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The measurement of lubricant-film thickness using ultrasound

By R. S. Dwyer-Joyce[1], B. W. Drinkwater[2] and C. J. Donohoe[1,2]

[1]*Department of Mechanical Engineering, Mappin Street, University of Sheffield, Sheffield S1 3JD, UK*

[2]*Department of Mechanical Engineering, University Walk, University of Bristol, Bristol BS8 1TR, UK*

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Ultrasound is reflected from a liquid layer between two solid bodies. This reflection depends on the ultrasonic frequency, the acoustic properties of the liquid and solid, and the layer thickness. If the wavelength is much greater than the liquid-layer thickness, then the response is governed by the stiffness of the layer. If the wavelength and layer thickness are similar, then the interaction of ultrasound with the layer is controlled by its resonant behaviour. This stiffness governed response and resonant response can be used to determine the thickness of the liquid layer, if the other parameters are known.

In this paper, ultrasound has been developed as a method to determine the thickness of lubricating films in bearing systems. An ultrasonic transducer is positioned on the outside of a bearing shell such that the wave is focused on the lubricant-film layer. The transducer is used to both emit and receive wide-band ultrasonic pulses. For a particular lubricant film, the reflected pulse is processed to give a reflection-coefficient spectrum. The lubricant-film thickness is then obtained from either the layer stiffness or the resonant frequency.

The method has been validated using fluid wedges at ambient pressure between flat and curved surfaces. Experiments on the elastohydrodynamic film formed between a sliding ball and a flat surface were performed. Film-thickness values in the range 50–500 nm were recorded, which agreed well with theoretical film-formation predictions. Similar measurements have been made on the oil film between the balls and outer raceway of a deep-groove ball bearing.

Keywords: lubricant-film measurement; ultrasound; elastohydrodynamic lubrication; film-thickness measurement

1. Introduction

The durability of machine elements such as gears, bearings and seals relies on the integrity of the lubricant film separating the contact surfaces. If the film fails, these surfaces contact and friction, wear and seizure can occur. The thickness of an oil film depends on the lubricant properties, the geometry of the bearing surfaces and the operating conditions. Typically, conformal contacts, such as those in journal bearings and thrust pads, operate in the hydrodynamic lubrication regime with films of the order of 1–100 μm. Counterformal contacts, such as those in rolling bearings and gears, operate in the elastohydrodynamic (EHD) regime with films less than 1 μm thick. The lubricant film should be thick enough to ensure separation of the surfaces, but not so thick that excessive pumping losses are incurred.

There are many methods available for predicting the thickness of fluid films formed between surfaces based on Reynolds's equation (Hamrock 1994). These are now routinely used in the design of bearing components. However, the measurement of lubricant films, and certainly the online monitoring of film thickness in a component, has proved more difficult.

Existing techniques for lubricant-film measurement fall into two categories: electrical methods and optical methods. Early work (El-Sisi & Shawki 1960) concentrated on the determination of the film thickness by measurement of its electrical resistance. In later studies, measurements of the film capacitance (Astridge & Longfield 1967) proved more effective and the use of micro-transducers gave enhanced resolution (Hamilton & Moore 1967). These methods require electrical isolation of the contact elements and either give a composite film-thickness value over the whole bearing surface or require the positioning and alignment of small surface-mounted sensors. The application of optical methods requires either that one of the contact elements is transparent or that it contains a transparent window. Optical interferometry (Cameron & Gohar 1966) has proved successful in the measurement of EHD films and, more recently, boundary films (Johnston *et al*. 1991). The use of lasers to fluoresce a lubricant film has also been used to determine film thickness (Richardson & Borman 1991). However, the requirement for transparency has meant that these methods are rarely used outside the laboratory.

In this paper, a novel approach has been investigated in which the response of a lubricant film to an ultrasonic pulse is used to determine the film thickness. The method is such that it can be used, with appropriate signal processing, over a wide range of lubricant-film thickness. Also, because it is essentially non-invasive, it can be used to give online film measurement in real bearing components.

2. Ultrasonic reflection from a layer

When ultrasound is incident on a boundary between two different media, some of the energy is reflected and some transmitted. The reflection and transmission behaviour of displacement waves at the boundary is dependent on the acoustic properties of the two media. The proportion of any incident signal reflected (or the reflection coefficient, R) is given by the well-known equation

$$R = \frac{z_1 - z_2}{z_1 + z_2}, \tag{2.1}$$

where z is the acoustic impedance of the media (given by the product of density and speed of sound) and the subscripts refer to the two media. The proportion of the incident signal transmitted (or transmission coefficient, T) can then be calculated as

$$T = 1 - R. \tag{2.2}$$

If ultrasound is incident on a multi-layered system, then the signal transmitted or reflected is the superposition of the result of the application of (2.1) and (2.2) at each boundary. Figure 1 shows an ultrasonic beam incident on a typical lubricated contact, which is a three-layered system consisting of steel–lubricant–steel. The steel

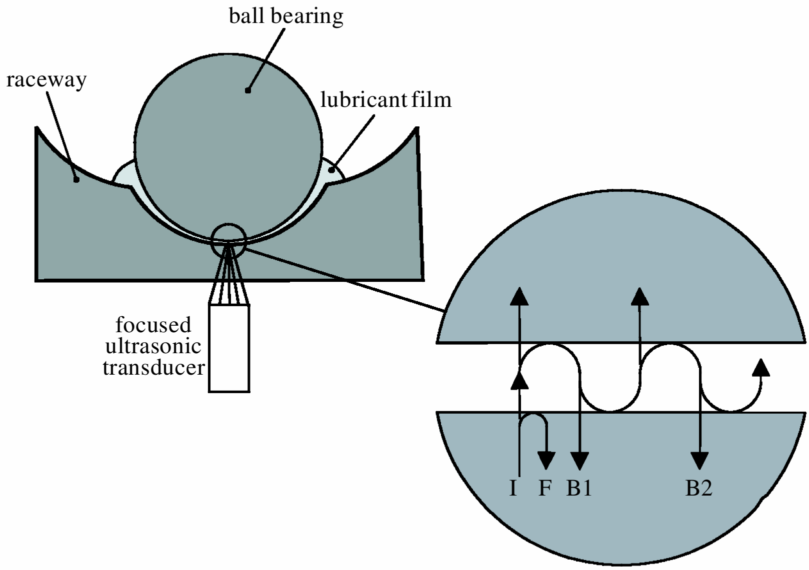

Figure 1. Schematic of an ultrasonic beam incident on a lubricated contact.

either side of the lubricant represents the bearing elements, such as the bush and shaft in a journal bearing, or the ball and raceway in a rolling-element bearing. As the steel sections are large when compared to the wavelength of the ultrasound, they can be modelled as semi-infinite. If the lubricant layer is sufficiently thick, or the ultrasonic wave packet small enough, then the reflections from the top and bottom bearing surfaces are discrete in time. This means that if the speed of sound in the lubricant is known then the thickness of the lubricant film can be determined by measuring the time-of-flight (ToF) between the two reflections. The amplitude of these reflected pulses can also be calculated from (2.1) and (2.2), assuming that the losses in the lubricant are small. This ToF approach is commonly used for thickness gauging of metallic components and for corrosion monitoring. As the lubricant film becomes thinner, the ToF approach becomes less accurate until, for very thin layers, the reflected pulses overlap and it becomes impossible to distinguish the discrete reflections. Typically, even the thickest lubricant films are less than 50 μm thick, and so the ToF approach is rarely applicable.

