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Applications of dry film lubricants for polymer gears

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ABSTRACT

Polymer gears can be run without lubrication, however, high running temperatures, driven by high contact friction, mean that the operating life of these gears, especially in medium to high power transmission applications, tends to be low and limited by wear. This paper describes an attempt to control friction and wear by reducing the running temperatures by using a series of solid lubricant coatings deposited on flanks of the polymer gear teeth. Four potential coatings were selected, viz. molybdenum disulphide (MoS_2), graphite flake, boron nitride (hexagonal) and poly-tetra-fluoro-ethylene (PTFE). Each coating was used with both reinforced and unreinforced poly-ether-ether-ketone (PEEK) and unreinforced polyamide (PA). Tests were carried out on coated-coated, coated-steel and coated gears running against uncoated gears. Wear rates (in the form of weight loss) and running temperatures were recorded. Results indicate that PTFE provided the greatest reduction in frictional forces and that failure mechanisms were predominately in delamination of the coatings and abrasive wear.

1. Introduction

Developments in materials and gear technology have resulted in the increasing use of polymers in machine elements. Their low cost (when injection moulded), light weight, resilience and their ability to operate under dry, unlubricated conditions all provide potential benefits. Polymer gears, when used in moderate power transmission applications without lubrication, are required to have potentially conflicting tribological (low friction, high resistance to wear) and mechanical properties (tooth stiffness, flexural strength). This situation is further complicated by the complex loading and contact phenomena that change throughout the meshing cycle. As the transmissible power levels increase, problems of surface temperatures arise due to the frictional losses between mating gear teeth [1,2].

Historically, the low modulus of polymers was considered to be desirable since both noise and contact forces were reduced during motion. However, recent research on polymer gear noise has shown that friction also plays a very important role in noise generation and wear [3,4]. In addition to this, the modulus of a material can significantly alter the path of contact between gear teeth, promoting premature and extended contact [5]. In light of this, efforts have been made to increase the modulus of polymers by fibre reinforcement.

The inclusion of reinforcing fillers can alter the wear mechanism. Typically, the exposure of a reinforcing element results in abrasive wear as the filler damages the matrix of composite counterface. Hooke et al. [2] investigated the effect of glass fibre reinforcement on the wear properties of polyamide discs (and gears). It was found that the reinforcement was dispersed within the matrix and once the matrix layer was removed, they observed that the most dominant form of wear was abrasion. In an attempt to balance required tribological and mechanical properties, Cropper [6] moulded an unreinforced frictional layer, which was moulded around an undersized reinforced gear, producing a dual phase gear. Extensive testing indicated that frictional wear was still an issue and aggressive abrasive wear of the outer tribological layer meant that the life expectancy of these gears was low.

The temperature dependence of polymer gears is well documented [1,7,8]. Currently, the understanding and characterisation of many types of gear failure mechanisms are based on a material's response to the combination of temperature and load applied to the system. There have been many attempts to reduce temperatures in polymer gears by reducing surface friction, with varying degrees of success. These include the use of PTFE filled polymer composites to reduce temperature and wear; although the high filler content required reduced the mechanical properties of the gears [9,10]. Tooth geometry has also been modified to reduce the specific line loads carried by individual teeth

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(increasing the contact area between teeth), and by the introduction of cooling holes throughout the gear body [7,11]. These modifications are thought to reduce the fatigue and wear resistance of individual teeth.

Wear accounts for the majority of polymer gear failures and has been widely reported [10,12–15]. Rao et al. [10] showed that controlling the friction generated at the meshing point has a positive effect on the wear resistance of polymer gears. All authors suggest that reducing flash temperatures reduces gear wear. The wear of polymers is significantly different to that of metals; they have much lower modulus values, lower strengths and melting points. In addition, a lower thermal conductivity in polymers often means that thermal effects are significant as noted in the previous paragraph.

The importance of friction and running temperatures for polymer gears is highlighted as well as many techniques are employed to control these parameters. However, very little research has been conducted on modifying the tooth flanks surfaces, in particular the application of dry film lubricants to reduce by friction and temperature. There is currently very little literature covering solid lubricant coatings on polymer substrates. Kosydar et al. [16] investigated the wear of boron nitride layers on a polyurethane substrate in comparison with both an uncoated version and a graphite-embedded version. Pin-on-disc wear tests showed that the boron nitride coatings effectively decreased the coefficient of friction of the polymer up to four times compared with that for uncoated polyurethane and polyurethane coated with graphite. However, there was no discussion of the wear mechanisms involved in this investigation.

Studies on hard coatings deposited onto soft substrates can provide an insight into possible material interactions when coated systems are subjected to a mechanical load. Holmberg et al. [17] investigated the tribology of thin coatings. Holmberg et al. concluded that a hard coating, when deposited onto a softer substrate, can decrease friction and wear by preventing ploughing. However, it was also found that thin coatings are susceptible to fracture caused by substrate deformation despite typically exhibiting residual compressive stresses. Samur et al. [18] found that the wear resistance of ceramic coatings on a polymer substrate, when subjected to a reciprocating Al_2O_3 10 mm diameter ball, increased with increasing hardness.

Finally, Nozawa et al. [19] looked at the tribological properties of polymer sheets adhered to steel gear teeth. They found that the tribological properties were improved and noise was reduced, although the tests were limited by the weak adhesion of the polymer sheets to the steel gear. Thus, development of a polymer gear with a solid-lubricant coating could potentially reduce the frictional losses, interfacial temperatures and ultimately the wear of polymer gear teeth.

This paper therefore describes an attempt to control the tribological properties of dry-running polymer gears through the application of dry film lubricants applied directly to the gear tooth flanks. The mechanical characteristics ought to be maintained by the applications of the coatings, as well as improving power transmission levels in comparison with other forms of frictional control and with uncoated gears.

