IOP Conf. Series: Materials Science and Engineering **174** (2017) 012036 doi:10.1088/1757-899X/174/1/012036

The behaviour of lubricated EHD contacts subjected to vibrations

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Abstract. Machine components containing contacts working in elastohydrodynamic (EHD) conditions are often subjected to vibrations. These may be originated from the mechanism or machine the contact is part of, the surrounding environment and within the contact itself. The influence of vibrations upon the behaviour of elastohydrodynamic films has been studied experimentally in a number of papers, but a comprehensive study of the effect of the parameters of the oscillatory motion upon the film thickness has not been carried out yet. In this study the authors evaluate the effect of the frequency of the oscillatory motion upon the EHD film thickness. Optical interferometry is used to measure lubricant film thickness in a ball-on-flat disc arrangement. A high – speed camera records the interferometric images for later analysis and conversion into film thickness maps. The disc runs at a constant angular velocity while the ball is driven by the traction forces developed in the EHD film. In steady state conditions, this would ensure pure rolling conditions, however in the present investigation the ball is subjected to harmonic vibrations in a direction perpendicular to the plane of the film. The contact under study is lubricated by basic oils and the temperature is kept at a constant value of 60^{o}C. The aim of this paper is to understand how vibrations influence the lubricant film formation.

1. Introduction

The elastohydrodynamic (EHD) regime occurs in lubricated non – conformal contacting surfaces, where, apart from the hydrodynamic action of convergent geometry, the elastic deformation of the surfaces and the increase of lubricant's viscosity with pressure govern the formation of the film. Many experimental [1 – 7] and numerical [8 – 11] studies have been carried out during the past six decades or so, which allowed the evaluation of the contribution of various parameters upon the behaviour of the lubricant film. This is valid for steady state conditions where the geometry of surfaces and various parameters are constant during the operation of the machine component however full steady state conditions are almost never encountered in practice. Classic examples of non – steady state or transient EHD conditions are cams and gears mechanisms, where rapid changes of geometry and load take place. Even rolling element bearings, normally associated with smooth, steady running experience transient conditions at the start up and shut down of the motion and in devices which inherently required variation of speed (for example the bearings of stepper motors).

Studies of the effect of transient conditions upon EHD films have received more attention during the past two decades. Many aspects of transient EHD have been approached such as elastic rebound and cyclic squeeze of a sphere on a flat surface [12 – 14], variation of geometry including artificially induced features [15 – 18], rapid change of speed, including sudden halting or start of motion [19 – 23]. These studies include both experimental and numerical approaches.

Elastohydrodynamic contacts subjected to vibrations can also be described as working in non – steady sate conditions of lubrication. The way the EHD film responds is very important in terms of the control of vibrations and noise of the whole system. Vibrations can be originated inside the contact itself, for example due to irregularities of surfaces. This situation was studied by Dareing and Johnson [24] who used a two –disc machine with the surface of one disc corrugated. Studying the effect of lubricant film on the damping of the system they found that the film has a significant contribution to the total dumping, especially at lower loads. They concluded that the fluid film contributed to damping via a squeeze film mechanism.

Another type of vibrations elastohydrodynamic contacts can be subjected to, are structural vibrations generated by shocks in loading [25, 26], or transmitted from the machinery the contact is part of [27]. It was found that a combination of lubricant entrainment and squeeze generates fluctuations in the film thickness. Parallel experiments and dynamic simulation of the whole system, including the contact has shown that the contribution of the dynamic response of the EHD contact itself is dominant. It is worth mentioning here that dampened vibrations similar to those obtained by a sudden variation of the applied load are obtained when the speed changes rapidly [28, 29]. This proves that the contacting surfaces and the lubricant can be seen as a mass, spring, damper system which responds when suddenly pushed out of a steady state condition.

