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ADAPTIVE INFLUENCE OF PRESSURE CHANGE OF THE HYDRAULIC SYSTEM OF THE HYDRAULIC DRIVE

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The efficiency of hydrogenated machines of various technological purposes depends to a large extent on the temperature of the environment. Temperature is the main factor in the efficiency of the hydraulic drive, which changes the physical properties of the working fluid: viscosity, moisture content, gases and mechanical impurities, which determine its modulus of elasticity. Efficiency is significantly reduced both at low temperatures and at high temperatures. The high-temperature mode of the hydraulic drive reduces the technical and operational performance of hydraulic machines (construction and road, mountain, agricultural and other self-propelled machines). High temperature is one of the most important factors that intensively affect the aging of the working fluid. An increase in the temperature of the lubricant above 70° C leads to a decrease in the lubricating capacity and premature wear of the equipment, an increase in the leakage of the lubricant, a decrease in its viscosity, thermal destruction with the formation of tarry deposits on the internal surfaces of the hydraulic equipment. Intensive wear is observed due to a decrease in the lubricating capacity of the friction surfaces, a decrease volumetric efficiency of hydraulic machines, changes in the linear dimensions of heated elements, as a result, the accuracy of technological operations deteriorates and the resource of hydraulic equipment is significantly reduced. All this leads to a decrease in the parameters of productivity and reliability of machines. Hydrogenated machines operated at elevated temperatures require a number of measures to increase heat transfer and stabilize the temperature regime. This article examines the issue of determining the given modulus of elasticity of the hydraulic system of a volumetric hydraulic drive using the example of the experimental model of the TT4M-23L tracked crawler tractor and the relationship of the given modulus with pressure and temperature. An equation is calculated that allows establishing the relationship between pressure and the given modulus of elasticity of the hydraulic system.

Key words: hydraulic system, induced modulus of elasticity, volumetric hydraulic drive, temperature, pressure.

F. 10. Fig. 2. Ref. 11.

1. Problem formulation

The development of the industrial complex depends on the level of mechanization, in particular on the technical level of agricultural tractors. The structural perfection of wheeled and crawler tractors, which is the basis of attached technological equipment, determines the technical level of the range of machines. It should be noted that almost all agricultural machines (harvesting and dredger, dredgerless dredger, forest loaders, for collecting felling residues, etc.) are equipped with a volumetric hydraulic drive of attached technological equipment, and recently, a crawler drive mechanism [1-2].

The use of a volumetric hydraulic drive as a power transmission of transmissions of domestic wheeled and crawler tractors requires separate research, especially at the time of start of movement and acceleration, since overcoming the forces of inertia imposes additional loads that occur in these operating modes.

2. Analysis of last researches and publications

The study of the operating conditions of tracked machines and their modes of operation is devoted to the work of many scientists. All these studies can be divided into two directions.

The research of the first direction is the determination of quantitative and qualitative characteristics of operating conditions. Such studies include the works of G. K. Vynogorov, I. A. Skiba, V. M. Kholoyov, and G. I. Shelepaev, V. M. Shitov, in which the assessment of the soils in terms of patency and the possibility of operating tracked machines on them during year [1-2]. Quantitative and qualitative determination of the operating conditions of tracked machines of the 40 kN class in the eastern regions of the country in the works



of M. E. Minchenko [3]. As a result of these studies, average or limit values for the necessary design of tractors were obtained.

The second direction of research is establishing the dependence of the modes of operation of tracked machines on individual operating factors. In the works of Yu. L. Kovalev, V. F. Poletaikin, and G. M. Anisimov, the effect of obstacles overcome by crawler tractors on the load in the suspension elements of crawler forestry tractors was studied [4-5]. Zaichyk M.I., Makhov G.A., Muravyov A.V. established that the turn of a tracked machine is one of the most loaded modes of movement of tracked machines. E. K. Striltsev [6] studied the influence of the speed of movement and the volume of rolling bundles on the interaction of the crawler drive of the TT-4 rolling tractor with forest soils.

