Application of Squeeze Film Dampers

The basics of design and use of squeeze film dampers in supercritical turbines.

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Squeeze film dampers have traditionally been used to overcome stability and vibration problems that are not adequately handled with conventional style bearings. One of the key design features in a squeeze film damper configuration is the introduction of support flexibility and damping in the bearing/support structure. This translates to lower transmitted forces and longer bearing life, particularly for machinery that is designed to operate at supercritical speeds. Machinery that runs above the first critical speed is classified as super-critical, and constitutes an increasing number of the new high performance machinery manufactured today. Higher efficiency, and lower bearing transmitted forces, are some of the advantages of running super-critical. Lower damping (not more damping) will also result in lower transmitted forces. However, since some damping is required to allow for safe traversing of the critical speed, an optimum value must therefore be determined to satisfy the two rather conflicting requirements. This is why an optimization process is generally required to determine the "right" amount of damping.

This article describes some of the commonly used squeeze film dampers, and introduces one of the more novel damper designs. Examples of the optimization process required in the design and selection of squeeze film dampers and their applications, to solve stability and critical speed problems, are also demonstrated.

Squeeze Film Dampers Without a Centering Spring

This is the simplest of the squeeze film damper configurations. The outer race of a rolling element bearing, or the outer bearing shell in the case of a fluid film bearing, is allowed to float and whirl in a clearance space between the bearing outer diameter and the housing inner diameter. As shown in Figure 1, the outer race, or bearing shell in the case of fluid film bearing, forms the damper journal, which is allowed to whirl, but is prevented from spinning by a "loose" anti-rotation mechanism. This "loose" anti-rotation configuration is necessary to allow the damper journal or outer race to whirl or orbit (not spin) in a precession motion, squeezing the oil in the small clearance space and generating an oil film pressure and, subsequently, a damping force.

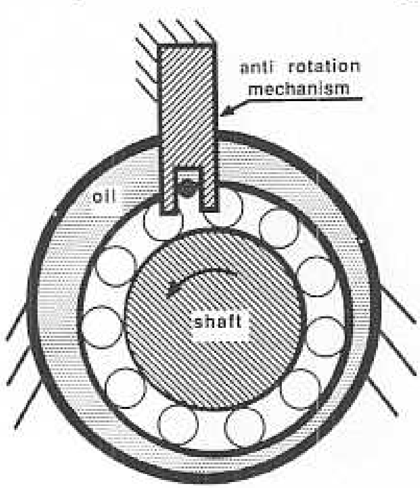

Fig.1 *Squeeze film damper without a centering spring*

The absence of a mechanical centering spring in this design configuration means that the damper journal will be bottomed out at start-up. As the speed increases and the shaft starts to whirl, the damper's journal (bearing shell outer surface) will lift off. The oil film in a squeeze film damper does not produce stiffness (i.e., support a static load) like conventional fluid film bearings. However, the damper does develop stiffness-like behavior. This stiffness is due to the cross-coupled damping coefficients, which exhibit stiffness-like (spring) characteristics.

The non-centered damper is one of the most non-linear of the squeeze film damper designs. There are two basic mechanisms that are responsible for this non-linear behavior. The first of the two non-linear mechanisms is attributed to the non-linear characteristics produced by the cross-coupled damping coefficients. The second source of non-linear behavior present with this type of damper comes as a direct consequence of the bottoming out of the damper journal. This generally occurs at high side loads, or due to excessive unbalance forces. The bottoming out of the damper journal, which is very likely with this design due to the absence of a centering spring, will result in a bi-linear spring behavior. This non-linear behavior can be inferred from the sub-synchronous and super-synchronous vibration characteristics often noted on this type of dampers. In some cases, the impact force generated when the damper journal bottoms out excites the lowest natural frequency of the rotor. In the case of flexible casings and support structures, the resonance frequencies of the structure can also be excited.

The non-centered dampers are commonly used on aircraft gas turbines, light weight process compressor rotors, and automotive turbochargers. In aircraft engines, their use has been limited to the smaller engines, where the use of a conventional style centering spring (squirrel cage spring) is difficult to implement due to space limitations.

