A broadband variable fluid damper with frequency selective valves for spacecraft micro-vibration isolation

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Abstract. In order to improve the vibration isolation performance of current three-parameter damper using bellow and fluid damping, a concept of fluid damper with variable damping and secondary variable stiffness was proposed, a prototype has been designed with several frequency selective valves to adjust the damping characteristics at specific frequency bands, especially that in the low-frequency range. Followed mathematical modelling and simulation of the damper have been carried out, the results show that the designed damper has obtained excellent damping characteristics both in the high- and low-frequency ranges, the micro-vibration isolation efficiency of the damper under realistic excitation of a Control Moment Gyroscope used in a manned spacecraft reaches a significant 71.03 %.The proposed damper concept and designed prototype have laid the foundation for further testing and optimization of high-end isolators for modern spacecrafts.

Keywords: variable fluid damper, frequency selective valve, broadband, damping characteristics, micro-vibration isolation, spacecraft.

1. Introduction

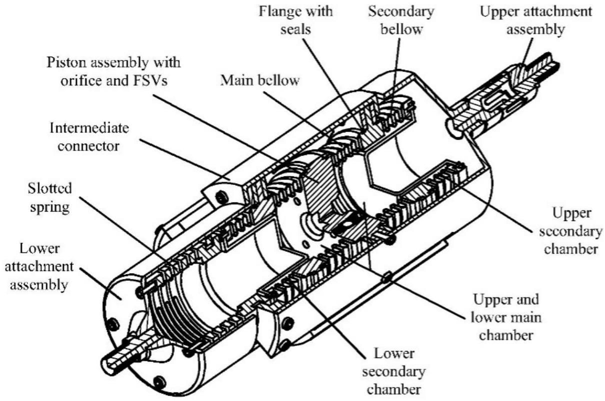
Fluid dampers are widely used in spacecrafts for impact and/or vibration mitigation during launching and on orbit [1, 2] or landing [3]. Damper using bellow and fluid damping originated from the two-parameter D-Strut developed by the early American Honeywell company for NASA, subsequently, many studies have been carried out and technical progress has been made. For example, various three-parameter dampers with secondary chambers [4-8] have been proposed. Compared with that of two-parameter dampers, the vibration isolation performance of three-parameter dampers has been significantly improved in the high-frequency range [1, 8], but the vibration isolation performance in the low-frequency range, including the resonance peak area, has not been improved.

A concept of a broadband fluid damper with variable hydraulic damping and secondary variable stiffness was first proposed, and a prototype has been designed with several frequency selective valves (FSVs) to adjust the damping characteristics at specific frequency bands, especially that in the low-frequency range. Mathematical modelling and simulation of the damper have been carried out, and the results show that the fabricated damper has obtained excellent vibration reduction performance both in the high- and low-frequency ranges, the proposed damper concept, designed prototype and research results in this study have laid the foundation for further testing and product optimization for modern spacecraft.

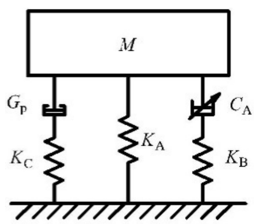
2. Design of a broadband variable fluid damper

As shown in Fig. 1(a), a type of fluid damper with variable damping and secondary variable stiffness was designed, the damper has four chambers filled with silicone oil, i.e., the upper and lower main chambers and the upper and lower secondary chambers. When the piston is actuated by spacecraft vibrations, the silicone oil will be forced through the damping systems embedded in the piston and produce damping forces. Fig. 1(b) illustrates that the damper has five parameters, which has an obvious advantage over the current three-parameter damper by enabling the damping element C_A to be variable.

Fig. 2 demonstrates the damping system embedded in the piston assembly. The fluid damping system includes a constant orifice and three FSVs, the three FSVs are arranged in a circular pattern around the constant orifice. The three FSVs are all low-frequency valves and designed to be normally open, but their natural frequencies are different. If the outside excitation frequency coincided with the natural frequency of a FSV, the valve ball of the FSV would resonant and close the valve, so the total damping effect of the damper would be increased, thus, the three FSVs would obviously improve the damping performance of the damper in the low-frequency band.

