Understanding the Contributions of Surface Morphology Transitions and the Phenomenon of Thermoelastic Instabilities on the Torque Output of a Carbon/Carbon Multi-Plate Clutch During Race-Start Conditions

This paper is a revised version of a paper entitled 'Carbon/carbon friction pair performance – towards an understanding of surface morphology and coefficient of friction variation in a high energy clutch application' presented at EuroBrake 2013, Dresden, Germany, 17–19 June, 2013.

This paper is a revised version of a paper entitled 'Performance of a Carbon/Carbon Composite Clutch During Formula One Race Start Conditions' presented at the International Conference on Automotive Composites (ICAutoC) 2016, Lisbon, Portugal, 21-23 September 2016.

Part of the work presented in this paper is also presented in a paper entitled 'Characterisation of a Carbon/Carbon Multi-Plate Clutch for a High Energy, Race Car Application' published in the International Journal of Vehicle Performance, Volume 2, No.3, 2016.

Abstract

The torque output of a carbon/carbon multi-plate Formula One clutch during race starts has proved to be both unstable and inconsistent. A one-dimensional heat transfer model utilising a Taguchi analysis suggested that a close interdependency exists between surface temperature, surface morphology and coefficient of friction thus affecting torque output stability. Friction surface examination showed that a non-uniform friction surface height may lead to thermoelastic instabilities and effective friction radius migration which directly affects torque output. A single clutch-plate interface dynamometer confirmed the formation of narrow (~2mm), high-temperature (1300-1650℃) hot bands during tests replicating race start conditions, indicating the establishment of thermoelastic instabilities. The hot bands did not move radially during single engagements but did migrate between successive engagements indicating that torque instability during single engagements is due to surface morphology effects alone whilst torque inconsistency between engagements is due to both surface morphology and effective friction radius migration effects.

Keywords: Carbon/carbon composites, Formula One, F1, multi-plate clutch, torque instability, torque inconsistency, surface morphology, hot banding, thermoelastic instabilities, friction radius migration

1. Introduction

A brief history of the development of carbon/carbon composite multi-plate clutches for Formula One (F1) applications has been presented by both Gibson and Taccini (1989) and Lawrence et al. (2006). The first carbon/carbon clutch was introduced in 1982 and in 1987 the first race victory using a carbon/carbon clutch was achieved and carbon/carbon multi-plate clutches have been standard technology within F1 ever since. The high coefficient of friction (COF) that the material offers, combined with its low density (offering weight saving opportunities) and high maximum operating temperature (>2000℃), make it an ideal clutch plate friction material for F1 applications.

Inagaki (2000) describes the basic structure of a carbon/carbon composite as consisting of a filler and matrix where the filler is carbon fibre and the matrix is a type of resin. The matrix and carbon fibres are brought together in an uncured preform within a mould (Strong, 2008) which is then cured and non-carbon atoms driven off via the process of pyrolysis. Liquid or gas deposition or infiltration methods are then used to impregnate additional resin into the porous composite. Both processes can take several days or even weeks. Several cycles of pyrolysis and deposition/infiltration are carried out before a final carbonisation stage which produces a composite that is almost entirely carbon and minimally porous. Carbon/carbon composites with specific properties can be produced by using different resins and fibre precursors, with the fibres arranged in specific directions (through the use of 2D or 3D weaves) along with careful control of the composite impregnation and carbonisation. Carbon/carbon composite production can be very expensive due to the high energy demands as production can take up to six months. The clutch plates in this investigation are made up of polyacrylonitrile (PAN) carbon fibres impregnated using chemical vapour infiltration (CVI). Although the specific resin type is unknown, the composite is a commercially available product used in many race car brake and clutch applications.

A literature review summarising research into carbon/carbon composite brake materials has been presented by Blanco et al. (1997) who note that published papers are limited due to many of the research findings being protected by patents as a result of much of the research in this area being carried out by industry. This review also highlights that most published research is associated with the development of aircraft brakes, a point that is reiterated by Savage (2009). However, much of the research finding can be related to multi-plate clutches as they are similar in construction and application to aircraft brakes in that friction is generated via disc-on-disc contact and they are both used in high energy applications (aborted take-offs for aircraft brakes)..

The friction performance of carbon/carbon composite materials in ambient conditions has been investigated by several authors (Blanco et al., 1997; Kasem et al., 2008; Krkoska and Filip, 2008; Yen and Ishihara, 1996a) and, in relation to the variation of the friction coefficient with surface temperature, the same general trends have been observed. Along with surface morphology examination, three distinct friction regimes have been identified. Initially type I morphology (thin, smooth film) is dominant due to water being present in the friction surface resulting in a low COF value. At approximately 150℃, the water is desorbed from the surface and type II morphology (rough, powdery debris) is formed resulting in a rapid increase in COF (along with a rapid rise in surface temperature). 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 but at a higher value than the COF at ambient temperature when type I morphology was dominant. The relationship between the surface morphology, COF and temperature is however more complex and the friction behaviour of carbon/carbon composites has also been shown to be strongly affected by the level of energy input (Francois et al., 2007; Hutton et al., 1998; Kasem et al., 2010; Yen and Ishihara, 1994, 1996a), friction surface condition (Lee et al., 1999), ambient conditions (Chen et al., 1996; Kasem et al., 2008; Tanner and Travis, 2005; Yen and Ishihara, 1996a, 1996b) along with fibre type and orientation (Byrne and Wang, 2001; Hao et al., 2014; Hutton et al., 2001).

The phenomenon of thermoelastic instabilities (TEI) In sliding contact can also have an effect on a friction material and its performance. Barber (1969) states that any irregularities on the contacting surfaces of sliding solids will cause a non-uniform contact pressure distribution. This then leads to a non-uniform heat input distribution which leads to preferential thermal expansion in the areas of high temperature. In these areas the contact pressure will then increase, concentrating the heat input and causing further temperature increase and greater thermal expansion. The thermal distortion thus tends to exaggerate the original surface irregularities. Through the development of a computational model Barber (1969) also showed that height differences of as little as 1nm are sufficient to initiate TEI.

Zhao et al. (2008) and Abdullah et al. (2014, 2015) predicted contact localisation due to TEI in carbon/carbon multi-plate clutches using axisymmetric finite element models despite the friction surfaces were modelled as initially perfectly flat. In their models the magnitude of heat flux input was proportional to the COF, contact pressure and sliding velocity across the friction interface. Wear was not included in the models. The simulations predicted rapid contact loss at the inner and outer radii due to radial expansion of the clutch plates eventually leading to hot bands being formed as contact was then localised towards the centre of the friction surfaces. Hot bands between the central radii and the outer radii of the clutch plates were predicted ny the simulations carried out by Zhao et al. (2008) whilst contact pressure localisation at the centre of the friction surfaces leading to the formation of central hot bands were predicted by the simulations carried out by Abdullah et al. (2014, 2015). The differences may be due to the fact that their respective axisymmetric models simulated several friction interfaces and so the expansion at one friction interface may affect the expansion of another leading to hot bands being formed at different radial locations. The material properties may also affect the amount of expansion and in turn how the hot bands develop once TEI has been established.

