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Safety estimation of high-pressure hydraulic cylinder using FSI method

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Abstract: Hydraulic cylinder is a primary component of the hydraulic valve systems. The numerical study of hydraulic cylinder to evaluate the stress analysis, the life assessment and the performance of operation characteristics in hydraulic cylinder were described. The calculation of safety factor, fatigue life, piston chamber pressure, rod chamber pressure and the change of velocity of piston with flow time after the beginning of hydraulic cylinder were incorporated. Numerical analysis was performed using the commercial CFD code, ANSYS with unsteady, dynamic mesh model, two-way FSI (fluid－structure interaction) method and $k-\varepsilon$ turbulent model. The internal pressure in hydraulic cylinder through stress analysis show higher than those of the yield strength.

Key words: two-way fluid－structure interaction; high-pressure hydraulic cylinder; durability life; safety estimation; dynamic mesh

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1 Introduction

Hydraulic systems are widely used around the world for different applications. Hydraulic system is used to transfer energy between different parts of a system, for example in vehicles, plant, industrial machine, etc. Hydraulic system is a device for the mechanical work converting the mechanical energy into fluid energy by applying pressure energy of the working fluid then converting a fluid energy into mechanical energy again via the actuator. Valve actuator, one of various kinds of actuators, plays an important role in hydraulic system for controlling valve with high pressure and its structure is determined by the source of power. The valve actuators are classified by supplied power, such as electric, pneumatic, and hydraulic actuators. Each actuator has unique features, so one must consider operating environment, such as control, stability, cost, etc.

Recently, there has been a tendency of using an electro－hydraulic actuator that combines electric actuators and hydraulic actuators. Fig. 1 shows electro－hydraulic control actuator system. Electro－hydraulic actuators replace typical hydraulic systems with self-contained actuators operated solely by electrical power. In typical hydraulic systems, additional system using the hydraulic pump for pressuring working fluid was required for operation. However, in the electro－hydraulic system, two different systems were combined and

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simplified, so that the stability and reliability were improved.

Fig. 1 Electro – hydraulic actuator system

Electro – hydraulic actuator is used on process valves with strict requirements for dynamics, positioning forces and safety. It had been originally developed for the aerospace industry, but now it is used for various industrial fields. The hydraulic cylinder is one of the components of an electro – hydraulic actuator. It consists of two fluid chambers and a piston with a rod. A hydraulic cylinder has two small ports: one is the inlet, the other is the outlet. As shown in Fig. 2, one port is located in the piston chamber and the other is located in the rod chamber.

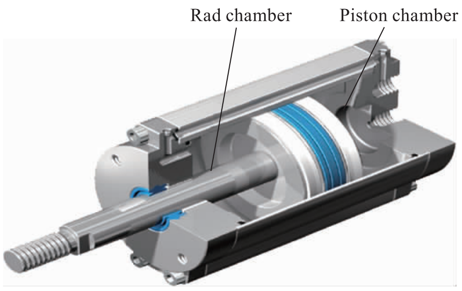


Fig. 2 Double acting hydraulic cylinder

When pressurized oil is forced into the piston side chamber, the cylinder will take an outward power stroke and the oil in a rod chamber is pushed out to oil tank. When the oil flow is reversed and the pressurized oil from the pump enters the rod chamber, the cylinder will then take an inward power stroke in a reversed direction and the oil in the piton chamber is pushed out to oil tank. The oil flow to the double acting cylinder is usually controlled by a reversible pump. One of the limitations of a cylinder is that it offers a fixed length of piston stroke.

To improve the performance and structural stability, many studies have been carried out. JUNG et al. [1] simulated the motion of this rod with fluid – structure interaction (FSI) method before manufacturing for design verification and pump sizing. JAMBHRUNKAR et al. [2] described the numerical study of hydraulic cylinder to evaluate the performance of cushioning in a cylinder. The study incorporates the calculation of cushion chamber pressure, change of velocity of piston with flow time after the beginning of cushioning in cylinder and the values obtained by computational analysis are compared with experimental results. HIREMATH et al. [3] studied a new approach for modeling the FSI of servo valve component actuator. TORBACKI[4] executed numerical strength and fatigue life analysis applied to hydraulic cylinder. MEIKANDAN et al. [5] studied theoretical analysis of tapered pistons in high-speed hydraulic actuators. Here an attempt is made to systematically analyze three types of profiles commonly considered for high-speed actuators, using Reynolds differential equation. Analyses are made using the finite element method. XUAN et al. [6] studied dynamic friction behaviour of hydraulic cylinder. Using hydraulic cylinders with different packing materials and sizes, friction characteristics are experimentally investigated under different magnitudes of external load. MARCZEWSKA et al. [7] studied practical fatigue analysis of hydraulic cylinders and some design recommendations. NICOLETTO et al. [8] studied the failure of a heavy-duty hydraulic cylinder. Alternative designs were developed and demonstrated to achieve a considerably longer operational life. MUVENGEI et al. [9] studied a simplified bond graph model of the inter-actuator interactions in a multi-cylinder hydraulic system. In this study, the structural safety of the hydraulic cylinder, which is a high-pressure valve actuator component, was evaluated. And this assessment is to be reflected in the design[10].

