Design of Viscoelastic Support for Vibration and Torque Reduction of Accelerating Rotors

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Abstract - *Rotors need to run up to and coast down from their operating speed. The rates of acceleration and deceleration required are often specified by the manufacturers of such rotors. However, it has been well understood that higher the acceleration rate, lower is the peak vibration due to passage through critical speed and it is expected that the torque required to pass through the critical speed should also fall. These are certainly desirable characteristics. However other considerations limit the acceleration rate (for eg., capacity of prime mover). An option left with the designer for reducing the transient flexural vibration and the vibratory torque requirement is to use viscoelastic support. The advantage of viscoelastic supports is that depending upon the material chosen, the storage modulus and loss coefficient of the support changes with frequency. The present work attempts to design appropriate viscoelastic supports by selecting from standard viscoelastic materials as well as suggesting futuristic (but realistic) materials for reducing the vibration and torque required for passage through critical speed using numerical simulation.*

Key Words: **Rotor, Acceleration, Vibration, Torque, Viscoelastic support design**

1.INTRODUCTION

Rotor dynamics aims to design rotors which operate with minimum vibration. This can be either achieved by reducing the excitation or by design of the rotor system. Introducing non-rotating damping is an established method followed during rotor design to reduce vibration. Unfortunately, anti-friction bearings can provide almost no external damping in contrast to hydrodynamic bearings. So during use of anti-friction bearings external damping should preferably be provided separately. Use of viscoelastic materials as pads has been proposed way back in 1981 by [1]. [2] has provided a theoretical discussion on vibration reduction of rotor using viscoelastic support where the frequency dependency of stiffness and damping has been considered. Further [3] has designed optimum viscoelastic support considering the dual criteria of minimizing rotor response and maximizing stability limit. They have analyzed for both rolling element bearing and cylindrical journal bearing and have considered the internal damping of the shaft and gyroscopic effects. The work proposed by [3] has been further extended in [4] to the case where viscoelastic polymeric materials are inserted on the stator of bearing in a sector form and length of such sector has also been determined. [5] uses viscoelastic polymeric supports as rings and the authors have performed both theoretical and experimental work. A purely experimental activity has been reported in [6] where plain and corrogated natural rubber sheet has been introduced beneath bearing housing. [7] and [8] have done ANSYS analysis and experimental investigation of viscoelastic supports using PVC. All this indicates that use of viscoelastic supports is an active area of study.

On the other hand all rotors run up to and coast down from their operating speed with certain rates of acceleration and deceleration. Analysis of Laval rotor subjected to acceleration by considering the rotor to be torsionally stiff as well as flexible has been given in [9] and the results of [9] in a simplified manner has been given by [10] in his text or the subject. The work of [9] has been extended in [11] to multi-degree of freedom, torsionally stiff anisotropic rotors where gyroscopic effects have been taken into account. From these study of rotors subjected to acceleration (during start-up), it has been observed that higher the acceleration rate, lower the maximum vibration during passage through critical speed. So, the torque required to pass through critical speed should also be lower as the vibratory torque falls. It is important to note that insufficient torque can stall an accelerating rotor as the vibratory torque needed to pass through the critical speed has not been provided. So it becomes imperative that higher the acceleration rate, lower is the vibration and vibratory torque requirement. But higher acceleration rates also increase the contribution of the torque required to accelerate the rotor considered as a rigid body. So it is easy to understand that an optimum acceleration would bring down the torque requirement. Similar observation holds for rotors mounted on viscoelastic support. But with proper selection of viscoelastic support, the vibration response will fall so abruptly that the required acceleration rate will steeply come down and so will the torque requirement. Design of appropriate viscoelastic supports for reducing the vibration and torque required for accelerating rotors during passage through critical speed using numerical simulation is the objective of the present work.

2.ANALYSIS

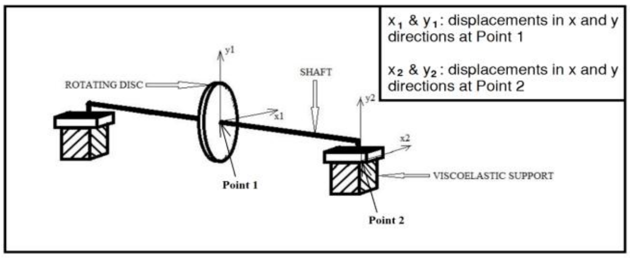

Fig -1: A schematic diagram of the rotor rig to be analyzed

The rotor system consists of a Jeffcott rotor with a massless shaft and two identical rigid bearings supported on identical viscoelastic support. The shaft is isotropic (circular).