The aim of the present investigation is to establish the cause of the unstable and inconsistent torque output of a carbon/carbon multi-plate clutch reported by many F1 teams. The following section (Section 2) describes the construction and results of a 1D heat transfer model incorporating a Taguchi Design of Experiment analysis used to assess the influence of material properties and system inputs on clutch-plate friction surface temperatures. Section 3 presents the results of white light interferometry and contact profilometry measurements carried out on both new and used clutch-plate friction surfaces. Section 4 describes the development of a unique dynamometer used to investigate in detail the performance of a single clutch-plate pair under typical race start conditions and the results of the dynamometer tests are presented in Section 5. Finally, in Section 6 the combined results of the numerical and experimental studies are discussed before the overall conclusions are summarised.

2. 1D Numerical Analysis

2.1 1D Heat Transfer Model

A one-dimensional (1D) heat transfer model was constructed using Matlab to enable a Taguchi Design of Experiment analysis to be carried out to determine which factors of the clutch plate engagement system and clutch-plate material properties have the greatest influence upon the maximum friction surface temperature reached.

As discussed above, the friction surface temperature of the carbon/carbon clutch plate affects the surface morphology whereby at a low temperature type I morphology is present which transitions to type II as the surface temperature increases. With this morphology change, the coefficient of friction also increases. This then increases the rate of energy dissipation at the friction surface which will cause the temperature to rise. In turn this will affect the surface morphology state and hence the coefficient of friction. Figure 1 shows this potential close interdependency.

[FIGURE 1]

The 1D model uses a finite difference approach as discussed by Incropera and Dewitt (2002) and finite difference equations presented by Limpert (1999) using the following assumptions to simplify the model:

- The clutch plates are identical in material and geometry
- Each clutch plate receives an equal proportion of the total energy dissipated at each clutch plate interface
- The clutch plate friction surfaces are perfectly flat
- Convective heat losses are negligible
- Material properties remain constant at all temperatures

The first two assumptions simplify the model in that only one clutch plate needs to be modelled. Assuming that the clutch-plate surfaces are perfectly flat means that the heat flux input at the clutch plate interface can be assumed to be uniform over the entire friction surface and the method presented by Limpert (1999) is therefore valid for this application. It is reasonable to assume that convective heat losses are negligible for two reasons. Firstly the clutch plates are mounted in an assembly in which very little air can circulate. Secondly, assuming that the clutch plate surfaces are perfectly flat, no convective losses can occur at the clutch plate interface or over the back faces of the plates due to full area contact between adjacent plates. Finally, the assumption that material properties are constant with temperature is necessary as the main purpose of the 1D heat transfer model is to facilitate a Taguchi analysis which requires the properties to be set at discrete but constant levels within each simulation.

The heat flux input was calculated based on the kinetic energy lost during a time step which is a function of the torque applied to the clutch plates. The uniform pressure model (Spotts, 2004) was used to calculate the torque generated as this model assumes that the clutch plates are perfectly flat and also accurately aligned resulting in the pressure and clamp load being uniform over the entire surface. The alternative uniform wear model (Spotts, 2004) was deemed to be unsuitable as not enough was known about the wear mechanism of the clutch plates to justify its use.

2.2 Taguchi Analysis

Taguchi (1987) defines a Design of Experiment approach as ‘the system by which one can efficiently and reliably evaluate all possible methods being considered for use in reaching a particular objective’. This technique is often used in manufacturing and production processes in order to improve quality by finding the optimum combination of factors which give the highest quality product where the quality is measured by defining certain objectives. In turn, this also gives an indication of which factors, when changed, have the greatest influence on the quality of the product. This feature of the Design of Experiments approach has been used to determine which factors of the clutch plate engagement system and material properties are most influential on friction surface temperature. The factor which has the largest influence on surface temperature will also have the largest influence on surface morphology and hence coefficient of friction due to the close interdependency shown in Figure 1.

The first step is to decide how many factors are to be investigated and at how many finite levels these factors are to be set at. A standard orthogonal array can then be chosen which outlines the experiments to be carried out. For seven factors each set at two discrete levels, an L_8 orthogonal array is appropriate. Table 1 shows the L_8 orthogonal array as presented by Taguchi (1987).

[TABLE 1]

If each column is considered to represent an experimental factor which can be set at either level 1 or 2, this orthogonal array presents eight experiments where each experiment is run with the seven factors at eight different combinations of levels. By doing this, the critical output can be measured for each experiment and statistical analysis can then be carried out to assess which factor has the largest influence on the critical output.

For the clutch plates, Table 2 shows the factors considered with the maximum friction surface temperature being the critical output. Table 3 shows the level 1 (minimum) and level 2 (maximum) values used. Some of these factors are physical values that can be controlled during race starts or through vehicle design. The COF and material property values have been obtained from literature sources (Byrne and Wang, 2001; Chen et al, 1996; Lee et al, 1997; Lee et al, 1999; Tanner and Travis, 2005; Zhao et al, 2008) where the values shown in Table 3 are simply the minimum and maximum values of these parameters obtained from these references.

[TABLE 2]

[TABLE 3]

The Signal-to-Noise (S/N) ratio is used to optimise the robustness of a product or process. Fowlkes and Creveling (1995) discuss the four properties of the ideal S/N ratio:

1. The S/N ratio reflects the variability in the response of a system caused by noise factors.
2. The S/N ratio is independent of the adjustment of the mean.
3. The S/N ratio measures relative quality because it is to be used for comparative purposes.
4. The S/N ratio does not induce unnecessary complications, such as control factor interactions, when the influences of many factors on product quality are analysed.

It is properties 3 and 4 that make the S/N ratio particularly useful for the clutch plate application. The S/N ratio will allow the relative influence of each factor to be compared independently. The S/N ratio can be calculated from the mean square deviation (MSD) using Equation 1:

$$S/N = -10\log(MSD) \tag{1}$$

The output that is most important in this investigation is the maximum friction surface temperature and hence it is this output that was used as the signal. To determine which factor has the biggest influence on the signal, the following procedure was carried out using the Matlab 1D heat transfer model:

1. Run all 8 simulations (experiments) shown in Table 2 and record the maximum predicted friction surface temperature.
2. For the first factor, take the four simulations with that factor set at level 1 and calculate the mean and variance of the maximum friction surface temperature.
3. Calculate the S/N ratio using these values.
4. Repeat steps 2 and 3 for that factor at level 2.
5. Calculate the difference between the S/N ratio at level 1 and level 2 to give the ‘Main Effect’ of that factor.

