FUNCTIONAL DIAGRAM FOR COMPARATIVE ANALYSIS OF ELECTROMAGNETIC BALL VALVES

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ABSTRACT: The main objective of this paper is to perform a comparative numerical analysis of an electromagnetic normally-closed direct-acting ball valve with two types of seats: cylindrical and conical. A functional diagram is developed using the MATLAB/Simscape/SimHydraulics programming language. The principal elements of the functional diagram are: the fixed-displacement hydraulic pump, the ball valve which is driven by an electromagnetic actuator, the pressure-relief valve, pressure and volumetric flow rate measuring devices and the hydraulic pipes. Two sets of analyses are performed; the first one will study the influence of the effects that are associated with the hydraulic pipes on the ball valve with conical seat. The second set of analyses will evaluate the influence of the seat (cylindrical and conical) on the principal functional characteristics of the ball valve: the valve flow factor, the volumetric flow rate through the ball valve and through the pressure relief valve and the valve pressure drop.
KEY WORDS: electromagnetic ball valve, functional diagram, MATLAB, Simscape, SimHydraulics.

1 INTRODUCTION

The electromagnetic valves are wide used as control devices by switching on or off the fluid flow in hydraulic systems. There are many classification criterions of electromagnetic valves, such as: the variable orifice geometry (ball with cylindrical seat, ball with conical seat, needle and poppet); the valve type according to the de-energized state (normally-closed valve NC, normally-open valve NO); the acting type (direct-acting valve, internally piloted valve). Recent developments are focused on optimization models of the valves (Chen, 2012), as well as on the cavitations flow through the valve (Valdes et al., 2014). All the relevant electromagnetic ball valve specific data (including the valve flow factor, K_v) can be found on the product data sheet. However, there is a strong correlation between the electromagnetic actuator characteristics, the ball valve parameters and the global parameters of the hydraulic system. In (Zahariea, 2014), there was found that the hydraulic system is related with a hysteresis phenomenon obtained on the valve flow factor analysis. In this paper, a functional diagram for modeling an electromagnetic ball valve with both cylindrical and conical seat will be presented. The simulation will be performed with and without the hydraulic system effects, in order to observe the influence of these effects on the functional characteristics of the ball valve with conical seat.

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The most important effects associated with the hydraulic circuit are: the fluid inertia, the fluid compressibility, the viscoelastic properties related with the pipe internal diameter to pressure variations, and the pressure losses caused by the linear and local resistances.

The second set of numerical simulation will have as the main objective the comparative analysis of the electromagnetic normally-closed direct-acting ball valve with two types of seats: the cylindrical seat and the conical seat.

2 MATLAB/SIMHYDRAULICS FUNCTIONAL DIAGRAM

To better understand the electromagnetic ball valve behavior, as well as to analyze the basic parameters (flow rate and pressure) of the hydraulic system which are close correlated with the valve dynamics, in (Zahariea, 2014), a functional diagram for modeling a normally-closed direct-acting ball valve with cylindrical seat has been presented. Based on this functional diagram, in Figure 1 is presented a new version: the functional diagram for simulation of the electromagnetic ball valve with conical seat.

The functional diagram has been developed using the MATLAB/Simscape/ SimHydraulics computing environment (Zahariea, 2010). Simscape/Sim Hydraulics (MathWorks, 2014) are computational toolboxes inside MATLAB programming language, which provides functional elements for modeling and simulating general physical systems (basic hydraulic, mechanical, electrical, thermal and pneumatic systems), respectively more complex and specialized hydraulic systems.

