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HYDRAULIC SYSTEM PROTECTION AGAINST CATASTROPHIC LINE FAILURE USING NEWLY DEVELOPED SAFETY VALVE

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Abstract

Hydraulic hose and line failures in the field cause an extreme hazard to operators, work crews, and the environment. This happens more than is reported on a daily basis. Hydraulic line breaks are a good reason for the increased fluid replacement bills of nearly all industrial and mobile hydraulic equipments. Sadly, we can fairly say up to now, there has been no cost effective solution introduced to the industry to resolve this hazard effectively until now. In this paper, a patent pending device is proposed as a realistic and feasible solution to offer protection when catastrophic accidental line breaks occur. The device consists of a modified spool design, found in velocity fuses with an optimized dynamic surge chamber. A mathematical model to describe the valve's dynamic response has been developed. This mathematical model has been used to assure the feasibility of the idea by simulating the valve's response over a wide range of working pressures. The agreement between the simulation results and the prototype lab experiments validates the mathematical model. Results are presented and discussed. As this is pending patent, the article focuses on the feasibility of the idea, the valve diagrams are simplified and some critical data are omitted for the protection of the owner.

Keywords: leakage, hydraulic line, hose, safety, valve

1 Introduction

Hydraulic line breaks, while the hydraulic power supply is running can cause serious harm to the machine operators, environment and damage equipment as well.

Several incidents have been reported by Hazard Information Bulletin (1977) and Ryan (1984) indicates number of deaths and plants shutdown due to fire accidents. Investigations show that the origins of these accidents were due to high pressure hydraulic line breaks.

Currently hydraulic velocity fuses are used in sub systems as load lock devices. Brezonick (2000) and Hitchcox (2007) have introduced in their articles the importance of using such velocity fuses for the hydraulic load safety. As shown in Fig.1, for example, where individual cylinders experience catastrophic load failure due to hose rupture, load will freely fall for a moment causing accelerated flow through the velocity fuse that shuts off and locks the cylinder in place preventing free fall of the load.

Despite limiting the hazard by locking the vertical load, the drawback is that the hydraulic power unit is still pumping fluid through ruptured hose and the surrounding environment.

If the operator does not shut off the engine or equipment in time, the pump will continue to vent supplying fluid out of the system until the reservoir is empty or cavitation of the pump. In many cases, the operator is the one in danger and cannot shut off the equipment after being exposed to this hazard.

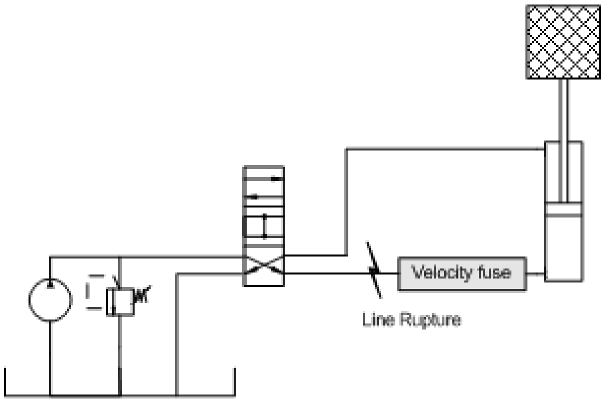

Fig. 1: *Velocity fuse to protect subsystem*

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This continuation of system pumping fluid results in expensive repair costs for equipment, increased liability for injury and environmental damage along with possible fines. Hose ruptures spray the heated fluid into atmosphere and environment which may severely burn workers, or causing jerk reactions leading to greater harm to workers attempting to get out of the way.

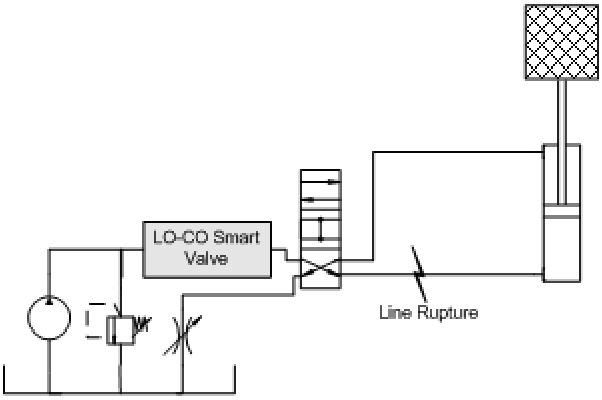

Fig. 2: *Smart valve to protect the full system*

Current velocity fuses, commercially used for holding vertical loads in case of line rupture, are not therefore capable of protecting the full system if located on the downstream side of the pump.

During normal operation, occasional surges occur that have prematurely caused current velocity fuses to shut off the flow. Therefore premature shut off of the flow during normal operation surges makes current fuses unusable for protecting the entire system from catastrophic hose or line failure.

A great need has thus existed for a reliable, smart safety shut off valve that can withstand system surges without prematurely shutting off the supply pressurized fluid. Most importantly while handling surges, it has to be able to shut off supply pressure during a catastrophic line break.

The newly developed safety valve is such an invention, patent pending, to protect the full operational hydraulic system, and sub-systems where work is being performed. This article will use the generic name “safety valve” hereafter.

First and foremost, this safety valve is able to distinguish between hydraulic line ruptures and normal operational surges (regardless if using variable or Fixed volume pumps) even when a tandem center directional control valve is selected.

This safety valve eliminates the need for expensive electronic feedback, flow, and pressure devices, by using the natural physics of fluid pressure and flow characteristics. Thereby being much more cost efficient.

2 Safety Valve Operation & Construction

As shown in Fig. 2, the safety valve can be placed downstream of hydraulic power supply line (Pump) immediately after system relief valve, well before the operational control valves for subsystems.

During normal operation when the directional valve shifted to different positions, what makes the safety valve handle the surges is through the optimized return line back pressure and valve chamber size.

As shown in Fig. 3, the safety valve has a housing, spool and spring device that closes when flow rates accelerated. As we have mentioned earlier, the challenge of the valve is to distinguish between actual hydraulic line rupture and normal working conditions.

The smart part of the valve is an optimized surge chamber added to the valve. The mission of the chamber is to disorientate fluid flow which then acts as a time delay mechanism regulating the valve shutting off.

Optimized chamber volume and internal construction stops fuse from shutting off until unsafe condition such as a ruptured line occur. Not until a ruptured line occurs does this chamber quickly reduce the differential pressure allowing the spool to instantly shut off fluid flow. Figures 4 to 8 show the following; respectively:

- Sleeve Critical Dimensions
- Spool Critical Dimensions
- Spring Critical Dimension
- Assembly Drawing, Fully Opened
- Assembly Drawing, Fully Closed

