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THERMAL-HYDRAULIC LUMPED PARAMETERS MODEL OF SERVOVALVE TO PREDICT PERFORMANCE SENSITIVITY TO TEMPERATURE

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ABSTRACT

This communication deals with the thermal-hydraulic modeling and simulation of electrohydraulic actuators to predict the performance sensitivity to temperature. The work reported focuses on the simulation of the servovalve second stage within the AMESim software environment. In the first part, the influence of the static characteristic of this stage on the actuator performance is pointed up. Then, two different models are developed. The fist one is isothermal while the second one considers that all the heat generated by pressure losses is passed to the fluid. The analysis of simulated leakage flow, flow gain and pressure gain displays the influence of temperature and the importance of the type of model. Some extensions of the presented model are proposed to include in detail the heat exchange with ambiance in upcoming work.

KEY WORDS

AMESim, Servovalve, Simulation, Static performance, Thermal effect

NOMENCLATURE

a : Equivalent leakage laminar coefficient due to servovalve and jack ($m^3/s/Pa$)
A : Global leakage coefficient ($m^3/s/Pa$)
e : Position static error (V)
f : Equivalent viscous friction coefficient (Ns/m)
F_b : Break away force of the jack (N)
F_e : External force (N)
G_{sv} : Servovalve transfer function denominator (-)
$\dot{h}$: Enthalpy flow rate (W)
I : Servovalve current (A)
K_A : Amplifier gain (A/V)
K_f : Force factor (m/s/N)
K_l : Position loop gain (1/s)
K_P : Position sensor gain (V/m)
K_{PI} : Pressure/current valve gain (Pa/A)
K_{PX} : Pressure/opening valve gain (Pa/m)
$\overline{K}_{PX}$: non- dimensional pressure gain (-)
K_{QI} : Flow/current valve gain ($m^3/s/A$)
K_{QP} : Flow/pressure valve gain ($m^3/s/Pa$)
K_{QX} : Flow/opening valve gain ($m^3/s/m$)
$\overline{K}_{QX}$: non- dimensional flow gain (-)
m : Mass of moving assembly (Kg)
$\dot{m}$: Mass flow rate (Kg/s)
P_n : Valve pressure drop (Pa)
P_r : Return pressure (Pa)
P_S : Supply pressure (Pa)
Q : Mean valve flow (m^3/s)
Q_l : Spool leakage flow rate (m^3/s)
Q_n : Nominal flow rate (m^3/s)
r_h : Hydraulic stiffness (N/m)

r_l : Load stiffness (N/m)
s : Laplace variable (1/s)
S : Active section of jack (m^2)
V_t : Chamber volume at mid stroke (m^3)
V_X : Tension representative load position (V)
V^*_X : Tension representative load position setpoint (V)
X : Valve spool opening (m)
X_n : Valve spool full stroke (m)
Z_l : Load position (m)
β : Effective Bulk modulus (Pa)
γ : Constant (-)
ΔP : Jack pressure difference (Pa)
ε_d : Position error due to external force (m)
ξ_h : Hydraulic damping ratio (-)
ξ_{sv} : Servovalve damping ratio (-)
τ_e : Equivalent time constant (s)
τ_h : Hydraulic time constant (s)
ω_h : Hydraulic natural pulsation (rad/s)
ω_{sv} : Servovalve pulsation (rad/s)

INTRODUCTION

Many electrohydraulic actuators are used in a wide range of environment temperature. A typical example is found in aerospace where the flight control actuation system has to operate from –40°C to +70°C. The modeling, design and control synthesis of such actuators is widely documented without considering the temperature effect [1] and [2]. The servovalve, being a key component of such actuators, has also generated a lot of research work to allow analysis and performance prediction. The hydraulic characteristic of the main spool valve influences a lot the actuator performance. For this reason, many authors have developed improved models being able to reproduce, by simulation, this characteristic [3], [4], [5] and [6].

