

Feasibility Studies on Newly Conceptualized Inter-shaft Squeeze Film Damper (ISSFD) Rings for Vibration Attenuation



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Abstract Squeeze film dampers (SFDs) have become very essential for high-speed turbomachinery as means of vibration energy dissipating mechanism. However, SFDs are not very common for inter-shaft bearing applications and are still in the R&D phase. It is observed that the application of SFDs for inter-shaft bearing systems is not being pursued in a real practical sense, as it becomes extremely difficult to realize a compatible component facilitating the radial squeezing mechanism—where squeeze film oil could be introduced—thus resulting in squeeze film damping. The gap that could be made available between the inner spool and inner race or the outer spool and the outer race of the inter-shaft bearing also being very little further complicates the application of a conventional squeeze film damper in the inter-shaft bearing plane. In this research work, newly conceptualized inter-shaft squeeze film damper (ISSFD) rings are fabricated and tested for evaluating their damping potential characteristics in dedicated instrumented test rig/s fabricated for the purpose. Parametric experimentations are conducted in a single-spool test rig as a proof of concept in attempting to quantitatively evaluate the damping potential of ISSFD rings and hence the suitability of their applications in inter-shaft bearing plane of two-spool system. This research study very clearly indicated the damping contribution of the ISSFD rings, and the performance of the system improved in terms of

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substantial reduction in shaft vibration amplitudes. The study also clearly indicated that the ISSFD rings contribute toward the stiffness in the bearing plane also, and as a result, the rigid body critical speed gets shifted. Parametric experimental studies brought out the effect of different geometric parameters on the stiffness and damping contribution of ISSFD rings.

Keywords Inter-shaft squeeze film damper (ISSFD) ring · Gas turbine engine · Two spools · Critical speed · Stiffness · Damping factor

1 Introduction

As SFDs have become an inevitable component in modern gas turbine engines, researchers are emphasizing on different aspects of SFDs from the point of view of improving their damping potential in the space-constrained gas turbine systems. While applications of SFD have become very common, the necessity of the same in the inter-shaft bearing plane applications, although being felt, however has still not been realized due to paucity of research from practical application point of view. Attempts however have been made by several researchers to conceptualize possible configurations of SFD suitable for inter-shaft bearing plane with practically no application potential.

Li et al. [1–3] have calculated steady-state unbalance response of the rotor system for different shaft speeds and have investigated damping effects of an inter-shaft squeeze film damper with different radial clearances under various levels of rotor unbalance. Their suggestion is that, for designing ISSFD, it should be necessary to consider the operating speed of the rotor below the threshold speed of the rotor system. Shafei et al. [4–6] have shown that the inter-shaft bearing dampers are stable supercritically only with a configuration in which the oil film does not rotate. Their ISSFD in this case is mounted in a set of double-decker rolling-element bearings with a non-rotating squeeze film. They have introduced squeeze film in between two sleeves, one on the outer race of the inner bearing and the other on the inner race of the outer bearing. These races are prevented from rotation using squirrel cages. The two sleeves are docked together to prevent their relative rotation and allow only a relative whirl between them. Defaye et al. [7] experimentally found that the efficiency of an ISSFD depends on the type of the crossing mode and on the direction of rotation of both shafts. They have suggested that an additional research effort is necessary in order to propose an optimal ISSFD technology. The influence of various parameters such as the shaft bending mode, the unbalance value and the radial clearance of the ISSFD must still be investigated. Gupta et al. [8, 9] analyzed ISSFD using Reynolds equation. They have tested an improved and inherently stable inter-shaft squeeze film damper based on El-Shafei's design in a test setup having two rolling element bearings; one each mounted on LP and HP rotor shafts. They have considered two design modifications for analysis and tested them experimentally. Jayaraman and Arunkumar [10] have discussed the concept of flexible supports/mounts which are

mounted in between the bearing and the shaft to bring down the rigid rotor critical speeds.

