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Vibration response of multiple rotor systems with crack and misalignment

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Abstract: *In earlier research on the vibration signatures of a crack or misalignment fault have been typically been attempted considered as fault in one time. The conditions of simultaneous formation of crack and misalignment were ignored. The continuous formation crack and misalignment leads to the formation of fatigue crack in the rotor shaft. Fatigue cracking of the rotor shaft is the type of rotor fault that could result in catastrophic failure of the rotor shaft. The aim is to investigate the steady-state vibration response of misaligned multiple rotor with a defect of crack and misalignment fault. First where the crack and misalignment fault will be formed simultaneously was examined. Along with both torsional and axial vibration, the detailed analysis is carried out to overcome the fault. The influences of misalignment are examined with the help of parameters such as misalignment level, type of misalignment, crack size and crack location. The vibration analysis is done with the help of Ansys software.*

Keywords: *misalignment, crack, multi-fault modeling, misalignment level, type of misalignment*

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I. INTRODUCTION

In large rotating machinery, such as turbine, generator, and aero engine, the rotor is one of the most important parts. Appearance and expansion of fatigue crack in large rotating machines may lead to catastrophic failures. Due to the catastrophic failure fatigue crack will occur in the rotor shaft. Online detection of cracks is very important to engineers working in the area of the machine dynamics. However online detection is very difficult since the crack, due to gravity, opens and closes alternatively through rotation and the variation of rotor system's parameters caused by the small crack are so tiny that they are not easy to detect. The dynamic behavior of a cracked rotor is a subject of current interest. Rotors in industrial installations have various faults such as misalignment in shaft drive line, temporary bow, stiffness asymmetry etc. But it cannot be completely removed. Although efforts are always made to keep them within the acceptable limits. During the course of time one or more of these faults either grow from their residual levels or creep into the system if absent initially. Also the prolonged existence of one fault may lead to the development of another fault. The presence of more than one fault is quite possible for real rotors. The presence of one rotor fault sometimes influence the dynamics of the other fault and the rotor system exhibits different vibration characteristics. Therefore investigation on the vibration response of the multiple faults in the multiple rotor system has to be done with varying the parameters

II. EQUATIONS OF MOTIONS

The figure (a) shows the schematic of the coupled rotors connected by a three-pin bush-type flexible coupling and simply supported on rigid bearings. Simplification on the bearing modeling has an advantage in that the observed vibration behavior and symptoms could directly relate to the fault without worrying about the possible masking effect from the bearing. The finite element (FE) model of the coupled rotor system is

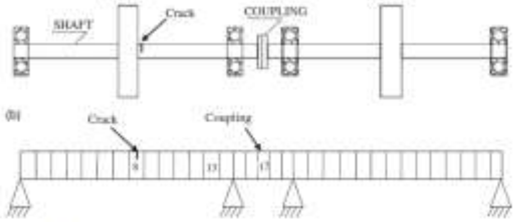


Fig. 1 (a) Coupled rotor system and (b) FE model of the system

shown in Fig. 1(b). Rotor-1 is assumed to have a transverse fatigue crack. The misalignment is considered between the two rotor shafts (i.e. between coupling halves). The shafts of diameter D are supported on the rigid supports with a span length of L. Discs of mass m are placed at the centre of the span on both the shafts. Disc and coupling half inertias (i.e. translational and rotational both) are lumped at their respective node locations. The misalignment effect is simulated by introducing the misalignment forces and moments at the coupling nodes. The crack is located at the centre of the rotor shaft in Fig. 1. The effects of crack location are also studied by changing the crack location to element. The stiffness matrix of the beam element at the crack location is replaced with the stiffness matrix of the crack element. For this purpose, the crack stiffness matrix. The equations of motion can be written in generalized form
as

$$[\mathbf{M}]\{\ddot{\mathbf{q}}\} + [\mathbf{C}]\{\dot{\mathbf{q}}\} - \omega[\mathbf{G}]\{\dot{\mathbf{q}}\} + [\mathbf{K}]\{\mathbf{q}\} = \{\mathbf{F}\}$$

where [**M**], [**C**], [**G**], and [**K**] are the mass, damping, gyroscopic, and stiffness matrices, respectively. [**C**] = αd [**K**] + βd [**M**], the constants αd =0.8132 rad/s, and βd = 7.185E-5 s/rad are found assuming modal damping ratios of 0.01 and 0.05 for the first two modes. {**F**} is the external excitation force vector due to unbalance and misalignment. {**q**} is the generalized co-ordinate of the system. ω is the rotational speed. The FE model used for studying the vibration response of a misaligned coupled rotor system is used here with the inclusion of a fatigue crack.

