IOP Conf. Series: Materials Science and Engineering **292** (2018) 012039 doi:10.1088/1757-899X/292/1/012039

Analysis of material parameter effects on fluidlastic isolators performance

Q Y Cheng[1], J H Deng, Z Z Feng and F Qian

Science and Technology on Rotorcraft Aeromechanics Laboratory, China Helicopter Research and Development Institute, Jingdezhen 333001, China

[1] E-mail: qy_cheng@163.com

Abstract. Control of vibration in helicopters has always been a complex and challenging task. The fluidlastic isolators become more and more widely used because the fluids are non-toxic, non-corrosive, nonflammable, and compatible with most elastomers and adhesives. In the field of the fluidlastic isolators design, the selection of design parameters of fluid and rubber is very important to obtain efficient vibration-suppressed. Aiming at getting the property of fluidlastic isolator to material design parameters, a dynamic equation is set up based on the dynamic theory. And the dynamic analysis is carried out. The influences of design parameters on the property of fluidlastic isolator are calculated. The material parameters examined are the properties of fluid and rubber. Analysis results showed that the design parameters such as density of fluid, viscosity coefficient of fluid, stiffness of rubber (K1) and loss coefficient of rubber have obvious influence on the performance of isolator. Base on the results of the study it is concluded that the efficient vibration-suppressed can be obtained by the selection of design parameters.

1. Introduction

Helicopter vibration is a critical aspect of helicopter design and a major reason for extended lead times during the aircraft development phase. Control of vibration in helicopters has always been a complex and challenging task. Increasing demands for expanding the flight envelop of helicopters, such as nap of earth flying, high speed, high maneuvers, coupled with the need to improved system reliability and reduce maintenance costs have resulted in more stringent vibration specifications.

Various methods have been applied to vibration control in the engineering field [1-6]. Traditionally, passive isolators and dampers are used to attenuate mechanical vibrations. The traditional approach to passive vibration isolation is to install relatively soft springs or elastomeric isolators to provide a low primary natural frequency [7-10]. These isolators would also incorporate sufficient damping to control resonant response. Soft systems with primary natural frequencies well below the N/rev exciting frequency are required to achieve isolation. Such systems result in large relative motion between the pylon and the airframe due to static loads. Natural frequencies low enough to isolate N/rev vibration would have static (1G) deflections up to 0.50 inches. Since flight controls and power transmission drive shafts cross this interface it is advantageous to keep the relative motion as small as practical. An effective method for isolating the N/rev vibration that did not allow large relative motions between the pylon and airframe was needed.

Fluidlastic products can also provide the long, predictable service life typical of elastomeric rotor bearings, because all of the relative motion across a Fludlastic device is accommodated by shear of the elastomer. There are no sliding seals, bushings, or bearings exposed to the environment [10-11]. Over a long service life, they typically show gradual degradation in appearance, and benign failure modes that allow for replacement long before their performance has been compromised. These characteristics also allow for on-condition replacement eliminating fixed service life intervals [12]. Many fluidlastic products cannot be distinguished from a conventional bonged rubber part by their external appearance.

There are many design parameters such as the density of fluid, viscosity coefficient of fluid, stiffness of rubber (K1) and loss coefficient of rubber that can affect the property of fluidlastic isolator. Each of these parameters is important considerations in the design of a fluidlastic isolator and affects property of fluidlastic isolator to a greater or lesser degree.

The present study is aimed at analyze the influence of material design parameters on property of fluidlastic isolator. The dynamic equation is set up based on the theory of dynamics. And the dynamic analysis is carried out. The influences of design parameters on the property of fluidlastic isolator are calculated.

2. Mathematical model

Figure 1 and figure 2 shows the cross-section and the mechanical model of Fluidlastic Isolator.

