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Problems of the starting and operating of hydraulic components and systems in low ambient temperature

Part III
Methods of determining parameters for correct start-ups of hydraulic components and systems in low ambient temperatures

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ABSTRACT

More and more intensive technology development makes it possible to produce modern hydraulically driven devices which secure high reliability of operation in various conditions. To design such devices, investigations are performed, including experimental tests, which are very expensive and time consuming. In the design process of a new hydraulically driven machine or system, designers frequently have to solve problems concerning its operation in different weather conditions, for instance in low ambient temperatures. Due to the complexity of the phenomena of concern, the serviceability of the designed hydraulic component or system in those conditions is most frequently assessed based on numerical calculations, much cheaper than the experiments. The article presents three methods: experimental, analytical, and that based on computer simulation, applied for assessing the serviceability of cold hydraulic components supplied with hot working medium being a source of a so-called thermal shock.

Keywords: hydraulic machines, hydraulic drives, diagnostics, hydraulic systems

INTRODUCTION [4, 5]

Two articles entitled "Problems of the starting and operating of hydraulic components and systems in low ambient temperature", which were published in Polish Maritime Research No. 4/2008 and 1/2009, presented the results of experimental investigations of hydraulic components and systems in the abovementioned conditions. Basic parameters which affect the correct start-up of a hydraulic component in thermal shock conditions were defined. These parameters include: the flow rate Q of the working medium (most frequently oil), the oil temperature T_{ol} and the initial temperature of the component T_{ot} which most frequently corresponds to the ambient temperature.

In the experimental investigations the time-histories of flow rates and oil pressures at component inlet and outlet were recorded. Based on these data the operation of the examined component was analysed, and conclusions were made about its correctness.

An important factor affecting the serviceability of the components in negative-temperature conditions is its structural properties. Numerous phenomena take place during the component start-up in thermal shock conditions. Among them, the factor which most affects the faultless operation of the component is the changes in the dimensions of axial and radial clearances between the cooperating elements. Methods which make it possible to determine the areas of correct and incorrect operation of the hydraulic component during its start-up in thermal shock conditions base on analysis of changes of the effective clearance during the transient process of component elements heating.

Determining parameters of correct start-up of the component based on the results of experimental investigations (recorded time-histories of oil pressures and flow rate) is expensive and time consuming. The areas of correct start-up parameters can be assessed not only from experimental investigations, which may lead to component damage, but also using numerical methods, which are recommended as cheaper and faster.

THERMAL PROCESSES IN COLD HYDRAULIC SYSTEM COMPONENTS DURING THEIR START-UPS IN THERMAL SHOCK CONDITIONS

The start-up of a cold hydraulic component supplied with the working medium, usually the mineral oil of higher temperature, is a source of unsteady heat flow $\dot{\Phi}_\alpha$ from the oil to component elements. The temperature of the component elements will change to reach a new steady-state level, as the system (component) response to the input signal in the form of a flow rate jump of the hot oil having the temperature T_{ol} (Fig. 1). During this time the component will absorb some portion of heat, arriving at a new internal energy level.

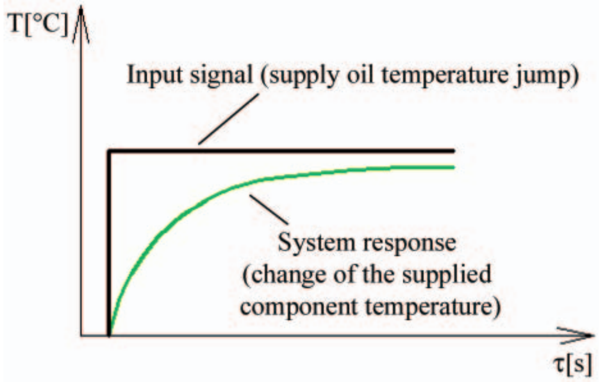

Fig. 1. System response (component temperature) to the set input signal in the form of a flow rate jump of the hot oil having the temperature T_{ol}

The hydraulic system consists of a number of components having different functions. Three basic groups of components, differing by the structure, can be named, which are: supply, control, and executing components.

The article presents thermal balances for two systems of hydraulic components, typical from the point of view of the examined problem, which are pumps and spool valves.

THERMAL BALANCE OF A HYDRAULIC PUMP IN THERMAL SHOCK CONDITIONS

It was assumed in the pump heating model that external leakage flows are absent, i.e. the mass flow rates of the working medium at the pump inlet and outlet are equal to each other:

$$\dot{m}_1 = \dot{m}_2 \tag{1}$$

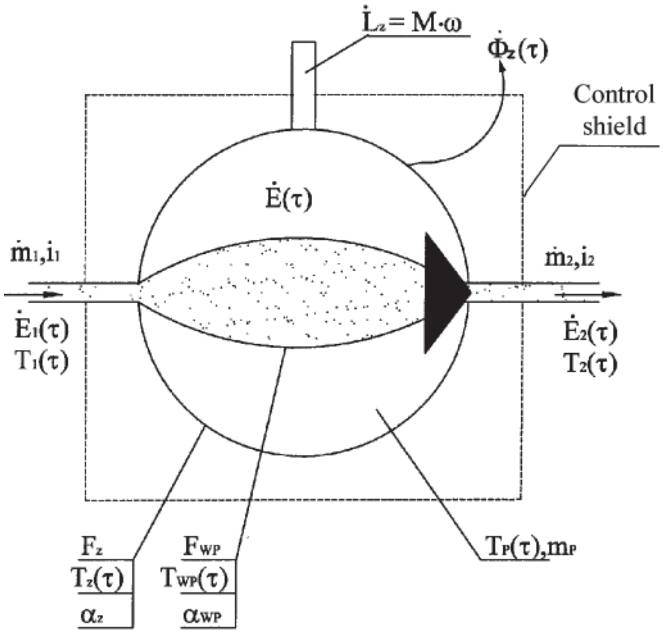

Fig. 2. Parameters used for determining energy balance in hydraulic pump

According to the first law of thermodynamics, the energy E_1 delivered to the hydraulic pump during its start-up is the sum of the energy extracted from the system E_2 and the system energy increase ΔE:

$$E_1 = E_2 + \Delta E \tag{2}$$

The energy delivered with the working medium flowing through the pump and that extracted from the pump in time dτ (Fig. 2) consist of:

- work done for delivering and extracting the elementary mass flow rate flux $dm = \dot{m} d\tau$ of the fluid having the specific volume v and pressure p, i.e.:

$$dL = p \cdot v \cdot dm \tag{3}$$

- internal energy:

$$dU = dm \cdot u \tag{4}$$

- kinetic energy:

$$dE_k = dm \cdot \frac{w^2}{2} \tag{5}$$

where:
w – average fluid flow velocity

- potential energy:

$$dE_p = dm \cdot g \cdot h \tag{6}$$

where:
h – height of the position of the elementary mass of fluid with respect to the reference level.

