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EXPERIMENTAL INVESTIGATION OF CLEARANCE EFFECTS IN A REVOLUTE JOINT

Alessandro TASORA, Edzeario PRATI, Marco SILVESTRI

Department of Industrial Engineering, University of Parma
43100, PARMA, Italy
Phone +39 0521 905903, Fax +39 0521 905705,
E-mail: tasora@ied.unipr.it, prati@ied.unipr.it, silvestri@ied.unipr.it

ABSTRACT

This paper deals with the effects of clearances in revolute joints. Experimental data is acquired from a testbed made of a four-bar mechanism of crank-rocker type, with motorized crank and clearance between rod and rocker. Since the revolute joint pairs can be changed, different values of clearance can be tested. Also the radius of the crank, the length of the truss and the angular speed of the crank can be changed, in order to investigate different settings. During test sessions, rocker and rod accelerations are acquired with accelerometers, and output graphs are compared to the numerical results coming from off-line multibody simulations. Surface wear on the axle and bushing of the revolute joint is measured either with a micrometer tool, either with a three-dimensional surface roughness tester. Wear does not affect the entire surface of the shaft, but mostly happens on specific spots, as predicted by our numerical model.

INTRODUCTION

Clearances in revolute joints, for instance in four bar linkages and in slider-crank mechanisms, may cause rapid wearing, undesired vibrations, noise and low motion precision.
Many articles address the need of numerical models which can realistically reproduce the odd motion of mechanisms with clearances [1][2].
In literature, most experimental works (for example [3],[4],[5]) deal with recording impact events, if any, in joints with clearance. In the present paper we present both the effects of clearances on accelerations and the outcomes of such impacts and vibrations on joint wear.
Experimental acceleration data is compared to numerical simulations provided by a known model [6], which can handle separations and impacts in clearance pairs. A plain implementation of such model [7] tends to overestimate the oscillatory effects, therefore we developed a more advanced model [8] which features dissipative effects caused by friction, and which introduces a more advanced impact model.
Measures on wearing can be useful to suggest predictive models for surface deterioration: we propose to use the Reye hypothesis to record the amount of work of the friction forces in polar coordinates. In fact measured wear does not affect the entire surface of the axle, but mostly happens on specific spots, as predicted by our numerical model that shows high-speed

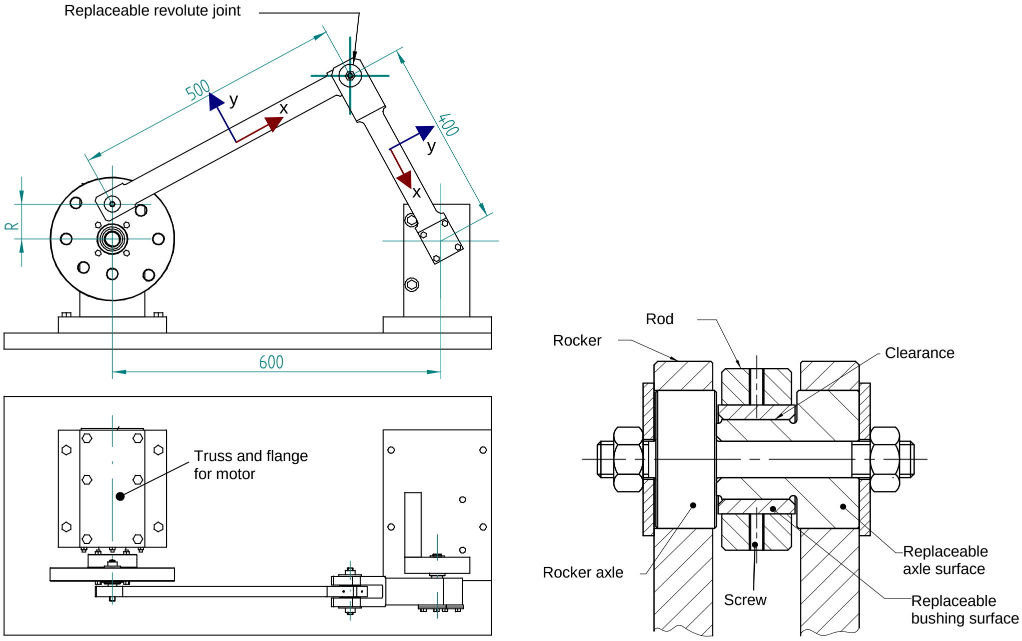

Figure 1: Design of the experimental four-bar linkage.

oscillatory motion of the contact point over two zones of the clearance surfaces.

