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EXPERIMENTAL SIMULATION OF ELASTOHYDRODYNAMIC ELLIPTICAL CONTACT IN SCREW COMPRESSOR ROTOR TRIBO PAIR

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Abstract

In screw compressor, the premature failure of rotor/roller contact has been observed intermittently in the industrial application. Therefore, the elliptical contact in screw compressor is investigated experimentally to study failure of tribo contacts. In pin-on-roller tribo-tester, the load capacity of lubricants is simulated by conducting experiments in controlled way to determine the cause of failure of rollers/rotors contacts. A plan of experiments based on Taguchi technique was used to acquire the data in a controlled way. The continuous presence of the lubricant film depends on the contact area, the load on the contact area, speed, operating temperature, surface finish, and the oil viscosity. Based on the results of the surface morphology, it has been suggested that the test specimens have great possibility to exhibit higher friction and higher wear. This implies that the thin lubricating film formed due to the inadequate hydrodynamic action. To mitigate the problem of wear and friction in the rotor contact, commercial oil has been used without additive and Tribo-performance results are reported.

Keywords: Elliptical Contact, Coefficient of Friction, WSD, Surface Morphology, Pin-On-Roller Tribo Tester.

1. INTRODUCTION

The screw compressor, a type of gas compressor, operates as per the rotary-type positive- displacement mechanism. It is commonly employed in applications demanding large volume and high pressure of air. It is used as a replacement of piston compressors, either in heavy industries or in high power air tools, with capacities ranging from 7.5 KW to 37 KW[1]. An oil-injected screw compressor often finds application in industries such as petrochemicals, mining, pharmaceuticals, wastewater treatment, rubber and plastics, automotive, apparel, agriculture, power generation, general manufacturing & many more [2]. In most cases, except for an increase in noise and vibration levels, total rotor failure is often catastrophic. Even though they are properly and efficiently lubricated these components are susceptible of failure. Rigorous attempts have been made to identify and rectify the failures so that they do not reoccur in the system [3,4]. These components often fail due to wear, scuffing, fretting and fatigue [5,6]. The failures are predominantly dependent on the operating conditions, improper and insufficient lubrication and the roughness of the contacting surfaces. The failure of these components is often marked by the failure of lubricating film. In this context attempts have been made to study the failure of lubricating film to assess the failure criterion of the mechanical components. Horng [5] studied the friction and wear behaviour of separated sliding steel surfaces under boundary lubrication. The results revealed that the maximum contact friction and wear is

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observed on the onset of boundary lubrication, beyond which the value of friction and wear is relatively low even after increase in the contact load. Jiang and Barber [7] investigated the failure of reaction film which accounts for mechanical wear. The result indicates that alteration in type of additives and their concentration can enhance scuffing load capacity in high speed and high load gear applications. The model proposed by the authors can also aid in determining the minimum number of scuffing tests using P-V curve. Ghosh and Sadeghi [8,9] simulated wear of rough surfaces, where the surface was treated as a collection of asperities of different radii and heights. It was predicted that surfaces with high roughness, kurtosis and positive skewness exhibited higher wear rates. Prakash et al. [10] studied the nature of scuffing in boundary lubricated sliding contacts with subsurface plastic deformations. The lubricant performance in boundary lubrication regime was found to have profound role in safeguarding the surface from severe deformation and micro- cracks. Owing to which the asperities got flattened resulting in surface smoothening. Xue et al. [11] and [12] proposed a method for predicting scuffing failure in spur gear pairs. The study revealed that the scuffing load capacity is greatly influenced by dynamic condition. Evans and Snidle [13] analyzed the influence of transverse roughness at the contact edges in EHL line contacts both analytically and experimentally. It was reported that the side flow of oil from the contacts reduced the film thickness, consequently resulting in scuffing failure. Olver et al. [14] investigated the effect of micropits and associated cracks on the film thickness distribution in EHL contact. The results revealed the presence of thinner films over larger region at negative slide roll ratio. Dong Zhu et al. [15] presented an approach for the prediction of pitting life based on newly developed mixed EHL model. The results revealed that on optimum surface topography is essential for improving the pitting life. Li et al. [16] predicted the influence of crack formation on fatigue lives of spur gear contacts operating under mixed lubrication conditions. Xu et al. [17] examined the friction behaviour and mechanism of composite coatings under different tests. It was observed that the interaction of subsurface and surface cracks resulted in spalling of coatings which is the main reason for fatigue failure. Since there are very limited information concerning the wear and friction of the compressor elements. Therefore, attempts were made in the current study to investigate mixed lubrication phenomena in order to better understand the failures of lubricated rotors / rollers contact in the screw compressor.

2. BACKGROUND

In the present paper, the problem related to lubrication related failures of rotor contacts of the screw compressors has been analyzed. Rotary screw compressors rely on the movement of two paired spiral rotors or rollers to create their streamlined airflow. Each rotor is spiral in design and responsible for taking in air and fluid, moving and pressurizing these elements from their initial intake opening. The screws rotate together to compress air, trapping and holding it within their curved divots and preventing its release. Rotary screw compressors use oil on chamber cylinder walls and bearings to keep machine parts lubricated and running smoothly. This is known as an “oil bath” or “oil flooded” lubrication maintenance. For rotary screw air compressors, the rotors are among the most

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critical as components. They are the vital component in compression process- without which, these air compressors simply couldn't function or exist as they do. Rotors or rollers come in pairs and are built inside cylinders. When the rotors rotate at high, smooth, sweeping speeds, they create pipeline for suctioned air to move through, compress and then discharge out with ultra-pressure. Interlocking twin rollers are the heart for reliable functioning of rotary screw compressor. The unavoidable wear and tear is the outcome of continuous running of industrial compressors. The rotors/rollers contact observed in the screw compressor is shown in Figure 1(a-b). Using the concept of reduced geometry, the rollers in contact can be represented using a pin on roller forming an elliptical contact as shown in Figure 2. The rotors are partially submerged in the lubricant reservoir. The lubricant used in this application is commercial lubricant HLP 46 (6.34 cSt at 100°C). As the lubricants protect rotors from failures, the reliability of rotors is affected by the lubricating oil used and its viscosity [18,19]. Fleming et.al.[20] reported that the different parameters of lubricating oil such viscosity, limiting shear stress etc. affects the functioning of rotor contacts. In this study, pin-on-disk (vane-roller) contacts lubricated with HLP(N) 46 lubricant tribological characteristics have been assessed by varying the sliding speed normal load and temperature. The models developed basically confine to the elliptical contact representing the simplified geometry of rotors contacts in the screw compressor is shown in the Figure 2.

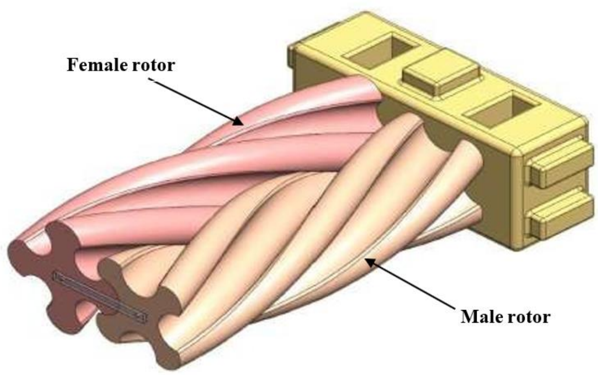

Figure 1: a) Schematic view of rotors contacts in the screw compressor model

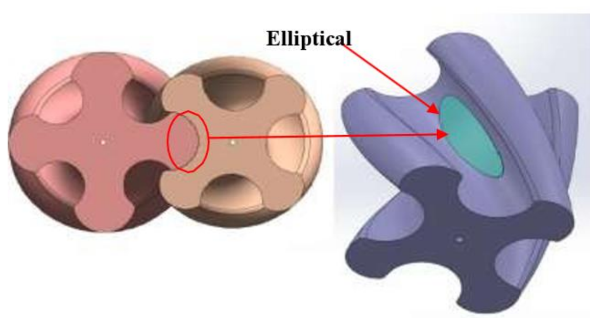

Figure 1: b) Rotors contacts in the screw compressor model represents elliptical conjunction

