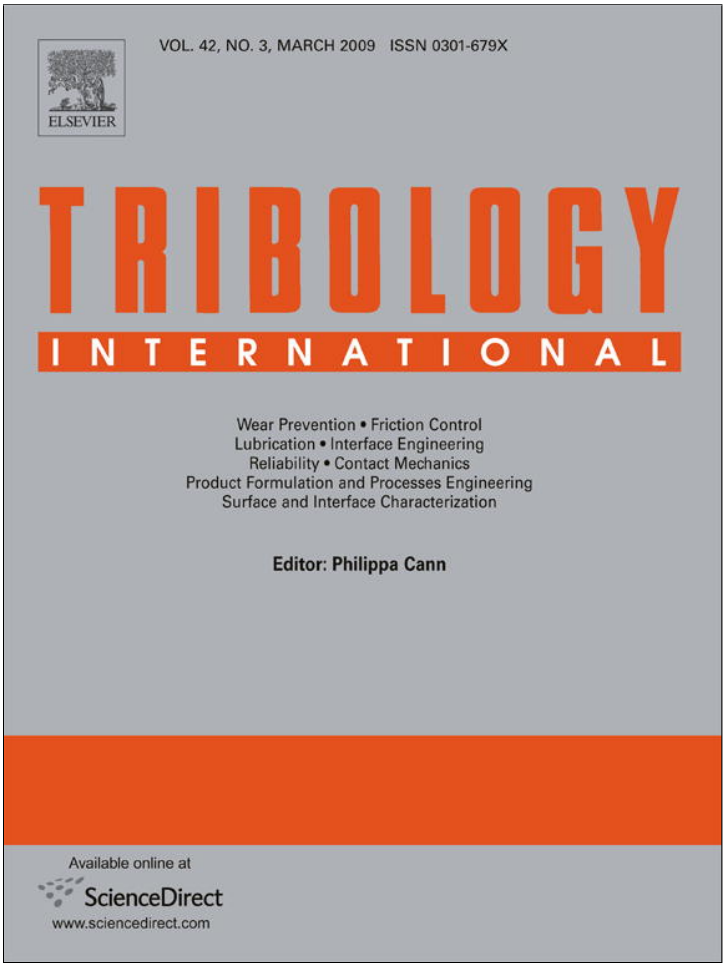
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# Application of fullerene-added nano-oil for lubrication enhancement in friction surfaces

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ABSTRACT

This paper presents the friction and antiwear characteristics of nano-oil composed of refrigerant oil and fullerene nanoparticles in the sliding thrust bearing of scroll compressors. The nano-oil was evaluated using a sliding thrust bearing tester. The friction coefficient of fullerene nano-oil at the lower normal loads ($<\sim$1200 N) under the fixed orbiting speed ($\sim$1800 rpm) was $\sim$0.02, while that of pure oil was $\sim$0.03, indicating that the fullerene nanoparticles dispersed in the base refrigerant oil improved the lubrication property by coating the friction surfaces. However the difference between friction coefficients for both nano-oil and pure oil was found to be negligible at higher normal load conditions ($>\sim$1200 N), indicating that the nanoparticles in the base oil have little effect on the enhancement of lubrication between the friction surfaces. The friction coefficient of nano-oil at various speeds of the orbiting plate in the sliding thrust bearing was found to be less than that of pure oil over the entire orbiting speed ranges between 300 and 3000 rpm. This is presumably because fullerene nanoparticles, which were inserted between the friction surfaces, improved the lubricating performance by increasing the lubricant oil viscosity and simultaneously preventing direct metal surface contacts.



## 1. Introduction

Scroll compressors are widely used in air conditioners for small- and middle-sized rooms due to their higher efficiencies as well as their lower vibration and noise characteristics. The compressors are known to give better performance than other compressors such as reciprocating or rotary types [1]. Fig. 1 shows a cross-sectional view of a low-pressure type scroll compressor. The pressure of the space between the fixed and orbiting scrolls is higher than that of the overall space of the hermetic compressors. This pressure difference is firmly pressed, and subsequently causes a normal load to the thrust bearing. Even though the friction loss in the thrust bearing makes up the largest part of the energy losses in scroll compressors [2], the characteristics of these friction losses have not been studied widely. It is important to understand the lubrication mechanism of the thrust bearings and the behavior of the orbiting scroll in such scroll compressors so that one can reduce the friction losses in the thrust bearing if the reliability and durability of scroll compressors is to be improved.

Ishii et al. [1] investigated the lubrication mechanism at the sliding thrust bearing of scroll compressors by conducting lubrication tests in a closed vessel pressurized with refrigerant R-22, and focusing on understanding the key factors in the lubrication for enhancing the compressor's performance. They found that lubrication at the sliding thrust bearing improved due to the pressure difference and formation of a wedge at the sliding surfaces of the thrust slide bearing. Okaichi et al. [2] performed hydrodynamic lubrication analysis and determined the bearing shapes of the journal bearing to give low friction losses with high reliabilities. They also tried to find the ideal thickness of the oil film and suggested that the bearing outer diameter and the bearing width should be 16 and 22 mm, respectively, to achieve the best lubrication performance. Fukuta et al. [3] reported that when the concentration of the refrigerant in the refrigeration oil stored in the compressors increased, the viscosity of the lubricant decreased significantly, and subsequently caused changes to the lubrication quality. The viscosity of the lubricant is related to molecular cohesive forces; low molecular cohesive force, which can only maintain low load support capacity of the lubricant, may eventually result in poor lubrication.