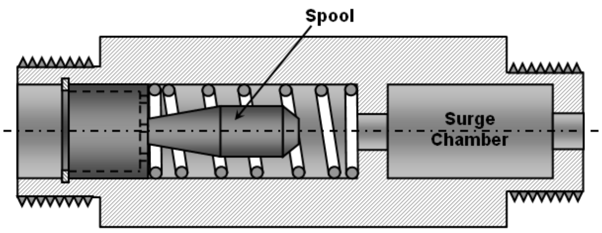

Fig. 3: *Safety valve construction*

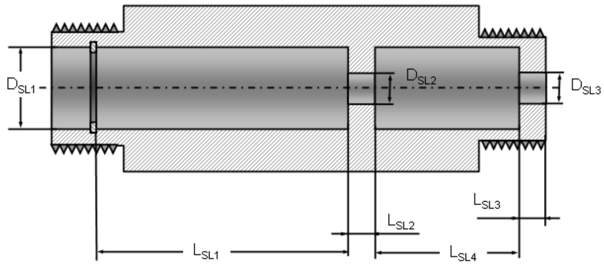

Fig. 4: *Sleeve Critical Dimensions*

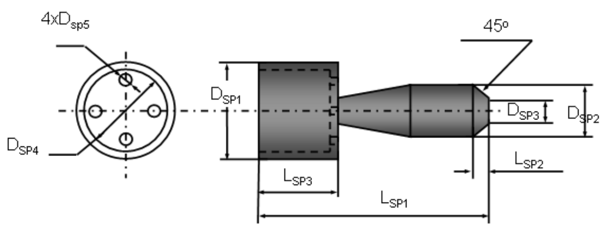

Fig. 5: *Spool Critical Dimensions*

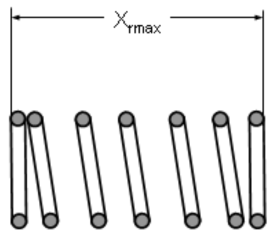

Fig. 6: *Spring Critical Dimension*

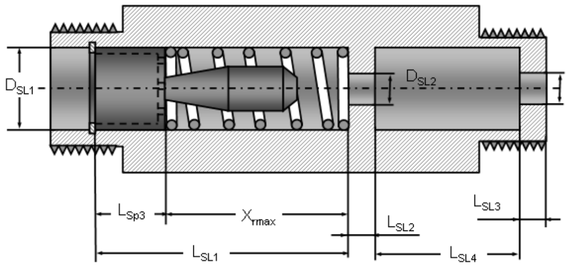

Fig. 7: *Assembly Drawing, Fully Opened*

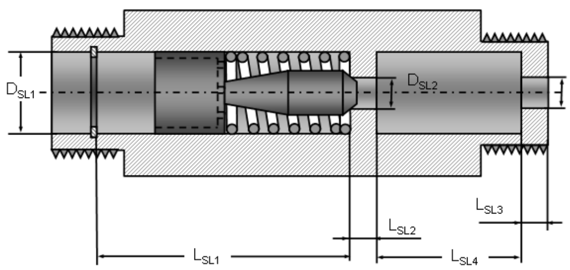

Fig. 8: *Assembly Drawing, Fully Closed*

3 Safety Valve Mathematical Model

Developing mathematical model requires kinematic study followed by a dynamic study. Figure 9 shows that the spool travels a maximum displacement S_{max} between the fully opened to the fully closed position in the positive direction shown in the figure. Based on the spool and sleeve dimensions, at the fully closed position, the face of the spool advances to the inlet of the surge chamber a distance L_{SP4}. The following four equations are used to find the value of S_{max} and L_{SP4} geometrically in terms of other known dimensions.

$$0 <= S <= S_{max} \tag{1}$$

$$S_{max} = L_{SL1} + L_{sp4} - L_{SP1} \tag{2}$$

$$L_{sp4} = (D_{SL2} - D_{SP3})/2 \tag{3}$$

Assuming that the spool face is chamfered on 45 degree, as shown in Fig. 10, then

$$D_{SP3} = D_{SP2} - 2L_{SP2} \tag{4}$$

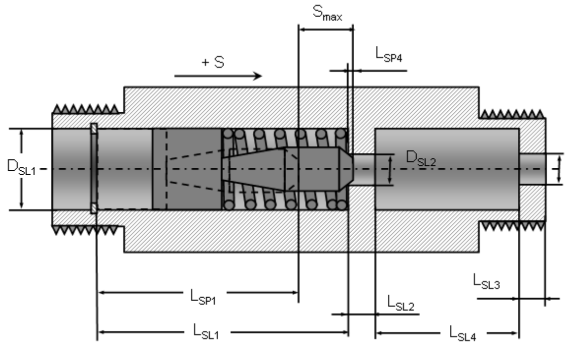

Fig. 9: *Safety valve spool displacement*

In the following section, force balance equation will be developed. Figure 11 shows the spool in an intermediate state between the fully opened and fully closed positions. Neglecting the flow forces, in this situation the spool is balanced under the effect of pressure forces and spring forces.

$$S'' = \frac{1}{M_{SP}}\left[P_1A_1 - P_2A_2 - P_3A_3 - k_f S' - k_x S\right] \tag{5}$$

$$S' = \int S'' dt \tag{6}$$

$$S = \int S' dt \tag{7}$$

Viscous friction force exists only during the spool travel from any current position to a new balanced position. The following equations describe the dynamics of the spool during its travel between two balanced positions.

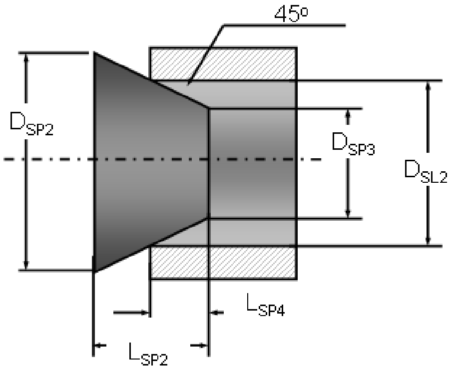

Fig. 10: *Valve spool-sleeve geometrical relations*

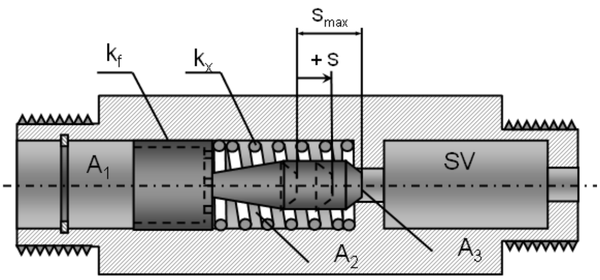

Fig. 11: *Safety valve spool force balance*

As shown in Fig.11, the area A_1 is the spool projected area that is affected by the pressure at the upstream side of the spool. The area A_2 is the spool projected area that is affected by the pressure at the downstream side of the spool in the spring chamber. The area A_3 is the spool face projected area when the spool is advancing in the inlet of the surge chamber. Whenever the spool face does not advance to the inlet of the surge chamber, the area A_3 is considered zero because the spool face is still affected by the pressure of the spring chamber.

The area A_1 is the fixed effective area at the upstream side of the spool, it is calculated by Eq. 8. The area A_2 the effective area at the downstream side of the spool, is a variable value based on the spool position relative to the sleeve and is calculated by Eq. 9. Before the spool face reaches the inlet of the surge chamber, both A_1 and A_2 are equal since A_3 is considered zero at that time. Once the spool advances to the inlet of the surge chamber, the area A_2 decreases by the value of A_3 affected now by the surge chamber pressure and it can be calculated by Eq. 10.

$$A_1 = \frac{\pi}{4}(D_{SP1}^2 - 4D_{SP5}^2) \quad (8)$$

$$A_2 = A_1 - A_3 \quad (9)$$

$$\begin{aligned} &if\ S <= (S_{max} - L_{SP4})\ then\ A_3 = 0 \\ &if\ S > (S_{max} - L_{SP4})\ then\ A_3 = \frac{\pi}{4}D_3^2 \end{aligned} \quad (10)$$

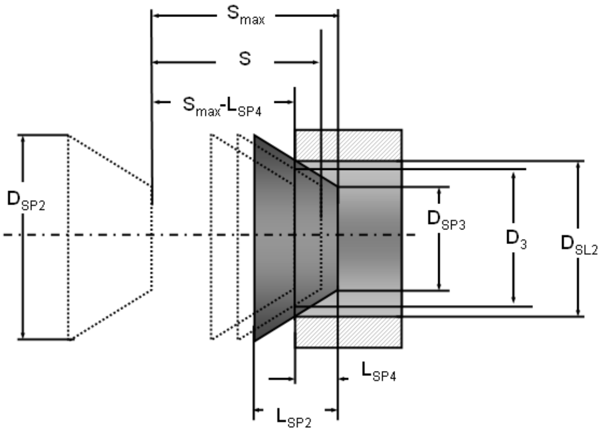

Fig. 12: *Safety valve spool force balance*

As shown in Fig. 12, the value of D_3 is the instantaneous diameter of the spool chamfering at the inlet of the surge volume. The value of D_3 is calculated by Eq. 11 and it varies from a minimum value of D_{SP3} to a maximum value of D_{SL2}.

$$D_3 = D_{SP3} + 2\left[S - (S_{max} - L_{SP4})\right] \quad (11)$$

Figure 13 shows the safety valve located in a schematic circuit. The circuit consists of a pump, discharging a constant flow rate Q_P equal 20 L/min. A relief valve is used in parallel to the pump supply line and adjusted at 250 bar.

