Analysis of Design Parameters Effects on Vibration Characteristics of Fluidlastic Isolators

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Abstract. The control of vibration in helicopters which consists of reducing vibration levels below the acceptable limit is one of the key problems. 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 is very important to obtain efficient vibration-suppressed. Aiming at getting the effect of design parameters on the property of fluidlastic isolator, a 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. Dynamic analysis results have shown that fluidlastic isolator can reduce the vibration effectively. Analysis results also showed that the design parameters such as the fluid density, viscosity coefficient, stiffness (K1 and K2) and loss coefficient have obvious influence on the performance of isolator. The efficient vibration-suppressed can be obtained by the design optimization of parameters.

1. Introduction
The control of vibration in helicopters which consists of reducing vibration levels below the acceptable limit is one of the key problems. 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 [1-5]. 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 [6-9]. 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 Fluidlastic device is accommodated by shear of the elastomer. There are no sliding seals, bushings, or bearings exposed to the environment [9]. 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 [10, 11]. Many fluidlastic products cannot be distinguished from a conventional bonged rubber part by their external appearance.
The present study is aimed at analyze the influence of 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. The method is used to analyze the actuator optimization selection of helicopter.

2. Mathematical Model

Figure 1 shows the schematic diagram of fluidlastic isolator. The fluidlastic isolator consists of an inner cylinder and an outer cylinder concentrically bonded together with elastomers to form two chambers that are joined by a tuning port through the inner cylinder. The elastomer serves both as the seal for the chamber and the compliant strain member in the isolator. The chambers and the tuning port are completely filled with an inviscid fluid. As the inner cylinder moves up or down, producing an inertial force that cancels the elastomeric spring force at a discrete frequency.

A mechanical model used to analyze the dynamic performance of fluidlastic device is shown in Figure 2.

![Figure 1. Cross-section of a fluidlastic Isolator](image1)

![Figure 2. Mechanical Model of Isolator](image2)

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

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

(1)

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

(2)

\[
[m(\ddot{x}_0 + \ddot{x}_2) + c\dot{x}_0] A_p - A_0 + K_2 x_d (A_p - A_0) = f A_0
\]

(3)

Where, \( f \) is the load that the M2 imposed on the lever mechanism. And \( A_p \) is defined as:

\[
A_p = A_e A_d / A_d
\]

(4)

The factors are defined as:

\[
R_1 = \frac{A_e}{A_0} = \frac{x_0}{x_1 - x_2}
\]

(5)

\[
R_2 = \frac{A_e}{A_0} = \frac{x_0}{x_d}
\]

(6)

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

\[
f = \left[ m R_1 \ddot{x}_1 - m (R_1 - 1) \ddot{x}_2 + c R_1 (\dot{x}_1 - \dot{x}_2) \right] (R_1 - 1) + K_2 \frac{R_0}{R_2} (x_1 - x_2) \left( \frac{R_0}{R_2} - 1 \right)
\]

(7)

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

\[
M_1 \ddot{x}_1 + m \ddot{x}_1 R_1^2 + c \dot{x}_1 R_1^2 + K_1 (1 + i \eta) x_1 + K_2 \frac{R_0^2}{R_2^2} x_1 = m R_1 (R_1 - 1) \ddot{x}_2 + c \dot{x}_2 R_1^2 + K_1 (1 + i \eta) x_2 + K_2 \frac{R_0^2}{R_2^2} x_2
\]

(8)
\[ M_2 \ddot{x}_2 + m(R_1 - 1)^2 \ddot{x}_2 + c_2 R_2^2 + K_1 (1 + i\eta) \dot{x}_2 + K_2 \frac{R_2^2}{R_1^2} x_2 = mR_1(R_1 - 1)\dot{x}_1 + c_1 R_1^2 + K_1 (1 + i\eta) x_1 + K_2 \frac{R_1^2}{R_2^2} x_1 \quad (9) \]

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

\[
\begin{bmatrix}
R_2^4 M_1 + R_2^2 R_1^2 m - R_2^4 m R_1 (R_1 - 1) \\
-mR_1 (R_1 - 1) R_1^2 - R_2^4 M_2 + R_2^4 m (R_1 - 1)^2
\end{bmatrix}
\begin{bmatrix}
\dot{x}_1 \\
\dot{x}_2
\end{bmatrix}
+ \begin{bmatrix}
R_2^4 c - R_2^4 c \\
-R_2^4 c
\end{bmatrix}
\begin{bmatrix}
\ddot{x}_1 \\
\ddot{x}_2
\end{bmatrix}
= \begin{bmatrix}
0 \\
0
\end{bmatrix}
\quad (10)
\]

