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ON LINE FAULT DETECTION FOR SERVOPROPORTIONAL VALVES

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Abstract. *This paper regards a model developed to aid online fault detection applied to single solenoid servoproportional spool valves. It includes the most common failures, its effects on the valve's behavior and how to identify undesirable behavior on an economically feasible way. Servoproportional valves are nowadays used in several applications in industry, aerospace and military fields. Some of these uses are critical-mission, where the valve must not fail without previous warning to prevent or mitigate financial losses, equipment damage and risk to personnel. Even in less demanding applications, it is very useful to be able to quickly locate a failure in the hydraulic circuit, which is a task that may demand up to 80% of the time used in corrective maintenance. Also, a valve which has not lost its function but is not operating at optimal performance may affect the entire hydraulic system's performance, making it hard for engineers to locate and solve the problem. Total failure can readily be detected using the signals available on the valve's electronics. A simple comparison between the command signal and the spool position signal can indicate to the user that the valve is no longer working properly. But, when that happens, the valve has already lost its function, and the effects of this on the system may be severe. However, since the spool positioning is on closed loop on these valves, incipient failures that can be overcome by the controllers cannot be easily detected. A model based on the valve's behavior is presented, which represents the valve's correct behavior under certain operation conditions, allowing the detection of failures that haven´t been foreseen by the model builder, as it would happen in a fault model. Differences between the model estimates and the actual measurements can be interpreted as the valve being in poor conditions, but it does not necessarily mean that the valve has lost its functions. To generate the estimates the valve inputs – pressure on the lines, command signal, and spool position – are fed to the model, and the model output is a healthy valve's force needs to overcome the internal forces and comply with the command signal. This estimate, given in the form of the maximum and the minimum current on the solenoid, is compared to a measurement of the valve solenoid's actual current. In this manner, the results of this research will make possible the development of a system that is able to detect when a valve is contaminated or suffering from wear before it actually loses its capability to operate. This yields great benefits both for the equipment maintenance effectiveness, for the system's energy efficiency and for the actuator's accuracy. It can also help prevent damage to other components, since most failures are caused by contaminated fluid, and this fluid runs through most of the hydraulic circuit.*

Keywords*: hydraulics, proportional valves, fault detection.*

1. INTRODUCTION

Electrically modulated hydraulic control valves (ISO, 1998), particularly servoproportional spool valves, are nowadays used in several applications in industry, aerospace and military fields. Some of these uses are critical-mission, where the valve must not fail without previous warning to prevent or mitigate financial losses, equipment damage and risk to personnel. Even in less demanding applications, it is very useful to be able to quickly locate a failure in the hydraulic circuit, which is a task that may demand up to 80% of the time used in corrective maintenance (Rodrigues *et al.*, 2003). Also, a valve who has not lost its function, but is not operating at optimal performance may affect the entire hydraulic system's performance, making it hard for engineers to locate and solve the problem (Bhojkar, 2004).

A third maintenance dimension called predictive maintenance, besides corrective and preventive maintenance, is then available when one is able to assess the condition of on-service equipment in order to predict when maintenance should be performed (Bhojkar, 2004). This approach is more cost effective over preventive maintenance because tasks are performed only when needed, there is less off-line time and less risk of damage and/or injury, since failures that may occur between scheduled preventive maintenance actions can be detected. Also, unnecessary tampering with the equipment might introduce failures in the system due to mistakes made by technicians. These facts justify the development of a valve fault detection method, in order to improve valve maintenance and the reliability of the hydraulic system where it is installed.

A fault detection process isolates the source of a system malfunction through the gathering and the analysis of information about the current state of the system obtained through measurement, testing and other sources of information, such as the user (Fenton *et al.*, 2002). Since the detection is usually performed in embedded systems with other functions, such as control, the fault detection process should be as simple as possible. The attempt to detect all possible failures would make the process unnecessarily complex and would increase its response time. That is why embedded fault detection systems are made to quickly detect the most common faults. On the other hand, running complex tests periodically would not compromise the system's safety (Bowers, *et al.*, 1989).

An intelligent monitoring system is usually composed by a sequence of subsystems that perform tasks including signal and data acquisition, signal conditioning, representation and evaluation to respond to the failure (Souza, 2004).

