International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 08 Issue: 08 | Aug 2021 www.irjet.net p-ISSN: 2395-0072

Rotor Dynamic Analysis of Horizontal Multistage Centrifugal Pump Rotor

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Abstract - *Rotordynamics is a field under mechanics, deals with the studying the vibration characteristics of rotating structures. In recent days, the study about rotordynamics has gained more importance within pump industries. The main reason is pump consists of many rotating parts constitutes a complex dynamic system. While designing rotors of high speed pump machineries, it is of prime importance to consider rotordynamics characteristics in to account. Considering these characteristics at the design phase may avoid the pump rotor from severe catastrophic failures.*

The present work focuses with the modal analysis of pump rotor. A rotor dynamic model is developed by using Ansys APDL commands. The main aim of this work is to build a pump rotor model in Ansys and Dyrobes to evaluate its critical speeds for different bearing conditions. In this study, lateral vibration of pump rotor is investigated analytically under two different bearing conditions: Rigid Bearings, Flexible Bearings, Analytical solution is carried out using Dunkerley's method. The unbalance response analysis is also carried out for the pump rotor using Ansys and Dyrobes. It is concluded that analytical and numerical results are found to be good agreement. The results of the unbalance response analysis for the pump rotor obtained from Ansys and Dyrobes are well agreed with each other. This work helps in understanding, modeling, simulation and post processing techniques for rotordynamics analyses of pump rotor using Ansys.

Key Words: Rotordynamics, Natural Frequency, Dry critical speed, Wet critical speed, finite element method.

1. INTRODUCTION

Centrifugal pumps are employed to carrying the fluids by converting the rotational kinetic energy in to the hydraulic energy with high pressure. Usually, electric motors are used for supplying rotational energy to the pump. The most general form of centrifugal pump is considered as volute pump. In this kind of pumps, liquid enters in to the pump at the middle location of a rotating impeller. The rotating impeller leads a much radial acceleration of the liquid from the centre of the impeller to the circumferential pump casing. This generates a vacuum at the middle location of the impeller, consequently repeated entrance of much process liquid and this liquid exhaust the pump by means of a discharge port.

Whenever the rotors rotate at very high speed they develop resonance. This resonance can be defined as "It is the conditions at which the harmonic forces are excited at their natural frequencies and it produces the extreme vibration in the rotors". This vibration with greater displacements begins the rotors to bend and twists significantly and it guides to permanent breakdown of the rotating system. Also, the deflection of shafts in incongruous manner has a greater chance to have a collision with the adjacent components at its closer proximity and cause severe unrecoverable damages. Hence the determination of these rotordynamics characteristic is much important [7-8].

2. LITERATURE SURVEY

Naveena. M. and Suresh P. M. [1], in this paper authors studied and analyzed the critical speed of multistage centrifugal pump rotor by using ANSYS. And also, the authors obtained analytical undamped critical speed by using Rayleigh's method. They compared the results obtained from ANSYS with the Analytical calculations and RBTS (Rotor Bearing Testing Software). Unbalance response analysis was also conducted by the authors to know the maximum displacement of rotor at the resonance points.

The author William D. Marscher [2], his paper delineates the fundamentals of centrifugal pump rotordynamics in a manner that is intentional to be machinery end user friendly. The main proposals were described in simply comprehensible expressions. The investigation and testing options was expressed in summarized manner.

The author S. Gopalakrishnan et al. [3], in the present paper researched about the Dry and Wet critical speed of centrifugal pump for different wearing rings under different diametral clearances. He conducted the experiment with Byron Jackson 2x3x111/2GSJA, two stage process pumps. He derived simplified equations for the calculation of the effect of the fluid gap, Lomakin mass, damping for smooth and grooved rings. The Lomakin effect decrease as the clearance opens up due to wear was revealed by the author. Thus, the centrifugal pump critical speed can reduce after several hours of working. The results are validated by measurements on a test pump. The author illustrated model computations for a typical multistage boiler feed pump in this research.

