Rotordynamic Analysis of Bolted Disk-Drum Rotor with Contact Nonlinearity

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ABSTRACT: This paper investigates the non-linear behavior of bolted disk-drum joint in a rotor subjected to bending loads during operation, using Finite element analysis (ANSYS- Workbench). The variation in contact area due to these bending loads results in non-linear deformation.The non-linear behavior of bending stiffness is studied from bending moment versus deflection plot obtained under static condition for different bolt preloads using ANSYS Workbench. The results predicted by finite element simulation are validated analytically using MATLAB. Considering the non-linear behavior of bending stiffness, rotor dynamic analysis of bolted and without bolted (continuous) disk-drum rotor is performed. For any given operational speed range, by varying the number of bolts under the condition of with and without bolted joints, the modal analysis is carried out to study the effect of bolted joints on modal frequencies of the disk-drum rotor bearing system. These responses are validated analytically using MATLAB. Also to study the effect of bolted joints on critical speeds, the Campbell diagram is obtained under the condition of with and without bolted joints for different number of bolts. Finally Harmonic analysis is carried out to determine the rotor whirl amplitude at critical speeds obtained for with and without bolted joint.

KEYWORDS -Bolted joint, Campbell diagram, Harmonic excitation, Nonlinear, Rotordynamic analysis, Unbalance response.

I. INTRODUCTION

The introduction of the paper should explain the nature of the problem, A Gas Turbine (GT) engine consists of various components and these are grouped accordingly based on their functionality. The rotating shafts, also known as rotors, are used to transmit the energy from the exhaust gases in downstream of turbine region to the upstream compressor region of the engine. The rotors also transmit power to other auxiliary components needed for functioning of the engine. These rotors are held intact by bearings at specified locations to isolate with the static parts and maintain stability under extreme operating conditions.

Bolted joints are frequently used in gas turbine engines and its structures to remove the complexity in the architecture of the engine. The most common application is to connect different rotor disc to the drum by means of bolted joints.The frictional contact at the joint interface induces nonlinear behavior of the bending stiffness in a bolted disk-drum joint configuration, which force these drums to flex in and flex out under the operating condition. The friction at contact interface in the joint inducing nonlinearity is more of a practical phenomenon and is very significant in rotor dynamic analysis during the design of rotor system. Generally effect of nonlinearity is neglected during such an analysis to keep things simpler. A simplified analysis disregarding the nonlinearity will yield erroneous results which predict an oversimplified behavior. Also the change in mating contact area and rigidity would vary with the operating cycle and excessive movement due to loading and offloading conditions of the joint structure could also cause severe wear and catastrophic failures.

Determining the joint stiffness and studying the dynamic behavior under such conditions is of biggest concern in accurate design of rotor system. To find a solution for the above quoted problem statement; an attempt is to be made for studying the nonlinear behavior of the bending stiffness. With the need for cost effectiveness and life expectancy, dynamic analysis should be performed to study the effect of such a bolted

joint on dynamic behavior of rotor-bearing system and compare the findings with the results obtained from the same rotor-bearing system without a bolted joint.

This present study is put forward to predict the dynamic characteristics of the rotor system such as determining the critical speeds, harmonic responses for a given operating conditions. These responses are obtained for with bolted and without bolted conditions and are correlated to understand the dynamic behavior.

1.1 Rotordynamics

Rotordynamics deals with study of mechanical devices consisting of rotating parts, also known as rotors, rotating with significant angular momentum about a fixed axis. In rotordynamics, resonant condition is said to be attained as the system frequency equals the rotational frequency of the rotor caused by unbalance excitation. This particular rotational speed is called critical speed. The amplitude of the rotor vibration exhibits a peak in the vicinity of the critical speed. Thus the critical speed assists in determining the rotor rigidity and flexibility conditions. Hence the rotordynamic study intensifies the understanding of rotor vibration phenomena and assists in eliminating the undesirable vibrations and improve the efficiency of the rotating machinery.

Rotordynamic analysis is not merely carried out during design stage, but also essentially used in testing and during actual operation of the machine to obtain the behavior of the machinery for preventive maintenance to identify the problematic mechanical signature, i.e., the vibration spectrum.

1.2 Nonlinearity in bolted joints

The rigidity of any joint structures and its contact state influence the rotor dynamic behaviour and also the responses due to mass unbalance. The joint stiffness is not constant at lower values of external loads and behaves nonlinearly up to the critical transition loads. When the external load exceeds these transition load, the stiffness value will increase and slightly attains a constant value, linearly [1]. The rigidity will vary as the external loads changes and affect dynamics of the joint structure.This behaviour is influenced from the contact state at the joint interface and bolt preloads. The contact state varies with the geometry and the loading condition.

