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Accurate Identification of Performance for Rotor-Bearing Systems Using the Modified Modelling Under Gyroscopic Effect

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ABSTRACT

The rotor-bearing system of modern rotating machines constitutes a complex dynamic system. The challenging nature of rotordynamic problems have attracted many scientists and engineers whose investigations have contributed to the impressive progress in the study of rotating systems. The purpose of the present paper is to investigate the effects of modal parameters on the noise produced by rotor-bearing systems under gyroscopic effect. To do this, we study reaction force in left and right bearing under gyroscopic effect in rotating machinery with high speed of rotation using modal data. We find modal parameter of modal in experimental part validate with simulation using ANSYS 12., and study effect of mass eccentricity of the rotor on the noise of the bearing are investigated, and the simulation results are presented advanced modelling and simulation techniques; active vibration controls; malfunctions and condition monitoring aspects through the graph of the bending stress with respect time of the bearing for various rotational speeds of the rotor.

Keyword-**Rotor-Bearing, Modelling, Reaction force, Bending stress, Gyroscopic effect.**

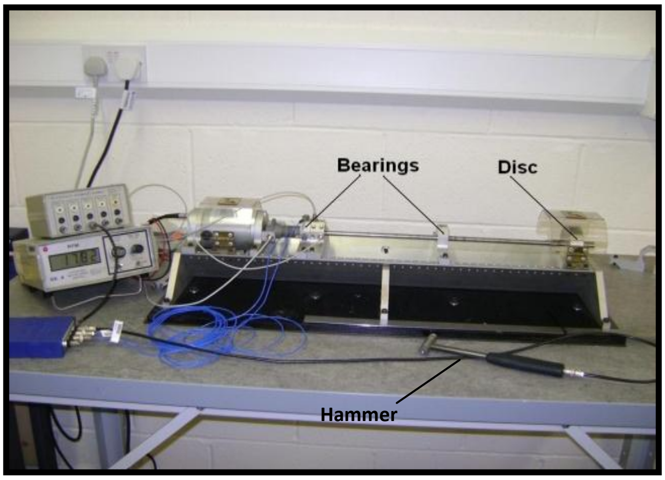
I. INTRODUCTION

The bearings used for supporting rotating machinery are one of the crucial elements by which the safe operation of the machinery is ensured. In recent years, with continuing demands for increased performance, many rotating industrial machines are now being designed for operation at high speed, a trend which has resulted in increased mechanical vibration and noise problems. Many researchers have studied the vibration characteristics of bearings [1–3], but there is relatively little information regarding their modified modelling under gyroscopic effect; (A gyroscope Fig.(1) is a device that can be used to maintain orientation based on the principles of angular momentum. It is a mechanism by means of which a rotor is journeyed to spin around an axis) [4,5]. However, there have been no studies on the effects of design parameters on the noise of rotor-bearing systems. In practice, it is very important to know how much the bearing noise can be [6]. However, there have been no studies on the effects of design parameters on the noise of rotor-bearing systems. In practice, it is very important to know how much the bearing noise can be reduced by design parameters such as bearing width, radial clearance, oil viscosity, mass eccentricity of the rotor, and so on. In other words, it is very important to know what parameters are dominant on bearing noise. It is also expected that [6,7].
The modal properties of the bearing can provide diagnostic information on abnormal phenomena of the rotor-bearing system. For example, if the frequency characteristics .The purpose of the present paper is to investigate the effects of modal parameters on the noise of rotor bearing systems. With the advancement in high-speed machinery and increases in their power/weight ratio, the determination of the rotor dynamic characteristics through reliable mathematical models gains prime importance. The advancement in modern instrumentation and computational capabilities has helped in implementing simulation techniques of these complex models. Modern machinery is bound to fulfill increasing demands concerning durability as well as safety requirements. On-line condition monitoring strategies are becoming increasingly commonplace in a greater range of systems [8,9].

Rotors are structures with special properties due to their rotation (causing e.g. the gyroscopic effect), due to their bearings (fluid film bearings, magnetic bearings) and in many cases due to surrounding fluids (seal forces). Therefore rotordynamics requires special engineering tools although the structural properties of the rotors and their supports could well be modelled by any general finite element program [4,10].

The recent development of magnetic bearings, which are now more and more introduced in industrial applications of turbomachines, required an extension of existing rotordynamic tools to model the specific characteristics of this bearing type and the controllers [8,9&10].

II. METHODS



Picture.1 Experimental setup for the modal testing.

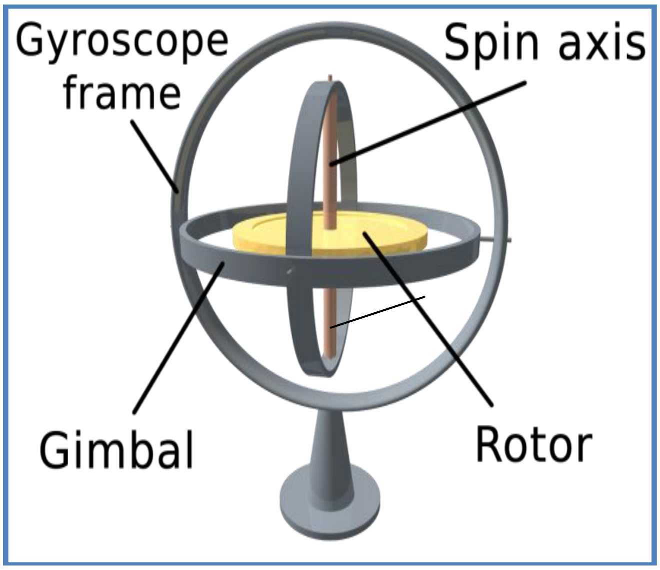


Fig.1 The gyroscopic effect [4,5].

2.1 Equations of motion

The general equations of motion for a multi-degree of freedom vibratory system shown in picture (1), may be written as [10,11]:-

$$[M]\{\ddot{q}(t)\}+[[G+C](\Omega)]\{\dot{q}(t)\}+[[B]+[K](\Omega)]\{q(t)\}=\{F(t)\} \quad \text{.. (1)}$$

$$Y_i(x)=\begin{cases} \dfrac{P\,a\,x}{6EIL}(x^2-l^2); & 0\le x\le l \\ \dfrac{P\,(x-l)}{6EIl}[a(3x-l)-(x-l)^2]; & (l\le x\le l+a) \end{cases} \quad \text{....(2)}$$

Table (1) Defintion of parameter for gyroscopic setup.

	Rotor Dia.	0.01 m		
	P	0.8 kg	P=M*9.81	0.007848KN
	X	0.24 m		
	a	0.24 m		
I=π*d^4/64	I	4.91E-10	MASS MOMENT OF INERTIA	

Table (2) Calculations natural frequency & stiffness of the system before rotation.

