Stability and Vibrationl Analysis of a Flexible Damped Rotor System using Finite Element Method

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Abstract— *Dynamic stability of a damped flexible Rotor system is studied in this work. Analysis of a damped rotating shaft with multiple discs are carried out by, varying the rotor offset positions in order to investigate the effect on critical speed and frequency of the system. From this analysis, it is observed that by varying the rotor offset positions, the modal mass participation varies and which in turn cause a variation in the critical speed and frequency of the system. This study also extends a detailed evaluation of damped rotor stability. Through this analysis it is observed that the system become unstable beyond its critical speed. The stabilizing effects of anisotropic bearing stiffness and external damping are also demonstrated. The effect of unbalance in the rotating machinery is evaluated in the last section.*

Keywords— ***Analysis, Bearing, Critical speed, Internal damping, Rotor, Unbalance.***

I. INTRODUCTION

The predictive maintenance concepts with vibration measurements have a greater application in rotor dynamics [1]. The future possibility of vibration based condition monitoring of rotating machines are described by Jyoti K.Sinha et.al. [2]. A.W.Lees et.al [3] seeks to give an overview of the recent developments in vibration-based condition monitoring which has considerable practical importance. Vibration based condition monitoring has become well accepted and widely used to identify faults in rotating machines. A finite element dynamic modal for rotor bearing system, which accounts gyroscopic moments and anisotropic bearings are developed using a modal truncation method by Y.A. Khulief et.al [4]. D.Combescure et.al [5] presents a refined finite element modeling used for dynamic analysis of large rotating machines. The stability domain of an internally damped flexible spinning shaft, which is driven by a non-ideal source, is studied by S.S. Dasgupta et.al [6]. J.K. Sinha et.al [7] proposes a method that can reliably estimate both the rotor unbalance and misalignment from a single machine run-down. An alternative balancing methodology for rotating machinery is presented by T.S. Morais et.al [8]. Chun-biao Gan et.al [9] extended the nonparametric modeling technique to the uncertain Jeffcott rotor with disc offset, and the random matrix model is established. The parametric instability of flexible rotor-bearing system under time-periodic base angular motions is analyzed by Qinkai Han et.al [10].

Rotor dynamic instability, a self-excitation of the rotor-bearing system, can occur without prior warning and has catastrophic potential. Instabilities pose a serious challenge to the designer, since there are many different mechanisms to be dealt with. Therefore, it is necessary to analyze the dynamic characteristics of the rotor system to guarantee the rotating machinery is running safely. Finite element simulation of rotor-systems can be improved by modeling internal damping effects. This provides an added dimension in studying turbo machinery stability and can assist the engineer in proper selection of bearing characteristics to insure safe high-speed operation.

II. FINITE ELEMENT MODELING OF FLEXIBLE ROTOR-BEARING SYSTEM

A rotating structure generally consists of rotating parts, stationary parts, and bearings linking the rotating parts to the stationary parts and/or the ground. The model consists of two rigid discs mounted on a uniform flexible mass less shaft supported by two identical flexible bearings at each side. The damping of the shaft and bearings are taken into account. The shaft is modeled with 9 nodes and 8 elements equal spaced every 0.05 meters along the shaft. The disks were modeled on two nodes of the shaft and the moments of inertia and masses were calculated for each disk and input as real constants for each element. The bearings are modeled at the beginning and end of the shaft. The stiffness and damping constants in the lateral directions are input as real constants. The effect of internal viscous damping is also incorporated into the finite element model.

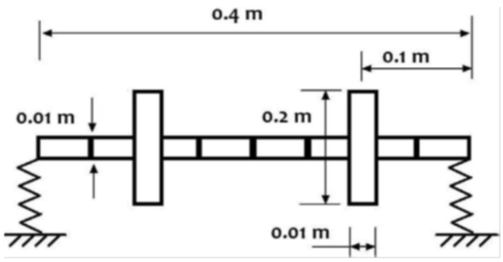

Fig. 1: Flexible two rotor system

2.1. The disk

The disk is modeled with Mass 21 element and is defined by a single node. The degrees of freedom can be extended up to six directions: translation in the x, y and z directions and rotation about the x, y and z axes. The rotary inertia effects can be included or excluded and element can be reduced to a 2D capability. If the element has only one mass input, it is assumed that mass acts in all coordinate directions. In this paper, two disks are identical with m_d=2.4Kg, I_d=0.006 Kgm^2, I_p=0.012 Kgm^2.

