Controlling motion of metal bascule structures by fluid power system (exemplified by lifting of bascule bridge span)

A V Ashcheulov

Peter the Great St. Petersburg Polytechnic University, 29, Polytekhnicheskaya str., St. Petersburg, 195251, Russia

E-mail: atsheulov_av@spbstu.ru

Abstract. This article discusses the dynamics of power processes upon motion of bascule metal structures peculiar for many lifting heavy-duty mechanisms. Due to the forces of inertia, during the acceleration of the mechanism with a load, fluctuations in power parameters occur: efforts; stress; pressures that can lead to the destruction of the metal structures of machines. Due to high alternating load oscillations, the working capacity of machinery decreases about 3 times, herewith, this problem is traced for machinery in various industries. Using the motion of large-tonnage span of bascule bridge as the example, at the design stage it is proposed to apply simulation modeling in order to determine optimum law of drive control, in particular: the law of variation of flow rate in fluid power systems. The article presents the results of simulation of hydraulic lifting mechanisms of a single-wing bascule bridge. Two basic algorithms of proportional control are considered: by the form of the function in the form of a trapezoid and by the form of the cosine function. The results indicate the possibility of significantly reducing the amplitude of fluctuations in load parameters during acceleration of mechanisms and, thereby, to increase the service life of heavy machines. To achieve such results, it is recommended to change the existing algorithms for automatic control of the speed of the mechanisms of heavy machines. Despite the fact that this issue has been studied quite well in cybernetics, it still remains relevant in applied technical specialties, in the design of modern heavy machines. In project organizations, along with constructive design methods, digital, computational design methods should be actively developed.

1. Introduction

Numerous hoisting and road building machines are equipped with mechanisms with bascule metal structures and the fluid power system [1]. Observation of operation of these mechanisms has demonstrated that motion at a constant speed results in increased dynamics upon acceleration and deceleration. In the spaces of hydraulic cylinders, the pressure drops reach high amplitudes. The coefficient of dynamicity $K_{dyn}$ [2]:

$$K_{dyn} = \frac{\Delta p_{max}}{\Delta p_{stat}},$$

where $\Delta p_{max}$ is the maximum pressure drop in hydraulic cylinders upon acceleration, $\Delta p_{stat}$ is the pressure drop in hydraulic cylinders in the static state. According to the author's results it can reach 1.4 and higher. Figure 1 illustrates a program window of the metering system (monitoring system) of bascule bridge. The window shows plots of pressure variations in spaces of hydraulic cylinders and
plot of the rotation angle of the lifting mechanism of the bascule bridge. The presented transition processes are the responses to consecutive activation (deactivation at the stroke end) of two pumps in the hydraulic system: boosting and main pumps. This is the relay stepwise volumetric (machine) control, when it is possible to vary only the steps and time of activation of the second step [3]. The first pressure surge in the discharge line is caused by activation of the boosting pump, the second surge occurs after activation of the main pump. In this case, the oscillation amplitude of the second pressure surge is higher than the first one, since there appear resonant effects of oscillation superposition. However, opposite situations are possible, this depends on the time of activation of the boosting and main pumps.

These oscillations in the range of operation pressure are not dangerous for the fluid power system [4]. However, the pressures in spaces of hydraulic cylinders are equivalent to forces, thus, the pressure oscillations evidence alternating loads in mount fittings of hydraulic cylinders and metal structures, which in the case of high oscillation amplitude can influence significantly their strength [5]. The following facts were detected in practice at some bridges: in the gaps between the rotation axle and the center of counterweights under the layers of old paint, the cracks in metal structures were detected. One significant accident in the 1980-s should be mentioned, when a metal structure at one bridge was completely destroyed and all large-tonnage counterweight fell into a navigable waterway [6].

In order to prevent such accidents, the guide for bridge design includes specified coefficients of load dynamicity, which should not exceed 1.2 [7]. It is possible to achieve the specified coefficient of dynamicity with relay stepwise control by further decrease in motion speed (decrease in flow rates of boosting and main pumps of the hydraulic system). However, this is a dead end, since it would increase the time of bridge lifting, which is unallowable. For bridges of the main waterway, the predetermined lifting time is 2 min, and for the reserved waterway, it is 4 min. The time is also specified for hoisting mechanism of hydraulic facilities (gate drives). The same problem exists for other machines, where increase in performances is highly important.