Harisha. S. and Y. J. Suresh [4], In this research paper authors estimated the critical speed of multistage centrifugal pump rotor and margins of separation between critical speed and the rotor operating speed. He made torsional analysis using Ansys tool. From the lateral analysis he concluded that there is no resonance noticed in between the speed of 25% of operating speed to 125% of operating speed. And from torsional analysis he showed that the shaft of pump, the coupling and the shaft of motor as single unit and they are unrestricted from any torsional critical speed as per the API 610 11th edition criterion.

Deepak Srikrishnanivas [5], this present work deals with FEA modelling, analysis and development of critical speed. In this study the natural frequencies are obtained for 0 - 2000rpm speed and harmonic analysis has carried out between the speed ranges 0 - 1000rpm for sample rotor model by using Ansys and compared with Dyrobes software. The author also made modal and unbalance response analysis on RM12 Jet engine rotor using Ansys & Dyrobes. He analyzed a case study of potential improvement in modeling approach of RM12. In this case study Eigen frequencies were extracted for first 15 modes at 30000 rpm.

3. OBJECTIVES

1. To determine the natural frequencies of horizontal multi stage centrifugal pump rotor.
2. To determine the Dry & Wet critical speeds of horizontal multistage centrifugal pump rotor and determine the mode shapes at these speeds.
3. Make an unbalance response analysis of a rotor in order to calculate rotor displacement and the forces acting on the rotor supports that are caused due to pump rotor imbalance.

4. METHODOLOGY

•Create a modeling of pump rotor by using Ansys APDL batch file commands.
•The bearings are introduced at the ends of the rotor.
•Pump impellers are considered as point mass for analysis point of view.
•Assigning the properties of rotor, bearings and dampers for developed model.
•Applying the loads and constraints.
•Analyze the finite element model of pump rotor using Ansys.
•Analytical solution is carried out using Dunkerley's method.
•Validating the analytical results with Ansys results.

5. MATHEMATICAL MODELS AND PROPERTIES FOR PUMP ROTOR

5.1 Dunkerley's Model for 'Dry' Critical Speed of Pump Rotor

This method is semi-empirical which gives approximate results and it is used when diameter of the shaft is uniform. According to Dunkerley's method a shaft is subjected to number of point load and natural frequency of rotor is given by [9]

$$f_n=\frac{1}{2\pi}\sqrt{\frac{g}{\delta_1+\delta_2+\delta_3+\dots+\delta_n}} \quad \text{Hz} \qquad (1)$$

Where,
f_n= Natural frequency of shaft in Hz
g= acceleration due to gravity in N/m²
δ= deflection of shaft due load in m

5.2 Effect of Bearing Damping

The characteristic equation is given by [8]

$$\left(\frac{\lambda}{\omega_n}\right)^3+\frac{N+1}{2\zeta N}\left(\frac{\lambda}{\omega_n}\right)^2+\left(\frac{\lambda}{\omega_n}\right)+\frac{1}{2\zeta N}=0 \qquad (2)$$

From above equation $(\lambda/\omega_n)_1$= -A is a negative real root and $(\lambda/\omega_n)_{2,3}$= -B ± i C is two complex conjugate roots with negative real part.

The frequency of the damped free precession is [8]

$$\omega_d = C\,\omega_n \qquad (3)$$

Where,
'C' is the imaginary component

5.3 Unbalance Response for Centrifugal Pump Rotor

The unbalance value permitted for the pump rotor can be calculated by using the following formula [1].

$$U_{per}=\frac{9549 * G_b * W}{n} \quad \text{in g.mm} \qquad (4)$$

Where,
W = Weight of rotor in Kg
G_b = Balancing quality grade as per ISO-1940 = 6.3 (for pump rotor)
n = Operating speed in rpm
As per API 610 11th edition (Para I.1.4)
The equation for total unbalance value is given by

