

Rotodynamic Analysis of High Speed Rotor with Integral S-shaped Squeeze Film Damper

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ABSTRACT: In order to spin at high speeds with eccentric heavy loads, the new designs of turbojet engines use rotor system working with integrated dampers. These rotors are always subjected to large unbalance during engine operational conditions. An inventive 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 Disc Flexi Rotor (MDFR) system designed and simulated. This shaft configuration consists of identical 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 ISSFD as well as integrated shaft system. By using current ISSFD integral with shaft of one inch diameter, critical speed is greatly increased and hence it reduces whirling of shaft during high speed operation. This ISSFD also takes care of greater misalignment within the squeezing clearance. This method of approach can be used in the analysis of high speed inter-shaft bearing and its influence on the twin spool rotor application.

KEYWORDS: Integral S-shaped Squeeze Film Damper, High Speed Rotor, Inter Shaft, Rotodynamic Analysis

I. INTRODUCTION

Vibration is repetitive problem in rotating machinery where the 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. 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 phenomenons. 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

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

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. However ISSFDs needs wide-range of computation to verify their dynamically forced performance and testing to establish their consistency for high performance turbomachinery. 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 direction. The aircraft industry has developed a number of effective short dampers for its wider range of use.

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. RELATED WORK TO CONTROL INSTABILITIES IN ROTORS

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 [1988], Childs [1993]. 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, [2002]. Zeidan et al. [1996] give a history of the SFD in jet engines and detail design practices for successful SFD operation in commercial turbomachinery. Adilleta and Della Pietra [2002] provide a comprehensive review of the relevant analytical and experimental work conducted on SFDs. San Andrés and Delgado [2007] discuss more recent SFD experimental research and present a mechanically sealed SFD impervious to air entrainment.

David P. Fleming [1986] 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 centreline. 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

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(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

weight. However, due to residual unbalances present in both rotors, the intershaft bearing is submitted to hard working conditions. A supplementary damping might 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, [2008]. 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, [2003].

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 ISSFDs 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, 2015].

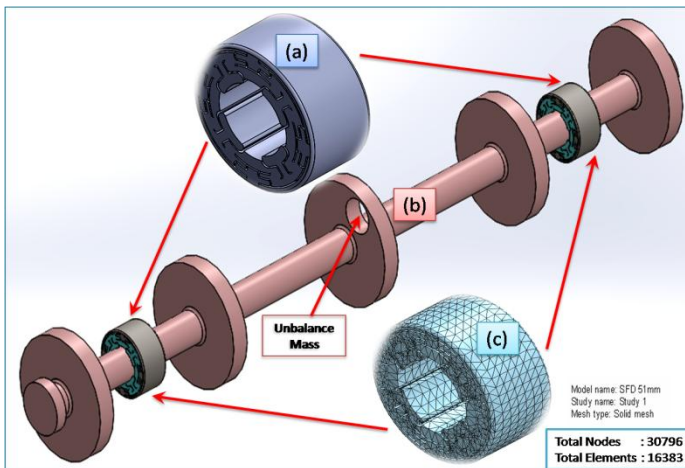
III. SPECIAL FEATURES OF ISSFD INTEGRATED WITH HIGH SPEED ROTOR

1. *Centre Under Static Load* : The present design centers the rotor under static load. It overcomes the deficiencies of O-ring dampers by providing centring 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.

IV. TYPICAL ISSFD MODELLING FOR MULTI DISC ROTOR

The ISSFD design can be manufactured through electrical discharge machining (EDM) for practical use. Integral “S” shape springs connect an outer and inner ring with 0.5mm cap throughout the slotted profile, and a squeeze film damper land extends between each set of springs. Bearing pads are housed in the inner ring as shown in Fig. 1a and twin ISSFD fits with simply supported location on central disc unbalanced MDRF which is shown in Fig. 1b. 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.

The 3D modelling of SSFD and MDRF has been made using SolidWorks. ISSFD has been simulated using SolidWorks simulation tool. These geometry models have 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. 1c shows 3D solid mesh of SSFD with finite element for analysis. Table 1 Shows model property which is used in FEM analysis.