6. Repeat steps 2-5 for the other factors.

The results of this procedure are shown in Figure 2. The analysis showed that of the race start input parameters, the initial rotational speed of the clutch plates has the largest influence on the maximum friction surface temperature whilst of the carbon/carbon material properties, thermal conductivity has the largest influence. However, of all the parameters considered, the results clearly show that the COF has by far the largest influence on friction surface temperature and hence on surface morphology and COF. This analysis therefore suggests that the close interdependency shown in Figure 1 does indeed exist and that the only way to gain full control of the COF and achieve a stable, consistent torque output is through careful control of the surface temperature so that large changes of the carbon/carbon surface morphology do not occur during race start conditions.

[FIGURE 2]

3. Friction Surface Examination

The friction surfaces of both new and previously race-conditioned clutch plates were examined using white light interferometry and contact profilometry techniques to gain an understanding of how frictional work might affect the surface profile of the clutch plates. The exact history of the race-conditioned clutch plates was unknown beyond that they had been used for at least one race weekend.

A Veeco Wyko NT3300S interferometer was used where the clutch plates were placed underneath the white light beam such that the centre of the friction surface (radially) was scanned. Vision32 software was used to process the measured data. A magnification value closest to unity was used to allow a large area to be scanned using a vertical scan step size of 10μm. Any effects due to tilt were removed and a median filter applied to obtain the surface roughness parameter values. The roughness parameters of interest are defined in Table 4.

[TABLE 4]

A Form Talysurf 120L Contact Profilometer was used to measure the clutch-plate friction surface profiles. The clutch plates were placed on the traverse unit and the stylus moved to contact the friction surface at the inner radius. A trace was then recorded from the inner to outer radius. Talymap Gold software was used to process the data. Any effects of tilt and noise were removed and a low-pass Robust Gaussian filter with a 0.8mm wavelength cut-off was applied to isolate any waviness profile from the short wavelength roughness profile. The waviness parameters of interest are listed in Table 5.

[TABLE 5]

3.1 New Clutch Plates

The results of the roughness analysis indicated a circumferential variability in all the roughness parameters with some areas having high roughness and deep voids with other areas showing the opposite. This suggests that even with processing and machining of the clutch plates, neither a perfectly flat nor smooth friction surface is achieved. Despite this variability, all the scans showed a similar trend to that shown in Figures 3 and 4 whereby there is no evidence of any initial bands or large surface perturbations and, regardless of the maximum void depths, the areas the voids occupy are relatively large.

[FIGURE 3]

[FIGURE 4]

As the previous analysis suggested that the new clutch-plate friction surfaces had no waviness profile, only one surface profile trace was carried out using contact profilometry at an arbitrary circumferential position. Figure 5 shows a typical profile trace where it can clearly be seen that there is no distinct waviness profile: However, the trace shows a clear surface peak towards the outer radius which could lead to localised contact.

[FIGURE 5]

3.2 Race-Conditioned Clutch Plates

In comparison to the new clutch plates, the average roughness value of the race-conditioned clutch plates was lower. Combined with the increase in negativity of the average skewness value (R_{sk}), this suggests that many of the original surface asperities have been worn away and the surface has become smoother as the result of frictional work. Both the average values of maximum void depth (R_v) and maximum peak height (R_p) were smaller in magnitude than for the new clutch plates and the size of void and peak areas as a proportion of the total surface was significantly smaller. This supports the idea that wear debris formed by wear of the surface asperities could be filling in the deep voids. This is illustrated in Figures 6 and 7 which show typical scans of the used clutch plate friction surfaces.

Figures 6 and 7 also show that there appears to be a regular undulation of the friction surface in the radial direction that was not seen with the new clutch plates. This could have been caused by banding effects leading to localised contact and hence wear in a finite number of bands across the friction surface. Once these bands have worn away, contact may move to another band and eventually a regular wavy profile may be formed on the surface.

[FIGURE 6]

[FIGURE 7]

Contact profilometry results showed that in comparison to the new clutch plates, all used clutch plates showed distinct waviness profiles that permitted a waviness profile wavelength (W_{sm}) value to be calculated. The profiles showed that the average peak height and average valley depth of the waviness profiles were approximately 0.8μm and 0.7μm respectively with an average spacing between the peaks of 2.34mm equating to 6.84 wavelengths per 16mm wide friction surface. Figure 8 shows a typical trace for the used clutch plates showing a clear waviness profile. It can also be seen that the shape of the waviness profile is roughly sinusoidal.

[FIGURE 8]

4. 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) are 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 of 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 9. 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 9]

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℃ 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℃. 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) (Williams, 2009). 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 6 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.

[TABLE 6]

5. SCID Results

5.1 Torque Outputs

Figures 10a and 10b 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 10a 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 10b 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 10b 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 10]

5.2 Surface Temperatures

The clutch-plate friction surface temperatures recorded by the thermal imaging camera showed a vast difference between those recorded during the lowest speed/load combination (A2) and the highest speed/load combination (B4). Figure 11 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 11 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℃ whilst the maximum temperature of ~550℃ occurred just beyond the mid-radius position. The temperature difference of around 75℃ is too small for the region of higher temperature to be classed as a distinct hot band.

[FIGURE 11]

Figure 12 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 11, Figure 12 shows a distinct hot band on the friction surface. The extremely high temperatures (1300-1575℃) existed in a narrow band approximately 2mm wide where the temperatures outside of the band were at around only 500-600℃.

[FIGURE 12]

For all engagements carried out at the 7000rpm/1000N speed/load combination (A2), a similar surface temperature profile to that shown in Figure 11 was observed with maximum temperatures of 550℃-600℃. A similar surface temperature profile as that shown in Figure 12 was measured for all clutch-plate engagements carried out at 1400N for both speed combinations (A4 & B4) with maximum temperatures of 1300℃-1650℃.

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:

$$T = P\mu r_e \quad (2)$$

Upon examining the evolution of the hot band shown in Figure 12 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.

6. Discussion

The Taguchi analysis carried out using the 1D heat transfer model showed that of all the factors investigated, the coefficient of friction (COF) had by far the largest influence upon the maximum friction surface temperature generated. This result suggests that the close interdependency between surface temperature, surface morphology and COF shown in Figure 1 does exist and therefore the COF could be controlled by controlling the surface temperature.

