Development of an algorithm for determining the main quality indicators of the transition process on the example of a LUDV hydraulic system

S Godorozha¹, A Protopopov¹ and M Ryabinin¹²

¹Bauman Moscow State Technical University, 5 Second Baumanskaya Street, Moscow, 105005, Russian Federation

²E-mail: ryabinin@bmstu.ru

Abstract. When optimizing hydraulic systems, various algorithms are used, which are based on the control quality indicators selected as target functions. This is why there is a need to automate the calculation of these indicators. The article considers the main indicators of the quality of the transition process, a mathematical model of the LUDV hydraulic system is compiled and its calculation is performed in two mathematical modeling packages in order to confirm the accuracy of the initial data. An algorithm for determining the main quality indicators of the transition process has been developed, which makes it possible to simplify and speed up the optimization process of the considered LUDV-hydraulic system.

Introduction

In the literature [1]–[14] when optimizing hydraulic systems, various algorithms are used that rely on control quality indicators selected as target functions. Therefore, there is a need to automate the calculation of these indicators.

According to sources [15]–[16], the main indicators (assessments) of the quality of the transition process can be divided into two groups:

- direct assessment;
- indirect (integral) estimates.

Direct estimates include (Fig. 1):

- transition time \( t_p \);
- time to reach the first maximum \( t_m \);
- time of first approval \( t_i \);
- the maximum dynamic error

\[
\delta_{\text{max}} = x_{\text{max}} - x_c;
\]

- maximum overshoot

\[
\sigma_{\text{max}} = \frac{x_{\text{max}} - x_c}{x_c}\cdot100\%;
\]

and the maximum dynamic error \( \delta_{\text{max}} \) and maximum overshoot \( \sigma_{\text{max}} \) together, they determine the assessment of "oscillation".
Among indirect (integral) estimates, the most widely used ones are:

- **Linear integral estimation** $I_1$
  \[ I_1 = \int_0^\infty \delta(t) \, dt, \]
  where $\delta(t)$ — the deviation of the controlled variable $X(t)$ from the set value $X_\infty$,
  \[ \delta(t) = X_\infty - X(t); \]

- **Quadratic integral estimation** $I_2$
  \[ I_2 = \int_0^\infty \delta^2(t) \, dt; \]

- **ITAE (Integral of Time and Absolute Error)** $I_3$
  \[ \text{ ITAE } = \int_0^\infty t \cdot |\delta(t)| \, dt; \]

- **Modified quadratic estimation** $I_4$
  \[ I_4 = \int_0^\infty t \cdot \delta^2(t) \, dt. \]

These estimates are indirect because their absolute value is not related to numerical indicators of the quality of the transition process. They can serve as a generalized indicator when comparing two or
more variants of systems: the variant in which the value of the integral estimate is less is considered the best.

The mathematical model of the system assumes a certain type of transition process of the controlled variable. The integral indicator is selected in such a way that it better characterizes the quality of the system. In other words, there are recommendations for selecting integral indicators for each type of transition process.

Let's look at some of the recommendations: the linear integral estimate $I_1$ can only be used for obviously non-oscillatory (aperiodic) processes. For oscillatory transients, the $I_2$ estimate is used. However, the choice of variable parameters at the minimum $I_2$ usually gives an oscillatory transition process with a rather large oscillation. In cases where such a process is unacceptable, we proceed to the integral criteria $I_3$ and $I_4$.

Mathematical model of the system
As an example, let's consider the definition of the main indicators of the quality of regulation on the drive of two hydraulic cylinders with a LUDV hydraulic system (Fig. 2).

Fig. 2. Schematic diagram of a hydraulic drive with two hydraulic cylinders
1 — adjustable pump; 2 — flow regulator for differential pressure; 3, 11 — control spool; 4, 10 — pressure compensator; 5, 9 — stabilizing spring of low rigidity; 6, 8 — Executive hydraulic cylinder; 7 — valve "or«; $p_p - p_1^* = p_p - p_2 = \Delta p_{\text{regulator}}; p_{LS} = p_{\text{max}} = p_2$. 
The operation of the system can be described by the following equations:

\[
\begin{align*}
Q_{dg} &= Q_{dg} + A_{cp} \frac{dy_{cp}}{dt} + \frac{W_c}{B} \frac{dp_c}{dt}; \\
Q_p &= Q_c + \frac{W_p}{B} \frac{dp_p}{dt}; \\
p_p \cdot a_x = p_{1S} \cdot a_y + C_c \left( x_s + x_{so} \right) + m_s \frac{d^2 x_s}{dt^2} + K_s \frac{dx_s}{dt}; \\
p_c \cdot A_{cp} &= p_p \cdot A_r + C_{cp} \left( y_o + y_{cp} \right) + M_c \frac{dy_{cp}}{dt} + K_{cp} \frac{dp_c}{dt}; \\
(p_{stn} - p_r) \cdot a_n = C_n \cdot \left( x_{stn} + x_{sn} \right) + m_n \frac{d^2 x_n}{dt^2} + K_n \frac{dx_n}{dt}; \\
(p_{stn} - p_r) \cdot a_n &= C_n \cdot \left( x_{stn} + x_{sn} \right) + m_n \frac{d^2 x_n}{dt^2} + K_n \frac{dx_n}{dt};
\end{align*}
\]

where \(Q_{dg}\) — flow rate of the working fluid through the filling slot TSR of the flow controller, \(m^3/s\) (Fig. 3);

