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Research paper

Analytical and Numerical Investigation to the Effect of Single Transverse Crack on the Dynamic Behavior of Rotor-Bearings System

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Abstract

This paper focused about the effect of single crack on the response and critical speed for rotor shaft which has a disc lies not at mid span supported by two journal bearings. First, found the dynamic behavior and parameters of journal bearings (stiffness and damping) and their Sommerfeld number with varies rotating speeds. The speed changing on eccentricity of journal with shaft center and on Sommerfeld number. Second, investigated the effect of changing of crack's depth on the maximum response and critical speed for single cracked rotor. The comparison has been done between analytical and numerical results for uncrack and single cracked rotor by showing the discrepancy between them by using MATLAB & ANSYS for analytical and numerical methods respectively.

Index Terms— rotor dynamics, cracked rotor, single crack, transverse crack, critical speed and dynamic behavior of the rotor-bearings system

1. Introduction

Since the rotors are wide utilized in industry like steam turbines, gas turbines, fans and stator of electric engines. The development in power and increasingly troublesome structure of turbomachines joined by higher necessities to their dependability. To build the life of rotors is additionally one of the principle focuses to show signs of improvement quality, current computational strategy is utilized to decide quality and unwavering quality attributes Nagaraju and Srinivas, 2014. Rotor elements is a critical part of the control of elements that relates to the conduct of a tremendous arrangement of rotational machines, Maurice, 2001. The two vital parameters in elements of rotor are reaction sufficiency and basic speed. The basic speed of rotor is normally characterized as the rotational speed of rotor at which vibration because of rotor unbalance is a nearby most extreme [1]. The basic speed of rotor upheld on the liquid film short diary orientation can't be characterized as on account of an inflexible bearing rotor, in light of the fact that the solidness and damping coefficients esteems are relying upon the rotor speed [2]. Therefore, it is in every case better to examine the unbalance reaction to find the basic rates Won Lee, 1993.
The connection between oil film solidness [3], dampers and rotor turn speed for short diary has been examined by finding the consistence grid [4] coefficients for single broke shaft with rotational movement to discover firmness network of split component [5]. By utilizing hypothesis of exhaustion break in break mechanics [6], two things have been cultivated in hypothetical examination by utilizing MATLAB, the location of splits by knowing the difference in basic velocities and reaction of the broke shaft [7] and contrasted and basic rates and reaction of the un split shaft, realizing the split profundity by knowing the reaction of split district. The numerical investigation of vibration performed by utilizing FEA with chose BEAM188 component for shaft and COMBI214 component for orientation in the ANSYS programming.

2. Fundamental Equations

2.1 Fluid Film Bearings

There are numerous parameters and material science wonders that control the rotors separated from stationary structures however the principle contrasts are the liquid film bolsters [8], on the off chance that we need to comprehend the rotor elements as in Fig.1. In the past they were trusting that the grease in the depression of the bearing will diminish the rubbing and limit the misfortunes, and afterward they found that the liquid [9] film doing numerous things more than the misfortunes of the grating. In the case of taking a gander at Fig.2, the bearing focus C and the diary focus $\tilde{C}$ will shape a frame of mind of the bearing and makes the edge with the vertical load (W), the leeway h will change between two qualities [10].
From the bearing geometry, speed, unconventionality, weight and frame of mind point [11] Sommerfeld inferred such parameter to give a sign about the bearing whimsy as, Michael, et al,.2012.

$$S = \frac{\mu DL\, N}{W} * \left(\frac{r}{C_l}\right)^2 \quad (1)$$

The radial and tangential forces F_r, F_t is.

$$F_r = -\frac{D\,\Omega\,\mu L^3\ \epsilon^2}{2h^2\,(1-\epsilon^2)^2} \quad and \quad F_t = -\frac{\pi\ D\,\Omega\,\mu L^3 \epsilon}{8h^2\,(1-\epsilon^2)^{3/2}} \quad (2)$$

The force F_t faces the descending gesture and the power lost is $F_t * \Omega D/2$, the subsequent strength on the bearing is opposite to the applied load on the rotor.