Many other works have investigated the lubrication characteristics of nanoparticle-added lubricating oil. For example, Ginzburg

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**Nomenclature**

| | |
|---|---|
| $v$ | relative velocity between two frictional plates ($m\,s^{-1}$) |
| $P$ | normal pressure ($kg\,s^{-2}$) |
| $X$ | inner surface of orbiting plate |
| $Y$ | middle surface of orbiting plate |
| $Z$ | outer of orbiting plate |
| $D$ | diameter of the plate |

*Greek letters*

| | |
|---|---|
| $\eta$ | dynamic viscosity ($kg\,m^{-1}\,s^{-1}$) |

*Subscripts*

| | |
|---|---|
| 1 | pure oil |
| 2 | nano-oil |
| in | inner location of frictional surface |
| out | outer position of frictional surface |

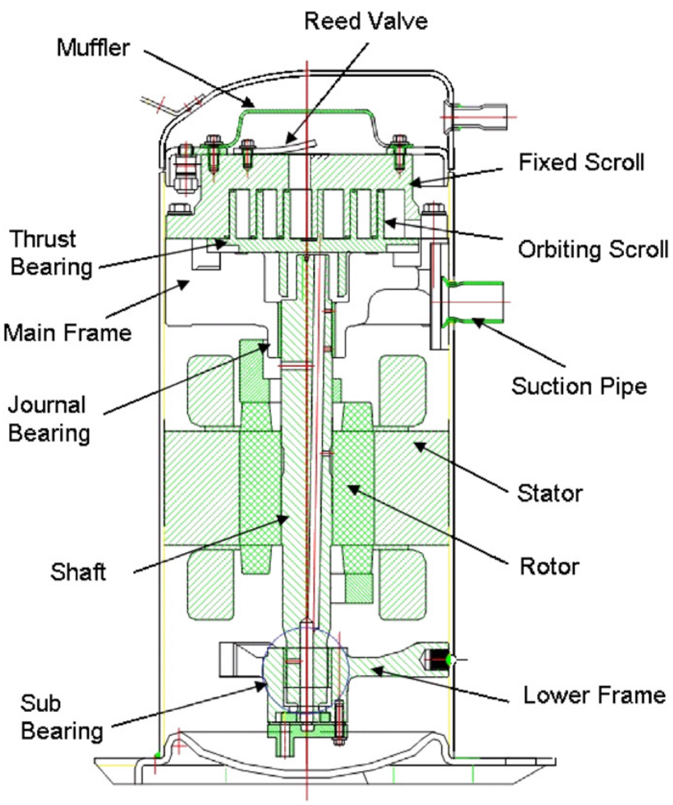


**Fig. 1.** Cross-sectional view of a low-pressure type scroll compressor.

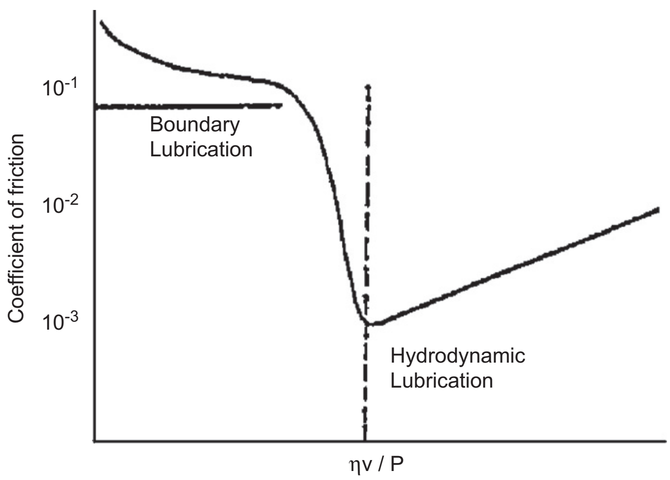


**Fig. 2.** Typical Stribeck diagram.

et al. [4] investigated the antiwear effect of fullerene-added I-40A industrial oil. The fullerene-oil-made film was formed on the copper foil surface of a steel roller which exhibited outstanding antiwear properties so that it enhanced the bearing strength of the tribotechnical unit and stabilized its operation at a fairly low friction coefficient. Talke [5] evaluated the contact from the viewpoints of tribology and placement on the Stribeck diagram which is frequently used as a convenient tool to analyze the lubrication and wear performances. In tribology, three lubrication regimes are often defined, namely, boundary, mixed, and hydrodynamic lubrication regions. Fig. 2 denotes the Stribeck diagram that is expressed in terms of the friction coefficient, which in turn is a function of velocity ($v$), dynamic viscosity ($\eta$), and normal pressure ($P$). Since most previous studies on fullerene additive to industrial oils were conducted in non-refrigerant conditions, the fundamental study of the friction characteristics of refrigerant oil under refrigerant conditions and various operating conditions of scroll compressors are difficult to find in the existing literature.

In this paper, the friction and antiwear characteristics of nano-oil composed of refrigerant oil and carbon nanoparticles in the sliding thrust bearing of scroll compressors are investigated using a sliding thrust bearing tester for evaluating the lubrication characteristics. The lubrication tests were conducted by measuring the friction surface temperature and friction coefficient at the sliding thrust bearing, as a function of normal load and orbiting speed. Also, the friction surfaces of the sliding thrust bearing were observed in order to examine the profile and image of the wear.

