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Operating Limits for Acoustic Measurement of Rolling Bearing Oil Film Thickness

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Abstract

An ultrasonic pulse striking a thin layer of liquid trapped between solid bodies will be partially reflected. The proportion reflected is a function of the layer stiffness, which in turn depends on the film thickness and its bulk modulus. In this work measurements of reflection have been used to determine the thickness of oil films in elastohydrodynamic lubricated (ehl) contacts. A very thin liquid layer behaves like a spring when struck by an ultrasonic pulse. A simple quasi-static spring model can be used to determine the proportion of the ultrasonic waves reflected.

Experiments have been performed on a model ehl contact between a ball and a flat surface. A transducer is mounted above the contact such that the ultrasonic wave is focused onto the oil film. The reflected signals are captured and passed to a PC for processing. Fourier analysis gives the reflection spectrum which is then be used to determine the stiffness of the liquid layer and hence its thickness. In further testing, an ultrasonic transducer has been mounted in the housing of a deep groove ball bearing to measure the film generated at the outer raceway as each ball passes. Results from both the ball-flat and ball bearing measurements agree well with steady state theoretical ehl predictions. The limits of the measuring technique, in terms of the measurable rolling bearing size and operating parameters, have been investigated.

Introduction

The durability of a rolling element bearing depends crucially on the formation of a film of lubricant between the rolling elements and the raceways. If this film fails to form, by for example lubricant starvation or supply interruption, the surfaces of the bearing components will come into contact; surface damage, wear, or in extreme cases seizure, can result. In normal operation, elastohydrodynamic oil films are formed between the opposing surfaces. The counter-formal nature of the contact results in high pressure, elastic deformation at the contact, and an increase in the viscosity of the lubricant. The films formed are usually thin (sub-micron) but are thick enough to separate the asperities on opposing smooth bearing surfaces.

The theoretical prediction of the formation of ehl films is well advanced. The regression equations of Hamrock and Dowson and [1981] provide a suitable method for the determination of film thickness in steady state Newtonian ehl smooth surface contacts. In recent years there have been many numerical studies to determine the effects of transient conditions, surface roughness, and non-Newtonian lubricant behaviour. The most flexible methods for ehl film thickness measurements have been by optical means. In the work of Cameron and Gohar [1966], optical interferometry was used to determine film thickness as well as spatial film variation. The use of lasers to fluoresce a lubricant film has also been used to determine film thickness [Richardson and Borman 1991]. However, the requirement for transparency (i.e. the need for an optical window in the machine element) has meant that these methods are rarely used outside the laboratory.

In this work a method based on the transmission of sound through the machine element, and its reflection at the oil film, is proposed as a method for measuring oil film thickness. The method offers non-invasive film measurement; does not require the electrical isolation of the surfaces, or an optical window.

Background

Ultrasonic Reflection from a Thin Liquid Layer

When ultrasound is incident on a boundary between two different media, some of the energy is reflected and some transmitted. The proportion of any incident signal reflected (known as the reflection coefficient, *R*) is given by:

$$R = \frac{z_1 - z_2}{z_1 + z_2} \tag{1}$$

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.

If ultrasound is incident on a layered system then some of the wave will be reflected at the front face of the layer and some at the back face. There will be a series of reflections as the wave bounces inside the layer (shown schematically in figure 1). If the lubricant layer were sufficiently thick (or the ultrasonic frequency high enough) then these reflections are discrete in time. So that if the speed of sound in the lubricant is known, the thickness of the lubricant film can be determined by measuring the time-of-flight between the two reflections. For thin layers, the reflected pulses overlap and it becomes impossible to distinguish the discrete reflections.

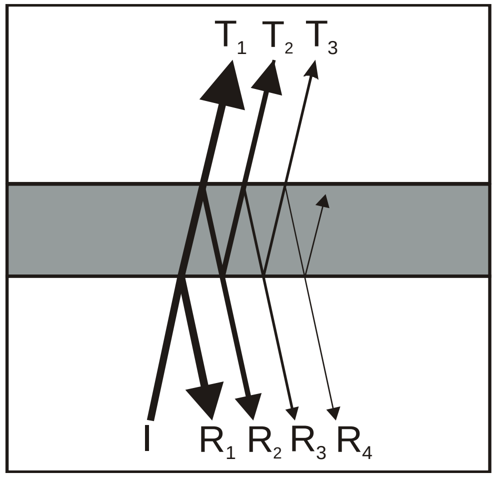

Figure 1. Schematic illustration of the reflection of an ultrasonic wave from a layer between two media.

In this work an alternative method is used. If the lubricant film thickness is very thin then the layer behaves like a spring and the reflection of the wave depends on the spring stiffness.

By considering the equilibrium of forces and compatibility at the boundaries of the layer during the passage of the wave, it is possible to show the dependence of the reflection on interface stiffness [Kendal & Tabor 1971]. In the same way, Tattersall [1973] demonstrated that the reflection coefficient of a layer was given by the expression:

$$R = \frac{z_1 - z_2 + i\omega(z_1 z_2 / K)}{z_1 + z_2 + i\omega(z_1 z_2 / K)} \tag{2}$$

where K is the stiffness of the layer and ω is the angular frequency (ω=2πf) of the incident wave. For identical materials either side of the lubricant film (z_1=z_2=z) this reduces to:

$$R = \frac{1}{\sqrt{1 + \left(K/\pi f z\right)^2}} \tag{3}$$

This relationship has been used to determine the stiffness of dry contacts study [Kendall & Tabor 1971, Krolikowski & Szczepek 1991, Drinkwater et al 1996]. In this case 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. The approach is equally applicable to a homogenous layer of reduced stiffness such as a thin oil film.

The Stiffness of a Lubricant Film

The stiffness of a layer, expressed per unit area, is defined by:

$$K = -\frac{\mathrm{d}\,p}{\mathrm{d}\,h} \tag{4}$$

where p is the applied pressure, and h is the approach of the surfaces (in other words the film thickness). For a liquid the bulk modulus is defined by:

$$B = -\frac{\mathrm{d}\,p}{\mathrm{d}V\,/V} \tag{5}$$

If the sound wave is large compared to the layer thickness, then a liquid layer is constrained to deform across it thickness only (i.e. the fluid area remains constant). Then $\mathrm{d}V/V=\mathrm{d}h/h$, and;

$$B = -h\frac{\mathrm{d}\,p}{\mathrm{d}\,h} \tag{6}$$

Combining (4) and (6) gives;

$$K=\frac{B}{h} \tag{7}$$

The speed of sound through a liquid depends on the bulk modulus and density, ρ (see for example Povey [1997]):

$$c = \sqrt{\frac{B}{\rho}} \tag{8}$$

The layer stiffness is also sometimes written as [Hosten 1991]:

$$K = \frac{\rho c^2}{h} \tag{9}$$

Combining (3) and (7) and rearranging, gives the film thickness in terms of the reflection coefficient and properties of oil and surrounding media.

$$h = \frac{B}{\pi f z}\sqrt{\frac{R^2}{1-R^2}} \tag{10}$$

The film thickness is thus a function of the stiffness of the oil (the density is not a contributing factor), and the acoustic impedance of the surrounding material. The reflection coefficient varies with the wave frequency, f but clearly the film thickness measured must not. It is important to note that it is the bulk modulus of the oil in the contact that must be used here; and that in elastohydrodynamic contacts this can be significantly different to that in the bulk.