Figure 1. Major specifications of fluid power of Troicky Bridge monitoring system.
2. Materials and methods

A distinctive feature of the research is the use of the author's methodology, which provides for the use of experimental data not of local bench tests of individual machine units, but of data from diagnostic systems (monitoring systems) of full-fledged machines that are directly in operation. We are talking about methods of full-scale tests, but not experimental machines, but industrial, operating models of machines. At the same time, the researcher does not need to write test methods, it is not required to install little reliable non-standard sensors, but it is necessary to study the design of monitoring systems and adapt digital information from the so-called on-board controllers, incl. and "black boxes", which are currently equipped with most critical machines. It should be noted that, first of all, information from monitoring systems is intended to ensure the safe operation of machines. But it also contains a large amount of information that can and should be used by design organizations. First of all, through this information, a feedback is established between the stage of machine operation and the stage of its design. In particular, this information reveals deficiencies in the operation of machines. The accumulation of this information allows you to form design standards and make forecasts for the durability of machine components. Using this information allows us to develop adequate digital models of complex machines.

The data of the monitoring systems installed on movable bridges allowed the author to identify and analyze the problem of the dynamics of the acceleration of the lifting mechanism considered in the article. The same data were used in simulation mathematical modeling for comparison with the calculated dependencies, for correcting the digital model, as well as for clarifying the values of "inaccurate" initial data.

Simulation mathematical modeling was carried out by traditional methods of composing and solving mixed systems of differential equations.

Solution to such sets of differential equations could be performed in general form by numerical integration using any certified application software. The author used the Model Vision proprietary software of StPPU.

3. Concept of motion

Solution to the mentioned problem should be searched at the machinery design stage. Proportional control of the pump flow rate of the fluid power system should be applied. However, it is insufficient to select an adjustable pump and a regulator type for the design of the hydraulic system, it is required to determine a mathematical equation, according to which the pump regulator should operate, it is referred to as the control law by experts in the field of automatic control. The control law is determined by solution to the optimization problem of motion control, it is obtained by mathematical simulation modelling of machinery, and this is beyond the frames of designing. Therefore, numerous Russian designing bureaus do not provide simulation modelling of their machinery due to insufficient competence in this field. It should be mentioned that the time for designing activities is restricted and sometimes additional time for complicated verifications is not stipulated. On the contrary, leading foreign companies, using CAD tools, carry out simulation modelling of hoisting mechanisms at the design stage, sometimes gaining competitive advantages in the form of smooth motion of their mechanisms without hydraulic shocks.

Simulation modelling begins with development of the mathematical model of the machine and stipulates for determination of a set of differential and algebraic equations of parameter interrelation. For instance, for a bascule bridge, it is possible to write the differential equation of motion of cylinder rod in the form of Lagrange equation of the second order. Herewith, the weight of the metal structure is reduced to the weight of movable cylinder rod, and to use equations of mechanism kinematics and equations of hydraulics, accounting for elastic properties of working fluid and the body of the hydraulic cylinder. In addition, it would be reasonable to consider for the volume of the total hydraulic system, that is, the equations of major pipelines. The equations of controlling hydraulic devices should also be considered, if they participate in control of the motion. The order of mathematical model depends on the complexity of the considered system. A general procedure of development of such
models is described in the Doctoral thesis by the author. At the second step, the solution to the set of equations is programmed. At the third step, the adequacy of predicted model is verified. If the adequacy is not high, then it should be provided by adjustment of initial data, in particular, by varying inaccurate parameters of the mathematical model. The procedure of adequacy achievement for the predicted model is described in other works by the author. Only a high degree of adequacy of the predicted model allows one to consider the results of simulation modelling as reliable.