In dynamic conditions, where the parts are in motion, the forces and the motion are transferred by means of contact. Hence it is important to consider the force distribution, related deformation and friction at contact before carrying out any analysis. This contact also behaves like a geometrical constraint restricting the displacement of the body. In general, the contact surface is a variable function of the load applied and the contact surface remains constant only in the case of stationary contact. Considering the theoretical aspect of contact condition, the correlations between the normal components and the tangential components are different. Nonlinearity due to contact arises whenever contact or mating is present, because, in most of the cases, the geometry of the contact zone is unknown. The nonlinear behavior depends on the contact status. The status might change directly with the load or might be determined by several external causes. When the frictional forces are considered, the contact problem becomes specially complicated due to dissipation of energy.The presence of frictional slippage at the interface dissipates the energy relative to the contact state in the joint interface.

The concept of nonlinearity is more of a practical concern because the displacements behave as a nonlinear function of the applied loads. In a theoretical way of solving such a problem many assumptions are made to disregard the nonlinearity and simplify the solution process. The analysis of tangential components increases the complications and requires exhaustive inspection. Eventually, FE methods involving comprehensive numerical techniques are required to solve the contact problems.

II. RELATED WORK

The presence of bolted joints in any structure will certainly influence the overall stiffness of the system. The frictional contact that exists between the joined components will be continuously varying along with the mating contact area during the operating cycle. Hence determining the joint stiffness and implementing the same during the dynamic analysis of the rotor system will be a more realistic approach of solving the problems pertaining to vibration in rotor systems [1].

The stiffness of the joined structure will be a function of number of bolts as well as the pre-tension effect unveiled by each bolt. Such a study on various parameters influencing the joint stiffness along with its nonlinear behavioral study using FE method and experimental approach yields feasible results. Also sensitivity analysis of mode shapes and critical speeds using Finite Difference Methods reveals the variation of joint stiffness proportional to externally applied loads and the geometrical configuration affecting the dynamic behavior [2].

The integrity of any gas turbine can be appropriately assessed by evaluating its dynamic behavior. This is a highly complex scenario that prompts the use of FE methods. ANSYS program developed on FE platform reduces the complexity in determining the dynamic behavior. The natural frequencies and associated critical speeds are determined by developing the Campbell diagram. ANSYS provides different environment such as APDL, ANSYS Workbench etc. through which similar results can be obtained [3].

Rotordynamic study involves both qualitative and quantitative study of the rotor-bearing system. When a rotor body rotates with significant angular momentum, resonance conditions are achieved at particular speeds called critical speeds. These critical speeds are found out by a plot of Natural frequency vs. Rotational speed. This becomes a complex phenomenon for complex shaped rotors. Hence simplified rotors are such as Jeffcott rotors can be widely used to understand the dynamics of rotating structures. Eventually Jeffcott rotors fail to provide a precise behavior quantitatively, but explain the qualitative behavior in an effective manner for an oversimplified rotor model than for a realistic rotor [4].

In the Finite element method to determine the dynamic behavior analytically, involves formulating the problem based on Euler-Bernoulli theory of beams considering 2-D beam element. The rotor is discretized into these beam elements. The rotation of the beam induces the gyroscopic effect which is a major concern in rotor dynamics. Determining the global Stiffness matrix, gyroscopic matrix & Mass matrix and solving the governing dynamic equation considering the unbalance mass as the excitation force will provide the harmonic behavior the discretized rotor system [5].

Nonlinearity is more of a practical and complex scenario. Hence for simplifying the analysis process the effects of nonlinearity is generally neglected. The nonlinearity problem can have many sources such as Material nonlinearity where in the law of elasticity is invalid, Geometric nonlinearity arising from large displacement and strain values, and nonlinearity due to the presence of contact zone which is generally unknown. Also the Frictional contact exhibits more complexity as the energy is dissipated [6].

III. ANALYTICAL MODELING OF BOLTED DISK-DRUM JOINT

Reference [1] is used for analytical modeling of the bolted disk-drum joint. This includes the mathematical expressions to determine the bending stiffness of the bolted joint and methodology to obtain the modal frequencies of the rotor system analytically. Based on these expressions MATLAB codes are scripted for both static and dynamic analysis.

