MODELLING AND ANALYSIS OF A DIRECTIONAL CARTRIDGE VALVE

Rafael Rivelino da Silva Bravo
Centro Federal de Educação Tecnológica de Campos
Industrial Center – Automation Department
Rua Doutor Siqueira, 273. Parque Dom Bosco. CEP 28030-130 – Campos dos Goytacazes – R.J.
Phone: 55 22 2733 3255 – Fax: 55 22 2733 3079 – e-mail: rbravo@cefetcampos.br

Victor Juliano De Negri
Universidade Federal de Santa Catarina
Technological Center – Mechanical Engineering Department
Laboratory of Hydraulic and Pneumatic Systems – LASHIP
Campus Universitário – Cx. P. 476 – CEP 88040-900 – Florianópolis – S.C. – Brazil
Phone: 55 48 3319396 / 3317714 – Fax: 55 48 3317615 – e-mail: victor@emc.ufsc.br
Home Page: http://www.laship.ufsc.br

Abstract: This work presents the study of a prototype of directional cartridge valve. The cartridge valves are widely used in hydraulic systems accomplishing the most varied functions according to the chosen configuration. The prototype was supplied by the company BOSCH REXROTH GROUP, which maintains partnership with the Laboratory of Hydraulic and Pneumatic Systems of the Mechanical Engineering Department of UFSC. The non-linear modeling of the component is done, and the results are obtained through the computational simulation using the software MATLAB. Through a test bench, the static and dynamic models were validated, making possible to obtain the coefficients and to understand the behavior of the component. The analysis of the results makes possible the identification and determination of the influence of the main parameters, as well as it indicates the actions that need to be done for the design improvement.

***Keywords**. cartridge valves, hydraulic systems, hydraulic valves.*

1. INTRODUCTION

In the early of 70's, the technology of the cartridge valves began to be developed at the European west and until now many researches correlated to this area. The cartridge valves can be constructively classified as slip-in and screw-in. The slip-in valves are used in high power hydraulic systems operating with flow rates among 200 to 7000 lpm. In this category, the called logical elements are included. On the other hand, the screw-in valves are usually available for limited ranges of flow rate and pressure, operating with flow rates from 2 to 140 lpm, and with continuous pressure varying around 350 bar or intermittent pressures oscillating about the 420 bar.

In this paper the study of a directional screw-in valve is accomplished. The mathematical modeling of the component is done using fundamental equations of the classic mechanics and of the fluid mechanics, seeking to accomplish the physical description and the behavioral analysis of the component. To complete the work, the prototype is analyzed experimentally aiming to determine gains and empiric coefficients of the equations and also to make comparisons among the theoretical and experimental results. The validation of the theoretical model makes possible to use it to analyze others components with similar characteristics.

2. VALVE DESCRIPTION

Under the concept of directional valves are included the valves that command the starting, stopping and changing of the direction of movement of cylinders and motors. In this paper will be studied a seat directional valve, with direct actuation by solenoid and return of the piston accomplished by spring action. The valve operates in only two positions – open and closed – and the flow rate can occurs in both directions (Fig. 1).

The operation principle can be explained as following: when the solenoid is energized, a force is generated, moving the nucleus and consequently the piston of the valve, unblocking the passage of A for B or of B for A, according to the design of the circuit. When the solenoid is de-energized, the resulting force (the spring force added by the flow force and the force due to the static pressure) acts on the piston blocking the passage A–B of the valve, characterizing this version as normally closed (NC). This model is especially used in applications where a closing without leaks in both flow directions is required. The lateral force due to the static pressure in the port B does not cause influence to the opening or closing of the valve, considering the annulling of the radial forces due to the symmetry of the holes in the sleeve of the valve (Fig. 1b).

