Modelling and vibration analysis of rotor system with misaligned bearing

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Abstract: A misaligned rotor system, which refers to its practical rotating center line determined by supports offset from the theoretical center line due to possible manufacturing error or improper installation, often suffers bad operating state or structure damage risk. In the present work, vibrations of a rotor system with misaligned bearing and its reaction forces under different misaligned offsets are investigated. A new simulation model of a rotor test rig by use of the multi-body dynamic analysis software-MSC.ADMAS is proposed, considering different unbalance and misaligned operating conditions.The simulation results reveal that the misalignment has obvious effects on the vibrations of the rotor system and the reaction forces of misaligned bearing. While the misaligned system presents more complex behavior of 2×, 3× rotating frequencies besides the fundamental 1× one at higher rotating speed by comparing with that in the subcritical speed ranges. Additionally, axial vibrations can be aroused by misalignment, the peak of which will increase but the the amplitude of reaction forces will decrease along the misaligned offset.

Keywords: misaligned rotor system; flexible supports; simulation model; rotor vibration; bearing reaction

1 Introduction

Misalignment will bring out abnormal vibration for rotating machinery, and the secondary effects of misalignment often accompany, such as elevated temperature at or near the bearings, excessive amount of lubricant leakage at the bearing seals, and even premature bearing, seal, or coupling failures [1]. In spite of frequent observations in practice and unlike other malfunctions, only a few researchers have paid attention to misaligned rotor system, especially to the characteristics of misaligned bearing.

In addition, a majority of the recent studies on misalignment have mainly analyzed coupling misalignment, discussed the effects of its structure and type on the vibration of connected rotor system. However, different authors report different vibration signatures which are widely promoted as a useful tool for studying misalignment. And

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there is no clear-cut picture of signature characteristics uniquely attributable to misalignment in the literatures on this subject.

Gibbons [2] modelled the reaction forces and moments of misaligned coupling. Palazzolo [3] reported the gear couplings may produce large static forces and moments with misalignment, and then derived the model. Xu and Marangoni [4, 5] derived reaction torques of misaligned hook coupling, proposed corresponding experiment, and showed that the vibration responses due to coupling misalignment mainly occur at the even integer multiples of the rotating speed. Sekhar and Prabhu [6] derived and introduced reaction forces and moments due to flexible coupling misalignment in the model and the numerical and experimental work evaluated the 2× vibration response as to be an important characteristic feature of misalignment. Ganeriwala [7] carried out a series of experiments by use of Spectur Quest Machiner Fault Simulator under varying misaligned conditions. Lee and Lee[8] derived a dynamic model for misaligned rotor-ball bearing systems by treating the reaction loads and deformations at the bearing and coupling elements as the misalignment effect, and found that the whirling orbits tend to collapse and the natural frequency of the misaligned rotor system increases largely as the misalignment increasing. Hu et al. [9] designed a rig supported on four bearings which was suited for misalignment identification experiments, and illustrated some predicted effects of lateral alignment changes on the rig vibration behavior. Hussain and Redmond [10] examined the effect of misalignment on the lateral and torsional responses of two rotating shafts with theoretical and numerical analysis and concluded that misalignment can be a source of both torsional and lateral excitations. In the reference [11], it is shown that excitation at twice synchronous speed is developed and an expression for the magnitude and phase of the response is derived as to a rigidly coupled rotors mounted on idealized linear bearings. Redmond [12] presented a representative shaft misalignment model supported by three bearings, and the numerical results showed that only parallel misalignment may produce system dynamic response while angular misalignment may generate purely static forces and displacements by theoretical analysis without other excitation sources. In the reference [13], experimental studies were performed on a rotor dynamic test apparatus and in some case the misalignment response was hidden and did not show up in the vibration spectrum. The authors have dealt with rigid coupling misalignment of a hyperstatic shaft-line equipped with journal bearings in the reference [14] and their simulation results showed that super-harmonic components were the most remarkable effects of rigid coupling misalignment.

The main objective of the present work is to investigate effects of misalignment on the vibration of rotor system, the reaction forces of misaligned bearing. A new model of a rotor system is proposed by use of MSC.ADMAS, considering the unbalance and misaligned conditions. And then, the effects of misalignment on the vibrations of rotor and the reaction forces of misaligned bearing are investigated with simulation analysis.

