

A study of thermal effects in EHL rheology and friction using infrared microscopy

Abstract

Infrared microscopy is used to obtain through-thickness oil temperature measurements from EHL contacts between different surface materials (steel, silicon nitride and zirconia) for the lubricants Santotrac 50 and PAO4. The measurement technique was first adapted to overcome focussing issues due to the partially transparent zirconia surface. Results were used to infer in-contact rheological behaviour of the lubricants. Santotrac 50 shows significant shear localization under all conditions with the position of the shear heating zone being highly affected by the contact surfaces' thermal properties. For PAO4, the shear profile depends on the contact surfaces' thermal properties with moderate to high surface conductivities favouring uniform shearing, whereas highly insulating surfaces (zirconia) cause shear localisation at the surface for both lubricants. These results are used to interpret friction measurements and show how the thermal properties of surfaces can be used to control rheology and friction. This paper is prefaced by a review of thermal EHL theory upon which our analysis is based.

Keywords: *Rheological behaviours, Thermal properties, Elastohydrodynamic (EHD) Lubrication, Infrared technique.*

Short Abstract: Infrared microscopy is used to obtain oil temperature measurements from EHL contacts between different surface materials for the lubricants Santotrac 50 and PAO4. Results were used to infer rheological behaviour of the lubricants. Santotrac 50 shows significant shear localization under all conditions and the position of the shear heating zone is highly affected by the thermal properties of surfaces. For PAO4, the shape of the shear profile depends on the properties of the contact surfaces with moderate to high surface conductivities favouring uniform shearing, whereas highly insulating surfaces (zirconia) cause shear localisation at the surface. These results are used to interpret friction measurements and show how the thermal properties of surfaces can be used to control rheology and friction.

Nomenclature

α	heat partition factor
β	temperature-viscosity coefficient
γ	shear strain rate
τ	shear stress
τ_e	Eyring stress threshold value
τ_{mean}	average corrected shear stress
$\tau_{e(iso)}$	isothermal shear stress
η	viscosity of oil
η_0	initial viscosity at bulk temperature
η_p	Newtonian dynamic viscosity at low shear rate
κ	thermal conductivity
μ	Traction coefficient
ρ	density of sample
σ	heat capacity of sample
χ	thermal diffusivity
a	contact radius
Br	brinkman value
c	heat capacity of sample
G_p	elastic shear modulus
h	film thickness
h_1	distance from ball surface to maximum temperature position
$\bar{P}$	average contact pressure

1	P_e	peclet number
2		
3	q	heat generation
4		
5	q_{ball}	heat transfer to ball surface
6		
7	q_{disc}	heat transfer to dis surface
8		
9	R	distance separating the locations of heat input the temperature rise
10		
11	srr	slide roll ratio
12		
13	T_{oil}	temperature of oil
14		
15	T_{surf}	temperature of contact surface
16		
17	T_{ball}	ball surface temperature
18		
19	T_{disc}	disc surface temperature
20		
21	U_s	sliding speed of contact surface
22		
23	U_{disc}	disc surface speed
24		
25	U_{ball}	ball surface speed
26		
27	U	entrainment speed of oil
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29		
30		
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1. Introduction

The counter form nature of the contacting surfaces in common rolling/sliding components such as gears, rolling bearings and cams are responsible for high contact pressures, which cause appreciable elastic deformation of the components accompanied by an increase in viscosity of the lubricating fluid. These conditions enable Elastohydrodynamic lubrication (EHL) whereby entrained fluid generates sufficient pressure to separate the surfaces, preventing metal-metal contact and reducing friction and wear. Shearing of this film gives rise to frictional resistance which is partly responsible for the inefficiency of machines with Elastohydrodynamically lubricated components.

Frictional heating plays a critical in determining the friction forces in EHL contacts since the resulting temperature rise affects the rheological properties [1] and degradation of the oil film [2][3]. This temperature rise also affects lubricant viscosity and hence fluid film thickness which also in turn affects the friction. This thermal feedback is modulated by the lubricant's thermal properties and the surface materials. Because of its key role, temperature rise is often measured to provide information regarding the lubrication mechanism in sliding contacts. This information is a vital complement to numerical models used to prevent component failure and optimise performance. These numerical models include CFD ones to understand EHL rheological behaviours [4][5][6][7], rheological ones to predict friction [8][9] and mathematical ones to predict the contact surface thermal properties [10] and temperature effects [11][12]. In summary, the temperature rise is the major detection index of friction heat generation and has played an indispensable role in tribology research during the last three decades [13][14].

The exact mechanisms involved in and the location of this energy transformation in EHL contacts are very difficult to predict. This is due to the extreme conditions (pressures of GPa and shear rates of 10^7), small areas (around 80 - 100 μm in radius) of the contacts, combined with the fact they are located between moving surfaces. And, as described above, thermal feedback occurs since shearing of the lubricant produces heat and the resulting increase in temperature, affects fluid viscosity.

The aim of this study is to better understand how friction heating is linked to temperature rise and how this affects lubrication performance. A particular focus is given to the influence of lubricant thermal properties and the tribological components. Our approach is an extension of that developed by in Reddyhoff's [15][16] and, more recently, Lu and co-workers' studies [17].

First, a detailed review of thermal EHL theory is presented, which is the basis for the subsequent analysis of experimental data. Such a review is currently absent from the tribology literature. Using this theory, the measured average temperature distribution within the film are inferred from the measured temperatures.

2. Background

2.1 Effects of thermal properties of material on heat transfer

Most contact surface temperature analysis has come about as a result of the pioneering work of Blok [18] published in 1937. Blok calculated the maximum rise in surface temperature on a semi-infinite contact body due to a uniform square heat source under conditions of both high and low Peclet number (defined as $P_e = \frac{U_s a}{2\chi}$).

Blok assumed that the average rise in temperature over the whole contact surface was relatively low due to the friction being low under oil lubrication conditions.

In 1942, Blok's work was continued by Jaeger [19], by solving the point heat source problem. In the latter's study, heat convection was ignored and the solution extended to point contact domains of different shapes. Then in 1959, Carslaw and Jaeger [20] developed the steady-state rise in temperature model caused by a stationary heat source with the time assumed as infinite:

$$T_{steady} = \frac{q}{2\pi\kappa R} e^{-\frac{U_s(R-x)}{2\chi}} \quad \text{Equation. 1}$$

where $R = x^2 + y^2$ is the distance separating the points of heat input and temperature rise, κ is the thermal conductivity, χ is the thermal diffusivity defined as $\chi = \frac{\kappa}{\rho\sigma}$, where σ and ρ are the heat capacity and density of samples, respectively. Integrating Equation 1 over the radial distance from the centre x gives,

$$T_{mean} = \frac{q}{4a\kappa} \quad \text{Equation 2}$$

where a is the contact half width. Archard [21] derived equations for the rise in contact surface temperature in the between both elastic and plastic deformed components. He did this using a theoretical model alike that of Blok and Jaeger. However, Archard stated that the mean temperature rise in Equation 2 is no longer applicable at higher Peclet values as there is insufficient time to establish a steady state temperature distribution on fast-moving surfaces. In this case, the mean temperature rise for Peclet values higher than 5 is

$$T_{mean} = \frac{0.31q}{\kappa a} \sqrt{\frac{\chi}{U_s a}} \quad \text{Equation 3}$$

Later, Tian and Kenndy [22][23][24] expanded the temperature rise to the entire range of Peclet values based on models using a Green's function method.

It should be noted that in Equation 2 and 3, q represents the rate of heat supply to one contact surface which cannot usually be calculated directly from measurements since the partition of the into each body of the generated heat is unknown. This is because contacting surfaces are often made of different materials moving at different speeds, so the flow of heat is not divided equally between them. Under dry conditions when the surfaces are in contact, there should be no discontinuity in the temperature profile across the contact area, and in

this case, the partition of heat can be obtained by matching the temperatures of corresponding points on opposing surfaces. Based on this, in 1959, Ling [25] developed a quasi-iterative method which the transform solutions were used to compute the temperatures on each surface. The temperatures obtained can then be used to calculate the heat transferred to each surface. In 1974, Kennedy and co-workers applied the finite element method to develop equations for the temperature of contact surfaces[26] [27][28]. More recently, Spikes [29] used a numerical algorithm based on Carslaw and Jaeger's equation to determine the heat partition for random contact shapes by matching the temperatures of each point in the contact with the heat source method given in Equation 4 (a) and (b).

$$q_{kl} = (\sum_{ij} C_{ij}^{kl})^{-1} T_{ij} \quad \text{Equation 4 (a)}$$

$$C_{ij}^{kl} = \frac{1}{2\pi\kappa} \iint_A \frac{\exp[-\frac{\Delta U}{2\kappa}\{x_k - x_i + R\}]}{R} dx'_k dy'_l \quad \text{Equation 4 (b)}$$

where C_{ij}^{kl} is the coefficient relating the temperature rise at point i,j (T_{ij}) with a heat input and point k,l (q_{kl}).