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

\[
f_c = F e^{i\omega t}
\]

\[
x_1 = X_c e^{i\omega t}
\]

\[
x_2 = X e^{i\omega t}
\]

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

\[
X_1 = \frac{\Delta_1}{\Delta_2} = \frac{\omega^2 m (R_1 - 1) R_1^2 - K_2 R_2^2 - j c_1 R_1^2 \omega^2 - K_1 (1 + i\eta) R_1^2}{\omega^2 m R_1^2 + K_2 R_2^2 - j c_1 R_1^2 \omega^2 + \omega^2 M_1 + K_1 (1 + i\eta) R_1^2}
\quad (12)
\]

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

\[
T = \frac{X_1}{X} = \left| \frac{\Delta_1}{\Delta_2} \right|
\quad (13)
\]

When the \( X_i, \Delta_i, c, \eta \) are all equal to zero, the frequency of undamped isolation system can be obtained:

\[
f = \frac{1}{2\pi} \sqrt{\frac{K R_2^2 + K_1 R_1^2}{(R_1 - 1) R_1 R_2 m}}
\quad (14)
\]

The amplitude of vibration can be written as:

\[
X = \frac{R_1 R_2^2 F}{K_1 (1 + i\eta) R_1^2 + K_2 R_1^2} - \frac{m(R_1 - 1)^2 R_2^2 \omega^2 + j c_1 R_1^2 R_2^2 \omega}{K_1 (1 + i\eta) R_1^2 + K_2 R_1^2}
\quad (15)
\]

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

\[
K^* = \frac{F}{X} = \frac{K_1 (1 + i\eta) R_1^2 + K_2 R_1^2}{R_1 R_2^2 - m(R_1 - 1)^2 R_2^2 \omega^2 + j c_1 R_1^2 R_2^2 \omega}
\quad (16)
\]

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

Main rotor speeds depend on the size of the helicopter but are typically low (160 to 450 RPM) because they are limited by tip speed and noise constraints. Since the number of rotor blades is generally between two and seven, N/rev frequencies are usually between 12 and 30 Hz. The frequency be focused in this paper is about 25 Hz.

The transmissibility versus stiffness (K1) curve is shown in Figure 3. 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.

The influence of K2 on the Transmissibility of vibration is shown in Figure 4. The results show that there is the same trend of transmissibility curves at different K2. The transmissibilities at the three stations are all less than 0.1 between 20Hz and 30Hz. It is means that the fluidelastic isolators can provide efficient vibration-suppressed.
It is also observed that the K2 has obvious influence on the resonance frequency. The resonance frequency of isolator increased with K2. So we can adjust the K2 to make the frequency of isolator suit the excitation. This is an advantage of the fluidilastic isolators.

**Figure 3.** The influence of K1 on the Transmissibility

The influence of loss coefficient on the Transmissibility of vibration is shown in Figure 5. 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.

**Figure 4.** The influence of K2 on the Transmissibility

**Figure 5.** Loss coefficient vs transmissibility

**Figure 6.** Viscosity coefficient vs transmissibility

**Figure 7.** The density versus transmissibility
The influence of fluid viscosity coefficient on the Transmissibility of vibration is shown in Figure 6. The results show that the viscosity coefficient has obvious influence on the transmissibility of vibration. With the decrease of the viscosity coefficient, the transmissibility of vibration decreased. It means that the efficiency of vibration-suppressed is improved.

The transmissibility versus fluid density curve is shown in Figure 7. 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 mass of fluid. When the nature frequency of isolator is varied from excitation frequency, the Transmissibility will be increased.

### 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) Dynamic analysis results have shown that fluidlastic isolator can reduce the vibration effectively. The transmissibility can reach 0.1 between 20Hz and 30Hz. The performance of isolator can be designed by the design parameters.

2) The K1, K2 and loss coefficient have obvious influence on the performance of isolator. When the K1 increase to some value, the transmissibility will be attain to minimum value. The transmissibility of vibration decreased with the increase loss coefficient. So the selection for K1, K2 and loss coefficient is very important to get a good effect of vibration-suppressed.

3) The nature frequency and transmissibility of vibration are all affect by the fluid performance. To design a fluidlastic isolator with higher static stiffness and well vibration-suppressed efficiency, the density of fluid should be higher value. However the fluid viscosity coefficient should be lower value.

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