EXPERIMENTAL SETUP

A four bar mechanism has been built, as depicted in Figure 1. This device exploits some special features that allow an easy measurement of wearing and accelerations. For instance, the revolute joint between rod and rocker can be quickly replaced, so that different clearances can be tested, each time using new (not deteriorated) bushing pairs.

Also, special care has been paid in avoiding clearances in other joints, where special pre-charged ball-bearing joints have been adopted in order to limit noise and unpredictable effects on measurements.

We made sets of revolute pairs with clearance ranging from 0.1mm to 1.0 mm over a base

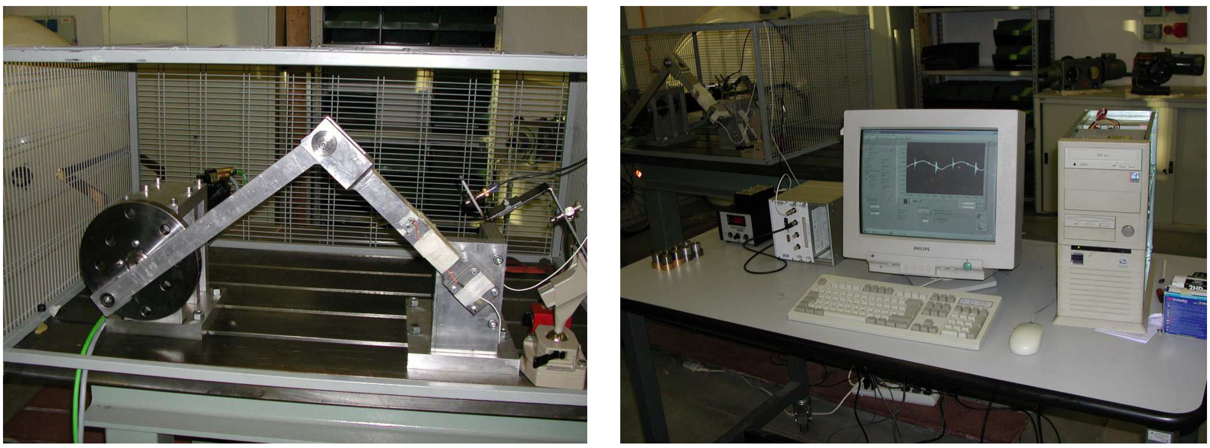

Figure 2. Instrumented four-bar mechanism and data acquisition system.

