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To cite this article: Wenshan Zhang *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **604** 012020

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Dynamic research of piston/piston assembly in labyrinth reciprocating compressor

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Abstract. The piston of labyrinth reciprocating compressor is often in the eccentric running state during operation, which leading to serious gas leakage and reduced compressor volume efficiency. To optimize the performance of labyrinth reciprocating compressor, the main reasons of piston eccentric operation are analyzed in detail in this paper. Based on the theory of multi-body dynamic and influence of piston bending and vibration, a rigid-flexible coupling multi-body virtual simulation model of piston/piston rod assembly is established by using numerical analysis software. The factors causing eccentric running of piston in cylinder are investigated and the eccentric running track of piston in cylinder and its dynamic characteristics are obtained.

1. Introduction

The radial motion law of the labyrinth piston in the cylinder and its dynamic behavior have a direct impact on the design of the gap value, the distribution of the labyrinth flow field and the volumetric efficiency of the compressor. To solve the problem of low volumetric efficiency of the current labyrinth compressor, it is necessary to first clarify the motion characteristics of the piston in the cylinder and its radial motion law. Due to the complex motion relationship of the piston rod/piston rod assembly system, various coupling factors need to be involved. Up to now, there have been relatively few reports on the movement characteristics of the piston in the cylinder. Only some scholars have conducted it with single influencing factor. Such as G. K solved the one-dimensional radial vibration equation of the piston and obtains the eccentricity of the piston [1]. Aiming at the vibration characteristics of the piston assembly, Lindi Wang and other scholars theoretically derived the mathematical model of the amplitude calculation and analyzed the natural vibration frequency of the piston assembly and its influence on the piston amplitude [2]. Xinyu Dong et al. used the modal extraction method in the large-scale finite element software ANSYS to analyze the modal analysis of the piston rod and optimize the installation position of the guide bearing. However, the reciprocating motion of the piston/piston rod assembly is composed of a plurality of components and their mutual motion, and the motion relationship between the components is coupled, which is a complex nonlinear motion system. The above research is only far from enough to study the motion state of the piston in the cylinder from the single factor or the motion characteristics of a certain component. In order to better study the motion characteristics of the piston/piston rod assembly, this paper uses the multi-body dynamics theory to establish a rigid-flexible coupling virtual simulation model of the piston/piston rod system to study the dynamic characteristics of the piston under the influence of multi-factor coupling factors and regularity.



2. Multibody dynamics theory

With the development of the current machinery industry, the traditional classical mechanics cannot meet the needs of the study of complex systems. Explore more reasonable research methods and build more realistic models to reflect the actual system becomes a new demand. A multi-body system is a complex mechanical system that is connected by multiple objects through a motion pair. The fundamental purpose of multibody system dynamics is to apply computer technology for dynamic analysis and simulation of complex mechanical systems. It is a branch of new discipline based on classical mechanics. The dynamics simulation of mechanical systems can often be used to study the relationship between the displacement, velocity, acceleration of each rigid body of the system and its force or moment. The multi-body dynamics simulation is to construct a complete dynamic system by constructing a series of rigid bodies and flexible bodies by articulating their mutual constraints [3]. The hinge mainly constrains the relative motion between the rigid bodies. The use of multi-body dynamics to analyze the dynamic characteristics of complex mechanisms has become an effective and reliable technical method.

The multi-body system studied in the calculation of multi-body system dynamics can be divided into multi-rigid system, flexible multi-body system and rigid-flexible hybrid multi-body system according to the mechanical properties of the objects in the system. A multi-rigid system refers to a system that can ignore the elastic deformation of an object in the system and treat it as a rigid body. This type of system is often in a low-speed motion state; a flexible multi-body system refers to a large range of objects that appear in the system during motion. The coupling of motion and the elastic deformation of an object, so that the object must be treated as a flexible body.

3. Modeling method

3.1. Multi-rigid modeling theory

During the movement of the piston/piston rod assembly, each component will produce a certain elastic deformation after being subjected to the force. In theory, all components are subject to force deformation. However, this doubles the workload, occupies a lot of computer resources, and the computational accuracy is not significantly improved. Normally, components that do not need to be considered for deformation or small deformation are directly used in the rigid body model, and do not affect the accuracy of the calculation results. In this paper, components such as pistons, guide bushings, crossheads, connecting rods, crankshafts, slides, etc. are treated as rigid bodies.

