International Journal of Mechanical Engineering and Robotics Research

ISSN 2278 ñ 0149 www.ijmerr.com
Vol. 3, No. 1, January 2014

Research Paper

ROTOR DYNAMIC ANALYSIS OF STEAM TURBINE ROTOR USING ANSYS

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Rotor dynamics is a field under mechanics. Mainly deals with the vibration of rotating structures. In recent days, the study about rotor dynamics has gained more importance within steam turbine industries. The main reason is steam turbine consists of many rotating parts constitutes a complex dynamic system. While designing rotors of high speed turbo machineries, it is of prime importance consider rotor dynamics characteristics in to account. And the world we are living in today is pushing the technology harder and harder. The products need to get better and today they also need to be friendlier to the environment. To get better products we need better analysis tools to optimize them and to get closer to the limit what the material can withstand. The modeling features for rotor and bearing support flexibility are described in this thesis. By integrating these characteristic rotor dynamics features into the standard FEA- modal, harmonic and transient analysis procedures found in ANSYS we can analyze and determine the design integrity of rotating equipment. Some ideas are presented to deal with critical speeds calculation using ANSYS. This Thesis shows how elements BEAM188 and COMBI214 are used to model the shaft and bearings, respectively.The purpose of a standard rotor dynamics analysis of Steam turbine rotor is to enable an engineer to characterize the lateral dynamics design characteristics of a given design. With the Campbell plots, we can determine critical speeds and system stability. These techniques, along with a same modeling and results are also calculated from TMS-050 to verify ANSYS result with testing result for unbalance response.

Keywords: Ansys, Critical speed, Rotor, Rotor dynamics, Steam turbine, TMS-050, Vibrations

INTRODUCTION

Steam turbine plant is an integral part of thermal power station. Therefore development, construction and improvement of steam turbine are an important field of development of power industry. Growth in power and more complicated design of turbo machines are accompanied by higher requirements for their reliability. To increase operational life of turbo machines is also one of the main tasks of quality improvement. In this connection at present, when developing and mastering the steam turbines, modern computational and experimental methods are used to determine strength and reliability

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characteristics. Rotor dynamics is the branch of engineering that studies the lateral and torsional vibrations of rotating shafts, with the objective of predicting the rotor vibrations and containing the vibration level under an acceptable limit. The principal components of a rotor-dynamic system are the shaft or rotor with disk, the bearings, and the seals. The shaft or rotor is the rotating component of the system. Basically there are three forms of vibrations associated with the motion of the rotor: torsional, axial and lateral. Torsional vibration is the dynamics of the shaft in the angular/rotational direction. Normally, this is little influenced by the bearings that support the rotor. Axial vibration is the dynamics of the rotor in the axial direction and is generally not a major problem. Lateral vibration, the primary concern, is the vibration of the rotor in lateral directions.

The bearings play a huge part in determining the lateral vibrations of the rotor. In this thesis, we will study the basic concepts of the lateral rotor dynamics of turbo machinery. With ever increase in demand for larger size and velocity in modern machines, Rotor Dynamics became more and more an important subject in the mechanical engineering design. It is well know that torsional vibration in rotating machines, reciprocating machines installation and geared system, whirling of rotating shaft, the effect of flexible bearing, instabilities due to asymmetric cross-section shafts, hydrodynamics bearings, hysteresis, balancing of rigid and flexible rotor can be understood only on the basis of rotor dynamics studies. Rotor dynamics is an extremely important branch of the discipline of dynamics that pertains to the operation and behavior of a huge assortment of rotating machines. The purpose of a standard rotor dynamics analysis and design audit is to enable an engineer to characterize the lateral dynamics design characteristics of a given design. While analysis of some rotating equipment may require analysis specific to the unit, a general method has emerged for performing the standard lateral analysis.

Fundamental Equation

The general form of equation of motion for all vibration problems is given by,

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{f\} \quad ...(1.1)$$

Where,

[M] = symmetric mass matrix

[C] = symmetric damping matrix

[K] = symmetric stiffness matrix

[f] = external force vector

[u] = generalized coordinate vector In rotordynamics, this equation of motion can be expressed in the following general form [3],

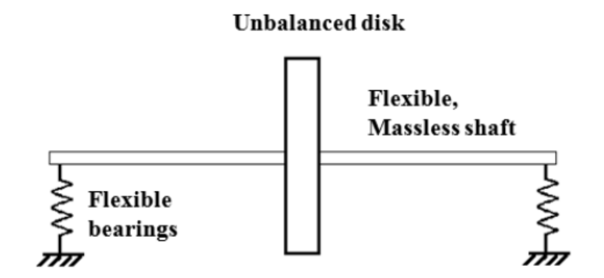
$$[M]\{\ddot{u}\} + ([C] + [Cgyro])\{\dot{u}\} + ([K] + [H])\{u\} = \{f\} \quad ...(1.2)$$