For a practical lubricant film, the challenge is to extract the thickness information from the reflections that are overlapping in the time domain. There are a number of approaches that can be adopted. For example, Tsukahara *et al*. (1989) used the surface waves generated by an acoustic microscope to characterize thin layers. In this paper, two alternative approaches are used. Firstly, the through-thickness resonances of the lubricant layer are measured and interpreted via a continuum model. Secondly, if the lubricant-film thickness is so small that the frequency of the first through-thickness resonance is above the measurable range, the amplitude of the reflected signal can be measured and interpreted using a spring interface model. These approaches are now discussed in detail.

Many authors (Hosten 1991; Kinra *et al*. 1994; Pialucha & Cawley 1994) have described continuum models of the multi-layered system response to ultrasonic waves. Based on continuity of stress and strain at each boundary in the multi-layered sys-

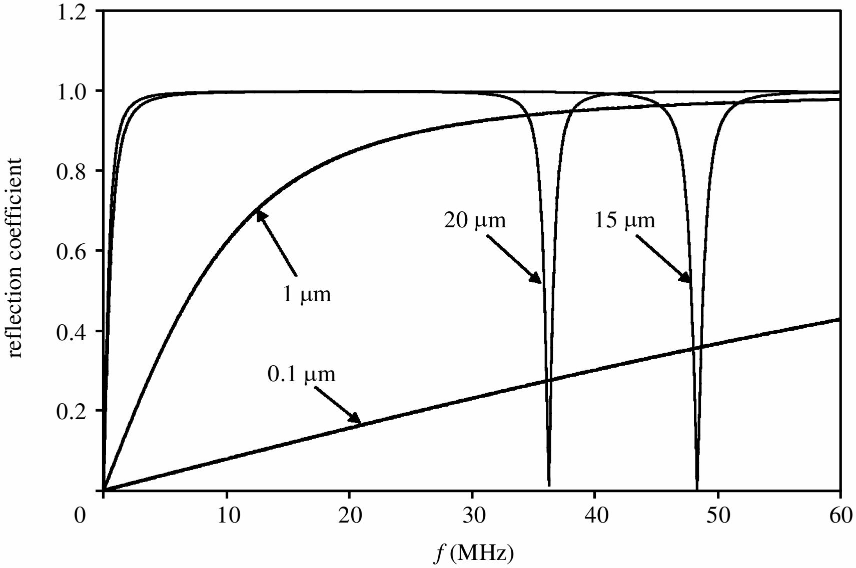

Figure 2. Continuum-model predictions of the reflection-coefficient spectrum from a layer of mineral oil between two steel elements.

Table 1. *Acoustic properties of lubricating oil* (*Shell Turbo T68*) *at room temperature and ambient and EHD pressures* (*estimated using the method of Jacobson & Vinet* (*1987*))

(Also included for comparison: water and steel properties.)

	density ρ (kg m^{-3})	bulk modulus B (GPa)	wave velocity c (m s^{-1})
oil at atmospheric pressure	876	1.84	1450
oil at 0.8 GPa	1002	12.6	3550
oil at 1.5 GPa	1044	21.2	4500
water at atmospheric pressure	1000	2.06	1483
steel	7923	172	5840

tem, these models can be used to predict the reflection and transmission of plane waves through a given system. Figure 2 shows a continuum-model prediction of the reflection coefficient spectrum for 0.1, 1.0, 15 and 20 μm thickness mineral oil layers between two steel half-spaces. The material properties used in this calculation are shown in table 1. This is essentially the *impulse response function* of the system, i.e. it gives the response of the system for an incident wave of unit amplitude. In figure 2, the resonant frequencies are seen as sharp minima in the reflection-coefficient spectrum. The minima seen at 36.3 MHz corresponds to a lubricant-film thickness of 20 μm.

Pialucha *et al.* (1989) used a continuum-model approach to show that the resonant frequencies of an embedded layer are related to its thickness, h, and acoustic properties by

$$h = \frac{cm}{2f_m}, \tag{2.3}$$

where c is the speed of sound in the lubricant layer, m is the mode number of the resonant frequency and f_m is the resonant frequency (in Hz) of the mth mode. It can be seen from consideration of (2.3) that higher resonant frequencies correspond to thinner lubricant films. The thinnest film measurable by this approach is then limited by the upper operating frequency. In practice, the frequency is limited to below 40–60 MHz by the attenuation of the ultrasonic pulse in the bearing materials. This means that, if the lubricant layer is below 10 μm, the resonant frequency will be above the measurable range.

The region below the first resonance of a simple mass-spring system is known as the stiffness-controlled region because the deflection of the mass is dependent purely on the applied load and the spring stiffness. In the same way, it can be shown (Hosten 1991) that the interaction of ultrasound with a very thin layer (which would have a very high first resonant frequency) is governed by the stiffness of that layer. The stiffness per unit area, K, of a thin layer is given by

$$K = \frac{\rho c^2}{h}, \tag{2.4}$$

where ρ is the density of the layer and c the speed of sound in the layer. This spring stiffness can then be used in a quasi-static model of the interaction of ultrasound with the lubricant layer. In this way, Tattersall (1973) demonstrated that the reflection coefficient of the layer was given by the expression

$$R = \frac{z_1 - z_2 + \mathrm{i}\omega(z_1 z_2/K)}{z_1 + z_2 + \mathrm{i}\omega(z_1 z_2/K)}, \tag{2.5}$$

where z_1 and z_2 refer to the acoustic impedance of the materials either side of the lubricant layer. For identical materials either side of the lubricant film ($z_1 = z_2 = z'$), this reduces to

$$R = \frac{1}{\sqrt{1 + (K/\pi f z')}}. \tag{2.6}$$

Drinkwater *et al.* (1996) used the above analysis to determine the stiffness of dry contacts in which the interface between two rough surfaces, composed of air gaps and regions of asperity contact, acts like a layer of reduced stiffness. Measurement of the reflection coefficient and hence interface stiffness can give information about the degree of conformity of the contacting surfaces.

Figure 3 shows a comparison between the spring-model prediction and the continuum-model prediction for the reflection-coefficient spectra from a mineral oil layer surrounded by steel. It can be seen that the agreement between the two models at frequencies below resonance is excellent. In fact, within the accuracy of the computer, the predictions are identical if $f < \frac{1}{2} f_1$. Combining equations (2.4) and (2.6), the thickness of the layer is related to the reflection coefficient by

$$h = \frac{\rho c^2}{\pi f z'} \left(\frac{R^2}{1 - R^2} \right). \tag{2.7}$$

From (2.7), it can be seen that as R approaches unity, so the calculated thickness approaches infinity. In practical measurement, this means that the spring approach loses accuracy as the reflection coefficient approaches unity. The authors have found that, for practical purposes, $R < 0.9$ gives satisfactory results.