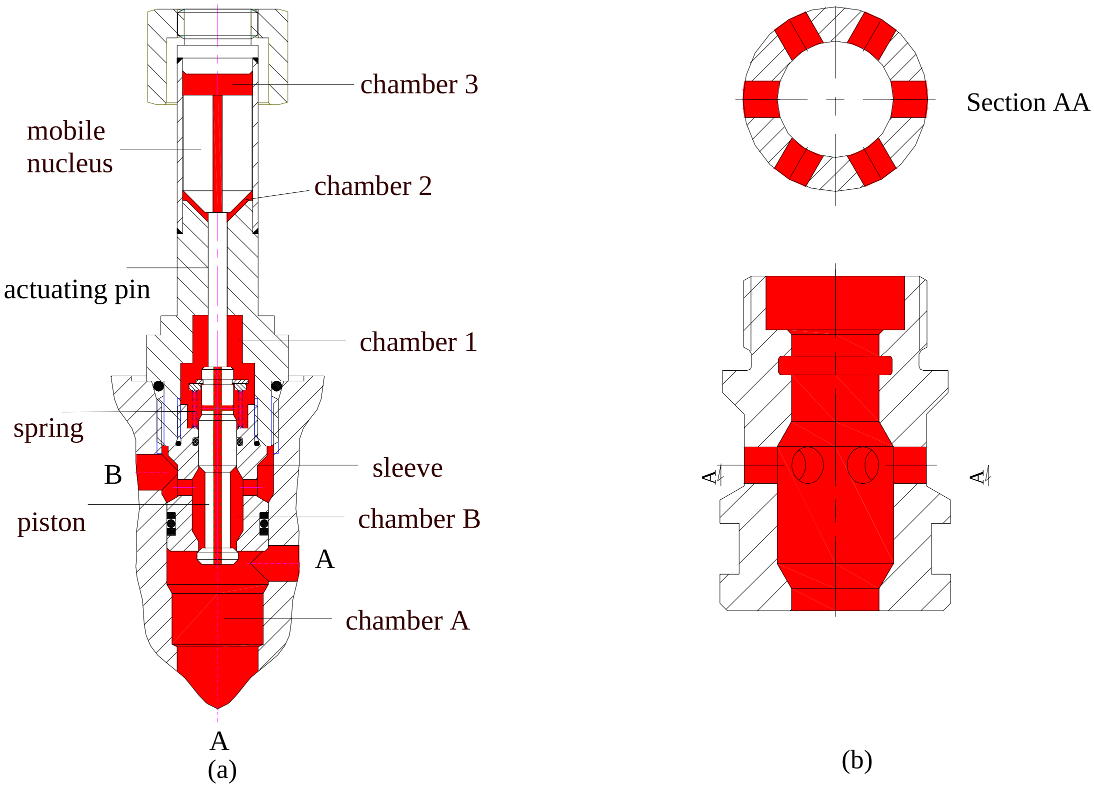


Figure 1: a) Scheme of the cartridge valve; b) Holes symmetrically distributed on the valve sleeve

It can be observed in Figures 1(a) and 2(a) that the piston has a concentric channel, allowing the fluid passage for the internal chambers of the valve and providing the lubrication of the mobile elements, including the solenoid. The solenoid used is characterized by having the actuation pin and the nucleus immersed in the hydraulic fluid that it circulates through the valve, being the coil and the fixed nucleus isolated of the fluid. With this conception, a continuous flow is allowed around the nucleus, improving the dissipation of the heat generated in the coil. The actuation pin transmits the force between the solenoid and the piston, allowing to open the valve as well as to push the nucleus for the initial position after the solenoid de-energizing. The pressure compensation is accomplished through the channel that allows the passage of the fluid for the superior area of the piston, allowing a reduction of the forces related to the actuating pressure, after the opening of the valve.

3. MATHEMATICAL MODELING

The modeling presented below is based on the non-linear equations that govern the electro-mechanical behavior of the valve, including the physical characteristics of the component. Specifically, are applied Continuity equation, Momentum equation, and Flow Rate equation in orifices (De Negri, 2001 and von Linsingen, 2001), as well as the measurement of several parameters, such as mass of the pieces, spring stiffness, time constant of the solenoid, and dimensions of areas and orifices.

Aiming to analyze the dynamic behavior of the component, it was considered as variables the displacement and speed of the piston, the flow rates, and the pressures in the internal chambers of the valve. Besides these one, it was considered the mass of the mobile parts, the input flow rate, the friction coefficients, the areas of the orifices, the flow force, the spring force, and the effect of the backstop. These factors were included as parameters in the system, and some of them can vary or not on the time according to predetermined conditions. The Figure 2 presents the valve in study, being the displacement of the piston considered positive in the direction to open the control orifice.

