Design of a magnetorheological damper with large damping force and nonlinear modeling at high velocity

Jian YAN and Longlei DONG
State Key Laboratory for Strength and Vibration of Mechanical Structures, Xi'an Jiaotong University, Xi'an, Shaanxi 710049, China

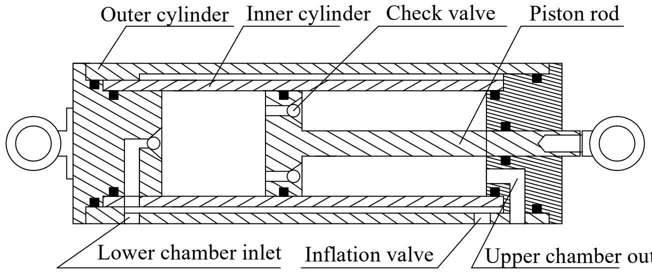
Jian YAN and Longlei DONG
School of Aerospace Engineering, Xi'an Jiaotong University, Xi'an, Shaanxi 710049, China

Abstract

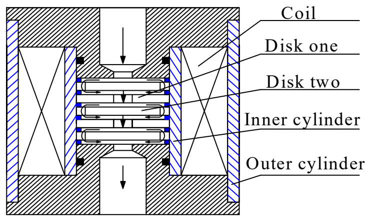
A method to design a magnetorheological damper with large damping force in a limited space is proposed, and a nonlinear model at high velocity is established. First, a magnetorheological damper with radial damping gap is designed and fabricated in this paper. Secondly, the magnetorheological damper is tested. The maximum damping force is close to 25kN. The damping force calculation model of the magnetorheological damper with radial damping gap at high velocity is obtained. Finally, by comparing the model with the experimental results, the accuracy of the damping force calculation model at high velocity is verified.

1 Design of magnetorheological damper

In this paper, a magnetorheological damper with radial damping gap is designed based on the stacking of two different types of disks. Figure 1 (a) and figure 1 (b) respectively show the schematic diagram of the hydraulic cylinder and the damping generator. The damping force target value of the designed damper is 25kN.



(a). Schematic diagram of cylinder structure of damper.



(b). Damping generator.

Figure 1: The principle of magnetorheological damper

2 Nonlinear modeling of the magnetorheological damper

Because the Bingham model[1] cannot accurately describe the shear thinning effect at a high shear rate, a behavior index n is introduced into the Herschel-Bulkley model[2]. The relationship between the shear stress τ and the shear rate $\dot{\gamma}$ is:

$$\tau=\tau_y(H)\operatorname{sgn}(\dot{\gamma})+K\dot{\gamma}^n \tag{1}$$

where τ_y is the yield stress; H is the magnetic field strength; K is the consistency index.

Considering the shear thinning of magnetorheological fluid at high velocity, the damping force increases nonlinearly with the change of velocity. Therefore, a parametric power function model can be established as follows:

$$F_r = N\left[\frac{2KA_p\left(\frac{A_p}{\pi}\frac{2n+1}{n}\right)^n\left(r_2^{1-n}-r_1^{1-n}\right)v^n}{h^{2n+1}(1-n)}+\frac{\pi\tau_y(D^2-d^2)(r_2-r_1)(1+av^b)}{2h}\right] \tag{2}$$

where N is the number of gaps between the disks; D is the diameter of the piston; d is the diameter of the piston rod; a and b are the parameters to be identified; r_1 and r_2 are the inner radius and outer radius of the radial damping gap, respectively; h is the height of the radial damping gap ; A_p is the effective area of the piston head.

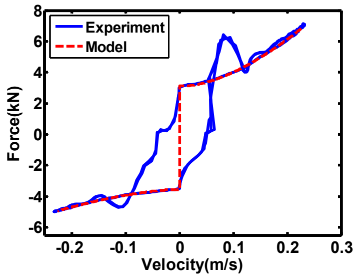
When the velocity is high, the damping force caused by minor losses need to be considered. The total tensile (F_T) and compression (F_C) damping forces are:

$$\begin{aligned} F_T &= \frac{6\pi\tau_y\left(D^2-d^2\right)\left(r_2-r_1\right)}{2h} + \frac{\pi\left(D^2-d^2\right)a_1 v^{b_1}}{4} - F_0 + f \qquad v \geq 0 \\ F_C &= -\frac{6\pi\tau_y d^2\left(r_2-r_1\right)}{2h} - \frac{\pi d^2 a_2 (-v)^{b_2}}{4} - F_0 - f \qquad v < 0 \end{aligned} \tag{3}$$

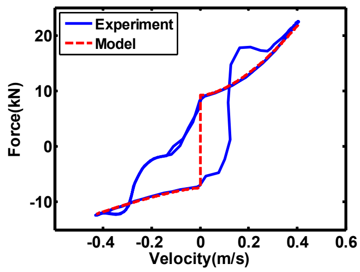
where F_0 is air cavity force; f is friction force.

3 Comparison of model results with experimental results

The designed magnetorheological damper is tested and compared with the model results. The comparison results of the model and the experiment show the accuracy of the established model. In addition, it can be seen from Figure 2 that the maximum damping force of the designed damper is close to 25kN.The results of the comparison are as follows:



(a). (0.6Hz, 0.06m, 1A).



(b). (1.4Hz, 0.05m, 3A).

Figure 2: Damping force versus velocity

4 Conclusion

A magnetorheological damper that has a large damping force is proposed. The established damping force calculation model solves the problem that the damping force of the radial damping gap is difficult to predict at high velocity. The research provides an effective design idea for the design of the magnetorheological damper with large damping force, and provides a damping force calculation method of the magnetorheological damper with radial damping gap at high velocity.

Acknowledgements

The author wishes to acknowledge that the experimental device is provided by State Key Laboratory for Strength and Vibration of Mechanical Structures.

References

[1] Wang M, Chen Z, and Wereley N M, Magnetorheological damper design to improve vibration mitigation under a volume constraint, *Smart Mater.Struct*. 2019, 28(11): 114003.

[2] Manjeet K and Sujatha C , Magnetorheological valves based on Herschel–Bulkley fluid model: modelling, magnetostatic analysis and geometric optimization, *Smart Mater.Struct*. 2019, 28(11): 115008.