PRELIMINARY STUDY OF AN INNOVATIVE PROPORTIONAL VALVE CONCEPT

Alisson Dalsasso Corrêa de Souza, alisson@emc.ufsc.br
Victor Juliano De Negri, victor@emc.ufsc.br
Federal University of Santa Catarina – LASHIP - Postal Box 476 – 88049 470 – Florianópolis – SC – Brazil

Abstract. This paper presents an analysis of current developments in directional continuous control valves that are being proposed in several research centers, as well as the constructive and behavioral characteristics of an innovative concept called IRV (Inertial Rotating Valve). Directional continuous control valves, commercially known as proportional directional valves and servo valves, are the most important component in the control of electrohydraulic positioning systems and, consequently, their static and dynamic behavior substantially affects the performance of such systems. In the design of traditional valves, with slide spool, special care should be taken regarding the presence of static friction and hydraulic stick, in addition to the symmetry of the control orifices, which cause phenomena such as hysteresis, dead zone, reversion error and, consequently, errors in the controlled flow rate. In this regard, the performance characteristics of the concept principles that are being widely proposed are discussed in this paper. Following this, the dynamic modeling of the IRV, which explores the effects of fluid inertia without having positioning of movable parts, is presented. Finally, simulations with this model implemented in Matlab are compared with the experimental results of the valve prototype, demonstrating the feasibility of this project and indicating improvements for future studies.

Keywords: hydraulic control valves, innovative design, simulation

1. INTRODUCTION

The search for more efficient hydraulic control systems (faster, with smaller errors and greater repeatability, etc.) is continually ongoing. This can be verified by recent publications on controller design, for example Cruz and Ferreira (2003), Davliakos and Papadopoulos (2005), Kim and Lee (2006). For the same purpose, as Section 2 will show, innovations in traditional valves have also been accomplished in many research centers and industries.

Differently from these solutions, the purpose of the research presented in this paper is to solve dynamic and static behavior problems related to hydraulic control systems through a new form of flow control.

The importance of addressing this issue in such manner lies in the fact that, according to Andrighetto (1996), in spite of advances in technology, the use of proportional valves still requires an additional effort for the adjustment of the controllers and obtention of better dynamic behaviors, mainly as a function of their significant nonlinearities. So far, these problems are still relevant.

In this context, the concept to be explored may result in more effective position and speed control systems, allowing the industrial requirements, which are becoming ever more demanding, to be satisfied.

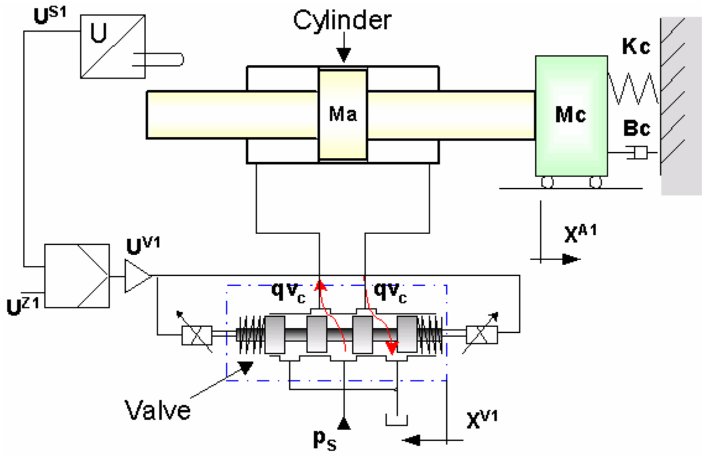
2. INNOVATIONS IN DIRECTIONAL CONTINUOUS CONTROL VALVES

Directional continuous control valves are known in industrial markets as proportional directional valves and servo valves. Two-, three- and four-way types are commercially available. As shown in Fig. 1, the most common directional continuous control valve uses a spool-like internal part to direct the flow. This form of construction requires that a movable part is moved, starting from a rest condition and reaching a new rest condition, involving its acceleration and subsequent deceleration.

Therefore, the alternated rest and movement conditions implicate the occurrence of composed friction forces, which cause phenomena such as hysteresis, dead zone, reversion error and, consequently, errors in the controlled flow rate.

However, the traditional valves have been improved through technological development, so that they can be used in several industrial applications. As a result, there are proven designs that incorporate the innovations into digital electronics, allowing modern on-board electronics for a direct connection to field bus systems. In its turn, many innovations for hydraulic valves can be observed, ranging from new ideas for the required electro-mechanical transformer to completely new mechanical designs.

As a first example of innovation, Fig. 2 provides a partly cross sectional view of the piezo-driven spool valve being developed at IFAS (Institute for Fluid Power Drives and Controls). As stated by Murrenhoff (2003), the valve spool is flow force compensated to minimize forces required from the piezo drive. All control electronics are integrated into the valve housing and using two piezo drives the system is thermally compensated. –3dB frequency ranges at 340 Hz and the -90° frequency approaches 270 Hz for 50% input signal. This development shows that a remarkable dynamic performance is possible with this directly driven valve. A further advantage is the low power compensation for the piezo drive while holding a position. On the other hand, a lack of reliability, high costs and small useful stroke have been presented as arguments disfavoring the use of piezo actuators.



