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Shaft Surface Roughness and its Influence on the Wear of the Lip Sealing Edge

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A paper presents first stage of the research on the role of the shaft surface roughness in lubrication and wear of the oil lip seals. Surface roughness of the shafts was characterized by means of the selected roughness parameters. Short term tests of the oil lip seals revealed that torque as well as the oil lip wear are greatly affected by the ratio of the profile height and average radius of curvature of all shaft irregularities, ordinates distribution across the profile height as well as by mean slope of the profile. It was found that some of the oil lip seals had uneven wear track over lip perimeter. These seals had lower power consumption than those with symmetrical lip wear. The lack of the symmetry in contact conditions over the lip perimeter renders the relation between the shaft surface roughness and the lip wear to be not so distinct.

Keywords: oil lip seal, roughness, wear, torque.

1. INTRODUCTION

It is well known nowadays that the shaft surface roughness affects the wear, tightness as well as frictional losses of the oil lip seals. Symons [1] found that the oil lip seals operating on the plunge ground shafts had three times higher power consumption than those running on the peened shafts with steel balls. Hence, Symons recommended the shaft surfaces with big radii of the irregularities curvature and with the small side inclination.

Qu [2] investigated the influence of the shaft surface roughness on the leakage; he found that the oil lip seals were tight then, when the chosen surface roughness parameters had the following values: skewness $R_{sk} > 0.4$; kurtosis $R_{ku} < 4.0$; reduced depth of the valleys $R_v < 3.8$ μm and the roughness average $0.25 \leq R_a \leq 0.5$ μm. However, Horve reported [3] that the oil lip seals with the smooth surface of the sealing edge were more often leaking than those with rough surface. It follows, that the shaft surface irregularities should not only create but also maintain the proper texture of the sealing edge surface during oil lip seal operation. Following this thesis Gawliński found [4] that in the one and the same oil lip seal operating on the plunge ground shaft, one can meet the lip sectors with and without the traces of the wear (Fig. 1).

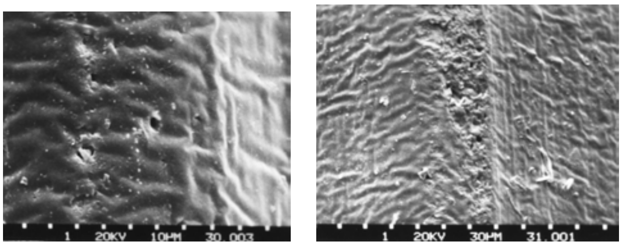

Fig. 1. An outlook of the sealing edge sectors under ESM in the oil lip seal operating on the plunge ground shaft through 200 hours, a – unworn surface, b – worn surface

The measurements have revealed that the relative position (expressed in terms of the contact width of the lip and the shaft) of the contact region along the sealing edge can vary over the lip perimeter within two or three contact widths.

This variation of the contact position along the sealing edge affects the contact pressure and hence, the wear rate of the lip. Therefore it is of practical interest to choose such shaft surface roughness which can ensure expected texture of the sealing edge in spite of variation of the contact pressure in the circumferential direction.

2. OBJECT OF INVESTIGATIONS AND TEST METHODS

The object of investigations were oil lip seals with ID equal to $d = 43$ mm. Mean radial force amounted to $P_r = 34.2$ N and its standard deviation $\sigma = 1.5$ N. Mean value of the contact width of the lip with the shaft was $a = 0.07$ mm while its standard deviation $\sigma = 0.01$ mm.

There were eight steel shafts with hardened external layer; each of them was plunge ground. The shafts were so ground in order to get different surface profiles. The surface roughness was measured by means of Taylor Hobson profilometer Talysurf laser 3D in three places over the shaft perimeter and on two levels (heights) of the shaft. All roughness parameters of the original surfaces are presented in Table 1.

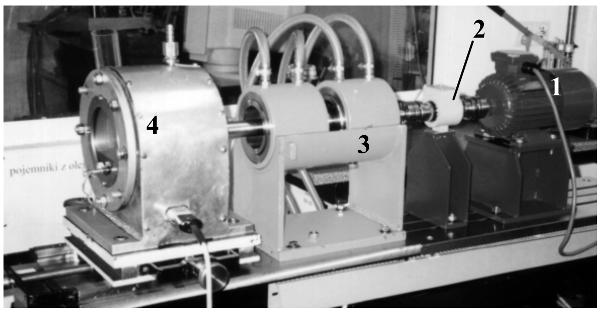

Fig. 2. Test rig for power consumption evaluation, 1 – electric motor, 2 – torquemeter, 3 – areostatic bearings 4 – test chamber

The same profilometer was used to map the profile of the examined oil lip seals. The profiles of both original and worn sealing edges of the lips have been determined at two

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places over lip perimeter. Comparison of the profiles before and after the tests makes easy an estimation of the wear rate of the lip.

