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MODEL OF THE RATE OF FLOW OF HYDRAULIC FLUID LEAKING THROUGH PRECISION PAIRS IN THE ELECTROHYDRAULIC CONTROLLER OF AN AUTOMATIC TRANSMISSION

MODEL NATĘŻENIA PRZEPŁYWU PŁYNU HYDRAULICZNEGO PRZEZ NIESZCZELNOŚCI PAR PRECYZYJNYCH STEROWNIKA ELEKTROHYDRAULICZNEGO AUTOMATYCZNEJ SKRZYNI BIEGÓW

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Summary

The conditions of operation and functions of automatic transmission fluid (ATF) in the automatic transmission (AT) have been presented. It has been shown that the ATF quality has a significant impact on the correct functioning of an electrohydraulic AT controller. The phenomenon of ATF flow through gaps (clearance) in precision hydraulic pairs, referred to as "internal leakage" and characterized by "internal leakage rate", has been analysed. It has been shown that excessive

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"internal leakage", caused by excessive wear of hydraulic parts and low oil viscosity, may result in a fluid pressure drop in the AT hydraulic system and in incorrect functioning of the transmission unit.

A model of the rate of internal leakage of hydraulic fluid through precision pairs in distribution valves has been proposed. In the model, an equation has been used that defines the rate of internal leakage in a precision hydraulic pair with a concentric annular gap. The model assumptions adopted and the test conditions have been specified. The values of clearance between components of the precision pairs in the electrohydraulic controller were determined by measurements carried out on a real controller that had been normally operated for quite a long time. The related outer diameters of pistons and spools and inner diameters of spool valve sleeves and hydraulic dampers were measured. Multi-variant tests were carried out on the model, with determining the total leakage values for the AT hydraulic system controller with different values of the degree of wear of the precision pairs and at different values of fluid temperature (viscosity and density) for the conditions of vehicle drive in gears "1" and "2".

Keywords: automatic transmissions (AT), hydraulic system, electrohydraulic controller, precision hydraulic pair, internal leakage, ATF flow rate

Streszczenie

Przedstawiono warunki pracy oraz funkcje płynu ATF w automatycznej skrzyni biegów (ASB). Wykazano, że istotny wpływ na prawidłową pracę sterownika elektrohydraulicznego ASB ma jakość płynu ATF. Przeanalizowano zjawisko przepływu płynu ATF przez szczeliny (luzy) hydraulicznych par precyzyjnych „nazywane natężeniem przepływu nieszczelności wewnętrznej" lub „przeciekiem wewnętrznym". Wykazano, że nadmierne „przecieki wewnętrzne", spowodowane nadmiernym zużyciem elementów hydraulicznych i niską lepkością oleju, mogą być powodem spadku ciśnienia płynu w układzie hydraulicznym ASB i przyczyną jej nieprawidłowego jej działania.

Zaproponowano model natężenia przepływu przez nieszczelności wewnętrzne hydraulicznych par precyzyjnych elementów rozdzielczych. Wykorzystano zależność na natężenie przepływu przez nieszczelności wewnętrzne hydraulicznej pary precyzyjnej ze szczeliną pierścieniową koncentryczną. Sformułowano założenia modelu i warunki badań. Wartości luzów par precyzyjnych sterownika elektrohydraulicznego określono podczas pomiarów na obiekcie rzeczywistym - po przebiegu eksploatacyjnym ASB. Pomierzono odpowiednie średnice tłoczków i suwaków oraz średnic wewnętrznych otworów cylindrycznych suwaków rozdzielczych i tłumików hydraulicznych. Przeprowadzono wariantowe badania modelu określając sumaryczne przecieki sterownika układu hydraulicznego ASB z różnymi wartościami stopnia zużycia par precyzyjnych i temperatury (lepkości i gęstości) płynu dla warunków jazdy na przełożeniu „1" i „2".

Słowa kluczowe: automatyczne skrzynie biegów (ASB), układ hydrauliczny, sterownik elektrohydrauliczny, hydrauliczna para precyzyjna, nieszczelności wewnętrzne, model natężenia przepływu płynu ATF.

1. Introduction

In most cases, the modern automatic transmission (AT) is a maintenance-free unit, giving the vehicle user much of driving comfort thanks to the following advantages:

- The transmission of this type is not provided with a conventional clutch; hence, the driver does not have to operate a clutch pedal when driving.
- No manual gear change is needed; thanks to this, the driver may keep both his/her hands on the steering wheel and thus the safety and comfort of driving is improved.
- The vehicle can be driven with very low speeds (creep) in urban traffic congestions.
- Gears are changed practically without momentary loss of the driving force.
- The transmissions of this type are characterized by high durability and reliability and by low failure frequency.

The AT is a device with a high degree of complexity of its construction. It is controlled by means of a complicated hydraulic system, whose major parts are electrohydraulic controller, hydraulic pump with a filter, hydraulic actuators operating the clutches and multiple-disc wet brakes, as well as torque converter and automatic transmission fluid (ATF) cooler [17].

In the electrohydraulic controller, there are several dozen distribution valves and each of them may include a few precision pairs, such as spool (piston) combined with a cylindrical hole in the valve body. A schematic diagram of the hydraulic control system of the ZF4HP-24 transmission has been presented in Fig. 1, where the ATF flow directions for the configuration of the first (D/1) gear have been shown.

In the automatic transmission, the ATF is used as the working medium, which operates in an unfavourable oxidizing environment and is subjected to intensive shearing; these ATF working conditions cause the physicochemical characteristics of the fluid to change during the vehicle operation [2, 4, 6, 10]. The ATF simultaneously performs the functions of hydraulic fluid, working medium in torque converters, gear oil, oil for wet clutches, and coolant for AT components.

The maintaining of the ATF pressure at an appropriate level is a prerequisite for efficient functioning of the hydraulic control system and, in consequence, for correct operation of the automatic transmission. The pressure, in turn, depends to a significant extent on the processes of wear of AT hydraulic system components and ATF ageing. A decline in the kinematic viscosity of the ATF in comparison with that of the fresh fluid results in an increased rate of internal ATF leakage through precision pairs in the distribution valves of the electrohydraulic controller and through other pairs of components interfacing with each other or seals operating under pressure. A decline in the kinematic viscosity of the ATF also impairs the volumetric efficiency and, in consequence, the delivery rate of the gear pump [9]. After a period of AT operation and in specific conditions of vehicle motion, a situation may occur that the demand for the working fluid supply, increased due to high fluid leakage through gaps in precision pairs, would exceed the delivery of the hydraulic pump. In such a case, the course of the working processes in the hydraulic control system would be disturbed, which would result in AT malfunction.