$$U = 4 * U_{per} \quad \text{in g.mm} \qquad (5)$$

5.4 Properties of Pump Rotor

Table 1 Properties of the pump rotor model

Elements	Parameter	values
Shaft	Length	1.5 m
	Outer diameter	0.05 m
	Inner diameter	0.0 m
	Poisson's ratio (υ)	0.3
	Density (ρ)	7800 Kg/m³
	Young's Modulus(E)	$2E^{11}$ N/m²
	Shear Modulus (G)	7.6 E^{10} N/m²
Disk	Outer diameter	0.46 m
	Inner diameter	0.05 m
	Thickness	0.03 m
	Diametral Inertia(Id)	0.5171 Kg.m²
	Polar Inertia(Ip)	1.0283 Kg.m²
Bearings	For rigid bearings Stiffness(Kxx, Kyy)	$1E^{10}$ N/m
	For flexible Bearings stiffness (Kxx, Kyy)	$4E^6$ N/m
Dampers	Damping coefficient(Cxx, Cyy)	2000 Nsec/m

6. FE MODELING OF PUMP ROTOR

6.1 Modelling of Rotor

The centrifugal pump rotor is modelled using BEAM188 elements. The entire pump rotor of Ansys model is same as that of Dyrobes model. The keyoption for the BEAM188, KEYOPT (3) = 2 is chosen for the elements modelled under the general portion and stiffness portion. The KEYOPT (3) signifies the element behavior and the keyoption value '2'represents that the component is build up by means of quadratic shape functions [6].

6.2 Modelling of Impellers

For this analysis, blade masses are included in the FE model as a concentrated point mass and it represents the total mass of disk at each and every stages of pump. The point masses can be modeled by means of MASS21 component consists its rotary inertia option activated [6].

6.3 Modelling of Bearings

There are two bearings are used in this analysis and these bearings are developed as linear isotropic bearings by means of COMBI 214 elements. The COMBI 214 elements are expressed in the plane parallel to YZ plane. Therefore, the DOF's of these elements are in UY and UZ direction [6].

6.4 Loads

Only for unbalance response analysis, an unbalance distribution has been assigned to the middle rotor. These unbalances are represented as forces acting in the two directions perpendicular to the spinning axis [5].

6.5 Constraints

The nodes of the rotor are constrained in axial and torsional directions, to avoiding axial and twisting movements of the rotor respectively. The bearing nodes are fixed in all direction. Hence, these nodes are constrained in axial as well as in torsional direction [5].

Fig.1 FE model of pump rotor

7. RESULTS AND DISCUSSIONS

7.1 Model Analysis of Pump Rotor

In the present investigation the numbers of eigenfrequency analyses are carried out on the centrifugal pump rotor model. The investigation is carried out with speed range in between 0 rpm to 4000 rpm through an increment of 50 rpm.

Table 2 Eigen frequency comparison of pump rotor at 0 rpm under dry state

Mode No	Frequency in Hz From Ansys	Frequency in Hz From Dyrobes	Operating speed in rpm	Margin (%)	Ratio (Ansys/ Dyrobes)
1	13.226	13.241	2000	60.32	0.999
2	13.226	13.241		60.32	0.999
3	82.068	82.343		146.20	0.996
4	82.068	82.343		146.20	0.996

Table 3 Eigen frequency comparison of pump rotor at 4000 rpm under dry state

Mode No	Frequency in Hz From Ansys	Frequency in Hz From Dyrobes	Operating speed in rpm	Margin (%)	Ratio (Ansys/ Dyrobes)
1	13.149	13.162	2000	60.55	0.999
2	13.304	13.317		60.08	0.999
3	50.287	50.205		50.86	1.00
4	133.93	133.88		301.79	1.00

Table 4 Eigen frequency comparison of pump rotor at 0 rpm under wet state

Mode No	Frequency in Hz From Ansys	Frequency in Hz From Dyrobes	Operating speed in rpm	Margin (%)	Ratio (Ansys/ Dyrobes)
1	12.518	12.532	2000	62.44	0.998
2	12.518	12.532		62.44	0.998
3	73.329	73.953		119.98	0.991
4	73.329	73.953		119.98	0.991

