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# Basic tests on circular eccentric cam-follower pairs

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## 1. Introduction

Cam-follower pairs are difficult to simulate both numerically and experimentally because of their continuously variable conditions of load, speed and geometry.

A new test rig has been designed and realized for investigations on contact force and lubricant film thickness and shape [1] in cam-follower contacts.

In this work, some preliminary experimental results obtained with the apparatus are presented.

## 2. Test rig

The experimental apparatus, that reproduces the cam follower contact, is shown in Figure 1. It is able to simultaneously measure all the contact-force components as well as film thickness and shape. Optical interferometry is used for the latter.

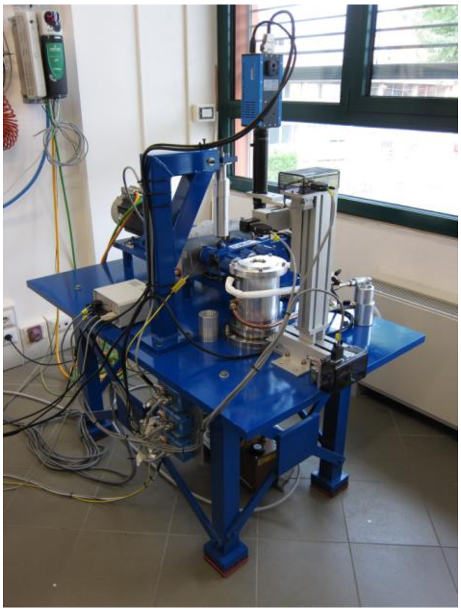

*Figure 1 Pictures of the test rig for cam-follower pairs*

Two different configurations are possible with fixed or moving follower. The follower position is maintained steady when optical interferometry has to be performed, (this is the case of this study), meanwhile the cam can simultaneously rotate and move up and down being coupled with a shaft borne on a rocker arm.

The cam is driven by an electrical brushless motor through an epicyclic gear system with gear ratio equal to 1. Thanks to the use of an adaptor, the follower can be easily changed in order to test different type of material and surface coatings or using glass discs with semi-reflective coating in order to perform optical interferometry. The adaptor is retained in a a novel six axes dynamometer, based on piezoelectric sensors, specifically designed and manufactured for the test rig [2, 3].The dynamometer is used for the measurement of all contact force components, along three different normal directions (normal to the follower, parallel to the cam axis and a third direction orthogonal to the previous two).

The contact between cam and follower is granted by an interchangeable spring, that also provides preloading.

The lubricant is directly furnished to the contact area by a low pressure oil system and it is maintained to the required temperature by a thermostat. The lubricant temperature is measured at the inlet by the means of a K thermocouple.

The data acquisition system, developed on a National Instruments® C-Rio platform, permits the simultaneous and synchronous sampling of contact force and angular position of the cam. All the data are sampled at rate 10kHz for channel with a resolution of 24 bits.

A high-speed camera coupled to a microscope is used for recording the interference images from which the film thickness can be evaluated. The optical system is mounted on a computer-controlled XYZ positioning device so that it is able to follow the contact area during the cam's rotation. A specific program has been developed in Labview® for the movements along three axes.

## 3. Test materials

The cams used for the tests are circular eccentric type with spherical surface. The cams, made of C40 steel, have a radius of curvature of 20 mm and an eccentricity of 2 mm. In Figure 2 a picture of a cam is shown.

*Figure 2 Picture of a cam used during tests.*

Cams without coatings and with different roughness were used, as reported in Table 1. Two followers were used, a steel one for the evaluation of the contact forces only (TEST1), and a glass one, with a semi-reflecting coating (Cr layer 145Å thick plus protection $SiO_2$ layer with 600Å thickness), for the interferometric measurements (TEST2), Tab.1.

*Table 1 Characteristics of the tested cams and followers.*

| Name | Ra Cam (μm) | Ra Follower (μm) | Cam curvature radius (mm) |
|---|---|---|---|
| TEST1 | 0.8 | 1 | 20 |
| TEST2 | 0.2 | 0.02 | 20 |

The test oil was a standard motor oil, a SAE 5W-40, whose viscosity at test temperature is about 0.13 Pas. During the tests it was continuously filtered and refilled and its inlet temperature was maintained constant at 25±1°C.

