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# Multi-Physics Experimental Investigation into Stator-Housing Contact Interface

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## Abstract

The shrink-fitting of housings on to electrical machine stators is a common, semi-permanent and low-cost method of assembly. As the stator-housing interface lies in the main heat extraction path, an ideal shrink-fit should provide the necessary holding torque, present minimal thermal contact resistance and remain mechanically and thermally stable over the operating temperature range and life of the electrical machine. The optimal design of such a shrink-fit represents a multi-physics problem requiring, among other data, accurate coefficient of friction and thermal contact conductance information. However, these parameters are influenced by many factors including interface pressure, surface preparation and temperature, and are therefore difficult to predict unless experimental methods are adopted. To this end, this paper presents two independent experimental apparatus designed to measure the pressure dependent coefficient of friction and thermal contact conductance between typical housing and electrical steel materials under in-service conditions.

## 1 Introduction

The housing of a rotating electrical machine serves the three main purposes of shielding the active components from the external environment, providing reaction torque through appropriate anti-rotation and mounting features and providing adequate cooling via fins or liquid cooling channels, Fig. 1, [1]. In rotorcraft and aerospace applications where mass is a critical design driver, it is desirable to minimise the housing mass whilst maintaining the necessary holding torque, thermal performance and mechanical integrity. One approach is to use an appropriate shrink-fit to provide the necessary holding torque which can enable physical anti-rotation features (pins, keys etc) and their associated stress concentration, complexity and cost to be eliminated, Fig. 1, and can ultimately lead to a reduction in the housing wall thickness and overall mass. As the stator-housing interface lies in the main heat extraction path of the electrical machine, an ideal shrink-fit should, in addition to providing the necessary holding torque, present minimal thermal contact

resistance and remain mechanically and thermally stable over the operating temperature range and life of the electrical machine. The identification of the optimum shrink-fit pressure and therefore part dimensions and choice of manufacturing method represents a multi-physics optimisation problem requiring, at a minimum, accurate coefficient of friction and thermal contact conductance information, [2]. However, the friction and thermal contact conductance at the interface of two materials are complex phenomena influenced by many factors including interface pressure, surface roughness, waviness and flatness, surface deformation, cleanliness and temperature, and are therefore difficult to predict, [2-4], hence reliable and repeatable experimental methods are highly desirable. To this end, this paper presents two independent experimental apparatus designed to measure the pressure dependent coefficient of friction and thermal contact conductance between typical housing and electrical steel materials under representative in-service conditions.

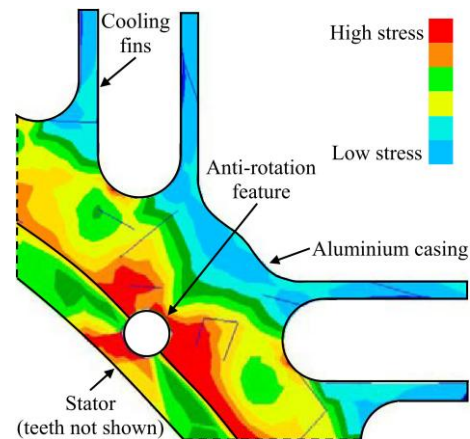


Figure 1: Stress map of a stator and housing utilising physical anti-rotation features on the stator-housing interface.

## 2 Principal of Shrink-Fits

A shrink-fit is a semi-permanent assembly system between two components providing a low-cost method for fastening parts and is widely used in industry, with applications to cutting tool holders, wheels and bands for railway stock, gears, turbine disks, locating ball and roller bearings and electrical machines. The underlying principle involves

establishing a pressure between the inner diameter of a hub (housing) component and the outer diameter of a shaft (stator) component through interference in dimensions at their radial interface, as illustrated in Fig. 2. The stator and housing are typically assembled by pressing the components together after expanding the housing by heating or in some cases contracting the stator by cooling to achieve the necessary clearance. At which point, the whole assembly is returned to the operating temperature whereupon the resulting interface pressure maintains the part location, provides resistance to tension and compression and allows transmission of a torque through friction, [2-3].