This work was done to assess the impact of increased clearance (wear) in precision pairs within distribution valves and of reduced ATF viscosity on the values of the rate of internal fluid leakage in the electrohydraulic controller and the rate of leakage through other pairs of components interfacing with each other and operating under pressure, with using the model having been built and the data obtained from measurements carried out on the precision pairs in a real system.

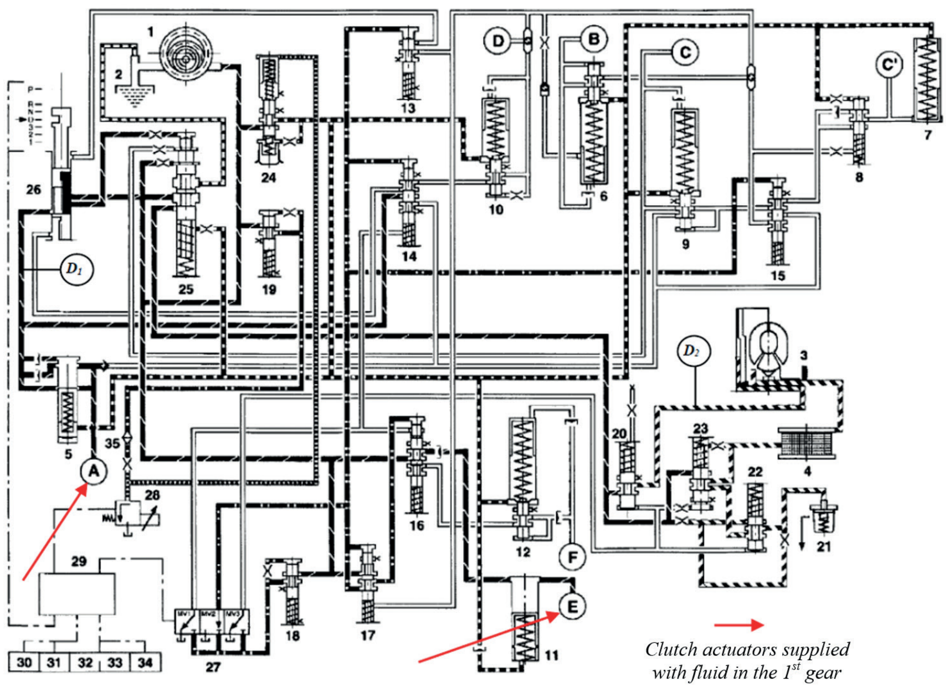


Fig. 1. Schematic diagram of the hydraulic control system of the ZF4HP-24 automatic transmission, with the ATF flow routes having been shown for the selector lever being set to "D" and for the 1st gear being implemented:
1 – hydraulic pump; 2 – hydraulic filter; 3 – torque converter; 4 – ATF cooler; 5 – hydraulic damper of clutch "A"; spool distribution valves: 6 – of clutch "B" with a hydraulic damper; 8 – of brake "C1"; 9 – of brake "C" with a hydraulic damper; 10 – of brake "D" with a hydraulic damper; 12 – of brake "F" with a hydraulic damper; 13 – of the reverse gear; 14, 15, and 16 – of the functions of gear changes 1↔2, 2↔3, and 3↔4, respectively; 17 – of the "kick-down" function; 18 and 19 – pressure distribution valves; 20 – of the torque converter; 22 – lubrication pressure control valve; 25 – main pressure distribution valves; 26 – of the drive mode selection system, mechanically operated: 7 – hydraulic damper of brake "C′"; 11 – hydraulic damper of clutch "E"; 21 – overflow valve of the lubrication system; 23 – lock-up clutch control valve; 24 – pressure-modulating valve; 27 – set of on-off solenoid valves MV1/MV2/MV3; 28 – proportional solenoid valve MV4; 29 – electronic AT (EAT) controller [13]

2. Hydraulic gaps

In moving joints between two parts interfacing with each other, e.g. between a piston and a cylinder, there is a space, often referred to as "gap" and precisely defined by the fit (clearance) provided according to the system assembly requirements. In result of the running-in and then the normal operational wear, the clearance and thus the gap width increases.

As mentioned above, the clearance in the precision hydraulic pair increases during the operation of a hydraulic unit. The value of this clearance consists of the initial (assembly) clearance and the clearance growth resulting from abrasive wear of the precision elements of the pair:

$$c_r = c_m + \Delta c_e \tag{1}$$

where: c_m – assembly radial clearance [mm];

Δc_e – growth in the radial clearance [mm], caused by the operational abrasive wear.

The following factors have an impact on the actual radial clearance, referred to as effective radial clearance and existing during the normal operation of a precision hydraulic pair [14, 15]:

- action of pressure on the control element (mechanical deformation), causing a reduction in the clearance value Δc_c;
- thermal expansion of the distribution valve spool, causing a reduction in the clearance value Δc_{Ts};
- thermal expansion of the electrohydraulic controller body (sleeve), causing a growth in the clearance value Δc_{Tk}.

The pressure-induced clearance change (Δc_c) may be ignored because of low values of the pressure (≤ 1 000 kPa). The impact of thermal expansion of components of precision hydraulic pairs on changes in the radial clearance should be considered significant because the operational temperature T_e of the fluid and transmission components is within a range of 70 90 °C. The difference ΔT between the temperature of measurement of the radial clearance c_r (carried out at an ambient temperature of T_o = 20 °C) and the working temperature of components of precision hydraulic pairs may be 70 °C. Therefore, the equation describing the effective radial clearance (c_{re}) in a precision hydraulic pair has the following form [14, 15]:

$$c_{re} = c_r + \Delta c_{Tk} - \Delta c_{Ts} = c_r + \left(\frac{(\beta_k \cdot D_k \cdot \Delta T_k) - (\beta_S \cdot d_S \cdot \Delta T_S)}{2}\right) \tag{2}$$

where: Δc_{Tk}, Δc_{Ts} – clearance changes resulting from the thermal expansion of the controller body and distribution valve spool, respectively [mm];

β_k, β_S – coefficient of linear thermal expansion of the material of the controller body and distribution valve spool, respectively [1/°C];

D_k, d_s – diameter of the hole in the controller body and of the distribution valve spool, respectively [mm];

ΔT_k, ΔT_S – difference between the actual working temperature of the controller body and distribution valve spool, respectively, and the temperature of these parts during dimension measurements.