3. Aim of the researches

The purpose of the work is to substantiate the parameters and modes of operation of the volumetric hydraulic drive of transmissions of agricultural tractors on the example of the experimental model of the TT4M-23L tracked crawler tractor.

4. Results of the researches

The three-dimensional hydraulic drive of tracked and wheeled drives has received the widest application, therefore, in the technical literature, considerable attention has begun to be paid to it [7-8]. Many structural parameters and operational factors affect the performance of the hydraulic drive of transport vehicles, among which the elasticity of the hydraulic equipment and the working fluid should be highlighted. These values determine the dynamic characteristics of the hydraulic drive of the technological equipment of the machine, reduce the accuracy of positioning and the uniformity of the movement of vehicles, the durability of the hydraulic equipment [9-10]. In the proposed article, the relationship between the given modulus of elasticity of the system and the pressure in the hydraulic drive of the track mechanism of the crawler tractor is considered, using the example of the TT4M23L forest transport machine, fig. 1.

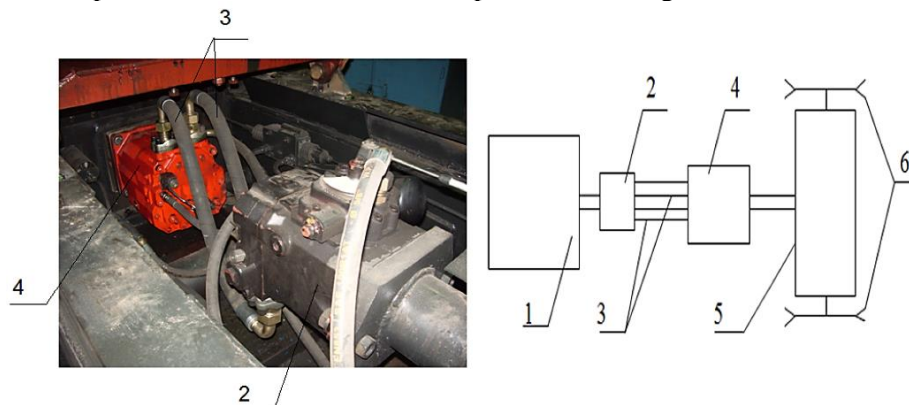


Fig. 1. Hydraulic drive of the travel mechanism:

1 – internal combustion engine; 2 – hydraulic pump HPV 135-02MI; 3 – pipelines; 4 – hydraulic motor HMV 280-02E1; 5 – mechanical gearbox; 6 – leading stars

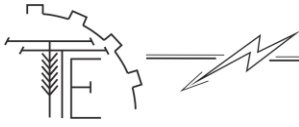
The elements of the volumetric hydraulic drive, which connects the pump and the hydraulic motor of the transmission drive, are: pipelines, high-pressure hoses and working fluid, which have elastic properties. The presence of elastic elements significantly affects the transient processes and operating modes of the hydraulic drive. When the pressure increases, the working fluid is compressed, and pipelines and hoses are deformed. This affects the flexibility of the hydraulic system in general.

Let's determine the dependence of the volume deformation of the elements on the increase in pressure. To do this, we sum up the potential energy of fluid compression U_p , the expansion of the metal pipe walls U_T , and the expansion of the sleeves U_P and get:

$$U = U_p + U_T + U_P;$$

To determine the two members of the right-hand side, we use the Zhukovsky equation:

$$U_p = \frac{\pi r_0^2 l}{2 \cdot E_p} \cdot \Delta p^2;$$



$$U_T = \frac{\pi r_T^3 l_T}{\delta_T E_T} \cdot \Delta p^2.$$

where r_T – is the inner radius of the pipeline at $\Delta p = 0$; l – length of sections of pipelines and sleeves; l_T – the length of the steel pipe; E_p – modulus of elasticity of the working fluid; E_T – modulus of elasticity of metal pipelines; δ_T – pipeline wall thickness; r_0 is the inner radius of the pipeline or sleeve in which the working fluid is at $p = 0$.