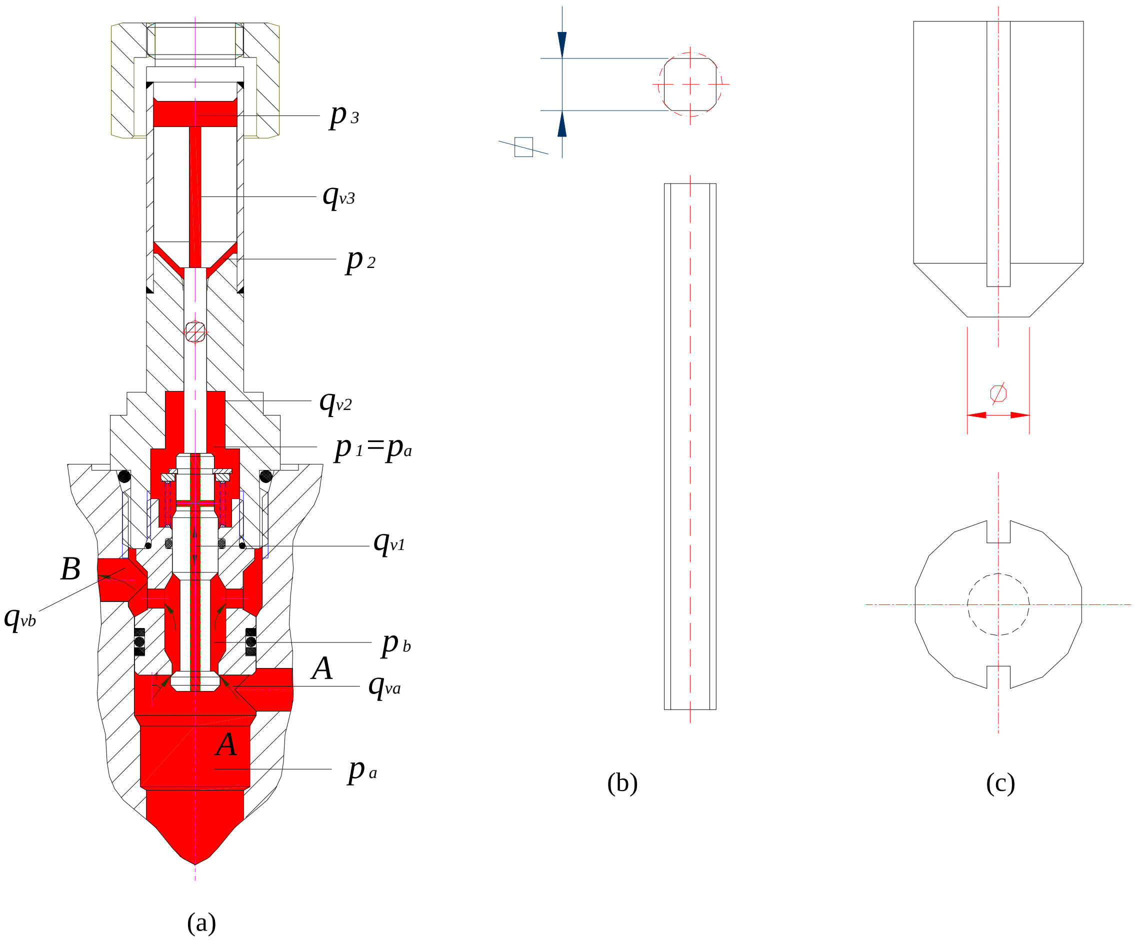


Figure 2: a) Variables related to the cartridge valve; b) Actuating pin; c) Mobile nucleus.

3.1. Chamber A

$$q_{vs} - q_{va} - q_{v1} = \frac{V_a}{\beta_e}\frac{dp_a}{dt} \tag{1}$$

$$q_{va} = C_{da}.A_A.\sqrt{\frac{2}{\rho}(p_a - p_b)} \tag{2}$$

Where:
q_{vs} = supply flow rate [m³/s]
q_{v1} = flow rate to the chamber 1 [m³/s]
V_a = volume of the chamber A [m³]
β_e = effective bulk modulus of the fluid [Pa]
q_{va} = flow rate throughout the control orifice [m³/s]
p_a = supply pressure at the port A [Pa]
p_b = pressure at the chamber B [Pa]
C_{da} = discharge coefficient related to control orifice [1]
A_A = area of the control orifice [m²]

3.2. Chamber B

$$q_{va} - q_{vb} = \frac{V_b}{\beta_e}\frac{dp_b}{dt} \tag{3}$$

$$q_{vb} = C_{db}.A_{BT}\sqrt{\frac{2.p_b}{\rho}} \tag{4}$$

Where:
V_b = volume of the chamber B [m³]
q_{vb} = flow rate at the port B [m³/ s]
A_{BT} = area of the restriction at the output B [m]
C_{db} = discharge coefficient related to restriction [1]

The area A_{BT} represents a variable restriction installed in the output pipe aiming the adjustment of the pressure in the chamber B and to facilitate the comparison among the theoretical and experimental results.

3.3. Internal chambers of the valve

The chambers 2 and 3 are located below and above the movable nucleus of the solenoid, respectively. During the solenoid energizing, the nucleus advances compressing the fluid in chamber 2 and expanding the fluid in chamber 3. When the solenoid is de-energized, the inverse fact can be observed. Due to the movement of the pieces and to the differential pressure, the hydraulic fluid flows among the chambers through two channels with square geometry (Fig. 2c) located in the periphery of the movable nucleus. Moreover, the oil feeds the chamber 1 through the concentric channel that crosses the piston. Subsequently, the fluid surrounds the actuating pin to arrive to the chamber 2. The difference between the geometry of the pin (Fig. 2b) and the circular geometry of the internal channel of the valve through where the pin slides allows the flow for the movable nucleus of the solenoid, creating conditions similar to the laminar flow in a annular section. The passages between the chambers produce an effect as a viscous friction affecting, consequently, the time response.

