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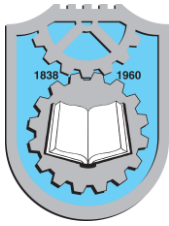
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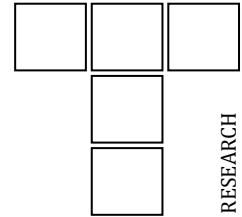
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Wear Performance of Commercial Polyoxymethylene Copolymer and Homopolymer Injection Moulded Gears

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ABSTRACT

Polymer gears are an effective solution to lightweighting, which are highly demanded in the automotive industry. Acetal is one of the most widely used polymer gear materials. In this study, two commercial grades of acetal, homopolymer (POM-H) based and copolymer (POM-C) based, were injection moulded into gears with their wear performance compared. Noticeable differences were discovered in failure mechanism, and thermal and mechanical characteristics, which led to a difference in performance prediction. The service life of over two million cycles was expected under a torque up to 10 N m, with POM-H gears having 35% better service life than POM-C. The differences in the properties of POM-H and POM-C should be considered in future industrial applications such as the replacement for metal gears.

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1. INTRODUCTION

Polymer gear development has shown high potential to replace metal gears, with advantages such as lower weight, lower cost, higher efficiency, and work without lubrication. Previous research revealed that polymer gears could provide about 70% in mass reduction, 80% improvements in inertia, and up to a 9% reduction in fuel consumption over the use of metal in automotive engineering [1]. However, due to a lack of research on material

performance, polymer gear applications are currently limited. Further research and implications for the development and application of acetal gears are therefore timely.

As an engineering thermoplastic resin, and potential substitute for metals, acetal, also known as polyacetal or polyoxymethylene (POM), has many advantages such as high chemical resistance, friction wear resistance, fatigue resistance (especially when normalised with respect to the modulus of elasticity) and

creep resistance [2]. It is applied widely across applications in automotive, medical, electronics, and household product sectors [3-6]. POM was first introduced by the 1953 Nobel Laureate Hermann Staudinger [7]. As a formaldehyde-based thermoplastic, POM is thermally unstable and decomposes into formaldehyde gas at low temperatures. In 1960, Dupont Company developed a commercial homopolymer acetal (POM-H) through a condensation reaction of polyformaldehyde and acetic acid to increase its thermal stability and named it Delrin™. In 1962, a copolymer acetal (POM-C) was developed using the reaction of trioxane, a cyclic trimer of formaldehyde, and a cyclic ether, by Celanese Company [8]. The differences in structure can be seen in Figure 1(a) and 1(b) and are reflected in different properties whereby POM-H has aligned uniform chains which allow for the formation of large crystalline domains. On the other hand, the random comonomer of the copolymer helps block the whole structure even under thermal influences.

Previous studies of POM polymer offered a comparison between POM-H and POM-C. Tajima and Itoh [2] studied the creep rupture strength of the two polymers with different molecular weights. Their work reported that POM-H showed higher creep-rupture resistance than POM-C due to its higher molecular weight and tensile strength. However, POM-C performed better in fatigue resistance and yield strength than POM-H. Archodoulaki *et al.* [9] researched the degradation behaviour of POM which indicated that POM-C was more resistant against thermal oxidation than POM-H. Besides the different chain structures, POM-C had a better thermal aging property than that of POM-H. Hertzberg *et al.* [10] studied the fatigue crack propagation of various polymers and concluded that acetal had the highest fatigue resistance found in semi-crystalline polymers to date and that the POM-H version performed better than copolymer. Stohler and Berger [11] found that different chemical reinforcement methods of preventing degradation could be applied, these included capping of end groups by esterification to POM-H while adding epoxides as a comonomer to POM-C.

As POM is one of a number of materials of interest in polymer gears, previous work has been summarised in recent literature reviews of the field e.g. [12-14]. More specific and pertinent previous studies include aspects such as gear meshing mechanisms [15-17], materials [18-20], and thermal analysis [21,22]. Interestingly many previous studies that have evaluated the performance of POM in gear applications, did not report the type of POM [23,24], i.e. homo- or co-[16,17,25,26]. In addition, most previous studies have assumed commercial POM-H and POM-C as one and the same, indistinguishably, for performance of these polymers in wear and fatigue applications like gears. Limited information has been presented around the comparison between these two types of commercial polymers. Unfortunately, this makes direct comparisons of performance very difficult as it ignores the effects of additives to the polymer formulation which can drastically effect properties such as friction, crystallinity, and tensile strength. The aim of this paper is to evaluate the viability of commercial POM-H and POM-C gears and to compare their performance under wear and fatigue in a gear application relative to previous findings. These performance tests were conducted in a novel test rig, designed and manufactured at the University of Warwick, to study the effects of misalignment on polymer gear contact and to continuously measure the wear of the gear surfaces under constant load conditions.

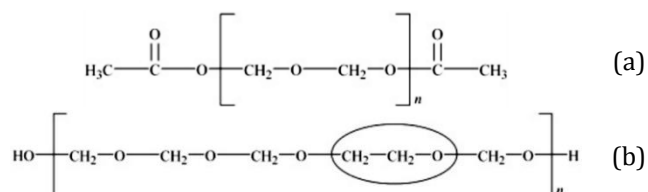
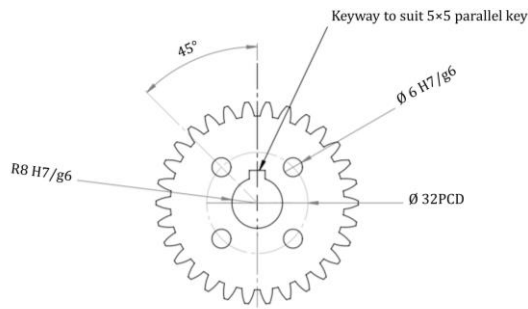


Fig. 1. Chemical structure of polyoxymethylene (a) homopolymer and (b) copolymer [8].

