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DYNAMIC CALCULATION OF THE FORCES ACTING ON THE HYDRAULIC CYLINDER

Khayrullo Baynazarov

Andijan machine-building institute, Andijan, Uzbekistan
Correspondence: avtodoroj2012@gmail.com

Abstract

Accurate calculation of the force that implements the lifting process by moving the hydraulic cylinder for the reliable and stable operation of the hydraulic drive of the trailer lifting device, also, directly related to it, it is important to choose the parameters of other elements of the hydraulic system correctly.
The article identifies and theoretically studies the main factors acting on the transient process and provides a dynamic calculation of the hydraulic cylinder in the working process of the trailers unloading device.

Keywords: transient process, hydraulic cylinder, rod, rod driving force, inertia force.

Introduction

In the production of modern, high-capacity, energy-efficient and resource-saving trailers designed to carry a variety of loads, it is important to improve the design of the hydraulic cylinder, the main body of the hydraulic system of their unloading device. During the overturning of the trailers, the hydraulic cylinder stock is affected by the load generated by the load through the platform on the one hand, and by the load coming from the hydraulic pump on the other hand. In carrying out the overturning process completely, reliably, safely and evenly, it is necessary to take into account the influence of all the dynamic factors generated during the operation of the hydraulic cylinder. The article [1] presents a mathematical model of the change in the mass of the load in the process of operating the lifting and tipping device of trailers. Working process of the tipping device consists of tipping the loads, which is considered a complex process. In this process, their mass is constantly changing (decreasing) in a uniform or uneven state until the loads are completely spilled. Also, the resistance force falling on the lifter shaft also decreases regularly. Such a process leads to the acceleration of the lifting shaft and the generation of dynamic forces in the trailer structure. Taking these forces into account is important in the perfect design of trailer structures. The article [3] theoretically sets out the transition process of the mechanical system (platform) of the trailers lifting and tipping device and their equations of motion. The article [10] presents the results of an experimental research to substantiate the basic parameters of the unloading device of the trailers.

Materials and Methods

In trailers with a carrying capacity of 4 ÷ 10 tons, taking into account that one-way, 2-stage telescopic hydraulic cylinders can fully perform the overturning process, the following calculations are performed on these hydraulic cylinders.

Kinematic analysis of the unloading device. The hydraulic cylinder (AB) is hinged to the platform (CB) on a (B) base, which is common to them, and together they form the unloading device of the trailer. Hence, the hydraulic cylinder and the platform are the main elements of the device (figure 1). The hydraulic cylinder is mounted on the frame via the base and the platform is mounted on the frame via the base C, which only rotates around their supports, so their degree of freedom is 1. The rod can move forward along its axis (x axis) relative to the hydraulic cylinder body. This means that its degree of freedom is equal to 1. The trajectory of the rod is linear with respect to the hydraulic cylinder body and curved with respect to the base. In computational work, it is assumed that its motion is linear.

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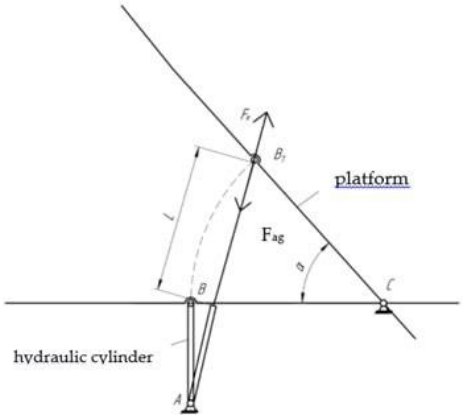


Figure 1. Kinematic scheme of the trailer unloading device

Dynamic analysis of the hydraulic cylinder of the unloading device. In structures directly connected to the hydraulic cylinder external load, for the position where the x axis of the coordinate system lies on the hydraulic cylinder axis, the differential equilibrium equation of linear forward motion of the rod can be constructed according to the Dalamber principle:

$$F_{in} = m \cdot \frac{d^2(L)}{dt^2} = F_{dr} - F_{res} \quad (1)$$

Where: m - is the sum of the masses of all moving parts in the process (m_1 - is the mass of the load, m_2 - is the sum of the masses of the body with the platform, m_3 - is the mass of the moving parts of the hydraulic cylinder, assuming that point B in the kinematic scheme is the center of gravity will be $m = m_1 + m_2 + m_3$); L - is the path of the stock; t - is the time spent on the road; F_{dr} - is the force acting on the stock; F_{res} - is the sum of all the forces resisting the stock movement.