## 4. Experimental procedure

The dynamometer's calibration was carried on by a specific equipment, applying known pure forces and torques by dead weights. The characterization procedure and the calibration curves are reported in [2, 3]. The notation of the contact forces/torque components used in this work is defined in Figure 3.

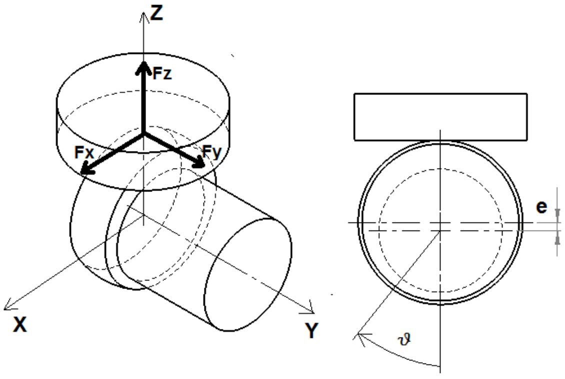

*Figure 3 Definition of the notation of the components of the contact forces, and zero point for the cam rotation angle .*

For each test specified in Tab.1, several data for different cam rotation regimes were acquired.

After the desired preload was applied by the spring with the cam positioned in contact with the follower on its base circle, the different cam rotation speeds previewed by the test procedure are reached. For each step of velocity (50, 100, 200, 250, 300, 350 and 400 rpm for the TEST1, 400 and 500 rpm for the TEST2), at least 5 cam rotation cycles are sampled. Due to the short duration of a single test, the drift effect introduced by the charge amplifier of piezoelectric sensors can be ignored.

The data angular resolution can be easily calculated: it goes from 0.03 deg at 50 rpm to 0.24 deg at the maximum velocity reached, 400 rpm. All the data were saved in a HD for their post processing.

## 5. Data manipulation

The voltage signals acquired from the six-axes dynamometer are converted in real time into the force and torque components in the acquisition program in Labview® by means of the calibration matrix. The relevant data can hence be visualized during the test with no delay, thanks to the simultaneous sampling capacity of the dAQ channels.

The data are stored either unscaled (electrical signal amplitude) or scaled (force and torque). Some data manipulation is performed on the acquired data using a number of Matlab® scripts. Besides the signal scaling, using the calibration matrix, they realize operations like spectral analysis, filtering and plotting.

In Figure 4 an example of unfiltered and filtered signals is shown. The zero point for the cam angle is taken on the base circle, in correspondence to zero lift (see also Figure 3).

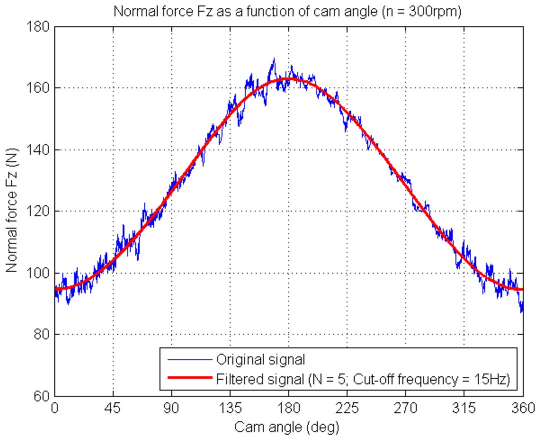

*Figure 4 Examples of $F_z$ unfiltered (blue) and filtered (red) signal acquired during the TEST1 at 300 rpm.*

The noise affecting the signal, which amplitude increases with the cam's speed, is mainly due to the transmission realized by an epicyclical gear system with helical gears (input and output gears with 40 teeth, idle gear with 22 teeth).

A numerical low pass 5th order Butterworth filter was implemented in Matlab®, using different cut-off frequencies, in order to obtain filtered signal for the contact force/torque.