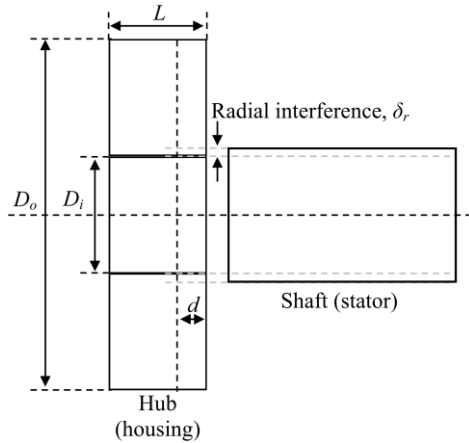


Figure 2: Schematic of a typical shrink fit.

The holding torque capability,  $T_h$ , of a shrink fit is given by (1) where,  $D$ ,  $L$ ,  $P$  and  $\mu$  are the diameter at the shrink-fit, the axial length of the contacting surfaces, the interface pressure and the static coefficient of friction respectively, [3].

$$T_h = \frac{\pi}{2} D^2 L P \mu \quad (1)$$

In order to accurately calculate the holding torque of a shrink-fit it is necessary to obtain a precise value for the static coefficient of friction between the constituent component materials. However, this property varies with interface pressure, material hardness and elasticity, surface roughness, surface conditions (dry, lubricated, oxidised, etc), environment (temperature, humidity) and loading rate, [2-5]. The pressure and contact area are themselves affected by geometrical inaccuracies caused by the chosen manufacturing method used to produce the component surfaces. In addition, the coefficient of friction is a highly variable property compared to other properties such as yield strength or Young's Modulus of Elasticity.

The stator-housing shrink-fit lies in the main heat extraction path and as such can have a considerable impact on the cooling capability of the electrical machine, [1]. The thermal interface resistance,  $R_c$ , between the stator and the housing is given by (2) where  $A$  is the area of the interface and  $h_c$  is the thermal contact conductance, [4]. As with the coefficient of

friction, the thermal contact conductance is highly dependent upon the in-service conditions and material properties, [4-5].

$$R_c = \frac{1}{h_c A} \quad (2)$$

Therefore, direct experimental measurement is highly desirable in order to obtain accurate coefficient of friction and thermal contact conductance data for a specific set of interfacing materials under a representative set of in-service conditions.

### 3 Coefficient of Friction Measurement

The frictional force required to initiate relative movement between two surfaces is given by (3) where,  $F$ ,  $\mu$  and  $N$  are the maximum static frictional force, the static coefficient of friction and the normal force at the interface respectively, [6].

$$F = \mu N \quad (3)$$

This relationship is valid for conditions of low normal loads and is contact area independent. The situation for shrink-fits is more complex. First, the maximum frictional force can either be radially translated to a maximum holding torque or axially translated to a maximum holding force, [2]. The coefficient of friction is highly dependent on the radial pressure at the interface which can be determined analytically using Lamé's equations for thick cylinders along with the interfacing material properties and the associated component dimensions, [6]. The mechanism of slippage in shrink-fits initiates at the surfaces closest to the applied force or torque, Fig. 2, with a depth  $d$  and then propagates along the contact surface axially. Failure of a shrink-fit assembly is assumed when complete slip occurs along this interface due to insufficient radial pressure and frictional resistance, [6]. Previous work, [2-3], utilised the relationship given in (1) for cylindrical parts in contact to measure the coefficient of friction upon reaching a maximum holding torque. However, measuring the coefficient of friction between homogenous aluminium and laminated electrical steel material samples presents problems in terms of cost and manufacturing given the difficulty of making laminated cylindrical components to a defined range of shrink-fit pressures and conditions. Therefore, an alternative linear measurement method is employed.

