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PROPORTIONAL VALVE WITH AXIAL FLOW AND ROTATIONAL METERING

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ABSTRACT

This paper presents a new concept hydraulic valve that tries to overcome a well-known poser affecting the pilot operated proportional valves, the flow forces. Despite of the traditional compensated profile spool valves, the basic idea is to design a valve that has as few mobile surfaces as possible. This assumption modifies the traditional valve design method and opens to new possibilities for the proportional valves. The solution presented in this paper uses an axial flow valve, where the oil gets through the valve across its axis, with two rotating surfaces causing a rotational metering. The result of this new design approach shows several advantages with respect to the common spool valves, such as the extremely compact size and the device versatility. This particular valve can realize the majority of the functions achievable using a two-way two-position proportional valve piloted by two pressure signals (for example a pressure compensated valve); the axial flow and the “built-in” metering edges yield the possibility to produce this valve as a cartridge component, whit all the advantages incidental to this type of devices.
Some Computational Fluid Dynamics Analysis confirm the prediction of a low affection of this valve by flow forces, this attitude makes the axial Flow and Rotational Metering Valve particularly suitable for the local compensation in Flow Sharing Load Sensing distributors.

KEY WORDS

Valve, Flow Forces, Metering, Axial Flow

NOMENCLATURE

A_{flow} : area upon which p_{flow} acts
$A_{flow,A}$: A_{flow} (ARM valve)
$A_{flow,s}$: A_{flow} (spool valve)
A_{pil} : area upon which the pilot pressures act
F_1, F_2 : force of the spring 3, 4 (spool valve)
F_{flow} : flow forces modulus
F' : T_{flow} reduced at R_{coil}
p_1, p_2 : pressure in volume 6, 5 (spool valve)
p_{err} : over-pressure due to flow forces
$p_{err,A}$: p_{err} (ARM valve)
$p_{err,s}$: p_{err} (spool valve)
p_{flow} : mean pressure recovery on spool walls
$p_{flow,A}$: p_{flow} (ARM valve)
$p_{flow,s}$: p_{flow} (spool valve)
p_{LS} : Load Sensing line pressure
p_{m1} : pressure upstream the local compensator
p_{v1} : pressure downstream the local compensator
R_{coil} : mean radius of the coil grooves (ARM valve)
R_{flow} : maximum distance between the valve axis and $A_{flow,A}$
T_{flow} : flow torque (ARM valve)
x : spool stroke (spool valve)
x_r : reducer stroke (ARM valve)
α : coil angle (ARM valve)
ϕ : rotor angular stroke (ARM valve)

INTRODUCTION

The majority of the power control systems can be represented as a block of devices which meter the area connecting two volumes; this devices are variable orifices and can be piloted in several ways such as: manual, pilot-pressure operated, electro-actuated and so on. Whatever the piloting method used, the final configuration of the device is affected by some “secondary effects” caused by the nature of the device itself. One of these secondary effects is that caused by flow forces.

This paper recalls the nature of the flow forces, some well-known method to reduce their impact on the device behavior and a new way to overcome the problem.
A new concept valve will be presented with an extremely low flow forces effect and a brand new axial flow design, some CFD Analysis will show how much this solution is effective in reducing the “over-compensation” of a two-way pilot operated valve .
Finally, some positive secondary aspects will be shown, such as the extreme flexibility of the device, that ensures a reduction of the production costs by the realization of multiple configurations using the same set of components.

