Journal of Tribology

Discussion

An Adaptive Squeeze-Film Bearing[1]

David W. Lewis.[2] This paper shows a nice balance between the theoretical and the experimental, each complementing the other. The findings suggest a very broad scope of additional work that needs be done. Specifically, the parametric study of all sorts of components that are found in rotating machinery. For example, control of one pad attitude angle in a tilt-pad bearing, the purposeful changing of lubricating oil temperature (hence, viscosity) in bearings, the bang-bang control of preload in fluid film bearings, variable support stiffness of bearings possibly for emergency use only (as in shut down of gas turbines), to name a few.

The authors suggest using the device in a location that may be inaccessible in many machines. The cross-coupling that exists in certain equipment may allow applications that, even though not optimally located, will still be fruitful.

Some of our own work confirms the unexpected that active feedback control devices can reduce transmitted forces. This reduction of both maximum rotor amplitude response and the transmitted forces should hasten the application of these concepts to large and/or critical rotating machinery.

The costs of implementing bang-bang systems versus some of our continuous multichannel analog or digital feedback control systems are coming closer together. But the savings promised by these systems, by reduction of necessary clearances for getting through the critical speeds, through increased efficiencies will ensure their growth in applications in the future.

D. L. Taylor.[3] Adaptive active vibration control for rotor bearing systems has been studied in increasing detail recently. It is interesting to see an investigation of the more easily realized passive adaptive control systems.

Although optimum adaptive active control always seems to lead to continuous variation of the control, optimum passive control frequently leads to the "bang-bang" variation discussed by the authors. Their findings are consistent with those of studies into adaptive passive vehicle suspensions. The advantages of implementation which have been found in that field also apply to rotor systems. These advantages are leading to changes in suspension design and should spur further experimental and development work in rotor control.

In the interest of other researchers who may pursue these topics, a few questions can be posed.

Generation of simulated results for multiple degree of freedom models is sensitive to the unbalance distribution chosen. Perhaps the authors could comment on selection of the test unbalance. Was this actually chosen arbitrarily (by a random number generator) and then checked to see that it excited the modes in the operating range, or was some reverse procedure used? A similar question can be posed concerning the location of the midspan damper. Was this chosen by some formal procedure or simply picked and checked afterwards to avoid nodal locations?

Another question concerns the experimental determination of the damping coefficient, Figs. 10, 11. At any particular supply pressure, the effective damping coefficient is needed over a limited frequency range (Fig. 5(*a*)). Should this restricted range be used in the generation of the Schroeder phased harmonic signal?

Would the authors please elaborate on the interpretation of the variation in estimated damping immediately after $t = t1$? Does this variation reflect an actual variation in the damping coefficient, or is it an artifact of the identification algorithm? Considering the physics of the situation, the new pressure distribution should stabilize almost instantly. This is consistent with the discusser's common sense but does not agree with the authors' interpretation. Perhaps this area would be appropriate for future investigations. Given a sinusoidal input, a transient response plot during the switching period could be very illuminating.

Finally, the results indicate a potential advantage to this type of design. This discusser hopes soon to see some papers (both experimental and analytical) on various physical questions of passive adaptive design. Other variations are possible (opening and/or closing extra lubricant feeds; novel ways to change effective land length; step variations in lubricant viscosity; and others). Also, the damper could react to changes in unbalance, load, or other operating parameters in addition to speed. The authors seem to have opened a very rich lode for future study.

R. Stanway.[4]

With many modern rotating machines operating in supercritical regions it is vitally important that effective schemes are available for the control of lateral vibrations [1]. In this paper the authors have described the application of a controlled passive element—an adaptive squeeze-film damper—which is capable of adapting to changing conditions

[1] By C. R. Burrows, M. N. Sahinkaya, and O. S. Turkay, published in the January issue of the JOURNAL OF TRIBOLOGY, Vol. 106, No. 1, pp. 145–151.

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in a nonlinear ("bang-bang") fashion. Because of this inherent nonlinearity it is not intuitively obvious what performance gains can be achieved or indeed how the design of a suitable controller should be approached. The authors have produced results from a digital simulation study to show the application of the adaptive damper to vibration control of a light, flexible transmission shaft and to indicate how damping levels can be chosen.

Perhaps the most interesting result to emerge from the investigation is the demonstration that when damping is applied at the support bearings then the optimised passive damper is superior in overall performance to the adaptive version. If similar results could be demonstrated for other rotor-bearing configurations then the role of adaptive dampers would be put into clearer focus. I would certainly urge the investigators to consider other types of rotor configuration—in particular a flexible rotor carrying heavy discs, such as that described in reference [2].