## 2. Experimental

Fig. 3 shows the schematic view of a sliding thrust bearing tester for evaluating the lubrication characteristics in scroll compressors. It was simply designed with two major plates which were the orbiting plate and the fixed plate. The surface between the orbiting and fixed plates replaced the frictional surface of the original thrust bearing in the scroll compressors. The sliding thrust bearing tester consisted of a closed test chamber, an air cylinder, two load cells, a servo motor, oil and refrigerant suppliers and heaters. The lubricant oil (4GSI, SUNISO) was supplied by an oil pump to the frictional surfaces from the inside of the friction surface to the outside. Two plates were located in a closed chamber. The inner space of the chamber was kept pressurized, at 5 bar, by using refrigerant gas R-22 in order to maintain the equivalent pressure condition in scroll compressors. A balance weight was attached to the eccentric shaft to reduce vibration. The normal load was applied via an air-cylinder system and controlled by a proportional-integral-derivative (PID) controller, which accurately controls the pressure of the air. The exact value

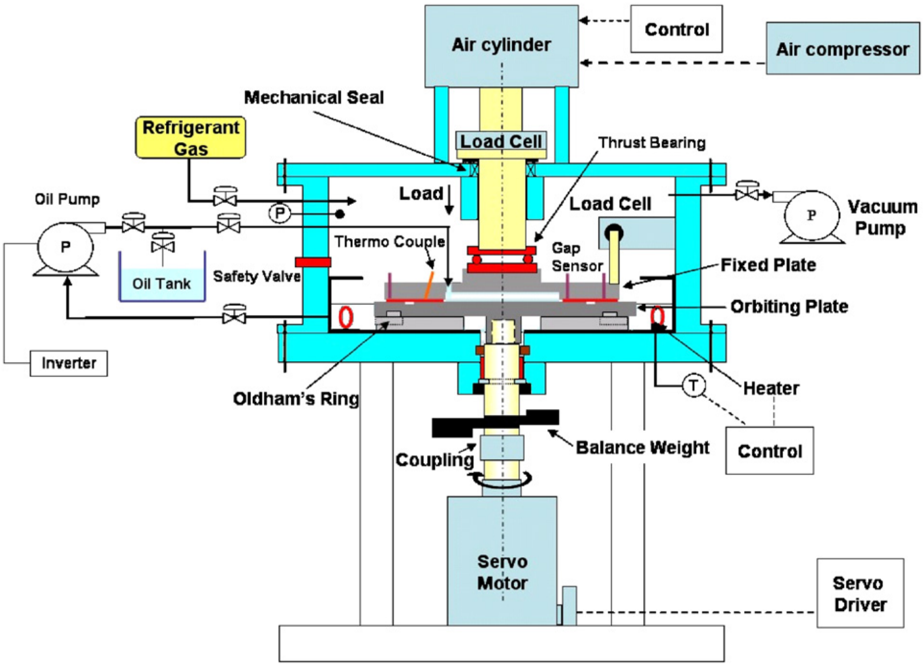


**Fig. 3.** Schematic of a sliding thrust bearing tester for evaluating the characteristics of the friction of pure oil and nano-oil.

of the applied normal load was measured by a load cell located under the air cylinder. The orbiting speed was controlled by an inverter in the servomotor. The frictional force was significant and it was measured by another load cell located in a closed chamber. The drag force acted on the frictional surface due to the orbiting motion of the orbiting plate and by the action of the normal load. The force made the fixed plate rotate in the same direction as the orbiting plate but the fixed plate was not able to rotate because it was fixed to the load cell which itself was fixed to the wall of a closed chamber. Therefore, the frictional force acting on the surface was measured by the load cell. The temperature of the frictional surface was also measured by two thermocouples attached to the fixed plate. The gap between the fixed plate and the orbiting plate was measured by inserted gap-sensors, which measured the distance between two solid materials at the micrometer scale. The frictional coefficient was measured as a function of orbiting speed and normal load.

Fig. 4(a) presents a simplified model of the fixed and orbiting plates showing the axial forces and the distribution of the pressure acting on the friction surfaces in the sliding thrust bearing of the scroll compressor. The apparent contact area is $6.86 \times 10^{-3}\,m^2$ calculated from the information given in Fig. 4(a). The inner and outer diameters of the fixed plate are 58 and 110 mm, respectively, and the diameter of the orbiting plate is 133 mm. The maximum contact pressure is 583.1 kPa when the normal force (or acting load) is 4000 N. The pressure distribution along the fixed surfaces decreases with increasing plate diameter, with the maximum value at the inner surface, causing a wedge formation between the friction surfaces. Fig. 4(b) is the result of simple computer simulation using the ANSYS WORKBENCH software to predict the pressure distribution along the orbiting surfaces. The simulation results indicate that the inner orbiting plate is more loaded than the outer orbiting plate at the same axial force, which is likely to be the reason for the formation of the wedge. The pressure distribution was adjusted by the magnitude of the axial force through loading of the air cylinder. The material used to make the fixed and orbiting plates was gray cast iron and their surface roughness was 1.3 and 2.1 μm, respectively.

Table 1 shows the major specification of the lubrication tests in the thrust-bearing tester used in this study. Before the lubrication tests were performed, the temperature of the supplied oil was allowed to reach 80 °C in order to give the equivalent operating conditions where the temperature of oil was about the same as in actual scroll compressors. The normal force and the orbiting speed were controlled up to 4000 N and 3000 rpm, respectively. The friction coefficients and the gap between the two plates were measured as a function of the orbiting speed in the range between 300 and 3000 rpm at the normal force of 3200 N. These tests were performed using both pure oil and nano-oil. The physical properties of mineral oil used were as follows: density = $0.915\,g/cm^3$ and kinematic viscosity = $54.6\,mm^2/s$ at 40 °C and $6.06\,mm^2/s$ at 100 °C.

Table 2 shows the types of test oils used in this study. Fullerene as a nanoparticle additive was added to the mineral oil in different volume fractions. Nano-oils I and II were the mixture of the mineral oil and fullerene at the volume fraction of 0.1% and 0.3%, respectively. Also carbon nanotubes were added to the mineral oil with a volume fraction of 0.1% to carry out various tests; these included suspension stability tests, extreme-pressure tests, and measurement of relative viscosities as a function of volume fraction.