The continuity equation for chambers 2 and 3 can be presented as:

$$q_{v2} - q_{v3} = \frac{dV_2}{dt} + \frac{V_2}{\beta_e}.\frac{dp_2}{dt} = -A_2.\frac{dx_v}{dt} + \frac{V_2}{\beta_e}.\frac{dp_2}{dt} \tag{5}$$

$$q_{v3} = \frac{dV_3}{dt} + \frac{V_3}{\beta_e}.\frac{dp_3}{dt} = A_3.\frac{dx_v}{dt} + \frac{V_3}{\beta_e}.\frac{dp_3}{dt} \tag{6}$$

Where:
q_{v2} = flow rate to the chamber 2 [m³/ s]
p_2 = pressure on chamber 2 [Pa]
V_2 = volume of the chamber 2 [m³]
q_{v3} = flow rate to the chamber 3 [m³/ s]
p_3 = pressure on the chamber 3 [Pa]
V_3 = volume on chamber 3 [m³]

The flow rate to the internal chambers can be estimated through the analysis of the laminar flow across an annular clearance for the chamber 2, and across a square section for the chamber 3, as mentioned previously. When the clearance f_r is very smaller than the hydraulic diameter D_h of the sleeve, and equally very smaller than the length L of the piston, it can admit that the behavior of the flow is similar to that happens among plane plates (von Linsingen, 2001). Therefore, the equation of the flow rate to the chambers 2 and 3, can be written as:

$$q_{v2} = \frac{\pi . D_p . f_{r2}^{\ 3}}{12 . \mu . L_p} (p_a - p_2) - \frac{\pi . D_p . f_{r2}}{2} \frac{dx_v}{dt} \mathrm{sgn}(\dot{x}_v) \tag{7}$$

$$q_{v3} = 2 . \frac{w_3^{\ 4}}{28{,}4 . \mu . L_z} (p_2 - p_3) - \frac{\pi . D_z . f_{r3}}{2} \frac{dx_v}{dt} \mathrm{sgn}(\dot{x}_v) \tag{8}$$

Where:
D_p = medium diameter of the actuating pin [m]
f_{r2} = radial clearance between actuating pin and the valve body [m]
L_p = pin length [m]
μ = absolute viscosity [N.s/m²]
w_3 = width of the square section of the flow channel [m]
L_z = length of passage to the chamber 3 [m]
D_z = diameter of the mobile nucleus [m]
C_{v3} = leakage coefficient [Pa.m³/ s]
f_{r3} = radial clearance between mobile nucleus and fixed nucleus [m]

3.4. Modeling of the electromagnetic actuator

The simplest model of a solenoid is a resistor in series with a linear inductor. The relationships between tension and current are obtained applying the Kirchoff's law to the electric tension.

$$V_m = L_b . \frac{dI_m}{dt} + R_b . I_m \tag{9}$$

$$F_m = K_f . I_m \tag{10}$$

Where:
F_m = total force available in the mobile nucleus [N]
K_f = gain of the solenoid [N/A]
V_m = tension on the coil [V]
L_b = inductance of the coil [H]
I_m = electric current [A]
R_b = resistance of the coil [Ω]

3.5. Movement equation of the Valve:

$$M_v.\frac{dx_v^{\;2}}{d^2t}+B_v.\frac{dx_v}{dt}+K_v(x_v+x_{vi})=F_m+F_R-p_a.A_e-p_2.A_2+p_3.A_3-F_f+F_{BT} \qquad (11)$$

where:
$M_v = M_e + M_n$ [kg]
M_v = total mass to be displaced by the solenoid force [kg]
M_e = mass of the piston {kg]
M_n = mass of the mobile nucleus [kg]
B_v = damping coefficient [N.s/m]
x_v = linear displacement of the piston [m]
x_{vi} = pre-load of the spring [m]
K_v = spring stiffness [N/m]
F_R = resultant flow force [N]
A_e = actuating area of the pressure in A [m²]
A_2 = area of the chamber 2 [m²]
A_3 = area of the chamber 3 [m²]
F_f = friction force [N]
F_{BT} = backstop force [N]

3.6. Flow Forces

Based on the application of the balance of momentum to the fluid mass present in the chamber B, it is observed that the flow of fluid inside the valve provokes forces on the mobile piston and that, therefore, it should be overcome by the solenoid (Blackburn et al, 1960; von Linsingen, 2001).

In a general way, the sum vector of the forces actuating on the fluid in a control volume it is constituted by the surface force, including the pressure forces that act in the input and output sections, and by the reaction force that the wall apply on the fluid. The resulting force is that the fluid applies on the wall, being same and of contrary sign by force of reaction of the wall.