#### 3.1 Experimental Apparatus

An overview of the linear friction experimental test apparatus is shown in Fig. 3. The apparatus is so arranged as to move, under a controlled strain rate, a flat aluminium sample (Al, 6082-T6) representing the housing inside diameter, relative to a static sample of laminated electrical steel, Fe, perpendicular to the aluminium sample. The bespoke test rig is built around a Roell Amsler uniaxial test machine, conventionally used for testing the mechanical properties of materials, for example generating stress-strain curves. The load representing the shrink-fit pressure is provided by a simple screw and ball

mechanism and is measured by an in-line calibrated load cell. The electrical steel samples are 10 x 10 x 10 mm cubes enabling the laminations to be aligned parallel or perpendicular to the direction of slip in order to give representative values for the coefficient of friction for holding torque (where the laminations are orientated in line with slip direction) or axial holding force (where laminations are aligned perpendicular to the slip direction). The use of laminated cube samples offer four faces to be measured per unit which reduces the time and cost to produce the samples. The load measured by the uniaxial testing machine represents the frictional load required to initiate slip and is used to deduce the coefficient of friction from (3) for a given normal load.

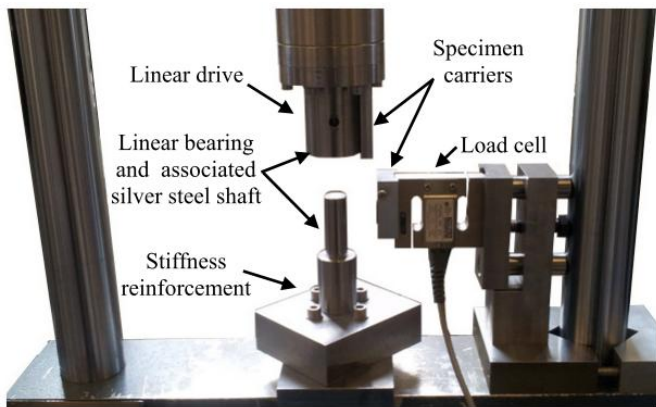


Figure 3: Coefficient of friction measurement apparatus.

An example load-displacement curve for 3 tests undertaken on laminated silicon iron electrical steel, (SiFe, 270M35) is shown in Fig. 4. The points of slip are highlighted as the maximum forces experienced before failure of the interface. These values are recorded as the maximum frictional force,  $F$ , for use in (3), given that the normal force,  $N$ , is set to correspond to the required interface pressure, in this case 2.5MPa.

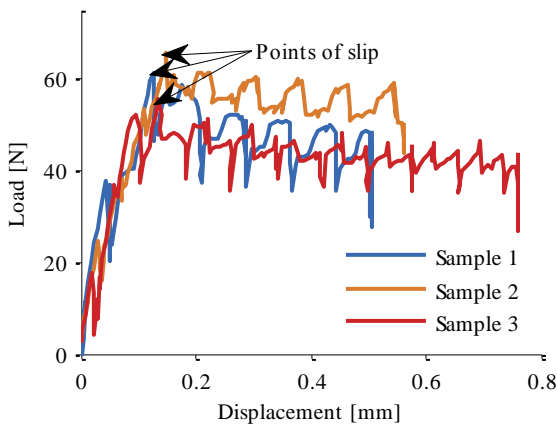


Figure 4: Example load-displacement plot for 3 tests at 2.5MPa interface pressure between SiFe and aluminium (across lamination direction).

## 4 Thermal Contact Conductance Measurement

In the analysis of heat conduction through multi-layer solids it is often assumed that perfect contact exists between each point on the mating surfaces. However, microscopically every surface exhibits asperities which form peaks and troughs across the interface as illustrated in Fig. 5. Hence, only a fraction of the total contacting area contributes to heat conduction while the remaining area typically behaves as thermally insulating voids, [4,7]. Under applied load plastic or elastic deformation of the contacting surfaces may occur, plastic deformation tends to increase the contacting area and therefore improves the heat conduction across the interface. In addition, interstitial material such as grease can fill the voids in the contact area and improve heat transfer. As a result, thermal contact conductance is a complex phenomena which is highly dependent upon the surface conditions and environment, hence, experimental measurement under representative in-service conditions is essential for the accurate prediction of the thermal performance of a shrink-fit.

