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THEORETICAL AND EXPERIMENTAL STUDY OF THE PRESSURE BEHAVIOR ON HYDRAULIC POSITIONING SYSTEMS

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This paper deals of the theoretical and experimental study of the dynamic behavior of the cylinder chambers pressures on loaded hydraulic positioning systems, focusing on helping in the choice of the match of symmetric and asymmetric valves with symmetric and asymmetric cylinders. This study is based on the behavioral analysis of the critical pressure conditions in the cylinder chambers, with different loads, analyzing the occurrence of critical points that can cause damages to the system. A noticeable fact related to the configuration of hydraulic positioning systems is the unexplained combination of symmetric valves with asymmetric cylinders in several industrial and academic applications, being this match not recommended by the manufacturers of hydraulic equipment. In this context, is justified a research about the combination of valves and cylinders, being this subject not much explored or comprehended, and the lack of data to support that choice ends up in difficulties to the design of hydraulic positioning systems. The experiments conducted in this research allowed the observation of the behavior of different matches and led to the comparison between them and their advantages and disadvantages. The occurrence of pressures spikes below zero and above the supply pressure were events observed as prejudicial to the system and the first, that may lead to cavitation, marked the maximum load permissible for each match, giving the designer practical results that point out the best combination for a certain load, when applied to the studied equipment. It was noticed that maintaining the same system parameters, system effective moving mass and load, a simple change in the match of valves and cylinders can generate or not critical points. The results obtained through this study can lead to the identification of the static and dynamic characteristics required for the hydraulic positioning system components in several configurations. Moreover, it also allowed a better understanding of the effects of limit pressures in some configurations of valve and cylinder under different loads, supporting the choice of the most adequate alternative for the electro-hydraulic positioning systems design. This paper also presents the results of the comparison between symmetric and assymetric valves when associated to an assymetric cylinder.

***Keywords**: Hydraulic positioning system, Cavitation, Pressure behavior*

1. INTRODUCTION

Hydraulic systems as they are known nowadays have been used in work industry since the beginning of the industrial revolution, but a lot of mechanical and electronic technologies have been aggregated to the modern hydraulic components. Moreover, the intensive use added to industrial and academic studies yielded a better understanding of their behavior. However, in technical and academic literature the limits of interconnection of valves and cylinders concerning their area symmetry are not well known. This paper studies the identification of such limits and reports representative phenomena, aiding the designer on picking the most adequate match of valve and cylinder for the development of systems of better performance.

There is a wide applicability of hydraulic systems that use proportional and servo valves. These have been mainly destined to high power control where reliability, speed and efficiency are required. Nevertheless, the problems in sizing and selecting the actuator system (valve and cylinder), in the current stage of technological research, is not adequately solved, especially concerning the preliminary choice of components.

A noticeable fact related to the configuration of hydraulic positioning systems is the combination of symmetric valves with asymmetric cylinders in several academic researches, as in Virvalo (2001), that deals of the closed loop hardware environment for design, tuning and test of servo control system; in Kim and Won (2001), that deal with modeling and control of a hydraulic active suspension; in Habibi et al. (1994), that carry out an analysis of a hydraulic actuator for multivariable control of industrial robot. These mentioned researches such as many others use the combination above mentioned and get feasible results in position control, even though the technical literature indicates this match as not appropriate. So, it is noticeable that this subject must be more deeply investigated and that limits where it is possible or not to use a given combination of cylinder and valve should be identified.

In this context, a broad study about the combination of valve and cylinder was carried out, combining symmetric and asymmetric valves with asymmetric and symmetric cylinders. For each match the validation of the model comparing simulation results with those from the experimental setup was made at first, and then the results of varying the spring rate and the spring pre-load displacement were analyzed, searching for the load limit in each situation as a way to contribute to form the design criteria for the valve and cylinder match, discovering the influence of the pressures in a certain configuration (SZPAK, 2008).

This paper presents theoretical and experimental analysis of critical operational conditions of the combination of symmetric and asymmetric valves with asymmetric cylinders comprising a hydraulic position control system. To carry out this analysis, a MATLAB/Simulink block diagram was developed to describe the model of the hydraulic positioning system behavior under certain loads, generating results that were suitable for the analysis of critical conditions concerning the pressures in the cylinder chambers. The model was validated through the comparison between experimental and simulated results. With this procedure, data regarding to the choice of the combination of valves and cylinders was obtained, allowing the achievement of a better system performance. To present the matter studied in this research, this paper presents and analyzes the results of the experiments conducted combining the asymmetric cylinder with the asymmetric and symmetric valves and compares these results side by side. The results presented are theoretical and experimental and more details regarding to the models can be found in SZPAK (2008).

2. EXPERIMENTAL SETUP

The experimental equipment was designed and constructed by LASHIP, Laboratory of Hydraulic and Pneumatic Systems (LASHIP) of the Mechanical Engineering Department of the Federal University of Santa Catarina with the purpose of studying and designing proportional hydraulic systems. It consists of: HPCU (Hydraulic Power and Conditioning Unit); Data Acquisition System VXI and Computers with the software Labview. At the workstation are available: asymmetric and symmetric cylinders; asymmetric and symmetric valves; pressure and position transducers and a loading system that allows the application of springs with different pre-load displacements and spring rates. The workstation is shown in Fig. 1, with one of the configurations studied in this paper, composed by a double action asymmetric cylinder, an asymmetric proportional valve, pressure transmitters, a pressure reducing valve and the HPCU.

