Analytical, Experimental and Numerical Investigation of Effect of Bearing Parameters on Critical Speed of Rotating Shaft

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***Abstract*—A shaft always pass through one or more critical speed as it accelerates or decelerates. While operating at critical speed shaft vibrates at large amplitude which can cause damage. As at critical speed resonance occur, shaft must be designed to pass quickly through this critical speed. Different bearing parameters leads to different bearing stiffness. Therefore, it is necessary to analyze the effect of bearing and its parameters on the critical speed of the shaft. This paper incorporates experimental investigation of effect of different bearing parameters on critical speed of a rotating shaft-disk system. The experimental setup has been developed to study effect of bearing parameters. The study involves determination of natural frequency analytically and experimentally followed by modal analysis in ANSYS APDL. Experimental analysis has been carried out to determine the actual critical speed of shaft with the help of state-of-the-art four channel Adash VA4 Pro FFT analyzer platform along with data acquisition system and electronic instrumentation. Modal analysis in ANSYS APDL has been carried out with two approaches.**

***Index Terms*—Critical speed, Universal Testing Machine, Stiffness, FFT analyzer, ANSYS APDL.**

I. INTRODUCTION

There is always a demand for speeds higher than critical speed of machine in various industrial applications such as turbines, compressors, electric motors, and pumps, in which a heavy rotor is mounted on a lightweight, flexible shaft that is supported in bearings. To achieve this higher speed, identification of critical speed is necessary. Analytically the critical speed of shaft depends on length of shaft, modulus of elasticity, diameter of the shaft and end conditions. In the analytical method of calculation of critical speed of the shaft, the shaft is assumed to be simply supported or fix-fix ends depending on the type of bearing. But experimentally there is some effect of bearing, as bearing also has its own stiffness which is not considered in the analytical method. Therefore, it is essential to consider the effect of bearing parameters on critical speed of the shaft. In this case the bearing parameters such as, outer diameter, bearing width, type of bearing are generally considered. The lowest rotational frequency that causes an object to vibrate at one of its natural frequencies is known as its first critical speed. An object may have an infinite number of critical speeds, but the most important one for engineers to consider is the first one. Some rotating objects are designed to operate below their first critical speed, but many rotate above this speed. As long as the rotational speed passes quickly through the critical speed, this should not have a significant effect on the life of the rotating machine. Issues can occur, however, if an object is designed to rotate at its critical speed, the resulting vibrations could cause the machine to break. Therefore, accurate identification of critical speed is essential.

II. LITERATURE SURVEY

There are many procedures available to balance rotating systems. Many investigators have studied and developed new methods for balancing of rotors. Now-a-days the influence coefficient method and modal balancing method are the two major balancing methods that are used extensively. The influence coefficient method requires vibration data (Phase angles and vibration values) measured at different locations on a bearing, at a single speed. Many

investigators have studied and developed new methods for critical speed of shaft. The first known rotor model used for a consistent study of the dynamic behavior and vibration of structures was published in 1895 by German engineer August Foppl. This model consisted of single disk, centrally located on a shaft of constant circular cross-section and with un-damped rigid bearings placed at each end of the shaft. Hota and Vakharia [1] presented a practical understanding of how a shaft vibrates, and examined issues that affect vibration. They have also presented the synchronous critical speeds of rotor-bearing systems. The majority of attention is on the unbalance excitation of natural frequencies with whirling and spinning in same direction. This paper helps to learn the basic modes of vibrations and predicting critical speed by practical approach and also concludes some important practical recommendations on rotor-bearing design so that unique critical speed situation may be avoided. B. K. Takle [2] derived relationship between natural frequencies and shaft diameter using graphical approach. Effect of self-weight of shaft on natural frequency has been discussed. Naveena and Suresh [3] carried research on evaluation of the Lateral critical speeds using ANSYS® software by plotting Campbell diagrams and is compared with theoretical calculations. Forced response analysis has been conducted to know the peak amplitudes at the resonance points. The rotor has been designed and analyzed by using the guidelines as per API 610 standards. Korody Jagannath.

