SOME ASPECTS REGARDING LOAD-SENSING SYSTEMS

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INTRODUCTION

Improving the characteristics of fluid power systems must not entail rendering fluid power more complicated, more expensive, or more maintenance intensive. Reducing energy consumption is an important trend in development of fluid power.

Load-sensing control systems have been developed recently with aim of realizing hydraulic circuits in which it is possible to control hydraulic actuators subjected to variable loads reducing energy consumption.

Energy saving is the biggest advantage with load-sensing. But load-sensing also means: longer hydraulic component life; oil flow regulation fast and precise; the system needs only a small cooler or no cooler at all; the number of pumps can be reduced and the system generates less noise.

Load-sensing is a primary control method. For energy saving it is also efficient secondary control. It is used not only for saving but also to recover energy.

1. STRUCTURE OF LOAD-SENSING SYSTEMS

There are two types of load-sensing systems:

-using pump with fixed displacement;
-using pump with variable displacement.

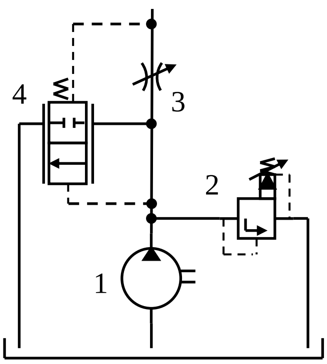

Figure 1. Load-sensing system with fixed displacement pump, *1 – pump; 2 – relief valve; 3 – throttle valve; 4 – pressure compensator.*

The system in figure 1 is a system with fixed displacement pump and pressure compensator. The pressure valve works as a pressure limiting valve.

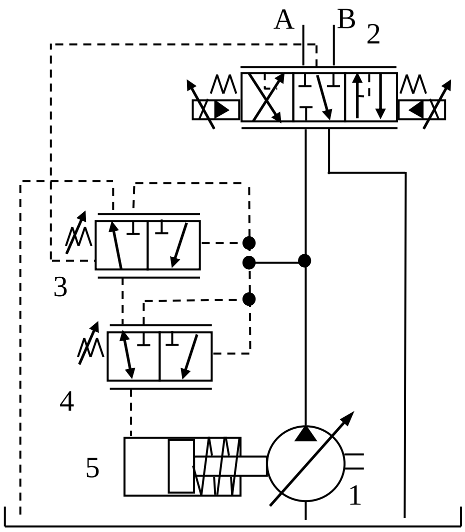

Figure 2. Load-sensing system with variable displacement pump, *1 – pump; 2 –proportional valve; 3 and 4 – pressure compensator, 5 – control system of the pump displacement.*

For the system in figure 2, the discharge of the variable displacement pump is changed by the pump control. The maximum pressure is maintained by a pressure relief valve.

2. ENERGY BALANCE FOR LOAD-SENSING SYSTEMS

In figure 3 one can see the energy balance for the two structures of load-sensing system.

As one can see, the structure in figure 2 is better from the point of view of energy consumption.

System in figure 1 is one which improves the energy balance and it is less complicated.

System in figure 2 uses a variable displacement pump, which has longer response time and a low mechanical efficiency because of the inertia of the pump elements, but the hydraulic energy balance is better than that of the system in figure 1.

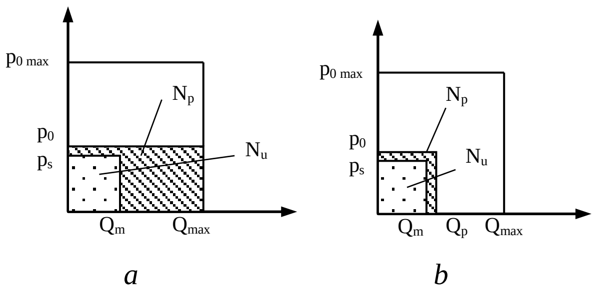

Figure 3. Energy balance for the load-sensing systems, *a – for the system in figure 1; b – for the system in figure 2.* N_p *is lost power,* N_u *is useful power.*

3. A SYSTEM WITH LOAD-SENSING LOOP

We propose a load-sensing system with an association including the calculus block BC1 and the relief valve SLP, with continuously variable set value of the pressure.

At the beginning of each working cycle the pressure $\boldsymbol{p_0}$ of the source is adjusted at the maximum value.

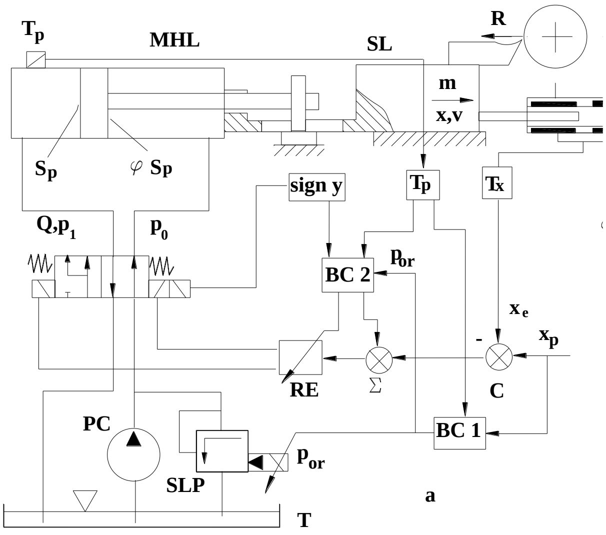

Figure 4. A system with load-sensing loop.

