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Dynamic Response and Diagnosis of Wear Fault in Plunger Pump Crosshead Guide Plate Based on Multibody Dynamics and Hydraulic Co-simulation

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Abstract: In the high-pressure process systems of the petrochemical industry, piston pumps are continuously used under extremely harsh conditions such as ultra-high pressure, large flow rate, long-term continuous heavy load, and media containing corrosive particles. The key friction pairs at the power end are prone to wear failure. This paper takes a new type of five-cylinder piston pump as the research object. Based on the multi-body dynamics theory and the L-N nonlinear spring-damping contact force model, a combined simulation of the hydraulic system and multi-body dynamics is established to study the vibration response of the crosshead guide plate wear fault under hydraulic conditions. The results show that when the liquid pressure exists, the maximum collision force between the crosshead and the guide plate increases with the continuous increase of the gap. When the gap is greater than 0.6mm, the collision force increases significantly; the vibration energy is concentrated near the second-order natural frequency of the housing (about 325 Hz), and the spectral peak increases with the increase of the gap. The study reveals the characteristic parameter variation rules of the crosshead guide plate wear fault, providing a theoretical basis for online monitoring and fault diagnosis of piston pumps in the petrochemical industry.

Keywords: Plunger pump; crosshead guide plate; wear failure; multibody dynamics; liquid load.

1. Introduction

The petrochemical industry is an important part of the national energy and materials strategy, covering multiple fields such as oil refining, ethylene production, fertilizers, and fine chemicals. In the process of petrochemical production, the plunger pump, as the core power equipment for high-pressure fluid transportation, is widely used in key processes such as hydrogenation feedstock, high-pressure reactor feedstock, and pipeline transportation. Facing extremely harsh working conditions such as ultra-high pressure, large flow rate, strong medium corrosion, and long continuous operation cycles, the plunger pump is almost always operating at full capacity. The key components of the power end, such as the crankshaft, connecting rod, crosshead, and plunger, are prone to wear, loosening, and even fracture. If not detected in time, it will not only seriously affect production efficiency and increase maintenance costs, but also may cause medium leakage, equipment damage, and even major safety accidents such as fire and explosion, becoming one of the key bottlenecks restricting the long-term safe operation of petrochemical plants. Among them, the crosshead and guide plate constitute the most important sliding friction pair of the plunger pump power end. Under long-term heavy-load reciprocating motion, guide plate wear is almost inevitable. The gap caused by wear changes the force state of the crosshead, causes the center of the plunger to deviate, and further accelerates the wear of the crankshaft and connecting rod bearings as well as the partial wear failure of the cylinder liner and sealing components. Therefore, in-depth study of the dynamic response laws of crosshead guide plate wear faults is of great theoretical significance and engineering value for the fault diagnosis and predictive maintenance of petrochemical plunger pumps.

In the field of reciprocating mechanical dynamics and fault diagnosis, foreign scholars have conducted systematic research earlier. In the aspect of contact-collision theory, Hunt and Crossley (1975) interpreted the recovery coefficient as the damping in vibration impact and proposed a classic nonlinear damping model [1]. Khulief and Shabana (1987) proposed a continuous contact force model for impact analysis of flexible multi-body systems, combining Hertz contact theory with Coulomb friction model [2]. Lankarani and Nikravesh (1990) further proposed an improved nonlinear contact force model considering hysteretic damping, which can accurately describe the energy dissipation during the collision process [3]. Vu-Quoc et al. (1999) based on the elastic-plastic theory proposed a positive pressure-displacement model applicable to spherical contact, considering the influence of plastic deformation [4]. Kraus et al. (1997) proposed a tangential spring damping friction model for friction contact modeling in dynamic simulation [5]. In the modeling of the dynamics of mechanisms with clearance, Huang et al. (1986) proposed a system dynamics equation based on the Lagrange multiplier method, considering Coulomb friction, static friction, impact, and the influence of constraint addition and deletion on the dynamic behavior of the mechanical system [6]. Chang and Huston (2001) based on the Newton-Euler equations and nonlinear dynamics theory, built a collision simulation framework for multi-body systems, and proposed a method for efficiently calculating the magnitude and direction of contact forces [7]. In the specialized research on reciprocating plunger pumps, Wu et al. (2014) conducted harmonic response analysis and experimental tests on a three-cylinder fracturing pump, and studied the vibration characteristics under different working conditions [8].