In the elementary time dτ the external work dL_z is delivered to the machine, and the heat $d\Phi_z$ is extracted from it to the environment. The internal energy of the pump changes by dE.

Using the balance form of the first law of thermodynamics we arrive at the equation:

$$dm_1\left(u_1 + p_1 \cdot v_1 + \frac{w_1^2}{2} + g \cdot h_1\right) + dL_Z = = dE + d\Phi_Z + dm_2\left(u_2 + p_2 \cdot v_2 + \frac{w_2^2}{2} + g \cdot h_2\right) \tag{7}$$

Components and differences of the kinetic energy, which depends on the flow velocities: w_1 on the suction side of the pump and w_2 on its pressure side, are negligibly small compared to other terms in Equation (7). Moreover, the positions of the inlet and outlet connectors are most frequently on similar levels, $h_1 \approx h_2$. Taking that into account, the energy balance equation can be simplified by eliminating the potential energy from it, as its differences are close to zero. After these simplifications Equation (7) can be written as:

$$dm_1(u_1 + p_1 \cdot v_1) + dL_Z = = dE + d\Phi_Z + dm_2(u_2 + p_2 \cdot v_2) \tag{8}$$

After introducing the enthalpy, defined as $i = u + p \cdot v$, to Equation (8) we arrive at:

$$\dot{m}_1 \cdot i_1(\tau) + \dot{L}_Z(\tau) = \dot{E}(\tau) + \dot{\Phi}_z(\tau) + \dot{m}_2 \cdot i_2(\tau) \tag{9}$$

where:
$i_1(\tau)$, $i_2(\tau)$ – enthalpy of the fluid at inlet and outlet of the control shield.

The hydraulic pump consists of two groups of elements, which are mobile and fixed elements. The simplified scheme of the pump, (Fig. 3), presents the flow of energy fluxes affecting the process of heating of pump elements.

The difference in energy fluxes: $\dot{E}_1(\tau)$ at pump inlet and $\dot{E}_2(\tau)$ at pump outlet depends on:

- thermal energy flux, $\dot{\Phi}_z(\tau)$, flowing from the outer pump surface to the environment. This quantity is defined by the equation:

$$\dot{\Phi}_z(\tau) = \alpha_z \cdot F_z \cdot (T_z(\tau) - T_{ot}) \tag{10}$$

where:
α_z – heat transfer coefficient between the outer pump surface F_z and the environment
$T_z(\tau)$ – average temperature of the outer pump surface
T_{ot} – ambient temperature.

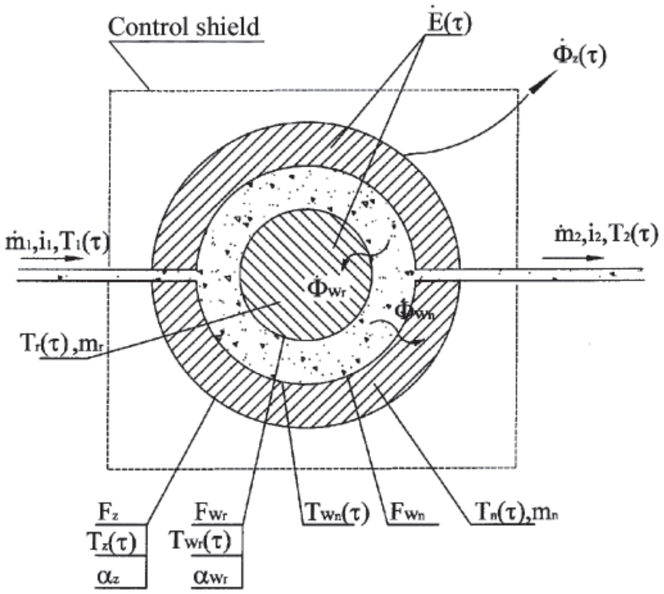

Fig. 3. *Model of cold pump heating by the flowing hot fluid*

- flux of thermal energy which increases the internal energy of the oil in closed pump chambers and the thermal energy of pump elements:

$$\dot{E}(\tau) = \dot{E}_{Zol}(\tau) + \dot{E}_P(\tau) \tag{11}$$

$\dot{E}_{Zol}(\tau)$ – flux of thermal energy that increases the internal energy of the oil in closed pump chambers:

$$\dot{E}_{Zol}(\tau) = V \cdot \rho_{ol} \cdot c_{ol} \frac{dT_{Zol}}{d\tau} \tag{12}$$

where:
- V – total volume of pump chambers and passages
- ρ_{ol} – oil density
- c_{ol} – specific heat of oil
- $\dot{E}_p(\tau)$ – flux of thermal energy that increases the thermal internal energy of structural pump elements:

$$\dot{E}_P(\tau) = \sum_{i=1}^{n} m_i \cdot c_i \cdot \frac{dT_i}{d\tau} \tag{13}$$

where:
- m_i – mass of the i–th pump element
- c_i – specific heat of the material of the i–th pump element
- T_i – average temperature of the i–th pump element.

- mechanical power delivered to the pump:

$$\dot{L}_Z(\tau) = \omega(\tau) \cdot M(\tau) \tag{14}$$

Part of the energy $L_Z(\tau)$ delivered to the pump is converted to heat as a result of energy losses caused by leakage flows, flow resistance, and friction. We can assume that the heat generated as a result of pump losses is totally absorbed by the oil.

The flux $\dot{\Phi}_\alpha(\tau)$ of the thermal energy flowing from the oil to the outer pump surface is equal to the flux $\dot{E}(\tau)$ of the thermal energy which increases the internal energy of pump elements and the flux $\dot{\Phi}_z(\tau)$ of the thermal energy flowing from the outer pump surface to the environment:

$$\dot{\Phi}_\alpha(\tau) = \dot{E}(\tau) + \dot{\Phi}_z(\tau) \tag{15}$$

The flux of the thermal energy flowing from the oil to pump elements is equal to:

$$\dot{\Phi}_\alpha(\tau) = F_{WP} \cdot \alpha_{WP} \cdot [T_{ol} - T_{WP}(\tau)] \tag{16}$$

where:
- F_{WP} – heat exchange area between the flowing oil and the pump
- α_{WP} – heat transfer coefficient between oil and pump
- T_{ol} – average oil temperature in the pump
- $T_{WP}(\tau)$ – average temperature of the heat exchange surface between the flowing oil and the pump.

