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Flow-Force Analysis in a Hydraulic Sliding-Spool Valve

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Ključne riječi

Analiza strujnih sila
Hidraulički ventil sa uzdužnim klipom
Kompenzacija strujnih sila
Rub klipa izlaznog strujanja
Rub klipa ulaznog strujanja
Strujne sile
Strujni tok fluida

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The necessary forces for the control of a hydraulic sliding-spool valve are normally determined by the stationary axial momentum forces of the oil flow through a valve. These forces increase with an increase in the volume flow and the pressure difference. Because of the limited actuating-force potential of normal electromechanical actuators, the direct actuation of conventional valves is limited to small dimensions of valves. To make possible the use of directly actuated valves also for higher hydraulic power and to utilize the direct valve actuation also for higher flows and pressures, the flow forces acting on the valve piston in the axial direction must be compensated. This is possible with an appropriate flow stream of fluid through the valve through which the axial component of the flow forces in a conventional valve is reduced. This research work presents an analysis and compensation of the flow forces through an appropriate construction of a hydraulic piston valve of nominal size 6.

Analiza strujnih sila u hidrauličkom ventilu sa uzdužnim klipom

Izvornoznanstveni članak

Potrebne sile za upravljanje hidrauličkim ventilom ovise u velikoj mjeri od stacionarnih aksijalnih impulznih sila strujanja ulja kroz ventil. Ove sile povećavaju se s porastom strujnog toka ulja i razlike tlakova. Zbog ograničenih sila za upravljanje ventilom, koje omogućuju uobičajeni elektromehanički pretvornici, kod konvencionalnih ventila direktno je upravljanje ograničeno na male dimenzije ventila. Da bi omogućili upotrebu direktno upravljanih ventila i za velike strujne tokove i tlakove, potrebno je smanjiti strujne sile koje djeluju na klip ventila. To se može postići primjernim vođenjem strujanja fluida kroz konvencionalni ventil tako da dođe do smanjenja odnosno kompenziranja aksijalnih komponenata visokih sila strujnog toka u ventilu. U okviru ovog istraživačkog rada analizirane su, i s odgovarajućom konstrukcijom minimizirane, strujne sile u jednom hidrauličkom ventilu s uzdužnim klipom dimenzije NG 6.

1. Introduction

Hydraulic drives (power trains) are nowadays used in many technical applications in which they are confronted with continuously increasing demands, especially those related to hydraulic power and the dynamics of systems. More intensive use of microelectronics in recent times has strengthened the existing application fields of the hydraulic drive and, what is even more important, has made possible completely new fields of application. However, despite the extended possibilities of the system techniques, it will also be very important in the future to pay more attention to the design and construction improvements of individual hydraulic components.

Basically, fluid power valves can be classified as non-continuously operating (e.g., switching and shut-off valves) and continuously operating valves (e.g., proportional and servo valves) [2]. In previous years, continuously operating valves have become more and more important for use in hydraulic circuits because they offer more functions and savings in necessary elements in comparison with simple directional control valves [9]. In the case of proportional and servo valves, the valve position is continuously adjustable and proportional to the electrical input signal. An electrically controlled valve consists of an electromechanical transducer and of a hydraulic power part. The role of the electromechanical transducer is to convert electrical quantities like voltage, current or charge into analogous quantities like force or position.

Another classification of fluid power valves concerns the way the main valve is controlled. In this regard it is possible to distinguish directly controlled valves from indirectly or pilot-operated valves. In comparison with

Symbols/Oznake

F_M	- force of an electromechanical actuator, N - sila elektromehaničkog pretvornika
F_R	- friction force on the sliding spool, N - sila trenja koja djeluje na klip
F_{Str}	- flow force, N - strujna sila
F_F	- spring force, N - sila opruge
F_a	- accelerationg force, N - sila ubrzanja
F_{ax}	- axial force, N - aksijalna sila
Q	- flow rate, 1/min - strujni tok
p	- pressure, bar - tlak
Δp	- pressure difference, bar - razlika tlakova
p_A, p_B	- pressure in the service port, bar - tlak u priključcima potrošnika
p_o	- delivery pressure, bar - snabdjevni tlak
p_T	- pressure in the return (tank) port, bar - tlak u priključku za spremnik
v	- mean stream velocity, m/s - srednja brzina strujnog toka
v_1, v_2	- stream velocity at the control edge, m/s - brzina strujnog toka na krmilnim rubovima klipa
l	- length of the accelerating of volume, m - duljina uljnog stuba za ubrzanje
A_o	- control edge cross-section, m^2 - presjek otvorenja krmilnog ruba klipa
A_{vc}	- stream cross-section, m^2 - presjek strujnog toka
y	- sliding-spool stroke, m - pomak klipa ventila
D	- sliding-spool diameter, m - promjer klipa ventila
α_D	- flow-rate coefficient - koeficijent strujnog toka
ρ	- density, kg/m^3 - gustoća
ε	- stream angle, degree - kut strujanja, º

indirectly controlled valves, directly controlled valves have many advantages, especially with respect to energy. A directly controlled valve does not need any hydraulic pilot valve because the sliding-spool element is directly actuated by the actuator. For this reason all the problems connected with pilot operation, such as high power losses, complicated and costly design of pilot stages, contamination danger and dependence of the valve dynamics on the pilot pressure are removed. The problem occurs because the direct control of hydraulic valves using electromechanical actuators is only practical for small nominal sizes of valves. This happens mainly because of the limited actuating-force potential of electromechanical actuators.