2. Material and methods

2.1. Gear materials and geometry

Tests were conducted on injection-moulded polymer gears with two separate geometries, shown in Fig. 1. The first gear geometry is the Birmingham standard employing a 20° pressure angle and is described more fully in [3], whilst the second employs a 30° pressure angle [3]. The mechanical properties of the polymer gear materials used are listed in Table 1, the materials were:

- Unreinforced polyamide (PA) (gears with a 20° pressure angle)
- Unreinforced poly-ether-ether-ketone (PEEK 450G) (gears with a 30° pressure angle)
- Carbon fibre reinforced PEEK 450CA30 (containing 30 wt% carbon fibre and gears with a 30° pressure angle)

2.2. Specification of coatings

Four industrially available coatings were selected, molybdenum disulphide, graphite flake, boron nitride (hexagonal) and PTFE powder. The composition of each dry-film lubricant is shown in Table 2. Note that the percentage of the dry film lubricant per coating is a theoretical estimate based on the densities of the raw materials for each coating.

To ensure that the coatings were sufficiently adhered to the polymer, cross hatch cuts of a predetermined depth were made on the coating. ISO certified adhesive tapes were then used to test adhesion quality. Once adhesion quality had been assessed, the tooth flanks were prepared using aluminium oxide grit. The coatings were then applied using a conventional air-atomising spray gun at an ambient temperature of 16–18 °C. The curing schedule was as follows and is according to an empirical schedule developed at Indestructible Paint Ltd:

- 10 min flash off at 16–18 °C following spraying

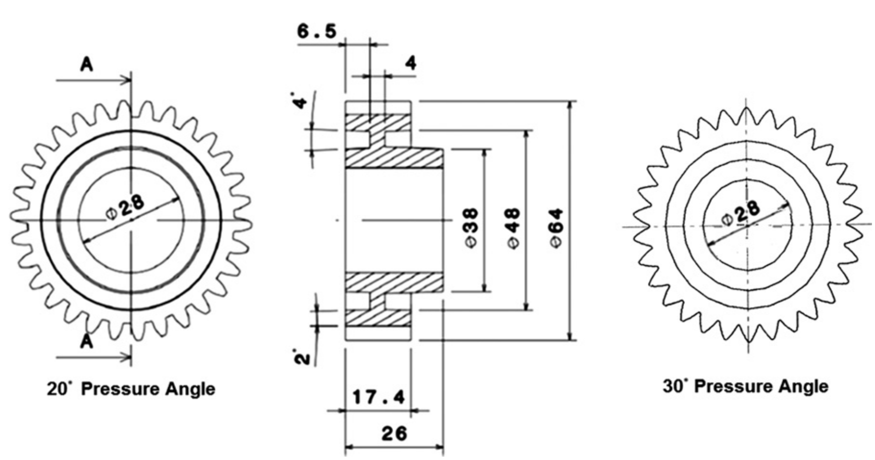

Fig. 1. Gear geometry showing both 20° and 30° pressure angle gears.

Table 1
Material properties of the gear materials.

Material	Polyamide (PA66)	Polyetheretherketone (PEEK 450G)	PEEK 450CA30	Test method
Tensile strength, yield 23 °C	82 MPa	100 MPa	260 MPa	(ISO 527)
Tensile elongation (Break, 23 °C)	25%	45%	1.7%	(ISO 527)
Tensile modulus 23 °C	3.0 GPa	3.7 GPa	25 GPa	(ISO 527)
Flexural modulus 23 °C	2.8 GPa	4.1 GPa	23 GPa	(ISO 178)
Melting point	263 °C	343 °C	343 °C	(ISO 3146)
Glass transition temperature	66 °C	143 °C	143 °C	(ISO 3146)
Heat deflection temperature (1.8 MPa)	74 °C	152 °C	336 °C	(ISO 75A-F)
Thermal conductivity	0.3 W/m K	0.29 W/m K	0.95 W/m K	(ISO 22007-4)

Table 2
Density, hardness, wt% of DFL, vol% of DFL within the dry film and for the four test coatings.

DFL type	Density (g/ml)	Vickers hardness (Hv)	DFL (wt%)	DFL (vol%)
Molybdenum disulphide	1.56	65	47	11.6
Graphite	1.39	60	43	22
Boron nitride	1.38	72	43	22.7
PTFE	1.39	5.9–6.5	46.9	25.3

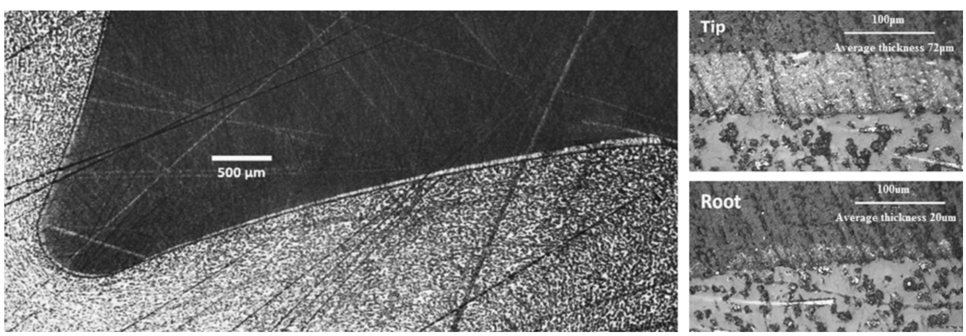

Fig. 2. Showing the variation in coating thickness along the gear tooth flank (note: coating thickness was determined after sectioning and polishing).

Table 3
Thickness of coatings.

Coating	Thickness at root (μm)	Thickness at tip (μm)	Average thickness (μm)
PTFE	15	34	24.5
Boron nitride	44	123	83.5
Graphite flake	20	72	46
MoS_2	14	52	33

- 1 h at 190 °C in an air circulating oven
- Cooled and de-masked prior to visual inspection for contaminants in the dry film

To establish the coating thickness, test gear samples were mounted in a cold set resin and measured using a microscope. Variation in coating thickness was observed along the flank of the gear teeth as shown in Fig. 2. Table 3 shows the coating thickness for each material.