$$F = \sqrt{F_r{}^2 + F_t{}^2} = \frac{\pi\ D\,\Omega\,\mu\ L^3\ \epsilon}{8h^2\,(1-\epsilon^2)^2}\left(\left(\frac{16}{\pi^2}-1\right)\epsilon^2+1\right)^{1/2} \quad (3)$$

In the event that the heap on bearing is known, the changed Sommerfeld number is given by Yukio and Toshio,**, 2012.**

$$S_s = \frac{D\,\Omega\,\mu L^3}{8\,F\,h^2} \quad (4)$$

The vertical resultant power is normal, where the heap is because of the rotor weight; for this situation, the position a diary takes in the bearing guarantees that the heap is without a doubt vertical [13]. On the off chance that the greatness of this heap is known, the bearing flightiness might be gotten by revamping Eq. (3) to give Eq. (5), where S_s from Eq. (4) is called adjusted Sommerfeld number or Ocvirk number and is known for a specific speed, load, and oil consistency, **Michael, et al,.2012**.

$$\epsilon^8 - 4\epsilon^6 + \left(6 - S_s{}^2(16-\pi^2)\right)\epsilon^4 - \left(4+\pi^2 S_s{}^2\right)\epsilon^2 + 1 = 0 \quad (5)$$

The values of eccentricity ratio $\epsilon\ is\ equal\ \frac{\check{C}-C}{h}$ always occupied among 0-1 so the assessment of ϵ has been initiate by repetition method from 0 to 6000 RPM by supercomputer program of MATLAB.

At the point when a straight bearing model is utilized in machine, the uprooting ought to be checked to be little on the grounds [14] that a direct examination does exclude any requirements on the relocation, we considered just short bearing so the grids is 2×2 for firmness and damping networks [15] could be found as, **Michael, et al,.2012**.

$$K = \frac{F}{h}\begin{bmatrix} a_{xx} & a_{xy} \\ a_{yx} & a_{yy}\end{bmatrix} = \begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy}\end{bmatrix} \quad (6)$$

$$c = F/(h\times\Omega)\begin{bmatrix} b_{xx} & b_{xy} \\ b_{yx} & b_{yy}\end{bmatrix} = \begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy}\end{bmatrix} \quad (7)$$

Where $h_o = \frac{1}{(\pi^2(1-\varepsilon^2)+16\epsilon^2)^{3/2}}$ (8)

$$a_{xx} = h_o \times 4(\pi^2(2-\varepsilon^2)+16\epsilon^2) \quad (9)$$

$$a_{xy} = h_o \times \frac{\pi((\pi^2(1-\varepsilon^2)^2)-16\epsilon^4)}{\epsilon\sqrt{1-\epsilon^2}} \quad (10)$$

$$a_{yx} = -h_o \times \frac{\pi(\pi^2(1-\varepsilon^2)(1+2\varepsilon^2)+32\epsilon^2(1+\varepsilon^2))}{\epsilon\sqrt{1-\epsilon^2}} \quad (11)$$

$$a_{yy} = h_o \times 4(\pi^2(1+2\varepsilon^2)+\frac{32\epsilon^2(1+\varepsilon^2)}{(1-\varepsilon^2)} \quad (12)$$

$$b_{xx} = h_o \times \frac{2\pi\sqrt{1-\epsilon^2}}{\epsilon}\frac{(\pi^2(1+2\epsilon^2)-16\epsilon^2)}{} \quad (13)$$

$$b_{xy} = b_{yx} = -h_o \times 8(\pi^2(1+2\epsilon^2)-16\epsilon^2) \quad (14)$$

$$b_{yy} = h_o \times \frac{2\pi((\pi^2(1-\varepsilon^2)^2)+48\epsilon^2)}{\epsilon\sqrt{1-\epsilon^2}} \quad (15)$$

The firmness framework isn't symmetric, along these lines, hydrodynamic direction is anisotropic backings in to the machine. The MATLAB PC program has been intended to think about the relations among whimsy and changed Sommerfeld number, at that point getting the connection between adjusted Sommerfeld number and firmness and Sommerfeld with damping. We consider the impact of dynamic powers following up on the orientation, for the most part the power relocation connection is nonlinear at the same time, its gave that adequacy of resultant is little, so can accept a direct power removal connection. We think about short bearing (L/D<1), where the framework are 2×2 , the firmness and damping networks might be written in shut shape in term of flightiness and load as appeared in Eqs.(6) and (7). The portrayal [16] of spring and damper of COMBI214 is 2-measurements component with longitudinal pressure and pressure ability as appeared **Fig.3**.

