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THE ANALYSIS OF RUNNING EFFICIENCY OF VALVE UNITS IN DIFFERENTIAL MORTAR PUMP

Методика розрахунків ефективності роботи клапанів заснована на аналізі конструкції клапанних вузлів і визначенні часу їх відкриття й закриття. Кут спрацьовування розглянутий як універсальний показник, що дозволяє порівнювати між собою насоси різних конструкцій з різними технологічними характеристиками, що й дозволяє вдосконалити їхню конструкцію шляхом зменшення цього показника.

Ключові слова: *диференціальний розчинонасос, кут спрацьовування клапанів, клапанний вузол, оптимізація конструкції клапанів.*

1. Introduction

The main requirement in the performance of construction work is a constant increase in the efficiency and quality of their conduct. The final stage of the construction of any facility is the carrying out of finishing works performed to provide the surfaces of building structures with protective and decorative properties with the help of mortars of various composition and properties. In most cases, as finishing works, plastering is used, which can't be carried out without using mortar pumps.

A wide market for meeting the needs of the construction industry provides a large range of construction machinery for mechanization of manual labor in construction, including among mortar pumps. But when creating new, more efficient designs of mortar pumps, or when considering existing structures, it is necessary to focus on energy saving, and therefore, high efficiency of their work.

The main part of the mortar pump, which determines the reliability of operation, is its valve. Therefore, assessing the efficiency of valves is one of the ways to reduce energy consumption during pump operation, which is justified the relevance of the studies.

2. The object of research and its technological audit

The object of research is a differential mortar pump, which has a horizontal pump column and is designed for pumping construction mortar mixtures of various mobility to the places of their mechanized application to the treated surfaces.

Fig. 1 is a diagram of a differential mortar pump with a horizontal arrangement of the pump column.

The mortar pump works as follows. The movement of the piston 1 is carried out by the crank mechanism 2, which is driven into rotation by an asynchronous squirrel-cage motor via a gear drive. When the differential piston moves forward (towards the crankshaft), the suction valve 3 opens and the mortar flows through the suction nozzle 4 into the working chamber. At this time, the pressure valve 6 is closed, so the mortar from the compensation chamber 7 is pushed into the delivery pipe 8. As the piston moves back, the suction valve closes, the mortar from the working chamber 5 opens through the discharge valve into the compensation chamber. One part of this mortar goes to fill the compensation chamber, expands, and the other enters the discharge pipeline. When the ratio of the cross-sectional area of the piston and its rod is 2:1, the amount of supplied mortar when the piston moves forward is the same as when it moves backwards, that is, the pump operates according to the principle of double action, providing a low-momentum feed. The piston rod of the differential piston 9 is simultaneously crosshead; it slides along the support of the crosshead 10, which senses the lateral force, thus unloading the rod shaft seals. To increase the service life of the shaft seals, a washing chamber 11 is provided. However, the feed irregularity coefficient K_H=1.57 for the differential mortar pump design is theoretical and does not take into account the feed losses. When developing and operating a mortar pump, it is important to minimize these losses.

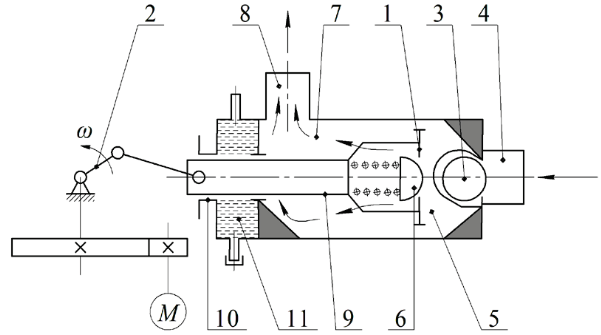

Fig. 1. Diagram of a differential mortar pump

Fig. 2 shows the theoretical curves of the time variation of the mortar supply.

Now the disadvantages of existing mortar pumps are:
– for single-piston – non-uniformity of the mortar supply in the suction and discharge strokes, which leads to the need to use additional devices to reduce the magnitude of the pulsation of the supply pressure – the compensators;

– for two-piston – increased metal consumption, overall dimensions and complexity of the design, reduces the efficiency of their use and increases the possibility of failure.