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

Fig. 1 SSFD and Multi Disc Flexible Rotor (a) SSFD solid model (b) Integral ISSFD with unbalanced MDRF (c) SSFD 3D solid mesh and Table. 1 Model properties for FEM analysis

V. 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 maximum displacement/flexibility about 12µm as well as 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 displacement as shown in Fig. 3a and stress levels as in Fig. 3b. 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

$$K_r = F_r / \delta_r = 1000 \text{ N} / 0. 01206 \text{ mm} = 82.92 \times 10^3 \text{ N/mm} \quad (1)$$

$$K_a = F_a / \delta_a = 1000 \text{ N} / 0. 250 \text{ mm} = 4 \times 10^3 \text{ N/mm} \quad (2)$$

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

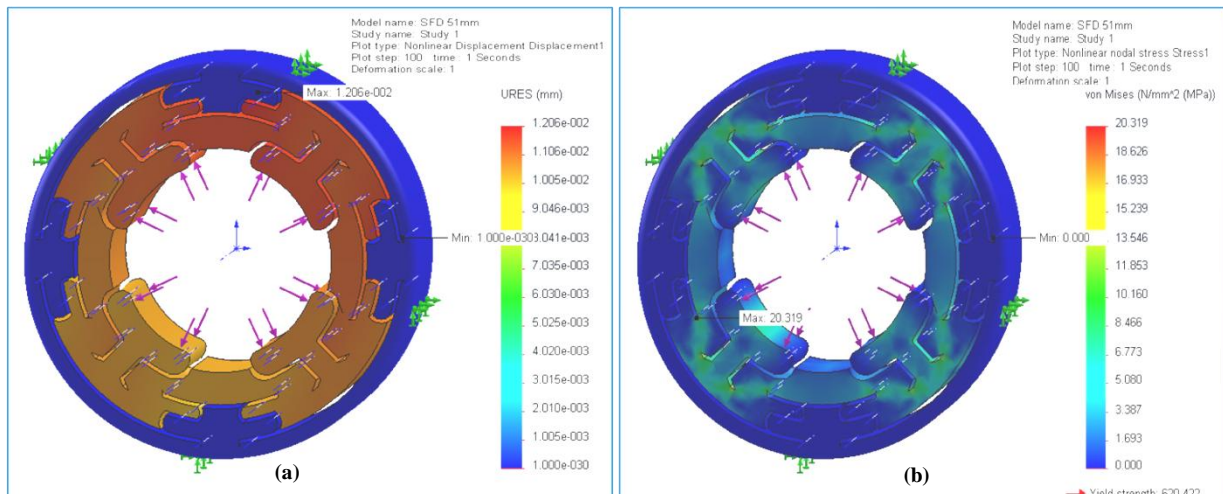


Fig. 3 Results of Displacement and Von Mises Stress (a) Application of force direction displacement (b) Stress levels

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

VI. GENERAL MATHEMATICAL APPROACH FOR CRITICAL SPEED ANALYSIS

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 critical parameters

- (a) Stiffness of the shaft and it's support
- (b) Total mass of shaft and attached parts
- (c) Unbalance of the 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 MDFR is necessary.

Critical Speed Equation : The popular two methods applies 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.

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

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.

Critical Speed of Long Span Rotor :

(a) Deflection from shaft weight $\delta_{st1} = \frac{5WL^3}{384EI} = 1.99 \times 10^{-6} \text{ m} \quad (4)$

(b) Deflection from load only $\delta_{st2} = \frac{WA(3L^2-4A^2)}{24EI} = 4.338 \times 10^{-6} \text{ m} \quad (5)$

(c) Total maximum static deflection $\delta_{st} = \delta_{st1} + \delta_{st2} = 1.99 \times 10^{-6} + 4.338 \times 10^{-6} = 6.37 \times 10^{-6} \text{ m} \quad (6)$

from Eqs. 3 Safety Factor 25%, therefore critical speed for long span (N_C) / maximum operation speed = $11848 \text{ rpm} \times 0.75 = 8886 \text{ rpm}$. The schematic view of ISSFD with self centred load (w) acting on equispaced disc rotor as shown in Fig. 4a.

VII. INTEGRATED ISSFD WITH MDFR CONFIGURATION

The required ISSFD with multi disc rotor system is made to do integrated simulation. This will facilitate rotodynamics on multi disc rotor with central unbalance mass systems displacement and synchronous whirl motion. The Fig. 4b shows detailed dimensions of disc locations connected with ISSFD. Integrated simulation of MDFR with newly designed ISSFD were carried out in SolidWorks Motion tool in order to find out critical speed of the system and its displacement amplitudes.

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 $200 \times 10^8 \text{ kg/m}^2$, I is moment of inertia in m^4 and L is length of shaft in 0.49 m .

