Fluidlastic Absorbers for Lagwise Loads Reduction of Rotor Blades

Dong Han
Associate Professor
Science and Technology on Rotorcraft Aeromechanics Laboratory
Nanjing University of Aeronautics and Astronautics
29 Yudao Street, Nanjing 210016, China
(Tele: +86-25-84896444)
(e-mail: donghan@nuaa.edu.cn)

Edward C. Smith
Professor
Vertical Lift Research Center of Excellence
The Pennsylvania State University
231D Hammond Building, University Park 16802, USA
(Tele: +1-814-8630966    Fax: +1-814-865-6246)
(e-mail: ecs5@engr.psu.edu)

ABSTRACT
To reduce the troublesome second harmonic lagwise load in stiff in-plane rotor blades, tuned fluidlastic absorbers are proposed to be embedded in the blade cavity. The aeroelastic simulation of the coupled blade and fluidlastic absorber system is based on the generalized force formulation. The results indicate that placing an embedded fluidlastic absorber in the blade chordwise direction can reduce the second harmonic lagwise root bending moment by more than 85% at different steady flight states, which means fluidlastic absorbers are an effective means to control this load. The corresponding stroke of the absorber is limited within 2% blade chord length. Increasing tuning port area ratio is an effective means to reduce the stroke with relatively small influence on the performance of the absorber. The effects of the tuning frequency of the absorber, the damping, initial position, forward speed, and flight altitude on the performance of the absorber are also studied.

NOMENCLATURE

\( A \)  
blade cross-sectional area

\( a_p \)  
displacement of prime mass

\( a_{p0} \)  
initial displacement of prime mass

\( a_t \)  
displacement of tuning mass

\( a_{t0} \)  
initial displacement of tuning mass

\( C_D \)  
fuselage drag coefficient

\( C_H \)  
rotor drag coefficient

\( C_{M_f} \)  
rotor rolling moment coefficient

\( C_{M_{sy}} \)  
fuselage rolling moment

\( C_{M_{yp}} \)  
fuselage pitching moment coefficient

\( C_T \)  
thrust coefficient

\( C_W \)  
weight coefficient

\( C_y \)  
rotor side force coefficient

\( C_{y_f} \)  
fuselage side force coefficient

\( c_p \)  
damping

\( F \)  
force

\( G \)  
tuning port area ratio

\( h \)  
vertical distance from helicopter mass center to hub center
**1 INTRODUCTION**

How to improve rotorcraft performance, such as forward speed, endurance, range, and flight altitude, is always a hot topic in this field. Some advanced rotorcraft such as V-22, A160, and X2 have demonstrated significant performance improvement compared with traditional helicopters. One of the common features of the rotor system is the deployment of rigid rotors designed to operate at variable rotor speeds. Rigid rotors have higher lagwise stiffness, which can incur higher lagwise loads level compared with soft in-plane or articulated rotors [1]. The second harmonic (2/rev) lagwise load generally increases significantly with forward speed [2]. With the reduction of rotor speed, the fundamental lagwise frequency ratio of stiff in-plane blades crosses 2.0, which can lead to large 2/rev lagwise loads [3]. The second harmonic lagwise load may be a paramount factor for the consideration of blade fatigue and strength due to the high amplitude in forward flight.

A number of concepts for blade or hub loads control have been explored recently. Wilbur et al. conducted the test to reduce rotor vibratory loads by blade active twist [4]. Both rotating and fixed frame loads could be dramatically affected, and 60% to 95% reduction of the fixed system loads was demonstrated in the test. Mannchen and Well used individual blade control to reduce helicopter vibration [5]. The results showed the significant effectiveness of the reduction of the hub loads and vibration. Kim et al.
utilized multiple trailing edge flaps to control the flapwise blade loads and pitch link loads [6]. About 30% maximum flapping bending moment and 40% pitch link peak could be reduced. Min et al. utilized Gerney flaps to control rotor vibratory loads [7]. The proper deployment of Gerney flaps could reduce about 80% of the 4/rev normal force vibrations. Han and Smith utilized embedded chordwise elastomeric absorbers for lagwise loads reduction of rotor blades [8]. The results showed significant lagwise loads reduction.

Austruy et al. utilized an embedded spanwise Coriolis absorber in a rotor blade to reduce the in-plane vibratory hub loads [9]. The simulations showed that over 85% 4/rev longitudinal and lateral hub shears could be reduced. Colella et al. applied higher harmonic control to reduce the loads at the tiltrotor wing root [10]. The results indicated significant 3/rev loads reduction at the blade root. These concepts primarily concentrate on hub loads or higher harmonic blade loads control. Few concepts focus on lower harmonic blade loads control.