\[
\begin{align*}
\dot{m}_{\text{red}} \cdot \dot{x}_{\text{hc}} &= P_{\text{hc}}^A \cdot S_{\text{hc}}^A - \left( F_{\text{ht}}^A + \frac{M_{\text{bs}}}{h} \right) \cdot \text{sign}V - \frac{M_L(\psi)}{h} - \frac{M_w(\psi, t) \cdot n}{h} \\
\dot{x}_{\text{hc}} &= \frac{Q_{\text{hc}}^A}{S_{\text{hc}}^A} = V \\
x_{\text{hc}} &= L_{\text{hc}} - x \\
S_{\text{hc}}^A &= S_{\text{hc}}^W = \frac{\pi}{4} \left( D_{\text{piston}}^2 - d_{\text{rod}}^2 \right) \\
D_{\text{piston}}^\text{conv} &= \sqrt{\frac{4 \cdot \sum S_{\text{hc}}^A}{\pi}} \\
\hat{P}_{\text{hc}}^A &= \frac{Q_{\text{hc}}^A - S_{\text{hc}}^A \cdot \dot{x}_{\text{hc}} - Q_{\text{hc}}^A}{K_{\text{cont}}^A} \\
K_{\text{cont}}^A &= \frac{\Delta V_{\text{A}} + S_{\text{hc}}^A \cdot \dot{x}_{\text{hc}}}{E_{\text{red}, \text{hc}}} \\
E_{\text{red}, \text{hc}} &= \frac{E_{\text{fluid}}}{1 - \frac{D_{\text{piston}}^\text{rel}}{E_{\text{hc}}}} \\
x &= \sqrt{r^2 + c^2 + 2 \cdot r \cdot c \cdot \cos(\gamma - \psi)} \\
h &= \frac{r \cdot c \cdot \sin(\gamma - \psi)}{x}
\end{align*}
\]

where \( m_{\text{red}} \) is the weight of moving parts including the metal structure reduced to the rod of the conventional hydraulic cylinder; \( x_{\text{hc}} \) is the rod stroke; \( P_{\text{hc}}^A \) is the pressure in the work space; \( S_{\text{hc}}^A \) is the surface area of the working space of the conventional hydraulic cylinder; \( L_{\text{hc}} \) is the length of the hydraulic cylinder with the extended rod; \( X \) is the current length of the hydraulic cylinder; \( O_{\text{hc}}^A \) is the flow rate of working fluid supplied to the working space; \( V \) is the rod linear speed; \( D_{\text{piston}} \) is the piston diameter; \( d_{\text{rod}} \) is the rod diameter; \( F_{\text{ht}}^h \) is the dry friction force in the hydraulic cylinder; \( M_{\text{bs}} \) is the moment of friction of mechanical units of the bascule span; \( M_L(\psi) \) is the moment of forces from positional loads (weight, icing, etc.); \( M_w(\psi, t) \) is the moment of forces of wind loads; \( n \) is the coefficient of the wind direction; \( \psi \) is the rotation angle of the metal structure; \( h \) is the arm of force action on the rod with respect to the rotation axis; \( \delta_{\text{hc}} \) is the wall thickness of the hydraulic cylinder;
\( \Delta V_A \) is the dead capacity in the hydraulic cylinder; \( E_{hc} \) is the elasticity module of the hydraulic cylinder wall, \( E_{fluid} \) is the module of fluid bulk elasticity; \( r, c, \gamma \) are the kinematic dimensions of the mechanism.

4. Results

At the first stage of predictions by the simulation model, the pressure was determined in the spaces of hydraulic cylinders according to the existing algorithm (relay stepwise volumetric control), however, at the bridge lifting time equaling to 2 min. Such lifting time can be achieved by increase in the flow rates of boosting and main pumps. The predicted results are illustrated in Fig. 2. The results of analysis are described below.

\[
\begin{align*}
\text{Figure 2.} & \quad \text{Predicted pressure drop in spaces of hydraulic cylinders of Trinity Bridge as a function of time.} \\
&W \quad \text{At the second stage of predictions, the simulation model was used for determination of pressure in the spaces of hydraulic cylinders according to the trapezoid law and according to the cosine law of pump flow rates for two lifting times: 4 min and 2 min:} \\
&W \quad \begin{cases} 
Q = K_Q \cdot t \\
Q = \frac{Q_0}{2} \left(1 - \cos \left( \frac{\pi \cdot t}{t_{acc}} \right) \right) 
\end{cases} \\
&W \quad \text{at} \quad 0 \leq t \leq t_{acc}, \\
&W \quad \text{where} \quad K_Q \quad \text{is the coefficient of proportionality for the trapezoid law of the flow rate;} \quad t_{acc} \quad \text{is the time of acceleration.} \\
&W \quad \text{The respective predicted dependences are illustrated in Figures 3 and 4.}
\end{align*}
\]
\[ Q = K_Q \cdot t \]

\[ Q = \frac{O_0}{2} \left[ 1 - \cos \left( \frac{\pi \cdot t}{t_p} \right) \right] \]

