Process equipment pump-controlled hydraulic drive: improving energy efficiency

IOP Conf. Series: Materials Science and Engineering **643** (2019) 012098 doi:10.1088/1757-899X/643/1/012098

Process equipment pump-controlled hydraulic drive: improving energy efficiency

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Abstract. The paper addresses the issues of stepless speed control of the hydraulic drive output link by means of frequency control, throttle control and pump-throttle control; they are compared, and their applicability in various types of hydraulic drives is analyzed [1, p.619]; a mathematical model of losses arising in a volumetric rotary hydraulic machine with volumetric (pumping) control and frequency management is given; a mathematical model is formed on the basis of which the dynamic characteristics of removable hydraulic drive with frequency control with varying parameters of external load and operating factors will be studied. The paper is based on theoretical research of the frequency-controlled hydraulic drive that was carried out [2, p.41]. As a result of the research, a schematic solution and a mathematical model of the developed hydraulic drive have been proposed. The main structural and technical and economic indicators are: a change in the angular velocity of the rotor of the machine in a given range, and an increase in the energy efficiency of the drive.

1. Introduction

The purpose of the work is to increase the energy efficiency of process equipment. A modern approach to the development of competitive process equipment is to improve the energy efficiency of the volumetric hydraulic drives used in it. In this case, the hydraulic drive designer faces the task to realize its main advantage: stepless speed control of the output link. To this end, different regulation methods are used. They are: volume-throttle control, pump-throttle control and frequency control of the hydraulic drive output link speed [3, p.368].

Volumetric hydraulic drives with throttle and pump-throttle control are the most widely used today. The frequency control method has found its application mainly in stationary pumping systems with dynamic pumps, the shafts of which are driven by asynchronous electric motors (AEMs). Successful application of frequency regulation in dynamic hydraulic drives led to the implementation of this method of regulation in power volumetric hydraulic drives. The practical implementation of this method of regulation is also due to the advent of compact and inexpensive devices: frequency converters [4, p. 84].

However, the operation modes of the volumetric-rotary pumps as constituent parts of power volumetric hydraulic drives are fundamentally different from dynamic pumps. The most important differences are as follows:

1. Rotor acceleration under almost instantly increasing load, reversal of sign and values of the load, frequent starts, stops and reverses, shock loads, drive operation in a wide range of speeds up to small and ultra-low leads to deterioration of the pump flow and cooling of the AEM [5, p.329];

2. Unlike dynamic hydraulic drives, when developing modern volumetric hydraulic actuators, there is a choice of available alternatives that are effective in terms of functionality and efficiency of methods for wide-range control of the output link speed.

Thus, there are at least three functional analogs of the volume hydraulic drive output link speed control competing with each other: volume-throttle control, pump-throttle control, and frequency control.

Hydraulic drives with pump-throttle control are the most common; they are based on adjustable pumps.

The throttle control principle has a number of advantages and disadvantages. Throttle control advantages are:

1. Minimal cost;
2. Insignificant mass-dimensional indicators;
3. Either air or liquid cooling systems can be used, or two pumps of different capacity, with their periodic shutdowns possible.
4. Throttle control can be used as a group drive or as an executive module, i.e. this control method is universal.

Throttle control disadvantages are:

1. Relatively low efficiency;
2. Increased heat dissipation.

The pump control method without the use of throttle apparatus is inoperative in group hydraulic drive, since load change on any of the consumers will result in pressure change in the pressure hydraulic line, and this in turn will lead to the disruption of all other consumers of the drive [6, p. 511]. Therefore, in a drive with pump control, throttle hydraulic equipment is installed for each consumer, due to which the flow is controlled and the pressure remains almost unchanged. Such a hydraulic drive is called pump-throttle.

Also, frequency regulation cannot be used in group hydraulic drive without throttle hydraulic equipment for each hydropower consumer. Such a drive is called hydraulic drive with frequency-throttle control.

