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A Comparative Study of Joint Clearance Effects on Dynamic Behavior of Planar Multibody Mechanical Systems

Abstract

Clearance always exists in actual joint due to many uncertainties such as machining tolerance, assemblage and load deformation. The main purpose of this paper is to propose a computational and experimental study on the dynamic characteristics of a planar multibody mechanical system with joint clearance. For this purpose, a suitable dynamic model plays a crucial role in simulating the overall performance of the mechanical systems. To describe the interaction in joints with clearance, the normal contact model is established based on the Lankarani–Nikravesh contact force model, while the friction effect is conducted using the Coulomb friction model. Meanwhile, the experimental platform is set up and a planar mechanism with clearance joint is employed as a model mechanism. The obtained results demonstrate that the dynamic behavior of the mechanism can be effectively predicted by this method. Furthermore, the effects of the rotational speed of crank shaft and size of joint clearance on the dynamic response of the mechanism are investigated.

Keywords

Multibody dynamics; revolute joint clearance; contact model; dynamic response.

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1 INTRODUCTION

With the development of aviation and automotive industry, the high-strength and low-plasticity materials are widely used, which causes the part manufactured with large batch and precision. In order to satisfy the requirements of clean and green production for manufacturing, pressing machine is widely employed to manufacture pressed metals and it is an effective machine tool for metal forming. Due to manufacturing tolerances, material deformation and assemblage, clearance joints unavoidable exist in the high speed press mechanical systems. In general, the effects of clearance joints are always neglected in dynamic analysis of mechanical system. However, the existence of revolute joint clearance causes excessive wear, noise and impact dynamic load. Moreover, it has serious effect on the dynamic performances and dynamic stability analysis of mechanism. Therefore, it is essential to establish an adapted model of a high speed press system to investigate the effects of clearance joints on dynamic behavior of machine (Liu and Tian, 2011, 2012; Machado et al., 2012; Bai et al., 2014; Varedi et al., 2015; Pereira ta al., 2015).

In the recent decades, the dynamic responce of multibody system with joint clearance has been investigated by many researchers analytically and experimentally (Flores et al., 2009; Liu et al., 2012; Zhang et al., 2013). Some of them focus upon that the multibody mechanical system is modeled with a realistic revolute joint. Dubowsky (1978) studied the influence of joint clearance and flexibility on dynamic characterics of planar mechanisms. The results showed the influence of impact force on the elastic mechanical system with clearance by experiment. Ravn (1998) proposed a continuous analysis methodology for planar mechanical system with clearance joints. The continuous analysis method was combined with the contact force model to depict joint clearance in revelute joint. The experimental data was used to demonstrate the validity of the presented analysis method. In addition, Schwab et al. (2002) focused on the influence of revolute joint clearance on the dynamic behavior of mechanical system and machines. The slider-crank mechanism was implemented as an example to conduct the dynamic analysis. The results of this study indicated that the connection of links with the lubrication of the joint smoothed the fluctuation value of the contact force. A comparative study, which combined simulation and experimental investigation on the dynamic behavior of the slider-crank mechanism with joint clearance, was conducted by Khemili and Romdhane (2008). Flores (2010) analyzed and discussed the influence of parameters on dynamic behavior of the mechanical system with multiple clearance joints. The results obtained in the paper represented that the clearance size and the operating condition played an important role in accurately predicting the dynamic behavior of planar mechanical system. The numerical simulation results were compared with experimental data, which revealed that the method could effectively describe the dynamic response of the mechanical systems. And then, Tian et al. (2011) proposed a new methodology to model and describe the multibody mechanical system with joint clearance. They also observed that the methodology could effectively describe the misalignment of journal inside the bearing. Based on the contact model, Bai and Zhao (2013) also presented a general computational approach for the modeling and dynamic analysis of planar mechanical system with clearance joint. This methodology could illustrate the dynamic characterics and wear phenomenon of clearance joint in the multibody systems. Furthermore, Muvengei and Kihiu (2013) investigated the dynamic characterics of a rigid-body multibody systems considering two-clearance revolute joints. This study suggested that the location of clearance joint played a crucial role on dynamic analysis of the planar multibody system with joint clearance. Erkaya et al. (2014) studied the kinematic and dynamic behavior of the planar four-bar mechanicak system with joint clearance and flexible linkage. They found that the flexibile linkage played an important role on the vibration of mechanical systems. Meanwhile, Erkaya and Uzamay (2015) carried out a numerical and experimental investigation to analyze the influence of joint clearance on partly compliant articulated mechanism. It was clearly shown from the results that the flexibility of flexural pivot had the suspension effects to minimize the undesired outputs of mechanism with clearance joint. The obtained experimental results showed the dynamic behavior of the multibody mechanism system with joint clearance.

It is important to note that the above mentioned references mainly focus on the dynamics model of planar mechanism system considering the modes of contact, because a suitable contact force model can make analysis more accurate. Some studies (Machado et al., 2012; Bai et al., 2015) have conducted the influence of clearance on dynamic characterics of mechanical systems considering contact–impact model. Lankarani and Nikravesh (1994) proposed a continuous contact force model for the contact-impact analysis of a two-particle collision, which considered the energy loss in the impact process. This model was employed by many researchers for different problems. Based on the contact force model, Flores and Ambrosio (2006) presented a methodology to model contact-impact forces of contact zone in multibody system with joint clearances. They discussed the influence of friction on the mechanism based on the elastic Hertz theory and classical Coulomb's friction law, which presented that the wear phenomenon occurred in the contact process. Meanwhile, Flores and his co-workers (2010) presented a general and comprehensive approach to deal with the transitions between non-contact and contact situations in multibody dynamics effectively. A demonstration case provided the results that illustrated the validity of the presented approach. Moreover, a new contact force model of revolute joint with clearance for planar mechanical systems was developed by Bai and Zhao (2013), which was built based on the Lankarani-Nikravesh contact force model and the improved elastic foundation model. In this paper, the hybrid contact force model was analyzed and compared with the existing contact models. The numerical simulation results illustrated that the proposed hybrid contact force model was an effective methodology to predict the dynamic behavior of planar mechanical systems with joint clearance. Tian (2013) provided a new method for analysis of rigid-flexible multibody systems with elastohydrodynamic lubricated cylindrical joints. They obtained a reduction in the contact force by introducing flexible body. Gummer and Sauer (2014) investigated the effects of joint clearance on the dynamic behavior of a slider-crank mechanism into RecurDyn. The results showed that the method was suitable for describing a slider-crank mechanism. Alves and Peixinho (2015) also conducted a comparative study on the most relevant existing viscoelastic contact force models. The results indicated that the prediction of the dynamic characterics of contacting solids strongly depended on the selection of the contact force model. And then, Ma et al. (2015) studied the effects of joint clearance on the dynamic behavior of planar multibody systems. The contact force model was established based on the Lankarani-Nikravesh contact force model and modified Coulomb friction model. They also observed that the interaction force increased as the increasing of clearance size and crank angular velocity, which could provide helpful suggestions for the designing of the classical slider-crank mechanism with joint clearance.

