Controlling rotordynamic response without squeeze-film dampers (SFDs)

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Abstract. SFDs are widely used in rotating machinery to provide damping in order to control rotordynamic response. Although popular, under certain conditions SFDs pose problems such as causing non-synchronous vibration arising from unbalance forces interacting with fluid-film forces affected by cavitation. Furthermore, in the interests of moving towards oil-free rotating machines, the need arises to find alternative means of rotordynamic response control.

In choosing a new vibration control technology, it is first necessary to consider certain general, configuration-independent criteria. For example, does the actuation method provide a limited stroke (e.g. piezoelectric or giant magnetostrictive) or is the stroke a "motorised" solution (e.g. an ultrasonic motor directly driving the actuator or a pump acting to vary the fill level of closed deformable volumes with incompressible fluid) Is the work per stroke per unit mass of the actuator material sufficient to provide the maximum stroke and force required for the control? What is the bandwidth of the actuator? In the case of electromagnetic actuation, what is the coupling factor? Can the elements of the actuator withstand the high temperatures of the operating environment? Is the solution an active or passive one? What are the fatigue properties of the materials used in the actuator? These are some of the questions that need to be considered when evaluating a new control method.

Once the significant properties have been identified, it is necessary to consider each of these in the context of the intended application. If one considers the actuation type, in the limited stroke case it will be required for the actuation to take place at synchronous frequency and the work per stroke per unit mass will determine the quantity of material required. For some applications – particularly aero-engines - one seeks to minimise overall mass and therefore materials with high values of work per stroke per unit mass are attractive. By contrast, in the case of motorised solutions the frequency of operation of the actuator may be orders of magnitude greater than synchronous speed. Since the main source of rotordynamic vibration arises from rotor unbalance – which causes vibration at synchronous speed – the control is also required (at least) at synchronous speed. Therefore, the actuation method should have a bandwidth equal at least to this speed. The coupling factor for piezoelectric and magnetostrictive actuators will determine the amount of energy that should be supplied for a given requirement of work, and therefore high values of this property are desirable. Whether the vibration control method is active or passive will also influence the overall weight and complexity of the machine.

Damping is not the only method by which vibration may be controlled. For example, if it is possible to alter the natural frequencies of the rotor-stator system as the system is approaching a resonance, then the build up of substantial vibration can be avoided as the system effectively never sits on a resonance.

In the present work, various actuation technologies such as piezoelectric, magnetostrictive, electromagnetic, shape memory alloys and ultrasonic motors have been considered. Some of these technologies are considered in more detail as they show more potential of being implemented in a relatively uncomplicated configuration, in the intended application.

Keywords: SFD, damping, response, vibration, control, active control
1. Introduction
The SFD has been a popular means of rotordynamic vibration control since the 1970s, and is widely employed in aircraft gas turbine engines today. The damper manifests as an oil film between the damper journal (a sleeve fitted to the outer race of the bearing) and the stator. Rotation of the journal is prevented by mechanical means, although it is free to undergo whirling motion. The hydrodynamic damping forces generated in the squeeze-film during whirling of the journal serve to oppose the motion of the bearing by dissipating energy from the system.

A paper by Zeidan et. al. [1] presents a detailed account of SFDs, where the authors discuss the historical development of SFDs, theoretical models and experimental results and design procedures. The above paper provides a total of 50 references relating to various aspects of SFDs.

SFDs are well known for their non-linear behaviour. This gives rise to multiple unbalance response solutions for certain speeds. It has been found that the minimum and maximum of these solutions are stable, and thus when operating in this regime the system is in bistable operation. Rabinowitz and Hahn present their work on a theoretical study of the steady-state performance and stability of SFD supported flexible rotors in two separate papers [2, 3]. In [2], the authors conclude that bistable operation can be avoided if the design parameters are chosen appropriately and in [3], the parameters that give rise to instability are discussed. In both the above papers, it is noted that pressurization of the SFD is a prerequisite to stability. The above authors presented a further paper which presents experimental results which validate their previous theoretical work [4]. Zhu et. al. analyse the multiple-solution response of a flexible rotor using the synchronous circular centred-orbit motion solution, numerical integration method and slow acceleration method in [5]. The authors conclude that there are three basic forms for the multiple-solution response: resonant, isolated bifurcation and swallowtail bifurcation multiple solutions. References to other analysis methods are given in [5]. An extremely fast method of obtaining the unbalance response of flexible rotors on SFDs is presented in [6], where non-linear forces from the SFD are included in the analysis. The authors report twenty-five orders of magnitude time savings as compared to numerical integration and iterative methods. The stability and bifurcation of unbalance response of flexible rotors on SFDs is analysed in [7], where the cavitation phenomenon of SFDs is included in the model. It is found that unpressurised SFDs promote non-synchronous vibrations.

The multiple response phenomenon under certain conditions means that it is not possible to accurately predict the unbalance response under these conditions, as more than one stable solution can exist. The non-synchronous vibrations that arise under certain conditions due to SFDs give rise to alternating stresses in the rotor that can lead to progressive fatigue, and eventually failure. Both of the above are notable disadvantages of using SFDs to control rotordynamic vibration.