Embedded chordwise absorbers have some unique advantages, low profile (embedded devices), low drag and low weight penalty (utilizing existing leading edge mass). Elastomeric absorbers have been utilized to provide damping to soft in-plane or articulated rotor blades [11-14] and reduce the lagwise 1/rev and 2/rev loads of stiff in-plane rotor blades [8]. The large stroke of elastomeric absorbers for lagwise loads reduction is a major obstacle for the application in rotor blades due to the dimensional restriction of the blade cavity [8]. The large stroke of elastomeric absorbers introduces large local lagwise bending moment, and the strength of the local structure should be strengthened to bear this large moment. Fluidlastic technology combines fluids with bonded elastomeric elements to provide unique vibration isolation and damping capabilities [15-16]. Fluidlastic absorbers can be designed to have desirable high static stiffness and high dynamic mass at a specific frequency. This can help to reduce the stroke for absorbing the same amount of energy compared with elastomeric absorbers. This work concentrates on the 2/rev lagwise load reduction of stiff in-plane blades by placing fluidlastic absorbers at the blade tip. System modeling is derived based on the generalized force formulation, including a rotor model, a fluidlastic absorber model, a fuselage model and a propulsive trim method. The achieved lagwise loads will be analyzed to investigate how much loads can
be reduced with as small stroke as possible, as seen in Fig. 1. Parametric effects are explored to investigate the influence on the performance of the fluidelastic absorber.

2 MODEL

The system modeling primarily includes a rotor model, a fluidelastic absorber model, a fuselage model and a propulsive trim method.

2.1 Rotor Model

This rotor modeling includes a moderate deflection composite beam model, blade kinematics, rotor aerodynamics and an induced velocity model.

2.1.1 Composite Beam Model

Composite materials are widely applied in manufacturing helicopter rotor blades. A sophisticated moderate deflection composite beam model is utilized to describe the anisotropy characteristics of composite beams[17]. This model is based on the assumptions of small strain and moderate rotations of beam cross section. The variation of the strain energy can be written as

$$\delta U = \sum_{i=1}^{n} Q_{i}^{k} \delta q_{i}$$

$$= \int \left( F_{x} \delta u_{x} + M_{x} \delta \phi + M_{y} \delta \kappa_{y} + M_{z} \delta \kappa_{z} \right) dL$$

(1)

The axial linear strain and the second-order moment strain measure are

$$u'_{x} = u + \frac{1}{2} \left( v^{2} + w^{2} \right)$$

(2)

$$\kappa =\begin{pmatrix} \kappa_{x} \\ \kappa_{y} \\ \kappa_{z} \end{pmatrix} = \begin{pmatrix} -C_{w} w^{n} + S_{w} v^{n} \\ S_{w} w^{n} + C_{w} v^{n} \end{pmatrix}$$

(3)

where, $C_{w}$ denotes $\cos \theta$, and $S_{w}$ is $\sin \theta$. $\theta$ equals the elastic twist angle plus the pretwist angle. With the nonlinear coupling terms between extension and torsion, the section loads are

$$\begin{bmatrix} F_{x} \\ M_{x} \\ M_{y} \\ M_{z} \end{bmatrix} = \begin{bmatrix} S_{uw} + \frac{1}{2} \phi' S_{uw} k_{p}^{2} \\ S_{wp} \\ S_{ww} \end{bmatrix} \begin{bmatrix} u'_{x} \\ \kappa_{x} \\ \kappa_{y} \\ \kappa_{z} \end{bmatrix}$$

(4)

feathering hinge or some combinations of these hinges. Rotor blades also rotate around a rotor shaft. For the convenience of the description of the kinematics of the rigid motions, the rotation is usually defined in the local rotating frame. The rotation about rotor shaft, rigid flap, rigid lag and feathering motion are described as generalized coordinates.

