Modeling and characteristic analysis of variable reluctance signal variation of rolling bearing outer ring fault

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ABSTRACT The bearing fault diagnosis based on instantaneous angular speed (IAS) has been developed rapidly. However, it is difficult to identify the weak fault of bearings by using the variable reluctance sensor (VRS) as an encoder because of the lack of the high-frequency information in the modulation frequency of IAS. To overcome this shortcoming, a novel method based on the raw signal of the VRS is proposed for bearing diagnosis in this paper. The mechanism of bearing local fault signal response is analyzed by establishing the bearing outer ring fault model and the sensor signal response model, which are explored and verified by the fault simulation experiments. The results show that the gearwheel tooth-pass frequency is modulated by the outer ring fault characteristic frequency, and the defect size can be estimated by the double peaks in the high-frequency band. Moreover, compared with the acceleration signals, the defect size estimations based on the VRS are closer to the actual defect size, which verifies the effectiveness of the proposed method.

INDEX TERMS Bearing fault diagnosis; Dynamic modeling; Fault characteristics; Variable reluctance sensor

I. INTRODUCTION
Rolling bearing is the key component of rotating machinery, which has been widely used in the fields of energy, chemical industry, and aerospace. However, it is prone to failure because of strict working conditions, long-term operation, and other influencing factors [1-3]. Therefore, a variety of signals, such as vibration, current, temperature, strain, acoustic emission and encoder, have been employed to monitor the condition of bearings[4].

Among them, vibration is the most widely used, but it should be noted that the vibration signal is easily affected by the transmission path, leading to the poor signal-to-noise ratio. In addition, the modulation effect caused by speed fluctuation is coupled with the modulation effect of local fault, which makes the fault difficult to identify [5, 6]. Hence, much more complex signal processing algorithms are required, such as variational mode decomposition [7], spectrum kurtosis [8], sparse filtering [9], and so on. Compared with vibration, the transmission path of instantaneous angular speed (IAS) is simple and the signal-to-noise ratio is high. Therefore, it is of great significance to study the bearing fault diagnosis method based on IAS.

Recently, bearing fault diagnosis based on IAS has been developed rapidly. Renaudin et al. [10] proposed an alternative way of bearing condition monitoring based on the IAS measurement, and proved that local faults like pitting in bearing generate small angular speed fluctuations which are measurable with optical or magnetic encoders. Moustafa et al. [11] found that the IAS has the ability to solve the quantitative diagnosis of rolling bearing under low speed and heavy load through experiments. Bourdon et al. [12] analyzed the variation characteristics of IAS waveform caused from the grating encoder, and verified the characteristics of bearing fault under different working conditions through experiments. To reveal the variation law of IAS, Gomez et al. [13, 14] introduced periodic disturbance into bearing torsional degree of freedom modeling, and proposed a bearing IAS modeling method, which was verified by the fault data of wind power transmission system.
However, the IASs mentioned above rely highly on high-resolution encoders, but the signal from the variable reluctance sensor (VRS) has less angle resolution. If using the VRS as an encoder, the modulation frequency of VRS signal has no enough high-frequency information to identify bearing weak fault [15, 16]. Figure 1 shows the working principle of the VRS. When the gearwheel rotates, the relative position of the probe top to the tooth top changes alternately. The magnetic circuit in the iron core changes periodically, generating the induced voltage is generated in the coil. Hence, the induced voltage has a good monitoring ability for the motion state of the rotating shaft [17]. Addabbo et al. [18] established the voltage response model of VRS and applied the original voltage signal to the health monitoring of rotor system. Laczenko et al. [19] proposed the rotating blade observation method, and detected cracks of the aeroengine blade according to the induced voltage signal generated by the VRS.

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![Magnet, Coil, Iron core, Gearwheel](image)

**FIGURE 1.** The schematic diagram: (a) The variable reluctance sensor; (b) The variable reluctance signals

To sum up, the diagnosis method based on the original signal of the VRS has been effectively verified in energy capture [20], speed measurement [21], and condition monitoring [18], but has not been discussed in bearing diagnosis. In addition, the raw signal of the VRS can reflect the change of bearing condition. There will be an obvious sudden change in the IAS, as a local fault occurs in the bearing. And the change will be reflected in the voltage signal of the VRS. However, there is a lack of mechanism research to reveal the relationship between bearing faults and raw signal characteristics.

