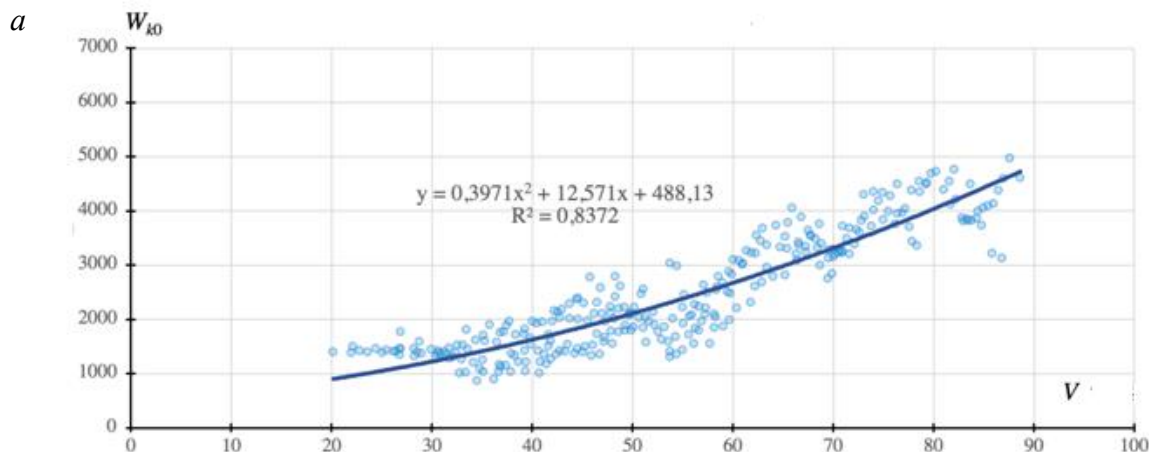


Figure 4. – Characteristic of the main resistance to motion of the investigated wagon model 12-196-01: *a* – in loaded condition; *b* – in empty condition

The use of a more advanced navigation receiver as a primary data recorder can guarantee an increase in the method's accuracy. The difference in values compared to the dynamometric method of determining resistance to motion averaged 15% and may be associated with measurement errors in both methods, as well as the peculiarities of the technical condition of the tested wagon. In the future, to eliminate the influence of the specific sample's technical condition, consideration is given to increasing the number of tested wagons by the coasting method to 3-5.

3. Conclusions.

As experimental data accumulates in the future, it may be possible to dispense with full-scale tests of wagons to determine motion resistance, provided the hypothesis of the identical nature of mechanical resistance for specific models of freight wagon trucks is confirmed.

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AERODYNAMIC FEATURES OF THE TWO-TIER ROTOR OF THE WIND UNIT AND ITS DRIVING TORQUE

Dekhkonov U.G.¹, Najmiddinov I.B.¹, Karabayeva M.U.¹, Azamov K.S.¹

¹Namangan Engineering - Construction Institute, Namangan, Uzbekistan

E-mail: znaniyasila7@yandex.ru

Abstract: The article analyzes the aerodynamic features of the rotor structure of the wind turbine. It is based on the laws of aerodynamics. It is shown that this feature is the main indicator in the design of the rotor structure, of moment gives the equations of a twisting propelling rotor with a vertical axis with a curved wing arising under the pressure of the wind flow. The relationship between dynamic, kinematic and geometric parameters is reflected in the equation.

Keywords: Aerodynamics, air, drag coefficient, differential equations, numerical method, wind, rotor.

1. Introduction

Based on the results of the conducted scientific research, we come to the following conclusions in order to design an effective, independent wind direction vector, large-diameter, small-speed and high-power wind turbine rotor construction suitable for the wind regions of our Republic and our technological capabilities [1-3].

The bottom line is that a rotor producing 6 blades of energy must have the useful efficiency of a 3-blade rotor. Also, the aerodynamic properties of the rotor should be high, i.e., the process of passive air absorption behind the wing should be accelerated and the coefficient of resistance of the nose should be increased, so that the wind flow can be quickly moved away from the rotor by transferring its kinetic energy [4-6].

Let the height of the prototype rotor be h m (Fig. 6). We divide it into two equal parts with a horizontal plane P at a height of $h/2$ m. We turn its separated second - upper tier part 1800 around the X axis in the counterclockwise direction. A top view of both of their parts is shown in Figures 2a and 2b. It can be seen from them (Figure 1 shows a 4-bladed rotor) that the proposed design is considered in the case of a 3-bladed rotor. It can be seen that the rotor layer shown in Fig. 7a moves counterclockwise, and the rotor layer shown in Fig. 7b moves clockwise: the active vane 2 of the shaft 1 moves under the wind pressure received by the closed plates 3, vanes 4 and 5 is in a passive position, because the wind pressure affects the plates 6 from the rear and moves them to the vane position [7-9]. It can be seen from the figure that the asset.

2. Statement of the problem

It can be seen from the figure that the center of the active wing 3 has a width of open surface 7 that passes the wind flow. The wings are interconnected by steel ropes 8 at the lower and upper ends of the wings. This allows the wing to operate without jerks in the instantaneously changing wind speed, it isolates the wings [10-11].

This process, which happened around the shaft 1 of the lower tier, takes place in the second tier of the rotor around the shaft 9, with only the opposite movement.

If we put the second layer of the rotor on top of the first layer of the rotor, the rotor shown in Figure 3 is formed.

In this case, the assembly process is as follows: the shaft 9 is passed through the shaft 1, and with the help of links with a joint (radial bearing), the upper and lower tiers of the rotor are ensured to rotate in opposite directions (Fig. 3). The vertical position of the shafts is ensured as follows: the shaft 9 is mounted on the radial-support bearing 10 in the lower part, and the shaft 1 forms a hinged joint with the help of the radial bearings 11 and 12.

In turn, the shaft 1 is fixed to the frame frame 13 with the help of radial bearing 14 and support bearing 15.

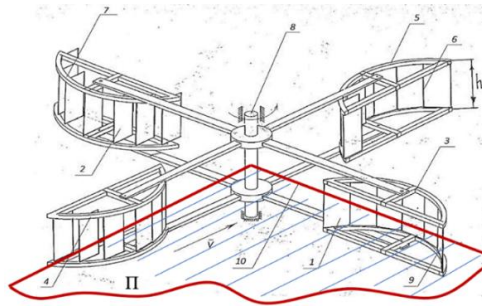


Figure 1. A single-stage rotor with an open center of the blade

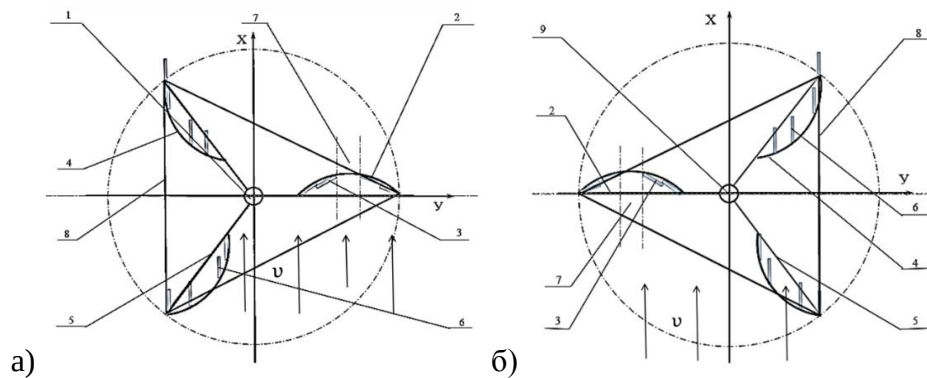


Figure 2. Horizontal of the two layers of the rotor

Another important aspect of the design is that the center of each wing is left open to let the wind flow through (Figure 4). If we pay attention to the figures, we can estimate that the aerodynamic processes of the wind flow are different in rotor constructions with a wing consisting of a central open part of the working surface 7 and a wing with a fully closed working surface [12-14].

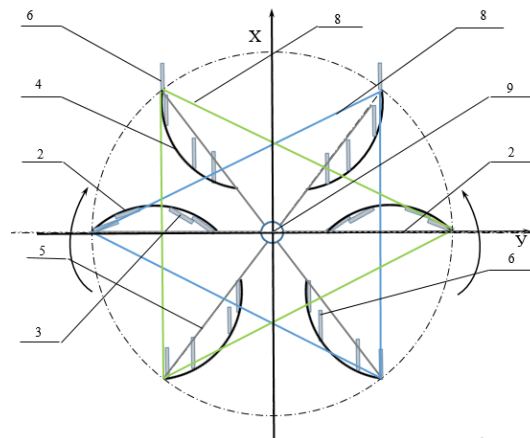


Figure 3. - 3-assembled view of the rotor

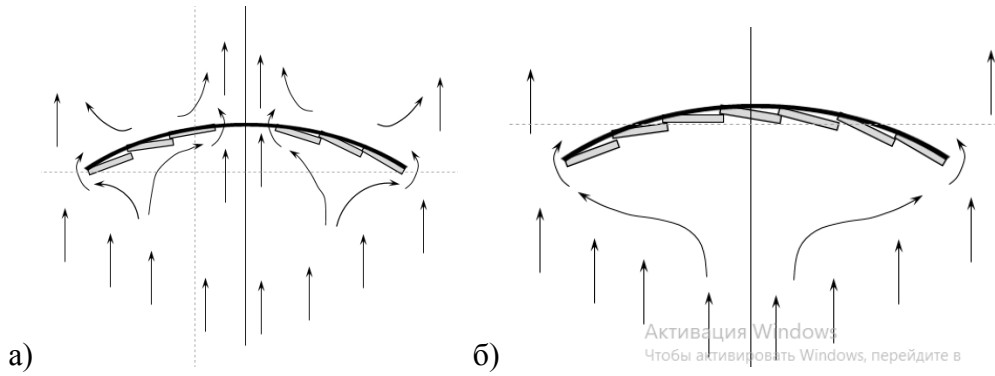


Figure 4: a) open and b) closed surface wings

In the following, we define the mathematical expression of the driving moment for the rotor design

It is known that the rotor may operate in four distinct positions, each associated with a corresponding change in the driving force applied to the wings during one cycle of operation. In this paper, we will analyze the second and third positions and derive their respective expressions.

An active cathode with the center plate relay open. According to it, the wind flow enters along the concave surface of the wing and flows along the central zone. While the central opening helps to suck the static air behind the wing in its direction, the wind stream flowing through the peripheral zones sucks the trailing static air close to it. In this way, the value of the air mass flowing over the wing surface increases [15].

3. Numerical results and their analysis

The aerodynamic process on the surface of the rotor with the central plate open can be analyzed as follows: according to Bernoulli's equation,

$$p_1 + \frac{\rho \cdot v_1^2}{2} + z_1 \cdot \rho g = p_2 + \frac{\rho \cdot v_2^2}{2} + z_2 \cdot \rho \cdot g \quad (1)$$

Here, p_1 and p_2 are the static pressure on the surfaces, $\frac{v^2}{2}$ is the aerodynamic pressure, and z is the height of the surfaces through which the wind flows. Since in our conditions the wind current flows over a surface of uniform height, we can write equation (1) as follows:

$$p_1 + \frac{\rho \cdot v_1^2}{2} = p_2 + \frac{\rho \cdot v_2^2}{2} \quad (2)$$

Therefore, the pressure difference, which is the main indicator of energy production, will be equal to equation (3) [18].

$$p_1 - p_2 = \frac{\rho \cdot v_1^2}{2} - \frac{\rho \cdot v_2^2}{2} \quad (3)$$

The essence of this equation is that in the zone of increased kinetic pressure, the static pressure decreases, and vice versa, the static pressure increases. Due to the almost motionless air behind the wing, the static pressure increases, and it tends to the areas of the wing that pass the wind flow, creating a relative vacuum behind the wing - that is, this process leads to the equation of non-self (nerazryvnosti sredy) of the flow ($S_1 \cdot v_1 = S_2 \cdot v_2$) according to the fact that the wind speed in the open part of the center of the wing increases slightly.

Based on the above-mentioned scientific research and certain technical laws, the proposed new design of the rotor of the wind turbine is shown in Figure 5.



Figure 5: Rotor of wind turbine "Samarkand", № IAP 2022 0106

The second position occurs when the second wing is restricted from moving in the direction of the wind at point B of the first wing, in shape 1. The third position is when the second wing travels from point A of the first wing relative to the direction of the wind, in shape 2.

The thrust moment they are generated by the rotor when in position 2 is influenced by the angle at which the wings are oriented relative to the oncoming wind. This angle significantly impacts the thrust generated, which is determined by how the operating surface of the second wing varies. The first wing is always fully operational. [16]. This is the statement of the second wing:

$$l \cdot \sin \varphi = r \cdot \sin(180 + \delta)$$

Equality begins with finding a solution. Precisely,

$$\delta = \arcsin\left(\frac{l}{r} \sin \varphi\right)$$

when equals the value, the angle of inclination of the first wing in the starting position begins with angles at the corners of the first and second wings. As a result, the forming surface of the second wing at a α_{s4} angle generates no driving torque. In this case, α_{s4} increases as φ increases, and its value can reach 90 degrees. Consequently, it is completely blocked by the second wing. In Form 1, α_{s4} represents an angle. The challenge lies in determining how α_{s4} varies when the angle of inclination changes.

Occurs when the value is. When the values of the φ and $\varphi+1200$ angles on the second wing increase, the angle of inclination of the first wing at the initial position begins to move toward the second wing. As a result, the surface formed at the base of the corners of the second wing doesn't create driving torque. In this case, the α_{s4} angles increase by φ and their value reaches 900. This is fully covered by the second wing when in form 1. In form 1, the α_i angles are represented as number α_{s4} , where the question of determining the change in value from φ to α_{s4} when the angle moves from [5] answered.

$$\text{from the drawing } \frac{R}{\sin(\angle O_1 O S_{\alpha_4})} = \frac{R}{\sin(\angle O_1 O A + \angle A O_1 O S_{\alpha_4})} = \frac{R}{\sin(\angle O O_1 S_{\alpha_4})} \text{ or}$$

$$\frac{R}{\sin(\beta + \delta_1 - (\varphi + \frac{2\pi}{3}))} = \frac{r}{\sin(\alpha_3 + \alpha_{s4})} \text{ and in this } \delta_1 = \delta + \frac{2\pi}{3} \text{ considering that}$$

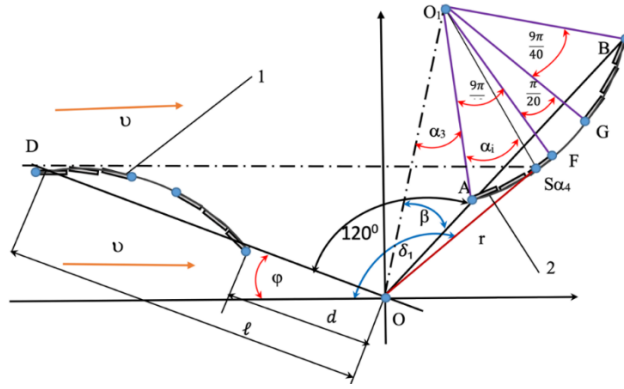


Figure. 6. Regarding the calculation of the torque of the Rotor 2nd wing in position 2

$\sin(\alpha_3 + \alpha_{s4}) = \frac{r}{R} \sin(\alpha + \beta - \varphi)$ and from it we find the final equation:

$$\alpha_{s4} = \arcsin\left(\frac{r}{R} \sin(\alpha + \beta - \varphi)\right) - \alpha_3$$

Where it is known that expressions r , α_3 , δ and β are represented by the following equations [17]:

$$r = \sqrt{R^2 + |OO_1|^2 - 2|OO_1| \cdot \cos[\alpha + \alpha_3]}, \quad \alpha_3 = \arccos\left(\frac{|O_1O|^2 + R^2 - d^2}{2 \cdot R \cdot |O_1O|}\right)$$

$$\delta = 2 \cdot \arctg \frac{-B1 \pm \sqrt{B1^2 - 4A1 \cdot C1}}{2A1}, \quad \beta = \arctg \frac{|O_1C|}{l \cdot k + a}$$

To this equation and Fig bazlanib 2-wing for holding events based on the conditions of [18]:

$$0 \leq \alpha_{s4} \leq \alpha_F$$

$$M_2 = \frac{1}{2} \cdot c \cdot \rho \cdot h \left[\int_{\alpha_{s4}}^{\frac{9\pi}{40}} (\vartheta \cdot \sin \delta_1 - \omega \cdot r)^2 \cdot r \cdot d\alpha + \int_{\frac{11\pi}{40}}^{\frac{\pi}{2}} (\vartheta \cdot \sin \delta_1 - \omega \cdot r)^2 \cdot r \cdot d\alpha \right]$$

$$\alpha_F < \alpha_{s4} < \alpha_G$$

$$M_2 = \frac{1}{2} \cdot c \cdot \rho \cdot h \cdot \left[\int_{\frac{11\pi}{40}}^{\frac{\pi}{2}} (\vartheta \cdot \sin \delta_1 - \omega \cdot r)^2 \cdot r \cdot d\alpha \right]$$

$$\alpha_G \leq \alpha_{s4} \leq \alpha_B$$

$$M_2 = \frac{1}{2} \cdot c \cdot \rho \cdot h \cdot \left[\int_{\alpha_{s4}}^{\frac{\pi}{2}} (\vartheta \cdot \sin \delta_1 - \omega \cdot r)^2 \cdot r \cdot d\alpha \right]$$

Here are the values derived from rotor design α_F , α_G and α_B : 13, 14 and 15. Based on the number of equations, it can be concluded that the second-stage wing undergoes three homogeneous dynamic transformations during its second position.

The driving moment of the rotor tier wing in position 3:

Once the tier 2 fan advances behind wing 1, modification φ transfers it to position 3. At the same moment, wing 2 begins to expand again in the direction of the wind from point A, which is its bottom section.

$d \cdot \sin \varphi \geq r \cdot \sin(180^\circ - \delta_1)$ marks the start of state 3 of the limitations. Even in this instance, it will seem homogenous with the following equation:

$$\alpha_{s5} = \alpha_3 - \arcsin\left(\frac{r}{R} \sin(\alpha + \beta - \varphi)\right)$$

The arc-shaped surface formed on the angular base α_{S5} enters an active functioning state, whereas the surface built on the angular basis α_{S4} remains passive. This is critical for determining the limit of the integral.

$$M_2 = \frac{1}{2} \cdot c \cdot \rho \cdot h \cdot \left[\left(\int_0^{\alpha_{S5}} (\mathcal{G} \cdot \sin \delta_1 - \omega \cdot r)^2 \right) \cdot r \cdot d\alpha \right]$$

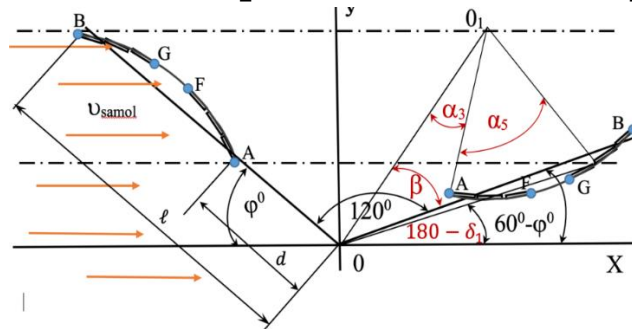


Figure. 7. We calculate the torque of rotor wing 2 in position 3 using the following conditions ($v_{shamol} - v_{the wind}, \alpha_{s5} - \alpha_5$):

And in the range $\alpha_G \leq \alpha_{S5} \leq \alpha_{SB}$ to

$$M_2 = \frac{1}{2} \cdot c \cdot \rho \cdot h \cdot \left[\int_{\alpha_G}^{\alpha_{S5}} (\vartheta \cdot \sin \delta_1 - \omega \cdot r)^2 \cdot r \cdot d\alpha \right]$$

The equations will be suitable. It is worth noting that in scenario 2, angle α_{S5} will represent the integral's beginning coordinate, α_{S5} as well as its end coordinate. However, in the $\alpha_G \leq \alpha_{S5} \leq \alpha_B$ range, the surface moment caused by the wing autofocus arc will be relatively tiny, because the surface is nearly parallel to the wind direction.

The result is that the second wing undergoes two homogenous dynamic changes at the third position of the tier. The term "tier" indicates that the rotor is divided into two tiers. We studied the second wing of one tier here. The second wing of the second tier will have the same characteristics, but with a 600 delay [19].

These equations do not require answers in engineering calculations; instead, they allow you to calculate with the required precision utilizing computer-based approximate solution methods.

4. Conclusions

The rotor operates unchanged in the direction of rotation, regardless of changes in the direction of the wind vector.

The pressure of the wind molasses on the wing area is controlled by changing the angular speed of the rotor, while ensuring safety from breakage.

The rotor drag coefficient reaches up to 1.4.

The open part of the wing allows the relatively static air to be sucked out behind the wing.

The rotor generates energy like a six-wing rotor, the efficiency is like a three-wing rotor.

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USING REGRESSION ANALYSIS FOR DEMAND FORECASTING IN RAILWAY FREIGHT TRANSPORTATION IN KAZAKHSTAN AND EVALUATING THE QUALITY OF THE FORECASTS

Djabbarov Sh.¹, Abdullaev B.¹, Inagamov S.¹, Abdusaid Y.¹

¹Tashkent State Transport University, Tashkent, Uzbekistan

E-mail: shuhratassistant@gmail.com

Abstract. Study finds time series analysis improves demand forecasting accuracy for railway freight transportation in Kazakhstan. The accuracy of demand forecasting is crucial for efficient resource planning in organizations, particularly in the railway freight transportation industry where many factors can influence future volume and turnover. However, traditional expert methods may no longer be sufficient to meet modern requirements. This article compares the accuracy of demand forecasts made using autoregressive integrated moving average (ARIMA) and expert methods, and finds that time series analysis using ARIMA can significantly improve accuracy. The study suggests that integrating time series analysis into enterprise practice can benefit transportation industry enterprises in Kazakhstan by improving demand forecasting and resource allocation, resulting in improved business success.

Keywords: demand forecasting, railway freight transportation, regression analysis, quality assessment of forecasts.

1. Introduction.

Accurate demand estimation is crucial for effective planning and decision-making in any organization, providing input for departments such as marketing, production, distribution, and finance [1]. For transport companies like Joint Stock Company “National Company “Kazakhstan Temir Zholy” (KTZ), forecasting future demand is critical for success and provides input for planning and control of functional areas such as transport operations planning, marketing, and finance [2]. KTZ is a significant player in the rail transportation industry, with over 21.000 km of railway lines, a fleet of approximately 46.000 freight cars, more than 2.000 passenger cars, and over 1.8 thousand locomotives. Accurate demand estimation helps businesses plan and organize accordingly [3]. The demand for railway transportation services is calculated based on the volume of cargo transportation in tons multiplied by the distance transported in kilometers, also known as freight turnover. Tariff freight turnover, which considers the shortest distance between the loading and unloading points, is used to estimate demand and is the basis for calculating future revenues from freight traffic. To

generate accurate forecasts, various forecasting methods have been developed using qualitative and quantitative approaches, including mixed or combined models [4].

Numerous studies have focused on constructing models to describe the demand for rail services, including [2, 5-8]. However, most of these studies concentrate on analyzing cities where demand for rail transport is driven by urban sprawl and deteriorating road transport conditions. Fewer models have been developed to assess the functioning of larger national rail networks, such as those in Sweden and India [9, 10]. Additionally, [10] reviewed the application of data envelopment analysis (DEA) in the transport sector. [10] propose various models to forecast demand in the regular passenger transport industry and compare them to choose the best one.

The objective of this study is to identify parameters of a mathematical model of rail cargo transport performance based on historical data to make reliable forecasts of future demand. The study investigates the national (Kazakhstan) railway system, proposes several models dedicated to this type of empirical data, establishes selection criteria, identifies the best model, and assesses its accuracy and effectiveness. Demand analyses and forecasts are crucial for developing transport policies, but demand data are not always available due to a lack of appropriate mathematical models for generating forecasts. Thus, it is essential to analyze railway systems in various countries to select appropriate methods for forecasting transport performance.

2. Materials, methods, and object of study.

KTZ relies on an individual using MS Excel and Access for demand planning of cargo transportation via rail. However, this method has limitations and can lead to deterioration in the quality of forecasts. To improve the process, KTZ plans to automate it using modern software such as SAP HANA and IBS SPSS Modeler (hereafter, SPSS). A pilot study was conducted using specialized software to forecast future traffic and freight turnover.

The study involved collecting data from KTZ systems and macroeconomic indicators, creating and testing a model using 2012-2016 test data, generating a monthly forecast for 2017, and evaluating the forecast's quality using the MAPE or MAE method. The quality of the forecast was compared between the manual forecast made by Marketing and Tariff Policy Department (hereafter, MTPD) experts and the forecast made by specialized software.

$$MAPE = \frac{1}{N} \sum_{t=1}^N \frac{|Z(t) - X(t)|}{Z(t)} \cdot 100\%. \quad (1)$$

$$MAE = \frac{1}{N} \sum_{t=1}^N |Z(t) - X(t)| \quad (2)$$

SPSS, a specialized software for data analysis and forecasting, was chosen due to its top ranking in the Gartner 2017 Data Science platform category. Three techniques were utilized to determine the most effective approach: ARIMA model, neural net model, and autofitting. In the experiment, five years from 2012 to 2016 of monthly historical data on traffic volume for 13 aggregated cargo categories were analyzed using SPSS.

The correlation between the macro indicators and historical data was analyzed using special tools in SPSS. The best forecast generated by SPSS (ARIMA) was compared with the actual data for 2017 and the MTPD forecast generated by the old method described in the introduction section of the article.

3. Results and discussions.

The research results and experiment are shown in the graph below (figure 1), comparing freight traffic for all goods and cargo types. The green line shows the forecast of MTPD KTZ experts, the blue line shows the ARIMA forecast, and the red line shows the actual traffic in 2017. The ARIMA forecast closely matches the actual traffic from the third month of 2017, while the expert forecast differs significantly. The MAPE indicator for the expert forecast was 9.2%, while for ARIMA it was 2.0%, indicating the superior quality of the ARIMA forecast.

The ARIMA model improved coal traffic forecasting for KTZ due to the high correlation between traffic and macro indicators, as well as coal traffic seasonality. IBM SPSS Modeler's "predictor screening" feature can identify crucial predictors for accurate forecasting by analyzing data correlation and forecasts from leading agencies.

The study used 260 indicators from various data sources and acquired macro indicators with the right format and frequency for correlation analysis. Using these methods reduces forecast and planning time from 3 months to 3-5 days, develops more forecasting scenarios, reduces average monthly forecast errors, and enables accurate calculation of demand volume and timing. Combining automation and MTPD experts' knowledge leads to significant progress in demand forecasting and business planning at KTZ.

Here are some key areas for improvement when using forecasting systems with ARIMA models:

1. Inadequate data can lead to less accurate forecasting results, as seen in some cases where IBM SPSS forecasting results were inferior to KTZ forecast results. When historical data is insufficient, the "expert" forecasting method may be the only alternative.

2. While the system can generate automatic forecasts, corrections by an expert may still be necessary to fully comprehend the forecast and evaluate its reasonableness. Any changes made by an expert to the "automatic" forecast must be documented for future training.

3. Using a larger number of predictors can potentially improve the accuracy of the forecast. However, limited macro- and microeconomic indicators in the Republic of Kazakhstan are a natural limitation of the system.

4. Maintenance and updates are required for any data analysis and forecasting system to receive and upload new traffic data and predictors. The benefits of using the system should outweigh the costs of maintaining it.

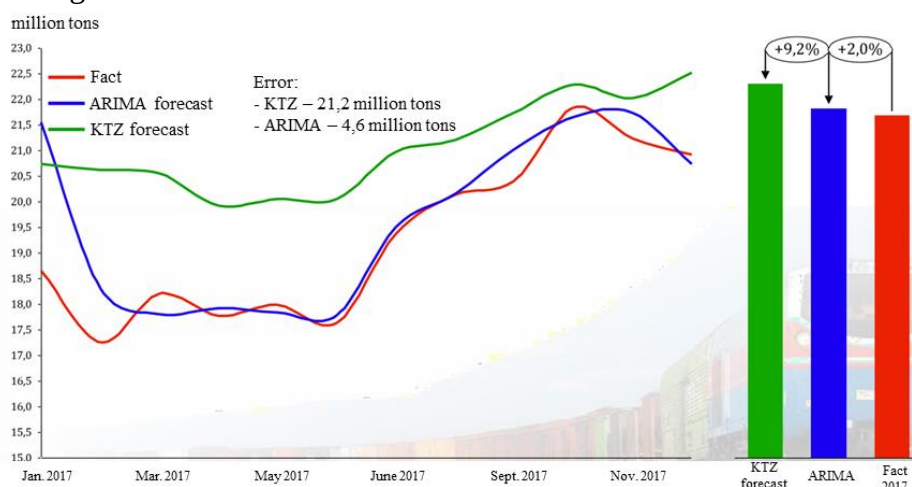


Figure 1. – Cumulative traffic forecast for all nomenclature and message types compared to the 2017 fact and MTPD KTZ forecast

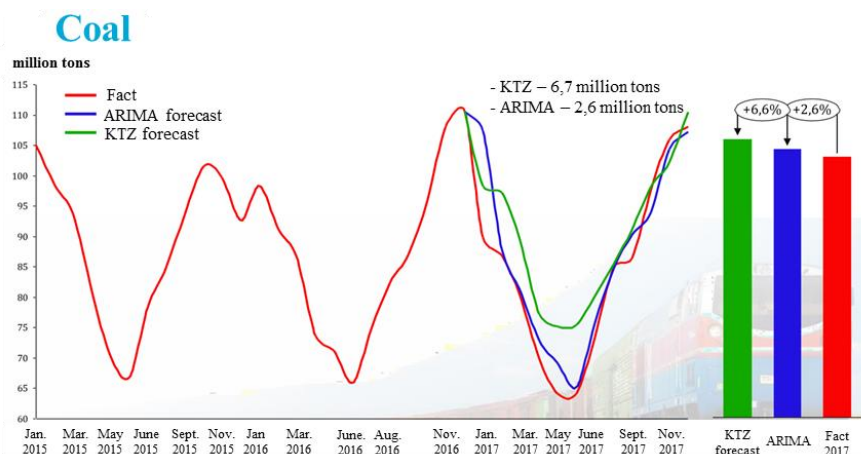


Figure 2. – Comparison of fact with MTPD experts' forecast and SPSS simulation result (ARIMA model) on the total coal transportation volume in all modes of communication

4. Conclusions.

The study evaluated two forecasts for freight transportation accuracy in Kazakhstan Temir Zholy (KTZ) using historical data and macroeconomic indicators analyzed and forecasted with IBM SPSS Modeler. The forecasts were compared to actual data for 2017 and forecast data by MTPD experts. Modern mathematical and statistical models like SPSS were comparable to those created by experts, particularly for goods with complete data and historical transportation volume. The study suggests implementing these models in Kazakhstan's largest enterprises, including the transport industry, which can save up to 8,000 man-hours of work. The study's methodology, results, and recommendations were implemented in practice at KTZ. The study led to the launch of the "Integrated Planning System" project.

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STUDYING THE PROCESS OF SEPARATION OF COTTON RUSHANKA SEEDS IN THE WORKING AREA OF THE SEPARATOR

Djuraev A¹., Juraev D.D².

¹Tashkent Institute of Textile and Light Industry, Tashkent, Uzbekistan.

²Navoi Department of the Academy of Sciences, 210100 Navoi, Uzbekistan.

E-mail: dadaxonjurayev91@mail.ru

Abstract. The article provides a design diagram and operating principle of the developed design diagram of a separator for separating rumen seeds, including cotton seeds. The advantages of the developed separator for separating cotton seeds are presented. In addition, the results of research to substantiate the parameters of the working area of the recommended effective, resource-saving design of the separator for separating cotton seeds, the analysis of the processes of separation of cotton seeds in the working area of the separator is considered based on theoretical studies using impact theory. Formulas have been obtained for calculating the speeds of the separated parts of cotton seeds. Analyzes of the obtained graphical dependencies of the system parameters substantiate their recommended values.

Keywords: separator, rushanka, cotton seeds, beater roller, multifaceted mesh drum, separation, momentum, kinetic energy, mass, friction coefficient, tilt angle, separation efficiency.

1. Introduction.

Currently, research work is being widely carried out aimed at improving the designs of separator machines that are used for processing cotton seeds and extracting feed oils from them, as well as creating new ones in the field of light and food industries. In this direction, it is important to improve the design of the Bitter separator machine to obtain effective raw oils from seed waste, to create new effective designs of working bodies, including a universal working drum of the separator, according to an analysis of the theory of machines and mechanisms [1,2].

One of the important issues of the production network is to improve the design of the separator that separates cotton seeds and base the parameters on an analysis of the dynamics of the interacting machine units [3].

Based on experimental theories, we will get acquainted with the working bodies of the improved design of the separator and analyze the influence of the multifaceted surface of the working drum and the rashanka in the process of separating the rashanka of cotton seeds into its working area.

The known design of the Bitter separator [4] contains two parts, in the first part, in two parallel slowly rotating perforated cylindrical drums, with rapidly rotating shafts with beaters coaxially located in the drums, the kernel is knocked out from the cut seeds and husks, and in the second part, on a shaking sieve the kernel and husk are separated. Both main parts of the machine are mounted on a common frame. This design was improved [5].

The essence of the design is that the design of the separator for separating cotton seeds (Bitter separator) contains two horizontally installed rotating mesh drums, inside which beater shafts with beaters mounted on them pass [6]. The beater shafts are installed eccentrically and the drum axes rotate in the opposite direction. Under the drums there is a sieve, under which there is an inclined solid tray for exiting the machine through the sieve, which moves back and forth in the direction of movement of the rushanka. In this case, the beater shafts are made in the form of a pipe with a through hole, and the beaters are mounted on the shaft by means of rubber bushings and secured with a threaded connection. In this case, the beaters are installed on the shaft in rows with alternating beater plates horizontally and vertically. In addition, the holes of the drum grids are divided into three sections, differing in the sizes of the grid holes; these hole sizes decrease from the beginning to the end of the drum, and the holes are made in the form of a truncated cone. In this case, the largest diameter of the conical holes of the screens are located in the outer surface of the drums [7,8]. The taper of the holes is made decreasing from the beginning to the end of the drum. It should be noted that the separator drive includes one electric motor and all working parts receive corresponding movements from a common electric motor. It is important to substantiate the operating parameters of the working bodies of the separator for separating cotton seeds, which ensures the efficiency of separation [9,10].

2. Materials, methods, and object of study.

Calculation scheme and mathematical model for separating cotton seeds in the working area. Theoretical studies of the process of separating cotton seeds in a separator were considered in the static mode of movement of a multifaceted mesh drum. When the multifaceted mesh drum rotates against the rotation of the beating shaft, the impact force of the impulse of the seed crushers on the mesh surface increases, which increases the effect of separating the seed crushers. In static mode, the mesh drum will not collapse. In this case, the effect of separating the Rushan seeds will be minimal.

In Figure 1 shows a design diagram of the separation zone of cotton seeds. According to the design diagram, the flat working area of the mesh drum is installed at an angle α to the drum axis.

This angle can vary widely, especially when the mesh drum rotates.

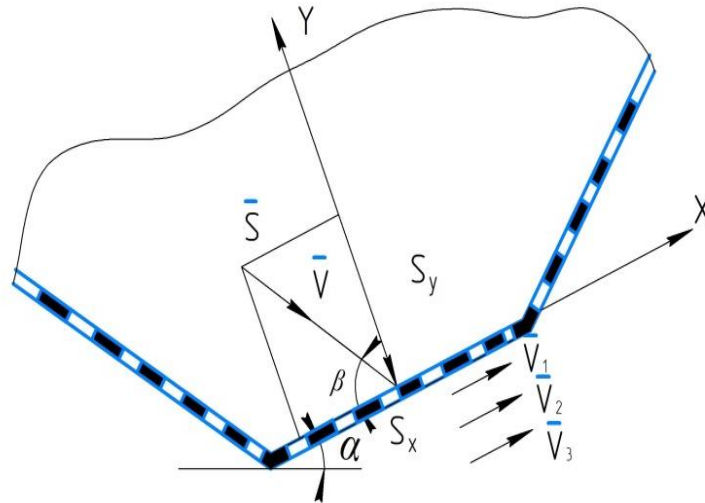


Figure1. – Calculation diagram of the separation zone of cotton seeds in the separator

The angle between the vector of the impulse force of the cotton seed rushanka and the surface of the mesh is angle β . Moreover, taking into account the components along the X and Y axes of the pulse [11, 12]:

$$\bar{S} = \bar{S}_x + \bar{S}_y \quad \bar{S}_x = S \cos \beta; \quad \bar{S}_y = S \sin \beta; \quad S = \sqrt{S_x^2 + S_y^2}; \quad S_x = f S_y \quad (1)$$

It should be noted that when parts of cotton seeds interact with the mesh surface, the change in kinetic energy turns into an impulsive force of interaction [13,14]. In this case, if the part in the general form of separation of cotton seeds is in "n"-e number of parts and taking into account the fact that the conditions of movement of these parts are parallel to the x axis due to sufficient depreciation by their layer of parts of the Rushanki, we can write:

$$\begin{aligned} \frac{1}{2} m_1 (V_1^2 - V^2) &= S \frac{\bar{V}_1 + \bar{V}}{2}; \\ \frac{1}{2} m_2 (V_2^2 - V^2) &= \frac{S}{2} (\bar{V}_2 + \bar{V}_1); \\ &\dots\dots\dots \\ &\dots\dots\dots \\ \frac{1}{2} m_n (V_n^2 - V^2) &= \frac{S}{2} (\bar{V}_n + \bar{V}) \end{aligned} \quad (2)$$

where, $m_1, m_2, \dots, m_n$ are the masses of cotton seeds separated by frequent rumples; $V_1, V_2, \dots, V_n$ - speeds of the separated parts of the rushanka; V is the speed of the Rushanka parts before interacting with the mesh; S_1, S_x, S_y – impulse and its components along the X and Y axes; f is the friction coefficient of the divided part of the seed on the mesh surface.

At the same time, taking into account the depreciation of the separated parts of the Rushanki due to their interaction with the layer of Rushanka parts on the surface of the mesh, that is:

$$V_{1_y} = 0; V_{2_y} = 0; \dots, V_{n_y} = 0; \quad m = m_1 + m_2 + \dots + m_n \quad (3)$$

where, $V_{1y}, V_{2y}, \dots, V_{ny}$ and $m_1, m_2, m_3, \dots, m_n$ are the components of the velocity and mass of the parts of the crashes vertical to the surface after the impact. It should be noted that the recovery coefficients of the impact parts of the rushans on the mesh surface along the “y” axis are chosen equal to zero (in fact, there is no rebound). Therefore, it is important to determine the speeds of the parts of the rushans along the “x” axis, which ensure not only their effective movement on the surface of the mesh drum and their effective separation.

Taking into account (1), we rewrite the system of equations (2) in the following form:

$$\begin{aligned} m_1 V_1^2 - m_1 V^2 &= VS \cos \beta_1 + SV_1 \cos \beta_1; \\ m_2 V_2^2 - m_2 V^2 &= VS \cos \beta_1 + V_1 S \cos \beta_1; \\ &\dots\dots\dots \\ &\dots\dots\dots \\ m_n V_n^2 &= m_n V^2 = VS \cos \beta_1 + V_n S \cos \beta_1; \end{aligned} \quad (4)$$

Taking into account the friction between the parts of the cotton seed roller and the mesh flat surface of the drum, according to (1), we have [15, 16]:

$$\begin{aligned}
V_1 &= V + \frac{S}{m_1} (1 - f_1) \cos \beta_1; \\
V_2 &= V + \frac{S}{m_2} (1 - f_2) \cos \beta_1; \\
V_3 &= V + \frac{S}{m_3} (1 - f_3) \cos \beta_1; \\
&\dots\dots\dots \\
&\dots\dots\dots \\
V_n &= V + \frac{S}{m_n} (1 - f_n) \cos \beta_1;
\end{aligned}
\tag{5}$$

It should be noted that the efficiency of dividing rushans into parts is estimated by the average value of the divided mass [17,18]:

$$m_{av} = \frac{m_1 + m_2 + m_3 + \dots + m_n}{n}; \quad (6)$$

Moreover, the smaller the mep value, that is, the larger the n value, the greater the efficiency of Rushanka separation.

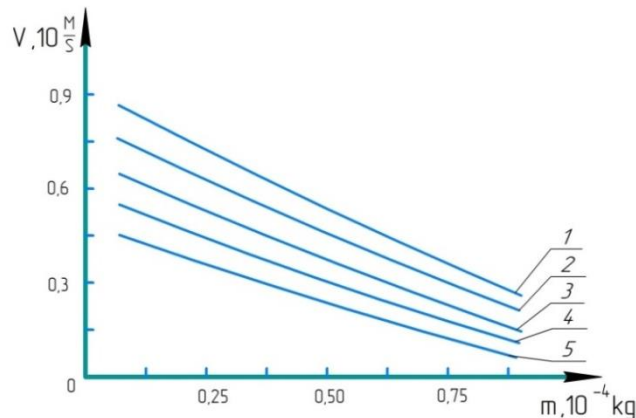
3. Results and discussions.

To solve the problem of interaction between parts of cotton seed runners, the values of their linear velocities were determined depending on their mass, impulsive force and impact angle. Calculations were carried out with the following even values of the parameters: $m=(0,15\div0,1)\cdot 10^4\text{kg}$; $f=(0,12\div0,3)$; $\beta=(40^\circ\div60^\circ)$; $S=(0,1\div0,3)\cdot 10\text{N}$.

The process of separating cotton seeds from the cotton seed is carried out continuously by the impact of the beater on the cotton seed and its separated parts. In this case, the direction of the force of influence on the crushers occurs on the helical line by installing the beaters and the angle of installation of their plates on the shaft on the in-line line. Therefore, the mixed Rushanka particles will be more crushed. To increase the productivity of the separator, it is important to increase the linear speed of the parts of the crushers after their impact interaction with the multifaceted mesh surface of the drum. At the same time, the greater the mass of the Rushanka particle, the lower the speed of its movement. In Fig. Figure 2 shows graphical dependences of the change in the linear velocity of the Rushanka particle after its interaction with the mesh on the change in the mass of the particle and on the change in the coefficient of friction of the Rushanka part on the mesh surface. Analysis of these

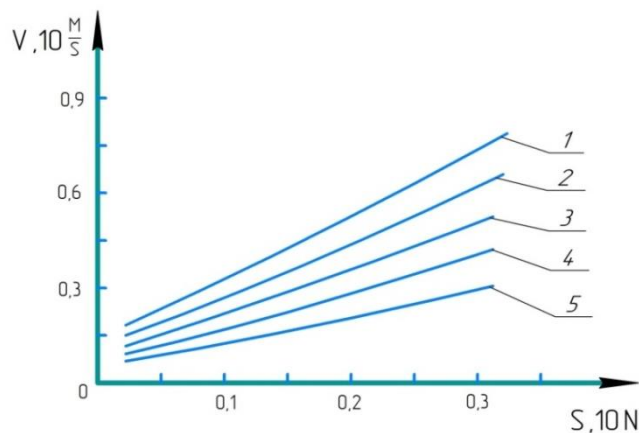
dependencies allows us to conclude that with an increase in the mass of the cotton seed ragwort particle from $0.1 \cdot 10^{-4}$ kg to $0.75 \cdot 10^{-4}$ kg with a friction coefficient $f_1=0.1$, the linear velocity of the rushanka particle after interaction with the netting decreases from $0.88 \cdot 10$ m/s to $0.22 \cdot 10$ m/s according to a linear relationship. If you increase the friction between the Rushanka particle and the mesh surface to 0.3, then the value of its speed decreases from $0.43 \cdot 10$ m/s to 0.72 m/s according to a linear pattern. This is explained by the fact that an increase in the mass of the Rushanka parts leads not only to an increase in friction with the mesh surface, but also to a decrease in its movement. Recommended parameter values are: $m \leq (0.25 \div 0.6) \cdot 10^{-4}$ kg; $f_1 \leq (0.15 \div 0.1)$.

The overall average speed of movement of parts of the cotton seed roll along the multifaceted mesh surface of the separator drum, both along the axis of the drum and in the transverse direction, also depends on the strength of the impulsive force on the part of the beater plates [19].



$$1 - f_1 = 0.10; 2 - f_1 = 0.15; 3 - f_1 = 0.2; 4 - f_1 = 0.25; 5 - f_1 = 0.3.$$

Figure 2. – Graphic dependences of the change in the linear speed of the Rushanka particle after its interaction with the mesh on the change in their masses and on the change in the coefficient of friction of the Rushanka part on the mesh surface



$$1 - \beta_1 = 60^\circ; 2 - \beta_1 = 55^\circ; 3 - \beta_1 = 50^\circ; 4 - \beta_1 = 45^\circ; 5 - \beta_1 = 40^\circ.$$

Figure 3. – Graphic dependences of the change in the linear speed of part of the cotton seed after interaction with the mesh surface on the change in the impulse strength and the angle of impact

In Figure 3 shows the dependence of the change in the linear speed of part of the cotton seed after interaction with the mesh surface on the change in the impulse strength and the angle of impact. According to the analysis of graphical dependencies, it can be stated that with an increase in impulsive force from 0.5 N to 3.1 N, the linear speed of Rushanka particles increases according to a linear pattern from 1.6 m/s to 7.5 m/s at $\beta_1=60^\circ$. With a decrease in the angle of influence of the force S to $\beta_1=40^\circ$, the linear speed V_1 increases from 0.64 m/s to 3.1 m/s proportionally.

It should be noted that the vertical component S_y leads to an increase in friction between the mesh and the crushed particles, and the horizontal component also leads to an increase in the velocity values V_1 . Therefore, the recommended values are $\beta=(45^\circ \div 50^\circ)$; $S=(0.25 \div 0.35) \cdot 10$ N.

4. Conclusions.

An effective separator design for separating cotton seeds is recommended. The results of theoretical studies of the interaction of cotton seeds with the surface of multifaceted mesh surfaces of the snare drum based on impact theory are presented.

Based on theoretical studies of the technology for separating cotton seeds, formulas were obtained for calculating their linear speed after interaction with the mesh surface. Recommended values for system parameters have been determined.

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DYNAMIC ANALYSIS OF THE MACHINE UNIT FOR CLEANING COTTON FROM SMALL POLLUTIONS

Djurayev A.¹, Yuldashev K.², Nizomov T.²

¹Tashkent Textile and Light Industry Institute, Tashkent, Uzbekistan

²Andijan Machine-Building Institute, Andijan, Uzbekistan

E-mail:temurbeknizomov@gmail.com

Abstract. In the article recommendation done piles curve has been with a drum cotton small from impurities cleaner should be conducted mechanism including dynamic and mathematical models of the machine assembly given. Pile headset and drive shaft angle speeds and screw driver of moments change laws received Speeds in this and screw driver moment vibration coverage work to the product, to the rubber bushing and moment of inertia to change dependency graphics build them analytical on the basis of the machine unit Peg headset parameters and work modes recommendation values determined.

Keywords: cotton, lint contamination, drum, inclined grid, electric motor, rotor, machine unit, operational law, loading, technological resistance, torque, angular velocity, humidity, purity, dissipation, cleaning efficiency.

1. Introduction.

The cleaning unit of the cotton gin includes supply rollers, four pile drums arranged in a row in the body, mesh surfaces under them and an auger hopper. The piles of the drum with input piles are bent by the radius R_1 from 60 to 80 mm along the direction of rotation of the drums. The bending radius of each drum pile is made 15% larger than the bending radius of the previous drum pile.

The proposed structural scheme of the cleaning unit from small impurities of the cotton cleaning unit is shown in Fig. 1. Cleaner body 1, supply rollers 2, drums with piles 3,4,5,6, mesh surfaces 12 under them, curved piles 7,8,9,10, drums 3,4,5,6 in rows along the axis, rotating sequentially set to match. At the bottom of the cleaner there is an auger 11 for removing impurities.

The cleaner works in the following order. The piled drums 3, 4, 5, 6 are dragged by the curved piles 7, 8, 9, 10 through the mesh surfaces 12 through the rollers 2 providing the cotton raw material. In this case, the separated impurities fall through the mesh surface 12 holes. The cotton that falls on the first drum 3 is sufficiently shredded. For hanging it, pegs 7 will have the smallest radius of curvature. The radius of curvature of the piles 8 of the second drum 4 is 15% larger than that of the first drum. The same procedure was also carried out in drums 5, 6 and 9, 10. In this case, the pieces of cotton falling on the last 6 drums will be sufficiently shredded. Therefore, the radii of curvature of its 10 piles were taken as maximum. In this case, the cleaning efficiency will be high and the seed damage will be reduced.

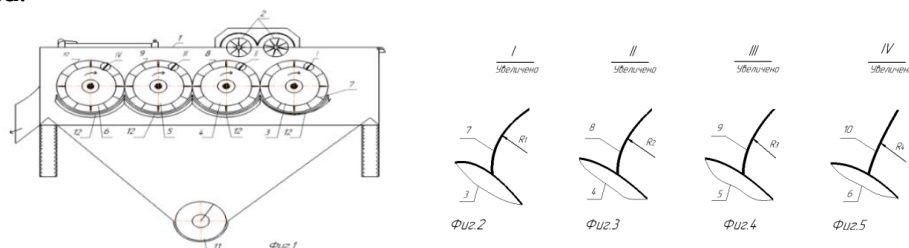


Figure 1. – (No. IAP 07541) Recommended pile drum schemes.

2. Materials, methods, and object of study.

Currently cotton small from impurities cleaner piled up drums movement asynchronous electricity drivers and strap transmissions through done is increased. The pile drum is stable in this movement laws strap extension singularity dissipative features and cotton technological resistance in consideration machine unit system according to account The scheme is shown in Fig.2. [1, 2].

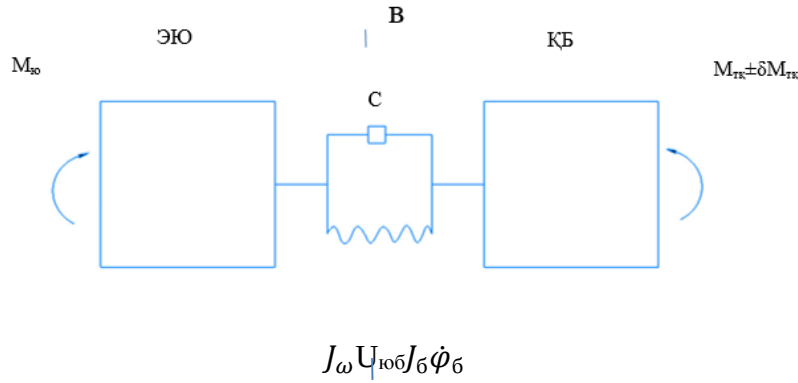


Figure 2. – Dynamic model of the dust collector machine unit

Suitable respectively duster machine ag regatta mathematical model Using Lagrange's 2nd order equations i received Asynchronous in this electricity mechanical static characteristics of the driver through account received , with tape extension singularity and dissipativ coefficients through in consideration received [3, 4]:

$$M_{ю} = \frac{2(1+\Sigma)M_k}{\frac{S}{S_k} + \frac{S_k}{S} + 2\Sigma}; \Sigma = \frac{1}{\sqrt{r^2 + r_u^2}} S = \frac{\dot{\varphi}_0 - \dot{\varphi}_H}{\dot{\varphi}_0};$$

$$S_k = \left(\frac{M_k}{M_H} + \sqrt{\left(\frac{M_k}{M_H} \right)^2 + 1} \right) \cdot \frac{\dot{\varphi}_0 - \dot{\varphi}_H}{\dot{\varphi}_0} \quad (1)$$

Here: $M_{ю}, M_H, M_k$ - driving torque on the shaft of an asynchronous electric drive, its nominal and critical values; S, S_k - slip coefficient of the electric conductor and its critical value; $\dot{\varphi}_H, \dot{\varphi}_0$ - angular speeds of the driver rotor in nominal and ideal mode; r, r_u -active and inductive resistances in short circuit. Cleaning machine unit movement expressive diffrents ial equations system [5-7]:

$$J_{ю} \ddot{\varphi}_{ю} = M_{ю} - b(\dot{\varphi}_{ю} - U_{ю\delta} * \dot{\varphi}_{\delta}) - c(\varphi_{ю} - U_{ю\delta} * \varphi_{\delta});$$

$$J_{\delta} \ddot{\varphi}_{\delta} = U_{ю\delta} [b(\dot{\varphi}_{ю} - U_{ю\delta} * \dot{\varphi}_{\delta}) + c(\varphi_{ю} - U_{ю\delta} * \varphi_{\delta}) - (M_0 + M_1 \sin \omega t \pm \delta M_0)] \quad (2)$$

3. Results and discussions.

The values of the initial parameters of the main parameters [8, 9], asynchronous electric drive $N=2.2\text{kW}$, $n=45\text{rpm}$ $f_c=50\text{ Hz}$ $\varphi_c=0.86$; $\omega_0=157\text{ c}^{-1}$ $\eta=0.835$

$\omega_n=98.91\text{ c}^{-1}$; $S_N=0.054$; $S_k^1=0.193$; $n=2$; $n_k=440\text{ rpm}$ $U_{юк}=1.0$; $J_{ю}=0.028\text{ MM}^2$;

$J_{\delta}=0.192\text{ MM}^2$; $C=(300\div 400)\text{ Nm/rad}$; $\delta=(2.4\div 2.8)\text{ Nm/rad}$; $M_{тк}=24.5+1.5\sin \omega t \pm \delta M_{тк}$ the values of angular velocities and torques of the screw driver on the shaft of the pile drum and its first electric drive shaft are given.

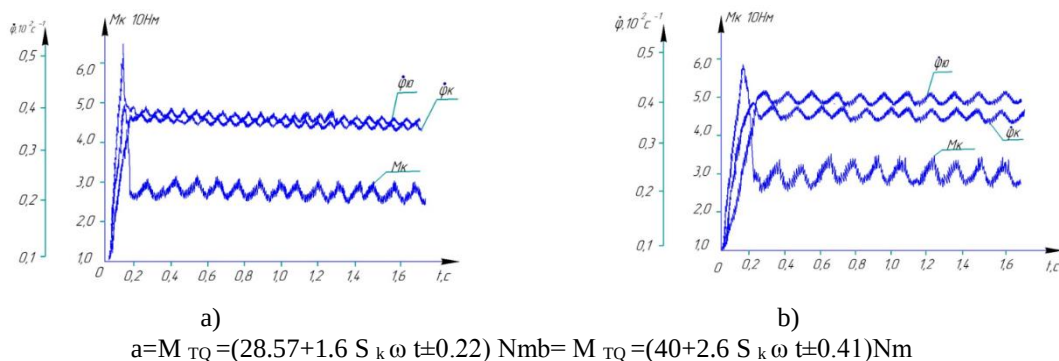


Figure 3. – Patterns of change of angular velocities and torques of a piled drum shaft piled gear

According to the analysis of the obtained laws, the driving shaft and pile set of the cotton cleaner from small impurities oscillate with certain frequencies and amplitudes around the average values of the angular speeds and torques of the screwdriver. [10,11] it can be seen from the obtained results that the random constituent effect is also changed by the law of motion. Angular velocities corresponding to the increase in values of $\dot{\varphi}_{ю}$ productivity, i.e. technological resistance moment

$M_{TK}\dot{\phi}_K$. The mutual spatial displacement of the rubber bladder varies depending on the coefficient of friction.

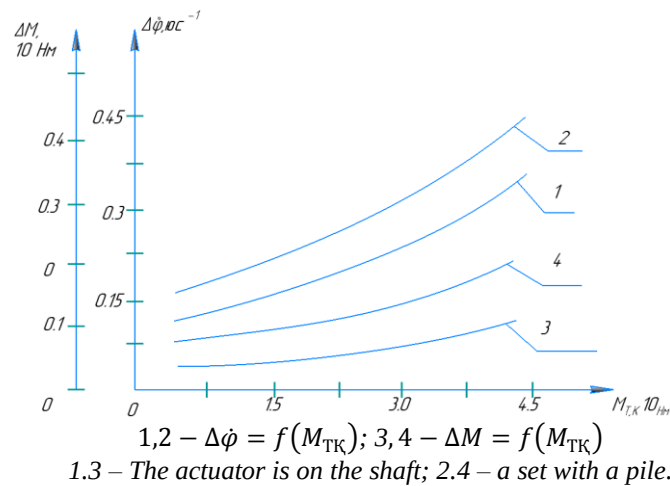


Figure 4. – Shows graphs of angular velocities and torques of the piled drum drive shaft and piled gear as a function of technological resistance

When the constructed graphs are analyzed connections, when the technological resistance of cotton M_{TK} increases from 12 N/m to 45 N/m, the values of the angular velocity of the piled gasket increase from 1.72 C^{-1} to 4.5 C^{-1} in the nonlinear connection, and $\Delta\dot{\phi}_{i0}$ and its values increase from 1.37 C^{-1} to 3.52 C^{-1} in a non-linear manner. It should be emphasized that the angular velocity of the piled gasket is smaller than the angular velocity of the drive shaft by about $(2.1 \div 2.4) \text{ C}^{-1}$, the main reason for this is due to the dissipativeness in the rubber bladder, certain energy is absorbed. [12.13.14], respectively, when the range of torsional torque on the drive shaft increases from 0.63 N/m to 1.5 N/m, the values of ΔM_q increase from 1.1 N/m to 2.45 N/m. That is, the torque range of the bolted gasket is higher because the resistance is directly applied to the bolted gasket. It should be emphasized that $\Delta\dot{\phi}_K$ If the value is too large, it can increase the damage to the fiber seed by increasing the pulse power, therefore, based on the results of experimental studies, it is desirable to have the technological resistance $M_{tq} \leq (35 \pm 38) \text{ N/m}$ to ensure that it is in the range of $\Delta\dot{\phi}_K \leq (2.2 \pm 2.4) \text{ C}^{-1}$ is appropriate.

Composed drum rubber bushing with curved pegs based on the equations circular bikrlık coefficient values 150 Nm/rad to 400 Nm/rad $\Delta\dot{\phi}_K$ values decreased from 3.74 C^{-1} to 0.86 C^{-1} , respectively $\Delta\dot{\phi}_{i0}$ values only decrease from 0.09 C^{-1} to 0.58 C^{-1} . That's why for rotating rubber bushing uniqueness increase with loading increases. Because to the system relatively resistance, that is rubber bush rotation deformation k require more power does In this M_K values from 3.8 N m to 42 N m if increases in nonlinear bonding M_{i0} values from 2.1 N /m to 29.8 N /m increased goes So cotton small from impurities cleaner piled up headset corner speed vibration coverage defined values provide for content curve peg drum rubber bushing circulation corner coefficient values in the range (320÷350) N m/rad to be to the goal is appropriate.

It is rubbery bushing brand 7IRP-13-46 from rubber preparation recommendation will be done

It is known rotary worker organs movement speed one in the plane to be provide for their moments of inertia values is increased. But spend it power increases.

$$\begin{aligned} -\Delta\dot{\phi}_{i0} &= f(c); & 2 - \Delta\dot{\phi}_H &= f(c); \\ 3 - \Delta M_{i0} &= f(c); & 3 - \Delta M_K &= f(c); \end{aligned}$$

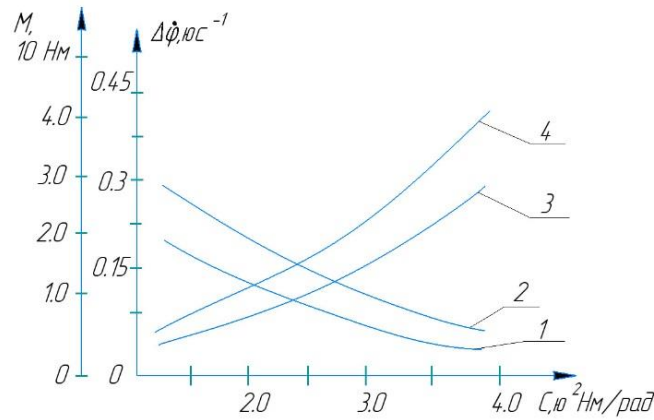


Figure 5. – Shows graphs of dependence of angular velocities, vibration ranges, and torques of the recommended cleaning pile drum drive shaft and pile gasket on the coefficient of rotation of the drum rubber bushing

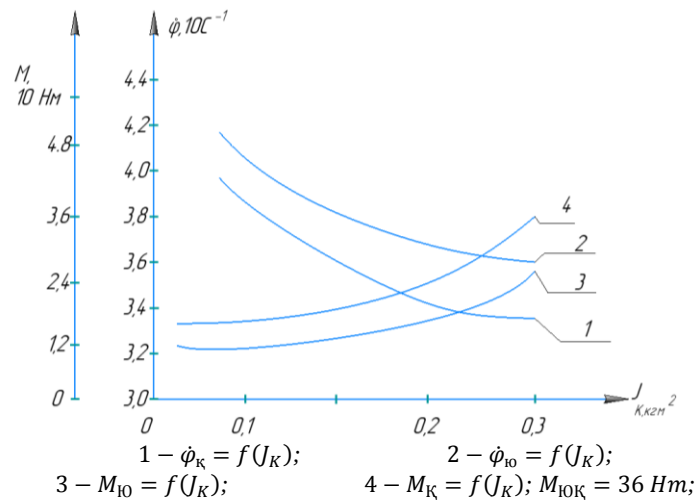


Figure 6. – Graphs of dependence of the angular velocities and loadings of piled drum drive shaft and piled gear on inertia moment of piled gear are given

Is increased from $\dot{\phi}_K 0.07 \text{ kg m}^2$ to 0.31 kg m^2 , the values decrease from 38.2 C^{-1} to 34.5 C^{-1} , and correspondingly, $\dot{\phi}_{10}$ the values decrease from 40.8 C^{-1} to 35.7 C^{-1} in the nonlinear connection. But spend power J_q increase at the expense of increase screw driver torque increases. Including M_K values from 18.6 N m to 36.5 N m increase M_K values from 17.2 N m to 24 N m shot it's boring. Pile drum angle k rate above 40 C^{-1} to be provide moment of inertia for $(2.9 \div 3.2) \text{ kg m}^2$ between choose recommendation will be done.

4. Results of comparative production tests.

A pilot copy of the recommended cotton cleaner was prepared and comparative production tests were conducted. The test results are given in the table. Construction view is shown in figure 6.



Figure 7. – Recommended Curved Pile Drum View

Table 1.

Results of comparative production tests

Demonstrators.	Line 2 of the current UXK unit	Line 1 of the newly modernized UXK unit
Incoming cotton raw material.		
Moisture.	8,5 %	8,5 %
The amount of pollution is massive.	3,0	3,0
Refine draw cotton material.		
Moisture.	8,5 %	8,3 %
The amount of pollution is massive.	0,57	0,40
Carcleaning efficiency.	82%	89,2%
Mechanical damage of the seed.	1,35 %	1,28 %

As a result of the conducted experiments, when the drum structure with curved piles recommended for cleaning cotton was used in the cotton ginning enterprise of Andijan region "VodiySanoatFahri" LLC, compared to the existing structure, the moisture content of cleaned cotton decreased by 0.17%, and the mechanical breakdown of seed decreased by 0.07%. and it was determined that the cleaning efficiency increased by 7.2%.

A number of recommendations are given based on the results of the research. [15,16,17,18,19,20].

5. Conclusions.

Cotton small from the dirty ones of the cleaner curve with a pile rubbery bushing drums Recommended construction done. Pile drum walking mechanism including dynamic analysis of the machine assembly based on peg drum and drive shaft laws and downloads determined . Results analytical based on peg set a and parameters of the rubber bushing recommendation values determined.

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CONSTRUCTING THE DIFFERENTIAL EQUATION OF GYROSTATIC MOTION IN LAGRANGE VARIABLES

Keldiyorova M.A.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T. Urazbaev, Tashkent, Uzbekistan*
E-mail: Keldiyorovamashhuraa@gmail.com

Abstract. *The movement of the body on another surface is supported by everything, including in engineering, transport, and do not lose yourself. He contributes to the development of analytical mechanics, stability derivation, differential equations, differential geometry methods. The study of the motion of a rigid body - a gyrostat rotating with a constant angular velocity around a vertical axis - on an absolute rough surface. His and his differential equations in several variables simplify the equation, verify the existence of recovery and the stability of this and hold.*

Keywords: *gyrostat, motion, equation, rigid body, Lagrange variables, Voronets equation.*

1. Introduction.

Differential equations of gyrostat motion in the research work are formulated in other variables, unlike the following works [1, 2]. To determine the position of the gyrostat, the coordinates of the center of mass and the Euler angles were obtained.

The equation of motion of a gyrostat on a rotating surface. We study the movement of a gyrostat (shell, flywheel) with a geometric and dynamic axis of symmetry under the influence of gravity on a horizontal surface. The horizontal surface rotates around the vertical axis with a constant angle speed. The gyrostat rests on this surface with its non-edge convex surface. Suppose that the surface of the gyrostat trajectories only one point with the horizontal basis surface. The axis of rotation of the flywheel is fixed to the gyrostat shell and has an arbitrary direction. In the following literature [3-6], the differential equations of motion of a rigid body with rotating flywheels are derived. The equation of motion of a gyrostat without a dynamic axis of symmetry on a stationary horizontal surface and the priority of its stable motion are presented in these scientific papers [7,9]. In these scientific works, it was shown that the conditions of asymptotic priority in relation to variables depend on nonholonomic connections and the direction of rotation of the gyrostat shell and the shape of the surface of the shell at the trajectory point. In the following works [1,3,6], the differential equations of motion are formulated using the general theorems of dynamics, and in the works [2], they are formulated in the form of Lagrange, Hamilton and Raus equations.

In this research work, the differential equations of motion are formulated using the Voronets equation for nonholonomic systems. The equations of the priority of stable motion, the center of mass of the gyrostat and the line drawn by the trace of the trajectory point with its surface plane are determined.

2. Study method.

In this research work, the differential equations of motion are formulated using the Voronets equation for nonholonomic systems. The equations of the priority of stable motion, the center of mass of the gyrostat and the line drawn by the trace of the attempt point with its surface plane are determined. The chirostat center-of-mass differential equation is integrated into the add-on Maple analytical software for playing exposure, and graphs for various traction parameters are provided.

3. Research results and discussions.

The equation of motion of a gyrostat on a rotating surface. We study the movement of a gyrostat (shell, flywheel) with a geometric and dynamic axis of symmetry under the influence of gravity on a horizontal surface. The horizontal face rotates around the vertical axis with a constant angular speed Ω . The gyrostat rests on this face with its non-edge convex surface. Suppose that the surface of the gyrostat attempts only one point with the horizontal reference face. The axis of rotation of the flywheel is fixed to the gyrostat shell and has an arbitrary direction. In the following literature [3-6], the differential equations of motion of a rigid body with rotating flywheels are derived. The equation of motion of a gyrostat without a dynamic axis of symmetry on a stationary horizontal surface and the priority of its stable motion are presented in these scientific papers [7, 9].

In these scientific works, it was shown that the conditions of asymptotic priority in relation to variables depend on nonholonomic connections and the direction of rotation of the gyrostat shell and the shape of the surface of the shell at the test point. In the following works [1,3,6], the differential equations of motion are formulated using the general theorems of dynamics, and in the works [2], they are formulated in the form of Lagrange, Hamilton and Raus equations. In this research work, the differential equations of motion are formulated using the Voronets equation for nonholonomic systems. The equations of the priority of stable motion, the center of mass of the gyrostat and the line drawn by the trace of the trajectory point with its surface plane are determined. To derive the gyrostat equation of motion, we introduce a fixed coordinate system $Oxyz$ centered on the axis of rotation, where the Oz axis points up along the vertical axis of rotation and the Oxy axes lie in the horizontal plane. We also introduce a moving coordinate system centered at the center of gravity G and oriented along the gyrostat central principal inertia axes. In this case, the tensor of moments of inertia (quantity that characterizes the distribution of masses in the body) will consist of only diagonal elements, that is, the centrifugal moments of inertia will be zero. Here the axis is oriented along the axis of symmetry of the gyrostat. To determine the position of the gyrostat, we enter the following generalized coordinates. Cartesian coordinates x, y represent the center of gravity of the gyrostat, Euler angles represent the spatial position of the coordinate system fixed by the gyrostat, and the flywheel represents the pure rotation angle around its axis. Gyrostat kinetic and potential energies. Let us assume that the gyrostat is stationary - moving on a plane rotating with a Ω - constant angular speed, and that the flywheel has an axis of pure rotation in an arbitrary direction. As we noted above, let the gyrostat have a dynamic axis of symmetry, and let the flywheel system (gyrostat - shell - flywheel) be located at the center of gravity. Let us denote the mass of the gyrostat by m , A and C are the central equatorial and polar moments of inertia of the gyrostat with the flywheel when it is suspended, and B is the moment of inertia of the flywheel in relation to the axis of pure rotation. We denote by $\omega_\xi, \omega_\eta, \omega_\zeta$, the projections of $O_1\xi\eta\zeta$ in the moving $\omega_x, \omega_y, \omega_z$ coordinate system of the gyrostat, and the projections $Oxyz$ in the fixed coordinate system of the gyrostat. $\delta_1, \delta_2, \delta_3$ - be the direction cosines of the shaft axis of pure rotation in the coordinate system fixed by the gyrostat. Let's calculate the gyrostat kinetic energy T^* using the relative motion dynamics formula [8].

$$T^* = \frac{1}{2} m V_{0_1}^2 + m \begin{vmatrix} \omega_\xi & \omega_\eta & \omega_\zeta \\ \xi_c & \eta_c & \zeta_c \\ V_\xi & V_\eta & V_\zeta \end{vmatrix} + \frac{1}{2} A_1 (\omega_\xi^2 + \omega_\eta^2) + \frac{1}{2} C \omega_\zeta^2 + \frac{1}{2} B \dot{\alpha}^2 + \\ + B \dot{\alpha} (\delta_1 \dot{\theta} + \delta_2 \dot{\psi} \sin \theta + \delta_3 (\dot{\phi} + \dot{\psi} \cos \theta)), \quad (1)$$

and the potential energy is as follows $\Pi = mgf(\theta)$

$$L^* = \frac{1}{2}m(\dot{x}^2 + \dot{y}^2) + \frac{1}{2}(A + mf'^2)\dot{\theta}^2 + \frac{1}{2}C(\dot{\varphi} + \dot{\psi}\cos\theta)^2 + \frac{1}{2}A\sin^2\theta\dot{\psi}^2 + \frac{1}{2}B\dot{\alpha}^2 + B[\delta_1\dot{\theta} + \delta_2\dot{\psi}\sin\theta + \delta_3(\dot{\varphi} + \dot{\psi}\cos\theta)]\dot{\alpha} - mgf(\theta) \quad (2)$$

Nonholonomic coupling equations. At the point where the surface touches the gyrostat, there are two nonholonomic coupling equations:

$$\begin{aligned} \dot{x} &= B_{14}\dot{\phi} + B_{13}\dot{\theta} + B_{15}\dot{\psi} + B_1 \\ \dot{y} &= B_{24}\dot{\phi} + B_{23}\dot{\theta} + B_{25}\dot{\psi} + B_2 \end{aligned} \quad (3)$$

where $B_{\mu\rho}$ is the distance from the center of gravity to the plane of support – z is equal to, $B_{\mu\rho}$ – coefficients of nonholonomic coupling equations.

We write differential equations of gyrostat motion in Voronets form in Lagrangian variables

$$\frac{d}{dt} \frac{\partial L}{\partial \dot{q}_\rho} - \frac{\partial L}{\partial q_\rho} - B_{\eta\rho} \frac{\partial L}{\partial q_\eta} = \frac{\partial T_0}{\partial \dot{q}_\mu} (\Omega_{\mu\beta} \dot{q}_\beta + \omega_{\mu\rho}) + Q_s + B_{\eta s} Q_\eta \quad (4)$$

So, θ, φ, ψ and α systems of second-order equations are formed.. Because the system has four degrees of freedom, there are two nonintegrable nonholonomic coupling equations, but the number of generalized coordinates is six. If we perform calculations and compactions for each term, the exact representation of these differential equations of motion in Lagrange variables will be as follows:

$$\begin{aligned} a_{11}\ddot{\theta} + (-m\chi\chi'\dot{\phi} - m\Omega(\chi'(-y\cos\psi + x\sin\psi) - m\Omega(-y\sin\psi - x\cos\psi)f'(\theta)) + \\ m\chi_1'(\theta)f'(\theta) + m\chi_1(\theta)f''(\theta) + 1/2 C \sin\theta))\dot{\phi} + mff'\Omega - m(f'\rho + \\ + \Omega(-\rho(y\cos\psi - x\sin\psi)))\dot{\psi} - m(\chi\rho - f'\chi')\dot{\phi} - m\Omega(y\cos\psi - x\sin\psi)f'' + \\ + 1/2 \dot{\theta}^2 \frac{\partial a_{11}}{\partial \theta} + B\delta_1\ddot{\alpha} - mgf' - m\Omega^2 f(x\sin\psi - y\cos\psi) = Q\delta_1 \end{aligned} \quad (5)$$

$$\begin{aligned} a_{22}\ddot{\phi} + a_{23}\ddot{\psi} + (A\sin\theta\cos\theta - C\cos\theta\sin\theta + mf'(\theta)f''(\theta))\dot{\phi} - \\ - (m\chi_1'(\theta)f'(\theta) + m\chi_1(\theta)f''(\theta) + 1/2 C \sin\theta)\dot{\psi} + m(f\chi_1 + \chi'f'\cos 2\psi + \\ + (m\Omega\chi(y\sin\psi - x\cos\psi) + m\Omega f'\chi\sin 2\psi - m\Omega\chi_1(-y\sin\psi - x\cos\psi))\dot{\psi} - \\ - m\chi'\Omega(y\cos\psi + x\sin\psi) - m\chi_1\chi'\cos 2\psi)\dot{\theta} - m\Omega^2\chi(x\cos\psi - y\sin\psi) - \\ - m\chi'f\sin 2\psi\dot{\theta}^2 + B\delta_3\ddot{\alpha} = Q\delta_3 \end{aligned} \quad (6)$$

$$\begin{aligned} a_{33}\ddot{\psi} + a_{23}\ddot{\phi} + (m\Omega(y\cos\psi - x\sin\psi)f'' + m\chi_1(\theta)\dot{\psi} - \\ - (m\chi_1'(\theta)f'(\theta) + m\chi_1(\theta)f''(\theta) + 1/2 C \sin\theta - mf\chi_1 - \\ - (m\Omega\chi(y\sin\psi - x\cos\psi) - m\Omega f'\chi\sin 2\psi - m\Omega\chi_1(y\sin\psi + x\cos\psi))\dot{\phi} + \\ + m\Omega(f' + f'')(y\cos\psi + x\sin\psi) - mf'(f' + f'')\cos 2\psi)\dot{\theta} - \\ - mf'(f' + f'')\cos 2\psi\dot{\phi}\dot{\psi} + B\dot{\alpha}(\delta_2\cos\theta - \delta_3\sin\theta) - m\Omega f(-y\cos\psi + x\sin\psi) - \\ - m\Omega ff' + B\ddot{\alpha}(\delta_2\sin\theta + \delta_3\cos\theta) + mf(f' + f'')\sin 2\psi\dot{\theta}^2 + \\ + m\Omega^2 f'(x\cos\psi + y\sin\psi) = Q(\delta_2\sin\theta + \delta_3\cos\theta) \end{aligned} \quad (7)$$

$$B[\ddot{\alpha} + \ddot{\theta}\delta_1 + (\sin\theta \cdot \delta_2 + \cos\theta \cdot \delta_3)\ddot{\psi} + \delta_3\ddot{\phi} + (\cos\theta \cdot \delta_2 - \sin\theta \cdot \delta_3)\dot{\psi}\dot{\theta}] = Q \quad (8)$$

This system of equations, together with the coupling equations, forms a closed system of equations and accurately represents the position or motion of the gyrostat on a surface rotating with a constant angular velocity.

Trajectories drawn by the center of mass of the gyrostat in different parameters. In this case, trajectories consist of spreading spirals, circular oscillating motion graphs.

$$\begin{aligned} \frac{\ddot{x}(t)}{\ddot{a}t} &= -5y(t) + \cos(5t), & \frac{\ddot{y}(t)}{\ddot{a}t} &= 5x(t) + \sin(5t); \\ \frac{\ddot{y}(s)}{\ddot{a}t} &= 5x(s) + \sin(5s), & \frac{\ddot{x}(s)}{\ddot{a}t} &= 1, & \frac{\ddot{x}(s)}{\ddot{a}t} &= -5y(s) + \cos(5s). \end{aligned} \quad (9)$$

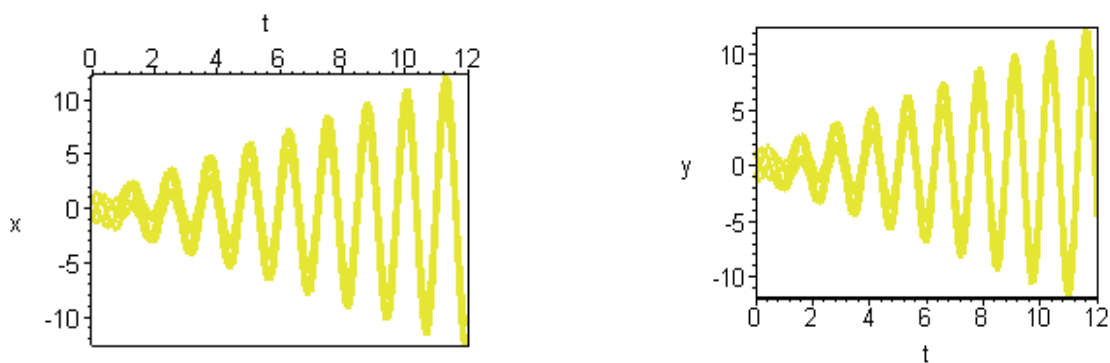


Figure 1. – Trajectories drawn by the center of mass of the gyrostatt in different parameters

4. Conclusions.

In this research work, the motion of a gyrostatt moving on a horizontal plane rotating with a constant angular velocity is studied. It is assumed that the gyrostatt and the plane have a single contact point, no slip.