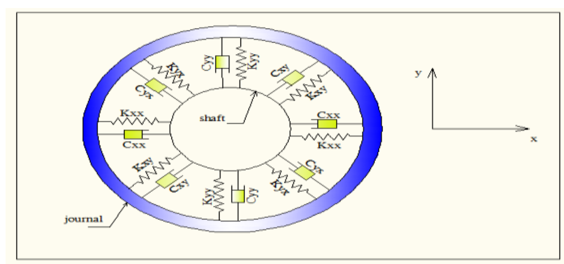

Figure 3: Springs and dampers with cross coupling of oil film journal bearing COMBI214 element geometry.

2.2 Dynamic Equations

The idea of rotor elements has exhibited by utilizing [17] the rotor which has circle lies at un rise to separate from orientation, the direction that taken with the rotor is short diary bearing L< D and $A \neq B$ as shown in **Fig.1**

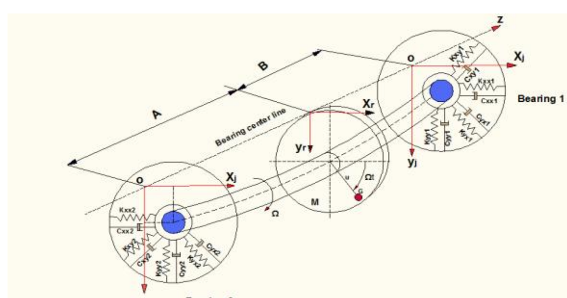

Figure 1: The bearings that taken with the rotor springs and damper representation

The rotor comprises of long adaptable shaft with adaptable diary bearing on the two finishes the direction has bolster solidness $K_{xx}, K_{yy}, K_{xy}\ and\ K_{yx}$ related with restraining $C_{xx}, C_{yy}, C_{xy}\ and\ C_{yx}$ in the two finishes in bearing 1 and bearing 2 as appeared in Fig.3, there is a plate m_d and the mass of shaft is m_s, the corresponding mass of the rotor is, **Michael, et al,.2012**.

$$m = \frac{17}{35} m_s + m_d \tag{16}$$

The focal point of gravity of plate is balanced from the pole geometry focus by an unconventionality, the movement of circle focus is portrayed by two translational removals (*xr*, *yr*).

In the event that have breathing of break case (opening and shutting), it is anything but difficult to work with directions lies on the rotor and pivots with it, at that point the decrease in firmness could be determined in both ζ and η bearings and exchange the solidness network to the settled directions with joining framework latency to solidness for finding the condition of movement of settled directions (x,y) as shown in **Fig.4**

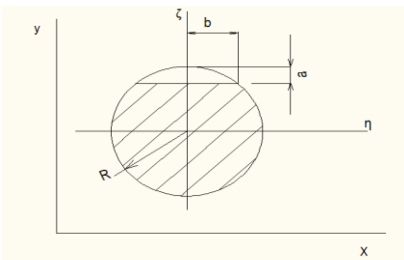

Figure 4: Cross section of cracked area

The solidness network for turning facilitates (ζ,η) for uncracked shaft is[K], the decrease of firmness because of split is is$[K_c(\theta)]$, θ is angle between rotor response and crack axis, the stiffness of cracked rotor is

$$[K] - [K_c(\theta)] = [K_{cr}] \tag{17}$$

The transformation matrix is used to convert the stiffness matrix from rotating coordinates to fixed coordinates **John and Michael, 2002.**

$$[A] = \begin{bmatrix} \cos(\Omega t) & \sin(\Omega t) \\ -\sin(\Omega t) & \cos(\Omega t) \end{bmatrix} \tag{18}$$

To get the cracked stiffness matrix as follows.

$$[K_{cr}] = [A]^T[K][A] - [A]^T[K_c(\theta)][A] = [K] - [K_c(\theta,t)] \tag{19}$$

∴ The stiffness matrix of cracked rotor with fixed coordinates is

$$[K_{cr}] = [K] - [K_c(\theta,t)] \tag{20}$$

Actually the total deflection of the system is consisted of static and dynamic deflection.

$$y = y_{st} + y_{dy} \tag{21}$$

Differentiate Eq. (21) twice with respect to time.