In contrast to the extensive study on the influence of clearance joint on the dynamic characteristics of multibody mechanism systems, devoting to the dynamic analysis of the slider-crank mechanism system with balancing mechanism is more limited. Erkaya and Uzmay (2012) were among the few researchers who investigated the effects of balancing and flexibile linkage on the kinematic and dynamic analysis of a slider-crank mechanism with clearance by experiment. They also performed the optimization study of mechanical system for reducing the vibration effects. Zheng and Zhou (2014) built the dynamic model of a high speed press system with consideration of flexible link and clearance joints. They proposed an effective dynamic model to study the effects of clearance joints on a high speed press system. Unfortunately, experimental investigation of joint clearance effects on the dynamics analysis of the mechanical systems was not performed in their study. Chen et al.

(2016) studied the effects of flexible linkage on the dynamic behavior of the presse system with joint clearance into ADAMS. Although they conducted the investigation of experiment for the mechanical system, the mechanism considered in their study was with lower driving speed (less than 200 rpm). However, the driving speed in most mechanical systems is always high, so the dynamic model might not be suitable for a high speed press system. Then, it is essential to study the effects of clearance joint on the dynamic characterics for a high speed press system. Moreover, the repeatable precision of the lowest position for the main slider plays a crucial role on the design objective of the high speed press system. Therefore, it is essential to conduct the numerical and experimental studies of the planar multibody systems with joint clearance.

The primary purpose of this work is to investigate the dynamic characteristics of planar mechanical systems with joint clearance based on a continuous contact force model, which accounts for energy loss during the contact process. Moreover, we conduct the dynamic simulations of the planar multibody mechanical systems and compare the simulation result with experimental data. Research results demonstrate and validate the efficiency of the dynamics model. Furthermore, the numerical results of the planar mechanical systems with clearance are presented and discussed. The method and procedure adopted throughout this work are proposed and analyzed with the help of a numerical simulation of a planar multibody mechanical systems with clearance problem.

This paper is organized as follows. Section 2 defines the composition and principle of the mechanism with clearance for a high speed press. The contact force model and dynamics model of the mechanism are presented in Section 3. In Section 4 the experimental test rig is described and the parameters of experiment and calculation are given. Section 5 investigates the dynamic responce of planar multibody mechanical systems with joint clearance. Finally, the conclusions are given in Section 6.

2 DESCRIPTION OF A HIGH SPEED PRESS MECHANISM SYSTEM

Fig. 1 describes a typical working principle of the high speed press mechanism, which consists of driving device, transmission mechanism and base. When the mechanism operates, the movement of crank shaft is controlled by motor. Meanwhile, the reciprocating motion of main slider is driven by linkages. In order to balance the inertia force, the counterweight slider and additional linkage are designed at the opposite side of crank shaft. In addition, the processing of forging piece is conducted by the energy transferring (Chen et al., 2016). The constitution of the transmission mechanism for the closed high speed press is shown in Fig. 2(a), which is similar to a slider-crank mechanical system with balancing mechanism. In the model, the main linkage and main slider are connected by the revolute joint with clearance. Furthermore, the dynamic response of the slider-crank mechanism is affected by the revolute joint clearance.

3 DYNAMIC MODEL OF MECHANISM

3.1 Kinematic and Dynamic Analysis of Mechanism

A planar multibody mechanical system with joint clearance is shown in Fig. 2. Due to the existence of clearance, the linkage can move freely on a plane and the slider only moves in vertical direction

(Bai and Zhao, 2012). Then, it is necessary to study the influence of joint clearance on the dynamic characteristics of planar multibody system.

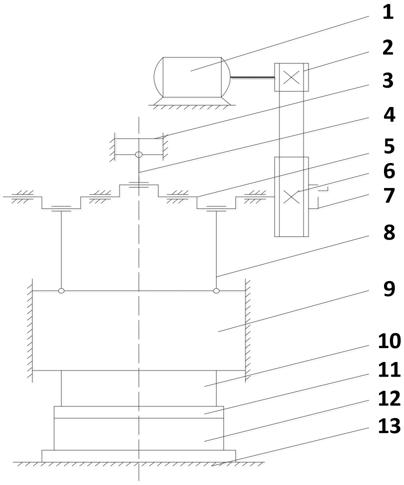


Figure 1: Schematic representation of press. 1—Motor; 2—Motor wheel; 3—Counterweight slider; 4—Additional linkage; 5—Crank shaft; 6—Fly wheel; 7—Clutch; 8—Main Linkage; 9—Slider; 10—Upper die; 11—Lower die; 12—Worktable; 13—Foundation.