The sum vector of the external forces provokes the variation of the momentum, which can be expressed by the steady state and transient flow forces, that is:

$$\sum\vec{F}=\vec{F}_S+\vec{F}_P=\vec{F}_S-\vec{F}_R=\vec{F}_{esc\,p}+\vec{F}_{esc\,t} \qquad (12)$$

Where:
F_S = surface force [N]
F_P = reaction force [N]
F_R = resulting force [N]
F_{escp} = steady state flow forces [N]
F_{esct} = transient flow forces [N]

The surface force on the port B has not axial component, being neglected. On the other hand, the surface force on the port A acts on the control orifice and depends of its specific conditions. Therefore, considering the high speed of the fluid on the orifice, the static pressure is locally reduced and, additionally, the flow section is very small. Consequently, the surface force on the Eq. (12) can be considered neglecter.

Being considered the control volume shown in Fig. (3a), the components of the flow force can be written accordingly below (Bravo, 2002).

a) Steady state flow force F_{escp}

The equation that depict the steady state flow force, can be expressed as:

$$F_{esc\,p} = 2.Cd_a.A_A.(p_a - p_b).\cos\varphi_a \tag{13}$$

b) Transient flow force F_{esct}

The transient flow force appears due to the acceleration of the mass of the fluid in the control volume. To provoke this acceleration, the pressure in the beginning of the chamber needs to be higher than the pressure in the opposite extremity, meaning that there is temporary variation between the input and output pressures in the chamber A. The transient flow force can be estimated by:

$$F_{esct} = -Cd_a.K_{ga}.L_a.\mathrm{sgn}(q_{va})\left(\sqrt{2.\rho.(p_a - p_b)}.\frac{dx_v}{dt} + x_v.\sqrt{\frac{\rho}{2.\Delta p}}.\frac{d(p_a - p_b)}{dt}\right) \tag{14}$$

Where:

φ_a = angle of the fluid jet [degree]

L_a = length of the chamber B [m]

$K_{ga} = \frac{\partial A_A}{\partial x_v}$ = geometric gain of the control orifice [m]