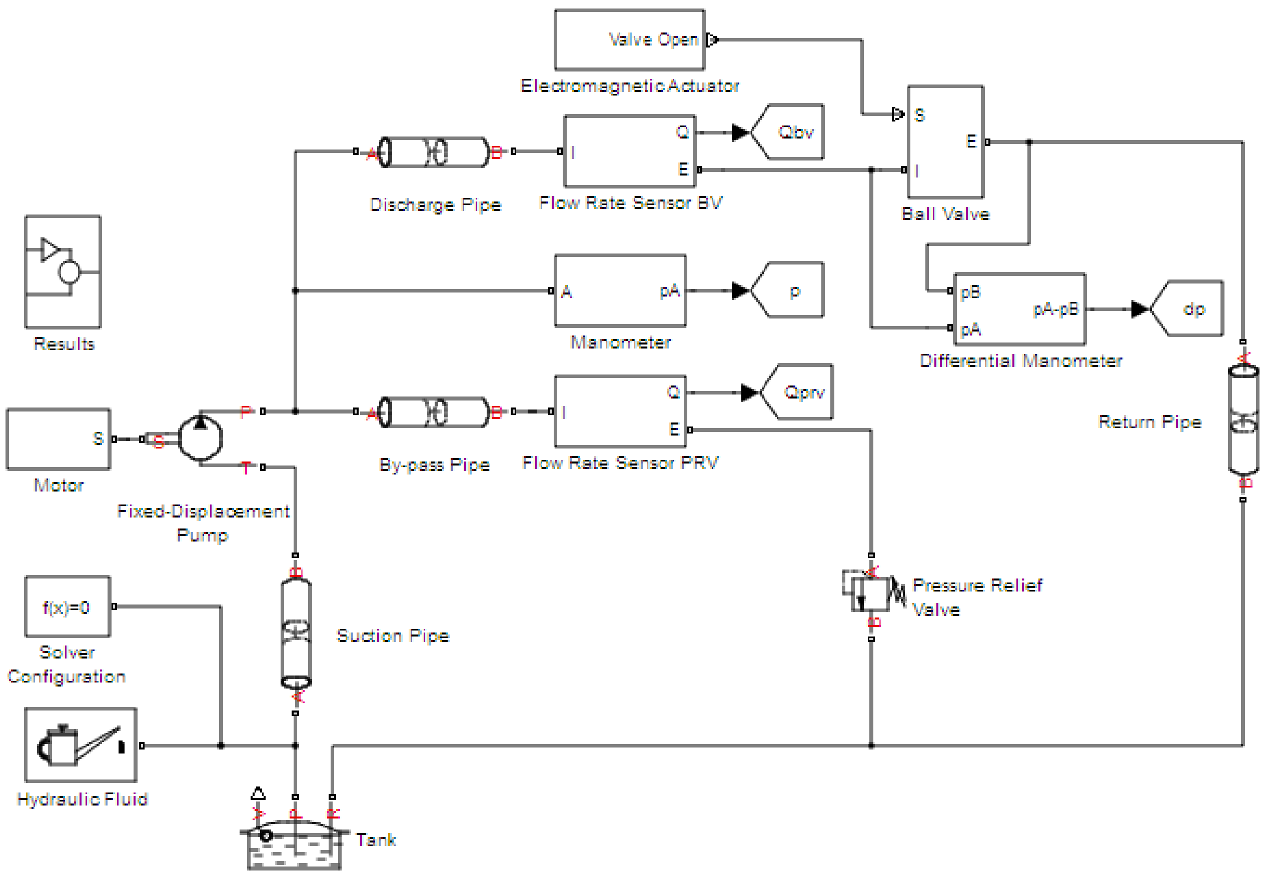
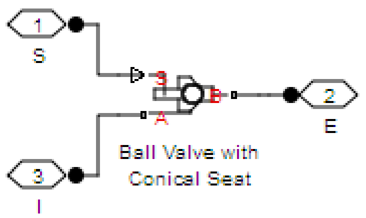
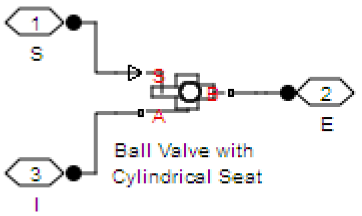


Figure 1. Functional diagram for modeling the electromagnetic ball valve with conical seat

The electromagnetic normally-closed direct-acting ball valve is simulated using the "Ball Valve" functional subsystem, which is presented in Figure 2,a for the conical seat and in Figure 2,b for the cylindrical seat. There are some common characteristic parameters of both ball valves (with cylindrical seat and conical seat): the valve ball diameter=6 mm, the initial opening=0.01 mm and the leakage area=10^{-12} m^2.



a) ball valve with conical seat



b) ball valve with cylindrical seat

Figure 2. Functional subsystem for modeling the ball valve

However, there are also some specific characteristic parameters; the orifice diameter=5 mm, the cone angle=120° and the discharge coefficient=0.75 for the ball valve with conical seat, meanwhile for the ball valve with cylindrical seat these specific parameters are the cylindrical seat diameter=5 mm and the discharge coefficient=0.65.