Table 5 Eigen frequency comparison of pump rotor at 4000 rpm under wet state

Mode No	Frequency in Hz From Ansys	Frequency in Hz From Dyrobes	Operating speed in rpm	Margin (%)	Ratio (Ansys/ Dyrobes)
1	12.457	12.468	2000	62.62	0.999
2	12.580	12.592		62.26	0.999
3	44.897	44.634		34.69	1.00
4	119.78	119.31		259.34	1.00

7.2 Critical Speed and Campbell Diagram Analysis

In the present investigation, numbers of eigenfrequency analyses are carried out on the centrifugal pump rotor model. The investigation is carried out with speed range in between 0 rpm to 4000 rpm through an increment of 50 rpm with several load steps. The critical speed can be obtained where the excitation line intersects with the natural frequency lines, which is shown in Campbell diagram. The Campbell diagrams are shown in below fig.2 and 3

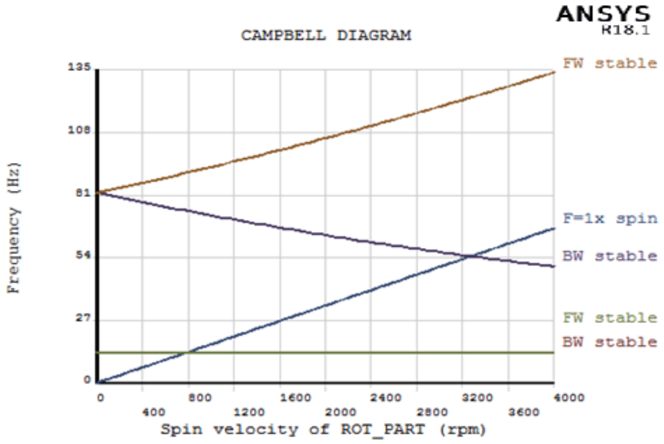


Fig.2 Campbell diagram for pump rotor for rigid bearings from Ansys

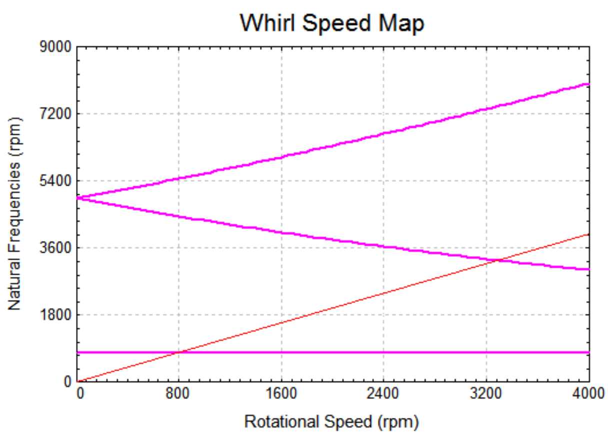


Fig.3 Campbell diagram for pump rotor for rigid bearings from Dyrobes

The figure 2 & 3 represents the Campbell diagram. It is examined from the Campbell diagrams, that the forward whirl frequencies increase and the backward whirl frequencies decreases with increase in rotational speed. The Campbell diagram extracted from Ansys gives more information. It presents the type of whirling as well as the stability of the system for each mode. The Campbell diagram extracted from Dyrobes does not give any similar information. On comparison of Campbell diagram, it can be observed that both plots have similar modal curves which indicates that the analysis result obtained from Ansys are good in agreement with the Dyrobes result. The analytical results are compared with the Ansys results.

Table 6 Dry critical speed comparison of pump rotor

Sl. No	Mode No	Analytical Dry Critical speed in rpm	Dry Critical speed by ANSYS in rpm	Dry Critical speed by Dyrobes in rpm	Error In %
1	1	831	793.56	794.49	4.71
2	2	3324	3290.63	3348	1.01

