The Frictional Behaviour of Rubber on Ice

Daniel Higgins

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Declaration

I declare that this thesis has been composed by myself and except where otherwise stated the work contained herein is my own.

Daniel Higgins.
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Abstract

A study of the friction between styrene butadiene rubber and ice has been undertaken. The instrument used to perform the measurements (a pin-on-disc tribometer) was designed, constructed and developed specifically for this task and so has been optimised for this tribosystem. The versatile and compact design is easily accommodated in a domestic freezer and has been utilised to measure friction in both steady-state and transient regimes.

The experimental component of the study is comprised of two separate and novel parts. Firstly, steady-state (constant temperature, load and speed) measurements of the sliding friction are placed on a friction map in speed-temperature space. This enables the frictional behaviour of this complex system to be presented in a clear and uncomplicated fashion. The magnitude of friction measured within the ranges of temperatures and speeds is determined by both an ice-driven thermal friction process and the viscoelastic properties of the rubber. Observations of the ice wear surfaces are made using low temperature scanning electron microscopy (LT-SEM), the morphologies of which are linked to the temperature-speed position on the map at which the samples were worn.

The second experimental section relates to the transient nature of friction as it changes from static to sliding regimes. This study uncovered the temperature, pressure and force rate dependencies that can be utilised to increase the (typically) very low friction on ice. Application of this aspect of rubber-ice friction is particularly important to the automotive industry where transient regimes are widely used in safety and performance enhancement systems such as anti-lock braking systems and active stability control.
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Introduction

One of the most important components on your car are the tyres. They are the only part of the vehicle in contact with the road, so no matter how well designed the chassis geometry, or how much power the vehicle has, it cannot be utilised without sufficient grip to the road. As well as delivering the performance the vehicle is capable of, they are of utmost importance to driving safety. These factors become much more prominent when it comes to driving in winter conditions.

As well as improving tyre technology, active traction systems (TCS) are increasingly being implemented into vehicles, not only to control the power delivery of high powered vehicles, but also to aid tyre grip on slippery surfaces. As more market niches are being created, traction control systems need to be implemented in different ways, by calibration through test programmes and utilising computational simulations.

Computational models are now commonplace in automotive design. They allow simulations to be run on an entire virtual vehicle even before a prototype is produced, reducing the costs of engineering, and assessing the implementation of control systems, such as traction control, anti-lock braking systems (ABS), or the application of both in an electronic stability programme (ESP®). All of these systems can be programmed and simulated virtually before verification during winter testing, but this can only be done successfully if accurate empirical data is available. For example, the relatively basic ABS system has been found to actually increase braking distances if incorrectly calibrated compared to a non-equipped vehicle.

All major automotive companies implement a comprehensive winter testing programme at specifically designed facilities in northern Europe and North
America, allowing application, calibration, and performance of their products to be verified in extreme conditions. Overall, this is an essential and worthwhile undertaking, especially when programming and calibrating control systems. The dynamic tests usually take place on specially prepared frozen lakes, but as these measurements are performed in an open air environment, changes in conditions from day to day make data reproducibility difficult.

Laboratory tests, while not always applicable to specific vehicle behaviour, can be used effectively to ascertain the physics behind the elemental behaviour of, for example, a single tyre tread block. As computational simulations are basically an extrapolation of small-scale behaviour to full-scale systems, it was envisaged that laboratory tests could be used successfully to predict the behaviour of tyres, and hence vehicles, on icy surfaces. Although it will not replace winter testing, laboratory research could greatly improve the accuracy of system programming and selection process for tyre trials, saving valuable time before this expensive testing process commences.

The technology in winter tyres has improved greatly over the past decade or so. The performance of quality winter tyres over “all-season”, summer, and even studded tyres is marked, and this has led to legislation to be put in place to enforce winter tyres, in countries such as Finland, Latvia, and Sweden making them compulsory during the winter months. This increase in popularity has heightened the degree of research being carried out in this field, with more companies than ever producing winter-oriented tyres, something that could not be afforded when the demand was not there. A lot of emphasis has been put on footprint design, with aggressive tread patterns and siped blocks to promote mechanical grip on ice, but it has also been discovered that some rubber compounds promote significantly more grip than others. Laboratory experiments can easily ascertain basic compound friction on ice, so more fundamental rubber friction research can give a good indication of the prospective tyre grip available from the material.

In this investigation, an understanding of the basic underlying principles of rubber and ice friction is established through references to a number of scientific publications and technical texts (Chapter 2). The concepts of several frictional
mechanisms is introduced, dependent on both rubber and ice properties, and their bearing on how this may affect the grip produced by tyres on icy surfaces. It is established that this tribological system is particularly complicated in nature with strong dependencies on temperature, pressure, and speed, factors which are regarded as non-determining in classical friction works.

To perform accurate laboratory measurements, the design and development of a suitable instrument was undertaken (Chapter 3). Investigation into designs implemented by previous researchers as well as those commercially available showed that specific design would be more productive than modifying a commercial instrument to suit this particular system. This way, the motion control and force measuring systems could be tailored to ensure the high degrees of accuracy and flexibility, essential when exploring both the steady-state and dynamic frictional properties of this tribosystem.

Chapter 4 contains steady-state friction experiments and an indirect investigation of the interfacial ice layer using low temperature scanning electron microscopy (LT-SEM). Friction measurements show that there is significantly higher friction than may be expected on ice at temperatures close to the glass transition temperature ($T_g$) of the rubber. Multi-parameter friction measurements presented in a novel fashion (friction map) clearly show areas in which high and low friction is prevalent. Using this map, samples worn under conditions of significantly different friction were investigated using LT-SEM, uncovering the particular friction mechanisms responsible.

The experiments presented in Chapter 5 regard the transient behaviour of the rubber-ice tribosystem. Until recently, most research on this subject has been based around the steady-state behaviour, i.e. sliding at constant speeds at fixed temperatures and pressures. Comparisons made between the elemental study presented here and full-scale tyre behaviour are favourable, indicating an interfacial process is responsible for frictional behaviour common at both scales.

Part of the work carried out in Chapter 5 was presented at the 19th International Association of Vehicle Systems Dynamics Symposium "Dynamics of Vehicles on
Roads and Tracks” in August and September 2005 at the Politecnico di Milano, Italy. Of the 200 papers presented at the conference, the Best Paper Award was conferred upon our research (V. Ivanović, J. Deur, M. Kostelac, Z. Herold, M. Troulis, C. Miano, D., Hrovat, J. Asgari, D. Higgins, J. Blackford, V. Koutsos, “Experimental Identification of Dynamic Tire Friction Potential on Ice Surfaces”, In XIX IAVSD Symposium, Politecnico di Milano, Milan, Italy, 2005 (to appear in 2007). This paper combines small-scale fundamental lab testing of rubber-ice friction with measurements made on a tyre/wheel/vehicle scale. We identified that specific transient behaviour between static and sliding friction is a fundamental property of the ice-rubber interface (not a property arising from vehicle dynamics; though observed in vehicle dynamics).

Furthermore, work carried out in Chapters 4 and 5 are being prepared for journal publication.

Throughout this work, there has been some repetition of key aspects relating to each individual section. Chapters can, for this reason, be read independently of the main body of work.
Chapter 2.

Rubber-ice friction literature review and background theory

The subject of the study is the rubber-ice tribosystem, but as both materials are inherently non-conformal to classical friction and contact theory, it is important to look at the materials separately in order to decipher the individual dependencies. The knowledge gained in reviewing the published works in this area was essential when analysing the experimental data in later chapters of this thesis.

2.1 Friction basics

The basic concept of the coefficient of friction originates from the empirical observation that the friction force is proportional to the applied vertical load. This linear relationship holds for most materials over a wide range of operating conditions and is independent of these conditions, and of contact area. This is stated by Amontons in his Law of friction:

\[ F = \mu N \]

where the ratio of the friction force \( F \) to the normal load \( N \) is defined as the coefficient of friction, \( \mu \).

It has been accepted, however, that both the friction of ice (see Section 2.2) and that of rubber (see Section 2.3) do not strictly follow Amontons' law, and it was acknowledged by Wilkinson (Wilkinson (1953)) that the relationship between the friction force and the normal load is not constant, but depends on the prevalent pressure \( p \), speed \( v \) and temperature \( T \) conditions during measurement, i.e.

\[ F = f(p, v, T)N \]
It will be seen later in this chapter that these are not the only parameters of importance when assessing the magnitude of friction between these two materials, but it does give an indication that the energy transferred to the interface, through the environmental temperature or by work done in moving or deforming the materials has a significant effect on the frictional properties of the system. This is particularly important when looking at the friction of ice, which is often reduced to low values ($\mu < 0.1$) by the presence of a melt water layer at the interface. Energy transferred to the interface in the form of heat will melt the interfacial ice layer, producing melt water, so it could be suggested that higher levels of heat melts more water, leading to lower friction. This phenomenon is discussed in Section 2.2.

The friction of rubber also depends on the conditions of measurement, and factors such as the speed (frequency) and temperature of measurements have a very pronounced effect on friction. These effects and others regarding rubber friction are discussed in Section 2.3.

Overall, the tribological system of rubber-on ice is non-trivial and the dependencies on temperature, speed, and pressure are marked, as shown in the experimental chapters later in this work.
2.2 Ice properties

2.2.1 Temperature
As this friction system operates predominantly in the region close to the melting point of ice, it is clear that the ice properties play an important role on the resultant friction. As well as the sharp phase change occurring at 0 °C, other phenomena occur close to the melting point on which friction is dependent.

Assuming a constant, moderate sliding speed, at high temperatures, the ice melts due solely to the environmental temperature, resulting in low friction ($\mu < 0.1$). At sufficiently low temperatures the ice does not melt and behaves like a dry crystalline solid resulting in high friction ($\mu > 1.0$) (Roberts and Richardson (1981), Southern and Walker (1972)). At intermediate temperatures frictional heating melts a small amount of the ice at the interface, producing intermediate $\mu$. As the temperature increases, the effect of the combined ambient temperature and frictional heating is more prolific melting, hence decreasing $\mu$ with increasing temperature.

A study carried out by Akkok et al. (Akkok et al. (1987)) produced analytical models based on experimental results of preceding works. With regard to atmospheric temperature, the model predicting the coefficient of friction $\mu$ takes the form:

$$\mu \propto (T_f - T_a)^f$$

where $T_f$ is the characteristic frictional temperature of the interfacial lubricating layer (usually very close to the melting point), $T_a$ is the temperature of the approaching ice (or the atmospheric temperature) and $f$ is an exponent based on the numerical data. Favourable comparisons are made to data provided by Conant et al. (Conant et al. (1949)), Evans et al. (Evans et al. (1976)), Forland and Tatinclaux (Forland and Tatinclaux (1985)) and Oksanen and Keinonen (Oksanen and Keinonen (1982)). Although it is suggested by Akkok et al. that the relationship with rising temperature should be linear in theory (Evans et al. (1976)), it is more commonly non-linear (Barnes et al. (1971), Bowden (1953),...
This behaviour is attributed to the reduced hardness of ice at high temperatures and to the increased hydrodynamic drag effect as the temperature approaches 0 °C (Akkok et al. (1987)).

This rise in friction with falling temperature is apparent in all studies. With a continued fall in temperature, generally a plateau is reached, whereby a further fall in temperature does not yield a significant rise in friction. The temperature point of the discontinuity that signifies the onset of a plateau changes however from study to study. Bowden and Hughes report this point to be at ≈ −40 °C for ice on ice, −10 °C for ski wax on ice and as high as −5 °C for brass on ice and −2 °C for PTFE on ice (Bowden (1953), Bowden and Hughes (1939)). These different points allow conclusions to be made about the source of the low friction being frictional heating, but also correlations of the friction to the thermal properties of the sliders to be made (Bowden and Hughes (1939), Oksanen and Keinonen (1982)).

2.2.2 Contaminants

Impurities (salts) in the ice significantly affects the friction on ice (Roberts and Lane (1983)). Dissolved impurities in the water are not incorporated into the ice crystal lattice, but are “frozen out” of the crystals, resulting in highly concentrated pockets of frozen brine at the crystal interfaces, and bulk surface (Hobbs (1974)). The effect of brine at the interface is that the melting point of the ice is effectively lowered, so that lubrication due to frictional melting is initiated at a much lower temperature.

Roberts and Lane (Roberts and Lane (1983)) carried out a systematic study of this effect by using different salt species and concentrations. The most significant point gained from the investigation was that of the dramatic drop in friction values with even a small amount of salt present (a concentration of 0.01 M of sodium chloride (NaCl) retards the melting point of the ice surface by 8 °C). From Figure 2.1 it can be observed the clear independence of friction on speed when salt is present between the temperatures of −20 °C and 0 °C, compared to the speed dependence exuded on “pure” ice. The temperature at which high friction is regained corresponds well to the eutectic temperature (the lowest temperature at which the
solid and liquid phases can coexist, or simply the melting point of the salt solution) of the salt solution that, in the case of NaCl, is \(-21.2 \, ^\circ C\).

![Figure 2.1. Speed dependence of the friction of natural rubber on "pure" (solid lines) and salted (0.1 M NaCl, broken lines) ice for a range of sliding speeds from $10^{-6}$ to $10^{-2} \, \text{ms}^{-1}$ (Roberts and Lane (1983)).](image)

Raraty and Tabor also made this observation previously in experiments measuring the adhesion of ice to stainless steel (Raraty and Tabor (1958)). The salt in this case was a commercial product, Teepol (sodium dodecyl sulphate), which was applied to the stainless steel as a wetting agent, but actually changed the ice properties by forming a salt solution at the ice-steel interface. Again, the melting point was reduced to that of the eutectic of the Teepol-ice system.

These observations show the significant dependence of salt present in the tribosystem, and the clear need to ensure a clean environment when measuring ice friction.

Salts are not the only impurities that can greatly affect the friction on ice. An experiment performed by Gnörich and Grosch (see Figure 2.2) illustrates quite clearly the effect of ice dust (analogous to frost) and water on a highly polished ice track. Bowden and Hughes (Bowden and Hughes (1939)), who note that if a small amount of water was added to an ice track, the friction dropped significantly, had observed similar effects previously.
2.2.3 Age of the ice

From previous experiments carried out (Conant et al. (1949), Roberts (1981a), (1981b), Venkatesh (1975), Wilkinson (1953)) it was noted that the age of the ice track was of considerable importance to friction measurements. Measurements taken on fresh ice (just a few hours old) revealed low friction, but spurious and hence showed little repeatability, while once the ice was “conditioned” by bedding the track in for 2-3 hours, measurements were very repeatable (Conant et al. (1949), Venkatesh (1975)), even over several weeks (Wilkinson (1953)). Measurements taken after several months showed a general drop in friction, attributed to the concentration of ionic impurities at the surface due to sublimation of pure ice and adsorption of contaminants from the environment (Roberts (1981a)). This was verified by chemical analysis of the surface layer, showing that the sodium and chloride concentrations had risen by 6000 % and 2000 % respectively over the 10 months (Roberts (1981b)). As well as the drop in friction measurements, the trend of the $\mu$-$T$ curve had changed, showing similar characteristics to the findings of Roberts and Lane (Roberts and Lane (1983)) when artificially adding salt to the water.
2.2.4 Liquid-like layer

Observations made on warm ice ($T > -15 \, ^\circ C$) shows that it possesses a liquid-like layer of water at its surface (Golecki and Jaccard (1978), Roberts (1981a)). Experiments have shown that friction on ice in this temperature range is much lower than friction on lubricated solids (Roberts (1980), (1981a)). An explanation of this phenomenon is that the liquid-like layer exists due to the discontinuity of the bulk ice as it meets the interface causing an imbalance of molecular forces across the interface (Makkonen (1997)). This provides a much better boundary lubrication system than artificially lubricated ones (such as on water-lubricated glass), as the liquid-like layer remains bonded to the underlying ice substrate, but can be "squeezed away" under the normal pressure when no appreciable bond exists (Roberts (1981a)).

The thickness of this liquid-like layer when observed optically was several hundred $\mu$m at temperatures near to $0 \, ^\circ C$ and diminished at about $-4 \, ^\circ C$ (no dimensions at lower temperatures are reported), and can exist down to temperatures of $-20 \, ^\circ C$, and is dependent on atmospheric conditions and water purity (Ahagon et al. (1988)). The viscosity of the layer was found to be about ten times higher at $-1 \, ^\circ C$ than that of pure water (at $0 \, ^\circ C$), and to be twenty times higher at $-2 \, ^\circ C$. Experiments by Barer and others (Barer et al. (1980)) showed that Newton's law of viscosity could describe ice flow, and Ahagon and co-workers have concluded that drag flow of the liquid-like layer is the major source of friction on ice (Ahagon et al. (1988)).

Direct experimental evidence for the liquid-like layer on ice crystals has been provided by optical (Golecki and Jaccard (1978)) and mechanical (Valeri and Mantovani (1978)) means. The mechanical method identifies an appreciable liquid-like layer down to $-15 \, ^\circ C$, while optically it is detectible down to $-35 \, ^\circ C$. Both research groups note the steep viscosity rise with decreasing temperature close to and below the melting point, echoing the observations made by Ahagon et al. while measuring frictional properties.

According to the theory of Makkonen (Makkonen (1997)), it is postulated that at specific temperatures the liquid-like layer can be present on ice when in contact with one material, yet disappear when another material is in contact. This is based
on the fact that the surface energy of ice can vary considerably depending on the counterfacing material, and could have significant influence on static friction (breakaway) measurements.
2.3 Rubber properties
As shown previously, the ice properties can be very significant in determining the friction available, especially near to its melting point. However, when sufficiently cold, or at speeds low enough to avoid frictional melting of the surface ice behaves like any other solid material. In this situation, it can be the rubber properties that determine the levels of friction measured.

2.3.1 The behaviour of typical viscoelastic materials
The viscoelastic nature of rubber is what gives it, in general, such good frictional properties. Its ability to deform and take the shape of the counterfacing material allows better intimate contact with it and hence adhesion is increased by an enlarged real area of contact. Characterising the viscoelastic properties of polymers such as rubber is generally done using dynamic mechanical analysis (DMA).

![Diagram](image.png)

Figure 2.3. Force and response curves for a viscoelastic material with time. Definitions of applied stress ($\sigma$), measured strain ($\varepsilon$) and phase angle ($\delta$) indicated.

The method is carried out by applying a sinusoidal force to the sample and observing the corresponding response amplitude, which allows the complex modulus ($E^*$) of the material to be calculated (see Figure 2.3). Measuring the
phase shift between the applied load and response displacement (hysteresis) indicates the damping properties of the material. From this information, the storage and loss moduli ($E'$ and $E''$ respectively) can be resolved by plotting an Argand diagram, as shown in Figure 2.4.

$$E^* = E' + iE''$$

where $E' = \frac{\sigma_0}{\varepsilon_0} \cos \delta$

and $E'' = \frac{\sigma_0}{\varepsilon_0} \sin \delta$

**Figure 2.4. Argand diagram showing the relationship between complex ($E^*$), storage ($E'$) and loss ($E''$) moduli and phase angle, $\delta$.**

By varying the temperatures used for DMA over a sufficient range, it is possible to observe the glass transition temperature ($T_g$) of the material. $T_g$ is the point at which the material's viscoelastic properties change from glassy (predominantly elastic) to a rubbery (predominantly viscous), and is located at the temperature point at which $\tan \delta$ is at a maximum. At low temperatures, viscoelastic materials display glass-like properties and towards higher temperatures display rubber-like properties. This is due to increased molecular mobility within the material as the temperature is raised.

The loss factor ($\tan \delta$) is a characteristic used to describe the materials' damping properties, as it is the ratio of energy lost to energy stored (see Equation 4) and so describes the materials' ability to dissipate energy.

$$\tan \delta = \frac{E''}{E'}$$

(4)

Friction is generated by dissipative processes, so it can be said that the loss factor, $\tan \delta$, and hence $T_g$ is an important property when studying the friction of materials. Typical plots obtained from DMA are shown in Figure 2.5. $T_g$ is
obtained from the temperature corresponding to the maximum value of \( \tan \delta \) (\( \approx 248 \text{ K} \) or \(-24 \text{ °C} \) for the "5.4 phr" sample).

![Typical plot obtained using DMA showing storage modulus \( (E') \), loss modulus \( (E'') \) and loss factor \( (\tan \delta) \) versus temperature of four styrene-butadiene samples containing different levels of sulphur (Salgueiro et al. (2004)).](image)

Figure 2.5. A typical plot obtained using DMA showing storage modulus \( (E') \), loss modulus \( (E'') \) and loss factor \( (\tan \delta) \) versus temperature of four styrene-butadiene samples containing different levels of sulphur (Salgueiro et al. (2004)).

From time-temperature superposition principle for viscoelastic materials, it can be shown that the behaviour of viscoelastic materials in temperature space can be transposed onto time space (Berthoud and Baumberger (1998), Huemer et al. (2001)). This allows frequencies to be extrapolated for which the loss modulus and factor are at a maximum. This is applicable when studying the behaviour of materials at different frequencies (and speeds), a more common investigation than temperature range studies.

The mechanism controlling friction in rubbery materials has been defined by Grosch (Grosch (1963)) as having an adhesion component and a deformation component. Both components are viscoelastic in nature, but the adhesion
component is closely linked to the position of the materials maximum loss factor at
the temperature axis and the deformation component relates to the maximum loss
modulus.

The adhesion component is present in all friction experiments, and is exemplified
by measurements on highly polished surfaces. The deformation component only
becomes significant on surfaces with a characteristic roughness, such as those
described by Grosch on silicon carbide tape (Grosch (1963)). It was found that if
the characteristic roughness frequency corresponds to the frequency \( f \) at which
the loss modulus is a maximum (i.e. \( vf = \) characteristic asperity spacing), then
maximum friction can be attained. For smooth, solid surfaces (or surfaces with no
characteristic roughness), the maximum friction is generally attained at a
temperature that corresponds to the \( T_g \) of that material. This behaviour can be
demonstrated by using mastercurves.

The friction characteristics of rubber materials have been commonly presented on
mastercurves. This is a plot of the friction coefficient with respect to speed at
different temperatures, and transformed using the Williams-Landel-Ferry (WLF)
transform (Williams et al. (1955)) (see Equation 5).

\[
\log_{10} a_T = \frac{-8.86 \cdot (T - T_0)}{101.6 + (T - T_0)}
\]

where \( \log_{10} a_T \) is the horizontal shift of the data along the velocity axis, \( T \) is the
temperature at which the measurements have been carried out and \( T_0 \) is the
reference temperature (usually \( T_g + 50 \)). An example of a conventional plot of the
data and the transformed mastercurve for styrene butadiene rubber on “dry” ice are
shown in Figure 2.6 overleaf.
This sort of plot shows rubber friction as fitting a curve that describes its behaviour throughout a range of temperatures and speeds. The conventional plot shows the frictional behaviour with temperature and speed, but only at the specific parameters chosen by the experimenter. This is an ideal method used to compare different rubbers that does not require a huge amount of data, but describes the behaviour in a clear fashion. The one drawback is the need to transform the data mathematically, so the data shown on the mastercurve is not "real" data, but transformed, "virtual" data. The peak shown on the mastercurve corresponds to the maximum friction that would be obtained at $T_g$. The existence of a mastercurve is generally taken as evidence that the process that produces it depends on the viscoelastic properties of the rubber (Southern and Walker (1972)).

As well as the glass transition temperature of the rubber influencing the frictional properties of the material, several other properties have been noted as affecting the friction produced on the counterfacing substrates.
2.3.2 Hardness

The hardness of rubber is a matter of great debate between many researchers. Some claim that a harder rubber can produce higher friction on soft ice due to the ability of its asperities to penetrate the lubricating layer to make intimate contact with the underlying ice (Roberts and Lane (1983)). Others say that for the rubber to have high friction, it must be sufficiently soft to follow the ice topography and hence maximise the interfacial contact area, or reduce the lubricating layer to a minimum thickness (Ahagon et al. (1988), Conant et al. (1949), Pfalzner (1950), Venkatesh (1975), Wilkinson (1953)). Both of these theories are perfectly acceptable, but they may only be true within confined envelopes of temperature-speed-pressure parameters. In general, the softer the rubber, the higher the friction although this is not always the case, and it is suggested that an optimum hardness may be required for best friction over a range of temperatures (Roberts and Lane (1983)).

For example, assuming a thin lubricating water layer is present between a rubber slider and an ice surface, if the rubber is hard and dwells on the ice surface, the adhesion would be less than that of a softer material due to deformation of the softer material and the squeeze effects of normal pressure removing water from the interface (Persson (2001)). The mechanical macro-keying effect may become more significant for the harder rubber, since it will expand less in the lateral directions and the overall pressure will be higher, so the harder material may possess a higher friction coefficient. Taking the same materials at a higher sliding speed, the harder rubber would be likely to have less friction than the softer, due to the inability for the harder material to deform over asperities or reduce the thickness of the lubricating layer, and hence the softer material should provide a higher friction coefficient. These are processes that can depend on the apparent temperature, speed and pressure, as well the inherent rubber properties.

The difficulty in comparing the hardness is that in order to maintain the general properties of the rubbers (i.e. $T_g$, resilience, base polymer, etc.), but change the hardness for comparative tests, fillers are added, which can change the way the materials interact and behave (in general, rubbers with 30 % carbon black filler added show $\approx \frac{1}{3}$ of the friction of pure, unfilled gum rubbers (Gnörich and Grosch (1972), Grosch (1963), Southern (1974))). Comparative studies of this type
(Gnörich and Grosch (1972), Southern (1974)) concluded that the harder rubbers did have lower coefficients of friction, but they also had the higher levels of carbon black added. Whether this is a convoluted system is not clear, but plots published by Conant et al. (see Figure 2.7) and Pfalzner (Pfalzner (1950)) shows that although a trend does exist, “hardness is not decisive in every case” (Pfalzner (1950)) and the spread of data does suggest that there may be other factors of more importance.