The ball valve opening is defined using the "Electromagnetic Actuator" subsystem presented in Figure 3. The "Pulse Generator" block will generate a DC electric signal having the amplitude of 24 V, the period of 8 s, and the pulse width of 50% of signal period. This signal applied on the "2-Position Valve Actuator", will generate the valve opening signal. The main characteristics of the actuator are: the push-pin stroke=1.8 mm (and, as a consequence, the ball valve stroke will be 1.8 mm as well), the switching-on time=1 s, the switching-off time=1 s, the nominal signal value=24 V.

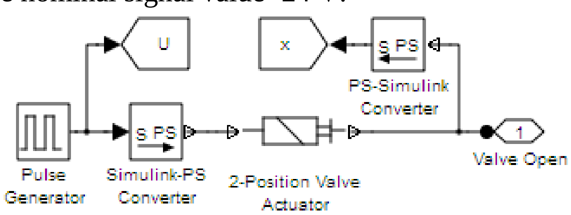


Figure 3. Functional subsystem for modeling the "Electromagnetic Actuator"

The input and the output signals of the electromagnetic actuator are presented in Figure 4. The voltage U [V] of the electric current applied to the electromagnetic actuator is a pulse signal which will generate a translational displacement x [mm] of the push-pin. There is a phase delay between these two signals, because of the switching-on and switching-off times of the actuator.

By changing the main characteristics of the actuator, different mechanical translational push-pin motion signals can be obtained. In Figure 5, the influence of the switching-on time is presented; three different values being considered: t_{on}=1 s as reference value, t_{on}=0.5 s and t_{on}=1.5 s as comparative values. Another important parameter is the switching-off time, in Figure 6 being analyzed the influence of this parameter with the same numerical values: t_{off}=0.5 s, t_{off}=1 s, and t_{off}=1,5 s.

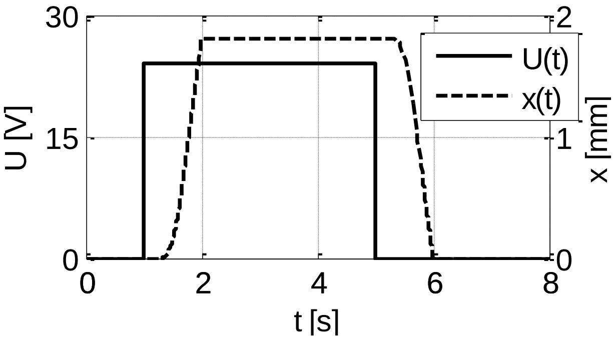


Figure 4. Input and output signals of the electromagnetic actuator

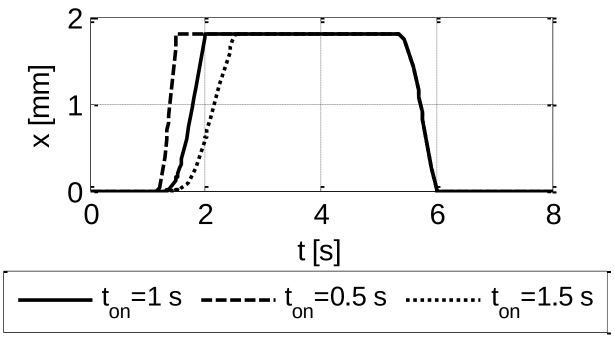


Figure 5. The influence of the switching-on time on the push-pin motion signal

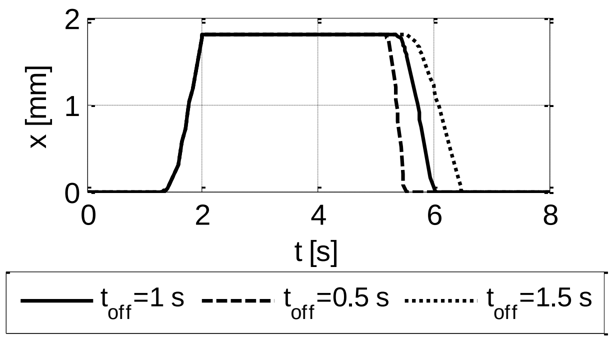


Figure 6. The influence of the switching-off time on the push-pin motion signal

The working fluid is defined with the “Hydraulic Fluid” functional element, which is set for Skydrol-LD4 at 30°C with relative amount of trapped air=0.005, density=985.312 kg/m^3, viscosity 14.0129 cSt and bulk modulus at atmospheric pressure and no gas=1.4664x10^9 Pa.