The forces acting on the rod are determined by the lifting and lowering stages of the platform (figure. 2).

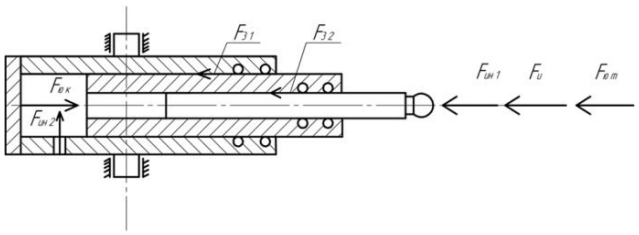


Figure 2. Scheme of dynamic calculation of the hydraulic cylinder.

Platform lifting stage. The inertial force generated at this stage of the platform can be determined as follows:

$$F_{in\ lif} = F_{in\ lif1} + F_{in2} \quad (2)$$

here: $F_{in\ lif1}$ - is the force of inertia generated by the acceleration of the masses of moving parts during the lifting of the platform, which can be defined as follows:

$$F_{in\ lif1} = (m_1 + m_2 + m_3) \cdot \frac{d^2(L)}{dt^2} \quad (3)$$

here: ϱ_o - is the density of the oil; d_x - is the diameter of the moving parts of the hydraulic cylinder; l is the path of the hydraulic cylinder steps.

The driving force of the hydraulic cylinder for the platform lifting stage is determined as follows:

$$F_{dr\ lif} = \frac{(m_1+m_2+m_3)\cdot g\cdot K_0}{K^n \cdot K_1^n} \quad (4)$$

here: g - is the acceleration of free fall of the body; α - is the angle of rotation of the platform; K - is the efficiency of the base C (sliding bearing); K_1 - is the efficiency of the base A (sliding bearing); n - is the number of supports; K_0 - is a coefficient that takes into account the mode of operation of the device, for mechanisms with variable loads and speeds of the working bodies during operation -$K_0 = 1{,}20 \ldots 1{,}5$.

The results of the theoretical analysis show that the resistance forces of the hydraulic cylinder work process depend on the following factors:

$$F_{res} = F_{p\ lif} + F_{sl} + F_{oil} \quad (5)$$

here: $F_{p\ lif}$ - the force pushing the hydraulic cylinder rod is determined by the angle of rotation of the platform (α) as follows:

$$F_{p\ lif} = \frac{(m_1+m_2+m_3)\cdot g\cdot \cos\alpha}{K^n \cdot K_1^n} \quad (6)$$

F_{sl} - is the sum of the resistance forces of the hydraulic cylinder sealing and cleaning sleeves and is defined as follows:

$$F_{sl} = K_{fr} \cdot \pi \cdot d_{sl} \cdot h_{sl} \cdot P_{hd\ s} \cdot n_{sl} \quad (7)$$

here: K_{fr} - is the coefficient of friction in the slip between the compactor and the compactor; d_{sl} - is compaction diameters of sealing and cleaning elements; h_{sl} - is compaction height of sealing and cleaning elements; $P_{hd\ s}$ - is the working oil pressure in the process; n_{sl} - is the number of sealing and cleaning elements. F_{oil} - is the internal friction force of oil and is defined as:

$$F_{oil} = 1{,}57 \cdot \mu \cdot d_x \cdot v_x \quad (8)$$

here: μ - is the coefficient of internal friction (coefficient dynamics) of the oil; d_x - diameters of oil flowing cylinders; v_x - is the oil flow rate.

Substituting the forces defined above into expression (1), we obtain the equilibrium equation for lifting the platform:

$$F_{in\ lif1} + F_{in\ 2} = F_{dr\ lif} - F_{p\ lif} - F_{sl} - F_{oil} \quad (9)$$

In the considered equations, the flow mode of the oil in the device hydraulic system is unstable (laminar-turbulent), the Reynolds number is critical, the temperature, density, viscosity of the working oil and the amount of insoluble air in it are constant.

However, the load mass (m_1) and the walkway of the stock (L) change during the process. This is because during the unloading process, the loads slide from the platform to each other or on the support surfaces, and the change in loads during these processes varies.