2.1 FE MODEL OF THE CRACKED ROTOR SEGMENT

The rotor segment with crack is represented by a beam element with six DOF per node (Fig. 1). Consider a rotor segment containing a transverse surface crack of depth a as

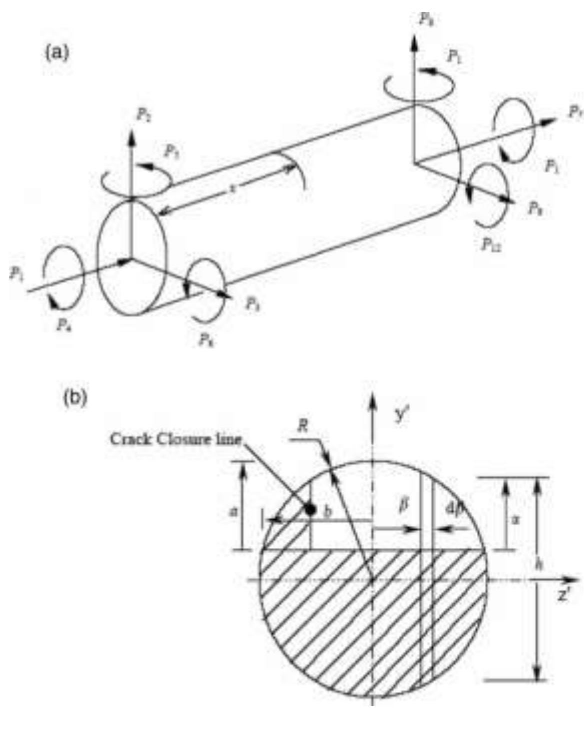


Fig.2 (a)Shaft FE and (b) Crack cross section.

Let the shaft element be of diameter D and length l. The element is under the action of shear forces P2, P3 and P8, P9, bending moment P5, P6 and P11, P12, axial forces P1 and P7, and torsional moments P4 and P10.The crack is situated at a distance x from the left end of the element.

To account for the crack effect, the stiffness matrix of a Timoshenko beam element is modified accounting for all the possible vibration coupling phenomena that exist in a cracked rotor. The beam element with modified stiffness matrix then fits into the complete FE assemblage representing a rotor bearing system and is used for study. complete mathematical derivation can be found.

Fig 3. Pro-E model of rotor shaft without crack

The figure 4 shows the Pro-E model of the rotor shaft with crack on the shaft

Fig 4. Pro-E model of the rotor shaft with crack on the shaft

2.2 CALCULATION OF MISALIGNMENT
FORCES AND MOMENTS

The figure 5 shows the schematic of the coupling halves with an exaggerated misalignment level.

