THE DYNAMIC STABILITY OF A LOAD SENSING DEVICE

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This paper was written in the memory of mat. eng. Gabriel Rădulescu – INOE 2000- IHP Bucharest, ROMANIA with whom I had an exceptional scientific collaboration over 25 years.
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Abstract: The paper is presenting a mathematical model for the dynamic behavior of a load sensing device. One is proposing a hydraulic model structure which provides a minimum energy consumption in order to obtain the optimum velocity for a hydraulic motor enclosed in a certain application. The paper studies the behavior of a system consisting of an adjustable axial piston pump and a flow controller. A case study is presented in the aim to verify the theoretical results.
Keywords: load sensing device, axial piston pumps

1. Introduction

Starting from the necessity to correlate the requested power in certain hydrostatic applications with the economical needs it has been developed a lot of devices used for adjustable axial piston pumps in the aim to minimize electrical energy consumption or fuel consumption at mobile applications. These devices became more and more sophisticated hence theoretical studies were performed [1-6] beginning with the mechanical ones, then combination between mechanical and hydraulic devices, servomechanisms and finally high electronic integration systems [12-15].

Studies concerning the efficiency of load sensing devices [7-11] was justified by the role of this specialized controllers which has been developed in order to provide the pump to offer the exact flow needed by the consumer i.e. the pump follows the application requested flow and adjust it (decreasing or increasing) by modifying it's capacity. The advantage of using load sensing devices is underlined by the modern tendency in mobile implements developing by using adjustable axial piston pumps with inclined disk in open systems. Load sensing devices as mechanical solution are useful in heavy duty applications where difficult and special conditions are to be fulfilled.

2. Motor speed variation devices

The necessity of load sensing devices rises from a brief comparison with classical motor variation speed systems concerning the amount of power losses.

2.1 Fixed capacity for motor and pump

This system is presented in figure 1 where the pump (1) has in derivation a three way speed controller consisting in: the valve (3) and the proportional distributer (4) which is conditioning the pressure drop $p_1 - p_2$.

$$N_u = p_2 Q_1 \,,\; N_{tot} = p_1(Q_1 + Q_2) \tag{1}$$

The efficiency relation is:

$$\eta = \frac{N_u}{N_{tot}} = \frac{p_2}{p_1}\frac{Q_1}{Q_1 + Q_2} = \frac{Q_1}{Q_{tot}}\left(1 - \frac{\Delta p}{p_1}\right) \tag{2}$$

where: $\frac{Q_1}{Q_{tot}} = \frac{n_M q_M}{n_P q_P}$ and $\Delta p = p_1 - p_2$.

In the hypothesis $q_M = q_P$ the power losses are:

$$\Delta N = \Delta N_u (1- \eta) \quad (3)$$

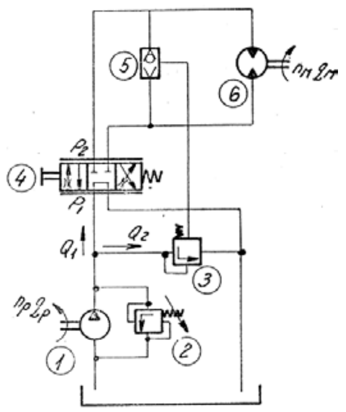

Fig.1 Fixed capacity motor and pump
1- Fixed cylinder pump; 2- Safety valve;3 -Normal closed valve with external control; 4- Proportional distributer; 5 -Selection valve; 6 - Fixed capacity motor.

Normal working parameters for this system are: Δp = 15 bar; p_1 =210 bar; n_M/n_P= 250/1500 = 0.1 and from relation (3) $\Delta N = 0{,}9N_{u.}$ This result is inconvenient for low power systems and unacceptable for high ones both because economical and technical reasons (the latter concerning the impossibility to dissipate the entire heat amount in the cooling system).

2.2 Adjustable capacity pump and fixed capacity motor

This solution improves the energetic situation by using an adjustable capacity pump equipped with servo-control as shown in figure 2. The capacity adjustment is controlled by the proportional distributor (6). The valve (5) provides a control flow Q_A from the control pump (4) flow The distributor (3) is selecting the motion direction for the motor (7).