The above mentioned equation (1.2) describes the motion of an axially symmetric rotor, which is rotating at constant spin speed Ω about its spin axis. This equation is just similar to the general dynamic equation except it is accompanied with skew-symmetric gyroscopic matrix, [C gyro] and skew-symmetric circulatory matrix [H]. The gyroscopic and circulatory matrices [C gyro] and [H] are greatly influenced by rotational velocity Ω. When the rotational velocity Ω, tends to zero, the skew-symmetric terms present in equation (1.2) vanish and represent an ordinary stand still structure. The gyroscopic matrix [C gyro] contains inertial terms and that are derived from kinetic energy due to gyroscopic moments acting on the rotating parts of the machine. If this equation is described in rotating reference frame, this gyroscopic matrix [C gyro] also contains the terms associated with Carioles acceleration. The circulatory matrix, [H] is contributed mainly from internal damping of rotating elements (XU Yang et al., 2004).

Theory

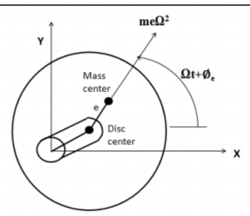
The concept of rotor dynamics can be easily demonstrated with the help of generalized Laval-Jeffcott rotor modal as shown in Figure 1.

Figure 1: Generalized Laval- Jeffcott Rotor Model



The generalized Laval-Jeffcott rotor consists of long, flexible mass less shaft with flexible bearings on both the ends. The bearings have support stiffness of KX and KY associated with damping CX and CY in x and y direction respectively. There is a massive disk of mass, m located at the center of shaft. The center of gravity of disk is offset from the shaft geometric center by an eccentricity of e. The motion of the disk center is described by two translational displacements (x, y) as shown in Figure 2.

Figure 2: End View of Laval- Jeffcott Rotor



When the rotor is rotating at constant rotational speed, Ω the equation of motion for the mass center can be derived from Newton's law of motion and it is expressed in the following form.

$$m\frac{d^2}{dt^2}\left(x + e\cos\left(\Omega t + \phi_e\right)\right) = -C_x\dot{x} - K_x x \quad \text{...(1.3)}$$

$$m\frac{d^2}{dt^2}\left(y + e\sin\left(\Omega t + \phi_e\right)\right) = -C_y\dot{y} - K_y y \quad \text{...(1.4)}$$

The above equations can be re-written as,

$$m\ddot{x} + C_x\dot{x} + K_x x = me\Omega^2\cos(\Omega t + \phi_e) \quad \text{...(1.5)}$$

$$m\ddot{y} + C_y y + K_y y = me\Omega^2\sin(\Omega t + \phi_e) \quad \text{...(1.6)}$$

Where, ϕ_e is the phase angle of the mass unbalance. The above equations of motions show that the motions in X and Y direction are both dynamically and statically decoupled in this model. Therefore, they can be solved separately.

Determination of natural frequencies

For this simple rotor model, the undamped natural frequency, damping ration and the damped natural frequency of the rotor model for X and Y direction can be calculated from

$$\omega_{nx} = \sqrt{\frac{K_x}{m}}, \qquad \zeta_x = \frac{C_x}{2m\omega_{nx}}, \qquad \omega_x = \omega_{nx}\sqrt{1-\zeta_x^{\,2}} \quad \text{...(1.7)}$$

$$\omega_{ny} = \sqrt{\frac{K_y}{m}}, \qquad \zeta_y = \frac{C_y}{2m\omega_{ny}}, \qquad \omega_y = \omega_{ny}\sqrt{1-\zeta_y^{\,2}}$$

Steady state response to unbalance

For single unbalance force, as present in this case, the ϕ_e can be set to zero. Therefore the equations (1.5) and (1.6) becomes,

$$m\ddot{x} + C_x\dot{x} + K_x x = me\Omega^2\cos(\Omega t) \quad \text{...(1.8)}$$

$$m\ddot{y} + C_y y + K_y y = me\Omega^2\sin(\Omega t) \quad \text{...(1.9)}$$

Then the solution for the response is,

$$|x| = \frac{me\Omega^2}{\sqrt{\left[\left(K_x - \Omega^2 m\right)^2 + \left(\Omega C_x\right)^2\right]}} \quad ...(1.10)$$

$$|y| = \frac{me\Omega^2}{\sqrt{\left[\left(K_y - \Omega^2 m\right)^2 + \left(\Omega C_y\right)^2\right]}} \quad ...(1.11)$$

MODELING AND DESIGN DATA INPUT

The modeling features for rotor and bearing support flexibility are described in this thesis, and shows how elements BEAM188, COMBI214 are used to model the shaft and bearings. And MASS21 used to model the additional masses.