Finally lubricated contacts can be subjected to forced vibration caused by cyclic, variable loading. Sakamoto and co – workers applied a pulsating load to a ball – on – flat lubricated contact and observed the variation of film thickness using the optical interferometry technique [30]. The load increased and decreased rapidly, in less than 50 milliseconds and the whole cycle was about 0.3 seconds. They found that during both, application and removal of load, oil entrapments form and travel like ripples through the contact. At larger entrainment speeds the formation of oil entrapment is suppressed.

Somehow surprising but there are not many experimental studies reporting on the behaviour of elastohydrodynamic contacts subjected forced vibrations and none, as far as the authors are aware to forced harmonic vibrations. The authors try to fill this gap in the present research where they apply forced, quasi harmonic vibrations to an elastohydrodynamic contact formed between a ball and a glass disc and evaluate the behaviour of the lubricant film.

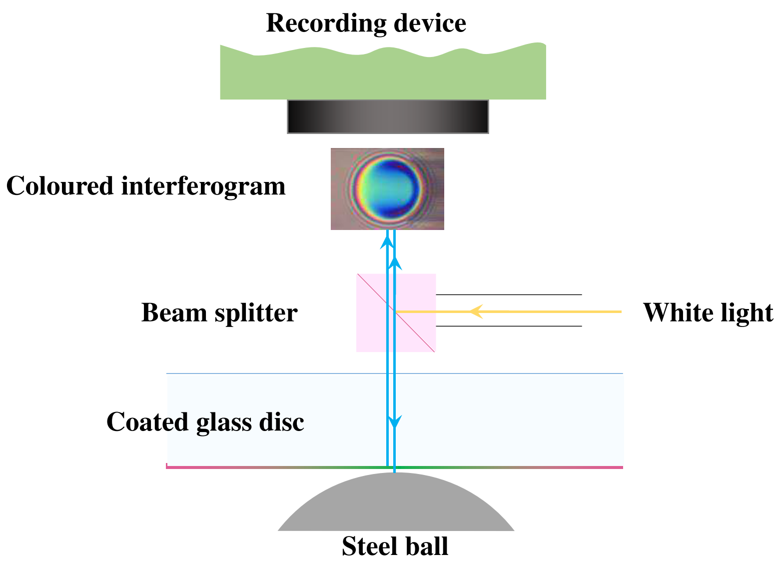


Figure 1. Principle of optical interferometry.

2. Experimental method and calibration

The experimental method widely used for quantitative, albeit modelled elastohydrodynamic contacts is optical interferometry. This technique was first used by Cameron and Gohar [1] and then employed extensively by other researchers in the field so it will only be briefly explained here. In the current arrangement the contact under study is formed between the flat surface of a glass disc and a steel ball, as seen in figure 1. The disc is coated on the contacting surface with a chromium and silica double layer. White light is used which results in film thickness maps in the form of coloured interferograms, corresponding to the wavelength of light which underwent constructive interference at a certain location on the contacting surface. In figure 1 an image of the EHD contact under study, showing the distinctive horse – shoe shape, can also be seen. The only remaining problem is to relate the colours in the image with a film thickness. This can be done in more than one way, but in this research the calibration of colour content versus film thickness is done following the procedure of the method called SLIM (spacer layer imaging method) [31].

SLIM takes advantage of the high precision of another method of measuring film thickness named Ultra – Thin – Film Interferometry (ULTRA) [2]. In ULTRA the central film thickness from a narrow band in the middle of the contact is measured. Using a test rig equipped with ULTRA, the film thickness for the test lubricant is measured in pure rolling conditions, at increasing entrainment speed. A correlation between film thickness and speed is thus obtained. In the second step the colour digital camera is attached to the rig. The spacer layer disc used in ULTRA is swapped with a glass disc coated with a thin (5 – 10 nanometres) thick chromium layer and making sure that the temperature is exactly the same as in the first test, the contact is run again through the same speeds, while coloured images are recorded. The variations of R, G, and B content with film thickness is extracted and are later used to convert the images of the contact under vibration into film thickness.