The energy flow control method using volumetric hydraulic drive with frequency control has several advantages compared with volumetric hydraulic drive with pump control. For example:

- increasing the value and a flatter dependence of the effective efficiency coefficient on the control parameter;
- a wider range of stepless speed control of the output link;
- the capability to place the power part of a volumetric hydraulic drive with an unregulated compact pump and hydraulic motor in the active zone of the process equipment;
- the ease of modernization and control system replacement, since the smooth adjustment of the angular velocity eliminates the use of variators, gearboxes, throttles, and other control equipment;
- hydraulic drive is equipped with simple and, therefore, more reliable hydraulic machines, which makes it possible to significantly increase the reliability and survivability of the system;
- the use of a frequency converter in a drive with a reversible pump and a motor allows energy recovery for both its own needs and to transfer energy to the electrical network;
- a high degree of unification of the pump and motor parts of the drive makes it possible to use not only identical reversible machines, but also to obtain various combinations of speeds, traction and adjustment characteristics at a given power, i.e. while maintaining the same drive AEM, through the use of reversible machines of different types, with different volumetric constants;
- hardware ease of upgrading of existing volumetric hydraulic actuators, by installing frequency converter, which makes it possible to make the drive without control adjustable, and also to further expand the control range.

However, volumetric frequency controlled hydraulic drive is associated with a number of problems. For example:

- the work of the drive motor to maintain the load when the rotor is stationary and the resulting problems with heat removal;
- physical processes features and negative phenomena that may occur during the transition of volume-rotary hydraulic machines to low and ultra-low angular velocities of the rotor, i.e. the problem of compatibility of volume losses with the pump flow [6, p.511].

Figure 1 shows the classification of hydraulic drives of stationary and non-stationary process equipment according to the output link movement type and layout type.

The classification makes it possible to determine the types of process equipment where the frequency method of output link speed control can be applied, and serves as the basis for comparison of this method with other methods.

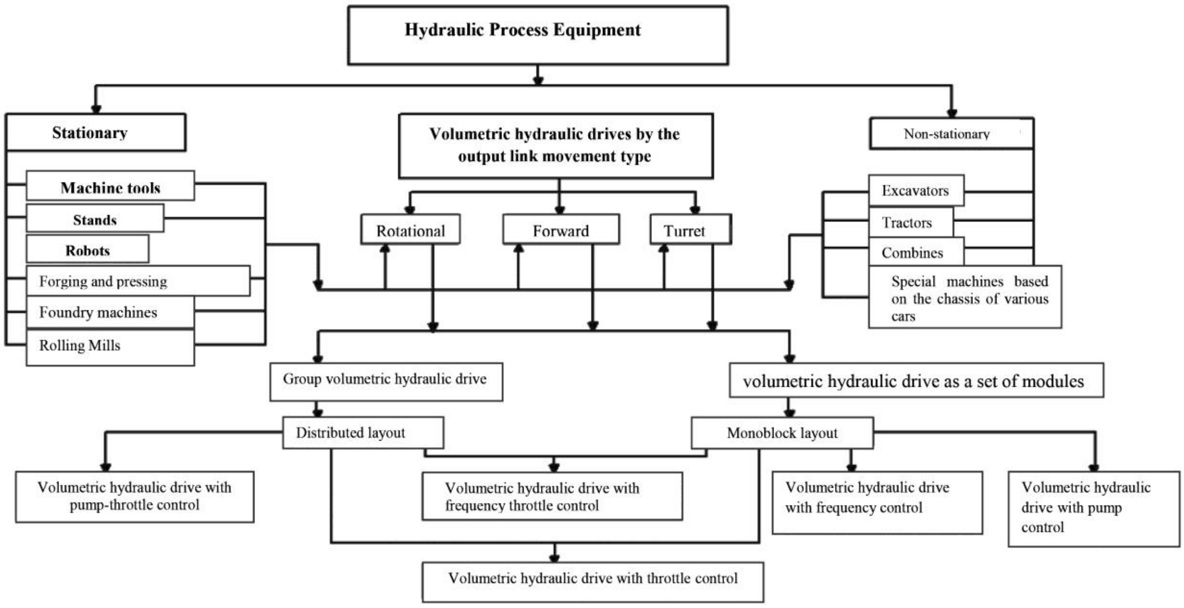

Figure 1. Process equipment hydraulic drives classification.