The traditional kinematics analysis is mainly based on the differential relationship between the displacement, velocity and acceleration of the moving parts and the space vector synthesis theory. The motion analysis in the multi-body dynamics system is based on the interconnection relationship between the components, that is, the motion sub-constraint equation between the moving parts, and solves the displacement, velocity and acceleration of the components [4]. Firstly, the position equation equivalent to the motion pair in the system is sought, and then the position constraint equation is derived to obtain the constraint equations of velocity and acceleration. Finally, the equations are numerically solved, and the corresponding coordinate transformation is performed to obtain the corresponding speed and acceleration in the coordinate system [5,6].

3.2. Flexible body modeling theory

In the reciprocating labyrinth compressor, the piston rod is the most critical bearing and guiding component, and its motion characteristics directly determine the radial motion law and dynamic characteristics of the piston. In addition, the restraining position of the guide bushing on the piston rod periodically changes with the crank angle. Therefore, in this paper, for the modeling of the piston rod, the rigid model cannot be used alone, and the bending effect and vibration characteristics of the piston rod under the action of the large periodic piston force must be considered at the same time, that is, the flexible model is adopted.

Unlike a rigid body, a flexible body is a deformed body. During the movement, there is a relative motion relationship between the points in the flexible body. Since any reference frame is impossible to completely consolidate with the deformed body, the position of the deformed body is accurately reflected. However, the reference frame cannot be consolidated with the deformed body but can only float in the deformed body [7,8]. Therefore, in the flexible piston rod model, the movement of any point on the piston rod is actual result of the coupling of the rigid body motion and the flexible deformation motion of the floating coordinate system.

3.3 Mathematical theory of slip motion

In a mechanical system, different moving parts are connected to each other to form different motion relationships. Influenced by factors such as design, manufacturing accuracy, assembly, and long-term wear, the gap is common between the motion pairs. When the mechanical system is moving, especially in the case of high-speed operation, due to the existence of the motion pair gap, the instantaneous moving parts may be separated and re-contacted. However, when the relative moving parts come into contact again, a strong impact is generated, and the speed, acceleration, and motion side reaction force of the parts are drastically changed, and noise and wear occur, causing changes in the dynamics of mechanical systems [9]. In this context, a sliding gap is provided between the guide bushing and the piston rod. Similarly, during the operation of the compressor, the relative movement between the piston rod and the guide bushing and the crosshead and sliding will inevitably lead to contact-collision. For the study of the contact-collision problem of the gap motion pair, it is generally considered that the moment of the collision of the moving component changes, although the component configuration does not change. The short-lived collision process actually apply an instantaneous unilateral constraint to the moving parts, and the collision of the boundary of the parts does not interfere.

In this paper, the contact collision model is used to divide the collision process between the gap motion pair and the moving component into two states: "free motion - contact deformation". According to the law of elastic damping between the contact surfaces of the components, the coupling relationship between the contact force and the deformation during the collision phase is derived. The equivalent spring damping model is used, and its generalized expression is Equation 1. Where F is Collision force of the collision component along the normal direction, C_1 C_2 is Damping factor, δ is Penetration distance of the contact point of the collision component along the normal direction, e is an index greater than or equal to 1, K is Contact stiffness.

$$F = K\delta^e + C_1\dot{\delta} + C_2\ddot{\delta} \quad (1)$$

At the moment of the initial contact of the component, that is, when the relative moving component transitions from the free state to the contact collision phase, the original constraint is not applicable to the collision phase. Firstly, the original constraint is invalidated, the constraint is replaced by the constraint force, and the equivalent spring damping model is introduced between the relative moving parts. According to the dynamic model established by the theory of contact force deformation, the degree of freedom of the relative moving parts is not directly related to the motion of the gap. This method can be used to transform the complex dynamics of the collision into a dynamic problem without topological changes.