3. Test rig and lubricants

Optical interferometry technique, seen in figure 1 was incorporated within a test rig capable of producing forced vibrations of the ball. A schematic of the test rig is shown in figure 2.

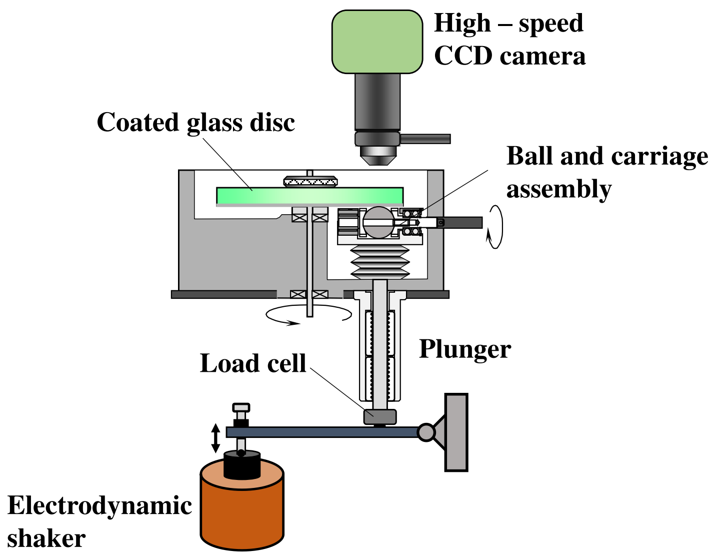


Figure 2. Experimental setup.

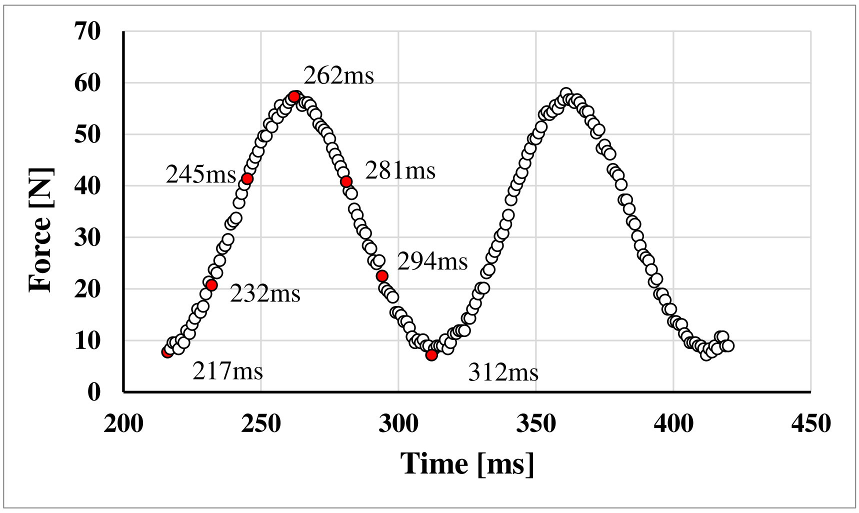
The glass disc shown is coated with a thin chromium layer (between 5 and 10 nanometres thick) and a silica layer (about 130 nanometres thickness) on the contacting surface. The disc is attached to a shaft which rotates at desired speed. The ball is also attached to a small shaft, but is free to rotate under the action of the friction force generated at the contact between the two. The shaft of the ball is supported by needle bearings on one side and a pair of angular contact bearings of the other, such that the total stiffness of these contacts is larger than that of the contact between the ball and glass disc. The ball carriage is attached to a bellow and actuated by a plunger guided by a pair of linear bearings fitted to the frame of the rig. The plunger is set in a harmonic motion by a shaker via a lever arm and a load cell. By connecting the load cell between the plunger and the lever, the force acting normal to the contact can be precisely measured, given the fact that the plunger is very rigid in compression.

The load was varied sinusoidally from zero (or close to zero) to about 50 N. For the pair of materials used this means a Hertzian pressure variation between zero and around 0.7 GPa. The frequency of the oscillatory motion of the ball was 10 Hz, 25 Hz, 50 Hz, and 100 Hz for these tests.

