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FSI simulation and experimental validation of nonlinear dynamic characteristics of a gas-pressurized hydraulic shock absorber

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Abstract. The structure, fluid and gas dynamic models of a gas-pressured hydraulic shock absorber are established to form a 3-D FSI (Fluid-Structure Interaction) simulation model, which can be used to predict the working characteristics and kinematical responses of the valve plates, as well as the flow characteristics, transient fluid pressure and velocity distribution of the shock absorbers. The working characteristics obtained from the simulation model are compared with the experimental results to prove the effectiveness of the simulation model. It is shown that the dynamic characteristics of the shock absorber can be predicted quite accurately by the 3-D FSI simulation model, which is meaningful to the understanding of the inner working mechanism of the shock absorber system.

1. Introduction

Ride comfort and handling of a car is directly influenced by the performance of automobile shock absorbers. There are many kinds of automotive shock absorbers, such as twin-tube, monotube, ER and MR shock absorbers. Monotube gas-pressurized hydraulic shock absorber is a new type of liquid-resistance shock absorber, which is developed in the 1960s [1]. A typical monotube gas-pressurized hydraulic shock absorber is composed by adding a floating piston in the compression chamber to form a gas chamber with high pressure inside, which is used to compensate the volume change caused by the piston rod [2].

In the traditional lumped-parameter models, simulation of shock absorbers' dynamic characteristics is carried out using semi-empirical formulas of fluid mechanics [3-6], which owes limited accuracy and can't accurately describe the dynamic characteristics of the shock absorber system. The FSI simulation model can describe the structure and fluid of the shock absorber more accurately [7-10]. Moreover, information of flow field and structure responses that can't be obtained from the traditional models can be obtained to help with the understanding of the inner working mechanism of the shock absorber.

By using the finite element method, an FSI simulation model of a monotube gas-pressurized hydraulic shock absorber designed for a special heavy-duty vehicle is established and analysed. Direct FSI method and Leader-Follower parameter moving mesh control strategy are used in the simulation model. The working characteristics, kinematical response of the valve plates, flow characteristics, transient fluid pressure and velocity distribution of shock absorbers are obtained from the model,



which is helpful to the understanding of the inner working mechanism of the monotube gas-pressurized hydraulic shock absorber

2. Mathematical model of the shock absorber

2.1. Fluid mathematical model

The motion of the oil and gas in the shock absorber is governed by the Navier-Stokes equations. The conservative forms for mass, momentums and energy equations in a fixed Cartesian coordinate frame of reference can be expressed as follows [11],

$$\begin{aligned}\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{v}) &= 0 \\ \frac{\partial \rho \mathbf{v}}{\partial t} + \nabla \cdot (\rho \mathbf{v} \mathbf{v} - \boldsymbol{\tau}) &= \mathbf{f}^B \\ \frac{\partial \rho E}{\partial t} + \nabla \cdot (\rho \mathbf{v} E - \boldsymbol{\tau} \cdot \mathbf{v} + \mathbf{q}) &= \mathbf{f}^B \cdot \mathbf{v} + q^B\end{aligned}\quad (1)$$

, where, ρ is the density, \mathbf{v} is the velocity vector, $\boldsymbol{\tau}$ is the stress tensor, \mathbf{f}^B is the body force vector of the fluid medium, E is the specific total energy, \mathbf{q} is the heat flux and q^B is the specific rate of heat generation.

Additional equations, i.e. the state equations, must be provided to obtain a closed system for solution variables.

Oil in the shock absorber can be modelled by a slightly compressible fluid model. Its state equations are expressed as follows,

$$\begin{aligned}\rho &= \rho_0 \left(1 + \frac{p}{\kappa} \right) \\ e &= C_v \theta\end{aligned}\quad (2)$$

, where, ρ_0 is the density at $p=0$, κ is the fluid bulk modulus of elasticity, C_v is the specific heat at constant volume, θ is temperature.

Gas in the shock absorber can be modelled by a compressible fluid model. Its state equations are expressed as follows,

$$\begin{aligned}p &= (C_p - C_v) \rho \theta \\ e &= C_v \theta\end{aligned}\quad (3)$$

, where, C_p is the specific heat at constant pressure.