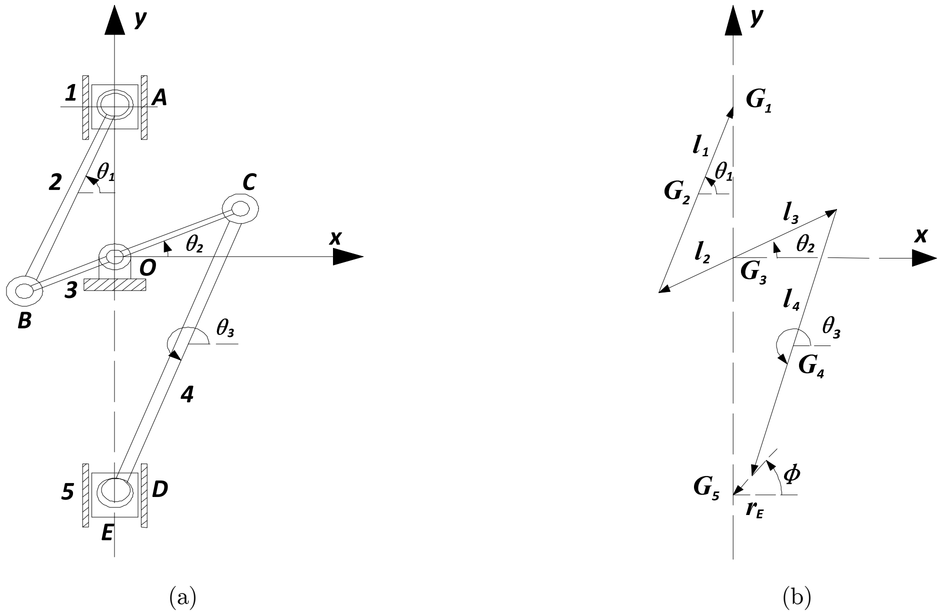


Figure 2: Slider-crank mechanism with revolute joint clearance: (a) schematic diagram; (b) vector diagram.

According to Fig. 2, G_1 is the mass center coordinate of the counterweight slider in the referential system, which is located in the mechanical system. G_2 and G_3 denote the coordinates of mass center for the additional linkage and crank, respectively. The mass center coordinates of main linkage and main slider are depicted by G_4 and G_5. x_{G_i} and y_{G_i} indicate the linear displacements of horizontal and vertical for the mass center in the mechanical system. The length of moving part and the size of clearance are defined by l_i and r_E, respectively. The position of mass center for moving part plays an important role, which is associated with velocity and acceleration (Erkaya, 2013). Therefore, the positions of mass centers are defined as follow:

$$\begin{Bmatrix} x_{G_1} \\ y_{G_1} \end{Bmatrix} = \begin{Bmatrix} l_1 cos(\theta_1) + l_2 cos(\pi + \theta_2) \\ l_1 sin(\theta_1) + l_2 sin(\pi + \theta_2) \end{Bmatrix} \tag{1}$$

$$\begin{Bmatrix} x_{G_2} \\ y_{G_2} \end{Bmatrix} = \begin{Bmatrix} l_{BG_2} cos(\theta_1) + l_2 cos(\pi + \theta_2) \\ l_{BG_2} sin(\theta_1) + l_2 sin(\pi + \theta_2) \end{Bmatrix} \tag{2}$$

$$\begin{Bmatrix} x_{G_3} \\ y_{G_3} \end{Bmatrix} = \begin{Bmatrix} l_{OG_3} cos(\theta_2) \\ l_{OG_3} sin(\theta_2) \end{Bmatrix} \tag{3}$$

In the same manner, the positions of mass center for the transmission mechanism are represented as:

$$\begin{Bmatrix} x_{G_4} \\ y_{G_4} \end{Bmatrix} = \begin{Bmatrix} l_3 cos(\theta_2) + l_{CG_4} cos(\theta_3) \\ l_3 sin(\theta_2) + l_{CG_4} sin(\theta_3) \end{Bmatrix} \tag{4}$$

$$\begin{Bmatrix} x_{G_5} \\ y_{G_5} \end{Bmatrix} = \begin{Bmatrix} l_3 cos(\theta_2) + l_4 cos(\theta_3) + r_E cos(\pi + \phi) \\ l_3 sin(\theta_2) + l_4 sin(\theta_3) + r_E sin(\pi + \phi) \end{Bmatrix} \tag{5}$$

where θ_1 denotes the angular position of additional linkage and θ_3 represents the angular position of main linkage. They are calculated as a function of θ_2 and ϕ in the following form:

$$\theta_1 = cos^{-1}\left(-\frac{l_2 cos(\pi + \theta_2)}{l_1}\right) \tag{6}$$

$$\theta_3 = cos^{-1}\left(-\frac{l_3 cos(\theta_2) + r_E cos(\pi + \phi)}{l_4}\right) \tag{7}$$

The velocities and accelerations of mass centers are given by the time-derivatives of position (Olyaei and Gahazavi, 2012).

$$\begin{Bmatrix} \dot{x}_{G_i} \\ \dot{y}_{G_i} \end{Bmatrix} = \begin{Bmatrix} \sum_{j=1}^{3} \dot{\theta}_j \frac{\partial x_{G_i}}{\partial \theta_j} + \dot{\phi} \frac{\partial x_{G_i}}{\partial \phi} \\ \sum_{j=1}^{3} \dot{\theta}_j \frac{\partial y_{G_i}}{\partial \theta_j} + \dot{\phi} \frac{\partial y_{G_i}}{\partial \phi} \end{Bmatrix} \tag{8}$$

$$\begin{Bmatrix} \ddot{x}_{G_i} \\ \ddot{y}_{G_i} \end{Bmatrix} = \begin{Bmatrix} \sum_{j=1}^{3} \ddot{\theta}_j \frac{\partial x_{G_i}}{\partial \theta_j} + \sum_{j=1}^{3} \dot{\theta}_j^2 \frac{\partial^2 x_{G_i}}{\partial \theta_j^2} + \ddot{\phi} \frac{\partial x_{G_i}}{\partial \phi} + \dot{\phi}^2 \frac{\partial^2 x_{G_i}}{\partial \phi^2} \\ \sum_{j=1}^{3} \ddot{\theta}_j \frac{\partial y_{G_i}}{\partial \theta_j} + \sum_{j=1}^{3} \dot{\theta}_j^2 \frac{\partial^2 y_{G_i}}{\partial \theta_j^2} + \ddot{\phi} \frac{\partial y_{G_i}}{\partial \phi} + \dot{\phi}^2 \frac{\partial^2 y_{G_i}}{\partial \phi^2} \end{Bmatrix} \tag{9}$$

where $\dot{\theta}_j$ is angular velocity of link and $\ddot{\theta}_j$ is the angular acceleration of link. i denotes the number of the moving links ($i = 1,2,3,4,5$) and j designates the number of the angular position (Tian et al., 2010). We can get the values of angular velocity and angular acceleration for linkages in the following equations:

$$\dot{\theta}_i = \dot{\theta}_2 \frac{\partial \theta_i}{\partial \theta_2} + \dot{\phi} \frac{\partial \theta_i}{\partial \phi} \quad (i = 1,3) \tag{10}$$

$$\ddot{\theta}_i = \ddot{\theta}_2 \frac{\partial \theta_i}{\partial \theta_2} + \dot{\theta}_2^2 \frac{\partial^2 \theta_i}{\partial \theta_2^2} + \ddot{\phi} \frac{\partial x_{G_i}}{\partial \phi} + \dot{\phi}^2 \frac{\partial^2 x_{G_i}}{\partial \phi^2} \quad (i = 1,3) \tag{11}$$