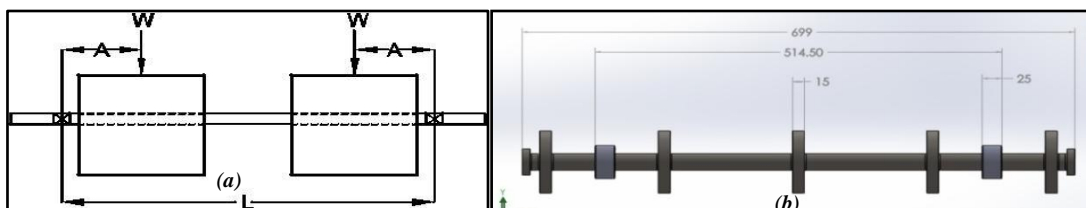


Fig. 4 MDR System with ISSFD configuration (a) Schematic view (b) Disc locations with ISSFD

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

Lumped Mass / Rayleigh's Method (Ref:- W. T. Thomson) : This method used to find natural frequency of lateral vibration of shaft carrying several masses (m_1, m_2, m_3) are extend to find the anticipated value of the fundamental natural frequency at discrete location deflections are w_1, w_2, w_3 in rotor system are given in Fig. 5.

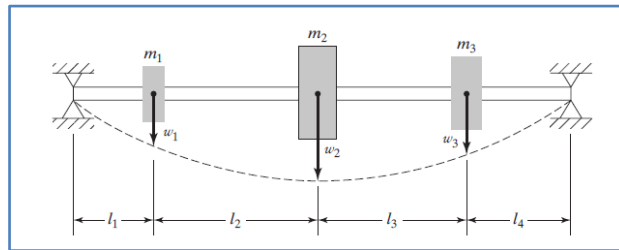


Fig 5. Rotor deflection due to multi disc at discrete locations

Fundamental natural frequency (ω) is given by

$$\omega = \sqrt{\frac{g(m_1w_1+m_2w_2+m_3w_3)}{m_1w_1^2+m_2w_2^2+m_3w_3^2}} \quad (7)$$

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

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

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/m³, E = 210Gpa and $I(\pi d^4/64) = 1.9174 \cdot 10^{-8}$, Mass of the disc = 0.925kg and mass of the shaft = 1.925kg. Therefore $m_1=0.925$ kg, $m_2=2.85$ kg and $m_3 = 0.925$ kg. Also $L_1=L_2=L_3=L_4 = 0.123$ m. w_1, w_2, w_3 are the deflection of the shaft at different locations found using Eqs. 8. Substituting above all parameters in Eq.7 we get $\omega = 600$ rad/s. Speed at which natural frequency occurs i.e. critical speed = $(600 \cdot 60) / 2\pi = 5730$ rpm

VIII. SIMULATION OF ISSFD WITH MDFR

The innovative 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 new approach shown in Fig. 6a results in controlled dislocation amplitude and shift in critical speed (15100 rpm) by using newly designed ISSFD. The whole solid model assembly simulated at high speed results in harmonic excitation are shown in Fig. 6b.

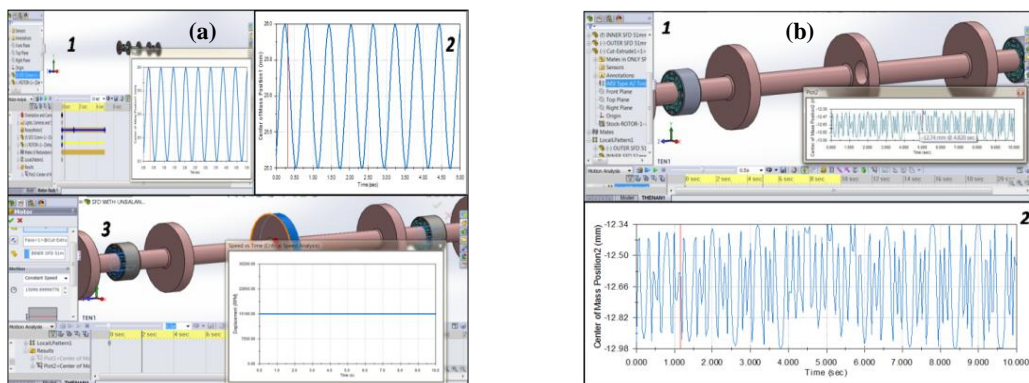


Fig 6. Simulation on whole rotor assembly (a) Critical speed attain at 15.1Krpm (b) Harmonic excitation at high speed