![Figure 2.7. The coefficient of friction versus hardness for different rubbers at -7 °C (Conant et al. (1949)).](image)

2.3.3 Resilience

As with hardness, there seems to be some confliction of conclusions regarding resilience (the ability of the material to use stored energy to retake its shape after a deforming load has been removed). Ahagon et al. (Ahagon et al. (1988)), Roberts (Roberts (1981a), (1981b)) and Wilkinson (Wilkinson (1953)) concluded that increasing the resilience of the rubber made further improvements to the friction measured on icy surfaces, whereas Pfalzner (Pfalzner (1950)) notes that low resilience appears to be favourable to high friction.

An overview of some of the basic theories of friction would suggest that resilience (the ability to recover energy after deformation) would be a good property to possess when trying to remain in intimate contact with the counterfacing substrate.
On the other hand, as friction is a dissipative process, higher resilience would suggest less dissipation, hence less friction. This is shown with reference to DMA (Figure 2.5), that the higher the resilience (indicated by the magnitude of $E'$) the lower $\tan \delta$, hence lower friction. Both of the observations may be correct within particular conditional envelopes, but this is obviously another parameter that requires further study to determine its importance in rubber-ice friction.

### 2.3.4 Base polymer type

The only reliable way to find out if a particular rubber compound provides good traction is to try it in the field. In practice, this is very expensive and hence the drive to test rubber compounds in the laboratory for comparative and verification studies. Variability between the same tyres on different days, and between tyres from the same batch adds to uncertainties in measurements (Williams and Troulis (2002)).

For these reasons, comparative studies between tyres of different base rubber composition and compounds have been common (Ahagon *et al.* (1988), Conant *et al.* (1949), Grosch *et al.* (1970), Niven (1958), Roberts and Lane (1983), Shimizu *et al.* (1992), Southern (1974)) due to the continuing need for tyre manufacturers to improve on performance and gain the competitive edge. Although outright performance is never the goal of these companies, being constrained with costs and wear rates, economy, etc. (Persson (2001)), some of the results are useful for ascertaining the dependency on compound type.

Whilst experiments on icy surfaces uncover differences of only 10 to 20 % between rubbers of different types, this is of great significance to vehicle handling (Roberts and Richardson (1981)). It is generally regarded that it is the ice surface that is the critical factor in governing the friction, but as this is the one parameter that cannot practically be modified, it is the rubber type and properties that need to be optimised.
2.4 Rubber-ice friction test methods

While it could be assumed that tests using the same materials and prevalent conditions would show the same trends and magnitudes of friction, it is found that different test methods and procedures can cause a wide variation in the friction measured between rubber and ice. How these factors can affect the magnitude of friction measurements is described in the following section.

2.4.1 Preparation of ice

Sample surface finish

The roughness of the ice has been noted to have a significant influence on the levels of friction measured. Buffing (polishing with fine emery paper) increased friction, the explanation given by Niven as allowing closer contact between the rubber and ice surfaces, increasing the real area of contact and thereby producing increased adhesion (Niven (1958)). This opinion is also stated by Roberts (Roberts (1981a)), who notes that on polished ice friction is higher for significantly higher speeds than on rough ice. Observations made by Shimizu et al. using tyres on ice of different roughness noted that smooth ice could produce friction coefficients of around 50% higher than those measured on rough ice (Shimizu et al. (1992)).

Roughness can also be caused by the presence of particles bonded to the ice surface. These can take the form of ice particles present from preparing the surface or debris from previous friction experiments, and (especially at low temperatures) frost. Niven notes that a decrease in friction could be seen caused by loose shavings of ice, which had not been removed thoroughly from the ice surface (Niven (1958)). This is echoed by Roberts (Roberts (1981b)) and Gnörich and Grosch (Gnörich and Grosch (1972)) who write that frost and ice crystals dramatically reduce friction levels.

For real life applications of this tribosystem (especially with regard to tyres on icy roads), it is paramount to say that clean, smooth ice surfaces do not occur on natural road surfaces (Gnörich and Grosch (1972)), but for a laboratory study, it is essential to control the roughness of the ice (and the rubber) surface to ensure reproducibility.
Conditioning
Several authors (Conant et al. (1949), Roberts and Richardson (1981), Southern and Walker (1972), (1974), Venkatesh (1975)) note of the need to condition ice samples before their use in friction experiments. This is generally carried out by polishing the ice using a rubber pad (Roberts and Richardson (1981), Southern and Walker (1972), (1974)) or by taking a set number of readings using the rubber sample until the measured friction has stabilised (Conant et al. (1949), Venkatesh (1975)). On fresh ice, friction measurements are variable (Conant et al. (1949), Southern and Walker (1972), (1974)), but once conditioned, reproducibility on ice has been noted to be good even over several weeks (Roberts and Richardson (1981), Southern and Walker (1974), Wilkinson (1953)).

Experiments carried out by Southern and Walker (Southern and Walker (1972)) to determine mastercurves (see Section 2.3.1) of several different rubber compounds found that while mastercurves were readily found for rubbers on conditioned ice, when using fresh ice the mastercurve was not present. They concluded that it was therefore not the rubbers’ viscoelastic properties that were deemed responsible for the observed behaviour on fresh ice, but a property of the ice itself.

A subsequent study (Southern and Walker (1974)) suggested that this irreproducibility was due to a structural weakness in fresh ice (causing ease of shear) that is removed by conditioning. It was found that re-cutting the ice did not remove the effect of conditioning, hence some tentative evidence seems to suggest that it is indeed a structural phenomenon.

Under the dry friction regime, conditioning of the ice is essential in obtaining reproducible results, but in the lubricated regime the difference in conditioned and non-conditioned ice is negligible, indicating that significant melting is removing the effects of conditioning (Southern and Walker (1974)).

Wilkinson (Wilkinson (1953)) notes that there is a degree of variation between identical ice samples made at different times that does not exist within tests carried out with the same ice sample. A survey carried out on the different ice structures led him to believe that this discrepancy could be due to these different “types” of
literature review

ice. A detailed explanation of the procedure used to make the ice samples is given, but it does not include any conditioning process before measurements were taken.

2.4.2 Preparation of rubber

Sample surface finish
The effect of cleaning the sample prior to testing on ice was studied by Wilkinson (Wilkinson (1953)). The result of this test was that the sample that had been cleaned by buffing with "Crocus cloth" (a fine grade of emery paper) produced a measured friction of approximately 10 – 20 % higher than that which had been used directly from the mould without this cleaning process. No description of the moulding process itself is provided, so some regard should be given to whether this effect is true to this system or whether it may be an effect due to contaminants such as release agents, oils, etc.

Conversely, with other materials, (1018 carbon steel) Calabrese et al. (Calabrese et al. (1980)) notes a slight increase in friction coefficient with increasing roughness. Forland and Tateno and Tabata and Tusima also observed increases, as a linear function with the roughness root mean square of the counterfacing material (stainless steel) (Forland and Tateno (1984), (1985)). These observations support the importance of the frictional ploughing contribution, and may be of importance to the rubber surface at high temperatures as the ice becomes soft relative to the rubber (and vice versa).

Sample shape
Previous study of the effect of sample shape showed that no significant dependencies were found on the shape of flat rubber pads on ice (Niven (1955), Venkatesh (1975)). Comparative experiments of flat pads, rounded pads (Conant et al. (1949), Wilkinson (1953)) and spherical rubber samples (Roberts and Richardson (1981)) were made and also noted little difference in the resultant friction values. Wilkinson did note, however, that when extremely thin (0.05 mm thick) samples of rubber were used, in both flat geometry and placed on the surface of a spherical mould, the measured friction was much less than that using a 2.5 mm thick sample. This was judged to be due to the inability of the rubber to maintain
intimate contact with the ice surface. Barquins and Roberts (Barquins and Roberts (1986)) also noted that when using rubber spheres of varying radius of curvature, the samples with greatest radius had greatest measured friction. This observation was regarded as evidence that the real area of contact between the sliders was of primary importance when measuring rubber friction on smooth surfaces.

With note to this observation, a comment made by Blaisdell and Borland (Blaisdell and Borland (1990)) regarding scaled models is that, the model dimension must be at least ten times any characteristic roughness so that scale problems are not an issue. This is of significant importance when directly comparing laboratory studies to real life tyre friction situations.

The shape of the samples may also be a useful tool in differentiating between the component of friction that can be attributed to edge effects and those derived from the interface. Samples with flat faces and sharp edges will undoubtedly have a significant proportion of friction attributed to the edge effect component, whereas this component using spheroidal samples will have little effect. This is particularly useful when comparing laboratory tests to tyre measurements: the tread patterns used in winter tyres are aggressively formed to make the most of the advantageous edge effects to mechanically key to the ice and snow surface.

2.4.3 Type of experimental setup
Experiments were carried out by Forland and Tatinclaux (Forland and Tatinclaux (1985)) to ascertain the dependence of the measured coefficient of friction on sample layout. Their equipment allowed friction measurements to be made using rubber-on-ice and ice-on-rubber geometries, and it was found that there were no appreciable differences between the magnitudes of friction measured using either geometry. This factor that is of critical importance when setting up protocols for our measurements, as both geometries are used in this study: steady state measurements use the ice-on-rubber geometry (see Chapter 4), while breakaway measurements (Chapter 5) are carried out using the rubber-on-ice layout.
2.5 Rubber-ice friction mechanisms and measurements

In the preceding sections, the effect on friction of several experimental parameters and material properties were shown for ice friction and rubber friction independently. This is essential before studying the effect of these parameters on the rubber-ice tribosystem as the characteristic behaviour of the materials can become convoluted under specific conditions and therefore any dependencies on a more important mechanism can become difficult to establish. This section shows how the observations reported for each material in previous sections affect friction when the two are combined.

2.5.1 The effect of temperature

The frictional resistance in materials sliding against ice is predominantly based on the surface temperature of the ice, as the higher the temperature, the larger the amount of melt water that can be produced. This can be determined by several factors, but the most palpable is that of environmental temperature. When the surface is dry, the resultant coefficient of friction is comparable to any other solid material, and in the case of rubber, can quite commonly be in excess of unity. As soon as a lubricating layer is introduced, this can fall to less than 0.03 (Bowden and Tabor (1950)).

Frictional resistance is developed from a source of energy dissipation (Ahagon et al. (1988)). With a system of rubber sliding on a hard substrate, wet or dry, it is the rubber which is the dominant source of energy dissipation, a theory which has been developed well by Grosch (Grosch (1963)). Grosch also used this theory to explain the frictional behaviour with ice as the substrate. In this study however, the temperatures were sufficiently low to avoid frictional melting. However, from measurements carried out by Ahagon et al. in a melting regime, it was concluded that “the energy loss process necessary for the frictional resistance takes place primarily in the fluid layer, and not in the rubber” (AhAGON et al. (1988)). From this statement, he goes on to say that the main requisite of the rubber is that it follows the topography of the ice surface as closely as possible, enabling shearing of the ice surface to be maximised, i.e. the rubber should be soft and resilient.
There are several authors (Ahagon et al. (1988), Conant et al. (1949), Gnörich and Grosch (1972), Niven (1955), (1958), Roberts (1980), (1981a), (1981b), (1981c), Roberts and Lane (1983), Roberts and Richardson (1981), Shimizu et al. (1992), Southern (1974), Southern and Walker (1972), (1974), Venkatesh (1975), Wilkinson (1953)) who have published data for rubber-ice systems. Mostly the authors have looked at rubber properties on ice, and therefore have attempted to avoid surface melting of the ice by using very low velocities and/or temperatures. Very little work has been reported for friction at high speeds ($v > 10$ ms$^{-1}$) on ice (Kozlov et al. (1994)).

For one group of researchers (based at the Natural Rubber Producers' Research Association, now the Tun Abdul Razak Research Centre (T.A.R.R.C. (2005))), measurements have mostly been based around obtaining mastercurves by the WLF transform for different rubbers on a "dry" ice surface. In doing so, we are provided with clear demonstrations that ice behaves just like any other solid surface. The equipment used by Gnörich and Grosch takes especially lengthy precautions as to maintain a frost-free, highly polished surface, by continually cleaning and polishing the ice, even during experiments. The result of this procedure is the high coefficients of friction measured ($\mu$ is in excess of 3.5 at $-10$ °C) (Gnörich and Grosch (1972)). The measurements fit well a mastercurve, except for measurements above $-5$ °C, where the ice properties mask the effect of the rubber compounds (i.e. even the precautions taken cannot prevent melt water at such elevated temperatures). The most interesting observation when avoiding melting of the ice is the behaviour of the different rubber compounds over the range of temperatures. Over exactly the same temperature range, the frictional behaviour of four different rubbers is shown. This behaviour is linked to the glass transition temperature of the different compounds (see Figure 2.8).
As shown previously, the glass transition temperature determines quite clearly the temperature at which the maximum friction is produced for a particular rubber compound. In this case, the ranking of $T_g$ from lowest to highest temperatures is as follows; BR, NR, SBR, NBR (the actual $T_g$ are not specified). Figure 2.9 corresponds to the data in Figure 2.8, but shows how the rubbers behave on another solid surface (glass). This goes some way in showing that mastercurves are also applicable on ice, with the difference being that friction is somewhat lost close to the melting point of the ice, where all four rubber compounds coefficient of friction drop significantly to a similar value (Ahagon et al. (1988)). This drop is due to the ice properties becoming dominant, negating the effect of the rubber compounds. The same effect has been shown in measurements by Southern and Walker (Southern and Walker (1972), (1974)).
The other significant author at the same institution is Alan Roberts. His work with regard to friction on ice is mostly based around optical observations of rubber hemispheres sliding on polished ice surfaces. In general, the measurements were carried out at low speed (<0.01 m/s), but observations have been made at higher sliding speeds (0.5 m/s) at −8 °C. At low speed, Schallamach waves of detachment were observed (an indication of solid contact (Roberts and Jackson (1975), Schallamach (1971))), but upon increasing the speed, the waves disappeared and melt water was clearly visible. In another publication (Roberts (1981a)) it is inferred that melt water sufficient to lower the friction significantly is produced at 0.1 m/s and −30 °C. This observation is significant, as it infers that melt water is readily produced at very low temperatures and fairly low speed. Previous findings may have to be re-addressed as to the (negligible) significance of melt water on rubber frictional behaviour.

As the temperature of the ice increases it also becomes softer (Barnes et al. (1971)). This enables the counterfacing slider to sink into the ice surface if it remains normally loaded on the ice. Sinking into the ice adds a mechanical

* Schallamach waves are produced in rubbery materials during sliding on hard substrates as a stress relief mechanism. Waves are caused by buckling of the rubber at the leading edge of the contact area due to excessive compressive stresses, and travel along the contact area (front to back) during sliding. This phenomenon has been intensively studied since their first discovery in 1971 by Schallamach (Schallamach (1971), Barquins and Roberts (1986)).
component to the friction by introducing a macroscopic edge effect, or mechanical keying, the extent of which depends on the relative hardness of the two materials (Petrenko and Whitworth (1999), Tabor (1987)). Observations indicate that for the rubber on ice tribosystem, the keying element becomes significant at temperatures over −10 °C (Roberts (1981b)), where the flow properties of the ice seem to change (Barnes et al. (1971)). Below −20 °C the creep rate of ice is greatly reduced, leading to the insignificant deformations observed, even after hours at relatively high normal load (Roberts (1981b)), so it can be assumed that below this temperature the macro keying component is negligible.

From a practical point of view it is difficult to maintain and control temperatures close to the melting point of the ice with regard to temperature dependence. The sharp decrease in friction close to the melting point shown by Bowden etc. applies also to friction with rubber. Close to the melting temperature the system is quite unstable due to the sharp phase change and this affects greatly the frictional behaviour.

2.5.2 The effect of speed
The coefficient of friction for most materials is usually independent of sliding speed (indeed Coulomb's Law states, "Kinetic friction is independent of the sliding velocity"), but for both ice and rubber friction is very speed dependent, but for different reasons.

Ice behaves like any other crystalline solid at low temperature, but as the temperature increases closer to its melting point (as is the case in this study) it melts by frictional heating causing a lubricating layer between itself and the counterfacing slider (Bowden and Hughes (1939)). For a fixed temperature, the effect of frictional heating increases with sliding speed so that the amount of melt water produced increases, lowering the coefficient of friction.

A model (designated the thermal control model) proposed by Evans et al. (Evans et al. (1976)) and developed by Oksanen and Keinonen (Oksanen and Keinonen (1982)) represents the friction behaviour of ice dependent on sliding speed.
The models are based on the assumption that friction is due solely to the thermal balance of the system's interfacial layer leading to viscous drag and the lubricating layer at the interface is fixed at 0 °C due to the latent heat of melting. Experimental studies have verified partially (Calabrese et al. (1980), Forland and Tatinclaux (1985)) and to a greater extent (Akkok et al. (1987), Southern and Walker (1974)) this model. Experimental data from Southern and Walker are shown in Figure 2.10 below.

\[ \mu \propto \frac{1}{\sqrt{v}} \quad (6) \]

For very low sliding speeds \((v < 10^{-2} \text{ ms}^{-1})\), an alternative model seems to describe the speed dependency well.

\[ \mu \propto \sqrt{v} \quad (7) \]

This is actually the inverse of the previous model, suggesting a "limiting envelope" exists in which Equation 6 is valid. This suggestion has been backed up experimentally by Barnes et al. (Barnes et al. (1971)) and Tusima (Tusima (1978)).
showing that the friction actually increases with speed up to a point determined by material properties and test conditions. This finding has been speculatively explained by adhesion and plastic deformation of the ice, possible at these very low speeds.

Southern and Walker (Southern and Walker (1974)) carried out limited measurements up to \( \approx 10 \text{ ms}^{-1} \) producing mastercurves to compare the three rubbers used, and Venkatesh (Venkatesh (1975)) up to 4 ms\(^{-1} \), again comparing several rubbers. Somewhat conflicting findings are the result, but as the method used for producing the mastercurves as done by Southern and Walker is effectively extrapolation, and not real measurements, caution must be taken comparing these results. The results reported by Venkatesh show some difference in the different rubber compounds, but very similar trends throughout the speed range. This signifies an ice-dominant system with the very slight differences in measurements due the specific rubber properties.

![Figure 2.11. Speed dependence of the friction of natural rubber on "pure" (solid lines) and salted (0.1 M NaCl, broken lines) ice for a range of sliding speeds from 10\(^{-5}\) to 10\(^{-2}\) ms\(^{-1}\) (Roberts and Lane (1983)).](image)

One particular observation of note is that of Roberts et al. (Roberts and Lane (1983), Roberts and Richardson (1981)) shown in Figure 2.11. When looking at rubbers on pure ice, it was found that Schallamach waves are generated if the sliding speed is sufficiently high. This is explained by the presence of the viscous
liquid-like layer producing high shear stress at high speeds, and low shear stress at low speeds. Generation of Schallamach waves requires a certain shear stress to be exceeded, so rapid sliding may lead to waves (Roberts and Lane (1983)).

2.5.3 The effect of pressure

One of the earliest theories on why ice is characteristically a low friction material is that of Reynolds (Reynolds (1900)) who suggests that the slipperiness may be due to the act of melting under pressure. He was sceptical about this theory, however, as it was suggested that ice would not be able to support the 10,000 lbf/in\(^2\) (\(\approx 69\) MPa) necessary to cause such melting. This is something that was quickly clarified, showing that indeed the yield strength of ice is much less than the pressure required to reduce its melting point enough to produce melt water for all but the highest temperatures (Bowden and Hughes (1939)).

According to classical theory, the coefficient of friction is independent of the normal force, and hence the contact area, but it has been shown that this is not the case for rubber on a smooth, solid surface (Schallamach (1952)). There have been several empirical models proposed for the relationship between the coefficient of friction and the normally applied load to rubber on hard substrates (Schallamach (1952), Thirion (1946), Venkatesh (1975), Wilkinson (1953)). The most widely accepted is that of Schallamach (Maeda et al. (2005)) which states that:

\[
\mu \propto \frac{1}{\sqrt[4]{p}}
\]

where \(p\) is the normal pressure applied to the surface by sliding body. This is in agreement with classical Hertzian elasticity theory (Barquins and Roberts (1986), Shigley (1986)), and basically states that the friction is directly related to the real area of contact between the samples.

Schallamach also makes reference to an earlier work by Thirion (Thirion (1946)), who found experimentally that friction was related to normal pressure by another similar model:
\[ \frac{1}{\mu} = a + bp \]  

where \( a \) and \( b \) are positive constants. It was noted, however, that the range of pressures used by Thirion was much higher than those selected by Schallamach. In his theory, Schallamach based his assumptions on Hertzian contact, which only holds for small deformations (and hence loads). The different model suggested by Thirion may hold for highly loaded systems, but with the decaying nature of the coefficient of friction with increased load, and the tendency to be independent of normal load with a sufficiently high load, the data can be claimed to be complementary within the experimental error of the respective systems.

### 2.5.4 Other parameters

Further studies of this system have shown dependencies on other physical phenomena such as electrical charge (Petrenko and Whitworth (1999), Roberts (1977), Schnurmann (1940)), prevalent environmental conditions such as humidity (Calabrese et al. (1980)) and dependencies of friction on the structure of ice, e.g. crystal orientation and grain size (Colbeck (1992), Forland and Tatinclaux (1985), Norheim et al. (2001), Shimizu et al. (1992)). These are interesting areas of further study, but fall outside of the scope of this project.
Chapter 3.

Tribometer design

Before the design process began, a review of previous tribometer designs was carried out. This involved both designs by commercial instrument companies and those by research groups, custom-built for their particular application. This gave a good indication of the features required for research in this area and of the general challenges encountered and how to overcome these successfully. In carrying out the review, it became apparent the lack of commercial instruments available for friction work on ice (the most design-restraining material) and hence the need for researchers to develop their own.

A design was developed having determined the criteria for the tribometer through this investigation and by the proposed test protocols. Budget and laboratory space were the constraints on the design, but the ultimate target was to produce an instrument that performed satisfactorily as proposed within these constraints. This meant modifications to the original design, tackling problems not seen in advance, and finding solutions to attain the criteria set before design.

Finally, the tribometer was tested on a well-known system (namely stainless steel on Teflon®) in order to verify that the instrument was operating satisfactorily and producing data of the high standard required. Once this had been attained, the instrument could be used reliably in the intended environment, i.e. in a freezer down to −30 °C.
3.1 **Review of previous tribometer designs**

The first step of almost all design projects start with a survey of products similar to that proposed. Within the field of instruments for scientific research there are predominantly two routes taken by researchers: one, to buy commercially available products and two, to design and build instruments to their own specification.

Commercial instruments are in general designed to be as versatile as possible. This allows a simple solution to many research projects with short setup times and efficient operation with user-friendly, proprietary software and helpful technical support. The main drawback with this approach is not providing the specialised capability required for detailed and original research work. Modifications can be made to these instruments to suit the specific conditions, but this is an expensive and time-consuming process.

Taking the route of designing and building their own instruments allow the researchers to tailor the tribometer to their individual needs, with user-specified measurement equipment and control methods. There are obviously drawbacks to this method, namely the long lead-time with the design process and solving the inevitable teething problems, but the benefits gained can be extremely beneficial to the quality of research.

### 3.1.1 Commercial tribometers

The following section provides an analysis of the more popular tribometers designed for scientific research and quality control of novel, as well as commercially produced materials and surfaces. A tabulated critique of these tribometers is included in Table 3-I overleaf.
<table>
<thead>
<tr>
<th>Tribometer and Manufacturer</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>CSM Tribometer (CSM Instruments SA (2003)) CSM Instruments SA, Switzerland.</td>
<td>General, versatile tester can be used in many configurations. Force measurement lever is designed as a frictionless force transducer. Can be used in linear reciprocating mode. Can calculate wear coefficients through track depth monitoring. Can be used in a controlled humidity/composition environment.</td>
<td>Forces deduced by measured deflections of arm – not direct force measurement. Maximum speed is 5.7 ms⁻¹ (12.8 mph). 100 Hz sampling rate. Proprietary software limited to standard regimes, specific specialised tests not possible without modification. Does not allow testing of cold samples.</td>
</tr>
<tr>
<td>CETR Micro-Tribometer (Centre for Tribology Inc. (2002)) Center for Tribology, Inc., CA, USA.</td>
<td>Claim to be the most advanced tribometers of their kind. High-end versatile materials tester. 6-D load and torque cells – 1 μN to 1 kN force range. 100 kHz data rate, 16-bit resolution. Spiral wear track. Can be used in linear reciprocating mode. Can calculate wear coefficients through track depth monitoring. Can be modified to perform a large array of materials tests. Built-in materials analysis.</td>
<td>Proprietary software limited to standard regimes, specific specialised tests not possible without modification. Does not allow testing of cold samples.</td>
</tr>
<tr>
<td>Micro Photonics Standard Tribometer (Micro Photonics Inc. (2006)) Micro Photonics Inc., PA, USA.</td>
<td>Fairly compact size (400 × 700 × 500 mm). Spiral wear track. Can be used in linear reciprocating mode. Can calculate wear coefficients through track depth monitoring. Electrical conductivity/resistance measurements. Can be used in a controlled humidity/composition environment.</td>
<td>10 mN friction force resolution. Maximum speed is 4.1 ms⁻¹ (9.2 mph). Proprietary software limited to standard regimes, specific specialised tests not possible without modification. Does not allow testing of cold samples.</td>
</tr>
<tr>
<td>CSM Nano Tribometer (CSM Instruments SA (2003)) CSM Instruments SA, Switzerland.</td>
<td>High force resolution (1 μN). Can be used in linear reciprocating mode. Specific software provided for parameter control and data acquisition. Can be used in a controlled humidity/composition environment. Low-temperature versions available with integrated sample cooling plate (no temperature ranges quoted).</td>
<td>Restricted force measurement of 10 μN to 1 N. 50 μN to 1 N normal load range requires small samples – may be too sensitive to surface finishes. Maximum track diameter is 20 mm. Maximum speed is 0.05 ms⁻¹ (0.11 mph). Proprietary software limited to standard regimes, specific specialised tests not possible without modification.</td>
</tr>
</tbody>
</table>

Table 3-I. Summary of commercially-available pin-on-disc tribometers.
**CSM Standard Tribometer** *(CSM Instruments SA, Switzerland.)*

The CSM Standard Tribometer (see Figure 3.1) is a typical pin-on-disc tribometer that has found popularity within a wide range of industries. Its modular design lends itself well to the versatile nature of commercial tribometers, allowing a linear reciprocating module as well as the standard pin-on-disc configuration. As well as the standard test types and controllable parameters, additional features that can be utilised include wear depth monitoring and electrical contact measurements.