O-Ring Supported Dampers

The simplest means of providing a centering spring in a squeeze film damper is through the use of elastomer O-rings. An illustration of this damper design is shown in Figure 2. The advantages of this design stem from its simplicity, ease of manufacture, and the ability to incorporate the damper into small envelopes. The relatively low radial space required makes it the preferred method to retrofit existing machines in the field. The O-ring doubles as a good end seal, which helps increase the effectiveness of the damper by reducing side leakage.

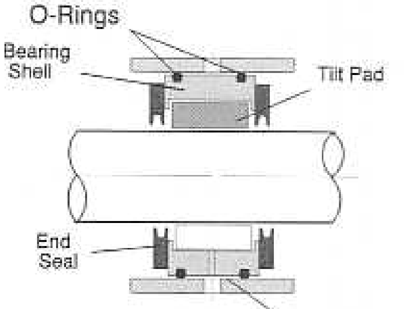

Fig.2 ***Schematic of an O-ring supported squeeze film damper***

Some of the disadvantages with this design are attributed to the limited range of stiffness that can be achieved with elastomers. Predicting the stiffness with a good degree of certainty is difficult in elastomeric materials, due to the material variance, and the influence of temperature and time on its properties. The elastomer stiffness and damping is strongly influenced by temperature. The O-ring design is also susceptible to creep, causing the damper to bottom out, which, as discussed above, may lead to a bi-linear spring behavior.

O-ring dampers are not capable of taking thrust loads, and cannot be easily manipulated for centering of the damper journal within the damper clearance space. One means of achieving some centering capability is through making the O-ring groove eccentric. This limitation makes them only suitable for use with lightweight rotors.

High speed and high pressure centrifugal compressors prone to stability problems frequently have been fitted with these O-ring dampers. The damper is installed in series with tilting pad fluid film bearings to enhance the stability of the compressor. Although most of their use has been primarily aimed at improving stability, they have also been used to reduce the synchronous response due to imbalance, or to shift the peak unbalance response outside the operating speed range.

The O-rings also provide a form of internal friction damping (hysteretic damping), in addition to the squeeze film damping (viscous damping) produced by the oil in the damper. The elastomer material of the O-ring limits the use of such dampers to mostly low temperature applications. The hysteretic damping from the O-rings has also been utilized in series with rolling element bearings, and gas bearings without any oil in the clearance space (dry O-ring damper). High speed dentist drills are an example where this configuration is commonly used. These drills are composed of air turbines running on ball or gas bearings, with elastomer O-ring supports for increased damping and improved stability.

Squirrel Cage Supported Dampers

This is the most commonly used squeeze film damper design, particularly in aircraft engines where its use is widespread. Most large aircraft gas turbine engines employ at least one, and in many instances, two or three of these dampers in one engine. A schematic of this damper is shown in Figure 3. A distinctive feature necessary with such a design (and apparent from the schematic), is the relatively large axial space required in comparison to the damper length. This is one of the major drawbacks of this damper design. The squirrel cage forming the centering spring for the damper quite often requires three to four times as much axial space as the damper itself.

Assembling the squirrel cage spring and centering the journal within the clearance space requires special tools and skills. The squirrel cage spring also complicates the damper end seal design and assembly. It is also very difficult to offset the spring assembly, in order to account for the gravity load due to the shaft weight. Maintaining parallelism between the damper journal and housing is another factor that adds uncertainty and complications to this design.

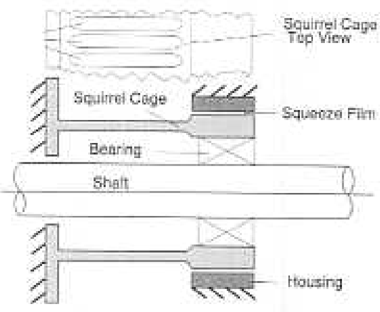

Fig.3 ***Schematic of a squirrel cage supported damper***

Integral Damper-Centering Spring

The new patented integral damper centering spring[1] is shown in Figure 4. The cantilevered support ribs, along with the sector they are supporting at both ends, form a centering spring element. The small gap between the sector and the outer ring forms the squeeze film damper clearance space. This new centering spring squeeze film damper design does not occupy any additional axial space beyond the existing length occupied by the bearing. This is a major advantage over conventional squirrel cage type damper supports, which in contrast require three to four times the axial space occupied by the damper or bearing. Furthermore, unlike the first two damper configurations (non-centered and O-ring supported dampers), this new design is capable of absorbing axial thrust loads without locking the damper's radial motion.