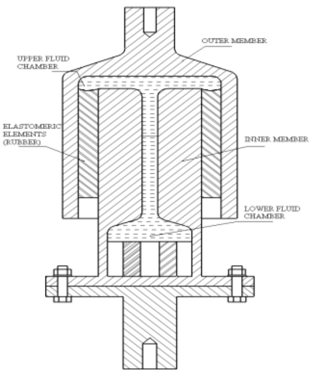

Figure 1. Cross-section of isolator

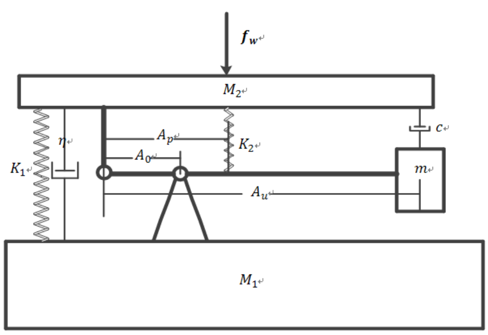

Figure 2. Mechanical model of isolator

According to the figure 1 and figure 2, the equations of motions (EOM) for the fluidlastic tuned isolator can be expressed as:

$$M_1\ddot{x}_1 = -K_1(1+i\eta)(x_1 - x_2) - f - [m(\ddot{x}_0 + \ddot{x}_2) + c\dot{x}_0] - K_2 x_d \tag{1}$$

$$M_2\ddot{x}_2 = K_1(1+i\eta)(x_1 - x_2) + f + K_2 x_d + c\dot{x}_0 + f_w \tag{2}$$

$$[m(\ddot{x}_0 + \ddot{x}_2) + c\dot{x}_0](A_u - A_0) + K_2 x_d (A_p - A_0) = fA_0 \tag{3}$$

Where, f is the load that the M_2 imposed on the lever mechanism. And A_p is defined as:

$$A_p = A_u A_0 / A_d \tag{4}$$

The factors are defined as:

$$R_1 = \frac{A_u}{A_0} = \frac{x_0}{x_1 - x_2} \tag{5}$$

$$R_2 = \frac{A_d}{A_0} = \frac{x_0}{x_d} \tag{6}$$

Taking Eq.(5) and Eq.(6) into the Eq.(3) ,We obtain the equation of f :

$$f = [mR_1\ddot{x}_1 - m(R_1 - 1)\ddot{x}_2 + cR_1(\dot{x}_1 - \dot{x}_2)](R_1 - 1) + K_2 \frac{R_1}{R_2}(x_1 - x_2)\left(\frac{R_1}{R_2} - 1\right) \tag{7}$$

Taking Eq.(7) into the Eq.(1) and Eq.(2) ,We obtain the following relations:

$$M_1\ddot{x}_1 + m\ddot{x}_1R_1^2 + c\dot{x}_1R_1^2 + K_1(1+i\eta)x_1 + K_2\frac{R_1^2}{R_2^2}x_1 = mR_1(R_1-1)\ddot{x}_2 + c\dot{x}_2R_1^2 + K_1(1+i\eta)x_2 + K_2\frac{R_1^2}{R_2^2}x_2 \quad (8)$$

$$M_2\ddot{x}_2 + m(R_1-1)^2\ddot{x}_2 + c\dot{x}_2R_1^2 + K_1(1+i\eta)x_2 + K_2\frac{R_1^2}{R_2^2}x_2 = mR_1(R_1-1)\ddot{x}_1 + c\dot{x}_1R_1^2 + K_1(1+i\eta)x_1 + K_2\frac{R_1^2}{R_2^2}x_1 \quad (9)$$

The Eq.(8) and Eq.(9) can be expressed as:

$$\begin{bmatrix} R_2^2M_1 + R_1^2R_2^2m & -R_2^2mR_1(R_1-1) \\ -mR_1(R_1-1)R_2^2 & R_2^2M_2 + R_2^2m(R_1-1)^2 \end{bmatrix}\begin{Bmatrix}\ddot{x}_1 \\ \ddot{x}_2\end{Bmatrix} + \begin{bmatrix} R_1^2R_2^2c & -R_1^2R_2^2c \\ -R_1^2R_2^2c & R_1^2R_2^2c \end{bmatrix}\begin{Bmatrix}\dot{x}_1 \\ \dot{x}_2\end{Bmatrix} + \begin{bmatrix} R_2^2K_1(1+i\eta)+R_1^2K_2 & -R_2^2K_1(1+i\eta)-R_1^2K_2 \\ -R_2^2K_1(1+i\eta)-R_1^2K_2 & R_2^2K_1(1+i\eta)+R_1^2K_2 \end{bmatrix}\begin{Bmatrix}x_1 \\ x_2\end{Bmatrix} = \begin{Bmatrix}0 \\ f_w\end{Bmatrix} \quad (10)$$