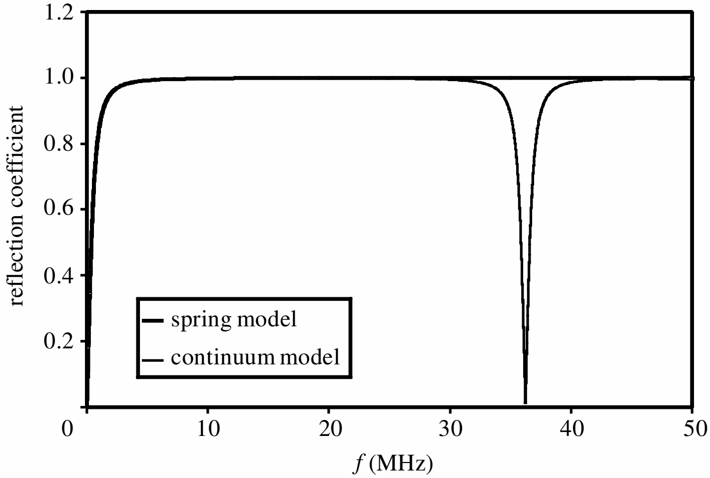

Figure 3. Comparison between spring-model and continuum-model predictions for the reflection-coefficient spectra from a mineral oil film between two steel elements.

3. Acoustic properties of a lubricant film

To determine the thickness of a lubricant film, using (2.7), it is necessary to know both the density of, and the speed of sound through, the lubricant. The speed of sound can readily be determined for oil in the bulk from a simple ToF measurement between a transducer and a flat reflector. For the oil used in these experiments (Shell Turbo T68), the acoustic properties at 20 °C are shown in table 1.

This value is suitable for the analysis of ambient fluid wedges or low-pressure hydrodynamic lubricant films. However, in EHD films, the lubricant is subjected to pressures high enough to cause solidification and a significant change in physical properties. The speed of sound through oil under EHD conditions is therefore very different to those in the bulk. The speed of propagation of a longitudinal wave through a medium depends on the density, ρ, and the bulk modulus, B, according to (see, for example, Povey 1997)

$$c = \sqrt{\frac{B}{\rho}}. \tag{3.1}$$

Jacobson & Vinet (1987) provide a model to determine the influence of pressure on the compression and bulk modulus of liquid lubricants. They give an equation of state to describe the behaviour of the lubricant under pressure, p,

$$p = \frac{3B_0}{x^2}(1-x)\mathrm{e}^{\eta(1-x)}, \tag{3.2}$$

and the bulk modulus under pressure is given by

$$B = \frac{B_0}{x^2}[2+(\eta-1)x-\eta x^2]\mathrm{e}^{\eta(1-x)}, \tag{3.3}$$

where B_0 is the bulk modulus at zero pressure, η is a lubricant-specific parameter and x is a function of the relative compression,

$$x = \sqrt[3]{\frac{\rho_0}{\rho_p}}, \tag{3.4}$$

Table 2. *Details of the transducers used in this work*

(The focal length and spot size (equation (5.5)) are determined for a centre-frequency wave in water.)

centre frequency f (MHz)	bandwidth (-6 dB points) (MHz)	element diameter D (mm)	focal length (in water) F (mm)	spot size (in water) d_f (mm)
10	4–17	12.7	76	921
25	17–32	6.35	54	522
50	40–75	6.35	25	121

where ρ_0 is the density at zero pressure and ρ_p the density at pressure p. These equations are used for the oil both in its liquid state and after it undergoes solidification, but employing different values of η and B_0.

Jacobson & Vinet used a high-pressure chamber to measure the compression of several liquid lubricants up to pressures of 2.2 GPa and determined the parameters for (3.2) and (3.3). For mineral oils, the bulk modulus increased from 1.5–1.7 GPa at atmospheric pressure to 20–30 GPa at EHD pressures, while the density increased by 20–30%. Therefore, from (3.1), the speed of sound within an EHD contact is likely to be some two to four times greater than that under ambient conditions.

Unfortunately, there are no test data available specifically for the lubricant used in these studies (Shell Turbo T68). However, parameters from a similar base oil (extracted from Jacobson & Vinet (1987)) have been used to determine the relationship between bulk modulus, density change and pressure. Table 1 shows the results of this analysis for two oil pressures, the lower corresponding to that experienced in the ball on flat apparatus, and the higher to that in ball-bearing contact.

4. Ultrasonic reflection measurement apparatus

An ultrasonic pulser-receiver (UPR) generates a high-voltage pulse that causes a piezoelectric transducer to be excited in mechanical resonance. The transducer then emits a broadband pulse of ultrasonic energy. As can be seen in figure 1, this ultrasonic pulse propagates through the bearing shell, is reflected from the lubricant layer and is received by the same transducer. Measurements were made using focused ultrasonic transducers with a range of different centre frequencies (10, 25 and 50 MHz). The transducers were immersed in a water bath and positioned so that the ultrasonic energy was focused through the water bath and bearing shell onto the lubricant layer. Table 2 details the bandwidth, geometry and focusing properties of the transducers used.

The pulse reflected from the lubricant layer was amplified and stored on a digital oscilloscope. The stored reflected signals were passed to a PC for processing. Figure 4 shows a schematic of the apparatus and data signals. Software (written in the LabView environment) was written to control the UPR, receive reflection signals from the oscilloscope, perform the required signal processing and display the required results.

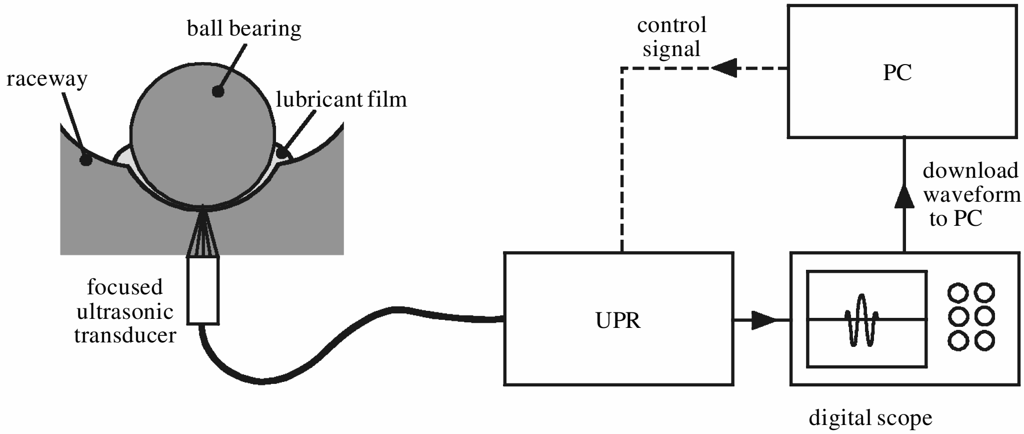

Figure 4. Schematic of ultrasonic pulsing and receiving system.