Y deflection=1.18E-03			
ω=89.99623 rad/sec		89.99623	rad/s
ω_n=89.99623 rad/sec			
f_n	14.32334486	Hz	
n	859.4006918	rpm	
ω=(k/M)^0.5		k=M*(ω_n^ 2)	
	K	6479.457131	N/m

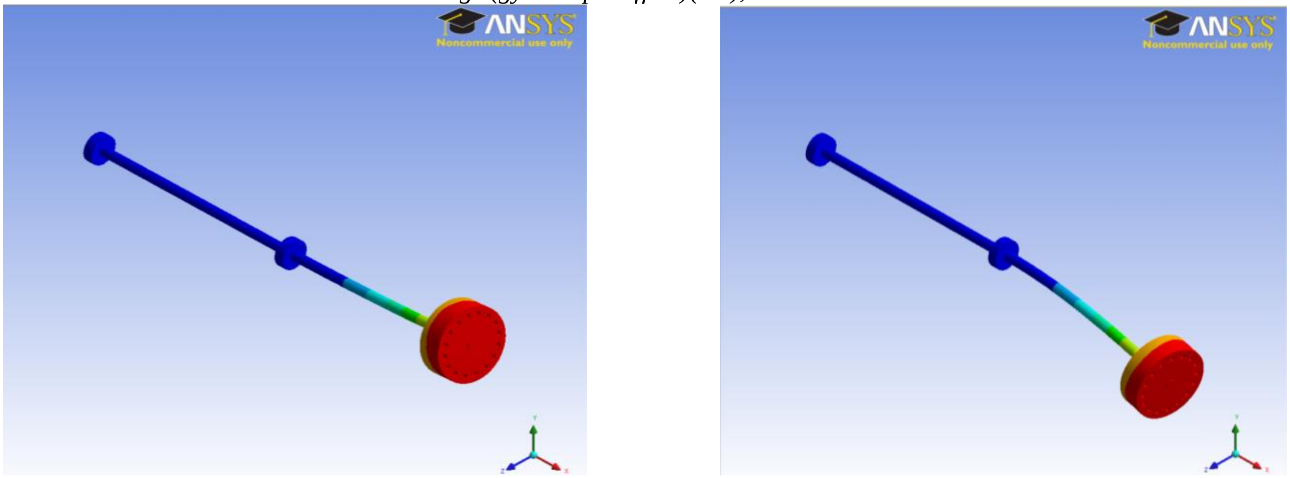
2.2 Imitation model in (ANSYS 12.)

A program has been written in (ANSYS 12),A model of rotor system with an overhung disc with multi degree of freedom (Y and Z directions) has been used to demonstrate the above capability see Fig.(2).Postprocessing commands(/POST1). Applying of gyroscopic effect to rotating structure was carried by using (CORIOLIS) command. This command also applies the rotating damping effect. Another command which was used in input file (SYNCHRO) that Specifies whether the excitation frequency is synchronous or asynchronous with the rotational velocity of a structure in a harmonic analysis; [10,12&13].

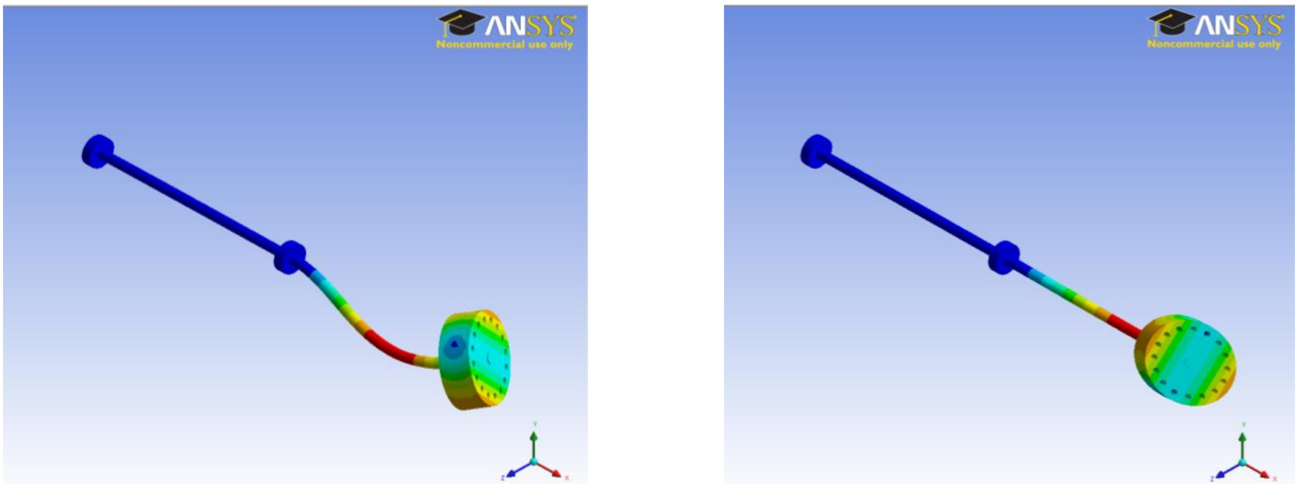
Fig. 2 Finite element model (gyroscopic geometry) ANSYS work bench (three dimensions).

2.2.1 The ANSYS Animation

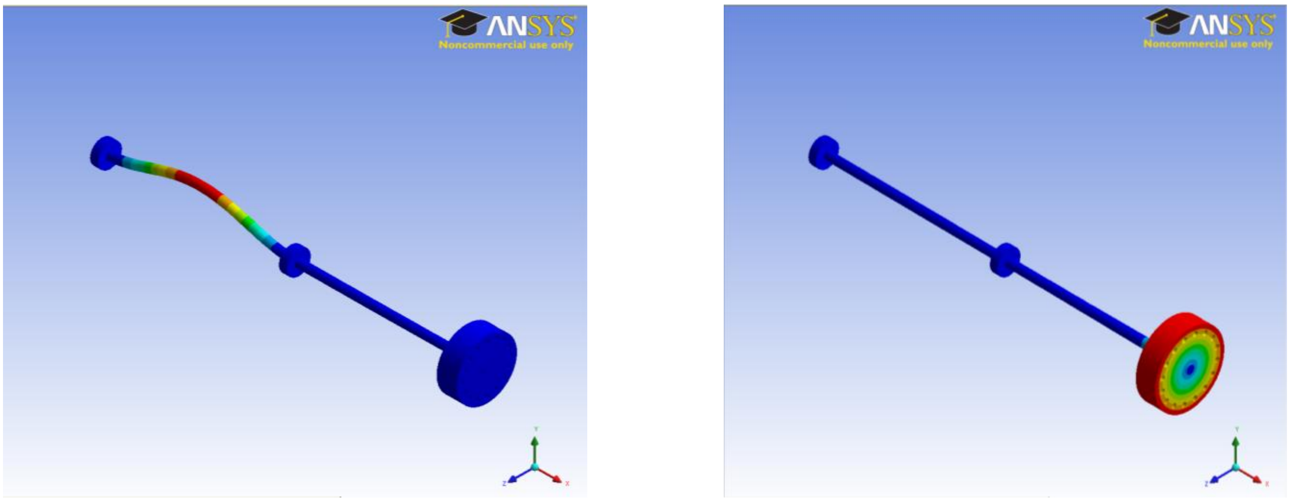
2.2.1.1 One disc in the end with two bearings (gyroscopic effect)(3D), ANSYS work bench



A-First mode shape.Natural frequency 15.47 Hz,(3-D).



