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A comprehensive method for joint wear prediction in planar mechanical systems with clearances considering complex contact conditions

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A comprehensive method to predict wear in planar mechanical systems with clearance joints is presented and discussed in this paper. This method consists of a system dynamic analysis and a joint wear prediction. As the size and shape of the clearance are dictated by wear and evolve with the dynamic response of the system, the contact between the journal and bearing could be conformal or non-conformal, which makes the contact conditions in clearance joints quite complicated. Therefore a modified contact force model is employed to evaluate the joint reaction force in this study. As the nonlinear stiffness coefficient is related to the physical and geometrical properties of contact bodies and varies with the deformation, this contact force model is applicable to different contact conditions between the journal and bearing. Furthermore, based on the Archard's wear model, the amount of wear can be quantified in the joint. And the geometry is updated to reflect the evolving contact boundary. Then, the wear process and the contact force model are integrated into the motion equations of the system to perform coupled iterative analyses between system dynamic response and joint wear prediction. In addition, a slider-crank mechanism is simulated as an example to demonstrate efficiency of the proposed method and to carry out a parametric study on mechanical systems considering joint wear. The influence of clearance size and driving power are discussed and compared respectively. The index of concordance is introduced to quantify contributions of contact pressure and sliding distance to wear rate under different types of journal motion. This study could help to predict joint wear in mechanical systems with clearances and optimize mechanisms in design.

wear, joint clearance, dynamics

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1 Introduction

In mechanical systems, components are interconnected by joints to allow relative motion. Due to manufacturing tolerance, assembly error and material deformation, clearances are inevitable in these joints which lead to the deviations of the system dynamic responses between the numerical predictions and the experimental measurements. There is a large amount of researches discussing the issue of the dynamic behavior of mechanical systems with clearance joints. A variety of problems have been considered: systems with rigid or flexible linkages [1,2], planar or spatial configurations [3,4], joint lubrication [5] and multiple clearances [6]. In these researches, joint wear is ignored.

Typically these joints consist of two components and they may be in contact and experience relative motion. It is therefore with no doubt that wear will occur on the contact surfaces which will further change the component profile. Depending on the operating conditions and wear amount at

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the joints, the performance of the system may also be adversely affected. Thus, a proper model for characterizing joint wear is necessary to quantify the loss of material from contact surfaces and to predict the system dynamic response. There have been a variety of papers involving in this issue. The finite element method which yields accurate results compared with experimental measurements has been widely used for modelling and simulating wear phenomenon [7]. Due to its computational complexity, this method is very time consuming. Tasora et al. [8] carried out an experiment to investigate wear in a revolute clearance joint. The experimental measurements agreed quite well with the theoretical results obtained using the Reye's hypothesis. They also pointed out that wear did not affect the entire surface of the shaft, but mostly happened on specific spots. Mukras et al. proposed a procedure to analyze planar multi-body systems in which wear was present at one or more revolute joints [9,10]. Besides, a comparison between elastic foundation model and contact force model in wear analysis was made to discuss the differences in predictions of wear depth and wear profile [11]. Li et al. [12] proposed an improved practical model for wear prediction of revolute clearance joints. In their study, the clearance was assumed as a massless link and the Archard's wear model was employed to estimate the wear depth in the joints. Flores et al. [13] developed a methodology for studying and quantifying the wear phenomenon in revolute clearance joints based on a continuous contact force model and the Archard's wear model.

It should be highlighted that dissimilar to traditional wear analyses in two contact bodies, joint wear prediction in mechanical systems with clearances has its own characteristics. On the one hand, the contact locations between the two components in a clearance joint are not uniform but concentrate on specific regions which are determined by the system response. Besides, as wear on the contact surfaces of the journal and bearing evolves, the system dynamic response may also change. And the joint reaction force may be altered, which probably influences the wear process as well. Thus, it is quite important to perform the joint wear analysis in the system dynamic framework. In other words, the wear estimation in clearance joint is closely associated with the entire system and it should be a system level prediction where the system dynamic analysis and the joint wear prediction are coupled. On the other hand, the size and shape of the clearance are dictated by wear. As the wear evolves, the component could not retain its original shape. The non-conformal contact may probably turn into a conformal contact and vice versa. The profiles of the two components are not regular and contact conditions between the journal and bearing become quite complicated. Therefore, a contact force model which has a wide scope is needed to predict the joint reaction force and wear depth accurately. It is intuitive that the consideration of the two aspects listed above will promote the precision of wear prediction in planar mechanisms with clearances. However, the two characteristics have been scarcely combined comprehensively in previous researches.

In this paper, a comprehensive method to predict wear in mechanical systems with clearance joints is presented. The analyses of the dynamic response and the wear process are considered simultaneously. Moreover, a modified contact force model is adopted to calculate the joint reaction force which could be applied to the complicated contact conditions in clearance joints. This contact force model and the wear process are then integrated into the motion equations of the system to predict wear in a system level. In addition, a parametric study on joint wear in mechanical systems is performed to discuss the effect of operating conditions and system configurations on the wear phenomenon which helps to reduce wear in mechanical design.

2 Modeling of revolute clearance joints

In ideal mechanical systems, it is assumed that the connecting points of two bodies linked by a revolute joint are coincident. When a clearance is introduced into a revolute joint, however, the two points are separated and the journal is constrained to move within the bearing boundary. Figure 1 depicts a revolute joint with clearance, where the difference in radii between the bearing and the journal defines the radial clearance [14]. When the journal and bearing are in contact and experiencing relative sliding, a contact force

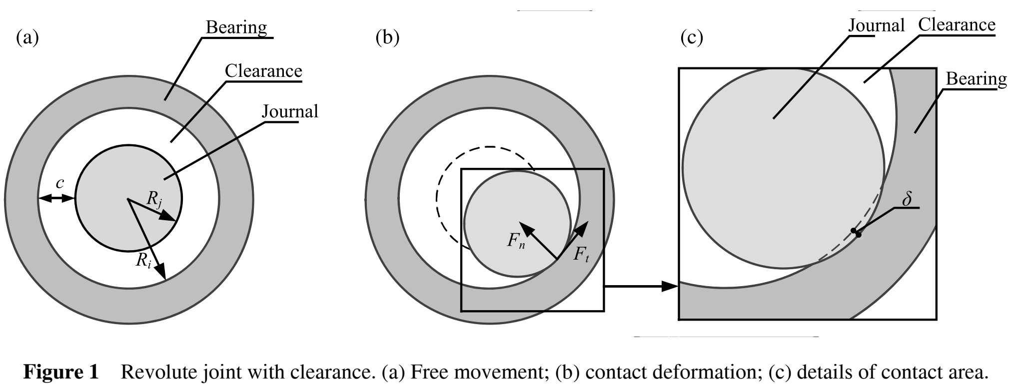

Figure 1 Revolute joint with clearance. (a) Free movement; (b) contact deformation; (c) details of contact area.

and a friction force will be applied to the contact plane which are defined as force constraints.