The friction coefficient is calculated as the ratio between the friction force amplitude, measured in *x* direction, and the normal contact force amplitude, measured in *z* direction (referring to Figure 3).

## 6. Experimental results

The plot of lift and its first and second derivative respects the cam rotation angle, for the cams used in the tests, is reported in Figure 5.

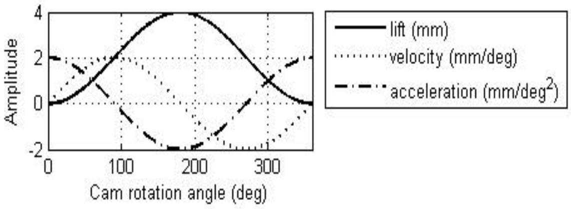

*Figure 5 Lift and first and second derivative of the lift respect the rotation angle for the tested cams.*

The maximum value of the elastic force, depending on the restoring spring, is in correspondence to the cam nose (180 deg). The dynamic effect, depending to acceleration, has also its maximum amplitude by the cam nose, but in opposition of phase with respect the elastic force. The total amount of the normal contact force is made summing the two above mentioned effects [5]. The first derivative of the lift correspond to the displacement of the cam follower contact point [4]. The two points of the inversion of the motion, verify on the flank of the cam, for 90 and 270 deg.

### *6.1. Contact forces measurement*

A waterfall graph of the normal contact force $F_z$ measured in a cam cycle for the different TEST1 rotation speeds is reported in Figure 6: it shows how for higher rotation speeds. the $F_z$ maximum value, such as the amplitude of its dynamic part, decreases. The same effect was obviously observed for $F_z$ in TEST2.

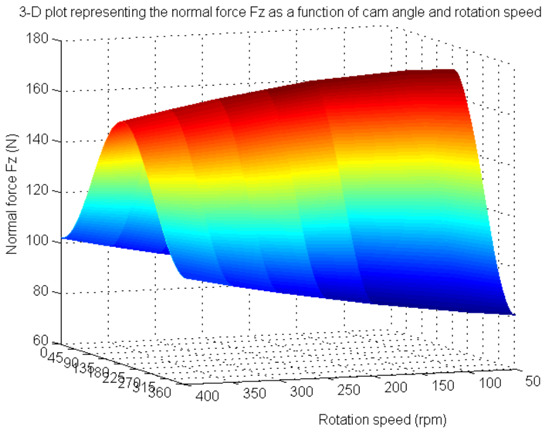

*Figure 6 Waterfall graph for the normal contact force $F_z$ measured in TEST1.*

This trend, depending on the increasing of the inertial effects, in opposition of phase respect the elastic force of the spring, is shown in Figure 7: the maximum of $F_z$ passes from 170 N at 50 rpm to 155 N at 400 rpm, in good accordance to the results obtained with the simulation model of the test rig presented in [5].

The TEST1 friction force $F_x$ (2D plot for the different velocity in Figure 8) decreases with the increase of the cam rotation speed, along all cam rotation angle. It can be noticed the angular displacement of the maximum of the $F_x$ at rising of the rotation speed. It passes from about 170 deg (next to the cam's nose) at 50 rpm, to 130 deg at 400 rpm (see Figure 10). The TEST2 friction force for 400 rpm is illustrated in Figure 9: its magnitude is smaller than the one of the TEST1 because of the lower roughness of surface in contact. The displacement of the maximum value also occurs in this test. This trend was present also in the tests carried out in the study reported in [4].

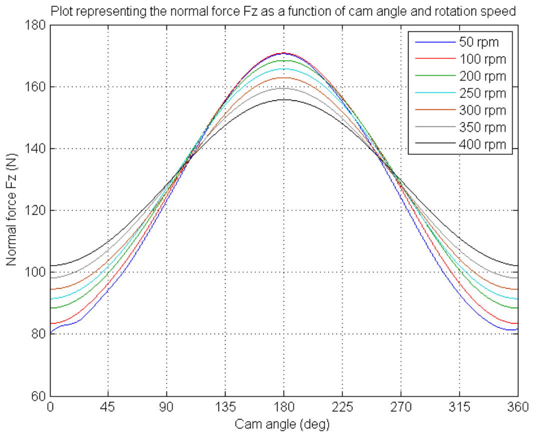

*Figure 7 Normal contact force $F_z$ versus cam rotation angle measured in TEST1 for increasing speed.*