In a hydraulic valve there are the so-called disturbance forces, which are effective against solenoid or control forces. They consist of flow forces, frictional forces, pressure forces and inertial (mass) forces (Figure 1). The frictional force FR consists of a Coulomb friction part, produced by radial forces, and the fluid friction, resulting from the movement of a piston [7, 12]. For normal operation of a hydraulic sliding-spool valve it is necessary to avoid the Coulomb friction forces because they cause the dead-band span (hysteresis) of a valve and consequently demand more force potential from the electromechanical actuator. As a reliable countermeasure for the reduction of the Coulomb friction forces, circumferential grooves have proved to be the most effective [11].

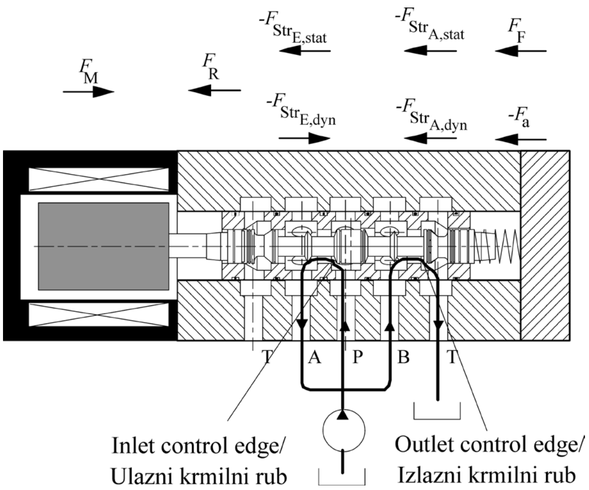

Figure 1. Forces acting on the valve spool (4-chamber structure)

Slika 1. Sile na klipu ventila (4- komorna struktura)

The viscous (Newton) friction forces soften the movement of the sliding spool (piston) and therefore help to stabilize the dynamic behavior of the valve. The reduction of the inertial (mass) forces can be achieved only through the reduction of the sliding spool mass and of the solenoid rotor mass. However, this measure has a positive influence only on the dynamic behavior of the valve. On the other hand, stationary flow forces play a much more important role in the control of higher hydraulic powers, especially when utilizing the direct valve actuation or directly controlled fluid power valves [8].

In order to make the most effective use of the advantages of the compensation method using the abbreviation of the flow stream from the two design principles (fourchamber and five-chamber construction) the four-chamber construction of a valve has been chosen for our research.

This research work presents an analysis and compensation of the flow forces through the appropriate construction of a 4/3-directional hydraulic proportional control piston valve of nominal size 6. The valve was designed using the valve spool-socket configuration in a four-chamber design. The research comprises the analysis and optimisation, e.g., the flow-force compensation of each flow resistance (e.g., two inlet edges and two outlet edges of the valve piston) using the flow-stream abbreviation method.

2. Flow forces and flow-force compensation methods

The expression "flow forces" can be understood as forces that emerge with a change of direction and/or velocity of the flow stream. It is necessary for the valve piston to stay in a balanced condition so that the external reaction forces act on it in the opposite direction. As there are high pressure differences in a hydraulic valve and the flow stream abbreviations occur at high flow velocities, flow forces become very apparent.

In the stationary operation of a hydraulic valve, there are some forces acting on the sliding spool of the valve, which are caused by the momentum of the fluid flowing through the valve. Figure 2 presents a schematic structure of an inlet and outlet edge of a non-compensated hydraulic valve with a valve spool and a socket. Due to the higher pressure potential before the control edge of the sliding spool and the socket and the variable cross-section of an orifice, the fluid-flow velocity at the control edge is always higher than at the valve opening with a relatively large and constant cross-section. According to Bernoulli, an area of lower pressure develops in this region. The pressure value is different for each direction of the fluid flow and acts differently on the circumferential area of the sliding spool. Therefore, a force develops, which results from the pressure distribution in the control chamber of a sliding spool. This force acts in the closing direction of the sliding spool, which means in the opposite direction to the actuator.

The control edge of a sliding spool, which acts as a resistance through which the fluid flows into the slidingspool chamber, is defined as the inlet edge. On the other hand, the control edge where the fluid leaves the chamber is defined as the outlet edge.

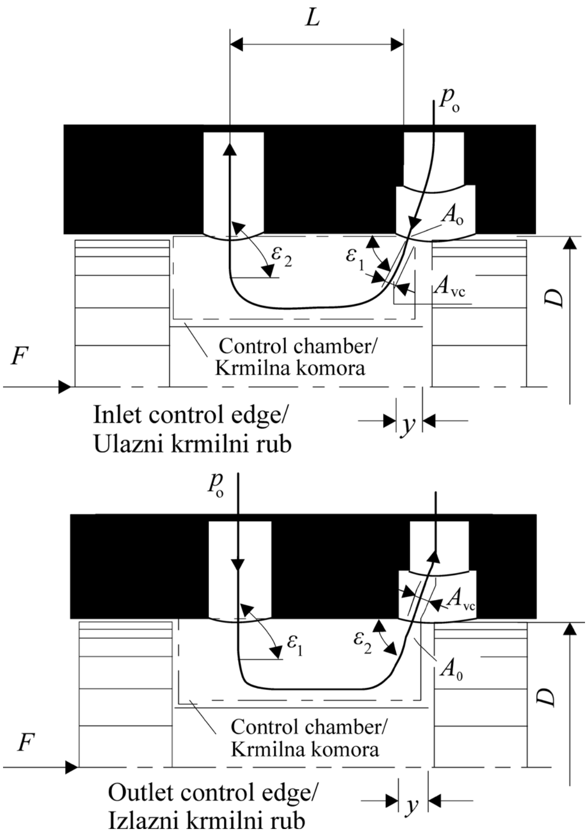

Figure 2. Flow forces acting on the valve spool

Slika 2. Sila strujanja na klipu ventila

The entry angle of the inlet edge depends on the edge rounding and on the radial clearance, and takes the value of ε_1= 69 ° when taking into consideration the ideal conditions in a valve (the clearance and the rounding radius= 0) [2]. At the same time the valve opening y must be much smaller than the upstream chamber. A calculation of the flow force can be done with simplification accepting the law of conservation of momentum for the inlet as well as for the outlet edge. Regarding the acting flow forces, an important difference between the inlet and outlet edge is the fact that the fluid flow at the outlet edge reaches its highest velocity and momentum outside the sliding-spool chamber. In view of that, the compensation of the flow force in this case, using the abbreviation of the fluid stream, is much more complicated and costly.

