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Squeeze Film Damping for Aircraft Gas Turbines

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ABSTRACT

Modern aircraft gas turbine engines depend heavily on squeeze film damper supports at the bearings for abatement of vibrations caused by a number of probable excitation sources. This design ultimately results in light-weight construction together with higher efficiency and reliability of engines. Many investigations have been reported during past two decades concerning the functioning of the squeeze film damper, which is simple in construction yet complex in behaviour with its non-linearity and multiplicity of variables. These are reviewed in this article to throw light on the considerations involved in the design of rotor-bearing-casing systems incorporating squeeze film dampers.

NOMENCLATURE

B – bearing parameter
c – radial clearance
C_c – critical damping coefficient
C_d – rotor damping coefficient
e_u – rotor unbalance eccentricity
F_u – unbalance force ($m_D e_u \omega^2$)
g – gravitational acceleration
L – squeeze film damper length
m_B – mass lumped at bearing

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m_D – mass lumped at disk
R – squeeze film damper radius
U – unbalance parameter
W_D – disk weight ($m_D g$)
$\bar{W}$ – gravity parameter
a – mass ratio
ξ – damping ratio
μ – lubricant viscosity
ω – rotor angular speed
ω_c – rotor critical speed
Ω – speed ratio

1. INTRODUCTION

A large number of rotor dynamics studies arise from the problems encountered in the design and operation of gas turbine engines used for aircraft propulsion. The task of the designer of a rotor-bearing layout is well summed up by the logic diagram of Fig. 1 due to Sternlitch[1]; the additional considerations when designing an aviation gas turbine would be of blade-loss impact and aircraft manoeuvres. The design and

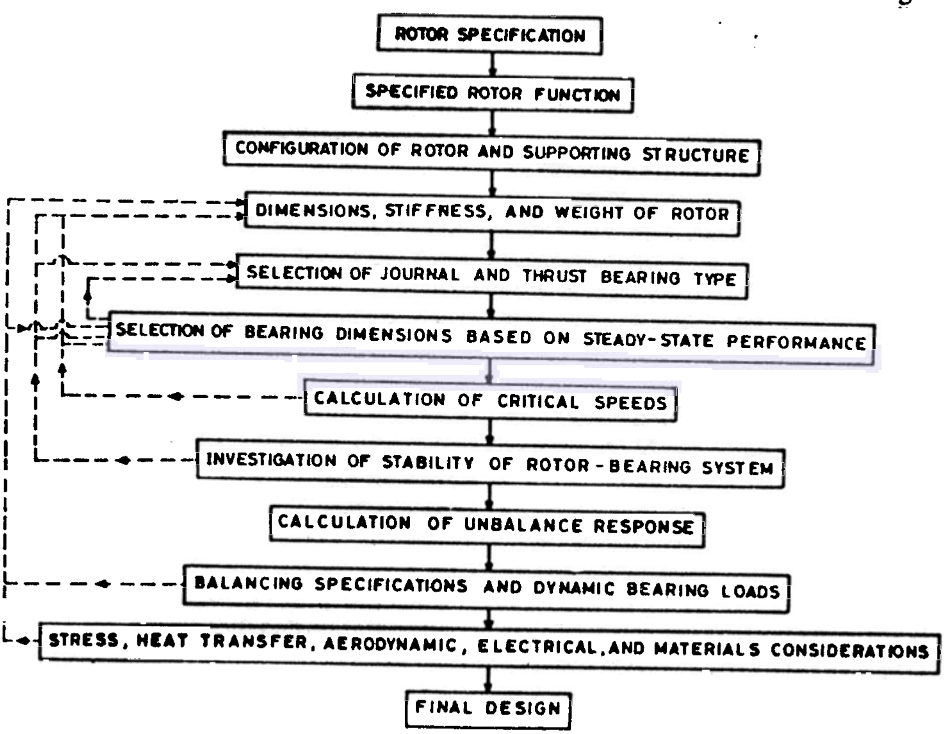

Figure 1. Logic diagram of design procedure[1]

development efforts for such a system have to accomplish the objective of smooth passage through the various 'critical states' involved in the operating speed range with minimum weight as the supreme dictum. Squeeze film damping of rotor supports for attenuation of rotor vibrations and for minimum bearing transmissibility, therefore, has become a regular feature of aircraft gas turbine rotor layouts ensuring reduced rotor-stator rubs, longer bearing life and consequently an efficient, reliable and light-weight engine.

The rotor-bearing design technology depends considerably on the analytical tools emerging from the research efforts in the field. The object of this article is to make a critical appraisal of the state-of-the-art in rotor dynamics research as applicable to aviation gas turbine engines, with special reference to squeeze film damping.

A preliminary appreciation regarding soft mounted, damped engine rotor systems may be gained through the articles by Magge[2] and Brown[3]. A detailed classification of rotor dynamics problems and various methods in vogue will be found in the report by Sane and Shende[4]. The areas of research may be grouped as follows :

i) Critical speed and mode shape analysis;
ii) Steady-state response to rotor unbalance;
iii) Balancing specifications;
iv) Time transient response to shock loads;
v) Stability and sustained transient behaviour against self-excitation (for example, due to rotor hysteresis[5] and aerodynamic cross coupling[6]); and
vi) Squeeze film dampers and performance of damped systems.