2 Simulation model based on ADAMS

2.1 Test rig

The test rig illustrated in Fig. 1 consists of a rigid shaft, two flexible supports. The rigid shaft is connected with motor by a flexible coupling. The two support assembly are identical. The size and material parameters of the test rig are listed in Table 1.

The whole stiffness of support point is combined by the stiffness of bearing and its flexible support. It is supposed that the axial, lateral and angular stiffness coefficients are 1×10^7N/m, 1×10^7N/m and 1×10^4Nm/rad respectively, according to the practical measurements.

Fig. 1. Photo of test rig

Table 1. Parameters of Test rig

Parameters	Value
Length of shaft	0.480m
Diameter of shaft	0.036m
Young modulus of shaft	2.06×10^{11}Pa
Poisson ratio of shaft	0.3
Density of shaft	7.85×10^3kg/m^3

2.2 Simulation model

A new simulation model of the test rig shown in Fig. 1 by use of the multibody dynamic analysis software- MSC.ADMAS is implemented and proposed in this section, and the effects of misalignment on the vibration and reaction force of bearing are analyzed based on the simulation model as follows.

In the simulation model, the shaft is proposed as rigid, the bearings (B1, B2) of rotor test rig are simplified as three springs with constant stiffness and damping respectively in the horizontal, vertical and axial directions at normal condition. As to the misaligned condition, the misaligned bearing(B2) is modeled as two groups of bearings in parallel with a assumed distance 10mm in the horizontal and vertical directions, and the shaft with right upward is taken as two parallel shafts with offset Δ. The unbalance can be easily set by adding a rigid ball on the disc with certain mass and certain distance from the geometrical center of the disc.

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The initial conditions are assumed as follows: the imbalance is set to 150g.cm the length of misaligned shaft is set to 100mm, and the stiffness and damping coefficients of two bearings are set to 1×10^7N/m and 1×10^4N.s/m according to that of test rig.

The established Adams model of the test rig is shown in Fig. 2.

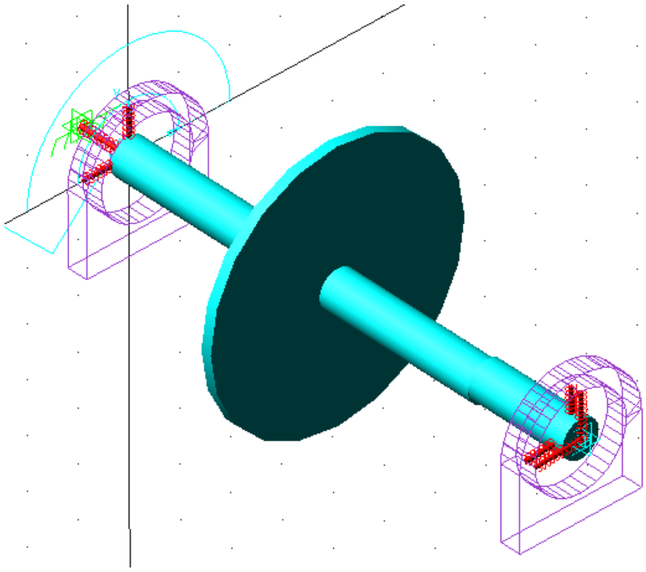


Fig.2. Simulation model of misaligned rotor system based on ADAMS

2.3 Results and discussions

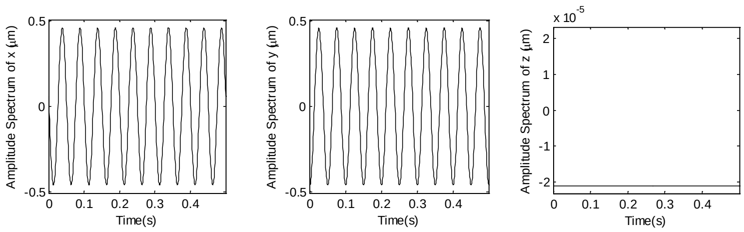
Based on the proposed simulation model in the above section, vibrations of the rotor system and the reaction forces of its misaligned support under different misaligned offsets are investigated. The misalignment offset values for B2 are set as 0-3 mm, and the rotating speeds of 20 Hz and 40 Hz in the sub-critical and super-critical ranges are chosen respectively. The four different operating conditions are choosen in Table 2.