To overcome the shortcoming mentioned above, the VRS signal response model of bearing fault was established. The main contributions of this works are as following: (1) A response dynamics model of VRS considering the fault size is established. The bearing fault characteristics of VRS signals are given through model and fault simulation experiments, providing a theoretical basis for fault identification; (2) The error of fault size estimation based on the high frequency of the raw signal is analyzed. The feasibility of the VRS in bearing quantitative diagnosis is discussed.

The remainder of this paper is organized as follows: Section 2 analyzes the mechanism of the VRS, and the bearing fault dynamic model and magnetoelectric voltage response model are established. Section 3 discusses the VRS signal response of rolling bearings with outer ring fault and analyzes the time domain and frequency domain characteristics of the signal. Section 4 verifies the effectiveness of the model through fault simulation experiments, and Section 5 draws the conclusion.

II. MODELING OF VARIABLE RELUCTANCE SENSORS RESPONSE TO BEARING FAULT

In this section, the instantaneous angular speed model and magnetoelectric signal voltage response model are necessarily established for bearing local fault to study the response characteristics of variable reluctance sensor for bearing fault.

A. DYNAMIC MODELING OF BEARING

In the horizontal rotor system, the rolling element rotates alternately in the load area and un-load area, and contacts with the inner ring, outer ring and cage respectively. We define the contact deformation forces of the outer ring, inner ring and cage along the radial direction of the rolling element as $N_o, N_i,$ and $N_e,$ and the friction forces along the tangential direction as $F_o, F_i,$ and $F_e.$ Due to the deformation, the actual center of the inner ring $O_i$ will be lower than the theoretical center $O_i$, and there is an angle between the line $O_iO_e$ and the line $O_iO_o$, which is defined as the bite angle $\alpha$. Then, the stress analysis of rolling element in load area and un-load area are shown in Figure 2.

![Bearing stress analysis](image)

**FIGURE 2.** Bearing stress analysis: (a) the load area; (b) the un-load area

$$\alpha = \arcsin \left( \frac{\delta \sin \phi_o}{R_o + R_i - \delta_v} \right)$$  \hspace{1cm} (1)

where $\delta$ is total deformation, $\delta = \delta_o + \delta_v,$ and $\delta_v$ is the deformation of the rolling element at the angular position.
φ₀ in the absolute coordinate system. Rᵢ is the radius of the inner raceway, Rₖ is the radius of rolling element.

According to the balance of rolling body rotation torque and rolling body revolution force, the differential equation can be obtained as follow

\[ J \ddot{\phi}_r = (F \cos \alpha - N_s \sin \alpha - F_r)(R_k - \delta_r) - F_r \]

(2)

where \( J \) is the moment of inertia of rolling element, \( \theta \) is the rotation angle of the rolling element. Similarly, the moment balance equation of cage and bearing inner ring can be obtained:

\[ J \ddot{\phi}_c = -R_r \sum_{i=1}^{n} N_{ij} \delta_i m_i g \sin(\phi_i) \]

(4)

where \( J_c \) is the moment of inertia of cage, \( J_r \) is the moment of inertia of inner ring. \( K_r \) is the torsional stiffness of inner ring, \( R_r \) is the revolution radius of rolling element. \( n \) is the number of rollers. \( \phi_i \) and \( \phi_r \) are the angular positions of cage and inner ring in the absolute coordinate system respectively. \( N_{ij} \) is the force exerted by the cage on the \( j \)th rolling element, \( F_r \) and \( N_r \) are the friction and support force exerted by the \( j \)th rolling element on the inner ring.

