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Design Of Two-Stage On/Off Cartridge Valves For Mobile Applications

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Abstract

Cartridge valves are widely used in mobile applications, where they are screwed in manifolds, to realize opportune circuit layouts. These valves are quite simple in operation but require a sophisticated design in order to meet all the requirements needed in the mobile machines. Typically, the design process is developed realizing a first design concept and some prototypes and experimentally testing them; after this, the designer chases the optimal performances requested to the valve with a trial and error approach on the prototypes, involving high time and cost resources. In this paper an alternative design procedure is proposed, which involves dedicated simulations to analyze the main critical issues regarding the cartridge valve object of the study. Modelling and simulations here have been considered as steps into the design process of a new valve, which satisfies the requirements and well adapt to the necessities to operate at higher flow and pressure levels without compromising its performances. In that way, the number of prototypes, realized to validate the numerical results and verify the design process, has been considerably reduced, together with related time and costs.

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Keywords: Cartridge valves, design, simulation, dynamic behaviour, flow forces, pressure losses, tightening torque.

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1. Introduction

This paper studies the design of an electrohydraulic on/off cartridge valve used in mobile applications, with the aim of improving the performance characteristic of the valve, allowing the managing of increased pressure and flow rate levels, without changing the valve size and worsening its performances.

Cartridge valves are especially useful in mobile equipment where space limitation can be a critical issue. Electrohydraulic cartridge valves can be easily proportionally controlled to manage a load, or they can perform an on/off logic control to open /close some sections of the circuit or retain the load when needed. For the easiness and flexibility they offer in the control, also in complex hydraulic systems, they are widely used in off-road vehicles. In this kind of applications, moreover, a great attention is given to efficiency of hydraulic systems, which strongly affects the fuel consumption and pollutant emissions of the vehicle. Many works in this field have been developed lately, see for example [1] and [2], where the authors analyzed the case of an excavator, or [3] and [4], where the authors focused on the hydraulic circuit of an agricultural tractor. Since the problem of efficiency in these systems is so critical (efficiency of hydraulic systems in mobile applications is around 21%, according to [5]!), in the traditional valve controlled systems many researchers have proposed to use multiple proportional cartridge valves to substitute the traditional spool valves, thus lowering the metering pressure losses in the system and allowing intelligent control of the loads. In that way the first independent metering valves with 4 or 5 proportional cartridge valves were designed, which substitute the metering action of the traditional single spool distributor used in mobile applications to control the loads [6, 7, 8]. Furthermore, On/Off cartridge instead of proportional cartridge valves may be used, following the new concept of “digital hydraulics”, where a number of on/off valve opportunely controlled realize “discrete” metering characteristics, guaranteeing fast responses and lower pressure losses with respect to the traditional spool valves [9].

Cartridge valves are quite simple in operation but require a sophisticated design in order to meet all the requirements needed in the mobile machines. The first requirement regards the fluid-dynamic behaviour of the valve: the objective to improve efficiency of the systems and to decrease fuel consumption leads to the request of valves able to manage flow introducing very low pressure losses. Since the valves considered in the paper are solenoid operated valves, an optimized design may also lead to less energy needed to operate the valve, and this is an aspect much appreciated also for the minor heating up of the solenoid itself. A 3D Computational Fluid Dynamic analysis (CFD) has been realized in this work to have a better insight on the flow within the valve and to analyze opportune geometries for the poppet and the body able to reduce the flow forces acting on the main poppet. This approach has already proven to be reliable and useful in the design of optimized valve geometry, as shown for example in [10, 11, 12] and also in [13, 14] and in [15]. The second requirement is related to the time response of the valve, which has to be as small as possible, in the order of 50÷70 ms: the time response strongly depends on the behaviour of the pilot stage and again on fluid-dynamic characteristics of the main poppet. A delicate system of orifices and internal passages in the valve affects the time response and has to be conveniently calibrated. A 0D (lumped parameters) model has been developed and presented in the paper to study these aspects, allowing also to have a better insight in the valve internal elements behavior: the valve analyzed has a pilot and a main stage and the model allows to observe the movements of the two in different operating conditions and for different design sets, studying the effect of the chosen design parameters on the entire valve dynamic response. This approach is often followed by researchers, for example by [16], [17], [18], for its effectiveness and for the difficulty (in terms of time and cost needed) to experimentally measure the valve internal variables. A third requirement is the need for the valve to resist to dynamic loads and vibration without unscrewing from the manifold. To achieve this, an opportune tightening torque has to be chosen, which has to be the optimal compromise between the anti-unscrewing requirement and the small deformations induced on the body of the valve by the tightening torque itself. Due to the low tolerances in fact, the deformations on the valve body may hinder the main poppet movement inside the valve. This topic is not so often addressed in literature, where the untightening torque is instead studied for bolts or fasteners, for example in [19], [20]. A structural analysis with Finite Element Method (FEM) has been constructed and is presented in this paper in a way to study the deformation of the valve body and the stresses, to verify the ability of the main poppet to move smoothly inside the valve. The results coming from these analyses has allowed to guide the design process of a new cartridge valve, able to manage an extended range of flow rate and pressure levels and respecting the target performances requested in mobile applications.