The flow force acting on a sliding spool can be determined by calculating the change of the momentum in the control volume of the sliding-spool chamber [5], using the assumption that the fluid stream is non compressible, two-dimensional, free of losses (friction free) and non-rotational. The stationary share of the flow force for the inlet edge can be calculated from the axial momentum component:

$$F_{\mathrm{ax}_{\mathrm{stat}}} = \rho \cdot Q \cdot \left(v_2 \cdot \cos\varepsilon_2 - v_1 \cdot \cos\varepsilon_1 \right). \qquad (1)$$

The non-stationary share of the flow force can be calculated using the length l of the accelerated oil quantity in the control volume of the sliding-spool chamber:

$$F_{\mathrm{ax}_{\mathrm{instat}}} = -\rho \cdot l \cdot \frac{\mathrm{d}Q}{\mathrm{d}t} \cdot . \qquad (2)$$

The non-stationary part of the flow force describes the force necessary to accelerate the fluid mass in the slidingspool chamber. The sign of this part depends on the respective fluid-flow direction.

The mean velocity v at the stream edge depends on the oil density and the pressure difference Δp:

$$v = \sqrt{\frac{2}{\rho}} \cdot \Delta p \,. \qquad (3)$$

The volume flow Q depends on the outlet coefficient α_D, the opening cross-section A_o and on the fluid-flow velocity v:

$$Q = \alpha_D \cdot A_o \cdot v, \qquad (4)$$

where $A_o = p \cdot D \cdot y$ and $\alpha_D = A_{vc} / A_o$,

A_o is the control edge opening cross-section, mm^2

A_{vc} is the fluid-flow cross-section, mm^2.

Different possibilities for the compensation of the flow forces in hydraulic sliding-spool valves were examined in the 1950s by different authors [3-4, 6]. The first classification of the suggested techniques was made by Backé [1]. He classified the compensation methods according to the following criteria:

- reduction of the axial forces by influencing the jet angle,
- compensation of the axial forces by achieving a pressure drop in the valve,
- compensation of the fluid-flow forces by an abbreviation of the oil stream.

More detailed research on the compensation method of the flow forces in hydraulic sliding-spool valves using the abbreviation of the oil stream has been done in recent years [8, 10]. This compensation method is also the main research topic presented in this article.

The inlet and the outlet edges of the valve are considered separately in the first part of the research because different compensation measures are used for each control edge following different conditions existing at the inlet and outlet edges.

The result of this is that flow forces at the inlet edge are compensated by using the method of two-dimensional fluid-stream abbreviation and at the outlet edge by using the three-dimensional fluid-stream abbreviation.

By applying the flow-force compensation at the inlet control edge it is intended to compensate the axial momentum component of the fluid stream entering the sliding-spool chamber with the outlet stream of the fluid. The law of the conservation of momentum shows that for compensation of the decreased fluid-flow velocity at the outlet cross-section, an appropriate outlet angle must be achieved. This can be done by an appropriate design of the sliding spool, i.e. by using a conical form of the sliding spool, as presented in Figure 3.

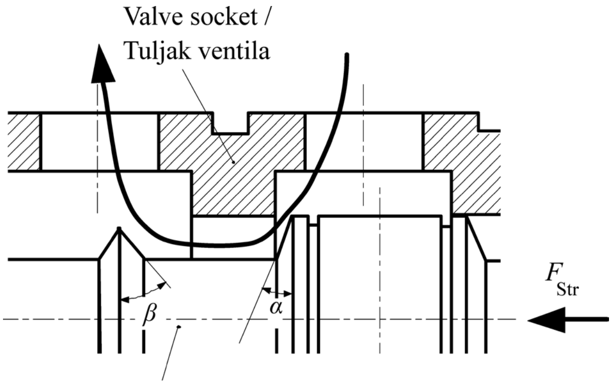

Figure 3. Spool geometry of the inlet edge

Slika 3. Geometrija klipa kod ulaznog strujanja fluida

A consequence of this measure is the fact that the larger part of the fluid stream abbreviation, which acts in the closing direction of the sliding spool in a conventional form of spool, takes place inside the valve socket and acts, therefore, on the valve socket walls. Using this measure, the flow forces acting in a closing direction of the sliding spool are greatly reduced. The extent of this reduction of the axial fluid-flow forces depends mostly on the angles α and β.

The flow-force compensation at the outlet control edge is much more complicated in comparison with the inlet edge. Because of the high fluid-flow velocity of the fluid jet, the conditions for an effective compensation measure of the axial flow force are more convenient behind the outlet control edge. In this case it is necessary to conduct the outlet oil jet back to the sliding spool. For this reason, a specially shaped design of the sliding spool and the valve socket is required. Figure 4 gives one of possible designs of the sliding spool and the valve socket.