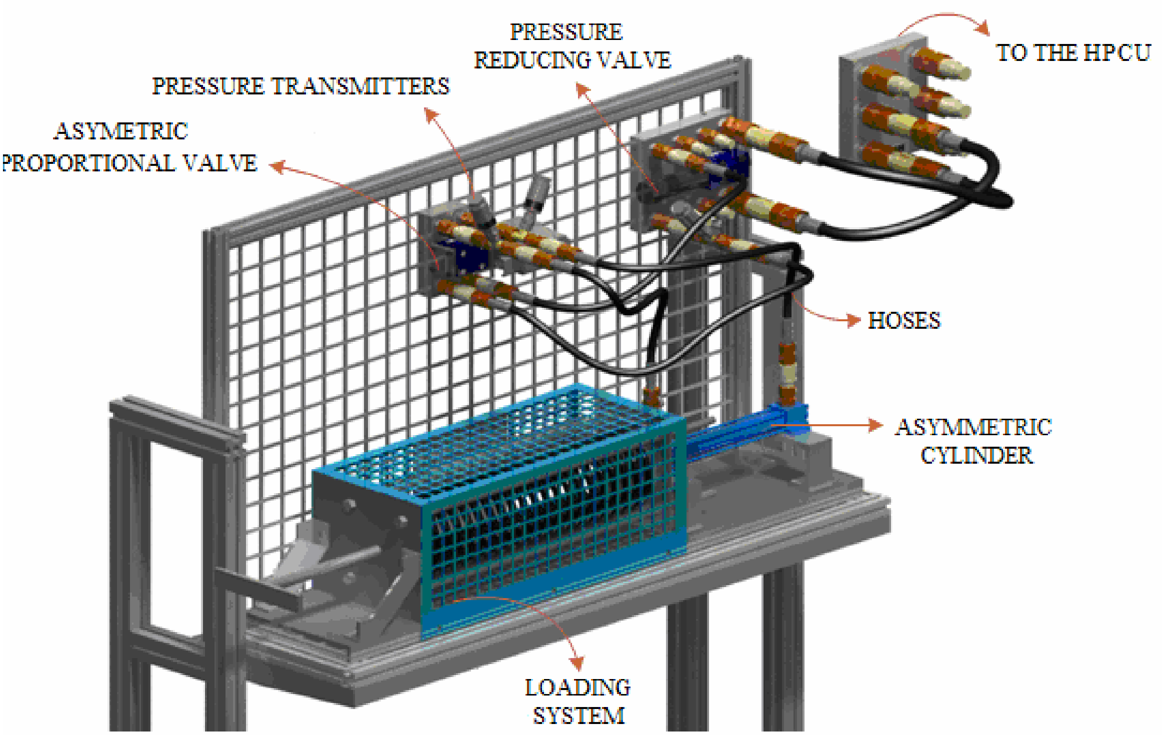


Figure 1. Work apparatus with a combination of asymmetric cylinder (AC) and asymmetric valve (AV).

The HPCU was configured to operate with a maximum pressure of $8\,MPa$; temperature of $45\,°C$; and a flow rate of approximately 30 lpm plus a $50\,l$ accumulator. The directional asymmetric proportional valve (AV) used was the Bosch Rexroth 4WREE 6 E1-08-22, with a metering orifice area ratio of $r_A^V = 2$, with a nominal voltage of $\pm 10\ V$; nominal flow rate of 8 lpm; internal leakage of 500 $cm^3/\min$; natural frequency of 439.8 rad/s and damping ratio of 0.8. With these values it was possible to obtain the parameters necessary to the model, obtaining the flow rate coefficient K_V of the valve, that equals to $1.333\text{x}10^{-7} m^3/s.\sqrt{Pa}$, and the internal leakage coefficient K_{Sim} of the valve, that equals to $7.0\text{x}10^{-10} m^3/s.\sqrt{Pa}$. The directional symmetric proportional valve (SV) used was the Bosch Rexroth 4WRPEH 6

C3B12L, with a metering orifice area ratio of r_A^V=1,with a nominal voltage of $\pm 10 V$; nominal flow rate of $12\, lpm$ @ 10 bar; internal leakage of $300\, cm^3/\min$; natural frequency of $439.8\, rad/s$ and damping ratio of 0.7. With these values it was possible to obtain the parameters necessary to the model, obtaining the flow rate coefficient of the valve that is 7.559x10^{-8} $m^3/s.\sqrt{Pa}$ and the internal leakage coefficient of 3.73x10^{-10} $m^3/s.\sqrt{Pa}$. The asymmetric cylinder (AC) used CDT3MS22518200 has a piston of $200\, mm$ of length; an areas ratio of r_A^A=2, where $r_A^A = A_1^A / A_2^A$ being A_1^A=4.91x10^{-4} m^2 and A_2^A=2.37x10^{-4} m^2. More information on the equipment, how the values were obtained and the valve model using K_V and K_{Sin} parameters can be found in SZPAK (2008) and in INÁCIO PEREIRA (2006).

The model used consisted in a set of equations that represent the dynamic behavior of the electrical, mechanical and hydraulic subsystems, combined through logical and mathematical operations according to the characteristics of its components.

Figure 2 shows the schematic diagram of the hydraulic positioning system used in the studies.

