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The wear of PEEK in rolling–sliding contact – Simulation of polymer gear applications [☆]



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ABSTRACT

The wear and friction in the pitch region of the centre of polymer gear teeth are not well understood. The transition around this point of the tooth between rolling and sliding has an important effect on the durability of polymer gear drives and can be simulated using a twin-disc configuration. This paper investigates the rolling–sliding wear behaviour of two poly-ether-ether-ketone (PEEK) discs running against each other with a simplified method of analysing and understanding the dynamic response of high performance polymeric gear teeth.

Tests were conducted without external lubrication over a range of loads and slip ratios, using a twin-disc test rig. The wear and friction mechanisms were closely related to surface morphology, with changes in crystallinity correlating with the severity of operating conditions. Observed failure mechanisms were also related to the structure of the contact surfaces, and included surface melting and contact fatigue.

Overall the PEEK discs were capable of running at low slip ratios for both low and high loads. Their performance reduced with an increase of the slip ratio. The results presented can be used in conjunction with the design process to allow the PEEK to be engineered for a specific high performance gear contact conditions.

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1. Introduction

With a growing awareness of engineering polymers, there is increasing application of polymers and polymeric composites to machine elements. The ability to economically manufacture and run unlubricated contacts at increasing temperatures (through the use of high temperature polymers such as poly-ether-ether-ketone (PEEK) is making their application more desirable.

The majority of published work on the tribology and wear of non-conformal polymer pairs relates to the performance of gears. For a pair of gears the dominant operating parameters such as sliding velocity and load, and the geometric parameters such as module and curvature of the contacting surfaces vary with the contact position on the tooth profile. Consequently, gear action is a very complicated process to understand. An alternative method of studying gear action is to apply the same load and speed conditions to a much simpler geometry. An example of such a simulation is the use of two cylindrical discs loading against each other in edge-to-edge contact, each rotating at different speeds. By varying the relative speeds of the discs (i.e. changing the ratio of sliding to

rolling velocity, the so-called ‘slip ratio’) and the normal load, the conditions experienced by gear teeth in contact may be approximated [1]. The simulation of gears can never be truly representative however, since changes in sliding velocity, tooth flexibility and differences in the thermal conditions make contact conditions very different. In addition, slip ratios in twin disc testing tend to be limited to around 30%, whereas slip ratios above 30% are found throughout the majority of the gear meshing cycle.

Nevertheless, twin-disc tests provide fundamental information about materials behaviour in rolling–sliding motion under non-conformal contact. They should be seen as complementing the applied information from direct gear testing and as contributing to a more fundamental understanding of polymer tribology under rolling–sliding conditions [1–5].

Previous work, using the twin-disc configuration, has compared the tribological performance of a range of engineering polymers and their composites, namely polyoxymethylene (POM) [1,2], polyamide 46 (PA46) [3], polyamide 66 (PA66) [2,4–6], glass–fibre reinforced PA66 [2,7,8], PA66 and POM filled with 20 wt% of polytetrafluoroethylene (PTFE) [9], and short fibre, aramid and carbon-reinforced PA66 [3]. These materials were tested over a range of rolling speeds and slip-ratios to study their wear and frictional properties and their potential damage mechanisms.

It was found that initially the fibre grades were capable of withstanding a high loads; however, as the surface layer of the material was removed and the reinforcing filler was exposed, the

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wear rate increased. This limited the materials use to high load, low cycle applications. In the unreinforced materials, friction was shown to correlate well with both temperature and wear.

The mechanical behaviour of PEEK is widely reported [10–13]. These results are not directly relevant to the non-conformal contact found in engineering components such as gears [1,2,14]. However, they do give an indication of the high performance of the material.

Literature covering twin disc testing of PEEK is limited. Avanzini et al. looked at the mechanical response of both filled and unfilled PEEK when run against a steel counterface in a rolling contact. They found that transverse cracks on the surface formed deep radial cracks in the material and that wear rate was related to pressure, velocity and hardness [15]. Similarly, Berer et al. investigated PEEK in rolling contact discussing the influence of lubricant and pre-cracks on the surface pitting behaviour of the material [16]. However, both investigations were limited as the discs were only run in rolling (i.e. both discs rotating at the same speed), thus the use of this in the simulation of gear tooth contact is limited.

Despite this, PEEK would appear to be an ideal gear material; its high relative thermal index for mechanical contact with impact (a parameter determining the maximum service temperature at which the critical properties of the material such as toughness and impact strength will remain within acceptable limits for a given mechanical application with impact over a long period of time [17]) suggests that it is capable of withstanding the high temperatures and stresses associated with high performance polymeric gearing without significant thermal degradation. Therefore, this paper will investigate the rolling–sliding wear behaviour of two unreinforced poly-ether-ether-ketone (PEEK 450G) discs running against each other, and the use of this as a simplified method of analysing the dynamic response of high performance polymeric gear teeth.