The hydraulic power unit is composed by the fixed displacement pump, the tank and the pressure relief valve. The parameters of the hydraulic pump are: the fixed displacement D=5x10^{-6} m^3/rad, the volumetric efficiency η_v=0.92, the total efficiency η_t=0.8, the nominal pressure p_n=100 bar, the nominal speed n_n=1500 rpm, the nominal kinematic viscosity ν_n=18 cSt. The inlet section of the hydraulic pump is connected with the tank through the suction pipe with the pipe diameter d_s=10 mm, the pipe length L_s=1 m, the aggregate equivalent length of local resistances L_{se}=0.1 m.

The fixed displacement pump is driven by an ideal motor that will generate constant velocity regardless of the torque exerted on the system. The functional diagram of the ideal motor is presented in Figure 7. The speed measuring unit is set to [rpm], using the “Simulink-PS Converter n”. The motor speed signal is generated using the “Signal Builder” block and the pump shaft speed is simulated using the “Ideal Angular Velocity Source” functional element.

The motor speed will start from zero and will rise to 1500 rpm at 0.3 s, having five characteristic intermediate values: 475 rpm at 0.05 s, 900 rpm at 0.1 s, 1200 rpm at 0.15 s, 1375 rpm at 0.2 s and 1450 rpm at 0.25 s (Figure 8). After 0.3 s, the speed motor will remain constant at 1500 rpm.

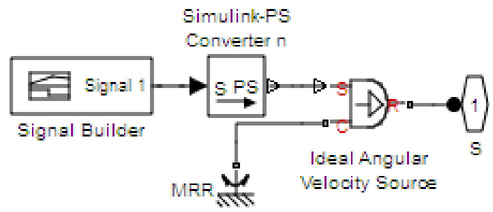


Figure 7. Functional subsystem for modeling the pump motor

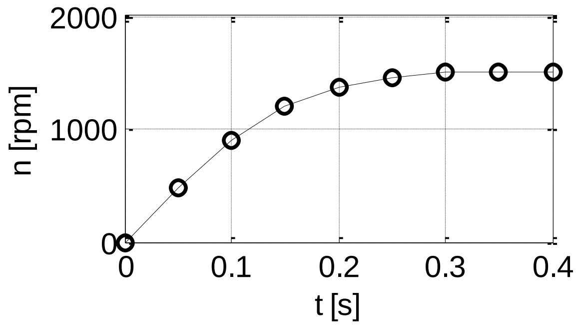


Figure 8. The speed motor curve

The pressure-relief valve will protect the hydraulic elements when the pressure exceeds the design limit allowing the fluid to be diverted through the by-pass pipe back to the tank.

The principal parameters of the pressure relief valve are: the relief valve pressure setting is 90 bar, the relief valve regulation range is 5 bar, and the maximum passage area is equivalent with a 5 mm diameter orifice. When the relief valve pressure setting is exceeded, the relief valve will open and therefore the fluid will be discharged to the tank through the by-pass pipe with the pipe diameter d_b=10 mm, the pipe length L_b=1 m, the aggregate equivalent length of local resistances L_{be}=0.1 m.

In order to measure the fluid flow rate diverted through the by-pass pipe, a fluid flow rate sensor identified by the name “Flow Rate Sensor PRV” is installed on that pipe. The functional diagram of the flow rate sensor is presented in Figure 9. The measuring process is performed by the “Hydraulic Flow Rate Sensor” functional element, which is an ideal flow meter inserted through the hydraulic conserving ports A (associated with the sensor positive probe) and B (associated with the sensor negative probe) along the hydraulic pipe. The physical signal port Q will output a signal proportional with the volumetric flow rate, which is positive if the fluid will flow from A to B, and negative if fluid will flow from B to A. The measuring unit of the fluid flow rate, which is set in this case to [m^3/h], is controlled using the functional element “PS-Simulink Converter Q”.