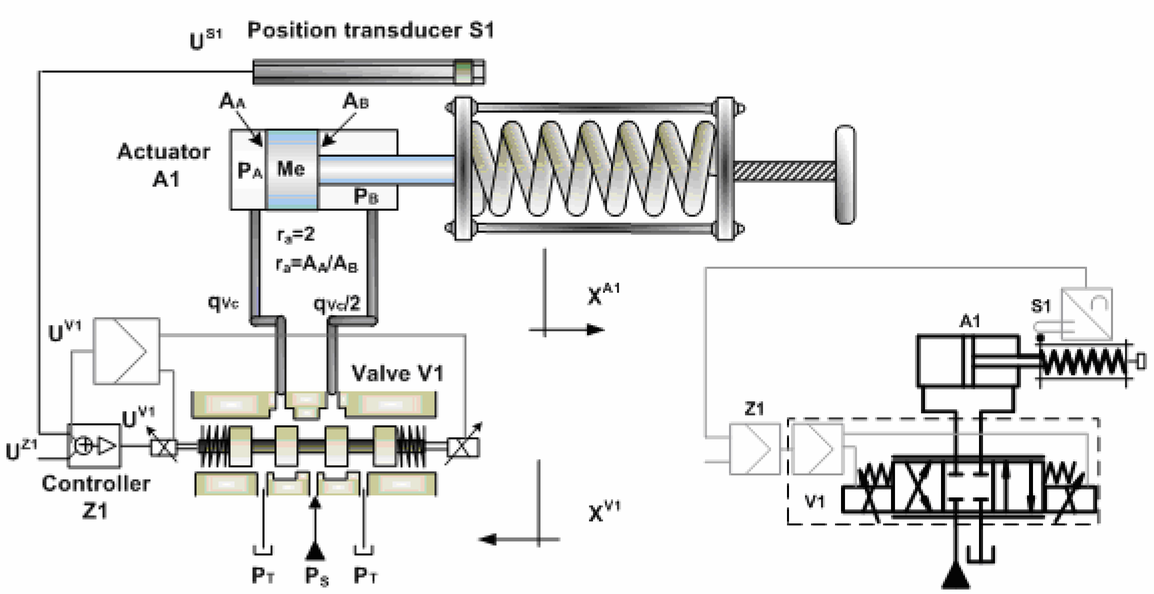


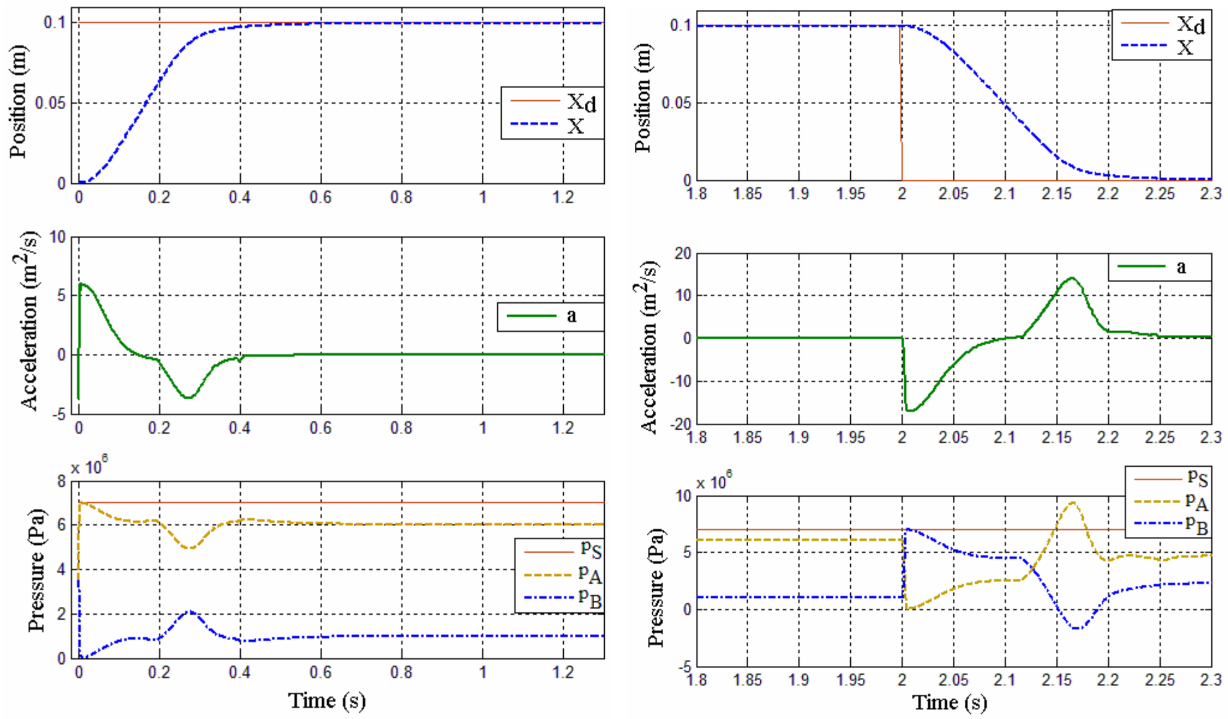
Figure 2. Hydraulic positioning system composed by an asymmetric valve (AV) and an asymmetric cylinder (AC).

3. CRITICAL PRESSURE CONDITIONS IN THE CYLINDER CHAMBERS

Through several simulations in Matlab/Simulink for the different configurations of valves and cylinder, it was detected that the critical conditions happen in two specific situations that occur when the acceleration reaches its maximum value both in the forward movement and in the reverse movement of the cylinder. Depending on the force applied and the configuration of the system, the pressure in the cylinder chambers can reach values above the supply pressure (p_S) and below zero (manometric pressure) creating conditions for the occurrence of cavitation in the cylinder chamber. Parameters like mass, spring rate and pre-load force from the spring have a direct influence on the critical conditions occurrence.

The maximum positive acceleration ($a_{MAX}(+)$) occurs at the beginning of the forward movement and at the end of the reverse movement of the cylinder, where p_B reaches its minimum value, many times close to or below zero, and p_A reaches its maximum value, sometimes overcoming the supply pressure p_S. When the maximum negative acceleration ($a_{MAX}(-)$) occurs, an inversion of the phenomenon described above is observed, where p_B has its maximum value, sometimes overcoming p_S, and p_A reaches pressures next to or below zero.

Figure 3 shows an example of the pressure behavior in the cylinder chambers in the forward movement of the cylinder for a better comprehension of the system behavior. In this simulation, there is a combination of a symmetric valve with a symmetric cylinder with a mass of 156 kg and no pre-load displacement on the spring. The supply pressure is 7 MPa, the spring rate is 5982 N/m, the controller proportional gain is 5 and the amplitude of the command signal is 5 V. The graphs of Fig.3 show the result to an step signal input, besides the pressure peaks in p_A and p_B.



(a) (b)

Figure 3. Example of the system response for a step input (a) forward movement of the cylinder, (b) reverse movement of the cylinder.

