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High-speed tribology of PFPEs with different functional groups and molecular weights coated on DLC

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Abstract

In this article, we study the friction and wear durability of perfluoropolyether (PFPE) with different functional groups and molecular weights (MW) for a range of disk rotational speeds (500–7200 rpm or 1.2–17.33 m/s). A 4 mm diameter silicon nitride ball under a normal load of 4 g was employed as slider against PFPE lubricated diamond like carbon (DLC) film on magnetic hard disk. The coefficient of friction increases with increasing speeds, to certain extent, but it decreases for the higher speeds. At very high speeds, the fluctuations in the coefficient of friction of low MW PFPEs were larger than those of high MW PFPEs. The optical microscope image of the ball after sliding showed that evaporation might have occurred more easily in low MW than in high MW when sliding speed was increased due to the frictional heat generated at the interface. The wear lives of Z-lube (carboxyl group at both ends) and Z-dol are significantly higher than AS1 (alkoxy silano group at both ends) at low speed (1.2 m/s). In comparison to low MW PFPEs, high MW PFPEs show better wear durability at higher rotational speeds.

KEY WORDS: Head Disk Interface (HDI), high-speed tribology, PFPE

1 Introduction

A hard disk is a magnetic storage device that can store digital data by applying a magnetic field with a flying read/write head (electro-magnetic sensor). In order to prevent corrosion of the magnetic layer, a diamond like carbon (DLC) layer is coated onto the magnetic layer of the disk. An ultra thin layer of perfluoropolyether (PFPE) is used as a typical lubricant in magnetic disk drive industry to reduce the friction and wear of the head disk interface in order to minimize any possibility of disk failure [1]. The areal storage density of a hard disk has been increasing for the past few decades at a rate of 60% per year and has now reached more than 100 Gb/in^2 [2]. In order to increase the areal density, the flying height (FH) between the slider head and the disk must be as low as possible. A FH of less than 10 nm (approaching 5 nm or even less) is required to get recording density >100 Gb/in$_2$ [3]. A PFPE film of 2 nm thickness or even less is used for maintaining the distance of around 10 nm or less between the magnetic layer on the disk and the read–write sensor on the slider head [4–6]. Faster data access time is also required with increasing areal data density. By increasing the disk rotational speed, we can get faster data access. Therefore, the speed of the hard disk has increased in recent years from 3600 rpm to 10,000 rpm and even higher speeds are anticipated in the near future [7]. Reducing the head disk interface (HDI) spacing will promote the probabilities of intermittent or sustained collisions between the head and the disk. Increasing the rotational speeds will generate high flash temperature at the collision points leading to lubricant degradation. In order to improve the tribological performance to withstand such critical conditions, new lubricants are required to be developed.

Tambe and Bhushan studied the effects of load/unload speed and disk speed (1800–10,000 rpm) on load/unload performance and reported that the probability of head-disk contacts would increase before stable air bearing film formation at higher rotational speeds [8]. Zhao et al. [7] investigated the effect of disk acceleration and velocity on tribology of HDI and found that stiction decreased with increasing acceleration. Sinha et al. studied tribology of PFPE lubricant layer with varying speed and found catastrophic failure of the lube and the carbon overcoat due to disk vibration at very high speed [9]. Lee et al. investigated the generation of contaminant particles as a function of rotational speed and found that the particle generation rate increased with increasing speeds [10]. However, to the best of our knowledge, there has been no systematic study on the effect of sliding speed on friction and wear performance of PFPE lubricants with different functional groups and molecular weights (MW) at HDI. In this article, we study the effect of rotational speeds on the friction and wear characteristics of PFPE lubricants listed in Table 1 at disk rotational speeds from 500 rpm to 7200 rpm. In addition to data storage, this study will also be beneficial to other areas where PFPE has been proposed as a lubricant. One such area is micro-electro-mechanical systems (MEMS) [11, 12].

2 Experiment

2.1 *Dip coating of PFPE lubricants*

About 2.5-inch DLC coated disks (~0.35 nm roughness, measured with an atomic force microscope (AFM)) were used in our experiments. A solution of PFPE lubricant with H-Galden ZV 60 $HCF_2O–(CF_2O)_p–(CF_2CF_2O)_q–CF_2H$ (purchased from Ausimont INC) was prepared. The concentration of the lubricant in the solution was 0.2 wt%. Dip coating technique was used for lubricant application on to the magnetic hard disks. The duration of dip coating time was 30 s. The disk was withdrawn at a constant speed of 2.4 mm/s. After dip-coating, the disks were kept in a clean room for 72 h before any test was carried out.

2.2 *Contact angle measurement*

VCA Optima Contact Angle System (AST product, Inc., USA) was used for the measurement of contact angles with distilled water droplets. A water droplet of 0.5 µL was used for contact angle measurements. A total of five independent measurements were conducted randomly on each side of the disk and an average value was taken for every sample. The deviation of contact angle was within ±3°.

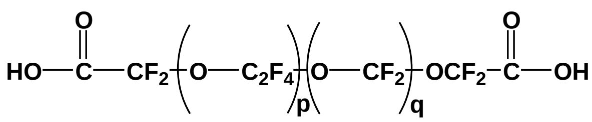
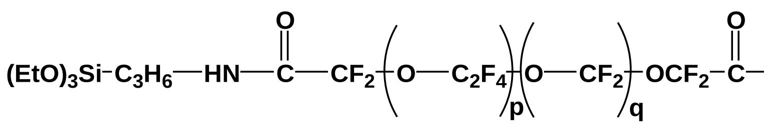
2.3 *Lubricant thickness measurement*

The thickness of the lubricants was measured by an Optical Surface Analyzer (Candela's 5100 OSA) (manufactured by Candela Instruments, USA). The OSA method relies on the detection of the reflected rays from the disk; the % reflectivity of the lubed disk depends on the original reflectivity of the disk and the lube thickness. Thus, there is a decrease in the disk reflectivity after lubrication of the disk. In this article, we used a known thickness of a disk coated with Z-dol 4000 as a reference disk, measured the % reflectivity change of the reference disk due to the lubricant and converted the number into a ratio of % reflectivity to the known lube thickness. The thickness of the reference disk was earlier measured using a lube thickness calibration method mentioned in ref. [9]. We assumed that the % reflectivity/thickness ratios of all other lubricants used in the present study were nearly the same as Z-dol 4000 since they all had the same backbone chemical structure. The % reflectivity of unknown thickness of other lubricants was examined by OSA and converted to known thickness by using the % reflectivity/thickness ratio of Z-dol 4000.