III. EXPERIMENTAL INVESTIGATION

Experimental Setup

The experimental setup for analysis of critical speed of shaft is given in figure 1. It consists of shaft, coupling, bearings, central bearing adapter, electric motor and weight attached at central adapter. The experimental setup has 3 HP motor with speed of 1440 rpm. It has been made to operate the motor with the help of VFD (Variable Frequency Drive) at above 1440 rpm. The shaft of electric motor connected to another shaft with the help of jaw type coupling. Figure 1 also shows accelerometer for vibration velocity measurement on bearing and photocell for speed measurement.

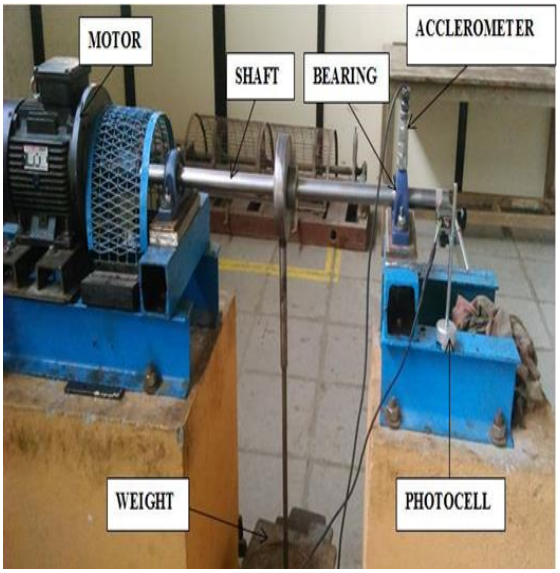

Figure 1 Experimental Setup

IV. BEARING SELECTION

Different bearing parameters leads to different bearing stiffness. Therefore it is necessary to analyze the effect of bearing parameters on critical speed of the shaft. The bearings with different parameters given in Table 1 have been selected to analyze effect of bearing stiffness on critical speed of shaft.

Table 1 Types of Bearing with Dynamic Load Capacity.

Sr. No.	Bearing Type	Bearing Designation (SKF)	Dynamic load (N)	D (mm)	B (mm)
1.	Deep groove ball bearing single row	6206	19500	62	16
2.	Deep groove ball bearing single row	6306	29600	72	19
3.	Self-aligning ball bearing	1307EKTN9	26500	80	21

V. DETERMINATION OF BEARING STIFFNESS.

Different bearing parameters leads to different stiffness, hence to analyze the effect of bearing, it is necessary to calculate stiffness of bearing. The bearing stiffness is calculated on Universal Testing Machine (UTM) as given in figure 2. To measure the stiffness on UTM one tool as shown in figure 3 is prepared to apply the radial load on bearing.

Table 2 Stiffness of Selected Bearings

Bearing No.	Bearing Type	Bearing Stiffness (N/m)
6206	Deep groove ball bearing	50x106
6306	Deep groove ball bearing	31.25x106
1307EKTN9	Self-aligning ball bearing	12x106

VI. EXPERIMENTATION TESTING STRATEGY ADOPTED.

The critical speed of developed shaft rotor setup is calculated analytically. The critical speed for 63 kg load determined using analytical equations is 1600 rpm. To determine critical speed experimentally, following strategy has been adopted:

1. Measuring vibration amplitude levels of both, drive end side and non-drive end side bearings at constant load of 63 kg.
2. Taking velocity frequency spectrums close to 1600 rpm with constant load of 63 Kg in vertical direction of both drive end side and non-drive end side bearings by using FFT analyzer.
3. Experimental identification of critical speed of shaft by comparing maximum vibration amplitude in vertical direction on both drive end side and non-drive end side bearings at constant load of 63 kg.

The vibration amplitude levels are measured in vertical direction of both end bearings by using FFT Analyzer as shown in figure 4. The input to the FFT is given by an accelerometer. During experimentation velocity in mm/s is measured in vertical directions on both drive end and non-drive end bearing.