THE NATURE OF THE FLOW FORCES

The flow forces

Neglecting the contribution due to changes in flow rate [1], flow forces can be considered a “static” phenomena that occurs every time a fluid is forced to pass through a restriction; due to the continuity equation, whenever a fluid passes through a striction it must increase its velocity, but its total energy is tends to remain constant, neglecting piezometric contribution, the total energy is given by the sum of the kinetic and the hydrostatic part.
The fluid must increase its velocity near the striction, and therefore it must convert some of its hydrostatic energy into the kinetic; this means that near a restriction the fluid accelerates and its static pressure decreases, generating a “local pressure loss”. On the other hand, once crossed the striction, the fluid decreases its velocity re-converting the kinetic energetic part into the hydrostatic one, generating a “local pressure recovery”.
This phenomena does not cause problems or disturbs hydraulic devices themselves, but is the first cause of the Flow Forces; the local pressure losses or recoveries cause forces when they occur near a surface. If the surface on which the pressure losses and recoveries act is a fixed surface, it doesn’t occur cause major troubles, and the behavior of the devices is not affected. But if it occurs on mobile surfaces, which can move along the same direction where the pilot pressure works, their force and the pilot force will sum in ruling the configuration of the device.

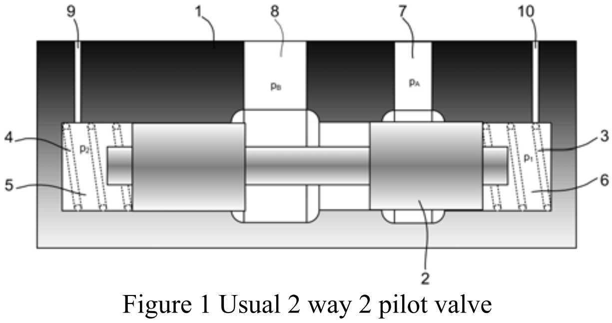

Figure 1 Usual 2 way 2 pilot valve

Let us consider a common spool valve, which has the task to meter the flow area between two volumes (A and B) in function of two pressure signals (p_1 and p_2) and the force of two springs (F_1 and F_2). Referring to Figure 1, this valve is made up by a body (1) with two port (7) and (8) communicating with the volumes A (at pressure p_A) and B (at pressure p_B) respectively. The ports (9) and (10) connect the pilot chambers (5) and (6) with their respective pilot intake (at pressure p_2 and p_1, respectively); the springs (4) and (5) complete the device logic providing the forces F_2 an F_1 respectively.
Linking the volumes (5) and (6) with several pressure intake it is possible to create the majority of the working logic in metering the flow area between two volumes.
The flow area is ruled by the axial position of the spool, and the axial position of the spool is determined by the equilibrium of the forces acting on the spool itself. To understand the relevance of the Flow Forces, an estimate of the spool equilibrium is advisable.
Let’s suppose that p_A is greater than p_B and the purpose of this valve is to meter the flow area between volume A and volume B in function of the pressures p_1 and p_2.
At the beginning ($p_1 = p_2$) the spool in the position represented in Figure 1, and upstream the spool the pressure is p_A, while downstream the metering edge the pressure is p_B (even between the spool walls). Increasing p_2 the spool moves right, opening the flow area between A and B; pretending that the flow forces don’t exist, the equilibrium equation of the spool is given by Eq. (1):

$$p_1 \cdot A_{pil} + F_1(x) = p_2 \cdot A_{pil} + F_2(x) \qquad (1)$$

Where A_{pil} is the area on which the pilot pressures works (supposed equal on both sides); as it can be seen in Eq. 1, the spool position x is given only by the pressures p_1 and p_2 and the two springs.
If we look more accurately, it is easy to recognize the striction through which the fluid must pass to reach volume B starting from A. The fluid must accelerate near the striction (right wall of the spool) causing a local pressure loss, and it must decrease its velocity near the left wall of the spool, generating a local pressure recovery.
Both the pressure loss and recovery throw off balance the spool with an axial force directed from right to left that tries to close the connection between volume A and B; the correct spool equilibrium equation is given in Eq. 2):

$$p_1 \cdot A_{pil} + F_1(x) + F_{flow} = p_2 \cdot A_{pil} + F_2(x) \qquad (2)$$

F_{flow} is function of a large number of parameters [2] [3] such as the flow area, the shape of the area, the jet angle, the pressure gradient, the flow rate and so on; it’s very easy to understand that due to the flow forces the spool position is function of several undesired factors that

affect the device logic.
The usual countermeasure is the inlet/outlet profile compensation; adopting this design method we can reach two advantages, first of all it is possible to reduce the momentum variation of the fluid (under-acting the local pressure losses and recoveries), secondary the local pressure variation can be dispatched near a fix wall, cancelling their effect. Unfortunately, the non-linearity of the local pressure variation phenomena makes this approach effective in some device configuration and totally ineffective in others.

