FSI and Modal Analysis of Elastic Ring Squeeze Film Damper for Small Gas Turbine Engines

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Abstract: The high speed gas turbine is a power plant developed for modern aircrafts. It is widely used and developed because it can meet the high power to weight ratio. The rotor system of modern small gas turbine works above the critical speeds. Hence, there is a stricter requirement for the control and isolation of vibration magnitude under heavy unbalance load and passing through critical speeds. An advanced oil film damper known as Elastic Ring Squeeze Film Damper (ERSFD) built with orifice pattern has better dynamic characteristics, vibration-isolating effect, simple structure, high reliability, enhanced damping effect when compared with conventional SFD. This ERSFD and rotor components are analyzed using COMSOL to find its mode shapes and ERSFD's profile pressure distribution at orifice under the ring subjected to vibration. These analyses made using COMSOL structural and CFD modules followed by eigenfrequency response and thin film flow. The above study results on ERSFD which enhances the controlled flow at identified locations. It also helps in designing small gas turbine engine high speed rotor system operating at super critical speeds.

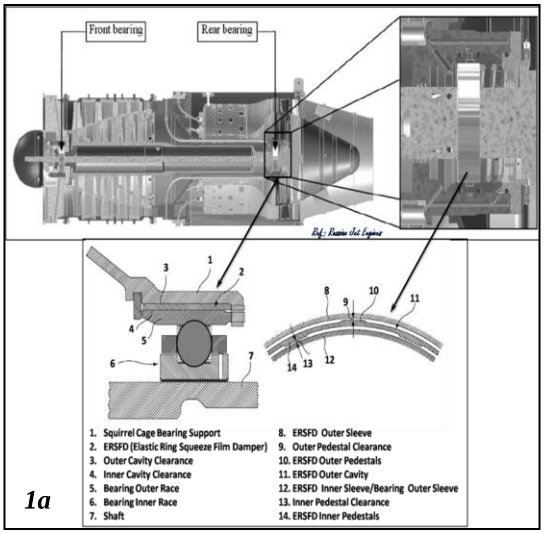
Keywords: *Elastic Ring Squeeze Film Damper, GT engine, Fluid Structure Interaction and Modal Analysis*

1. Introduction

Elastic Ring Squeeze film damper (ERSFD) is current interest of study. The damping effect occurs on ERSFD when a structure moves in close proximity to another surface, in effect with alternately stretching and squeezing incompressible fluid that may be present in the space between the moving structures. Reynolds published the squeeze film effect in his study [8] on lubrication. This effect was an important mechanism for the generation of pressure in a lubricating film together with the wedge effect. The squeezed fluid can act as a mass, spring and damper, having a significant effect on the dynamics of the oscillating structures. The primary goal of a fluid film damping system is to limit the vibration of a given structure by dissipating the energy to the fluid within the cavity [1-4].

The present bearing support structure has a flexi-centering elastic ring comprising of non-contact cavities and of metal to metal contact pedestal pads radially protruding from inner and outer circumferential surfaces of the ring. Flexible portions of the ring are defined between the contact pedestals and have a radial thickness less than that of the ring at the contact pedestal pads, such as to permit elastic deflection of the ring in a radial direction between the contact pedestals. These inner and outer pedestals are used to hold anti rotation sleeve mounted on bearing outer race with bearing housing resulting in the stiffener in bearing plane. Orifice openings provided within the flexible portions of the ring are sized and configured to permit unrestricted smooth oil flow and the flow pattern to further enhance the damping [1, 3].

The design for the ERSFD rotor bearing system is commonly determined by their basic geometric parameters such as the length, diameter, bearing dimension ratio, concentric pressure ratio, and the relevant vibration control device and its dimensions (i.e. capillary diameter). Modeling and numerical simulations of ERSFD have been used in designing and investigating ERSFD systems dynamics to control rotor vibrations and transmitted forces to the base structure [6-8]. This ERSFD rotor bearing type has better dynamic characteristics than other anti-whirl configurations, i.e., good suppression of whirl, good damping at critical speeds, overall good performance, wide range of design parameters, and moderate cost reduction when compare with active dampers. The detailed assembly view in aero engine is shown in Fig 1[a, b]. The modeling, analysis using COMSOL is explained in below chapters.