In that case, a nonholonomic coupling equation is formed from the fact that the velocity of the instantaneous center of velocity is equal to zero. Based on it, differential equations of non-holonomic mechanical motion, especially Voronets, were studied.

The following results were obtained in this work:

- nonholonomic coupling and differential equation of motion for a body (gyrostatt) with arbitrary convex surface, dynamic and geometric symmetry axis is formulated;
- using the Voronets equation, the conditions for the presence of stationary motion are defined in both Lagrange and Raus variables;
- gyrostatt center of mass motion trajectory equations are defined;
- the differential equation of induced motion is formulated and the first approximation terms are extracted;
- priority conditions are determined using Lyapunov's first approximation theorem;
- the differential equation of the center of mass of the gyrostatt is integrated using the Maple analytical calculation program in the EC dialog box, and graphs of movement trajectories for various parameters are presented.

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FRICITION IN ROLL MODULES OF TECHNOLOGICAL MACHINES

Khurramov Sh.R¹, Abdukarimov A².

¹Tashkent University of Architecture and Civil Engineering, Tashkent, Uzbekistan

²Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan

E-mail: shavkat-xurramov59@mail.ru

Abstract. The study is devoted to solving the problem of determining the patterns of friction stress distribution in the roll module contact zone of technological machines. A mathematical model was developed that describes the patterns of friction stress distribution in the contact zone of roll modules. It was established that the ratio of friction stress to normal stress in the contact zone of roll modules is not constant and changes at each point of the contact zone. It was revealed that the ratio of friction stress to normal stress depends on external forces acting on the roll supports, geometric and kinematic parameters of the roll module, and deformation properties of the strip and roller coating.

Keywords: roller machines, two-roller modules, contact stresses, friction coefficient, friction stress model.

1. Introduction.

The technology of processing materials using roller machines has become widespread in various industries.

The basic element of roller machines is a roll module formed by two rollers and the material being processed.

Technological processes in roll modules are conducted as a result of the contact interaction of the processed material (a strip) with the rollers. During the interaction in the roll modules, friction forces arise in the contact zone between the strip and the rollers. The magnitude of friction forces in roll modules has a great influence on the energy-power and kinematic parameters of the technological process, determines the permissible angle of nip of the strip and the advance of the strip, that is, plays an extremely important role [1].

Quantitative measures of friction force are function $f(x)$ that relates friction stress $t(x)$ to normal stress $n(x)$ distributed in the contact zone (in the zone of strip deformation), according to the following formula [2]:

$$t(x) = f(x)n(x) \quad (1)$$

and the proportionality coefficient f connecting the friction force and the normal force by the Amonton-Coulomb law according to the following formula [2]:

$$T = fN. \quad (2)$$

Function $f(x)$ is called the “Friction Stress Model”, and the proportionality coefficient f is known as the coefficient of sliding friction (or coefficient of friction) [2].

The friction coefficient is a certain dimensionless physical constant, the value of which is obtained only experimentally, by direct measurement of forces T and N , and by determining these forces as a quotient:

$$f = \frac{T}{N}. \quad (3)$$

The dependence is quite correct [2], since it follows from the Amonton-Coulomb law; forces T and N are integral characteristics of the process. However, the use of coefficient f instead of function $f(x)$ is theoretically unjustified and has not been experimentally confirmed and, naturally, the following condition

$$t(x) = fn(x) \quad (4)$$

is neither a model of friction stress nor a law of friction [3-8]. For this reason, theoretical solutions obtained using condition (4) are approximate. Consequently, they do not provide high accuracy and reliability in predicting the parameters of the roll module [9].

An accurate theoretical determination of the friction force is possible only if there is accurate data on the value of the friction coefficient and a correct mathematical model of friction stresses in roll modules.

When theoretically determining friction stresses in the contact zone, at least the influence of the friction coefficient, normal contact stresses, and sliding velocity must be considered.

Taking into account the above, for the theoretical determination of friction stresses in metal rolling, a friction stress model was proposed in the following form [10]:

for the lag zone:

$$t_{0x} = -fn_{0x} \frac{h_0}{h_0 - h_\gamma} \left(\frac{h_\gamma}{h_x} - 1 \right); \quad (5)$$

for the advance zone:

$$t_{1x} = -fn_{1x} \frac{h_1}{h_\gamma - h_1} \left(\frac{h_\gamma}{h_x} - 1 \right), \quad (6)$$

where $t_{0x}, n_{0x}, t_{1x}, n_{1x}$ – are the values of friction stress and normal contact stress in the contact zone section under consideration, in the lag and advance zones, respectively; h_x, h_γ – are the values of strip thickness in the considered and neutral sections of the contact zone; h_0, h_1 – are the values of the strip thickness in the sections of the entrance and exit from the contact zone, respectively; f – are the values of the friction coefficient.

In roll modules of technological machines, one or both rollers have an elastic coating. In roll modules, the contact interaction between the strip and the roller coating occurs along the roll contact curves. In this case, the pattern of the friction stress distribution depends on the shape of the roll contact curves [11-14].

Works [15-16] are devoted to the study of friction stresses in roller modules of technological machines. However, the results of these works do not reveal the patterns of friction stress distribution.

The study aims to develop a friction stress model for roll modules of technological machines, considering the kinematics of the contact zone and the shape of the roll contact curves, which will improve the accuracy of friction determination.

The research is devoted to solving the problem of determining the patterns of friction stress distribution in the contact zone of a symmetrical roll module, consisting of a processed material with a thickness of δ_1 , two drive rolls with radii R and elastic coatings with a thickness of H (Fig. 1).

2. Materials and methods.

Since we are considering a symmetrical roll module, we will investigate the friction stresses of the top roller.

In a roll module under static conditions, the point of maximum compression of the strip is located on the line of centers (on the line connecting the centers of the rolls). Under dynamic conditions, due to the action of reactive forces, this point shifts from the line of centers to the beginning of the contact zone. It was revealed that this displacement in the roll modules of technological machines is insignificant, so we can assume that the point of maximum compression of the strip lies on the line of centers [17].

Based on this, we divide the contact zone relative to the line of centers into zones of compression and recovery (deformation). Suppose that in the contact zone, according to the kinematics, there are lag and advance slip zones, separated by a neutral point. In a two-roll module, the neutral point is located in the compression zone [17]. Then the contact zone consists of compression sections with lagging slip 1, compression sections with advance slip 2, and recovery zone with advance slip 3.

We solve the problem posed in polar coordinates.

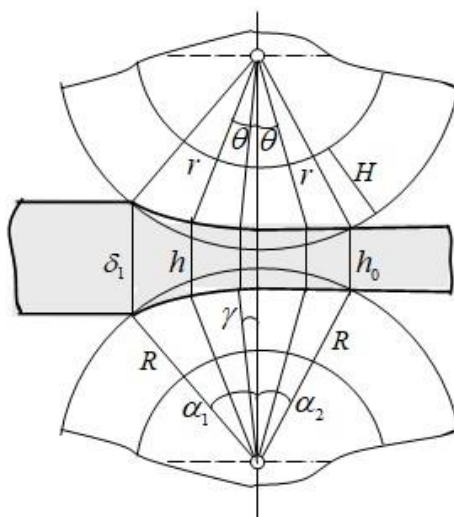


Figure1. – Scheme of a roll module of technological machines

Let the roll contact curve be given by the following equation:

$$r = r(\theta_i), i = \overline{1,3}, -\varphi_1 \leq \theta_1 \leq -\gamma, -\gamma \leq \theta_2 \leq 0, 0 \leq \theta_3 \leq \varphi_2,$$

where i – is the index for the section number, φ_1, φ_2 – are the contact angles, γ – is the neutral angle (in the zone of deformation of the strip).

The shape of the roll contact curve is determined primarily by the deformation properties of the strip materials and the material being processed, which are characterized by the compression line during interaction and the line determining the recovery of its deformation. These lines define the relationship between the deformation of the material and the stress under compression and recovery.

Materials such as fabrics, paper, leather, wool, cotton, and others are processed in roll modules; rubber and polyurethanes of various brands, technical cloth made from felt of various origins are used to coat the rollers. In the technical literature and articles written by researchers involved in establishing the nature of deformation of such materials, a power-law relationship between stress and strain under compression and recovery is mainly given [17-19].

In [17], mathematical models of the shape of roll contact curves were obtained when the deformation properties of materials of the strip and roller coating are specified by power-law relationships between strain and stress under compression and recovery in the following form:

for the compression zone:

$$r(\theta) = \frac{R}{1+k_1} \left(1 + k_1 \frac{\cos \varphi_1}{\cos \theta} \right), -\varphi_1 \leq \theta \leq 0, \quad (7)$$

for the advance zone:

$$r(\theta) = \frac{R}{1+k_2} \left(1 + k_2 \frac{\cos \varphi_2}{\cos \theta} \right), 0 \leq \theta \leq \varphi_2, \quad (8)$$

where

$$k_1 = \frac{2H}{\delta_1} m_1, k_2 = \frac{2H}{\delta_2} m_2, \quad (9)$$

here δ_2 – is the thickness of the strip when it exits the contact zone; m_1, m_2 – are the indicators that determine the ratio of the strain rates of the strip materials and the roller coating, respectively, under compression and recovery.

3. Results.

We rewrite equalities (5) in polar coordinates:

$$t(\theta) = -f n(\theta) \frac{h(-\varphi_1)}{h(-\varphi_1) - h(-\gamma)} \left(\frac{h(-\gamma)}{h(\theta)} - 1 \right).$$

Then for section 1 of the roll contact curve, we have

$$t(\theta) = -fn(\theta) \frac{\delta_1}{\delta_1 - h(-\gamma)} \left(\frac{h(-\gamma) - h(\theta)}{h(\theta)} \right), -\varphi_1 \leq \theta \leq -\gamma. \quad (10)$$

From Figure 1, it follows that

$$h(\theta) = \delta_1 - 2(r(\theta) \cos \theta - R \cos \varphi_1), -\varphi_1 \leq \theta \leq -\gamma.$$

Then, considering equality (7), we obtain

$$h(\theta) = \frac{1}{1+k_1} (\delta_1(1+k_1) - 2R(\cos \theta - \cos \varphi_1))$$

or in the first approximation

$$h(\theta) = \frac{1}{1+k_1} (\delta_1(1+k_1) - R(\varphi_1^2 - \theta^2)). \quad (11)$$

From here we find

$$h(-\gamma) = \frac{1}{1+k_1} (\delta_1(1+k_1) - R(\varphi_1^2 - \gamma^2)). \quad (12)$$

Substituting equalities (11) and (12) into equalities (10), after transform, we find

$$t(\theta) = fn(\theta) \frac{\delta_1(1+k_1)(\theta^2 - \gamma^2)}{(\varphi_1^2 - \gamma^2)(\delta_1(1+k_1) - R(\varphi_1^2 - \theta^2))}, -\varphi_1 \leq \theta \leq -\gamma. \quad (13)$$

Formula (13) describes the law of friction stress distribution in section 1 of the contact zone. In section 2, the phenomena of compression and advance slip of the strip occur.

Therefore, first, from dependence (6), we obtain:

$$t(\theta) = -f_1 n(\theta) \frac{h(0)}{h(-\gamma) - h(0)} \left(\frac{h(-\gamma) - h(\theta)}{h(\theta)} \right), -\gamma \leq \theta \leq 0, \quad (14)$$

where $f_1 = -f(0)$ ($0 \leq f_1 \leq f$).

From equality (10), we have

$$h(\theta) = \frac{1}{1+k_1} (\delta_1(1+k_1) - R(\varphi_1^2 - \theta^2)), -\gamma \leq \theta \leq 0, \quad (15)$$

$$h(0) = \frac{1}{1+k_1} (\delta_1(1+k_1) - R\varphi_1^2). \quad (16)$$

Substituting equalities (12), (15), and (16) into equalities (14), after transform, we find

$$t(\theta) = -f_1 n(\theta) \frac{(\delta_1(1+k_1) - R\varphi_1^2)(\gamma^2 - \theta^2)}{\gamma^2(\delta_1(1+k_1) - R(\varphi_1^2 - \theta^2))}, -\gamma \leq \theta \leq 0. \quad (17)$$

Formula (17) describes the law of friction stress distribution in section 2 of the contact curve zone.

In section 3, the phenomena of compression and advance slip of the strip occur. Therefore, first, from dependence (6), we obtain:

$$t(\theta) = -n(\theta) \left(f_1 + (f - f_1) \frac{\delta_2}{h(0) - \delta_2} \left(\frac{h(0) - h(\theta)}{h(\theta)} \right) \right), 0 \leq \theta \leq \varphi_2. \quad (18)$$

By analogy with equality (11), we obtain:

$$h(\theta) = \frac{1}{1+k_2} (\delta_2(1+k_2) - R(\varphi_2^2 - \theta^2)), 0 \leq \theta \leq \varphi_2, \quad (19)$$

$$h(0) = \frac{1}{1+k_2} (\delta_2(1+k_2) - R\varphi_2^2). \quad (20)$$

Substituting equalities (19) and (20) into equalities (18), after transform, we find

$$t(\theta) = -n(\theta) \left(f_1 + (f - f_1) \frac{\delta_2(1+k_2)\theta^2}{\varphi_2^2(\delta_2(1+k_2) - R(\varphi_2^2 - \theta^2))} \right), 0 \leq \theta \leq \varphi_2. \quad (21)$$

Formula (21) describes the law of friction stress distribution in section 3 of the contact zone.

Thus, a mathematical model was obtained that describes the patterns of friction stress distribution in the contact zone of roll modules of technological machines.

In formulas (14), (17) and (21), stresses $t(\theta)$ and $n(\theta)$ reflect the stresses acting on the strip at the points of the contact zone. At these points of the contact zone, there are also stresses $t_1(\theta)$ and $n_1(\theta)$, acting on the roller coating. According to Newton's law, stresses $t(\theta)$, $n(\theta)$ and $t_1(\theta)$, $n_1(\theta)$ are equal in magnitude and opposite in direction.

Previously, a mathematical model was obtained that describes the patterns of friction stress distribution $t_1(\theta)$ in the contact zone of roll modules of technological machines, which has the following form [20]:

$$t_1(\theta) = n_1(\theta) \begin{cases} tg(\theta - \psi_1(\theta) + \xi), & -\varphi_1 \leq \theta \leq 0, \\ tg(\theta - \psi_2(\theta) + \xi), & 0 \leq \theta \leq \varphi_2, \end{cases} \quad (22)$$

where $\psi_i(\theta) = \arctg \frac{r'_i(\theta)}{r_i(\theta)}$, $\xi = \arctg \frac{F}{Q}$, F – is the horizontal reaction of the roll supports, Q – is the pressure force of the roll clamping device.

Considering equalities (7) and (8), we find

$$tg \psi_i = \frac{k_i \cos \varphi_i}{\cos \theta + k_i \cos \varphi_i} tg \theta. \quad (23)$$

At the neutral point, we have $t_1(-\gamma) = 0$, therefore, $tg(-\gamma - \psi_1(-\gamma) + \xi) = 0$ or $tg(-\gamma - \psi_1(-\gamma)) + tg \xi = 0$, since the values of $(-\gamma - \psi_1(-\gamma))$ and ξ are close to zero.

Then, considering equality (23) and assuming that $\sin \gamma \approx \gamma$, $\cos \gamma \approx 1$, we obtain:

$$-\gamma + \frac{k_1 \cos \varphi_1}{1 + k_1 \cos \varphi_1} \gamma + \xi = 0. \quad (24)$$

From here we find $\gamma = (1 + k_1 \cos \varphi_1) tg \xi$ or

$$\gamma = (1 + k_1 \cos \varphi_1) \frac{F}{Q}. \quad (25)$$

From equalities (21) and (22), we have

$$f_1 = \frac{t(0)}{n(0)} = \frac{t_1(0)}{n_1(0)}$$

or considering equality (23)

$$f_1 = \frac{F}{Q}. \quad (26)$$

Analysis of formulas (14), (17) and (21) shows that the ratio of friction stress to normal stress depends on external forces acting on the roll supports, geometric and kinematic parameters of the roll module, and deformation properties of the strip and roller coating.

4. Conclusions.

A mathematical model was developed that describes the patterns of friction stress distribution in the contact zone of roll modules of technological machines.

It was determined that the ratio of friction stress to normal stress in the contact zone of roll modules of technological machines is not constant, and changes at each point of the contact zone.

It was revealed that the ratio of friction stress to normal stress depends on external forces acting on the roll supports, geometric and kinematic parameters of the roll module, and deformation properties of the strip and roller coating

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STUDYING THE MOVEMENT OF COTTON RAW MATERIAL IN THE WORKING CHAMBER OF THE STONE CAP

Kosimov Kh.Kh., Muradov R.M.

Namangan Institute of Textile Industry, Namangan, Uzbekistan

E-mail: xusanboy_8407@mail.ru

Abstract: *Theoretical study of the process of operation of ginneries in the development of high-efficiency structures of the ginneries, which is a key element of the device for air transport of raw cotton. Based on the results-based recommendations, it is possible to fully retain the heavy impurities in the cotton and to preserve the natural properties of the fiber and seeds. Heavy contaminants fall into the pocket without reaching the mesh surface of the working chamber. Fine mixes hit the working surface of the net. After the hit, the back will move down towards the pocket. The retraction coefficient of the mesh curved plate or the elastic property of the plate material have a large effect on the impact process. After separating the cotton piece from the return curved mesh plate, i.e. the movement after the impact process, a differential equation was constructed.*

Keywords: *cotton, seeds, plate, heavy mixes, working chamber, grid, stone, elastic, drum, feather.*

1. Introduction. The composition of cotton has a significant impact on the efficiency of the equipment installed in the technological process of ginneries, their continuous operation. Heavy contaminants in cotton can damage the working parts of cleaning machines and the teeth of gin and linter saws during its processing. Such a change results in damage to the seed and fibers during the separation of the fiber from the seed in the gin machine.

In cotton ginning machines, the cleaning efficiency is reduced, and more cotton particles are added to the waste. Heavy contaminants can also cause a fire to break out as a result of the impact of the processing machine on the metal working bodies. Therefore, the fact that heavy mixtures fall into the working chambers of cotton processing machines has always been in the focus of scientists and experts in the field, and they have been looking for ways to keep heavy mixtures completely in the air-carrying device [1].

Consider current trends, innovations, and methods in the field of design and modeling of drive systems used in technology, in various machines and industries.

The study in [2] is devoted to developing a new compact drive with adjustable stiffness. The transmission precision of this drive is based on a planetary gear train with a rocker link. A prototype of the drive was built and its performance and the algorithm for identifying stiffness were experimentally substantiated.

In [3], a new type of drive was developed that increases the energy efficiency of a technical system due to torque variability and gravity compensation. This drive is effective when used in the design of robotic systems.

In [4], the authors of the article solved the problem of the permissible six-dimensional position of the working body of serial mechanisms with low mobility using the inverse kinematic modeling method.

The authors of reference [5] developed an experimental method for analyzing the vibration of systems with straight bevel gears. Using the developed analytical method, cracks in the teeth of bevel gears were studied.

In [6], the durability of plastic gears of vehicles was investigated and analyzed based on the ISO standards. To determine the resonant frequency, plastic gears were vibration tested. Based on the research results, specific recommendations were issued to improve the durability of vehicle drives.

Reference [7] studied the dynamics of a planetary bearing of the separator based on a complex model that considers gearing and gear contacts. It was determined that the centrifugal force of roller rotation, which causes impact force, has a greater influence on the separator.

In [8], the balancing of the gear mechanism of a cardan drive was considered and studied. Theoretically, it was determined that the mechanism could be balanced using springs. Robotic manipulators were experimentally tested on a prototype of the developed balancer with one degree of freedom.

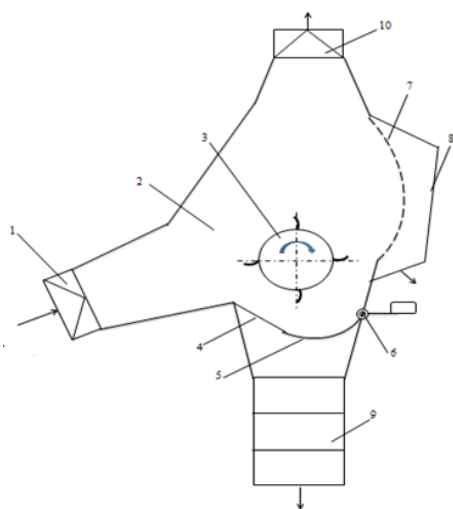
The study in [9] examined gear contacts considering their transmission efficiency. For this purpose, a method for treating the surface of gears was developed. The research results showed improved gear performance achieved through flow machining of gear surfaces.

In [10], the dynamic performance of planetary gears was studied. For this purpose, a matrix method for searching for planetary gear configurations was developed. As a result of applying this method, it became possible to ensure the required gear ratios of the planetary mechanism.

The authors of article [11] proposed a new graphical method for analyzing gear transmission to construct a geometric model and determine the rotation velocity of planetary drive elements. This graphical method simplifies the design stages of planetary gears. Based on the analysis of the reviewed recent publications [2–11], it was determined that when designing and calculating a differential transmission mechanism for a roller technological machine, it is necessary to consider gear ratios, torque of the working parts, balance of the generated power, losses in gear engagement, and transmission efficiency.

2. Materials, methods, and object of study.

Linear grinders are the first device that affects the technological and quality characteristics of the transported cotton, and serve to clean the cotton from foreign bodies, as well as heavy mixtures during air transportation. Our research has shown that the raw cotton hits the walls of the liner with a certain force. These bumps lead to fiber defects and a decrease in seed quality (Fig. 1) [12–15].



1-inlet pipe; 2- working cameras; 3- elastic-coated blade drum; 4-router; 5-segment shaped plate; 6-hinged mechanism; 7-elastic mesh surface; 8.9-pocket; 10- outlet pipe

Figure 1. – Stone device of an improved new design

To reduce the impact force, a device with a mesh curved plate that receives and softens the impact force is installed in the working chamber of the stone (Fig. 2) [16–18].

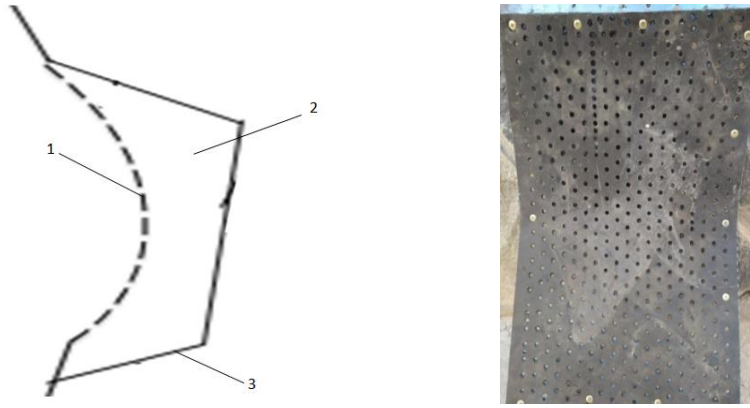
Theoretical bases were created by studying the condition of the particles in the crushing chamber and determining some structural and technological parameters of the device [19–22].

We divide the process of beating a piece of cotton into two stages. The first stage continues until the velocities of the hitting objects equalize [23–27]. Both bodies have the same U -velocity. Then, the process moves to the second stage, which indicates the end of the stroke. In this case, the segment V_1 acquires a velocity. The reflective mesh surface acquires a velocity of V_2 . According to the law of conservation of momentum.

$$m_1 V_1 - m_1 U = m_2 V_2 - m_2 U \quad (1)$$

Here: V_1, V_2 - the speed of the cotton ball and the reflector after the impact; m_1 - piece weight; m_2 - the mass of the mesh plate reflector of the object to be struck; S_1 and S_2 the impulse reaction of bodies during loading.

$$U = \frac{m_1 V_1 + m_2 V_2}{m_1 + m_2} \quad (2)$$



1-mesh surface; 2- pocket; 3-lid
Figure 2. – Various surfaces mounted on an elastic base in a clamping device

It is $V_1 = U = U_H \cos \alpha_0 \cdot \cos \varphi_0$ known from the initial conditions that for the impact process, the equation forms a value perpendicular to the surface of the body forming the velocity of the body being struck. In this case ($V_2 = 0$) has no return speed. Considering all the steps:

$$U = \frac{m_1 \cdot U_H \cdot \cos \alpha \cdot \cos \varphi_0}{m_1 + m_2} \quad (3)$$

Let us consider the 2nd stage of the collision process. It is known that during a shock, the force pulse changes to a magnitude n , which is called the recovery coefficient. We write the law of conservation of momentum separately for each body [28–30].

$$S_{11} = -nS_1, \quad S_{12} = -nS_2 \quad (4)$$

Here: S_{11} and S_{12} are the impact reaction momentum of the bodies during loading (in stage 2 of the impact process), the n -impact coefficient.

Substituting the actions of the pulses, we obtain:

$$\begin{cases} m_1 U_1 = m_1 U = -n(m_1 V_1 - m_1 U) \\ m_2 U_2 = m_2 U = -n(m_2 V_2 - m_2 U) \end{cases} \quad (5)$$

$V_2 = 0$ We form the final equation taking into account:

$$\begin{cases} U_1 = (1+n)U - n \cdot V_1 \\ U_2 = (1+n)U \end{cases} \quad (6)$$

Substituting operations U and V_1 , we obtain the following from the last equation:

$$\begin{aligned} U_1 &= (1+n) \frac{m_1 U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0}{m_1 + m_2} - n U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 = \\ &= U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 \left[\frac{(1+n) \cdot m_1}{m_1 + m_2} - n \right] = \frac{m_1 - n \cdot m_2}{m_1 + m_2} \cdot U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 \\ U_2 &= (1+n) \frac{m_1 U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0}{m_1 + m_2} = \frac{m_1(1+n)}{m_1 + m_2} \cdot U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 \end{aligned} \quad (7)$$

Thus, the return velocities of the fragments form the final equation as follows:

$$\begin{aligned} U_1 &= \frac{m_1 - n \cdot m_2}{m_1 + m_2} U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 \\ U_2 &= \frac{m_1(1+n)}{m_1 + m_2} U_H \cdot \cos \alpha_0 \cdot \cos \varphi_0 \end{aligned} \quad (8)$$

It can be seen from the equations that the cotton piece has a low velocity because the return mass exceeds the piece mass. In this case, the higher the initial velocity U_H , the smaller the angle α_0 of inclination φ_0 of the inlet pipe and the smaller the return, the higher the velocity of the post-impact cotton piece. The maximum return speed after the stroke is taken $\varphi_0 = 0$, when the speed of the cotton

swab corresponds to the normal direction of the mesh surface at the point of impact [2, 3].

3. Results.

The differential equation for the motion of a piece of cotton after it has been separated from the reciprocating curved mesh plate, i.e., after the impact process.

According to Newton's second law, the system of differential equations of motion of a piece of cotton and additional heavy mixtures, separated from the reciprocating curved mesh plate, is written as follows.

The differential equation for the motion of a piece of cotton after it has been separated from the reciprocating curved mesh plate, i.e., after the impact process.

$$\begin{cases} m_1 \dot{V}_x + kV_x + k_x(U_{x0} - V_x)^2 = 0 \\ m_1 \dot{V}_y + kV_y - k_y(U_{y0} - V_y)^2 = m_1 g \end{cases} \quad (9)$$

$$\begin{cases} m_{1t} \dot{V}_{xt} + k_{xt}(U_{x0} - V_{xt})^2 = 0 \\ m_{1t} \dot{V}_{yt} - k_{yt}(U_{y0} - V_{yt})^2 = -m_{1t}g \end{cases}$$

where: m_1 , m_{1t} - mass of cotton and heavy mixtures, respectively, k - coefficient of elasticity of the cotton piece, k_x, k_y - coefficient of aerodynamic resistance of air acting on the cotton piece horizontally and vertically, k_{xt}, k_{yt} - aerodynamics of air acting on heavy mixtures horizontally and vertically, respectively resistance coefficient.

The initial conditions of the system of differential equations:

$$V_x(0) = U_{x0}; V_y(0) = U_{y0}; V_{xt}(0) = U_{xt0}; V_{yt}(0) = U_{yt0}. \quad (10)$$

Since the movement of a piece of cotton after separation from the return curved mesh plate is nonlinear, the system of differential equations is solved in the Maple-17 program in a numerical way.

The results obtained under the initial conditions of the system of differential equations are shown in Figure 2.

As can be seen from the graphs, the cotton and heavy mixtures that entered the stoning chamber continued to move upwards and the cotton particles continued to move upwards and the heavy mixtures downwards into the pocket. It is known that heavy mixtures fall into the pocket without reaching the working surface of the working chamber.

At a small value of the velocity return coefficient $-n$ of the lattice curve, the impact process is soft, while at a large value it is sharp.

4. Discussion. The physical and mechanical parameters of the mixtures in the movement of cotton and heavy-duty mechanical system stone in the working chamber were determined.

After separating the cotton piece from the return curved mesh plate, i.e. the movement after the impact process, a differential equation was constructed.

5. Conclusion. As can be seen from the graphs, the cotton and heavy mixtures that enter the stoning chamber continue to move upwards and the cotton particles continue to move upwards and the heavy mixtures downwards into the pocket.

Heavy contaminants fall into the pocket without reaching the mesh surface of the working chamber. Fine mixes hit the working surface of the net. After the hit, the back will move down towards the pocket.

The retraction coefficient of the mesh curved plate or the elastic property of the plate material have a large effect on the impact process.

At a small value of the velocity return coefficient $-n$ of the lattice curve plate, the impact process is soft, while on a large value it is sharp.

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METHODS FOR PARAMETRIC RELEASE OF MECHANICAL SYSTEMS CONTAINING SERVO-CONSTRAINTS

Khusanov K.¹

¹National Research University-Tashkent Institute of Irrigation and Agricultural Mechanization Engineers, Tashkent, Uzbekistan

E-mail: xusanov1954@gmail.com

Abstract. The derivation of a differential equation of motion for a system with non-ideal holonomic servo-constraints is examined in this article, considering the parametric release from servo-constraints. For such a system, extended equations of motion were obtained, which contain normal and shear components of the servo-constraint reactions. As an example, we consider the problem of a gyroscope in a gimbal with holonomic releasing servo-constraints. A technique is proposed for determining the reaction forces of servo-constraints, which ensures stabilization of motion relative to the manifold defined by holonomic servo-constraints.

Keywords: servo-constraint, parametric release, shear, non-ideal, gyroscope, control.

1. Introduction.

N.G. Chetaev [1] and A.G. Azizov [2,3] developed a method for the parametric release of mechanical systems considering the presence of servo-constraints. Releasing the system means making changes that do not limit the variety of its states while making each state more independent. This process expands the range of available accelerations for the system during real motion. In this case, all possible states and accelerations acceptable for a free system are also considered acceptable for a released one.

Research conducted by A.G. Azizov is devoted to developing the methods for the analytical dynamics of systems including servo-constraints. His works proposed a constructive approach to determining reaction forces in servo constraints, considering possible inconsistencies and allowing the recovery of broken constraints. To study such systems, he applied the theory of parametric release, developed by N.G. Chetaev and V.I. Kirgetov [4,5] for a certain class of mechanical systems.

In a series of publications [6-12], mechanical systems were studied that were subject to various forces, including nonlinear positional, potential, and corrected dissipative and gyroscopic forces. In these studies, conditions under which the rate of change of the parameter ensures the stability of the equilibrium position for both linear and highly nonlinear systems were identified; here dissipative forces were described by the Rayleigh function. It was demonstrated that for any values of the system parameters (for example, the mass of the loads, the stiffness of the springs), it is possible to achieve stability of the equilibrium position by adding an auxiliary load with a damper through a nonlinear spring. The proposed approach to the analysis of stability and stabilization of mechanical systems with nonlinear positional forces is based on the decomposition of equations of disturbed motion into two isolated subsystems. Conditions were also established for stabilizing systems with nonlinear nonstationary potential forces by applying forces of different structures.

Research conducted in [13,14] examines the dynamics of systems with Béguin servo constraints in two versions: when the constraints are implemented through controlled forces and when they are implemented by controlling the inertial properties of the system.

The dynamics of such systems is a richer and more diverse field compared to the classical dynamics of holonomic and non-holonomic systems.

2. Materials, methods, and object of study.

To derive a differential equation of motion of a mechanical system with non-ideal holonomic servo-constraints considering parametric release, we take servo-constraints as invariant relations arising from the obtained differential equations of motion. Therefore, in the presence of disturbances that violate these conditions, the question arises of considering the release from servo-constraints and solving problems of stabilizing motion relative to the manifold defined by servo-constraints. Using this approach, equations of motion were compiled for a holonomic system, providing for release from servo constraints.

Let us consider a system whose position is determined by generalized coordinates $\tilde{q}_1, \dots, \tilde{q}_n$ and restrictions are imposed in the form of servo constraints:

$$f_\alpha(\tilde{q}_1, \tilde{q}_2, \dots, \tilde{q}_n, t) = 0 \quad (\alpha = 1, 2, \dots, a) \quad (1)$$

The resulting system, while remaining holonomic, has $n-a$ degrees of freedom and from equations (1) parameters $\mathbf{a}$ can be expressed through the rest $n-a$ and the latter can be taken as the generalized coordinates of the new system, i.e.:

$$\tilde{q}_i = \tilde{q}_i(q_1, q_2, \dots, q_k, t) \quad (i = 1, 2, \dots, n ; k = n - a) \quad (2)$$

the variation of coordinates is interconnected by the following conditions:

$$\delta \tilde{q}_i = \sum_s \frac{\partial \tilde{q}_i}{\partial q_s} \delta q_s \quad (s = 1, 2, \dots, k).$$

3. Statement of the problem.

To implement continuous parametric release, additional independent quantities η_p ($p = k+1, \dots, n$) are introduced that characterize the degree of release of the system from servo constraints. Then, these quantities η_p are considered as the left sides of the servo-constraint equations, calculated under the real movement of the system.

The possibility of changing parameters to implement continuous release in the dynamics of systems with servo constraints is of interest, first, from the point of view of determining the reactions of servo constraints. These reactions provide the necessary changes in parameters, including stabilization of the system's motion relative to the servo constraints.

For the variety of allowable positions of the released system and options, we obtain the following dependencies:

$$\tilde{q}_i = \tilde{q}_i(q_s, \eta_p, t), \quad \delta \tilde{q}_i = \sum_s \frac{\partial \tilde{q}_i}{\partial q_s} \delta q_s + \sum_p \frac{\partial \tilde{q}_i}{\partial \eta_p} \delta \eta_p. \quad (3)$$

We decompose the reaction forces of servo constraints (1) into normal and shear components in the following form:

$$R_i = \sum_\alpha \lambda_\alpha \frac{\partial f_\alpha}{\partial \tilde{q}_i} + \sum_\tau \mu_\tau \frac{\partial \tilde{q}_i}{\partial q_\tau}, \quad (\tau = k+1, \dots, n) \quad (4)$$

and write the d'Alembert-Lagrange principle in generalized coordinates $\tilde{q}_1, \dots, \tilde{q}_n$:

$$\sum_i \left(\frac{d}{dt} \frac{\partial T}{\partial \dot{\tilde{q}}_i} - \frac{\partial T}{\partial \tilde{q}_i} - Q_i - R_i \right) \delta \tilde{q}_i = 0, \quad (5)$$

where

$$T = T(\tilde{q}_i, \dot{\tilde{q}}_i, t), \quad Q = Q(\tilde{q}_i, \dot{\tilde{q}}_i, t).$$

4. Results.

Substituting equalities (3) into equations (5), we obtain

$$\frac{d}{dt} \frac{\partial \tilde{T}}{\partial \dot{\tilde{q}}_s} - \frac{\partial \tilde{T}}{\partial \tilde{q}_s} = \tilde{Q}_s + \sum_\tau \mu_\tau a_{s\tau}, \quad (s = 1, 2, \dots, k) \quad (6)$$

$$\frac{d}{dt} \frac{\partial \tilde{T}}{\partial \dot{\eta}_p} - \frac{\partial \tilde{T}}{\partial \eta_p} = \tilde{Q}_p + \lambda_p + \sum_\tau \mu_\tau b_{p\tau}, \quad (p = k+1, \dots, n) \quad (7)$$

$$\text{where } \tilde{T} = \tilde{T}(q_s, \eta_p, \dot{\eta}_p, q_s, t), \quad a_{s\tau} = \sum_i \frac{\partial \tilde{q}_i}{\partial q_s} \frac{\partial \tilde{q}_i}{\partial q_\tau}, \quad b_{p\tau} = \sum_i \frac{\partial \tilde{q}_i}{\partial \eta_p} \frac{\partial \tilde{q}_i}{\partial q_\tau}.$$

Thus, we obtain two systems of differential equations (6) and (7); Eq. (6) makes it possible to determine movement $\tilde{q}_i = \tilde{q}_i(t)$, and Eq. (7) makes it possible to determine the reactions of servo constraints (1) under their exact implementation.

Let us consider it using the example of a gyroscope installed in a gimbal. In the publications [15,16], devoted to the analysis of the behavior of a gyroscope with an electric motor, the assumption is made that the speed of rotation of the rotor relative to the symmetry axis remains absolutely stable. This assumption of rotational speed stability can be interpreted as introducing a special constraint on the gyroscope placed in a gimbal.

The condition for the stability of the angular velocity of eigenrotation can be considered as a servo constraint imposed on a gyroscope in a gimbal:

$$\dot{\gamma} = \omega = \text{const}, \quad (8)$$

where ω – is the constant angular velocity of the rotor's eigenrotation; γ – is the angle of rotation of the rotor relative to the inner race.

We present the kinetic energy of a gyroscope in a gimbal suspension in the following form [16]:

$$\tilde{T} = \frac{1}{2} \{ [A_2 + (A + A_1) \cos^2 \beta + C_1 \sin^2 \beta] \cdot \dot{\alpha}^2 + (A + B_1) \dot{\beta}^2 + C(\dot{\alpha} \sin \beta + \dot{\gamma})^2 \}$$

where α – is the angle of rotation of the outer race relative to the base, i.e. relative to a stationary object on which the gyroscope is located in a gimbal; β – is the angle of rotation of the inner race relative to the outer one; A_1, B_1, C_1 – are the moments of inertia of the inner race relative to the x_1, y_1, z_1 -axes, respectively, connected to the inner race; A, C – are the equatorial and polar moments of inertia of the gyroscope rotor; A_2 – is the moment of inertia of the outer race relative to the x_2 -axis connected to the outer race of the gimbal.

The left side of the constraint (1) is taken as the release parameter

$$\eta = \gamma - \omega t - \gamma_0 (\omega = \text{const}, \gamma_0 = \text{const})$$

the kinetic energy of the system is represented in the following form:

$$\tilde{T} = \frac{1}{2} \{ [A_2 + (A + A_1) \cos^2 \beta + C_1 \sin^2 \beta] \cdot \dot{\alpha}^2 + (A + B_1) \dot{\beta}^2 + C(\dot{\alpha} \sin \beta + \dot{\eta} + \omega)^2 \}$$

Equations (6) and (7) are written as:

$$\begin{aligned} \frac{d}{dt} \frac{\partial \tilde{T}}{\partial \dot{\alpha}} - \frac{\partial \tilde{T}}{\partial \alpha} &= \sum_{\tau} \mu_{\tau} a_{\tau 1} \quad (\tau = 1, 2) \\ \frac{d}{dt} \frac{\partial \tilde{T}}{\partial \dot{\beta}} - \frac{\partial \tilde{T}}{\partial \beta} &= \sum_{\tau} \mu_{\tau} a_{\tau 2} \\ \frac{d}{dt} \frac{\partial \tilde{T}}{\partial \dot{\eta}} - \frac{\partial \tilde{T}}{\partial \eta} &= \sum_{\tau} \mu_{\tau} b_{\tau 3} + \lambda \end{aligned}$$

or

$$\begin{aligned} [A_2 + (A_1 + A) \cos^2 \beta + (C_1 + C) \sin^2 \beta] \cdot \ddot{\alpha} + (C_1 - A_1 - A) \sin 2\beta \dot{\alpha} \dot{\beta} + \\ + (\dot{\eta} + \omega + \sin \beta \dot{\alpha}) C \cos \beta \dot{\beta} + C \sin \beta \ddot{\eta} = \mu_1, \quad (9) \\ (A + B_1) \ddot{\beta} + \frac{1}{2} (A + A_1 - C - C_1) C \cos \beta \dot{\beta} + C \sin \beta \ddot{\eta} - C \omega \cos \beta \dot{\alpha} = \mu_2 \\ C \ddot{\eta} + C \sin \beta \ddot{\alpha} + C \cos \beta \dot{\alpha} \dot{\beta} = \lambda \end{aligned}$$

With exact implementation of constraint (1), from equation (9), we obtain:

$$\begin{aligned} [A_2 + (A + A_1) \cos^2 \beta + (C + C_1) \sin^2 \beta] \cdot \ddot{\alpha} + (C_1 - A - A_1) \sin 2\beta \dot{\alpha} \dot{\beta} + \\ + (\omega + \sin \beta \dot{\alpha}) C \cos \beta \dot{\beta} = \mu_1 \quad (10) \\ (A + B_1) \ddot{\beta} + \frac{1}{2} (A + A_1 - C - C_1) \sin 2\beta \dot{\alpha}^2 - C \omega \cos \beta \dot{\alpha} = \mu_2 \\ C \sin \beta \ddot{\alpha} + C \cos \beta \dot{\alpha} \dot{\beta} = \lambda \end{aligned}$$

The results we arrived at are consistent with the equations determined in [15]. In that study, the right-hand sides of equations include both external and internal moments of forces N_{x2} , L_{y1} , M_{z1} , which are assumed known. This is because such moments can be caused by dry or viscous friction in bearings, tension in wires, or torque from motors.

The first two equations from the system of equations (10) are identical to the ones obtained in [16], but without considering the parametric release. If we assume that the servo link designated as (1) is ideal, then its response will be as follows:

$$\lambda = \frac{C}{2} \sin 2\beta \dot{\beta} \left[\frac{[2(A + A_1 - C) - C] \sin \beta \dot{\alpha} - C\omega}{A_2 + (A_1 + A) \cos^2 \beta + (C + C_1) \sin^2 \beta} \right] + C \dot{\alpha} \quad (11)$$

Let equations (10) be the equations of motion of a gyroscope with a holonomic releasing servo link (1). The following relations are taken as a particular solution:

$$\alpha = \alpha_0, \quad \beta = \beta_0, \quad \eta = 0, \quad \lambda = 0, \quad \mu_1 = \mu_2 = 0 \quad (12)$$

By introducing disturbances

$$\alpha = \alpha_0 + x_1, \quad \beta = \beta_0 + x_2, \quad \eta = x_3, \quad \lambda = u_3, \quad \mu_1 = u_1, \quad \mu_2 = u_2 \quad (13)$$

and substituting them into equations (9), we obtain:

$$\begin{aligned} & [A_2 + (A + A_1) \cos^2(\beta_0 + x_2) + (C + C_1) \sin^2(\beta_0 + x_2)] \ddot{x}_1 + (C_1 - A - A_1) \sin[2(\beta_0 + x_2)] \dot{x}_1 \dot{x}_2 + \\ & + C \cos(\beta_0 + x_2) (\dot{x}_3 + \omega) + \sin(\beta_0 + x_2) \dot{x}_1 \dot{x}_2 + C \sin(\beta_0 + x_2) \ddot{x}_3 = u_1 \\ & (B_1 + A) \ddot{x}_2 + \frac{1}{2} (A + A_1 - C - C_1) \sin[2(\beta_0 + x_2)] \dot{x}_1^2 - C \cos(\beta_0 + x_2) \dot{x}_1 \dot{x}_3 - \\ & - C \omega \cos(\beta_0 + x_2) \dot{x}_1 = u_2 \end{aligned} \quad (14)$$

$$C \ddot{x}_3 + C \sin(\beta_0 + x_2) \ddot{x}_1 + C \cos(\beta_0 + x_2) \dot{x}_1 \dot{x}_2 = u_3$$

Expanding the input trigonometric functions into series in x_2 powers, (not containing x_2 in a power higher than the first one), we obtain the equations of disturbed motion of the gyroscope in the first approximation:

$$\begin{aligned} L \cdot \ddot{x}_1 + N \cdot \dot{x}_2 + M \cdot \ddot{x}_3 &= u_1 \\ G \cdot \ddot{x}_2 - N \cdot \dot{x}_1 &= u_2 \\ C \cdot \ddot{x}_3 + M \cdot \ddot{x}_1 &= u_3 \end{aligned} \quad (15)$$

where

$$\begin{aligned} L &= A_2 + (A + A_1) \cos^2 \beta + (C + C_1) \sin^2 \beta \\ N &= C \omega \cos \beta_0, \quad M = C \sin \beta_0, \quad G = B_1 + A \end{aligned}$$

Under conditions $u_i = 0$ ($i = 1, 2, 3$), zero solution to system (15) is unstable or stable, but not asymptotically stable. Indeed, the characteristic equation of the system has the following form:

$$\begin{vmatrix} L\lambda^2 & N\lambda & M\lambda^2 \\ -N\lambda & G\lambda^2 & 0 \\ M\lambda^2 & 0 & C\lambda^2 \end{vmatrix} = 0$$

or

$$\lambda^6 (GLC - M^2 G) + CN^2 \lambda^4 = 0 \quad (16)$$

In order for the system under consideration to be stable in the first approximation, it is necessary that all the roots of the equation in terms of variable λ^2 be real and negative. Solving this equation in terms of variable λ^2 will give the following result:

$$\lambda^4 = 0, \quad \lambda^2 = -\frac{CN^2}{G(LC - M^2)}CN^2$$

Since $M^2 < LC$, then equation (16) has a pair of purely imaginary roots $\pm \sqrt{CN^2 / G(LC - M^2)} \cdot i$. In this case, both stability (not asymptotic) and instability can occur [17-20].

To achieve asymptotic stability, we involve controls u_i .

By introducing the following notation

$$x_i = y_{2i-1}, \quad \dot{x}_i = y_{2i} \quad (i = 1, 2, 3)$$

and making the necessary calculations, we obtain equations in the normal form:

$$\begin{aligned} \dot{y}_1 &= y_2, \quad \dot{y}_2 = -\frac{NC}{LC - M^2} y_4 + \frac{C}{LC - M^2} u_1 - \frac{M}{LC - M^2} u_3 \\ \dot{y}_3 &= y_4, \quad \dot{y}_4 = \frac{N}{G} y_2 + \frac{1}{G} u_2 \end{aligned} \quad (17)$$

$$\dot{y}_5 = y_6, \quad \dot{y}_6 = -\frac{NM}{LC - M^2} y_4 - \frac{M}{LC - M^2} u_1 + \frac{L}{LC - M^2} u_3$$

Let us study the controllability of a gyroscope as a control object with three controls.

Let us create a matrix [21]

$$K = \|Q, PQ, P^2Q, P^3Q, P^4Q, P^5Q\| \quad (18)$$

The determinant of a matrix composed of matrix columns Q and (PQ) , is $C(GL + MN) / G^3(LC - M^2)$ and, therefore, the rank of matrix (18) is six. The latter means that a gimbal gyroscope as a control object with non-ideal releasing servo constraints (1) is completely controllable.

5. Analysis of results.

To study the controllability of a gyroscope using the normal components of the reactions of servo constraints (1), we create a matrix of the form (18), in which matrix K_1 , of a 6x6 format, is square and contains six rows and columns.

$$P = \begin{vmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{NC}{M^2 - LC} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & \frac{N}{G} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{NM}{CL - M^2} & 0 & 0 \end{vmatrix} \quad Q = \begin{vmatrix} 0 & 0 & 0 \\ \frac{C}{LC - M^2} & 0 & \frac{M}{M^2 - LC} \\ 0 & 0 & 0 \\ 0 & \frac{1}{G} & 0 \\ 0 & 0 & 0 \\ \frac{M}{M^2 - LC} & 0 & \frac{L}{LC - M^2} \end{vmatrix}$$

$$PQ = \begin{vmatrix} \frac{C}{LC-M} & 0 & \frac{M}{M^2-LC} \\ 0 & \frac{NC}{G(M^2-LC)} & 0 \\ 0 & \frac{1}{G} & 0 \\ \frac{CN}{G(LC-M^2)} & 0 & \frac{MN}{G(M^2-LC)} \\ \frac{M}{M^2-LC} & 0 & \frac{L}{LC-M^2} \\ 0 & \frac{MN}{G(LC-M^2)} & 0 \end{vmatrix}$$

The determinant of matrix K_1 is zero, which indicates that its rank is below six, emphasizing the limited controllability of the gimbal gyroscope due to standard responses to servo constraints.

This implies that for complete controllability of a system with servo constraints, it is necessary to consider possible non-idealities of these constraints. Otherwise, to achieve controllability, adding new constraints or introducing various forces (dissipative or gyroscopic ones) will be necessary.

6. Conclusions.

Based on these principles, the following algorithm of the system release can be derived. To release a system with an allowable set of states determined by equations (2), it is necessary to transform the system so that the allowable set of states of the new system is specified by equations (3). In these equations, η_p are additional independent parameters that replace the functions from equations (2), provided that $\eta_p = 0$.

$$K_1 = \begin{vmatrix} 0 & \frac{M}{M^2-LC} & 0 & \frac{MCN^2}{G(M^2-LC)^2} & 0 & \frac{MC^2N^2}{G^2(M^2-LC)^2} \\ \frac{M}{M^2-LC} & 0 & \frac{MCN^2}{G(M^2-LC)} & 0 & \frac{N^4MC^2}{G^2(M^2-LC)^3} & 0 \\ 0 & 0 & \frac{MN}{G(M^2-LC)} & 0 & \frac{MCN^3}{G^2(M^2-LC)^2} & 0 \\ 0 & \frac{MN}{G(M^2-LC)} & 0 & \frac{MCN^3}{G^2(M^2-LC)^2} & 0 & \frac{N^3MC^2}{G^3(M^2-LC)^3} \\ 0 & \frac{L}{LC-M^2} & 0 & -\frac{N^2M^2}{G(M^2-LC)} & 0 & -\frac{N^4M^2C}{G^3(M^2-LC)^3} \\ \frac{L}{LC-M^2} & 0 & -\frac{N^2M^2}{G(M^2-LC)} & 0 & -\frac{M^2CN^4}{G^2(M^2-LC)} & 0 \end{vmatrix}$$

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STUDYING THE THERMAL CONDUCTIVITY OF MATERIALS.

Lyamin A.V.¹, Nurmukhamedov A.M.¹, Nosirov M.I.¹

¹Tashkent institute of chemical technology, Tashkent, Uzbekistan

E-mail: lyamin_2020_tkti@mail.ru

Abstract: The article is devoted to the determination of the coefficient of thermal conductivity of materials by the method of continuous heat flow. Two types of materials were tested in the experiment: textolite and fluoroplast, which are both heat-insulating and antifriction. The experiment was conducted at the department of "Fundamentals of mechanics and engineering graphics" of the Tashkent institute of chemical technology on a standard laboratory stand. The results of the experiment are presented in the form of summary tables and a graph.

Keywords: thermal conductivity, coefficient of thermal conductivity, heat flow, temperature gradient.

1. Introduction.

The thermal conductivity of materials in the construction of modern structures is essential and important, since comfort and warmth in the living room will depend on which materials were chosen as construction materials. Currently, basalt fiber slabs produced in Uzbekistan, among others, are beginning to be used everywhere in Tashkent during the construction and reconstruction of residential premises [1–9].

In this regard, we consider it necessary and expedient to conduct a comprehensive study to determine their thermal conductivity in laboratory or production conditions before manufacturing and installing such building insulation materials.

2. Methods of research.

To conduct experiments to determine the thermal conductivity coefficient of various materials, the department of "Fundamentals of mechanics and engineering graphics" has a corresponding experimental installation. On this type of equipment, studies can be carried out to determine the temperature gradient of various solid materials of round shape and limited thickness. This experimental equipment in the fields of science: thermal engineering and hydraulics, is manufactured by "Zarnitsa" in the Russian Federation (Kazan). Similar laboratory equipment in this area is manufactured in Germany (Hamburg) by GUNT [10–14].

This article provides a methodology for conducting a study on the thermal conductivity of samples made of textolite and fluoroplast. These materials are widely used both in general mechanical engineering and in the construction of various structures.

Recall that the thermal conductivity of materials is the ability of a material body to transfer heat from its more heated parts to less heated ones through the chaotic movement of particles. Consider a layer of a solid substance with a temperature of T_0 enclosed between two parallel plates located at a distance of Δh from each other. Let the temperature of the lower plate rise instantly from T_0 to T_1 at time $\tau=0$, which does not change at subsequent times. As a result, the temperature profile inside the layer begins to change over time and, at sufficiently high temperatures, a stationary linear temperature distribution is established. In stationary conditions, to maintain the temperature difference (temperature gradient) $\Delta T=T_1-T_0$, a constant heat flow Q is required. For sufficiently small values of ΔT , the ratio is valid:

$$\frac{Q}{F} = \lambda \cdot \frac{\Delta T}{\Delta h} \quad (1)$$

According to expression (1), the rate of heat transfer through a unit surface area of the layer F is proportional to the temperature difference at a distance of Δh . The coefficient of proportionality is called the coefficient of thermal conductivity of the material [15–18].

Equation (1) is also valid in cases where the space between the plates is filled not only with a solid, but also with a liquid or gas, provided that there is no convection and radiation. Thus, this ratio describes the process of thermal conductivity in solids, liquids and gases.

If the local rate of heat transfer through a unit of the layer surface (heat flux density) in the positive direction of the "y" axis is denoted by q_y , then at $\Delta h \Rightarrow 0$ the ratio (1) takes the form:

$$q_y = -\lambda \cdot \frac{dT}{dy} \quad (2)$$

This equation is a one-dimensional formulation of Fourier's law of thermal conductivity. It is valid if the temperature depends on only one "y" coordinate. Thus, the following formulation can be given to the law of thermal conductivity: the density of the heat flux due to thermal conductivity is proportional to the temperature gradient. The minus sign in equation (2) means that heat is spreading in the direction of decreasing temperature.

When experimentally determining the coefficient of thermal conductivity, as a rule, they strive to create a one-dimensional temperature field. So, in relation to the one-dimensional temperature field of flat, cylindrical and spherical layers under boundary conditions of the first kind, the equation for determining the thermal conductivity coefficient looks like this:

$$\lambda = \frac{K \cdot Q}{t_1 - t_2} \quad (3)$$

where: Q is the heat flux (in W); t_1 & t_2 are the temperatures of the outer and inner surface of the layer ($^{\circ}\text{C}$); K is a coefficient depending on the shape and size of the test sample (m^{-1}) [19–23].

It follows from formula (3) that in order to determine the thermal conductivity coefficient of the material under study, it is necessary to measure in stationary mode the heat flux Q passing through the sample under study and the temperatures of its isothermal surfaces. Formula (3) describes the temperature distribution in solids, as well as in liquids and gases in the absence of heat transfer methods other than thermal conductivity. In the case of a temperature dependence of the thermal conductivity coefficient, formula (3) can be used provided that a small temperature difference occurs in the sample under study.

Despite its methodological simplicity, the practical application of stationary thermal conductivity methods to determine the appropriate coefficients is associated with difficulties in creating a one-dimensional temperature field in the studied samples and taking into account heat losses.

In addition, stationary methods require considerable time for conducting experiments due to the duration of the installation's transition to a stationary thermal regime.

In the study of thermal insulation materials with low thermal conductivity ($\lambda \leq 2,3 \text{ W/m}\cdot\text{K}$), the method of an unlimited flat layer has become widespread, when a sample of the material under study is given the shape of a thin plate. To create a temperature difference, one surface of the plate is heated, and the other is cooled using devices between which the test sample is clamped.

When choosing the geometric dimensions of the studied samples of materials with low thermal conductivity, it is necessary to fulfill the condition: $\delta \leq \left(\frac{1}{7} \dots \frac{1}{10}\right) \cdot D$ (where D is the diameter of a round plate or the side of a square), which ensures the one-dimensionality of the temperature field. Thermal insulation is used to eliminate heat losses from the side surfaces of the sample.

The disadvantages of the method include the difficulties associated with the elimination of thermal resistances that occur at the points of contact of the sample with the surfaces of the heater and refrigerator. The error in determining the thermal conductivity coefficient due to the contact resistance can reach 10-20% with a sample thickness of 1,5-3,0 mm and becomes even greater with an increase in the thermal conductivity of the material under study. To reduce the contact thermal resistance of the surface of the sample and heat exchangers, they are subjected to careful processing, and significant compressive forces are created to ensure good contact.

The experimental stand "Study of thermal conductivity of materials" is made in the form of a desktop stand equipped with a horizontal work surface for the arrangement of the studied laboratory modules and a vertical work surface on which the control and control unit is located (Fig.1). The measuring system of the experimental installation has the ability to output the temperature regime to a personal computer in the form of graphs.

The main module consists of two laboratory samples "4", which are placed between the heater "1" and two refrigerators "2" (Fig.2). The necessary density of contact of the studied samples with hot and cold surfaces is provided by the use of the device "5". To reduce heat losses, the heater has a casing "3".

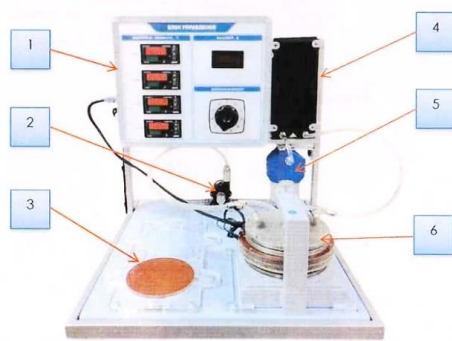
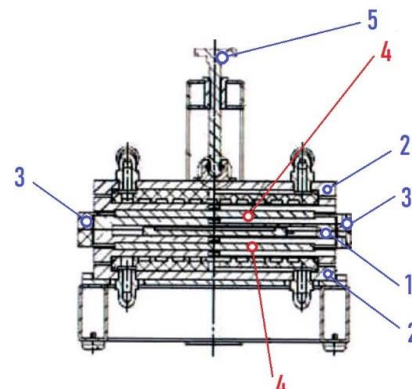


Figure 1. General view of the experimental installation:
1 – control and display module; 2 – centrifugal pump; 3 – set of laboratory samples; 4 – air-water heat exchanger; 5 – water tank; 6 – module "determination of thermal conductivity coefficient".



A



B

Figure 2. Image of the main module (position #6, Fig.1):
A – exterior view; B – sectional view; 1 – heater; 2 – refrigerator; 3 – thermal insulation casing; 4 – test sample; 5 – clamping screw.

The main module consists of two refrigerators and one heater (see Fig.2-B). Refrigerators are placed at the bottom and top, and the heater is in the middle of the prefabricated structure. A sample of the test material (specially treated disc) is placed between each refrigerator and the corresponding side of the heater. For a tight fit of the samples to the walls of the refrigerator and heater, the module

is equipped with a screw clamp. For temperature measurement, the module is equipped with 7-point temperature sensors with Pt100 calibration.

The stages of the experiment can be represented as the following algorithm:

1. Determination of the average temperature values of the inner hot t_{ht} and outer cold t_{cd} surface of the samples according to the formulas:

$$t_{ht} = \frac{t_1 + t_2 + t_3 + t_4}{4} \quad \text{– the average surface temperature of the samples from the heater side;}$$

$$t_{cd} = \frac{t_5 + t_6}{2} \quad \text{– the average surface temperature of the samples from the refrigerator side.}$$

$$t_{av} = \frac{t_{ht} + t_{cd}}{2} \quad \text{– the average temperature between the hot and cold side of the test sample.}$$

2. Determination of the heat flux Q passing through the samples: $Q = Q_{ht} - Q_{hf}$, where $Q_{ht} = U^2/R$ is the heat flux from the heater; $Q_{hf} = \frac{5\pi\lambda_{cs}}{2} \left(\frac{d_{ht} + d_{cs}}{d_{cs} - d_{ht}} \right) (\delta_{ht} + \delta_{cs})(t_{ht} - t_{cs})$ - heat losses on the casing.

3. Determination of the coefficient of thermal conductivity according to the formula: $\lambda = \frac{K \cdot Q}{t_{ht} - t_{cd}} = \frac{Q \cdot \delta}{2F \cdot (t_{ht} - t_{cd})}$, where: $K = \frac{\delta}{2F}$ – a coefficient that takes into account the shape of the sample; δ - sample thickness; $F = \frac{\pi d^2}{4}$ – the surface area of the sample. The installation parameters required for calculations are shown in Table 1.

4. After determining the value of " λ " for two different materials, it is necessary to plot the dependence of the thermal conductivity coefficient on the average temperature.

5. Comparison of the found thermal conductivity coefficients with the corresponding theoretical values. The values of " λ " for some materials are shown in Table 2.

Table 1.

Individual parameters of the experimental mounting

Name of the parameter	Numerical value
Heater resistance, R	280 Om
Diameter of the heater, d_{ht}	0,17 m
Heater thickness, δ_{ht}	0,02 m
Sample diameter, d	0,14 m
Sample thickness, δ	0,005 m
Thermal conductivity coefficient of the casing (material is glass-textolite), λ_{cs}	0,3 W/(m·K)
Outer diameter of the casing, d_{cs}	0,19 m
Thickness of the casing, δ_{cs}	0,024 m

Table 2.

The actual thermal conductivity coefficients of some materials

Material	λ_0 , W/(m·K)
Asbestos ($\rho=500 \text{ kg/m}^3$)	0,107
Asbestos cement	0,088
Wool felt	0,047
Textolite	from 0,23 to 0,34
Fluoroplast	0,255

3. The results of the experiment.

The main results of the experiment are presented in tabular form (see Tab.3 and Tab.4) and the dependency graph (Fig.3).

Table 3.

The results of measuring the thermocouple readings at the experimental mounting

№	U, V the voltage on the heater	Thermocouple readings, °C							t_{ht} , °C	t_{cd} , °C	t_{av} , °C	Type of material
		T1 heating	T2 heating	T3 heating	T4 heating	T5 cooling	T6 cooling	T7 casing				
0	the room temperature	18,2	18,1	18,0	18,0	17,8	18,7	20,3	18,0	18,2	18,1	Fluoroplast
1	100	26,0	28,2	23,4	23,4	26,1	26,3	23,4	25,2	26,2	25,7	
2	150	36,4	39,5	31,0	31,0	25,8	25,9	25,4	34,4	25,8	30,1	
3	200	51,2	59,7	42,2	42,3	25,1	25,1	29,7	48,8	25,1	36,9	
4	250	71,6	84,4	58,4	58,7	25,0	25,0	36,6	68,2	25	46,6	Textolite
0	the room temperature	28,1	28,0	28,4	28,5	25,6	26,4	25,0	28,2	26	27,1	
5	100	31,0	33,0	29,1	29,1	21,9	21,9	24,2	30,5	21,9	26,2	
6	150	36,7	41,3	32,2	32,3	21,7	21,8	24,7	35,6	21,75	28,6	
7	200	46,0	54,0	38,2	38,3	21,9	22,0	26,5	44,12	21,9	33,0	

8	250	61,2	73,3	49,6	49,9	22,5	22,7	30,6	58,5	22,6	40,55	

Table 4.

Heat flow measurement results

№	U, V the voltage on the heater	Q_{ht} , W heating	Q_{hf} , W losses	Q, W the total heat flow	t_{ht} , °C	t_{cs} , °C T7	$t_{ht} - t_{cs}$ °C	λ , W/(m·K) the coefficient of thermal conductivity	Type of material
0	the room temperature				18,075	20,3			Fluoroplast
1	100	35,71429	3,450546	32,26374	25,25	23,4	1,85	-5,739033	
2	150	80,35714	16,92633	63,43082	34,475	25,4	9,075	1,2427645	
3	200	142,8571	35,71781	107,1393	48,85	29,7	19,15	0,7623121	
4	250	223,2143	59,07894	164,1353	68,275	36,6	31,675	0,6409329	
0	the room temperature				28,25	25,0			Textolite
5	100	35,71429	11,84377	23,87052	30,55	24,2	6,35	0,4516231	
6	150	80,35714	20,37687	59,98027	35,625	24,7	10,925	0,7074663	
7	200	142,8571	32,87345	109,9837	44,125	26,5	17,625	0,8116991	
8	250	223,2143	52,03796	171,1763	58,5	30,6	27,9	0,7803322	

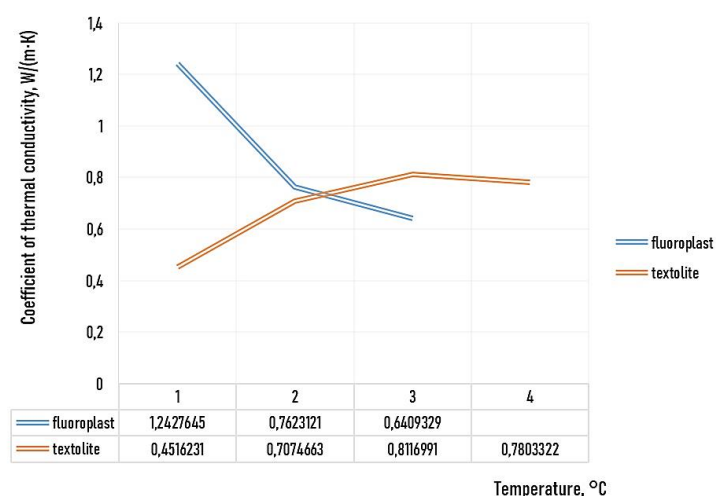


Figure 3. Graph of the dependence of the thermal conductivity coefficient on the average temperature.

4. Discussion of the results.

According to the results of the conducted studies, it can be seen that the theoretical values of the thermal conductivity coefficients of textolite and fluoroplast differ greatly from each other. So, according to Table 2, for textolite this value ranges from 0,23 to 0,34 W/(m·K), which is 2,3 times less than the obtained average experimental values; for fluoroplast, the reference values range from 0,255 W/(m·K), which is as much as 3,5 times less than the obtained average experimental values. Thus, it makes sense to re-conduct studies for different temperature and time ranges.

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METHODS FOR DETERMINING AND IMPROVING PARAMETERS AFFECTING WAGONS

Makhmudova Sh.A.¹, Akhmedova D.A.¹

¹Tashkent State Transport University, Tashkent, Uzbekistan

E-mail: maxmudovash88@gmail.com

Abstract: Analysis of researches on the dynamics of rotation of a pair of wagon wheels on the slope of the hill, despite the fact that there are enough works on the existing method of calculating the height of the sorting hill, the method of calculating the braking distance of the wagon on the sections with a stopping point of the hill under the influence of longitudinal wind is studied in this article. at the same time, the determination of the speed of the wagon entering the next section from the stopping place was studied.

Keywords: force, moment, railroads, braking zone, rotating masses, construction

1. Introduction

In scientific research, scientists have presented the identification of the parts of the rail shaft, which were broken by fatigue. This was achieved by studying malfunctions in the train car axle. To do this, a wagon model was considered to obtain dynamic forces acting on the wagon axis. By studying the dynamic forces and moments acting on the axis, the critical part of the axis was determined to predict the broken region. In addition, the change in the safety factor, which depends on the speed of the wagon load, is discussed taking into account passengers [1]

In recent times, as a result of the growing demand for cargo transportation by freight railroads, scientists and engineers are developing cost-effective loading capacity strategies. Including such strategies as increasing the load on the vehicle axle. The path may be technically unsuitable for additional loads and stresses. but there is an exception, the design of the track structure for heavy transport is based on the traditional track design with ballast. However, heavy vehicles are characterized by load, speed, traffic intensity, etc., which differ from normal transportation conditions [2].

The development of training does not stop there, as the focus is on fatigue problems specific to the railway industry. formulation of the problem of stimulated representations. Apart from these problems, the problem is related to bearings, axles and then [3].

One of the reasons for the destruction is the destruction of engineering components. In recent years, a large amount of data has been collected and up to 50 parameters are known that influence fatigue behavior. Engineers discuss the most commonly studied parameters such as process effects (export, corrosion, relative slip between components, temperature and time effects, surface traction and normal surface pressure between components)[4].

The purpose of the study was to develop and analyze experimental methods for measuring the forces acting on the side frame of bogies with the sides of the wheel pair when the car is moving. Using the final method, theoretical studies were carried out on the temperature-strained state of the rear frames of freight car bogies under the action of a wheel pair on the axial opening. as well as the method of carrying out the finite element method, it was possible to avoid the installation location of strain gauges [5].

2. Research Methods

According to the D'alambert principle, parallel to the rail line, $Ox'y'z'$ to the Ox' axis of the inertial calculation system according to the mathematical description of the motion of the carriage in the braking zones on the sections of the sorting hill for the absolute motion of the rigid body on the projection, the acceleration of the carriage in the equally decelerated motion has a negative sign

$$|a_{kTi}| = \frac{|\Delta F_{Ti}|}{M_{pro}} 10^2, \quad (1)$$

where $|\Delta F_{Ti}|$ – is the resultant force due to which slippage of wagon wheel pairs along the rolling surface of the track and on the brake tires of the wagon retarder in the braking zones (BZ) of sections with a parking space (PS) occurs, kN:

$$|\Delta F_{Ti}| = F_{xi} + |F_{ci}|; \quad (2)$$

$|a_{kTi}| = a_{kTi} \cdot \text{sgn} \Delta F_{1Ti}$ – with the modulus function if $|\Delta F_{1Ti}| < 0$ if, $|a_{kTi}| = -a_{kTi}$, will be.

(1) from the expression, $|\Delta F_{Ti}| < 0$ or $|F_{ci}| > F_{xi}$. If the condition is observed, the acceleration of the car in the braking zones of the sections with a stop at the initial speed $V_{HTi} > 0$ will be equally slowed down to the moment $V = 0$.

In accordance with the existing methods of calculating sorting peaks, we express the formulas (1) and (2) in the usual concepts and generally accepted notations.

It is possible to present the force F_{xi} due to the entry of the wagon into the braking zones (BZ) of the sections of PS, G_1 from the force of non-rotating parts (carriage, wagon body) and wagon weight in shares:

$$F_{xi} = k_{x0i} G_1, \quad (3)$$

here i – the number of profile sections of the sorting hill ($i = 1, \dots, 9$), $k_{x0i} - G_1$ is the coefficient describing the force F_{ex} taking into account the longitudinal wind force projection force F_{xi} as a percentage of G_1 , i.e. in the following form:

$$k_{x0i} = i_{ax} + k_{ax} \approx 0.014 + 0.004 = 0.018.$$

We note that the value of F_{BX} is neglected because it is too small: $F_{BX} \ll G_1$ (for example, $3.2 \ll 794 \text{ kN}$ [6]).

Power of various resistances $|F_{ci}|$ due to its components, the speed of entry of the wagon into the braking zone according to $V_{BX.Ti}$, the adhesion force of the wagon wheel to the brake pad of the wagon retarder F_{brak} the sliding friction force of the wheel pair relative to the brake tires (main wear) $F_{fri.ski}$ or F_{oi} ni, the environment, The resistance force F_{sn} from snow and ice can be determined as a percentage of the weight G_1 of the non-rotating parts (carriage, wagon body) and the weight of the freight wagon, as well as the F_{xi} force.

For example, $F_{brak} = k_{brak} G_1$, here $k_{brak} \approx 0.024 - G_1$; $F_{oi} = k_{oi} G_1$ factor taking into account the F_{brak} power in share from $F_{ch} = k_{ch} G_1$ where $k_{oi} \approx 0.25 - G_1$; factor taking into account the power F_{oi} from $k_{sn} G_1$ in share, where $0.0075 - G_1$ – factor taking into account the power F_{sn} from in share.

The result is the power of various resistances $|F_{ci}|$ can be represented as:

$$|F_{ci}| = (k_{brak} + k_{oi} + k_{cb} + k_{ch}) G_1 \quad (4)$$

As a final result, in accordance with [7], we can write formulas (1) and (2) in the following form in generally accepted notations:

$$|a_{kTi}| = a_T (i_{t0xi} - |w_{Ti}|). \quad (5)$$

Here $a_m = \text{const}$ – conditional determination of its linear acceleration of the car in the braking zones of SP (stopping place) sections in the uniformly decelerated movement, m/s^2 :

$$a_m = \frac{G_1}{M_{pro}} 10^3 \quad (6)$$

For example, at $G_1 = 794 \text{ kN}$ and $M_{pro} = 8,269 \cdot 10^4 \text{ kg}$: $a_T = 8,953 \text{ m/s}^2$; $i_{t0xi} - (5)$ is a dimensionless quantity conditionally describing the profile slope of the hill in the SP sections, taking into account the effect of the longitudinal wind projection force F_{BX} according to the formula

$$i_{t0xi} = i_{TXi} + k_{BX} \quad (7)$$

$k_{BX} - G_1$ is a dimensionless quantity that contributes to the accelerated movement of the wagon as a percentage of, taking into account the effect of a small value of the longitudinal wind projection F_{BX} on the Ox axis. If the force F_{BX} is not taken into account, then $k_{BX} = 0$, $|w_{Ti}|$ – a dimensionless quantity and/or difference conditionally describing the relative resistance of various resistances to movement in the braking zones of sections SP, taking into account the effect of longitudinal wind, $|w_{Ti}|$

$$w_{Ti} = k_{brak} + k_{oi} + k_{cb} + k_{ch} \quad (8)$$

Here $k_{\text{brak}} = F_{\text{brak}}/G_1$, $k_{0i} = F_{\text{fri}}/G_1$ or $k_{0i} = F_{0i}/G_1$, $k_{\text{cb}} = F_{\text{cb}}/G_1$, $k_{\text{ch}} = F_{\text{ch}}/G_1$ Here, $-F_{\text{brak}}$ forces in braking of the car retarder, coefficients taking into account the forces $F_{\text{fri}} = F_{0i}$ from the main resistances, F_{cb} forces from the environment and wind, and F_{ch} forces from snow and snow, in the percentage G_1 of the wagon's non-rotating parts and the weight of the wagon.

It should be noted separately that the relative resistance to movement of various resistances in the braking zones of SP sections $|w_{\text{ti}}|$ in practice, it is a non-constant quantity ($|w_{\text{ti}}| \neq \text{const}$), and its increase occurs when the speed of the car decreases.

For the convenience of calculations, the resultant force due to car braking at SP sections $||\Delta F_{\text{ti}}||$ we can write (see formula 4) in the following form:

$$|F_{\text{ti}}| = (k_{x0i} + (k_{\text{brak}} + k_{0i} + k_{\text{cb}} + k_{\text{ch}}))G_1.$$

Here, based on the previously defined mathematical models, the time taken by the wagon to travel the wheel base section of the first stop on the hill under the influence of the longitudinal wind was calculated.

Wagon Calculations of the rotation speed of wagon wheel pairs were performed on the considered section of the hill based on the known values of the movement time.

It should be noted that the results of previous known studies on the rolling dynamics of a pair of wagon wheels on the profile of the sorting hill are not clear [8].

Taking this into account, the proposed method of constructing a mathematical model of the accelerated carriage movement in the acceleration sections of the hill under the influence of a small longitudinal wind is described in detail.

In this case, the effect of the longitudinal wind projection on the horizontal and side of the wagon (for example, in the South-West direction) is taken proportional to the surface of the side of the wagon facing the wind, that is, $F_{\text{rb},x} = f(A_{\text{r}})$ ба $F_{\text{rb},y} = f(A_{\text{b}})$.

According to the method proposed by the authors, the rotation speed and movement time of the wagon wheel pair in the previous section at the first brake position of the sorting hill under the influence of longitudinal wind were determined in [9]. An example of calculating the speed and time of rotation of a pair of wagon wheels under the influence of longitudinal wind on the section before the first stop is given.

In section 1BZ, the results of calculation of the speed and movement times of the wagon after braking in this section due to the effect of longitudinal wind are presented in the non-stop movement of the wagon, as well as for a special case, i.e., when the wagon decelerator is located on the "horizontal area".

The results of the analytical study of the movement of the carriage on the slope of the second acceleration section of the sorting hill under the influence of longitudinal wind on the 1BZ section with braking and in the arrow zone are described in [10]. For a special case, i.e., when the car is located in the "horizontal area" of the decelerator, the acceleration of the hill under the influence of a small headwind and 1BZ sections, the research results of the car movement are presented.

A refined example of calculation of the rotation time and speed of a pair of wagon wheels from the slope of the hill in the wheelbase length of the section braking zones under the influence of the longitudinal wind, according to the proposed method, was found to have escaped the attention of the researchers.

The purpose of this paragraph is to determine the acceleration of a car moving with equal acceleration down the slope of a hill, taking into account the effect of a small longitudinal wind, using the proportionality of the forces acting on the "car-road" system.

Then, using the formulas of the speed and path of body movement known from ordinary physics, develop a sequence of calculating the rotational movement and speed of a pair of wagon wheels in the length of the wheel base of the sorting hill section.

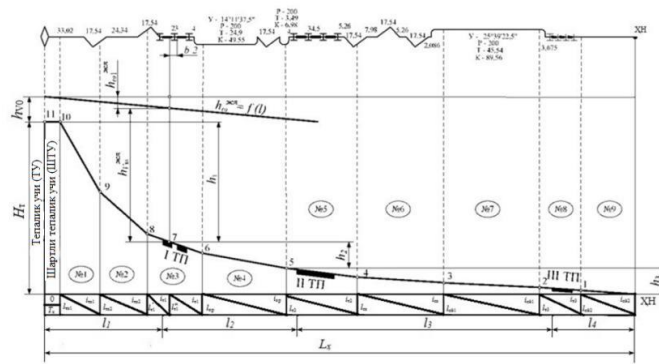


Figure 1. - Calculation drawing of the longitudinal profile of the hill descent

Table 1

Existing methods of determining the path, speed and travel time of the wagon in sections with a stop

№	Calculation expression	Published organization or author	Year of development	Factors considered
1	-	"RTY" JSC on the railway and in all countries	1938 y.	Power of braking devices
2	$v = v_{o3} - a_3 t$ $x(t) = v_{o3} t + \frac{1}{2} a_3 t^2$ $t_{ti} = \frac{v_{nti}}{a_{kti}}$	Turanov H.T., Gordienko A.A., Plahotich I.S.	2016 y.	$v, x(t)=l, t$

State the issue. It is required to obtain an accurate example of the calculation of the time of the wagon movement and rotation speed according to the simplified method at the length of the wheelbase of the wagon at on the sorting hill.