The $4 \times 4$ transformation matrix is utilized to describe the motion of rigid rotation [18]. The position vector of an arbitrary point, as seen in Fig. 2, rotating around an axis $\mathbf{v}$ with a translation distance $\mathbf{d}$ is

$$\mathbf{R} = \mathbf{d} + \mathbf{Ar}$$

(5)

This expression can be applied to describe any rigid rotation around a prescribed axis. Unless otherwise stated, the right-hand-rule
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has been observed for all coordinate systems
in this work. The above expression can be
written as

\[
\begin{bmatrix}
 R \\
 1
\end{bmatrix} = T_{4x4} \begin{bmatrix}
 r \\
 1
\end{bmatrix} = \begin{bmatrix}
 A & d \\
 0 & 1
\end{bmatrix} \begin{bmatrix}
 r \\
 1
\end{bmatrix}
\] (7)

where, the vector of the axis is

\[
v = \begin{bmatrix}
 v_x \\
 v_y \\
 v_z
\end{bmatrix}
\]

and the rotation matrix is

\[
A = I + \tilde{v} \sin \theta + \tilde{v}^2 (1 - \cos \theta)
\] (8)

In the above expression, the sign '~' denotes

\[
\begin{bmatrix}
 -v_3 \\
 v_2 \\
 v_1 \\
 v_3
\end{bmatrix}
\] (9)

The advantage of the notation in Eq. 7 is the
combination of the translation and rotation in a single matrix.

If a blade has several rigid rotations (for example, \(m\)), the position vector after the elastic deformations and rigid rotations can be written as

\[
\begin{bmatrix}
 R \\
 1
\end{bmatrix} = H \begin{bmatrix}
 d \\
 1
\end{bmatrix} = T_{n} \cdots T_{2} \cdots T_{1} \begin{bmatrix}
 r_0 \\
 1
\end{bmatrix}
\] (10)

The transformation matrix \(T_i\) denotes a rigid rotation (\(i > 0\)). The determination of the rigid rotations is according to the rotor type.

The methodology derived by Zheng [19] to calculate the kinetic energy is employed to describe the nonlinear coupling characteristics between elastic deflections and rigid rotations. The variation of the kinetic energy of the blade is

\[
\delta T = \sum_{i=1}^q \int_{A} Q_i^T \delta q_i = \sum_{i=1}^q \int_{A} \rho \frac{\partial \mathbf{R}}{\partial q_i} \cdot \frac{\partial \mathbf{R}}{\partial q_i} dAdl \delta q_i
\] (11)

and the tangent mass, damping, and stiffness matrices introduced by the kinetic energy are

\[
M_{ij}^T = \frac{\partial Q_i^T}{\partial q_j} = -\int_{A} \rho \frac{\partial \mathbf{R}}{\partial q_i} \cdot \frac{\partial \mathbf{R}}{\partial q_j} dAdl
\] (12)

\[
C_{ij}^T = \frac{\partial Q_i^T}{\partial q_j} = -\int_{A} \rho \frac{\partial \mathbf{R}}{\partial q_i} \cdot \frac{\partial \mathbf{R}}{\partial q_j} dAdl
\] (13)

\[
K_{ij}^T = \frac{\partial Q_i^T}{\partial q_j} = -\int_{A} \rho \left( \frac{\partial \mathbf{R}}{\partial q_j} \cdot \frac{\partial \mathbf{R}}{\partial q_i} + \mathbf{R} \cdot \frac{\partial^2 \mathbf{R}}{\partial q_j \partial q_i} \right) dAdl
\] (14)

The generalized force, and the matrices introduced by the kinetic energy, can be obtained from the position vector \(R\) in Eq. 10.

2.1.3 Aerodynamics

The velocity of an arbitrary point on the pitch axis with respect to the local airflow is

\[
\begin{bmatrix}
 U_R \\
 U_T \\
 U_p
\end{bmatrix} = H^T \begin{bmatrix}
 0 \\
 \mathbf{R} - \mathbf{V}_\infty + T_{qpv} \\
 v_j
\end{bmatrix}
\] (15)

where, the air flow velocity vector \(\mathbf{V}_\infty\) is defined in the inertial frame. The relative velocity of an arbitrary point on the blade pitch axis to the local air flow is determined by the flight state and the blade kinematics described in the kinetic energy section. According to the angle of attack and the resultant velocity, the lift coefficient \(C_l\), drag coefficient \(C_d\) and pitching moment coefficient \(C_m\) of the airfoil segment is determined by Leishman-Beddoes aerodynamics with dynamic stall [20]. Drees’ inflow model is utilized to determine the distribution of induced velocity over the rotor disk [21].