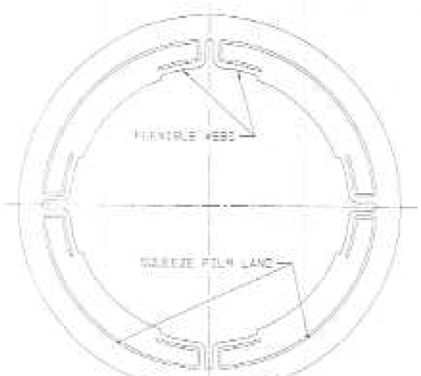

Fig.4 ***Schematic of an integral centering spring squeeze film damper***

The integral design makes manufacturing, assembly, and inspection much easier and more reliable than any other configuration. The squeeze film gap can be made with precision utilizing this concept. Wire Electric Discharge Machines (EDM) provide an excellent means of obtaining the desired clearance with high precision and repeatability, maintaining excellent parallelism between the damper journal and housing. Damper retrofit in process type equipment, which are required to meet API specifications dictating a split bearing configuration, can be easily accommodated with this new concept. This design, unlike conventional style dampers, can be easily provided in a split configuration as shown in Figure 5.

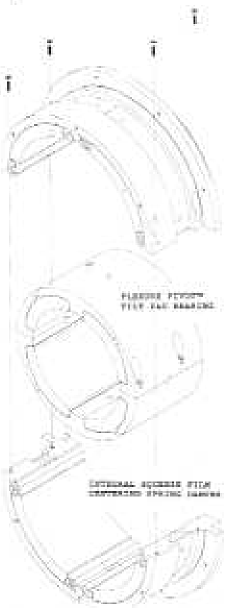

Fig.5 *Schematic of a split integral squeeze film damper*

Optimization for Improved Stability

In process equipment, squeeze film dampers are used primarily to improve rotordynamic stability. They are usually used as a last resort, due to the complexity and drawbacks inherent in conventional dampers. The difficulty associated in accurately predicting damper performance due to cavitation[2], is another reason for their limited use. The new damper design with the integral centering spring overcomes many of these difficulties as shown by the following example.

This example demonstrates the importance of the centered damper in suppressing the sub-synchronous vibration exhibited by the relatively heavy rotor shown in Figure 6. This rotor was supported by O-ring type dampers, but continued to exhibited sub-synchronous vibrations after the damper retrofit. The stability without the squeeze film damper is very low, as indicated by the low logarithmic decrement predicted for the first forward mode shown in Figure 7. The rotor is much more flexible than the bearings, resulting in very small motion at the bearing supports and consequently low effective damping. The bearings are virtually at a node, and therefore are ineffective in suppressing the sub-synchronous vibrations.

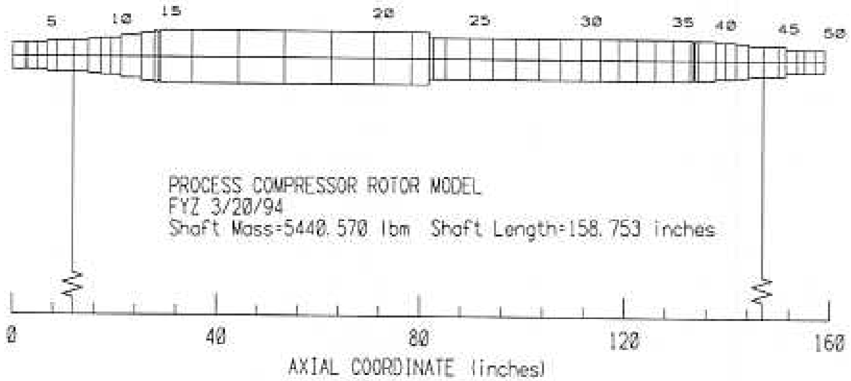

Fig.6 *Rotor model for a typical heavy process compressor*

The use of a squeeze film damper in series with the tilting pad bearings will introduce flexibility and damping to the bearing support. The damping provided by the squeeze film damper however, must be optimized as shown in Figure 8. Low values of damping may not be sufficient, and high damping may lock the damper and reduce the effective damping, as evidenced by the reduction in the logarithmic decrement at high damping levels.