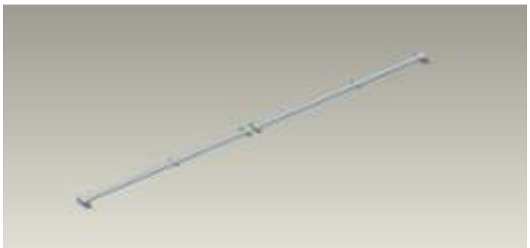


Fig.5 misalignment coupling shaft with rotor

III. VIBRATIONANALYSIS USING ANSYS SOFTWARE

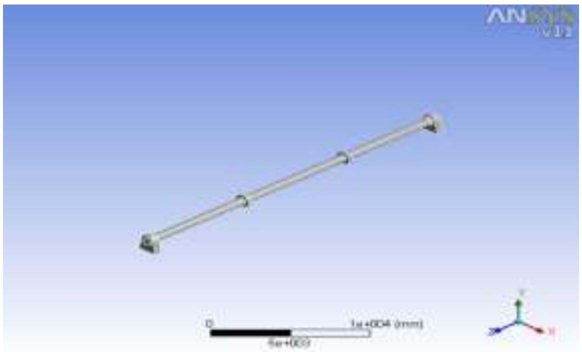


Fig.6 Meshed diagram through ANSYS .without crack

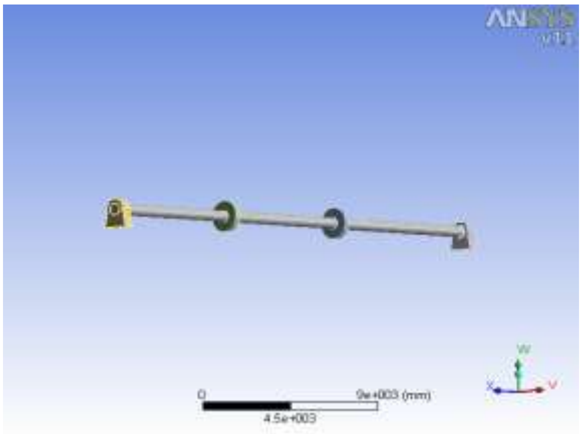


Fig.7 Meshed diagram through ANSYS .with crack

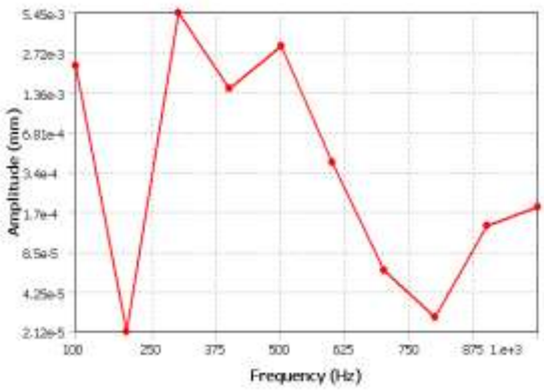


Fig 8 frequency VS Amplitude for the rotor without crack

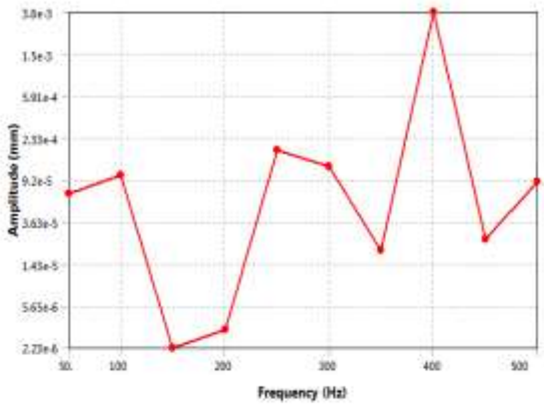


Fig .9 frequency VS Amplitude for the rotor with crack

IV. RESULT AND DISCUSSION

Strain Energy Calculation

The strain energy stored by the cracked shaft element is calculated by varying the a/D ratio from 0.25,0.28,0.30,0.32 and 0.36 with varying the length of crack from 0.025m,0.225m,0.475m,0.725m,0.975m.

(1) When a/D ratio is 0.25 with varying the length the tabular column shows the amount of strain energy stored in both uncracked and cracked element

(2) Table 6.1.1 shows the Strain energy with a/D ratio is 0.25 with varying the length

Displacement Node	Strain energy stored in uncracked shaft	Strain energy stored in cracked shaft				
		Length of crack				
		0.025m	0.225m	0.475m	0.725m	0.975m
u1	6.06E-05	8.03E-05	7.65E-05	7.08E-05	7.01E-05	6.89E-06
u2	8.19E-05	7.60E-05	6.86E-05	6.40E-04	6.36E-04	6.02E-05
u3	7.65E-04	6.85E-04	6.67E-04	5.90E-04	5.20E-04	6.80E-05
u4	0.0265	6.76E-05	6.54E-04	6.32E-05	6.12E-05	5.80E-06
u5	7.27E-04	7.01E-04	6.78E-05	6.56E-05	6.30E-05	6.60E-06
u6	5.86E-04	6.25E-05	6.02E-05	6.15E-05	6.02E-05	5.60E-06

When a/D ratio is 0.28 with varying the length of crack the tabular column shows the amount of strain energy stored in both uncracked and cracked element

Table 6.1.2 shows the Strain energy with a/D ratio is 0.28 with varying the length

Displacement Node	Strain energy stored in uncracked shaft	Strain energy stored in cracked shaft				
		Length of crack				
		0.025m	0.225m	0.475m	0.725m	0.975m
u1	6.06E-05	5.90E-06	5.60E-05	5.02E-05	4.56E-06	4.12E-06
u2	8.19E-05	7.06E-05	6.56E-05	6.06E-05	5.45E-06	5.03E-06
u3	7.65E-04	6.03E-04	5.76E-04	5.08E-04	6.07E-05	6.00E-05
u4	0.0265	6.12E-04	5.08E-04	5.00E-05	4.86E-05	4.56E-05
u5	7.27E-04	6.86E-04	6.06E-04	5.96E-05	5.65E-05	5.08E-05
u6	5.86E-04	4.26E-04	4.03E-04	3.76E-05	4.02E-04	3.93E-05

When a/D ratio is 0.30 with varying the length of crack the tabular column shows the amount of strain energy stored in both uncracked and cracked element.