The contact force happens in the surface between journal and bearing, which causes the process of contact or impact (Flores and Ambrosio, 2004). In order to evaluate the dynamic characteristics of mechanism, the equations of motion are defined by the balancing of inertial forces and torques, as shown in Fig. 3. Thus, the equations of motion are expressed by:

$$\begin{Bmatrix} F_{(i-1)i_x} + F_{(i+1)i_x} \\ F_{(i-1)i_y} + F_{(i+1)i_y} \\ \sum M_{G_i} \end{Bmatrix} = \begin{Bmatrix} m_i \ddot{x}_{G_i} \\ m_i \ddot{y}_{G_i} \\ J_{G_i} \ddot{\theta}_{(i-1)} \end{Bmatrix} \tag{12}$$

where $F_{(i-1)i_x}$ and $F_{(i+1)i_x}$ denote the joint forces. M_{G_i} and J_{G_i} are the inertial moment and the mass moment of inertia. m_i is the mass of moving part.

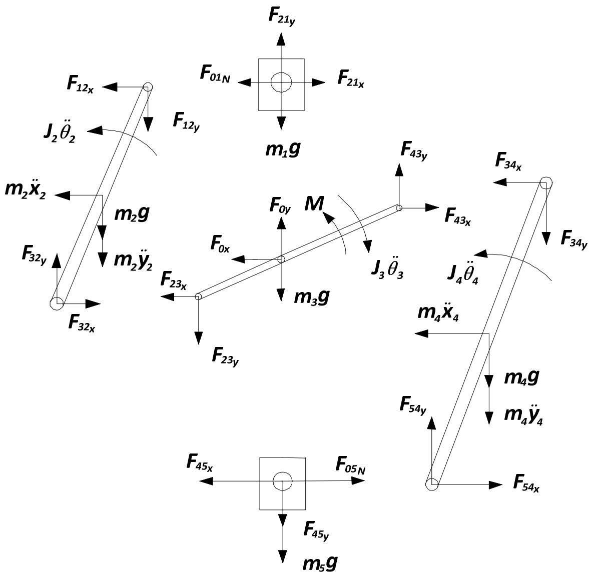


Figure 3: Force representation of transmission mechanism.

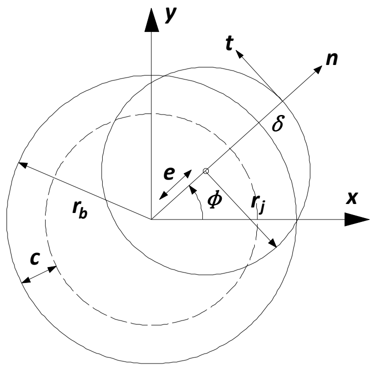


Figure 4: Schematic view of the contact.

3.2 Relative Motion Between Journal and Bearing

The motion equations are achieved in the section 3.1, which proposes the traditional model in multibody system with clearance. The contact process of journal and bearing can cause the contact-impact force, which can be obtained by normal and tangential contact forces. And relative position between the journal center and the bearing center is calculated by eccentricity vector, which can determine that the state of the bodies is contact or not. Therefore, the model of clearance joint can describe the influence of clearances effectively, which is built based on force interaction (Daniel and Cavalca, 2011). The components of the eccentricity vector can be given by:

$$\begin{Bmatrix} e_x \\ e_y \end{Bmatrix} = \begin{Bmatrix} ecos(\phi) \\ esin(\phi) \end{Bmatrix} \tag{13}$$

The magnitude of eccentricity vector can be defined by:

$$\mathrm{e} = \sqrt{e_x^2 + e_y^2} \tag{14}$$

As proposed in Fig. 4, the deformation caused by the impact between journal and bearing is computed according to:

$$\delta = e - c \tag{15}$$

Moreover, the eccentricity ratio is calculated as:

$$\varepsilon = \frac{e}{c} \tag{16}$$

According to angular position of journal inside bearing, the angle function can be written as:

$$\phi = tan^{-1}\left(\frac{e_y}{e_x}\right) \tag{17}$$

The derivate of angle with respect to time is evaluated as:

$$\dot{\phi} = \frac{e_x \dot{e}_y - e_y \dot{e}_x}{e^2} \tag{18}$$

In order to obtain the relative velocity of the bodies interact, it is necessary to compute the point along the eccentricity line during the process of contact occurs (Reis et al., 2014). The position of point can be expressed as:

$$\begin{Bmatrix} x_k^c \\ y_k^c \end{Bmatrix} = \begin{Bmatrix} x_k + r_k cos(\phi) \\ y_k + r_k sin(\phi) \end{Bmatrix} \qquad k = j, b \tag{19}$$

where r_k represents the radius of journal and the bearing. The relative velocity of the point is derived as:

$$\begin{Bmatrix} \dot{x}_k^c \\ \dot{y}_k^c \end{Bmatrix} = \begin{Bmatrix} \dot{x}_k - \dot{\phi} r_k cos(\phi) \\ \dot{y}_k + \dot{\phi} r_k sin(\phi) \end{Bmatrix} \qquad k = j, b \tag{20}$$

Furthermore, the relative scalar velocity of body is computed as:

$$\begin{Bmatrix} \dot{x}_c \\ \dot{y}_c \end{Bmatrix} = \begin{Bmatrix} \dot{x}_j - \dot{x}_i \\ \dot{y}_j - \dot{y}_i \end{Bmatrix} \tag{21}$$

