POLISH MARITIME RESEARCH 3 (103) 2019 Vol. 26; pp. 39-46
10.2478/pomr-2019-0044

DIAGNOSTIC MODEL OF CRANKSHAFT SEALS

Piotr Bzura
Gdańsk University of Technology, Poland

ABSTRACT

The paper presents a research stand being a diagnostic model of radial lip seals used, among others, on crankshafts of piston combustion engines in order to identify the correctness of their operation. The possibility of determining the technical condition of lip seals on the basis of the proposed coefficient of correctness of operation has been described. The basic features of seals influencing their correctness of operation were also described, along with examples of determining the durability limits of lip seals. A modified version of the friction node of the T-02 four-ball apparatus is presented. It allows to check the correctness of sealing lips operation as well as to test the compatibility between the steel shaft, sealing lip and sealed lubricating oil. It was shown that the test results con-firmed the usefulness of the hypothesis that the quality of oil affects the durability of sealing lips and their coefficient of correctness. Additionally, attention was paid to the possibility of analyzing the pumping effect affecting the transition of the seal-shaft system from the state of partial suitability S_2 to the state of full suitability S_1 or to the state of unfitness S_3, and because the change in the state of such a system is random, it requires a probabilistic analysis.

Keywords: marine gas turbine, inlet air fogging, applicability

INTRODUCTION

Dynamic crankshaft seals of internal combustion piston engines work properly when there is sufficient lip-to-shaft pressure to ensure permanent compression and maintenance of the thin-layer grease between lip and shaft. In addition, the lip pressure is supported by the oil pumping effect, which is the result of the interaction between the deformed lip and the rotating shaft. Thus, it can be concluded that this tribological system of any internal combustion piston engine consists of its steel crankshaft, its contact sealing lip and the lubricating oil that exists between these elements.

These crankshaft seals have two functions: they maintain the required lubricant layer and prevent contamination from entering from outside (moisture and water and dry particles of various materials, including dust, sand, dirt or other particles such as those produced in manufacturing processes), which can directly damage the crankshaft bearings of the motor.

In the operating practice of this type of seals it is important to estimate the durability of their operation, which is characterized by the minimal leakage of lubricating oil. Therefore, the durability of any such seal, which is characterized by the correctness of its operation when there is a minimal leakage of oil, can be assessed by means of a post-rightness of operation coefficient. Due to the fact that the preparation of operational tests in this respect requires obtaining preliminary results of laboratory tests, so such tests have been planned and carried out. In these studies, a simplified model of the above mentioned seal was applied, developed on the basis of the friction node of the modified T-02 four-ball apparatus. The results of these tests are presented in the article, the main purpose of which is to present the method of developing the results of tests carried out at the friction node of the modified T-02 four-ball apparatus using a relational diagnostic model.

The presented relational diagnostic model, whose general graphic form is shown in Fig. 1, fulfils two basic functions:

- theoretical is a special (due to the sealing contact zone adopted as the research objective) reflection of the modeled system,
- practical it is a tool enabling the analysis of discrete stochastic processes in states and continuous in time, the values of which are distinguished such states as: S_1, S_2 and S_3 (Fig. 1).

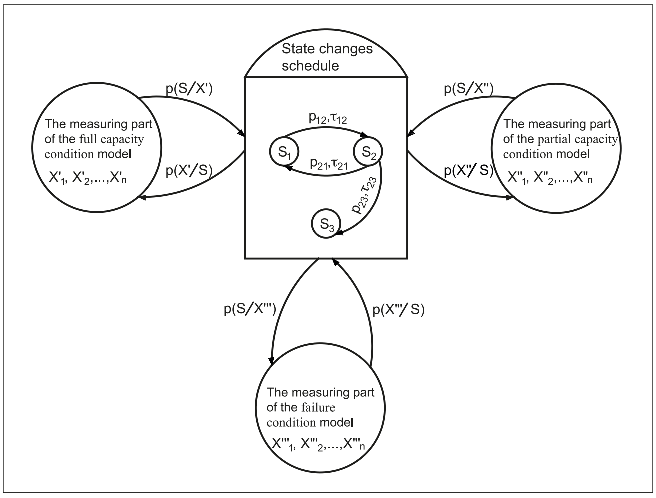


Fig. 1. Rational model of changes in the condition of sealing systems: S_1 – full capacity condition; S_2 – partial capacity condition; S_3 – failure condition; p_{ik} – probability of process transition from S_i to S_k condition; τ_{ik} – duration of S_i condition under condition of transition to S_k condition; i, k = 1, 2, 3; $X'_1, X'_2,.., X'_n$ – observational indicators indicating full capacity status; $X''_1, X''_2,.., X''_n$ – observational indicators indicating incomplete condition status; $X'''_1, X'''_2,.., X'''_n$ – observational indicators indicating zero condition