It can be seen that the coating deposition method resulted in a variation in the thickness of the coating from root to tip, the coatings being thicker at the tip. Scanning electron microscopy was used to further examine the deposited coatings (Fig. 3). The topography of the various DFL coatings were visually different and surface characteristics such as cracks and porosity can be identified. Finally, the average surface roughness of coated gear tooth flanks was measured using an optical interferometer (Table 4).

2.3. Experimental methods

A test procedure has been developed to consider the influence of meshing conditions on the performance of coated polymer gears. Tests were conducted at 1500 rev min^{-1} with an applied load of 7 Nm. Coated test gears of the same geometry were run in mesh against a similarly coated gear and compared to bench mark uncoated gears. If a particular DFL coating proved successful in reducing wear and running temperatures, then further permutations of the successful coating combinations were conducted. These included uncoated/uncoated, uncoated steel, coated/ uncoated, coated/ steel and coated/coated gear meshes and allowed the efficacy of the coating to be fully assessed.

Experiments were conducted on a power re-circulating, closed loop test rig developed specifically for testing polymer gears (Fig. 4) and as described in Hoskins et al. [3]. Temperatures were measured using non-contacting, K type, infra-red thermocouples. Three of these thermocouples are shown in Fig. 4 adjacent to the

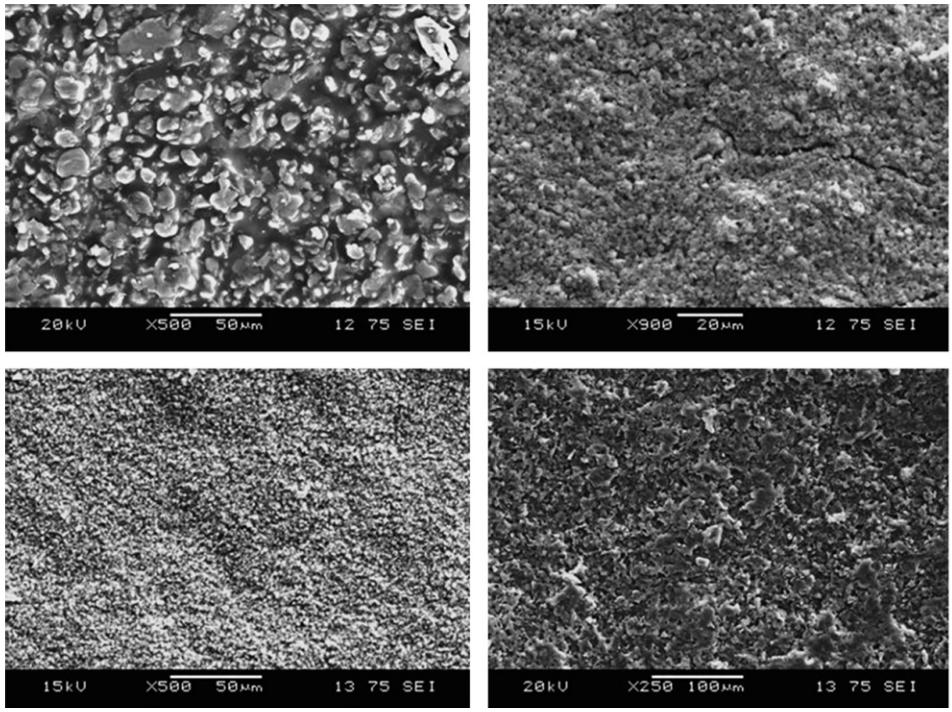

Fig. 3. The morphology of as processed coatings deposited on polymer gears. Clockwise from top left: PTFE, graphite, MoS2, boron nitride.

Table 4
Average surface roughness of deposited coatings.

Coating	Surface roughness (μm)
PTFE	1.93
Boron nitride	1.28
MoS_2	1.15
Graphite Flake	1.04

test gears; the centre probe being used to measure the temperature at the pitch point of the gears.

It has not been possible to measure the coefficient of friction between the meshing gears directly using the experimental rig selected for this study. However, the effectiveness of the selected DFL's in reducing frictional losses could be inferred from measured running temperatures.

Temperature dominates all aspects of plastic gear performance and thus limits their operational range. It has a detrimental effect on mechanical material properties and has an influence on all of the major polymer gear failure modes. Heat is generated during the meshing of polymer gear teeth through a complex combination of mechanisms:

- Hysteresis loses, generated as a result of viscoelastic deformation and mainly converted into heat.
- Frictional heating, caused by the kinematics of the meshing cycle (i.e. the combined rolling and sliding).
- Heat conducted through drive shafts of the gears.
- Ambient radiation.

In most applications in which polymer gears are used, ambient radiation and conduction through the drive shafts represents a negligible contribution to overall heat generation and can be all but eliminated through careful design of application.

Frictional heating has been shown to be the far most dominant mechanism of heat generation. Koffi et al. [20] first modelled the components of heat generation per unit facewidth for a polyamide 6/6 gear. Fig. 5 shows the results of the model for both frictional and hysteretic heating. It predicts that heat generated by hysteresis will be much lower compared with that generated from friction. This is shown 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 [5]. Both Hooke et al. and Kukureka et al. [2,15] confirm this for polyamide roll/slide twin disc tests. Thus it can be assumed for the experiments described in this paper that frictional heating is much greater than other mechanisms of heat generation in the test gears. With the constant load, rotational speed and laboratory conditions, differences in measured temperatures are attributed to a change in the sliding friction between the meshing tooth flanks.