2. Materials and methods

Fig. 1 shows the twin-disc test rig that was used for unlubricated, rolling–sliding tests. Two cylindrical discs were mounted on spindles contained in a friction block and a pivoted loading block. An electric motor (1) provided an input speed of 1000 rpm, using two toothed belts (2) and a pair of speed change gears (7). The discs were driven at a controlled speed with the relative slip ratio between the contacting discs adjusted by altering the gear ratios. The system is capable of simulating the non-conformal contact found in common machine components such as gears and cams. Loads were applied to the system by a weight (5) attached to the upper pivoted loading block (4) to provide a normal force between the two discs. The lower block was mounted on vertical leaf springs and strain gauges were used to determine the sliding frictional force by noting the tangential force on the lower disc. Wear of the samples was approximated by detecting the displacement of the upper block using a linear variable displacement transducer (LVDT) to record the displacement of the disc centres.

The test discs were machined from an unreinforced PEEK 450G extruded bar [17].

All specimens were polished to a surface roughness of approximately 5 μm whilst maintaining the cylindricity of the discs. Fig. 2 shows the sample geometry.

The test discs were placed on the shafts in the twin-disc test rig and tests were run for a range of loads and slip ratios, shown in Table 1. During the tests, the friction, wear and temperature data for the system were collected.

The tribological and mechanical properties of polymers are much more sensitive to temperature than those of metals; therefore it is necessary to establish the maximum contact temperatures so

that their effects on the wear behaviour can be considered. In this investigation, a Fluke Ti25 infrared camera (temperature range $-20\text{ }^{\circ}\text{C}$ to $350\text{ }^{\circ}\text{C}$ and accuracy $\pm 2\text{ }^{\circ}\text{C}$ or 2%) was used to measure temperature.

To determine whether the surface of the discs has undergone thermal ageing/enthalpic relaxation during testing, the crystallinity of the samples was investigated using differential scanning calorimetry (DSC). The Perkin Elmer DSC 7 unit used in this investigation consisted of a sample and a reference cell, both of which have separate heaters and platinum resistance temperature sensors. The heaters are coupled so that the differential power needed to maintain the two cells at the same temperature, can be measured [18,19]. This allowed the mass fraction of the crystalline phase in the polymer to be established, knowing that the theoretical heat of fusion for the pure crystalline phase of PEEK was (130 J/g).

Finally, the wear on the contact surfaces was characterised using optical methods and a JEOL JSM-6060 scanning electron microscope (SEM).

3. Theory

3.1. Gear kinematics

In polymeric gears, the compliance of the material means that the path of contact extends beyond the theoretical path for a perfectly stiff gear mesh (Fig. 3) [20]. When the path of contact is extended, the load distribution between meshing teeth and the sliding velocity are also altered.

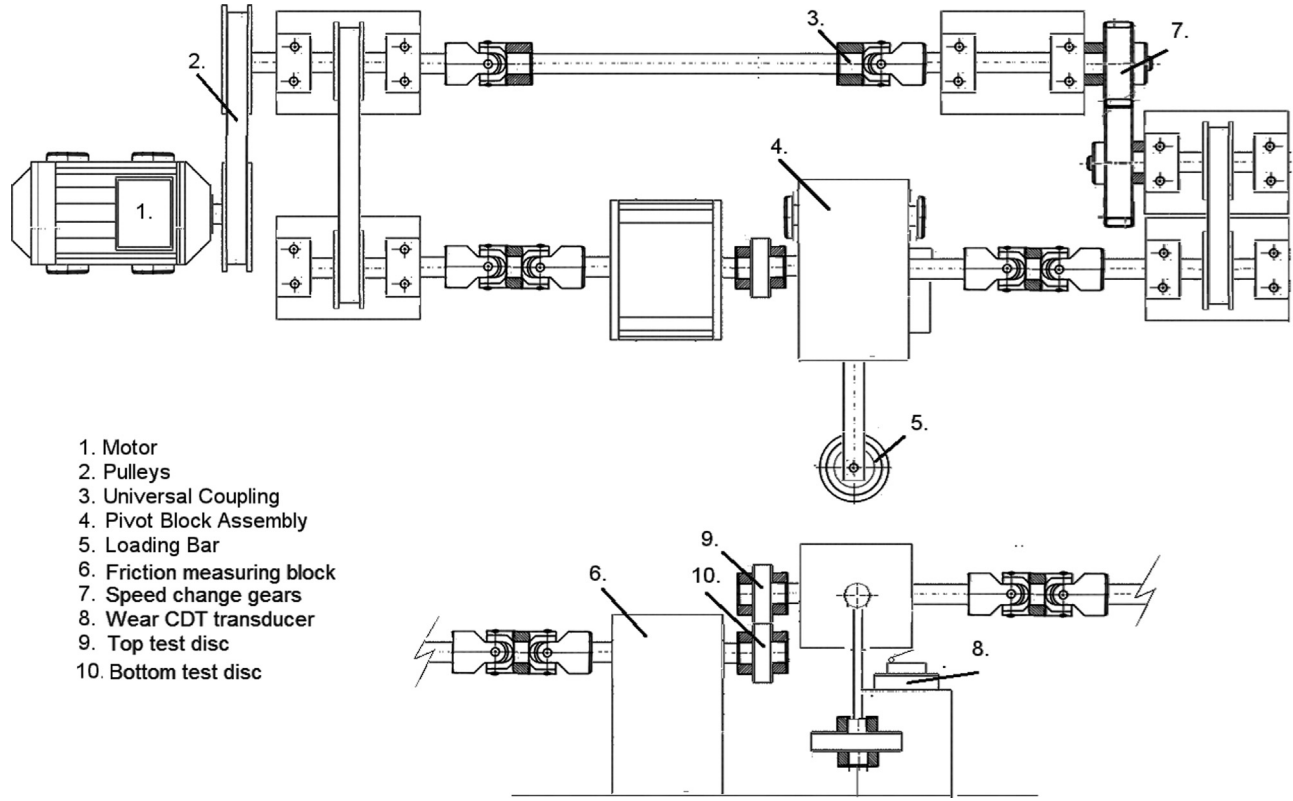
The slip ratio of a pair of meshing gears refers to the ratio of the sliding velocity at the point of contact, to the rolling velocity. For two discs, rotating at V_1 and V_2 respectively, the slip ratio can be described by Eq. (1)

