Stationary Structures

The papers in this section are from a session with the same title chaired by William D. Cowie.

Control of Gas Turbine Stator Blade Vibrations by Means of Enamel Coatings

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This paper describes the application of a high temperature enamel coating to part of the surface area of the stator vanes of a jet engine, to increase the damping and thereby reduce aerodynamically induced resonant vibration failures. The investigation showed that such a coating can increase the damping level by a very significant amount, particularly when it is initially very low as when the blades are stalled, for example. The use of enamel coatings is a relatively new approach to resolving jet engine vibration problems, and many engineering problems have to be overcome before all high temperature vibrations are controllable. Some of the engineering problems are discussed.

Nomenclature

b = breadth of beam
E_D = real part of complex Young's modulus of damping material
f_n = nth resonant frequency
f_{1B} = first bending mode frequency ($= f_1$)
f_{1T} = first torsional mode frequency ($= f_2$)
h_m = thickness of metal beam
h_D = thickness of coating
L = length of beam
ρ_m = density of beam material
ρ_D = density of coating material
η_s = modal loss factor
η_D = loss factor of coating material

Introduction

THE purpose of this paper is to describe the application of enamel coatings to reduce vibration levels in the stator vanes of a high performance jet engine. Vibratory stress levels in several stator stages, including the 10th and 13th, were found to be excessively high because the vanes were operating in a stalled condition so that the levels of aerodynamic damping were very low. One of the possible approaches was to apply a suitable glass-like coating over part of the external surface of each vane to increase the damping, through the energy dissipating capacity of the coating under cyclic strain at high temperature, above the softening temperature but below the melting point.

Tests were first conducted to determine the vibration modes, modal damping, and resonant frequencies of the vanes in a section of the tenth stage. From this information, it was thought to be possible that a damping treatment applied in the form of a thin coating on the vane surface near the attachment points to the outer (o.d.) and inner (i.d.) shrouds might be successful in achieving some increase in damping levels, if the correct material were selected for the coating. To verify this possibility, tests were conducted using a coating consisting of a polymeric room-temperature damping material. The results of these tests showed that modal loss factors in the lowest frequency mode could be increased from about 0.00025, an extremely low figure, to over 0.010. This represented a factor of 40 by which the damping was increased. Other modes showed comparable, though smaller, increases.

Tests were then conducted, using vibrating cantilever beams rather than the stator blades, to find a coating material having the appropriate high temperature damping behavior, and to determine the complex modulus proper-

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Index categories: Structural Composite Materials (Including Coatings); Aircraft Vibration.

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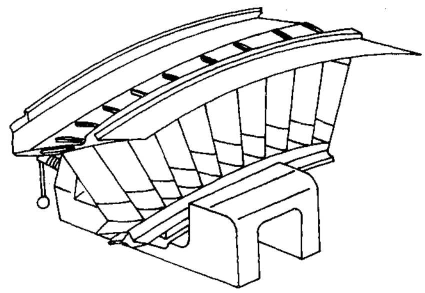

Fig. 1 Sketch of stator section in vise.

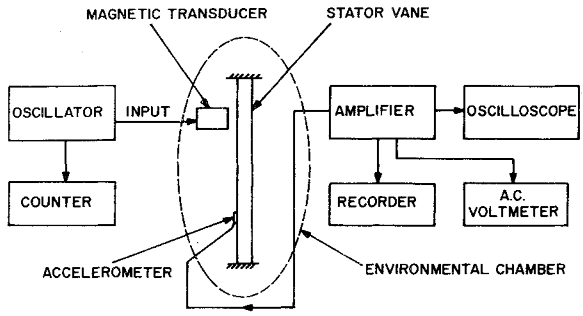

Fig. 2 Block diagram of test system.

ties of this material. The damping behavior of the porcelain enamel coating was evaluated over a wide range of frequencies and temperatures, and these properties compared qualitatively with those of the room temperature material at its peak damping temperature. The appropriateness of the selection having been determined, tests were then conducted to measure the effect of the enamel coating on the dynamic response of a single vane, under laboratory conditions, to verify the damping levels attainable. The coating material was then applied directly to the vanes of the 13th stage stator ring which was then incorporated in an engine and subjected to endurance testing. The thirteenth stage was coated because, within the period of the investigation, revised temperature estimates showed that the temperature originally expected in the 10th stage were actually occurring in the thirteenth. The engine tests showed that an erosion resistant coating, such as magnesium zirconate, was needed over the enamel. This step is well within the state-of-the-art, was applied in an experimental test, and will be incorporated in future engine tests.