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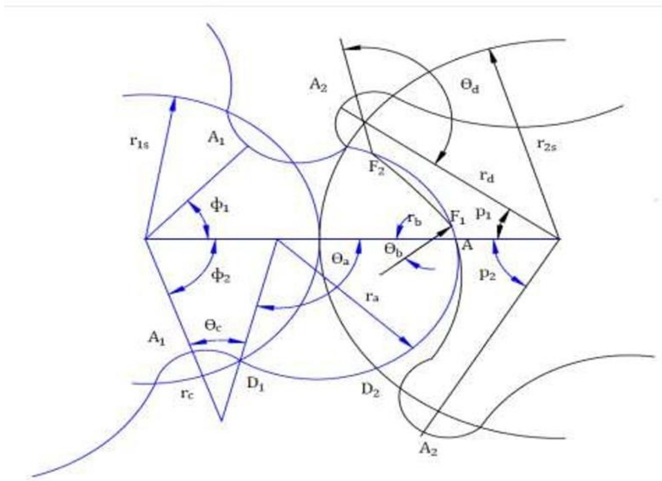

Figure 2: Simplified geometry representation of rotors contacts in the screw compressor

3. EXPERIMENTAL STUDY

To avoid wear and tear that comes with continuous, industrial compressor running and use. Since rotors are the heart of compressor's very existence, regularly inspecting and testing their productivity is essential. An orthogonal array and ANOVA was employed to investigate the influence of normal load, speed and temperature on lubricated rotors/rollers sliding wear of the test specimens.

3.1 Plan of experiments

For designing of such experiments, standardized orthogonal array' is used. Designing is based on the degree of freedom (DOF) of the orthogonal model DOF either greater or equal to the addition of wear parameters for an array model. Taguchi is the traditional and robust approach for designing which decides the susceptibility of model performance. A mathematical least square technique called ANOVA i.e. analysis of variance [24]. The need for statistical ANOVA is to evaluate significant design parameter which influences the wear features. ANOVA shows the significance of various parameters and their interactions at the confidence interval of 95%. The interaction between the parameters have statistical significance but do not have any physical significance which means the error is greater than the percentage contribution of interaction. For the confirmation of the experimental conclusion, the error between the experimental and predicted value is small.

The experiments were performed with the motive to correlate the effect of operating parameters namely, load(N), speed(rpm) and temp (°C) with contact friction(μ) and wear. The contact friction and wear (in terms of wear scar diameter) were obtained for different combination of operating parameters as determined using orthogonal array as presented in Table 1.The Selection criteria of orthogonal array was based on the degrees of freedom which should be greater than or equal to the summation of contact friction and

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wear[21].The experiments were conducted in accordance with the level of parameters generated in each row of orthogonal array. Twenty-four (24) different tests were conducted in accordance with the developed orthogonal array. The contact friction and wear results thus were obtained then subjected to ANOVA analysis.

In this present study, pin-on-roller tribometer contact geometry is used. It is a replica of vane-on- roller contact geometry used in compressor. Vane is replaced with spherical ended pin test specimen. Therefore, the radial clearance position and the major components for the rotary piston compressor are illustrated in Figure 1. The roller contact area with the pin (vane), with partly reciprocal motion controlled by the load, operating speed, viscosity of lubricant and so on, moves forward into the direction of revolution. The reciprocating motion area is subject to a very severe lubricated state because a lubricant film cannot be formed at zero relative velocity and the friction heat cannot be removed effectively from the area. Wear takes place primarily between pin (valve) and disk (roller), valve and cylinder slot and roller and cylinders in the rolling piston compressors (Jonsson (6) and Cho et al. (7)). The wear between the roller and the pin (vane) is the most critical among these sliding pairs [22].

Table 1: Taguchi orthogonal array design for coefficient of friction and wear

Test	Load (N)	Speed (rpm)	Temp (°c)	Coefficient of friction(μ)	Avg. W.S.D. (mm)
1	50	200	25	0.0814	0.382
2	50	200	25	0.1104	0.448
3	50	200	25	0.0537	0.481
4	50	400	50	0.068	0.412
5	50	400	50	0.0952	0.503
6	50	400	50	0.0772	0.576
7	50	600	75	0.0881	0.651
8	50	600	75	0.0989	0.695
9	50	600	75	0.1101	0.666
10	100	200	50	0.1173	0.673
11	100	200	50	0.1055	0.695
12	100	200	50	0.11056	0.703
13	100	400	75	0.0973	0.698
14	100	400	75	0.1071	0.687
15	100	400	75	0.0996	0.672
16	100	600	25	0.1183	0.683
17	100	600	25	0.1188	0.658
18	100	600	25	0.1136	0.681
19	150	200	75	0.1221	0.628
20	150	200	75	0.1274	0.658
21	150	200	75	0.123	0.675
22	150	400	25	0.1279	0.719
23	150	400	25	0.1191	0.731
24	150	400	25	0.1049	0.691
25	150	600	50	0.1479	0.752
26	150	600	50	0.1491	0.695
27	150	600	50	0.1493	0.731

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3.2 Experimental setup and procedures

The pin-on-roller tribotester used for the experimentation along with its contact geometry is shown in Figure 3. The EN-31 make spherical ended steel pins of 8mm diameter in contact with EN-31steel roller has been used as the test specimens. The pin and the roller are arranged in such a manner that the axis of pin axis of pin specimen is perpendicular to the axis of rotation of the roller. The spherical ended of the pin rests on the roller along its circumference. The roller is assembled on the driver shaft firmly and is able to rotate at the desired speed. On the contrary the pin is stationary and forms an elliptical contact. The roller is partially submerged in the lubricant reservoir. The roller while rotation carries the lubricant along its circumference thus forming thin lubricant film at the contact. The contact is loaded with the help of pneumatic bellows at the of the holder. The test rig is computer controlled and operating parameters are controlled through appropriate experimental programme. The contact friction is continuously monitored during the test duration with help of data acquisition system. At the end of the test, the pin specimen were removed cleaned for wear measurement. The wear scar obtained on the pin specimen was using industrial stereo zoom apochromatic microscope and the wear scar diameter measuring. The worn out surfaces was then used analyzed using SEM and optical profilometry, EDX to investigate mode and mechanism of wear the possibility of boundary film formation if any and determine the morphology of worn out surface. Initially 24 test specimens were used for conducting the experiments of one-hour duration each. In order to eliminate the human and experimental error, each of the test was repeated twice and after examining the specimens, data of eight test specimens were sensored out.

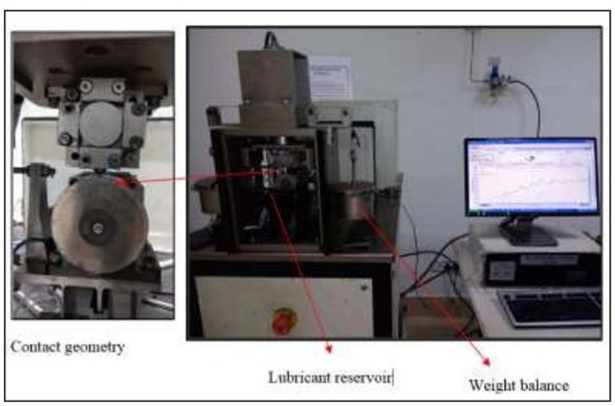

Figure 3: Experimental set up pin-on-roller and contact geometry

3.3 Materials

3.3.1 Lubricant

In the present experimental work, commercial oil (HLP 46) has been used to lubricate the scaled down of screw compressor. The physico-chemical characteristics of the lubricant selected are as given in the table 2.

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3.3.2 Test specimens

The test specimens i.e. the spherical ended pin and the roller were fabricated using forged EN-31 steel (1% C. Cr steel). The surface finish and hardness of the test specimen was limited to N3-N4 grade and 60 ± 2 HRC, respectively. Material composition of the test specimen is given in Table 3.