When the pressure inside the hydraulic system is lower than the set pressure of the relief valve, the pressure relief valve will be closed and the fluid discharged by the pump will flow through the main hydraulic circuit composed by: the discharge pipe, the ball valve and, after that, back to the tank through the return pipe. The principal parameters of the discharge pipe are: the pipe diameter d_d=10 mm, the pipe length L_d=5 m, the aggregate equivalent length of local resistances L_{de}=0.1 m. The principal parameters of the return pipe are: the pipe diameter d_r=10 mm, the pipe length L_r=5 m, the aggregate equivalent length of local resistances L_{re}=0.1 m

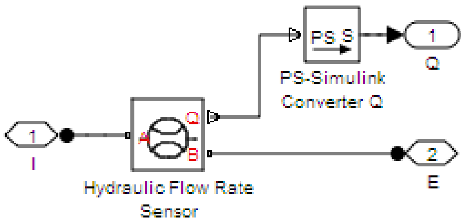


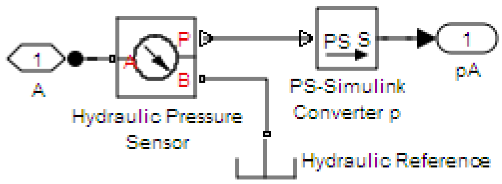
Figure 9. Functional subsystem for modeling the “Flow Rate Sensor PRV”

The second fluid flow rate sensor identified by the name “Flow Rate Sensor BV” is installed on the discharge pipe. The functional diagram of this second flow rate sensor is identical with that one presented in Figure 9, including the measuring unit which is also set to [m^3/h]. However, the output signals of these two flow rate sensors are different: Q_{bv} for the flow rate through the ball valve and Q_{prv} for the flow rate through the pressure relief valve.

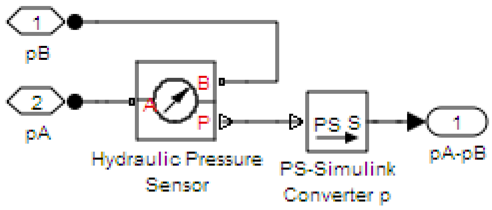
Measuring process of the discharge pressure and the pressure drop in the ball valve are performed using one manometer and one differential manometer, which are presented in Figure 10 (a, for the manometer and b, for differential manometer). Both manometers have the measuring unit set to [bar] by means of the functional element “PS-Simulink Converter p”. Moreover, both manometers will measure the hydraulic differential pressure between two connecting points: $\Delta p=p_A-p_B$ and will output a signal proportional with this differential pressure through the physical port P.

For the case presented in Figure 10,a, the input port B of the manometer is connected with the “Hydraulic Reference” functional element, which will simulate the atmospheric pressure, p_{at}. The input port A of the manometer is connected with the discharge section of the pump. Thus, for this case the manometer will measure the discharge pressure of the pump (p_d-p_{at}).

For the second case, presented in Figure 10,b, the two input ports A and B are associated with the sensor positive probe (upstream relative to the ball valve, p_{up}), respectively with the sensor negative probe (downstream relative to the ball valve, p_{down}). Thus, for this case, the manometer will measure the pressure drop in the ball valve (p_{up}-p_{down}).



a) manometer



b) differential manometer

Figure 10. Functional subsystem for modeling the pressure measurement devices

3 NUMERICAL RESULTS

Two sets of analyses are performed; the first one will study the influence of the effects that are associated with the hydraulic pipes (the fluid inertia, the fluid compressibility, the viscoelastic properties related with the pipe internal diameter to pressure variations, and the pressure losses caused by the linear and local resistances) on the ball valve with conical seat. In this case, two types of simulations are performed; the first one for an ideal case, without the effects that are associated with the hydraulic circuit and the second one for the real case, with these effects. Both these simulations will refer to a ball valve with conical seat.

The second set of analyses will evaluate the influence of the seat type (cylindrical and conical) on the principal ball valve functional characteristics: the valve flow factor, the volumetric flow rate through the ball valve and through the pressure relief valve and the valve pressure drop.

One of the most important characteristic parameters of the ball valve, no matter the type of the seat, is the flow factor, K_v:

$$K_v = Q\sqrt{\rho/(1000\Delta p)} \qquad (1)$$

where Q [m^3/h] is the volumetric flow rate, ρ [kg/m^3] is the fluid density and Δp [bar] is the pressure drop in the ball valve.

3.1 Influence of the hydraulic pipes

In order to avoid the oversizing of the ball valve, observed in (Zahariea, 2014), for the stroke of the ball valve with conical seat, a value of 1.8 mm has been used.