The calculus block BC1 receives the evolution of the load pressure during a rotation of the part.

The equilibrium at the MHL is:

$$\boldsymbol{m} \cdot \ddot{\boldsymbol{x}} + \boldsymbol{h} \cdot \dot{\boldsymbol{x}} = \varphi \cdot \boldsymbol{S_p} \cdot \boldsymbol{p}_0 - \boldsymbol{p}_1 \cdot \boldsymbol{S_p} - \boldsymbol{R} \quad (1)$$

where: $\boldsymbol{m}$ is the mass of the assembly, $\boldsymbol{h}$ is the rate of viscous friction, $\boldsymbol{Sp}$ is the area of the piston, $\boldsymbol{R}$ is external load.

It results the load pressure:

$$\boldsymbol{p}_1 = \varphi \cdot \boldsymbol{p}_0 - \frac{1}{\boldsymbol{S_p}}\left(\boldsymbol{m} \cdot \ddot{\boldsymbol{x}} + \boldsymbol{h} \cdot \dot{\boldsymbol{x}} + \boldsymbol{R}\right) \quad (2)$$

The pressure drop on the active edge of the valve is $\boldsymbol{\Delta p = p_0 - p}$, for negative displacements $\boldsymbol{y}$ of the spool of the valve, and $\boldsymbol{\Delta p = p_1}$ for positive values of $\boldsymbol{y}$.

We consider that the condition for the maximum efficiency is the work with pressure drops on the active edges greater than a preset value $\boldsymbol{\Delta p_a}$. It must be evaluated the rational pressure of the source, for the working cases with $\boldsymbol{y}$ positiv, assuming that $\boldsymbol{p_{1\,min}}$ appears in this situations with:

$$\boldsymbol{p}_{0r} = \boldsymbol{p}_0 - \frac{1}{\varphi} \cdot \boldsymbol{p}_{1max} + \frac{1}{\varphi} \Delta \boldsymbol{p_a} \quad (3)$$

This relation is the necessary condition for obtaining a minimum set pressure drop $\boldsymbol{\Delta p_a}$ on the active edge.

For negative values for $\boldsymbol{y}$:

$$\boldsymbol{p}_{0r} = \frac{1}{1-\varphi} \cdot \boldsymbol{p}_{1max} - \frac{\varphi}{1-\varphi} \cdot \boldsymbol{p}_0 + \frac{1}{1-\varphi} \cdot \Delta \boldsymbol{p_a} \quad (4)$$

The equations (3) and (4) may be used only if the load pressure do not change essentialy with the pressure of the source.

Finally, we have the value for $\boldsymbol{p_{0r}}$ as the maximum between the two arised from (3) and (4), which is sent to the relief valve as set value of pressure.

We can state that the algorithm for BC1 is: after initial set value for $\boldsymbol{p_{0r}}$ at maximum and the first turn of the part, from the registered values of $\boldsymbol{p_1}$ are determined $\boldsymbol{p_{1\,min}}$ and $\boldsymbol{p_{1\,max}}$ and then, in concordance with sign y, is calculated the value for the rational pressure $\boldsymbol{p_{0r}}$, which is sent to the relief valve and will be the pressure of the source for all the working cycles with similar conditions.

4. SECONDARY CONTROL HYDRAULIC CONCEPT

The load-sensing (LS) technology was generally accepted due to its better energy utilization. However, in most applications on the market the hydraulic systems are sub-optimized for some specific function and there are still a lot of problem to solve about controllability, overall system efficiency and damping of oscillations.

In addition to the primary controlled systems, constant pressure systems could be applied. This concept characterize secondary controlled systems, where the hydraulic output units are connected to a constant pressure rail. Displacement control of the secondary units, support direct control of the output torque or force to the load.

A secondary controlled system is characterized by a pressure coupling, since the high pressure is maintained at a quasi-constant level and the flow is transferred without throttling from the primary side. By connecting a hydraulic accumulator on the high pressure side, energy can be recovered when lowering or decelerating a load.

As long as the high pressure can be held at a quasi-constant level there will be no interference between the secondary controlled units. During braking of the load the secondary unit will work as a pump and feed fluid into the high-pressure rail and this energy is stored in the accumulator

5. CONCLUSIONS

System in figure 1 has a load-sensing loop which influences the pressure of the relief pressure valve. This is a load-sensing which improves the energy balance and it is less complicated.

System in figure 2 uses a variable displacement pump which has longer response time and a low mechanical efficiency because of the inertia of the pump elements but the hydraulic energy balance is better than that of the system in figure 1.

In this paper, it has been proved that hydrostatic systems with saving energy have the possibilities to be a very strong competitor.

Increased overall system efficiency is of great importance, since this is the way to reduce the total consumption. Control strategies also need further improvements to fully utilize the advantages of hydrostatic systems. Secondary control can be used for both drive trains and working hydraulics (lifting loads etc.). Brake energy storage capability in hydraulic accumulators also gives great advantages in energy consumption reduction.

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Recomandat spre publicare: 28.01.2009