A Study in Vibration of a Large-Scale Hydraulic Cylinder Actuator via Numerical Simulation

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Abstract

This paper presents a study in vibration field of a large-scale hydraulic cylinder actuator via numerical simulation. A model of hydrostatic hoisting machine which is built with dynamical parameters is executed for implementation and performance evaluation. The research has applied theories of vibration and fluid mechanics in calculation, analysis and modeling. Compressibility and viscous characteristics of hydraulic oil which have impacts on system dynamics are considered. Therefore, the mathematical model of this system is equivalent to a set of mass-spring-damper in investigation. The research results deduce a practically technical prediction in design and development of hydraulic systems using large-scale cylinders.

Keywords: Large-scale hydraulic cylinder, Hydraulic actuator, Vibration, Numerical simulation.

1. Introduction

Hydrostatic hoisting machines using large-scale hydraulic cylinder are used widely in water resource control, hydropower plants, great hoisting systems. In operation, heavy load with mega dimensions and complicated geometry is hoisted/lowered or hanged at any transition position. Hence a great inertia and hydraulic cylinder with oil fulfilled become a vibration mechanism. The vibration of hydraulic actuator may combine with bulky load bending and mechanical structure shaking to cause resonance, which phenomenon may make system stuck or more seriously destroy whole system. This problem and related topics have been mentioned in some recently published researches such as: Ning Chenxiao et al. [1] denoted some typical resources of vibration and noise of hydraulic hoist; Riccardo Bianchi et al. [2] took a frequency-based approach of payload oscillation reduction in load-handling machines; Hao Feng et al. [3] had a modelling study on stiffness characteristics of hydraulic cylinder under multi-factors...However, these works did not focus deeply on vibration characteristics of large-scale cylinder.

In this work, a mathematical model of a hydraulic actuator using large-scale cylinder in hoisting machine is developed. Dynamic characteristics of the actuator are identified. Compressibility and viscous of hydraulic oil are considered basing on fluid mechanics and vibration theory. Then vibration performance of hydraulic actuator is obtained via numerical simulation as presented hereunder.

2. The system description

Figure 1 shows a typical diagram of hydraulic actuator in hoisting machine. It includes: a large-scale hydraulic cylinder vertically placed working as main actuator component, a counter balance valve for negative load hanging, a 4/3 flow directional control valve, a pressure control valve and hydraulic unit including hydraulic pump driven by electrical motor and a check valve for protecting pump without opposite flow.

The operation of the system consists of hoisting load, lowering load and hanging load at an any middle position. In hoisting load process, hydraulic pump sucks oil from tank, then push oil through check valve, 4/3 valve (b mode), check valve of counter balance valve to lower chamber of cylinder. High pressure in lower chamber makes piston rod move upward thus pull the load. For hanging load at middle position, counter balance valve is set at a pressure value which is higher than pressure value generated by load. In lowering load process, (4/3 valve in a mode), pressure of oil pumped makes pressure setting in counter balance reduce in propotion to the stated ratio of valve, allow oil in lower chamber of cylinder drain to tank.

There are many factors which can make system vibrate during these operation stages such as variable external forces, bulky load bending, stiffness of oil, counter balance setting, rotating at high speed of pump … However, this research just focuses on self-

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vibration of actuator under heavy load, stiffness and viscous damping of hydraulic oil.