Because the stress states of the rolling element in the load area and un-load area are completely different, it is necessary to clarify the range of the bearing area, which is the limit angle of load distribution \( \phi_{\max} \)

\[ \phi_{\max} = \pm \arccos\left(\frac{P_o}{2 \delta_r}\right) \]

(6)

where \( P_o \) is the bearing clearance, \( \delta_r \) is the radial deformation. The relationship between deformation and load can be described as

\[ Q = K \delta^2 \]

(7)

where \( K \) is contact stiffness [22], it can be expressed as

\[ K = \left[\frac{1}{(1/K_p)} + (1/K_p)^{1/2}\right]^{-2} \]

(8)

where \( K_p \) and \( K_p^\prime \) are the contact stiffness between the rolling element and the inner and outer ring respectively, which were defined as \( K_p = 2.15 \times 10^5 \sqrt[\rho]{\rho}(\delta)^n \) by Harris, where \( \rho \) is the curvature. This paper studies cylindrical roller bearing, \( n=1.11 \) [23]. The relationship between deformation and load is shown in Figure 3. We can obtain the relationship between load and deformation through force balance

\[ F_r = \int_{-\phi_{\max}}^{\phi_{\max}} \phi Q_{\max} (\delta_{\max} - \phi)^2 \cos \phi d\phi \]

(9)

where \( F_r \) is the load acting on the bearing in the rotor system, mainly the mass of the rotating shaft, and \( Q_{\max} = Q_{\max} (\delta_{\max} - \phi)^2 \).

According to the position of the rolling element, the forces of the inner and outer rings on the rolling element are discussed respectively. The force of the rolling element in the load area can be expressed as

\[ \begin{align*}
N_i &= Q_p \\
F_i &= \mu N_i \\
N_o &= N_i \cos \alpha + F_r + G \cos \phi_o \\
&- (F_i + F_r) \sin \alpha + m_r \phi_o \delta_r^2 \\
F_o &= \mu N_o
\end{align*} \]

(10)

The stress of the rolling element in the no-load area is related to the speed. When \( F_r + G \cos \phi_o + m_r \phi_o \delta_r^2 \geq 0 \), the rolling element contacts the outer ring, and the force is shown in (11). When \( F_r + G \cos \phi_o + m_r \phi_o \delta_r^2 < 0 \), the rolling element contacts the inner ring, the force is shown in (12).

\[ \begin{align*}
N_i &= 0 \\
F_i &= 0 \\
N_o &= N_i \cos \alpha + F_r + G \cos \phi_o - (F_i + F_r) \sin \alpha + m_r \phi_o \delta_r^2 \\
F_o &= \mu N_o
\end{align*} \]

(11)

\[ \begin{align*}
N_i &= 0 \\
F_i &= 0 \\
N_o &= -(F_i + F_r) \cos \alpha + m_r \phi_o \delta_r^2 \\
F_o &= \mu N_o
\end{align*} \]

(12)

In (10), (11), and (12), \( F_r \) and \( N_i \) are the friction force and support force between the cage and the rolling element respectively. They are expressed as
\[
\begin{align*}
\frac{N_c}{K_c R_m} (\phi_c - \phi_b) \\
F_c = \mu_c N_c
\end{align*}
\]
(13)
where \(K_c\) is the spring stiffness, and \(K_c\) is 10\(^8\) N/m [24]. \(\mu_1\), \(\mu_2\), and \(\mu_c\) are the friction coefficient [25]. It is worth noting that the contact position between the rolling elements and the cage depends on their relative position.

In this study, spalls with angle \(\theta_s\), depth \(d\), and width \(l\) are considered as the located outer ring defects, as shown in Figure 4, and \(d > \delta_{\phi}\).

**FIGURE 4.** The contact between the rolling element and local fault

When the rolling element rolls entry the pit defect, the contact force suddenly changes with the change of the contact deformation. To simplify the calculation, we keep the contact stiffness in (8) unchanged and obtain contact forces in different states according to the deformation of contact defects. Then the deformation \(\delta_{\phi}\) at any angle can be expressed as

\[
\delta_{\phi} = \delta_{\phi} - f(\phi_b) \quad \phi_c \in (\phi_b - \theta_d, \phi_b + \theta_d)
\]
(14)
where \(\phi_b\) is the absolute space angle of the \(j\)th roller, and \(f(\phi_c)\) can be expressed as

\[
f(\phi_c) = \begin{cases} 
\delta_{\phi} & d > \delta_{\phi} \\
\delta_{\phi} - \Delta d & \text{other}
\end{cases}
\]
(15)

According to (14) and (15), the rolling element collides with the edge of the defect, the contact deformation will suddenly change, resulting in the sudden change of torque, and the IAS of the rotating shaft is changed.