The evaluation of the acquired data from the system is important to the decision making process since it generates the information necessary to determine whether the system is in good conditions or not (Souza, 2004). This important task may be done using techniques such as artificial intelligence, behavior or failure models, signal analysis, process history, statistical methods and others (Fenton *et al.*, 2002, Grimmelius, 1999). The model based approach requires extensive effort during the development phase, and is computer intensive when used on-line, but is adaptable to other similar equipment through parameter and model adjustments. The model's development process also adds knowledge over the system under study (Grimmelius *et al*., 1999). For these reasons, the model approach has been chosen to generate the necessary information for decision making.

Following the need to create a simple fault detection process that can be embedded in the valve's electronics, a research on the valve's most common failures was made.

2. COMMON FAILURES IN SPOOL VALVES

Electrically modulated hydraulic control valves should have a service life of several million cycles when operated with adequate filtered fluid within the boundaries of its nominal specifications. However, its service life is greatly influenced by operational conditions, such as the environment where it is installed, fluid contamination, the use of dither, pressure and voltage spikes, etc.

Among the most common failures are the ones caused by contaminated fluid and degraded components due to excessive wear (Bhojkar, 2004). In order to gain some knowledge on how these failures affect the valve and what should be monitored to detect such a situation, a few of the most common faults are shown below.

2.1. Solenoid fault

After the solenoid reaches the magnetic saturation current level, additional current through the solenoid will only cause it to generate heat. This increases its impedance and therefore decreases the current through the solenoid for a certain voltage. This also reduces its ability to generate mechanical force. In extreme cases the core may fracture or the coil's inductance may be permanently altered (Glibisco, 2002). When the solenoid burns it becomes an open circuit, and no current goes through no matter the voltage applied. This causes the valve not to respond to the command signal, and therefore no spool movement will be possible.

2.2. Spring fault

A broken or fatigued spring may cause the spool to drift, since it is subject only to both the solenoid and to the steady-state flow forces. A valve with a broken spring is not able to comply with the command signal, even though the solenoid is working properly.

2.3. Fluid fault

According to Vijlee (2003), up to 75% of the failures in hydraulic systems are caused by contamination generated or added to the system. The undesirable effects include frequent component replacement, loss of movement repeatability, parameter alteration, fluid degradation and others. Among the most common contaminants are:

2.3.1. Solid particles

The gap between the spool lands and the sleeve of control valves ranges from 1 to 25 µm, according to the valve's size and design. That is why control valves are usually the most sensible components in a hydraulic circuit. Particles around that size can silt or become wedged in the spool or sleeve, leading to erratic movements, jamming, and permanent damage to lands, orifices and sleeve (Park, 1997). Contaminants such as sand, metal particles, polish compound, and other residues can cause wear and premature failure, making this kind of contaminant one of the critical factors to affect the service life and the reliability of hydraulic systems (Merrit, 1967, Tessman and Hong, 1998, Doddannavar and Barnard, 2005). These contaminants may come from the external environment or may come from the wear of components of the system.

How long it will take for this to occur depends on the amount and size of particles, and on the pressure differential, varying from a few seconds to a few hours if the valve is centered (Merrit, 1967, Park, 1997). Particles ranging from 5µm to 40 µm may quickly jam a valve, and occurrences even during the commissioning of the machine have been reported (Park, 1997).

In the long run, particle contamination may cause increased friction between the spool and the sleeve, increasing hysteresis, scratching the spool's surface and eroding edges, increasing flow and non-linear behavior in the center

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position. This state demands more from the solenoid, who has to operate at higher current levels to move the spool, increasing both threshold and flow gains, and decreasing pressure gains. When the valve is extremely damaged its responses may become slower, unstable or fail completely due to the clogging of orifices or to the failure of the solenoid (Merrit, 1967, Tessman and Hong, 1998, Park, 1997). The service life may be reduced from years to months, or even weeks (Park, 1997).

2.3.2. Water

Water may be dissolved in the fluid up to the saturation point, and then the excess presents as a second phase (free water) or an emulsion. How much water can the fluid hold will depend upon the kind of fluid, additives, pressure and temperature (Bauer and Day, 2007, Parker Training, 1999). Water contamination can lead to failures such as corrosion of surfaces, accelerated abrasive wear, metal fatigue, and increased friction due to viscosity loss among others. These effects will lead to the increase of the force necessary to move the spool. Even greater complications may occur when the temperature decreases, since the fluid holds less dissolved water under these conditions, and when the freezing point is reached, the formation of ice crystals may affect the system's functions (Parker Training, 1999).

