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CONSIDERATIONS REGARDING THE HYDRODOYNAMIC STRESSES FOR DIRECTIONAL SPOOL VALVES

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Abstract: *The dynamic analyzes of directional spool valves as well as the employed coefficients is an objective of great importance for understanding their running performances. The paper presents a synthesis of the manner in which must be determined for directional spool valves, the working characteristics taking into consideration the pressure and flow capacity losses. The established relations as well as the realized analyzes put into evidence most of the elements which must be taken into account for the design, fabrication and running of those devices.*

Keywords*: Directional spool valves, impulse hydrodynamic forces, angle, edge.*

1. Introduction

By way of operation hydraulic directional valves, no matter how they work, are subject to random dynamic stress produced by hydrodynamic forces induced by the hydraulic flow regime [1, 3, 4, 8, 15, 16, 17, 18]. Hydrodynamic parameters which influence the stress value are mainly the pressure, the flow and respectively the speed of the oil through the interstices created between the spool shoulders and the hole in the valve body. Because the application range is very wide, it is very important to know the hydrodynamic behavior of valves, depending on the pressure level, the flow rate, and the type of actuation scheme used [1, 3, 4, 8, 15, 16, 17, 18].

2. Impulse hydrodynamic forces

When passing the fluid through the valve sections as a result of spool displacement with the $\mathbf{Y_S}$ distance, an impulse force emerges which opposes its movement and seeks to bring it back to its original position. The hydrodynamic impulse force has two components: a stationary component, associated with the instantaneous flow through the spool, and a transient component determined by the transient variation of the flow rate [1, 3, 4, 8, 15, 16, 17, 18].
In Fig.1 is shown the flow distribution into the spool - valve body assembly for two $\mathbf{Y_S}$ positions (openings) made between the shoulder of the spool and the hole in the valve body, which lead to different passage angles through the annular space of the spool valve.

For spools with straight edges and for certain values of the ratio $\frac{\mathbf{Y_S}}{\mathbf{J}}$ (Fig.2), Von Misses sets the values of the angles under which the jet enters or exits relative the axis of the valve, for θ = 21 - 69°. For θ = 21...45°, the ratio is $0 < \frac{\mathbf{Y_S}}{\mathbf{J}} < 1.0$, respectively $\mathbf{Y_S} = \mathbf{J}$, for θ = 45° (Fig.1). The deduction is based on the impulse theorem applied for a non-stationary flow in the annular control volume of the axial section "**abcdef**" (Fig.2) in the form:

$$\frac{d}{dt}\sum(M_s\ \vec{Y}_s) = \sum\vec{F}_e \tag{1}$$

which becomes:

$$\sum\vec{F}_e = \int_{cd}\rho\ \vec{Y}_s\ dQ - \int_{ab}\rho\ \vec{Y}_s\ dQ - \int_{abcdef}\frac{\partial(\rho\ \vec{Y}_s)}{dt}\ dW_d \tag{2}$$

where: $\mathbf{W_d}$ - the volume element in the ring zone of the spool.

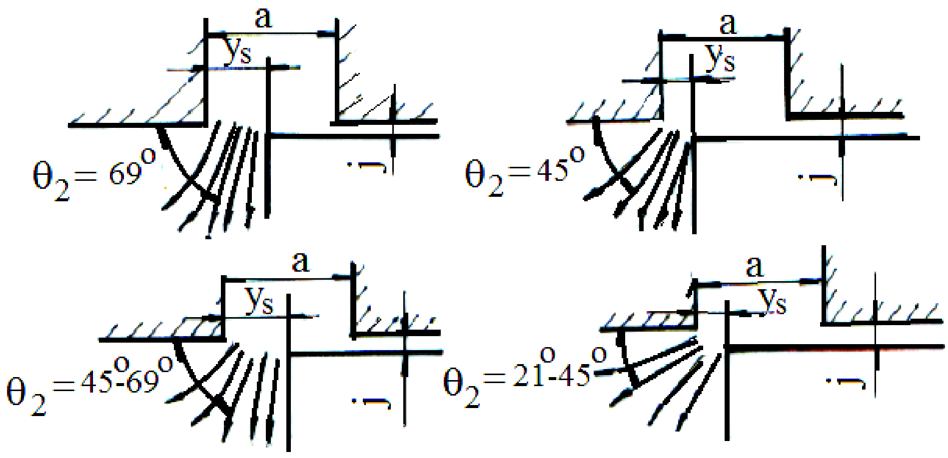

Fig.1

From the point of view of the power required for the displacement of the spool, it is useful to know only the axial component of the hydrodynamic impulse force on the hydraulic spool. For this purpose, equation (2) is projected along the axis of symmetry of the spool (OX axis), the radial forces being canceled due to axial symmetry. Thus, we obtain the force with which the liquid acts on the spool (in two hypotheses) (Fig.2, correlated with Fig.1):