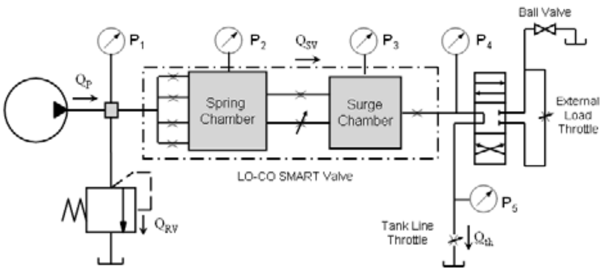

Fig. 13: *Safety valve in the system*

If the relief valve is not cracked, the flow passing through the safety valve Q_{SV} will be equal the pump supply flow rate Q_P. If the relief valve is cracked, flow rate Q_{RV} will pass through it based on a static characteristic of the valve. The flow rate Q_{SV} enters the spring chamber in the safety valve sleeve via four holes in the spool that are represented by four fixed orifices. The spring chamber is connected with the surge chamber with fixed and variable orifices; one of them is considered at a time. Before the spool face enters the inlet of the spring chamber, the variable orifice is considered fully closed. Once the spool face enters the inlet of the surge volume, the fixed orifice is considered fully closed.

The surge chamber is connected in line with the directional valve with a fixed orifice that represents the outlet of the surge chamber. Two variable throttles are used, one to simulate the external load and the other to apply back pressure in the return line. Five pressure values are monitored, P_1 is the pump pressure, P_2 is the pressure of the spring chamber, P_3 is the pressure of the surge chamber, P_4 is the load pressure including the pressure drop across the directional valve and the tank line pressure P_5.

The following assumptions are considered for the next part of the valve modeling to reduce the complexity of the model:

Leakage from the upstream side of the spool to the spring chamber is negligible

Oil compressibility considered only in the surge chamber

All the orifices are considered as sharp-edged short

The following very well known equation will be used to describe the nonlinear relation between the differential pressure across any sharp-edged short orifice and the flow rate passing through it.

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho_f}} \quad (12)$$

Since the fluid specific gravity is the ratio between fluid density and water density, then Eq. 12 can be rewritten to use the commonly used constants as follows

$$Q = C_d A \sqrt{\frac{2\Delta P}{SG \cdot \rho_w}} \quad (13)$$

Equation 13 can be used to find the differential pressure in terms of the flow rate as follows.

$$\Delta P = \frac{SG \cdot \rho_w \cdot Q^2}{C_d^2 A^2} \quad (14)$$

By applying Eq. 14 on the tank line throttle, neglecting the flow resistance in the hydraulic line between the throttle and the tank, pressure P_5 is

$$P_5 = \frac{SG \cdot \rho_w \cdot Q_{th}^2}{C_d^2 A_{th}^2} \quad (15)$$

Fluid flow through the tank line throttle equals the safety valve flow Q_{SV} if a hydraulic motor is used. If a differential area cylinder is used, the cylinder area ratio must be considered in calculating the return flow.

The tank line throttling area is a simple variable circular area function of the throttle diameter as follows.

$$A_{th} = \pi D_{th}^2 / 4 \quad (16)$$

The pressure P_4 is the cumulative load pressure and is calculated as follows.

$$P_4 = P_5 + \Delta P_{DV} + P_{EL} \quad (17)$$

Where ΔP_{DV} is the pressure drop across the directional valve and P_{EL} is the pressure equivalent to the external load.

Applying the continuity equation on the surge chamber results in the pressure P_3 as follows.

$$Q_{34} = C_d A_{34} \sqrt{\frac{[P_3 - P_4]}{SG \cdot \rho_w}} \quad (18)$$

$$A_{34} = \frac{\pi}{4} D_{SL3}^2 \tag{19}$$

$$P_3 = \frac{B}{V_s} \int (Q_{SV} - Q_{34}) dt \tag{20}$$

Where V_s is the volume of the surge chamber, B is the oil bulk modulus. The values Q_{SV} and Q_{34} are the flow in and out of the surge volume respectively.

Pressure in the spring chamber is calculated by the following equation.

$$P_2 = P_3 + \frac{SG \cdot \rho_w \cdot Q_{SV}^2}{C_d^2 A_{23}^2} \tag{21}$$

Where A_{23} is the throttling area between the spring chamber and the surge volume chamber port. As it is mentioned earlier, spring chamber is connected with the surge chamber using fixed and variable orifices; one of them is considered at a time. As shown in Fig. 14.a, the area A_{23} will be the fixed area of the surge chamber inlet, expressed as follows.

$$if\ S <= (S_{max} - L_{SP4})\ then\ A_{23} = \frac{\pi}{4} D_{SL2}^2 \tag{22}$$

As shown in Fig. 14.b, the area A_{23} will be a variable area based on the spool position relative to the surge chamber inlet. As shown in Fig. 14.c, the area in this case will be a frustum of a right cone that has the dimensions D_1, D_2 and L. Instantaneous values of these dimensions are found geometrically as in the Eq. 23, 24 and 25, respectively. The area A_{23} then can be expressed by Eq. 26.

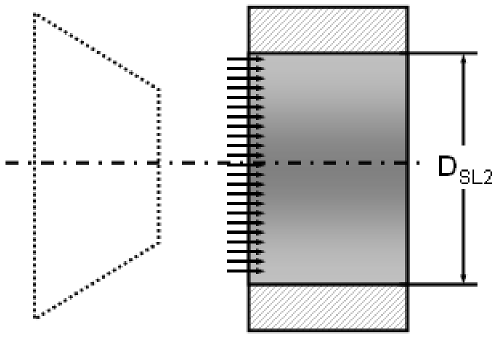

Fig. 14.a: *Throttling area A_{23}, case 1*

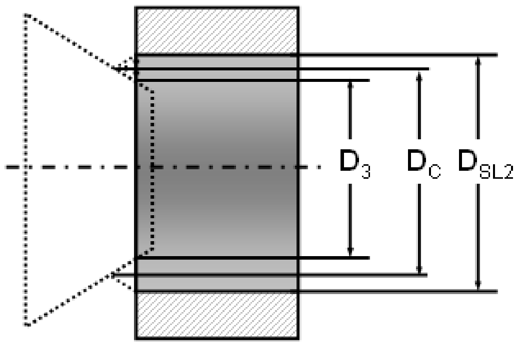

Fig. 14.b: *Throttling area A_{23}, case 2*

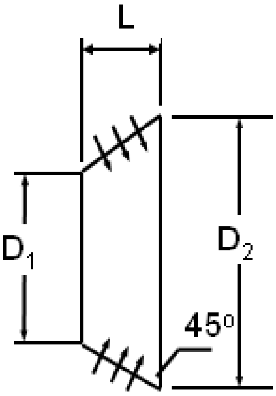

Fig. 14.c: *Calculation of throttling area A_{23}, case 2*

$$D_1 = 0.5 D_{SL2} + 0.5(D_{SL2} - D_3) \tag{23}$$

$$D_2 = D_{SL2} \tag{24}$$

$$L = 0.25\ (D_{SL2} - D_3) \tag{25}$$

If $S > (S_{max} - L_{SP4})$ then

$$A_{23} = \frac{\pi}{2} [\ D_1 + D_2] L \sqrt{2} \tag{26}$$

Equations 23 to 25 are valid only for a spool chamfered on 45 degree as mentioned earlier.