2.3.3. Air

The air may be dissolved or free in the hydraulic fluid. Though it causes less trouble when dissolved, it may cause jamming, erratic functioning and other undesirable effects when it is trapped inside a valve (ATOS Electrohydraulics, 2006). The air may come from leaks in the system, from maintenance interventions, or from fluid turbulence in the sump.

2.3.4. Varnish and slug

Varnish and slug are generated by the degrading of the hydraulic fluid, which may happen as a result of natural aging, overheating, or the presence of contaminants such as water, air, solid particles, among others. It may be dissolved or free in the fluid. The fluid's capacity to hold them dissolved depends greatly on the fluid's temperature (Doddannavar and Barnard, 2005). The varnish may accumulate in low flow and low temperature areas of the system, while areas with a high flow or high temperature are not affected. This kind of contaminant affects particularly valves that are not used for a certain period, like the ones used in emergency systems or during the cold start of a machine with degraded fluid (Atherton, 2007).

These residues may clog orifices and grooves making the valve sluggish. In such a state the valve experiences jamming and performance loss. Also, solid particles may accumulate on the sticky surface, creating an abrasive surface that accelerates the wear of the moving parts of the system (Atherton, 2007).

As seen above, in electrically modulated hydraulic control spool valves, the most common failures may cause the increase of friction between the spool and the sleeve due to wear, abrasion, varnish and particle accumulation, that end up altering the valve's frictional characteristics (Fey, 1987). Those cause the stick and slip phenomenon, where the spool moves and stops several times during its trajectory, causing the actuator to move irregularly, and may even jam the valve's spool. For that reason it is important to have some estimate of the forces that would normally be present. Since the force applied on the valve by the solenoid is fairly proportional to the current through its coil (Linsingen, 2003), an estimate of normal current levels can be used for comparison with the actual current levels through the solenoid to detect faults. Knowing the current that should be applied at the solenoid at each spool position may help the operator or monitoring system to realize when the valve is not working properly even before the fault could be noticed through other of the valve's signals, such as spool position.

3. SERVOPROPORTIONAL VALVE MODEL

From the previous section one can perceive that, along with the spool positioning, the magnitude of the forces involved in the spool movement is affected in most of the failures mentioned in the researched literature. Servo-proportional valves are built with solenoids stronger than the needs to overcome the valve's internal forces in order to be able to move even in the presence of certain contaminants, or at higher pressure differentials. This is why the difference between the current levels of a healthy valve are so different from the levels of a contaminated or damaged valve, since the controller uses the extra power when it is not able to move the spool. Therefore, a model to estimate the internal forces through the magnitude of the current that should be applied at each spool position at certain operation conditions will be developed. In order to do so, a closer look at the forces involved in the spool movement should be taken.

The axial force necessary to move the spool can be divided in forces necessary to accelerate the spool and anything else that moves with it, forces necessary to overcome friction and the forces due to the flow through the valve, also called flow induced forces or Bernoulli forces (Merrit, 1967). There is also the force necessary to overcome springs used to center the valve or to act against the solenoid.

Regarding the flow forces, most of the force generated by the fluid is due to the steady-state flow force. That is related to the variation of the quantity of motion of the fluid as it leaves the valve's chambers. It is normal to the cross section of the *vena contracta* (the region where the jet area becomes a minimum and therefore the speed of fluid is maximum), which is typically at around 69° from the center axis of the spool if there is no radial clearance between the spool and the sleeve (Merrit, 1967). Therefore, this force has a lateral component that pushes the spool against the sleeve or body of the valve, and an axial component that tends to shut the valve closed. Commonly the ports of the valve are located symmetrically around the spool, leading the components of the lateral force to compensate each other. However, the axial forces are not compensated.