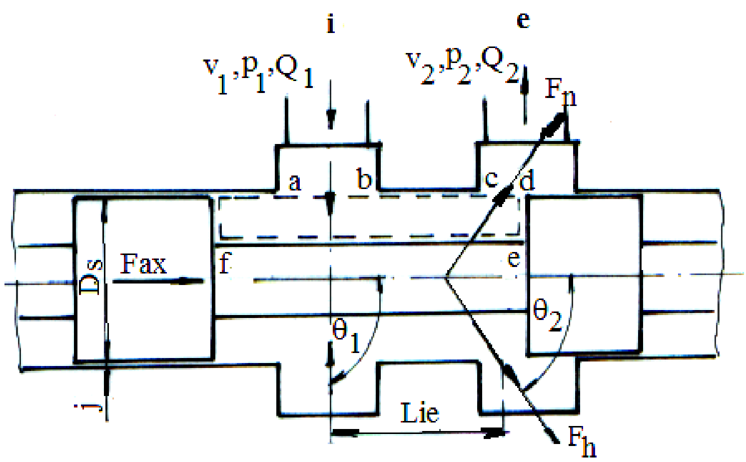

Fig. 2. Highlighting hydrodynamic forces in the spool - valve assembly

a) the fluid inlet is done from **i** to **e**, at the control edge of the spool:

$$F_{hax} = \rho\, \dot{Y}_{s_2}\ Q \cos\theta_e - \rho\, \dot{Y}_{s_1}\ Q \cos\theta_i - \rho\, L_{ie} \frac{dQ}{dt} \tag{3}$$

where to: $\boldsymbol{\theta_e} = \boldsymbol{\theta}_2 = 90\,^\circ$; $\mathbf{cos\,\theta_2} = 0$

$$\boldsymbol{\theta_i} = \theta_1 + 180^{\mathbf{o}}\ ;\quad \mathbf{cos\,\theta_i = cos(\theta_1} + 180^{\mathbf{o}}\mathbf{) = -cos\,\theta_1}$$

we will have:

$$\mathbf{F_{hax} = \rho\, Q\, \dot{Y}_{s1}\ cos\ \theta_1 - \rho\, L_{ie} \frac{dQ}{dt}} \tag{4}$$

b) the fluid inlet is done from **e** to **i,** at the control edge of the spool:

$$\mathbf{F_{hax} = \rho\, \dot{Y}_{s1}\, Q\, cos\,\theta_e - \rho\, \dot{Y}_{s2}\, Q\, cos\,\theta_i + \rho\, L_{ie} \frac{dQ}{dt}} \tag{5}$$

where to: $\mathbf{\theta_e = \theta_1;\quad \theta_i = \theta_2 + 180^\circ = 270^\circ\ ;\quad cos\ \theta_i = 0}$ we will have:

$$\mathbf{F_{hax} = \rho\, \dot{Y}_{s1}\, Q\, cos\,\theta_1 + \rho\, L_{ie} \frac{dQ}{dt}} \tag{6}$$

In both cases, the stationary component of the hydrodynamic impulse force has the same value and acts in the direction of closure of the wiper slot. The angle $\boldsymbol{\theta_1}$ depends on the size of the radial play and the ratio $\mathbf{\frac{Y_S}{J}}$ and varies within the limits $\boldsymbol{\theta}$ = 0°... 69° (Fig.1). Considering that the valve is working on a pressure $\mathbf{\Delta p}$ = const., we will have:

$$\mathbf{\dot{Y}_{s1} = C_v \sqrt{\frac{2\,\Delta p}{\rho}}} \tag{7}$$

From [1] it follows:

$$\mathbf{Q = C_d\, \pi\, D_s\ Y_s \sqrt{\frac{2\,\Delta p}{\rho}}} \tag{8}$$

from which, the stationary component of the axial hydrodynamic force, $\mathbf{F_{haxs}}$, under the conditions of constant pressure operation, becomes:

$$\mathbf{F_{haxs} = \rho\, Q\, \dot{Y}_{s1}\, cos\ \theta_1 = 2\, C_d\, C_v\, \pi\, D_s\ Y_s\ \Delta p\, cos\ \theta_1 = K_{hp}\ Y_s} \tag{9}$$

respectively

$$F_{haxs} = \rho\, Q\, \dot{Y}_{S1}\, \cos\theta_1 = B_F\ Q \sqrt{\Delta p} \tag{10}$$

where different forms of the hydrodynamic force coefficients are presented in the expressions (9) and (10).