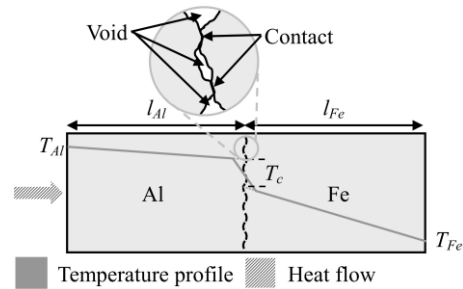


Figure 5: Illustration of Al and electrical Fe solids in contact at a microscopic level.

### 4.1 Experimental Apparatus

The thermal contact conductance across a material interface can be measured experimentally using a simple steady-state heat flow meter as illustrated in Fig. 5 along with an intuitive analogy to the electrical domain, [8].

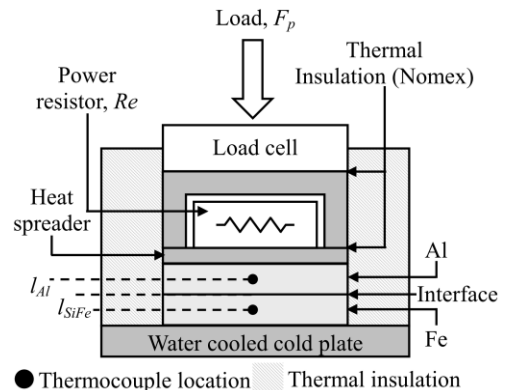


Figure 5: Steady-state heat flow meter.

Material samples are placed in contact and housed between a water cooled cold plate and a heated aluminium fixture. A static load is applied to the samples by a hydraulic press capable of applying up to 11 MPa of interface pressure. The

load is measured by an in-line 50kN Kistler load cell. In order to approximate one-dimensional heat flow, the experimental setup is thermally insulated using low thermal conductivity foam, in addition, the load cell and load transfer structure surrounding the heater are thermally insulated from the power resistor heater and heat spreader using 0.76 mm Nomex 410. It is assumed that the heat dissipated from the heater flows across the material sample interface and is extracted from the system by the water cooled cold plate. Thermocouples are embedded at known positions within the aluminium and the laminated electrical steel samples in order to measure the temperature differential across the interface. The thermocouples are calibrated over a 10 - 90 °C temperature range using a fixed temperature bath with a stability of 0.01 °C resulting in a thermocouple accuracy of  $\pm 0.1^\circ\text{C}$ . Under steady-state experimental conditions the applied load (interface pressure),  $F_p$ , supplied electrical power,  $Q$ , and temperature difference,  $T$ , are recorded using an Agilent 34972A data acquisition unit.

The thermal contact conductance,  $h_c$ , is determined through an electrical analogy where the sample material interface is represented as a thermal equivalent circuit, as illustrated in Fig. 6, [4,7]. The thermal resistance of the interfacing materials  $R_{Al}$  and  $R_{Fe}$  are defined by (4) where  $l$ ,  $A$  and  $k$  are the length, cross-sectional area and thermal conductivity respectively. The effective thermal resistance of the interface,  $R_c$ , is defined by (2).

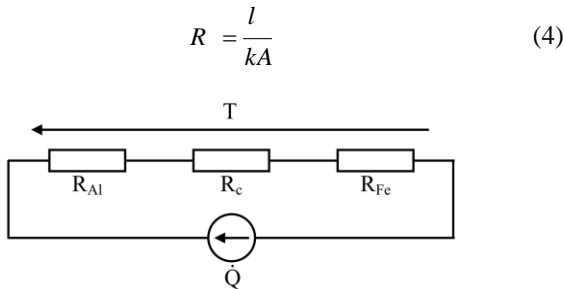


Figure 6: Thermal equivalent circuit of Al-Fe interface.