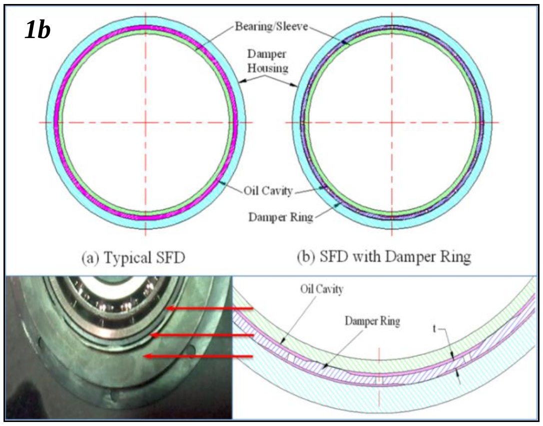


Figure: 1[a, b] Current Technology ERSFD Detailed Assembly View

2. Modeling of ERSFD

The required conceptual models (2D drawings and complex 3D models) are made with geometry parameters obtained from numerical models using modeling software. Then using this imported models, modal analysis carried out to find natural frequency of the components and used in COMSOL-CFD modeling and solid mechanics simulation software to find pressure distribution and to analyses fluid structure interactions. This 3D model of test rig small engine rotor system with dual ERSFD is illustrated in Fig. 2 with an exploded view of ERSFD components for better understanding.

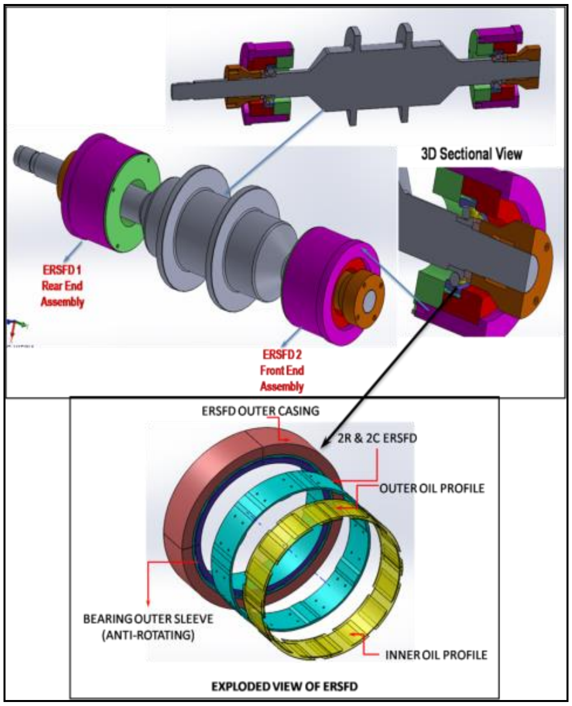


Figure: 2 3D Model of Test Rig Engine Rotor System with an Exploded Dual ERSFD

The Fig 3a shows fine mesh modelling in COMSOL for the test shaft and ring components of modal analysis. These are physics controlled (auto) **-** free tetrahedral mesh which are used in order to obtain efficient simulations when investigating different setups. Fig 3b shown, also physics controlled (auto) **-** free triangular mesh generated for FSI analysis.

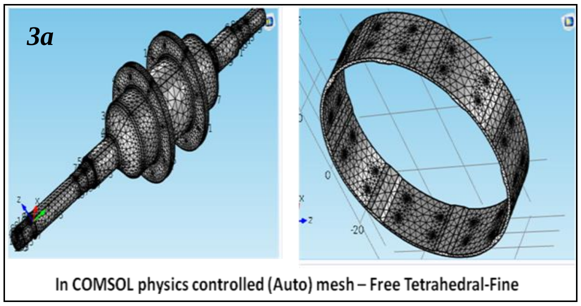
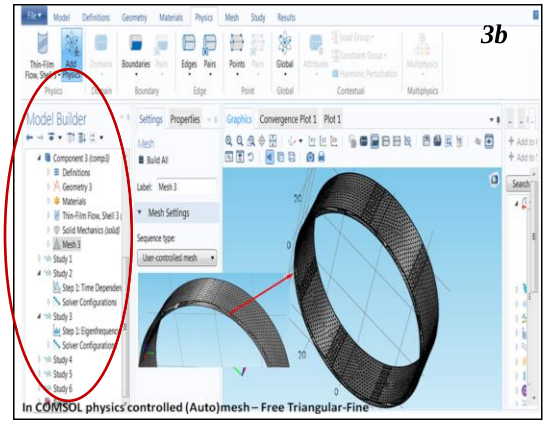