Getting rid of flow forces

To show a new way to reduce Flow Forces modulus, it's helpful to express F_{flow} as the ratio between a "mean pressure recovery on spool walls" and the area on which this recovery acts (normalized with respect to the spool displacement), $F_{flow} = p_{flow} * A_{flow}$; the reduction of F_{flow} can be reached by the reduction of its factors.
The first action is to minimize p_{flow} (the mean pressure variation) reducing the fluid momentum variation. This approach implies the minimization of changes in direction of the fluid streamlines in passing through the valve. A spool valve constraints the fluid to perform two variations of direction (from radial to axial and then from axial to radial). The elimination of the direction variation of the fluid velocity suggests the adoption of a valve that makes the fluid to pass through without changing its direction, such as a purely axial flow valve. The increase and the decrease of the fluid velocity across the striction is not removable (until the continuity equation holds), so it's expected that p_{flow} will not drop to zero.
To explain exactly which actions can be undertaken to minimize A_{flow}, it is necessary to focus on A_{flow} definition. The pressure losses and recovery are dangerous when they're near a surface, but this surface must be able to generate a non zero work along the permitted displacement of the device. More precisely, referring to Figure 1, a pressure local variation can be near three type of walls:

- A fixed surface (such as a body wall), in this case the pressure variation doesn't work, because the wall doesn't move
- A cylindrical spool surface (such as the lateral spool surfaces), in this case the pressure variation doesn't work, because the surfaces are always orthogonal whit respect of the permitted displacement
- A plane spool surfaces (such as the vertical spool surfaces), in this case the pressure variation works, because the force generated by itself is directed as the permitted movement.

Reducing A_{flow} means lowering the size of the areas on which the pressure variations act, but means also skewing that surfaces with respect to the displacement direction. Common spool valve sizes are more or less proportional to the maximum flow rate allowed, this means that trying to reduce A_{flow} in common spool valves is a drudgery; the minimization of A_{flow} is constrained by the need of an edge made up by mobile surface whose sizes are approximately equal to the base of a cylinder housing the spool.
One possible way to minimize A_{flow} is to use edges that move orthogonally with respect to the piloting devices (diaphragms, or rotational meterings) .

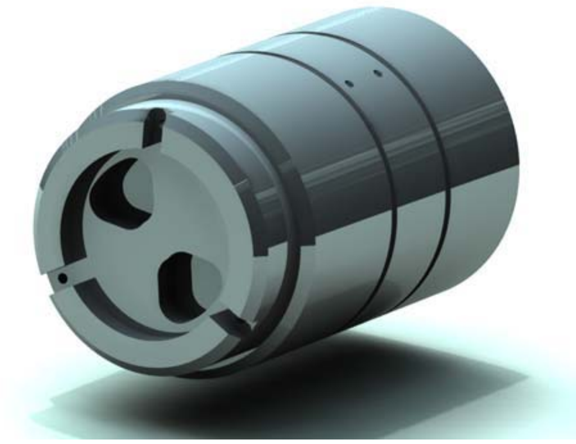

Figure 2 The ARM valve

THE PROPORTIONAL VALVE WITH AXIAL FLOW AND ROTATIONAL METERING

Trying to design a low-flow-forces valve, a new valve was conceived, named Proportional Valve with Axial Flow and Rotational Metering (ARM), and patented as Patent IT TO2007A000518 [4]. This valve almost insensitive to flow forces and allows an axial fluid flow. This latter aspect makes the component particularly suited for cartridge design.
The basic idea is to meter the pass through regulating two areas. The size of these areas is ruled by the relative rotation between two components (a *stator* and a *rotor*) shaped in order to commute from the "fully closed" position to the "fully open" through a relative rotation of a given angle φ. Pilot pressures and possibly springs act on an element called *reducer*. Its position rules the angular displacement between the stator and the rotor.