## 3. Results and discussion

### 3.1. Physical properties of nano-oil

Fig. 5 shows the suspension stability of nano-oils evaluated by the ultraviolet spectrophotometer, which measured UV intensity of the nano-oil solution by evaluating the rate of nanoparticle sedimentation. The relative concentration was calculated by the ratio of the particle concentration intensity of the suspension at each time the measurement was taken divided by the initial concentration intensity of the suspension. A value of 1.0 at the relative concentration means excellent stability of the colloid solution without particle sedimentation. Rapid precipitation was observed for nano-oil III, which is the mixture of mineral oil and

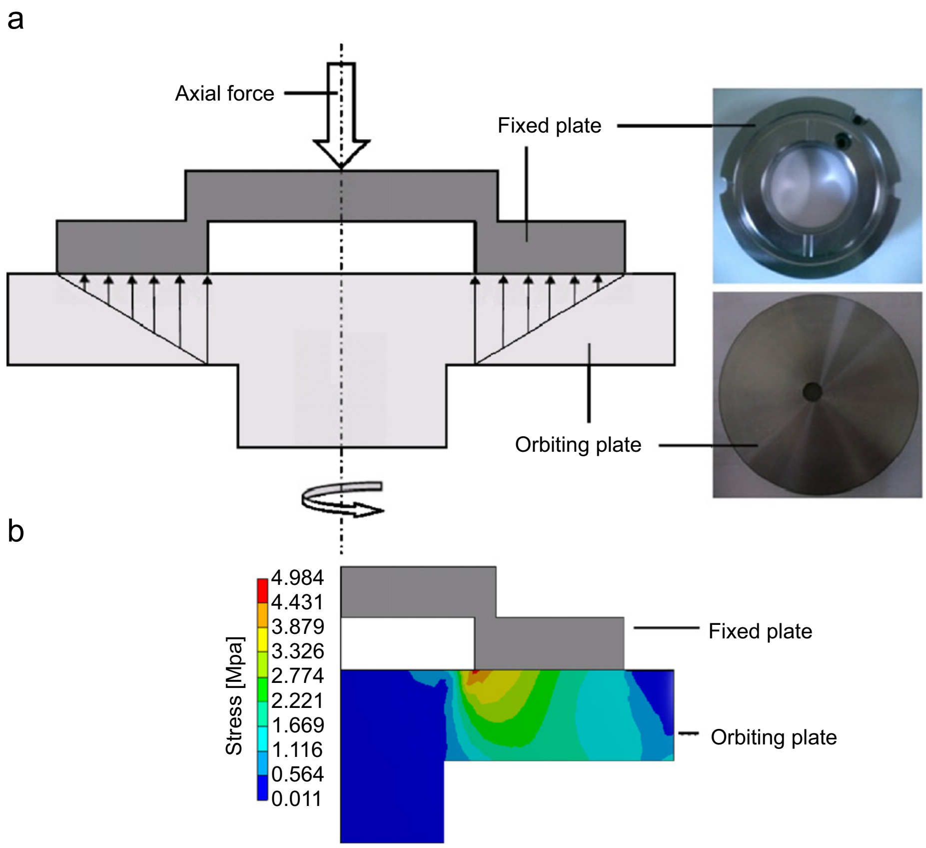


**Fig. 4.** Simplified model of the fixed and orbiting plates showing the axial force and the pressure distribution acting on the friction surfaces in the sliding thrust bearing of scroll compressors. (a) Simplified model and (b) simplified model simulated by ANSYS WORKBENCH.

**Table 1**
Major specification of the lubrication tests in the sliding thrust bearing tester in this study

| | |
|---|---|
| Axial force (N) | 0–4000 |
| Orbiting speed (rpm) | 0–3000 |
| Orbiting radius (mm) | 3 |
| Refrigerant oil | Pure oil, nano-oil |
| Refrigerant | R-22 |

**Table 2**
Types of pure oil and compositions of nano-oil

| Oil type | Solvent | Particle fraction |
|---|---|---|
| Pure oil | Mineral oil | Fullerene 0 vol% |
| Nano-oil I | Mineral oil | Fullerene 0.1 vol% |
| Nano-oil II | Mineral oil | Fullerene 0.3 vol% |
| Nano-oil III | Mineral oil | CNT 0.1 vol% |

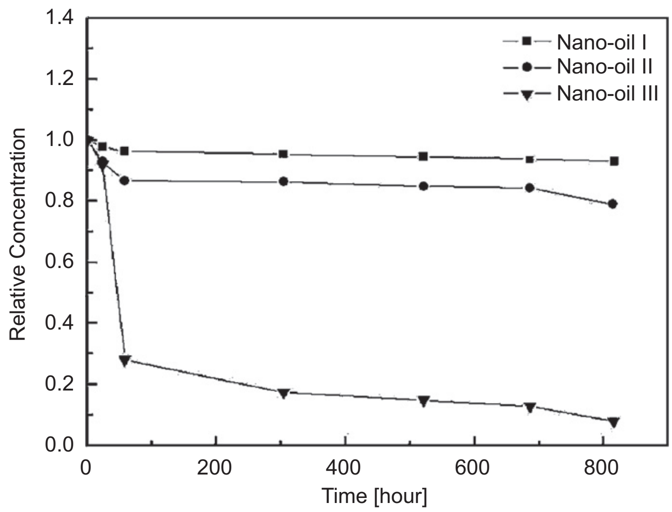


**Fig. 5.** Suspension stability of various nano-oils measured by UV–Vis spectrophotometer.

carbon nanotubes. After 58 h, the relative absorption dropped as much as 72%, indicating that nano-oil III was heavily agglomerated in the refrigerant oil suspension. On the other hand, little nanoparticle precipitation was observed in the nano-oil I and II suspensions for the mixture of mineral oil with fullerene. Even after 686 h, the relative absorption of nano-oils I and II dropped by only 6% and 16%, respectively, indicating that these nanoparticles had produced excellent stable suspensions.