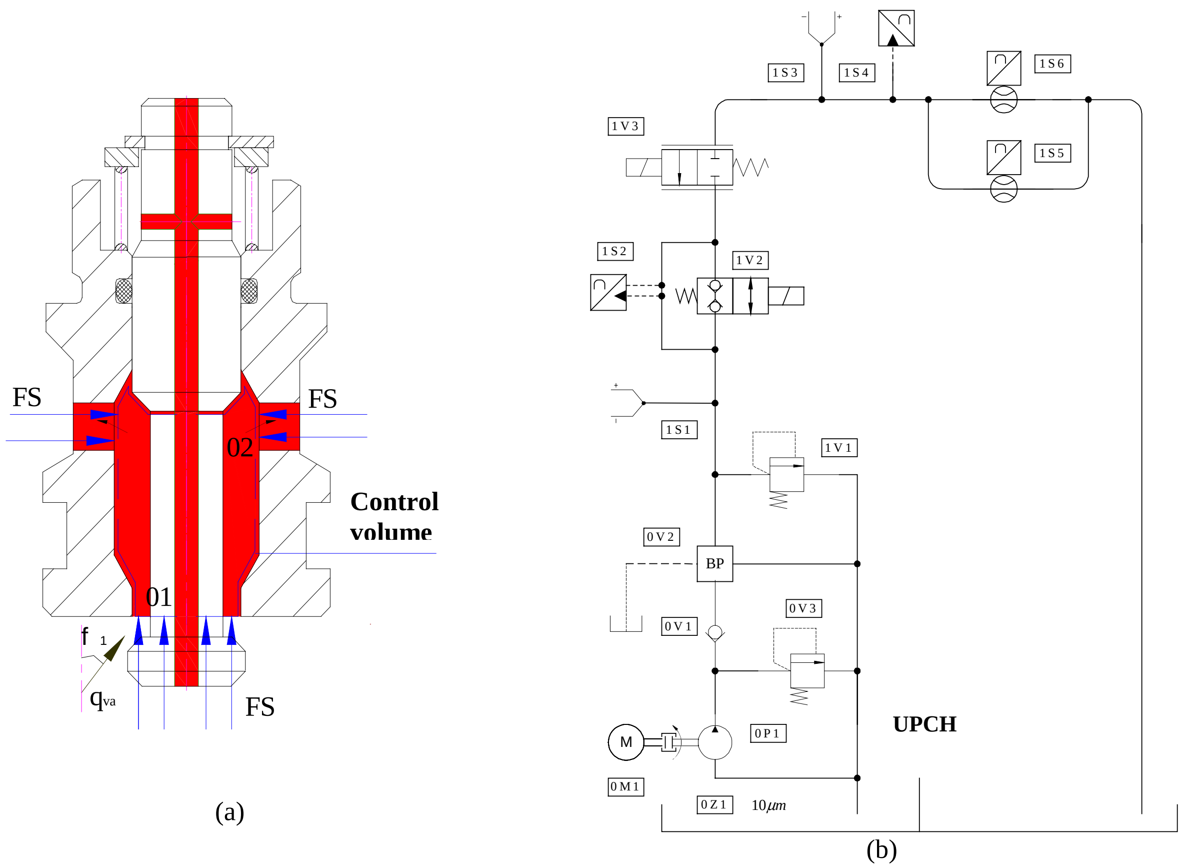


Figure 3. (a) Control volume, (b) hydraulic circuit used in the experiment

4. TESTS DESCRIPTION

The hydraulic circuit for test of the directional valve (1V2) it is presented in Fig. (3b) and it follows the recommendations from the International Standards ISO 4411 (ISO, 1986) and 6403 (ISO, 1989). The settling of the flow rate and supply pressure is accomplished through a manifold containing proportional valves (BP) 0V2 installed in UPCH (Power and Conditioning Hydraulic Unit). During the tests, the pressure of the system is controlled through the valves 1V1 and 1V3, being the flow rate measured through the transducers 1S5 and 1S6 and the differential pressure through the transducer 1S2. The hydraulic fluid is a mineral oil ISO VG 32, being the temperature maintained in 40 ± 2°C. A data acquisition and control system was used to obtain the dynamic characteristics. The displacement of the piston was accomplished through a micrometrical table and the measure of the displacement force through a load cell. As described in Bravo (2002), some adaptations in the body of the valve were made seeking to improve the measurement conditions.

5. RESULTS ANALYSIS

5.1. Steady State

The equation of the flow rate (Eq. 2) was used to obtain the discharge coefficient C_{da}. During the test, the differential pressure across the valve was maintained constant and the valve was opened gradually. It is observed through the Fig. (4a.) that happened an abrupt inversion in the behavior of the discharge coefficient where a value reduction was followed by a sudden increasing of the discharge coefficient. This behavior is function of the theoretical values of the area of the control orifice A_A that reaches its maximum value in a specific position. From this condition, the flow area does not change with the piston displacement. However, the value of the discharge coefficient grows in consequence of the continuous increasing of the flow rate, which is generated by the change in the behavior of the flow in the input region of the valve.