Although scholars have made significant progress in the aforementioned fields, the existing research mainly focuses on static structure analysis or single system dynamics simulation. There is a lack of systematic revelation regarding the dynamic response patterns, vibration characteristic parameters changes, and frequency band distribution characteristics of the crosshead guide plate wear fault under liquid load conditions. Moreover, the related research on plunger pumps in the petrochemical industry is even more scarce. Therefore, this paper takes a certain new type of five-cylinder plunger pump as the research object. Based on the Lankarani Nikravesh nonlinear spring-damping contact force model and multi-body dynamics theory, a hydraulic system joint simulation model is established. The dynamic simulation research on the deterioration process of the wear gap of the crosshead guide plate from 0.2 mm to 0.7 mm is carried out. The variation laws of the displacement of the crosshead axis center, the contact collision force and the vibration response of the housing under the liquid load are analyzed. The fault characteristic parameters are extracted, aiming to provide a theoretical basis for the online monitoring and diagnosis of the crosshead guide plate wear fault of the plunger pump.

2. Modeling and Simulation Methods

2.1. Research Subjects and Structural Overview

This paper takes the new five-cylinder plunger pump as the research object, which is mainly composed of the power end and the hydraulic end. The power end includes key moving components such as the crankshaft, connecting rod, crosshead, and crosshead guide plate; the hydraulic end includes the plunger, pump body, suction valve and discharge valve. Under long-term reciprocating and heavy-load conditions, the crosshead and guide plate are prone to generating wear gaps, which can cause increased impact, vibration, and even failure such as jamming or fracture. To accurately describe the failure mechanism, this paper adopts the multi-body dynamics + hydraulic joint simulation method for research.

2.2. Crosshead Guide Plate Wear Failure Mechanical Model

2.2.1. Kinematic Description of Moving Pair with Clearance

The crosshead and guide plate form a moving pair, and after wear, a radial clearance e is formed. The movement of the crosshead changes from ideal linear motion to free motion with oscillation and offset, as shown in the moving pair between the crosshead and the guide plate in Figure 1.

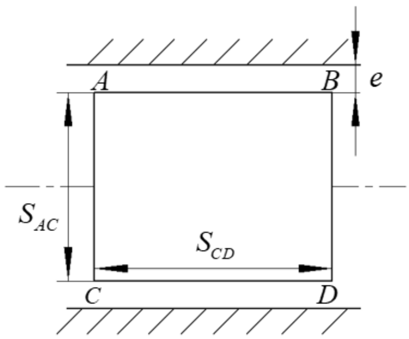


Figure 1. shows a moving pair with a gap between the crosshead guide plates

In the transmission system of the five-cylinder plunger pump, there are five crank-slider mechanisms. Although the initial phase between the crank of two adjacent hydraulic cylinders differs by 144°, their motion laws are the same. Therefore, this article only conducts research and analysis on a single crank-slider mechanism. Additionally, to facilitate the research and analysis, the single crank-slider mechanism is simplified to a crank-slider mechanism within a plane, and the following assumptions are made for the worn crank-slider mechanism:

1) Assume that the interaction forces between all moving components are concentrated forces.

2) Do not consider the existence of clearance in the motion pairs except for the crosshead guide plate, and all motion pairs are well lubricated.

3) Assume that the center position of the crosshead pin is the centroid position of the crosshead.

The schematic diagram and mechanical model of the mechanism for the crosshead guide plate wear fault are shown in Figure 2.

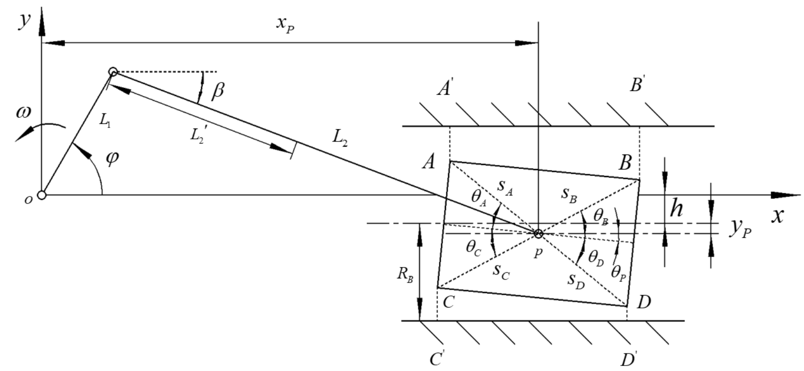


Figure 2. Schematic diagram of the mechanism for the crosshead guide plate wear fault