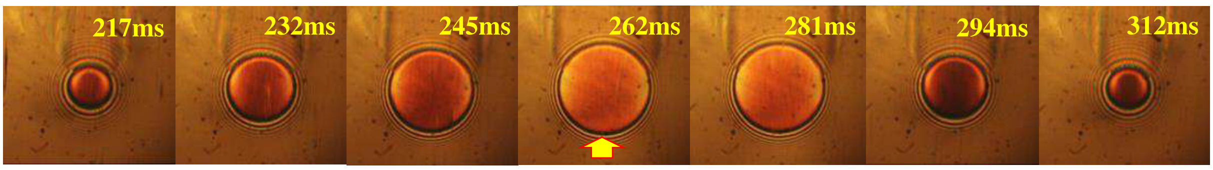
More than one lubricant was tested in this research, but for limitations of space only the results for a synthetic base oil, poly – alpha – olefin (PAO40) are shown. Poly – alpha – olefins are hydrocarbon synthetic oils widely used in a wide variety of applications because they have well characterized chemical composition and many steady state experimental data are available in literature. The dynamic viscosity of this oil is 0.396 Pas at 40^{o}C and 0.039 Pas at 100^{o}C. The temperature of the lubricant was kept at 60^{o}C. This was chosen in order to get steady state film thickness thick enough for the transient effects to be quantified. In order to get a good number of images for each cycle of oscillation, even at larger frequencies, a speed of 2000 frames per second was set on the camera.

4. Results and discussion

The tests were carried out at entrainment speeds with values between 0.1 m/s to 0.5 m/s however it was found that because of the too large film thickness obtained at the larger speeds, data obtained at those conditions was not usable. For this reason only results for the lower speed will be shown in this paper. Figures 3 to 6 show the variation of the load and images of the elastohydrodynamic contact at certain points during the cycle.