Table 2. Operating conditions

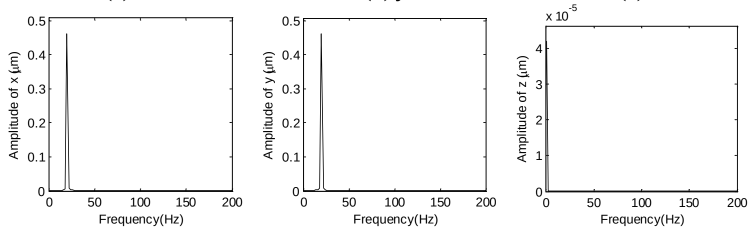
Cases	Misalignment conditions	Rotating speeds	Illustrations
Case 1	Normal condition (Δ=0mm)	20Hz, 40Hz	Figs.3,7,11,15
Case 2	1mm offset at B2 (Δ=1mm)	20Hz, 40Hz	Figs.4,8.12,16
Case 3	2mm offset at B2 (Δ=2mm)	20Hz, 40Hz	Figs.5,9,13,17
Case 4	3mm offset at B2 (Δ=3mm)	20Hz, 40Hz	Figs.6,10,14,18

Vibrations of misaligned rotor

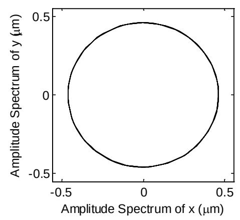
The vibration responses of shaft (*x*- vertical, *y*- horizontal, *z*- axial) near B2 under the different conditions shown in Table 2 are simulated, and the obtained results are illustrated in Figs. 3-10.



(a) *x* (b) *y* (c) *z*

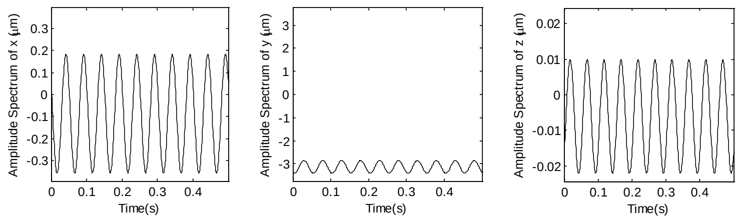


(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*

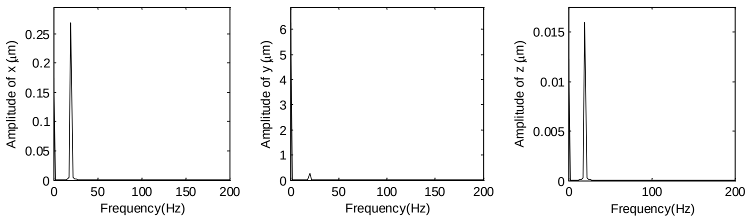


(g) orbit

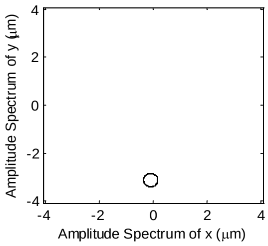
Fig. 3. Vibration responses of rotor under normal condition at rotating speed 20Hz



(a) *x* (b) *y* (c) *z*

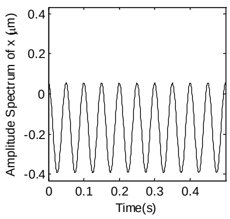


(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*

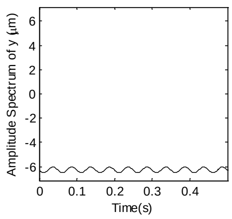


(g) orbit

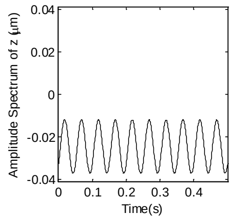
Fig. 4. Vibration responses of rotor with offset 1mm at rotating speed 20Hz