In Fig. 3a and Fig. 3b one can observe the following behavior for the pressures p_A and p_B:

-Forward movement:

- p_A is maximum, when the cylinder has $a_{MAX}(+)$;
- p_B is minimum, when the cylinder has $a_{MAX}(+)$;
- p_A is minimum, when the cylinder has $a_{MAX}(-)$;
- p_B is maximum, when the cylinder has $a_{MAX}(-)$.

-Reverse movement:

- p_A is minimum, when the cylinder has $a_{MAX}(-)$;
- p_B is maximum, when the cylinder has $a_{MAX}(-)$;
- p_A is maximum, when the cylinder has $a_{MAX}(+)$;
- p_B is minimum, when the cylinder has $a_{MAX}(+)$.

On the sequence the study of the different matches between symmetric and asymmetric valves with asymmetric cylinders is presented, observing how do the chamber pressures behave for different values of load mass and external force applied to the system, verifying in which configurations the pressures above the supply pressure or below zero can occur.

3.1. Symmetric valve (SV) with double action asymmetric cylinder (AC)

Figure 4 shows the validation of the model for SV+AC through the comparison of the real system response with the non-linear model results with proportional gain of 5, step input of $8V$ and without the spring load. It shows the behavior of pressures p_A and p_B where one can observe that the model makes a good description of the real behavior, mainly regarding to the spike values. As shown in SZPAK (2008) the theoretical and experimental position results are correlated proving the model representativeness.

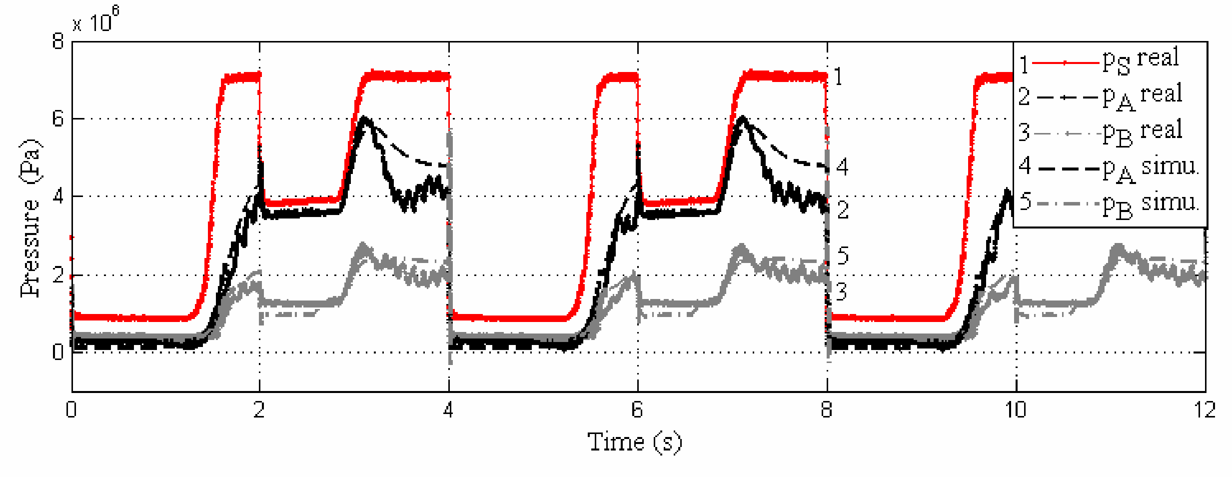


Figure 4. Theoretical and experimental results of the AC+SV system without spring.

In Fig. 5 a simulation with a load mass of 300 kg, proportional gain of 5, input signal with an amplitude of 5 V, and without spring pre-load displacement; one can observe the effect of springs with different stiffnesses. Springs with rates equal to 2.67x10^{-3}, 6.10x10^{-3} and 1.64x10^{-2} were used. As the spring stiffness increases, the pressure peaks and the probability of occurrence of critical points decrease. Figure 5d is a zoom on the Fig. 5b between 0 and 0.6 showing only the pressures in the chamber A of the cylinder and shows that negative pressure peaks can occur with lower spring loads.