The contact force model describing the impact-contact phenomenon is one of the important contents of mechanical systems with clearance joints. The use of a proper model directly affects the precision of the simulation results. Several expressions have been proposed to model the impact-contact phenomenon between the journal and bearing where the Hertz contact model has been used in a number of researches. However, it may lead to erroneous results when the conformity is large [15]. And as stated before, due to the complicated contact conditions in clearance joints, the contact force model for wear prediction should be applicable to both the conformal and non-conformal contacts between the journal and bearing. Thus, in this paper, a modified contact force model is employed to calculate the contact force in a revolute clearance joint for wear prediction. This model is based on the Lankarani-Nikravesh model and the improved elastic foundation model which extends the scope of previous approaches and is applicable to the complicated contact conditions. Detailed discussions and comparisons with experimental results of this model are available in refs. [15,16].

Using this model, the force normal to the contact plane can be expressed as

$$\begin{cases} F_n = K\delta^n + D\dot{\delta}, & \delta \geqslant 0, \\ F_n = 0, & \delta < 0, \end{cases} \tag{1}$$

where $K\delta^n$ represents the elastic force term, $D\dot{\delta}$ accounts for the energy dissipation term, δ is the relative penetration depth, $\dot{\delta}$ is the normal impact velocity, D represents the damping coefficient, and K is the nonlinear stiffness coefficient which can be evaluated as

$$K = \frac{1}{8}\pi E^* \sqrt{\frac{2\delta\left(3\left(R_i - R_j\right) + 2\delta\right)^2}{\left(R_i - R_j + \delta\right)^3}}, \tag{2}$$

where E^* is compound elastic modulus and R_i, R_j represent the bearing and journal radii respectively. It is obvious that the nonlinear coefficient is related to the material property, geometry of the components, clearance size and deformation of the contact bodies. Furthermore, the coefficient is no longer a constant but varies with the relative penetration depth to reflect the contact process. The damping coefficient is expressed as

$$D = \frac{3K\left(1 - c_e^2\right)\mathrm{e}^{2(1-c_e)}\delta^n}{4\dot{\delta}^{(-)}}, \tag{3}$$

where c_e is the coefficient of restitution and $\dot{\delta}^{(-)}$ is the normal relative velocity before impact.

This modified contact force model is applicable to both the conformal and non-conformal contacts between the journal and bearing in wear prediction. It is also suitable for different coefficients of restitution. Moreover, the numerical results were compared with the experimental ones for the dynamic response of a mechanical system with a clearance joint, which shows good calculation precision of the model [16]. Therefore, the nonlinear contact force model extends the application scope and overcomes the disadvantages of the previous models. It is quite precise for wear predictions in revolute joints with clearances in mechanical systems.

The tangential contact characteristic of clearance is represented using tangential friction force model where the Coulomb's friction law is widely used. However, this law does not take parameters such as relative tangential sliding velocity and material properties into consideration. And the use of Coulomb's friction law may lead to numerical difficulties for its strong nonlinear properties during the transition from sticking to sliding or vice versa. In order to avoid the sudden change of the tangential friction force, the relation between the relative tangential velocity and the friction coefficient can be modified into [17]

$$\mu = \begin{cases} \mu_d, & |V| > V_d, \\ \mu_s \sin\left(\dfrac{\pi}{2}\dfrac{|V|}{V_s}\right), & |V| < V_s, \\ \dfrac{\mu_s + \mu_d}{2} + \dfrac{1}{2}\left[(\mu_s - \mu_d)\cos\left(\pi\dfrac{|V| - V_s}{V_d - V_s}\right)\right], & V_s \leqslant |V| \leqslant V_d, \end{cases} \tag{4}$$

where V, V_s and V_d are the relative tangential velocity, stick-slip switch velocity and static-sliding friction switch velocity respectively. μ_s and μ_d are the static and sliding friction coefficients.

3 Wear model for revolute clearance joints

When the two components in a revolute clearance joint are in contact and experience relative tangential motion, wear would occur which directly leads to progressive loss of material from the surfaces of the components. Most of previous approaches for wear prediction are based on the Archard's wear model. This model correlates wear volume with physical and geometrical properties of contact bodies such as the applied load, sliding distance and hardness of the material. Moreover, it has been successively applied to the wear prediction for cam and follower components [18,19], revolute joint components [20] and helical gears [21]. In this paper, the Archard's wear model is used for the wear prediction in a revolute clearance joint in mechanical systems. The Archard's wear model is expressed as [22]

$$\frac{V_w}{s} = \frac{kF_n}{H}, \tag{5}$$

where V_w is the wear volume, s is the relative sliding distance, k is the dimensionless wear coefficient, F_n represents the normal contact force and H is the hardness of the softer material.

For wear estimation in a revolute clearance joint, wear depth is more convenient than wear volume to show the effect of wear on the dimensions of the components. Thus, assuming that the deformation of asperities on the contact surface is plastic, and the actual contact area is proportional to the normal contact force, eq. (5) can be re-written as

$$\frac{V_w}{s} = kA_a = \frac{kF_n}{H}, \tag{6}$$

where A_a is the actual contact area. Dividing eq. (6) by A_a yields

$$\frac{h}{s} = \frac{kP}{H}, \tag{7}$$

where h represents the wear depth and P is the contact pressure. It can be concluded that the wear depth is closely related to the distribution of the contact pressure and the relative sliding distance. For the application of this model in an iterative procedure, its differential form can be expressed as

$$\frac{\mathrm{d}h}{\mathrm{d}s} = \frac{kP}{H}. \tag{8}$$

The wear depth can be obtained by using a forward finite difference approach. The updating wear depth can be evaluated as

$$h_i = h_{i-1} + kP_i \Delta s_i / H, \tag{9}$$

where h_i, P_i and Δs_i are the wear depth, the contact pressure and the relative sliding distance at the ith cycle respectively, and h_{i-1} is the wear depth at the $(i-1)$th cycle. Thus, the wear depth at every time step can be calculated and the final wear depth equals to the depth accumulated.