Since $y_{st} = constant$ that means $\dot{y}_{st} = 0$

$$\dot{y} = \dot{y}_{dy} \ \& \ \ddot{y} = \ddot{y}_{dy} \tag{22}$$

$$[M]\{\ddot{y}_{dy}\} + \big[[D] + [G]\big]\{\dot{y}_{dy}\} + \big[[K] - [K_c(\theta,t)]\big]\{y_{st} + y_{dy}\} = Q_u + W \tag{23}$$

Where

Q_u: Is the out of balance force, W: is the weight of the rotor.

The gyroscopic and damping have been respected in the skew symmetric lattice [G] and in symmetric positive semi-distinct matrix[D].

at the point when the damping network [D] in rotor is axisymmetric then will create skew symmetric commitment to uncracked framework [K] , alluding to Eq. (23) as an entire condition, when the case is consistent express the avoidance of the rotor for one transformation, $[K_c]$ changes, the $[K] > [K_c(\theta,t)]$,

$$[K]\{y_{st}\} = W \tag{24}$$

Then equation (23) could be written as.

$$[M]\{\ddot{y}_{dy}\} + \big[[D] + [G]\big]\{\dot{y}_{dy}\} + \big[[K] - [K_c(t)]\big]\{y_{dy}\} = Qu \quad 25$$

Eq. (25) is nonlinear because $[Kc(t)]$ is nonlinear and changes with the time t. That is meaning, it depends on speed and crack depth **John and Michael, 2002**.

To locate the basic speed, the consonant reaction because of unbalance mass can be determined at plate area where the most extreme reaction happens and in this way the basic speed will be the rotor speed at the greatest reaction relocation Rao, 2011. For symphonious movement condition progresses toward becoming. As got the two conditions of movement.

$$m\ddot{x}_r + K(x_r - x_j) = m_u\, e\, \Omega^2 \cos \Omega t \tag{26}$$

$$m\ddot{y}_r + K(y_r - y_j) = m_u\, e\, \Omega^2 \sin \Omega t$$

and

$$\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix}\begin{Bmatrix} x_j \\ y_j \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix}\begin{Bmatrix} i\Omega x_j \\ i\Omega y_j \end{Bmatrix} + \begin{bmatrix} K & 0 \\ 0 & K \end{bmatrix}\begin{Bmatrix} x_j \\ y_j \end{Bmatrix} = \begin{bmatrix} K & 0 \\ 0 & K \end{bmatrix}\begin{Bmatrix} x_r \\ y_r \end{Bmatrix} \tag{27}$$

Where $x_j = Be^{i\Omega t}$, $\dot{x}_j = i\Omega Be^{i\Omega t} = i\Omega x_j$

$y_j = De^{i\Omega t}$, $\dot{y}_j = i\Omega De^{i\Omega t} = i\Omega y_j$

$$K_{xx} = (K_{xx1} + K_{xx2}), \quad K_{xy} = (K_{xy1} + K_{xy2})$$

$$K_{yy} = (K_{yy1} + K_{yy2}) \quad K_{yx} = (K_{yx1} + K_{yx2})$$

$$C_{xx} = (C_{xx1} + C_{xx2}) \quad C_{xy} = (C_{xy1} + C_{xy2})$$

$$C_{yy} = (C_{yy1} + C_{yy2}) \quad C_{yx} = (C_{yx1} + C_{yx2})$$

The above conditions of movement give signs about the movements in x and y headings are both decoupled if there should be an occurrence of static and dynamic for this model subsequently they can be comprehended independently to discover reaction adequacy in x and y bearings whenever.