In Figure 11, there is presented, for the ideal case, the dimensionless valve flow factor K_v/K_{vmax} with respect to the dimensionless ball valve stroke x/x_{max}100 [%], where 0% stroke correspond to the valve full close and 100% stroke correspond to the valve full open and K_{vmax} and x_{max} are the maximum values of the valve flow factor and ball valve stroke. It can be observed that the valve flow factor reaches its maximum value at x/xmax≅100%, which leads to the conclusion that the ball valve stroke is correctly designed.

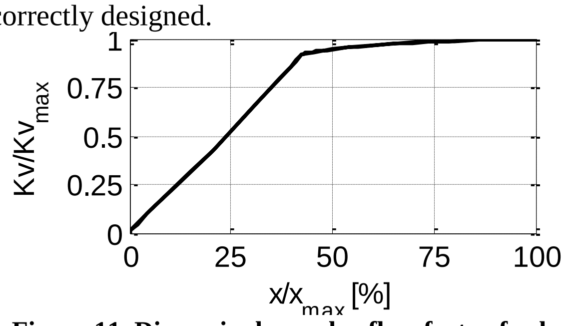


Figure 11. Dimensionless valve flow factor for ball valve with conical seat, (ideal case)

The dimensionless valve flow factor K_v/K_{vmax} with respect to the dimensionless ball valve stroke x/x_{max}100 [%] for the real case is presented in Figure 12.

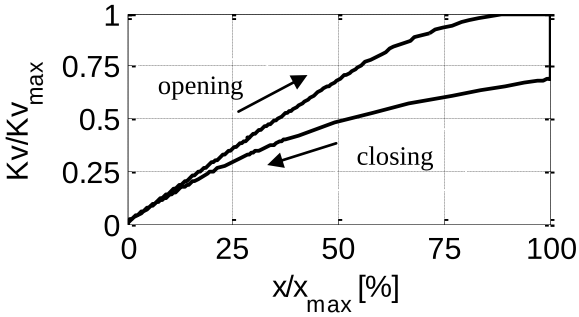


Figure 12. Dimensionless valve flow factor for ball valve with conical seat (real case)

For the ideal case presented in Figure 11, there is only one curve for both operating phases of the valve (the opening and the closing stroke times).

On the other hand, if the hydraulic system effects are considered, as shown in Figure 12, a hysteresis-like phenomenon can be observed related to the valve flow factor, because there are two different curves, one for opening stroke time and the other for the closing stroke time

The volumetric flow rate through the ball valve with conical seat is presented in Figure 13 for the real and the ideal cases. For the real case, a peak with a high overshoot at the end of the opening stroke time can be observed.

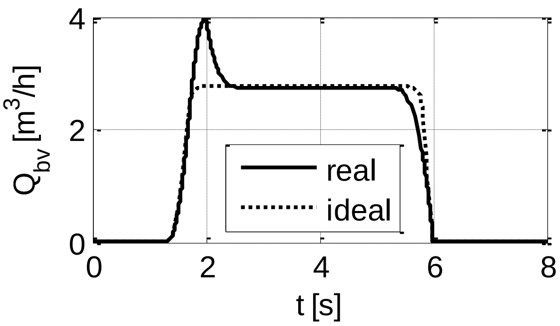


Figure 13. Volumetric flow rate through the ball valve with conical seat

In Figure 14, there are presented the curves for volumetric flow rate through the pressure relief valve. High peaks can be observed too, but in this case at the end of the closing stroke time of the ball valve with conical seat, when pressure rise until the maximum value. These peaks are evolving into a high oscillating under-damped dynamics during all the valve closing period.

As for the pressure drop in the ball valve with conical seat, which is represented in Figure 15, the same under-damped response can be observed, but with smaller overshoots.