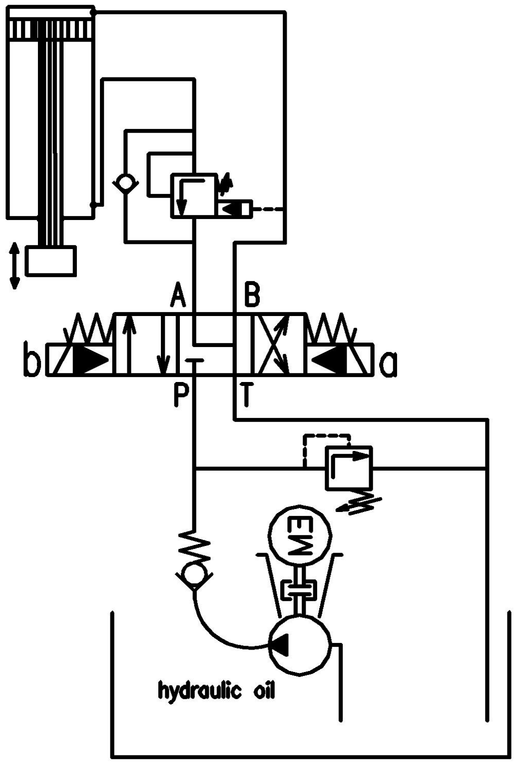

Fig. 1. Diagram of hydraulic actuator in hoisting system

3. The system dynamics and modeling

In this section, the research develops the mathematically characteristic equations of hydraulic cylinder actuator then model it in MATLAB/Simulink for convenient investigation. The hydraulic system model is established upon solid and fluid mechanics according to the aforementioned diagram. Governing equations of hoisting and lowering load based on Newton's Law are shown as:

$$m\ddot{x} = P_L A_L - P_U A_U - mg - \beta_c \dot{x} + \sum F \qquad (1)$$

$$Q_1 - A_L \dot{x} = \frac{V_L}{E} \dot{P}_L \qquad (2)$$

$$A_U \dot{x} - Q_2 = \frac{V_U}{E} \dot{P}_U \qquad (3)$$

where m is the equivalent mass of piston consisting of piston rod, load and auxiliary parts; P_L, V_L, A_L, P_U, V_U, A_U are pressure, oil volume and effective stress areas in lower and upper chamber of cylinder, respectively; g is gravity; Q_1 and Q_2 are input/output flow rates with assumption of no leakage. β_c is viscous damping coefficient of oil, x is displacement of piston; ΣF is sum of external force impacting on equivalent mass.

In stage of hanging load, Eq. (2) and Eq. (3) are neglected. The equivalent mass may oscillate around equilibrium position. The volumes of oil in chambers become a spring with a certain stiffness. It leads Eq. (1) to:

$$m\ddot{x} = P_L A_L - P_U A_U - mg - \beta_c \dot{x} - kx + \sum F \qquad (4)$$

where k is stiffness of system. When hydraulic cylinder stops suddenly at any middle position while hoisting or lowering load, it causes vibration due to inertial characteristics. In vibration theory, it is simply in the form of equation:

$$m\ddot{x} + c\dot{x} + kx = F \qquad (5)$$

with c is system damping coefficient. If $F=0$, it is free damped vibration. Otherwise, it is forced damped vibration. The vibration has interactive influence on whole mechanism/system in which hydraulic cylinder is driving actuator. Resonance can occur if frequency of actuator equals to natural frequency of bulky load, mechanical structure and so on. Anyways, this phenomenon may destroy machine, system and structure seriously. Identifying whole parameters of governing equation let us be able to investigate the dynamics of the actuator.

The stiffness of actuator is generated by compressibility of hydraulic oil in the chambers. Hydraulic oil is compressible due to the formula:

$$\frac{\Delta V}{V} = -\frac{\Delta P}{E} \qquad (6)$$

with ΔV is the change in volume, V is original volume, ΔV is pressure increase, E is elasticity of oil, a minus sign is used due to volume decrease. This compressibility of hydraulic oil can be considered as a spring (illustrated in Figure 2) which stiffness is defined as:

$$k = \frac{\Delta P}{\Delta L} = \frac{\Delta P \times A}{\frac{\Delta V}{A}} = \frac{\Delta P \times A^2}{\frac{\Delta P \times V}{E}} = \frac{A^2 \times E}{A \times S} = \frac{A \times E}{S} \qquad (7)$$

where $\Delta L = \Delta V/A$ is the cylinder displacement, A is effective area, S is longitudinal length of cylinder chamber. Because diameter of cylinder is much bigger than hose, then the research neglected effect of a small oil volume in hoses. Furthermore, the elasticity module of steel is nearly 200 times larger than the elasticity of oil. Hence in this research, hydraulic cylinder and piston rod are considered as rigid bodies without deformation.