A squeeze film damper is essentially a simple device comprising of an oil film interposed between the non-rotating but nutating outer race of a rolling element bearing and its housing as shown in Fig. 2. With its two implementations – with and without a centralising spring (Figs. 3 (a) and (b)) – it has been incorporated in many complex multi-spool rotor-bearing-casing structures giving successful results in face of almost all the problems of vibration. The complexities involved in the design analysis

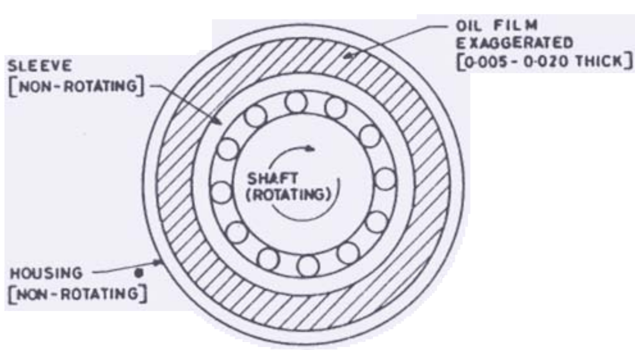

Figure 2. Elements of a squeeze film damper[3].

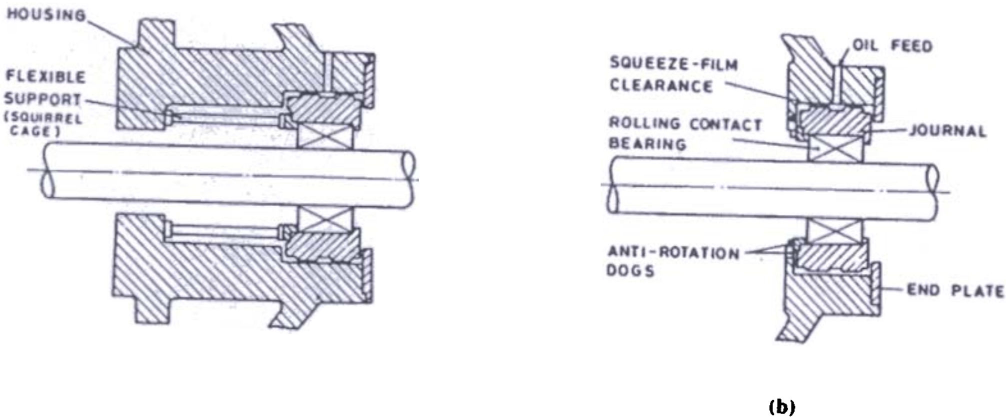

Figure 3. (a) Squeeze film damper with centralising spring and (b) Squeeze film damper without centralising spring.

of a projected rotor-bearing system, make a close co-ordination between the design/development and analytical efforts inevitable. It is, therefore, interesting to take stock of the developments in the investigations of squeeze film-damped systems.

2. ROTOR-BEARING LAYOUTS OF AIRCRAFT GAS TURBINE ENGINES

To highlight the variations in structural configurations in which squeeze film dampers appear, a few typical rotor arrangements together with related analytical findings are discussed below with a comparative outlook.

i) One of the early investigations in multi-spool configurations is by Hibner[7] on a model shown in Fig. 4(a) idealised as seen in Fig. 4(b). Squeeze film damping is applied to both the spools with respect to the casing near the turbine disks in conjunction with centralising springs and the steady-state response shows considerable reduction in peak amplitudes.

ii) Hong Bao Lin[8] has also analysed a twin-spool rotor model with the turbine end bearing of outer spool squeeze film-damped. The analysis shows amplitude attenuation as well as reduced bearing loads due to damping, with centralising spring (Fig. 5).

iii) One more rather complex system[9] where squeeze film damping has been effectively used on turbine end bearings (nos. 6 and 7) is shown in Fig. 6.

iv) Qihan Li[10] *et al.* have analysed a model of twin-spool rotor with intershaft squeeze film damping (Fig. 7 (a) and (b)). The advantage of this arrangement is

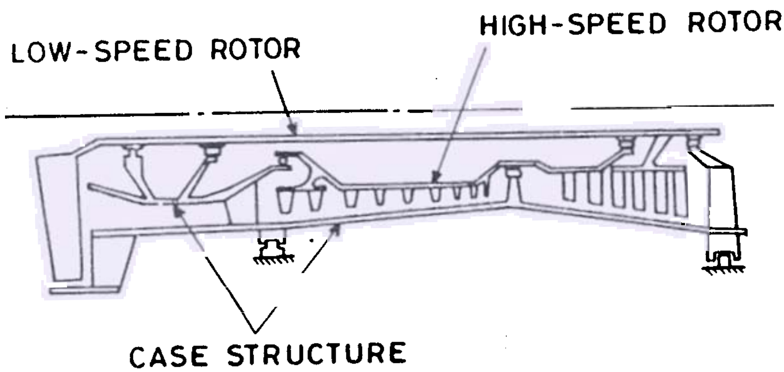

Figure 4(a). Twin-spool rotor layout[7].

