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Dynamic Analysis of Multi Mass Counter Rotor System with Integral S-shaped Squeeze Film Damper

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Abstract: In order to reduce the mass, the new designs of turbojet engines use rotors working with higher speeds. These rotors are subjected to large unbalance during engine operational conditions. An additional damping might be necessary in order to enable working at rotation speeds higher than the critical ones. In order to achieve this objective, Integral S-shaped Squeeze Film Damper (ISSFD) and Multi Mass Counter Rotor (MCR) system designed and simulated. This shaft configuration consists of twin ISSFD in simply supported location and unbalance in the central disc. However, due to the rotation of the unbalance shaft system, instabilities similar to those met in classical hydrodynamic bearings can appear in ISSFD. This paper emphasis on dynamic analysis of newly designed Integral S-shaped Squeeze Film Damper as well as integrated shaft system. By using this ISSFD integrated with shaft of 25 mm diameter, critical speed is greatly increased and hence it reduces whirling of shaft during operation at high speeds. This ISSFD also takes care of misalignment about ~ 320 µm within the squeezing clearance. This method of approach can be used in the analysis of inter-shaft bearing and its influence on the double spool rotor application.

Keywords: Integral S-shaped Squeeze Film Damper, Counter Rotor, Inter Shaft, Dynamic Analysis

I. INTRODUCTION

Vibration is repetitive problem in rotating machinery where in a shaft rotates in bearings. Vibration intentionally reduced by using damping devices including an oil film in an annular space between the outer race of a bearing and the housing. When the shaft moves off the bearing axis, the oil film exerts a damping force on the shaft. These damping devices are known as “squeeze film dampers.” In practice, squeeze film dampers are used in gas turbine engines to dampen the whirling vibration of rotors. The ability of SFD's is to reduce the amplitude of engine vibrations and to decrease the magnitude of the force transmitted to the engine frame makes them perfect rotor support *[3-5]*. Also, the energy removed in the SFD's enhances the stability of the rotor bearing system.

Whirl vibration results from a number of different operational phenomenon’s. For example, synchronous whirl is caused essentially by centrifugal forces acting on a mass unbalanced shaft. The shaft is generally mass unbalanced because the geometric and inertial axes of the shaft are not identical due to machining tolerances and material imperfections, wear and tear, and in some instances due to residues on the shaft from the working fluid in a turbomachine. When the synchronous whirl frequency coincides with a natural frequency in the rotor-bearing system, the system’s vibration amplitudes can become excessive. The system is then said to be in resonance. The system's natural resonant frequency is generally referred to as its critical speed.

Another operational phenomenon is a self-excited whirl. This occurs when whirl vibration is caused by elements within the rotor-bearing system and may cause the system to become unstable. Such self-induced vibration rapidly increases in amplitude and sudden failure to the bearing system. Other causes of shaft instability include aerodynamic induced excitations which may originate from pressure variations around the circumference of impellers and seals and from material hysteresis, rubbing between rotating and stationary parts and other such activity common to rotating equipment.

The present ISSFD isolates above said critical issues in high speed rotodynamic systems. Integral S-shaped squeeze film dampers (ISSFDs) slot in elastic supports which enhance dynamic performance and reduces geometry, offer accuracy, split construction allowing easier assembly and inspection *[6]*. However ISSFDs needs wide-range of computation to verify their dynamically forced performance and testing to establish their consistency for high performance turbomachinery *[2]*. These squeeze film dampers generally can be classified into two broad categories: short dampers and long dampers. Short dampers are those dampers for which the short bearing approximation to the Reynolds equation applies. The short bearing approximation is justified if the damper is relatively short in the axial direction such that the flow in the damper is substantially axial rather than substantially circumferential. Accordingly, for short dampers the pressure gradient in the axial direction is larger than the pressure gradient in the circumferential

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direction. The aircraft industry has developed a number of effective short dampers for its wider range of use *[1, 2]*.