Rotor

BEAM188 Element Description: BEAM188 is suitable for analyzing slender to moderately Stubby/thick beam structures. BEAM188 is a linear (2-node) or a quadratic beam element in 3-D. BEAM188 has six or seven degrees of freedom at each node, with the number of degrees of freedom depending on the value of KEYOPT(1). When KEYOPT (1) = 0 (the default), six degrees of freedom occur at each node. These include translations in the x, y, and z directions and rotations about the x, y, and z directions. When KEYOPT (1) = 1, a seventh degree of freedom (warping magnitude) is also considered. This element is well-suited for linear, large rotation, and/or large strain nonlinear applications.

Figure 3: Beam Geometry

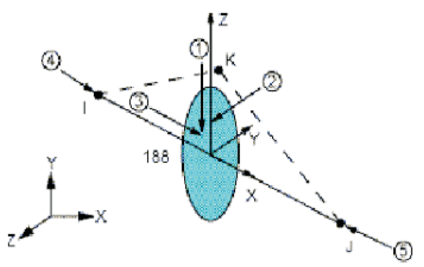


Table 1: Sectional Proprieties

Section No.	Length(L) (mm)	Diameter(D) (mm)	Temp °
1	09.00	75	60
2	11.00	71	60
3	116.50	63	60
4	18.00	136	60
5	54.00	63	60
6	18.00	110	60
7	33.50	100	60
8	40.00	100	60
9	54.50	100	60
13	06.00	125	197
14	02.00	125	207
15	12.5	125	250
16	28	125	260
17	52.5	125	280
18	133.5	245	480
19	36	362	486
20	53.8	180	313
21	12.65	166.3	455
22	12.5	140.8	449
23	16.29	168.8	444
24	12.5	143.3	438
25	16.65	171.3	432
26	12.5	145.3	438
27	17.01	173.8	419
28	12.5	148.3	412
29	17.36	176.3	405
30	12.5	150.8	398
31	17.72	178.8	390
32	12.50	153.3	382
33	18.08	181.3	373
34	12.50	155.8	364
35	19.14	183.8	354
39	53.00	125	200
40	1.00	125	106
41	39.00	105	60
42	54.50	100	60
43	40.00	100	60
44	77.00	100	60
45	25.00	85	60
46	29.50	60	60
47	10.00	70	60
48	31.00	105	60
49	1.00	105	60

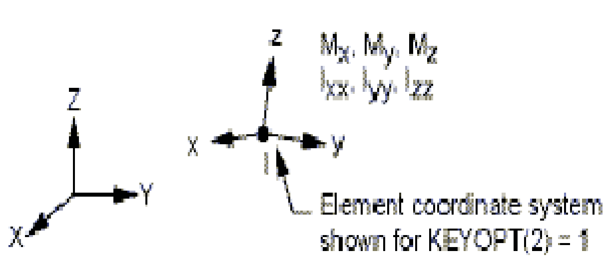
Where, L = Length of the each section (mm)

D = Diameter of each section (mm)

Blades (Additional Disks Masses)

MASS21 Element Description: MASS21 is a point element having up to six degrees of freedom: translations in the nodal x, y, and z directions and rotations about the nodal x, y, and z axes. A different mass and rotary inertia may be assigned to each coordinate direction.

Figure 4: Mass 21 Geometry



Two bearings used in this thesis, one at front side (at section 08) and other at rear side (at section 43). And "COMBI-214" Element used for Modeling of the Bearing in ANSYS. COMBI214 Element Description: 2-D Spring-Damper Bearing. COMBI214 has longitudinal as well as cross-coupling capability in 2-D applications. It is a tension compression element with up to two degrees of freedom at each node: translations in any two nodal directions (x, y, or z). COMBI214 has two nodes plus one optional orientation node. No bending or torsion is considered. The spring-damper element has no mass.

Table 3: Bearing Details

	Length (mm)	Diameter (mm)	Section Location	Type of bearing
Front Bearing	40	100	8	Tilting pad
Rear Bearing	40	100	43	Tilting pad

Table 4: Properties for 1st Bearing

Speed (RPM)	$K_{XX} \times 10^3$	K_{XY}	K_{YX}	$K_{YY} \times 10^3$	$C_{XX} \times 10^3$	C_{XY}	C_{YX}	$C_{YY} \times 10^3$
671.3	27003	0	0	29272	155.26	0	0	193.35
1342.6	32701	0	0	36101	152.46	0	0	185.05
2013.9	38414	0	0	42945	149.66	0	0	176.75
2685.3	44127	0	0	49789	146.86	0	0	168.44
3356.6	49833	0	0	56618	144.05	0	0	160.14
4292	57749	0	0	66146	140.11	0	0	148.52
4703.2	61289	0	0	70336	138.45	0	0	143.54
5361.3	66893	0	0	77090	135.75	0	0	135.34
5610.3	69023	0	0	79633	134.71	0	0	132.43
6411.7	75818	0	0	87804	131.39	0	0	122.57
7205.9	82545	0	0	95859	128.07	0	0	112.61
8014.7	89459	0	0	104093	124.65	0	0	102.75
8825.1	96395	0	0	112413	121.33	0	0	92.89
9637.3	103276	0	0	120663	118	0	0	82.92
10403.1	109674	0	0	128443	114.68	0	0	73.07
11000	115566	0	0	135608	111	0	0	63.12
12000	125388	0	0	147551	107	0	0	51.02
13000	135209	0	0	159494	102	0	0	38.25