In this case, according to the known value of the length of the wheel base braking zones on the hill, the time of movement of the wagon t_{1kb} , given the given initial speed V_{01kb} , before braking of the decelerator of the car can be determined.

According to the value of t_{1kb} , the rotation speed of the wagon wheel pair $V_{1kb}(t_{1kb})$ can be determined.

Proposed sequence for solving the problem of the movement of the wagon in the length of the wheelbase of on the hill. At the length of the wheelbase of 1SP on the hill, the sequence of calculation of the speed of the wagon under the influence of the longitudinal wind is the same as in [11-13] and is performed in the following sequence:

1. In solving the problems of carriage movement from the sorting hill, the simple rules of theoretical mechanics were used: the basic principle of D'alambert in the coordinate form, as well as the formulas for determining the speed and path of the body's movement, known from the course of physics.

2. In the conditions of the problem and the accepted initial conditions, the case of the wagon entering the 1SP section from the slope of the sorting hill with the specified initial speed V_{0kb} is considered. When a single wagon enters the 1SP section of the hill, the wagon is mainly subjected to external forces in the form of the weight of the loaded wagon, as well as longitudinal forces in the form of F_{rBX} and F_{rB} tests the projection of aerodynamic drag force against the wind.

3. When forming the calculation model of the movement, in pure rolling of the wheels relative to the rolling surface, the rotation of the wheel pair along the rail line is $F_{fri,x}$ and the effect of the wind on the wagon is from the side. It is assumed that the sliding friction on the side of the rail line is F_{rBy} , that is, $F_{fri,x} = F_{\tau} = F_{fri,k} + F_{fri}$.

3. Analysis and results

Accordingly, the contact-free mechanics (found by applying the rail line principle) in the proposed computational model of carriage movement on the 1SP section of the hill 3 will look like the picture.

In this case, using the basic principle of D'alambert, as well as the plot of hill 1SP for a known value of the length of the wheelbase of the car, the speed and path formula of the body $l_{1kb} = x_{1kb}(t_{1kb})$ the time at which the movement of the car takes place at the given initial speed before braking of the retarders at hill 1BZ is t_{1kb} , and according to the value of t_{1kb} , the car speed $V_{1kb}(t_{1kb})$ was determined.

The proposed method for calculating the rotational speed and movement time of a pair of wagon wheels at the wheelbase length of 1BZ of the hill. A new method of calculating the rotation speed and movement time of a pair of wagon wheels under the influence of longitudinal wind at the wheelbase length of 1BZ of the hill is performed in the following sequence:

1. When designing the sorting hill, the length of the wagon wheelbase for the platform is assumed to be $l_{bx} = 3,5 \text{ m}$, taking into account the entrance length of the wagon to this section $l_{kb} = 10,5 \text{ m}$. Then the length of the considered section of the hill is $l_{1kb} = l_{kb} + l_{bx} = 14 \text{ m}$. In this case, the slope of the hill is 14 thousand, $\psi_{03} = 0,014 \text{ rad}$ ($\sin(\psi_{03}) = 0,014$, $\cos(\psi_{03}) = 1$) [4].

2. The projection of the longitudinal wind force on the car is calculated along the longitudinal Ox and transversal Oy_1 axis in the following form, kN :

$$F_{rBx} = 0.5A_r; F_{rBy} = 0.5A_b,$$

here $0.5 - 1 \text{ m}^2$ is the specific pressure, kN/m^2 , A_t is the surface of the sides of the freight wagon, m^2 . For example, if $A_r = 6.384$ or $A_b = 27.36 \text{ m}^2$, then $F_{rBx} = 3.192$ and $F_{rBy} = 13.68 \text{ kN}$

3. The rolling friction force caused by the weight of the freight wagon G_z projection is $F_{fri.k}$. Also, the wind resistance force F_r along the Oz axis is determined as the component product F_{rBx} of the reaction of the links (rail line). According to Coulomb's law, this force is equal to:

$$F_{fri.k} = f_0(G \cos(\psi_{03}) + F_{rBx} \sin(\psi_{03})),$$

here f_0 is the given coefficient of rolling friction taking into account the number of wheels on the trolley, rolling friction (on the bearing rings and on the wheel rail), usually $f_0 = 0.0001$ is taken. For example, if $f_0 = 0.0001$, the weight of the freight car, taking into account the inertia of the rotating masses (pair of wheels) $G = 908 \text{ kN}$.

When $G = 650 \text{ kN}$, $\cos(\psi_{03}) = 1$, $F_{rBx} = 3.192 \text{ kN}$, $\sin(\psi_{03})$, $F_{fri.x} = F_{fri.k} = 0.094 \text{ kN}$. The frictional force of the car sliding due to the wind from the side of the car is $F_{fri.k}$. This force, as a component product of the reaction of the links (rail line), is equal to the following according to Coulomb's law:

$$F_{fri.x} = F_{fri} = f_{ck0} F_{rBy} = f_{ck0} F_{rBb},$$

here f_{ck0} is the coefficient of friction of the wheel flange when sliding the rail line to the side surface (usually $f_{ck0} = 0.25$ is taken). For example, if $f_{ck0} = 0.25$, $F_{rBy} = F_{rBb} = 13.68 \text{ kN}$, $F_{fri.x} = F_{fri} = 3.42 \text{ kN}$.

For example, if $k_{cp} = 0.0003$ and $G = 908 \text{ kN}$, then $F_{cp} = 0.272 \text{ kN}$ will be.

4. All the forces acting on the rolling wagon under the influence of the headwind at a constant value are calculated, kN :

- on the 1BZ section of the hill, the forces that ensure the movement of the wagon and "move" the wagon as a projection of the longitudinal wind F_{rBx} force and the weight of the loaded wagon in the direction of the wagon rolling along the Ox_1 axis are found in the form:

$$F_{sd.x1} = G \sin(\psi_{03}) + F_{rBx} \cos(\psi_{03}),$$

here G is the weight of the freight car, taking into account the inertia of the rotating masses, kN .

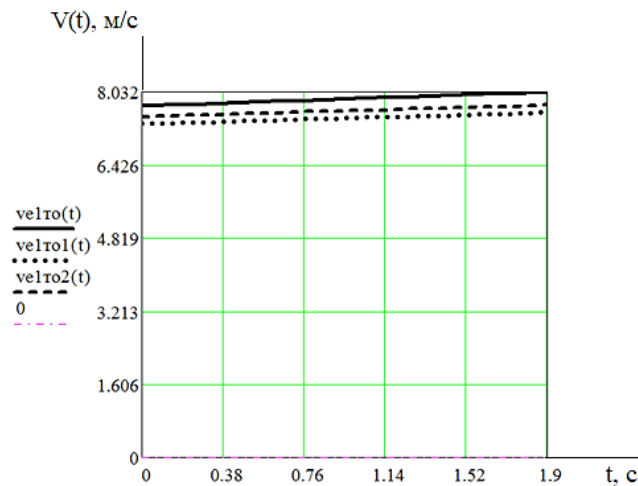


Figure 2. - The dependence graph of the speed of the wagon after braking at an arbitrary moment of movement Time on the hill 1BZ plot.

As you can see, the graphical dependence of the time of movement of the hill from the length of the 1BZ plot after braking $l_{jto} = l_{1to}$, for example in l_{2to} and l_{3to} , the appearance of the speed of rotation of the Wagon Wheel pair from the time of movement will be the same as shown in Fig.2. w_{okt} (t_{tok}) analysis of graphical dependencies, $t = 1.901$ s for a period of time only when the environmental resistance is affected by f_{sr} , the wagon speed decreases to a trivial value (fig. 2), the impact of wind from the sidewall of the carriage ranges from F_{rvy} (8.033 m/s (≈ 28.9 km/s) to 7,593 m/s (≈ 27.3 km/s)), i.e. by about 5.5%. In this, the wagon covers a distance of ≈ 15 m. Therefore, at the time of performing practical calculations, it is not necessary to take into account the wind effect F_{rvy} by the side of the wagon of a small value.

4. Conclusions and suggestions

1. Taking into account the inertia of the rotating masses (for example, $a_{1kb} = 0.168$ m/s²) the length of the Wagon Wheel Base from the suspension site at l_{1kb} in an equally accelerated movement of the wagon, the calculated value of its linear acceleration made it possible to calculate the time of l_{1kb} at a known value of the distance traveled by during this time, the wagon on the given plot of the Hill performs an equally accelerated movement. t_{1kb} time made it possible to calculate the wagon speed of $V_{KB}(t_{kb})$ at the end of the plot under consideration (for example, $V(t_{1kb}) = 7,935$ m/s or 28.6 km/s).

2. In the parking lot plot, excluding rotating masses (e.g. $a_{1t} = 2,556$ m/s²) the calculated value of the linear acceleration of the wagon in an equally slowed motion, the known value of the time in braking is in TT (e.g. $t_{1t} = 2.0$ s), as well as for the built-in construction of the wagon decelerator.

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NUMERICAL STUDY OF FINE DEBRIS PARTICLES APPROACHING RAW COTTON IN PNEUMATIC CONVEYOR SYSTEM OF COTTON HARVESTER

Matchanov R.D.¹, Yuldashev A.T.², Kuldoshev D.A.², Djuraeva N.B.²

¹JSC "Agrokhim", Tashkent, Uzbekistan

²Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan

E-mail: don_02@mail.ru

Abstract. The article presents the results of experimental and computational-numerical studies of the mechanism of saturation of raw cotton with small debris particles when moving in the pneumatic conveyor system of a cotton harvester. The results of a numerical study of the influence of air flow speed on the movement of debris particles and raw cotton in the pipeline of a pneumatic transport system were obtained. The distance between lobules of raw cotton and small debris particles was determined, at which the debris particles reach the lobule and, penetrating inside the cotton fruit, saturate it. The article also presents the results of experimental studies conducted on a two-row cotton harvester MX-1.8, which confirmed that the saturation of raw cotton with small debris particles when moving in a pneumatic transport system depends on the airflow velocity and the distance of approach of the cotton lobules with small debris particles. The results of numerical calculations were obtained in the form of graphs using the MathCAD 15 programming environment.

Keywords: cotton harvester, pneumatic conveyor system, raw cotton, small debris particles, airflow, numerical calculation, mathematical model.

1. Introduction.

Currently, in Uzbekistan, great importance is attached to the collection of raw cotton by cotton-harvesting machines with horizontal and vertical spindles. One of the main requirements of machine picking is to collect cotton with the least amount of impurities. If the content of small debris particles in raw cotton is more than 12%, then one-time cleaning provided by the cleaning technology is insufficient to reduce the impurity in raw cotton, as a result of which the frequency of cleaning increases. As studies conducted by the Central Scientific Research Institute of Chemical Industry have shown, with an increase in the frequency of cleaning raw cotton using saw cleaners, used in fiber cleaning shops, the content of litter in the cotton fiber reduces. However, the proportion of skins from the stems of cotton plants and knocked down seeds in cotton fiber increases. As a result, the amount of fiber defects increases, which reduces the quality of products produced by textile enterprises. In this regard, the removal of debris particles from raw cotton and cotton fiber is a labor-intensive technological process.

As is known, under increasing impurity, the quality of raw cotton deteriorates and the costs of its processing increase [1-6]. Ultimately, this leads to an increase in the cost of cotton fiber. The studies conducted by many famous scientists are devoted to reducing the impurities in raw cotton during machine harvesting [7-17]. The scientists focused their efforts on the influence of the quality of defoliation, the design features of the spindles and harvesting apparatus, and technological

operating modes, on the impurity of raw cotton with debris particles during machine harvesting. Models were substantiated for assessing the effect of the main parameters of vertical-spindle harvesting devices and changes in the parameters of the field's agricultural background on the average impurity of harvested raw cotton using empirical formulas.

The results of experimental studies [18] confirmed that the amount of fine litter when raw cotton is conveyed from the corridor to the hopper increases up to 50%. Using the receiving chamber developed [19, 20], a reduction in clogging to 30% was achieved. However, the mechanism of saturation of raw cotton with small debris, as well as the reduction in the impurity of raw cotton based on data on the morphological composition of debris elements, have not been fully studied. Therefore, the need to study the mechanism of saturation of raw cotton with debris particles in the pneumatic conveyor system of a cotton harvester is an urgent scientific problem.

This research aims to study the mechanism of saturation of raw cotton with small debris in the pneumatic conveyor system of a cotton harvester and to develop prerequisites for its reduction.

2. Materials, methods, and object of study.

Experimental studies were conducted on a double-row cotton harvester MX-1.8. Computational studies of the saturation of raw cotton with small debris were performed based on modeling the movement of cotton lobules and debris in a pneumatic transport system using the *MathCAD15* programming environment.

3. Results and discussions.

To determine the close-in movement of cotton lobules and small debris, the model shown in Fig. 1 was applied.

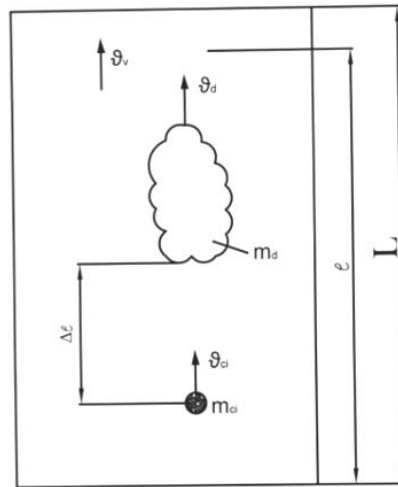


Figure 1. – **Model of the movement of cotton lobules and small debris particles**

We investigate the saturation of raw cotton with debris particles in the pipeline of a pneumatic transport system. According to Fig. 1, to simulate the movement of raw cotton and small debris as they move in the pipeline of a pneumatic transport system, we take the following indicators:

- L – distance from the receiving chamber to the hopper ($L = 4.5 \div 5.0$ m);
- ℓ – the distance of closing-in between the debris particles and the cotton lobules;
- $\Delta \ell$ – the initial distance from debris particles to cotton lobules ($\Delta \ell = 0.5 \div 1.2$ m);
- m_d – mass of a cotton lobule according to [10] $m_d = 1.0$ g;
- m_{ci} – mass of debris particles according to [3] $m_{ci} = 1.0$ mg;
- V_v – airflow speed, m/s;
- V_d – speed of a cotton lobule, m/s;
- V_{Ci} – speed of debris particles, m/s;
- V_{sd} – soaring speed of a cotton lobule, m/s;
- V_{sCi} – soaring speed of small debris particles, m/s.

The speeds of cotton lobules (V_d) and small debris particles (V_{SCi}) are determined as:

$$V_d = V_v - V_{sd} \quad (1)$$

$$V_{Ci} = V_v - V_{SCi} \quad (2)$$

According to [4], the experimental and calculated soaring speeds of a cotton lobule are: $V_d = 4.69 \div 5.02$ m/s. In the cotton ginning industry, the delay coefficient was experimentally determined, which is equal to the ratio of the cotton speed to the air speed [3]:

$$k = 0.7 \div 0.75 \quad (3)$$

Let t_0 be the time of closing-in of small particles with cotton lobules when passing distance ℓ from the receiving chamber to the hopper, which is L . Then the time of approach t_0 for the lobule is:

$$\frac{\ell}{V_d} = t_0, \quad (4)$$

for small debris particles, it is:

$$\frac{(\Delta\ell + \ell)}{V_{Ci}} = t_0. \quad (5)$$

Equating expressions (4) and (5), we obtain:

$$\frac{\ell}{V_d} = \frac{(\Delta\ell + \ell)}{V_{Ci}}, \quad (6)$$

hence

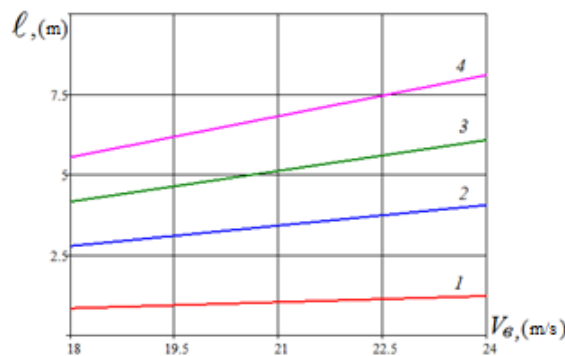
$$\ell = \frac{V_d \cdot \Delta\ell}{(V_{Ci} - V_d)}, \quad (7)$$

or

$$\ell = \frac{(V_v - V_{sd}) \cdot \Delta\ell}{(V_v - V_{SCi}) - (V_v - V_{sd})}. \quad (8)$$

To perform a numerical study according to expression (8), we take the following values of air speed $V_v = 18 \div 24$ m/s, the soaring speed of debris particles $V_{SCi} = 0.3$ m/s, distance from debris particles to a cotton lobule $\Delta\ell = 0.3 \div 2$ m. Considering the above data, a graph was plotted in the *MathCAD 15* programming environment, presented in Fig. 2.

Analyzing the graphs in Fig. 2, we see that when the airflow speed $V_v = 18 \div 24$ m/s and the distance of small debris particles to raw cotton lobules $\Delta\ell = 0.3 \div 2$ m change, the distance of approach of small particles to raw cotton lobules is $\ell = 0.83 \div 8.08$ m.

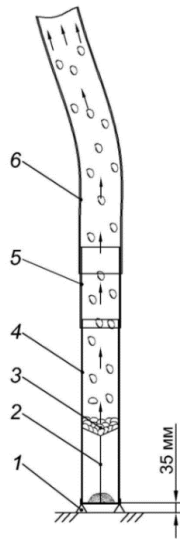


1 – $\Delta\ell = 0.3$ m/s, 2 – $\Delta\ell = 1.0$ m/s,
3 – $\Delta\ell = 1.5$ m/s, 4 – $\Delta\ell = 2.0$ m/s

Figure 2. – Graph of changes in the closing-in distance (m) of debris particles and cotton lobules depending on the air speed V_b (m/s) at different values of the distance from debris particles to raw cotton lobules $\Delta\ell$ (m)

When the distance $\Delta\ell = 0.3 \div 1.0$ m and the airflow speed $V_v = 18 \div 24$ m/s change, the saturation of raw cotton with small debris particles is ensured. For $\Delta\ell = 1.5$ m and $V_v \leq 20.4$ m/s, raw cotton is also saturated with small debris particles. For $\Delta\ell = 2.0$ m for all values of airflow velocity $V_v = 18 \div 24$ m/s, raw cotton is not saturated with small debris particles since the distance of closing-in of particles and cotton lobules satisfies the inequality $\ell \geq L$.

To confirm the numerical studies, full-scale tests were conducted, as shown in Figs. 3, 4, and 5. Let us present the procedure for conducting full-scale tests. With the engine of the MX-1.8 cotton picker running and the centrifugal fan turned off, we place(in a glass pipe) hand-picked raw cotton (Fig. 3, b) and small debris particles, the distance between which is 0.3 m. Air intake is ensured due to the gap in stand 1 (Fig. 3, a).



1 – pipe stand to provide air suction, 2 – holder for bundles of raw cotton and small debris,
3 – bundle of raw cotton, 4 – glass pipe, 5 – plastic pipe, 6 – corrugation

Figure 3. – **Schematic representation of a glass pipe (a) and its experimental version (b) for full-scale testing**

Next, we turn on the centrifugal fan. Under the operating mode of the centrifugal fan, the air suction speed in the pipeline is 22 m/s. Under the influence of the airflow, raw cotton with small debris particles is discharged into the hopper of a cotton harvester (Fig. 4).



Figure 4. – **Hopper of a cotton harvester with scattered bunches of raw cotton**

Scattered bunches of raw cotton are collected from the hopper, and the saturation of cotton lobules with small debris particles is analyzed. Figure 5 shows raw cotton lobules before and after testing. Analysis of the results obtained confirmed that, despite the fact that the distance between small debris particles and raw cotton lobules was 0.3 m, a certain amount of debris particles reached the cotton lobules and penetrated the inner area of the cotton fruit.



Figure 5. – Raw cotton lobules before testing (a) and after testing saturated with small debris particles (b)

4. Conclusions.

1. The raw cotton impurities during machine harvesting are determined by the degree of saturation of the raw cotton lobules with small debris in the pipelines of the pneumatic conveyor system of the cotton-harvesting machine.

2. It was determined that at an initial distance from the raw cotton lobule in the pipeline equal to $0.3 \div 2.0$ m, the distance of approaching the debris particles and the cotton lobules varies from 0.83 m to 8.08 m.

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DYNAMIC MASS-INERTIAL CHARACTERISTICS OF THE ACTION OF A THREE-POINT ENGINE OF A MOBILE MACHINE TAKEN INTO ACCOUNT OF CHANGING LOADS

Matmurodov F.M.¹

¹Chirchik Higher Tank Command Engineering School, Chirchik, Uzbekistan

E-mail: matmurodov@yahoo.com

Abstract. The paper examines the dynamic characteristics of the spatial motion of an engine with a passive type damper. Systems of spatial differential equations for transverse-longitudinal and angular vibrations are described. It was determined numerically that at various values the damping coefficient and stiffness of the engine damping devices. The vibrations along the coordinate x, y, z axes of the engine support along the coordinate axes with the existing and improved damper were experimentally studied.

Keywords: engine, damper, inertial characteristics, dynamics, stability, transverse-longitudinal vibration, angular vibration.

1. Introduction.

In the 21st century, engine designers have achieved significant increases in torque in the lower rev range. But at the same time, the load on engine parts also increases, despite the fact that there is a requirement to reduce their weight. All this increases the torsional deformation of the crankshaft, as well as the risk of resonance phenomena in mobile vehicles, including armored personnel carriers, tanks and cars.

Vibrations of mobile vehicles and their components occur due to excessive engine speed and uneven roads when driving. Reducing and optimizing engine vibrations is considered a scientific and technical problem; therefore, in this work we are exploring the issues of optimizing the vibration-resistant parameters of mobile machines.

The damper is needed to smooth out and compensate for torsional vibrations of the engine crankshaft. Such vibrations occur in the engine due to the uneven impact on the shaft from the connecting rod and piston group. On the one hand, the damper protects the engine from damage due to rotational resonance of the crankshaft. On the other hand, it protects belts and drive components from jerks that occur due to uneven rotation of the shaft.

When using a conventional all-metal crankshaft pulley instead of a damper pulley, vibrations are transmitted to the auxiliary equipment, causing them to fail prematurely. In addition, this can even affect engine performance and subsequently call for overhaul.

Car owner complaints that may indicate failure of the pulley-damper may include complaints of unusual vibrations in the cabin, whistling drive belt, as well as unusual noise or knocking, the frequency of which varies with engine speed.

According to a statistical survey, over 80% of drivers who have installed additional shock absorber units on their cars are satisfied with their work. At the same time, there is a risk that you will fall into the opposite camp - every fifth car owner either did not feel the difference before and after installation of the damper, or was dissatisfied with its work.

It should be noted at once that this equipment is optional, its installation on the vehicle is not obligatory. It is positioned as a means of increasing the comfort of being in the car, helping to reduce driver fatigue (if it is a mini steering damper) and reducing the load on the engine (engine vibration damper).

Creation of an effective vibration protection system for the driver of a wheeled tractor is an actual task aimed at improving the driver's labor, ensuring safety. Acceleration on the seat and noise level in the driver's cab of a wheeled machine is strictly regulated by international standards. Therefore, world manufacturers of wheeled machines constantly improve vibration protection systems and introduce new noise-absorbing materials in the design of driver's cabs. Thus, studies [1-4] have determined the ranges of natural vibration frequencies of various human organs (Fig. 1), and also established the distributions of vibration frequencies by elements of the wheeled tractor structure (Fig. 2).

F.Dine and M.Mitschke pay special attention to low-frequency vibrations (1...8 Hz), which have the most harmful effect on the driver during driving and cause occupational diseases.

Studies have established that the natural frequency of human oscillations on the seat of modern wheeled machines is 1...3 Hz, and the permissible values of mean-square accelerations and amplitudes of vertical movements of the seat should not exceed 8.5 m/s² and 0.02 m respectively, which is stipulated by GOST 12.1.012-90 and EEC Council Directive No. 78/764.

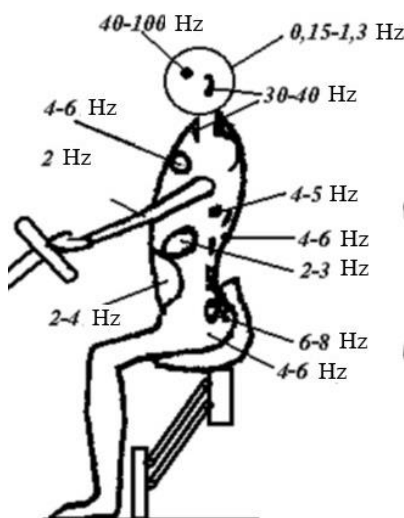


Figure 1. – Frequency distribution of natural vibrations

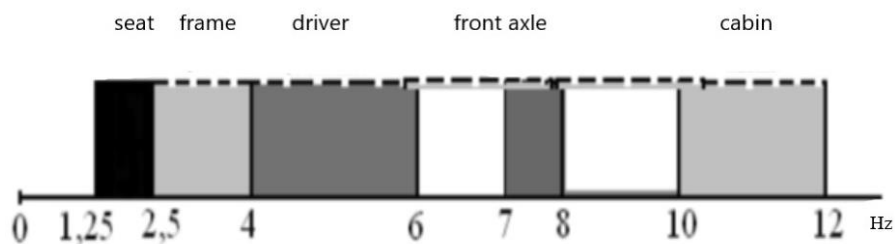


Figure 2 -Distribution of vibration frequencies by elements of wheeled machine structure

2. Materials, methods and objects of research.

There are numerous publications on theoretical and experimental studies of ball ABD, for example, works [5-10], in which the conditions of existence of modes of autobalancing of rotors with different unbalance are considered. In the majority of theoretical works on the basis of mathematical models obtained by the Lagrangian method, the autobalancing modes at steady-state rotational frequencies are analytically investigated with the construction of their stability regions in the space of design parameters on the example of rotors with double-ball ABD. At the same time, in theoretical studies, as a rule, the influence of gravity and non-viscous friction forces is not taken into account.

Numerical modeling of unsteady rotation modes of single-disk rotors with an autobalancing device (ABD) is considered in [13-16]. And in [13] the force of gravity is additionally considered, and in the source [14] - only the rolling friction force of balls in the ABD cage. In [15] the results of parametric analysis of the dynamics of a single-disk rotor with a two-ball ABD obtained by numerical solution of the dimensionless equations of motion without taking into account the forces of gravity and non-viscous friction of the balls are given, and in the source [16] the transitions of the rotor speed with a "non-ideal" cage of a two-ball ABD through the critical frequency are modeled. If the mechanism of occurrence of the autobalancing effect by ball ABU at supercritical rotational frequencies of the rotor and its absence at precritical frequencies are studied and described, in particular, in the sources, then, judging by available publications, the influence of the dynamic behavior of balls in the cage on the dynamics of the rotor at its transient and various stationary modes of rotation, including taking into account the forces of gravity and non-viscous friction of balls, remains poorly investigated.

3. Results of the study.

Let us describe the systems of differential equations spatial transverse longitudinal and angular vibrations [17-20]:

-considering the mass parameter

$$\begin{aligned} M_{dv}\ddot{x}_c + h_x\dot{x}_c + c_x x_c - h_{x\beta}\dot{\beta}a_{x\omega} - c_{x\beta}\beta b_{x\beta} + \ddot{s}_u s_{xa} &= 0; \\ M_{dv}\ddot{y}_c + h_y\dot{y}_c + c_y y_c - h_{y\alpha}\dot{\alpha}a_{y\omega} - c_{y\alpha}\alpha b_{y\alpha} + \ddot{s}_v s_{ya} &= 0; \\ M_{dv}\ddot{z}_c + h_z\dot{z}_c + c_z z_c - h_{z\gamma}\dot{\gamma}a_{z\omega} - c_{z\gamma}\gamma b_{z\gamma} + \ddot{s}_e s_{za} &= 0; \end{aligned}$$

here s_{xa} , s_{ya} , s_{za} - control coefficients of axial x, y, z angular acceleration; $a_{x\omega}$, $a_{y\omega}$, $a_{z\omega}$ - control coefficients of axial x, y, z angular velocity; $b_{x\beta}$, $b_{y\alpha}$, $b_{z\gamma}$ - control coefficients on axial x, y, z angles.

-considering inertial parameters

$$\begin{aligned} J_{u_c}\ddot{\alpha} + (J_{u_c} + h_a)\dot{\alpha} + c_\alpha\alpha - J_{u_c v_c}\dot{\beta} - (J_{u_c v_c} - c_\Sigma\omega)\dot{\beta} + h_{y\beta}\dot{y}_c a_{\beta\omega} - c_{y\beta}y_c b_{\beta\beta} + y_a \ddot{s}_u s_{\beta a} &= 0, \\ J_{v_c}\ddot{\beta} + (J_{v_c} + h_\beta)\dot{\beta} + c_\beta\beta - J_{v_c e_c}\dot{\alpha} - (J_{v_c e_c} - c_\Sigma\omega)\dot{\alpha} + h_{x\alpha}\dot{x}_c a_{\alpha\omega} - c_{x\alpha}x_c b_{\alpha\alpha} + x_a \ddot{s}_v s_{\alpha a} &= 0, \\ J_{e_c}\ddot{\gamma} + (J_{e_c} + h_\gamma)\dot{\gamma} + c_\gamma\gamma - J_{e_c u_c}\dot{\beta} - (J_{e_c u_c} - c_\Sigma\omega)\dot{\beta} + h_{z\gamma}\dot{z}_c a_{\gamma\omega} - c_{z\gamma}z_c b_{\gamma\gamma} + z_a \ddot{s}_e s_{\gamma a} &= 0, \quad (1) \end{aligned}$$

here $s_{\beta a}$, $s_{\alpha a}$, $s_{\gamma a}$ - regulating coefficients of angular acceleration at angles α, β, γ ; $a_{\beta\omega}$, $a_{\alpha\omega}$, $a_{\gamma\omega}$ - regulating coefficients of angular velocity at angles α, β, γ ; $b_{\beta\beta}$, $b_{\alpha\alpha}$, $b_{\gamma\gamma}$ - regulating coefficients of axial dimensions x, y, z at angles α, β, γ . $y \propto x \propto \beta \propto \gamma$

$$\begin{aligned} \ddot{s}_\xi + \frac{h_a}{m}\dot{s}_\xi + m\ddot{x}_a \sum_{j=1}^n \sin\theta_j \sin\alpha_j - m\ddot{y}_a \sum_{j=1}^n \cos\theta_j \sin\alpha_j - m\ddot{z}_a \sum_{j=1}^n \sin\theta_j \cos\alpha_j &= 0, \\ \ddot{s}_\eta + \frac{h_a}{m}\dot{s}_\eta - m\ddot{x}_a \sum_{j=1}^n \sin\theta_j \cos\alpha_j + m\ddot{y}_a \sum_{j=1}^n \cos\theta_j \cos\alpha_j - m\ddot{z}_a \sum_{j=1}^n \cos\theta_j \sin\alpha_j &= 0, \\ \ddot{s}_u + \frac{h_a}{m}\dot{s}_u - m\ddot{x}_a \sum_{j=1}^n \cos\theta_j \cos\alpha_j - m\ddot{y}_a \sum_{j=1}^n \sin\theta_j \cos\alpha_j + m\ddot{z}_a \sum_{j=1}^n \cos\theta_j \cos\alpha_j &= 0, \quad (2) \end{aligned}$$

где $c_x = c_y = c_z = c_L + c_R + c_p$; $c_\alpha = c_\beta = c_\gamma = c_L l_{cL}^2 + c_R l_{cR}^2 + c_p l_{cp}^2$; $c_{y\alpha} = c_{x\beta} = c_{z\gamma} = c_L l_c + c_R l_{cR} + c_p l_{cp}$; $h_x = h_y = h_z = h + h_R + h_p$ $h_\alpha = h_\beta = h_\gamma = h_L l_{cL}^2 + h_R l_{cR}^2 + h_p l_{cp}^2$; $h_{y\alpha} = h_{x\beta} = h_{z\gamma} = h_L l_c + h_R l_{cR} + h_p l_{cp}$;

$$\begin{aligned} M_s &= M_r + m_\Sigma = M_r + m_0 + nm; A_s = A_r + M_r z_G^2 + (m_0 + nm) z_a^2; \\ J_{u_c} &= A_s + \sum_{j=0}^n m_j R_j^2 \sin^2\theta_j; J_{v_c} = A_s + \sum_{j=0}^n m_j R_j^2 \cos^2\theta_j; \end{aligned}$$

$$J_{e_c} = A_s + \sum_{j=0}^n m_j R_j^2 \sin^2 \theta_j;$$

For the present example, we conditionally consider

$$J_{u_c} = J_{v_c} = J_{e_c} = A_s + 1/3 M_{dv} R^2 \sin^2 \theta; J_{u_c v_c} = J_{e_c u_c} = J_{e_c v_c} = A_s + 1/3 M_{dv} R^2 \sin \theta \cos \theta;$$

$$J_{u_c v_c} = A_s + \sum_{j=0}^n m_j R_j^2 \sin \theta_j \cos \theta_j; J_{e_c u_c} = A_s + \sum_{j=0}^n m_j R_j^2 \sin \theta_j \sin \theta_j;$$

$$J_{e_c u_c} = A_s + \sum_{j=0}^n m_j R_j^2 \cos \theta_j \cos \theta_j;$$

$$C_\Sigma = C_r + \sum_{j=0}^n m_j R_j^2 = C_r + m_0 R_0^2 + n m R^2;$$

$$\theta_j = \omega t + \alpha_j; x_a = x_c + z_a \beta; y_a = y_c + z_a \alpha; z_a = z_c + x_a \gamma; \quad (3)$$

$$s_u = \sum_{j=0}^n m_j R_j \cos \varphi_j; s_v = \sum_{j=0}^n m_j R_j \sin \varphi_j; s_e = \sum_{j=0}^n m_j R_j \cos \varphi_j;$$

$$s_\xi = \sum_{j=0}^n m_j R_j \cos(\alpha_j + \psi_j); s_\eta = \sum_{j=0}^n m_j R_j \sin(\alpha_j + \psi_j); s_u = \sum_{j=0}^n m_j R_j \cos \varphi_j; s_u = s_v$$

$$= s_e = M_{dv} R \sin \theta;$$

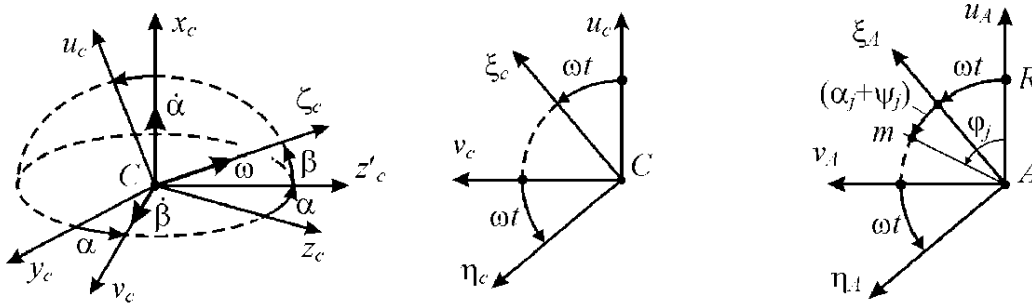
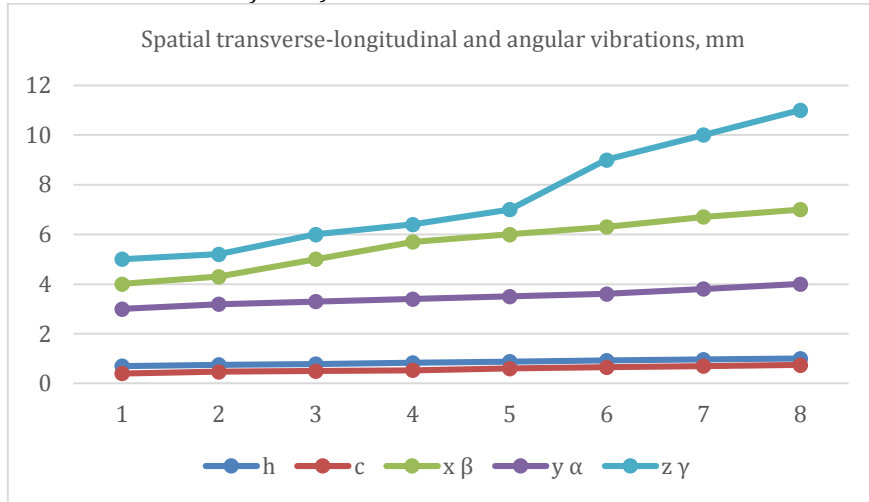


Figure 3. – Kinematics of spatial vibration motion of the motor

Equations (1) describe the transverse and angular vibrations of the system in a stationary coordinate system. Equations (2) describe the autobalancing motion of the system.

Moments of inertia with respect to axes rigidly connected to the motor:

$$m_\Sigma = \sum_{j=0}^n m_j = m_0 + n m; M_s = M_r + m_0 + n m;$$



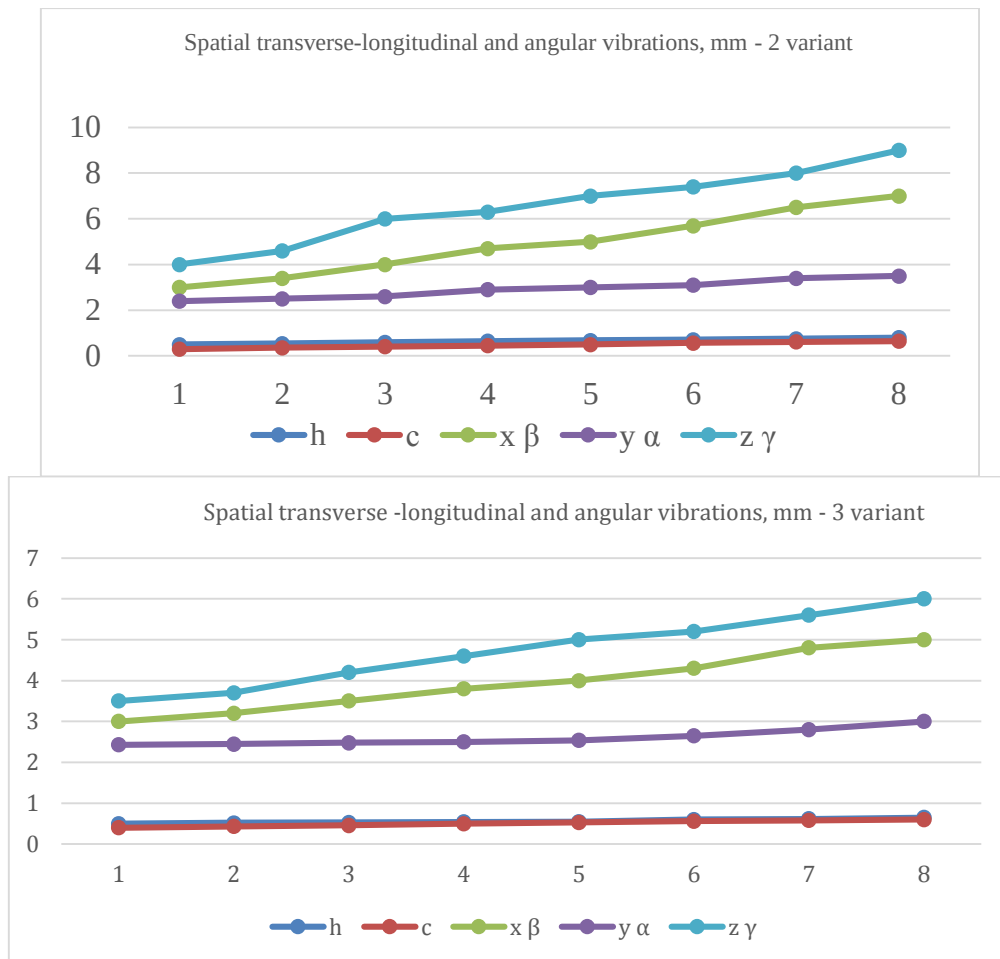


Figure 4. – **Dynamic mass-inertia characteristics of three-axis motor of mobile machine on different damping coefficient and stiffness**

Numerical experiments have been carried out dynamic mass-inertia characteristics of a three-axis motor of a mobile machine with different damping coefficient and stiffness (Fig.1). On the first variant, the existing damping devices of the motor are considered. From the figure it can be seen that, at eight times the damping ratio and stiffness of the motor damping devices gradually increases 0.4 to 1, and the transverse-angle vibrations $x\beta$, $y\alpha$, $z\gamma$ also increases from 3 to 11 mm.

In the second damping embodiment, the damping and stiffness ratio gradually increases 0.3 to 0.8, and the transverse-angle vibrations $x\beta$, $y\alpha$, $z\gamma$ also increase from 2.4 to 9 mm. In the third damping variant, the damping ratio and stiffness gradually increases 0.4 to 0.65, and the transverse-angle oscillations $x\beta$, $y\alpha$, $z\gamma$ also increase from 2.43 to 6 mm.

We present experimental measurements of the ejection in x, y, z coordinate axes over the left engine mount in coordinate axes with the existing and improved damper.



Figure 5. – **Advanced multi-component dampers**

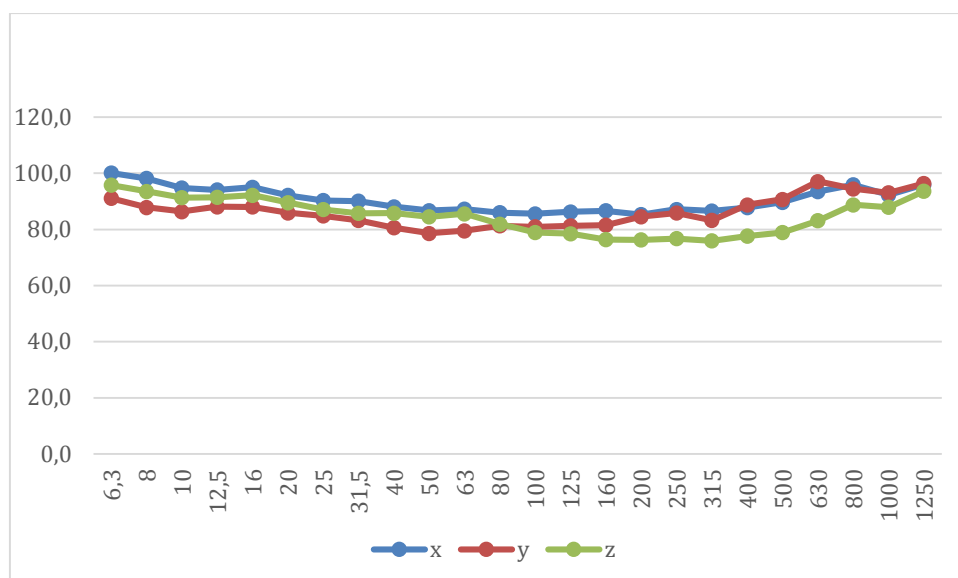


Figure 6. – **Vibration above the left engine support in coordinate axes with existing damper**

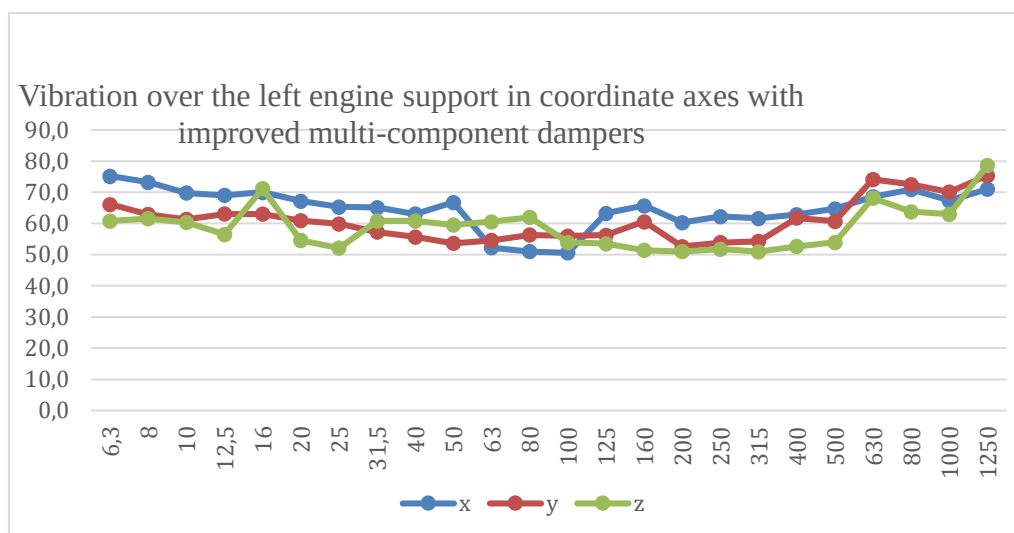


Figure 7. – **Vibration over the left engine support in coordinate axes with improved multi-component dampers**

The vibration above the left engine mount in coordinate axes with the improved damper is about 25% less vibration than with the existing damper.

4. Conclusions.

It is determined by numerical experiment that at different values of the damping coefficient and stiffness of the engine damping devices change 0.4 to 1, and the transverse-angle vibrations $x\beta$, $y\alpha$, $z\gamma$ also grow from 3 to 11 mm.

The third variant of damping devices is considered to be a rational parameter, in which the damping and stiffness coefficient varies 0.4 to 0.65, and the transverse-angle oscillations $x\beta$, $y\alpha$, $z\gamma$ are averaged, the oscillations are about 4.5 mm per second.

The most effective option for neutralizing engine vibrations is the use of a torque vibration damper (also called a crankshaft pulley with damper). These are common in diesels, but are also found in powerful gasoline engines. As a result of the installation, the auxiliary equipment is largely relieved of dynamic loads.

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ANALYTICAL DYNAMIC SIMULATION OF MULTI-HIERARCHICAL MECHANICAL SYSTEM AND ENGINE SHAFT-DAMPER-CASE OF MOBILE MACHINE USING MULTIMASS RHEOLOGICAL MODELS

Matmurodov F.M.¹

¹Chirchik Higher Tank Command Engineering School, Chirchik, Uzbekistan

E-mail: matmurodov@yahoo.com

Abstract. The article analytically models a multi-hierarchical mechanical system based on the method of multi-mass rheological models. Using this model, the ways of use are shown in the form of an example, an engine shaft-damper-housing of a mobile machine was obtained and modeled. The developed model makes it possible to find the amplitude-frequency characteristics of relative movements, vibration velocity and vibration acceleration of moving mechanisms of military mobile equipment.

Keywords: mobile technology; hierarchical mechanical system; multi-mass rheological model; motor shaft; damper; frame; armored personnel carrier; moving; vibration velocity; vibration acceleration.

1. Introduction.

The creation of energy-saturated mobile machines is accompanied by an increase in their dynamic load from external influences, which are oscillatory in nature with variable frequency and amplitude. This causes the occurrence of increased oscillatory processes in the system soil - propulsion - motor transmission unit, and reduces productivity, worsens controllability, smoothness and quality of performance of specified technological operations, and also leads to the destruction of the soil structure and its compaction [1,2]. Dynamic processes arise as a result of uneven soil resistance and surface unevenness, as well as physical and mechanical properties and other external influences. Transformed in the transmission, all these influences are summed up and supplied to the engine crankshaft in the form of a certain function of the moment of resistance $M_c = f(t)$. V.P.Boltinsky identified two ways to reduce the influence of dynamic loads on the effective parameters of the engine. The first - operation is that during operation the tractor is aggregated in such a way that the engine load does not exceed a certain set value. The second is a change in the design of individual mechanisms of the tractor or engine to reduce the influence of fluctuations in external loads on the operational and fuel-economic indicators of the machine-tractor unit. The largest number of studies on this problem are devoted to the second method of reducing the dynamic load of external influences. For these purposes, various elastic damping devices are installed in the clutch, transmission, drive wheel drives, on power take-off shafts, towed and mounted devices of mobile machines [4]. The work [5,6] indicates that the introduction of elastic damping elements into the clutch reduces torsional vibrations of the engine, however, dynamic loads from external influences are reduced slightly. The most effective way to reduce the uneven traction resistance of aggregated agricultural machines, uneven field surfaces and physical and mechanical properties of the soil is to install elastic-damping drives closer to the drive wheels of mobile machines. The object of study was a universal row-crop tractor with various agricultural machines and transport units equipped with rigid axle shafts with a diameter of 50 mm and elastic-damping drives. To carry out the tests, the following were accepted as the main initial parameters characterizing the experimental conditions:

- load on the tractor hook, which was changed by loading a dynamometer laboratory based on a metrological vehicle and engaging various gears.

- the rigidity of the drives was changed by installing elastic-damping drives of the driving wheels (which were installed on the axle shafts, in front of the wheel gearbox).

The tractor was tested, comparative traction tests were carried out on a concrete road, it was equipped with sensors and strain gauge units and instruments:

- induction sensor on the drive wheels and the track measuring wheel;
- strain gauge couplings for measuring torques on axle shafts;
- accelerometers installed on drive axles;
- a flow meter to determine fuel consumption;
- time was determined by oscilloscope marks.

In shaft supports whose operating speeds are close to or higher than their natural frequency, additional damping is necessary to limit the amplitude of oscillations in near-resonance and resonance

modes when using rolling bearings. Currently, to reduce the level of vibration, elastically damper shaft supports with hydrodynamic dampers have found widespread use in mobile machines [7,8].

In [9–13], the authors developed designs and studied the parameters of roller technological machines. The rational parameters of working bodies and actuators of machines were theoretically determined. The technological parameters of mechanical processing of fibrous materials, using raw hides as an example, were experimentally determined and scientifically substantiated.

The use of dynamic vibration absorbers is the most efficient at sharp tuning when the frequency of external action coincides with the frequency of antiresonance. The article examines a method to determine the boundary frequencies of angular vibration damping[14-16]. The vibration absorber is an elastic coupling device of a planetary-type with weights, rigidly mounted to satellites. The obtained dependences can be used for solving the inverse problem of determining the tuning parameters of the absorber by boundary frequency values. The article presents a transfer function and amplitude frequency characteristics of the system with an elastic main drive shaft and a dynamic vibration absorber. The comparison of the amplitude-frequency characteristics of the system with an absorber and the amplitude-frequency characteristics of the system without an absorber demonstrate the efficiency of the proposed method of vibration damping

Theoretical and practical experimental investigations reveal that the spectral tightness is one of the basic characteristics of the road profile determine the forces acting on moving vehicle[17].. Although it became possible to establish the connections among them.

The moving mechanisms of military mobile vehicles, as a rule, operate at operating speeds exceeding the first, and sometimes even the second critical speed. The dynamic state of the engine, characterized by the spectrum of natural frequencies, the level of stator vibration, the amplitude of shaft oscillations, as well as the forces and dynamic stresses acting on the machine elements, is determined by the rigidity and damping in the supports. Therefore, it is necessary to optimize these parameters in order to meet the tactical and technical requirements for the product. The main reason for vibration in an internal combustion engine is the unstable operating mode when transmitting power to the transmission. In this case, excessive vibrations of mechanisms, including the engine shaft, and the housing on a military mobile vehicle, appear. In this case, damping means are considered an important way to solve this problem. And their analytical modeling is an urgent task.

2. Research methods.

Analytical modeling was carried out using the method of mathematical statistics, the method of complex amplitudes, and the principal moment of inertia theorem. To construct the calculated dependencies, the oscillation equations of the relaxation damping and sequential damping system were compiled.

3. Research results and discussions.

To model the calculated dependencies, it is necessary to compose oscillation equations for the damping system of the n-th hierarchical links under the influence of an exciting force $F_b = \Delta\omega^2 \sin\omega t$ [18-20]:

$$\begin{aligned} -m_k \ddot{x}_k + c_1 x_k &= K_1(\omega_1) x_k; \\ -m_d \ddot{x}_d + (c_2 + u_{dk} b_2 \omega_2)(x_d - x_k) &= 0 \\ -m_3 \ddot{x}_3 + K_3(\omega_3)(x_3 - x_d) &= 0 \\ \text{-----} \\ -m_{n-1} \ddot{x}_{n-1} + K_{n-1}(\omega_{n-1})(x_n - x_{n-1}) &= 0 \\ -m_n \ddot{x}_n + K_n(\omega_n) x_n &= F_b \end{aligned}$$

where $K_i(\omega_i) = \frac{c_{i+1}(c_i + u_{(i-1)i} b_i \omega)}{c_{i+1} + c_i + u_{(i-1)i} b_i \omega}$ – dynamic stiffness, $u_{(i-1)i}$ – gear ratio.

We dynamically simulate oscillations of a 3-level damping system under the influence of an exciting force (fig. 1)

$$\begin{aligned} -m_k \ddot{x}_k + c_1 x_k &= K_1(\omega_1) x_k, \\ -m_d \ddot{x}_d + (c_2 + u_{dk} b_2 \omega_2)(x_d - x_k) &= 0, \end{aligned}$$

$$-m_v \ddot{x}_v + K_v(\omega_v)(x_v - x_d) = F_B$$

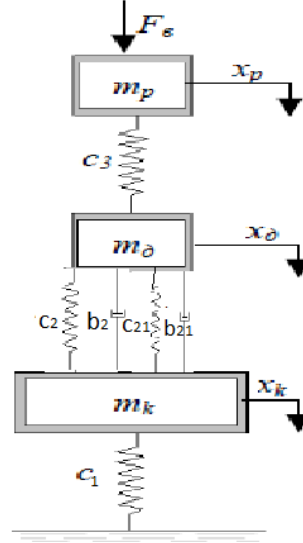


Figure 1. – Equivalent design diagram for damping a mobile machine engine

If a mechanical system has relaxation damping, it also has additional damping, and then damping x_d has the form

$$-m_d \ddot{x}_d + (c_2 + u_{dk} b_2 \omega_2)(x_d - x_k) + (c_{21} + u_{d21k} b_{21} \omega_{21})(x_{d21} - x_k) = 0$$

Let us introduce dimensionless parameters, denoting $k_1 = c_2/c_1$, $k_2 = c_3/c_2$, $\dots$, $k_{n-1} = c_n/c_{n-1}$; $m_{dk} = m_d/m_k$, $m_{vk} = m_v/m_k$, $\dots$, $m_{nk} = m_n/m_k$; $\eta_i = \omega_i/\omega_k$; $Q_i = (c_i/c_1)/\omega_k$; $\omega_k = \sqrt{c_1/m_k}$.

Applying the method of complex amplitudes, we obtain the damping oscillation equations in the following form

$$\begin{aligned} -m_k \eta_k^2 X_k + c_1 X_k &= K_1(\eta_1) x_k, \\ -m_d \eta_d^2 X_d + (c_2 + u_{dk} b_2 \eta_2)(X_d - Q_k) &= 0, \\ -m_3 \eta_3^2 X_3 + K_3(\omega_3)(X_3 - X_d) &= 0, \end{aligned}$$

$$\begin{aligned} -m_{n-1} \eta_{n-1}^2 X_{n-1} + K_{n-1}(\omega_{n-1})(X_n - X_{n-1}) &= 0, \\ -m_n \eta_n^2 X_n + K_n(\omega_n) X_n &= F_B, \end{aligned}$$

where $K_i(\eta_i) = c_i \omega_i^2 \frac{k_i(1-k_{i+1})(k_{i+1}Q_i^2 + \eta_i^2) + u_i k_i k_{i+1}^2 Q \eta_i}{Q_i^2(1+k_{i+1})\eta_i^2}$ – dynamic stiffness is a function of dimensionless frequency η ; X_k , X_d , X_n – complex amplitudes of movement of the body, damper and executive bodies.

According to this method, recording movement in complex form $x = X e^{i\omega t}$, allows you to transform the original equations to a system of linear equations.

As an example, using the complex amplitude method, we obtain the vibration equations of a 3-level damping system

$$-m_k \eta_k^2 X_k + c_1 X_k = K_1(\eta_1) x_k, \quad (1)$$

$$-m_d \eta_d^2 X_d + (c_2 + u_{dk} b_2 \eta_2)(X_d - Q_k) = 0, \quad (2)$$

$$-m_v \eta_v^2 X_v + K_v(\omega_v)(X_v - X_d) = 0, \quad (3)$$

Having solved equations (1)-(3), we determine the amplitudes of displacement of the body, damper and shaft as modules of the corresponding complex amplitudes

$$x_k = \frac{\Delta \eta_k^2}{m_k} \sqrt{[C^2(\eta_k) + D^2(\eta_k)]/B(\eta_k)},$$

$$x_d = \frac{\Delta \eta_d^2}{m_k} \sqrt{\frac{Q_d^2 k_d^2 + (1+k_d - \eta_d^2)^2 + k_d^2 k_v^2 \eta_d^2}{B(\eta_d)[Q_d^2(1+k_d)^2 + \eta_d^2]}},$$

$$x_v = \frac{\Delta \eta_v^2}{m_k} \sqrt{[(C^2(\eta_v) + 1 - \eta_v^2)^2 + D^2(\eta_v)]/B(\eta_v)}$$

$$x_{vo} = |x_v - x_k| = \frac{\Delta\eta_v^2}{m_k} |1 - \eta_v^2| / \sqrt{B(\eta_v)},$$

$$x_{do} = |x_d - x_k| = \frac{\Delta\eta_d^2}{m_k} |1 - \eta_d^2| / \sqrt{\frac{Q_d^2 + \eta_d^2}{B(\eta_v)(1+k_d)^2\eta_d^2}},$$

$$B(\eta_i) = [C(\eta_i)(1 - \eta_i^2 - m_{vk}\eta_i^2) + m_{vk}\eta_i^2(1 - \eta_i^2)]^2 + [D(\eta_i)(1 - \eta_i^2 - m_{vk}\eta_i^2)]^2,$$

$$C(\eta_i) = \frac{k_i(1-k_{i+1})(k_{i+1}Q_i^2 + \eta_i^2)}{Q^2(1-k_i) + \eta_i^2},$$

$$D(\eta_i) = \frac{k_i k_{i+1}^2 Q_i \eta_i}{Q_i^2(1-k_i) + \eta_i^2}.$$

Using the above, you can find the numerical values of the amplitude-frequency characteristics (AFC) of the housing, damper and motor shaft of a mobile machine x_k, x_d, x_v .

4. Conclusions.

From the analysis of the dynamics of a three-mass system, the following conclusions can be drawn that on the frequency response of the model's structural elements there are two resonances, the position and magnitude of which are determined by the mass-inertial and elastic-damping characteristics of the system. The compiled analytical dynamic model allows us to find the frequency response of relative movements, vibration velocity and vibration acceleration of the body, damper and engine shaft of a mobile machine.

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IMPROVEMENT AND RESEARCH OF A MACHINE FOR FIBROUS MATERIALS STAKING

Nabiev A.M.¹, Tsoy G.N.¹, Rakhimov F.R.¹, Rozaxinov A.S.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Uzbekistan Academy of Sciences*

E-mail: a.nabiev@mail.ru

Abstract. *An improved design of a drum staking machine of non-throughput type is proposed. The modified machine is designed for staking and snuffing the leather fabric of fur sheepskins and medium-sized hides. The designs of the working knives and the clamping device of the staker were improved. The parameters of the clamping device were studied. Equations are derived for calculating the reaction forces arising on the supports of the L-shaped clamping device. The reaction forces on the supports of the lever clamping device of machine are determined. Dependence graphs of the reaction force on the support on the angle formed between the initial and working positions of the pedal and the elastic element of the clamping device were obtained. The results of the study allow the optimal choice of the lifting angle of the working pedal of the staker's clamping device.*

Keywords: *staking machine, working drum, knives, clamping device, levers, reaction forces.*

1. Introduction.

To accelerate the production and export of finished leather and footwear products with high added value from leather raw material based on its deep processing, small industrial zones specialized in leather processing and production of finished leather footwear and fur products, are being created in the regions of Uzbekistan.

It is known that in the leather and fur industry, staking machines of throughput and non-throughput types are used [1]. In particular, the non-throughput drum staking machine is designed for staking and snuffing the leather fabric of fur sheepskins and medium-sized skins (for example, astrakhan fur, lambskin, etc.).

In the production of fur products, the RM-2M machine (Russia) is used, designed for staking and snuffing the leather fabric of sheepskins and medium-sized skins (astrakhan fur, lambskin).

The MRK machine (Russia) is also known for staking the leather fabric of fur skins [1]. By staking down the leather fabric, the required elastic-plastic properties are obtained.

A significant disadvantage of staking machines of this type is the relatively low level of quality of the knife due to its frequent repeated sharpening in tanneries and fur factories, and the low resource of the movable elastic support [2–5].

2. Materials, methods, and object of study.

An improved machine was developed for staking and snuffing the leather fabric of fur skins. The design of the working knives and the clamping device were improved in the developed unit. Figure 1 shows a general view of the staking machine, its side view. Figure 2 shows the front view of the machine and Figure 3 shows section A-A of the machine. Figure 4 shows a scheme of the knife of the staking machine. Figures 5 and 6 show improved designs of staking machine knives [6–9].

The working tools of the drum staking machine are flat steel knives, rigidly fixed in the radial grooves of a barrel-shaped cast-iron drum, the shaft of which rotates in bearings mounted on the machine frame.

The working edge of the knife has a semicircular contour, and the sharpening is one-sided. To increase the efficiency of staking down the processed fur skin, it is proposed to make the working edge of the knives segmented, as shown in Figures 5 and 6. Moreover, on the knife in Figure 6, the segments are tilted by approximately 11° relative to the segments in Figure 5.

The knives shown in Figures 5 and 6 are installed in the grooves of the drum in an alternating order.

Thus, when the drum rotates, the knives alternate by protrusions and depressions of the segments, and act on the processed fur skin, stretching it not only in length, but also in width from the middle of the drum to its ends, increasing the yield over the usable area [10–13].

The proposed staking machine, shown in Figure 1, consists of frame 1, staking drum 2, feeding table 3, in the middle part of which there is a longitudinal cutout in which movable elastic support 4 is placed, made in the form of a felt belt protected by leather, pedal 5 for supplying and pressing the table with the fur skin on it to the staking drum and dust suction device 6, folding safety shield 10 and drive 7 [1, 14–18].

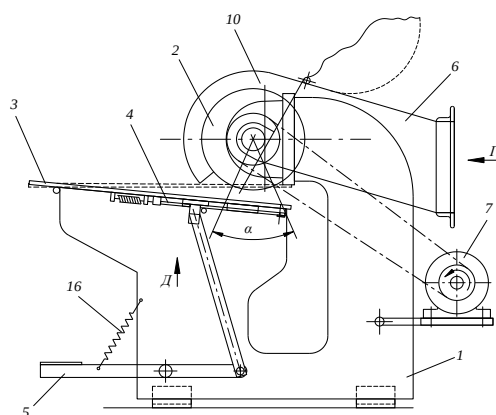


Figure 1. – General view of the drum staking machine

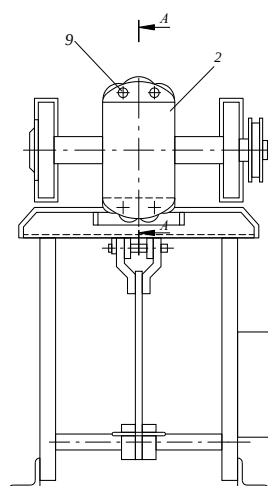


Figure 2. – Front view of the drum staking machine

Knives 8 (Figure 3) of the staking drum (10 pieces) are rigidly fixed in the radial grooves of the drum with bolts 9.

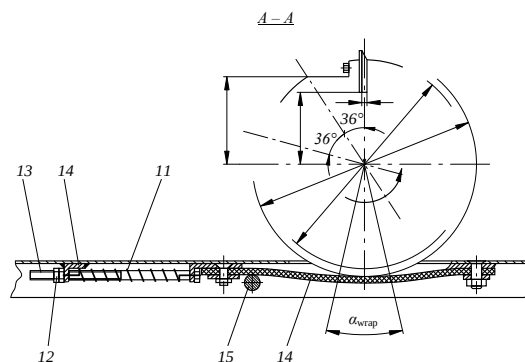


Figure 3. – Front view of the drum staking machine

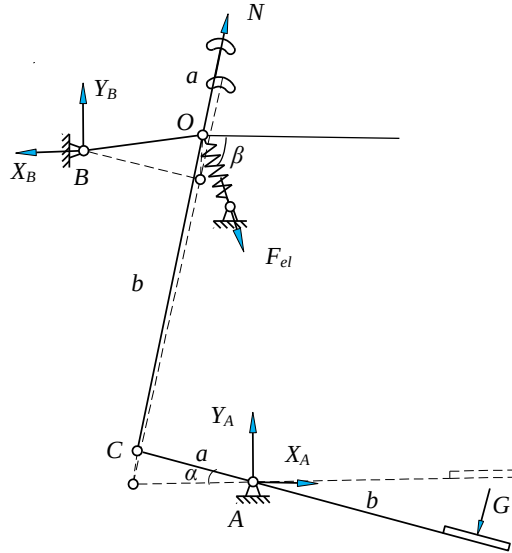


Figure 7. – Scheme for calculating the lever contour of the clamping device of the staking machine

According to the design scheme in Figure 7 of the lever contour of the clamping device of the staking machine, the static equilibrium conditions are used to determine the reaction forces arising on supports A and B:

$$\begin{aligned}
 \sum F_{kx} &= X_A - X_B + F_{el} \cdot \cos \beta - G \cdot \sin \alpha + N \cdot \sin \alpha = 0 \\
 \sum F_{ky} &= Y_A + Y_B - G \cdot \cos \alpha - F_{el} \sin \beta + N \cdot \cos \alpha = 0 \\
 \sum M_O &= -G \cdot (b + a) + Y_A \cdot (a - b \cdot \sin \alpha) + X_A \cdot (b \cdot \cos \alpha + a \sin \alpha) - \\
 &\quad - Y_B \cdot a \cos \gamma - X_B \cdot a \sin \gamma = 0
 \end{aligned} \tag{1}$$

here, X_A , X_B and Y_A , Y_B – are the reaction forces arising on the supports of the clamping device at points A and B; F_{el} – is the spring elastic force; G – is the force created by the operator on the support at point A; N – is the resulting force that occurs during the movement of the lever contour of the clamping device of the staking machine; a and b – are the lengths of the levers attached to the support at point A; α – is the angle formed between the lever AD and the X-axis; β – is the angle formed between the spring location and the X-axis.

Since the system of equations (1) consists of three equations and has four unknowns, it is impossible to solve this system of equations. Therefore, to determine these unknowns, the method of dividing the lever contour into parts was used (Figures 8, a, b).

For each part of the divided lever contour of the staking machine (according to Figures 8, a, b), equilibrium equations were derived using the static equilibrium condition.

According to scheme (a) in Figure 8, we will compose the equilibrium equations for the section located above the separation point C:

$$\begin{aligned}
 \sum F_{kx} &= X_C - X_B + F_{el} \cdot \cos \beta + N \cdot \sin \alpha = 0 \\
 \sum F_{ky} &= Y_C + Y_B - F_{el} \sin \beta + N \cdot \cos \alpha = 0 \\
 \sum M_O &= -Y_C \cdot b \cdot \sin \alpha + X_C \cdot b \cdot \cos \alpha - Y_B \cdot a \cos \gamma - X_B \cdot a \sin \gamma = 0
 \end{aligned} \tag{2}$$

Using scheme (b) in Figure 8, we will compile equilibrium equations for the section located below point C:

$$\begin{aligned}
 \sum F_{kx} &= X_A - X_C - G \cdot \sin \alpha = 0 \\
 \sum F_{ky} &= Y_A - Y_C - G \cdot \cos \alpha = 0 \\
 \sum M_A &= -G \cdot b + X_C \cdot a \cdot \sin \alpha - Y_C \cdot a \cdot \cos \alpha = 0
 \end{aligned} \tag{3}$$

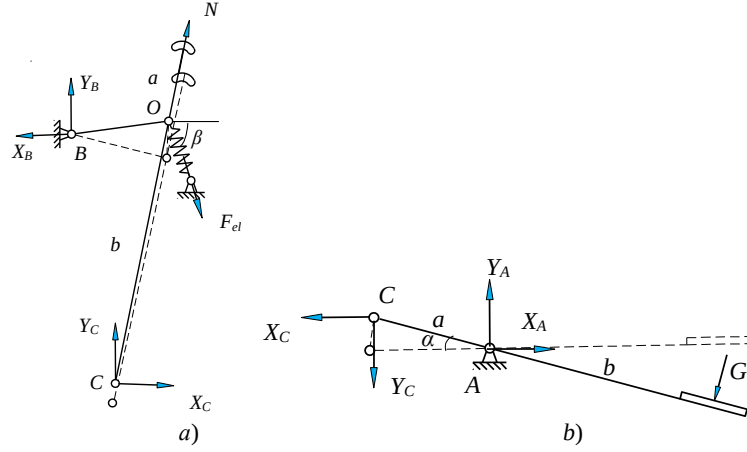


Figure 8. – Scheme for dividing and calculating the thrust (a) from the pedal (b) of the clamping device of the staking machine

Let us write the system of equations (2) in the following order:

$$X_C - X_B = -F_{el} \cdot \cos \beta - N \cdot \sin \alpha \quad (4)$$

$$Y_C + Y_B = F_{el} \sin \beta - N \cdot \cos \alpha \quad (5)$$

$$-Y_C \cdot b \cdot \sin \alpha + X_C \cdot b \cdot \cos \alpha = Y_B \cdot a \cos \gamma + X_B \cdot a \sin \gamma = 0 \quad (6)$$

Let us write the system of equations (3) in the following order:

$$X_A - X_C = G \cdot \sin \alpha \quad (7)$$

$$Y_A - Y_C = G \cdot \cos \alpha \quad (8)$$

$$X_C = Y_C \cdot \operatorname{ctg} \alpha + \frac{G \cdot b}{a \sin \alpha} \quad (9)$$

From equations (4) and (5), we obtain the following relations:

$$X_B = X_C + F_{el} \cdot \cos \beta + N \cdot \sin \alpha, \quad (10)$$

$$Y_B = F_{el} \sin \beta - N \cdot \cos \alpha - Y_C. \quad (11)$$

Substituting equations (10) and (11) into equation (6), we obtain the following relationship:

$$\begin{aligned} Y_C \cdot (a \cos \gamma - b \cdot \sin \alpha) + X_C (b \cdot \cos \alpha - a \sin \gamma) = \\ = (F_{el} \sin \beta - N \cdot \cos \alpha) \cdot a \cos \gamma + (F_{el} \cdot \cos \beta + N \cdot \sin \alpha) \cdot a \sin \gamma \end{aligned} \quad (12)$$

Substituting equation (9) into equation (12), we obtain the following equation:

$$\begin{aligned} Y_C \cdot (a \cos \gamma - b \cdot \sin \alpha) + Y_C \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma) + \frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma) = \\ = (F_{el} \sin \beta - N \cdot \cos \alpha) \cdot a \cos \gamma + (F_{el} \cdot \cos \beta + N \cdot \sin \alpha) \cdot a \sin \gamma \end{aligned}$$

By integrating the above equation, we can determine the value of the unknown reaction force.

$$Y_C = \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha) - \frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \quad (13)$$

Substituting equation (13) into equation (9), we obtain the following relation:

$$\begin{aligned} X_C = \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \cdot \operatorname{ctg} \alpha - \\ - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \cdot \operatorname{ctg} \alpha + \frac{G \cdot b}{a \sin \alpha} \end{aligned} \quad (14)$$

To determine the value of the reaction force arising at point A and directed along the OX-axis, we substitute equation (14) into equation (7) and write the following:

$$X_A = \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \cdot \operatorname{ctg} \alpha -$$

$$- \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma) \cdot \operatorname{ctg} \alpha}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \frac{G \cdot b}{a \sin \alpha} + G \cdot \sin \alpha$$

$$(15)$$

To determine the value of the reaction force arising at point A and directed along the OY-axis, we substitute equation (13) into equation (8) and write the following:

$$Y_A = \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} -$$

$$- \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + G \cdot \cos \alpha$$

$$(16)$$

The value of the total reaction force arising at point B is determined by the following equation:

$$R_A = \sqrt{X_A^2 + Y_A^2}.$$

$$R_A = \sqrt{\left(\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \cdot \operatorname{ctg} \alpha + \frac{G \cdot b}{a \sin \alpha} - \right.}$$

$$\left. - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \operatorname{ctg} \alpha + G \cdot \sin \alpha \right)^2 +$$

$$+ \left(\frac{a \cos \gamma \cdot ((F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha))}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right)^2$$

$$+ \left(- \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + G \cdot \cos \alpha \right)^2$$

$$(17)$$

We can simplify equation (17) and write it in the following form:

$$R_A = \sqrt{\left(\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \right.}$$

$$\left. - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right)^2 (1 + \operatorname{ctg} \alpha)^2 +$$

$$+ 2 \cdot G \cdot \left(\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \right.}$$

$$\left. - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right) \cdot \frac{b \cdot \cos \alpha}{a \sin^2 \alpha} +$$

$$+ 4 \cdot G \cdot \left(\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \right.}$$

$$\left. - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right) \cdot \cos \alpha +$$

$$+ \left(\frac{G \cdot b}{a \sin \alpha} \right)^2 + 2 \cdot \frac{G^2 \cdot b}{a} + (G \cdot \sin \alpha)^2 + (G \cdot \cos \alpha)^2$$

$$(18)$$

By introducing a series of trigonometric substitutions into equation (18), we obtain the following equation.

$$\begin{aligned}
R_A = & \left[\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{(a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)) \cdot \sin \alpha} - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{(a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)) \cdot \sin \alpha} \right]^2 + \\
& + 2 \cdot G \cdot \left[\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right] \cdot \frac{b \cdot \cos \alpha}{a \sin^2 \alpha} + \\
& + 4 \cdot G \cdot \left[\frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \right] \cdot \cos \alpha + \\
& + \left(\frac{G \cdot b}{a \sin \alpha} \right)^2 + 2 \cdot \frac{G^2 \cdot b}{a} + 1
\end{aligned} \tag{19}$$

Equation (19) allows us to calculate the reaction forces arising at point A.

3. Results.

Based on the obtained equation (19) and using the calculation and graphic program Maple 2018, we plot the following graph (Figure 9) with the set values $a=16$ cm, $b=32$ cm, $F_{el} = 50$ N, $N = 140$ N, $G = 70$ N, $\gamma=(1/18) \pi$.