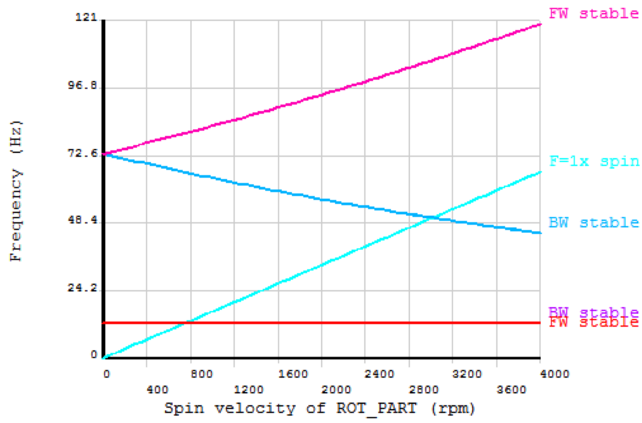


Fig. 4 Campbell diagram for pump rotor for flexible bearings from Ansys

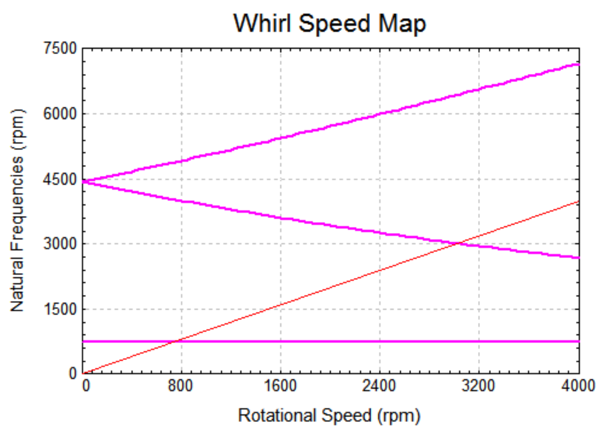


Fig.5 Campbell diagram for pump rotor for flexible bearings from Dyrobes

Table 7 Wet critical speed comparison of pump rotor

Sl. No	Mode No	Analytical Wet Critical speed in rpm	Wet Critical speed by ANSYS in rpm	Wet Critical speed by Dyrobes in rpm	Error In %
1	1	783.6	751.08	751.93	4.32
2	2	3134.4	3026.58	3039.09	3.56

International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056
Volume: 08 Issue: 08 | Aug 2021 www.irjet.net p-ISSN: 2395-0072

7.3 Mode Shapes

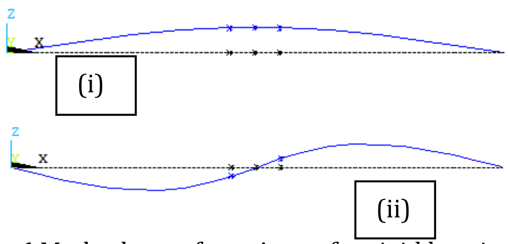


Fig.6 Mode shapes from Ansys for rigid bearings at critical speed (i) 793.56, (ii) 3290.63

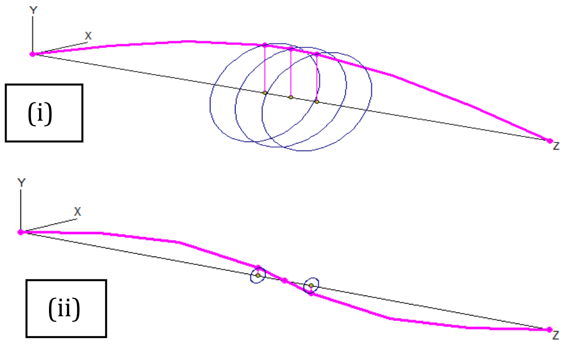


Fig.7 Mode shapes from Dyrobes for rigid bearings at critical speed (i) 794.49, (ii) 3348

The figure 6 & 7 depicts the mode shapes of centrifugal pump rotor for different dry critical speed which are obtained from Ansys and Dyrobes.