![Figure 3.1. The CSM standard tribometer (CSM Instruments SA (2003)). Note the modular design of the device with interchangeable disc attachment, adjustable disc radius and load applied by adding masses above the pin sample. On the left, the device is shown with the environmental hood attached.](image)

The useful features of this tribometer are based on its versatile design, with reciprocation and track-depth monitoring possible. The instrument can also be used within its own environmental chamber and a high temperature module can be added. The load cell can be used on many sample sizes and shapes, although the design of this item could be criticised: the calibration of the measurement arm is dependent on the homogeneity of the arm’s deflection under different conditions. At elevated temperatures, for example, the non-linearity of the arm will be emphasised as higher deflections will occur, and measurement of this deflection is critical to the accuracy of the tribometer. The distance between load cell and contact point is significant, and this can only be detrimental to measurements
through unmeasured deflection of the pin.

The speed range of the instrument, while being the highest of the commercial tribometers described here at 5.7 ms\(^{-1}\), does not meet specifications necessary for comparisons to automotive tyres*. CSM state that although the instrument can be accommodated in a cold climate, there is no guarantee that the device will operate in a calibrated manner, as the instrument was not designed to do so.

The software, while user-friendly and powerful in its operation is limited to standard wear and friction tests. Modification to this, while possible is not user-programmable and will need to be done by the company’s engineers.

**CETR Micro-Tribometer (Center for Tribology, Inc., CA, USA.)**

This high-end versatile materials tester (see Figure 3.2) is based on a standard platform, but can be modified to perform a large array of materials tests including tribometry (pin-on-disc, four-ball, disc-on-disc, etc.), scratch tests, wear tests, and indentation tests. This versatility is augmented by the ease of transfer between the various modular drive modes and by simple to use, but high performance measurement and control software (100 kHz data rate, 16-bit resolution). With regards to the tribometry role of this tester, the six-dimension load cells (meaning normal, longitudinal and lateral forces are measured in real-time, as well as the corresponding torques) can be changed depending on materials, load, etc. to allow force measurements within ranges from 1 \(\mu\)N to 1 kN. The instrument also has built-in materials analysis equipment (acoustic emission sensors, atomic force microscopy, digital video) that can be used *in situ* during tests.

This instrument is claimed to be the most advanced device of its type, and with the array of multi-testing modules and instruments onboard, it’s easy to see why. Again, this device has a linear reciprocating mode and track depth monitoring, but it also has the added feature of providing a spiral wear track. This is especially

* At the initial specification stage of the project, discussions with the industrial partners concluded that a range of speeds typical to vehicle testing should be covered by the tribometer. This included skidding due to wheel lock-up at high speeds (\(\approx 30\) ms\(^{-1}\)) as well as the small, rolling velocities present due to tyre deflections. We have since realised that these specifications are not necessary for tyre comparisons to be made successfully.
useful when looking to test on virgin material, something that may be important on ice measurements as the surface structure can change when tested repeatedly. Six-dimensional load cells are used on this instrument; a useful (and expensive) feature that can compensate for the distance between contact point and pin sample holder. No low temperature testing is mentioned.

Figure 3.2. The CETR Micro-Tribometer (Centre for Tribology Inc. (2002)). This instrument takes a different layout than the CSM tribometer allowing a more compact, upright design. This layout provides more room between measurement head and disc for non-standard samples.

**Micro Photonics Standard Tribometer** (*Micro Photonics Inc., PA, USA*)

The Micro Photonics Standard Tribometer (see Figure 3.3) has a distinct resemblance to the CSM Tribometer in appearance and in features. It allows environmental control, reciprocation and track depth monitoring, as well as the added features of electrical measurements and spiral wear track capability. The speed range again does not meet the criteria set for automotive comparisons (≈ 30 ms\(^{-1}\)) with maximum speed of 4.1 ms\(^{-1}\), and the friction force resolution of 10 mN is less than ideal.

One useful feature of this tribometer is the ability to use non-standard test pieces with the clamp system on the disc holder. This easily allows tests of non-adhered and non-uniform materials to be performed. Consideration of balancing of the disc must be addressed if high speeds are to be used to avoid excessive vibration or
damage to the tribometer. Once again, no cold temperature testing is allowed for, but a high-temperature module is available.

Figure 3.3. The Micro Photonics Standard Tribometer (Micro Photonics Inc. (2006)). Very similar machine to the CSM Tribometer using same layout and offering many of the same features. One useful feature is the ability to mount non-standard samples on the disc plate.

CSM Nano Tribometer (CSM Instruments SA, Switzerland.)
The final commercial tribometer is the CSM Nano Tribometer (see Figure 3.4). This differs from the previous three by falling into the market between standard laboratory tribometers and atomic force microscopy (AFM), and is designed to allow “simulation of low load tribological contacts; e.g., in micro electromechanical systems, microsystems and other devices” (CSM Instruments SA (2003)). The technology used for force measurements and movement of samples is derived from AFM, but the device is much larger than these instruments, allowing more conventional samples to be used.

This is the only device that has the capability of testing in a low temperature environment, although this is limited to a cooling stage, and not a refrigerated environment. Being such a small device, extremely high force resolution is provided (1 μN), although this is limited to the range of 50 μN to 1 N. This may be too small for rubber on ice experiments, as the instrument will be sensitive to surface roughness, and so would require extremely well prepared samples, and in
doing so may not be comparable to a real-life system. The size and drive technology again limits the ultimate velocity range which is $0.05 \text{ ms}^{-1}$.

Figure 3.4. The CSM Nano Tribometer (CSM Instruments SA (2003)). This device falls in the gap between a standard laboratory tribometer and an atomic force microscope. The force measurement technique uses a similar method to that of an AFM with fibre optic sensors measuring the displacement of a glass spring cantilever probe. Shown on the right is the linear reciprocating module.

The features on this device were too specialised for use on the tribometer, although it is useful to see the sensing technology on display here. The fibre-optic load cell is a useful feature, and may be a valuable way to isolate vibrations for future iterations of the tribometer.

3.1.2 Custom-built tribometers
In general for laboratory work there are two types of tribometer; linear and pin-on-disc. Linear types have the advantage of having a more realistic geometry, but tend to be used for very low-speed and adhesion-type tests, whereas pin-on-disc instruments are used for higher speed, steady-state tests. A summary of the advantages and disadvantages of each type of custom-built tribometer used by researchers is given in Table 3-II overleaf.
<table>
<thead>
<tr>
<th>Research Group and tribometer type</th>
<th>Advantages</th>
<th>Disadvantages</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blaisdell and Borland (Blaisdell and Borland (1990)) Linear, constant contact type</td>
<td>Samples were 10x the maximum roughness feature size – meets size requirements for small-scale tests. Scatter in measurements was 3% and 11% for dynamic and static friction respectively. Digital data logging system.</td>
<td>Conducted in a cold room – changes in temperature takes a long time, measurement equipment must work at low temperatures. Maximum μ limited to 1.0. Fixed speed of 0.06 ms⁻¹. Large displacements required of &gt; 1 m means the apparatus commands a large footprint. 20 Hz sampling rate. Large samples (46x53 mm) required large loads (222 N).</td>
</tr>
<tr>
<td>Huemer et al. (Huemer et al. (2001)) Linear, lift-off type</td>
<td>Normal pressure programmable from 50 kPa to 800 kPa. Temperatures variable between −25 °C and 36 °C. Can be used on soft surfaces such as compressed snow. 3-D load cells. Digital data acquisition.</td>
<td>Conducted in a cold room – changes in temperature takes a long time, measurement equipment must work at low temperatures. Distance in contact limited to 200 mm. Maximum speed of 1 ms⁻¹. Precision positioning of samples are required to ensure intimate contact.</td>
</tr>
<tr>
<td>Gnörich and Grosch (Gnörich and Grosch (1972)) Pin-on-disc, tangential force measurement</td>
<td>Ice can be swept clean/polished during experiments. Temperature range of −30 °C to 0 °C.</td>
<td>Ice-cooled in alcohol bath. Operates at low speeds – maximum speed 0.05 ms⁻¹. Rubber sample not cooled. Pressure range is 9.8 kPa to 98 kPa.</td>
</tr>
<tr>
<td>Roberts et al. (Roberts (1981a), Roberts and Lane (1983), Roberts and Richardson (1981)) Pin-on-disc, tangential force measurement</td>
<td>Domestic deep freeze used for convenience. Large temperature range of −60 °C to 0 °C. Pressure range of 0.1 to 1 MPa. Ice surface machined in situ ensuring true running surface relative to rubber samples.</td>
<td>Displacement of load cell in tangential direction – induces off-track movement. Maximum speed of 1 ms⁻¹. Temperature measured by mercury thermometer – remote reading difficult.</td>
</tr>
<tr>
<td>Venkatesh (Venkatesh (1975)) Pin-on-disc, tangential force measurement</td>
<td>Instruments kept outside the cold chamber – don’t have to be cold conditions capable. Temperatures down to −40 °C possible. Pressures up to 392 kPa possible. Differing rubber sample geometries used.</td>
<td>Displacement of load cell in tangential direction – induces off-track movement. 395 mm Ø disc – large inertia to accelerate and accommodate in cold environment. Ice “pre-loaded” during testing – this may detrimentally alter the state of the ice. Load cell mounted considerable distance above the point of contact. Maximum speed of to 4 ms⁻¹ recorded. Temperature measured by mercury thermometer – remote reading difficult.</td>
</tr>
<tr>
<td>Spring, Pihkala and Leino (Spring et al. (1985)) Pin-on-disc, rotating force measurement</td>
<td>Movement due to load cell displacement is restricted to the track path – no lateral forces applied to pin. Normal forces measured by a load cell. Speed range of 0 ms⁻¹ to 13.5 ms⁻¹ (largest seen).</td>
<td>Flatness of the ice is dependent on the instrument being level. 4% (dynamic) and 5% (static) measurement accuracy. Large sample size – high normal forces required. 0.9 m diameter ice track. Temperature range of −20 °C to 20 °C. Load cell is a spring balance – not frictionless.</td>
</tr>
</tbody>
</table>

Table 3-II. Summary of custom-built tribometers.
**Linear tribometers**

The linear devices have two general types of operation. One type has a linear carriage (containing the ice sample) moving underneath relative to the fixed rubber sample to which the force sensor is connected (Blaisdell and Borland (1990), Conant *et al.* (1949)) (see Figure 3.5). The samples are in contact throughout the acceleration – constant velocity – deceleration phases. The size of these devices means accommodation in a cold room is necessary. This is an expensive option, and also means that as the signal conditioning or data logging hardware need to be in close proximity, hence they also may need to be in the cold room. When carrying out tests over a wide range of temperature, experiments are time consuming due to the time taken to accomplish changes in temperature.

![Diagram of linear tribometer](image)

**Figure 3.5.** The linear tribometer used by Blaisdell and Borland (Blaisdell and Borland (1990)) for measuring the friction of aircraft tyre samples on sanded ice. The rubber sample stays stationary above the moving ice tray. Normal loads applied to sample by masses and displacement and friction forces measured using digital instruments.

The specifications of the Blaisdell and Borland tribometer (see Figure 3.5) may have been adequate for their particular application, but a maximum fixed speed of 0.06 ms$^{-1}$ and a sample rate of 20 Hz is not high enough for the purposes of this study.

The second type (Huemer *et al.* (2001)) has a similar layout (see Figure 3.6), but with the added feature that the rubber sample is not in contact during the acceleration and deceleration phases and is “touched down” during the constant
velocity phase. This has the advantages of allowing relatively higher velocities as less energy is expended during acceleration as well as eliminating pre-sliding effects such as formation of frost and heating/cooling of samples. Being a linear system, however, this higher velocity capability only amounts to a maximum speed of 1 m/s, again much less than the 30 m/s preferred for automotive comparisons.

Useful features of this tribometer are the range of normal pressures possible, controllable by pneumatics, from 50 kPa to 800 kPa. A three-dimensional load cell measures forces allowing monitoring of normal and lateral forces, as well as the friction force itself. This touch-down method of applying normal forces is also very useful when performing tests on soft, or non-stable materials such as snow, where forces applied over a time prior to testing may affect the surface condition.

Figure 3.6. The touch-down type tribometer as used by Huemer et al. (Huemer et al. (2001)). The rubber and ice are held apart until the specified speed is reached before being pressed together with a force applied by a pneumatic cylinder.

Pin-on-disc tribometers
The pin-on-disc instruments also fall into two categories. The first type (Gnörich and Grosch (1972), Roberts (1981a), Roberts and Lane (1983), Roberts and Richardson (1981), Venkatesh (1975)) has a fixed point from where the friction
force is measured at a tangent to the centre of rotation of the spinning disc (as in Figure 3.7). The second group of devices (Oksanen and Keinonen (1982), Southern (1974), Spring et al. (1985)) measure the friction force from an arm, which is free to rotate about the centre of rotation of the disc (as in Figure 3.8). The second group of instruments have the advantage of keeping the force measurement in the path of the pin, so no radial component is applied as the force sensor deflects (see Figure 3.9).

Figure 3.7. Custom built pin-on-disc tribometers as used by Roberts and Richardson (Roberts and Richardson (1981)) and Gnörich and Grosch (Gnörich and Grosch (1972)) respectively. These devices both use tangential friction force measurement and are loaded by applying masses above the pin sample. Note the use of optics in the Roberts and Richardson tribometer for in situ observation of the rubber sample deformations, and the polishing disc and felt slider on the Gnörich and Grosch tribometer to remove frost from the ice sample.
Figure 3.8. Pin-on-disc tribometers with rotating force transducers as used by Southern (Southern (1974)) and Oksanen and Keinonen (Oksanen and Keinonen (1982)) respectively.

Figure 3.9. Methods of friction force ($F$) measurement used on custom-built pin-on-disc tribometers. Friction force measured in path of pin on disc (b) means no radial displacement due to deflection of the force transducer, as is experienced on tribometers using a tangential measurement axis (a).
The pin-on-disc tribometers have a layout much better suited to velocity-temperature-pressure measurements than the linear tribometers described previously. The tribometers' compact size allows easy accommodation in domestic freezers (with the exception of the (Gnörich and Grosch (1972)) tribometer) and the velocity range is predominantly larger than the linear types.

The use of domestic freezers is extremely useful with regards to cost and experimental protocol. Being small in size allows quick freezing times and for series of experiments at different temperatures to be carried out in short periods of time. The size also permits data and control hardware to be located outside of the freezer, meaning that more expensive cold-temperature hardware need not be used.

A flat, uniform surface is easily achievable using pin-on-disc tribometers as demonstrated by Gnörich and Grosch (Gnörich and Grosch (1972)) who were able to employ a self-cleaning and polishing mechanism to their tribometer to level the surface and to remove frost during experimentation. This is specific to their testing protocol, but it is a useful method to prepare the ice before testing, ensuring there is no runout* in the ice surface.

A feature of the tribometer used by Roberts et al. (Roberts (1981a), Roberts and Lane (1983), Roberts and Richardson (1981)) is the optical setup used to observe Schallamach waves in situ during friction experiments. These were some of the first direct optical measurements to be taken on ice.

Some of the drawbacks uncovered while appraising these tribometers are the widespread use of very simple load cells for measuring the friction forces. The Roberts (Roberts (1981a), Roberts and Lane (1983), Roberts and Richardson (1981)) and Gnörich and Grosch (Gnörich and Grosch (1972)) tribometers both use simple, strain gauge-based load cells with primitive logging hardware, although this is understandable given the age of this research. Another point is the use by Gnörich and Grosch of an alcohol cooling system meaning while the ice was cooled in a controllable manner, the rubber samples were not cooled and were free to warm up in the atmosphere.

* Runout is the lateral displacements of an object as it rotates due to misaligned axes of disc and rotation, eccentricity or imbalance.
In terms of load cell design, the magnitude of deflection of the pin should be kept to a minimum for accuracy of positioning and losses due to friction. It is, however, advantageous to allow the small, necessary movement to occur in a circumferential direction, i.e. in the wear track of the disc (see Figure 3.9). This is where the second group of tribometers are at an advantage (albeit a relatively small one).

From the perspective of the positioning (velocity control) system, again the load cell should have minimal deflection, as unmeasured movement of the load cell is detrimental to the positioning and hence motion control of the tribometer. The drawback with having small load cell deflections is the lack of sensitivity, something essential when measuring the small friction forces usually experienced on ice.
3.2 Design criteria

3.2.1 Temperature range and regulation
The main criterion was to provide accurate measurement of friction forces of rubber on ice. This automatically dictated that the environment around the vicinity of the ice must maintain the sample in a steady state, i.e. any change in the structure or phase of the bulk ice material is not permitted except due to frictional processes during testing. Ice is the more sensitive sample to temperature (as it melts at modest temperatures) so the main precautions were taken to keeping it stable. The rubber sample is also maintained at a mutual freezing temperature to protect the ice surface once contact is made.

The range of temperatures in which most vehicles experience icy surfaces in northern Europe rarely falls below −30 °C, therefore the range of temperatures of most significant interest is −30 °C to 0 °C. Domestic freezers cover this temperature range conveniently, so this was chosen as the temperature-controlled environment. This choice also allows “normal temperature” signal conditioning and data acquisition hardware to be used (as it can be situated outside of the freezer, but still in close proximity to the tribometer), although sensors attached to the tribometer have to operate in the tribometer’s operational range.

The choice of using a domestic freezer also put limits on the size of the tribometer. It was decided to use a small chest freezer with internal dimensions of ≈ 500 × 500 × 700 mm, allowing adequate room for a tribometer of reasonable size plus room for samples and other associated equipment such as an ice surfacing tool (see section 3.4), thermocouples, etc.

3.2.2 Sample size and shape
For the information gathered to be comparable to real tyre situations the size and shape of the samples must be appropriate. From an analysis point of view, tyres can be viewed on many scale levels from full size to (as is the case with brush-type models (Deur et al. (2004))) asperity contact points. A useful comment made by Blaisdell and Borland (Blaisdell and Borland (1990)) is that when comparing
laboratory studies to real life tyre situations, the model scale must be at least ten times any characteristic roughness so that scale problems are not an issue.

As the main interest in this study is to determine rubber-ice interaction at a small, interfacial scale, effects of tyre tread patterns are outside its scope. It was proposed, therefore, that rubber sample scales on the sub tread block size (i.e. < 200 mm²) were most appropriate and be un-patterned in nature (assuming initially a rubber-on-ice geometry). Ice preparation (see section 3.4) allows surface roughness to be much smaller than 0.1 mm, so based on the thoughts of Blaisdell and Borland (Blaisdell and Borland (1990)), samples of thickness greater than 1 mm are acceptable models of “real” geometry.

The ice samples were proposed to be produced from water frozen into a dish-shaped container that forms the disc sample. In reality, ice can take many forms on a driving surface, from very thin films on asphalt (e.g. “black ice”) to in excess of 200 mm on frozen lakes in Scandinavia and model surfaces should be able to replicate this for relevance when comparing. A two-dimensional model was constructed to calculate the stress within ice and to ascertain the extent to which constraint within a dish affects the stress distribution. Two sets of boundary conditions were selected (see Figure 3.10): (a) ice was given a thickness of 1 metre to replicate an almost unrestricted scenario and (b) the ice was fixed within a vertical boundary of 5 mm to replicate the ice-in-dish scenario. Both sets of boundary conditions had the same normal pressure applied of 400 kPa to a 10 mm long face.

Comparing the results of the two simulations it can be seen that restraint within a dish cuts off the extension of the stress distribution as the wall is approached, but the shape of the distribution is repeated relatively accurately. Of particular importance to friction measurements is the distribution at the interface between rubber and ice, and the transition of this is almost identical from high stress at the interface to virtually zero outside the contact area. The zone of maximum stress in (b) was found to be below the surface, at around half of the ice depth. This is not ideal, but as it is believed that the effect of internal ice pressure is insignificant to friction measurements this is not a major concern. The result of these models was
to choose an ice depth of 6 mm to try to alleviate this stress concentration, but still allow the ice to be quickly frozen during preparation. From a thermodynamic point of view, the ice should ideally be infinitely thick, but as this is obviously not possible, and with the normal stresses becoming insignificant at relatively small depths, it was decided that 6 mm should be adequate.

Figure 3.10. The normal stress distribution within (a) unconstrained ice and (b) ice constrained in a 5 mm deep aluminium dish caused by a stationary rubber block under a normal pressure of 400 kPa.

Previous study of the effect of sample shape showed that no significant differences were found in overall shape of rubber pads on ice (Niven (1955), Venkatesh (1975)). Comparative experiments of flat pads, rounded pads (Conant et al. (1949), Wilkinson (1953)), and spherical rubber samples (Roberts and Richardson (1981)) were made and also noted little difference in the resultant friction values. Wilkinson did note, however, that when extremely thin samples of rubber were used, both in flat geometries, and adhered to the surface of a hard spherical mould, the friction dropped significantly, judged to be due to the inability of the rubber to maintain intimate contact with the ice surface.
From these findings, it was decided to use flat rubber cylinders of 2 mm in thickness. These are easy to produce, and represent fairly well the geometry found on tyre treads.

3.2.3 Force ranges
The most critical component of the tribometer is the method of accurately measuring the friction forces. The low friction associated with ice ($\mu = 0.03$ (Bowden and Tabor (1950))) means that friction forces in a representative laboratory experiment could be expected to be as low as 20 to 30 mN. Conversely, coefficients of friction exceeding 30 have been measured under very specific conditions (Barquins and Roberts (1986)). Amontons law states that the friction force is the product of the normal force and the coefficient of friction, so assuming a representative contact pressure of 300 kPa on a 12 mm$^2$ contact patch, the predicted maximum friction force is $\approx 15$ N, based on a representative maximum coefficient of friction of 4 (Gnörich and Grosch (1972), Southern and Walker (1972)). The resolution of this force measurement should be adequate to measure the 1 mN minimums that may occur (based on the low friction of ice close to its melting point (Roberts (1981a), Southern and Walker (1972))).

The method of measuring the force must be essentially friction-free, as any frictional losses in the force measurement system will be realised as unnecessary error in friction measurement due to the ice and rubber.

It was decided that normal loads should be applied by means of applying known masses directly above the point of contact of the pin on the disc. The load can be easily selected and can be assumed to be unchanging throughout experiments. This removes the need for a control system or a second load cell as (if the surface is suitably prepared) no significant deviations should occur to this fixed weight through accelerations, inertias, etc. The weights are chosen to accurately apply representative contact pressures to the samples, and this will of course depend on the size and geometry of the samples used. Using the earlier pin sample size example of 12 mm$^2$ and the representative pressure of 300 kPa, the required mass would be approximately 400 grams. Varying the sample geometry would require a corresponding change in load, so the tribometer was designed to accommodate this.
3.2.4 Velocity range and control
Although the majority of published data suggest that friction is more prominent at low speeds, a comparison to data of a vehicle skidding at high speed may be required. This criterion would set the maximum speed of the drive system to reach $\approx 30 \text{ m/s}^{-1}$ (67.5 mph), but allowing good resolution at low speeds and static to sliding friction transition possible.

Gaining good speed resolution and high speeds is usually not possible simultaneously as resolution is usually produced as a percentage of the full-scale output. To obtain both of these targets, a speed reduction system such as a gearbox is required. This allows the speed to be reduced, while maintaining the speed resolution. Obviously, in terms of resolution, the higher, the better, but this is again determined by budget. For this study, good resolution at high speed is not so important, but smoothness in rotation is favoured. At low speeds, where friction speed dependence is highest, the resolution should be good, probably in the region of $0.01 \text{ m/s}^{-1}$.

As well as providing a range of steady-state velocities, the control system for the disc motion should allow transitions in speed, i.e. programmable accelerations and decelerations. This is essential when studying the transient effects proposed as experimental measurements (see Chapter 5).
### 3.2.5 Summary of design criteria

<table>
<thead>
<tr>
<th></th>
<th>Temperature</th>
<th>Sample</th>
<th>Force</th>
<th>Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Ice</td>
<td>SBR</td>
<td>Friction</td>
<td>Normal</td>
</tr>
<tr>
<td>Range</td>
<td>-30 to 0 °C</td>
<td>&gt; 50 mm diameter</td>
<td>&lt; 10 mm diameter</td>
<td>0 to 15 N</td>
</tr>
<tr>
<td>Accuracy</td>
<td>± 1 °C</td>
<td>-</td>
<td>± 1 mN</td>
<td>Calibrated</td>
</tr>
<tr>
<td>Measurement</td>
<td>K-type thermocouple, averaged pair</td>
<td>-</td>
<td>Strain gauges in Wheatstone Bridge configuration</td>
<td>Optical encoder feedback</td>
</tr>
<tr>
<td>Control</td>
<td>Domestic freezer, temperature controlled thermostatically</td>
<td>Surface machined on tribometer for minimum relative runout</td>
<td>Cylindrical, contact surface prepped before experimentation</td>
<td>Calibrated masses, static loading</td>
</tr>
</tbody>
</table>

3.2.1 chapter section relating to particular parameter

Table 3-III. Summary of tribometer design criteria for both measurement and control parameters.
3.3 Design development
A modular approach has been taken in the design of the tribometer, allowing the best solution for each component to be combined in the final design. The use of modules allowed development and replacement of separate components to develop performance and eliminate problematic mechanisms, both mechanical and electrical. The components of the tribometer have been reduced to five main categories: drive system, friction force measurement, normal force application, signal conditioning and data acquisition, and control software.