2.2. The flexible shaft

The shaft is modeled with the element Beam 188, is a two noded beam element in 3D with tension, compression, torsion and bending capabilities and is developed based upon Timoshenko beam theory. The element has six degree of freedom at each node, translation in the nodal x, y and z directions and rotation about the nodal x, y and z axes. This beam element consists of different section shapes so that it can be modeled with desired section shapes and there by real constants for the chosen sections are automatically included.

2.3. Bearing support

The bearings are flexible in nature and are fixed to the base. Bearing supports are modeled using Combin14 element, has longitudinal or torsional capability in one, two, or three dimensional applications. The longitudinal spring-damper option is a uniaxial tension-compression element with up to three degrees of freedom at each node: translations in the nodal x, y, and z directions. No bending or torsion is considered. The torsional spring-damper option is a purely rotational element with three degrees of freedom at each node: rotations about the nodal x, y, and z axes.

III. VALIDATION

In order to verify the validity of the analysis code presented in this paper, we will compare our analysis results based on our analysis code with the results in the Ref.[10]. The results of the whirling frequencies without base movements from the reference journal are reproduced by using the analysis code

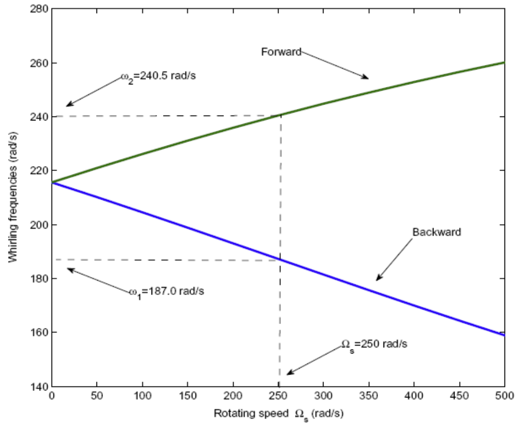

Fig. 2: Whirling frequencies of the system without base movements from the reference journal

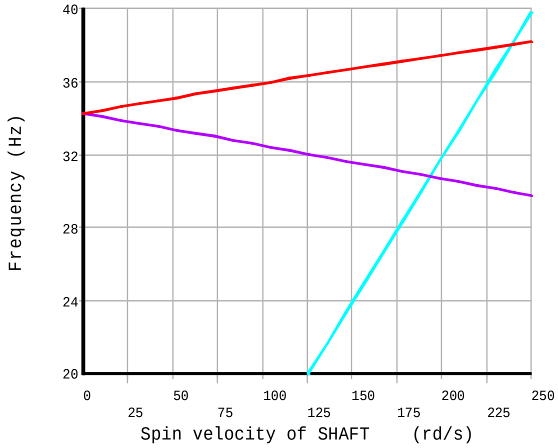

Fig. 3: Whirling frequencies of the system without base movements obtained in this work

Table 1: Whirling speed

	Forward Whirling Speed (rad/sec)	Backward Whirling Speed (rad/sec)	Forward Whirling Frequency (Hz)	Backward Whirling Frequency (Hz)
Values from the Reference Journal	240.5	187	38.27	29.76
Values obtained by using Analysis code	240.3	187	38.25	29.76

IV. COMPUTATIONS AND DISCUSSIONS

4.1. Analysis of Flexible rotor System with different Rotor offset Positions

Calculation and analysis of critical speed is one of the important tasks of rotor dynamics, which helps to choose the balance velocity of the rotor systems. In this section the analysis of the critical speed of the flexible rotor system with multiple discs are carried out. The analysis is carried out by placing the rotor discs at various nodes. analysis taking the gyroscopic effect due Here the analysis is carried out by placing the rotor disks on symmetrical positions, that is, the analysis is carried out by placing the rotors at the nodes 2&8, at the nodes 3&7, at the nodes 4&5 to study the effect of rotor positions on the critical speed and hence the stability of the system. In this analysis Campbell diagrams and unbalance response diagrams are used to analyze the critical speed.