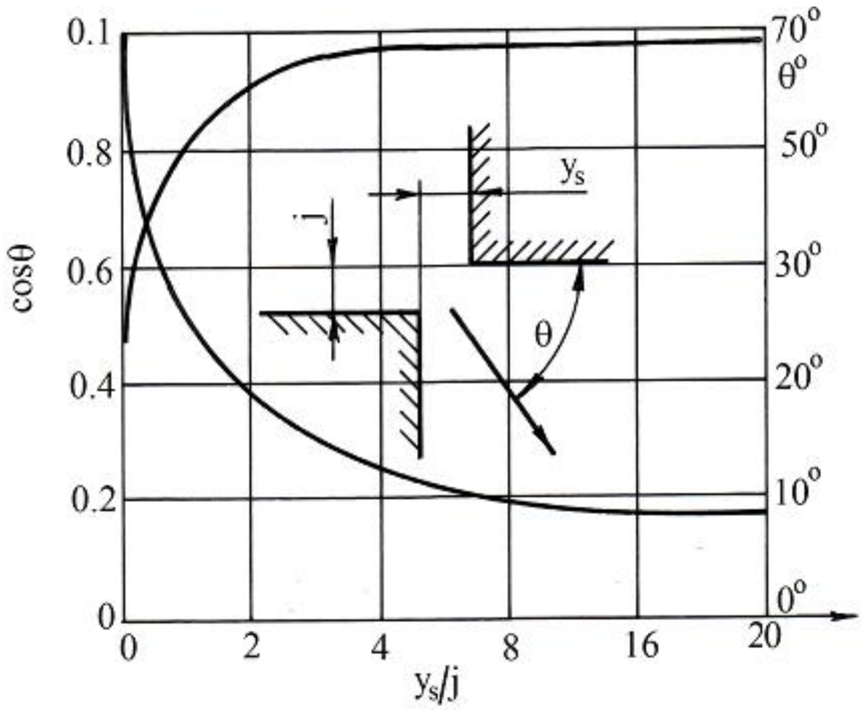

Fig.3

Under constant flow operation regime, we will have:

$$F_{haxs} = \frac{\rho\, Q^2 \cos\,\theta_1}{C_d\;\pi\, D_s\;Y_s} = K_{hq}\,\frac{1}{Y_s} \tag{11}$$

Here we noted: **C_d, C_v** - flow and speed coefficients (**C_d** = 0.61 – 0.7; **C_v** = 0.95 – 0.98); **K_{hp}**, **K_{hq}**, $\mathbf{B_F = \sqrt{2\,\rho}\cos\theta_1}$ - coefficients of axial hydrodynamic forces. In the general case, for **n_m** circuits that pass through the valve, relations coeficienţii forţelor hidrodinamice axiale (9, 10) become:

$$F_{haxs} = 2\, n_m\; C_d\; C_v\;\pi\, D_s\;\, Y_S \Delta p \cos\,\theta_1 \tag{12}$$

$$F_{haxs} = \frac{\rho\, n_m\;\, Q^2 \cos\,\theta_1}{C_d\;\pi\, D_s Y_s} \tag{13}$$

For the real situation, where the radial play is also considered, the hydrodynamic force becomes (Fig.2):

$$F_{haxs} = 2\, n_m\; C_d\; C_v\;\pi\, D_s \Delta p\,\sqrt{Y_s^2 + J^2}\;\cos\,\theta_1 \tag{14}$$

where:

$$\mathbf{\frac{Y_s}{J} = \frac{1 + \frac{\pi}{2}\sin\,\theta_1 - \ln\left[tg\frac{\pi - \theta_1}{2}\right]\cos\,\theta_1}{1 + \frac{\pi}{2}\cos\,\theta_1 + \ln\left[tg\frac{\pi - \theta_1}{2}\right]\sin\,\theta_1}} \tag{15}$$

The relations (9 ... 14) show that in the case of feeding a hydraulic resistance (specific to the spool - body valve assembly) at constant pressure (Fig.4c), the hydrodynamic force **F_{haxs}** increases with the **Y_S**, opening up to the saturation of the source, until all the flow provided by the pump passes