On the opposite side, a few references are found dealing with the influence of temperature on actuator performance. Most of the work in this field is performed to simulate the variation of the fluid temperature at the hydraulic system level caused by the energy losses in components. The case of the electro-hydrostatic actuator is studied in [7]. For more details, a lumped parameters model for hydraulic components is presented in [8].

As a result, there is no study that combines the two approaches (closed loop performance vs. temperature).

Therefore, a research work is in progress at INSA Toulouse to develop thermal-hydraulic models of electrohydraulic actuators. As a first step, the servovalve second stage work is reported in this communication, with special consideration to the variation of the gains associated with the hydraulic static characteristic of a spool valve.

IMPORTANCE OF SERVOVALVE STATIC PERFORMANCE

As it was seen before, the servovalve is a major component in electrohydraulic actuators. Moreover, it has a large effect on the control loop performance because of its own performance. Before dealing with it dynamics, it is worth reproducing with accuracy the influence of temperature on the flow/current gain K_{QI} , the flow/pressure gain K_{QP}, the pressure/current gain K_{PI} and the valve leakage flow rate Q_l (the flow/pressure gain can be calculated from the other gains). The importance of these gains is pointed up below, considering the common case of proportional position control (see Appendix).

Stability
The loop gain K_l is limited by stability conditions. When the load stiffness is much smaller than hydraulic one ($r_l << r_h$), it is defined by Eq (1) as:

$$K_l \leq \gamma \xi_h \omega_h \tag{1}$$

The value of constant γ must bet lower than 2, depending on the selected the stability criteria [3]. As a result the stability will be described in Eq (2):

$$K_A \leq \gamma \xi_h \omega_h (1 + Af / S^2) S / K_{QI} K_P \tag{2}$$

As $A = a + K_{QP}$, Eq (2) points up the influence of the flow/pressure gain K_{QP} on the actuator stability. On the other hand, according to Eq (2), the control gain is also influenced by flow/current gain K_{QI} , especially around the null opening region. As there is generally no internal leakage at the jack level, the value of a is mainly due to the servovalve leakage.

Rapidity
The closed loop rapidity is measured by the equivalent closed loop first order time constant τ_e [3], the inverse of K_l . So, the rapidity is also affected by a, K_{QP} and K_{QI}.

Accuracy
The servovalve also contributes to fix the loop precision [2]. Due its pressure/current gain K_{PI}, it introduces a static error e that can be calculated by studying the applied forces on the jack piston at the equilibrium position, as described in Eq (3):

$$e \geq \frac{F_b}{S K_{PI} K_A} \tag{3}$$

Besides, the position error due to the disturbing external force is got from the Appendix as Eq (4):

$$\varepsilon_d = \frac{K_f}{K_l} F_e = \frac{A}{S.K_A.K_{QI}.K_P} F_e \tag{4}$$

The load error is affected by both flow/pressure and flow/current gains. Finally, it is pointed up that the servovalve gains K_{QI}, K_{PI} and K_{QP} have a great influence on the closed loop accuracy.

Consumption

In fact, the ideal servovalve with perfect geometry doesn't exist. At the null opening region, a leakage flow Q_l occurs between the spool valve and the valve body [6]. It is due to the clearance between the valve spool and the valve body that combines with the roundness of the orifices edge. Moreover, the leakage flow has its maximum value at the neutral spool position, which is the average one in position control applications. Therefore, the valve leakage which dominates within the null region is an important contributor to the permanent losses in a hydraulic circuit. Besides, this leakage flow dissipates energy that generates heat, warming the oil.

These considerations point up the importance of the servovalve static gains K_{QI}, K_{PI} , K_{QP} and Q_l on the actuator performance. For this reason, it is of prime importance to develop efficient virtual prototyping simulation models to evaluate how the performance is influenced by temperature conditions.