Table 2: Physico-chemical characteristics of commercial lubricant and test specimen

S.no.	Characteristics Lubricant	Characteristics Lubricant range
1	Kinematic Viscosity @ 40°C, cSt	43-48
2	Viscosity Index, Min	98
3	Flash Point(COC), °C, Min	200
4	Pour Point, °C, Max	(-)15
5	Emulsion Test D-1401, 40-37-3, Minutes, Max	20
6	FZG, Rating Stage, Failure	10

Table 3: Material composition of test specimen

Composition of EN31	EN31(%)
C	0.9
Si	0.1-0.35
Mn	0.3-0.75
S	0.05
P	0.05
Cr	1.0-1.6
Modulus of elasticity(Gpa)	202
Poisson’s ratio	0.3
Hardness (Gpa)	7.45

4. RESULTS AND DISCUSSION

The effect of the load, speed and temperature on wear and friction in contact was planned according to Taguchi orthogonal array design as shown in Table 1. The results are explained by analyzing friction and wear loss of test sample. Wear loss of specimens describe by interpreting the metallographic images after wear test and then evaluate the wear loss with different parameters by Taguchi method. In the succeeding section, tribological performance characteristics, surface morphology, EDX analysis and statistical analysis has been discussed.

4.1 Effect of operating parameter on tribo-performance of HLP-46 oil lubricated contacts

4.4.1 Influence of load

In order to understand the variation in contact friction behaviour with load for the continuous motion, tests were carried out at a different operating conditions under lubricated condition. The contact friction (μ) behavior of lubricated test sample is shown in Figure 4(a-e). The contact friction (μ) increases monotonically with an increase in load.

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However, with an increase in contact load, COF values increase. In the case of WSD, there is also a comparable trend. As the load increases from 50N-150N, the contact friction of lubricated test specimens increases. This is due to viscosity of the lubricant, that forms the thinner oil film between the sliding surfaces in contact. It seems that, 150N, for the operating load was more severe than the other load conditions in pin–roller contact. From table 1, seen that contact friction increases with increasing load. The lowest contact friction was obtained at 10N,200rpm and 25°C. At every test conditions, friction peaks can be observed indicating failure of lubricating film. The lubricating film get distorted resulting in direct metal to metal contact which in turn increases the value of friction coefficient. From figure intermittent peaks has been observed which means initiation of wear scars occurs. This is complex phenomenon and it lies upon material properties and contact friction condition. This can be attributed that the lubricant film becomes very thin and unable carry maximum load. Under this circumstance friction shoots up. Intermittent spikes are often seen in the entire period of the experimentations. During early stage of experimentation, sharp rise in friction also noticed due to static friction because inertia force. Under this situation the commercial oil is not capable of forming the adequate lubricant film to separate the mating surfaces of test specimens.

4.1.2 Influence of Speed

The effect of speed on the HLP(N) 46 tribo-performance has been examined in the range of 200 to 600 rpm. The COF variation for the entire experiment period is illustrated in Figure 4(a-e). The figure shows that COF increases as contact speed increases. In the case of wear behavior, it is found that the WSD increases with increased contact speed. Table 1 provides a comparative evaluation of COF and WSD at various speeds. As speed increases, the centrifugal force increases, which leads to the formation of a thin lubricating film and thus contacting bodies are not separated. This results in the observation of a higher COF value. WSD increases with speed increases were observed. At 200 rpm, increment of WSD and COF is 48.49% and 15.16%, respectively and at 600 rpm this increment is 62.68% and 21.12%. The friction changes with lower and higher speeds are noticeable, since it is governed primarily by the centrifugal force, but a substantial change is occurred in WSD at 200 and 600rpm. However, there exists a slight increase in WSD has been observed for TS7 than that of TS9, TS14 and TS11, especially at 150 N, 600rpm and 75. The results truly reveal that the area of contact is higher in case of TS11 and therefore larger wear is observed in this case. Wear rate of material increase with the increasing load parameters due to the generation of higher heat between the mating test specimens surface as a consequence of higher load[8].

4.1.3 Influence of temperature

Temperature influence on the tribo performance of HLP(N)46 has been investigated through variations in the lubricant temperature between 25°C and 75°C. Figure demonstrates the variation of COF for lubricated contacts at various temperatures over the entire duration of the experiment. From the figure it is noted that the COF rises marginally with temperature increase. In the same way, the WSD increases with an increase in the temperature in case of wear behavior. Table 1 provides a comparative

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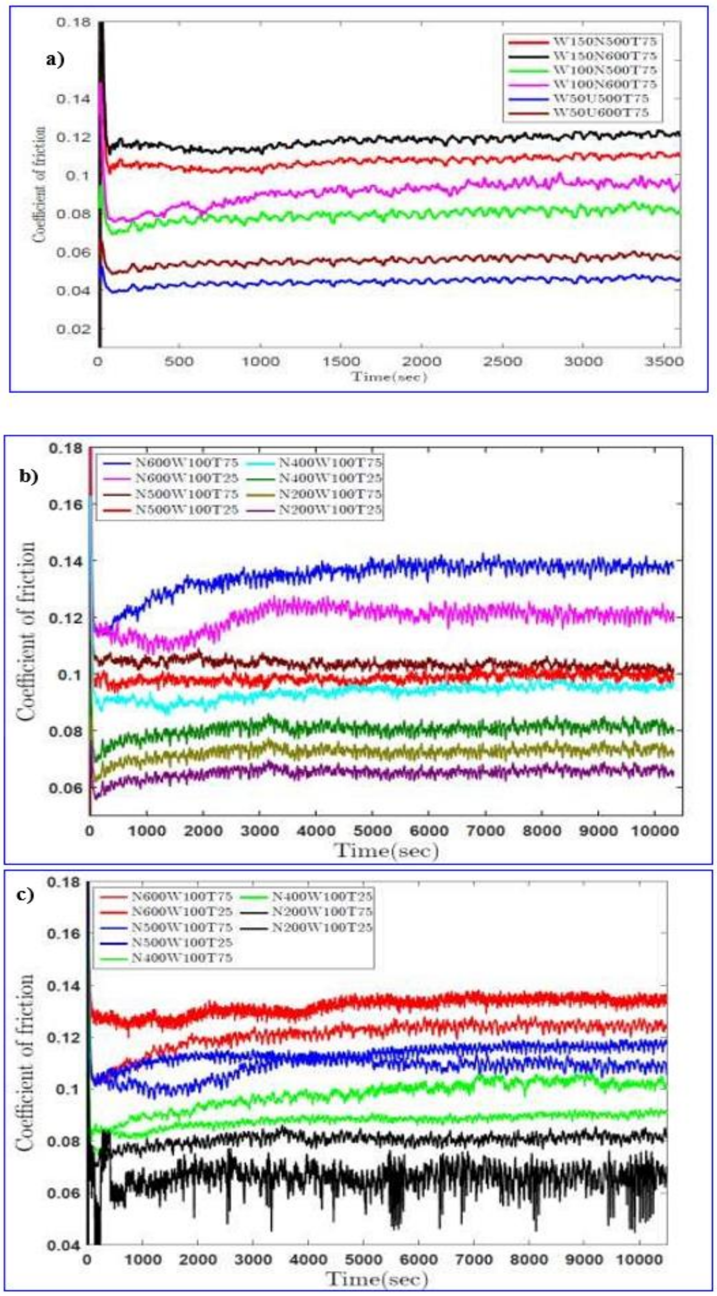
evaluation of COF and WSD for lubricated contact at various temperatures. The results indicate that the increase of the COF at 50°C and 75°C is 11% and 16% respectively. The WSD has shown that wear increases by a substantial 43.47% at 25°C, which is increased to 52.5% at 75°C. Wear and friction decreases marginally at a higher temperature by 3.7% and 9.7%, respectively and 3.6% and 8.4%, respectively at lower temperature.



