Impact of undissolved gas on dynamic processes in the fluid drive for well testing and servicing research

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Abstract. This paper concerns topical issues because power oils for a hydraulic drive are the mixture of liquid and undissolved gas. This mixture can arise when tanking and during dynamic processes because of the different dissolution and evolution rates of gas in a pressure fall in some spots of power fluid flow. That is why this paper deals with the impact of the gas factor on the behavior pattern of fluid. When describing the processes fluid drive was considered as a lumped parameters system.

1. Introduction

Compressibility of power fluid effects on the behavior pattern of a fluid drive. The modulus of compression assesses quantitively the compressibility of the fluid. Non-stationary hydrodynamic processes in comparison to thermal processes routinely run much faster which allows putting to use the localized adiabatic modulus of compression of fluid in the behavior pattern of a fluid drive [1]. Power fluids of fluid drive constitute a two-phase mixture of liquid and undissolved gas (typically air). This mixture can arise when tanking and during dynamic processes because of the different dissolution and evolution rates of gas in a pressure fall in some spots of power fluid flow [1]. As air compressibility is considerably larger than liquid compressibility, modulus of compression of two-phase mixture inferior to modulus of compression of liquid, notably it essentially makes itself felt under low-pressure conditions [2]. Undissolved air increases compliance of fluid drive and contributes to pressure rise retarding in actuating elements that exert a significant impact on the speed of operation of the controlling system. Researches [1-5] are on dynamic processes of fluid-operated machines. Papers [1-5] give a fuller list of researches on the dynamic processes of fluid-operated machines. Paper [4] shows interdependence between modulus of compression of fluid drive and pressure, however dependence between modulus of compression and gas factor was not taken into consideration.

Papers [1, 2, 5] are on the model of dependence between modulus of compression of the mixture (liquid and undissolved gas) and pressure and gas factor.

\[ B_{\text{mix}} = \frac{v_l + v_g}{B_l v_l + B_g v_g} \]  

where \( V_l \) – liquid phase volume; \( V_g \) – gas phase volume; \( B_l \) – compression modulus of liquid; \( B_g \) – compression modulus of gas.
For the practical application of characteristics entered in equation (1), it is easier to express them in terms of calculating those characteristics under atmospheric pressure $p_0$. The utmost suitable for practical application formula is developed in research [2].

$$B_{\text{mix}} = \frac{(1-\alpha_g)A(B_{L,0}+AP_0)/(B_{L,0}+Ap) + \alpha_g(P_0/p)^\frac{1}{n} - \alpha_g}{B_{L,0}+Ap} \frac{A(B_{L,0}+AP_0)/(B_{L,0}+Ap) + \alpha_g(P_0/p)^\frac{1}{n}}{A(B_{L,0}+AP_0)/(B_{L,0}+Ap) + \sqrt{\frac{B_{L,0}+Ap}{A}} \frac{\alpha_g(P_0/p)^\frac{1}{n}}{A+\alpha_g}}$$

(2)

where $\alpha_g$ – volumetric content of gas under atmospheric pressure; $A$ – coefficient that depends on liquid type and temperature; $B_{L,0}$ – compression modulus of liquid under atmospheric pressure; $n$ – polytropic index.

2. Research on the impact of a gas factor on dynamic processes in the hydraulic power unit for oil and gas wells development

Research on the impact of a gas factor on dynamic processes of the fluid drive is considered an example of a hydraulic power unit for oil and gas wells development UPA-60A (Multi-Operated Lifting Unit). The fluid power system of that unit operates as follows.

Oil flows through suction hoses from tank to pumps where through pressure pipeline with backpressure valves installed on it flows to reverse hydraulic valve. Pump-motors have a valve chest safeguarding system against thrasing. Oil goes from reverse hydraulic valve either to hydraulic valve controlling hydraulic cylinders (hydraulic jacks) or to a hydraulic valve controlling motor group operation.

As the motor group works with deactivated hydraulic cylinders and under more loaded unsteady conditions, research impact of a gas factor on dynamic processes of a fluid drive of the hydraulic power unit for oil and gas wells development UPA-60A with motor group powered on. In such a case, research on the impact of a gas factor on dynamic processes of the fluid drive may be conducted on an expedited basis of UPA-60A unit shown at figure 1.

**Figure 1.** Fluid power system under research: 1 – tank; 2 – pump-motor with valve chest; 3 – backpressure valve; 4 – filter; 10 – hydraulic valve; 9 – throttle valve; 13 – rotor engine; 20 – manometer; 17 – safety valve.
During setting up a mathematical model of the fluid drive of hydraulic power unit UPA-60A take up following idealizations. The rotational speed of the pump shaft is constant, only if the driving motor has a governor component and ensures constant speed of output shaft. The length of hydraulic circuits is comparatively inconsiderable, for this reason, wave processes influencing on dynamic processes of fluid drive can be neglected. Length of intake hydraulic circuits (from tank to pumps) is not large thus its liquid resistance can be neglected.