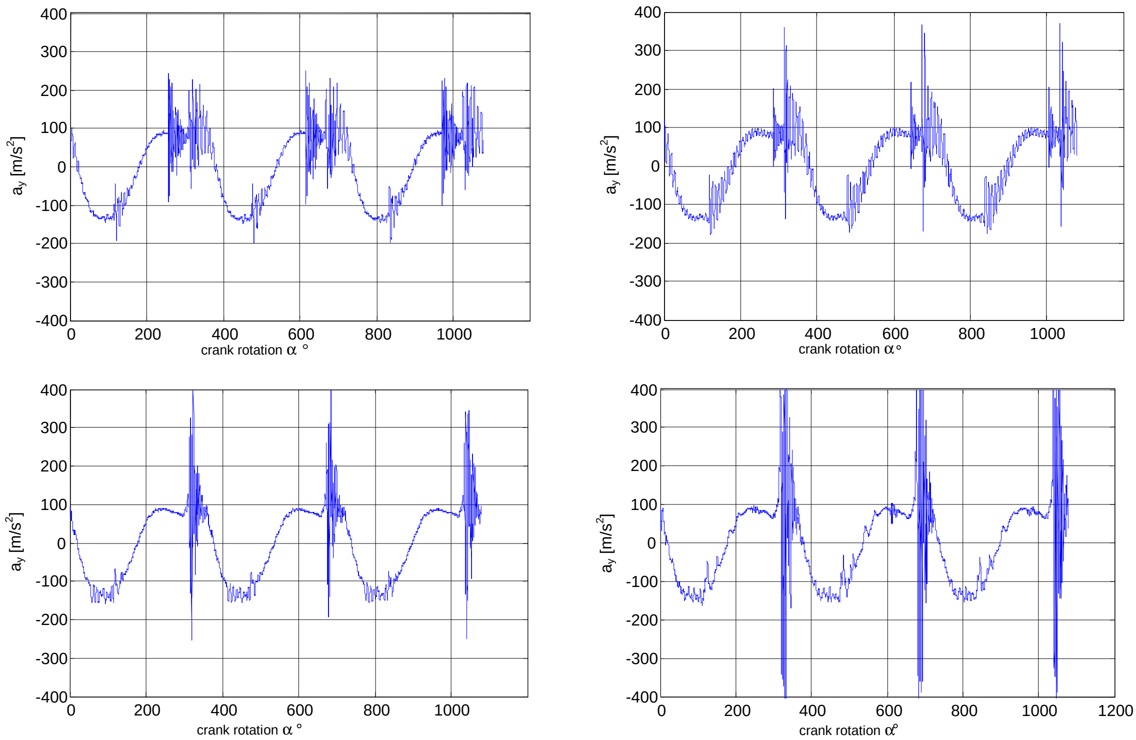

Figure 3. Case of rotation speed = 500 rpm, crank radius = 65 mm: rod Y-acceleration for growing clearances (from up-left: 0.1 mm, 0.2 mm, 0.5 mm, 1 mm).

diameter of 22 mm, using different materials (aluminum, steel, bronze). For each bushing, some sets of radial holes have been machined as references to allow precise measurements of metal wearing in polar coordinates.
In such four-bar mechanism, the crank acts also as a flywheel, and it is connected to a servomotor with resolver feedback. The speed of rotation is digitally controlled, and can be adjusted in the 100-1000 rpm interval. The radius of the crank can be changed between 65 mm, 75 mm, 85 mm and 95 mm. The length of the truss can be modified as well.
Accelerometers can be mounted in different positions, both on the rod and on the rocker. An inductive proximity sensor is mounted on the truss: it detects the upper dead position of the rocker, acting as a trimmer for data acquisition.

EFFECTS ON ACCELERATIONS

Accelerations have been acquired with two Dytran accelerometers: most experiments did not exceed the ±1000 g range of the 3032A1 model, while the 3200B (featuring a ±10000 g range) was just a backup in case of unexpected highly impulsive effects.
The maximum sampling rate of the acquisition board (20 kHz) has been used in order to capture high-frequency effects. Sampling windows last at least 3-4 revolutions of the mechanism and have been repeated many times, both because vibration plots aren’t perfectly periodic and because highly non-linear phenomena such as impacts and friction may occasionally cause unpredictable outcomes.
Accelerometers were applied in four positions, that is in both centers of mass of rod and rocker, aligned to X and Y directions of the bars.

