INFLUENCE ON ENERGY EFFICIENCY OF MECHANISMS OPERATED WITH HYDRAULIC CYLINDERS

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ABSTRACT

During the operation of mechanisms operated by hydraulic cylinders significant energy losses occur. They is due to the loss of flow, pressure and friction losses that occur in the system. This paper highlights the connection between the kinematics diagram, hydraulic diagram and system efficiency. Energy losses that occur in hydraulic systems of excavators are analyzed.

KEYWORDS: : hydraulic cylinders, mechanism, energy efficiency

1. INTRODUCTION

Evaluating energy losses of the analyzed system starts from the power source. In this case the source of flow is an axial piston pump with variable displacement control, for constant power , controlled by an automatic flow control device. The device limits the maximum power produced by the hydraulic pump (HP, Figure 1), in order not to exceed the maximum power produced by the engine (TE).
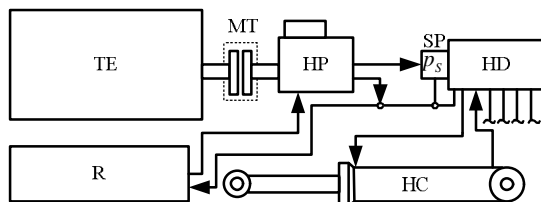

Fig. 1. The schematic diagram of the mechanical and hydraulic system

It is known that:

- The combustion engine has an efficiency that depends on load and speed (η_{TE});

- Mechanical transmission has its own mechanical efficiency (η_{MT});

- The efficiency of the pump depend by the mechanical and hydraulic instantaneous efficiency (η_M and η_H).

The hydraulic oil supplied by the hydraulic pump meets on the trail a series of hydraulic resistance whose value depends on:

- Spatial geometry of elements which limit body of fluid;

- Hydraulic oil viscosity, that depend on oil composition and temperature.

Pressure drops on hidraulic resistances lead to significant energy loss. If the operating pressure exceeds the maximum safety value, p_S, the pressure valve opens and sends a flow to tank.

Exceeding pressure at safety valve, which has been set, depends on hydraulic system and of forces acting on active hydraulic cylinder. The size of these forces depends on the soil strength and geometry of digging equipment

The inert weights of equipments elements can lead to overcome the maximum pressure. Usually they appear at the beginning and end of the race. Some properly chosen parameters for kinematic of equipment, correlated with hydraulic scheme, can reduce the energy losses.

It is taken into account and the fact that the hydraulic oil return to tanks is made with energy consumption due to pressure loss.

Even the loss of hydraulic fluid from hydraulic cylinders that are beyond the pressure overload should be included in the equipment efficiency during the digging.

All energy losses, regardless of how they are produced, lead to lower overall efficiency of the hydraulic excavator, of the work

equipment in accomplishing the tasks of digging.

2 CALCULATION OF INTERNAL LOSS OF FLOW FOR A VARIABLE FLOW HYDRAULIC PUMP

Most excavators use as source of energy heat engines. Their power characteristic allows only modest excess of power produced and that only for short periods of time. Therefore, to limit the maximum mechanical power transmitted are used hydraulic pump power regulator. Variable flow pumps are used with axial piston. The achieved flow is described by the relation $Q_t = \frac{1}{2} \cdot \pi \cdot D^2 \cdot R \cdot z \cdot n \cdot sin(\alpha)$ or $Q_t = K \cdot sin(\alpha)$.

At small angles $sin(\alpha) \cong \alpha$ (in radians) results $Q_t = K \cdot \alpha$.

Increasing pressure beyond a certain point p_b (at which maximum power , N is reached) leads to reduced flow. Figure 2 shows the diagram of operation of such pumps.
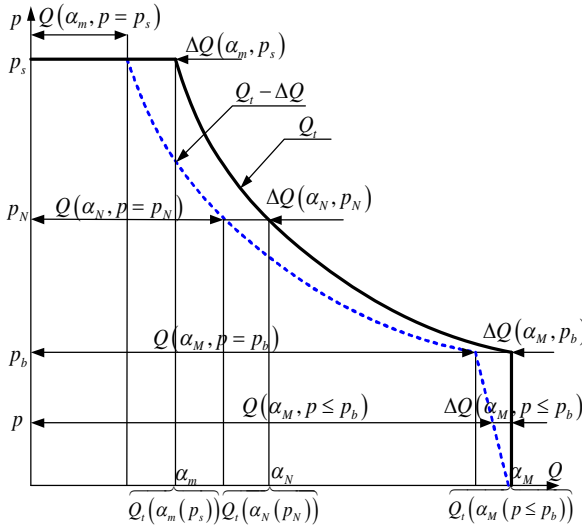

Fig. 2. Flow diagram at a pump with variable flow at constant maximum power.