3. Leakage in automatic transmissions

In the precision pairs incorporated in the distribution valves of the hydraulic control system, appropriate assembly clearance should be provided between pair components. In result of the running-in and then the normal operational wear, the clearance increases. The difference in the hydraulic fluid pressure across the hydraulic gap causes the fluid to flow (leak) through the gap (clearance) between elements of the precision hydraulic pair, which is an unfavourable phenomenon from the point of view of losses of the fluid stream. This flow is referred to as "internal leakage" and its intensity is defined by the "rate of internal leakage" [23]. There are many such pairs in the automatic transmission, where undesirable ATF flow (internal leakage) takes place. A single distribution valve may include a few precision pairs, which mean a number of hydraulic gaps between plenums to which hydraulic fluid flows at different pressure values.

The sum of the rates of internal leakage at all the elements of the precision hydraulic pairs that are involved in the fulfilment of a specific function of the system is referred to as total rate of internal leakage in the electrohydraulic controller.

In certain conditions of vehicle motion, a situation may occur that the demand for the working fluid supply, increased due to high fluid leakage through gaps in precision pairs, would exceed the delivery of the hydraulic pump. If this is the case, the course of working processes of the hydraulic control system may be temporarily disturbed (by pressure drops), which would result in AT malfunction.

There are also some good points of the internal leakage through the gaps. A very small ATF flow in the friction area lubricates the parts of a precision hydraulic pair. The ATF flows from the plenum with a higher fluid pressure to the plenum with a lower fluid pressure. The flow is laminar and the flow rate depends on gap geometry and ATF properties. The leakage rate is directly proportional to the pressure drop across the gap. Assuming the hydraulic valve spool being ideally cylindrical, the existing formulas that define internal leakages have been determined from empirical tests. In the said formulas, the roughness of the interfacing surfaces has not been taken into account because only one grade of tolerance, obtainable from the machining type used (lapping), is assumed for all precision hydraulic pairs. The existing theoretical models make it possible to determine the total rate of internal leakage in precision hydraulic pairs of an electrohydraulic AT controller in various operation states, with taking into account the wear of precision pairs and the ATF properties. The equation for the rate of internal leakage q_{ve} in a precision hydraulic pair with a concentric annular gap (Fig. 2) has the following form, according to Blackburn J. F., Reethof G., and Shearer J. L. [23]:

$$q_{ve} = 10^6 \cdot \frac{(p_l - p_0)\cdot\pi\cdot d_1\cdot c_r^3\cdot\rho\left(\frac{p_l+p_0}{2}\right)}{12\cdot\mu\cdot l_c\cdot\rho(0)}\ [\mathrm{mm^3/s}] \tag{3}$$

When the value of the rate of internal leakage in a precision hydraulic pair is being determined, the temperature-dependent change in kinematic viscosity and the thermal expansion of the hydraulic fluid (density change) should be taken into account.

At pressures within a range of 100-30 000 kPa (0.1-30 MPa), as prevailing in AT hydraulic systems [12], the ATF density does not significantly change; therefore, changes in this density related to pressure variations may be ignored [23]. At such an assumption, the equation (3) for the rate of internal leakage in a precision hydraulic pair may be simplified as follows:

$$q_{ve} = 10^6 \cdot \frac{(p_l - p_0)\cdot\pi\cdot d_1 \cdot c_r^3}{12\cdot\nu\cdot\rho(0)\cdot l_c} \tag{4}$$

where: p_l – pressure on the supply side [kPa];

p_0 – pressure on the leakage side [kPa];

$(p_l - p_0)$ – pressure drop across the gap [kPa];

d_1 – outer diameter of the spool (shaft) of the precision hydraulic pair [mm];

c_r – radial clearance of the precision hydraulic pair [mm];

$\rho(0)$ – ATF density at the atmospheric pressure [g/cm^3];

$\rho\left(\frac{p_l+p_0}{2}\right)$ – ATF density [g/cm^3]

μ – dynamic ATF viscosity; $\mu = \nu \times \rho$ [Pa×s]; ν – kinematic ATF viscosity [mm^2/s];

l_c – length of the area of contact between the cylindrical part of the spool (shaft) and the cylindrical surface of the sleeve (hole) of the precision hydraulic pair [mm].

The value of the rate of internal leakage q_{ve} decreases linearly with rising kinematic viscosity ν and increases with the third power of the radial clearance c_r.

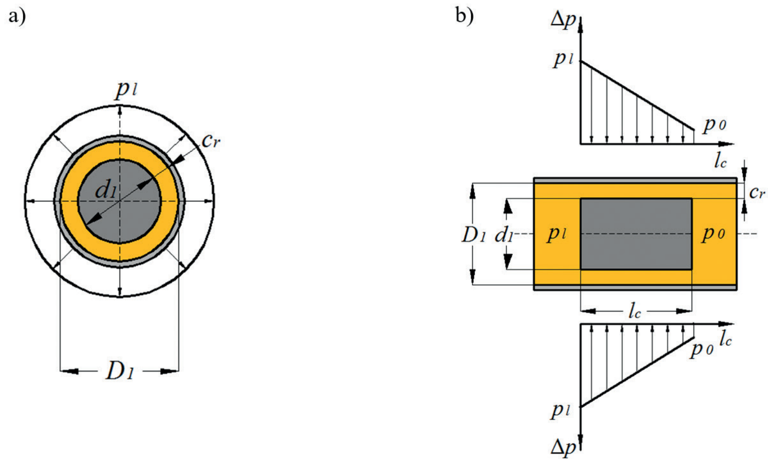


Fig. 2. Schematic diagram of a precision hydraulic pair with a concentric annular gap and pressure distribution in the gap: a) cross-section of the gap; b) longitudinal section of the gap; D_1 – inner diameter of the sleeve; d_1 – outer diameter of the spool; p_l – pressure upstream of the gap (supply side); p_0 – pressure downstream of the gap (leakage side) [23, 25]