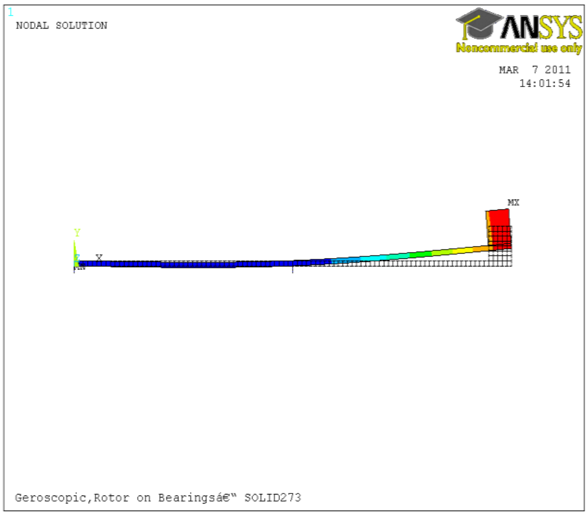
B-Second mode shape.Natural frequency 217.01Hz,(3–D).

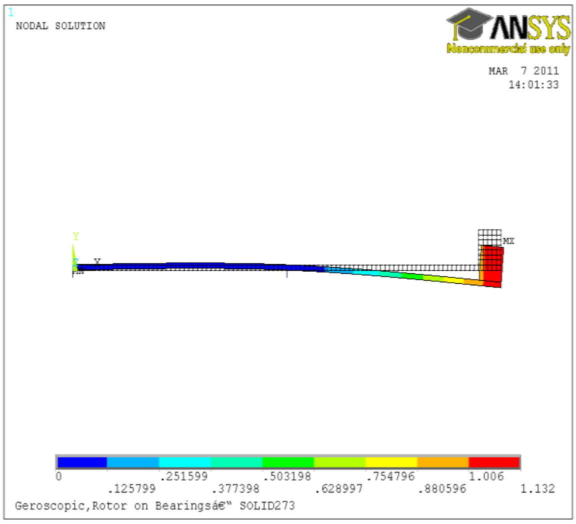


C-Third mode shape.Natural frequency 508.06Hz. D-Fourth mode shape.Natural frequency 626.85Hz.

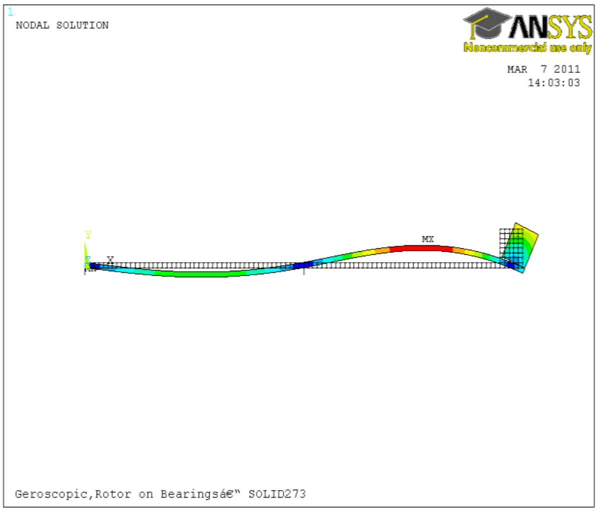
Fig.3 Finite element method simulations (FEM),different mode,ANSYS workbench;

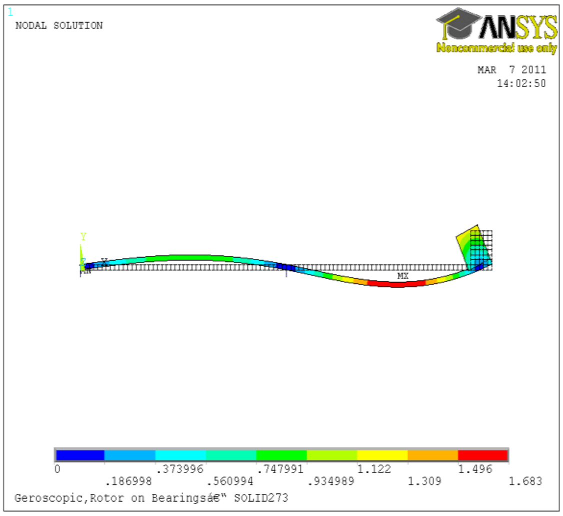
2.2.1.2 One disc in the end with two bearings (Gyroscopic effect),(2D) ANSYS APDEL



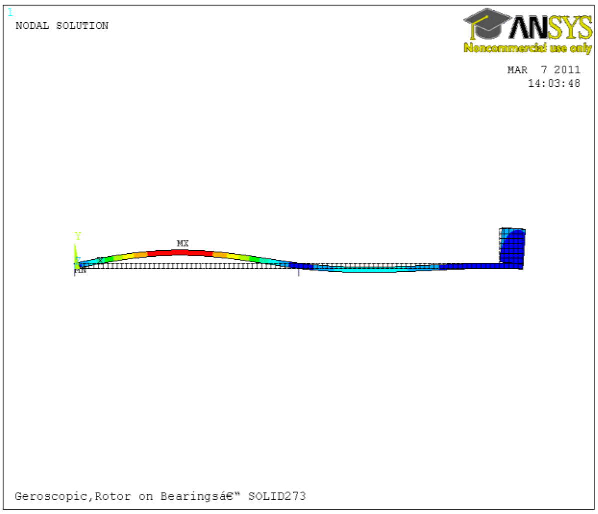


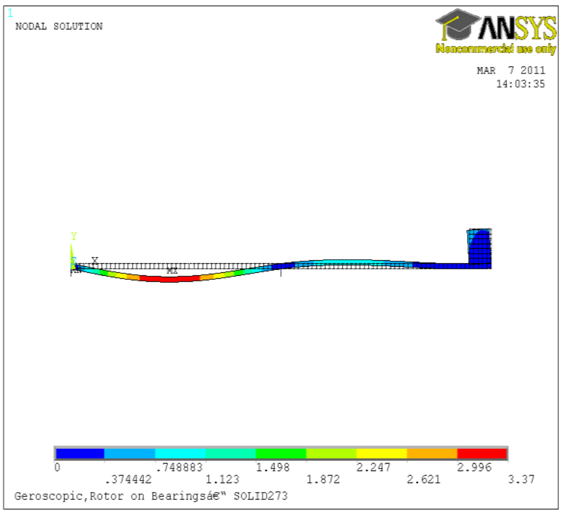
A- First mode shape.Natural frequency 15.703 Hz,(2-D).