Considering a linear dynamic model without any bolted joint, the equation of motion for the rotor system without bolted joint can be expressed as:

$$M\ddot{\delta} + (C + \Omega G)\dot{\delta} + K\delta = f \tag{1}$$

Where, the 'M' is the mass matrix, 'C' is the damping matrix, 'G' is the Gyroscopic matrix and 'δ' is the displacement matrix. Now by considering nonlinear model, the bending stiffness matrix 'Kbe' is added into corresponding global stiffness matrix 'K'. Dynamic equation of rotor for modal analysis considering bolted joint is:

$$M\ddot{\delta} + (C + \Omega G)\dot{\delta} + (K + K_{be})\delta = f \tag{2}$$

IV. MODELING AND ANALYSIS

3.1 Solid modeling using CATIA v5

The rotor system considered for analysis has 3 disks and 2 drums and are joined using M6 bolts conforming to ISO standard. To determine the bending stiffness of the bolted joint i.e. for static analysis the CAD model is as shown in Fig.1. This includes only the middle disk bolted to the drums. The CAD model for dynamic analysis involves all the 3 disks as shown in Fig.2
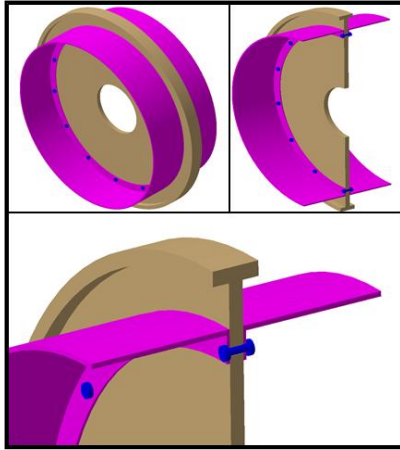
Fig. 1. CAD model of rotor with bolted joint
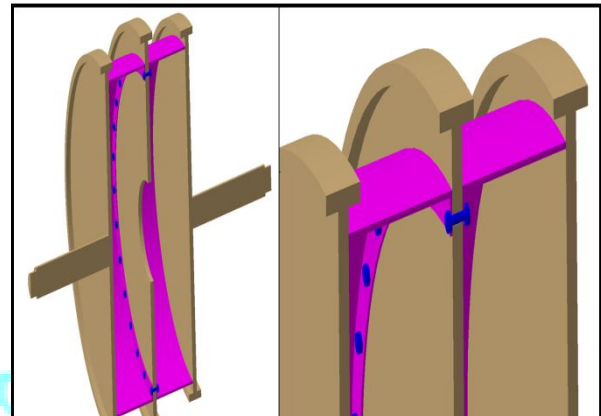(for static analysis)

Fig. 2. CAD model of 3 rotors with bolted joint
(for dynamicanalysis)

3.2 FE Modeling & Analysis using ANSYS Workbench

The solid model of a bolted joint rotor, is imported to ANSYS Workbench and meshed using 3D-solid tetrahedron elements. The total number of nodes were 46874 and total number of elements were 23202. Initially all parameters are investigated with bolt preload value of 100N and then with a preload of 200N.

3.2.1 Bending Stiffness behaviour (Static Analysis)

In this paper the bending stiffness is investigated for different combination of bolt numbers and preload values. The boundary conditions applied are shown in Fig.3 and as follows: The bolt pretension, in each bolt is assigned a preload of 100N at first and then 200N. All the DOFs of the nodes on the right side surface of right side drum are constrained to form a fixed support. This represents the rigidity provided from the welding of drum to the outer disks. A bending moment, applied on the left drum shell surface indicates the bending load on the rotor drum.
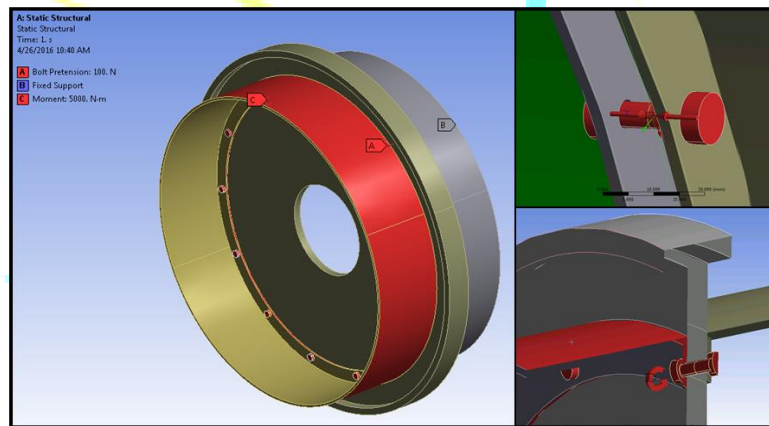