If the gap is sufficiently narrow (capillary), where the radial clearance is 4-20 μm, the phenomenon of obliteration occurs. For the popular hydraulic fluids, the maximum thickness of the boundary layer at which the gap may be obliterated is 4-5 μm. In the case of fluid flow through narrow gaps, such a layer may build up in a significant part of the nominal cross-section of the gap. The intensity of the obliteration process depends to a considerable degree on the time of the hydraulic element being under fluid pressure and on the value of the pressure drop across the gap (the intensity increases with this drop). Working movements of the hydraulic element destroy the gap obliteration. Total obliteration of the gap only occurs in very narrow gaps, less than 5 μm wide. For wider gaps ($c_r \geq 20$ μm), a decrease in the effective cross-section of the gap is observed, which causes a reduction in the rate of flow of the hydraulic fluid through the gap. The intensity of the obliteration process increases with growing contamination of the hydraulic fluid. In gaps with a width exceeding 20 μm, the obliteration process practically does not occur and it may be ignored when the rate of ATF flow through the gap in a precision pair is calculated [25].

4. Mathematical model

At present, mathematical modelling has become a research tool increasingly often used to analyse the working processes that take place in hydraulic systems [22, 24, 26]. The impact of wear of precision pairs in the AT hydraulic system on the values of the rate of internal ATF leakage may be assessed by analysing the results of testing a model of internal leakage in the electrohydraulic AT controller.

The value of the rate of internal leakage provides information that makes it possible to determine how much of the instantaneous delivery of the hydraulic pump is lost for the internal ATF leakage in the electrohydraulic controller.

Fig. 3 presents, as an example, a structural model of internal leakage in the electrohydraulic AT controller in the configuration of the 2nd gear. The structural models for the configuration of individual gear ratios are built by separating the elements from the complete model of the electrohydraulic AT controller that currently are in contact with the hydraulic fluid under pressure. The substructures of this type may be mathematically described in respect of internal leakage for the currently interfacing surfaces of precision pairs in a specific hydraulic element.

The following assumptions were made when the internal leakages in the electrohydraulic controller were modelled:

- In the model, stationary states of the positions of spools in individual distribution valves had to be represented.
- The equations representing the variations of pressures (p_s, p_m) in the other sections, which were separated from the main pressure p_g, had to be determined analytically (due to the impossibility of carrying out measurements on the real system).
- The equation describing the pressure in the outlet controller ports had to be formulated theoretically, as it was adopted at the design stage, i.e. this pressure should not exceed 10 % of the value of the main pressure p_g.

- For all the precision hydraulic pairs, an assumption was made that the hydraulic fluid leaked through concentric annular gaps.
- The phenomenon of obliteration of the gap was not taken into account in the calculations in consideration of high values ($c_r > 20$ μm) of the clearance in the precision hydraulic pairs as measured on real objects and insignificant values (not exceeding 1 000 kPa) of the pressure drop across the hydraulic gap.
- The impact of thermal expansion of the materials used on the radial clearance in the precision hydraulic pairs (effective clearance c_e) was taken into account.
- The temperature of the hydraulic elements was assumed as identical with that of the ATF.

Furthermore, the ATF supply method and the positons of distribution valve spools in the electrohydraulic controller body are taken into account in the structural models. Based on this, the total internal leakage through a precision hydraulic pair may be described by a mathematical equation.

In the system configuration of the 2nd gear, the equation for the total leakage in the hydraulic system takes the following form:

$$\Sigma q_{D/2} = \Sigma q_{p_{g2}} + \Sigma q_{p_{s_2}} + \Sigma q_{p_{m_2}} + \Sigma q_{p_{k_2}} \ [\mathrm{mm^3/s}] \qquad (5)$$

where: $\Sigma q_{p_{g2}}, \Sigma q_{p_{s_2}}, \Sigma q_{p_{m_2}}, \Sigma q_{p_{k_2}}$ – total leakages from the sections fed with fluid under the main, control, modulated, and torque converter pressures, respectively.

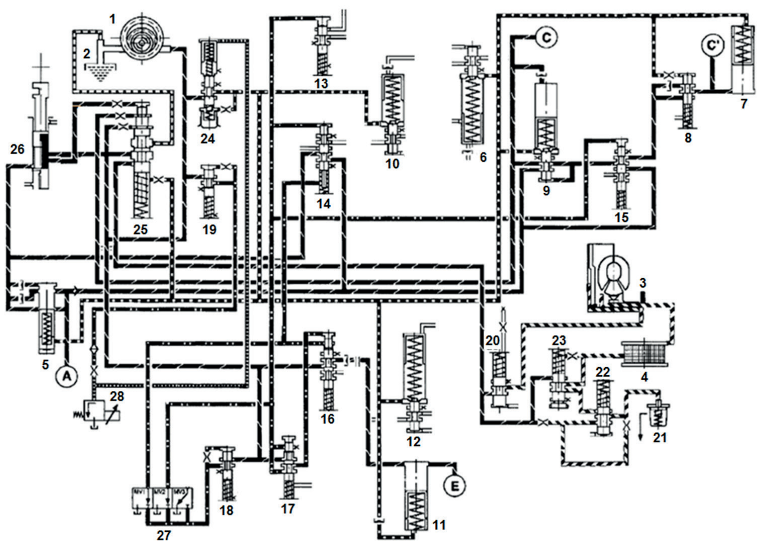


Fig. 3. Structural model of internal leakage in the electrohydraulic AT controller in the configuration of the 2nd gear (for the meaning of the reference numbers of individual components of the hydraulic control system see Fig. 1) [13]

