A Diagnosis method of the small end fault on reciprocating compressor connecting rod

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Abstract. The connecting rod is the key moving part of a reciprocating compressor, of which the stress state is extremely complicate and the wear fault of the small end is always a bottleneck problem in the field of fault monitoring and diagnosing. This paper is aimed to present a new method to diagnose the above wear fault. Firstly, a contact model of a clearance in the revolute joint of the small end of a connecting rod bearing (SECRB) was established and a multi-body simulation tool was utilized to simulate the slider-crank mechanism with a clearance, from which the dynamic influence of wear gap in SECRB of a slider-crank mechanism was obtained. Based on the study above, we extracted the characteristics of the wear fault of SECRB and then proposed a brand new approach to monitoring and diagnosing this wear fault by analyzing the angle domain of vibration signals. The availability was verified by conducting an experiment on a reciprocating compressor. And the experimental results show that this method can not only accurately diagnose the wear fault of SECRB but also approximately estimate its severity. This study laid a foundation for the online monitoring and early warning of this fault.

1. Introduction
Reciprocating compressor is one of indispensable key equipments in many sectors of the national economy, such as chemical process, gas transmission, power engineering, and other important occasions. However, due to the attributes such as complex structure, a lot of vibration sources, and many vulnerable parts of reciprocating compressors, serious accidents may happen frequently, which always cause irreversible consequences. The connecting rod as one of the key moving parts has complex motion states and poor working conditions, as a consequence of which, the wear faults often occur in its small end. In the early stage of the failure, dynamic behaviors of the unit will not be impacted seriously, but when the wear deteriorates to a certain degree, it may result in many malignant faults, such as cylinder knocking, scuffing, hitting, and even piston rod fracture. Therefore, researching the methods that can monitor and diagnose the wear faults of SECRB in the early stage on online has important theoretical significance and practical engineering application values.
Nowadays, the studies, surrounding the diagnosis methods of wear faults of mechanical systems, are mainly divided into two aspects[1]: oil analysis and vibration signal analysis, (both of which have some inherent drawbacks). The former can be valid in the detection of wear faults of the unit, but it can’t locate where the wear occurs, making it difficult to guide the maintenance effectively, and apply it in the on-line monitoring system currently. In recent years, the latter has made great processes with the development of the dynamic theory of mechanical systems with clearance joints, on which many researchers have done a lot of theoretical and experimental investigations. Plores et al. [2-4] studied and compared the models of dry friction and lubrication and the models of rigid body and flexible body, and did a parametric study on the dynamic response of planar multi-body systems with multiple clearance joints. Schwab et al[5] compared the revolute joint clearance models in the dynamic analysis of rigid and elastic mechanical systems, and presented a procedure to estimate the maximum contact force during collision in the impact model. Liu[6] presented a more direct and effective formula to describe the properties of the contact in the cylindrical joint with clearances based on the FEM analysis. Wenqing Lu [7] and Troy Feese et al [8] researched the fault prognosis methods for reciprocating machinery. However, most of those research works were aimed at the dynamic impact of different clearance models and the optimization of these models. The reports that apply the slider-crank mechanism with clearance joint in SECRB to actual reciprocating compressor, on whose piston exerted constantly changing gas force in working condition, and put forward an effective wear fault diagnosis method of SECRB, which can be applied to the engineering practice, have not been seen.

In this paper, a theoretical model of clearance joint in the SECRB was established based on the modern contact theory, and a practical model in a real reciprocating compressor was built in a multi-body dynamic simulation tool, in which many simulations were performed with the changing gas forces exerted on the piston. Based on the simulation results, from which the characteristics of the wear fault of SECRB were extracted, a new approach to monitor and diagnose the wear fault of SECRB through analyzing the angle domain of vibration signals has been put forward. In order to verify the validity and practicability of this method, an experiment was conducted on a real reciprocating compressor, from which the simulation entity model sizes were obtained. The experimental results reveal that this method is simple and effective to diagnose such wear faults, predicting strong practical value for engineering applications.