When the vibration load is a harmonic excitation, the load and displacement can be written as follows:

$$\left.\begin{aligned} f_w &= Fe^{j\omega t} \\ x_1 &= X_1e^{j\omega t} \\ x_2 &= X_2e^{j\omega t} \end{aligned}\right\} \quad (11)$$

The Eq.(11) is substituted into the Eqs.(10). We obtain the following relations:

$$\frac{X_1}{X_2} = \frac{\Delta_1}{\Delta_2} = \frac{\omega^2m(R_1-1)R_1R_2^2 - K_2R_1^2 - jc\omega R_1^2R_2^2 - K_1(1+i\eta)R_2^2}{\omega^2mR_1^2R_2^2 - K_2R_1^2 - jc\omega R_1^2R_2^2 + \omega^2M_1R_2^2 - K_1(1+i\eta)R_2^2} \quad (12)$$

From the equation (12), the Transmissibility of vibration can be written as:

$$T = \left|\frac{X_1}{X_2}\right| = \left|\frac{\Delta_1}{\Delta_2}\right| \quad (13)$$

When the X_1, Δ_1, c, η are all equal to zero, the frequency of undamped isolation system can be obtained:

$$F_r = \frac{1}{2\pi}\sqrt{\frac{K_1R_2^2 + K_2R_1^2}{(R_1-1)R_1R_2^2m}} \quad (14)$$

The amplitude of vibration can be written as:

$$X = \frac{R_1R_2^2F}{\left[K_1(1+i\eta)R_2^2 + K_2R_1^2\right] - m(R_1-1)^2R_2^2\omega^2 + jcR_1^2R_2^2\omega} \quad (15)$$

From the equation (15), the dynamic stiffness of isolation system can be obtained:

$$K^* = \frac{F}{X} = \frac{\left[K_1(1+i\eta)R_2^2 + K_2R_1^2\right] - m(R_1-1)^2R_2^2\omega^2 + jcR_1^2R_2^2\omega}{R_1R_2^2} \quad (16)$$

According to the static equation, the static stiffness of isolation system can be also obtained:

$$K = K_1 + K_2\frac{R_1^2}{R_2^2} \quad (17)$$

3. Results and Discussion

3.1. Stiffness of fluidlastic isolators

A civil helicopter is chosen for this study. The N/rev frequency of this helicopter is close to 25Hz.

The high loads and very small motions of high stiffness isolators are even more challenging. The effectiveness of a tuned isolator in a system can be estimated by measuring the dynamic stiffness (K*) over the frequency range of interest. The K* value at the tuned (N/rev) frequency is a good indicator of the effectiveness of the isolator in the system.

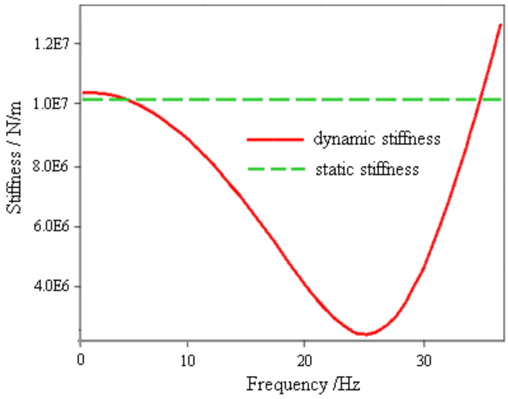

Figure 3. Fluid isolator stiffness versus frequency.