With thick continuous line is represented the theoretical variation of the function $p(Q)$ Variation of $p(Q)$ affected by loss of flow $\Delta Q(p)$ is represented by dashed line.

$$N = Q_t\left(\alpha_M\left(p_b\right)\right) \cdot p_b = Q_t\left(\alpha_m\left(p_s\right)\right) \cdot p_s = = Q_t\left(\alpha_N\left(p_n\right)\right) \cdot p_N \tag{1}$$

Are known:

-The yield for pump at maximum working pressure $\eta_H\left(\alpha_m, p_s\right)$;

-The maximum mechanical power, $N = ct.$, transmitted to pump at nominal speed n. Pressure change between p_b (begin to reverse tilt) and p_s, the opening pressure of the valve.

By imposing certain operating speeds it is obtained the required flow $Q_t\left(\alpha_M\left(p_b\right)\right)$.

By imposing the condition that the power is constant, $N = ct.$,it results in pressure p_b.

$$p_b = \frac{N}{Q_t\left(\alpha_M\left(p_b\right)\right)} \tag{2}$$

By limiting the maximum pressure (functional and safety reasons), according to the kinematics scheme and hydraulic scheme, results the following flows

$$\begin{cases} Q_t\left(\alpha_m\left(p_s\right)\right) = \dfrac{N}{p_s} \\ Q_t\left(\alpha_N\left(p_N\right)\right) = \dfrac{N}{p_N} \end{cases} \tag{3}$$

From the mathematical expression of hydraulic efficiency

$$\eta_H\left(\alpha_m, p_s\right) = 1 - \frac{\Delta Q\left(\alpha_m, p_s\right)}{Q_t\left(\alpha_m\left(p_s\right)\right)} \tag{4}$$

we obtain the expression of the flow loss

$$\Delta Q\left(\alpha_m, p_s\right) = \left(1 - \eta_H\left(\alpha_m, p_s\right)\right) \cdot Q_t\left(\alpha_m\left(p_s\right)\right). \tag{5}$$

Because the angle of tilting for elements of pomp is relatively small, **hydraulic flow losses can be considered that depend only on the pressure**, being proportional to it, as it can be seen in Figure 3. These losses occur inside the hydraulic pump and are drained to the hydraulic oil tank.
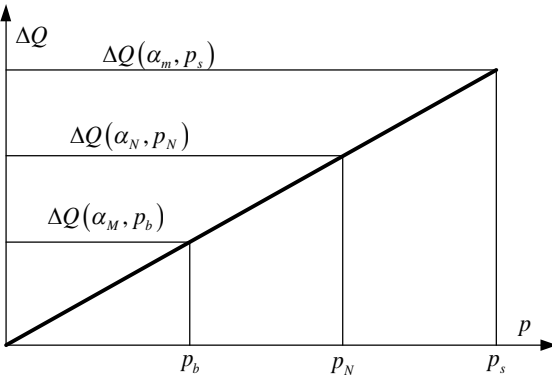

Fig. 3. Influence of pressure on flow losses

As a consequence

$$\frac{\Delta Q(\alpha_m, p_s)}{p_s} = \frac{\Delta Q(\alpha_N, p_N)}{p_N} = \frac{\Delta Q(\alpha_M, p_b)}{p_b} \qquad (6)$$

and

$$\Delta Q(\alpha_N, p_N) = \frac{p_N}{p_s} \cdot \Delta Q(\alpha_m, p_s) \qquad (7)$$

Starting from

$$\eta_H(\alpha_N, p_N) = 1 - \frac{\Delta Q(\alpha_N, p_N)}{Q_t(\alpha_N(p_N))} \qquad (8)$$

and using relations (1, 6), it follows

$$\eta_H(\alpha_N, p_N) = 1 - \left(\frac{p_N}{p_s}\right)^2 \cdot (1 - \eta_H(\alpha_m, p_s)) \qquad (9)$$

relationship that satisfies the requirements of the end of interval for pressure variation.

Because the pressures $p_b \le p_N \le p_s$ and using the relation (5) in p_b resulting

$$\eta_H(\alpha_M, p_b) = 1 - \left(\frac{p_b}{p_s}\right)^2 \cdot (1 - \eta_H(\alpha_m, p_s))$$
$$= 1 - \frac{\Delta Q(\alpha_M, p_b)}{Q_t(\alpha_M(p_B))} \qquad (10)$$

Therefore, the loss of flow from hydraulic pump, at maximum flow and p_b, it is

$$\Delta Q(\alpha_M, p_b) = \left(\frac{p_b}{p_s}\right)^2 \cdot (1 - \eta_H(\alpha_m, p_s)) \cdot \frac{N}{p_b} \qquad (11)$$