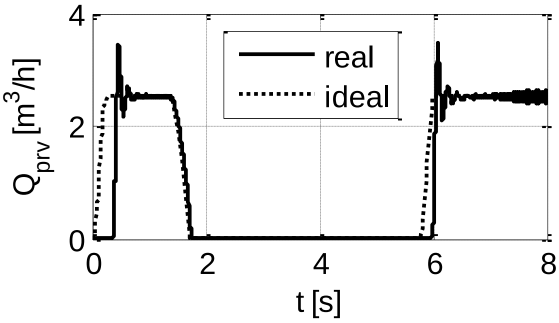


Figure 14. Volumetric flow rate through the pressure relief valve

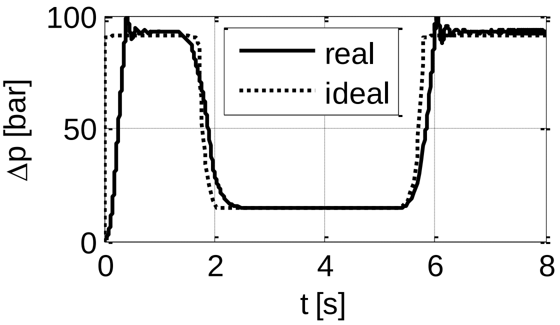


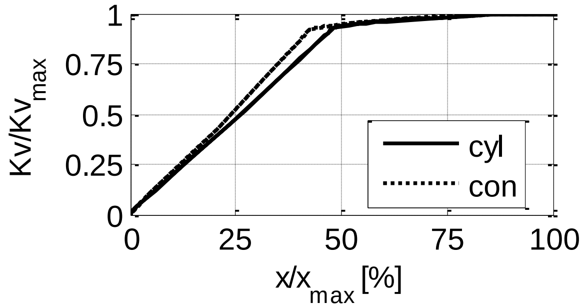
Figure 15. Pressure drop in the ball valve with conical seat

3.2 Influence of the ball valve seat

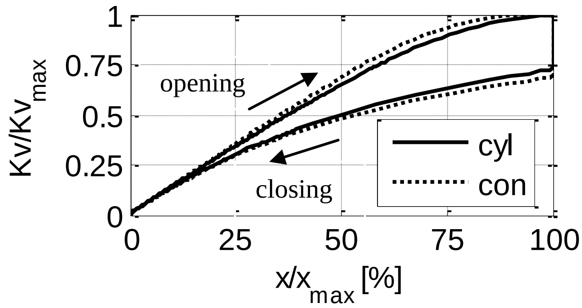
For studying the influence of the ball valve seat, two simulations has been performed with the same values for all of the geometric and functional parameters, except for the seat type which has been imposed first, cylindrical and second, conical. This analysis has been performed twice, one for the ideal case and the second for the real case (from the point of view of the effects that are associated with the hydraulic pipes).

The comparative analysis of the dimensionless valve flow factor K_v/K_{vmax}, with respect to the dimensionless ball valve stroke $x/x_{max}100$ [%] for ball valve with cylindrical and conical seats is presented in Figure 16,a for the ideal case and Figure 16,b for the real case. There can be observed some differences on both ideal and real cases. For the ideal case the valve flow factor is greater for the conical seat case, meanwhile for the real case this conclusion is valid only for the opening stroke time, for the closing stroke time the valve flow factor being greater for the ball valve with cylindrical seat.

The comparative analysis of the volumetric flow rate through the ball valve with cylindrical and conical seats with respect to the time is presented in Figure 17,a for the ideal case, and Figure 17,b for the real case. There is very difficult to observe some differences, except for the real case, where the fluid flow rate at the end of the opening stroke time and during the fully opening period of time is greater for the ball valve with conical seat.