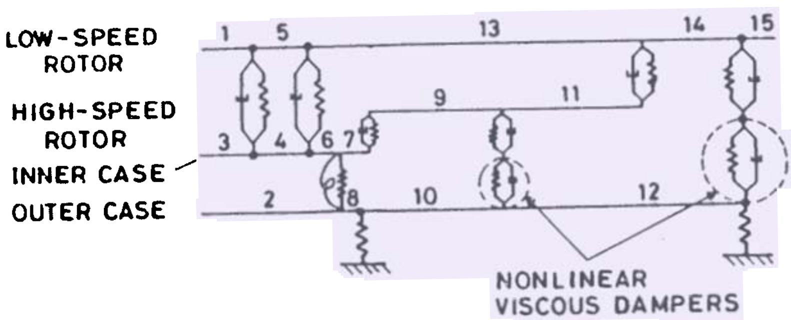

Figure 4(b). Idealised twin-spool[7].

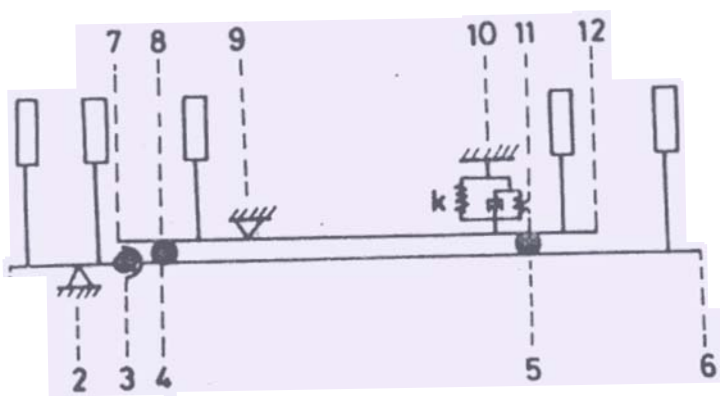

Figure 5. Twin-spool rotor layout[8].

claimed to be elimination of static support structure in the aerodynamic flow path resulting in improved engine efficiency, performance and reliability. The damper seems to be effective in reducing peak amplitudes together with the loading of the intershaft as well as the adjacent rotor-bearing supports.

The complexity of construction as also the possibility of cross-excited rotor instability, however, warrant caution while adopting this arrangement. The rotor instability in this case is further investigated by Qihan Li[11] *et al.* by a transient analysis under blade loss condition which indicates a limit on the permissible unbalance.

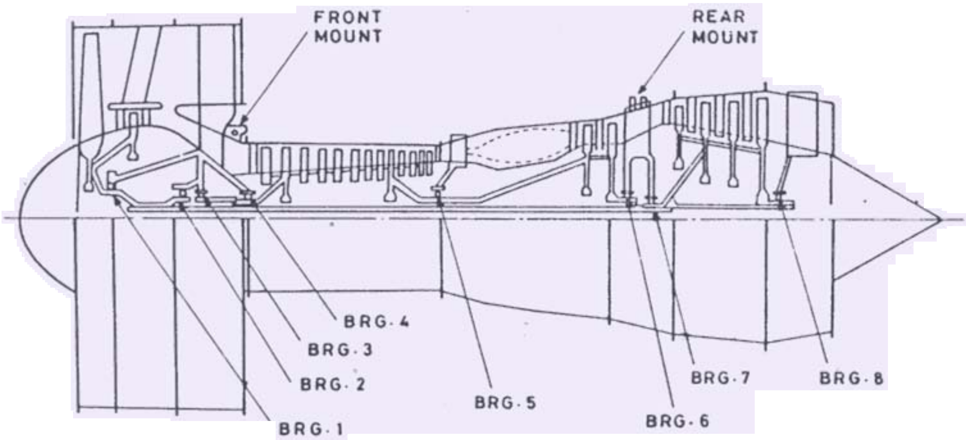

Figure 6. Twin-spool rotor-casing[5]

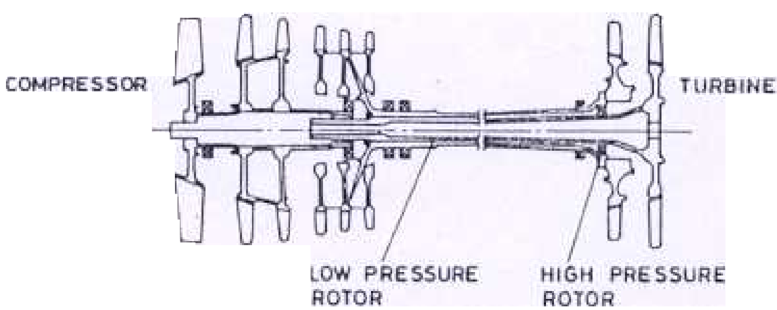

Figure 7(a). Rotor layout with inter-spool damping[10].

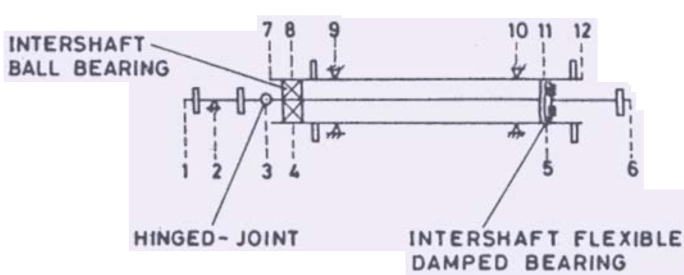

Figure 7(b). Idealised rotor with inter-spool damping[10].

The above investigations are all commonly carried out by the transfer matrix method of analysis due to its convenience. Further, the damper is commonly idealised as one with cavitated or ruptured (π) film (discussed later). The efficacy of squeeze film damping in these analyses is generally proved by steady-state response although a transient evaluation is always desirable. Not much work in respect of transient analysis of complex multi-spool configurations is reported probably owing to the heavy costs of computations. In terms of efficient method and generality of reasoning the articles by Nelson[12] *et al.* and Li and Gunter[13] are of interest.