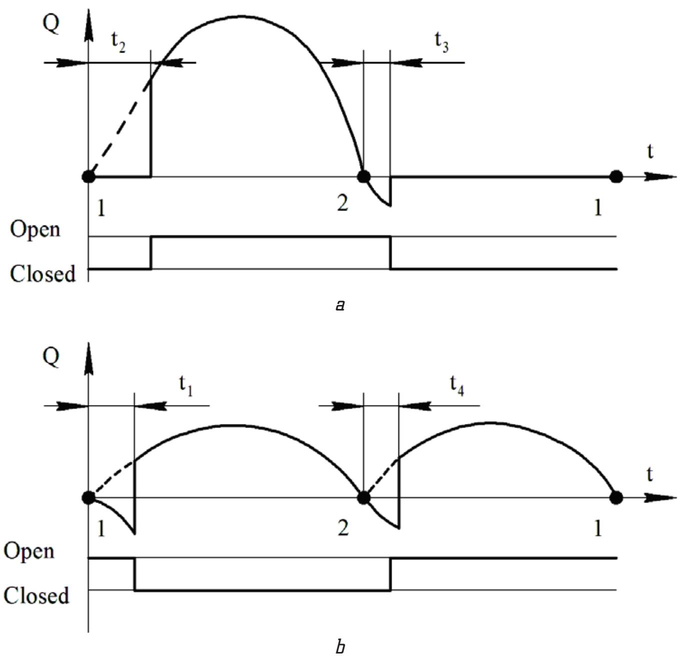

Fig. 2. Theoretical curves of the change in time of flow, which passes through: *a* – discharge nozzle; *b* – suction socket of mortar pump

Therefore, the main direction of improving the design is the combination of efficiency of two-piston mortar pumps with the simplicity of single-piston design by improving valve assemblies.

3. The aim and objectives of research

The aim of research is determination of the effect of the structural dimensions of the valve assemblies of the differential mortar pump with the horizontal location of the pump column on the numerical value of operation angle of its valves. Minimization of this parameter indicates a minimum loss of pumped mortar and, accordingly, a reduction in energy costs.

To achieve this aim it is necessary:

1. To investigate the conditions of time for opening and closing both the suction and delivery valve units.

2. To find the rational geometric dimensions of the valve units, based on which to obtain mathematical dependencies with the determination of operation angle of the valve units.

3. To check, in practical experience, the objectivity of the theoretical values obtained for the operating angle of the valve units.

4. Research of existing solutions of the problem

Investigation of the existing problem begins with studying and taking into account the rheological characteristics of mortars [1, 2].

The authors of [3] propose an effective design of a mortar pump, but no recommendations are made on the choice of geometric characteristics to improve the efficiency of its operation.

Among the attempts to solve this problem, it should be noted [4, 5].

Attempts to investigate the relationship between the geometric parameters of the working chamber of a mortar pump and the volume efficiency, the magnitude of which determines the pump efficiency of the pump, is considered by the authors in [6, 7].

Piston pumps, when pumping liquids through pipelines, create an uneven movement regime [8], which is somewhat corrected by the use of air caps. The higher the uniformity of the mortar supply through the pipelines, the less probable the mortar leakage [9], the less resistance to the movement of the mortar through the pipelines, the less the wear of the pipelines from alternating loads and abrasion, the better surface is obtained after the mortar is suctioned.

Increasing the number of floors of construction limits the use of air caps through the removal of air contained in them and the need for periodic pumping. At this time, one of the promising designs among piston pumps is the differential scheme of a mortar pump with a flow piston [10].

The differential scheme of the mortar pump [11] significantly reduces the impulsivity of the feed, providing a feed irregularity coefficient $K_H=1.57$, in comparison with the single-action mortar pump, in which $K_H=3.14$, and allows dispensing without air caps.

Thus, the results of the analysis allow to state that, in spite of a sufficiently large volume of existing studies on the mortar pump effectiveness, there are no specific methods for calculating the geometric parameters of their working chamber.

5. Methods of research

During the implementation of the research, methods of physical and mathematical modeling and similarity theory are used. To perform calculations and plotting a personal computer with a free mathematical program «SMath Studio Desktop» is used. Research results are registered in the free office package of the «LibreOffice» programs.

6. Research results

Let's analyze the expected losses, considering the operation of the differential pump column shown in Fig. 1.