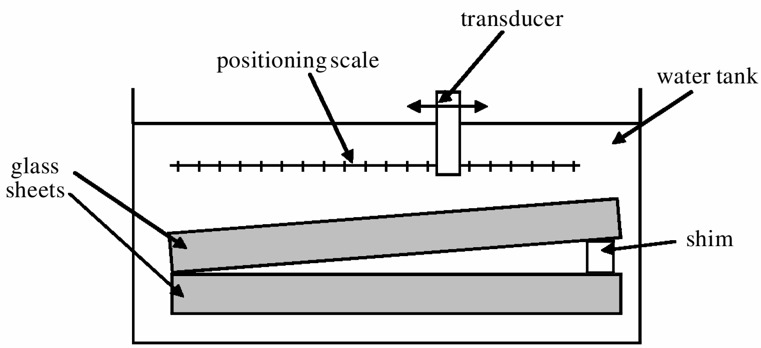

Figure 5. Schematic of the apparatus used for generating thin liquid films between two flat surfaces.

5. Measurement of a stationary liquid layer

Initial experiments were performed on thin liquid layers between stationary solid surfaces. Two configurations were chosen: a water wedge between optical glass flats, and an oil film between two eccentric cylinders. The former apparatus was used to produce a wide range of measurable liquid-layer thickness. The latter was designed to investigate the effects of the curvature of the solid surfaces and the use of a mineral oil as the liquid medium.

(*a*) *Liquid-wedge apparatus*

Shims of various thickness were used to separate sheets of float glass, which were then immersed in a water tank, as shown in figure 5. The ultrasonic transducers were also immersed in the water tank and mounted above the liquid wedge on a computer-controlled xy positioning frame. The ultrasonic pulsing and receiving was controlled by the apparatus as shown in figure 4.

The first step in the procedure was to determine the frequency characteristics of the incident ultrasonic pulse. The bottom glass plate was removed and the pulse reflected

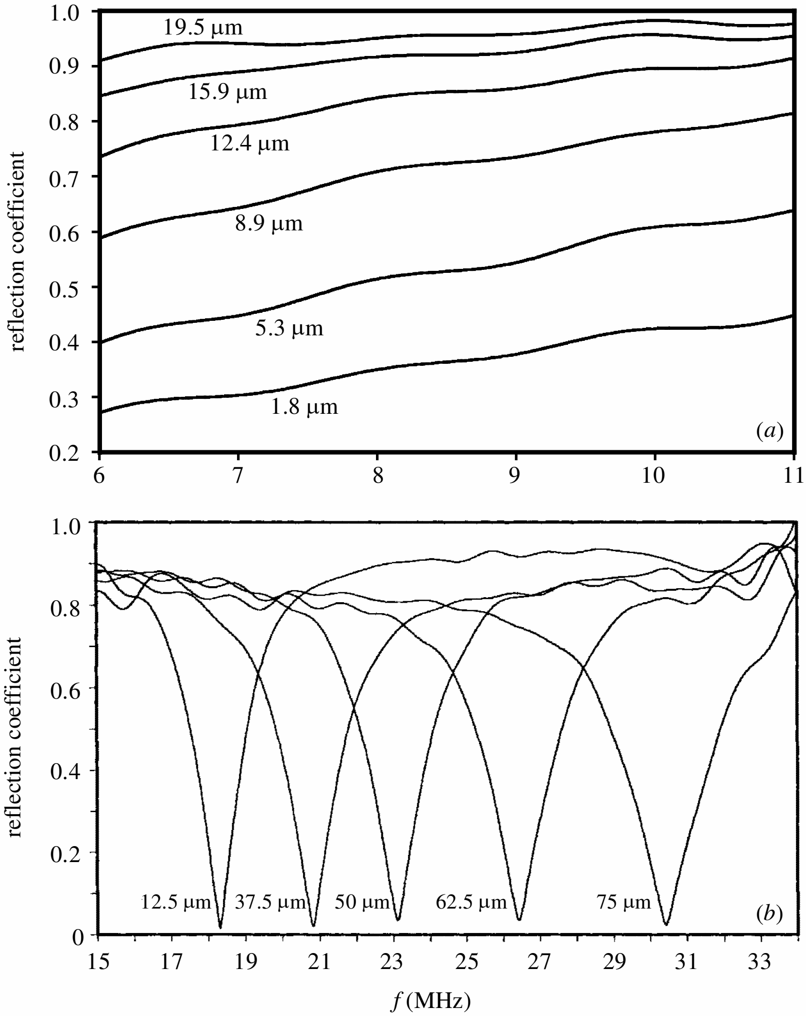

Figure 6. Measured reflection-coefficient spectra for a range of fluid film thicknesses using (*a*) a 10 MHz and (*b*) a 25 MHz centre-frequency transducer.

from the glass–water interface recorded. This time-domain signal was converted to the frequency domain via a fast Fourier transform and stored as a reference, A_{ref}. In this way,

$$A_{\mathrm{ref}}(f) = I(f)R_{\mathrm{gw}}, \tag{5.1}$$

where $I(f)$ is the frequency-response characteristic of the incident pulse and R_{gw} is the glass–water reflection coefficient given by (2.1). The bottom plate was then repositioned and measurements of the pulse reflected from the fluid film were recorded at various locations along the fluid wedge. This measurement signal, A_{meas}, can be written as

$$A_{\mathrm{meas}}(f) = I(f)R_{\mathrm{l}}(f), \tag{5.2}$$

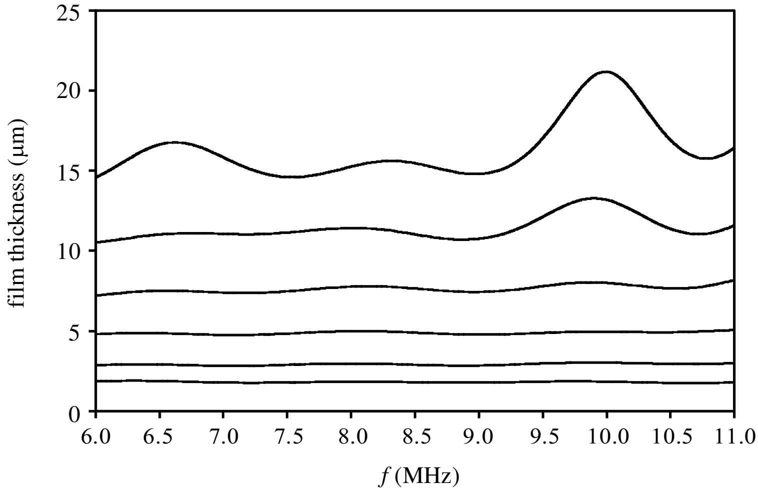

Figure 7. Film-thickness results determined by the application of the spring model to the reflection spectra of figure 6*a*.

where $R_l(f)$ is the reflection-coefficient spectrum of the lubricant layer, which can then be obtained by combining (5.1) and (5.2),

$$R_l(f) = \frac{A_{\mathrm{meas}}(f)}{A_{\mathrm{ref}}(f)} R_{\mathrm{gw}}. \tag{5.3}$$

Figure 6 shows two sets of reflection-coefficient spectra of liquid layers in a liquid-wedge experiment recorded using two different wide-band transducers for a range of film thickness. The thickness of the liquid layer at any measuring location was estimated from the geometry of the fluid wedge formed by the shim.