2.2.2. Contact and Collision Mechanics Model

When there is a gap between the crosshead and the guide plate, the collision force mainly consists of the normal collision force and the tangential friction force during the collision between the crosshead and the guide plate. Due to the wear of the crosshead guide plate, the movement of the crosshead on the guide plate changes from reciprocating motion to free motion, and the motion state becomes very complex. The Coulomb friction force model is an empirical model used to describe the friction force generated in adhesive or sliding contact. Therefore, this model is adopted to describe the tangential friction force during the collision between the crosshead and the guide plate. The simulation model of Coulomb tangential friction force is:

$$f_i = -\mu_d F_i \operatorname{sgn}(v), i = A、B、C、D \tag{1}$$

In the formula:

μ_d -Coefficient of sliding friction;

F_i -Collision force at each collision point of the two ends of the crosshead, kN;

$Sgn(v)$ -Symbol function related to the direction of the crosshead speed.

$$sgn(v) = \begin{cases} 1, & v > 0 \\ -1, & v < 0 \end{cases} \tag{2}$$

The Lankarani-Nikravesh nonlinear spring-damper collision force model based on the Hertz contact force model takes into account the kinetic energy loss of the components after the collision and the energy loss caused by the system's own damping. It is very suitable for situations where there is energy loss during the collision. Therefore, the Lankarani-Nikravesh nonlinear spring-damper collision force model can be adopted. The normal contact collision force formula is:

$$F_i = K\delta_i^n + C\dot{\delta}_i,\ \ n = 1.5, (i = A, B, C, D) \tag{3}$$

In the formula:

K -Stiffness coefficient, N/mm;

C -The total damping coefficient during the collision process, N.S/mm;

δ_i -The penetration depth of the collision, mm;

$\dot{\delta}_i$ -Relative collision speed, m/s.

Expression of the stiffness coefficient:

$$K = \frac{4\sqrt{R_i}}{3\pi\left(\frac{\left(1-\mu_i^2\right)}{E_i} + \frac{\left(1-\mu_j^2\right)}{E_j}\right)} \tag{4}$$

In the formula:

R_i -The curvature radius of the hypothetical spherical surface, in millimeters;

μ_i、μ_j -The Poisson's ratios of the crosshead and the guide plate respectively;

E_i、E_j -The elastic modulus of the crosshead and the guide plate, GPa.

The damping coefficient is:

$$C = \mu\delta^n \tag{5}$$

In the formula:-Lag damping coefficient,

$$\mu = \frac{3K\left(1-C_e^2\right)}{4\dot{\delta}^{(-)}} \tag{6}$$

C_e -Recovery coefficient, $C_e = 1-\alpha\dot{\delta}^{(-)}$ where α represents the material characteristic parameter;

$\dot{\delta}^{(-)}$ -Initial speed at the time of collision, m/s.

When the edges or vertices of the two ends of the crosshead collide with the inner surface of the guide plate, formula (3) is applicable. However, when the crosshead is in contact with the guide plate or in a free state of separation, no collision force is generated. In such cases,Fi=0. In conclusion, the contact collision normal force simulation model is:

$$F_i = \begin{cases} K\delta_i^n + D\dot{\delta}_i, & \delta_i > 0 \\ 0 \quad , & \delta_i \le 0 \end{cases} \ \text{(i=A,B,C,D)} \tag{7}$$

2.3. Modeling of Hydraulic System and Joint Simulation

2.3.1. Construction of the AMESim Hydraulic Model

AMESim is a software developed by LMS specifically for the modeling, simulation and dynamic analysis of mechanical and hydraulic systems. It is widely used in various fields such as aerospace, heavy machinery, and vehicles. It includes complex system modeling and simulation for mechanical, fluid, thermal analysis, control, and electromagnetism. Therefore, in this paper, the AMESim simulation software is selected to model and build the hydraulic system of the plunger pump, and the established hydraulic system model of the plunger pump is shown in Figure 3.