2. Contact calculations

Due to the needs of the assembly and the relative rotation, there is a certain gap in the cylindrical joint, which is so small that it hardly makes any influence on the dynamic response on mechanical system with good lubrication conditions[9]. However, this gap will enlarge when wear become seriously caused by various reasons such as bad lubrication conditions. Meanwhile, the dynamic response on the mechanical system can be significantly enhanced[9]. Therefore, in order to study the increase impact, a cylindrical clearance joint contact model should be established.

2.1. Model of cylindrical clearance joint

Figure 1 shows the cylindrical clearance joint in real mechanism of the reciprocating compressor reciprocating compressor. The radial clearance can be defined as the difference between the radius of
bearing and journal and given by:

\[ e = R_b - R_j \]  \hspace{1cm} (1)

where \( R_b \) is the radius of bearing and \( R_j \) is the radius of journal.

**Figure 1.** Cylindrical clearance joint in real mechanism of the reciprocating compressor

The existence of the gap makes two more degrees of freedom, that is, two less restrictions. The center of the journal and the center of the bearing are not coincident, except for some special cases. Instead, bearing is limited in a certain area, namely, within the boundary of the bearing. The eccentricity ratio \( \varepsilon \) can represent the deviation degree of bearing center, which is a dimensionless physical quantity. And it can be given by:

\[ \varepsilon = \frac{e}{c} \]  \hspace{1cm} (2)

where \( e \) represent the eccentricity and \( c \) represent the radius clearance.

2.2. Contact force model

Many kinds of contact models have been put forward in previous literatures. The most widely applied one was Hertz contact model, which, however, did not take energy dissipation into consideration in the process of collision, and was only applicable to completely elastic impact. Lankarani and Nikravesh[10] established a new contact model, which has been widely used in dynamic simulation of mechanical system with clearance joint by means of dividing the normal contact force into elastic part, represented by the first term in Eq.(3), and dissipative part, represented by the second term in Eq.(3). This model can be described as

\[ F_n = K \delta^m + D \dot{\delta} \]  \hspace{1cm} (3)

where \( K \) represents the contact stiffness, \( D \) is the hysteresis damping coefficient, \( \delta \) is the depth of relative penetration, and \( \dot{\delta} \) represents the relative impact velocity. The exponent \( m \) is usually chosen for 1.5 for metallic contact[10]. The contact stiffness can be given by:
where $\sigma_b$ and $\sigma_j$ are the material properties and can be given by:

$$\sigma_z = \frac{1 - v_z^2}{E_z}, (z = i, j)$$  \hspace{1cm} (5)$$

In Eq.(5) $v_z$ is Poisson’s coefficient and $E_z$ is the Young’s modulus.

Hysteresis damping coefficient can be expressed as:

$$D = \frac{3K(1-c_z^2)}{4} \delta^{\gamma}$$  \hspace{1cm} (6)$$

where $c_z$ represents the coefficient of restitution and $\delta^{\gamma}$ is the initial impact velocity. However, it is hard to determine these two parameters, so a modified approach[11] was used in this paper, given by:

$$D = d \left| \frac{\delta}{\dot{\delta}} \right| \left| \frac{\dot{\delta}}{\delta} \right| m_2 \delta^{m_3}$$  \hspace{1cm} (7)$$

where damping coefficient $d$ can be chosen by the user.

