Rotating Structures and Components

The papers in this section are from two sessions. The first session was chaired by Major Larry Gross, Ph.D., and Allen C. Royal. The second session was chaired by Eric E. Abell.

Encapsulated Tuned Dampers for Jet Engine Component Vibration Control

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Nomenclature

b	= breadth of beam
E	= Young's modulus
E_D	= real part of complex Young's modulus of damping material
$(EI)_e{}^*,(EI)_e$	= effective (complex/real) flexural rigidity of beam
E_m,E_D	= Young's modulus of metal outer layer/viscoelastic core
f_n	= resonant frequency in nth mode (Hz)
G	= factor by which centrifugal loading exceeds unit gravity
h	= thickness of wall
I_D,Im	= second moment of area of viscoelastic core/metal outer layer about neutral axis
L	= length of beam
$p(x,t)$	= time dependent load per unit length
t	= time
x	= axial coordinate along beam, with root as origin
y	= transverse displacement of point on beam
β	= nondimensional parameter
η_s,η_D	= loss factor of composite beam/damping material
ξ	= $[b(\rho h)_e\omega^2L^4/(EI)_e]^{1/4}$
	= first eigenvalue of cantilever beam ($\simeq$12.36)
$(\rho h)_e$	= effective mass per unit length of composite beam
ρ_D,ρ	= density of damping material/metal outer layers
ω_o	= resonant frequency at zero rotational speed
ω_r	= resonant frequency (rad/sec)

Theme

MANY jet engine components suffer from serious vibrational resonance problems induced by the engine environment. Examples are stator vanes, inlet guide vanes, combustor cans, hollow shafts and turbine[1] and compressor blades. This paper describes a preliminary investigation of a tuned damper, utilizing damping material encapsulation to withstand centrifugal load and erosion problems. The tests were conducted at room temperature for convenience, but the results are equally applicable to higher temperatures if the appropriate damper materials are used (high-temperature enamels and metals). Specific tests on control of one mode of a turbine blade show a great reduction in vibration levels resulting from proper tuning of the damper.

Contents

The damper being examined consists of a hollow metal cantilever beam, of rectangular inner and outer section, the cavity of which is completely filled with a material which dissipates energy under cyclic strain. The damper is attached to the system to be damped, such as a hollow turbine blade, as shown in Fig. 1. Clearly, the size of the damper is of critical importance. The damper is a fairly complex device, and the state of strain in the core damping material is also very complex when the damper vibrates. The tests were conducted with the core material consisting of a commercially available stiff elastomeric damping material.‡ The properties of the material, in the form of graphs of the Young's modulus E_D and loss factor η_D vs temperature at various frequencies, were already known.[2,3] The material was then electrochemically coated with a copper layer to achieve the encapsulation. The dampers were clamped in a fixture and vibration tests conducted to determine modal damping and resonant frequencies as a function of temperature and damper

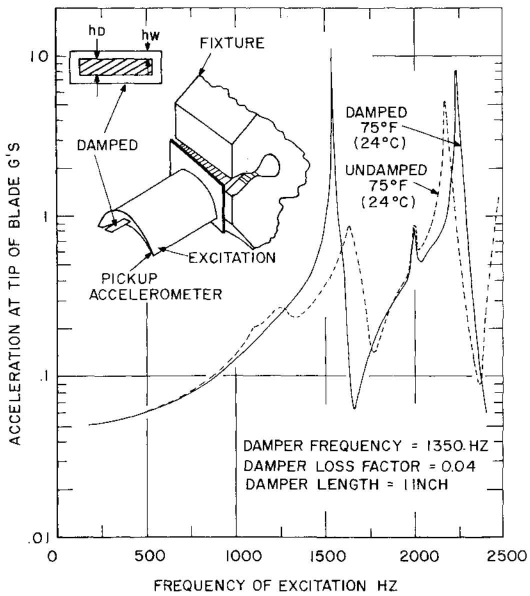

Fig. 1 Test system for measuring blade response with damper attached.

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‡LD400, a polymeric damping material made by Lord Mfg. Co., Erie, Pa.