2.4 *Tribological tests*

Frictional coefficient and wear durability of different lubricants with various end groups and MWs of PFPEs on magnetic hard disk were tested on a custom-built high speed ball-on-disk tribometer. Si3N4, silicon nitride balls (~5 nm roughness) with 4 mm diameter were used as the slider. The sliding balls were held in a ball holder that was screwed at the tip of a force-sensing double (orthogonal) cantilever. Normal load was measured by recording the displacement of the cantilever with a laser beam in vertical direction. Vertical displacements of the cantilever (holding the slider ball) were recorded continuously with a laser displacement transducer and the change in the voltage output from the transducer was subsequently converted to normal load. This cantilever also had 4-bridge strain gauge attached that could provide friction force due to lateral (horizontal) movement of the cantilever during the test. A fixed normal load of 4 g was used for friction and wear tests. The friction measurements were carried out at the disk radii of approximately 20 mm and 23 mm. Rotational speeds used were 500, 1500, 2500, 4000, 5000, 6000 and 7200 rpm. The values of coefficient of friction and wear durability were taken as an average of three tests. All measurements were made at 23 ± 2 °C and relative humidity of 55±5%in a clean booth (Class 100) environment.

Table 1. *Chemical structure of PFPE lubricants.*

Lubricant	Molecular weights	Chemical structure
Z-dol	4000	$OH\text{-}CH_2CF_2O\text{-}(O\text{-}C_2F_4)_p\text{-}(O\text{-}CF_2)_q\text{-}CF_2\text{-}CH_2\text{-}OH$
Z-lube	3000, 4000 & 5000	
AS1	3000, 4000 & 5000	

where p:q = 2/3. Z-lube has carboxyl group at both ends and AS1 has a polar alkoxy silano group at both ends.

For each test, we used a new ball and the ball surface was imaged using an optical microscope after every test and after cleaning with acetone to check whether or not the ball surface had worn. In this article, we focus on two comparisons. The first one is the comparison of the coefficient of friction and wear durability of all lubricants listed in Table 1 at 1.2 m/s (500 rpm), and the latter is the comparison of the coefficient of friction and wear durability of the more durable lubricants, Z-dol 4000 and Z-lube (3000, 4000 & 5000 MW), at various speeds up to 100,000 cycles of sliding at a constant applied load. To understand the mechanism of lubrication and the nature of lubricant transfer to the slider, the surface morphologies of the worn surfaces were observed using JEOL JSM-5600 LV scanning electron microscopy (SEM).

3 Results and discussions

3.1 Contact angle

The water contact angles, surface energies and thicknesses of AS1 (silane coupling agent at both ends of PFPE) (3000, 4000 & 5000 MW), Z-dol 4000 and Z-lube (3000, 4000 & 5000 MW) are shown in Table 2. The thicknesses of all lubricants are in the range of 2–4.7 nm. Unlubricated DLC has shown a water contact angle of 66°, which is lower than for any other lubricated sample. The Young equation for the contact angle h of a liquid L on a plane surface S is given as [13]

$$r_L \cos\theta = r_S - r_{SL}$$

where r_S = surface free energy of solid S; r_L = surface free energy of liquid L; r_{SL} = interfacial free energy. The detailed explanation of the calculation of surface energy from contact angle values, as used in this work, is mentioned in reference [14]. Figure 1 shows the relation between contact angle and surface energies. The water contact angles for AS1 were higher than both for Z-lubes and Z-dol because AS1 could adhere to the surface more strongly because of the presence of terminal alkoxy silano group as a polar group at both terminals of PFPE [15]. Higher contact angle means lower surface energy. The MW does not show any effect on water contact angles in the case of AS1 while the contact angle increases as the MW increases in the case of Z-lube. For the case of AS1 the polar group is adsorbed on to the surface and the outermost part is covered with PEPE group, therefore the contact angles are independent of the MW. But for Z-lube, lubricant molecules seem to be isotropic compared with AS1, and the contact angle is determined by fluorine content, which becomes higher with increase in MW. Because of the presence of a functional group at both ends (carboxyl group in Z-lube and hydroxyl group in Z-dol), MW has clear effect on contact angle of Z-lube.

Table 2. *Water contact angles and surface energies of different lubricants.*

Lubricant	Contact angle of water (°)	Surface energy(mJ/m²)	Thickness (nm)
AS1 3000	97	16.6	2.6
AS1 4000	94	18.7	2.8
AS1 5000	98	17.7	3.1
Z-dol 4000	83	18.3	2[9]
Z-lube 3000	83	19.5	4.7
Z-lube 4000	84	19.3	2.9
Z-lube 5000	93	16.6	3.4

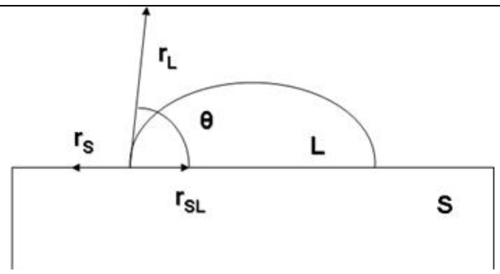


Fig. 1 *Relation between contact angle and surface energies.*

3.2 *Initial coefficient and wear life at 1.2 m/s (500 rpm) rotational speed*

Figure 2(a) and (b) show the initial coefficient of friction and wear durability of every lubricant sample mentioned in Table 1. We calculated an average value of first 400 cycles of 3 experiments and reported as an initial coefficient of friction. When the sample was worn out, there were large fluctuations in the coefficient of friction immediately followed by a clear wear track on the disk and the corresponding numbers of cycles were defined as wear durability of that sample.