From the literature review, it is understood that a very few researchers have made attempts with regard to realizing SFDs for inter-shaft bearing applications. Most of these attempts do not seem to be feasible from practical application point of view, primarily because of the space constraints in the inter-shaft bearing plane and secondly because of the instability problems that exist in the existing SFD designs pertaining to oil film rotation. It is very clear from the above discussions that the efforts on developing ISSFD from practical application perspective are very scanty. As such an attempt has been made here to develop practically feasible ISSFD suitable for space-constrained gas turbines. Research effort in here basically includes conceptualization of ISSFD rings (suitable for space-constrained gas turbines), fabrication and testing for their damping potential.

2 Newly Conceptualized ISSFD Rings

Figure 1 shows schematic representation of two-spool configuration of a typical gas turbine engine in which a conventional squeeze film damper (SFD) is mounted on bearing-3. Bearing-4 represents an inter-shaft bearing, and the gap that could be made available between high-pressure (HP) spool and low-pressure (LP) spool is as low as 3–7 mm, and hence, it is a real challenge to develop a suitable squeeze film damper with a non-rotating oil film. In order to mount SFD in the inter-shaft bearing plane in such a small gap, the maximum radial space available for the damper should be well within 7 mm facilitating the squeezing action. Also, the necessity of very thin grooves for oil film needs to be created in the ring that would create squeezing action under time-dependent loading conditions. In a real practical sense, it should be a compatible component facilitating the radial squeezing mechanism, where squeeze film oil could be introduced, thus resulting in squeeze film damping.

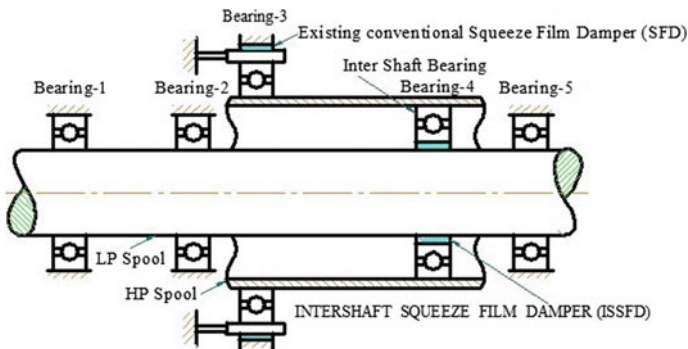


Fig. 1 Schematic of a typical two-spool gas turbine engine

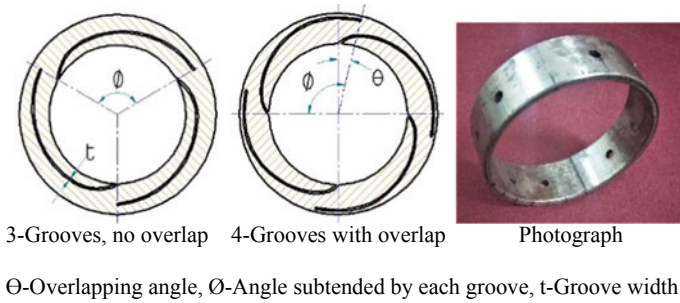


Fig. 2 Schematic/photograph of newly conceptualized ISSFD ring configurations

Figure 2 shows the schematic/photograph of the newly conceptualized ISSFD rings. The possible geometric parameters that could be varied include number of grooves, angle subtended by each of the groove/groove angle and overlapping angle of the grooves apart from groove width (nominal oil film thickness). Groove width was kept constant for the present study, while experiments have been conducted in a single-spool rig using newly conceptualized ISSFD. The assembly of ISSFD in a two-spool inter-shaft bearing (bearing-4) could be either with the LP spool or with the HP spool. It is preferable to have the ISSFD assembled with the LP spool from the point of view of assuring the presence of oil in the groove due to centrifugal action as shown in Fig. 1. The conventional squeeze film damper as well as the proposed inter-shaft squeeze film damper has rigid boundaries.

3 Test Rig and Instrumentation

To explore the suitability of ISSFD rings, two test rigs—a static test rig and a single-spool dynamic test rig—were designed and fabricated, and experiments were conducted. Basically, a static test rig is designed to evaluate the variation of static stiffness along the circumferential direction of ISSFD rings. A single-spool dynamic test rig is designed to demonstrate the damping potential of ISSFD rings defined in terms of vibration attenuation and also facilitating parametric studies.