The current source,  $Q$ , represents the heat flow across the interface, supplied by a known electrical power dissipation in the heater, Fig. 5. The temperature,  $T$ , across the interface represents the temperature rise due to the heat flow. Solving the equivalent circuit for  $h_c$  gives (5) where  $A$ ,  $l_{Al}$ ,  $l_{Fe}$  are measured material sample dimensions, Fig. 5,  $k_{Al}$  and  $k_{Fe}$  are the thermal conductivity of the interfacing materials (previously measured using a similar experimental setup), [8], the temperature difference,  $T$ , and electrical heating power,  $Q$ , are experimentally measured.

$$h_c = \frac{1}{\frac{TA}{Q} - \frac{l_{Al}}{k_{Al}} - \frac{l_{Fe}}{k_{Fe}}} \quad (5)$$

As with the coefficient of friction measurement, material samples in aluminium, Al, and laminated electrical steel, Fe, are employed for experimental testing, however, in this case

the samples measure 66 x 66 x 7 mm, as illustrated in Fig. 7. A steel support structure is employed to prevent delamination of the electrical steel samples under high compressive load. The laminations are orientated along the direction of heat flow, as is typical of an electrical machine. Relatively small volume material samples are used in order to reduce manufacturing costs and minimise the time required to make a steady state measurement.

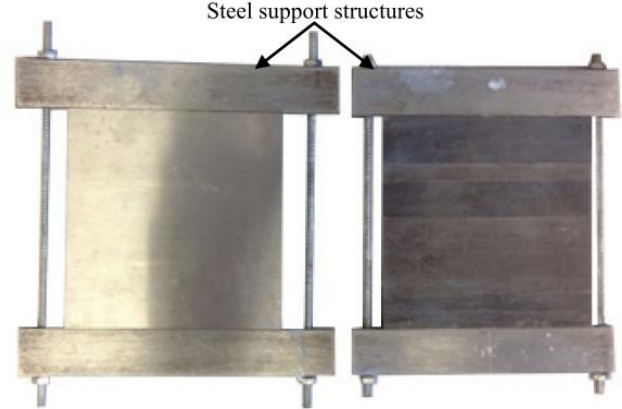


Figure 7: SiFe material samples ground (left) and Wire Electrical Discharge Machining (EDM) finish (right).

## 5 Experimental Results

Aluminium, Al, (6082-T6) and silicon iron, SiFe, (270M35) material samples were produced with two common surface finishes, a ground finish and a wire cut Electrical Discharge Machining (EDM) finish, Fig. 7. The material sample surfaces were measured using a Talysurf surface roughness tester, giving an average surface roughness of 0.02  $\mu\text{m}$  for the ground surfaces and 0.4  $\mu\text{m}$  for the EDM surfaces. The coefficient of friction and thermal contact conductance between ground-ground and EDM-EDM finished Al and SiFe samples were measured using the experimental apparatus detailed in Sections 3.1 and 4.1 in an interface pressure range of 0 - 10 MPa.

### 5.1 Coefficient of Friction

The measured coefficient of friction between Al and SiFe is shown in Fig. 8 for ground-ground and EDM-EDM surface interfaces. Each point on the graph is an average of three test results in order to reduce measurement error. The coefficient of friction measurements are undertaken with the laminations aligned with the slip direction, and are therefore representative of a stator-housing assembly providing electromagnetic reaction torque. As expected, the higher surface roughness of the EDM finish results in a higher value of friction, and therefore for the same interface pressure, would result in a higher holding torque capability than the ground surface. Interestingly, the coefficient of friction of the laminated SiFe and aluminium material combination appears to be increasing with interface pressure for both surface finishes over the 0 - 10MPa pressure range measured. Usually, over a much higher pressure range, the expectation is



that the coefficient of friction for two metals in contact would reduce with increasing pressure, [9-10]. This negative correlation is illustrated in Fig. 9, which includes historical data for SiFe samples with an EDM surface finish in contact with Al measured using an alternative technique, [2-3].

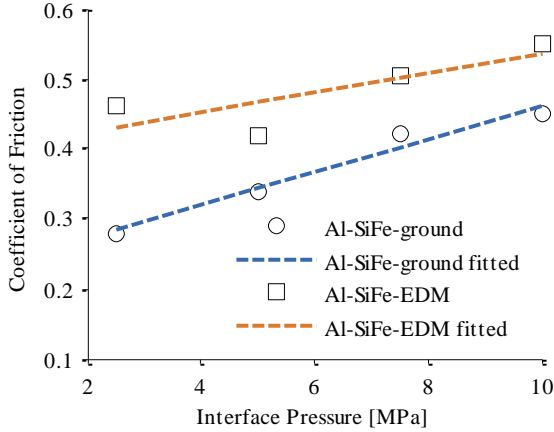


Figure 8: Coefficient of friction as a function of interface pressure for ground-ground and EDM-EDM finished laminated SiFe on Al (laminations in line with slip direction).