Wear of the test gears was measured by mass loss using two precise analytical balances. To do this the meshing gears were stopped periodically, (noting their relative positions), then removed and after weighing they were replaced as closely as possible to the original positions. This was done to minimise the loss of a transfer film whilst weighing and to minimise the disruption to the meshing action of the individual teeth. To account for changes in moisture content of the test gears, the mass of a control gear, mounted on the pivot block assembly during running, was measured at the same time as the test gears. Wear rate (as a percentage mass loss, R%) was calculated using Eq. (1).

$$R\%_i = \frac{\left(\frac{M_i}{M_0}.P_0 - P_i\right)}{P_0} \cdot 100 \tag{1}$$

Where P_0 is the original mass of the gear, P_i is the current gear mass, M_i is the current control gear mass and M_o is the original mass of the control gear. Alternative methods of measuring wear, such as by measuring the displacement of the loading arm, have been shown to be inaccurate [21].

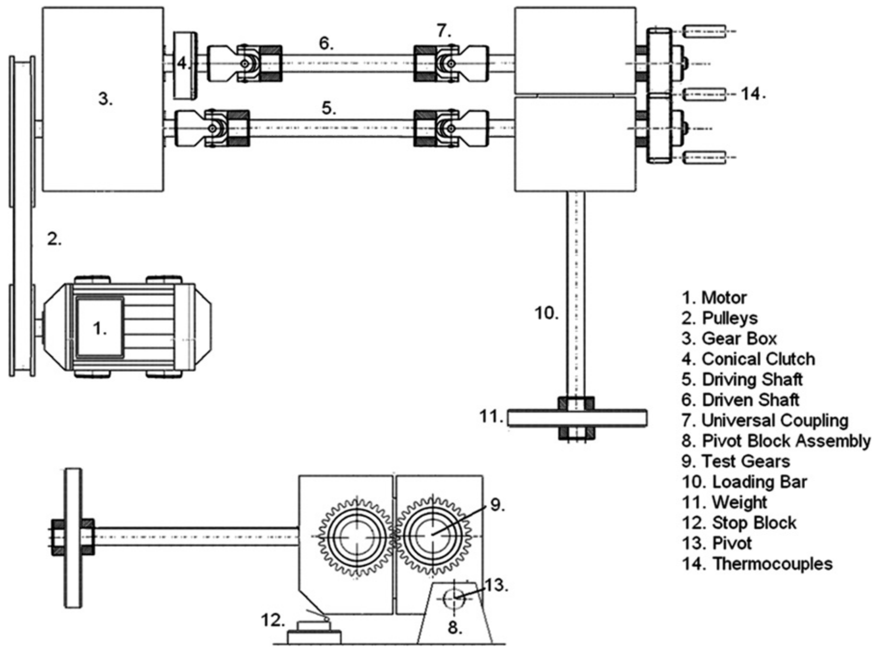

Fig. 4. Power re-circulating, closed loop test rig.

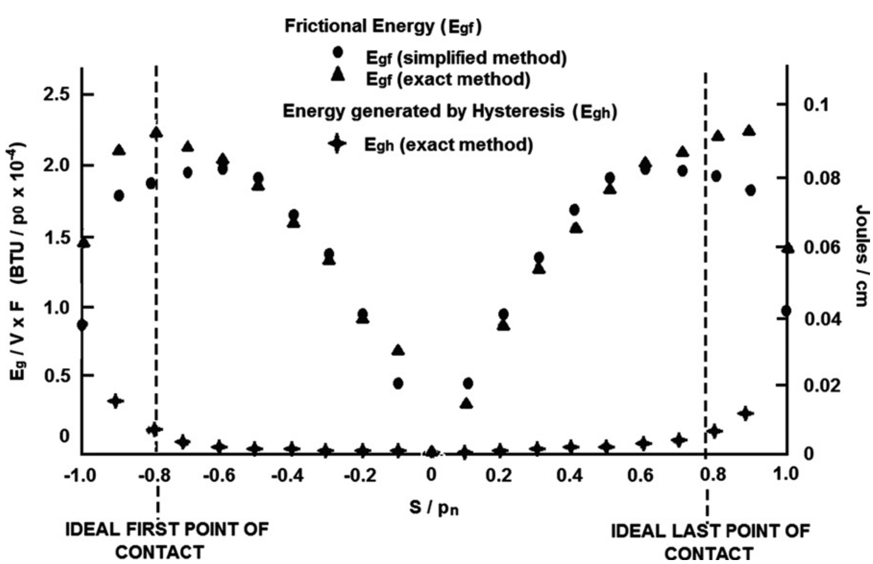

Fig. 5. A comparison of the calculated energy created from friction and hysteresis for a pair of meshing polyamide gears (note: Gears modelled with 20 teeth, transmitting 53 Nmm with a storage modulus of 2.07 GPa, a loss modulus of 0.05 and a co-efficient of friction of 0.1) (after 19).

3. Results and discussion

Table 5 provides hardness values for the three polymeric gear materials tested. In comparison to the Vickers Hardness values shown in Table 2, the polymers used in this study are harder than PTFE yet softer than the three other industrially available DFL coatings.

3.1. Polyamide

Fig. 6 shows both the relative weight loss and the mesh temperature against the number of cycles for meshing uncoated and coated polyamide gears as well as two identifiable regions of wear and temperature. The addition of DFL coatings to the polyamide gear surfaces reduces the measured temperatures up to 30 °C compared to the uncoated gears, reducing the running temperature to below the glass transition temperature of the material (T_g=47 °C (0.15% $H_2$0)) [22].

Table 5
Polymer material hardness (note that values have been scaled from durometer results for comparison).