The nominal contact area and contact pressure were 2.94×10^{-10} m^2 and 199 MPa respectively, as calculated using Hertzian contact model. The mechanical properties of DLC (Young's modulus: 120 GPa and Poisson's ratio: 0.2) and Si_3N_4 ball (Young's modulus: 310 GPa and Poisson's ratio: 0.22), as provided by the supplier were used for the calculation of the contact area and the contact pressure.

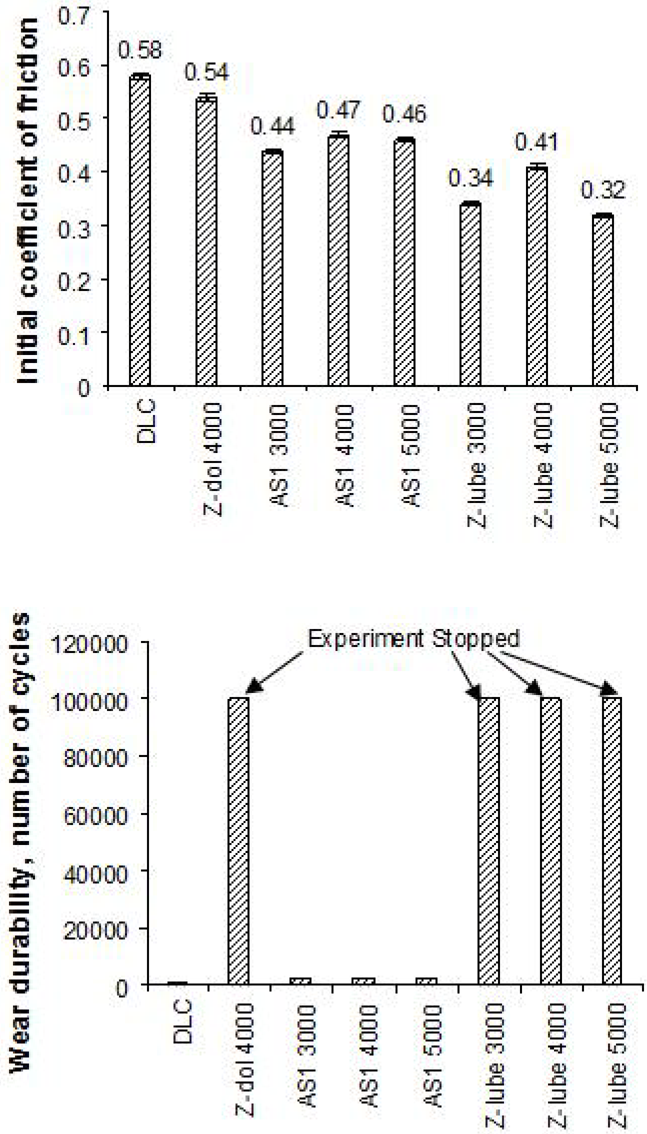


Fig. 2 *(a).Initial coefficient of friction (an average of first 400 cycles) at a normal load of 4 g and at a rotational speed of 1.2 m/s (500 rpm) for unlubricated disk and disks with different lubricants. (b). Wear durability of unlubricated disk and disks with different lubricants at a normal load of 4 g and at a rotational speed of 1.2 m/s (500 rpm).*

The coefficient of friction of DLC was the highest and its wear durability was the shortest (~400 cycles) in comparison with other lubricated samples. After lubrication of the DLC surface, the coefficient of friction dropped slightly but the wear durability increased to a great extent. For AS1, wear durability was ~2500 cycles and for Z-dol and Z-lubes, no wear occurred until 100,000 cycles when the tests were stopped. The MW did not show any effect on the wear durability of both Z-lube and AS1. The key property of a good lubricant is its replenishment capability, that is, to flow back and recover the area where the lubricant is depleted due to repeated slider-disk contact [16]. The replenishment strongly depends on the surface energy and the mobility of the lubricant. Since, the contact angles of AS1 lubricants have been higher than those of Z-lube and Z-dol, their (AS1 lubricants) surface energies and mobility are lower than those of Z-lube and Z-dol. It is not a problem when the rotational speed is low and the lubricant replenishes faster than it is depleted, because the durability of the interface is high due to sufficient lubricant reflow [16]. In this condition, the MW does not affect the coefficient of friction or the wear durability of lubricants.

After every experiment, the ball surface is observed under an optical microscope and it has been found that the ball surfaces slid against DLC and AS1 (after test) were identical. Figure 3 shows SEM image of bare disk (DLC) and optical image of Si_3N_4 ball after 400 cycles of sliding at 1.2 m/s (500 rpm). We can see a clear wear track and debris along the wear track on the disk and there is presence of small particles (DLC) on the ball surface. But for Z-dol and Z-lubes (for all MW), we did not observe any wear track on the disk after sliding for 100,000 cycles whereas the ball surface has shown the presence of some liquid, which must be the lubricant transferred from the disk surface to the ball. Tribological phenomena are greatly influenced by the frictional heating at the interface. Therefore, in order to check the thermal stability of the lubricants, we conducted differential scanning calorimetry (DSC) on a commercial machine (TA instruments Differential Scanning Calorimetry DSC 2910). This test showed that Z-lube and Z-dol were thermally stable up to a temperature of 500 °C whereas AS1 showed clear sign of thermal degradation beyond 360 °C.

The possible reason for not observing any wear for Z-dol and Z-lubes until 100,000 cycles at 1.2 m/s (500 rpm) could be that the rotational speed was less than the critical replenishment rates and the lubricants were thermally stable. Also, the modified PFPE (Z-lube) anchors the rubbing surface strongly, which leads to better lubricity [17].