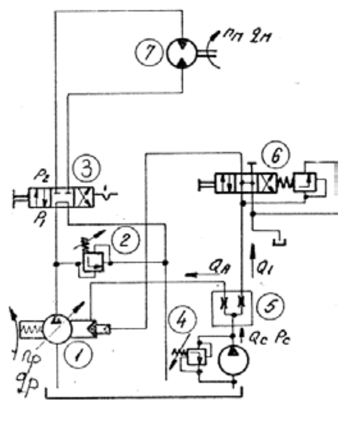

Fig. 2 Adjustable capacity pump and fixed capacity motor
1- Adjustable capacity pump equipped with servo control; 2- Safety valve; 3-Distributor; 4- Control pump; 5- Priority valve; 6- Proportional distributor; 7 - Fixed capacity motor.

The adjustable pump output flow is:

$$Q_P = Q_0 \Psi , \quad (4)$$

where: Q_0 is the maximum flow rate and Ψ□the governing factor with values between [0,1].

The useful and total power in this case are:

$$\mathbf{N_u = p_2 Q_0 \Psi \, , \, N_{tot} = p_1 Q_0 \Psi + p_c Q_c} \, . \tag{5}$$

The power loss is:

$$\mathbf{\Delta N = N_{tot} - N_u = Q_0 \Psi (p_1 - p_2) + p_c Q_c} \, . \tag{6}$$

and finally:

$$\Delta N = \frac{\Delta p}{p_2} N_u + p_c Q_c \, . \tag{7}$$

For the same parameters as shown at 2.1 and for $p_c = 0.25\ p_2$, $Q_2 = 0.1 Q_0\ \Psi$, the result is $\Delta N = 0.225 N_u$.This solution can be used for mean and high power systems with the following remarks: the motor drive is depending on two energy sources: the pumps (1) and (2) – hence two power sourced; there are two control systems with parallel working: one for reversing the motor motion (3) and one for controlling the pump (6).

2.3. Load sensing device

This system is presented in figure 3. The output flow pump (1) which drive the motor (5) is passing through the proportional distributor (3). When (3) is closed the pump is controlled by the pressure p_1 on minimum flow. For maximum flow gap of (3) is controlled at maximum flow. The flow regulator (2) is controlled by the pressure drop $\Delta p = p_1 - p_2$.

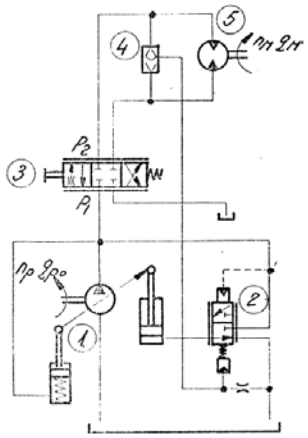

Fig.3 Load sensing device
1- Adjustable cylinder pump; 2- Flow controller; 3- Proportional distributer; 4- Selection valve; Fixed cylinder motor.

Using the same algorithm presented at 2.1 and 2.2 the useful powers are:

$$\mathbf{N_u = p_2\, n_M\, q_M \, , \quad N_{tot} = p_1\, q_{0P} \Psi_{nP}} \, , \tag{8}$$

where: q_M = ct. , $q_P = q_{P0}\ \Psi$ are the capacities of the motor and pump. Considering the maximum possibilities for the system: $q_M = q_{0P}$ and $n_M = n_P\ \Psi$, the ratio between powers is:

$$\frac{p_1}{p_2} = \frac{N_{tot}}{N_u} \text{ and } \frac{p_1 - p_2}{p_2} = \frac{N_{tot} - N_u}{N_u} \tag{9}$$

and finally the loss is:

$$\Delta N = N_u \frac{\Delta p}{p_2} . \tag{10}$$

For the numerical case study $\Delta N = 0.075\ N_u$. In figure 4 one presents the power variation ΔN in respect with useful power N_u between the minimum and maximum working power values.