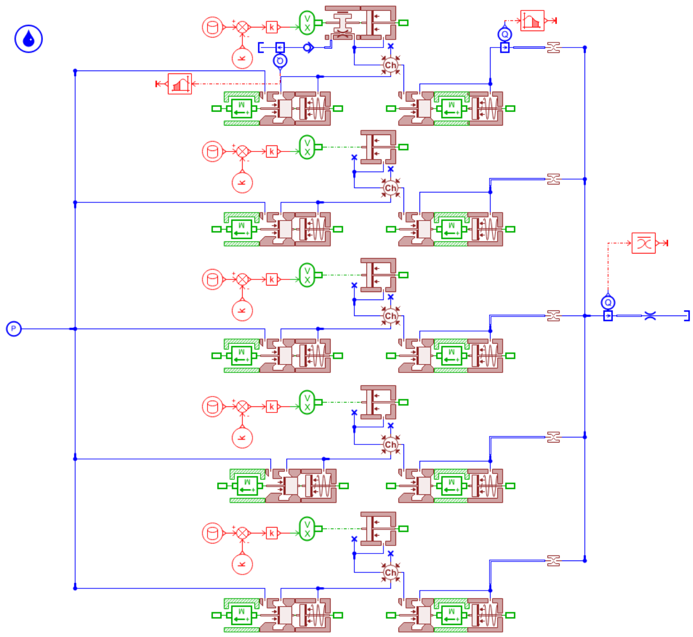
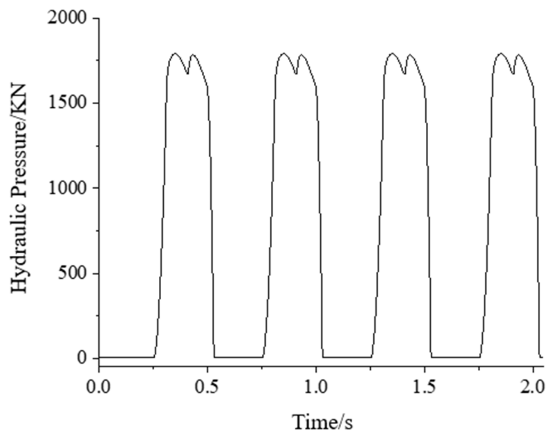


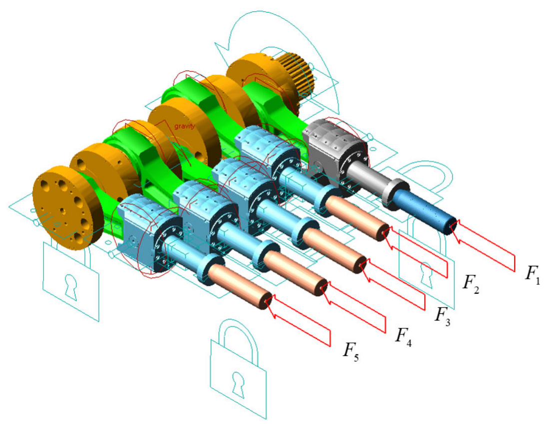
Figure 3. Hydraulic System Model of the Plunger Pump

2.3.2. Hydraulic - Multi-body Dynamics Combined Simulation

During the actual operation of the plunger pump, due to the reciprocating linear motion of the plunger, the pump cylinder undergoes a periodic process of liquid intake and discharge. The presence of the liquid causes a liquid pressure on the plunger end face. Additionally, during the reciprocating motion of the plunger, it generates frictional force with its sealing components, which acts in the opposite direction of its motion. The magnitude of the frictional force is related to various factors and is difficult to calculate precisely. At the same time, it is much smaller than the plunger force. Therefore, this frictional force can be ignored. Using the hydraulic model in this chapter, the variation curve of the pressure each plunger experiences over time is calculated, and the variation curve of the liquid pressure over time is also calculated, as shown in Figure 4(a). The liquid pressure is imported using the spline function provided by Adams, and it is assumed that the liquid pressure acts as a concentrated force on the center of the plunger end face. After adding the liquid pressure, the model is shown in Figure 4(b). Keeping the wear gap of the crosshead guide plate at 0.2 - 0.7 mm, perform dynamic simulation to analyze the vibration response of the crosshead guide plate wear fault under the liquid load.