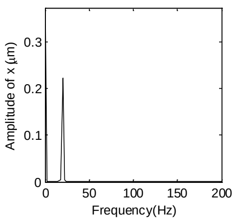
(a) *x*



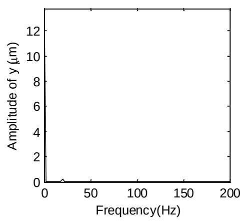
(b) *y*



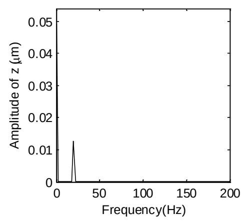
(c) *z*



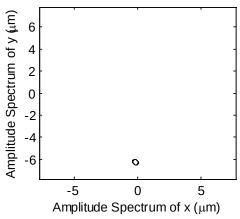
(d) spectrum of *x*



(e) spectrum of *y*

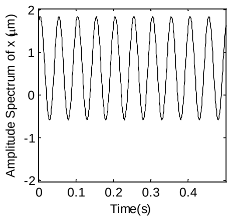


(f) spectrum of *z*

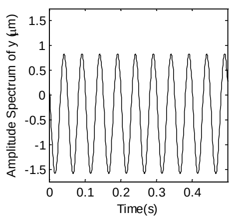


(g) orbit

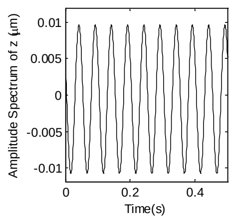
Fig. 5. Vibration responses of rotor with offset 2mm at rotating speed 20Hz



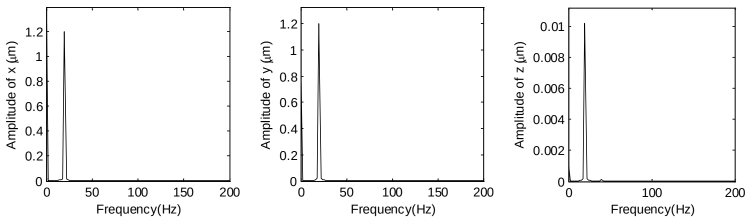
(a) *x*



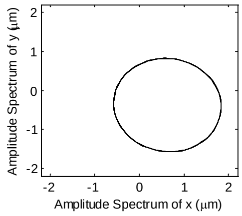
(b) *y*



(c) *z*

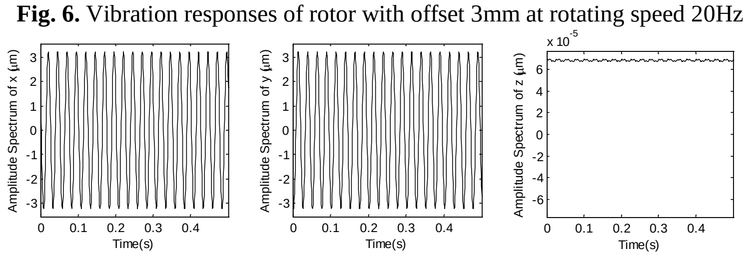


(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*

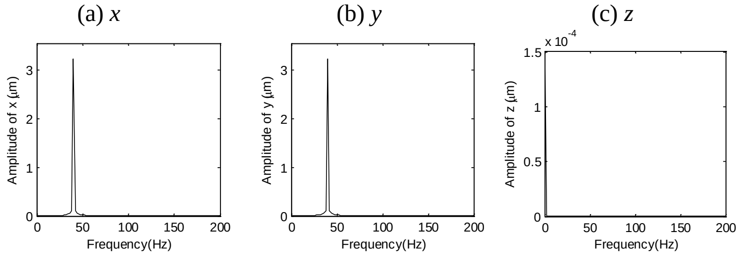


(g) orbit

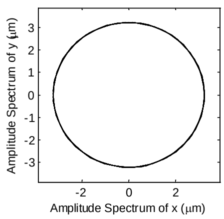
Fig. 6. Vibration responses of rotor with offset 3mm at rotating speed 20Hz



(a) *x* (b) *y* (c) *z*



(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*



(g) orbit

Fig. 7. Vibration responses of rotor under normal condition at rotating speed 40Hz