The flux $\dot{\Phi}_\alpha(\tau)$ of thermal energy flowing from the oil to mobile elements $\dot{\Phi}_{Wr}(\tau)$ and fixed elements $\dot{\Phi}_{Wn}(\tau)$ of the pump is equal to:

$$\dot{\Phi}_\alpha(\tau) = \dot{\Phi}_{Wr}(\tau) + \dot{\Phi}_{Wn}(\tau) \tag{17}$$

$$\dot{\Phi}_\alpha(\tau) = \sum_{j=1}^{m} F_{Wrj} \cdot \alpha_{Wrj} \cdot [T_{ol} - T_{Wrj}(\tau)] + \sum_{k=1}^{z} F_{Wnk} \cdot \alpha_{Wnk} \cdot [T_{ol} - T_{Wnk}(\tau)] \tag{18}$$

where:
- r – index defining the mobile element
- n – index defining the fixed element
- m – number of surfaces of mobile elements
- z – number of surfaces of fixed elements.

THERMAL BALANCE OF THE COLD CLASSICAL AND PROPORTIONAL SPOOL VALVE, AND THE HYDRAULIC SERVOVALVE DURING SYSTEM START-UP IN THERMAL SHOCK CONDITIONS

In the spool valves which are most frequently used in hydraulic systems the oil flows through channels connected with the pump 1, executing components 2 and 3, and the outlet to the tank 4 (Fig. 4). During the system start-up in thermal shock conditions, when the hot working medium suddenly flows through the cold valve, the increase of the internal energy of the valve is observed.

The thermal balance for the hydraulic valve with the oil flowing inside (Fig. 4) can be described using the equations in which the energy fluxes of the medium flowing in $\dot{E}_1(\tau) + \dot{E}_3(\tau)$ and out $\dot{E}_2(\tau) + \dot{E}_4(\tau)$ are taken into account:

$$dm_1(u_1 + p_1 \cdot v_1 + \frac{w_1^2}{2} + g \cdot h_1) + dm_3(u_3 + p_3 \cdot v_3 + \frac{w_3^2}{2} + g \cdot h_3) = dE + d\Phi_Z + dm_2(u_2 + p_2 \cdot v_2 + \frac{w_2^2}{2} + g \cdot h_2) + dm_4(u_4 + p_4 \cdot v_4 + \frac{w_4^2}{2} + g \cdot h_4) \tag{19}$$

Let us assume that the heights of the valve ports are $h_1 \approx h_2 \approx h_3 \approx h_4$, and the flow velocities are $w_1 \approx w_2 \approx w_3 \approx w_4$. After introducing the enthalpy according to the definition $i = u + p \cdot v$ to Equation (19) we arrive at:

$$(\dot{m}_1 \cdot i_1 + \dot{m}_3 \cdot i_3) \cdot d\tau = (\dot{m}_2 \cdot i_2 + \dot{m}_4 \cdot i_4) \cdot d\tau + dE + \dot{\Phi}_Z \cdot d\tau \tag{20}$$

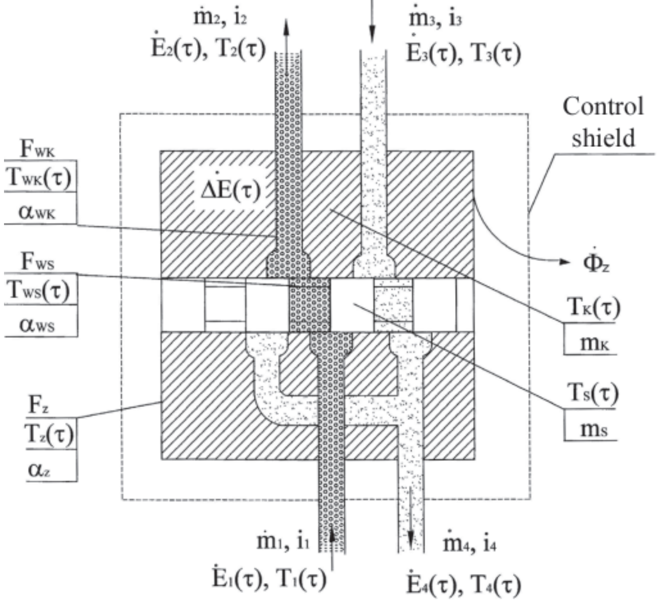

***Fig. 4.** Model of thermal energy flow from the hot oil to spool valve elements: $\dot{E}_1(\tau)$ - flux of energy flowing into spool valve channel P; $\dot{E}_2(\tau)$ - flux of energy flowing out of spool valve channel A; $\dot{E}_3(\tau)$ - flux of energy flowing into spool valve channel B; $\dot{E}_4(\tau)$ - flux of energy flowing out of spool valve channel T; $\dot{\Phi}_z(\tau)$ - flux of energy passed to the environment; $\dot{m}_1$, $\dot{m}_3$ - mass flux of oil flowing into the spool valve; $\dot{m}_2$, $\dot{m}_4$ - mass flux of oil flowing out of the spool valve; $T_1(\tau)$, $T_3(\tau)$ - oil temperature at inlet; $T_2(\tau)$, $T_4(\tau)$ - oil temperature at outlet; i_1, i_3 - inlet enthalpy of the fluid; i_2, i_4 - outlet enthalpy of the fluid; $T_S(\tau)$ - temperature of the spool; $T_{WS}(\tau)$ - temperature of the spool surface being in contact with the flowing oil; F_{WS} - heat exchange surface between oil and spool; α_{WS} - heat transfer coefficient between the oil and the spool surface; $T_{WK}(\tau)$ - temperature of the inner housing surface; F_{WK} - heat exchange surface between the oil and the housing; α_{WK} - heat transfer coefficient between the oil and the surface of housing; α_z - heat transfer coefficient between the outer surface and the environment*

If we assume that the temperature difference between the outer surface and the environment is close to zero in the initial system start-up stage, then the flux of energy flowing from the outer valve surface to the environment becomes also close to zero:

$$\dot{\Phi}_z(\tau) = \alpha_z \cdot F_z \cdot (T_z(\tau) - T_{ot}) \approx 0 \qquad (21)$$

With the relation (21) being taken into account in Equation (20), the increase of the internal energy dE of the spool valve elements is given by the equation:

$$dE = (\dot{m}_1 \cdot i_1 + \dot{m}_3 \cdot i_3)d\tau - (\dot{m}_2 \cdot i_2 + \dot{m}_4 \cdot i_4)d\tau \qquad (22)$$

Assuming that the mass flow fluxes $\dot{m}_1 = \dot{m}_2$ and $\dot{m}_3 = \dot{m}_4$, Equation (22) can be written in the form:

$$dE = \dot{m}_1(i_1 - i_2)d\tau + \dot{m}_3(i_3 - i_4)d\tau = \dot{\Phi}_\alpha \, d\tau \qquad (23)$$

The increase of the internal energy of spool valve elements (spool and casing) depends on the working medium enthalpy differences between channels 1 and 2, and between channels 3 and 4.