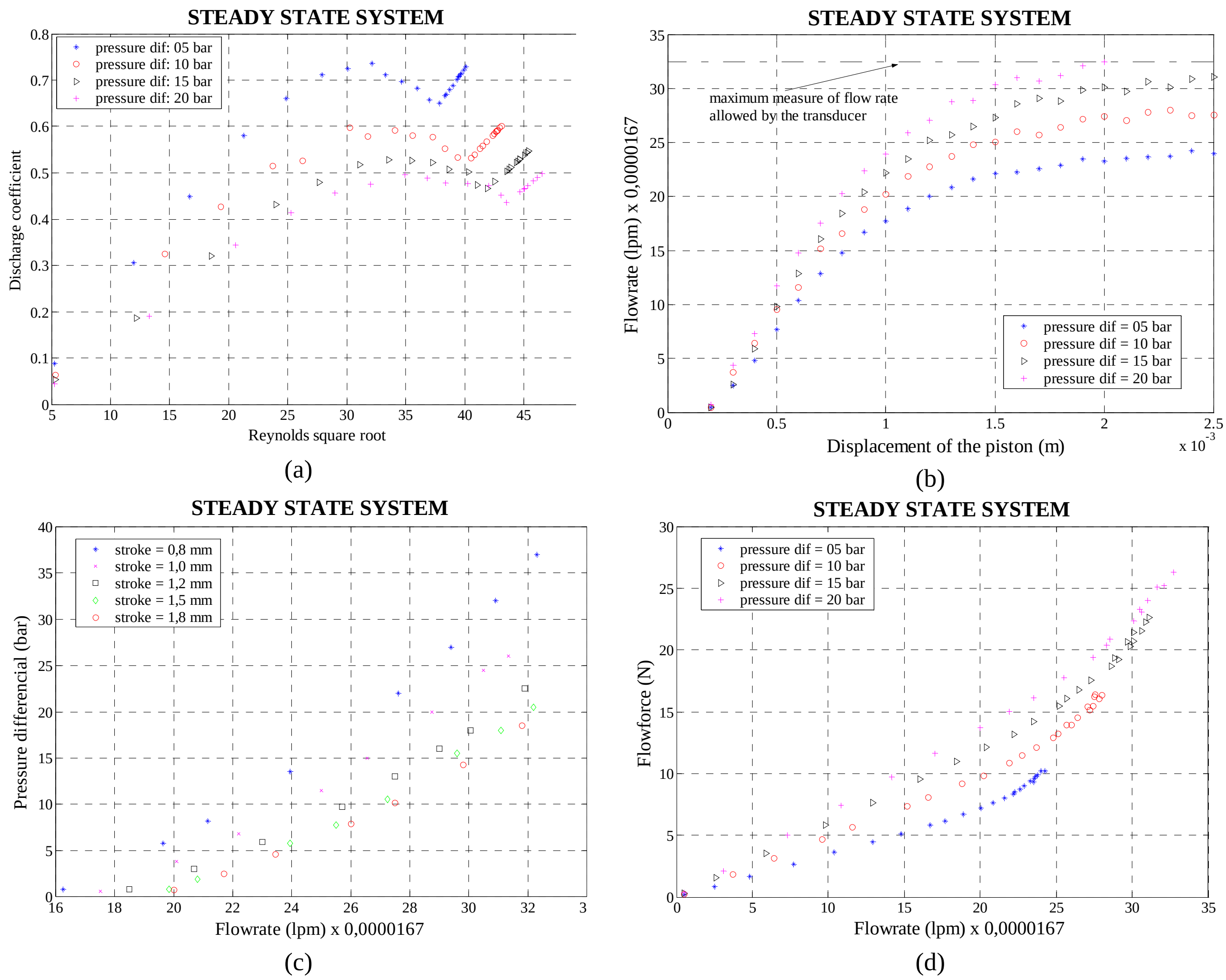


Figure 4. Static response of the valve: (a) discharge coefficient, (b) flow rate × piston displacement, (c) flow rate × pressure drop, (d) flow force ×x flow rate.

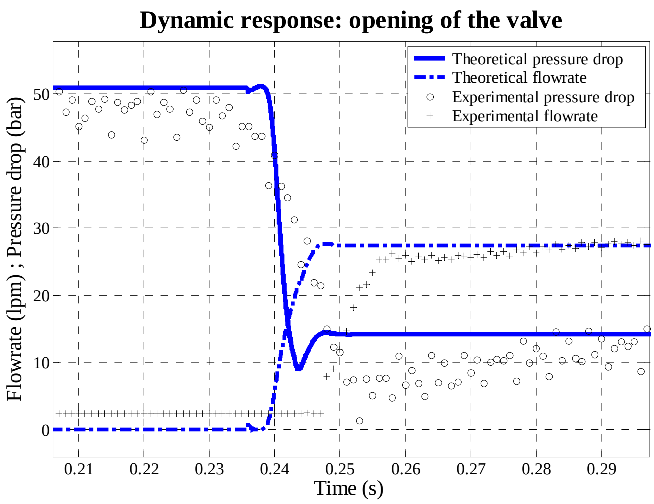
The graphic of flow rate versus piston displacement (Fig. 4b.) aids to choice of the opening stroke. From a displacement of approximately 1,37 mm there is not more changing in the area of the control orifice. For displacements above 2,0 mm the flow rate is almost constant under a specific differential pressure.

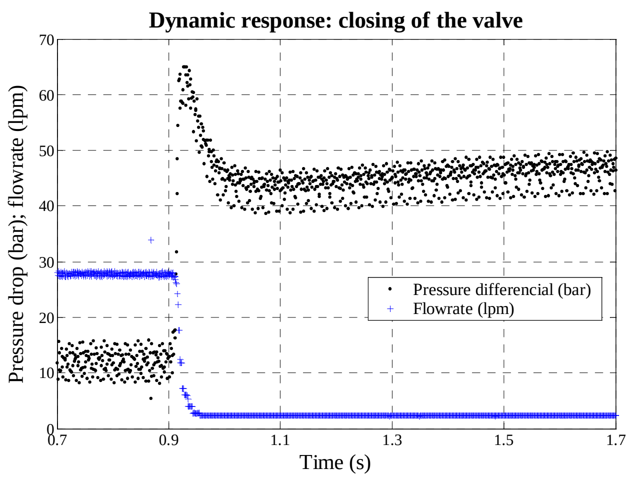
A typical graph of product catalogs is shown in Fig. (4c.) and specifies the relationship between the flow rate and the differential pressure on the component. Through this information, the user can select the better valve for the planned work condition. For the obtaining of the curve, the tests were accomplished commanding the valve through the solenoid and changing the length of the actuating pin, allowing the variation in the stroke of opening x_v of the valve. As expected, for largest strokes the valve presented smaller pressure losses. Can be observed that for low flow rates, for instance below 16 lpm and displacement of 0,8 mm, it was not possible to measure of the differential pressure in the valve. For this range, the measurement by the differential pressure transducer was not reliable because the noise presence. Figure (4d.) presents the flow force versus the flow rate for different values of pressure drops on the valve. The opening of the valve was gradually increased with displacements from 0,2 to 2,5 mm.