Long dampers are those dampers for which the long bearing approximation to the Reynolds equation applies. The long bearing approximation is justified if the damper is relatively long in the axial direction such that the flow in the damper is substantially circumferential rather than axial. Therefore, for long dampers, the pressure gradient in the circumferential direction is larger than in the axial direction. Long dampers are effective at attenuating the amplitude of vibration of the engine at the critical speed. In practice, short dampers are effective at attenuating the magnitude of the vibratory force transmitted to the engine frame at the operating speed. The present ISSFD is made as shorter type to control vibration magnitude efficiently.

II. LITERATURE SURVEY

The most commonly recurring problems in rotor dynamics are excessive steady state synchronous vibration levels and sub synchronous rotor instabilities. The first problem may be reduced by improved balancing, or by introducing modifications into the rotor-bearing system to move the system critical speeds out of the operating range, or by introducing external damping to limit peak amplitudes at traversed critical speeds. Sub synchronous rotor instabilities may be avoided by eliminating the instability mechanism, by rising the natural frequency of the rotor-bearing system as high as possible, or by introducing damping to increase the onset rotor speed of instability, Vance and Childs et al.

Lightweight, high performance engines exhibit a trend towards increased flexibility leading to a high sensitivity to imbalance with large vibration levels and reduced reliability. Squeeze film dampers (SFDs) are essential components of high-speed turbomachinery since they offer the unique advantages of dissipation of vibration energy and isolation of structural components, as well as the capability to improve the dynamic stability characteristics of inherently unstable rotor-bearing systems. SFDs are used primarily in aircraft jet engines to provide viscous damping to rolling element bearings which themselves have little or no damping. One other important application is related to high performance compressor units where SFDs are installed in series with tilting pad bearings to reduce (soften) bearing support stiffness while providing additional damping as a safety mechanism to prevent rotor dynamic instabilities. In addition, in geared compressors, the SFD assists to reduce and isolate multiple frequency excitations transmitted through the bull gear, for example San Andrés et al.

Zeidan et al. give a history of the SFD in jet engines and detail design practices for successful SFD operation in commercial turbomachinery. Adilleta and Della Pietra et al provide a comprehensive review of the relevant analytical and experimental work conducted on SFDs. San Andrés and Delgado et al discuss more recent SFD experimental research and present a mechanically sealed SFD impervious to air entrainment.

David P. Fleming et al proposed a new damper concept characterized by two oil films called as dual clearance squeeze film damper for high load conditions. Under normal conditions, in only one low-clearance film is active, allowing precise location of the shaft centerline. Under high unbalance conditions, both films are active, controlling shaft vibration in a near-optimum manner, and allowing continued operation until a safe shutdown can be made.

In order to reduce the embarked mass, the new designs of turbojet engines use softened rotors working with higher speeds. Following this trend, the use of intershaft bearings in dual-spool turbojets is then recommended for reducing weight. However, due to residual unbalances present in both rotors, the intershaft bearing is submitted to hard working conditions. A supplementary damping might then be necessary in order to enable working to rotation speeds higher than the critical ones. The adopted solution consists of introducing a thin fluid film between the intershaft bearing and one of the two rotors in order to provide the necessary damping. However, due to the rotation of the inner and of the outer rings, fluid film instabilities similar to those met in classical hydrodynamic bearings can appear. Experimental investigation of several designs of the intershaft bearing damper is carried out and of their influence on the dynamic behaviour of a double spool rotor is studied C. Defaye et al.

In spite of the many successful applications, industry often recognizes that the design of SFDs is based on overly simplified predictive models that either fail to incorporate or simply neglect unique features (structural and fluidic) that affect the damper dynamic force performance. Actual damper performance can range from erratic to non-functioning depending on the operating conditions. Issues such as lubricant cavitations or air entrainment are of fundamental interest, San Andrés and Diaz et al.