2. Cartridge Valve Description

The valve studied is controlled through a pilot stage operated by a solenoid: the pilot stage acts on a needle, the main stage opens as a consequence of the decreasing of pressure in the back chamber of the poppet when the needle of the pilot stage is moved away from its seat. In Fig. 1a, the main elments constituting the valve are shown:

- on the right side, within a core tube 1, a plunger 2 can move to the rigth against a spring 3, when the coil of the solenoid (not represented) is activated by the current and attracts the plunger itself. Together with the plunger, when the secondary and internal spring 4 is totally compressed and the magnetic force has increased to an opportune value, the plunger is able to carry the needle 5 away from its seat.
- on the left side, the valve body 6 allows the valve to be screwed in hydraulic manifold thanks to a threaded part 8 and houses eigth radial holes 9 to allow the fluid flowing through the valve; within the body valve, a poppet 10 can move away from the seat letting the fluid flowing from one port to the other. This movement is possible only when the back chamber 11 of the poppet is depressurized because the pilot needle opens a passage to the tank for the fluid in the chamber.

In Fig. 1b, the four check valves are shown (12, 13, 14, 15), which allow the operation of the cartridge valve in both the flow directions. A first critical issue for the design of this kind of valve is related to the position of the poppet when the valve opens: as much the poppet moves away from the seat as much the pressure loss for the fluid passage will be lower, but there is a limit in terms of the solenoid force, which may not be high enough to open fully the needle, and hence also the main poppet. A second issue of the valve is the time response: not considering the solenoid response delay, which has not been modelled in this work, the time response of the valve strictly depends on the movement of the plunger and the needle (the choice of springs 3 and 4 is hence critical) and on how much rapidly the pressure rises and decreases on the back chamber 11, i.e. on the dimensions of passages 16, 17 and 18 and on the overall volume of the back chamber and also of the dead volume in the pilot stage. In the A to B flow direction orifice 16 and check valve 12 are responsible for the building of pressure in chamber 11 (valve closing), while orifice 17 and check valve 13 for the lowering of pressure (valve opening); in the B to A flow direction, orifice 18 and check valve 14 enable to pressurize chamber 11 determining the opening of the valve and orifice 17 and check valve 15 discharge pressure in chamber 11 and let the valve closing.

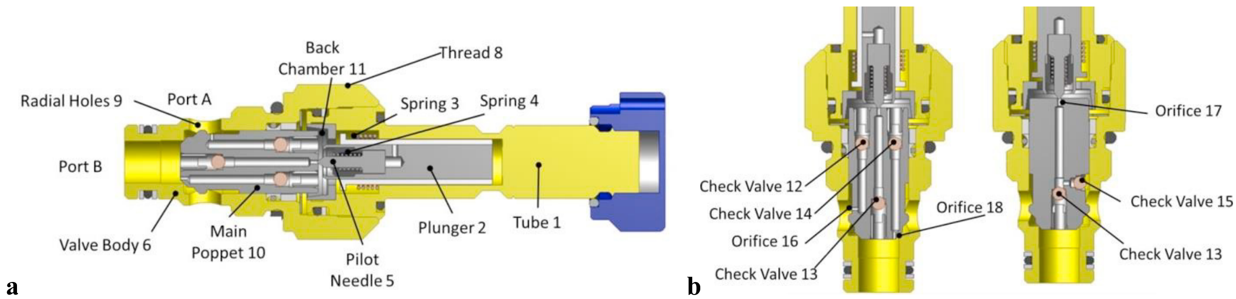

Fig. 1. CAD cross view of the cartridge valve (a); details of the main poppet (b)