3.3 Friction and wear at high speeds

Since we speculated that the critical replenishment rate of AS1 is low and the thermal stability is also poor, we did not conduct further tribological test on AS1 at higher rotational speeds. Figure 4 shows the coefficient of friction of Z-lube and Z-dol lubricants at various speeds. We conducted experiments up to 100,000 cycles of rotation and no wear had occurred except for Z-lube (3000 & 4000 MW) at 17.33 m/s (7200 rpm). Our results show that the coefficients of friction for Z-lube increase with increasing speeds initially but later it decreases with increasing speeds. Stick-slip happens due to the friction force which changes as a function of speed. The high speeds can overcome stick-slip phenomenon easily than the low speeds due to the dynamic action giving lower coefficient of friction. Other effect at high rotational speed is the high frictional energy dissipation at the interface leading to lowering of the viscosity of the lubricant. Low viscosity can facilitate easy sliding and hence low coefficient of friction. From the Stribeck curve, a decreasing coefficient of friction indicates the change in the lubrication regime from boundary to mixed lubrication.

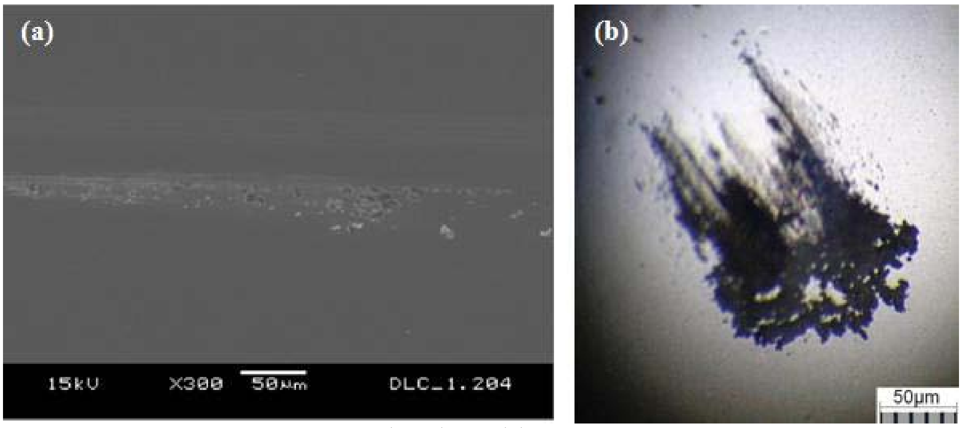


Fig. 3 *(a) SEM image of the wear track of bare disk (DLC) and (b) an optical image of Si3N4 sliding ball at 1.2 m/s (500 rpm) after 400 cycles of sliding.*

Therefore, we can say that the lubrication enters a mixed lubrication regime at higher speed. This may be possible as long as the lubricant still behaved as liquid-like without having gone any chemical degradation due to frictional heating. This change in the nature of the interface due to chemical degradation will also change the lubrication regime from mixed to asperity interaction (solid–solid) with corresponding increase in the coefficient of friction.

The coefficient of friction of Z-dol decreases with increasing rotational speeds. As we mentioned in section 3.2, as long as the rotational speed is low and the lubricant is able to replenish, no wear will occur. In the range of 3.61–9.63 m/s (1500–4000 rpm), the coefficient of friction of Z-dol is lower than that of Z-lube of all MW. In this range, the lubricant mobility is a strong function of functional end group of PFPE rather than its MW [18]. Within the same functional end group, Z-lube 5000 has higher coefficient of friction than Z-lube (3000 & 4000 MW) in the range of 3.61–6.02 m/s (1500–2500 rpm). Beyond 6.02 m/s (2500 rpm), the coefficient of friction of Z-lube 5000 is lower and mobility is not the key factor to reduce friction. We can conclude that the surface mobility is not the only factor for effective lubrication. Bonded layer between the lubricant and carbon overcoats (DLC), which helps to resist asperity penetration, and thus prevents solid-solid contact, is also essential [16]. Although higher MW has lower mobility, their ability to adhere strongly to the substrate (DLC) can reduce the friction. This property has played a greater role at high speeds. Also, high MW lubricant can withstand higher interface temperature with little reduction in viscosity. By observing the wear tracks on disks coated with Z-lube (3000 & 4000 MW) tested at the rotational speeds below 17.33 m/s (7200 rpm), we can infer that the critical replenishment rate of Z-lube (3000 & 4000 MW) is low for the rotational speed of 17.33 m/s (7200 rpm) and thus they wear out soon. Their wear durability is about 30,000 cycles at 17.33 m/s. Figure 5 shows the wear durability of Z-lube (3000, 4000 & 5000 MW) and Z-dol at 17.33 m/s. No wear occurred for disks coated with Z-lube 5000 and Zdol 4000 for 106 (one million) cycles at 17.33 m/s when the experiments were stopped. Figure 6 shows the ball surfaces of Z-lube 5000 and Z-dol 4000 after running one million cycles at 17.33 m/s and it is clear that the lubricants are still in liquid state. No wear track was observed on the disk surfaces even when observed under FESEM (Field Emission Scanning Electron Microscopy).