The lateral and torsional equations of motion of the rotor are:-

$$mx_1''+K_s(x_1-x_2)+C_s(x_1'-x_2')-me(\phi''\sin\phi+\phi'^2\cos\phi)=0 \quad (1a)$$

$$my_1''+K_s(y_1-y_2)+C_s(y_1'-y_2')+me(\phi''\cos\phi-\phi'^2\sin\phi)=0 \quad (1b)$$

$$I_p\phi''+eK_s(y_1-y_2)\sin\phi-eK_s(x_1-x_2)\cos\phi+eC_s(y_1'-y_2')\sin\phi -eC_s(x_1'-x_2')\cos\phi=T \quad (2)$$

where, m = mass of the disc; e = eccentricity; ϕ = angular displacement ($^1/_2\alpha t^2$) ; ϕ'= angular velocity (αt); ϕ''= angular acceleration (α); t = time (in seconds); K_s = stiffness of the shaft; C_s = damping coefficient; T = torque required ; I_p = mass moment inertia of the rotor disc

The equilibrium equations of the viscoelastic support considered massless are :-

$$^1/_2K_s(x_1-x_2)+{}^1/_2C_s(x_1'-x_2')=Fv_x \quad (3a)$$

$$^1/_2K_s(y_1-y_2)+{}^1/_2C_s(y_1'-y_2')=Fv_y \quad (3b)$$

where, Fv_x = force in x-direction; Fv_y = force in y-direction

The material properties of the viscoelastic support are assumed to be represented by a four element model as shown in Fig -2.

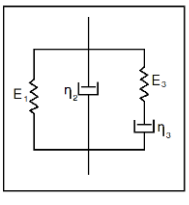

Fig -2: Four-element viscoelastic model

The constitutive equation of viscoelastic material in uniaxial loading using the four-element model is :-

$$\sigma=(\tilde{\beta}+\tilde{\gamma}D+\tilde{\varphi}D^2)/(1+\tilde{\alpha}D)*\varepsilon \quad (4a)$$

where, σ = stress; ε = strain; D = differential operator; E_1, E_3, η_2&η_3=springs and dampers of four-element model (Fig -2); $\tilde{\alpha}=\eta_3/E_3$; $\tilde{\beta}=E_1$; $\tilde{\gamma}=\eta_2+\eta_3+(E_1*\eta_3/E_3)$; $\tilde{\varphi}=\eta_2*\eta_3/E_3$;

$$F=(\bar{\bar{\beta}}+\bar{\bar{\gamma}}D+\bar{\bar{\varphi}}D^2)/(1+\bar{\bar{\alpha}}D)*x \quad (4b)$$

where, F = force; x = displacement; A = area of viscoelastic support ; L = length of viscoelastic support; $\bar{\bar{\alpha}}=\tilde{\alpha}$; $\bar{\bar{\beta}}={}^A/_L*\tilde{\beta}$; $\bar{\bar{\gamma}}={}^A/_L*\tilde{\gamma}$; $\bar{\bar{\varphi}}={}^A/_L*\tilde{\varphi}$

So, the force deflection relation for viscoelastic support is:-

$$Fv_x=(\bar{\bar{\beta}}+\bar{\bar{\gamma}}D+\bar{\bar{\varphi}}D^2)/(1+\bar{\bar{\alpha}}D)*x_2 \quad (5a)$$

$$Fv_y=(\bar{\bar{\beta}}+\bar{\bar{\gamma}}D+\bar{\bar{\varphi}}D^2)/(1+\bar{\bar{\alpha}}D)*y_2 \quad (5b)$$