2.2 Rheology from temperature measurements

Rheology, under full film elastohydrodynamic conditions, concerns the dependence of shear stress (or viscosity) on shear strain rate. Under EHL conditions, the fluids normally sustain high pressures which can be as high as several gigapascals. Such high pressures, in conjunction with sub- μm film thickness and m/s sliding speeds, results in rheology that present several special phenomenon such as shear thinning [30], phase transitions from liquid to glassy state and rate-independent behaviour [31]. Previous experimental results have revealed two types of rheology behaviour under non-Newtonian conditions, namely viscoelastic and shear heating – the latter being important in this investigation of thermal effects.

2.2.1 Shear heating

2.2.1.1 Shear bands

In 1967, Plint [32] suggested inhomogeneous shear in EHL contacts due to a localisation of shear heating, temperature rise and viscosity reduction at the central position across the thickness of the film. This phenomenon known as shear localization or shear bands can be explained as follows. Most lubricants have poor thermal conductivity especially when in a glassy-state, which causes oil film heat generation to result in a rapid temperature rise in the location where shear rate is greatest. Moreover, according to Roelands viscosity definition [33], the viscosity reduces approximately exponentially with temperature rise. These conditions clearly favour the film shearing in the centre, rather than uniformly though the thickness and this phenomenon can generate an extremely high temperature rise.

In 1977, Winter [34] summarized the dimensionless parameters such as Nahme, Brinkman and Griffith numbers to define shear band properties. The Brinkman number is the ratio between heat generated by viscous dissipation and heat delivered by molecular conduction as shown in Equation 5 (a) and (b),

$$Br = -\frac{\tau^2 h^2}{\eta_0 k} \frac{1}{\eta} \frac{d\eta}{dT} \quad \text{Equation 5 (a)}$$

$$Br = -\frac{\eta U^2}{\kappa} \frac{1}{T_{surf} - T_{oil}} \quad \text{Equation 5 (b)}$$

where h is the film thickness, τ is the shear stress, η is the viscosity at a specific temperature and pressure, U is the sliding speed of oil, T_{oil} and T_{surf} are the temperatures of oil and contact surface. The Brinkman number in the form of Equation 5 (a) has been widely used in [35] [36][37], with the application of the coefficient for temperature-viscosity $\beta = -\frac{1}{\eta} \frac{d\eta}{dT}$. The form in Equation 5 (b) is rarely used in literature but provides a convenient way of identifying fluid behaviour if the temperatures of oil and contact surfaces are known.

The value of Brinkman number was verified by series of experimental studies starting with Poehlein [38] in 1968. Although his research was based on Newtonian fluids, Poehlein pointed out the critical value that can be used as a guide for measurements and simulations of non-Newtonian fluids. For instance, he proposed that the viscosity remains constant when the Brinkman number is maintained at a value that is lower than 3.508, whereas above this value shear banding is expected to occur. When the Brinkman value is lower than π^2 , Winter [39] pointed out that the viscosity varies linearly with temperature. It was also proved by Murthy [40] that, at high Brinkman values, viscous dissipation occurs uniformly and the temperature increases in a proportional manner. When the Brinkman value is lower than 10, the temperature increase remains almost constant, as indicated by Winter. However, when the Brinkman number exceeds a critical value of 10, the thermal behaviour of the liquid undergoes a drastic change. Thus, the possibility of observing shear localization in non-Newtonian flow depends on the sliding speed and temperature difference between oil and contact surfaces.

The shear localization phenomenon at high Brinkman numbers was observed experimentally by Bair and co-workers [41], using an optical ray tracing particle image velocimetry technique. They showed that pressure is a key factor influencing shear localization. More specifically, shear bands were shown to be caused by lubricant inhomogeneity due to low thermal conductivity in the glassy-state and a viscosity which reduces exponentially with temperature in the shearing zone. A multilayer lubricant film is formed with a thin hot shearing zone between two solid-like portions of the film. Shear bands under EHL conditions occur along different directions and have been compared by Bair [35] with three models. This proves that the shear band localization is an intermolecular and random phenomenon caused by the lubricant being mechanically stressed to or near its ultimate strength.

The shear bands may occur at high strain rate or high shear stress even near pure rolling conditions. Heyes and co-workers [42] and Gattinoni et al. [43] used the Non-equilibrium molecular dynamic (NEMD) simulation to study a simple Leonard-Jones liquid at high pressure with plane Couette flow and predicted the shear banding occurring away from the contact surfaces as the rest of the liquid crystalized to gain mechanical strength. Meanwhile, the full behaviour from Couette to CL (central localised state) and Plug Slip tending to Solid has been mapped and the temperatures profiles were delivered with formulate. Ewen et al [44] continued the NEMD simulation with multiple lubricants and found that all of the liquids under investigation display central localization induced by high pressure with the same contact surfaces. They also show that the shear banding results in a lower friction coefficient and a lower gradient of the shear stress versus log (strain rate) curve, when compared to simple Couette flow.

Yagi and co-workers [45][46] used an infrared technique and successfully measured the oil temperatures and sliding surfaces to investigate the impact of oil film temperature distributions on film thickness by varying inlet temperatures and slide roll ratios. Matching calculating to measured temperatures showed that Couette flow was occurring.

Wong and co-workers [47][48][49] proved the existence of shear bands at high pressure with the use of **luminescence velocimetry** methods by observing through-thickness velocity profiles of lubricants. Using the uniform shear stress assumption across the oil film, they showed a phase transition and lubricant viscosity rise occurring at high pressure and generating a non-linear velocity profile.

2.2.1.2 Viscous heating and temperature rise of oil film

The contact surfaces' thermal properties control the rate of heat transfer and this effects oil temperature and the variation of viscosity which lead to a temperature gradient across the oil film.

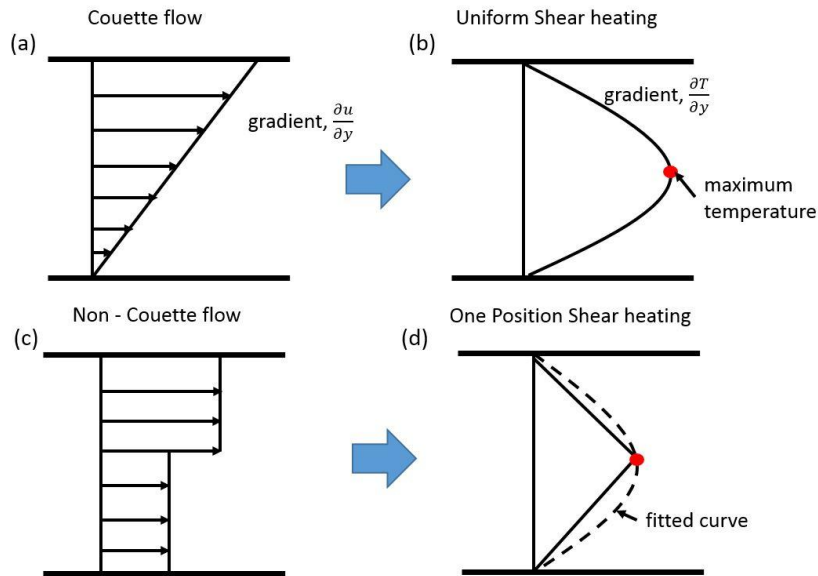


Figure 1. Simplified extremes of Rheological behaviours depending on different flow properties.

Analytical models of the heat flow in an EHL contact for both uniform and localised shear heating have been developed in the current study to calculate the temperature profiles across oil film as a function of thickness. These models can be compared with experimental results to determine the type of rheological behaviour occurring in the film under different conditions. The variations in heat generation and transfer are governed by the energy equation,

$$c\rho \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = \kappa \frac{\partial^2 T}{\partial z^2} + \eta \left[\left(\frac{\partial u}{\partial z} \right)^2 + \left(\frac{\partial v}{\partial z} \right)^2 \right] - \frac{T}{\rho} \frac{\partial \rho}{\partial T} \left(u \frac{\partial p}{\partial x} + v \frac{\partial p}{\partial y} \right) \quad \text{Equation 6}$$

where c , k , η and ρ are the heat capacity, thermal conductivity, viscosity and density of lubricants, u , v and w are the speed of fluids in three dimensions. The energy equation contains four terms, which from left to right represent: heat convection, heat conduction, heat generation due to shear and compression. As described above, the heat transferred by convection and generated by compression can be ignored as they are outweighed by

conduction and shear respectively. The shear heating in the direction perpendicular to sliding speed can also be ignored due to the low speed in this direction. Moreover, the heat generation can be replaced by frictional heating, which simplifies the energy equation to,

$$\eta \left(\frac{\partial u}{\partial z} \right)^2 = \frac{\bar{\tau}_i^2}{\eta_i} = -\kappa \frac{\partial^2 T}{\partial z^2} \quad \text{Equation 7}$$