From figure 6*b*, it can be seen that, at higher film thicknesses, resonant frequencies (observed as minima in the reflection-coefficient spectra) are visible. When these resonances occur, the layer thickness can be readily determined from (2.3). For thinner liquid layers and/or when using lower-frequency transducers, the resonance is not observed as it occurs beyond the frequency range of the transducer. However, the shape of the reflection-coefficient spectrum below resonance can be used to predict the thickness of the liquid layer though the application of the spring model (equation (2.7)).

The stiffness of the layer can be measured over a range of frequencies with one transducer. As the stiffness at a single frequency is sufficient to enable prediction of the fluid-layer thickness, there is some redundancy in the data that can be used as a method of checking the accuracy of the spring model. Figure 7 shows the data given in figure 6*a* converted to film thickness via (2.7) over the bandwidth of the transducer. From figure 7, it can be seen that, for the lower film-thickness values, the curves are horizontal. This indicates a good measurement, as the film thickness cannot be a function of the frequency with which it is measured. However, for the top curve, the model does not correctly convert the reflection coefficient to a constant layer thickness. This is because, as the reflection coefficient tends to unity, so the spring model (equation (2.7)) becomes prone to numerical error. For the next curve down,

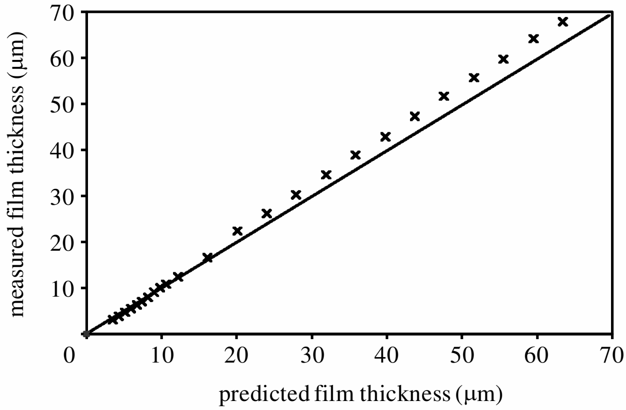

Figure 8. Film-thickness values measured by ultrasonic reflection compared with predictions of the thickness determined from the geometry of the fluid wedge.

at the higher frequencies, the same happens. This represents a limit of the spring-model approach, and to measure these thicker films either lower frequencies should be used, so R is lower, or higher frequencies used and the location of resonances determined.

Figure 8 shows ultrasonic measurements of the thickness of the liquid film compared with the geometrical prediction of the layer thickness. The data below 25 μm correspond to measurements performed using the spring model, while the data above 25 μm have been determined from the measurement of film resonances. Agreement is good (and a straight line is observed) but, in practice, the measured liquid-layer thickness tends to be slightly larger than that of the shim. This is likely to be caused by either some thin residual of water between the shim of the glass plate or by deformation of the shim where it has been cut. Each of these effects would tend to make the liquid layer thicker and hence explain the difference seen in figure 8.

(*b*) *Annular oil-film apparatus*

Figure 9 shows the apparatus to create an oil wedge between a ring and a shaft. The ring and shaft were made from aluminium. The ring–shaft interface had a nominal diameter of 70 mm and a diametral clearance of 100 μm. The shaft was spring loaded against the ring. By changing the position of the loading point, the gap thickness in the region above the transducer could be varied. The gap was filled with mineral oil (Shell Turbo T68) through a small filler hole. A 25 MHz focused transducer was located in a water bath beneath the assembly. The transducer was positioned so that the wave was focused on the fluid layer.

Figure 10 shows a plot of the film thickness determined from the reflection measurements for various angular positions of the loading point relative to the ultrasonic measurement position. In this case, all the films were measured using the film-resonance method. The theoretical width of annular gap, h, between eccentric cylinders is obtained from geometry,

$$h = R_1\left[1 - \left(\frac{R_1 - R_2}{R_1}\right)^2 \sin^2\phi\right]^{1/2} - R_2 + (R_1 - R_2)\cos\phi, \tag{5.4}$$

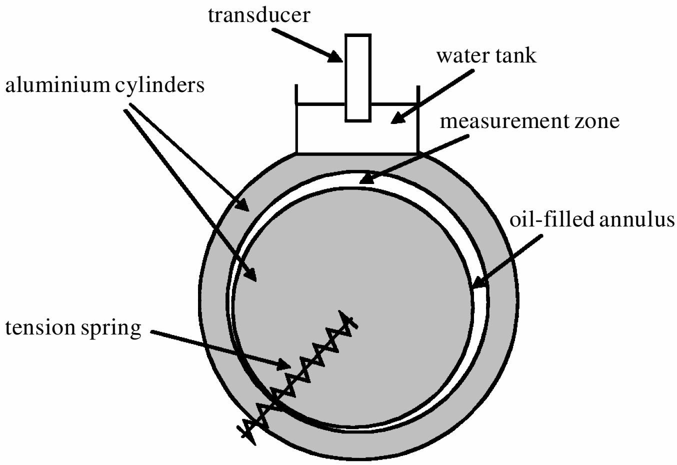

Figure 9. Apparatus used to create an annular oil film for ultrasonic-reflection experiments.

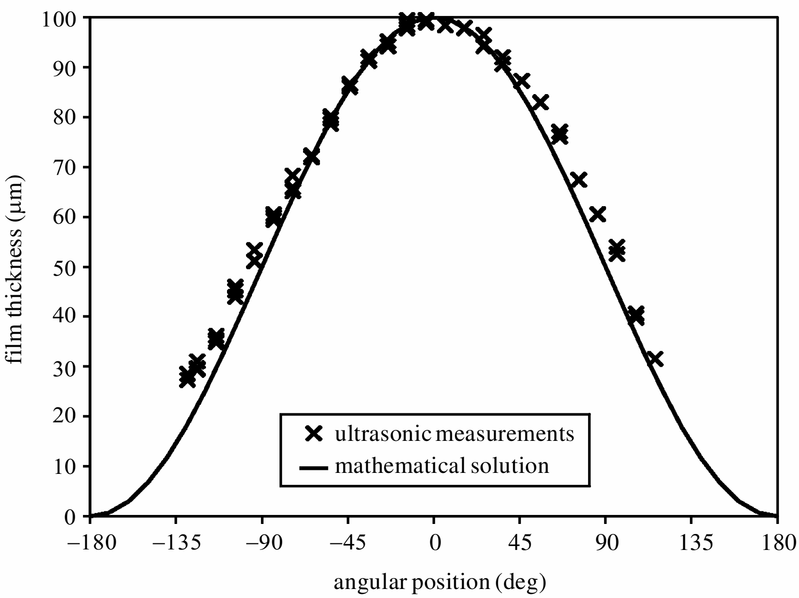

Figure 10. Film-thickness values measured by ultrasonic reflection for an annular film between two eccentric cylinders. Experimental results are compared with the geometrical solution.

where R_1 and R_2 are the radii of the bush and shaft, respectively, and φ is the angle measured from the point of contact. Equation (5.4) is included in figure 10 for comparison, and a good correlation was observed.