2. Theory

To solve the issues of competitiveness of a volumetric hydraulic drive with frequency or frequency-throttle control, it is necessary to evaluate the positive aspects and assess the disadvantages of such drives, provided that they are functionally equivalent to their counterparts. As objects for comparison, volumetric drives with frequency and pump control were chosen.

The performance indicators of these drives include:

- forward and reverse stroke with infinitely variable speed control of the output link in the achievable ranges;
- output link shutdown;
- work with positive and negative loads on the output link;
- feasibility of the executive part in the power range from 2 to 100 kW;
- the presence of one output link speed control channel;
- the same degree of dynamic stability for all compared drives;
- shutdown of the output link under load and when the main power is disconnected;
- in case of power failure, a sequence of movements by the output link with setting in a predetermined position;
- the presence of a range of stepless speed control of the output link, defined as the ratio of the output link speed to the maximum value:

$$\Delta\omega = \frac{\omega_M^{min}}{\omega_M^{max}} \tag{1}$$

where ω_M^{min} is minimal rotation speed;

- total power or energy loss: $\delta = 1 - \eta$;

where η is full drive efficiency.

- ease of drive layout on the process equipment;
- possibility of spatial separation of individual parts of the drive;
- specific cost-effective indicators: m_N – specific gravity, i.e. drive mass per power unit; $\$_N$ – specific cost, i.e. the ratio of the cost of purchasing drive components to output power; L_N is the specific overall size, i.e. drive geometrical size related to output power;
- resource and survivability of the drive;
- speed, dynamic accuracy and dynamic stiffness;
- dead zone;
- fire and explosion hazard, safety of operation and the need for special training of personnel;
- noise and vibration indicators;
- ease of formation of standard series on the application and type of movement of the output link of the drive within each value of the AEM power and / or pump;
- the presence of backlash;
- the ability to perceive alternating and shock loads;
- the degree of protection of the drive from external factors;
- the possibility of expanding the functionality using the basic circuit design of the drive;
- disposal degree and ease.

3. Results

To establish the preferred field of application of a variable frequency hydraulic drive, as compared with its main functional analogue of a pump controlled hydraulic volumetric hydraulic drive, it is necessary to solve a number of problems:

- to perform classification and analysis of possible ways to control volumetric hydraulic drives;
- to find rational structures for constructing a volume hydraulic drive with frequency control;
- to create physical and mathematical models that allow to carry out numerical experiment to study performance characteristics of volume hydraulic drive with frequency control in a wide range of rotor angular velocity;
- to identify the features of physical processes and explore possible negative phenomena that occur during the transition of the volume-rotary machine to the low and ultra-low angular velocity of the rotor;
- to identify the main types of hydraulic machines that would be preferable for use in a volumetric hydraulic drive with frequency control.

Energy efficiency and cost-effectiveness of the drive energy circuit depends on the degree of energy perfection of the pump and hydraulic motor as energy converters [7, p.387]. The width of the control range, while maintaining the maximum possible energy performance of the hydraulic machine, determines not only the quality of the drive, but often the functional suitability of this power system as part of the process equipment. Therefore, when considering the general problem of comparing the advantages and disadvantages of volumetric hydraulic drives with pumping and frequency control, special attention should be paid to energy and control indicators.