The most opportune method for a mathematical model of fluid drive is the Cauchy method

\[
\frac{d\omega_p}{dt} = \frac{1}{J_{dr,p1}} \left[ M_{dr.en} - V_p (p_{m1} + \Delta p_1 + \rho g z_{m1} - p_t)/\pi \cdot \eta_{h,p} \right],
\]

where \( \omega_p \) – angular rotational velocity of pump shaft; \( t \) – time; \( J_{dr,p1} \) – summary inertia moment of drive actuator rotor and power takeoff rotor reduced to pump shaft and calculated with regard to inertia moment of pump rotor; \( M_{dr.en} \) – driving engine torque reduced to pump shaft; \( V_p \) – pump capacity per revolution; \( p_{m1} \) – input pressure of hydraulic motor; \( \Delta p_1 \) – supply line pressure loss; \( \rho \) – liquid density; \( g \) – free fall acceleration; \( z_{m1} \) – height of inlet fitting center of hydraulic motor, calculated from fluid free surface in tank; \( p_t \) – pressure on fluid free surface in tank; \( \eta_{h,p} \) – hydromechanical coefficient of pump efficiency.

\[
\frac{d\omega_m}{dt} = \frac{1}{J_{dr}} \left[ V_m(p_{m1} - p_{m2})\eta_{h,m}/2\pi - M_{dr} \right],
\]

where \( \omega_m \) – angular rotational velocity of hydraulic motor shaft; \( J_{dr} \) – inertia moment of work member reduced to hydraulic motor shaft and calculated with regard to inertia moment of hydraulic motor rotor; \( V_m \) – hydraulic motor capacity per revolution; \( p_{m2} \) – output pressure of hydraulic motor; \( \eta_{h,m} \) – hydromechanical coefficient of hydraulic motor efficiency; \( M_{dr} \) – torque from load action reduced to hydraulic motor shaft.

\[
\frac{dp_{m1}}{dt} = \left( 2Q_p - 2Q_v - \frac{V_m \omega_m}{2\pi} \left[ 1 + \left( \frac{1-\eta_{vol.m}}{\eta_{vol.m}} \right) \frac{p_{m1}}{p_{nom,m}} \right] \right) E_{11}/(V_{1l} + V_{1m}),
\]

where \( Q_p \) – pump capacity; \( Q_v \) – safety valve flow rate; \( \eta_{vol.m} \) – volumetric efficiency of hydraulic motor; \( p_{nom,m} \) – rated input pressure of hydraulic motor; \( E_{11} \) – reduced compression modulus in supply hydraulic line; \( V_{1l} \) – volume of supply hydraulic line; \( V_{1m} \) – volume of hydraulic motor space connected to supply hydraulic line.

\[
\frac{dp_{m2}}{dt} = \left( \frac{V_m \omega_m}{2\pi} - Q_l \right) E_{12}/(V_{2l} + V_{2m}),
\]

where \( Q_l \) – flow rate of working fluid in drain hydraulic line; \( E_{12} \) – reduced compression modulus in drain hydraulic line; \( V_{2l} \) – volume of drain hydraulic line; \( V_{2m} \) – volume of hydraulic motor space connected to drain hydraulic line.

\[
\frac{dh}{dt} = v_v,
\]

where \( h \) – valve stroke; \( v_v \) – speed of valve opening.

\[
\frac{dv_v}{dt} = \left[ S_v (p_{v1} - p_{v2})\psi_v - F_{sp} \right]/\mu_v,
\]

where \( S_v \) – cross-sectional area of valve supply canal; \( p_{v1} \) – input pressure of valve; \( p_{v2} \) – output pressure of valve; \( \psi_v \) – experimental coefficient of valve; \( F_{sp} \) – spring pressure; \( \mu_v \) – orifice coefficient of valve opening.

\[
J_{dr,p1} = J_{dr,p} + \rho_l \frac{4}{\pi d_t^2} \frac{V_m}{2\pi} \left[ 1 + \left( \frac{1-\eta_{vol.m}}{\eta_{vol.m}} \right) \frac{p_{m1}}{p_{nom,m}} \right] \frac{V_p}{m_{h,p}}
\]
where $J_{dr, p}$ – summary inertia moment of drive actuator rotor and power takeoff rotor reduced to pump shaft, calculated with regard to inertia moment of pump rotor; $l_t$ – length of supply line; $d_l$ – supply line bore.