2. SAMPLE PREPARATION AND EXPERIMENTS

2.1 Materials

Commercially available injection moulding grades of POM-H (Delrin® 500T BK602, Wilmington, Delaware, USA) and POM-C (Hostaform C9021, Irving, Texas, USA) were purchased from DuPont and Celanese. The geometry and specifications of the tested gears can be found in Figure 2.



Parameters	Gear Category	Module (mm)	Tooth Number	Pressure Angle (°)	Face Width (mm)	Tooth Thickness (mm)	Contact Ratio	Profile Standard
Value	Spur Gear	2	30	20	15	3.14	1.67	DIN 3960

Fig. 2. Gear geometry and specifications.

2.2 Gear injection moulding

The gear specimens were manufactured through injection moulding (IM). An ENGEL 60T IM machine (ENGEL U.K. Ltd, UK) and a customised-designed two-plate mould (Figure 3(a) and 3(b)) were used for gear manufacturing. The mould contained a heating-cooling system to control the temperature of the cavity, and a set of ejection pins to automatically remove gears from the mould at the end of each IM cycle. To enable the molten polymer to flow uniformly, the gear part was center gated (Figure 3(c)). After removing the sprue, location holes and pinholes were introduced with a cutting machine tool. No further post-treatment was carried out before testing.

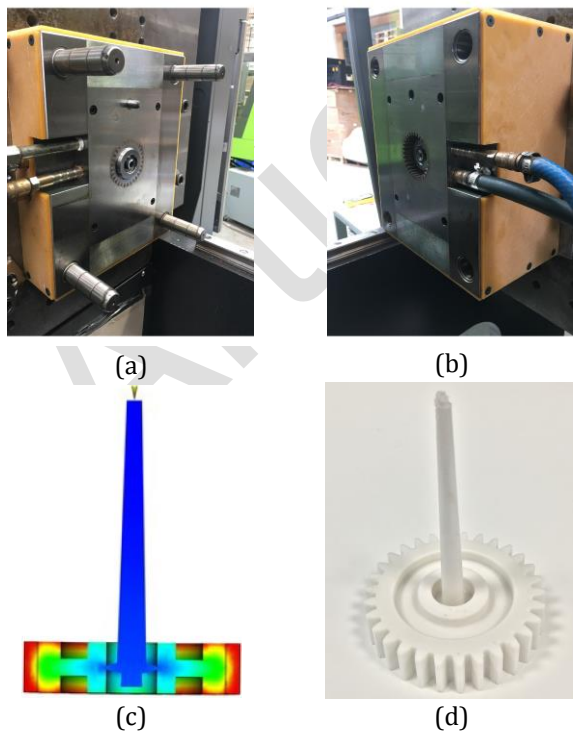


Fig. 3. (a) and (b) - Gear moulding tool; (c) - Gear cross section from Moldflow; (d) - Injection moulded gear sample with sprue intact.

As per the manufacturer's recommended material preparation, the material was predried for 4 hours at 100 °C. It was then IM as per recommended processing conditions, which can be found in Table 1. IM parameters were optimised in the middle of recommended ranges [27,28]. However, the holding time was determined by weighing the sample until the sample weight reached its maximum as per good moulding practice, allowing the system to reach a steady state. To apply changes to the IM process parameters on the samples, two complete material purging trials were carried out and the first 25 manufactured gear samples were discarded on each setting. A sample of manufactured gear with the sprue intact can be seen in Figure 3(d). The average shrinkage of the material was 1.7%, this was offset by designing a slightly larger mould cavity. After the plastic sprue bar was removed, and the location hole and pinholes cut, the polymer gears were ready for testing.

Table 1. IM procedure parameters.

Parameters	POM-C Gear	POM-H Gear
Melt Temperature (°C)	200	205
Mould Temperature (°C)	100	50
Injection Speed (mm/s)	45	45
Hold Pressure (MPa)	70	70
Holding time (s)	35	35
Cooling time (s)	75	75

2.3 Wear performance and material characterisation

To test the real-time wear performance of polymer gear pair, a test rig (Figure 4) was designed and manufactured at the University of Warwick, UK. It could measure the gear wear continuously with different settings such as load, rotation speed, and alignment style. A linear variable differential transformer (LVDT) was attached to monitor the gear wear. This recorded the data at a rate of 60 Hz through a data-logging system linked to a computer. To decrease the influence of the sensor on gear meshing, the gear wear was measured indirectly through the rotation degree of the pivot block. Full design and measuring details of the test rig and relevant tests were in references [17,29,30].