Table 2 shows the results of extreme-pressure tests of the pure oil and the nano-oil. These extreme-pressure tests were based on the ASTM D3233 standard and involved measuring the breaking pressure of the oil film at the surface–surface contact [6]. The extreme pressure of the nano-oil having 0.1 and 0.3 vol% of carbon nanoparticles increased to 225% compared with that of pure oil, indicating that the oil films made by nano-oils I and II are much stronger than that produced by the pure oil. This is because the nano-oils have less opportunity to contact the metal surface because of the presence of the nanoparticles, while pure oil is in direct contact with the metal surfaces.

Figs. 6(a) and (b) show the relative and kinematic viscosities of nano-oils as a function of added volume fraction of nanoparticles

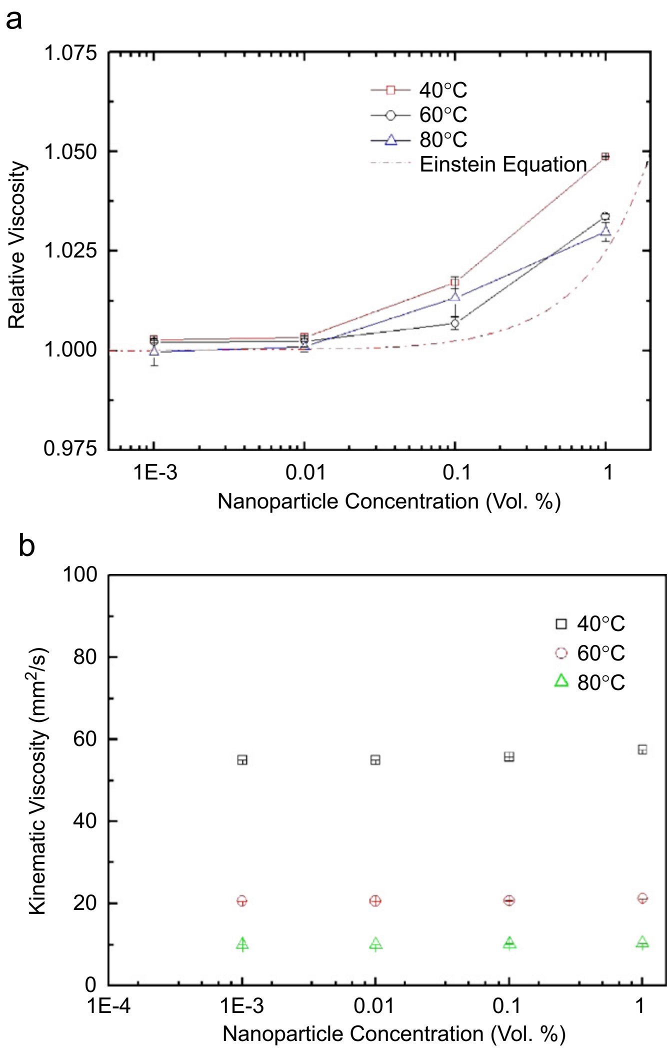


**Fig. 6.** Viscosity of nano-oil as a function of nanoparticle concentration and oil temperature ranging from 40 to 80 °C. (a) Relative viscosity and (b) kinematic viscosity.

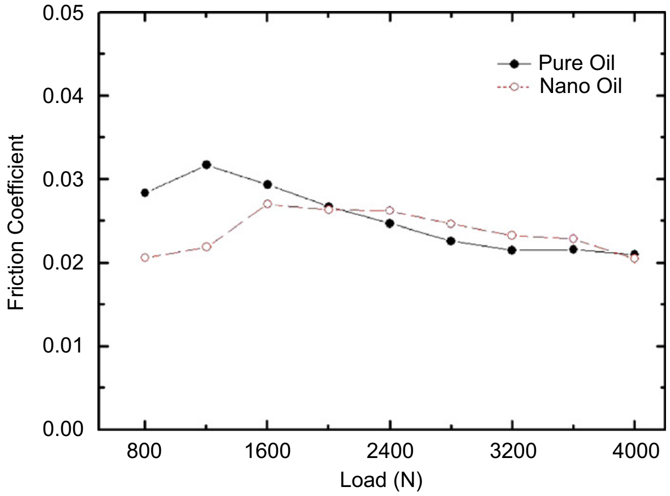


**Fig. 7.** Lubrication test results on the friction coefficient as a function of the normal axial force in the sliding thrust bearing tester at the fixed orbiting speed of 1800 rpm for the cases of pure oil and 0.1 vol% nano-oil.

in suspension for temperature ranging from 40 to 80 °C, respectively. The relative viscosity of the nano-oils was calculated by taking the ratio of the kinematic viscosity of nano-oil to the kinematic viscosity of pure oil. There was no significant change in the relative viscosity of the nano-oil under low volume added fractions up to about 0.01 vol% as shown in Fig. 6(a); in fact an approximate 5% increase in the relative viscosity of the nano-oil was observed when the volume fraction of the nanoparticles was over 0.01 vol%. The viscosity of nano-oil only increases slightly when the amount of nanoparticle additives is increased because the viscosity dissipation occurs due to the flow around a single sphere, or the Brownian movement of the particles. At very low particle volume fractions (<10 vol%), Einstein reported that the viscosity of a suspension of hard spherical particles could be predicted by the following simple linear formula:

$$\eta = \eta_s(1 + 2.5f) \tag{1}$$

where $f$ is the particle volume fraction. Eq. (1) has been modified by Batchelor [12], to show that the viscosity of the suspension increases with increasing particle volume fraction when $f$ is less than 10 vol%. As seen in Fig. 6(b), the kinematic viscosity is a strong function of oil temperature, as expected.