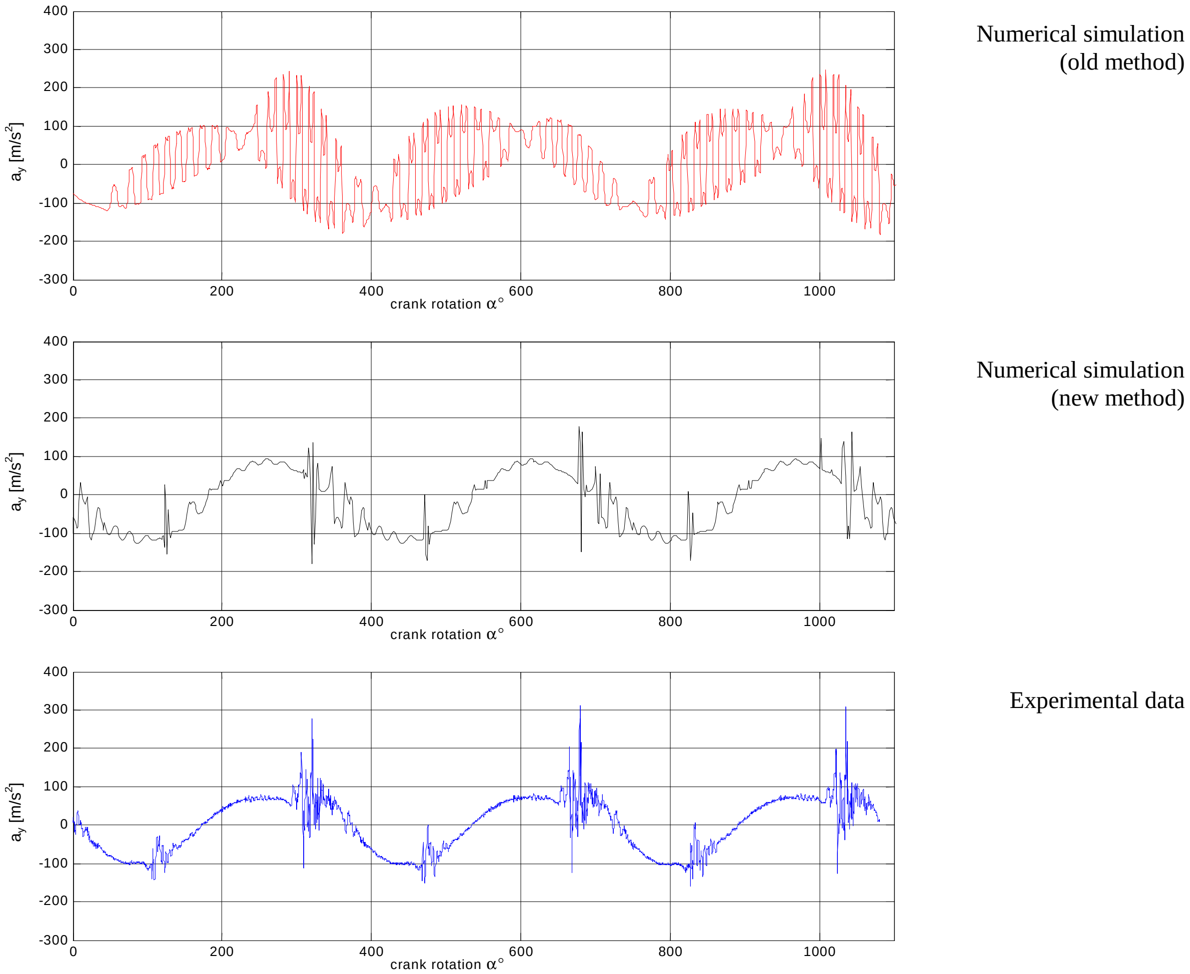

Figure 4. Rod Y-acceleration: Comparison between old simulation model, improved simulation model, experimental data (speed = 500 rpm, clearance = 0.5 mm, crank = 65mm).

Acquisitions have been performed for many combinations of the following parameters.

- Rod-rocker clearance: 0.1 mm, 0.2 mm, 0.5 mm, 1.0 mm.
- Rotation speed: 250 rpm, 500 rpm, 750 rpm, 1000 rpm.
- Radius of crank: 65 mm, 75 mm, 85 mm, 95 mm.

As expected by numerical and theoretical investigations, increasing clearance causes increasing amplitude of oscillations on accelerations (however decreasing high-frequency content), while increasing rotation speed and radius of crank increase both amplitude and high-frequency content of such disturbs (see Figure 3).

Experimental data have been compared with numerical results coming from a simulation software written in Matlab™. This one uses a widely accepted model which can simulate the loss of contact in clearance and successive impacts [6],[7]. The comparison shows the pitfalls of that implementation: friction is not taken into account either for axle-bushing sliding either for tangential effects on impacts, hence overestimating the oscillatory effects on accelerations. Therefore we implemented a mode advanced simulation method that introduces friction between shaft and bushing either for continuous contact, either for impacts (using the recent Chatterjee-Ruina model for tangential impulsive effects [10]). Our new software [7], written in C++ for higher computational speed, gave results which are much more similar to the experimental data, as shown in Figure 4.

Loss of contact in clearance, if any, depends non-linearly on many parameters such as geometry and masses of the system, rotation speed, clearance, friction and so on. Such event is quickly followed by a small impact between axle and bushing, and it is recorded as a clear peak in acquired accelerations. If impacts were really impulsive as in Poisson theory [9], accelerations should have a single (infinite) peak, for an abrupt discontinuity in speeds of the rigid bodies. However, our experimental results show that impacts in clearances give rise to peak accelerations followed by high frequency disturbs which extinguish in few milliseconds (see Figure 5). This most likely happens because the structures cannot be considered perfectly rigid, but should be deemed as elastic bodies (mostly for bar and rocker, but maybe also for crank and truss) whose natural frequencies are excited by impacts. In fact, analysis performed with FEM methods suggested that elastic effects should be taken into account when more precise numerical simulations of post-impact vibrations are needed, but introducing the elastic phenomena in our simulation software would increase a lot the computational time of the method.