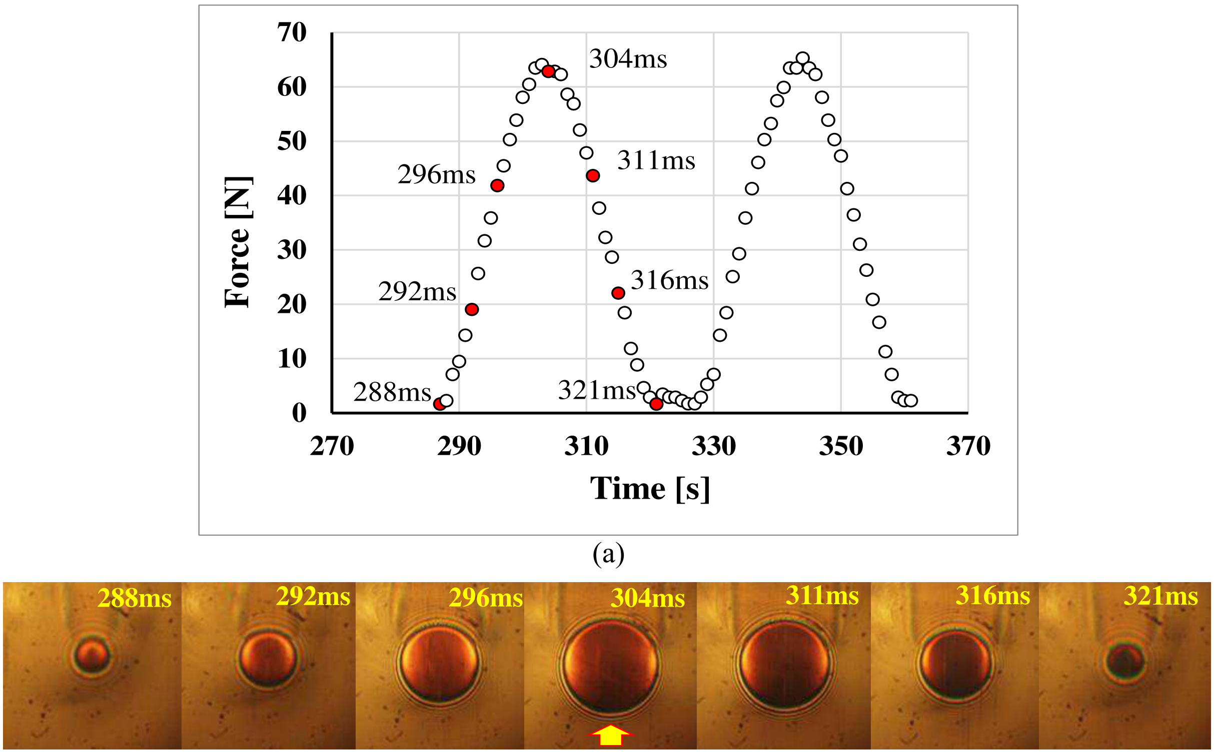


(a)



(b)

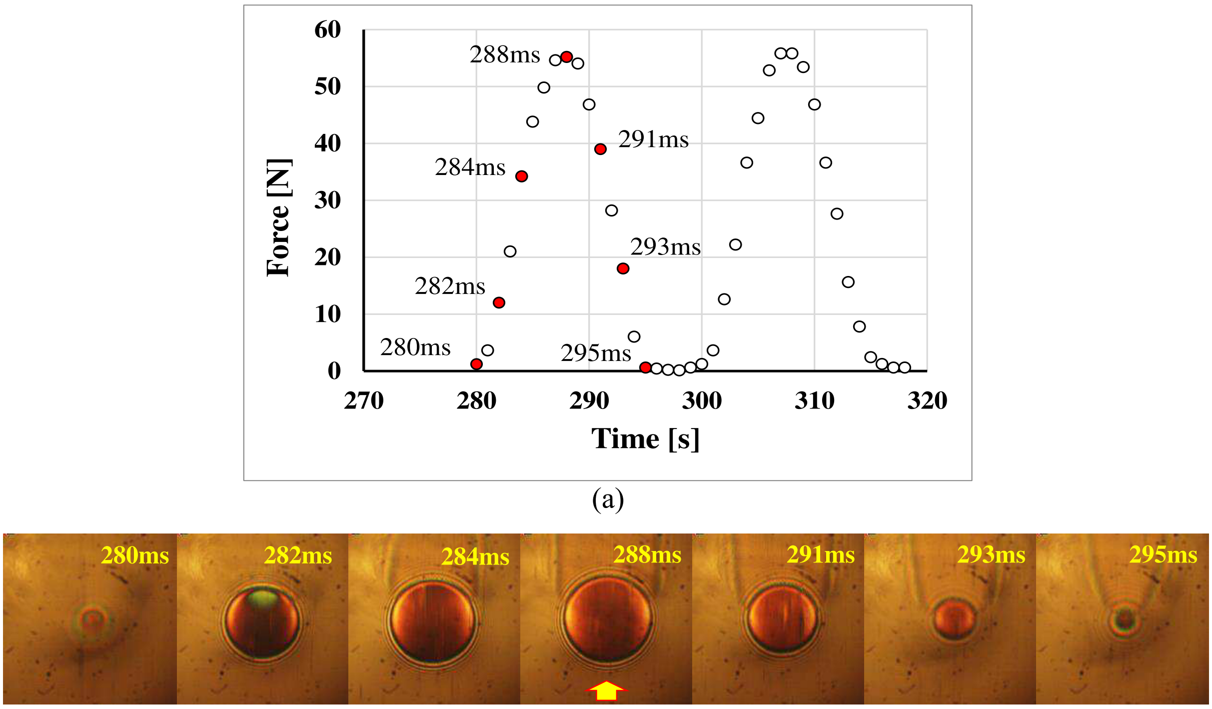
Figure 3. Load variation and interferograms of the contact for 10 Hz, 0.1 m/s entrainment speed.