The dynamic equation of rolling bearing with outer ring fault can be expressed as

\[
J_\omega \ddot{\theta} = (F_c \cos \alpha - N_c \sin \alpha - F_\omega)(R_b - \delta) - F_\omega
\]

\[
+ (F_c \sin \alpha + N_c \cos \alpha)(R_b - \delta)
\]

\[
m_ \omega \dot{\phi}_b = F_c \cos \alpha - N_c \sin \alpha + F_\omega - m_ \omega g \sin \phi_b
\]

\[
J_\omega \dot{\phi}_b = -R_m \sum_{j=1}^{n} (F_c \cos \alpha - N_c \sin \alpha) - \delta_{\phi} m_ \omega g \sin(\delta_{\phi})
\]

\[
J_\omega \dot{\phi}_b = T_x - \delta_{\phi} m_ \omega g \sin(\delta_{\phi}) - \delta_{\phi} m_ \omega g \sin(\delta_{\phi})
\]

where \(\delta_{\phi}\) and \(\delta_{\omega}\) represent the centroid offset of cage and inner ring caused by the outer ring defect.

**B. MODELING OF THE VRS**

The model of variable reluctance sensor in reference [14] is used in this study, which can be expressed as

\[
V_o = \frac{PM}{\mu_c A_w (R_m + l_w(t)/(\mu_c A_w))^2} \frac{dl_w(t)}{dt}
\]

where \(PM\) and \(R_m\) are parameters related to permanent magnets, \(PM = (B_r / \mu_r) / l_m\) and \(R_m = l_w / (\mu_c A_w)\). \(l_w\) is magnetic circuit related to the permanent magnet, and \(A_w\) represents the cross-sectional area of permanent magnet. In (17), we can find that the output voltage is directly proportional to the change rate of the external magnetic circuit \(l_w(t)\).

The magnetic circuit always moves in the direction of minimum magnetoresistance. The external magnetic circuit changes periodically with the position of each tooth of the gearwheel relative to the probe, with the movement of the rotating shaft, as shown in Figure 5. Without considering the machining error of each gear, the external magnetic circuit can be expressed as

\[
l_w(t) = \frac{a_0}{2} + a_1 \cos(\omega_\tau t) + a_2 \cos(\omega_{gear} t)
\]

(18)

where \(\omega_\tau\) is the rotation frequency of inner ring, \(\omega_{gear}\) is the gearwheel pass-through frequency, \(\omega_{gear} = \omega_\tau \cdot N_{gear}\).

**FIGURE 5.** Schematic diagram of magnetic circuit changing: (a) right at the top of the tooth; (b) right at the bottom of the teeth

### III. SIMULATION ANALYSIS

[4] VOLUME XX, 2022

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A. SIMULATION PROCESS AND PARAMETERS

The cylindrical roller bearing commonly used in certain equipment is used to verify the validity of the model. The bearing model used in the modeling is the same as that installed in the test device, which is NF308. The voltage response solution steps are shown in Figure 6.

Step 1: Input bearing parameters and variable reluctance sensor parameters.

Step 2: Calculate the spatial positions of each roller $\theta$ and the speed of each element at time $t$, such as $\phi_0$, $\phi_e$, and $\phi_i$.

Step 3: Calculate the force between rollers and inner ring, outer ring and cage at time $t$, including support force, friction force and centrifugal force.

Step 4: Calculate the instantaneous angular acceleration of each element at time $t$, such as $\dot{\theta}$, $\dot{\phi_0}$, $\dot{\phi_e}$, and $\dot{\phi_i}$.

Step 5: Calculate the voltage $V_t$ of VRS according to the current inner ring speed $\dot{\phi_i}$.

Step 6: Cycle steps 2 to 5 until $t \geq T$.

Step 7: Output the signal of the VRS.