The conditional probability presented in Fig. 1 can be presented in general terms by means of formulae:

$$P\,(S\,/\,X) = \frac{P\,(S \cap X)}{P\,(X)} \tag{1}$$

$$P\,(X\,/\,S) = \frac{P\,(S \cap X)}{P\,(S)} \tag{2}$$

Therefore, it should be assumed that the identification of a given technical condition of a lip seal can be determined only taking into account the probability of its occurrence.

The engine crankshaft seal, which will be the subject of research, is a combination of components (sealing lip, crankshaft and sealed substance, i.e. engine lubricating oil), which should ensure maximum reduction of oil leakage.

The above sealing ability is the main characteristic (property) of this type of sealing system and is measured by the mass of the substance escaping through the seal (effluent rate) in relation to the permissible mass of the leaking substance [1, 2, 3].

According to the author, the sealing ability can be assessed by means of the "W_{PD}" coefficient of performance, which can be determined according to the following relation:

$$W_{PD} = 1 - \frac{D_S}{D_{S(dop)}} = 1 - \frac{1}{D_{S(dop)}\Big|_{t_0}^{t_r}} \int_{t_o}^{t_r} D(t)dt \tag{3}$$

At the same time, according to [4, 5, 6, 7]:

$$D_S = LS \cdot \tau \tag{4}$$

where:

D_S, $D_{S(dop)}$ – actual and permissible 'harmful' effects as appropriate;
τ – duration of one crankshaft revolution;
L_S – "harmful work" during one revolution;
t_r – estimated useful life;
$D(t)$ – a function that describes how the 'harmful' effect changes over time;
t_o – is the moment from which the operation of the tested sealing system commenced.

In conditions of full suitability of the sealing system for operation, i.e. when the system is in the state of full suitability S_1: $D_S < D_{S(dop)}$, the coefficient of correctness of operation $W_{PD} > 0$.

In conditions of incomplete usability, i.e. when the system is in the state of partial usability S_2: $D_S = D_{S(dop)}$, the coefficient of correctness of operation $W_{PD} = 0$.

Under fault conditions, i.e. the system is in a state of failure S_3: $D_S > D_{S(dop)}$, the coefficient of correctness of operation $W_{PD} < 0$.

The proposed diagnostic parameter, the W_{PD} coefficient of correctness of operation enables to check, among other things, the limits of measurement of the strength (durability) of sealing lips, whose basic features will be described in the next chapter.

DYNAMIC SEALING MECHANISM

The view of the cross-section of the traditionally used lip seal and its scheme with the selection of the main parameters of the construction structure is shown in Fig. 2. The nomenclature used in this figure for its description is the same as that used by Johnston in his work [8].

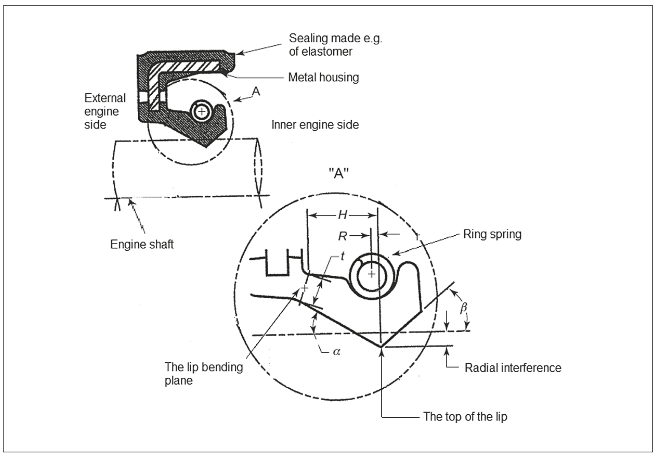


Fig. 2. View of the cross-section and the scheme of the lip seal with an indication of the basic parameters of its construction structure: A – structural detail; R – distance between spring axis and lip apex; H – lip length; α – lip external angle; β – lip internal angle; t – thickness of bending plane