For sleeves, we apply the Lamé equation:

$$\Delta r = \frac{\Delta p r_p}{E_p} \cdot \left[\frac{r_H^2 + r_p^2}{r_H^2 - r_p^2} + \mu \right] \quad (1)$$

where r_H , r_p – outer and inner radius of the sleeve; E_p – modulus of elasticity of the sleeve; μ is Poisson's ratio; Δr_p is an increase in the inner radius with an increase in pressure by Δp .

Substitute in equation (1)

$$\frac{r_H^2 + r_p^2}{r_H^2 - r_p^2} + \mu = C_w;$$

We will get:

$$\Delta r_p = \frac{\Delta p r_p}{E_p} \cdot C_w. \quad (2)$$

As is known, the potential energy of deformation of pipeline walls can be determined in the general case [11]:

$$U = 2 \cdot \pi \cdot l \cdot \int \Delta p d(\Delta r). \quad (3)$$

Substituting the value of Δr_p from formula (2) to (3), we get:

$$U_p = \frac{2 \cdot \pi \cdot r_p^2 \cdot l_p}{E_p} \cdot C_w \int \Delta p d \cdot p.$$

After integration:

$$U_p = \frac{\pi \cdot r_p^2 \cdot l_p \cdot C_w}{E_p} \cdot \Delta p^2.$$

Then the potential energy of compression of the working fluid, expansion of the walls of metal pipes and expansion of the walls of the sleeves will be equal to:

$$U = \left[\sum \left(\frac{\pi \cdot r_T^3 \cdot l_T}{\delta_T \cdot E_T} + \frac{\pi \cdot r_T^2 \cdot l_T}{2 \cdot E_p} \right) + \sum \left(\frac{\pi \cdot r_p^2 \cdot l_p \cdot C_w}{E_p} + \frac{\pi \cdot r_p^2 \cdot l}{2 \cdot E_p} \right) \right] \cdot \Delta p^2$$

We will replace

$$\begin{aligned} \sum \left(\frac{\pi \cdot r_T^3 \cdot l_T}{\delta_T \cdot E_T} + \frac{\pi \cdot r_T^2 \cdot l_T}{2 \cdot E_p} \right) &= A \\ \sum \left(\frac{\pi \cdot r_p^2 \cdot l_p \cdot C_w}{E_p} + \frac{\pi \cdot r_p^2 \cdot l}{2 \cdot E_p} \right) &= B \end{aligned}$$

and we get:

$$U = (A + B) \cdot \Delta p^2$$

Applying the Kostigliano-Ostrogradsky theorem, according to which the derivative of the potential energy of deformation with respect to one of the independent external forces is equal to the displacement corresponding to this force, we can write:

$$\frac{\partial U}{\partial p} = \Delta Q,$$

where ΔQ – is an increase in the volume of the system under pressure.

So:

$$\frac{\partial U}{\partial p} = \Delta Q = 2 \cdot (A + B) \cdot \Delta p. \quad (4)$$

Expression $2(A+B)$ is transformed and replaced by:

$$\pi \cdot r_H^2 \cdot l_H = \Delta Q,$$

where Q_H – is the volume of this section of the pipeline, then we get:

$$2 \cdot (A + B) = \left[\sum \left(\frac{Q_T \cdot r_T}{\delta_T \cdot E_T} + \frac{Q_T}{2 \cdot E_p} \right) + \sum \left(\frac{Q_p \cdot C_w}{E_p} + \frac{Q_p}{2 \cdot E_p} \right) \right] \quad (5)$$

Note that

$$\sum Q_T + \sum Q_p = Q,$$

where Q – is the total volume of the system.