A commonly found design feature among the various configurations is damper location adjacent to the turbine disk. As a well known fact, the turbine portion of the rotor is more prone to degradation of balance during use, due to factors like distortion, shaft warpage owing to thermal gradients and erosion of blades. Self-excitation-induced rotor instability is also experienced predominantly by the turbine[5,6]. Structurally the turbine shaft, being more flexible relative to the compressor rotors of drum construction, is likely to be responsible for occurrence of rotor critical speeds in the operating speed range. This is vindicated by the mode shapes observed in the critical speed analyses of the above models involving large deflections at the turbine end.

One more aspect of gas turbine rotor construction is that owing to the high temperatures and possible alignment deviations, a functional clearance as an allowance between the bearing and its housing may be inevitable particularly in case of multi-bearing system. In fact, before the advent of squeeze film dampers, operation of rotor bearings in a dry clearance (causing severe vibrations) was one of the problems under analytical treatment[14]. The filling of this clearance space with oil constitutes a squeeze film damper which also simultaneously enables mitigation of the many rotor vabration problems.

The raison d'etre for squeeze film damping in a gas turbine rotor layout is, therefore, clear and location of squeeze film damper in a configuration may be decided through a judicious study of the vibration mode shapes vis-a-vis operating speed range and the critical speeds involved, in addition to the above practical considerations.

3. SQUEEZE FILM DAMPER BEHAVIOUR

The squeeze film damper poses several problems as regards prediction of its behaviour, inspite of its simple construction. Beginning with the introduction of the concept by Cooper[15], many investigators have contributed substantially to the subject and it would be appropriate to summarise the developments in this field.

Though essentially non-linear in character, when used with a centralising spring to give a steady-state circular orbit, a squeeze film damper may be modelled in a linear fashion with constant value of damping coefficient until the journal excursion is limited within 40 per cent of the radial clearance. This assumption may be used in preliminary analysis of critical speeds and unbalance response while working out the layout and optimum damping, etc. Non-linear transient computations for steady-state amplitudes and forces cannot be avoided in case of dampers without centralising spring. The quasi-linear approach developed by Holmes and Dogan[16,17] appears to be promising as a partial substitute for this costly computational technique.

The governing equation for hydrodynamics of a squeeze film damper is the Reynold's equation in polar-cylindrical co-ordinates[18] in common with a dynamically loaded journal bearing. Booker[18] also summarises the three widely used approximate solutions of the Reynold's equation, i.e., Sommerfeld long bearing with no axial pressure flow; Ocvirk short bearing with no circumferential flow, and the Warner's finite bearing – all three solutions with π or 2π films which represent a ruptured and full film respectively, the difference being due to the oil supply pressure vis-a-vis the subatmospheric cavitation pressure. The assumptions underlying Reynold's equation together with these idealisations of boundary conditions have enabled the investigators of squeeze film damping to make the problem tractable. Particularly the short bearing solution with π film has been used extensively in simulation of the squeeze film, both with and without centralising spring, although the experimental findings indicate a somewhat higher degree of damping than predicted by theory. However, with growing need for better understanding, refinements in modelling of squeeze film damping are being introduced on the following lines.

i) Simandiri and Hahn[19] have introduced the model of a partially pressurised film so that conditions like $\pi <$ film spread $< 2\pi$ may also be handled.

ii) Marmol and Vance[20] have developed a finite difference model with effects of boundary conditions due to piston ring seals (Fig. 8(a)), radial and axial 'O' ring seals (Fig. 8(b)), no end seals, inlets and inlet feedback, etc. included. Their important findings are that with piston ring seals the leakage is so appreciable that the damper behaviour is like a short bearing. On the other hand, the 'O' ring seal with its flow restriction acts like a 'long bearing', which means heavy damping as also experimentally discovered by Feder[21] *et al.*

iii) Adams[22] *et al.* have also demonstrated a sophisticated finite difference model. More recently Samaha and Sankar[23] have demonstrated the use of finite element models of squeeze film accounting for all possible discontinuities.

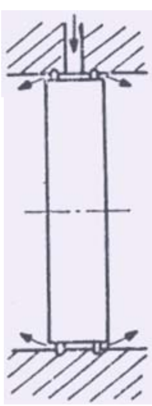

Figure 8(a). Piston ring seals.

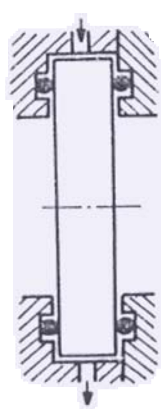

Figure 8(b). Axial 'O' ring seals.

iv) Some secondary effects like journal eccentricity[24], oil inertia have been also recently tackled. Cookson and Dainton[25] have worked experimentally on optimisation of end plate clearance (Fig. 3(b)) of a damper.

While the need for accurate modelling of the squeeze film damper for exact simulation of non-linear rotor dynamics cannot be denied, it may be emphasised that for general evaluation the more economically computable approximate solutions of Reynold's equation–provided the boundary conditions are reasonably observed – are adequate. As an example, the Warner finite bearing solution used in the non-linear analysis is found to simulate the piston ring sealed damper satisfactorily as observed experimentally[4,26]. Barrett[27] *et al.* have also indicated suitability of finite bearing approach.