The change in the mass of the sliding load (m_1) during the process for its upper half is as follows:

$$m_1 = \frac{\rho \cdot В \cdot [2 \cdot А \cdot Б - А^2 \cdot tg(90° - \varphi_0 + \alpha)]}{2} \quad (10)$$

for the lower half of the load is determined as follows:

$$m_1 = \frac{0{,}5 \cdot \rho \cdot В \cdot Б^2}{tg(90° - \varphi_0 + \alpha)} \quad (11)$$

here: ϱ - is the density of the load; А, Б, В А - are respectively the height, width, length of the load; φ_0 - is the natural slope angle formed when the load is poured in a heap.

The walkway of the stock (L) is determined by the following equation:

$$L = 1{,}95 \cdot ВС \cdot \sin\frac{\alpha}{2} \quad (12)$$

here: BC - is the distance from the base axis of the platform to its center of gravity [3].

Results

Substituting the above values of inertia, driving and resistance forces acting on the hydraulic cylinder into expression (2), the system of differential equations of the hydraulic cylinder for the platform lifting stage during load lifting can be written as follows:

$$F_{ин\,к} = \begin{cases} [0{,}5 \cdot \rho \cdot В \cdot [2 \cdot А \cdot Б - А^2 \cdot tg(90° - \varphi_0 + \alpha)] + m_2 + m_3 + \rho_м \cdot 0{,}785 \cdot d_х^2 \cdot 1{,}95 \cdot ВС \cdot sin\frac{\alpha}{2}] \cdot \frac{d^2(L)}{dt^2} = \\ \left[\frac{[0{,}5 \cdot \rho \cdot В \cdot [2 \cdot А \cdot Б - А^2 \cdot tg(90° - \varphi_0 + \alpha)] + m_2 + m_3] \cdot g]}{К^n \cdot K_1^n}\right] \cdot (K_0 - cos \propto) - K_{иш} \cdot \pi \cdot d_х \cdot n_з \cdot P_б \cdot n_з - 1{,}57 \cdot \mu \cdot d_х \cdot v_х, \\ 0° < \alpha \le \varphi_0 - arctg\frac{А}{Б} \\ \frac{0{,}5 \cdot \rho \cdot В \cdot Б^2}{tg(90° - \varphi_0 + \alpha)} + m_2 + m_3 + \rho_м \cdot 0{,}785 \cdot d_х^2 \cdot 1{,}95 \cdot ВС \cdot sin\frac{\alpha}{2}] \cdot \frac{d^2(L)}{dt^2} = \\ \frac{\left[\left(\frac{0{,}5 \cdot \rho \cdot В \cdot Б^2}{tg(90° - \varphi_0 + \alpha)}\right) + m_2 + m_3\right] \cdot g}{К^n \cdot K_1^n} \cdot (K_0 - cos \propto) - K_{иш} \cdot \pi \cdot d_х \cdot n_з \cdot P_б \cdot n_з - 1{,}57 \cdot \mu \cdot d_х \cdot v_х, \quad \varphi \ge \alpha \ge \varphi_0 - arctg\frac{А}{Б} \\ m_2 + m_3 + \rho_м \cdot 0{,}785 \cdot d_х^2 \cdot 1{,}95 \cdot ВС \cdot sin\frac{\alpha}{2}] \cdot \frac{d^2(L)}{dt^2} = \frac{m_2 \cdot m_3 \cdot g}{К^n \cdot K_1^n} \cdot (K_0 - cos \propto) - K_{иш} \cdot \pi \cdot d_х \cdot n_з \cdot P_б \cdot n_з - 1{,}57 \cdot \mu \cdot d_х \cdot v_х, \\ \varphi < \alpha < 45° \end{cases}$$