![FIGURE 6. Voltage solution flow of variable reluctance sensor](image)

The parameters of NF308 bearing are shown in Table 1. The variable reluctance sensor is Delong DZ, and its permanent magnet material is Y30BH, with a length of 16.4 mm, a width of 6.95 mm, and a height of 2.5 mm. So, the air gap $l_a = 2.5$ mm, the cross-sectional area of the permanent magnet $A_m = 17.375$ mm$^2$, $\mu_n \equiv 1$ and $B_g = 0.38 \sim 0.42$ T. Assume that the number of teeth of the gearwheel installed on the transmission shaft is 25. There is a rectangular fault with a length of 10 mm, a depth of 0.1 mm, and a width of 1 mm directly below the outer ring.

![TABLE 1 NF308 BEARING PARAMETERS](image)

| Parameters                      | Value  |
|---------------------------------|--------|
| The number of the rollers $n$   | 12     |
| The diameter of the roller (mm) | 12.0   |
| Pitch diameter (mm)             | 65.5   |
| Clearance (μm)                  | 40     |
| Density (kg·m$^{-3}$)           | 7810   |

The fourth-order Runge Kutta method is used to solve the magnetoelectric signal response. The inner ring speed is 330 rpm, the simulation step is 0.0001s, and the simulation time is 2 s. Set the vertically downward direction as $0^\circ$, the common angle speed and rotation speed of the rolling element in the initial state are 0, and the angular speed of cage revolution is 0.

B. RESULT ANALYSIS

Figure 7 shows the voltage signal of the VRS of the normal bearing. The variable reluctance signal of normal bearing has obvious harmonic characteristics. The energy in the spectrum is mainly the gearwheel pass-through frequency, $f_g = f_s \cdot N_{gear}$, and its harmonics, and are modulated by rotation frequency.

In Figure 8, a similar phenomenon is also observed in the VRS signal of the outer ring fault. It is worth noting that the gearwheel tooth-pass frequency is also modulated by the outer ring fault characteristic frequency in the spectrum. In addition, there is a frequency band separated by the outer ring fault characteristic frequency at 400 - 900 Hz, as shown in Figure 9. The double peak phenomenon is observed in the time domain of the filtered signal, and the interval between the double peak is 0.002 s. In the modeling, we have assumed that there will be significant changes in deformation of rolling bearings when they entry and exit point pit. Therefore, the interval can be expressed as:

$$t = \frac{2l_f D_p}{\pi(D_p^2 - d^2)f_s}$$  \hspace{1cm} (18)

where $l_f$ is the fault size, $d$ is the diameter of rolling element, $D_p$ is the diameter of the pitch diameter, and $f_s$ is the rotation frequency. Then, the time of the bearing passing through the pit is 0.002 s, when the bearing is at a shaft speed of 5.5 Hz. Therefore, we can evaluate the bearing fault size by the time interval of double peak.
A. TEST INTRODUCTION

The effectiveness of the simulation results is verified through the fault simulation experiment. The fault test bed is shown in Figure 10. The motor is connected with the sliding bearing seat through the coupling. The experimental bearing seat is installed on the sliding bearing seat through the cone sleeve structure. At the same time, a radial loading device and an axial loading device are set to simulate the bearing radial force and the axial positioning of the bearing respectively. The gearwheel is fixed at the shoulder of the shaft through three locking screws. The gearwheel material is DT4 and the number of teeth is 47, as shown in Figure 11 (a). The variable reluctance sensor adopts the Delong DZ series and is fixed on the test bench through the magnetic base. The probe is facing the center of the gearwheel, and the gap between the probe and the tooth top is 1 mm through the feeler gauge. The accelerometer is mounted on the bearing housing and synchronously measured with the VRS signal. The acquisition card is DT9837B and the sampling frequency is set to 10 kHz. A rectangular fault with a length of 10 mm, a depth of 0.5 mm, and a width of 1 mm was processed on the bearing outer ring by laser, as shown in Figure 11 (c). The test speed is set to 330 rpm.