Under harmonic load, the expressions for storage modulus and loss coefficient of the viscoelastic material are :-

$$E=(\tilde{\beta}-\omega^2\tilde{\varphi}+\tilde{\alpha}\tilde{\gamma}\omega^2)/(1+\tilde{\alpha}^2\omega^2) \quad (6a)$$

where, E = storage modulus; ω = frequency (in rad.sec^{-1})

$$\eta=(\tilde{\alpha}\tilde{\varphi}\omega^3+\omega\tilde{\gamma}-\tilde{\alpha}\tilde{\beta}\omega)/(\tilde{\beta}-\omega^2\tilde{\varphi}+\tilde{\alpha}\tilde{\gamma}\omega^2) \quad (6b)$$

where, η = loss coefficient

Using Eqs. (1a), (3a) and (5a) and a constant acceleration rate(α) , a 4th order differential equation in terms of x_1 to represent the rotor response is obtained :-

$$m\{^1/_2(C_s\bar{\bar{\alpha}})+\bar{\bar{\varphi}}\}x_1'''' + [m\{^1/_2(K_s\bar{\bar{\alpha}})+{}^1/_2(C_s)+\bar{\bar{\gamma}}\}+C_s\bar{\bar{\varphi}}]x_1'''+ [m\{^1/_2(K_s)+\bar{\bar{\beta}}\}+C_s\bar{\bar{\gamma}}+K_s\bar{\bar{\varphi}}]x_1''+(C_s\bar{\bar{\beta}}+K_s\bar{\bar{\gamma}})x_1'+K_s\bar{\bar{\beta}}x_1 - me[\{^1/_2(K_s)+\bar{\bar{\beta}}\}\{\alpha\sin(^1/_2\alpha t^2)+(\alpha t)^2\cos(^1/_2\alpha t^2)\}+\{^1/_2(K_s\bar{\bar{\alpha}})+{}^1/_2(C_s)+\bar{\bar{\gamma}}\}\{3\alpha^2t\cos(^1/_2\alpha t^2)-(\alpha t)^3\sin(^1/_2\alpha t^2)\}+\{^1/_2(C_s\bar{\bar{\alpha}})+\bar{\bar{\varphi}}\}\{3\alpha^2\cos(^1/_2\alpha t^2)-6\alpha^3t^2\sin(^1/_2\alpha t^2)-(\alpha t)^4\cos(^1/_2\alpha t^2)\}]=0 \quad (7)$$

where, x_1'''' = 4th order derivative of x_1 w.r.t. time;
x_1''' = 3rd order derivative of x_1 w.r.t. time ;
x_1'' = 2nd order derivative of x_1 w.r.t. time;
x_1' = 1st order derivative of x_1 w.r.t. time

Similarly using Eqs. (1b), (3b) and (5b) a similar equation in terms of y_1 may be obtained and this is not shown here for brevity.

After obtaining the rotor response the evaluation of torque from Eq. (2) requires finding of x_2, y_2 and their first order derivatives. x_2 and y_2 can be expressed in terms of x_1 and y_1 using Eqs. (1), (3) and (5). However it has been found easier to derive equations of the form Eq. (7) for both x_2 and y_2, so that may be solved separately.

3.VALIDATION

Eq. (7) and alike has been cast in state-space form ie, $\dot{z}$ = f(z) where {z} is the state defined by :- $\{z\} = \{x_1 \;\; x_1' \;\; x_1'' \;\; x_1'''\}'$

where {}' means transpose of matrix {}

Ode45 of MATLAB which is a non-stiff solver based on fourth/fifth order Runge Kutta method has been used to solve the equation. Thereafter torque has been evaluated along the lines described earlier.

The process of validating the method and the code is to convert the viscoelastic support into a rigid support and comparing the results obtained with those in [10]. While doing so one should avoid using unduly large values to avoid possible numerical problems.

The specification of the rotor system, shown in Fig. 1 are :-

Length of the shaft = 1 m; Diameter of the shaft =$18.9*10^{-3}$m; Mass of the disc = 10 kg; *E* of shaft material (steel) = $2*10^{11}$ N. m^{-2}; Height of viscoelastic support = 5 cm; Cross-sectional area of viscoelastic support = 4.5*8 cm^2

A rotor rig similar to the one specified is present in the Dynamics lab of the department.

The values of the parameters $\tilde{\alpha}$, $\tilde{\beta}$, $\tilde{\gamma}$ and $\tilde{\varphi}$ for rubber was obtained by fitting the $E-\omega$ curve as given in [12]. The value of $\tilde{\beta}$ was then increased to 20 times (to increase stiffness) keeping all other parameters same. The values thus obtained were :

$\tilde{\alpha} = 0.72 * 10^{-4}$ sec; $\tilde{\beta} = 2.6 * 10^8$ N. m^{-2}; $\tilde{\gamma} = 2*10^4$ N. sec. m^{-2}; $\tilde{\varphi} = 0.55$ N. sec^2. m^{-2}