Ultrasonic Reflection Apparatus

An ultrasonic pulser receiver (UPR) generates a voltage signal that causes a piezoelectric transducer to be excited. The pulse characteristics are specified in a control signal sent from the PC at the start of the measurement series. The transducer emits a broad band frequency pulse that is reflected from the interface and received by the same transducer.

In this work wide band immersion transducers were used. A spherical curved lens bonded to the piezo-electric element causes the wave to be focused. Depending on the experiment, two transducers were used, with central frequencies of 50 and 25 MHz. The former provides a higher resolution and so was used for smaller contact patch sizes. The transducer is immersed in a water bath and positioned so that the wave is focused on the lubricant layer.

The reflected pulse is amplified and stored on a digital oscilloscope. The stored reflected signals are passed to the PC for processing. Figure 2 shows a schematic diagram of the apparatus and data signals. Software (written in the LabView software environment) has been written to control the UPR, receive reflection signals from the oscilloscope, perform the required signal processing, and display appropriate results.

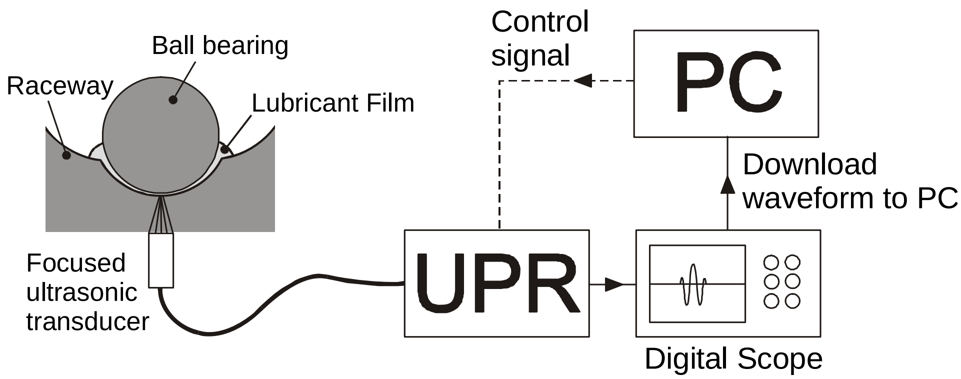

Figure 2. Schematic representation of ultrasonic pulsing and receiving system.

The procedure starts with the recording of a reflected signal when the surfaces are out of contact (i.e. at a steel/oil interface). A Fourier transform is performed to give an amplitude frequency spectrum. This steel/oil reflection spectrum is then divided by the reflection coefficient for a steel/oil interface (R=0.95 as obtained from equation 1) to give the steel/air reflection (i.e. the incident signal). Then a given lubrication condition is created and reflected signals are recorded. Each reflected signal is Fourier transformed and then divided by the reference spectrum. This gives a reflection coefficient spectrum. This is then processed using equation (10) to give film thickness directly.

Measurements from a Model EHL Contact

Initial experiments were performed on a simple ball on flat contact. A conventional optical ehl apparatus has been modified to hold a transducer as shown in figure 3.

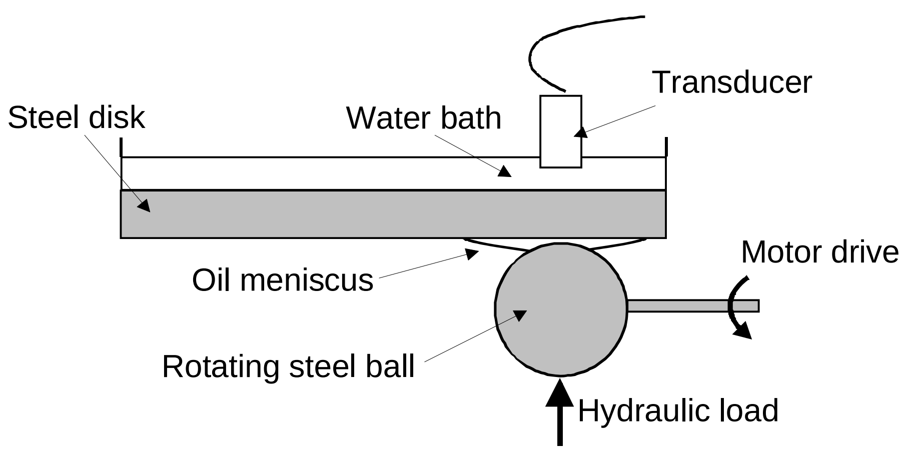

Figure 3. Schematic diagram of the experimental apparatus used to generate an elastohydrodynamic film for a ball sliding on a flat.

A steel ball is supported on rollers and hydraulically loaded onto the underside of a steel disk. The ball is rotated at constant speed by an electric motor through a gear box and quill shaft. The steel disk is held stationary such that the ball is completely sliding against the disk underside. The ball is flooded with a mineral oil (sheet Turbo T68), which is entrained into the contact to form a film. A 50 MHz focusing transducer is mounted above the contact in a water bath (the water is required to couple the transducer to the steel disk). The transducer is positioned directly above the contact region and at a distance such that the wave is focused on the ball/disk interface.

Figure 4 shows a series of pulses (as amplitude against frequency spectra) reflected from this contact. The reference spectrum is that which is reflected back when the ball is not in contact (and subsequently divided by the reflection coefficient for steel-oil to give the incident pulse). The next three spectra are recorded when the ball is in contact and an oil film is formed as the ball the sliding speed is increased.