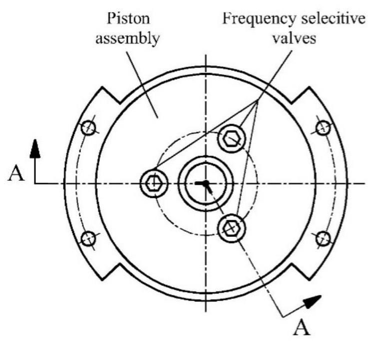


a) A broadband variable fluid damper

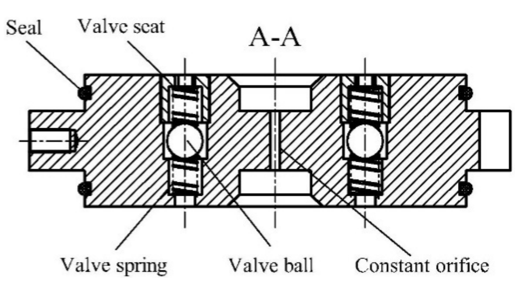


b) Mechanical model of the damper

Fig. 1. Prototype design and mechanical model of a broadband variable fluid damper



a) Top view of the piston assembly



b) A-A cross-section

Fig. 2. The damping system with constant orifice and FSVs in the piston assembly

3. Modelling the characteristics of the damper

Fig. 3 demonstrates the physical principle or model of a broadband variable fluid damper designed in the last section for micro-vibration reduction of a Control Moment Gyroscope (CMG) used in a manned spacecraft.

The stiffness of the two main bellows is given by:

$$K_A = M_C(2\pi f_{n1})^2, \tag{1}$$

where M_C is the CMG mass and f_{n1} is the first-order natural frequency of the isolation system.

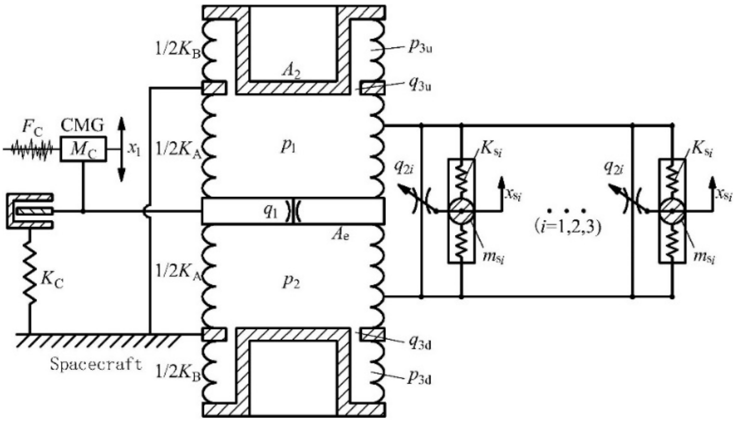


Fig. 3. Physical model of a broadband variable fluid damper used for micro-vibration reduction of a Control Moment Gyroscope (CMG) used in a manned spacecraft

The minimum vibration transmissibility [6] of a isolation system with basic three-parameter damper can be expressed as:

$$T_{min} = \sqrt{\frac{N+2}{\left(1-\frac{2(N+1)}{N+2}\right)^2 + \frac{1}{N+1}\left(N+1-\frac{2(N+1)}{N+2}\right)^2}} = \frac{N+2}{N}, \tag{2}$$

where N is the stiffness ratio of secondary bellow to main bellow, i.e., $N = K_B/K_A$, the recommendation value of N is 1-2. In addition, stiffness of the slotted spring [9, 10] in the damper can be expressed by:

$$K_C = \frac{4Eah^3}{nR^3} \Big/ \left[\frac{3}{2}\pi - \frac{12}{\pi} + \frac{1+\nu}{\gamma}\left(\frac{3}{2}\pi - \frac{2}{\pi} - 4\right)\right], \tag{3}$$

where E, ν are respectively the elastic modulus and the Poisson's ratio of the spring material, the others are all geometric parameters of the spring.

Under micro vibrations of the CMG, the dynamics of the isolation system with the fluid damper can be expressed as:

$$M_C\ddot{x}_1 + A_e sgn(\dot{x}_1)(p_1 - p_2) + K_A x_1 = \begin{cases} F_C(t) - K_C\left[x_1 - sgn(x_1)G_p\right], & |x_1| > G_p, \\ F_C(t), & |x_1| \le G_p, \end{cases} \tag{4}$$

where x_1 is the displacement of the CMG, A_e is the equivalent area of the bellow which is actuated by fluid, p_1 and p_2 are the pressures of the upper and lower main chambers, $F_C(t)$ is the disturbance force function of the CMG, G_P is the gap for the slotted spring to function.