2.2. Solid mathematical model

The motion of the solids of the shock absorber is governed by motion equations,

$$\rho_s \frac{\partial^2 \mathbf{u}}{\partial t^2} = \nabla \cdot \boldsymbol{\tau}_s + \mathbf{f}_s^B \quad (4)$$

, where, ρ_s is the density, \mathbf{u} is the displacement vector, $\boldsymbol{\tau}_s$ is the stress tensor, \mathbf{f}_s^B is the body force vector of the solid medium.

2.3. FSI mathematical model

In FSI analyses, the fluid boundaries are influenced by the deformation of the solid domain, while the fluid forces are applied onto the solid domain through the FSI boundaries.

Direct method is used in the simulation model, in which the fluid and solid solution variables are fully coupled. The fluid and solid equations are combined in one system, and linearized in a matrix system. The matrix system can be written as,

$$\begin{bmatrix} \mathbf{A}_{ff} & \mathbf{A}_{fs} \\ \mathbf{A}_{sf} & \mathbf{A}_{ss} \end{bmatrix} \begin{bmatrix} \Delta \mathbf{X}_f^k \\ \Delta \mathbf{X}_s^k \end{bmatrix} = \begin{bmatrix} \mathbf{B}_f \\ \mathbf{B}_s \end{bmatrix} \quad (5)$$

, where, \mathbf{X} is the solution vector of the coupled system, \mathbf{A} and \mathbf{B} are the linearized matrixes.

The convergence of the coupled system is controlled by the stress and displacement tolerances, relaxation factors and convergence criteria.

3. FSI model of the shock absorber

The monotube gas-pressurized hydraulic shock absorber used in this paper is designed for a special heavy-duty vehicle, which needs to meet the large stroke and damping force requirements. A simplified model of the damper is shown in Figure.1. The valve system includes a piston, two valve plates and two compression springs. As the piston is forced to move in and out along the fluid-filled cylinder, a pressure difference is developed across the piston causing the fluid to flow through the valves on the piston. The piston moves in the cylinder during the compression stroke, causing the volume decrease and pressure increase in the compression chamber, as well as the volume increase and pressure decrease in the rebound chamber. The piston moves out the cylinder during the rebound stroke, causing the volume increase and pressure decrease in the compression chamber, as well as the volume decrease and pressure increase in the rebound chamber. The volume change caused by the piston rod is compensated by the air chamber.

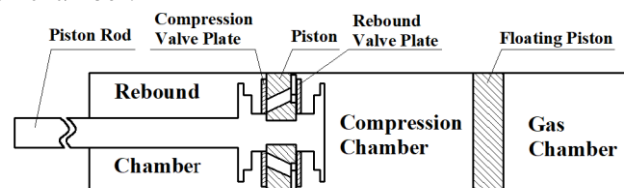


Figure 1. Simplified structure of the shock absorber.

The fluid geometry model is obtained from Boolean operation of the structure. In order to improve the FEA model accuracy and calculation efficiency, 1/6 FEA models of structure and fluid are established for FSI dynamic simulation analysis. Hexahedral meshes are used for the structure model except the piston body, whose structure is too complex to be meshed with hexahedral elements, so the tetrahedral meshes are used. In order to limit the number of finite elements, only structure associated with the FSI surfaces of the piston body is meshed. Hexahedral meshes are used for the fluid model. Throttling gaps are generated when the valve plates are opened, which should be reserved between the valve plates and the piston in the FEA model. In order to capture the drastically changed pressure and velocity in the gap, the gap is discretized into four layers of meshes to meet the deformation here. The average element size is only 0.05mm at the gap. Pressure and velocity in the gas, compression and rebound chamber is relatively uniform, so elements with relatively large size are adopted at these regions. The established structure and fluid FEA models are shown in Figure.2.