4 Dynamic analysis of mechanical systems with clearance joints

In order to predict joint wear in a mechanical system correctly, it is quite necessary to perform dynamic analysis of the mechanical system with clearance joints. As the wear at the contact interface evolves, the joint forces will be changed, which directly leads to the variation of the dynamic response of the system. Moreover, in wear prediction at a system level, it may not be possible to determine the correct contact locations without performing a dynamic analysis.

Because kinematic constraints are substituted by the contact force constraints in the clearance joints, the analysis of mechanical systems with clearance joints involves a set of differential and algebraic equations which are a combination of the motion equations and the kinematic constraint equations. The motion equations for a mechanical system with clearance joints can be expressed as

$$\begin{cases} \boldsymbol{M}\ddot{\boldsymbol{q}} + \boldsymbol{C}\dot{\boldsymbol{q}} + \boldsymbol{K}_g \boldsymbol{q} + \boldsymbol{\Phi}_q^{\mathrm{T}} \boldsymbol{\lambda} = \boldsymbol{F} + \boldsymbol{F}_c, \\ \Phi(\boldsymbol{q}, t) = 0, \end{cases} \tag{10}$$

where $\boldsymbol{M}$, $\boldsymbol{C}$ and $\boldsymbol{K_g}$ represent the system mass matrix, generalized damping matrix and generalized stiffness matrix respectively. The vector $\boldsymbol{q}$ contains the displacements, and $\boldsymbol{\Phi_q}$ is the Jacobian matrix of constraint equation $\boldsymbol{\Phi}(\boldsymbol{q}, t)$=0. The vectors $\boldsymbol{\lambda}$ and $\boldsymbol{F}$ are the Lagrange multipliers and generalized forces respectively. It should be noted that the generalized forces of the system excluding the generalized contact forces in clearance joints are defined as $\boldsymbol{F}$. Meanwhile, the generalized contact forces corresponding to the contact between the journal and bearing are defined as $\boldsymbol{F_c}$ which manifest the effect of the clearance and can be expressed as

$$\boldsymbol{F}_c = u(\delta)(\boldsymbol{F}_n + \boldsymbol{F}_t), \tag{11}$$

where $\boldsymbol{F_n}$ and $\boldsymbol{F_t}$ are the normal contact force vector and tangential friction force vector respectively, and $u(\delta)$ is introduced as a unit step function. When an impact or a contact occurs between the journal and bearing, it could automatically impose force constraints on the motion of the system. The expression of the unit step function is expressed as

$$u(\delta) = \begin{cases} 0, & \delta < 0, \\ 1, & \delta \geqslant 0, \end{cases} \tag{12}$$

where the relative penetration depth δ determines the value of the contact force.

5 A comprehensive method for wear prediction in planar mechanical systems with clearances

In this section, a comprehensive method is presented to predict joint wear in mechanical systems with clearance joints. This comprehensive method could reflect the mutual effect of the wear process and the dynamic response. In this procedure, the wear regions will be determined by the contact and relative sliding between the journal and bearing. Then, the contact boundary in the normal direction of the contact surface will be altered by the wear depth estimated. Once the geometry updated, the dynamic response of the system will be influenced and result in the change of the wear regions subsequently.

Furthermore, in order to determine profiles of the journal and bearing at every time step with low computational cost, the surface is divided into several sectors. The wear depth in each sector could be evaluated by computing the sliding distance and contact pressure. Then extrapolation is employed to update the profile at every time step.

The steps of the comprehensive method for wear prediction in planar mechanical systems with clearances are shown as follows.

(1) Define the initial configurations of the system and the initial conditions to start the simulation.

(2) Define the parameters associated with the joint such as the journal and bearing radii, the hardness of the material. Start the simulation at the time t_0.

(3) Check the relative motion of the two components in the joint. If they come into contact, evaluate the normal and tangential forces given in eqs. (1) and (4) respectively. Otherwise, proceed with the motion equations of the system given in eq. (10) straightly.

(4) Compute the current and accumulated wear depth given in eqs. (8) and (9) to update the component profile. Then, proceed with the motion equations of the system in eq. (10).

(5) Obtain new positions and velocities of the system.

(6) Update the time. Go to step (3) with a totally new set of parameters of the joint obtained previously until the simulation is completed.

6 Case study

In this section, a planar slider-crank mechanism is performed as an example to illustrate the method presented before and to carry out a parametric study on the mechanical system with joint wear under different operating conditions namely the clearance size and the driving power.

Figure 2 shows the configuration of the selected slider-crank mechanism. The mechanism consists of two ideal revolute joints, one between the crank and the ground, and the other between the crank and the connecting rod, an ideal transitional joint between the slider and the ground, and a clearance joint between the connecting rod and the slider. The geometric and inertial properties of the slider-crank mechanism are listed in Table 1. In Table 2, the material

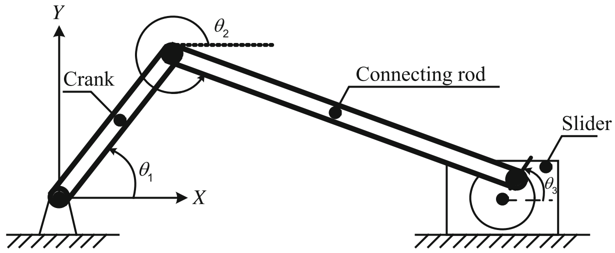

Figure 2 A slider-crank mechanism with a clearance revolute joint.

properties and dimensions for the joint components are provided. It should be noted that a smaller value for the wear coefficient is adopted to accelerate the wear for simulations purposes which is widely used in related researches [13]. Furthermore it is assumed that the journal is much harder than the bearing so that no appreciable wear occurs on its surface and, as a result, the journal always retains its original shape.