Experimental Investigation

Vibration Response Characteristics of Stator Vanes

To determine the vibration response behavior of a typical stator vane, a ring section containing ten vanes and the (i.d.) and (o.d.) shrouds was clamped in a vise as shown in Fig. 1. The center vane was excited by a magnetic transducer driven by an oscillator, and the response was picked up by a miniature accelerometer (about 0.2 g mass). A block diagram of the test system is shown in Fig. 2. A typical undamped, and a damped, response spectrum, showing acceleration amplitude versus frequency for constant force input, is illustrated in Fig. 3. It is seen that significant modes of vibration occurred, for the tenth stage stator ring segment, at about 2750, 6094, 7611, and 12198 Hz. The mode at 2750 Hz was identified, by means of laser holographic techniques, as the first bending mode (1B), while that at 6094 Hz was the first torsion mode (1T). The other modes were not identified, though they are clearly important and were investigated in the damping tests. Figure 4 shows the sensor locations.

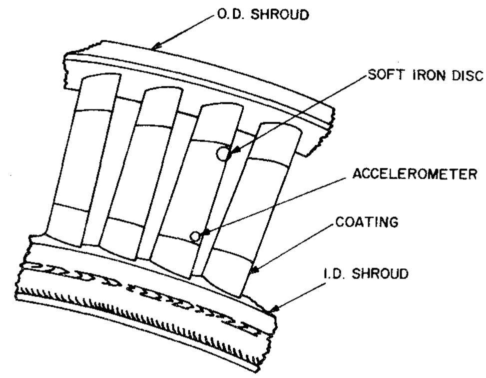

Fig. 4 Sketch of vanes with room temperature damping treatment.

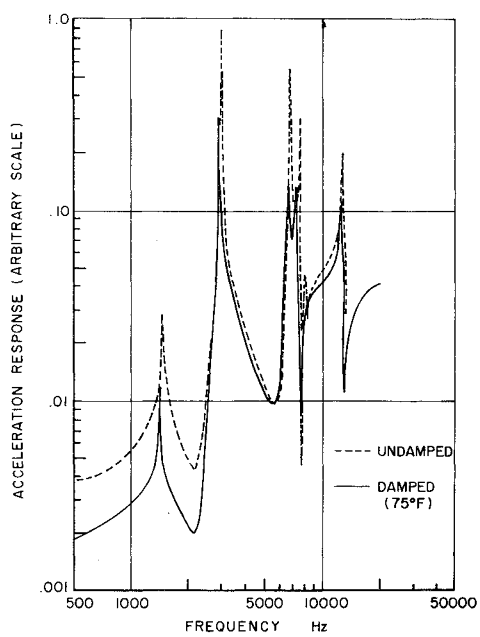

Fig. 3 Typical response spectrum.

Effect of Room Temperature Coating on Vane Response

To determine the effect of a damping coating on the modal damping of the vanes, it was simplest first to conduct tests at room temperature using a coating material having damping behavior qualitatively similar to that of the enamel, but near room temperature instead of at high temperature where the initial tests would be far more difficult to conduct. This room temperature damping material[1] was available in the form of damping tiles (LD-400,