The oil lip seals were run on the test rig presented in Fig. 2. The shaft of the test rig is supported in two aero-static bearings; it is driven by an electric motor with the angular speed up to 1000 rad/s. The torque was measured with aid of the inductive torquemeter with the measuring range (0 – 2) Nm. Time of the oil lip seal operation was 20 hours; the fluid to be sealed was mineral oil SAE 30 at temperature of 60 °C.

3. CHARACTERISTIC OF SHAFT SURFACE ROUGHNESS

The following parameters have been chosen to characterize the surface roughness of the shaft to be tested: R_a, S_m, Δ_a, λ_a, R_{sk} and R_{ku}. Average roughness R_a parameter was used because of its common application. Parameter S_m characterizing mean peak spacing along the mean line of profile influences tangential displacement of the lip; moreover, it is believed that it also influences vibrations of the external layer of the lip. Average profile slope Δ_a changes exactly its value with the wear process as well as it is a good measure of the surface roughness anisotropy. Average wavelength λ_a of the surface profile takes into account the wavelength and peak-to-valley height. It can be calculated as follows: $\lambda_a = 2\pi \frac{R_a}{\Delta_a}$; one can assume that it affects lubrication condition at lip-shaft interface.

Statistical parameters: skewness R_{sk} can be treated as the asymmetry coefficient of the ordinates distribution across the profile height while kurtosis R_{ku} is the measure of the sharpness of the amplitude distribution curve. Both these parameters influence the real contact conditions between rubbing surfaces.

Furthermore, the authors of this paper used an average radius $\bar{r}$ of all irregularities which has been calculated by program Profile specially prepared to investigate the surface profiles. This radius is the geometrical mean of the radii determined in axial r_A and circumferential r_C directions:

$$\bar{r} = \sqrt{r_A \cdot r_C}\ . \tag{1}$$

The knowledge of the average radius $\bar{r}$ makes easy calculation of the ratio $\frac{R_t}{\bar{r}}$ being used in tribology as a measure of stress resulting from penetration of the metal irregularity into the rubber surface. Parameter R_t is the height of the surface profile. Program Profile allows to determine also the approximation constants ν and b of the upper part of the bearing ratio curve:

$$\eta = b \cdot \varepsilon^{\nu}\ , \tag{2}$$

where η is the material ratio at given relative approach ε to the profile.

The constants ν and b have been used to find so-called critical relative approach ε_{crit} [5] at which all metal irregularities contact the surface of the sealing edge of the lip. It can be calculated using following equation:

$$\varepsilon_{crit} = \frac{1}{(b \cdot \nu)^{1/\nu - 1}}\ . \tag{3}$$

Table 1. Mean values of the original shaft surface roughness parameters determined in axial (A) and circumferential (C) directions.

Shaft number		Roughness parameters								
		R_a, µm	S_m, µm	Δ_a, deg	λ_a, µm/deg	R_{sk}	R_{ku}	$\bar{r}$, µm	$R_t/\bar{r}$	ε_{crit}
1.2	A	0.47	54	8.4	0.4	–0.28	3.1	9	0.4	0.27
	C	0.24	498	0.9	1.9	–0.6÷+0.3	3.4		0.23	0.26
2.2	A	1.10	61	10.5	0.7	–0.48	3.1	6	1.4	0.20
	C	0.37	495	1.2	1.1	–0.37÷+0.50	3.4		0.5	0.80
3.2	A	0.51	58	8.7	0.4	–0.41	3.2	16	0.27	0.35
	C	0.27	330	0.9	2.2	+0.16	3.0		0.12	0.44
4.2	A	0.26	65	4.5	0.4	–1.87	8.6	19	0.15	0.10
	C	0.17	305	0.6	1.7	–0.69÷+0.14	3.2		0.07	0.26
5.2	A	0.43	83	3.1	0.86	–1.21	4.4	26	0.12	0.12
	C	0.13	198	1.2	0.74	–1.4	7.9		0.07	0.34
6.2	A	0.46	157	3.0	1.0	-0.74	3.3	19	0.19	0.10
	C	0.11	262	1.1	0.71	–3.63	25.3		0.11	0.27
7.2	A	0.49	34	10.1	0.32	+0.31	2.3	17	0.17	0.50
	C	0.08	797	0.5	1.1	–1.9÷+1.8	17.4		0.10	0.27
9.2	A	0.36	76	5.3	0.5	–0.49÷+0.66	3.6	14	0.24	0.14
	C	0.11	148	0.9	0.9	–0.14÷+0.53	4.3		0.08	0.28