In the dynamic simulation, the gravity is taken as in the negative Y direction and the motion of the system is defined in a vertical plane. The initial conditions used are based on the results of a kinematic analysis of the slider-crank mechanism in which all the joints are assumed to be ideal.

6.1 General characteristics of joint wear in the slider-crank mechanism

In this section, the radial clearance size of the mechanism is set to be 0.02 mm, and the crank rotates at a constant angular velocity of 2000 r/min. This mechanism is defined as a standard slider-crank mechanism in order to make comparisons with following mechanisms under different operating conditions. Representative results of the dynamic analysis and the wear prediction are shown in Figures 3–7.

In Figures 3 and 4, the dynamic response of the standard mechanism is obtained for two full crank rotations after steady-state has been reached. It is obvious that the dynamic behavior of this system tends to be non-linear. This idea is supported by the high frequency oscillation in the slider acceleration curve. The oscillation is directly related to the

Table 1 Geometric and inertial properties of the slider-crank mechanism

No.	Body	Length (m)	Mass (kg)	Moment of inertia (kg m^2)
1	Crank	0.05	0.30	0.00025
2	Connecting rod	0.12	0.21	0.00100
3	Slider	-	0.14	0.00010

Table 2 Parameters used in the dynamic simulation

Parameter	Value	Parameter	Value
Journal radius	9.8 mm	Sliding friction coefficient	0.03
Coefficient of restitution	0.9	Young's modulus	207GPa
Stick-slip switch velocity	0.001 m/s	Poisson's ratio	0.3
Static-sliding friction switch velocity	0.0015 m/s	Dimensionless wear coefficient	1.736×10^{-4}
Static friction coefficient	0.04	Hardness of the material	2.17×10^{9} Pa

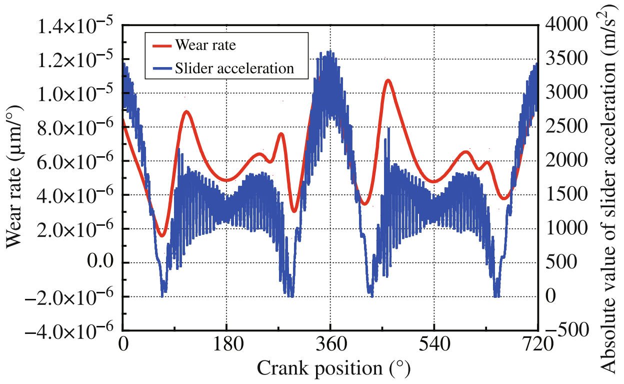

Figure 3 (Color online) Slider acceleration versus wear rate (0.02 mm/2000 r/min).

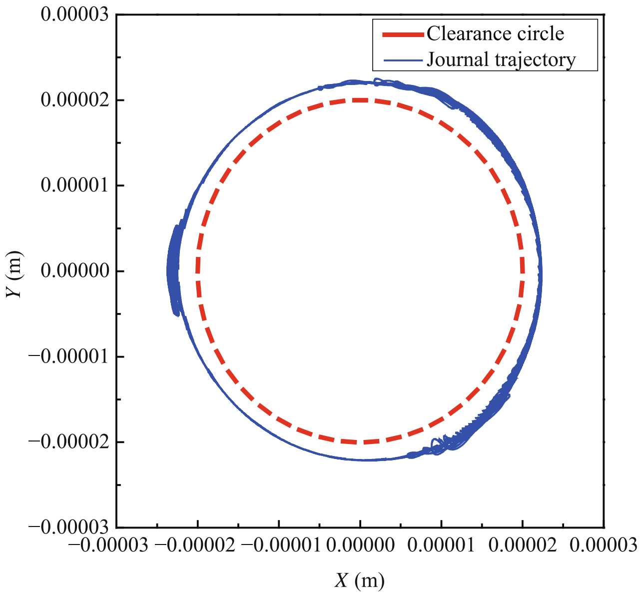

Figure 4 (Color online) Journal trajectory (0.02 mm/2000 r/min).

contact between the two components of the clearance joint and is propagated throughout the mechanism which will lead to vibration of the system. Furthermore, the dynamic response of the system repeats itself from cycle to cycle. This behavior is expected since the journal and bearing are in continuous contact shown in Figure 4 where the free flight and impact motion of the journal do not exist. It also seems that the entire surface of the bearing experiences continuous contact. So it is hard to determine the wear region directly from the dynamic analysis.

In Figure 3 the wear rate is illustrated against the slider acceleration. As the acceleration is closely related to the joint reaction force, this plot reveals that the variation of the wear rate is in accordance with that of the contact force. The effect of two factors namely the contact pressure and sliding distance on the wear phenomenon will be discussed in detail in the final section.

In addition, the variation of the bearing radius for 40 full crank rotations is shown in Figure 5. It is obvious that the

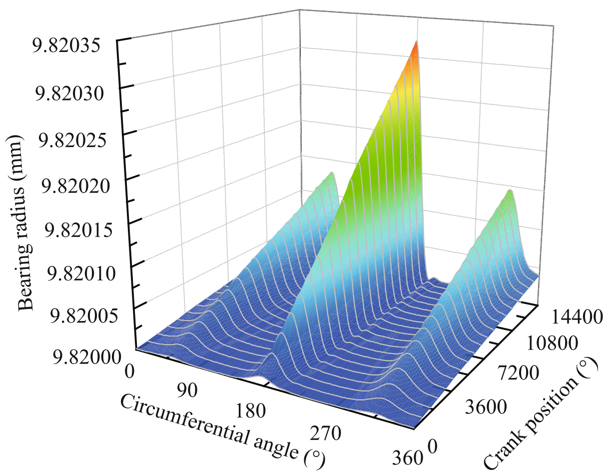

Figure 5 (Color online) Variation of bearing radius for 40 cycles (0.002 mm/2000 r/min).