Figure: 3a Auto mesh for Modal Analysis



In COMSOL physics controlled (Auto)mesh – Free Triangular-Fine

Figure: 3b Auto mesh for FSI Analysis

In a COMSOL model builder (Fig 3b), the above shown 3D models (Fig 2) are imported into 3D geometry and materials are assigned depending on analysis type. Then physics interface of solid mechanics/CFD-FSI (tffs) is added with boundary conditions followed by physics controlled (auto) mesh formation Fig 3 [a, b]. The study such as eigenfrequency and max mode FSI are incorporated to find mode shapes on shaft, elastic rings and pressure distribution on ERSFD.

3. Computational Methods

Modal Analysis : An eigenfrequency analysis finds the eigenfrequencies and modes of deformation of test rotor and ring components. The eigenfrequencies f in the structural mechanics field are related to the eigenvalues $\boldsymbol{\lambda}$ solved through

$$f = \frac{-\lambda}{2\pi i} \qquad .....1$$

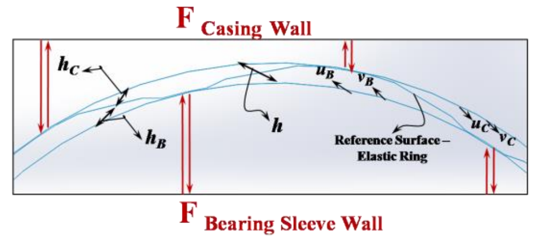
CFD/FSI (tffs) : The following modified Reynold’s equation, pressure difference and the forces on the noncontact/contact surfaces due to excitations are analytically found using below mentioned simplified equation in COMSOL solver.

$$\frac{1}{r}\frac{d}{dr}\left(\frac{r{h_0}^3}{12\mu}\frac{dp_f}{dr}\right) = V_\omega$$

modified Reynold's equation 2

Where v_w is the velocity of the wall with the boundary conditions $p_f = 1$ atm

The tff, interface is used to model the oil film on an ERSFD ring. The model is 3D since the film pressure varies radially on the pocketed oil surfaces. When Thin-Film Flow (tff) is assigned to a boundary, this boundary represents a reference surface in the ERSFD ring. In practice a small gap (≅ 300 μm) exists at the boundary and two impermeable structures, the wall and the oil cavity surfaces, are located either side of it. Fig 4 shows the configuration of the bearing sleeve and the case wall in an ERSFD problem, and describes term used in the interface. In COMSOL the edge complexity is neglected during mesh formation near narrow region heights. A region of descriptive view of the fluid structure shell model is shown in Fig 4. This annulus model is used to perform CFD-FSI (tffs) analysis with equation 2.



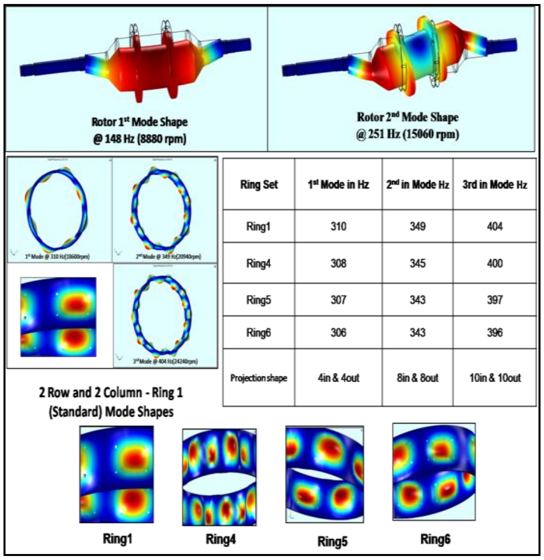
h = total size of the gap is (h_C+h_B)
h_C = height of the casing wall above the reference surface
h_B = height of the bearing sleeve wall above the reference surface
$u_{C,B}$ = displacement of the casing & bearing sleeve wall above the reference surface
$v_{C,B}$ = velocity of the casing & bearing sleeve wall above the reference surface
$F_{C,B}$ = force or energy dissipation due to pressure difference in oil

Figure: 4 Fluid Structure Shell Model

During FSI, the input variation in oil surface height (initial thickness 0.05mm) w.r.t time is varied in steps of 0.01, 0.05, 0.075, 0.1 and 0.15mm from the reference surface.