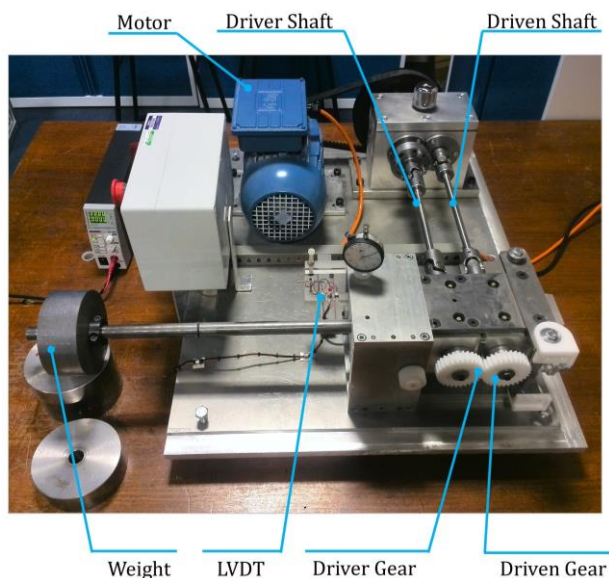


Fig. 4. Polymer gear test rig.

The morphology of the worn surface of the gear teeth was observed using a scanning electron microscope (SEM) ZEISS Sigma (Carl Zeiss AG, Germany). To reduce surface charging during SEM imaging, a thin film of gold was deposited onto the samples using an Au/Pd evaporation system. The images were captured using either secondary electron (SE) mode or the in-lens detector. The gun voltage was set to 5.00 kV and the high current set to OFF to avoid burning the plastic sample.

In order to compare the elastic properties of POM-H and POM-C, dynamic mechanical analyses of the polymers were performed using a dynamic mechanical analyser (DMA 242 E Artemis, Netzsch, Inc, Selb, Germany). Flat specimens (16 mm in gauge length, 8 mm in width, and 2 mm in thickness) were heated from 20 to 180 °C. The tests were performed in single cantilever mode at a frequency of 16.67 Hz, maximum dynamic force of 10 N, and static force of 2 N, displacement of 0.05 mm, and a heating rate of 20 °C/min.

While DMA analysis can reveal the mechanical properties related to the temperature, a FLIR C2 (Flir Systems, Inc, USA) thermal camera was used to generally monitor the gear teeth surface temperature in the linear phase during gear meshing. While the temperature was monitored, only one point was recorded which was the point when the temperature stabilised during the linear phase.

3. RESULTS AND DISCUSSION

3.1 Wear performance

The wear and fatigue performance of the POM gears were evaluated using the rig shown in Figure 4. In this study, all the tests were set at a rotation speed of 1000 rpm (16.67 Hz) and no misalignments were set for the gear meshing. In each test, a pair of the same type of gears meshed against each other. Those tested gears operated at a centre distance of 60 mm. The load was 9 N m and the test was kept running until the teeth broke. Further analysis revealed that both types of acetal gears had a relatively high endurance, which was reflected by a very low wear rate. As the driver gears usually broke first, they were regarded as the focus of polymer gear failure. Thus, all the research was focused on the driver gears in this study.

The smoothed results of the two types of acetal gears can be seen in Figure 5, this was calculated from the mean values of 6 experiments, with the error shown as the shaded regions until gears broke. The end of gear service life was recorded when the gear teeth broke or when a tooth jumped out from the mesh point. This end of service life is illustrated as the steep rise in Figure 5. The POM-H gears had an average of 35% longer service life than POM-C gears. For the POM-H gears, the wear could be divided into three phases: running-in, linear wear, and finally high wear. The gear pair failed to work as the gear teeth became softer and jumped out from meshing under high temperatures. This matched the conclusion in other studies [31]. POM-C gears show a similar pattern to POM-H gears, with running-in and linear wear phases, however, the POM-C gears had a relatively higher wear rate in both phases. When it came to the final phase, the curve of the POM-C gears steepens near the teeth breakpoint. The c part of the curve had a very short range on the horizontal axis. When comparing it to the curve of the POM-H gears, the curve shape of the POM-C gears showed a much more rapid wear increase before the breaking point. At this point a crack formed, making the LVDT (Figure 4) display a sudden increase in readings. This final part of the curve can be called a crack phase rather than a final massive wear phase, as the gear failed.

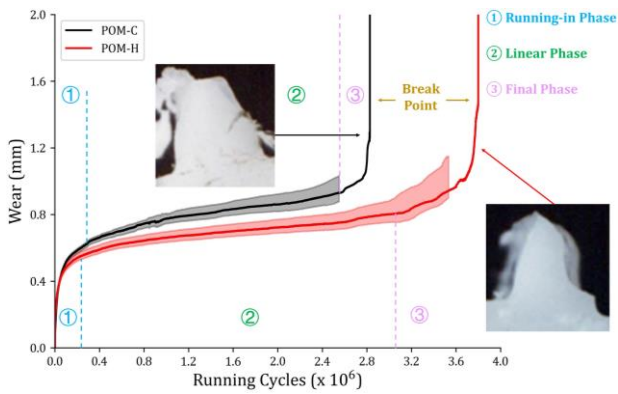


Fig. 5. Wear degradation of POM gears.