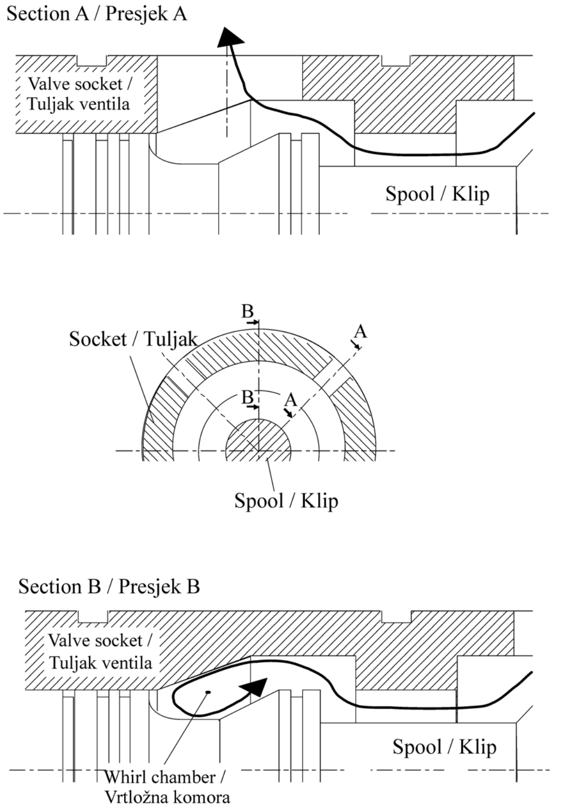

Figure 4. Spool and valve socket geometry of the outlet edge

Slika 4. Geometrija klipa i tuljka kod izlaznoga strujanja iz ventila

The upper part of the figure (section A) presents how the oil can flow directly out of the sliding-spool chamber. In this case, no compensation of the flow force is possible. The lower part of the figure (section B) presents the whirl chamber and the abbreviation of the fluid flow in it. The compensation of the flow force is a consequence of such an oil-jet abbreviation in the whirl chamber. This means that only that part of the oil flow remaining in the whirl chamber contributes to the flow-force compensation.

3. Experimental research

The valve, optimized in this research, should be designed for a pressure range from 0 to 350 bar with a nominal flow rate of at least 40 L/min and the maximum value of the flow force should not exceed a value of 30 N. The valve should be designed with a sliding-spool valve-socket configuration. The main task of the valve socket is to feed the oil streaming from the external circumferential groove to the sliding spool and also away from it under determined control-edge geometry. Figure 5 presents the flow-force-compensated valve socket and the sliding spool.

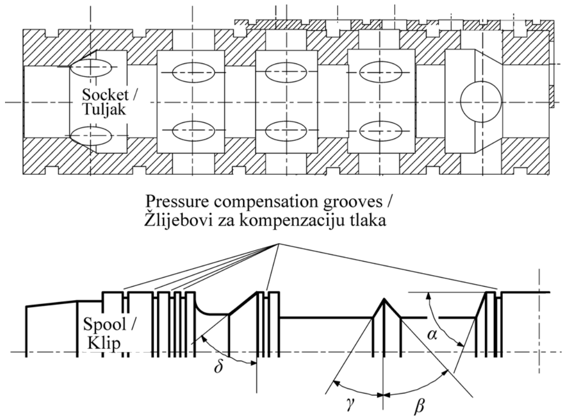

Figure 5. Valve socket and spool (flow force compensated)

Slika 5. Tuljak i klip ventila (kompenzirane sile strujanja)

The geometry of the sliding spool depends on the length of the valve body and the valve socket, on the width of the inner circumferential grooves or the distance between the control edges, as well as on the control-edge geometry.

The flow-force compensation of the sliding spool is the result of a combination of the angles Alpha (α) and Beta (β), influencing the inlet control edge, as well as Gamma (γ) and Delta (δ), influencing the outlet control edge. To get the optimal flow-forces compensation, it is necessary to analyse different sliding-spool geometries and the combination of the above-mentioned angles. Different geometries of five differently designed sliding spools analysed in this research are presented in Table 1.

Table 1. Spools with the angle geometry

Tabela 1. Klipovi ventila sa geometrijom kutova

Spool No. / Klip broj	Angle / kut			
	Degree / ° α	Degree / ° β	Degree / ° γ	Degree / ° δ
1	60	30	30	30
2	30	40	60	45
3	30	50	60	60
4	70	20	20	45
5	75	15	15	60

A realistic evaluation of the experimental research of the flow-force compensation in a valve can only be achieved when the research results of the flow-force-compensated sliding spool are compared with the results obtained with the non-compensated, conventional sliding spool. The geometry of the conventional, non-compensated sliding spool of standard form is presented in Figure 6. Also, in this case, the geometry and the control-edge position depend on the geometry of the valve socket.

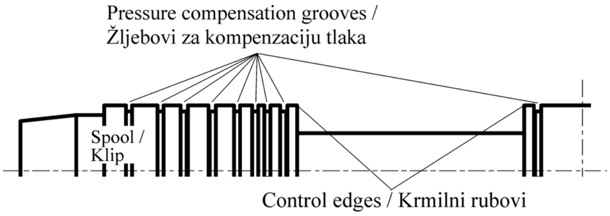

Figure 6. Non-compensated spool

Slika 6. Nekompenzirani klip ventila

3.1. Test bench

The effect of the flow-force compensation measures can be evaluated by comparing the experimentally investigated control characteristics of the valve, like the flow force versus the spool stroke and flow rate versus the spool stroke. The experimental investigation is made for the whole valve as well as for a single control edge separately. Also, the flow rate versus the pressure difference is measured for the complete valve (all control edges together).