wear does not affect the entire bearing surface in the same manner but concentrates on three specific regions especially around 180°. And the wear rates in the three severe wear regions are substantially different. In order to reveal the discrepancies, the variation of the bearing radius for 10, 20, 30 and 40 full crank rotations is plotted against the circumferential angle in Figure 6. Three severe wear regions can easily be found around 50°, 180°and 320° with peaks of the bearing radius, demonstrating that the clearance of the joint is no longer regular after wear. These three severe wear regions also correspond to the thick part of the journal trajectory in Figure 4. It also can be seen that initially the difference of the bearing radius is not distinct. However, as the wear evolves, the wear rate around 180° becomes drastically larger compared with other regions. In Figure 7, the radii of the three regions are expanded against the crank position. The contact location alternates among the three regions which is indicated by the staircase like shape for the radius curve. And the wear rate around 180° is obviously larger. The wear pattern presented here also coincides with the

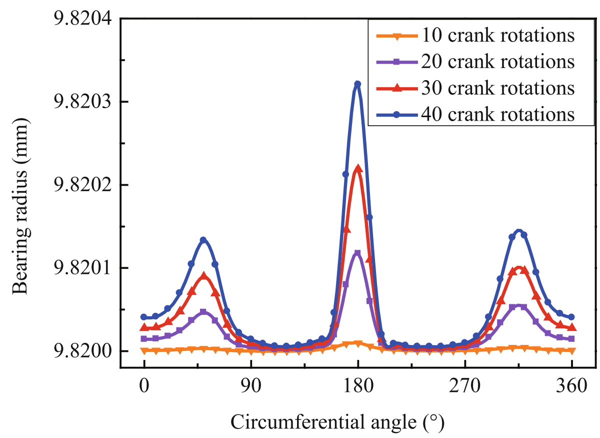

Figure 6 (Color online) Variation of bearing radius with crank rotation (0.002 mm/2000 r/min).

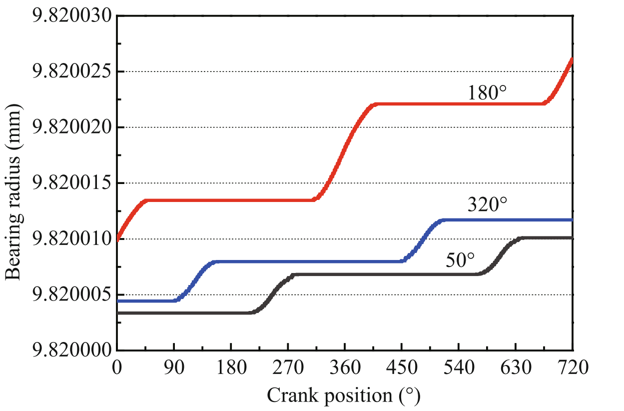

Figure 7 (Color online) Variation of bearing radius in specific regions (0.002 mm/2000 r/min).

conclusion drawn before.

From the analysis above, the general characteristics of joint wear in the slider-crank mechanism can be obtained that wear is not uniform but concentrates on some specific regions. The distribution is based on the frequency of contact in different ranges and is closely related to the dynamic response of the system. Moreover, wear rate also varies along the surface which will further change the profile of the bearing. Thus, performing dynamic analyses is critical for the prediction of correct contact locations and wear regions in mechanical systems with clearances for joint wear predictions.

6.2 Effect of clearance size on joint wear

In this section, the effect of clearance size on joint wear in the slider-crank mechanism will be investigated. The radial clearance sizes are altered to be 0.2 and 0.5 mm respectively, and the crank speed remains 2000 r/min. Then, comparisons of dynamic response and wear prediction will be made among the mechanisms with the new parameters and the standard mechanism discussed before. Representative results are illustrated in Figures 8–11 which are obtained for two full crank rotations after the steady state has been reached.

The journal trajectories for different radial clearance sizes are illustrated in Figure 8. Different from the journal trajectory shown in Figure 4, due to the increase of the clearance size, three types of journal motion are captured, namely, the free flight mode, the impact mode and the continuous contact mode. The variation of the journal motion directly leads to the change of the dynamic response of the system. This conclusion can be proved by the slider accelerations shown in Figure 9, a large number of acceleration peaks in the curve are associated with the impacts between the journal and bearing which is substantially different from the slider acceleration illustrated in Figure 3.

Furthermore, it is worth noticing that different clearance sizes imply the use of different scales for journal trajectories plotted. Comparing the journal trajectories in Figures 4 and 8, it can be concluded that the relative penetration depth between the two components in the joint becomes deeper with the increase of the clearance size which leads to noticeable larger slider accelerations in Figures 3 and 9. This is because a larger clearance gives the journal more free flight time, thus more energy is accumulated and finally results in a deeper penetration. As explained before, the acceleration is closely related to the joint reaction force, thus the magnitude of the wear rate also becomes larger with the increase of clearance size shown in Figure 9. The wear processes are illustrated in Figure 10. It is clear that the regions experiencing wear are similar but the wear rate

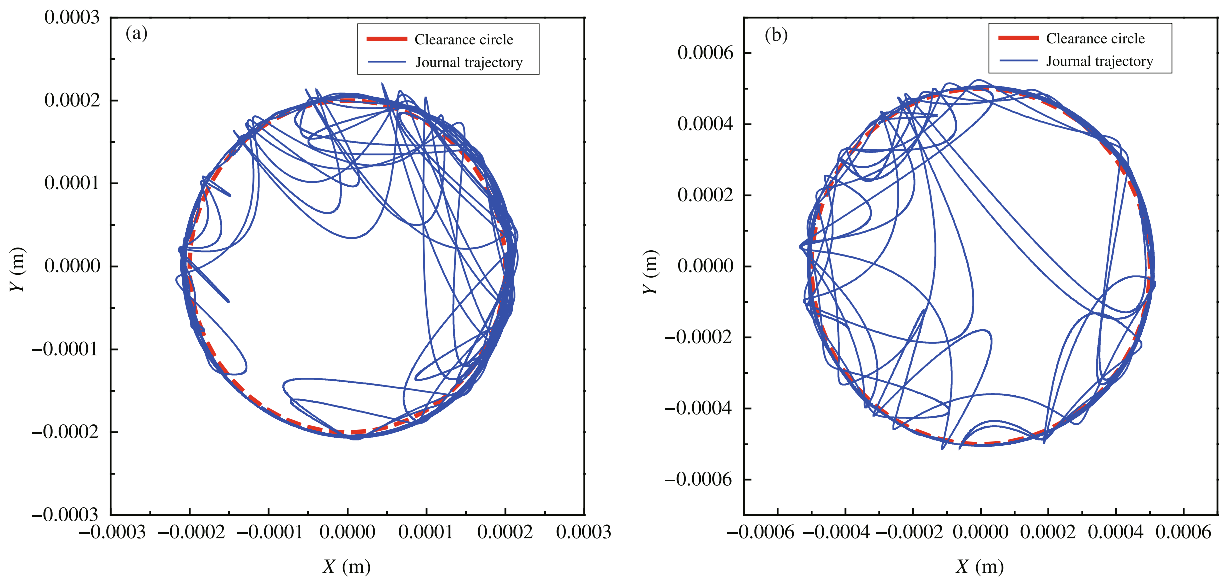

Figure 8 (Color online) Journal trajectory for different radial clearance size. (a) 0.2 mm; (b) 0.5 mm.