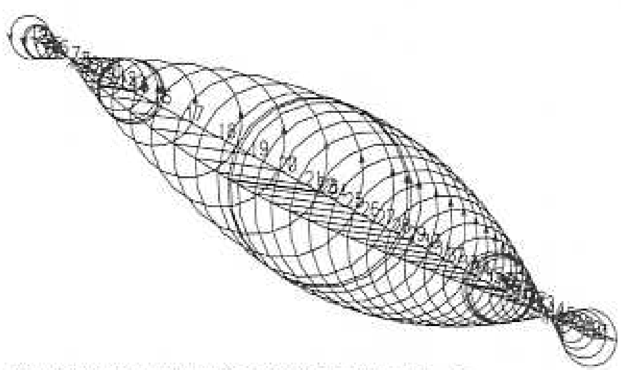

Fig.7 *First forward mode on fluid film tilt pad bearings.*

Using the optimum damping value, and varying the squeeze film support stiffness, showed that a more flexible spring support will allow motion at the bearing, resulting in more effective damping, and thus suppression of the instability and sub-synchronous vibrations. However if the damper bottoms out, as was the case with the O-ring damper, the stiffness will increase and the damper becomes ineffective. The damper is very effective when the damper journal (tilt pad bearing shell OD) can be held close to the centered position within the damper clearance space. This is very difficult to accomplish with the conventional O-ring dampers, particularly in the case of relatively heavy rotors.

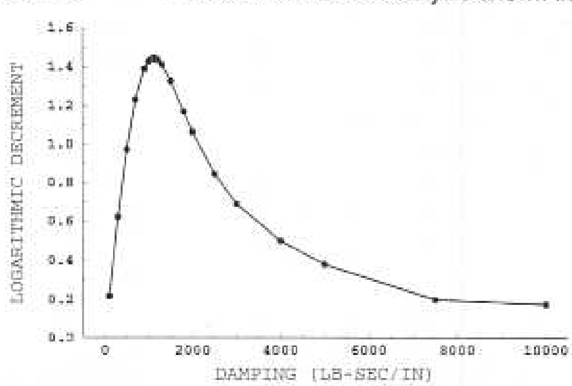

Fig.8 *Damping optimization for improved stability.*

The use of an integral centering spring damper configuration allows precise location of the damper journal, and realization of the required stiffness value. In order to achieve the low stiffness values while still maintaining lower stresses, the damper configuration shown in Figure 5 was utilized. The "S" shaped flexible elements were required to keep the stiffness and stresses low, due to the relatively heavy rotor weight as evident by the finite element stress analysis shown in

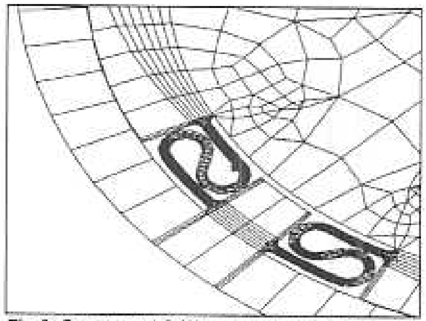

Fig.9 Stress and Stiffness Analysis Using ANSYS.

Figure 9. The wire EDM technology allows the production of such a damper device, which can be easily designed with an offset to compensate for the deflection due to rotor weight.

The stability of the rotor-bearing system was greatly improved with the optimized squeeze film damper. This is evident from the relatively high logarithmic decrement for the first forward mode shown in Figure 10. The flexibility introduced with the integral centering spring allows for motion at the bearings, thus making the damping more effective.