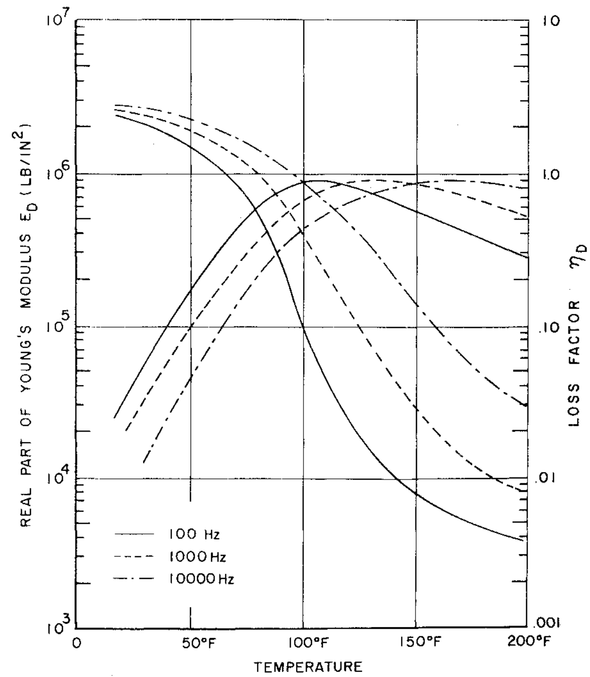

Fig. 5 Complex modulus properties of room temperature damping material.

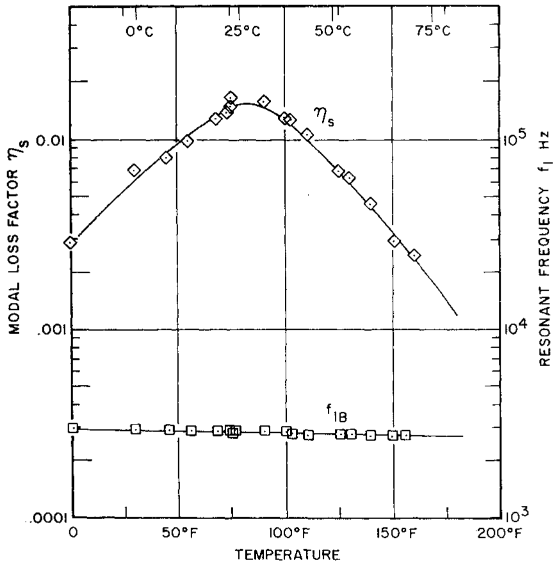

Fig. 6 Modal damping and resonant frequency of vane in first mode (1B).

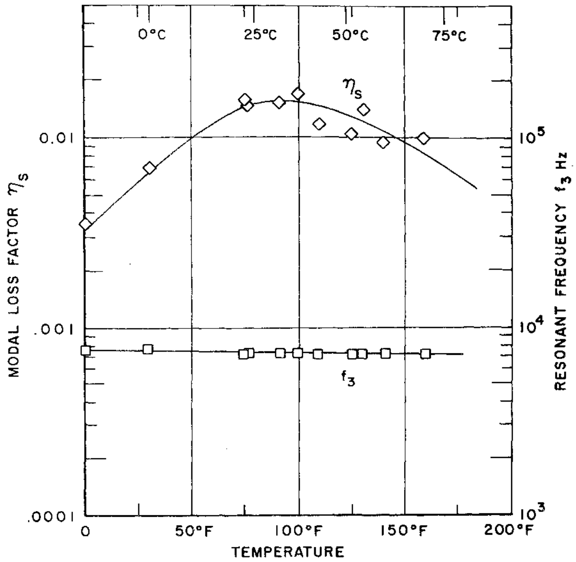

Fig. 8 Modal damping and resonant frequency of vane in third mode.

Lord Mfg. Co.), in a minimum thickness of 0.02 in. (0.508 mm). The material was applied to the vane surface by means of a stiff adhesive in strips 0.5 in. wide (12.7 mm) at the o.d. and i.d. ends of each vane and on both the convex and concave sides. The complex modulus properties of the material, determined from vibrating beam and other test techniques,[1,2] are shown in Fig. 5 as a function of temperature for several frequencies. The modal damping and resonant frequencies of the damped vanes were measured in the same way as for the undamped case. Figures 6–8 show graphs of the modal loss factor η_s and resonant frequency f_n for the first three modes, as a function of temperature. It is seen that the damping is increased significantly in each case.