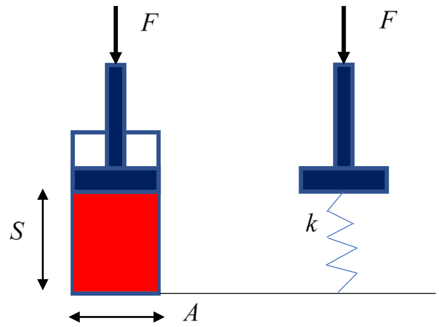

Fig. 2 Equivalent model for stiffness of hydraulic oil

The other parameter in oscillation system is damping coefficient. Identification of damping coefficient of hydraulic actuator is not easy due to complexity of variable of factors. According to Gabriela Koreisova [4], there is resistance against a straight motion of piston in cylinder caused by viscous friction. It is the resistance of the solid energy bearer and linearly dependent on the velocity of motion following formula:

$$F_b = bv \tag{8}$$

in the equation, b (Ns/m) is viscous friction or a viscous damping coefficient. For simplifying very complicated problems, a piston/cylinder configuration shown in Figure 3 is used and damping coefficient identification is implemented upon following assumptions:

- A piston with length l and radius r_1 in a cylinder with radius r_2 and the gap $2\delta=d_2-d_1$.
- Hydraulic oil fulfills annulus gap, dynamic viscosity $\mu=\rho\upsilon$. Radius of the gap $r=0.5(r_1+r_2)$ and $\delta<<r$.
- The distribution of hydraulic oil is linear in annular gap.
- The leakage is zero because of seals.

Hence, Gabriela indicated that damping coefficient for the friction force can be calculated as:

$$\tau = \frac{F_b}{S} = -\mu\frac{dv}{dr} \tag{9}$$

and the strain rate can be expressed due to linear distribution of the fluid speed:

$$\frac{dv}{dr} = \frac{v}{\delta} \tag{10}$$

We get the formula for the friction force as:

$$b = \pi\rho\upsilon\frac{dl}{\delta} \tag{11}$$

By substituting constructional parameters:

$$k_\delta = \frac{d}{\delta}\ ;\ k_l = \frac{l}{d} \tag{12}$$

We have

$$b = \pi\rho\upsilon k_\delta k_l d \tag{13}$$

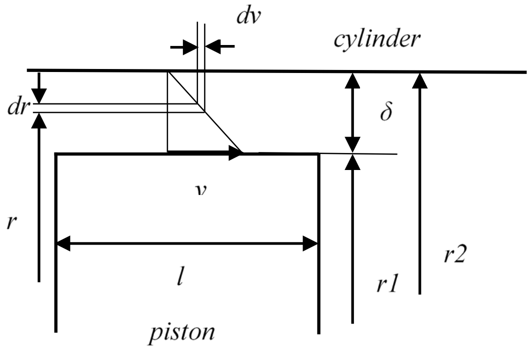

Fig. 3 Dimensions of the piston and cylinder

In the above equation, the constructional parameters are in the range of:

$$k_\delta = 2.10^3 \div 4.10^4\ ;\ k_l = 0.5 \div 1.5 \tag{14}$$

with k_δ can be chose proportionally to cylinder diameters. Because the hydraulic force in circuits can be expressed as a product of the pressure and active area. It leads to the hydraulic damping force:

$$F_{bH} = \Delta pS = bv = b\frac{Q}{S} \tag{15}$$

with Δp is pressure attenuation and it can be defined as:

$$\Delta p = \frac{b}{S}v = \frac{b}{S^2}Q = R_bQ \tag{16}$$

When the active piston area is in the no- rod chamber, the linear resistance is:

$$R_b = \frac{b_1}{S^2} = \frac{16}{\pi}\rho\upsilon\frac{l}{\delta D^3} \tag{17}$$

D is the cylinder bore diameter. Similarly, when active piston area is in piston rod chamber (d is rod diameter), the linear resistance is:

$$R_b = \frac{16}{\pi}\rho\upsilon\frac{d.l}{\delta(D^2-d^2)^2} \tag{18}$$

There is other damping coefficient against motion. It relates to leakage flow rate. However, this factor is neglected because of sealing with caulking. Sealing elements also cause friction which depends on normal pressing force on cylinder bore and friction coefficient between rubber seal and steel. This friction should be considered in calculation as a working pressure attenuation. This value is identified in engineering handbook and technical guide from manufacturers.