3.2 Dynamic mechanical property

To compare the two polymers and support the wear results, single cantilever mode dynamic mechanical analysis (DMA) was used to characterise the mechanical properties of POM-H and POM-C. The average storage modulus (E'), is the energy stored by the material through elastic deformation [32]. This is higher in POM-H than POM-C from 20 °C to 160 °C, which covers working conditions from room temperature to POM melting temperature (Figure 6(a)). A thermal camera was used to check the gear surface temperature with an emissivity value of 0.92 [33]. Similar Like from other research [34], in the linear wear phase, the surface temperatures of both POM-H and POM-C gears at the contact area was stabilised at about 110 °C (Figure 6(b)), meaning that the gear pair operated under this temperature for most of the time. The characterisation of elastic properties with DMA helps to support the results of wear from Figure 7, because previous studies had

established a correlation between elastic properties and wear properties of engineering polymers. For instance, Lancaster [35] showed a set of relationships between wear resistance on a polymer-metal system of polymers and other mechanical properties. Their study, covering a wide range of polymers, including commercial POM-H and polyamides, suggested that polymers with higher elastic modulus tend to perform better in wear applications. Similar correlations between elastic properties and wear have been shown in other previous studies [35-37]. Also, POM-H maintains a higher loss modulus than POM-C throughout the whole testing temperature range (Figure 6(c)). This means POM-H has generally better impact resistance. As the loss modulus represents the viscous part of the sample [36,37], the increase of the loss modulus value of POM-H from around 105 °C leads to a more viscous performance and resultant higher crack resistance, which means at the stabilised working temperature (around 110 °C), there is a relatively big difference on viscous performance between POM-H and POM-C gears. This results in the final breaking format of gear tooth softening rather than gear tooth break. At the same time, the loss modulus value of POM-C reduces with temperature, which leads to crack formation and direct gear tooth breakage due to cracks.

The significant difference in elastic properties and wear properties of the homo- and copolymer material grades here is likely due to the differences in their molecular structures, highlighted in Figure 1.

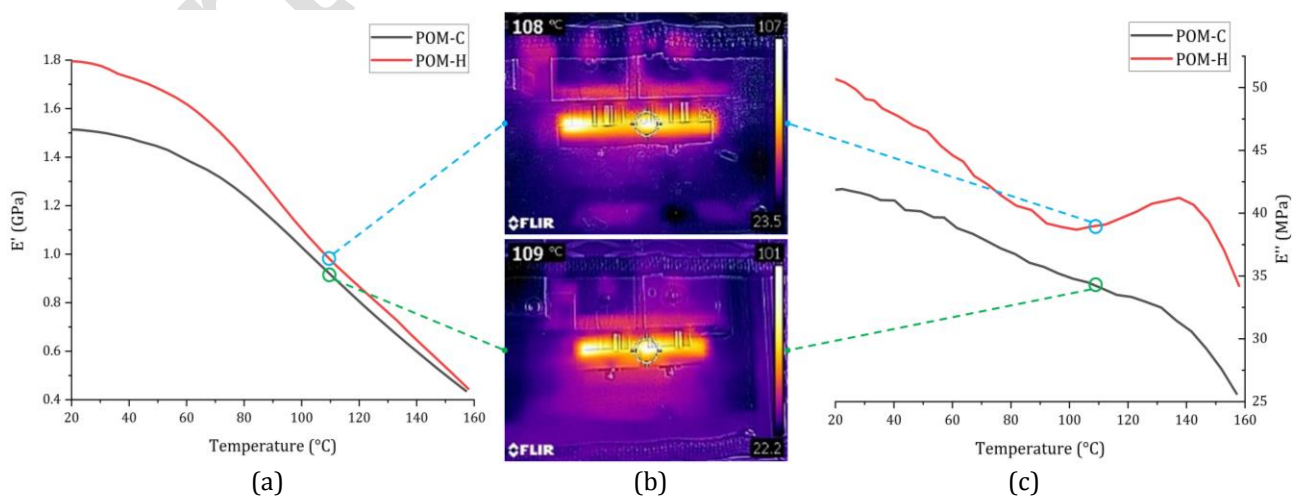


Fig. 6. DMA results of POM-C and POM-H: (a) Average storage modulus; (b) Thermal image; (c) Loss modulus.

Given that the structure of POM-H presents a more uniform backbone, it is likely to have more stable, organised crystalline domains as the polymer solidifies after IM [38,39]. Also, the uniform structure of POM-H allows for higher entanglement between crystalline domains. Correspondingly, the acetic acid molecules along the backbone of the POM-C are detrimental to the formation of crystalline structures. This difference in structure ultimately leads to higher elastic and wear properties in POM-H compared to POM-C [38,39].

3.3 Failure mechanism

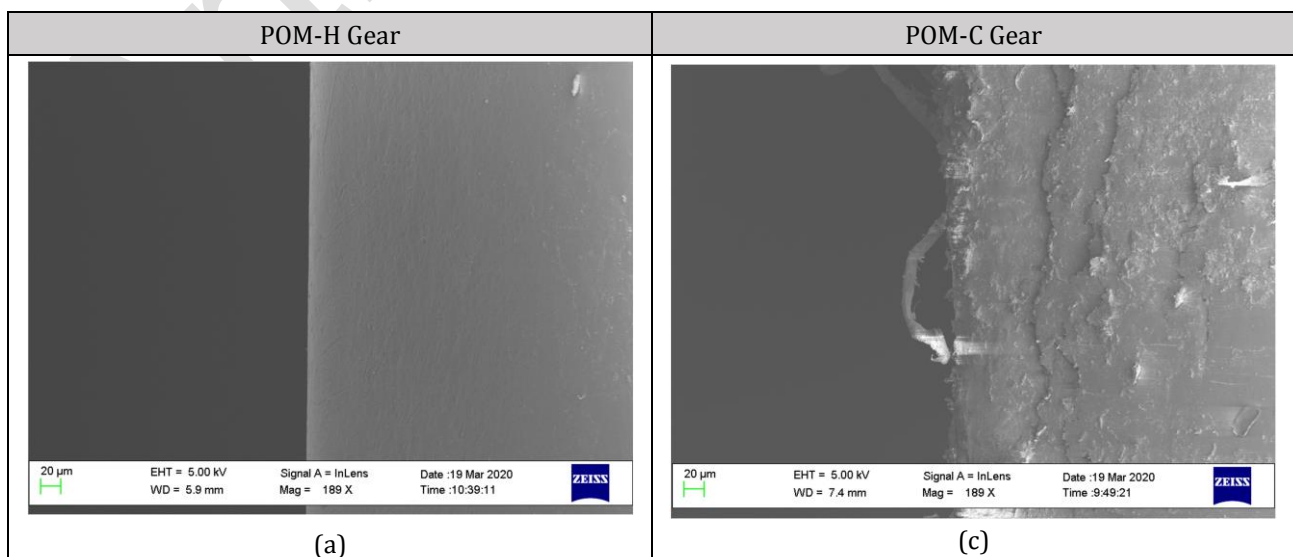
To investigate the failure mechanisms of the gears, a high-resolution scanning electron microscope (SEM) was used. Differences existed between each area (tip, root, and pitch line) of the gear teeth contact surface of the two acetals.