However, because of the close interdependency between surface temperature, surface morphology and COF, it is very difficult to achieve this and there is therefore no quick fix to the issue. The analysis did however show that varying the initial rotational speed or inertia could help to the control the surface temperature but this may not be possible depending on the vehicle design or regulations. The thermal conductivity of the carbon/carbon composite could also be used as a mechanism for controlling surface temperature but this analysis assumes that the factors investigate are independent of each other. In reality it is likely that the material properties are strongly interdependent such that the ideal combination cannot actually be achieved in practice and hence the main effect of the thermal conductivity is in fact much less than predicted by this analysis and is hence not a solution to the problem.

As well as the variability associated with the surface morphology and COF, the white light interferometry and contact profilometry analyses of new and race-conditioned clutch plates showed that thermoelastic instabilities may also have an effect on torque output. A non-uniform friction surface will lead to localised contact and through localised heating thermal expansion and wear, thermoelastic instability will be induced. This will lead to contact being isolated to discrete bands on the friction surface where the radial location of the band will be the effective friction radius thus influencing the torque output (Equation 2). This result was reiterated by the race-conditioned clutch plates showing distinct waviness profiles.

The results of the single clutch-plate interface dynamometer (SCID) tests showed that at low speed/load combinations, the torque output of the clutch-plate pairs was stable and consistent but with a low peak value and hence long engagement times. At the speed/load combination representative of race start conditions, the torque was unstable and inconsistent but with a high peak torque output and short engagement times. At the low speed/load combinations no hot bands were observed using the thermal imaging camera. However at the speed/load combination representative of race start conditions the existence of distinct, narrow (~2mm), high temperature (1300-1650℃) which represent the locati on of the effect friction radius were observed The hot bands and hence effective friction radius were not observed to migrate during single engagements but did migrate between successive engagements. This result along with the torque output results suggests that torque instability during engagements is due to surface morphology effects alone whilst torque inconsistency is due to a combination of both effective friction radius migration and surface morphology effects.

For the race start application, a consistent torque output is critical as the instability can be dealt with via clutch and throttle control from the driver as long as the driver knows what to expect. The results of this investigation suggest that a greater level of consistency may be achieved by eliminating the potential for effective friction radius migration through the clutch-plate design and geometry. This would not however achieve 100% consistency as surface morphology effects would still have an influence upon torque output and therefore these effects would also need to be controlled and minimised.

7. Conclusions

A one-dimensional heat transfer model incorporating a Taguchi analysis showed that the coefficient of friction has the largest influence on the maximum clutch-plate friction surface temperature reached during engagements. The results of the analysis suggested that a close interdependency exists between surface temperature, surface morphology and coefficient of friction for these carbon/carbon clutch plates.

White light interferometry and contact profilometry analyses of new and race-conditioned clutch plates showed that new clutch plates had non-uniform surface profiles which could lead to the phenomenon of thermoelastic instabilities being established. Race-conditioned clutch plates showed distinct waviness profiles suggesting that contact and frictional work is isolated to a discrete numbers of bands across the friction surface. At which band contact is predominant will influence the effective friction radius and hence torque output.

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℃). The SCID showed that the 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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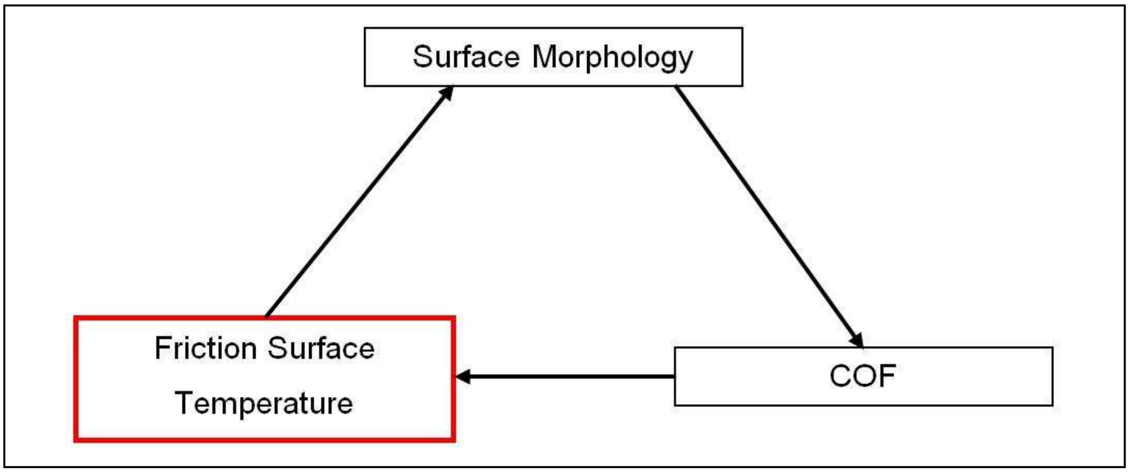


Figure 1 - Possible Friction Surface Temperature-Morphology-COF Relationship

	Column	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Experiment Number (EN)								
1		1	1	1	1	1	1	1
2		1	1	1	2	2	2	2
3		1	2	2	1	1	2	2
4		1	2	2	2	2	1	1
5		2	1	2	1	2	1	2
6		2	1	2	2	1	2	1
7		2	2	1	1	2	2	1
8		2	2	1	2	1	1	2