For the configuration of the 2nd gear, the equation for the total leakage from the section fed with fluid under the main pressure p_g, for hydraulic dampers (i = 5, 7, 11) and spool distribution valves (i = 8, 9, 14, 15, 16, 17, 20, 23, 25, 26), has the following form (cf. Fig. 4):

$$\Sigma q_{p_{g2}} = \Sigma q_{p_{g2}5} + \Sigma q_{p_{g2}7} + \Sigma q_{p_{g2}11} + \Sigma q_{p_{g2}8} + \Sigma q_{p_{g2}9} + \Sigma q_{p_{g2}14} + \Sigma q_{p_{g2}15} + \Sigma q_{p_{g2}16} + \\ + \Sigma q_{p_{g2}17} + \Sigma q_{p_{g2}20} + \Sigma q_{p_{g2}23} + \Sigma q_{p_{g2}25} + \Sigma q_{p_{g2}26} \ [\mathrm{mm^3/s}] \tag{6}$$

As an example, for hydraulic damper "5" in the configuration of the 2nd gear, the equation for the total leakage from the section fed with fluid under the main pressure has the form:

$$\Sigma q_{p_{g2}5} = q_{p_{g2}5-1} = 10^6 \cdot \frac{(p_g - p_m)\cdot\pi\cdot d_{1-5}\cdot c_{re1-5}^3}{12\cdot\upsilon\cdot\rho(0)\cdot l_{c1-5}} \ [\mathrm{mm^3/s}] \tag{7}$$

In contrast, for distribution valve "8" in the configuration of the 2nd gear, the equation for the total leakage from the section fed with fluid under the main pressure has the form:

$$+(10^6 \cdot \frac{(p_g - 0{,}1\cdot p_g)\cdot\pi\cdot d_{2-8}\cdot c_{re2-8}^3}{12\cdot\upsilon\cdot\rho(0)\cdot l_{c2-8}}) \ [\mathrm{mm^3/s}] \tag{8}$$

The equations for the total leakage from the section fed with fluid under the main pressure p_g for hydraulic dampers (i = 7, 11) and spool distribution valves (i = 9, 14, 15, 16, 17, 20, 23, 25, 26) in the configuration of the 2nd gear may be written in a way as presented above.

The same approach was adopted to formulate the equations for total leakages $\Sigma q_{p_{s_2}}$, $\Sigma q_{p_{m_2}}$, $\Sigma q_{p_{k_2}}$ from the sections fed with fluid under the control, modulated, and torque converter pressures, respectively.

5. Determining the important dimensions of precision hydraulic pairs

For the values of the total rate of internal ATF leakage in the electrohydraulic controller to be modelled, certain geometrical characteristics of the precision hydraulic pairs in the controller, important from the point of view of internal leakage, had to be determined. With this objective in view, the electrohydraulic controller was dismantled and the following dimensions were measured:

- outer diameter of distribution valve spools and hydraulic damper pistons, d_i [mm];
- inner diameter of cylindrical holes (sleeves) of spool distribution valves and hydraulic dampers, D_i [mm];
- length of the area of contact between the cylindrical part of the spools (pistons) in distribution valves and hydraulic dampers and the interfacing surfaces of cylindrical holes, l_c [mm].

Schematics of typical precision hydraulic pairs in an electrohydraulic AT controller have been shown in Fig. 4.

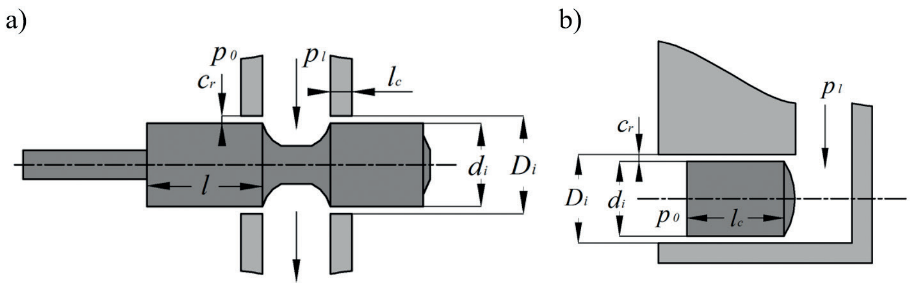


Fig. 4. Schematics of precision hydraulic pairs in an electrohydraulic controller: a) spool distribution valve; b) hydraulic damper; p_0, p_l – pressures; l – length of the cylindrical part of the spool

The outer diameters of distribution valve spools and hydraulic damper pistons and the inner diameters of the sleeves were measured with the use of a toolroom microscope УИМ-21 (UIM 21) with an accuracy of 0.001 mm. Based on this, the radial clearance c_r of precision hydraulic pairs was calculated from the formula:

$$c_{r(i)} = \frac{(D_i - d_i)}{2} \text{ [mm]} \tag{9}$$

The diameters were measured in two lateral planes (1-1 and 2-2) and in two directions perpendicular to each other (x1-x2 and y1-y2), as shown in Fig. 5.

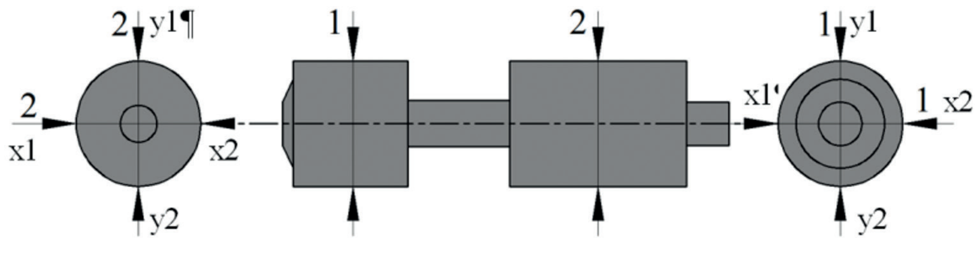


Fig. 5. Schematic presentation of the diameter measurement places for the spool of distribution valve 6 (reference number as shown in Figs 1 and 3)

Each of the measurements was repeated twice. The length l_c of the area of contact between the cylindrical hole and the interfacing spool surface in distribution valves and between the piston and the interfacing sleeve surface in hydraulic dampers was measured by a dial calliper with 0.02 mm minimum graduation of the dial indicator at an ambient air temperature of t_H = 20±0.5 °C.

Example measurement results obtained for selected precision pairs of the electrohydraulic controller (Fig. 6) have been given in Table 1.

