Abstract: The present work is a study of the effects of air in hydraulic control system. The study is based on models of: the mixture of air with hydraulic fluid, the servo hydraulic control system and experimental identification. The air mixture model is to evaluate the impact on the fluid properties, the hydraulic system’s one is to assess the impact of changes in the fluid properties in the system performance and identification to estimate the amount of air in the system.

Keywords: hydraulic, identification, modeling.

1. PURPOSE

The hydraulic systems are broadly used in the industry due to its ability to adapt to a wide range of purposes and its proven robustness. Pneumatic systems have quite the same features but the transfer media (gas) increases the difficulties to tune the system to the requirements due its compressibility. Both types of system had completely different kinds of implementations and tuning due to behavior of the medias (gas – compressible and fluid - non compressible) even with almost the same utilization. The present work study effects of the gas (compressible) in systems designed to hydraulic fluids (non-compressible).

Jinghong et al [1] in 1994 made a study on how the effective bulk modulus varies with pressure, their study had almost the same motivation but the focus was only on the media properties.

Wang et al [2] in 2008 proposed a method to control the bulk modulus with a vacuum generator. The apparatus used and the method were just for new design, their focus was mainly to increase the bulk modulus, not on how to deal with low bulk modulus media.

Akkaya [3] in 2006 made studies on how the bulk modulus interferes on the performance of a hydraulic transmission. His study was based on models and simulation of the system. He shows that introducing variations on the media some sort of controllers (PID) is not able to stabilize the system.

Cho et al [4] in 2001 proposed a method of estimation of air content in the oil media from experiment. The estimation was from the comparison of pressures upstream and downstream of automatic transmission.

This work is motivated from the requirements of recent hydraulic control systems and from the difficulty to determine the quality of bleeding maintenance procedures.

The focus is to evaluate the effect of the air in servo valve control hydraulic system performance and verify this method in real application data.

2. METHODS

The study is based on models of: the mixture of air with hydraulic fluid, the servo hydraulic control system and experimental identification. The air mixture model was to evaluate the impact on the fluid properties, the hydraulic system’s one was to assess the impact of changes in the fluid properties in the system performance and identification to verify the performance with entrapped air.

System Description

All evaluations are on a dual hydraulic stage servo valve with a flapper/nozzle in the first stage and 3 way spool valve. The valve controls the pressure on set of 5 pistons actuators from 0 to 20.7 MPa (3000psi). The pistons have a return spring that is fully extended with 1.2 MPa (170psi). The Figure 1 shows a schematic of the valve.

The control valve and actuator are modeled from basic physics equations as the flow equations and Newton laws. Linearizing those equations considering only the first order of the Taylor expansion series and collecting the higher order derivative we have:

\[ \dot{x}_f = \frac{1}{I} \left( K_m \cdot \dot{i}_m - B_f \cdot \dot{x}_f - K_f \cdot x_f \right) \]  

\[ \dot{p}_1 = \frac{\rho_1}{V_1} \left( A_2 \cdot x_0 + K_s \cdot p_1 + K_{f2} \cdot x_f \right) \]
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Where \( K_1 = \frac{\partial \xi}{\partial \frac{\partial v}{\partial}} \) and \( K_{11} = \frac{\partial \xi}{\partial x_f} \)

\[
\dot{P}_2 = \frac{\beta_f}{V_f} (A_2 \dot{x}_f + K_2 P_2 + K_{12} x_f) \tag{3}
\]

Where \( K_2 = \frac{\partial P_2}{\partial x_f} \) and \( K_{12} = \frac{\partial P_2}{\partial x_f} \)

\[
\ddot{x}_f = \frac{1}{M_f} (-A_1 P_1 - P_2 A_2 + A_2 P_2 - C_0 \dot{x}_f - K_0 \dot{x}_f) \tag{4}
\]

\[
\dot{P}_b = \frac{\beta_b}{V_b} (-A_p \dot{x}_b + K_p \dot{x}_b + K_0 \dot{x}_b) \tag{5}
\]