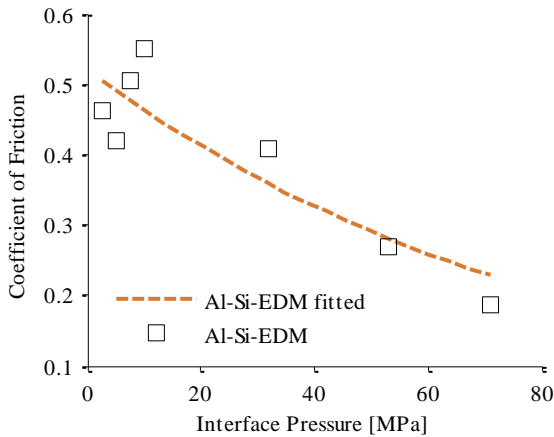


Figure 9: Coefficient of friction as a function of interface pressure for EDM-EDM finished laminated SiFe on Al (laminations in line with slip direction).

## 5.2 Thermal Contact Conductance

In the present thermal contact conductance measurement apparatus setup, it is assumed that the heat supplied electrically by the heater only flows across the material interface and that no leakage is present. In reality some leakage will occur across the interface between the hydraulic press and the heated fixture and to the surrounding environment, therefore the use of heat flux sensors or metering blocks is desirable to improve accuracy by measuring the heat flux crossing the boundary directly, [4,7]. In order to confirm the validity of the current experimental setup, the thermal contact conductance across the ground-ground interface of two Aluminium samples was measured and shown to lie within 15% of values reported in the literature at 2 MPa, as shown in Fig. 10.

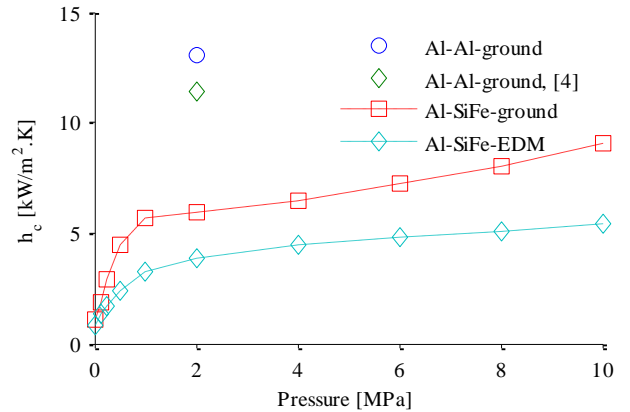


Figure 10: Thermal contact resistance as a function of interface pressure.

Fig. 10 presents the thermal contact conductance across the interface of ground-ground and EDM-EDM Al and SiFe material samples as a function of interface pressure. It is evident that the thermal contact conductance increases rapidly with interface pressure in both the ground and EDM cases until approximately 1.5 MPa and 2 MPa respectively where the thermal contact conductance begins to plateau and the rate of improvement with interface pressure becomes small. The difference in surface roughness of the ground, 0.02  $\mu\text{m}$  and the EDM, 0.4  $\mu\text{m}$  samples accounts for the difference in thermal contact conductivity with the smoother surface having a larger effective heat transfer area over the interface, [4,7].

