PERFORMANCE OF A CARBON/CARBON COMPOSITE CLUTCH DURING FORMULA ONE RACE START CONDITIONS

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Summary: A single clutch-plate interface dynamometer (SCID) was designed and commissioned to facilitate friction and thermal performance testing of single clutch-plate pairs. Narrow (~2mm) high-temperature (1300-1650°C) hot bands were observed during SCID tests at high rotational speeds and clamp loads typical of race starts. Migration of the hot bands occurred between successive engagements but not during single engagements. Scanning electron microscope (SEM) images showed that the thickness of the friction film formed on the friction surfaces appeared to become thicker as the amount of friction work increased. A thermomechanically coupled finite element analysis (TCFEA) was then developed by coupling Matlab and Abaqus to simulate the thermo-mechanical response of a single clutch-plate pair during SCID testing. With allowance for wear, the model predicts high degrees of contact localisation resulting in a single distinct hot band of comparable temperature and width to those recorded during SCID testing.

1 INTRODUCTION

Blanco et al. [1] present a literature review summarising research into carbon/carbon (c/c) composite brake materials. They note that published papers are limited due to much of the research in this area being carried out by industry and is thus protected by patents. This review also highlights that most published research is associated with the development of aircraft brakes, a point reiterated by Savage [2]. Aircraft brakes are similar in construction to the multi-plate c/c clutch used in this investigation in that friction is generated via disc-on-disc contact. The fact that they are also used in high energy applications such as aborted take-offs means that much of the research findings can be related directly to multi-plate clutches.

The friction performance of c/c composite materials in ambient conditions has been investigated by several authors [1, 3, 4, 5] and the same general trends have been observed in relation to the variation of the friction coefficient (COF) with surface temperature. This, together with surface morphology examination, has enabled three distinct friction regimes to be identified. Initially the COF is low as type I morphology (thin, smooth film) is dominant due to water being present in the friction surface. At approximately 150°C, the water is desorbed from the surface and a rapid increase in COF and temperature is observed as type II morphology (rough, powdery debris) is formed. As energy continues to be dissipated at the friction surface, type II morphology is converted to type III morphology (thick, smooth film). The COF falls from its peak value and remains fairly constant at a higher value than the COF at ambient temperature. The relationship between surface morphology, COF and temperature is however more complex and the friction behaviour of c/c composites has also been shown to be strongly affected by the level of energy input [3, 6, 7, 8, 9], friction surface condition [10], ambient conditions [3, 5, 11, 12, 13] and fibre type/orientation [14, 15, 16].

In sliding contact, the phenomenon of thermoelastic instabilities (TEI) can also affect friction material performance. Barber [17] states that any irregularities on the contacting surfaces of sliding solids will cause a non-uniform contact pressure distribution. The heat input distribution mirrors this non-uniformity and lead to preferential thermal expansion in the high-temperature areas. The contact pressure then increases in these areas, concentrating the heat input and causing further temperature and thermal expansion increases. The thermal distortion thus tends to exaggerate the original surface profile. Barber's [17] computational model showed that a height difference of as little as 1nm is sufficient to initiate TEI.

The aim of the present investigation is to establish the cause of the unstable and inconsistent torque output during race starts of a PAN-CVI carbon/carbon multi-plate clutch, as reported by many F1 teams.

2 SINGLE CLUTCH-PLATE INTERFACE DYNAMOMETER (SCID)

A unique single clutch-plate interface dynamometer (SCID) was developed to facilitate friction and thermal performance testing of single clutch-plate pairs, eliminating the complexities of the full multi-plate clutch system. The SCID was designed to replicate typical race start rotational speeds, clamp loads and total energy dissipation levels. In the SCID, unlike in the race car where the initially stationary clutch plates (gearbox side) would be brought up to the same speed as the rotating clutch plates (engine side) during the race start, one clutch plate in the pair is non-rotating and the rotating clutch plate is brought to rest.

Thermocouples were unsuitable for measuring friction surface temperatures in this application for several reasons. Full area contact between the clutch plates negated the use sliding thermocouples whilst embedded thermocouples can only measure temperatures close to the friction surface but not of the actual friction surface. The inherent measurement lag of thermocouples also meant that any rapid transient thermal events may not have been captured. A FLIR X6540SC high-speed thermal imaging camera was therefore used to measure the friction surface temperatures. The setup of the camera is shown in Figure 1. A Ø12mm hole was machined through the non-rotating assembly so that, via use of an infrared mirror set at 45°, the friction surface of the rotating clutch plate could be viewed directly by the thermal imaging camera. The Ø12mm hole is centred at the mean geometric radius of the friction surface allowing 75% of the 16mm wide friction surface to be imaged.