2 Theoretical analysis

In order to make the analysis of the hydraulic system, it is assumed the following conditions:

1) Supply inlet pressure of hydraulic pump is constant as p_1 and outlet pressure of p_2 is zero.

2) Pressure distribution at each chamber of the cylinder is uniform.

3) Cavitation does not occur.

Hydraulic cylinders are usually subjected to many internal and external forces. Fig. 3 shows a schematic diagram of hydraulic cylinder connected to a load.

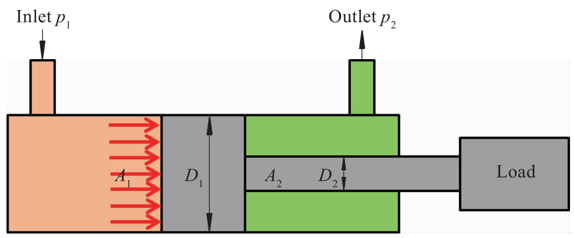


Fig. 3 Schematic of hydraulic cylinder

Frictional force is neglected because it is too small than the supply pressure. Here A_1 and A_2 can be obtained by Eqs. (1) and (2).

$$A_1=\frac{\pi D_1^2}{4}, \tag{1}$$

$$A_2=\frac{\pi(D_1^2-D_2^2)}{4}. \tag{2}$$

Eq. (3) shows power output by hydraulic pressure.

$$p_1A=F, \tag{3}$$

Flow rate discharged from the hydraulic cylinder can be calculated by Eq. (4).

$$Q=Av, \tag{4}$$

where v is the piston velocity.

3 Numerical analysis

The 3-D modeling which is based on 2-D drawings was conducted by using SolidWorks as shown in Fig. 4.

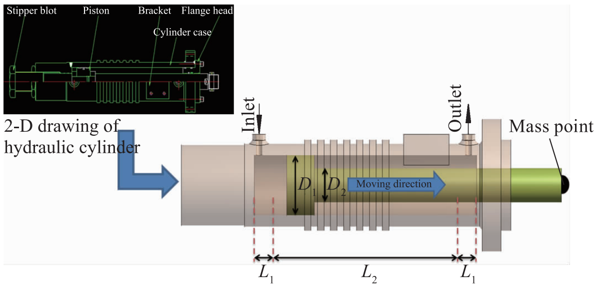


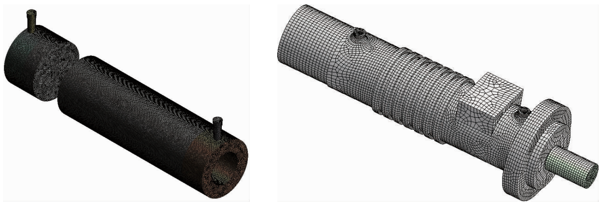
Fig. 4 3-D modeled hydraulic cylinder

In order to analyze the structural deformation with the internal pressure change of a hydraulic cylinder, a two-way FSI analysis was performed. Moreover, hydraulic cylinder system has a piston which moves in the fluid region and the piston movement is determined by flow in the hydraulic cylinder. Dynamic mesh method is also conducted to describe a motion of the piston regarding these flow characteristics. Detailed dimensions and boundary conditions are specified in Tab. 1. L_1 is free stroke, L_2 is a stroke length, W is weight of mass point.

Tab. 1 Geometric details and boundary conditions applied in this study

Parameters	p_1/MPa	p_2/MPa	W/kg	D		L	
				D_1/mm	D_2/mm	L_1/mm	L_2/mm
Values	35/45/55	0.2	400	160	90	70	630

In order to evaluate the structural stability of the hydraulic cylinder in the proof pressure (55 MPa), operating pressure was increased until the proof pressure. The properties of oil, which are used in the simulation are as follows: density $\rho=889$ kg/m^3, viscosity $\mu=1.06$ kg/m · s. The properties of steel are $\rho=200$ kg/m^3, Young's modulus $E=200$ Pa, Poisson's ratio $\nu=0.3$, tensile yield strength $\sigma_s=2.5\times10^8$ Pa, tensile ultimate strength $\sigma_b=4.6\times10^8$ Pa. Grid systems were generated using ANSYS Meshing. Fluid domain is prepared using the re-meshing method, which is one of the dynamic mesh methods. Re-meshing method is possible to use only tetrahedron grid. Fig. 5a shows the grid system of fluid domain which consists of tetrahedron. The grid system of structural domain consists of hexahedron as shown in Fig. 5b. The number of elements in the fluid domain and structural domain are about 3 000 000 and 60 000, respectively. Numerical analysis was performed using the commercial CFD code, ANSYS 15.0.