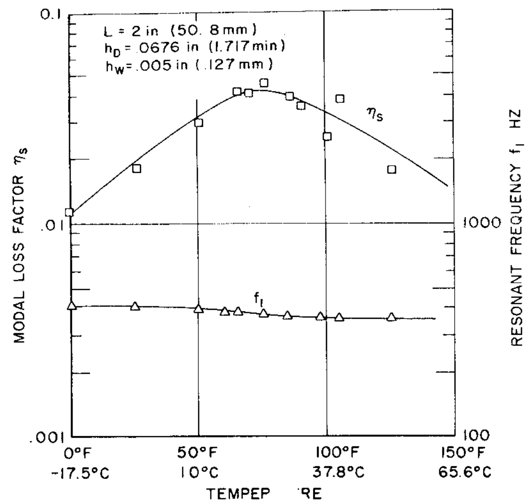

Fig. 2 η_s **and** f_1 **vs temperature for a typical damper.**

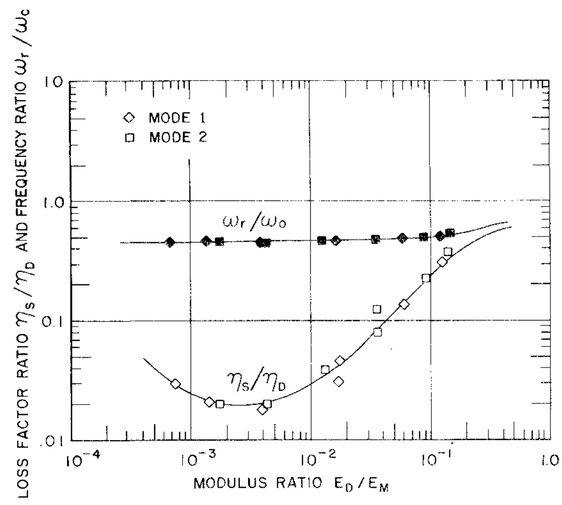

Fig. 3 η_s/η_D **and** ω_r/ω_o **vs** E_D/E_m.

length. The principal dimensions of the damper are the core thickness h_D, the wall thickness h, the breadth b, and the length L. The important nondimensional geometrical ratios are h/b, h/h_D and h/L. The dimensions chosen were h_D = 0.0676 in. (1.717 mm), h = 0.005 in. (0.127 mm), and L = 1 and 2 in. (25 mm) and b = 0.247 in. (6.27 mm). A typical graph of η_s and f_1 vs temperature for a typical damper is shown in Fig. 2. From this data, and the properties of LD-400, graphs of η_s/η_D and ω_r/ω_o vs E_D/E_m were plotted as in Fig. 3. It is seen that two regimes of high damper loss factor occur, one where E_D/E_m is high (>0.05), in which bending deformations dominate, and another for E_D/E_m low ($<2 \times 10^{-4}$), where shear mechanisms dominate. Figure 1 shows the test system used to measure the blade response with the damper attached. In the undamped case, the damper was removed. This figure also shows typical response spectra for the damped and undamped turbine blade at room temperature. It is seen that the two peaks usually associated with the application of a tuned damper to a single degree of freedom system[4,5] are clearly discernible and that the amplitude of the first mode, to which the damper was tuned, is considerably reduced.

A simple analysis was used to estimate the composite modal loss factor and natural frequency of such a beam, based on the classical equation of motion for transverse displacement of a thin beam in bending, with the usual simplifying assumptions made. For the equivalent homogeneous Euler-Bernoulli beam, the effective flexural rigidity is

$$(EI)_e^* = (EI)_e(1 + i\eta_s) = E_m\left[\frac{2b}{3}\left\{\left(\frac{h_D}{2} + h\right)^3 - \left(\frac{h_D}{2}\right)^3\right\} + \frac{4h}{3}\left(\frac{h_D}{2}\right)^3\right] + E_D(1 + i\eta_D)\frac{2}{3}(b - 2h)\left(\frac{h_D}{2}\right)^3$$

where the first term represents the stiffness of the encapsulated layer and the second term that of the coating. Finally, from the classical normal mode solution of the Euler-Bernoulli equation, the fundamental frequency of the composite beam is given by

$$b(\rho h)_e\omega_r^2L^4/(EI)_e = \xi_1^4$$

$$\therefore \omega_r^2 = \xi_1^4(EI)_e/b(\rho h)_eL^4$$

where

$$(\rho h)_e = \rho_D h_D(1 - 2h/b) + 2\rho h + 2\rho h_D(2h/b)$$

and

$$\eta_s = Im(EI)_e^*/Re(EI)_e^*$$

If the damper is assumed to rotate, in the turbine blade, at a uniform speed and in an approximately uniform centrifugal field over its length, then the modified Euler-Bernoulli equation of motion can readily be shown to be

$$(EI)_e^*(\mathrm{d}^4y/\mathrm{d}x^4) + (\rho h)_e bLG(1 - x/L)(\mathrm{d}^2y/\mathrm{d}x^2) - (\rho h)_e bLG(\mathrm{d}y/\mathrm{d}x) + (\rho h)_e b(\mathrm{d}^2y/\mathrm{d}t^2) = p(x,t)$$