B-Second mode shape.Natural frequency 216.8 Hz,(2–D).





C-Third mode shape.Natural frequency 507.39Hz,(2-D).

Fig.4 Finite element method simulations (FEM),different mode, ANSYS APDEL;

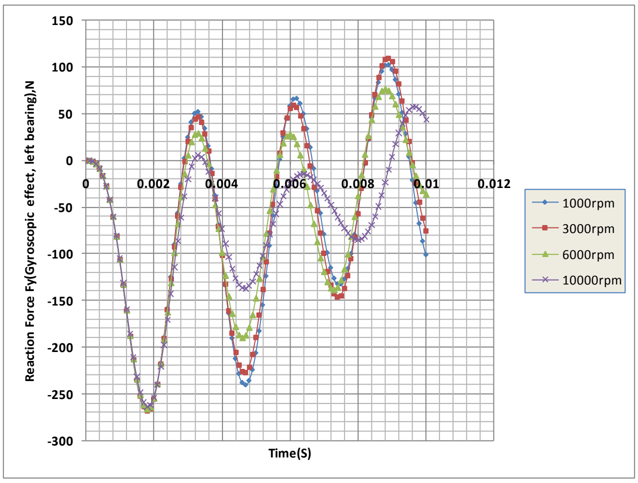
2.3 Test setup

The rotor consisted of a shaft with a nominal diameter of 10 mm, with an overall length of 610 mm.Two journal bearings, RK4 Rotor Kit made by Bentley Nevada (the advanced power systems energy services company), could be used to extract the necessary information for diagnostic of rotating machinery, such as turbines and compressor. The test rotor is shown in picture (1). Basically; Been testing the process will be conducted on the rotary machine as the project is based on rotary dynamics reach practical results for the purpose of subsequently applied machinery rotary by using (Smart office program),and then do the experimental testing using the impact test, installed fix two accelerometer(model 333B32),sensitivity (97.2&98.6) mv/g in Y&Z direction and roving the hammer(model 4.799.375,S.N24492) on each point for the purpose of generating strength of the movement for the vibration body and the creation of vibration for that with, creating a computer when taking reading in public that he was dimensions and introducing it with the data within the program (Smart office)[14&15&16].Configuration for testing on the machines with rotary machine the creation of all necessary equipment for that purpose with the design geometry wizard[17].

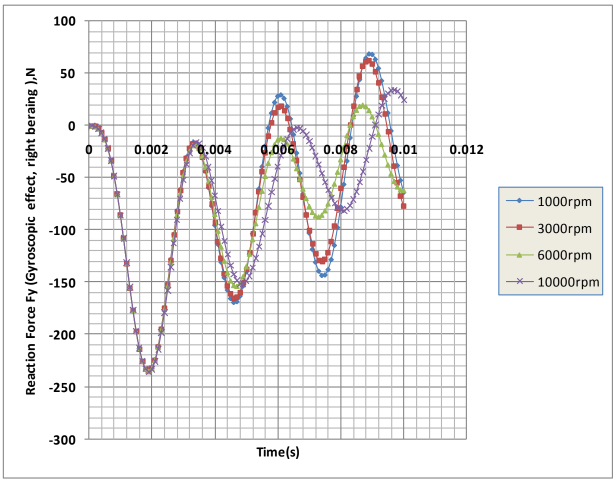
III. RESULTS (TABLES&FIGURES)

3.1 Response forces in the left and right bearings (gyroscopic effect)

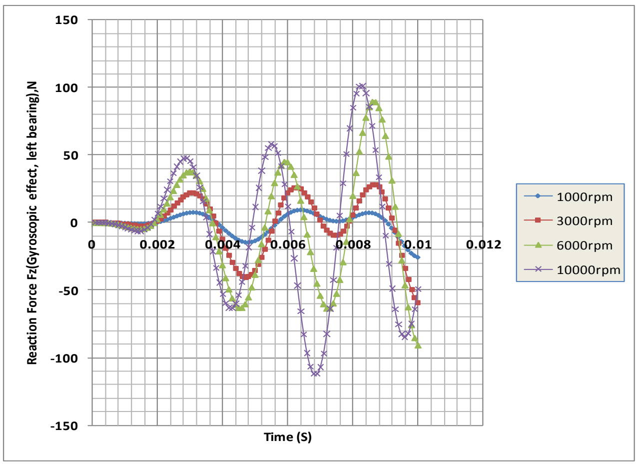
We find the relation between the reaction forces with respect time by using further simulation, can we see from the Fig.(5-A,B,C,D),the performance of reaction forces in the right and left bearings with different speed of rotations:-



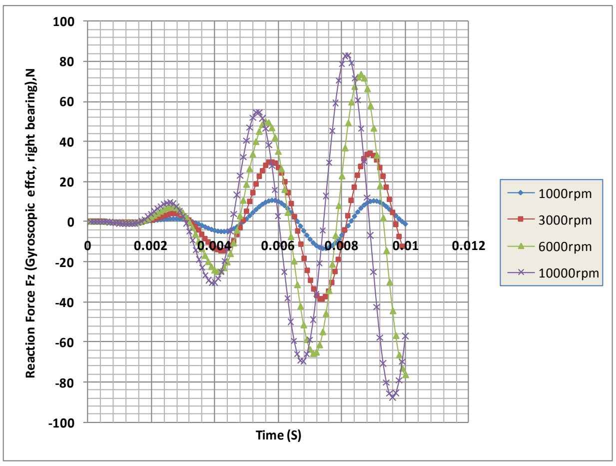
A-Reaction force(Fy) left bearing.



B-Reaction force(Fy) right bearing.



C-Reaction force (Fz) left bearing.