From the above formulas, we can calculate stiffness and damping coefficient in hydraulic actuator. However, the coefficient k in Eq. (5) is not a constant due to the position of piston in cylinder and Bulk modulus of oil; the coefficient c generally recognized as a constant for each configuration of hydraulic cylinder and type of hydraulic oil. Solving Eq. (5) via analytical calculation is unsuitable. The equations are expressed and solved in Simulink model shown in Figure 4.

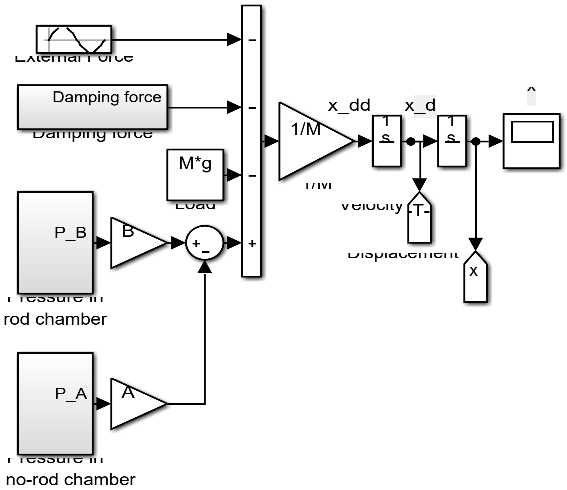

Fig. 4. Simulink model of system

4. The system simulation and results

In the previous section, a mathematical model of system is developed and built up in MATLAB/Simulink. This section shows numerical simulation and response of system corresponding to initial conditions. The actuator parameters are shown in Table 1:

Table 1. The paramters of system

Quantity/Factor	Value (unit)
Equivalent Load	~50010 (kg)
Hydraulic cylinder stroke	10 (m)
Cylinder bore diameter	0.3 (m)
Piston rod diameter	0.2 (m)
Hoisting/Lowering speed	2 (m/minute)

According to acquisition of dynamic calculation, the stiffness and damping coefficient of system depend on certain position of piston and characteristics of hydraulic oil. Therefore, the research takes investigation at some position of piston displayed by value H- distance between piston and lower bottom of cylinder. Specifications of 3 typical kinds of hydraulic oil used in simulation are in Table 2:

Table 2. The specifications of hydraulic oil

Hydraulic oil	Specifications
ISO VG32	- Density: 844.4 (kg/m^3) - Viscosity: 15.9869 (cSt) - Bulk modulus 1266530000(Pa)
Skydrol LD-4	- Density: 961.873 (kg/m^3) - Viscosity: 7.12831 (cSt) - Bulk modulus 1242850000(Pa)
Skydrol 500B-4	- Density: 1016.6 (kg/m^3) - Viscosity: 6.95191 (cSt) - Bulk modulus 1331860000(Pa)

In each case, hydraulic cylinder starts to hoist or lower the equivalent load at an initial position and stops after 5 seconds. Performance of system in hoisting process is shown in Figure 5, Figure 6 and Figure 7. Figure 8, Figure 9 and Figure 10 are 3 cases of system performance in lowering process.

Amplitude and phrase of vibration in each case are plotted corresponding to each type of hydraulic oil. Difference in density, Bulk modulus and viscosity of hydraulic oils makes difference in frequency, steady state error in position control. The difference in performance of system also depends on the position of piston that defines the stiffness of vibration mechanism. In other word, it is a multi-variable function. Via acquired data statistics, numerical results fit well with governing equations, hence can indicate characteristics of system such as damped natural frequency approximately, magnification factor (if any) and so on. These are useful for avoiding unexpected phenomenon as destructive resonance in system design and components selection.