(a) Liquid pressure variation curve



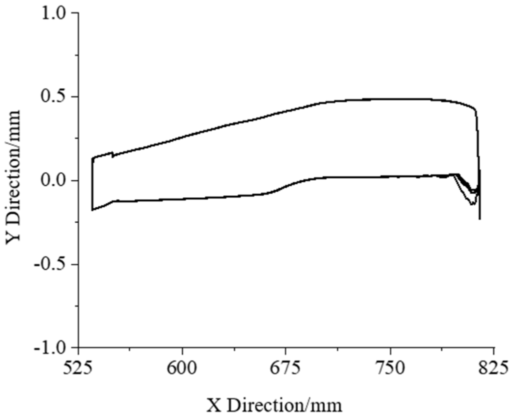
(b) Diagram of Liquid Pressure Addition

Figure 4. Changes in liquid pressure and the location where it is applied

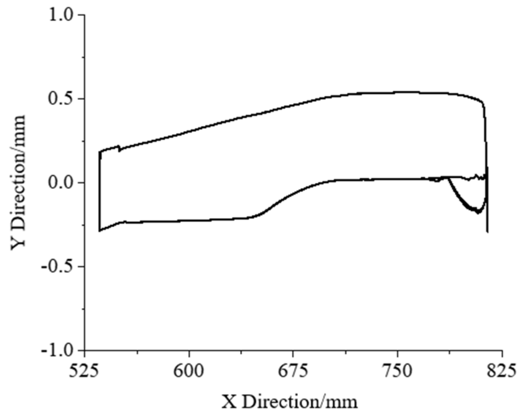
3. Results and Analysis

3.1. Analysis of the axial displacement of the crosshead shaft under liquid load

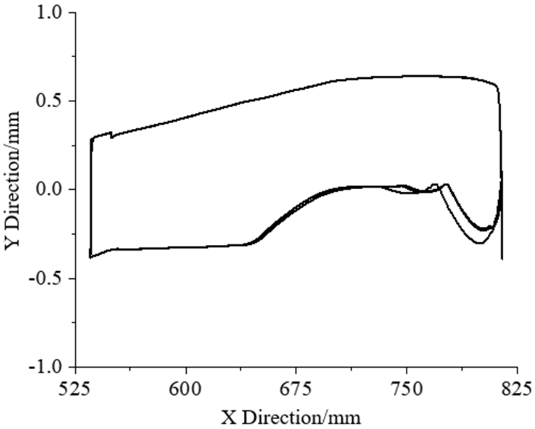
When analyzing the displacement trend of the crosshead axis, due to the gap between the crosshead and the guide plate, the crosshead sinks. Therefore, the displacement of the crosshead axis center in the XY plane was extracted for analysis, and the result is shown in Figure 5. Among them, the maximum value in the X direction is the far stop point and also the starting point, and the axis trajectory rotates counterclockwise.



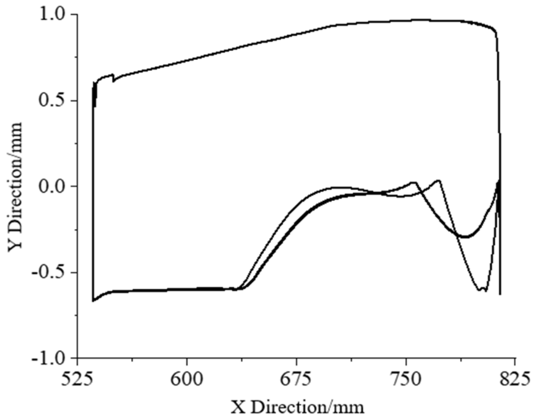
(a) e = 0.2mm Axial displacement in the Y direction



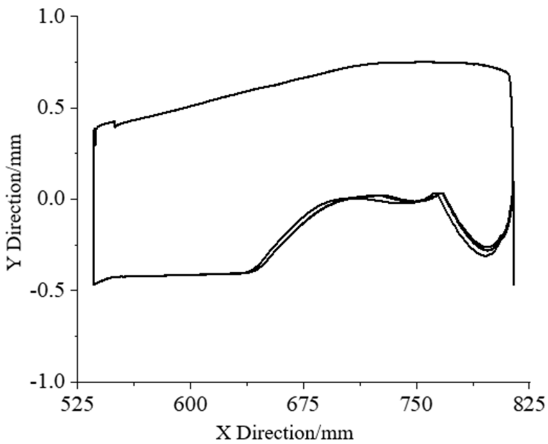
(b) e = 0.3mm Axial displacement in the Y direction