Figure 2 FFT Analyzer

VII. DEEP GROOVE BALL BEARING (6206)

The vibration RMS velocity values for 6206 bearing has been measured on drive end and non-drive end bearings as shown in Table 3.

Speed RPM	Drive End Velocity (mm/s)	Non-Drive End Velocity (mm/s)	Drive End 1XRPM (mm/s)	Non-Drive End 1XRPM (mm/s)
1300	1.21	0.458	0.242	0.106
1400	1.42	0.527	0.365	0.235
1450	1.22	0.678	0.384	0.221
1500	2.09	1.41	1.116	1.084
1520	2.37	1.47	1.189	0.985
1530	2.22	1.66	1.229	1.251
1550	2.39	1.34	1.44	1.231
1560	2.55	1.76	1.521	1.45
1580 (Critical Speed)	2.78	1.98	1.623	1.478
1590	2.38	1.49	1.312	1.213
1600	2.32	1.36	1.373	1.242
1670	2.20	1.23	0.954	0.929

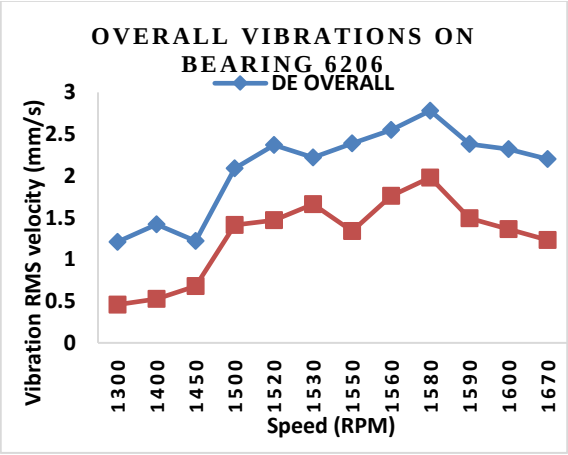

Figure 3 Vibration Amplitude vs. Speed in RPM for 6206 bearing

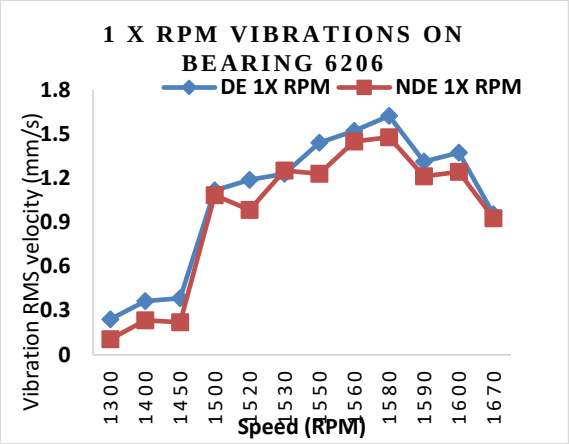

Figure 4 1xRPM Vibration Amplitude vs. Speed in RPM for 6206 bearing

Figure 3 and figure 4 shows overall vibration amplitude at drive and non-drive end and vibration amplitude at 1xrpm. Both these graphs show that, maximum vibration amplitude is at 1580 rpm which is experimental critical speed due to effect of 6206 bearing.

Deep Groove Ball Bearing (6306)

Table 4 shows the vibration RMS velocity values for deep groove ball bearing (6306).

Table 4 FFT Readings for 6306 bearing

Speed RPM	Drive EndVelocity (mm/s)	Non-Drive EndVelocity (mm/s)	Drive End 1XRPM (mm/s)	Non-Drive End 1XRPM (mm/s)
1300	1.42	1.07	0.669	0.617
1400	2.15	1.69	1.5	1.16
1445	4.31	2.13	1.5	1.432
1500	4.13	3.28	2.45	2.444
1550	3.77	3.06	2.366	2.315
1570 (Critical Speed)	4.80	3.66	2.525	2.57
1580	4.00	2.94	2.477	2.393
1600	4.21	3.34	2.161	2.083

From Table 4 it is clear that maximum vibration amplitude is at 1570 RPM. Thus the experimental critical speed value is 1570 RPM for 6306 bearing. The graph is plotted between overall vibration velocity values vs. RPM as shown in figure 7. Also graph is plotted between 1xRPM vibration velocity values vs. RPM as shown in figure 8. From both these graphs it is clear that maximum peak obtained is at 1570 RPM.