\(Q_{dg}\) — working fluid flow through the drain slot TSR flow regulator, \(m^3/s\);

\(A_{cp}\) — area of the piston cavity of the control cylinder, \(m^2\);

\(y_{cp}\) — moving the control piston of the regulator, \(m\);

\(W_c\) — volume of the cavity in the control line of the controller, \(m^3\);

\(B\) — interval reduced volumetric modulus of elasticity of a liquid, \(B = 10^9\ Pa\);

\(p_c\) — pressure in the control line of the regulator, \(Pa\);

\(Q_p\) — pump flow, \(m^3/c\);

\(Q_1, Q_2\) — the cost of the working fluid to consumers, \(m^3/c\);

\(W_p\) — volume of the pressure line cavity, \(m^3\).
$p_p$ — pump pressure, Pa;

$p_{LS}$ — line pressure $LS$, $p_{LS} = \max\left(p_1, p_2\right)$, $p_1$, $p_2$ — pressure caused by the load on the hydraulic cylinder rods, Pa;

$a_s$ — the area of the end face of the spool, m$^2$;

$C_s$ — spring stiffness of the regulator spool, N/m;

$x_{s0}$ — preload of the regulator spool spring, m;

$m_s$ — the mass of the spool, kg;

$K_s$ — coefficient of viscous friction of the regulator spool, N·c/m;

$x_s$ — offset of the regulator spool relative to the neutral position, m;

$A_r$ — area of the control cylinder rod cavity, m$^2$;

$C_{cp}$ — spring stiffness of the control piston, N/m;

$y_0$ — preload of the control piston spring, m;

$M_{cp}$ — the reduced mass of the piston, kg;

$K_{cp}$ — coefficient of viscous friction of the control piston, N·c/m;

$p_{c1}, p_{c2}$ — pressure control the test section, Pa;

$p_T$ — pressure in the drain line, Pa;

$a_{s1}, a_{s2}$ — area of the spool end of the working section, m$^2$;

$C_{s1}, C_{s2}$ — stiffness of the return spring of the spool of the working section, N/m;

$x_{s10}, x_{s20}$ — preload of the spool return spring, m;

$x_{s1}, x_{s2}$ — displacement of the spool of the working section from the neutral position, m;

$m_{s1}, m_{s2}$ — weight of the spool of the working section, kg;

$K_{s1}, K_{s2}$ — coefficient of viscous friction of the spool of the working section, N·c/m;

$p_{1}, p_{2}$ — pressure in front of the compensator, Pa;

$a_{c1}, a_{c2}$ — the area of the end face of the compensator, m$^2$;

$C_{c1}, C_{c2}$ — compensator spring stiffness, N/m;

$x_{c1}, x_{c2}$ — offset of the compensator from the neutral position, m;

$x_{c0}$ — preload of the compensator spring, m;

$m_{c1}, m_{c2}$ — compensator weight, kg;

$K_{c1}, K_{c2}$ — coefficient of viscous friction of the compensator, N·c/m;

$\mu$ — flow coefficient;

$b_{c2}$ — total width of the compensator's throttling windows, m;

$\rho$ — the density of working fluid, kg/m$^3$;

$W_{c1}, W_{c2}$ — volume of the cavity in front of the compensator, m$^3$;

$A_{p1}, A_{p2}$ — hydraulic cylinder piston area, m$^2$;

$F_1(t), F_2(t)$ — force on the hydraulic cylinder rod, N;

$M_{1}, M_{2}$ — reduced weight of the hydraulic cylinder piston, kg;

$y_{1}, y_{2}$ — displacement of the hydraulic cylinder from the extremely retracted position, m;

$K_{p1}, K_{p2}$ — the coefficient of viscous friction of the hydraulic cylinder piston (the calculation of the coefficient has a number of specific features and is not considered in this paper), N·c/m;

$W_{1}, W_{2}$ — volume of the cavity in front of the hydraulic cylinder, m$^3$. 
Result of calculation. Data analysis

To confirm the accuracy of the source data, the system of differential equations describing the operation of the drive was solved in the VisualStudio integrated development environment in C++ and in the MatLAB/Simulink structural modeling package. The Runge-Kutta method was used to solve both cases.