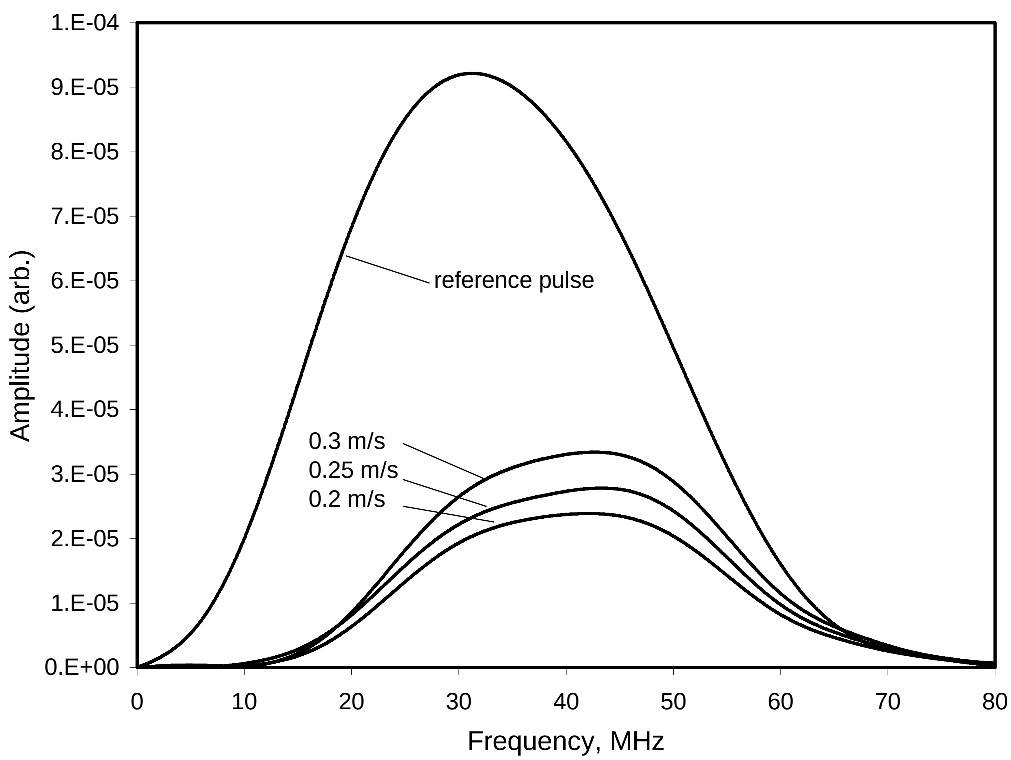

Figure 4. Four reflected pulses in the frequency domain. The reference pulse is reflected from a steel oil interface (with no ball in contact). The three other pulses are recorded as the ball sliding speed is increased.

Each of the spectra recorded at an oil film is divided by the reference spectrum to give the reflection coefficient spectrum; this is plotted in figure 5. A single reflected spectrum (normalised by the maximum amplitude) is also included is to indicate the bandwidth of the transducer. Within the frequency range 25 to 55 MHz there is a clear frequency dependence of the reflection coefficient. Outside of this range the signal energy is low, so the reflection coefficient is subject to noise.

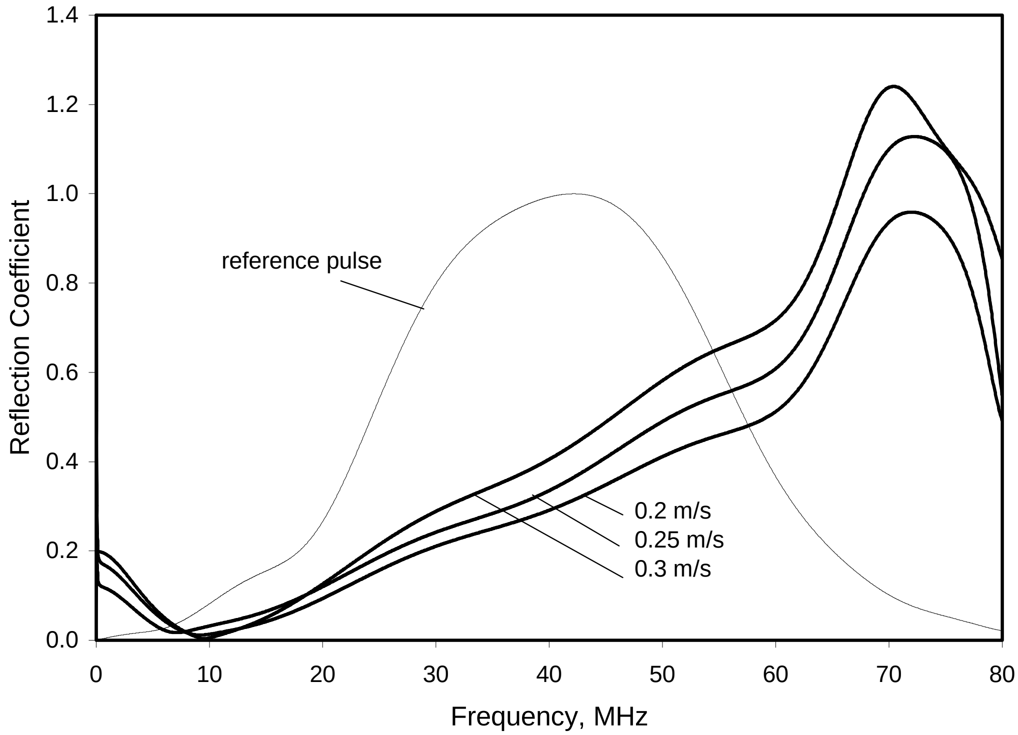

Figure 5. Reflection coefficient spectra for the three pulse reflected from oil films (obtained by dividing each pulse spectra of figure 4 by the reference spectrum).

Equation (3) is then used to determine the stiffness of the film measured at each of these frequencies from the reflection coefficient (as shown in Figure 6). It can be seen that the frequency dependence is now almost all removed. The calculated film stiffness should be independent of the measuring frequency.

