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Letter to the Editor

Turbomachine blade damping

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1. Introduction

Blade fatigue failures are very common in turbomachinery. They are excited by flow path interference between the stator and rotor stages at nozzle passing frequency and its harmonics. When the frequency of any of these harmonics coincide with any one of the natural frequencies, resonance occurs. The high stresses at resonance lead to fatigue problems. The only way by which the resonant stresses can be controlled is by dissipation of the vibratory energy. Thus damping becomes an important aspect of the blade design.

For a turbine blade the means by which this vibratory energy could be dissipated is by material damping, Coulomb damping at the interfaces, gas dynamic damping and possibly impactive damping. All these damping mechanisms are complex and non-linear in nature. Generally the aerodynamic damping is very low, and when there is no impact damping, the material and Coulomb damping between interfacial slipping surfaces are the main sources of dissipation of energy. A designer usually depends on past experience and test results to estimate the available damping in limiting the resonant stresses. Attempts have been made to quantify the material damping using Lazan's damping law or by contact elements using 2D models, see Ref. [1]. However, an application of these procedures to determine the equivalent viscous damping values in turbine blades is not successfully implemented.

Based on experimental work, Rao et al. [2] quantified blade damping as a function of strain amplitude in each mode of vibration for different speeds. This non-linear model of damping has been successfully used in evaluating the resonant stresses, see Ref. [3]. However, it is very expensive to conduct such tests for each blade stage. Here a method is illustrated to estimate material damping, friction damping as well as combined material and friction damping using a finite element code, ANSYS.

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2. Material damping

Rowett [4] investigated the torsional damping properties of certain grades of steel shafting and suggested an early form of stress damping law,

$$E_H = J\sigma^n, \tag{1}$$

where E_H is energy loss per unit volume, J is a constant of proportionality, σ is the local stress and n is a damping exponent.

The total energy dissipated by material hysteresis is obtained by integrating the above stress relationship throughout the structure. Lazan [5] conducted comprehensive studies into the general nature of material damping and presented damping results data for almost 2000 materials and test conditions. Lazan's results show that the logarithmic decrement values increase with dynamic stress, i.e., with vibration amplitude, where material damping is the dominant mechanism.

To determine the resonant stress of a turbine blade passing through a critical speed, the most practical approach is to use a viable commercial code such as ANSYS. Damping plays an important role in determining this stress value accurately and experimentally, Rao et al. [2] demonstrated a method of quantifying the equivalent viscous damping in each mode at different rotational speeds. Rao et al. [6] used such a model to obtain the resonant stresses. A procedure to determine the equivalent viscous damping by computational means is demonstrated below.

2.1. Non-linear viscous damping due to material hysteresis

A simple rectangular blade (Sandvik O&T steel) $5\,\text{cm} \times 1\,\text{cm}$ with a T root is considered for the study. The blade length is 30 cm. The contact surfaces on either side are $5\,\text{cm} \times 1\,\text{cm}$. The disk is taken to be 14 cm long and 25 cm wide with 1 cm thickness for the purpose of modelling and the contact surfaces are 4 cm from the top of the disk. The blade root is taken as $15\,\text{cm} \times 5\,\text{cm}$. The blade and its root are modelled using 606 SOLID 45 elements [7]. The interfacial surfaces between the blade root and disk are modelled as 20 CONTACT 173 and 20 TARGET 170 contact elements. A friction coefficient 0.4 is assumed between the contact elements. The material density is $7800\,\text{kg/m}^3$. The elastic modulus is taken as 210 GPa. The material damping property is taken from Ref. [8]

$$D = J\left(\frac{\sigma}{\sigma_e}\right)^n, \tag{2}$$

where D = specific damping energy kN m/m^3/cycle, $J = 16.0$, $n = 2.3$, and σ_e = fatigue strength = 630 Mpa.

Using Lazan's law above and the material properties, the coefficient J and exponent n, the specific damping energy, total damping energy and the total strain energy are calculated by integrating over the entire volume. The strain amplitude at a reference point near the root of the blade is monitored throughout the analysis. The loss factor, ratio of total damping energy and total strain energy is obtained. It is then converted to equivalent viscous damping for the strain amplitude considered at the reference point. The mode shape is then modified to have different values of strain amplitude at the reference point and the specific damping energy, total damping energy and the total strain energy are correspondingly updated. Thus, the equivalent viscous