Based on the relief valve status, all pump supply flow or part of it Q_{SV} is passing through four holes to reach the spring chamber. Pump pressure is then calculated as follows.

$$P_1 = P_2 + \frac{SG \cdot \rho_w \cdot Q_{SV}^2}{4 C_d^2 A_{12}^2} \tag{27}$$

Where

$$A_{12} = \frac{\pi}{4} D_{SP5}^2 \tag{28}$$

The article is focused on the feasibility of the invention, so an assumption of a relief valve linear static characteristics are considered to predict the status of the relief valve and the flow passing through it, Q_{RV}. As shown in Fig. 15, at any operating pump pressure P_1, Q_{RV} is calculated as follows.

$$P_{CO} = P_{CR} \left[1 + (OV/100) \right] \tag{29}$$

$$\text{if } P_1 \le P_{CR} \rightarrow Q_{RV} = 0 \tag{30}$$

$$if\ P_{CR} < P_1 < P_{CO} \rightarrow Q_{RV} = p \left[\frac{P_1 - P_{CR}}{P_{CO} - P_{CR}} \right] \tag{31}$$

$$if\ P_1 \ge P_{CO} \rightarrow Q_{RV} = Q_p \tag{32}$$

$$Q_{SV} = Q_P - Q_{RV} \tag{33}$$

Where, OV, P_{CR} and P_{CO} are the relief valve percentage override, cracking pressure and cut off pressure, respectively.

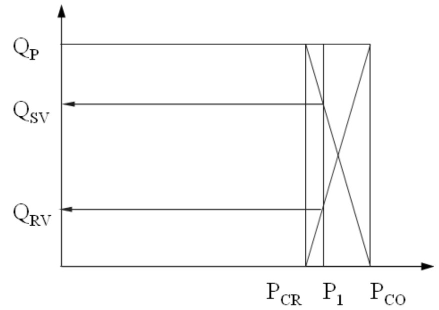

Fig. 15: *Relief valve linear static characteristic*

4 Safety Valve Performance Simulation

Matlab-Simulink model has been developed based on the previously developed mathematical model. Several runs were made to simulate the valve performance, using three different operating pressures. Simulation results for operating pressure 100 bar are presented in Fig. 16 and Fig. 17. For operating pressure 135 bar, results are presented in Fig. 18 and Fig. 19. For operating pressure 175 bar, results are presented in Fig. 20 and Fig. 21.

Each simulation run started with the assumption that the pump is off and turned on at the time zero, while the directional valve connects the pump to the load throttle. Simulation results show that the safety valve spool oscillate for about one second before reaching a steady state balanced position at about 75 % of its saturation limit. This balanced position is the result of two balanced forces, one being the differential pressure drives the spool to close and the other is the spring force drives the spool to open. Figures 17, 19 and 21 show how the safety valve balanced position is independent of the operating pressure and depends on the flow passing through it, with constant of 20 L/min. Figures 16, 18 and 20 show that, the safety valve is kept open allowing for the pump flow to feed the system regardless of the operating pressure.

From the steady state loading condition, at time 3, the pump is unloaded by switching the directional valve to the neutral tandem center. Figures 16, 18 and 20 show that the safety valve remains open and was smart enough to understand that this is a normal unloading operation. The pump and load pressures are reduced to little bit above the return line pressure. The valve remains open with some spool oscillation before it returns to same previous steady state condition. From the steady state unloading condition, the pump is loaded again at time 6 by switching the directional valve from the neutral to a loaded condition. Simulation results show that the valve remains open and performed similar to the first stage of the simulation.

From the steady state loading condition, at time 9, line rupture was simulated by reducing the load pressure suddenly to the atmospheric pressure; and the flow in the return line becomes zero. In this condition, a surge flow is developed at the upstream side of the safety valve spool that drives it to reach its saturation limit and shuts off the pump flow supply line. Accordingly, pump pressure increases to the relief valve setting value of 250 bar. Same trend of results are seen on Fig. 16, Fig. 18 and Fig. 20.

Simulation results assure the feasibility of the safety valve function. In the following section, the model is further validated based on laboratory testing.