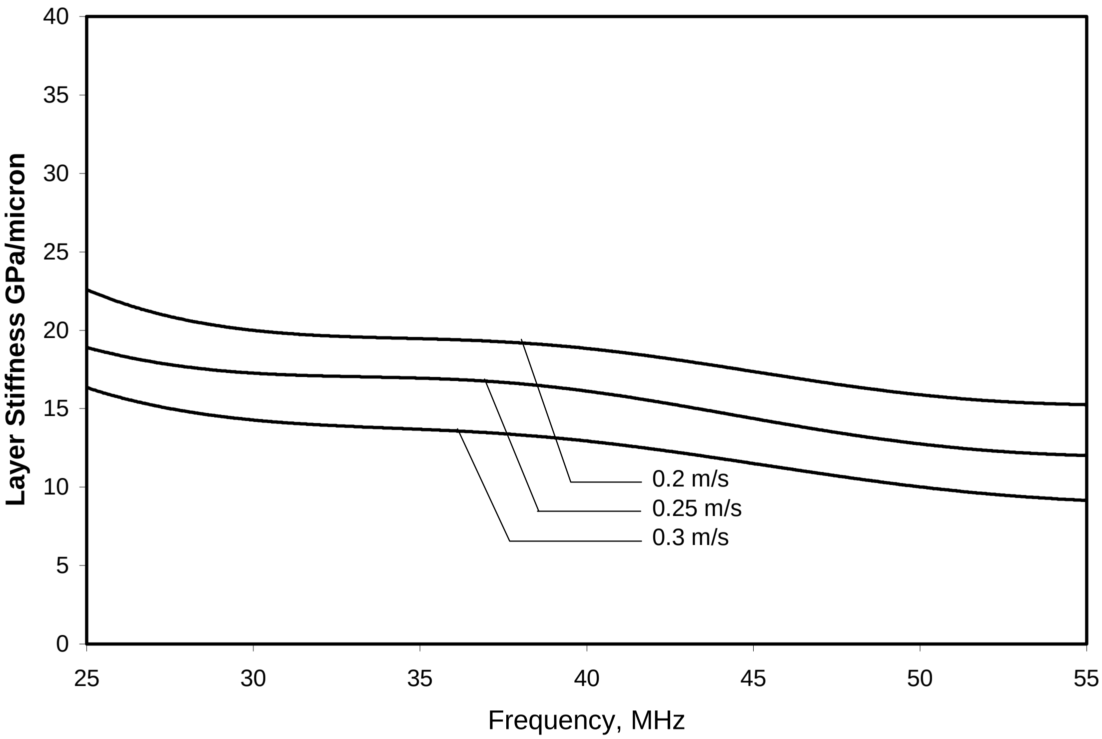

Figure 6. Layer stiffness against frequency calculated using the spring model (equation 3) from reflection data for three ball rotational speeds.

To obtain the film thickness from the stiffness, the bulk modulus of the oil in the contact is needed (equation 7). It is simple to obtain the modulus of the oil in the bulk by measuring the speed of sound (by timing a pulse travelling over a known distance). For T68 at room temperature the speed of sound turns out to be 1450 m/s and from equation (8), given that the density is 876 kg/m3, gives a bulk modulus of 1.84 GPa. However, the oil in the contact is under high pressure and its modulus changes significantly

Jacobson and Vinet [1987] provide a model to determine the influence of pressure on the compression and bulk modulus of liquid lubricants. Their results are based on the use of a high pressure chamber to measure the compression of several liquid lubricants up to pressures of 2.2 GPa. For mineral oils the bulk modulus increased from around 1.5-1.7 GPa at atmospheric pressure to 20-30 GPa at EHD pressures, whilst the density increased by about 20% to 30%. The modulus determined from a pressure cell will be the isothermal bulk modulus. Whereas the acoustic velocity is a function of the adiabatic bulk modulus. In this work the wave power is very small (and the wave is passing through a liquid) so little heating is expected. For this work it is a reasonable approximation to compare the two values.

Unfortunately, there is no test data available specifically for the lubricant used in these studies (Shell Turbo T68). Instead, a different approach has been used. The theoretical solution for the film thickness in a point elastohydrodynamic contact given by Hamrock and Dowson [1981] is used. The central film thickness, h_c is expressed in terms of five non-dimensional parameters:

$$H_c = 2.69 U^{0.67} G^{0.53} W^{-0.067} \left(1 - 0.61 e^{-0.73 k}\right) \qquad (11)$$

where: $H_c = \dfrac{h_c}{R_x}$, $W = \dfrac{P}{2E^* R_x^2}$, $U = \dfrac{\eta_o u}{2E^* R_x}$, $k = 1.03\left(\dfrac{R_y}{R_x}\right)^{0.64}$, $G = 2\alpha E^*$

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 bulk modulus of the oil in the contact is then chosen so as to minimise the least square error between the experimentally measured film thickness and that determined from equation (11). A value of B=25.8 GPa gives the best fit to the data (with a correlation coefficient of R^2=0.88). This value of the bulk modulus is in keeping with the data of Jacobson and Vinet for a mineral oil under pressure. Figure 7 shows the correlation between the measured and theoretical data using this single value of B.

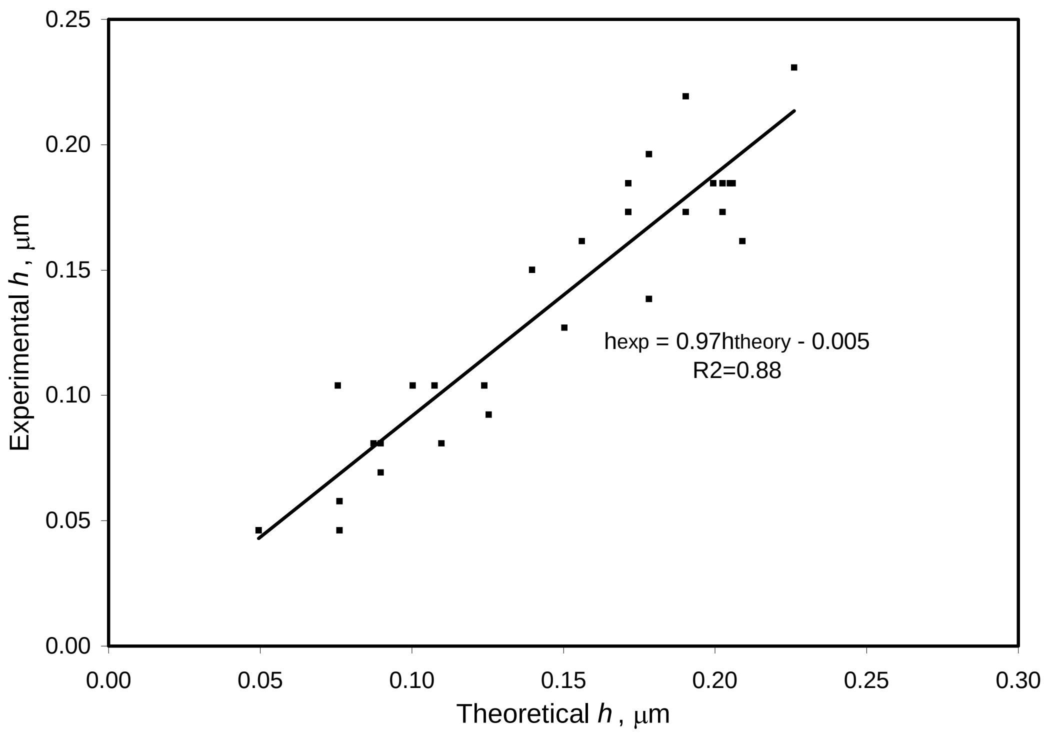

Figure 7. Measured film thickness compared against theoretical predictions (equation 11). A single value of the bulk modulus of the oil in the contact has been chosen to minimise the least square error.

The variation of measured film thickness with speed for two different sized contacts is shown in figure 8 (the bulk modulus again is set to 25.8 GPa).