(a)

(b)

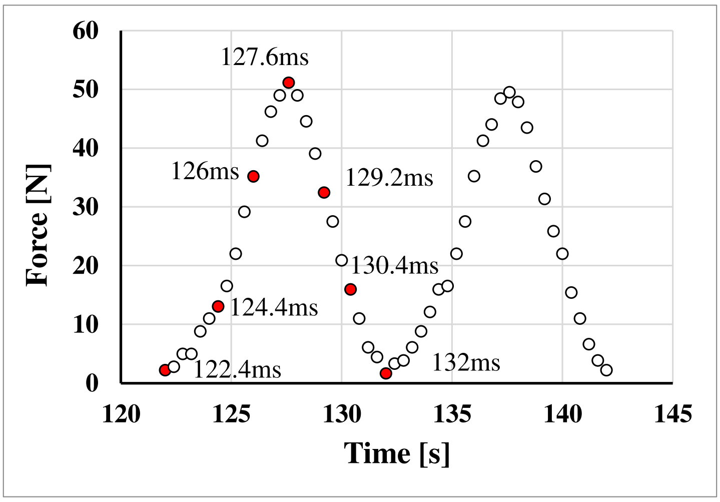
Figure 4. Load variation and interferograms of the contact for 25 Hz, 0.1 m/s entrainment speed.



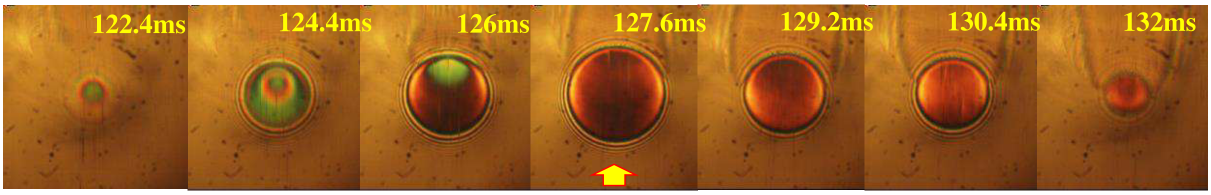
(a)

(b)

Figure 5. Load variation and interferograms of the contact for 50 Hz, 0.1 m/s entrainment speed.



(a)



(b)

Figure 6. Load variation and interferograms of the contact for 100 Hz, 0.1 m/s entrainment speed.

As seen in these images at lower frequencies of vibration, which imply relatively small rate of variation of load, the film thickness does not show any special features, which leads to the conclusion that in these conditions steady state film treatment of the lubrication is satisfactory. At 50 Hz and especially at 100 Hz however, pronounced deviations from the classic flat film thickness region can be observed. These deviations are larger during the load increase phase. As it is seen in the first two images, the contact diameter increases nearly four times over a two – millisecond interval. This sudden increase of contact area implies a sudden expansion of the convergent zone in the inlet, which, in turn means that thicker film is forced into the contact. The thicker lubricant intake forces the surfaces to further deform elastically thus it is accompanied by an increase of pressure. Although calculations have not been carried out this additional elastic deformation of the surfaces is accompanied by local increase of the contact pressure. In real – life applications this may translate into reduced fatigue durability. In these tests, during the 100 Hz tests, it was noticed that the lowest durability of the silica coating was obtained at the thickest films, which may be explained by the pressure spikes due to fluid entrapment. Waves of the film thickness can be observed during the load decrease phase, but they are crescent shaped and very shallow.

The variations over half cycle of central and minimum film thickness are shown in figures 7 and 8. For the lowest frequency, at 10 Hz, the central film thickness is roughly equal to the central film thickness on a steady – state test that is about 130 nm. In the 100 Hz test the values of the film thickness during the first 2 – 4 milliseconds after the beginning of the load phase, the film thickness is much larger than the steady – state values. At 220 nanometres the central film thickness is about sixty percent larger than the steady state film.