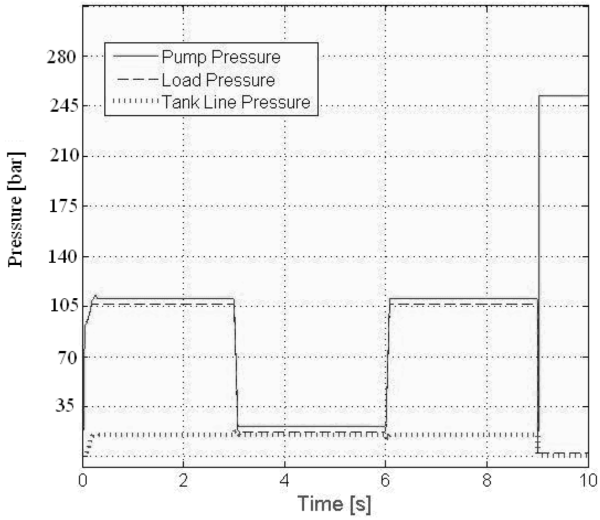

Fig. 16: *Simulated pressures at P_{load} = 100 bar*

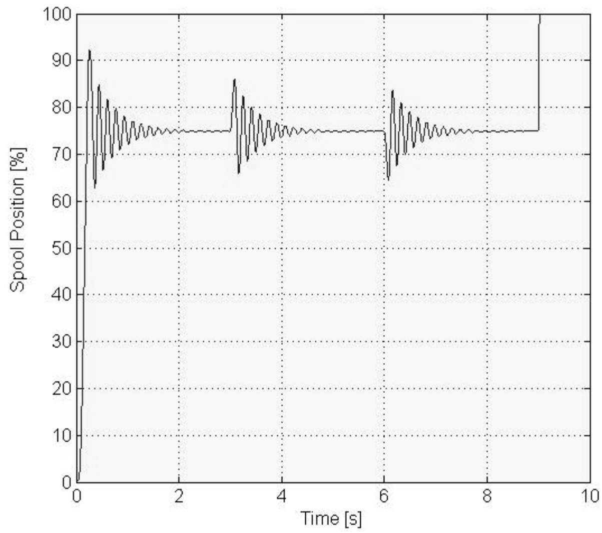

Fig. 17: *Simulated spool position at P_{load} = 100 bar*

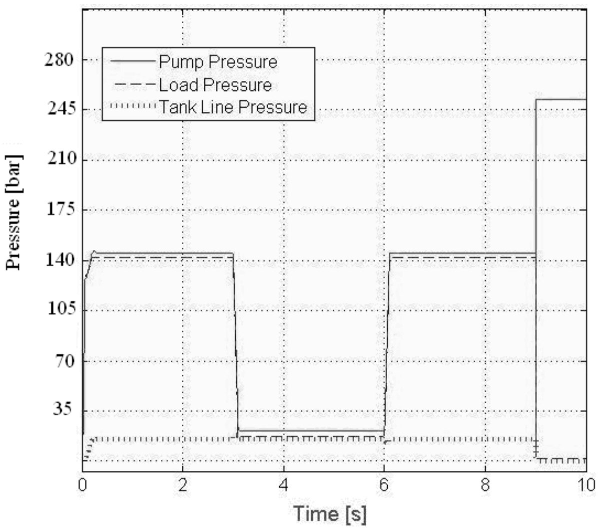

Fig. 18: *Simulated pressures at P_{load} = 135 bar*

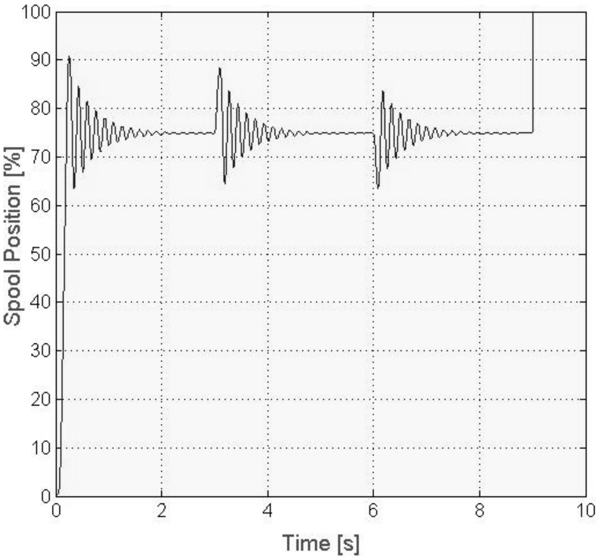

Fig. 19: *Simulated spool position at P_{load} = 135 bar*

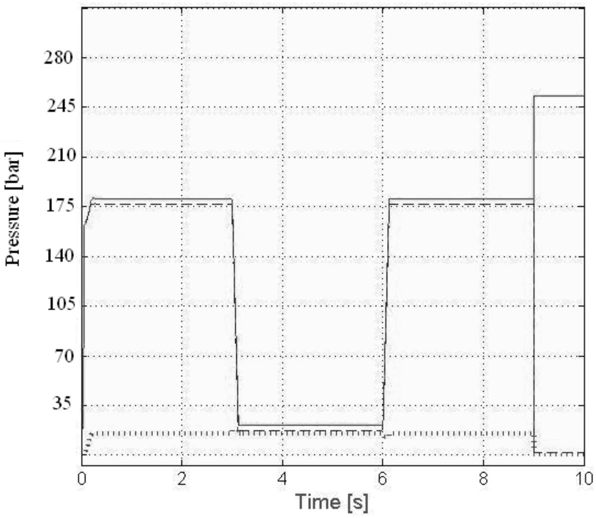

Fig. 20: *Simulated pressures at P_{load} = 175 bar*

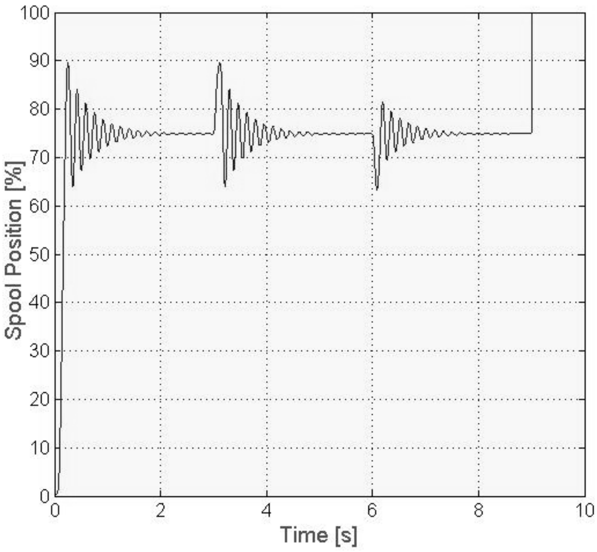

Fig. 21: *Simulated spool position at P_{load} = 175 bar*

5 Experimental Test and Validation

As shown in Fig. 22, a test hydraulic circuit was designed based on the mathematical model in Fig. 13.

The physical test circuit required the same working conditions or 20 L/min flow and 250 bar pressure relief setting. As shown in Fig. 23, the pump is supplying flow to the upstream side of safety valve and was measured by a flow meter.

Pump flow continues through the safety valve to the reservoir through the tandem position of the directional valve. As shown in the figure 23, data logger was used with a set of three pressure sensors recording pump pressure P_1, load pressure P_4 and tank line pressure P_5.

Two needle valves were implemented, one as a return line throttle and the other to simulate external load. A manual ball Valve is implemented to simulate a line rupture when the pump is under loading conditions.

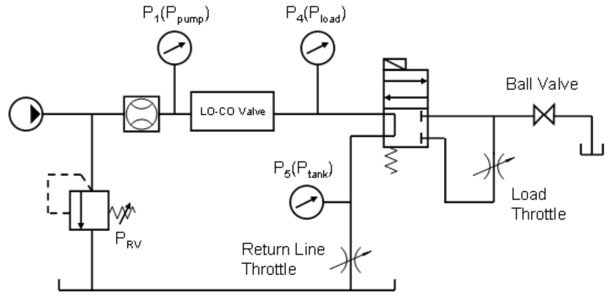

Fig. 22: *Test hydraulic circuit*

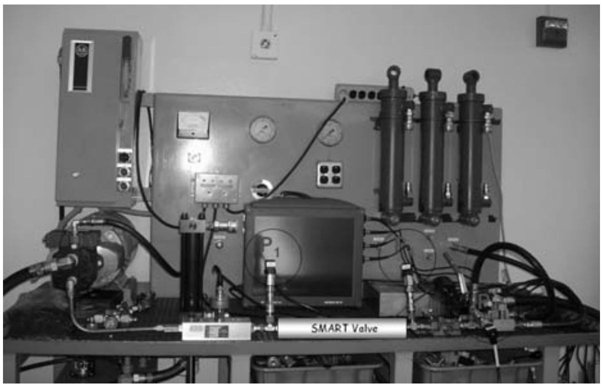

Fig. 23: *Test setup and experimental measurement*

The test stand has been used to test the feasibility of the safety valve prototype. In some circumstances in controlled test, it was not necessary for a return line throttle, as the internal resistance of return line was already optimized for this operation. The current test data were taken when the tank line pressure P_5 was adjusted at 14 bar. Using manually controlled sensitive needle valves on the load line and the tank line made it easy to set the load pressure. The following load pressures were tested, 117 bar, 150 bar and 175 bar.

The following test protocol was also repeated three times at the aforementioned load pressures. Pump is turned on while the directional valve solenoid is energized to connect the pump delivery line to the load throttle. Once the steady state was achieved, the directional valve solenoid is de-energized to unload the pump through the directional valve tandem center. The collected results using the data logger have been presented in Fig. 24, 30 and 36. Results show that the safety valve performed smartly and it was kept open as in normal pump unloading conditions. The surge chamber and the tank line pressure were optimized to achieve this situation. Pump and load pressures are then reduced to approximately tank line pressure value.

The following test protocol was repeated three times at the aforementioned load pressures. Pump is turned on while the directional valve solenoid is energized to connect the pump delivery line to the load throttle. After spool reached steady state and safety valve is open, the shut off valve is opened suddenly to simulate line rupture. What is demonstrated in the following paragraphs is how the Matlab simulation predicted accurately how the prototype would work. The actual hydraulic test circuit did just that, validating each other.

The collected results using the data logger have been presented in Fig. 27, 33 and 39. Results show that the safety valve performed smartly and it shuts the pump oil supply line when there is a line rupture condition. Pump pressure reaches the relief valve setting value, 250 bar. Load pressures and tank pressure reduced to the atmospheric value since the flow becomes zero.

In order to validate the mathematical model, the Matlab-Simulink model was used to simulation the safety valve response to the same pressure change and timing conditions at which the physical test ran. Figures 25 and 28 show the simulation results with the pump normally unloaded and load pressure adjusted at 117 bar.

Similarly, Fig. 31 and 34 when the load pressure adjusted at 150 bar and Fig. 37 and 40 when the load pressure adjusted at 175 bar. Figures 26 and 29 show the simulated safety valve spool position when the pump normally unloaded and Fig. 29 in the case of line rupture, respectively, when the load pressure adjusted at 117 bar. Similarly, Fig. 32 and 35 when the load pressure adjusted at 150 bar and Fig. 38 and 41 when the load pressure adjusted at 175 bar.

The very good agreement between the simulation and test results fairly validates the model and confirms the feasibility of the idea. Test results show the effect of the relief valve dynamics as a pressure overshooting during the safety valve shutting off the pump oil supply line. Simulation results did not show that because it considered only the relief valve static characteristics.