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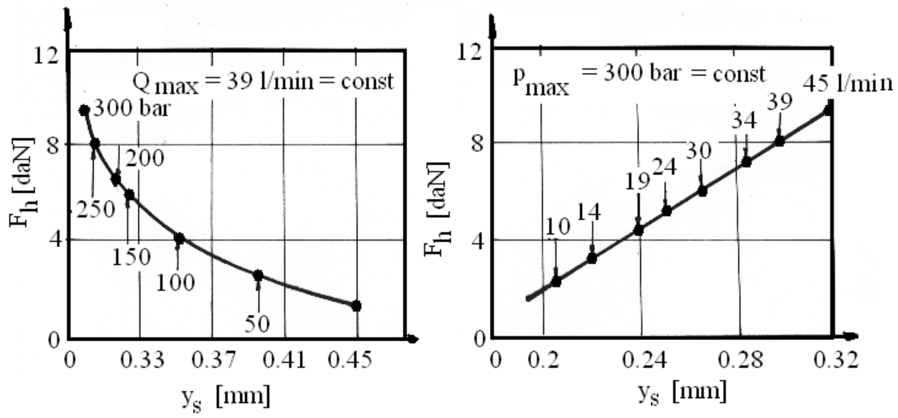
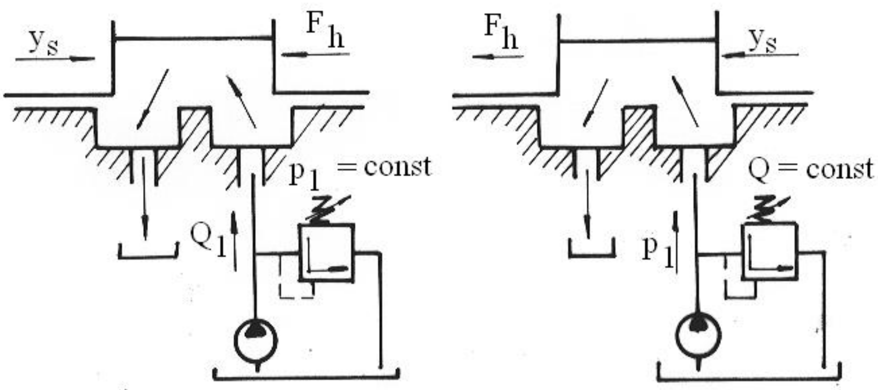
by resistance, which causes the safety valve to be closed and as result the constant feeding pressure condition to be canceled (Fig.4b).
The stationary component $\mathbf{F_{haxs}}$ of the axial hydrodynamic force on the spool always tends to close the slot of the throttle and is calculated with the same relation regardless the location of the hydraulic spool control edge.
The relations (9 ... 14) show that in the case of feeding a hydraulic resistance at constant flow (Fig.4d), the hydrodynamic force $\mathbf{F_{haxs}}$ increases with the decrease of $\mathbf{Y_S}$ opening up to the saturation of the source, when the safety valve opens and a part of the pump flow is discharge to the tank, thus canceling the feeding condition at constant flow (Fig.4a).
Therefore, at reaching the saturation limits, a hydraulic resistance feeded at constant pressure passes into a steady flow feed and vice versa.
The values of the stationary hydrodynamic force component, calculated with relations (9 ... 14) for a DN 10 distributor, are shown in Figures 4a and b. For small openings, the curves can be approximated by straight lines. For the larger $\mathbf{Y_S}$ openings, the impulse hydrodynamic force decreases because the pressure $\mathbf{\Delta p}$ on the control edge becomes smaller.



a) under constant flow regime; b) under constant pressure regime

Fig.4



c) d)

Fig.4

The compensation of hydrodynamic forces can be achieved by:

- properly fitting of the spool and of the valve body (Fig.5). The hydrodynamic force becomes:

$$\mathbf{F_{haxs} = \rho\, Q - \dot{Y}_{s1}\ \cos\ \theta_1 - \rho\, Q\, Y_{s2}\ \cos\ \theta_2 = \rho\, Q\, \dot{Y}_{s1}\left[1 - \frac{\dot{Y}_{s2}\ \cos\ \theta_2}{\dot{Y}_{s1}\ \cos\ \theta_1}\right]} \qquad (16)$$

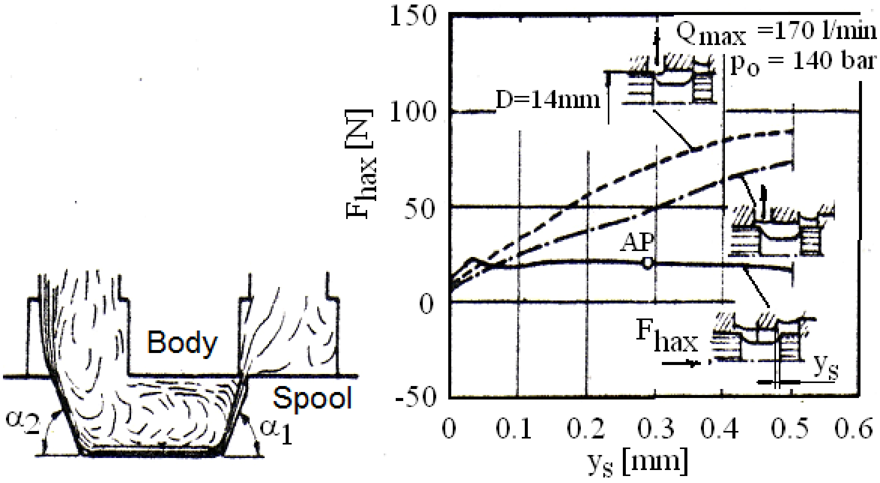


Fig.5.a. The effect of the fluid flow on the hydrodynamic force for the inlet and outlet edge