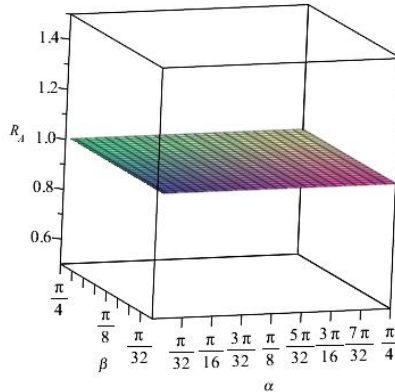


Figure 9. – Graph of changes in reaction forces on the support at point A at the angle formed between the initial and working positions of the pedal and the spring of the clamping device of the staking machine

By substituting expression (14) into equation (10), we can find the horizontal component of the reaction force created at point B:

$$\begin{aligned}
X_B = & \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \cdot \text{ctg} \alpha - \\
& \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \cdot \text{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \frac{G \cdot b}{a \sin \alpha} + F_{el} \cdot \cos \beta + N \cdot \sin \alpha
\end{aligned} \tag{20}$$

Let us simplify equation (20) and write it in the following form:

$$\begin{aligned}
X_B = & \frac{a \cos \gamma F_{el} \sin \beta \operatorname{ctg} \alpha - a \cos \gamma N \cos \alpha \operatorname{ctg} \alpha + a \sin \gamma F_{el} \cdot \cos \beta \operatorname{ctg} \alpha + a \sin \gamma N \cdot \sin \alpha \operatorname{ctg} \alpha}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \\
& \frac{\frac{G \cdot b}{\sin \alpha} \cos \gamma - \frac{G \cdot b^2}{a} - \frac{G \cdot b}{\sin \alpha} \operatorname{ctg} \alpha \cdot \sin \gamma - \frac{G \cdot b}{\sin \alpha} \sin \gamma \cdot \operatorname{ctg} \alpha + N \cdot \sin \alpha \cos \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \\
& + \frac{F_{el} \cdot \cos \beta a \cos \gamma - F_{el} \cdot \cos \beta b \cdot \sin \alpha + F_{el} \cdot \cos \beta \operatorname{ctg} \alpha \cdot b \cdot \cos \alpha - F_{el} \cdot \cos \beta \operatorname{ctg} \alpha \cdot a \sin \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \\
& + \frac{-N \cdot \sin \alpha b \cdot \sin \alpha + N \cdot \sin \alpha \operatorname{ctg} \alpha \cdot b \cdot \cos \alpha - N \cdot \sin \alpha \operatorname{ctg} \alpha \cdot a \sin \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \\
X_B = & \frac{a F_{el} \cos \gamma (\sin \beta \operatorname{ctg} \alpha + \cos \beta) + b F_{el} \cos \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \\
& + \frac{a N \cos \gamma (\sin \alpha - \cos \alpha \operatorname{ctg} \alpha) + b N \cos 2\alpha + \frac{G b \cos \gamma}{\sin \alpha} - \frac{G b^2}{a}}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \quad (21)
\end{aligned}$$

By substituting expression (13) into equation (11), we can find the vertical component of the reaction force created at point B [23–25]:

$$\begin{aligned}
Y_B = & F_{el} \sin \beta - N \cdot \cos \alpha - \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \\
& + \frac{a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha) - \frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \quad (22)
\end{aligned}$$

Let us simplify equation (22) and write it in the following form [29–32]:

$$\begin{aligned}
Y_B = & F_{el} \sin \beta - N \cdot \cos \alpha - \frac{a \cos \gamma \cdot (F_{el} \sin \beta - N \cdot \cos \alpha) + a \sin \gamma \cdot (F_{el} \cdot \cos \beta + N \cdot \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \\
& - \frac{\frac{G \cdot b}{a \sin \alpha} (b \cdot \cos \alpha - a \sin \gamma)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} = \\
= & \frac{a F_{el} \sin \beta \cos \gamma - b F_{el} \sin \beta \cdot \sin \alpha + b F_{el} \sin \beta \operatorname{ctg} \alpha \cdot \cos \alpha - a F_{el} \sin \beta \operatorname{ctg} \alpha \cdot \sin \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \\
& - \frac{a N \cos \alpha \cos \gamma - b N \cos \alpha \cdot \sin \alpha + b N \cos \alpha \operatorname{ctg} \alpha \cdot \cos \alpha - a N \cos \alpha \operatorname{ctg} \alpha \cdot \sin \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} - \\
& - \frac{a F_{el} \cos \gamma \sin \beta - a N \cos \gamma \cos \alpha + a F_{el} \sin \gamma \cdot \cos \beta + a N \sin \gamma \cdot \sin \alpha}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} = \\
= & \frac{-a F_{el} \sin \gamma (\sin \beta \operatorname{ctg} \alpha + \cos \beta) + a N \sin \gamma (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) - \frac{G b}{\sin \alpha} \sin \gamma}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} + \\
& + \frac{b F_{el} \sin \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) + \frac{G b^2}{a} \operatorname{ctg} \alpha - b N \cos \alpha (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)} \quad (23)
\end{aligned}$$

The value of the total reaction force arising at point B is determined by the following equation

$$R_B = \sqrt{X_B^2 + Y_B^2}.$$

$$R_B = \frac{\sqrt{A}}{a \cos \gamma - b \cdot \sin \alpha + \operatorname{ctg} \alpha \cdot (b \cdot \cos \alpha - a \sin \gamma)}; \quad (24)$$

here

$$\begin{aligned}
A = & 2a^2 NF_{el} \cos 2\gamma (\sin \beta \operatorname{ctg} \alpha + \cos \beta) (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) + a^2 F_{el}^2 (\sin \beta \operatorname{ctg} \alpha + \cos \beta)^2 + \\
& + a^2 N^2 (\sin \alpha - \cos \alpha \operatorname{ctg} \alpha)^2 + b^2 F_{el}^2 (\sin \alpha - \cos \alpha \operatorname{ctg} \alpha)^2 + \\
& + 2b^2 NF_{el} (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) (\cos \beta \cos 2\alpha + \sin \beta \cos \alpha) + \\
& + b^2 N^2 (\cos^2 2\alpha + \cos^2 \alpha (\sin \alpha - \operatorname{ctg} \alpha \cos \alpha)^2) + \frac{G^2 b^4}{a^2} (1 + \operatorname{ctg}^2 \alpha) - 2 \frac{G^2 b^3}{a \sin \alpha} (\cos \gamma + \operatorname{ctg} \alpha \sin \gamma) + \\
& + \frac{G^2 b^2}{\sin^2 \alpha} + 2ab \left(\frac{N \sin \gamma (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) - F_{el} \sin \gamma (\sin \beta \operatorname{ctg} \alpha + \cos \beta)}{-F_{el} \sin \gamma (\sin \beta \operatorname{ctg} \alpha + \cos \beta)} \cdot \left(\frac{F_{el} \sin \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) - N \cos \alpha (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)}{-N \cos \alpha (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)} \right) \right) + \\
& + 2abG \sin \gamma \left(\frac{N \sin \gamma (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) - F_{el} (\sin \beta \operatorname{ctg} \alpha + \cos \beta)}{a} \operatorname{ctg} \alpha - \frac{\sin \gamma}{\sin \alpha} \right) + \\
& + 2b^2 G (F_{el} \sin \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) - N \cos \alpha (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)) \left(\frac{b}{a} \operatorname{ctg} \alpha - \frac{\sin \gamma}{\sin \alpha} \right) + \\
& + 2abG \cos \gamma \left(\frac{F_{el} (\sin \beta \operatorname{ctg} \alpha + \cos \beta) - N (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)}{\sin \alpha} - \frac{b}{a} \right) + \\
& + 2b^2 G (F_{el} \cos \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) + N \cos 2\alpha) \left(\frac{\cos \gamma}{\sin \alpha} - \frac{b}{a} \right) + \\
& + 2ab \cos \gamma (F_{el} (\sin \beta \operatorname{ctg} \alpha + \cos \beta) - N (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha)) \left(\frac{F_{el} \cos \beta (\cos \alpha \operatorname{ctg} \alpha - \sin \alpha) + N \cos 2\alpha}{\sin \alpha} \right)
\end{aligned} \tag{25}$$

4. Discussions.

From Figure 9, it is clear that at $\alpha=0^\circ$, i.e. at the initial position of the pedal, no reaction force arises at the support at point A. At all other angles, the value of the reaction force is 1·1022 N. This means that the value of the reaction force does not depend on the change in the angle between the initial and working positions of the spring installed on the clamping device of the staking machine. From Figure 10, it is clear that the reaction force created on the support at point B does not depend on the change in the angle β between the initial and working positions of the spring installed on the clamping device of the staking machine. Here the reaction force varies depending on the angle α formed between the initial and working positions of the operating pedal of the machine. When the angle between the initial and working positions of the pedal is $\alpha=0$, the value of the reaction force is also zero.

Similarly, using the calculation and graphic program Maple 2018, we plot the following graph (Figure 10) based on equations (24) and (25) with the set values $a=16$ cm, $b=32$ cm, $F_{el}=50$ N, $N=140$ N, $G=70$ N, $\gamma=(1/18)\pi$.

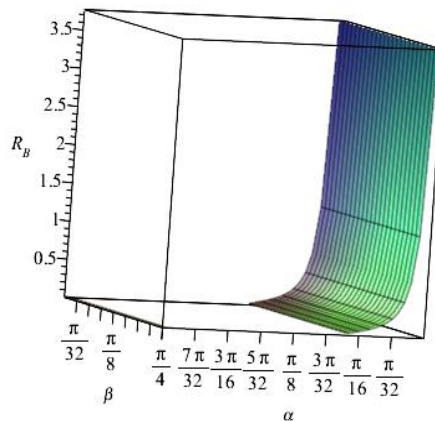


Figure 10. – Graph of changes in reaction forces on the support at point B at the angle formed between the initial and working positions of the pedal and the spring of the clamping device of the staking machine

5. Conclusions.

It was determined that the maximum value of the angle between the initial and working positions of the pedal cannot exceed 13.1 degrees, which is appropriate from the point of view of meeting ergonomic requirements for the design of the staking machine. This research aimed at the

development of small and medium-sized businesses and private entrepreneurship, specializing in the production of fur and leather products.

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DEVELOPMENT OF DIGITAL MODELS OF CARRIAGES AND PREPARATION OF BIG DATA FOR THE DEVELOPMENT OF THE ALGORITHM FOR PREDICTING THE REMAINING SERVICE LIFE OF CAR UNITS AND PARTS.

Namozov S.B.¹, Fedorova V.I.²

¹Tashkent state transport university, Tashkent, Uzbekistan

²JSC «ASRIRT», Moscow, Russia

E-mail: sohibnamozov956@gmail.com

Abstract. The paper presents the main results of multivariate modeling of the movement of gondola cars on bogies with axle loads of 23.5 tf (bogie 18-100) and 25 tf (bogies 18-9855 and 18-194-1) in various combinations of track design and technical condition of both cars, and the railway track. Based on the modeling results, recommendations were made for optimizing the running parts in terms of reducing wear on the wheel flanges, and a functional diagram was provided for assessing the service life of the running parts of cars.

Keywords: freight car, chassis, resource, bogie, digital model.

1. Introduction.

The tasks of preventing derailments associated with the failure of individual parts and components of the car and their cyclic durability, which allows maintaining stable and reliable operation in the entire range of modes, developing methods and diagnostic tools that allow organizing the maintenance and repair of cars and their parts according to their technical condition.

Nowadays, the issues of improving the reliability, efficiency, maneuverability and maintainability of freight rolling stock are relevant. No less important issues are extending the service life of components and parts, and expanding the permissible operating modes of freight cars.

The tasks of preventing derailments associated with the failure of individual parts and components of the car and their cyclic durability, which allows maintaining stable and reliable operation in the entire range of modes, developing methods and diagnostic tools that allow organizing the maintenance and repair of cars and their parts according to their technical condition.

In turn, an important issue is the development of the required volumes of transportation on time with a given level of safety, reliability and economic efficiency, which is the main task of the railways.

Providing transportation may consist of a set of measures to increase the weight and/or length of the train, and/or increase the axle load, increasing the capacity of station throats, other measures leading to an increase in the route speed of freight trains, etc., which has already been generally done on Russian railways: double trains weighing 7,100 tons are running, new series of 2ES6 locomotives have been put into operation, and train schedules are being analyzed and optimized.

To select the optimal scheme for mastering given volumes of traffic, it is necessary to conduct a comprehensive technical and economic analysis, including linking the method of increasing transport and throughput capacity with changes in the load of the track infrastructure, the choice of the type and design of the track, its maintenance schemes (maintenance and repair strategy), with the actual and forecast volumes of traffic, the possibility of providing transportation with traction (linking with the platform of tractive rolling stock).

2. Materials, methods and object of study.

The implementation of this approach with the required quality, timing and complexity of solutions is impossible without the use of modern digital technologies for modeling forecasting and analysis - a digital twin of the system being optimized.

The “carriage-track” system is an ideal sample for building a digital twin, but research on the whole of this system and experience shows that the task is quite difficult, as it includes a large number of variables, both from the side of the car and from the side of the track, and in addition, there is a “third” body, which entails a large number of calculations and analysis.

As part of the implementation of the R&SW plan of JSC ‘Russian Railways’ on the topic “Optimization of interaction in the wheel-rail system, development and updating of regulatory documents of JSC ‘Russian Railways’ in the framework of increasing the bearing capacity and service life of the upper track structure,” a matrix of multivariate calculations was developed, described in [1-7].

The development of dynamic models of freight cars was carried out in the “Universal Mechanism” software package [8].

3. Results and discussions.

The freight car model is built using the system of bodies approach, according to which the mechanical system is represented as a set of absolutely rigid bodies connected through hinges and power elements.

The model is parameterized, that is, it allows changes in geometric and physical parameters.

Contact interactions of elements are carried out using power elements that describe the contact of individual points of one body with the surface of another. This power element allows you to simulate gaps between contacting elements, entry and exit from contact, sliding and adhesion modes of friction force, as well as self-oscillatory processes.

In relation of being applied to the model of the friction system of a three-element bogie, the interaction of the wedge with the friction bar and the bolster is described by 16 elements of this type. The wedge model takes into account the gap in the transverse direction, the overestimation and underestimation of the wedge, and possible wear of the friction surfaces.

The wedge model allows for changes in its design, allows modes of locking, sliding and self-oscillation, and realistically describes the process of redistribution of friction forces when damping vertical and transverse vibrations. It is possible to set different values of contact stiffness and friction coefficients for different friction pairs.

The model of the heel assembly is built using 12 contact power elements and takes into account the possibility of the heel being torn off from the heel bearing, the body shifting on the heel, changing the radius and possible wear. Each axle box unit includes 10 contact interactions for transmitting vertical load from the sidewall and modeling the longitudinal, transverse clearance and limiting the rotation of the sidewall around the vertical axis. The car model is formed by 19 absolutely rigid bodies (body, two bolsters, 4 side frames, 8 wedges, 4 wheel pair sets) and has 114 degrees of freedom (6 degrees of freedom for each body). The structural representation of the model is based on the use of the subsystem method. The subsystem tree is presented in Figure 1.

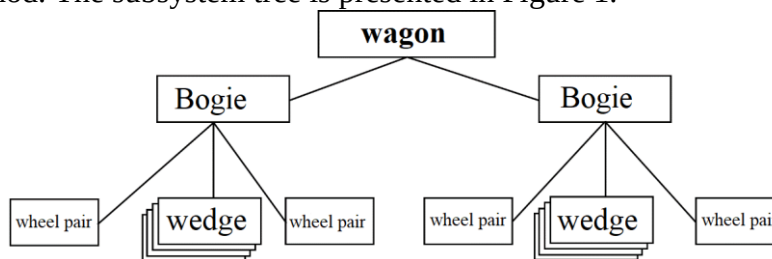


Figure 1. – Structure of the model subsystems

To study the influence of a freight car on the railway track, an analysis of the car fleet was carried out, which is presented in [1]. Thus, the following models of cars were selected from the total fleet, which are shown in Table 1.

Table 1.

Selected car prototypes for multivariate modeling

Car model	Car number in depot, units.	% of Car number in depot
Gondola car 12-132	139295	15.47
Gondola car 12-9853	79222	6.30
Gondola car 12-196	52704	4.19
Tank 15-1547	20461	4.08
Tank 15-9993	339	0.04
Hopper 19-9549	13180	1.05
Platform 13-6964-01	12377	0.99

To carry out the work, initial data were generated for computer modeling of car movement and analysis of the influence of interaction parameters in the wheel-rail system, and initial variables were determined for outputting the results.

According to the requirements of GOST 33211-2014 [9] for a freight car and checking the adequacy of the developed model, the following indicators of dynamic qualities were assessed:

- coefficient of vertical dynamics (dynamic addition) of the sprung parts;
- ratio of frame force to static axial load;
- safety factor against wheel derailment under the condition of rolling the wheel flange onto the rail head.

In addition to the abovesaid, for a freight car, in accordance with the requirements of GOST 33211-2014, the horizontal and vertical accelerations occurring on the platform were assessed; coefficient of dynamic addition of unsprung parts and additionally derived parameters from PC UM:

- the power of friction forces on the wheel flange MWear;
- total force in the transverse direction (lateral force between the wheel and the rail) Y(L);
- total force in the vertical direction (vertical force between the wheel and the rail) Q(V);
- work of friction forces on the skating circle AWear1;
- work of friction forces on the flange of the wheel AWear2;
- distance traveled (used for plotting and analysis);
- frame force in the wheel pair N;
- acceleration on the axle box in the vertical and horizontal direction;
- contact patch area Area;
- maximum contact pressure Pressure;
- maximum shear stress TauMax;
- approach angle of the wheelset yaw;
- force between the bolster and the friction wedge;
- forces arising in the springs under the bolster and wedge;
- force between the axle box body (adapter) and the side frame in the vertical, longitudinal and transverse directions.

The cars were accepted in two types of condition: empty and loaded, and their technical condition was also taken into account: new, moderately worn and maximally worn.

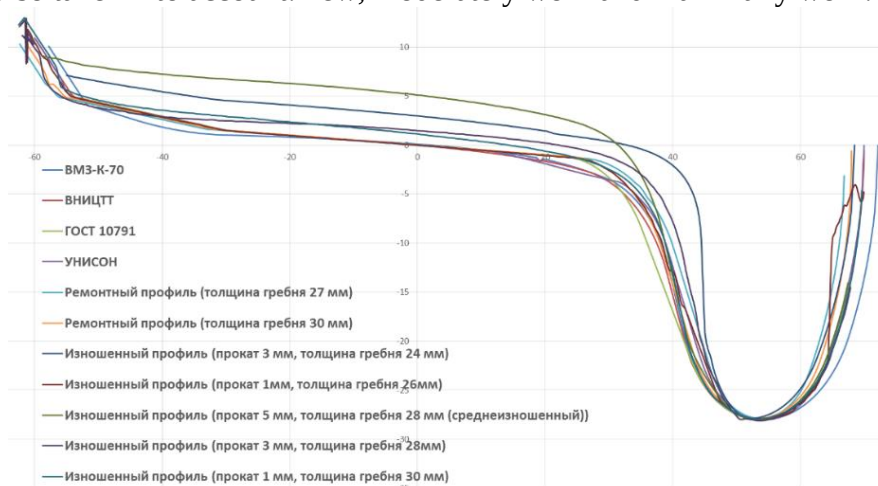


Figure 2. – **Wheels Profiles for multivariant modeling**

For assessing the reliability of the dynamic indicators obtained during modeling, the calculation results were compared with experimental data obtained during running dynamic tests and on the impact on the being tested track.

The model of a loaded gondola car was verified using the maximum values of dynamic indicators obtained during modeling and the impact on the track of indicators when compared with experimental data. Figure 3 shows, as an example, an oscillogram of frame forces when moving in the straight section of track at the speed of 90 km/h.

After verification of the models, calculations were carried out, Figures 4-6 show examples of the obtained indicators of the work of friction forces, stability coefficient and lateral forces.

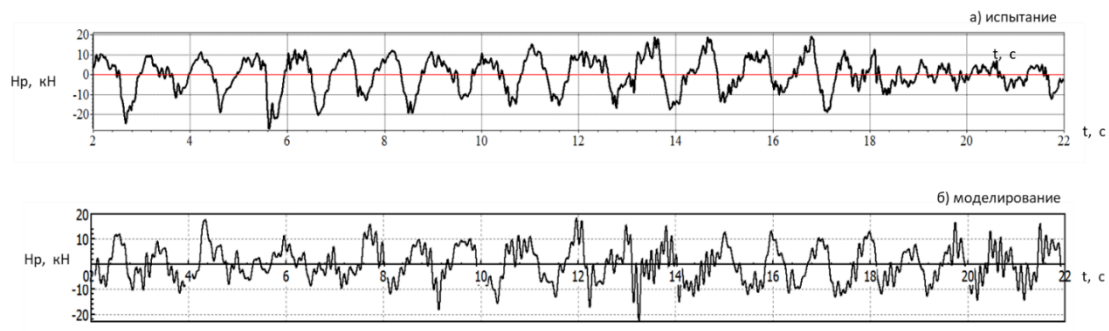


Figure 3. – Oscillogram of frame forces (H_p) of a loaded car body equipped with cassette bearings when moving in a straight section of track at a speed of 90 km/h during dynamic running tests (a) and during computer modeling (b)

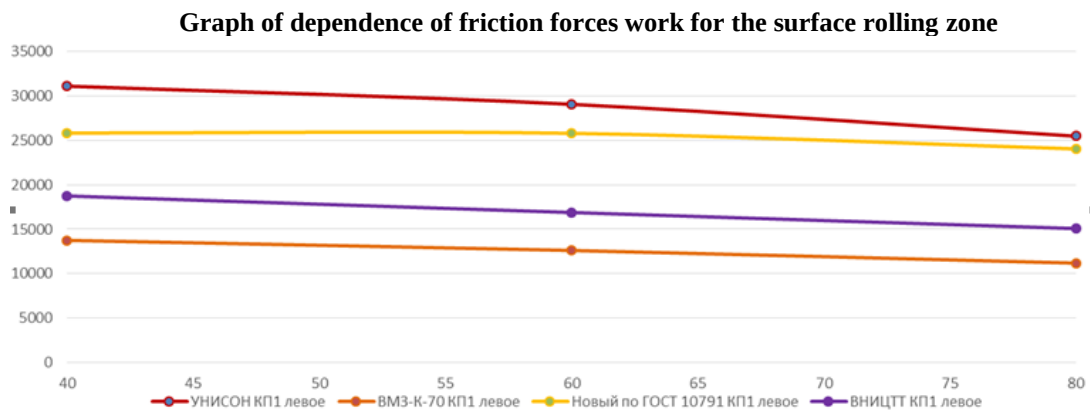


Figure 4. – Work of friction forces on the rolling surface depending on the wheel profile in a curve with a radius of 650 m on new P65 rails (gondola car on bogies 18-100 gross)

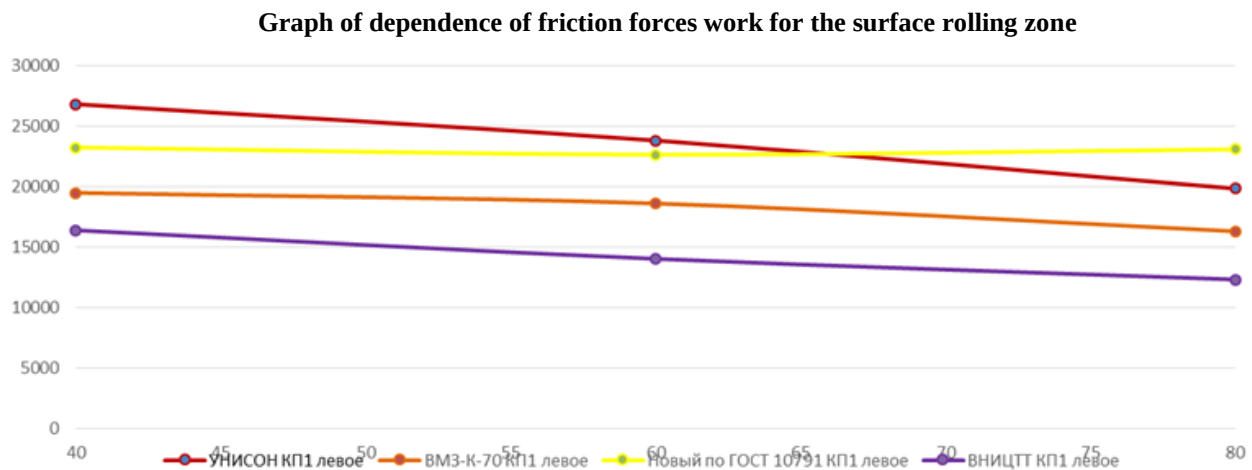


Figure 5. – Work of friction forces on the rolling surface depending on the wheel profile in a curve with a radius of 650 m on new P65 rails (gondola car on bogies 18-9855 gross)

a

b

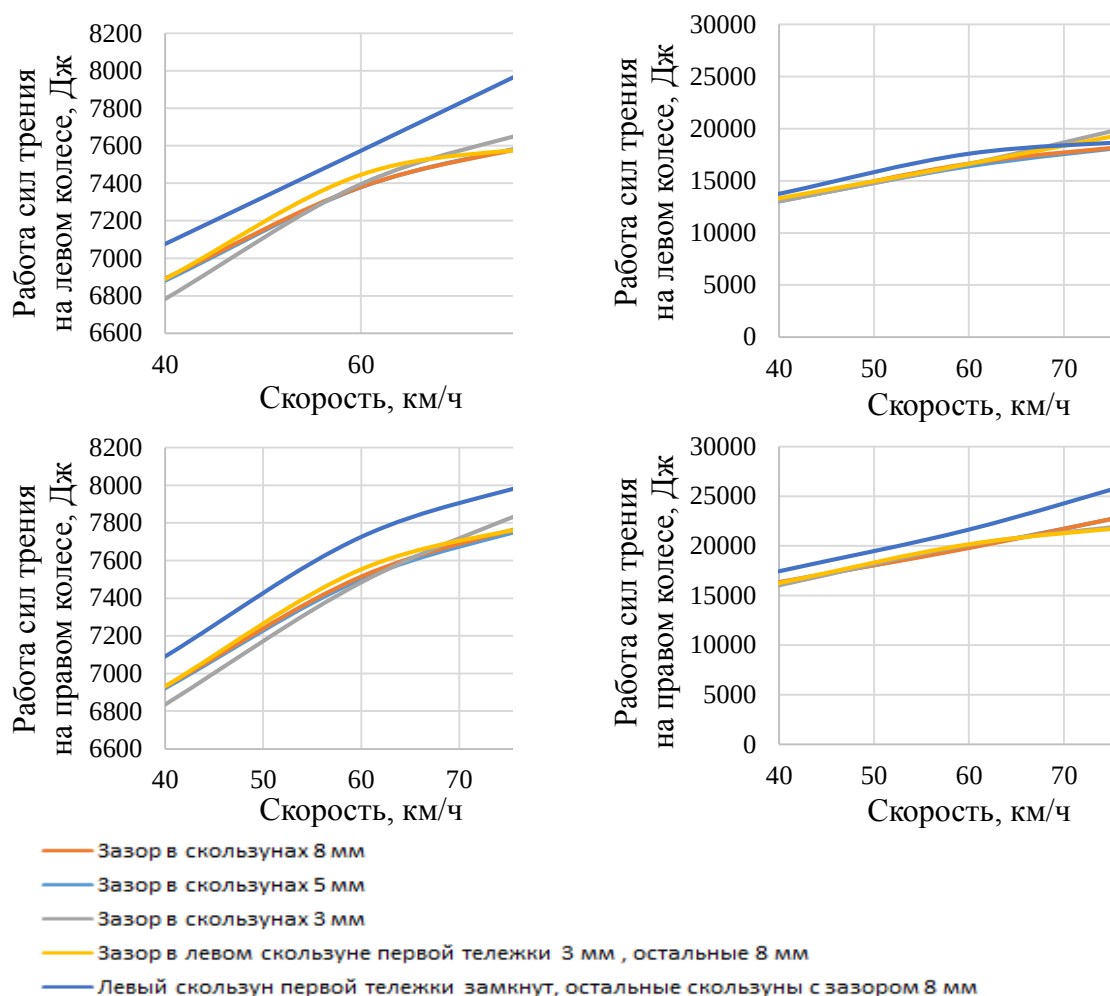


Figure 6. – Work of friction forces on the left and right wheels depending on the speed of movement of the car on bogies 18-100 along the straight section of the track in the area: a – rolling surface; b – ridge

4. Conclusions.

The material presents a small part of the calculations performed. The remaining results will be presented in the report.

The main conclusion can be drawn that the wear of car wheels is influenced by up to 20-30%, depending on the type of bogies:

- the thickness of the wheel flanges and the arrangement of wheel pairs with different wheel profiles in the car (running/non-running in the direction of travel), as well as the profile of the wheel and, accordingly, the rail;
- parameter of the horizontal rigidity of the connection of the side frames with the bolster, in particular for bogies of type 18-100;
- static deflection of the spring suspension, it must be increased when the car is packaged to 20-25 mm through the use of spring suspension with a bilinear characteristic;
- type of side supports (sliders) - use sliders in constant contact with elastic elements in the form of springs, with self-lubricating sliders. The moment preventing the trolley from turning relatively to the loaded body at axial loads of up to 23.5 tf should be in the range of 0.9-1.3 tf·m, for an empty body 0.3-0.4 tf·m. For cars on bogies providing an axle load of up to 25 tf, the moment with the body loaded should be in the range of 1.4-1.6 tf·m;
- the overestimation of the friction wedges should not exceed 2 mm, which will ensure a relative friction coefficient of at least 0.07. It is allowed to install polyurethane friction linings with a friction coefficient of about 0.12 between the wedge and the bolster.

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CONVERSION OF A PASSENGER WAGON INTO A DYNAMOMETRIC WAGON FOR RAILWAYS OF THE REPUBLIC OF UZBEKISTAN

Rahimov R.V¹, Khikmatov F.F¹, Zafarov D.Sh¹, Galimova F.S¹.

¹Tashkent State Transport University, Tashkent, Uzbekistan

E-mail: rakhimovrv@yandex.ru

Abstract. The article consistently outlines the stages of the research carried out on the conversion of a passenger wagon produced in the Republic of Uzbekistan into a wagon for special-technical purposes for operating it as part of passenger and freight trains. The test results of a prototype dynamometric wagon under the action of workloads on it allow us to conclude that the proposed design complies the established requirements in terms of strength, the stress-strain state of the nodes do not exceed the maximum permissible values.

Keywords: passenger wagon, dynamometric wagon, body, frame, load, stress, strength, testing.

1. Introduction

In the Republic of Uzbekistan, in order to create a unified railway network, a number of major projects are being implemented and large-scale measures are being taken to build tracks and improve the infrastructure of railway transport, equip the rolling stock fleet with wagons and locomotives with improved technical and economic characteristics [1]. In order to increase the volume of freight traffic, with the existing shortage of rolling stock and the lack of reserves for increasing the carrying capacity and carrying capacity of the country’s heavily loaded sections of the railway network, it is necessary to search for new directions for the development of railway transport, namely, the improvement of existing and construction of new railways, the integration of modern rolling stock into operation [2-4]. Therefore, the development of new and improvement of existing structures of rolling stock for special-technical purposes is one of the most major aspects, which will not only ensure the safety of train traffic when operating them on mountainous sections of the railway track, but will also allow

choosing the optimal locomotive traction for train operation, determining the longitudinal-dynamic coupling forces of wagons and locomotives.

The purpose of the research is to develop a dynamometric wagon for special-technical purposes, which, by design and technical condition, must comply the necessary level of safe movement for its operation as part of freight and passenger trains.

2. Analysis of designs of special-technical purpose wagons

In order to determine the main problems, the current state and promising directions for the development of the fleet of wagons, the designs of domestic and foreign passenger wagons and wagons of special-technical purpose are analyzed in detail. With all the variety of available structures of rolling stock and theoretical and experimental researches carried out to improve their designs, at the moment the literature has not considered and has not been studied at a sufficient level the development and improvement of the design of dynamometric wagons for special-technical purposes based on passenger wagons intended for their operation as part of freight trains. Based on the analysis, for further theoretical calculations and experimental researches, the design of the body of the passenger wagon model 61-907 [5, 6] produced by JSC “TashVSRZ” (Tashkent) was selected in order to improve and convert it into a dynamometric wagon of special-technical purpose for operation as part of both passenger and freight trains.

3. Researches of the loading of the passenger wagon body structure

In order to research the loading of the body structure of the passenger wagon model 61-907, a three-dimensional model of the body was developed using the SolidWorks 2021 software package to improve and convert it into a dynamometric wagon for special-technical purposes (figure 1).

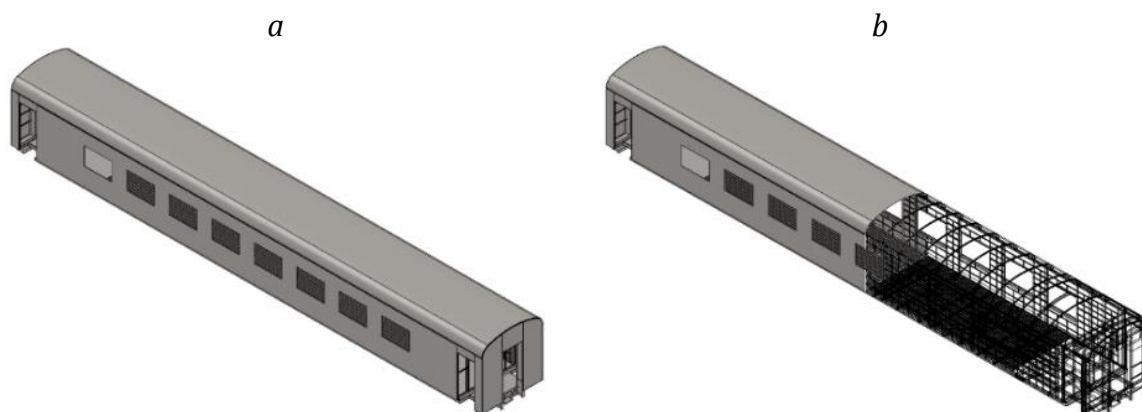


Figure 1. – General view of the spatial model of the passenger wagon body structure (a) and its metal structure (b)

To carry out theoretical research, the wagon body model developed in the SolidWorks 2021 software package was imported into the ANSYS Workbench 2021 software, on the basis of which an updated finite element model of the passenger wagon body was developed (figure 2), which allows taking into account the features of the variants of the design decisions taken in the calculations. Namely, changing the geometric dimensions of the body and frame elements (center sill, lateral longitudinal and transverse beams, braces and other elements), which will allow us to reasonably choose the optimal thicknesses of the reinforcing elements of the passenger wagon depending on the operating conditions.

To determine the loading capacity of the wagon body structure, stress distribution in the body elements and reinforcement zones, several transverse measuring sections with virtual measurement points installed on them were provided in the design model of the body to obtain individual stress values at each of the points.

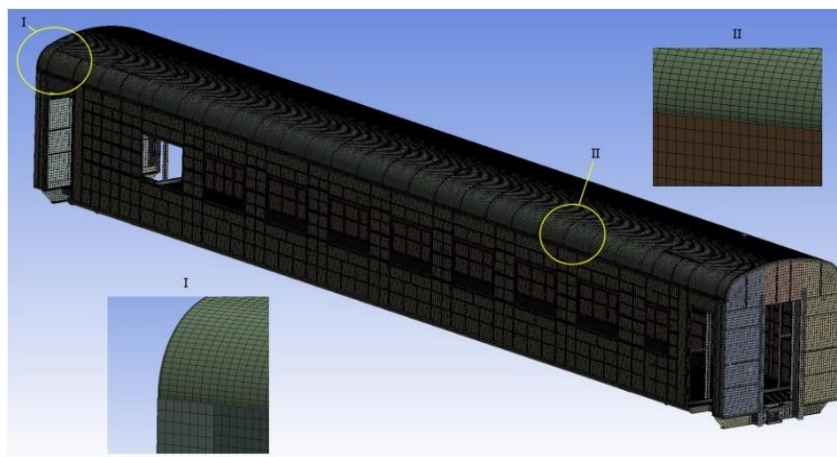


Figure 2. – Calculated finite element model of a passenger wagon body

As a result of the strength calculations performed using the finite element method in the ANSYS Workbench 2021 software, carried out according to the requirements of [7-10] under the influence of longitudinal forces applied to the front and rear stops of the auto-coupling equipment of the wagon body, the values of maximum stresses at virtual measurement points were obtained. Diagrams of the stress distribution according to the Mises theory on the elements of the wagon body in the first design mode are shown in figure 3.

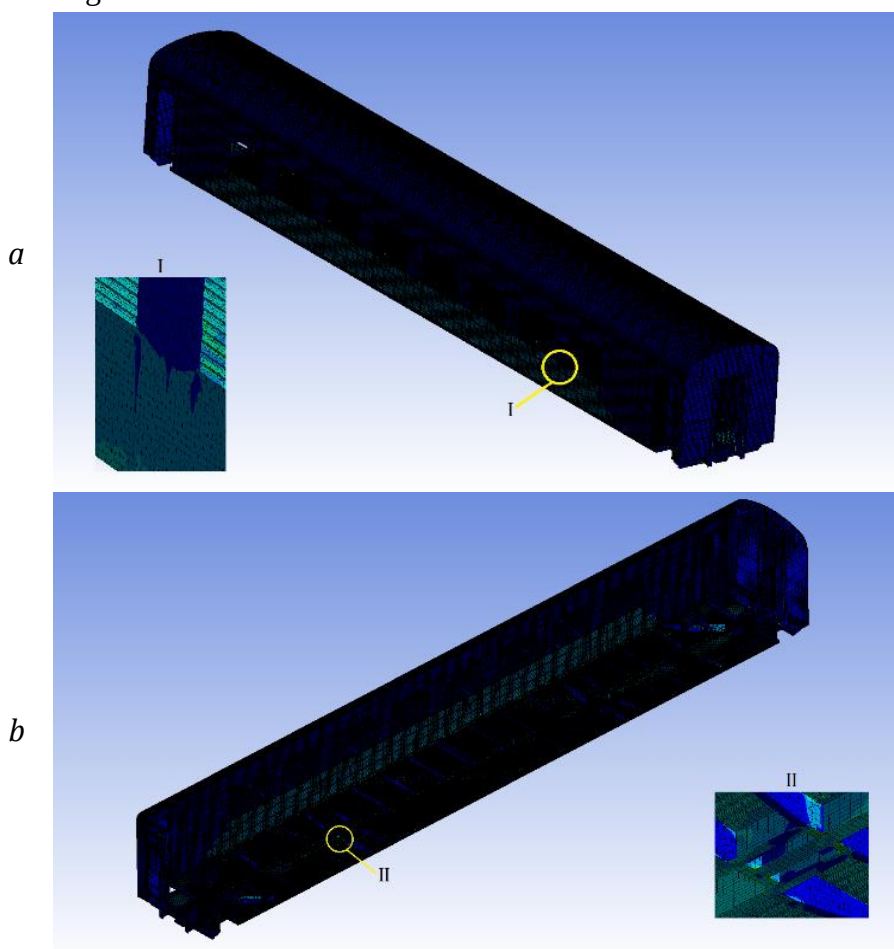


Figure 3. – Stress distribution fields in the elements of the wagon body at the I design mode (with the central interaction of auto couplings, compression force of 3.5 MN): *a* – top view; *b* – bottom view

The obtained dependences of the maximum equivalent stresses arising in the body elements allowed us to determine that the stress changes on the elements of the side walls and roof with an increase in longitudinal forces are insignificant, and their values do not exceed the established permissible values. At the same time, the stresses arising in the elements of the wagon frame in some

parts exceed the established permissible values, in this regard, to determine the places requiring reinforcement in the future, researches were carried out directly on the wagon frame.

According to the performed researches of the loading of the wagon frame under the action of longitudinal forces, depending on the design mode with a value from 1 to 3.5 MN applied to the front and rear stops of the automatic coupling equipment, the maximum stress values in each of the virtual measurement points were obtained. Diagrams of the stress distribution according to the Mises theory on the elements of the wagon frame with the central interaction of the auto couplings are shown in figure 4.

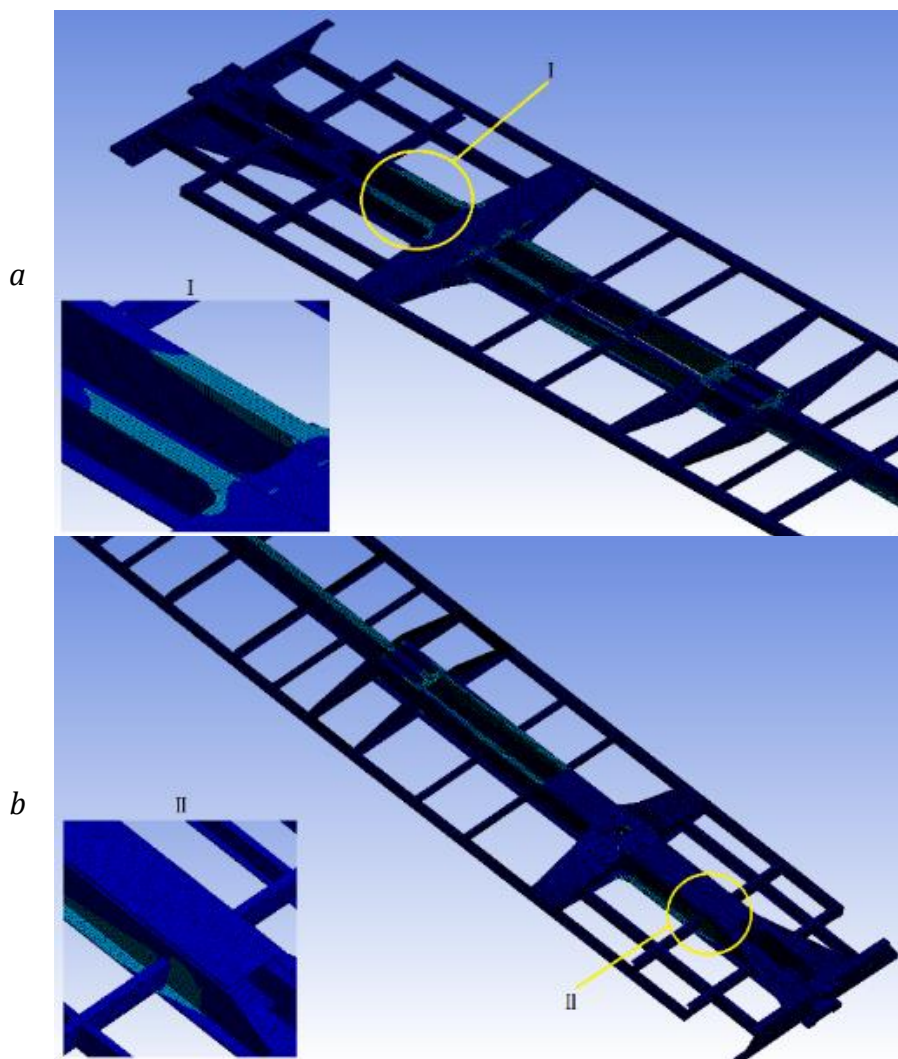


Figure 4. – Stress distribution fields in the elements of the carriage frame at the I design mode (compression force of 3.5 MN):
a – top view; b – bottom view

Analysis of the research results shows that the maximum stress value in the frame elements, mainly in the center sill, is 523.62 MPa, which, according to the first design mode, exceeds the established permissible stress of 276.3 MPa. Therefore, for the operation of the developed wagon as part of freight trains with increased longitudinal loads, it is necessary to provide for the reinforcement of the most loaded elements of the frame of the passenger wagon model 61-907 to improve its metal structure in order to convert it into a dynamometric wagon for special-technical purposes.

Based on the researches carried out, it was determined that in order to comply the established requirements, the center sill of the frame of the specially-technical dynamometric wagon in the central part, made of I-beam No. 30, must be reinforced on both sides along the entire length with steel plates 8 mm thick (figure 5) connecting the upper and lower shelves of the I-beam on both sides of the wall.

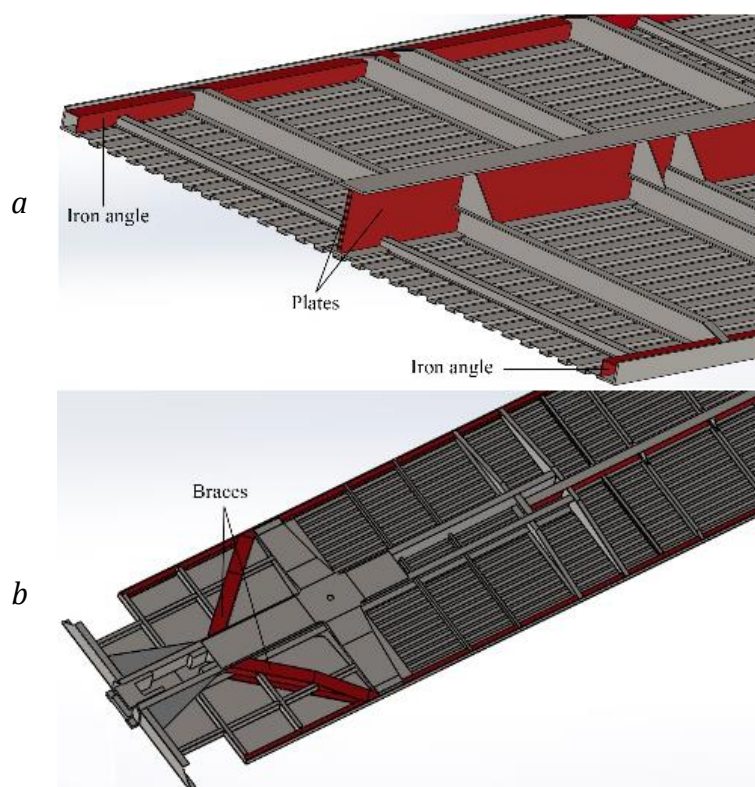


Figure 5. – Wagon frame with reinforcing elements

To reduce stresses on the center sill in the cantilever part of the frame, it is necessary to provide special beams (braces) from the I-beam, fixed on one side to the center sill, and on the other to the lateral longitudinal beams of the frame, which will allow part of the longitudinal forces acting through the stops of the automatic coupling equipment to be distributed to each of the lateral longitudinal beams of the frame. In this case, the beam should have a narrowing at the points of attachment to the binding without changing the cross-section profile.

In addition, the lateral longitudinal beams of the frame located between the doorways along the entire length must be reinforced from the inside in the places between the transverse beams with an equal iron angle measuring $75 \times 75 \times 5$ mm.

Thus, as a result of complex researches using the finite element method, an improved design of the passenger wagon body is proposed for converting it into a dynamometric wagon for special-technical purposes, characterized in that the center sill made of an I-beam is reinforced on both sides with steel plates connecting the upper and lower shelves of the I-beam on both sides of the wall to the side the longitudinal beams of the frame are welded on the inside with equal corners between the transverse beams along the entire length of the lateral longitudinal beams, the cantilever part of the frame has braces fixed on one side to the center sill, and on the other to the lateral longitudinal beams of the frame.

4. Theoretical researches of the design of a dynamometric wagon for special-technical purposes

In order to assess the strength and stability of the proposed design of the body of a dynamometric wagon for special-technical purposes on the basis of an improved design of the passenger wagon body, as well as its driving dynamic qualities and impact on the railway track, theoretical researches were carried out [11, 12].

The strength of the dynamometric wagon body was evaluated in accordance with the requirements of [7-10] and as a result of the calculation performed using the finite element method, equivalent stresses were obtained, according to the Mises theory, arising in the elements of the dynamometric wagon body under the first, second and third design modes. It is established that the strength of the proposed design of the dynamometric wagon complies the established requirements. The following results were obtained:

- in the I design mode, the maximum equivalent stresses in the frame elements were 209 MPa (75.7% of the allowable stresses), in the side wall frame elements – 130 MPa (58.9% of the allowable stresses), in the end wall frame elements – 112 MPa (43.9% of the allowable stresses) and in the roof frame elements – 125 MPa (51 % of permissible stresses);

- in the II design mode, the maximum equivalent stresses in the frame elements were 137 MPa (49.6% of the allowable stresses), in the side wall frame elements – 118 MPa (53.5% of the allowable stresses), in the end wall frame elements – 99 MPa (44.9% of the allowable stresses) and in the roof frame elements – 118 MPa (48.2% of permissible stresses);

- in the III design mode, the maximum equivalent stresses in the frame elements were 120 MPa (60% of the allowable stresses), in the side wall frame elements – 130 MPa (78.8% of the allowable stresses), in the end wall frame elements – 39 MPa (23.6% of the allowable stresses) and in the roof frame elements – 151 MPa (91.5% of the permissible stresses).

In addition, researches were carried out to assess the stability of the body elements and the driving qualities of the dynamometric wagon and its impact on the railway track, the results of which established that the developed dynamometric wagons complies normative requirements [9, 10].

5. Experimental researches of a dynamometric wagon for special-technical purposes

To carry out experimental researches, a program and methodology of static tests for compressive/tensile strength of a dynamometric wagon for special-technical purposes have been developed. According to the developed program and methodology, a prototype dynamometric wagon of a special-technical purpose model 61-907-DM was subjected to static strength tests, according to [10], the purpose of which was to assess the stress-strain state of the elements of the wagon body under the action of statically applied longitudinal compressive and tensile test loads.

During tests on the basis of JSC “Tashkent plant for the construction and repair of passenger wagons”, a stand for loading wagons with longitudinal static loads was used to create longitudinal forces on the frame, strain gauges BBF200-10AA-A(11)-BX30 were used to register deformations, measurements were carried out by the hardware-software complex MIC-185 with recording processes on a hard disk the disk of a personal computer.

Before applying longitudinal loads for compression and stretching of the wagon, strain gauges were applied to the researched points of the load-bearing elements of the wagon body (figure 6), obtained from the results of calculations.

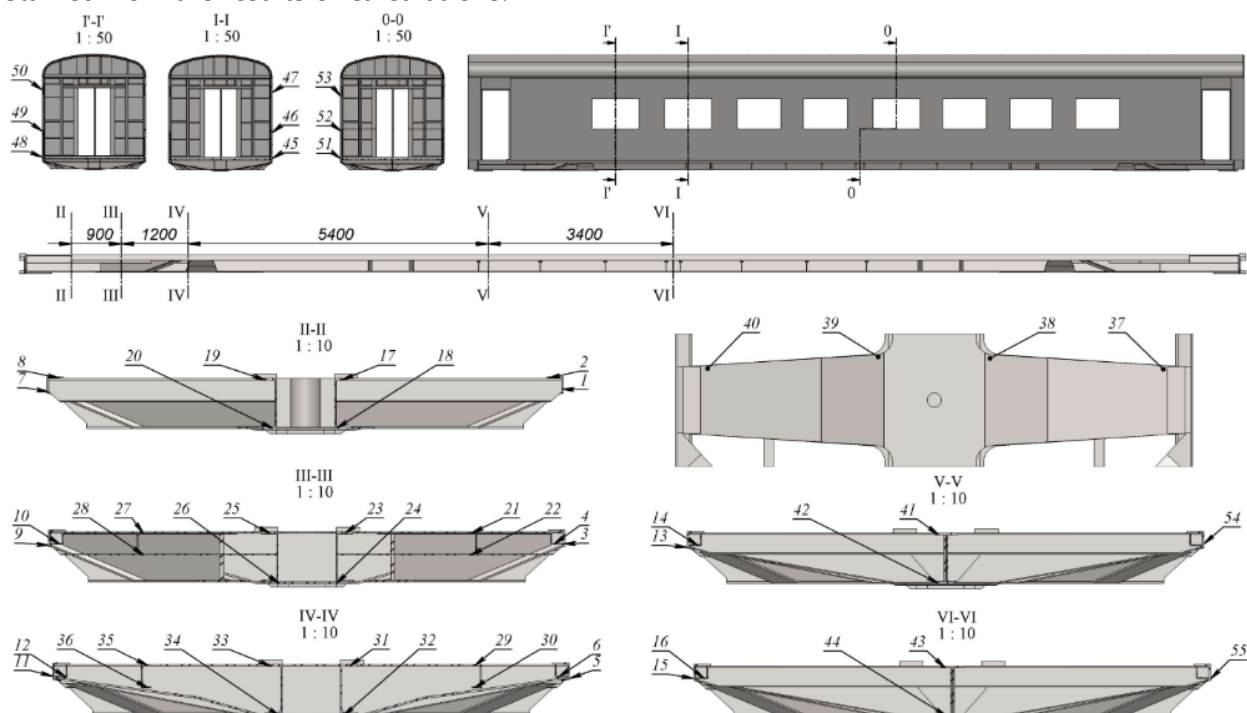


Figure 6. – Scheme of the installation of strain gauges on the body and frame of the prototype wagon

The tests were carried out in the following sequence:

- application of a longitudinal static load through the elements of the automatic coupling device with a force of 1.0 MN and 2.5 MN (stretching)
- application of a longitudinal static load through the elements of the automatic coupling device with a force of 2.5 MN (stretching) and 3.5 MN (compression) using jacks mounted on the loading stand (figure 7).

As a result of the performed static strength tests of the prototype dynamometric wagon model 61-907-DM, maximum stress values were obtained on the following elements (figure 8): on the center sill in the reinforcement termination zone – 217.8 MPa; on the center sill in the central part – 180 MPa; on the pivot beam – 99 MPa; on the body elements – 93 MPa.



Figure 7. – General view of a prototype dynamometric wagon model 61-907-DM on a stand for loading with longitudinal static loads

Based on the obtained results of complex tests of a prototype dynamometric wagon for special-technical purposes model 61-907-DM under the action of statically applied longitudinal compressive and tensile test loads, it was determined that the proposed design of the dynamometric wagon complies the established requirements and meets the strength conditions, the resulting stresses do not exceed the maximum permissible values. In addition, no visible damage or residual deformations were found in the design of the prototype wagon.

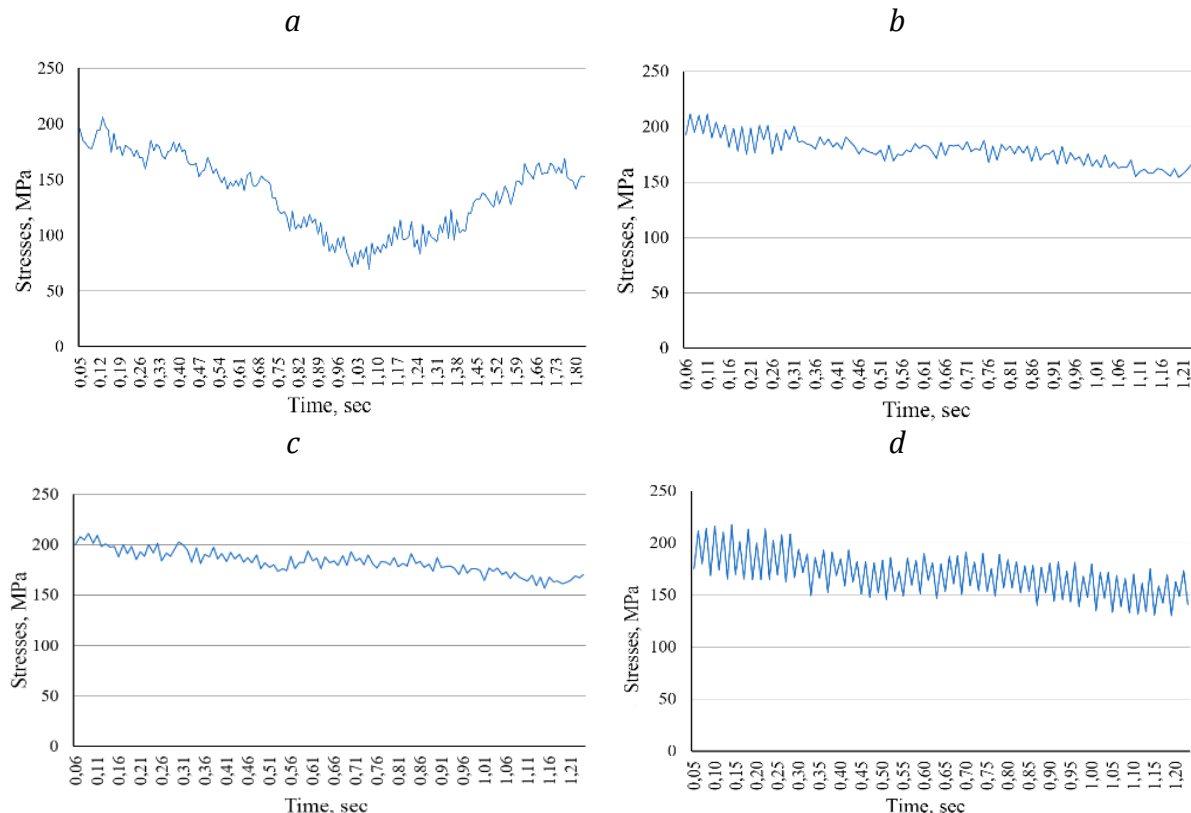


Figure 8. – The maximum values of the received stresses on the body elements of the dynamometric wagon:
a – strain gauge № 13; b – strain gauge № 15; c – strain gauge № 16; d – strain gauge № 23

6. Conclusion

Thus, on the basis of the complex theoretical and experimental researches carried out, it was determined that the design of the developed dynamometric wagon of a special-technical purpose model 61-907-DM satisfies the strength conditions and can be operated as part of both passenger and freight trains. The developed design of the body of a dynamometric wagon for special-technical purposes on the basis of an improved design of the body of a passenger wagon model 61-907 was introduced at JSC “Tashkent Plant for the construction and repair of passenger wagons” during the construction of a dynamometric wagon for special-technical purposes model 61-907-DM.

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MODERN ALUMINUM COMPOSITES AND ALLOYS FOR CAR BUILDING. PART 1.

Rakhimov R.V.¹, Mansurov Yu.N.¹, Wang Zhengwei², Lingying Ye²

¹Tashkent State Transport University, Republic of Uzbekistan

²Central South University of China, China

E-mail: yulbarsmans@gmail.com

Abstract. The article presents an analysis of the use of aluminum alloys in the field of railway transport. Typical high-strength aluminum alloys used in high-speed trains are presented. The unique properties of aluminum alloys are analyzed. The main problems that arise when using them are also mentioned. Key production technologies such as casting, molding, and welding are analyzed. Future improvement directions for better application of aluminum-based alloys in railway applications are summarized.

Keywords: aluminum, alloys, composition, structure, properties, railway transport

1. Introduction.

With the rapid development of human civilization, consequent air pollution and greenhouse gas (GHG) emissions have been threatening humanity for many years. Energy conservation and emissions reduction are a growing priority in the development of the transport industry.

A number of approaches can be used to improve energy efficiency and reduce CO₂ emissions, such as reducing aerodynamic drag, powertrain losses, tire rolling resistance and weight [1-5]. Among these options, lightweight construction is currently considered one of the most effective solutions. In recent years, weight reduction has become a major research topic in the transportation industry around the world [6–10].

Changing the material seems to be a promising option for lightening the design. Traditionally, steel, both low-carbon steel and stainless steel, is used as the main material for the manufacture of parts and transport structures.

Today, titanium, aluminum and magnesium are promising lightweight metal materials as alternatives to steel and cast iron [12-15]. Moreover, aluminum exhibits balanced characteristics, such as light weight, good corrosion resistance, good manufacturability, high specific strength and relatively low cost. The density of aluminum alloy is only a third that of steel. By optimizing the design by replacing the material, the total weight of the car body when using aluminum is reduced by 50%. This degree of reduction deserves efforts for engineering application in, for example, car building [16-19].

Although aluminum constitutes ~8 wt.% (by mass) of the Earth's crust, aluminum's high affinity for oxygen, as well as the stability of aluminum oxides and silicates, have long prevented its use as a basis for materials (alloys in particular). For this reason, aluminum only became economically viable for use at the end of the nineteenth century [20-25].

Aluminum alloys were originally used in the aviation industry, which is an important part of the transportation industry. It was first used on the fuselage of the Junker F13 with the invention of the alloy in 1920. It began to be used in the 80s of the last century in the production of trains to increase their speed. It was widely used in the automotive industry, as soon as it was necessary to obtain cars with speeds above 200 km/h. The development and application of aluminum alloys is booming, especially thanks to the huge expansion of high-speed rail lines in China over the past 20 years.

The use of relatively new lightweight materials has led to the development and application of a number of new manufacturing processes such as friction stir welding and laser welding. This technological advancement has promoted the use of aluminum alloys not only in high-speed trains but also in urban rail vehicles, which continuously contributes to energy conservation and climate change [3,7,11,26].

This work, consisting of two parts, presents the relationship between the chemical composition, structure and performance properties of aluminum alloys (the first part) and the use of key technologies for processing aluminum alloys for the production of high-quality parts and structures for railway transport, in particular, high-speed trains (the second part).

The purpose of the study is to establish the possibility of using aluminum-based alloys, including those with a high content of impurities, to use modern technologies for their processing to

ensure comfortable travel in an environmentally friendly mode of operation of railway transport, as well as to stimulate the interaction of materials scientists and transport workers, engineering workers of the real world. sector of the economy to improve technical analysis with a view to subsequently actively promoting technological innovation in transport in general.

To achieve the goal, the following tasks were solved:

- a theoretical analysis of the existing use of aluminum-based alloys, including those from scrap and waste in transport, was carried out;
- the relationship between the chemical composition of alloys, their structure and properties has been established;
- experimentally studied the influence of modern processing technologies of aluminum alloys on their performance properties.

Characteristics of aluminum alloy for transport. The special operating conditions of trains and the unique properties of aluminum alloys contribute to their combination, which contributed to the popularization of high-speed trains and the in-depth use of aluminum alloys in the transport industry. For engineering applications in railway transport, aluminum has the following advantages: reducing the net weight of the vehicle, while increasing the strength and rigidity of, for example, a passenger car, while simultaneously significantly reducing weight and meeting safety requirements (aluminum alloy body is 30–50% lighter than become).

Aluminum alloy has excellent heat resistance. Although the melting point of aluminum (660 °C) is significantly lower than that of steel (1539 °C), the heat resistance of the body is related not only to the melting point of the material, but also to the thermal conductivity of the material. Compared with steel, aluminum alloy has superior thermal conductivity and better heat dissipation.

Aluminum alloy is easy to process, produce and maintain. Through the development and application of large hollow aluminum profiles and complex profiles, aluminum welding technology is constantly being improved, and vehicle manufacturing technology is becoming more mature. Aluminum alloy parts are easy to replace and suitable for all types of surface treatment. The workload requirements for the production of railway car bodies are also significantly lower than for the production of steel bodies.

The price of aluminum alloy is moderate. The high price of aluminum material increases the production cost of vehicles, but aluminum alloys also make vehicles lightweight, resulting in increased efficiency, lower energy consumption and lower maintenance costs. Thus, the analysis showed that if the cost of steel structures is 100%, then aluminum structures are 52%. The cost of restoring decommissioned structures is 100% for steel and 480% for aluminum. From the perspective of comprehensive economic benefits, the use of aluminum vehicles is economical and reasonable. Therefore, the final cost of using aluminum alloy is moderate.

Typical aluminum alloys for high-speed train car bodies are shown in Table. 1.

Table 1

Typical aluminum alloys for high-speed train car bodies [2-6, 8-11]

3	Brand	State	Mechanical properties			Application area	Note
			σ_B, MPa	$\sigma_{0.2}, \text{MPa}$	$\delta \%$		
5000	5083	O	$\geq 275 \sim 350$	$\geq 125 \sim 200$	≥ 16	Front trim	Plate
6000	6005A	T6	≥ 270	≥ 225	≥ 8	Under frame, sidewall, roof, end wall, beam	Plate, 3 ~ 5 mm
	6005A	T6	≥ 255	≥ 215	≥ 6	Under frame, sidewall, roof, end wall, beam	Section, 3~5 mm
	6082	T6	≥ 310	≥ 260	≥ 10	Under frame, end beam	Plate, 3~6 mm
	6106		≥ 250	≥ 200	≥ 8	End wall	Chapter
7000	7B05	T4	≥ 315	≥ 195	≥ 11	Under frame, end beam	P, 2,9 ~ 6,5
	7B05	T5	≥ 325	≥ 245	≥ 10	Sub frame, end beam, traction beam	Chapter

2. Application (results of practice).

In railway transport, aluminum alloys are mainly used in the manufacture of the car body and its auxiliary structures: parts of the structure, including the roof, sub frame, end and side walls.

However, different aluminum designations are chosen for different parts of the body based on their properties. Both extruded profile and plate are used. World experience shows that extruded profiles account for about 70% of the total weight of an aluminum alloy body, plates account for about 27%, and castings and forgings account for about 3%. On the other hand, a trend has emerged according to which aluminum alloys are beginning to be used not only in the body, but also in other structures, such as gearboxes and axle boxes.

3. Body part.

The development of aluminum alloy materials and large extruded profiles opens the way to the modernization and light weighting of railway vehicles. Thin plates and hollow complex thin-walled profiles have been successfully developed. In Japan, 6N01 alloy (alloy 6005) was developed with better extrusion, welding and corrosion properties to produce porous complex thin-walled hollow sections, widely replacing 7N01 and 7003 profiles as floor, side and roof body structures. In Western Europe, aluminum alloy body is mainly made of 6005A extruded profiles, the main reason is that the extrusion performance of 6005A is better, the production process is more simplified, and the stress corrosion problem of 7000 series alloy can be avoided. Application of typical aluminum alloys in high-speed trains Shinkansen 300 series is shown in Fig. 1.

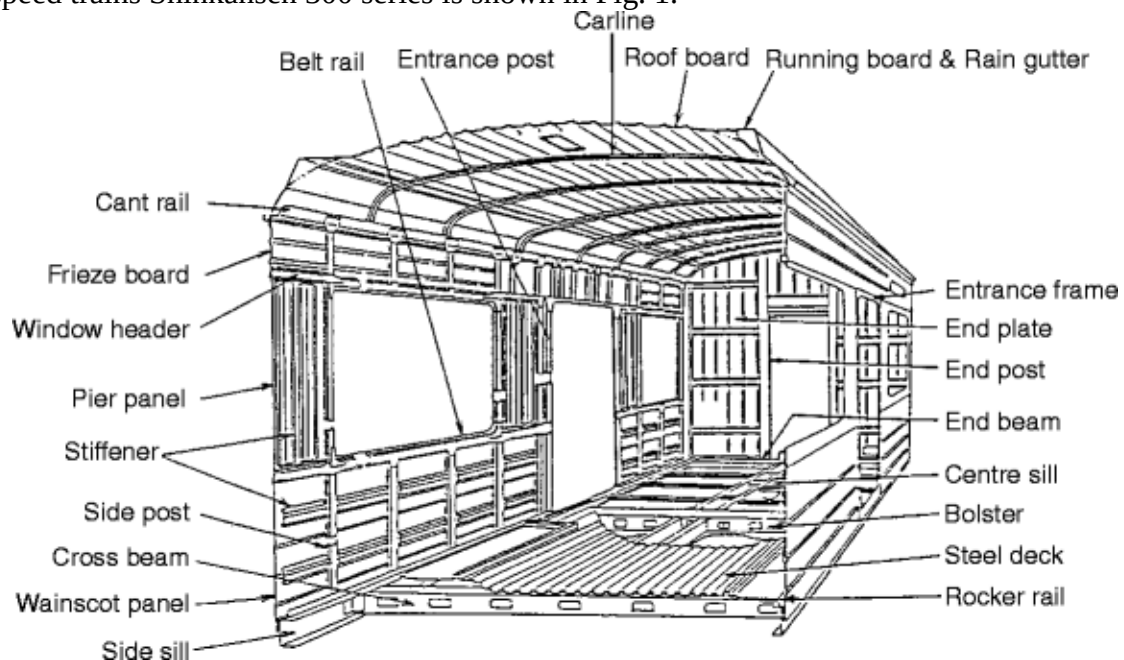


Figure 1. – Application of aluminum alloys in the body of a high-speed trains [22-26]

A typical sidewall of a high-speed train, consisting of extruded sections, is shown in Fig. 2. However, when it comes to a locomotive, the structure is completely different. To achieve optimal aerodynamic characteristics when moving at high speed, the locomotive was given a streamlined design, as shown in Fig. 3.

This unique shape makes it impossible to produce relatively regular and straight sections. Therefore, the beam-slab structure became the optimal option for the head car.

The frame is designed to meet rigidity and strength requirements to support the front skin against plastic deformation. Before fixing the skin, it is welded to hundreds of aluminum plate beams. After this, the skin is divided into small pieces according to the principle of good workability. Each part is deformed to a certain shape depending on the profile of the structure. Then the parts are fixed to the frame one by one.

4. Transmission.

To further reduce the weight of the train by reducing the weight of the body is clearly not enough, since the car body makes up only about 20% of the total weight of the train. Key bogie components, including the traction motor, wheelsets, frame and braking system, have attracted the

attention of lightweight advocates. Lightening the transmission can help reduce unsprung weight and rail wear or damage.

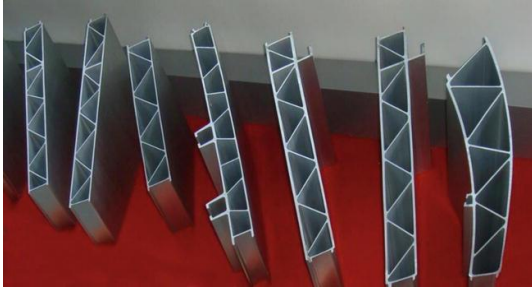


Figure 2. – **Typical extruded profiles for high-speed trains.**



Figure 3. – **Streamlined head car of a high-speed train**

The high-speed train gearbox is made by casting aluminum rather than wrought alloy due to its complex and uneven thickness. Low pressure casting is widely used in non-ferrous alloy casting because of the high feeding pressure and temperature gradient, as well as stable filling, which can effectively improve casting density and product yield.

AlSi7MgA and AlCu4Ti alloys are commonly used as casting materials for gearboxes due to their good fluidity, low thermal expansion and shrinkage percentage. Typical aluminum gearboxes of CRH series high-speed trains are shown in Fig. 4. These same alloys are commonly used as casting materials for gearboxes due to their good fluidity, low thermal expansion and shrinkage percentage.

5. Axle box.

The axle box is one of the important load-bearing parts of the three-wheeled trolley and the transmission transmission unit. The left part of the axlebox is installed on the axle journal through a rolling bearing, and the right swing arm is connected to the positioning rotary seat on the frame through an elastic unit. When the train moves, the axlebox experiences vertical force, longitudinal force and lateral force. Thus, the condition of the axle box housing bearing is complex, and its structure and operational stability are very important for the safe operation of the train.



Figure 4. – **Typical aluminum gearboxes for CRH series high-speed trains**

7050 aluminum alloy forgings have high strength and toughness, which can significantly reduce unsprung weight. The weight of a forged aluminum product is reduced by 62.5% compared to a traditional carbon steel product. Therefore, forging of axle box made of aluminum alloy is widely used on high-speed trains. A 3D model of an aluminum alloy axle box is shown in Fig. 5a. A typical finished aluminum alloy high-speed train axle box is shown in Fig. 5 B.

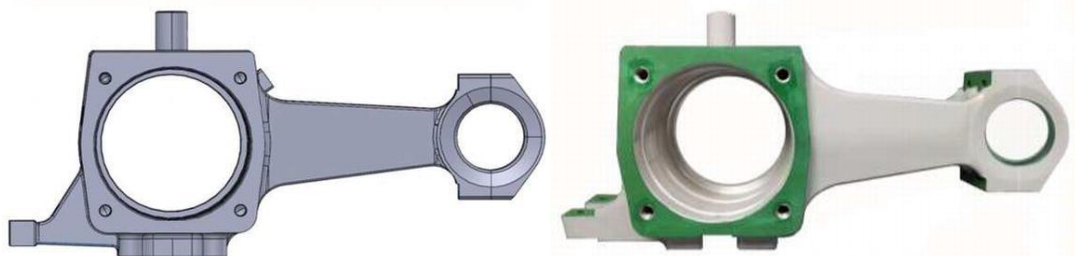


Figure 5. – **3D model and experimental finished aluminum alloy axle box**

An experimental study of the relationship between the chemical composition, structure and properties of aluminum-based alloys made it possible to compile a correspondence table and select various aluminum-based metal systems that make it possible to manufacture parts and structures of various types of cars (Table 2).

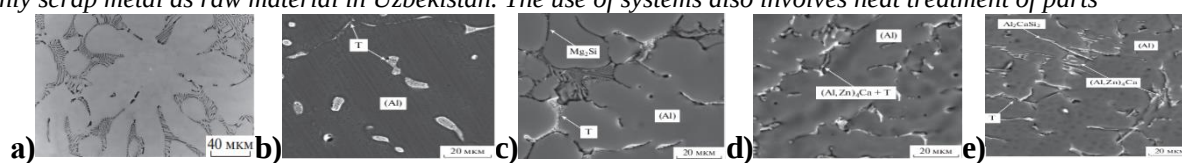
As an example in Figure 6 shows the structures of aluminum-based metal systems with components introduced into the aluminum composition together.

Table 2

Recommendations for the selection of aluminum-based metal systems for railcar construction

Main purpose	Type of semi-finished products	Alloy system
Tank boilers for transportation of chemically aggressive and/or highly clean cargo	Sheets, plates, profiles, rods, pipes	Al-Mg-Cu-Fe-Si-Mn-Zn-Ti
Sheathing, stiffeners and other parts of freight car bodies	Sheets, plates, profiles, rods, pipes	Al-Mg-Mn-Fe-Si-Zn-Ti-Cu-Be
Structural elements of freight car bodies	Profiles, panels, pipes	Al-Mg-Cu-Zn-Si-Cr-Fe-Mn-Ti
Load-bearing structural elements of freight car bodies	Profiles, pipes	Al-Zn-Cu-Mg-Mn-Zr-Fe-Si

Note: metal alloy systems allow the use of the ratio of components and impurities, taking into account the use of mainly scrap metal as raw material in Uzbekistan. The use of systems also involves heat treatment of parts



a) Al-5.5Mg-2.5Si-1Li alloy (SM); b) – Al-Zn-Mg; c) – Al-Zn-Mg-Ca; d) – Al-Zn-Mg-Si; e) – Al-Zn-Mg-Si-Ca

Figure 6. – Microstructures of the studied alloys with elements introduced

Thus, based on the results of studies of aluminum-based metal systems, it has been established that in order to manufacture products for elements of railway rolling stock, for example, freight car bodies, it is recommended to use the alloys of the systems indicated in Table. 4. The specific development of new alloys from scrap and waste for the carriage building of Uzbekistan is possible based on test results to determine mechanical properties, which is a short-term task for the team of authors.

6. Conclusions.

From the results of the study it follows that for the carriage building of Uzbekistan, (taking into account the availability of only secondary raw materials in the industrial production of the country, it is necessary and possible to use metal systems based on aluminum. It must be borne in mind that the most acceptable is the Al-Mg system.

The information presented in the work is useful for the development of aluminum-based alloys with the main alloying element magnesium and a high content of impurities, in particular the most harmful - iron, silicon and copper, which could be widely used for the manufacture of parts for railway rolling stock.

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MODERN ALUMINUM COMPOSITES AND ALLOYS FOR CAR BUILDING. PART 2.

Rakhimov R.V.¹, Mansurov Yu.N.¹, Zhengwei Wang², Lingying Ye²

¹Tashkent State Transport University, Tashkent, Uzbekistan

²Central South University of China, China

E-mail: yulbarsmans@gmail.com

Abstract. Improving the production technology of railway transport, improving its design and economic performance indicators require the use of aluminum alloys in the field of railway transport, which has a number of problems, the solution of which will contribute to the improvement and possible development of a number of key production technologies. In the work, based on an analysis of world experience, models of rolling stock have been compiled and key technologies used in the production and repair of carriage building products have been proposed.

Keywords: carriage building, materials, technologies, design, durability of carriages

1. Introduction.

The most urgent task of modern railway transport in Uzbekistan is the localization of production in order to manufacture its own car-building products, which would allow the transport industry to abandon the import of finished products, components, and materials in demand by the industry [1-5].

In the first part of the work, the authors proposed the composition and structure of light materials using aluminum as an example, the use of which will significantly improve the quality of car-building products and bring them into compliance with international requirements [6-10]. To do this, you also need to understand the trends in the development of technologies in the global carriage building, since it is known that so far, both repair and production of new products in the carriage building of Uzbekistan are based on the use of both materials and processing technologies of the last century. Traditionally, steel, both low-carbon steel and stainless steel, is used as the main material for the manufacture of parts and structures of railway transport.

In this work, the second part of the study, options are presented for the use of key technologies for processing aluminum alloys for the production of high-quality parts and structures for railway transport, in particular, high-speed trains (second part).

The purpose of the study is, based on the possibility of using aluminum-based alloys, including those with a high content of impurities, established by the authors, to propose the use of modern technologies for their processing to ensure comfortable travel in an environmentally friendly mode of operation of railway transport, as well as to stimulate interaction between materials scientists and transport workers, engineering workers in the real sector of the economy to improve technical analysis with a view to subsequently actively promoting technological innovation in the industry.

To achieve this goal, the following tasks were solved:

- the achievements of global industrial production of carriage building products were theoretically studied and critically analyzed in comparison with analogues in Uzbekistan;
- experimental studies of corrosion protection of products of the domestic railcar industry were carried out and technologies were proposed that increase the corrosion resistance of railcars, including after major repairs;
- improved technologies for casting, welding and corrosion protection of domestic car-building products have been proposed, allowing the use of lightweight aluminum-based materials in industry.

2. Materials and research method.

The work used aluminum grade A99, alloys (including model ones) based on it using aluminum alloys with various elements [11-16].

The alloys for the study were prepared in a laboratory electric resistance crucible furnace. The melting process was carried out at a temperature of 720-740°C, casting temperature - 720±5°C.

The compositions of the studied alloys are given in Table. 1.

A methodological feature of the work is that for the first time in research practice, multielectrode arrays were used to quickly screen corrosion inhibitors at various concentrations and pH, placing identical pairs of AA2024-T3 wires in a large number of separate reaction cells containing different inhibitory solutions, applying a potential of 100 mV between the two wires in each cell (Fig. 1).

Table 1.

Concentration ranges of components of metal systems based on aluminum, introduced separately or together, % by weight

Zr, Cr, Sc, Ti, W, Sr	Sn, Pb	Fe	Li*	Mn*	Zn*	Cu*	Mg*	Si*	Al
tenths	0,1-0,6	0,1-2	до 2	0,1-1,2	0,5-6	0,2-5	0,5-7	0,2-7	ост.

*Note: chemical elements marked * are components that form the basis of the most commonly used metal systems*

The current between both electrodes was used to determine the polarization resistance. Using the setup, many different combinations of inhibitors and their synergistic behavior were assessed for one metal, and the possibility of using the setup to assess the effect of pH on the effectiveness of inhibitors was demonstrated [17-20].

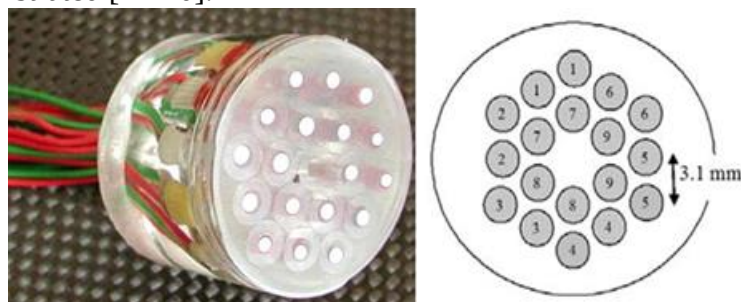


Figure 1. – Multi-metal multi-electrode for corrosion testing

The basic idea of the proposed method, using a combination of different pairs of metals assembled together as a multielectrode (ME), is to test a combination of nine pairs (or as many as you are interested in) of different metals in one electrode configuration, connected via a multiplexer to a potentiostat /galvanostat [21-26].

The measurements consist of applying a voltage of 100 mV between a selected pair of the same metals inside the ME and measuring the current flowing between them, then repeating the same procedure for other pairs of metals. This setup was used to rapidly evaluate (without removing ME from solution) the concentration range over which a particular inhibitor or combination of inhibitors provided corrosion protection.

3. Results and their analysis.

Casting - an important process in the manufacture of complex structures such as gearboxes (see Part 1). Experimental testing of casting technologies in laboratory conditions showed that the main problems of the casting process, for example, of gearboxes, are as follows:

- in the thin-walled zone of the product there are often no cold partitions, the porosity and density of other defects are increased in order to meet the requirements for the operation and quality of the appearance of the casting;
- the surface of the box, flange, hanger and other key parts are free of excessive shrinkage porosity, shrinkage cavity and other defects to meet the internal quality requirements of radiographic inspection of castings;
- high requirements are required for the processing of pinholes and unprocessed surfaces, especially for castings made of aluminum alloys, in which pinhole defects easily occur;
- in the production of aluminum alloy gearboxes for high-speed electric trains, common casting defects are porosity, pinholes and shrinkage cavities.

To eliminate these defects, it is necessary to take two measures based on controlling the air content in the molding sand. First, the air ventilation inside the casting should be improved by opening more air holes. Secondly, the thickness of the coating should be guaranteed to reduce surface voids in the sand by using a coating with high thermal conductivity, such as zircon powder. The filling pressure and holding pressure are increased accordingly to increase the resistance to gas penetration into the metal liquid.