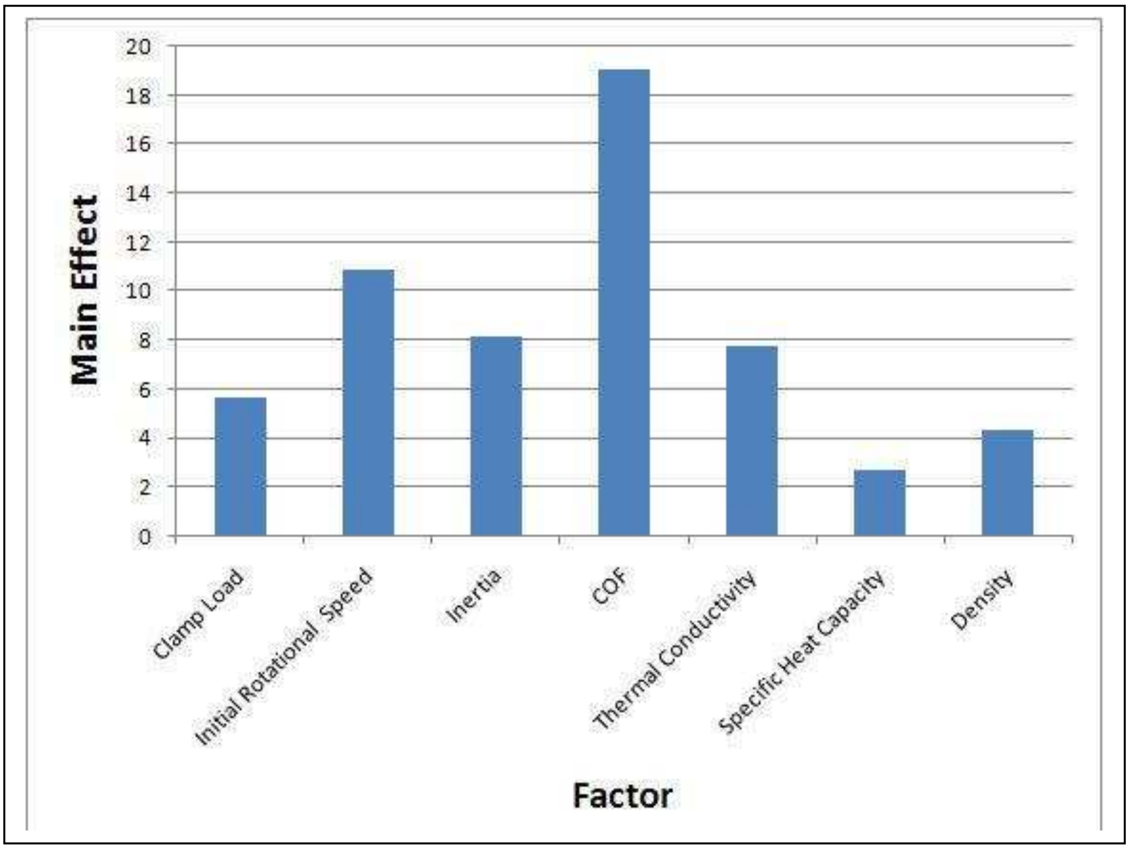
Table 1 – L_8 Orthogonal Array

	Factor	Clamp Load (N)	Initial Rotational Speed (rpm)	Inertia (kgm^2)	COF	Thermal Conductivity ($Wm^{-1}K^{-1}$)	Specific Heat Capacity ($Jkg^{-1}K^{-1}$)	Density (kgm^{-3})
EN								
1		1	1	1	1	1	1	1
2		1	1	1	2	2	2	2
3		1	2	2	1	1	2	2
4		1	2	2	2	2	1	1
5		2	1	2	1	2	1	2
6		2	1	2	2	1	2	1
7		2	2	1	1	2	2	1
8		2	2	1	2	1	1	2

Table 2 – Factors Investigated in Design of Experiment Analysis

Factor	**Level**	
	1	**2**
Clamp Load (N)*	1000	2000
Initial Rotational Speed (rpm)*	8000	10000
Inertia (kgm^2)*	0.025077	0.033436
COF**	0.2	0.5
Thermal Conductivity (Wm^{-1}K^{-1})**	10	60
Specific Heat Capacity (Jkg^{-1}K^{-1})**	700	1900
Density (kgm^{-3})**	1600	1900

Table 3 – Factor Values Used in Design of Experiment Analysis
*Factor values can be set during race starts or through vehicle design
**Factor values obtained from literature (Byrne and Wang, 2001; Chen et al, 1996; Lee et al, 1997; Lee et al, 1999; Tanner and Travis, 2005; Zhao et al, 2008)



Main Effect
20
18
16
14
12
10
8
6
4
2
0
Clamp Load
Initial Rotational Speed
Inertia
COF
Thermal Conductivity
Specific Heat Capacity
Density
Factor

Figure 2 – Main Effect Results of 1D Heat Transfer Model

Parameter	Name	Definition
R_a	Average Roughness	The average surface deviation from the mean surface height. A value of 0 would indicate a perfectly flat surface.
R_p	Maximum Peak Height	The height of the largest surface peak above the average surface height.
R_v	Maximum Valley Depth	The depth of the largest surface valley/void below the average surface height.
R_t	Maximum Peak-to-Valley Distance	$R_t = R_p - R_v$
R_{sk}	Skewness	A measure of the surface asymmetry about the mean surface height. Negative skewness indicates a predominance of valleys and positive skewness indicates a spiky surface.

Table 4 – Surface Roughness Parameters Measured Using White Light Interferometry

Parameter	Name	Definition
W_p	Maximum Peak Height	Height of the largest peak above the mean friction surface height
W_v	Maximum Valley Depth	Depth of the largest valley below the mean friction surface height
W_z	Maximum Peak-to-Valley Distance	$W_z = W_p – W_v$
W_{sm}	Waviness Profile Wavelength	Mean distance between waviness profile peaks. The number of peaks is therefore equal to $16/W_{sm}$ (friction surface is 16mm wide)

Table 5 – Surface Waviness Parameters Measured Using Contact Profilometry

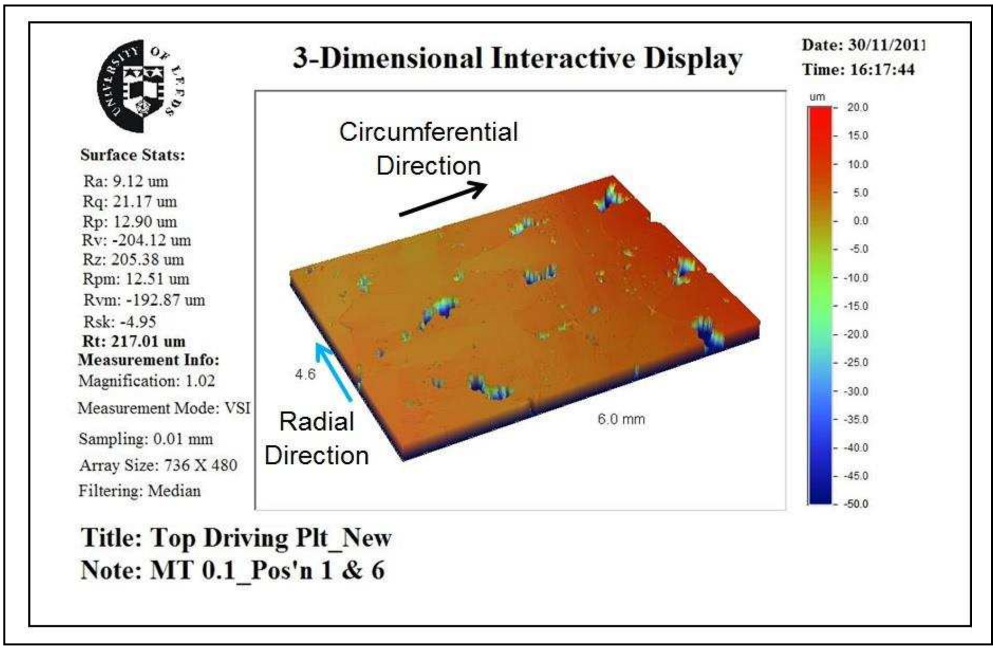


Figure 3 – Processed Image of Typical New Clutch-Plate Friction Surface (Isometric View)