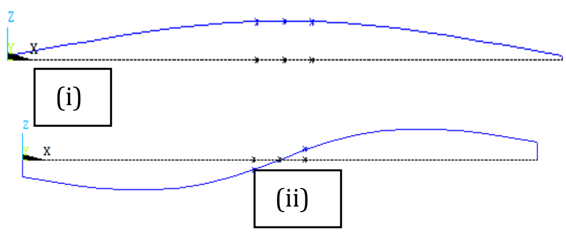


Fig.8 Mode shapes from Ansys for flexible bearings at critical speed (i) 751.08, (ii) 3026.58

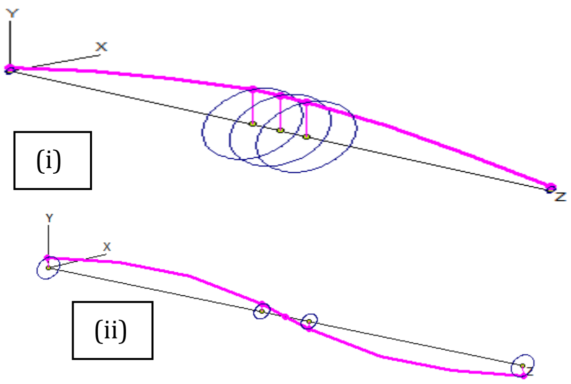


Fig.9 Mode shapes from Dyrobes for flexible bearings at critical speed
(i) 751.93, (ii) 3039.09

The Fig. 8 and Fig.9 are indicating the mode shapes for different wet critical speeds which are obtained from Ansys and Dyrobes.

7.4 Unbalance Response Analysis

A harmonic analysis is performed on the pump rotor model for the applied unbalance loading between the speed range 0 to 1000 rpm with increment of 50 rpm. The results of maximum displacement of the pump rotor and the maximum force transmitted through the bearings are determined from this analysis. The maximum amplitude and bearing force are expected at critical speeds. The total unbalance is 0.002764 Kg.m and this unbalance is applied as the force input to the pump rotor model. The total unbalance is calculated from the formula as specified by API610 and it is applied to the disk node in the FE model. To examining the present analysis, total unbalance value is applied to the 2nd impeller station since largest displacement is notices at second impeller. The results of the maximum displacement of the pump rotor obtained from the Ansys and Dyrobes are presented in the following figures.

a. Maximum Displacement of the Pump Rotor at Midspan

The maximum displacement of the pump rotor obtained from the Ansys and Dyrobes are shown in below Figures 10 & 11.