It is clear that many design parameters are involved in determining the performance of the valve, and that the classical method of the design, based on the experience and on the trial and error approach, leads to the realization of a high number of prototypes to be experimentally tested to obtain a satisfying result. Time and costs related to this approach are not acceptable anymore in the modern market of fluid power systems. That's way modelling and simulation have been applied with different tools and objectives, during the design process of the new valve. In particular, the aim of the design was here to extend of 30% the range of operation of the valve (higher flow rate and pressure values), in a way to cover the more and more demanding applications of fluid power systems in off-road vehicles, while maintaining the same size for the valve.

3. Lumped parameters model of the cartridge valve

A lumped parameters model of the cartridge valve allows to study the dynamic behavior of the valve (time responses, eventual vibration or fails in the operation), maintaining under control also the stationary characteristics (pressure losses). Starting from the old architecture of the valve, some suggestions coming from the simulations have been applied in the design of the new valve. The lumped parameter model has been realized in LMS Imagine.Lab AMESim environment [21], which allows to model hydraulic components in detail with the Hydraulic Component Design library. The model is shown in Fig. 2, with the main elements labeled as in Fig. 1. The solenoid is not physically represented, but the model receives as input the magnetic force as function of the plunger position, as it is experimentally measured on the test rig.

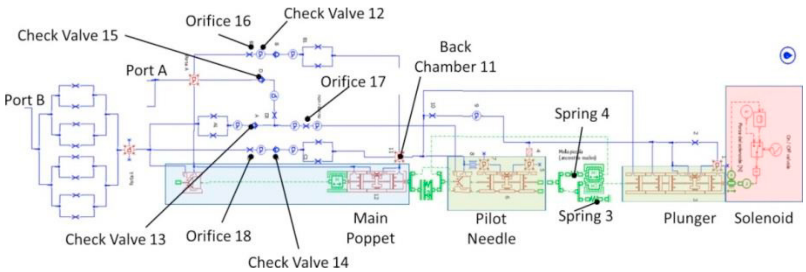

Fig. 2. AMESim model of the valve.

The flow contributions Q through orifices 16, 17, 18 are modelled as in Equation (1), where A is the flow area, p is the wet perimeter of the orifice section, Δp the pressure drop across the orifice, ρ the fluid density and μ the dynamic viscosity, v the fluid velocity, D_h the hydraulic diameter C_d the discharge coefficient:

$$Q = C_d \cdot A \cdot \sqrt{\frac{2 \cdot |\Delta p|}{\rho}} \cdot sign(\Delta p),\ C_d = C_\infty \cdot \tanh\left(\frac{2 \cdot \lambda}{\lambda_{cr}}\right),\ \lambda = \frac{\rho \cdot v \cdot D_h}{\mu},\ D_h = \frac{4 \cdot A}{p} \tag{1}$$

In Equation (1) the discharge coefficient C_d is defined as a function of the actual Reynolds number value λ computed during simulation, of the critical Reynolds number value λ_{cr}, i.e. the value for which the flow is changing from laminar to turbulent, and of the asymptotic value of the discharge coefficient C_∞, which is the value reached by the discharge coefficient in fully turbulent condition. In Fig. 3, the pressure losses measured on the test rig on four prototypes of valves are plotted against the numerical results, showing a good comparison; pressure losses and flow rates are non – dimensioned with respect to the maximum values tested. The time responses of the valve during the opening and closing operations were obtained on the test rig, measuring the pressure transient at the inlet port of the valve: being a normally closed valve, the opening time has been taken as the time interval needed to the pressure at the inlet to drop of the 10% with respect to the maximum pressure considered in the test; the closing time, is the time interval needed for the pressure at the inlet to grow again of the 10% with respect to the pressure of the outlet (tank). The average time response of the valve during opening and closing (the average was done on 10 prototypes tested) has been used as a reference for the numerical time response, in a way to verify the correctness of the settings in orifices 16, 17 and 18. At this point, having verified the reliability of the dynamic model, different operating conditions were tested, verifying that the model is able to depict the trend in the valve behavior (i.e. the time response is changing in the numerical model as it does in the real valve). Some critical issues have been analyzed for this version of the cartridge valve, looking at the displacements of the moving elements and at the pressure inside the back chamber (all variables that are difficult to be measured on a test rig, because they need miniaturized sensors and a prototype of the valve opportunely modified for the purpose). The first issue was the fact that the main poppet finds its equilibrium position before having reached the total displacement that it can travel, when the solenoid is activated both in A to B and in B to A configuration. This issue depends on the depressurization of the back chamber 11 and hence on orifice 17. To better control the dimension of the orifice during the poppet manufacturing, it was chosen to adopt a shorter poppet in which orifice 17 is nearer the "discharge port" and hence a longer needle, having its seat moved towards the poppet end, as shown in Fig. 4. The poppet is lighter, orifice 17 is better positioned and it is expected that the poppet can move further when the valve opens. This means less pressure loss with the same flow, or, changing point of view, it allows the valve to operate with higher flow rate with the same pressure loss.