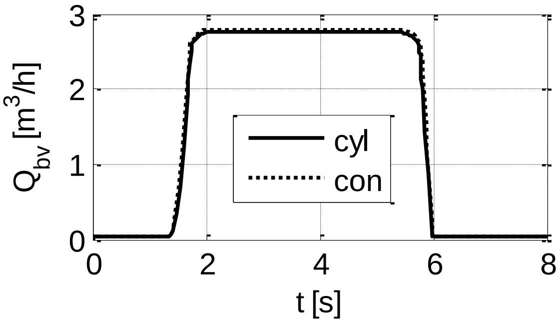


a) ideal case

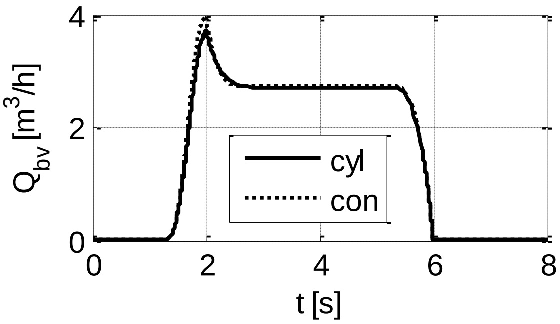


b) real case

Figure 16. Comparative dimensionless valve flow factor



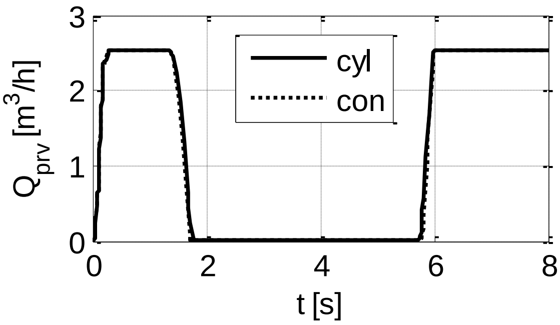
a) ideal case



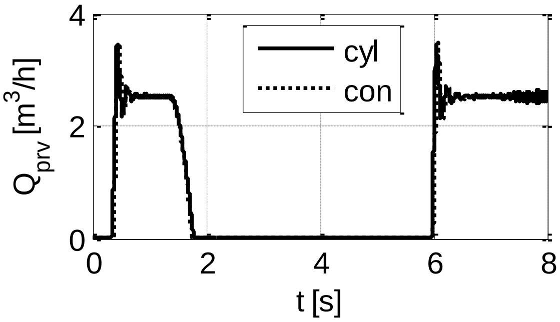
b) real case

Figure 17. Comparative volumetric flow rate through the ball valve

The comparative analysis of the volumetric flow rate through the pressure relief valve from the hydraulic system of the ball valve with cylindrical and conical seats with respect to the time is presented in Figure 18,a for the ideal case, and Figure 18,b for the real case. There are no relevant differences between the ball valve with cylindrical seat and conical seat.



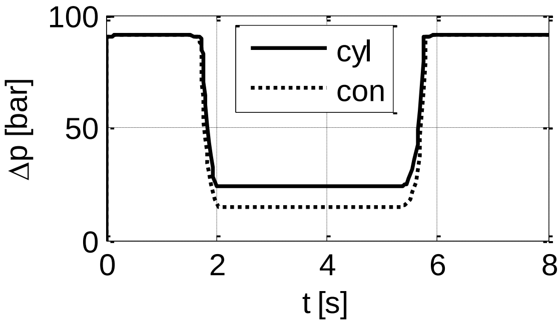
a) ideal case



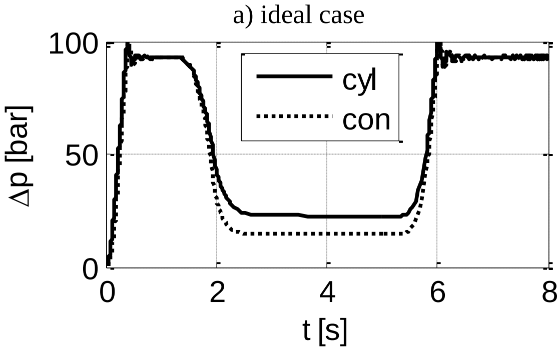
b) real case

Figure 18. Comparative volumetric flow rate through the pressure relief valve

The comparative analysis of the pressure drop in the ball valve with cylindrical and conical seats with respect to the time is presented in Figure 19,a, for the ideal case and Figure 19,b, for the real case. For this analysis, one can observe important differences, especially during the time when the ball valve is fully open, when the pressure drop in the valve is greater for the cylindrical seat.



a) ideal case



b) real case

Figure 19. Comparative pressure drop in the ball valve

4 CONCLUDING REMARKS

The valve flow factor diagram obtained for the ball valve with conical seat for the ideal case corresponds to a valve with quick opening stroke time (Figure 11). However, when the same valve is analyzed on a real hydraulic circuit, it can be observed a considerable delay, approximately 25…30%, in the volumetric fluid flow rising process (Figure 12). The hysteresis-like phenomenon obtained on the valve flow factor analysis is related strictly with the hydraulic circuit and has been also observed on the ball valve with cylindrical seat (Figure 16,b).

Considering the hydraulic system effects, it has been observed the tendency to move the valve flow factor curve of the ball valve with conical seat away from the ideal shape, toward a square root characteristic, as well as for the ball valve with cylindrical seat (Figure 12 and Figure 16,b).

The more evident difference between the ball valve with cylindrical and with conical seat can be observed on the pressure drop in the valve (Figure 19), as well as on the volumetric fluid flow through the ball valve (Figure 17), especially during the fully opening period of time. This conclusion can be explained by the lower local hydraulic loss of the fluid flow through the ball valve with conical seat, comparing with the cylindrical seat version.

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