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Volumetric efficiency of the pump can be determined as follows [8, p.46]:

$$\eta_{v.p.} = 1 - \frac{q_{lk} + q_{comp}}{Q_p^T} \tag{2}$$

where q_{lk} is leakage of fluid from pressure line to intake line (leakage); q_{comp} is fluid loss due to its deformation (compressibility); Q_p^T is theoretical pump flow.

$$Q_p^T = V_{v.p}^* \overline{e_p} \left[\omega_p\right] \overline{\omega_p} \tag{3}$$

where $V_{v.p}^*$ is volume constant for 1 rad turn of the hydraulic machine shaft; $\overline{e_p}$ is relative dimensionless regulation parameter; $[\omega_p]$ and $\overline{\omega_p}$ is angular and dimensionless angular velocity of the pump shaft.

Assuming the fluid flow mode in the gap is laminar, the flow rate of the leakage can be represented as:

$$q_l k = k_v lfp = k_\mu / \mu p = (k_l k \delta^3 p) / \mu \tag{4}$$

where k_{vlf} is volumetric loss factor in the hydroline $k_{vlf} = \frac{k_\mu}{\mu}$;$k_{lk}$ is coefficient of proportionality, the same for all geometric similar machines (determined empirically); μ is fluid viscosity index; p is pressure drop in cavities; δ is equivalent gap, the energy loss on which is equal to the energy loss in all the gaps of a volume-rotary machine; k_μ is volume loss factor.

Fluid loss due to its compression is determined by the Hooke deformation equation for the volume compressed by the fluid:

$$q_{comp} = \frac{V_{comp.p}^*}{E_f} \omega_p p \tag{5}$$

where $V_{comp.p}^*$ is compressible volume reduced to 1 rad pump shaft turn; E_f is fluid elasticity modulus.

The compressible volume is the sum of the non-displaceable volume and the working volume therefore:

$$V_{(comp.p)}^* = (V_{comp.non-disp} + V_w) \frac{\omega_p}{2\pi E_f} = (V_{comp.p} + V_{v.p} \bar{e}_p) \frac{\omega_p}{2\pi E_f}. \tag{6}$$

Then the equation for the volumetric efficiency of the pump will take the form:

$$\eta_{v.p.} = 1 - p\left[\frac{2\pi k_{on}}{V_0 \overline{e}} + \frac{1}{2\pi E_f}\left(1 + \frac{V_{comp.non-disp}}{V_0 \overline{e_p}}\right)\right] \tag{7}$$

For a controlled pump operating at almost constant angular shaft velocity $\omega_{н} = [\omega_{н}] = const$:

$$\eta_{v.p.} = 1 - \frac{p}{\overline{\overline{e_p}}}\left(\frac{2\pi k_{vlf}}{V_0 \omega_p} + \frac{k_V}{2\pi E_f}\right) - \frac{p}{2\pi E_f} \tag{8}$$

where k_V is the coefficient characterizing the ratio of non-displaced and working volumes.

If there is frequency control [9, p.94], then $\overline{e_p} = 1$ and ω_p becomes the adjustable value. In this case, the volumetric efficiency is:

$$\eta_{V.p.} = 1 - \frac{2\pi p}{\left[\omega_p\right]\overline{\omega_p}} - \frac{k_{vlf}}{V_0} - \frac{1+k_V}{2\pi E_f} \tag{9}$$

Obviously, the smaller value k_V corresponds to the more perfect machine.

Determination of pump mechanohydraulic efficiency:

$$\eta_{m.h.p.} = \frac{M_p^m}{M_p^m + M_{d.f.} + M_{v.f.} + M_{c.m.}} = \left(1 + \frac{M_{d.f.}}{M_p^m} + \frac{M_{v.f.}}{M_p^m} + \frac{M_{c.m.}}{M_p^m}\right)^{-1} \tag{10}$$

where $M_{d.f.}$ is dry friction moment; $M_{v.f.}$ is viscous friction moment; $M_{c.m.}$ is charge moment; M_p^t is theoretical moment on the pump shaft; $M_p^t = \frac{V_0\overline{e_p}}{2\pi}p$.