6 Conclusion

Catastrophic High Pressure Fluid Exposure by hose and line failures represent an extreme hazard to the environment, work crews and the equipment. Premature spool closure is why current velocity fuse cannot be used within normal operating systems, especially when the system is using a tandem or open center control valve. This hazard can now be eliminated using safety valve, which is patent pending. Safety valve performed smartly and shut off the pump oil supply line in case of line rupture. The valve is able to distinguish between normal pump unloading condition and line rupture. The smartness of the valve is achieved by optimizing the integrated surge chamber volume with a corresponding tank line pressure. In some circumstances hydraulic back pressure in the return line is already available such as provided in pressurized reservoir's for example. The chamber acts as a delay mechanism and maintains the differential pressure that acts against spool until the differential pressure and volume is depleted due to catastrophic rupture before allowing spool to close. This paper documents the study to confirm the feasibility of the safety valve idea. A mathematical model has been developed using Matlab-Simulink software. The developed model was utilized in simulating the valve response to various pressure change conditions in a typical hydraulic circuit. The fruitful benefit of developing such model is offering a flexible tool to optimize the valve design parameter for future tests and different circuits or systems. The agreement between the simulation results and the prototype test results validate the model and confirm the feasibility of the safety valve operating principle, over the most typical commercially working pressure range. Further work is going on to optimize design parameters of safety valve with various sizes.

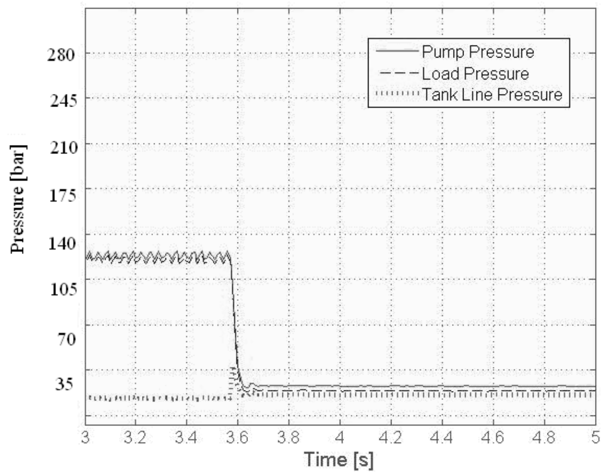

Fig. 24: *Actual pressures when switch from P_{load} = 117 bar to P_{tank}*

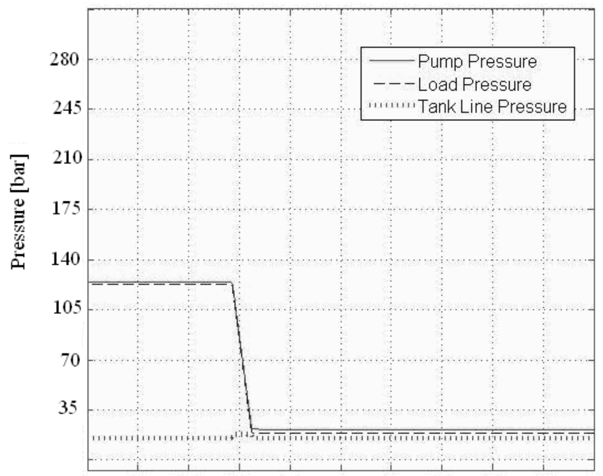

Fig. 25: *Simulated pressures when switch from P_{load} = 117 bar to P_T*

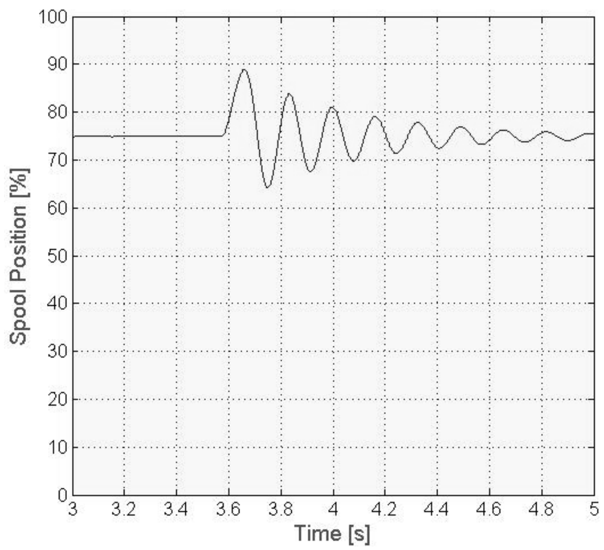

Fig. 26: *Simulated spool position when switch from P_{load} = 117 bar to P_T*

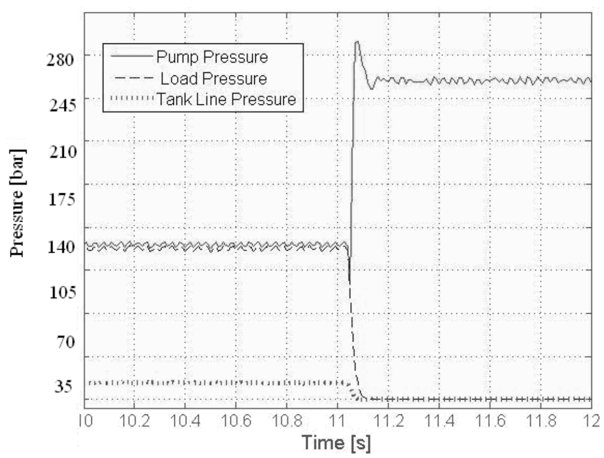

Fig. 27: *Actual pressures when switch from P_{load} = 117 bar to P_{atm}*

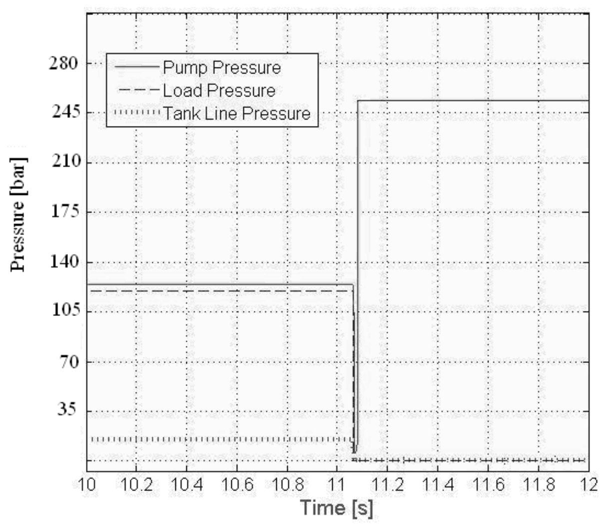

Fig. 28: *Simulated pressures when switch from P_{load} = 117 bar to P_{atm}*

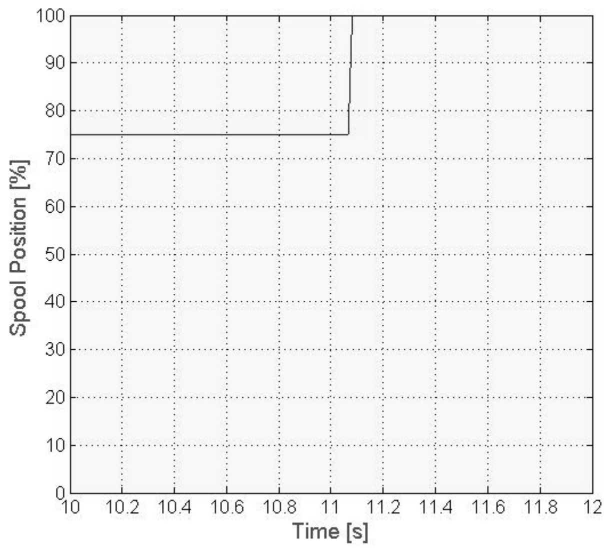

Fig. 29: *Simulated spool position when switch from P_{load} = 117 bar to P_{atm}*