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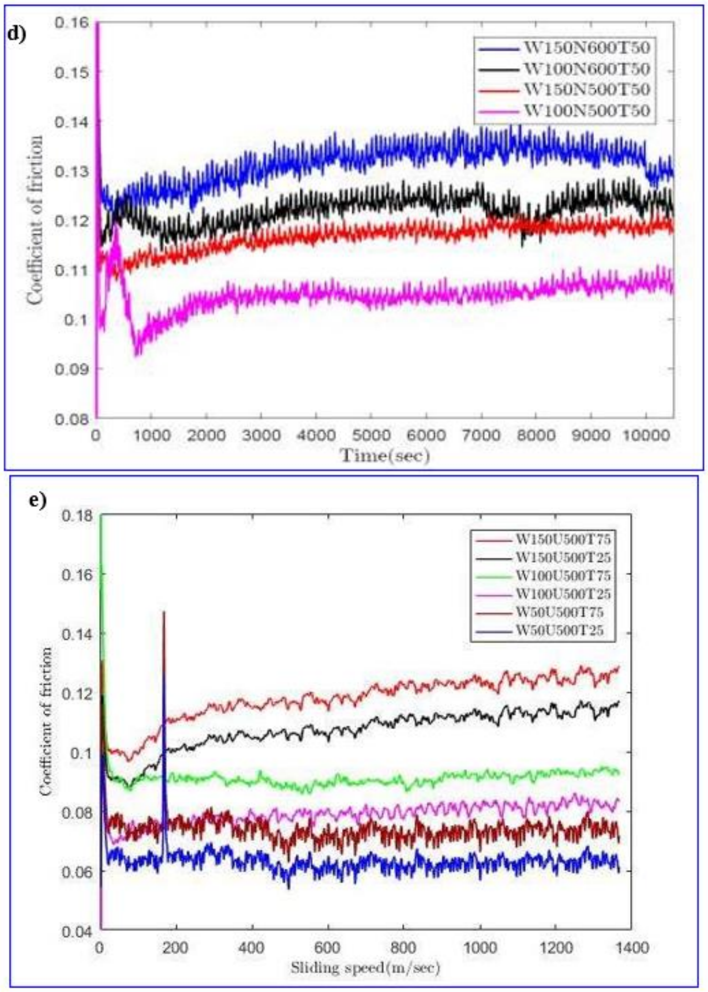

Figure 4: Influence of test conditions parameters on the friction behavior of commercial oil

4.2 Surface morphology

Micro-level inspections were carried out in the test specimens to detect the nature of the failure. SEM images of worn out test specimens is illustrated in Figure 5. The wear scars on the surface of worn out test specimens reveal marks of scratches along sliding/rolling direction with comparatively deep scratches at the center of contact surface. The scratching marks are observed on the surface of test specimens due to the ploughing action of asperities. At 150 N, 600 rpm and 75°C, the damage to surface is observed to reveal scoring marks that are parallel to sliding direction. In Figure 6, TS 9 and TS 10 indicates the minor marks of rubbing action in the sliding direction. Also, in Figure 5, TS2 and TS 4 signifies the rubbing type of wear with micropits. The wear process in the test specimens due to the gouging, plastic deformation and reinforcement particles will crush into very minute particles and form a very thin sub surface layer (designated as MML) between the work hardened pin and the counter face which is unable to withstand

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high stresses and increases the sliding wear. In Figure 5, TS 11, TS14 and TS7 depicted evidence of worn out surfaces due to typical subsurface micro cracking i.e. flaky debris resulting from higher intensity of elastic/plastic deformation. The wear track of worn out test specimen was in particular sliding grooves with typical reference to micro-ploughing. In Figure 5, TS11 shows smooth and featureless normal polishing wear with extended sliding pits and grooves appeared on the worn test specimen surface. Figure 5, TS1 and TS14 clearly indicates the small amount of surface delamination and formation wear grooves for the higher operating test condition. Figure 5, TS4, TS7, TS9 and TS11 shows numerous plowing grooves parallel to the sliding direction were seen on the wear tracks. SEM analysis indicates the pits on the worn out test specimens lubricated with commercial oil and reveal that pit size varying is more severity. The evolution of surface spalling or flaking test specimens has been caused a significant increase in friction, with the corresponding temperature effects in test specimens. The isolated pits form at the beginning of the flaking and these joints are quickly created in bands of pit and the samples are gradually destroyed as illustrated in Figure 5. There is further deterioration due to the load on the edges of the pits. In addition, the wear rate and temperature are increases owing to the metal particles in the lubricant. The rise in temperature further deteriorates the characteristics of the lubricant resulting in the failure being accelerated.

Figure 6 depicted the wear scar diameter differences after the rolling test of the lubricant. The anti-wear characteristics of lubricants are generally assessed according to the wsd, as the larger the wear scars the more serious it is. The wsd of TS2, TS3 TS4 and TS7 are larger than TS11 and TS14.At operating condition 50N,200rpm and 25°C wear rate of material is less. The WSD increases with a decrease in the oil film at higher load and speed.

Taylor Hobson Talysurf Series 2 stylus instrument is used for identifying the surface topography of the worn out test specimens. 3D Surface topography done on different worn out test specimens are illustrated in the Figure 7. The surface topography of the worn out test specimen is quantified on the basis of minimum width of the circular wear tracks. The profilometry examination of the worn out test specimen reveal the wear rate keeps on increasing with increase in the test conditions parameters. Quantitative analysis of surface irregularities is done by measuring roughness values using profilometer.

The surface roughness parameters $S_q, S_a, S_{sk}, S_{ku}, S_p, S_v$ and $S_z$are obtained from the study as mentioned in the Table 4 . Among the different surface roughness parameters, S_a is the most frequently noted parameter which is also a poor indicator of surface roughness [8]. It has been found that when S_a was lower than 1μm the surfaces were less wear reduction exhibit in the materials. Considered that all $S_a$values obtained in the current study were above 1.25μm and below 3.31μm, which presents wear and friction increases with increased roughness of the worn out test specimen surfaces. It was therefore inferred that these contact surfaces would experience higher wear and friction, resulting in failure of components.

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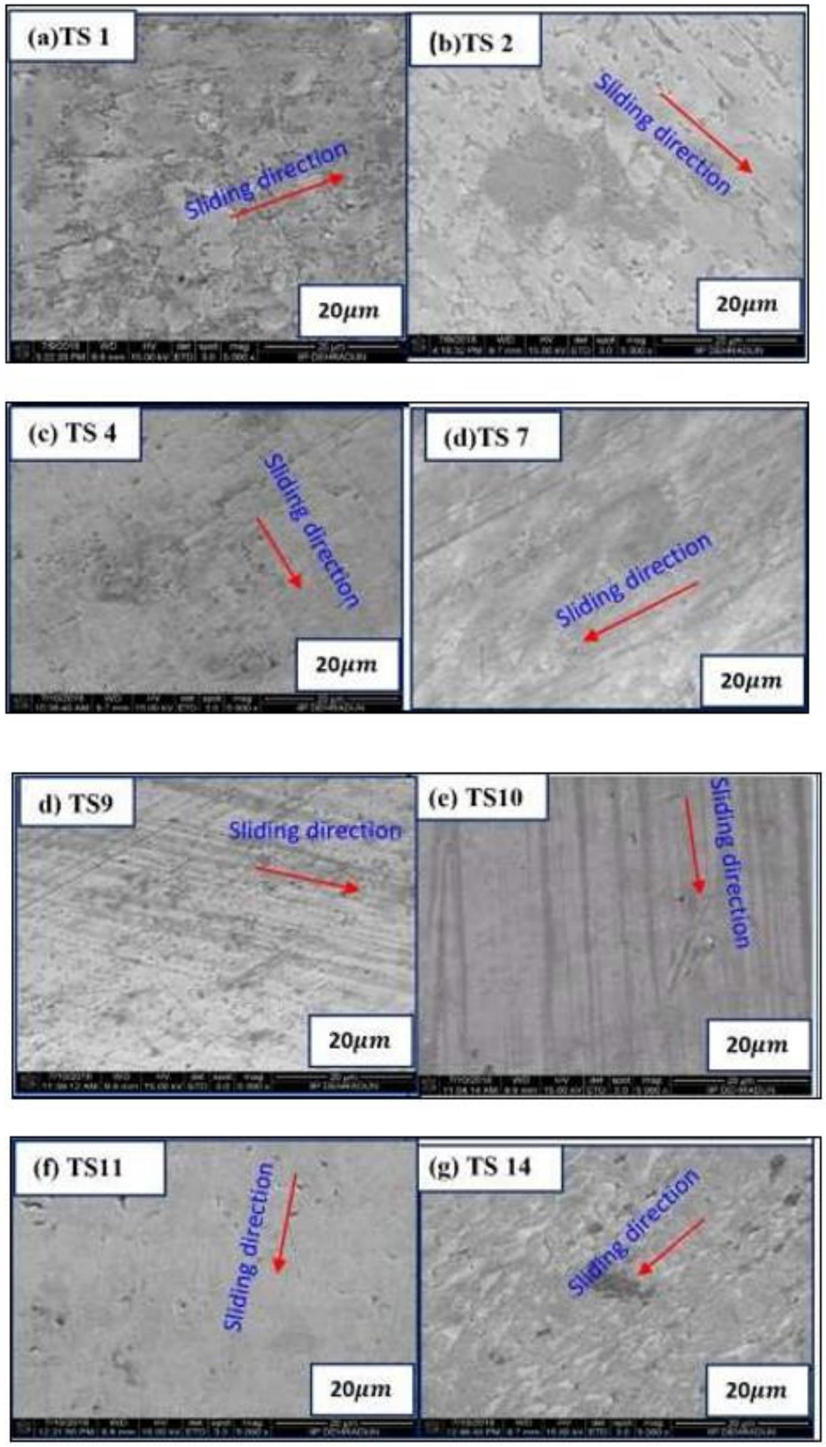