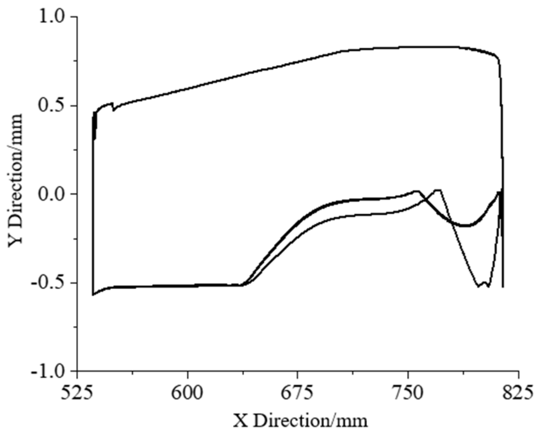
(c) e = 0.4mm Axial displacement in the Y direction



(d) e = 0.5mm Axial displacement in the Y direction



(e) e = 0.6mm Axial displacement in the Y direction



(f) e = 0.7mm Axial displacement in the Y direction

Figure 5. Graph showing the axial displacement variation of the crosshead within the XY plane

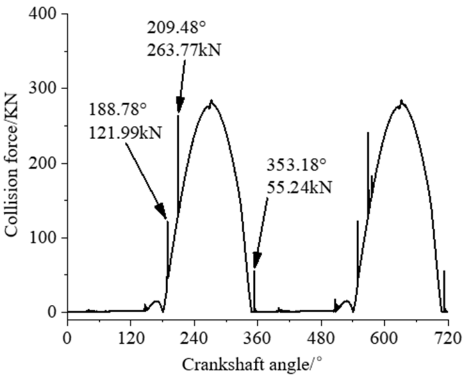
From Figure 5, we can see:

1. The Y-direction displacement of the crosshead axis within the XY plane increases with the continuous increase of the clearance. Near the far stop point and the near stop point, there is a sinking phenomenon, and the sinking range near the near stop point is larger than that near the far stop point.

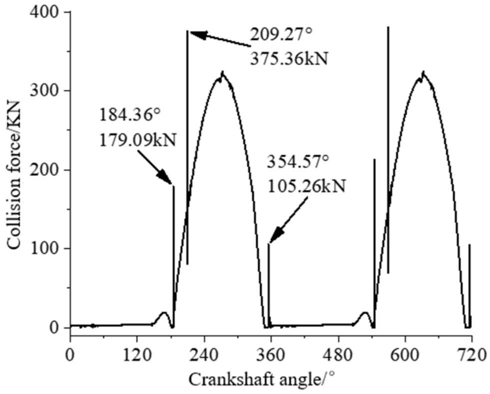
2. The displacement of the crosshead axis in the Y direction undergoes the following processes: first, a drop followed by a rise at the far stop point (starting point); then, a slow drop in the middle part; then, a sinking near the near stop point; and finally, a momentary drop followed by a slow rise during the return process.

3.2. Analysis of Contact Collision Forces of Crosshead under Liquid Load

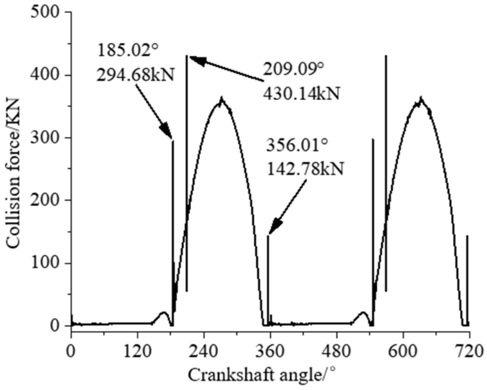
When there is a gap in the large head cap, the collision force between the large head cap and the crankshaft will cause vibration of the housing in all directions. Therefore, the resultant force of the collision force is extracted for analysis. Taking 0.05mm as the increment, the variation trend of the contact force as the gap e increases from 0.2mm to 0.7mm is analyzed, as shown in Figure 6.