Flexible rotors operating in high speed and pressure application are particularly at risk to rotordynamic instabilities and high amplification factors through critical speed changeover. By using ISFDs in the rotor system the ratio of energy transmitted to the bearing location is maximized and therefore can drastically improve the damping ratio of system, Bugra Ertas et al.

III. TYPICAL ISSFD FOR MULTI MASS ROTOR

The ISSFD design can be manufactured through electrical discharge machining (EDM). Integral “S” shape springs connect an outer and inner ring with 0.5mm cap throughout the slotted profile *[4]*, and a squeeze film damper land extends between each set of springs *[1]*. Bearing pads are housed in the inner ring (Figure 1; Images 1&2). The unique design allows for high-precision control of concentricity, stiffness and rotor positioning, and it produces superior damping effectiveness by separating stiffness from damping *[2]*.

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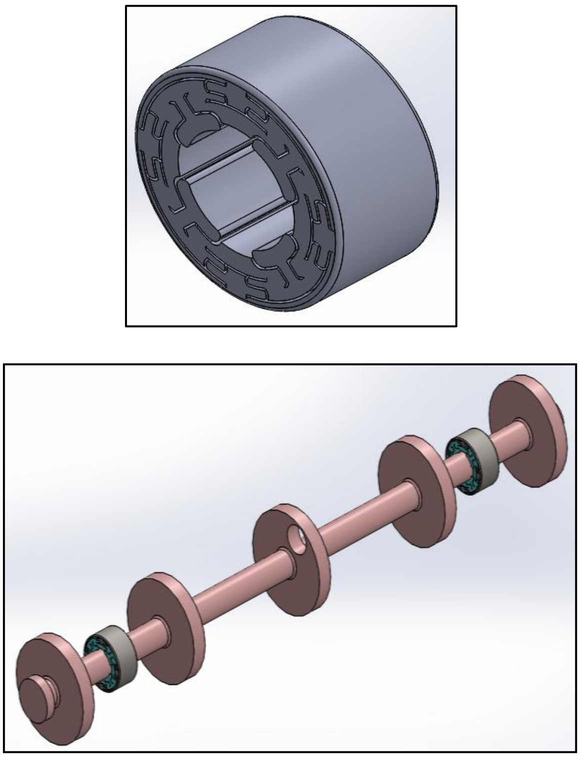

Fig. 1. SSFD and Multi Mass Flexible Rotor

IV. INTEGRATED ISSFD FEATURES

1. Centre Under Static Load

The present design centers the rotor under static load. It overcomes the deficiencies of O-ring dampers by providing centering without sag, linear stiffness unaffected by temperature or age, and acceptance of high radial loads.

2. Improve Stability

By introducing flexibility into the rotor/bearing system and providing optimum damping, ISSFD technology maximizes the energy dissipation at the bearing locations and significantly improves the stability of the system.

3. Shift Critical Speeds and Reduce Amplification Factor

ISSFD shift critical speeds and significantly reduce the amplification factor. With the reduction in amplification factor, machine seal clearances can be tightened to reduce gas or steam leakage.

4. Reduce Dynamic Bearing (Transmitting) Forces

ISSFD reduces the dynamic load that is transmitted to the bearings, which reduces pedestal vibration and increases bearing life, particularly for rolling element bearings. For fluid film bearings, the technology can mitigate pivot wear and reduce babbitt fatigue.

5. Decrease Unbalance Sensitivity

ISSFD helps in reducing the sensitivity to unbalance, protecting impellers and seals from rubbing and increasing maintenance intervals.

V. MODELLING AND SIMULATION OF SSFD

The 3D modelling of SSFD and MFS has been made using SolidWorks. ISSFD has been simulated using SolidWorks simulation tool. The geometry SSFD and MRS model has a significant influence in its performance therefore need of adopting the standard procedure to model. This computer graphics software is used for modelling, design and simulation. Fig. 2 Shows 3D solid mesh of SSFD with finite element for analysis. Table 1 Model properties.