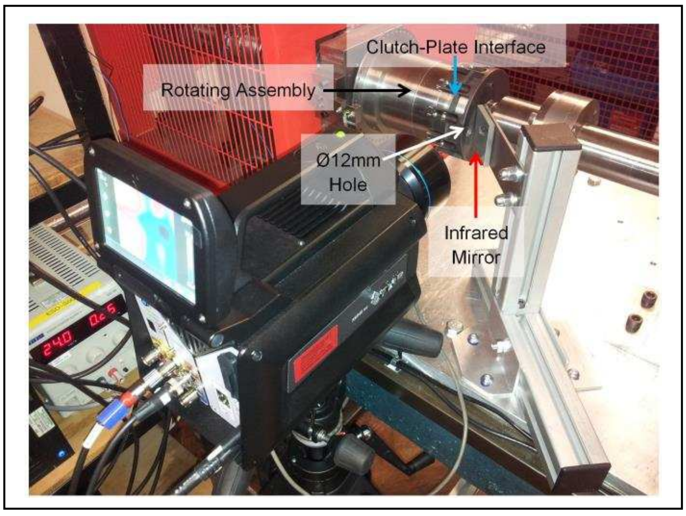

Figure 1 – Thermal Imaging Camera Setup for SCID

The FLIR X6540SC thermal imaging camera is able to measure temperatures of up to 1500°C but cannot do so continuously from room temperature. The camera's 300-1500°C temperature range was used in this investigation as high transient temperatures were anticipated. Due to the noise associated with using this temperature range, the lowest temperature that the camera was able to measure was 450°C. As the emissivity of the clutch-plate material was unknown, an emissivity value of 1 was assumed in order to process the results. An emissivity value of 1 effectively produces the lowest possible temperatures that the clutch plate friction surfaces may achieve (i.e. a conservative measurement) [18]. The true emissivity of the clutch plates is likely to be close to unity as the black carbon/carbon material has a very dark appearance so the assumption of an ideal black body is reasonable.

Table 1 shows the combinations of initial rotational speed and clamp load investigated and the nomenclature used to identify them. For each speed/load combination, a new clutch-plate pair was used with seven consecutive engagements carried out for each pair. The clutch plates were allowed to cool for ten minutes between engagements. The speed/load combination B4 (8000rpm/1400N) is representative of typical race start conditions.

Initial rpm/Total Energy Dissipation (kJ)	**Clamp Load (N)**		
	1000	**1200**	**1400**
7000/6.74	A2	A3	A4
8000/8.80	B2	B3	B4

Table 1 – Initial Rotational Speed/Clamp Load Combinations Used in SCID Tests

3 SCID RESULTS

3.1 TORQUE OUTPUTS

Figures 2a and 2b show the torque outputs for clutch-plate pairs A2 and B4 which were respectively subjected to the lowest and highest speed/load combinations. For both figures, the first 0.8-1.0s of the time axis represents the build-up of the clamp load and hence the delay before the torque ramps up from zero at about 0.2s. At the low speed/load combination the torque output was both stable and consistent but at the high speed/load combination the torque output was both unstable during single engagements and inconsistent between engagements. It is important to note that the timescales for each figure are different and it can clearly be seen that the engagement times for the low speed/load combination tests were much higher than for the high speed/load combination. This is in essence the performance problem associated with the clutch in that stable, consistent torque comes with the drawback of long engagement times but high torque, and hence short engagement times, results in unstable and inconsistent torque output. As the initial rotational speed, clamp load or both were increased, the stability and consistency of the torque output was observed to decrease.

In relation to the surface morphology regimes of the carbon/carbon clutch plates, the stable torque output shown in Figure 2a suggests a predominance of type I morphology resulting in a low COF throughout the engagement. This indicates that the power dissipation level at this particular low speed/load combination was insufficient to bring about a change from type I to type II morphology. The torque output behaviour shown in Figure 2b however suggests that the type I morphology has been converted to type II and then to type III. This indicates that the power dissipation level at this particular high speed/load combination was sufficient to initiate changes in surface morphology. This transitional behaviour was particularly evident during Engagement1 in Figure 2b where the torque output was initially low but rose sharply during the course of the engagement (at ~1.6s), indicating that type II morphology had formed which greatly increases the COF.