Where \( K_p = \frac{\partial P_b}{\partial x_b} \) and \( K_0 = \frac{\partial P_b}{\partial x_b} \)

\[
\ddot{x}_b = \frac{1}{M_b} (A_p \dot{x}_b - B_p \dot{x}_b - K_0 \dot{x}_b) \tag{6}
\]

The system has only one input, the system current \((i_m)\) and only the piston chamber pressure \((P_b)\) as output characterizing a Single Input Single Output model (SISO). Considering \( x \) and \( u \) states and input vectors respectively where:

\[
x^T = [\dot{x}_f \ x_f \ P_1 \ P_2 \ \dot{x}_v \ x_v \ P_b \ x_p \ x_b] \tag{7}
\]

\[
u = [i_m] \tag{8}
\]

Where:
- \( x_f \): flapper velocity
- \( x_v \): flapper displacement
- \( P_1, P_2 \): chamber pressure (1 and 2)
- \( x_v \): spool velocity
- \( x_p \): spool displacement
- \( P_p \): Piston pressure
- \( i_m \): coil current
- \( x_p \): piston velocity
- \( x_b \): piston displacement

And the state space representation as bellow:

\[
\dot{x} = Ax + Bu \tag{9}
\]

\[
y = Cx + Du \tag{10}
\]

The matrices \( A, B, C \) and \( D \) are from the system states equations are:

\[
A = \begin{bmatrix}
\frac{B_f}{V_f} & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & K_{11} & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & K_{12} & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & -K_b & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & K_b & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix} \tag{10}
\]

\[
B = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix} \tag{11}
\]

\[
C = \begin{bmatrix}
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix} \tag{12}
\]

Experimental Identification

To determine the bulk modulus the lowest order of the system the better. Figure 2 shows the hydraulic diagram of the complete system available.

The simpliest system to identify is the one from PT 3A (or B) to PT 4A (or B). This one should be only a first order system regarding the lines and media stiffness. The system has the transducers described in Table 1.

Table 1 - System Transducers

<table>
<thead>
<tr>
<th>TAG</th>
<th>Description</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>PT 1</td>
<td>System Pressure</td>
<td>(0-5000)+/-93.5</td>
</tr>
<tr>
<td>PT R</td>
<td>Return Pressure</td>
<td>(0-1000)+/-18.7</td>
</tr>
<tr>
<td>PT 2</td>
<td>Piston System Pressure</td>
<td>(0-5000)+/-93.5</td>
</tr>
<tr>
<td>PT 3A</td>
<td>L/H Control Pressure</td>
<td>(0-3000)+/-30.0</td>
</tr>
<tr>
<td>PT 3B</td>
<td>R/H Control Pressure</td>
<td>(0-3000)+/-30.0</td>
</tr>
<tr>
<td>PT 4A</td>
<td>L/H Real Pressure</td>
<td>(0-3000)+/-30.0</td>
</tr>
<tr>
<td>PT 4B</td>
<td>R/H Real Pressure</td>
<td>(0-3000)+/-30.0</td>
</tr>
<tr>
<td>I 1</td>
<td>L/H Valve Current</td>
<td>(0-33)+/-3.0</td>
</tr>
<tr>
<td>I 2</td>
<td>R/H Valve Current</td>
<td>(0-33)+/-3.0</td>
</tr>
<tr>
<td>VP</td>
<td>Hyd Pump Voltage</td>
<td>(0-40)+/-0.1</td>
</tr>
<tr>
<td>IP</td>
<td>Hyd Pump Current</td>
<td>(0-170)+/-1.2</td>
</tr>
</tbody>
</table>

All the pressure transducers are piezoelectric devices installed in a Wheatstone circuit. The voltage and currents are measured in the direct forms.

The transducers provide an analogic signal and its conversion is on an AD Card. The card is set to digitalize the signal with a zero order holder and it applies a low pass filter of fourth order at 50 Hz.

The data acquisition is on 400Hz, system dynamics with natural frequencies over 200Hz (810 rad/s) are not reliable according Nyquist. In order to have a more robust analysis only dynamics less than 51Hz (321 rad/s) are considered.

Previous data shows that the focus of the identification shall be on frequencies in the range until 50Hz. Moreover, frequencies below 0 Hz must not be regarded, as modes of the piston are not assessed due to the extraction of the piston filling and emptying times.