Material	Hardness		
	Vickers	Rockwell, M	Rockwell, R
PEEK 450 G	26.1–28.5	95–105	119–131
PEEK 450 CA30	49.2–54.7	100–110	121–127
PA 66	16.6–24.8	79–87	114–126

A direct correlation between surface temperature and wear is also evident; the PTFE coating reduces the temperature by approximately 30 °C compared to the uncoated gears during steady state conditions, and also minimises the wear over the same time period. In addition, the graphitic coating provides a significant reduction in surface temperature. Despite that not all

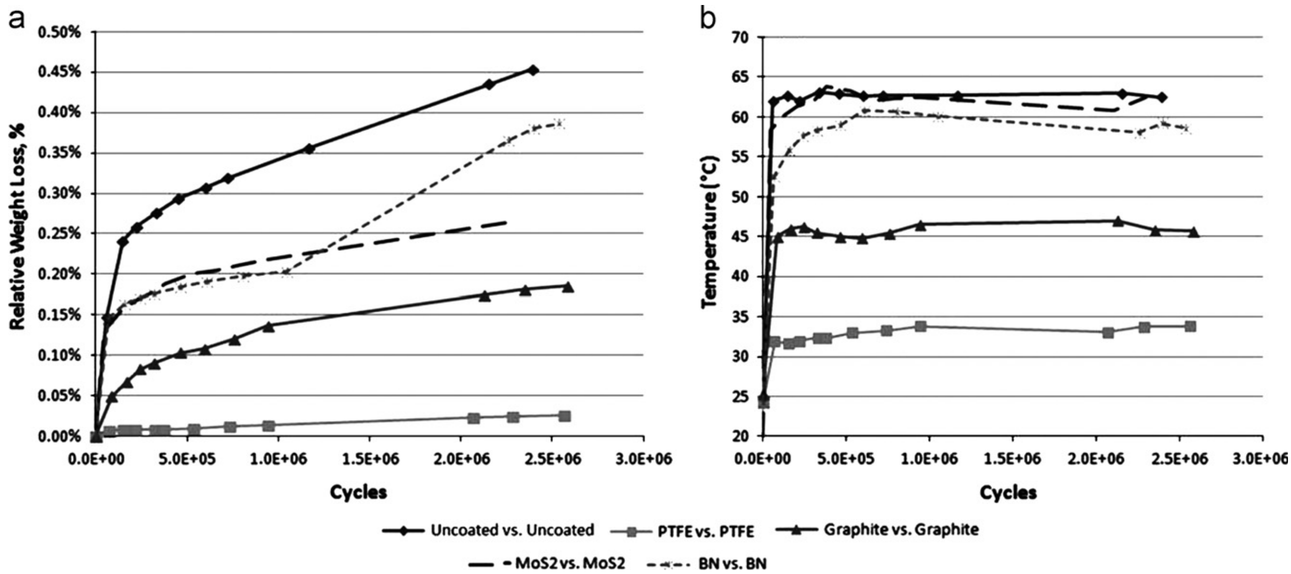

Fig. 6. (a) Wear rate and (b) temperature vs. number of cycles for coated polyamide gears.

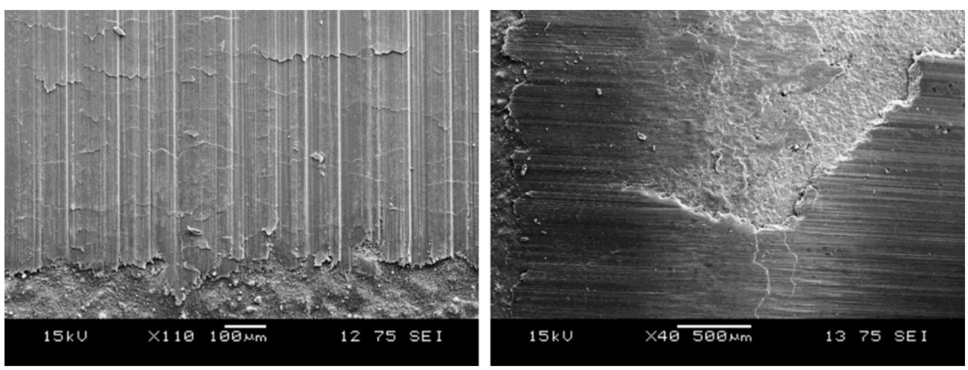

Fig. 7. SEM observation of graphite vs. graphite polyamide contacts.

coatings reduced the temperature significantly, all four of the coatings reduced the wear rate (although the importance of this may be lessened as the coating is completely removed). The boron nitride coating initially had a significant effect on the wear rate of the gear mesh. However, as seen in Fig. 6a and observed on the gears, after 1×10^6 cycles a combination of delamination of the coating and abrasive wear meant that the coating was almost completely removed. This is likely to be a result of the brittle nature of the coating in comparison with the compliant substrate.

The microstructure of the tooth flank of graphite vs. graphite polyamide contact is shown in Fig. 7. Wear tracks are clearly visible in the coating; a result of abrasion during the motion between the sliding gear teeth. The presence of surface cracks (visible in the as processed specimen, Fig. 3) is likely to increase the permeability of oxygen moisture into the coating, thereby reducing its strength, coating adhesion and associated lubricating performance. The ability of polyamides to moisture absorption is well documented in the literature [22,23]. Jia et al. showed that an increase in the moisture (%H_2O) dramatically reduces the glass transition temperature for the material. Once the gears are running, the surface cracks can be seen to increase in size and frequency and failure of the coating occurs when delamination of the two surfaces results in dynamic meshing forces and adhesion between the two contacting surfaces. The combination of forces and adhesion mean that the coatings are removed. This delamination suggests that the interfacial adhesion between the coating and the substrate is not sufficient for the dynamic forces involved.

Fig. 8. Pitch line fracture in a boron nitride coated PA gear run in mesh with a steel gear.

The reduction of temperature in polyamide is imperative in the prevention of pitch line fractures [8]. High temperatures were generated by both a steel and polyamide gear coated in boron nitride. During the meshing cycle temperatures well in excess of the T_g of 50 °C (i.e. ~72 °C after 1500 cycles) were measured causing the gear tooth to fail by pitch line fracture (Fig. 8). PTFE was shown to be the most successful coating, reducing the meshing temperatures and wear for both similar and dissimilar material contacts. A full summary of the results for PTFE is given in Table 6.