According to Archard [21], two idealised types of heat generation can be considered, depending on whether shearing occurs at a single, localised position (non-Couette flow) or whether shearing and heat generation occurs uniformly across the oil film (Couette flow). These are shown in Figure 1. For the case of localised heat generation (Figure. 1 (b)) and considering the Fourier's energy law [50], the heat transfer to each surface can be presented as,

$$q_{ball} = \alpha q = \kappa \frac{T_{max} - T_{ball}}{h_1} = \alpha \tau (srr * U) \quad \text{Equation 8 (a)}$$

$$q_{disc} = (1 - \alpha)q = \kappa \frac{T_{max} - T_{disc}}{h - h_1} = (1 - \alpha) \tau (srr * U) \quad \text{Equation 8 (b)}$$

where α is the heat partition factor (which can be obtained from Equation 4 in section 2.1 inputting the experimental results of T_{ball} and T_{disc}), h_1 is the distance from ball surface to maximum temperature position while h is the oil film thickness. In addition to this, unknown values of h_1 and T_{max} can be calculated from Equation 8 (a) and (b) according to,

$$h_1 = (1 - \alpha)h - \frac{k(T_{ball} - T_{disc})}{q} \quad \text{Equation 9 (a)}$$

$$T_{max} = T_{ball} + \frac{h(1 - \alpha)\alpha q}{k} - \alpha(T_{ball} - T_{disc}) \quad \text{Equation 9 (b)}$$

Equation 8 results in the linear temperature distribution shown in Figure 1 (d). Furthermore, if as Archard assumed, the temperature rise of the two bounding surfaces are equal, Equation 9 (a) and (b) becomes,

$$T_{max (local shear)} = \frac{T_{ball} + T_{disc}}{2} + \frac{hq}{4k} \quad \text{Equation 9 (c)}$$

$$T_{average(local shear)} = \frac{T_{ball} + T_{disc}}{2} + \frac{hq}{8k} \quad \text{Equation 9 (d)}$$

For the case when heat results from viscous losses that are uniformly distributed throughout the oil film, the shear stress is approximately constant with the strain rate and the rate of heat generation is inversely proportional to viscosity. In this case, the temperature profile across the oil film can be obtained from the integration of Equation 7:

$$T = -\frac{\bar{\tau}^2}{k\eta} \frac{z^2}{2} + Az + B \quad \text{Equation 10 (a)}$$

$$A = \frac{T_{disc} - T_{ball}}{h} + \frac{\bar{\tau}^2 h}{2k\eta} \quad \text{Equation 10 (b)}$$

$$B = T_{ball} \quad \text{Equation 10 (c)}$$

where A and B are constant parameters and can be found from the boundary conditions prescribed by the temperature of the contact surfaces. This reflects how oil temperature is affected by the transfer of heat to surroundings. The heat input in Equation 10 is the mean value over the whole contact area with the uniform shear stress and the application of an average effective oil viscosity η . The resulting parabolic temperature distribution from this form of heating is shown in Figure 1 (b).

If it assumed that the temperature rise of the contact surfaces are equal to each other, Equation 10 can be integrated to give the temperatures that arise to the uniform shearing of the film:

$$T_{\max(Couette)} = \frac{T_{ball} + T_{disc}}{2} + \frac{hq}{8k} \quad \text{Equation 10 (d)}$$

$$T_{\text{average}(Couette)} = \frac{T_{ball} + T_{disc}}{2} + \frac{hq}{12k} \quad \text{Equation 10 (e)}$$

The above approach uses one of two equations for the temperature rise within a lubricant depending of the location of the shear heating. These equations (9 and 10) consist of two terms; the mean surface temperature rise based to the heat transfer through contact surfaces which can be deduced from Equation 1 of Carslaw and Jaeger's equation, and the average temperature rise of oil film caused by shear heating deduced from Equation 7. Equations 9 and 10 can be combined and written in the more compact form according to Archard:

$$T = \frac{\tau(srr * U)}{\sqrt{2\pi k \rho c}} \sqrt{\frac{2a}{U}} + \frac{h}{bk} \tau(srr * U) \quad \text{Equation 11}$$

where U is the entrainment speed, a is the contact area radius, c , ρ , and κ are the thermal capacity, density and thermal conductivity of lubricant, h is the thickness of the film and b is a constant having a debated value which depends on where the heat produced in the film (value of b also varies depending on whether it is the average or maximum temperature within the film that is being calculated) as given by Equations 9 and 10. Our experiments measure the average film temperature and therefore a value of 12 indicates uniform shearing within the film and a value of 8 indicates shear localisation at a single position. Instead, if the temperature being used is that of the central position where $z = \frac{h}{2}$, a value of 8 indicates uniform shearing and a value of 4 suggests shear localised at a central position. Equation 11 has been widely used in various studies, *e.g.* by Johnson [50], Spikes [51] and Rahnejat [52], with different values of b ranging from 4 to 24.

3. Methodology

With the IR technique developed by Winer [53], Spikes [54], Reddyhoff [51] and Lu [17], it is possible to obtain 2D maps of the temperature of the contacting surface and the through-thickness average temperature of the EHD film accurately while accounting for the variation in oil emissivity with temperatures and film thickness. A conventional EHD rig was used to produce the sliding/rolling contact to be used for the temperature measurements. Here, an EHL contact was generated by pressing 19.05 mm diameter ball specimen rotating in lubricant against a rotating sapphire disc. The ball and disc were independently controlled to give a range of slide-roll ratios (defined as $SRR = \frac{U_{disc} - U_{ball}}{(U_{disc} + U_{ball})/2}$, where U_{ball} and U_{disc} are respectively the speed of the ball and disc surfaces, positive SRR always indicates $U_{disc} > U_{ball}$ in this research). It should be noted that the circular contact in these experiments represent those found in ball bearing components, rather than line type contacts found in cams and gears.

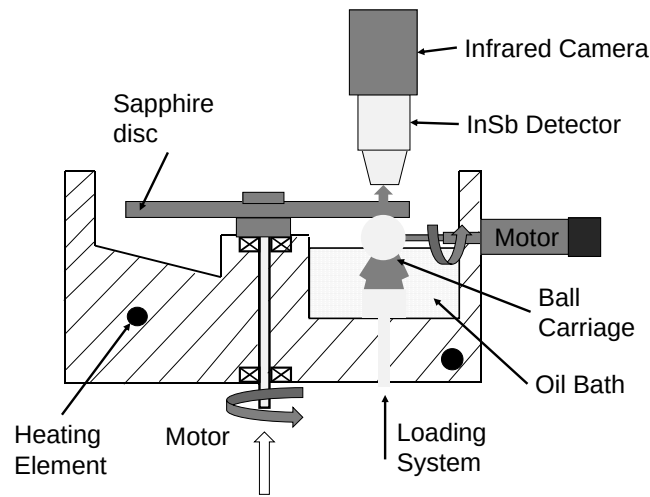


Figure 2. Schematic of experimental apparatus.

Ball and disc surface temperature distributions were obtained using an infrared camera located above the EHD setup which focused onto the contact as shown in Figure 2. The camera contains an InSb detector with a spatial resolution of $5.6 \mu\text{m}/\text{pixel}$ and detecting IR wavelengths between $3 \mu\text{m}$ to $5 \mu\text{m}$.

In order to characterise the effect of surface thermal properties on rheological behaviour, ball specimens of different materials were used to provide a wide range of thermal conductivity and diffusivity values. AISI 52100 steel, silicon nitride (with 90% Si_3N_4) and zirconia (with 94.8% ZrO_2 and 5.2% Y_2O_3) were chosen for the reasons discussed below. The mean and maximum Hertzian pressures of 0.583 and 0.926 GPa were obtained for different specimen material combinations by controlling the applied load. As an unavoidable consequence, the contact half width varied from 80 to 100 μm between the different materials tested. Throughout the testing an infrared transparent disc (100 mm diameter and 3.5 mm thickness) made from single crystal Al_2O_3 was used as the counter specimen. Table 1 shows the thermal and mechanical properties of ball, alongside test lubricants' thermal properties. Steel was chosen since its thermal conductivity and diffusivity are similar to that of sapphire. Si_3N_4 was chosen due to its widespread use as a ball bearing material in industrial high speed/high pressure

applications for reasons of high strength, thermal stability and low density. The use of Si_3N_4 implies higher contact pressure, thermal conductivity and diffusivity compared to steel and sapphire. Finally, zirconia was used as it has similar mechanical properties to steel but also extremely low thermal conductivity and diffusivity.

Table 1. Thermal and mechanical properties of specimens and lubricants.

	Steel	Si3N4	Zirconia	Sapphire	Santotrac 50	PAO4
Young's Modulus	210	310	210	345		
Poisson ratio	0.3	0.27	0.32	0.3		
Load	20	12	20			
Hertz contact diameter	100 μm	80 μm	100 μm			
Conductivity (W/m.K)	21	30	3	23.1	0.104	0.17
Heat Capacity(J/kg.k)	420	760	278	753	1867	2219
Density (kg/m3)	7860	3440	5680	3985	889	820
Diffusivity (mm2/s)	6.36	11.47	1.90	7.7	0.06	0.09

Two base oils were tested in combination with the different ball specimens. Firstly, the traction fluid Santotrac 50 was chosen for its viscosity (24.973 mPa.s at 40 °C) and high traction behaviour, to transmit high power at moderate contact pressure. PAO4 was also tested, which is a widely used synthetic lubricant produced by polymerization of alkenes. Compared with Santotrac 50, PAO4 has lower traction and less thermal stability under EHL conditions. The infrared radiation characteristics of the test samples were measured using FTIR and are presented in Figure 3. The spectral range of the lubricant is 3.3 to 3.6 μm , and the sapphire disc emits radiation longer than 4.3 μm [55]. Detailed information regarding the infrared microscopy technique used to obtain the thermal measurements of oil and component surfaces and temperature profiles across the oil film can be found in Jia and co-worker's research [17].