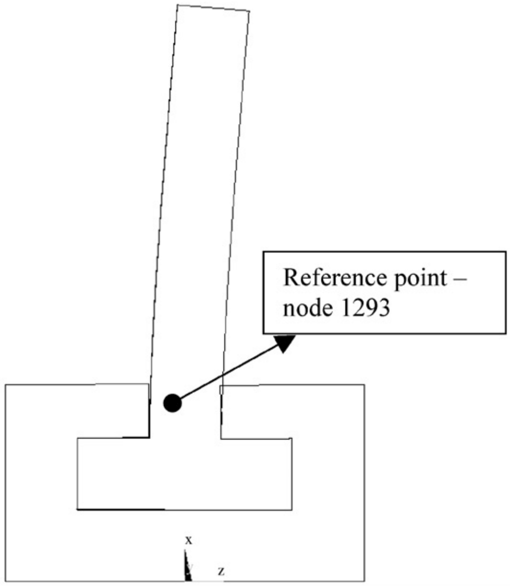

Fig. 1. The chordwise mode 383.67 Hz of the blade at 200 r.p.m.

damping is determined as a function of strain amplitude at the reference point for the given mode and speed. The same procedure can be applied to obtain the equivalent viscous damping values for other modes at the same speed. Here, only one mode is considered to illustrate the procedure. The steps in calculation are illustrated below for 200 r.p.m.

2.2. 200 r.p.m.

With the help of ANSYS, the natural frequencies and orthonormal mode shapes are first determined. For determining the material damping characteristics the contact elements are closed. The first two natural frequencies are 82.6 and 383.67 Hz. For a similar case, Rao et al. [9] considered the chordwise mode to determine friction damping, also see Ref. [1]. Here the second mode corresponding to the chordwise direction is considered to illustrate the procedure. This mode shape is shown in Fig. 1.

To estimate the damping in each mode the following procedure is adopted.

Total damping energy:

The total damping energy D_0 (Nm) is given by

$$D_0 = \int_0^v D \, \mathrm{d}v, \tag{3}$$

where v is the volume.

Loss factor η:

$$\eta = \frac{D_0}{2\pi W_0}, \tag{4}$$

where W_0 is the total strain energy (Nm).

Equivalent viscous damping [10]:

$$C = \frac{\eta K}{\omega}, \tag{5}$$

where C is the equivalent viscous damping (N s/m), ω is the natural frequency (rad/s), K is the modal stiffness (N/m).

To obtain the equivalent viscous damping at other strain amplitudes a program is written in Microsoft Excel taking advantage of the proportionality of stress and strain energy with the strain amplitude. For increased strain amplitudes, the orthonormal reference strain amplitudes, stress and strain energy are multiplied by a factor F to obtain the equivalent viscous damping C_e at various strain amplitudes. The procedure is best illustrated by the following numerical calculations: natural frequency in the first mode, $\omega_n = 2411.93$ rad/s; orthonormal strain amplitude at the reference point—node 1293, see Fig. 1, $\varepsilon = 0.269826$; total damping energy, $D_0 = 214235.4$ Nm; strain energy, $W_0 = 2\,908\,699$ Nm (ANSYS output); loss factor, $\eta = D_0/(2\pi W_0) = 0.011722$; equivalent viscous damping (N s/m), $C_e = \eta K/\omega_n = 28.2733$, damping ratio, $\xi = C_e/(2\sqrt{Km}) = 0.0058611$, where $m = 1$ and $K = (\omega_n)^2$.

If the strain amplitude is multiplied by a factor $F = 0.1$ then the corresponding strain amplitude is $\varepsilon' = \varepsilon F = 0.269826 \times 0.1 = 0.0269826$; the total damping energy is $D_0' = D_0 \times F^{2.3} = 214235.4 \times 0.1^{2.3} = 1073.72$; the strain energy is $W_0' = W_0 \times 0.1^2 = 2908699 \times 0.1^2 = 29086.99$; the loss factor is $\eta' = D_0'/(2\pi W_0') = 5.875 \times 10^{-3}$; the equivalent viscous damping is $C_e' = \eta' K/\omega_n F^2 = 0.14$, where $K = \omega_n^2$; and the damping ratio is $\xi = C_e F^2/(2\sqrt{Km}) = 0.0029375$.

The damping characteristic thus obtained is shown in Fig. 2. In a similar manner the damping characteristics can be obtained at other speeds. Since the chordwise mode is not significantly affected by centrifugal load, the damping characteristic remains almost the same as given in Fig. 2.