Table 2: Desk Input Data

Disk No.	Mass of Disk (kg)	Equivalent Diameter	Section Location
1	1.8147	215.02	22
2	1.8254	216.94	24
3	1.8714	220.19	26
4	1.9142	223.36	28
5	1.9589	226.51	30
6	2.0073	229.88	32
7	2.0554	233.13	45
8	2.1049	236.5	49

Bearing Details

Figure 5: Combi 214 Geometry

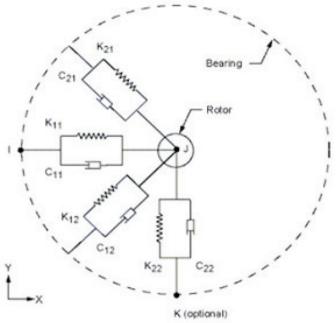


Table 5: Properties for 2nd Bearing

Speed (RPM)	$K_{XX} \times 10^3$	K_{XY}	K_{YX}	$K_{YY} \times 10^3$	$C_{XX} \times 10^3$	C_{XY}	C_{YX}	$C_{YY} \times 10^3$
664.5	26729	0	0	28975	155.26	0	0	193.35
1329	32369	0	0	35734	152.46	0	0	185.05
1993.5	38024	0	0	42509	149.66	0	0	176.75
2658	43679	0	0	49283	146.86	0	0	168.44
3322.5	49326	0	0	56043	144.05	0	0	160.14
4248.4	57163	0	0	65474	140.11	0	0	148.52
4655.4	60666	0	0	69622	138.45	0	0	143.54
5553.3	68322	0	0	78824	134.71	0	0	132.43
6346.6	75047	0	0	86911	131.39	0	0	122.57
7132.6	81706	0	0	94885	128.07	0	0	112.61
7933.2	88550	0	0	103035	124.65	0	0	102.75
8735.5	95415	0	0	111270	121.33	0	0	92.89
9539.3	102226	0	0	119436	118	0	0	82.92
10297.4	108559	0	0	127137	114.68	0	0	73.07
11000	114392	0	0	134230	111	0	0	63.12
12000	122724	0	0	144363	107	0	0	51.02
13000	131057	0	0	154495	102	0	0	38.25

Rotor Material Properties

Table 6: Material Property

Young Modulus 'E	2.1× 1011 N/m2
Poisson Ratio (µ)	0.25
Density 'ñ'	7800 kg/m3

Table 6: Rotor with Meshing

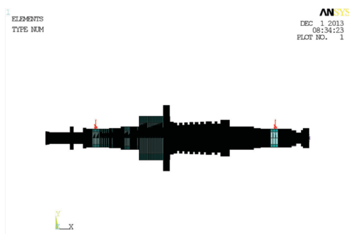
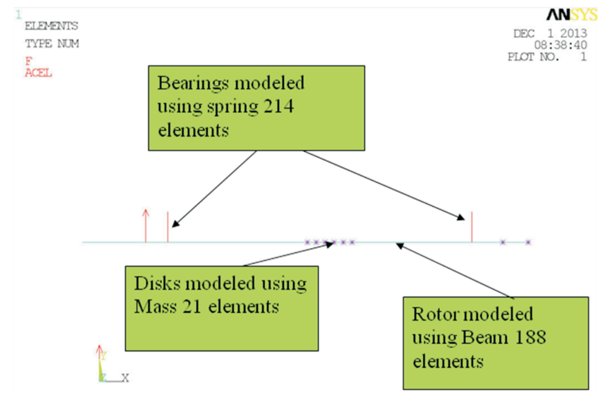


Figure 7: Finite Element Model of Rotor



RESULTS AND DISCUSSION

The different analysis carried out to the rotor dynamic integrity of the steam turbine rotor under given loads.

Analysis Types

A. Modal analysis

B. Harmonic analysis

C. Transient analysis

A. Modal Analysis

Figure 8, Figure 9, Figure 10 and Figure11 shows fist four mode shape and damped natural frequency at the operating speed (11800 rpm) by Ansys.