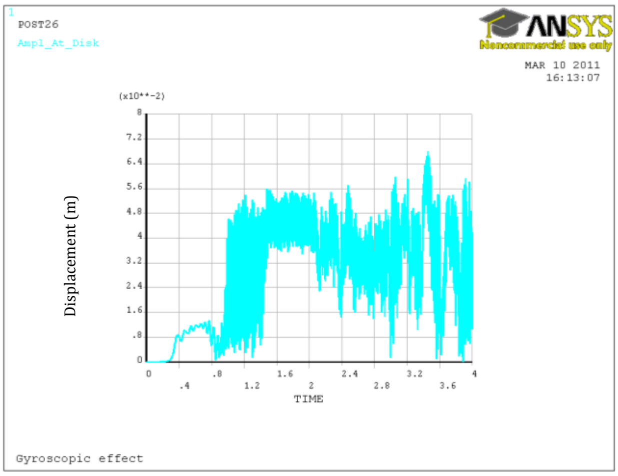
D-Reaction force(Fz) right bearing.

Fig.5 Relation between reaction force bearings versus time at different speed of rotation(gyroscopic effect);

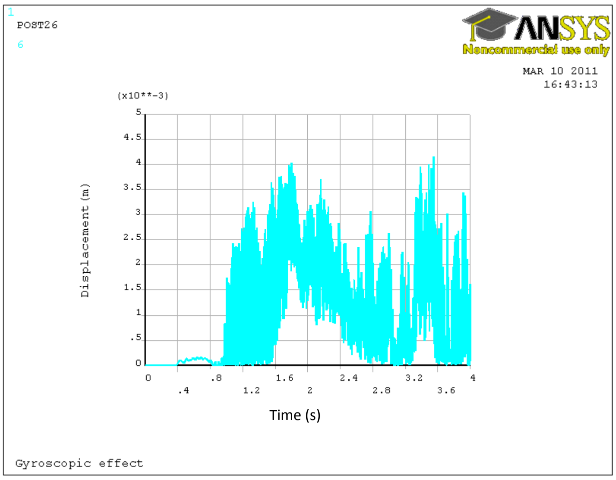
3.2 Unbalance effect

3.2.1 Unbalance with add mass (simulation result)

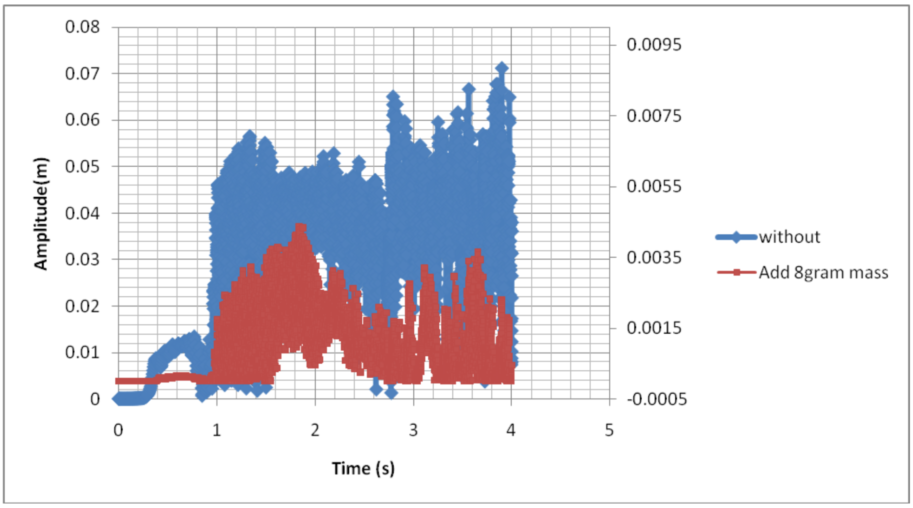
In this set simulation, unbalance loading is applied to the system to be at the optimum phase angles of $Ø=90^{\circ}$ *and* $Ø=270^{\circ}$ respectively .ANSYS simulation of the set shown in Fig.(6).



A- Displacement versus time before add mass.



B-Displacement versus time after add,8 gram mass.



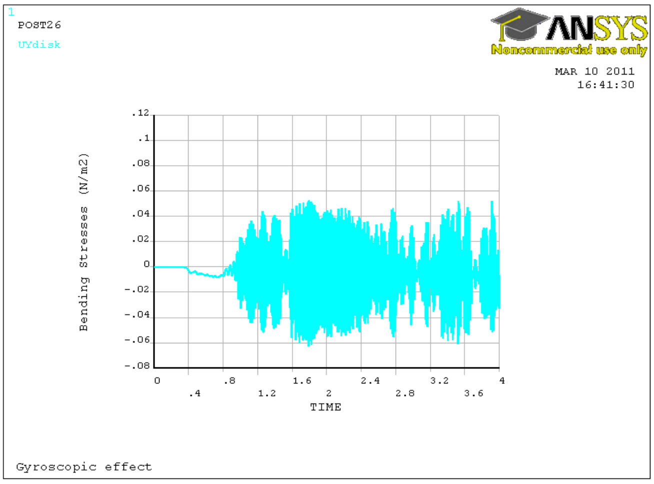
C-Merge comparison.

Fig. 6 The Amplitude versus time,(A-With out load,B-After add 8 gram mass&C-Merge);

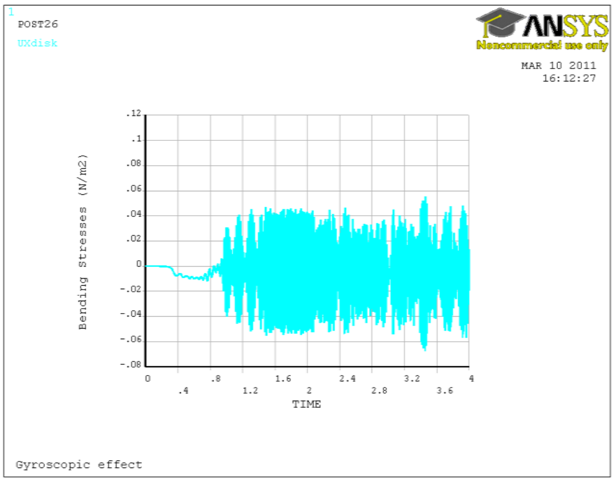
3.2.2 Behaviour of bending stresses with unbalance when add mass

We discover the relation between the bending stress versus time(second),see Fig.(7–A,B),the performance of bending stresses at gyroscopic effect in the middle when add 8 gram mass in the disc at phase angles of $Ø=90^{\circ}$ *and* $Ø=270^{\circ}$ respectively.