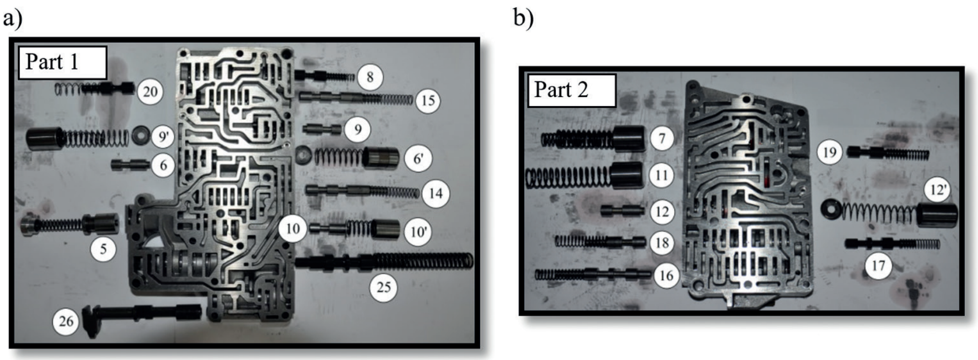
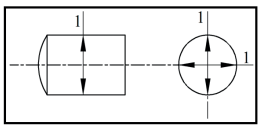
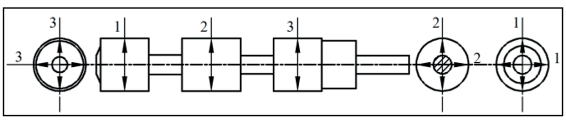


Fig. 6. Electrohydraulic controller blocks: a) Part 1; b) Part 2

Table 1. Diameters and contact area lengths measured and radial clearance values calculated

<table>
<tr><th>Component No</th><th>Component name</th><th colspan="6">Measurement and calculation results</th></tr>
<tr><td rowspan="4">12'</td><td rowspan="2">Hydraulic damper of brake "F"</td><td>Place of measurement</td><td>d [mm]</td><td>D [mm]</td><td>c_r [mm]</td><td>l [mm]</td><td>l_c mm</td></tr>
<tr><td>1</td><td>19.945</td><td>19.993</td><td>0.024</td><td>27.0</td><td>27.0</td></tr>
<tr><td>Component view</td><td colspan="6">Place of damper measurements, schematic</td></tr>
<tr><td>
Material: steel</td><td colspan="6">
</td></tr>
<tr><td rowspan="6">14</td><td rowspan="4">Spool distribution valve of gear change 1↔2</td><td>Place of measurement</td><td>d [mm]</td><td>D [mm]</td><td>c_r [mm]</td><td>l [mm]</td><td>l_c mm</td></tr>
<tr><td>1</td><td>9.949</td><td rowspan="3">19.993</td><td>0.024</td><td>27.0</td><td>27.0</td></tr>
<tr><td>2</td><td>9.947</td><td>0.0205</td><td>10.3</td><td>2.5</td></tr>
<tr><td>3</td><td>9.947</td><td>0.0205</td><td>10.2</td><td>2.5</td></tr>
<tr><td>Component view</td><td colspan="6">Place of valve measurements, schematic</td></tr>
<tr><td>
Material: steel</td><td colspan="6">
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</table>

6. Analysis of the results of testing the model of internal leakage in the controller

The calculations of internal leakage through gaps between the interfacing surfaces of precision pairs in the controller were carried out for several ATF temperature values and for various degrees of wear of controller components. With changes in the ATF temperature Te, the values of kinematic viscosity ν and density ρ of the hydraulic fluid and of the effective radial clearance c_{re}, which are taken as input data for the calculations of the rate $\sum q_i$ of internal ATF leakage, change as well. Individual variants taken into account at the modelling of internal leakage through gaps between the interfacing surfaces of precision pairs in the electrohydraulic controller have been presented in Table 2.

In the first variant (**W1**), the following assumptions were made:

- The operational ATF temperature value was adopted as 80 °C.
- The values of the effective radial clearance c_{re} in precision hydraulic pairs were assumed as identical with those determined by measurements of real controller components but adjusted for the impact of the ATF temperature to be taken into account.
- The kinematic viscosity of the fresh hydraulic fluid was adopted.

In the second variant (**W2**), the kinematic viscosity value ν was changed to that obtained from measurements carried out on a sample of the hydraulic fluid used in the transmission normally operated in a vehicle that covered a mileage of S = 106 315 km [8].

The third variant (**W3**) differed from the first one (**W1**) in that for all the precision hydraulic pairs in the controller, the radial clearance values were increased by a hypothetical degree of wear equal to z_a = 10 %.

In the variants from fourth (**W4**) to eighth (**W8**), the radial clearance values were increased for all the precision hydraulic pairs in the controller by a hypothetical degree of wear equal to z_a = 50 %.

The hypothetical degree of wear za defines the percentage increase in the radial clearance in a precision hydraulic pair in relation to the clearance determined by measurements and calculations of diameters of spools (pistons) di and sleeves Di in the electrohydraulic controller body. The radial clearance in a precision hydraulic pair in the controller, raised by the hypothetical degree of wear za, is defined by an equation:

$$c_{r(i)\,z_a\%} = \frac{(D_i - d_i)}{2} + \frac{z_a}{100} \cdot c_{r(i)}\ [\mathrm{mm}] \tag{10}$$

In the model tests, the radial clearance values were increased for all the precision hydraulic pairs in the electrohydraulic controller by a hypothetical degree of wear za in relation to the clearance values determined by measurements carried out on real precision pairs.

The properties of the automatic transmission fluid, which should meet the specifications of DEXRON ATF IID, according to the operation manual of the transmission under test [13, 18, 21], were modelled for the temperature range T_e = –40-100 °C with using data given in the HIPOL ATF IID product data sheet [20]. When determining approximate changes in the values of kinematic viscosity v and density ρ of the ATF IID hydraulic fluid with temperature (Fig. 7), the graphs and equations published in [16, 19] were employed.