It was decided to use a domestic chest freezer as the environment control system, given the small footprint possible with a pin-on-disc tribometer and the ease of use a commercial freezer provides. This allows temperatures of down to $-35$ °C, low enough to provide the full range of measurements proposed. Such a simple temperature control system allows design complexity to be reduced with refrigeration or ethanol cooling plates not required to keep the ice at a stable temperature. The one drawback with this simple solution is that the electronic control and measurement components used on the tribometer have to operate in this environment, and so in some cases are specifically designed for low temperatures.

3.3.1 Drive system
From an ideal point of view, a linear motion system would be perfect for friction research, for as soon as something slides, little control of direction is possible (and changes of direction is an important effect in dynamic friction measurements). This is clear to see when looking at winter driving conditions: in the UK, a significant number of winter accidents occur due to the inability to control (i.e. steer) the vehicle once traction has been lost (48% - DOT 2003 (Tyres & Accessories 2004) (Etyres (2006))). Pin-on-disc systems inherently apply a radial acceleration to the motion because the pin is following a circular track on the disc instead of a straight line.

For laboratory experiments however, a linear system is notoriously difficult to implement due to size restrictions and control systems, and the range of speeds producible is much less for a given footprint. It is not a surprise therefore that the
majority of laboratory friction test equipment is of the pin-on-disc type. These systems can easily produce accurate high-speed pseudo-linear motion in a very compact unit by replacing the linear track with a flat disc. With speeds of up to 30 ms\(^{-1}\) required, a linear system would not be feasible so pin-on-disc is the obvious choice.

The drive component chosen for the tribometer is a DC servomotor manufactured by Faulhaber Group GmbH & Co KG (model number 3863 024C). It has an operating temperature range of \(-30{\,}^\circ{\text{C}}\) to \(+125{\,}^\circ{\text{C}}\) and is of coreless design allowing high torque (up to 1.25 Nm) and efficiency (85%) in a compact package. The maximum permissible speed is 6700 rpm and axial loading of 6 N is allowable. The motor is fitted with an optical encoder, also supplied by Faulhaber, with a 1024 pulse per revolution resolution.

This is one of the few servomotors with high specification that is operationally rated at temperatures below \(-20{\,}^\circ{\text{C}}\), essential with the tribometer being used in a freezer. A high-efficiency motor is also important to ensure excess heat is not produced close to the ice, changing its temperature during testing. The small size also ensures less heat production than a larger-bodied motor. A high specification also allows a versatile design strategy, with axial loading possible directly on the motor, without the need for additional support. Calculations show (see Appendix I) that the specification of this motor produces favourable torque over a high range of speeds. One problem with electric motors, however is the lack of power at low speeds, a problem compounded by the fact that high ice friction is produced at the lowest sliding speeds. This can be overcome by using a small disc radius, although this means that the nonlinearity due to disc diameter is emphasised, something that is not desirable. The other option is to use a gearbox, diminishing the range of speed, but boosting the torque output and the low-speed velocity resolution.
3.3.2 Friction force measurement

Figure 3.11. A generic force transducer on a pin-on-disc type tribometer. The main components are the pin sample, and the pin sample holder, which are connected to the tribometer body with a measurement arm. Relative coordinate axes are indicated by \(x, y, \) and \(z\).

Force sensors (a schematic diagram of a generic sensor is shown in Figure 3.11) have always posed a problem for tribologists. Friction forces are caused due to the motion of one sliding component over the other, so inherently this motion must occur for a friction force to be measured. The friction force may appear to be constant for a fixed load-temperature-velocity regime, but this force is the sum of interactions at many scale levels, most of which are dynamic processes. When trying to measure this force, it is important to separate the friction processes and the measurement technique so they are mutually exclusive. Force measurement devices almost universally require movement of one form or another, so it is essential that this movement causes minimal effect to the friction of the system that is being studied. It is this aspect that causes problems in measuring what is essentially a "static" force, and requires consideration with regards to the overall instrument dynamics.

With the difficulty in measuring forces at a small contact point (\(\approx 1 \text{ mm}^2\) if planar-spherical contact geometry is assumed), due to geometry restrictions the forces are
usually measured “close to” the contact, which in reality is an appreciable distance from the interface. The combination of the friction force and this distance provides a moment around the centre of the transducer rotation. Depending on whether this centre is leading or lagging the contact point, the moment induces an impulse that reduces or increases the normal load respectively, hence changing the measured friction. Figure 3.12 below illustrates this coupling effect.

![Figure 3.12. Front elevation of a generic force transducer showing the moment (M) produced by friction force (F) about the centre of rotation (C). The coupled impulse is proportional to the off-centre distance (s).](image)

The inline position (where $s = 0$) means that although a moment is still being produced around C, a negligible impulse occurs in the normal direction for small rotations.

A second form of coupling occurs when the axis of deflection of the measurement arm does not correspond exactly to the plane of the disc sample surface (see Figure 3.13). Usually cantilever-type force transducers are designed to deflect in the measurement axis yet remain stiff in the normal axis. If the axis of measurement does not lie parallel to the surface (where the friction force acts) “jacking” occurs as the off-axis deflection of the transducer acts to lift the load off or press down on the surface, hence reducing or increasing the normal force. In practice, this effect would produce juddering of the transducer as the friction force fluctuates with high and low normal loads as it passes over asperities.
Figure 3.13. Front elevation of a force transducer showing the "jacking" motion caused by misaligned axes of transducer deflection (X) and friction force (F). The extent of "jacking" is proportional to the misalignment angle (α).

As well as causing coupled forces, off-centre friction forces cause torsion in the measurement arm. Torsions are difficult motions to measure, and although within the elastic limit of the arm material torsions are proportional to the corresponding x-axis and z-axis components, they are undesirable. With these characteristics of existing force transducers it was decided to eliminate these problems as much as possible. To do this, the centre of rotation of the force transducer had to be aligned in both x-axis and z-axis to the contact point on the interface. As demonstrated in the preceding diagrams, this is not straightforward using a single cantilever design, so a dual cantilever arrangement was devised. Figure 3.14 shows a schematic drawing of the dual cantilever design, showing how the combined cantilevers allows the centroid (or centre of rotation) to be positioned directly at the contact point of the pin on the disc sample surface.

Figure 3.14. End view of the dual cantilever force transducer. The centroid of the transducer coincides with the contact point of pin on disc so no moment is produced by the friction force (F). Deflection must take place in the plane of the disc surface, acting collinear to F.
Now that the deflection of the measurement arm is controlled in a manner that does not affect the source of what is being measured, a device is required to convert deflection into a force reading. The two methods of achieving this are to measure the deflection of the pin at the point of contact, or to measure the induced strain of the measurement arm.

To measure the deflection at the pin, a displacement measurement device would be required. This could take the form of inductive devices (linear variable differential transformers, or LVDT), resistive devices (potentiometers) or digital encoder-type devices. The main problem with these devices is that of internal friction. Although some manufacturers state their transducers are "friction free" they are not completely free of frictional forces due to the magnetic effect of coils etc. within the device. Bearing in mind the small magnitude of some of the forces expected from rubber on ice, any friction within the force transducer will have a detrimental effect on the accuracy of the instrument.

The second method is measuring the strain induced by the friction force on the measurement arm. Some of the most versatile devices for doing this are strain gauges. These are passive, resistive devices that can be calibrated to produce a representative voltage output for a given strain input. Because of their simple operation, they can provide extremely high resolution in measuring small forces as well as exceptionally high forces, depending on the installation method and mechanism.

There are several types of strain gauges depending on environment, scale, magnitude of strain etc. The main governing criteria with regard to the tribometer are that of low temperature and high moisture. The highest resolution can generally be attained using semiconductor type gauges, but historically, electronic devices (i.e. semiconductors) and low temperatures tend not to mix: electronic devices such as integrated amplifiers that operate at low temperatures are given military ratings such are their complexity and cost. This limits the choice to metal resistive gauges. These are not as highly sensitive as semiconductor type, but are not affected in operation by low temperatures. When used in a full Wheatstone Bridge configuration provides a highly sensitive force transducer. Figure 3.15
shows a typical full bridge strain gauge circuit diagram.

![Wheatstone Bridge Circuit Diagram](image)

**Figure 3.15.** The Wheatstone Bridge strain gauge configuration with fixed supply voltage ($V_{in}$). Compressive ($-R_g$) and tensile ($+R_g$) strain gauges work in harmony to increase the force transducer sensitivity, producing a representative output voltage ($V_{out}$) of the strain input.

The dual cantilever design of the force transducer lends itself well to the Wheatstone Bridge strain gauge layout, providing two tensile and two compressive surfaces for any deflection direction (the four vertical faces of the measurement arms, see Figure 3.16).

A full bridge system is essential when working within a range of temperatures, as gauges can be very sensitive to temperature change. Using four identical gauges within the freezer means that any resistance change due to temperature will act on all of the gauges, the step resistance change is balanced on all four arms of the bridge circuit and so no temperature dependence is shown in the output voltage.

Having four active gauges increases the sensitivity of the force transducer to twice that of a two active gauge transducer. Placing the gauges as close as possible to the root of the cantilevers means the gauges are subjected to the maximum strain for a given deflection, again optimising sensitivity. Higher sensitivities can be produced by having less rigid measurement arms, at the detriment of larger displacements at
the pin end of the arms. Therefore there is a tuning process of attaining the best possible signal without the displacement detracting from accurate velocities at the interface. Providing a simple method of changing the arms in the design allows this tuning process to be as easy as possible. There is also a natural frequency issue by using this method. This should not correspond to particular disc frequencies as resonance has a detrimental effect on friction force measurements.

One problem that is associated with strain gauges when used to monitor dynamic forces is that of electrical noise pick up. This is because of the low operational voltages associated with the gauges, so any ambient noise will be of a significant magnitude compared to the strain gauge signal, hence a low signal-to-noise ratio. This can be filtered out using software, but it is undesirable and all efforts have been taken to boost the signal to raise the signal-to-noise ratio. One way is by using less stiff arms (within reasonable limits to prevent excess deflections, i.e. max deflection is 3 to 4 mm), hence allow higher strains therefore larger signals. Again, this has to be carried out carefully with regard to the adverse effects of excessive deflection.

The second (additional) method is to boost the signal by using an instrumentation
amplifier as close as possible to the point of measurement. This method allows the signal to be boosted without the need for large pin deflections or high input voltage, which can be undesirable with regards to temperature compensation. By placing the amplifier as close as possible to the strain gauges (maximum of around 100 mm), pick up of ambient noise will only take place through the strain gauges (plus the short connecting wires to the amplifier). The amplifier (Burr-Brown® model number INA15AU) is of "military standard", meaning it can operate at the low temperatures experienced in the freezer environment (it is rated to —40 °C). The boosted signal from the amplifier will be subject to the same ambient noise, but as the gain factor of the amplified signal is around 100 times that of the gauge signal, the signal-to-noise ratio is significantly improved, providing a much cleaner force signal.

3.3.3 Normal force application
As discussed previously, it is crucial to keep the normal force constant throughout friction experiments. Even if all measures are taken to remove coupled forces due to instrument dynamics, changes in terrain still exist on the disc sample, which invariably change the magnitude of the normal force. This means that the method of applying this load must adjust for topography changes. This is particularly important in pin-on-disc systems, as any runout of the disc will affect the force measurement producing periodic maxima and minima. It is possible to create a servo system providing feedback to adjust for the terrain change, although this would prove to be unnecessarily expensive for the performance required. The simplest alternative solution is to allow the force transducer to rotate about a pivot, allowing the pin to rise and fall over undulating terrain without affecting the normal load (significantly). Of course rates of change in topography causes fluctuations in normal load as the pin is accelerated vertically over the surface, but this should only be significant at high velocities if all efforts are taken to ensure the disc surface is as flat as possible.

Although adjustment for terrain is essential, it is also important to restrain the sensor in movement in the radial direction and tangential directions to maintain velocity and ensure friction-free measurement respectively.
Normal loads could be applied to the pin in two ways: using an actuator to apply the force or loading with known masses. Actuators can take several forms: linear electrical actuators with motor driven or voice coil driven pushrods, hydraulic or pneumatic cylinders, lead screws, etc. Applying forces by actuators has the benefit of control over normal forces during experimentation, should this be deemed necessary, and would add an extra dimension to possible test types. They can also be inertia-free, either by design or programming which is something that may be beneficial on undulating terrain. The main drawback however is again the cost factor of implementing the actuator and control system and complexity of maintaining and calibrating such a system when normal forces are so critical in the friction measurements.

Static loading by known masses is the most straightforward and cost effective method to implement and calibrate, and so this has been chosen for the final design. The loads can be precisely maintained and quickly changed to the required specification and allow excellent repeatability for series of experiments. There are no predetermined limits to the loads that can be placed and resolution can be extremely high by using commercial weighing scales to calibrate before experimentation.

By using known weights, it is necessary to consider the mass of the structure supporting the normal load, as this will add a contribution to the normal load. The most satisfactory method is the use of a counterweight. As the measurement arm rotates about the pivot the counterweight compensates for the mass of the arm, maintaining constant weight acting on the pin. The positioning of the counterweight does not impede the motion of the supporting structure, and also allows the pivot to be swung through 90° to allow easy access to the disc sample.

### 3.3.4 Signal conditioning and data acquisition

To deal with the high data rates required from the force transducer and to retrieve velocity data from the motor encoder a high performance data acquisition system is required. Temperature readings are also required to observe the experiment temperature, and to monitor temperature changes occurring throughout experiments. Specialised serial- and USB-based equipment is readily available for...
such tasks that can be used on simple PC hardware, but for three different data sources and types different hardware can prove to be problematic. As the sources of data are linked, it is important to be able to synchronise the data to see event dependencies and transients, something that can be difficult to do at high sample rates. Data is often buffered to attain high sample rates, and sent to the PC in “packets” which have to be synchronised post-event. It was therefore decided that data acquisition should be carried out using a single product, or a family of products by the same manufacturer. Motion control of the drive system should also be possible in correspondence to the data acquisition, so that specific tests such as static-sliding transients are possible with experiment efficiency. The only product that could achieve this requirement in a simple to use environment was LabVIEW® and the associated hardware made by the National Instruments Corporation.

LabVIEW® is a graphics-based programming environment that is popular with academia and industry alike for data acquisition and signal processing. It allows fast set up times with specialised routines for common tasks with the ability to be fully customisable for specific applications. National Instruments also produce a range of signal conditioning and acquisition hardware, fully compatible with and controllable within the LabVIEW® environment. Although more expensive than similar products by other companies, using National Instruments hardware allows easy integration of different sections of the tribometer, and eliminates the possibility of compatibility issues between manufacturers.

Signal conditioning is essential for transferring small-signal analogue data to the PC without the chaotic effect of picking up extraneous noise, and preparing the signal for data logging on the PC. The data is filtered and amplified before being transferred to the analogue-to-digital converter (ADC), hence ensuring a high signal-to-noise ratio. The data acquisition system consists of several components, shown in Figure 3.17.

The data acquisition and control system is split into two divisions: the motion control and velocity measurement section, and the analogue data acquisition section. Within the two divisions, there are actuators and sensors, analogue signal conditioning, and digital data and control steps.
Figure 3.17. Tribometer control and data acquisition hardware. There are two divisions of hardware; motion control (dotted line) and data acquisition (solid line), each requiring a separate PCI card to be inserted into the host PC.

The two main components associated with motion control in the tribometer are the DC servomotor and the optical encoder. To operate accurately, the servomotor requires position/velocity feedback as well as power. The power is supplied by a servo motor drive (NI MID-7652) and the encoder provides the feedback. The encoder also provides position and velocity data that can be monitored via software. Commands sent from the PC to control velocity are processed by the motion control PCI card (NI PCI-7342) and sent to the servo motor drive.

Signal conditioning for the analogue data section (i.e. force transducer and temperature sensors) is based around the National Instruments SCXI® platform. This consists of a stand-alone, modular chassis into which specific signal conditioning modules can be placed. National Instruments produce a multitude of individual conditioning modules depending on what measurement and data types each customer might want to use. In this case, analogue force and temperature measurements are required so two modules are needed in the SCXI® chassis. The SCXI® chassis (NI SCXI-1000) is mains powered, providing a filtered power supply to each of the four modules it can accommodate. For force measurement,
strain gauges are used, and National Instruments produce an eight-channel full bridge module (NI SCXI-1520 and NI SCXI-1314 terminal block) specifically for this purpose. This module amplifies and filters the small signal from the strain gauges, as well as providing a filtered excitation voltage to power the full bridge circuit. For temperature measurements, K-type thermocouples are used, and the suitable module (NI SCXI-1112) provides channels for up to eight thermocouples. This module provides a feature called “cold-junction compensation” which allows the erroneous temperature offset associated with thermocouple connections to be eliminated by monitoring the temperature of the thermocouple connection.

As personal computers cannot deal with analogue data, an ADC is required between the PC-based logging and control stage and the signal conditioning unit. This piece of hardware converts conditioned analogue data of between —10 and +10 volts into digital data. The resolution of the ADC is described in terms of bits and is a measurement of the accuracy to which the converter can represent analogue data in a digital form, the higher the resolution, the higher the accuracy of replication. The NI PCI-6052E ADC which was selected for use with the tribometer has a 16-bit resolution, meaning it divides the total analogue input voltage into $2^{16}$ (binary data, 1/0 to 16 levels) discrete values or 65536 “quantization levels”. This is the maximum theoretical resolution possible, but in practice noise affects the actual resolution, and diminishes it to the level of the apparent noise magnitude. Theoretically, with full 16-bit resolution, the full-scale accuracy would be 0.31 mV, but with signal-to-noise ratios of the ADC taken into account, the hardware is rated to 4.75 mV full-scale. Although this decreases the accuracy of the ADC by more than a factor of ten, the resolution is still extremely high (accurate to 0.024%), adequate enough to accurately measure the large range of force readings expected on ice. Assuming the minimum force to be measured is (as specified earlier) 1 mN, the accuracy of the ADC allows maximum forces of over 4.2 N to be measured within a single range. If larger forces are expected, the input range can easily be configured to allow larger voltages to be measured to a lower resolution, and vice versa for small forces. This ease of configuration was one of the main criteria for selecting this particular ADC, allowing a vast array of tests under different conditions to be run with the minimum of effort.
Another feature of this ADC is the high sample rate available. National Instruments "guarantee" a 333k samples/s sample rate for any given channel on this ADC. For multi-channel sampling, the whole sample rate is divided by the number of channels used, so if 3 channels are used, the maximum available rate is 111k samples/s. This high rate allows events of small duration to be identified, something that is extremely important in the transition from static to sliding friction. Assuming two channels are used, measurements at time intervals of 6 μs are possible allowing extremely accurate measurements of the peak static force to be made.

This hardware allows each of the sensors and motor to be controlled and read, but in order to do this in a controlled and useful manner, software has to be written.

### 3.3.5 Control software

The LabVIEW® environment is a graphical-based programming system allowing readings to be taken from sensors, control of actuators and logging of data in a logical manner. Conforming to an entire system of software and hardware produced by National Instruments allows straightforward configuration of equipment from within the programming environment. As described previously, the hardware provides configurable, versatile interfaces to measure and condition signals, and it is this configurability that makes the hardware so powerful in research instruments such as the tribometer. It is also essential to be able to configure the tribometer for different testing regimes, as there can be two orders of magnitude difference in μ depending on the conditions of the test. In general, measurements over a large range result in lower accuracy than over a smaller, constrained range. When the magnitude of, for example, force measurements can be pin pointed to fall within a known range, the hardware can be configured accordingly to provide much higher resolution within this range hence a higher degree of accuracy. This is a much more straightforward proposition than designing appropriate amplification or measurement hardware for each range of measurements to be taken. It also allows interchangeable or replacement components to be set up quickly for use.

When writing the measurement and control system the main hurdle to overcome
was the inability of LabVIEW® to carry out simultaneous tasks within a single program window. Initially, it was impossible to read data and control motion in a reliable fashion, but with experience gained by using the program, it was possible to find alternate routes to the ultimate goal. Independent subroutines (or "vis" ("Virtual Instruments")) can be created to run in the background during execution of the main control window. This approach provides a powerful interface for data acquisition and control of the tribometer. With independent subroutines, however, it is essential to synchronise the two separate data files to allow meaningful cross-references during analysis of data. The time stamp recorded for each data file is derived from a common system clock, so once synchronisation is initiated time alignment is achieved for all data. Providing a system-wide start command to execute both subroutines, after which no user commands are required, does this effectively. A flowchart of the LabVIEW® control system is shown in Figure 3.18. Note the only simultaneous command is the start trigger, after which the subroutines run completely independently.

The programming structure of LabVIEW® is such that there are two windows:

- the graphical user interface (GUI), or the front end, which the operator interacts with by entering parameters and issuing commands, and
- the command block diagram, or back end, which is constructed by the programmer and forms the architecture of the control system.

The important one with regards to the smooth operation of the control and measurement system is the block diagram, as it is this that determines the configuration, logic and calibration of all measurement data and control commands. The GUI has essentially two functions: (i) a way to control the tribometer by accessing the most important functions of the block diagram and (ii) to allow the operator to interact with the instrument in an aesthetic fashion and to display the data in a useful manner. It is also possible to set a level of security on the tribometer so that "operators" can use the tribometer and change test parameters accordingly, but cannot alter the control system or critical parameters, hence no harm may be done to the instrument. This is especially useful in a multi-operator environment, where preferences can be set, and setup easily reverted when needed.
Figure 3.18. A flowchart of the measurement and control system used for the tribometer. The measurement and motion subroutines run independently, produce separate data files and are synchronised by an initial start command.

The main GUI for the tribometer is shown in Figure 3.19. This gives the operator access to the main functions of the tribometer, i.e. setup of initial motion control parameters (wait, velocity (C3), acceleration (C1)), OK (start) button, real time test duration indicator (Elapsed Time) and the Stop button. Data functions of the tribometer such as sampling rate, sensor channels to log, calibration, ADC gain factors, etc. are available to access through the respective subroutine GUIs and block diagrams. All GUIs and block diagrams are shown in Appendix III.
Figure 3.19. The GUI of the main control .vi. Functions that are controlled from this window are Start (OK Button), wait, velocity (C3) and acceleration (C1) motion parameters and Stop. Elapsed test time is indicated in seconds. Other tribometer functions are controlled through other similar GUIs.

LabVIEW® has an extremely large number of functions (.vis) available for use, some very useful and others only useful for extremely specialized applications. Only some of the more common .vis were required to program the tribometer control application successfully. A screen shot of the main block diagram is shown in Figure 3.20. This block diagram corresponds to the main GUI (Figure 3.19) for controlling the basic operation of the tribometer. Each of the controls on the GUI has corresponding functions in the block diagram (note replication of control/function names in both GUI and block diagram), but not all functions of the block diagram are accessible through the GUI. This allows an uncluttered front panel to the instrument with only the most important parameters available as well as providing a certain level of non-administrating operator restriction.
Figure 3.20. The block diagram of the main control .vi. Shown are the two subroutines within their respective true/false loops, the start (OK Button) trigger control, initial conditions for the motion control system and the stop control with "Elapsed Time" and "stop" conditions.

The .vis shown within the true/false loops are the corresponding motion control and measurement subroutines outlined in Figure 3.18. As explained, these .vis run independently, each having their corresponding block diagrams and GUIs (Appendix III). These provide the "backbone" of the control system, logging data gathered from and providing motion control of the tribometer.

Programming the control and data system in LabVIEW was an ongoing process from the design stage through to during data collection by running experiments. The learning curve was relatively straight: initially it was relatively easy to collect sets of data from the corresponding sensors and to control the motion of the disc, but integrating these functions into one structure proved to be more difficult than expected.
National Instruments have a useful online knowledge base (NI Developer Zone), with solutions to common problems addressed and example code posted by users in academia and industry. This was a very useful database and looking at this resource and the code posted solved many of the more simple problems. Studying the methodology used by these other programmers enabled the tribometer application to be developed successfully and to find the solution to the problem of integrating all of the control and data acquisition components.
3.4 Final design

As stated previously, the tribometer design is based around a series of modular elements, i.e. motion system, friction force measurement, normal force application, signal conditioning, and control system. The final design is combining these elements to work as a complete system, i.e. the tribometer. The main criterion for this final assembly is that the tribometer operates as a complete system, with each of the constituent parts performing as they would in isolation. Careful consideration has been given to minimise electrical and mechanical interference between elements allowing them to perform to their maximum potential. The modular design approach was taken to allow easy exchange of components and modification to the tribometer, so the final design reflects this by allowing all components to be easily and quickly removed and replaced.

Figure 3.21. The final tribometer instrument design. The main components are labelled.

The structure of the tribometer first of all supports the force transducer and the disc sample. The force transducer rotates about its axis to allow for terrain undulation...
of the disc surface. This is the only degree of freedom the force transducer possesses, as deflection of the transducer itself is critical to the accuracy of force measurement, and movement of the transducer is therefore not permitted. This movement is obtained by mounting the force transducer using a pair of opposing angular contact bearings (to support the axial forces produced by the friction of the pin on disc) and an instrument ball bearing (see Figure 3.22). This arrangement allows free rotation about the transducer pivot, but axial displacement is controlled by the angular contact bearings and an adjustable thrust plate to remove play due to wear, etc.

Figure 3.22. Cross section of the bearing housing showing the pair of opposing angular contact bearings designed to support friction forces but allow axial rotation to compensate for terrain undulation.

Normal loads are applied to the pin by loading masses of known weight to the top of the pin sample beam. They are held in place by a specially designed mass rack, including a clamp to stop the masses vibrating during testing which introduces mechanical noise into the force measurements. The force transducer incorporates adjustable and replaceable arms. These can be changed in length and replaced by thinner/thicker ones, should that be deemed necessary. A counter weight was included on the force transducer to balance the weight of the transducer and the
mass carrier. The counterweight is adjustable so that it can be moved to balance the different weight of different cantilever arms and pin sample sizes. The bearing housing is rigidly attached to the tribometer chassis, but its position can be adjusted to select the track diameter on the disc.