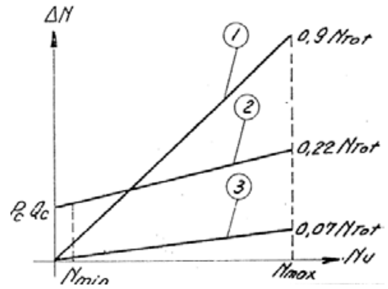

Fig.4 Power variation ΔN for all three case studies
1- Speed controller system; 2- Servo control regulation system; 3 – Load sensing regulation system

In figure 4 one can observe the difference of about 40-45% between mean working power values of the speed controller regulation system and load system regulation system. Basically the load sensing controller represents the association between an adjustable capacity pump and a flow controller. This configuration can be improved with other controllers according to specific application.

3. Mathematical model for the load sensing device

The hydraulic model for the studied sensing device is consisting of an adjustable axial piston pump and a flow controller as shown in figure 5. The characteristics are presented in figure 6. In figure 7 is presented the constructive diagram for the device.

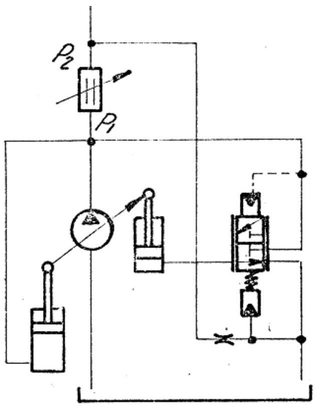

Fig.5 Hydraulic model of the sensing device

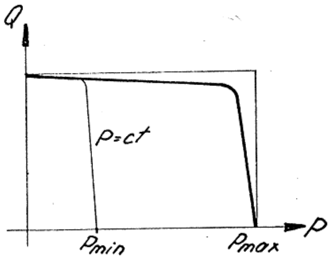

Fig.6 The characteristics of the load sensing device

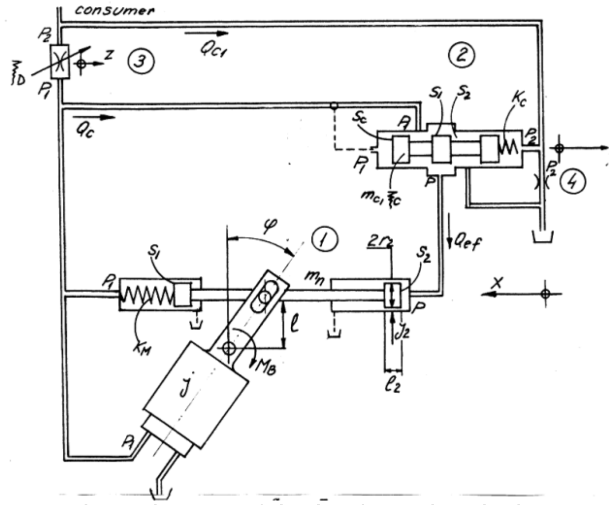

Fig.7 Diagram of the load sensing device
1 – The pump servomotor (S_1, S_2 – driving sections, K_M – spring constant, J – Cylinder block inertia moment, M_B = cylinder block driving couple, m_M – servomotor mass, l – lever arm length, $2r_2$ – drive diameter piston, j_2 – piston cylinder clearance);
2 – Flow Controller (m_c - slide-valve mass, S_1, S_2 – passing sections, S_c – control section, K_c control spring constant, ζ_C – hydraulic losses coefficient in the distributor);
3 –Hydraulic resistance (ζ_l – hydraulic losses coefficient);
4 – Nozzle (d – diameter, l – length).

The mathematical model is build on the next hypothesis:
a- The study is performed at constant consumer load (p_1 = ct.);
b- Friction forces are neglected;
c- Flow waste between the distributor opposite chambers are neglected (low pressure difference);
d- The compressibility effectis considered for a long drossel-distributor distance.