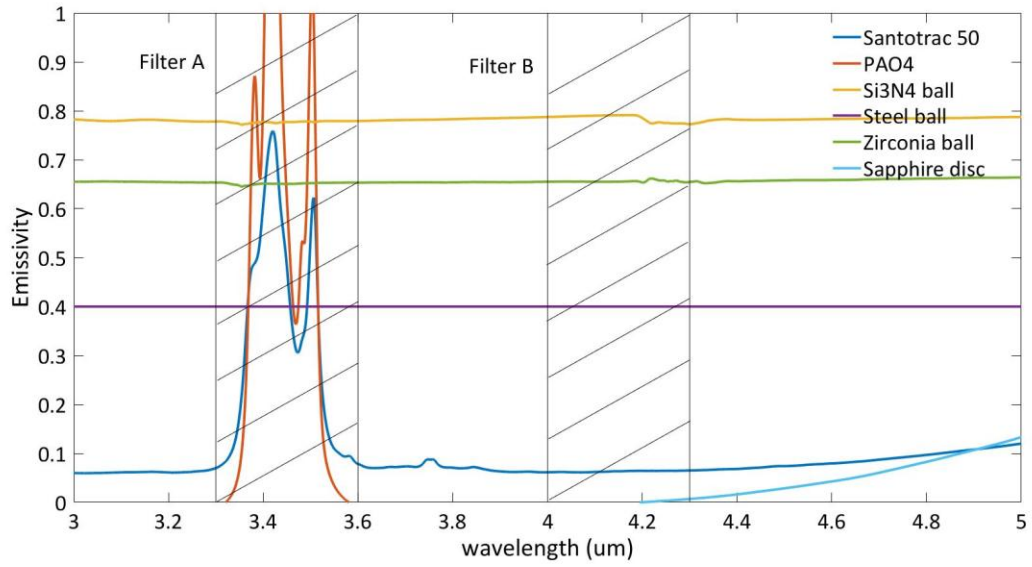


Figure 3. Emissivity/absorptivity spectra from: specimens, test lubricants and filters.

4. Results and Discussion

4.1 Emissivity behaviour

A material's emissivity is used to define the radiation from a sample compared with an ideal black body for a given temperature. Using the infrared spectroscopy technique described previously [17], the emissivity of the two test lubricants was obtained for a range of **mean** film thickness and **average oil** temperatures. This was done for a wavelength range from 3.3 - 3.6 μm with film thickness varying from 60 to 120 nm and temperature varying from 35 to 80 °C as shown in Figure 4. **This method involved recording the infrared radiation from the contact using two bandpass filters and input this into Planck's law equation to obtain emissivity. This was done for a range of calibration film thicknesses and temperatures since emissivity varies depending on these parameters (a detailed description can be found in [17]).** The data points in Figure 4 were obtained from calibration tests at different temperature and film thickness (achieved by adjusting the entrainment speed), while the solid lines are computed by fitting empirical equations which resulted in following constants:

$$\varepsilon_{\text{Santotrac } 50} = h \times 1e^{-9} \times (0.01 + 5e^6 \times e^{-0.07 \times T_{\text{oil}}}) \quad \text{Equation 12 (a)}$$

$$\varepsilon_{\text{PAO4}} = h \times 1e^{-9} \times (0.01 + 5e^6 \times e^{-0.069 \times T_{\text{oil}}}) \quad \text{Equation 12 (b)}$$

The curve fitting lines present a very good agreement with experimental results with an R squared value greater than 0.9 for film thickness from 80 nm to 120 nm. The discrepancy between experimental results and empirical curve fitting lines, **especially at lower film thickness conditions, may** be attributed to noise in the infrared camera measurements due to the oil film being only nanometres in thickness. **Effect of film thickness and temperature on emissivity is interesting and has not been studied in detail. In figure work, comparison with FTIR could provide insights for this.**

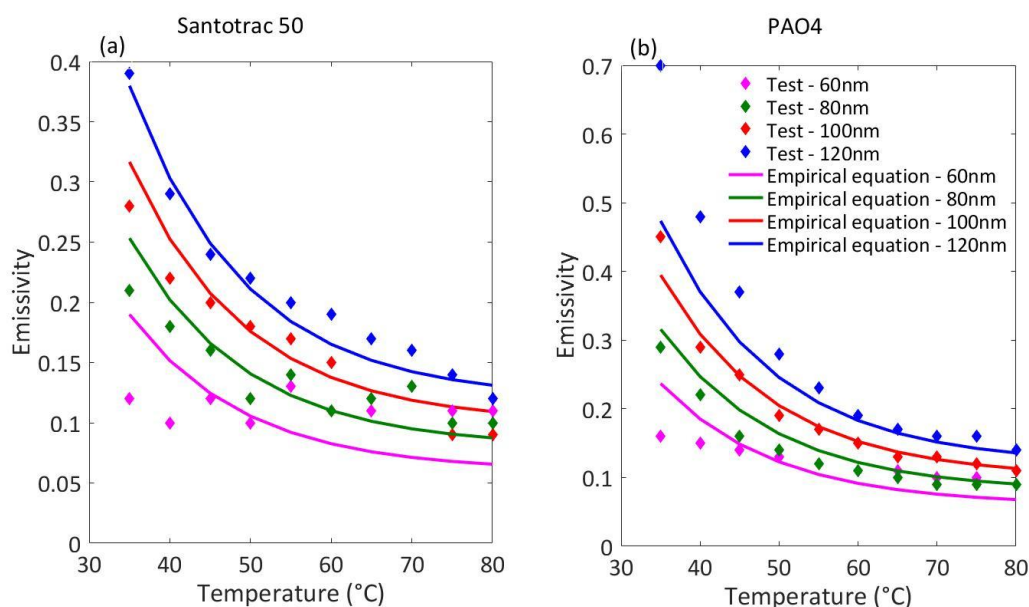


Figure 4 Emissivity of oil film over a range of temperature and film thicknesses with the curving fitting lines and experimental results in the case of wavelength range of 3.3 to 3.6 μm for film thickness varying from 60 to 120 nm and temperatures from 35 to 80 °C.

The emissivity values shown in Figure 4 present integrated values over the detected spectral range as shown in Figure 3. For both Santotrac 50 and PAO4 emissivity decreases rapidly to a constant value at higher temperatures which proves the theory suggested by Schopp [56]. This behaviour is due to infrared polarization effects caused by the alignment of molecules which can lead to the change of intensities of band structures. It stated by Schoen and co-workers [57][58] that for short chains or low weight molecular lubricants, more gauche bands are generated with increasing pressure and this could explain the higher emissivity of PAO4 compared to Santotrac 50, especially at high pressures.

4.2 Friction Behaviour

Friction was measured independently by an MTM2 (PCS Instrument, UK), under the conditions of 0.583 GPa average contact pressure of and 40 °C inlet oil temperature for all the tests. All tests were conducted with SRR varying from 0 to 1.5 with the entrainment speed controlled to ensure the film thickness of the oil was always 100 nm. As shown in Figure 5 (a) and (b), the friction behaviour is remarkably affected by the different lubricants. The friction of PAO4 is relatively low comparing with the traction fluid due to the molecular structure of the PAO4, which comprises a high proportion of flexible linear alkyl chains. These linear chains exhibit less interaction with neighbouring aligned chains and introduce more room for molecules to adapt to the applied stress. For the PAO4, the data has a predictable trend with friction increasing linearly with log (strain rate), according to Eyring studied by other experimental [59] and simulation research [44]).

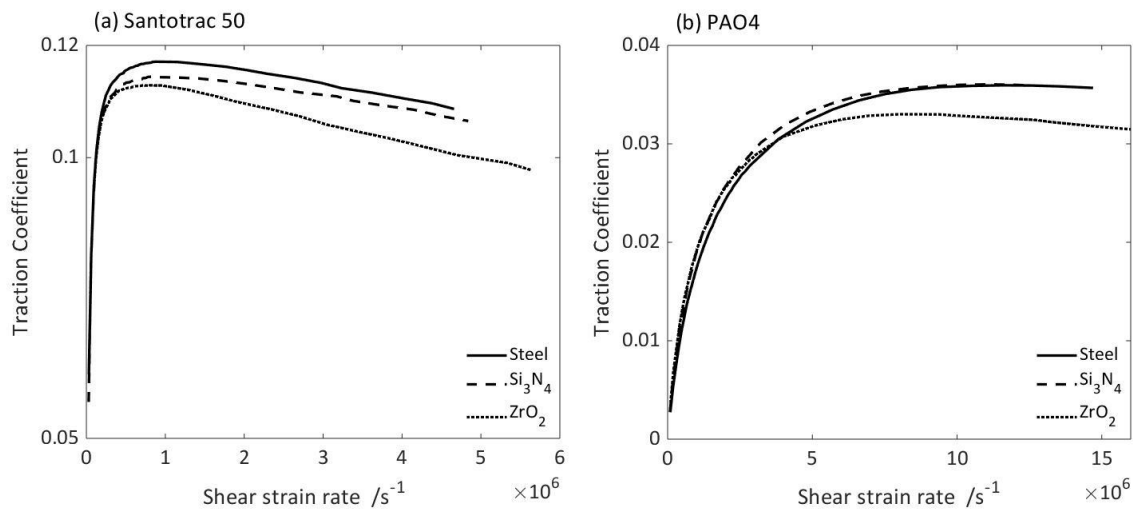


Figure 5. Average friction coefficient versus strain rate at mean pressure = 0.583 GPa and bulk oil temperature = 40 °C. The figures on the left presents the results for Santotrac 50 and right PAO4.