The key to eliminating pinholes is mainly based on controlling the hydrogen content of the liquid aluminum. The refining process (which is not used in domestic casting) can reduce the amount

of oxidation inclusions and hydrogen content in liquid aluminum, and thus effectively reduce the tendency to form micropores.

As for the shrinkage cavity at the top of the gearbox, its effectiveness has been proven by simulating the solidification process using the licensed program MAGMAsoft. The location of the cooling block and riser can be optimized to ensure that the top riser feed channel is unblocked with reasonable temperature distribution.

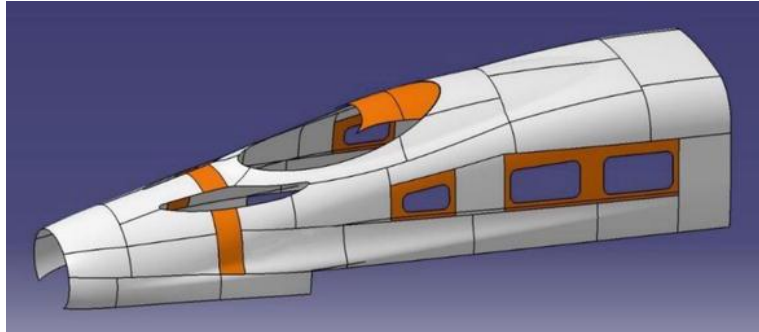


Figure 2. – Front panel of a high-speed train

Molding process. Four different technologies are available for the front skins. The most commonly used is the hammer press, where it is convenient to use a hammer machine to make targets. However, the tolerance on the dimensions of the manufactured product largely depends on the experience of the worker. Once the panels are installed on the structure frame, any further changes to the geometric characteristics can only be made using hand tools such as a hammer. This manual process makes high repeatability of manufactured components virtually impossible. The second method is the expansion-stretching process. This applies to curved profile panels, but only to a certain extent. In addition, a rotary press is used, which is mainly designed to produce panels with small and uniform curvature, and for panels with complex and small curvature, a process called forming press is used, which uses a mold to obtain a given shape.

As shown in Fig. 2, the front skin of a typical high-speed train is divided into approximately 70 small parts, which are joined together using a 170-meter welding line. Since each panel is limited to just 1 meter in length, the manufacturing process becomes labor intensive and inefficient for producing significant numbers of small components. Front panel product quality is also degraded due to increased residual stresses resulting from the uneconomical and complex assembly process.

To ensure the accuracy of each part's assembly, a finite element analysis is required to simulate the process of skin pull-out and elastic return based on a flexible multi-point matrix. In order to improve the computation efficiency and obtain satisfactory computation accuracy, a dynamic explicit algorithm is used to model the image process, and a static implicit algorithm is used to calculate the elastic return. In Fig. Figure 3 shows the distribution of MISES stresses and equivalent plastic strain for the formation of a solid part. The molding process is designed based on the simulation results, which can save experimental time and improve adaptability to various products, as shown in Figure 4.

Welding. Due to the special thermophysical properties and welding characteristics of aluminum alloy, such as low melting point, thermal conductivity, high electrical conductivity, expansion and shrinkage, high content of alloying elements compared to carbon and stainless steel [2,5,11,19], when welding aluminum alloys Pores, cracks, lack of penetration, large welding deformations, and slags form much more easily. Assembly accuracy, quality and productivity can be greatly affected by the welding process. The welding process is therefore crucial in the production of high-speed trains, for example. When choosing a welding method, the following aspects should be considered:

- it is known that aluminum alloys are coated with a dense oxide film, which easily adsorbs moisture and transfers hydrogen to the melt bath. Aluminum can also turn into oxide slag present in the weld, which affects its performance. Therefore, it is very important to remove the oxide from the base metal and the surface of the aluminum alloy grooves before welding;

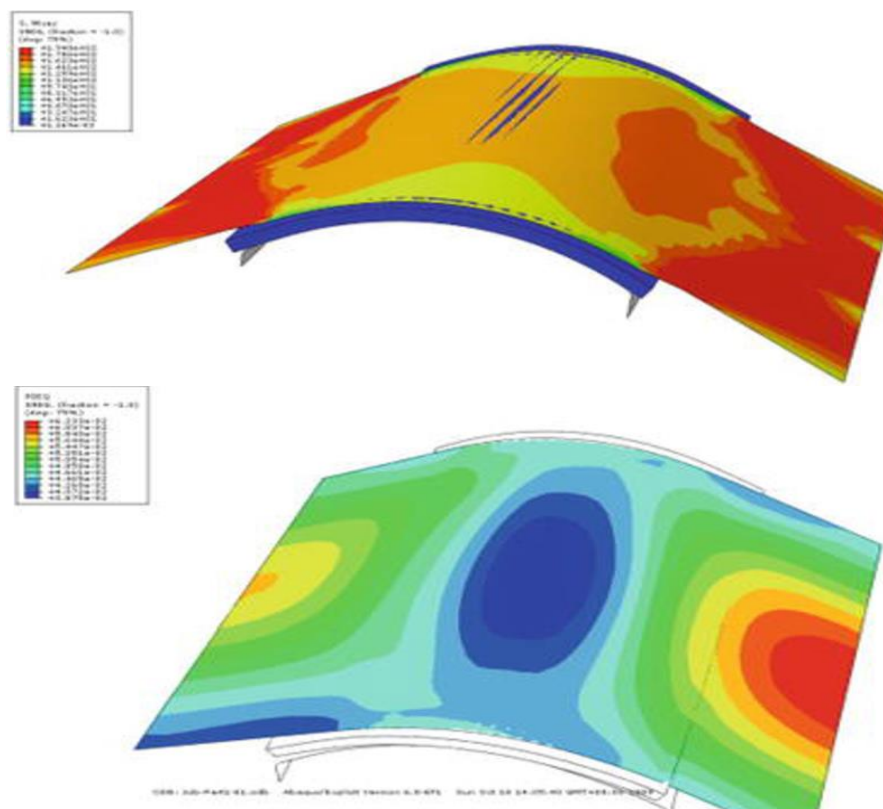


Figure 3. – Distribution of MISES stresses and equivalent plastic strain for forming a solid part

- it has been established that the thermal conductivity of aluminum alloy is five times higher than that of low-carbon steel. A high power or concentrated energy welding heat source should be the preferred option. And preheating is necessary for welding thick sheets;



Figure 4. – Sheathing based on a flexible multi-point matrix - Chinese carriage building workshop

- it has been established that the coefficient of thermal expansion and the rate of shrinkage during cooling of an aluminum alloy are two times higher than that of steel. Therefore, welding deformation when melting aluminum alloy is serious. Deformation control measures such as reverse deformation and reinforcement limitation should be considered;

- Aluminum alloy material contains various kinds of alloying elements, and the loss of alloying elements during fusion welding easily leads to a decrease in joint strength and corrosion resistance, and intergranular cracks are easily formed in the weld metal and heat-affected zone. The tendency to crack should be taken into account.

Based on the findings obtained, three typical welding processes are presented that can be widely used in the production of trains in Uzbekistan, in particular high-speed ones.

Pulse welding is the most widely used and developed welding method for aluminum alloys, which is characterized by large thermal power, high linear energy, good energy concentration and good shielding effect. The above characteristics are suitable for welding aluminum alloy due to its unique thermophysical properties. Pulse technology can be used to control wire melting, improve arc stability, and achieve liquid metal jet transition at low average current, which is suitable for all-position welding.

Considering the effect of cathode sputter welding on the removal of aluminum oxide film, a DC positive electrode must be used for pulse welding. Semi-automatic pulse welding is suitable for welding short irregular welds, while conventional long straight welds are usually welded automatically with laser tracking.

Typical defects include poor formability, burn-through, over-welding, cracking, pores and slag as shown in Fig. 5.

Friction stir welding. In addition to traditional MIG welding, friction stir welding (FSW) is widely used. It has been found that this process and the pulsed welding process are incompatible. Welds made using pulse welding technology cannot be used for friction stir welding, so the use of this technology is limited.

However, FSW has shown the following unique advantages:

- low manufacturing cost - welding consumables such as electrode, wire, flux and shielding gas are not required during the welding process.
- there is no need to remove the oxide film before welding, which reduces cleaning time and increases production efficiency.
- good welding quality. The temperature of friction stir welding is relatively low, so the change in the microstructure of the heat-affected zone is small, and the residual stress is low, resulting in low deformation.
- connection efficiency is higher compared to pulse welding.
- environmentally friendly process, safe. There is no pollution, smoke, radiation, etc. Less energy consumption. Because friction stir welding relies solely on the rotation and movement of the welding head, it saves more energy than fusion welding or even conventional friction welding.
- high welding efficiency. Long welds, large sections and in different positions can be welded simultaneously.

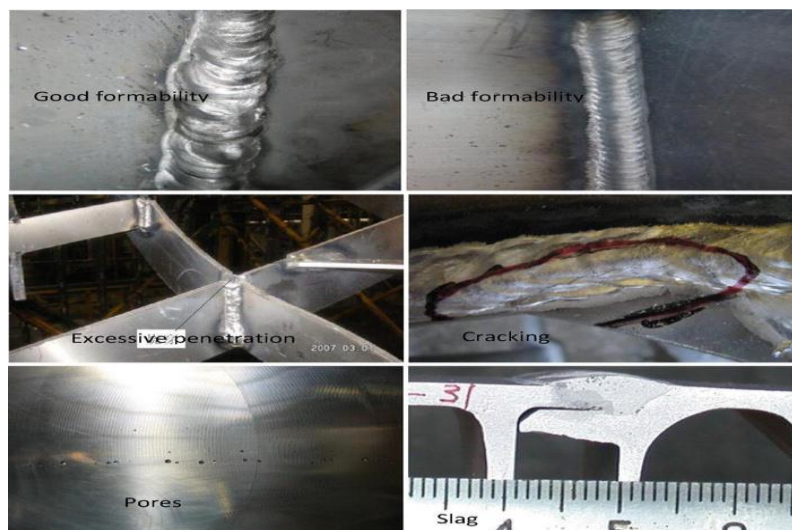


Figure 5. – Typical aluminum alloy weld defect

The above advantages contribute to its application in the field of railway transportation. FSW has been found to produce a flatter weld than pulse welding, which requires minimal post-processing (Figure 6). The defects of friction stir welding joints mainly include holes, non-welded joints, protrusions and grooves. Defects arise mainly due to the fact that during the welding process different parts of the weld metal are subjected to different thermomechanical processes, which leads to overheating or insufficient fluidity of plastic materials.

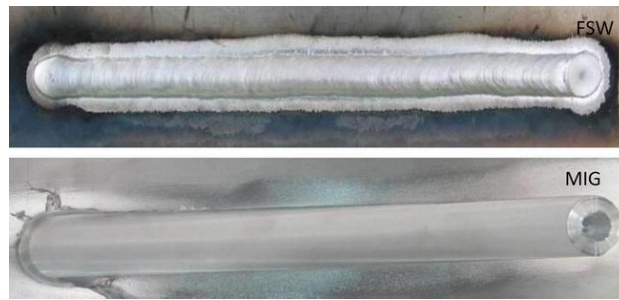


Figure 6. – Comparison of MIG and FSW welds

The top of the weld seam is subject to strong friction and stirring effect from the stirring needle and the shaft shoulder at the same time, even if the welding speed is very high or the speed of the stirring head is not high enough, it can still achieve a certain heat input and form a defect-free joint.

In the middle of the weld, the heat input is less than from the top, but the heat transfer is also less than from the top and bottom, so the total heat absorption is greater than from the top and bottom, and the degree of softening of the material is the highest. The heat input at the bottom of the weld is lowest and the power output is highest. Therefore, welding defects will appear at the bottom of the weld if the process parameters are incorrectly selected or the size of the welding tool is inappropriate.

Laser welding. Laser welding is a new welding technology that has a broad development perspective, contributing to the production of high-quality welded joints.

Laser welding uses both laser beam and arc, which features high welding speed, stable welding process, high thermal efficiency, and provides larger welding gap. The weld pool for laser welding is smaller. Compared with pulse welding, laser welding shows lower heat input, smaller heat affected zone and smaller working deformation.

Based on the characteristics of concentrated heat source, strong penetration and arc welding, a new hybrid laser-pulse welding joint and groove design was developed through experimental optimization and verification. Compared with traditional welding, the angle of the upper groove is smaller.

The wide range of applications and high efficiency of laser welding increase its competitiveness in terms of investment costs, shorter production times, lower production costs and increased productivity.

Figure 7 shows a comparison between hybrid welding and laser welding. Laser and pulse welded hybrid welds are flat, reflecting good formability.

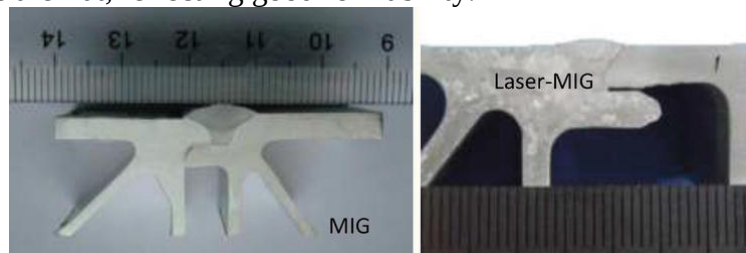


Figure 7. – Comparison of hybrid MIG and laser-MIG welds

Regarding design, the use of aluminum alloys also ensures greater aesthetics. Authors from China kindly provided the surfaces of car bodies manufactured at the country's car-building factories (Fig. 8), which look superior compared to traditional steel ones.



Figure 8. – Painted external and internal surface of an aluminum car body

Anti-corrosion process. The increasing speed of the train allows it to experience contact with different environments in a matter of hours. Environmental change is very complex when a train moves from Uzbekistan to other countries, from high latitudes to low latitudes, from low altitude to high altitude. The weather can change dramatically from sunny to rainy. Temperatures can vary from sub-zero to 60°C. Humidity can vary from very dry to very wet (~100%). The air can change from fresh and clean to polluted. It may contain dust, ox sulfide, ox nitride or chloridione. These ingredients can cause corrosion of the aluminum alloy, which will adversely affect the safety and long-term reliability of the train, especially when it is traveling at speeds greater than 200 km/h.

Aluminum is known to exhibit good corrosion characteristics because it can form a passive film in the atmosphere. However, corrosion resistance is also threatened by alloying elements and aggressive environmental factors. Pitting, galvanic, and stress corrosion are common types of corrosion in aluminum engineering structures exposed to atmospheric conditions.

In railway transport, an organic coating system is used to protect aluminum from corrosion. To work in different environments, the outer surface coating system is different from the inner surface. The outer coating system is used to combat harsh environmental conditions, while the inner coating combats moisture condensation. Therefore, higher demands are placed on the external coating system. They were evaluated through a series of accelerated corrosion tests, including a salt spray test, a humid heat test, and a high and low temperature test.

To protect the body from an aggressive environment, a technology has been developed consisting of preliminary sandblasting, epoxy primer, polyurethane putty, polyurethane layer, polyurethane topcoat and varnish.

To protect against internal moisture, a technology has been developed using cleaning, etching coatings, anti-corrosion primer and polyurethane top coating. When connecting aluminum components to other alloys, pre-treatment of the surface with the application of an anti-corrosion primer is necessary to ensure physical isolation from each other and protection against galvanic corrosion.

4. Conclusions.

Technologies are presented that allow the use of aluminum alloys in the field of railway transport. Technologies for the production of trains from unique aluminum alloys are shown.

Instructions have been developed and presented for the use of materials and technologies for the production of a railway vehicle, including the car body, gearbox and axle box steering rod.

Key production technologies such as casting, forming, and welding and corrosion protection are analyzed.

Using this information, industrialists will be able to apply advanced materials, especially aluminum alloys in railcar construction and continuously contribute to the sustainable development of human civilization through technological innovation.

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SELECTION OF PIPING MATERIAL WITH LOW AERODYNAMIC RESISTANCE FOR A COTTON HARVESTER

Rizaev A.A.¹, Khudjaev M.K.², Ganjajev Sh.A.¹

¹Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan

²Tashkent State Technical University named after Islam Karimov, Tashkent, Uzbekistan

E-mail: mukhiddinkhudjaev@gmail.com

Abstract. Calculations were conducted based on the laws of continuum mechanics and the material for the pipe system of cotton harvesters with the lowest aerodynamic resistance was determined. A cotton harvester's corrugated suction pipeline was made from this material, and the kinematic characteristics of cotton as it moved through the piping system were measured in field conditions. The results of calculations to determine the material for the pipeline and field tests for the serial and proposed modified designs of cotton-picking machines are presented. Machines with resource-saving working parts were recommended for mass production.

Keywords: pipeline material, cotton picker, aerodynamic resistance, field tests.

1. Introduction.

To improve the operation of machines for cotton harvesting and processing, sufficient theoretical and experimental studies were conducted [1-4]. The study of raw cotton motion through a pipeline and the material of this pipeline with low aerodynamic resistance makes it possible to improve the harvesting processes and transportation of cotton using a pneumatic conveyor unit in cotton processing enterprises.

The movement of raw cotton and the air mixture through a pipeline of a variable cross-section as a multi-speed heterogeneous medium was studied [5]. Applying Coulomb's law of dry friction, a closed system of equations of one-dimensional motion was obtained, consisting of the equation of motion of the mixture components and the law of conservation of mass. Equations of cotton porosity, air pressure, and component velocities were obtained to describe cotton transportation through a pipeline using a pneumatic conveyor unit at cotton processing plants. The multi-speed model of the motion of cotton-air mixture as a heterogeneous medium in pneumatic transport pipelines more correctly describes the process of airflow and material under pneumatic transportation and makes it possible to propose its use in calculations of pneumatic transport and its design. The resulting model is universal and can be used for calculating pipelines of variable cross-sections used in other industries.

Based on the analysis of theoretical studies on aeroelastic vibrations of cotton-air mass in the pipelines of cotton harvesters, a mechanical and mathematical model of aeroelastic vibrations of the cotton-air mass in straight pipelines of the pneumatic system was developed and a calculation algorithm was compiled [6]. In contrast to previously conducted work, the transportation process considered depends on changes in the longitudinal and transverse displacements of the center of gravity of the mass of a bundle of cotton, caused by pulsations of airflow pressure.

An analysis of the pneumatic conveyor system of a cotton-picking machine was conducted in [7] and the results of experimental studies on the motion of a continuous medium in the pipelines of the pneumatic transport system of cotton pickers were obtained. Based on the analysis of the measurement of airspeed in the pipelines of operating MX-2.4 designs with a row spacing of 90 cm, to improve the operation of the pneumatic transport system, a modernization of the semi-mounted MX-2.4 cotton picker design with a row spacing of 60 cm was proposed. A refinement of the piping scheme corresponding to a final velocity in the receiving chamber of the cotton picker of less than 6 m/s was recommended. Studies were conducted to determine the aerodynamic resistance for only one type of material of the suction pipeline.

Research to determine aerodynamic drag was performed in various fields of industries and as a result of these studies, many problems in these industries were solved. The process of aerodynamic resistance of the input zone of dust collector in a vortex dust and gas separator was studied and the results of experiments were presented in [8]. The aerodynamic resistance of coolants and heat exchangers during their design was determined [9]. The aerodynamic resistance in single- and multi-row finned bundles with an exhaust shaft was determined and a method for calculating air speed was developed; the influence of external airflows on the gravitational thrust created by the shaft was

revealed [10]. Research on aerodynamic resistance leads to a reduction in vehicle fuel consumption [11, 12]. Determining aerodynamic drag is important for optimizing performance in such a field as biking [13].

The above studies were conducted for existing designs of assembly apparatus and their suction pipelines.

This article presents the results of calculations to determine the material for the pipeline of cotton pickers with the lowest aerodynamic drag readings and the test results to determine the kinematic characteristics of the movement of raw cotton along the pipeline of a serial production and along the pipeline of a modified design proposed by the authors. A recommendation is given for mass production of this type of machines with resource-saving working parts.

2. Materials, methods and object of study.

Research methods are based on the laws of continuum mechanics and techniques of conducting experimental research.

Improving the operation of the pneumatic part of cotton-picking machines generally improves the quality of work and productivity of the cotton picker. To achieve this goal, it is necessary to determine the pipeline material with the lowest aerodynamic resistance readings and conduct tests to determine the kinematic characteristics of the cotton motion in a pipeline made from this material. Comparison of the results of the kinematic characteristics of cotton motion through the pipeline of a serial cotton harvester with the results of the proposed design leads to specific conclusions and recommendations.

Based on the known laws of aerodynamics of continuous media [14], calculations were conducted on the aerodynamic resistance of the pipeline of the pneumatic conveyor system of a cotton picker. To determine the resistance of the pipelines of the cotton picker to the airflow, we use formula $\Delta P \approx \xi \rho V^2 l / (2d)$, where ξ is the resistance coefficient of the pipeline, V is the air speed in the pipeline, m/s, l is the length of the pipeline, mm (which varies within $l = 1160 \dots 2130$ mm), d is the internal diameter of the pipeline, mm ($d = 170 \dots 180$ mm). First, we determine the Reynolds number for a characteristic speed of 20 m/s: $Re = Vd / \nu = 20 \cdot 0,172 / 1,8 \cdot 10^{-5} = 1,9 \cdot 10^5$.

The pipeline resistance coefficient for different pipes is determined by different formulas. Let us calculate the resistance coefficient of the pipeline and, using it, the value of pressure loss for several types of material from which the pipeline of the pneumatic conveyor system of a cotton picker could be made.

The resistance coefficient for smooth pipes is determined by the Blasius formula: $\xi = 0,3164 / Re^{0,25}$; after calculations, we obtain $Re^{0,25} = 20,9$ and $\xi = 0,0154$. Thus, the pressure loss for smooth pipes is $\Delta P = 0,154 \cdot 1,25 \cdot 400 / 2 \cdot 1000 / 172 = 22,4$ Pa.

We calculate using the formula for determining the resistance coefficient for glass pipes and obtain $\xi = 0,312 / Re^{0,25} = 0,0149$; the pressure loss for glass pipes for this value of ξ is $\Delta P = 0,49 \cdot 1,25 \cdot 400 / 2 \cdot 1000 / 172 = 21,6$ Pa.

We calculate using the formula for determining the resistance coefficient for polyethylene pipes and get $\xi = 0,25 / Re^{0,226} = 0,016$; the pressure loss for polyethylene pipes is $\Delta P = 0,16 \cdot 1,25 \cdot 400 / 2 \cdot 1000 / 172 = 23,2$ Pa.

For non-smooth pipes, the resistance coefficient is determined by formula $\xi = 0,11(k_{uu} + 68 / 4Re)^{0,25}$. Here k_{uu} is the equivalent roughness (for brass pipes $k_{uu} = 0.0015 \div 0.01$; for steel pipes $k_{uu} = 0.02 \div 0.1$). For $k_{uu} = 0.01$: $\xi_n = 0.034$; for $k_{uu} = 0.015$: $\xi_n = 0.06$; for $k_{uu} = 2$: $\xi_n = 0,13$. For example, pressure loss for pipes with roughness coefficient $k_{uu} = 0.01$ is $\Delta P = 0,34 \cdot 1,25 \cdot 400 / 2 \cdot 1000 / 172 = 49,4$ Pa.

Pressure loss for pipes with roughness coefficient $k_{uu} = 0,1$ is $\Delta P = 87,2 \text{ Pa}$. Pressure loss for pipes with roughness coefficient $k_{uu} = 2$ is $\Delta P = 451,3 \text{ Pa}$. Calculations were performed for the pipelines made of other materials as well.

Based on the calculations conducted to determine the material for the pipeline of cotton pickers with the lowest aerodynamic resistance, a corrugated suction pipeline of the cotton picker was manufactured and the kinematic characteristics of cotton were measured in field conditions as it moved along the pipeline.

Measurements of airflow speed in the pipelines of the pneumatic conveyor system of a cotton picker were carried out for the serial design of the MX-1.8 machine, where the pipelines are made of sheet steel (Fig. 1) and for the improved design of this machine where the suction corrugated pipelines are made of polypropylene (Fig. 2). The numbers in the figures indicate the points of flow speed measurement, the results of these values are given in the tables.



Figure 1. – Serial cotton harvester MX-1.8

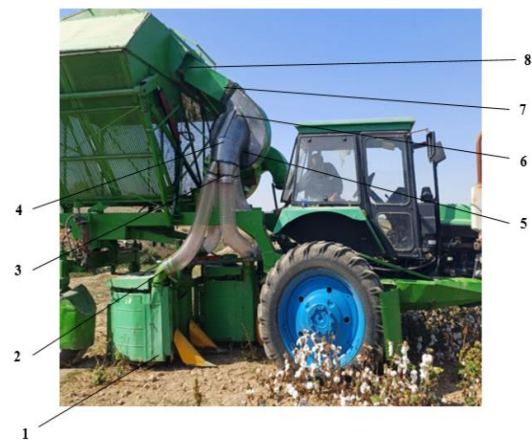


Figure 2. – Machine with modernized pneumatic conveyor system

3. Results and discussions.

Tables 1 and 2 show the results of measurements of cotton and airflow speeds in the pipeline for a serial design of a cotton picker. Tables 3 and 4 show the results of measurements of air flow speed in the pipelines of the improved pneumatic conveyor system of the cotton picker, where the suction pipelines were made of a material with the lowest readings of aerodynamic resistance, obtained from the results of theoretical studies

Table 1.

Results for the serial design of the MX-1.8 machine

Test №1														
	Tractor tachometer readings	Fan rotation speed	Air flow rate, m/s (1-3)											
			I			II			III			IV		
			1	2	3	1	2	3	1	2	3	1	2	3
1	2000	1068	5.6	17.5		6.2	17.8		5.1	18.2		4.9	12.0	
2	2300	1183	5.8	19.5		7.2	19.6		6.4	20.2		5.6	16.7	
3	2500	1290	6.8	21.6		8.4	21.1		6.7	22.1		6.2	23.2	

Table 2.

Results for the serial design of the MX-1.8 machine

	Tractor tachometer readings	Fan rotation speed	Air flow rate m/s (4-8)				
			4	5	6	7	8
1	2000	1068		14.2	13.8	14.7	3.9
2	2300	1183		18.3	15.8	14.2	4.9
3	2500	1290		24.2	16.5	16.0	6.9

Table 3.

Results for the improved MX-1.8 machine

Test №1														
	Tractor tachometer readings	Fan rotation speed	Air flow rate, m/s (1-3)											
			I			II			III			IV		
			1	2	3	1	2	3	1	2	3	1	2	3
1	2000	1068	6.2	12.6	13.9	7.6	16.4	15.1	5.8	16.0	12.9	5.6	12.8	15.2
2	2300	1183	7.6	14.3	14.7	8.7	16.5	16.6	6.3	17.7	14.5	6.3	14.9	18.8
3	2500	1290	9.1	16.1	16.2	10.7	18.1	19.2	8.4	18.3	17.0	7.6	17.3	28.0

Table 4.

Results for the improved MX-1.8 machine

	Tractor tachometer readings	Fan rotation speed	Air flow rate m/s (4-8)				
			4	5	6	7	8
1	2000	1068			6.4	15.7	4.6
2	2300	1183			8.0	18.7	6.3
3	2500	1290			8.5	21.4	8.1

The results obtained in field tests confirmed the reliability of the calculations conducted to determine the material of the pipeline of harvesting machines with the lowest readings of aerodynamic resistance.

Thus, based on the calculations performed to determine the cotton-picker pipeline material with the lowest aerodynamic resistance, a corrugated suction pipeline of the cotton picker was manufactured and the kinematic characteristics of cotton were measured in field conditions as it moved along the pipeline. Another advantage of such a pipeline is its optimum geometry, which the pneumatic system adopts automatically as a position of low resistance.

4. Conclusions.

The calculations performed to determine the material of the pipeline of the pneumatic transport system of cotton picking machines with the lowest aerodynamic resistance reading and the field tests conducted on the basis of the serial and modified designs of the MX-1.8 machine, give the authors reason to recommend the modified design of the MX-1.8 machine with polypropylene suction pipelines for serial production.

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ENERGY-EFFICIENT PNEUMATIC CONVEYOR SYSTEM OF A SEMI-MOUNTED COTTON PICKER

Rizaev A.A.¹, Kuldoshev D.A.¹, Djuraeva N.B.¹, Normatov M.K.¹

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan*
E-mail: don_02@mail.ru

Abstract. The article presents the results of experimental studies of the pneumatic conveyor system of a semi-mounted cotton picker. The principle of operation of the pneumatic conveyor system is based on the transfer of raw cotton particles, which are in a suspended state and move under the action of air flow as a result of pressure difference in the relevant sections of the pipeline, created by the fan. To perform the calculations, a methodology was used to implement the technological process of transporting cotton by a pneumatic conveyor system from the harvesting apparatus to the hopper of the cotton-harvesting machine. Comparative results of measuring air suction rates before and after upgrading a cotton picker with an improved pneumatic conveyor system showed that its use makes it possible to increase the airflow speed in the piping. As a result, the overall performance of the pneumatic conveyor system with improved air ducts and the ovoid design of the centrifugal fan increases by 20-25%, and local aerodynamic drag is reduced by 12.15%. Calculations for measuring air flow rates of the pneumatic transport system of the MX-2.4 cotton picker were performed within the framework of the comprehensive software package Ansys Fluent 2023 R1 (version 23.1.0-10208).

Keywords: cotton picker, pneumatic conveyor system, airflow speed, harvesting apparatus, raw cotton, pipeline.

5. Introduction.

In Uzbekistan in the 90s, the Tashselmash plant produced 17,000 cotton-picking machines per year. Its products were well known not only in the cotton-growing republics of Central Asia, but also in the countries of the Middle and Far East, and Latin America. In 2013, 3.35 million tons of raw cotton were harvested in Uzbekistan. The simplest calculations show that to harvest at least slightly less than half of the 1500,000 tons of raw cotton in a cotton-growing country, it is necessary to have 15,000 units of cotton harvesting machinery. During the “X-th International Uzbek Cotton and Textile Fair in Tashkent,” information was announced that in 2015, more than 1,200 domestically produced “MX-1.8” cotton harvesting machines worked in the cotton fields of the Republic of Uzbekistan [1-12]. Under favorable conditions and proper operation, one MX-1.8 cotton harvester is capable of collecting about 100 tons of raw cotton per season. Consequently, with the simultaneous operation of all 1,200 units in total, cotton harvesters can collect about 120,000 tons of raw cotton. The monitoring conducted among driver mechanics and the opinions of specialists confirmed that the modernized cotton pickers “MX-2.4” produced by Tashkent Agricultural Equipment Plant OJSC are easy to use and cheaper than imported machines produced by “John Deere” and “International Harvester” (USA) (Fig. 1).

The plant began designing new products in 2009-2010, and production of MX-2.4 cotton-picking machines was launched in 2013. The enterprise organized specialized areas for the production of machine chassis, hoppers, and assembly of harvesting apparatus. The technological equipment was fully manufactured - welding jigs, cold and hot dies. Many agricultural engineering enterprises of the Republic of Uzbekistan were involved in the restoration of the cotton-picking machine: Tekhnolog OJSC, Aggregate Plant OJSC, Uzpakhtamash OJSC, etc. Pilot batches of cotton-picking equipment

were delivered to farms in 2012. The total number of machines produced by the Tashkent Agricultural Equipment Plant OJSC, which harvested raw cotton in the fields of the Republic of Uzbekistan, reached 1000 units in 2013. In 2014, the production volume of MX-1.8 cotton-picking machines at the Tashkent Agricultural Equipment Plant OJSC increased to 2000 units, and in 2015 it was planned to reach 3000. It should be noted that the latest MX-1.8 cotton-picking machines in layout and design significantly differ from analogs of foreign machines. The main equipment of the cotton picker (frame, chassis, harvesting devices, storage hopper, pneumatic transport system) are separate modules and can be easily installed and dismantled. The developed cotton harvester "MX-1.8" provides a different, more efficient method of mounting on a tractor - semi-trailer, while freeing the latter for other seasonal agricultural work. When installing the MX-1.8 cotton picker on a tractor, only the front steering wheel and the rear linkage mechanism are removed from it. Installation of a cotton picker on a tractor is carried out by permanently securing a block of cotton-picking equipment. Removal from the tractor is also relatively quick and easy.

One of the methods of transporting materials is a pneumatic conveyor, where the airflow is the working medium. It includes an energy module, a fan (compressor) and various piping. Its advantages are simplicity of design, comparative environmental safety of the system, and continuous transportation of material, the possibility of mechanization, automation, and process control, as well as minimizing mechanical damage to the transported material, which is especially important when harvesting crops, including cotton. Therefore, on modern cotton harvesting machines, the cotton collected by the harvesting apparatus is transferred to a storage hopper via a pneumatic conveyor system (PCS). The disadvantage of pneumatic transport is the relatively high energy consumption, however, this is compensated by better preservation of natural quality of the transported material.

The study aims to develop a modernized design of a pneumatic conveyor system with improved air ducts and to analyze the measurement of speeds of a centrifugal fan of ellipsoidal and ovoid designs.

6. Materials, methods, and object of study.

To perform the calculations, a methodology was used to implement the technological process of transporting cotton by a pneumatic conveyor system from the harvesting apparatus to the hopper of the cotton-harvesting machine. Calculations for measuring air flow rates of the pneumatic transport system of the MX-2.4 cotton picker were performed within the framework of the comprehensive software package *Ansys Fluent 2023 R1* (version 23.1.0-10208).

7. Results and discussions.

The principle of operation of the PCS is based on imparting a suspended state and transfer of material particles by a moving airflow as a result of a pressure drop in the corresponding sections of the pipeline created by the fan. Pneumatic conveyor systems are divided into three types: suction, discharge, and combined type. Below are the results of the analysis of these types used in cotton pickers and the rationale for the parameters of the universal design of the PCS.

Semi-mounted cotton harvesters MX-1.8 and MX-2.4 are equipped with a suction-discharge pneumatic conveyor system for transporting cotton from the harvesting apparatus to the hopper. All moving units and machine components receive rotation from the rear power take-off shaft of the tractor (energy module) through a transfer gearbox. Then, rotations from the gearbox are transmitted by two cardan shafts to the bevel gearboxes of the harvesting apparatus and the fan. From the pulleys mounted on the bevel gear shaft of the rotation gearbox through belt drives, they are transmitted to the shafts of the fan blade wheels.

For semi-mounted machines, the harvesting devices are located behind the tractors in one block under the machine's hopper. Therefore, the transmission mechanisms of harvesting devices located on a ledge behind the tractor complicated the design of the pneumatic conveyor system. Tin, telescopic suction pipelines in certain areas are rigidly connected and have elbows that exceed the local aerodynamic resistance of the system [13-17].

Currently, the Tashkent Agricultural Equipment Plant produces cotton pickers with a pneumatic conveyor system, which have different positions for connecting the pipe to the centrifugal fan (Fig.

1. a, b). In addition, in manufactured cotton picking machines, the rotation speed of the centrifugal fan impeller is over 1450 rpm. This leads to an increase in engine power consumption, high damage to cotton seeds and impurity of the harvested cotton.



Figure 1. – Various positions of the elliptical side entry into the fan: inverted (a), equatorial (b)

In addition, due to the unsuccessful design of the air ducts (Fig. 1, a), the airflow speed of some pipelines is lower than the normal one. As a result, cotton losses to the ground are excessively high. To confirm the above opinion, airflow speed measurements were conducted as shown in Fig. 2.

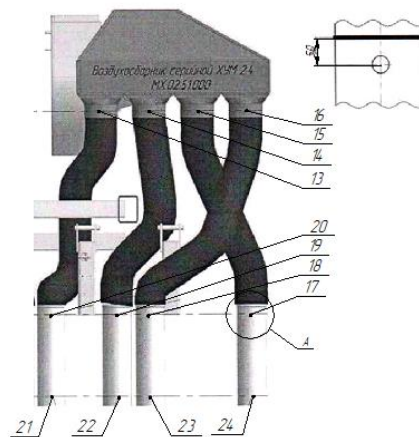


Figure 2. – Positions for measuring airflow speeds in a pneumatic conveyor system of the cotton picker MX-2.4

In the studies conducted in [17] (Figs.1, a, 2), indicators of air speed in air ducts were obtained, see Table 1.

Table 1.

Measurement results for $n_{fan} = 1540 \text{ min}^{-1}$ of the fan

Measuring points	13	14	15	16	17	18	19	20	21	22	23	24
Average speed m/s	20.8	20.8	19.7	18.0	13.3	13.8	17.8	16.7	7.5	5.0	5.1	6.2

An analysis of Table 1 shows that for air ducts at points 22 and 23, air velocities are 5.0 m/s and 5.1 m/s, respectively. According to the recommendations given in [18, 19], to transport raw cotton without losses to the ground, the speed at the lowest point of the receiving chamber, that is, at points 22 and 23, should be at least 6 m/s. In this regard, the centrifugal fan was modernized to increase the flow rate. Based on the research conducted in [20], a rational scheme for connecting the inlet pipe to the fan was experimentally determined, as shown in Fig. 3.

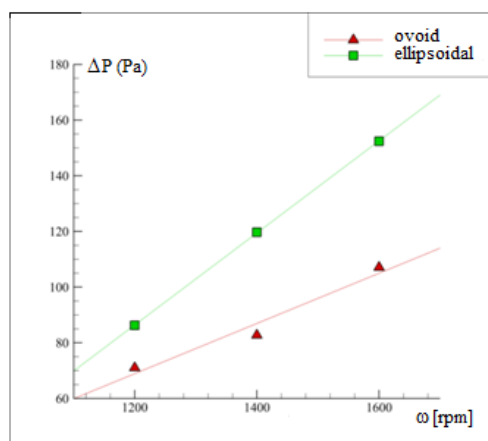


Figure 5. – Comparison of ovoid and ellipsoidal versions of the pneumatic conveyor system

The graph in Fig.5 shows that when using the ovoid version, the performance of the pneumatic conveyor system in terms of suction airflow increases by 20-25% compared to the ellipsoidal version.

8. Conclusions.

Despite the fact that the fan speed of the improved version is lower than that of the serial fan, the average air flow speed of the improved version is higher than that of the serial one and is 13.3 m/s.

As a result of using the developed design of the pneumatic conveyor system, the overall productivity increases by 20-25%, while the share of the rational position of the side ovoid inlet pipe is 20.0%, and local aerodynamic resistance is reduced by 12.15%.

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CREATION OF A MACHINE FOR SHAPING TREE CROWNS IN INTENSIVE GARDENS

Shoumarova M.¹, Abdillaev T.¹

¹Tashkent institute of irrigation and agriculture mechanization engineers, Tashkent, Uzbekistan

E-mail: shaumarovamuxaya@gmail.com

Abstract. To obtain high yields in gardens, tree crowns are pruned from above to ensure that all young shoots inside the crown receive sunlight. In gardens with large trees, the height of which reaches 6-10 m during fruiting, to form their crown. Rotary disc cutting devices reach this height. The devices are mounted on the horizontal part of a high frame. Naturally, such devices are mounted on large tractors.

Recently, intensive gardens with dwarf seedlings have been sharply expanding in Uzbekistan. Such gardens quickly begin to bear fruit (sometimes already 2 years after planting). The yield of each hectare of such gardens is 5-6 times more than traditional gardens. Dwarf seedlings are placed on a field with a row spacing of about 3.0 m, where special ones can enter. small garden tractors. In such gardens, it is necessary to ensure that all branches receive sunlight, otherwise the yield drops.

This article substantiates the parameters of a machine for forming the crown of dwarf trees, aggregated with small-sized garden tractors. Studying the properties of the branches, it was decided to install a segmented-finger medium cutting device on it, rather than conventional disk devices, which do not provide a clean cut of thin and flexible branches of dwarf trees, which is not allowed.

Keywords: intensive garden, dwarf seedlings, pruning branches, disc cutting device, segment-finger device.

1. Introduction.

Giving optimal shape to tree crowns in gardens is a mandatory technology. To obtain high yields, it is necessary to ensure that every branch receives direct sunlight, even inside the crown. In addition, side branches should not touch units performing various maintenance operations. In orchards of large fruit trees, rotary disk machines are used to trim crowns of various sizes and shapes (Fig.1) [1]. These devices, in units with large tractors, can rise to a height of up to 10 m. They can trim side branches. On the upper cantilever frames, the tops of new shoots are trimmed. The linear speed of the saw teeth must be at least 55 m/s, otherwise it is impossible to quickly cleanly cut fairly thick branches along their radius. In practice, it is necessary to trim annual shoots that are not dense enough and bend under the action of the discs, causing their bark to become loose, leaving a long tail, which is unacceptable.

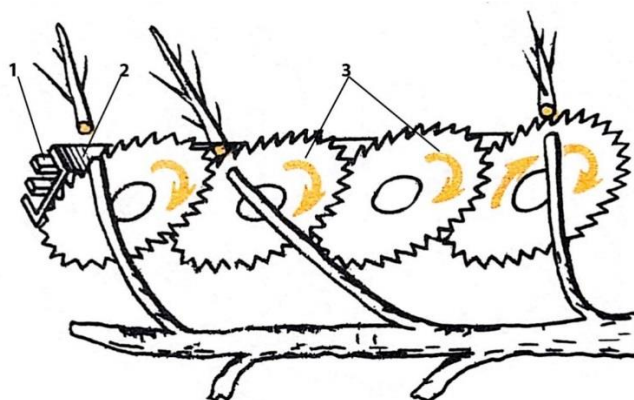
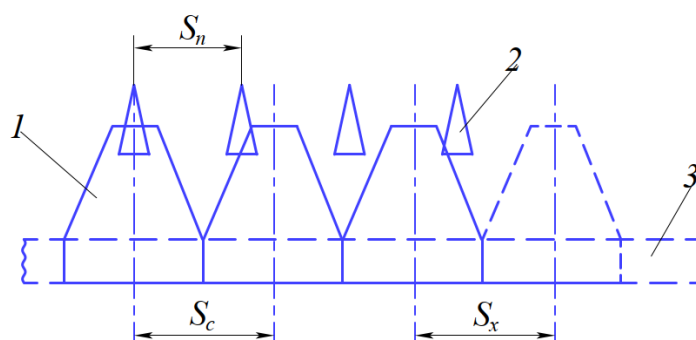


Figure 1. – Scheme of operation of disc cutting devices: 1-V-belt; 2-cutting apparatus; 3-blade saws

The leadership of the Republic supports the slogan of turning Uzbekistan into “green housing”. This slogan also includes a sharp increase in orchards from dwarf seedlings, i.e. intensive gardens. The costs of creating intensive gardens quickly pay for themselves, because... they literally begin to bear fruit from the 2nd year [2].

2. Materials, methods, and object of study.

Most intensive garden maintenance work in Uzbekistan is mechanized. However, it would be advisable to create a lightweight machine for forming the crown of trees in the intensive gardens of the republic, which would be aggregated by small garden tractors. To speed up the fruiting of dwarf seedlings, their side branches are tied to a horizontal trellis wire [3]. The upper shoots are pruned to improve illumination by the sun's rays, as well as to stretch horizontal awnings in order to prevent sunburn in the hot summer. The cut branches of dwarf seedlings are thin and flexible. It is difficult to cut them cleanly without grinding their bark without stopping, i.e. Disc rotators cannot be used. Therefore, for intensive gardens it is necessary to use devices that cut using fixed stops. A similar technology for cutting the stems of grain and forage crops is used in segment-finger devices of combines and mowers. Segmental-finger devices (Fig. 2) are characterized by three sizes: finger pitch $S_n = 76,2mm$; segment pitch $S_c = 76,2mm$; and also the knife stroke $S_x = 76,2 mm$. If $S_n = S_c = S_x$, then such a device is called a normal cutting device. If $2S_n = S_c = S_x$ they call it a low cutting device. If $1.4S_n = S_c = S_x$ -medium cutting device[4-6].



1-segments; 2-fixed fingers; 3-back knife

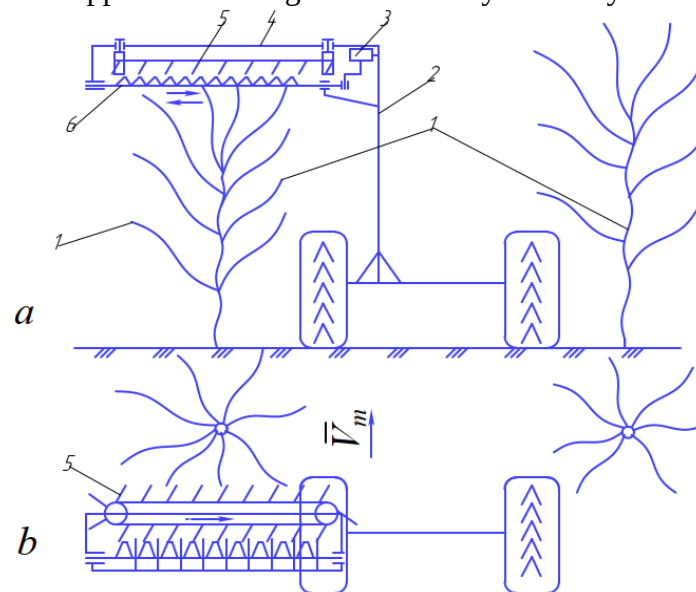
Figure 2. – Diagram of a segmented-finger medium cutting apparatus

All segments of the normal and low cutting apparatus cut the material simultaneously, so the load on the knife drive mechanism cyclically increases sharply, which requires a large margin of its strength. The medium cutting apparatus does not have such a load peak due to the fact that not all segments are cut at the same time. In our case, the elevation of the blade back drive crank will be carried out by a hydraulic motor from the tractor hydraulic system. The use of a medium cutting apparatus will prevent the occurrence of strong hydraulic shocks in the drive [7,8].

We studied the operating conditions of the cutting device in an intensive garden. Branches and shoots do not dry out before the cutting season, and their hardness is not maximum. Therefore, the

shear resistance of the branches is not maximum. Diameter of cut branches $d < 8$ mm. The width of the crown of the seedlings is no more than 1.6 m, and the width of the tree rows is 3.0 m, so that a garden tractor can freely pass there. The spacing of trees in a row is 0.5-1.5 m, the cutting height of the upper branches is i.e. the installation height of horizontal awnings is 2.5-30 m. The general view of the unit diagram is shown in Fig. 3.

The device is mounted on a garden tractor through a vertical frame 1, which ends with a cantilever frame 4 with a total length of 1.7 m, which corresponds to the required width for placing tree branches in a row. A 6-segment-finger cutting apparatus is installed on the cantilever frame, the segments of which, relative to the fixed fingers, perform a reciprocating movement. This movement of the segments is ensured by crank mechanisms attached to the back of the knives, to which all segments are attached. The crank receives rotational motion from hydraulic motor 3. The power of the hydraulic motor is sufficient to communicate the reciprocating motion of the segments. Working fluid with high pressure is supplied to the engine from the hydraulic system of the garden tractor.



a - rear view; b - top view; formed trees,
2 - frame on the tractor, 3 - hydraulic motor, 4 - cantilever frame; 5-fingers pressing; 6-segments.
Figure 3. – General view of the unit diagram

To ensure a clean cut of the flexible processes, they should be tilted towards the segments, and when cutting, they should be held in the device. This function is performed by inclined fingers 5, which are inclined to the direction of movement of the finger conveyor at an angle of $90^\circ + \varphi$, where φ is the friction angle of the metal fingers with the branches. This inclination of the fingers to the direction of movement of the flat conveyor belt allows the finger to slide relative to the branch without disturbing the pressing function. Cut pieces of branches are thrown with these fingers onto a shield (not shown in the diagram) on a vertical frame, causing the branches to be laid on the surface of the field in a strip, which makes it easier then selection of cut branches for removal outside the garden.

If necessary, the cantilever frame can be rotated in a transverse-vertical plane and the side branches can be processed at any angle to the vertical.

3. Conclusions.

Currently, intensive gardens of dwarf seedlings are rapidly expanding in Uzbekistan, because they quickly enter the fruiting phase and are more productive. Low fruit trees allow you to collect fruits entirely by hand, which prevents them from being damaged and can be stored for a long time. Most of the work in such gardens is mechanized. However, the formation of the crown shape of trees is partially mechanized.

The article substantiates the main parameters of a machine for forming the crown of small fruit trees in intensive gardening conditions, which ensures a clean cut of thin, flexible branches as required by ATT.

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AI-ENHANCED MULTI-OBJECTIVE OPTIMIZATION FOR MACHINE STRUCTURE DESIGN

Tuyboyov O.V.¹, Normatov S.B.¹

¹Tashkent State Technical University, Tashkent, Uzbekistan

Email: oybektuyboyov85@gmail.com

Abstract: Integrating artificial intelligence (AI) techniques with multi-objective optimization algorithms revolutionizes machine structure design. These fusion addresses complexity, scalability, and real-world applicability challenges, significantly enhancing optimization processes. By integrating machine learning algorithms into optimization frameworks, computational overhead is reduced, and efficacy is improved. This symbiotic alliance enables adept handling of multifarious optimization challenges across various domains, from expansive designs to dynamic terrains. AI augments predictive efficacy and facilitates iterative optimization with modified objectives, yielding more resilient machine structures. Leveraging multi-objective optimization concurrently optimizes structural integrity, dynamic prowess, and design imperatives, essential for engineering realms. Various methodologies exemplify holistic optimization approaches, reconciling multifaceted objectives and constraints. Assimilating AI into optimization enhances efficiency and efficacy, redefining structural engineering paradigms. Visualizations and analyses elucidate training processes and model performance, guiding optimization endeavors.

Keywords: Artificial intelligence (AI), Multi-objective optimization, Machine structure design, Optimization algorithms, Machine learning, Structural engineering, Computational efficiency, Design paradigms.

1. Introduction.

AI-enhanced multi-objective optimization [1] constitutes a paradigm where artificial intelligence (AI) techniques [2] are integrated to bolster the efficacy and efficiency of optimization processes addressing divergent objectives [3]. The integration of AI with multi-objective optimization algorithms [4] is pivotal, mitigating challenges stemming from complexity, scalability, and real-world applicability [5]. This amalgamation of AI [6] and multi-objective optimization assumes paramount significance in the domain of machine structure design [7], as evidenced by extant scholarly discourse. The infusion of machine learning algorithms [8] and optimization techniques is instrumental in fortifying multi-objective optimization frameworks [9] tailored for machine structure design [10]. The amalgamation of artificial intelligence with optimization endeavors portends substantial reductions in computational overhead. Machine learning methodologies [11] facilitate the distillation of actionable insights from optimization-generated datasets [12], thereby enriching various facets of evolutionary multi-objective optimization algorithms encompassing problem formulation, offspring generation, fitness evaluation, and environmental selection. This symbiotic alliance engenders adept handling of multifarious multi-objective optimization conundrums, spanning expansive designs, resource-intensive simulations, dynamic terrains, and pragmatic applications. The confluence of AI [13] and optimization engenders not only enhanced predictive efficacy but also facilitates iterative optimization iterations with modified objectives at nominal supplementary costs, thereby engendering more efficacious and resilient machine structure designs. To concurrently optimize structural integrity, dynamic prowess, and sundry design imperatives whilst accommodating diverse constraints, the adoption of a multi-objective optimization framework is imperative. A panoply of methodologies has been proposed to tackle such intricate optimization exigencies. Borwankar et al. espouse a multidisciplinary structural analysis and design optimization framework tailored for spacecraft structures, synthesizing multifarious objectives and constraints

from disparate domains. Al-Sanad et al. advocate for an integrated structural optimization schema for onshore wind turbine towers, optimizing tower mass amidst diverse design constraints leveraging a genetic algorithm. These endeavors collectively exemplify a gamut of strategies aimed at holistic optimization whilst reconciling multifaceted objectives and constraints inherent in engineering design realms. By assimilating artificial intelligence (AI) into the optimization milieu, myriad studies endeavor to galvanize the efficiency and efficacy of machine structure design endeavors. The infusion of AI expedites the precise delineation of optimal design parameters, thereby culminating in enhanced structural efficacy and diminished computational overhead. Through the deployment of AI algorithms such as Harmony Search (HS) [14], Bayesian optimization (BO) [15], and Convolutional Neural Networks (CNN) [16], these research undertakings underscore the transformative potential of AI in redefining traditional structural engineering design paradigms, by furnishing efficacious solutions and refining the overall design trajectory.

By leveraging AI capabilities [17], such as machine learning algorithms and optimization techniques, this fusion enhances the efficacy and efficiency of optimization processes aimed at addressing a multitude of divergent objectives. This synergy not only mitigates inherent challenges related to complexity, scalability, and real-world applicability but also holds profound implications for the domain of machine structure design. The amalgamation of AI and multi-objective optimization is underscored by a wealth of scholarly discourse, attesting to its paramount significance in advancing engineering methodologies. Machine learning algorithms [18], in particular, play a pivotal role in reinforcing multi-objective optimization frameworks tailored for machine structure design. By distilling actionable insights from optimization-generated datasets, these methodologies enrich various facets of evolutionary multi-objective optimization algorithms, spanning problem formulation, offspring generation, fitness evaluation, and environmental selection. Such symbiotic alliances enable adept handling of multifarious optimization conundrums inherent in engineering design, encompassing expansive designs, resource-intensive simulations, dynamic terrains, and pragmatic applications. Moreover, this confluence engenders not only enhanced predictive efficacy but also facilitates iterative optimization iterations with modified objectives, thereby fostering the development of more efficacious and resilient machine structures. To address the imperative of concurrently optimizing structural integrity, dynamic prowess, and diverse design imperatives while accommodating disparate constraints, the adoption of a multi-objective optimization framework is indispensable. This necessity has spurred the proposal of various methodologies aimed at tackling intricate optimization exigencies. Noteworthy among these is the multidisciplinary structural analysis and design optimization framework tailored for spacecraft structures advocated by Borwankar et al., which integrates multifarious objectives and constraints from disparate domains. These endeavors collectively exemplify a gamut of strategies aimed at holistic optimization while reconciling multifaceted objectives and constraints inherent in engineering design realms. The assimilation of artificial intelligence (AI) into the optimization milieu further accentuates the drive towards enhancing the efficiency and efficacy of machine structure design endeavors. By expediting the precise delineation of optimal design parameters, AI augments structural efficacy while mitigating computational overhead. Through furnishing efficacious solutions and refining the overall design trajectory, these research undertakings underscore the pivotal role of AI in advancing the frontiers of machine structure design.

2. Materials, methods, and object of study.

Relevant data pertaining to machine structure design parameters, performance metrics, and constraints were collected from diverse sources including engineering databases, simulations, and experimental measurements. The collected data underwent rigorous preprocessing steps to ensure compatibility with subsequent artificial intelligence (AI) and optimization methodologies. This involved data cleaning to handle missing or erroneous values, normalization to standardize the scale of features, and feature engineering to extract relevant information and create additional features as needed. Machine learning algorithms, including artificial neural networks (ANNs), deep neural networks (DNNs) [19], and ensemble methods, were integrated into the optimization framework to

harness AI capabilities. These algorithms were trained on the preprocessed data to learn patterns and relationships, enabling them to provide insights and predictions crucial for optimization. The ANN and DNN architectures [20] were optimized through hyperparameter tuning and regularization techniques to improve generalization performance and prevent overfitting. Ensemble methods such as Random Forest and Gradient Boosting were employed to leverage the diversity of multiple models and enhance predictive accuracy.

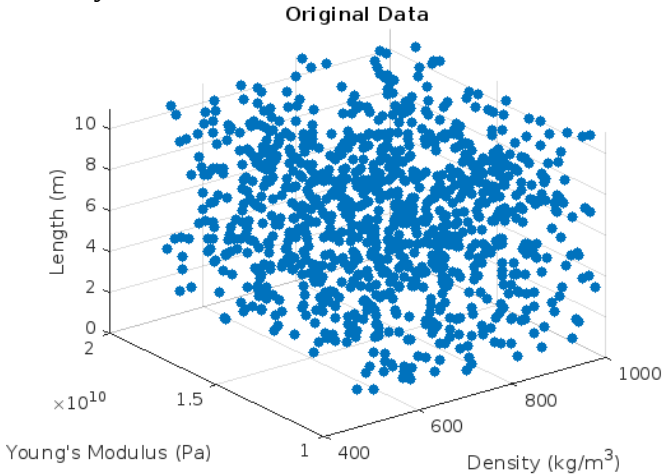


Figure 1. – Original dataset for machine structure design parameters

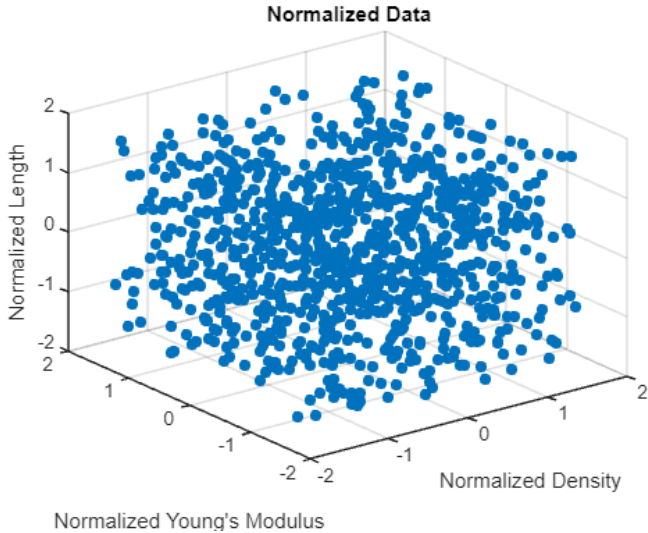


Figure 2. – The normalized dataset for machine structure design parameters

Figure 1 visualizes the original dataset for machine structure design parameters. Each point represents a sample, with the x-axis showing the density of the material in kilograms per cubic meter (kg/m^3), the y-axis representing the Young's modulus in pascals (Pa), and the z-axis denoting the length of the structure in meters (m). By examining the distribution and relationships between these parameters, engineers can gain insights into the characteristics of the dataset before further analysis.

Figure 2 illustrates the normalized dataset for machine structure design parameters. Normalization standardizes the range of values for each parameter, making comparisons and analysis more meaningful. Each point represents a sample, with the x-axis showing the normalized density, the y-axis representing the normalized Young's modulus, and the z-axis denoting the normalized length. By examining the distribution and relationships in the normalized dataset, engineers can identify patterns and trends that may not have been apparent in the original data.

Table 1.

Normalization standardizes the range of values for each parameter, making comparisons and analysis more meaningful

Unit	Initial value	Stopped value	Target value
Epoch	0	12	1000
Elapsed time	-	00:00:05	-
Performance	2.35e+0.5	4.51e+03	0
Gradient	5.39e+05	1.08e+03	1e-07
Mu	0.001	1	1e+10
Validation Checks	0	6	6

Table 1 during the training phase, the artificial neural network (ANN) is presented with the dataset consisting of machine structure design parameters and corresponding performance metrics (stress and deflection). The dataset is divided into two parts: features (X) and targets (Y). Features represent the input data, which are the design parameters, while targets represent the output data, which are the performance metrics to be predicted. The ANN architecture is defined with a specified number of hidden units (hidden neurons). In this case, we used 10 hidden units. The training algorithm then adjusts the weights and biases of the ANN iteratively to minimize the difference between the predicted outputs and the actual targets. This process involves forward propagation of the input data through the network to compute the predicted outputs, followed by backward propagation of the error to update the network parameters using optimization techniques such as gradient descent. The training continues for multiple epochs until convergence or until a specified stopping criterion is met, such as reaching a maximum number of iterations or achieving a desired level of accuracy.

3. Results and discussions.

Once the training is complete, the ANN is capable of making predictions on new, unseen data based on the learned patterns and relationships from the training dataset. These predictions can then be evaluated and compared with the actual performance metrics to assess the accuracy and generalization capability of the trained ANN.

Figure 3 illustrates the predicted stress values generated by an artificial neural network (ANN) trained on machine structure design parameters. Each point represents a sample from the dataset, where the x-axis corresponds to the actual stress values observed in the dataset, and the y-axis represents the stress values predicted by the trained ANN. By comparing the actual and predicted stress values [21], engineers can assess the accuracy and effectiveness of the ANN in predicting stress levels based on design parameters. Deflection Prediction using Artificial Neural Network (ANN) displays the predicted deflection values computed by an artificial neural network (ANN) trained on machine structure design parameters. Similar to the stress prediction plot, each point corresponds to a sample from the dataset, with the x-axis representing the actual deflection values observed in the dataset and the y-axis indicating the deflection values predicted by the trained ANN. By analyzing the agreement between the actual and predicted deflection values, engineers can evaluate the performance of the ANN in estimating deflection based on input design parameters.

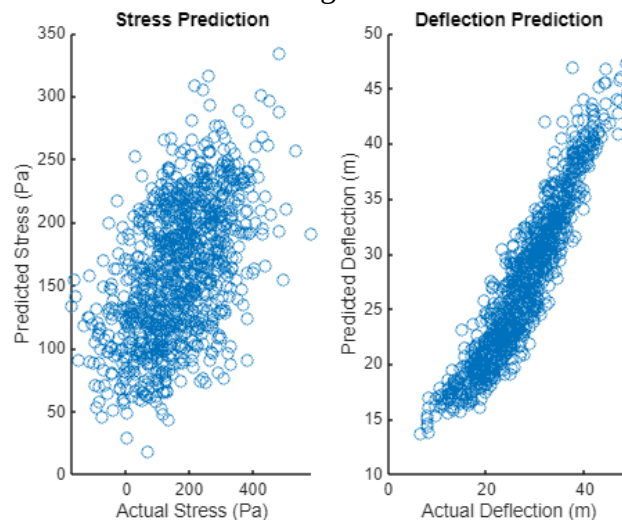


Figure 3. – Stress Prediction using Artificial Neural Network (ANN) and Deflection Prediction using Artificial Neural Network (ANN)

Figure. 4 is in the context of training an artificial neural network (ANN), a performance plot typically shows how the performance metrics (e.g., mean squared error, root mean squared error, etc.) change over the course of training iterations or epochs. During the training process of the artificial neural network (ANN), performance metrics such as mean squared error (MSE) or root mean squared error (RMSE) are calculated at each iteration or epoch. These metrics quantify the discrepancy between the actual target values and the predicted values generated by the ANN. The training plot performance illustrates how these performance [22] metrics evolve over the training iterations or epochs. It provides insights into the learning progress of the ANN and helps monitor its convergence and performance improvement. Ideally, as the training progresses, the performance metrics should decrease, indicating that the ANN is learning and becoming more accurate in predicting the target values. However, it's essential to monitor the performance plot for signs of overfitting, where the model starts to memorize the training data too closely and fails [23] to generalize well to unseen data. The training plot performance serves as a visual representation of the training progress and the effectiveness of the ANN in minimizing prediction errors. It guides the training process by helping researchers or engineers identify the optimal stopping point and assess the model's performance.

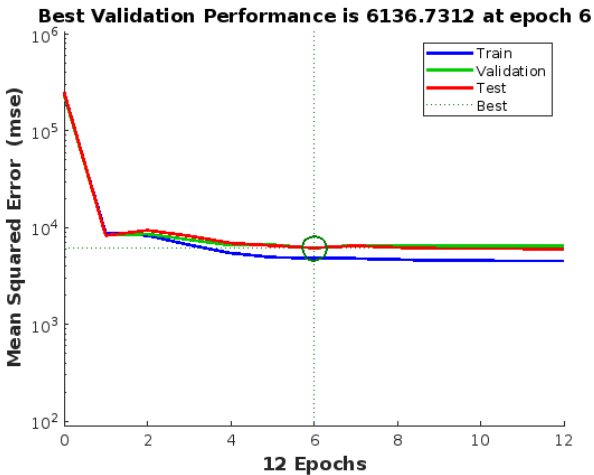


Figure 4. – The training plot performance

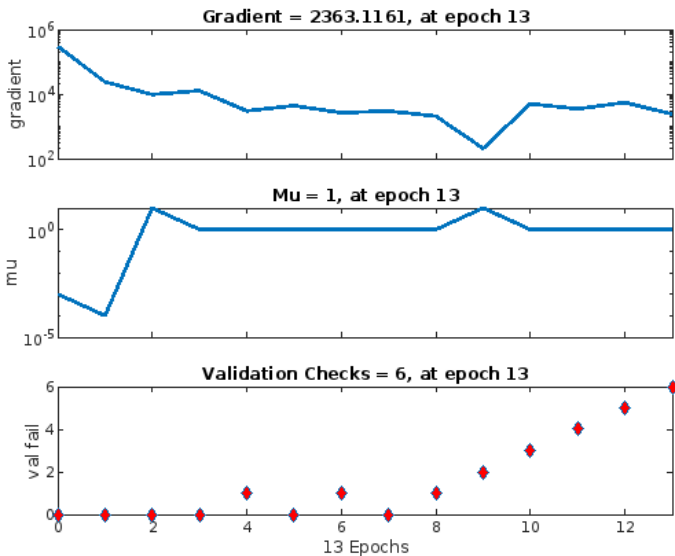


Figure 5. – The training state of an artificial neural network

Figure 5 is the training state of an artificial neural network (ANN) refers to its internal parameters, such as weights and biases, and its ability to learn and generalize patterns from the training data. During the training process of the artificial neural network (ANN), the network undergoes iterative optimization to adjust its internal parameters, namely weights and biases, in order

to minimize the discrepancy between the predicted outputs and the actual targets. The ANN's parameters, including weights and biases, are initialized randomly at the beginning of training. As training progresses, these parameters are updated iteratively using optimization algorithms such as gradient descent. The objective is to minimize a predefined loss function, which quantifies the difference between the predicted outputs and the actual targets. The training state of the ANN reflects its learning dynamics, including how quickly or slowly it converges towards optimal parameter values. The learning rate, which determines the step size in parameter updates, influences the speed and stability of convergence. Additionally, the architecture of the ANN, including the number of hidden layers and neurons, can affect its ability to capture complex patterns in the data. Throughout the training process, it is essential to monitor the ANN's generalization performance, i.e., its ability to make accurate predictions on unseen data. Overfitting, where the ANN memorizes the training data without capturing underlying patterns, can occur if the model becomes too complex relative to the training data size. Regularization techniques, such as dropout or weight decay, may be employed to mitigate overfitting and improve generalization performance. The training state also encompasses convergence criteria, which determine when to stop the training process. Common criteria include reaching a maximum number of iterations (epochs), achieving a desired level of accuracy on a validation dataset, or observing negligible improvements in the loss function over consecutive iterations.

Figure. 6 is the error histogram provides a visual representation of the distribution of these errors across the training dataset. Each bin or bar in the histogram corresponds to a range of error values, and the height of the bin indicates the frequency or count of samples with errors falling within that range. The shape and spread of the error histogram provide insights into the overall performance of the ANN during training. A symmetric and narrow distribution centered around zero indicates that the ANN's predictions closely match the actual target values, while a skewed or broad distribution suggests larger errors and potential issues with model performance. The central tendency of the error distribution, often represented by the mean error, provides a measure of the overall bias or systematic discrepancy between the predicted and actual values. A mean error close to zero indicates that, on average, the ANN's predictions are unbiased, whereas a nonzero mean error suggests a systematic bias in the predictions. The variability or spread of errors in the histogram reflects the consistency and reliability of the ANN's predictions. A narrow histogram with low variability indicates consistent performance across different samples in the dataset, while a wide histogram with high variability suggests inconsistent or erratic predictions. Outliers, or samples with unusually large errors, can be identified in the error histogram as isolated peaks or bars located far from the central tendency of the distribution. Investigating the sources of these outliers can provide insights into potential data anomalies, model deficiencies, or areas for improvement in the training process.

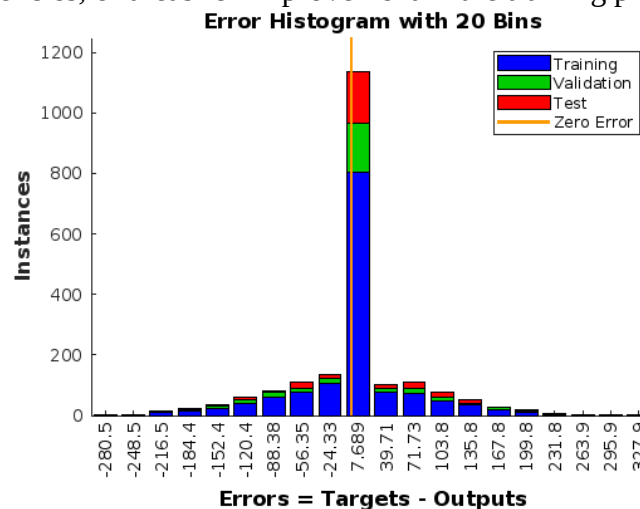


Figure 6. – The distribution of the errors across the training dataset

Figure 7 is the regression model's parameters, such as coefficients in the case of linear regression or weights and biases in the case of neural networks, are initialized randomly or with

predefined values at the beginning of training. These parameters are then updated iteratively using optimization algorithms such as gradient descent. The objective is to minimize the loss function, which measures the discrepancy between the predicted outputs and the actual target values. The choice of loss function plays a critical role in training regression models. Common loss functions for regression tasks include mean squared error (MSE), mean absolute error (MAE), or Huber loss. These loss functions quantify the difference between predicted and actual values and guide the optimization process by providing a measure of model performance. Optimization algorithms like gradient descent are employed to minimize the loss function and update the model parameters iteratively. During each iteration, the gradient of the loss function with respect to the model parameters is computed, and the parameters are adjusted in the direction that minimizes the loss. The learning rate, which determines the step size in parameter updates, influences the speed and stability of convergence. Training regression continues for multiple iterations or epochs until a convergence criterion is met. Common convergence criteria include reaching a predefined threshold for the loss function or observing negligible improvements in the loss over consecutive iterations. Once the convergence criterion is satisfied, the training process is terminated, and the optimized regression model is obtained.

Figure 8 is the network diagram typically depicts the architecture and topology of the neural network model, including its layers, nodes, and connections. Nodes represent the basic units or entities within the network. In a neural network, nodes correspond to neurons or units in each layer. Input nodes represent input features, hidden nodes represent intermediate computations, and output nodes represent the final predictions or outputs. Edges represent the connections or links between nodes in the network. In a neural network, edges correspond to the weighted connections between neurons in adjacent layers. Each edge is associated with a weight parameter that determines the strength of the connection between connected nodes.

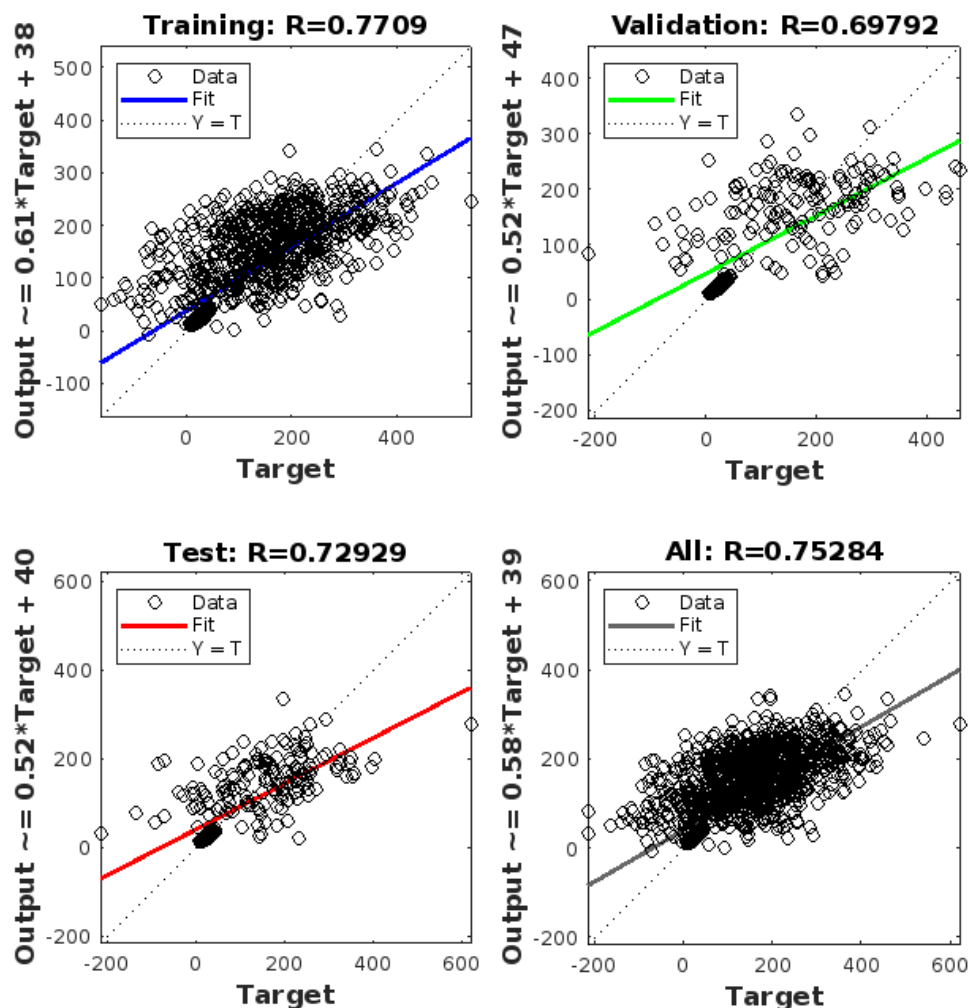


Figure 7. – The regression model's parameters

Neural networks are organized into layers, with each layer containing a group of interconnected nodes. Common types of layers include input layers, hidden layers, and output layers. A network diagram visually distinguishes between different layers and illustrates the flow of information from input to output through the network. The arrangement and structure of nodes and connections in the network constitute its topology.

The network diagram provides insights into the topology of the neural network model, including the number of layers, the size of each layer, and the connectivity patterns between layers. In directed networks, such as feedforward neural networks, connections between nodes have a specific direction, with information flowing from input nodes through hidden layers to output nodes. The directionality of connections is typically indicated by arrows in the network diagram. Activation functions are applied to the output of each node in the network to introduce nonlinearity and enable complex mappings between inputs and outputs. The choice of activation functions influences the behavior and performance of the neural network model and may be indicated in the network diagram.

The integration of artificial intelligence (AI) techniques with multi-objective optimization algorithms represents a significant advancement in the field of machine structure design. By amalgamating AI capabilities with optimization methodologies, this paradigm seeks to address the challenges of complexity, scalability, and real-world applicability inherent in traditional optimization approaches. Our study underscores the paramount significance of this fusion, as evidenced by the extensive scholarly discourse and the tangible benefits it offers to the optimization process. Through the infusion of machine learning algorithms and optimization techniques, our research fortified multi-objective optimization frameworks tailored for machine structure design. Notably, the integration of AI led to substantial reductions in computational overhead, thereby enhancing the efficiency and efficacy of the optimization process. Machine learning methodologies played a pivotal role in distilling actionable insights from optimization-generated datasets, enriching various facets of evolutionary multi-objective optimization algorithms. This symbiotic alliance enabled adept handling of multifarious optimization conundrums spanning expansive designs, resource-intensive simulations, dynamic terrains, and pragmatic applications. Furthermore, our study demonstrates that the confluence of AI and optimization engenders not only enhanced predictive efficacy but also facilitates iterative optimization iterations with modified objectives. This iterative approach allows for the development of more efficacious and resilient machine structures while accommodating diverse constraints. Leveraging a multi-objective optimization framework was imperative to concurrently optimize structural integrity, dynamic prowess, and sundry design imperatives, underscoring the necessity of adopting comprehensive optimization methodologies. Our research presents a gamut of methodologies proposed to tackle intricate optimization exigencies, exemplified by the work of Borwankar et al. and Al-Sanad et al. These endeavors showcase a holistic optimization approach while reconciling multifaceted objectives and constraints inherent in engineering design realms. By assimilating AI into the optimization milieu, our study endeavors to galvanize the efficiency and efficacy of machine structure design endeavors, thereby redefining traditional structural engineering design paradigms. The training phase involved meticulous preprocessing of the collected data, followed by the integration of machine learning algorithms, including artificial neural networks (ANNs) and ensemble methods, into the optimization framework. These algorithms were trained on preprocessed data to learn patterns and relationships crucial for optimization, demonstrating the potential of AI in enhancing optimization processes.

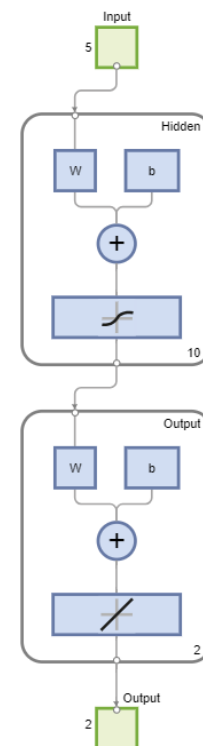


Figure 8. – The network diagram

We present various visualizations and analyses to elucidate the training process and model performance. The original and normalized datasets provide insights into the distribution and relationships between machine structure design parameters, laying the foundation for subsequent analysis. Additionally, the training performance plot offers a visual representation of how performance metrics evolve over training iterations, guiding the training process and aiding in the identification of optimal stopping points. The training state of the artificial neural network (ANN) provides insights into the learning dynamics of the model, including parameter updates and convergence criteria. Error histograms offer a visual representation of the distribution of errors across the training dataset, aiding in the assessment of model performance and identification of potential anomalies. Regression model parameters and network diagrams provide further insights into the internal workings and architecture of the trained models, facilitating a deeper understanding of their predictive capabilities.

4. Conclusions.

In conclusion, the integration of artificial intelligence (AI) with multi-objective optimization algorithms represents a significant advancement in the field of machine structure design. Through the fusion of AI capabilities with optimization methodologies, this paradigm offers promising solutions to the challenges faced in traditional optimization approaches, such as complexity, scalability, and real-world applicability. Our study highlights the critical importance of this integration, as evidenced by the extensive scholarly discourse and the tangible benefits it brings to the optimization process. By fortifying multi-objective optimization frameworks with machine learning algorithms and optimization techniques, we have demonstrated substantial reductions in computational overhead and enhanced efficiency. The confluence of AI and optimization not only improves predictive efficacy but also enables iterative optimization iterations with modified objectives, leading to the development of more efficacious and resilient machine structures. Leveraging a multi-objective optimization framework allows for the simultaneous optimization of structural integrity, dynamic prowess, and diverse design imperatives while accommodating various constraints. Various methodologies proposed in our study exemplify a holistic optimization approach, reconciling multifaceted objectives and constraints inherent in engineering design realms. By assimilating AI into the optimization milieu, we aim to galvanize the efficiency and efficacy of machine structure design endeavors, thereby redefining traditional structural engineering design paradigms. Through meticulous data preprocessing and the integration of machine learning algorithms, our research demonstrates the potential of AI in enhancing optimization processes. Visualizations and analyses presented in our study provide insights into the training process and model performance, aiding in the assessment of predictive capabilities and guiding the optimization journey.