And the relative normal and tangential velocities to plane of collision are defined as following (Flores and Ambrosio, 2004):

$$v_N = \dot{x}_c cos(\phi) + \dot{y}_c sin(\phi) \tag{22}$$

$$v_T = -\dot{x}_c sin(\phi) + \dot{y}_c cos(\phi) \tag{23}$$

3.3 Modeling of Contact Force

The contact force model plays an important role in the dynamic analysis of mechanical system during the contact-impact process. The well-known contact force model of Lankarani and Nikravesh is widely employed by elasticity contact theory, which takes into account the effects of elasticity and damping. Moreover, the influence of damping relates to the dissipated energy during the impact process, along with the dissipative effect correlated with the Coulomb friction on the contact surface (Wang et al., 2014). Thus, the contact between journal and bearing can be modeled by the L-N contact force model:

$$F_N = K\delta^n + D\dot{\delta} \tag{24}$$

where δ denotes the relative penetration and the exponent n equals to 1.5 for metallic contacts. The generalized parameter K depends on the material properties and radius. K can be calculated as (Rahmanian and Ghazavi, 2015):

$$K = \frac{4}{3\pi(h_1 + h_2)}\sqrt{\bar{R}} \tag{25}$$

where

$$\bar{R} = \frac{R_1 R_2}{R_1 + R_2} \quad , h_i = \frac{1 - \nu_i^2}{\pi E_i} \quad (i = 1,2) \tag{26}$$

The quantities R_i and h_i are the radius and the material parameter. ν_i and E_i are the Poisson's ratio and the Young's modulus of each sphere, respectively (Zhao and Bai, 2011). Furthermore, the hysteresis damping coefficient is defined as:

$$D = \frac{3K(1 - c_e^2)}{4\dot{\delta}^{(-)}} \delta^n \tag{27}$$

where $\dot{\delta}^{(-)}$ is the initial impact velocity and c_e denotes the restitution coefficient. Substituting the parameters of K and D from Eqs. (25), (26) and (27) into Eq. (24), the contact force can be given by:

$$F_N = K\delta^n \left[1 + \frac{3(1 - c_e^2)\dot{\delta}}{4\dot{\delta}^{(-)}}\right] \tag{28}$$

Furthermore, the friction force is always a complex phenomenon in the multibody systems, which has the influence on the efficiency and the motion of mechanism. In recent years, many researchers have established different friction models (Wang and Tian, 2016). As is well known, the friction force model employed in the numerical simulation is based on Coulomb's friction law, which can effectively describe the highly nonlinear phenomenon (Erkaya, 2012). It is calculated by:

$$F_T = -c_f c_d F_N \frac{\upsilon_T}{|\upsilon_T|} \tag{29}$$

where c_f represents the friction coefficient, F_N describes the normal force and υ_T denotes the relative tangential velocity. The dynamic correction coefficient c_d is written as:

$$c_d = \begin{cases} 0 & if \quad |\upsilon_T| \le \upsilon_0 \\ \dfrac{\upsilon_T - \upsilon_0}{\upsilon_1 - \upsilon_0} & if \quad \upsilon_0 \le |\upsilon_T| \le \upsilon_1 \\ 1 & if \quad |\upsilon_T| \ge \upsilon_1 \end{cases} \tag{30}$$

where υ_0 and υ_1 can be given tolerances for the tangential velocity of surfaces in contact (Flores, 2010).

4 EXPERIMENTAL STUDY

In this work, it is extremely difficult to measure the performance parameters such as contact force and penetration depth of journal bearing under conditions of different clearance sizes. Therefore, the experimental tests with the revolute joint clearance of 0.1 mm are conducted as a case study to verify the proposed method. The slider accelerations of the mechanical system with respect to various driving speeds are analyzed by the proposed computation method. If the numerical result is in accordance with that acquired by experimental test under the same operating conditions, the method will be validated.

Fig. 5 depicts that the experimental test rig is set up to obtain testing results of the multibody mechanical system at different driving speeds. The experiment consists of testing platform and testing system (Flores et al., 2011; Erkaya and Uzmay, 2010). The composition of testing platform is based on the transmission mechanism of a high speed press system, which includes motor, crank shaft, fly wheel, linkages, main slider and balancing mechanism. The main slider of the high speed press system, as shown in Fig. 1, is chosen as test object. Due to the acceleration cannot be showed directly, the data processing system is installed into the PC to facilitate the analysis of the testing data. Moreover, the acquisition and analysis systems are established into the BeeTech software, which can transform data at any time. In order to collect the acceleration data accurately, we calibrate the acceleration sensor before testing. Furthermore, the experiment is conducted under the unloading condition and two periods of acceleration are employed to comparisons between numerical results and experimental data, which is more effective to depict the validity of proposed methodology. The parameters of kinematic and dynamic for the planar multibody system are presented in Table 1.

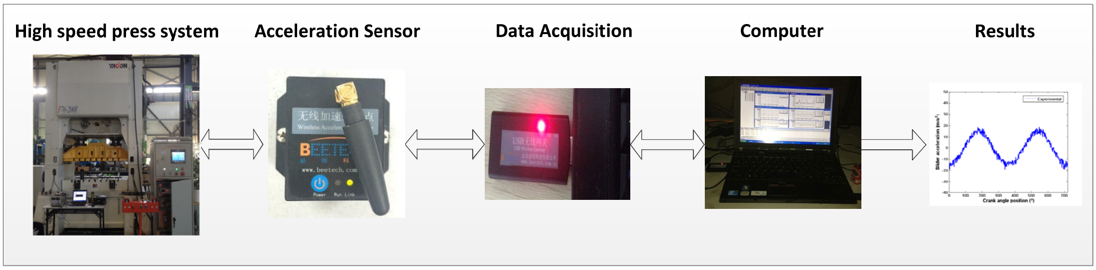


Figure 5: Block diagram for experimental measurement of main slider acceleration.