\[
Q_p \approx \frac{V_p \omega_p}{2\pi} \left[1 - \left(\frac{\Delta p_l + \rho g z_{m1}}{p_{nom, p}}\right)(1 - \eta_{vol, p})\right],
\]

where $\eta_{vol, p}$ - volumetric coefficient of pump efficiency.

\[
Q_v \approx \mu_v \pi \left[d_v - \left(h \sin 2\beta / 2\right)\right] \sin \beta \left[2\left(p_{m1} + \Delta p_l + \rho g z_{m1} - p_t\right) / \rho\right]^{1/2};
\]

where $d_v$ – diameter of valve supply canal; $\beta$ - cone-generating angle of valve opening.

\[
Q_l \approx \mu_l S_l \left[2\left(p_{m2} - p_t + \rho g z_{m2}\right) / \rho\right]^{1/2}.
\]

$\mu_l$ - discharge coefficient of drain line; $S_l$ - cross-sectional area of drain line; $z_{m2}$ - height of outlet fitting center of hydraulic motor, calculated from fluid free surface in tank.

Quoted result of calculation mathematical model of a fluid drive of hydraulic power unit UPA-60A with load variation from the minimum value to ceiling value in a linear fashion is in figures 2, 3, 4. Figure 2 shows the dependence diagram of the rotational speed of the pump shaft and time under the different quantity of undissolved gases in power fluid. Figure 3 shows the dependence diagram of input pressure of hydraulic motor and time under the different quantity of undissolved gases in power fluid. Figure 4 shows the dependence diagram of output pressure of hydraulic motor and time under the different quantity of undissolved gases in power fluid. There is an initial period of parameter variation in fluid drive wherein parameter fluctuations with peak amplitude take place is in figures 2, 3, 4.

\[\omega, \text{rad/s}\]

\[0 \quad 0.1 \quad 0.2 \quad 0.3 \quad t, \text{s}\]

**Figure 2.** Dependence diagram of the rotational speed of the pump shaft and time with 3 different values of undissolved gases in power fluid: — 0 %; ---- 3 %; ···· 9 %.

Due to diagrams increase in the content of undissolved gas in power fluid influences on the behavior of characteristics of the fluid drive. In the process of increase in the content of undissolved gas from 0 to 9 % frequency of oscillations of the rotational speed of hydraulic motor shaft decreases and time of transient (figure 3) increases, input (figure 4) and output (figure 5) pressure of hydraulic motor behave similarly.
Figure 3. Dependence diagram of input pressure of hydraulic motor and time with 3 different values of undissolved gases in power fluid: — 0 %; ---- 3 %; ····· 9 %.

Figure 4. Dependence diagram of output pressure of hydraulic motor and time with 3 different values of undissolved gases in power fluid: — 0 %; ---- 3 %; ····· 9 %.

3. Conclusion
Accounting result suggests that in the process of the hydraulic motor shaft load increase by executive device there being of undissolved gas in power fluid leads to decreasing of the frequency of oscillations of the rotational speed of the hydraulic motor shaft, decreasing input and output pressure of hydraulic motor and increasing the time of transient.

The greatest influence of undissolved gas in power fluid appears during the initial period of hydraulic motor shaft rotation.
Mathematical model of the fluid drive of hydraulic power unit UPA-60A allows researching the influence of gas factor on changing of dynamic processes in fluid drive and to choose best values of drive unit characteristics at the design stage.

References
[1] Popov D N 2002 *Mechanics of Fluid and Pneumatic Actuators* (Moscow: Bauman Moscow State Technical University Press)
[2] Metlyuk N F, Avtushko V P 1980 *Dynamics of Fluid and Pneumatic Actuators of Vehicles* (Moscow: Machinery Construction)
[3] Mandrakov E A, Nikitin A A 2014 *Dynamics of Hydraulic Systems* (Moscow: INFRA-M; Krasnoyarsk: Siberian Federal University Press)
[4] Korobochkin B L 1976 *Dynamics of Machine Hydraulic Systems* (Moscow: Machinery Construction)
[5] Prokopyev V N 1972 *Dynamics of Fluid Actuator* (Moscow: Machinery Construction)
[6] Gorbeshko M V 1997 Development of Mathematical Models for the Hydraulic Machinery of Systems Controlling the Moving Components of Water Development Works *Hydrotechnical construction* **31** (**12**) 745–50
[7] Parr E A 1998 *Hydraulics and Pneumatics: a technician’s and engineer’s guide* 2nd ed (Oxford: Butterworth-Heinemann)
[8] Mobley R K 1999 *Fluid Power Dynamics* (Oxford: Butterworth Heinemann)
[9] Rabie M G 2009 *Fluid Power Engineering* (New York City: McGraw Hill Professional)