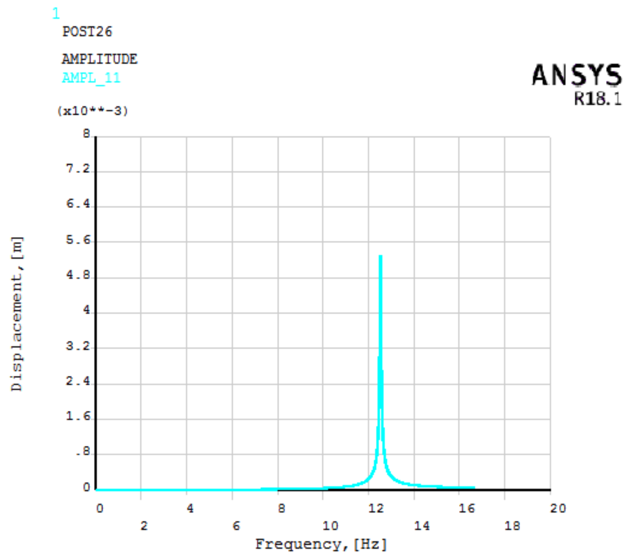


Fig.10 Maximum response of pump rotor from Ansys

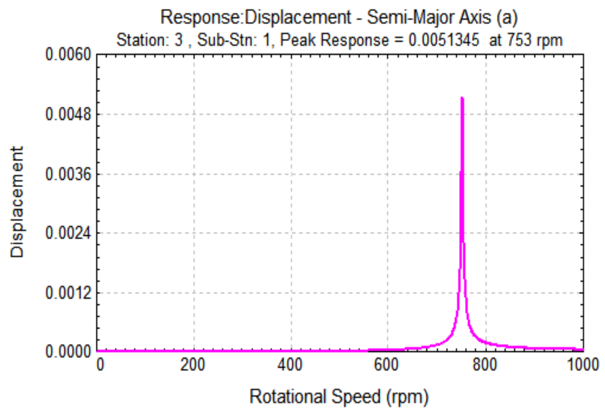


Fig.11 Maximum response of pump rotor from Dyrobes

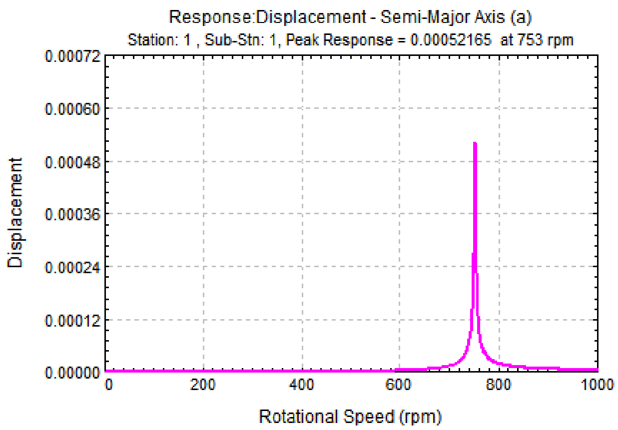
From both figures 10 and 11, it is observed that the maximum displacement of the pump rotor is occurred at the middle disc location and their comparisons are presented in the below table 8

Table 8 Comparison of maximum displacement of rotor at midspan

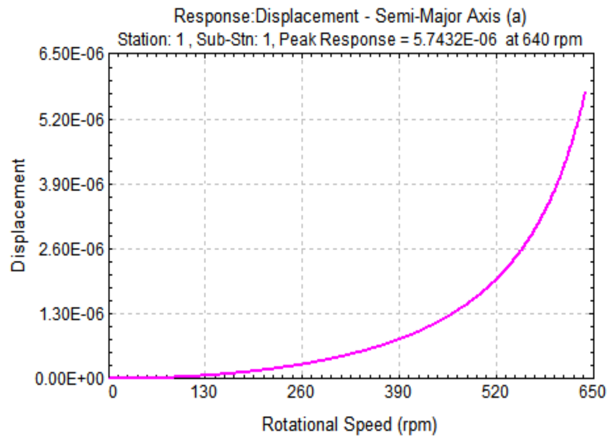
Response of the rotor	Ansys	Dyrobes	Ratios
Maximum displacement of the rotor, m	0.00528	0.00513	1.02
Critical speed in rpm	751.2	753	0.99

b. Maximum Displacement of the Pump Rotor at Bearing Location

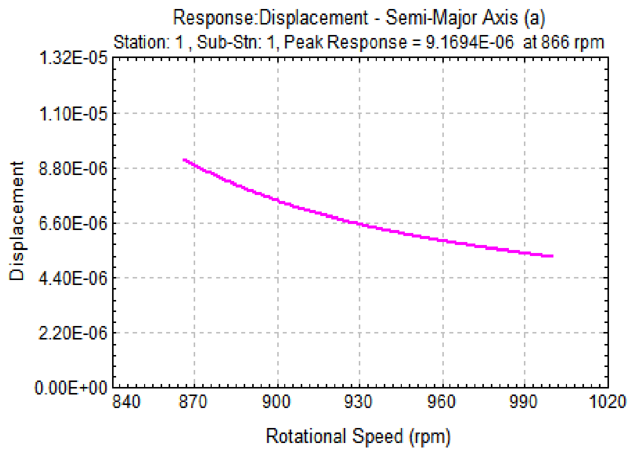
The maximum displacement of pump rotor at bearing location near the first critical speed, below and above the first critical speed is obtained from Ansys and Dyrobes are shown in the below figures.