Table 6
Temperature and wear rates for PTFE coated Polyamide.

Gear material	Running temperature °C	Initial wear rate (% weight loss/1 × 10^5cycles)	Steady state wear rate (% weight loss/1 × 10^5cycles)	Total wear (% weight loss after 2 × 10^6cycles)
UC vs. UC	63	0.1622	0.0080	0.43
UC vs. Steel	52	0.0787	0.0053	0.22
PTFE vs. UC	40	0.0226	0.0027	0.07
PTFE vs. Steel	41	0.0220	0.0036	0.08
PTFE vs. PTFE	33	0.0089	0.0009	0.02

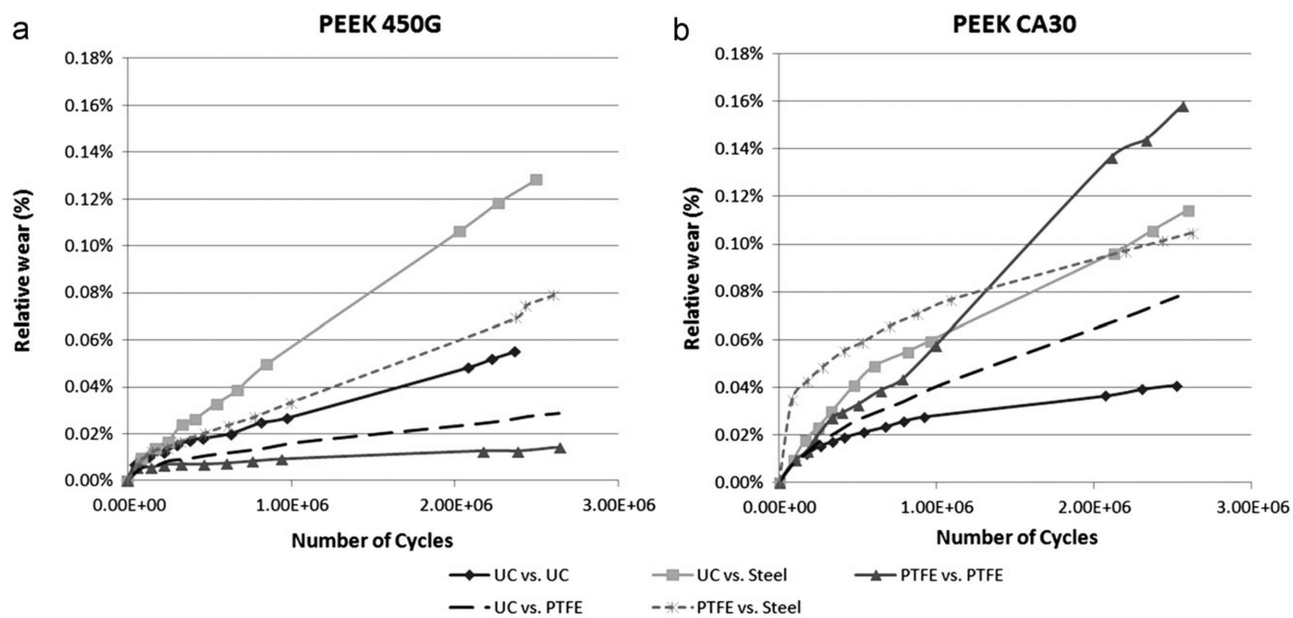

Fig. 9. (a) Wear rate and (b) temperature vs. number of cycles for coated as processed PEEK gears.

Table 7
Temperature and wear rates for PTFE coated as processed PEEK.

Material mesh	Running Temperature (°C)	Initial wear rate (% weight loss/1 × 10^5 cycles)	Steady state wear rate (% weight loss/1 × 10^5 cycles)	Total wear (% weight loss after 2 × 10^6 cycles)
UC vs. UC	35	0.0153	0.0020	0.045
UC vs. Steel	36	0.0113	0.0050	0.105
PTFE vs. UC	32	0.0062	0.0010	0.023
PTFE vs. Steel	37	0.0108	0.0025	0.060
PTFE vs. PTFE	32	0.0097	0.0003	0.015

When mated against a more rigid counterface it is usually only the polymer that exhibits deformation. For these materials, whilst the steady-state wear rate is reduced for the uncoated polymer-steel gears, when the coated polyamide is run against a steel counterface the combination of increased meshing forces and localised deformation of the polymer results in an increased wear rate.

3.2. Poly-ether-ether-ketone (PEEK)

Fig. 9 shows the relative wear and temperature against the number of cycles for coated PEEK. It is evident that the PTFE coating reduces both the temperature and the wear and a full summary of the results for PTFE coated PEEK gears is given in Table 7. However, other coatings were not able to influence the gears positively during the running-in period, despite the combination of the increased radius of flank curvature and reduced compliance of the gear material being expected to reduce the contact stresses. Nevertheless, the steady-state wear rates for all of the coatings except MoS_2 were lower compared with those of the uncoated gears.

Fig. 10 compares the wear of the coated unreinforced and coated reinforced PEEK materials. It can be seen that the coatings on the reinforced PEEK material are not as tribologically beneficial as those deposited on the unreinforced material. Therefore no further tests were conducted. The tribological effects of carbon and glass reinforcement have been extensively researched [2, 6, 12 and 13]. Therefore, the effectiveness of the coatings for reducing friction will be dependent on them preventing reinforcing fibres, in this case carbon fibres, from becoming exposed on the gear tooth flanks. In the reinforced PEEK, the harder coatings were initially able to support the load. As the coatings were broken down and the reinforcing carbon-fibres became exposed, abrasive damage increased. This is shown by the abrasion tracks that are visible on the coating in Fig. 11. This also shows that the thickness of the coating was dramatically reduced and that delamination of the coating occurred exposing the carbon fibres in the polymer matrix. Delamination was the most common form of failure in the coatings deposited on the reinforced material. The performance