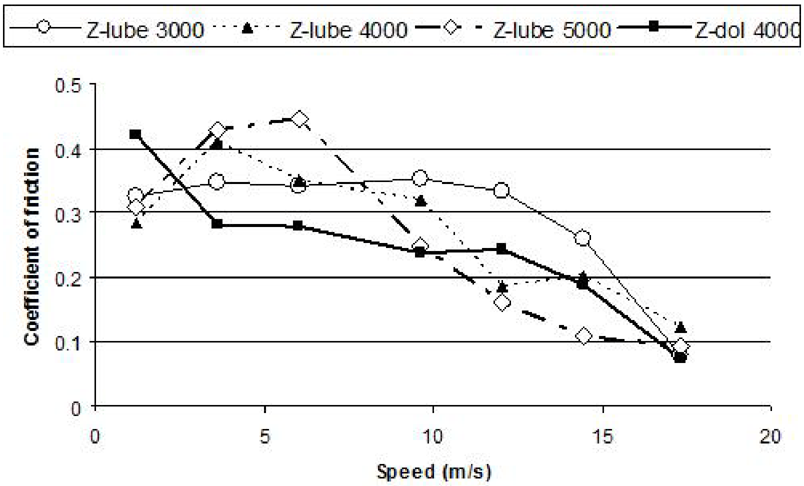


Fig. 4 *Coefficient of friction as a function of rotational speed for 4 mmSi3N4 sliding against Z-lube (3000, 4000 & 5000 MW) and Z-dol 4000 for first 400 cycles.*

From the study of friction and wear at high speeds, we can conclude that in order to get the best performance of a lubricant as a function of rotational speed, there are two key factors: first, the relation of functional end group versus rotational speed, and second, the relation of MW versus rotational speed. According to our results, in the range of 3.61–9.63 m/s (1500–4000 rpm), Z-dol 4000 is the best. Beyond 9.63 m/s, Z-lube 5000 is more preferable as far as the coefficient of friction is concerned. For the same functional end group for Z-lube, low MW is suitable for low rotational speeds and high MW is suitable for high rotational speeds. Before using any lubricant, we have to know the optimal replenishment rate of that lubricant to get the lowest friction and longest wear durability.

3.4 *Initial and final coefficients of friction*

During rotation on the same track for a large number of cycles (such as 100,000 cycles), transfer of the lubricant from disk to ball surface is unavoidable. The lubricant amount on the disk surface is reduced and the ability to resist sliding is decreased as sliding progresses. However, it is not obvious that the transfer of lubricant from the disk surface to the head results in an increase or decrease in the coefficient of friction. Figure 7 shows the comparisons of initial and final coefficients of friction of Z-lube (3000, 4000 & 5000 MW) at various speeds 1.2–14.45 m/s (500–6000 rpm). The initial coefficient of friction was defined as an average of first 400 cycles and the final coefficient was an average of last 400 cycles (99,601–100,000 cycles) of disk rotation. According to our results, there was no overall change in the frictional coefficient due to the sliding process for Z-lube (all MW) and the variations are within the experimental error.

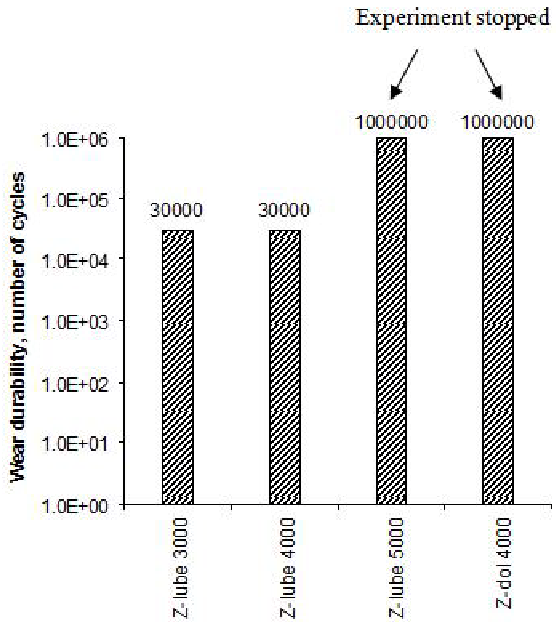


Fig. 5 *Wear durability of Z-lube (3000, 4000 & 5000 MW) and Z-dol 4000 at a normal load of 4 g and at a rotational speed of 17.33 m/s (7200 rpm).*

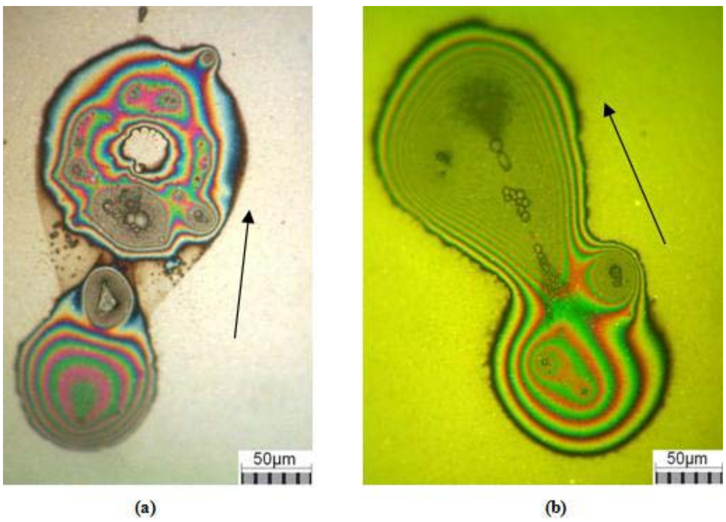


Fig. 6 *Optical microscopic image of Si3N4 ball surfaces against (a) Z-lube 5000 and (b) Z-dol 4000 at 17.33 m/s (7200 rpm) for 106 (one million) cycles. Solid arrows show the direction of disk sliding.*

3.5 *Fluctuations in the friction reading*

Figure 8 shows the fluctuations of the coefficients of friction for Z-lube 3000 at various speeds. There are large fluctuations at high sliding speeds (14.44 & 17.33 m/s) in comparison with low sliding speeds. Wear of the disk initiated followed by large fluctuation for Z-lube (3000 & 4000 MW) at 17.33 m/s around 30,000 cycles. Similarly, for Z-lube 5000 and Z-dol 4000, their fluctuations in the coefficient of friction values at high speeds are higher than at lower speeds. But in comparison with Z-lube (3000 & 4000 MW), the changes of their fluctuations at various speeds are not significant. At low speed, there is a small temperature rise with time during sliding but at high speed, the temperature rise is significant. The elevated temperature can degrade and solidify the lubricant. The extent of thermal degradation of lubricant mainly depends upon the functional end groups and the MW. A lubricant with high MW has superior resistance to thermal effects and high viscosity even at high shear rates whereas that of lower MW has inferior resistance to thermal degradation due to friction-generated heat at high speeds. As a result, there are large fluctuations and wear occurrences in the case of Z-lube (3000 & 4000) compared with Z-lube 5000. Stable-unstable range of velocities for Z-dol 4000 and Z-lubes are shown in table 3. Stable means when the fluctuation of coefficient of friction is within ±0.15 whereas if the fluctuations are larger than ±0.15, then it is assumed unstable.