The ARM valve as a local compensator

Similarly to the common spool valves, the ARM valve can realize a large number of logical operations according as the arrangement of the pilot pressure intakes. For the sake of understanding, suppose that the ARM valve is used as a pressure compensating valve in a Load Sensing Flow Sharing System (LSFS).
Referring to Figure 3, the task of this valve is to keep the pressure in p_{m1} at the Load Sensing value p_{LS}. The volume p_{m1} is connected directly to the supply port of the pump (supposed to be an LS compensated pump), from p_{m1}, the fluid passes through the compensator in

order to decrease the pressure, until it reaches the section load pressure. In order to perform this operation, the compensator meters the passing area until the pressure p_{m1} is kept equal to p_{LS}. Moreover, if p_{m1} is the highest pressure among all the sections, the shuttle valve S opens and the LS line reaches the pressure p_{m1}.
To perform this specific task, the ARM valve is made up by seven elements. Referring to Figure 4, these elements are: the stator (or the cartridge) (1), the rotor (2), a blocking bolt (3), the reducer (4), two spacers (5) and (6) and a spring (7).

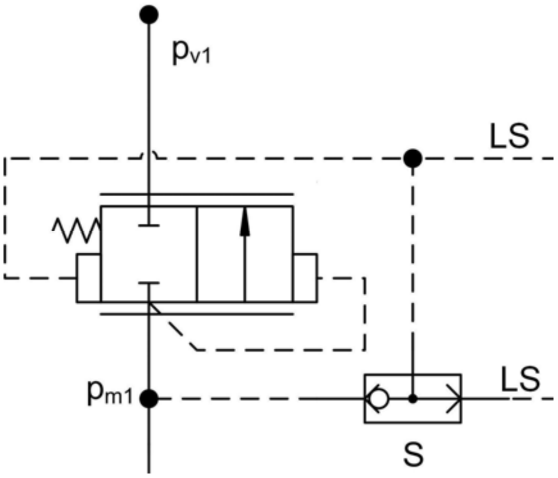

Figure 3 An LSFS local compensator hydraulic scheme

The cartridge (1) has a tubular shape, with a closed base in which two holes are shaped, in order to control a controlled area through which the fluid flows. Three straight grooves are machined inside the cylindrical surface of the cartridge allowing the stator to engage with the reducer (4), creating a prismatic coupling. Some holes are drilled both on the base and on the skin of the cartridge, so that the upstream pressure p_{m1} and the LS pressure p_{LS} can reach their chambers (the volumes from which the holes take the pressure signals are not represented, because they are obtained in the body of the valve housing the ARM). The ARM valve can integrate the function of the shuttle valve, in fact it can select the pressure signal of the highest load and send it on the LS line. This feature is obtained by means of three more holes drilled on the skin of the cartridge connecting the p_{m1} signal chamber to the LS when p_{m1} is higher than p_{LS}.
The rotor (2) is cylindrical as well. It has a closed base where two holes are located, in order to control a controlled area through which the fluid flows. These holes are made so as to meter (together with those on the cartridge) the passing area. The rotor is shaped to enter totally inside the cartridge. Three coil grooves are machined on the external skin of the rotor, thus allowing the rotor and the reducer to engage to each other. Holes and grooves are positioned so as to be in the "fully closed" configuration when the reducer is at its maximum displacement on the upstream side.
The blocking bolt (3) constrains the rotor to be fixed with respect to the cartridge along its axis, but permits the relative rotation of the two elements.
The reducer (4) is a flat ring that couples the cartridge and the rotor. Since the reducer is positioned between the stator and the rotor, it separates the volumes between the two elements into two chambers where it is possible to bring the two desired pressure signals.
The particular coupling choice of this device ensures that an axial translation of the reducer (with respect to the cartridge) causes an axial rotation of the rotor (with respect to the cartridge as well).
The axial position of the reducer univocally set the angular position of the rotor and, thereby, the metering of the valve. The axial position of the rotor is a function of the two pressures in the pilot chambers and of the spring force.