The speed of the disc rotating around its own geometrical center is called the rotational speed of the rotor, which is a given value during the running process. The rotating speed of disc around the center of the shaft is called the whirl speed of the rotor. The line, when the rotational speed and whirl speed of the rotor become equal is the 1X line represented in the Campbell diagram. Fig 4 to Fig 6 shows the Campbell and unbalance response diagram for the rotors at different rotor offset positions. The intersection point of 1X line and the curves in the Campbell diagram corresponds to the critical speed of the System. From the Campbell diagram it is clear that the net effect of the gyroscopic factor would increase the whirl speed in forward direction and decrease the whirl speed in the backward direction as shown in Table 2.

Results from the Campbell diagram and unbalance response curve shows that forward whirl speed is critical but the backward whirl speed is not. This is because at backward whirl speed the steady state unbalance response curve shows no peak and no phase shift. This can be illustrated from the Fig 4 to Fig 6. The reason for this behavior is the axial symmetry of this two disk rotor system. It is because that the backward mode vector is orthogonal to the unbalance force vector and as a result energy cannot be fed into the backward whirl. When the rotor is not axially symmetric both forward and backward whirl speed become critical speeds. The results shows that the critical speed of the system increases with the increase in the degree of rotor offset positions. The critical speed of the flexible rotor system varies in the range of 1206rpm to 3257rpm, when the positions of the rotor discs are changed from the center of the shaft towards the end of the shaft. The effect of gyroscopic factor on the rotor positions and hence the critical speed of the system is also investigated, and the results shows that the position of rotor discs within the system have a greater influence in the action of gyroscopic force on the system. When the rotor discs are placed at the center of the shaft, effect of gyroscopic force on the critical speed is very less, a change of 1.08 % only. Whereas, when the rotor discs are positioned at the end of the shaft, the effect of gyroscopic force become very critical, a change of 57.87% on critical speed is obtained. In the case of a flexible two rotor system the effect of gyroscopic force on the critical speed of the system become more prominent when the rotor discs are positioned at the end positions of the shaft, rather than at the center position.

4.2. Effect of Internal damping in the stability of flexible two rotor system

This analysis extends to a detailed evaluation of damped rotor stability of a flexible two rotor system. In order to perform the analysis, consider a damped flexible two rotor system from the previous analysis with maximum critical speed. Here the effect of internal viscous damping is incorporated into the finite element model to evaluate the damped rotor stability. The internal damping is given to the system as a material property. In this work we are providing a material damping co efficient of 0.0002/s.

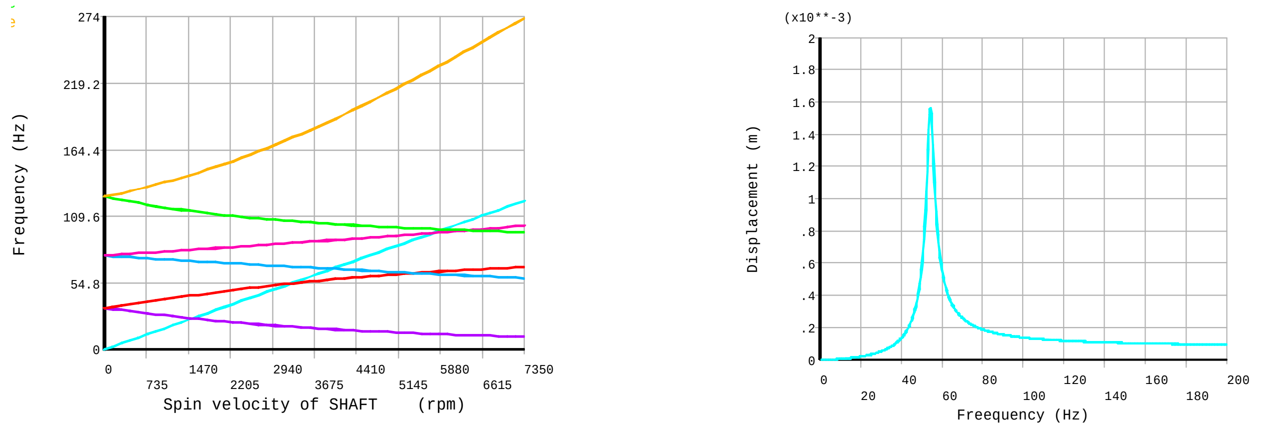

Fig.4: Campbell and Unbalance response diagram when rotors are placed on node 2 and 8