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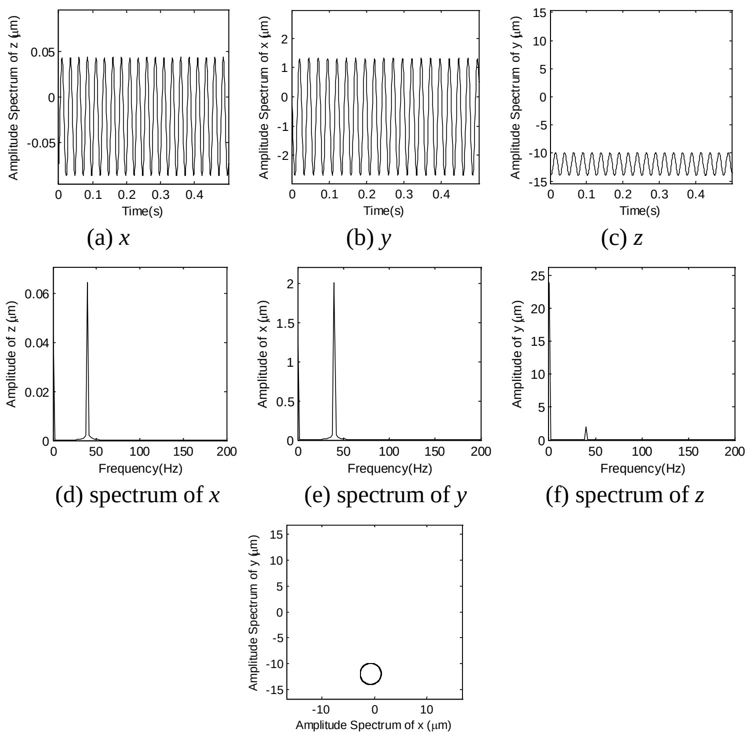
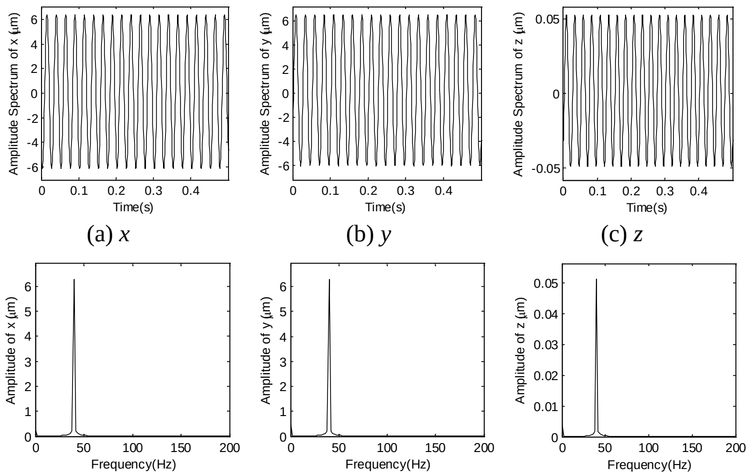


(a) *x* (b) *y* (c) *z*

(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*

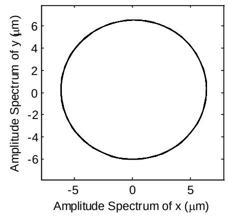
(g) orbit

Fig. 8. Vibration responses of rotor with offset 1mm at rotating speed 40Hz



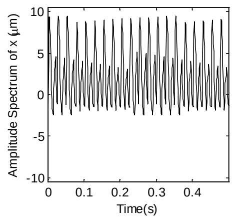
(a) *x* (b) *y* (c) *z*

(d) spectrum of *x* (e) spectrum of *y* (f) spectrum of *z*

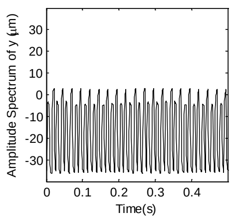


(g) orbit

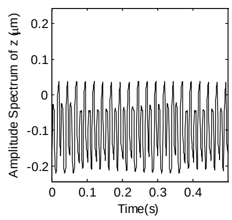
Fig. 9. Vibration responses of rotor with offset 2mm at rotating speed 40Hz



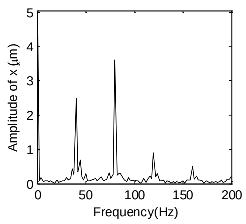
(a) *x*



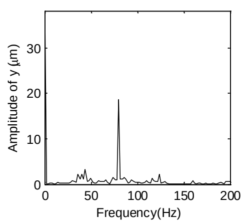
(b) *y*



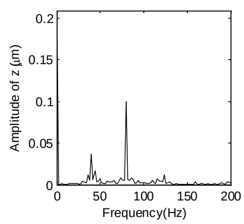
(c) *z*



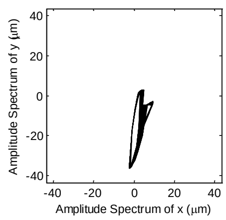
(d) spectrum of *x*



(e) spectrum of *y*



(f) spectrum of *z*



(g) orbit

Fig. 10. Vibration responses of rotor with offset 3mm at rotating speed 40Hz

It is can be clearly seen from the simulation results in Figs. 3-10 that misalignment has an evident influence on the vibration of the rotor system, but the effects are distinguish in the sub-critical and super critical speed ranges. With the same misaligned offset Δ, the system presents more complex behavior of 2×, 3× rotating frequencies besides the fundamental 1× one at a higher rotating speed. Accordingly, the orbit varies normal ellipse form to irregular.