4. Compressor model

In the reciprocating compressor, the unbalanced reciprocating inertial force of the body is transmitted to the casing through the main bearing, causing vibration of the casing. Since the vertical structure is employed, the reciprocating inertial force acts vertically on the foundation and is more easily absorbed by the foundation. Han Tao and other related scholars analyzed the dynamic response of the casing of the current reciprocating labyrinth compressor. The results show that the vibration amplitude of the labyrinth compressor casing is less than 0.02 mm [10]. This kind of small-range displacement vibration of the casing has little influence on the motion characteristics of the piston and can be ignored. In order

to simplify the model, the influence of the vibration of the casing on the piston movement is not considered. The radial movement of the piston is mainly studied from the crank-link mechanism itself.

The main components of the crank-link mechanism of the reciprocating labyrinth compressor are: piston, piston rod, crank, connecting rod, crosshead, slide, guide bushing. The piston as a guided member is suspended from the top of the piston rod, and the piston rod serves as the most critical guiding and bearing member. During the movement of the compressor, the force and motion characteristics of the piston rod directly determine the movement of the piston in the cylinder. The main reasons for the eccentric operation of the labyrinth compressor piston can be summarized as the following three aspects [11]:

1. During the design, manufacture and installation of the compressor, there is a radial gap between the crosshead and the slide, the piston rod and the guide bearing. When the compressor is running, the crosshead reciprocates in the chute, and it also oscillates along the radial gap, thereby causing the piston mounted on the top end of the piston rod to be radially deflected.
2. As the main bearing member, the piston rod is restrained by the crosshead and the guide bushing and is equivalent to the outwardly extending beam structure with one end fixed at the end. Under the action of a large periodic piston force, a slight deflection of the piston causes the piston force to produce a periodically varying radial disturbance in the radial direction of the piston. Under the radial disturbance force, the piston rod may be bent and deflected.
3. The piston/piston rod assembly not only withstands large periodic gas force loads, but also with radial power and unbalanced inertia forces. These cyclical loads may cause a strong resonance of the piston/piston rod assembly over the range of compressor speeds, further exacerbating the piston's trajectory from the cylinder centerline.

During the actual operation of the compressor, the eccentricity of the piston and its piston rod is mainly the result of the coupling of the above three factors. Based on multi-body dynamics and computer simulation software, this paper takes an actual compressor as an example to establish a multi-factor coupling virtual model of the labyrinth piston/piston rod assembly and explores the radial motion of the piston under multi-factor coupling law. The key guide piston rod is treated as a flexible body.

The main parameters of the target compressor are shown in Table 1.

Table 1. The parameters of the compressor

Category	Vertical
Type of action	Double
No. of cylinder	1
Stroke	90 mm
Rotational speed	600 r/min
Crank radius	45 mm
Piston rod length	420 mm
Connecting rod length	200 mm
Suction/Discharge pressure	0.1/0.5 MPa(G)
Clearance between crosshead and slid-way (single side)	0.1 mm
Clearance between piston rod and guide bearing (single side)	0.05 mm

The flexible body model of the piston rod is established by the finite element software ANSYS. The model includes the bending stiffness, mass, centroid, frequency and vibration mode of the first 10 steps of the piston rod. The bending effect and vibration characteristics of the piston rod are considered. The three-dimensional solid model of other moving parts is established, and the three-dimensional model is

assembled in the ADAMS analysis software according to the actual connection mode of the crank-link mechanism, and the corresponding motion pair is added at the connection point. The assembled multi-body model is shown in Figure 1.