Finally, we observed that in many cases the intense disturbs on accelerations such as in Figure 3 are not even generated by contact loss and impacts, because oscillations in position of contact point during continuous sliding are enough to explain such impulse-like effect.

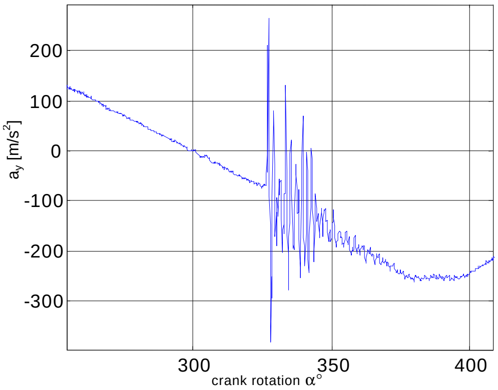

Figure 5. Example of post-impact vibrations (at center of mass of rocker)

EFFECTS ON SURFACE WEAR

We performed long-term tests to measure the distribution of surface wear on axles and bushings, using different clearances, speeds and metals.

In Figure 6 it is shown the deterioration of the axle after a test with 600000 revolutions, and it is easy to see how the effect is different on distinct zones of the surfaces.

In detail, after many tests, we evaluated the metal removal from axles and bushings using a custom measuring system, so that we were able to obtain polar plottings of the wearing (Figure 7).

In general, the material removal is concentrated on two distinct zones, on the opposite sides of the axle (approximately about Φ_5=90° and Φ_5=270°). There is a significant correspondence with the fact that numerical simulations show that the contact point oscillates rapidly inside the bushing at the dead-positions of the rocker (i.e. when rocker inverts rotation direction)

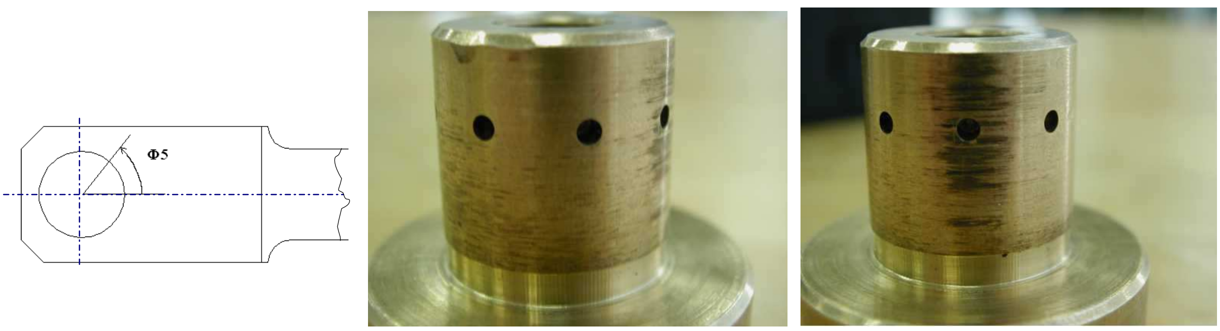

Figure 6. Surface wearing on axle, for Φ_5=0° and Φ_5=90°.
(example of 600000 revolutions, clearance 0.5 mm, rotation speed 500 rpm)