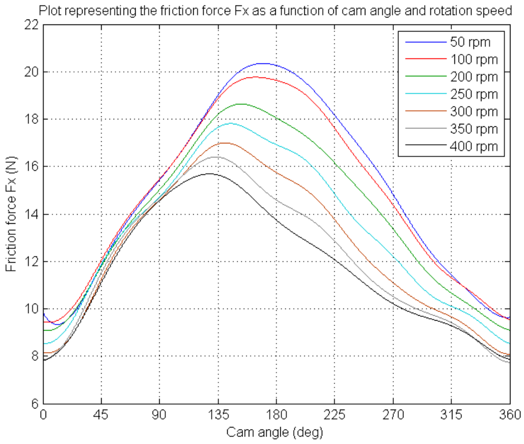

*Figure 8 Friction force $F_x$ versus cam rotation angle measured in TEST1 for increasing speed.*

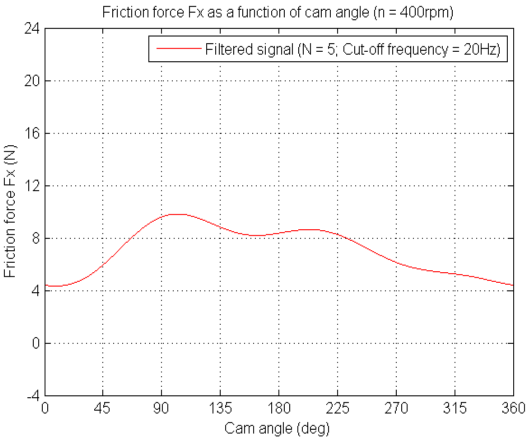

*Figure 9 Friction force $F_x$ versus cam rotation angle measured in TEST2 at 400rpm.*

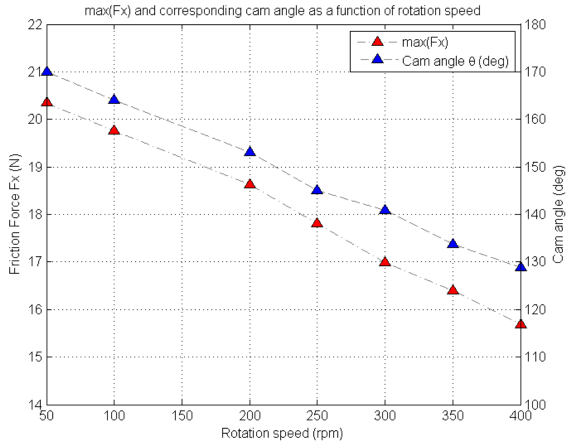

*Figure 10  Friction force maximum value (red triangle) and its angular position (blue triangle) versus cam rotation speed (TEST1).*

The friction coefficient $\mu$ calculated for TEST1 is plotted in Figure 11. It shows a decrement with speed increasing and an asymmetry with respect the cam's nose. The highest value is near the first inversion of the motion of the contact point, during the up side of the lift curve, and this effect become more evident for increasing speed. The friction coefficient calculated at 400 rpm for TEST2 speeds is in Figure 12.

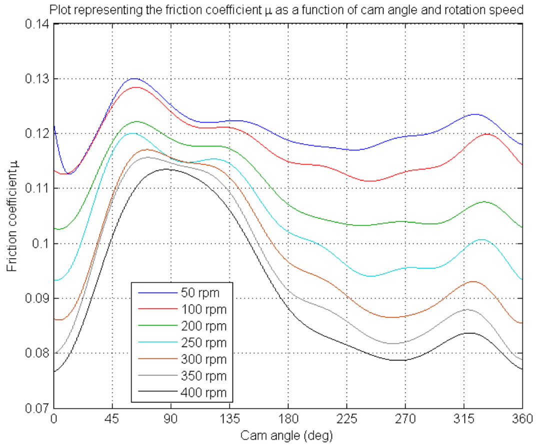

*Figure 11  Friction coefficients versus cam rotation angle in TEST1 for increasing speed.*