A discussion on the practical aspects of damper design/parameter selection is desirable. If the damper location and an optimum value of damping coefficient (on linear basis) are worked out for a layout, then the question is how one might achieve the value in practice. As mentioned above, the end sealing arrangement would decide the approximation concerning the bearing type (long, short or finite). The end plate seal (Fig. 3(b)) with adjustment of clearance probably offers right type of controllable design. Between the two extremes of the side plate clearance (from equal to damper radial clearance to zero) the boundary conditions vary from short to long[25].

As mentioned by Adams[22] *et al.* the designs with axial 'O' ring seals are common in military applications and the piston ring seal is popular among commercial aircraft engines. The 'O' ring seal tends towards rigidity owing to the long bearing behaviour resulting in a high degree of damping. The absence of leakage flow in this design may hamper the damper heat dissipation. The piston ring seals, besides permitting copious leakage are known to exert an additional frictional force upon damper pressurisation. Out of the remaining parameters which decide the amount of damping, the geometrical ones – diameter, length and clearance – are easily controllable, although the clearance may be affected somewhat by thermal expansions. The supply pressure (usually[22] 3.5 to 4 ata) decides the angular spread of cavitation and therefore the damping coefficient value. As will be seen later, the 2π film is best suited for a centralised damper and an uncentralised damper cannot be feasible without cavitation. Oil viscosity is likely to vary during engine operation if not controlled artificially. In fact Rabinowitz and Hahn[28] suggest use of oil temperature as a control parameter for damping.

One more aspect of oil film force which has received rather limited attention is the occurrence of tension in oil film where subatmosperic or negative pressures are indicated in a 'cavitated' film. This process is observed by Humes and Holmes[29] as also by Sane[26] *et al.* Humes and Holmes have even incorporated this phenomenon in an approximate way in their analytical treatment of the damper. The phenomenon is not yet understood fully but in some cases it might affect the film forces substantially.

In the light of the above discussion it may be concluded that the squeeze film damper as a part of a rotor configuration has to operate with a number of variables which are only partially controllable or understood. In a design exercise, therefore, the designer should evaluate a system within a wide range of damping values, say, by linear analysis and subsequently when the location and the required degree of damping are ascertained, a more realistic non-linear analysis to examine the undesirable effects and other extreme possibilities may be undertaken. The uncentralised damper seems to be more difficult to tackle and suitable formulations with quasi-linear analysis are desirable.

4. SQUEEZE FILM DAMPED ROTOR MODELS

To be able to appreciate the working of a squeeze film damper from system design point of view, cognizance of the following analytical and experimental studies on simplified models and the generalities derived from them may be taken. Though these models are too simple as compared to the complex configurations of aircraft gas turbine engines, it may be possible to structurally simplify the sections of the latter for comparison with the former in order to acquire qualitative design insights.

i) Mohan and Hahn[30] have analysed a symmetric rigid rotor with squeeze film damped bearings (Fig. 9) in conjunction with centralising springs assuming short bearing π film solution for the damper. Simandiri and Hahn[19] have extended this model to include effect of pressurised oil supply ; they have also experimentally validated[31] all their theoretical findings.

ii) A squeeze film-damped flexible symmetric rotor with centralising springs is investigated by Rabinowitz and Hahn[32] theoretically as well as experimentally[28], highlighting the advantages of pressurisation. They have also done an optimisation study on this model for pressurised damper[33] . This model is also studied by Changsheng Zhu[34] *et al.* for subsynchronous whirls and by Rabinowitz and Hahn[35] for stability.

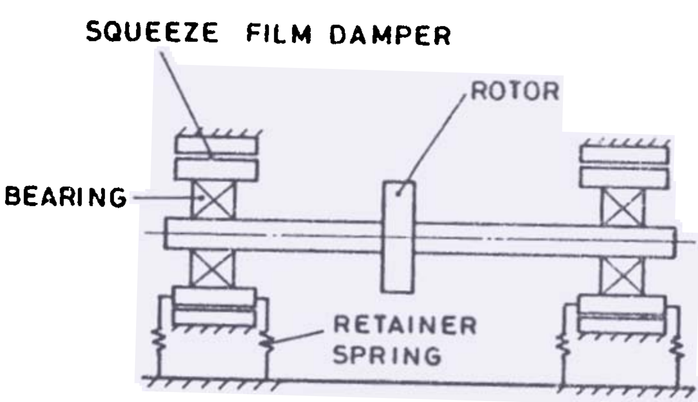

Figure 9. Rigid rotor with centralising springs

iii) Rigid symmetric rotor without centralising spring for damper (Fig. 10) is investigated by Holmes[36], Cookson and Kossa[37] for unbalance response theoretically. Some special aspects of this problem like pressurisation are studied by Humes and Holmes[29].

iv) The above configuration with rotor flexibility added is studied by Cookson and Kossa theoretically[38] as well as experimentally[39]. The subsynchronous orbits of this rotor are investigated by Changsheng Zhu[34] *et al.* and Meng Guang[40] *et al.*

v) Investigations in squeeze film-damped asymmetric models are fewer. The models analysed by Holmes and Dogan[16] (Fig. 11) and Sane and Shende[4] (Fig. 12) may be mentioned in this context although the two are different in concept.