Legend:

U^{S1}: voltage related to displacement transducer;
U^{V1}: drive voltage related to valve;
U^{Z1}: set-point voltage;
x^{A1}: actual position of the cylinder rod;
x^{V1}: actual position of the valve spool;
qv_c: control flow rate;
p_s: supply pressure;
Ma: mass of the cylinder;
Mc: mass of the load;
Kc: spring constant;
Bc: damper constant.

Figure 1. Diagram of the positioning hydraulic system.

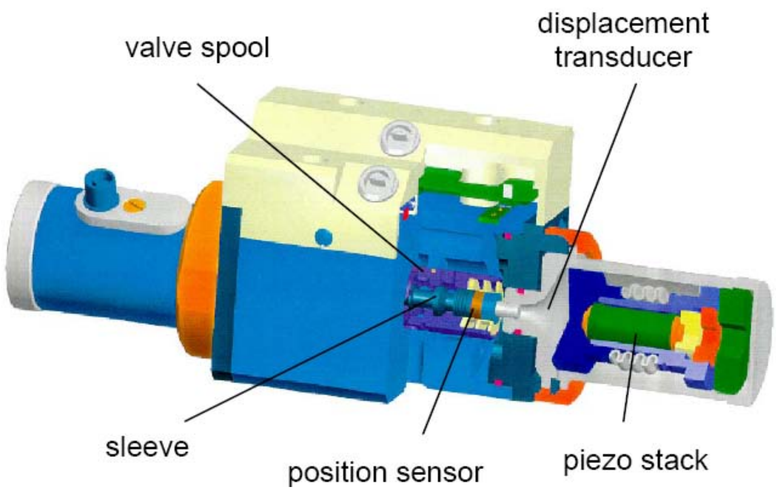


Figure 2. Directly controlled piezo valve with hydraulic displacement transducer (Murrenhoff, 2003).

In the following example (Fig. 3) the cross sectional view of a design that uses four individually adjustable metering edges is shown. As a consequence, the difficulty of requiring four matching metering edges with precision grinding involved for good static valve characteristics, for a 4-way valve setup, is overcome. According to Branz (2001), the stepper motor in the upper right part of Fig. 3 turns the shaft in order to displace the center body of the valve. The four individually adjustable spools are attached to this body each providing one metering edge. The shaft to the left side of the valve (lower center part of Fig. 3) provides a mechanical feedback from the actuator to the valve. In conclusion, with this setup the lapping condition can be set to the demand of the application.

Another development using stepper motors to directly drive a valve spool is depicted in Fig. 4. As indicated by Murrenhoff (2003), this valve is used in mobile hydraulic applications and the stepper motor replaces conventional proportional solenoids. The stepper motor drives the spool via a toothed bar and offers the advantage that no spool displacement sensor is required. This is possible because the stepper motor exhibits a detente torque providing a rigid system between the possible steps. Holding a position does not require electric power input. In this regard, the stepper motor performs similarly to the piezo drive and has the same advantage over the conventional proportional solenoid.

As a solution at lower cost, a particular type of two-stage valve developed by Vickers Inc. termed a “Valvistor” has an embedded internal hydraulic feedback that provides an efficient flow-force compensated proportional flow valve. As reported by Prasetiawan *et al.* (2001), the valve uses a small pilot circuit to drive the larger main flow, similar to the behavior of an electronic transistor; hence the name. Other advantages to this type of valve include a fast response, repeatability, and low hysteresis. Using a pulse-width-modulated (PWM) signal, the pilot valve controls a relatively small flow in the pilot circuit shown in Fig. 5. The pilot flow generates a pressure difference across the main poppet, and consequently causes it to move. The main poppet position determines how much fluid flows from the inlet to the outlet through the main circuit in Fig. 5. This position is fed back by the longitudinal slot on the poppet. Notably, the pilot flow is routed so as to converge with the main flow which increases the steady state efficiency, and at the same time affects the dynamic performance. In brief, a -90° frequency approaches 100 Hz and can operate at up to 160 L/min.

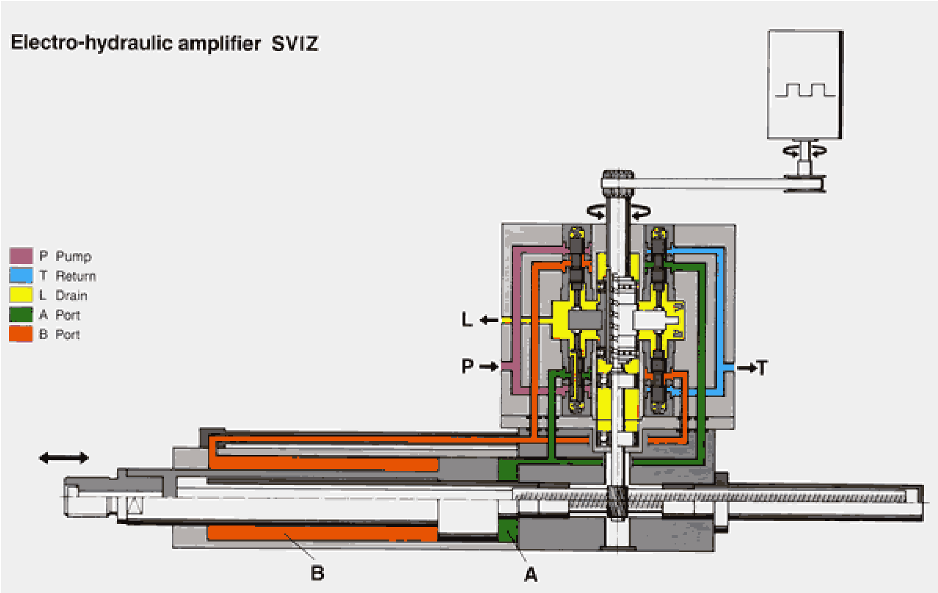


Figure 3. Servo valve with individually adjustable metering edges (Branz, 2001).