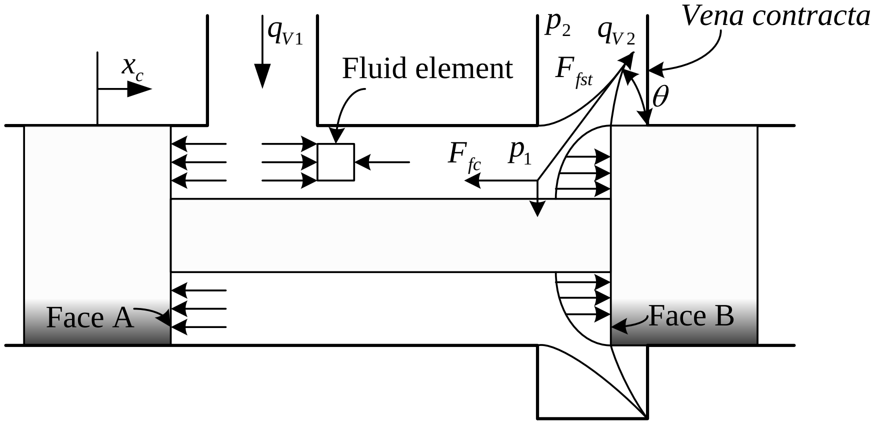

Figure 1. Flow forces on a spool valve due to flow leaving a valve chamber (Merrit, 1967)

The jet flow force can be expressed for each metering orifice as (Linsingen, 2003, Stringer, 1976, Merrit, 1967):

$$F_{fst} = \frac{\rho \cdot q_{V2}^2}{cc \cdot A_0} \tag{1}$$

Where:

F_{fst} is the steady-state flow force [N];

ρ is the fluid's mass density [kg/m³];

q_{V2} is the volumetric flow rate of fluid leaving the valve's chamber at the *vena contracta* [m³/s];

A_0 is the metering orifice area [m²];

cc is the contraction coefficient that relates the area at the metering office to the area at the *vena contracta* [no dimension].

This force is applied over the fluid. Consequently, there is a reaction force on the spool, and its axial component is:

$$F_{fc} = -F_{fst} \cdot \cos\theta \tag{2}$$

Where:

F_{fc} is the axial component of the steady-state flow force, a reaction to the jet flow force [N];

θ is the jet angle at the *vena contracta* [no dimension].

Assuming that the fluid inside small valve chambers is incompressible and that the leakage is negligible when compared to total flow, one can assume that the amount of fluid entering the chamber is the same amount leaving the chamber. Therefore (Merrit, 1967, Stringer, 1976, Linsingen, 2003):

$$q_{V1} = q_{V2} = q_V = cd \cdot A_0 \cdot \sqrt{\frac{2}{\rho} \cdot (p_1 - p_2)} \tag{3}$$

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Where:
q_{V1} is the volumetric flow rate of fluid entering the valve's chamber [m³/s];
q_V is the flow rate through the valve's chamber [m³/s];
cd is the discharge coefficient that accounts for cv and cc [no dimension];
p_1 is the pressure at the inlet of the valve's chamber [Pa];
p_2 is the pressure at the outlet of the valve's chamber [Pa].
The flow through the chamber can also be represented by (Furst, 2001):

$$q_V = Kv \cdot \frac{x_c}{x_{cn}} \cdot \sqrt{(p_1 - p_2)} \tag{4}$$

Where:
x_c is the valve's spool displacement [m];
x_{cn} is the nominal spool displacement, at which the Kv is obtained [m];
Kv is the valve's flow rate coefficient [m³/s.√Pa].
Both equations can be related by (Furst, 2001):

$$Kv = cd \cdot A_0 \cdot \sqrt{\frac{2}{\rho}} = \frac{q_V}{\sqrt{(p_1 - p_2)}} \tag{5}$$

From Eq. (1), Eq. (2) and Eq. (4):

$$F_{fc} = \frac{Kv \cdot x_c \cdot \sqrt{2 \cdot \rho} \cdot (p_1 - p_2) \cdot \cos\theta}{x_{cn}} \tag{6}$$

Because this force is directly proportional to spool displacement, it is analogous to a centering spring, although it also depends on pressure differential and flow (Stringer, 1976, Merrit, 1967).

The forces above are a significant contributor to the force required to stroke the valve's spool (Merrit, 1967), and are the reason why valves stroked by an electromagnetic device have size limits, once this kind of devices have distinct force limitations. Bigger valves must be hydraulically operated in a two-stage configuration, or have some kind of steady-state flow force compensation to overcome these forces. Therefore, for the valve's internal forces one has the equation below:

$$F_c = m_e \cdot \ddot{x}_c + B_e \cdot \dot{x}_c + K_m \cdot x_c + F_{fc} \tag{7}$$

Where:
F_c is the force necessary to move the valve's spool [N];
m_e is the equivalent moving mass [kg];
B_e is the equivalent viscous friction coefficient [N.s/m];
K_m is the spring rate [N/m].

