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THE INLET FLUID TEMPERATURE INFLUENCE ON THE MULTI-MODE HYDRAULIC DRIVE OPERATING

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Abstract
The possibility of implementation of controlled temperature changing, cooling processes, in cyclic hydraulic system via computing simulation was considered. The features of load and the Power changes during operating modes influence on fluid temperature. The algorithm of calculation and its implementation to control cooling processes were presented. The simplified thermal dynamic cyclic calculation of hydraulic system was developed. Obtained simulating results and insuring by experiment were presented,

Keywords: cyclic hydraulic drive system, operating mode, thermal flow, fluid temperature.

INTRODUCTION

Using of mechatronic devices in hydraulic units with sensitivity to temperature changes, which causes changes of viscosity and density of fluid, gives possibility to raise operating efficiency of hydraulic systems. For example, stabilization of the rational temperature height (value) in several operating modes [1-4]. Instead of the direct temperature stabilization, controlling via computer simulations makes possible to create systems with pre-adjustment [5-7]. Predefined technological cycles and operating modes in industrial hydraulics are pre-conditions for creating such systems. The main idea is to preset heat-transfer circuit configuration by throttles, valves and additional devices adjustment, according to the following system operating mode.

EXPOSITION

The researching goals are to set up the method of calculation the fluid temperature change in operating modes and development of numerical model, to find out time of temperature stabilization, of cyclic hydraulic drives. In case of denoted goals, were been considered: diagrams of multi-mode systems, with throttling, and their features; developing of simplified algorithm, to find out time of the fluid temperature stabilization, and its using in case of the multi-mode cyclic system. Also, the comparison of simulation results, via algorithm, and experimental data was been made and analyzed.

As boundary conditions, the calculating model includes analyzed operating processes of hydraulic systems these can be represented by several cycle charts, among which system are switching [1, 2, 4, 8, 9]. Additionally, pumping system is considered as constant displacement pump (n=const, [rpm]).

The proposed method and the algorithm includes middle values of Power for each operating tact, instead of middle Power per cycle, for using in the heat-calculation. As assumption, the hydraulic Power of pumping system N_p separates on three main parts:

N_A - Power consumed by drives;

N_{Hyd} - Power, used to overcome the hydraulic line resistance (incl. orifices, throttles, valves, etc.);

$N_{\Delta Q}$ - (over)Power, returned to tank through pressure relief valves.

The Power per tact:

$$N_p = N_A + N_{Hyd} + N_{\Delta Q} = p_p \cdot Q_p = \\ = F_{ext} \cdot Q_A / S_{ef} + \Delta p_{l,\varsigma} \cdot Q_A + p_p \cdot (Q_p - Q_A), \quad (1)$$

herein p_p - pressure at the pump outlet, Q_p - pump flow rate, F_{ext} - middle external load for tact, Q_A - consumed flow, S_{ef} - drive crosssection area, $\Delta p_{l,\varsigma}$ - sum of pressure drops in channels.

According to assumption - the input Power are constant, so external load value and output drive velocity Q_A / S_{ef} have main influence on the fluid temperature, due to changes of heat-flow Power N_{heat}:

$$p_p \cdot Q_p = const = F_{ext} \cdot Q_A / S_{ef} + \Delta p_{l,\varsigma} \cdot Q_A + p_p (Q_p - Q_A) \quad (2)$$

or

$$p_p \cdot Q_p - F_{ext} \cdot Q_A / S_{ef} = \Delta p_{l,\varsigma} \cdot Q_A + p_p (Q_p - Q_A) = N_{heat} . \quad (3)$$

The Power of heat-flow in fluid is a instantaneous energetic characteristic, when the fluid temperature is integral criterion of this Power. Increment heat quantity $\theta_i, [J]$ per operating tact makes possible to see heat spreading over chart, obtained from the Power spreading among tacts:

$$\begin{cases} F_{\max} : \theta_i = [\Delta p_{l,\varsigma}(t_{oil}) \cdot Q_A + p_p (1 - \Delta \widetilde{Q}) Q_p] \cdot T_{takt} \\ (Q_A / S_{ef})_{\max} : \theta_i = [\Delta p_{l,\varsigma}(t_{oil}) \cdot Q_p] \cdot T_{takt} \end{cases} . \quad (4)$$

Another assumption says, that heat-flow Power at the moment of devices switching and/or load disappearing, including inertia, might being changed from min to max value, in range of the system Power. However, in the most cases, tact duration are determined by technological processes. But average value of useful load can change, though the input Power still is constant. Further, for any tact, in which two or several drives are active, its duration can be changed and as average load, so as average Power, can changes in tens times.

The decrement of heat quantity in fluid is defined by several factors. One of the most using is convective heat-transfer between system and environment [10 - 13]. For any tact, decrement can be described by thermal flow q_{mni}, between fluid and environment, and heat-transfer surface area F_i:

$$q_{moi} = F_i \cdot q_{mni} = \alpha \cdot F_i \cdot (t_1 - t_2) , \quad (5)$$

herein $t_1, [{}^0C]$ - fluid temperature; $t_2, [{}^0C] = const$ - environment temperature, α - thermal diffusivity.