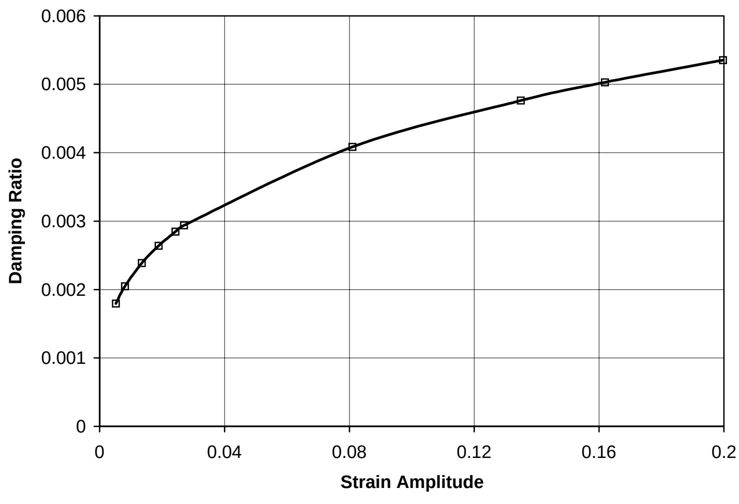

Fig. 2. Material damping given as equivalent viscous damping ratio as a function of reference strain amplitude for the chordwise mode.

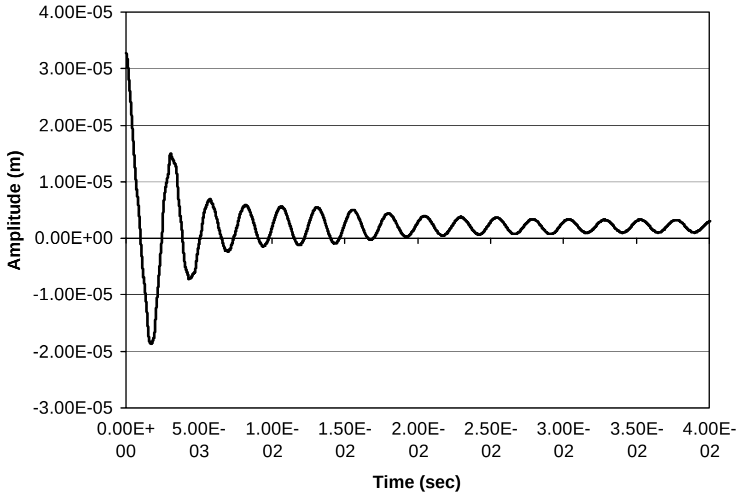

Fig. 3. Free vibration decay due to friction at 200 r.p.m.

3. Friction damping

To determine the equivalent viscous damping from friction at the interfaces between the disk and root, the omega load corresponding to the desired speed is given, e.g., 200 r.p.m. An impulse, say 60 N, is then applied at the blade tip in the required (chordwise) direction to excite the chordwise mode. The response thus obtained is used to determine the friction damping as a function of reference point displacement. A typical decay curve obtained is shown in Fig. 3. The damping due to friction is calculated from this response decay curve.

Initially, when the displacements are large, the damping is high and the decay is rapid. Subsequently the decay envelope is a straight line following Coulomb's law. Using this decay curve, the equivalent viscous damping in the system as a function of the strain amplitude at the reference point is generated. Fig. 4 gives the equivalent viscous damping as a function of the reference displacement at 200 r.p.m. At higher speeds, the interfacial slipping disappears as the contact elements get closed. The free vibration decay curve at 1000 r.p.m. is given in Fig. 5. It can be clearly seen that there is practically no damping in the system due to friction.

4. Combined material and friction damping

To determine the combined effect of material and interfacial friction, the average equivalent viscous damping from the material damping run is included in the response calculation due to friction as discussed above. The resulting net equivalent viscous damping values are then obtained to define the non-linear damping behavior of the bladed-disk system.

The results from the above investigation are presented in graphical form in Fig. 6.

As can be seen from Fig. 6, the influence of friction gradually disappears as the speed increases and only material damping is available at high speeds. The effect of material damping over friction at a low speed, 200 r.p.m. is shown in Fig. 7.

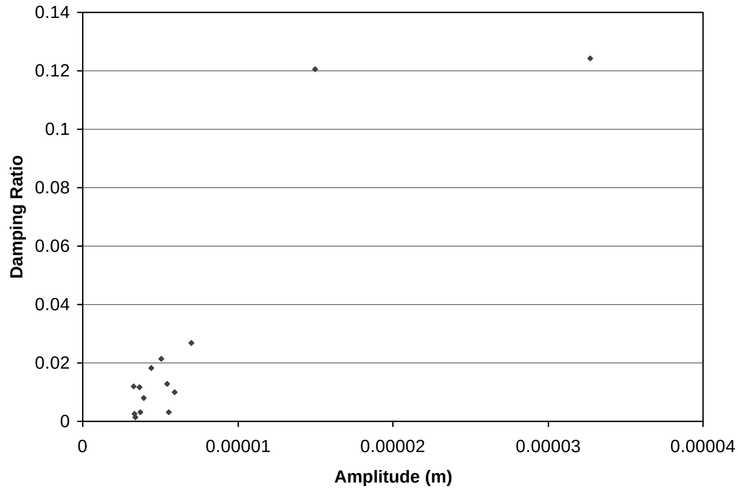

Fig. 4. Equivalent viscous damping as a function of amplitude at 200 r.p.m. due to interfacial friction.