Fluidlastic isolators must also be designed to handle a specific range of input motions. The dynamic stiffness versus frequency curve is shown in figure 3. The dynamic stiffness varied with the frequency. The dynamic stiffness varies from nominally 1.03E7 N/m statically (Frequency is equal to 0 Hz) to approximately 1.91E6 N/m at 25 Hz. The effective dynamic stiffness at 25Hz is less than 20% of the static stiffness. So, the “rigid” isolator can still provide effective performance. However, the static stiffness is relatively high. It is advantageous to keep the relative motion between the pylon and airframe as small as practical.

3.2. Effect of fluid properties

The transmissibility versus fluid density curve is shown in figure 4. The results show that the alteration trend of transmissibility influenced by fluid density is similar to that affected by K1. There is also an optimal value exists for fluid density to get a good effect of vibration-suppressed.

According to the equation (14), the nature frequency of isolator is decreased with the increase of mass of fluid. When the nature frequency of isolator is varied from excitation frequency, the Transmissibility will be increased.

The influence of viscosity coefficient of fluid on the Transmissibility of vibration is shown in figure 5. The results show that the viscosity coefficient has obvious influence on the transmissibility of vibration. Compared to the loss coefficient, Compared the viscosity coefficient with the loss coefficient, the influence on Transmissibility of vibration is relatively higher.

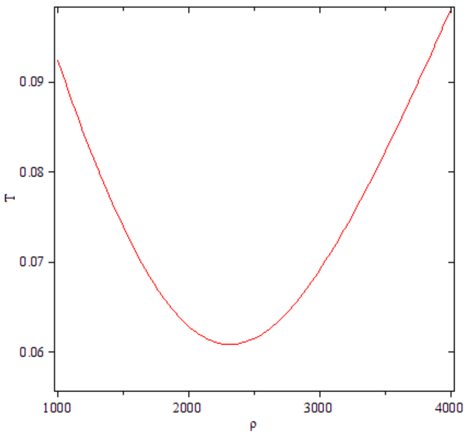

Figure 4. Density versus transmissibility.

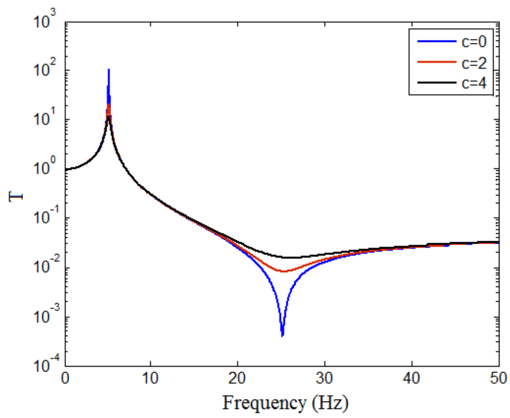

Figure 5. Viscosity coefficient versus transmissibility.

3.3. Effect of rubber properties

The transmissibility versus stiffness of rubber (K1) curve is shown in figure 6. It is observed that the K1 has obvious influence on the transmissibility. When the K1 increase to some value, the transmissibility will be attain to minimum value. This is means that the selection for K1 is very important to get a good effect of vibration-suppressed.

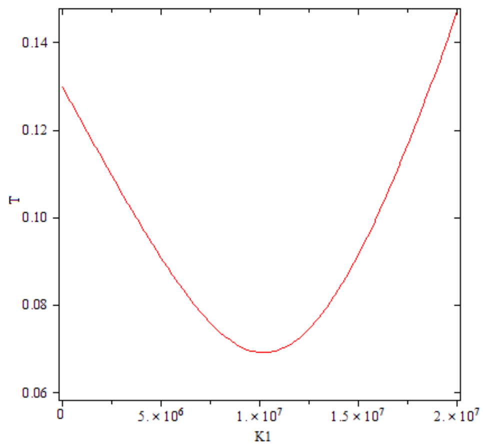

Figure 6. K1 versus Transmissibility.