THERMAL-HYDRAULIC MODELING

In order to develop such servovale models, two approaches are proposed as a first step, based on extreme cases. In the first one, it is supposed that the heat transfer between the fluid and its surroundings is perfect. So, all the heat caused by pressure losses passes towards the ambiance, no heat being exchanged with the hydraulic fluid. Consequently the fluid keeps its temperature constant. Therefore, this case is presented by an isothermal model, called below HM for hydraulic model.

In the second extreme case, we suppose that there is no heat transfer with the entourage. The energy generated by the pressure losses is stored by the fluid, which temperature changes during the work. So, a model with a variable temperature fluid is used, being called THM for thermal-hydraulic model.

At the present stage, the work focuses on the second stage of the servovalve (common cylindrical spool valve) that is the aim in this paper. As a limitation, the model does not reproduce any heat exchange with the bushing (fluid at constant temperature or fluid storing all the energy dissipated by pressure losses). The models are developed within the *AMESim* software [9] that offers interesting facilities and libraries dedicated to thermal-hydraulic simulation.

Typically, the thermal-hydraulic library offers numerous models of components and basic elements.

Opposite to the hydraulic library in which there is neither enthalpy flow rate nor temperature variables, the thermal-hydraulic components exchange hydraulic and thermal power variables, as described in Figure (1):

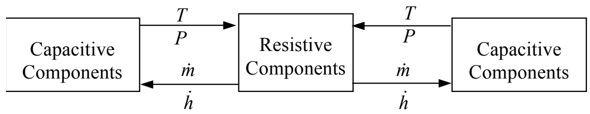

Figure 1. The basic variable flow for thermal hydraulic elements

In capacitive components, as volumes, the temperature and the pressure are computed from the enthalpy and mass flow rates inputs at ports of these components. The resistive ones, as the orifices, are the components in which the enthalpy and mass flow rates are evaluated from the temperatures and pressures inputs at their ports [9]. To avoid any causality conflict, it is necessary to connect them alternatively.

Consequently, the thermal-hydraulic model of the spool valve is built from the standard model of cylindrical spool with slot orifices, as shown on Figure (2). The orifices model includes rounded edges, radial clearance and Reynolds number effects.

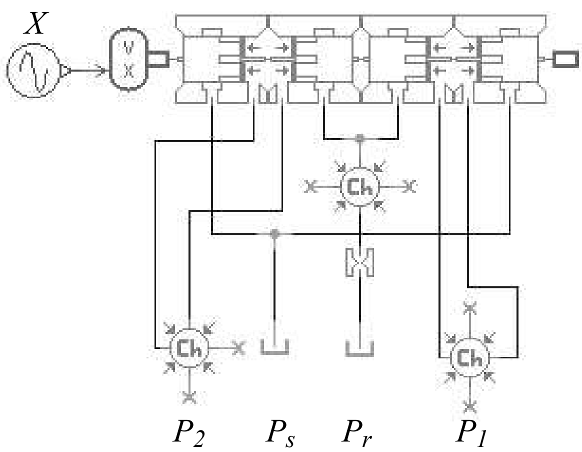

Figure 2. Spool valve model in AMESim for pressure gain and leakage flow characteristics

The servovalve under study is typically a MOOG 76101 delivering 9.8 l/mn at 70 bars pressure drop with the following geometry: full stroke 213µm, diameter 6.9mm, 4 slots of 3 mm, identified rounded edge radius 15µm, identified radial clearance 2µm and identified underlap 15µm.

Furthermore, the two models can be run in parallel to point up the differences between pure hydraulic and thermal-hydraulic models.

SIMULATION AND ANALYSIS

To point up the performance at very low temperature, the models are run at different fluid temperatures leading to a variation of oil viscosity in the range 30-4000 cst.

Leakage flow rate (*Q_l*)

The leakage flow rate can be measured by reproducing in simulation the experimental setup, Figure (3). The total flow rate returned from the valve to the tank is plotted versus the valve opening, as the spool makes a complete cycle around its neutral position at null and with closed use ports.