In addition, by referring to Fig. 3, the fluid continuity equation in the damper under vibrations can be formulated by:

$$A_e sgn(\dot{x}_1)\dot{x}_1 = q_1 + q_2 + q_3, \tag{5}$$

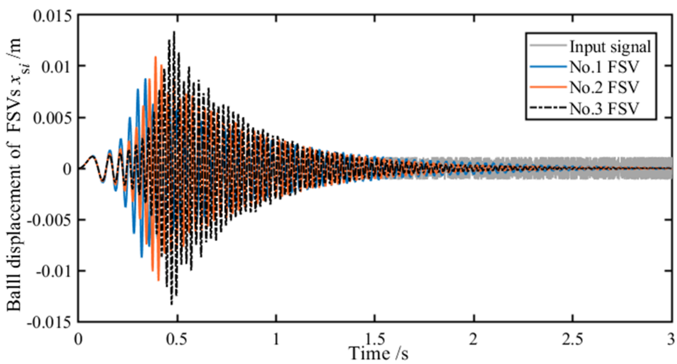
where q_1, q_2 and q_3 are respectively the flow rates passing through the constant orifice, passing through the FSV system and forced into the upper or lower secondary chambers, q_1, q_2 and q_3 can be formulated by:

$$\begin{cases} q_1 = C_{d1}\left(\frac{\pi}{4}{d_1}^2\right)\sqrt{\frac{2}{\rho}sgn(\dot{x}_1)(p_1 - p_2)}, \quad q_2 = \sum_{i=1}^{3} C_{d2}A_{xi}\sqrt{\frac{2}{\rho}sgn(\dot{x}_1)(p_1 - p_2)}, \\ q_3 = \frac{\pi d_3 \delta^3}{12\mu L}(p_1 - p_{3u}), \quad \mathrm{sgn}(\dot{x}_1) = 1, \quad q_3 = \frac{\pi d_3 \delta^3}{12\mu L}(p_2 - p_{3d}), \quad \mathrm{sgn}(\dot{x}_1) = -1, \end{cases} \tag{6}$$

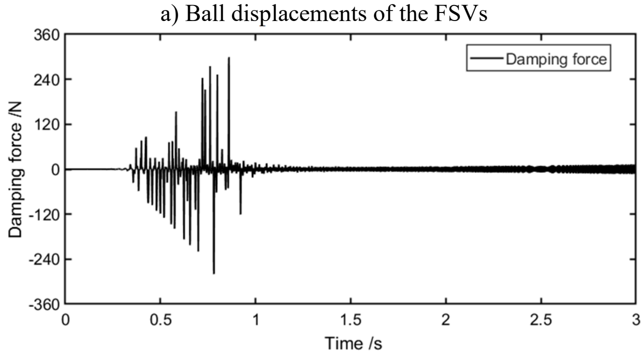
where C_{d1} and C_{d2} are the discharge coefficients of the constant orifice and the FSVs, A_{xi} ($i = 1\sim3$) are flow areas of the FSVs, p_{3u} and p_{3d} are the pressures of the upper and lower secondary chambers, μ is the dynamic viscosity of the silicone oil, the others are geometric parameters of the annular passage between the main chamber and the secondary chamber.

4. Simulation and analysis

With the mathematical model developed, MATLAB/Simulink models for the fluid damper and the above vibration isolation system have been established for simulation. In the simulation process, some of the main parameters and values include $M_c = 34$ kg, $K_A = K_B = K_C = 1.3745\times10^6$ N/m, $A_e = 7.49\times10^{-4}$ m^2, $d_1 = 2.5\times10^{-3}$ m, $C_{d1} = 0.80$, $C_{d2} = 0.60$, natural frequencies of the three FSVs are respectively 25 Hz, 32 Hz and 40 Hz, and some of the results obtained are shown by Figs. 4-6.