### 3.2. Lubrication tests for the nano-oils

Fig. 7 shows the lubrication test results for the nano-oil and the pure oil. The friction coefficients were measured as a function of the normal force under the orbiting speed of 1800 rpm and normal forces up to 4000 N. The loads are increased in steps of 400 N, from 800 to 4000 N and the friction coefficients are the time-averaged values for each test condition. The standard deviations for the average values are within 0.37% under the normal load of 1200 N. When the normal load exceeds 1200 N, the standard deviations increase by up to 20%. Therefore, the difference between the friction coefficients of pure oil and that of nano-oil is not within normal measurement errors. Fig. 7 shows that the friction coefficient of carbon nano-oil at the normal load less than 1200 N was 0.02, while the friction coefficient of pure oil was 0.03. This indicates that less metal contacts appear to occur with the presence of nanoparticles in the oil suspension. The higher extreme pressures corresponding to higher normal load conditions are known to reduce the possibility of metal contacts [7]. Although the lubrication regime is in the mixed lubrication range, there are areas where extreme-pressure conditions exist locally [8]. Therefore the increase of extreme pressures can reduce the friction coefficient and it has been verified that the extreme pressure of nano-oil is two times higher than that of pure oil (see Table 2). The lower friction coefficient of nano-oil, compared to that of pure oil at normal forces less than ~1200 N, results from the lower metal contacts at the same load. After the friction coefficient reaches the maximum value, it began to decrease, because both oils make the orbiting and fixed plates polished to some extent at the lower normal force conditions. Also the maximum friction coefficient of 0.02 for the nano-oil occurred at ~1600 N, while the maximum friction coefficient of 0.03 of pure oil occurred at ~1200 N. This indirectly indicates that the nano-oil gives better lubrication than pure oil at the lower normal force conditions (<~1600 N) because the lower maximum friction coefficient represents the worst condition for the lubrication performance.

Fig. 8 shows the lubrication test results of the friction coefficients as a function of the orbiting speed using the sliding thrust bearing tester at the normal force of 3200 N. The fixed plate was pressurized at a fixed normal force of 3200 N and the orbiting speed of the orbiting plate was decreased from 3000 to 300 rpm. Fig. 8(a) shows that the friction coefficient of the nano-oil was

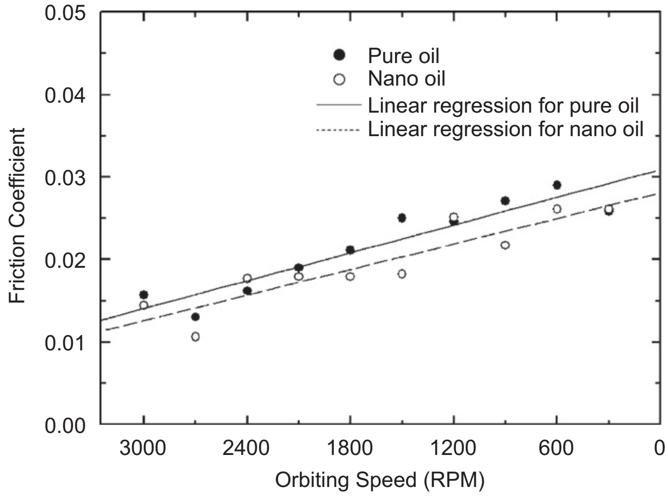


**Fig. 8.** Lubrication test results on the friction coefficient in the sliding thrust bearing tester at the fixed normal force of 3200 N for the cases of pure oil and 0.1 vol% nano-oil.

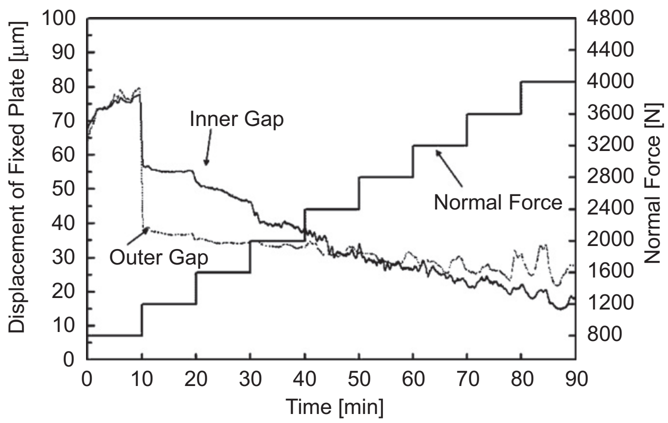


**Fig. 9.** Displacement of the fixed plate as a function of the normal axial force in the sliding thrust bearing tester at the fixed orbiting speed of 1800 rpm.

0.019, while that of pure oil was 0.022 under the conditions of refrigerant gas R-22 at the pressure of 5 bar. According to the Stribeck diagram shown in Fig. 2, a significant decrease in rotating speed results in the boundary lubrication regime, because the metal contact is known to frequently occur, and so the lubrication performance is poor. The friction coefficient of the nano-oil is less than that of pure oil over the entire orbiting speed ranges between 3000 and 300 rpm. This is because fullerene nanoparticles, which were inserted between the friction surfaces, improved the lubrication performance by increasing viscosity and preventing contact between the metal surfaces.