Considering the preceding requirements, after the selection of an operation range without piston stroke, a band pass filter of 0 to 50Hz is desired. As the data acquisition is
on separated channels, a re-sampling is also desired to reduce the phase lag effect.

**Air Level Estimation**

The segregation by the air amount in the system is made by the type of bleeding procedure. It consists in segregate the system due to the expected efficiency of the procedure applied in the system after a component replacement or system opening. Actually three kinds are used after a part replacement or system opening:

1. **No bleeding**: any special procedure to bleed the air.
2. **Complete bleeding**: all the system lines are cycled and all the oil is replaced with an unpressurized external cart with a low-pressure flow.
3. **High pressure bleeding**: the oil is replaced with high-pressure flow.

This method assumes that the procedures 1 to 3 are listed in the effectiveness order.

**Identification Method**

The system identification process follows the Ljung [5] recommendations after the properly data selection and filtering. The process begins with a simple model and its complexity increases gradually until the desired result is achieved. The main goal is to achieve the required result with the less complex model.

The evaluation of the model representation is made by the difference between the actual results and predicted by the model. FIT function performs this calculation as follows:

\[
FIT = \left(1 - \frac{|Y - \bar{Y}|}{|Y - \bar{Y}|}\right) \times 100
\]

Where \(Y\) is the actual result, \(\bar{Y}\) is the predicted by the model and \(\bar{Y}\) is the average value. The validation is done preferably in another data packet.

The assessment of complexity is given by the diagram of poles and zeros and by de bode diagram. In the diagram of poles and zeros is verified the existence of poles and zeros that are very close and therefore subject to cancellation. In the bode diagram of the model and of the residue model is verified the presence of dynamics not represented by the model.

**Considerations on data selection**

In order to improve the identification performance any non-linearity is desired. As the system is a linearized model, a large variation in any system parameter is also not desired. Therefore, the beginning and the ending of the pressure application have to be avoided.

System supply pressure and other minor effects are not evaluated in this work.

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**Bulk modulus variation**

To evaluate the real stiffness in a hydraulic flow we have to consider not only the fluid bulk modulus but also the effects of the structure or flexible parts and of the free air.

Oil bulk modulus has variation regarding the amount dissolved and entrained air. The more the air fraction in the mixture lowers the bulk modulus. Entrapped air may be considered a Perfect Gas and assuming any heat transfer in the process (adiabatic) the bulk modulus can be calculated as:

\[
\beta_p = \frac{V_g}{V_t} (1.4P)
\]

The previous equation is valid for units on the International Standard. Where \(V_g\) and \(V_t\) are gas and total volumes respectively and \(P\) is the system pressure.

Thin walled vessels, where the inner diameter is almost the outer, like hydraulic tubing have the following bulk modulus:

\[
\beta_t = \frac{T.E}{D}
\]

Where \(E\) is the metal’s young modulus, \(T\) is the wall thickness and \(D\) is the tube diameter. The valve structure is much stiffer than the hydraulic lines so it is not considered. More detail of the previous equation can be found in Merritt (1967) [6]

So the effective bulk modulus can be determined with:

\[
\beta_e = \frac{1}{\beta_g} + \frac{1}{\beta_t} + \frac{1}{\beta_f} (1.4P)
\]

Rewriting the equation (2) with the Laplace transform to put into evidence the chamber pressures modes of vibration we have a first order differential equation as below.

\[
P_1 = \frac{K_1}{\tau_1, s + 1}
\]

Where the time constant \((\tau_1)\) is described below. The relation between the time constant and the bulk modulus is clearly identified.

\[
\tau_1 = \frac{1}{\beta_1} \frac{\nu_1}{\sqrt{2P}} \left(\frac{A_p C_m p}{P_0 - P_t} + \frac{\pi C_d P (\nu_1 + X_1)}{\sqrt{P_1 - P_c}}\right)
\]

**3. RESULTS**

Figure 3 presents the variation of the effective bulk modulus with volume of entrapped air. It is possible to see that with low pressures even small amounts of air have high influence on the effective bulk modulus. On the other side as the fraction of entrapped air increases, the pressure has less weight on the stiffness.