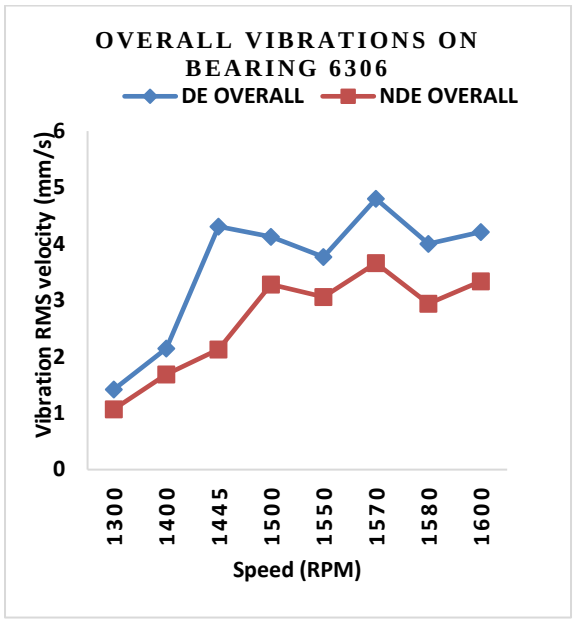

Figure 5 Vibration Amplitude vs. Speed in RPM For 6306 bearing

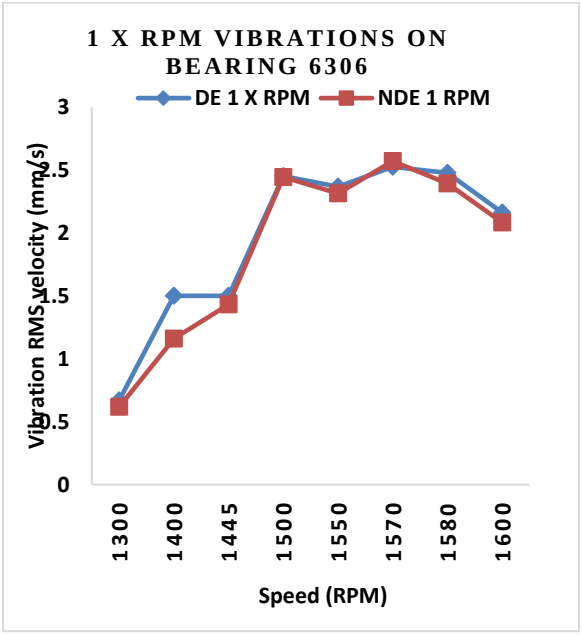

Figure 6 1xRPM Vibration Amplitude vs. Speed in RPM for 6306 bearing.

SELF-ALIGNING BALL BEARING (1307EKTN9)

Table 5 shows the vibration RMS velocity values for 1307EKTN9 bearing.

Table 5: FFT Readings for 1307EKTN9 bearing

Speed RPM	Drive End Velocity (mm/s)	Non-Drive End Velocity (mm/s)	Drive End 1XRPM (mm/s)	Non-Drive End 1XRPM (mm/s)
1300	1.96	0.791	0.507	0.413
1400	2.61	0.767	0.79	0.465
1500	2.86	0.917	0.81	0.51
1530	3.03	1.11	0.86	0.589
1550 (Critical Speed)	2.99	0.911	0.825	0.569
1570	1.96	0.791	0.507	0.413