Analysis of the Table 1 contents allows to draw following conclusions concerning the surface profiles determined in axial (A) and circumferential (C) directions:

1. There is high rate of the surface anisotropy especially with respect to R_a , S_m , Δ_a and $\frac{R_t}{\bar{r}}$ parameters;

2. Unexpectedly, skewness R_{sk} determined in circumferential direction has, in the case of some shafts, negative values. It means that the most of the profile ordinates is situated above mean line. There are some shafts (e.g. No: 5.2; 6.2) where skewness in axial direction is bigger than in circumferential one;

3. Non-typical, as for plunge ground surfaces, are the values of kurtosis being practically the same in both directions for the surface profiles of the shafts No: 1.2; 2.2; 3.2 while for the shafts surface No: 5.2; 6.2 and 7.2 kurtosis in circumferential direction is far bigger than in axial one;

4. The mean values of the radii $\bar{r}$ of all irregularities are small in the case of all shafts what leads to the relatively high values of $\frac{R_t}{\bar{r}}$ ratio.

4. RELATION BETWEEN TORQUE AND SHAFT SURFACE ROUGHNESS

There was geometrical and material similarity of the oil lip seals to be examined. Therefore, one can assume that the torque will be affected by the shaft roughness as well as by the contact conditions at lip-shaft interface. Speaking about contact conditions one means, first of all, position of the contact area along the sealing edge of the lip as well as its position with respect to the shaft surface. The torque values recorded at the beginning of the oil lip seals operation on the particular shafts were following in Table 2.

Table 2. The torque values recorded

Shaft number	1.2	2.2	3.2	4.2	5.2	6.2	7.2	9.2
Torque; Nm	0.36	0.27	0.23	0.39	0.38	0.38	0.24	0.36

It is typical that high torque values have those seals which operate with the shafts surface having relatively small profile slope Δ_a in axial direction and low critical relative approach ε_{crit} . This latter means that there is high probability of the dense contact between the lip and shaft even at low contact pressure.

Time of twenty hours of the continuous operation was sufficient to change the surface texture of the lip as well as the lip profile due to the war process. It resulted in the change of the contact pressure as well as of the lubrication conditions between the lip and the shaft. The shaft surface roughness, on the other hand, experienced only slight changes in that time interval. The torque values recorded after 20 hours of operation at constant speed of $n = 1000$ rpm are presented in Fig. 3 and Fig. 4; torque is determined here as a function of the shaft speed. Comparison of the torque values at the beginning and at the end of the tests shows that the frictional losses decreased irrespective of the shafts surface roughness; torque decreased by factor of two at the speed of 1000 rpm.

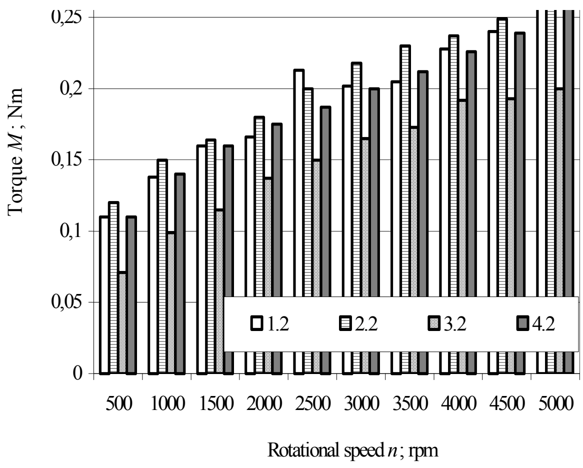

Fig. 3. Torque dependence on the shaft speed after 20 hours of continuous operation