The forces necessary to accelerate the effective moving mass (the mass of the valve's moving parts plus some fluid contained in the drain ports at the ends of the spool) and to overcome the dampening forces provided by the transient flow forces that occur at the change of flow rate and the viscous friction forces of the fluid adjacent to the valve's moving parts are, in most of the valve's movements, considerably smaller than the spring and steady-state flow forces, and can be neglected in the model for simplicity's sake. Their effect, however, is important, since they limit the valve's performance and response time and may demand the solenoid's maximum available force, and for this reason measures must be taken in the fault detection process to avoid false fault detections. This yields a much simpler, easier to calculate model, giving speed to the process. In a solenoid valve all these forces must be overcome solely by the solenoid (s). Therefore, the force produced by the solenoid (s) must be equal to the forces necessary to stroke the spool. Since the force generated by the solenoid (s) can be described in a simplified way by the equation (Stringer, 1976):

$$F_s = K_{Fi} \cdot i_s \tag{8}$$

Where:

F_s is the force produced by the solenoid [N];

K_{Fi} is the solenoid's force - current coefficient [N/A]

i_s is the current on the solenoid [A].

Substituting Eq. (8) in Eq. (7) and considering steady-state conditions (Stringer, 1976):

$$x_c = \frac{K_{Fi}}{K_{me}} \cdot i_s \tag{9}$$

3.1. Model applied to the studied valve

The Hydrus, HR01 servoproportional single solenoid valve is a prototype valve recently designed at LASHIP/UFSC. Since there is no field data on failures and the magnitude of the internal forces at such state, the choice for the model based approach was reinforced, since it is based on the valve's correct behavior.

Being a single solenoid valve, the HDR01 has only one spring, as seen in Fig. 2. This spring counteracts the solenoid, and to this force the steady-state flow force is added. At the beginning of the solenoid's displacement, when the current and the force yielded by the solenoid is zero, the valve port A is fully open and connected to the supply port, while the port B is connected to the return port. After the center position, where all ports are closed, the valve connects the port B and the supply port at the same time it connects the port A and the return port. Since the steady-state flow forces tend to close the valve, while the port A is connected to the supply port the steady-state flow forces help the solenoid against the spring. After the center, the steady-state flow forces act against the solenoid along with the spring.

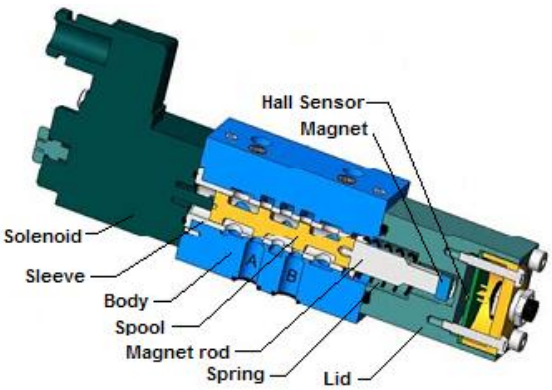

Figure 2. Hydrus HR01 servoproportional valve

As it typically occurs, the proportional solenoid has non-linear regions at the beginning and at the end of its displacement. These regions were avoided to make the valve proportional, but the non linearity offsets the force curve from zero, since the solenoid only starts generating force after a certain current is applied. Also, the solenoid presents hysteresis on its force versus current relation, as a result of the remanence phenomenon. Therefore it is necessary to add a linear coefficient to Eq. (8) in order to compensate the force offset. This coefficient has two values in order to account for the hysteresis of the solenoid, delimiting the maximum and minimum force yielded for a given current level. The force actually generated by the solenoid at a certain current will depend on its previous movements and on which way it is moving. Hence:

$$F_s = K_{Fi} \cdot i_s + b_s \tag{10}$$

Where:

b_s is the solenoid's force x current curve linear coefficient [N].