The design of the sliding mount for the force transducer allows replacement by a servo-controlled linear bearing system. It was envisaged that this could be beneficial (at a later date) for extended tests on virgin ice material. The addition of a second servomotor would allow a spiral track to be programmed meaning that the pin could run on “new ice” throughout the experiment, instead of following the same track. This projected system was not utilised, and instead solid aluminium slides were used to check the system prototype and therefore keep costs down.

The disc and motor assembly was also attached rigidly to the chassis, although it was decided to manufacture a custom gearbox to reduce the speed of the disc for low-speed measurements. The servomotor has maximum power available at the intermediate speed of 3400 rpm, with power tailing off to a minimum at highest and lowest speeds. A gearbox allows the maximum power to be available at a lower speed (selectable through different gear ratios), where friction will be highest. The speed reduction also has the effect of better speed control as the encoder is attached to the motor. At, for example, a 20:1 gear ratio, the speed of the sample disc will be twenty times slower than the motor, so a step correction at the motor would give a twentieth of a step correction at the disc, hence smoother velocity profile. The gearbox was designed to be removable so that if higher speeds are required, the gearbox can be bypassed, and the disc can be mounted directly onto the motor, to give a full range of speeds (at a lower speed resolution).

The chassis was fabricated from aluminium to resist corrosion in the damp environment. There are 88 separate components, bolted together onto the base plate. The simple layout allows modifications to be made easily, with components being easy to relocate, e.g. to allow larger discs or different pin samples to be used.

There were two criteria when designing the sample mounts on the tribometer. Firstly, the samples had to be held as rigidly as possible to the instrument so that
compliance of the rubber or ice creep was allowed to affect the corresponding friction mechanisms without being too excessive making the deflection unrealistic. Secondly, the samples must be easily removed and replaced as the samples wear or the ice melts. This was of particular importance when carrying out observation of worn ice samples using low temperature scanning electron microscopy (LT-SEM) (see Chapter 4). With these in mind, quick-release mounts would be ideal, but the scale of the pin sample in particular meant that space was the governing parameter.

The very basic grub screw method was used to support both pin and disc samples. The pin sample was based around the sample stubs used in LT-SEM with view to carrying out work using this instrument at a later date. The stubs are aluminium cylinders, 10 mm in diameter and 6 mm high. It was decided that bonding the pin material onto the stubs would give a good sample 'module' as well as being easily transferred into the LT-SEM when required. A grub screw through the sample cross member on the force transducer holds the stub in place, and allows quick and easy removal.

The disc sample holders (for ice) were custom built dishes of 90 mm in diameter. These can be mounted onto the final drive from the gearbox and directly onto the shaft of the servomotor by a grub screw. A collet was manufactured to assure that the disc was balanced when used at the high speeds when mounted on directly the motor. The disc itself was machined with recesses to provide good mechanical keying to make sure the ice did not slip at higher temperatures. Separate sample holders were made for bonding rubber to when using the ice-on-rubber geometry. (Teflon® disc samples were machined from a cylinder of Teflon® so did not require a sample holder/plate.)

Before experiments, it was important to assure that the ice disc sample surface was flat relative to the pin so that terrain undulations due to runout were avoided where possible. When preparing the ice in the sample disc, water was poured into the dish and allowed to freeze. If the dish was perfectly horizontal and flat relative to the axis of the force transducer, runout would be avoided due to the Earths gravitational pull. This prerequisite is avoided by using an ice-machining tool, designed to both level the surface of the ice from pebbling and ensure there is no
runout relative to the pin. The tool is a simple lathe machining tool, mounted onto a custom pivoted arm on the tribometer chassis. The motion of the arm allows the ice surface to be machined to a specific height and assures it is flat relative to the pin sample (see Figure 3.23).

Figure 3.23. Diagram of the tribometer with the specially designed ice machining tool attached. The cutter can be moved radially across the plane of the ice surface to ensure it is flat relative to the force transducer.

In terms of the electronics and sensing equipment, connection was fairly straightforward. The force transducer and K-type thermocouples were connected to the PCXI-1000 signal conditioning unit, which was situated outside the freezer. The two thermocouples were located as close as possible to the point of contact between pin and disc without interfering with the system. Power was supplied to the force transducer by the signal conditioning unit.
The servomotor has its own power supply, controlled by the MID-7652 servo amplifier. Feedback was supplied to and power received from the servo amplifier for the digital encoder.

The final specification of the tribometer is shown in Table 3-IV. Reference to the design criteria in Table 3-III can be made, showing that the criteria were met for all parameters, except for that of friction force accuracy. This is the main reason for the modifications explained in section 3.5.
<table>
<thead>
<tr>
<th>Temperature</th>
<th>Sample</th>
<th>Force</th>
<th>Velocity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Disc</td>
<td>Pin</td>
<td>Friction</td>
<td>Normal</td>
</tr>
<tr>
<td>Range</td>
<td>−33 to 0 °C</td>
<td>&lt; 90 mm diameter</td>
<td>&lt; 10 mm diameter</td>
</tr>
<tr>
<td>Accuracy</td>
<td>± 1 °C</td>
<td>1 mm</td>
<td>1 mm</td>
</tr>
<tr>
<td>Measurement</td>
<td>K-type thermocouple, averaged pair</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Control</td>
<td>Ice surface machined on Domestic freezer, tribometer, SBR hemispherical, temperature controlled thermostatically</td>
<td>Ice sheet bonded to SBR cylindrical, machined disc, contact surface contact surface prepped before experimentation</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 3-IV. Summary of final design specification of the tribometer for both measurement and control parameters.
3.5 Modifications

After the tribometer was built, preliminary tests were carried out to check the operation of the instrument and to determine what modifications needed to be carried out to meet the criteria set before design. These tests showed that most systems performed satisfactorily, although the force signals, despite the consideration given during the design phase, were noisier than expected.

The absolute friction force values averaged over the fairly short time of a second or two seemed to correspond fairly well to data previously published (Conant et al. (1949), Roberts (1981b), Wilkinson (1953)) for steady state experiments, but the level of the noise was \( \approx 200\% \) of the absolute friction force values, which was deemed too substantial. There were two sources of this noise: (i) airborne and mains line-sourced electrical noise and (ii) mechanical noise from the gearbox. The following sections explain how this problem was rectified through modifications to the drive system, measurement devices and by modifying the tribometer layout.

3.5.1 Drive system

The mechanical noise from the gearbox was the source that was deemed to be the most difficult to rectify. The gearbox was an integral part of the final design allowing selection of several ranges of speeds, dependent on experimental protocol. The use of gears was not perceived to be a problem before the tribometer was built, but it became an obvious problem after preliminary testing.

The type of gearing used seems to have been the main problem: worm and wheel setups tend to have reasonable anti-backlash characteristics, but the inclusion of spur gears with no anti-backlash features was a major flaw. The first step was to eliminate this problem by removing the entire gearbox running tests with the disc sample mounted directly onto the servomotor. The specification of the servomotor allowed direct loading of the motor and off-axis loads, so risk of damaging the motor was minimal. This trial run showed that not only was the noise reduced significantly, but also the torque of the motor was satisfactory to run friction experiments without the need to boost power by using a gearbox. On discovering this, it was decided to discard the use of a gearbox completely, requiring some
modifications to the structure of the tribometer to accommodate the new layout.

By using this revised layout, the need to change gear ratios to perform tests over different velocity ranges was not required. This was especially useful when performing breakaway tests (see Chapter 5), where consideration of the ultimate velocity was not a precondition when defining the test protocol.

The new layout allowed the height of the motor/disc assembly to be adjusted relative to the force transducer. This was required as when the ice disc sample wore (or sublimed), the transducer was not located exactly horizontal, as was planned. Also, when changing samples to either Teflon® or the ice-on-rubber geometry, the transducer tended not to be horizontal. Providing an adjustable mount allows easy and accurate adjustment.

A second observation when testing without the gearbox was a high frequency noise emitted by the servomotor. This is due to resonance within the motor/encoder assembly. As the encoder is a digital device, it positions the motor in a stepwise fashion. If the motor is not in the exact position the control system commands, advancing or retarding the motor by a number of encoder steps corrects the motor position. If the servo control system is not perfectly set up, an oscillation can occur between “−1” and “+1” step when trying to set the position to “0” causing an audible squeal. The frequency of this oscillation depends on the torsional properties of the motor-encoder coupling, which in the tribometer resulted in a frequency of ≈ 9000 Hz. This is a higher frequency than the sample rates used to collect data, but by the Nyqvist sampling theorem* it would still affect readings to some degree, so a band-stop filter was put on this frequency to attempt to remove this superfluous noise.

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* Nyqvist (or Nyqvist-Shannon) sampling theorem states that analogue data at frequencies higher than 2B Hz (where B is the Nyqvist bandwidth) cannot be precisely reconstructed from digital data, but the effect of analogue noise at a higher frequency than twice the sampling rate exists but at a different frequency in the reconstructed digital data.
3.5.2 Force measurement
Noise in the force signal was caused from external sources, but the technology used to measure the forces (i.e. strain gauges) did not perform as required. Even with careful installation and the use of an operational amplifier to boost signals, the noise picked up from airborne and mains line sources was deemed too substantial. With less stiff arms (1.2 mm), the noise was tolerable at ≈ 10 %, but this meant excessive deflection of the pin, and resonance occurred readily.

It was decided that the only way to solve this problem without major changes to the design of the tribometer was to replace the dual cantilever design with a commercial load cell. Although the technology used with most load cells is also strain gauges, their compact design and manufacture to high tolerances produces a device that is much less prone to noise pick-up.

The load cell (Entran ELFS-T3M) was chosen based on its small size, versatile configuration and high performance at low temperatures. The latter of these was the main criteria, as most of the load cells looked at were not capable of being used in a cold, wet environment. Another reason for choosing this particular cell was the ability to specify the range of operating temperatures and forces. Entran assured that the load cell could be temperature-compensated for a particular temperature range, increasing the force data reliability. The temperature range was specified as 0 to –40 °C, while the load range was chosen as 25 N. Very small displacement (>0.013 mm, full scale) and a natural frequency much higher than the sampling range means that mechanical noise due to resonance is eliminated.

The tribometer was modified to accommodate the new load cell (see Figure 3.24) based on the criteria set in section 3.3.2, so on receipt of the load cell preliminary tests could be carried out to assess its performance. These tests showed excellent results, producing negligible mains line or airborne noise pick-up. A direct comparison of data obtained using the strain gauges on the dual cantilever and the load cell is shown in Figure 3.25.
Figure 3.24. Diagram of the tribometer with the Entran load cell fitted. A support arm onto which the load cell is mounted replaces the dual cantilever load cell. The other end of the load cell is mounted onto a redesigned pin sample holder and normal loads are applied directly onto this.

The load cell offers significant noise reduction over the strain gauges. Looking at the signal in Figure 3.25, between 0 and 1400 samples, where the velocity is 0 ms\(^{-1}\), the signal from the load cell shows noise amplitude of around 0.04 N, a 90% reduction on that of the strain gauges. During steady-state sliding, the strain gauges tended to pick up noise from the servo motor hence the exaggerated noise in the 2000 to 5000 samples range. The load cell is relatively unaffected by the motor, so variations in friction and terrain across the surface of the disc can be easily observed, indicated by the periodic signal recorded during sliding, something that is not possible using the strain gauge transducer.

Calibration of the load cell is straightforward, applying masses directly to the load cell by way of a specially designed mass carrier – linearity of the load cell
surpasses the specified 1% quoted on the data sheet, both in tension and compression.

Figure 3.25. Comparison of friction data obtained using the dual cantilever force transducer (strain gauges) and the Entran load cell. Both data sets were taken under similar conditions (-20 °C, 0.3 ms⁻¹ with normal load of 1.96 N). Note periodic waveform produced by the load cell trace indicates one revolution of the sample disc.

3.5.3 Vibration isolation

As well as noise picked up from electric sources, vibrations from the laboratory have detrimental effects on friction force measurement. These are typically low-frequency oscillations (below 10 Hz) and are caused by movement of people, machinery, etc. in close proximity to the laboratory. The oscillations were not particularly significant in magnitude, but the frequency range corresponded with those that were being measured on the tribometer (≈ 100 to 600 rpm of the sample disc) so isolation was essential.

Again, these signals can be filtered out at the data acquisition stage by means of a simple high-pass filter, but removal of the physical source is much better practice than through artificially filtering the resultant noise. The simplest solution to this
problem was by the use of bonded rubber isolating blocks between the 
measurement section and the ground (see Figure 3.26). These are specifically 
designed to isolate low frequency sources such as machinery, automotive 
components, etc. from users, and are reasonably efficient in doing so. This 
approach was also taken to the isolation of the motor from the measurement 
section.

Figure 3.26. The final layout of the tribometer. Additions to the design with regard to reducing 
vibrations are the rubber isolators and a large steel base block.

In conjunction with the rubber isolators, a large block of steel was placed between 
the ground and the tribometer (Figure 3.26). This provides a large inertial 
resistance to vibration, and with the compliance of the rubber isolators (vibrations 
are not transmitted efficiently, and so are damped out of the system) provides an 
excellent and inexpensive isolating platform.

Engineering drawings of all components are included in Appendix IV.
3.6 Verification

Once the modifications had been carried out and the tribometer was performing satisfactorily tests were carried out to determine the repeatability of results and accuracy in forward and reverse directions. The load cell was calibrated by the manufacturer in tension, although it is manufactured to perform equally in tension and compression. The only way to determine the dynamic performance of the cell was to run preliminary tests, and in order to monitor it was operating correctly it was tested at room temperature, out of the freezer.

Obviously ice could not be used, so it was substituted with a system with similar coefficients of friction that was stable at room temperature: stainless steel on Teflon®. Comparisons to absolute friction values are not made, as this is not the system under study, and the Teflon® was of unknown composition*, but the performance of the tribometer was of most interest.

The pin sample was a polished stainless steel ball bearing of 8 mm diameter. The disc was 80 mm diameter and 20 mm thick mounted directly onto the motor shaft by means of a steel collet and a worm screw. The surface of the disc was abraded with 800-grit silicon carbide paper and cleaned with dry tissue paper. The normal load was applied directly to the pin by masses of known weight and ambient temperature was 24 °C for all tests.

Normal load dependence tests were run at a constant velocity of $2.83$ ms$^{-1}$ by loading the pin incrementally up to a maximum of 5.9 N, then unloaded incrementally back to a minimum of 0.98 N. Velocity dependence tests were performed in forward and reverse directions from 5 rpm to 1000 rpm ($0.014 \leq v \leq 2.83$ ms$^{-1}$) in a random manner. The steel ball was cleaned after every run to remove the debris of Teflon® that had been picked up. Before, during and after the experiments, control readings at 0 rpm were taken to find the “zero” friction force reading in order to offset forces where necessary. It was found that the “zero” force did not change significantly during tests, so these zeroes were averaged and

* Teflon® is a trademark DuPont uses for the compound polytetrafluoroethylene (PTFE) and other similar compounds. DuPont lists coefficients of friction for Teflon® materials as ranging from 0.05 to 0.4, depending on composition (DuPont (2006)).
taken as the force base. The results of these tests are shown in Figure 3.27 and Figure 3.28.

For the initial few seconds while the disc accelerated from rest to its target speed the force trace increased with velocity until a plateau was reached at constant velocity. Running the disc for 15 seconds at a fixed speed and averaging the force trace over the range 5 to 10 seconds where the velocity was constant sampled the data.

Figure 3.27. Results of preliminary tests showing the dependence of the friction force on normal load of stainless steel on Teflon®. The ratio of friction force to normal load (and hence, the gradient of the $F-W$ curve) gives the friction coefficient (Amontons law) as $\approx 0.16$. The speed for all tests was $2.83 \text{ ms}^{-1}$.

As Teflon® does not change state under the low normal loads applied in these tests, the coefficient of friction would be expected to be constant over the given set of normal forces ($0.98 \leq W \leq 5.9 \text{ N}$). The data plotted in Figure 3.27 shows this behaviour. The data is linear and the compactness of data in the loading and unloading phases shows there is little hysteresis or non-linearity. The linear shape
shows there is no load dependency on the coefficient of friction, hence no bulk failure nor phase changes are taking place.

![Graph showing friction force vs sliding speed](image.png)

**Figure 3.28.** Results of preliminary tests using the tribometer at room temperature on stainless steel and Teflon®. Tight grouping of data in forward and reverse directions shows calibration is valid in tension and compression modes. The normal load was 1.96 N.

The tight grouping of data in both forward and reverse directions in Figure 3.28 shows that the load cell is calibrated in both directions. The difference in readings at \( \approx 1.2 \text{ ms}^{-1} \) can be regarded as anomalous, as it is the only point at which a significant discrepancy is shown. It also shows that the tribometer layout is not directional, i.e. it does not favour a particular direction, nor shows differing behaviour by reversing the sliding direction. The friction force measured at 2.83 ms\(^{-1}\) is 0.31 N, and as the normal load is 1.96 N, the resultant coefficient of friction is \( \approx 0.16 \), the same as was found over the entire load range in Figure 3.27.
3.7 Summary

Before experiments began, a suitable instrument had to be designed due to the budget and the conditions under which the instrument had to operate. A study of the devices on the market for tribology research and those designed and built by ice friction researchers was carried out.

Parameters based on previous work on rubber-ice friction were identified and design criteria set for the design of a custom tribometer. The comprehensive design process resulted in a fully operational prototype tribometer, which was consequently assessed and modified accordingly. Modifications included replacing an in-house built friction force transducer with a custom-built load cell from a specialist supplier due to the performance based on preliminary testing.

Verification experiments to assess performance of the tribometer were carried out using Teflon® and stainless steel. The results of these tests showed that the tribometer was capable of reproducible steady-state friction tests in forward and reverse directions and over a normal load range of up to 6 N. Absolute friction coefficients were in line with those values stated by a material manufacturer (DuPont (2006)).
Chapter 4.

Steady state friction

Both the friction of rubber and of ice is heavily dependent on speed and temperature at which the measurements are taken, so in the past the friction could be displayed as a function of each parameter, or as a transformed representation of two parameters, such as mastercurves for rubber. The method presented here displays the friction of the rubber-ice tribosystem over a range of speeds and temperatures on a map, showing simultaneously the friction of this system at a particular speed and temperature. LT-SEM was used to observe the wear surface of the ice samples and links between the wear surface morphology and measured friction are made. The mechanisms driving friction are related to both properties of the ice and of the rubber, dependent on the temperature-speed position on the map.

4.1 Introduction

The tribological pairing of rubber and ice is an extremely significant system with regards to motor vehicle safety and performance in cold climates. While rubber is regarded to have especially good frictional properties, ice is considered extremely slippery as it has a coefficient of friction ($\mu$) much lower than other crystalline solids due to lubricating melt water formed by frictional heating. $\mu$ of rubber sliding on dry, smooth surfaces can exceed 30 under specific conditions (Barquins and Roberts (1986)), while $\mu$ for ice can be less than 0.03 at temperatures near its melting point (Bowden and Tabor (1950)). The pairing of these two materials therefore produces a complex system that exhibits both high and low friction within specific temperature-speed zones on the friction map.

The friction of both ice and rubber has been shown experimentally to be dependent on temperature and speed. The friction of ice varies with speed (Barnes et al. (1971), Bowden and Hughes (1939), Evans et al. (1976)) and for sufficiently high
speeds ($>\approx 0.01\text{ ms}^{-1}$) frictional heating melts the ice at the interface to produce a lubricating layer of melt water (Bowden and Hughes (1939), Bowden and Tabor (1950), Evans et al. (1976)). The thickness of this layer increases with sliding speed, resulting in lower $\mu$ with increasing speed (Ahagon et al. (1988)). Ice melts more readily at temperatures close to its melting point, so $\mu$ decreases with temperature for a given speed (Bowden and Tabor (1950), Evans et al. (1976)). At low speeds and temperatures ($<\approx 0.01\text{ ms}^{-1}$ and $-10^\circ\text{C}$ respectively) the frictional heat generated is not sufficient to melt the ice to form a continuous lubricating layer. Under these conditions, deformation of asperities (Barnes et al. (1971)) and adhesion through sintering of ice resulting in asperity growth (Maeno and Arakawa (2004)) (or a combination of both processes (Kennedy et al. (2000))) govern the friction of ice.

The viscoelastic properties of rubber also change with temperature and speed, resulting in $\mu$ having temperature and speed dependencies (Grosch (1963)). Generally, when rubber behaves softest (at high temperatures or low frequencies), the ability to readily deform under normal loads allows very good adhesion to the counterfacing substrate. Conversely, if excessively soft compounds are used, the rubber has less cohesive strength, so can fail under modest loads resulting in low $\mu$ (Roberts (1981a)). At low temperatures (or high frequencies) the rubber is much stiffer, transferring loads more efficiently, but this stiffness produces less adhesion due to the reduced ability to deform; hence low $\mu$. The resultant of these conflicting behaviours is found by calculating the tan $\delta$ of the material by dynamic mechanical analysis (DMA). The maximum dissipation is indicated by a maximum tan $\delta$, the temperature point of which corresponds to the glass transition temperature ($T_g$) of the rubber, where internal friction is highest, producing the maximum $\mu$ (Grosch (1963), Tabor (1974)).

Friction curves of previous studies of the rubber-ice tribosystem exhibit a significant temperature dependency with $\mu$ for a fixed speed generally high ($\mu > 1$) below $-20^\circ\text{C}$ with a minimum ($\mu < 0.1$) close to the ice melting point at $0^\circ\text{C}$ (Gnörich and Grosch (1972), Roberts (1981c), Roberts and Richardson (1981)). There is also a pronounced speed dependency with low friction ($\mu < 0.3$) at both very low and high speeds with marked maxima of high $\mu$ at intermediate speeds.
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(Venkatesh (1975), Wilkinson (1953)) the position of which changes with temperature, normal pressure and the rubber compound type.

A useful method for looking at the frictional properties of materials with multiple parameter dependencies has been the production of friction maps (Lim (1998), Lim and Ashby (1987), Marmo et al. (2005)). These can show data of multiple parameter experiments in a clear form and are useful in identifying the zones of high and low friction in temperature and speed space. Other methods have been used to show multi-parameter friction dependencies, most notably that of mastercurves (Gnörich and Grosch (1972), Southern and Walker (1972)). Mastercurves are a representation of rubber viscoelastic properties defined as a function of the Williams-Landel-Ferry transform (Williams et al. (1955)), hence those produced for the rubber-ice tribosystem have generally avoided frictional melting of the ice, as this is a product of a different physical process and hence masks the viscoelastic contribution to friction. Friction maps, on the other hand, do not require such transformations or precautions to avoid frictional melting and present friction data in a much more robust fashion, clearly identifying areas of high and low friction throughout the range of parameters chosen.

The magnitude of the friction is a direct effect of the governing friction mechanisms. These mechanisms are difficult to observe using direct (optical microscopy (Roberts (1981c), Roberts and Richardson (1981))) or indirect (inspection using crossed-polarizers after wear (Norheim et al. (2001))) observation methods due to the small size of some of the features. Recent advances in low temperature scanning electron microscopy (LT-SEM) have allowed detailed indirect measurements and observation of friction-produced features on wear surfaces (Marmo et al. (2005)). Observation of these features allows insight to the frictional processes that produce them.

Friction experiments were performed over the range of temperatures –33.3 °C to –1.0 °C and speeds 0.003 ms\(^{-1}\) to 2.62 ms\(^{-1}\) and a friction map was produced by contouring the \(\mu\) values over the speed-temperature space. Ice samples within zones of significantly different frictional behaviour on the map were observed using LT-SEM to identify characteristic features that may uncover specific friction mechanisms.
4.2 Experimental

4.2.1 Sample preparation
Ice hemispheres attached to cylindrical custom made aluminium sample stubs were used for all tribology experiments. The ice was frozen directly onto the aluminium stubs from tap water at temperatures as close as possible to the temperature of experimentation. Water droplets of $70 \mu l$ in volume were placed into a 7 mm diameter by 1 mm deep bevelled recess in the stub using a syringe and allowed to freeze. Surface tension and expansion during freezing of the water droplets produced the domed profile of the ice samples. The domes were not perfectly hemispherical, but tended to have a peaked appearance, due to the expansion of water during freezing erupting and breaking the surface forming a conical peak (see Figure 4.1). The bevelled edge allowed the ice to remain locked in place during tribology experiments and during preparation for LT-SEM.

![Figure 4.1](image)

**Figure 4.1.** An elevation of the ice sample showing the aluminium sample stub with a bevelled recess to lock the ice in place during experimentation. The conical peak formed on the domed ice sample during sample preparation is due to water expansion breaking the hemispherical surface.

Images of the freshly prepared ice samples were taken using LT-SEM to observe the features present before they were worn. Two images of the samples are shown in Figure 4.2. The conical peak feature is clearly seen in these images on top of the predominantly hemispherical ice dome (Figure 4.2(a)). The base diameter of the cone is $\approx 2$ mm, relatively small compared to the 7 mm diameter of the dome. The
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The polycrystalline structure of the ice is also shown (Figure 4.2(b)). The grains are irregularly shaped, and sized between 50 μm and 500 μm in diameter. The surfaces of the samples were covered in small, euhedral frost crystals, something that could not be easily avoided, but it is not expected that these were significantly detrimental in any way to the friction measurements as they would quickly become worn away during the first stages of sliding.

Figure 4.2. LT-SEM images of a freshly prepared ice sample. (a) The conical peak can be seen protruding from the hemispherical body of ice, the base of which is indicated with a broken line. The surface of the ice is covered with small ice crystals, formed as frost between freezing and taking the images. (b) The second image shows a close-up of the ice surface showing clear grain boundaries (also, note the typical hexagonal structure of the frost crystals).