4.1 The flow controller

For the flow controller the continuity equation is:

$$Q_c = S_c\dot{y} = \frac{S_{01}}{\xi_c}\sqrt{\frac{2}{\rho}}\sqrt{p_1 - p} \quad (11)$$

where: $S_1 = S_{01}$ (1+ y/y_0), y_0 – the total initial displacement and $S_{01} = \pi d_c y_0$ the equivalent surface. If it is accepted the relation $2y_0p_1 >> y_p$ then using the approximation:

$$Q_c - S_c\dot{y} = \frac{S_{01}}{\xi_c}\sqrt{\frac{2p_1}{\rho}}\left(1 - \frac{p}{2p_1} + \frac{y}{y_0}\right), \text{ where } Q_{01} = \frac{S_{01}}{\xi_c}\sqrt{\frac{2p_1}{\rho}} \quad (12).$$

Finally the equations are:

$$m_c\ddot{y} + K_c y = S_c(p_1 - p_2)$$
$$y(0) = y_0; \quad \dot{y}(0) = 0 \quad (13).$$

$$\sqrt{p_1 - p_2} \cong \sqrt{p_1}\left(1 - \frac{p_2}{2p_1}\right)$$

4.2. The hydraulic resistance

The hydraulic resistance could be a mechanical controlled drossel mounted in series with a slide valve distributor, which provides a pressure drop and also reverse the motor motion direction or is a slide valve distributor with pressure regulator device. For the latter solution the output flow pump is:

$$\mathbf{Q = nq = nq_{max} \sin\phi / \sin\phi_{max}}; \tag{14}$$

for lower ϕ values $\sin\phi = \phi$ and relation (14) becomes:

$$\mathbf{Q = nq = nq_{max}\,\phi / \phi_{max}}; \tag{14'}$$

From geometrical conditions for the pump displacement: $x = l(\text{tg}\,\phi_{max} - \text{tg}\,\phi)$ and at lower angle ϕ the value is: $x = l(\phi_{max} - \phi)$. In this case the pump flow is:

$$Q \cong Q_{max}\frac{\varphi_{max} - \frac{x}{l}}{\varphi_{max}} = Q_{max}\left(1 - \frac{x}{x_{max}}\right), \text{ where } \varphi = \varphi_{max} - \frac{x}{l}, \tag{15}$$

where: $\phi = \phi_{max} - x/l$.
The hydraulic resistance input flow is:

$$Q - Q_c = \frac{S_3}{\xi_D}\sqrt{\frac{2}{\rho}}\sqrt{p_1 - p_2} \text{ where } S_3 = S_{03}\left(1 - \frac{z}{z_0}\right) \tag{16}$$

if neglecting the waste flow between the piston (S_1) and the cylinder in the hypothesis p_1 = ct. The regulating process is starting from the drossel open position and the reducing consumer motion possible by closing it. In this conditions and for the same approximation for pressure difference as mentioned before relation (16) is:

$$Q - Q_c = \frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}}\left(1 - \frac{p_2}{2p_1} - \frac{z}{z_0}\right) \text{ where: } Q_{03} = \frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}} \tag{17}$$

$$Q_{max}\left(1 - \frac{x}{x_{max}}\right) - Q_c = \frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}}\left(1 - \frac{p_2}{2p_1} - \frac{z}{z_0}\right) \tag{18}$$

The flow through nozzle(4) is:

$$Q_{c1} + S_c\dot{y} = \frac{d^4}{128\eta l}(p_2 - 0) \text{ where; } R_{02} = \frac{d^4}{128\eta l} \tag{19}$$

One consider $Q_{c1} = \varepsilon(Q - Q_c)$ where ε = ct. is (1…2)%, and using (17):

$$Q_{c1} = \varepsilon\frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}}\left(1 - \frac{p_2}{2p_1} - \frac{z}{z_0}\right) \tag{20}$$

Finally using (19) and (20) one obtain a relation between z and y:

$$\varepsilon\frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}}\left(1 - \frac{p_2}{2p_1} - \frac{z}{z_0}\right) + S_c\dot{y} = R_{02}p_2, \text{ where } Q_{03} = \varepsilon\frac{S_{03}}{\xi_D}\sqrt{\frac{2p_1}{\rho}} \tag{21}$$