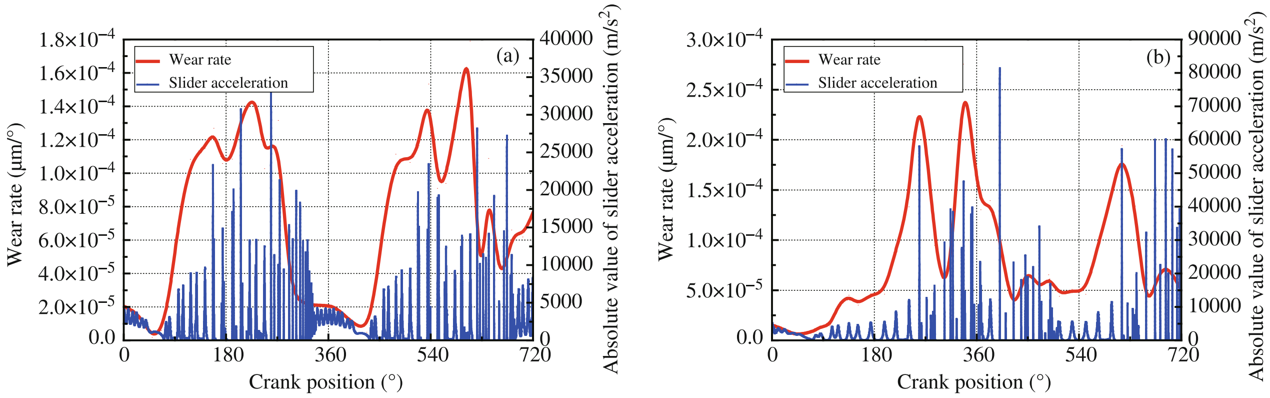

Figure 9 (Color online) Slider acceleration versus wear rate for different radial clearance size. (a) 0.2 mm; (b) 0.5 mm.

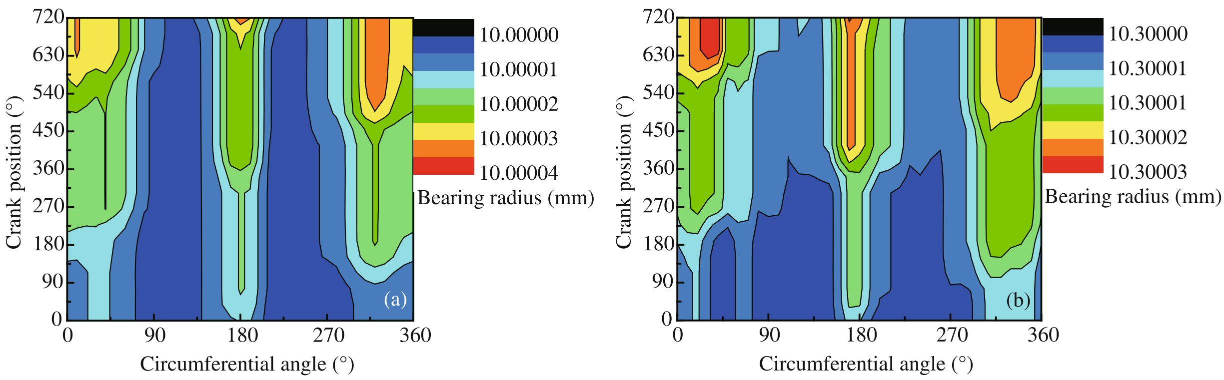

Figure 10 (Color online) Variation of bearing radius for different radial clearance size. (a) 0.2 mm; (b) 0.5 mm.

differs from each other. A larger clearance size could accelerate the wear process. A comparison of wear depth accumulated among different mechanisms is illustrated in Figure 11. From the plotted results, two conclusions can be drawn. Firstly, with the increase of the clearance size, the differences of wear depths between severe wear regions become smaller, which means a relatively large clearance could produce a more uniform distribution of wear. Secondly, as stated before, a larger clearance corresponds to a much larger joint reaction force, however, the wear depth may not be affected seriously in this case. This is because although the magnitude of contact force is much higher with a larger clearance, the duration of contact between the journal and bearing is quite short. Thus, the incremental sliding distance is also small. Since the contact pressure and sliding distance are two factors to determine the joint wear, the wear depth will not be affected seriously. This reason can also be proved by the slider acceleration presented in Figure 9. With increasing of the clearance size, it takes a longer time for the journal to move between two impacts, so less collisions could occur between the journal and bearing which is demonstrated by the number of peaks in the acceleration curve.

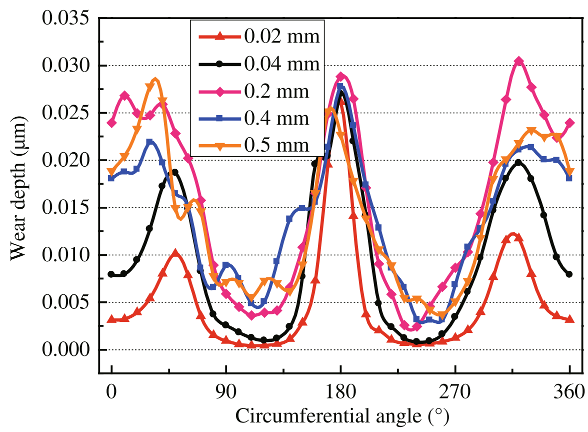

Figure 11 (Color online) Wear depth accumulated for different clearance size.