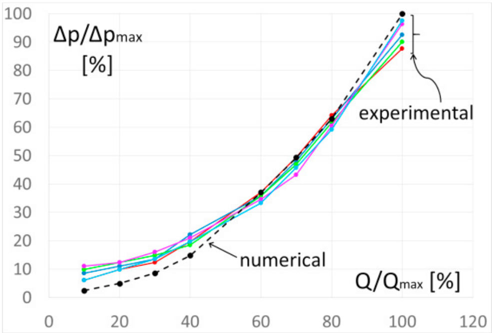

Fig. 3. Experimental and calculated pressure losses in the valve main poppet.

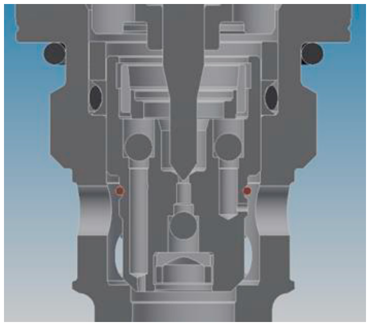

Fig. 4. Cad cross section of the new valve: detail of the needle and poppet.

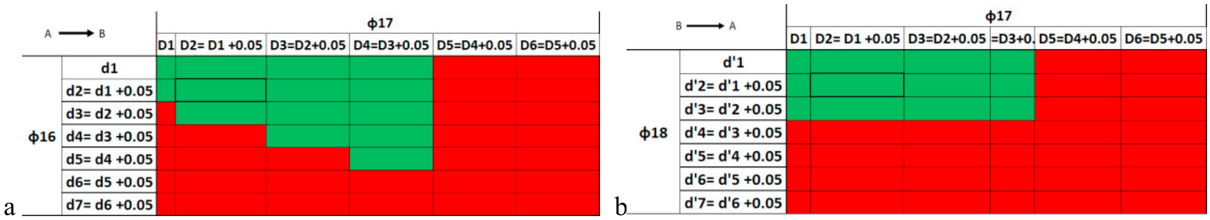

Fig. 5. Tables of the pairs of orifices 17 and 16 (A to B flow (a)) and orifices 17 and 18 (B to A flow (b)); green boxes mean that the pair of diameters chosen lead to a good behaviour of the valve in terms of time response and pressure losses, red boxes choices instead lead to valve problems.

A second issue is the correct choice of the diameters of orifices 16, 17 and 18. For some values within the manufacturing tolerances of the orifices diameters the main poppet travels a very little displacement and pressure losses become excessively high. For the new version of the valve, shown in Fig. 4, to better understand the role of the orifices diameters values, a campaign of numerical simulations was performed, varying the diameters of the orifices and checking both the time response and the pressure loss of the valve. In the tables of Fig. 5, the green boxes indicate that the valve opens with low pressure losses (equal to the target fixed for the new valve by the manufacturer) while the red boxes indicates that the valve doesn't open or close at all or the pressure losses are too high or the time response is too large. Starting from the reference values of the diameters of the old version of the valve, there's a little interval available for the choice of the diameters and tolerances in manufacturing that has to be strictly respected. If the orifice 17 is too big, being responsible for the chamber 11 depressurization, the time response in opening is very small but the main poppet fails in closing in an opportune time interval. If the orifices 18 or 16 are too big, the closing time is perfect but there is the risk that the main poppet doesn't move sufficiently (high pressure loss) or even at all. Time responses for the valve at the maximum flow rate spans from 30 ms to 60 ms for the opening and from 60 ms to 100 ms for the closing, when changing the diameters of orifices.