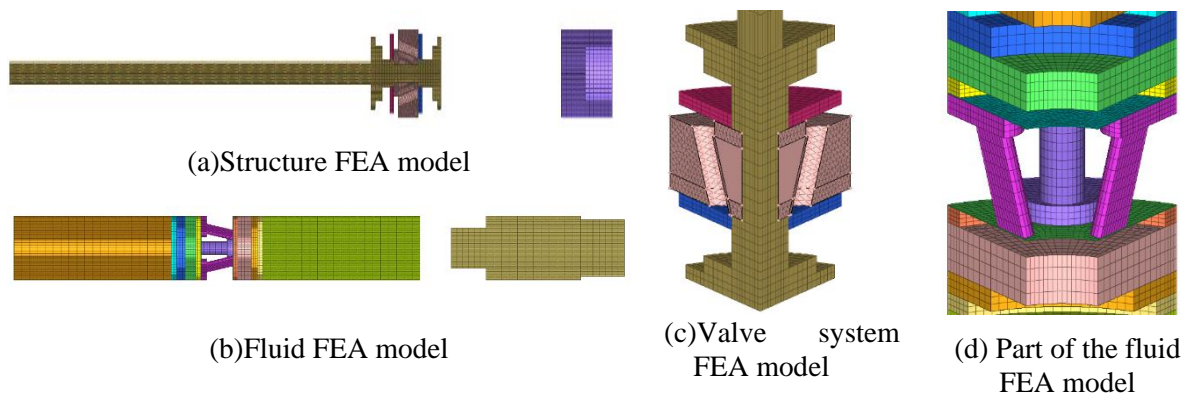


Figure 2. FEA mesh models.

In the structure model, a displacement excitation of the sine wave form with the frequency 5 Hz and amplitude 32 mm is applied to the piston rod, and a sinusoidal displacement excitation of 0.1Hz is applied to the piston rod in the beginning 2.5 seconds of the simulation process to ensure the formal simulation's starting excitation speed is zero, which is helpful to the convergence of the FSI analysis. The displacement excitation is shown in Figure.3. The spring finite elements are used to replace the structures of the compression springs. The designed stiffness characteristics and the experimental results are shown in Figure.4, which shows great agreements with each other. There are two contact pairs defined in the structure model, i.e. the valve plates surfaces and the piston surfaces. FSI interfaces include surfaces of the valve plates, flow holes' surfaces of the piston and both side surfaces of the floating piston.

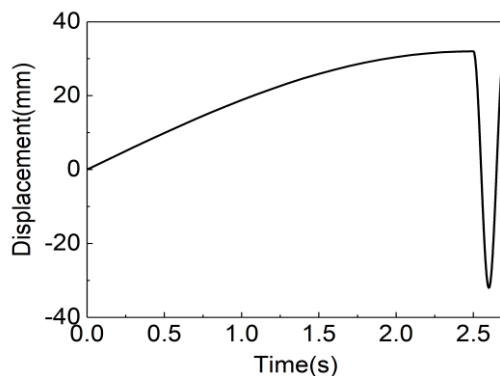


Figure 3. Input displacement excitation.

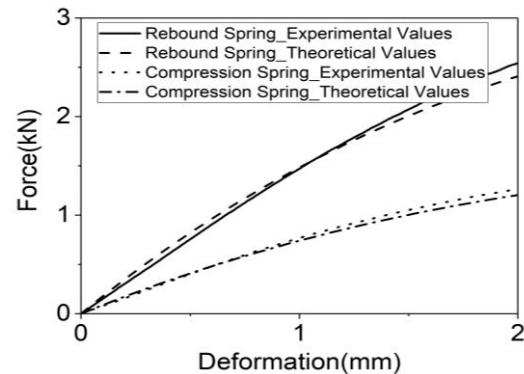


Figure 4. Spring stiffness characteristics.

In the fluid model, the oil is modelled as a slightly compressible fluid, and the gas is modelled as a low-speed compressible fluid. The material parameters needed in the FSI analysis are shown in Table 1. What's more, Leader-Follower method is used for parametric dynamic moving mesh control, initial pressure of 2 MPa is applied to the fluid model and FSI interfaces corresponding to the structural model's FSI interfaces are defined.

Table 1. Material parameters.

Materials	Solid	Oil	Gas
Density/(kg/m ³)	7800	832	/
Viscosity/(Pa · s)	/	0.017	22
Elastic modulus /(GPa)	210	1.5	/
Poisson's ratio	0.3	/	/
C_p /(J/(kg · K))	/	/	968.5
C_v /(J/(kg · K))	/	/	764.84

4. Results and discussion

4.1. Model validation

To verify the reliability of the results of the FSI simulation model, the experimental and simulation work diagrams are presented in Figure.5.