Complex Modulus Properties of High Temperature Enamel

The material selected for application in the engine endurance tests was a commercially available porcelain enamel (CV-16845, Chicago Vitreous Corp.), having a softening region around 950°F (510°C), which was the expected operational temperature. The complex modulus properties of the enamel were evaluated using well established vibrating beam test techniques.[3,4] The test system is illustrated in Fig. 9. A typical graph of modal loss factor η_s and frequency f_r for one particular coated beam is illustrated in Fig. 10. From this data, and similar data for other modes and beams, one can determine the Young's modulus E_D and loss factor η_D of the material using well-known analyses.[3,4] Typical graphs of E_D and η_D vs temperature for several frequencies are illustrated in Fig. 11. From these results, it is seen that the enamel behaves, around 950°F (510°C), in a qualitatively similar way to the room temperature damping material around 75°F (24°C), although the enamel has the advantage of being somewhat stiffer. It was therefore concluded that the observed damping properties of the enamel should lead to adequate levels of damping in the stator vanes at high temperature. The coating was applied in a nominal thickness of 0.010 in. (0.254 mm) to the o.d. and i.d. areas of a single 10th stage vane and to an entire 13th stage. The width of the coating was 0.50 in. (12.7 mm) for the 10th stage vane and 0.25 in. (6.35 mm) for the thirteenth stage, at each end. Since the 13th-stage vanes were about half the length and thickness of the tenth stage vanes, the relative dimensions of the treatment did not change significantly. Figure 12 shows a photograph of the enamel coating on the vanes.

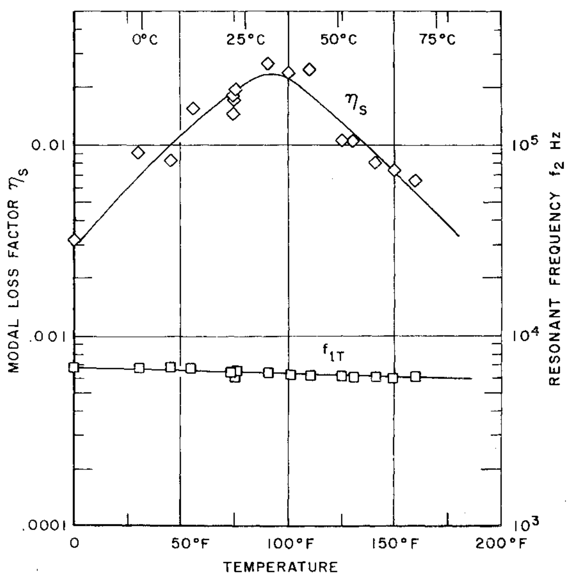

Fig. 7 Modal damping and resonant frequency of vane in second mode (1T).

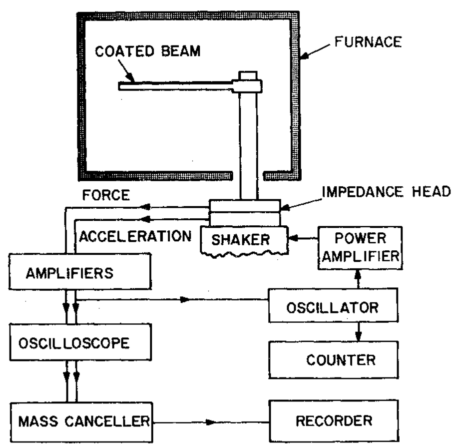

Fig. 9 Complex modulus measurement system.

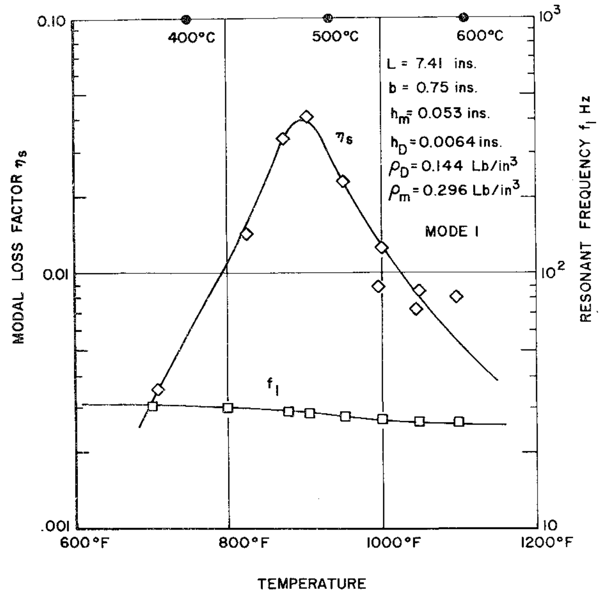

Fig. 10 Modal damping and resonant frequency of enamel coated beam.