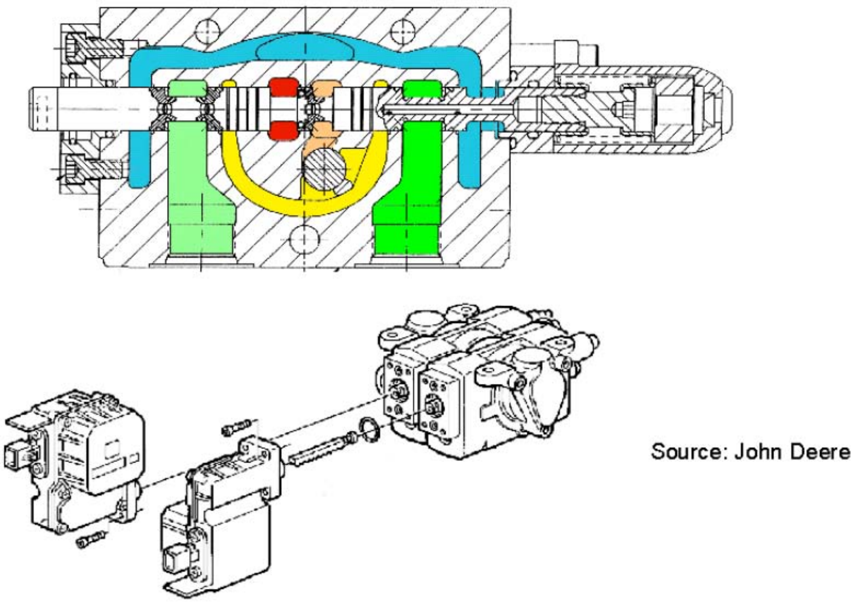


Figure 4. Stepper motor driven valve (Murrenhoff, 2003).

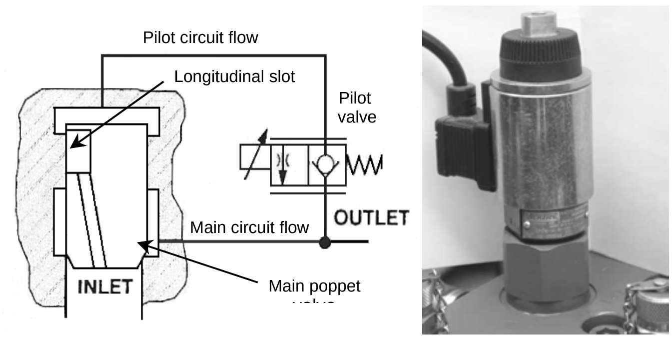


Figure 5. Valvistor diagram (left side) and photo of an actual valve (right side) (Prasetiawan *et al*., 2001).

Through an analysis of the concepts shown above, one can note that the improvements are based on mechanical principles of servo and proportional valves conceived several decades ago. For this reason, such improvements are evidently welcome, however, restricted.

Differently of the solutions mentioned above, Zaun (2004) reports an exception making use of an electrorheological (ER) fluid in an unconventional valve concept. The ER fluid is applied as a pressure media, which changes its behavior when an electric field is applied. In this case, the sheer stress of the fluid increases proportionally to the field strength and it appears as a viscosity change. This proportional relationship can be used to design servo valves without mechanical metering edges.

Figure 6 shows the circuit diagram of such a system. Pressure p_1 increases while pressure p_2 drops by activating valve 1A and valve 2B. This pressure difference causes the cylinder piston to move to the right. Activating valve 1B and 2A results in the reverse movement.

Figure 7 depicts the static flow valve characteristic with a programmed positive overlap and with constant supply pressure. According to Murrenhoff (2002), the ER valves do not require any hardware adjustments to change the lapping condition. Another advantage is the dynamics of the ER effect which can be used for high response servo hydraulic drives. A cylinder drive with such a control system has been investigated at IFAS and exhibited an excitation frequency of 400 Hz by means of a closed loop position control.