A further disadvantage is that the damper force generated in the SFD will always have a fixed phase relative to the velocity – i.e. it is in phase with the velocity. The authors recognise this as a constraint placed on the control force. The present paper investigates the possibility of achieving better vibration reduction by employing active control forces, where the phase of the forces may not be fixed under all conditions.

2. FE Modelling of a rotor in bearings employing SFDs
In this section, a rotor-stator system is defined, and its behaviour under unbalance excitation explored. The damping of this simple machine is based on SFDs, whose behaviour is modelled here as a linear viscous damper in parallel with the squirrel-cage stiffness (although it is recognised that a more accurate model of an SFD entails a non-linear model) connecting the bearing outer housing to the stator. The bearing outer housing is assumed to be rigidly fixed to the rotor, and it therefore undergoes the same motion as the rotor.

The rotor model is based on the usual linear, second order equations shown below:
Where $\mathbf{f}(t)$ is the applied forcing vector (in this case unbalance), $\mathbf{q}(t)$ is the response vector of the system, $\mathbf{M}$, $\mathbf{C}$, $\mathbf{G}$ and $\mathbf{K}$ are the global mass, damping, gyroscopic and stiffness matrices of the system respectively.

The unbalance response of the system in the frequency domain may be calculated as:

$$
\mathbf{q}(\Omega) = \left( \mathbf{K} + i\Omega(\mathbf{C} + \Omega\mathbf{G}) - \Omega^2\mathbf{M} \right)^{-1} \mathbf{f}(\Omega)
$$

The term within brackets is the dynamic stiffness matrix, and will subsequently be referred to as $\mathbf{E}$.

2.1 Rotor model

A picture of the FE model that will be used in this paper is shown below:

![Simple overhung rotor](image)

**Figure 1 - FE model of a simple overhung rotor in bearings**

The model is based on beam elements, where each element has two nodes and each node has 4 Degrees of Freedom (DOFs) – translations in the $x$ and $y$ directions and rotations about these directions. In addition to the stiffness properties of the shaft elements, inertial and gyroscopic contributions from the shaft elements and the discs are incorporated into the model (the discs themselves are assumed rigid). Furthermore, the squirrel cage stiffness and damping from the SFD are applied at the bearings.

The parameters that define the model are stated below:

| Material Properties of the rotor shaft | Dimensions of the rotor components (mm) |
|--------------------------------------|----------------------------------------|
| Young’s Modulus – 211 GPa            | Shaft (solid) – $\varphi 30$ throughout |
| Poisson’s Ratio – 0.3                 | Disc 1 (node 4) – $\varphi 300$ OD, $\varphi 30$ ID, 60 thickness |
| Density (shaft and discs) – 7810 kgm$^{-3}$ | Disc 2 (node 13) – $\varphi 350$ OD, $\varphi 30$ ID, 30 thickness |

**Bearing Properties**

| Left bearing stiffness – $4 \times 10^7$ N/m | Left bearing damping coefficient – $7.202 \times 10^7$ Ns/m |
|---------------------------------------------|-----------------------------------------------------------|
| Right bearing stiffness – $12 \times 10^7$ N/m | Right bearing damping coefficient – $7.202 \times 10^7$ Ns/m |
For simplicity, it is assumed that the bearing stiffness and damping properties are isotropic. A distribution of unbalance forcing along the length of the rotor is now defined. This will be kept fixed in all of the subsequent modelling. The magnitude of the unbalance forcing is chosen such that the machine will experience deflections at the bearings that are large enough such that the bearing outer housing starts to make contact with the stator. This allows one to model the situation when the hydrodynamic force generated in the SFD is insufficient to drive the rotor back to its axis. It is assumed in this model that the SFD thickness is 0.1 mm. Therefore, when the bearing outer housing makes contact with the stator, the bearing stiffness is iteratively increased to reflect this. The following two figures show the unbalance response at the bearings and the increase in effective lateral stiffness when damper journal-stator contact is made:

![Uncontrolled response at the bearings](image1)

**Figure 2 - Rotor response at bearings (SFD case)**

![Variation of effective lateral stiffness with speed (uncontrolled)](image2)

**Figure 3 - Increase in effective lateral stiffness to reflect contact of journal with stator**

A “cost function” can now be defined, which is a measure of the overall rotor vibration. In our case, this may be defined as the sum of squares of translations at all the nodes (in the FE sense) of the rotor. This provides a convenient means of comparing the performance of the SFD with that of an active solution. The following figure shows the cost function as a function of speed for the SFD case:
3. FE Modelling of a rotor in actively controlled bearings

In this simulation, the same model as above is used, with the only change that the SFD is now removed (i.e. the damping value above is now set to zero) and active control forces are applied at the bearings such that the cost function for each individual speed is brought down to its minimum possible value. The optimum active control forces can be computed through the following procedure based on the assumption of a linear model:

From equation (2), we have that

$$q = E^{-1} [f + f_c]$$  \hspace{1cm} (3)

Where the additional term $f_c$ contains the control forces. Multiplying both sides by the diagonal weighting matrix $W$, we have

$$Wq = WE^{-1} [f + f_c]$$  \hspace{1cm} (4)