In the experimental investigation of the flow-force compensation measures in the valve it is necessary to vary the geometrical parameters of the valve as well as the pressure difference at the valve. For a practical realization of the experimental investigation an appropriate flexible experimental setup is built, as shown in the figure. The discharge rate of the experimental system amounts to Q_{max}=250 L/min, at a maximum pressure of p_{max}=315 bar. Together with the delivery-pressure control valve, the control and analysis electronics, the pressure difference Δp can be controlled and kept constant during each individual investigation. This is very important for high flow rates, where pressure losses on the test bench increase significantly and consequently reduce the pressure differences. To avoid such a deviation of an actual pressure value from a set pressure value, the delivery-pressure control valve (Company: MOOG, Type: 72D-159) is used and controlled over the controller and consequently the set pressure value is kept constant.

For measuring the flow rate, a gear-measuring motor VS 2 made by the company VSE was used. The pressure sensors (company: Kistler, Type: 4075A500, range: 0—500 bar) for measuring the pressures p_0, p_A, p_B and p_T were mounted directly on the tested valve. The axial force was measured using a force sensor (company: Burster Präzisionalmeßtechnik, type 8523-500 N, range ±500 N). The spool position was measured using a potentiometric position sensor (company: Burster, type 8712—25).

All the sensors and the associated electronics used in the investigations were calibrated at the beginning of the measuring cycle and had a measuring error smaller than 0,1 %. The measuring data were immediately stored and analysed using the Lab View measuring software. Each investigation was repeated five times under identical measuring conditions and at the end the mathematical mean value was calculated. All final measuring results represent, therefore, the measured mean values of five investigations for each parameter with a measuring

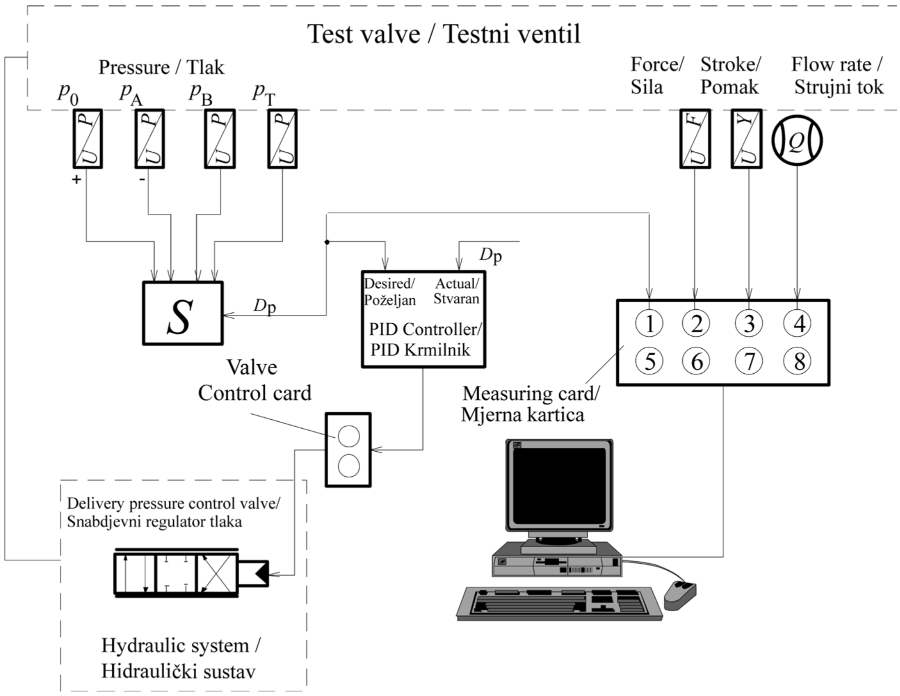

Figure 7. Test bench for measuring flow forces

Slika 7. Mjerno mjesto za mjerenje sila strujanja tolerance smaller than 2 %.

3.2. Flow-force investigations with modified spool geometry

Figures 8 and 9 show the measurement results of a non-compensated (conventional) and a compensated sliding spool at the pressure difference Δp=100 bar.

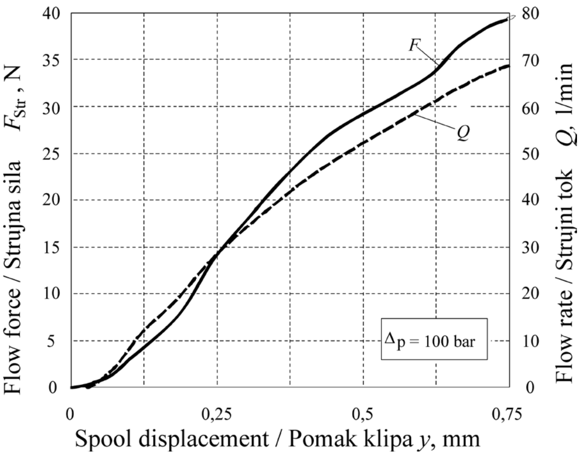

Figure 8. Flow force and the fluid flow, non-compensated spool, Δp = 100 bar, the whole valve (P- > B- > A- > T)

Slika 8. Sila strujanja i tok struje, nekompenzirani klip, Δp = 100 bar, cijeli ventil (P- > B- > A- > T)