Therefore, associating Eq. (6), Eq. (9) and Eq. (10), regarding the characteristics of the valve one has:

When $x_{c\min} \leq x_c \leq x_{c\max}$:

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$$K_{Fi} \cdot i_s + b_s = K_m \cdot \left(x_c + x_{m0}\right) - \frac{Kv \cdot \sqrt{2 \cdot \rho} \cdot \Delta p \cdot \cos\theta}{x_{cn}} \cdot \left(x_{c0} - x_c\right) \qquad (11)$$

Where:

x_{c0} is the center of the spool's displacement, the position that closes the valve [m];

$x_{c\,\min}$ is the minimum spool displacement for which the valve is linear [m];

$x_{c\,\max}$ is the maximum spool displacement for which the valve is linear [m].

x_{m0} is the valve's spring's initial displacement [m];

Equation (11) can be expressed as:

$$i_s = \left[\frac{\sqrt{2 \cdot \rho} \cdot Kv \cdot \cos\theta \cdot \Delta p}{x_{cn} \cdot K_{Fi}} + \frac{K_m}{K_{Fi}}\right] \cdot x_c + \left[\frac{-\sqrt{2 \cdot \rho} \cdot Kv \cdot \cos\theta \cdot x_{c0} \cdot \Delta p}{x_{cn} \cdot K_{Fi}} + \frac{K_m \cdot x_{m0} - b_s}{K_{Fi}}\right] \qquad (12)$$

The Eq. (12) can be rearranged into:

$$i_s = a_i \cdot x_c + b_i \qquad (13)$$

Where:

a_i is the angular coefficient of the current estimate model [A/m];

b_i is the linear coefficient of the current estimate model [A].

These coefficients can be expressed as:

$$a_i = \left[\frac{\sqrt{2 \cdot \rho} \cdot Kv \cdot \cos\theta}{x_{cn} \cdot K_{Fi}}\right] \cdot \Delta p + \left[\frac{K_m}{K_{Fi}}\right] \qquad (14)$$

And:

$$b_i = \left[\frac{-\sqrt{2 \cdot \rho} \cdot Kv \cdot \cos\theta \cdot x_{c0}}{x_{cn} \cdot K_{Fi}}\right] \cdot \Delta p + \left[\frac{K_m \cdot x_{m0} - b_s}{K_{Fi}}\right] \qquad (15)$$

Both Eq. (14) and Eq. (15), however, can be further divided in angular and linear coefficients regarding Δp forming the equations below.

$$a_i = a_{a_i} \cdot \Delta p + b_{a_i} \qquad (16)$$

$$b_i = a_{b_i} \cdot \Delta p + b_{b_i} \qquad (17)$$

Where:

a_{a_i} is the angular coefficient that adjusts a_i to the pressure drop on the valve [A/m.Pa];

b_{a_i} is the linear coefficient that adjusts a_i to the pressure drop on the valve [A/m];

a_{b_i} is the angular coefficient that adjusts b_i to the pressure drop on the valve [A/Pa];

b_{b_i} is the linear coefficient that adjusts b_i to the pressure drop on the valve [A].

From Eq. (13), Eq. (16) and Eq. (17) a Simulink block diagram has been designed.

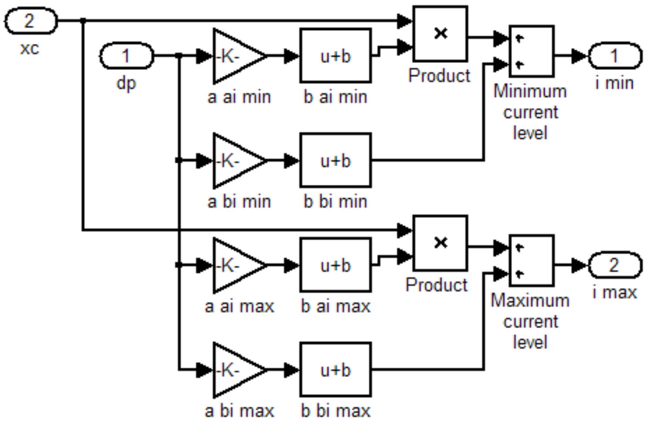

Figure 3. Solenoid current estimation model

The block diagram in Fig. 3 estimates the maximum and the minimum current at the solenoid at a certain spool position for a given pressure drop on the valve. It is supposed to estimate the steady-state current, but since the spring forces are considerable, it can be used to estimate with a reasonable margin of error the transient current after the first moments of great acceleration and solenoid and valve response, since the estimates change as the spool moves. Under steady-state conditions the current value is to be around the range created by the two estimated limits, or slightly above or below if hunting occurs due to the static friction of the spool, which is a non-linear characteristic not accounted for in the model.

The model expressed in Fig. 3 is fit for a symmetrical proportional valve. For an asymmetrical valve, two of these block diagrams should be used in order to change the Kv value as the valve changes the flow direction.