$$\text{slip ratio}(\%) = 2 \frac{V_2 - V_1}{V_1 + V_2} \quad (1)$$

Restrictions on the test rig limit the obtainable slip ratio to below approximately 30%. Therefore, twin-disc tests traditionally assess a material's mechanical response to conditions found in the region surrounding the pitch point of gears, where the velocity of sliding is low (see Fig. 4). Sukurmaran et al. modelled a polymer–metal gear contact using a twin disc set-up. It was shown that the wear rate was low and that the main mode of failure was plastic flow [22]. However, there was little consideration of how simulating plastic gears would be different from modelling steel tooth contact.

Fig. 4 shows the theoretical slip-ratios for a pair of meshing gear teeth without deflection and the influence of the extension in the path of contact. The theoretical position of each twin-disc test will model different meshing positions on a simulated gear tooth depending on the extent to which deflection affects the system. In addition, from Fig. 4 it can be seen that slip ratios of up to 30% obtainable using twin disc tests, will simulate both the regions surrounding the pitch-point and the theoretical first point of contact when large deflections occur (note that the gear teeth used in this investigation were based on the Birmingham standard geometry, i.e. a spur gear, 2 mm module, 30 teeth, 20° pressure angle, 1:1 ratio, [23]).

For steel discs, twin-disc testing can be used to simulate the conditions found at the pitch point of a steel gear tooth. However, this kinematic model shows that when large deflections occur, due to the high compliance of polymeric materials, the path of contact is extended; thus allowing tooth conditions to be modelled for premature contact.



1. Motor
2. Pulleys
3. Universal Coupling
4. Pivot Block Assembly
5. Loading Bar
6. Friction measuring block
7. Speed change gears
8. Wear CDT transducer
9. Top test disc
10. Bottom test disc

Fig. 1. Schematic view of the twin-disc test rig.

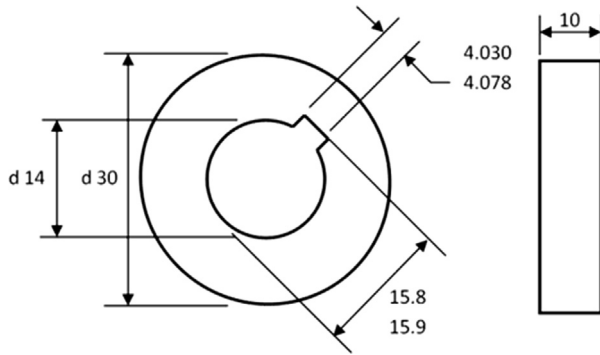


Fig. 2. Twin disc test geometry (dimensions in mm).

3.2. Interfacial temperature

The predicted temperature generated at the interface between two contacting discs can be modelled as the sum of the prevailing temperatures at that point (Eq. (2))

$$T_{\text{interface}} = T_{\text{flash}} + T_{\text{bulk}} \quad (2)$$

where T_{flash} is the instantaneous temperature due to frictional heating, and T_{bulk} is the measured bulk temperature of the two bodies, combining ambient temperature and physical heating effects.

Assuming that a system is based on steady-state sliding where parameters such as heat intensity, sliding speed, and normal pressure are regarded as constant, the flash temperature can be approximated using Blok's equation [24] (Eq. (3))

$$\delta T_{\text{flash}} = \frac{1.11 \mu F [V_1^{1/2} - V_2^{1/2}]}{b \sqrt{2k\rho c a}} \quad (3)$$

Table 1
Test conditions.

| Load | Slip ratio | | |
|------|------------|--------|--------|
| | 3.92% | 14.29% | 28.57% |
| 400 | ✓ | ✓ | ✓ |
| 300 | | ✓ | ✓ |
| 200 | | | ✓ |
| 100 | | | ✓ |

where the semi-width of contact between the two contacting surfaces, a , can be calculated as

$$a = \sqrt{\frac{4FR}{b\pi E}} \quad (4)$$

and where

$$\text{contact modulus: } \frac{1}{E} = \frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \quad (5)$$

$$\text{relative radius of curvature: } \frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} \quad (6)$$

In Eqs. (3)–(6), μ is the coefficient of friction, F is the normal load, V_1 and V_2 are the rolling velocities of the disc surfaces, b is the width of the discs, k is the thermal conductivity, ρ is the density of the material, c is the specific heat capacity, and R_1 and R_2 , E_1 and E_2 , and ν_1 and ν_2 are the respective radii, Young's modulus and Poisson's ratio of the top and bottom discs.