The flux of the thermal energy flowing from the oil to spool valve elements is equal to:

$$\begin{aligned} \dot{\Phi}_\alpha(\tau) = \alpha_{WK} \cdot F_{WK} \cdot [T_{ol} - T_{WK}(\tau)] + \\ + \alpha_{WS} \cdot F_{WS} \cdot [T_{ol} - T_{WS}(\tau)] \end{aligned} \qquad (24)$$

METHODS FOR DETERMINING CLEARANCES BETWEEN COOPERATING HYDRAULIC COMPONENT ELEMENTS IN THERMAL SHOCK CONDITIONS

Changes in the effective clearance between the cooperating hydraulic component elements during the start-up depend on a number of factors, including: assembly clearance, load of the component, ambient temperature, oil temperature and flow rate.

If the effective clearance values calculated for the start-up of the hydraulic component in thermal shock conditions are positive ($l_e > 0$), we can conclude that the component will work faultlessly in those conditions.

Methods which make it possible to determine component serviceability (calculate clearances between the cooperating elements) are given in Fig. 5. These methods include experimental, analytical, and computer simulation based methods.

Experimental method

The experimental method of determining effective clearance changes between the cooperating hydraulic component elements during the process of component heating requires a number of series of experimental tests for a given component structure scale. The hydraulic components must be prepared for investigations in thermal shock conditions, including possible measurements of temperatures of their elements and the oil temperature. Temperature sensors are to be placed in both the fixed and mobile elements of the component. If the temperature of the mobile elements of the examined component cannot be measured, model investigations of the process of their heating are to be done.

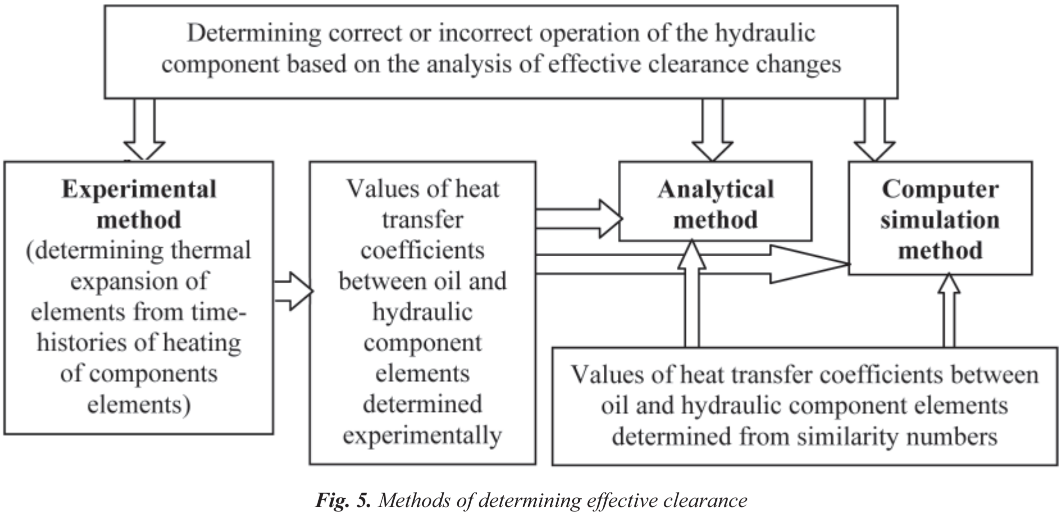

***Fig. 5.** Methods of determining effective clearance*

The experimental investigations of the hydraulic component in thermal shock conditions for selected start-up parameters (flow rate, ambient temperature, and oil temperature) and the initial clearance, will produce the time-histories of temperatures of hydraulic component elements, pressures and flow rates. The collected data, which include the time-histories of temperatures of elements and assembly clearances, will make the basis for determining the correct operation area depending on the initial temperature difference between the oil and the hydraulic component, and the temperature differences between the cooperating component elements.

The effective clearance l_e between the cooperating elements of not loaded component depends on the assembly clearance l_m and the thermal expansion difference Δl_t:

$$l_e = l_m - \Delta l_t \tag{25}$$

When the seizure of the mobile elements in the fixed elements takes place, the effective clearance [3, 4, 5] is equal to zero ($l_e = 0$).

After introducing the equations which describe the thermal expansion differences Δl_t of the hydraulic component elements we get:

$$l_m - \beta_r \cdot h_r \cdot (T_r - T_0) + \beta_n \cdot h_n \cdot (T_n - T_0) = 0 \tag{26}$$

where:

- β_r, β_n – linear thermal expansion coefficient of the mobile element (r) and fixed element (n)
- h_r, h_n – height of the mobile element, height of the opening in the fixed element
- T_r, T_n – average temperature of the mobile element and fixed element, respectively
- T_0 – temperature of the element height measurement

After further transformations of Equation 26 a formula is obtained which describes the temperature of the mobile element, at which the effective clearance disappears:

$$T_r = \frac{\beta_n \cdot h_n}{\beta_r \cdot h_r} T_n + \frac{l_m + \beta_r \cdot h_r \cdot T_0 - \beta_n \cdot h_n \cdot T_0}{\beta_r \cdot h_r} \tag{27}$$

Assuming that the component elements are made of materials revealing similar values of linear thermal expansion coefficients, ($\frac{\beta_n \cdot h_n}{\beta_r \cdot h_r} \cong 1$), Equation 27 can be written as:

$$T_r = T_n + B \tag{28}$$

where:

$$B = \frac{l_m}{\beta_r \cdot h_r}$$

Equation 28 can be used for determining the temperature difference between the mobile element and the fixed elements, at which the effective clearance disappears, i.e. when $l_m = \Delta l_t$:

$$T_r - T_n = \Delta T_{r-n} = B \tag{29}$$

In Equation 25 the temperature difference between the mobile element and the fixed elements depends most of all on the assembly clearance l_m. The smaller the assembly clearance, the smaller the permissible temperature difference between the mobile and fixed elements cooperating with each other at which the effective clearance disappears.