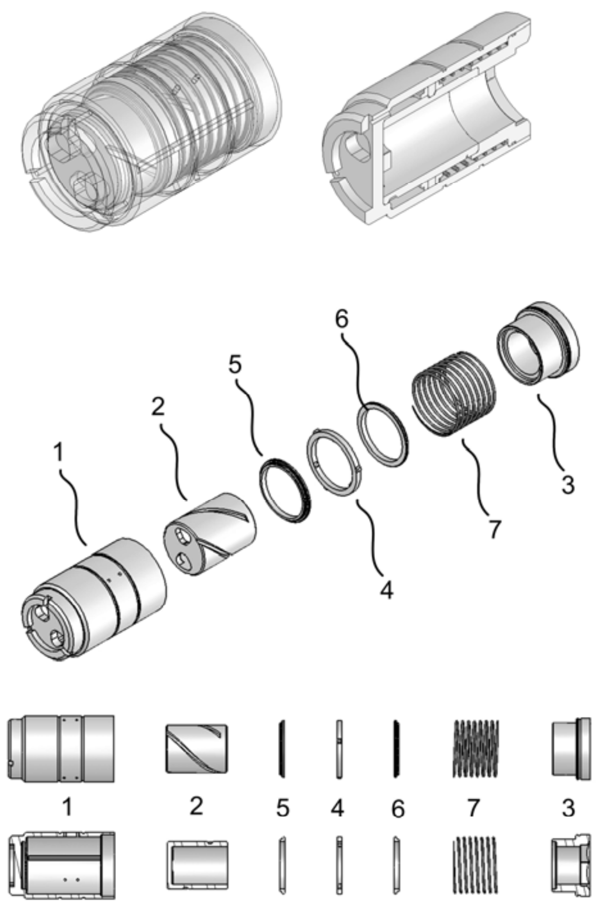

Figure 4 The ARM valve arranged as an LSFS local compensator

The spacers (5) and (6) work as end-stops for the reducer axial displacement and the spring (7) places the reducer against the upstream end-stop, so as to place the

ARM valve in "fully closed" configuration when all pressures are zero (requested feature in local compensator for LSFS valves).

ARM local compensator action

The ARM valve, arranged as previously described and positioned downstream a metering edge, can perform the same identical function of a traditional spool-based local compensator, integrating the selection of the maximum pressure signal (shuttle valve S in Figure 3).

When the section is not activated, the ARM valve is in the "fully closed" configuration (the spring pretension applies a very small threshold pressure). The upstream pilot chamber is subject to the upstream pressure p_{m1}, whilst the downstream chamber is at the LS pressure p_{LS} (the pressure of the highest load of all the active sections). When the section is active, the supply flow finds a closed port (ARM "fully closed"), and then the pressure p_{m1} starts rising. As soon as p_{m1} exceeds p_{LS} the reducer moves backwards (towards the downstream of the valve).

The more the reducer moves towards, the more the rotor rotates. In fact, named α the coil angle, if the reducer moves backwards a stroke x_r, the rotor covers an angle ϕ given by Eq 3:

$$\phi = \frac{x_r}{\tan(\alpha)} \frac{1}{R_{coil}} \tag{3}$$

The pressure upstream the valve works to gradually line up the holes of the cartridge and of the rotor (opening the valve). The ARM enters its metering condition, and $p_{m1} = p_{LS}$ (neglecting the spring and the flow forces). If p_{m1} is the highest pressure among all sections, p_{LS} can't counterbalance it, therefore the reducer moves totally backwards reaching the downstream end-stop. In this position there is an open link between the upstream pilot chamber and the LS line, inducing $p_{LS} = p_{m1}$.