A number of general observations may be made from the above studies, which cover a fairly wide field :

(a) The performance of a squeeze film-damped system may be analysed with the help of the following non-dimensional parameters.

Bearing parameter $B = \mu RL^3/m_B\, \omega_c c^3$,
Unbalance parameter $U = F_u/m_D c\omega_c^2$,
Gravity parameter $\bar{W} = W_D/m_D c\omega_c^2$,
Mass ratio $\alpha = m_B/m_D$,
Damping ratio $\xi = C_d/C_c$,
Speed ratio $\Omega = \omega/\omega_c$.

Only B, U, $\bar{W}$ are relevant to a rigid rotor but all six parameters are involved in the analysis of a squeeze film-damped flexible rotor.

(b) The squeeze film damper without centralising spring has the advantages of cheapness and simplicity of configuration as compared to the centralising design. The former type implies freedom for the rotor to find its own position while the latter permits circular orbits. The two types represent two different design philosophies and also problems of different nature.

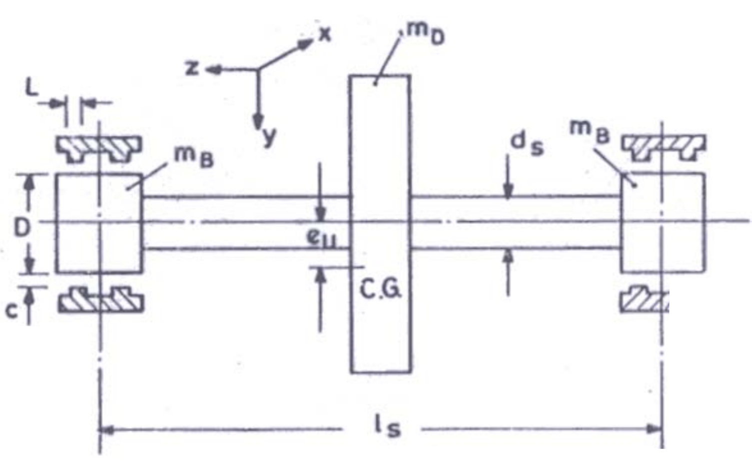

Figure 10. Rig/flexible rotor without centralising springs.

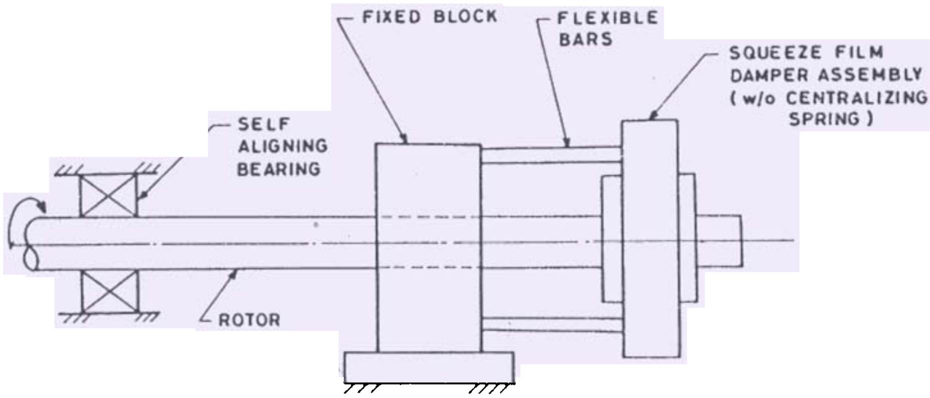

Figure 11 Unsymmetric rotor with squeeze film damping[1]

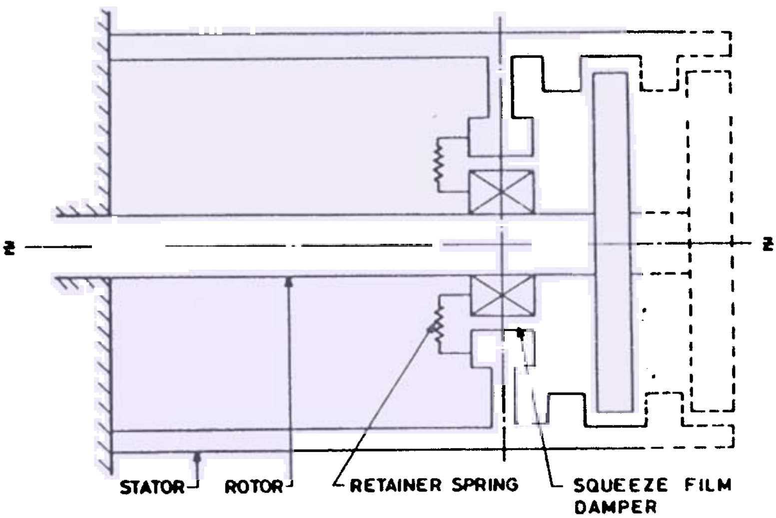

Figure 12. Overhung flexible rotor-casing model[4].

By using centralising springs, the natural frequencies are artificially reduced so that the critical speeds so introduced are traversed well before the normal operating speeds are reached. In the uncentralised design, the natural frequencies are not reduced and only the vibrations while running through the already existing natural frequencies are reduced.

(c) The centralised damper, when functioning with a cavitated film has the possibility of bistable operation which is explained by the two possible configurations of bearing centre, journal centre and rotor mass centre for equilibrium, owing to the obliquity of the film force as depicted[15] in Fig. 13. The higher mode is characterised by very large values of rotor amplitude and force transmitted by the bearing and is undesirable. The bistable mode is observed in case of both, the rigid and flexible rotors and it limits the maximum safe unbalance with which the rotor can operate when passing through a critical speed. Pressurisation is the answer to this problem.