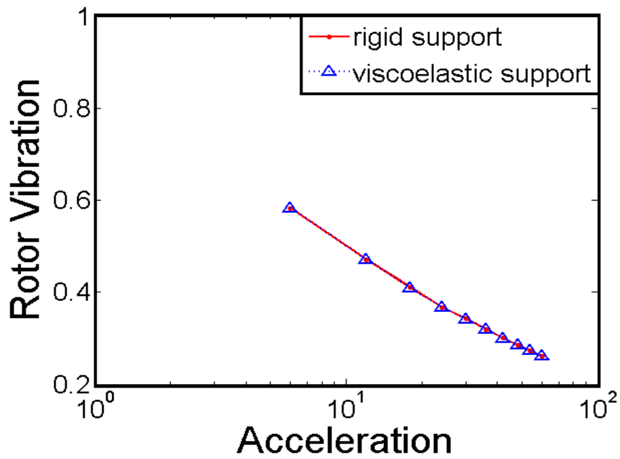

Chart -1: Rotor vibration at different acceleration rates obtained by using a rigid support and a viscoelastic support which has been made rigid by changing the values of the parameters of the four element model of the support

The rotor vibration is plotted in Chart-1 against α(acceleration). The rotor vibration reported is the ratio of the maximum vibration for a certain α (with or without viscoelastic support) to the vibration at critical speed for α=0 (without using viscoelastic support). A small value of non-rotating damping (ζ = 0.01) [10] has been taken so that the vibration at critical speed, though high, does not become infinity. The blue dashed line (with Δ) plotted uses the values of the parameters of the four element model mentioned earlier whereas the red line (with o) uses the formulation for rigid support as given in [10]. The lines exactly coincide, thus validating the process.

4.SIMULATION

For simulation the same rotor, as that of validation has been employed. For employing viscoelastic support, six different viscoelastic materials have been chosen. Details regarding these materials follow.

i. Material-1: The material is GE SMRD (at 150^0F) [12]. The parameters of the four element model has been so chosen that the E – ω and η - ω curves of the material match with the curves given in [12].
ii. Material-2: The material is similar to Ployisobutylene (PIB) (at 100^0F) [12]. Here the parameters have been so extracted that the E – ω curve matches with the curve of the stated material. However although the η - ω curve obtained has similar nature to the curve given, yet its range is a bit wider than that given in [12].
iii. Material-3: The material is similar to Soundcoat DYAD 601 (at 150^0F) [12]. The details regarding the nature of $E-\omega$ and η - ω curves is similar to that mentioned for Material-2.
iv. Material-4: The material resembles 3M ISD-113 (at 100^0F) and 3M 468 (at 150^0F) [12]. This means that the parameters chosen produce E – ω and η - ω curves which fairly match with that of the stated materials.

It needs to be stated that using a four element model, the authors could only fit the $E-\omega$ and η - ω curves of the stated material with a single set of values of $\tilde{\alpha}$, $\tilde{\beta}$, $\tilde{\gamma}$ and $\tilde{\varphi}$ (for each material) over the whole range of ω (1 Hz – 10^4 Hz) to the extent stated above.

v. Material-5 and Material-6 may be considered as futuristic material which may be developed using appropriate polymers. However it needs to be stated that the range of *E* values of either of the materials is realistic in the sense that there exists materials in [12] whose *E* values resemble the stated E values.

Further for all materials considered, the η values never exceed 1 and in most of the range remain much below 1. Materials-1,2, 3&4 have similar η curve. For these curves the maximum value of η, which is 1, reaches at a frequency not below 1000Hz and in all of them η falls thereafter. Most industrial rotors operate at a speed between 3000rpm (50Hz) and 6000rpm (100Hz). It may be noted that at 3000rpm η values of Material-1 to Material-4 are around 0.05 and at 6000 rpm the values are around 0.1. So in the operating range none of these materials may be termed as 'highly damped'. However, the η variation of Material-5 and Material-6 is different, but not unrealistic. The values of the parameters of the four-element models of these viscoelastic supports have been given in Table-1 and the variation of storage modulus and loss coefficient with frequency has been represented in Chart-2 and Chart-3.

Table -1: Parameters of the four element models of all the six materials along with the range of E values in the frequency range of 1Hz-1000Hz.