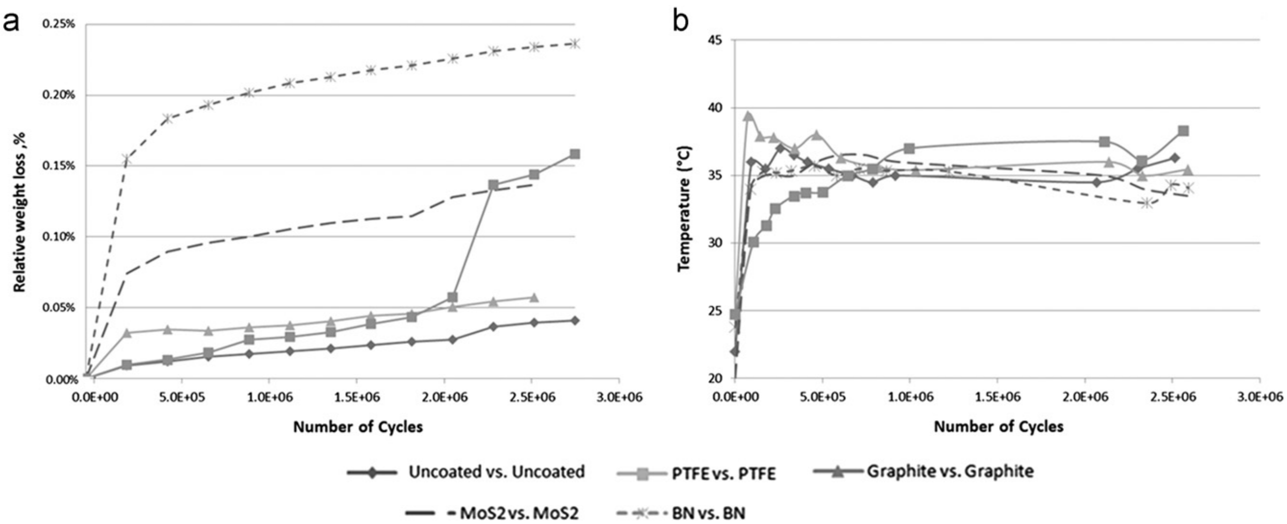

Fig. 10. (a) Wear rate and (b) temperature vs. number of cycles for coated reinforced PEEK gears.

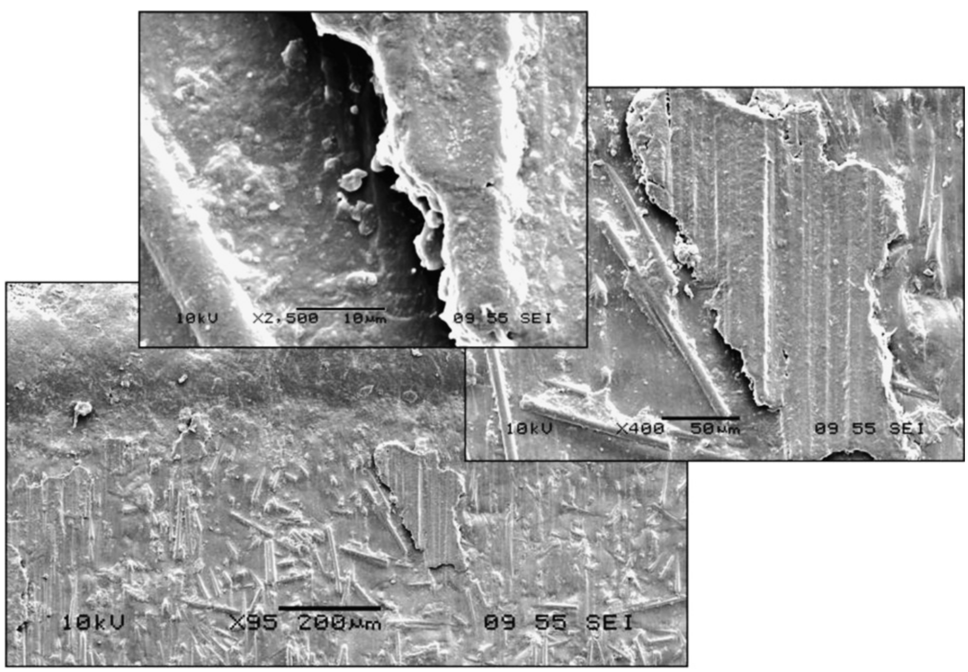

Fig. 11. Delamination and abrasive wear of the BN coating from carbon fibre reinforced PEEK substrate at increasing levels of magnification.

of the PTFE coating decreased significantly when applied to a reinforced composite material. Since the coating was softer than the substrate, the coating was scratched very easily, transferring a proportion of the coating to the counterface. However, a large majority of the coating was lost (i.e. after approximately 1×10^6 cycles, with an increased wear rate beyond this).

3.3. Discussion

Fig. 12 consolidates the observed results and outlines how the coating hardness affects the wear and friction mechanisms. The effects of loading, surface roughness, debris and the sliding speed are all included.

The thickness of the coating affected the wear and friction mechanisms in all of the tests. Fig. 3 showed the variation in coating thickness from the root to the tip of the gear and so it is necessary to separate the wear and friction mechanisms observed in these areas of the gear flank.

The specific loading of individual gear teeth is primarily influenced by the geometry of the gears. An increased pressure angle reduces deflection, increasing contact forces. It was observed that the harder coatings when deposited on the 20° pressure angle PA gears tended to fail by delamination (Fig. 11). This occurred when the interfacial adhesion between the coating and substrate was exceeded by the stresses induced by tooth deflection. However, with softer coatings the wear debris was not always discharged from the system; instead, it was transferred to the counterface. Sliding speed changed the degree of material transfer from the PTFE coating. At the pitch point, due to the pure rolling motion, the softer coatings are not scratched away, but instead transferred to the counterface as shown in Fig. 12.