Figure 8: Mode Shape 1

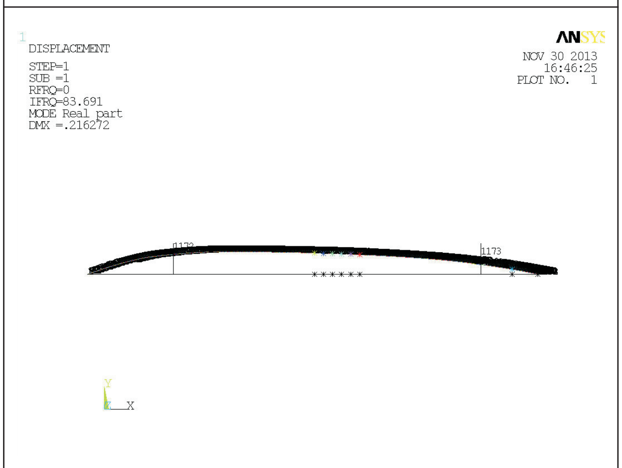


Figure 9: Mode Shape 2

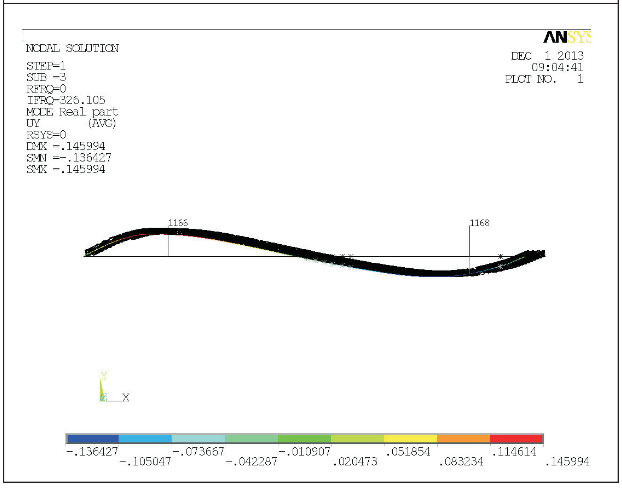


Figure 10: Mode Shape 3

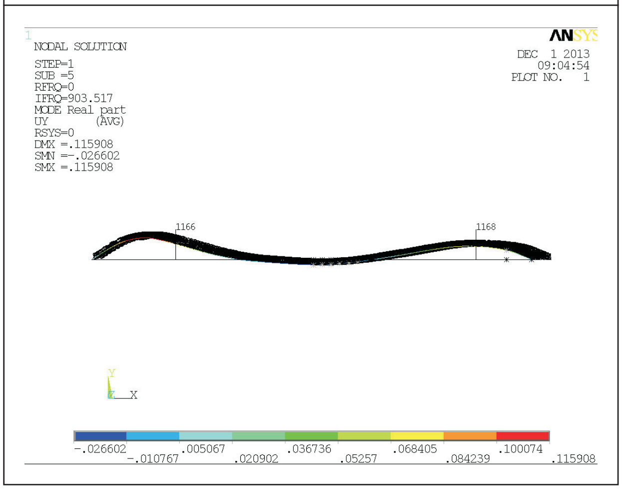


Figure 11: Mode Shape 4

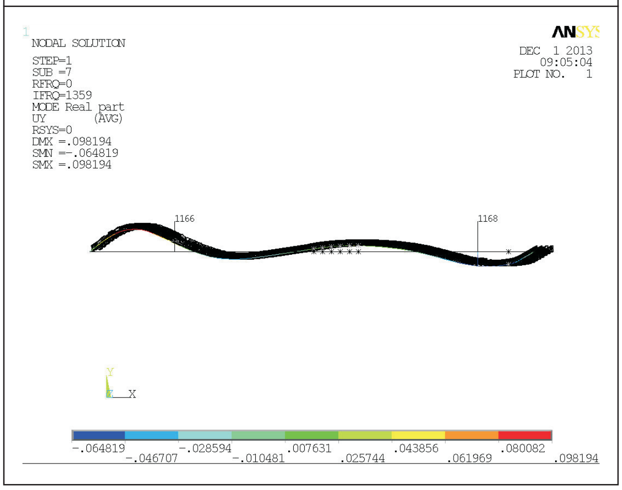


Table no 8 Shows comparisons of the two different tools for undamped natural frequency (Hz) at the operating speed (11800 rpm).

Table No.8 Undamped natural frequency (Hz) at the operating speed.

Table 8: Undamped Natural Frequency (Hz) at the Operating Speed

Mode Number	Ansys	TMS-050
1	83.6	84.0
2	326.1	329.1
3	903.5	901.5
4	1359	1361

Figure 12. Shows Critical speeds of the system throughout the full range speed. We can find out the natural frequency of the system by interpolating.