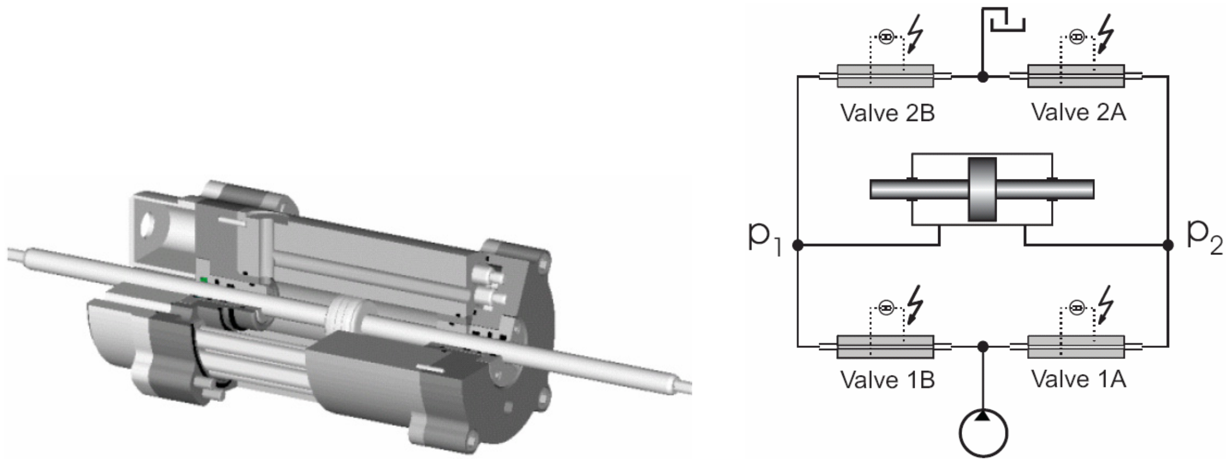


Figure 6. Servo valve (Murrenhoff, 2002) and actuator using an electrorheological fluid (Zaun, 2004).

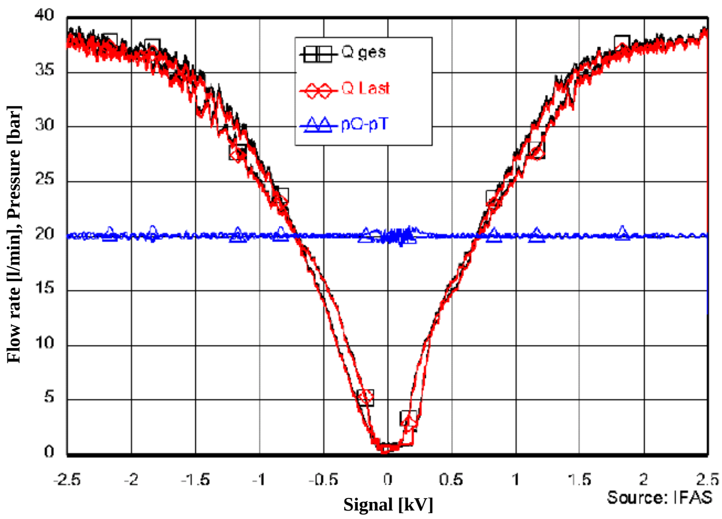


Figure 7. Pressure-flow characteristic for positive overlap (Murrenhoff, 2002).

Based on a technology still in development, this last design reveals the interest in a concept that does not demand the positioning of movable parts.

In view of this, the development of new conceptions of valves may offer a solution for the progress of the hydraulic control systems. Thus, the development of an original flow control form, materialized in a hydraulic valve, would represent a real innovation over proven proportional valves and servo valves in the current market.

3. CONCEPT OF THE INERTIAL ROTATING VALVE

This conception substantially differs from the existent solutions for controlling the fluid flow, since it is carried out without requiring the positioning of movable parts to promote the opening of the control orifices. To be precise, it is accomplished by the control of the aperture time of these orifices (De Negri, 2004).

As a starting point, one can consider the IRV as being composed of one rotor and one chamber[1] (V2) situated before the interior of the valve, as shown in Fig. 8. To make things easier, we will assume that the rotor has only one radial orifice (actually, it has four orifices). Therefore, in this configuration, two circular orifices must be aligned to permit free passage of fluid. The first refers to the traverse area of the pipe while the second refers to the area of the radial orifice of the rotor.

Accordingly, the opening and the closing of the fluid passageway (control orifice) is indicated in Fig. 9. In this case, the speed of the condition alternation (opened, closed or partially opened/closed) is proportional to the frequency of the rotor. The calculation of such area can be carried out as outlined in Merritt (1967).

As described in De Negri (2004), the functional principle of the IRV is founded on the occurrence of the alteration in the fluid flow due to the variation in the control orifice area. In other words, the anterior fluid masses of such control orifice are accelerated when the passageway area is increased and decelerated when it is decreased, in order to modify both the velocity and the flow rate of fluid. Consequently, considering a given time interval which corresponds to a complete operation cycle, the period that the control orifice is open is proportionally related to the average flow rate through the orifice.

Finally, the flow rate control can be obtained through the fluid accelerating or decelerating on either the anterior or posterior fluid volume of the control orifice.

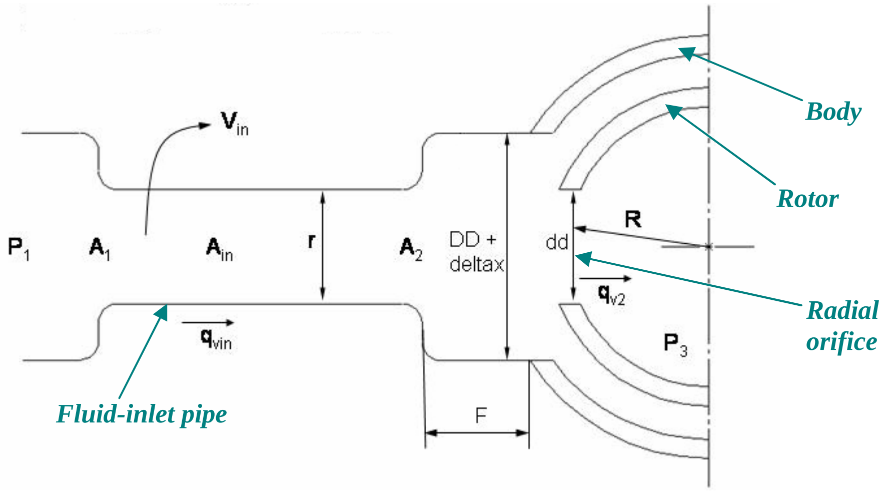


Figure 8. Sketch of the parts that comprise the IRV.