4. CFD Analysis

The Computational Fluid Dynamic Analysis has been used first to determine the flow parameters for the three orifices (C_∞ ,λ_{cr}) and to evaluate the flow efflux through the main poppet. In this section, we focus the attention on the main poppet geometry and the modification introduced to reduce the flow forces in the new valve, thus allowing a greater displacement for the main poppet and a lower pressure drop through the valve.

The analysis was performed using the commercial software Autodesk [22]: the 3D model considered is constituted by a portion of the valve body and by the main poppet, as shown in Fig. 6, for the original geometry of the poppet on the left, and the new one on the right, which has a sort of fin nearby the conical part of the poppet. This fin is supposed to better guide the flow through the 8 radial holes, which are manufactured on the valve body around the poppet, thus lowering the flow forces. The geometry and position of the fin clearly determine the effectiveness on the fluid forces on the poppet.

The mesh is unstructured, with the exception of a thin layer of structured mesh near the walls; a sensitivity analysis of the mesh size was performed, finding the right compromise between the convergence of the results and the computational time with a mesh of 643133 elements (119951 nodes). The general purpose standard k-ε turbulence model was chosen for its robustness. The fluid is mineral oil with a kinematic viscosity of 46 cSt at 40 °C and density of 860 kg/m^3, as the one used in the test rig. As boundary conditions, the flow rate is imposed at the inlet and the pressure at the outlet (tank pressure). The analysis is performed keeping the main poppet opened at a certain displacement and analysing the flow through the poppet itself; the residual force on the poppet due the fluid pressure is calculated as the flow force on the poppet.

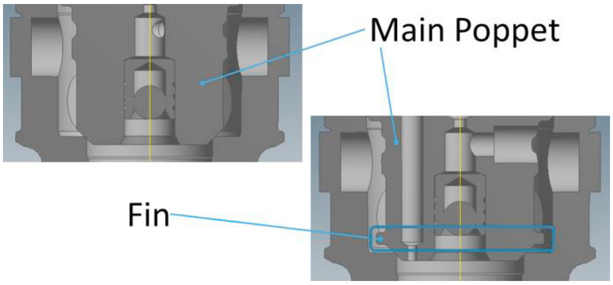

Fig. 6. Old (a) and new (b) geometry of the valve main poppet.

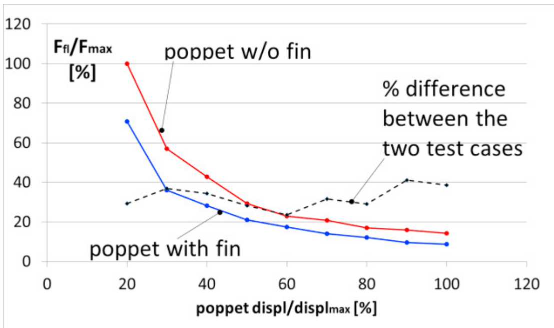

Fig. 7. Non-dimensioned flow forces vs non-dimensioned poppet displacement.

The Fig. 7 shows the flow forces trend with respect to the poppet displacement (both non-dimensioned with respect to their higher values). An average reduction of the 30% in the flow forces allow the poppet to move further and pressure losses to be consistently reduced; in that way the valve may operate with higher flow rate values that the nominal ones, as requested in modern applications.

5. Structural Analysis

The project of the new cartridge valve aims not only to manage higher flow rate with acceptable pressure losses, but also to extend the range of the maximum pressure managed by the valve. For this reason, first of all static and dynamic structural analyses based on Finite Element Method (FEM), again with [22], were developed, assessing the capacity of the valve body to resist the maximum pressure load in static and dynamic loading configuration. There is, moreover, an important issue to be addressed, which is described in this section, regarding the deformation of the valve body due to the tightening torque applied: the cartridge valve in fact is screwed on a manifold with a tightening torque that has to be opportunely calibrated. The torque has to be sufficiently high to prevent untightening of the valve due to vibration that typically involve hydraulic systems that equip both industrial than mobile machinery; on the other side, the deformation on the valve body due to the tightening torque has to be within the manufacturing tolerances and has not to hinder the movement of the main poppet within the valve.