The weighting matrix can be used to assign different levels of importance to rotor deflections at individual nodes. In the present analysis, all translations have been assumed equally important. Contributions due to rotations have been ignored by placing zeros in the corresponding entries of the weighting matrix. Now, multiplying equation (4) by its Hermitian transpose, an expression for the weighted sum of squares (i.e. the cost function) is obtained:

$$cost = [f + f_c]^H E^{-H} W^H W E^{-1} [f + f_c]$$  \hspace{1cm} (5)

Since control forces occur only at the bearings, a constraint may be applied to $f_c$ to reflect this. Viz.,

$$f_c = Sp$$  \hspace{1cm} (6)

Substituting (6) into (5), and differentiating with respect to $p$ and subsequently multiplying by $S$ gives the optimum control force vector as:

$$f_c = -S (S^H E^{-H} W^H W E^{-1} S)^{-1} (S^H E^{-H} W^H W E^{-1} f)$$  \hspace{1cm} (7)

Once the optimal control forces have been found, the new (controlled) response is computed. This is shown in the following figure:
It can be seen from the above figure that there still is bearing-stator contact. The cost function in this case is shown below.

At this stage, it is appropriate to explain the basis on which the damper values in the SFD case have been chosen. The cost functions for the entire speed range were calculated in the case of many values of damping coefficient, and in each case the maximum cost function from the speed range was taken. A graph of maximum cost function vs. damping coefficient is shown below:
It is found after further simulation that the damping coefficient value giving rise to a minimum value of the maximum cost function is approximately 7202 Ns/m, which is the value that has been used in this work.

It is important to note that the magnitude of control force that the actuator can provide may fall short of the value computed for the optimal control force. This effect has been incorporated into the present model, and it is found that if no such limitation is placed, no bearing-stator contact is observed in the controlled response. The following figure shows the magnitude limitation of the control force, which is chosen to be 50N in the present model.

![Figure 8 - limitation of control force magnitude](image)

4. **Comparing the performance of SFDs with that of an active solution**

A comparison between the vibration control performance of the SFD and the actively controlled case can now be made based on the cost function that each case gives rise to. The following figure shows the ratio between the cost function in the controlled case to that in the uncontrolled case:

![Figure 9 - fraction between controlled and uncontrolled cost functions](image)

It is evident from the above figure that the vibration level with active control forces implemented is much lower than present when using SFDs. One may also compare the magnitude of the damper forces
in the SFD case with the actuator forces in the controlled case. This comparison is shown in the following figure:

![Figure 10 - comparing damper force (SFD case) with actuator force (controlled case)](image)

From the upper half of the above figure, it can be seen that for 0 – 1000 rpm, the damper and active control forces are similar, whereas after this speed the damper force is substantially higher. The lower half of the figure shows that in 0 – 1500 rpm, the control force magnitude is slightly higher and above this speed the damper force is, once again, substantially higher. Therefore one may draw the conclusion that overall, the magnitude of active control forces required for optimum control is lower than that of the damping forces.

The main advantage of using active control forces as compared with SFDs is the freedom to choose the phase of the force with respect to the displacement or velocity. In SFDs, the damping force is always in phase with the velocity. The freedom to change the phase of the control force for various speeds enables one to optimise it for each value of speed in the range, whereas in the case of the SFD, optimisation will only be possible at a single speed.

5. Some candidate technologies and design concepts for active control of vibration

Electro-mechanical devices such as piezoelectrics and magnetostrictives are widely used in actuation, and could provide a viable source of rotordynamic vibration control. It is generally acknowledged in the literature that direct implementation of piezoelectric stacks or magnetostrictives in rotordynamic vibration control is usually infeasible, due to the inherently low stroke produced by these devices. Therefore, some form of mechanical amplification is essential, should actuators of these types be considered for rotordynamic applications.

Vibration control may also be achieved by electromagnetic means. This may be implemented either in the form of an Active Magnetic Bearing (AMB) or an Active Magnetic Damper (AMD). A design for the latter is presented in [8], where the authors implement their design in a scaled test rig producing results that demonstrate a reduction in vibration as a result of applying the AMD. The authors then go on to assess their design on a computational model of a real aero-engine, and have obtained promising results.
6. Conclusions

It has been shown through the simulation results that have been presented in this paper that better rotor-dynamic response control can be achieved by replacing the SFD with an actively controlled solution. This is partly because with the latter, not only is it possible to optimise the control forces for each individual running speed, but it is also possible to vary the phase of the control force with respect to either the displacement or velocity, whereas the damping force generated by an SFD will always be in phase with the velocity.

It is acknowledged that the actuator providing the active control forces may not always be able to generate the required magnitude of control force, simply due to physical/material limitations. This effect has been shown in the relevant simulation results above. However, by choosing the specifications of the actuator such that under most conditions, the demands for optimum control can be met, satisfactory overall control performance can be achieved.

The AMD has been mentioned above as a control method that has been tested and delivered promising results. The authors of the present paper believe that it may be possible to find a cheaper, simpler alternative using piezoelectric technology to achieve the same objectives.

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