5.2. Dynamic Response

Figure (5a.) shows the flow rate and pressure drop curves for transient conditions during the opening of the valve. The pressure drop in the experiment is measured through the differential pressure transducer 1S2 (Fig. 3b). The dynamic response is observed when the solenoid is energized through an on/off switch. For accomplishment of the dynamic tests, the closing spring had to be removed because the insufficient force of the solenoid. Consequently, the valve closing resulted of the interaction between the static pressure and flow force. Comparing the results, it is observed that the theoretical model presented a faster response than the experimental test. During the test, the valve needed 18 to 19 ms to reach 91% of the total value of the flow rate. However, it was necessary 140 ms to achieve the steady state. The theoretical model presented a time response of 13 ms to reach the maximum value of the flow rate and 19 ms to reach the steady state.

Among some reasons that could cause the difference between the results, the most probable is the insufficient force of the solenoid. Besides, the magnetic characteristics of the solenoid were not modeled, and the inductance was treated as a linear parameter. A delay at the reading from the flow rate transducer could be provoked by the limited dynamic response of the instrument. It can explain the died time in the flow rate response related to the pressure response during the experiment, once the pressure measured already presents a reduction from the instant 0,235s, indicating that is happening the opening of the valve.





(a.) (b.)

Figure 5. (a) Dynamic response: (a.) opening and (b.) closing of the valve.

The Figure (5b.) shows the time response for the closing of the valve during the experiment. As the valve was operating without the spring, the closing at the moment of the de-energizing of the solenoid was caused essentially by the steady state flow force described in Eq. (13) and in Fig. (4c.). The spending time to close the valve was of the order of 35 ms.

Comparing the obtained dynamic results, it is possible to conclude that:

- The solenoid is inadequate, once it does not supply enough force to move the valve correctly. Consequently, the time of real opening is higher than the expected;
- The dynamic response of the valve is very high. However it was only verified when the closing spring was removed. The changing of the solenoid should correct the delays noticed in normal conditions of operation.
- Concerned to the comparison among the theoretical and experimental results, it is observed that the further substitution of the solenoid will give better results than the ones already reached in the opening of the valve. Beyond, it is observed that the format of both theoretical and experimental curves of flow rate and pressure are very similar, showing that the model is a good representation of the physical phenomena.

6. CONCLUSION

The study of mathematical models for hydraulic components design is essential for the knowledge of its static and dynamic behaviors. Through the interrelationship among results from simulations and experiments, it is possible to reduce both time and cost during the component development. As observed along this work, the parameters were obtained more precisely and easily compounding theoretical and practical activities.

For the modeling, fundamental equations of both classic mechanics and fluid mechanics were applied. The non-linearity’s inclusion in the proposed models, such as, geometry of the control orifice, pre-load of the spring, friction and the effect of the backstop, allowed more effective physical descriptions by the theoretical model.

The experimental activity, necessary to validation of the model of the directional valve, made possible to obtain important parameters. This one, being used in the theoretical model, allowed quite similar results when compared with the tests. However, some divergences on the results were found, suggesting modifications on the prototype to allow its correct operation. In short, the detailing level adopted in the modeling was suitable for the description of the behavior of the valve and for the analysis of the variables and involved parameters.

7. REFERENCES

Blackburn, J. F.; Reethof, G.; Shearer, J. L., 1960, "Fluid Power Control", MIT Press, 710 p.

Bravo, R. R. S., 2002, "Modelagem e Análise de Válvulas Cartucho", Universidade Federal de Santa Catarina, Florianópolis, 131 p. (Master Thesis)

De Negri, V.J., 2001, "Sistemas Hidráulicos e Pneumáticos para Controle e Automação: Partes I, II e III", Universidade Federal de Santa Catarina, Florianópolis, 158p.

Ge, S. S., Lee, T. H., Ren, S. X., 1999, "Adaptive Friction Compensation of Servo Mechanism". In: International Conference on Control Applications. Hawai, USA, p. 1175-1180.

ISO, 1986, "ISO 4411 - Hydraulic fluid power - Valves - Determination of pressure differential/flow characteristics', Switzerland, 11 p.

ISO, 1989, "ISO 6403 - Hydraulic fluid power - Valves controlling flow and pressure - Test methods", 1. ed. rev., Switzerland, 29 p.

Mccloy, D. Martin, H. R., 1973, "The control of fluid power", London: Longman Group Limited, 1973, 367p.

Merrit, H. E. "Hydraulic Control System", John Wiley & Sons, Inc, New York, 1967.

von Linsingen, I. "Fundamentos de Sistemas Hidráulicos". Florianópolis: Ed. da UFSC, 2001, 399p.