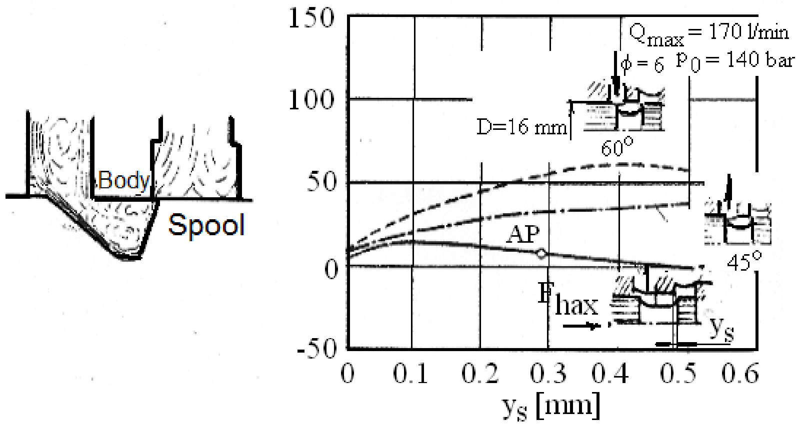


Fig.5.b. The influence of spool angle in the flow area on the impulse hydrodynamic forces

The data from Figures 5a and b suggest that by selecting the contour of the spool the constructor has the possibility to control the hydrodynamic forces on the spool both in size and direction, but within certain limits. This is also possible by further processing of the spool shoulders, which will lead to a corresponding adjustment of the hydrodynamic forces for the inlet and outlet edges.
Typical output angle is $\boldsymbol{\theta_1} \approx 69°$. The lower values of θ_1 slightly influence the hydrodynamic force, having a more pronounced effect at the large openings of the spool. Biger angular values for $\boldsymbol{\theta_1}$ will lead to detachment of the flow within the valve and high pressure losses due to direction changes, which require lower angle values for $\boldsymbol{\theta_2}$. This method is an effective means of compensation, although it is not possible to define an optimal profile of the spool or valve body.

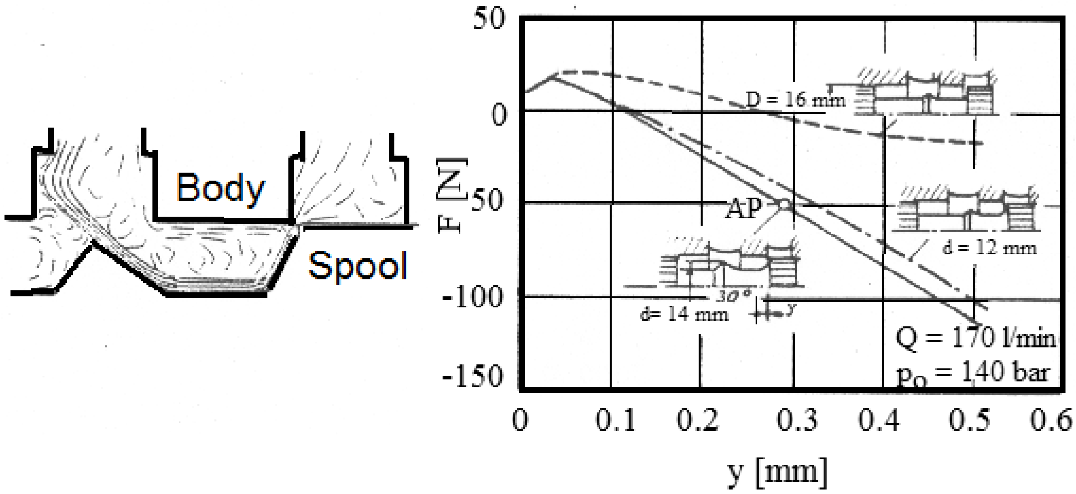


Fig.5.c. The compensation of hydrodynamic force on a flow edge through a conical shoulder in the spool chamber