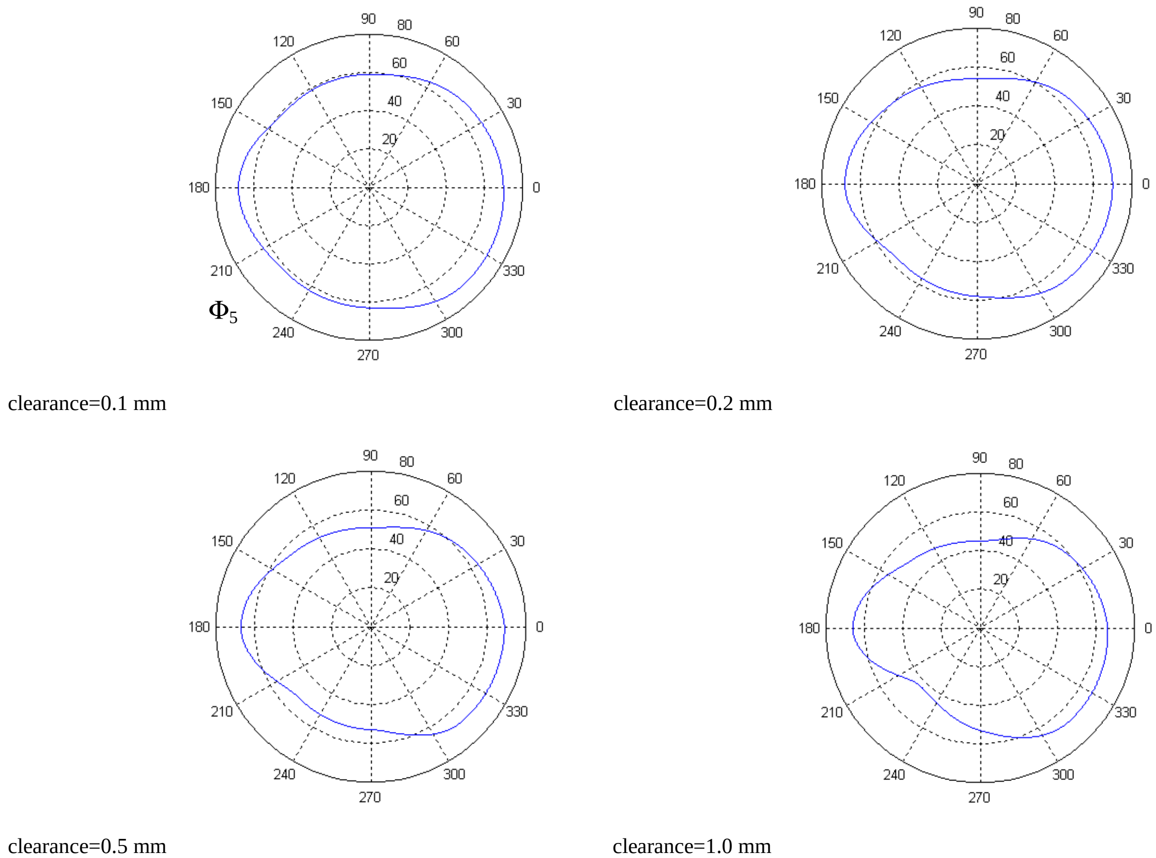

Figure 7. Wearing (subtracted to a circle of 80 μm) after 100000 revolutions at 500 rpm.

hence working as a fast-paced scrubbing device on the two zones. Such oscillating motion of the contact point at the dead-positions is validated also by the experimental results over accelerations.

Moreover, we measured the roughness of bushings and shafts surfaces using a three dimensional tester. Again, the most appreciable wearing is available at the two opposite zones about Φ_5=90° and Φ_5=270°: the close-up 3D diagram provided by this device shows noticeable traces of surface erosion where the contact point is subject to the high-frequency movements (see Figure 8).

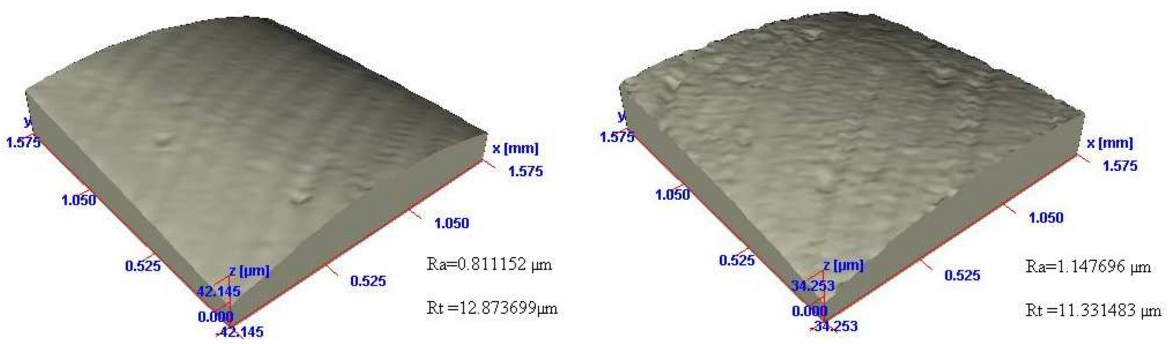

Figure 8. Measured surface roughness, using 3D tester, for Φ_5=0° and Φ_5=90°.