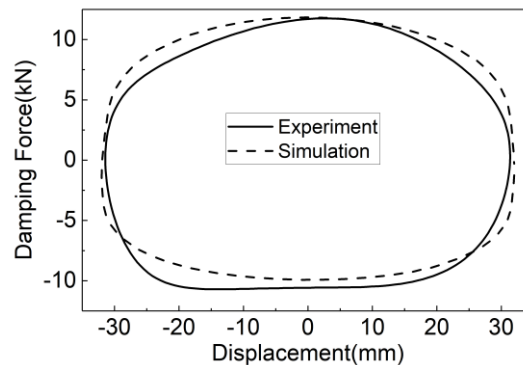


Figure 5. Work diagrams of the shock absorber

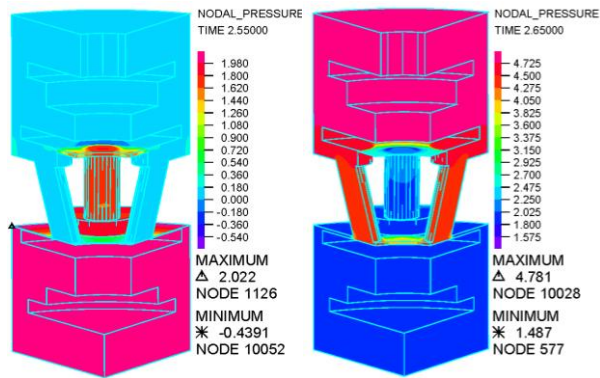
As can be observed in Figure.5, the simulation result shows good agreements with the experimental result. The friction between the pistons and the cylinder inner surface is neglected in the FSI model. Moreover, the experimental result is affected by the machining error and measurement error. Above factors make the simulation work diagram a fuller, better symmetric curve.

4.2. Flow characteristics

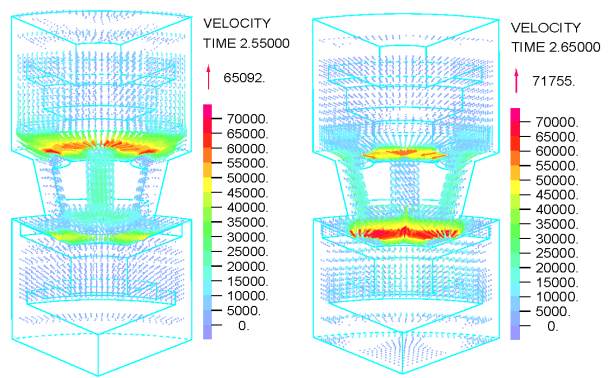
It is hard to obtain transient fluid pressure and velocity distribution in the absorber by experiments. However, FSI dynamic analysis can help get the fluid pressure and velocity distribution at anytime and anywhere. Fluid pressure and velocity distribution around valve system at the maximum velocity points of the compression and rebound strokes are shown in Figure.6 and Figure.7.

Figure.6 shows the pressure gradient is mainly concentrated in the throttling gap between the piston and the valve plates. Negative pressure zone appears in the compression stroke, which means cavitation happens in this area. It should be noted that if the pressure were negative, the oil in the actual shock absorber would become gas medium. Pressure distribution far away from the valve system shows small pressure gradient, the variation in the pressure of the compression and rebound chambers are shown in Figure.8. As can be observed in Figure.8, pressure in the rebound chamber changes drastically, pressure in the compression chamber changes little.

As can be observed in Figure.7, the maximum velocity value in the compression stroke is 65m/s, which occurs at the throttling gap between the piston and compression valve plate. The maximum velocity value in the rebound stroke is 71.8m/s, which occurs at the throttling gap between the piston and the rebound valve plate. Therefore, an excitation of 1m/s at the piston rod can lead to nearly 70 times velocity magnitude in the shock absorber.