3.1 Static Test Rig

The static test rig facilitates loading of ISSFD ring and measurement of deflection along the load line. While Fig. 3a shows the schematic of the static test rig, Fig. 3b shows the photograph of the same. The mandrel along with the ISSFD ring is supported between two V-blocks which are mounted on the base plate. The loading

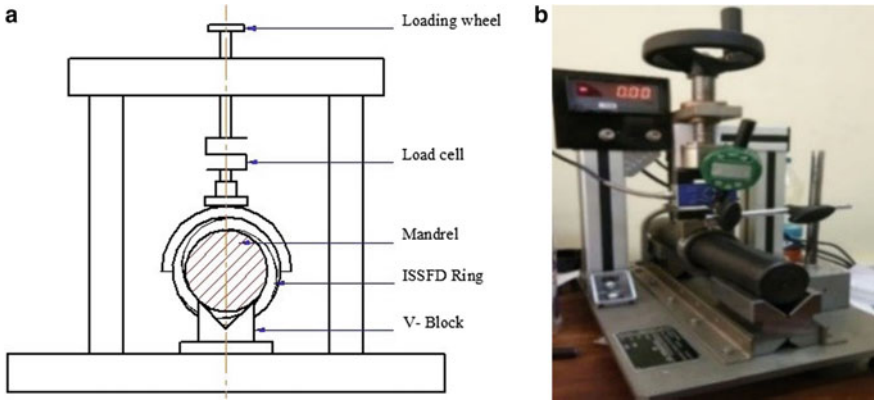


Fig. 3 a Schematic of static test rig, b photograph of static stiffness test rig

unit includes the screw rod arrangement for application of load. The instrumentation includes measurement of load and deflection along the loading direction.

3.2 Single-Spool Dynamic Test Rig

This test rig essentially consists of a horizontally configured shaft with a variable speed motor capable of running up to 16,000 rpm. The horizontally configured shaft is supported on two bearings, with one of them supporting the ISSFD ring. An oil pump provides supply of squeeze film oil, and provision of radial holes in the ring ensures continuous presence of oil in the grooves. Instrumentation basically includes measurement of speed and vibration amplitudes in two mutually perpendicular directions. Facility in the rig is provided to mount any desired amount of unbalance on either side of the bearing mounted with ISSFD ring to excite the system. While Fig. 4a shows the schematic of dynamic test rig, Fig. 4b shows the photograph of the same.

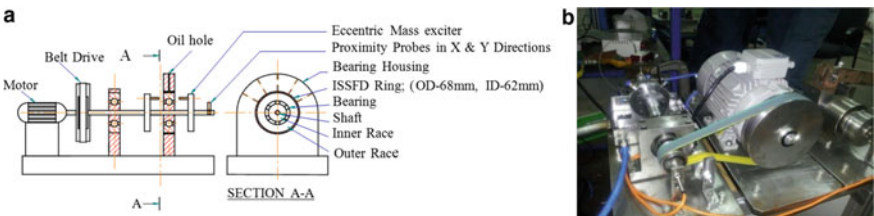


Fig. 4 a Schematic of the dynamic test rig, b photograph of the dynamic test rig

4 Results and Discussions

Experiments, parametric in nature, were conducted to evaluate the stiffness offered by the ISSFD rings (under static condition) and damping potential in terms of reduction in shaft vibration amplitudes. The system was run supercritical in order to bring out the effectiveness of the damper (as experienced by the ISSFD ring when mounted on a typical inter-shaft bearing) and hence the presence of ISSFD ring. Tests were conducted on three and four-grooved ISSFD rings, the geometric details of which are provided in Table 1.

4.1 Static Stiffness

Static stiffness details were obtained for all the four ISSFD rings with varying number of grooves, angle subtended and overlapping angles. For all the cases, the groove width was kept constant at 300 μm . Figure 5a indicates the typical load deflection pattern for a three-grooved ISSFD ring with 135° subtended angle clearly indicating relatively lower stiffness at relatively lower loads followed by a sharp increase in stiffness (very little deflection change) after certain level of loading.

This sharp change in stiffness is due to the initiation of metal to metal contact (groove closure) after a particular loading level. Figure 5b schematically shows the loading of a typical ISSFD ring. The circumferential stiffness variation is as shown in Fig. 5c. Table 2 provides the details of the stiffness values for a three-grooved ISSFD ring with 135° subtended angle.