Name	Values of $\tilde{\alpha}, \tilde{\beta}, \tilde{\gamma}$ and $\tilde{\varphi}$	Range of E from 1Hz to 1000Hz
Material-1	$\tilde{\alpha} = 3.67*10^{-5}$ sec $\tilde{\beta} = 3.79*10^{7}$ N. m^{-2} $\tilde{\gamma} = 7.96*10^{3}$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	38.5 N. mm^{-2} to 49 N. mm^{-2}
Material-2	$\tilde{\alpha} = 2.91*10^{-5}$ sec $\tilde{\beta} = 1.5*10^{6}$ N. m^{-2} $\tilde{\gamma} = 2.52*10^{2}$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	1.54 N. mm^{-2} to 1.75 N. mm^{-2}
Material-3	$\tilde{\alpha} = 3.67*10^{-5}$ sec $\tilde{\beta} = 1.69*10^{6}$ N. m^{-2} $\tilde{\gamma} = 3.56*10^{2}$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	1.75 N. mm^{-2} to 2.1 N. mm^{-2}
Material-4	$\tilde{\alpha} = 5.81*10^{-5}$sec $\tilde{\beta} = 3.79*10^{5}$ N. m^{-2} $\tilde{\gamma} = 7.96*10$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	0.385 N. mm^{-2} to 0.497 N. mm^{-2}
Material-5	$\tilde{\alpha} = 5.81*10^{-3}$sec $\tilde{\beta} = 6*10^{5}$ N. m^{-2} $\tilde{\gamma} = 2*10^{4}$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	0.63 N. mm^{-2} to 3.5 N. mm^{-2}
Material-6	$\tilde{\alpha} = 5.81*10^{-3}$ sec $\tilde{\beta} = 1.897*10^{5}$ N. m^{-2} $\tilde{\gamma} = 6.32*10^{3}$ N. sec. m^{-2} $\tilde{\varphi} = 0.55*10^{-5}$ N. sec^{2}. m^{-2}	0.203 N. mm^{-2} to 1.19 N. mm^{-2}

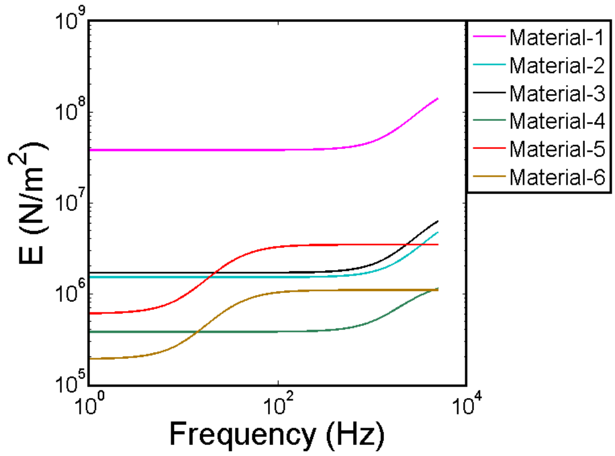

Chart -2: Variation of storage modulus (E) with frequency for the 6 materials to be used as viscoelastic support

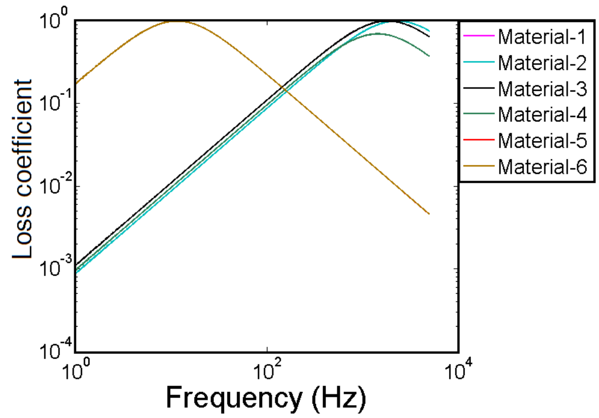

Chart -3: Variation of Loss coefficient (η) with frequency for the 6 materials to be used as viscoelastic support

Simulation has been done for all the 6 viscoelastic materials used as supports and the time response of the rotor has been obtained for different acceleration rates with and without the use of viscoelastic supports. It has been observed that the use of any of the 6 viscoelastic supports causes a reduction as well as shift of the peak value of rotor amplitude ratio (R_D, where $R_D = {|x|}/{e}$) and the torque required. Chart-4 shows the effect of use of a viscoelastic support(Material-4) on R_D and torque required for α = 6 rad. sec^{-2}, which is the least value of α considered and hence is the most crucial point of observation. As is evident from the graphs, the use of the viscoelastic support causes the maximum value to shift from t=14 seconds to t=9 seconds and the peak value of rotor amplitude ratio and torque required is also reduced considerably.