The variation of the work done by the aerodynamics is
\[
\delta W_1 = \sum_{i=1}^{N_r} Q_i^e \delta q_i = \left[ \begin{array}{c} \delta R_i \cdot \frac{\partial R_i}{\partial q_i} + M_i \cdot \frac{\partial a_i}{\partial q_i} \end{array} \right] \delta q_i \, dt \quad (16)
\]

It should be noted that the aerodynamic force vector \( F_A \), aerodynamic moment vector \( M_A \), and angle vector \( a_s \) are defined in the inertial coordinate frame.

2.2 Fluidlastic Absorber Model

The configuration and mathematical model of a fluidlastic absorber are shown in Fig. 3. The basic mechanism is described in Ref. 22. The fluid in the absorber acts as a kinematic amplifier and its velocity is \( (G-1) \) times that of the prime mass. The relation between the displacement of the prime mass and tuning mass is

\[
a_t = a_{t0} - (G-1)a_p \quad (17)
\]

Large tuning port area ratio can be utilized to amplify the kinetic energy absorbed by the absorber. In this way, it can significantly reduce absorber mass and decrease the stroke. According to the mathematical model of the fluidlastic absorber, the non-rotating natural frequency is

\[
f = \frac{k_p}{\sqrt{m_p + (G-1)^2 m_t}} \quad (18)
\]

For the same weight of the absorber, larger tuning port area ratio is corresponding to larger spring stiffness for the same natural frequency. The spring acts on the prime mass and the fluid has no effect on the static displacement of the absorber. In this way, the static displacement of the prime mass can be reduced significantly due to the large spring. Under rotation, the absorber at the blade tip must bear large centrifugal force.

The fluid is encapsulated in the container, which can limit the variation of the stroke and mass center of the absorber. The absorber is treated as a one degree-of-freedom rigid body with chordwise motion. The position vector of the prime mass and the tuning mass can be written as

\[
\begin{align*}
\{ R_p \} & = T \begin{bmatrix} r_p \\ 1 \end{bmatrix} = T \begin{bmatrix} 0 \\ a_p + a_{p0} \\ 0 \\ 1 \end{bmatrix} \\
\begin{bmatrix} R_t \\ 1 \end{bmatrix} & = T \begin{bmatrix} r_t \\ 1 \end{bmatrix} = T \begin{bmatrix} a_t + a_{t0} \\ 0 \\ 1 \end{bmatrix}
\end{align*}
\]

(19) (20)

The contribution from the absorber to the generalized force, tangent mass, stiffness and damping matrices can be derived by substituting the above two expressions in Eq. 11 to Eq. 14. The elastic potential energy of the absorber is \( \frac{1}{2} k_p a_p^2 \) and the work done by the viscous force is \( \frac{1}{2} c_r a_p^2 \).

2.3 Equations Of Motion

By using Hamilton’s principle, the implicit nonlinear dynamic equations of motion based on the generalized force formulation include four parts: blade elastic potential energy, blade kinetic energy, work done by vector \( M_A \) and angle vector \( a_s \) are defined in the inertial coordinate frame.

![Image of mechanical and mathematical configuration of a fluidlastic absorber](image-url)
the aerodynamics and the energy stored by the fluidlastic absorber. The equations of motion are
\[ Q_i^T(q) + Q_i^T(q, \dot{q}, \ddot{q}, t) + Q_i(q, \dot{q}, t) = 0 \]
\[(i = 1, \cdots, n) \]
An Implicit-Newmark integration method is utilized to calculate the periodic responses [23]. This unconditionally stable implicit scheme permits the use of large time integration step. This step is determined by the considerations of accuracy.

2.4 Propulsive Trim

For the propulsive trim analysis, the fuselage is treated as a rigid body with aerodynamic forces and moments on it. Three pitch controls and two rotor attitude angles are the input variables to solve the three force and two moment equilibrium equations [21], as seen in Fig. 4. The control input vector is
\[ \mathbf{x} = \{ \theta_0, \theta_1, \theta_2, \alpha_s, \phi_i \}^T \]
and the output or target vector is
\[ \mathbf{y} = \{ y_1, y_2, y_3, y_4, y_5 \}^T \]

The equilibrium equations in non-dimensional form are
\[ \begin{align*}
y_1 &= C_T - C_W \\
y_2 &= C_D + C_H - C_T \alpha_s \\
y_3 &= C_Y + C_{Yr} + C_T \phi_i \\
y_4 &= C_{M_T} + C_{M_{Yr}} + C_W (\bar{h} \alpha_s - \bar{x}_{CG}) - \bar{h} C_D \\
y_5 &= C_{M_X} + C_{M_{Yr}} + C_W (\bar{h} \phi_s - \bar{y}_{CG}) + \bar{h} C_{Yr}
\end{align*} \]

Newton–Raphson method is utilized to calculate the trim values in hover and forward flight.