Once the coating substrate had become exposed, the wear of the gears generally accelerated. This was most significant in the

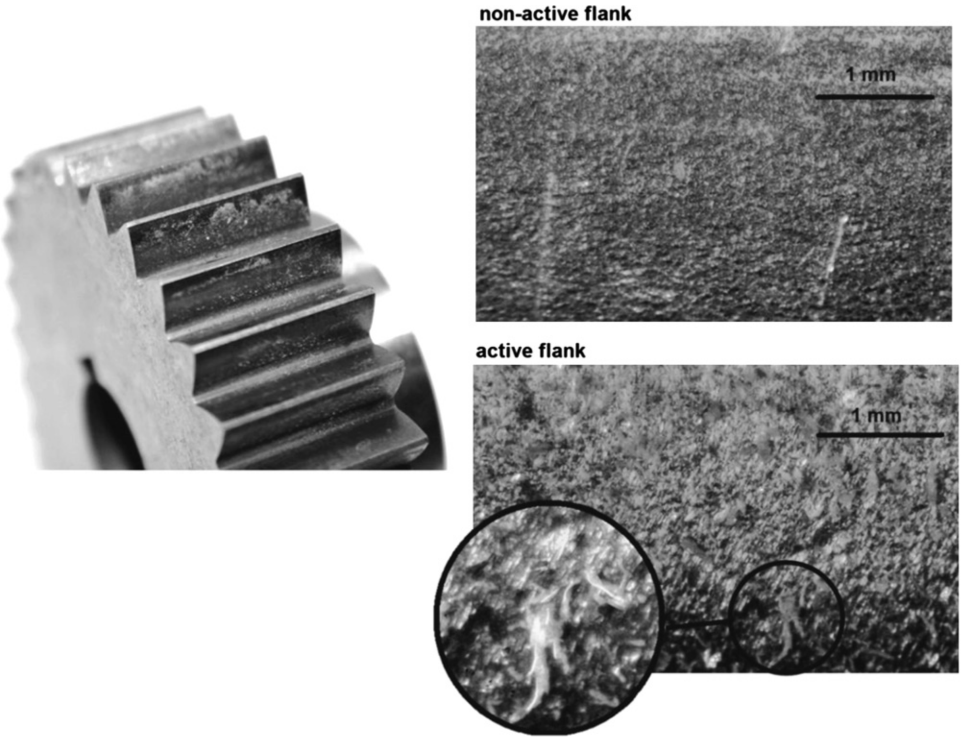

Fig. 12. Soft coating deposited on the steel gear counterface.

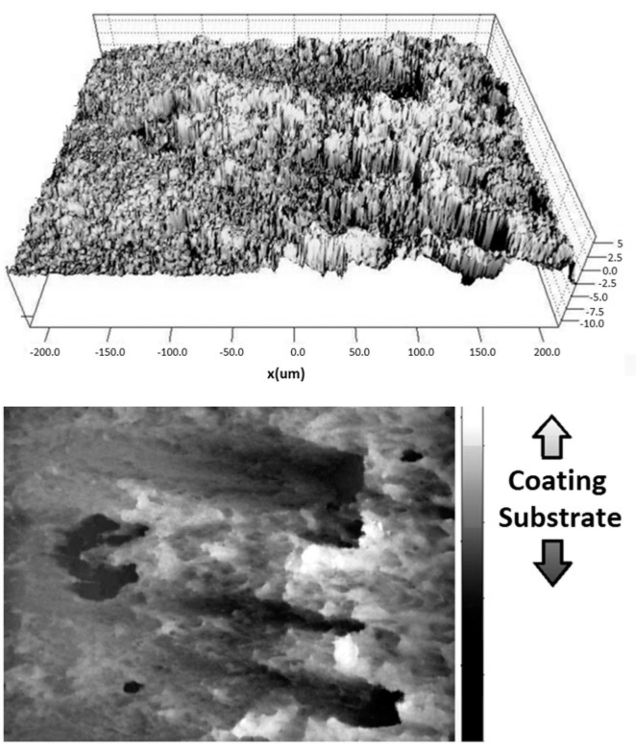

Fig. 13. Surface roughness of graphite on PA (Ra = 1.30 μm) showing areas where the coating has been removed from the surface.

reinforced material. However, the as processed surface roughness was not as significant as the running-in period which served to remove the outer layers of the coatings. Fig. 13 shows an optical interferometer image of the graphite coating on PA. The coating roughness becomes insignificant when compared with the delaminated areas of the coating.

4. Conclusions

This paper has described an attempt to improve the contact properties between dry-running mating polymer gear teeth using a variety of dry film lubricants. The use of graphite and PTFE coatings reduced the friction, associated running temperatures and wear in polymer gears. These improvements could potentially lead to an increase in the power transmission and gear life of the materials.

PTFE provided the most significant improvements in working life with a reduction in running temperature of 30 °C and a reduction in wear of over 90% during the test duration compared with the uncoated polyamide equivalents. The reduction of high local temperatures in the vicinity of the pitch line could potentially reduce the frequency of pitch-line fracture. In the PEEK gears the effectiveness of dry-film lubricants was limited, with coatings deposited on a reinforced substrate predominantly failing due to the abrasive action of the reinforcing filler.

There are limitations of using a dry film lubricant coating. Wear is an accepted factor when using polymeric gears; however, the wear can become significant if used for transmitting high loads. Therefore it is possible that the coating loses its effectiveness over time as the wear exceeds the coating thickness. However, a proportion of the coating may be transferred to the outer layer of the gear as it wears, effectively re-forming the lubricating layer.

In conclusion, the application of a dry-film lubricant coating has been shown to significantly improve the working life of both polyamide and PEEK gear sets. Similar properties can be achieved by the use of a lubricating filler; however, this has been shown to decrease the matrix strength. The use of a low cost, highly effective, external coating is much more desirable.

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