The input step effect in the calculation is the area of the filling slots of the spool distributors. The output signal is the speed of the hydraulic cylinder rods. Disturbing effects are forces on the rods of hydraulic cylinders.

The calculation results are presented as graphs (Fig. 4, 5). It is worth noting that the Runge-Kutta method does not allow solving the system of nonlinear differential equations described above with a time step of more than $10^{-7}$ s.

Fig. 3. Pump regulator diagram

Fig. 4. Graph of changes in the speed of the hydraulic cylinder rod over time (VisualStudio)
The dependencies shown in figures 4 and 5 are almost identical: they have the same indicators and type of transition process.

**Processing of calculation results. Defining the main indicators of transition quality**

So, the dependences obtained from the calculation have the form of an oscillatory transition process. In accordance with the recommendations presented above, it is advisable to choose an integral assessment of ITAE.

The definite integral in the expression for the indicator ITAE is calculated using the Simpson formula for a uniform grid, which has high accuracy in comparison with other numerical methods for calculating the definite integral.

The results of calculating the above indicators for the previously obtained dependences of the speed of the hydraulic cylinder rod on time (Fig. 4) are shown in figures 6-9, where \( z[1] \) is the speed of the hydraulic cylinder rod 6, \( z[2] \) is the speed of the hydraulic cylinder rod 8.

**Fig. 5.** Graph of changes in the speed of the hydraulic cylinder rod over time (MatLAB/Simulink)

**Fig. 6.** During the transition process

**Fig. 7.** Maximum dynamic error
The system calculation results obtained in various mathematical modeling packages (VisualStudio and MatLAB/Simulink) are identical. The developed algorithm for determining the main quality indicators of the transition process will optimize the components included in the system to improve the quality of regulation.

References
[1] N Sosnovsky and D Ganieva 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 012016
[2] I Kolodin and M Ryabinin 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 012018
[3] V Brusov et al 2020 IOP Conf. Ser.: Mater. Sci. Eng. 779 012020
[4] Y Vul et al 2020 IOP Conf. Ser.: Mater. Sci. Eng. 779 012021
[5] B Kulakov and D Kulakov 2020 IOP Conf. Ser.: Mater. Sci. Eng. 779 012027
[6] D Vdovin et al 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 012025
[7] B Kulakov and D Kulakov 2019 IOP Conf. Ser.: Mater. Sci. Eng. 589 012036
[8] Wang, X., Tong, S., Ge, J., & Zhang, J. (2014). Study on dynamic performance of EFMC system in hydraulic excavator. ZhongguoJixieGongcheng/China Mechanical Engineering, 25(15), 2030-2037. doi:10.3969/j.issn.1000-6392. doi:10.1109/AUS.2016.7748194
[9] Ai, C., Lin, J., Kong, X., & Zhao, X. (2016). Design and research of new large flow three-way pressure compensation valves. GaojishuTongxin/Chinese High Technology Letters, 26(5), 498-503. doi:10.3772/j.issn.1002-0470.2016.05.010
[10] Yang, X., Zhao, X., Ai, C., Peng, G., Wang, Z., Dou, J., . . . Li, Y. (2016). Design and research of new large flow three-way pressure compensation valve. Paper presented at the AUS 2016 - 2016 IEEE/CSAA International Conference on Aircraft Utility Systems, 967-972. doi:10.1109/AUS.2016.7748194
[11] Polishchuk, M., Suyazov, M., &Opashnyansky, M. (2020). Study on numerical analysis of dynamic parameters of mobile walking robot. Journal of Mechanical Engineering and Sciences, 14(1), 6380-6392. doi:10.15282/jmes.14.1.2020.14.0499
[12] Zhang, C. (2020). PD plus dynamic pressure feedback control for a direct drive stewart manipulator. Energies, 13(5) doi:10.3390/en13051125
[13] Zhao, B., Guo, W., Ge, L., Hao, Y., &Quan, L. (2020). Experiment study on operation characteristics of new flow self-balancing pump controlled asymmetric hydraulic cylinder. JixieGongchengXuebao/Journal of Mechanical Engineering, 56(8), 257–264. doi:10.3901/JME.2020.08.257
[14] Wang, X., Ge, L., Zhao, B., Hao, Y., Quan, L., & Mu, X. (2020). Energy efficiency characteristics of cable shovel lifting system driven by hydraulic-electric hybrid system. NongyeJixieXuebao/Transactions of the Chinese Society for Agricultural Machinery, 51(4), 418–426. doi:10.6041/j.issn.1000-1298.2020.04.049
[15] Voronov A.A., Titov V.K., Novogradov B.N. Fundamentals of the theory of automatic
regulation and control. Textbook for universities. M. "Higher School", 1977
[16] Popov D. N. Dynamics and regulation of hydraulic and pneumatic systems. Textbook
for engineering universities. M., "Engineering", 1976. — 424 p.