4.3. Adjustable capacity system

The inertia mass for adjustable capacity system is $M = m_M + j/l^2$ where m_M is the piston mass, J – static momentum for the pump block. The necessary momentum for rotating the pump block (for p_1 = ct.) is $M_B = C_B p_1$ =ct. Neglecting the viscous friction it results:

$$M\ddot{x} + K_M x = pS_2 - p_1 S_1 - \frac{M_B}{l} \quad (22)$$

The continuity equation is:

$$Q_M = Q_{PM} + Q_{KM} + \dot{x}S_2 \quad (23)$$

Must take into account the losses between the piston S_2 and the cylinder:

$$Q_{pM} = \frac{f(M_2)}{\eta l_2} p, \text{ where } f(r_2) = \frac{\pi r_2^2}{2} j_2 (j_2 + r_2 - \frac{j_2}{\ln(1 + \frac{j_2}{r_2})}) = ct \quad (23)$$

and finally if:

$$C_{M2} = \frac{f(r_2)}{\eta l_2} \quad (23')$$

the equation is:

$$\mathbf{Q_{PM} = C_{M2} p} \quad (24).$$

The compressed flow must be considered because V_2 cannot be neglected even the distance between distributor to adjustable capacity system is short: $Q_{KM} = \frac{V_2}{\beta}\frac{dp}{dt}$. Considering $V_2 = S_2$ x corresponding to a mean value for x given by $\bar{x} \cong \frac{1}{3} x_{max}$ finally it results: $Q_{KM} = \frac{1}{\beta}\bar{x}S_2 \frac{dp}{dt} = \frac{S_2 x_{max}}{3\beta}\frac{dp}{dt}$, where $C_{KM} = S_2\, x_{max} /(3\beta)$. The compressed flow is:

$$\mathbf{Q_{KM} = C_{KM}\, dp/dt.} \quad (25)$$

Using the relations (23) to (25), and because $Q_M = Q_c - S_c\dot{y}$ the final equation is:

$$Q_c - S_c\dot{y} = C_{M2}p + C_{KM}\frac{dp}{dt} + \dot{x}S_2. \quad (26)$$

The mathematical model for the dynamic behavior of the load sensing device is described by the equations (12), (13), (21), (22) and (26). In order to apply Laplace transformation $\mathcal{L}[y(t)] = Y(s);\ \mathcal{L}[x(t)] = X(s)$ first step is consisting in the separation of the term ($Q_c - S_c y$) between equations (12) and (26), then using the same method for p_1 between (13) and (22) and differentiation the obtained relation. Second step is substituting the last two relations in the first derived one and the result is:

$$\frac{X(s)}{Y(s)} = \frac{\frac{C_{KM}M}{S_2}s^3 + \frac{M}{S_2}(C_{M2} - \frac{Q_{01}}{2p_1})s^2 + (\frac{C_{KM}M}{S_2} + S_2)s + \frac{K_M}{S_2}(C_{M2} + \frac{Q_{01}}{2p_1})}{\frac{S_1 m_c C_{KM}}{S_2 S_c}s^3 + \frac{S_1 m_c}{S_2 S_c}(C_{M2} - \frac{Q_{01}}{2p_1})s^2 + \frac{S_1 K_c C_{KM}}{S_2 S_c}s + (\frac{C_{M2}S_1K_c}{S_2S_c} + \frac{Q_{01}}{y_0} - \frac{Q_{01}S_1K_c}{2p_1S_cS_2})} \quad (27)$$