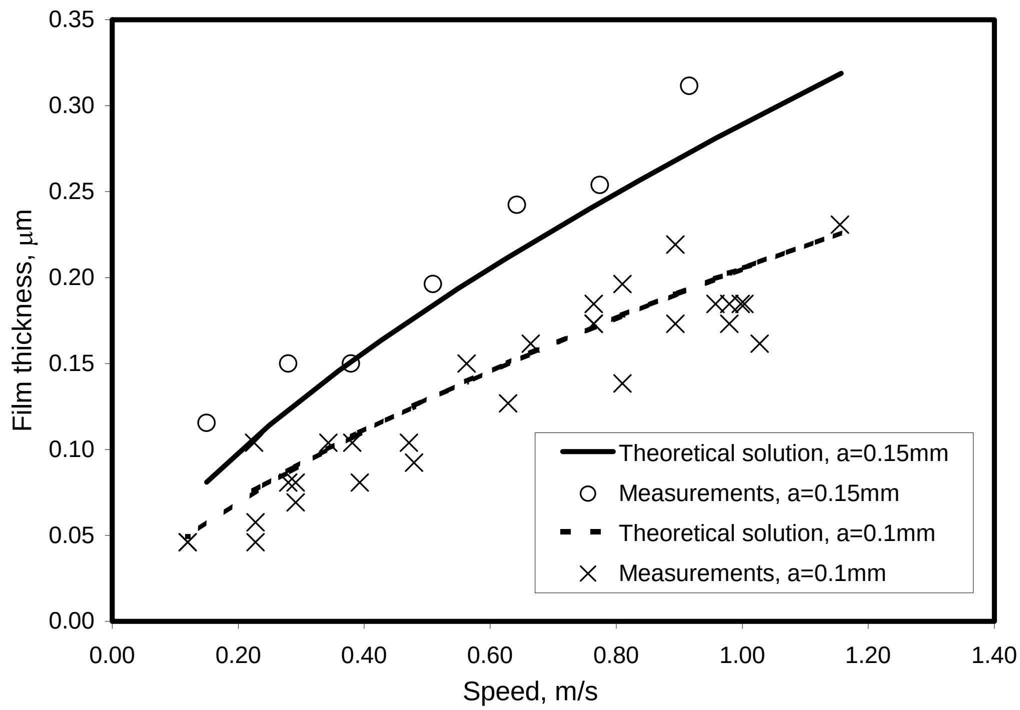

Figure 8. Measured film thickness against speed for two contacts at different loads. Results are compared with the theoretical solution (equation 11).

Measurements from a Rolling Element Bearing

Figure 9 shows a diagram of apparatus developed to measure the film thickness in a rolling element bearing. The bearing is a 6410 deep groove ball bearing and is loaded hydraulically and rotated by an electric motor. A 25 MHz transducer is positioned in a small water bath through machined in the bearing bush. The outer race is in contact with the water but not modified in any way. The transducer is spherically focusing and positioned in the water bath such that the wave is focused on the interface between the ball and the outer raceway. The transducer is carefully positioned so that the focused spot falls within the ball/raceway contact region. The bearing cavity is flooded with a mineral oil (shell Turbo T68).

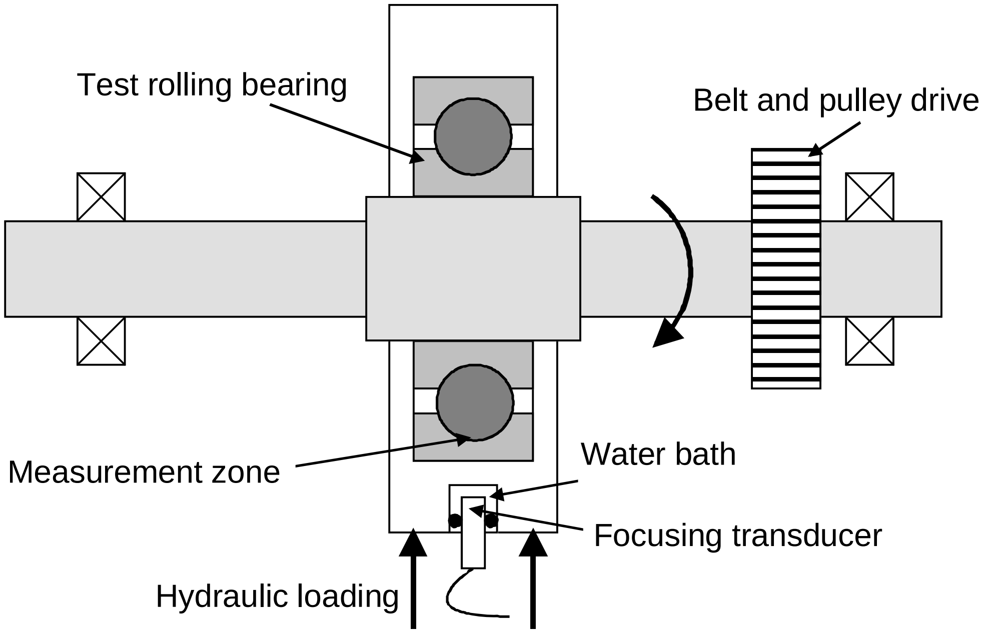

Figure 9. Schematic of apparatus used for ultrasonic measurements of film thickness in rolling element bearings.

A reflected pulse is recorded when the bearing is stationary and there is no ball located over the measurement region (although the cavity is still filled with oil). This signal (when corrected by the steel/oil reflection coefficient) is then used as the reference signal. The bearing is then rotated and the transducer set to continuously pulse at a repetition rate of 10 kHz.

When a pulse strikes the outer raceway bore, when there is no ball at that location, the pulse is largely reflected (R=0.95). However when a ball is over the measurement zone a much greater proportion of the wave is transmitted and reflection coefficients are significantly reduced. A gate is set on the digital oscilloscope to capture and store only these reduced amplitude reflected pulses (typically set at R<0.7). The pulse with the minimum amplitude as the ball passes over is used. This ensures that a signal is not processed when the contact is half positioned over the measurement zone. This would give a signal back from a film of varying thickness and the assumptions of the spring model stiffness (equation 7) would no longer be valid. It would perhaps be a better, but more involved, solution to trigger the pulse such that it is only received when a ball is directly overhead.

A variety of bearing loads and speeds were tested. However, because the temperature of the lubricant varies throughout these tests it is not possible to present the data as film thickness against speed (or load) curves, because the viscosity is different at each point. Instead the data is presented as the measured film thickness plotted against a prediction of the film thickness from the solution of Hamrock & Dowson [1981] (see figure 10). The viscosity of the oil used in the prediction is determined at the appropriate temperature (as measured by a thermocouple in the oil close to the contact region). The bulk modulus is that determined from the experiments on the ball on flat apparatus above. Strictly a different value of the bulk modulus should be used at each bearing load; further work is needed in this area.