To reduce the effects of the curvature of the lubricated interface, the transducers used in this experiment focus the wave onto the oil layer. The spot size of the focused wave varies with frequency according to the relationship (Silk 1984)

$$d_{f(-6\ \mathrm{dB})} = \frac{1.028Fc}{fD}, \tag{5.5}$$

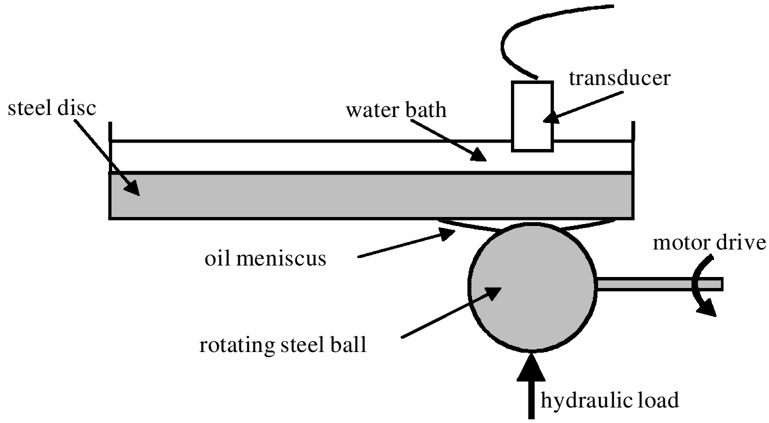

Figure 11. Schematic of the experimental apparatus used to generate an EHD film for a ball sliding on a flat.

where d_f is the diameter of the spot size (where the signal has reduced to -6 dB of its peak value), F is the transducer focusing length, f is the wave frequency, c is the speed of sound in water and D is the diameter of the piezoelectric element. For a 25 MHz frequency wave, this corresponds to a spot size in water of 522 µm. This is small compared with the radius of the shaft/bush apparatus. So, essentially, the transducer is receiving signals back from a flat region of constant film thickness and the curvature has little effect. The good agreement between experimental and theoretical liquid-layer thickness shown in figure 10 further validates this hypothesis.

6. Measurement of dynamic lubricating films

An EHD lubricant film forms between rolling or sliding non-conformal contacts. Typically, the thickness of the oil layer is in the region of 0.1–1 µm. This is within the range of the possible measurement by ultrasonic means and within the spring-model regime of ultrasonic response. In this work, EHD films have been measured in a model rig containing a single stationary lubricated contact and in a rolling-element bearing where the lubricant film is transient (with respect to the transducer).

(*a*) *Ball on disc EHD lubrication apparatus*

Figure 11 shows the apparatus used to generate a single EHD lubricated contact. A steel ball is supported on rollers and hydraulically loaded onto the underside of a steel disc. The ball is rotated at constant speed by an electric motor through a gear box and quill shaft. The steel disc is held stationary such that the ball is completely sliding against the underside of the disc. This apparatus is a modified 'optical elastohydrodynamic' apparatus (originally developed by Cameron & Gohar (1966)), where the semi-reflective rotating glass disc has been replaced by a station-

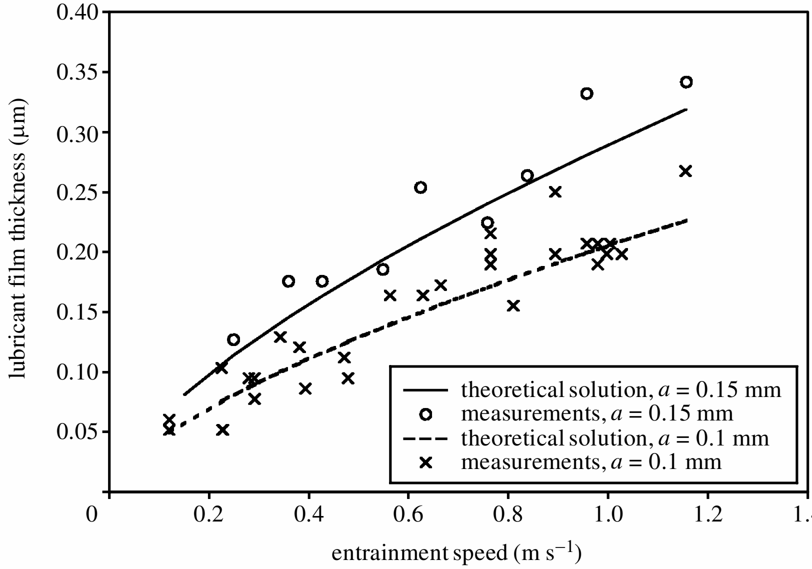

Figure 12. EHD lubricant-film-thickness variation with ball-sliding speed for two loads. Ultrasonic-reflection measurements are compared with a theoretical solution.

ary steel disc, and the microscope replaced by an ultrasonic transducer. The ball–disc contact was flooded with a mineral oil (Shell Turbo T68), which is entrained into the contact to form an EHD film. A 50 MHz focused transducer was mounted above the contact in a water bath. The transducer was positioned directly above the contact region and at a distance such that the wave is focused on the ball–disc interface.

Reflected pulses were recorded from the disc–ball interface as the ball rotational speed and load were varied. A reference spectrum was obtained by recording the pulse reflected when the ball was out of contact with the disc but the disc was still flooded with oil. As in the wedge experiment, the reference and lubricant-layer measurements were used in (3.2) to obtain the reflection-coefficient spectrum of the lubricant layer. A series of reflection coefficient spectra were then recorded from the disc–ball contact as the sliding speed was increased. The spring model (equation (2.7)) was used to determine the film thickness from these spectra. An automated LabView interface was written to record the spectra, divide by a reference and convert the data into film thickness (as shown in figure 6). As discussed in § 3 above, the density and speed of sound of the oil used in (2.7) are those at contact pressure (in this case, 0.8 GPa) and not those in the bulk.