Figure 5: SEM images of worn out test specimens

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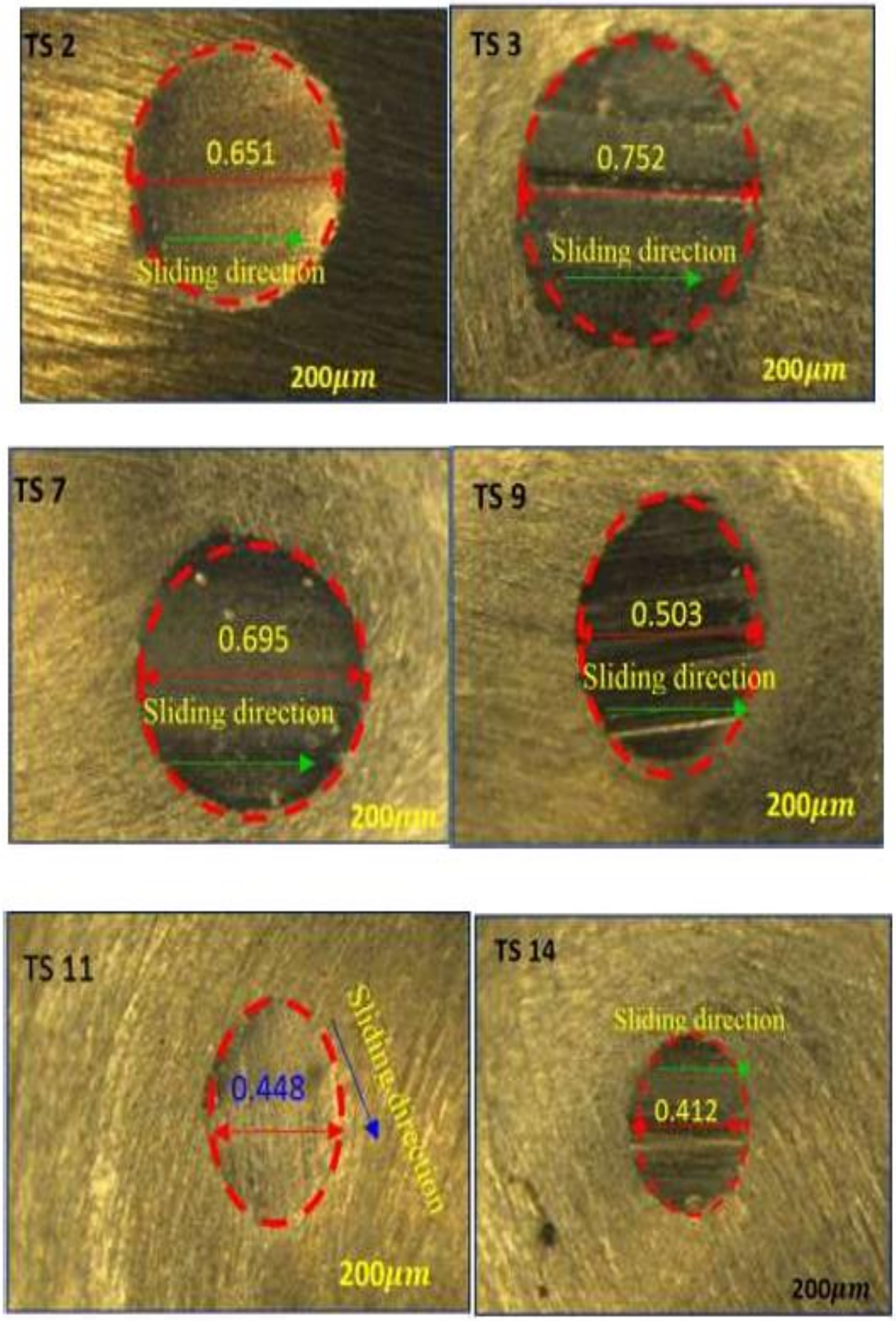

Figure 6: Optical images of worn test sample surfaces

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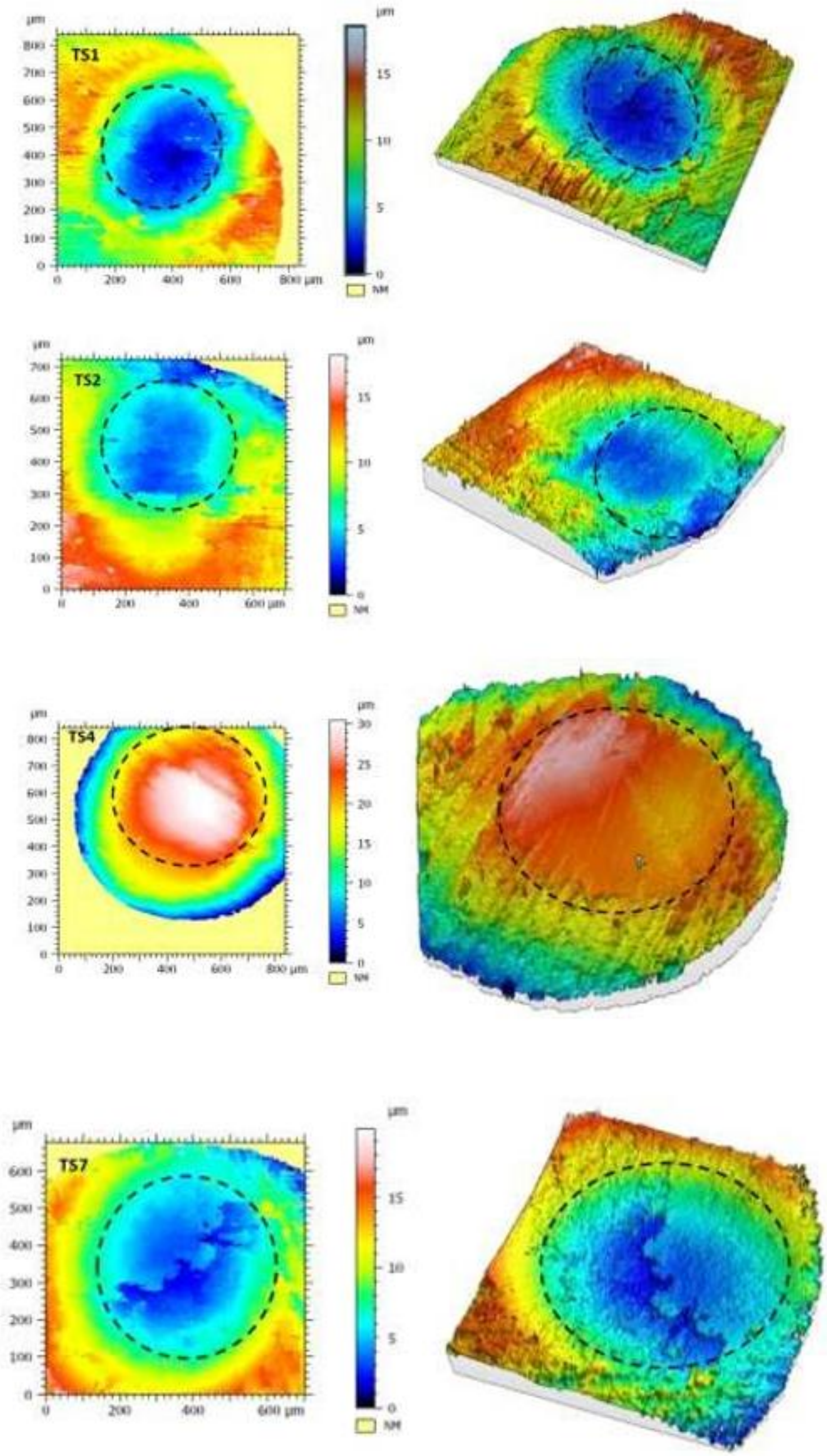