Figure 12: Critical Speed Map by TMS-050

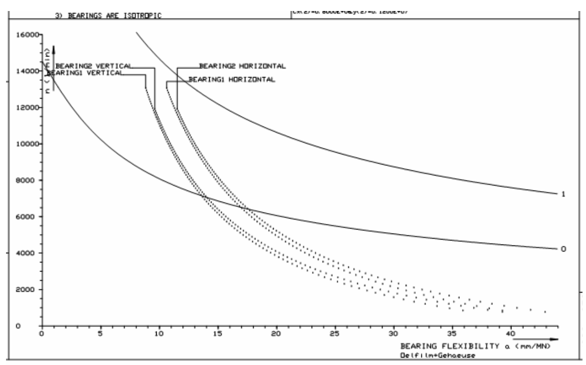


Table 9: Damped Natural Frequency (Hz) at the Operating Speed

Mode Number	Ansys	TMS-050
1	81.5	81.2
2	85.6	85.1
3	221.6	227.6
4	464.9	451.9

Table No.9 Show comparison of the damped natural frequency (Hz) of two different tools at the operating speed (11800 rpm).

Critical speed and Campbell diagram analysis.

In this analysis, a number of Eigen frequency analyses are performed on the steam turbine rotor model for the speed range starting from 0 rpm to 12000 rpm with an increment of 150 rpm using multiple load steps.

Figure 13: Damped Critical Speed by Ansys

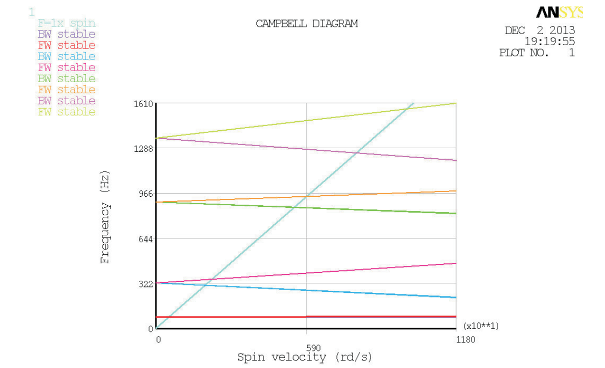
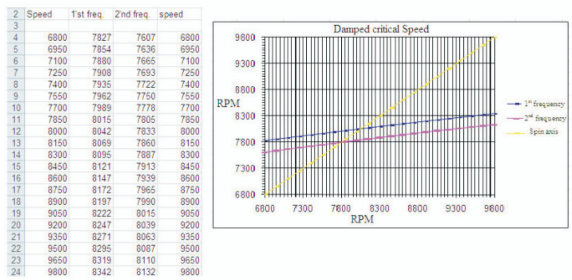


Figure 14: Damped Critical Speed by TMS-050



2	Speed	1'st freq.	2'nd freq.	speed
3				
4	6800	7827	7607	6800
5	6950	7854	7636	6950
6	7100	7880	7665	7100
7	7250	7908	7693	7250
8	7400	7935	7722	7400
9	7550	7962	7750	7550
10	7700	7989	7778	7700
11	7850	8015	7805	7850
12	8000	8042	7833	8000
13	8150	8069	7860	8150
14	8300	8095	7887	8300
15	8450	8121	7913	8450
16	8600	8147	7939	8600
17	8750	8172	7965	8750
18	8900	8197	7990	8900
19	9050	8222	8015	9050
20	9200	8247	8039	9200
21	9350	8271	8063	9350
22	9500	8295	8087	9500
23	9650	8319	8110	9650
24	9800	8342	8132	9800

Figure 13 show damping critical speed (Campbell diagram) of the system from Ansys. Figure 14 Show critical map method to find out damped critical speed of the system by TMS-050 software.

Often, rotor critical speeds correspond to natural frequencies of the system. Steam turbine rotor is supported by two tilting pad bearings. Typically, stiffness and damping coefficients of the bearing are varied with rotating speed, and in this case, natural frequencies of the system are varied. When a natural frequency equals to the rotating speed, the rotating speed is called critical speed. TMS-050 series software gives only numerical damped natural frequency corresponding to the speed. In order to find out damped critical speed it is necessary to convert this numerical data into graphical representation.

Table 10: Damped Critical Speed

Critical Speed	Ansys rpm	TMS-050 rpm
1	7451	7780
2	8145	8045

Table No.10 shows damped critical speed of two different tools Ansys and TMS-050

B. Harmonic Analysis

In this section, it will show unbalance response of the system at the bearing location by apply unbalance force at the center position to find out displacements which is very sinusoidal at the same known frequency and comparison of all the result with shop test result.