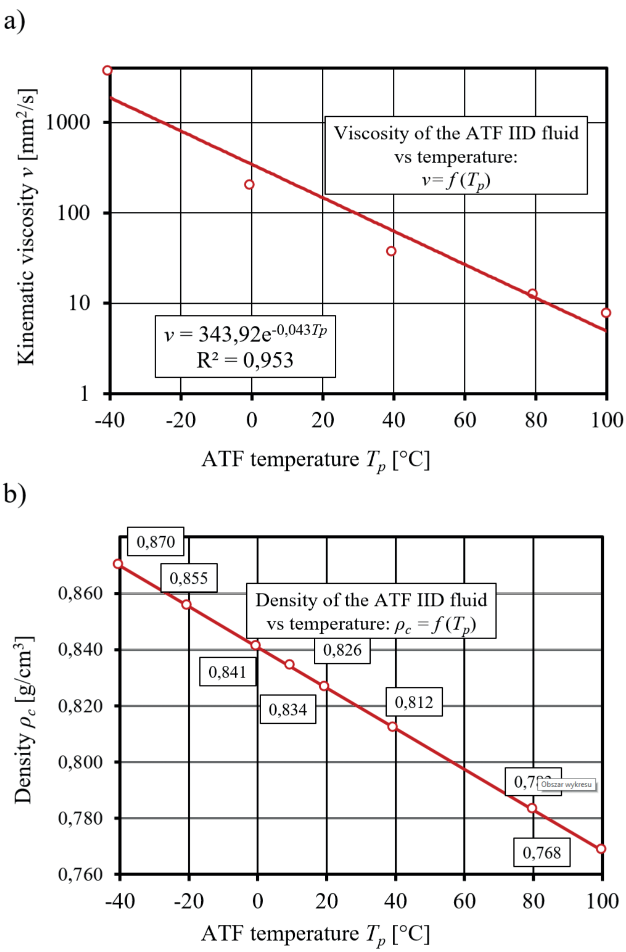

Fig. 7. Characteristics of the ATF IID fluid vs temperature:
a) kinematic viscosity v; b) density ρ [19]

The model test results in the form of results of calculation of the total rate of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs in the electrohydraulic controller of automatic transmission ZF4HP-24 in the conditions of the minimum vehicle acceleration test (drive in gear "1", acceleration, "1→2" gearshift process, drive in gear "2") lasting for t_p = 9 s have been presented in Figs 8 and 9 and in Table 2.

The "1→2" gearshift process begins at the instant of t_p = 3.5 s and ends after the instant of about t_p = 4.8 s. In the graph, this is reflected in a rapid growth in the delivery rate Q_p of the hydraulic pump (up to 33 dm³/min), directly related to the engine speed growth during acceleration in gear "1". Following the engagement of gear "2" (t_p 4.8 s), the pump delivery rate stabilizes at a level of about 33 dm³/min, as the vehicle is now accelerated in gear "2". The hydraulic gear pump in the automatic transmission is directly driven by the engine crankshaft; therefore, changes in the pump delivery rate directly depend on changes in the engine speed.

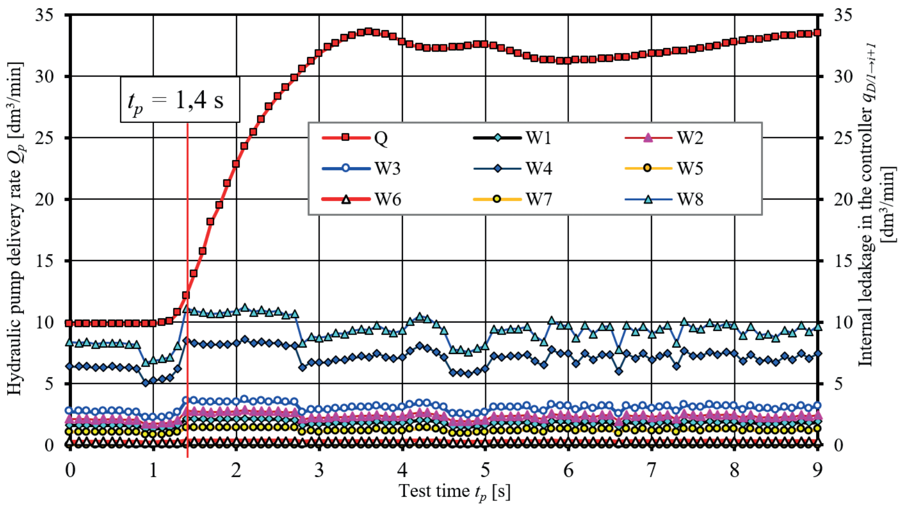

Fig. 8. Time history of the internal leakage rate $q_{D/1\to i+1}$ in the electrohydraulic controller in the conditions of the minimum vehicle acceleration test in gears "1" and "2", for the calculation variants from W1 to W8

Table 1. Diameters and contact area lengths measured and radial clearance values calculated

Variant No	W1	W2	W3	W4	W5	W6	W7	W8
ATF temperature T_e [°C]	80	80	80	80	-40	20	40	100
Kinematic viscosity ν [mm²/s]	11.1	11.1	11.1	11.1	1 922.7	145.5	61.5	4.65
Density ρ [g/cm³]	0.783	0.783	0.783	0.783	0.870	0.826	0.812	0.768
Hypothetical degree of wear z_a [%]	0	0	10	50	50	50	50	50
Internal leakage rate $q_{D/1\to i+1}$ [dm³/min]	0.9-2.16	1.67-2.81	2.16-3.64	5.09-8.46	max. 0.021	max. 0.38	max. 1.41	6.67-11.23
Relative delivery rate of the hydraulic pump [%]	17.7	23.1	29.9	69.5	0.172	3.12	11.6	92.2

The pump delivery characteristic curve was determined for the needs of the model tests, based on real geometric dimensions of the pump (the pump displacement was calculated) and on engine speed values measured during experimental field tests. The nature of changes in the leakage rate depends on the time history of the main pressure p_g in the hydraulic system. The operational values of the main pressure p_g in the hydraulic system of the AT under test were within a range of 400-900 kPa (0.4-0.9 MPa), depending on the system setting. The said curve was obtained from experimental field tests and implemented in the form of the input data used in the model tests.

Regardless of the modelled variant of the electrohydraulic controller operation, the values of the rate of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs changed in a stepwise manner during the whole test period of t_p = 9 s, but remained at a specific average level. For variants **W6**, **W3**, and **W8**, the average values of the rate of internal leakage through precision hydraulic pairs were q_D = 0.32 dm³/min, q_D = 3.2 dm³/min, and q_D = 9.3 dm³/min, respectively.