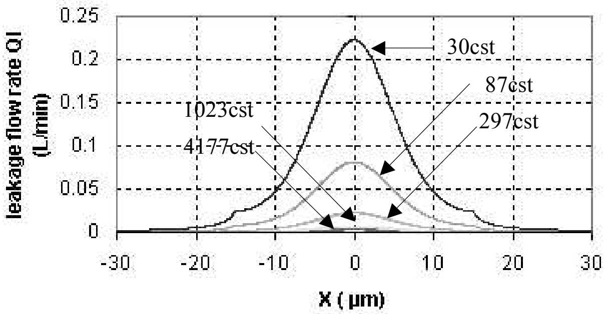

Figure 3. Leakage flow rate with different temperatures (HM and THM similar)

The leakage flow rate values got from the hydraulic and thermal hydraulic models are compared for relative spool openings of 0%, 2.5% and 5% and at the same work temperature. As displayed by Figure (4), the thermal-hydraulic model has a leakage flow slightly higher than the hydraulic model.

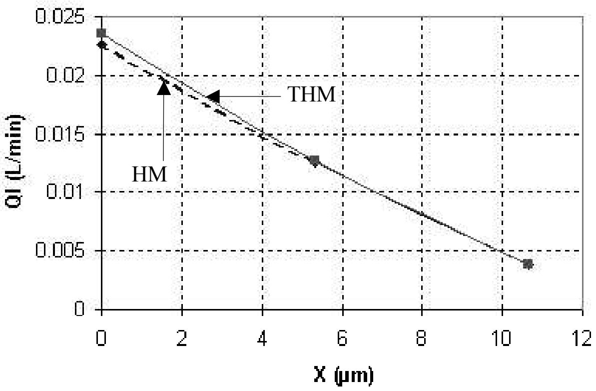

Figure 4. Leakage flow for different spool opening (at 300 cst)

Furthermore, the plot of the leakage flow rate as a function of the temperature for various spool openings, is displayed on Figure (5):

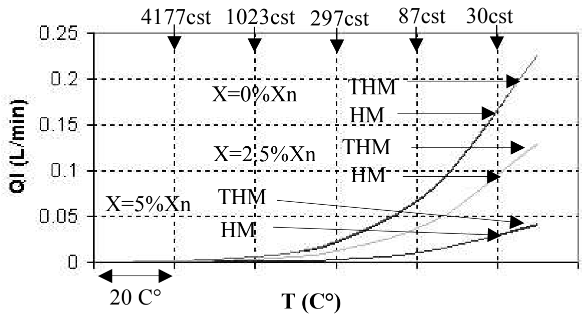

Figure 5. Leakage flow rate vs. temperature

Pressure/opening gain (*K_{PX}*)

Once again, the pressure/opening valve gain K_{PX} is simulated by reproducing the experimental process: the pressure difference at use ports ($\Delta P=P_1-P_2$) is plotted as a function of the spool opening that is varied at very low frequency while the use ports are blocked. Moreover, the non-dimensional pressure gain $\overline{K}_{PX}$ is expressed in Eq (5):

$$\overline{K}_{PX} = K_{PX} \,.\, X_n \;/\; P_n \tag{5}$$

The valve being supplied at constant pressure. $\overline{K}_{PX}$ is plotted on Figure (6) vs. temperature for two spool openings of 1% and 3% of spool full stroke.

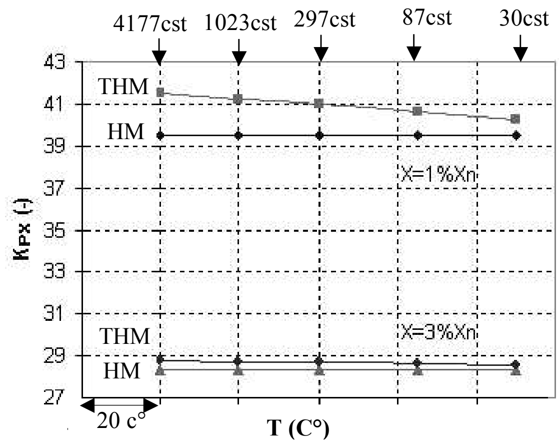

Figure 6. Non- dimensional pressure gain vs. temperature for two spool openings

According to this figure, it is observed that $\overline{K}_{PX}$ somewhat decreases with the temperature rise for the thermal-hydraulic model, whereas it remains constant whatever the temperature in hydraulic model. The thermal-hydraulic values are always higher than the hydraulic model ones but the difference is not significant (less than 5%).