Fig. 9 shows the results of the displacement of the fixed plate as a function of the normal force using the sliding thrust bearing tester at the orbiting speed of 1800 rpm and for normal forces up to 4000 N. The displacement of the fixed plate decreases with increasing normal force. Initially the displacement of the inner fixed plate was higher than that of the outer fixed plate up to the normal force of 2800 N. This is because the lubrication oil is designed to be injected from the inner fixed plate to the outer fixed plate in the sliding thrust bearing [9]. However, above 2800 N, transversely, the displacement of the inner fixed plate is lower than that of the outer plate, indicating that the wedge of the fixed plate had formed at the higher normal force conditions ($>$2800 N).

### 3.3. *Wear analysis of frictional surfaces*

Fig. 10 shows an image of the wear and surface roughness of the orbiting plate at the orbiting speed of 1800 rpm and for normal forces up to 4000 N, for a 90 min test period. The black colored circles in the image of the orbiting plate appear on the inside of the fixed plate, and are believed to be a mark from oil carbonization due to the wedge phenomenon occurring between the friction surfaces in the sliding thrust bearing.

Table 3 shows the surface roughness of the orbiting plate operated at the orbiting speed of 1800 rpm and for normal forces up to 4000 N for a 90 min test period. The surface roughness of the orbiting plate for pure oil distinctively decreases from 2.5 µm in the middle area, $Y_1$, to 1.1 µm in the inside area, $X_1$, while for

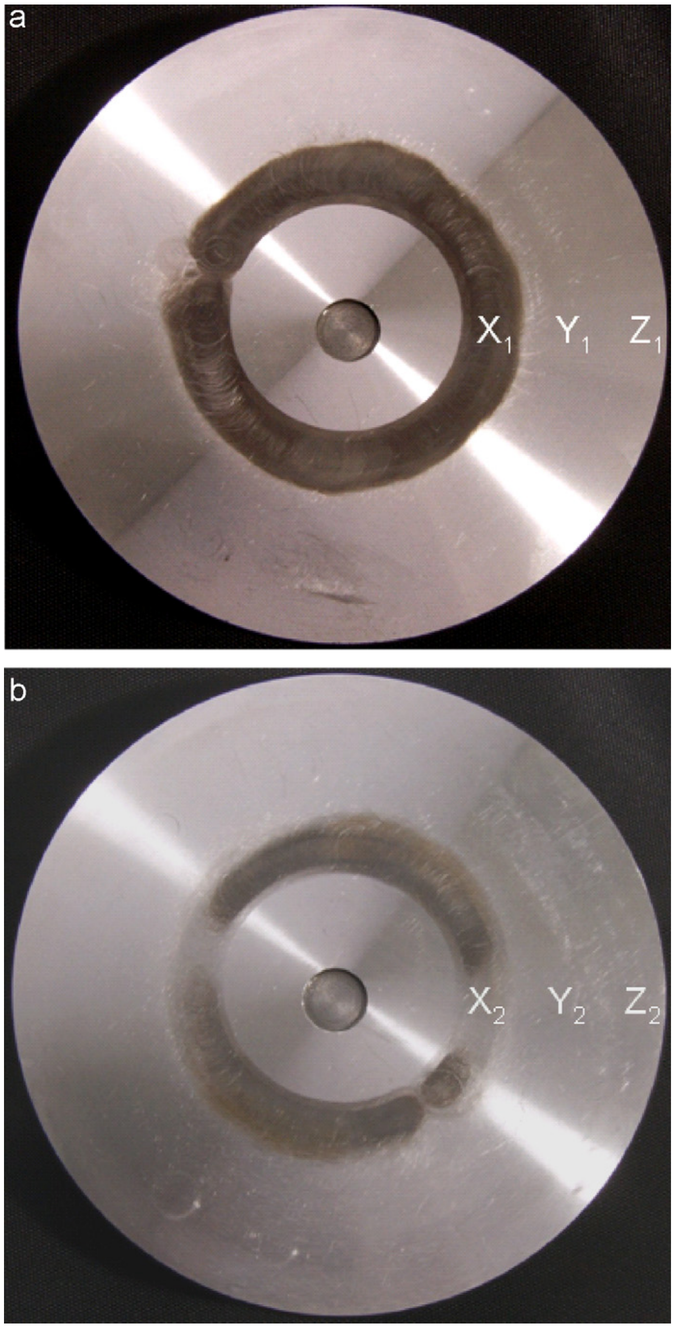


**Fig. 10.** Images of wear and surface roughness of the orbiting plate operated at the orbiting speed of 1800 rpm and for normal forces up to 4000 N for a 90 min test period with (a) pure oil and (b) nano-oil I.

**Table 3**
Results of extreme-pressure tests of pure oil and nano-oil based on ASTM D3233

| Oil type | Extreme pressure (N) |
|---|---|
| Pure oil | 1176 |
| Nano-oil I | 2646 |
| Nano-oil II | 2646 |

**Table 4**
Surface roughness of the orbiting plate at the orbiting speed of 1800 rpm and for normal forces up to 4000 N for a 90 min test period (locations of $X_1$ $X_2$, $Y_1$, $Y_2$, $Z_1$, and $Z_2$ are shown in Fig. 10)

| Location | Surface roughness (μm) | |
|---|---|---|
| | Pure oil | Nano-oil |
| $X_1$, $X_2$ | 1.1 | 1.0 |
| $Y_1$, $Y_2$ | 2.5 | 1.1 |
| $Z_1$, $Z_2$ | 1.3 | 1.2 |

nano-oil it decreases from 1.1 μm at $Y_2$ location to 1.0 μm (see Fig. 10) at the $X_2$ location. It is assumed that carbon nanoparticles were coated on the frictional surfaces so that the presence of nanoparticles between the frictional surfaces seemed to prevent metal contact. As more nanoparticles were added to the coated area, the microstructure of the coating became finer and more compact [10]. According to previous works by Ginzburg et al. [4], and Rapoport et al. [11], it was found that (i) the fullerene molecules accelerate the self-restoration of the polymeric tribo-film damaged in the course of the mechanochemical degradation and (ii) the fullerene particles suspended in the nano-oil have spherical structure and play a role as ball-bearings in the friction surfaces, which were identified by the lower friction coefficient of nano-oil compared with pure oil. Therefore the addition of fullerene in oil can reduce the carbonization of the oil and lower the friction coefficient (Table 4).