After the piston arrives at the rear end position 1 (the piston is maximally removed from the crankshaft and according to Fig. 1, located in the rear right position), the piston stops and begins to move forward. There is a change in pressure in the chambers of the pump – in the compensation chamber, the pressure begins to gradually increase, and it decreases in the working chamber. From the compensation chamber, the mortar, located there, begins to flow towards the lower pressure – into the working chamber, causing the discharge pressure to sink in the discharge pipeline. The mortar, flowing, carries with it the locking element of the discharge valve, the spring of which maximizes the speed of its movement. Moving, the closing element closes the discharge valve, having spent for this time t_1. The pressure in the compensation chamber after the discharge valve closes continues to grow and the process of mortar pumping into the discharge pipeline is restored. Simultaneously, a vacuum is formed in the working chamber. The higher the suction pressure, $p_{s.min}$ should be, the greater the time t_3 until the suction valve is opened. The horizontal arrangement of the pump column contributes to a reduction in the suction pressure $p_{s.min}$.

After the piston has reached the extreme forward position of the end position 2, the piston starts to move back. There is a redistribution of pressures – the pressure drops in the compensation chamber, which causes the discharge pressure to settle in the discharge pipeline, and the pressure begins to increase in the working chamber. The mortar in the working chamber begins to be forced into the suction pipe, dragging the locking element of the suction valve behind it, which is facilitated by the gravitational force when the valve moves towards the suction nozzle. Moving, the closing element closes the suction valve, having spent for this time t_3.

Increasing the volume of the compensation chamber causes the mortar, which is in the discharge pipeline, to flow back to the compensation chamber. This occurs until the mortar in the working chamber is compressed to a pressure higher than the pressure in the compensation chamber. In this case, the discharge valve is opened, having spent for this time t_2, and the mortar is pumped into the compensation chamber and the discharge pipe.

It should be noted such features of the valves:

– the flow rate of the mortar closes the suction valve, twice as high as for the discharge valve, since with the same speed variation the cross-sectional area of the piston is twice as large as the difference between the areas of the piston and the rod. This, accordingly, causes a faster closing of the suction valve;

– the process of closing the discharge valve occurs on a movable socket, the direction of movement of which contributes to the reduction of the closing time of the valve;

– the suction valve is located in the flow of the pumped mortar and the injection is in the «shadow» of the rod, has a significant diameter (70 mm) compared to the piston diameter (100 mm). This causes a «washout» of the discharge valve and increases the closing time.

From the analysis of Fig. 2, it is possible to determine the reasons for the uniformity of the feed:

– the outflows associated with the closing time of the valves. Reducing the closing time of the valve by reducing the length of the pullback can't always give the desired result, since with a retraction length below the optimal, the hydraulic resistance of the valve increases that is deteriorated the pump operation. At the same time, the filling capacity of the working chamber deteriorates for the suction valve, which further reduces the uniformity of the feed;

– unfilledness of the working chamber. Increased value of $p_{s.min}$ impairs the filling capacity of the working chamber, increasing the opening time of the suction valve, and partially filled the working chamber causes «failure» of the discharge pressure by increasing the opening time of the discharge valve.

The analysis of losses during the operation of a differential mortar pump has made it possible to develop a design of valve units of a mortar pump that can provide the same minimum level of losses in the forward and reverse motion of a differential piston. This allowed, respectively, to ensure the mortar pumping in equal portions and reduce the pulsation level. The use of a differential mortar pump together with a rubber-fabric injection pipeline allows for an almost uniform supply of the mortar.

From the analysis of the diagrams of pressure changes at different points of the injection pipeline, it is evident that when the rubber hoses are used as a pressure pipe, the pulsations of the pumped mortar decrease. At a distance of 10 m from the pump, pulsation is practically not felt.

Let's consider the reasons that cause the opening – closing of the pump valves. The entire path of the ball movement during the valve actuation time for closing consists of two parts:

$$h = h_p + h_o, \tag{1}$$

where h – the height of the lifting balls above the socket; h_p – the path traveled by the mortar in the cylindrical cavity where the ball is located, during the valve actuation time; h_o – the path traveled by the ball together the mortar according to mortar during the valve response time.