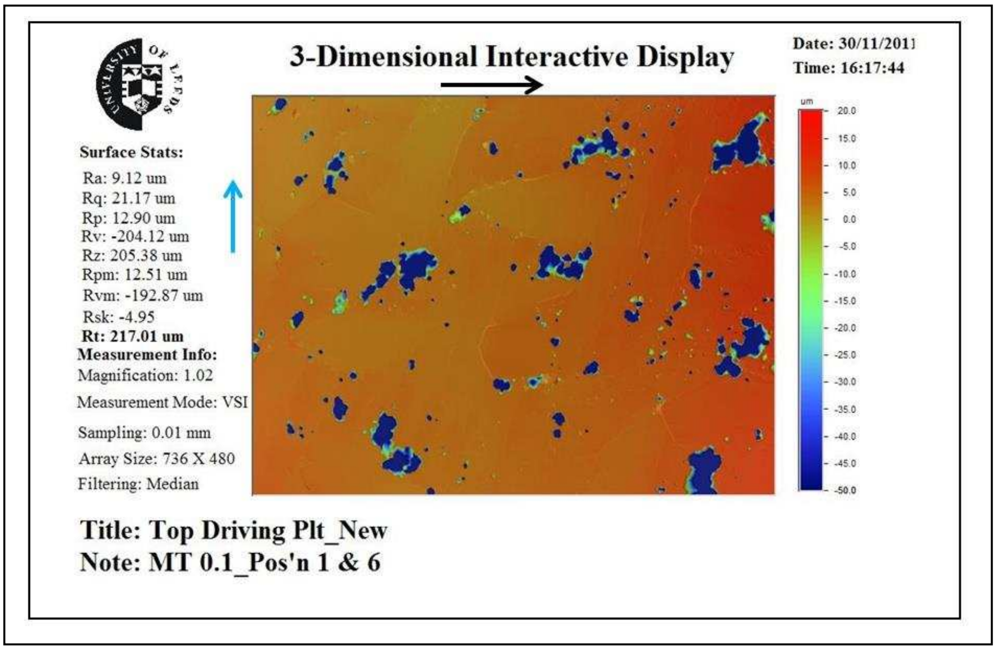

3-Dimensional Interactive Display
Date: 30/11/2011
Time: 16:17:44
Surface Stats:
Ra: 9.12 um
Rq: 21.17 um
Rp: 12.90 um
Rv: -204.12 um
Rz: 205.38 um
Rpm: 12.51 um
Rvm: -192.87 um
Rsk: -4.95
Rt: 217.01 um
Measurement Info:
Magnification: 1.02
Measurement Mode: VSI
Sampling: 0.01 mm
Array Size: 736 X 480
Filtering: Median
um
20.0
15.0
10.0
5.0
0.0
-5.0
-10.0
-15.0
-20.0
-25.0
-30.0
-35.0
-40.0
-45.0
-50.0
Title: Top Driving Plt_New
Note: MT 0.1_Pos'n 1 & 6

Figure 4 – Processed Image of Typical New Clutch-Plate Friction Surface (Plan View)