The substrate used in all experiments was cured (at 150 °C for 25 min) styrene butadiene rubber (SBR, serial number 325005) (with carbon black) disc with styrene levels of 31% and of 2 mm in thickness. The $T_g$ of the SBR was found to be about $-22$ °C by carrying out DMA at the representative parameters of 1 Hz frequency and 0.05 mm amplitude (DMA results are provided in Appendix V). The rubber was bonded to an aluminium disc of 70 mm diameter using epoxy adhesive (Araldite®) and the rubber surface prepared by abrading with 800-grit silicon carbide paper and wiping clean with tissue paper and a small amount of ethanol to remove the rubber debris. The rubber surface was routinely cleaned by the aforementioned method before each experiment to remove any excessive frost and debris that may have been produced.
4.2.2 Friction measurements
The tribology experiments consisted of applying a fixed sliding speed to the ice sample for a period of 15 seconds. The rubber and ice samples were in thermal equilibrium (± 1.5 °C) before the experiment was carried out. Running multiple series of isothermal (± 1.5 °C) experiments over the range of speeds generated data throughout the speed-temperature space: the individual speeds were set in a random order to eliminate wear patch size and temperature build-up effects. This is particularly important within zones of high dissipation where the heat produced, especially in the rubber, will tend to melt the ice at an artificially high rate if the experiments are carried out in progressively ascending/descending speeds. By performing the measurements in a random fashion, heat is allowed to be conducted away from the interface during runs with less dissipation of heat from the surface and a more true measurement can be expected. The friction trace became constant after the first 3 seconds of the experiment and friction forces were determined to be the mean force measured over this steady-state plateau.

As the ice samples wear over time, changing their contact area, and hence pressure, they were replaced routinely, the regularity dependent on the speed-temperature parameters. At high speeds and high temperatures the ice wore much more quickly, and needed replacing after only a very few runs. At low temperatures and low speeds, the sample was fairly slow wearing, therefore did not require replacement as frequently. It was found from preliminary experiments that the samples had limits of wear within which the results were very reproducible.

When a new sample was placed into the tribometer, it was worn for approximately 30 seconds at 200 rpm (0.52 ms⁻¹) after which it was sufficiently “conditioned” to have a high degree of confidence in the measurements. Once the wear patch had worn to approximately 2 mm² it was found that the measurements were no longer repeatable, so the ice sample was discarded and replaced with a fresh one. At high temperatures, the conditioning period was reduced accordingly, and less test runs were possible before a fresh sample was required.

A new, unconditioned ice sample was made for all wear surface LT-SEM observations. All samples used for LT-SEM observation were not conditioned, as
the process would have worn the samples excessively at high temperatures, therefore all were worn for the experiment duration of 15 seconds only.

4.2.3 Wear surface observation
After tribology experiments, the resultant wear surfaces were observed using a Hitachi S4700 LT-SEM fitted with a Gatan Alto 2500® cryo-preparation system. The ice samples were tested on the tribometer housed adjacent to the LT-SEM in the domestic chest freezer. Samples were immediately removed from the tribometer’s sample holder and placed in the dovetail specimen-carrier (see Figure 4.3) that slides into the LT-SEM stage. The specimen-carrier was stored in the freezer while experiments took place to ensure thermal equilibrium between ice sample and carrier.

![Figure 4.3. The capped specimen carrier with aluminium stub in position for observation of worn ice samples in the LT-SEM. The cap was required to transport the quenched ice sample to the cryo-preparation system without excessive build up of frost destroying the wear morphology of the ice surface.](image)

The temperature of the samples must be reduced to liquid nitrogen temperature before entering the LT-SEM. This has proved problematic, as frost from atmospheric water vapour immediately covers the sample. Usually, a sublimation process can be adopted to remove the frost, but obviously this is not appropriate for ice surface observations as sublimation would destroy the surface morphology.
To prevent extraneous frost formation contaminating the worn surface, the specimen carrier is fitted with a custom cap. The carrier cap covers the sample (and is big enough to ensure it does not come into contact with the wear surface) while it is quenched in liquid nitrogen and transported from the chest freezer into the LT-SEM. This was found to significantly decrease the formation of frost as the frost formed on the outside of the cap, preserving the features on the wear surface. Once inside the LT-SEM preparation system, and under vacuum, the samples were coated with a 60/40 gold/palladium alloy of 6-8 nm thickness.
4.3 Results

4.3.1 Ice-rubber friction map

Plotting each of the 225 tribology experiments in speed-temperature space and contouring for coefficient of friction values produced a friction map (see Figure 4.4). The coefficient of friction was contoured by interpolating the data using a mesh with 0.6 °C temperature spacing and 0.1 ms⁻¹ speed spacing.

![Friction map plot of the coefficient of friction (μ) of ice on SBR with sliding speed and temperature. Magnitude of μ indicated by colour bar and contour lines on map. Letters A – E indicate the position on the map of the samples observed with LT-SEM.](image)

Areas of high and low friction are indicated according to the colour bar to the right of the map, with the maximum μ (1.14) situated within the zone of high friction at 0.013 ms⁻¹ and around −25 °C and minimum μ (0.01) situated at 0.005 ms⁻¹ and −1.2 °C and at 2.62 ms⁻¹ and −1.0 °C. At high temperatures (> −7 °C), the coefficient of friction is independent of speed with contour lines indicating a friction coefficient of less than 0.2 for all but the very lowest of speeds. The zone
of high friction seems to have a strong temperature dependency with $\mu$ values falling rapidly either side of the maximum at around $-25\, ^\circ C$.

4.3.2 Wear surfaces

Wear surfaces were observed using LT-SEM on five representative ice samples from different zones within the friction map, and exhibiting different frictional behaviour. Fresh ice samples were produced before testing under the specific conditions selected based on the topography of the friction map. The wear surface is the area on the ice sample that is in intimate contact with the counter facing SBR sample during sliding. Presented are results from observations of these five samples (Samples A to E). The temperature, speed, coefficient of friction and area of contact patch worn are summarised in Table 4-I.

<table>
<thead>
<tr>
<th>Sample</th>
<th>temperature ($^\circ $C)</th>
<th>speed (ms$^{-1}$)</th>
<th>coefficient of friction, $\mu$</th>
<th>area of contact patch (mm$^2$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>$-5$</td>
<td>0.005</td>
<td>0.30</td>
<td>1.85</td>
</tr>
<tr>
<td>B</td>
<td>$-25$</td>
<td>0.005</td>
<td>0.85</td>
<td>0.33</td>
</tr>
<tr>
<td>C</td>
<td>$-33$</td>
<td>0.008</td>
<td>0.30</td>
<td>0.26</td>
</tr>
<tr>
<td>D</td>
<td>$-20$</td>
<td>2.1</td>
<td>0.35</td>
<td>1.1</td>
</tr>
<tr>
<td>E</td>
<td>$-5$</td>
<td>2.1</td>
<td>0.12</td>
<td>12.2</td>
</tr>
</tbody>
</table>

Table 4-I. Summary of experimental parameters for ice samples observed using LT-SEM.
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Figure 4.5. Sample A worn at -5 °C and 0.005 ms⁻¹. Direction of sliding indicated by arrow. (a) Large wear surface (1.85 mm²) still retaining peak feature created during preparation of ice sample. Large quantity of debris accumulated at leading edge of wear patch. (b) Peak feature deformed and protruding in direction of transport due to generation of melt water. (c) Frost grains entrained at leading edge are rounded and show significant sintering. (d) Wavelet features present on the ice correspond to abrasion pattern on the counterfacing rubber sample (see Figure 4.6).

The morphology of an ice sample (Sample A) has been produced at high temperature -5 °C and at low speed 0.005 ms⁻¹ and is shown in Figure 4.5. The wear surface has a circular shape with entrained debris at the leading edge, located at the top of Figure 4.5(a). The peak at the centre of the wear zone (Figure 4.5(b)) was formed before wear occurred while the sample was being prepared (see Figure 4.1), and its presence after wear has taken place indicates that at this temperature the rubber was acting resiliently and was able to deform around the protrusion. The trailing edge of this peak shows signs of deformation through the flow of melt water causing an extruded plane in the direction of transport (Figure 4.5(b)). The rounded morphology of the surface is evidence that a relatively thick layer of water formed through frictional heating of the ice surface. The rounding and high degree
of sintering of entrained particles at the leading edge is an indication of extensive frictional heating during wear (Figure 4.5(c)). Wavelet features on the surface (Figure 4.5(d)) are believed to be formed as a consequence of the partially worn rubber sample passing over the surface of the ice. Typical abrasion patterns (Iwai et al. (2005), Muhr and Roberts (1988)) of similar wavelength are observed also on the rubber wear surface (see Figure 4.6).

![Image](image.png)

**Figure 4.6.** LT-SEM image of a worn rubber sample. Note the abrasion pattern of wavelets perpendicular to the direction of transport, indicated by the arrow. The spacing of the waves correspond quite closely to the spacing of the wavelet features observed on the counterfacing ice sample in Figure 4.5(d).

The wear surface on Sample B (Figure 4.7) was produced within the zone of very high friction, and close to the point of maximum friction on the friction map, at -25 °C and 0.005 ms⁻¹. There is an accumulation of debris at the leading edge (at the top of Figure 4.7(a)) of the wear surface. The patterning within the wear surface has a porous texture most likely formed by deformation of entrained ice grains. The smeared, multi-layered features shown on Figure 4.7(b) demonstrate successive layers of this slurry mixture being deposited at intervals during sliding and indicate deformation of entrained ice debris. The entrained ice grains at the leading edge are rounded in appearance (Figure 4.7(c)). The worn surfaces of these entrained grains at the leading edge have the same features as the main wear surface, making it difficult to define the boundary between debris pile and wear surface. There is no indication of an extensive or continuous lubricating liquid
water layer, hence the high friction associated with this position on the friction map.

Figure 4.7. Sample B worn at −25 °C and 0.005 ms⁻¹. Direction of sliding indicated by arrow. (a) Small wear surface (0.33 mm²) with debris accumulation at front and sides of surface. (b) Worn material is deformed and shows a porous texture — material deposited at intervals having the same wear features. (c) Entrained frost grains at the leading edge show the same patterning as on the main wear surface.

Sample C (Figure 4.8) was tested at low temperature (−33 °C) and at low speed (0.008 ms⁻¹), exhibiting lower friction than Sample B. The transport direction is upwards in Figure 4.8(a) with the entrained frost crystals gathered at the leading edge, shown at the top of the figure. Wear grooves are clearly observed on the wear surface running parallel to the direction of transport (Figure 4.8(b)). The wear morphology is principally formed by the deformation and fracture of the ice with overlying layers of ice crystals drawn into the wear surface at the leading edge, shown by the small angular fragments in the debris and micro cracking of the
surface (Figure 4.8(c)). These layers can be observed by looking at the trailing edge of the wear surface where stacked strata of plastically deformed material are extruded in tongue like sheets (Figure 4.8(d)). This formation suggests that wear has taken place through shearing of solid ice and not through melting and water transport. As the wear has taken place with the rubber below its \( T_g \), the rubber behaves like a stiff glass and hence its ability to deform is diminished resulting in lower friction than measured on Sample B.

Figure 4.8. Sample C worn at -33 °C and 0.008 ms\(^{-1}\). Direction of sliding indicated by arrow. (a) Small wear patch (0.26 mm\(^2\)) with debris accumulation at front and sides of patch. (b) Wear grooves running parallel to direction of transport. (c) Micro cracking and scuff features along flanks – indicative of deformation and fracture of the ice. (d) Debris deformed in tongue-like sheets at trailing edge of the wear surface.
Sample D (Figure 4.9) is a sample tested under high speed (2.1 ms⁻¹) and relatively low temperature (−20 °C) conditions. The features shown on these images are very similar to those of ice wear on a hard substrate (steel) (Marmo et al. (2005)), explained by the hard behaviour of rubber at low temperatures and high frequencies (and speeds) (Mark et al. (1994)). Wear striations are clearly visible on the surface in the direction of transport (Figure 4.9(a)). Thin, flat layers are present on the surface indicating melt water being swept back in the direction of transport and deposited by solidification at intervals (Figure 4.9(b)). The wear surface shows signs of melting with “sausage-shaped” ridges present, as identified in Marmo et al. (Marmo et al. (2005)) as being formed by refreezing of melt water (Figure 4.9(c)). There is a distinct boundary around the wear surface with frost debris entrained at the leading edge, indicating melting is limited to a thin layer at
the interface and not transferred to the surrounding debris. The trailing edge (Figure 4.9(d)) has a cohesive mass of plastically deformed ice, formed through the removal of material at the surface and deposited at the trailing edge.

Figure 4.10. Sample E worn at -5 °C and 2.1 ms⁻¹. Direction of sliding indicated by arrow. (a) Significant melting producing large flood plane (12 mm²) with a meniscus at leading edge. (b) Trailing edge of flood plane refreezing in successive stacked plates. (c) New refrozen structure produced with large rounded features. Large grains shown are typical for a slow-freezing system.

Sample E (Figure 4.10), worn at high temperature (-5 °C) and high speed (2.1 ms⁻¹), shows the significance of frictional heating in generating a significant amount of melt water at temperatures close to the melting point of ice. The debris deposited at the trailing edge has been swept back a long distance beyond original contact surface (Figure 4.10(b)), indicating that a significant amount of water has been melted through frictional heating processes. A meniscus clearly observed at the leading edge confirms the large amount of melt water present (Figure 4.10(a)).
Grain sizes of refrozen water are significantly bigger (20 μm to 100 μm) than those previously observed for slower speeds (Figure 4.10(c)). This may be due to heat built up in the rubber disc allowing the ice to refreeze at a slower rate than has occurred at slower speeds, or more likely, the thickness of the melt water layer is significantly greater than at slow speeds, hence the greater thermal mass of water present takes longer to freeze leading to the larger grains observed. The melt water debris has refrozen and been deposited as stacked plates at the trailing edge showing deposits have been made at intervals during the experiment (Figure 4.10(b)).
4.4 Discussion

4.4.1 Comparison to other steady state friction measurements

Direct comparisons to previous studies of rubber on ice friction is difficult given the varying properties of different rubber compounds and the dependence of friction on these properties and the different test methods used by the researchers. Figure 4.11 shows a comparison of two selected sets of our $\mu$-temperature data to those found by three studies undertaken with comparable conditions (Conant et al. (1949), Roberts (1981b), Wilkinson (1953)). Parameters from each study are summarised in Table 4-II.

![Figure 4.11. Comparison of the single-speed data with respect to temperature to those found by previous rubber-ice friction studies (Conant et al. (1949), Roberts (1981b), Wilkinson (1953)).](image_url)
The lubricating layer of water due to frictional heating produces low friction on ice and as this is dependent on pressure, speed and temperature, therefore differences between studies are expected. Two sets of data from this work plotted in Figure 4.11 at different speeds (0.01 m s^{-1} and 0.13 m s^{-1}) but identical pressure and temperature shows how the friction can contrast. Comparisons to data from (Gnörich and Grosch (1972), Southern and Walker (1972)) has not been made due to the specific techniques (in situ polishing of the ice surface to remove frost and avoiding frictional melting) used to find large friction values ($\mu_{\text{max}}$ was 3.5 and 4 respectively). While the data we compare to were not produced under identical conditions, the comparison is good for all data from 0 °C to around −15 °C. Below −15 °C there is quite a large deviation between data. All data rises with falling temperature (until \approx −15 °C) from a minimum at 0 °C. Data from studies by Conant et al. (Conant et al. (1949)) and Roberts (Nitrile) (Roberts (1981b)) compare favourably with those here in both magnitude and trend, showing a rise to an apparent maximum followed by a fall in $\mu$ with decreasing temperature. Data from Roberts (Isoprene) (Roberts (1981b)) and Wilkinson (Wilkinson (1953)) also rise with falling temperature, but do not exhibit the maximum within the temperature range reported.

The local maximum is due to the glass transition temperature of the rubbers used in the friction experiments. The source of this high friction is the highly dissipative
nature of rubber due to the competing behaviour of viscous ($E''$) and elastic ($E'$) components at $T_g$ (Grosch (1963), Mark et al. (1994)). This means if $T_g$ lies within the temperature range studied, a $\mu$ maximum is expected close to $T_g$ with $\mu$ values falling either side of this temperature point. If $T_g$ lies below the range of temperatures studied, then a rising $\mu$ trend would be expected with falling temperature. This explains the differences in trends shown in Figure 4.11, of the studies by Roberts, noting the $T_g$ values for the rubbers in Table 4-II and the presence of a peak for nitrile rubber ($T_g = -25 \degree C$) and not for isoprene ($T_g = -67 \degree C$).

The speed dependence shown in results from tribology experiments compares favourably with the literature (Evans et al. (1976), Southern and Walker (1974)). The friction measurements were carried out at isothermal intervals, controlling the sliding speed and executing the individual constant speed measurements in a random order. This method was required to reduce the effects of heat build up and differences in surface area in the samples over consecutive tests. By doing this, the speed dependent (normal pressure independent) behaviour of the tribosystem was examined for different temperatures.

Referring to Equation 6 from the literature review, the thermal control model

$$\mu \propto \frac{1}{\sqrt{v}}$$

proposed by Evans et al. (Evans et al. (1976)) holds for the friction data measured within this study (above $v = 0.01 \text{ ms}^{-1}$). Isothermal ($\pm 1.5 \degree C$) data has been transformed according to this model and plotted in Figure 4.12, the linear function of each line is given in the form

$$\mu = \frac{A}{\sqrt{v}} + B$$

where $A$ and $B$ are constants, dependent on temperature.
Figure 4.12. Coefficient of friction ($\mu$) plotted against the reciprocal square root speed ($v^{-1/2}$) for (top) all isothermal friction data ($\pm 1.5$ °C), and (bottom) data at $-2.8$ °C and $-22$ °C ($\pm 1.5$ °C) showing the linear fit functions for the data, indicated adjacent to the respective data set.

The slope of each line $(A)$ depends on the temperature at which the data was collected, i.e. for data collected at $-2.8$ °C, the slope was $0.014 \text{ m}^{1/6} \text{ s}^{-1/2}$, while for $-
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22 °C it was 0.115 m\(^{1/6}\) s\(^{-1/6}\). The data plotted in Figure 4.12 is for speeds above 0.01 m s\(^{-1}\), below which the data falls away from the linear trend shown to a lower value than predicted by the thermal control model. This speed value (0.01 m s\(^{-1}\)) corresponds to the speed suggested by Bowden et al. (Bowden and Hughes (1939), Bowden and Tabor (1950)) at which frictional melting is initiated, below which no melting occurs by frictional mechanisms alone (but this value will change depending on the normal load applied to the system (Akkok et al. (1987))). From this assumption, it is clear that the friction values measured below this speed are predominantly driven by rubber-ice friction properties, and not melt water lubrication. Hence, the thermal control model is only valid for rubber-ice friction within zones in which the dominating friction mechanisms are melt water-property based, as is true of the systems on which the model was developed.

![Figure 4.13](image-url)

**Figure 4.13.** Plot of \(A\) (the slope of the \(\mu - v\) plots in Figure 4.12) against temperature compared to \(\tan \delta\) (at 1 Hz, 0.05 mm amplitude) data for the rubber sample (325005) as measured using DMA.
It was also found that the slope of each line was closely related to the dynamic properties of the rubber (tan $\delta$ data) collected when performing DMA on the samples. A plot of $A$ and of tan $\delta$ against temperature is shown in Figure 4.13.

When plotted in this form, it is clear that the trend shown by the parameter $A$ is similar to that of tan $\delta$ with temperature. With reference to earlier work by Evans et al. (Evans et al. (1976)), it is suggested that the differences in slope of $A$ are due to the different thermal conductivity of the materials used for their experiments. It could be suggested that the trend shown here may be due to a change in thermal conductivity (particularly within close proximity to the surface layer) with temperature if this system follows the thermal control model fully. A higher thermal conductivity would allow heat to be removed from the interface at a higher rate, maintain a lower net temperature of the ice surface, and hence produce less melt water. It has been shown some time ago (Schallamach (1941)) that the thermal conductivity of rubber does change with temperature within this temperature range, the dependence being monotonic and decreasing with temperature. Furthermore, the change is not substantial at less than 10% within the temperature range studied here (Schallamach (1941)). Although we cannot exclude the conductivity change influence on the behaviour, the non-monotonic relationship points to a different direction.

Below the minimum speed for which the thermal control model seems to be valid, the dependence on the rubber properties is clearly observed in Figure 4.4. Dependencies on deformation and adhesion components have been found previously to depend on the loss modulus ($E''$) and loss factor (tan $\delta$) of the rubber respectively (Grosch (1963)). For the lowest speeds shown on the map the maximum friction has been measured, in accordance with the thermal control theory for ice, but the temperature position of the highest friction contours are dependent on viscoelastic properties of the rubber. For the most part, the contribution of $E''$ can be assumed to be negligible as the ice sample did not have a characteristic roughness, or asperity spacing and as such can be regarded as a smooth surface. The relationship of the friction to tan $\delta$ can therefore be considered the most significant, as this is the adhesion component and has been
identified to be of most importance when sliding on smooth surfaces (Grosch (1963)).

This clearly seems to be the case. When comparing the DMA data (Figure 4.13) to that shown on the friction map (Figure 4.4), the clear tan δ peak is mirrored on the map, especially at low speeds. As well as at the very lowest speeds, it is of importance higher up the y-axis, but with diminishing influence as the speed increases and frictional melting becomes the dominant mechanism. At temperatures below $T_g$, the combination of the rubber behaving more stiffly and with less dissipation results in a general fall in friction with distance from the peak. Above $T_g$, the rubber is softer, which should lead to higher friction, but with the readiness of the ice to melt with increasing temperature, the system exhibits lower friction.

As shown, the friction between rubber and ice has essentially a dependency on heat generation, therefore multi-parameter studies have been used to define the behaviour. Unlike mastercurves, the friction map displays real data and shows the prevalent friction levels produced over a wide range of temperature and sliding speed. No transforms are used to deduce the estimated coefficient of friction, and mathematical models valid exclusively for both ice and rubber and covering different frictional mechanisms are represented on one, clear diagram.

4.4.2 Wear surface morphologies
The wear mechanisms observed using LT-SEM on the ice samples can be related to the position of the sample on the friction map. Melt water produced by frictional heating during wear of Samples A (Figure 4.5) and D (Figure 4.9) causes sufficient lubrication to make $\mu$ around 0.3. Although these samples were worn in different zones of the friction map, the combination of ambient temperature and speed produces the heat necessary to melt a significant lubricating film.

Sample E (Figure 4.10) was worn in an area of the map producing highest frictional heating, with high ambient temperature and speed. The LT-SEM images of this sample clearly show the extensive melting of the sample to produce a large flood plane, lubricating the interface significantly to lower $\mu$ to 0.12.
Sample B was worn in the zone of maximum friction at $-25 \, ^\circ C$ and $0.005 \, \text{ms}^{-1}$ and exhibits very high $\mu$, compared to the others, of 0.85. Lowest frictional heat is generated at lowest speeds, shown by the highest $\mu$ observed at the base of the map (Figure 4.4). At these speeds (given a sufficiently low temperature) the heat is not significant enough to produce a continuous lubricating layer, so ice behaves like a dry crystalline solid material. This lack of continuous lubrication revealed by the LT-SEM images in Figure 4.7 indicates that there is a link between the high friction in this zone of the map and the properties of the SBR close to its $T_g$. The peak in $\mu$ is only significant at low speeds because as speed increases, the higher dependency on rubber properties is lost due to lubrication, as observed on Sample D (Figure 4.9).

As shown with Sample B, at low speeds and temperatures, there is no lubricating film apparent, and the same is evident when looking at Sample C. The morphology suggests deformation of entrained frost particles and features such as scuffing and micro cracks typical of dry friction. The low friction measured in this area of the map is due to the SBR behaving more like a glassy solid, losing its ability to deform around asperities effectively at temperatures below the $T_g$ and resulting in a lower $\mu$ of around 0.3.

Wavelet features such as those observed on Sample A (Figure 4.5(d)) correspond in wavelength to abrasion features present on the surface of the rubber sample (Figure 4.6). Patterning on both surfaces may be an indication of a peeling process such as Schallamach waves (Roberts (1981c), Schallamach (1970)), but is more likely to be a result of wear of the rubber. Rubber wear features such as these are well known features associated with crack propagation in the rubber as it deforms during sliding, and is the main wear mechanism of rubber materials (Iwai et al. (2005)). The corresponding wavelets observed on the ice are probably caused due to the motion of the rubber over the ice, or an impression imparted as it refreezes. This is an interesting observation, but whether this process contributes significantly to friction is a matter for further investigation.
4.5 Conclusions

A friction map was produced to identify friction regimes dependent on speed and temperature. Coefficient of friction values within the map range over two orders of magnitude from 0.01 to 1.14.

A distinct zone of high friction was measured for low speeds around -25 °C. This is due to the proximity of its glass transition temperature, measured to be around -22 °C using dynamic mechanical analysis, hence the highly dissipative nature of the rubber was displayed. Samples examined using low temperature scanning electron microscopy from the high friction zone show no evidence of a continuous, lubricating fluid film. The wear morphology has a porous texture with evidence of shear deformation of ice debris indicating friction derived from adhesive forces between ice and rubber.

Low friction measured at high temperatures and high speeds are due to the presence of a lubricating layer of melt water produced by frictional heating. Morphology features such as small, refrozen ice crystals in the interface layer are indicative of this liquid water layer being produced during sliding.

Samples worn below the glass transition temperature show evidence of deformation through mechanical shear of ice debris. This indicates the rubber’s inability to deform as at higher temperatures, hence acting as a hard substrate resulting in lower friction than measured at the glass transition temperature.

Data obtained through steady state friction measurements fit those produced by previous studies of rubber and ice tribosystems. Above a threshold sliding speed of 0.01 ms\(^{-1}\) the friction of ice on rubber fits the thermal control model developed for ice friction systems. Below this speed, the system exhibits friction mechanisms dependent closely on the dynamic behaviour of the rubber. Friction data fitting both mechanisms are present on the friction map.
Chapter 5.