Therefore, the predicted contact temperature can be estimated through use of Eq. (2) and the measured bulk temperature.

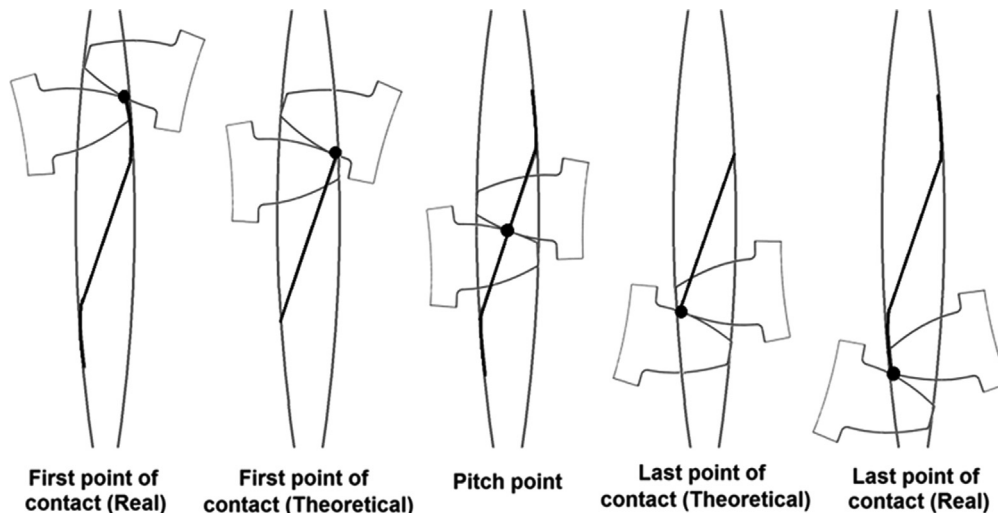


Fig. 3. The real and theoretical paths of contact for a pair of meshing polymer gear teeth (after Dearn et al. [21]).

4. Results and discussion

4.1. Wear mechanisms and topographical analysis

As shown in Fig. 5, all tests displayed three distinct regions of wear which could be correlated with both the temperature and friction profiles for each test. This is a common observation in both twin-disc and gear tests [1–5,7,9,14,25], and represents: a ‘running-in’ period, a period of steady-state running, and a period of increased wear signifying the end of the test specimen’s lifecycle.

SEM images of the worn disc surfaces are shown in Fig. 6. The images show a 14% slip-ratio test which induced dramatically-increased temperatures and coefficients of friction. This test eventually resulted in localised damage on the disc surface producing significant amounts of wear.

From Fig. 6 it can be seen that the disc material softened in the high severity tests. This is a result of the discs being thermally overloaded, and also because of yielding (permanent deformation). In addition, localised damage can be seen on the surface of the disc. It is thought that this is a consequence of the high temperatures generated at the point of contact forcing the expulsion of wear debris. A proportion of this debris is subsequently trapped in the mesh, damaging the surface of the disc. Following cyclic fatigue and adhesion, this failure can then be seen on the surface of the disc.

It is important to understand the influence of crystallinity on the wear mechanisms observed. The simulated position of each twin-disc test on a gear tooth correlates with a specific contact condition. If the severity at this point exceeds that required for in-service crystallisation or a change in crystallinity to occur, optimising methods to change the loading and temperature at this point may be required to improve the efficiency of the geared system.

Fig. 7 shows the relative test severity correlated with the average wear rate. The severity of the test was calculated as the average thermal output from the contact integrated over time, allowing the wear rates to be quantified based on the operating conditions. It was found that the high slip-ratio, high load tests were the most severe and thus produced the greatest wear rate and temperature increase. The low slip-ratio, low load tests produced the lowest wear rate and temperature increase, and were thus considered to be the least severe.

The high severity tests showed localised damage on the surface (as shown in Fig. 6). However, several different wear mechanisms were observed in the tests.

In the later stages of the high-severity tests as the material began to soften, the discs began to expel wear debris as material was transferred between the surfaces of the discs. This is associated with interfacial wear, where the energy from the sliding component of friction due to contact is dissipated in a thin region adjacent to the surface.

Once the discs began to expel wear debris, the temperature became less steady. Wear debris was forced into the surface of the material by the contact pressure. It can be seen from the polished sample in Fig. 8 that there is a definite change in the material around the centre of the sectioned disc. This area is associated with the highest temperatures during running and is symptomatic of cohesive wear [26].

Overall, when operating under severe loading conditions PEEK shows considerably less wear than other reported materials [2]. Although the wear rates in some of the more severe tests are high, there was very little visible wear at lower temperatures. Even above the glass transition temperature the material maintained its mechanical strength until approximately 250 °C, when the surface began to soften.

4.2. Measured temperature and friction

Fig. 9 shows the temperature profile for the 28% slip-ratio, 200 N test. It can be seen that the maximum temperature (101.2 °C) is generated at the meshing point of the two discs. The temperature variation across the disc (line B) represents a flash temperature increase at the point of contact and the proportion of heat dissipated from the discs during rotation. It can also be seen that the upper disc is running hotter than the lower disc; this is due to the top disc rotating at a higher speed thereby having less time to cool down between rotations. The variation in temperature from the centre to the outer edge of the disc (line A) is due to a combination of convection, conduction and radiation from the disc to the surrounding area [27].