It can be seen from Fig. 8 that a critical period of operation of the electrohydraulic controller lasted until the test time came to a value of t_p = 1.4 s. The pump delivery rate, increasing (from the instant of t_p = 1 s) with the engine speed, reached then a value of Q_p = 12.18 dm³/min, which only slightly exceeded the value of the rate of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs. For variants **W8** and **W4**, the internal leakage rate values were q_D = 11.2 dm³/min and q_D = 8.46 dm³/min, respectively.

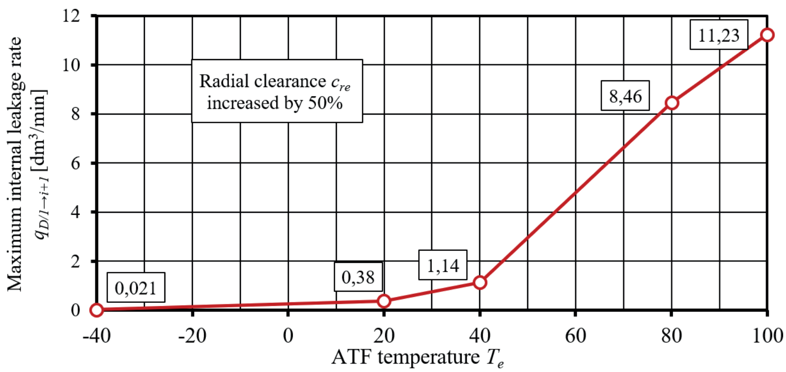

Fig. 9. Maximum values of the rate of internal leakage $q_{D/1\to i+1}$ in the electrohydraulic controller vs ATF temperature Te for the test conditions covering drive in gear "1", acceleration, "1→2" gearshift process, and drive in gear "2", with an assumption of significant wear of the interfacing surfaces of the precision hydraulic pair (i.e. with the radial clearance values c_{re} increased by 50 %)

The values of the rate of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs rise with growing ATF temperature (declining viscosity of the fluid) and with increasing wear of the precision hydraulic pairs (growing radial clearance).

When the ATF temperature rises from –40 °C to 100 °C (and the fluid viscosity drops accordingly), the temperature of components of precision hydraulic pairs rises as well; then, without any change in the degree of wear of these components (z_a = 50 %), the rate of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs increases, too; this increase becomes much steeper when the fluid temperature exceeds t_p = 40 °C (Fig. 9). When the fluid temperature reaches a level of t_p = 100 °C, the rate of internal leakage through precision hydraulic pairs rises tenfold to reach a range of $q_{D/1}$ = 6.67-11.23 dm^3/min, which makes more than 92 % of the hydraulic pump delivery rate Q_p for the test time of t_p = 1.4 s.

If the ATF temperature remains unchanged (at a level of t_e = 80 °C), the rate of internal leakage is determined by the wear of components of precision hydraulic pairs. With increasing degree of wear z_a of the interfacing surfaces of the hydraulic pairs, the rate of internal leakage increases as well; for z_a = 50 %, the value of this rate rises fourfold to a level of $q_{D/1}$ = 5.09-8.56 dm^3/min, i.e. to more than 70 % of the hydraulic pump delivery rate Q_p.

7. Recapitulation

The model tests were carried out to determine the impact of abrasive wear of the interfacing surfaces of electrohydraulic controller components and of ATF temperature T_e (i.e. fluid viscosity) on the total rate $q_{D/1\rightarrow i+1}$ of internal leakage in precision hydraulic pairs of an electrohydraulic controller.

A high difference in the hydraulic fluid pressure across the hydraulic gap between the interfacing surfaces of precision hydraulic pairs causes the fluid to flow (leak) through the gap (clearance) between elements of the precision hydraulic pair. This is an unfavourable phenomenon from the point of view of losses of the fluid stream; it is referred to as "internal leakage" and its intensity is described by "internal leakage rate". The sum of rates of internal leakage at all the elements of the precision hydraulic pairs that are involved in the fulfilment of a specific function of the system (total rate of internal leakage in the electrohydraulic controller) may cause a drop in the main pressure p_g below the minimum value that is necessary for the hydraulic control system of the automatic transmission to function correctly.

The value of the total rate of internal leakage in the electrohydraulic controller chiefly depends on the fluid viscosity (i.e. temperature and ageing) and the wear of elements of hydraulic pairs. The values of rates of internal leakage through gaps between the interfacing surfaces of precision hydraulic pairs grow with rising fluid temperature, declining fluid viscosity, and increasing wear of the precision pairs.

An increase in ATF temperature T_e (i.e. a drop in the fluid viscosity) and, in consequence, in the temperature of components of the precision hydraulic pairs, without any change in the degree of their wear, causes the internal leakage to grow. Nevertheless, the rate of this growth definitely rises only when the fluid temperature exceeds t_p = 40 °C. At the fluid temperature of about t_p = 100 °C, the rate of internal leakage through precision hydraulic pairs becomes several times as high as that observed for the cool fluid and it may cause a drop in the main pressure p_g below the minimum value necessary for the electrohydraulic controller and the automatic transmission to function correctly.

The highest rates of internal leakage through precision hydraulic pairs in an electrohydraulic controller occur when the pairs reach a significant degree of operational wear (z_a = 50 %) and at high temperatures. In such a situation, the internal leakage makes a great part (even 92 %) of the instantaneous hydraulic pump delivery rate Q_p, which may result in a drop in the main fluid pressure p_g below the minimum value required for correct functioning of the AT hydraulic control system.

The leakage through precision pairs in the controller, resulting from operational wear and reduced ATF viscosity, combined with internal leakage in other system components and with lowered volumetric efficiency η_{vp} of the pump, may cause a drop in the main ATF pressure p_g below the value that is indispensable for, and ensures, efficient functioning of the AT hydraulic control system.

The model tests were carried out for the AT working conditions representing the vehicle drive in the first and second gear only. It is advisable to carry out model tests for the automatic transmission being configured to operate in each of the gears and for various conditions of normal vehicle drive, including creeping and maximum acceleration over the whole range of transmission ratios.

The full text of the article is available in Polish online on the website http://archiwummotoryzacji.pl.

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