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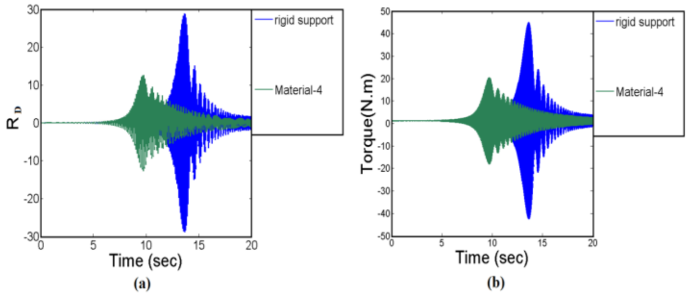

Chart -4: Time response of (a) rotor amplitude ratio (R_D) (b) torque required due to use rigid support and viscoelastic support -- Material-4 at α = 6 rad. sec^{-2}

After simulation, the rotor vibration and torque has been plotted against α in Chart-5 and Chart-6. The rotor vibration reported is the ratio of the maximum vibration for a certain α (with or without viscoelastic support) to the vibration at critical speed for α=0 (without using viscoelastic support). In either case a small value of non-rotating damping (ζ=0.01) has been taken [10] so that the vibration at critical speed though high does not become infinity. This non-dimensional variable is a suitable parameter for the measurement of the severity of vibration. This representation is similar to the one used during validation.

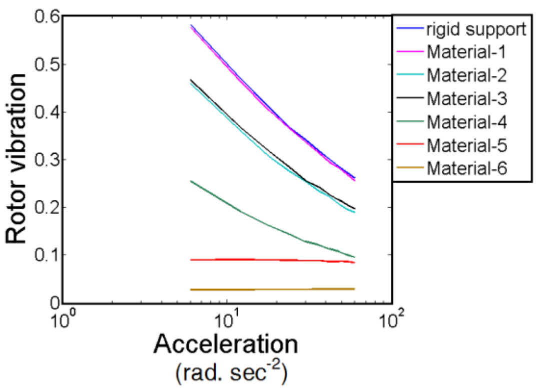

Chart -5: Effect on Rotor Vibration due to use of different viscoelastic supports at different acceleration rates

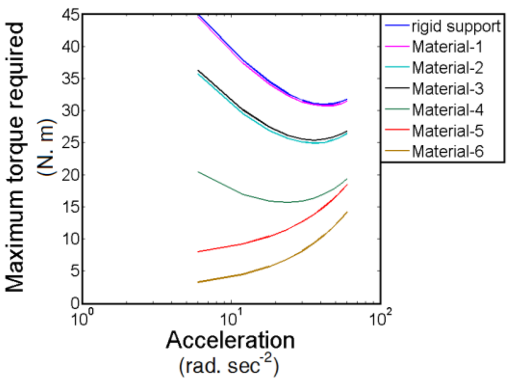

Chart -6: Effect on Maximum Torque Required due to use of different viscoelastic supports at different acceleration rates

Chart-5 and Chart-6 plots the vibration and torque curves for all the materials in a single plot. So these curves show the effect of choice of materials on rotor performance.

In Chart-5 with increase in α, the fall of vibration ratio is obvious [10]. The effect of viscoelastic support decreases at high α as the vibration ratios are already very small. For low values of α, the effect of viscoelastic support is considerable. At α = 6 rad. sec^{-2}(which is the cardinal point to compare the performances), for Material-1 the vibration ratio is lowered by about 1% as compared to that for rigid support, for Material-2 it is lowered by 21%, for Material-3 it is lowered by 20%, for Material-4 56%, 85% for Material-5 and 95% for Material-6. The improvement is so considerable for Materials-4,5 & 6 that the vibration ratio at α=6 rad. sec^{-2} (least) is smaller than that of rigid support at α =60 rad. sec^{-2} (maximim).