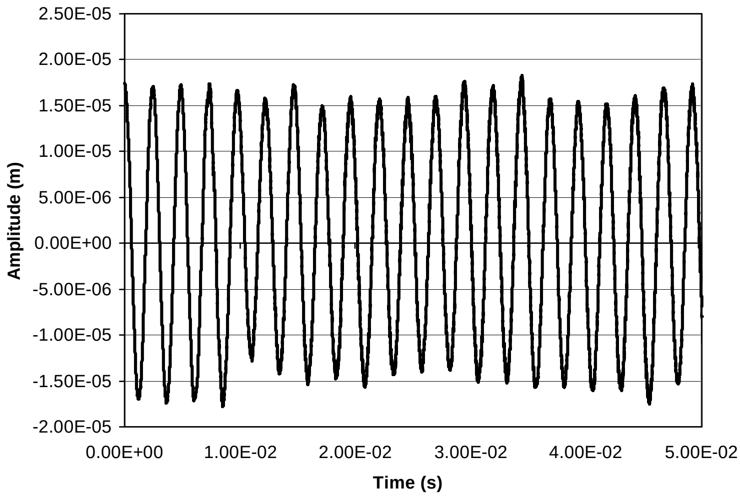

Fig. 5. Free vibration decay at 1000 r.p.m. due to interfacial friction.

5. Conclusions

A method of determining equivalent viscous damping ratio for different rotational speeds and modes as a function of displacement or strain at a reference point in a blade is presented. This method can be adopted using a suitable finite element code, e.g., ANSYS.

It is shown that the effect of friction disappears with increasing speed and that the viscous damping ratio increases with strain in the blade.

The non-linear model developed here can be easily applied to determine the resonant response very accurately and thus improve the current design capabilities in blade life estimation. The capability to assess damping in the blade from finite element codes avoids costly tests at the initial stages of design.

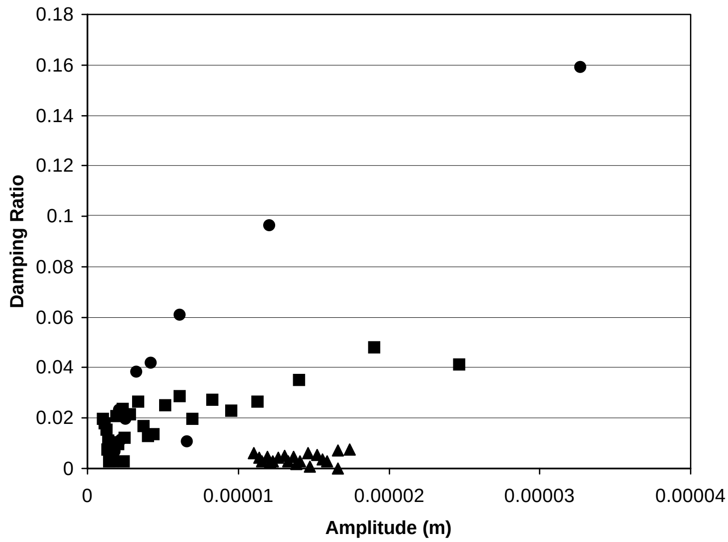

Fig. 6. Combined friction and material damping results for 200, 500 and 1000 r.p.m: ●, 200 r.p.m; ■, 500 r.p.m; ▲, 1000 r.p.m.

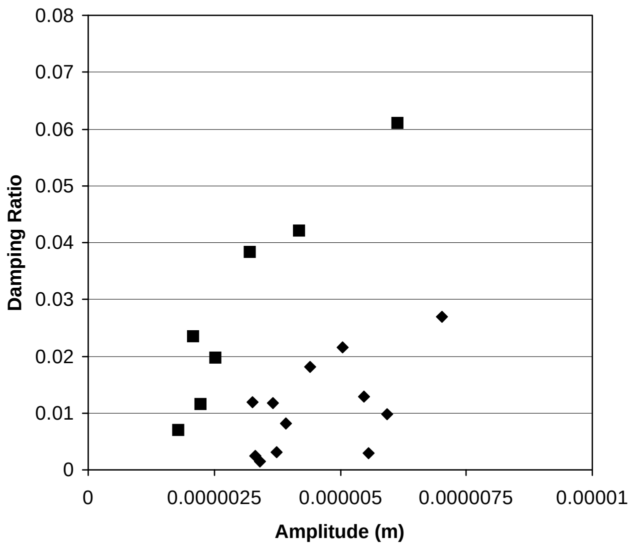

Fig. 7. The effect of material damping over friction damping at 200 r.p.m: ♦, friction damping; ■, friction and material damping.

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