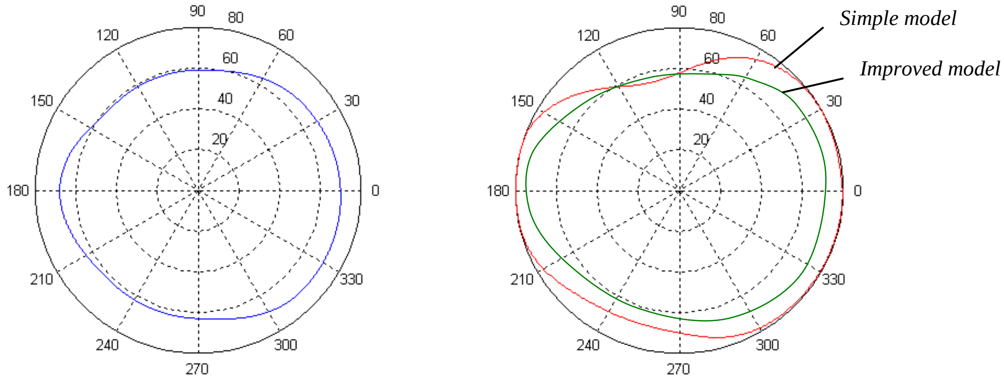

Figure 9: Experimental wear of shaft (subtracted to a circle of 80μm) on the left, and numerical simulation of wear with Reye hypothesis on the right (simple and improved model).

The facts above suggest adopting the Reye hypothesis as a way to predict the surface wear in polar coordinates. Under the assumption of such model, the volume V of the removed material is proportional to the work L_f of the friction forces. When the contact happens inside a sector covering an arc (α_i , α_k), the removed material is:

$$V\Big|_{(\alpha_i,\alpha_k)} \propto L_f\Big|_{(\alpha_i,\alpha_k)} \tag{1}$$

The simulation software has been improved in order to keep track of the work of friction forces on different sectors of the shaft, so that a polar diagram can be plotted after some rotations (at least 4÷5 rotations are needed to obtain an average result).
In detail, an array with n cells is created, with each cell representing a sector. For each k-th step Δt of the simulation, for contact point falling inside a sector (α_i,α_{i+1}) with friction force F_f and contact sliding speed v, the following update is performed:

$$V_k\Big|_{(\alpha_i,\alpha_{i+1})} = C\cdot\left(\Delta s\cdot F_f\right)\Big|_{(\alpha_i,\alpha_{i+1})} \cong C\cdot\left(v\cdot\Delta t\cdot F_f\right)\Big|_{(\alpha_i,\alpha_{i+1})} \tag{2}$$

The total removed volume V on a sector (α_i,α_{i+1}), at the end of the numerical simulation, is the sum of all V_k volumes removed at each integration step:

$$V\Big|_{(\alpha_i,\alpha_{i+1})} = \sum V_k\Big|_{(\alpha_i,\alpha_{i+1})} \tag{3}$$

Given axle width b and original radius r, the radius after wearing $r*$ can be estimated:

$$r*\Big|_{(\alpha_i,\alpha_{i+1})} = r - \frac{V\Big|_{(\alpha_i,\alpha_{i+1})}}{b} \tag{4}$$

Introducing an appropriate constant C in Equation 2, in Figure 9 it is shown how the predicted wearing is comparable enough to the experimental data, especially if the model is improved with a method that takes into account that the contact does not happen on a precise point, but rather on a discrete area.

CONCLUSIONS

A special experimental device has been built for measuring the effect of clearance in revolute joints of a four-bar mechanism, both in terms of accelerations and surface wearing. This has been done for different speeds, different clearances and different crank lengths.

Accelerometers have been placed in distinct directions and sampled data has been compared to numerical results, hence validating a new simulation software which adopts an advanced contact model for higher realism.

Thank to a custom measuring system and to a three-dimensional surface tester, the wearing has been measured in terms of removed material and roughness, thus motivating the development of a method that can predict the wearing of axles and bushings in a context of numerical simulations.

Future developments may embrace the development of a more detailed numerical model where the elasticity of the parts is taken into account, and the high-speed video recording of the mechanism.

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