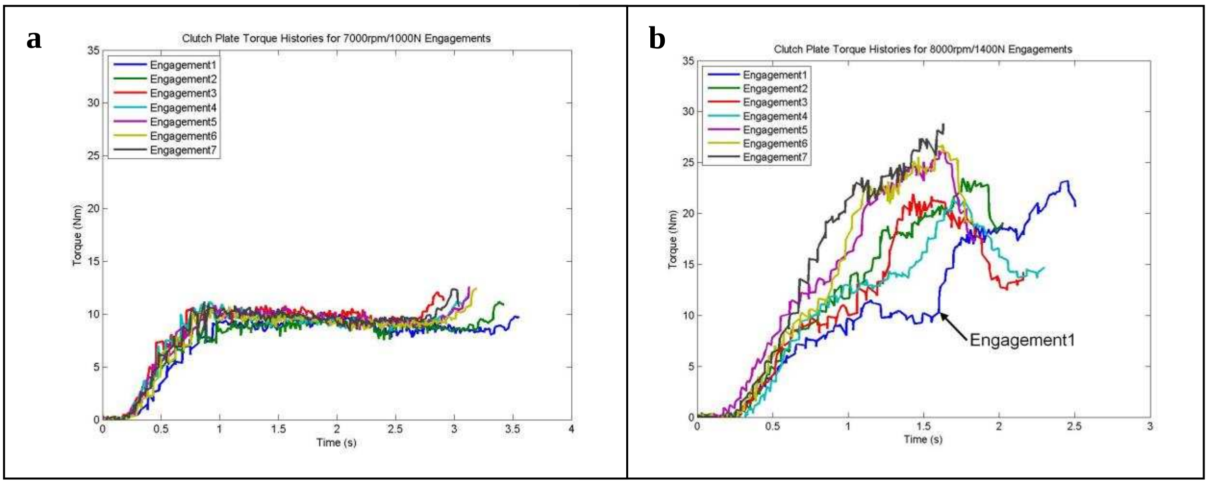

Figure 2 – SCID Torque Output Results for (a) 7000rpm/1000N and (b) 8000rpm/1400N Speed/Load Combinations

3.2 TEMPERATURE MEASUREMENTS

The clutch-plate friction surface temperatures recorded by the thermal imaging camera showed a vast difference between recorded during the lowest speed/load combination (A2) and the highest speed/load combination (B4). Figure 3 shows the clutch-plate friction surface temperature profile for Engagement4 at 7000rpm/1000N (A2) when the maximum temperature was recorded. The white circle superimposed on the image indicates the circumference of the Ø12mm viewing hole. Sp1 and Sp2 (label partially covered by temperature scale) respectively indicate the location of the innermost and outermost radial points viewed whilst Sp3 indicates the point where the maximum temperature occurred.

It can clearly be seen from Figure 3 that the surface temperature profile at this point was non-uniform across the radial span of the friction surface. Towards the inner radius, the friction surface temperature was ~475°C whilst the maximum temperature of ~550°C occurred just beyond the mid-radius position. The temperature difference of around 75°C is too small for the region of higher temperature to be classed as a distinct hot band.

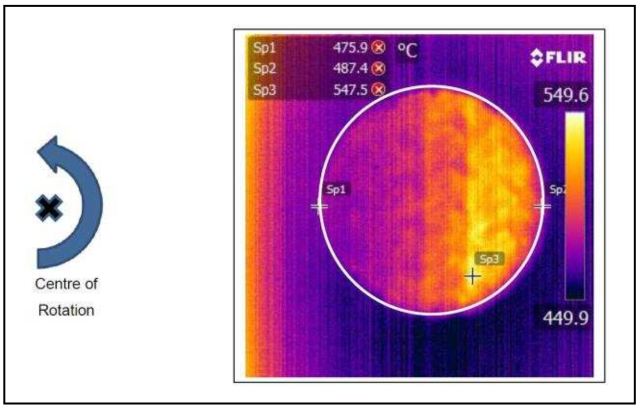

Figure 3 – Maximum Clutch-Plate Friction Surface Temperature Recorded During Engagement4 at 7000rpm/1000N Speed/Clamp Load Combination (A2)

Figure 4 shows the temperature profile for Engagement5 at 8000rpm/1400N (B4) when the maximum temperature was recorded. In contrast to the surface temperature profile shown in Figure 3, Figure 4 shows a distinct hot band on the friction surface. The extremely high temperatures (1300-1575°C) existed in a narrow band approximately 2mm wide where the temperatures outside of the band were at around only 500-600°C.

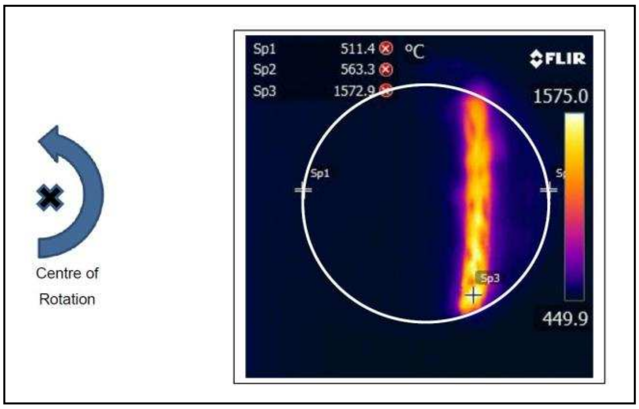

Figure 4 – Maximum Clutch-Plate Friction Surface Temperature Recorded During Engagement5 at 8000rpm/1400N Speed/Clamp Load Combination (B4)

For all engagements carried out at the 7000rpm/1000N speed/load combination (A2), a similar surface temperature profile to that shown in Figure 3 was observed with maximum temperatures of 550°C-600°C. A similar surface temperature profile as that shown in Figure 4 was measured for all clutch-plate engagements carried out at 1400N for both speed combinations (A4 & B4) with maximum temperatures of 1300°C-1650°C.