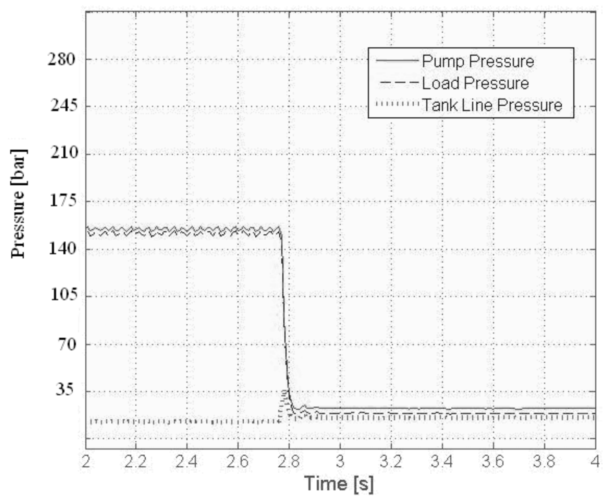

Fig. 30: *Actual pressures when switch from P_{load} = 150 bar to P_T*

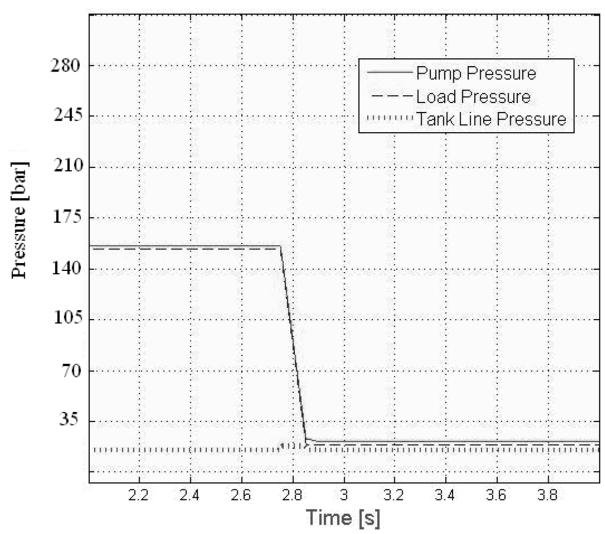

Fig. 31: *Simulated pressures when switch from P_{load} = 150 bar to P_T*

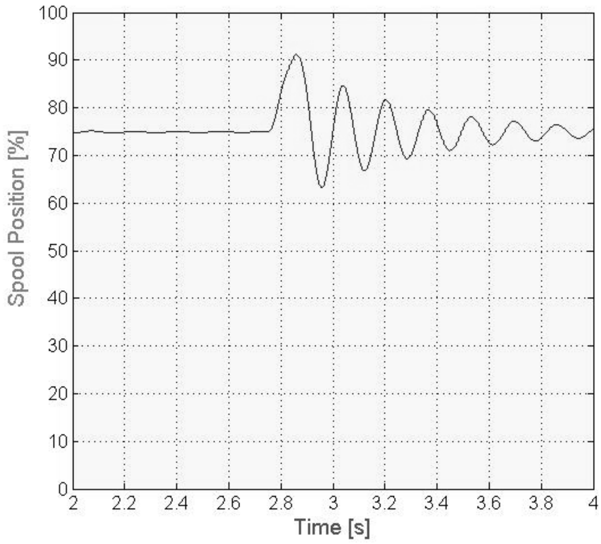

Fig. 32: *Simulated spool position when switch from P_{load} = 150 bar to P_T*

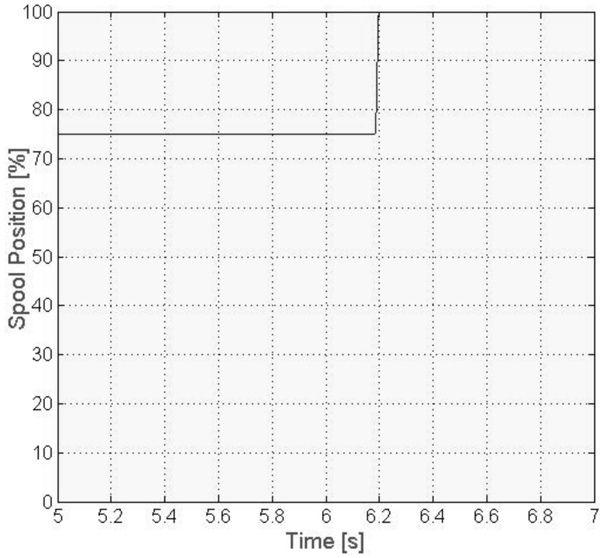

Fig. 35: *Simulated spool position when switch from P_{load} = 150 bar to P_{atm}*