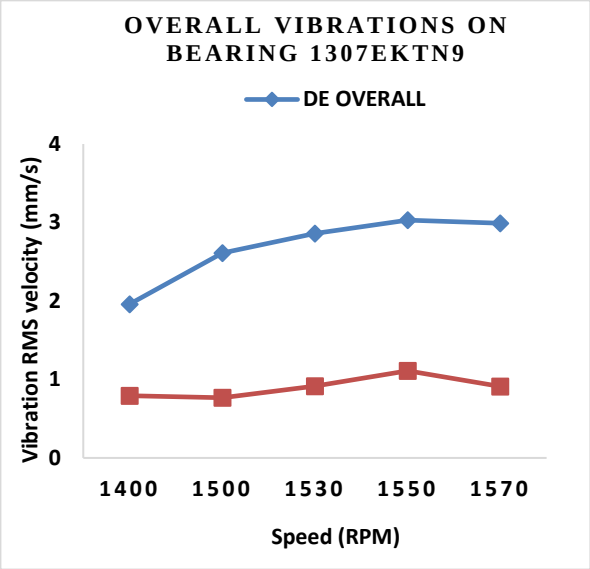

Figure 7 Vibration Amplitude vs. Speed in RPM for 1307EKTN9 bearing.

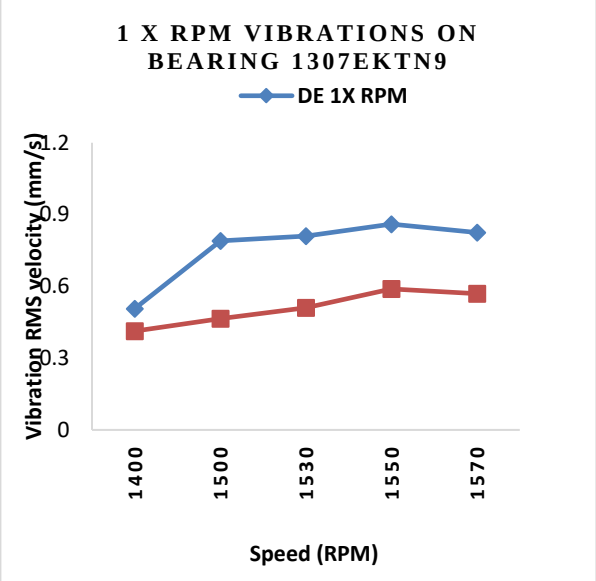

Figure 8 1xRPM Vibration Amplitude vs. Speed in RPM for 1307EKTN9bearing

The figure 9 shows graph of overall vibration RMS velocity versus rpm at drive and non-drive end. The graph of peak at 1xrpm versus rpm at drive and non-drive is given in figure 10. From both these graphs it is clear that maximum peak obtained at 1550 rpm.

Overall Summary

For all types of bearings vibration readings have been taken in vertical directions because stiffness of bearing measured on universal testing machine is in the vertical direction. It is not possible to measure the stiffness of bearing in horizontal direction on universal testing machine as it requires separate fixture. Vertical vibration values are considered as vertical bearing stiffness is known. Also for all types of bearing critical speed value decided experimentally from overall vibration readings and vibration peak obtained at 1x RPM value.

VII. FINITE ELEMENT ANALYSIS INVESTIGATION.

For modal analysis, shaft is model by using BEAM 188 element, figure 11 shows shaft model in ANSYS APDL.

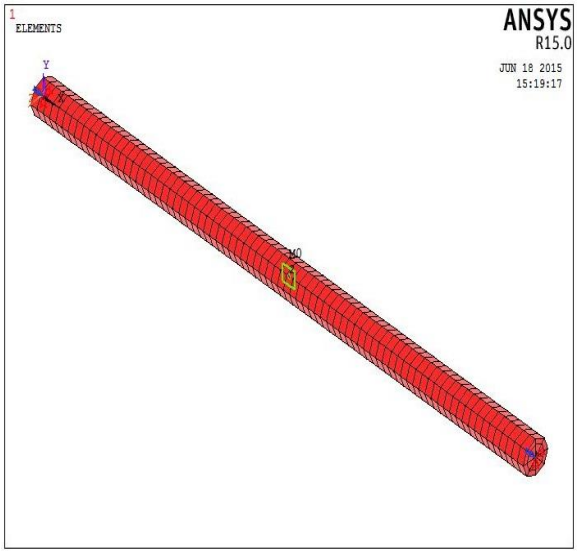

Figure 9 Shaft Model in ANSYS APDL

To apply the load at center of shaft, MASS 21 element is used and boundary conditions used as a simply supported beam that gives different natural frequencies as shown in Table 6.