The very high localised hot band temperatures observed during the clutch-plate engagements representative of race start conditions, suggest that the majority (if not all) of the friction work is done in the areas of the hot bands. It is therefore reasonable to assume that the radial position of the hot band represents the location of an effective friction radius (EFR). Torque (T) is directly proportional to clamp load (P), COF (μ), and EFR (r_e) and hence if the EFR migrates, the torque output will vary even if the COF remains constant:

$$T = P\mu r_e \tag{1}$$

Upon examining the evolution of the hot band shown in Figure 4 it was observed that the hot band formed at approximately 1s into the clutch-plate engagement and persisted at a high temperature for approximately 0.6s. Given that the clutch-plate engagement took less than 2s, the hot band persists for a significant period of the engagement. The critical observation however was that the hot band, and hence EFR, did not migrate radially during the engagement, a result observed for all clutch-plate engagements at 1400N (both speeds).

However, the hot bands were observed to migrate between successive engagements to the extent that their radial location was different for all seven engagements for clutch-plate pairs A4 and B4. For this particular clutch plate design, the outer radius dimension is almost 50% greater than the inner radius dimension and hence if the EFR migrated from near the inner radius to near the outer radius between successive engagements, a torque increase of almost 50% would result even if the surface morphology and hence COF remained constant.

The lack of EFR migration during single engagements showed that thermal expansion remains dominant during the very short engagement times. However, wear of the friction surfaces must occur for the EFR to migrate between successive engagements. The level of wear during a single engagement is insufficient to overcome thermal expansion but the wear then results in a surface recess once the clutch plates have cooled and this forces contact to be established elsewhere during the subsequent engagement.

4 FRICTION SURFACE CHARACTERISATION

Prior to SCID testing, a circumferentially arbitrary region of each clutch plate friction surface was imaged using scanning electron microscopy. A Carl Zeiss EVO MA15 SEM was used at a magnification value of x100. Several small regions of the clutch plate friction surfaces were scanned using SmartSEM and then stitched together using SmartStitch to produce images of the friction surface across the full radial width. Backscattered electrons (BSE) produced the best contrast images. The same regions of the clutch plates were imaged after SCID testing which allowed a direct comparison of the surface structure to be made.

Figure 5 shows the SEM images for the stationary clutch plate of clutch-plate pair A3 before and after SCID testing. Several areas have been highlighted which clearly illustrate changes that occurred due to friction work. A visible friction film was established on the friction surface after SCID testing and it was observed that the apparent film thickness increased with as either the initial rotational and/or clamp load were increased and hence the friction film thickness was also observed to increase with the maximum friction surface temperature recorded.