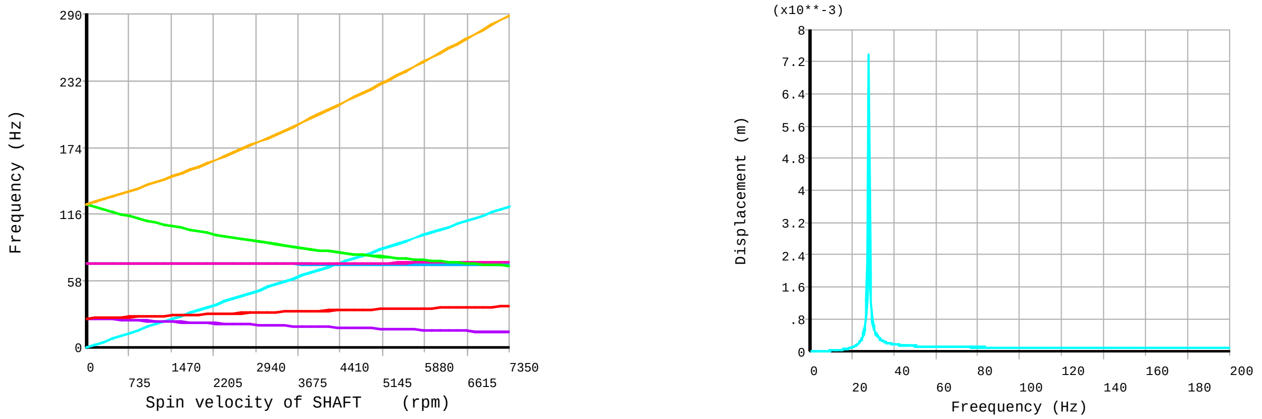

Fig.5: Campbell and Unbalance response diagram when rotors are placed on node 3 and 7

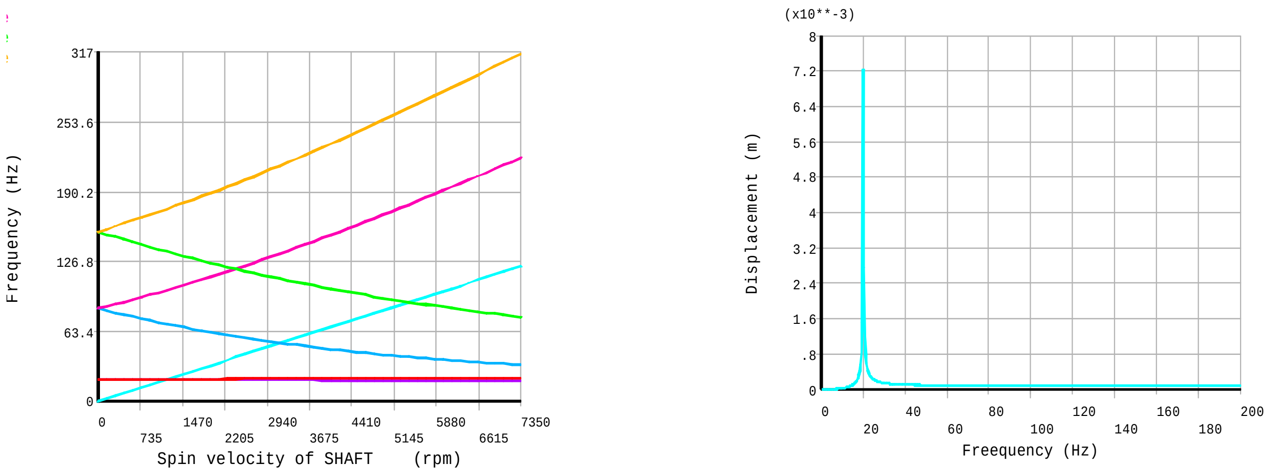

Fig.6: Campbell and Unbalance response diagram when rotors are placed on node 4 and 5

Table 2: Results from Campbell diagram and unbalance response curve

Rotor Position	1st Forward Critical Speed (rpm)	1st Backward Critical Speed (rpm)	Unbalance response peak (rpm)
2,8	3257	1540	3264
3,7	1688	1349	1680
4,5	1206	1179	1200

Table 3: Effect of gyroscopic factor on critical speed

Rotor Position	First Mode frequency (Hz)	1st Critical Speed (rpm)	Critical Speed without gyroscopic effect (rpm)	Percentage change (%)
2,8	34.386	3257	2063	57.87
3,7	25.049	1688	1502	12.38
4,5	19.986	1206	1193	01.08

Internal damping is due primarily to friction at rotor component interfaces. When the power inserted by the internal damping exceeds the power extracted by the external damping, that is, support bearing damping and external damping, the rotor will become unstable.