The contact surfaces with different materials have similar friction trends over the whole strain rate range for both lubricants. According to Eyring's model, below the critical strain rate, friction rises linearly with strain rate suggesting constant viscosity. Under these low strain rates, the friction for the three materials present very similar values for both lubricants. This is because shear heating is low at low strain rate and because the contact pressure is equal between the three material contacts. However, significant differences can be observed at high strain rates, especially between the zirconia and other materials. The measured friction of Santotrac 50 shows a shear thinning regime for a material case. The high friction and low critical strain rate for Santotrac 50 are caused by neighbouring molecules sliding over one another subject to molecular stiffness and energy barriers [43]. The critical strain rate can also be observed in PAO4 but with very high value of strain rate. At strain rates above the critical strain rate transition, the zirconia contact shows much lower friction than the other two materials. While the friction of steel and silicon is stable, the friction of zirconia presents a decreasing trend. This can be attributed to the shear localisation at the top surface of zirconia ball as discussed below. It is possible that because of the extremely low thermal conductivity and diffusivity of the zirconia ball, the shear is generated very close to the zirconia ball surface, and with increasing strain rate, the heat is accumulated to the lower surface of oil film and results in the reduced friction shown for PAO4. This demonstrates how EHL

friction can be reduced by controlling the thermal properties of the material within the contact. Further considerations about the properties and phenomenon of zirconia will be discussed in the followed sections.

4.3 Thermal Behaviours Validation of Materials

Figure 6 shows the comparison between heat generation obtained from friction measurements ($q_{total} = \mu \times \bar{P} \times \bar{U}$) and heat generation obtained from the infrared temperature measurements, for all contact conditions tested. The latter being calculated from $q_{total} = q_{ball} + q_{disc}$, in which the heat components have been calculated using Carslaw and Jaeger's equation (Equation 4) inputting the measured ball and disc surface temperatures.

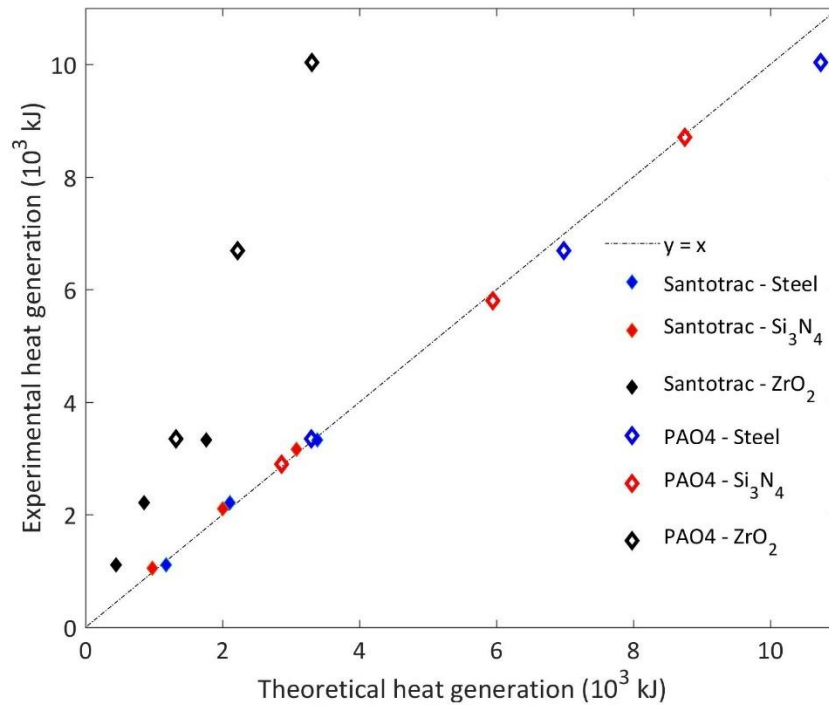


Figure 6. Heat generation of contact surfaces measured from the friction coefficient of steel, silicon nitride and zirconia surface lubricated by Santotrac 50 and PAO4 on MTM2, comparing with theoretical heat generation of contact surfaces calculated using Carslaw and Jaeger's method (Equation 4). The dashed line is the trend line for heat generation.

Except for the zirconia, the results from the thermal analysis and friction measurements show similar values, which goes some way to confirm the validity both of Equation 4 and the experimental measurements of temperature and friction coefficient.

The zirconia, which is characterised by considerably lower thermal conductivity and diffusivity than steel and silicon nitride, shows a very large discrepancy when comparisons are made with the heat generation calculated from friction measurements. This is attributed to the temperature distributions of zirconia not being captured correctly by the infrared camera due to the near-infrared radiative properties of zirconia, as studied by Dombrovsky and co-workers [60]. It has been proven that zirconia is a highly effective thermal resistance material in that only its outer micrometre layers become heated, even when the temperature exceeds 2000 K, by Petrov [61]. Liebert [62], Jantschner [63] and Manara et al. [64] have used zirconia as a thermal barrier coating characterising its infrared-optical properties and heat transfer coefficients. Their results showed extremely low

infrared emittance and high transmittance for coating thicknesses lower than 250 μm , especially at the measurement wavelength range from 3 – 5 μm which is the detection range of the infrared camera used in this research. This combination of high thermal resistance and high infrared transparency explains why the high temperatures of zirconia surfaces were not detected experimentally.

Despite this limitation, it is still possible to obtain the correct temperature of the zirconia surface as follows. First, the heat transferred to the lower zirconia surface can be found from $q_{zirconia} = q_{total} - q_{sapphire}$ where $q_{sapphire}$ is the heat transferred to the upper sapphire obtained by applying Carslaw – Jaeger's equation (Equation 4) to the measured surface temperature of the sapphire disc, and q_{total} is the total generated heat, which can be deduced from measured MTM traction coefficient ($q_{total} = \mu \times \bar{P} \times \bar{U}$). Once the heat transfer to the zirconia surface is found in this way, it can be input into Carslaw – Jaeger's equation to give the required surface temperature. Note: this approach is justified by good agreement between the heat obtained from measured friction and the heat obtained from infrared measurements for the steel and silicon nitride surfaces in Figure 6.

4.4 Lubricant rheology behaviour

Figure 7 shows how the average temperatures of the contact surfaces and oil film depend on SRR between 0 to 1.5. All the tests were controlled at the same average contact pressure of 0.583 GPa and the environment temperatures were 40 $^{\circ}C$. It should be noted that, unlike the contact surfaces of steel and Si_3N_4 , the temperature values of the zirconia contact surface presented in Figure 7 were obtained using the heat balance method described above, rather than being measured directly.