6.3 Effect of driving power on joint wear

In this section, influence of driving power on joint wear in the slider-crank mechanism will be studied. In order to make comparisons with the standard mechanism analyzed before, the radial clearance size remains to be 0.02 mm, but the crank speed is altered to be 30 and 600 r/min respectively. The results are shown through Figures 12–15 which are obtained for two full crank rotations after the steady state has been reached.

The journal trajectory inside the bearing is shown in

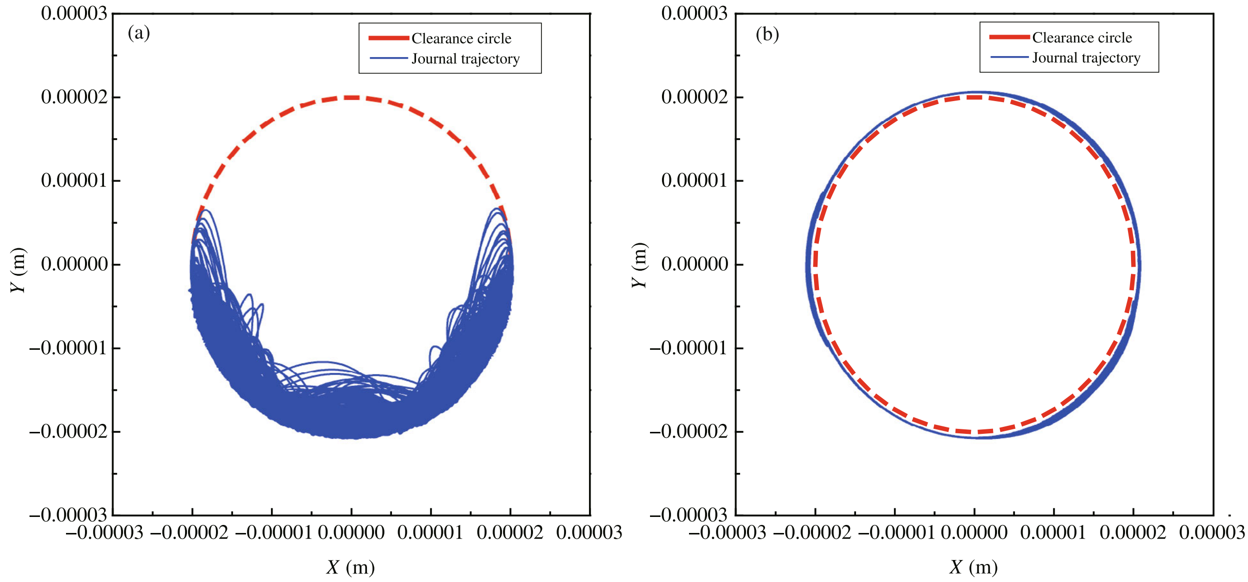

Figure 12 (Color online) Journal trajectory for different crank speed. (a) 30 r/min; (b) 600 r/min.

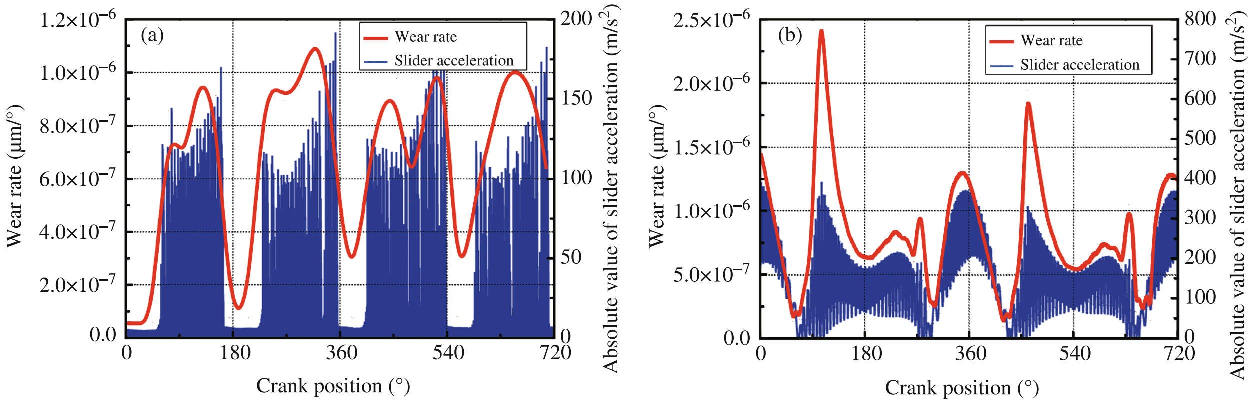

Figure 13 (Color online) Slider acceleration versus wear rate for different crank speed. (a) 30 r/min; (b) 600 r/min.

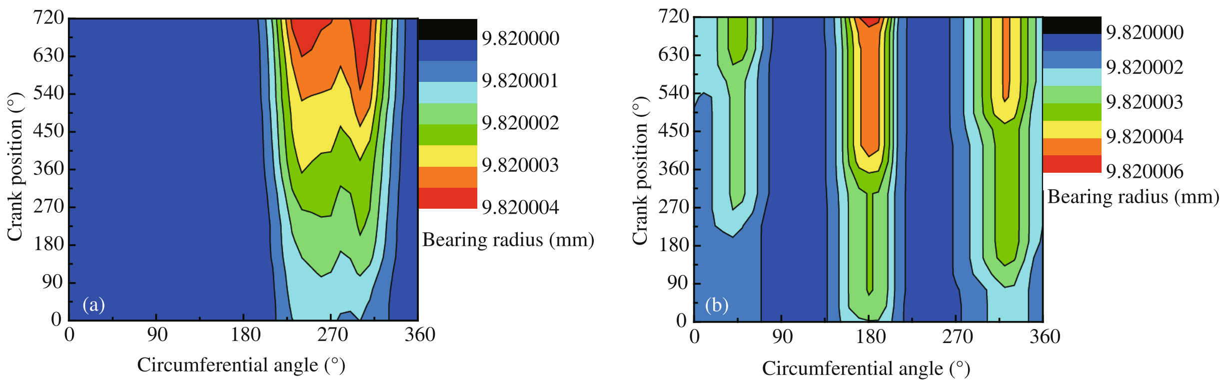

Figure 14 (Color online) Variation of bearing radius for different crank speed. (a) 30 r/min; (b) 600 r/min.