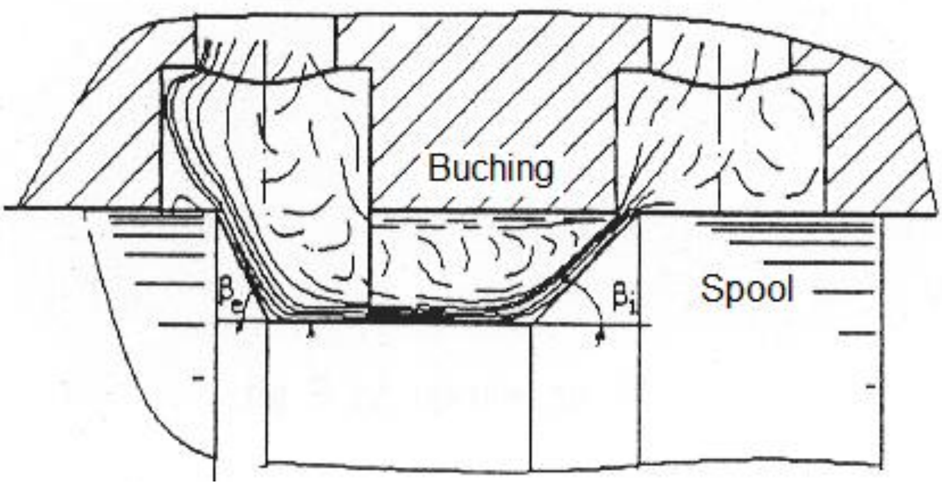


Fig. 6

The method of jet deviation at inlet and outlet spool edges aims to compensate the axial component of the impulse hydrodynamic forces caused by the inlet jet by means of the flow at the outlet, or vice versa, which is done by the geometry of the spool shoulders. The jet in the hydraulic resistance (Fig.5) breaks a little in the spool chamber, which, at angles $\theta > 60^0$, allows for good compensation.
Experimentally, it has been demonstrated that a sufficient reduction of hydrodynamic forces can be achieved for different geometries of the spool - valve body assembly (Fig.5.b). By concentrating the fluid flow near the spool rod, the desired flow of the current is possible with relatively small shoulders of the spool.
The combination of channel bushing and spool with an additional 30 ° angle in the outlet area leds at the action of the force in the direction of opening the spool (induced impuls hydrodynamic force) (Fig.5.c). The insertion of a double conic threshold allows the use of the inertia force of the "sticky" jet to divert it, providing a sufficient flow section to the tank (Fig.6).

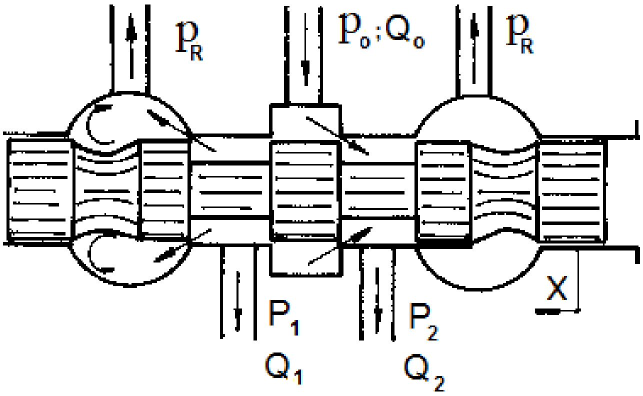


Fig. 7

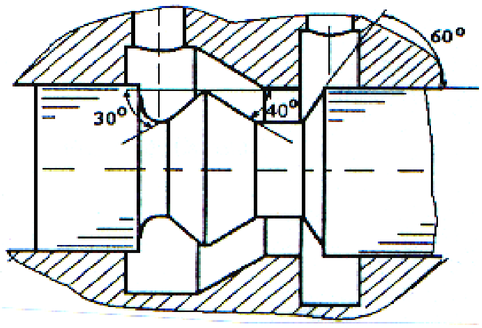


Fig. 8

The corresponding profiling of both the spool and the valve body, Figs. 7-8, has the effect of reducing axial hydrodynamic forces but is technologically high in complexity. For this case, the resulting force will have the value:

$$F_{hd\,ax} = Q\,\dot{Y}_S\,\rho(\cos\,\theta_1 - \cos\theta_2) \qquad (17)$$

According to this relation if $\cos\,\theta_2 > \cos\,\theta_1$, the axial hydrodynamic force becomes negative, provide a positive effect on the spool control system, i.e. it tends to move the spool in the direction of the opening of the adjustment section (participates in the opening or closing of the spool). Increasing the angle θ_1 can be done by means of a bevelled edge on the spool (Fig.6). As the angle tends to values θ_1 > 69°, the axial component of the hydrodynamic force acts in the radial direction, leading to its compensation, ie the value of the hydrodynamic force tends to zero (Fig.9).