Table 6 Frequency in ANSYS APDL

Sr. No.	Time/Frequency (Hz)	Speed(rpm)
1	26.225	1573.5
2	370.17	22210.2
3	654.14	39248.4
4	1007.1	60426
5	1014	60840
6	1941.9	116514
7	2548.6	152916
8	2609.3	156558
9	3147.1	188826
10	3153.9	189234

ANSYS gives first natural frequency as 26.225 Hz (1573.5 RPM) which is a critical speed of shaft for this orbit plot as shown in figure 12.

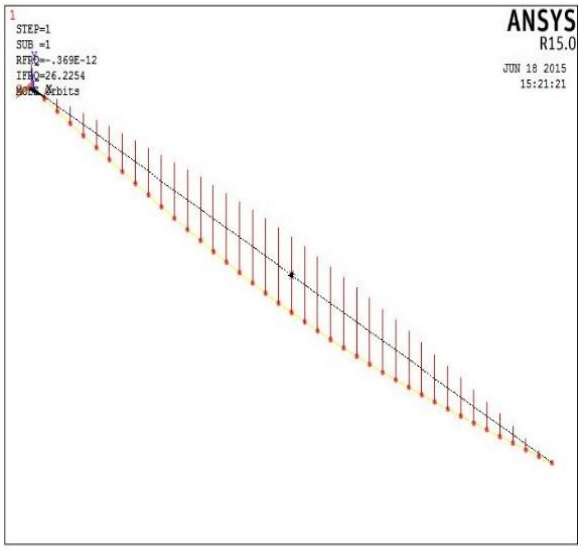

Figure 10 Orbit Plot at Frequency of 26.225 Hz

Numerical Result with Bearing Stiffness

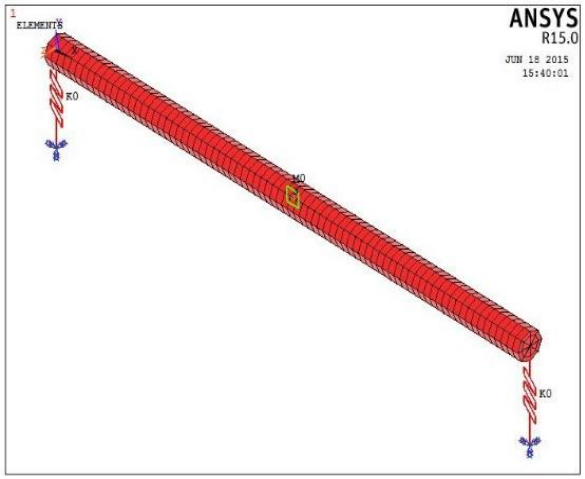

Figure 12 ANSYS Model with Bearing Stiffness

Figure 12 shows the ANSYS APDL model to calculate the natural frequency of shaft with bearing stiffness. Here shaft is modeled using BEAM 188 elements and for bearing combination element i.e. COMBIN 14 element is used, this element allows stiffness and damping characteristics in one direction. To apply load at center of shaft, MASS 21 element is used. Table 7 shows the critical speed results at different bearing stiffness. It shows that as bearing stiffness increases the critical speed also increases.

Table 7 Critical Speed at Different Bearing Stiffness (ANSYS)

Bearing No.	Bearing Type	Bearing Stiffness (N/m)	Critical Speed	
			Hz	RPM
6206	Deep groove ball bearing	50x106	25.99	1559.7
6306	Deep groove ball bearing	31.25x106	25.85	1551
1307EKTN9	Self-aligning ball bearing	12x106	25.22	1513.2

VIII. ANALYTICAL INVESTIGATION

As all the bearing stiffness values have been determined, the result obtained by analytical method in MATLAB are shown in Table 8.