A completely bled system tends to have the oil bulk modulus and the lines contributions are minor because their
value differs in two orders of magnitude from the others. As a rule of thumb, the effective bulk modulus is the gas bulk modulus for that pressure.

Figure 3 presents the bulk modulus variation on chambers from 0 to 100% of the volume of air. It is possible to see the transition between the oil bulk modulus to the air’s one.

Figure 3 - Bulk Modulus Variation with Fraction of Entrapped Air

On small chambers, like in the valves, it is possible to have 100% of entrapped air on the lines and reservoir reasonable values are fairly lower than this.

The single mode evaluation

Considering the whole variation of the bulk modulus presented before we shall have a huge variation in the system performance. Figure 4 shows how the pressure chamber time constant changes with this variation.

Figure 4 - Time Constant ($\tau$) Variation with Bulk Modulus ($\beta$)

As the bulk modulus the time constant has also a variation of several orders of magnitude.

The Sensitivity to the Bulk Modulus

As the model’s parameters are from estimative they are subjected to errors, these are evaluated in this sensitivity analysis. A sensitivity analysis is to identify the effect of these parameter’s errors in the system dynamic. In our case, this analysis focuses on the identification of the affected poles of each parameter.

The evaluation consists in calculate the system eigenvalues (poles) for the values in the expected range of variation. The following pictures are from 200 evaluations of the system poles. The evaluations are from the minimum to the maximum values, each of the evaluation is presented either in: blue, cyan, green yellow and red according Table 2.

Table 2 - Sensitivity Analysis Plot Colors

<table>
<thead>
<tr>
<th>Evaluation Number</th>
<th>Plot Color</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 to 40</td>
<td>Blue</td>
<td>XXX</td>
</tr>
<tr>
<td>41 to 80</td>
<td>Cyan</td>
<td>XXX</td>
</tr>
<tr>
<td>81 to 120</td>
<td>Green</td>
<td>XXX</td>
</tr>
<tr>
<td>121 to 160</td>
<td>Yellow</td>
<td>XXX</td>
</tr>
<tr>
<td>161 to 200</td>
<td>Red</td>
<td>XXX</td>
</tr>
</tbody>
</table>

As stated before, the bulk modulus is the system characteristic that have the highest influence of the amount of air in the system. This also affects the whole system dynamic as shown in the next figures.

On Figure 5, Figure 6 and Figure 7 the bulk modulus of the whole system fluid varies from 0.35 GPa to 1.5 GPa. This affects almost all the system poles mainly in the natural frequency. As the bulk modulus increases, the natural frequency also increases.

The main effect in on the spool’s poles that has the natural frequency more than doubled. Although this variation does not affect the whole system dynamic, as these poles are too fast.

On Figure 6 is detailed the first order pole with the lowest natural frequency, in this case the natural frequency goes from 200 to 900 rad/s this variation (~4x) is almost the same order of the bulk modulus. In this case, it should be the delay due to the compression of the oil in the tubes between the valve and the pistons.

Figure 5 - Pole variation with 0.35 GPa < $\beta$ < 1.5 GPa - High Frequency
Identification Results

The same procedure is applied to generate the data in the tables below. Note that there is a narrow variance between points, regardless of the method of bleeding. All could be represented with only 2 poles and 1 zero ARX model with an efficiency of 90. The natural frequency is around 200 rad/s and the damping factor around 0.20. Even among the sides A and B is not noticed a considerable difference. With the approximation of the system time constant ($\tau$) by the inverse of the natural frequency ($\frac{1}{\omega_n}$) and comparing these results with Figure 4 - Time Constant ($\tau$) Variation with Bulk Modulus ($\beta$) it is possible to affirm the even the simplest procedure achieves a bulk modulus higher than 1 GPa ($\tau = \frac{1}{200} = 0.005s$).

It shows that the even with the simplest bleeding procedure, there is no major effects on performance regarding the bulk modulus in this condition. This condition means the commanded pressure (300psi) and the transfer function.