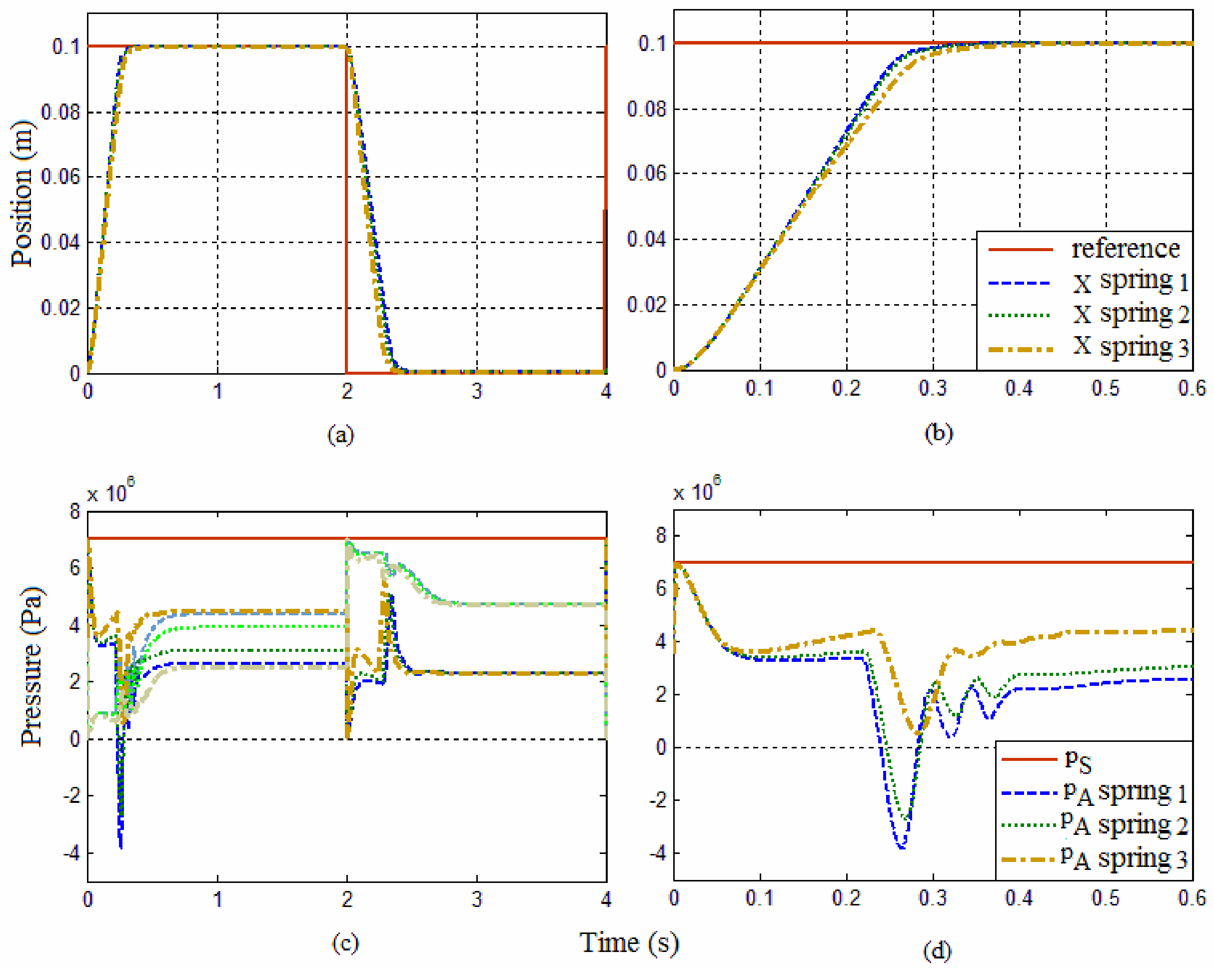
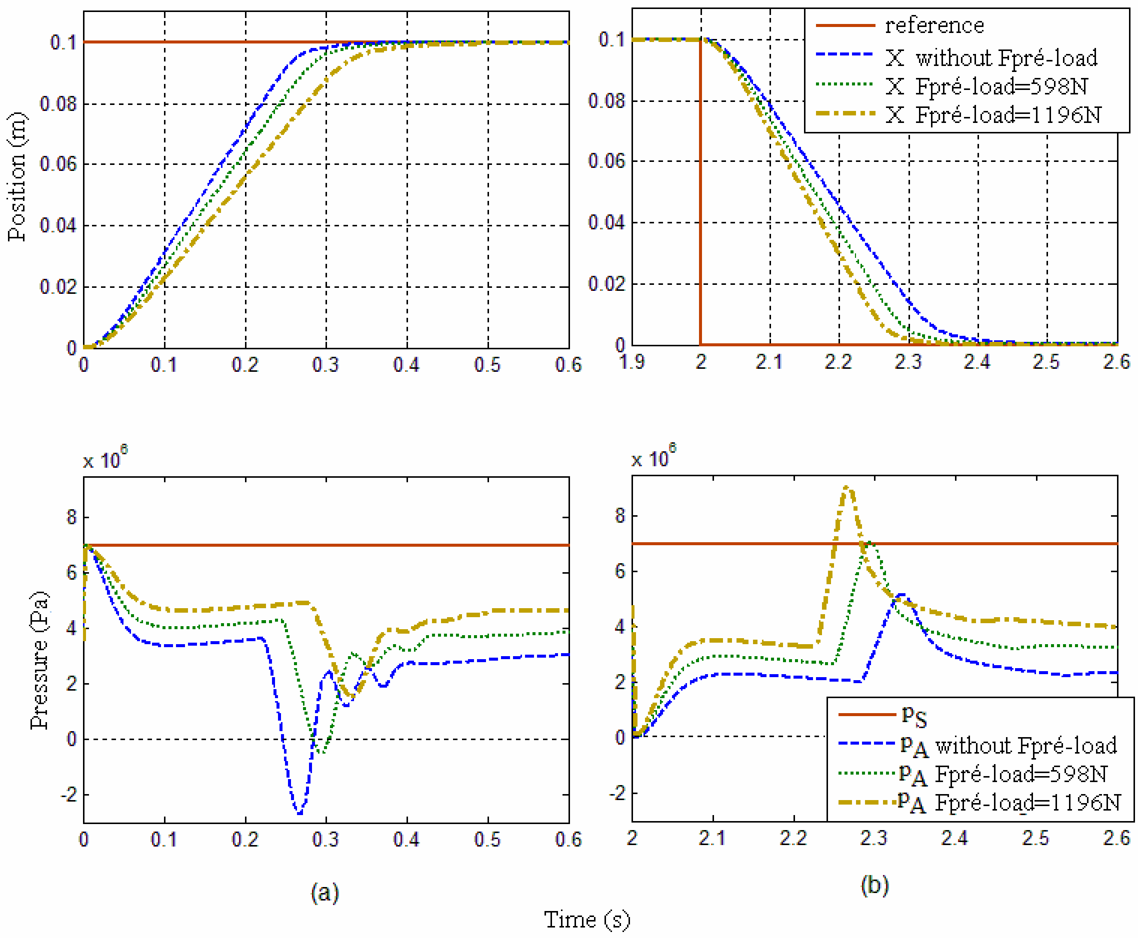


Figure 5. Theoretical results for the SV+AC system with the three different springs. (a) Position, (b) Pressure, (c) Zoom of the position in the cylinder forward movement, (d) Pressure A in the forward movement of the cylinder.

The effect caused by increasing the spring pre-load force (fc) was also analyzed. In Fig.6, the AC+SV system with proportional gain of 5, amplitude of 5 V and spring rate of 5982 N/m, one can notice that when the spring pre-load force (fc) is increased, the pressure p_A tends to overcome p_S in the cylinder reverse movement and reach below zero in the forward movement of the cylinder.