For the calculation of flow losses, for pressure p_N, it is considerd the identity

$$\eta_H(\alpha_N, p_N) = 1 - \left(\frac{p_N}{p_s}\right)^2 \cdot (1 - \eta_H(\alpha_m, p_s))$$
$$= 1 - \frac{\Delta Q(\alpha_M, p_b)}{Q_t(\alpha_M(p_B))} \qquad (12)$$

from where it results

$$\Delta Q(\alpha_N, p_N) = \left(\frac{p_N}{p_s}\right)^2 \cdot (1 - \eta_H(\alpha_m, p_s)) \cdot \frac{N}{p_N} \qquad (13)$$

Consequently, on the interval $p_b \le p_N \le p_s$, flows are

$$\begin{cases} Q(\alpha_m, p_s) = \dfrac{N}{p_s} \cdot \eta_H(\alpha_m, p_s) \\[2mm] Q(\alpha_N, p_N) = \dfrac{N}{p_N} \cdot \eta_H(\alpha_N, p_N) \\[2mm] Q(\alpha_M, p_b) = \dfrac{N}{p_b} \cdot \eta_H(\alpha_M, pb) \end{cases} \qquad (14)$$

For pressures $p \le p_b$ the loss of flow decreases linearly from $\Delta Q(\alpha_M, p_b)$ in p_b to 0, as seen in Figure 2. From

$$\frac{p}{\Delta Q(\alpha_M, p \le p_b)} = \frac{p_b}{\Delta Q(\alpha_M, p_b)} \qquad (15)$$

it results

$$\Delta Q(\alpha_M, p \le p_b) = \frac{p}{p_b} \cdot \Delta Q(\alpha_M, p_b), \qquad (16)$$

relationship that satisfies the requirements of the ends of the interval.

2. ENERGY LOSS DUE TO PRESSURE DROP ACROSS THE HYDRAULIC CIRCUITS

Pressure drop in hydraulic installations, from the hydraulic pump to the consumers, depends on the size of hydraulic resistance and the flowing kinds. They may differ in various parts of the hydraulic circuit, even if the flow is the same. Because the distances between local hydraulic resistances are small it is unlikely to witness the return to laminar flows.

The determination of these pressure drop can be made:
- By classical analytical calculus [1];
- By numerical simulation using specialized software [3].

A delicate problem in carrying out the calculations is related to determining the working pressure of the hydraulic pump. Mechanical resistances encountered by consumers during work lead to a certain value of pressure. Exit pressure at hydraulic pump will be higher that the working pressure of the consumer, due to loss of hydraulic pressure on the hydraulic circuits. And the pump pressure value depends on the product flow. Developed flows cause consumers to achieve speeds that influence the mechanical resistance.

Hydraulic oil is evacuated from consumers at higher pressure than the pressure in the tank. Exhaust flow size depends on the size of the consumer received flow and internal flow losses.

In the case of machinery working the soil, does not exist mathematical models describing the interaction of the bucket with very various soil types. Therefore, determining the forces of digging is difficult.

One possible solution is to use maximum forces. They are developing in the direction of digging. Maximum digging forces are calculated considering related restrictions introdused by:

- The kinematic schemes;
- The hydraulic schemes;
- The machine stability;
- The adherence of the machine.

4. CONCLUSIONS

Making machines with increased energy efficiency is possible through a closer coordination between digging equipment parameters and characteristics of hydraulics.

The mechanism of excavation equipment and hydraulic components can be improved separately . However, their lack of correlation may introduce significat energy losses.

Performing numerical simulations using consistent models can significantly contribute to increasing energy performance. For their realization are required mathematical models that interpret reality more accurately .

REFERENCES

[1] **Deacu L., Banabic D., Radulescu M.M., Ratiu C**., *Tehnica hidraulicii proportionale*. Cluj-Napoca, Editura Dacia, 1989.

[2] **Hauk N.** *Contributii teoretice si experimentale privind comportarea in regim dinamic a cilindrilor hidraulici montati pe excavatoare*. Doctoral dissertation – Universiti "Dunarea de Jos" Galati, 1998.

[3] **Hauk N**. *The possibilities of increasing efficiency and confidence of the calculations for hydraulic cylinders provided with cushioning system at end of race.* The annals of "Dunarea de Jos" University of Galati, fascicle XIV Mechanichal Engineering, ISSN 1224-5615, 2012

[4] **Janosevic D., Mitrev R., Andjelkovic B., Petrov .P,** *Quantitative measures for assessment of the hydraulic excavator digging efficiency,* Journal of Zhejiang University-SCIENCE A (Applied Physics & Engineering) 2012 13(12):926-942

[5] **Poole A. D., Booker J. D.** Design methodology and case studies in actuator selection. Mechanism and Machine Theory , Vol. 46. Issue 5

[6] **Soo-Jin Lee, Pyung-Hun Chang**, *Modeling of a hydraulic excavator based on bond graph method and its parameter estimation*, Journal of Mechanical Science and Technology 26 (1) (2012) 195~204

[7] **Voinea, R., Deciu, E., Dragomirescu, C.**, *Technische Mechanik*, Ed. ALMA Craiova, 2009.

[8] http://www.boschrexroth.com.