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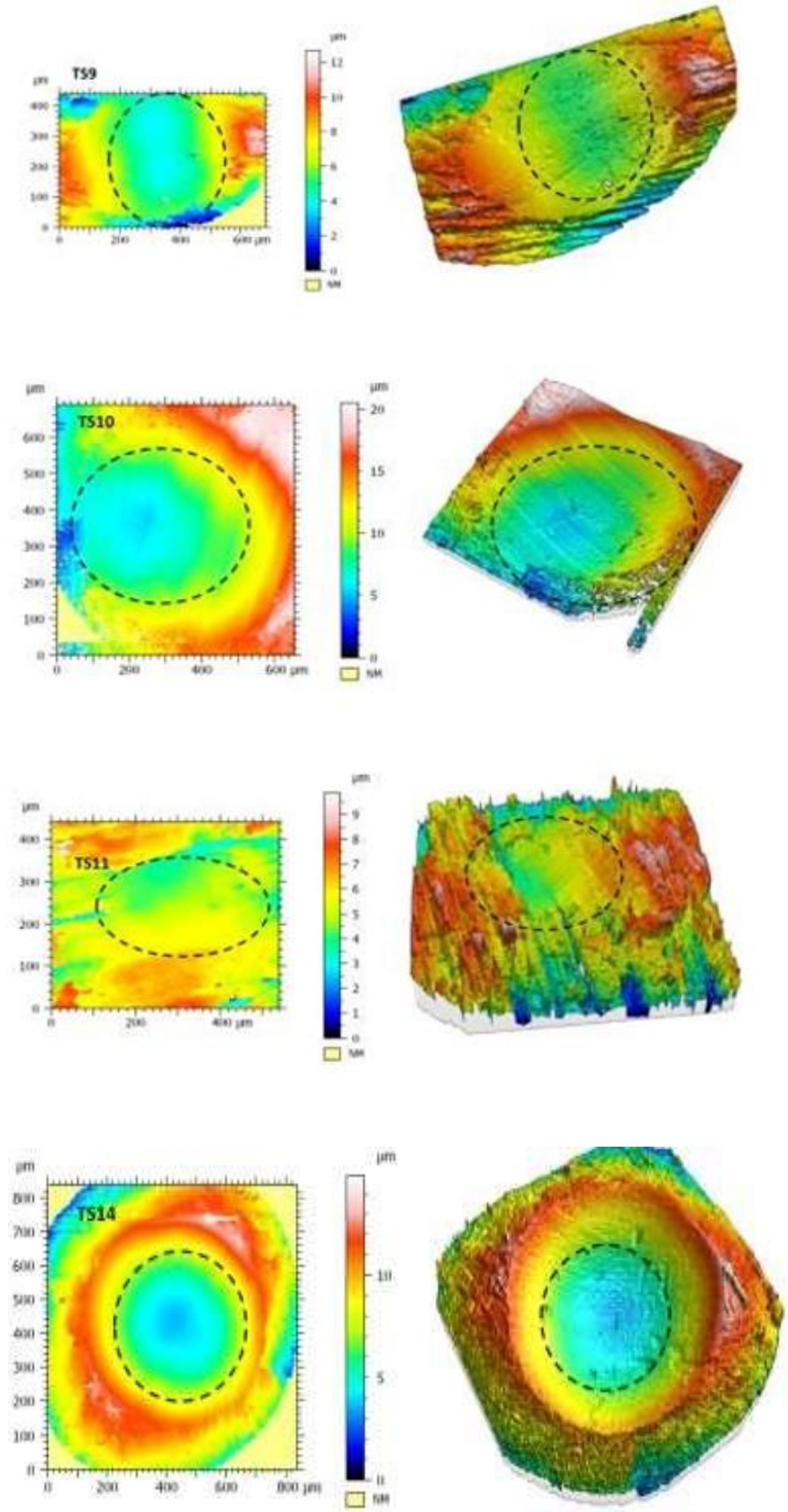

Figure 7: 3D surface profile of the used test specimen surfaces

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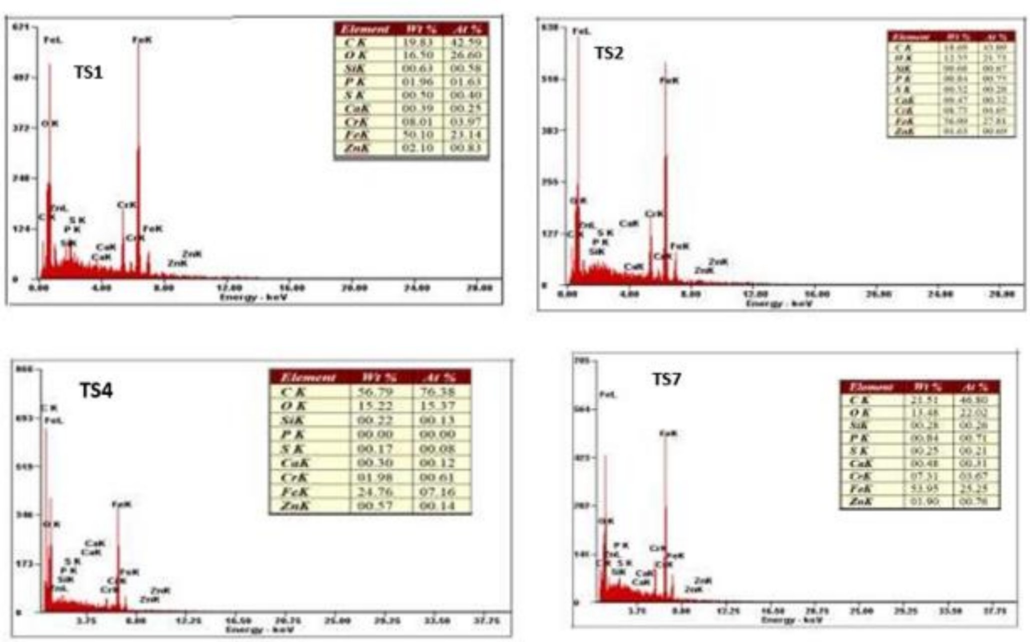
Table 4: Surface roughness parameters of used test specimens

No. of expt.	$S_q(\mu m)$	$S_a(\mu m)$	$S_{sk}(\mu m)$	$S_{ku}(\mu m)$	$S_p(\mu m)$	$S_v(\mu m)$	$S_z(\mu m)$
1	3.51	2.94	-0.128	2.11	10.4	8.12	18.5
2	3.63	3.13	0.151	1.88	9.76	8.43	18.2
3	2.00	1.64	0.36	2.73	6.64	6.03	12.7
4	2.95	2.43	-0.584	2.72	6.96	10.6	17.6
5	3.56	3.02	0.327	2.13	11.9	8.02	19.9
6	3.92	3.31	0.407	2.2	9.61	10.9	20.5
7	1.56	1.25	-0.343	3.03	4.32	5.54	9.85
8	2.40	2.01	-0.233	2.23	6.57	8.29	14.9

4.3 EDX analysis

The EDX analysis of the worn out test specimens surface reveals the presence of compositions of the worn material such as CK, OK, FeK SiK, PK, CaK, CrK indicating the formation of thin boundary films. Due to presence of inadequate boundary film, the worn out test surface shows more wear and causes higher value of friction. Composition of lubricated worn out test specimen has been revealed through the EDX/EDS technique is presented in the Figure 8. CK, OK, FeK SiK, PK, CaK, CrK are chemically corrosive additives that have a strong affinity for the contacting surface and form thick films which prevents metal to metal contact. EDX pattern confirms the presence of CK, OK, FeK SiK, PK, CaK, CrK nanoparticles. The presence of CaK, CrK nano-particles increase the polarity on the surface of test specimen due to absorption of moisture from the environment resulting into increase of settling rate of these nano-particles in the lubricating oil Therefore, the rate of settling of these particles can be reduced by enhancing the dispersion stability[23]. OK and FeK were found, mainly due to the formation of iron oxide in the test sample.