The specificity of shaft sealing in the engine hull results from the asymmetry of the seal geometry [8, 9]. The geometry of this seal is characterized by a specific elastomer angle on the oil side of the seal, which is steeper than the angle on the ambient side. Typical design angles, when the seal is in the free state, are in the range of 40–45° for the β attack angle (on the hull-to-body side of the engine) and 25–30° for the α angle on the envelope side (outside the engine hull). After being installed on the shaft, these angles will change by about 10°, taking the values in the work-state: $\beta = 50°$ and $\alpha = 20°$. A further influence on the asymmetrical distribution of stress is the position of the seal spring. The cross-sectional axis of this spring is moved away from the top of the lip by the size R, usually by 10% of the length of the edge of the length H. This shift is directed towards the inside of the seal. This position of the spring in relation to the seal is important, so that the correct value of the direct edge load can be ensured and the asymmetrical geometry of the seal can be maintained during operation. Moving the axis of the seal by R size ensures the required pressure to the shaft in the external direction without the possibility of excessive lip deformation.

There are two factors that have a significant influence on the sealing performance of lip seals. They provide a permanent pumping of oil that enters the "contact zone" between the lip of the seal and the crankshaft of the engine on the oil side of the seal. This phenomenon is known as inward pumping of oil. The first of these factors is described as a micro-mechanism [8], which provides both load support for oil layer formation and inward pumping (Figure 3), and the second as a macro mechanism which primarily supports the pumping effect.

This micro mechanism has been the subject of many studies and analyses in the last three decades, the most important of which are the results obtained by Salanta [10]. These studies show that the proper effect of oil pumping depends on two important characteristics of the seal:

1. Asymmetrical geometry.
2. The texture of the elastomer surface in the contact zone after the seal has been applied.

The correct functioning of the seal to provide the required shaft sealing in each engine shaft depends significantly on both of these characteristics.

Once the seal is installed, the lip of the seal adheres to the crankshaft, creating a sealing or contact zone with a width of typically 0.2–0.4 mm. When the motor is started, a different texture is visible as a result of lip abrasion in the moulded surface. This is due to the fact that the abrasive material of the lip seal is used to expose the surface with clear thickening or micro-shifting of its layers. Such roughness is deformed by the rotation of the motor shaft so that it is oriented at an angle to the shaft axis, as shown in Fig. 3 [9, 11, 12].

These thickened bumps become micro-elastohydrodynamic wedges that allow both the surface of the elastomer seal to be raised and the oil to be pumped when the engine shaft rotates in both directions. However, as the contact surface on the outside of the engine hull is more extensive than on the inside, the effect of pumping into the inside of the engine hull is stronger than pumping outwards. Therefore, the seal

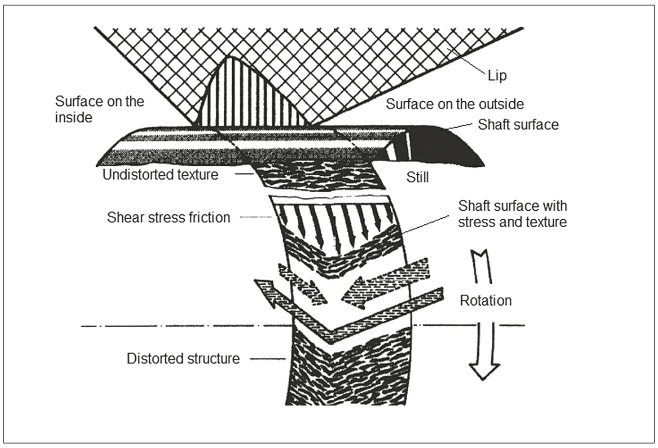


Fig. 3. Diagram of contact lip distortion in the form of irregularities at an angle to the shaft axis

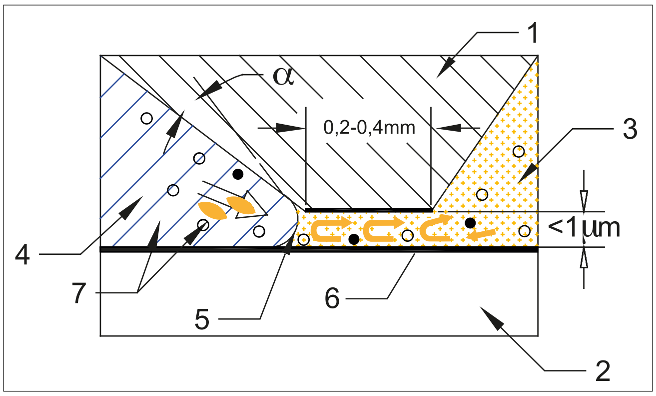


Fig. 4. Flow diagram of the oil film under the sealing lip: 1 – sealing lip, 2 – crankshaft, 3 – inner side (lubricating oil), 4 – outer side (air), 5 – concave meniscus, 6 – flux of recycled oil, 7 – impurities, α – wetting angle (oil moistens the lip walls of the seal)

has a tendency to return oil and because of that prevents oil leakage outside the engine hull. This action is shown schematically in Fig. 4 [8, 9, 12, 13].