The axial vibration of the rotor system can be aroused by misalignment, and the peak of axial and radial vibrations will increase along the misaligned offset Δ ranging from 0mm to3mm. The further comparisons of the vibration amplitudes of 1× frequency at rotating speeds of 20 Hz and 40 Hz are collected and listed in Table 3.

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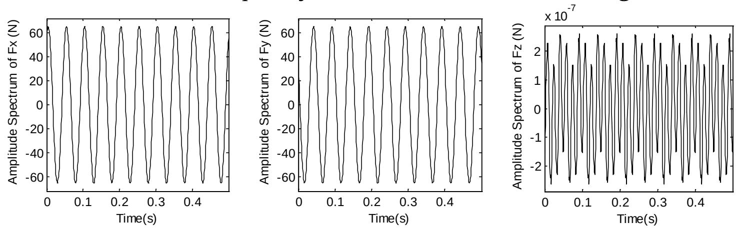
Table 3. Vibration amplitudes of 1× frequency with different misaligned offsets and rotating speeds

Offset\Speeds	20Hz			40Hz		
Δ(mm)	*x*(μm)	*y*(μm)	*z*(μm)	*x*(μm)	*y*(μm)	*z*(μm)
0	0.46	0.46	0.00	3.22	3.22	0.00
1	0.27	0.28	0.01	2.00	2.04	0.06
2	0.22	0.23	0.01	6.28	6.28	0.05
3	1.20	1.20	0.01	2.48	3.20	0.04

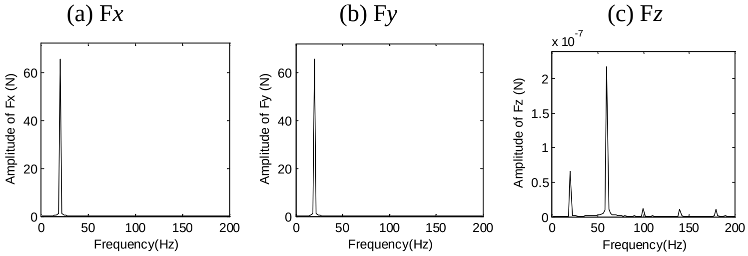
At rotating speed of 20 Hz, vibration amplitudes of rotor system in three directions have the increasing trend along the misaligned offset Δ. At rotating speed of 40 Hz, the amplitude of 1× frequency increases firstly and then decreases, but the peak response increases dramatically.

Reaction forces of misaligned bearing

The reaction forces of B2 (F*x*- vertical force, F*y*- horizontal force, F*z*- axial force) under the different operating conditions shown in Table 2 are simulated, and the obtained results in time and frequency domains are illustrated in Figs. 11- 18.

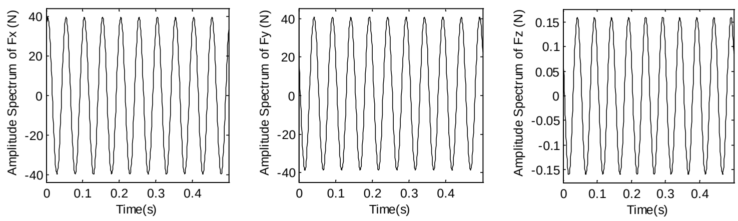


(a) F*x* (b) F*y* (c) F*z*

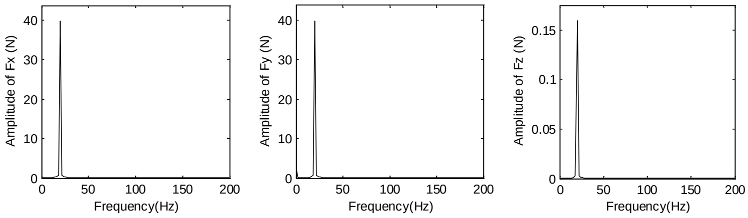


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig.11. Reaction forces of B2 under normal condition at rotating speed 20Hz