Figure 12 shows the measured variation in film thickness determined from the reflection coefficient spectra as the speed is increased for two load cases (giving contact radii, a, of 0.1 and 0.15 mm). For this lubrication case, a theoretical prediction of the oil-film thickness can be obtained from the regression equations of Dowson & Higginson (1966). The central film thickness, h_c, is expressed as a regression equation consisting of five non-dimensional parameters,

$$H_c = 2.69U^{0.67}G^{0.53}W^{-0.067}(1 - 0.61e^{-0.73k}), \tag{6.1}$$

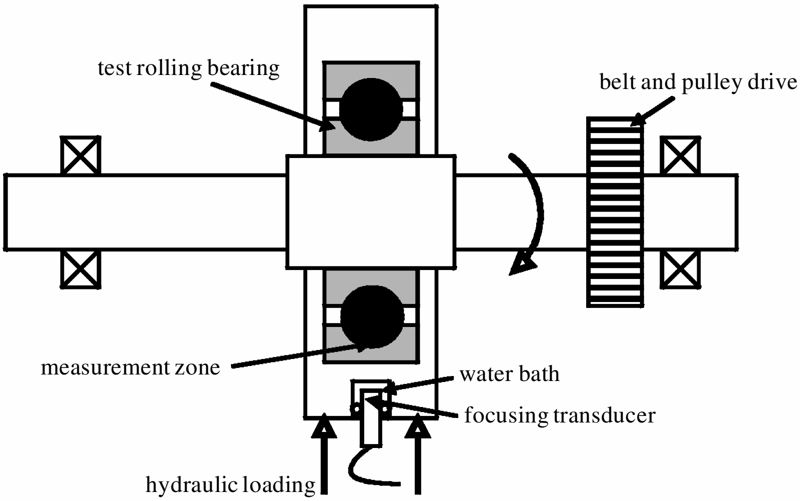

Figure 13. Schematic of apparatus used for ultrasonic measurements of film thickness in rolling-element bearings.

where

$$H_c = \frac{h_c}{R_x}, \qquad W = \frac{P}{2E^* R_x^2}, \qquad U = \frac{\eta_0 u}{2E^* R_x},$$

$$k = 1.03\left(\frac{R_y}{R_x}\right)^{0.64}, \qquad G = 2\alpha E^*,$$

where E^* is the reduced modulus, R_x and R_y are the reduced radii in the parallel and transverse directions, u is the mean surface speed, P is the applied contact load, η_0 is the viscosity of oil in the inlet and α is its pressure-viscosity coefficient.

The predictions of (6.1) for the two load cases are plotted in figure 12. The agreement between the theoretical solution and the experimental data is good. Typically, the measured data are repeatable to within $\pm 20\%$. The oil film generated by the apparatus is not completely stable. To directly observe the lubricant film, the steel disc was replaced by the glass disc and the film monitored visually. A certain amount of film fluctuation was observed during normal running (caused by a slight processing of the ball on its supporting rollers). Notwithstanding, the results are encouraging because this indicates that both the integrity of the ultrasonic method, and also that the speed of sound of the lubricating oil, compares well with predictions from high-pressure measurements.

(*b*) *Rolling bearing lubricant-film measurement*

The duration of the ultrasonic pulses used in this work was short. Spatially, the pulse width was typically three wavelengths in length. A 25 MHz wave travelling in steel results in a pulse duration of 0.1 μs. Therefore, it is feasible to record the film thickness in a rolling element moving past the focal point of an ultrasonic transducer.

The approach has been used to measure film thickness in a 6410 deep-groove ball bearing. Figure 13 shows a schematic of the bearing and the positioning of the transducer. The 25 MHz transducer (see table 2 for details) was located in a small

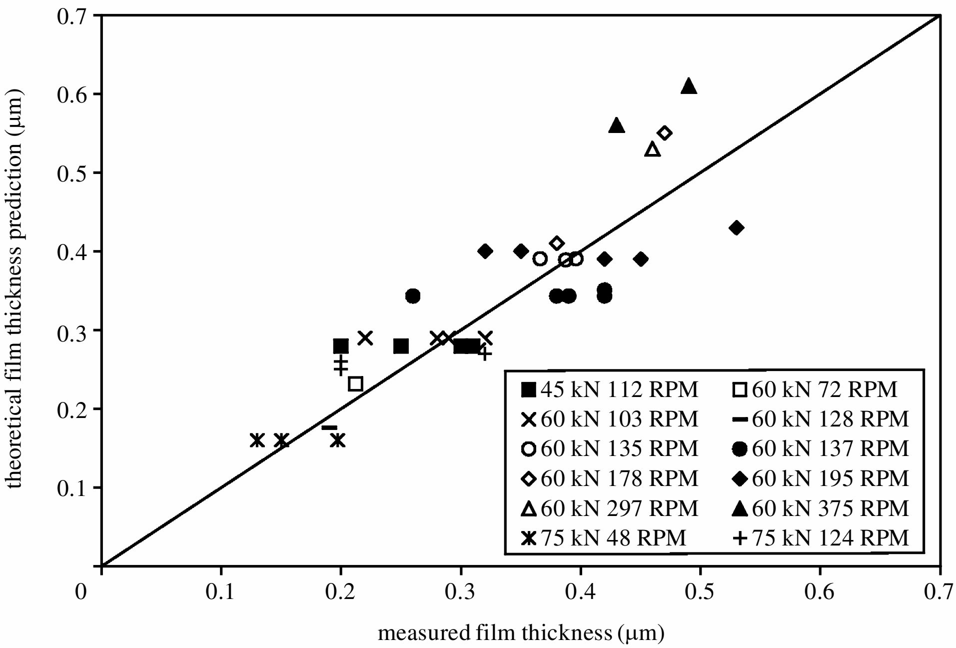

Figure 14. Comparison of rolling-element-bearing film thickness measured by an ultrasonic means with EHD theoretical solution.

water bath machined out of the bearing bush. The transducer was spherically focused and positioned in the water bath so that the wave was focused on the contact between a ball and the raceway. The bearing was loaded hydraulically, and rotated by an electric motor. The bearing cavity was flooded with a mineral oil (Shell Turbo T68). A thermocouple was positioned on the outer raceway near the ultrasonic measurement point. The bearing was loaded to give a contact eclipse between the ball and the outer raceway of semi-major and semi-minor axes 11.5 and 0.8 mm, respectively. The bearing was then rotated and the transducer set to continuously pulse at a repetition rate of 10 kHz.