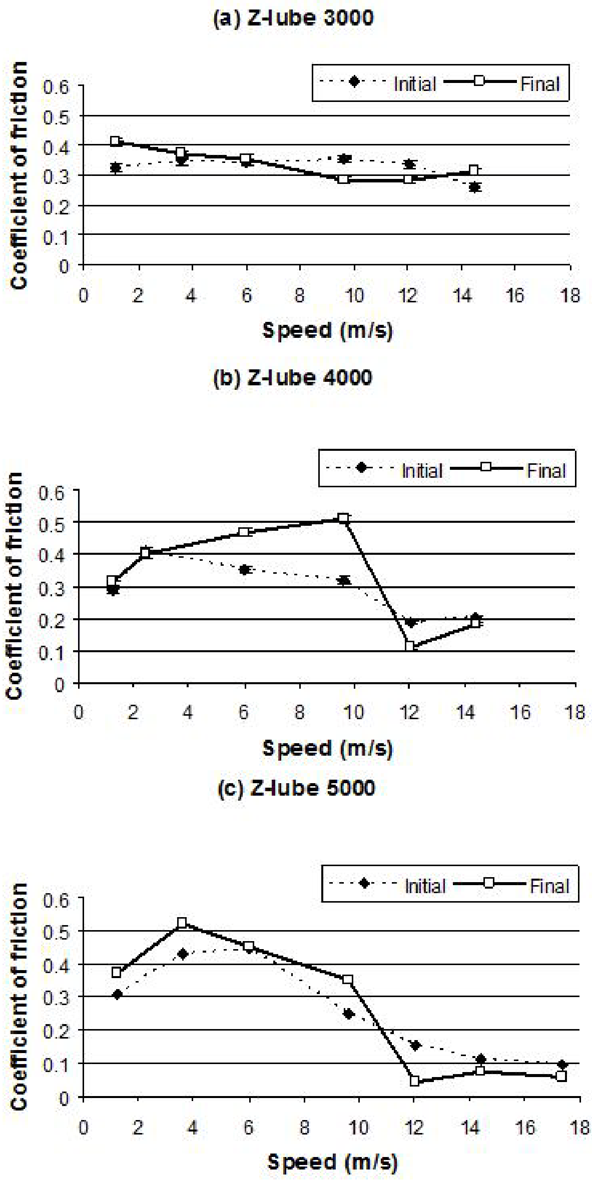


Fig. 7 Initial and final coefficients of friction as a function of rotational speeds for (a) Z-lube 3000, (b) Z-lube 4000 and (c) Z-lube 5000.

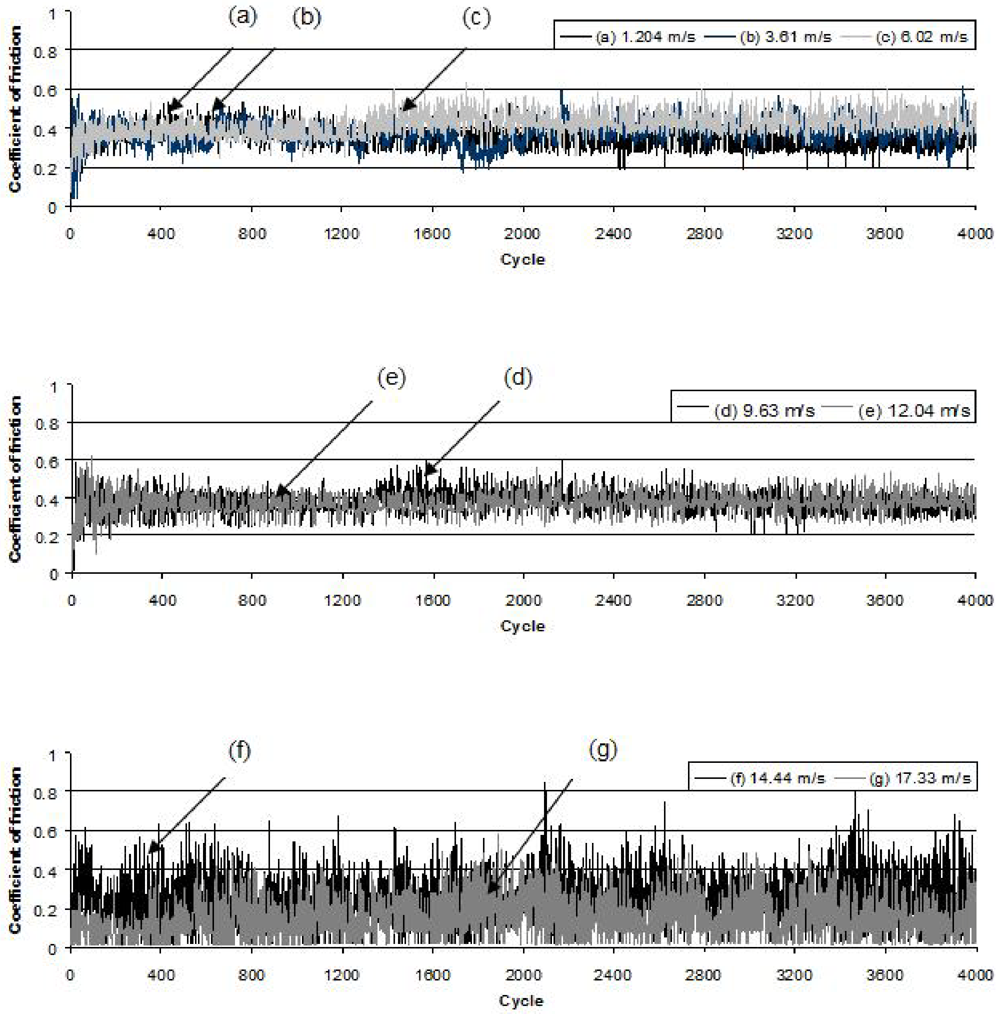


Fig. 8 *Fluctuations of coefficient of friction for Z-lube 3000 at various speeds for first 4000 cycles.*