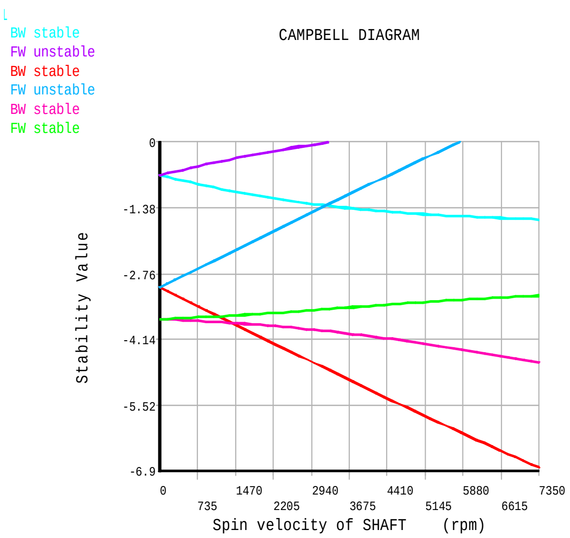

Fig.7: Campbell diagram of damped rotor system with isotropic bearing

Results from the Campbell diagram, in Fig.7 shows that the characteristic instability of first mode of the system occurs at a speed of 3257rpm, which is the first critical speed of the system. The first mode remain unstable for higher spin speeds, along with the second forward mode becoming unstable at second critical speed, that is, at 5811rpm. In order to improve the threshold speed of instability and to make the system stable over higher rotational speeds, two methods are employed. Addition of isotropic bearing damping into the system stabilize the first and second mode of vibration in the system.

An isotropic bearing damping of $C_{yy} = C_{zz} = 1\times10^2$ Ns/m is added to evaluate the effect of external damping in the threshold speed of instability. From the Campbell diagram as shown in Fig.8 it is clear that the threshold speed of instability of the rotor system is greatly improved. The addition of external damping makes the first mode of the system stable to 7973rpm. The system becomes unstable at 6739rpm, where the second mode becomes unstable. By improving the external damping we can improve the threshold speed of instability.

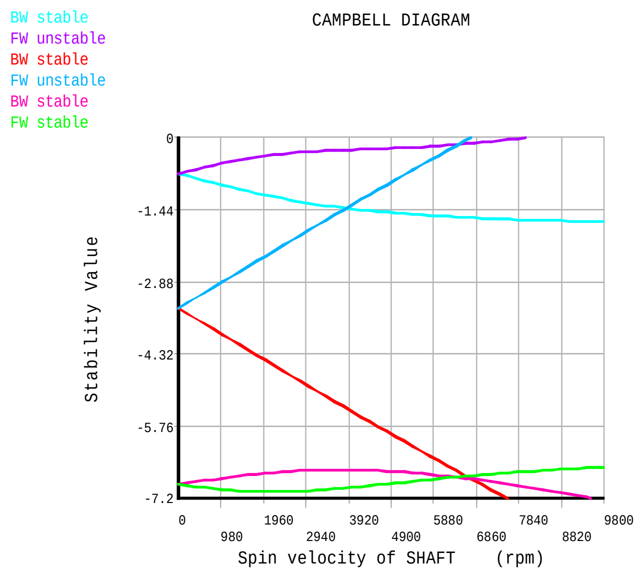

Fig.8: Campbell diagram of damped rotor with external damping

Removal of isotropic bearing stiffness and the addition of anisotropic bearing stiffness, can improve the stability of the system to a great extent. An anisotropic bearing stiffness of $K_{yy}=1\times10^6$ N/m and $K_{zz}=0.8\times10^6$ N/m is added into the system to evaluate the effect of anisotropic bearing stiffness in the threshold speed of instability. From the Fig.9, values from the Campbell diagram shows that anisotropic bearing stiffness improves the threshold speed of instability of the system. The system is stable up to 6820rpm, where the second mode of the system becomes unstable. The first mode is stable over 10000rpm.