2.3 Modeling and Design Data Input

The displaying highlights for rotor and bearing help adaptability are depicted in this papers and show how component BEAM188, COMBI214 are utilized to demonstrate the pole with heading and MASS21for circle to demonstrate the majority. The firmness and damping of the two orientation with cross coupling bearings and solidness with damping of the pole has been considered in this investigation incorporating their varieties with the changing of rotational turn speed to get precise outcomes. Bar component (BEAM188) is useful for breaking down slim to decently thickset or thick shaft structures, it is a straight (2 – hubs) as appeared in Fig.5, beam188 component has six or seven degrees of opportunity at every hub, with the quantity of degrees of opportunity relying upon the KEYOPT(1) esteem, when KEYOPT(1) = 0 at every hub, these incorporate interpretations in the x,y and z bearings and revolutions about x,y and z tomahawks, when KEYOPT(1) =1, a seven level of opportunity (twisting greatness) is additionally considered, this component is appropriate for direct vast connection or potentially substantial strain nonlinear applications, MASS21 component for lighted mass plate is a point component having up to six level of opportunity uprooting in x,y,z headings and turn about x,y and z tomahawks. The cross segment territory in split district as component has less cross segment zone than alternate segments on the pole then we entered the estimations of solidness and damping of diary heading for every 1000 rpm from 0 rpm to 6000 rpm taking the variable estimations of unbalance mass, an alternate mass and turning inactivity might be doled out to each organize course, Nagaraju and Srinivas, 2014.

Tables 1 and 2 speak to the estimations of firmness and damping of heading with the varieties of turn speed so can speak to the COMBI214 component in ANSYS for each bearing and at any speed, the outcomes are from MATLAB program which has been intended to illuminate a lot all things considered. The benefits of damping coefficients diminished with rotational speed till to 2000 RPM then its change little and its around considered steady qualities however the cross coupling esteems keep diminishing while the solidness likewise change quickly till to 2000, is appear to be steady and firmness in cross coupling bearings isn't equivalent and differed little sums as appeared in Figs. (6) and (7).

Table 1: Properties of Bearing number 1, (Stiffness and damping).

Speed RPM	**Eccentricity ratio**	$K_{xx}1 * 10^3$	$K_{xy}1 * 10^3$	$K_{yx}1 * 10^3$	$K_{yy}1 * 10^3$	$c_{xx}1 * 10^3$	$c_{xy}1 * 10^3$	$c_{yx}1 * 10^3$	$c_{yy}1 * 10^3$
500	0.4419	3010	1646.8	-5273.5	3307.5	92.808	-58.21	-58.21	171.52
1000	0.2938	3199.2	3584.5	-5992.5	2345.2	78.512	-30.726	-30.726	104.39
1500	0.2158	3275.2	5448.3	-7221.7	2032.2	74.329	-20.916	-20.916	86.99
2000	0.1690	3311.0	7291.4	-8682.6	1895.6	72.552	-15.839	-15.839	79.988
2500	0.1382	3330.2	9133.1	-10272	1825.0	71.689	-12.737	-12.737	76.550
3000	0.1167	3341.4	10965	-11927	1784.4	71.157	-10.640	-10.640	74.581
3500	0.1008	3348.5	12804	-13636	1759.0	70.870	-9.1422	-9.1422	73.406
4000	0.0887	3353.2	14634	-15365	1742.2	70.641	-8.0095	-8.0095	72.593
4500	0.0791	3356.6	16475	-17128	1730.4	70.532	-7.1259	-7.1259	72.080
5000	0.0714	3359.0	18303	-18893	1721.9	70.411	-6.4174	-6.4174	71.668
5500	0.0650	3360.8	20149	-20685	1715.6	70.377	-5.8368	-5.8368	71.418
6000	0.0597	3362.2	21973	-22466	1710.7	70.288	-5.3523	-5.3523	71.165

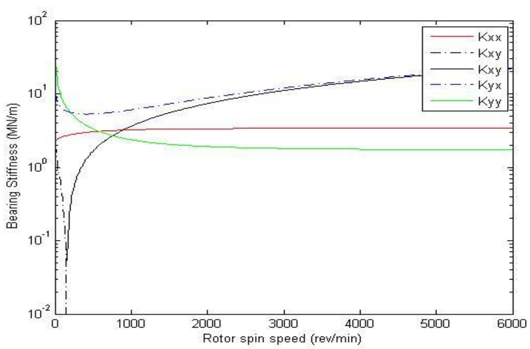

Figure 6: Stiffness of the journal bearing verses spin speed of rotor.