Because of drives, which are active in tact and works in the same cycle over and over again, the hydraulic line lines and system structure, hence the heat-transfer surface area, for the same tact, is set up as constant value. So, the heat-transfer has main depend on temperature difference [13]:

$$q_{mni} = (\lambda_k / \delta_k) \cdot (t_1 - t_2) , \quad (6)$$

herein $\delta, [м]$ - heat-conductor thickness.

Considering (5) and (6), several factors were set as constant for a tact duration such as (F, t_2, λ, δ). Fluid temperature raising, main source of the heat-transfer Power, is negative factor, and depends on fluid features and operating conditions [9, 14].

As result of analyses, the heat-transfer equilibrium cannot be reached in acting system due to dynamic processes, such as heating by resistances, valves, external load, etc.

For example, increment heat quantity per cycle is 5 kW, and heat-transfer surface area is 0.5 m^2. For typical design devices and conditions, for thermal flow between the surface and environment, thermal diffusivity is $\alpha = 0.02\ kW/m^2 \cdot K$ [13]. The fluid temperature might increase up to 500 °C, theoretically, but it depends on tact duration, system precondition at tact beginning, and the Power spreading over chart.

In addition, there can be used controlled heat-transfer processes to provide less dynamic difference between input thermal energy and thermal loss to environment. Taking in account thermal features of fluid, the rational structured chart of cooling processes with pre-adjustment opportunity can be developed.

The data for the cooling chart calculation are system and drives condition in defined operating mode:

$$q_i = \sum_{i=1}^{j} q_{ГА_i} + \sum_{i=1}^{k} q_{l_i} + \sum_{i=1}^{m} q_{М_i} - \sum_{i=1}^{n} q_{mo_i} , \quad (7)$$

herein $\sum_{i=1}^{j} q_{ГА_i} = \sum_{i=1}^{j} N_{ГА_i}$ - thermal flow due to the Power drops on hydraulic devices;

$\sum_{i=1}^{k} q_{l_i} = \sum_{i=1}^{k} N_{l_i}$ - thermal flow due to line resistance; $\sum_{i=1}^{m} q_{M_i} = \sum_{i=1}^{m} N_{M_i}$ - thermal flow due to orifices; $\sum_{i=1}^{n} q_{mo_i} = q_{mo_1} + q_{mo_2} + \ldots + q_{mo_n}$ - thermal losses over cycle.

Simplified calculation of thermal loss to environment includes conductivity of hydraulic pipes, valves and tank. Calculation of thermal losses based on [13] is:

$$q_{moi} = F_i \cdot q_{mni}, [W], \tag{8}$$

herein $F,[m^2]$ - surface area of thermal flow to environment; $q_{mn},[W/m^2]$ - thermal flow through homogeny flat conductor.

Quantity of $q_{mn}, [W/m^2]$ calculated in condition of stable environment temperature:

$$q_{mn_i} = \frac{\lambda_i}{\delta_i}(t_1 - t_2), \left[W/m^2\right], \tag{9}$$

herein $\lambda, \left[W/m \cdot {}^0C\right]$- thermal conductivity; $\delta, [м]$ - homogeny flat conductor thickness; $t_1, [{}^0C]$ - fluid temperature; $t_2, [{}^0C] = const$ - environment temperature.

Thermal conductivity of hydraulic devices are taken as average value of thickness, according to equivalent mass of housing.

Thermal energy θ_i , $[J]$, that is received by fluid at fixed period:

$$\theta_i = q_i \cdot t_i, \tag{10}$$

herein t_i,[sec] - period of the fluid over shifting in hydraulic lines on current operating mode.

According to number of active drives and hydraulic lines, the quantity of obtained thermal energy was defined by [7]. This energy leads to increase the temperature of devices and fluid, which existing in working channels [15]:

$$\theta_i = c \cdot m_i \cdot \Delta t_i, \tag{11}$$

herein $c, [J/(kg \cdot \text{sec})]$ - fluid thermal capacity; $m, [kg]$ - mass of over shifted fluid in channels; $\Delta t_i = t_2^i - t_1^i, [{}^oC]$ - increment value of temperature, since it inlet to and till it leaves channels .

Temperature increment is obtain by recalculation the [11]:

$$\Delta t_i = \theta i / (c \cdot m_i), \tag{12}$$

Heated fluid, when it left the channels, goes to tank, and fluid thermal energy diffusing into tank content and effects on its temperature. Hence, fluid temperature at sucking channel of pump is changed and goes to system. This temperature was obtained by:

$$t_T = ((t_T^{init} + \Delta t_i) + ((n-1) \cdot t_T^{init}))/n, \tag{13}$$

herein t_T - current temperature in tank, t_T^{init} - initial temperature of fluid at sucking channel.

Calculation was made according to (7) – (13) and it was based on cyclic recalculation of the system steady condition on tact, according to algorithm (Fig.1).