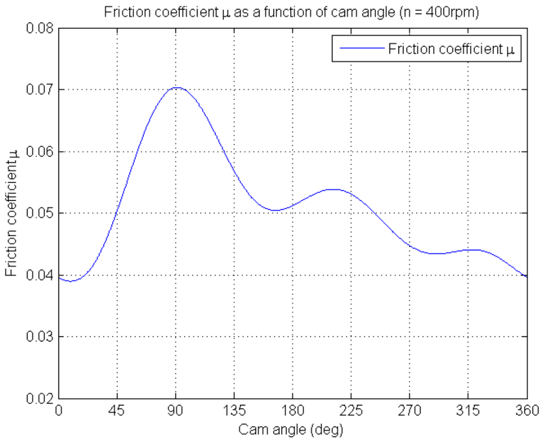

*Figure 12  Friction coefficient at 400rpm for TEST2.*

### 6.2. Torque measurement

The cam driving torque, directly measured thanks to the dynamometer, is shown in Figure 13 (unfiltered in blue, filtered in red). The negative offset is due to the 80 N preload. The torque takes negative values for the whole up side of the lift curve, then it changes its sign at approximately 215 degrees (35 degrees after the cam nose, where the maximum lift occurs), and it becomes negative again at 325 degrees.

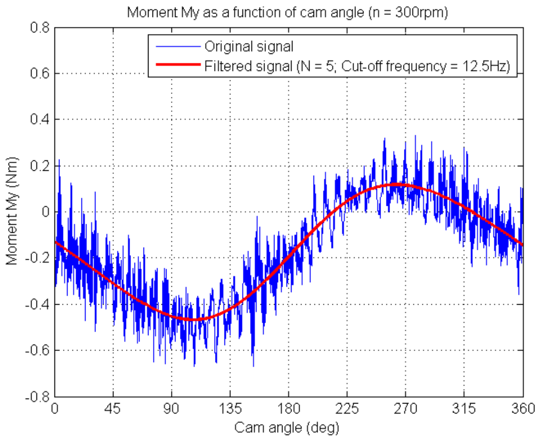

*Figure 13  Unfiltered (blue) and filtered (red) cam driving torque at 300 rpm for TEST1.*

During negative torque phases, the cam follower tends to brake the motor, while during positive phase it gives back work into the drive system. The change of sign is due to the contribution of the normal contact force that in the half part of the cam cycle has a positive moment; its lever is maximum (2 mm) at the 270 degree position (where the motion of the cam follower contact point inverts its direction).

### 6.3. Interferometric images

In Figure 14 two interferometric images recorded at 400 rpm in TEST 2 at 220 deg of cam rotation angle are shown. Some problems due to scratches on semi-reflecting layer provoked by severe preload applied to the contact before the cam starting to rotate are visible above all in right side image.

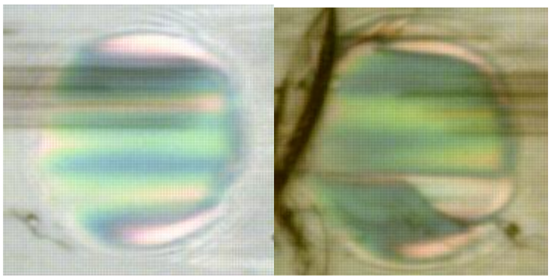

*Figure 14  Examples of interferometric images recorded at 400 rpm in TEST 2 at 220 deg of cam rotation angle.*

Another problem is connected with the non uniform roughness of the cam surface, that shows the traces of the grinding in some angular position.

New tests are being carried out with a lighter preload and cams characterized by uniform surface roughness.

## 7. Conclusions

A number of preliminary tests performed with a new test rig for cam-follower lubricated pairs show the capabilities of the apparatus for force and film thickness measurements.

First methodologies for numerical data manipulation have been identified

## 8. Acknowledgment

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## 9. References

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