The third step is separating the term p_2 between equations (21) and (22) where $Q_{03} = p_1(R_{02} + Q_{03}/(2p_1))$. Using the transformation $\mathcal{L}[y(t)] = Y(s);\quad \mathcal{L}[z(t)] = Z(s)$ the result is:

$$W(s)=\frac{\varepsilon M S_c Q_{03}}{z_0 S_1 m_c^2\left(R_{02}+\varepsilon\frac{Q_{03}}{2p_1}\right)}\frac{s^3+\omega_{11}s^2+\omega_{12}^2 s+)\omega_{13}^2}{(s^2+\omega_{21}s+\omega_{22}^2)(s^3+\omega_{11}s^2+\omega_{22}^2 s+\omega_{33}^3)}, \qquad (28)$$

$$W_0(s)=\frac{\varepsilon M S_c Q_{03}}{z_0 S_1 m_c^2\left(R_{02}+\varepsilon\frac{Q_{03}}{2p_1}\right)} \qquad (28')$$

and ω_{ij} are the transfer function coefficients.

Case study:

The theoretical model developed in the paper is verified by a case study. It is presented an adjustable axial piston pump with the following data: q = 63 cm^3/rot, n_{nom} = 1500 rot/min, Q_{nom} = 1575 cm^3/s and working pressure 210 bar. One can calculate the values of the transfer equation:
$\omega_{11} = \omega_{31} = 1.28\ 10^{-5}$ l/s; $\omega^2_{12} = 672$ l/s^2 ; $\omega^3_{13} = 9.5\ 10^{-3}$ l/s^3; $\omega_{21} = 7.4\ 10^3$ l/s,
$\omega^2_{22} = \omega^2_{32} = 2\ 10^5$ l/s^2; ; $\omega^3_{33} = 6{,}2\ 10^4$ l/s^3. Neglecting $\omega_{11} = \omega_{31}$ for low values, relation (28) is:

$$W(s)=W_0\frac{s(s^2+\omega_{12}^2 s)}{(s^2+\omega_{21}s+\omega_{22}^2)(s^3+\omega_{22}^2 s+\omega_{33}^3)}, \qquad (29)$$

and the characteristic equation is:

$$\mathbf{s^5+\omega_{21}s^4+2\,\omega^2_{22}s^3+(\omega^3_{33}+\omega_{21}\,\omega^2_{22})\,s^2+(\omega^4_{22}+\omega_{21}\,\omega^3_{33})s+\omega^2_{22}\,\omega^3_{33}=0}, \qquad (30)$$

where the coefficients are: a_0 =1, a_1 =7.4 10^3, a_2 = 10^5, a_3 = 3.7 10^8, a_4 =10^{10}, a_5 =2.6 10^9.

The system stability

The system stability can be studied using various stability criteria. Because of the approximations made for the values of the coefficients, is not recommended to use the alternate roots criterion. One prefer Hurwitz stability criterion. The specific conditions for the characteristic equation are:

$$\mathbf{a_{1,5}>0},\quad \Delta_2=\begin{vmatrix}a_1 & a_3\\ a_0 & a_2\end{vmatrix}>0 \quad \text{and} \quad \Delta_4=\begin{vmatrix}a_1 & a_3 & a_5 & 0\\ a_0 & a_2 & a_4 & 0\\ 0 & a_1 & a_3 & a_5\\ 0 & a_0 & a_2 & a_4\end{vmatrix}> . \qquad (31)$$

The conditions for a stabile system are fulfilled:

$$\Delta_2\,\Delta_{21}-\Delta^2_{22}>0\ ,\ \Delta_{21}>0,\ \Delta_2\,\Delta_{21}>\Delta^2_{22} \qquad (32).$$

5. Conclusions

The necessity of load sensing devices is justified by a brief comparison with classical motor variation speed systems concerning the amount of power losses. This solution can be used for mean and high power systems with the following remarks: the motor drive is depending on two energy sources - the pumps and - hence two power sources; there are two control systems with parallel working - one for reversing the motor motion and one for controlling the pump. The analysis of the dynamic behavior of the device was performed by applying the Laplace transformation on the mathematical model. Deriving the coefficients of the characteristic equation stability conditions was analysed using Hurwitz criterion.

The compatibility between the theoretical model and the real loading sensing device confirm the accuracy of the problem solution. Developing the proposed mathematical model is possible to improve the dynamic behavior of the hydrostatic system equipped with adjustable axial piston pumps.

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