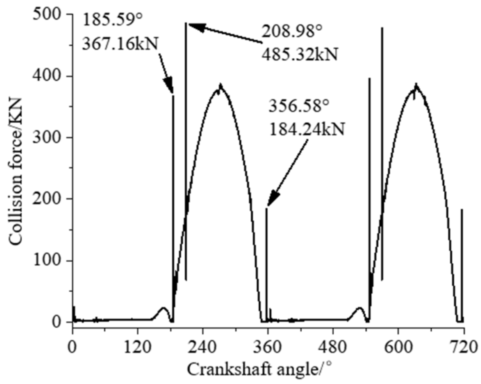
(a) Waveform diagram of collision force when e = 0.2mm



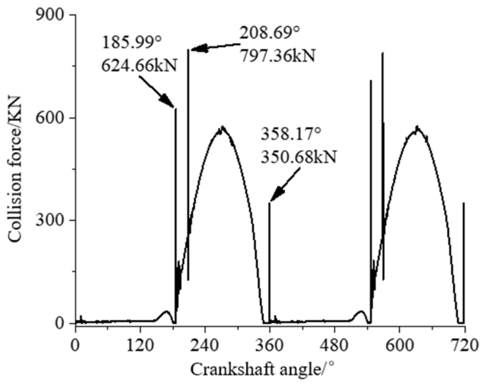
(b) Waveform diagram of collision force when e = 0.3mm



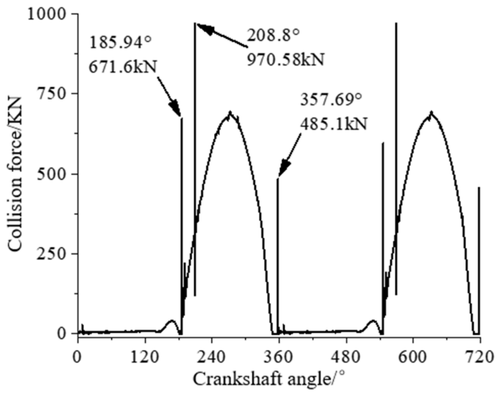
(c) Waveform diagram of collision force when e = 0.4mm