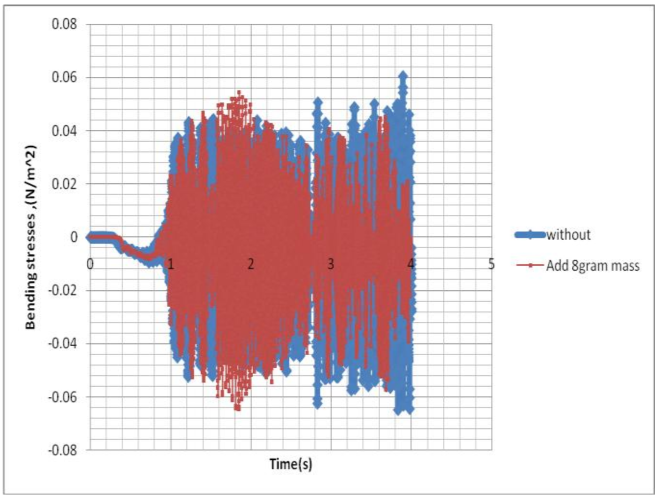
The bending stresse decreases in both direction of motion (Y, Z),see Fig (7-C,D) that mean reduce the reaction force in the bearing to make the bearing long save life.



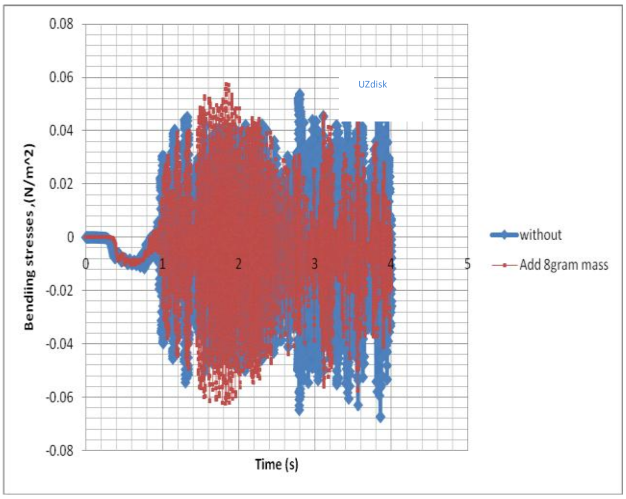
A-Bending stresses in Y direction before add mass.



B-Bending stresses in Y direction after.

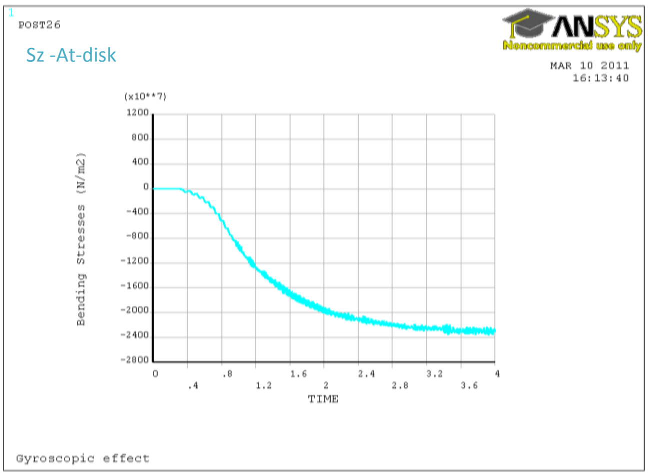


C-Merge in Y direction.

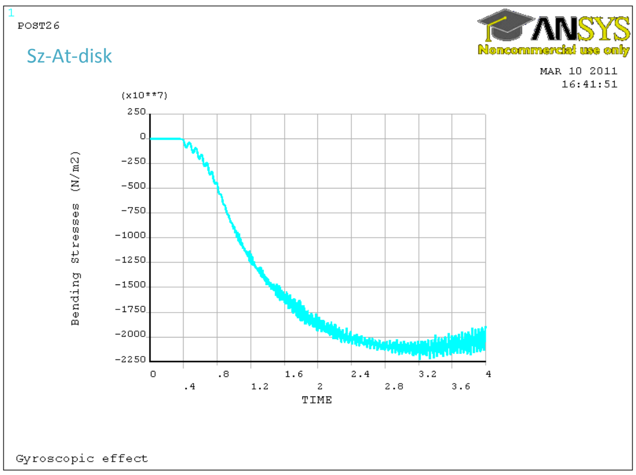


D-Merge in Z direction.

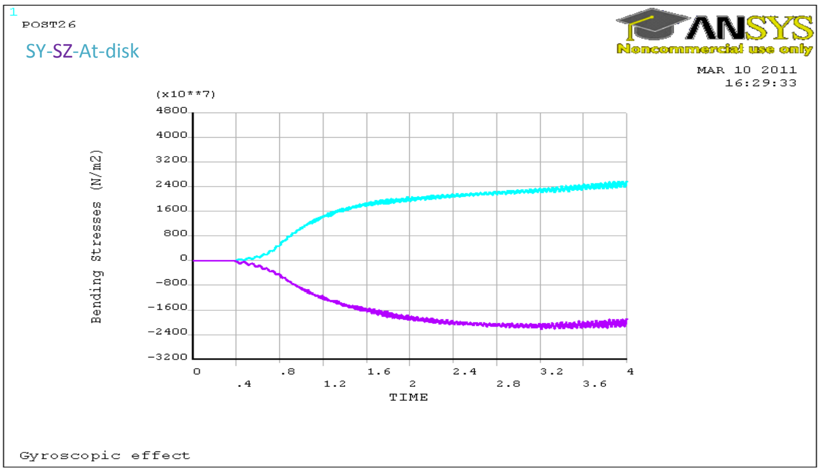
Fig.7 Relation between the bending stresses versus time (sec.);



A-Sz before add the mass .



B-Sz after add 8 gram mass.



C- Sy –Sz at disc.
Fig.8 Bending stresses sample in Y and Z direction (gyroscopic effect);

3.3 Discover damping ratio(ζ) from modal analysis

We discover the damping ratio (ζ) for different mode shape by cur fitting [11,18&19], (multi degree of freedom system) in experimental part, (Table 3) and see Fig.(9).

Table (3)
Natural frequency and damping ratio (ζ) for gyroscopic effect rang (0-500) Hz,(experimental part).