Figure 6a, b shows the corresponding results obtained for other rings, while Table 3 shows relevant results.

Figure 7 shows the variation of average stiffness for three-grooved ISSFD rings for different values of subtended angles. By increasing the subtended angle/groove length and hence the overlap angle of grooves of ISSFD ring, the average stiffness value decreases.

Table 1 Geometric details of ISSFD rings

No. of grooves	3 grooves			4 grooves
Groove angle	135°	150°	165°	105°
Overlap angle	15°	30°	45°	15°
Outer diameter	68 mm	68 mm	68 mm	68 mm
Inner diameter	62 mm	62 mm	62 mm	62 mm
Average groove width	300 μm	300 μm	300 μm	300 μm

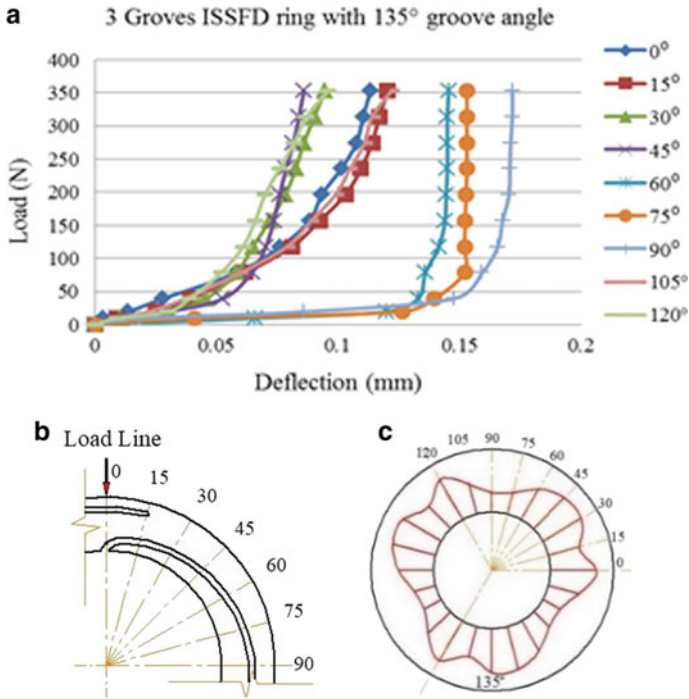


Fig. 5 a Load versus deflection pattern for three-grooved ISSFD ring with 135° subtended angle, b loading along zero degree for three-grooved 135° ring, c circumferential variation of stiffness

4.2 ISSFD Performance Evaluation

The effectiveness of the presence of newly conceptualized ISSFD rings was quantitatively estimated primarily from the point of view of vibration attenuation and shift in the rigid rotor critical speeds as a result of stiffness contribution from ISSFD squeeze film oil. Experiments were conducted on three- and four-grooved rings with different groove subtending angles (and hence overlapping angles). Figure 8 indicates the variation of shaft amplitude as a function of speed with and without the presence of oil for a three-grooved ISSFD ring with 135° subtending angle for two levels of unbalance. It is very clear that the presence of oil has attenuated the vibration levels at all speeds. It can also be observed that there is a shift in the damped critical speed (from 12,600 to 13,800 rpm) indicating the addition of stiffness in the bearing plane. Similar results were obtained when the unbalance levels were increased from 164.92 g-mm to 235.74 g-mm and that there is no change in the critical speed levels. Vibration amplitudes have attenuated by about 35% at both levels of excitation. Contribution of damping factor ‘ξ’ is estimated using half power method, and there is an increase in the value of ξ by 70.15% for 162.94 g-mm unbalance level and 78.43% for 235.74 g-mm unbalance level. This is due to relatively increased squeeze

Table 2 Stiffness values for three-grooved ISSFD ring with 135° groove angle

Three-grooved ISSFD ring with 135° groove angle (average $K = 3514.08$ N/mm)										
Loading line position (°)	0	15	30	45	60	75	90	105	120	
Stiffness (N/mm)	3202.08	3268.74	4618.46	4582.46	3662.82	2287.04	2063.03	3316.93	4625.75	