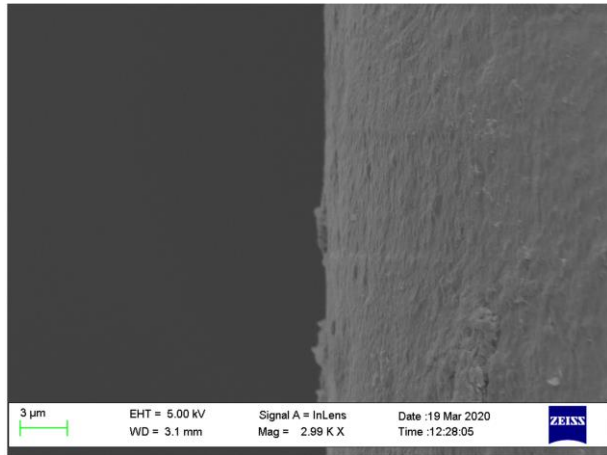
At the tip area, the POM-H gear teeth look smooth at relatively low magnification of 189 times (Figure 7-1(a)). Little wear can be seen on the surface. Alternatively, when looking at the POM-C gear teeth under the same magnification, the tip area is deformed showing large flaw regions (Figure 7-1(c)). The POM-H gear teeth only display clearly as a rough and melted worn surface when viewing at a higher magnification such as 2.99k times (Figure 7-1(b)).

The root area also shows different patterns. For the POM-H gear teeth, the surface is relatively smooth with the melted material flow (Figure 7-1(d)). For the POM-C gear teeth, the grooves are visible with lots of debris (Figure 7-1(e)).

The most significant difference happens around the pitch line, marked as the blue double-sided arrows in Figure 7-2(f)–(h). For the POM-H gear teeth, the pitch line area is clean without much wear (Figure 7-2(f)). As in the theory of gear meshing, at the pitch lines of the meshing gears, the gears have no sliding but only rolling against each other [40]. The two sides surrounding the pitch line have much more melted debris than other areas, as shown in Figure 7-2(g). This is because, on each side of a pitch line, the gears slide against each other, with friction leaving more melted debris. The motion of the gears during meshing makes gears have an S-shaped profile, as stated by other authors [40]. In the pitch line image of the POM-C gear teeth, there exists abrasive wear all around the pitch line as pitting [41] and a crack occurs there (Figure 7-2(h)). Finally, the gear breaks because a crack develops on any one of the gear teeth.

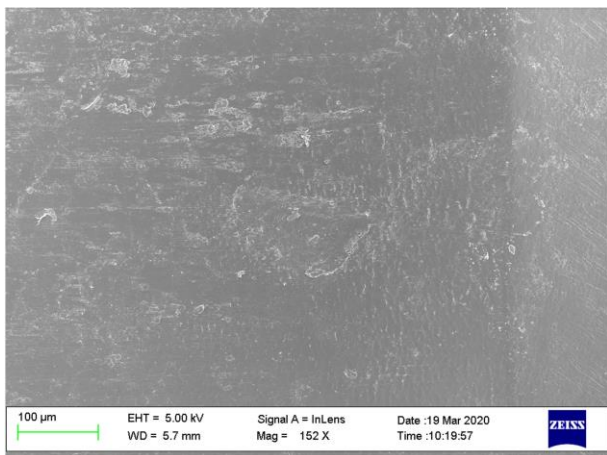
In general, there are no cracks visible beneath the surface of POM-H gear teeth, and the pasted debris demonstrates the polymer deformation (Figure 7-2(i)). As also stated by other researchers [42,43], the polymer becomes softer first, then is stretched and finally detached. The main cause of wear is adhesion (Figure 7-2(k)). While for the POM-C gear, the smear is the most noticeable on the teeth (Figure 7-2(j)), indicating the wear mechanism when the polymer wears away from the gear. The smear occurs when the gears start running and then are stretched and become more noticeable during the gear meshing. Finally, it reaches a point that the polymer breaks and peels off the leading edge of the smear (Figure 7-2(l)). Similar results were found by others [23,44].