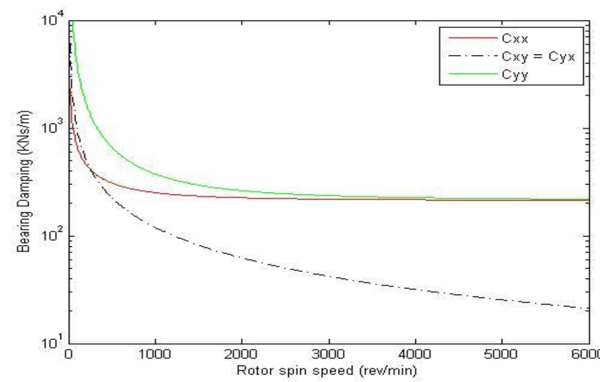

Figure 7: Rotor spin speed verses damping of fluid film bearing

Table 2: Properties of bearing number 2, (Stiffness and damping).

Speed RPM	**Eccentricity ratio**	$K_{xx}2 * 10^3$	$K_{xy}2 * 10^3$	$K_{yx}2 * 10^3$	$K_{yy}2 * 10^3$	$c_{xx}2 * 10^3$	$c_{xy}2 * 10^3$	$c_{yx}2 * 10^3$	$c_{yy}2 * 10^3$
500	0.3703	2251.4	1760.4	-3956.9	2008.4	85.47	-43.38	-43.38	132.91
1000	0.2289	2364.4	3663.7	-5025.7	1504.7	75.68	-22.658	-22.658	102.74
1500	0.1624	2401.8	5527.7	-6496.4	1361.4	73.096	-15.318	-15.318	79.99
2000	0.1249	2417.7	7385.5	-8131.4	1303.3	72.096	-11.556	-11.556	76.079
2500	0.1012	2425.7	9237.4	-9842.1	1274.7	71.588	-9.2718	-9.2718	74.169
3000	0.0849	2430.2	11094	-11601	1258.6	71.337	-7.739	-7.739	73.142
3500	0.0731	2433.0	12944	-13381	1248.7	71.156	-6.6406	-6.6406	72.488
4000	0.0641	2434.9	14806	-15189	1242.2	71.069	-5.8144	-5.8144	72.118
4500	0.0571	2436.1	16655	-16997	1237.7	71.007	-5.1708	-5.1708	71.817
5000	0.0515	2437.1	18494	-18802	1234.5	70.901	-4.6553	-4.6553	71.558
5500	0.0469	2437.7	20330	-20611	1232.1	70.811	-4.2331	-4.2331	71.355
6000	0.0430	2438.3	22193	-22450	1230.3	70.823	-3.8811	-3.8811	71.281

3. Results and Discussion

Table 4 demonstrates the component of the chose model. For Critical Speed in this examination, the primary Eigen recurrence investigation are done on the split rotor display for speed extend from 0 to 6000 rpm utilizing various load step, the basic Eigen recurrence of the rotor demonstrate comparing to various rotational turn speeds are plotted. The rotor is upheld by two tilting cushion short orientation, firmness and damping coefficients of course are changed with turn speed and for this situation the regular recurrence of the framework is fluctuated, if the normal recurrence equivalent to pivoting turn speed, the speed is called basic speed.

Table 5 shows the first analytical critical speed of uncracked model is 6400 RPM and 6150 RPM found by numerical with 3.91% error between them. **Table 6** shows the maximum amplitude response is 0.0409mm analytically and 0.0396mm numerically with 3.18% error between analytical and numerical. From **tables 5 & 6** are clear the total increasing in response from uncracked to 0.8R depth is 34% analytically and response increases only 4.88% from uncracked to 0.2R depth. The critical speed decreasing 17.18% from uncracked to 0.8R depth and only 3.12% decreasing from uncracked to 0.2R.

The comparison has been done for two methods by plotting the frequency response of uncracked and cracked shaft. It's clear that the amplitude response increases while the critical speed decreases with increasing the crack depths as shown in **Fig. 8** for analytical analysis and in **Fig. 9** for numerical analysis.