3.1 COMSOL Modal Analysis

The shaft and ERSFD ring components depicted in Fig 2 is part of a dual ERSFD bearing supported rotor system and it is subjected to both mechanical loads under centrifugal force due to unbalance mass and thermal loads due to hot oil supply. The geometry for this rotor has been created with CAD software, and it is import into COMSOL for analysis. The objective requirement of modal based frequency response analysis carried out using structural mechanics for undamped response of shaft and undamped response of the rings. The obtained results are shown in Fig. 5.



Ring Set	1st Mode in Hz	2nd in Mode Hz	3rd in Mode Hz
Ring1	310	349	404
Ring4	308	345	400
Ring5	307	343	397
Ring6	306	343	396
Projection shape	4in & 4out	8in & 8out	10in & 10out

Figure: 5 Modal Analysis Results of Shaft and Different ERSFD Ring

3.2 CFD/Fluid Structure Interaction Analysis on ERSFD Ring

This Fluid Structure Interaction (FSI) model is used to compute the damping force acting on a ERSFD ring. Also to compute pressure distribution over the entire ring during the ring subjected to high frequency response. These analyses were carried out by interaction between fluid flow and structure via CFD under thin film flow (tffs) between vibrating narrow surfaces [5]. The metallic elastic ring projected inner pedestals are in close proximity or in contact to a stationary surface of bearing outer race sleeve. Similarly the projected outer pedestals are in contact to an outer casing. The damping results from the squeezing of a thin film of oil between the two oil surfaces separated by metallic elastic ring. The squeezing action forces out the oil, between the pedestals surfaces (non-contact oil film cavities) resulting in a damping force that acts to prevent mechanical contact between the two surfaces. The opposite effect takes place when the surfaces move away from each other as oil is drawn back into the ERSFD. The above said action is called cavitation effect between bearing outer sleeve, elastic ring and outer casing.

This type of dampers has many advantages like two layers of hydrodynamic effects to dissipate energy via inner layer through flexible centering structure orifices to the outer layer. The major non-linearity issue with the ordinary SFD's are controlled via fluid structure interactions through configured flow patterns. This thin flexible metal structure contributes together with dual layered thin film hydrodynamic effects aid in smooth centering of heavy rotor system at their shaft axis during high unbalance and frequency of oscillatory condition.

4. Results

This model examines the effect of the periodic motion of the ERSFD on the flow developed, including the pressure in the oil and the resulting damping forces. Small amplitude motion is analyzed using a linear frequency domain simulation. A nonlinear transient analysis is performed for small 0.01mm to large amplitude motion 0.15mm. This result shows to hold high load carrying capacity and high damping effect under high amplitude of vibration. The Fig. 6 shows pressure intensity profile (for its respective mode shapes) also intense place of flow orifice pattern locations on annulus ring.