The infrared temperature profiles shown in Fig. 10, illustrate how the temperature of the contacting discs, run at 14% slip-ratio and 400 N load, progressed over time. The test samples began to expel large amounts of wear debris after 80 min. This can be related to the temperatures exceeding the Relative Thermal Index (RTI) of PEEK for mechanical use with impact (180 °C) [17].

Through use of Eq. (2) the maximum flash temperature for the test combinations can be found. Table 2 summarises these results. It is important to note that the expulsion of wear debris from the interfacial region between the discs reduced the bulk and flash

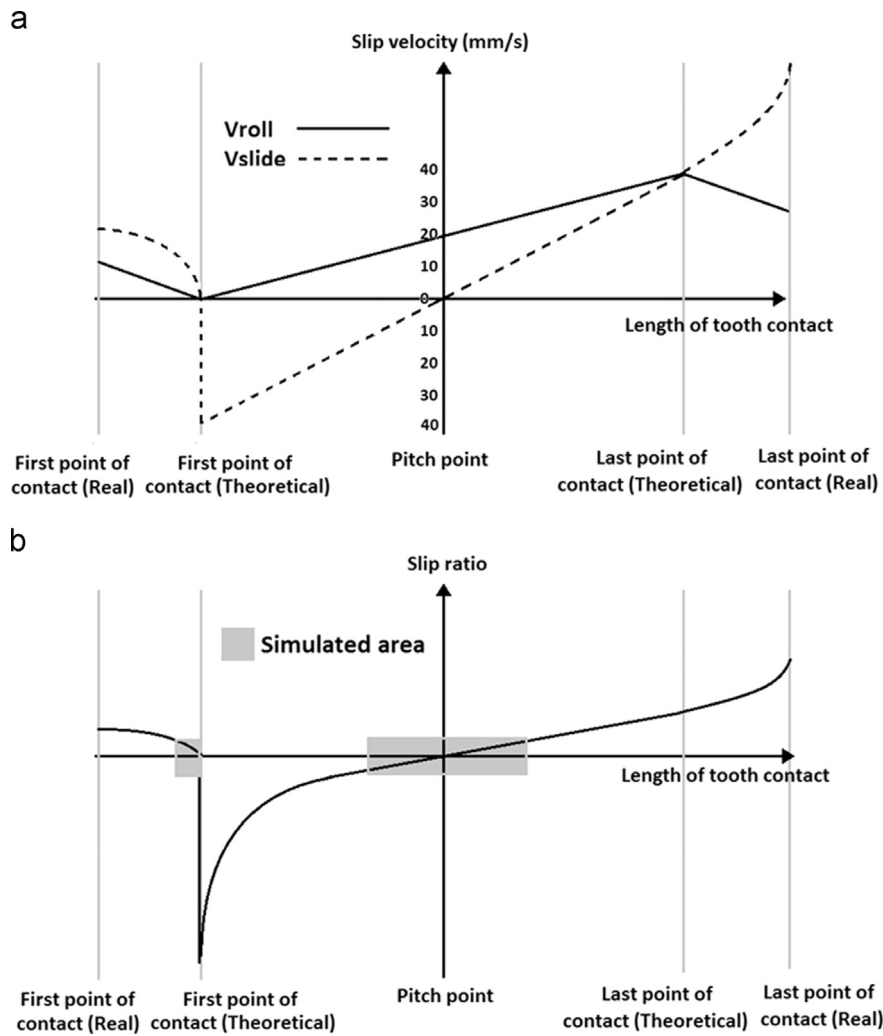


Fig. 4. (a) Shows the sliding velocity (V_{slide}), rolling velocity (V_{roll}), and (b) the slip ratio for a polymer gear contact: grey areas representing simulated areas when using twin disc testing.

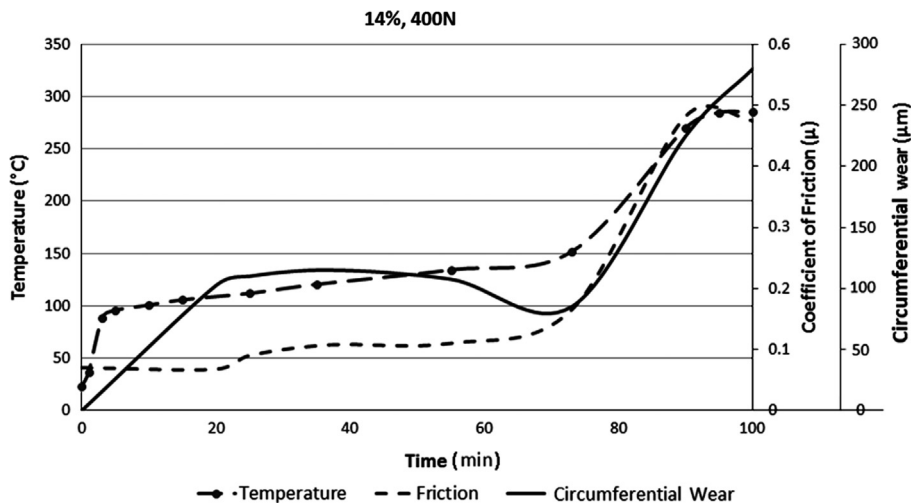


Fig. 5. Temperature, friction and wear results for 14% slip ratio and 400 N test.

temperatures, and so there was not a strong correlation between this flash temperature and test severity. However, it is possible to see that as slip-ratio and load increased, so too did the flash temperature.