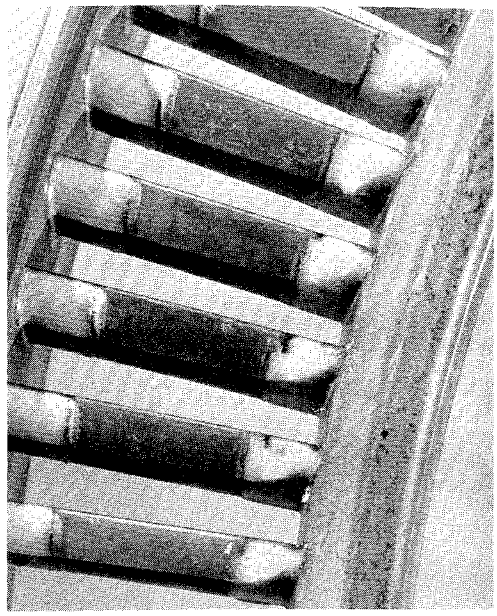

Fig. 12 Photograph of enamel coating on thirteenth stage stator vanes.

Effect of High Temperature Coating on Vane Damping

A vibration response investigation was also conducted on a single stator vane, coated with the enamel in a thickness of 0.01 in. (0.254 mm) and width 0.50 in. (12.7 mm), in a furnace, with a high temperature strain gage used to measure the response. Figure 13 shows the measured damping in the first bending mode as a function of temperature. The test system used was identical to that shown in Fig. 9, apart from the use of the strain gage. Tests were also conducted with a further magnesium zirconate coating applied, by plasma spraying, over the enamel in a thickness of 0.008 in. (0.203 mm). Measured damping is shown in Fig. 13.

Discussion and Conclusions

This investigation has shown that a damping treatment can be developed to reduce vibration levels in stator vanes. The amount of damping attainable has been established, for one type of stator geometry, using room temperature and elevated temperature dynamic response tests. A porcelain enamel coating has been identified, and its damping properties measured, which is suitable for application around 950°F (510°C).

More effort is clearly needed to fully develop a coating appropriate for every stage of an engine or engines. One would not apply such a coating unless the presence of an unduly low level of damping necessitated such an approach, and problems can be expected to occur at various temperatures, some well below 950°F and some well above. Such problems may occur whenever the aerodynamic damping is low, for example when the vanes are stalled, and when the excitation is too high. Creep problems are not expected to occur in service, since the vanes are stationary, but some flow and/or erosion can be expected if the temperature rises unduly. If, in any particular instance, it is found that such flow or erosion does occur to a significant degree, it can be minimized by coating the enamel in turn with a very thin metal and/or ceramic layer, by plasma spraying or electrodeposition.

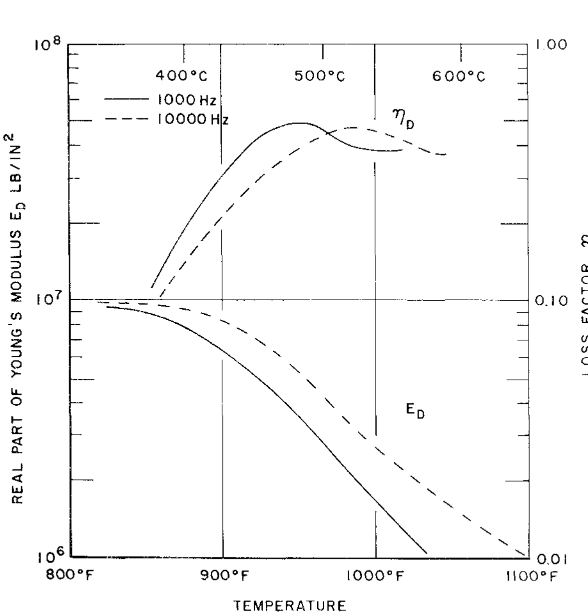

Fig. 11 Complex modulus properties of enamel.