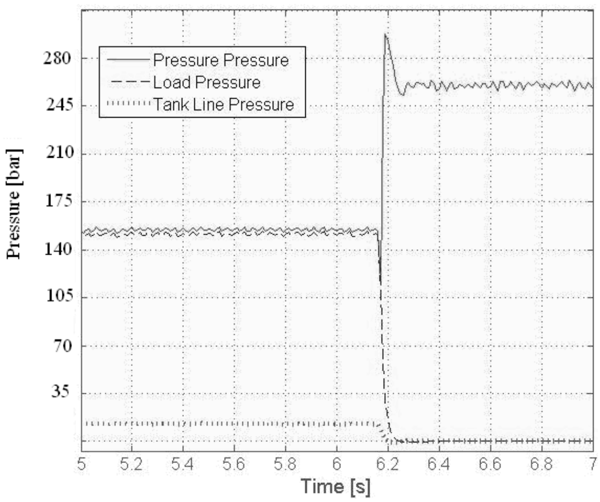

Fig. 33: *Actual pressures when switch from P_{load} = 150 bar to P_{atm}*

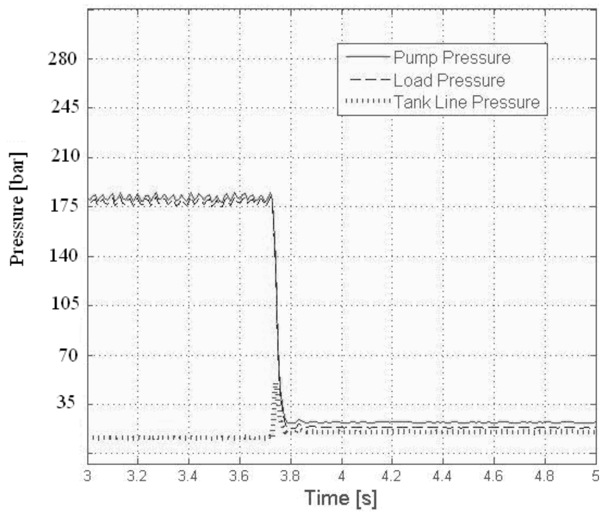

Fig. 36: *Actual pressures when switch from P_{load} = 175 bar to P_T*

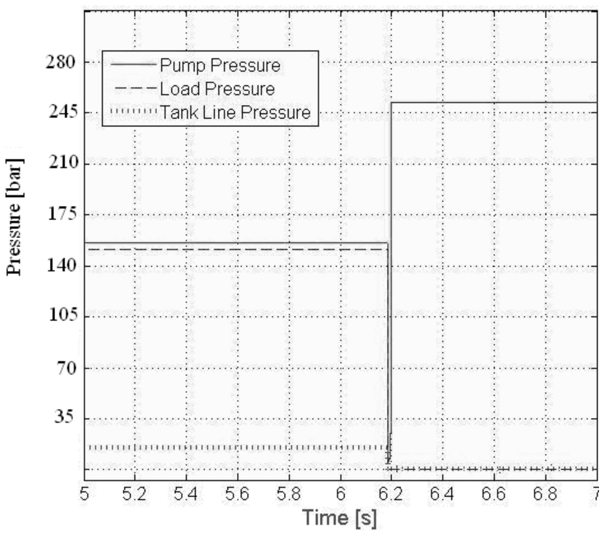

Fig. 34: *Simulated pressures when switch from P_{load} = 150 bar to P_{atm}*

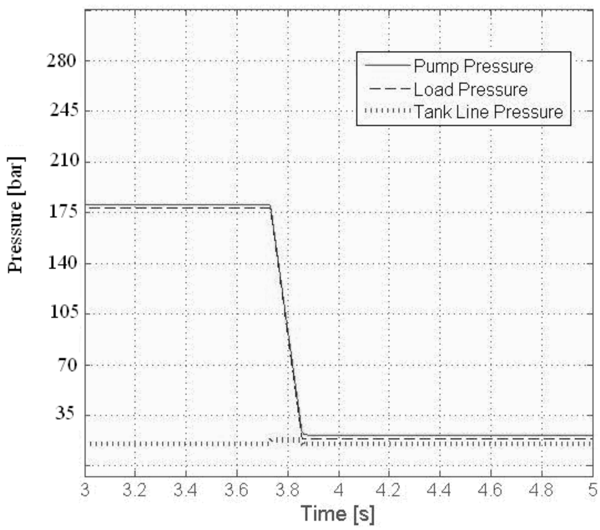

Fig. 37: *Simulated pressures when switch from P_{load} = 175 bar to P_T*

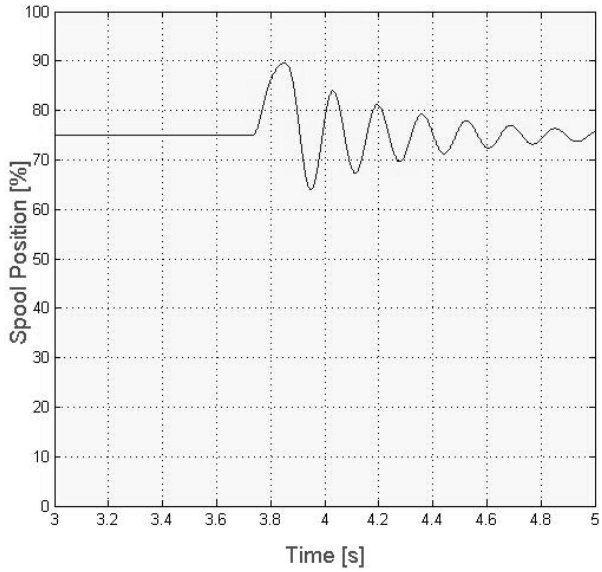

Fig. 38: *Simulated spool position when switch from P_{load} = 175 bar to P_T*

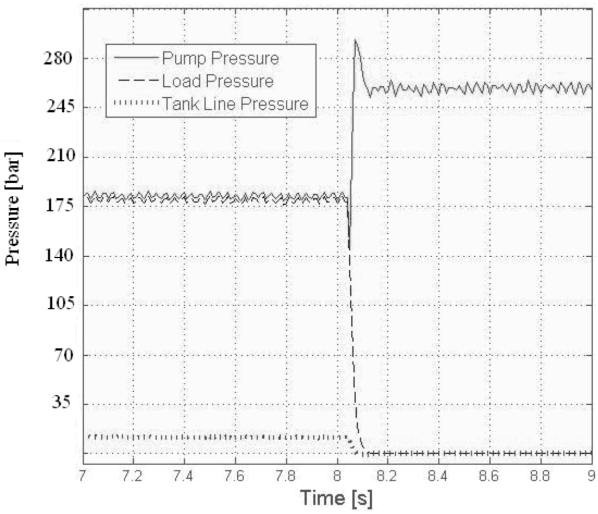

Fig. 39: *Actual pressures when switch from P_{load} = 175 bar to P_{atm}*

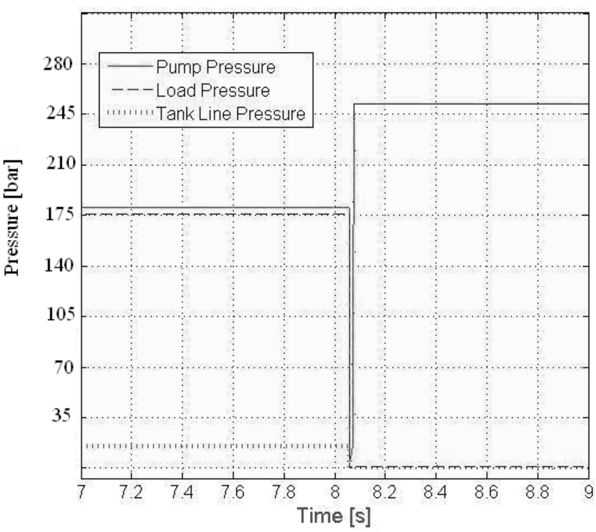

Fig. 40: *Simulated pressures when switch from P_{load} = 175 bar to P_{atm}*

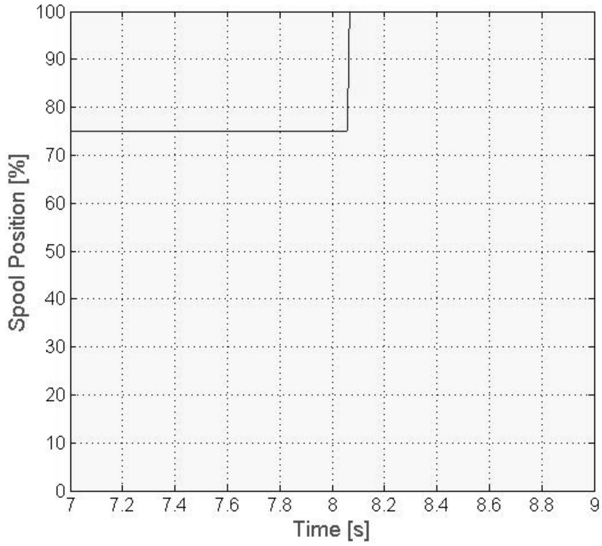

Fig. 41: *Simulated spool position when switch from P_{load} = 175 bar to P_{atm}*

Nomenclature

ΔP	General pressure difference
ΔP_{DV}	Pressure drop across te directional valve
ρ_f / ρ_w	Fluid density / water density
A	General area
A_1	Projected area at the upstream side of the spool
A_{12}	Fixed restricted area between upstream side of the spool and the spring chamber
A_2	Projected area at the downstream side of the spool
A_{23}	Variable restricted area at the entrance of the surge chamber
A_3	Spool face projected area when the spool advancing in the inlet of the surge chamber
A_{34}	Fixed restricted area at the outlet of the surge chamber
A_{th}	Throttling area of the tank line throttle
B	Fluid bulk Modulus
C_d	Discharge coefficient
D_{SL1}	Sleeve geometrical diameter
D_{SL2}	Sleeve geometrical diameter
D_{SL3}	Sleeve geometrical diameter
D_{SP1}	Spool geometrical diameter
D_{SP2}	Spool geometrical diameter
D_{SP3}	Spool geometrical diameter
D_{SP4}	Spool geometrical diameter
D_{SP5}	Spool geometrical diameter
D_{th}	Throttling diameter of the tank line throttle
k_f	Viscous friction coefficient
k_x	Spring stiffness
L_{SL1}	Sleeve geometrical length
L_{SL2}	Sleeve geometrical length
L_{SL3}	Sleeve geometrical length
L_{SL4}	Sleeve geometrical length
L_{SP1}	Spool geometrical length

L_{SP2}	Spool geometrical length
L_{SP3}	Spool geometrical length
L_{SP4}	Spool geometrical length
M_{sp}	Spool mass
OV	Relief valve % pressure override
P_1 (P_{pump})	Pump pressure
P_2	Safety valve spring chamber pressure
P_3	Safety valve surge chamber pressure
P_4 (P_{load})	Load pressure including pressure drop across the directional valve and return line pressure
P_5 (P_{pump})	Back pressure before tank line throttle
P_{CO}	Relief valve dead head pressure
P_{CR}	Relief valve cracking pressure
P_{EL}	Pressure equivalent to the external load
Q	General flow rate
Q_{34}	Flow from surge chamber to the system
Q_p	Flow source (from the pump)
Q_{RV}	Flow through the relief valve
Q_{SV}	Flow into surge chamber from the spring chamber
Q_{th}	Flow through tank line throttle valve
S (S_{max})	Instantaneous (maximum) spool displacement
S'	Spool velocity
S''	Spool acceleration
SG	Fluid specific gravity
V_s	Volume of the surge chamber
X_{rmax}	Spring maximum (Free) length

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