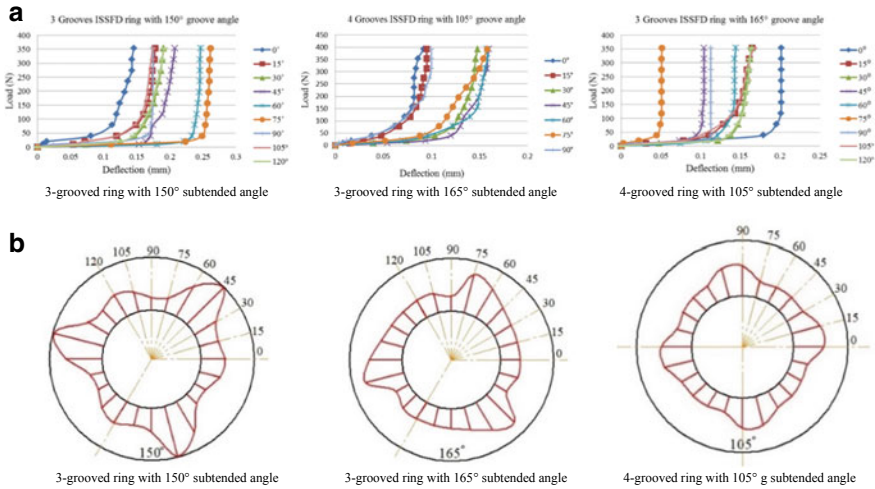


Fig. 6 a Load versus deflection curves for three-grooved and four-grooved rings, b circumferential variation of stiffness for three-grooved and four-grooved rings

velocity as a result of increased vibration levels at higher unbalance levels. Figures 9, 10 and 11 show the corresponding results obtained for other ISSFD rings.

5 Conclusion

An attempt has been made to newly conceptualize inter-shaft squeeze film damper systems in space-constrained gas turbines from practical application perspectives. The conceptualization essentially aims at realizing the presence of oil films in a suitably designed ring compatible to be accommodated in practical gas turbines. The geometrical variation considered in the newly conceptualized ISSFD ring includes number of grooves provided in the ring, angle subtended by each of the groove (and hence the overlapping angle) and groove width which actually would be the average squeeze film thickness. A dedicated instrumented test rig was designed and fabricated, wherein the speeds ranging up to 16,000 rpm could be realized. Arrangements were provided in the rig to vary unbalance levels and also the rigid rotor critical speeds. Detailed parametric experimentation was conducted to obtain the effect of different geometric parameters on the performance of the system (defined in terms of vibration attenuation) apart from estimating the stiffness of rings under static condition. The study very clearly indicated the high potential of the application of the newly conceptualized ISSFD rings for practical gas turbines in view of their suitability for space-constrained situations. The qualitative evaluation indicated substantial damping contribution (and hence the vibration reduction) and moderate stiffness contribution and hence shift in rigid body critical speeds.

Table 3 Stiffness values for three-grooved and four-grooved ISSFD rings

3 grooved ISSFD ring with 150° subtended angle (average $K = 3045.13$ N/mm)										
Loading line position (°)	0	15	30	45	60	75	90	105	120	
Stiffness (N/mm)	2201.63	2445.11	2972.11	6420.02	4405.96	1746.1	1733.87	2483.36	2980.07	
3 grooved ISSFD ring with 165° subtended angle (average $K = 2758.97$ N/mm)										
Loading line position (°)	0	15	30	45	60	75	90	105	120	
Stiffness (N/mm)	1357.32	2363.151.11	2470.18	2951.11	4096.5	4941.51	1788.45	2388.58	2473.86	
4 grooved ISSFD ring with 165° subtended angle (average $K = 2704.63$ N/mm)										
Loading line position (°)	0	15	30	45	60	75	90	105	120	
Stiffness (N/mm)	3218.69	3540.13	2300.27	2204.45	2142.09	2314.53	3212.26	-	-	

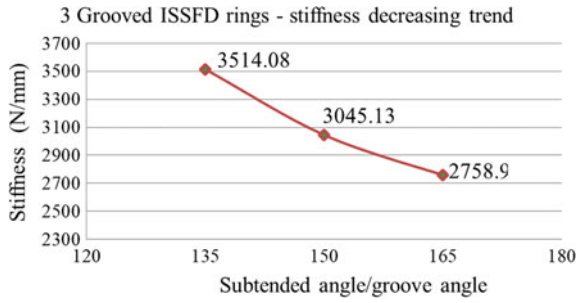


Fig. 7 Variation of average stiffness value with variation in subtended angle/groove length and overlap angle of grooves