Table 3. *Stable-unstable range of velocities for Z-dol 4000 and Z-lubes.*

Lubricant	**Range of velocities (m/s)**	
	Stable	**Unstable**
Z-dol 4000	1.24~17.33	-
Z-lube 3000	1.24~12.04	14.44~17.33
Z-lube 4000	1.24~12.04	14.44~17.33
Z-lube 5000	1.24~17.33	-

3.6 Optical images of ball surfaces

Figure 9 shows the optical images of the surfaces of the sliding ball against the disks coated with Z-lube 3000 after being spun at various speeds for 100,000 cycles. Figure 9 (i) (far-right bottom-most image) shows the ball surface of figure 9(g), after cleaning with acetone. There was no wear of the Si3N4 ball at any speeds. The lubricant is transferred from the disk to the ball in all cases. By observing figure 9, we can understand the transition of lubricant from liquid state to solid state with increasing speeds. In other words, the elevated temperatures generated at high speeds dry up the lubricant with possible degradation as the sliding speed is increased. Moreover, the degraded molecules behave like solid. A few particles were generated at low speeds but the extent of particle generation increased with rotational speed [10]. These particles may come out from lubricant or disk surface and they may act as third-body and lead to abrasion between at the head-disk interface. The asperities of these particles may penetrate into the coating layer leading to large scale coating failure. If the bonded layer is not strong enough, the particles can penetrate through the lubricant layer and lead to wear of DLC substrate. Z-lube (3000 & 4000 MW) cannot firmly coat on DLC in comparison with Z-lube 5000. This is the reason for early wear occurrence in Z-lube (3000 & 4000 MW) at 17.33 m/s (7200 rpm). Moreover, higher MW lubricant is thermally more stable than lower MW lubricant, which is essential at high rotational speed. By comparing figure 9(g) & (h), we can infer that the ball has reached DLC substrate through Z-lube 3000 at 17.33 m/s. Higher MW lubricants such as Z-lube 5000 and Z-dol 4000 maintain liquid-like behaviour even at high sliding speeds. The other effect to be considered would be the thickness of the lubricant, however, the above deductions are still valid as the lubricant thickness range was within a very controlled range of 2–4.7 nm.