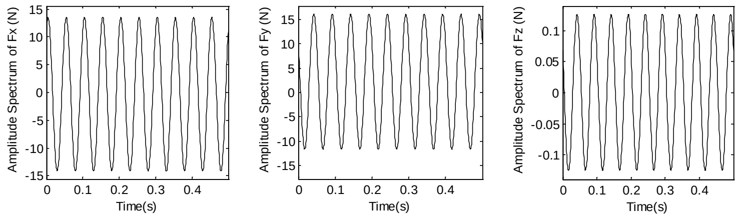


(a) F*x* (b) F*y* (c) F*z*

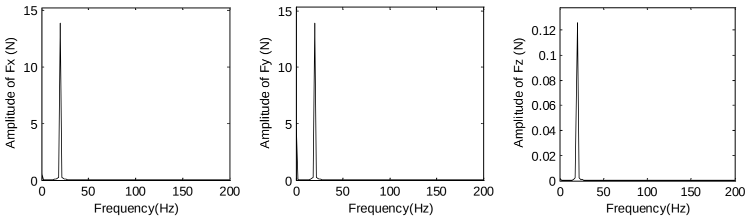


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 12. Reaction forces of B2 with offset 1mm at rotating speed 20Hz

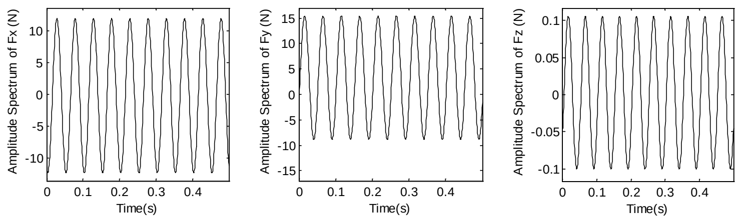


(a) F*x* (b) F*y* (c) F*z*

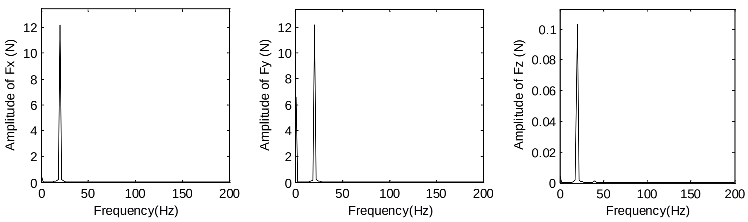


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 13. Reaction forces of B2 with offset 2mm at rotating speed 20Hz



(a) F*x* (b) F*y* (c) F*z*



(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 14. Reaction forces of B2 with offset 3mm at rotating speed 20Hz

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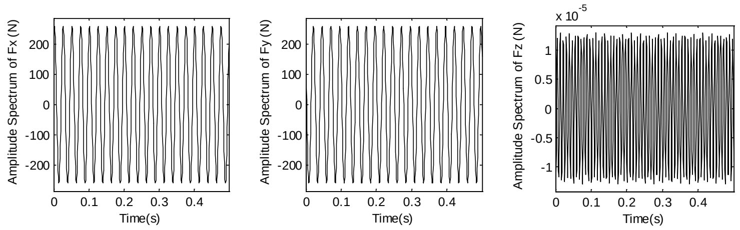
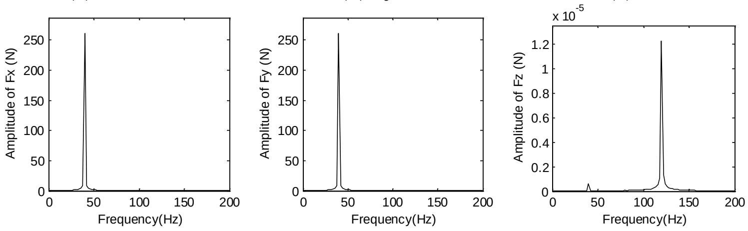
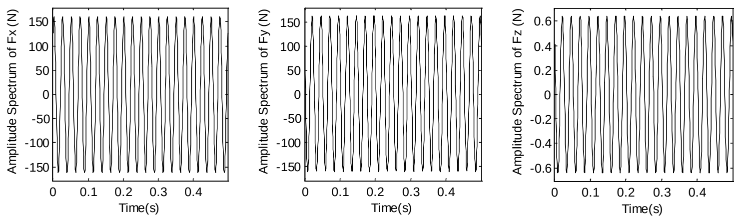


(a) F*x* (b) F*y* (c) F*z*

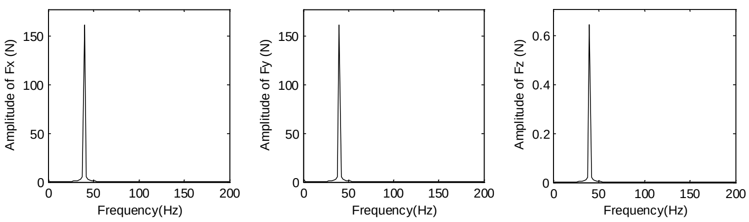


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig.15. Reaction forces of B2 under normal condition at rotating speed 40Hz