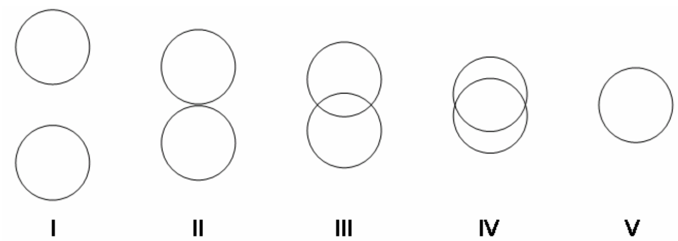


Figure 9. Gradual alignment of orifices of the IRV (I - not aligned, V - fully aligned) (Climaco and De Negri, 2004).

3.1. Prototype of the Inertial Rotating Valve

Starting from the information obtained through the simulation of a preliminary modeling, which is given in the next section, a conception of the inertial rotating valve was obtained for evaluating the feasibility of the constructive solutions development. This conception has been materialized in a prototype, main components of which, along with the final characteristics and geometric appearance, are given below (see Fig. 10).

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[1] This chamber is not the same as the chambers shown in next section. In fact, as the reader will see in Section 2.2, the purpose is to represent the compressibility of the fluid.

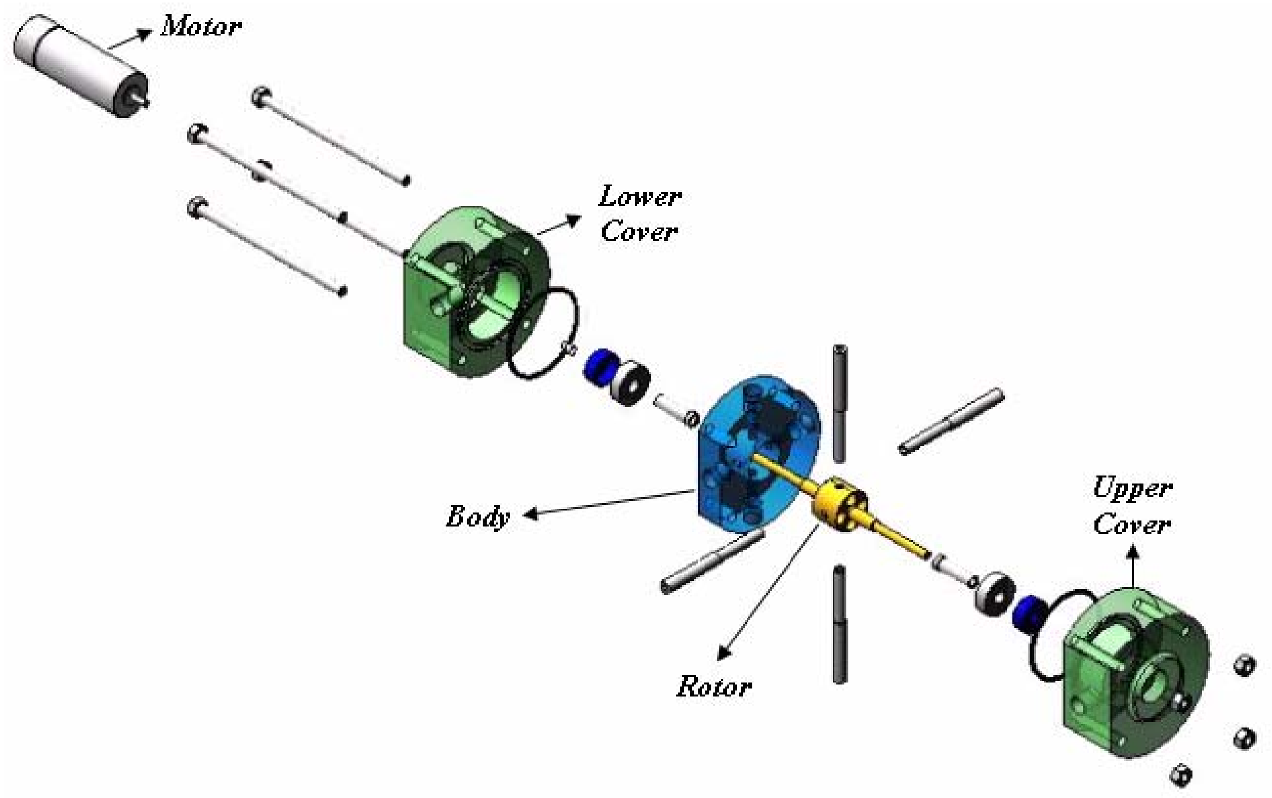
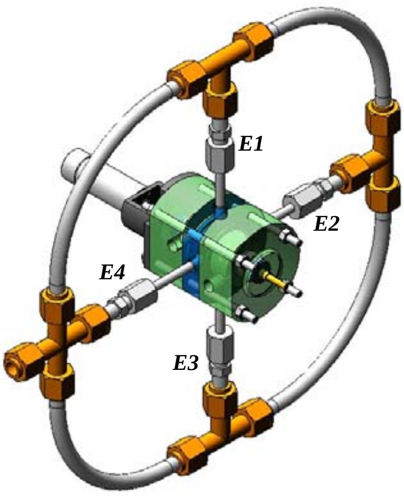


Figure 10. Main components of the prototype of the IRV (exploded view) (De Negri, 2004).