Temperature dominates all aspects of the performance and design of polymer machine elements; it has an influence on the failure modes found in components such as gears, limiting the operation to lower temperatures. For the tested PEEK discs, heat is

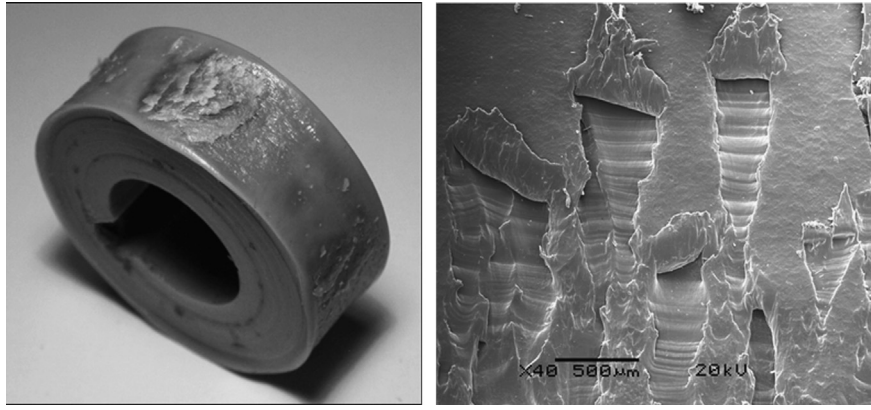


Fig. 6. Surface damage in 14% slip ratio and 400 N tests.

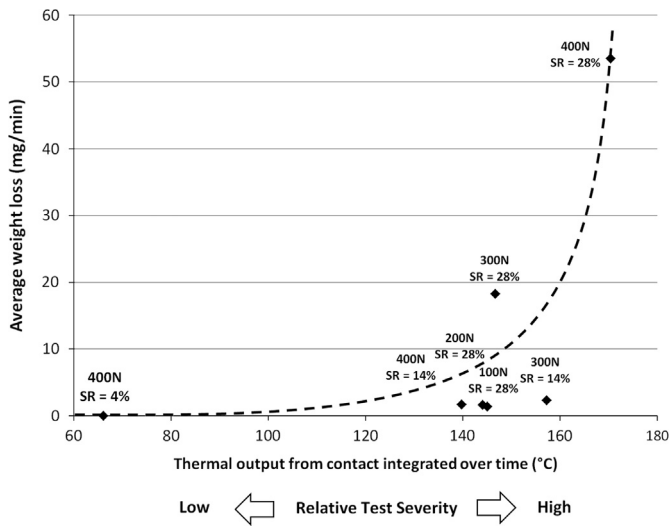


Fig. 7. Average wear rate versus test severity for twin disc tests.

generated and dissipated through a complex combination of mechanisms. These are:

- Heat generation through hysteresis losses, generated as a result of viscoelastic deformation and which is mainly converted into heat.
- Frictional heating, caused by the slip ratio that has been applied to the test discs.
- Heat conducted through drive shafts of the discs.
- Heat loss through ambient radiation.

Koffi et al. first modelled the components of heat generation for a polyamide 6/6 gear [28]. It was shown that frictional heating was responsible for the largest proportion of the heat generated. This was found to be the case except for pure rolling (i.e. at the pitch point of the gears, where sliding velocities are zero) and during tooth contact outside the line of action where tooth deformation is increased [20]. Both Hooke et al. and Kukureka et al. confirm that this is the case for polyamide roll/slide twin disc tests [1,2,7].

Fig. 11 shows the temperature plots for all test conditions and the relationship between measured friction and temperature. It can be seen that frictional heating is one of the main contributing factors to the heat generated by the discs. In addition, a rapid increase in the coefficient of friction can be seen as the temperature goes beyond the glass transition temperature of the material. This transition temperature, and its effect on friction and wear, has been seen in polymer twin discs and fretting tests [2,29].

After an initial period of testing, the material was observed to discolour, possibly due to thermal degradation, becoming predominantly darker as temperatures increased. This phenomenon was also observed by Rae et al. who described a discoloration in PEEK as temperatures exceeded 200 °C [30]. Following the discoloration of the test samples, the observed wear rate substantially increased.

Operation near or above the glass transition temperature of PEEK (143 °C), and below the principal crystalline melt transition (335 °C), will induce enthalpic relaxation. Crystallinity measurements were taken following the tests to see if there is any evidence of this (Fig. 12). It was found that crystallinity values of 30–35% were present in all tests that exceeded the glass transition temperature of the material. These values are 3–8% higher than those measured on untested samples, suggesting that enthalpic relaxation had occurred. This is supported by the conclusions of Ostbery and Seferis who described the effects of annealing above the glass transition temperature on the crystallinity of PEEK [31]. Additionally, the influence of molecule orientation as a result of the high shearing friction forces present in this series of testing can lead to additional enthalpic relaxation during operation [32]. However, due to the increased wear rate at the end of testing, much surface material was removed as debris. Thus, the crystallinity measurements are, of necessity, only able to provide limited information relating to this stage of the testing.