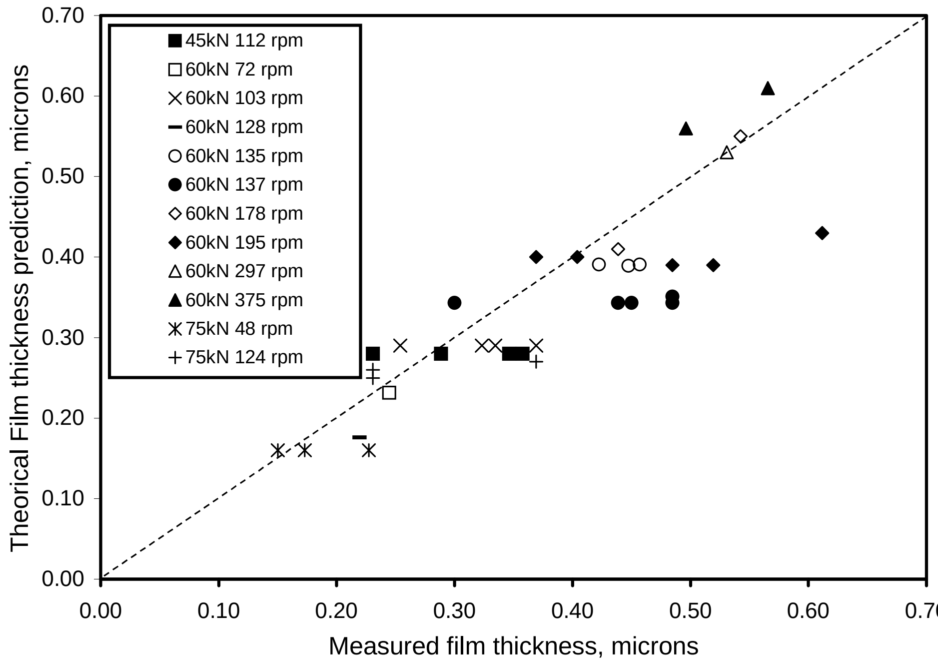

Figure 10. Comparison of rolling element bearing film thickness measured by an ultrasonic means with EHD theoretical solution (the dashed line shows the locus of exact agreement).

The results show reasonable agreement. The spread along the x-axis shows the variation in film for nominally the same bearing operating conditions (load, speed, and oil temperature). The measured film thickness varies by around ±20%.

Limits of Applicability

The successful operation of this method relies on the following:

(a) the spring model is an appropriate method for interpreting the reflected signal

(b) the ultrasonic wave must be able to penetrate through the bearing raceway

(c) the focused spot size must be smaller than the contact patch dimensions

(d) the contact must be in position for long enough for a pulse to be reflected

Each of these requirements will be investigated in turn, in order to investigate what limits they put on the type of bearing films that can be measured.

Spring Model Validity

The spring model which describes the reflection of ultrasound at a thin layer relies on two assumptions; firstly that the layer is thin and may be considered as a spring of negligible mass and damping, and secondly, that the opposing surfaces are smooth and parallel.

If the layer is thick compared with the ultrasonic wavelength then mass effects become important. At higher frequencies and thicker films there is the appearance of resonances in the reflection spectrum and the mass of the layer is important. The frequency at which a resonance occurs is given by [Pialucha and Cawley, 1994]:

$$f_{res} = \frac{cm}{2h} \tag{12}$$

where m is the mode number. The speed of sound is a function of both stiffness and density; so clearly the resonance depends on both stiffness and mass. From equation (12), the first resonance (m=1) will occur at 2.25 GHz, 225 MHz, and 22.5 MHz for films of thickness 1, 10, and 100 µm respectively. For thin ehl film, therefore, the measurement frequencies would have to be very high before any mass effects are observable.

The effects of surface roughness are harder to quantify. The stiffness (equation 7) is calculated on the basis of a smooth parallel-sided oil film. Surface roughness can have two effects; firstly there may be asperity contact, and secondly the gap width will vary. If rough surface contact occurs then the reflection will depend on the combined stiffness of both the liquid and the solid part. The scale of the roughness is so small, in a bearing component, compared to the sound wavelength, that individual asperities will not scatter ultrasound. The stiffness of the liquid part will therefore be controlled by the mean gap thickness (i.e. the spacing between the two rough surface mean lines).

Attenuation in the Bearing Raceway

Some of the ultrasonic wave will be attenuated as it passes through the raceway material. If the attenuation is excessive then the reflected signal cannot be distinguished from background noise. Typically, porous materials and those with high hysteresis attenuate significantly. The amount a material attenuates ultrasound is defined by an attenuation coefficient, α where:

$$I_x = I_0 e^{-\alpha x} \tag{13}$$

where I_x is the amplitude of a wave, initially of amplitude I_0, after it has passed through a thickness of material of x [Krautkrämer and Krautkrämer 1975].

A series of experiments were performed to determine the attenuation coefficient for a typical bearing steel. A wave was passed through a bearing steel roller (from end to end). The first and second reflections were compared. The former has passed through 2l (where l is the length of the roller), whilst the latter has passed through 4l. Knowing I_{2l} and I_{4l} simultaneous equations can be set up to determine α for each test frequency. Empirically the attenuation coefficient is frequently found to vary with the square of the frequency [Krautkrämer and Krautkrämer 1975]. A best fit to the test data for the bearing steel used in these experiments gave α=1.5f^2 dB/m.

If we allow a conservative reduction of the incident signal of say, 90%, equation (13) can be used to determine the thickness of bearing steel that can be successfully penetrated. Figure 11 shows the maximum bearing raceway thickness that can be penetrated by a given ultrasonic frequency.

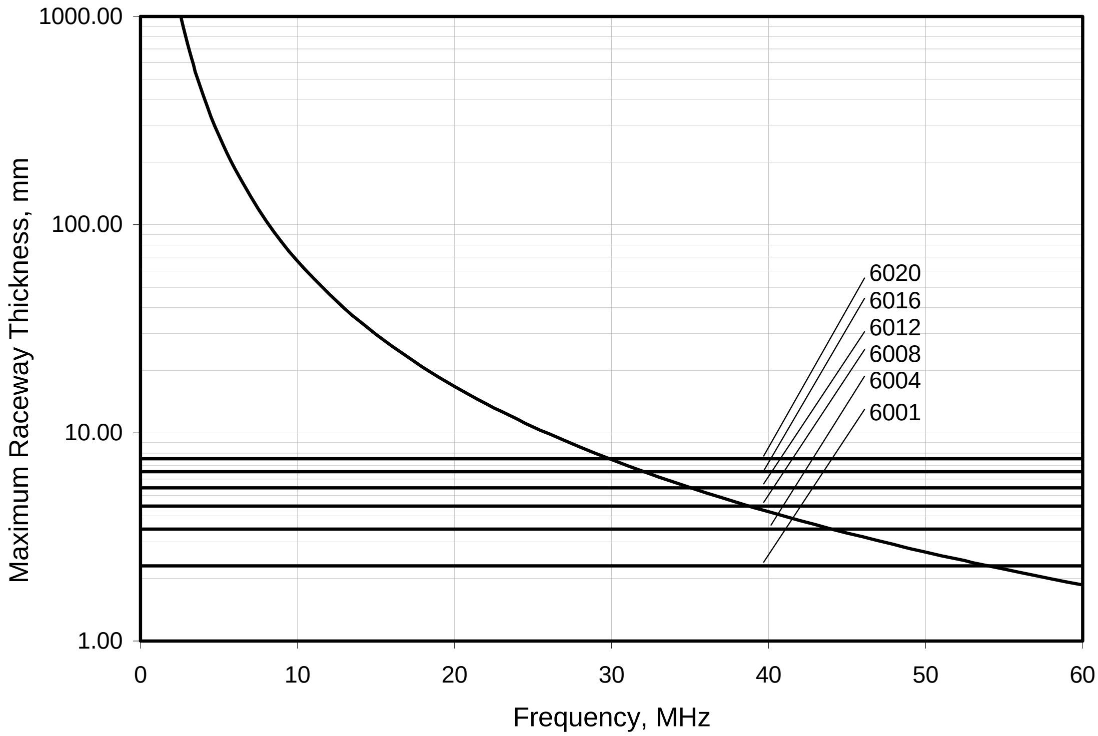

Figure 11. The maximum bearing steel raceway thickness that can be penetrated by an ultrasonic wave and still retain less than a 90% signal loss. Raceway dimensions from some of the 60XX series of ball bearings are over plotted.