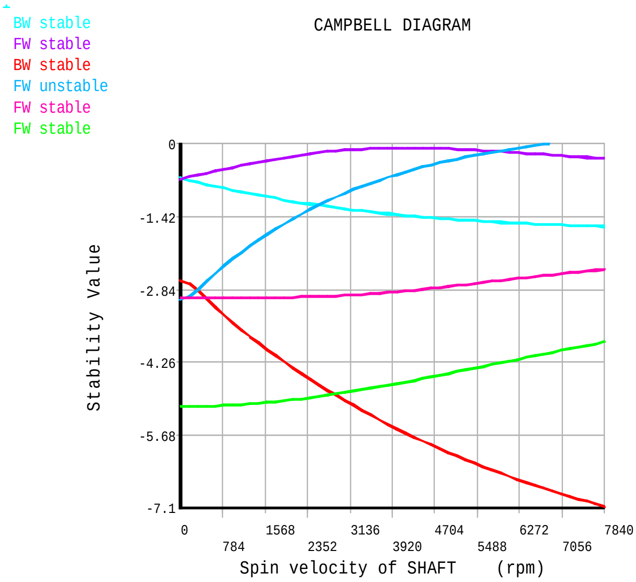

Fig.9: Campbell diagram of damped rotor system with anisotropic bearing

The benefit of using anisotropic bearing is obtained as a result of elliptical orbit formed by asymmetric bearing stiffness. The cross coupled force puts maximum amount of work into the system when the orbit is circle and this work reduces when the orbit become more and more elliptical, when the orbit is become more asymmetric. As the whirl orbit become more and more elliptical, the work input from the cross coupled force decreases towards zero, whereas the work taken away by the damping does not. Stability is achieved by increasing the ellipticity of the orbit. The asymmetry of the bearing improves the stability of the system.

Table 4: Comparison of threshold speed

Damped rotor system	Threshold speed of instability (rpm)
With Isotropic bearing stiffness	3257
With Isotropic bearing stiffness along with damping	6739
With anisotropic bearing stiffness	6820

4.3 Effect of unbalance in the vibration of damped rotor systems

Transient analysis is carried out to study the effect of unbalance in the vibration a damped rotor system. Model used for the analysis is same as that used for damped stability evaluation. An unbalance force is generated by adding a small unbalance mass of about 2gm into the system. The unbalance force is provided to the system in 3 conditions. The unbalance response is plotted by the use of vibrational analysis, by varying the rotational speed of the rotor from 0 to 7000rpm with in 4sec.

4.3.1 Unbalance force due to an added unbalance in any one of the rotor disk

The analysis is carried out by adding an unbalance in any one of the two rotor discs. Because of this unbalance force system excited in both its first and second mode of vibration as shown in Fig.10. At zero rpm there is no force from unbalance excitation, so the unbalance response curve starts out with no response and the amplitude of vibration grows slowly during the running process. The system vibrates with maximum amplitude and heavy noise, when reaching its critical speeds. The unbalance response amplitude gradually reduces to a constant value above the critical speed.it is because below the critical speed, the unbalance acts to pull the disk out into an orbit that grows increasingly large with the speed. Once the disk achieves this state, further increase in speed does not change the amplitude until the effect of the next mode are observed.

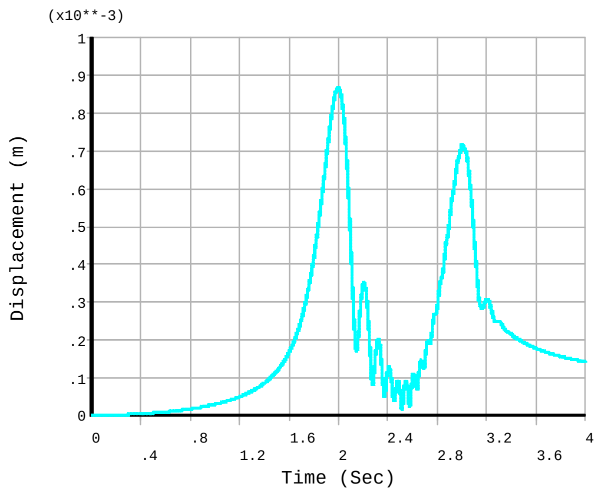

Fig.10: Unbalance response curve due to a single unbalance force

4.3.2 Unbalance force due to symmetrical unbalances

The analysis is carried out by adding symmetrical unbalances at both rotor discs of the damped two rotor discs. Fig.11 shows the unbalance response of a flexible two rotor system with the addition of two symmetrical unbalance in the system. From this figure it is clear that the system vibrates only in first mode and there is no excitation in the second mode of vibration,. When symmetrical unbalance forces are there in the same direction, which will suppress the second mode of vibration and the system continue to vibrate only in its first mode. So the excitation of the system, only occurs on reaching the first critical speed of the system, but the amplitude of vibration is very high in this condition, followed by large number of vibrations in the first critical speed range