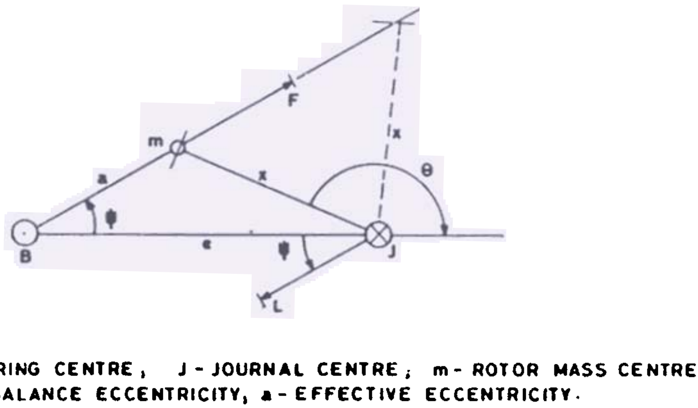

Figure 13. Bistable rotor operation[15].

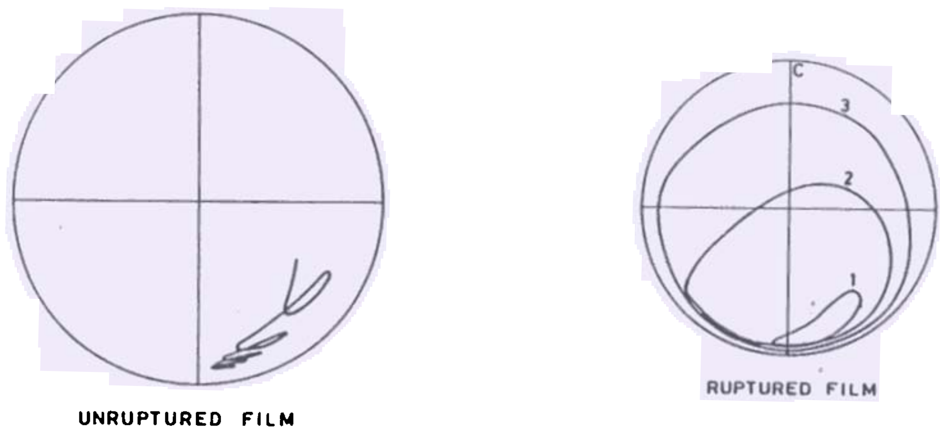

Figure 14. Typical steady-state orbits of a rotor without centralising spring.

(d) The damper without centralising spring, however, cannot support the weight of the rotor unless there is film rupture, since the 2π film has no stiffness component (in phase with displacement) and the rotor falls towards the bottom of the damper cavity (Fig. 14) if the damper is pressurised. The steady-state orbits of such a rotor with ruptured film are non-circular as shown in Fig. 14. The damper orbit is closer to the centre and tends to be circular for smaller values of $\bar{W}$ and the possibility of formation of a stable orbit depends on value of B relative to U ; for small values of B the transient rotor excursion seems to be never ending[36].

One of the classical problems due to non-linearity in case of the rotor with uncentralised damper is subsynchronous whirls at 1/4, 1/3, 1/2 of operating speed (Fig. 15) depending on the various combinations of operating parameters B, U, $\bar{W}$, a, ξ and Ω. Asynchronous precession of rotor is always undesirable owing to the occurrence of alternating stresses ; centralising spring in a damper usually helps to avoid this possibility[34].

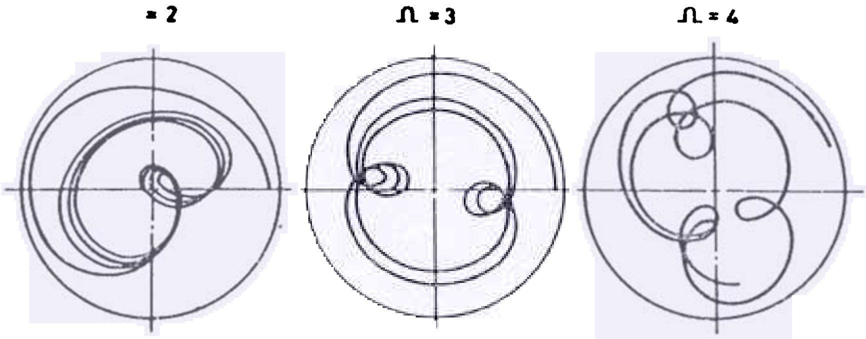

Figure 15. Typical subsynchronous orbits of a squeeze film damped rotor.

(e) The amount of damping in a centralised damper can be specified by means of a linear damping coefficient and by the criteria of low critical speed amplitude and low bearing transmissibility at operating speed, the value of damping coefficient can be optimised. One more criterion for optimum damping could be that the mode shape does not vary with insertion of oil film[9].

Owing to the many complexities associated with the design and operation of an uncentralised damper, it is not easy to optimise the damper design. The degree of damping should not be excessive so that the damper operates like a rigid link between the rotor and stator with no benefit, neither should the damping be deficient in which case large vibrations of bearing with respect to its housing may occur, even though the rotor vibrations may be kept to reasonable limits.