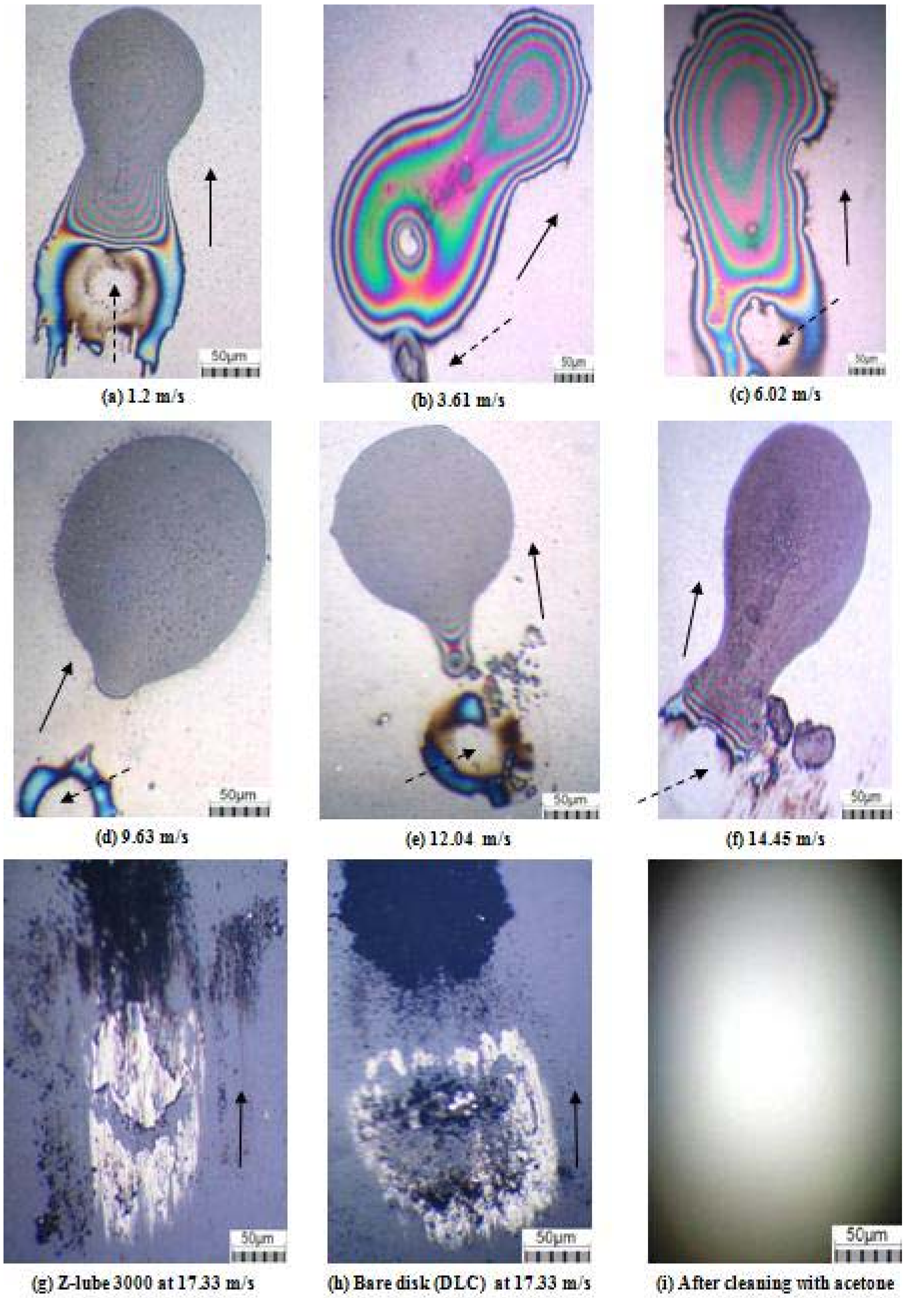


Fig. 9 *Optical microscopic image of Si3N4 ball surfaces against Z-lube 3000 after frictional test. The solid arrows show the direction of disk sliding and dotted arrows show the position of contact point between the disk and the silicon nitride ball surface. (a–f) show the ball surfaces at various speeds run until 100,000 cycles, (g) shows DLC particles from disk were adhered to ball surface at very high speed 17.33 m/s (7200 rpm) about 30,000 cycles, (h) shows the ball surface against bare disk (only DLC) at very high speed 17.33 m/s about 400 cycles and (i) shows image after cleaning the surface of (g) with acetone.*

4 Conclusions

In this article, we study the effect of rotational speeds on friction and wear durability of different PFPE lubricants coated on magnetic hard disks. From this study, the following conclusions are drawn:

1. AS1, which has a polar alkoxy silano group at both ends, has higher water contact angle, lower surface energy and lower wear durability (at 1.2 m/s) than for other PFPEs which have functional groups at both ends of their molecules. Z-lubes and Z-dol have higher surface energies than AS1, enough mobility and hence they can replenish the depleted area much easily. And, the thermal stability of AS1 is lower than both Z-lube and Z-dol. Therefore, no wear occurred at 1.2 m/s until 100,000 cycles for Z-lubes and Z-dol when the test was stopped.
2. The increase in rotational speed strongly affects the friction and wear durability of both functional groups, and the results may be summarized as:

    - At low speeds, the coefficient of friction increases and then decreases beyond a critical speed. This critical speed strongly depends upon the functional end groups and MW of PFPE.
    - Z-dol 4000 is better than Z-lube at low speeds. At high speeds, Z-lube 5000 is preferable due to lower coefficient of friction.
    - For the same functional group, low MW PFPE is suitable for low speeds and high MW is suitable for high speeds. That means mobile layer is important for low speeds, and, bonded layer with high viscosity of the mobile part is important for high speeds.
    - At 17.33 m/s, the wear durability of Z-lube (3000 & 4000) is about 30,000 cycles. No wear occurred at 17.33 m/s until 106 (one million) cycles of sliding for Z-lube 5000 and Z-dol 4000.

3. The increasing speeds have an advantage to reduce friction until certain sliding speed. Beyond this range, there is large fluctuation due to stick-slip phenomenon. Lubricants degrade due to high temperature at HDI and evaporation may occur easily in low MW than in high MW PFPE. At this stage, if the bonded layer cannot withstand the penetration of particles, the wear of the disk occurs.

Finally, results of this study would be very helpful in designing lubricants for high-speed engineered parts for high precision applications where thickness of the lubricant cannot be tolerated more than a few nanometer.

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References

1. B. Bhushan, Tribology and Mechanics of Magnetic Storage Devices, 2nd ed (Springer Verlag, New York, 1996).
2. Y.L. Zhu, B. Liu, Y.H. Li and Q.F. Leng, IEEE Trans. Mag. 35(5) (Sept, 1999) 2403.
3. A.K. Menon, (1999), "Critical requirements for 100 Gb/in^2 head/disk interface" ASME Proceedings of the Symposium on Interface Technology Toward 100 Gb/in^2 9:1–9.
4. A.M. Homola, IEEE Trans. Mag. 32(3) (1996) 1818.
5. A. Khurshudov and R.J. Waltman, Wear 251 (2001) 1124.
6. Z. Zhao and B. Bhushan, Tribol. Lett. 6 (1999) 129.
7. Q. Zhao, F.E. Talke and S.W. Meeks, IEEE Trans. Mag. 35(5) (Sept, 1999) 2406.
8. N.S. Tambe and B. Bhushan, Microsyst. Technol. 8 (2002) 409.
9. S.K. Sinha, M. Kawaguchi, T. Kato and F.E. Kennedy, Tribol. Int 36 (2003) 217.
10. D.Y. Lee, J. Hwang and G.N. Bae, Microsyst. Technol. 10 (2004) 103.
11. S. Sundararajan and B. Bhushan, J. Vac. Sci Technol. A 19 (2001) 1777.
12. H. Liu and B. Bhushan, Ultramicroscopy 97 (2003) 321.
13. T. Young, Philosophical Transactions of the Royal Society of London, Vol. 95 (1805) 84 [Also see Frederick M. Fowkes, "Attractive forces at interfaces" Industrial and engineering chemistry 56, 12 (December, 1964) 40–52].
14. C.J. van Oss, Interfacial Forces in Aqueous Media (Marcel Dekker, New York, 1994).
15. S.-K. Lee, M. Inoue, H. Kondo and H. Hanaoka, SID 02 Digest 33(1) (May, 2002) 830.
16. X. Ma, J. Gui, B. Marchan, M.S. Jhon, C.L. Bauer and G.C. Rauch, IEEE Trans. Mag. 35(5) (Sept, 1999) 2454.
17. H. Kondo, Y. Hisamichi and T. Kamei, J. Magn. Magn. Mater. 155 (1996) 332.
18. S.K. Deoras, S.-W. Chun, G. Vurens and F.E. Talke, Tribol. Int. 36 (2003) 241.