Table 8 Critical Speed at Different Bearing Stiffness (MATLAB)

Bearing No.	Bearing Type	Bearing Stiffness (N/m)	Critical Speed	
			Hz	RPM
6206	Deep groove ball bearing	50x106	26.26	1575.6
6306	Deep groove ball bearing	31.25x106	26.13	1567.8
1307EKTN9	Self-aligning ball bearing	12x106	25.56	1533.6

It is seen that results obtained in MATLAB analysis are in line with those of ANSYS analysis results.

IX. COMPARISON OF EXPERIMENTAL, NUMERICAL AND ANALYTICAL RESULTS

One of the most useful tools available to anyone evaluating the rotor-dynamics of a rotor bearing system is the un-damped critical speed map. The critical speed map allows to completely quantifying the possible critical speed of machine. It does not necessarily reveal anything about the true critical speed frequencies. It is used to show how various changes in the bearing parameter will affect the critical speed. As different bearing parameter contribute to different bearing stiffness, it results in change in critical speed. Table 9 shows the comparative results obtained by ANSYS, experimental and analytical analysis of critical speed for different bearing stiffness.

Table 9 Comparison of Results

Bearing Designation	Bearing Stiffness (N/m)	Analytical (MATLAB) Critical Speed (RPM)		Numerical (ANSYS) Critical Speed (RPM)		Experimental Critical Speed (RPM)
		With Bearing Stiffness	Without Bearing Stiffness	With Bearing Stiffness	Without Bearing Stiffness	With Bearing Stiffness
6206	$50x10^6$	1575.6	1600	1559.7	1573.5	1580
6306	$31.2x10^6$	1567.8		1551		1570
1307EKTN9	$12x10^6$	1533.6		1513.2		1550

X. CRITICAL SPEED MAP COMPARISON

Figure 14 shows the critical speed map graph for experimental, analytical and numerical APDL results. It also shows that the experimental critical speed values are in line with the analytical and FEM results.

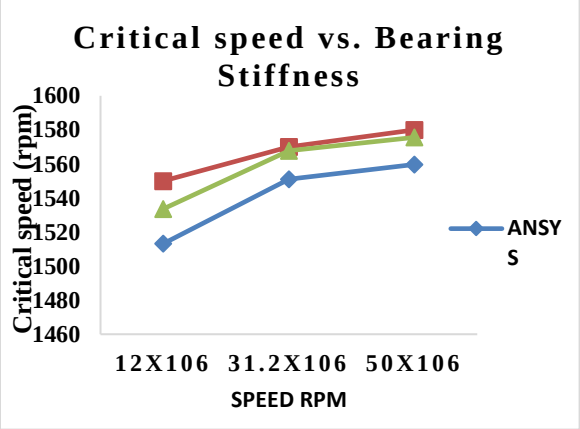

Figure 13 Critical Speed Map Comparison

XI. CONCLUSION

The analytical, experimental and numerical analysis has been carried out on different types of selected bearings to analyze the effect of bearing parameters on critical speed of a rotating shaft. As different bearing parameters contribute to different bearing stiffness, effect of bearing stiffness on critical speed is plotted as given in figure 14. The results show that the critical speed increases with the bearing stiffness.

The modal analysis is carried out in ANSYS APDL without considering the effect of bearing stiffness and also with considering the effect of bearing stiffness. In this analysis COMBIN14 element is used to analyze the effect of stiffness of bearing on critical speed. The results of this modal analysis shows that the critical speed without considering effect of bearing stiffness is 26.225 Hz (1573.5 RPM) and the critical speed decreases if effect of bearing stiffness is considered as given in Table 9.

The analytical, Numerical and Experimental analysis results reveal that it is required to consider the effect of stiffness of bearings; along with the effect of stiffness of shaft; in the determination of critical speed of rotating shaft.

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