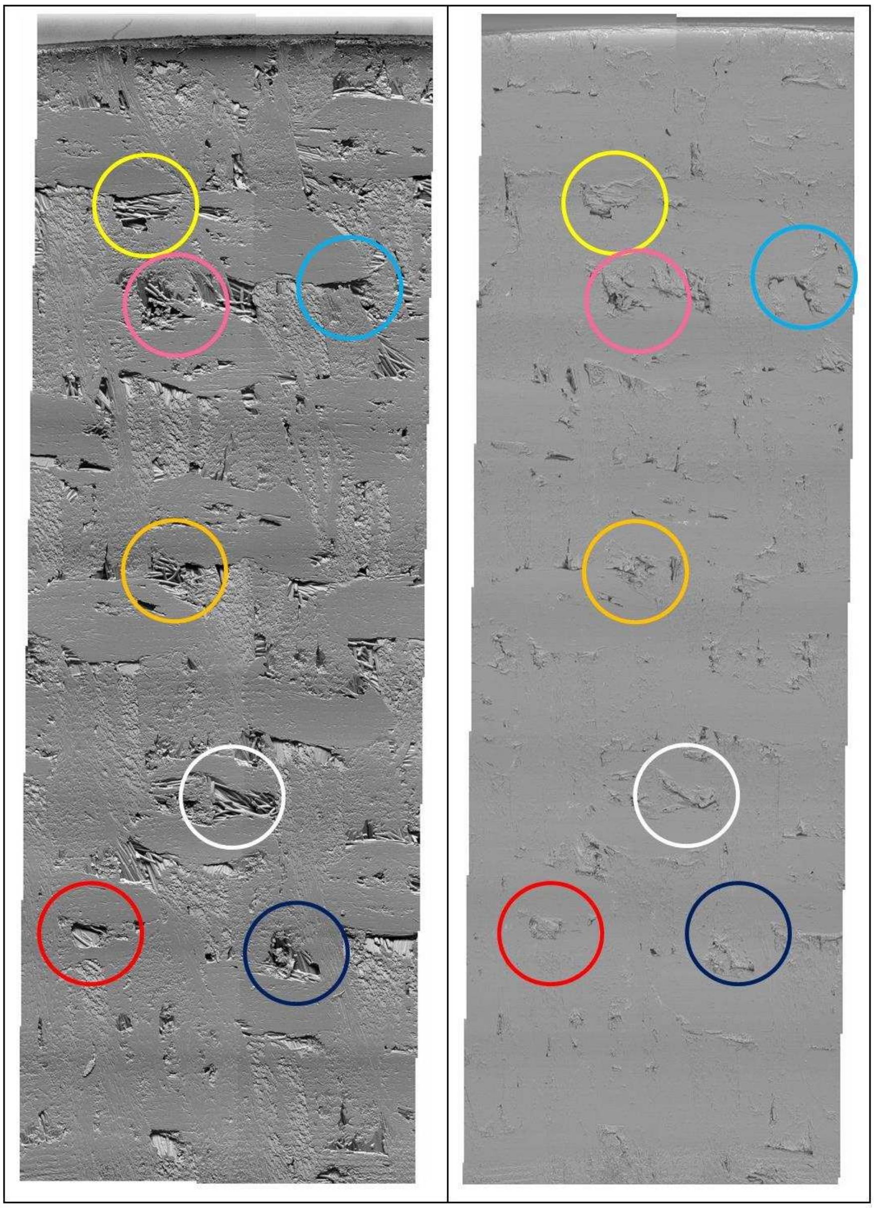

Figure 5 – SEM Images of Clutch Plate A3-DN Before (Left) and After (Right) SCID Testing (x100 Magnification)

It was also observed the carbon fibres appeared more distorted after tests at higher speed/load combinations and hence maximum friction surface temperature than for the lower speed/load combinations. For clutch-plate pairs A4, B3 and B4, distinct wear tracks were formed on the clutch-plate friction surfaces which could clearly be seen under visible light. The wear tracks on clutch-plate pairs A4 and B4 were more distinct than on B3 which correlates with the maximum temperatures recorded for clutch-plate pairs A4 and B4 being higher than those recorded for clutch-plate pair B3. Figure 6 shows an SEM image of the wear track on the stationary plate of clutch-plate pair A4 where distorted fibres have been highlighted. In comparison to the fibres shown in Figure 5 for stationary clutch plate A3 before SCID testing (left), the top sections of the fibres shown in Figure 6 appear to have been sheared off. At the high hot band temperatures observed, oxidation of the carbon/carbon composite material would be expected but an increased level of abrasive wear would also occur due to the greatly increased normal loads arising from the increased levels of thermal expansion in the area of the hot band. It is therefore unclear whether the wear tracks were the result of oxidation, abrasive wear or a combination of both.

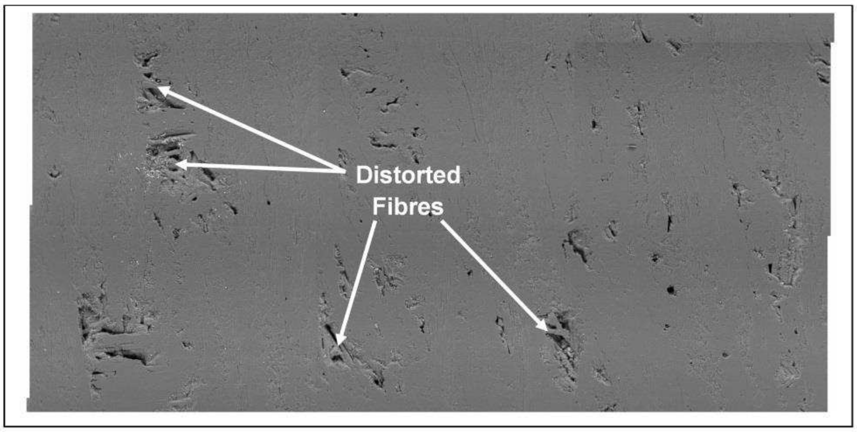

Figure 6 – SEM Image of Wear Tack on Friction Surface of Clutch Plate A4-DN After SCID Testing (x100 Magnification)

5 THERMOMECHANICALLY COUPLED FINITE ELEMENT ANALYSIS (TCFEA)

The TCFEA was developed by coupling the commercially available software packages Matlab and Abaqus to simulate the mechanical and thermal response of the clutch plates during SCID engagement tests. The first stage in the analysis was to construct an axisymmetric finite element model of the clutch plates in Abaqus. Figure 7 shows a schematic of the finite element model used. The analysis type used was coupled temperature displacement using quad CAX4T coupled temperature-displacement elements.