5. Conclusions

The wear mechanisms of poly-ether-ether-ketone running against itself in non-conformal, unlubricated rolling-sliding contact have been investigated over a range of loads and slip-ratios. Overall, the possibility of using PEEK in low slip ratio conditions, for both low and high loads, has been demonstrated with high temperature operation being possible despite an increase of wear. It has been shown that wear, friction and temperature increase as the slip ratio and the load are increased. However, the wear rates are significantly lower than for other polymers tested using the twin-disc configuration. Failure mechanisms observed on the contact surfaces included surface melting and contact fatigue failures particularly in the more severe high slip-ratio, high load conditions.

To conclude, the use of these results in conjunction with the design process can aid in the development of more effective, highly loaded, polymeric gear systems. It has been shown that the reduction of load and slip-ratios help to reduce the generated temperature, and its associated effects around the pitch-point and the region of premature contact. Thus, the possibility of optimising polymeric gear tooth geometry to reduce the slip-ratio away from

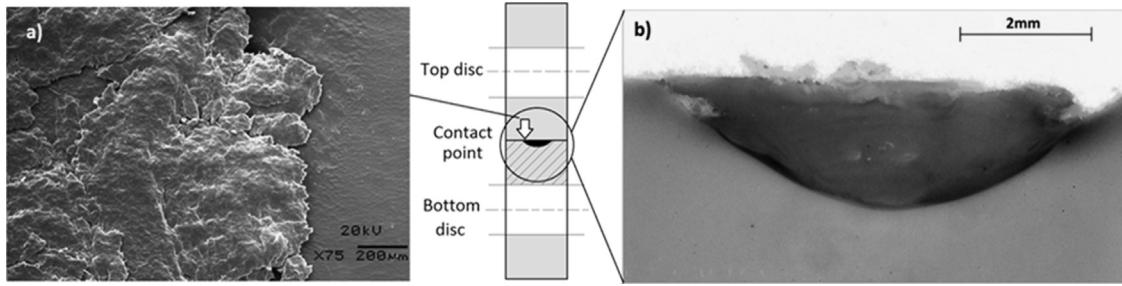


Fig. 8. (a) (Left) the wear debris on the disc surface, and (b) (Right) the cohesive region in the centre of the sectioned and polished disc associated with the highest temperatures.

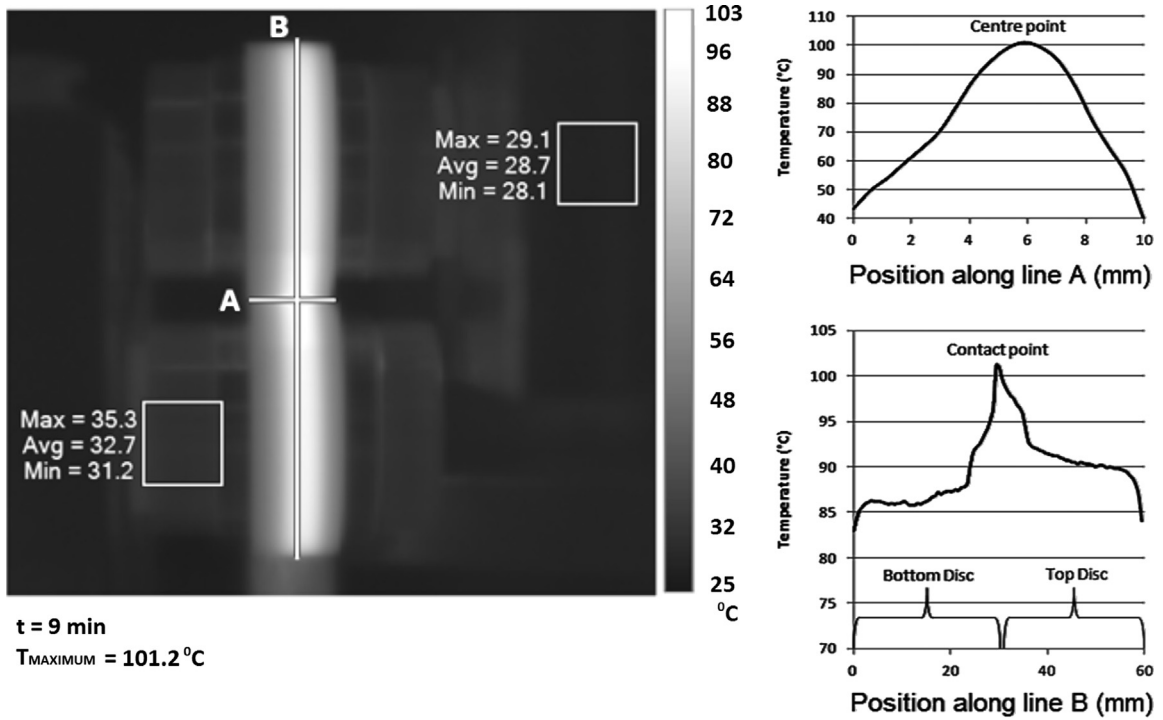


Fig. 9. Infrared temperature profile for 28% SR, 200 N test at 9 min, and temperature plots showing measured temperature variation during disc rotation.