Fig. 3. FE model with boundary conditions (for static analysis)

To generate a contact pair the contact and target surfaces are assigned. The contact between the disk and the drum flange represents a rigid-to-flexible and surface-to-surface contact. Here the target surface is generally preferred to be of the rigid disk and the contact surface is from the flexible drum. In case of the contact between the bolt head surface and the inner surface of the drum flange, a flexible-to-flexible contact is represented where both the bodies in contact undergo deformation. Out of these two, bolt head surface is taken as target surface and drum flange surface will become the contact surface to form a flexible-to-flexible contact pair.

For different combination of bolt numbers, bolted joints with 30 bolts and 14 bolts are modeled. In a 30 bolted model, out of 62 contact pairs, the two contact pairs between the disk and the drum is considered to be a frictional joint having a co-efficient of friction as 0.2. The remaining 60 contact pairs between 30x2 bolt heads

and drum flange are considered as bonded contact. This is because the bolt pretension load which is normal to the contact surface will be very high compared to the tangential forces over the contact surface. Hence the frictional forces are neglected at these 60 interfaces.

3.2.2 Dynamic Analysis

For dynamic analysis the boundary conditions applied are shown in Fig.4 and as follows: To restrict the rotor in axial direction, the ends of the rotor shaft is fixed. The bearing are represented by elastic supports having a stiffness of $2x10^7$ N/m. The Bolt pretension, in each bolt is assigned a preload of 100N at first and then 200N. In dynamic analysis, the 30 bolted rotor model will have 2 additional contact pairs i.e. a total of 64 contact pairs. The two additional contact pairs are between left & right disks and the respective rotor drum shells. This pair represents a bonded contact indicating the welded joint between left & right disks and their respective rotor drums. The remaining 62 contact pairs are the same as in static analysis.

For rotordynamic analysis, to obtain the Campbell diagram, and perform the modal analysis, a rotational velocity is applied on the rotor disks as shown in Fig.5. The axis of rotation also called as spin axis is taken to be the x-axis. The operational speed range for the rotor under study is considered to be 1000 to 12000 rpm (104.72 to 1256.6 rad/s) and the velocity is incremented with a step of 1000 rpm.
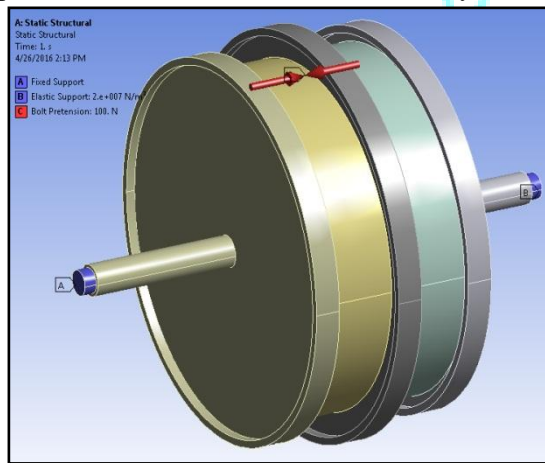

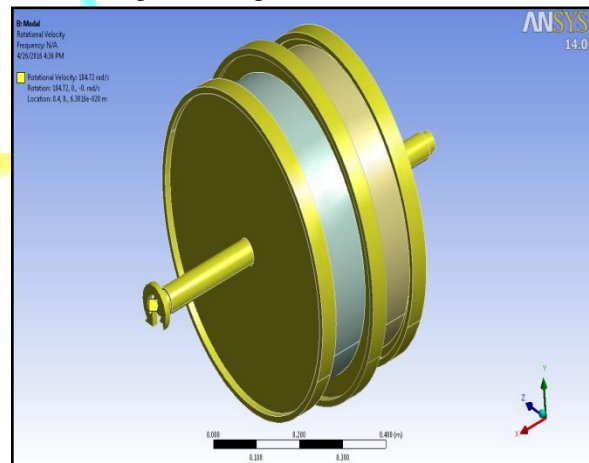Fig. 4. FE model with boundary conditions
(for dynamic analysis)