It is not easy to obtain an exact solution for this equation. However, if we expand $y(x)$ and $p(x)$, for the case of harmonic loading, as generalized eigenfunction series and neglect all terms except the first (low modal coupling), we can readily arrive at the approximate solution for the specific set of boundary conditions:

$$\xi^4 = 12.36 + 1.566\beta$$

$$\therefore \omega_r/\omega_o = (1 + 0.1267\beta)^{1/2}$$

where $\beta = (\rho h)_e bLG/(EI)_e$

When the damper rotates uniformly in a centrifugal field, the enamel will behave as a classical viscous fluid with high viscosity and eventually a steady state condition will be reached in which the pressure within the enamel, which will have to be withstood by the metal walls, varies linearly from zero at the tip to a maximum at the root. The peak pressure $\hat{p}$ is simply

$$\hat{p} = G\rho_D L$$

This variable pressure over the walls of the damper will cause them to bend outwards and a tendency to rupture will occur. The walls will have to be designed to withstand such loads.

It has been shown in this paper that an encapsulated tuned damper can be designed, with a sufficiently high loss factor (Fig. 2) to be effective in controlling the vibration levels in a jet engine component, a turbine blade in this case. The effects of centrifugal loading, both in altering the tuning of the damper and in causing failure of the damper under "hydrostatic loads" have been discussed. The real world engineering problems of making a high temperature encapsulated damper, utilizing engine-compatible enamels and metal coatings, instead of the room temperature elastomer used here, have yet to be fully addressed and will require considerable effort in the future in order to achieve the most practical configurations. The promise of such a damper is that if it can be made to work in a real jet engine environment, a great deal of auxiliary structure now used externally to the components will in due course be eliminated, with considerable savings in weight and improvement in efficiency.

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High-Speed Rotor Dynamics—An Assessment of Current Technology for Small Turboshaft Engines

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An extensive study was made to determine current needs for research in rotor dynamics to solve problems encountered in small high-speed turboshaft engines for helicopter and aircraft propulsion. The purpose of this paper is to report the state-of-the-art for this area as completely and concisely as possible. The present and past philosophy of rotor-bearing system design including the impact of the demand for front drives, is discussed. Methods for critical speed prediction and high-speed balancing are reviewed. The trend to higher speeds is seen to require consideration of new approaches to balancing through flexural modes. The major parameters available for control by the designer are shown to be the bearing support properties, and recommendations are made for improving the accuracy of prediction of these properties. Nonsynchronous excitation is categorized according to the mechanisms producing the forces, and a need is shown for better methods to identify the resulting whirling and vibration, since several of these motions are potentially unstable. Finally, reasons are given for the predominant use of rolling-element bearings in these engines, and the potential for special applications of oil-film and gas bearings is discussed.

I. Introduction

MODERN design of small turboshaft engines is characterized by ever increasing power weight and power/size ratios. The increase in performance is being obtained in part from higher shaft speeds, which has heightened the importance of rotor dynamics considerations in the design and development process.

In particular, there are a number of problem areas connected with rotor dynamics which are peculiar to the special requirements of rotor-bearing systems of small turboshaft engines for helicopter or aircraft propulsion. These special requirements are: 1) Increasingly higher shaft speeds. Increased airflow and power output can thus be obtained without an increase in physical size requirements. 2) Small frontal area and light weight. 3) Front drive. It is usually desired to locate the power takeoff shaft out through the front of the compressor section. 4) Maintainability. Individual components making up the rotor-bearing assembly should be easily replaceable. 5) Long life. It is desired to increase the engine operating life significantly. Frequency of overhauls should be reduced.

Even if only taken one at a time on an individual basis, these requirements would generate significant problems for the rotor-bearing design engineer. Taken together simultaneously, these requirements are a severe challenge to meet. For example, the combination of requirements 1 and 2 has resulted in a blade tip clearance problem in small engines. The very short length blades require extremely small tip to housing clearances to maintain high aerodynamic efficiency and high power output. This in turn requires very small rotor shaft excursions to avoid blade-housing interferences, a design condition which is incompatible with low dynamic bearing loads at high speeds.

An extensive study was made by the authors for the U.S. Army (USAAMRDL, Eustis Directorate, Propulsion Technical Area) to determine the current needs for research in rotor dynamics. From the standpoint of Army aviation, there are two broad objectives to be met in solving rotor dynamics problems through a program of directed research. One objective is to reduce the magnitude and frequency of rotor dynamics-related failures and required redesign efforts in propulsion hardware development programs. Another objective is to improve the reliability and maintainability of future engines in the field through reduction of vibration and dynamic bearing loads.

The purpose of this paper is to report the current state-

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