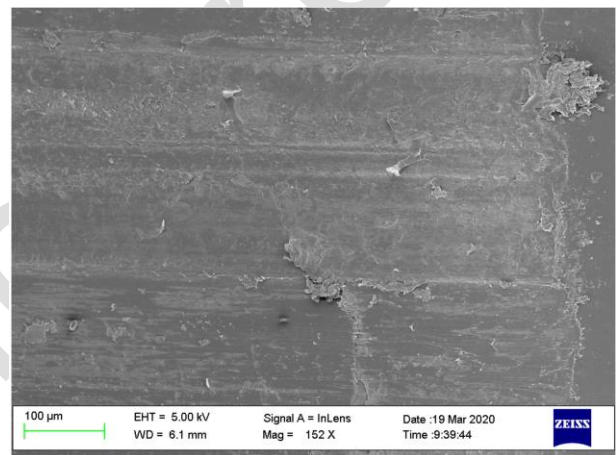


(b)

Tip



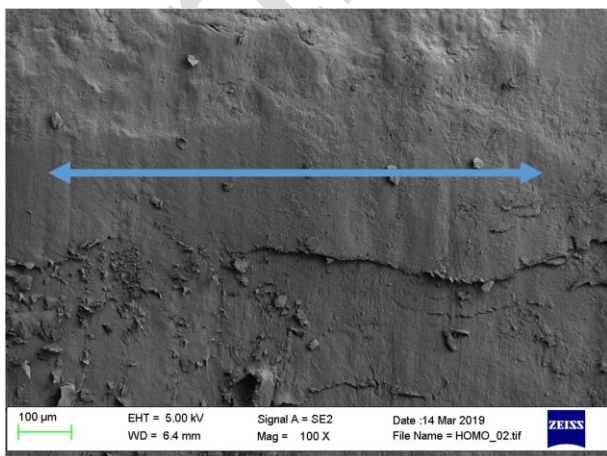
(d)



(e)

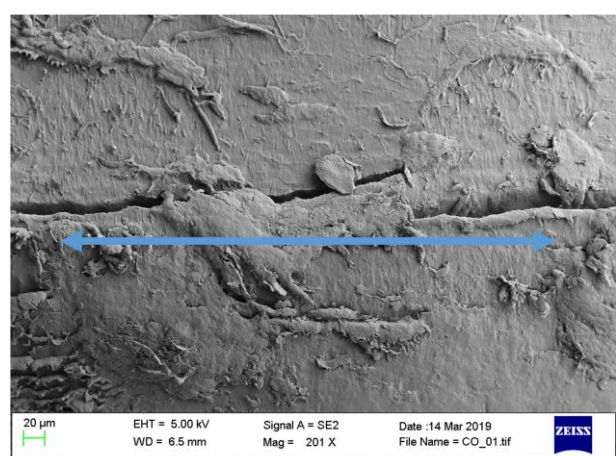
Root

POM-H Gear



(f)

POM-C Gear



(h)