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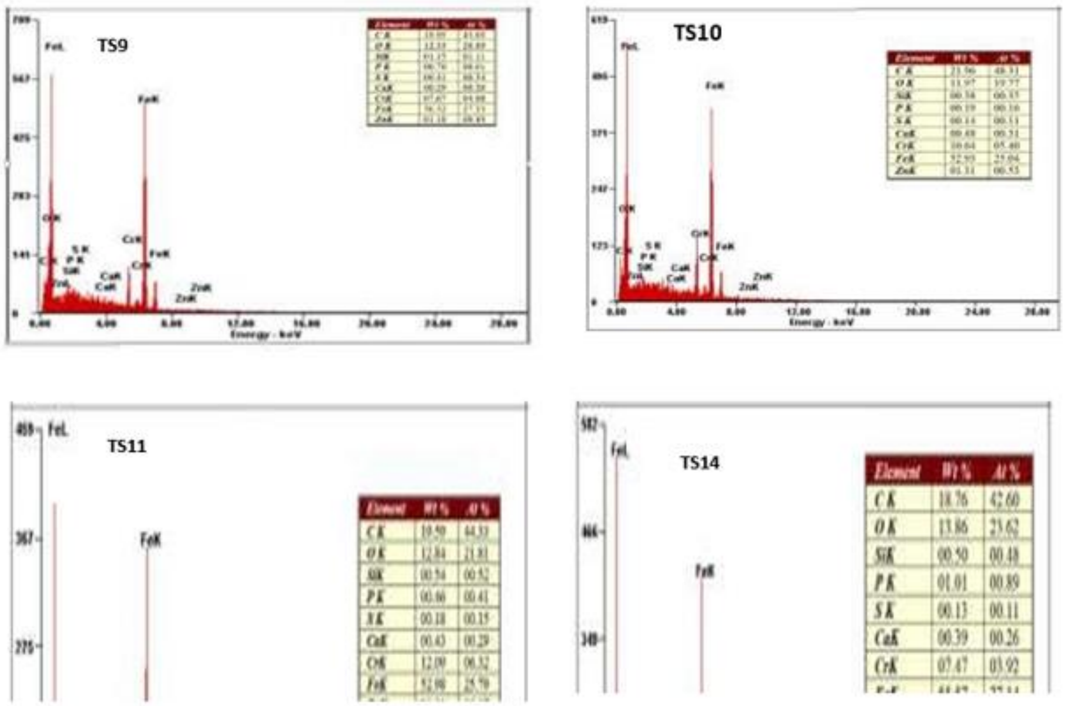

Figure 8: EDX analysis of different worn test specimen surfaces after pin-on-roller test

4.4 Statistical analysis

Statistical ANOVA technique is used to study different design parameter which substantially affects the wear characteristic. This technique is used to predict the optimal combinations of different process parameters. Using this analysis, the level of confidence and level of significance is maintained at 99% and 1%, respectively [21]. Effect of (i) load, (ii) speed and (iii) temperature on COF and wear of test specimens has been investigated through the ANOVA analysis. Results obtained from ANOVA analysis of test specimens are presented in the Tables 5 and 6. The different combination of above factors have substantial effect on wear and friction coefficient of the worn out test specimens. The percentage contribution(p)of each shown in column 5 of Table5. It can be seen from Table 5 the linear interaction of load (p=57.4%), temperature p= (40.9%) and speed (p=37. 9%).Further, square interaction of load (p=58%), speed (p=61.2%) and temperature (p=63.9%) and 2-Way Interaction load and speed (p=74.3%), load and temperature p= (52.7%) and speed and temperature (p=96. 2%).From Table 6 that the linear interaction of load (p=70.1%), temperature p= (40.9%) and speed (p=39. 3%) can be noticed. Further, square interaction of load (p=53.55%), speed (p=27.3%) and temperature (p=61.2%) and 2-Way Interaction load and speed (p=65.5%), load and temperature p= (71.59%) and speed and temperature (p=62.2.3%). ANOVA analysis in Table 5 and 6 designates the p- value of each factor on the total variation showing their degree of effect on the result.

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It can be clearly observed from Figure 9, that the results obtained from the tests that the WSD increases with increase in test condition parameters (W, U and T). The surface plots used to represent increment in the WSD with different combination of operating test condition parameters. It can be noticed that the wear of worn out specimens increases with increase in temperature of the commercial oil. With the increase in the temperature of the lubricant film, the viscosity of the lubricant becomes lower resulting in thin formation of boundary film which may have detrimental effect on tribo-performance. From Figure 10, it can be seen that the rise in COF with the test conditions operating parameters. It can be clearly noticed from obtained results of different test conditions, indicates that the COF increases. Also, it can be seen from the Figure 10 that at higher load and commercial oil are incapable to produce thick lubricant film and which results in sudden rise in friction has been seen at early condition of test. The frictional heat generated is increase with increases lubricant temperature. In these circumstances, the chances of direct metal-to-metal contact is primarily high. The increase in friction and wear rate of material is more due to the thin hydrodynamic lubricant film occur at the vicinity of worn out test surfaces. This can be attributed that the static friction, which is affected due to inertia forces, operating test condition and lubrication conditions.