At a frequency of 25 MHz, it would be possible to test through 10 mm of bearing steel. But, if the measurement frequency is 50 MHz, then only 3 mm is possible. The approximate thickness of the outer raceway for a selection of deep groove bearings is plotted on figure 11. This indicates that for the larger size bearings, the measurement frequency will be limited to the lower ranges.

Spatial Resolution

In this work focused transducers are used where a spherical lens bonded to the piezo element focuses the emitted wave. The spot size of a focused transducer is given by (Silk 1984):

$$d_{f\,(-6dB)} = \frac{1.028Fc}{fD} \tag{14}$$

where F is the transducer focal length, c is the speed of sound in the focusing medium, and D is the diameter of the piezo element. This expression is plotted in figure 12 for a typical transducer of crystal size 7.5 mm and focal length 25 mm. This plot shows what frequency would be required to achieve a given spot size. This spot should fall entirely within the central film thickness region of the contact. As a first approximation therefore, a measurement could be achieved if this spot size is less than the minor semi-axis of the ball raceway elliptical point contact ($d_f \leq a$). It is not possible to focus the wave to less than the wavelength in the focusing medium, λ (where $c=f\lambda$). This represents a physical minimum to the spatial resolution and is also shown on figure 12 for a wave travelling in steel.

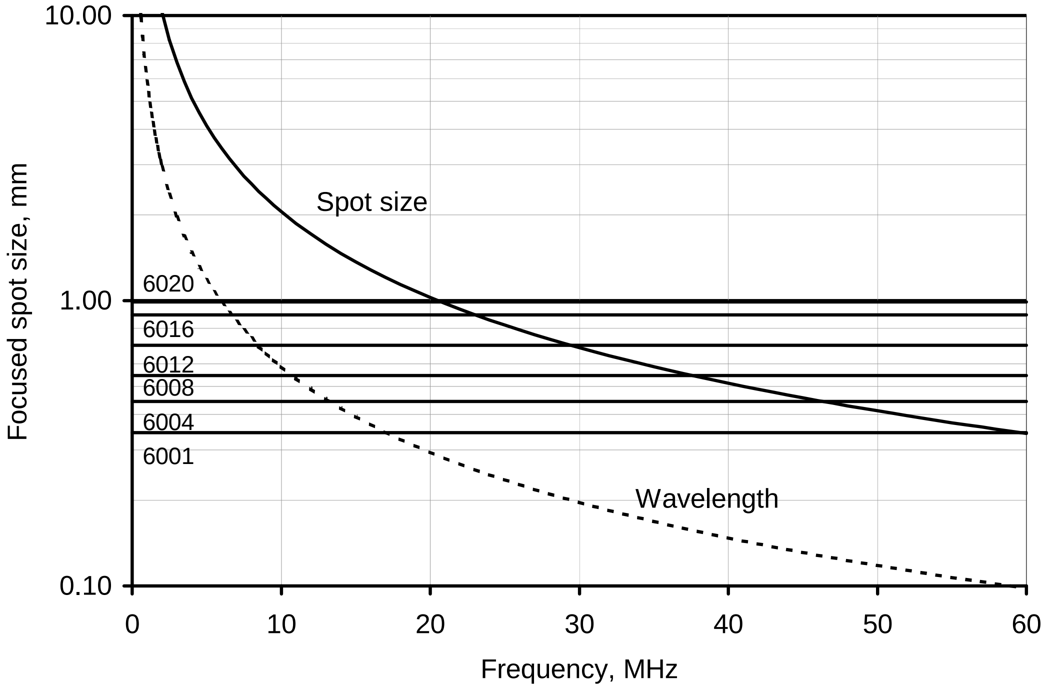

Figure 12. Spot size variation with frequency (for an ultrasonic transducer of spherical focal length 25 mm and crystal diameter 7.5 mm)

Over-plotted in figure 12 are the minor semi-contact widths calculated for some of the 60XX series of ball bearings (loaded to their maximum dynamic load rating, *C*). It is evident that for the larger bearing sizes most frequencies will be suitable; whilst for the smaller instrument bearings higher frequencies are required.

Speed Limit

In these experiments a pulse repetition rate of 10 kHz was used. There is thus a limiting ball speed such that two successive pulses should fall within a contact patch as the ball moves over the measurement zone. The relationship between the limiting ball speed and the size of a measurable contact patch is plotted in figure 13 for several pulse repetition rates. Again, contact patch sizes from bearings in the 60XX series are over plotted.