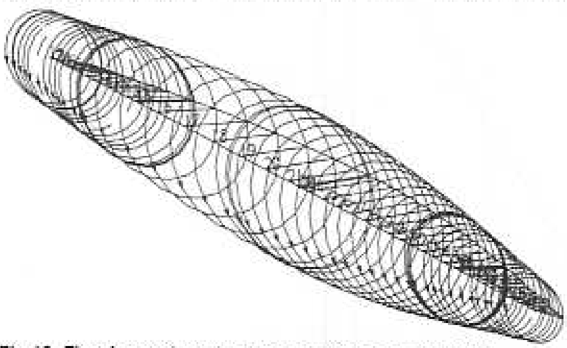

Fig.10 First forward mode after including a squeeze film damper in series with the tilt pad bearing.

Control of Critical Speeds

The squeeze film damper in series with a fluid film bearing also provides a means for improving the synchronous response, an advantage rarely utilized with process type equipment. The tight vibration restrictions with the latest API specifications, and the even more restrictive specifications dictated by the users, makes the squeeze film damper a very attractive and practical alternative to other costly and time consuming designs. The squeeze film damper will help in reducing the sensitivity to unbalance, which often results in significant delays during shop testing and commissioning of process type equipment. The following example shows how the squeeze film damper softer support can shift the critical speed down, removing it from the operating speed range.

The undamped critical speed for a process steam turbine is shown in Figure 11. This turbine's operating speed range was too close to the first critical speed, which forced operation at higher speed than necessary for process requirements, consuming more steam in order to maintain acceptable vibration levels. A stiffer bearing, evident from the undamped critical speed map, will not help in this case. Higher bearing stiffness will have a minor effect on the critical speed if any, and will shift the critical closer to the running speed, which is not the direction sought in this application. The oil film stiffness in the existing bearing cannot be made any softer since the sleeve bearing configuration in use is one of the softest fluid film bearings available. Therefore, a drastic reduction in the bearing support stiffness can only be achieved by incorporating a squeeze film damper in series with the fluid film bearings. The response comparison shown in Figure 12 demonstrates the distinct advantages made possible with the squeeze film damper.

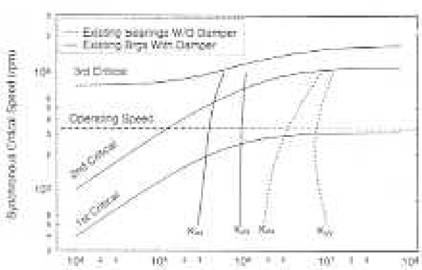

Fig.11 Undamped critical speed map.

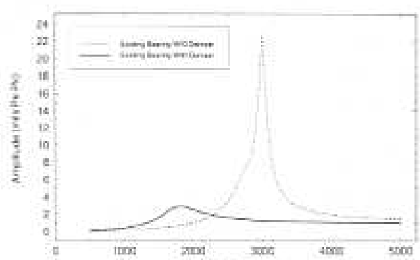

Fig.12 Response improvement with squeeze film damper supports.

Summary and Conclusions

The use of squeeze film dampers in problem process machinery has tainted it as a "bandage" solution, and many users shy away from its use for this reason. Another reason for their limited use is the difficulty in accurately predicting the performance, particularly with O-ring supported dampers.

These deficiencies with conventional dampers have been addressed with the novel integral damper centering spring design. One of the key attributes of this design is the ability to accurately predict its stiffness and damping characteristics. It can also be easily adapted to work with fluid film bearings in a split configuration, as required by the API specifications. The stability with existing fluid film bearings can therefore be extended with this new damper concept. This should allow operation at higher speeds, and possibly with longer spans between bearings. The balance limitations and threshold speeds on many of the existing machinery can be extended.

The ability to center such a damper in applications where relatively heavy rotors are used has proven to be very advantageous. The limited axial space present with most machinery does not hinder its applicability. Furthermore, the accuracy in predicting the stiffness and damping is another very desirable feature that has been missing with conventional damper designs. ■

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References

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