Breakaway measurements

The measurements presented in this chapter were taken using the tribometer in a dynamic regime, increasing the tangential force to the SBR samples in static contact with the ice until breakaway occurred and hence sliding began. It was found that by increasing the rate of tangential force, greater breakaway forces were measured. It is expected that this phenomenon could be incorporated into traction control as a means to generate extra traction (DTFP) on icy surfaces.

5.1 Introduction

The friction of rubber on ice has significant importance within regions of cold and temperate climate especially with regard to vehicle safety and performance. While both laboratory and field studies have been carried out using this system for steady-state research, little has been addressed with regard to parameter transients.

It has recently been shown experimentally that a significant rate-dependent friction force exists between vehicle tyres and ice surfaces for abrupt transients in motion (Ivanović et al. (2005)). For a transient in force rate (or torque) a maximum friction force is generated corresponding to the transition of the rubber from the static to the sliding friction regime, or breakaway force. Tests performed with model vehicles have shown that this rate-dependent dynamic tyre friction potential (DTFP) can amount to more than two times the peak value of the (low rate) static friction if an abrupt force transient is applied (Ivanović et al. (2005)). The DTFP has been shown to have a logarithmic growth dependency on the force (torque) transient rate.

In the study presented here this tyre behaviour has been extended to small-scale samples to analyse the behaviour parametrically in a more controllable
environment. Research in the field of steady state rubber on ice friction have found dependencies on individual parameters such as temperature (Bowden (1954), Conant et al. (1949), Gnörich and Grosch (1972), Roberts and Lane (1983)) and normal pressure (Southern and Walker (1972), Venkatesh (1975)). Reviews of time-dependent friction between other materials (steel on steel (Olsson et al. (1998), Richardson and Nolle (1976))) have found that the parameters of most importance are the rate of tangential loading and stationary contact time. Significantly higher breakaway forces were measured at longest contact times and at lowest rate of applied force. All of these parameters were postulated to have an effect on the breakaway force due to material properties so are included in this study.

To measure this time-dependent frictional force, breakaway tests were devised. These experiments by definition include a period of static contact followed by a tangentially applied force until sliding is initiated. The initial conditions of the test could determine the magnitude of the breakaway force, as sliding does not occur before the maximum force is reached. Hence, we could assume that effects discussed in past work (unrelated to breakaway testing) such as adhesion and mechanical keying play a major role in the breakaway force (Barnes et al. (1971), Bowden (1953), Maeno and Arakawa (2004)).

The dependence of adhesion on dwell time can be explained by the theory of plastic deformation and sintering of asperities under normal loading to increase the real area of contact. Adhesion is proportional to the real area of contact, and dependence on temperature, time, pressure and contaminants has been widely accepted (Maeno and Arakawa (2004)).

The mechanical keying component of the breakaway force can be divided into two distinct parts: micro keying and macro keying. Macro keying is the degree to which the rubber sample sinks into the ice surface, the displaced ice providing a “wall” to resist motion. Micro keying occurs at the interface where the material in contact deforms at the asperity scale to take the profile of the other. If the hardness properties are significantly different between sliders this can also cause “ploughing” of the softer material by harder asperities and gives rise to an
additional, micro mechanical force (Barnes et al. (1971), Bowden (1953), Maeno and Arakawa (2004)).

Furthermore, as we have seen in Chapter 4 the behaviour of the rubber-ice tribosystem at low velocities is greatly affected by the viscoelastic properties of rubber. Since for the breakaway experiments we have an initial regime of very low (nominally zero) velocities we expect that the rubber properties will play a significant role as well.

This particular aspect of frictional behaviour is of significant interest in the vehicle control systems field, where the contribution of occasional steady state friction is most likely diminished by the more significant role of the dynamic friction potential.
5.2 Experimental

For all previous experiments, the tribometer was used in steady state mode, i.e. constant speed for a fixed period of time. The breakaway tests required the tribometer to run in a transient fashion, starting at standstill and accelerating at a known rate until sliding friction commenced. Initially, it was decided that feedback from the load cell would be used to control the motion of the drive system to allow a linear increase in friction force with time. Preliminarily tests showed that this was not realistic, due to the combination of noise in the friction force signal and instability at low speeds, so an alternative method was devised.

This method uses the experiment time data in an expression that produces an increase in friction force with time. Ideally, a linear force increase with time was preferable, but due to the constraints of the motion control system, this was not achievable. The force control used for experiments takes the form of a parabolic function, increasing the rate of force linearly with time, or

\[ F = \frac{a}{2} t^2 + c \]

where \( F \) is the friction force, \( t \) is time and \( a \) and \( c \) are constants. The constant \( a \) is the programmed parameter for the force rate \( (dF/dt) \), and \( c \) is 0, indicating that the breakaway test starts from rest for all tests. Explanations for the use of this function can be found in Appendix VI. By using this method, the applied force rate to the slider can be controlled, but this rate is indicative, and the actual force rates at the breakaway point are directly calculated from the experimental \( F-t \) curve during analysis of the data.

The tests were performed by applying the increasing tangential force with time to the rubber sample resting on ice until sliding (breakaway) was initiated (see the schematic diagram in Figure 5.1). Breakaway forces \( (F_{\text{max}}) \) were determined to be the maximum force exhibited within the breakaway period. \( t_{\text{dwell}} \) is the time the rubber sample rests on the ice before any tangential force is applied, while \( t_{\text{lag}} \) is the time from when force is applied to the time breakaway occurs. The velocity profile is the controlled parameter, therefore the motion system tries to follow the velocity.
profile programmed by the motion control software (NI-Motion®). The shape of this profile is parabolic to replicate the increasing friction force, but as breakaway occurs, the difference between the required and real positions results in an overshoot, shown as the spike in the velocity trace. The base of this spike indicates that sliding has commenced, so this is used as the trigger from which $t_{lag}$ is calculated.

Figure 5.1. Schematic diagram of breakaway test protocol showing controlled force rate ($dF/dt$) and corresponding disc speed with time. Note definitions of: breakaway force ($F_{max}$), dwell time ($t_{dwell}$), and lag time ($t_{lag}$).

The experiments were carried out using two protocols. Protocol I consisted of applying a controlled force rate after a constant initial dwell time of 12 seconds. The schematic diagram shown in Figure 5.2 explains the terminology and how the parameters chosen are related. Protocol I was also used for dwell time ($t_{dwell}$) dependency experiments. A range of times between 11 and 300 seconds were used, at a constant $dF/dt$ of $7.5 \pm 1$ Ns$^{-1}$. 
Figure 5.2. Schematic diagram of breakaway force ($F_{\text{max}}$) with time for a typical test using Protocol I. The initial dwell time ($t_{\text{dwell}}$) is fixed for all force rates before force is applied. The total breakaway test time ($t_{\text{breakaway}}$) is determined by the sum of $t_{\text{dwell}}$ and the rate-dependent lag time, $t_{\text{lag}}$, i.e. $t_{\text{breakaway}}$ for rate1 is $t_1$ and $t_{\text{breakaway}}$ for rate2 is $t_2$. Note $t_{\text{dwell}}$ is constant for all force rates.

Protocol II is a modification to Protocol I to allow the time in stationary contact ($t_{\text{breakaway}}$) between the SBR and ice to be (approximately) the same, hence any dependency on this parameter can be decoupled from the force rate dependency. To do this, the period of dwell ($t_{\text{dwell}}$) before force application has to be controlled so that tests using high force rates (with inherently shorter lag times) and those using low force rates (with longer lag times) breakaway at similar experiment durations. Figure 5.3 shows a schematic diagram of the methodology.

The controlled $t_{\text{dwell}}$ was calculated by running preliminary tests to determine the dependency of $t_{\text{lag}}$ on force rate based on the range of proposed force rates. A fairly consistent non-linear trend was found, so the breakaway tests were run over the range of force rates controlling $t_{\text{dwell}}$ to provide a breakaway time of approximately 15 seconds. Those tests that fell outside the $t_{\text{breakaway}}$ range 15 ± 3 seconds were rejected.
Figure 5.3. Schematic of breakaway force ($F_{\text{max}}$) with time for a typical test using *Protocol II*. The dwell time is controlled before the corresponding force rate is applied, i.e. the controlled dwell time is $t_2$ for a test using force rate, $\text{rate2}$. Note $t_{\text{breakaway}}$ is constant for all force rates.

Experiments were carried out using cylindrical samples of cured (150 °C for 25 min) styrene butadiene rubber (SBR, serial number 325003, styrene 23 % wt) mixtures containing carbon black of 10 mm and 4 mm diameter and 2 mm in height, lightly buffed using 800-grit silicon carbide paper and bonded to an aluminium stub of the same diameter. From dynamic mechanical analysis of the SBR samples, it was found that $T_g$ of this rubber was approximately −31 °C (full dynamic mechanical analysis data is provided in Appendix V), outside but close to the range of temperatures studied here. The temperature dependency on the rubber dynamic properties falls rapidly either side of this peak at $T_g$, therefore the nonlinearity of the rubber properties is reduced in the temperature range under investigation but cannot be excluded.

The ice was made from tap water frozen in an aluminium dish of 68 mm diameter and 5 mm deep, and was at least 24 hours old. The dish (see Appendix IV) was designed so that the ice was keyed to the dish and no movement was possible, even if a thin layer of water was present between dish and ice. Initially, the ice was machined flat using the specially designed cutting tool then a thin uniform layer of
the surface was melted with a hair drier to prepare the surface. The “glare” finish was then removed by light abrasion using P800 grade silicon carbide paper in the direction of motion used during experiments and the ice debris removed by wiping with dry tissue paper. As a final step, the SBR samples were run in steady state sliding contact on the ice at a constant speed of 200 rpm for 20 minutes while applying a normal load of 400 g at temperatures below −15 °C. This was found to improve repeatability by applying a “conditioning” to the ice. Before each test run commenced, the rubber element was cleared of ice debris using dry tissue paper.
5.3 Results

5.3.1 Protocol I

Temperature

Figure 5.4 shows how the breakaway force changes with temperature for a range of applied force rates from 1.7 Ns\(^{-1}\) to 28.7 Ns\(^{-1}\). Both sets of data were carried out under identical conditions with two samples of the same material using Protocol I. While Set 2 measurements were taken over the full force rate range, Set 1 was restricted in force rate to the range 3.7 Ns\(^{-1}\) to 7.1 Ns\(^{-1}\).

![Breakaway force with temperature for 10 mm SBR samples using Protocol I. Sets 1 and 2 were carried out using different samples of the same material. Set 1 has relatively restricted force rates (3.7 Ns\(^{-1}\) to 7.1 Ns\(^{-1}\)) compared to Set 2 (1.7 Ns\(^{-1}\) to 28.7 Ns\(^{-1}\)).](image)

In general there is a gentle rise in the breakaway force with decreasing temperature from 0 °C to around −20 °C, then a fairly flat plateau of maximum breakaway forces is observed until the lowest temperature tested (−29.7 °C). Set 1 displays overall slightly lower breakaway forces than Set 2, but still has the general rise in breakaway force with falling temperature trend.
**Force rate**

Figure 5.5 shows how breakaway force varies with applied force rate for four temperature ranges. Again, two data sets are plotted taken using two different samples of the same materials under identical conditions using Protocol I.

The data has been split up into the temperature ranges: $-1.3 \, ^\circ C$ to $-10 \, ^\circ C$; $-10 \, ^\circ C$ to $-15 \, ^\circ C$; $-15 \, ^\circ C$ to $-20 \, ^\circ C$; and $-20 \, ^\circ C$ to $-29.7 \, ^\circ C$. In general, the lowest $F_{\text{max}}$ are produced in the range $-1.3 \, ^\circ C$ to $-10 \, ^\circ C$, while the highest forces are produced in the range $-20 \, ^\circ C$ to $-29.7 \, ^\circ C$.

In addition, the highest breakaway forces produced are by the highest $dF/dt$. This is true for all temperature ranges. Within the parameters measured, the $F_{\text{max}}$ against $dF/dt$ data tends to fit a logarithmic trend with the curve steepest at low $dF/dt$ and becoming less steep with increasing $dF/dt$. 

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**Figure 5.5.** Breakaway force against force rate for four temperature ranges using 10 mm SBR samples and Protocol I. Data sets are the same as those shown in Figure 5.4. Points are individual breakaway tests and placed according to their test temperature, with indicative logarithmic trend lines fitting both data sets.
Dwell time

Results shown in Figure 5.6 clearly show the relationship between the time in stationary contact and the corresponding breakaway force generated at constant temperature (−20 °C) and $dF/dt$ (7.5 Ns$^{-1}$) using a rubber sample of 10 mm diameter. Samples were left to dwell on the ice surface for 12, 60 and 300 seconds and a breakaway test performed as described previously using Protocol I.

![Figure 5.6](image)

Figure 5.6. A plot of $F_{\text{max}}$ against $t_{\text{dwell}}$ for 10 mm diameter SBR samples on ice using Protocol I. Temperature and $dF/dt$ are constant at −20 °C and 7.5 Ns$^{-1}$ respectively. Individual points plotted as mean and error bounds shown as variance of 10 test runs per fixed dwell time.

A logarithmic trend is clear (of the form $F_{\text{max}} = 1.6 \log (t_{\text{dwell}}) + 0.13$) with $F_{\text{max}}$ values rising from ≈ 2 N at $t_{\text{dwell}} = 12$ s to ≈ 4 N at $t_{\text{dwell}} = 300$ s.

This trend shows the clear dependency of $F_{\text{max}}$ on the time in stationary contact of the SBR and ice samples. The inherent coupling effect of $t_{\text{lag}}$ with changing force rate means a contribution to $F_{\text{max}}$ is an experimental effect of (rather than a dependency on) the $dF/dt$ chosen. Figure 5.7 shows the variation of $t_{\text{lag}}$ with $dF/dt$ and hence the extent of this contribution in experiments using Protocol I.
For these experiments, at small force rates, $t_{lag}$ was reaching in excess of 15 seconds, and for high force rates, was in the region of 3 seconds. This assumes that although a force is being applied to the sliders, the interfaces are in relative stationary contact ($\pm 0.15$ mm), therefore this period ($t_{lag}$) is in effect an extension to the dwell time.

![Graph](image)

Figure 5.7. Plot showing the coupled nature of $t_{lag}$ with $dF/dt$ using 10 mm SBR samples and Protocol I. $t_{dwell}$ before force application is 12 seconds for all measurements. All tests carried out in the temperature range $-10 \degree C$ to $-22 \degree C$. (Line is indicative of trend.)

While Protocol I is useful to show the effects of both fixed rate and $t_{dwell}$ experiments, the nature of the protocol means that the breakaway time is varying with rate, which is an unwanted effect and might lead to overestimation of breakaway force at low force rates, so the actual dependency on force rate might be more abrupt than shown in Figure 5.5 for a given temperature range. To show the dependency of $F_{max}$ on $dF/dt$ alone (decoupled from $dF/dt$-related contact time) a new experimental protocol was developed (Protocol II).
5.3.2 Protocol II

Temperature
A plot of breakaway force against temperature using *Protocol II* is shown in Figure 5.8. A full range of applied force rates of 0.3 Ns$^{-1}$ to 39.7 Ns$^{-1}$ is plotted over temperatures of $-3.9$ °C to $-29.7$ °C producing a range of $F_{\text{max}}$ up to a maximum of 12 N occurring at $-17$ °C.

![Figure 5.8. Breakaway force with temperature for 4 mm SBR sample using *Protocol II*. Data set has force rates in the range 0.3 Ns$^{-1}$ to 39.7 Ns$^{-1}$. Breakaway time for all experiments was 15 seconds (± 3 seconds).](image)

The points show a trend similar to that of the *Protocol I* data (see Figure 5.4). A general rise in $F_{\text{max}}$ with a fall in temperature is evident, with the rise becoming less prominent for temperatures below approximately $-20$ °C.

Force rate
As for the *Protocol I* experiments in Section 5.3.1, the plot of $F_{\text{max}}$ against $dF/dt$ has been split into four temperature ranges and is shown in Figure 5.9. The ranges are $-3.9$ °C to $-10$ °C; $-10$ °C to $-16$ °C; $-16$ °C to $-20$ °C; and $-20$ °C to $-27$ °C.
At high and intermediate temperatures there is a significant non-linear relationship between the breakaway force and the force rate in both protocols, however at lower temperatures this does not hold for Protocol II. The effect of temperature on the breakaway of low force rates is more evident in Protocol II, with much higher breakaway forces being measured for the low force rates at low temperatures. This trend of diminishing dependence on force rates cannot attributed to the change of protocol but it has to be noted that Protocol II was implemented with a lower diameter sample (4 mm). The dynamic friction potential increase in the high temperature range can amount to 8 N (from 3.5 N at 0.3 Nm$^{-1}$ to almost 12 N at 39.7 Ns$^{-1}$) by applying a high $dF/dt$, a gain of over 250% is achievable on the friction available at low $dF/dt$. This gain is similar to the one observed in Protocol I for high temperatures. However, the effect of $dF/dt$ at low temperatures in Protocol II is less marked, with breakaway forces rising by less than 10% between low and high $dF/dt$. 

Figure 5.9. Breakaway force against force rate for four ranges of temperatures using 4 mm SBR samples and Protocol II. Points are individual breakaway tests and placed according to their test temperature.
5.4 Discussion

As discussed previously, the breakaway force might be derived from two sources: the surface effects of adhesion and asperity shearing at the interface, and the edge effects of mechanical keying between the sliders. A combination of both of these can determine the magnitude of the measured breakaway forces.

The degree of $F_{\text{max}}$ attributed to adhesion depends on the real area of contact between the counter facing samples. The mechanical keying component is composed of micro keying, macro keying and ploughing.

During experiments it was observed that the SBR sample made an impression on the ice surface when left in stationary contact. Above around $-10$ °C the sample made an appreciable indentation in the surface, becoming more pronounced with increasing temperature. Below $-10$ °C it was possible to see where the sample had been sitting, although there was no appreciable indentation. Below around $-20$ °C the sample did not make a mark on the ice, even for extended periods of contact. This indicates that macro keying will contribute to $F_{\text{max}}$ for high temperatures, most significantly between $-10$ °C and $-1$ °C above which the ice is too weak to support any significant normal forces.

While it is noted that the adhesion component of the friction force is related to the real area of contact and the shear strength of the asperities within, it is also noted by Barnes et al. (Barnes et al. (1971)) that this simple relationship does not take into account asperity junction growth (Maeno and Arakawa (2004), Tusima (1978)). The growth of junctions (i.e. asperity-to-asperity contact zones) is an effect that increases the real area of contact and can be caused due to simultaneous application of normal and tangential loading as well as sintering processes. It is conceivable that shear forces exerted during $t_{\text{lag}}$ could increase the area of contact at a similar rate as during $t_{\text{dwell}}$ where shear force is not applied, so this is assumed to be the case for Protocol II experiments.
5.4.1 The effect of temperature

Figure 5.4 and Figure 5.8 show the trend of breakaway force with temperature for Protocol I and Protocol II respectively. For both plots there is an overall non-linear trend with temperature, with the highest \( F_{\text{max}} \) occurring at the lowest temperatures, and the lowest \( F_{\text{max}} \) closest to the melting point. The non-linear trend is steepest at high temperatures, and becomes less so with falling temperature, possibly forming a plateau below around \(-20\ ^\circ\text{C}\). The trend is consistent with both protocols used.

Looking at the overall shape of the plots, they are quite similar to the findings of previous ice investigators unconcerned with friction transients when investigating steady state sliding ice friction (Bowden (1953), Bowden and Hughes (1939), Bowden and Tabor (1950), Evans et al. (1976)) and when studying ice surface properties (Bowden (1954), Golecki and Jaccard (1978), Hobbs (1974), Roberts (1981b), Valeri and Mantovani (1978)).

The two main points relating to the temperature of the ice from previous work are those of strength properties and the liquid-like layer. The strength of ice varies considerably with temperature, especially close to the melting point, and when contaminants are present. The shape of plots of friction against temperature for ice on ice reported by Raraty and Tabor (Raraty and Tabor (1958)) and of Bowden (Bowden (1953)) fits quite closely the data obtained here. The explanation by both authors is that the strength of ice becomes greater with falling temperature and the strength of the adhesive bond is limited by the shear strength of the ice, given sufficient adhesion. This behaviour is mirrored in the breakaway force plots shown in Figure 5.4 and Figure 5.8.

Because of the short time scales involved in this study, a relationship to the behaviour of a liquid-like layer is possibly a more appropriate comparison to make in this system. It has been reported and generally accepted that there is a "mobile, liquid-like and disordered" surface layer on the ice, weaker than the bulk material and capable of creep several orders of magnitude faster than the bulk (Roberts (1981c)). It is also reported that this liquid-like layer remains present when rubber is in contact, something that does not often occur with other surfaces (such as metal oxides), due to rubber's non-reactive nature (Makkonen (1997)).
layer has been found to exist in pure ice to temperatures even below those studied here (Golecki and Jaccard (1978), Valeri and Mantovani (1978)), but any impurities greatly affect the nature of the surface (Roberts and Lane (1983)). As the experiments here were carried out using tap water, ionic impurities will be present, so the effect of this layer could be highly significant. The level of impurities present in the ice may define the shape of the $F_{max}$ against temperature curve by separating the solid ice from the rubber surface at high temperatures and not allowing significant solid-solid adhesion to take place.

Although no significant sliding takes place in breakaway experiments, this shape of trend, when regarding ice friction behaviour with temperature, is generally related to melting of the surface through frictional heating. Breakaway experiments are basically a measure of static friction, so it could be assumed that no sliding takes place, hence frictional heating should not be the main cause of this behaviour. It has been found, however, that frictional heating can take place over very small displacements (around $\frac{2}{3}$ of an asperity diameter (Calabrese et al. (1980)), so force-induced sliding during $t_{lag}$ may be sufficient to induce melting and hence lower friction, especially at higher temperatures. It has also been suggested (Barnes et al. (1971), Maeno and Arakawa (2004), Tusima (1978)) that as ice has a low shear strength compared with most other materials that shearing within the ice may occur, given sufficient adhesion to the counterfacing material. This may also generate heat to weaken the ice sufficiently and produce the decreasing breakaway force measured above $-20$ °C. However, it is again unlikely that “sufficient” adhesion will have taken place during the 14 seconds of Protocol II experiments for the adhesion bond to have reached the cohesion strength of ice at that temperature. For this reason, bulk shear is unlikely in this system, and surface shear (either between rubber and ice or within the liquid-like layer, or a combination of both) is the more likely mechanism. We note that the liquid-like layer viscosity rise is more than 200 % from $-1$ °C to $-2$ °C, and can be ten times higher than that of pure water at 0 °C as suggested by Ahagon et al. (Ahagon et al. (1988)).

Temperature also affects the viscoelastic properties of the rubber samples used in these breakaway tests. (Full DMA data can be found in Appendix V for the rubber
used in breakaway tests). In our temperature range, both the SBR storage modulus, $E'$, and loss modulus, $E''$, increase with dropping temperature which might also contribute to higher forces at lower temperatures. The findings in Chapter 4 would suggest that the rubber property contributions are significant. DMA tan $\delta$ data for this rubber also show similar trends with temperature to that of Figures 5.4 and 5.8. Thus, it is not clear at this point the relative significance of the ice-related and rubber-related mechanisms.

As a system, all of these effects (strength changes, viscosity changes, viscoelasticity changes, small-scale frictional heating) may combine to produce the behaviour observed in Figures 5.4 and 5.8. The higher scatter in points in Figure 5.4 compared to those in Figure 5.8 is most likely due to the non-constant time effect of Protocol I, with Protocol II results in Figure 5.8 showing the more true behaviour of breakaway force with temperature.

### 5.4.2 The effect of force rate and time in stationary contact

The plots shown in Figure 5.5 (Protocol I) and Figure 5.9 (Protocol II) have been divided up into temperature ranges because, as shown previously, the temperature has a significant effect on the breakaway force ($F_{\text{max}}$). All temperature groups show a non-linear trend with the rate of change of $F_{\text{max}}$ decreasing with increasing $dF/dt$.

The behaviour of the system resembles the mechanical response of viscoelastic materials (for example see Léger et al. (Léger et al. (2001)) where transient SBR friction on glass is reported). This behaviour is demonstrated most clearly for high temperatures in Protocol II (see Figure 5.9), with a significant increase in $F_{\text{max}}$ shown for increasing $dF/dt$.

The data obtained using Protocol I is fairly similar throughout the temperature range but this is not true for Protocol II. The effect of $dF/dt$ is very significant for high temperatures using Protocol II and minimal for low temperatures. The fact that in Protocol I there is a significant change of breakaway force in all temperature ranges and in Protocol II there is almost no change in breakaway force at low temperatures is counter-intuitive and cannot be attributed to the change of
protocol. The main caveat of Protocol I is that low force rates lead to longer breakaway times and consequently to an overestimation of the breakaway force at low force rates which theoretically leads to less steep curves. Due to this reason, steeper curves are to be expected in Protocol II, which is clearly not the case especially at low temperatures. If the main source of the behaviour originates purely from the viscoelastic time- and temperature-dependent properties of rubber the behaviour should have been similar.

We believe that a crucial point is that for Protocol II a smaller diameter sample has been used, resulting in higher pressures (for the same load) and affecting the presence of the liquid-like layer at the interface, and hence adhesion between rubber and ice.

For the temperature range used in these experiments, it has previously been shown that a liquid-like layer may exist on the surface of the ice, even when a rubber slider is in contact. Normally in sliding friction the liquid-like layer is not affected significantly by squeeze effects because of the pressure generated hydrodynamically. In breakaway experiments, however, the static pressure applied to the interface during \( t_{\text{breakaway}} \) may cause some of the layer to be squeezed out of the interface layer, artificially thinning it.