## 5.3 Discussion

The mechanical output torque capability,  $T_{mo}$ , of an electrical machine can be estimated using (6) where  $D_r$ ,  $L$  and  $\sigma$  are the rotor diameter, active length and the air-gap shear stress respectively. The air-gap shear stress is given by (7) where  $B$ ,  $Q$  and  $K_u$  are the magnetic loading, electric loading and a dimensionless factor relating to the practical utilisation of the magnetic field and current sheet in a particular machine configuration, respectively. Typical values of  $\sigma$ ,  $B$ ,  $Q$  and  $K_u$  for various small to medium electrical machine types are given in Table 1, [11].

$$T_{mo} = \frac{\pi}{2} D_r^2 L \sigma \quad (6)$$

$$\sigma = K_u B Q \quad (7)$$

Machine Type	$K_u$	B [T]	Q [kA/m]	$\sigma$ [kPa]
Brushed DC	1.00	0.70	20	14.0
Induction (IM)	0.81	0.57	32	14.7
Inverter fed IM	1.00	0.57	32	18.4
Synchronous	1.00	0.64	47	30.4
Brushless DC	0.94	0.90	50	42.3
Switched reluctance	1.29	0.30	50	19.4

Table 1: Typical design values for various electrical machine types, [11].

The stator-housing shrink-fit must provide a holding torque,  $T_h$ , (1), greater than the motor output torque  $T_{mo}$ , (6), giving (8) where  $D_s$  is the stator outer diameter. Defining the split ratio (ratio of rotor to stator diameter) as  $\delta$  in the range  $0 > \delta < 1$ , (8) reduces to (9).

$$T_{mo} = \frac{\pi}{2} D_r^2 L \sigma < T_h = \frac{\pi}{2} D_s^2 L P \mu \quad (8)$$

$$\delta^2 \sigma < P \mu \quad (9)$$

In the worst case (analytically) where the split ratio,  $\delta$ , is close to 1, (9) shows that the product of the shrink-fit interface pressure,  $P$ , and the coefficient of friction,  $\mu$ , must exceed the air-gap shear stress of the electrical machine type under consideration. The fitted curves presented in Fig. 8 indicate that the  $P\mu$  product exceeds the maximum air-gap shear stress presented in Table 1 at a shrink-fit interface pressure of 125 kPa for the EDM finish and 217 kPa for the ground finish. However, at such low interface pressures the thermal contact conductance is comparatively low, Fig. 7. A factor of 3 improvement in thermal contact conductance is achieved with an interface pressure  $> 2$  MPa, conveniently resulting in a substantial factor of safety on the required holding torque. As the electrical machine temperature rises during operation, the stator and housing components will undergo thermal expansion in accordance with (10), where  $\Delta$ ,  $\alpha$ ,  $D$  and  $T$  are the change in diameter, the linear coefficient of expansion, the original component diameter and the change in temperature respectively.

$$\Delta = \alpha D T \quad (10)$$

As the linear coefficient of expansion of aluminium is greater than that of SiFe, the shrink-fit pressure at the stator-housing interface will reduce with increasing temperature. Therefore, the ambient temperature shrink-fit must be designed to operate at an interface pressure  $\gg 2$  MPa such that the interface pressure will remain  $> 2$  MPa up to the maximum operating temperature of the electrical machine and maintain good thermal and mechanical holding performance. However, if the target shrink-fit pressure is too high at the ambient temperature then the temperature required to expand the housing over the stator during assembly may have a detrimental effect on the mechanical properties of the materials. In addition it has been shown that core loss increases with shrink-fit pressure which is detrimental to the efficiency of the machine as well as the thermal performance, [12].

## 6 Conclusion

Experimental apparatus designed to measure the coefficient of friction and thermal contact conductance in the range 0 - 10 MPa under representative in-service conditions is presented. Analysis shows that shrink-fit pressures  $> 10$  MPa, although not uncommon in general engineering practice, are difficult to justify for small and medium electrical machine housings. It is evident that increased sample sizes and higher

resolution in the 0 - 10 MPa interface pressure range is required in order to fully characterise the complex friction and thermal contact conductivity properties.

Given sufficiently accurate pressure dependent coefficient of friction, thermal contact conductance and core loss data, it is possible to determine an optimal shrink-fit pressure which will provide adequate holding torque, present minimal thermal contact resistance and minimise the effect on core loss. Thereby enabling an overall reduction in the required volume and mass of housing material. To this end, future work will focus on improving the accuracy of the experimental apparatus, investigating the temperature sensitivity of the coefficient of friction and thermal contact conduction, [13], and on the measurement of a statistically significant number of material samples.

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