Fig. 5. FE rotor model with rotational velocity

3.2.3 Harmonic Analysis

For harmonic analysis, the excitation force considered is from the unbalance mass present in the middle rotor disk. The unbalance force is given by $F_b = mr\omega^2 = F_0\omega^2$, where 'm' is the unbalance mass, 'r' is the location of unbalance in the disk and 'ω' is the rotational velocity. This centrifugal unbalance force can be resolved along y and z axes as shown in Fig.6.In this study, an unbalance mass of m=0.1kg, present at a radius of r=0.25m in the middle-disk is considered. Hence the unbalance force of $F_0 = 2.5x10^{-2}$ N is applied on the middle disk at a radius of 0.25m as shown in Fig.7.
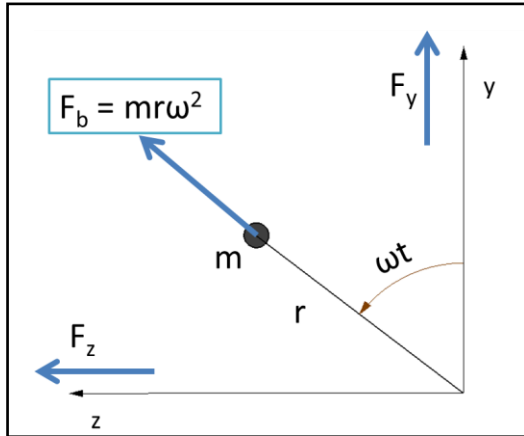
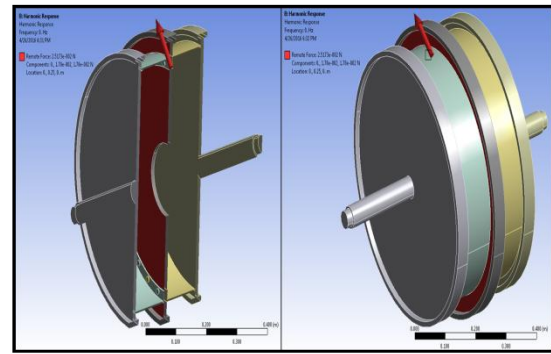Fig. 6. Unbalance excitation force in the rotor disk

Fig. 7. Unbalance force on middle disk of the bolted joint rotor

V. RESULTS AND DISCUSSION

4.1 Bending stiffness behavior

The total deformation and von-misses stress distribution areas are shown in Fig. 8 & Fig. 9 respectively. For different bending loads, the total deformation 'δ' is measured at the outer edge of the drum flange and the relative rotation angle 'Φ' is calculated. The respective bending stiffness is calculated as $K_{be}=M / \Phi$. As in Fig.10, the plot of M vs Φ, shows a nonlinear behavior of the bending stiffness. As the bolt preload is increased the slope of the line i.e. the stiffness increase. This is because the addition of preload reduces the deformation or opening up of the drum at the joint interface and prevents the bolted joint to deform under applied load. Also the increase in number of bolts induces more rigidity to the bolted joint.
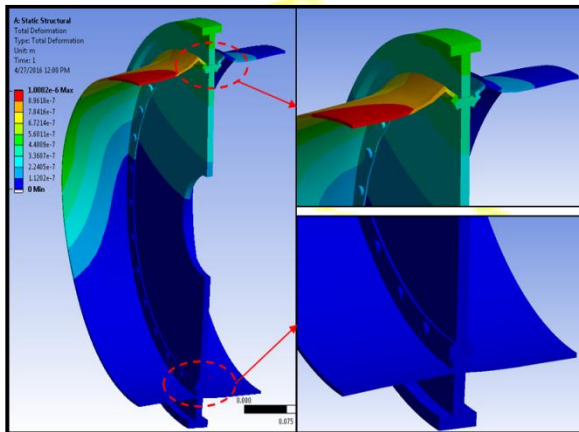

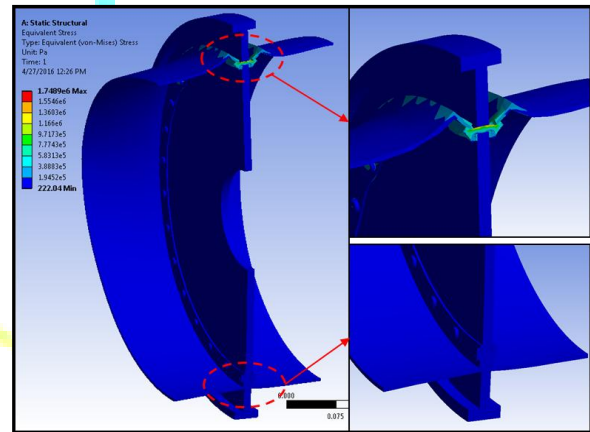Fig. 8. Total deformation of bolted disk-drum joint