Figure 12. When the crank speed is 30 r/min, the trajectory exhibits complex profiles, in which rebounds often take place after impacts instead of continuous contact. Moreover, it is obvious that the contact locations concentrate on the bottom of the bearing. In contrast, the journal is in permanent contact with the bearing wall and the whole bearing surface experiences contact under the crank speed of 600 r/min, which is similar to the case of 2000 r/min. It can be reasonably inferred that the wear regions are different in these cases due to the variation of the contact locations. Furthermore, the

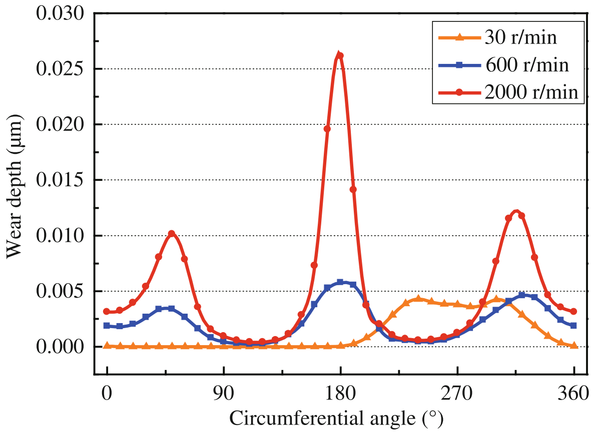

Figure 15 (Color online) Wear depth accumulated for different crank speed.

relative penetration depth between the journal and bearing is quite small in Figure 12 and it increases with a higher crank speed shown in Figure 4. The reason is that the angular velocity of the crank implies the power input to the system, thus a higher speed leads to a deeper penetration and a larger joint reaction force, which can be confirmed in the corresponding acceleration curves. Magnitude of slider acceleration increases drastically when more power is input to drive the system. Owing to the sharp rise in acceleration, the wear rate also increases and the variation of its magnitude is in accordance with that of the acceleration shown in Figures 3 and 13. It should be highlighted that when contact between the two components is lost, the wear rate is zero due to the contactless condition. However, in order to make the curves more clear and illustrate qualitative results, the envelopes of the wear rate curves are drawn in these figures. From the variation of bearing radius illustrated in Figure 14, wear regions can be easily distinguished. When the crank speed is low, the region experiencing wear locates around 270°. With the increase of the crank speed, the regions expand to a more uniform distribution and the wear rate also rises. In order to make the differences more clear, a comparison of wear depth accumulated is shown in Figure 15. When the crank speed decreases, the discrepancies of wear depths in main regions shrink, which means less power input to the system will reduce the wear phenomenon. In addition, a relatively low crank speed will make the contact locations between the journal and bearing limited on the bottom of the bearing surface, thus make the wear region more concentrated.

6.4 Implications

Different from the traditional analysis of wear between two contact bodies, the joint wear analysis in a mechanical system with clearance is a more complex issue. According to the Archard's wear model, contact pressure and sliding distance are the two main factors to determine the wear rate. However, under some operating conditions, due to the flight motion of the journal in a clearance joint, contact may not be continuous. Besides, with the evolution of the relative penetration depth, the contact pressure also changes. So contributions of contact pressure and sliding distance are different in these cases and vary with the types of journal motion. In this section, the index of concordance is introduced to quantify the contributions, which helps to further reveal the implicit relations among the contact pressure, sliding distance and wear rate. This analysis also helps to reduce wear in the clearance joint and optimize the mechanism in design. Regarded as the most widely used method for concordance measure, the cosine matching function (CMF) is employed in this study to evaluate the index of concordance among different variables. A larger value of the concordance corresponds to a stronger correlation between two variables measured. Detailed information of this function is provided in ref. [23]. The result is listed in Table 3 where the *P*-*w* concordance and s-w concordance represent the concordance between the contact pressure and wear rate, and the concordance between the sliding distance and wear rate respectively.

From Table 3, it can be concluded that once impact occurs between the journal and bearing, the *P*-*w* concordance becomes larger than the *s*-*w* concordance, which implies that the contact pressure dominates the wear rate in this case. But it is quite the opposite with the above results when the journal is in permanent contact with the bearing. This is because rebounds often take place after impacts instead of continuous contact when the journal collides with the bearing, which makes the contact duration quite short. Furthermore, the tangential sliding distance is also small in each impact. Thus, the contact pressure is the main contributor to

Table 3 Relations evaluated by CMF

Operating condition		Motion state	*P*-*w* concordance	*s*-*w* concordance
Clearance size (mm)	Crank speed (r/min)			
0.02	30	Flight and impact	*0.7791	0.7461
0.02	600	Continuous contact	0.8916	*0.9571
0.02	2000	Continuous contact	0.8684	*0.9677
0.2	2000	Flight and impact	*0.8338	0.8237
0.5	2000	Flight and impact	*0.8075	0.7495

the wear. In contrast, when the journal permanently contacts with the bearing wall, the magnitude of the contact pressure is similar to the impact case if the differences in relative penetration depth and initial collision velocity are small. However, due to the dynamic response of the system, the journal moves constantly along the bearing surface. Thus, it is reasonable to conclude that the sliding distance dominates the wear rate in this case.

7 Conclusions

The main conclusions of this paper are drawn below.

(1) A comprehensive method for joint wear prediction in planar mechanical systems with clearances considering complex contact conditions is presented in this paper. This method consists of a system dynamic analysis with clearance joints and a wear prediction, where a modified contact force model is employed to evaluate the joint reaction force and the Archard's model is adopted for wear prediction.

(2) The effects of clearance size and input power on joint wear in a slider-crank mechanism are investigated using the proposed comprehensive method. A relatively large clearance could produce a more uniform distribution of wear. And although the magnitude of the joint reaction force is much higher in this case, the total wear depth may not be affected seriously. With the increase of the crank speed, the wear regions expand to a more uniform distribution and the wear rate also rises.

(3) According to the calculation of CMF, under different types of journal motion, the contact pressure and incremental sliding distance have different contributions to wear rate. The contact pressure dominates the wear rate once impact occurs between the journal and bearing. On the contrary, the incremental sliding distance dominates the wear rate when the two components are in permanent contact.

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