Simulations characteristics	value	Simulations characteristics	value
Young's modulus (GPa)	210	Length of main crank (m)	0.015
Density (kg/m^3)	7860	Length of main linkage (m)	0.350
Friction coefficient	0.01	Length of additional crank (m)	0.025
Poisson's ration	0.3	Length of additional linkage (m)	0.560
Restitution coefficient	0.9	Mass of main linkage (kg)	412.53
Radius of the solids in contact (mm)	70	Mass of main slider (kg)	2700.71
Clearance (mm)	0.1	Mass of additional linkage (kg)	79.50
Mass of crank (kg)	453.87	Mass of counterweight slider (kg)	1150.26

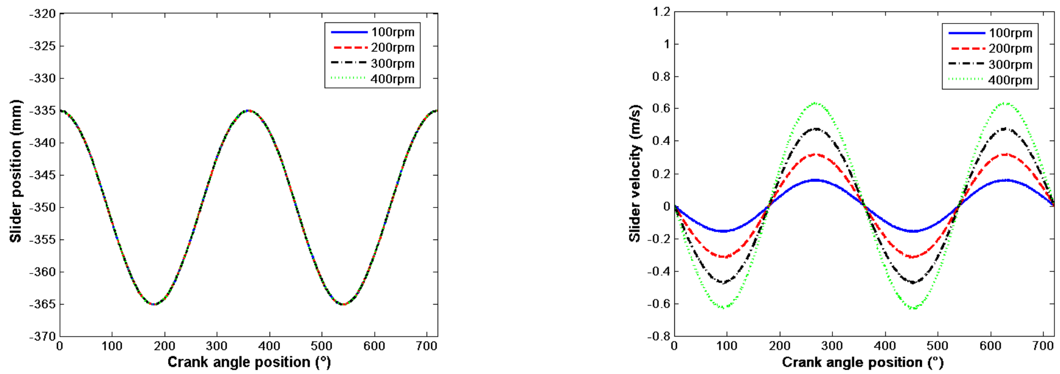
Table 1: Simulations characteristics.

5 RESULTS AND DISCUSSIONS

The purpose of this section is to study the effects of clearance joints on dynamic response of the mechanism in high speed press system. The planar mechanism is employed as simulation model to analyze the dynamic responce of planar multibody system with joint clearance. Then, the numerical results at different driving speeds are compared with that of the experimental data to demonstrate the computational efficiency of the presented method. In addition, the results of the numerical simulation with different clearance sizes are compared in the section. The results reveal the effects of these parameters on dynamic characteristics of a high speed press mechanical systems.

5.1 Simulation and Validation with Different Driving Speeds

To investigate dynamic behavior of the mechanism with clearance joint at different driving speeds, the model is selected as the case study, as shown in Fig. 2(a). The rotational speeds are 100 rpm, 200 rpm, 300 rpm and 400 rpm, respectively. The clearance size is 0.1 mm and the initial center positions of journal and bearing are coincident.



(a) (b)

Figure 6: Dynamic behavior of slider for different driving speeds: (a) Displacement of slider, (b) Velocity of slider.

Fig. 7 shows that the numerical results based on the dynamic model in this paper agree quite well with experimental data at different driving speeds, indicating that the presented dynamic model is an effective method to model planar multibody system with joint clearance. Then, the differences between the experiment and simulation for dynamic behavior plotted in Fig. 7 could be illustrated as follow: In the first place, the friction between the slider and guideway is ignored. In addition, the misalignment and lubrication between journal and bearing, which are always employed in actual mechanism system, cannot be considered in the simulation. In the formulation for the mathematic model of clearance joint, the assumption is made that the joint flexibility is neglected. Furthermore, the values of restitution and friction coefficient play the important roles in the numerical results. However, the values of the restitution and friction coefficient used in the work are based on the published data. The same issue occurs and has been observed by many other researchers (Bai and Zhao, 2012; Ma et al., 2015).

Fig. 7 also depicts the acceleration of the main slider with the clearance size of 0.1 mm at different driving speeds. Looking at the curves of slider acceleration, the peak value increases slowly at the lower speed. However, the results of Fig. 7(g) are compared with that of Fig. 7(a), which presents that the effects of the rotational speeds for the acceleration value increases scarcely at a higher driving speed. Moreover, it is clear from the figure that the fluctuation value of acceleration in Fig. 7(a) is lower than the values of others. This phenomenon appears due to the increase of contact velocity for journal and bearing and the continuous contact motion is the main period in motion period. Meanwhile, the results indicate that the oscillation phenomenon still exists in the operating time of mechanism system. Furthermore, the acceleration of the main slider can be changed by the existence of clearance.