Now replace:

$$\sum Q_T = \beta_1 \cdot Q;$$

$$\sum Q_T = \beta_2 \cdot Q,$$

where $\beta_1 + \beta_2 = 1$; then expression (5) will take the form:

$$2 \cdot (A+B) = Q \cdot \left(2 \cdot \beta_1 \sum \frac{r_T}{\delta_T \cdot E_T} + \frac{1}{E_P} + \sum \frac{2 \beta_2 C_w}{E_P} \right)$$

Next, we introduce the notation:

$$2 \cdot \beta_1 \sum \frac{r_T}{\delta_T \cdot E_T} + \frac{1}{E_P} + \sum \frac{2 \beta_2 C_w}{E_P} = \frac{1}{E_T'}$$

where E_T' – is the induced modulus of elasticity of the hydraulic system, then:

$$2 (A+B) = \frac{Q}{E_T'}$$

and formula (4) can be written as follows:

$$\Delta Q = \frac{Q}{E_T'} \cdot \Delta p. \quad (6)$$

We will obtain an expression that determines the deformation of the system under the action of pressure Δp , taking into account the elasticity of the working fluid, pipelines and hoses.

Now let's consider the increase in the stroke of the pistons of the hydraulic motor X of the drive of the tractor transmission with a constant supply of working fluid, fig. 2.

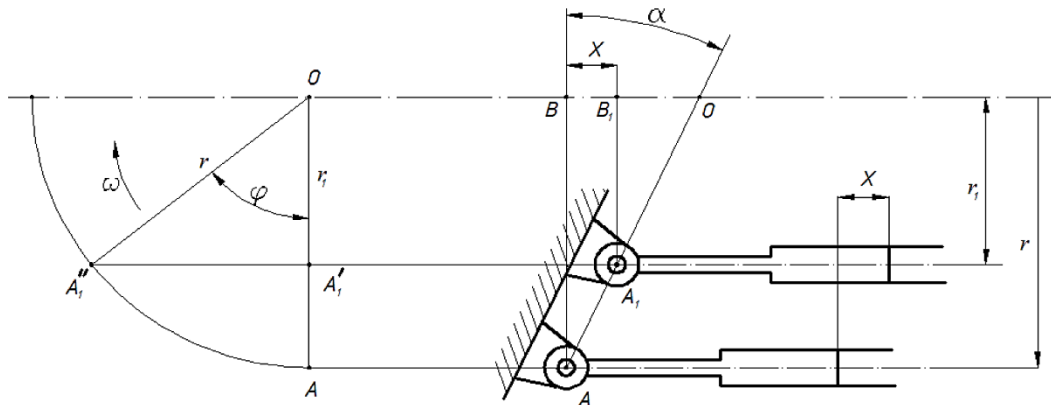


Fig. 2. Schematic diagram of the hydraulic motor

The increase in the stroke of the piston during the X rotation of the rotor from the dead position to the angle can be determined from the triangles OBA and OB1A1:

$$X = OB - OB_1;$$

$$OB = r \cdot \operatorname{tg} \alpha ;$$

$$OB_1 = r_1 \cdot \operatorname{tg} \alpha ;$$

so:

$$X = (r - r_1) \cdot \operatorname{tg} \alpha ,$$

where α – is the angle of inclination of the puck; r – is the radius of the rotor circle, where the centers of the pistons are located.

From the triangle OA1'A1'':

$$r_1 = r \cdot \cos \varphi ,$$

where $\varphi = \omega t$; t is time, then:

$$X = r \cdot \operatorname{tg} \alpha - r \cdot \cos \varphi \cdot \operatorname{tg} \alpha ;$$

$$X = r \cdot \operatorname{tg} \alpha (1 - \cos \varphi). \quad (7)$$

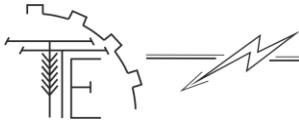
Since Z pistons participate in the operation of the hydraulic motor, equation (7) will take the form:

$$X = r \cdot \operatorname{tg} \alpha (1 - \cos Z \cdot \varphi).$$

The volume of oil supplied by the pump during time dt will be $q \cdot dt$, and the total increase of the pistons of the hydraulic motor during time dt will be equal to dx . Then the compression of the oil and the deformation of pipelines and hoses during the time dt will be:

$$X = r \cdot \operatorname{tg} \alpha (1 - \cos Z \cdot \varphi).$$

or



$$dQ = \left(q - F \cdot \frac{dx}{dt} \right) \cdot dt, \quad (8)$$

where F – is the total cross-sectional area of the hydraulic motor plungers.
Based on (6), we write:

$$dQ = \left(q - F \cdot \frac{dx}{dt} \right) \cdot dt, \quad (9)$$

Solving (8) and (9) together, we get:

$$dp = \frac{E_r}{Q} \left(q - F \cdot \frac{dx}{dt} \right) \cdot dt. \quad (10)$$

With the total displacement of the plungers on X , we have:

$$Q = Q_0 + X \cdot F,$$

where Q_0 – is the initial volume of the hydraulic system.

When integrating equation (10), we get:

$$Q = Q_0 + X \cdot F,$$

term C – is determined from the initial condition $t = 0$, $X = 0$ and $p = 0$, then $C = 0$. Therefore:

$$p = \frac{E_r}{Q} (qt - X \cdot F) \cdot dt;$$

or

$$p = \frac{I}{2 \cdot (A+B)} \cdot (qt - X \cdot F).$$

5. Conclusions

The resulting equation allows establishing the relationship between pressure and the specified modulus of elasticity of the hydraulic system. The equation was used in the design of the volumetric hydraulic drive of the transmission of the experimental model of the TT4M-23L crawler tractor and the study of dynamic processes.

The proposed approach can be used in relation to schemes of similar hydraulic drives of machines and mechanisms.

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АДАПТИВНИЙ ВПЛИВ ЗМІНИ ТИСКУ ГІДРАВЛІЧНОЇ СИСТЕМИ ГІДРООБ'ЄМНОГО ПРИВОДУ

Ефективність гідрофікованих машин різного технологічного призначення великою мірою залежить від температури довкілля. Температура є головним чинником ефективності роботи гідроприводу, яка змінює фізичні властивості робочої рідини: в'язкість, вміст вологи, газів та механічних домішок, що визначають модуль її пружності. Ефективність значно знижується як за низьких температур, так і при високих. Високотемпературний режим гідроприводу знижує техніко-експлуатаційні показники гідрофікованих машин (будівельно-дорожніх, гірських, сільськогосподарських та інших самохідних машин). Висока температура одна із найважливіших чинників, які інтенсивно впливають на старіння робочої рідини. Підвищення температури мастила вище 70° С призводить до зниження змащувальної здатності та передчасного зносу обладнання, збільшення витоків мастила, зниження його в'язкості, температурної деструкції з утворенням смолистих відкладень на внутрішніх поверхнях гідрообладнання. Спостерігається інтенсивне зношування внаслідок зниження змащувальної здатності тертьових поверхонь, зниження об'ємного ККД гідромашин, зміна лінійних розмірів елементів, що нагріваються, в результаті погіршується точність виконання технологічних операцій і значно зменшується ресурс гідрообладнання. Все це веде до зниження параметрів продуктивності та надійності машин. Гідрофіковані машини, що експлуатуються в умовах підвищених температур, вимагають проведення низки заходів щодо збільшення тепловіддачі та стабілізації температурного режиму. В даній статті розглядається питання визначення наведеного модуля пружності гідравлічної системи об'ємного гідроприводу на прикладі дослідного зразка гусеничного трельовального трактора ТТ4М-23Л та взаємозв'язок наведеного модуля з тиском та температурою. Розраховується рівняння, що дозволяє встановити взаємозв'язок тиску та наведеного модуля пружності гідравлічної системи.

Ключові слова: гідравлічна система, наведений модуль пружності, об'ємний гідропривід, температура, тиск.

Ф. 10. Рис. 2. Літ. 11.

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