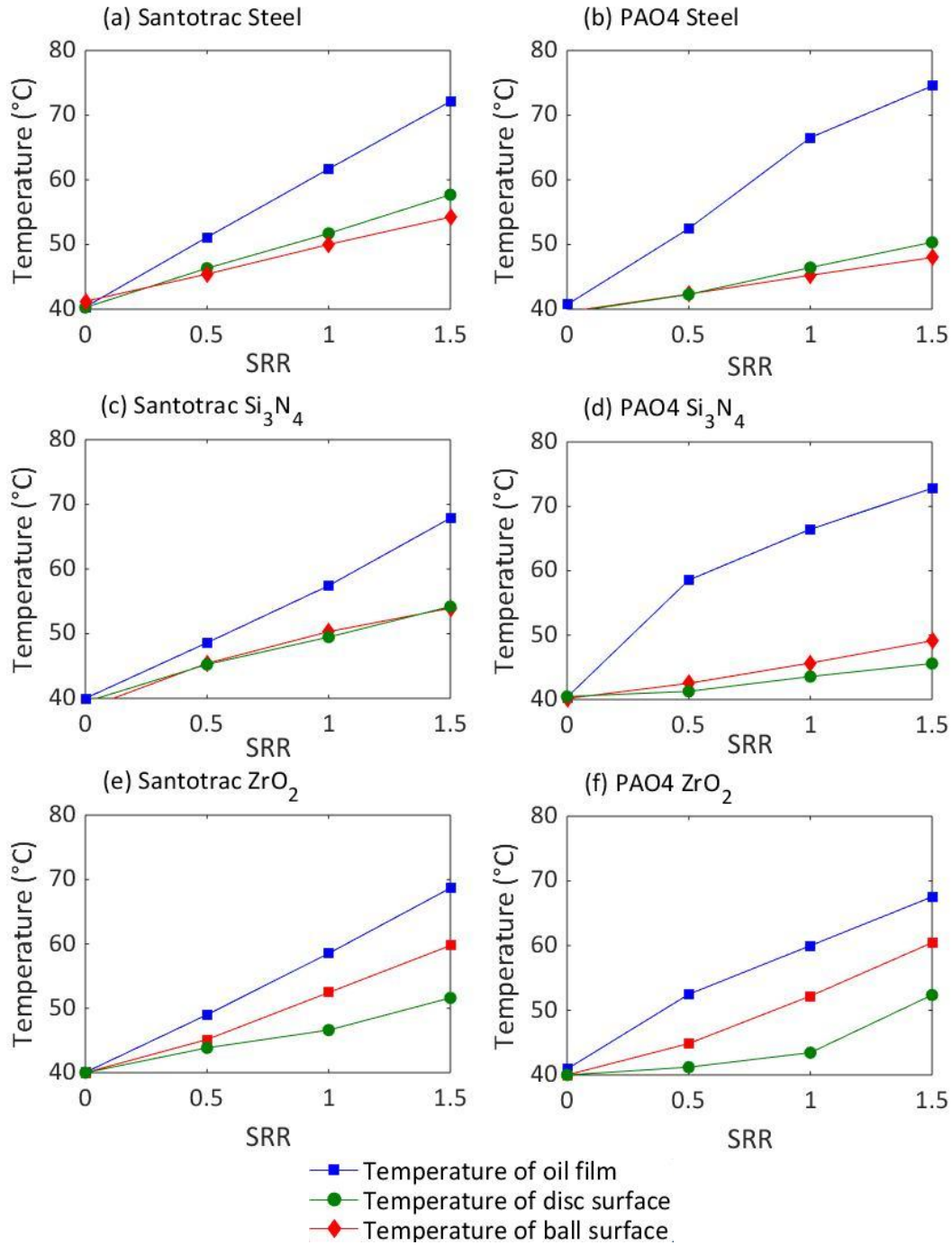


Figure 7. Temperatures of contact surfaces and oil vs. SRR from pure rolling to 150%, carried out at an average contact pressure of 0.583 GPa and 40 °C inlet oil temperature for all tests. The figures on the left column are the results for Santotrac 50 and the right column are for PAO4. a/b ball specimen is steel, c/d ball specimen is Si_3N_4 , e/f ball specimen is zirconia. The blue, green and red lines represent respectively the temperatures of oil film, ball and disc surface.

It is observed that the mean oil temperature is significantly greater than both contact surfaces, which results from the low oil thermal conductivity. This agrees with Manton and co-workers' study [65] and simulation study of Clarke and co-workers [66][67]. It is also obvious that there is a large difference between the temperatures of the sapphire disc and zirconia ball surface. The temperatures of zirconia ball are always higher

than the temperatures of sapphire disc, because the low thermal conductivity and high thermal resistance properties of zirconia compared to sapphire, as studied by Kennedy [68].

4.4.1 Average film temperature

As described above, the heating in an EHL film can be generated by either uniform shearing and localised shearing along a horizontal plane and the method deduced by Archard [21] can be used to calculate the temperature of the oil film based on each of these mechanisms. To test which of these shearing mechanisms dominates, figure 8 presents the average oil temperature rise comparisons between experimental results and theoretical calculations for SRRs up to 150%, obtained for a 0.583 GPa mean contact pressure and oil inlet temperature of 40 °C. The temperature rises measured through experiments are shown as blue points in the figure whereas the red points indicate the theoretical temperature rise calculated based on the assumptions of uniform and local position shear heating. More specifically, the through-thickness average temperature that results if the oil is shearing *uniformly* due to Couette flow (red points) is calculated by applying Equation 9 (d) to the measured surface temperatures while the through-thickness average temperature that results if the oil is shearing along a localised plane (green points) is calculated by applying equation 10 (e) to the measured surface temperatures. The errors between the average temperature rise measured by experiments and theoretical calculations are also shown in Figure 8 as red and green bars.

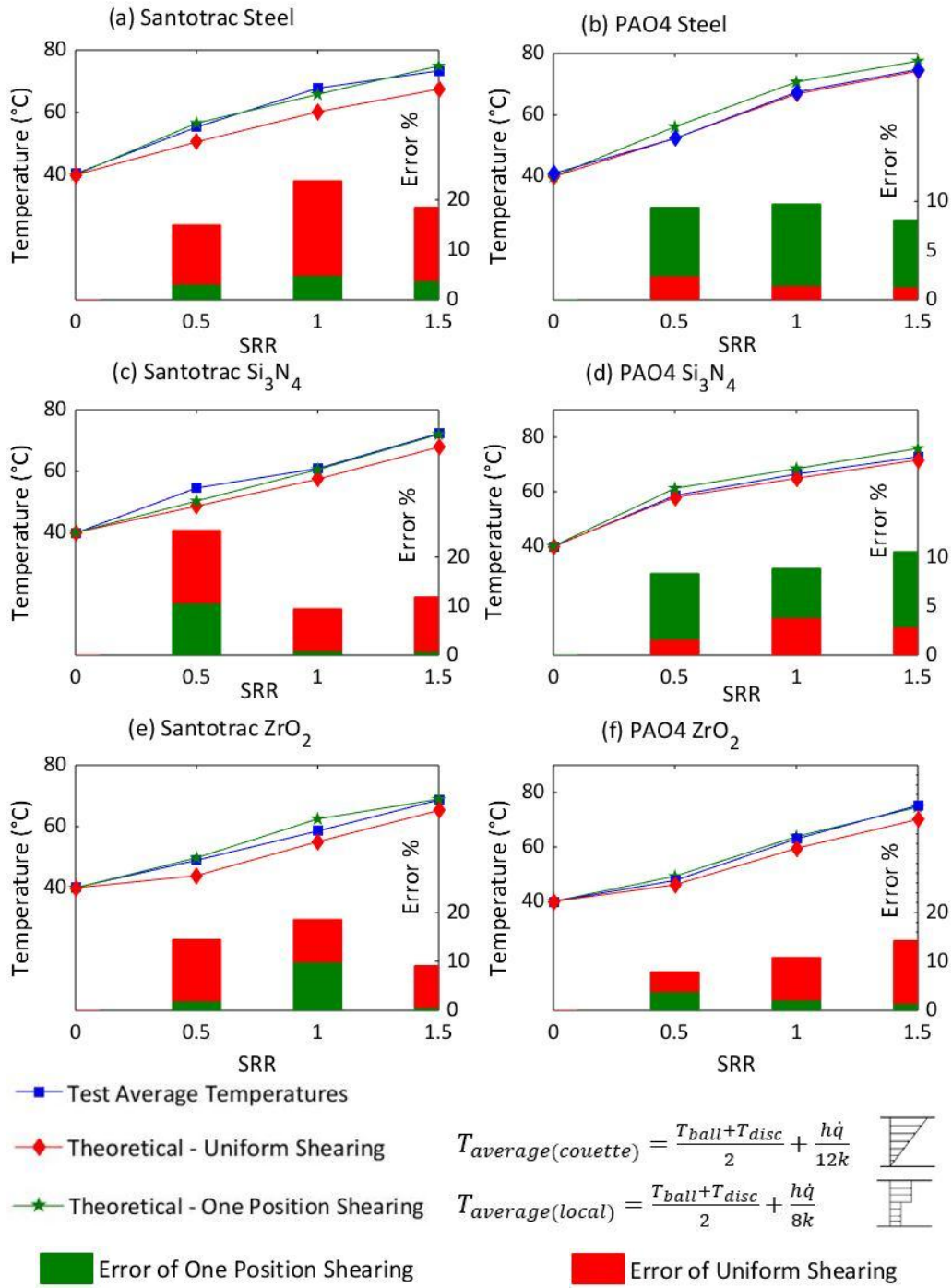


Figure 8. Through-thickness averaged oil temperature rise comparisons between experimental results and theoretical calculations with the SRR up to 150% carried out at an average contact pressure of 0.583 GPa and 40 °C inlet oil temperature for the tests. Figures on the left column are the results for Santotrac 50 and the right column are for PAO4. a/b ball specimen is steel, c/d ball specimen is Si_3N_4 , e/f ball specimen is zirconia. The blue, red and orange lines represent respectively the experimental results, and the theoretical results with uniform and local shearing.

For Santotrac 50, there is generally good agreement between measured average oil film temperatures and those calculated assuming shear (and hence heat generation) is localised in a single position (Figure 8 (a), (c) and (e)), with only small discrepancies for some slide roll ratios.