**Figure 3.** Predicted pressure drop in spaces of hydraulic cylinders of Trinity Bridge as a function of time at 4 min lifting time.

\[ Q = K_Q \cdot t \]

\[ Q = \frac{O_0}{2} \left[ 1 - \cos \left( \frac{\pi \cdot t}{t_p} \right) \right] \]

**Figure 4.** Predicted pressure drop in spaces of hydraulic cylinders of Trinity Bridge as a function of time at 2 min lifting time: a) at \( Q = K_Q \cdot t \); b) at \( Q = \frac{O_0}{2} \left[ 1 - \cos \left( \frac{\pi \cdot t}{t_p} \right) \right] \).

5. **Discussion**

Comparative analysis of Figures 1 and 2 demonstrated that with two-fold decrease in the total lifting time, the coefficient of dynamicity \( K_{dyn} \) increased by 20% from 1.33 to 1.7. The oscillation amplitudes of pressure drop reached the presets of safety valves of the hydraulic system (80 kg/cm\(^2\)). Oscillations of pressure (forces) were so high that they had no time to decay during overall lifting time.

Analysis of predictions in Figures 3 and 4 demonstrated that in the case of lifting time of 4 min, the trapezoid law of flow rate variations resulted in the coefficient of dynamicity equaling to 1.08, and the cosine law resulted in the coefficient of dynamicity equaling to 1.10. In the case of the lifting time of 2 min, the trapezoid law of flow rate variations resulted in the coefficient of dynamicity equaling 1.16, and the cosine law resulted in the coefficient of dynamicity equaling to 1.19.

6. **Conclusion**

Conventional relay control of motion speed of the heavy metal structure of the bascule type results in increased dynamics of the process, since acceleration coincides with overriding of maximum loads. The coefficients of dynamicity significantly exceed norms, which increases risks of destruction of metal structures. In order to provide an uninterrupted operation, the machinery works at lower rates,
which decreases machinery capacities (according to statistics, the rates of different machines are reduced three times!). A similar situation can be observed not only upon the reciprocal, but also upon rotational motion of mechanisms [8].

It is required to design drives, the fluid power system in particular with proportional control and to carry out additional dynamical predictions at the design stage: simulation modelling of mechanism motion aiming at determination of optimum laws of speed control. At present, these predictions are supported by verified software and prediction procedures [9]. In addition, it is possible to involve specialized researching companies in simulation modeling.

An optimization criterion should be presented by the minimum coefficient of dynamicity of load on the mechanism upon the specified time of mechanism motion.

For the fluid power system the flow rates can be varied either according to trapezoidal law or according to cosine law. Both laws are implemented by modern pump regulators and/or hydraulic valves with proportional control [10].

References
[1] Alexanderov M P 2000 Hoisting machines, textbook for universities (Moscow: MGTU named after Bauman, Vysshaia shkola) 552 p
[2] 2013 GOST 32579.1-2013 Hoisting cranes. Principles for the formation of design loads and load combinations. Part 1. General provisions
[3] Bashta T M 1972 Hydraulic drive and hydropneumatic automation (Moscow: Mashinostroenie) p 320
[4] Galdin N S 2009 Hydraulic machines, volumetric hydraulic drive: a tutorial (Omsk: SibADI) 272 p
[5] Sokolov S A 2011 Structural mechanics and metal structures of machines: a textbook (St Petersburg.: Politekhnika) 450 p
[6] 55th anniversary of the Alexander Nevsky Bridge - SPb GBU Mostotrest. Retrieved from: https://mostotrest-spb.ru/jubilees/most-aleksandra-nevskogo-91.
[7] 1991 Movable bridge design: Guide book (Lengiprotransmost; Transport, Moscow)
[8] Ashcheulov A V, Shestopalov A A, Khoroshanskii A E and Kochetkov A V 2015 Operation of drill rigs with top drive systems Chemical and Oil-and-Gas Engineering 214-17
[9] Ashcheulov A V, Filipovskii V M and Meshkovskii E O 2016 Optimization of Regulator Parameters of the Automatic Regulation System of the Rotation Frequency of the Valve Engine of the Top Drive System of Bore Installation Int. J. of Applied Engineering Research 11(22) 11055-11059
[10] Sveshnikov V K (ed) 1999 Hydraulic equipment of mobile machinery: Guidebook (Parker Hannifin Corporation)