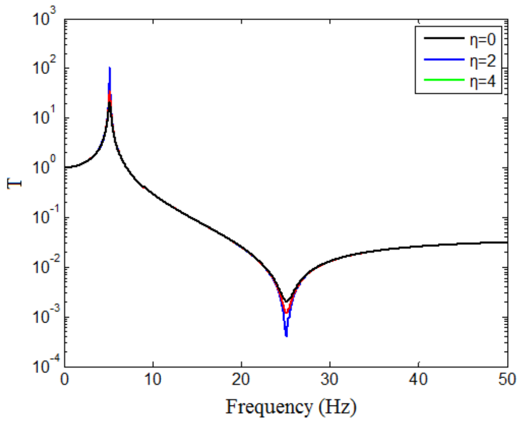

Figure 7. Loss coefficient versus Transmissibility.

The influence of loss coefficient of rubber on the Transmissibility of vibration is shown in figure 7. The results show that the loss coefficient has obvious influence on the transmissibility of vibration. With the increase of the loss coefficient, the transmissibility of vibration decreased. It is means that the efficiency of vibration-suppressed is improved. However, the result is also show that the loss coefficient has no influence on the resonance frequency of isolator.

4. Conclusions

The dynamic equation is set up based on the theory of dynamics. The dynamic analysis is carried out. The influences of design parameters on the property of fluidlastic isolator are calculated. Findings are listed below:

1) The design parameters such as the density of fluid, viscosity coefficient of fluid, stiffness of rubber (K1) and loss coefficient of rubber have significant influence on the performance of isolator. The efficient vibration-suppressed can be obtained by the selection of design parameters. The frequency of isolator can be also designed by the design parameters.

2) The properties of fluid have obvious influence on the performance of isolator. The nature frequency of isolator is affected by fluid density. And it is decreased with the increase of fluid mass. The viscosity coefficient and density of fluid are all have effect on the transmissibility of vibration. The selection for density and viscosity coefficient is very important to get a good effect of vibration-suppressed.

3) The rubber performance has some influence on properties of isolator. The nature frequency of isolator is increased with stiffness of rubber. The transmissibility of vibration is influenced by the stiffness (K1) and loss coefficient. And transmissibility is increased with loss coefficient. So the well vibration-suppressed efficiency can be obtained by selection of K1 and loss coefficient.

References

[1] Crews S T 1987 Rotorcraft Vibration Criteria A New Perspective *Proceedings of the 43rd Annual Forum of the American Helicopter Society* (St. Louis MO)
[2] Strehlow H, Mehlhose R, Znika P and Roth D 1990 Review of MBB's Passive and Active

Vibration Control Activities *The Royal Aeronautical Society in London*

[3] Heilmann J, Swanson D, Askari B A 2003 Vibration Attenuation Through the Use of Active Frahms *American Helicopter Society 59th Annual Forum* (Phoenix Arizona)

[4] Roget B, Chopra I 2004 Wind Tunnel Testing of an Individual Blade Controller for a Dissimilar Rotor *Presented at the American Helicopter Society 60th Annual Forum* (Baltimore, MD)

[5] C T Hugin, C Hatch 2006 Global control of helicopter vibrations using a semi-active vibration control system *47th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Con* (Newport, Rhode Island)

[6] Brown T J, Mccloud J L 1980 Multicyclic control of a helicopter rotor considering the influence of vibration, load, and control motion *In: 36th annual National Forum of the AHS* (Washington. D C: AHS)

[7] Gupta B P, Logan A H 1984 Wood E R. Higher harmonic control for rotary wing aircraft *In: AIAA/AHS/ASE Aircraft Design Systems and Operations Metting* (San Diego, California)

[8] Halwes, Dennis R 1975 Nodal Beam Inertial Flexure U.S. Patent 3 858-831

[9] Flannelly, W G 1964 Dynamic Anti-resonant Vibration Isolator U.S. Patent 3,322,379

[10] Smith M R., Redinger W S 1999 The Model 427 Pylon Isolation System *American Helicopter Society 55th Annual Forum* (Montreal, Quebec, Canada)

[11] McGuire, Dennis P 1994 Fluidlastic Dampers and Isolators for Vibration Control in Helicopters *American Helicopter Society 50th Annual Forum* (Washington, DC)

[12] Dennis P McGuire 2006 Active Vibration Control Using Fluidlastic Pylon Struts *American Helicopter Society 62nd Annual Forum* (Phoenix, Arizona)