Flow /opening gain (K_{QX})
The use ports are connected together. The flow passing through is plotted on Figure (7) as a function of the spool opening that is varied at very low frequency. Due the very high viscosity, there is an important change in the flow gain.

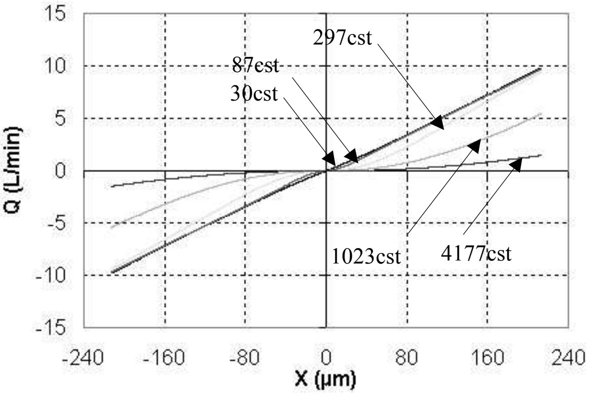

Figure 7. Flow gain for different temperatures

Two regions can be identified on the flow gain curve. The first one corresponds to the small valve opening in the surroundings of null region, in which the control position actuators operate. The second one is found for large openings in which the actuators operate when moving.

In order to develop system level models, it is proposed to express the flow gain K_{QX} as a continuous function, which its parameters vary with temperature. As defined in Eq (6), it includes a hyperbolic tangent function that reproduces the change of gain around the null region:

$$X = X_0 \tanh \frac{Q}{Q_0} + kQ \qquad (6)$$

where X_0 and Q_0 are the coordinates of the two regions asymptotes intersection point. And k is the asymptote slope at large openings.

The non-dimensional flow gain $\overline{K}_{QX}$ is then calculated for these two domains by Eq (7):

$$\overline{K}_{QX} = K_{QX} \,.\, X_n \,/\, Q_n \qquad (7)$$

The nominal flow Q_n is equal to 9.8 L/min for a valve pressure drop of 70 bar.

As expected under laminar flow conditions, at small openings the flow gain increases very significantly with temperature, Figure (8):

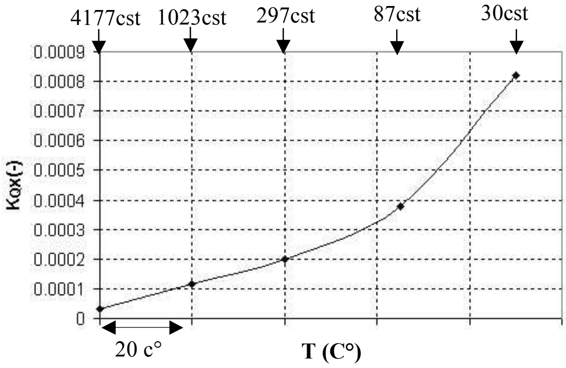

Figure 8. Flow gain at small spool valve opening

DISCUSSION

Considering the above results, it first appears that the valve leakage flow raises with temperature, increasing the global leakage coefficient A of the equivalent linear model of the actuator in the vicinity of the null opening. Adding, the thermal-hydraulic model gives slightly higher values than the hydraulic model because of raised temperature due to the energy losses. Secondly, the null region flow gain also grows significantly with temperature, producing high changes in the effective loop gain. Thirdly, the pressure gain remains quite constant with temperature. Only a slight reduction is found when using the thermal-hydraulic model.