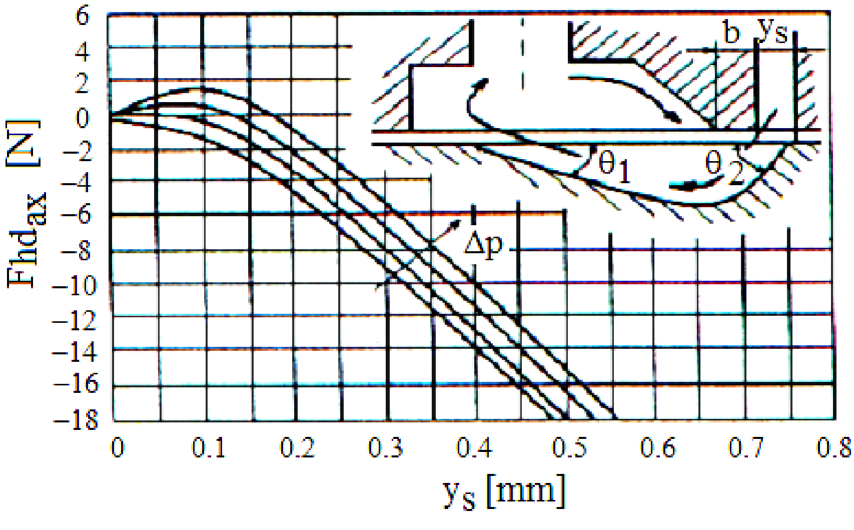
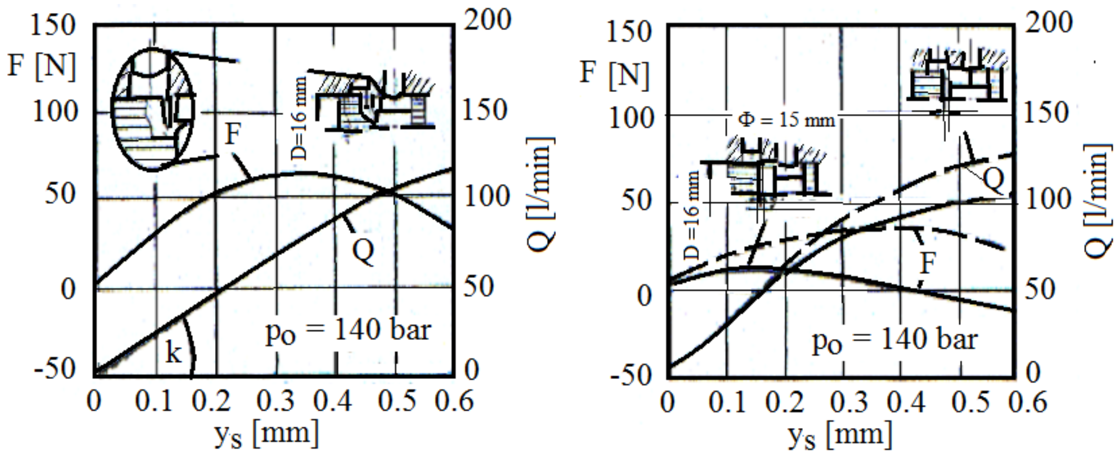


Fig. 9

The compensation of the impulse hydrodynamic forces by changing the angle of the jet or the resistance force is achieved by dividing the total section of the current into several partial sections, which will operate in succession (Fig.10.a). This solution leads to large spool strokes and nonlinear features, plus: the high cost price and the phenomenon of clogging. At a high amplifier coefficient of the valve, limited compensation is achieved (Fig.10.a).
The introduction of a hydraulic resistance force (Fig.10.b), by creating a pressure drop compared to the first solution, that of the corresponding profiling of the spool and the valve body, highlights

that the first method (Fig.10.a) is more efficient. The pressure loss that occurs by the insertion of an additional step on the spool shoulder (Fig.11) is due to the discontinuous section enlargements (determined with the Borda-Carnot relations) and the friction on the spool and bush walls. This solution results in an increase of impulse hydrodynamic force compensation and is usually only used to reduce the impulse hydrodynamic forces for the input edge.



a) b)

Fig. 10. Influence of jet angle and resistance force
a) methods of compensating the impulse hydrodynamic forces
b) fluid recirculation on the spool shoulders.

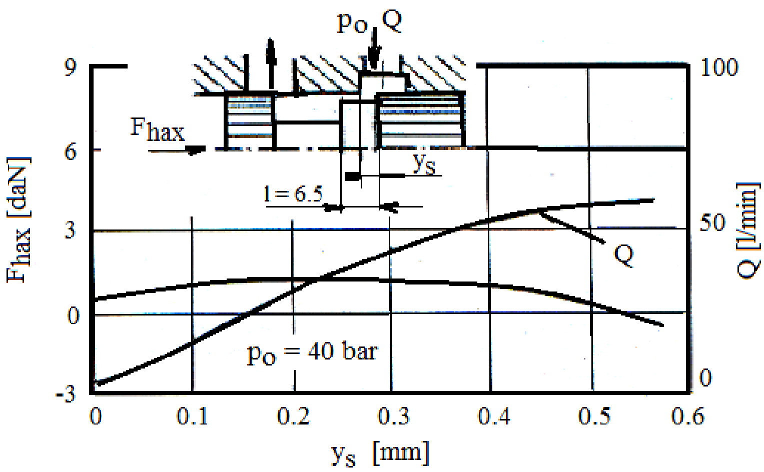


Fig. 11

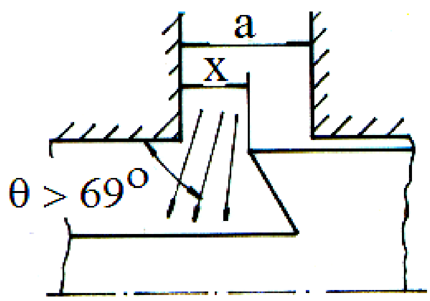


Fig. 12

Another component of the hydrodynamic forces is the transient one, caused by the instability of the stationary flow regime, the transient regime or the resonance phenomenon. Flow instability occurs when passing from one flow regime to another, accompanied by transient phenomena.