1 The temperature profiles of Santotrac 50 suggest that inhomogeneous shear exists for all of the contact surface
2 material combinations tested. The temperature rise caused by energy dissipation leads to a viscosity reduction at
3 the maximum temperature position across the oil film, which in turn extenuates shear localization. The
4 occurrence of shear localisation and frictional heating explains the friction decrease of Santotrac 50 for all three
5 contacts after the transition strain rate shown in Figure 5. This phenomenon caused by pressure and temperature
6 rise from frictional heating is also proved by similar research on velocity profile measurements [47] and
7 molecular dynamics simulations [44].
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11 In contrast with Santotrac 50, the measured average temperatures of PAO4 oil films are close to the calculations
12 based on the assumption of uniform shear heating (Figure 8 (b) and (d)) except for the zirconia (Figure 8 (f)).
13 This suggests that for the mean contact pressure of 0.583 GPa, a Couette flow profile develops within the PAO4
14 oil film with a constant velocity gradient for the contact surfaces of steel and silicon nitride ball as shown in
15 Figure 1. The heat generated by shearing is therefore uniformly distributed across the oil film. It is possible to
16 calculate the Brinkman value using Equation 5 (b) and inputting the average temperatures of oil film and contact
17 surfaces. In all Santotrac tests, for all the contact surface materials used, the Brinkman values vary from 3.89 to
18 6.165, which are all higher than the critical point of 3.5. This supports the prediction of the existence of shear
19 bands in this traction fluid film. For PAO4, Brinkman values with the contact surfaces of steel and silicon
20 nitride ball vary from 1.04 to 3.695 which in most cases suggest no shear localisation, and again consistent with
21 the flow deduced from temperature measurements.
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29 The average temperatures of the PAO4 oil films with the zirconia ball present a different rheological behaviour
30 compared with those of the contact consisting of steel and silicon nitride. As shown in Figure 8 (f), the
31 experimental results show a good agreement with the calculation assuming shear is localised in a single position
32 instead of uniform shearing. This explains the different behaviours of friction coefficient delivered by zirconia
33 contact surface shown in Figure 5 (b) which decreases as the slide – roll ratio increases more steeply than for the
34 other two materials.
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39 These results show that, when calculating the mean temperature of the oil film with the commonly used
40 Archard's equation (equation 11), the value of the constant b should be 8 when considering Santotrac lubricated
41 contacts (due to shear localisation) and 12 when considering PAO lubricated contacts (due to uniform shear).
42 The exception to this is PAO the lubricated contact with zirconia surface as this also favours localised shear and
43 hence a value of b of 8.
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48 **4.4.2 Through thickness temperatures**

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50 In this section, the through-thickness temperature profile is deduced from the measured surface and mean oil
51 film temperatures. The following methods assume that the temperature profile varies as a polynomial equation
52 whose shape is defined by a combination of the energy equation and heat transfer law. These correspond to
53 localised (non-Couette flow) or evenly distributed shear heating (Couette flow). Three assumptions were made:
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- 58 1. The temperature varies with film thickness as a third order polynomial form as shown in Equation 13.
59 Unlike the theoretical methods described in Section 2.2.2.2, the cubic polynomial is used from now on to
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increase the accuracy/flexibility and predict more realistic temperature profiles, (*i.e.* the heat generation will not be based on the simplified rheological behaviours shown in Figure. 1). The use of a third order polynomial is possible because the average temperature of oil is used as a further constraint (in addition to the measured surface temperatures) rather than being used as a validation as was the case above.

$$T_{oil} = a \times z^3 + b \times z^2 + c \times z + d \quad \text{Equation 13}$$

where z is the vertical height from ball surface to disc surface and a, b, c, d are constants to be determined.

2. Compression heating is negligible [69] and therefore shear heating is the only heat generation mechanism.

$$q_{ball} + q_{disc} = q_{total} = \mu \times \bar{P} \times \bar{U} \quad \text{Equation 14}$$

where $\bar{U}$ is the sliding speed, $\bar{P}$ is the mean contact pressure, q_{disc} and q_{ball} are the heat transfer rate to disc and ball surfaces and μ is the traction coefficient measured directly from MTM2 rig.

3. Generated heat is conducted rapidly into the contact surfaces, no heat convective heat transfer is allowed. In this case, the temperature profiles are analysed differently, depending of the shear distribution shown by Figure 8.

- 3.1. For the localised shear heating which can be used for the traction fluid Santotrac 50, the heat conduction is given by

$$\kappa_{oil} \times \frac{T_{max} - T_{ball}}{Z_{Tmax}} = q_{ball} \quad \text{Equation 15 (a)}$$

$$\kappa_{oil} \times \frac{T_{max} - T_{disc}}{h - Z_{Tmax}} = q_{disc} \quad \text{Equation 15 (b)}$$

where κ_{oil} is the test lubricant thermal conductivity, Z_{Tmax} is the height within the film at which the temperature is a maximum, T_{max} is the maximum temperature across oil film, T_{disc} and T_{ball} are the disc and ball surface temperatures.

- 3.2. As shown by Figure 8, the PAO4 lubricant exhibits a Couette flow profile for steel and silicon nitride. In these cases, assumption 3.1 is replaced by one which states that heat generated by shear heating is uniformly distributed across the oil film.

$$-\kappa \frac{d^2 T_i}{dz^2} = \frac{\bar{\tau}_i^2}{\eta_i} \quad \text{Equation 16}$$

where $\bar{\tau}_i$ is the uniform shear stress and η_i represents the viscosity for each layer varying with temperature which can be obtained from Roeland's Equation.

For the conditions of zirconia, due to the lack of temperature measurements, Equation 4 was combined with Equation 14 and 15 to calculate temperature of zirconia, T_{max} and Z_{Tmax} in advance.

According to this method, given values of average oil film temperature, maximum temperature location and the temperatures of two contact surfaces, the temperature profiles as the cubic polynomial shape as described in Equation 13 can be plotted.

The oil film was divided into ten layers from the ball to the disc surface during each analysing process (note: this approach is valid provided the thickness of each layer is greater than that of a single lubricant molecule). The temperature profile was calculated by a MatLab script using the equations described above. More detailed information of deducing the temperature profiles of oil film can be found in the previous publication by Lu et al. [17].

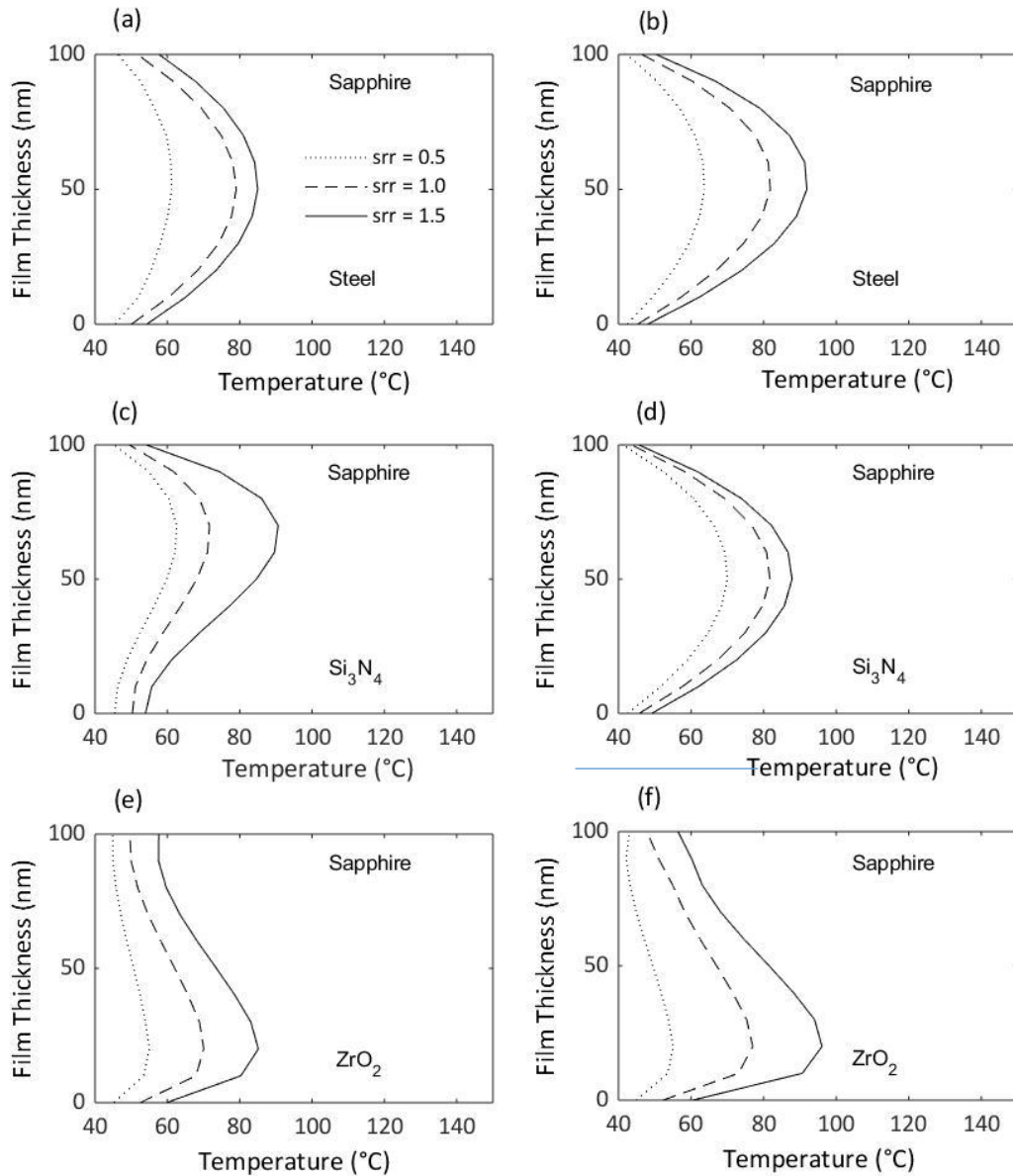


Figure 9. Temperature profiles across the oil film at different SRRs for each test lubricants and ball specimens. The figures on the left column are the results for Santotrac 50 and the right column are the results for PAO4. a/b Ball specimen is steel, c/d ball specimen is Si₃N₄, e/f ball specimen is zirconia.