Further lowering of flow forces

We had already described the straightforward approach to reduce flow forces sensitivity of the valves, namely an action on the mean pressure local variation p_{flow} and the area on which it acts A_{flow}. At this point, since the principles of the ARM valve are known, it is possible to show how this device can reduce further the flow forces sensitivity by a smart kinematic chain that makes flow forces "work badly".

The regulation error introduced by the flow forces can be expressed by the pressure $p_{err} = F_{flow} / A_{pil}$. p_{err} is the pressure that unbalances the system adding itself to the pilot pressure that moves the devices towards the "fully closed" direction. So p_{err} is a significant error indicator that considers the flow forces modulus, but also how them works on the device. Using p_{err}, instead of the usual F_{flow}, we can consider not only the influence of the valve shape on the regulation error, but also the influence of the piloting device of the valve.

In case of a common spool valve $F_{flow} = p_{flow} * A_{flow}$, so $p_{err,s}$ value is given by Eq. 4:

$$p_{err,s} = (p_{flow,s} A_{flow,s}) / A_{pil,s} \tag{4}$$

In the ARM valve, some additional considerations must be applied:

- $p_{flow,A}$ acts on walls that rotate, instead of translating
- $p_{flow,A}$, supposed constant, working on A_{flow} causes an axial torque $T_{flow} = p_{flow,A} * A_{flow,A} * (R_{flow} / 2)$, where $(R_{flow} / 2)$ is the mean radius on which p_{flow} acts
- named R_{coil} the radius of the coil of the rotor ($R_{coil} > R_{flow}$), the torque T_{flow} is countervailed by a tangential force $F' = T_{flow} / R_{coil}$
- named α the coil angle, the axial force needed to generate a tangential force equal to F' is F_{flow} and its value is $F_{flow} = F' * \tan(\pi/2-\alpha)$

As a consequence, in case of ARM valve, $p_{err,A}$ is given by Eq. 5:

$$p_{err,A} = \left(\frac{p_{flow,A} A_{flow,A} (R_{flow} / 2)}{R_{coil}} \right) \frac{\tan(\frac{\pi}{2} - \alpha)}{A_{pil,A}} \tag{5}$$

For the sake of comparison between a spool valve and the ARM valve, some hypothesis on the ARM performances must be introduced:

- $R_{coil} = R_{flow}$
- $A_{flow} = 0.6 * A_{pil}$ for both valves
- $\alpha = 45°$, so $\tan(\pi/2-\alpha)=1$
- $p_{flow,\,s} = p_{flow,\,A}$

Using these assumption $p_{err,s} = 2 * p_{err,A}$. More realistic hypothesis are: $R_{coil} = 1.3 * R_{flow}$, $A_{flow,A} = 0.2 * A_{pil,A}$, $\alpha = 45°$, $p_{flow,\,s} = 8 * p_{flow,\,A}$, these new assumptions show that a common spool compensator is up to ten times more sensitive to flow forces than an ARM valve.

CFD ANALYSIS

A preliminary numerical analysis campaign was performed in order to verify the performance of the ARM valve. It focused on the flow forces and the metering characteristic of the new valve. The analysis shows that the ARM valve is scarcely affected by the flow forces, due to its particular rotating metering edges and its kinematic chain, that dramatically decreases the force that countervail the valve pilot pressure.

The following example allows the reader to benchmark a common spool valve with respect to the ARM valve. In this case the boundary conditions are the upstream and downstream pressures (p_{m1} = 300 bar, p_{v1} = 100 bar) and the pressure gap is 200 bar. Figure 5 is shows the pressure contour over the only surfaces that can perform a non-zero work on the pilot system. Some local pressure drops (about 50 to 100 bar) are on these surfaces (blue colored), but it is easy to notice that these zones are very small, inducing a small torque T_{flow} of 0.14 Nm that, thanks to the kinematic coupling, causes only 0.5 bar p_{err}.