In order to determine the temperature difference ΔT_{ol-ot} between the hot oil and the cold component, the following set of equations is to be solved:

$$\begin{cases} T_r - T_n = \Delta T_{r-n} = B \\ \Delta T_{ol-ot} = a \cdot \Delta T_{r-n} \end{cases} \tag{30}$$

The constant "a" is to be read from the diagram (Fig. 6) or calculated.

After solving the set of equations the relation was obtained which can be used for calculating permissible temperature difference $\Delta T_{ol-ot(MAX)}$ between the hot oil and the cold component:

$$\Delta T_{ol-ot(MAX)} = a \cdot B \tag{31}$$

When the temperature difference ΔT_{ol-ot} between the oil and the hydraulic component is smaller at the initial time than the permissible temperature difference $\Delta T_{ol-ot(MAX)}$ (Fig. 6), the start-up of this component will take a faultless course in thermal shock conditions.

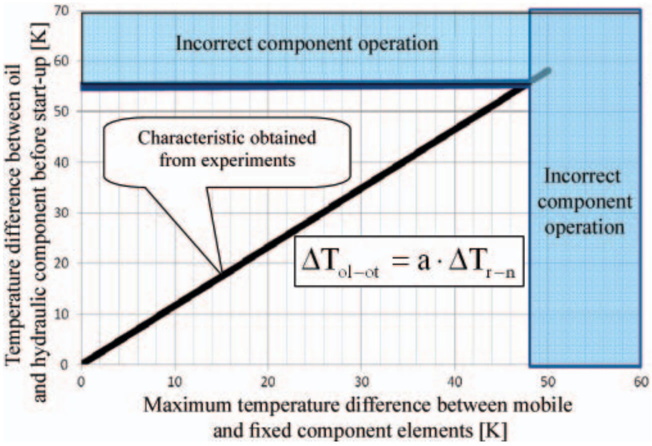

Fig. 6. *Experimental assessment of the maximum temperature difference between the oil and the component, at which the component will work faultlessly in thermal shock conditions*

Analytical method

Due to its disadvantages, the experimental method cannot be used in all cases. For instance, measuring temperatures of mobile elements is, generally, extremely difficult. Therefore basic parameters of faultless system start-up should be determined using the analytical method or the computer simulation method.

An advantage of the analytical method is its ability to determine the correct operation area using simple equations. Time consumption is smaller than when using complicated computer codes.

In order to determine with the aid of the analytical method the serviceability of the hydraulic component started in thermal shock conditions we have to know the time-histories of average temperatures of mobile and fixed elements of the component.

Figure 7 shows the time-histories of the average temperatures of the fixed elements T_n and their surfaces T_{Wn} which are in contact with the flowing oil after the beginning of the start-up. In the initial time interval of heating of the fixed elements, from $\tau = 0$ to τ', the disordered heat transfer takes place [1, 3, 12, 13].

In the next interval, $\tau > \tau'$, the nature of temperature changes at particular points of fixed elements (T_n , T_{Wn}) depends of the shape, dimensions, and thermophysical properties of the element and on heat transfer conditions on the element surface. This is the time of well-ordered heat transfer. After time τ_u the steady state of temperatures in fixed elements is reached.

The time τ' of initial heating of component elements a disordered manner (Fig. 7) is short, as compared to the entire time τ_u of the heating process and is not of high importance for the mathematical description of the phenomenon. To determine the effective clearance with the aid of the analytical method, the author made use of the well-ordered state of heating of component elements.

The temperature time-history $T_n(\tau)$ of the fixed elements can be calculated, for initial conditions (T_{ol} , T_{ot}), making use of the energy balance for those elements:

$$\dot{\Phi}_n(\tau) = \alpha_{Wn} \cdot F_{Wn} \cdot [T_{ol} - T_n(\tau)] \cdot \psi_n = \\ = m_n \cdot c_n \cdot \frac{dT_n}{d\tau} + \dot{\Phi}_z(\tau) \quad (32)$$

where:

- m_n – mass of fixed elements
- $\psi_n = \frac{T_{ol} - T_{Wn}}{T_{ol} - T_n} = \frac{\vartheta_{Wn}}{\vartheta_n}$ – coefficient of non-uniform temperature distributions in fixed elements, taking into account relations between the temperature differences:
  - ϑ_{Wn} – difference between the average oil temperature (higher) and the average temperature (lower) T_{Wn} of the surface F_{Wn}
  - ϑ_n – difference between the average oil temperature (higher) and the average volumetric temperature (lower) of the fixed elements in the conditions of well-ordered unsteady heat transfer.

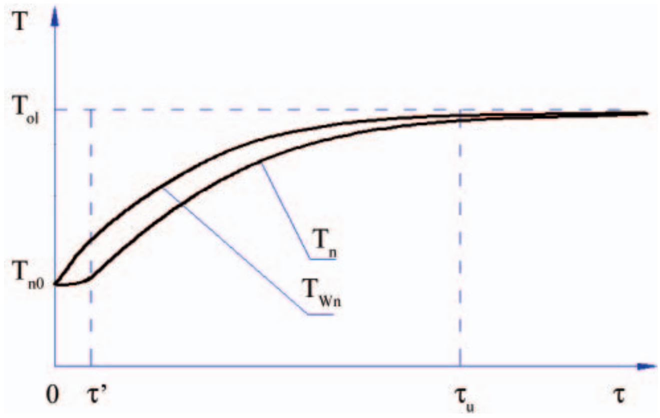

Fig. 7. *Time-histories of temperature changes in heated fixed elements. Within time interval from τ = 0 to τ = τ' disordered heat transfer takes place, while the interval τ = τ' to τ = τ_u – is characterised by well-ordered unsteady state, and above τ = τ_u – by steady state. T_{n0} is the initial temperature of the fixed element*

Some quantities in Formula (32), such as masses and heat transfer surfaces of component elements, and specific heats of materials, are known, while the others are to be determined experimentally. The results of experimental tests make the basis for determining the heat transfer coefficient α_{Wn} and the coefficient ψ_n of non-uniform temperature distribution in component elements.