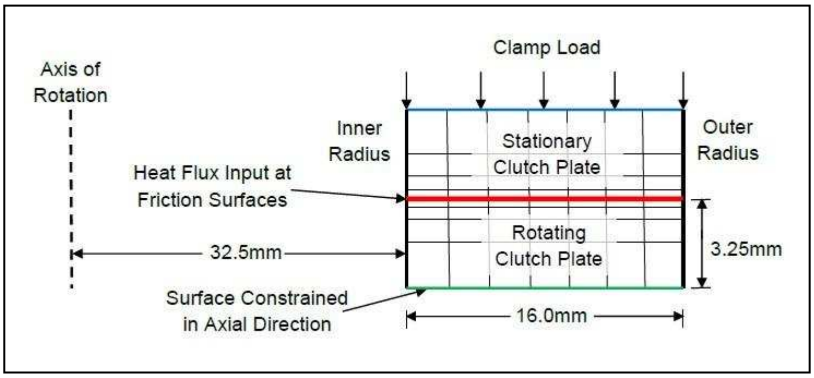

Figure 7 – Axisymmetric Finite Element Model of Clutch-Plate Pair (Both clutch plates are 3.25mm thick)

A bias ratio of 2.0 was used such that the mesh density in the axial direction was twice as dense at the friction surfaces than at the non-friction surfaces. A mesh of 201 nodes in the radial direction and 21 nodes in the through-thickness direction (per clutch plate) was used with a time step of 0.01s.

A simple relationship between temperature and COF was assumed in the analysis. These values, which have been taken from previous dynamometer tests, are listed in Table 2. The COF values are linearly interpolated between the reference values and the COF is assumed to remain constant at a value of 0.5 at temperatures above 1200°C.

Temperature (°C)	**COF**
25	0.20
200	0.25
400	0.30
600	0.35
800	0.40
1000	0.45
1200	0.50

Table 2 – Temperature-COF Reference Values Used in TCFEA

5.1 NUMERICAL PROCEDURE

Before the full analysis was carried out, Matlab was used to modify the Abaqus input (.inp) file to assign an individual surface definition to each friction surface element, define each node on the slave friction surface as an individual set, assign heat flux inputs to each defined friction surface and add print commands to output contact pressure and temperature results for the defined nodes to the Abaqus data (.dat) file.

Another Matlab script was then used to carry out the analysis in a loop. First the Abaqus job was submitted simulating a single time step and once it had been completed, the contact pressure and friction surface temperature results were read from the .dat file and stored in matrices within Matlab. The contact pressure was then used to calculate the normal force acting on each friction surface element which, along with the coefficient of friction calculated based on the surface node temperatures, was then used to calculate the torque generated by the clutch-plate pair. The rotational speed reduction of the rotating clutch-plate during the next time step was then calculated allowing the energy dissipation to be calculated. This was converted to a heat flux value which was apportioned across the friction interface to reflect the contact pressure distribution. An additional step was then appended to the Abaqus .inp file with the calculated heat flux input values and the Abaqus job resubmitted. This process was repeated until the rotating clutch plate had become stationary.

The friction surfaces were modelled as initially perfectly flat resulting in an initially uniform contact pressure across the friction interface. The initial speed and clamp load inputs were 8000rpm and 1400N, replicating the typical race start speed/load combination (B4) investigated during the SCID engagement tests. The clamp load application was modelled to ramp up over a time period of one second as it did during SCID testing and the initial temperature of the clutch plates was set to 25°C (298K). The analysis was carried out until the relative rotational speed of the clutch plates was reduced to zero.