(a) Fluid domain (b) Structural domain

Fig. 5 Grid system of fluid domain and structural domain

4 Results and discussions

4.1 Motion characteristics of hydraulic actuator

The hydraulic cylinders with various values of the inlet pressure are simulated under the same inlet diameter and outlet diameter. When the inlet pressure is $p_1 = 35$ MPa, the transient pressure p_t distributions of hydraulic cylinder were depicted, as shown in Fig. 6. The time is denoted as τ.

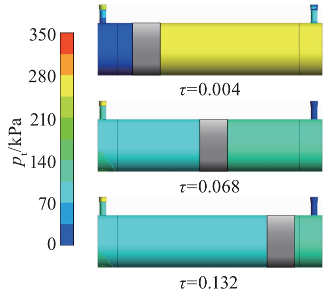


Fig. 6　Transient pressure distribution of hydraulic cylinder when inlet pressure is $p_1 = 35$ MPa

Fig. 7 shows the simulated the mass flow rate through inlet and outlet port.

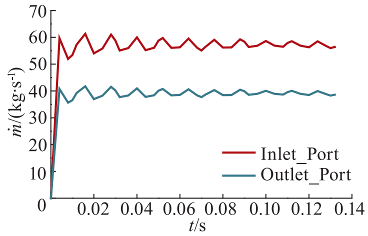


Fig. 7　Variation of mass flow rate at inlet and outlet ports

Fig. 8 shows the piston displacement and even here the above effect can be seen, where s is the piston displacement.

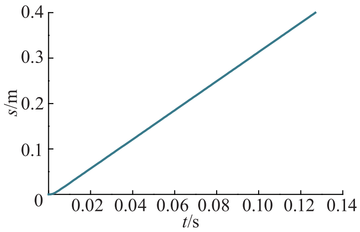


Fig. 8　Variation of piston displacement

The piston starts moving only when the pressure in the piston chamber overcomes the applied load. The piston velocity obtained from the finite element simulation is compared with that obtained by dividing the flow rate by the area of cross section of the piston. This is shown in Fig. 9. Even here, the piston velocity obtained through the finite element simulation matches well with that calculated result based on the fluid flow into the piston chamber and the piston area.

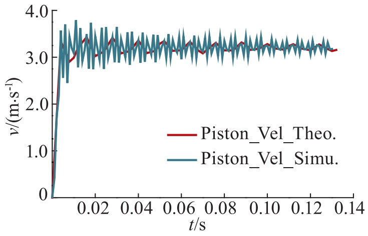


Fig. 9　Variation of piston velocity

4.2　The stress and deformation analysis by internal pressure

The thickness of the safe cylinder in the proof pressure was determined by using the following Barlow equation:

$$t = \frac{pD_1}{2\left(\frac{\sigma_s}{SF}\right)}, \tag{5}$$

Here, the cylinder thickness is denoted as t, the proof pressure is p, the cylinder inner diameter is D_1, the yield strength is σ_s and the safety factor is SF. From Eq. (5), the cylinder thickness can be calculated as 0.052 8 m when the safety factor is 3. However, the cylinder thickness of the newly designed model in this study showed 0.06 m, which is thicker than the theoretical value of 0.052 8 m. Therefore, it may be concluded that the designed hydraulic cylinder in this study is sufficiently safe. The value of the finite element analysis results was evaluated equivalent stress based on the von Mises yield criterion. Von Mises yield criterion is a theory that is most often used in the ductile material. The structural deformation analysis was carried out for structural safety assessment of the internal pressure on the hydraulic cylinder. The equivalent stress was analyzed with each time step, and we found the time step that indicated the maximum equivalent stress. The maximum equivalent stress taking place in the vicinity of port is shown in Fig. 10 for different inlet pressure conditions.

The maximum equivalent stress has a value higher than the yield strength that can induce a permanent deformation. The maximum equivalent stress was approximately 1.5 GPa when the inlet pressure was $p_1 = 55$ MPa, as shown in Fig. 10c. Similarly analyzed for deformation, the maximum deformation s is shown in Fig. 11.