Name	Natural Frequency (Hz)	Damping Ratio(ζ) %	Modal A[kg/s]
Mode1	15.137	75.773	1.387959e-04 +i6.447278e-05
Mode 2	216.51	26.637	0.000103579 +i2.700067e-5

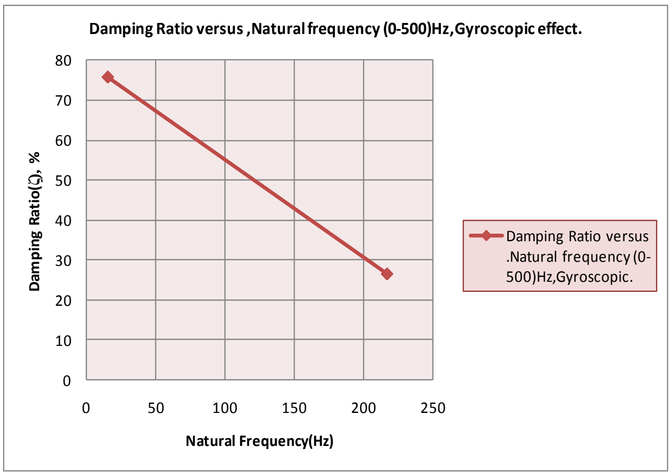


Fig. 9 Damping ratio(ζ) versus natural frequency (0-500)Hz,gyroscopic effect.

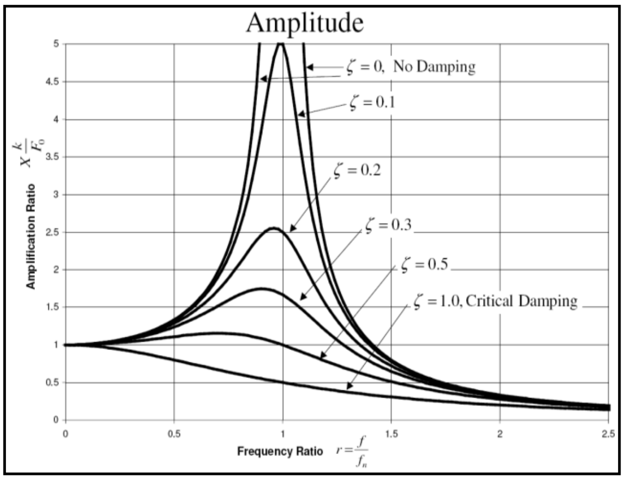


Fig 10.Variation of amplification ratio with r [16,20].

3.4 System identification and vibration monitoring in gyroscopic effect

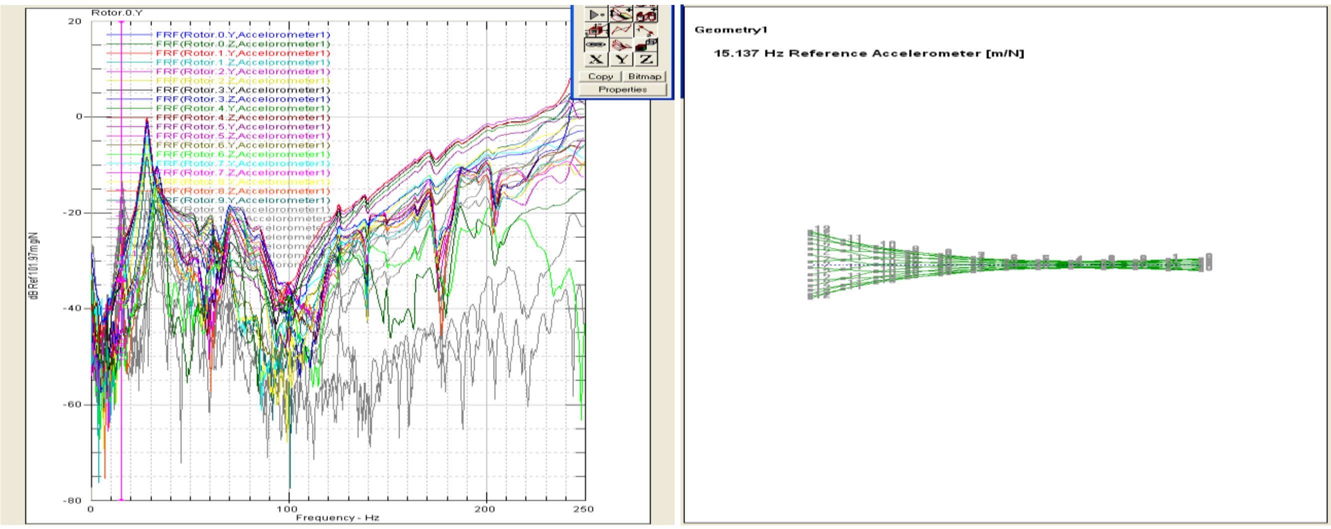


Fig. 11 Gyroscopic effect,(FRF) versus frequency (Hz),(first mode shape).Natural frequency 15.137 Hz.

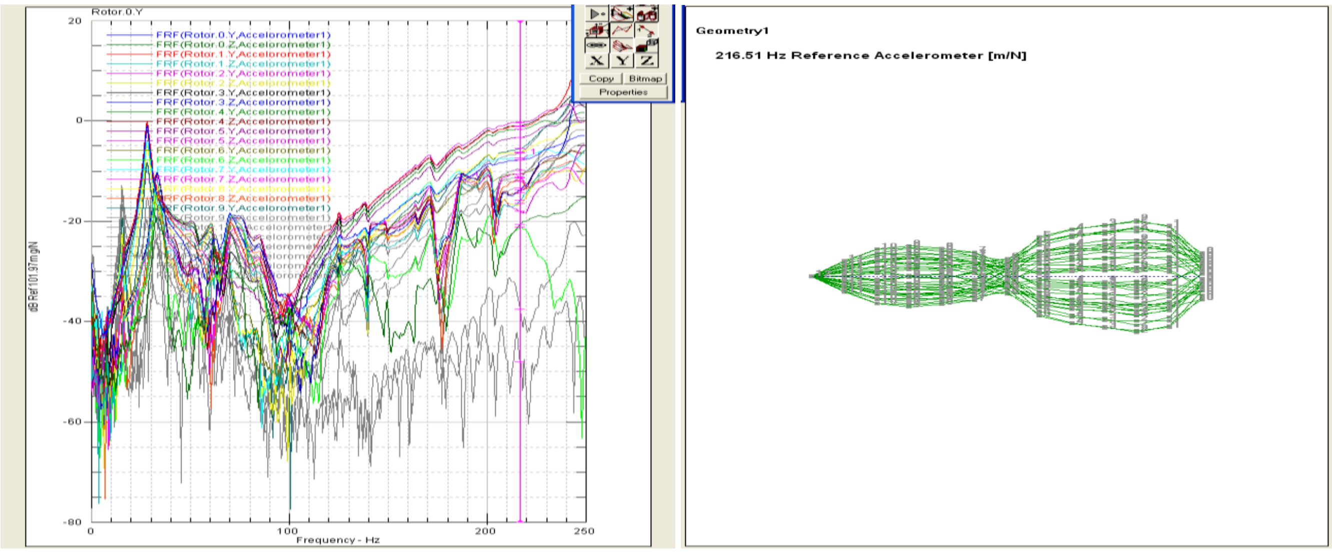


Fig.12 Gyroscopic effect,(FRF) versus Frequency (Hz),(second mode shape).Natural frequency 216.51Hz.