Figure 15: Unbalance Response at 1st Bearing

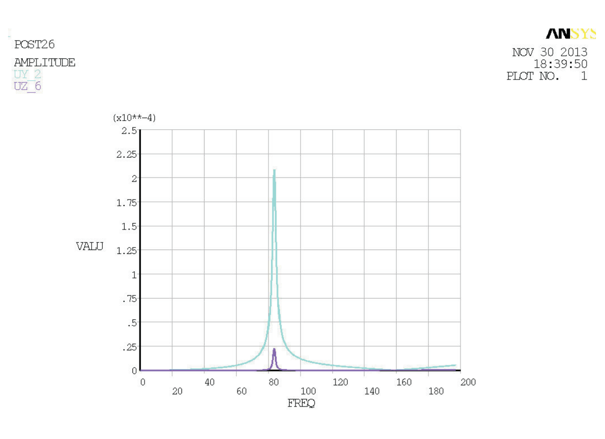


Figure 16: Unbalance Response at 2nd Bearing

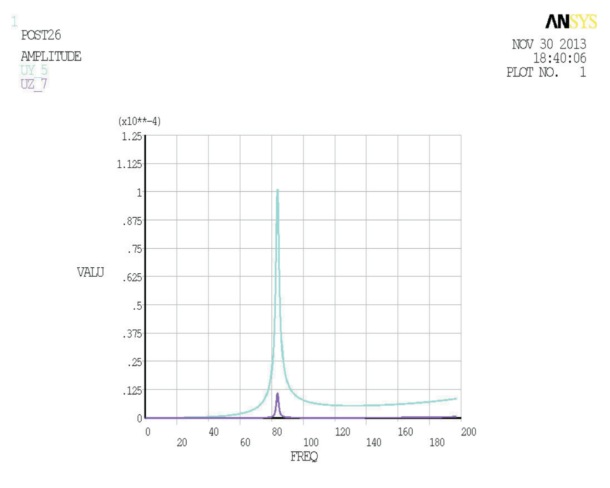


Figure 17: Unbalance Response at 1st Bearing

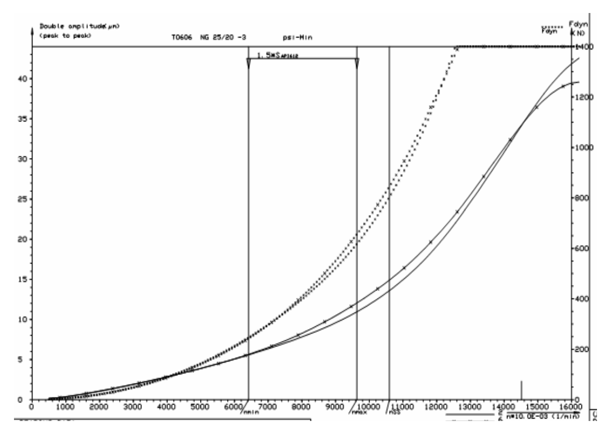


Figure 18: Unbalance Response at 2nd Bearing

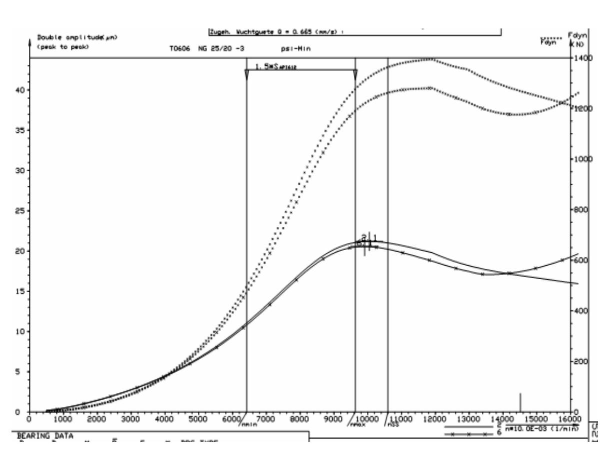


Figure 19: Unbalance Response at 1st Bearing

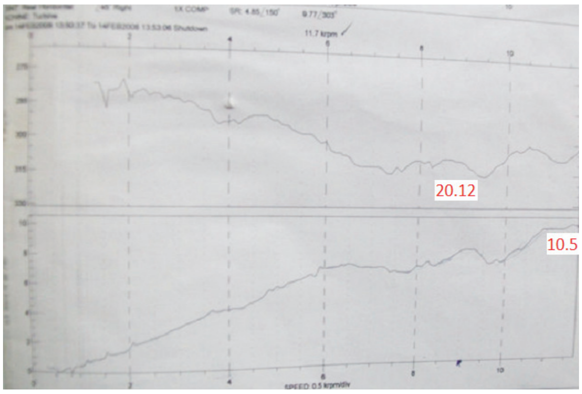


Figure 20: Unbalance Response at 1st Bearing

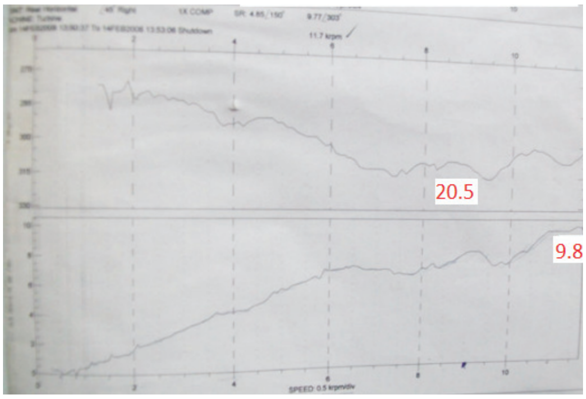


Figure 15 and Figure 16 shows unbalance response at bearing 1 and 2 respectively from Ansys , Figure 17 and Figure 18 shows unbalance response at bearing 1 and 2 respectively from TMS-050 and Figure 19 and Figure 20 shows experimental unbalance response at bearing 1 and 2 respectively.