(a)

(b)

Figure 6. Theoretical results for the SV+AC. (a) Position and pressure in the forward movement of the cylinder (b) Position and pressure in the reverse movement of the cylinder.

It is important to observe that the model does not consider the occurrence of cavitation, which means the fluid pressure may be below its vapor pressure without phase change. On a real system cavitation would probably be occurring under these conditions.

Table 1. Maximum mass values without possibility of cavitation occurrence. Results from simulations with symmetric valve and asymmetric cylinder.

SIMULATED RESULTS SYMMETRIC VALVE + ASYMMETRIC CYLINDER			
Spring 1	$F_{PreC} = \mathbf{0}\ N$	$F_{PreC} = \mathbf{216.84}\ N$	$F_{PreC} = \mathbf{523.68}\ N$
$M_1\ [kg]$	277	351	434
Spring 2	$F_{PreC} = \mathbf{0}\ N$	$F_{PreC} = \mathbf{598.21}\ N$	$F_{PreC} = \mathbf{1196.42}\ N$
$M_1\ [kg]$	354	561	875
Spring 3	$F_{PreC} = \mathbf{0}\ N$	$F_{PreC} = \mathbf{1608.3}\ N$	$F_{PreC} = \mathbf{3216.6}\ N$
$M_1\ [kg]$	635	2116	-

In Tab. 1 are compiled the results obtained for this match of valve and cylinder, where one is able to observe that, increasing the spring pre-load displacement and stiffness, also increases the load the system can support without the occurrence of critical points.

Therefore, it is noticed that this combination of valve and cylinder offers the system good operational conditions avoiding the occurrence of critical pressure points for a wider variety of springs with different stiffness coefficients and pre-loads displacements, and for higher mass values when compared to other combinations of cylinder and valve. It is shown that with the load force increase (changing the spring rate and its pre-load displacement), increases the mass that can be moved by the system. As it can be seen in Fig. 2, the direction of the spring force is opposite to the forward movement of the cylinder and therefore, the higher the spring force, the more the pressures will be further away from the situation of cavitation. For some sets of parameters, simulation errors occurred in MATLAB/SIMULINK due to numeric instability. It was found experimentally that for those values the actuator does not move due to the excessive load applied to the system. On those cases, no value was registered in the tables.

3.2. Asymmetric valve with double action asymmetric cylinder

Figure 7 below shows the validation of the model for the match AV+AC through the comparison of the real system values with the non-linear simulated model with proportional gain of 5, input signal with an amplitude of 8 V, spring rate of 16083 N/m and without pre-load displacement. It shows the pressures p_A and p_B in both situations, verifying the correspondence between the results generated through simulation and the ones observed in the real system.

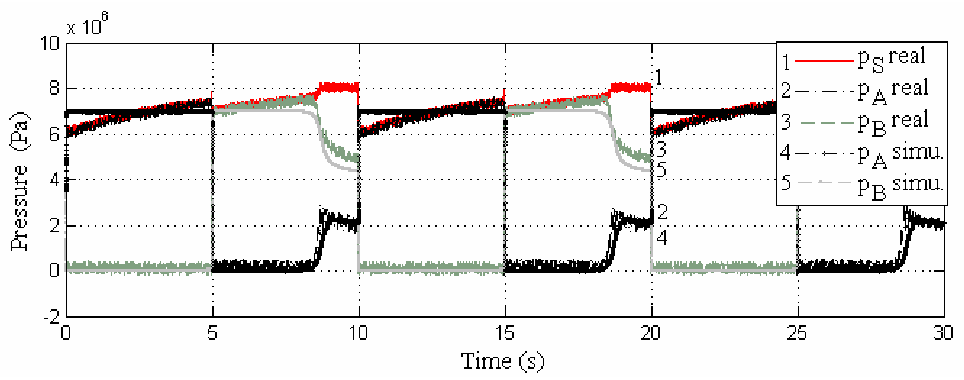


Figure 7. Graph of the real system results x non-linear simulated model with AC+AV.

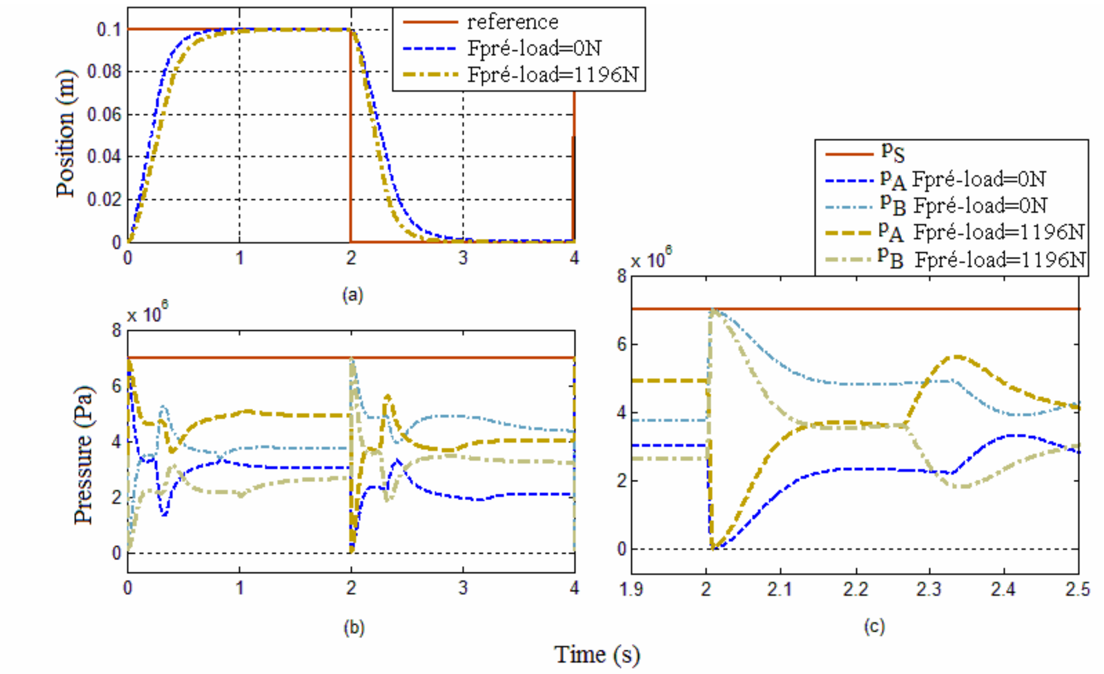


Figure 8. Graph of the non-linear model for AC + AV (a) Position (b) Pressure (c) Zoom of the pressure between 1.9 and 2.5 seconds.