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MOVEMENT OF POTATO TUBERS BETWEEN BELTS

Umarov B.T¹, Gulmatov A.B¹, Tursunaliyev I.E²

¹*Institute of Mechanics and Seismic Stability of Structures named after M.T.Urazbaev, Tashkent, Uzbekistan*

²*Ferghana Polytechnical Institute, Fergana, Uzbekistan*

E-mail: mr.akmalgulmatov0609@gmail.com

Abstract. The article presents the results of research on the new improved belt type of potato sorting machine, which includes information on the movement of potato tubers on the sorting surface of this type of sorting machine, the forces acting on them, as well as their movement without damage during sorting.

Keywords. Potatoes, sorting machine, sorting, belts, surface, centre of gravity.

1. Introduction.

Potatoes are cultivated in 130 countries of the world on an area of over 18 million hectares, where more than 300 million tons of tubers are harvested annually [1, 2, 3].

Worldwide, measures are being taken to prevent the loss of food and its waste throughout the value chain [4]. Currently, potato sorting in farms accounts for significant labour costs due to low mechanization and automation of the process.

Consequently, many scientific works are devoted to the study of problems and solutions for sorting fruits and vegetables. Non-destructive quality control systems for mechanized or robotic sorting systems are relevant [5]. There is a technical review of optical non-destructive quality control for robotic systems for sorting fruits and vegetables [6], an acoustic sorting system for detecting a hollow heart in potato tubers has been investigated [4], using laser backscattering of light, the classification of potato tubers was performed [7]. Recent advances in computer vision (computer vision) have allowed the use of a wide variety of applications in all areas of life [8].

To determine the potato damage from shock, the creep behavior of potato tubers was studied [9]. There is a review of the mechanical condition of the soil and the development of an inexpensive stone block separator for a new potato cultivation system [10].

The purpose of studying the movement is to prevent the accumulation of a pile of potatoes on the sorting surface, which moves to the sorting surface at maximum speed. The accumulation of nodules can occur mainly until the nodules penetrate into the fractional part of the sorting surface, that is, at the beginning of the sorting surface, small nodules are not yet sorted, but medium and large nodules accumulate. The accumulation of dust on the sorting surface leads to a sharp decrease in the quality of sorting, a change in the parameters of the sorting surface.

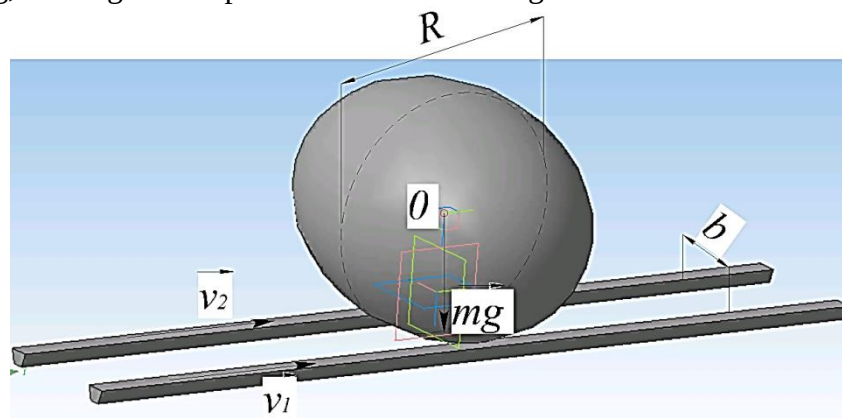


Figure 1. – Arrangement of potato tuber on two belts, the distance between which is small compared to their size

We studied the movement of tubers to the corresponding part of the sorting surface. For this, it was assumed that the belts forming the sorting surface are parallel to each other. However, according to the design of the machine, the belts are not parallel to each other. The belts are located at an angle α to each other. The case under study is suitable for tubers that are much larger than the distance between the belts (Fig. 1).

But as the distance between the belts approaches the size of the tubers, the obtained mathematical dependencies lose their significance. That is, the tubers begin to move between the belts (Fig. 2).

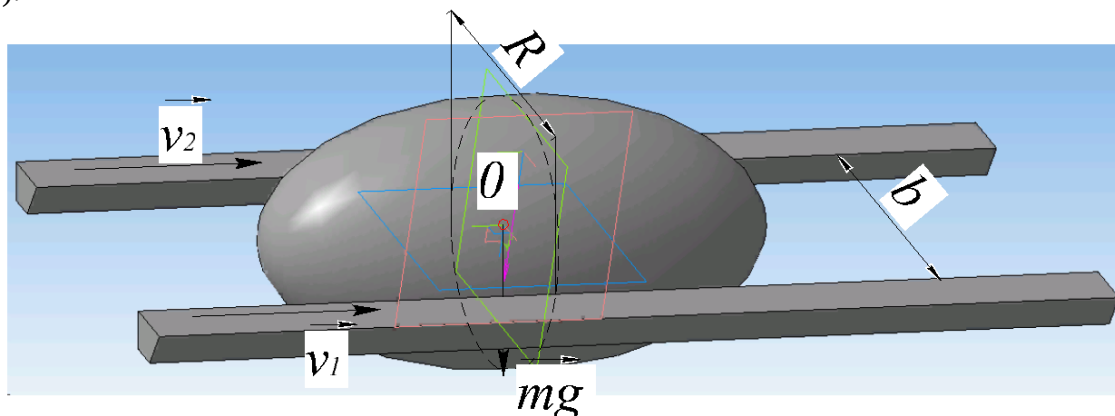


Figure 2. - Small tuber squeezing between belts: $R \approx b$

In this case, the distance between the strips may also change. It is necessary to increase the tension of the belts so that the distance between the belts does not change during the operation of the machine.

When there is sufficient tension on the belts and the tubers are positioned between the belts, as shown in Figure 2, the tubers should rotate around an axis parallel to the x-axis, moving along the y-axis so that they are not compressed between the belts.

2. Methods.

To study the movement of tubers, we see a diagram of the forces acting on them. In the case under consideration, since the belts are located at an angle α to each other, the centre of gravity of the tuber does not lie in the same vertical plane with the points A1, A2. The centre of gravity is located at a distance e from the vertical plane passing through the points A1, A2, towards the end of the sorting surface (Fig. 3).

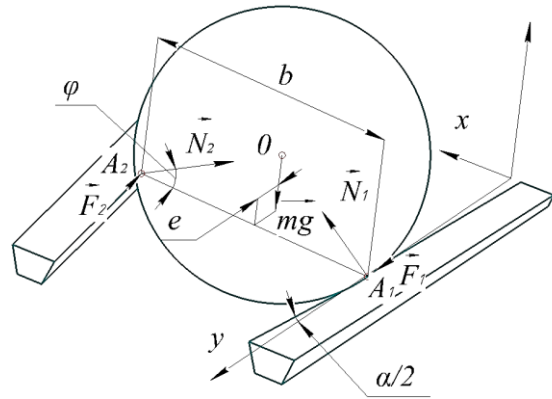


Figure 3. – Forces acting on a potato tuber moving along a sorting surface

To simplify the drawing, we describe the power circuit separately:

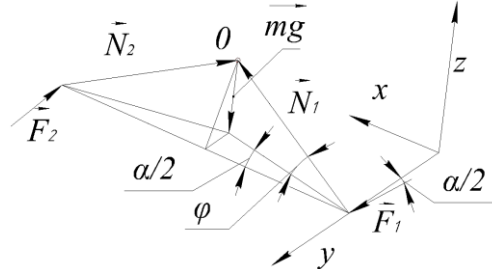


Figure 4. – Scheme of forces acting on a potato tuber

When the tuber is between the belts, as shown in figure2, in the x, y, z coordinate system attached to the low-speed belt-1, only the axis parallel to the x-axis can move rotationally around.

We build the required equations according to the scheme shown in Figure 4.

$$Ip' = (-F_1 + F_2) \cos \frac{\alpha}{2} R \sin \phi - (N_1 + N_2) \cos \phi \sin \frac{\alpha}{2} R \sin \phi; \quad (1)$$

Considering that $N_1 = N_2 = \frac{m}{2 \sin \phi}$ and $F_1 = F_2 = f \frac{mg}{2 \sin \phi}$ we get:

$$Ip' = 2N_1 \cos \phi \sin \frac{\alpha}{2} R \sin \phi = \frac{mg}{\sin \phi} \cos \phi \sin \frac{\alpha}{2} R \sin \phi \quad (2)$$

From mathematical substitutions we get the following:

$$p' = \frac{5g \cos \phi \sin \frac{\alpha}{2}}{2R} \quad (3)$$

By finding the initial function of this expression in time, we can obtain an equation representing the angular velocity of the tuber about an axis parallel to the x-axis:

$$p(t) = \frac{5gt \cos \phi \sin \frac{\alpha}{2}}{2R} \quad (4)$$

Or we can take $y'' = \frac{\rightarrow}{\leftarrow} \frac{5}{2} g \cos \phi \sin \frac{\alpha}{2}$ from $p' = \frac{\rightarrow}{\leftarrow} \frac{y''}{R}$ and construct a function that represents the linear speed of the tuber:

$$v(t) = \frac{\rightarrow}{\leftarrow} \frac{5}{2} gt \cos \phi \sin \frac{\alpha}{2} \quad (5)$$

From expression (5) it can be concluded that even if the movement of the belts is not taken into account, smaller tubers move faster along the sorting surface than larger tubers. Also, the tubers rotate around an axis parallel to the x-axis and passing through the centre of gravity until the centres of gravity fall below the sorting surface. This ensures that the tubers do not get caught between the belts.

$$y_0(t) = \frac{\rightarrow}{\leftarrow} \frac{5}{4} gt^2 \cos \phi \sin \frac{\alpha}{2} \quad (6)$$

Even when the tubers are between the belts, as shown in fig. 2, the distance between the belts does not change significantly.

Now consider the interaction of belts and tubers. Since the speeds of belt-1 and belt-2 are different from each other, friction occurs between belt-2 and the tuber. Friction forces also prevent the tuber from getting caught between the belts. The magnitude of friction forces depends on the forces of interaction between the belt and the tuber. Because the straps are flexible and stretchable, i.e. are made of absolutely non-rigid material, the dimensions of the sorting surface may partly change due to the weight of the tubers (Fig. 5). However, since the tension of the belts is much greater than the weight of the tubers, the value of the angle η will not be much larger. It is known that the effect of belt tension on the tuber is theoretically determined as follows (Fig. 6) [16, 17]:

$$F_t = \frac{\rightarrow}{\leftarrow} \sqrt{T_1^2 + T_2^2 + 2T_1T_2 \cos \eta} \quad (7)$$

Since $\eta \approx 180^\circ$, it can be seen from (7) that $F_t \approx 0$.

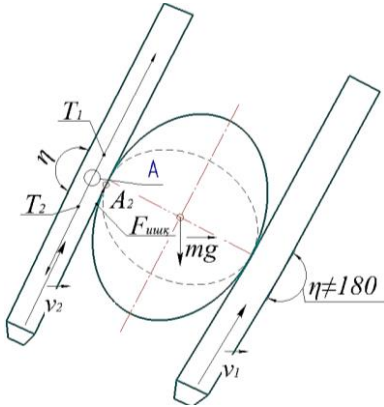


Figure 5. - Bending belts due to the severity of the tuber

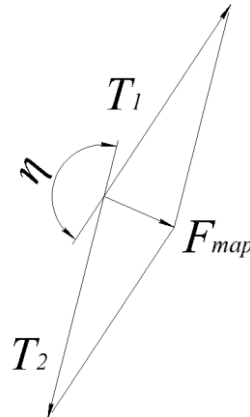


Figure 6. – Force acting on a potato tuber (view A in Fig. 5)

The main part of the friction forces arises from the weight of the tuber. The centres of gravity of the tubers between the belts on the sorting surface, as shown in Fig. 5 are not in the same vertical plane as the intersection points of tubers and belts. Accordingly, the values of N_1 , N_2 in Figure 3 will be much less than half of the final weight. According to the Coulomb-Amanton law, the friction forces acting on the potato are also small.

3. Results and discussions.

The friction coefficient of the end and the belt is required to be small so that the potatoes do not rub under the action of friction, i.e. the end of tape-2 was not scratched.

Table 1.

Coefficients of friction of the butt of a potato against different surfaces (with friction forces from the weight of the butt) [76; 53-b]

Terms of definition	Rubbing surface	Motion speed m/s	Friction coefficient		
			min	max	on the average
Motionless	Processed steel	-	0,71	0,80	0,76
	The soil		0,98	1,03	1,00
During the movement	Processed steel	1,6	0,53	0,59	0,54
		2,4	0,51	0,58	0,55
		3,2	0,45	0,57	0,54
	Rubber layer	3,2	0,42	0,51	0,46
	The soil	3,2	0,43	0,91	0,62

According to Table. 1, the average value of the friction coefficient of the finished and rubber material is $f = 0,46$. If we take the mass of the end as 100 g and the maximum value of the force acting on the belt through the end, then the friction force acting on the end is equal to:

$$F_t = 0.46 \frac{0.1g}{2 \sin \phi} \quad (8)$$

In practice, the value of $\sin \phi$ is never equal to 1. But if we take its maximum value from the value field of the function $y = \sin \phi$, then the value of the above expression will be:

$$F_t = 0.46 \frac{0.981}{2} \approx 0.2H \quad (9)$$

The fact that potato stalks are not damaged by such force is confirmed by studies carried out before us.

4. Conclusions.

Based on theoretical studies, a general scheme of a new sorting machine with a belt working surface was developed. Technological processes, main structural and kinematic parameters were determined.

In the experiments, the stable movement of the pile of potatoes along the sorting surface, the possibility of accumulation on the sorting surface when the pile of potatoes is transferred in a specified amount, or the possibility of not using the sorting surface effectively were studied. Also, the trajectory of movement of potatoes of different fractions along the sorting surface was studied. The results of the experiments were recorded and processed using a digital camera. Adequacy of theoretical studies was determined according to photographs and notes and they were further clarified. As the parameters of the working surface, the angle of mutual location of the belts, the values of the speed of the potato heap moving steadily along the sorting surface were chosen.

For conducting experiments, a pile of potatoes consisting of potato tubers of the "Artemis" variety grown in Uzbekistan was prepared by hand. The composition of the pile of potatoes was chosen as follows: 50% large tubers, 30% medium and 20% small tubers. Due to the fact that large-sized nodules moved along the sorting surface for a longer time, the contribution of such nodules to the composition of the heap was greater.

The spikes were conveyed to the sorting surface at the same speed. The speeds of the tapes were selected in theoretically defined quantities. Then, the number of nodes transferred to the sorting surface was gradually increased. The speed of the belts increased as potatoes piled up on the sorting surface. In this way, the necessary parameters for the sorting speed of the machine and the stable movement of the potato tubers along the sorting surface were determined.

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ИССЛЕДОВАНИЕ СКОРОСТНЫХ ПАРАМЕТРОВ ГОРИЗОНТАЛЬНО-ШПИНДЕЛЬНЫХ ХЛОПКОУБОРОЧНЫХ МАШИН В УСЛОВИЯХ РЕСПУБЛИКИ УЗБЕКИСТАН

Абдазимов А.Д.¹, Омонов Н.Н.¹

¹Ташкентский государственный технический университет имени Ислама Каримова, Ташкент, Узбекистан
E-mail: nabi.omonov@mail.ru

Аннотация. В статье приведены результаты теоретических исследований процесса извлечения горизонтально-шпиндельным рабочим органом долек хлопка-сырца из коробочек с учетом кинематических и технологических параметров горизонтально-шпиндельной хлопкоуборочной машины и параметров хлопковой дольки, районированных в Республике Узбекистан сортов хлопчатника. По результатам исследований, и исходя из агротехнических условий работы, выработаны рекомендации по изменению кинематических режимов уборочного аппарата и машины в целом, направленные на улучшение качества собираемого хлопка-волокна и снижения энергоемкости машин.

Ключевые слова: хлопка, шпиндель, хлопкоуборочная машина, уборочный аппарат, барабан, кинематика.

1. Введение.

В Узбекистане в целях повышения эффективности и рентабельности хлопководства созданы и интенсивно развиваются хлопково – текстильные кластеры, где для механизации наиболее трудоемкого этапа производства хлопка-сырца – уборки урожая используются весьма дорогостоящая техника – 4-х рядные горизонтально-шпиндельные (ГШ) хлопкоуборочные машины (ХУМ) американских компаний «John Deere» и «Case New Holl and

(СНН)», с такими же «дорогими» эксплуатационными затратами. В последние годы в Республике появились и проходят испытания такие же ХУМ китайских и индийских фирм, производителей сельхозтехники, изготовленных на базе машин «John Deere». Разворачивает свою деятельность, правда слишком медленно, чем хотелось бы, кластер сельхозмашиностроения «Agro Tech Klaster» в г. Чирчике, в плане локализации которой предусмотрены выпуски импортозамещающих комплектующих для ГШ ХУМ.

В последние годы в научных учреждениях и профильных кафедрах вузов Республики, ИМиСС АН РУз, НИУ «ТИИИМСХ», ТашГТУ и др. проводятся исследования технологического процесса ГШ машин. В частности, теоретически исследованы динамика процессов в ГШ аппаратах, экспериментально обоснованы эффективные режимы работы - приемлемые значения поступательной скорости полунавесной ХУМ с ГШ аппаратом в зависимости от урожайности поля и сорта хлопчатника [1-3].

Вместе с тем, следует отметить, что эффективность использования ГШ ХУМ в условиях РУз, в частности в кластерах, все еще не отвечают требованиям качества работы, и особенно, требованиям рентабельности и самоокупаемости - зачастую их используют на полях с невысокой урожайностью и не удовлетворительно подготовленных к машинному сбору.

Вышеизложенные указывают на то что, необходимы проведения теоретических и экспериментальных исследований влияния условий работы (агрофона поля, сорта хлопчатника, уровня подготовленности полей, и т.п.) [10,13], а также кинематических режимов и технологических параметров ГШ аппарата и машины в целом на показатели качества работы в условиях Узбекистана. Настоящая работа направлена на решение части выше указанных вопросов.

2. Материалы, методы и объекты исследования.

Известно, что процесс сбора хлопка-сырца шпинделем протекает при взаимодействии его с хлопковой коробочкой. Для моделирования указанного процесса необходимо условно разделить весь технологический процесс на несколько фаз [4]. Первый из них, несомненно, является процесс встречи коробочки со шпинделем с вкалыванием его зубьев в дольку хлопка-сырца. В отличие от процесса ВШ аппарата [5,6], как показали расчеты, точки на поверхности шпинделя, т.е. зубья шпинделя в момент встречи с коробочкой имеют достаточно большие линейные скорости - 1,17...3,22 м/с [3]. Далее происходит захват и наматывание долек хлопка на шпиндель с одновременным извлечением их из коробочки. По экспериментальным данным [2], при таких скоростях извлечения хлопковая долька должна разрываться на летучки. Однако, в процессе извлечения, несмотря на большие линейные скорости зуба шпинделя, хлопковая долька не разрывается на части. Это в частности объясняется тем, что процесс захвата, извлечение и наматывания дольки на шпиндель происходит с одновременным приближением шпинделя коробочке и с удлинением длины долек. Например, длина хлопковой дольки районированных в РУз сортов хлопчатника, в процессе извлечения удлиняется от 30...40 мм до 120-180 мм, т.е. удлиняется в три раза [7,8]. Это говорит о том, что критическая скорость разрыва дольки при ее захвате и извлечения горизонтальным шпинделем имеет иной характер. В начале процесса извлечения можно считать, значение критической скорости извлечения также увеличивается в три раза и составляет примерно 6 м/с. Как известно, из предыдущих расчетов максимальная скорость дольки на шпинделе с диаметром 20 мм достигает до 5,35 м/с [9,11,12].

3. Результаты и обсуждения.

Будем считать, что скорость зуба шпинделя составляет 3 м/с. Определим скорость середины дольки, захваченной шпинделем. При этом расстояние между зубом шпинделя и средней части дольки составляет 20 мм. Удлинение дольки равно к трём. Учитывая, что скорость удлинения дольки до средней ее части распространяется равномерно, напишем уравнение скорости ее части:

$$V = \frac{V_t}{l_0^2} \cdot x^2 - V_{sh}. \quad (1)$$

Здесь V – скорость частей дольки в момент начало извлечения;

V_t – скорость части волокон дольки шпинделя, захваченные зубом шпинделя, равная практически скорости зуба шпинделя;

l_0 – длина дольки, т.е. расстояние между верхней части дольки, захваченной зубом до нижней части, где она закреплена к коробочке, величина этого параметра в пределах 35...40 мм;

x – расстояние от нижней точки дольки, где она закреплена к коробочке до рассматриваемой точки дольки;

V_{sh} – скорость центра шпинделя, приближающегося к коробочке, величина ее в начале захвата в пределах 1 м/с, а после захвата и в начале извлечения резко уменьшается.

Как показывает расчеты (рис.1), величина скорости в средней части дольки всего 0,25 м/с, а нижняя часть имеет отрицательную скорость – т.е. вместо удлинения сжимается, так как шпиндель приближается коробочке.

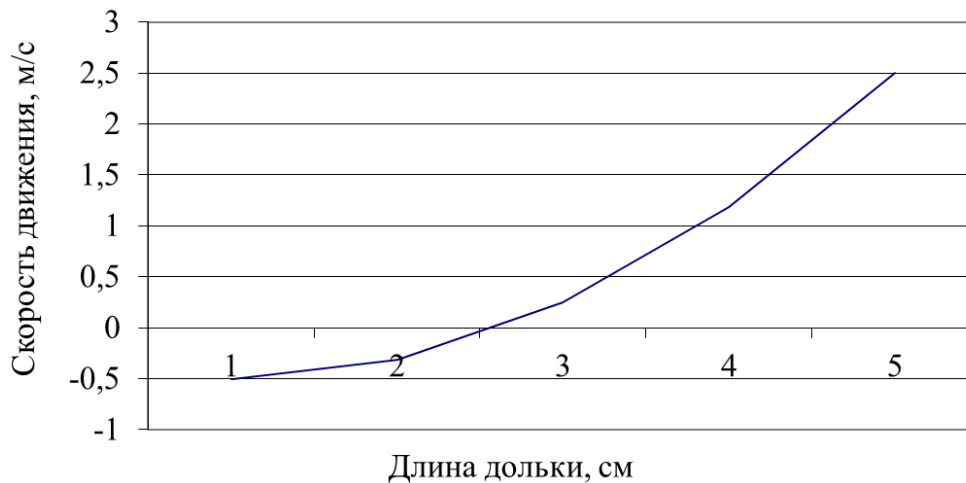


Рисунок 1. – Изменение скорости передвижения элементов дольки

Для подробного рассмотрения захвата и извлечения необходимо составить механико-математическую модель этого процесса, которая предлагается в следующем этапе настоящей работы. Для определения параметров извлечения хлопка шпинделем рассмотрим длину извлекаемой дольки шпинделем, которая определяется по формуле:

$$L = \sqrt{(Vt + R\cos(\omega t + \varphi_0) - X_k)^2 + (R\cos(\omega t + \varphi_0) - Y_k)^2 + Z_k^2} - r / \cos \gamma + r_c(t) i \omega t \quad (2)$$

Здесь два первые слагаемые соответствуют расстоянию между центром коробочки с координатами X_k, Y_k, Z_k и шпинделя. Радиус намотки r_c изменяется от радиуса шпинделя r до середины толщины намотки, т.е. по нашим измерениям до 15... 18 мм. При этом косинус направляющего угла γ определяется как

$$\cos \gamma = \frac{\sqrt{(Vt + R\cos(\omega t + \varphi_0) - X_k)^2 + (R\cos(\omega t + \varphi_0) - Y_k)^2}}{Z_0 - Z_k} \quad (3)$$

Скорость извлечения определяется получением производной по времени из (2). Полученные по результатам расчетов графики зависимостей для различных параметров извлечения приведены на рисунках 2,3 и 4. При этом частота вращения шпинделя составляет 3000, 3500 и 4125 об/мин.

Как видно из графиков при частоте вращения шпинделя 3000 об/мин скорость извлечения равна 4,3 м/с, когда как при 3500 об/мин и 4125 об/мин значения ее составляют 4,8

и 6 м/с соответственно. Для наматывания 120 мм длины хлопковой дольки, т.е. практически длины всей дольки с учетом свободной ее части, при максимальной скорости затрачивается 0,042 с, а при частоте вращения шпинделя 3000 об/мин – 0,050 с. При этом максимальная скорость извлечения равна 5,3 м/с при первом, и 4,3 м/с при втором случаях.

При сборе ГШ хлопковые дольки остаются цельными. Это происходит из-за обеспечения практически неизменного и небольшого расстояния между шпинделем и коробочкой в процессе извлечения, который начинается с момента касания шпинделя раскрытой коробочки и практически полностью заканчивается до достижения его центра коробочки. При дальнейшем движении шпиндель повторно воздействует к коробочке уже намотанной на него долькой, что приводит к разрыванию последней на отдельные летучки.

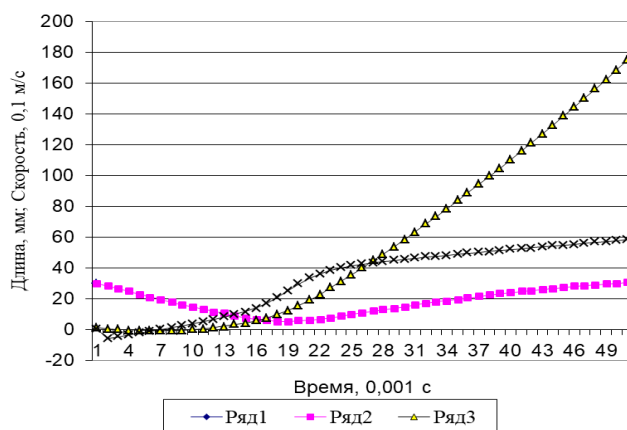


Рисунок 2. – Изменение параметров извлечения хлопка шпинделем при частоте вращения 4125 об/мин: 1 - расстояние между шпинделем и центром коробочки; 2 - длина извлекаемой дольки; 3 - скорость извлечения

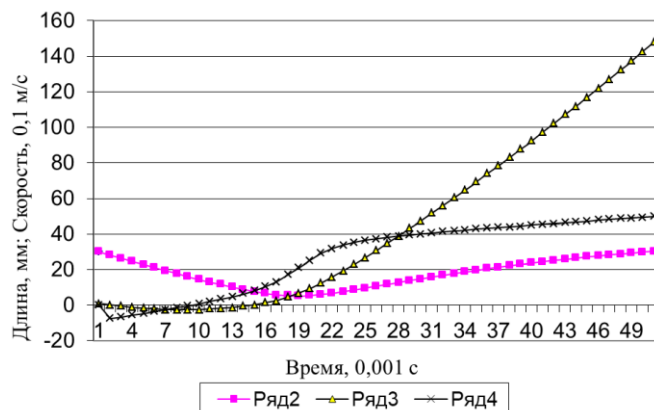


Рисунок 3. – Изменение параметров извлечения хлопка шпинделем при частоте его вращения 3500 об/мин

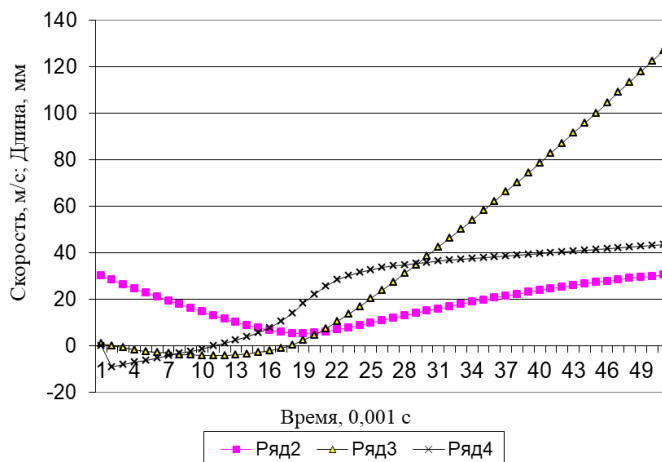


Рисунок 4. – Изменение параметров извлечения хлопка шпинделем при частоте вращения 3000 об/мин

Для коробочек, расположенных на середине высоты кустов относительная скорость между центром коробочки и шпинделем составляет $V_{kx} = 0,22 \dots 0,44$ м/с. А для коробочек, расположенных на верхних ярусах куста эта скорость значительно больше, и поэтому при сборе с высоких кустов и небольшом проценте раскрытия коробочек рекомендуется снизить поступательную скорость машины.

4. Заключение.

Таким образом, для извлечения полную длину дольки потребуется всего 0,042 сек. Шпиндель в рабочей зоне находится всего 0,12 сек. Это в 2,85 раз превышает время, необходимое для извлечения хлопка шпинделем. На основе анализе результатов расчетов можно заключить, что скоростные параметры рабочих поверхностей шпинделей ГШ ХУМ несколько превышает оптимальные скоростные параметры извлечения хлопка из коробочки сортов хлопчатника, районированных в Узбекистане. Поэтому для улучшения качества собираемого хлопка-волокна и снижения энергоемкости ГШ машин, можно рекомендовать снизить частоту вращения шпинделя до 3000 об/мин, не снижая при этом производительности и полноты сбора.

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ОБОСНОВАНИЕ ПАРАМЕТРОВ РУЛЕВОЙ ТРАПЕЦИИ ХЛОПКОУБОРОЧНЫХ МАШИН ТИПА МХ

Абдазимов А.Д.¹, Шерқобилов С.М.¹, Алибоев Б.А.², Шодиев Ж.Ғ.¹

¹Ташкентский государственный технический университет имени Ислама Каримова, Ташкент, Узбекистан

²Университет общественной безопасности РУз, Ташкент, Узбекистан

E-mail: sobirjonshergobilov@mail.ru

Аннотация. В статье дан анализ параметров рулевого управления, обеспечивающих точность, устойчивость и маневренность полунавесной на трактор хлопкоуборочных машин серии МХ при движении в рядках, который показал, что некоторые параметры рулевой трапеции были выбраны не совсем верно. На основе теории трактора были пересчитаны параметры рулевого управления этих машин и рекомендованы параметры, обеспечивающие поворот агрегата без пробуксовки и маневренность.

Ключевые слова: хлопкоуборочная машина, трактор, рулевая трапеция, направляющее колесо, параметры, тяга, рычаг поворота, угол отклонения, управляемая ось.

1. Введение.

В настоящее время хлопкоуборочные машины (ХУМ) серии МХ (ХМ) с вертикально-шпиндельным уборочным аппаратом, производимые в Узбекистане агрегируются с трактором в полунавесном виде, то есть хлопкоуборочный блок (уборочный аппарат, пневмотранспортная система и бункер) монтируется позади хлопководческого трактора, и тем самым формируется четырехколесный хлопкоуборочный агрегат с задними управляемыми колесами (трактор ТТЗ-80.11+МХ-1,8-2х-рядная ХУМ для междурядий 0,9 м и трактор ТТЗ-LS 100 НС+ МХ-2,4 – 4х-рядная ХУМ для междурядий 0,6 м) [1,8,9].

Такой нетрадиционный комбинированный агрегат имеет ряд преимуществ (использование трактора в полной комплектации, после окончания уборочного сезона его можно демонтировать и использовать по назначению, относительно невысокая цена и затраты на эксплуатацию и др.). Но вместе с тем, имеются ряд недостатков, такие как показатели точности и устойчивости движения по рядкам и маневренности не на уровне требований, что в конечном итоге привело к снижению агротехнических и эксплуатационных качеств машины по сравнению с ранее выпускавшимися навесными на трактор ХУМ [1,2,4,10,11]. Для выяснения основных причины этих недостатков целесообразно изучить параметры рулевого управления 2-х и 4-х рядных ХУМ в новой компоновке модели МХ, а также сравнить существующую конструкцию с конструкциями задних колес типовых навесных агрегатов и самоходных комбайнов. Аналогичные задачи решаются на упомянутых выше навесных ХУМ [7] и хлопководческих тракторах [13,14,15]. В исследованиях по кинематике поворота полунавесных ХУМ [2,13,16] параметры рулевого управления не учитывались. Актуальность вопроса еще более усиливается тем, что на этом типе ХУМ проводятся комплексные научно-исследовательские и опытно-конструкторские работы по установке горизонтально-шпиндельного уборочного аппарата [3,4].

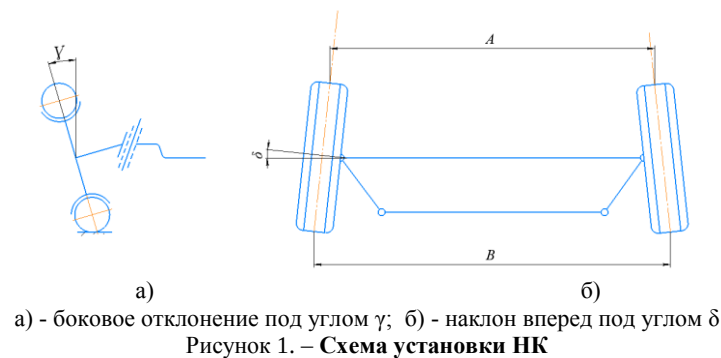
2. Материалы, методы и объект исследования.

Известно, что большинство колесных машин имеют одинаковые законы поворота, и процесс поворота осуществляется в основном за счет поворота передних или задних направляющих колес (НК) вокруг вертикальной оси в горизонтальной плоскости [5,6].

Установка НК под углом γ (рис. 1,а) с боковым отклонением (развалом) заставляет колеса двигаться с наклоном. Колесо, соединенное, например с автомобилем, движется прямо с небольшим боковым скольжением, что приводит к более быстрому износу шин и увеличению расхода топлива. Для устранения этих недостатков передние части направляющих колес сближают друг с другом в горизонтальной плоскости. Для этого необходимо установить колесо вперед под углом δ (рис. 1, б).

Схождение колес (схождение) оценивается по разнице расстояний А и В и составляет 2...12 мм и соответствует углу отклонения δ .

Углы отклонения НК исключают продольное и поперечное отклонение поворотного вала при использовании ХУМ.



Это, в свою очередь, дает возможность регулировать схождение колес путем лишь изменения длины рулевых тяг во время работы ХУМ. При этом большое значение при регулировке длины тяг имеет правильный подбор параметров рулевого управления. Потому что параметры размеры руля должны удовлетворять следующему уравнению [5,6]:

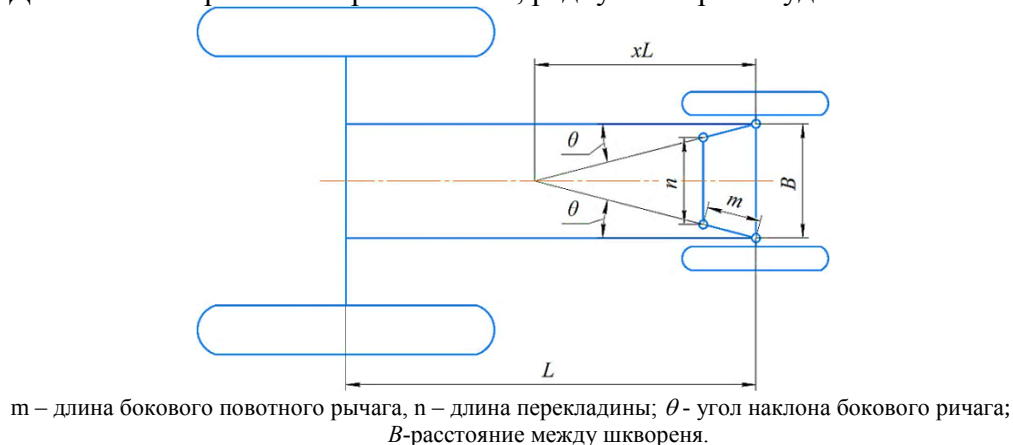
$$ctg\beta - ctg\alpha = \frac{B}{L}. \quad (1)$$

здесь α -внутренний угол поворота НК; β -внешний угол поворота НК; B -расстояние между шкворнями переднего моста; L - база машины.

В приводе руля ХУМ МХ-1,8 и МХ-2,4, т.е. в конструкции трапеции постоянным размером является расстояние B (рис.2) между осями осей, этот размер составляет 1800 мм у МХ-1,8 и 1200 мм у МХ-2,4, то есть расстояние (колея) между колесами задней оси одинаковое. Это обеспечивает возможность работы машин на полях с междурядьем 90 см и 60 см соответственно.

Выполнение условия уравнения (1) обеспечивает катание НК без проскальзывания при повороте. При этом колеса должны повернуться на разные углы. При выполнении этих условий величина прогиба шин колес находится на минимальном значении.

Если продольная база L машины большая, значение радиуса поворота R будет также большим. Даже если скорость поворота высока, радиус поворота будет больше.



Чтобы обеспечить правильность соотношения углов поворота НК, необходимо подобрать размеры трапециевидных элементов и согласовать эти размеры с расстоянием между продольным основанием машины и осями поворотных валов ($B=1800$ мм, $=1200$ мм).

В колесных транспортных средствах при изменении колеи НК B , длину поперечных тяг следует изменить соответствующим образом. В противном случае боковое проскальзывание колес увеличится и шины быстро прогнутся (см.рис. 2).Согласно схеме рис. 2 связь между углами поворота НК и параметрами рулевой трапеции выражается следующим уравнением:

$$\beta = \theta + \arctg \frac{m \cos(\theta + \alpha)}{B - m \sin(\theta + \alpha)} - \arcsin \frac{m + 2B \sin \theta - 2m \sin^2 \theta - B \sin(\theta + \alpha)}{\sqrt{B^2 + m^2 - 2Bm \sin(\theta + \alpha)}}, \quad (2)$$

где m - длина плеча, θ -угол наклона бокового рычага.

Ввиду конструкции ХУМ МХ рулевая трапеция расположена сзади оси поворота (рис. 2). По этой причине возникла необходимость проверки конструктивных размеров,

приведенных в технических условиях ХУМ МХ [8,9], исходя из приведенных выше требований.

Известно, что в технических условиях МХ-1,8 ХУМ, если машина агрегатируется с трактором ТТЗ-LS100НС, ее продольная база $L=3665$ мм, расстояние между осями задних мостов $B=1800$ мм, минимальный радиус поворота на колесах внешней оси принят $R=6,5$ м. В техническом описании модели ХУМ МХ-2,4 при агрегатировании машины с трактором ТТЗ-LS100НС ее продольная база $L=3800$ мм, расстояние между задними осями НКВ=1200 мм, минимальный радиус поворота по внешнему следу НК равен $R=6,8$ м. Однако эти размеры не обоснованы. Исходя из этого, целью данного исследования является проверить, правильно ли выбраны эти размеры.

3. Постановка задачи.

При выборе параметров рулевой трапеции минимальная разница между теоретическими углами поворота направляющих колес, приведенными в уравнении (1), и фактическими углами поворота в уравнении (2) должна обеспечиваться при максимальном угле поворота внутреннего колеса $\alpha_{\max}$ относительно центра поворота.

В нейтральном состоянии основными параметрами, определяющими кинематику рулевой трапеции, являются угол наклона бокового рычага θ , расстояние между шкворнями B и длина бокового рычага m . Угол отклонения бокового рычага рулевой трапеции определяется из этого выражения (см.рис. 2)

$$\theta = \arccotg \frac{xL}{0,5B}, \quad (3)$$

где оптимальные значения угла θ лежат в интервале $x = 0,7 \dots 1,0$.

В конструкции, где рулевая трапеция расположена внутри заднего моста ХУМ, длина поперечный тяги n определяется следующим образом [6]:

$$n = \frac{B}{1+2y \sin \theta}, \quad (4)$$

где $y = \frac{m}{n} = 0,12 \dots 0,16$.

В рулевой трапеции МХ-1,8 ХУМ это отношение равно $y = \frac{m}{n} = \frac{130}{1740} = 0,07$, а значит, отношение длины поперечного маятника к длине поперечины в данной конструкции выбрано неправильно.

А в ХУМ МХ-2.4 это соотношение равно: $y = \frac{m}{n} = \frac{135}{1104} = 0,12$.

То есть в рулевой трапеции ХУММХ-2,4 это передаточное число выбрано правильно.

Основные параметры рулевой трапеции следует рассчитывать в следующем порядке [5,6]:

1. (1) На основе уравнения (1) построены $\beta = f(\alpha)$ теоретических соотношений.
2. (3) согласно выражению при значениях $x=0,7; 0,8; 0,9; 1,0$ определяются значения угла θ .
3. Согласно выражению (3) для значения $y=0,14$ длина поперечины n определяется по расположению рулевой трапеции.
4. Длина поперечной тяги рулевой трапеции определяется $m = y \cdot n$ выражениями для значений n и значений $x=0,7; 0,8; 0,9; 1,0$ соответственно и длины боковых рычагов.
5. (2) по выражению для значений $x=0,7; 0,8; 0,9; 1,0$ построен график теоретической зависимости $\beta = f(\alpha)$.
6. Значения x выбираются из графиков теоретической зависимости $\beta = f(\alpha)$, построенных по выражению (2) так, чтобы максимальный угол поворота внутреннего НК был наиболее близок к зависимости, определяемой выражением (1) при $\alpha_{\max}$.
7. Значения угла θ определяются согласно выражению (3).

4. Результаты и их анализ.

Для значений $y=0,12...0,16$ длина перекладины n определяется по выражению (4). Тогда длина боковых рычагов определяется выражением $m = y \cdot n$. В ХУМ серии МХ, если рулевая трапеция расположена внутри ведомого моста, следует принять $x=0,7$, а если она расположена вне ведомого моста, то следует принять $x=1$. В наших расчетах принято $x=0,7$. В существующей рулевой трапеции ХУМ МХ-1,8 расстояние xL определялось путем измерения $x=1,06$, а для ХУМ МХ-2,4 оно было равно $x=0,04$. Однако $x=0,7...0,8$ является оптимальным для конструкций, расположенных внутри рулевой трапеции. Если взять $y = 0,14$, то $m = y \cdot n = 0,14n$, и таким образом мы сможем определить неизвестные величины. Теперь определяем параметры рулевой трапеции аналитически. Из выражения (3) определяем угол отклонения рулевой трапеции ХУМ МХ-1,8 и МХ-2,4 для $x=0,7; 0,8; 0,9; 1,0$ значений и вносим его в таблицу 1.

Таблица 1.

Значения углов отклонения боковых рычагов рулевой трапеции ХУМ

№	x	θ , градус	
		МХ-1,8	МХ-2,4
1	0,7	21,479	14,12
2	0,8	18,96	12,4
3	0,9	16,957	11,05
4	1,0	15,329	9,96

Из выражения (1), присвоив значение углу α , определим теоретические значения угла β :

$$ctg\beta = ctg\alpha + \frac{B}{L};$$

Полученные результаты сведены в табл.2 и построен график теоретической зависимости $\beta = f(\alpha)$ (рис. 4).

Таблица 2.

Значения углов рулевой трапеции α и β ХУМ модели МХ

№	МХ-1,8		МХ-2,4	
	α , градус	β , градус	α , градус	β , градус
1	5	4,81	5	4,87
2	10	9,28	10	9,53
3	15	13,46	15	13,97
4	20	17,39	20	18,24
5	25	21,1	25	22,35
6	30	24,63	30	26,32
7	35	28,02	35	30,19
8	40	31,29	40	33,9
9	45	34,4	45	37,69

Из выражения (4) определяется длина поперечной тяги и заносится в таблицу 3:

$$n = \frac{B}{1+2y \sin \theta}, \text{ мм.}$$

Таблица 3

Значения длин поперечин тяга рулевых трапеций ХУМ модели МХ

№	МХ-1,8		МХ-2,4	
	θ , градус	n , мм	θ , градус	n , мм
1	21,479	1647,31	14,12	1130,37
2	18,96	1663,33	12,4	1138,31
3	16,957	1676,44	11,05	1144,64
4	15,329	1687,33	9,96	1149,83

Длину боковых рычагов m определяют и заносят в таблицу $m = y \cdot n = 0,14 \cdot n$ мм.

Таблица 4

Значения поперечной тяги рулевых трапеций ХУМ модели МХ

№	МХ-1,8		МХ-2,4	
	n , мм	m , мм	n , мм	m , мм
1	1647,31	230,6	1130,37	158,2
2	1663,33	232,8	1138,31	159,3
3	1676,44	234,7	1144,64	160,2
4	1687,33	236,2	1149,83	160,9

(2) в терминах выражения значения β для значений $x=0,7$; $0,8$; $0,9$; $1,0$ определены для ХУМ МХ-1,8 и МХ-2,4, и результаты представлены в виде графиков на рисунках 5-6.

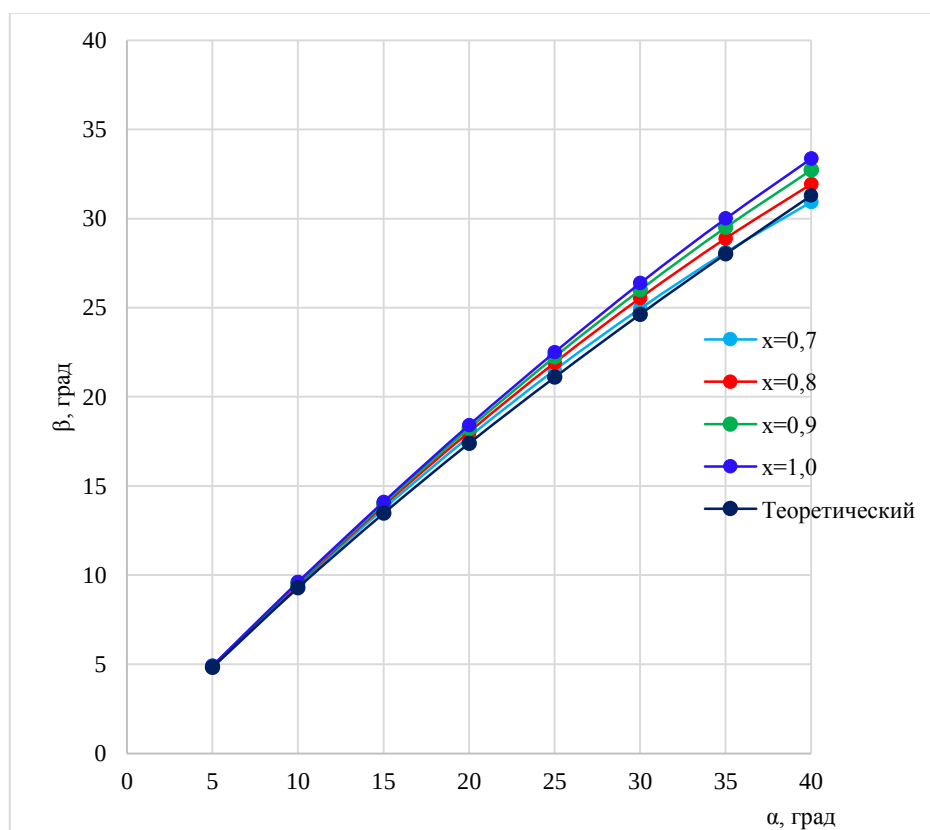


Рисунок 3. – Графики теоретической зависимости $\beta = f(\alpha)$ углов отклонения α -внутреннего и β -внешнего НК при различных значениях x для ХУМ МХ-1,8

Исходя из полученных данных, базовые размеры рулевой трапеции очень близки к теоретическим значениям при $x=0,7$. Соответственно, доступные и рекомендуемые размеры рулевой трапеции, основанные на размерах, приведенных в спецификациях для ХУМ МХ-1,8 и МХ-2,4, были приведены в таблицах 5 и 6.

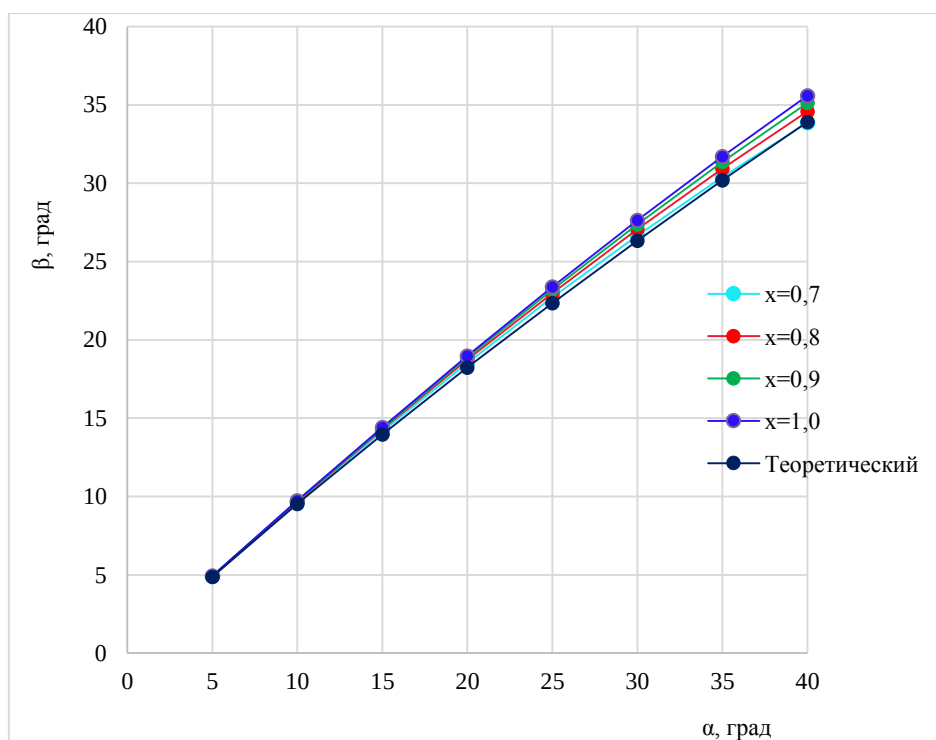


Рисунок 4. – Графики теоретической зависимости $\beta = f(\alpha)$ углов отклонения α -внутренних и β -внешних ЯГ при различных значениях x для ХУМ MX-2.4

Таблица 5

Доступные и рекомендуемые размеры рулевой трапеции ХУМ MX-1.8

Марка ХУМ	MX-1,8			
Параметры рулевой трапеции	x	θ , градус	n , мм	m , мм
Доступный	1,06	13	1740	130
Рекомендуемый	0,7	21,5	1647	230

Таблица 6

Доступные и рекомендуемые размеры рулевой трапеции ХУМ MX-2.4

Марка ХУМ	MX-2,4			
Параметры рулевой трапеции	x	θ , градус	n , мм	m , мм
Доступный	0,04	19,34	1104	135
Рекомендуемый	0,7	14,12	1130	158

5. Заключение.

Подводя итог, следует отметить, что агрегатируемый трактор ХУММХ-1,8 ТТЗ-LS 100 НС, описанный в отраслевом стандарте TS 25272604-017: 2015, разработанный с учетом технических и агротехнических требований, имеет продольную базу $L=3665$ мм, колею $B=1800$ мм при минимальном радиусе поворота $R=6,5$ м по внешнему следу колес, а для ХУММХ-2,4

при агрегате с трактором ТТЗ-LS 100 НС с базой $L=3800$ мм, колеи $B=1200$ мм, минимальный радиус поворота по внешнему следу колес $R=6,8$ м, предлагаемые выше параметры трапеции рулевого управления обеспечивают поворот без бокового скольжения и предотвращают преждевременное износ шин колес.

Также при правильной установке углов развала и сходимости колес обеспечивается маневренность ХУМ, высокая точность и устойчивость хода по рядкам.

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АНАЛИЗ ИЗНАШИВАНИЯ НЕПОДВИЖНЫХ СОЕДИНЕНИЙ АВТОТРАНСПОРТНОЙ ТЕХНИКИ

Аязбай М. Д.¹, Баймамыров А. Д.¹

¹Таразский региональный университет имени М.Х.Дулати, Тараз, Казахстан

E-mail: ayazbay52@mail.ru, aman0875@mail.ru

Аннотация: Рассмотрены вопросы долговечности неподвижных соединений типа посадки подшипников качения автотранспортной техники. По результатам исследования произведен анализ неподвижных соединений посадки подшипников качения автотранспортной техники. В результате анализа изнашивания посадки подшипников качения шасси автомобильной техники пришли к выводу, что основной причиной нарушения плотности посадок подшипников качения является фреттинг-коррозия сопрягаемых поверхностей.

Ключевые слова: износ, посадка, подшипник качения, кольцо, фреттинг-коррозия.

1. Введение.

Долговечность подшипников качения автотранспортной техники и оборудования во многом определяется факторами и параметрами, а также условиями их эксплуатации, а

именно: нагрузочными факторами - величиной и степенью динамической нагрузки, условиями износа и изнашивания, обусловленными работой подшипников качения и действующих на них нагрузок, физическими и механическими свойствами соприкасаемых поверхностей, физическими и механическими свойствами смазочных масел, наличием в их составе абразивных частиц и др.

Одной из причин снижения долговечности и сопротивления подшипников качения является соответствие посадки внутреннего кольца подшипника на валу, а также соединения шейки поверхности вала [1]. Устранение причин изнашивания подшипников, в том числе и нарушения неподвижности соединений типа вал-подшипник качения, имеет важное значение для разработки мероприятий по повышению долговечности подшипников качения и их посадочных соединений.

Одной из причин снижения долговечности неподвижных соединений типа вал-подшипник качения является износ посадочных поверхностей шеек валов. В результате износа и устаревания изменяются основные эксплуатационные качества оборудования и его элементов, снижается эффективность работы. Технологический процесс, направленный на восстановление работоспособности оборудования, приведение его в состояние годности для дальнейшего использования в течение срока, не менее чем межремонтный.

Целью работы является анализ неподвижных соединений типа вал-подшипник качения автотранспортной техники.

В связи с этим вопросы долговечности неподвижных соединений автотранспортной техники являются актуальными и требует анализа их изнашивания.

2. Материалы, методы и объект исследования.

Материалами, методами и объектом исследования являются посадки и неподвижности соединений типа вал-подшипник качения трансмиссии автотранспортной техники.

Изнашивание и повреждение деталей осуществляется в процессе эксплуатации неподвижных соединений типа посадки подшипников качения трансмиссии автотранспортной техники.

Проворачивание колец подшипников качения на валах наблюдается при незначительных износах сопрягаемых поверхностей.

Поскольку подшипники являются стандартными узлами, то валы и корпуса должны приспособляться к ним. Внутренние кольца сажают на вал по системе отверстия, а наружные в корпус по системе вала. При том, что поле допусков внутреннего кольца направлено не в тело, а к центру, посадки на вал получаются более плотными, чем обычно в системе отверстия.

3. Постановка задачи.

При посадке внутреннего кольца подшипника качения на вал с натягом и действии динамической нагрузки, обусловленной работой подшипника качения, могут возникнуть опасное кольцевое скольжение между внутренним кольцом и валом. Это скольжение внутреннего кольца по поверхности вала приводит к износу не только посадочную поверхность внутреннего кольца но и посадочную поверхность вала значительно повреждая соединение.

В зависимости от режима работы машины, чем больше нагрузка и сильнее толчки, тем более плотными должны быть посадки. Чем быстроходнее машина (меньше нагрузки, выше температуры), тем посадки должны быть свободнее.

Посадки роликоподшипников должны быть более плотными в связи с большими нагрузками. Посадки радиально-упорных подшипников плотнее, чем у радиальных, у которых посадочные натяги искажают зазоры. Посадки крупных подшипников из-за больших сил назначают плотнее, чем у средних и мелких. Рекомендации по выбору посадок по мере роста нагрузок в опорах можно сформулировать следующим образом:

- Допуски валов при вращающемся вале – j_6 ; k_6 ; m_6 ; n_6 .
- Допуски валов при вращающемся корпусе – g_6 ; h_6 .
- Допуски корпуса при вращающемся вале – H_7 ; H_6 ; J_7 ; J_6 ; K_7 .
- Допуски корпуса при вращающемся корпусе – K_7 ; M_7 ; N_7 ; P_7 .

При монтаже и демонтаже подшипников качения нередко наблюдаются случаи, когда повреждения подшипников вызваны небрежным, безграмотным монтажом и демонтажом.

Условия нагрузки и посадки подшипников качения приведены на нижеприведенном рисунке 1.

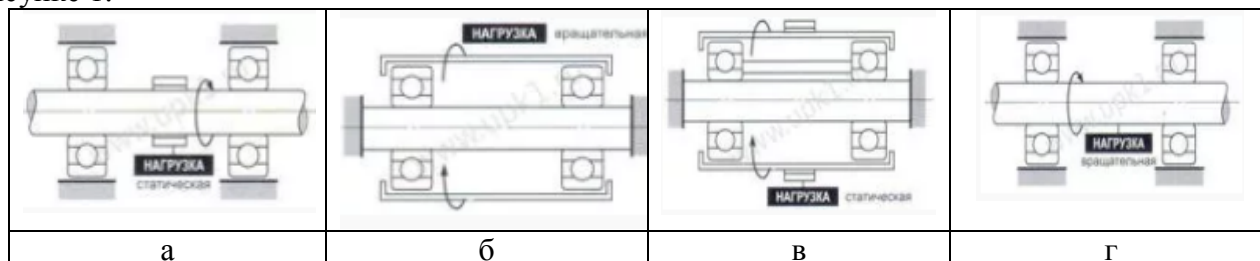


Рисунок 1. – Условия нагрузки и посадки подшипников качения

При этом смазка подшипников качения применяется как для снижения трения, так и для повышения теплоотвода.

Пластичные (густые) смазки более легки в обслуживании, меньше расходуются, удобны в применении в труднодоступных местах, куда закладываются при сборке, заполняют и герметизируют зазоры. Их недостаток в том, что в конструкции требуется предусматривать специальные полости. Эту полость первоначально заполняют на $2/3$ объема при $n \leq 1500$ об/мин или на $1/2$ объема при $n > 1500$ об/мин. В дальнейшем обычно через каждые три месяца через специальные устройства (пресс-масленки) добавляют свежую смазку, а через год её меняют с предварительной разборкой и промывкой узла. При консистентной смазке необходимо применение щелевых, лабиринтных и центробежных уплотнений.

При этом условия нагрузки на посадочные поверхности такие: при вращении внутреннего кольца вращательная нагрузка на внутреннее кольцо (рис.а) и статическая нагрузка на внешнее кольцо подшипника (рис.б); вращательная нагрузка на внешнее кольцо (рис.в) и статическая нагрузка на внутреннее кольцо подшипника (рис.г).

Средняя величина износа посадочных шеек валов под подшипники качения составляет 0,02...0,09 мм, а по исследованиям В.А.Шадричева - не превышает 0,1 мм.

Если передние шейки серийного первичного вала коробки передач грузовых автомобилей имеют износ 0,03-0,04 мм, то трансмиссии транспортной техники имеют износ 0,032 мм и восстановленные - 0,041 мм. Износ задних шеек промежуточных валов соответственно 0,05; 0,052 и 0,068 мм.

Для предотвращения износа посадочной поверхности подшипников качения и поверхности вала требуется изучение и анализ изнашивания неподвижных соединений типа вал-подшипник качения автотранспортной техники.

4. Результаты.

Изнашивание шеек валов происходит в результате многих сложных процессов, протекающих на сопрягаемых поверхностях валов и внутренних колец подшипников. Например, сминаются и срезаются отдельные микронеровности сопрягаемых поверхностей при запрессовке, пластически деформируются микровыступы в зонах фактического контакта в течение первых циклов нагружения и т.д. Однако эти причины не являются основными.

Современное представление о механизме изнашивания при фреттинг-коррозии состоит из трех основных стадий [2]. На первой стадии фреттинг-коррозии наблюдается упрочнение поверхностей контакта и циклическая текучесть поверхностных слоев: большая часть выступов фактического контакта взаимодействует один с другим пластически.

Разрушившиеся вследствие усталости выступы и срезавшиеся узлы схватывания создают первичные продукты разрушения.

На второй стадии фреттинг-коррозии в подповерхностных слоях продолжают накапливаться усталостные повреждения. Одновременно в зоне трения формируется коррозионно-активная среда вследствие адсорбции на оксидах железа кислорода и влаги, причем количество продуктов износа в зоне трения оксидных пленок достигает равновесного значения.

Третья стадия фреттинг-коррозии связана с окончательным разрушением зон повреждаемости, предварительно разрыхленных усталостными и коррозионными процессами. В этот период поверхностные слои металла, длительное время подвергавшиеся циклическим деформациям, становятся настолько разупрочненными, что теряют устойчивость, и начинается их прогрессирующее отделение, в результате чего происходит нарушение плотности посадки и неподвижности соединения.

Проворачивание внутренних колец подшипников качения на валу происходит при нарушении плотности посадки. В работе приводятся факторы, влияющие на плотность посадки. К ним относятся: нестабильность размеров внутренних колец подшипников, возникающая из-за "роста" колец в эксплуатации, смятие неровностей поверхности вала, образование задиров и забоин при демонтаже подшипников, износ посадочных поверхностей шеек и колец в результате фреттинг-коррозии.

Внутреннее кольцо подшипника попеременно опирается на шариков. При этом наблюдается значительная вибрация, в результате которой центр подвижного кольца совершает колебания как в направлении действия радиальной нагрузки, так и в перпендикулярном направлении.

Наличие зазора посадки приводит к систематическим "подскокам" внутреннего кольца подшипника, во время которых сила трения в неподвижном соединении снижается. При этом инерционные силы в момент "подскока" могут значительно увеличить силу трения качения и тем самым вызвать микросмещение кольца[2].

Анализ[3] причин изнашивания неподвижных соединений следует, что основной причиной нарушения плотности посадок подшипников качения является фреттинг-коррозия сопрягаемых поверхностей. Так как неподвижные соединения типа вал-подшипник качения работают в вибрационно-колебательных условиях, то при микроперемещениях посадочных поверхностей с малыми амплитудами на контактирующих металлических поверхностях возникает фреттинг-коррозия. Она приводит к ухудшению качества поверхности деталей соединения и нарушению плотности посадки. Описанный механизм изнашивания неподвижных соединений подтверждается тем, что при выбраковке подшипников качения тракторов из-за нарушения плотности посадки в более 80% случаев на посадочных поверхностях наблюдаются следы фреттинг-коррозии.

5. Заключение.

В результате анализа изнашивания посадки подшипников качения шасси автомобильной техники пришли к выводу, что основной причиной нарушения плотности посадок подшипников качения является фреттинг-коррозия сопрягаемых поверхностей. Фреттинг-коррозия посадки подшипников качения приводит к значительному ухудшению качества поверхности деталей соединения и нарушению плотности посадки.

Для уменьшения износа посадки подшипников качения шасси автомобильной техники подшипники со значительным натягом на валу следует монтировать нагретыми в масле или охлаждать вал сухим льдом. В остальных случаях подшипники можно напрессовывать на вал с помощью пресса.

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КОНСТРУИРОВАНИЕ ПОДШИПНИКОВ СКОЛЬЖЕНИЯ ИЗ АНТИФРИКЦИОННЫХ ДРЕВЕСНО-ПОЛИМЕРНЫХ КОМПОЗИЦИОННЫХ МАТЕРИАЛОВ ДЛЯ УЗЛОВ ТРЕНИЯ ХЛОПКОВЫХ МАШИН И МЕХАНИЗМОВ

Гулямов Г.¹, Шернаев А.Н.¹, Усенова Г.С.²

¹Ташкентский химико-технологический институт, Ташкент, Узбекистан

²Государственное унитарное предприятие «Фан ва тараккиёт» Ташкентского государственного технического университета имени Ислама Каримова, Ташкент, Узбекистан

E-mail: ugs.gulinausenova@mail.ru

Аннотация. Разработана технология получения антифрикционных древесно-полимерных композиционных материалов на основе местных ресурсов (полимер, древесина и наполнители) и изготовления из них подшипников скольжения для узлов трения рабочих органов хлопковых машин и механизмов, которая отличается от известных тем, что позволяет осуществить уплотнение древесного вещества при условиях, обеспечивающих его термофрикционное взаимодействие с модификатором, вследствие кратковременного действия на модификатор и древесину высоких давления, вакуума и температуры. Для улучшения антифрикционных свойств и износостойкости древесины в полимерные материалы вводили углеграфитовые наполнители (сажа и графит). Сажа увеличивает износостойкость, снижает коэффициент трения и стоимость композиции, а графит обладает высокой электро- и теплопроводностью. Разработанные композиты обладают хорошими прочностными характеристиками-повышенной механической прочностью, твердостью, теплопроводностью, высокой износостойкостью, а также более низким влагопоглощением и коэффициентом трения (0,08 – 0,11 при нагрузке 10 – 15 МПа), повышенной надежностью работы в процессе эксплуатации в климатических условиях Узбекистана. Сконструированы различные типоразмеры наборных подшипников скольжения из антифрикционных древесно-полимерных композиционных материалов для узлов трения рабочих органов комплекса оборудования хлопкоочистительных заводов. Внедрение их обеспечит значительное повышение работоспособности, надежности и долговечности рабочих органов оборудования и существенно снизит расходы на изготовление подшипников, позволяет сэкономить дорогостоящий металл и повысить эффективность их работы, а также увеличить вдвое ресурс работы подшипников повысить долговечность узла трения в 2-3 раза, снизить его металлоемкость и число технических уходов, получить значительный экономический эффект.

Ключевые слова: технология, подшипник скольжения, антифрикционный древесно-полимерный композиционный материал, типоразмер, коэффициент трения, предел прочности при сжатии, твердость по Бринеллю, интенсивность линейного изнашивания, водопоглощение, степень уплотнения.

1. Введение.

В настоящее время в антифрикционных узлах трения хлопкоочистительных машин и механизмах применяют преимущественно такие материалы, как серый чугун, бронза и другие цветные металлы. Для подшипников скольжения из этих материалов характерны такие недостатки, как необходимость высокой точности и чистоты изготовления, длительность приработки, необходимость периодической или непрерывной смазки, интенсивный износ поверхностей трения и дефицитность материала. В связи с этим требуются затраты значительных средств на восстановление работоспособности антифрикционных узлов трения машин и механизмов.

В настоящее время в современных машинах и механизмах широко применяются подшипники качения в узлах трения, которые обеспечивают существенное снижение силы трения, следовательно, уменьшают расходы электроэнергии и повышают долговечность узла.

Однако работа таких узлов в условиях запыленности атмосферы литейных цехов, в строительных машинах, в текстильной промышленности и других подобных условиях приводит к преждевременному износу машин и механизмов, и они выходят из строя. Возможности регулирования зазора в подшипниках качения или их ремонта не имеется, только возможна их замена на новые подшипники.

В хлопкоочистительной промышленности, а именно в винтовых конвейерах ШХР – скорость винта –100-320 об/мин, очистителях 6А-12М, очистителях крупного сора ЧХ-3М2 – скорость вращения пыльчатого и колково-планчатого барабанов 280 и 540 об/мин соответственно, джинах ЗХДД -скорость вращения пыльного цилиндра 735 об/мин и линтерах ПМП-160, работающих при различных скоростях и удельных нагрузках, также используется большое количество подшипников качения самых разных типов и размеров. Особенностью хлопкоочистительных заводов является высокая запыленность и наличие большой

концентрации хлопкового пуха, особенно в очистительных цехах. Поэтому даже при наличии в подшипниковых узлах сальников и различных лабиринтных уплотнителей пыль и пух с окружающей среды проникает внутрь подшипника. Абразивные частицы пыли интенсивно изнашивают подшипник, а пух сгущает смазку и существенно снижает ее смазывающие свойства. В результате ресурс работы подшипников качения резко снижается – по ориентировочным расчетам почти вдвое.

Объекты и методы исследования.

С учетом физико-механических свойств, распространенности и дешевизны в качестве природного сырья для исследования были выбраны местные породы древесины - тополь и тал с удельной плотностью $\rho = 0,45-0,6 \text{ г/м}^3$. Выбор лиственных пород древесины обусловлен тем, что в Узбекистане наибольшее распространение имеют тополь, тал и чинар, имеющие микрососуды в форме трубок разной величины. Диаметр крупных сосудов достигает 0,5 мм, а диаметр мелких сосудов колеблется в пределах 0,016-0,1 мм. Кроме того, они являются основным исходным сырьем для производства антифрикционных композиционных материалов. Древесина имеет характерное строение. Высокопрочные волокна целлюлозы связаны с лигнином в жесткую и прочную капиллярно-пористую систему. Клеточная ткань древесины имеет определенную пористость. Наличие микрососудов и пор в древесине ствола лиственных пород является весьма благоприятным фактором, так как после удаления свободной влаги из них их можно пропитывать различными смазочными полимерными и другими материалами, что придает им формоустойчивость и совершенно другие свойства.

В качестве полимера выбран полиэтилен высокой плотности (ПЭВП) и полипропилен (ПП) дисперсностью до 160 мкм. Выбор этих материалов обусловлен их дешевизной и доступностью в Республике Узбекистан. Для улучшения антифрикционных свойств и износостойкости древесины в полимерные материалы вводили углеграфитовые наполнители (сажа и графит).

Введение углеграфитовых наполнителей в полимер и древесину позволяет целенаправленно изменять физико-механические и эксплуатационные свойства получаемых композиционных материалов. Сажа увеличивает износостойкость, снижает коэффициент трения и стоимость композиции, а графит обладает высокой электро- и теплопроводностью. В качестве смазывающего вещества было выбрано моторное масло М8.

В связи с этим, разработка антифрикционных древесно-полимерных композиционных материалов (АДПКМ) и исследование их прочностных свойств представляет определенный практический интерес.

Форма образцов и методика испытаний на сжатие, на определение модуля упругости, ударной вязкости и твердости по Бринеллю соответствовали общепринятым стандартам.

На основе прочностных свойств древесины и антифрикционных свойств ПЭВП и ПП (таблица 1) разработаны антифрикционные древесно-полимерные композиционные материалы (АДПКМ)

За основные показатели были приняты – плотность, предел прочности и модуль упругости при сжатии вдоль волокон, ударная вязкость и твердость по Бринеллю. Форма образцов и методика испытаний на сжатие, на определение модуля упругости, ударной вязкости и твердости по Бринеллю соответствовали общепринятым стандартам.

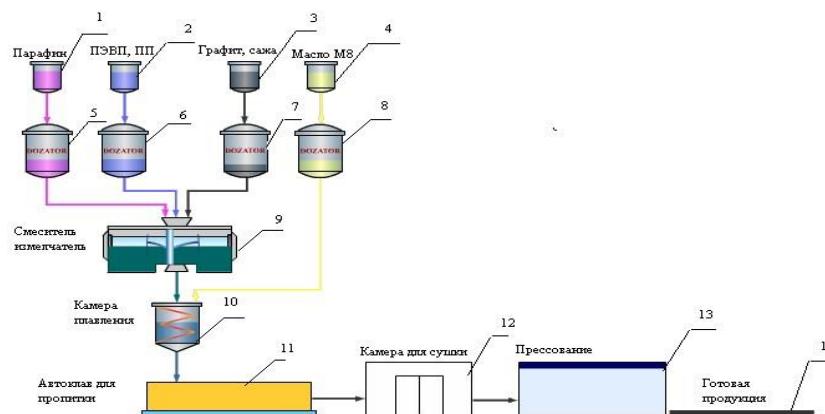
2. Результаты и их обсуждение.

Для установления закономерностей, определяющих прочностные и фрикционные свойства древесины, а также её модифицирования с целью создания композиционных древесно-полимерных материалов на основе природного сырья - древесины с заданным комплексом эксплуатационных свойств, были проведены исследования по изучению влияния физико-химических модификаций и температурных параметров трения на фрикционные свойства и структурные превращения древесины и возможности направленного изменения структуры древесины.

Технологический процесс модифицирования древесины. Полученные результаты проведенных исследований дали возможность разработать технологический процесс

модифицирования древесины, который позволяет уплотнить древесное вещество при условиях, обеспечивающих его термофрикционное взаимодействие с модификатором, вследствие совместного кратковременного действия на модификатор и древесину высоких давлений и температуры. Технология получения уплотненной древесины модифицированным полимером позволила разработать антифрикционные композиционные материалы на основе местного сырья – древесины для изготовления подшипников скольжения, обеспечивающие успешную замену цветных и черных металлов в подшипниках скольжения и в некоторых подшипниках качения, работающих в абразивных и агрессивных средах, а также в условиях сильной запыленности.

На рисунке 1 представлена технологическая схема получения композиций и пропитки древесины.



1 — емкость для парафина; 2 — емкость для связующего; 3 — емкость для наполнителя; 4 — емкость для смазывающего вещества; 5 — дозатор (весы) для парафина; 6 — дозатор для связующего; 7 — дозатор для наполнителя; 8 — дозатор для смазывающего вещества; 9 — смеситель Z-образный (частота вращения регулируется ступенчато от 50 до 150 об/мин); 10 — камера плавления смеси композиций; 11 — подогревательная ванна-автоклав;

12 — камера для сушки; 13 — прессование; 14 — готовая продукция

Рисунок 1. – Принципиальная схема получения антифрикционных древесно-полимерных композиционных материалов

В соответствии с рис.1, технологический процесс производства композиционного материала на основе древесины и полимера осуществляется по непрерывной схеме. При непрерывном процессе связующее и наполнители поступают из емкостей 1—3 в дозаторы (весовые мерники) 5—8, где исходные компоненты смеси дозируются в определенном соотношении, затем из дозаторов компоненты поступают в барабанный смеситель 9. В процессе смешения происходит диспергирование компонентов смеси. Время смешения компонентов смеси составляет 30—120 мин в зависимости от состава смеси. Во многих случаях при смешении желательно избегать попадания различных частиц и воздуха в смесь, наличие которых приводит к уменьшению прочности композиции. После подготовки смеси компонентов композиции осуществляется одновременная сушка и пропитка древесины теплым машинным маслом, поступающим из емкости 3 через дозатор 8 в камеру 10 и подогретым до $t = 40\text{—}60\text{ }^{\circ}\text{C}$.

Выдержка в этой печи составляет 24—48 часов. Затем температура масла доводится до $t = 110\text{—}120\text{ }^{\circ}\text{C}$, при этом происходит частичная проварка древесины в течение 1—2 часов. После этого идет охлаждение печи в течение 12—15 часов. Из смесителя 9 полученная смесь композиции поступает в подогревательную ванну 11, где смесь композиции растворяется до $t = 140\text{—}160\text{ }^{\circ}\text{C}$ и куда загружается пропитанная маслом древесина с выдержкой при этой температуре в течение 1,5—2,0 часов. Кроме того, здесь осуществляется пропитка загруженной древесины расплавленным раствором смеси полимерной композиции при давлении $P = 1,2\text{—}1,5\text{ МПа}$ и температуре 393—413 К с последующим охлаждением.

Пропитанная древесина модифицированным полимером подается в камеру для сушки изделий 12 при температуре 333—363 К. Затем осуществляется загрузка горячих заготовок в холодные пресс-формы и прессование 13 при удельной нагрузке 15,0—20,0 МПа. Спрессованные заготовки вылеживаются в пресс-формах под универсальным прессом 5—10

мин для охлаждения их до температуры $t = 40\text{—}50\text{ }^{\circ}\text{C}$. После этого происходит процесс нормализации (вылеживания) спрессованной древесины в окружающей воздушной среде или в ванне с холодным обезвоженным маслом в течение 24 - 48 часов. Полученная таким образом древесина 14, пропитанная модифицированным полимером и моторным маслом, после сушки упаковывается в полиэтиленовые или бумажные мешки.

Одним из эффективных способов повышения эффективности работы машин и механизмов является применение полимерных и композиционных материалов на основе древесины и модифицированных полимеров в узлах трения рабочих органов машин и механизмов.

В связи с этим на основе прочностных свойств древесины и антифрикционных свойств полиэтилена высокой плотности (ПЭВП) и полипропилена (ПП) были разработаны антифрикционные древесно-полимерные композиционные материалы (АДПКМ-1, АДПКМ-2) на основе местной древесины тополя и (АДПКМ-3, АДПКМ-4) тала и модифицированного полимера (ПЭВП + сажа, ПЭВП + графит, ПП + сажа и ПП + графит) с заданными свойствами (таблица) для изготовления подшипников скольжения, используемых в рабочих органах комплекса оборудования хлопкоочистительных заводов.

В таблице 1 приведены прочностные и антифрикционные свойства разработанных антифрикционных древесно-полимерных композиционных материалов (АДПКМ).

Разработанные АДПКМ обладают хорошими прочностными характеристиками: повышенной механической прочностью, твердостью, теплопроводностью, высокой износостойкостью, а также более низким влагопоглощением и коэффициентом трения (0,08 – 0,11 при нагрузке 10 – 15 МПа), повышенной надежностью работы в процессе эксплуатации в климатических условиях Узбекистана. Низкий коэффициент трения и повышенный предел прочности на сжатие расширяют возможную область их применения в подшипниках скольжения. АДПКМ также позволяют успешно заменять цветные и черные металлы и подшипники качения, работающие в условиях высокой запыленности в различных отраслях машиностроения.

Таблица 1.