Fig. 4  Forces and moments on the rotor and fuselage.

3 AEREOELASTIC ANALYSIS

The aeroelastic analysis of the blade-absorber system in forward flight is presented. Comprehensive parametric analysis is conducted to investigate the performance of the 2/rev absorber.

3.1 Baseline Parameters

The baseline blade is a non-uniform stiff in-plane blade, as seen in Fig. 5. The weight of the helicopter is 3000kg. The parameters of the rotor and the fluidlastic absorber are shown in Table 1 and Table 2. The blade mass density from 70% - 80% and 90% - 100% is 14kg/m, and the density in other places is 9kg/m. To keep the same gross weight of the two blade systems, the mass density of the blade with an absorber is uniformly 9kg/m, and the mass of the absorber is 6kg. The initial position of the absorber coincides with the pitch axis. When comparing the lagwise load, the fundamental lagwise frequency of
the two blades are the same by changing the lagwise stiffness of the blades. The non-dimensional rotating frequencies of the blade are listed in Table 3. In the fuselage model, only aerodynamic drag is considered in the aeroelastic analysis. The ratio of the equivalent flat plate area to the rotor disk area is 0.006.

![Configuration of rotor blade with and without absorber.](image)

**Table 1 Rotor Parameters**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor Radius</td>
<td>6m</td>
</tr>
<tr>
<td>Chord Length</td>
<td>0.4m</td>
</tr>
<tr>
<td>Number of Blades</td>
<td>4</td>
</tr>
<tr>
<td>Airfoil</td>
<td>NACA0012</td>
</tr>
<tr>
<td>Rotor Speed</td>
<td>300RPM</td>
</tr>
<tr>
<td>Pretwist</td>
<td>0°</td>
</tr>
<tr>
<td>Blade Mass</td>
<td>60kg</td>
</tr>
</tbody>
</table>

**Table 2 Parameters of the fluidelastic absorber**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Prime Mass</td>
<td>3kg</td>
</tr>
<tr>
<td>Tuning Mass</td>
<td>3kg</td>
</tr>
<tr>
<td>Critical Damping</td>
<td>1%</td>
</tr>
<tr>
<td>Tuning Port Area Ratio</td>
<td>11</td>
</tr>
<tr>
<td>Location</td>
<td>Blade Tip</td>
</tr>
</tbody>
</table>

**Table 3 Frequencies of the rotor blade**

<table>
<thead>
<tr>
<th>Mode</th>
<th>Baseline</th>
<th>With 2/rev absorber</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st flap</td>
<td>1.13</td>
<td>1.13</td>
</tr>
<tr>
<td>2nd flap</td>
<td>3.67</td>
<td>3.61</td>
</tr>
<tr>
<td>1st lag</td>
<td>1.54</td>
<td>1.54</td>
</tr>
<tr>
<td>1st torsion</td>
<td>4.82</td>
<td>4.82</td>
</tr>
</tbody>
</table>

### 3.2 Analysis Of The 2/Rev Absorber

The absorber is tuned to control the 2/rev lagwise root bending moment. For different tuning frequencies of the absorber and forward speeds, the 2/rev bending moments are shown in Fig. 6. With increase tuning frequency, the performance of the absorber increases first, and then decreases. At the forward speed 30km/h, 81.0% of the moment is reduced, when the absorber is tuned to 10.9Hz. At low forward speeds, the performance of the absorber is not very effective for these low loads states. It can reduce 93.2% 2/rev lagwise root bending moment at the speed 200km/h. The performance of the absorber improves significantly at high load states. The corresponding stroke of the absorber is shown in Fig. 7. The stroke increases with increasing forward speed, and decreases with increasing tuning frequency. When the absorber is tuned to 10.9Hz, the stroke is 1.51% the blade chord length, which is a relatively very small value compared with the blade cavity size. Fig. 9 and Fig. 10 depict the 1/rev and 3/rev lagwise root bending moments with forward speed for different tuning frequencies. These two figures illustrate that the tuning frequency has substantially small influence on the other harmonic lagwise loads.
Fig. 6  Influence of the tuning frequency on the 2/rev moment.

Fig. 7  Influence of the tuning frequency on the stroke.