3.2. Experiments

At first, experiments were made to prove the theoretical concepts and evaluate the need for models that account for temperature changes. In Fig. 4a the valve's behavior under different supply pressure levels can be seen, showing that current changes relatively linearly with spool displacement, and that the rate of this change varies with the pressure on the valve.

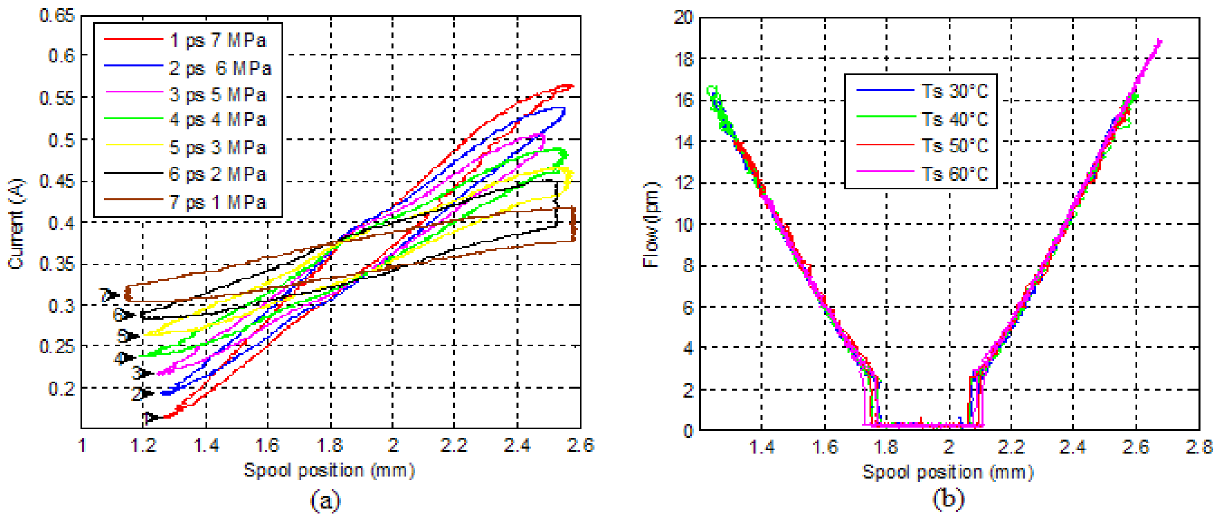

Figure 4. (a) Current on the solenoid x spool position at T_s = 40°C (b) Flow x spool position at p_s = 4 MPa

Also, the effect of hysteresis and threshold can be noted, since two different current levels are seen for the same spool position, depending on the way the spool is moving. Figure 4b shows how the flow remains the same with temperature changes of the magnitude used in the experiments. For greater temperature changes further compensations may be needed on the model.

Then the model has been experimented with theoretical values, extracted from manufacturer catalogs, and with experimental values, obtained from the experiments that originated Fig. 4a. Both obtained good results, however the

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experimental values have proven to be more accurate. Merrit (1967) warns to the uncertainty of the theoretical models, especially when the spool is close to the center, due to the changes in the jet angle at the *vena contracta* and other simplifications of non-linear factors. Values obtained experimentally may reduce some of the effects of these simplifications and variations that occur from valve to valve. However the model with theoretical values is still good enough for estimates, due to the great differences in the current noted when the valve is contaminated or worn.

The model experiments were made with port A connected to port B on a test bed following the recommendations of ISO 10770-1, with supply temperatures T_S ranging from 30°C to 60°C, and supply pressure p_S ranging from 3 to 5 MPa, running step signals ranging from 1,32 to 2,43mm and sinusoidal signals from 1,32 to 2,43mm with frequencies ranging from 1 to 20s. In Fig. 5a and Fig. 5b the current on the solenoid i_s can be compared to the model's minimum and maximum current estimates using experimental values and catalog/theoretical values to obtain the coefficients $a_{a_i}, b_{a_i}, a_{b_i}$, and b_{b_i}.