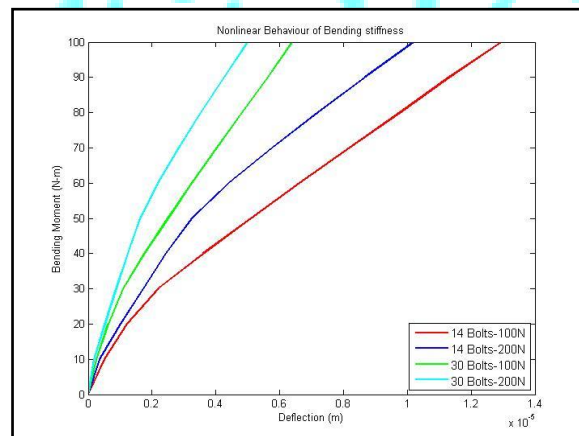Fig. 9. Von-Mises stress distribution

Fig. 10. Graph of bending moment M vs. relative rotation angle Φ

For a maximum bending load of 5000 N-m the bending stiffness was found to be 1.12×10^7 N/m from ANSYS simulation. A stiffness of 1.36×10^7 N/m was determined from MATLAB program. The stiffness values for different preloads and number of bolts are compared in TABLE I.

TABLE I.   BENDING STIFFNESS USING ANSYS AND MATLAB

Cases	14 Bolts, P=100N	14 Bolts, P=200N	30 Bolts, P=100N	30 Bolts, P=200N
ANSYS (N/m)	5.45×10^6	5.53×10^6	1.10×10^7	1.12×10^7
MATLAB (N/m)	5.74×10^6	5.96×10^6	1.23×10^7	1.36×10^7

4.2 Dynamic Analysis

4.2.1   Modal Analysis

The natural frequency of the rotor system for with bolt and without bolted disk-drum joint rotor obtained from ANSYS simulation and MATLAB program are given in Table II and Table III respectively.

TABLE II.   ANSYS MODAL FREQUENCIES (HZ) FOR DIFFERENT JOINT STATUS AND DIFFERENT BOLT NUMBERS

No. of Bolts	14 Bolts		30 Bolts	
Joint Status	Without Bolted	With Bolted	Without Bolted	With Bolted
1st Mode	36.15	36.15	35.35	35.34
2nd Mode	75.85	75.70	76.24	75.58
3rd Mode	75.97	75.79	77.06	76.01
4th Mode	81.76	81.43	94.13	92.97
5th Mode	93.03	92.67	95.39	93.97
6th Mode	105.50	105.18	102.69	102.69

TABLE III.   MATLAB MODAL FREQUENCIES FOR DIFFERENT JOINT STATUS AND DIFFERENT BOLT NUMBERS

No. of Bolts	14 Bolts		30 Bolts	
Joint Status	Without Bolted	With Bolted	Without Bolted	With Bolted
1st Mode	54.49	44.32	58.28	44.32
2nd Mode	59.17	46.17	61.48	46.32
3rd Mode	76.21	73.5	77.13	73.50
4th Mode	83.18	78.62	84.23	78.62
5th Mode	93.64	92.92	93.83	92.92
6th Mode	104.33	103.63	104.38	103.63

From the above two tables it can be seen that the presence of bolted joint in the rotor model reduces the rigidity making it a much more flexible rotor.