3.5 Contrast measured and predicted natural frequencies for gyroscope

All the result nearby each other between the experimental and simulation (ANSYS) for gyroscope without increasing the speed, see the result in (Table 4) and Fig. (14) for contrast.

Table (4)
Contrast between natural frequency (Hz),outcomes from experiment&ANSYS,(gyroscopic effect) at speed 30 rpm.

Mode Shape	ω_n (ANSYS)Gyroscopic (Hz)	Frequency Gyroscopic Experiment(Hz)	Error %
1	15.703	15.137	1.158007973
2	216.8	216.51	-0.133943005

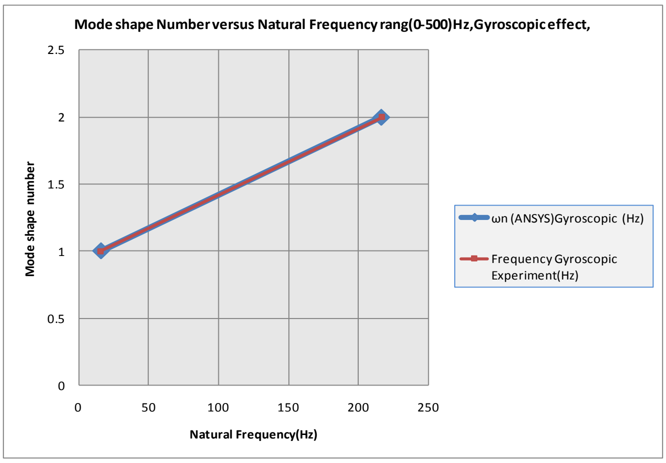


Fig. 13 Mode shape number versus natural frequency experiment and ANSYS,(gyroscopic effect).

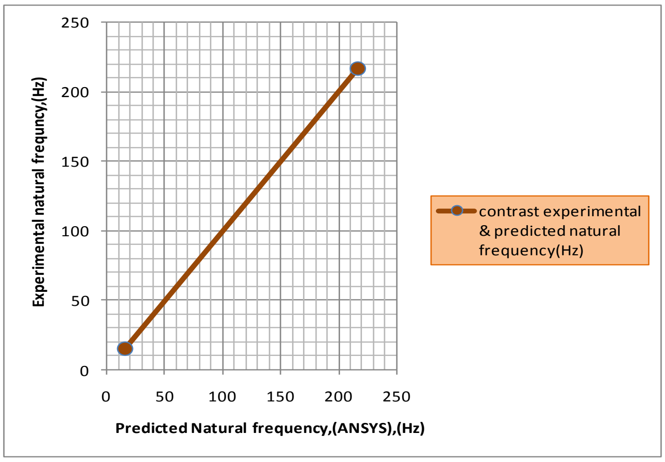


Fig. 14 Natural frequency(experiment versus ANSYS), (gyroscopic effect).

IV. DISCUSSION AND CONCLUSION

In this paper investigate the behaviour of bearing rotor system with gyroscopic effect has been cared out ,a simple mathematical model has been used, however more elaborate models based on a much large degree of freedom may be used based on suppleness or stiffness influence coefficients. The mathematical models may also be used to refine the measured data and help in removal of contaminated data. It is therefore feasible to create a mathematical model as a database for various systems for condition monitoring during their life time of the machines.

For further studies, there is no need to make more experiments about this study while ANSYS gives accurate results. We used (ANSYS) to find the relation between the reaction bearing forces (*N*) with respect time can we see Fig.(5-A,B,C&D).This performance in the right and left bearings with different speed we see when increasing speed of rotation the reaction force increasing for both right and left bearings when increasing the speed of rotation but from the figure above we see the maximum reaction force in Y direction in left and right bearings when the motor run up, after a few second is become decreases. While the reaction forces in Z direction is began increasing slowly in left and right bearings until reach maximum value when the speed is increasing.That mean we must take care to left bearing when run up the motor because this bearing carry maximum reaction force at the began.During study this performance of reaction force in both bearings can aid in the design of low-noise rotor-bearing systems and reduce the reaction force in the bearing to make the bearings long save life by lubracation.In order to investigate the effects of design parameters on the noise of rotor-bearing systems, the effects of radial clearance and width of bearing, lubricant viscosity, for various rotational speeds. It is found that, as a general rule, the noise of the bearing decreases as the lubrication viscosity increases, the width of the bearing increases, and the radial clearance of the bearing decreases.

The locations of the adding balance masses in suppressing the vibration amplitudes are decided to be at the optimum phase angles of $\varnothing=90^\circ$ *and* $\varnothing=270^\circ$ respectively.It was observed for each of the different eccentricity ratios studies. The critical adding mass ratios can also be predicted through its linear relationship with the eccentricity ratios,The simulation values obtained from the ANSYS see Fig.(6), this results showed that could reduce the vibration by reducing amplitude when add 8 gram mass in the angles show above; As a result, can reduce the vibration more effectively and modified method described in this paper to solve real-world engineering problems.

We discover the relation between the bending stress versus time(second),see Fig. (7) the behavior of bending stresses at gyroscopic effect when added 8 gram mass in the disc the bending stresse decreases in both direction of motion (Y,Z) that mean reduce the reaction force in the bearing to make the bearing long and save life .

From Table (3) detection damping ration (ζ) in experimental part for the first and second mode at speed 30 rpm, and we can see from Fig (9) the decreased the damping ratio caused increased natural frequency until reach maximum amplitude when the system reach resonance $\boldsymbol{\omega} = \boldsymbol{\omega}_{n,}$ when damping ration (ζ) approximately = 0),(free vibration) is clear in Fig.(10)[16,21].

From Table(4),contrast measured and predicted natural frequencies for gyroscopic effect all the outcome nearby each other between the experimental shown in Fig.(11),(12) and model simulation (ANSYS) shown in Fig.(3),(4)for gyroscopic effect without rising the speed,see the result in (Table 4) and is more clear in Fig. (13)&(14) for contrast. Plotting the experimental value against the predicted on for each of the modes included in the contrast shown in Fig.(14).In this way it is possible to see not only the degree of correlation between the two sets of results, but also the nature(and possible case) of any discrepancies which do exist.The points plotted should lie on or close to straight line of slope [17,22].

4.1 Summaries what have learned

A gyroscope is a device that can be used to maintain orientation based on the principles of angular momentum.
As a general rule, the noise of the bearing decreases as the lubrication viscosity increases, the width of the bearing increases, and the radial clearance of the bearing decreases.

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