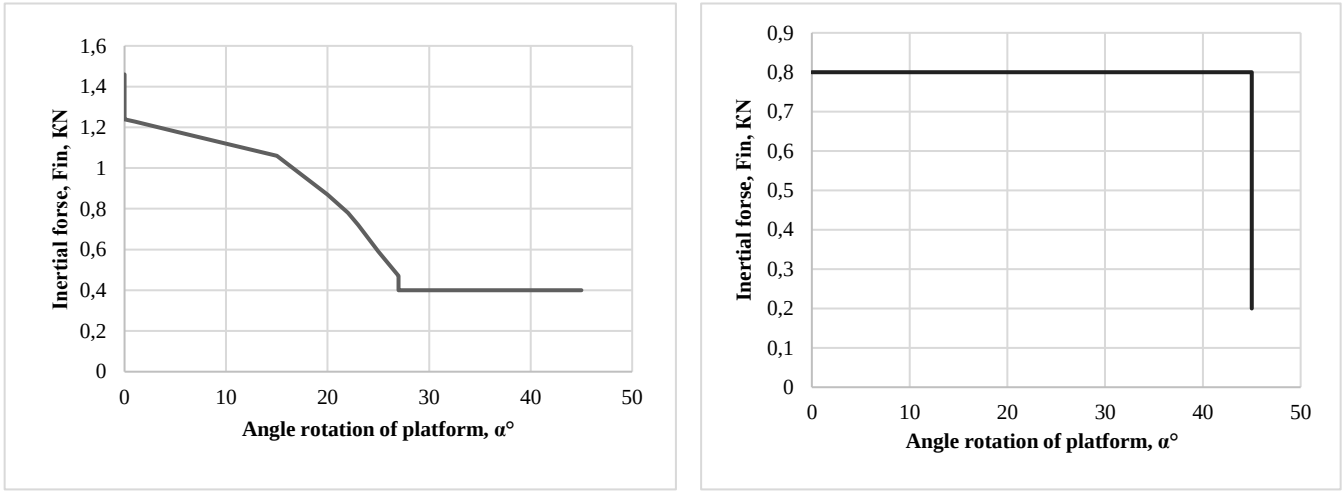


Figure 2. The lifting stage of the platform is a diagram of the change in the inertial force acting on the hydraulic cylinder with respect to the angle of rotation (α) in the example of wheat (left) and cotton (right).

The lowering stage of the platform. The inertial force generated at this stage is determined as follows:

$$F_{\text{in low}} = F_{\text{in low1}} + F_{\text{in 2}} \quad (14)$$

here: $F_{\text{in low1}}$ - is the inertial force generated by the acceleration of the masses of the moving elements during the lowering stage of the platform, defined as follows:

$$F_{\text{in low1}} = (m_2 + m_3) \cdot \frac{d^2(L)}{dt^2} \quad (15)$$

The driving force for the platform lowering stage is as follows:

$$F_{\text{dr low}} = \frac{(m_2+m_3)\cdot g}{K^n \cdot K_1^n} \quad (16)$$

The resistance force for the platform lowering stage is as follows:

$$F_{\text{res low}} = F_{\text{noz}} + F_{\text{sl}} + F_{\text{oil}} \quad (17)$$

where: F_{noz} - is the resistance force at which the working oil flows through the small hole in the nozzle, which is defined as follows:

$$F_{\text{noz}} = \frac{171557 \cdot \rho \cdot V \cdot f_{\text{oil}} \cdot d_x^2}{2 \cdot d_{\text{noz}}} \quad (18)$$

Putting the above-defined values of inertia, driving and resistance forces acting on the hydraulic cylinder in expression (3), the system of differential equations of the hydraulic cylinder during the lowering of the platform can be written as follows:

$$F_{\text{ин}} = \left(m_2 + m_3 + \rho \cdot 0{,}785 \cdot d_x^2 \cdot BC \cdot \sin\frac{\alpha}{2}\right) \cdot \frac{d^2(L)}{dt^2} = \frac{(m_2 + m_3) \cdot q}{K^n \cdot K_1^n} - \frac{171557 \cdot \rho \cdot V \cdot f_{\text{м}} \cdot d_x}{2 \cdot d_{\text{шт}}} - K_{\text{иш}} \cdot \pi \cdot d_x \cdot h_{\text{з}} \cdot P_{\text{б}} \cdot \eta_{\text{з}} - 1{,}57 \cdot \mu \cdot d_x \cdot v_x$$

here, $\alpha < \frac{\alpha_{\text{мах}}}{n_1}$ has $d_x = d_1$ and $\alpha > \frac{\alpha_{\text{мах}}}{n_1}$ has $d_x = d$ will be; n_1 - is the number of steps of the hydraulic cylinder.

Discussion

The analysis of the available technical literature, published scientific works and internet data shows that this issue has not been fully researched [1-8, 12].

Conclusions

The above system of equations is a system of differential equations, showing the change of inertia, driving and resistance forces relative to the angle of rotation of the rod during the transition, taking into account the characteristics of the load and working oil, the operating mode and design of the device and the basic parameters of the hydraulic cylinder.

They are recommended for use in design and research.

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