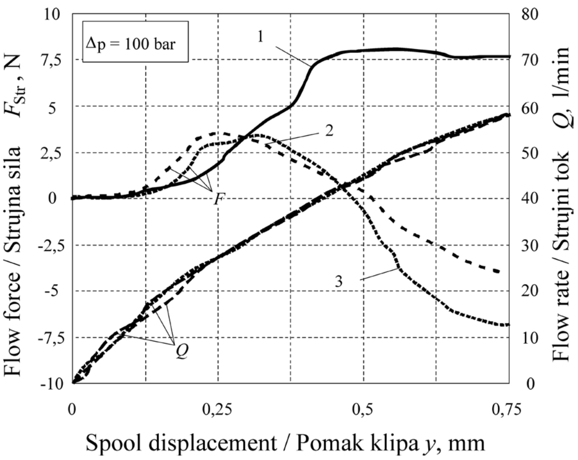

Figure 9. Flow force and the fluid flow, compensated spool (Nr. 1 – 3), Δp = 100 bar, the whole valve (P- > B- > A- > T)

Slika 9. Sila strujanja i tok struje, kompenzirani klip (br. 1 – 3), Δp = 100 bar, cijeli ventil (P- > B- > A- > T)

It is evident from the results that the compensated spool has a lower maximum flow force than the non-compensated spool. The flow force of the compensated sliding spool reaches, after a short steep increase, a maximum value, and then decreases again. The decline in the force can be explained by the overcompensation of the sliding spool. While sliding spools No. 2 and 3 (see Table 1) the effect of the flow force turns in the opposite direction. On the other hand, when spool No. 1 slides the flow force increases constantly with the increasing stroke of the spool (increasing the inlet/outlet cross-section) when compared with other spools. When comparing the flow-force characteristics of the individual control edges of the spool Nos. 1, 2 and 3, as presented in Figures 10 and 11, it is clear from the results that the overcompensation of the spool at the inlet control edge could be prevented by the enlargement of the angle α and the reduction of the angle β at the same time. At the outlet control edge, the same effect could be achieved with a reduction of the angle γ.

Based on the results, it is reasonable to investigate some additional sliding-spool geometries. As is evident from Table 1, the geometries of spools Nos. 2 and 3 are changed on the one half side of the spools and the new sliding spools are marked as spools Nos. 4 and 5. The following investigations of the modified sliding spools at the pressure differences 100, 200 and 250 bar show clearly the influence of the improvement of the sliding-spool geometries. In Figures 12 and 13 the results of the flow-force measurements for the whole valve are presented.

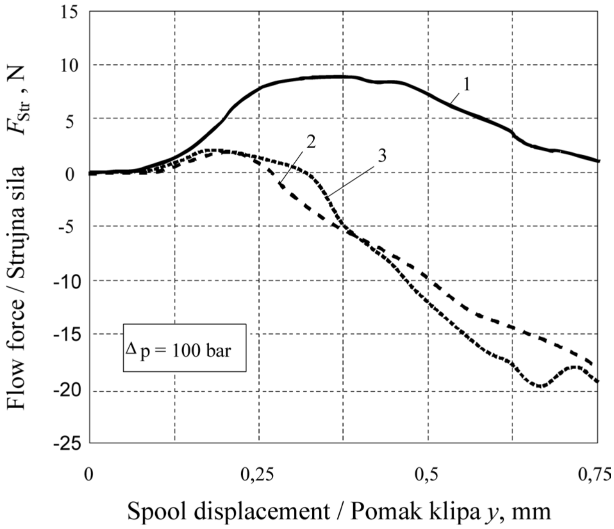

Figure 10. Flow-force measuring results, spool Nr. 1, 2 and 3, Δp = 100 bar, inlet edge (P- > A, P- > B)

Slika 10. Rezultati mjerenja sile struje, klip br. 1, 2 i 3, Δp = 100 bar, rub klipa ulaznog strujanja (P- > A, P- > B)

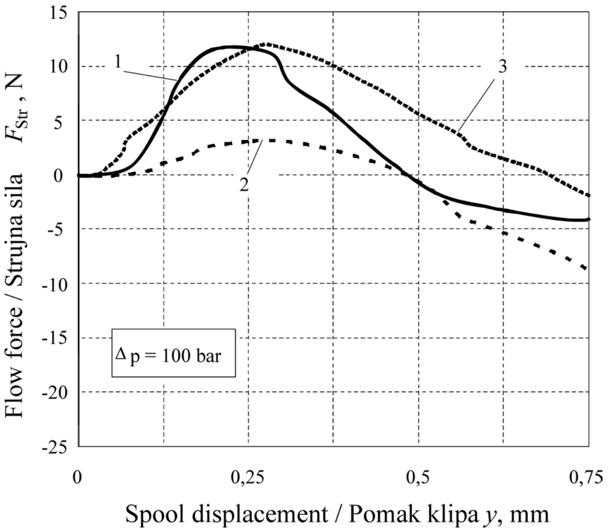

Figure 11. Flow-force measuring results, spool Nr. 1, 2 and 3, Δp = 100 bar, outlet edge (A- > T, B- > T)

Slika 11. Rezultati mjerenja sile struje, klip br. 1, 2 i 3, Δp = 100 bar, krmilni rub klipa izlaznog strujanja (A- > T, B- > T)

As seen from the results, the flow-force characteristics do not demonstrate any significant difference between both valve-spool modifications when the pressure difference is increased. Comparing these results with the one from Figure 9, it is evident that the flow-force characteristics are reduced with an enlarged inlet/outlet cross-section or, in other words, with an increased spool stroke, although this decrease is significantly smaller in Figures 12 and 13 than in Figure 9. Considering the single control edge it is evident that the outlet control edge is still overcompensated in spite of smaller maximum values of the flow force (Figure 14).