2.3. Friction force model

Friction force acts when there is relative motion with two objects, along the tangent direction. Coulomb’s friction law is the most simply and wildly used method to calculate the friction force in dry friction model. However, it is not continuous when the relative velocity equals to zero. In order to overcome this problem, a kind of representatively modified method[9] will be used to calculate the friction force in this paper, which is shown in Fig.2 and given by:

$$F_f = \begin{cases} -\text{sign}(v) \mu(v) |F_n| & \text{if } v < v_s \\ \text{sign}(F_f) \times \min(|F_n|, f_{\text{max}}) & \text{if } v \geq v_s \end{cases}$$  \hspace{1cm} (8)$$

In Eq.(8), $F_n$ represents normal contact forces, $f_{\text{max}}$ is the maximum friction forces, and $\mu(v)$ is friction coefficient, which is a function of the relative speed in tangent direction, given by Eq.(9).

$$\mu(v) = \begin{cases} \text{havsin}(v, 0, 0, v_s, \mu_s) & \text{if } v < v_s \\ \text{havsin}(v, v_s, \mu_s, v_d, \mu_d) & \text{if } v \geq v_s \end{cases}$$  \hspace{1cm} (9)$$

In this equation, ($\mu_0$, $\mu_1$) are Static and dynamic friction coefficient and ($v_0$, $v_1$) are threshold velocities.
3. Modeling and simulation results
A method for dynamic modeling and simulation was proposed and effectively verified by Gummer A and Sauer B in reference[11]. In this paper, the same method is adopted to establish model and do dynamic simulation for reciprocating compressor. The distinct differences are that a practical model of slider-crank mechanism with a clearance joint in a real reciprocating compressor is built and the simulations are performed with the changing gas forces, which can make a great influence on the dynamic behavior, exerted on the piston.

![Entity model of slider-crank mechanism with clearance of a real reciprocating compressor](image)

**Figure 2.** Entity model of slider-crank mechanism with clearance of a real reciprocating compressor

| Operating parameters       | Value         | Operating parameters       | Value         |
|----------------------------|---------------|----------------------------|---------------|
| Working speed              | 495 r/min     | Crank radius               | 0.09 m       |
| Connect rod length         | 0.45 m        | Cylinder radius            | 0.125 m      |
| The suction pressure       | 100 kPa       | The discharge pressure     | 300 kPa      |

The entity model set up in a multi-body dynamic simulation tool is shown in figure 2. The geometric dimensions of each component of the crank connecting rod mechanism and some physical parameters of a real reciprocating compressor are listed in table 1. In the simulation, contact forces are calculated according to Eq.(3), damping forces are calculated according to Eq.(7), and friction forces are calculated according to Eq.(8). All parameters required to set in the simulation are listed in table 2.

![In-cylinder gas pressure curve](image)

**Figure 3.** In-cylinder gas pressure curve

The changing gas forces acting on the piston of the reciprocating compressor can be got generally by two ways, that is, simulation and experimental test. This text used the latter to get gas forces. The relationship between gas pressure and crank angle is shown in figure 3.

In order to study the influence of wear clearance in SECRB of the reciprocating compressor on the mechanism dynamic behaviors, a lot of simulations of mechanism with different gap sizes joints or
ideal joints are performed. Crosshead is attached directly to the small end of the connecting rod through the crosshead pin, for which the crosshead dynamic response that affected by the gap will be more significant. Therefore, the dynamic impact on the crosshead needs to be focused on.

Table 2. Parameters used in the dynamic simulation

| Contact characteristics       | Value            | Contact characteristics       | Value            |
|-------------------------------|------------------|-------------------------------|------------------|
| Contact stiffness $k$         | 5603076/ N/mm    | Damping exponent $m2$         | 0.0297           |
| Stiffness exponent $m1$       | 1.5              | Indentation exponent $m3$     | 1.5              |
| Damping coefficient $d$       | 2003200          | Coefficient of restitution $e$| 0.9              |
| Static and dynamic friction coefficient ($\mu_0, \mu_1$) | 0.01            | Threshold velocities ($v_0, v1$) | 1.0 mm/s        |

The simulate results of displacement and velocity of the crosshead are shown in figure 4, from which we can discover that the displacement and velocity in clearance joint conditions are same with ideal joint condition, respectively. Thus, it is explainable that the dynamic impacts on displacement and velocity by clearance are very little.