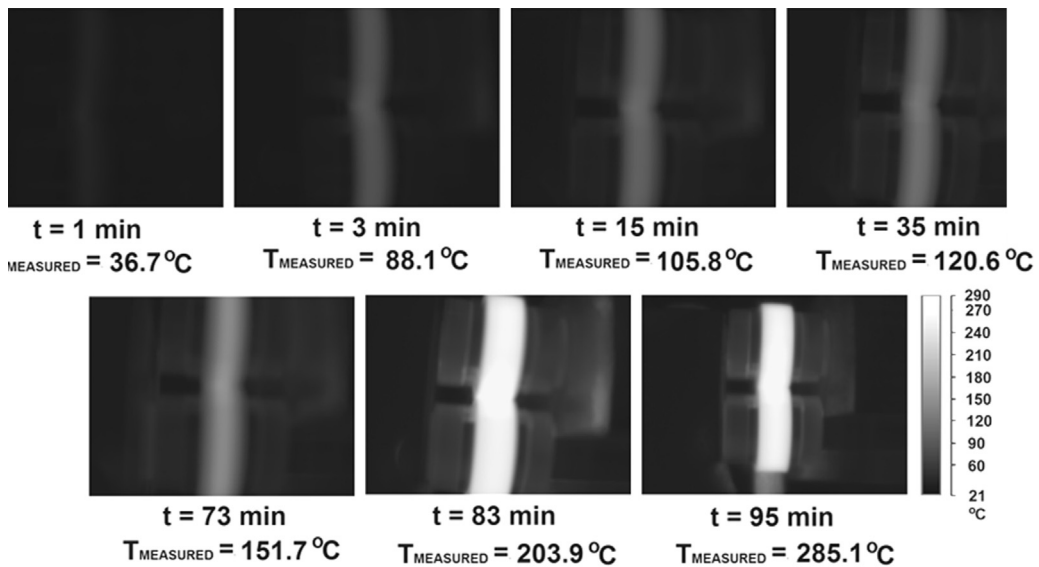


Fig. 10. Infrared temperature plots for 14% SR, 400 N test.

Table 2

Showing the estimated interfacial temperature (T_i) and associated flash temperature rise (T_f) for the test conditions.

| Load | Slip ratio | | |
|------|---|--|--|
| | 3.92% | 14.29% | 28.57% |
| 400 | $T_f=2\text{ }^\circ\text{C}$, $T_i=104\text{ }^\circ\text{C}$ | $T_f=14\text{ }^\circ\text{C}$, $T_i=299\text{ }^\circ\text{C}$ | $T_f=26\text{ }^\circ\text{C}$, $T_i=296\text{ }^\circ\text{C}$ |
| 300 | | $T_f=13\text{ }^\circ\text{C}$, $T_i=298\text{ }^\circ\text{C}$ | $T_f=21\text{ }^\circ\text{C}$, $T_i=302\text{ }^\circ\text{C}$ |
| 200 | | | $T_f=15\text{ }^\circ\text{C}$, $T_i=303\text{ }^\circ\text{C}$ |
| 100 | | | $T_f=9\text{ }^\circ\text{C}$, $T_i=295\text{ }^\circ\text{C}$ |

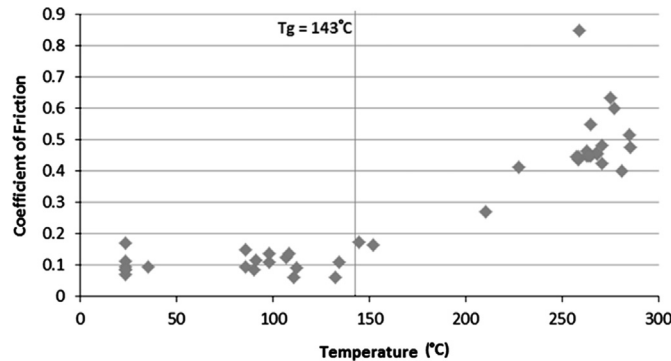


Fig. 11. Measured temperature versus coefficient of friction for all test conditions.

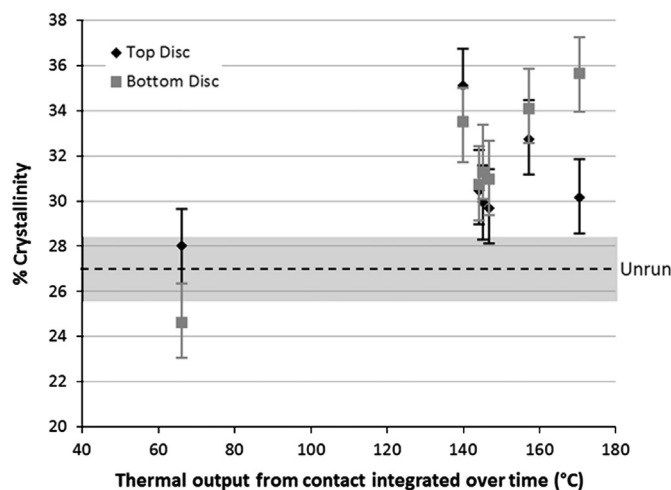


Fig. 12. Crystallinity of PEEK discs versus test severity following testing.

the pitch point will potentially increase the service life and reduce the occurrence of those wear mechanisms shown in the high slip-ratio tests.

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