First of all, according to the left side of Fig. 11, the fluid is supplied to the valve by conducts E1, E2, E3 and E4 simultaneously. Afterward, as exhibited in the right side of Fig. 11, the fluid flows to the interior of the rotor when its radial orifices are aligned with the orifices of the body.



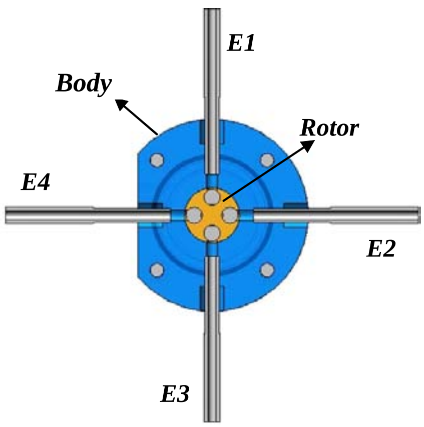


Figure 11. Collapsed view of the IRV with its admission conducts (left side) and alignment of the radial orifices of the IRV (right side) (De Negri, 2004).

The rotor, as well as the body, has four radial orifices for the admission of the fluid to its interior (see Fig. 12). In addition, the rotor has four axial orifices for the distribution of the fluid to the covers.

Therefore, according to Fig. 13, the fluid flows to the inner chambers of the covers through four axial orifices of the rotor. Finally, each cover includes one orifice where the fluid leaves the chambers and, as a result, the valve (see Fig. 13 and Fig. 11).

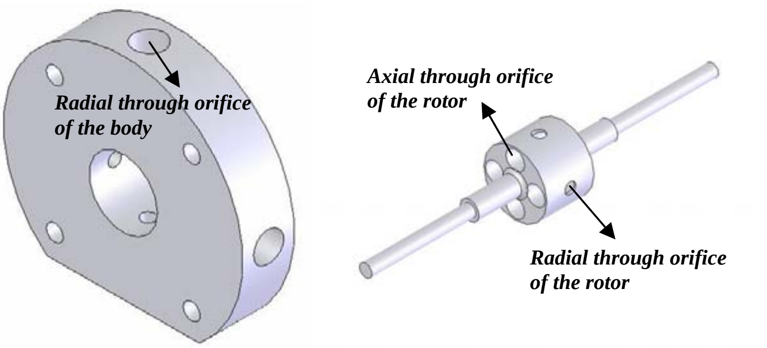


Figure 12. Axial and radial orifices of the IRV.

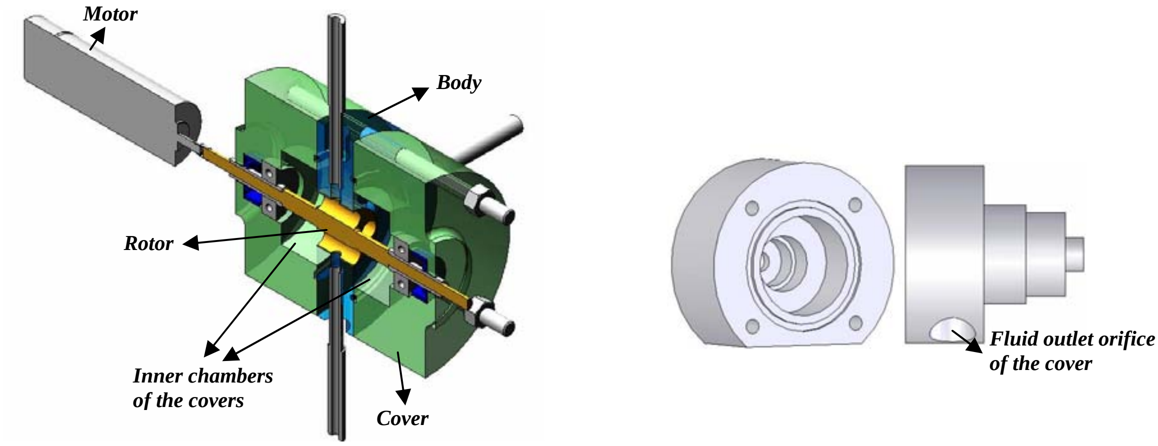


Figure 13. Inner chambers of the covers (left side) and fluid output orifices (right side) (Climaco and De Negri, 2004).

3.2. Preliminary Modeling of the Inertial Rotating Valve

The preliminary modeling of the IRV considered the sketch in Fig. 8 which was implemented in Matlab. However, we have assumed that the valve is composed of four inlet conducts and four radial orifices in the rotor, as shown in Fig. 11.