The flux $\dot{\Phi}_z$ of the energy passed to the environment, described by Equation (32), has been neglected. The thermal energy passed to the environment during the start-up time is small, as the outer temperature of the component is approximately equal to the ambient temperature $T_z(\tau) \approx T_{ot} \rightarrow \dot{\Phi}_z \approx 0$.

After some transformations of Equation (32) we get for fixed elements:

$$m_n \cdot c_n \frac{dT_n}{d\tau} = \alpha_{Wn} \cdot F_{Wn} \cdot (T_{ol} - T_n) \cdot \psi_n \quad (33)$$

Equation 33 can be rewritten to the form:

$$\frac{dT_n}{T_{ol} - T_n} = \frac{\alpha_{Wn} \cdot F_{Wn} \cdot \psi_n}{m_n \cdot c_n} d\tau \quad (34)$$

where:

$$\sigma_n = \frac{\alpha_{Wn} \cdot F_{Wn} \cdot \psi_n}{m_n \cdot c_n} \quad (35)$$

is the rate of heating of fixed component elements.

After relevant transformations of Equation 34 and introduction of relation 35 we arrive at:

$$\frac{d(T_{ol} - T_n)}{(T_{ol} - T_n)} = -\sigma_n \cdot d\tau \quad (36)$$

After integrating we get:

$$T_{ol} - T_n = (T_{ol} - T_{ot})e^{-\sigma_n \cdot \tau} \quad (37)$$

Having known the initial conditions: oil temperature T_{ol}, ambient temperature T_{ot} and the rate of heating of fixed elements σ_n we can calculate the temperature time-history $T_n(\tau)$:

$$T_n(\tau) = T_{ol} - (T_{ol} - T_{ot})e^{-\sigma_n \cdot \tau} \quad (38)$$

In the analytical method the rate of heating of fixed elements can be determined from Equation 35.

Temperature time-histories $T_r(\tau)$of mobile component elements can be calculated based on the initial conditions (T_{ol}, T_{ot}) and the rate of heating of mobile elements σ_r:

$$T_r(\tau) = T_{ol} - (T_{ol} - T_{ot})e^{-\sigma_r \cdot \tau} \quad (39)$$

The rate of heating of a mobile element of the hydraulic component can be calculated from the relation:

$$\sigma_r = \frac{\alpha_{Wr} \cdot F_{Wr} \cdot \psi_r}{m_r \cdot c_r} \quad (40)$$

where:

α_{Wr} – heat transfer coefficient between oil and mobile element
F_{Wr} – area of heat transfer of mobile element
m_r – mass of mobile element
c_r – specific heat of mobile element
Ψ_r – coefficient of non-uniform temperature distribution in mobile element.

The rate of heating eg. of fixed elements of the hydraulic component can be established experimentally having known current values of the temperature differences ϑ_{n1} and ϑ_{n2} for times τ_1 and τ_2 [13]:

$$\sigma_n = \frac{\ln \vartheta_{n1} - \ln \vartheta_{n2}}{\tau_2 - \tau_1} \quad (41)$$

The rate of heating of an element can be more precisely determined by measurements of more temperature differences and corresponding time moments, and by making a semi-logarithm diagram. In Fig. 8, of coordinates lnϑ and time τ, time-history of temperature difference is presented using the straight line equation y = ax + b, where a is the slope of this line and represents the rate of heating σ (σ = –a).

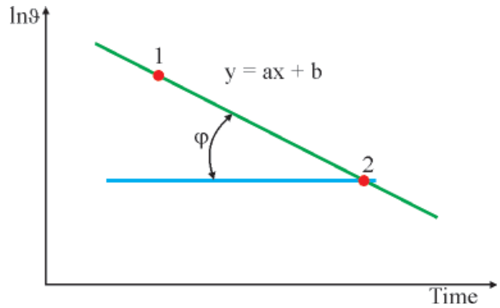

Fig. 8. *Determining the rate of heating*

Calculating coefficients of non-uniform temperature distribution in hydraulic component elements

The coefficients of non-uniform temperature distribution ψ_r, ψ_n in component elements (Equations 35, 40) can be determined using the relations given by Kondratev [1, 3, 7, 8].

The coefficient ψ depends on the Biot number (Bi):

$$\psi = 1 - 0.73 \cdot \xi^2 \tag{42}$$

where:

$$\xi = \frac{p}{p_\infty} \text{ – criterial number} \tag{43}$$

The criterial number p can be calculated from the formula (Bi < 1):

$$p^2 = \frac{L_0}{l} \cdot Bi \tag{44}$$

where:

L_0 – characteristic dimension defining the body, for instance: wall thickness, radius of a cylinder or a sphere

$l = V/F_w$ – linear dimension of a body as the ratio of its volume V to heat transfer surface F_w.

The value of the number p_∞ for fixed elements of hydraulic components can be roughly estimated based on a selected characteristic element having the shape of a plate or cylinder. For the plate the value of the number p_∞ is equal to $p_{\infty pl} = \pi/2$, while for a cylinder - $p_{\infty wa} = 2.4$ [1, 3, 7, 8].

For an arbitrary fixed element of the hydraulic component, the criterial number can be assumed from between those for a plate and a cylinder. The averaged value of the criterial number for the fixed element is equal to $\frac{p_{\infty wa} + p_{\infty pl}}{2} = p_\infty \cong 2$.

Calculating heat transfer coefficients between oil and surface of the hydraulic component elements

Calculating the heat transfer coefficients between the oil and the surfaces of component elements requires the following data: heat transfer area, mass of the element, specific heat of the element, coefficient of non-uniform temperature distribution in the element, and the rate of heating of the element.

Using Equations (35), (40), the heat transfer coefficients α_n , α_r between the oil and the surfaces of fixed and mobile elements of hydraulic components were calculated, among others, for: proportional spool valve PVG 32 made by Sauer Danfoss (Fig. 9), satellite motor SOK 100 made by Hydroster (Fig. 10).

Determining serviceability of the proportional spool valve PVG 32 with the aid of the analytical method

Proportional spool valves PVG 32 [5] are used in various hydraulic systems. They can consist of up to twelve sections. Using the analytical method we can determine the serviceability of the spool valve working in thermal shock conditions.

In order to determine the average temperatures of the heated elements in the spool valve PVG32 in thermal shock conditions, relations (38, 39), we should know the rate of heating of those elements. This quantity can be determined from formula (35). The required heat transfer coefficient can be determined from curves shown in Fig. 9.