Total Nodes	30796
Total Elements	16383

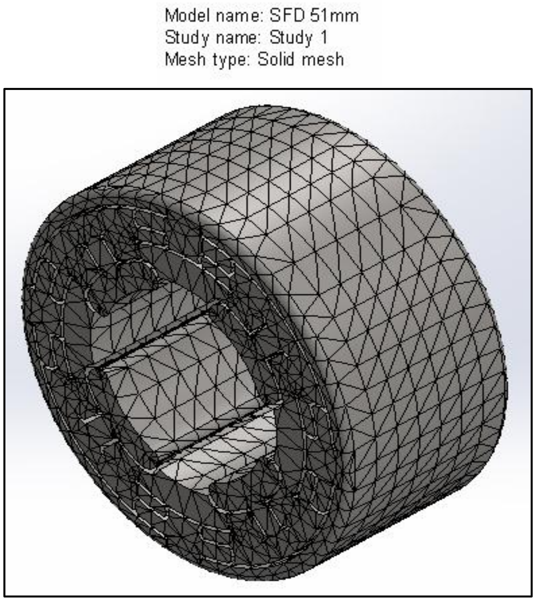

Fig. 2 SSFD Model with Solid Mesh

TABLE 1
MODEL PROPERTY

Model Properties	
Name	Alloy Steel
Model type	Linear Elastic Isotropic
Default failure criterion	Max von Mises Stress
Yield strength	620.422 N/mm^2
Tensile strength	723.826 N/mm^2
Elastic modulus	210000 N/mm^2
Poisson's ratio	0.28
Mass density	7850 g/cm^3
Shear modulus	79000 N/mm^2
Thermal expansion coefficient	1.3e-005 /Kelvin

VI. SSFD FEM ANALYSIS

FEM analysis was carried out in a SSFD to find dynamic load carrying capacity and flexibility condition under 1000 N load. The load was applied on to the 4 leg pads of the SSFD and outer ring made as a fixed ring boundary condition. The FEM results shows (Fig. 3) maximum displacement/flexibility about 12µm and von mises stress in maximum range of about 20.32 MPa at slotted corners of the SFD profile. The computed result yields SSFD system in application of force direction

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displacement as shown in Fig. 3. The displacement obtained by the FE calculation in the application of force direction for 0. 01206 mm, were obtained in the SSFD ring by rigidity calculation formula is

$K_r = F_r / \delta_r = 1000$ N / 0. 01206 mm = 82.92 x10^3 N/mm (1)
$K_a = F_a / \delta_a = 1000$ N / 0. 250 mm = 4 x10^3 N/mm (2)

The FEM result shown stiffness of SSFD is about 82.92 kN/mm which is 95% higher stiffer *[4]* than the allowable displacement limit of 250μm.