FIGURE 7. Variable reluctance signal of normal bearing: (a) the time waveform; (b) the spectrum

FIGURE 8. Variable reluctance signal of bearing with outer fault: (a) the time waveform; (b) the spectrum

FIGURE 9. Variable reluctance signal of outer ring fault: (a) the local spectrum; (b) the filtered time waveform

IV. TEST VERIFICATION
TABLE 2
FREQUENCY ORDER OF BEARING FAULT CHARACTERISTICS AT 60 RPM

| Type                               | Value |
|------------------------------------|-------|
| Outer ring fault characteristic order \((f_o/Hz)\) | 4.70  |
| Inner ring fault characteristic order \((f_i/Hz)\) | 7.28  |
| Rolling element fault characteristic order \((f_b/Hz)\) | 4.78  |
| Cage fault characteristic order \((f_c/Hz)\) | 0.40  |

B. DATA ANALYSIS

Figure 12 and Figure 13 respectively show the variable reluctance signals of normal bearing and outer ring fault under the un-load condition with a rotating speed of 330 rpm. In Figure 12, the main component of the VRS signal of the normal bearing is the gearwheel tooth-pass frequency and the sideband with the rotation frequency as the interval. Compared with the VRS signal of the normal bearing, we can find that the gearwheel tooth-pass frequency is modulated by not only rotation frequency, but also the characteristic frequency of outer ring fault in the variable reluctance signal of outer ring fault, as shown in Figure 13 (b). The experimental results are in good agreement with the simulation.

The Hilbert Demodulation was used to demodulate the variable reluctance signal to instantaneous frequency (IF), the results are shown in Figure 14. The obvious periodic impulses are observed in the IF time waveform. And there are the outer ring fault characteristic frequency and its harmonics in the IF spectrum. It means that we can identify the bearing fault directly through the sideband frequency of the gearwheel tooth-pass frequency, avoiding the problem of traditional resonance band positioning.

FIGURE 12. Normal bearing: (a) the time waveform; (b) the spectrum

FIGURE 13. The bearing with outer ring fault: (a) the time waveform; (b) the spectrum

FIGURE 14. The bearing with outer ring fault: (a) the time waveform; (b) the spectrum

Similarly, for the double peak phenomenon in the simulated filtered signal, the fast spectrum method was applied to the analysis of variable reluctance signal, as shown in Figure 15 (a). The optimal band corresponds to a band pass filter with \(f_c=3437.5 \text{ Hz}\) and 625 Hz bandwidth \((k = 3)\). Obvious periodic impulses are observed in the filtered signal, and the time interval of impulses is the time corresponding to the outer ring fault characteristic frequency. There are two small peaks in each large impulse, which is similar to the double peak of vibration signal [26]. However, it is worth noting that the double peak of vibration acceleration is caused by the sudden change of stress when the rolling element entries and exits the local defect, while that of the VRS signals is caused by the change of IAS.
The fault size is estimated for the time interval of double peak in Figure 15(c) by the method proposed in references [27, 28]. The square envelope is used to process the filtered signal, which is shown in Figure 16. In the Figure 16, $\beta_1$ is the angular extent between the two maxima points of the first and second rising events, which are the gentle decompression and recompression events when the rolling elements pass through the pit. Using (7), the relative contact deformation between a rolling element and both raceways at the entry and exit points of the defect $\delta_{\text{max}}$ is 3 $\mu$m, and the angular $\beta_1$ is 0.0049. Therefore, the total length of a defect can be estimated by

$$L = \sin(2\beta_1 + \beta_2) \times R = 1.1665 \text{ mm} \quad (19)$$

According the above method, mean and standard deviations for a number of events on the response of the defective bearing for various speeds was estimated as shown in Figure 17. It can be found that the defect size estimations based on the VRS are closer than that based on acceleration to the actual defect size.

V. CONCLUSION

The method based on the raw signal of the VRS is proposed for the bearing fault identification in this paper. The IAS model of bearing outer ring fault and the VRS voltage response model were established to reveal the relationship between the fault and the VRS signals, and the validity of the results was verified by simulation and experimental test. In the study, the gearwheel tooth-pass frequency is modulated by the outer ring fault characteristic frequency in the VRS signals, as the defect occurs in the bearing outer ring. And, the bearing outer ring fault can be identified by the sideband of the gearwheel tooth-pass frequency. In addition, the double peaks are observed in the filtered signal in a particular frequency band, which are used to the defect size estimations. The results show that the defect size estimations based on the VRS are closer than that based on acceleration to the actual defect size.

The future research will focus on the quantitative diagnosis of natural spalls and the gear wear fault detection based on the VRS signal.

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