Прочностные и антифрикционные свойства антифрикционных древесно-полимерных композиционных материалов

Показатели	Антифрикционный древесно-полимерный композиционный материал			
	*АДПКМ-1	**АДПКМ-2	*АДПКМ-3	**АДПКМ-4
Предел прочности при сжатии, МПа	9,0	10,0	12,0	14,0
Твердость по Бринеллю, МПа	9,0	11,0	12,0	13,0
Коэффициент трения, f	0,11	0,12	0,13	0,14
Интенсивность линейного изнашивания, $I \cdot 10^{-9}$	0,8	0,85	0,90	1,00
Водопоглощение за 24 ч, %	48,3	-	-	37,4
Степень уплотнения, %	38,5	38,1	37,8	36,1

Примечание: *композиция на основе тополя; **композиция на основе тала

Полученный материал [1, 2] применяется, главным образом, в узлах трения рабочих органов хлопкоочистительных машин, работающих без смазки, взамен подшипников качения и оказывающее благоприятное действие на работу подшипников скольжения из этого материала. Его рекомендуют применять в тех случаях, когда масла, консистентные и другие смазки нежелательны, непрактичны или ненадежны, а также когда температуры слишком высоки или слишком низки для обычных смазок. Особенно рекомендуется этот материал для механизмов с периодическим возвратно-поступательным или вращательным движением.

Использование подшипников скольжения [3, 4] из древесины, полимеров и ингредиентов, полученных из местных минеральных и вторичных сырьевых ресурсов, позволяет повысить вдвое их ресурс работы, снизить уровень шума, сократить расход смазочных материалов и трудоёмкость смазочных работ.

Такая технология позволяет использовать различные полимеры, смолы и наполнители для получения изделий с заданными объемными и поверхностными свойствами.

Отличительной особенностью этого антифрикционного древесно-полимерного композиционного материала является эффект само смазывания, что очень важно для машин текстильной, хлопкоочистительной, пищевой отраслей промышленности и сельского хозяйства.

Конструирование наборных подшипников скольжения. Анализ условий эксплуатации комплекса оборудования хлопкоочистительных заводов показывает, что изменение конструкции рабочих органов, в какой-то степени, позволяет повысить работоспособность и эффективность работы машин, но не устраняет, присущие им недостатки.

Исследование процессов, ответственных за изменение структуры и химической природы древесины, происходящих под воздействием механических напряжений в присутствии активных антифрикционных модификаторов, составляет новую основу поиска методов ее модифицирования. На основании этих исследований были разработаны композиционные материалы на основе древесины, полимеров и наполнителей для подшипников скольжения взамен подшипников качения.

Подшипники скольжения имеют ряд преимуществ, заключающиеся в их компактности, возможности нести большие нагрузки и в возможности восстановления и регулирования в условиях хлопкоочистительных заводов.

Основным недостаткам известных комбинированных подшипников скольжения из древесно-полимерных композиционных материалов относится наличие обоймы со сложными фиксирующими устройствами и не технологичность изготовления крупногабаритных древесно-полимерных втулочных вкладышей, которая устраняется выполнением в виде отдельных наборных древесных элементов, собранных в «бочку» или «ласточкин хвост» в металлической обойме.

Конструкции наборных подшипников скольжения наиболее отработаны и унифицированы, так как они давно и широко применяются в судостроении в качестве дейвудных подшипников [5].

Обычно наборные элементы изготавливают из продольных брусков трапецеидальной формы [6]. Если бруски набираются в «бочку». То сопряженная поверхность с металлической обоймой имеет кривизну того радиуса, что и внутренняя поверхность обоймы. При наборе элементов в «ласточкин хвост» их размеры и форма должны соответствовать размерам и форме установочного паза металлической обоймы. Выступающая часть древесного элемента задается в 1,5 раза больше допустимой величины износа подшипника, а глубина паза на 1,0-2,0 мм больше выступающей части наборного элемента. Толщина заготовок для изготовления наборных древесных элементов должна быть в 2,5- 3,0 раза больше расчетной. На рабочей поверхности наборных элементов по кромке снимаются фаски под углом 45° . Хотя снимаемая фаска значительно уменьшает опорную поверхность подшипника, однако она обеспечивает устойчивость материала при расточке посадочного размера, а образовавшиеся между ними пазы служат для аккумуляции смазки и вывода абразива из зоны трения. Для уменьшения зазора при усушке между элементами, набираемыми в «бочку», и для предотвращения их выпадения на внутренней поверхности неразъемной металлической обоймы устанавливают две или три упорные планки, а в разъемной – четыре или пять. Элементы должны плотно прилегать друг к другу, к поверхности обоймы и упорным планкам. Бруски в наборе укладываются так, чтобы возникающая сила при разбухании заклинивала их в металлической обойме.

Наборные подшипники скольжения из древесно-полимерных композиционных материалов растачиваются на токарных или расточных станках. При хранении внутренняя полость их заполняется влажными опилками или покрывается слоем консистентной смазки.

Подшипники, конструктивно выполненные в виде наборных элементов из древесно-полимерных композиционных материалов (рис.2), работающие на торец и собранные в «бочку» или «ласточкин хвост» в металлической обойме обладают лучшей формоустойчивостью и работоспособностью [2]. Однако формование их требует высокой точности геометрических размеров древесно-полимерных композиционных материалов и

металлической обоймы, специальных устройств для фиксации наборных элементов и их консервации, что резко удорожает изготовление и ремонт узлов трения.

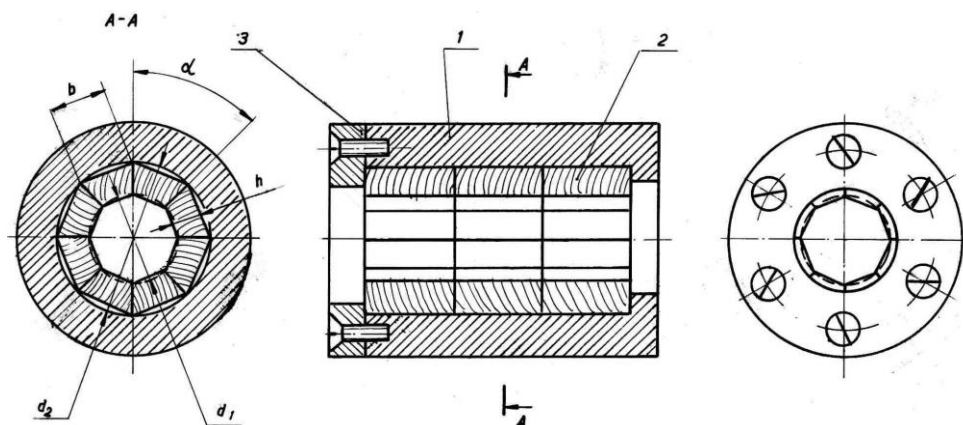
Втулочные вкладыши 2 с поперечным расположением волокон по отношению к их оси запрессовываются в металлическую обойму 1, с торцов на которой устанавливаются металлические ограничители 3, сохраняющие древесину в зажатом состоянии при разбухании в осевом направлении.

Формование древесных вкладышей для таких подшипников может быть осуществлено на устройстве [3].

Для стабилизации размеров древесно-полимерных композиционных материалов в комбинированных подшипниках скольжения эффективно совместное применение технологических и конструктивных мероприятий, таких как циклическое увлажнение и усушка древесного вкладыша в замкнутом контуре. Для этого вкладыш с радиальным расположением волокон запрессовывают в металлическую обойму, подвергают водной и термической обработке. При увлажнении древесина почти не разбухает, но при повышении температуры усыхает.

Во вторую металлическую обойму, снова увлажняется и сушится в масле. Имеющаяся при этом усушка устраняется запрессовкой вкладыша в третью рабочую металлическую обойму с меньшим диаметром. Такие комбинированные подшипники при работе в условиях переменной влажности обладают высокой формоустойчивостью.

Учитывая, что древесно-полимерные композиционные материалы [4] наиболее эффективно использовать в тихоходных материалоемких узлах трения, работающих в режиме самосмазки, для промышленного внедрения конструкции подшипников скольжения должны быть просты в изготовлении, обеспечивать технологичность монтажа и демонтажа без изменений основных существующих размеров узлов и способствовать повышению надежности и долговечности работы машин и механизмов. При этом разработка и внедрение новых конструкций подшипников скольжения из древесно-полимерных композиционных материалов должны производиться с учетом решения задачи рационального использования вторичных сырьевых ресурсов деревообрабатывающей промышленности.



1-металлическая обойма; 2-втулочные вкладыши из АДПКМ; 3-металлические ограничители

Рисунок 2. – Конструкция комбинированных подшипников скольжения из антифрикционных древесно-полимерных композиционных материалов (АДПКМ) для узлов трения рабочих органов хлопковых машин и механизмов

Разработка конструкций подшипников скольжения из композиционных материалов. Успешная разработка конструкций подшипников скольжения из композиционных материалов на основе древесины и модифицированных полимеров возможна только при всестороннем анализе работы узла трения хлопкоочистительных машин и механизмов, хорошем знании свойств выбранных и разработанных материалов, особенностей их поведения при различных эксплуатационных условиях, технологических возможностей конструкции.

Узлы трения машин и механизмов хлопкоочистительных заводов работают в сложных условиях в присутствии большого количества абразивных частиц, сильной запыленности воздуха, под действием атмосферных воздействий, статических и динамических нагрузок,

вибраций и т.д.

Одним из основных критериев при конструировании подшипников скольжения на основе древесины и полимеров является фактор (PV) - произведение значений Р-удельного давления на V-скорость скольжения [3]. Работоспособность подшипника скольжения, кроме того, определяется температурой трения, возникающей в зоне контакта. Температура трения влияет на величину и устойчивость коэффициента трения и на износостойкость пары. Оптимальный температурный режим подшипников скольжения ограничивается значениями теплостойкости древесины и полимерных материалов. При повышенной температуре происходит потеря физико-механических свойств материала.

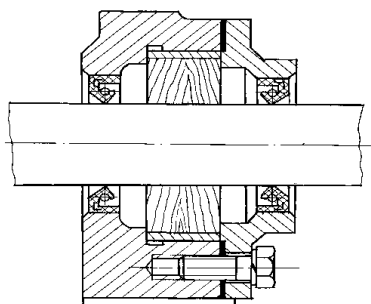


Рисунок 3. – Подшипниковый узел подшипника скольжения с антифрикционным древесно-полимерным композиционным материалом на основе тополя

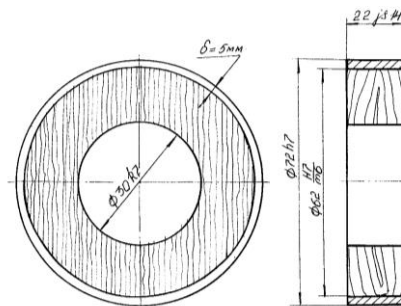


Рисунок 4. – Конструкция подшипника скольжения из антифрикционного древесно-полимерного композиционного материала на основе тала

На основе антифрикционного древесно-полимерного композиционного материала, установленная на питателях хлопкоочистительных машин (питатель джина, очистители мелкого и крупного сора). Втулка изготавливается из круглой древесины с припуском на механическую обработку и на осадку при прессовании. Далее она пропитывается ингредиентами по разработанной ранее технологии. После пропитки втулка помещается в металлическую обечайку и в таком состоянии устанавливается в пресс-форму для последующего её прессования. В этом подшипнике нагрузка также воспринимается торцами волокон в нижней части отверстия втулки.

Перспективность применения антифрикционных древесно-полимерных композиционных материалов. В последние годы одним из перспективных направлений повышения эффективности работы узлов трения машин является применение в них новых антифрикционных древесно-полимерных композиционных материалов на основе тополя или тала. Применение антифрикционных древесно-полимерных композиционных материалов в подшипниках скольжения целесообразно благодаря их высокой износостойкости, низкой стоимости, технологичности изготовления и малого количества отходов производства, хорошей демпфирующей способности, низкого коэффициента трения, возможности работы с ограниченной смазкой или без нее.

Опыт отечественного и зарубежного машиностроения показывает эффективность применения композиционных полимерных материалов в определенных условиях работы узлов трения машин и механизмов. Современная практика применения полимерных и композиционных материалов в антифрикционных узлах трения требует проведения поисковых работ по разработке оптимальных конструкций типовых подшипников скольжения из композиционных материалов с учетом специфических условий их работы.

Диапазоны применимости таких материалов обусловлены спецификой процессов, происходящих при трении, и физико-механическими свойствами композиционных полимерных материалов.

Поэтому встала задача создания новой технологичной конструкции подшипника скольжения. В основе этой конструкции лежит изготовление цельной втулки из модифицированной древесины (рис.4) на основе тала.

Втулка изготавливается из круглой древесины с припуском на механическую обработку и на осадку при прессовании. Далее она пропитывается ингредиентами по разработанной

ранее технологии. После пропитки втулка помещается в металлическую обечайку и в таком состоянии устанавливается в пресс-форму для последующего её прессования. После прессования подшипник имеет вид, представленный на рис. 4.

В этом подшипнике нагрузка также воспринимается торцами волокон в нижней части отверстия втулки. Для обеспечения этого положения в подшипниковом узле предусматривают специальные метки на обечайке.

Ниже приведены фотографии с изображением подшипниковых узлов подшипника скольжения из АДПКМ -1 на основе тополя (а) и ПС – подшипника качения (б) (рис.5), отдельных подшипников (рис.6).



Рисунок 5. – Подшипниковые узлы подшипника скольжения из АДПКМ -1 на основе тополя (а) и подшипника качения (б)

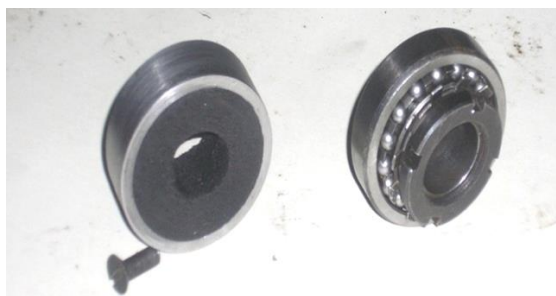


Рисунок 6. – Подшипник скольжения из АДПКМ-2 на основе тала (а) и подшипника качения (б)

В таблице 2 приведены типоразмеры разработанных подшипников скольжения из древесно-полимерных композиционных материалов для узлов трения комплекса оборудования хлопкоочистительных заводов.

Таблица 2.

Типовые размеры разработанных подшипников скольжения из антифрикционных древесно-полимерных композиционных материалов для узлов трения комплекса оборудования хлопкоочистительных заводов

Тип машин и механизмов	Размеры подшипников скольжения из древесно-полимерных композиционных материалов, мм			
	Внутренний диаметр подшипника		Внешний диаметр, D	Ширина, В
	подшипник по валу, d_v	подшипник по втулке, закрепленной на валу, d_{dn}		
Винтовые конвейеры Ø 320	55-60	-	90	40
Конденсор волокна 5KB	-	56	100	28
Сепаратор хлопка СС-15А ал)	-	56	100	28
Питающие валики и сорные шнеки к очистителям, джинам и линтерам	30	36	72	22

Обычно наборные элементы изготавливают из продольных брусков трапецеидальной формы. Если бруски набираются в «бочку», то сопряженная поверхность с металлической обоймой имеет кривизну того радиуса, что и внутренняя поверхность обоймы. При наборе элементов в «ласточкин хвост» их размеры и форма должны соответствовать размерам и форме установочного паза металлической обоймы. Выступающая часть древесного элемента

задается в 1,5 раза больше допустимой величины износа подшипника, а глубина паза на 1,0-2,0 мм больше выступающей части наборного элемента. Толщина заготовок для изготовления наборных древесных элементов должна быть в 2,5- 3,0 раза больше расчетной. На рабочей поверхности наборных элементов по кромке снимаются фаски под углом 45°. Хотя снимаемая фаска значительно уменьшает опорную поверхность подшипника, однако она обеспечивает устойчивость материала при расточке посадочного размера, а образовавшиеся между ними пазы служат для аккумуляирования смазки и вывода абразива из зоны трения. Для уменьшения зазора при усушке между элементами, набираемыми в «бочку», и для предотвращения их выпадания на внутренней поверхности неразъемной металлической обоймы устанавливают две или три упорные планки, а в разъемной – четыре или пять. Элементы должны плотно прилегать друг к другу, к поверхности обоймы и упорным планкам. Бруски в наборе укладываются так, чтобы возникающая сила при разбухании заклинивала их в металлической обойме.

Наборные подшипники скольжения из антифрикционных древесно-полимерных композиционных материалов растачиваются на токарных или расточных станках. При хранении внутренняя полость их заполняется влажными опилками или покрывается слоем консистентной смазки.

3. Выводы.

Подшипники конструктивно выполненные в виде наборных элементов из антифрикционных древесно-полимерных композиционных материалов, работающие на торец и собранные в «бочку» или «ласточкин хвост» в металлической обойме обладают лучшей формой устойчивости и работоспособности. Формование их требует высокой точности геометрических размеров древесно-полимерных композиционных материалов и металлической обоймы, специальных устройств для фиксации наборных элементов и их консервации, что резко удорожает изготовление и ремонт узлов трения.

Учитывая, что антифрикционные древесно-полимерные композиционные материалы наиболее эффективно использовать в тихоходных материалоемких узлах трения, работающих в режиме самосмазки, для промышленного внедрения конструкции подшипников скольжения должны быть просты в изготовлении, обеспечивать технологичность монтажа и демонтажа без изменений основных существующих размеров узлов и способствовать повышению надежности и долговечности работы машин и механизмов. При этом разработка и внедрение новых конструкций подшипников скольжения из древесно-полимерных композиционных материалов должны производиться с учетом решения задачи рационального использования вторичных сырьевых ресурсов деревообрабатывающей промышленности.

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УСЛОВИЯ ДИНАМИЧЕСКОЙ УРАВНОВЕШЕННОСТИ И ПРОЧНОСТИ КАНАТНОГО РАБОЧЕГО ОРГАНА КАМНЕРЕЗНОЙ МАШИНЫ

Мамасаидов М.Т.¹, Исманов М.М.¹

¹Кыргызско-Узбекский Международный университет, Ош, Кыргызстан

E-mail: cru_2011@rambler.ru; ismanov1970@mail.ru.

Аннотация. Данная работа посвящена определению условий динамической уравновешенности и прочности канатного рабочего органа камнерезной машины. Используются методы преобразования математических формул, анализа полученных зависимостей. Обоснованы условия динамической уравновешенности канатного рабочего органа в процессе резания камня, при $\varepsilon_{III} = 0$, $a_K = 0$ и определены зависимости соответствующих им режимных и конструктивных параметров. Выявлено, что для уменьшения силы натяжения режущего каната P_H , возникает необходимость увеличения величины скорости движения режущего каната U_K , при постоянной мощности привода канатного рабочего органа N_K и уменьшение суммарной силы сопротивления P_C в процессе резания камня. В свою очередь уменьшение диаметра каната d_T , также требует снижения P_C и увеличения параметра U_K . Сформулированы условия динамической прочности канатного рабочего органа и на их основе выработаны рекомендации к выбору их минимально допустимых геометрических параметров.

Ключевые слова: природный камень; процесс резания; динамическая уравновешенность; динамическая прочность; канатный рабочий орган; камнерезная машина.

1. Введение.

Высокая производительность и надежность работы камнерезных машин обеспечиваются при рациональных конструктивных параметрах и режимах резания природного камня.

Как показывает практика [1–4], для достижения высокой производительности, камнерезные машины эксплуатируют в ускоренных режимах работы, которые в большинстве случаев назначаются интуитивно, без глубокого научного обоснования. При этом процесс резания природного камня сопровождается вынужденными колебаниями рабочих органов этих машин, значительно влияющими на надежность и производительность их работы. Эти колебания возникают вследствие физических особенностей процесса резания камня и погрешностей изготовления основных узлов и деталей этих машин.

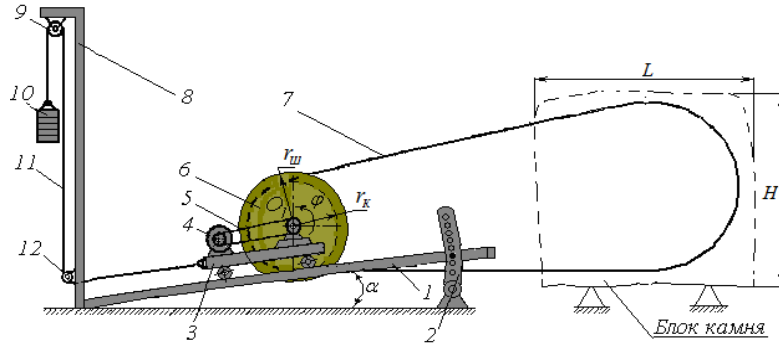
Основной причиной возникновения вынужденных колебаний, обусловленных погрешностями изготовления основных узлов и деталей камнерезных машин, являются: дефекты в передачах, неточности в изготовлении вращающихся деталей привода и рабочего органа; нерациональные конструктивные параметры основных деталей и узлов; недостаточная виброустойчивость, динамическая неуравновешенность и прочность нагруженных рабочих органов камнерезных машин. В связи с этим, возникает необходимость определения условий динамической уравновешенности и прочности рабочих органов камнерезных машин в процессе резания природного камня.

2. Результаты и обсуждения.

Рассмотрим камнерезную машину с канатными рабочими органами (рисунок 1). Проведенный анализ зависимостей режимных и конструктивных параметров канатных рабочих органов камнерезных машин показывают [5–9], что изменение величины углового ускорения рабочего шкива ε_{III} в значительных диапазонах вызывают дополнительных инерционных и динамических нагрузок на элементы канатного рабочего органа и другие узлы машины. Что приводит к появлению шума и вибрации камнерезной машины, снижению ее эксплуатационных показателей.

Следует отметить, что рациональный режим работы камнерезной машины с канатным рабочим органом (рисунок 1) обеспечивается при равномерном вращении рабочего шкива ($\varepsilon_{III} = 0$) и движении рабочей тележки камнерезной машины ($a_K = 0$). При этом шум и вибрация

камнерезной машины в процессе резания камня не значительное, а величины инерционных и динамических нагрузок на элементы канатного рабочего органа будут минимальными.



1 – шасси; 2 – стойки для регулирования угла наклона шасси; 3 – рабочая тележка;
4 – электродвигатель; 5 – клиноременная передача; 6 – рабочий шкив; 7 – алмазный канат; 8 – вертикальная рама; 9, 12 – шкивы; 10 – груз; 11 – трос.

Рисунок 1. – Конструктивная схема камнерезной машины с канатным рабочим органом

В результате исследования динамики канатного рабочего органа получены уравнения, характеризующие зависимости углового ускорения рабочего шкива ($\varepsilon_{ш}$) и линейного ускорения рабочей тележки (a_K) камнерезной машины от конструктивных и режимных параметров канатного рабочего органа [9]:

$$\varepsilon_{ш} = \frac{2[(M_{ш} - P_C \cdot r_K) \cdot m_M + (P_C - P_{II}) \cdot m_2 \cdot r_K \cdot \cos \alpha]}{m_1 \cdot r_{ш}^2 \cdot m_M + 2m_2 \cdot r_K^2 \cdot (m_1 + m_T + m_2 \sin^2 \alpha)}, \quad (1)$$

$$a_K = \frac{(P_{II} - P_C) \cdot (m_1 r_{ш}^2 \cdot m_M + 2m_2^2 r_K^2) + 2P_{II} \cdot m_2 r_K^2 \cdot (m_1 + m_3)}{m_M [m_1 r_{ш}^2 \cdot m_M + 2m_2 r_K^2 \cdot (m_1 + m_T + m_2 \sin^2 \alpha)]} + \frac{2m_2 r_K \cdot m_M \cdot (P_C r_K - M_{ш}) \cos \alpha}{m_M [m_1 r_{ш}^2 \cdot m_M + 2m_2 r_K^2 \cdot (m_1 + m_T + m_2 \sin^2 \alpha)]}. \quad (2)$$

Проведенный анализ зависимостей углового ускорения рабочего шкива ($\varepsilon_{ш}$) и линейного ускорения рабочей тележки (a_K) от режимных и конструктивных параметров канатного рабочего органа показывают [10], что изменение величин $\varepsilon_{ш}$ и a_K , соответственно в среднем составляют $0,01 \text{ рад/с}^2$ и $0,01 \text{ м/с}^2$. Откуда, принимая $\varepsilon_{ш} = 0$ и $a_K = 0$, из уравнений (1) и (2), после определенных преобразований определим необходимые условия динамической уравновешенности канатного рабочего органа камнерезной машины в следующем виде

$$\frac{M_{ш}}{r_K} = \frac{P_C (m_M - m_2 \cdot \cos \alpha) + P_{II} \cdot m_2 \cdot \cos \alpha}{m_M}, \quad (3)$$

$$\frac{2M_{ш} \cdot m_2 \cdot r_K \cos \alpha - P_{II} (m_1 \cdot r_{ш}^2 + 2m_2 \cdot r_K^2)}{m_M \cdot m_1 \cdot r_{ш}^2 + 2m_2 \cdot r_K^2 (m_2 - m_M \cos \alpha)} = \frac{P_C}{m_M}, \quad (4)$$

где m_M - масса камнерезной машины без направляющих рельсов, кг; m_2 и r_K - соответственно масса (кг) и радиус вращения (м) режущего каната; $M_{ш}$, m_1 и $r_{ш}$ - соответственно крутящий (вращающий) момент (Н·м), масса (кг) и радиус (м) рабочего шкива; m_T - масса рабочей тележки с приводом канатного рабочего органа, кг; α - угол наклона рельса рабочей тележки к горизонту, град.; P_C - суммарная сила сопротивления резанию, Н.

Суммарная сила сопротивления резанию (P_C) была выражена зависимостью вида [11]:

$$P_C = P_{II} (f + f_C) + \frac{M_{ш} \cdot \omega_{ш}}{v_K} \cdot \mu + \frac{\pi \cdot E \cdot d_T^2}{4l} \cdot \Delta l. \quad (5)$$

Подставляя значение P_C из (5) в (3), определим рациональную величину скорости резания (движения режущего каната):

$$v_K = \frac{M_{III} \cdot \omega_{III} \cdot \mu}{\left[\frac{M_{III} \cdot m_M - P_{II} \cdot m_2 \cdot r_K \cdot \cos \alpha}{r_K \cdot (m_M - m_2 \cdot \cos \alpha)} - P_{II}(f + f_C) - \frac{\pi \cdot E \cdot d_T^2}{4l} \cdot \Delta l \right]}. \quad (6)$$

Аналогичным образом, из (6), учитывая $v_K = r_K \cdot \omega_{III}$, определим рациональное значение радиуса вращения режущего каната (рабочего шкива):

$$r_K = \frac{M_{III} \cdot \mu}{\left[\frac{M_{III} \cdot m_M - P_{II} \cdot m_2 \cdot r_K \cdot \cos \alpha}{r_K \cdot (m_M - m_2 \cdot \cos \alpha)} - P_{II}(f + f_C) - \frac{\pi \cdot E \cdot d_T^2}{4l} \cdot \Delta l \right]}. \quad (7)$$

Определено, что для обеспечения динамической уравновешенности и эффективной работы камнерезной машины должны быть выбраны рациональные значения режимных и конструктивных параметров канатного рабочего органа в зависимости от угла наклона рабочей тележки α и коэффициента сопротивления камня относительному сдвигу (сцепление) или срезу μ .

Работоспособность основных деталей (ведущий шкив, трос, алмазная втулка и др.) канатного рабочего органа характеризуется критериями, важнейшими из которых являются их прочность, износостойкость и коррозионностойкость.

Следует отметить, что в процессе резания камня работоспособность канатного рабочего органа, прежде всего, определяется коэффициентом запаса прочности режущего алмазного каната [12, 13]:

$$k \leq \frac{P_{PP}}{P_H}, \quad (8)$$

где P_{PP} – предельно-допустимая сила, приводящаяся к разрыву режущего каната, Н; P_H – наибольшее усилие натяжения, действующая на ведущей ветви режущего каната, Н. Практикой обоснована следующая безаварийная величина коэффициента запаса прочности режущего каната $k > 4$ [12].

Предельно-допустимая сила P_{PP} , приводящаяся к разрыву режущего каната (ГОСТ 2688 – 80) зависит от его диаметра d_T . Например, при $d_T = 4,8$ мм, $P_{PP} = 16050$ Н, а когда $d_T = 5,1$ мм, $P_{PP} = 18150$ Н.

Наибольшее усилие натяжения режущего каната P_H действует на ведущей его ветке (рисунок 1) и всегда ограничивается установленной мощностью привода КРО N_K , через величину предельно возможной силы резания P_P [12, 13]:

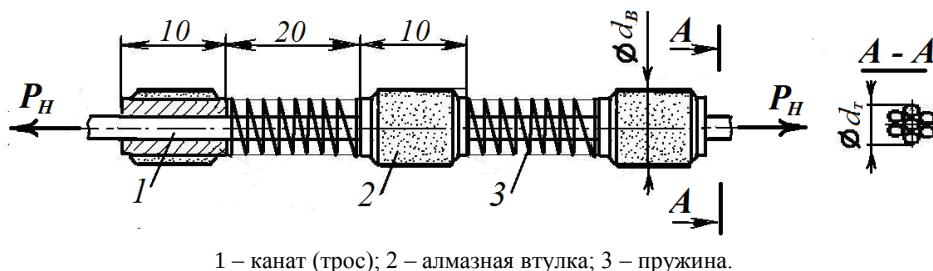
$$P_H = \frac{P_P}{k_\phi} \leq \frac{1}{k_\phi} \cdot \left(\frac{N_K \cdot \eta}{v_K} \right), \quad (9)$$

где $k_\phi = 0,61$ – коэффициент полезного использования рабочего контура режущего каната; η – к.п.д. привода КРО; v_K – скорость движения режущего каната, м/с.

Необходимо отметить, что работоспособность режущего каната определяется главным критерием – его динамической прочностью на растяжение (разрыв). Режущий канат при работе на растяжение считается прочным (рисунок 2), если максимальные фактические напряжения в поперечном сечении каната (троса) меньше предельных [14]:

$$\sigma_{max} = \frac{P_H}{\pi \cdot \left(\frac{d_T^2}{4} \right) \cdot k_3} \leq [\sigma]_o, \quad (10)$$

где σ_{max} - фактическое максимальное напряжение в поперечном сечении каната в процессе работы; $[\sigma]_o = 1800 \text{ Н/мм}^2$ - предельно допустимое напряжение в поперечном сечении каната; $k_3 = 0,6$ - поправочный коэффициент для определения фактической площади поперечного сечения каната (рисунок 2).



1 – канат (трос); 2 – алмазная втулка; 3 – пружина.

Рисунок 2. – Режущий алмазный канат

Принимая $P_H = \frac{1}{k_\phi} \cdot \left(\frac{N_K \cdot \eta}{v_K} \right)$ из условия (9) и подставляя в (10), определим зависимости диаметра каната в виде:

$$d_T \geq \sqrt{\left(\frac{4N_K \cdot \eta}{v_K} \right) \cdot \frac{1}{\pi \cdot k_\phi \cdot k_3 \cdot [\sigma]_o}}. \quad (11)$$

Из выражений (9) – (11) следует, что для уменьшения силы натяжения режущего каната P_H , возникает необходимость увеличения величины скорости движения режущего каната v_K , при постоянной мощности привода канатного рабочего органа N_K и уменьшение суммарной силы сопротивления P_C в процессе резания камня. В свою очередь уменьшение диаметра каната d_T , также требует снижения P_C и увеличения параметра v_K .

Таким образом, соблюдая условия динамической прочности (11) можно обоснованно приступить к разработке методики выбора рациональных параметров канатного рабочего органа камнерезных машин.

3. Выводы.

1. Определено, что для обеспечения динамической уравновешенности и эффективной работы камнерезной машины должны быть выбраны рациональные значения режимных и конструктивных параметров канатного рабочего органа в зависимости от угла наклона рабочей тележки α и коэффициента сопротивляемости камня относительному сдвигу (сцепление) или срезу μ ;

2. Обоснованы условия динамической уравновешенности канатного рабочего органа в процессе резания камня, при $\varepsilon_{III} = 0$, $a_K = 0$ и определены зависимости соответствующих им режимных и конструктивных параметров;

3. Выявлено, что для уменьшения силы натяжения режущего каната P_H , возникает необходимость увеличения величины скорости движения режущего каната v_K , при постоянной мощности привода канатного рабочего органа N_K и уменьшение суммарной силы сопротивления P_C в процессе резания камня. В свою очередь уменьшение диаметра каната d_T , также требует снижения P_C и увеличения параметра v_K .

4. Сформулированы условия динамической прочности канатного рабочего органа и на их основе выработаны рекомендации к выбору их минимально допустимых геометрических параметров.

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ДИНАМИЧЕСКИЕ МОДЕЛИ И ЗАКОНОМЕРНОСТИ НАГРУЖЕНИЯ СИЛОВЫХ ЭЛЕМЕНТОВ ГИДРАВЛИЧЕСКОГО КАМНЕКОЛЬНОГО ПРЕССА

Мамасаидов М.Т.¹, Исаев И.Э.¹

¹Кыргызско-Узбекский Международный университет, Ош, Кыргызстан

Email: cpu_2011@rambler.ru

Аннотация. В данной работе исследованы верхняя и нижняя траверсы станины гидравлического камнекольного пресса с целью определения необходимых условий его прочности для определения коэффициентов жёсткостей траверс. Используются методы составления расчетных схем и преобразования аналитических выражений. Проведены исследования динамики несущей станины пресса, тяжёлых шпилек (шпилька-гайка) и рабочего органа. Полученные результаты позволяют разработать методику выбора рациональных параметров верхней и нижней траверсы станины гидравлического камнекольного пресса и совершенствовать их конструкцию.

Ключевые слова: верхняя и нижняя траверсы станины, жёсткость, деформация, прогиб, гидравлический камнекольный пресс.

1. Введение.

Природный камень является одним из самых древних строительных материалов, которым пользовался ещё первобытный человек. Он обладает высокими декоративными свойствами, долговечностью и может сохранять свои физико-механические свойства под действием различных факторов. В связи с этим, изделия из природного камня с успехом применяются при облицовке зданий и сооружений, отделке набережных и фонтанов, для покрытия площадей и разграничения проезжей части улиц, изготовления ограждений в садово-парковой архитектуре [1,2,3].

Среди строительных изделий из природного камня особое место занимает так называемые «колотые изделия», получаемые путем колки камня вручную или механизированным способом с помощью камнекольных станков. Эти изделия нашли в градостроительстве, особенно в европейских странах.

Как показал анализ технологического процесса изготовления колотых изделий из камня ручным способом, на изготовления одной плиты затрачивается до 10 часов рабочего времени. Этим была обусловлена необходимость создания технологии механизированного процесса обработки камня, позволяющего снизить стоимость продукции и повысить производительность [1].

В настоящее время, интерес к обработке камня расколом возрос не только вследствие ее малой энергоемкости, но и благодаря меньшей стоимости получаемой продукции. Для производства колотых изделий в мировой практике существует ряд технических средств обработки камня расколом, среди которых приоритет имеют камнекольные станки. Их производят во многих развитых странах, в т.ч. такие ведущие фирмы как "Kameklu" (Франция), "Porsfeld" (ФРГ), "Park tul kompani" (США), "Stainex" (Италия), "Wan Wurden" (Голландия) и др.

Работы по созданию камнекольных станков проводились и ныне развиваются в нашей стране. В 1985-1990 годах в НИЦ "Импульс" АН Кирг. ССР (ныне Институт машиноведения НАН КР) были созданы камнекольные станки типа ПКА-400, ПКА-800, ПКА-3000, которые выпускались в г.Бишкек Заводом имени Ленина и с эффективностью применяются в различных камнеобрабатывающих предприятиях республики и за её пределами. Эксплуатация этих прессов на практике позволило выявить ряд их преимуществ, но также и некоторые недостатки: возникновение значительных динамических нагрузок; использование дорогостоящего гидравлического оборудования; большая металлоёмкость конструкции и сложность технологических операций при изготовлении оборудования.

Однако, производство колотых изделий очень эффективно, т.к. энергоемкость обработки камня расколом более чем в 7 раз ниже, чем технологии распиловки [1,2,7]. В ведущих странах мира, особенно в Европе, создание специализированных станков, производство и применение колотых строительных изделий камня развиты очень высоко [1].

В Кыргызско-Узбекском международном университете под руководством академика М.Т. Мамасаидова ведутся работы по созданию новой конструкции гидравлического камнекольного пресса типа ГКП-1. Преимуществом ГКП-1 перед его аналогом - гидравлическим камнекольным прессом типа ПКА является то, что приводом является простая и очень надёжная механическая передача. В ней меньше узлов и деталей, требующих специального изготовления, как у аналога. Все детали могут быть изготовлены в обычных механических цехах. Как известно из практики, гидравлический домкраты обладают большой надёжностью и КПД, долговечны, просты и дешевы в эксплуатации. Малый гидравлический пресс может быть использован непосредственно на рабочих местах, тогда как прессы типа ПКА работают только в стационарных условиях специализированного камнекольного цеха [1,2,3].

Таким образом, описанные достоинства гидравлического камнекольного пресса ГКП-1 обеспечивает ему упрощение конструкции и повышения эксплуатационных свойств по сравнению с большими прессами.

2. Результаты и обсуждения.

Данная работа посвящен результатам исследования динамики несущей станины пресса, стяжных шпилек(шпилька-гайка) и рабочего органа. Согласно этой методике расчет сводится к расчету стяжных шпилек и проверке напряжений затяжки в стойках. Кроме этого необходимо проверочный расчет верхней и нижней балок, а также стойки в опасных сечениях. При нагружении природного камня в процессе раскола, на силовые элементы (станину, стяжной шпильку, рабочие органы) камнекольного пресса действуют значительные реактивные силы, что приводят к накоплению в них потенциальной энергии деформации. В момент раскола t_p (рис.1) [1], когда реакция внезапно исчезает, накопленная энергия преобразуется в кинетическую энергию [1,2], вызывая колебания и возникновение динамических напряжений в элементах пресса.

Такие нагрузки приводят к уменьшению долговечности силовых элементов и ухудшению работоспособности ГКП-1 в целом.

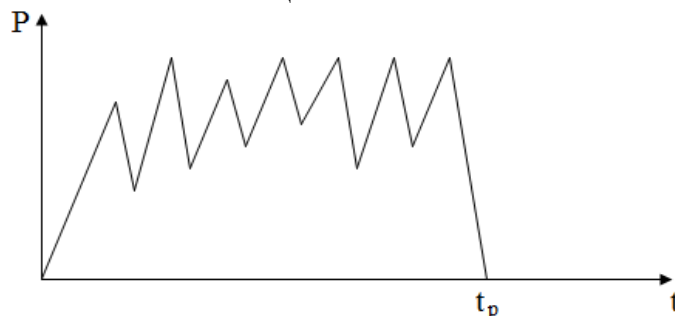


Рисунок 1. – Характер изменения нагрузки в процессе раскола образца камня [1]

Составим динамическую модель этих силовых элементов гидравлического камнекольного пресса (рис.1) на основе следующих допущений:

- масса рабочего стола сравнительно мала по сравнению с массами остальных силовых элементов пресса и поэтому ее можно не учитывать;
- жесткость пружины рабочего стола ничтожно мала по сравнению с жесткостью других силовых элементов пресса, а следовательно ее можно пренебречь;
- силы тяжести силовых элементов пресса можно не учитывать из-за малости их значений по сравнению с развиваемым усилием раскола (для ГКП-1 максимальное значение P принято равным 400кН).

В динамической модели силовых элементов ГКП (см. рис.2, а) учитываются лишь упругие и инерционные характеристики рабочего органа, стяжной шпильки и станины пресса. Здесь m_2 - масса верхней части станины (траверсы), $m_{ср}$ – средней части станины, учитывающие массы ее боковые стенок, $m_{ро}$ – масса рабочего органа, m_3 – масса нижней

части станины и нижнего рабочего органа (нижней траверсы), C_3 – жёсткость станины прессы, $C_{вш}$ и $C_{нш}$ – жесткости верхней и нижней частей затяжной шпильки.

На основе научно-обоснованных положений [2] можно построить упрощённую модель силовых элементов ГКП-1 (рис. 1.2,б), в которой: C_1 , C_2 – жёсткости верхней и нижней части затяжной шпильки и станины; m_2 – масса верхнего рабочего органа и средней части станины; m_1 , m_3 – массы нижнего рабочего органа и нижней части станины (траверсы).

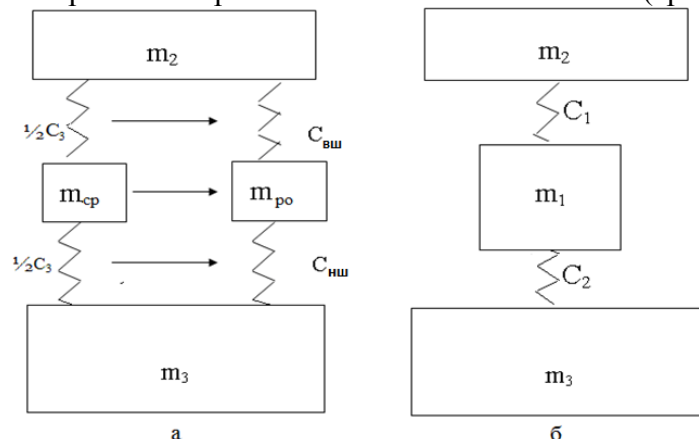


Рисунок 2. – Динамическая модель силовых элементов гидравлического камнекольного прессы ГКП-1: а- общая модель, б- упрощённая модель

Таблица 1

Упругие и инерционные характеристики силовых элементов гидравлического камнекольного прессы ГКП-1

М а с с ы, кг			Коэффициенты жёсткости, Н/м	
m1	m2	m3	C1	C2
537	93	2050	0,22*1011	0,05*1011

Упрощённая динамическая модель ГКП-1 описывается системой дифференциальных уравнений:

$$\begin{cases} m_1 x_1'' + c_1 (x_1 - x_2) - c_2 (x_1 - x_3) = 0; \\ m_2 x_2'' - c_1 (x_1 - x_2) = 0; \\ m_3 x_3'' - c_2 (x_1 - x_3) = 0 \end{cases} \quad (1)$$

где x_1 , x_2 , x_3 – координаты центров тяжести рассматриваемых масс.

Предполагая, что нижняя масса жёстко закреплена на фундаменте можно принять в третьем уравнении системы (1) $m_3 \rightarrow \infty$. Тогда решение получится без учёта последнего дифференциального уравнения системы, составленной относительно двух массовой модели. Однако для наиболее полного анализа колебания силовых элементов прессы остановимся на решении системы составленной для трёх массовой модели.

Для получения решения системы дифференциальных уравнений (1) введём новые переменные:

$$\begin{aligned} x_1 - x_2 &= y_1; \\ x_1 - x_3 &= y_2 \end{aligned} \quad (2)$$

Решая совместно систем уравнений (1) и (2) методом подстановки, получим дифференциальное уравнение четвёртого порядка с одной перемен-ной в виде

$$y_1^{IV} + y_1'' \left(\frac{c_1}{M_1} + \frac{c_2}{M_2} \right) + y_1 \left(\frac{c_1 c_2}{M} \right) = 0, \quad (3)$$

где

$$M_1 = \frac{m_1 m_2}{m_1 + m_2};$$

$$M_2 = \frac{m_1 m_3}{m_1 - m_3}; \quad M = \frac{m_1 m_2 m_3}{m_1 + m_2 - m_3}.$$

Характеристическое уравнение запишется в виде

$$r^4 + r^2 2h + k^2 = 0, \quad (4)$$

$$h = \frac{1}{2} \left(\frac{c_1}{M_1} - \frac{c_2}{M_2} \right), \quad k^2 = \frac{c_1 c_2}{M}$$

Корни характеристического уравнения (4) определяются как

$$r_{1,2} = \pm pi; \quad r_{3,4} = \pm qi,$$

$$p = \sqrt{h - \sqrt{h^2 - k^2}}; \quad q = \sqrt{h + \sqrt{h^2 - k^2}},$$

которые являются частотами колебаний системы.

Общее решение дифференциального уравнения (3) запишется в виде

$$y_1 = A \sin pt + B \cos pt + C \sin qt + D \cos qt \quad (5)$$

Для определения постоянных интегрирования уравнения (5) воспользуемся начальными условиями:

$$\begin{cases} y_1(0) = -\frac{P}{C_1}; y_1'(0) = 0; \\ y_2(0) = \frac{P}{C_2}; y_2'(0) = 0. \end{cases} \quad (6)$$

Откуда получим $A = 0, C = 0$,

$$D = \frac{P}{p^2 - q^2} \left(\frac{2m_2 + m_1}{m_1 m_2} - \frac{p^2}{C_1} \right);$$

$$B = -\frac{P}{C_1} - \frac{P}{p^2 - q^2} \left(\frac{2m_2 + m_1}{m_1 m_2} - \frac{p^2}{C_1} \right).$$

Подставляя полученные значения коэффициентов интегрирования в общее решение (5), имеем

$$y_1 = P \left[\left(\frac{1}{p^2 - q^2} \left(\frac{p^2}{C_1} - \frac{2m_2 + m_1}{m_1 m_2} \right) \frac{1}{C_1} \right) \cos pt + \right. \\ \left. + \frac{1}{p^2 - q^2} \left(\frac{2m_2 + m_1}{m_1 m_2} - \frac{p^2}{C_1} \right) \cos qt \right] \quad (7)$$

Из анализа уравнения (7) вытекает, что величина $P2/C1$ составляет 5% от значения $2m2+m1/m1m2$. Поэтому данное выражение y_1 можно упростить в виде

$$y_1 = P \left[- \left(\frac{2m_2 + m_1}{m_1 m_2 (p^2 - q^2)} + \frac{1}{C_1} \right) \cos pt + \frac{2m_2 + m_1}{m_1 m_2 (p^2 - q^2)} \cos qt \right] \quad (8)$$

Подставляя численные значения m_1 и c_1 из таблицы 1 в формулу (8), получим

$$y_1 = P(0,744 \cos pt - 5,31 \cos qt) \cdot 10^{-11} \quad (9)$$

Из этой расчётной зависимости видно, что первая гармоника совершает колебания с частотой $p=4000$ 1/с и меньшей, амплитудой чем вторая гармоника с частотой $q=17000$ 1/с и с амплитудой в 7,5 раз больше. Такую картину периодических колебаний двух гармоник можно наблюдать на диаграмме изменения деформации верхней части шпилки (рис.3).

Используя выражения (1) и (2) величину деформации нижней части шпилки определим по формуле

$$y_2 = P[k_1(k_4 - k_3 p^2) \cos pt + k_2(k_3 q^2 - k_4) \cos qt] \quad (10)$$

В данном уравнении: k_1 и k_2 определяют амплитуды колебаний двух гармоник верхней части шпилки и определяются из выражения (8), равные

$$0,744 \cdot 10^{-11} \text{ и } 5,31 \cdot 10^{-11} \text{ (м/Н)}; k_3 = m_1/C_2; k_4 = C_1/C_2 + C_1 m_1/C_2 m_2.$$

Из диаграммы на рис.3 видно, что колебание станины пресса носит периодический характер с частотами p и q . Максимальная величина деформации не превышает значения соответствующей моменту раскола блока камня.

Расчёт внутренних усилий шпилки в процессе раскола проведём по формулам

$$P_{ш1} = y_1 \cdot C_1, P_{ш2} = y_2 \cdot C_2 \quad (11)$$

Используя выражения (9)-10) и учитывая, что одна из гармоник колебаний верхней части имеет меньше амплитуду, окончательно получим

$$P_{ш1} = P \frac{C_1(2m_2 + m_1)}{m_1 m_2(p^2 - q^2)} \cos q t; \quad (12)$$

$$P_{ш2} = P C_2[k_1(k_4 - k_3 p^2) \cos p t + k_2(k_3 q^2 - k_4) \cos q t]. \quad (13)$$

Из этих выражений видно, что величина внутренних усилий в верхней и нижней части винта зависят как от коэффициентов жёсткости, так и от соотношения масс рассматриваемой модели.

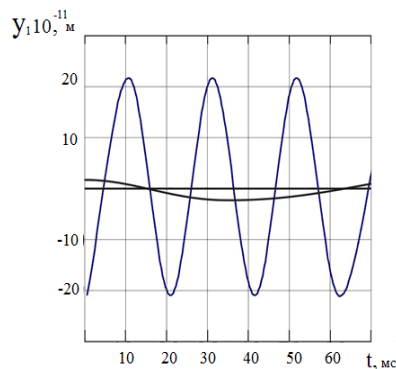


Рисунок 3. – Диаграмма изменения деформации верхней части затяжной шпилки в зависимости от времени

На рис.4 дана оценка амплитуды колебания затяжной шпилки от соотношения масс рабочего органа m_1 и верхней траверсы m_2 . Из этой диаграммы можно видеть, что с увеличением соотношения масс m_1/m_2 амплитуда колебаний уменьшается.

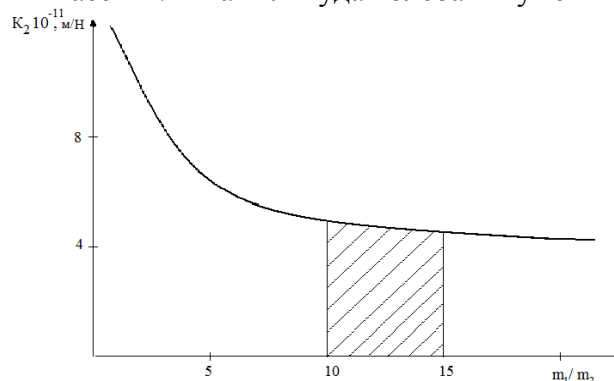


Рисунок 4. – Диаграмма изменения амплитуды колебания верхней части

Затяжной шпильки от соотношения масс рабочего органа (m_1) и верхней траверсы (m_2)

В диапазоне изменения m_1/m_2 от 1 до 10 наблюдается интенсивное уменьшение амплитуды колебаний K_2 от $15 \cdot 10^{-11}$ до $5 \cdot 10^{-11}$ (м/Н). Дальнейший рост коэффициента m_1/m_2 приводит к незначительному уменьшению амплитуды. Из полученных решений можно сделать вывод о том, что при конструировании гидравлического камнекольного пресса ГКП-1 необходимо подбирать параметры, удовлетворяющие соотношению масс m_1/m_2 в диапазоне от 10 до 15.

Таким образом, обобщая результаты исследования динамики гидравлического камнекольного пресса ГКП-1 позволили сформулировать научно-обоснованные рекомендации по выбору рациональных параметров силовых элементов (станины, затяжных шпилек и рабочего органа).

3. Выводы.

1. Обоснованы принципиальная схема и предпосылки к динамическому исследованию перспективного гидравлического камнекольного пресса на основе анализа их существующей теории и практики.

2. Установлено, что при расколе камня колебание элементов привода гидравлического камнекольного пресса происходит по гармоническим законам, а момент на валу электродвигателя ГКП-1 при подобранных характеристиках колеблется с частотой 250 с⁻¹ и амплитудой 60 (Нм).

3. Выявлено, что уменьшение коэффициента неравномерности хода двигателя требует увеличения моментов инерции элементов привода или увеличения их жесткости.

4. Разработана динамическая модель силовых элементов (станины, рабочего органа) гидравлического камнекольного пресса ГКП-1.

5. Установлено, что колебание силовых элементов гидравлического камнекольного пресса происходит по двум гармоническим законам, первая из которых имеет меньшую частоту и амплитуды в 7...8 раз.

6. Выявлено, что при конструировании гидравлического камнекольного пресса ГКП-1 необходимо подбирать соотношению масс рабочего органа (m_1) и верхней траверсы (m_2) ГКП-1 в диапазоне от 10 до 15.

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ИССЛЕДОВАНИЕ ПОТРЕБЛЯЕМОЙ МОЩНОСТИ ЭЛЕКТРОДВИГАТЕЛЯ СЕМЯОТВОДЯЩЕЙ ТРУБЫ

Мухаммадиев Д.М.¹, Эргашев И.О.², Абдусаломов М.С.¹, Жамолова Л.Ю.³

¹Институт механики и сейсмостойкости сооружений, Ташкент, Узбекистан

²Ферганский политехнический институт, Фергана, Узбекистан

³Ташкентский государственный аграрный университет, Ташкент, Узбекистан

E-mail: davlat_mm@mail.ru

Аннотация. В данной статье приведены материалы экспериментального исследования по изучению потребляемой мощности электродвигателя семяотводящей трубы. Результаты получены с использованием симметричного квази – D – оптимального плана с тремя факторами и построение уравнения регрессии этого процесса. Установлено, что минимальное значение потребляемой мощности электродвигателя семяотводящей трубы достигается до $y=1.278$ кВт при частоте вращения трубы $x_2 = -0.4$ (340 об/мин), плотности сырьцового валика в рабочей камере $x_1=1$ (286 кг/м³) и угле расположения отверстий на поверхности семяотводящей трубы $x_3=-1$ (50°).

Ключевые слова: волокноотделитель, перфорированная труба, электродвигатель, потребляемая мощность, уравнения регрессия.

1. Введение.

В технологических процессах хлопкоочистительной промышленности одним из основных показателей является потребляемая мощность электродвигателей [1]. Известно, что при создании и разработке новых конструкций машин важным фактором является изучения потребляемой мощности электродвигателей, которое определяет эффективность применения этих машин в производстве. Поэтому при проектировании семяотводящего устройства необходимо изыскать закономерность изменения потребляемой мощности электродвигателя, которая зависит от кинематических и конструктивных параметров семяотводящей трубы. Потому что этот показатель является основным при использовании семяотводящей трубы в процессе волокноотделения, а для определения этого показателя нами проведены экспериментальные исследования семяотводящей трубы со следующими факторами: плотность сырьцового валика, угол наклона отверстий трубы и частота вращения семяотводящей трубы [2-4].

2. Материалы, методы и объект исследования.

Для определения потребляемой мощности электродвигателя семяотводящей трубы изготовлено лабораторный стенд рабочей камеры с различными семяотводящими трубами (рисунок 1). Входные факторы исследования приняты: плотность сырьцового валика x_1 , частота вращения семяотводящей трубы x_2 и угол наклона отверстий трубы x_3 .



Рисунок 1. – Лабораторный стенд рабочей камеры с семяотводящей трубой



а – труба с углом перфораций 50°; б – труба с углом перфораций 56°; в – труба с углом перфораций 62°

Рисунок 2. – Трубы с различными перфорациями для семяотвода

Потребляемую мощность, необходимую для привода семяотводящей трубы и шнека, определяли по формуле

$$N = \frac{\sqrt{3} \cdot U \cdot I}{1000} \eta_{эл} , \quad (1)$$

где I – сила тока, потребляемая электродвигателем, А; U –напряжение сети, В; $\eta_{эл}$ – коэффициент полезного действия электродвигателя.

Для измерения потребляемой мощности электродвигателя использовали цифровой вольтметр/амперметр D85-2042A (AC80-500V) (рис. 3). Цифровой измеритель напряжения переменного тока D85-2042A может одновременно отображать переменное напряжение и силу тока. Счетчик имеет цветной ЖК-дисплей и нет необходимости в увеличении рабочей мощности и трансформаторе переменного тока. Техническая характеристика цифрового вольтметра/амперметра D85-2042A представлена в таблице 1 [5].



Рисунок 3. – Цифровой вольтметр/амперметр D85-2042A

Таблица 1.
Техническая характеристика цифрового вольтметра/амперметра D85-2042A

Наименование показателей	Значение
Диапазон измерения напряжение, В	80-500
Диапазон измерения тока, А	0-100
Точность измерения, %	0,5
Размер упаковки, мм	81×44×43
Вес с упаковкой, гр	80

Для изменения частоты вращения электродвигателя семяотводящей трубы использовали частотный преобразователь 5,5 кВт марки LEI5000-5R5-PB-3 (рис. 4), который позволяет экономить до 50% электроэнергии, повышает надежность привода за счет защиты электродвигателя от скачков напряжения и снижает износ. Техническая характеристика частотного преобразователя 5,5 кВт марки LEI5000-5R5-PB-3 представлена в таблице 2 [6].



Рисунок 4. – Частотный преобразователь 5,5 кВт с LEI5000-5R5-PB-3

Таблица 2.
Техническая характеристика частотного преобразователя 5,5 кВт с блоком управления LEI5000-5R5-PB-3

Наименование показателей	Значение
Мощность, кВт	5,5
Диапазон изменения напряжения, В	440±10%
Диапазон изменения частоты тока, В	0-60
Максимальная сила тока, А	13,0
Размер упаковки, мм	81*44*43
Вес с упаковкой, гр	80

При измерении плотности сырьевого валика использовали электронные весы “PortableElectronicScale” (рис. 5), фиксируя массу в килограммах, приходящуюся на 50х50 мм² площади, а далее разделив ее на ширину шахты (рис. 5). При этом нагрузка на весы не должна превышать 40 кг. Техническая характеристика электронных весов “PortableElectronicScale” приведена в таблице 3 [7].



Рисунок 5. – Электронные весы “PortableElectronicScale”

Таблица 3

Техническая характеристика электронных весов “PortableElectronicScale”

Наименование показателей	Значение
Диапазон измерений, кг	от 0.1 до 40
Диапазон измерений, Н	от 0.98 до 392.24
Цена деления, кг (Н)	0.01 (0.098)
Габаритные размеры, мм	90х25х175
Масса, кг	не более 0,145

В лабораторный стенд рабочей камеры семена поступали через шахту, а для создания сырьевого валика с различными плотностями сверху шахты устанавливали груз с различной массой (рисунок 1 и 5).

3. Постановка задачи и результаты.

Для выявления закономерности изменения потребляемой мощности электродвигателя семяотводящей трубой приняли симметричный квази –D– оптимальный план ($n=3$) с тремя

факторами (x_1, x_2, x_3) (таблица 4), а уровень надежности опытов 0.95 с ошибкой 3% и числом повторностей 3.

Таблица 4.

Рабочая матрица и результаты эксперимента по выявлению закономерности изменения потребляемой мощности электродвигателя семяотводящей трубы

№ опыта	Плотность сырьцового валика в рабочей камере x_1		Частота вращения семяотводящей трубы x_2		Угол расположения отверстий на поверхности семяотводящей трубы x_3		Результаты параллельных опытов по выявлению закономерности изменения потребляемой мощности электродвигателя семяотводящей трубы y , %			
	код	кг/м ³	код	об/мин	код	°	1	2	3	Среднее
1	0	278,5	1	375	1	62	1,364	1,342	1,342	1,349
2	0	278,5	-1	325	1	62	1,342	1,386	1,386	1,371
3	0	278,5	1	375	-1	50	1,281	1,271	1,276	1,276
4	0	278,5	-1	325	-1	50	1,298	1,303	1,293	1,298
5	1	286	0	350	1	62	1,413	1,408	1,403	1,408
6	-1	271	0	350	1	62	1,320	1,320	1,342	1,327
7	1	286	0	350	-1	50	1,287	1,298	1,276	1,287
8	-1	271	0	350	-1	50	1,398	1,448	1,476	1,441
9	1	286	1	375	0	56	1,342	1,298	1,298	1,313
10	-1	271	1	375	0	56	1,276	1,320	1,342	1,313
11	1	286	-1	325	0	56	1,364	1,298	1,320	1,327
12	-1	271	-1	325	0	56	1,347	1,337	1,342	1,342
13	0	278,5	0	350	0	56	1,303	1,293	1,298	1,298

В таблице 4 представлены рабочая матрица и результаты экспериментальных исследований по выявлению закономерности изменения потребляемой мощности электродвигателя семяотводящей трубы y от параметров x_1, x_2 и x_3 , а при обработке результатов эксперимента использованы программные средства «MSExcel» [8].

Уравнение регрессии симметрического квази-D-оптимального плана имеет вид:

$$y = b_0 + b_1 x_1 + b_2 x_2 + b_3 x_3 + b_{12} x_1 x_2 + b_{13} x_1 x_3 + b_{11} x_1^2 + b_{22} x_2^2 + b_{33} x_3^2 \quad (2)$$

Для выявления полиномиальной связи между переменными параметрами y и x_1, x_2, x_3 необходимо найти аналитическое выражение уравнение регрессии, а для этого из таблицы 4 необходимо определить коэффициенты $b=(b_0, b_1, b_2, b_3, b_{12}, b_{13}, b_{23}, b_{11}, b_{22}, b_{33})$ уравнения регрессии (2).

Коэффициенты b_i уравнения регрессии находим с использованием метода наименьших квадратов, который является решением системы уравнений:

$$X^T X \cdot b = (X^T Y), \quad (3)$$

где X – матрица размерности 13×10 , первый столбец которой составлен из 1 (x_0), а другие 9 столбцов – из значений $x_1, x_2, x_3, x_1 x_2, x_1 x_3, x_2 x_3, x_1^2, x_2^2, x_3^2$, т.е. матрица X имеет следующую структуру:

$$X = \begin{bmatrix} 1 & 0 & 1 & 1 & 0 & 0 & 1 & 0 & 1 & 1 \\ 1 & 0 & -1 & 1 & 0 & 0 & -1 & 0 & 1 & 1 \\ 1 & 0 & 1 & -1 & 0 & 0 & -1 & 0 & 1 & 1 \\ 1 & 0 & -1 & -1 & 0 & 0 & 1 & 0 & 1 & 1 \\ 1 & 1 & 0 & 1 & 0 & 1 & 0 & 1 & 0 & 1 \\ 1 & -1 & 0 & 1 & 0 & -1 & 0 & 1 & 0 & 1 \\ 1 & 1 & 0 & -1 & 0 & -1 & 0 & 1 & 0 & 1 \\ 1 & -1 & 0 & -1 & 0 & 1 & 0 & 1 & 0 & 1 \\ 1 & 1 & 1 & 0 & 1 & 0 & 0 & 1 & 1 & 0 \\ 1 & -1 & 1 & 0 & -1 & 0 & 0 & 1 & 1 & 0 \\ 1 & 1 & -1 & 0 & -1 & 0 & 0 & 1 & 1 & 0 \\ 1 & -1 & -1 & 0 & 1 & 0 & 0 & 1 & 1 & 0 \\ 1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix},$$

а вектор Y , составленный из 13 значений y_i , т.е.

$$Y = \begin{pmatrix} 1,349 \\ 1,371 \\ 1,276 \\ 1,298 \\ 1,408 \\ 1,327 \\ 1,287 \\ 1,441 \\ 1,313 \\ 1,313 \\ 1,327 \\ 1,342 \\ 1,298 \end{pmatrix}.$$

Если матрицу $X^T X$ переводим обратную матрицу $(X^T X)^{-1}$, тогда коэффициенты $b = (b_0, b_1, b_2, b_3, b_{12}, b_{13}, b_{23}, b_{11}, b_{22}, b_{33})$ уравнения регрессии

$$b = (X^T X)^{-1} (X^T Y) \quad (4)$$

При определении значений коэффициентов уравнения регрессии нами были выполнены следующие шаги с использованием матричных функций MSExcel [8]. Транспонирование – функция ТРАНСП(диапазон ячеек); б) Умножение матриц (частный случай – умножение матрицы на вектор) – функция МУМНОЖ(диапазон1; диапазон2); в) Вычисление обратной матрицы – функция МОБР(диапазон ячеек).

Приведем результатов матричных функций MSExcel.

а) транспонирования матрицы X :

$$Y = \begin{pmatrix} 1,349 \\ 1,371 \\ 1,276 \\ 1,298 \\ 1,408 \\ 1,327 \\ 1,287 \\ 1,441 \\ 1,313 \\ 1,313 \\ 1,327 \\ 1,342 \\ 1,298 \end{pmatrix}.$$

б) транспонирования матрицы Y :

$$Y = \begin{pmatrix} 1,349 & 1,371 & 1,276 & 1,298 & 1,408 & 1,327 & 1,287 & 1,441 & 1,313 & 1,313 & 1,327 & 1,342 & 1,298 \end{pmatrix}$$

в) умножения матриц X и транспонированной матрицы X^T , т.е. $X^T X$:

$$X^T X = \begin{pmatrix} 13 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 8 & 8 & 8 \\ 0 & 8 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 8 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 8 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 4 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 4 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 4 & 0 & 0 & 0 & 0 \\ 8 & 0 & 0 & 0 & 0 & 0 & 0 & 8 & 4 & 4 & 4 \\ 8 & 0 & 0 & 0 & 0 & 0 & 0 & 4 & 8 & 4 & 4 \\ 8 & 0 & 0 & 0 & 0 & 0 & 0 & 4 & 4 & 8 & 8 \end{pmatrix}$$

г) вычисления обратной матрицы $(X^T X)^{-1}$:

$$(X^T X)^{-1} = \begin{vmatrix} 1 & 0 & 0 & 0 & 0 & 0 & 0 & -0,5 & -0,5 & -0,5 \\ 0 & 0,125 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0,125 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0,125 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0,25 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0,25 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0,25 & 0 & 0 & 0 \\ -0,5 & 0 & 0 & 0 & 0 & 0 & 0 & 0,4375 & 0,1875 & 0,1875 \\ -0,5 & 0 & 0 & 0 & 0 & 0 & 0 & 0,1875 & 0,4375 & 0,1875 \\ -0,5 & 0 & 0 & 0 & 0 & 0 & 0 & 0,1875 & 0,1875 & 0,4375 \end{vmatrix}$$

д) умножая матрицы Y и транспонированной матрицы X^T , получим:

$$X^T Y = \begin{vmatrix} 17,350 \\ -0,088 \\ -0,088 \\ 0,154 \\ 0,015 \\ 0,234 \\ 0,000 \\ 10,758 \\ 10,589 \\ 10,758 \end{vmatrix}$$

После определения матриц $X^T X$ и $X^T Y$ можно определить коэффициенты b уравнения регрессии, используя (4), т.е. умножая матрицы $X^T Y$ на $(X^T X)^{-1}$:

$$b = \begin{vmatrix} 1,298 \\ -0,011 \\ -0,011 \\ 0,019 \\ 0,004 \\ 0,059 \\ 0,000 \\ 0,034 \\ -0,008 \\ 0,034 \end{vmatrix}$$

Тогда уравнения регрессии потребляемой мощности электродвигателя семяотводящей трубы y (кВт) в зависимости от плотности сырцового валика x_1 в рабочей камере ($\text{кг}/\text{м}^3$), частоты вращения семяотводящей трубы x_2 и угла расположения отверстий x_3 на поверхности семяотводящей трубы имеет вид:

$$y = 1,298 - 0,011 x_1 - 0,011 x_2 + 0,019 x_3 + 0,004 x_1 x_2 + 0,059 x_1 x_3 + 0,034 x_1^2 - 0,008 x_2^2 + 0,034 x_3^2 \quad (5)$$

4. Анализ результатов и обсуждение.

Анализ уравнения регрессии, т.е. значимость коэффициентов уравнения регрессии (5) по критерию Стьюдента с использованием параллельных опытов (таблица 4) приведены в таблице 5.

Проверяем однородность ряда скорректированных дисперсий (таблица 5):

$$G = \frac{S_{jmax}^2}{\sum_{j=1}^N S_j^2} = \frac{0,0016}{0,0057} = 0,275 < G_{1-g}(v_1, v_2) = G_{0,05}(2,13) = 0,3924 ,$$

где при $g=0,05$ число степеней свободы $v_1=m-1=3-1=2$ и $v_2=N=13$; N – число опытов; m – число параллельных опытов. При $G= 0,275$ ряд однороден. Тогда дисперсия воспроизводимости составит

$$S_B^2 = \frac{1}{N} \sum_{j=1}^N S_j^2 = \frac{0,0057}{13} = 0,0004; k_B = N(m-1) = 13(3-1) = 26 .$$

Для определения значимости коэффициентов найдем среднее квадратичное отклонение

$$S_{bi} = \sqrt{\frac{S_B^2}{nN}} = \sqrt{\frac{0,0004}{3 \cdot 13}} = 0,0033, \quad i=0, 1, 2, 3.$$

Таблица 5.

Результаты анализа уравнений регрессии

Номер опыта	f_N	Эмпирическая дисперсия		$\bar{y}$	$\hat{y}$	$\bar{y} - \hat{y}$	$(\bar{y} - \hat{y})^2$
		S_j^2	S_j				
1	3	0,0002	0,0127	1,3493	1,3320	0,0174	0,0003
2	3	0,0006	0,0254	1,3713	1,3540	0,0174	0,0003
3	3	0,0000	0,0050	1,2760	1,2934	-0,0174	0,0003
4	3	0,0000	0,0050	1,2980	1,3154	-0,0174	0,0003
5	3	0,0000	0,0050	1,4080	1,4327	-0,0247	0,0006
6	3	0,0002	0,0127	1,3273	1,3374	-0,0101	0,0001
7	3	0,0001	0,0110	1,2870	1,2769	0,0101	0,0001
8	3	0,0016	0,0395	1,4407	1,4160	0,0247	0,0006
9	3	0,0006	0,0254	1,3127	1,3054	0,0073	0,0001
10	3	0,0011	0,0336	1,3127	1,3200	-0,0073	0,0001
11	3	0,0011	0,0336	1,3273	1,3200	0,0073	0,0001
12	3	0,0000	0,0050	1,3420	1,3493	-0,0073	0,0001
13	3	0,0000	0,0050	1,2980	1,2980	0,0000	0,0000
Сумма	39	0,0057	0,2189	17,3503	17,3503	0,0000	0,0028

При вышеуказанных значениях критерии Стьюдента будут следующими:

$$t_0 = \frac{|b_0|}{S_{b_0}} = \frac{1,298}{0,0058} = 223,908 > t_{0,025;26} = 2,478, \quad t_1 = \frac{|b_1|}{S_{b_1}} = \frac{0,011}{0,0074} = 1,483 < t_{0,025;26} = 2,478,$$

$$t_2 = \frac{|b_2|}{S_{b_2}} = \frac{0,011}{0,0074} = 1,488 < t_{0,025;26} = 2,478, \quad t_3 = \frac{|b_3|}{S_{b_3}} = \frac{0,019}{0,0074} = 2,611 > t_{0,025;26} = 2,478,$$

$$t_{12} = \frac{|b_{12}|}{S_{b_{12}}} = \frac{0,004}{0,01} = 0,351 < t_{0,025;26} = 2,478, \quad t_{13} = \frac{|b_{13}|}{S_{b_{13}}} = \frac{0,059}{0,01} = 5,605 > t_{0,025;26} = 2,478,$$

$$t_{23} = \frac{|b_{23}|}{S_{b_{23}}} = \frac{0,000}{0,01} = 0 < t_{0,025;26} = 2,478, \quad t_{11} = \frac{|b_{11}|}{S_{b_{11}}} = \frac{0,034}{0,007} = 4,584 < t_{0,025;26} = 2,478,$$

$$t_{22} = \frac{|b_{22}|}{S_{b_{22}}} = \frac{0,008}{0,008} = 1,11 < t_{0,025;26} = 2,478, \quad t_{33} = \frac{|b_{33}|}{S_{b_{33}}} = \frac{0,034}{0,008} = 4,584 < t_{0,025;26} = 2,478.$$

Критическое значение Стьюдента $t_{\alpha/2; k_B}$ будет $t_{0,025;26} = 2,478$. Из 10 коэффициентов $b_5 - t_1, t_2, t_{12}, t_{23}, t_{22}$ незначимы, остальные 5 коэффициентов модели значимы. Поэтому $k_{ад} = N - (10 - 5) = 13 - 5 = 8$.

Адекватность модели (5) определяем, вычисляя дисперсию адекватности:

$$S_{ад}^2 = \frac{1}{k_{ад}} \sum_{j=1}^N (\bar{y}_i - \hat{y}_i)^2 = \frac{0,0028}{8} = 0,00334.$$

Расчетные значения критерия Фишера показывают, что модель (5) адекватная:

$$F = \frac{S_{ад}^2}{S_B^2} = \frac{0,00334}{0,0004} = 0,813 < F_{0,05;8;26} = 3,10 .$$

Гипотеза об адекватности принятой модели (5) не отвергается. Уравнение регрессии (5) адекватно с доверительной вероятностью 95%.

С учетом дисперсии, адекватности, критерия Стьюдента и Фишера уравнение регрессии потребляемой мощности электродвигателя семяотводящей трубы в зависимости от факторов x_1, x_2, x_3 имеет следующий вид:

$$y = 1,298 + 0,019 x_3 + 0,059 x_1 x_3 + 0,034 x_1^2 + 0,034 x_3^2 \quad (6)$$

5. Заключение.

В результате проведенных экспериментальных исследований по изучению потребляемой мощности электродвигателя семяотводящей трубы устроена математическая модель (6) в зависимости от плотности сырцового валика x_1 , частоты вращения семяотводящей трубы x_2 и угла расположения отверстий на поверхности семяотводящей трубы x_3 (рисунок 6-7).

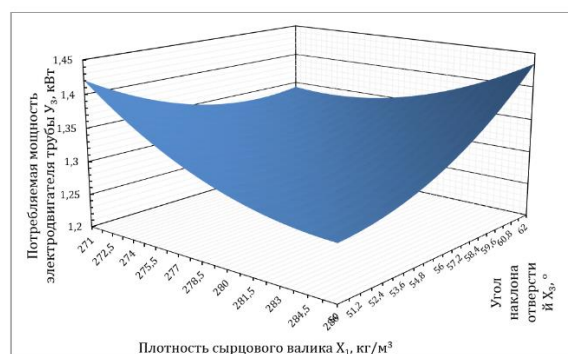


Рисунок 6. – Изменения потребляемой мощности электродвигателя в зависимости от плотности сырцового валика x_1 и угла наклона отверстий x_3

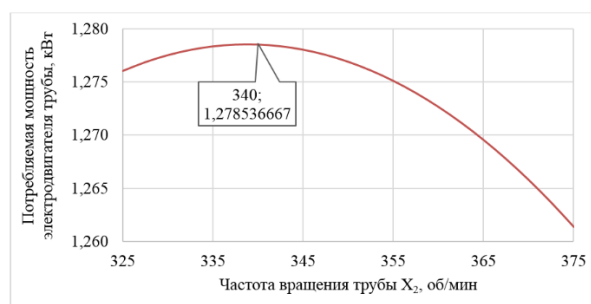


Рисунок 7. – Изменения потребляемой мощности электродвигателя в зависимости от частоты вращения трубы x_2

Установлено, что минимальное значение потребляемой мощности электродвигателя семяотводящей трубы достигается до $y=1.278$ кВт (рисунки 6 и 7) при частоте вращения трубы $x_2 = -0.4$ (340 об/мин), плотности сырцового валика в рабочей камере $x_1 = 1$ (286 кг/м^3) и угле расположения отверстий на поверхности семяотводящей трубы $x_3 = -1$ (50°).

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АНАЛИЗ КИНЕМАТИЧЕСКИХ ПАРАМЕТРОВ ШПИНДЕЛЕЙ И СЪЁМНИКОВ ГОРИЗОНТАЛЬНО-ШПИНДЕЛЬНЫХ ХЛОПКОУБОРОЧНЫХ МАШИН

Норкузиев О.С.¹

¹Институт механики и сейсмостойкости сооружений им. М.Т. Уразбаева, Ташкент, Узбекистан
E-mail: oqmirza@bk.ru

Аннотация: Статья посвящена исследованию и разработке теории моделирования технологического процесса горизонтально-шпиндельного аппарата, методике исследований по взаимодействию съёмника и шпинделей расчетно-графоаналитическим методом при использовании реальных материалов технологических параметров ХУМ и экспериментальных исследований в следующем порядке: анализ усредненных кинематических параметров шпинделей, подходящих к зоне съёма; вывод аналитических зависимостей для описания функций перемещений, скоростей и ускорений при взаимодействии шпинделей; разработка алгоритмов для численных исследований кинематических параметров шпинделей и съёмника. На основе методов аналитической геометрии и кинематики сложного плоскопараллельного движения была решена обобщенная задача для определения каждой точки шпинделя при контакте с дисковым съёмником уборочного аппарата, которая позволила анализировать движения шпинделя в циклическом процессе работы. В среде программирования MathCAD построены графики, описывающие закономерности абсолютного движения вершины зуба шпинделя и диска съёмника, вектор направления при различных скоростях работы хлопкоуборочной машины, выполняющих очистку поверхности шпинделя от хлопковых волокон.

Ключевые слова: горизонтально-шпиндельные хлопкоуборочные машины; рабочая камера; барабан; шпиндель; угловая скорость

1. Введение.

На всех этапах процесса машинного сбора при съёме хлопка с кустов шпинделями и транспортировании хлопкового волокна, оно не должно терять свое ценное текстильное качество при взаимодействии с рабочими органами уборочного аппарата. Как известно, эффективная работа ХУМ напрямую зависит от параметров рабочих органов (барабан, шпиндель, съёмник, воздушно-транспортная камера и т.п.) и оптимального построения технологии работы уборочного аппарата. Шпиндель и съёмник являются основными рабочими органами, которые вращаясь, осуществляют наматывание и съём хлопка. В хлопкоуборочной машине скорость вращения и траектория движения шпинделей являются ключевым фактором, влияющим на эффективность и качество сбора хлопка. В данной работе проводится кинематический анализ взаимодействия съёмника со шпинделями горизонтально-шпиндельного хлопкоуборочного аппарата в процессе съёма дисковыми съёмниками хлопкового волокна со шпинделей хлопкоуборочной машины.

2. Материалы, методы и объект исследования.

Расчётный анализ кинематики горизонтально-шпиндельного хлопкоуборочного аппарата проведен с учетом результатов ранее проведенных исследований [1-12]. На основе численных методов с использованием реальных технологических материалов по параметрам машин (Case 2022) и экспериментальных исследований проведен анализ процесса кинематики взаимодействия съёмника и шпинделя. Для численных расчетов принимаем технологическую схему работы горизонтально-шпиндельного аппарата (ГША) [3] (рис.1) и следующие первоначальные данные: качество интервалов разбивки угла поворота шпиндельного барабана $N_d=24$; частота вращения барабана $n_b=158$ об/мин; частота вращения съёмника $n_c=1458$ об/мин; длина поводка шпинделя $L_p=50,8$ мм; радиус барабана по центрам привода шпинделей $R_b=123.82$ мм; длина шпинделя от оси вращения вала шестерен кассеты до точки Б (начало активной части) $l_{кб}=74.84$ мм = O_iB_i ; расположение оси съёмника относительно координат барабана γ равно $\pi/6$; наружный радиус выступа съёмника, взаимодействующего со шпинделем, $R_m=94$ мм; внутренний радиус выступа съёмника, взаимодействующего со шпинделем, $R_n=71$ мм.