4.2.2   Critical speeds from Campbell diagram

Fig.11 shows Campbell diagram for a bolted rotor with 30 bolts. The variations in critical speeds obtained from Campbell diagram are presented in Table IV and Table V. It can be seen that critical speeds decrease due to softening of the joints. On the other hand, the increase in number of bolts increases the critical speed by strengthening the bolted joint and making the rotor system relatively rigid.
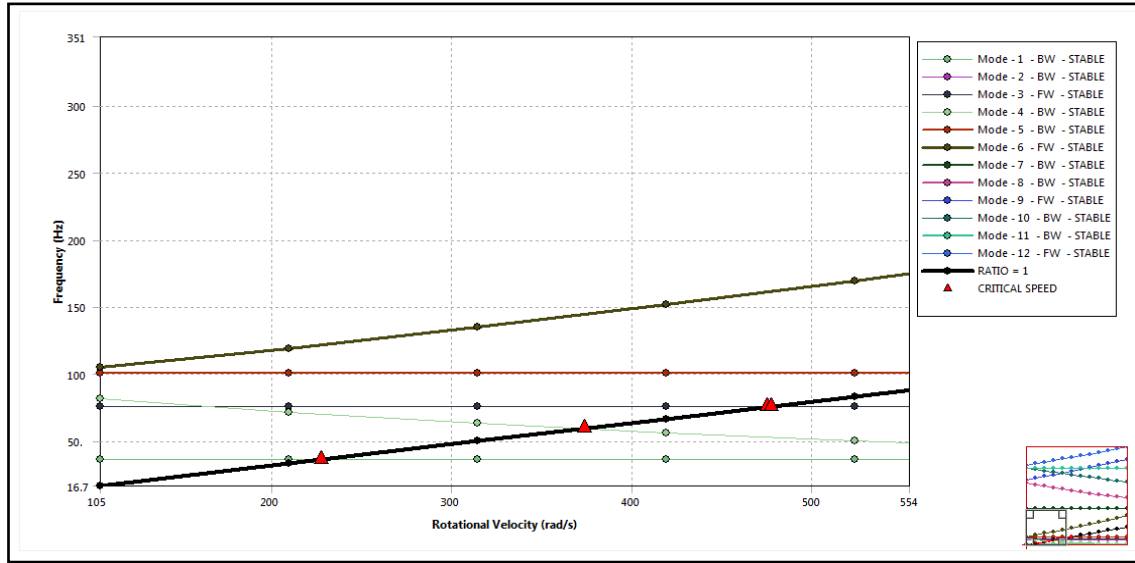
Fig. 11. Campbell diagram for a bolted rotor with 30 bolts

TABLE IV.   VARIATION IN CRITICAL SPEED IN BOLTED ROTOR WITH 14 BOLTS

| Mode # | Without Bolted | | | With Bolted | |
| | Whirl Direction | Critical Speed | | Critical Speed | |
		rad/s	Hz	rad/s	Hz
1	FW	227.16	36.17	227.16	36.17
2	BW	476.09	75.81	476.07	75.65
3	FW	478.21	76.15	477.19	75.99
4	BW	375.01	59.71	373.69	59.50
5	FW	647.7	103.14	631.38	100.54

TABLE V.   VARIATION IN CRITICAL SPEED IN BOLTED ROTOR WITH 30 BOLTS

| Mode # | Without Bolted | | | With Bolted | |
| | Whirl Direction | Critical Speed | | Critical Speed | |
		rad/s	Hz	rad/s	Hz
1	FW	222.09	35.36	221.40	35.25
2	BW	480.89	76.57	478.76	76.24
3	FW	484.59	77.16	482.41	76.82
4	BW	406.30	64.70	403.35	64.23
6	BW	677.27	107.85	645.12	102.73

4.2.3   Harmonic analysis

For the excitation due to the unbalance mass of 0.1kg at a radius of 0.25m in the middle rotor, the response amplitude of the outermost edge of the middle disk is recorded. The Amplitude vs. Frequency plot for the middle disk considering both the case of without bolted rotor and with bolted joint rotor are as shown in Fig.12 & Fig.13 respectively.The response amplitudes for different number of bolts are summarized in Table VI and Table VII.
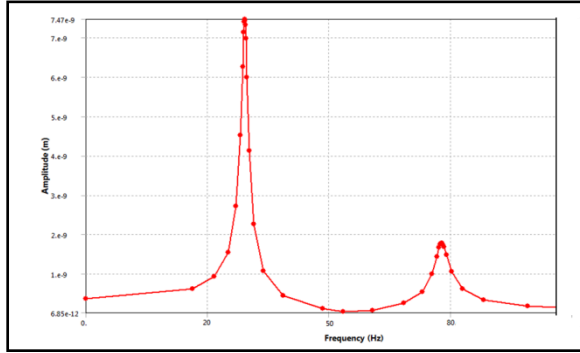
Fig. 12. Amplitude vs. Frequency plot for disk-drum rotor without bolted joint
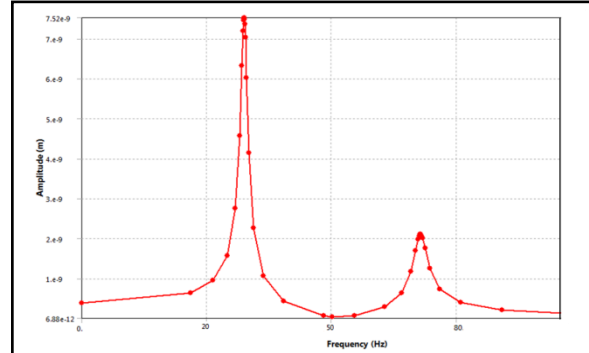