(f) While considering optimisation of damper design the problem of stability against self excitations and non-linear transient response should not be overlooked. Owing to the complex nature of the problems, the published work[41–43] as also inadequate design data do not enable inference of any general rules and the design exercise must include individual investigations into these non-linear aspects of rotor behaviour. Moreover, the effect of stator flexibility in series with squeeze film damper and comparison of performance of a squeeze film-damped rotor with that of a rotor with 'bearing in dry clearance' are also the points of practical interest. With these considerations an overhung rotor-casing model is discussed below.

5. OVERHUNG FLEXIBLE ROTOR-CASING MODEL – A CASE STUDY

The configuration seen in Fig. 12 consists of an overhung rotor-stator combination with interposition of a squeeze film damper between the two together with a parallel centralising spring. This model, in the light of the discussion in section 2 above, is selected for investigations[4] due to its practical significance in that while involving most of the salient features of the turbine portion of an actual engine, a simplicity essential for basic studies is also maintained. Under steady-state synchronous excitation, this rotor is assumed to execute circular orbit owing to the presence of the centralising spring. However, it is felt that the centralising spring is not essential for this assumption if the connection at the coupled end of the rotor is enough stiff. In such a case the

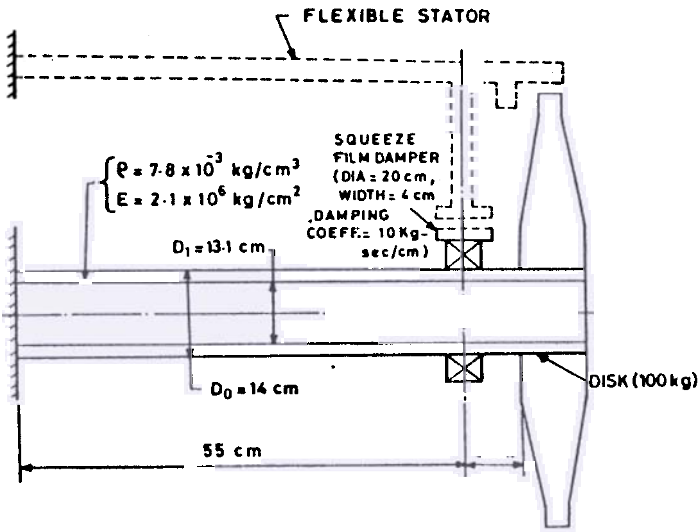

Figure 16. Overhung rotor for case study.

outrigger bearing will mainly perform the damping function. This model is analysed and software is developed for critical speed analysis ; unbalance response and optimisation of damping ; non-linear unbalance response in comparison with bearing operation in dry clearance and with rigid mounting ; effect of flexible stator on synchronous response with squeeze film damping ; linear stability analysis against rotor hysteresis and aerodynamic cross-coupling ; non-linear transient phenomena of blade loss and rotor acceleration, together with sustained transient whirls.

As a typical case, a large rotor of actual proportions as shown in Fig. 16 was investigated with squeeze film damping at the outrigger bearing in a comprehensive manner for all the above effects. This rotor has an operating speed of 10400 rpm and the first critical was found to be around 3500 rpm, the second being nearly 150,000 rpm – far removed from the operating speed range. It was found that the first critical was the primary mode of interest and application of squeeze film damper at the outrigger bearing near the disk gave the best results. The damping coefficient was optimised by linear analysis and the non-linear response of the rotor for steady-state and transient vibrations was determined with this value of damping coefficient obtained by means of two different designs of dampers – with π and 2π film. It was found that the 2π film damper is generally well suited to provide the attenuation and isolation desired; the π damper offers better stability.

6. CONCLUSIONS

A number of complex rotor systems have been designed in the recent times incorporating squeeze film damping for amelioration of vibration problems so that rotor-stator rubs are minimised, bearing life is increased and structures are protected from ill effects of vibration.

A squeeze film damper needs to be strategically located in a rotor layout for which free vibration mode shapes must be examined. In general it is found that the squeeze film damper location near turbine disk is most beneficial. Inter-spool squeeze film damping implies complex configuration and problems of rotor instability. Hence damping of turbine rotor with respect to the case structure appears to provide a secure solution.

Squeeze film dampers have many design variations depending on oil supply arrangements, end seals, pressurisation, etc. The two major implementations of squeeze film dampers are with a centralising spring and without one. These two types represent different design philosophies and problems. In general the dampers with centralising spring are more amenable to analysis and can be designed more confidently.

The degree of damping required in a layout may be decided by study of steady-state unbalance response determined by linear analysis. Squeeze film dampers without centralising spring cannot be linearised easily and a quasi-linear treatment may be necessary in their case. It is difficult to decide upon optimum damping for uncentralised dampers due to their essentially non-linear behaviour.

A number of simplified rotor-bearing configurations with squeeze film damping (with and without centralising springs) have been analysed theoretically and experimentally in the recent times. It may be possible to develop qualitative design insights for more complex systems through a study of these models.

Final design of a squeeze film-damped system can be ascertained only after a thorough non-linear check-up of its behaviour not only in response to rotor unbalance but also due to many other possible sources of excitation. The performance of such a system should not only be compared with rigid bearing mounting but also with bearing mounting in clearance which is a veritable non-linear possibility in aircraft gas turbine operation. Similarly effects of casing flexibility in series with a squeeze film damper need to be evaluated.

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