One experimental test of the spool valve PVG 32 was done for the following initial conditions: spool valve temperature -10 °C, oil temperature 37 °C, and flow rate 39 dm³/min. Based on those data the equations were obtained to describe time-histories of:

a) spool temperature, $T_S(\tau)$:

$$T_S(\tau) = 37 - 47e^{-0.106\tau} \text{ [°C]} \tag{45}$$

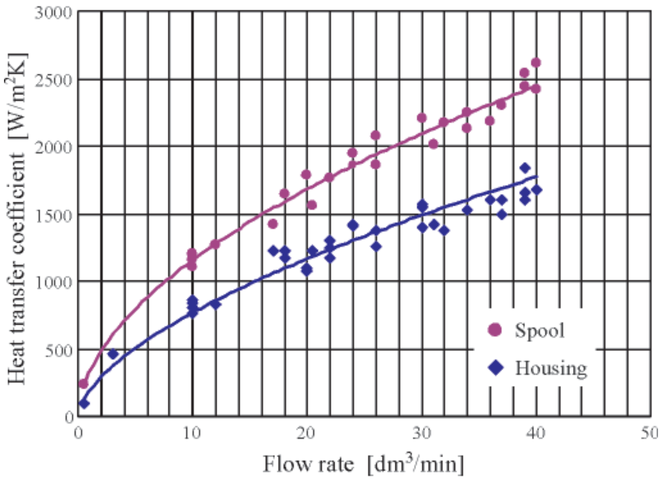

Fig. 9. Heat transfer coefficients between oil and surfaces of spool and housing of proportional spool valve PVG 32 vs. flow rate

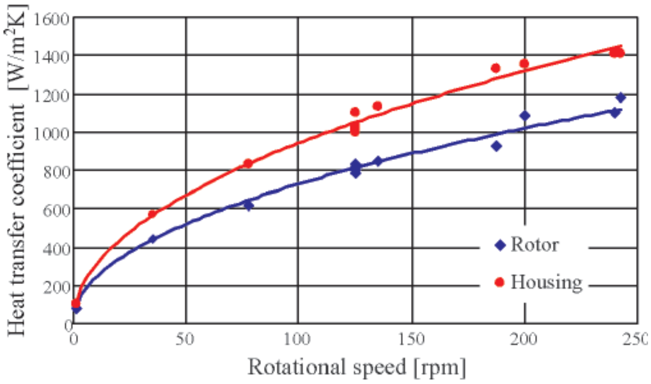

Fig. 10. Heat transfer coefficients between oil and surfaces of rotor and housing of satellite motor SOK 100 vs. rotational speed

b) housing temperature, $T_K(\tau)$:

$$T_K(\tau) = 37 - 47e^{-0.0105\tau} \text{ [°C]} \tag{46}$$

The heating rates in Equations (45) and (46) were determined experimentally.

The temperature time-histories calculated using those formulas for the spool and the housing are close to those obtained from experimental tests (Fig. 11).

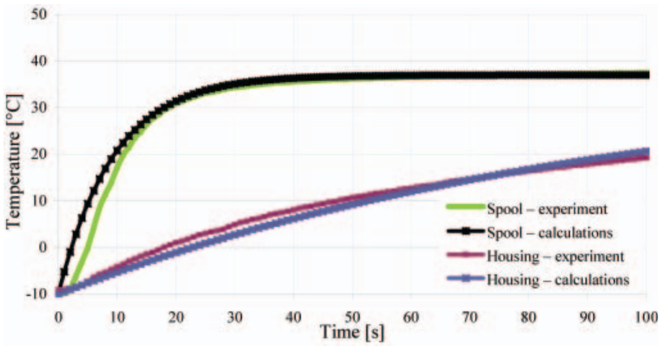

Fig. 11. Temperature time-histories of the spool and housing in spool valve PVG 32 obtained from experimental tests and calculated for initial conditions: spool valve temperature -10 °C, oil temperature 37 °C, flow rate 39 dm3/min

The equation of the clearance time-history between the spool and the housing during the spool valve start-up has the following form:

$$l_{suw} = l_m - \beta_S \cdot h_S \cdot [T_S(\tau) - T_0] + \\ + \beta_K \cdot h_K \cdot [T_K(\tau) - T_0] \tag{47}$$

After placing relations (38) and (39) into Equation 47 we get:

$$l_{suw} = l_m - \beta_S \cdot h_S \cdot [T_{ol} - (T_{ol} - T_{ot})e^{-\sigma_S \cdot \tau} - T_0] + \\ + \beta_K \cdot h_K \cdot [T_{ol} - (T_{ol} - T_{ot})e^{-\sigma_K \cdot \tau} - T_0] \quad (48)$$

where:
S – spool
K – housing.

Effective clearance changes between the spool and the housing having the nominal dimension $h \approx h_S \approx h_K$ is given by the relation:

$$l_{suw} = l_m + h \cdot [(\beta_K - \beta_S) \cdot (T_{ol} - T_0) + \\ + (T_{ol} - T_{ot})(\beta_S \cdot e^{-\sigma_S \cdot \tau} - \beta_K \cdot e^{-\sigma_K \cdot \tau})] \quad (49)$$

For instance, the equation of the effective clearance time-history between the spool and the housing in the single-section spool valve PVG 32 during the start-up from initial conditions: spool valve temperature -10 °C, oil temperature 37 °C, flow rate 39 dm^3/min, heating rates (experimentally determined) for the spool σ_S = 0.106 [1/s] and housing σ_K = 0.0105 [1/s], and the clearance 7 µm, can be written as:

$$l_{suw} = 7 + 0.018 \cdot \\ \cdot [-8.5 + 47 \cdot (11 \cdot e^{-0.106\ \tau} - 10.5 \cdot e^{-0.0105\tau})] \quad [\mu m]\ (50)$$

It results from the characteristics determined with the aid of formula (50) and given in Fig 12, that for from these start-up conditions incorrect spool valve operation will not take place as the minimum clearance for ΔT_{ol-ot} = 47 K is equal to 0.7 µm.

Permissible parameters of the hot oil supplied to the cold spool valve at which the correct valve operation will take place can be calculated. By increasing, in the theoretical analysis, the oil temperature in Equation (49) we get the characteristics of effective clearance changes for increasing temperature differences between the oil and spool valve, (Fig. 12).