Once a suitable vibration control strategy has been formulated, it remains to develop hardware for its implementation. Towards this end, the authors have demonstrated experimentally that it is feasible to switch the damping in the squeeze-film through "on-off" control of the inlet pressure of the lubricant. Attention has been restricted to a one-coefficient representation of the squeeze-film damping and such a representation is generally considered adequate for an isolator operating in the linear regime with a concentric damper-ring. However, in view of the wide range of lubricant supply pressures which are employed by the authors (and any unexpected behaviour which might result) it would be useful if experimental confirmation of the absence of significant cross-coupling effects could be produced. In this respect, it would be interesting to compare the tracking properties of the authors' frequency-domain identification algorithm with an alternative time-domain formulation [3]. The time-domain formulation appears particularly well suited for monitoring up to four squeeze-film damping terms from noisy measurements of the displacement responses to synchronous excitation.

Additional References

1 Stanway, R., and Burrows, C. R., "Active Vibration Control of a Flexible Rotor on Flexibly-Mounted Journal Bearings," ASME *Journal of Dynamic Systems, Measurement and Control*, Vol. 103, 1981, pp. 383–388.

2 Cunningham, R. E., "Steady-State Unbalance Response of a Three-Disk Flexible Rotor on Flexible, Damped Supports," ASME *Journal of Mechanical Design*, Vol. 100, 1978, pp. 563–573.

3 Stanway, R., "Identification of Linearised Squeeze-Film Dynamics Using Synchronous Excitation," *Proc. I. Mech. E.*, Vol. 197 (C), 1983, pp. 199–204.

Authors Closure

The authors are gratified by the response to the paper. Several important issues have been raised which point to the need for further work.

Professor Taylor's question about the choice of out-of-balance distribution is extremely important. The approach adopted by the authors was to allow the phase at each station to take one of eight values spaced at 45 degree intervals as determined by a random number generator. The magnitude of the mass unbalance was also determined using a random number generator with a step size of 5×10^{-6} kg m. As a check on the ability of this arrangement to excite all the system modes, the eigenvalues were computed and the response generated to ensure that a resonance occurred at each of the corresponding values of damped natural frequency.

The choice of station 3 for the damper was based upon the optimisation results reported by Dostal et al. [7].

There may be some misunderstanding concerning the relationship between figs. 5, 10, and 11 which prompted Professor Taylor's second question. Figures 10 and 11 are characteristics of the bearing as an isolated element, that is, they are independent of the rotor-bearing configuration. Figure 5 however relates to the characteristics of a particular rotor-bearing system and defines the optimum damping values for that system. Hence it is not clear what benefits would be gained by changing the excitation frequency bandwidth as a function of the supply pressure.

Professor Taylor has raised an interesting and important issue about the dynamics of the oil-film within the clearance following a step-change in supply pressure. This certainly calls for further experimental and theoretical investigation. What the authors' results demonstrate is that in the period t_1 to $t_1 + T$, there is a poor goodness of fit and this shows that the linear constant coefficient model is inadequate in this period. The high goodness of fit values are recovered by $t_1 + 2T$, hence the data collected in the time interval T immediately following the step-change in pressure is in some sense different to that collected before t_1 or after $t_1 + T$. Beyond this observation it is dangerous to speculate, and Professor Taylor is right in drawing attention to this problem. Although the question is important to a proper understanding of the phenomena exhibited within the bearing oil-film, its effect on vibration control is minimal because changes in the damping coefficient are required at the approximate fixed points where the response is insensitive to the damping value.

The authors share the contributor's enthusiasm for this field of study and are encouraged by his interest and complimentary remarks.

Professor Lewis has raised several interesting points and possibilities which demonstrate what a rich field of study it is. It offers the opportunity for control engineers and rotor-bearing specialists to collaborate in attacking a recurring source of problems in a wide class of machines.

Dr. Stanway correctly underlines the authors' conclusion that the results presented in the paper are limited to the particular configuration studied. The main contribution of the work is to present a methodology which does have general application. It is hoped to use the approach to examine other configurations as suggested by the contributor.

Although the results focus on the use of an uncoupled model for the bearing, this is justified by the experimental results published in reference [15].

Dr. Stanway's suggestion that alternative estimators should be compared is well taken. This should help resolve some of the issues raised by Professor Taylor.

Additional Reference

15 Sahinkaya, M. N., Turkay, O. S., and Burrows, C. R., "Minimisation of the Variance in Oil-film Damping Coefficient Estimates," Presented at the 1983 ASME Winter Annual Meeting. Paper 83-WA/DSC-6. (To be published in ASME *Journal of Dynamic Systems, Measurement, and Control*).