The force moments of dry and viscous friction depend on many factors [10, p.1629].

It is proposed to round the contact friction coefficient between hard surfaces, using the Boche formula [10, p.1630]:

$$k_f = \frac{a}{1+bv} \tag{11}$$

where a and b are empirical constants determined by the standard series of hydraulic machines; v is characteristic linear velocity of mutually moving contacting surfaces.

Then the moment of resistance is [11, p.168]:

$$M_{m.r.} = \frac{k_{f0}}{1+k_{f\omega}\omega} \cdot \frac{V_0\overline{e_p}}{2\pi} p \tag{12}$$

where k_{f0}, $k_{f\omega}$ are empirical coefficients of proportionality; k_{f0} characterizes rest friction; $k_{f\omega}$ characterizes movement friction.

This ratio can be used to simulate losses in a volumetric hydraulic machine only with the following amendments:

- according to this ratio, in the absence of a pressure difference between the pressure and suction pump nozzle (pressure and drain pipe of the motor), the torque of the dry friction force is zero, which is not true. In fact, due to friction in the shaft seals and the presence of the initial clamping of parts, the moment of initial contact friction forces equals 2-3% of the theoretical moment on the hydraulic machine shaft [15, p.1768];
- this ratio does not take into account the type and change of contact friction modes (dry, semi-dry, semi-liquid, liquid), depending on the speed and force of pressing parts [12, p.699].

With due amendments, the expression can be represented as follows:

$$M_{m.r.} = \frac{k_{f0}}{1+k_{f\omega}\omega} \cdot \frac{V_0\overline{e_p}}{2\pi}\left(p_{st.0} + p\right) \tag{13}$$

where $p_{st.0}$ is conditional differential pressure, equivalent to the action of all initial contact friction forces (0,2÷0,4 MPa).

The moment of high-speed viscous friction forces is divided into a moment not related to changes in the working volume of the machine, and a moment depending on the working volume [13, p.85]:

$$M_{v.f.} = k_{1vf} + \frac{V_0}{2\pi}\omega_1 + k_{2vf} + \frac{V_0}{2\pi}\overline{e_p}\omega \tag{14}$$

where k_{1vf} and k_{2vf} are empirical coefficients characterizing the individual components of viscous friction.

The moment of resistance due to the operation of the pump feed system is [14, p.422]:

$$M_{f.s.} = \frac{V_{0fs}}{2\pi\eta_{fs}} p_{cn} = \frac{k_{fs}V_0}{2\pi\eta_{fs}} p_{fs} \tag{15}$$

where V_{0fs} is volumetric constant load of the hand pump of the feed system; p_{fs} is pressure setting of the feed system overflow valve; η_{fs} is feed pump efficiency; k_{fs} is proportionality coefficient between the volume constant of the main and the feed pump ($k_{fs} = 0{,}2 \div 0{,}4$).

As a rule, the feed system is built on the basis of a gear pump, the total efficiency of which $\eta_{fs} = 0{,}42 \div 0{,}75$, and the forward pressure limited by the setting of the overflow safety valve is 2,5÷3 MPa. The feed pump supply is very tangible and can be compared with the supply of the main pump.

This makes it possible to introduce the following dependence of the main and feed pumps constants:

$$V_{0fs} = k_q V_0 \tag{16}$$

where k_q is proportionality coefficient (for the majority of regulated hydraulic machines it is 0,15÷0,20). Hence the following system of equations:

$$\begin{cases} \dfrac{M_{f.s.}}{M_p^m} = \dfrac{V_{0fs}}{2\pi\eta_{fs}} p_{fs} \dfrac{V_0\overline{e_p}}{2\pi} p = k_q \dfrac{p_{fs}}{\eta_{fs}\overline{e_p}p} \approx \dfrac{0{,}2\cdot 0{,}25\cdot 10^6}{0{,}72}\cdot\dfrac{1}{\overline{e_p}p} = 0{,}7\dfrac{1}{\overline{e_p}p} \\ \dfrac{M_{f.s.}}{M_p^m} = \dfrac{k_{f0}}{1+k_{f\omega}[\omega]\overline{\omega_p}}\cdot\dfrac{V_0\overline{e_p}}{2\pi}(p_{st.0}+p)\cdot\dfrac{2\pi}{V_0\overline{e_p}p} = \dfrac{k_{f0}}{1+k_{f\omega}[\omega]\overline{\omega_p}}\left(\overline{p_{st.0}}+1\right) \\ \dfrac{M_{v.f}}{M_p^m} = \dfrac{1}{2\pi}\left(k_{1vf}V_0[\omega_p]\overline{\omega_p} + k_{2vf}V_0\overline{e_p}[\omega_p]\overline{\omega_p}\cdot\dfrac{2\pi}{V_0\overline{e_p}p}\right) = \dfrac{k_{1vf}[\omega_p]\overline{\omega_p}}{\overline{e_p}p} + \dfrac{k_{2вm}[\omega_p]\overline{\omega_p}}{p} \end{cases} \tag{17}$$

where $\overline{p_{st.0}}$ is relative dimensionless pressure drop, equivalent to the action of all the initial factors of contact friction in hydrolines:

$$\overline{p_{st.0}} = \frac{p_{st.0}}{p} \tag{18}$$

In [15, p.1770] is shown a graph of the contact friction coefficient k_f dependence on lubricant film thickness.

Film thickness depends on the speed of surfaces relative movement.

According to the graph given, the friction coefficient corresponds to the beginning of the movement of the machine shaft and is in the range 0,15÷0,18. With moving surfaces, the coefficient value is significantly reduced. Therefore, the constant $k_{f\omega}$ can be defined as follows:

$$k_{f\omega} = \frac{k_{f0} - k_f^{min}}{k_f^{min}\omega_f} \tag{19}$$

where ω_f is the machine rotor angular velocity, at which contact friction is minimal k_f^{min}.

Take this linear velocity value $v = 0{,}2\frac{m}{s}$, a $k_f^{min} = 0{,}08$.

Take the linear size of the hydraulic machine active part [14, p.423]:

$$l_V = \sqrt[3]{V_0} \tag{20}$$

get the relationship between linear and angular velocity:

$$\omega = \frac{v}{l_V} \tag{21}$$

To quantify viscous friction coefficients k_{1vf} and k_{2vf} let us assume that all the losses established in the mechanohydraulic efficiency are associated exclusively with viscous friction, while there is no feed system, and no contact friction at all.

Then the mechanohydraulic efficiency can be determined as follows:

$$\eta_{m.h.p.} = \left(1 + \frac{M_{vf}}{M_p^m}\right)^{-1} = \frac{M_p^m}{M_p^m + M_{vf}} \tag{22}$$

And also:

$$\frac{M_{v.f}}{M_p^m} = \frac{k_{1vf}\omega_p}{\overline{e}_p p} + \frac{k_{2vf}\omega_p}{p} = \frac{\overline{\omega_p}\left[\omega_p\right]}{p}\left[\frac{k_{1vf}}{\overline{e}_p} + k_{2vf}\right] \tag{23}$$

For optimal mode, the pump control parameters are 1.

Consequently:

$$k_{1vf} + k_{2vf} = \left(\frac{1}{\eta_{m.h.p.}} - 1\right)\frac{p}{\omega_p} \tag{24}$$

If, according to [15, p.1771], to accept the statement about the equality of these coefficients, then they can be calculated easily.

4. Findings

Further analysis will be based on the numerical solution of these equations. To do this, it is necessary to consider three schemes of the hydraulic drive of technological equipment with similar mechanical and hydraulic characteristics, where the specified control methods are implemented. Then you can make a final conclusion about the most effective of them from the point of view of energy saving.

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