Using the method described above, temperature profiles were calculated, based on the measured surface and average oil temperature, and the results shown in Figure 9. The contact pressure was made equal for all tests to maintain the constant shear heating. While details of temperature profiles shown in Figure 9 are obviously governed by rheological properties of the fluids described above, the thermal properties of contact materials have a strong effect, especially for shear localization.

As shown in Figure 9, the oil film temperatures increase with slide roll ratios which prove that the shear heating is the main cause of temperature rise. The thermal properties of contact surfaces are also shown to have an important effect on the position of shear heating in the Santotrac 50 film (Figure 9 (a), (c) and (e)) which is in

agreement with the research of Habchi [70]. Moreover, in a new study by Ewen et al. [71], it was proven that the use of walls with different thermal conductivities lead to Asymmetric Melting (AM) for the fluids which would cause the shear localisation. Therefore, for the through-thickness temperature profiles deduced here, in the case when the steel ball with thermal conductivity close to sapphire disc is used, the heat is removed equally by the ball and disc. As a result, maximum lubricant temperature within the oil film is located at the centre of the oil film. When silicon nitride is used as the contact surface, as the thermal conductivity and diffusivity of the ball is higher than the sapphire disc, more heat is transferred to the ball.

In contrast, the zirconia ball has extremely low thermal conductivity and diffusivity compared to the sapphire disc and, thus, heat is instead transferred to the sapphire disc. Moreover, due to the near – infrared radiative properties of zirconia, there was no temperature rise observed on the ball surface during the experimental measurements, the temperatures of zirconia presented in Figure 9 were calculated with the methods described in section 4.4.2. In this case, a temperature discontinuity must be present on the top of the zirconia surface. Specifically, the heat must accumulate in a thin layer with thickness of a few nano-meters as studied by Petrov [61] and Keiteb [72].

It is interesting to note that for positive SRRs of Santotrac 50, the heat energy transferred to ball surface is 49.55% at a SRR of 50% and the decreases to 44.42% at slide roll ratio of 150%; this proves the theory put forward by Yang and co-workers [73] that heat conduction dominates when the sapphire disc runs at high speed.

The temperature profiles of PAO4 for contact surfaces with different thermal properties are shown in Figure 9 (b), (d). The profiles obtained with uniform shear distributed across the oil film with the contact surfaces of steel and silicon nitride show parabolic shapes with the oil film's maximum temperature being located near to the central position in all conditions. The temperatures of oil film still increase with sliding speed, however, the thermal properties of the surfaces show only marginal effects on the rheological behaviour of fluid. As described above, the temperature profiles of PAO4 generated by zirconia surface presented a different rheological behaviour of shear localisation, which is shown in Figure 9(f).

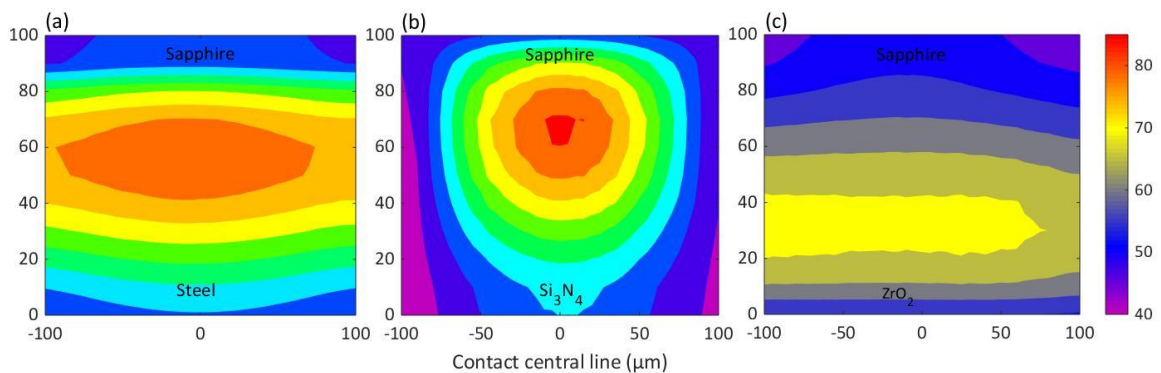


Figure 10 Through film thickness temperature maps for contact lubricated by Santotrac 50, for (a) steel ball, (b) silicon nitride ball, (c) zirconia ball.

Temperature distributions of the oil within the film for Santotrac 50 in the x (along the entraining direction) and z directions are presented in Figure 10 under the conditions of 100% slide roll ratio with contact surfaces of steel,

silicon nitride and zirconia. The method used to obtain these maps is also detailed in [17].. Figure. 9 shows that the maximum temperatures occur close to the centre of x axis, which confirms the assumption of parabolic heat flux distribution across the contact area due to a Hertzian contact pressure distribution proposed by Winer [74]. The contact with the silicon nitride ball and sapphire disc generates 45 °C of maximum oil temperature rise, while the temperature rise of the silicon nitride ball and sapphire disc surfaces are only 10 – 15 °C. Note that the large shear heating zone close to ball surface observed in Figure. 10 (c) for the zirconia ball with low conductivity. This shows a good agreements with the simulation carried out by Ewen et al. [71], Kim and Sadeghi [75] and Habchi and Vergne [76][77].

5 Conclusion

A combination of experimental tests and theoretical calculations have been used to investigate the thermal property effect of lubricants and sliding components on rheological behaviours of fluids due to frictional heating. Three ball materials and two lubricants were subjected to a range of slide roll ratios (0% - 150%). Results show that thermal properties are a dominant factor in affecting heat generation and friction; however, the exact relationship between these properties and the system response is complex.

The main conclusions can be summarised as follows:

1. PAO4 and Santotrac 50 emit IR radiation at the same wavelengths due to them both being the hydrocarbon lubricants. However, the PAO4 has higher emissivity and more radiation emits than Santotrac 50 according to the measurements of FTIR.
2. The zirconia presents infrared transparent radiative properties and perfect thermal resistance. This results in a micro-scale heated layer with thickness less than the measurement depth of the camera and hence restricts the measurements of zirconia contact surface temperatures. However, the temperatures of zirconia surface can still be obtained by applying Carslaw – Jaeger's equation and combining in a heat balance using friction coefficient measured by MTM2. This method makes the IR method more flexible and can be used in the future whenever one contact surface cannot be detected.
3. Shear heating within the oil film for contacts lubricated by Santotrac 50 occurs predominantly at a single position where the maximum temperature across the film located, which strongly suggests shear localisation. The PAO4 when lubricating contact surfaces of steel and silicon nitride, on the other hand, presented a uniform shear across the film with the heat generation being distributed evenly with maximum temperatures across the oil film in most conditions a located close to the midpoint.
4. It follows that, when calculating the average oil film temperature with the commonly used Archard's equation (equation 11), the value of the constant b should be 8 when considering Santotrac lubricated contacts and 12 when considering PAO lubricated contacts. The exception to this is PAO the lubricated contact with zirconia surface which should also have a value of b of 8.
5. In all cases, the thermal properties of the contact surfaces show an important effect on shear localization. With the extremely low thermal conductivity and diffusivity of zirconia, shear localisation occurs close to the surface of zirconia ball which is markedly different rheological behaviour compared to the contacts involving steel and silicon nitride. The silicon nitride ball, with its higher thermal conductivity and diffusivity than the sapphire disc, produced shear localisation close to sapphire disc. As the steel ball has the similar thermal conductivity and diffusivity with sapphire disc, the shear localisation is observed close to the central point of the oil film. The temperatures of the oil film have been found to increase with sliding speed.
6. The critical strain rate, where friction starts to reduce due to shear heating, has been observed for both lubricants using MTM measurements. Both oils give low friction when lubricating zirconia contact surfaces, which results from shear localisation due to the low surface thermal conductivity.

Implications of this study are that temperature and friction information of oil and contact surfaces can be used to understand rheological behaviour. It also shows that a materials and lubricants with different thermal properties

can be selected to control friction in practical applications. In addition to this, the obtained in-contact temperature data will be used in future to validate and inform CFD (*e.g.* [36][7]), numerical modelling (*e.g.* [75][76][78][79]) and molecular dynamics [71] simulations.

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