In Chart-6, as briefly qualitatively explained in the introduction, for rigid and viscoelastic supports (Materials-1,2,3&4), the torque first decreases and then increases so that a minimum point exists. However for Materials-5&6, it only increases. The reason is, for these cases, the vibratory torque is so low (because of low vibration) that its fall is hardly noticeable. The torque reduction for Material-1, Material-2, Material-3, Material-4, Material-5 and Material-6 is about 1%, 20%, 19%, 55%, 82% and 93% respectively.

It is now relevant to point out a few things:

a. All viscoelastic materials have considerable damping but one should not think that damping is the cause of fall of vibration due to use of the viscoelastic materials. If this would have been true then Material-1 to Material-4, which have similar damping characteristics, should not have shown such wide change of performance.

b. A quick conclusion that follows is lower the stiffness of the support, greater is the fall of vibration. This trend is clearly observed in increase of % fall of vibration in the sequence Material-1,Material-2 & 3, Material-4. Material-2 and Material-3 with similar E shows similar fall of vibration. But conversely it is seen that Material-5 has greater % fall of vibration than Material-4, inspite of it having higher E. Lastly, Material-6 has the best performance, but it posesses an E value greater or lesser than that of Material-4 depending upon the frequency range.

c. From the above two points it becomes imperative that neither increase of damping nor decrease of stiffness is the necessary cause for fall of vibration due to use of viscoelastic support. Rather one should conclude that it is the nature of the $E - \omega$ curve that permits an appropriately chosen viscoelastic material to reduce rotor vibration during run-up. Similar conclusions have been reached by [2] for rotor running at constant speed.

4.DESIGN DATA

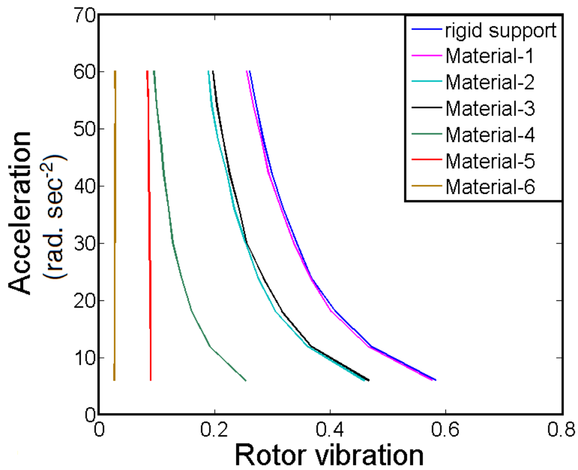

Chart -7: Effect on acceleration required by the system due to use of different viscoelastic supports for different permissible values of Rotor vibration

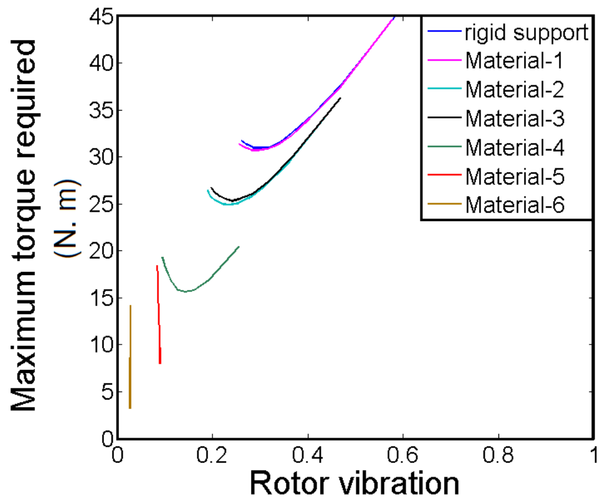

Chart -8: Effect on Maximum torque required by the system due to use of different viscoelastic supports for different permissible values of Rotor vibration

Chart-7 and Chart-8 recasts the information of Chart-5 and Chart-6 so that vibration ratio is plotted along x-axis and α & torque for various conditions is plotted along y-axis. The utility of these two curves to a designer is that, once he decides the vibration ratio, he knows what α and torque is required for the rotor bearing support system for proper selection of the prime mover. Alternatively, for a definite rotor bearing, prime mover system, an appropriately designed support may be provided. Curves for Material-5 and Material-6 are vertical implying that with change of α or torque, vibration ratio remains constant and obviously for such curves one needs to choose the least α.

5. CONCLUSIONS

The work studies dynamics of accelerating rotors on viscoelastic support. It discusses how for super-critical rotors, during start-up, the high vibration during passage through critical speed can be considerably reduced without using high acceleration and torque on the rotor but by employing properly designed viscoelastic supports.

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