In hydraulic installations, although changes in flow conditions occur rapidly, the instability is manifested at a low intensity, except when overlapping other causes. The phenomenon becomes important if the frequency of the transient phenomena is equal to that of the spool, in which case the resonance phenomenon appears. Flow instability may occur when the fluid jet flows directly into a larger chamber, the transient regime being given by the transient components of the hydrodynamic forces.

The transient component $\mathbf{F_{haxtr}}$ of the impulse hydrodynamic force is associated with the fluid inertia in the annular space of the control volume, due to fluid acceleration in the spool chamber and is given by:

$$\mathbf{F_{haxtr}} = \rho\,\mathbf{L_{ie}}\,\frac{\mathbf{dQ}}{\mathbf{dt}} = \rho\,\mathbf{L_{ie}}\,\mathbf{C_d}\,\pi\,\mathbf{D_s}\left\{\sqrt{2\,\rho\,\Delta p}\,\frac{\mathbf{dY_S}}{\mathbf{dt}} + \sqrt{\frac{\rho}{2\Delta p}}\left(\frac{\mathbf{d\Delta p}}{\mathbf{dt}}\right)\right\} \tag{18}$$

This component is oriented opposite to the flow direction of the fluid. The term representing the travel speed of the spool is important because it influences the speed of the movement, decreasing it in the case of "**a**", and increasing it in case "**b**". Under conditions of constant pressure, the second term can be neglected. Where from:

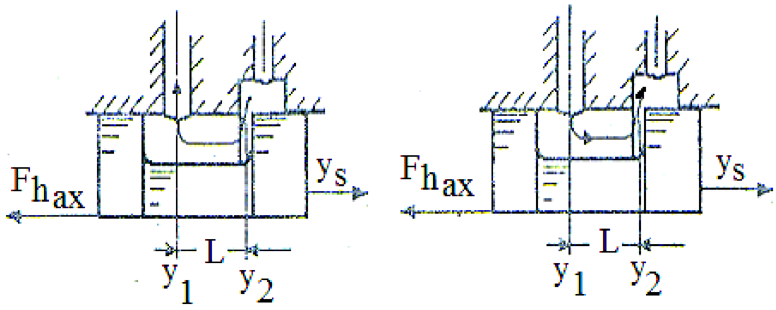
$$\mathbf{F_{haxtr}} = \rho\mathbf{L_{ie}}\,\mathbf{C_d}\,\pi\,\mathbf{C_d}\sqrt{2\,\rho\,\Delta p}\;\dot{\mathbf{Y}}_s = \mathbf{K_{htr}}\;\dot{\mathbf{Y}}_s \tag{19}$$

We can express the hydrodynamic force in the form:

$$F_{hax} = (K_{hst} \pm K_{htr})\dot{Y}_S \tag{20}$$

the sign (+) being for the case of the active edge set at the exit, and the sign (-) corresponds to the **inlet** location of the control edge.

From the above, it results that the static component of the hydrodynamic force is independent of the flow direction of the fluid through the adjustment section, and the transient component is dependent on the direction of the resistance flow through the fluid. If the $\mathbf{K_{htr}}$ coefficient is positive (Fig.13.a), the damping is positive, the spool is in static equilibrium, and if it is negative (Fig.13.b), the damping is negative and the spool enters a static imbalance. To improve the static balance condition of the spool, it is necessary to have the positive damping lengths greater than the negative lengths.



a) b)

Fig. 13

As the component $\mathbf{F_{haxtr}} \cong \dfrac{\mathbf{F_{haxs}}}{\mathbf{30}}$ is negligible in relation to the stationary hydrodynamic force, it can be approximated $\mathbf{F_{hax}} \cong \mathbf{F_{haxs}}$. However, for high pressure ($\mathbf{p_o} > 10\,\mathrm{MPa}$), hydrodynamic forces

can become important in both stationary and transient modes, and therefore it is necessary to compensate them.
If the hydraulic spool has several active edges, the hydrodynamic forces result from summing the forces associated with all the active edges. At the same time, when calculating the final hydrodynamic force at the hydraulic spool, all the forces associated with the control edges, components that tend to close the respective throttle slots, will be considered.

3. Conclusions

Hydrodynamic forces play an important role in the optimal operation of directional spool valves. An important role is played by the stationary component, which depends on the geometry of the edges of the spool and of the valve body bush, influencing the inlet and outlet flow of the liquid jet.
When the flowing passes from one regime to another, the transient component of the hydrodynamic force becomes important.
Hydrodynamic forces become important for a pressure greater than 10 MPa, in both stationary and transient mode, when compensation is required.

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