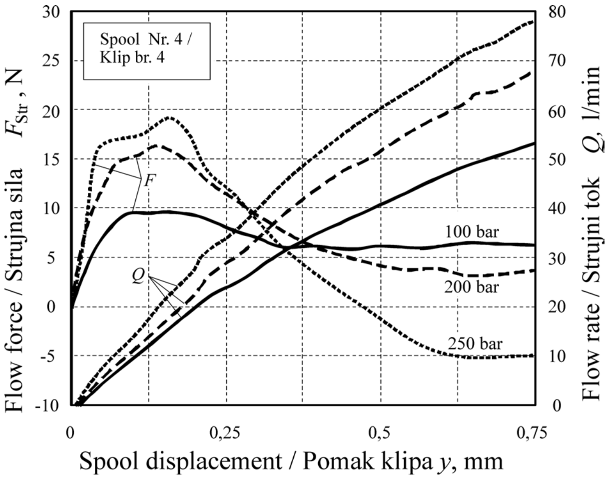

Figure 12. Flow-force measuring results, spool Nr. 4, the whole valve (P- > B- > A- > T)

Slika 12. Rezultati mjerenja sile strujanja, klip br. 4, cijeli ventil (P- > B- > A- > T)

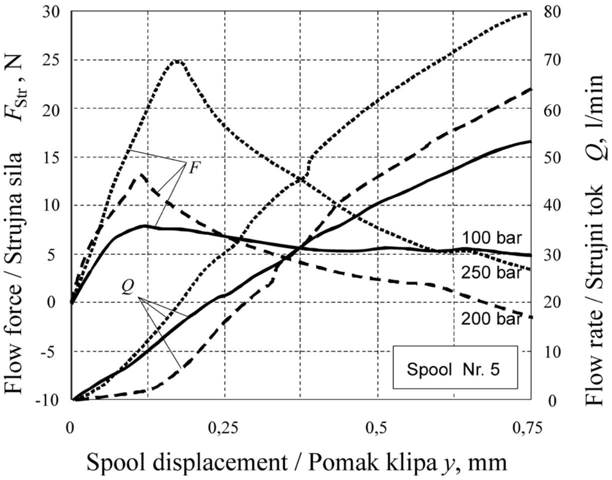

Figure 13. Flow-force measuring results, spool Nr. 5, the whole valve (P- > B- > A- > T)

Slika 13. Rezultati mjerenja sile strujanja, klip br. 5, cijeli ventil (P- > B- > A- > T)

3.3. Flow-force investigation with modified valve-socket geometry

Because the modification of the sliding-spool geometry regarding the outlet control edge still does not bring the expected success, the following investigations include the modification of the valve socket, which should result in more favourable flow-force characteristics. As only the part of the oil flow remaining in the whirl chamber contributes to the flow-force compensation (Figure 4), it must be accepted that more oil can flow out of the whirl chamber through the socket bores.

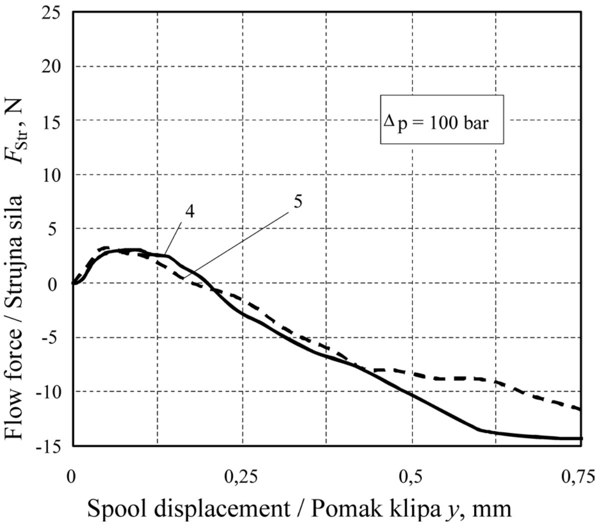

Figure 14. Flow-force measuring results, spool Nr. 4 and 5, Δp = 100 bar, outlet edge (B- > T, A- > T)

Slika 14. Rezultati mjerenja sile strujanja, klip br. 4 i 5, Δp = 100 bar, krmilni rub klipa izlaznog strujanja (B- > T, A- > T)

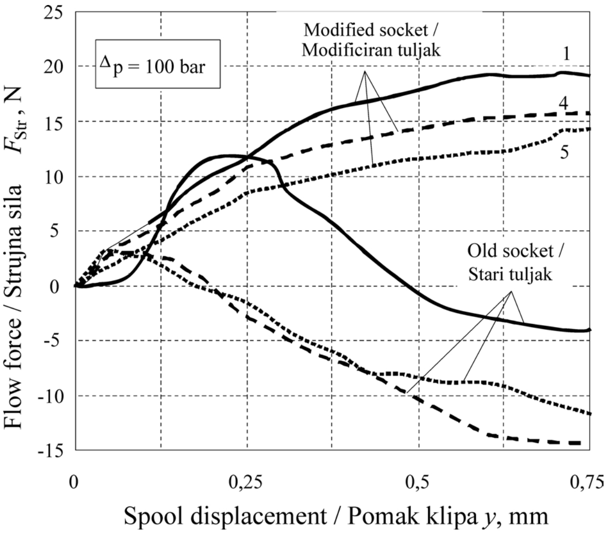

Figure 15. Flow-force measuring results, modified socket, valve piston Nr. 1, 4 and 5, Δp = 100 bar, outlet edge (B- > T, A- > T)

Slika 15. Rezultati mjerenja sile strujanja, modificiran tuljak, klip br. 1, 4 i 5, Δp = 100 bar, krmilni rub klipa izlaznog strujanja (B- > T, A- > T)

When comparing the measurement results of the outlet control edge with the sliding spools Nos. 1, 4 and 5 that have the modified valve socket, presented in Figure 15, it can be determined that the strong overcompensation of the first socket geometry is completely eliminated by the enlargement of the socket bores from the dimension of d_2=4 mm to the dimension of d_2=5 mm. The measuring results of the flow-force characteristics of the spools with the modified valve socket show completely different, but very desirable characteristics of the flow forces, when compared with the results where the old (non-modified) valve socket was used. The strong overcompensation is avoided, and at the same time the highest value of the flow force is not increased significantly. The three presented spools have very similar flow-force characteristics, in which spool No. 5 has the smallest value of the flow force.