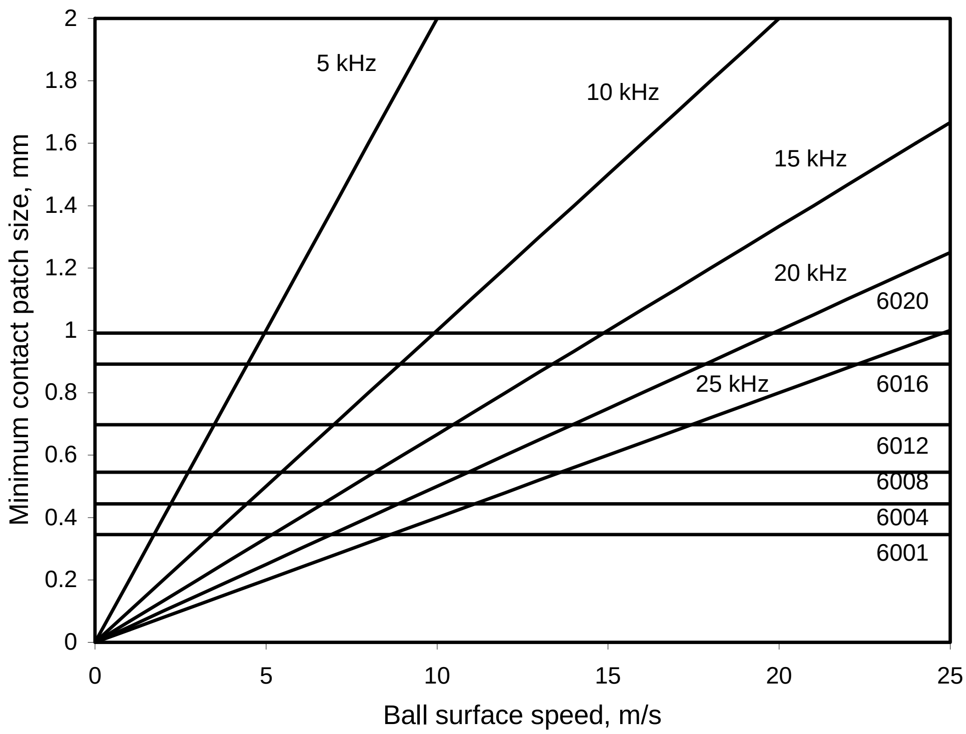

Figure 13. The minimum measurable contact patch width (2a) for a given ball surface speed. As the pulse repetition rate is increased smaller higher speed contacts can be measured.

The maximum possible repetition rate is determined by the requirement that the second incident pulse should not interfere with the first reflected pulse. How close these two pulses can be will be governed by the thickness of the bearing raceway.

An alternative approach would be to trigger the pulsing system so that pulses are only received when the ball is directly over the transducer. The width of a single pulse is typically twice its wavelength (the pulses are designed to be as short as possible). If that pulse is to be completely reflected from a contact then the contact width must larger (i.e. approximately $a \geq \lambda$). For frequencies in the range 10 to 50MHz travelling in steel, the wavelengths and thus theoretically minimum measurable contact widths, would be from 590 to 118 μm. Essentially this is similar to the spatial resolution limit; it is not possible to focus a pulse to a size less than its wavelength.

Practical Implementation

There are a number of practical problems associated in this approach. Firstly a transducer must be held at a fixed location in space, normal to the contact, and coupled to the outer raceway. Positioning the transducer takes some care, so that the wave can be focused directly on the contact location. A water bath has been built to couple the transducer to the bearing outer race. This is somewhat awkward to set-up and maintain during testing.

There is also a requirement for a reference signal (to deduce the incident ultrasonic wave amplitude). This means that a measurement must be recorded when a ball is not above the measurement location. In this work this was done when the bearing was stationary. In principle it would be possible to do this as part of a running test. Signals reflected back from a ball contact are divided by those recorded when no ball is present.

The output of a piezo-electric crystal varies with temperature. If during a test the transducer heats up, then the signal amplitude, and hence reflection coefficient, reduce, thereby under-predicting film thickness. Typically, signal reductions of about 10% were observed as the temperature varied during these tests. The transducers are pre-calibrated in an oven, so that temperature response is known before testing and can be accommodated.

The interpretation of the reflected signal requires that the bulk modulus of the oil in the contact is known (equation 10). For oil at ambient conditions this is an easy parameter to measure. However, for oil under high pressure it is not so straightforward. Here, the high-pressure modulus has been determined by fitting a theoretical model to experimental data from the ball on flat contact. This has then been used for later interpretation of ball bearing reflected signals. This approach is a useful way to determine the bulk modulus of lubricants under pressure. The contact essentially acts as a small high-pressure cell; and if the film thickness is assumed then the bulk modulus is obtained. For better accuracy of ball bearing film measurements, the relationship between bulk modulus and pressure could be determined by this method and an appropriate value used in each ball bearing test case.

Conclusions

1. Ultrasound reflects 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.
2. The response of the lubricant layer to an ultrasonic pulse has been modelled using a spring model approach. The reflection is then a function of the stiffness of the lubricant layer (i.e. its bulk modulus and thickness), the frequency of the ultrasonic wave, and the acoustic impedance of the surrounding solid bodies.
3. The elastohydrodynamic lubricant film that forms between a ball sliding on a flat surface has been measured by this means. Films down to 50 nm were recorded. By comparing the measured film thickness with theoretical predictions the bulk modulus of the oil in the contact was deduced to be 25.8 GPa.
4. The approach has also been used to measure film thickness in a rolling element bearing. The transducer is mounted on the bearing raceway and the pulse reflected back as the ball passes over is recorded. Measurements agree reasonably well with theoretical predictions although there is significant scatter in the data.
5. Limitations of the method depend on ultrasonic attenuation in the raceway, spatial resolution of the wave, and the speed of the contact patch. In principle the approach will work for most ball bearing sizes, although larger heavily loaded bearings, with a large contact patch, will be easier to measure.

References

Cameron, A. and Gohar, R., 1966, Theoretical and Experimental Studies of the Oil Film in Lubricated Point Contact, *Proc. Roy. Soc. Lond. A*, Vol. 291, pp. 520-536.

Drinkwater, B.W., Dwyer-Joyce, R.S. and Cawley, P., 1996, A study of the interaction between ultrasound and a partially contacting solid-solid interface, *Proc. R. Soc. Lond. A*, Vol. 452, pp. 2613-2628.

Hamrock, B. J. and Dowson, D., 1981, "Ball Bearing Lubrication, the Elastohydrodynamics of Elliptical Contacts", John Wiley & Sons, London.

Hosten, B., 1991, Bulk heterogeneous plane-wave propagation through viscoelastic plates and stratified media with large values of frequency-domain, *Ultrasonics*, Vol. 29, pp. 445-450.

Jacobson, B. O. and Vinet, P., 1987, A Model for the Influence of Pressure on the Bulk Modulus and the Influence of Temperature on the Solidification Pressure for Liquid Lubricants, *ASME J. Trib*., Vol. 109, pp. 709-714

Kendall, K. and Tabor, D., 1971, An ultrasonic study of the area of contact between stationary and sliding surfaces, *Proc. R. Soc. Lond. A*, Vol. 323, pp. 321-340.

Krautkrämer, J. and Krautkrämer, H., 1975, *Ultrasonic Testing of Materials,* Springer-Verlag, New York.

Krolikowski, J. and J. Szczepek, 1991, Prediction of contact parameters using ultrasonic method, *Wear,* Vol. 148, pp. 181-195.

Pialucha, T. and Cawley, P., 1994 The detection of thin embedded layers using normal incidence ultrasound, *Ultrasonics*, Vol. 32, pp. 431-440.

Povey, M.J.W., 1997, *Ultrasonic Techniques for Fluids Characterisation*, Academic Press, San Diego.

Richardson, D.A. and Borman, G.L., 1991, Using Fibre Optics and Laser Fluorescence for Measuring Thin Oil Films with Applications to Engines, *SAE Paper 912388*.

Silk, M.G., 1984, *Ultrasonic Transducers for Nondestructive Testing*, Hilger, Bristol.

Tattersall, H.G., 1973, The Ultrasonic Pulse-Echo Technique as Applied to Adhesion Testing, *J. Appl. Phys.D.*, Vol. 6, pp. 819-832.