Figure 4. Simulation results of crosshead: (a) displacement; (b) velocity

However, as shown in figure 5, high acceleration amplitudes occur at the reversal points of crosshead, indicating high impact forces, when compared to ideal joint simulation results. What’s more, the acceleration amplitudes become higher as the clearance size increasing, indicating higher impacts forces.

When wear clearance increases to a certain degree, forces and moment cannot be transmitted well, making the two connecting objects directly impact after crossing the gap. As a result, the impact forces will appear abnormal fluctuations and larger amplitudes. As shown in the simulation results, on one hand, it is on the reversal points of crosshead that forces transmit appear unstable and collisions happen between the two components, no matter how large the gap size is. On the other hand, the amplitudes of crosshead acceleration increase with the development of wear process.
4. Method presentations

It is known that mechanical parts are generally accompanied with high frequency vibrations when high frequency impacts are acting on. Thus, in wear conditions, crosshead will inevitably appear short-term high frequency vibrations, which can be transferred to the surface of the body though the directly connected slideways. Therefore, abnormal vibration signals can be obtained if a vibrating sensor has been installed on the surface right above of the crosshead.

Based on the above analysis, a new method for monitoring and diagnosing the wear fault of SECRB through analyzing the angle domain of vibration signals was put forward. First of all, the practical model of slider-crank mechanism with a clearance joint of a real reciprocating compressor should be established. Secondly, the in-cylinder gas forces should be obtained through experiments or simulations. Thirdly, dynamic simulations should be done in a multi-body dynamic simulation tool, the results of which should be analyzed to get the phase position where abnormal high acceleration

![Figure 5. Simulated crosshead acceleration for different diametric clearances](image)
amplitudes occur. Moreover, acceleration vibrating sensors should be installed on the surface right above the crossheads of the reciprocating compressor to monitor angle domain vibration signals. Finally, high frequency vibration signals with large amplitudes will appear at the same phase position got from dynamic simulations when wear of SECRB occurs.

5. Experimental verification
A reciprocating compressor on which an on-line monitoring system has installed is shown in figure 6. A lot of sensors were set up to capture various signals, and two kinds of them will be used to perform this verifiable experiment. One is the acceleration sensors installed on the right above surface of the crossheads and used to monitor their vibrations. The other one is the key sensor installed on the side of flywheel and used to determine the positions of the piston in the cylinder. Besides, the procedure we used to simulate the clearance joint in this text is as follows. Firstly, a crosshead pin 0.5 mm less than the standard one in diameter was employed to replace the original one. Then the reciprocating compressor was turned on to collect the experimental data for a period of time. Finally, the above steps were repeated again using a crosshead pin 0.7 mm less than the standard one in diameter.

Figure 6. Reciprocating compressor experimental platform

Vibration waveforms in normal operation and fault conditions are shown in figure 7, from which we can find that vibration waveforms appear obviously additional impacts at 0 and 180 degree in the fault conditions when compared to normal condition, and that the amplitudes of the vibration signals become larger with the gap size increasing. It is obviously that the fault characteristics gotten from experiment agree well with that extracted from simulation results, proving that the method put forward in this paper and based on vibration signal angle domain analysis is effective.
6. Conclusion

A contact model of a clearance in the revolute joint of SECRB was established and a multi-body simulation method proved available was utilized to simulate the slider-crank mechanism with a clearance. A new method to diagnosis the wear fault of SECRB of reciprocating compressors effectively is presented, which can also be suitable in on-line monitoring system. This method can not only accurately diagnose the wear fault of SECRB but also approximately estimate its severity. Since calculation of wear is really difficult and not the purpose of this paper, the method about how to calculate the wear size from acceleration is not done. Experiments were conducted to demonstrate the availability of the approach. The method is universal and can be adapted to almost all reciprocating compressors.

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