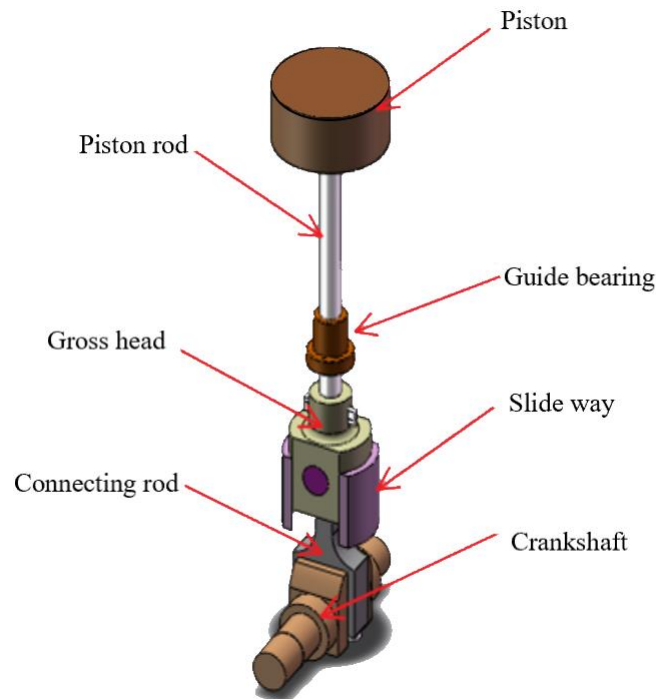


Figure 1. 3D simulation model

In the multi-body model, the reciprocating inertial force is automatically calculated during the simulation of the ADAMS software and does not require additional loading. The only external force that needs to be applied is the gas force, shown in Figure 2. In this paper, the gas force is loaded by calling and writing the Step function that comes with ADAMS. Based on the contact collision model, the contact force function of the slip gap pair is set between the crosshead and the slide, and between the piston rod and the guide bushing, which is equivalent to adding a nonlinear spring damping to the motion pair. Define the state of motion between them. In the ADAMS software, the Impact function is used to define the "contact-collision" motion of the crosshead and the slide.

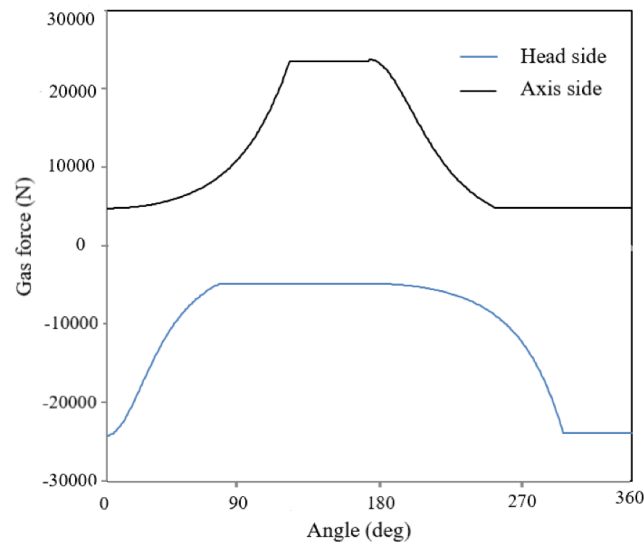


Figure 2. Gas force

5. Results and discussion

After setting the boundary conditions, the model is loaded according to the actual speed and pressure ratio, and the following results are obtained. Figure 3 shows the trajectory of the model compressor piston under multi-factor coupling. The eccentric motion of the piston mainly occurs in the plane of the connecting rod, and the piston is mainly biased toward the non-stress side of the crosshead during operation. The direction indicated by the arrow is the eccentric motion direction of the piston at various times throughout the stroke, which is consistent with the actual situation. It can be seen from the figure that the eccentric trajectories of the pistons in the cylinders all exhibit an "8" shape. When the piston approaches the top dead center, the distance between the crosshead and the guide bushing is reduced, and the bending resistance of the piston rod is the weakest, thereby causing the piston eccentricity to reach a maximum near the top dead center. At different speeds, due to the cyclical change of the piston rod stiffness, the running track of the piston still shows an "8" shape, and as the crankshaft speed increases, the piston eccentricity also increases.