To study this problem with a FEM approach, the elements to be considered are the valve body and a manifold or a sufficient portion of it. The chosen constraints were: a constraint of contact on the surface under the "head" of the body valve; again a constraint of contact on the single surfaces of the flanks in the thread that was reproduced in the

original CAD model. The mesh has been chosen setting the average and minimum dimensions of the element (respectively 0.05, 0.04), which are non-dimensioned factors, defined as relative to the maximum dimension of the valve. Going to lower values with these parameters means much more computational time with little differences in the results (differences for displacement are on the order of 0.01 μm).
The load can be imposed as an axial load F derived from the tightening torque T, using for example Equation (2), valid for 60° standardized thread [23]:

$$F = T \Big/ \left(0.16 \cdot p_{th} + 0.58 \cdot \mu_{th} \cdot d_2 + \mu_k \cdot \frac{(D_{head} + d_{head})}{4}\right) \tag{2}$$

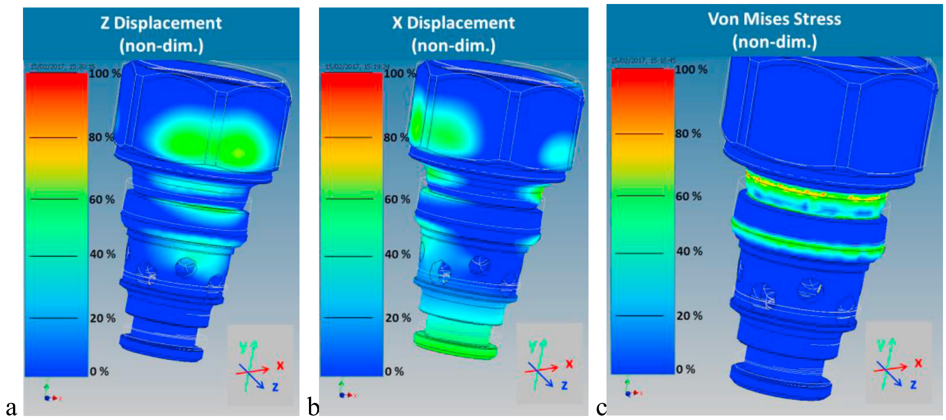

Fig. 8. Non dimensioned displacement in x and z axes (plane perpendicular to the symmetry axis of the valve) and Von Mises stress.

The axial load derived from equation strictly depends on the thread pitch p_{th}, flank diameter d_2, contact surface between the head of the valve and the manifold (D_{head} and d_{head} define the external and internal diameters of the annular ring surface under the valve head), and finally from the friction coefficients μ_{th} and μ_k in the thread and under the valve head. In Fig. 8, the results regarding the displacement along x and z directions are shown for the maximum value of tightening torque usually adopted for this size of cartridge valve: y axis is along the symmetry axis of the valve, x and z displacements allow hence to define the radial deformation of the valve body. Displacement is non-dimensioned with respect to the its maximum value, which is lower than 2 μm; the highest displacements values in the valve body are located nearby the 8 holes that allow flow entering and exiting the valve. The asymmetry in the deformation is given by the presence of the thread. These values of displacements give confidence that the main poppet can move smoothly inside the body valve. Moreover, the maximum Von Mises stress is much lower than the yield limit for the kind of steel considered, and located nearby the valve head. A little higher tightening torque is allowed, because the radial deformation of the body is within the manufacturing tolerances and the Von Mises stress is quite low.

6. Conclusions

In this work the design of an on/off cartridge electrohydraulic valve has been addressed by means of modelling and simulation. Many design parameters are involved in determining the performance of the valve, and the classical method of the design, based on the experience and on the trial and error approach, leads to the realization of a high number of prototypes to be experimentally tested to obtain a satisfying result. Time and costs related to this approach are not acceptable anymore in the modern market of fluid power systems. For these reasons, in this work the critical issues involved in the design of this kind of valve have been analyzed by means of different simulation approaches and tools, guiding the entire design process.

The dynamic behavior of the valve has been studied with a lumped parameter model, which also allowed having a better insight about the internal components movements and the design parameters that mostly affect the time response of the valve; in that way, both the problem of the time response and of the pressure loss through the valve were studied. The flow through the main poppet was addressed with a Computational Fluid Dynamic analysis, proposing a new geometry able to reduce the flow forces. Finally, by means of a Finite Element Model analysis, the problems of the deformation on the valve body due to the operating pressure and the tightening torque has been deepened. All these analyses contributed to the development of a new valve, able to manage greater flow rate and pressure, maintaining the same size and good dynamic behavior.

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