The modeling of the conducts (pipes) was achieved using the elastic model, which assumes that changing the momentum of the liquid causes deformations in the pipeline and compression of the liquid (see Eq. (1) and Eq. (2)). At the same time, variable orifices have been modeled as valves located in the end of pipes (see Eq. (3)).

$$\Delta p = p_1 - p_2 = \frac{fL\rho}{2DA^2} q_V \cdot |q_V| + \rho \frac{L}{A} \frac{dq_V}{dt} \tag{1}$$

where:
Δp = pressure drop across the pipe;
p_1 = pressure at position 1 in a pipeline (pipe inlet);
p_2 = pressure at position 2 in a pipeline (pipe outlet);
q_V = flow rate through the pipe;
f = Darcy-Weisbach friction factor;
A = pipe cross-sectional area;
D = pipe hydraulic diameter;
L = length of pipe between positions 1 and 2;
ρ = fluid density;
t = time.

$$q_{V1} - q_{V2} = \frac{dV}{dt} + \frac{V}{\beta}\frac{dp}{dt} \tag{2}$$

where:

q_{V1} = inlet flow rate into the chamber;

q_{V2} = outlet flow rate into the chamber;

V = volume of fluid in the chamber;

β = bulk modulus;

p = pressure of fluid in the chamber;

t = time.

$$q_V = C_d A_0 \sqrt{\frac{2\Delta p}{\rho}} \tag{3}$$

where:

q_V = flow rate through the orifice;

C_d = flow discharge coefficient;

A_0 = orifice passage area;

Δp = pressure drop across the orifice;

ρ = fluid density;

t = time.

Even being a simple modeling approach, the above equations were used to preview the dynamic behavior of the first prototype of the valve. One can verify that the flow rate (indicated as Qv2 in Fig. 14) changes in synchronicity with the orifice area and also observe the pressure behavior in the inlet chamber as a consequence of the opening and closing of the control orifice. Figure 15 shows the behavior with higher frequency when the maximum values of the flow rate decrease due to the inertial effect of the fluid.

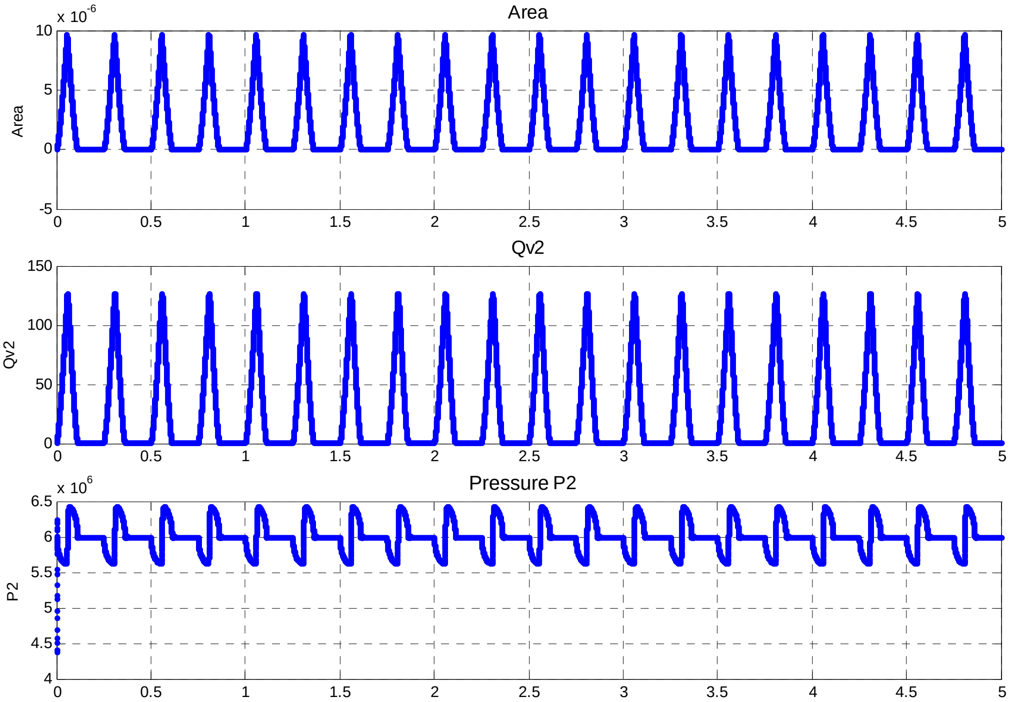


Figure 14. Simulation: 4 axial orifices in the rotor, 4 axial orifices in the body, 20 mm diameter rotor, 1 Hz frequency rotor.