In this case, the values h_p and h_o can be expressed as the product of the corresponding velocities v_p and v_o by the valve actuation time t:

$$\begin{aligned} h_p &= v_p \cdot t; \\ h_o &= v_o \cdot t. \end{aligned} \tag{2}$$

Substituting expressions (2) in formula (1), obtain:

$$h = v_p \cdot t + v_o \cdot t,$$

so,

$$t = \frac{h}{\left(v_p + v_o\right)}. \tag{3}$$

Taking into account expression (3), the first of formulas (2) will look as follows:

$$h_p = \frac{h \cdot v_p}{\left(v_p + v_o\right)}. \tag{4}$$

Theoretically, the entire mortar, which is in the lower part of the valve and which passed at its operating time, is expended for return leakage. On this basis, the size of the return leakage ΔV when closing the suction valve can be calculated by the equation:

$$\Delta V = \pi \frac{\pi}{4 \cdot d_v^2 \cdot h_p}, \tag{5}$$

where d_v – the diameter of the valve chamber.

Substituting the values of h_p in equation (5) by expression (4), let's obtain the final equation for calculating the return leakage of the mortar when the valve is actuated:

$$\Delta V = \frac{\pi}{\frac{4 \cdot d_v^2 \cdot h \cdot v_p}{\left(v_p + v_o\right)}}. \tag{6}$$

The speed of the mortar in the valve chamber is related to the speed of piston movement v_{ps} by the following relationship:

$$v_p = \frac{v_{ps} \cdot D_p^2}{d_v^2}, \quad (7)$$

where D_n – the piston diameter.

In equation (6), the quantities d_v and h are constructive parameters, and for a particular pump they do not change. The speed of the mortar v_p depends only on the speed of the piston and also does not depend on the properties of the pumped mortar. The main influence on the magnitude of the backward sources is provided by the speed of advance of the balls in the mortar V_0, which depends strongly on the mortar mobility.

In a mortar of high mobility, the speed V_0 of the ball immersion will be much greater than the mortar speed v_p, so the value of ΔV of return leakage will be smaller. As the mortar mobility is pumped, the speed V_0 will decrease, and the return leakage will increase. With a sufficiently thick mortar, the ball will sink into it under the action of its own weight so slowly that the velocity V_0 can be equated to zero. In this case, the height of the cylindrical volume expended on the return sources when the suction valve is closed will be equal to the height h of the lifting balls above the socket.

Let's consider in more detail the conditions of operation of the suction and discharge valves.

The suction valve (pos. 3 in Fig. 1) operates due to the pressure difference – the rarefaction of the relative atmospheric pressure, under which the receiving hopper is located. The suction is carried out through the suction pipe 4, and the horizontal arrangement of the pump working chamber positively affects the suction process. The difference in the valve actuation pressure, respectively, does not exceed 0.01 MPa, which causes the design requirements for this valve – the valve operating loss must be minimal. Such valve can't be spring-actuated. Valve operation is carried out with the speed of the pumped mortar v_p. To prevent «washing» of the balls, the valve is placed in the cage, which, moreover, being in the region of the gradient vector of the velocity gradient of the mortar, prevents the leakage of the mortar fragments. The additional relative speed V_0 is provided by the inclined valve guides.

The discharge valve 6 operates under more favorable conditions than the suction valve, since the pressure difference for its operation is practically unlimited and is due at least to the discharge pressure. By increasing the stiffness of the spring of the discharge valve, it is possible to significantly increase v_p, but this reduces the height of the valve lift and increases the rate of mortar flow through the valve slot. Such operating conditions of the valve can provoke the enthusiasm and wedging of a coarse fraction of the particles in mortar and the output of the mortar pump from an operational state. Therefore, the spring stiffness is chosen such that the height of the valve response is up to 10–12 mm. The discharge valve is located in the cavity of the differential piston, completely eliminating the «washout» of the locking element.

On the return leakage, each valve operates with a cylindrical volume of mortar equal in magnitude to the path l_n traversed by the piston for the total valve response time, and by the diameter – the piston diameter. From this volume, the annular volume of the differential empty piston housing D_{kp} is eliminated. Then equation (7) will have the form:

$$v_p = \frac{v_{ps} \cdot \left(D_n^2 - D_{kp}\right)}{d_v^2}. \quad (8)$$

Substituting the obtained value of the mortar speed into the formula (3), it is possible, taking into account the individual design dimensions and parameters of any mortar pump, determine the closing rate t_1, t_2, t_3 or t_4 of any valve in each of its operation cycles in accordance with Fig. 2. It is obvious that the outflows during the pump operation are determined by the total closing time of the successive suction and discharge valves, i. e.:

$$t_\Sigma = t_1 + t_2. \quad (9)$$

However, to compare the efficiency of the mortar pump, or to compare the leakage values, it is incorrect to compare neither valve closures nor leakage volumes. There is a parameter characterizing the efficiency of the valves, as the angle of their operation γ_v, or, in other words, the angle to which the crankshaft rotates during their closing. Comparing the operation angle of the valves, it is possible to obtain an estimate of the efficiency of the mortar pump, various in terms of design and technological parameters. To determine the valve angles γ_v (°), using the relations (3), (7) and (8) and knowing the response time of the valves t_Σ (c), let's use the dependence:

$$\gamma_v = 60 \cdot n_d t_\Sigma, \quad (10)$$

where n_d – frequency of double strokes of the piston, min^{-1}.

From the calculations performed on the design and technological parameters for the mortar pump, shown in Fig. 1, the following values are obtained:

t_1 – closing time of the discharge valve; $t_1 = 0.0022$ s;

t_2 – opening time of the suction valve, $t_2 = 0.0024$ s; $\gamma_v = 16.8°$.

The diagrams of the operation of this differential mortar pump are recorded when pumping the same lime-sand mortar with a mobility of 8 cm, which are shown in Fig. 3.

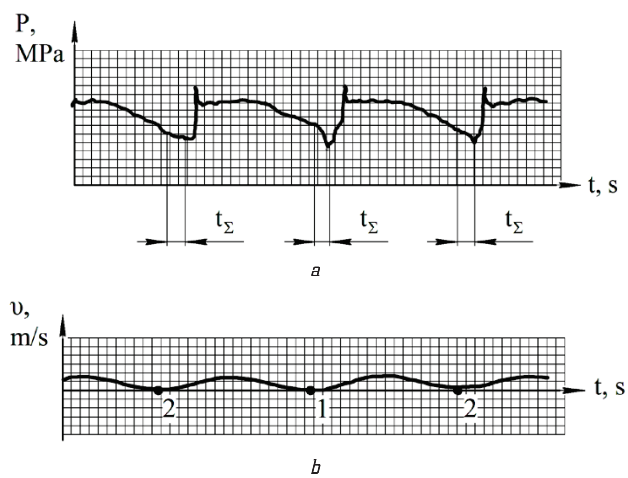

Fig. 3. Diagrams of the differential mortar operation: *a* – the pressure change; *b* – the speed change, where: 1 – the point of the rear end position of the piston, 2 – the point of the forward end position of the piston

To determine the response time of the valves, a diagram of the pressure change in the discharge nozzle is given in synchronism with the diagram of the change in the speed of piston movement. In the speed change diagram, it is easy to track the extreme forward and rear points in which the speed is zero, or, accordingly, the moment when the valves are opened or closed. The length between cycles, according to the selected scale, is 35 mm and corresponds to 180 degrees of rotation of the crankshaft of the pump. The completion of the valve response time is defined as the pressure set point on the pressure change diagram. In the adopted scale, the time interval is 3.8 mm. In terms of degrees, this is 17.3 degrees, which confirms the theoretical calculations.

7. SWOT analysis of research results

Strengths. Among the strengths of this research is the determination of the actual angles of operation of valve assemblies, based on the diagram of the concrete design of a mortar pump with a horizontal arrangement of the pump column. For example, for a lime-sand mortar with a mobility of 8 cm, the response time of the valves in both the forward and reverse motion of the differential piston is theoretically calculated to be 16.8°. The practical values obtained for the operating angle under the same conditions are 17.3°. The coincidence of the data confirms the effectiveness of the carried out analysis to minimize the magnitude of harmful return leakage when valve assemblies operate. This ensures the mortar pumping in equal portions in both the suction and discharge cycles and, correspondingly, a decrease in the pulsation level of the supply pressure. Mortar pumping with a reduced level of impulsivity causes a reduction in energy costs for the transportation process.

Weaknesses. The weak side of this research is so far an insignificant amount of the accumulated material of practical values of the operation angles of valve assemblies by piston mortar pumps for various design solutions. This calls for additional theoretical calculations to compare the effectiveness of their constructive solutions among themselves.

Opportunities. To the opportunities of the subsequent researches it is necessary to carry the computer automated analysis of the operation of valve assemblies as a general or separate operation by analyzing the diagrams of the operation of a differential mortar pump with automatic indication of the numerical values of their angles. The introduction of this perspective completely excludes the inefficient operation of the mortar pump as a whole and, accordingly, the prevention of non-conditional costs.