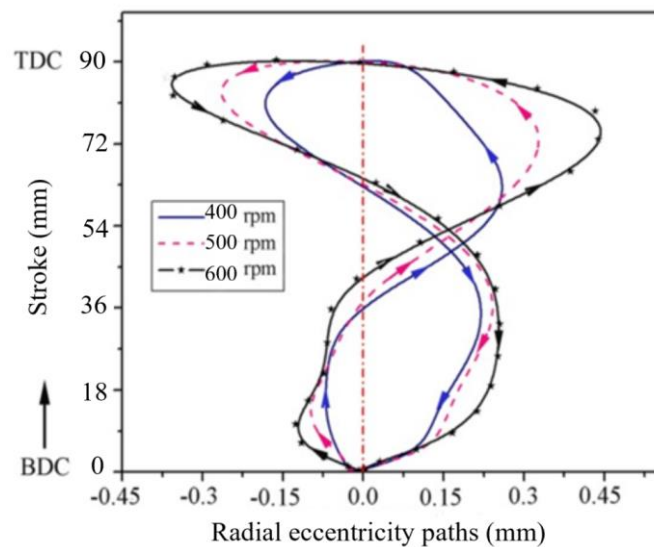


Figure 3. Piston's radial eccentric paths with different speed

Figure 4 shows the effect of gas force change on the trajectory of the piston. As the gas force on the piston increases, the bending deformation of the piston rod increases when the crankshaft speed is constant, so the piston eccentricity also increases. The gas force increases rapidly, and the span between the piston and the guide bushing increases when the piston approaches the top dead center (compression and exhaust process), causing the radial acceleration of the piston to change drastically and reach a peak. At the end of the bottom dead center, since the model compressor is double-acting, the axial force of the piston also rapidly increases to the peak value, so the radial acceleration of the piston also increases. However, at the bottom dead center, the span between the piston and the guide bushing is reduced, and the piston rod stiffness is increased, so the radial acceleration of the piston increase and the frequency of change is less than the top dead center.

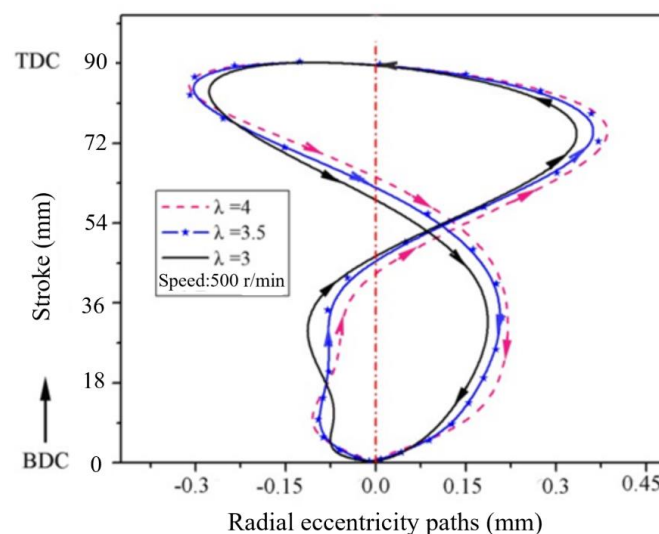


Figure 4. Piston's eccentric paths with different pressure ratios

Figure 5 shows the radial acceleration curve of the model compression piston at different speeds. As the speed increases, the radial acceleration increases.

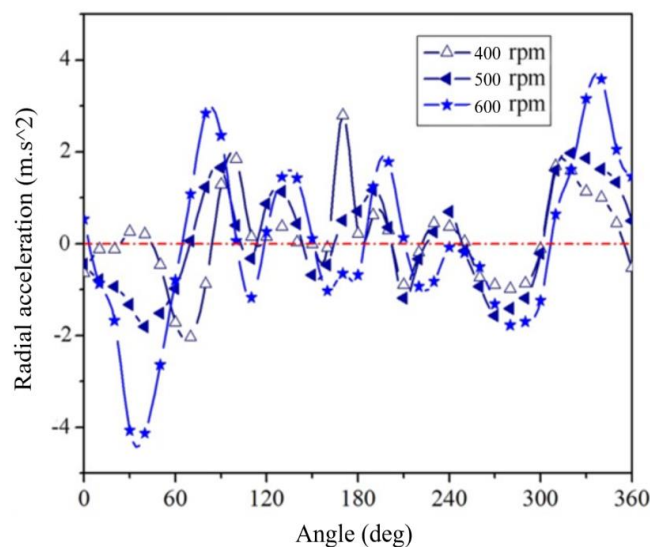


Figure 5. Piston's radial acceleration

6. Conclusion

In this paper, a rigid-flexible multi-body and multi-body virtual simulation model of piston/piston rod assembly is established by using ANSYS and ADAMS commercial numerical analysis software. The radial motion law and dynamic characteristics of the piston are studied. It has been found that the swinging, flexible bending and vibration of the piston rod can cause the piston rod to operate eccentrically. The eccentric running trajectory of the piston in the cylinder is characterized by an "8" shape, and the eccentricity increases as the rotational speed and the gas force increase.

Acknowledgments

The authors acknowledge the financial support of the National Science Foundation of China [No.51676150].

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