Table 6.1.3 shows the Strain energy with a/D ratio is 0.30 with varying the length

Displacement Node	Strain energy stored in uncracked shaft	Strain energy stored in cracked shaft				
		Length of crack				
		0.025m	0.225m	0.475m	0.725m	0.975m
u1	6.06E-05	7.60E-05	7.08E-05	7.00E-06	6.89E-06	6.52E-06
u2	8.19E-05	6.86E-05	6.54E-05	6.36E-04	5.32E-06	5.20E-05
u3	7.65E-04	6.50E-04	6.32E-04	6.03E-04	5.56E-05	5.35E-05
u4	0.0265	6.35E-04	6.22E-04	5.89E-05	5.66E-05	5.38E-05
u5	7.27E-04	7.08E-04	6.50E-05	6.32E-05	6.21E-05	5.60E-06
u6	5.86E-04	2.08E-	5.20E-	5.05E-	3.56E-	3.06E-

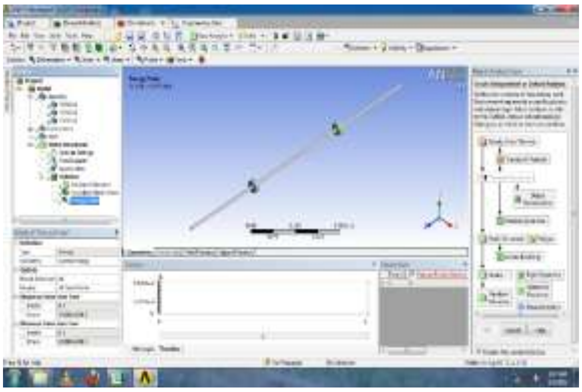
		04	05	05	05	06

When a/D ratio is 0.32 with varying the length of crack the tabular column shows the amount of strain energy stored in both uncracked and cracked element.

Table 6.1.4 shows the Strain energy with a/D ratio is 0.32 with varying the length

Displacement Node	Strain energy stored in uncracked shaft	Strain energy stored in cracked shaft				
		Length of crack				
		0.025m	0.225m	0.475m	0.725m	0.975m
u1	6.06E-05	5.67E-05	6.76E-06	5.05E-05	5.32E-05	5.06E-05
u2	8.19E-05	7.40E-05	5.60E-05	5.09E-06	4.04E-06	4.10E-06
u3	7.65E-04	7.40E-04	6.60E-04	4.40E-04	8.60E-05	7.60E-05
u4	0.0265	6.26E-04	4.20E-04	8.30E-05	8.12E-05	7.06E-05
u5	7.27E-04	6.86E-04	4.60E-04	6.80E-05	6.32E-05	6.02E-05
u6	5.86E-04	6.86E-05	5.60E-05	5.32E-05	6.43E-06	6.56E-06

CALCULATION OF STRAIN ENERGY USING ANSYS



The maximum Strain energy absorbed by the rotor system when there is no crack is 5.19×10^{-4}J,But the maximum strain energy absorbed by the rotor system by the presence of crack is 4.82×10^{-4} J at a/D ratio 0.25.

CALCULATION OF NATURAL FREQUENCIES

The Natural frequencies are calculated using Ansys software by varying the a/D ratio with varying the crack length.