Для выполнения расчетного исследования применены методы математического моделирования и проведен анализ технологических процессов в работе съёмника и шпинделей уборочного аппарата на основе аналитической механики, теории механизмов и сельскохозяйственных (хлопкоуборочных) машин. Теоретические исследования проведены на

основе классических методов моделирования закономерностей изменения исследуемых параметров с использованием среды программирования MathCad.

Ранее проведенный (1) кинематический анализ движения кассет шпинделей в рабочей зоне позволяет провести анализ кинематических параметров взаимодействия шпинделей и съёмников. Для этого проведен анализ конструкции ГША и сделан оттиск паза натурального привода кассет шпинделей (2) хлопоуборочной машины фирмы “Case 2022”. Оттиск паза сделан с точностью $\pm 1,4$ мм. Перед проведением исследований проведена следующая подготовка.

1. Оттиск паза привода перенесен на бумагу в масштабе 1:1. Графически с помощью циркуля и лекал построена кривая точек центра паза и осевая окружность центров кассет шпиндельного барабана $\phi 123.82$ мм с учетом [3].

2. Из условий, что шпиндели в рабочей зоне имеют единственное положение, когда ось шпинделя находится на одной прямой с радиусом барабана и занимает положение перпендикулярно оси рядка, поводок длиной 50.8 мм перпендикулярен шпинделю. А центр ролика привода поводка совпадает с кривой центра паза, где делалось построение каждого положения шпинделя. Исходя из этого положения, на рис.1. выбрана система прямоугольных координат относительно центра вращения шпиндельного барабана и паза привода кассет. Ось O_6Y_6 совпадала с постоянным положением шпинделя с положительными значениями Y_6 со стороны рабочей зоны. Положительные значения O_6Y_6 имели направления в сторону зоны увлажнения шпинделей. Отсчет угла поворота шпиндельного барабана начинался от положительных значений оси O_6Y_6 .

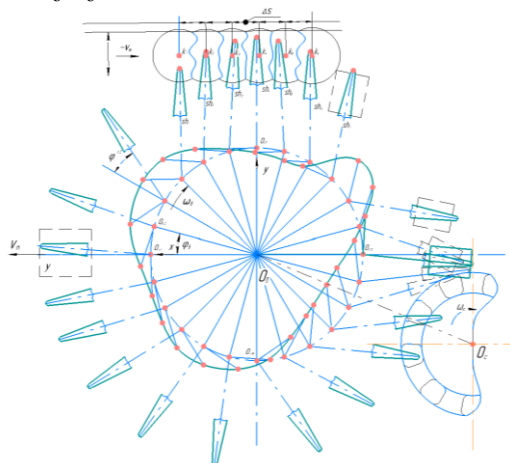


Рисунок 1. – Технологическое построение положений шпинделей и траектории паза кулачкового привода кассет горизонтального шпиндельного аппарата (ГША)

3. Проводились радиальные линии, соответствующие повороту барабана на угол $\pi/12$. Циркулем, радиусом 50.8 мм от положений поворота барабана делались зачески на кривой центра паза привода кассет. С помощью линеек с точностью ± 0.5 мм проведен замер координат этих пересечений. Полученные координаты в первом приближении явились основой дальнейшего анализа. Расчет параметров проводился аналитически, методом координат. Отдельные положения взаимодействия шпинделей и съёмника проводились графически, путем построения в масштабе 1:1. На рис 1, в качестве примера и иллюстрации показаны графические построения положения оси шпинделя, поводка, привода, кассет, барабана и съёмника в уменьшенном масштабе 1:2 .

3. Постановка задачи.

Отношение окружных скоростей шпинделей и барабана не изменяется при изменении частоты вращения вала двигателя, от которого осуществляется привод к ведущим колесам машины и к шпиндельным аппаратам. Но вследствие того, что при работе аппарата шпиндельная кассета поворачивается, вал шпиндельной кассеты также поворачивается, и частота вращения шпинделей увеличивается или уменьшается в зависимости от угла поворота

φ_{ki} (рис.1), расчетные значения углов поворота φ_{ki} кассет , усредненных значений угловых скоростей $\dot{\varphi}_{ki}$, ускорения $\ddot{\varphi}_{ki}$ отражены в таблице 1.

Таблица 1.

Расчетные значения углов поворота φ_{ki} кассет , усредненных значений угловых скоростей $\dot{\varphi}_{ki}$ и ускорения $\ddot{\varphi}_{ki}$

№	1	2	3	4	5	6	7	8	9	10	11	12
φ_k , rad/s	0	$\pi/12$	$\pi/6$	$\pi/4$	$\pi/3$	$5\pi/12$	$\pi/2$	$7\pi/12$	$2\pi/3$	$3\pi/4$	$5\pi/6$	$11\pi/12$
$\dot{\varphi}_k$, rad/s	0.236	0.960	0.960	0.786	0.524	0.262	0.00	-0.262	-0.524	-0.786	-0.786	0.736
$\ddot{\varphi}_k$, rad/s ²	31.42	22.9	-5.53	-13.81	-16.56	-16.56	-16.56	-16.56	-16.56	-8.28	48.1	68.15
№	13	14	15	16	17	18	19	20	21	22	23	24
φ_k , rad/s	π	$13\pi/12$	$7\pi/6$	$5\pi/4$	$4\pi/3$	$17\pi/12$	$3\pi/2$	$19\pi/12$	$5\pi/3$	$7\pi/4$	$11\pi/6$	$23\pi/12$
$\dot{\varphi}_k$, rad/s	0.236	0.960	0.960	0.786	0.524	0.262	0.00	-0.262	-0.524	-0.786	-0.786	0.736
$\ddot{\varphi}_k$, rad/s ²	-28.19	-25.75	-25	-13.1	6.6	-16.5	-26.45	33.1	30.35	17.6	-12.53	-15.22

Расчет кинематических параметров взаимодействия шпинделей и съёмников проводился по следующему алгоритму.

1. Проводился ввод исходных данных: номер i - той точки положения барабана N_d , X_{ki} , Y_{ki} значения координат центра паза K_i , соответствующей точкам привода кассет.

2. По теореме косинусов из треугольника $O_6K_iO_i$ (рис. 2) рассчитывались углы α_i , β_i для соответствующих положений барабана, при этом радиус до центра паза кулачка $K_i = \sqrt{x_{ki}^2 + y_{ki}^2}$, где R_6 , L_p – известные величины, а также рассчитывался угол поворота кассеты относительно вертикальной плоскости $\varphi_i = \frac{\pi}{2} - \alpha_i - \beta_i$.

3. По формулам рассчитывались параметры и координаты:

$$\omega_6 = \frac{\pi}{30} n_6 \text{ – частота вращения барабана;}$$

$$T_6 = \frac{60}{n_6} \text{ – время одного оборота барабана;}$$

$$D_6 = \frac{T}{N_6} \text{ – дискретизация времени;}$$

$T_i = Nt_i D_i$ определение времени поворота до i -ого положения; координаты i -ого положения по центрам привода O_i кассет шпинделей:

$$X_{oi} = R_6 \cos(\omega_6 T_i), Y_{oi} = R_6 \sin(\omega_6 T_i) \quad (1)$$

- координаты точек Б и В на оси шпинделя:

$$\begin{aligned} X_{6i} &= X_{oi} + l_{к6} \cos(\omega_6 T_i + \varphi_i); \\ Y_{6i} &= Y_{oi} + l_{к6} \sin(\omega_6 T_i + \varphi_i); \\ X_{bi} &= X_{oi} + l_{к6} \cos(\omega_6 T_i + \varphi_i); \\ X_{bi} &= X_{oi} + l_{к6} \sin(\omega_6 T_i + \varphi_i); \end{aligned} \quad (2)$$

4. В связи с тем, что координаты центра паза привода кассет замерены с недостаточной точностью, а их отклонения до 0.1 мм дают расхождение координат точки В до 9.7 м, проводилось уточнение координат центра паза кулачка, исходя из условия постоянства и достоверности радиуса барабана и длины поводка. Для этого определялась расчетная длина поводка L_p в каждом положении:

$$L_{pi} = \sqrt{(X_{ki} - X_{oi})^2 + (Y_{ki} - Y_{oi})^2} \quad (3)$$

Определялись расхождения между нормальной и расчетной длинами. По углу φ_i находились корректирующие величины по осям $X_6 Y_6$. После этого опять определялась расчетная длина поводка. Такое уточнение проводилось до совпадения длины поводка с

точностью до 0,0 мм. Результаты вычислений представлены в таблицах 2-7. После этого уточненные координаты взяты исходными данными вычислений.

5. Затем вычислялись координаты центра съёмника (т.О_с):

$$\begin{aligned} X_c &= A \cos(7\pi/6), \\ Y_c &= A \sin(7\pi/6). \end{aligned} \quad (4)$$

6. Далее проводился параллельный перенос координат к центру съёмника. $X_{бpi}$, $Y_{бpi}$ – координаты т. Б, $X_{вpi}$, $Y_{вpi}$ – координаты т.В. Определялся угол γ_i наклона оси шпинделя к оси О_с, X_c :

$$\gamma_i = \arctg(X_{ki} - X_{oi}) + (Y_{ki} - Y_{oi}), \quad (5)$$

7. Учитывался знак при вычислении корней квадратного уравнения.

8. Зная координаты точек Б и В, составлялись уравнения прямых относительно X_c и Y_c в виде вычисления коэффициентов при неизвестных и свободных членах уравнения линии оси шпинделя по формулам:

$$\begin{aligned} K_{xi} &= (X_{вpi} - X_{бpi})(Y_{вpi} - Y_{бpi}), \\ C_{xi} &= (X_{бpi} \cdot Y_{бpi} - Y_{вpi} \cdot X_{вpi}) : (Y_{вpi} - Y_{бpi}), \\ K_{yi} &= (Y_{вpi} - Y_{бpi})(X_{вpi} - X_{бpi}), \\ C_{yi} &= (Y_{бpi} \cdot X_{бpi} - X_{вpi} \cdot Y_{вpi}) : (X_{вpi} - X_{бpi}), \end{aligned} \quad (6)$$

9. Нахождение координат точек пересечения линии оси шпинделя с наружным R_m и внутренним R_n радиусами съёмника, взаимодействующих со шпинделем, проводилось совместным решением уравнений окружностей $X^2 + Y^2 = R_m^2$ и соответствующих уравнений линий оси шпинделя, путем подстановки уравнений линий в уравнения окружностей. В результате такой подстановки получены квадратные уравнения с действительными числами в виде выражений из коэффициентов и свободных членов уравнений оси шпинделя. Затем вычислялись дискриминанты квадратных уравнений по выражениям:

$$\begin{aligned} D_{myi} &= (2K_{xi} \cdot C_{xi})^2 - 4(1 + K_{xi}^2) \cdot (C_{xi}^2 - R_m^2), \\ D_{mxi} &= (2K_{yi} \cdot C_{yi})^2 - 4(1 + K_{yi}^2) \cdot (C_{yi}^2 - R_m^2), \\ D_{nyi} &= (2K_{xi} \cdot C_{xi})^2 - 4(1 + K_{xi}^2) \cdot (C_{xi}^2 - R_n^2), \\ D_{nxi} &= (2K_{yi} \cdot C_{yi})^2 - 4(1 + K_{yi}^2) \cdot (C_{yi}^2 - R_n^2). \end{aligned} \quad (7)$$

Вычисление координат точек пересечения радиусов съёмника с осевыми линиями шпинделей относительно центра съёмника проводилось по выражениям:

$$\begin{aligned} X_{моi} &= (-2K_{yi} \cdot C_{yi} + \sqrt{D_{mxi}}) : 2(1 + K_{yi}^2), \\ X_{мрi} &= (-2K_{yi} \cdot C_{yi} - \sqrt{D_{mxi}}) : 2(1 + K_{yi}^2), \\ Y_{моi} &= (-2K_{xi} \cdot C_{xi} + \sqrt{D_{myi}}) : 2(1 + K_{xi}^2), \\ Y_{мрi} &= (-2K_{xi} \cdot C_{xi} - \sqrt{D_{myi}}) : 2(1 + K_{xi}^2), \\ X_{ноi} &= (-2K_{yi} \cdot C_{yi} + \sqrt{D_{nxi}}) : 2(1 + K_{yi}^2), \\ X_{нрi} &= (-2K_{yi} \cdot C_{yi} - \sqrt{D_{nxi}}) : 2(1 + K_{yi}^2), \\ Y_{ноi} &= (-2K_{xi} \cdot C_{xi} + \sqrt{D_{nyi}}) : 2(1 + K_{xi}^2), \\ Y_{нрi} &= (-2K_{xi} \cdot C_{xi} - \sqrt{D_{nyi}}) : 2(1 + K_{xi}^2), \end{aligned} \quad (8)$$

где $X_{мо}$, $Y_{мо}$, $X_{но}$, $Y_{но}$ – координаты точек пересечения, лежащие со стороны барабана; $X_{мр}$, $Y_{мр}$, $X_{нр}$, $Y_{нр}$ – противоположные координаты точек пересечения. При этом учтены

следующие условия:

а) если все дискриминанты отрицательны, то точки пересечения отсутствуют, тогда координатам присваивались нули;

б) если дискриминанты внешнего радиуса положительны, а внутреннего отрицательны, то вычисляются только точки пересечения с внешним радиусом, при этом, исходя из значения угла γ (положительное или отрицательное), ставился знак перед квадратным корнем дискриминанта.

в) если все дискриминанты положительны, то вычисляются все точки пересечения с подстановкой знака перед квадратным корнем дискриминанта, исходя из значения угла γ .

10. Проводилось вычисление следующих величин расстояний:

L_{ci} – расстояние между осевой линией привода шпинделей по окружности барабана и точками пересечения осевой линии шпинделя с наружным радиусом съёмника, лежащими со стороны барабана;

L_{coi} – расстояние между осевыми линиями привода шпинделей по окружности барабана и точками пересечения осевой линии шпинделя с наружным радиусом съёмника, лежащими с дальней стороны от барабана;

L_{vci} – расстояние между осевой линией привода шпинделей по окружности барабана и точками пересечения осевой линии шпинделя с внутренним радиусом съёмника, лежащими со стороны барабана, по выражениям:

$$\begin{aligned} L_{ci} &= \sqrt{(X_{mpi} - X_{овci})^2 + (Y_{mpi} - Y_{овci})^2}, \\ L_{coi} &= \sqrt{(X_{moi} - X_{овci})^2 + (Y_{moi} - Y_{овci})^2}, \\ L_{vci} &= \sqrt{(X_{hpi} - X_{овci})^2 + (Y_{hpi} - Y_{овci})^2}, \end{aligned} \quad (9)$$

где $X_{овci}, Y_{овci}$ – координаты i -того положения по центрам привода шпинделей в системе координат в центре O_c съёмника.

Вычисление координат точек, в которых будут вычисляться модули и направления векторов скоростей в системе координат в центре съёмника: X_{ni}, Y_{ni} – расположенные на наружном радиусе съёмника (т. п. рис.2.). X_{vi}, Y_{vi} – расположенные между или во внутреннем радиусе съёмника (т. vi. рис.1.) и их же значения в системе координат в центре барабана $X_{nbi}, Y_{nbi}, X_{vbi}, Y_{vbi}$.

Вычисления проводились согласно следующим условиям:

а) если координаты точек пересечения радиусов съёмника с осевыми линиями шпинделей равны нулю, то есть нет пересечений, то в этих положениях шпинделей координаты для вычисления скоростей также равны нулю;

б) если есть пересечения с наружным радиусом съёмника и нет пересечений с внутренним радиусом и $L_{coi} < L_{кв}$, то значения координат для вычисления скоростей принимали значения координат пересечения с наружным радиусом съёмника;

в) если есть пересечения с наружным радиусом съёмника и нет пересечений с внутренним радиусом и $L_{coi} > L_{кв}$, то есть точки лежат между радиусом съёмника, то координаты точек V_i вычислялись методом деления отрезка в данном отношении по следующим выражениям:

$$\begin{aligned} \lambda_i &= L_{кв}(L_{coi} - L_{кв}). \\ X_{vi} &= L_{кв}(X_{oci} + \lambda_i X_{noi}): (1 + \lambda_i) \\ Y_{vi} &= L_{кв}(Y_{oci} + \lambda_i Y_{noi}): (1 + \lambda_i); \end{aligned} \quad (10)$$

г) из условия пересечения с осью шпинделя наружного и внутреннего радиусов съёмника проводилось перераспределение значений перечисленных координат для вычисления скоростей;

д) из условия пересечения наружного радиуса съёмника и выхода из пересечения с внутренним радиусом и при условии $L_{vo} > L_{кв}$, переписывались значения координат пересечений с наружным радиусом, а координаты точек V_i вычислялись аналогично пункту в;

е) из условия $L_{ci} > L_{кв}$, когда расстояние между осевой линией привода шпинделей по окружности барабана и точками пересечения осевой линии шпинделя с наружным радиусом съёмника, лежащими со стороны барабана больше, чем рабочая длина шпинделя $L_{кв}$, значениям координат точек для вычисления скоростей присваивались нули.

11. Расчет скорости $d\varphi_i/dt$ угла поворота шпинделя проводился дискретно для каждого положения шпинделя, то есть как разность углов, смежных положению деления на дискрету времени:

$$\frac{d\varphi_i}{dt} = (\varphi_{i+1} - \varphi_i): D_t = \varphi_i. \quad (11)$$

12. Расчет расстояний по оси шпинделя между осевой линией привода шпинделей по окружности барабана и точками следует из выражений:

$$\begin{aligned} L_{ni} &= \sqrt{(X_{ni} - X_{o6i})^2 + (Y_{ni} - Y_{n6i})^2} \\ L_{vi} &= \sqrt{(X_{vi} - X_{o6i})^2 + (Y_{vi} - Y_{n6i})^2} \end{aligned} \quad (12)$$

13. Вычисление совместных линейных скоростей барабана и шпинделя для точек n_i и v_i , по направлению координатных осей с условием, что при отсутствии пересечения скорости равны нулю, проводились по формулам:

$$\begin{aligned} V_{xni} &= -R_6 \omega_6 \sin(T_i \omega_6) - L_{ni}(\omega_6 + \dot{\varphi}) \sin(\omega_6 T_i + \omega_i), \\ V_{xvi} &= -R_6 \omega_6 \sin(T_i \omega_6) - L_{vi}(\omega_6 + \dot{\varphi}) \sin(\omega_6 T_i + \omega_i), \\ V_{yni} &= -R_6 \omega_6 \cos(T_i \omega_6) - L_{ni}(\omega_6 + \dot{\varphi}) \cos(\omega_6 T_i + \omega_i), \\ V_{yni} &= -R_6 \omega_6 \cos(T_i \omega_6) - L_{ni}(\omega_6 + \dot{\varphi}) \cos(\omega_6 T_i + \omega_i), \end{aligned} \quad (13)$$

Расчет модуля вектора суммарной линейной скорости барабана и шпинделя для точек n_i и v_i , осуществлялся по выражениям:

$$V_{6ni} = \sqrt{V_{xni}^2 + V_{yni}^2}, \quad V_{6vi} = \sqrt{V_{xvi}^2 + V_{yvi}^2}. \quad (14)$$

Расчет углов γ_{6i} направления векторов относительно оси O_6X_6 (положительное по часовой стрелке) – по $\gamma_{6i} = \arctg(V_{yi}/V_{xi})$.

14. Расчет радиусов съёмника до точек n_i и v_i , в системе координат относительно центра съёмника согласно выражениям:

$$R_{ni} = \sqrt{X_{ni}^2 + Y_{ni}^2}, \quad R_{vi} = \sqrt{X_{vi}^2 + Y_{vi}^2}. \quad (15)$$

15. Расчет модуля вектора скорости съёмника в точках n_i и v_i :

$$V_{cni} = \omega_6 R_{ni} V_{cvi} = \omega_c R_{vi}, \quad (16)$$

где $\omega_c = \frac{\pi}{30} n_c$ – угловая частота вращения съёмника;

16. Расчет углов γ_{ci} направления векторов относительно оси O_6X_6 (положительно по часовой стрелке) $\gamma_{ci} = \arctg(V_{yi}/V_{xi}) + \pi/2$ и проекция векторов скоростей съёмника на оси координат:

$$\begin{aligned} V_{xcni} &= V_{cni} \cos(\gamma_{cni}), \quad V_{ycni} = V_{cni} \sin(\gamma_{cni}), \\ V_{xcvi} &= V_{cvi} \cos(\gamma_{cvi}), \quad V_{ycvi} = V_{cvi} \sin(\gamma_{cvi}). \end{aligned} \quad (17)$$

17. Расчет разности углов между векторами скоростей:

$$\gamma_{rni} = \gamma_{cni} + \gamma_{6ni}, \quad \gamma_{rvi} = \gamma_{cvi} + \gamma_{6vi}. \quad (18)$$

18. Расчет модуля суммарного вектора скоростей съёмника и шпинделя в точках n_i и v_i проводился согласно выражениям:

$$\begin{aligned} V_{ni} &= \sqrt{V_{6ni}^2 + V_{cni}^2 + 2V_{6ni}V_{cni} \cos(\gamma_{rni})}, \\ V_{vi} &= \sqrt{V_{6vi}^2 + V_{cvi}^2 + 2V_{6vi}V_{cvi} \cos(\gamma_{rvi})}. \end{aligned} \quad (19)$$

19. Проводился расчет сумм проекций векторов скоростей шпинделя и съёмника с целью определения знака при вычислении углов:

$$\begin{aligned} R_{yn} &= V_{y6ni} + V_{ycni}, \quad R_{xn} = V_{x6ni} + V_{xcni}, \\ R_{yv} &= V_{y6vi} + V_{ycvi}, \quad R_{xv} = V_{x6vi} + V_{xcvi}, \end{aligned} \quad (20)$$

20. Углы наклона суммарных векторов скоростей определялись по проекциям векторов скоростей съёмника и шпинделя на оси координат, исходя из условий:

а) если сумма проекций на ось O_6X_6 меньше нуля, то углы равны:

$$\gamma_{rni} = \pi + \arctg(R_{yn}/V_{xn}), \quad \gamma_{rvi} = \pi + \arctg(R_{yv}/V_{xv}); \quad (21)$$

б) если сумма проекций на ось O_6X_6 больше нуля, то:

$$\gamma_{rni} = \arctg(R_{yn}/V_{xn}), \quad \gamma_{rvi} = \arctg(R_{yv}/V_{xv}); \quad (22)$$

Таким образом, углы наклона суммарных векторов скоростей положительны и отсчет ведется по часовой стрелке от положительного направления оси OX_6 .

4. Результаты и их анализ.

По вышеописанному алгоритму написана программа в среде программирования «MathCAD», позволяющая в диалоговом режиме изменять вводимые параметры и выводить промежуточные результаты расчетов. Такая программа может являться основной для дальнейшего усовершенствования в области расчетов кинематических взаимодействий горизонтально-шпиндельного аппарата не только в зоне съема, но и в рабочей зоне, а также в зоне увлажнения по всей окружности шпиндельного барабана.

Один из результатов вычислений приведен в таблице 1. Анализ кинематики показывает, что поворот кассеты шпинделей от воздействия привода в зоне съема происходит неравномерно. В результате этого угловая скорость поворота от максимальной на этом участке в рад/с уменьшается до нуля и затем достигает значения 10 рад/с, но в обратном направлении.

Значения угловых скоростей поворота кассеты шпинделей с угловой скоростью барабана (16.54 рад/с) даны по порядку в таблице 2, а результаты вычислений линейных скоростей и их направлений приведены в таблицах 3-7.

Таблица 2.

Угол поворота φ_i и угловая скорость поворота кассеты φ_i шпинделя ГША в плане уборочного аппарата

Номер положения барабана	Угол поворота φ_i градус	Угловая скорость φ_i градус
56	-44.26	-0.00
57	-48.79	-19.99
58	-52.74	-17.40
59	-56.55	-16.84
60	-59.04	-10.96
61	-61.46	-10.69
62	-62.89	-6.30
63	-63.35	-2.08
64	-62.27	4.23
65	-62.13	0.62
66	-61.17	4.23
67	-60.27	3.99
68	-59.52	3.27
69	-58.02	6.63
70	-54.59	15.16
71	-52.31	10.04

Таблица 3.

Модули суммарных линейных скоростей шпинделя для точек n_i и v_i и их направления

№	Номер положения барабана	V_{6ni} м/с	γ_{6ni} град.	V_{6ni} м/с	V_{6ni} град
1	56	0.00	0.00	0.00	0.00
2	57	1.88	-49.82	1.81	-46.46
3	58	2.01	-51.11	1.99	-50.08
4	59	2.04	-48.28	2.03	-47.89
5	60	2.26	-53.11	2.48	-58.85
6	61	2.26	-50.05	2.43	-55.05
7	62	2.45	-52.34	2.69	-57.78
8	63	2.68	-53.92	2.98	-59.27
9	64	3.13	-56.80	3.57	-61.73
10	65	2.95	-50.43	3.27	-54.76
11	66	3.30	-50.68	3.70	-54.69
12	67	3.42	-47.70	3.84	-51.45
13	68	3.54	-44.56	3.89	-47.51
14	69	4.13	-44.38	4.21	-44.89
15	70	0.00	0.00	0.00	0.00
16	71	0.00	0.00	0.00	0.00

Таблица 4.

Модули суммарных линейных скоростей съёмника для точек n_i и v_i и их направления

№	Номер положения барабана	V_{bni} м/с	γ_{bni} град.	V_{bni} м/с	V_{bni} град
1	56	0.00	0.00	0.00	0.00
2	57	14.62	153.18	14.62	176.73
3	58	14.62	142.93	13.84	176.13
4	59	14.62	136.31	12.97	172.29
5	60	14.62	129.63	11.78	167.80
6	61	14.62	124.27	11.04	152.75
7	62	14.62	118.76	11.04	136.32
8	63	14.62	113.04	11.04	124.57
9	64	14.62	106.41	11.04	112.83
10	65	14.62	100.75	11.04	103.75
11	66	14.62	94.25	11.04	93.57
12	67	14.62	87.29	11.04	82.72
13	68	14.62	79.56	11.78	72.65
14	69	14.62	68.75	11.16	67.05
15	70	0.00	0.00	0.00	0.00
16	71	0.00	0.00	0.00	0.00

Таблица 5.

Модули суммарных линейных скоростей съёмника для точек n_i и v_i и их направления

№	Номер положения барабана	V_{bni} м/с	γ_{bni} град.	V_{bni} м/с	V_{bni} град
1	56	0.00	0.00	0.00	0.00
2	57	12.92	156.4	13.36	182.04
3	58	12.68	145.4	12.55	182.69
4	59	12.59	137.5	1150	178.83
5	60	12.36	130.3	10.24	177.92
6	61	12.37	123.3	8.96	160.03
7	62	12.20	116.8	8.45	140.77
8	63	12.03	110.6	8.07	125.98
9	64	11.66	101.8	7.50	110.24
10	65	12.12	94.03	8.09	95.23
11	66	12.07	85.22	8.13	79.70
12	67	12.44	76.06	8.81	84.46
13	68	12.97	66.52	10.39	53.77
14	69	13.54	52.46	13.18	49.82
15	70	0.00	0.00	0.00	0.00
16	71	0.00	0.00	0.00	0.00

Таблица 6.

Модули суммарных линейных скоростей шпинделей и съёмника
для точек n_i и v_i и их направления

№	Номер положения барабана	V_{bni} м/с	γ_{bni} град.	V_{bni} м/с	V_{bni} град
1	56	0.00	0.00	0.00	0.00
2	57	12.92	156.44	13.36	182.04
3	58	12.68	145.14	12.55	182.69
4	59	12.59	137.05	1150	178.83
5	60	12.36	130.13	10.24	177.92
6	61	12.37	123.23	8.96	160.03
7	62	12.20	116.98	8.45	140.77
8	63	12.03	110.16	8.07	125.98
9	64	11.66	101.98	7.50	110.24
10	65	12.12	94.03	8.09	95.23
11	66	12.07	85.22	8.13	79.70
12	67	12.44	76.06	8.81	84.46
13	68	12.97	66.52	10.39	53.77
14	69	13.54	52.46	13.18	49.82
15	70	0.00	0.00	0.00	0.00
16	71	0.00	0.00	0.00	0.00

**Разность модулей суммарных линейных скоростей шпинделей и съёмника
для точек n_i и v_i и разности их углов**

Номер положения барабана	$V_{ni} - V_{vi}$ м/с	$\gamma_{rvi} - \gamma_{rni}$ град.
56	0.00	0.00
57	-0.45	25.60
58	0.13	37.55
59	1.09	41.78
60	2.12	47.79
61	3.41	36.80
62	3.75	23.79
63	3.96	15.82
64	4.16	8.26
65	4.03	1.20
66	3.94	-5.52
67	3.63	-11.60
68	2.58	-12.75
69	0.36	-2.64
70	0.00	0.00
71	0.00	0.00

Анализ полученных вычислений показывает, что скорость шпинделя в среднем на порядок меньше линейной скорости съёмника. Направления линейных скоростей шпинделя и съёмника в начале взаимодействия (положения 57, 58, рис 3,4) почти противоположены (суммарные угол $203^0 - 220^0$), а на выходе эти векторы сближаются до угла $102^0 - 103^0$.

Сравнение суммарных скоростей в точках n_i и v_i показывает, что разность их достигает 4.16 м/с. Однако в начале взаимодействия для первых двух положений такая разность минимальна, а максимальная разность приходится на вторую половину взаимодействия.

Направление векторов суммарных скоростей в точках, которые меняются следующим образом: в начале взаимодействия (положения 57, 58) вектора направлены почти параллельно оси OX_c (156.44 и 182.02 град); при выходе из взаимодействия вектора скоростей направлены в сторону барабана (рис. 2).

Анализ направлений векторов суммарных скоростей в точках n_i и v_i (таблица 7) показывает, что в начале взаимодействия, когда на шпиндель намотан хлопок, разность углов составляет $26.6^0 - 37.55^0$. То есть происходит разрыв цилиндра хлопка, намотанного на шпиндель.

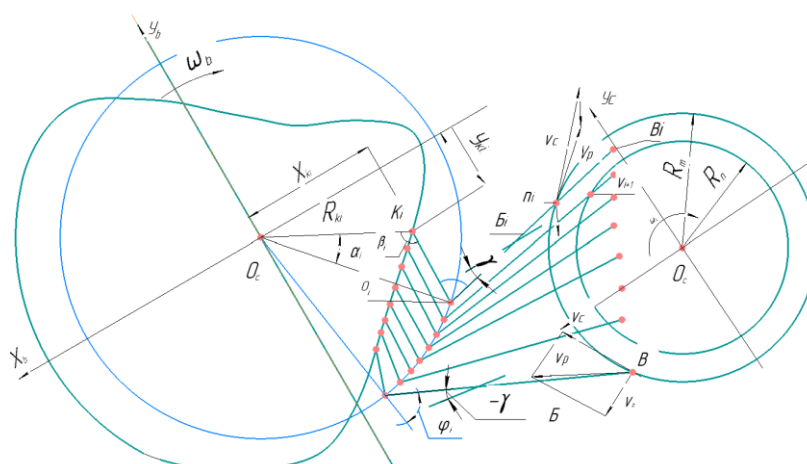


Рисунок 2. – Графоаналитическое построение положения для расчета кинематических параметров при взаимодействии шпинделей и съёмника

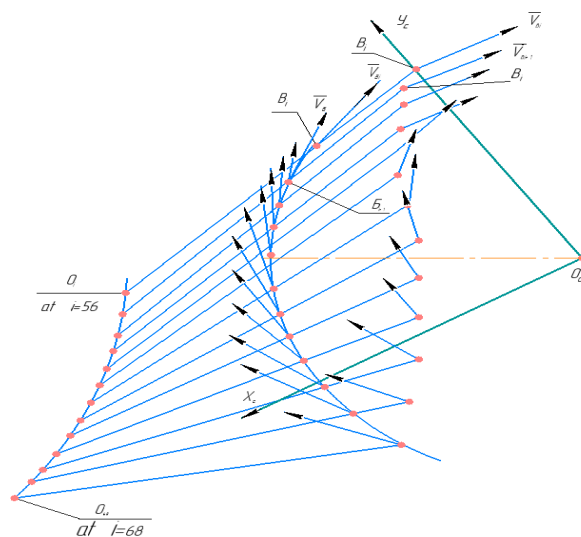


Рисунок 3. – План скоростей взаимодействия точек на поверхности шпинделей и съёмника

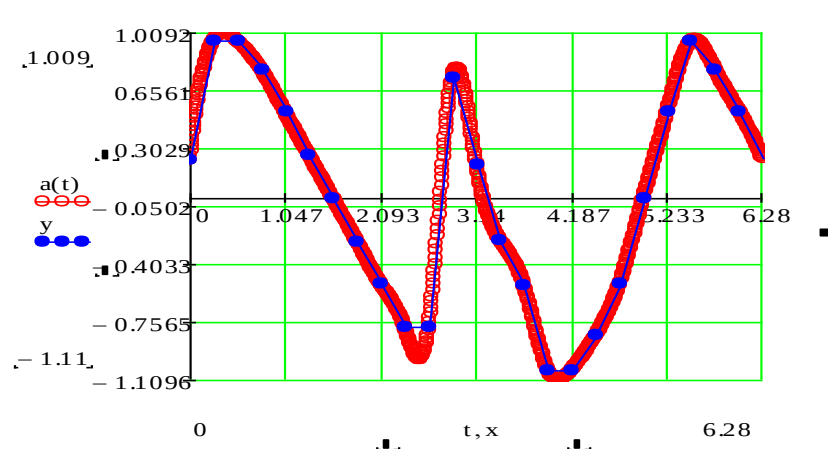


Рисунок 5. – Изменения угловой скорости вращения шпинделей за один оборот барабана в технологическом процессе работ ГША

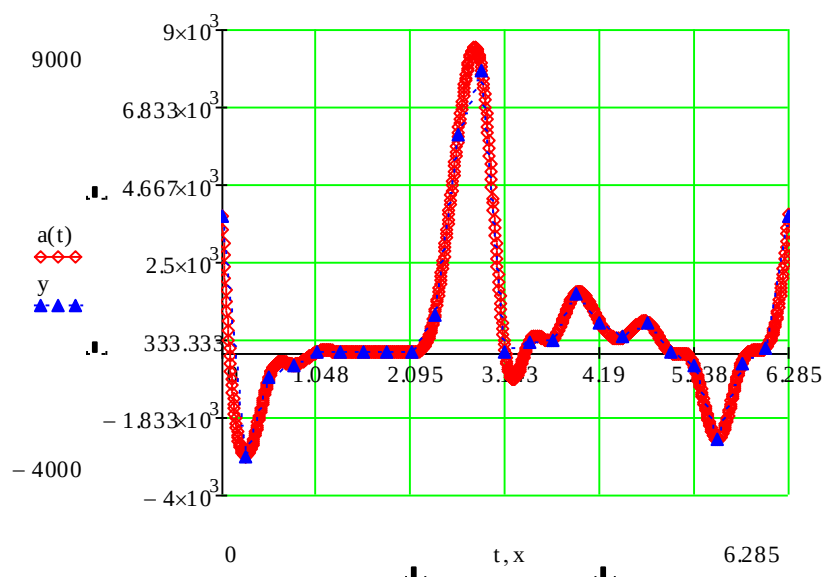


Рисунок 6. – Изменения угловой ускорения вращения шпинделей за один оборот барабана в технологическом процессе работ ГША

Используя координаты шпинделей и барабана (рис. 2) и таблицу 1 при изменении частоты вращения вала средняя частота вращения шпинделей изменяется в зависимости формы профиля направляющей дорожки кулачка и может увеличиваться или уменьшаться.

Построены графики угловой скорости шпинделей ω_{sh} (рис.5) и ускорения ϵ_{sh} (рис.6), используя данные полученных измерений координат расположения шпинделей одного цикла поворота шпиндельного барабана и угла поворота φ_{ki} (рис 2), расчетные значения углов поворота φ_{ki} кассет, усредненных значений угловых скоростей $\dot{\varphi}_{ki}$ и ускорения $\ddot{\varphi}_{ki}$ отражены в таблице 1.

4. Обсуждения.

В данной статье разработана методика анализа кинематических параметров шпинделя относительно диска съёмника, обеспечивающая оптимальное направление скорости съёма хлопка с переменнo вращающимися шпинделями в зоне съёма. По предложенной методике и алгоритму написана программа в среде “MathCAD”, которую в перспективе можно использовать для проектирования разрабатываемых рабочих органов (шпинделей и съёмников) горизонтально-шпиндельных хлопкоуборочных машин.

5. Выводы.

1. Составлена математическая модель кинематики изменения движения шпинделя рычажным кулачковым механизмом для привода шпинделей уборочного аппарата в зоне съёма горизонтально-шпиндельной хлопкоуборочной машины.
2. Предложена методика определения местонахождения шпинделя относительно диска съёмника, обеспечивающая оптимальное направление скорости съёма хлопка с переменнo вращающимися шпинделями в зоне съёма. Определены значения коэффициента обработки дисковым съёмником поверхности шпинделя по мере контакта со шпинделем, с учётом их длины и взаимного расположения планок съёмника относительно шпинделя.
3. По вышеописанному алгоритму написанная программа может являться основной для дальнейшего усовершенствования в области расчетов кинематики при технологическом цикле в зоне съёма, рабочей зоне, а также в зоне увлажнения.

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ИССЛЕДОВАНИЕ ТЕМПЕРАТУРЫ ПОДШИПНИКА СКОЛЬЖЕНИЯ ИЗ АНТИФРИКЦИОННОГО ДРЕВЕСНО-ПОЛИМЕРНОГО КОМПОЗИЦИОННОГО МАТЕРИАЛА И ИЗНОСА ПРИ РАЗЛИЧНЫХ УДЕЛЬНЫХ НАГРУЗКАХ И СКОРОСТЯХ

Шернаев А.Н.¹, Гулямов Г.¹, Лямин А.В.¹, Усенова Г.С.²

¹Ташкентский химико-технологический институт, Ташкент, Узбекистан

²Государственное унитарное предприятие «Фан ва тараккиёт» Ташкентского государственного технического университета имени Ислама Каримова, Ташкент, Узбекистан

E-mail: ugs.gulinausenova@mail.ru

Аннотация. Приведены результаты исследований зависимостей температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала на основе тала и тополя при различных удельных нагрузках и скорости $V=0,628$ м/с от времени эксплуатации. Разработан экспериментальный стенд по измерению износа подшипников скольжения. По результатам проведённых исследований рекомендованы области использования разработанных подшипников скольжения для ряда узлов трения хлопкоочистительных машин, удельная нагрузка которых не превышает $P=45$ МПа при скорости 0,628 м/с и $P=20$ МПа при скорости 1,256 м/с.

Ключевые слова: подшипник скольжения, антифрикционный древесно-полимерный композиционный материал, полипропилен, полиэтилена высокой плотности, тополь, тал наполнитель, графит, сажка, экспериментальный стенд, измерение, температура подшипника, удельная нагрузка, скорость

1. Введение.

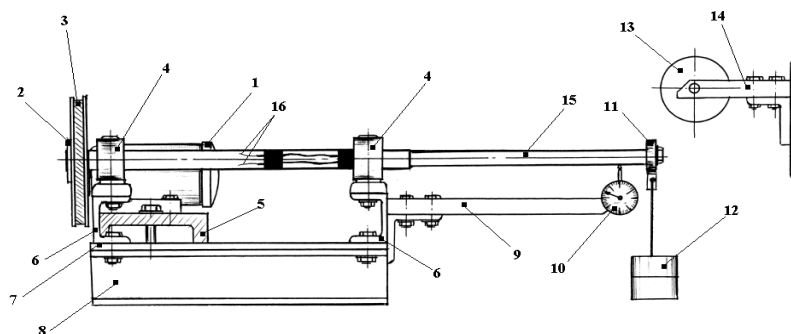
Одним из основных критериев при конструировании подшипников скольжения на основе древесины и полимеров является фактор - произведение значений удельного давления на скорость скольжения. Работоспособность подшипника скольжения, кроме того, определяется температурой трения, возникающей в зоне контакта. Температура трения влияет на величину и устойчивость коэффициента трения и на износостойкость пары, а также на возможность загорания хлопка-сырца при его взаимодействии с подшипником скольжения. Оптимальный температурный режим подшипников скольжения ограничивается значениями теплостойкости древесины и полимерных материалов. При повышенной температуре происходит потеря физико-механических свойств материала. В связи с этим, исследование температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала при различных удельных нагрузках и скоростях, возникающих в зоне контакта, является важной проблемой.

2. Объекты и методика измерения температуры.

В качестве объекта был использован подшипник скольжения из антифрикционного древесно-полимерного композиционного материала на основе полипропилена, полиэтилена высокой плотности, тополя, тала и различных наполнителей – графита и сажки, а также созданный экспериментальный стенд для измерения температуры подшипника при различных удельных нагрузках и скоростях. В процессе работы экспериментального стенда периодически производился замер температуры через каждые 10...15 минут. И если температура уже стабилизировалась, и её повышение маловероятно, то замер температуры прекращался (рис.1).

3. Результаты и их обсуждение.

Общий вид экспериментального стенда приведен на рис.1. Полученные результаты исследования приведены на рисунках 2-5. Из рисунков 2-3 (для тала) и 4-5 (для тополя) видно, что с увеличением удельной нагрузки температура подшипника увеличивается соответственно. Но после 40 минут работы при скорости $V=0,628$ м/с рост температуры прекращается, а при скорости $V=1,256$ м/с стабилизация температуры происходит значительно раньше (15 минут) для удельных нагрузок $P=8,9$ и $20,7$ МПа. При удельной нагрузке $P=44,5$ МПа рост температуры снижается, но очень медленно. Она достигает 100 и более °С, при которых работоспособность подшипника резко ухудшается.



1-электродвигатель ($n=1400$ об/мин; $N=0,25$ кВт); 2- шкив $\varnothing 150$; 3-ремень; 4-корпус подшипника; 5-стойка для электродвигателя; 6- стойка для корпуса подшипника; 7-установочная плита; 8-основание стенда (швеллер №12); 9- индикаторная стойка; 10-индикатор; 11-обечайка подшипника №205; 12-система грузов; 13-блок; 14-стойка блока; 15-вал $\varnothing 30$; 16-хромель-капельная термопара

Рисунок 1. – Общий вид экспериментального стенда

График температур для скорости $V=0,628$ м/сек

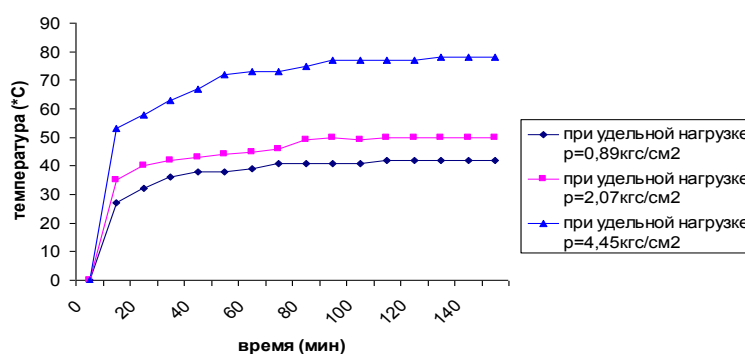


Рисунок 2. – Зависимость температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала на основе тала при различных удельных нагрузках и скорости $V=0,628$ м/с от времени эксплуатации

Для измерения величины линейного износа ПС на диаметр был использован индикатор.

Принцип измерения величины износа следующий:

- устанавливаем индикатор на индикаторную стойку, находящуюся под валом, на фиксированном расстоянии от корпуса ПС;
- нагружаем вал определённой постоянной нагрузкой равной 4кг;
- выставляем стрелку индикатора на ноль;
- осторожно разгружаем вал и перекидываем тот же самый груз ($m=4$ кг) через блок , после фиксируем положение стрелки индикатора на циферблате.

График температур при скорости $V=1,256$ м/сек

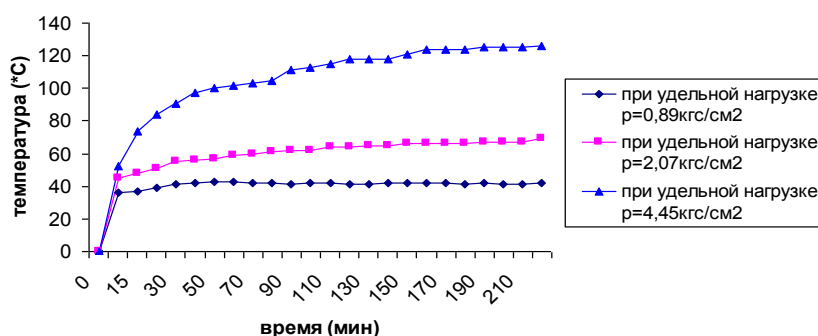


Рисунок 3. – Зависимость температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала основе тала при различных удельных нагрузках и скорости $V=1,256$ м/с от времени эксплуатации

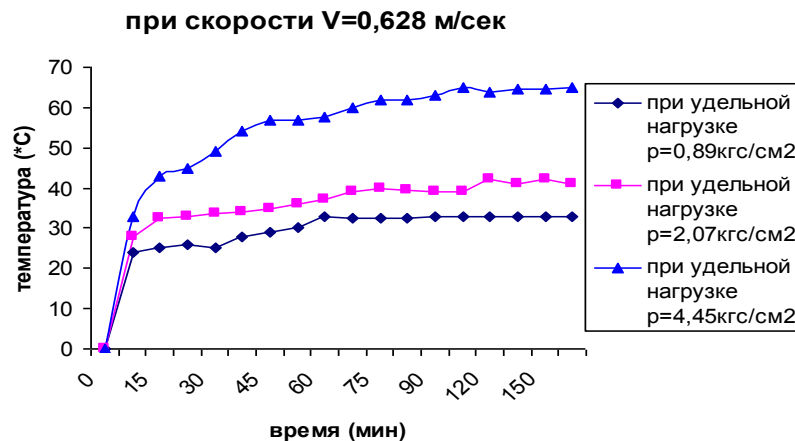


Рисунок 4. – Зависимость температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала основе тополя при различных удельных нагрузках и скорости $V=0,628 \text{ м/с}$ от времени эксплуатации

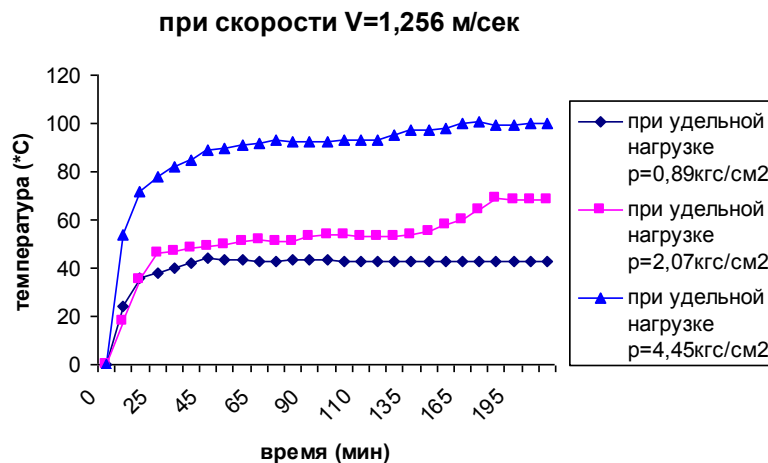


Рисунок 5. – Зависимость температуры подшипника скольжения из антифрикционного древесно-полимерного композиционного материала основе тополя при различных удельных нагрузках и скорости $V=1,256 \text{ м/с}$ от времени эксплуатации

Показания на индикаторе говорят об общей деформации системы. Для того, чтобы получить истинную величину износа на диаметр ПС (мм), необходимо из последних показаний деформации системы вычесть предыдущие показания и полученное число умножить на коэффициент $K=0,425$, который является следствием переноса зоны измерения от места непосредственного контакта вала с ПС к концу вылета рабочего вала.

Величина износа определяется из следующего соотношения:

$$И = (\Delta i - \Delta i-1) \cdot K,$$

где И – износ (мм);

Δi , $\Delta i-1$ – последующая и предыдущая деформация системы, мм;

$K = 0,425$ – коэффициент отдаления зоны замера показаний.

Для достоверности полученных измерений замеры деформации системы Δi проводились по 10 раз, после чего выводилось среднее значение Δi_{cp} (мм) по следующей формуле: $\Delta i_{cp} = \sum \Delta i / 10$, $n=10$, $i=1$.

Все данные по износу древесного подшипника периодически фиксировались и заносились в таблицу. Ниже приведены таблицы 1 и 2 с экспериментальными данными по износу для древесной втулки из тала.

Таблица 1.

**Экспериментальные данные по износу для ПС на основе тала
при скорости вращения вала $V=0,628$ м/с**

Подвешиваемый груз, кг	Удельная нагрузка, МПа	Общее время работы, ч	Износ внутреннего диаметра ПС за истекший период времени, мм
Без груза	0,89	6,0	0,029
m = 3,25	2,07	12,0	0,045
m = 9,75	4,45	12,0	0,029

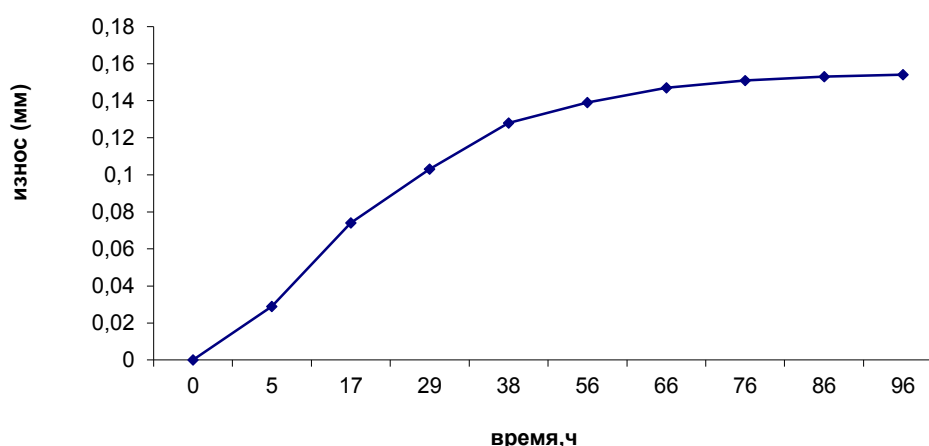
Таблица 2.

**Экспериментальные данные по износу для ПС на основе тала
при скорости вращения вала $V=1,256$ м/с**

Подвешиваемый груз, кг	Удельная нагрузка, МПа	Общее время работы, ч	Износ внутреннего диаметра ПС за истекший период времени, мм
Без груза	8,9	9,0	0,025
m = 3,25	20,7	9,0	0,045
m = 9,75	44,5	9,0	0,011

Из таблиц 1-2 видно, что более высокие значения износа при малых удельных нагрузках вызвано тем, что в этот период идёт процесс приработки подшипника. На рис.6-7 представлены графики износа по времени. Из этих графиков видно, что процесс приработки занимает 60 часов. Далее были проведены дополнительные испытания для контроля стабилизационного состояния процесса изнашивания. Этот же образец подшипника испытывался при удельной нагрузке 20,7 МПа в течении 10 часов. При этом износ составил 0,008мм при скорости вращения вала $V=0,628$ м/с, и 0,018 мм при скорости $V=1,256$ м/с. Эта область износа называется областью нормального износа.

Эксперимент проводился без дополнительного смазывания подшипника. В процессе работы экспериментальной установки подшипник частично зарекомендовал себя как самосмазывающийся, т.к. полимерная композиция сама выделялась из пор древесной втулки, и тем самым происходил эффект самосмазывания в зоне трения. Эта смесь полимерной композиции и образовала тот самый гляцевый уплотнённый слой на внутренней поверхности ПС. Но при больших нагрузках на подшипник и больших скоростях вращения вала, а также в процессе длительной работы подшипника без дополнительной смазки (около 60 часов), полимерная композиция истощается, и древесная втулка начинает высыхать и трескаться. В связи с этим, при аналогичных условиях работы подшипника, рекомендована периодическая подача смазочного материала в зону трения.

Рисунок 6. – Графики износа ПС при скорости $V=0,628$ м/с

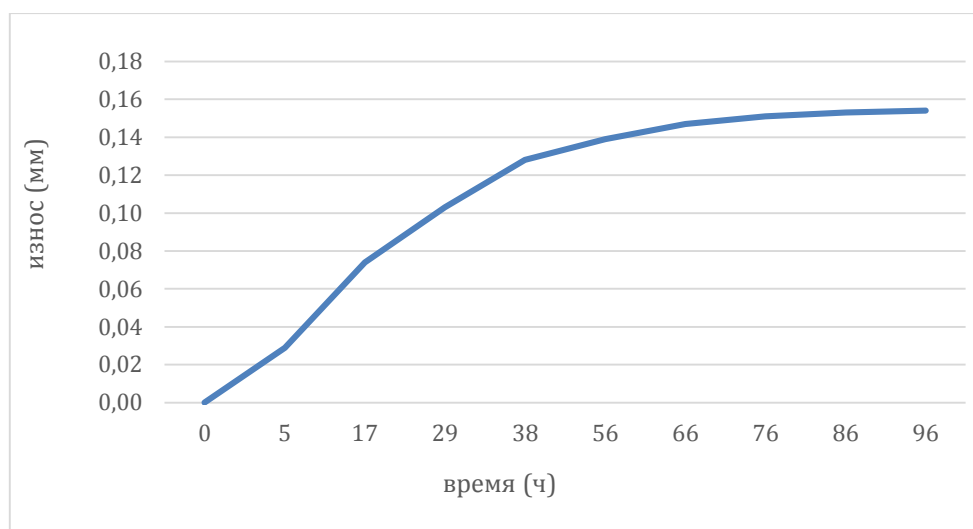


Рисунок 7. – Графики износа ПС при скорости $V=1,256$ м/с

По результатам проведённых исследований можно рекомендовать область использования разработанных подшипников скольжения для ряда узлов трения хлопкоочистительных машин, удельная нагрузка которых не превышает $P=45$ МПа при скорости $0,628$ м/с и $P=20$ МПа при скорости $1,256$ м/с. Работа подшипника эффективна при периодическом его смазывании через каждый 1 месяц работы.

Технология получения антифрикционных древесно - полимерных композиционных материалов на основе полиэтилена и полипропилена. В последние годы потребности народного хозяйства Узбекистана в конструкционных материалах для производства различных типов подшипников скольжения растут быстрыми темпами. Поэтому создание наукоемких технологий по разработке конструкционных и композиционных полимерных материалов из местного сырья, исключающих импорт подшипников качения, имеет существенное значение для экономики республики.

В качестве природного сырья может быть использована местная древесина (тополь и тал) и её отходы в деревообрабатывающей промышленности.

Древесина имеет характерное строение - высокопрочные волокна целлюлозы связаны с лигнином в жесткую и прочную капиллярно-пористую систему. Это определяет широкие возможности существенного улучшения свойств древесины путем направленного её армирования и наполнения с последующей специальной физико-химической обработкой и рационального сочетания древесины с другими материалами, в частности, с полимерами и ингредиентами, полученными на основе местных минеральных и вторичных сырьевых ресурсов. При этом представляется возможным создать большую группу дешевых недефицитных композиционных материалов с высокими конструктивными и антифрикционными свойствами, способных длительно работать в запыленных условиях Узбекистана и, таким образом, во многих случаях успешно заменить металлы в узлах трения машин и механизмов. При создании таких материалов и технологии их получения, а также изготовления из них изделий для машиностроительной промышленности, могут использоваться различные отходы производств при одновременном решении ряда вопросов, связанных с охраной биосферы и окружающей среды и улучшением условий труда работников различных отраслей промышленности, в том числе и хлопкоочистительной промышленности.

Наиболее приемлемым древесным материалом в качестве основы композиционного материала для подшипников скольжения, работающих в условиях Узбекистана, являются тополь и тал.

Известно, что в основе всех технологий производства древесно- полимерного композиционного материала на основе тополя и тала лежит подготовка сырой древесины к прессованию путем приданию ей пластичности. Учитывая, что с повышением температуры и влажности древесины её составные части – лигнин и гемицеллюлоза значительно

размягчаются и становятся менее вязкими, вследствие чего сопротивление сжатию (уплотнению) падает. Поэтому первоначально в технологии вводилась операция предварительной термообработки паром при низких температурах.

С целью повышения механических свойств, её водо- и влагостойкости древесины и для получения заготовок подшипников скольжения из тополя и тала, которые обладали бы свойствами самосмазывания, они пропитываются жидким минеральным маслом. При этом древесина сначала прогревается в горячей ванне до 95-115°C, находящийся в капиллярно-пористой системе, воздух расширяется и частично выходит наружу. Затем древесина помещается в ванну с холодным пропиточным составом, оставшийся при этом воздух уменьшается в объеме, создавая внутри неё вакуум, который заполняется модификатором. Мелкие древесины пропитываются в течение 60 мин в горячей ванне и переносятся на 60 мин в холодную. Этим способом хорошо пропитывается древесина влажностью до 10 %.

На основе проведенных исследований и сформулированных требований, предъявляемым к материалам подшипников скольжения, разработана технологическая схема пропитки древесины-тополя и тала машинным маслом и полимерной композицией. Она состоит из следующих этапов:

- пропитка теплым машинным маслом, подогретым до $t = 40-60^{\circ}\text{C}$ с выдержкой в печи до 24-48 часов. Затем температура масла доводилась до $t = 110-120^{\circ}\text{C}$, при которой происходила частичная проварка древесины в течение 1-2 часов. После этого идет охлаждение печи в течение 12-15 часов;

- загрузка пропитанной маслом древесины в ванну с горячей полимерной композицией, подогретой до $t = 140-160^{\circ}\text{C}$, с выдержкой при этой температуре в течение 1,5-2,0 часов, с последующим охлаждением;

- загрузка горячих заготовок в холодные пресс-формы и прессование при удельной нагрузке 15,0-20,0 МПа. Спрессованные заготовки выдерживались в пресс-формах под универсальным прессом 5-10 мин для охлаждения их до температуры $t = 40-50^{\circ}\text{C}$;

- нормализация (вылеживание) спрессованной древесины в окружающей воздушной среде или в ванне с холодным обезвоженным маслом в течение 24-48 часов.

Такая технология пропитки древесины с торца под давлением обеспечивает относительную легкость продвижения жидкости вдоль волокон древесины и вытеснение из сосудов воды и воздуха, высокую скорость наполнения сырой древесины. Она дает возможность изменять в заданном направлении структуру древесного вещества путем продавливания заготовки через канал переменного сечения при одновременной подаче потока нагретого модификатора в её торец под действием высокого давления. Проведенные исследования позволили разработать технологию получения композиционных материалов на основе древесины и модифицированных полимеров и технологию изготовления из них подшипников скольжения взамен подшипников качения самых различных типоразмеров, используемых в машинах и механизмах комплекса оборудования хлопкоочистительных заводов. Такая технология обеспечивает высокопроизводительное изготовление подшипников скольжения путем формования их на специальных пресс-формах. Учитывая, что антифрикционный древесно-полимерный композиционный материал (АДПКМ) наиболее эффективно использовать в тихоходных материалоемких узлах трения, работающих в режиме самосмазки, для промышленного внедрения конструкции подшипников скольжения должны быть просты в изготовлении, обеспечивать технологичность монтажа и демонтажа без изменений основных существующих размеров узлов и способствовать повышению надежности и долговечности работы машин и механизмов.

4. Выводы.

Таким образом по результатам проведённых исследований рекомендованы области использования разработанных подшипников скольжения для ряда узлов трения хлопкоочистительных машин, удельная нагрузка к $P=45$ МПа при скорости 0,628 м/с и $P=20$ МПа при скорости 1,256 м/с.

Разработана технологии получения антифрикционных композиционных материалов на основе древесины и модифицированных полимеров, полученных из местных минеральных и вторичных сырьевых ресурсов, и технологии изготовления из них подшипников скольжения для комплекса оборудования хлопкоочистительных заводов позволит повысить вдвое ресурс их работы, снизить уровень шума, сократить расход смазочных материалов и трудоемкость смазочных работ.

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ЭКСПЕРИМЕНТАЛЬНОЕ ИССЛЕДОВАНИЕ ДВИЖЕНИЯ МАЛОГО ГРУЗОВОГО АВТОМОБИЛЯ В УСЛОВИЯХ НЕРОВНЫХ ДОРОГ

Шодиев Ж.Г.¹, Бабашев К.А.²

¹ Ташкентский государственный технический университет имени Ислама Каримова, Ташкент, Узбекистан

² Узбекское агентство по техническому регулированию, Ташкент, Узбекистан

E-mail: javohir_1881@mail.ru

Аннотация. В статье представлен анализ результатов эксперимента по определению горизонтальных и вертикальных виброускорений поддрессоренной и неподдрессоренной массы широко используемого при перевозке мелкой сельскохозяйственной продукции автомобиля Labo класса N1, оснащенного устройством для исследования параметров вибрации, при его движении по специальной платформе с искусственной дорожной неровностью. Описан порядок статистической обработки представленных результатов эксперимента с помощью программы Microsoft Excel.

Ключевые слова: система подвески, акселерометр, искусственные неровности дороги, виброускорение, неподдрессоренная масса, поддрессоренная масса.

1. Введение.

Постановление Президента Республики Узбекистан ПҚ-115 от 5 марта 2024 года «О мерах по поддержке промышленного производства продукции в хозяйствах и использованию новых резервов в усадьбах и фермерских хозяйствах» Увеличился спрос на грузовики Labo класса N1. Причина в том, что для каждого хозяйства и фермера, выращивающего сельскохозяйственную продукцию и занимающегося ее глубокой переработкой, задача доставки выращенной им продукции на внутренние рынки является одним из актуальных вопросов.

Также в постановлении Президента Республики Узбекистан от 18 июля 2019 года «О дополнительных мерах по скорому развитию автомобильной промышленности Республики Узбекистан» предусмотрено внедрение методов управления, а также внедрение конкурентоспособного производства на внутреннем и внешнем рынках. Пунктом 3 данного решения предусмотрено развитие производства запасных частей и комплектующих в научно-исследовательских и опытно-конструкторских приоритетных сегментах.

Это, в свою очередь, требует совершенствования ходовых частей автомобиля Labo, производимого в нашей стране, с учетом действующих на них сил, дорожных и

эксплуатационных условий. Исследования показывают, что 80% поломок автомобилей, выпускаемых в нашей стране, происходят из-за выхода из строя ходовой части [1].

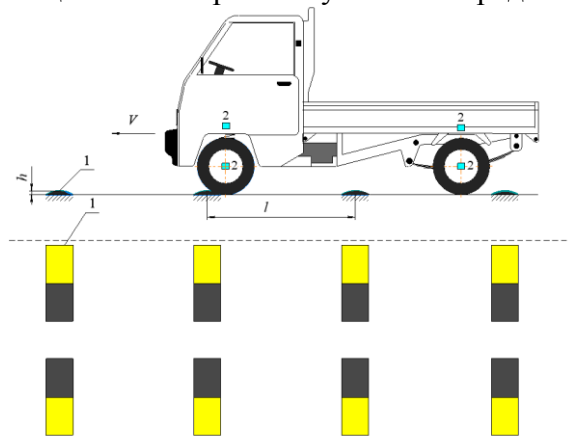
При движении автомобиля Labo с грузом по неровным сельскохозяйственным полям система подвески подвергается чрезмерным силам реакции. А поскольку этот автомобиль предназначен для езды по ровным дорогам, это становится причиной преждевременного выхода из строя системы подвески.

Для повышения подвижности автомобиля по неровным сельскохозяйственным дорогам большое практическое значение имеет проведение научных исследований их основных параметров по локализации элементов системы подвески. Ведь при проектировании каждого объекта важно провести исследования с учетом условий эксплуатации.

На основании вышеизложенного были проведены исследовательские работы по подбору параметров системы подвески, подходящих к условиям эксплуатации автомобиля [2,3]. В рамках исследования была разработана математическая модель движения автомобиля по неровным дорогам [4] и проведены экспериментальные исследования по проверке ее адекватности.

2. Материалы, методы и объект исследования.

Для экспериментальных исследований был подготовлен специальный полигон на основе параметров, учтенных в математической модели. В данном случае искусственные неровности дороги ИДН-400 были размещены на выбранном участке в 4 ряда с интервалом 1,38 м (рис. 1).



1- искусственная неровность; 2- акселерометр; V -скорость автомобиля; h -высота искусственной неровности; l - расстояние между искусственными неровностями.

Рисунок 1. – Искусственное поле неровностей дорог для экспериментальных исследований

Одним из важных пунктов эксперимента является обоснование параметров искусственной неровности дороги. Эти параметры выбираются на основании информации, приведенной в пункте 4.3.4 международного стандарта ГОСТ 32964-2014 «Требования к искусственным неровностям дорог, методы контроля». Согласно ему высота неровностей составляет $0,05 \div 0,07$ м, а ширина – 0,5 м. Расстояние между препятствиями выбирается в зависимости от скорости автомобиля. Например, если автомобиль должен двигаться со скоростью 5 км/ч (1,38 м/с), то расстояние между неровностями может составлять 1,38 м.

3. Постановка задачи.

Экспериментальное исследование проводилось в следующем порядке:

1. Датчики акселерометра марки GY-521 (MPU 6050) применяются на оси колес автомобиля (неподдрессоренная масса) и на части кузова, находящейся над колесами (поддрессоренная масса), с целью одновременной регистрации вибраций, возникающих в горизонтальном и вертикальном направлениях при проезде автомобиля по неровностям. Общее количество датчиков - 8, причем первый из 2-х датчиков, подключенных к одному контроллеру, был установлен на поддрессоренной массе автомобиля (кузов - К), а второй - на неподдрессоренной массе (колесо-Г). Номера датчиков и места установки указаны в Таблице 1 соответственно.

Таблица 1.

Количество пар датчиков и их места установки

Количество пар датчиков (контроллера)	1		2		3		4	
Место установки пар датчиков (контроллера).	Передний лево /право		Передний лево /право		Задний лево /право		Задний лево /право	
Номеров датчики	1.1	1.2	1.1	1.2	1.1	1.2	1.1	1.2
Место установки датчиков	Г	К	Г	К	Г	К	Г	К

2. Все датчики подключаются к компьютеру в салоне автомобиля с помощью специальной программы (предназначенной для приема сигналов и обработки данных) [7,8].



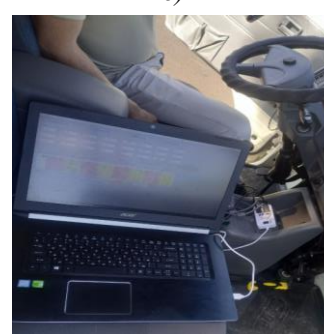
а)



б)



с)



д)

а- подготовка места искусственных неровностей дороги; б- датчик акселерометра марки GY-521 (MPU6050); с, д - оснащение испытуемого автомобиля прибором исследования параметров вибрации.

Рисунок 2. – Процесс экспериментальных исследований подвесок автомобилей

В ходе эксперимента автомобиль двигался со скоростью 5 км/ч (1,38 м/с) в направлении, перпендикулярном искусственной неровности дороги. В этом случае при приближении автомобиля к неровности компьютером активируется устройство, оснащенное акселерометром GY-521 (MPU 6050).



Рисунок 3. – Процесс тестирования

3. После того, как задние колеса автомобиля проезжают последнюю неровность, автомобиль останавливается и записанные результаты сохраняются.

4. Таким образом опыт повторяют не менее 5-6 раз.

5. Полученные результаты представлены в таблице 2.

4. Результаты.

Виброускорения автомобиля в течении 1 секунды при проезде неровностей дороги регистрировались отдельно для автомобиля и четырех колес. Таблица результатов эксперимента представлена для одного колеса.

Таблица 2.

Результаты экспериментальных исследований

Горизонтальное виброускорение переднего колеса автомобиля (неподдрессоренной массы)										
T, с	x1	x2	x3	x4	x5	x6	x7	x8	x9	x10...xn
0	0,07	0,07	0,07	-0,42	-0,31	-0,46	-0,30	-0,38	-0,37	-0,20
0,1	1,69	1,80	3,92	2,71	1,14	1,09	1,62	2,35	2,17	1,59
0,2	1,87	1,08	0,93	0,97	1,32	1,01	0,97	1,93	1,61	1,49
0,3	1,34	1,13	2,94	0,64	0,67	1,27	2,99	1,34	2,22	3,52
0,4	3,88	2,62	1,00	1,25	0,43	2,05	1,20	0,92	4,92	1,01
0,5	1,02	1,13	2,47	1,21	0,67	0,57	0,68	0,04	0,39	0,50
0,6	2,19	0,75	0,78	1,17	0,13	1,27	0,72	1,20	0,80	2,88
0,7	0,77	0,86	1,01	0,68	0,71	0,86	0,68	0,45	1,24	0,42
0,8	2,29	0,52	1,13	2,27	1,30	1,05	0,72	1,28	1,07	1,16
0,9	2,19	2,60	2,24	2,17	1,39	1,66	2,18	2,42	2,09	3,46
1,0	0,02	0,13	0,12	0,11	0,43	0,58	0,49	0,67	0,34	0,71

При обработке результатов эксперимента статистическая обработка проводилась по значениям виброускорений в каждом 0,1 секундном ряду. Для статистической обработки результатов использовалось программное обеспечение Microsoft Excel.

Вариационный ряд при T=0,1 секунды: 1,69; 1,80; 3,92; 2,71; 1,14; 1,09; 1,62; 2,35; 2,17; 1,59; 0,80; 0,78; 1,31; 2,72; 2,44; 1,14; 1,33; 1,02; 1,30; 0,88;.....

Среднее арифметическое значение вариационного ряда с 44 значениями виброускорений определялось по следующей формуле:

$$\bar{x} = \frac{x_1 + x_2 + x_3 + \dots + x_n}{n} = \frac{1}{n} \sum_{i=1}^n x_i \quad (1)$$

где n - номер вариационного ряда.

Среднеквадратичное отклонение:

$$\sigma = \sqrt{\frac{\sum_{i=1}^n (x_i - \bar{x})^2}{n-1}} \quad (2)$$

Коэффициент вариации:

$$V = \frac{\sigma}{\bar{x}} \cdot 100\% \quad (3)$$

Для проверки сомнительности полученных результатов используем критерий (критерий) Стьюдента. На основе расчетного значения критерия Стьюдента была определена формула:

$$t_p = \left| \frac{x_i - \bar{x}}{\sigma} \right| \quad (4)$$

где x_i — значение, которое считается подозрительным.

Для проверки подозрительности значений вариационного ряда из таблицы был взят критерий Стьюдента, степень свободы ряда составляла $v=n-1=43$, $\alpha=0,05$ и $t_T=2,08$.

Подозрительные значения проверяются по выражению (4), если рассчитанное значение мало по сравнению со значением критерия Стьюдента в таблице $t_p < t_T$, то подозрительное

значение не считается грубой ошибкой. Сомнительные значения из вариационного ряда исключаются, а для остальных строятся графики закономерности распределения.

Эмпирическую закономерность распределения проверяли на регулярность распределения значений виброускорения с целью представления их на графиках в виде многоугольника распределения. При этом были определены максимальные и минимальные значения виброускорения и определены их различия:

$$R = x_{\max} - x_{\min} \quad (5)$$

где $x_{\max}$ и $x_{\min}$ - максимальное и минимальное значения вариационного ряда.

Для определения диапазона значений использовалась следующая формула $i = \frac{R}{k}$ (6)

где $k = \sqrt{n}$ - количество групп [9,10].

5. Анализ результатов.

Для нахождения закона распределения брались среднее значение и повторяемость полученных интервалов исследуемых величин.

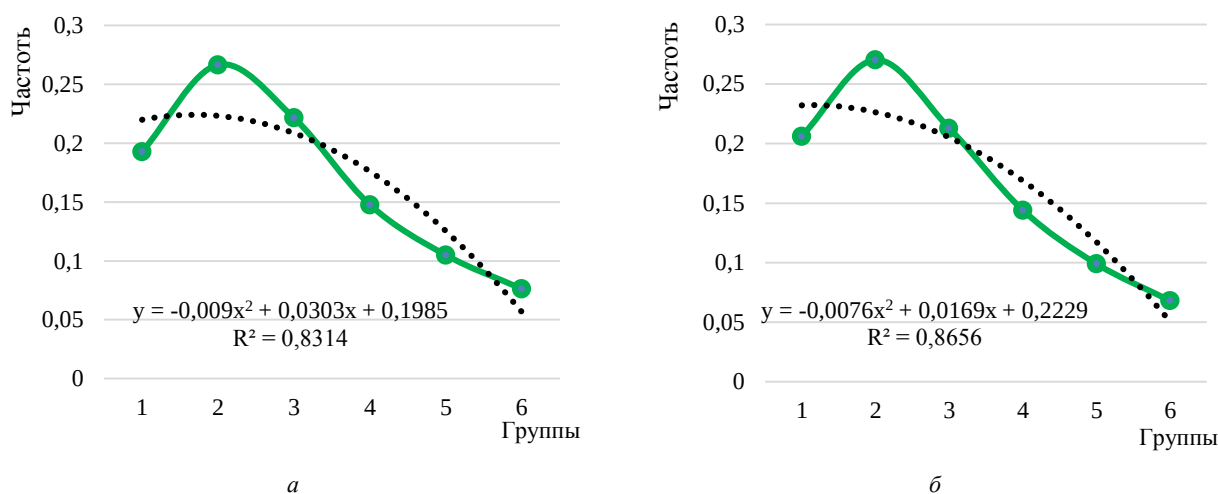


Рисунок 4. – Виброускорение передней (а) и задней (б) осей колес в горизонтальном направлении

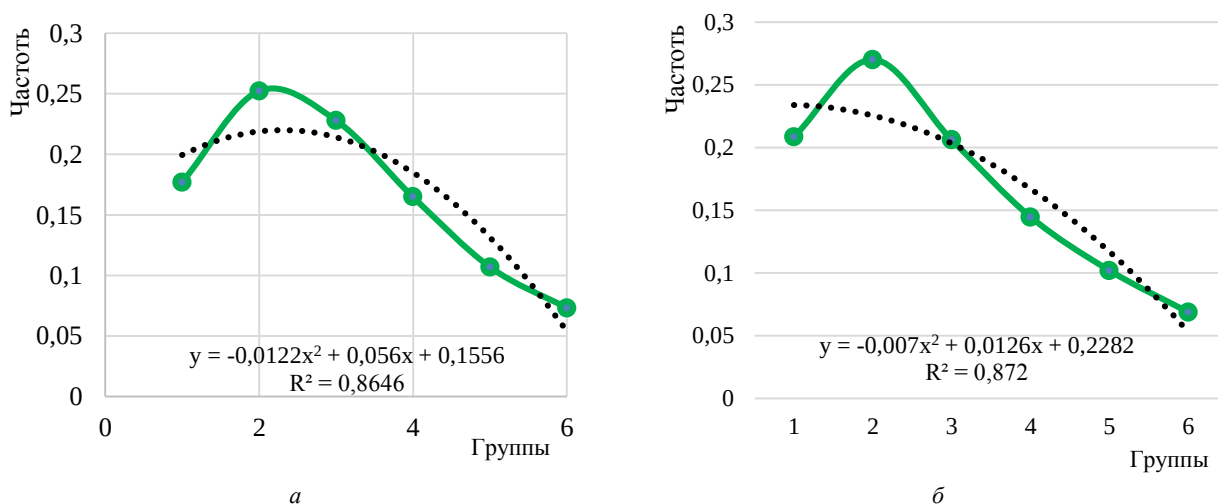


Рисунок 5. – Виброускорение передней (а) и задней (б) осей колес в вертикальном направлении

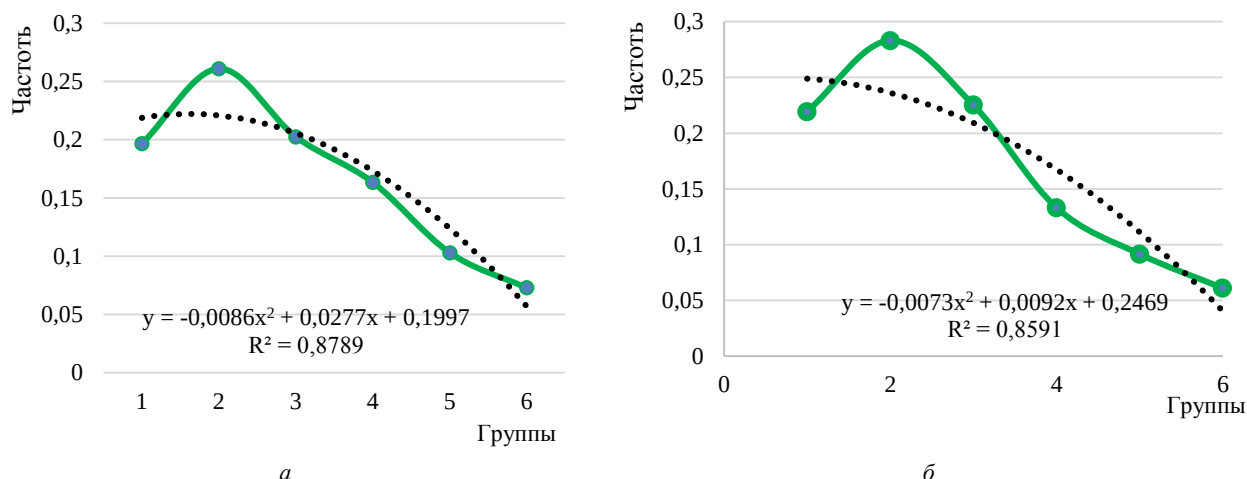


Рисунок 6. – Виброускорение автомобиля в горизонтальном (а) и вертикальном (б) направлениях

6. Заключение.

Путем построения эмпирических графиков распределения установлено, что распределение результатов эксперимента подчиняется закону распределения Вейбулла. Построение графического описания законов теоретического распределения аппроксимировалось путем сравнения его с эмпирическим графическим распределением и оценивалось по критерию совместимости. Коэффициенты соответствия для виброускорений передних и задних колес в горизонтальном направлении составили 0,83-0,8, в вертикальном направлении 0,86-0,87, а для виброускорения автомобиля в горизонтальном и вертикальном направлениях - 0,87-0,85.

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