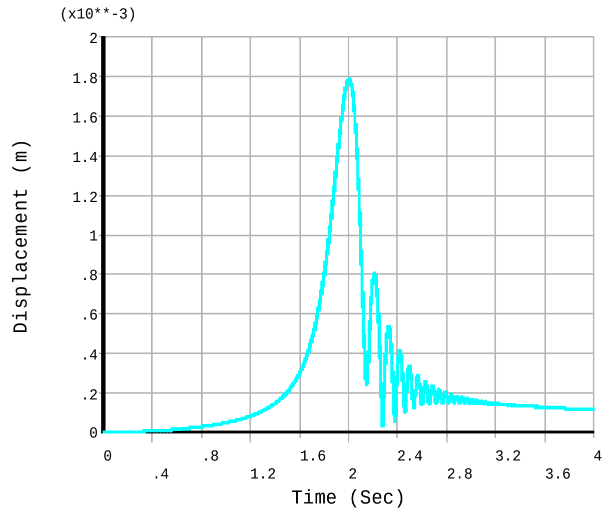

Fig.11: Unbalance response curve due to symmetrical unbalance force

4.3.3 Unbalance force due to symmetrical and opposite unbalances

The analysis is carried out by adding symmetrical and opposite unbalances at both rotor discs of the damped two rotor discs. Fig.12 shows the unbalance response of a flexible two rotor system with the addition two symmetrical and opposite unbalance in the system, causes the system to vibrate only in second mode there is no vibration in the first critical speed of the system. It is because symmetrical and opposite unbalance force on the system always tries to vibrate the system in its second mode and reduces the first mode of vibration. So the system excite only in its second critical speed range. So the large amplitude vibrations only occur in a higher speed range so the system can be safely operated in its first critical speed without any vibration and noise.

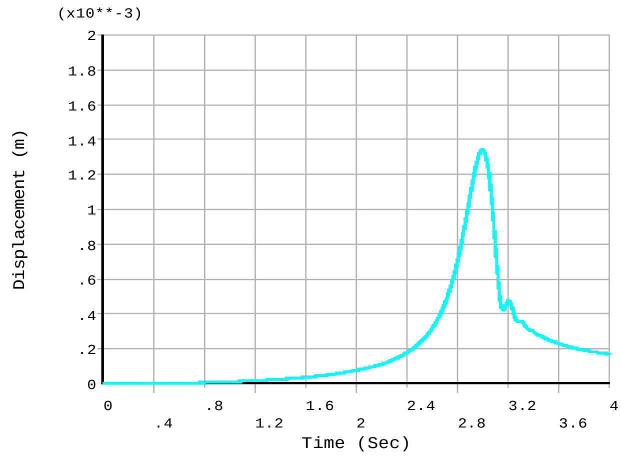

Fig.12: Unbalance response curve due to symmetrical and opposite unbalance

We can reduce the amplitude and vibration and stress generated on each element caused by unbalance with the aid of suitable external damping.

V. CONCLUSION

Gyroscopic effect has a greater influence in the critical speed of the system. Gyroscopic effect is more prominent, when the discs are positioned at the end of the shaft, which cause an increase of 58% in its critical speed. 63% of reduction in critical speed of two rotor system is obtained when the positions of discs are changed from the end of the shaft, towards the center of the shaft. The rotor will be more stable if the discs are placed towards the ends of the shaft and will be less stable if the discs are placed more towards the center of the shaft. The influence of internal damping makes the system unstable above its critical speed. The threshold speed of the system can be improved by using external damping and anisotropic bearing. When a system is rotating due to unbalance mass unnecessary vibration occurs, further it may generate excessive stress in machine elements. The mode of vibration and stress generated in the elements depends on the position of unbalance and the amplitude of vibration is become more severe, when the unbalance is higher.

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