5.2 WEAR EQUATION

The results of the SCID testing suggested that wear must occur in order for the effective friction radius to migrate between successive clutch-plate engagements. Zhao et al. [19] used a wear equation based on Archard's wear law [20] relating the incremental wear (Δh) to contact pressure (p), sliding velocity (u), time (Δt) and a material and application specific wear constant (k):

$$\Delta h = kpu\Delta t \tag{2}$$

However, it was also observed in the present SCID testing that wear tracks on the clutch plates became more apparent as the rotational speed and clamp load were increased and hence they also became more apparent as the maximum friction surface temperatures increased. The wear is likely to have been due to abrasion, oxidation or a combination of both. As oxidation wear would be expected to exponentially increase with temperature [13, 21, 22], an exponential term with relation to surface temperature (Z) and reference temperature (Z_o) was added to Equation 2 to give the wear equation used in the TCFEA:

$$\Delta h = kpu\Delta t e^{(Z/Z_o)} \tag{3}$$

The incremental wear calculated for each time step was then divided by the time step value to give a dimensional wear rate, w, during that time step as shown by Equation 4.

$$w = \frac{\Delta h}{\Delta t} \tag{4}$$

This wear rate was incorporated into the TCFEA through use of the adaptive meshing function in Abaqus. Using the velocity control method, each node along the friction interface was assigned an adaptive mesh definition during the stage of the TCFEA when a new step is added. The velocity control method allows a velocity to be assigned to each node such that the finite element model moves the node at that velocity to simulate wear whilst the node can still displace due to thermal and mechanical effects. The aggregate effects of wear, thermal expansion and mechanical loading are thus calculated for each node at the interface.

6 TCFEA RESULTS

Figure 8a shows the contact pressure distribution and surface temperature profile at 0.40s into the engagement. Contact is initially lost at the inner and outer radii due to radial expansion of the clutch plates. The effect of this loss of contact at the inner and outer radii is to concentrate heat flux towards the centre of the friction interface which affects the temperature profile and hence the thermal expansion of the friction surfaces. The contact pressure distribution is then influenced by the non-uniform thermal expansion across the friction interface and the phenomenon of thermoelastic instabilities is established, exacerbating the effect. After 0.40s two areas of high contact pressure have been formed as the result of contact becoming isolated to two small areas. This non-uniformity is mirrored in the temperature profile as more heat is input to these areas.

Initially the simulation was run without the allowance for wear and predicted a similar evolution of the contact pressure distribution and temperature profile as that shown in Figure 8a. The two hot bands formed at this stage were predicted to persist for much of the remainder of the clutch-plate engagement, However, in the simulation with allowance for wear, at the elevated surface temperatures wear becomes significant and the two contact areas shown in Figure 8a are worn away forcing contact to move further towards the centre of the friction interface. Contact is eventually isolated virtually to a single contact area approximately 1mm wide as shown in Figure 8b. This results in the formation of a single narrow high-temperature hot band of approximately 1550°C. The radial and through-thickness temperature gradients were similarly extreme to those along the friction interface with the bulk temperature of the clutch plates just 0.25mm directly below the hot band being approximately 800°C lower.

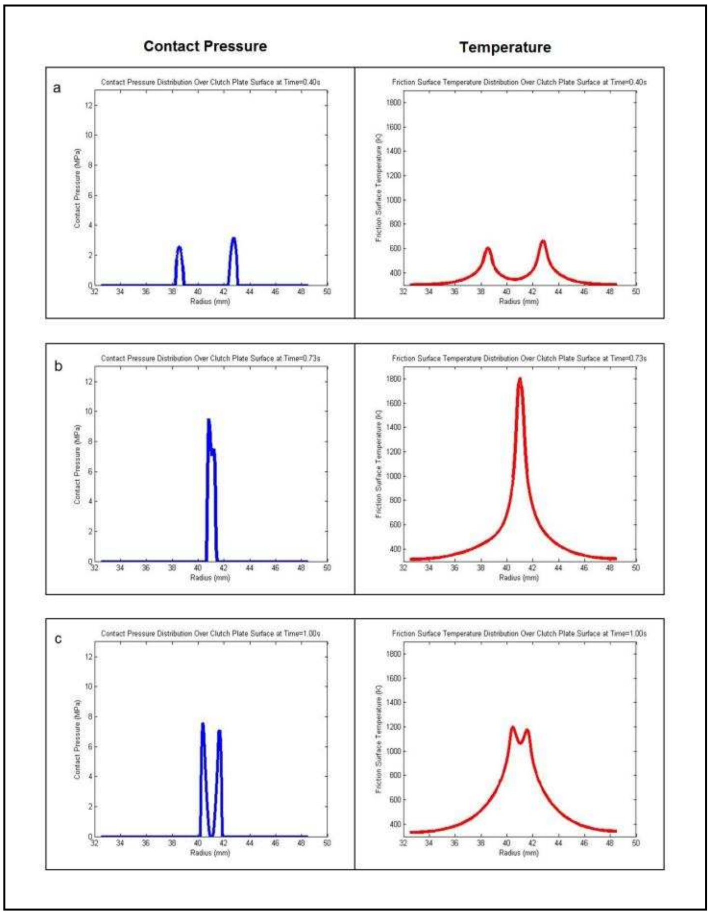

Figure 8 – Contact Pressure Distribution and Temperature Profile Across Friction Interface at (a) t=0.40s, (b) t=0.77s and (c) t=1.00s (Wear Model)

After 0.77s when the maximum friction surface temperature occurs, the amount of heat input becomes less than the rate of heat conduction away from the contact area, both radially and in the through-thickness direction. As a result, the temperature of the hot band reduces and the width of the hot band increases. The reduced level of thermal expansion combined with wear of the single central contact area leads to two contact areas being re-established, resulting in the contact pressure and temperature distributions shown in Figure 8c.