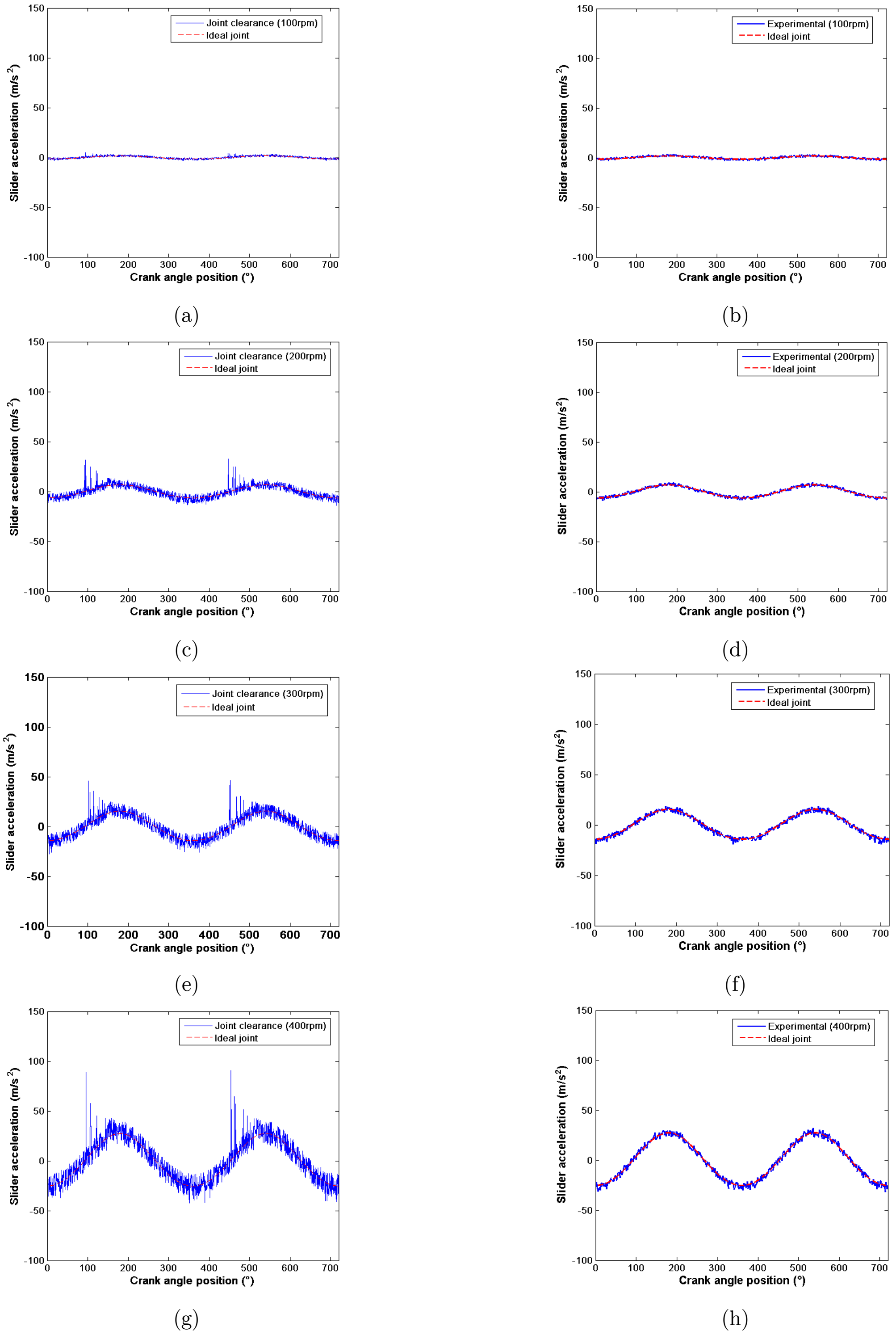


Figure 7: Numerical and experimental results of slider acceleration with the clearance size of 0.10 mm. The driving speeds are 100 rpm, 200 rpm, 300 rpm and 400 rpm, respectively. (a, c, e, g) are numerical simulation results and (b, d, f, h) are experimental results.

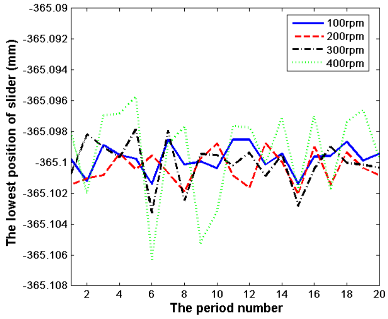
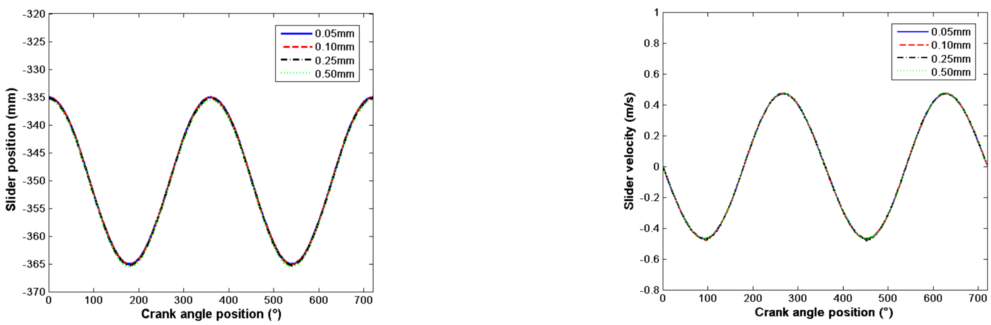


Figure 8: The repeatable precision of the minimum displacement with different driving speeds.

Fig. 8 presents the repeatable precision of the lowest position for the main slider at different driving speeds. When the driving speed is higher than 200 rpm in the calculation, the repeatable precision obviously drops, as shown in curves of the simulations. The value of the repeatable precision reaches the minimum at the angular velocity 400 rpm. It seems that the influence of the driving speed on the repeatable precision in the low-speed conditions become slighter due to the collision velocity of journal and bearing increases weakly. The value of fluctuation increases from 0.003 mm to 0.011 mm as the driving speed increases from 100 rpm to 400 rpm. The reason for this phenomenon may be that the deformation of bearing and contact-impact force increase with the increase of oscillation frequency. Although there is difference in the results of the repeatable precision, the deviation value of the lowest position for slider is smaller.