Threats. Threats to in the introduction of conducted research in production is due to the insufficient level of computerization of construction sites, including those equipped with additional equipment for recording diagrams of the performance of a differential mortar pump.

Thus, the SWOT analysis of the research results allows to determine the direction to achieve this goal by applying a computerized automated analysis of the operation of valve assemblies to confirm the effectiveness of their operation.

8. Conclusions

1. The conditions of the time spent on the closing of the valve are investigated, which is $\gamma_v = 16.8°$ for the mortar mobility of 8 cm. With an increase in mortar mobility, the valve response angle will decrease.

2. The mathematical dependencies for determining the operation angle of valve assemblies, depending on the geometric characteristics of the mortar pump and the rheological characteristics of the mortar are obtained. Analysis of the obtained data shows that the support for the valve closing or opening time is less indicative than the valve response angle.

3. In practice, by performing the experiments (Fig. 3, *a*), the valve closing angle is established, which is 17.3 degrees.

In addition, it should be noted that a shorter valve response time can be achieved by:

– reducing the height of the ball above the valve socket with the possibility of reducing the size of the pumped fraction;
– weight reduction of the suction valve balls without changing its dimensions;
– reducing the space of the pump chambers in places where the mortar speed gradient is small or zero.

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АНАЛИЗ ЭФФЕКТИВНОСТИ РАБОТЫ КЛАПАННЫХ УЗЛОВ ДИФФЕРЕНЦИАЛЬНОГО РАСТВОРОНАСОСА

Методика расчета эффективности работы клапанов основана на анализе конструкции клапанных узлов и определении времени их открытия и закрытия. Угол срабатывания рассмотрен как универсальный показатель, позволяющий сравнивать между собой насосы разных конструкций с различными технологическими характеристиками и позволяющий усовершенствовать их конструкцию, добиваясь уменьшения этого показателя.

Ключевые слова: дифференциальный растворонасос, угол срабатывания клапанов, клапанный узел, оптимизация конструкции клапанов.

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MATHEMATICAL MODELING OF VIBRATIONAL SYSTEMS FOR TRANSVERSE GRINDING BY WHEEL PERIPHERY

Розглянуто модель вібраційної системи шліфування, яка дозволяє простежити основні закономірності руху, одержати аналітичні залежності руху шліфувальних кругів, перехід системи через проміжні резонанси, шляхи перетворення й розсіювання енергії в системі. Дана модель створює передумови для підвищення ефективності процесу шліфувальної обробки.

Ключові слова: *вібраційна система шліфування, закономірності руху, шліфувальний круг.*

1. Introduction

When cutting of metals in the system machine – tool – device – part vibrations occur. They lead to a decrease in the quality of the machined surfaces. In order to apply effective methods of quenching these vibrations, it is necessary to develop rational methods for determining their amplitude-frequency characteristics.

One of the reasons for occurrence of mechanical vibrations during grinding operations is radial runout on the grinding wheel. Radial runout occurs when the grinding wheel is first mounted on the machine in the absence of straightening and balancing. Also, the radial runout increases after a certain time of the grinding operation due to the appearance of a ripple on the working surface of the wheel [1–4].

The statement of problems of the dynamics of vibrational systems can be very diverse. In the most general formulation, these problems are reduced to investigating the motion of the vibrating machine itself and the drive system – the vibrating machine.

The relevance of the research is creation of a mathematical model that will allow to investigate the motion of the grinding system and obtain dependencies for determining the main parameters of the system.

2. The object of research and its technological audit

The object of research is the vibrational grinding system.

The vibrational grinding system is a stochastic system, so it is very difficult to determine its state at any given time. One of the problem areas of the research object is the determination of the way in which the regularities of mutual influence of the main parameters of the system's motion will be taken into account.

The question of taking into account the energy dissipation in elastic elements in the study of the vibrations of an elastic system is quite complex, since internal friction depends on a number of factors whose influence is quite complex and practically not subject to direct account. Among the many hypotheses describing dissipative forces, the Kelvin-Voigt hypothesis has recently become most widespread.

With comparative simplicity, this hypothesis with sufficient accuracy characterizes the dissipative forces arising during the deformation of elastic bonds and ensures sufficient convergence with the experimental data [5]. Therefore, in this study let's use this hypothesis to describe the forces that arise in the elastic bonds of the vibrational system.