According to all these changes, the simulation predicts that the static error will increase a little because of losses with raising temperature. As well as, the rapidity and the stability both depend on the change rate of A and K_{QX} . Because they affect K_l together, their partially compensate.

It must be kept in mind that this analysis must now be validated by real experiments. This part is now under preparation while the other parts of the servovalve are being modeled and analyzed in parallel.

Another action to be performed lies in the extension of the *AMESim* thermal-hydraulic component library. A thermal port will be added on the model icon to allow connection with the thermal library components. This will allow to include the heat exchange between the fluid and the external components, in the models of the thermal-hydraulic elements.

CONCLUSION

In this paper, we presented a first step in the study on the influence of temperature on the closed loop performance of electrohydraulic actuators. The importance of the servovalve static characteristics has been first pointed up. Then, a thermal-hydraulic model of a spool valve, has been built using only available components form the software libraries. Two typical cases have been studied. The first isothermal one is similar to the commonly used for conventional hydraulic simulations. In the second one, all the heat generated by pressure losses is stored into the fluid, so its temperature varies during the work. The simulation was run at different temperatures to point up its influence on the servovalve static characteristics.

Finally, it has been shown that the change of fluid viscosity due to temperature has a high importance on both leakage flow and flow gain and a very little influence on the pressure gain. When the heat generated by the pressure losses at valve orifices is considered, only slight changes are observed on the simulated values, that differ by less than 5% from the ones got using the hydraulic model.

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APPENDIX

The actuator which is selected in this study is made of a symmetrical jack and a symmetrical servovalve with rigid anchorage and transmission, Figure (9):

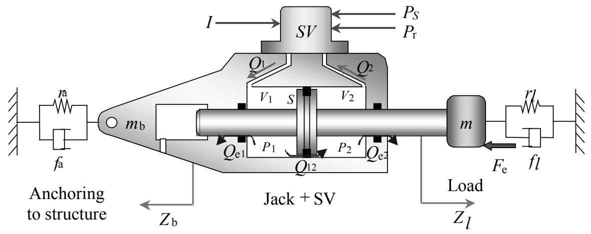

Figure 9. Basic model of an electrohydraulic actuator

Transfer function of an electrohydraulic actuator model can be roughly evaluated by making a linear analysis [3], with the consideration that the load stiffness r_l and its damping f_l are neglected, and that the jack piston and the load move together for same displacement Z_l without no connecting stiffness between them. So it will be as the following:

$$s\left(1+\frac{2\xi_h}{\omega_h}s+\frac{1}{\omega_h^2}s^2\right).Z_l=\frac{K_{QI}}{\left(1+\frac{Af}{S^2}\right)S\,G_{sv}(s)}I-\frac{K_f\,(1+\tau_h s)}{\left(1+\frac{Af}{S^2}\right)}F_e$$

with:

- Position static error $e=V_X^*-V_X=V_X^*-K_P.Z_l$

- Loop gain $K_l=\dfrac{K_A.K_{QI}.K_P}{\left(1+\frac{Af}{S^2}\right)S}$

- Hydraulic stiffness $r_h=\dfrac{2\beta S^2}{V_t}$

- Hydraulic damping ratio $\xi_h=\dfrac{\frac{Am}{S^2}+\frac{f}{r_h}}{2\sqrt{\left(1+\frac{Af}{S^2}\right)\frac{m}{r_h}}}$

- Hydraulic pulsation $\omega_h=\sqrt{\left(1+\frac{Af}{S^2}\right)\frac{r_h}{m}}$

- Load factor $K_f=\dfrac{A}{S^2\left(1+\frac{Af}{S^2}\right)}$

- Hydraulic time constant $\tau_h=\dfrac{S^2}{Ar_h}=\dfrac{V_t}{2A\beta}$

- Servovalve transfer function denominator

$$G_{sv}(s)=1+\frac{2\xi_{sv}}{\omega_{sv}}s+\frac{1}{\omega_{sv}^2}s^2$$