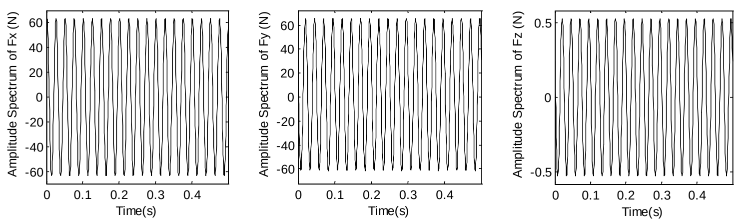


(a) F*x* (b) F*y* (c) F*z*

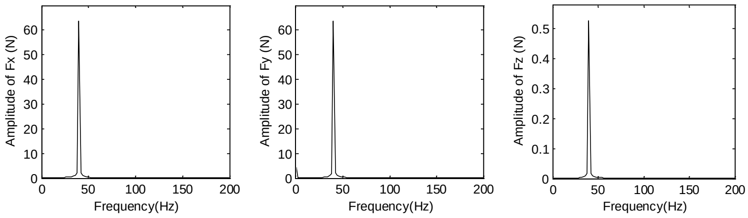


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 16. Reaction forces of B2 with offset 1mm at rotating speed 40Hz

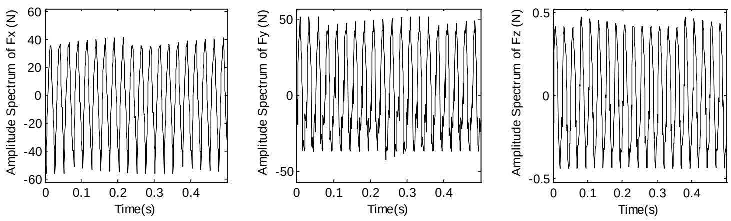


(a) F*x* (b) F*y* (c) F*z*

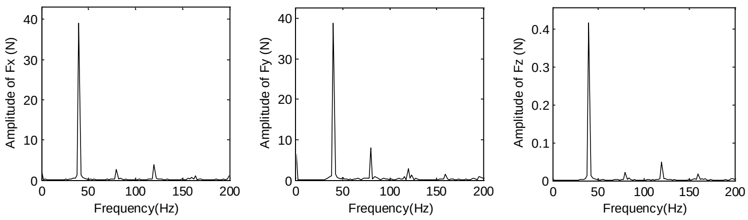


(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 17. Reaction forces of B2 with offset 2mm at rotating speed 40Hz



(a) F*x* (b) F*y* (c) F*z*



(d) spectrum of F*x* (e) spectrum of F*y* (f) spectrum of F*z*

Fig. 18. Reaction forces of B2 with offset 3mm at rotating speed 40Hz

It can be seen form the simulation results that the amplitude of reaction forces in radial and axial directions decrease along misaligned offset Δ and share the same treads. The further comparisons of the amplitudes of reaction forces at rotating speeds of 20 Hz and 40 Hz are collected and listed in Table 4.

Table 4. Amplitudes of reaction forces of 1× frequency with different misaligned offsets and rotating speeds

Offset\Speeds	20Hz			40Hz		
Δ(mm)	F*x*(N)	F*y*(N)	F*z*(N)	F*x*(N)	F*y*(N)	Fz (N)
0	65.60	65.56	6.60	260.81	260.7	6.23
1	39.73	39.71	0.16	161.3	161.2	0.64
2	13.86	13.89	0.13	63.54	63.51	0.53
3	12.18	12.15	0.10	39.06	38.83	0.42

3 Conclusions

In this paper, a new model of a test rig with misaligned bearing by use of MSC.ADMAS is proposed. And the vibrations of the rotor and the reaction forces of misaligned bearing under different misaligned offsets are investigated.

The results reveal that misalignment has an evident influence on the vibration of the rotor system, but the effects are distinguish in the sub-critical and super critical speed ranges. With the same misaligned offset, the system presents more complex behavior of 2×, 3× rotating frequencies besides the fundamental 1× one at higher rotating speed. It is also clearly found that axial vibrations of the rotor system can be aroused by misalignment, the peak of which will increase along the misaligned offset. While the amplitude of reaction forces of misaligned bearing in the radial and axial directions decrease along misaligned offset and share the same treads.

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