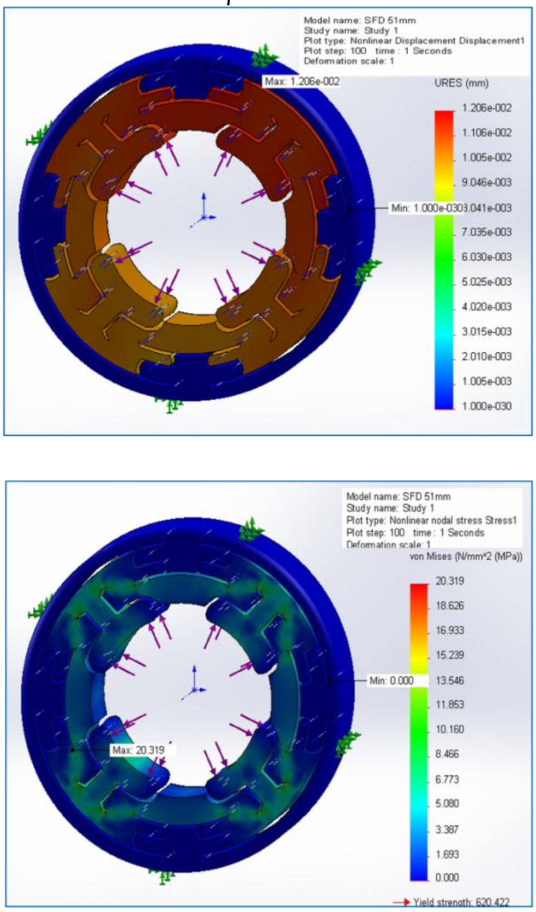

Fig. 3 Results of Displacement and Von Mises Stress

VII. ISSFD INTEGRATED SHAFT CONFIGURATION (MODEL AND ANALYSIS)

The required ISSFD with multi mass rotor system model is made to do integrated simulation as shown in Fig. 4. Integrated simulation of MCR with newly designed ISSFD *[4, 5]* is carried out in SolidWorks Motion tool in order to find out critical speed of the system and its displacement amplitudes.

VIII. GENERAL MATHEMATICAL APPROACH FOR CRITICAL SPEED ANALYSIS

8.0 Critical Speed of Shaft System

All rotating shaft, even in the absence of external load, deflect during rotation. The combined weight of a shaft and wheel can cause deflection that will create resonant vibration at certain speeds, known as critical speed. The magnitude of deflection depends upon the followings

a. Stiffness of the shaft and it's support
b. Total mass of shaft and attached parts
c. Unbalance mass with respect to the axis of rotation
d. The amount of damping in the system

Therefore, the calculation of critical speed (N_C) for MCR is necessary.

8.1.1 Critical Speed Equation

Two methods were used to calculate critical speed, Rayleigh-Ritz and Dunkerley equation. Both the equations are approximation to the first natural frequency of vibration, which is assumed to be nearly equal to the critical speed of rotation. In general, the Rayleigh-Ritz equation overestimates and the Dunkerley equation underestimate the natural frequency. The equation illustrated below is the Rayleigh-Ritz equation (Ref: KURGER); good practice suggests that the maximum operation speed should not exceed 75% of the critical speed.

$$N_C = \frac{30}{\pi}\sqrt{\frac{g}{\delta_{st}}}\ rpm \quad (3)$$

where g is acceleration of gravity (9.81 m/s^2) and δ_{st} is total maximum static deflection. Critical speed depends upon the magnitude or location of the load or load carried by the shaft, the length of the shaft, its diameter and the kind of bearing support.

8.1.2 Critical Speed of Long Span MCR

(a) Deflection from shaft weight (δ_{st1})

$$\delta_{st1} = \frac{5WL^3}{384EI} = 1.99 \text{ X } 10^{-6} \text{ m} \quad (4)$$

(b) Deflection from load only (δ_{st2})

$$\delta_{st2} = \frac{WA(3L^2-4A^2)}{24EI} = 4.338 \text{ X } 10^{-6}\text{m} \quad (5)$$

(c) Total maximum static deflection (δ_{st})

$$\delta_{st} = \delta_{st1}+\delta_{st2} = 1.99 \text{ X } 10^{-6}+4.338 \text{ X } 10^{-6} = 6.37 \text{ X } 10^{-6} \text{ m} \quad (6)$$

from Eqs. 3 safety factor 25%, therefore critical speed for long span (N_C) / maximum operation speed = 11848 rpm x 0.75 = 8886 rpm

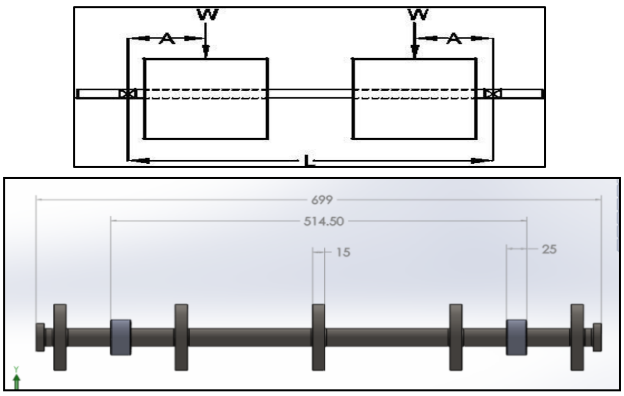

Fig. 4 Multi Mass Rotor System with twin ISSFD configuration

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where , w is weight of shaft in 4.7 kg, W is weight of wheel in 2.53 kg, E is modulus of elasticity for ANSI TYPE A2 steel is 200 $\times 10^8$ kg/m^2, I is moment of inertia ($\pi D^4/64$) in m^4 and L is length of shaft in 0.49 m.