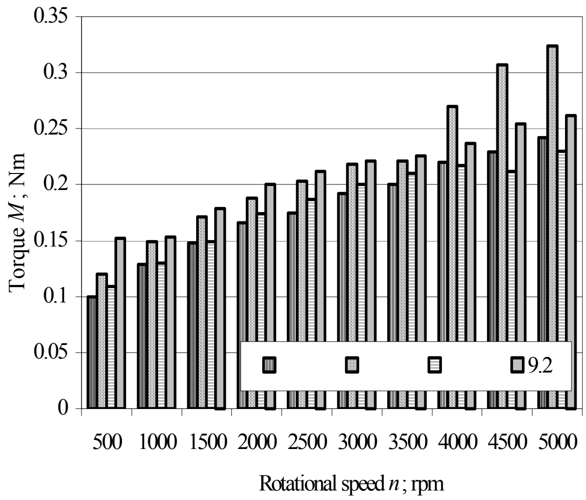

Fig. 4. Torque dependence on the shaft speed after 20 hours of continuous operation

It seems to be a bad surface feature when high ratio $\frac{R_t}{\bar{r}}$ is combined with mean value of the critical approach ε_{crit} of the rubbing surfaces. Seals running on the shafts 1.2; 2.2 show the highest values of the torque with increase of the shaft speed. The same effect can be met when the seal operates on the shafts with very small skewness R_{sk} and high kurtosis (shaft No: 6.2). High value of the critical approach ε_{crit} when combined with the low value of R_t/r ratio seems to be advantageous with respect to the friction losses; seals operating on the shafts 3.2; 5.2 and 7.2 distinguished themselves with low power consumption. However, one should keep in mind that the torque is affected by the contact conditions between the lip and the shaft, too. Therefore, the final conclusions about the surface roughness influence on the torque should follow an analysis of the sealing edge wear over the lip perimeter.

5. RELATION BETWEEN SHAFT ROUGHNESS AND LIP WEAR

Determination of the lip wear at two places over the lip perimeter is an efficient way to estimate the contact conditions existing at lip-shaft interface. In the whole population of the examined seals one could meet the seals with symmetrical as well as with asymmetrical wear trace over lip perimeter (Fig. 5). Asymmetrical worn profile manifests variation of the contact conditions in the circumferential direction, e.g. variation of the penetration depth of the shaft surface irregularities into lip surface.

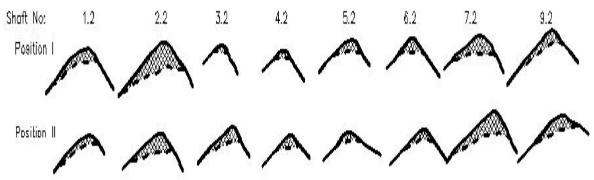

Fig. 5. Profiles of the sealing edge before (solid line) and after (broken line) 20 hours of the continuous oil lip seals operation on different shafts

Comparison of the lip profiles taken before and after the seal operation with corresponding torque shows the following regularities:

- the lowest torque usually corresponds to the asymmetrical (ununiform) lip wear (vide seals rubbing against the shafts No: 3.2; 5.2),
- higher torque values corresponds to the symmetrical (uniform) lip wear (seals rubbing against the shafts No: 2.2; 4.2; 6.2; 9.2).

The lack of the symmetry of the contact conditions in the circumferential direction takes responsibility that there is not clear relation between the shaft surface roughness and the lip wear rate. In general, high $\frac{R_t}{\bar{r}}$ ratio affects the lip wear (those running on 2.2; 7.2 shafts) especially then, when combined with high value of relative approach ε_{crit}. Low values of the skewness and high values of the kurtosis indicate the surface profile with big number of the ordinates situated above mean profile line. The oil lip seals rubbing against such surfaces of the shafts (e.g. 5.2; 6.2 shaft) experience low wear rate.

6. CONCLUSIONS

The present stage of the tests directed on the search of the optimal roughness of the shaft surface can be summarized with the following conclusions:

1. High torque values correspond to the shaft surfaces with low slope Δ_a and low values of the relative critical approach ε_{crit} ;
2. The efficient way to reduce power consumption is to lower considerably the ratio $\frac{R_t}{\bar{r}}$, to enlarge the value of the skew ness and to lower kurtosis value;
3. The lip wear leads to the significant reduction of power consumption;
4. The lack of the symmetry in the contact conditions in the circumferential direction results in lower power consumption;
5. High $\frac{R_t}{\bar{r}}$ ratio combined with high value of the relative critical approach ε_{crit} leads to the high wear rate of the lip.

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