The focal spot size of the transducer was *ca.* 500 μm. The maximum rotational speed of the bearing in these tests was 375 rpm. At this rotational speed, the region of focused ultrasound would remain completely within the lubricated contact for *ca.* 0.5 ms. This allowed approximately five discrete pulses of ultrasound to be reflected from the lubricant film, as each ball passed through the measurement zone. Initially, a reference reflected pulse was recorded when the bearing was stationary and there was no ball located over the measurement zone (although the cavity was still filled with oil). If a pulse is incident on an EHD film, its amplitude is reduced because part of the wave has been transmitted through the oil film into the ball. By continuously monitoring the signals reflected from the raceway measurements of the ball-raceway contacts were made. Of the five pulses reflected from the ball-raceway contact, the lowest amplitude pulse was selected for analysis. This pulse corresponds to the thinnest measured film thickness. It should be noted that the resolution of the focused transducer and the measurement interval are such that it was not possible to record the horseshoe-shaped minimum film-thickness constriction at the contact exit. The selected pulse was analysed, firstly using (5.3) to calculate the reflection

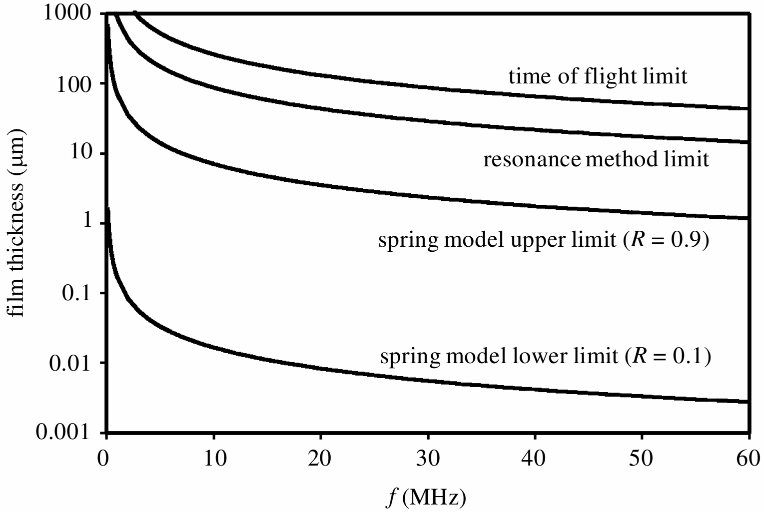

Figure 15. Limits of operation for film-thickness measurement by ultrasonic means.

coefficient and then using the spring model to predict the film thickness. The speed of sound in the oil in the contact is set to $c = 4400\ \mathrm{m\,s^{-1}}$ (determined using the method described in § 3).

Figure 14 shows the measured film thickness plotted against a prediction of the film thickness from the solution for an elliptical contact of Dowson & Higginson (1966) (equation (6.1)). The bearing and lubricant heats up during operation. The temperature of the entrained oil was monitored and used to determine an appropriate viscosity with which to calculate the film thickness in (6.1). This was why the results were not plotted as film thickness against speed (as in figure 11), because viscosity was varying throughout the test.

The results in figure 14 show good agreement between measured lubricant-film thickness and that predicted by the EHD regression equations. The spread along the x-axis shows the variation in film for nominally the same bearing operating conditions (load, speed and oil temperature). At this stage, it is not clear whether this scatter is a result of the measurement process or whether this is the actual film thickness in the bearing. It is worth noting that this measurement process could be improved if the exact position of the ball could be related to the ultrasonic measurement, so allowing a more precise measurement of the minimum film thickness.

7. Discussion: limits of operation

It is instructive to consider the limits of film-thickness measurement for a given ultrasonic frequency. The simplest method for determining the thickness of a lubricant layer, providing it is sufficiently thick, is by measuring the ToF through the layer. A reflection occurs at the layer front face and another at the back face. The thickness is readily obtained from the time between the arrival of these pulses multiplied by the acoustic velocity. For the method to work, the two reflected pulses must be sufficiently discrete, and so the propagation distance must be greater than the pulse width, and so the minimum film thickness detectable by this method is given by

$$h > \frac{nc}{2f}, \tag{7.1}$$

where n is the number of wavelengths in the ultrasonic pulse, typically around three. This measurement limit is plotted in figure 15 as the minimum frequency required to measure a given film thickness. Conventional ultrasonic non-destructive testing is generally restricted by losses in the propagating media (steel in this case) to frequencies below 40–60 MHz. From (7.1), this means that the ToF method is only practically useful for layers of thickness greater than 40 μm. This is thicker than that found on most commercial bearing systems.

A lubricant layer resonates at a frequency given by (2.3). This equation also gives the low thickness limit for this technique as the upper frequency output by a given transducer determined the lowest measurable layer thickness. This limit is also shown in figure 15 for the first through-thickness mode (i.e. $m = 1$). So, using the highest practical frequencies, films greater than 10 μm are measurable by the film-resonance method.

The upper limit of film thickness measurable using the spring-model approach is reached as R tends to unity in (2.7). Experience shows that measurements of R less than 0.9 yield satisfactory results. The lowest film thickness measurable will depend on the smallest possible value of R. This, in turn, will depend on the signal-to-noise ratio. A signal-to-noise ratio of 60 dB is typical for conventional ultrasonic non-destructive testing systems. The noise amplitude is thus $\frac{1}{1000}$ of the pulse amplitude. In the case of a measurement on a real bearing, the achievable signal-to-noise ratio will be reduced by the transducer-bearing reflection coefficient and by the attenuation in the bearing shell. This means that the pulse incident in the fluid film will be reduced from that output by the transducer by a factor of five (as a conservative estimate). Further, it is assumed that for reliable measurements, a pulse reflected from the oil layer should have an amplitude approximately 20 times that of the noise level. This results in a minimum measurable reflection coefficient of $R = 0.1$. The relationships between the film thickness and the transducer frequency, for the upper limit when $R = 0.9$ and the lower where $R = 0.1$, are shown in figure 15.

This indicates that, for a measuring frequency of 60 MHz, a lubricant film of 2 nm is measurable. This opens up the possibility of measuring boundary-type lubricant films between machine elements. However, the models presented here are designed for a parallel-sided oil layer completely separating the smooth bearing surface. If, at these low film thickness, metal-to-metal contact occurs, then transmission will occur at this location. In this case, some alternative method for interpreting reflection coefficients will be needed.

8. Conclusions

Ultrasound is reflected from the lubricant film between bearing surfaces. It is possible to deduce the thickness of the lubricant film by comparing the frequency spectrum of the reflected pulse with that of the incident pulse. The response of the lubricant layer to an ultrasonic pulse has been modelled using both a spring-model method and a continuum method. The spring model is suitable for thin lubricant films in which the wavelength of the ultrasound is much larger than the layer thickness. The continuum model is valid for all layer thicknesses and allows the prediction of the through-thickness resonant frequencies of the lubricant layers. Measurement of these through-thickness resonances also provides a means of measuring the layer thickness.

Experiments on fluid layers between flat plates and eccentric rings have demonstrated that ultrasound can suitably measure a wide range of liquid layer thickness. The EHD lubricant film that forms between a ball sliding on a flat surface has also been measured ultrasonically. The measured results agree well with theoretical predictions, taking into account the greatly increased speed of sound through the lubricant when it is under high pressure. Films in the range 50–500 nm were recorded.

The approach can also be used to measure film thickness in a rolling-element bearing. An ultrasonic transducer was mounted on the bearing raceway and the pulse reflected from the lubricant layer, generated as the ball passes, was recorded. Again, measured results agreed well with theoretical predictions.

The lower limit of measurement by this method depends on the signal-to-noise ratio. A continuum model showed that using frequencies up to 60 MHz permits the measurement of 2 nm thick films. There is no upper limit to film-thickness measurement. Thick films (greater than 40 μm) can be measured by a time-of-flight method. Films of intermediate thickness can be determined by measuring the frequency at which the film resonates.

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