8.2 Lumped Mass / Rayleigh's Method (Ref: W. T. Thomson)

This method used to find natural frequency of lateral vibration of shaft carrying several masses and extended to find the approximate value of the fundamental natural frequency of a discrete system as shown in fig. 5. Fundamental natural frequency (ω) is given by

$$\omega = \sqrt{\frac{g(m_1 w_1 + m_2 w_2 + m_3 w_3)}{m_1 w_1^2 + m_2 w_2^2 + m_3 w_3^2}} \qquad (7)$$

By strength of material concept deflection of shaft w(x) is given by

$$w(x) = \begin{cases} \frac{Pbx}{6EIl}(l^2 - b^2 - x^2); & 0 \le x \le a \\ -\frac{Pa(l-x)}{6EIl}(a^2 + x^2 - 2lx); & a \le x \le l \end{cases} \qquad (8)$$

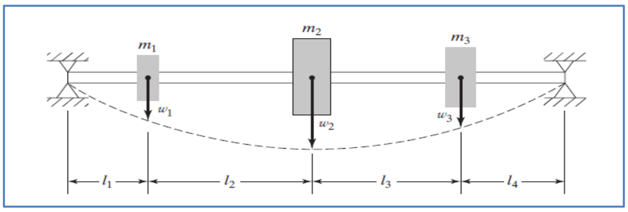

Fig 5. Shaft carrying MCR

Disc diameter=0.1m, disc thickness=0.015m, shaft diameter=0.025m, total length of shaft=0.49m. Assuming steel material density as 7850 kg/m^3, E=210Gpa and I= $(\pi d^4/64)=1.9174*10^{-8}$, Mass of the disc=0.925kg and mass of the shaft=1.925kg. Therefore m_1=0.925kg, m_2=2.85kg and m_3=0.925kg. Also $L_1=L_2=L_3=L_4$=0.123m. Substituting above in Eqs.7 & 8 we get ω=600 rad/s. Speed at which natural frequency occurs i.e. critical speed=600*60/2π= 5730 rpm

IX. COMPUTATIONAL ANALYSIS / RESULTS

The considered ISSFD configuration with MR system is designed, modelled and simulated its static and dynamic conditions in add on SolidWork Simulation and Motion software. This work emphasis in finding system natural frequency (critical speed) and amplitude variations in ISSFD shaft system *[3]*. This is new approach shows controlled displacement amplitude and shift in critical speed by using newly designed ISSFD. The solid model, controlled displacement and its critical speed is shown in Fig. 6 (Images 1 to 3).

9.1 Results

This new approach shows improvement in critical speed of the system about 15100 rpm by using integrated squeeze film damper which is ~70% more. This ISSFD can able to self align for deviation of 320 µm. Small displaced shaft centre and critical speed increment using ISSFD *[6]* with unbalance mass of 67 g at eccentricity radius 32 mm shown in Fig.6 (Images 1 to 3). Results of continuous run (0-40000 rpm) shown in Fig.7 (Images 1 & 2). Displacement analysis for different speeds shown in Fig.8 (Images 1 to 5). Enhancement in critical speed given in Table. 2