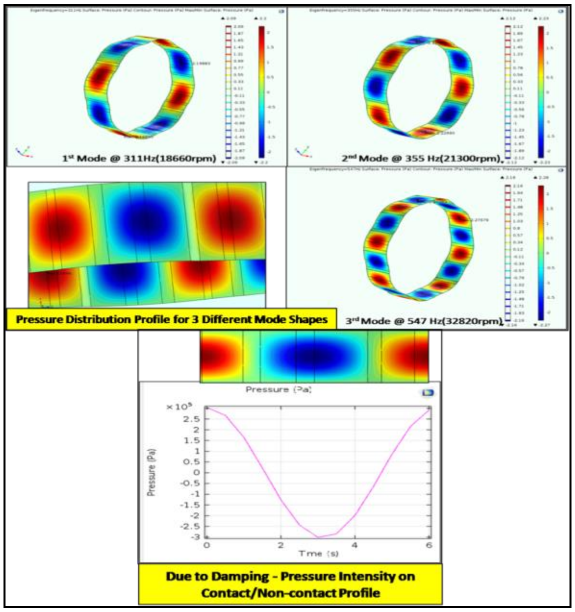


Figure: 6 FSI Analysis Result of ERSFD Ring

Above illustration shows that first mode (ω_n) is at 311Hz and has maximum and minimum displacement at four locations with pressure of +/- 2.22 bar alternatively. The second and third mode occurs at 355Hz & 547 Hz respectively with five and eight pressure locations of +/- 2.23 bar & +/- 2.28 bar. This shows influence vibration pattern on pressure distribution and damping. Also from Fig 6 it is found that at orifice locations in baseline configuration high pressure intensity is developed. It will confirm the precise orifice location for maximum damping. In general it can be said that providing orifice in proper location will support additional damping by means of pressure relief through orifice pattern like 2R-2C, 3R-2C and 2R-4C. Currently the 2R-2C has shown authenticated proof for enhancement in damping by simulation as well as with experimental results.

The pressure and load variations w.r.t time for various amplitude of vibrations (0.01, 0.05, 0.075, 0.1 & 0.15mm) are shown in Fig 7 [a, b].

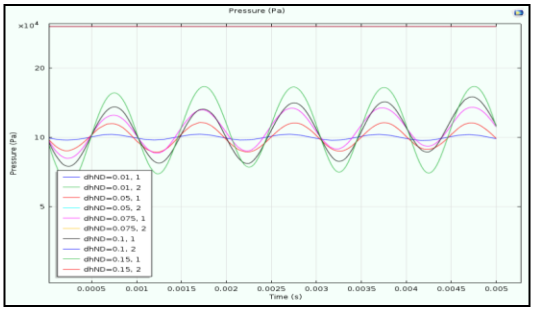


Figure: 7a Oil Film Damping Pressure vs Time for Different Values of Amplitude of Film Thickness

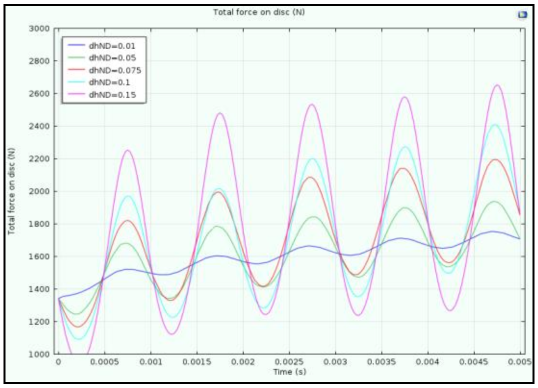


Figure: 7b Damping Load on the ERSFD vs Time for Different Values of Amplitude of Film thickness.

5. Conclusions

In this work using a COMSOL multiphysics solver, the modal and FSI on rotor and ERSFD effects were investigated. To this aim the software was applied for determining the mode shapes characterization on the test shaft and ERSFD's. The values of the alternate pressure intensity profile from max to min over the entire annular surface are numerically determined by solver. It has been shown that the numerical solutions obtained using COMSOL provides precise orifice patterns and locations. Aim of the ERSFD is to control the vibration amplitudes and reduce the transmitted forces to the base, caused by rotor imbalance. A CFD-fluid structure interaction technique was applied after structural analysis for maximum vibration reduction in the critical speed zone. FEM analysis using COMSOL provides crucial design inputs in terms of Eigen frequencies, deflection mode shapes and orifice locations to give maximum flow through openings resulting in required amount of damping.

6. References

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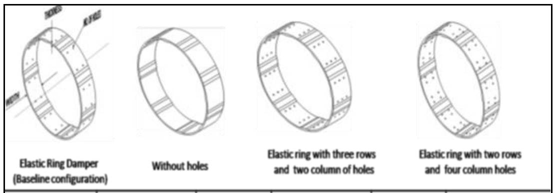
8. Appendix

Table 1: ERSFD hole pattern for Modal Analysis



Set	Thickness (mm)	No. of Rows	No. of holes per row	Width (mm)	Remarks
Ring1	0.9	2	16	15.6	Baseline
Ring4	0.9	Without hole		15.6	New configuration
Ring5	0.9	3	16	15.6	
Ring6	0.9	2	32	15.6	