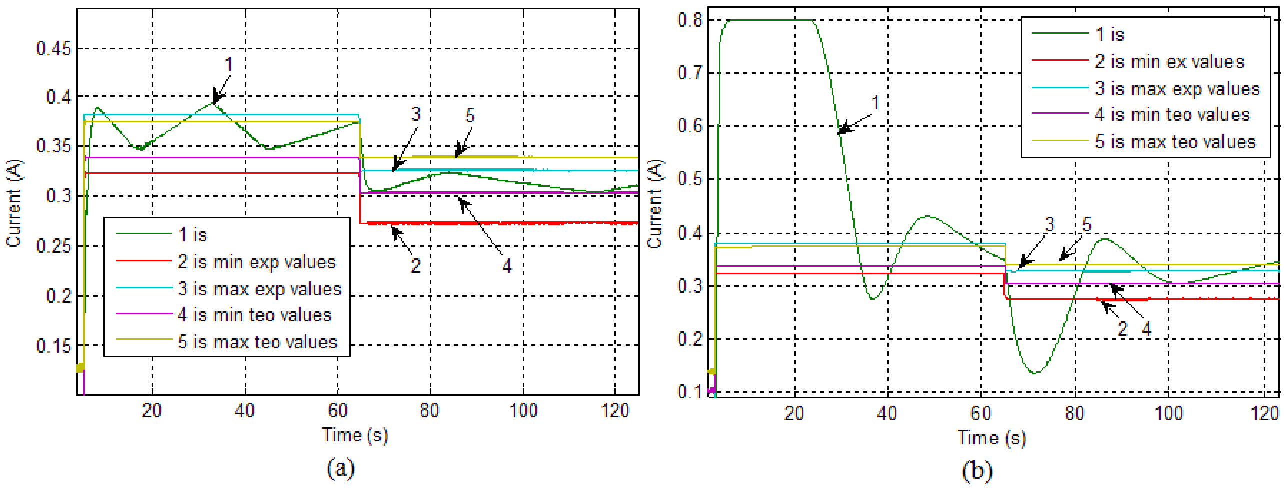

Figure 5. (a) Current on the solenoid for 1,85mm and 1,635 mm step signals at p_S = 4 MPa T_S = 40°C – healthy valve
(b) Current on the solenoid for 1,85mm and 1,635 mm step signals at p_S = 4 MPa T_S = 40°C – contaminated valve

On both sets of step experiments, great differences between the estimate and the actual current could only be noted on some experiments during 10% of the settling time after a step command was issued, or during the change of direction in the spool's displacement in the sinusoidal experiments. During these moments the controller ran an elevated current through the solenoid because the positioning error was maximum. Since the estimate is based on the spool position, the error between the estimate and the actual current was considerable. After the solenoid response time and spool acceleration, the error dropped to under 5% of the estimated value in the step signal experiments and to under 10% of the estimated value in sinusoidal experiments. This error exists because the model is made with steady-state condition equations. In the step signal experiments, during steady-state position the current remained within the boundaries of the estimates for most of the time, except when hunting occurred. Here some error was noticed due to the spool´s static friction, which forced the controller to raise the current to move the spool while the spool remained still.

After these experiments, the valve was contaminated. Three of its 12 grooves were contaminated with diesel engine adhesive. This emulates the accumulation of dirt or varnish on the grooves. The valve did not lose its function, and the controller was still able to position the spool. However, the valve's settling time was increased and the solenoid was far more demanded, yielding great differences between the estimates and the actual current even after the spool accelerated. The error between i_s and the estimates was greater than 100% of the estimated value during a considerable amount of time. Under these conditions a fault detection system would be capable of detecting the fault and warning the operator before a valve jam occurred. The jam eventually did occur under these conditions during a cold start after these experiments were made.

4. CONCLUSIONS

The model estimates adequately the current under several operation conditions. Along with other information commonly available on servo-proportional valves electronics, an efficient fault detection system can be designed using, for example, a simple rule based system. Although the model is based on steady-state equations, an error between i_s and the model's estimates greater than the typical error due to transient forces can be noted when the valve is in poor conditions, thus allowing the model to be used for diagnosis under transient conditions as well, provided the estimates made during the initial spool acceleration are disregarded.

The sensibility of the system will depend upon the quality of the transducers generating the data, on the quality of the valve and on the user's tolerance to faults. If the transducers have a low uncertainty, the valve is linear and well balanced, and the fluid is well filtered, the user can set the system to warn after small errors between the estimates and the actual current are detected. However, if the transducers have a higher uncertainty, the valve has variations in its parameters depending on spool position or larger manufacturing tolerances, and the system is run with average filtered fluid, a higher error level should tolerated before a fault is considered detected.

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