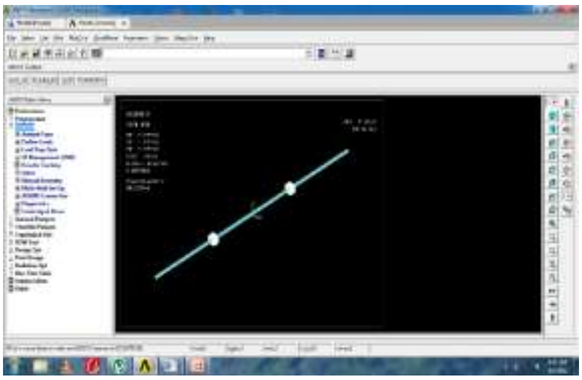


Fig 6.2.1 Meshed model of rotor system

The figure 6.2.2 shows the calculation of Natural frequency in Ansys

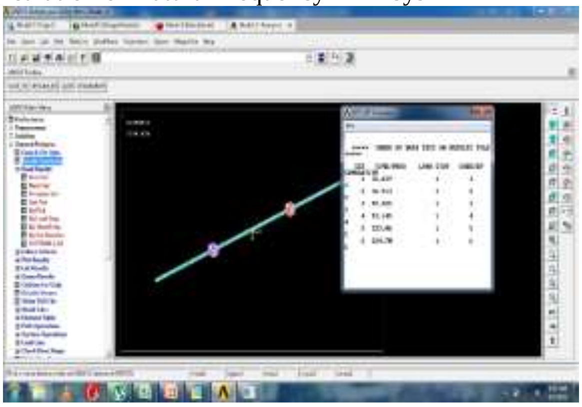


Fig. 6.2.2 Natural frequency in Ansys.

When a/D ratio is 0.25 with varying the length, the tabular column shows the natural frequencies for uncracked and cracked shaft element.

Table 6.2.1 shows the Natural frequency of a/D ratio is 0.25 with varying the length

Frequency in Hz	Uncracked	Position of the crack				
		0.025m	0.0225m	0.475m	0.725m	0.975m
1st frequency(HZ)	37.438	37.05	36.98	37.03	36.11	36.06
2nd frequency(HZ)	37.712	37.22	37.31	37.23	37.33	37.46
3rd frequency(HZ)	93.637	92.99	92.97	92.92	92.67	92.87
4th frequency(HZ)	94.32	93.41	93.27	93.47	93.08	93.17
5th frequency(HZ)	228.35	226.82	226.98	227.04	227.56	227.8
6th frequency(HZ)	229.97	228.81	228	227.79	227.68	227.6

(1) When a/D ratio is 0.28 with varying the length, the tabular column shows the natural frequencies for uncracked and cracked shaft element.

Table 6.2.2 shows the Natural frequency of a/D ratio is 0.28 with varying the length

Frequency in Hz	Uncracked	Position of the crack				
		0.025m	0.0225m	0.475m	0.725m	0.975m
1st frequency(HZ)	37.438	36.42	36.99	37.029	37.006	37.045
2nd frequency(HZ)	37.712	36.504	37.32	37.266	37.63	37.426
3rd frequency(HZ)	93.637	92.08	92.99	92.968	93.42	93.06
4th frequency(HZ)	94.32	93.43	93.386	93.464	93.399	93.46
5th frequency(HZ)	228.35	227.67	227.17	226.93	227.77	227.03
6th frequency(HZ)	229.97	228.21	228.24	228.22	227.36	228.3

(1) When a/D ratio is 0.30 with varying the length, the tabular column shows the natural frequencies for uncracked and cracked shaft element.

Table 6.2.3 shows the Natural frequency of a/D ratio is 0.30 with varying the length

Frequency in Hz	Uncracked	Position of the crack				
		0.025m	0.0225m	0.475m	0.725m	0.975m
1st frequency(HZ)	37.438	37.077	36.98	37.029	37.01	37.02
2nd frequency(HZ)	37.712	37.256	37.3	37.284	37.36	37.01
3rd frequency(HZ)	93.637	93.049	93.009	93.005	93.25	93.02
4th frequency(HZ)	94.32	93.332	93.324	93.489	93.07	93.28
5th frequency(HZ)	228.35	226.95	227.08	226.95	227.45	226.92
6th frequency(HZ)	229.97	228.66	228.21	228.05	227.38	228.025

V. CONCLUSION

Multi-fault modelling and vibration analysis of a rotor system with crack and misalignment has be modeled with rigid support. Steady-state, axial and torsional vibrations at the disc locations have been analyzed to study the effects with crack by varying the parameter as such as position of crack, length of crack and depth of crack.

The following conclusion were derived

- Strain energy decreases when the depth of crack increases with increase in length in the cracked shaft.
- Natural frequency decreases when the crack length is increase.
- Change in natural frequency is varied in random manner when compared with natural frequency of uncracked rotor system with cracked rotor system.
- The vibrations generated due to crack fault are much more than that was generated by the uncracked rotor system.

NOTATIONS

a - crack depth
a/D - crack depth ratio
A - shaft cross-section area
b - half-width of the crack
c - damping coefficient
[**C**] - damping matrix of the rotor system in stationary co-ordinate system
dy - parallel misalignment along y
co-ordinate direction
D - diameter of the shaft (rotor)
E - modulus of elasticity
F1, F2, FII, FIII - functions of crack parameters

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