(a)



(b)



(c)

Fig.12 Maximum deflection at bearing location (a) near critical speed, (b) below critical speed and (c) above critical speed

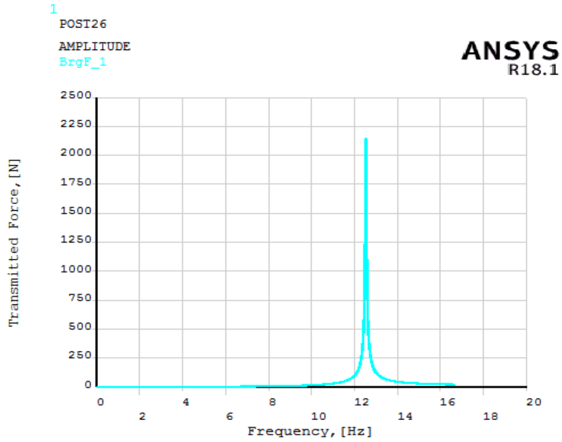
The figure 12(a) depicts the maximum displacement of pump rotor at bearing location near the critical speed. The results obtained from both Ansys and Dyrobes good agree with each other. The figures 12(a) & (b) illustrate the highest displacement of the rotor at bearing location with speed above and below the critical speed. And these deflections are compared with the Minimum diametral clearance specified by API 610 which is presented in Table 9

Table 9 Comparison of displacement of rotor with API610 clearance at bearing location

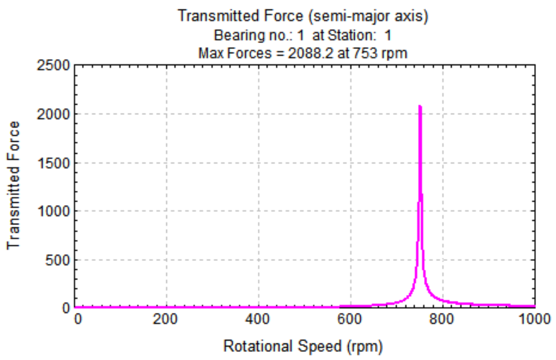
Diameter of rotating member in m	Minimum diametral clearance as per API610 in m	Dyrobes Maximum amplitude of vibration at bearing location in m	speed in rpm
0.05	0.00028	0.00000574	640
0.05	0.00028	0.00052	753
0.05	0.00028	0.00000916	866

c. Maximum Bearing Force of Pump Rotor

The maximum force transmitted through the bearings near the first critical speed obtained from Ansys and Dyrobes result are shown in the below figures



(a)



(b)

Fig.13 Maximum bearing force of pump rotor from (a) Ansys, (b) Dyrobes

The figure 13(a) and figure 13(b) depicts the maximum force transmitted to bearings near critical speed. The maximum bearing load obtained from Ansys and Dyrobes results are compared in the below table 10.

Table 10 Maximum force transmitted to bearing

Response of the rotor	**Ansys**	**Dyrobes**	**Ratios**
Maximum bearing force in N	2130	2088.2	1.02
Critical speed in rpm	751.2	753	0.99

8. CONCLUSIONS

The following conclusions are drawn from the above study,

•The natural frequencies of the centrifugal pump rotor for several rotational speeds are calculated. The natural frequencies gained from Ansys and Dyrobes model are close to each other. It is also observed that separation margin is also well as specified by API.

•From the critical speed and Campbell diagram analysis it is concluded that, the dry and wet critical speeds for the centrifugal pump rotors are identified. The Analytical critical speeds are less than 10% as specified by API 610 from the Ansys and Dyrobes results.

•From unbalance response analysis, the maximum displacement of the centrifugal pump rotor is noticed. It is observed that, at first critical speed the maximum displacement of the pump rotor is occurred at the middle disc location.

•The displacement at bearing location is also determined in this study. The maximum amplitude of deflection of rotor below and above the critical speed at bearing locations is less than the 35% of the diametric clearance at that location as specified in API 610 guidelines. but at critical speed the displacement of pump rotor is greater than the 35% of diametric clearance, it indicates if rotor runs at critical speed bearings will get damages and hence system.

•In this present study maximum force transmitted through the bearings at the critical speed is obtained from Ansys and Dyrobes and the results obtained from both Ansys and Dyrobes good agree with each other.

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