As the engagement progressed further, contact was re-established over an even larger area and the temperature gradients in both the radial and through-thickness direction reduce until the engagement is complete after 1.38s.

7 DISCUSSION

The stable, consistent torque behaviour observed during the lowest speed/clamp load combination (Figure 2a) suggests a predominance of type I surface morphology with no transition to type II morphology after water desorption. This is despite maximum friction surface temperatures of 550-600°C measured on the friction surfaces. The transition from type I to type II morphology would be expected at 150°C but it is likely that the short engagement time is insufficient to allow all the water to be desorbed.

The initial surface profile non-uniformities arising from machining of the clutch plates led to the phenomenon of thermoelastic instabilities occurring. The SCID results showed that as the initial rotational speed and/or clamp load is increased, the power dissipation at the friction interface increases and the effect of TEI and hence hot banding becomes stronger.

The TCFEA analysis however predicts high levels of contact localisation and hot banding even when the friction surfaces of the clutch plates are modelled as initially perfectly flat. Initial contact loss at the inner and outer radii leads to contact pressure increases towards the centre of the friction interface. For the non-wear model this results in two contact points either side of the centre of the friction interface leading to two distinct hot bands whereas the simulation including wear predicts a single centralised hot band. In this respect, the finite element model incorporating wear most closely replicates the SCID results.

The narrow, high-temperature hot bands (1300-1650°C) observed during SCID tests at the typical race start speed and load combination (8000rpm/1400N) indicate that a high degree of contact localisation occurs. The hot bands were measured to be approximately 2mm wide suggesting that as little as 12.5% of the friction surface areas may actually be in contact. One limitation with the thermal imaging camera was that it could not measure temperatures below 450°C. Considering this cut-off temperature, Figure 8b shows that the TCFEA simulation incorporating wear predicts a hot band also approximately 2mm wide where the model simulates the same speed/load combination used in the SCID tests. The maximum simulated temperature of 1550°C also agrees strongly with the SCID results and, as with the SCID results, the hot band was not predicted to migrate radially during a single engagement. Although the hot bands were shown and predicted to be approximately 2mm wide, Figure 8b shows that the actual contact area (predicted by the TCFEA) may momentarily be as little as 1mm wide.

The high degree of contact localisation during the race start condition SCID tests implies that the radial location of the hot band indicates an effective friction radius (EFR). The fact that neither the clamp load nor the EFR vary during a single engagement (according to both SCID and TCFEA results) shows that torque instability during a single engagement is due to surface morphology effects alone, leading to variations in COF (Equation 1). The torque inconsistency between engagements is however due to a combination of both surface morphology effects and effective friction radius migration. As well as influencing the EFR, if contact moves to a different area of the clutch-plate friction surface between successive engagements, it may move to an area with a different initial surface morphology, and hence COF, thus potentially resulting in unexpected torque output behaviour leading to a less than optimal race start.

The migration of the EFR between engagements observed during SCID tests indicates that wear of the friction surfaces is occurring. Without wear, the same contact point would persist between engagements and no EFR migration would occur. Due to the short clutch-plate engagement times (<3s), thermal expansion remains dominant over wear such that no EFR migration occurs during single engagements. However, once the clutch plates have cooled to ambient temperature, the wear that occurred at the previous contact point results in a surface recess; contact is thus established elsewhere during the next engagement which forces the change in EFR.

8 CONCLUSIONS

A single clutch-plate interface dynamometer (SCID) was designed and commissioned to facilitate friction and thermal performance testing of single clutch-plate pairs. The high energy levels involved during SCID tests representative of race start conditions resulted in narrow (~2mm) hot bands of very high maximum temperatures (1300-1650°C).

The TCFEA confirmed the observed hot banding behaviour. At race start input levels, the TCFEA has shown that wear of the friction surfaces causes contact to be isolated to a single contact area resulting in a single, distinct, 2mm wide hot band with a maximum temperature of ~1550°C. These hot bands are formed despite the friction surfaces being initially perfectly flat.

Both the SCID and TCFEA have shown that hot bands do not migrate during single clutch-plate engagements but do migrate between successive engagements. The clutch-plate torque output instability during single engagements is therefore due to surface morphology effects alone whilst the torque output inconsistency between successive engagements is due to a combination of both surface morphology effects and effective friction radius migration.

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