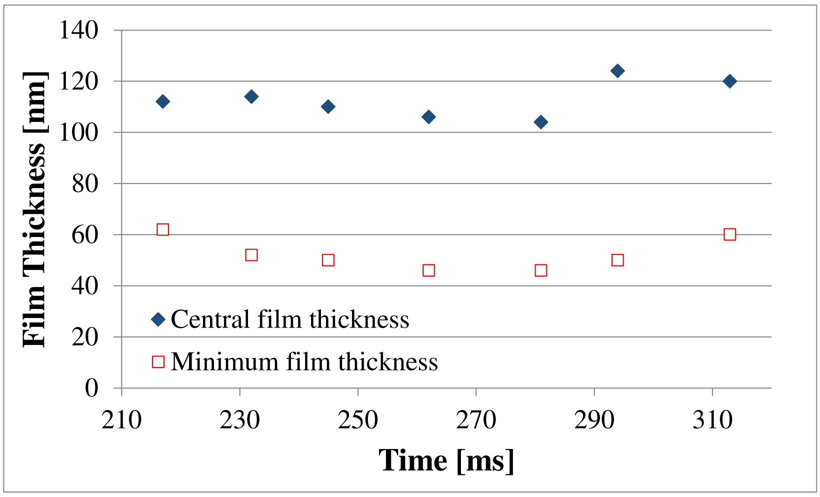


Figure 7. Central and minimum film thickness, 10 Hz.

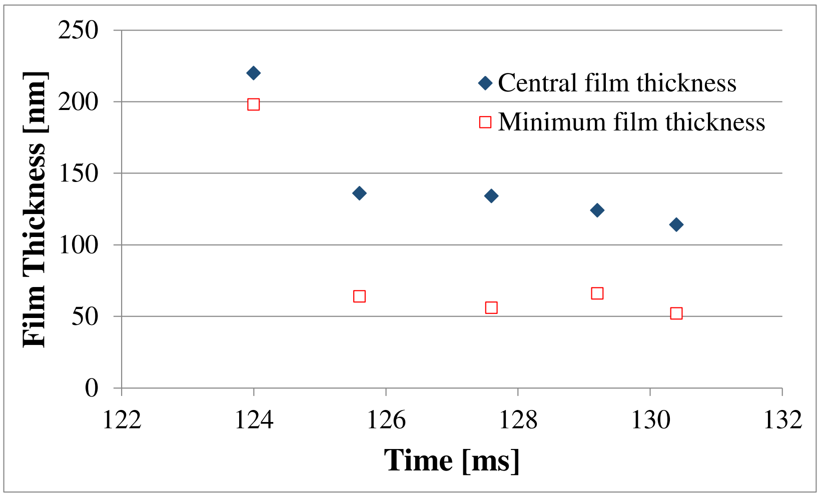


Figure 8. Central and minimum film thickness, 100 Hz.

5. Conclusions

This paper shows the results of an experimental investigation into the behaviour of elastohydrodynamic films subjected to harmonic, forced vibrations.

The elastohydrodynamic contact under study was a circular contact, formed between a glass disc and a steel ball in the presence of a synthetic base oil lubricant. The film thickness was measured by the optical interferometry. The use of white light resulted in coloured images of the EHD contact which were recorded by a high – speed camera and later analysed for film thickness at various locations on the contact area. The contact was subjected to frequencies from 10 Hz to 100 Hz while the entrainment speed was kept at 0.1 m/s.

It was found that at lower frequencies the film thickness shows very little change from the steady – state values. At 50 Hz and 100 Hz, however a very thick film formed few milliseconds after the beginning of the load increasing phase. As the load continues to increase the film thickness stabilises at around steady – state values and shows very shallow waves. The local increase of film thickness means increased elastic deformations of the contact and thus local variation of the pressure. It is expected that this local, large variations of pressure to be detrimental to fatigue life of the contact, in real – life applications.

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