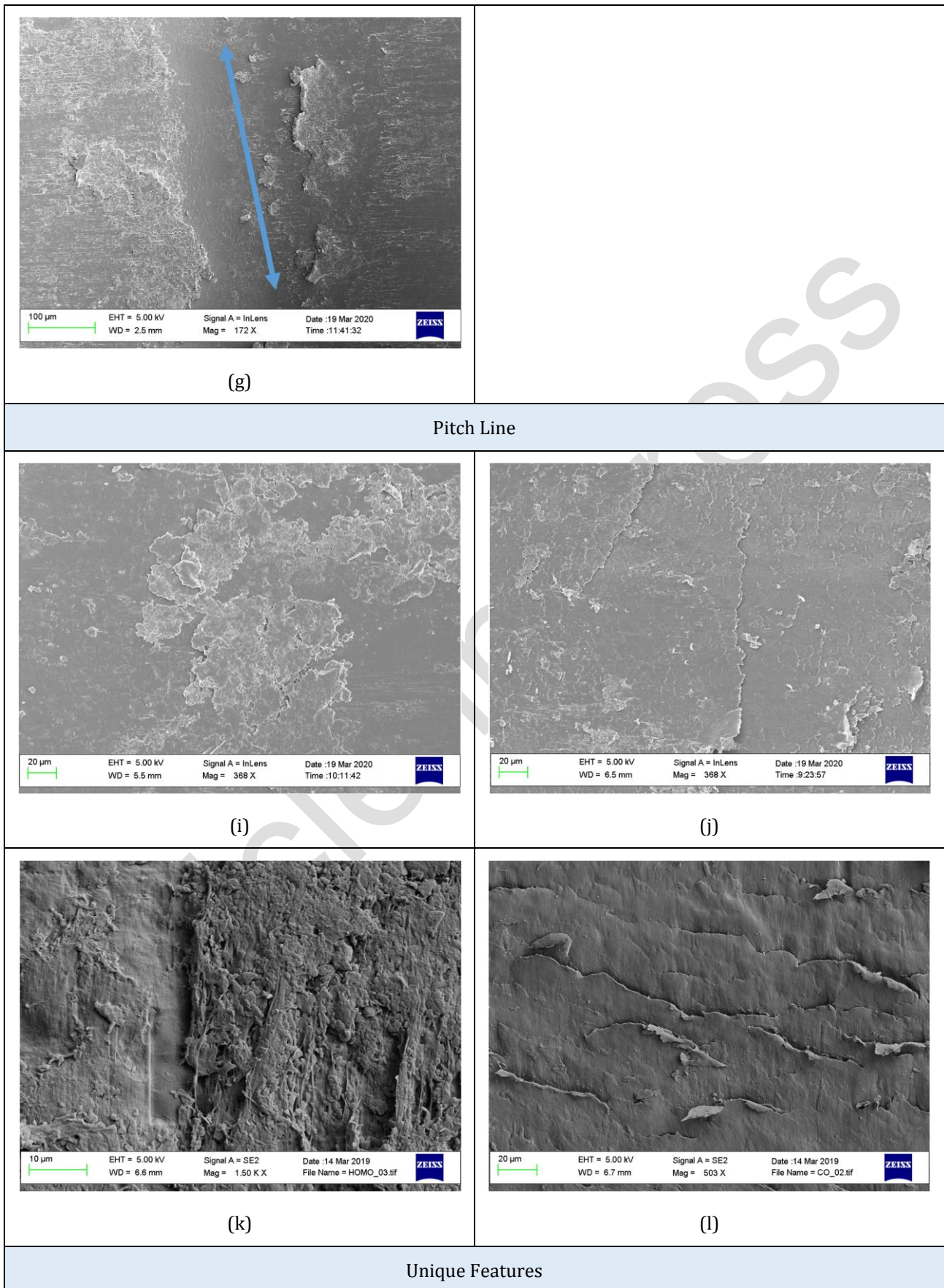


Fig. 7. Comparison of SEM.

3.4 Performance analysis

The wear rate from the acetal gear test results can be expressed as the ratio of change in geometry of the worn gear (in terms of length) to the number of revolutions (cycles) the gear undertook in the test.

Wear rates under this critical value of torque was studied by Friedrich [45], providing an equation of wear rate k_s as $k_s = \frac{V_w}{Fs}$, with V_w as the wear volume, F as the normal force and s as the sliding distance. By adapting the spur gear tooth profile, this equation can be further expressed as

$$k_s = \frac{Qbd}{2TN} \quad (1)$$

with Q as the wear depth, b as the gear face width, d as the gear pitch circle diameter, T as the torque, and N as the number of cycles the gear takes.

In previous research [31], it has been discovered that the critical value of k_s , above which the wear rate grows much faster, is related to the maximum gear contact surface temperature θ_{max} , consisting of ambient temperature θ_a , body temperature θ_b and flash temperature θ_f . This can be further expressed as

$$\theta_{max} = \theta_a + \theta_b + \theta_f = \theta_a + k_1T + k_2T^{3/4} \quad (2)$$

where

$$k_1 = \frac{c_r\mu}{bc\rho Z(r_a^2 - r^2)} \text{ and } k_2 = \frac{1.11\mu(V_1^{1/2} - V_2^{1/2})}{rb(2k\rho ca)^{1/2}} \quad (3)$$

with T as the transmitted torque, ρ as the specific gravity, k as the thermal conductivity, c as the specific heat, a as the half contact width, r_a as the outside radius, r as the reference radius, b as the tooth face width, V_1 and V_2 as the sliding velocities of the two contact gears [46].

c_r is a constant relating to the contact ratio of the two gears, as

$$c_r = \frac{1 + \text{contact ratio}}{4} \quad (4)$$

For Hertzian line contact [34], k_2 can be further expressed as a more basic form

$$k_2 = \frac{1.11\mu(V_1^{1/2} - V_2^{1/2})}{2r^{3/4}b^{3/4}(k\rho c)^{1/2}} \left(\frac{\pi E}{R}\right)^{1/4} \quad (5)$$

where E is the effective elastic modulus and R is the relative radius.

With gear information k_1 and k_2 given after tests along with temperatures recorded, the critical value of torque T , beyond which the wear rate shall increase rapidly, could be calculated. Then the corresponding wear rate k_s at the critical torque could also be observed using Equation (1). Then the prediction of gear performance below the critical torque can be interpreted using k_s .

For the current experiment that the maximum surface temperatures reaching the melting point were 178 °C for POM-H and 166 °C for POM-C respectively, similar to the previous research [31], using Equations (2-5), the corresponding predicted loading capacities were 11.5 N m for POM-H and 10.3 N m for POM-C gear respectively. As the incremental step loading method was approved to reveal the wear rate of various loads efficiently [31,34], it was applied for the two types of acetal gears at 1000 rpm from 6 N m with a step load increment of 1 N m until the gear broke. The results were calculated by the mean values of 6 repetitive experiments. The errors were plotted as shaded regions in Figure 8(a), and error bars in Figure 8(b). The wear curves were shown in Figure 8(a) and it could be seen that the POM-C gear broke at 11 N m and the POM-H gear failed at 12 N m. The wear rates were calculated using Equation (1) and the wear rate against load plot is shown in Figure 8(b). Similar to the pattern of the POM-H gear from 6 to 9 N m, the POM-C gear kept a relatively stable low wear rate at the load of 6 to 8 N m, and then wore more quickly until breaking. These tests prove that the above equations could offer good prediction on the gear transition torque for both POM-C and POM-H gears. It is found that the wear rate of POM-H gears tested at the load of 7 N m was lower than the value at 6 N m. We believe this is simply down to the inherent experimental error at this stage of the process because of start up, high wear rates and rapid temperature rises which all introduce variability.

For POM-H, it is possible to use Equation (1) to work out the wear rate k_s , and then the gear wear. However, for POM-C gears, before the factor of temperature, fatigue owns a more noticeable influence on the breakage of the gear teeth. In this situation, it might not be very useful to use k_s for gear wear prediction.