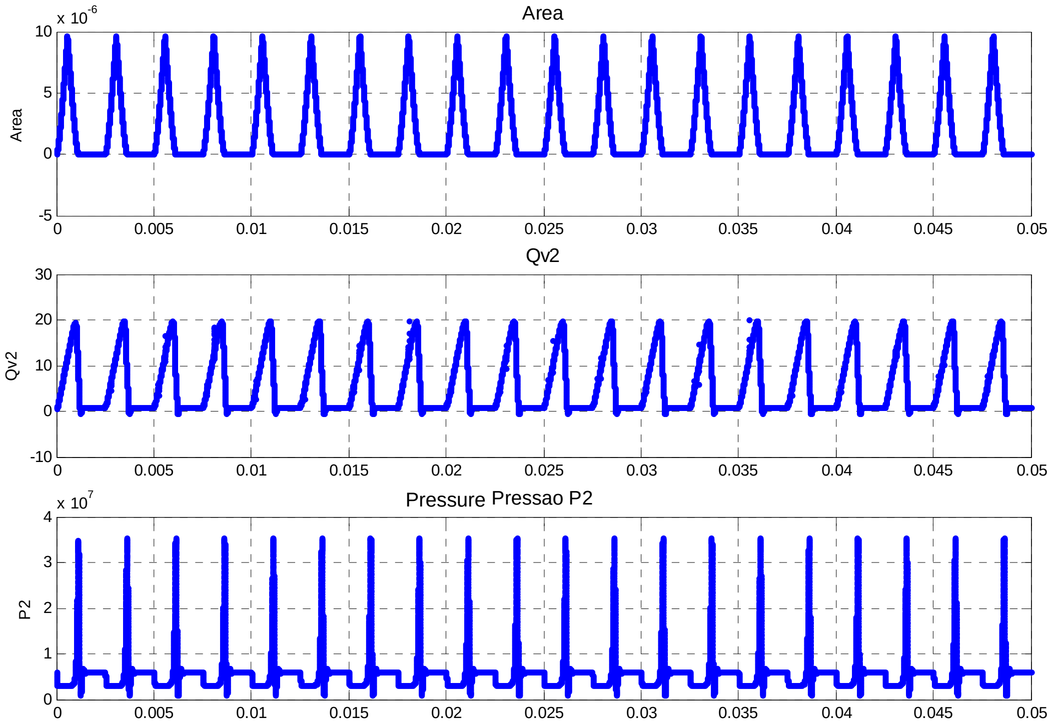


Figure 15. Simulation: 4 axial orifices in the rotor, 4 axial orifices in the body, 20 mm diameter rotor, 100 Hz frequency rotor.

As Fig. 16 exemplifies, simulations were performed for a wide range of parameters of interest, and it was concluded that the maximum flow rate is given by lower frequencies. On the other hand, the flow rate approaches zero at higher frequencies.

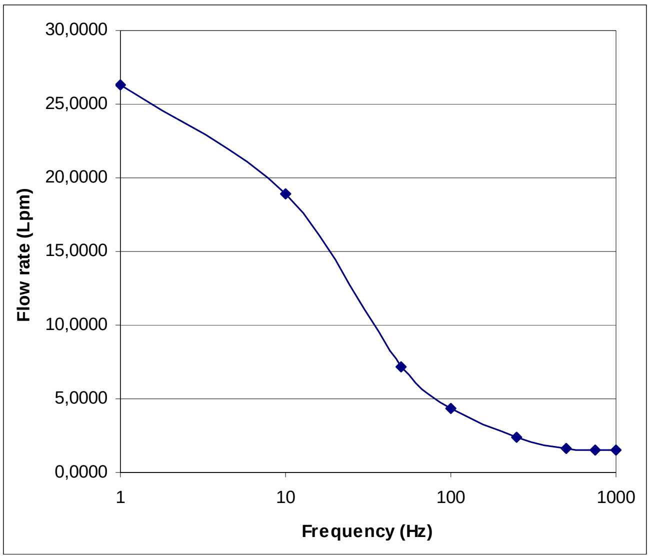


Figure 16. Decreasing outlet flow rate with increasing frequency of the rotor.

3.3. Inertial Rotating Valve Tests

Fig. 17 illustrates the results obtained from the tests performed on Inertial Rotating Valve. The general agreement between experimental data and simulations confirms the flow rate decreasement with increasing of frequency of the rotor.

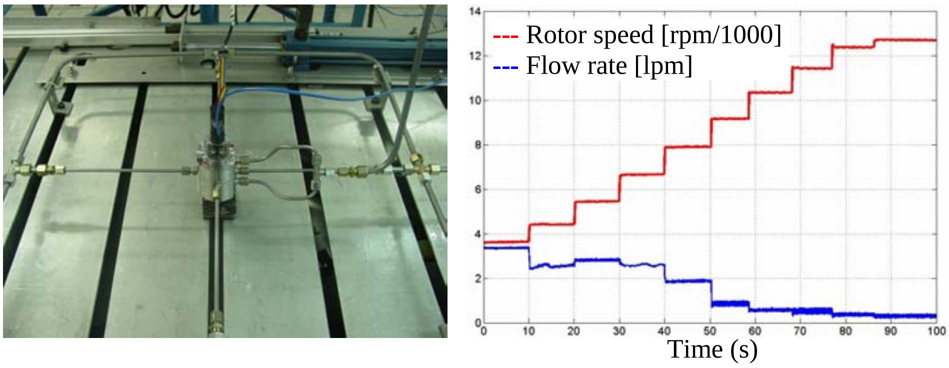


Figure 17. Workbench and experimental data of Inertial Rotating Valve tests.

4. CONCLUSIONS

Most part of innovations is intended to improve behavioral characteristics of classic sliding-spool valves. Thereby, their nonlinearities are reduced, but not eliminated. One way to overcome such problem is to develop a concept that does not demand the positioning of movable parts. With this in mind, the innovative concept presented in this paper was conceived with the specific purpose of validating the applicability of the inertial effect of the fluid to flow control.

After the manufacture of the IRV prototype, tests were carried out on a workbench directly interfaced with a data acquisition and control system. The results of the tests indicated the operation and feasibility of application of the valve.

The most important inference is that the IRV shows potential for use in hydraulic systems. However, pressure disturbances generated by the IRV must be dampened. In view of this, project changes need to be made. Also, a more realistic modeling of the IRV fluid flow is needed. With regard to the project and modeling improvements required, we intend to use ANSYS CFX computational fluid dynamics (CFD) software in the future.

In a goal-driven approach using this software, we will be able to study, quantify, and plot various performance simulation responses as a function of the design parameters for IRV parts, as well as assemblies. Also, we should able to evaluate and refine designs quickly and accurately.

Overall, we predict that IRV, in the not too distant future, will become an attractive flow control system, alongside the servo and proportional valves currently on the market. In order to achieve this objective, we already bought an ANSYS CFX license.

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