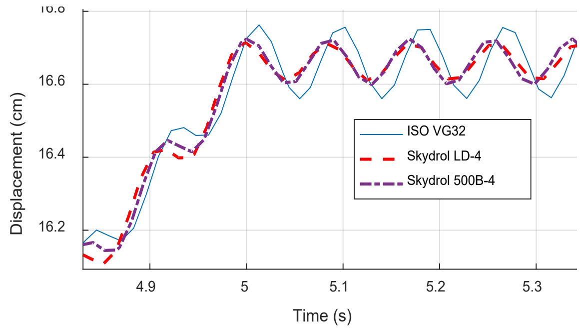

Fig. 5. Case 1: Start hoisting at H = 0, stop after 5s

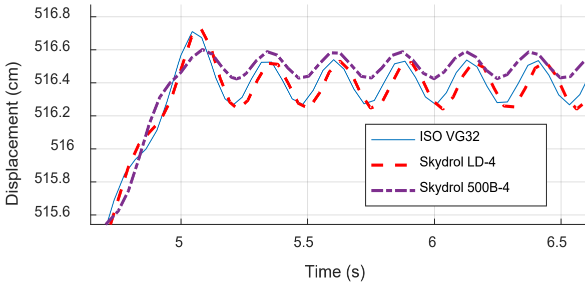

Fig. 6. Case 2: Start hoisting at H = 5m, stop after 5s

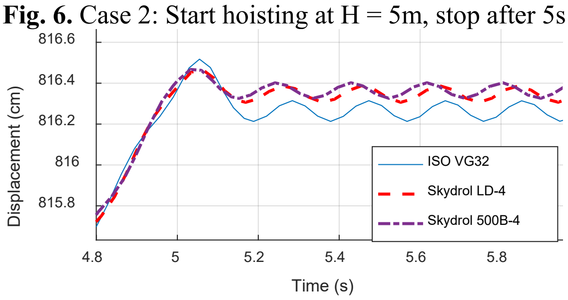

Fig. 7. Case 3: Start hoisting at H = 8m, stop after 5s

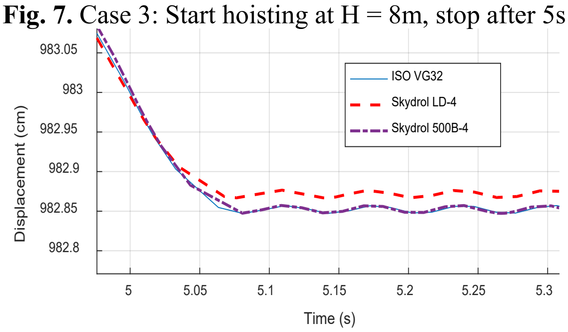

Fig. 8. Case 4: Start lowering at H=10m, stop after 5s

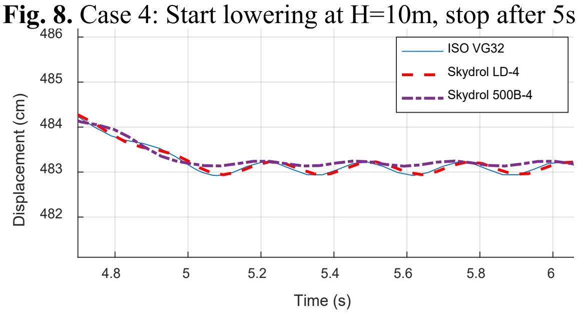

Fig. 9. Case 5: Start lowering at H=5m, stop after 5s

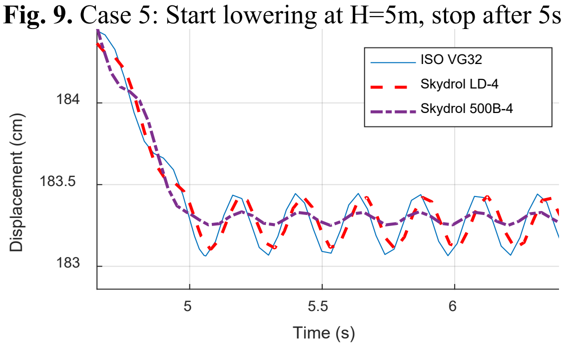

Fig. 10. Case 6: Start lowering at H=2m, stop after 5s

4. Conclusion

In this work a physics-based model of a large-scale hydraulic cylinder actuator has been established. Response of system that is complicated to be solved analytically can be obtained quickly with flexible input parameters. Investigation results via numerical simulation are suitable with theory. These results are useful for hydrostatic systems in early stages of design optimization, stability, etc. Therefore, this work contributes a serviceable base for prospective projects.

Acknowledgments

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