Table 5: Results of Response with crack depths (single crack) of $A \neq B$ rotor

Maximum amplitude response of $A \neq B$ model for single transverse crack					
	Uncracked	**Crack depth mm = a/R**			
Method		**20= 0.8R**	**15= 0.6R**	**10= 0.4R**	**5 = 0.2R**
Analytical response mm (change percentage)	0.0409	0.0430 (4.88%)	0.0464 (11.8%)	0.05438 (24.78%)	0.06201 (34%)
Numerical response mm (change percentage)	0.0396	0.0406 (2.46%)	0.0446 (11.2%)	0.0529 (25.14%)	0.0593 (33.2%)
Percentage of discrepancy between analytical and Numerical	3.18%	5.58 %	3.88 %	2.72 %	4.37%

Table 6: Results of Critical Speed with crack depths (single crack) of $A \neq B$ rotor

Critical Speeds of $A \neq B$ model for single transverse crack					
	Uncracked	**Crack depth mm = a/R**			
Method		**5 = 0.2R**	**10= 0.4R**	**15= 0.6R**	**20= 0.8R**
Analytical critical speed (change percentage)	6400 rpm	6200 rpm (3.12%)	6060 rpm (5.13%)	5800 rpm (9.37%)	5300 rpm (17.18 %)
Numerical critical speed (change percentage)	6150 rpm	6050 rpm (4.31%)	5800 rpm (5.69%)	5600 rpm (8.94%)	5400 rpm (12.2%)
Discrepancy Between Analytical and Numerical	3.91 %	2.42%	4.29%	3.45%	1.89%

- percentage change of response $= \left|\frac{cracked\ response - uncracked\ response}{uncracked\ response}\right| \times 100\%$
- percentage change of cr.speed= $\left|\frac{uncracked - cracked}{uncracked}\right| \times 100\%$

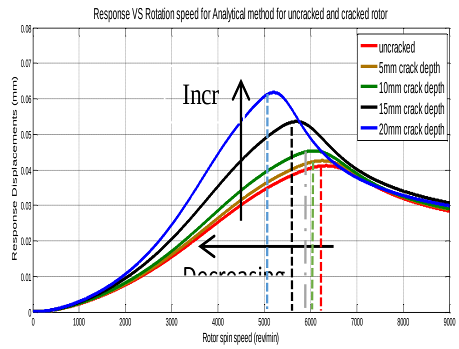

Figure 8: Analytical Response versus rotation speed for single cracked $A \neq B$ rotor

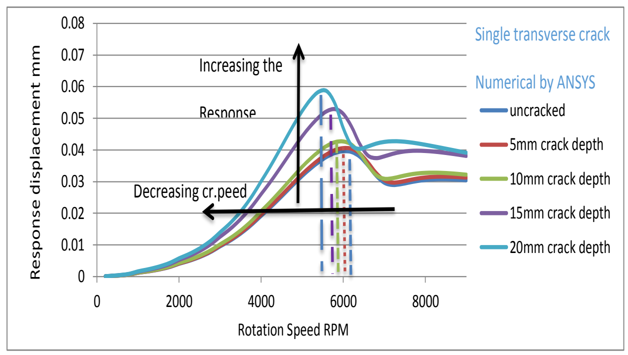

Figure 9: Numerical amplitude Response versus rotation speed for single cracked rotor of ($A \neq B$) model.

4. Conclusion

It tends to infer that;

1-The Eigen recurrence determined for broke rotor as a first basic speed by ANSYS, to get the essential recurrence which is the reverberation speed, it found 6000 rpm for a given components of rotor as in Table.4, reverberation speed is increasingly hazardous case to maintain a strategic distance from it by making its timeframe as little as conceivable to dodge disappointment.

2-The basic speed diminishes with expanding the split profundities for all models chose and the reaction increments with expanding the break profundities.

3- The effects of critical speed and response started as a sensible values and could not neglected at crack depths ≥ 0.3R, because the depths 0.2R shows no effect on critical speed and amplitude response and the 0.8R crack depth is the danger depth.

4- The discrepancy between analytical and Numerical within acceptable limits.

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