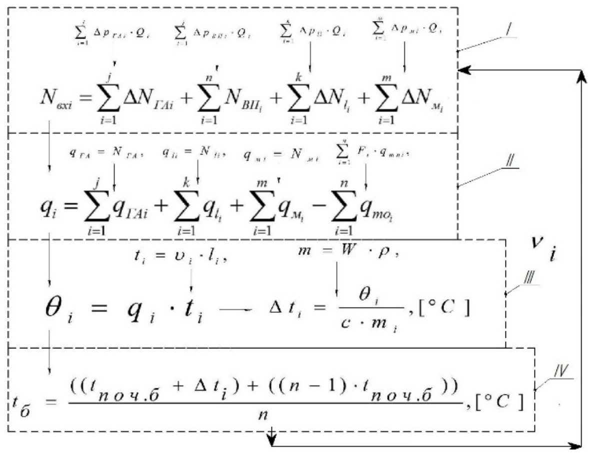

Figure 1*. Algorithm chart of the cyclic calculation of stabilization time and fluid temperature on one operating mode*

The test simulation of hydraulic system of the stamp-machine was made. It consists of 5 hydraulic cylinders: the 1st and 2nd cylinders are stamping actuators; the 3rd – distributor; the 4th and 5th - lifting cylinders. Data, which was used in simulation, are presented in Table 1.

The system chart is the same on each mode. The modes duration is 40 minutes, it's average value of every mode. The chart consists of 11 tacts and operating duration about 48…50 seconds. Difference among modes due to different stamping force, that is provided by the 1st and 2nd cylinders.

Table 1. *Hydraulic system data of operating stamping-machine chart*

Mode	Number of tacts	Mode duration, sec	Number of actuators	Cycle time, sec	The Power, W
1	11	2500	5	49±1	2921,216
2	11	2500	5	49±1	3827,968
3	11	2500	5	49±1	4734,72

Simulation series were made for each operating mode, without mode shifting. Results of simulation represented on Fig.2.a, initial temperature range - +15…+30 °C. The difference of stabilization period among modes corresponds to different Power losses in hydraulic lines due to viscosity changes. So the income thermal energy quantities are different too.

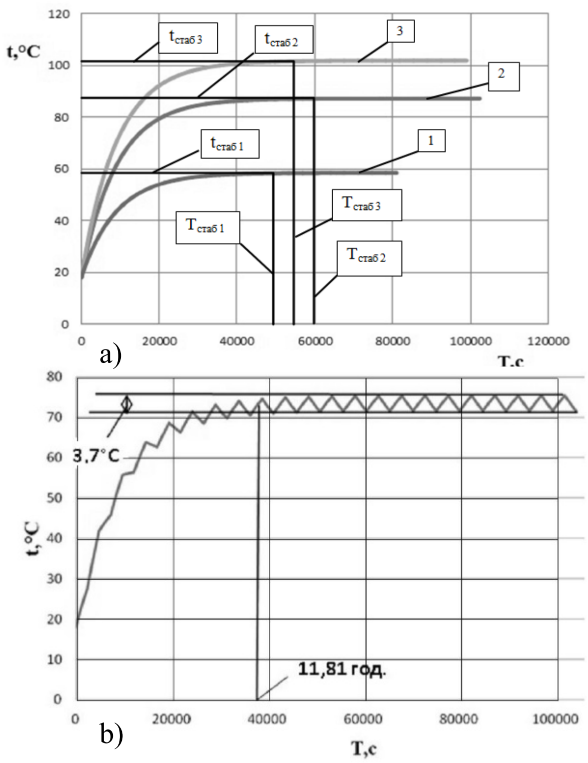

Figure 2. *Simulation series results:*
a) – fluid temperature characteristics on each operating mode; b) – fluid temperature characteristics of system operating on two modes.

Hence, results show that on each mode there is linear incrementing of temperature. The average duration of this region on each mode of the system is about 5% of temperature stabilization period.

Another simulation series are made for operating system work with mode switching (Fig.2.b). Presets for simulation were been: at first - 50 cycles on the 1st mode (about 40 minutes), hence – 50 cycles on the 3rd mode (40 minutes too), and repeat. Each mode duration is chosen to provide linear temperature incrementing, according to Fig.2.a.

Results of the system simulation with switching between two modes show that duration of the temperature stabilization period is co-equal to higher value of mode durations. Also the character of stabilized temperature non-linear, however it is periodical, due to different thermal losses and income thermal Power and processes of reaching thermal equilibrium during the mode.

Results of simulations were ensured by experimental series. Differences between: temperature increment quantities are less than 12 °C; periods of stabilization – less than 20%; stabilized temperature quantities – 6 °C.

CONCLUSION

It is shown that the simplified thermodynamic calculation algorithm in multi-mode cyclic systems allows to simulate and determine the stabilized fluid temperature and the time of stabilization.

Longitude periodic changes of operating modes in the hydraulic system, go to the constant fluctuations of temperature. The amplitude of the temperature fluctuations is close to the linear dependence on the Power changes and depends on the mode duration.

The proposed algorithm for a simplified thermo-hydraulic calculation can be used to develop an algorithm for controlling and implementing a chart of adjustment for cooling. diagram a pre-connecting/ disconnecting heat exchangers based on the calculation of the speed of temperature changes, the initial temperature value and the duration of next operating mode of hydraulic drives.

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