The desirable flow-force compensation of the outlet control edge, achieved in such way, is also reflected very strongly in the measurement results of the whole valve. Figure 16 shows the flow-force characteristics with spool No. 5 and with the modified valve socket, where the flowforce stroke and the flow-rate stroke characteristics are presented for different pressure differences and at a maximum flow rate of 50 L/min.

The results in Figure 16 prove that flow forces increase with the increasing pressure difference; however, the maximum value of the flow force remains smaller than the desirable value of 30 N. The curves are set together for two or three different linear parts and could therefore be estimated as suitable in terms of their controllability in the potential use of the valve. At the same time, the significantly reduced flow forces make it possible to use the same external electromechanical actuators for higher flow rates and for higher pressure differences. This could also result in using the direct control of hydraulic valves for larger nominal valve sizes.

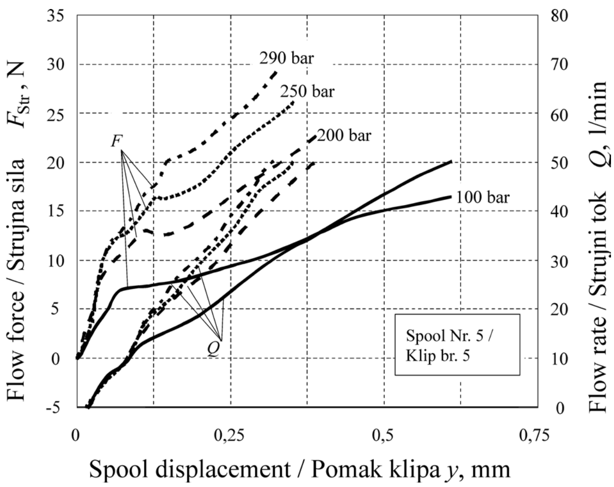

Figure 16. Force-force spool stroke and flow-rate spool stroke characteristics, spool Nr. 5, modified valve socket, the whole valve (P- > B- > A- > T)

Slika 16. Rezultati mjerenja sile i strujnog toka, klip br. 5, modificiran tuljak, cijeli ventil (P- > B- > A- > T)

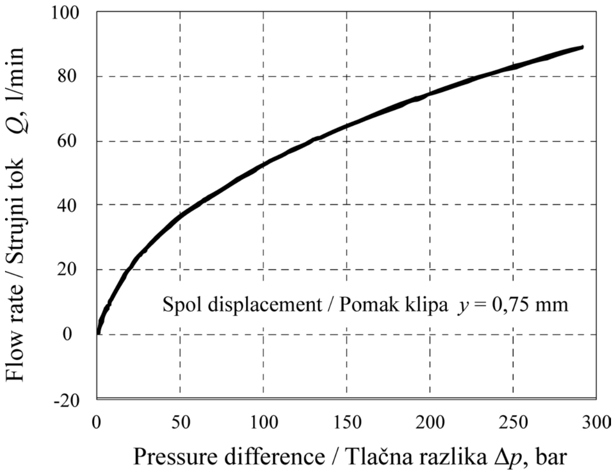

Figure 17. Flow rate versus pressure difference, measurin

Slika 17. Rezultati mjerenja strujnog toka u ovisnosti od tlačne diference

3.4. Nominal flow rate

It is desirable for the compensated valve to achieve a nominal flow rate of at least Q_{Nenn}= 40 L/min. The measured flow-rate pressure difference characteristic is presented in Figure 17. From the course of characteristics it is possible to realize that the nominal flow-rate value of the whole valve goes beyond the value of Q = 40 L/min at the pressure difference Δp = 70 bar or Δp = 35 bar per a single control edge and at the maximum spool stroke y = 0,75 mm. With this, the main goal is achieved.

4. Conclusion

Stationary axial momentum forces of the oil flow through a valve, which increase with the increasing flow rate and pressure difference, determine the external actuation forces required for controlling a sliding-spool valve. One of the goals in the research of hydraulic-valve development and optimization is also to enable the direct control of the valves for bigger nominal valve sizes (NVS). It is well known that the direct control of hydraulic valves is limited to smaller nominal valve sizes up to NVS 10, by using conventional electromechanical actuators, like solenoids.

This research shows the possibilities of using the direct control of the valves also for higher hydraulic powers and, consequently, the use of the advantages of the direct control also for higher flow rates and pressure differences. This is achieved by the reduction or compensation of the axial flow forces acting on the valve sliding spool in the closing direction.

The implemented flow-force investigations and the resulting modifications of the sliding spool and the valve socket prove that it is possible to reduce the flow force in the valve and to eliminate the non-desirable non-linear digressive flow-force characteristics by implementing some geometrical modifications to the sliding spool and the valve socket. All the control edges of the sliding spool are investigated separately and the characteristics of the flow forces in the valve are improved. Finally, an optimal spool-valve socket configuration is found, consisting of sliding spool No. 5 (see Table 1) and of the modified valve socket. This combination shows, compared with the noncompensated valve, significantly better results with clearly reduced maximum flow forces and with very convenient flow-force characteristics.

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