Table 11: Unbalance Response at 1st Bearing

Bearing Location	Ansys	TMS-050	Shop Test
1	10	12	10.5
2	20.5	21	20.12

Table 12: Unbalance Response at 2nd Bearing

Bearing Location	Ansys	TMS-050	Shop Test
1	20.15	22	20.5
2	12.5	13	9.8

C. Transient Analysis

In this section, it will show the response of a structure to arbitrary time-varying loads at the bearing location to find out stability of the system at the different operating speed with the seal effect. If the amplitude of the system is decrease with time, that means system is stable otherwise system is unstable. Also it will calculate log-decrement (ld) value of the system at different speed and comparison of the ld value.

Figure 21: Response at 1st Bearing

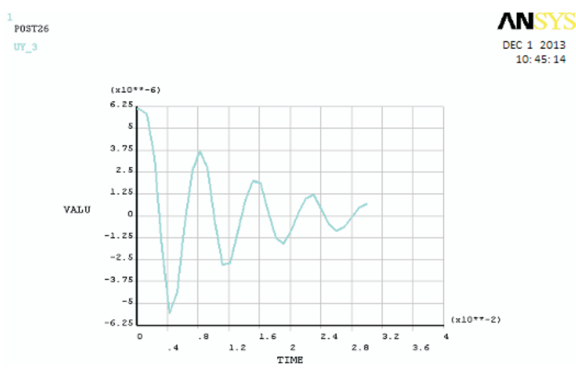


Figure 22: Response at 2nd Bearing

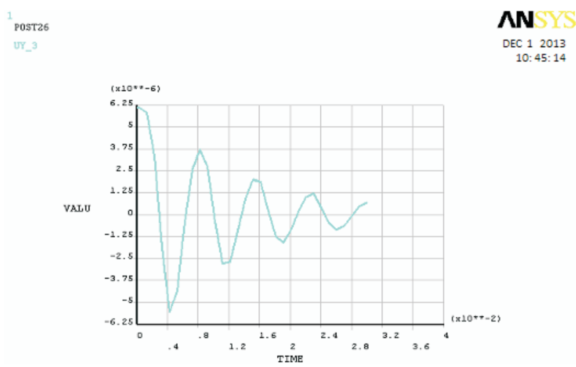


Figure 21 and Figure 22 shows the transient response at bearing 1 and 2 respectively.

Table 13: Comparison of ld Value

Speed (rpm)	Ansys (QR-damped)	TMS-050
9200	2.518	2.343
9350	2.482	2.285
9500	2.446	2.231
9650	2.348	2.178
9800	2.318	2.127
11800	1.997	2.447

Table 14: Comparison of ld Value

Speed (rpm)	Ansys (QR-damped)	TMS-050
9200	1.829	1.421
9350	1.781	1.367
9500	1.734	1.314
9650	1.603	1.262
9800	1.563	1.213
11800	1.132	0.918

Table 13 and Table 14 Show the comparison of the ld value at different operating speed for the 1st damped frequency and 2nd damped frequency respectively.

CONCLUSION

Thus the main objective of the thesis work to build and to perform Rotor dynamics analysis of steam turbine rotor model using Ansys is accomplished. It has been shown through simulations and comparisons, the results obtained from Ansys model and TMS-050 are in good agreement with each other. The Rotor made of multiple steps is modeled and analyzed for different boundary conditions in Ansys and TMS-050. The analysis summary is as follows.

From Eigen frequency analysis of steam turbine rotor model, Eigen frequencies of the steam turbine rotor for different rotational speeds are calculated. The Eigen frequencies obtained from Ansys and TMS-050 are closed to each other for most of the modes. The number of critical speeds calculated from Ansys model and TMS-050 is fair. The Campbell diagram generated from Ansys is very similar to critical speed diagram of TMS-050.

From harmonic analysis, the maximum displacement of the rotor and bearing load for the applied unbalance loading are determined. Peak values of the response curves obtained from Ansys and TMS-050 are relatively close to each other. And also the results obtained from two tools are closed to experimental results. From Transient analysis, the amplitude of response is decreases with increase the time, which means system is stable. And also log-decrement values calculated from Ansys and TMS-050 are good agreement.

Finally the results obtained from various analyses are under acceptable limits. So the system is safe for working with given bearing valves and rotor loads. Apart from these, Ansys software could be an effective tool for rotor dynamics calculation in many aspects. It has got some extra additional features than TMS-050. Ansys has the capability to handle more complex geometry.

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