Fig. 8  The first harmonic load for different tuning frequencies and forward speeds.
Fig. 9  The third harmonic load for different tuning frequencies and forward speeds.

Fig. 10  Influence of the damping of the absorber on the performance.

Fig. 11  Influence of the damping on the stroke.

Usually, large damping can decrease the vibration control ability of dynamic absorbers. The absorber is tuned to 10.9 Hz. The influence of the damping of the absorber on the performance and stroke is shown in Fig. 10 and Fig. 11. The lagwise root bending moment increases with increasing damping. That means the decrease of load control ability. It is necessary to reduce the damping of the absorber. The stroke of the absorber decreases with increasing damping and increases with increasing forward speed. Even for the cases with low damping and high forward speed, the stroke is less than 2% blade chord length, which is a small
value compared with the blade chord length. Fig. 12 depicts the influence of the initial position in blade lagwise direction on the performance of the fluidlastic absorber. The moment ratio used in the following is defined as the ratio of the root bending moment generated with the absorber to the moment generated by the blade without the absorber. It is obvious that the absorber offsetting 5% chord length near the blade leading edge can increase the performance by 5.3% at the forward speed 100km/h. If offsetting 5% chord length near the trailing edge, the moment ratio can decreases by 3.3%. At the forward speed 200km/h, the variation from 5% chord length to -5% chord length is 6.7%. The variation of the performance of the absorber is substantially small. When designing the setup position in the blade cavity, the blade-absorber system, it is necessary to consider the effect on blade flutter problem.

For different tuning port area ratios, the performance and stroke of the absorber are shown in Fig. 13, when the forward speed is 200km/h. The performance of the absorber decreases with tuning port area ratio. When the tuning port area ratio increases from 10 to 100, the moment ratio changes from 5.89% to 12.9%. The corresponding stroke ratio changes from 1.71% to 0.14%. The variation of the performance of the absorber is substantially small, and the stroke decreases significantly with tuning port area ratio. Large tuning port area ratio seems to be unnecessary due to the small stroke even at small tuning port area ratios.

Fig. 14 depicts the influence of the flight altitude on the performance of the absorber. The overall moment ratio is less than 15%, which means that the absorber can reduce the moment by more than 85% during the forward speed 100km/h to 200km/h and the flight altitude 0km to 3km. It is obvious that the variation of the performance of the absorber is limited. The flight state has substantially small influence on the performance. These flight speeds and flight altitudes usually contain most typical steady flight states, which means fluidlastic absorbers are an effective means to control the second harmonic lagwise load during different flight states.

For different tuning port area ratios, the performance and stroke of the absorber are shown in Fig. 13, when the forward speed is 200km/h. The performance of the absorber decreases with tuning port area ratio. When the tuning port area ratio increases from 10 to 100, the moment ratio changes from 5.89% to 12.9%. The corresponding stroke ratio changes from 1.71% to 0.14%. The variation of the performance of the absorber is substantially small, and the stroke decreases significantly with tuning port area ratio. Large tuning port area ratio seems to be unnecessary due to the small stroke even at small tuning port area ratios.
4 CONCLUSION

To reduce the second harmonic lagwise load of stiff in-plane rotor blades, fluidelastic absorbers are utilized to control the lagwise loads and decrease the stroke. The fluidelastic absorber is treated as a one degree-of-freedom rigid body. The rotor blade is treated as moderate deflection beam model undergoing coupled flap, lag and torsion deflections. A propulsive trim method is utilized to simulate the real flight state. The equations of motion of the system are derived based on the generalized force formulation. Comprehensive parametric analysis is conducted to explore the feasibility to control the blade lagwise loads, and reduce the large stroke. The investigation yielded the following conclusions:

1) The second harmonic fluidlastic absorber can reduce more than 85% corresponding harmonic lagwise root bending moment at high forward flight. The corresponding stroke of the absorber is less than 2% blade chord length.

2) The absorber has substantially small influence on the first and third harmonic lagwise loads.

3) The tuning port area ratio has substantially small influence on the performance of the absorber. Larger tuning port area ratio can introduce a little decreases of the performance of the absorber. Even with larger tuning port area ratio, more than 85% bending moment can be reduced. Increasing tuning port area ratio is an effective means to reduce the stroke of the fluidlastic absorber.

4) The position of the absorber in blade cavity has substantially small influence on the performance of the absorber.

5) The fluidlastic absorber works very well at different forward speeds and flight altitudes.

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