Fig. 13. Amplitude vs. Frequency plot for disk-drum rotor with 30 bolts

TABLE VI. COMPARISON OF RESPONSE AMPLITUDES FOR BOLTED DISK-DRUM ROTOR WITH 30 BOLTS

Response Peak	With Bolted			
	Amplitude (m)	Speed rad/s	Amplitude (m)	Speed rad/s
1st Peak	7.47×10^{-9}	222.09	7.52×10^{-9}	221.4
2nd Peak	1.77×10^{-9}	484.80	2.1×10^{-9}	482.50

TABLE VII. COMPARISON OF RESPONSE AMPLITUDES FOR BOLTED DISK-DRUM ROTOR WITH 14 BOLTS

Response Peak	With Bolted			
	Amplitude (m)	Speed rad/s	Amplitude (m)	Speed rad/s
1st Peak	1.01×10^{-7}	227.32	1.01×10^{-7}	227.16
2nd Peak	2.91×10^{-8}	477.01	2.92×10^{-8}	476.01

From the above two tables, it is quite clear that the presence of bolted joint in the rotor experiences relatively higher amplitudes of vibration at resonance frequencies(i.e. at critical speeds). Also from the comparision of the response plots in Fig.14, it can be seen that the amplitude peak slightly shifts towards the left as the rotor system exhibits lesser rigidity relative to the rotor system without bolted joints.
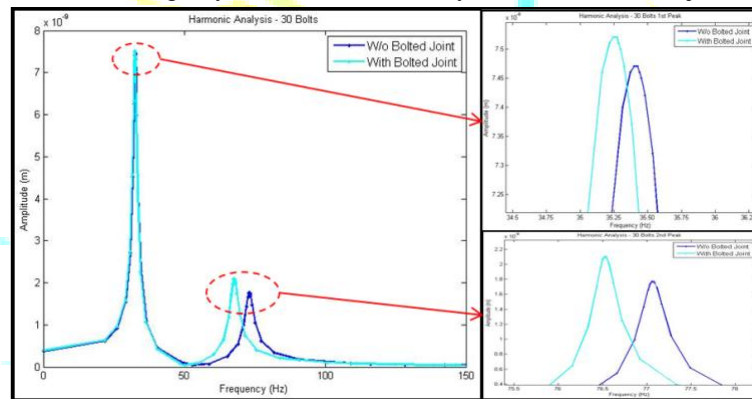

Fig. 14. Unbalance response comparison for rotor with 14 bolts

VI. CONCLUSION

The dynamic behaviour of a bolted disk-drum joint rotor discussed in this thesis considers the nonlinearity effect due to the frictional contact at the joint interface. Based on the analytical model used, the influence of bolted joint on dynamic parameters and the importance of these parameters are comprehended from the results obtained by both analytical and FEA methods. The FEA results obtained showed good agreement with analytical model. This study emphasizes the importance of understanding the nonlinear behavior of a bolted joint and its dependent parameters in any rotor-bearing system. As the consideration of nonlinear effects is more of a practical concern, the results obtained from this investigation are more of realistic values and are

reliable. Following are the vital conclusions that are drawn from this study.

1) The simulation results show that there is a decline in bending stiffness in case of a bolted disk-drum rotor. Hence the modal frequency decreases in the bolted model as against the rotor model without bolted joint.

2) From the analytical model used, the bending stiffness at the joint, displays a nonlinear behavior at lower bending loads, which is an obvious phenomena due to frictional losses in the contact zone as the tangential forces are predominant at lower bending loads.

3) The increase in number of bolts, increases the joint stiffness as these additional bolts try to minimize the deflection of the drum flange at joint interface. This additional stiffness causes the modal frequency to increase. Hence optimum number of bolts should be considered that satisfies the joint strength requirements and limits the joint deflection without much change in the modal frequency.

4) In a similar manner, the bolt preloads have little or no influence in modal frequencies. Increase in preload increases joint stiffness and hence the modal frequency seems to be increasing, but no significantly.

5) Consideration of frictional contact between the joined parts decreases the critical speeds. Though the critical speed increases comparatively, by increasing the number of bolts, there is an overall decline in critical speed due to the presence of bolted joint.

6) In the case of bolted rotor model, the amplitude peak from harmonic response seems to increases relatively at the resonance conditions due to slight decrease in the rigidity.

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