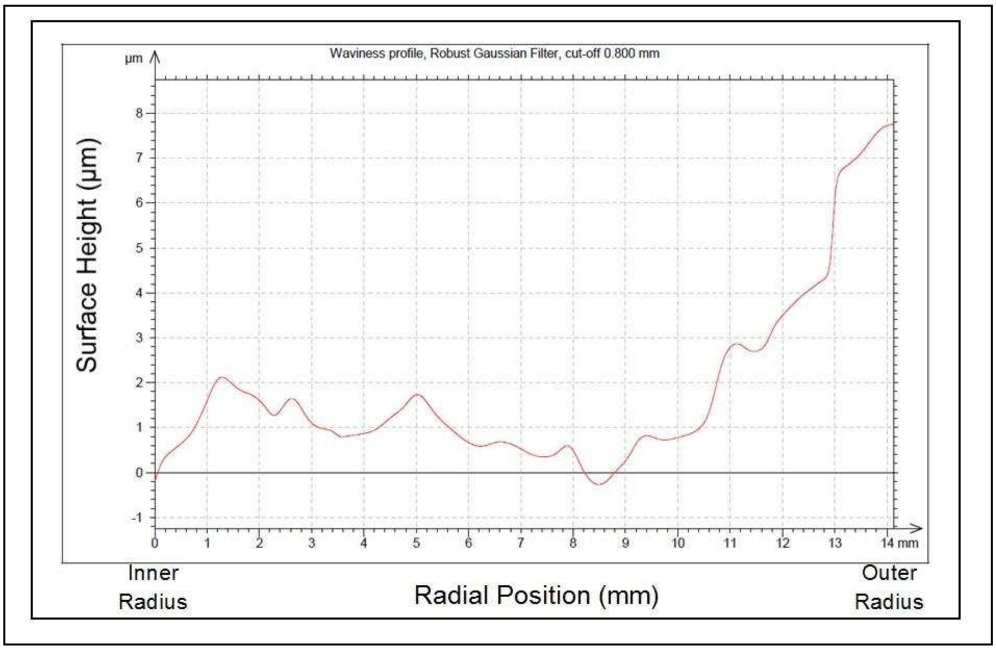


Figure 5 – Typical Friction Surface Profile of New Clutch Plate

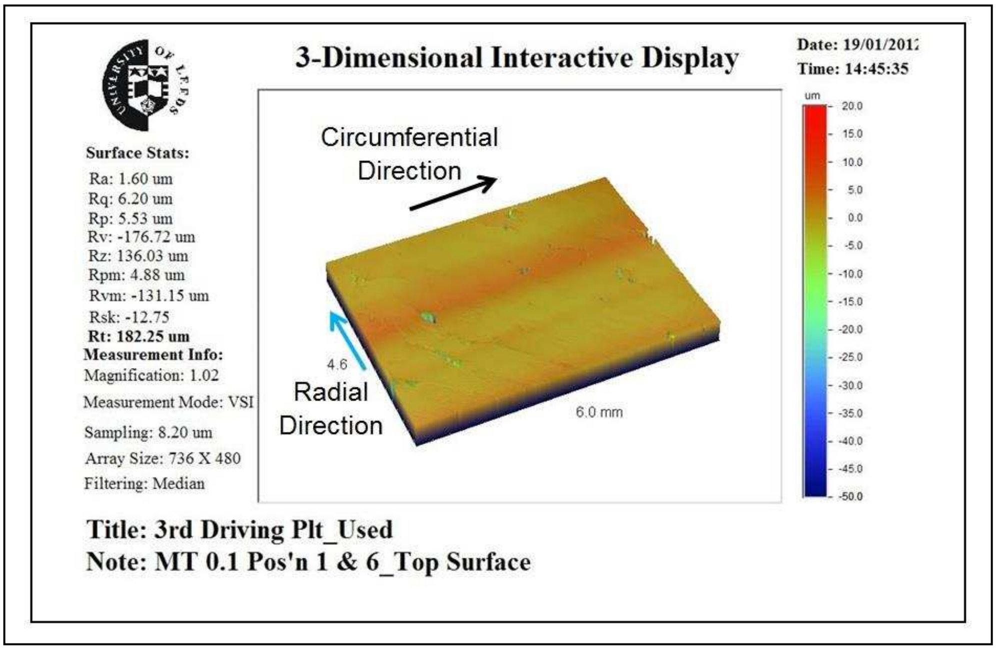


Figure 6 – Processed Image of a Typical Race-Conditioned Clutch-Plate Friction Surface (Isometric View)

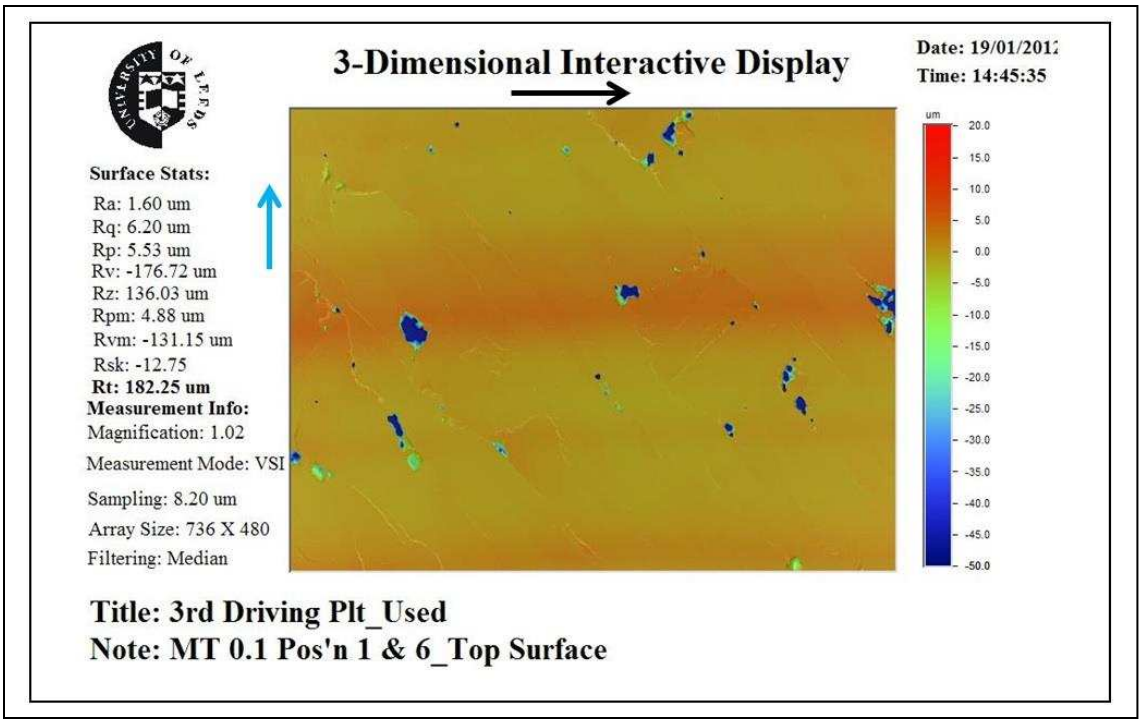

3-Dimensional Interactive Display
Date: 19/01/201
Time: 14:45:35
Surface Stats:
Ra: 1.60 um
Rq: 6.20 um
Rp: 5.53 um
Rv: -176.72 um
Rz: 136.03 um
Rpm: 4.88 um
Rvm: -131.15 um
Rsk: -12.75
Rt: 182.25 um
Measurement Info:
Magnification: 1.02
Measurement Mode: VSI
Sampling: 8.20 um
Array Size: 736 X 480
Filtering: Median
um
20.0
15.0
10.0
5.0
0.0
-5.0
-10.0
-15.0
-20.0
-25.0
-30.0
-35.0
-40.0
-45.0
-50.0
Title: 3rd Driving Plt_Used
Note: MT 0.1 Pos'n 1 & 6_Top Surface

Figure 7 – Processed Image of a Typical Race-Conditioned Clutch-Plate Friction Surface (Plan View)