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

IX. COMPARATIVE RESULTS

This innovative new approach shows enhancement 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 maximum deviation of 320 μm which is resulted from simulation. Very less displacement and critical speed increment using ISSFD with unbalance mass of 67 g at eccentricity radius 32 mm shown in Fig.6 (a-b). Results of continuous run (0-40000 rpm) is shown in Fig. 6b which shows controlled harmonic excitation within the max displacement limit. Variants in displaced amplitude is analysed at different speeds from simulation outputs are shown in Fig. 8 (a - e). Enhancement in critical speed is compared with other methods are given in Table. 2. The ISSFD simulated with MDRF is significantly shifted its critical speed about 70%. Therefore this ISSFD is effective in highly unbalance with multi disc overloaded situation.

Sl. No	Methods	Critical Speed in rpm
1	Long Span	5730
2	Rayleigh - Ritz	8886
3	Simulated with Innovative ISSFD	15100

Table . 2 Critical Speed for different methods

The variation in harmonic excitation at various speed is shown below. The speed at 0-10K rpm the no of fluctuations are more and band width is less as shown in Fig. 8a. Unsteady state is prominent at this lower speed rate. On the other hand as set of speed increases the fluctuation rate reduces and steady state band width increases. By means of ISSFD steady state is prominent at high speed rates which is shown in Fig. 8 (b - e). This phenomenon take place because of newly integrated ISSFD which controls and stabilise the unsteady occurrence within the displacement limit of 320 μm . This shows that the ISSFD dynamically self aligns and controls the uncertainty during high speed operation starts from 30-40K rpm as shown in Fig. 8e.

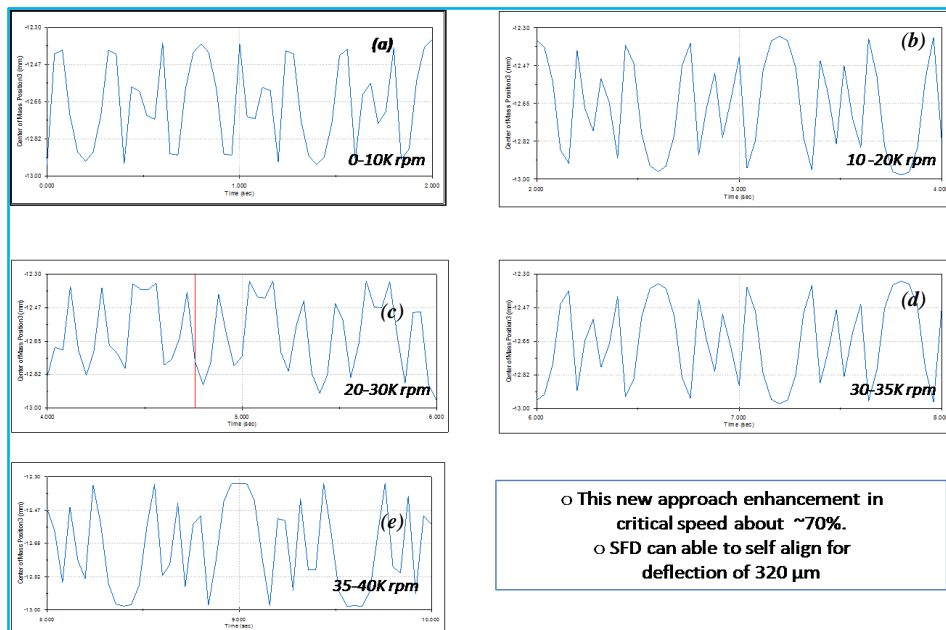


Fig 8. Displacement analysis at different speeds (a) More instable bandwidth at lower speed 0-10K rpm (b - e) More steady bandwidth attains at high speed rate starts from 10-20, 20-30, 30-35 and 35-40K rpm.

International Journal of Innovative Research in Science, Engineering and Technology

(An ISO 3297: 2007 Certified Organization)

Vol. 4, Issue 7, July 2015

X. CONCLUSIONS

The innovative complex profile ISSFD type (ID: 25.4 mm, OD: 51 mm) and MDFR 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 $\sim 320 \mu\text{m}$. This design and dynamic simulation method can be adapted to reduce high vibrational amplitudes and enhance good support damping of aero engine multi disc rotor system during high speed changeovers. The future work may be continued to analysis squeeze complex 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.

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