(d) Waveform diagram of collision force when e = 0.5mm



(e) Waveform diagram of collision force when e = 0.6mm



(f) Waveform diagram of collision force when e = 0.7mm

Figure 6. Graph of Force Variation during Collision

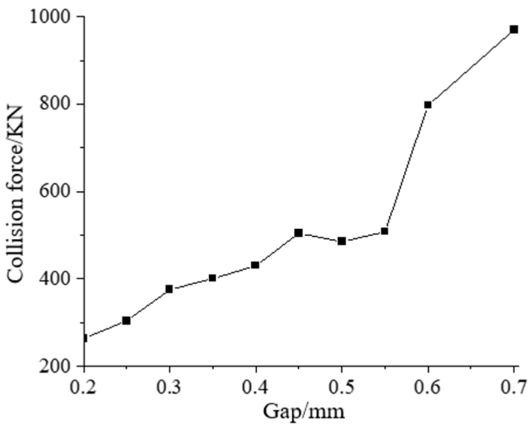


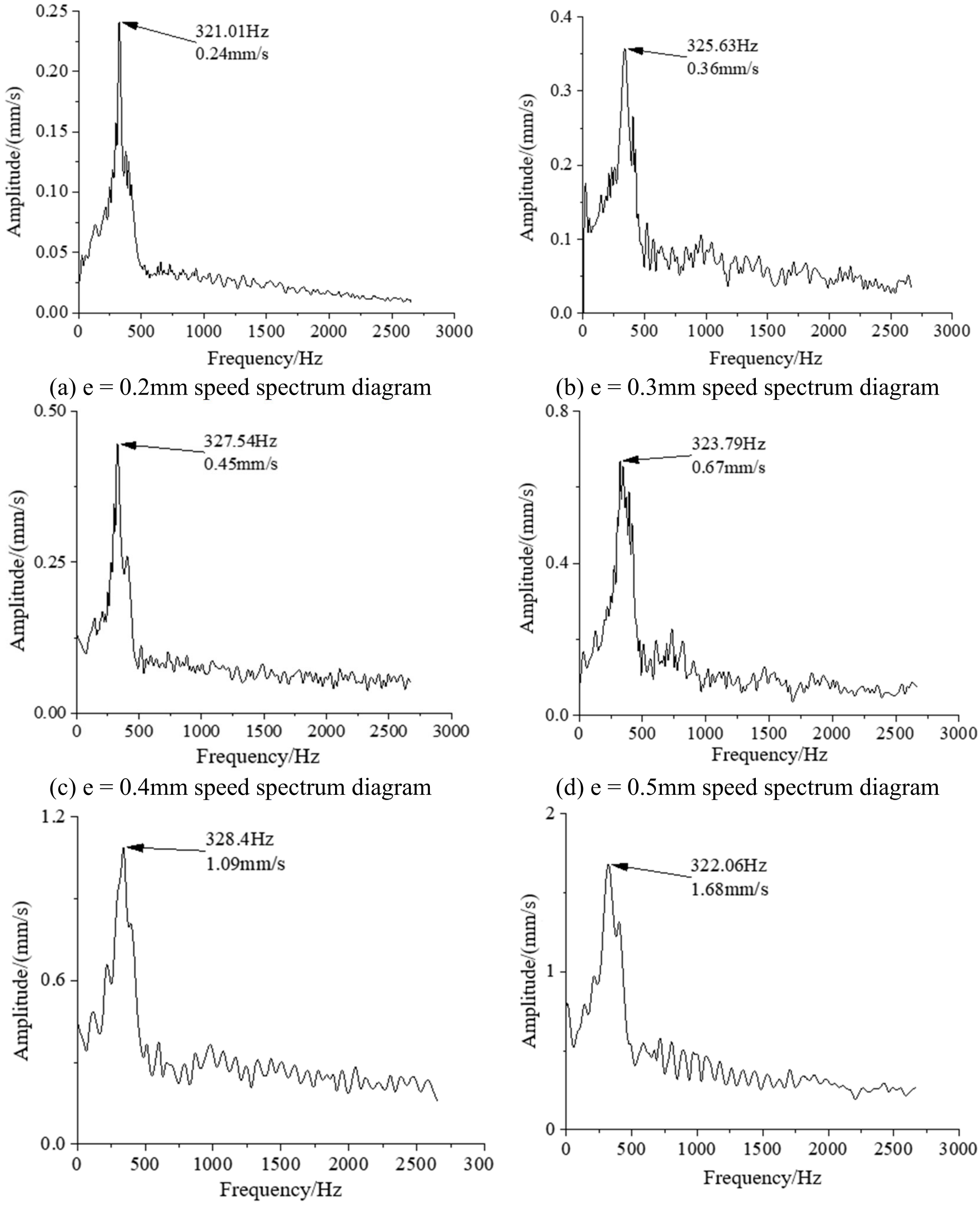
Figure 7. Peak Force Curve of Collision

From Figure 6, we can see:

When liquid pressure is present, the maximum collision force between the crosshead and the guide plate increases as the gap continues to widen. Referring to the collision force peak graph, as shown in Figure 7, it can be seen that the collision force significantly increases when the gap is greater than 0.6mm.

3.3. Analysis of shell vibration response under liquid loading

The vibration frequencies at each measurement point on the plunger pump housing were analyzed. With an increment of 0.05mm, the analysis was conducted on the gap ranging from 0.2mm to 0.7mm. The speed spectrum envelope diagram of the measurement points on the shell is shown in Figure 8.



(a) e = 0.2mm speed spectrum diagram

(b) e = 0.3mm speed spectrum diagram

(c) e = 0.4mm speed spectrum diagram

(d) e = 0.5mm speed spectrum diagram

(e) e = 0.6mm speed spectrum diagram

(f) e = 0.7mm speed spectrum diagram

Figure 8. Speed Spectrum Diagram on the Shell

The results show:

1. When the gap is 0.2mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 321Hz. When the gap is 0.3mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 325Hz. When the gap is 0.4mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 327Hz. When the gap is 0.5mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 323Hz. When the gap is 0.6mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 328Hz. When the gap is 0.7mm, the velocity spectrum of the measuring points on the shell mainly concentrates around 322Hz.

2. The presence of hydraulic pressure causes the crosshead guide plate to suffer wear and failure. In this case, the velocity spectrum on the shell is more concentrated around 320Hz.

4. Conclusion

Based on the multi-body dynamics and the model of moving joint contact and collision with clearance, combined with the joint simulation of the hydraulic system, this paper studies the dynamic response of the wear fault of the crosshead guide plate of the plunger pump under liquid load, and obtains the following conclusions:

1. When liquid pressure is present, the maximum collision force between the crosshead and the guide plate increases as the gap continues to widen. When the gap exceeds 0.6mm, the collision force significantly increases.

2. The vibration energy is concentrated near the second natural frequency of the housing (about 325Hz), and the spectral peak increases with the increase of the clearance.

3. Through the typical fault simulation of key components of the plunger pump, the characteristic data such as the peak value of the velocity spectrum when the crosshead guide plate undergoes a typical fault under the liquid pressure condition were obtained.

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