In the analysis of the pre-load effect on this combination of valve and cylinder, Fig. 8 shows how adding pre-load forces makes the system pressures more oscillatory, and how, depending on the pre-load force value, the system can reach pressures close to or below zero and above the supply pressure. In this case the controller gain is 5, the amplitude of the sinusoidal input signal is 5 V and the spring rate is 5982 N/m.

In Tab.2 are the results obtained for the match of a symmetric valve and an asymmetric cylinder.

Table 2. Maximum mass values that may be used without the possibility of cavitation occurrence. Results from simulations with an asymmetric valve associated to an asymmetric cylinder.

SIMULATED RESULTS ASYMMETRIC VALVE + ASYMMETRIC CYLINDER			
Spring 1	$F_{\Pr eC} = \mathbf{0}\ N$	$F_{\Pr eC} = \mathbf{216.84}\ N$	$F_{\Pr eC} = \mathbf{523.68}\ N$
$M_1\ [kg]$	1024	1229	1471
Spring 2	$F_{\Pr eC} = \mathbf{0}\ N$	$F_{\Pr eC} = \mathbf{598.21}\ N$	$F_{\Pr eC} = \mathbf{1196.42}\ N$
$M_1\ [kg]$	1227	1848	-
Spring 3	$F_{\Pr eC} = \mathbf{0}\ N$	$F_{\Pr eC} = \mathbf{1608.3}\ N$	$F_{\Pr eC} = \mathbf{3216.6}\ N$
$M_1\ [kg]$	1975	-	-

For this match, the mass values supported without occurrence of critical points are even higher and also increase as does the spring pre load force, but there are some pre-loads forces that cannot be used or the actuator will not be able to move.

4. COMPARISON BETWEEN SV+AC AND AV+AV SYSTEMS

After the individual behavior analysis, follows the comparison between them for the same set of parameters to verify which one is the most adequate choice in each case.

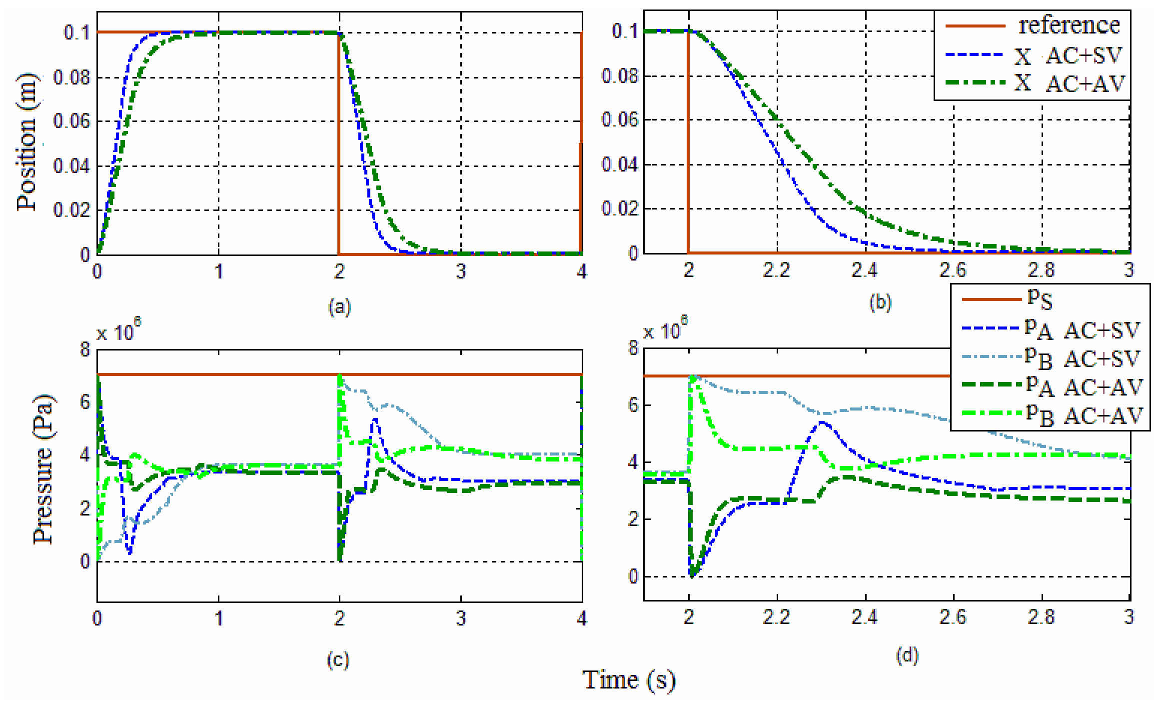


Figure 9. Comparison of the non-linear model answer for SV+AC x AV+AC. (a) Position, (b) Position reverse movement, (c) Pressures and (d) Pressure in the reverse movement.

Figure 9 shows the comparison of the position behavior in the forward and reverse movements of a symmetric valve and an asymmetric valve commanded by an asymmetric cylinder, both with a 400 kg mass, with spring 1 and with a pre-load force of 524 N. It is noticed that the response of the SV+AC setup is faster both in forward and reverse movements than the AV+AC setup (Fig.9a). However, when comparing the chamber pressures of the systems commanded by an AV and by a SV, one can notice more oscillation in pressures p_A and p_B during the transient regime with the SV+AC setup, as shows Fig. 9c and in Fig 9d. When the controller gain is adjusted to make the AC+AV setup meet the position result of the SV+AC match, the pressure peaks increase, but these peaks are still smaller than the ones observed in the SV+AC setup.