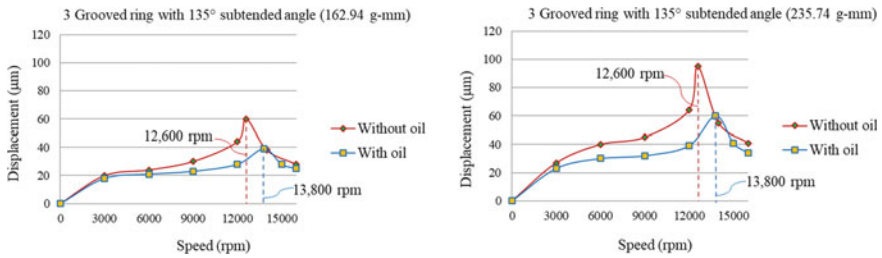


Fig. 8 Vibration amplitude for three-grooved ring with 135° subtending angle

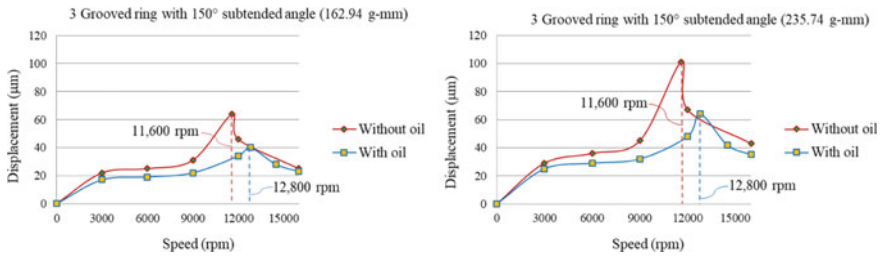


Fig. 9 Vibration amplitude for three-grooved ring with 150° subtended angle

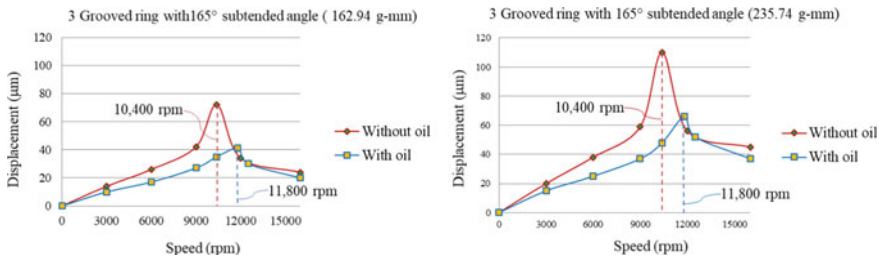


Fig. 10 Vibration amplitude for three-grooved ring with 165° subtended angle

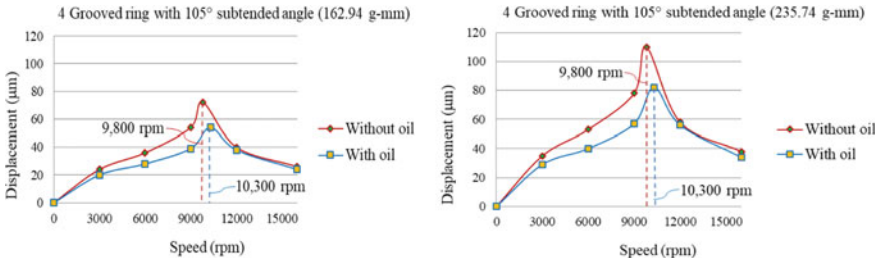


Fig. 11 Vibration amplitude for four-grooved ring with 105° subtended angle

Acknowledgements The authors immensely acknowledge the financial support received from: Gas Turbine Research Establishment (GTRE) under Gas Turbine Enabling Technology (GATET) scheme, DRDO, PO Box 9302, C. V. Raman Nagar, Bangalore-560 037 and B.M.S. College of Engineering, Bull Temple Road, Bangalore-560019.

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