Table Error! No text of specified style in document..3 - Result of PT3A/PT4A

<table>
<thead>
<tr>
<th>#</th>
<th>Bleeding</th>
<th>ARX</th>
<th>FIT</th>
<th>$\omega_n$ (rad/s)</th>
<th>Damp ($\zeta$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>89.4</td>
<td>202</td>
<td>0.13</td>
</tr>
<tr>
<td>2</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>93.1</td>
<td>221</td>
<td>0.27</td>
</tr>
<tr>
<td>3</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>93.2</td>
<td>212</td>
<td>0.21</td>
</tr>
<tr>
<td>4</td>
<td>Complete</td>
<td>2 pole 1 zero</td>
<td>89.3</td>
<td>183</td>
<td>0.21</td>
</tr>
<tr>
<td>5</td>
<td>Complete</td>
<td>2 pole 1 zero</td>
<td>89.0</td>
<td>182</td>
<td>0.20</td>
</tr>
<tr>
<td>6</td>
<td>High Pressure</td>
<td>2 pole 1 zero</td>
<td>92.5</td>
<td>206</td>
<td>0.25</td>
</tr>
<tr>
<td>7</td>
<td>High Pressure</td>
<td>2 pole 1 zero</td>
<td>90.7</td>
<td>203</td>
<td>0.22</td>
</tr>
</tbody>
</table>

Table Error! No text of specified style in document..4 - Result of PT3B/PT4B

<table>
<thead>
<tr>
<th>#</th>
<th>Bleeding</th>
<th>ARX</th>
<th>FIT</th>
<th>$\omega_n$ (rad/s)</th>
<th>Damp ($\zeta$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>89.8</td>
<td>198</td>
<td>0.28</td>
</tr>
<tr>
<td>2</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>93.1</td>
<td>224</td>
<td>0.33</td>
</tr>
<tr>
<td>3</td>
<td>Short</td>
<td>2 pole 1 zero</td>
<td>97.4</td>
<td>174</td>
<td>0.17</td>
</tr>
<tr>
<td>4</td>
<td>Complete</td>
<td>2 pole 1 zero</td>
<td>90.6</td>
<td>191</td>
<td>0.32</td>
</tr>
<tr>
<td>5</td>
<td>Complete</td>
<td>2 pole 1 zero</td>
<td>90.2</td>
<td>191</td>
<td>0.32</td>
</tr>
<tr>
<td>6</td>
<td>High Pressure</td>
<td>2 pole 1 zero</td>
<td>88.8</td>
<td>205</td>
<td>0.18</td>
</tr>
<tr>
<td>7</td>
<td>High Pressure</td>
<td>2 pole 1 zero</td>
<td>89.4</td>
<td>204</td>
<td>0.38</td>
</tr>
</tbody>
</table>

4. DISCUSSION

On the air in hydraulic control systems refers mainly to the effects of bulk modulus variation on the system performance.

The amount of air in the system has a direct effect on the system stiffness. The limits for this variation are the oil and the air bulk modulus. A difference in the stiffness is more evident on system with lower pressures when the air is present; as the pressure increases the level of air contamination became imperceptible.

Performance of hydraulic system is much correlated to this design stiffness. System response time may have a huge variation in performance while working with different bulk modulus as presented in the single mode evaluation.

The numerical analysis shows that the system tends to have a reduction of the response time, as it is stiffer. The reduction affects natural frequencies of seven (7) of the system poles (total of 9 in this model). Theoretical variation is of several orders of magnitude in the response time but considering only the slower pole it is of 50%.

The validation chapter has an attempt of determine the system bulk modulus from experimental data. All the points
presented a bulk modulus higher than 1GPa (natural frequency higher than 200 rad/s); it shows that all the bleeding procedures are effective for this line. In order to have a better precision in this determination future works shall focus on other system vibration modes, besides the actuation line, such as the actuator or the spool modes. Another way is by identification of experimental data with higher acquisition rates.

The main conclusion is that the higher the air fraction the lower the bulk modulus and this effect increase as the system operation pressure is decreased. On real system the maintenance does not have a precise control this way is still possible to avoid performance problems increasing the system pressure.

5. REFERENCES