Figure 10 shows that, for the same operational condition but without the pre-load force, p_A reaches values below zero in the forward movement of the cylinder with the use of a SV, creating the possibility of occurrence of cavitation as seen in Fig. 10a. Another factor seen are the oscillation of pressures p_A and p_B that are quite superior in the system with SV, as seen in Fig 10a and Fig 10b. So it is verified, based on the pressure behavior that, for a 400 kg mass and without any pre-load force, the system could operate without the occurrence of critical points only with AV. Adjusting the controller gain of AC+AV to obtain the same position result of AC+SV one can observe the possibility of occurrence of pressures below zero in this system as well, but still with smaller pressure peaks than in the AC+SV setup. In steady state there are no significant differences between these conceptions.

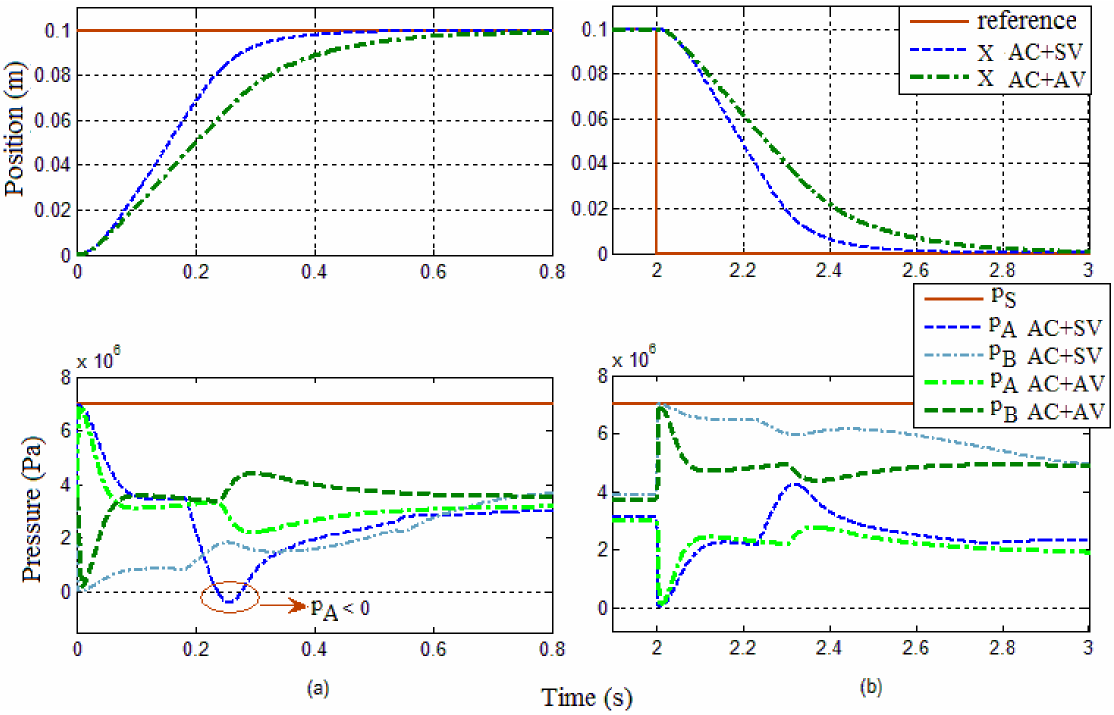


Figure 10. Comparison of the non-linear model answer for AC+SV x AC+AV. (a) Forward movement the cylinder and (b) Reverse movement of the cylinder.

For the same system parameters, the simple change in the match between valves and cylinder can create or not occurrence of critical points. Table 3 shows the maximum possible masses for the use of the matches previously mentioned of valves and cylinders without the occurrence of critical points with pressures below zero that can cause cavitation. On the top of the columns is indicated the spring used and the pre-load force applied on by it, and in the boxes are written the maximum values of mass in kg for which there is no possibility of occurrence of cavitation.

The imposed limits for the use of a certain conception are results from the observation of the pressures behavior in each simulation. In each situation it was pursued the limit value of mass that could be used without the occurrence of pressures below zero in the cylinder chamber, that could cause critical points.

Table 3. Maximum values of mass that can be used without creating the possibility of cavitation for different conceptions of valves and cylinders.

VALOR MAXIMUM MASS VALUE FOR CONCEPTIONS OF VALVES AND CYLINDERS										
		Spring 1			**Spring 2**			**Spring 3**		
F_{PreC} $[N]$		**0**	**217**	**524**	**0**	**598**	**1196**	**0**	**1608**	**3217**
SV+AC	M_1 $[kg]$	277	351	434	354	561	875	635	2116	-
AV+AC	M_1 $[kg]$	1024	1229	1471	1227	1848	-	1227	1848	-

5. CONCLUSIONS

The comparisons between simulations with the same parameters with different conceptions showed even more clearly that care is necessary to make a good choice once that, with little change of some parameters such as pre-load force and spring rate, adequate setups may turn into problematic ones.

The lack of data to aid on the correct choice of the match between valve and cylinder can create difficulties for the design of hydraulic positioning systems. It was to provide some support for such choice that this research was conducted.

Another fact noticed is that the conception that reached the best results in hydraulic positioning systems was the AV+AC match, that allowed, in almost all studied situations of pre-load force, the use of masses above one ton, in some cases almost reaching two tons, resulting in a stronger system.

This theoretical and experimental results lead to a better knowledge of the possible uses of valves and cylinders configurations under different loads, providing grounds for more adequate choices for the design of eletro-hydraulic positioning systems. Other combinations were studied and other comparisons were made and may be consulted in Szpak (2008).

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