Fig. 11 shows the glow discharge spectrometer (GDS) profiling for measuring raw atomic concentrations of the orbiting plate at the orbiting speed of 1800 rpm and for normal forces up to 4000 N for a 90 min test period. The GDS analysis revealed that the black color circles in the image of the orbiting plate, shown in Fig. 10, mostly consisted of iron and carbon. The amount of carbon on the surface lubricated with nano-oil was 5% in mass larger than for the pure oil-lubricated surface, which was expected because of the presence of fullerene nanoparticles in the nano-oil. As a carbonization criterion, an arbitrary figure of 90% mass concentration of iron was chosen to quantitatively determine the degree of carbonization of the iron orbiting plate. When the mass concentration of iron is over 90%, the material of the orbiting plate was considered to be very close to the pure iron plate, implying that the plate does not become very carbonized in the given depth. It can be seen in Fig. 11 that the 90% mass concentration of iron was reached at a depth of 1.7–1.8 μm for the pure oil and 0.7–0.8 μm for the nano-oil. This indirectly indicates that the nano-oil enhanced the characteristics of the antiwear and friction resistance in the sliding thrust bearing in the scroll compressors.

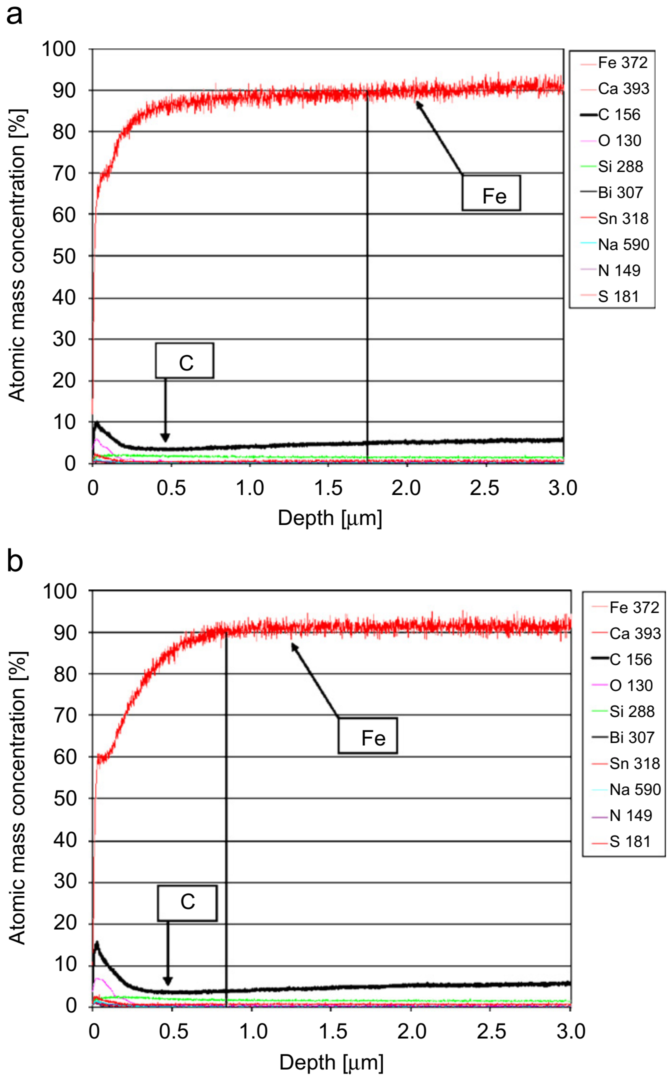


**Fig. 11.** GDS profiling for measuring raw atomic mass concentrations of the orbiting plate operated at the orbiting speed of 1800 rpm and for normal forces up to 4000 N for a 90 min test period with (a) pure oil or (b) nano-oil I.

## 4. Conclusions

In this work, the lubrication tests of a sliding thrust bearing in scroll compressors have been conducted in a closed chamber with the refrigerant gas (R-22) focusing on the lubrication characteristics of fullerene nanoparticle-dispersed oil. The friction coefficient and friction surface temperature for using both nano-oil and pure oil have been evaluated by varying the sliding thrust bearing operating parameters, including the normal forces and the orbiting speed of the friction plate. The pressure distribution along the fixed surface was analyzed using the ANSYS WORKBENCH software. It was found that the pressure distribution caused a wedge formation of the fixed plate, which was the key factor in the differences measured for the surface temperatures of the inner and outer plates. In addition, it has been shown that oil carbonization occurs within the orbiting plate. The lubrication properties of nano-oil were found to improve by observing its GDS profiling that the 90% mass concentration of iron was reached at a depth of 1.7–1.8 μm for pure oil and 0.7–0.8 μm for nano-oil. As a result of various friction tests, the presence of fullerene nanoparticles in the lubrication oil was found to improve the lubrication performance in the friction surfaces by reducing the metal surface contacts.

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