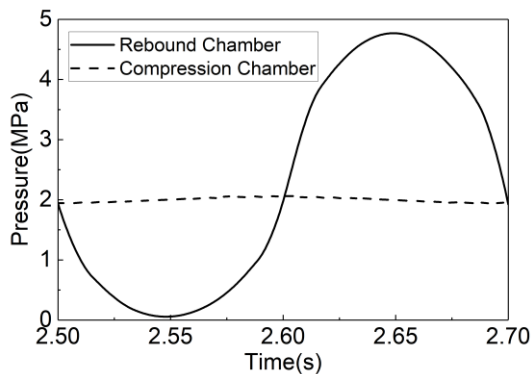
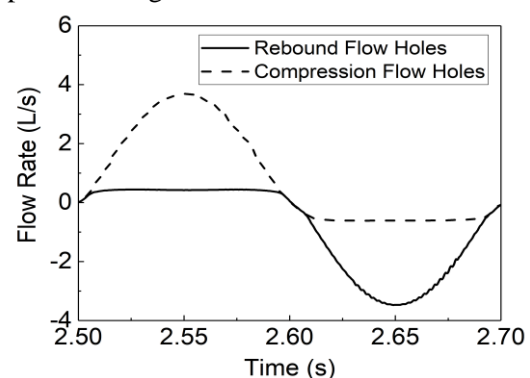
(a)Compression stroke (b)Rebound stroke

Figure 6. Fluid pressure distribution.

(a)Compression stroke (b)Rebound stroke

Figure 7. Fluid velocity distribution.

Figure.9 shows the variation in the flow rate through the compression and rebound flow holes. There is small amount of flow rate through the rebound flow holes during the compression stroke and small amount of flow rate through the compression flow holes during the rebound stroke, which are caused by the reserved gap between the piston and valve plates. This amount of flow rate can replace the amount of flow rate for constant flow channels and piston leakage.

**Figure 8.** The variation in the pressure of the compression and rebound chambers.**Figure 9.** The variation in flow rate.

4.3. Nonlinear dynamic response of the structure

Figure.10 shows the displacements of the valve plates, the maximum displacement of the compression valve plate is greater than the maximum displacement of the rebound valve plate, which is caused by the stiffness difference of the compression springs.

Figure.11 shows the displacements of the floating piston and the piston. As can be observed in Figure.11, there is a small phase difference between the displacement of the floating piston and the displacement of the piston, as the mass of the floating piston and oil compressibility are considered in the FSI model.

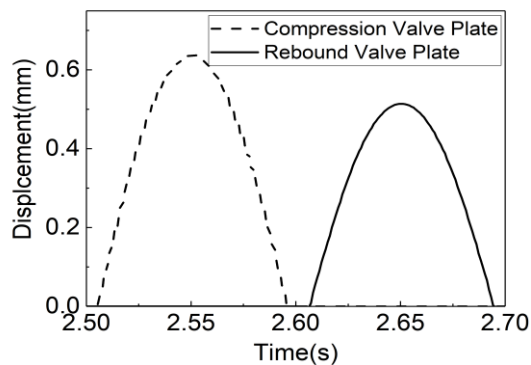


Figure 10. Displacement of the valve plates.

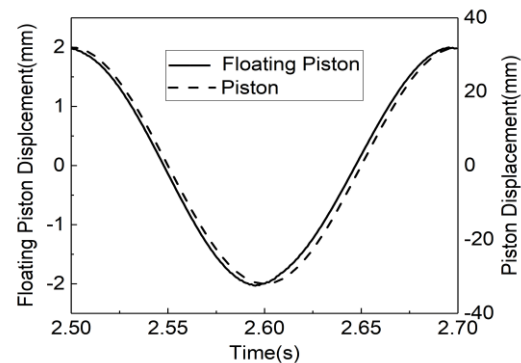


Figure 11. Displacement of pistons.

5. Conclusion

An FSI simulation model of a monotube gas-pressurized hydraulic shock absorber is built, in which the slightly compressible characteristic of the oil and the compressible characteristic of the gas are considered, direct FSI method and Leader-Follower moving mesh control strategy are used. The simulation work diagram shows very similar tendency with the experimental result, which verifies the reliability of the FSI model.

The following results can be obtained from the FSI model: the working characteristics, the flow characteristics which include transient fluid pressure and velocity distribution, as well as the variation in flow rate, the kinematical responses of the valve plates and floating piston, which are helpful to the understanding of the inner working mechanism of the monotube gas-pressurized hydraulic shock absorber system. This work is meaningful to the guiding and optimization of the design of shock absorbers.

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