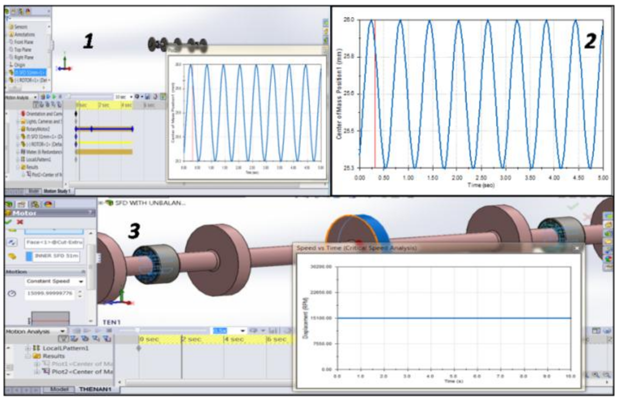

Fig 6. 3D MCR model, controlled displacement and critical speed

TABLE 2
CRITICAL SPEEDS BY DIFFERENT METHODS

Sl. No	Methods	Critical Speed in rpm
1	Lumped Mass (Ref: W. T. Thomson)	5730
2	Rayleigh - Ritz (Ref: KURGER)	8886
3	Computational Analysis with ISSFD	15100

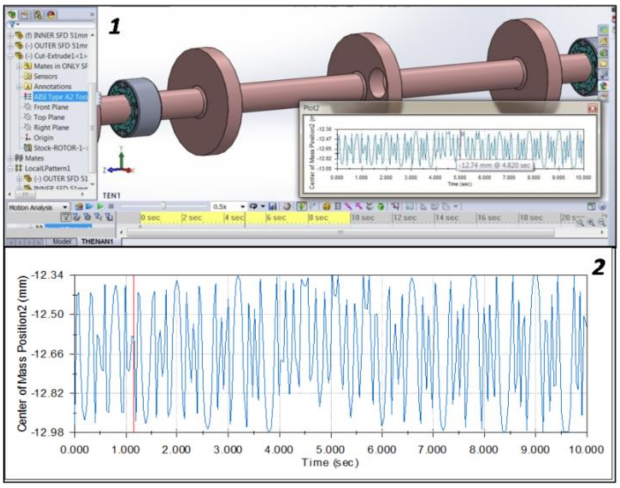

Fig 7. Results of continuous run (0-40000 rpm)

X. CONCLUSIONS

The newly considered complicated ISSFD type (ID: 25.4 mm, OD: 51 mm) and MCR is modelled using SolidWorks and simulated using motion analysis software. By using this ISSFD integrated with shaft of 25 mm diameter, critical speed is greatly increased and hence it reduces whirling of shaft during operation at high speeds. This ISSFD take care misalignment about ~ 320 µm. This design and dynamic simulation method can be adapted to reduce high vibrational amplitudes and enhance good support damping of aero engine multimass rotor system during high speed changeovers. The future work may be continued to analysis squeeze profile and its thickness. This method of approach can be used in the analysis of inter-shaft bearing and its influence on the double spool rotor application.

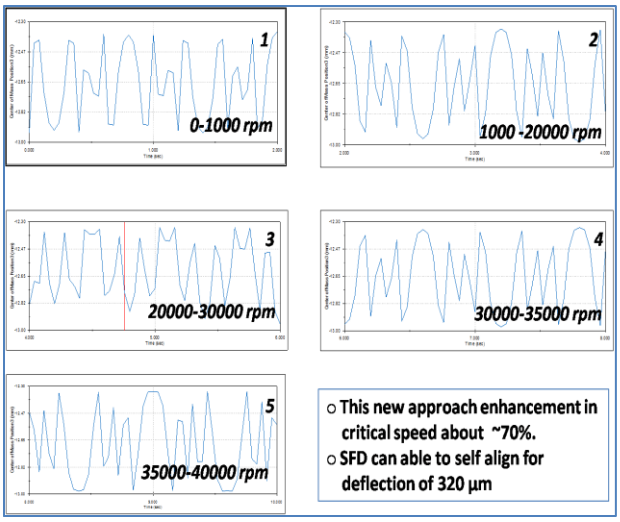

Fig 8. Displacement analysis for different speeds

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BIOGRAPHY

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