a) Ball displacements of the FSVs

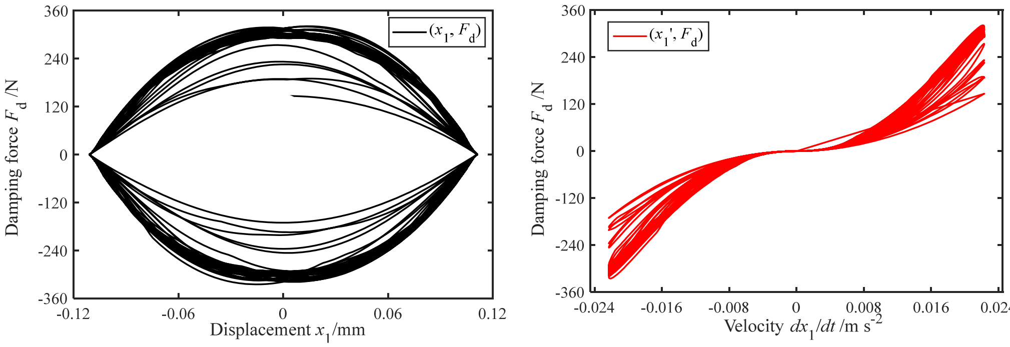


b) Damping force profile of the fluid damper

Fig. 4. Response characteristics of the fluid damper (simulation condition: sinusoidal chirp excitation with displacement amplitude ±0.1 mm, frequency range 0.1-500 Hz in 3 seconds)

Figs. 4(a) and (b) obviously demonstrate the response nature of the FSVs and the damper under sinusoidal chirp excitation condition. The three FSVs will close when the excitation frequencies coincide with their natural frequencies and thus improve damping force of the damper at these frequencies, because the three FSVs are all designed in the low-frequency range, so Fig. 4(b) shows that the damping forces in low-frequency range are obviously larger, and that in the middle to high frequency range are obviously smaller, in other words, the fluid damper has obtained larger damping coefficient at the low-frequency range and smaller coefficient at the middle- to high-frequency range, thus, the fluid damper has obtained excellent damping characteristics in a broadband.

Fig. 5 shows the damping characteristics of the fluid damper when subject to a sinusoidal excitation of displacement amplitude ±0.1 mm and frequency 32 Hz which coincides with the natural frequency of the No. 2 FSV. Fig. 5(a) and 5(b) demonstrate that the fluid damper has obtained stable and idea damping characteristics.



a) Force-displacement characteristics b) Force-velocity characteristics

Fig. 5. Damping characteristics of the fluid damper (simulation condition: sinusoidal excitation with displacement amplitude ±0.1 mm and frequency 32 Hz)

Fig. 6 shows the micro-vibration isolation performance of the fluid damper under realistic excitation of a CMG [11, 12] used in a manned spacecraft. It demonstrated that the micro-vibration of the CMG has been obviously mitigated, the Root Mean Square (RMS) value of CMG disturbance force 548.1 N has been reduced to 158.8 N after isolation, thus the vibration efficiency of the isolation system with the broadband variable fluid damper reaches a significant 71.03 %.

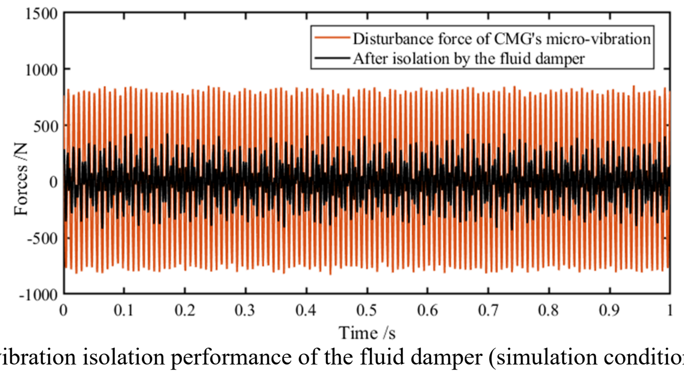


Fig. 6. Micro-vibration isolation performance of the fluid damper (simulation condition: experimental disturbance force profile of a CMG's macro-vibration)

5. Conclusions

1) A concept of a broadband fluid damper with variable hydraulic damping and secondary variable stiffness was proposed, and a prototype has been designed with several frequency selective valves to adjust the damping characteristics at specific frequency bands, especially that in the low-frequency range.

2) Mathematical modelling and simulation of the damper have been carried out, the results show that the designed damper has obtained excellent damping characteristics both in the high- and low-frequency ranges, the micro-vibration isolation efficiency of the fluid damper under

realistic excitation of a CMG used in a manned spacecraft reaches a significant 71.03 %.

3) The damper concept and designed prototype have laid the foundation for further testing and optimization of high-end isolators for modern spacecrafts.

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Data availability

The datasets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

Conflict of interest

The authors declare that they have no conflict of interest.

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