5.2 Influence of the Clearance Size

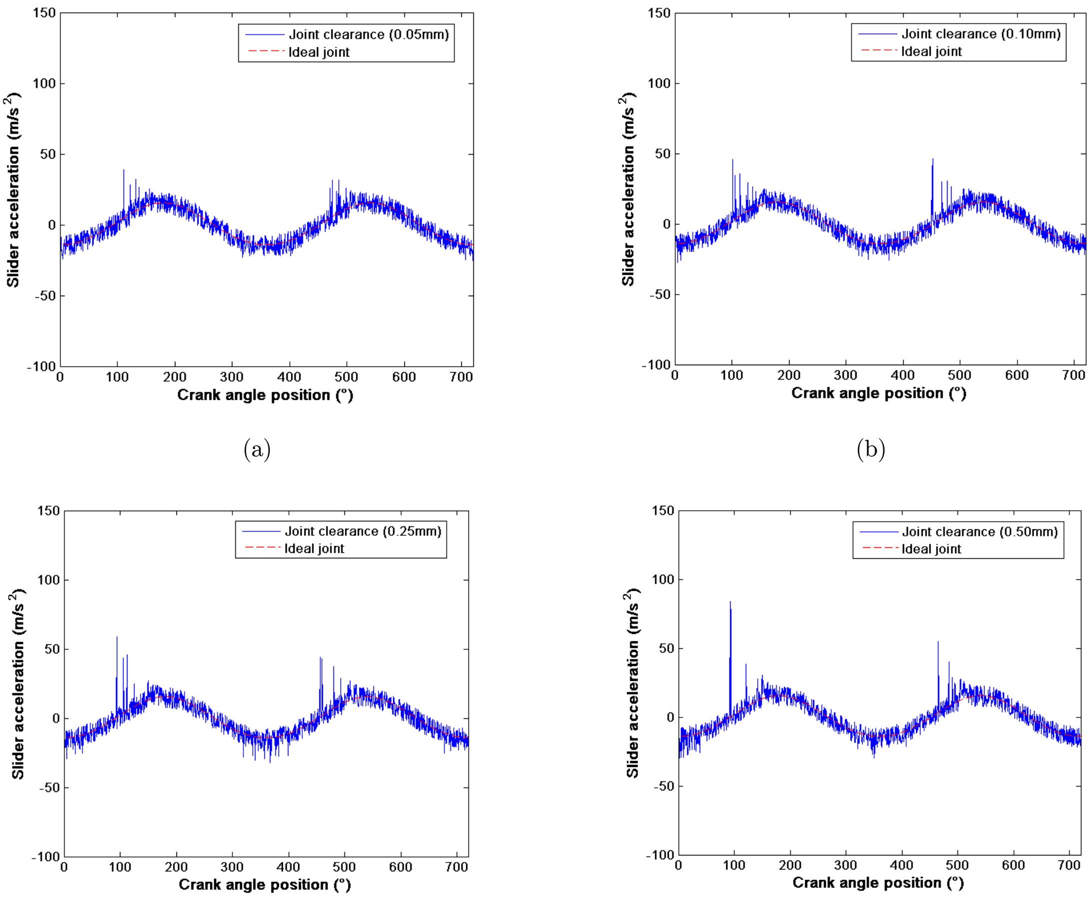
The dynamic analysis is always conducted based on the ideal joint at the constant running speed. In fact, the size of clearance joint slowly increases as long-term wear between journal and bearing. It means that the motion of planar multibody system with the various clearance sizes may provide significantly different dynamic responses. To investigate the influence of the clearance size on the dynamic performance of mechanism, the driving speed is 300 rpm. The values of revolute joint clearances are chosen to be 0.05 mm, 0.10 mm, 0.25 mm and 0.50 mm.



(a) (b)

Figure 9: Dynamic behavior of slider for different clearance sizes: (a) Displacement of slider, (b) Velocity of slider.

When the dynamic model is applied to the slider-crank model, the slider displacement and slider velocity are shown in Fig. 9. The results show that the displacement curves with different clearance sizes are similar. Then, the fluctuation value of velocity for the mechanism with the larger-size clearance is larger than others. When the clearance is 0.05 mm, the maximum deviation of velocity is 0.008 m/s. Then, the value is 0.012 m/s as the clearance increases to 0.50 mm. Meanwhile, the results also describe the peal value of velocity is increased from 0.476 m/s to 0.478 m/s. Furthermore, it is found that the smaller clearance size could smooth the dynamics curves. The reason for this phenomenon may be that the contact force of contact bodies decreases with the decline of clearance size.



(a) (b)

(c) (d)

Figure 10: Slider acceleration with different clearance sizes: (a) 0.05 mm, (b) 0.10 mm, (c) 0.25 mm, (d) 0.50 mm.

Fig. 10 describes how acceleration of the main slider is affected by clearance size. The numerical simulation results indicate that larger-size of clearance provides a higher acceleration. The maximum value of acceleration increases from 38.96 m/s to 83.53 m/s. If the size of clearance joint is increased to 1 mm, the difference may be more obvious. Therefore, the influence of joint clearance on dynamic response of mechanism cannot be neglected. Furthermore, the oscillation of acceleration and decrease of motion stability for mechanism happen due to the existence of clearance. The reason for this phenomenon is that the increase of clearance size causes the growth of contact force and the contact force is high-frequency oscillation as the same as acceleration. However, the oscillation frequency is weakened with increase of clearance size. The same results are proposed in reference (Flores, 2010).

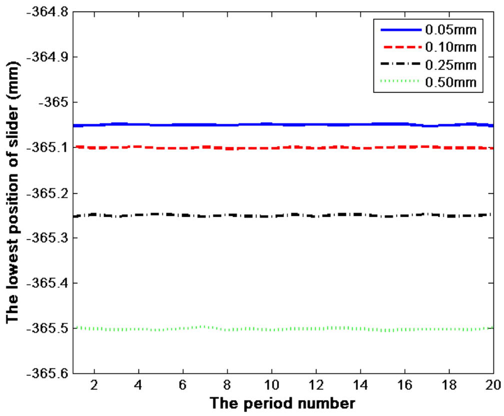


Figure 11: The repeatable precision of the minimum displacement with different clearance sizes.

Fig. 11 shows the repeatable precision of the lowest position for the mechanism with different clearance sizes. It is seen that the average position obviously declines with the increase of the clearance size. The results reveal that lowest position of slider for clearance size, comparing the value of the lowest position in Fig. 8, is more sensitive than the driving speed. Meanwhile, the smaller clearance can provide lower fluctuating value at the same rotational speed. When the clearance size is 0.05 mm, the fluctuating value of the lowest position of slider is 0.0045 mm. As the clearance size increases to 0.50 mm, the value is increased to 0.0047 mm. The increasing percentage of fluctuating value is 3.57%. This is a very interesting finding, which demonstrates that the clearance size effects the variation of the slider position.

6 CONCLUSIONS

A computational and experimental study is performed to study the influence of joint clearance on dynamic response of planar multibody systems. A general and comprehensive method for the dynamic modeling and analysis of mechanism with joint clearance is proposed in this work. With energy loss taken into account, the contact-impact model is employed by the Lankarani-Nikravesh contact force model to describe the contact response between journal and bearing. The numerical results based on the dynamic model agree quite well with experimental data, indicating that the proposed dynamics model can give an effective description of dynamic characteristic of mechanical system with joint clearance. In addition, the results highlight the influence of driving speed and clearance size on the mechanism performance.

The simulation result describes that the influence of clearance on dynamic response of mechanism cannot be neglected. The existence of clearance causes the obvious oscillation of slider position and slider acceleration. Meanwhile, the slider position of mechanism with clearance decline clearly and the amplitude increases as the increase of clearance size, which reveals that the existence of clearance will lead to shaky of the mechanism with high peaks. Moreover, the acceleration of mechanism with clearance is sensitive to variation of driving speed and clearance size. The numerical results could provide helpful suggestions for the optimization designing the kinematics and dynamics of planar multibody systems with joint clearance.

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