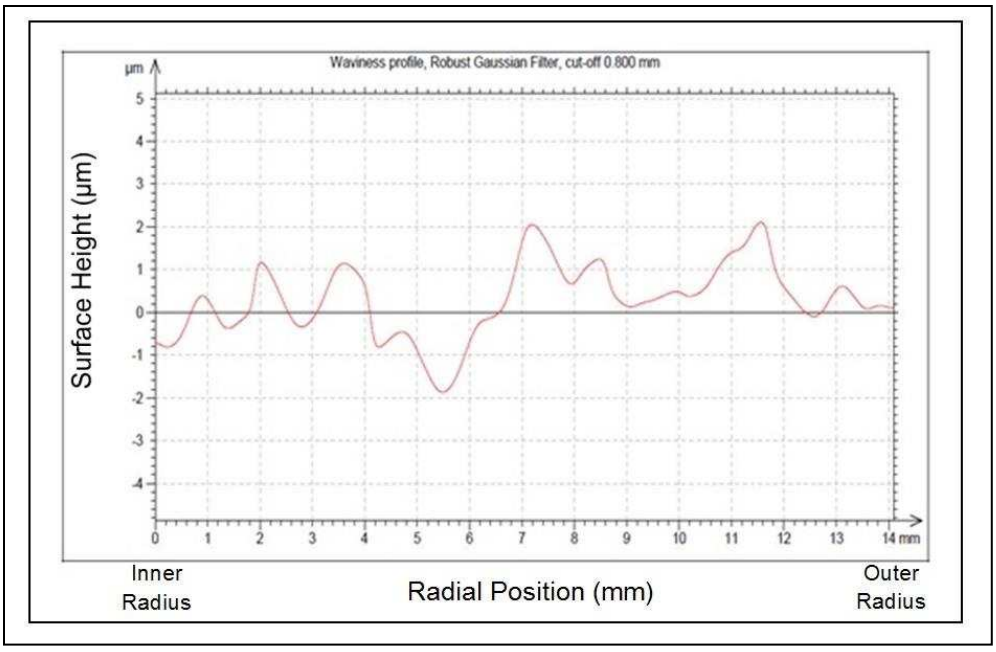


Figure 8 – Typical Friction Surface Profile of a Race-Conditioned Clutch Plate

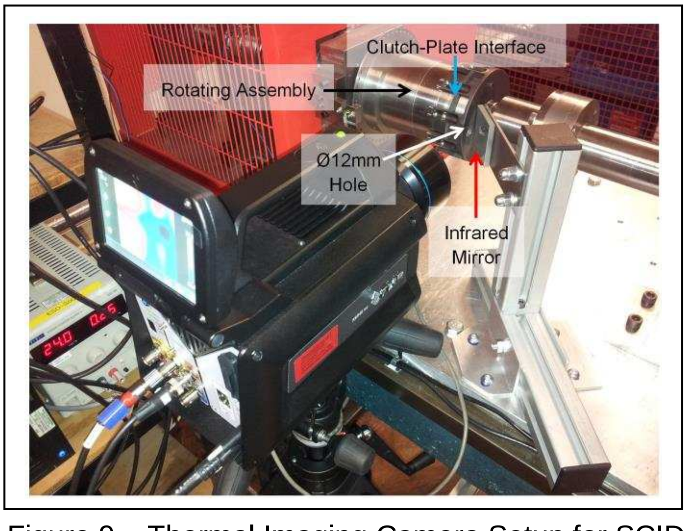


Figure 9 – Thermal Imaging Camera Setup for SCID

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

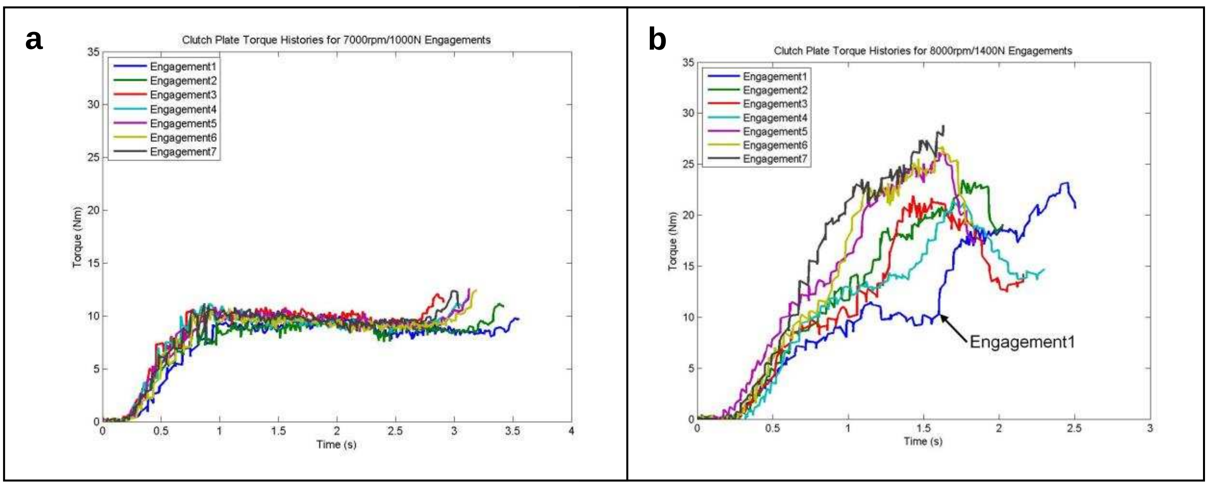

a
Clutch Plate Torque Histories for 7000rpm/1000N Engagements
Engagement1
Engagement2
Engagement3
Engagement4
Engagement5
Engagement6
Engagement7
Torque (Nm)
Time (s)
b
Clutch Plate Torque Histories for 8000rpm/1400N Engagements
Engagement1
Engagement2
Engagement3
Engagement4
Engagement5
Engagement6
Engagement7
Torque (Nm)
Time (s)
Engagement1

Figure 10 – SCID Torque Output Results for (a) 7000rpm/1000N and (b) 8000rpm/1400N Speed/Load Combinations

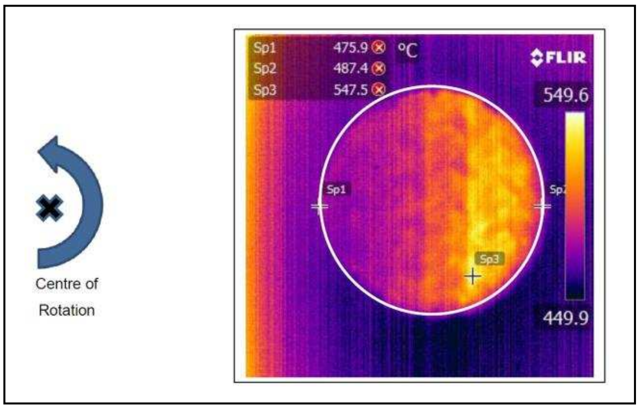


Figure 11 – Maximum Clutch-Plate Friction Surface Temperature Recorded During Engagement4 at 7000rpm/1000N Speed/Clamp Load Combination (A2)

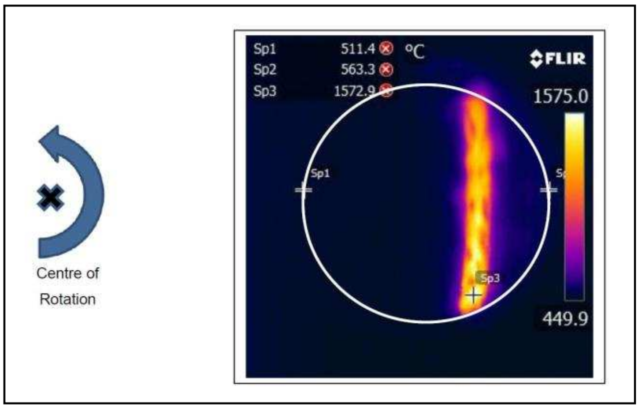

Centre of
Rotation
Sp1 511.4
Sp2 563.3
Sp3 1572.9
°C
FLIR
1575.0
449.9
Sp1
Sp3

Figure 12 – Maximum Clutch-Plate Friction Surface Temperature Recorded During Engagement5 at 8000rpm/1400N Speed/Clamp Load Combination (B4)