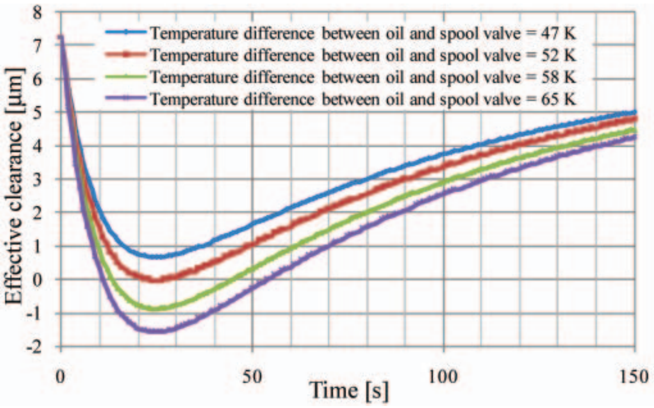

Fig. 12. *Time-history of effective clearance between spool and housing in spool valve PVG 32 for initial conditions: spool valve temperatures -10 °C, oil temperature 37, 42, 48, 55 °C, flow rate 39 dm^3/min*

When the initial temperature difference between the oil and the spool valve is equal to 52 K, the effective clearance will temporarily totally disappear at 25th second (Fig. 12).

When the temperature difference is 65 K (Fig. 12), clearance disappearance and the incorrect operation of the spool valve will be observed between second 12 and 55.

The clearance changes calculated using the analytical method are very close to those obtained from the experiment.

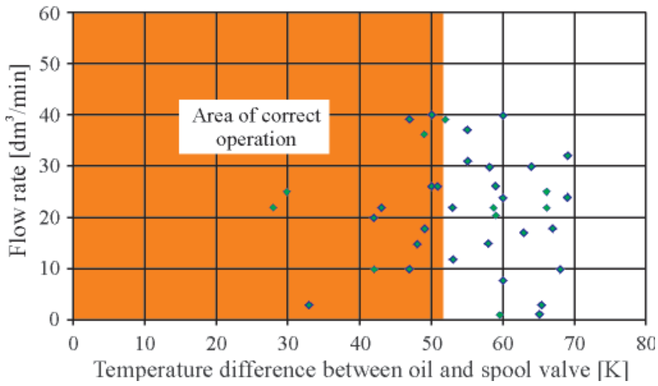

Fig. 13. *Areas of parameters of correct and incorrect operation of spool valve PVG 32 with clearance 7 µm*

When the temperature difference between the oil and the valve exceeds 52 K, the spool valve PVG32 operates incorrectly in thermal shock conditions (Fig. 13). Here, the calculated results are in full agreement with the experimental tests. In those tests faulty operation of the spool valve was observed when the temperature difference between the oil and the spool valve was larger than 54 K.

Characteristics of the computer simulation method

The computer simulation method for determining the serviceability of hydraulic components in thermal shock conditions makes use of computed temperature distributions in hydraulic component elements. These temperature distributions make it possible to calculate the effective clearance between cooperating component elements using relation (25).

Computing the effective clearances during the component start-up requires the following input data: initial temperature of the component, oil temperature and heat transfer coefficient (stepwise flow rate).

When deciding to solve this task, the author had to select a relevant numerical method.

There are a number of basic numerical codes which make use of the finite element method FEM [3, 6, 9, 10, 11], including: NASTRAN, ABAKUS, ANSYS, COSMOS/M and others.

The numerical calculations of temperature distributions were performed for a number of hydraulic components, including: satellite motors SOK 100 and SOK 1600 made by Hydroster, orbital motor GMR 160 made by Rexroth [3], axial piston pump PWK 27 made by Hydrotor, axial piston pump PV 16 made by Parker, gear pump PZ-2-K-6,3 made by Hydrotor, orbital motor TF 170 made by Parker, radial piston motor MR1000 made by Parker, gear motor PZ-2-K-10 made by Hydrotor, hydraulic cylinder CJ2F-50/28/250 made by Stalko, spool valve RE2510/101 made by Hydrotor, proportional spool valve PVG 32 made by Sauer Danfoss, and hydraulic servovalve 4WS2EM10 - 45/20 made by Rexroth.

Along with temperature distributions, the numerical calculations also produced the time-histories of average temperatures in component elements.

Comparing the numerical results with those obtained from experimental tests has made it possible to conclude that the time-histories of average temperatures of the examined elements are very close to each other.

SUMMARY AND CONCLUSIONS

The summary and conclusions concerning methods used for determining permissible start-up parameters for hydraulic components in low ambient temperatures is given below:

- A basic design factor, deciding about permissible start-up parameters of a hydraulic component in thermal shock conditions, is the effective clearance which defines dimensions of the clearances between the surfaces of the cooperating elements. The effective clearance can be calculated for an arbitrary component based on temperature time-histories of its mobile and fixed elements. If the effective clearance is positive, the component will work faultlessly.
- There are three methods of determining parameters for faultless start-up of the hydraulic component in low ambient temperatures, which are: the experimental method, the analytical method, and the computer simulation method.
- Using the analytical method we can determine the area of correct operation of an arbitrary not loaded hydraulic component during its start-up in thermal shock conditions, depending on the oil flow rate and the temperature difference $\Delta T_{ol-ot} = (T_{ol} - T_{ot})$ between the oil and the environment, according to the following inequality:

$$l_m - h \cdot [(T_{ol} - T_0) \cdot (\beta_r - \beta_n) + \\ + (T_{ol} - T_{ot}) \cdot (\beta_n \cdot e^{-\sigma_n \tau} - \beta_r \cdot e^{-\sigma_r \tau})] > 0 \tag{51}$$

  The variables in this inequality are: the oil temperature T_{ol}, the ambient temperature T_{ot} and the rates of heating σ_r , σ_n, of mobile and fixed elements. The rates of element heating σ, mainly depend on the average flow velocity of the fluid supplying the hydraulic component.
- The heat transfer coefficient between the oil and the mobile and fixed component elements depends on the average oil flow velocity. The author determined its values experimentally.
- The analytical method of determining faultless component operation can be used in engineering practice. Using this method the area of correct operation of a hydraulic component can be determined with high accuracy.
- The computer simulation method makes it possible to assess the serviceability of the component during its start-up in thermal shock conditions with higher accuracy than the analytical method. The computer simulation method base on the finite element method (FEM) for determining temperature fields in component elements.

The application of the computer simulation method for determining the process of elements heating and the serviceability of hydraulic components will be presented in Part 4 of the article.

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