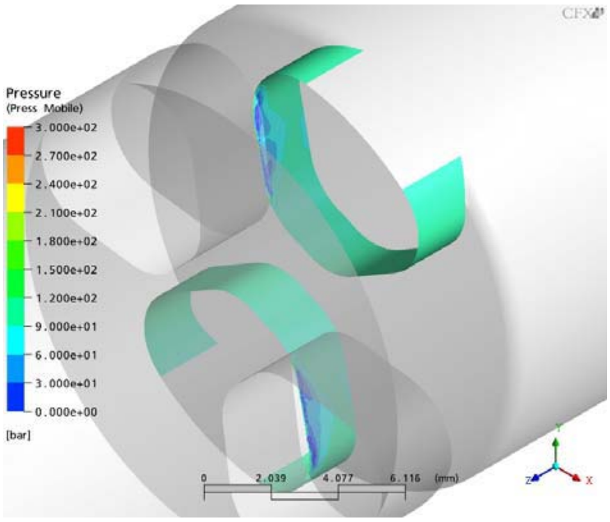

Figure 5 Contour plot of pressures on the ARM valve

Figure 6 shows the results of a spool compensator valve studied in the same boundary conditions (p_{m1} = 300 bar, p_{v1} = 100 bar). It is easy to notice that there are wider pressure recoveries on the mobile surfaces of the spool (red to green colored). These pressure recoveries generate a F_{flow} about 110 N, that means (in the specific case) a p_{err} close to 5 bar.

FURTHER ASPECTS

ARM valve features additional interesting characteristics both on functionality and machining.
It has an extremely compact design (axial dimension about 50 mm, radial dimension 25 mm, with nominal flow of 85 l/min and nominal pressure drop of 3 bar) including all inside components, making the design specially suitable for cartridge valves. The overall dimensions are comparable to those of a fitting and can be integrated in a valve port or hose fitting with almost negligible impact on block size.
Valve dimension allow a very fast reaction. The moving part weights few grams, compared to the usual 50-200 g of conventional spools. This feature keeps inertia effect to a minimum.
The fast action makes the proposed valve suitable also for critical applications where fast reaction is needed.
As to the production features, ARM valve can be easily configured to perform a large part of the conceivable control logic functions usually performed by 2-way valves using the same set of base components. A complete series of valves can be produced from a limited set of common components. For instance, using the same components, two valves with opposite pilot pressure logic can be produced, simply modifying the mounting. This achievement is not trivial, since in traditional valves the exchange of pilot pressure logic forces modifications on valve housing, with additional production costs.
Different functions can be performed simply varying position and number of drilled holes or using different spacers. All results can be achieved with negligible additional costs.
Preliminary numerical investigations, both CFD and dynamic, confirmed the positive advances that the component makes possible. Physical prototype production is being undertaken and will trigger an experimental functional qualification.

Courtesy of Walvoil S.p.A

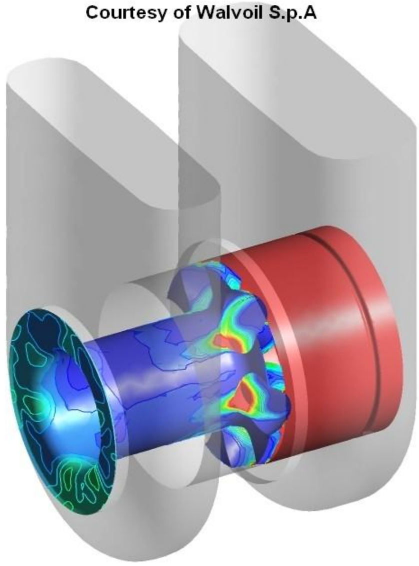

Figure 6 Contour plot of pressures on a spool compensator

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