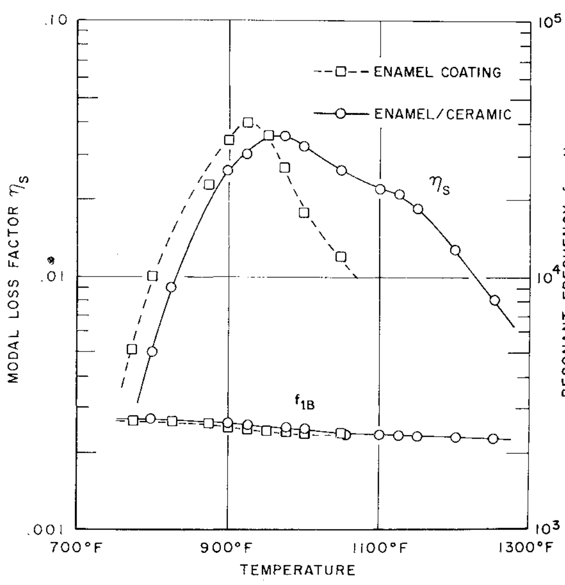

Fig. 13 Modal damping and resonant frequency for enamel coated vane in first bending mode.

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Modeling Engine Static Structures with Conical Shell Finite Elements

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The conical shell element with nonsymmetric loading and displacement capabilities has excellent possibilities for application to engine static structures. The major benefit would be a dramatic reduction in computer time as compared with a plate model, however, a severe limitation is the inability to combine this element with any other element types. This paper shows a technique that can be used to bypass this element compatibility problem. The inherent difficulty lies in the degree-of-freedom peculiarities of the conical shell element. The nonsymmetric motion of this element is accomplished by expanding each degree of freedom in a Fourier series with respect to the azimuthal coordinate. The technique presented in this paper sums this Fourier series for each degree of freedom and connects it to the appropriate degree of freedom for the nonshell portions of the structure by using functional constraints. The methods used to make the elements compatible and the computer time, displacement, and stress comparisons with standard plate and beam models will be shown.

Nomenclature

f = line load density
G = functional constraint matrix
K = stiffness matrix
m = maximum harmonic number
n = harmonic number
P = forces
q = functional constraint forces
R = coefficients of functional constraints
U = displacements
z = axial
θ = rotational displacements
ϕ = azimuth position

Subscripts

c = connected
cu = crossed terms
n = harmonic number
r = radial
u = unconnected
z = axial
ϕ = azimuthal
O = zeroth harmonic

Superscripts

$*$ = anti-symmetric displacements
sd = standard displacement system
fc = Fourier coefficient system

Matrix Notation

$\{ \}$ = column matrix
T = transpose
-1 = inverse

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Introduction

THE main static structural components of gas turbine engines are assemblies of shells, plates, and beam substructures. The problem of analyzing these structures has always been a difficult one to solve. Until the 1960's, classical solutions had to be combined with testing and experience to predict the structural characteristics. With the advent of finite element solutions in the sixties, the ability to analyze engine static structures was greatly enhanced. The analyst was no longer limited to the few classical solutions available and had greater generality in geometry, loading conditions, material properties, and boundary conditions. Unfortunately, with this enhancement came the added expense of a finite element solution. Once the computer programs are available, the major expenses of a solution are computer time and input preparation time.

Presently, engine static structures are being analyzed by constructing finite element models consisting of plates and beams. An analysis of a model of this type is relatively expensive because of the large number of degrees of freedom required for accuracy. This increases both preparation and computer time. This paper proposes a new method of modeling engine static structures that decreases both computer time and preparation time when compared with a plate and beam analysis of similar accuracy. The method uses conical shell finite elements with nonsymmetric loading and deflection capability for the shell portions of the structure. Normal plate and beam elements are used for the nonshell portions. The degree-of-freedom problems encountered when these elements are