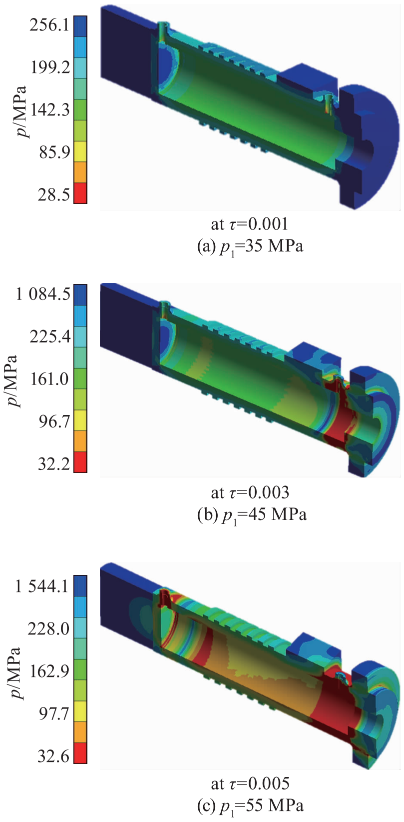


Fig. 10 The maximum von Mises stress distribution for different inlet pressure conditions.

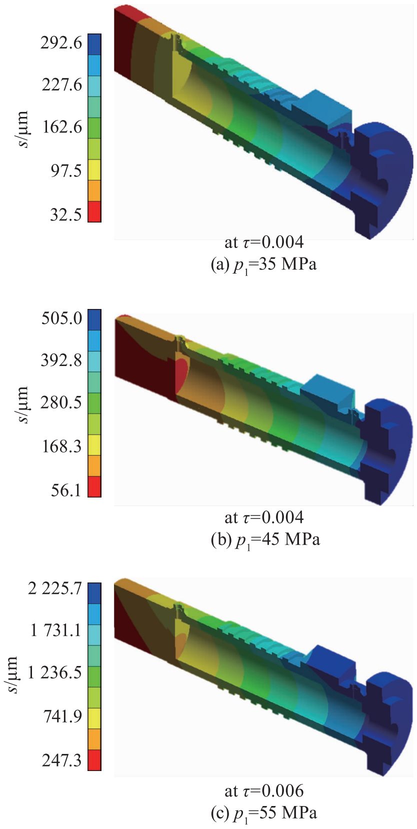


Fig. 11 The maximum total displacement distribution for different inlet pressure conditions

In addition, the maximum total displacement was approximately 0. 000 22 m when the inlet pressure was p_1 = 55 MPa, as shown in Fig. 11c. The results of a durability life N due to pressure change are shown in Fig. 12. The stress which exceeds the yield strength was calculated by numerical analysis. It is confirmed that reinforcement design concerning the hydraulic cylinder should be conducted to improve the durable quality. It is also confirmed that the flow region design of cylinder port, in which stress is concentrated, should be conducted smoothly because it can reduce the fluid resistance.

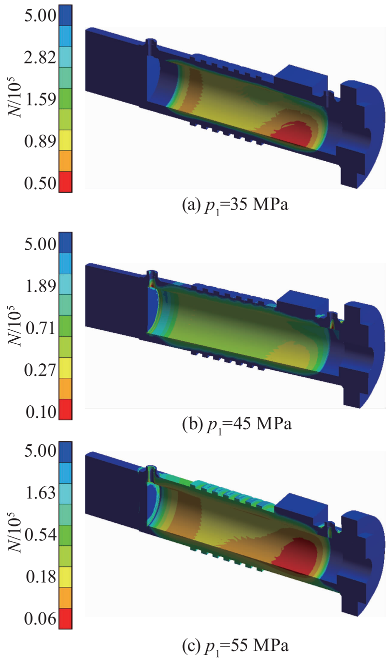


Fig. 12 Durability life distribution for different inlet pressures

5 Conclusions

1) Finite element approach was adopted to model the fluid – structure interaction of hydraulic cylinder. The dynamic of hydraulic cylinder was presented as the variation of chamber pressure, mass flow rate and piston velocity. The piston velocity obtained through the finite element simulation matches well with the results based on the fluid flow into the piston chamber and the piston area.

2) The fluid – structure interaction analysis was performed to depict the movement of the actuator. And

the stress, generated by hydraulic pressure, is analyzed in the cylinder. In result, the stress that exceeds the yield strength has occurred in some regions. Therefore, it must be designed to reconsider the thickness of the hydraulic cylinder.

3) High stress regions occurred because of the flow resistance generated in the vicinity of port. Thus, in order to reduce flow resistance, optimal design is required at the internal flow field configuration of the port. In addition, a design of the thickness of the hydraulic cylinder should be reinforced to withstand proof pressure.

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Nomenclature

A_1, A_2 Cross-sectional areas of cylinder and piston (m^2)

D_1, D_2 Diameters of cylinder and piston (m)

$\dot{m}$ Mass flow rate ($kg \cdot s^{-1}$)

p_1, p_2 Inlet and outlet pressure (Pa)

t Time (s)

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