The squeeze effect is more prominent for experiments using Protocol II, where the normal pressure is 310 kPa, compared to the 50 kPa of Protocol I. This pressure-induced expulsion of the liquid-like layer can lead to dry "friction" behaviour (shear rate independent) demonstrated by the decrease in curvature of the plots in Figure 5.9. We have already discussed that this cannot be attributed to the viscoelastic behaviour of the SBR samples because in this case the results of both protocols should compare well. Thus, it must be assumed that the deviation from non-linear behaviour at low temperatures is due to changes at the ice/rubber interface. The friction forces should be greater as the liquid-like layer becomes thinner (loss of lubrication effect leading to dry friction) or more viscous (Newton's law of viscosity). It is expected and has been shown that the liquid-like layer acquires lower thickness and higher viscosities at low temperatures (Ahagon et al. (1988), Golecki and Jaccard (1978)).
We can also suggest that the increased pressure of Protocol II squeezes out the liquid-like layer, allowing more intimate contact between rubber and solid ice surfaces. This contact allows stronger adhesion and keying to take place between samples, especially at low temperatures, hence giving rise to higher frictional forces. The weakness of the ice at higher temperatures allows the ice to creep within a thin interface layer, producing the behaviour observed in Figure 5.9.

The plot in Figure 5.6 clearly shows the dependence of $F_{\text{max}}$ with time in contact. Longer times result in higher breakaway forces as sintering processes cause keying and allow adhesion to take place more effectively over longer periods. With reference to force rate experiments, using Protocol I all experiments have equal time in stationary contact ($t_{\text{dwell}}$) but the highest force rates have lowest times in total contact ($t_{\text{breakaway}}$). Conversely, Protocol II experiments have constant $t_{\text{breakaway}}$ and hence, the highest force rate experiments have the longest time in stationary contact. If the time in stationary contact was an important parameter, the trend for $F_{\text{max}}$ against $dF/dt$ for Protocol II experiments should show significantly lower $F_{\text{max}}$ for low rates than for high rates, which is not the case, especially at low temperatures. It must be concluded that for the time scales and force rates of our experiments, contact time is not of significant importance to this system, or certainly that the contribution of $dF/dt$ outweighs the effects of contact time. One can actually see that the findings shown in Figure 5.6 suggest a relatively weak time dependence of breakaway force (maximum of 25%) within the range of contact times between 15 to 30 s.

### 5.4.3 The effect of contact pressure

When carrying out experiments, two sample sizes were used; 4 mm and 10 mm diameter both were 2 mm thick. Plots of $F_{\text{max}}$ against temperature for both sizes are shown in Figure 5.10. The comparison of these two sizes shows a marked difference in the breakaway forces, with the 10 mm sample having exhibited $F_{\text{max}}$ of around half of the 4 mm sample. It should be noted that the normal load in both cases was the same at 4 N, so the much larger friction forces evident is an effect of the respective pressure.
It is likely that for two samples bearing the same normal load, the initial real area of contact should be similar, independent of physical sample sizes, assuming purely elastic deformation. Once failure of these initial bearing asperities has occurred, the real area of contact will rapidly become dependent on the load per unit of projected area, i.e. force per unit of sample surface area, or apparent pressure. This is of course much higher for the small sample compared to the large sample (310 kPa compared to 50 kPa), so one might expect the creep rate to be much higher for the 4 mm samples leading to a higher real area of contact (Barnes et al. (1971)), more effective squeezing of the liquid-like layer from the interface leading to more extensive mechanical keying, higher adhesion strength and larger drag forces.
5.4.4 Comparison to winter tyre measurements

As part of the collaboration with Ford’s Premier Automotive Group (PAG) research, experimental data was provided for comparison with their ongoing winter handling programme. The data provided by them was for a model vehicle fitted with full-scale winter tyres using which they are able to carry out similar breakaway tests as described here. Data from such tests is shown in Figure 5.11. The protocol used by PAG means that comparisons can be made to the data shown in Figure 5.5.

The trend of $F_{\text{max}}$ against $dF/dt$ (notated as $\dot{F}_{\text{app}}$) shown in Figure 5.11 is very similar to the proposed trend shown in Figure 5.5 for the comparable temperatures of $-10 \, ^\circ C$ to $-16 \, ^\circ C$. The most significant rise in $F_{\text{max}}$ by increasing $dF/dt$ is shown for lowest $dF/dt$, with the slope of the curve decreasing with increasing $dF/dt$. A much higher dependency on $dF/dt$ is displayed for tyre measurements compared to those measured on the tribometer, and this may be due the differences in tyre behaviour compared to flat rubber samples.
The dependency of breakaway force on $t_{\text{dwell}}$ is shown also in the PAG tyre measurements, with a significant dependency on force rates shown for even short $t_{\text{dwell}}$ periods, in agreement with our measurements.

The favourable comparisons between the PAG data and those presented here are very useful in terms of vehicle system development. Whilst the data is different in magnitude, the trends uncovered using the tribometer are mirrored in the trends shown when using full-size winter tyres. This allows comparative tests to be made between, for example, dynamic traction control programs and fine tuning may be possible using the tribometer in parallel with numerical models to improve the very expensive full-scale testing carried out today.

The mechanisms and frictional behaviour shown in breakaway tests is of very high significance to vehicle winter handling. At the moment, most empirical data used in vehicle numerical models is steady state, but the significant increase in friction possible by incorporating a dynamic element may allow much better grip on icy surfaces. This could allow great performance and safety gains over the dynamics systems currently in operation.
5.5 Conclusions

The source of dynamic tyre friction potential (DTFP) has clearly been shown to be from surface interaction between the rubber element and ice. The source of the breakaway force is predominantly based on the interfacial behaviour of the system although viscoelastic contributions from the rubber cannot be excluded.

Temperature has a significant effect on the measured breakaway forces with highest forces being produced at lowest temperatures. This is a trend also found in steady state friction studies and can be attributed to the change in ice strength, interface layer viscosity and small-scale frictional heating with temperature.

Rate of applied (shear) force has a very significant effect, with a 250 % increase in $F_{\text{max}}$ possible at high temperatures by changing $dF/dt$ from low $(0.3 \text{ Ns}^{-1})$ to high $(39.7 \text{ Ns}^{-1})$ rates. At low temperatures the gain is not as significant with the rise in $F_{\text{max}}$ being 10 % within the range of $dF/dt$ studied. The source of this behaviour has been attributed to the diminished influence of the liquid-like layer at low temperatures and high pressures.

Contact time also has an effect on the magnitude of $F_{\text{max}}$, with greatest $F_{\text{max}}$ being measured for highest contact times, but this only becomes significant for prolonged periods of contact. As $F_{\text{max}}$ is (at least partly) an indication of the adhesion before breakaway occurs, longer contact times produce higher adhesion through real contact area growth with time. The processes that control this are sintering and creep of ice and deformation of rubber.

The effect of contact pressure on the creep of the interfacial ice layer, on rubber-ice adhesion and on the thickness of the liquid-like layer determines the level of breakaway forces measured for this system. $F_{\text{max}}$ at high force rates measured at low pressures of 50 kPa are around 50 % lower than those measured at the relatively high pressures of 310 kPa using the same normal load.

Favourable comparisons can be made to full-scale tyre data, showing that the processes that occur at the interface and control $F_{\text{max}}$ for small-scale samples are also responsible for the frictional behaviour of winter tyres.
Incorporating dynamic aspects into vehicle control systems could allow significant progress with regard to vehicle performance and occupant safety.
Chapter 6.

Conclusions and further work

6.1 Conclusions
A pin-on-disc tribometer was designed and built in-house and used to perform measurements of friction between styrene butadiene rubber and ice.

The design was modified as a result of problems identified during a preliminary test period, the modifications optimised performance for this tribosystem and for the environmental conditions used for this work.

The tribometer design was accomplished so that friction measurements could be taken using samples in two geometries, ice-on-rubber and rubber-on-ice. The former of these configurations was used for friction measurements leading to the production of a friction map; the ice pin samples being used for low temperature scanning electron microscopy (LT-SEM) observations to identify ice wear morphologies. The latter geometry was used when performing breakaway tests.

Friction mapping was used to represent steady-state friction coefficients as a function of environmental temperature and sliding speed. Friction maps allow experimental data to be displayed on a single diagram, giving the reader a clear indication of the magnitude of friction at any temperature-speed position within the range studied.

Previous methods used to represent friction data did not allow multi-parameter data to be displayed effectively. Mastercurves can demonstrate the viscoelastic nature of rubber friction very effectively on dry, unchanging surfaces, but requires transformation of data, and cannot show friction data as the dependency on viscoelasticity is lost and another mechanism takes over, such as friction melting as...
Conclusions

is the case in this tribosystem. Friction curves have been used to show the single parameter dependencies of friction such as speed or temperature, even with mechanism changes, but they cannot show multi-parameter dependencies as clearly nor robustly as the friction map method presented here.

Wear morphologies observed using LT-SEM were linked to the temperature-speed position on the friction map at which the sample was worn. Friction behaviour specific to rubber properties and to an ice-controlled thermal friction model were revealed to coexist over large speed and temperature ranges on the map. LT-SEM observation allows great insight to the mechanisms that control the magnitude of friction, especially at critical points on the map, such as the zone of maximum friction and transitions from frictional melting to dry friction regimes.

Identification of a Dynamic Tyre Friction Potential (DTFP) during measurements using full-scale tyres led to an investigation to determine the source of this previously unreported phenomenon. Measurements of transient friction in breakaway tests showed that breakaway friction can be boosted by increasing the rate of applied shear force on the interface. It was possible to replicate this phenomenon during experiments carried out on the tribometer and measurements were verified to data collected using the full-scale tyres.

As well as verifying data and determining that the DTFP was due to interfacial processes other dependencies were established. Greater increases in friction were measured for high temperatures, with the effect of force rate becoming less prominent at temperatures below −20 °C. Increasing the time in stationary contact and the normal pressure also increased the magnitude of the breakaway forces.
6.2 Outlook and perspective

The completion of this project has revealed that there is much more work that could be investigated in the field of rubber-ice friction. The project has established some important groundwork and met several key goals. Firstly the tribometer and experimental procedures were developed to the extent that further experiments can be readily executed. Steady state experiments provided the basis on which the tribometer and procedures were developed, and through modifications and fine-tuning, the group of transient experiments were facilitated.

The tribometer as a whole was developed to the extent where it would be easy for further work to be carried out using the current set up. There are however areas where the instrument could be improved with regard data accuracy and repeatability and experiment versatility. Firstly, there is a slight tendency for the servo motor (if kept electrically enabled) to raise the disc temperature by several degrees. Protocol allowed this to be minimised, but changes could be made to improve the situation. At the project initiation, it was envisaged that the tribometer would have 2 dimensions of movement, i.e. the pin could follow a spiral track on the disc instead of the circular track of the current layout. This would require a second servo motor and some slight modification to the tribometer, but all other equipment specified at the offset would enable this modification. This second modification would allow pseudo-linear movement by allowing the pin to traverse over virgin material, eliminating the memory effect that exists when travelling over a single, circular track.

Steady-state experiments leading to friction maps is a very valuable tool for material selection purposes based on the respective friction properties. As the most slippery (and hence dangerous) road conditions are at temperatures just below freezing, it is important to improve the tyre’s friction under these circumstances. Friction maps could be utilised to compare different rubbers over a range of temperatures and speeds, modifying and selecting ones that exhibit favourable properties for a wide range of conditions, while aiming to improve friction significantly within the temperature range close to freezing.
Further studies in this regime of interest could be of the use of thermal imaging equipment such as an infrared camera to measure heat dissipation from the contact area. This would be a very useful method to determine the thermodynamic effects of temperature build-up in the ice, a factor of slight concern with the current tribometer geometry. As it was established during the course of the project the importance of the thermal control model, thermal imaging would also be of interest in determining the thickness of the interfacial melt water layer for different experimental parameters.

Transient friction studies are very rare in the scientific literature although they can be the most valuable and relevant for many practical situations and applications of the rubber-ice system. As we have found, the reason for the scarcity of these studies could be the complex non-linear behaviour of all components within the rubber-ice system: rubber viscoelasticity, ice mechanical properties and the rubber-ice interface with the possible existence of the liquid like layer. We believe that with our transient regime experiments we have just opened the way for a multitude of systematic experiments for different experimental parameters in order to elucidate further the underlying physical mechanisms. As it has been shown, the experiments carried out in the laboratory can easily replicate the behaviour revealed when experimenting on full-scale tyres and vehicles. The laboratory experiments are faster, less costly and are more controllable in terms of temperature and speeds than full-scale measurements. There is obviously the slight non-conformity of using a pin-on-disc system and the memory effects that comes with it, but modifications to the tribometer as suggested and protocol development could allow very important behaviour to be uncovered and refined.

The main use of all of this data, both in steady state and transient regimes will be that of vehicle control systems and pre-production and development chassis modelling. These are processes that have become very refined and essential in the last decade or so, and although the models in particular have become very powerful tools, they must be validated with accurate data or the "rubbish in, rubbish out" scenario is realised. Of course there are no correct data or even representative data for all tyres and ice, but with careful protocol, materials and understanding very useful and powerful systems can be produced.
Of significant importance is that of the breakaway findings, and if a strong understanding is established, the phenomenon of DTFP could be incorporated into dynamic automotive traction and stability control systems to promote higher grip on icy road surfaces, something that is particularly important at high temperatures.
Chapter 7.

References


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Appendix I – Calculations regarding the limits of the motion control system

The motor chosen for the control system is a Faulhaber 3863 Series. The absolute limits of the motor are provided on the data sheet (see Appendix II.1), but the system is limited by the requirements of the torque load (i.e. the friction force at the radius of the disc). I have based the calculations on the previous friction measurements by Gnörich and Grosch (Gnörich and Grosch (1972)), assuming a maximum coefficient of friction, \( \mu \), of 4.

The torque that can be provided by the motor to rotate the friction force at a given disc radius is limited by the supply current from the National Instruments MID-7652 servo amplifier. This gives a total limiting torque of 133.2 Nm, up to a speed of \( \approx 6000 \) RPM, where the motor specification is the limiting factor. The disc radius is thus limited in size, dependent on the frictional load. A plot of this relationship is shown in Figure AI.1 with each line representing different \( \mu \).

![Figure AI.1](image_url)  
Figure AI.1. The limiting disc sample radius based on motor torque and the prevalent coefficient of friction, \( \mu \).
From the plot it is clear that the normal load is increased, the effective radius of the motion system becomes much less. This is a measure of the force that can be applied to start motion (especially important in breakaway tests), as once sliding occurs, $\mu$ will become much less, and so the effective radius is greatly increased.

When sliding occurs and the disc is therefore in motion, the limits of the system are governed by the power consumed to keep the torque load sliding. Again, the coefficient of friction will be much less than the assumed value of 4 once sliding has commenced, but this figure will be used to find the absolute limits of the system.

![Figure A1.2](image)

**Figure A1.2.** The limiting power for the motion control system used in the tribometer. Dashed lines are the system constraints, and the solid lines represent the power required for each torque load, assuming $\mu = 4$. Shaded areas are outside the respective component limits, leaving the white area the operational envelope of the tribometer.

Figure A1.2 represents the operating envelope for the tribometer (white area), governed by the motor power and the current provision by the servo amplifier. Sliding friction experiments can only be carried out within this envelope, hence when a high normal load is required, the sliding speed is limited by the speed of the motor as shown, i.e. for a 600 g normal load and prevailing $\mu = 4$, the limiting motor speed is 1570 RPM, but if using 400 g, speed up to 2336 RPM can be used.
Appendix II – Component data sheets

All.1. Faulhaber 3863-024C servo motor
All.2. Entran ELFS-T3M-25N load cell
# DC-Micromotors

Graphite Commutation

## Series 3863 ... C

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<th>012 C</th>
<th>018 C</th>
<th>024 C</th>
<th>036 C</th>
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<td>Efficiency</td>
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<tr>
<td>No-load speed</td>
<td>n₀</td>
<td>6 500</td>
<td>6 600</td>
<td>6 700</td>
<td>6 400</td>
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<td>No-load current (with shaft ø 6,0 mm)</td>
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<td>0,240</td>
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<td>Stall torque</td>
<td>Mₑ</td>
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<td>1 090</td>
<td>1 250</td>
<td>1 170</td>
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<td>Friction torque</td>
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<td>- rotor, max. permissible</td>
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<td>- axial at 3 000 rpm</td>
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<td>- axial at standstill</td>
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<tr>
<td>axial</td>
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<td>Housing material</td>
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## Recommended values

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<td>Current up to (thermal limits)</td>
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## Orientation with respect to motor terminals not defined

[Diagram]

For notes on technical data and lifetime performance refer to "Technical Information".
ELFM Series Load Cells

ELFM-B1, -B2 & T2

ELFM-B1
5 to 50 Lb
25 to 250 N

ELFM-B2
100 Lb
500 N

ELFM-T2E
5 to 100 Lb
25 to 500 N

ELFM-T2M
5 to 100 Lb
25 to 500 N

ELFM Series

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<th>Body Styles</th>
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<th>N Ranges (± for -T)</th>
<th>Overrange Limit (± for -T)</th>
<th>Output &quot;FSO&quot; Nom. (± for -T)</th>
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<td>25 to 250 N</td>
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EXCITATION: 5VDC 350 Ω
BRIDGE IMPEDANCE: Metallic Foil
SENSING TECHNOLOGY: Metal Foil
NON-LINEARITY: ±0.25%FSO
HYSTERESIS: ±0.5%FSO
DEFLECTION AT "FS": 0.025 mm to 0.075 mm
THERMAL ZERO SHIFT: ±0.02%FSO/C
THERMAL SENSITIVITY SHIFT (TSS): ±0.02%FSO/C
OPERATING TEMPERATURE: -50°C to 70°C
COMPENSATED TEMPERATURE: ±2%FSO typ
ZERO OFFSET AT 20°C (70°F): ±2%FSO
MODE: CALIBRATION: "Off-the-Shelf" Stocking Program

Body Material: Stainless Steel

Entrain Sensors & Electronics
USA: Fairfield, NJ
UK: Garston, Watford, Herts, England
Europe: Les Clayes-sous-Bois, France

www.entran.com
Options and Accessories:

COMPENSATED TEMPERATURE RANGES:
- STANDARD = 15°C to 70°C (60°F to 160°F)
- Z1 = -20°C to 40°C (0°F to 100°F)
- Z2 = 0°C to 60°C (32°F to 140°F)
- Z4 = 40°C to 90°C (100°F to 200°F)
- Z* = Non-standard, contact Entran

EXCITATION VOLTAGE:
- STANDARD = 5VDC
- V* = Non-standard Excitation, contact Entran

SPECIAL LEAD LENGTH:
- L00F
- L00M

SPECIAL MODULE LOCATION:
- M00F
- M00M

WATERPROOFING LEAD EXIT
Not available on ELFMB1 below 25lbs or 125N:
- X = Short term waterproofing. Limited to 105°C (220°F)

CONNECTOR WIRED TO LEADS:
- C = Microtech type male or equivalent (w/o mate)
- R = RJ Telephone type male (w/o mate)
- RQ = Pins to mate with MM50 screw terminals

WIRING COLOR CODE:
- STANDARD Green = + Output Signal
- White = - Output Signal
- W+ White = + Output Signal
- Alt. Color = - Output Signal

MATING CONNECTORS FOR CONNECTOR OPTIONS:
See Cable and Connector Bulletin

CALIBRATION:
- STANDARD Available
- -B: Compression
- -T: Tension
- -B: Compression calibration for -T body
- -T: Tension calibration for -T body

Model Number construction:

ELFM Series - B1 Housings - 50 Range - N Units - Z1/L2/M/B Options

- B1
- B2
- T2E, T2M
- (K used N=Newtons for 1000 L=Pounds)
- Ex.: 1K

C, R or RQ
- L00F or L00M
- M00F or M00M
- V*
- W+
- X
- Z1, Z2, Z4, or Z*

"Off-the-Shelf" Stocking Program

Mounting & Wiring:

LOAD CELL MODULE (internal on B1 & B2)
- Red + Input
- Green + Output (Option W+: White + Output)
- Black - Input
- White - Output (Option W+: Alt. Color - Output)

Entran®

ELFM LOAD CELLS

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Appendix III – LabVIEW® block diagrams and graphical user interfaces (GUIs)

The tribometer LabVIEW® control system comprises three main routines: the main .vi from which all other routines are started and synchronised (Figures AIII.1 and AIII.2), the data acquisition .vi (Figures AIII.3 and AIII.4) which is used to select which data should be logged, and the fashion it should be logged in, and the motion control .vi (Figures AIII.5 and AIII.6) which is used to control the speed, acceleration, time, etc of the motion control system. All three main GUIs and block diagrams are shown in pages 174-179.

Figure AIII.1. The block diagram for the main LabVIEW® .vi. This corresponds to the main GUI (see Figure AIII.2). Triggers for the two subroutines (motion control .vi and data acquisition .vi) are incorporated into this block diagram for synchronization purposes.
Figure AII.2. The main GUI from which the tribometer is operated. Once the sampling and motion parameters have been selected using the data acquisition and motion control GUIs respectively, the tribometer can be started and stopped using this control panel.
Figure All.3. The data acquisition .vi block diagram.
Figure AIII.4. The data acquisition .vi GUI. Friction forces are displayed as a waveform in the “Friction Force (N)” graphical display, while the environmental temperature is displayed on the “Freezer Temperature (degC)” indicator.
Figure All.5. The motion control .vi block diagram.
Figure AIII.6. The motion control .vi GUI using which the motion parameters of the tribometer are selected. The velocity as a function of time is programmed by entering the appropriate parameters into the “C1”, “C2” and “C3” controls. The real-time disc velocity is shown in the “Filtered Velocity” display.
Appendix IV – Engineering drawings for the tribometer

Engineering drawings for the main components of the tribometer are shown in pages 181-201. Dimensions to be taken as notated, assume as not to scale.
UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX
Φ To Fit Pulley Provided

Φ 6

Φ Bearing Press Fit

Φ 12

Φ 21.7

Φ 10
Note: Angles indicative only
Contact: Daniel Higgins
Tel: 07884182747
d.d.higgins@ed.ac.uk

SOLID EDGE
EDS-PLM SOLUTIONS

Tribometer Arm for Load Cell

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX

NAME | DATE
--- | ---
DRAWN | s973641 04/16/07
CHECKED | 
ENG APPR | 
MGR APPR | 

FILE NAME: Tribo5Mods.dft
SIZE: A4
SCALE: 1:1
WEIGHT: SHEET 7 OF 20

REVISION HISTORY
REV | DESCRIPTION | DATE | APPROVED
--- | --- | --- | ---

Tri-washer

Dimensions are in millimeters

ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX

UNLESS OTHERWISE SPECIFIED

NAME | DATE
---|---
S9734641 | 04/16/07

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Tri-washer

SIZE | DWG NO | REV
---|---|---
A4 | | 

FILE NAME: Tribo5Mods.dft

SCALE: | WEIGHT: | SHEET 10 OF 20
---|---|---
Two of required
REVISION HISTORY

REV | DESCRIPTION | DATE | APPROVED
--- | ----------- | --- | ---

NAME | DRAWN | CHECKED | ENG APPR | MGR APPR
--- | --- | --- | --- | ---

DATE | DRAWN NO.
--- | ---

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SOLID EDGE ACADEMIC COPY

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
ANGLES ±XX°
2 PL ±XXX 3 PL ±XXXX

FILE NAME: Tribo5Mods.dft

SCALE: WEIGHT: SHEET 18 OF 20
Please modify existing component to specified dimensions shown.
Appendix V – DMA data for styrene butadiene rubber

Figure AV.1. Dynamic rubber data for the SBR (serial number 325005) sample found using DMA in cantilever mode: single frequency (1 Hz), single displacement (0.05 mm). The peak in the tan δ against temperature plot indicates that $T_g = -22.4 \, ^\circ\text{C}$. 
Figure AV.2. Dynamic rubber data for the SBR (serial number 325003) sample found using DMA in cantilever mode: single frequency (1 Hz), single displacement (0.05 mm). The peak in the tan δ against temperature plot indicates that $T_g = -30.7 \, ^\circ C$. 
Appendix VI – Deriving an expression for replicating constant rate of force application

From previous tests carried out using the tribometer, it was decided that velocity feedback using the friction force or motor torque would not be possible due to extraneous noise. Therefore, another, theoretical method was derived for control of force application to the rubber element by using the velocity control system.

Friction forces in the tribometer (see Figure AVI.1) are measured using a load cell, which can be assumed to be spring-like in operation (viscous behaviour of rubber or ice is assumed minimal), therefore, Hooke's law is applied, which states:

\[ F = kx \]  

\[ t = tx \]  

\[ \text{load cell} \]  

\[ \text{rubber element} \]  

\[ t = tx+1 \]  

Figure AVI.1. Schematic diagram of the tribometer, the load cell represented as a Hookean spring. Deformations of \( x \) apply a force, \( F \), to the rubber element after \( x+t \) seconds.

For breakaway tests, ideally \( x \propto t \) so the friction force is increased linearly with time, therefore by taking the time derivative of (i):

\[ \frac{dx}{dt} = F' \]  

\[ \text{therefore } v = \text{constant, since } x \propto t \]  

\[ (ii) \]

where \( F' \) is the force rate and \( x \) is the strain of the load cell. This a constant velocity over the second stage of breakaway period, \( t_{\text{lag}} \), preceded by an instantaneous step velocity change and as this is not possible using the present
control system (because of the inability to maintain a very low steady speed), another expression is suggested:

\[ v = at \tag{iii} \]

where \( a \) is a constant defining the force application rate.

\[ \frac{dx}{dt} = at \tag{iv} \]

therefore substituting the integral into (i),

\[ F = \frac{a}{2}t^2 + c \tag{v} \]

This gives us a parabolic force application with time, instead of the constant rate requested, and the force rate is calculated as the tangent of the function at the breakaway point (where the slope becomes steeper and nearly linear), as shown in Figure AVI.2 below.

![Figure AVI.2](image)

Figure AVI.2. Plot of force with time for a typical breakaway test. The force rate \((dF/dt)\) is determined by calculating the tangent of the \(F-t\) curve at the breakaway point.