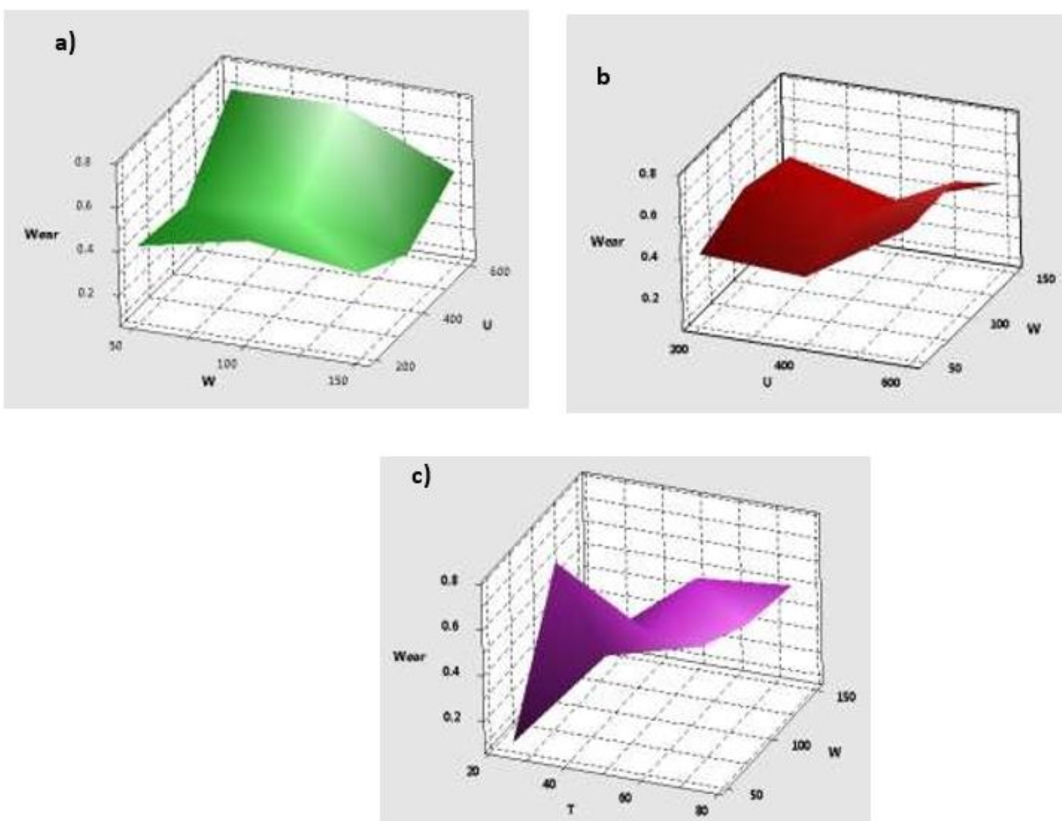

Figure 9: Surface plot for wear at different combination of operating conditions

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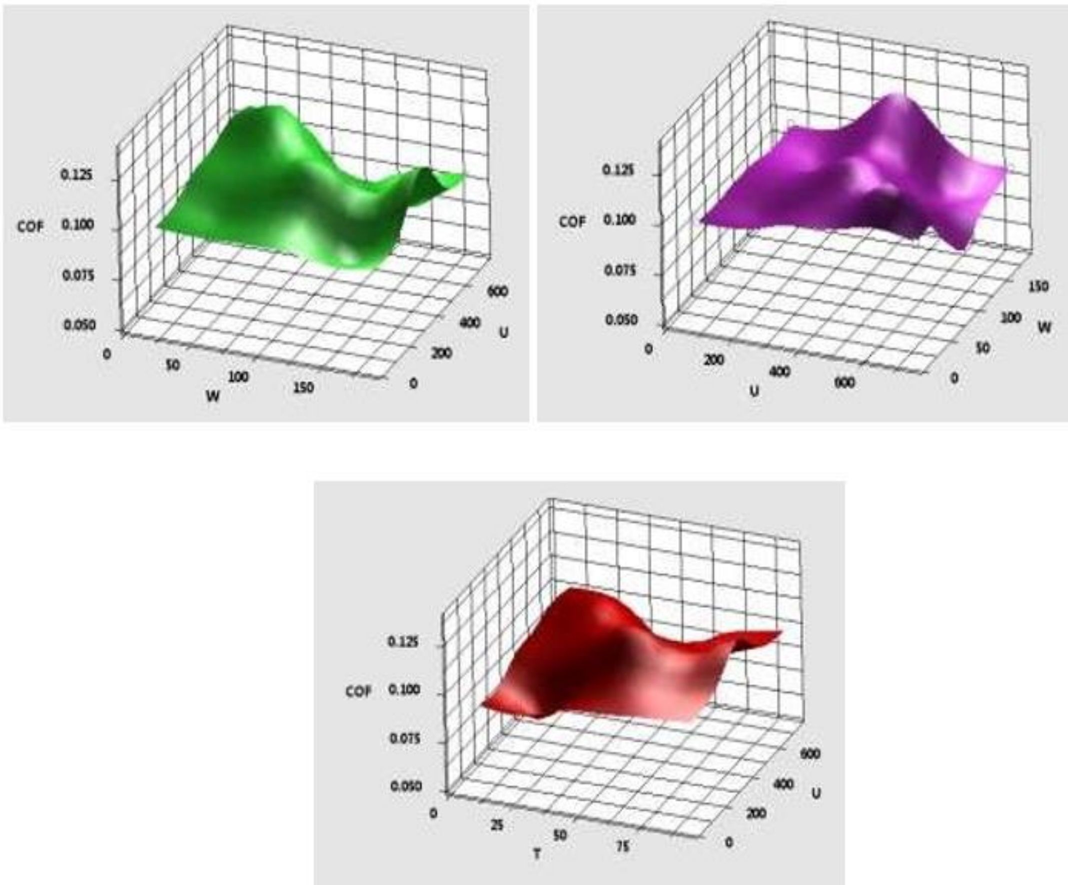

Figure 10: Surface plot for coefficient of friction at different combination of operating conditions

Table 5: Analysis of Variance (ANVOA) for friction

Source	DOF	Adj. SS	Adj.MS	F- Value	P- Value
Model	9	0.003821	0.000425	0.94	0.559
Linear	3	0.000582	0.000194	0.43	0.74
W	1	0.000163	0.000163	0.36	0.574
U	1	0.000418	0.000418	0.93	0.379
T	1	0.05633	0.05633	0.81	0.409
Square	3	0.002976	0.000992	2.21	0.206
W^2	1	0.002689	0.002689	5.98	0.58
U^2	1	0.000132	0.000132	0.29	0.612
T^2	1	0.000112	0.000112	0.25	0.639
2-Way interaction	3	0.000263	0.000088	0.19	0.896
$W \times U$	1	0.000054	0.000054	0.12	0.743
$W \times T$	1	0.000207	0.000207	0.46	0.527
$U \times T$	1	0.000001	0.000001	0	0.962
Error	5	0.002249	0.000450		

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Lack-of-Fit	3	0.001251	0.000417	0.84	0.585
Pure Error	2	0.000999	0.000499		
Total	14	0.00607			

Table 6: Analysis of Variance (ANVOA) for wear

Source	DOF	Adj. SS	Adj. MS	F- Value	P- Value
Model	9	0.319407	0.03549	0.51	0.82
Linear	3	0.128571	0.042857	0.62	0.633
W	1	0.011514	0.011514	0.17	0.701
U	1	0.060726	0.060726	0.87	0.393
T	1	0.05633	0.05633	0.81	0.409
Square	3	0.145979	0.04866	0.7	0.591
W^2	1	0.030831	0.030831	0.44	0.535
U^2	1	0.105388	0.105388	1.52	0.273
T^2	1	0.000477	0.000477	0.01	0.937
2-Way Interaction	3	0.044857	0.014952	0.22	0.882
$W \times U$	1	0.015625	0.015625	0.22	0.655
$W \times T$	1	0.01005	0.01005	0.14	0.719
$U \times T$	1	0.019182	0.019182	0.28	0.622
Error	5	0.347348	0.06947		
Lack-of-Fit	3	0.178868	0.059623	0.71	0.63
Pure Error	2	0.16848	0.08424		
Total	14	0.666756			

5.CONCLUSIONS

The tribological performance of commercial oil in elliptical contact geometry of rotors/rollers contact in the screw compressor has been studied in EHL regime. From this study of rotors/rollers contacts in the screw compressor has been indicated that the decrease in film thickness with increase in test condition parameters.

The viscosity of the lubricant becomes lower at elevated temperature resulting into thin lubricant films on the rotor mesh which in turn results in detrimental effect on the performance of rotors/rollers contact.

Consequently, at elevated temperature, the reaction kinematics is influenced resulting in decrease of adhesive forces and increase in the chemical activity.

The interaction of temperature, speed and load factor have a substantial effect on the tribo performance characteristics of tribo pair. The anti-friction and anti-wear behavior of commercial oil is influenced due to inadequate lubricant film in mating test specimen, revealed from the EDX and SEM analysis.

Surface morphology of worn out test specimens revealed the severity of failures. Commercial oil lubricated test specimens surface reported that the elasto- plastic yielding of the asperities and scratches are visible.

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Nomenclature

ANOVA	Analysis of variance
DF	Degrees of freedom
EDX	Energy dispersive X-ray spectroscopy
MML	Mechanically mixed layer
P	Percentage of contribution
S_a	Arithmetical mean height
S_{sk}	Skewness
S_{ku}	Kurtosis
S_p	Maximum profile peak height
S_q	Root mean square deviation
S_v	Maximum profile valley depth
SS	Sum of squares
SEM	Scanning Electron Microscope
μ	Coefficient of friction
WSD	Wear scar diameter

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