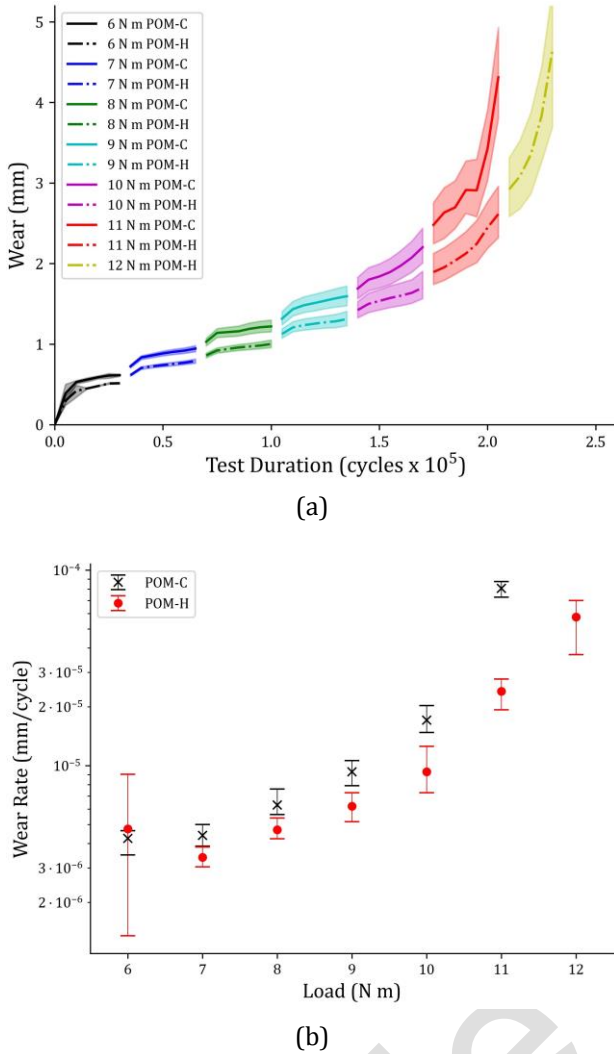


Fig. 8. Step loading tests results for acetal gears (a) Wear curve. Step changed per 3×10^4 cycles. (b) Wear rate.

From previous studies on fatigue propagation [10], the stress intensity factor played a key role in monitoring fatigue crack growth. As a complex and material-shape-dependent property, the stress intensity factor contains information on both the load applied to the cracking point and geometry influence. The growth rate of fatigue crack of gear teeth could be measured using the Paris–Erdogan equation

$$\frac{da}{dN} = C(\Delta K)^m \quad (6)$$

with a as the crack length, N as the load cycle, K as the stress intensity factor. C and m are two experimentally obtained material-dependent coefficients. da/dN is the crack growth and ΔK is the stress intensity factor range for one cycle given as $\Delta K = K_{\max} - K_{\min}$.

In past, there had been a variety of studies on gear fatigue related quantities, especially the stress intensity factor. A typical solution of stress intensity factor for gear fatigue [47] was shown as

$$K = \frac{6FLY}{bS^2} \sqrt{\pi a}, \quad (7)$$

where

$$Y = \left(\cos \phi - \frac{C}{L} \sin \phi \right) Y_m(\alpha) - \frac{S}{6L} \sin \phi Y_t(\alpha). \quad (8)$$

F is the load applied; L , S , b and C are length quantities including gear specimen dimensions; a is the crack length; $\phi = \arctan F_y/F_x$; $Y_m(\alpha)$ and $Y_t(\alpha)$ are shape factors where $\alpha = a/S$.

By extending K and F to a range for a cycle, it could be seen that $\Delta K = \frac{6\Delta FLY}{bS^2} \sqrt{\pi a}$, where ΔK is the stress intensity factor range and ΔF is the load range for a cycle.

With a pre-set F , it is possible to produce a logarithmic graph showing fatigue crack growth as a function of da/dN against ΔK [10]. With a sufficient number of tests, a more accurate pattern of fatigue could be obtained, helping to determine when and where the crack would occur, which could be further studied. This could be used in future predictions in the performance of POM-C gears.

4. CONCLUSIONS

Two types of commercial grade POM (POM-H and POM-C) gears were manufactured to compare the wear performance. The tests found a significant difference in the wear performance. POM-H gears had three wear phases, namely running in, linear wear, and final massive wear. The final massive wear was due to the temperature reaching the material melting point, and the teeth were therefore too soft to keep normal meshing, namely thermal failure. And similar results have been concluded in other references [34,48]. In contrast, POM-C gears have better performance in thermal aging, which lead to a finding that the POM-C gears shared the same first two phases with POM-H ones, but also had the third one as the crack phase rather than exhibiting thermal failure. This finding also matched with the conclusions in [9,10].

The final breakpoint was due to the fatigue cracks around the pitch lines of the gear teeth. DMA tests supported the above findings in the aspect of energy dissipation between the two materials. SEM images illustrated differences in the material failure mechanism, of which the most obvious one was that the POM-H illustrated material adhesion while the POM-C revealed mainly smearing. More tests were carried out with various loads from 6 N m until reaching their corresponding transition torque values and similar results were found in the above aspects. In general, the POM-H gear wear performance could be estimated with Equation (1) while the POM-C gear wear performance should refer to Equation (6). It can be concluded that the transition torque prediction for both POM-H and POM-C gears was still applicable using Mao's method [34]. In this study case, the POM-H gears performed about 35% better in service life on average than the POM-C gears. Considering the mechanics studied above, the polymer formulation based on the homopolymer had a better performance than the copolymer formulation for injection moulded polymer gears. This is a physical phenomenon due to differences in the molecular structure of POM-H and POM-C which leads to very different mechanical performances and failure modes. Even though the POM-C performs better in thermal aging when it comes to the gear application tested here, the crack and wear resistance play more important roles. Consideration on a case by case application basis may be applicable here.

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