During operation, changes in direction of rotation and loads on the engine crankshaft create new gaps in the structure of the sealing lip. The distribution of the absorption of the seal is then random and ensures proper tightness of the tribological system of the seal as long as the deformation of the structure is reversible. However, after a certain period of operation, the ability of the unevenness of this surface to return to a neutral position when the shaft is held in place will decrease.

As a result, these bumps will have a directional orientation that will always result in a more effective seal in the normal crankshaft rotation than in the reverse direction. This has been demonstrated experimentally [8] by testing the reversal of a new seal and then repeating these tests after several hundred hours of operation.

The correct functioning of lip seals depends not only on the constructional value of the seal, lip edge loading, etc., but also on the sealing material (its elasticity) and other factors, such as micro-inequalities of the engine crankshaft. In further considerations, the main factor influencing the correct functioning of the seal is presented, together with examples of how the performance of the seal is calculated, which determines the performance of the seal.

THE PARAMETERS OF THE PROPER FUNCTIONING OF THE SEAL

The most important factor negatively influencing the durability of the sealing lip is the heat generation at the point of contact with the shaft. "Harmful" measures to dissipate the heat generated by the friction are one of the key factors in limiting the life of the seal. The harmful effects of friction between the seal and the motor shaft can be demonstrated by considering that a typical 50 or 60 mm diameter shaft seal will consume up to 100 W of power [14].

In addition, the type of lubricating oil that comes into contact with the seal must also be taken into account, as its viscosity will have a direct effect on the dispersed operation ("harmful operation") produced in the friction node, which is formed by the sealing lip, lubricating oil and the engine crankshaft. Most of the generally available empirical data from empirical studies relate to typical lubricants, such as engine oil. This type of lubricant has a dynamic viscosity in the range of 0.005-0.01[Belt] at the typical working temperature for them t = 100°C. Lubricants with higher viscosity, such as transmission greases, generate higher shear stresses and thus cause higher friction. Examples supporting these observations are shown in Figure 5, where it can be seen that increasing viscosity in engine oils increases energy consumption by about 30%. Transmission greases, which may have higher viscosities, can further increase energy consumption, because the denser the viscosity of the grease causes a convex-acid increase in lower lip temperature.

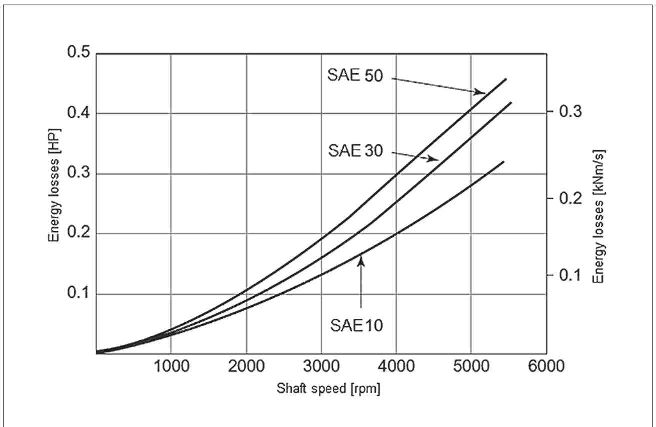


Fig. 5. Effect of viscosity on seal energy consumption. Source: BHR Group

Continuous dissipation of energy in the form of heat has a negative effect on the durability of the seal, which is equal to the value of the correct operation time until the loss of sealing efficiency. However, it is not easy to determine the actual seal life, as it depends on a number of factors, such as the operating temperature, the eccentricity of the spring seat, the speed of rotation, and the type of sealing substance and lubrication.

Figure 6 shows the curves of the estimated seal life, obtained by considering the main factors determining the seal life, such as the rubber material of the seal, lubricant and lip temperature [15].

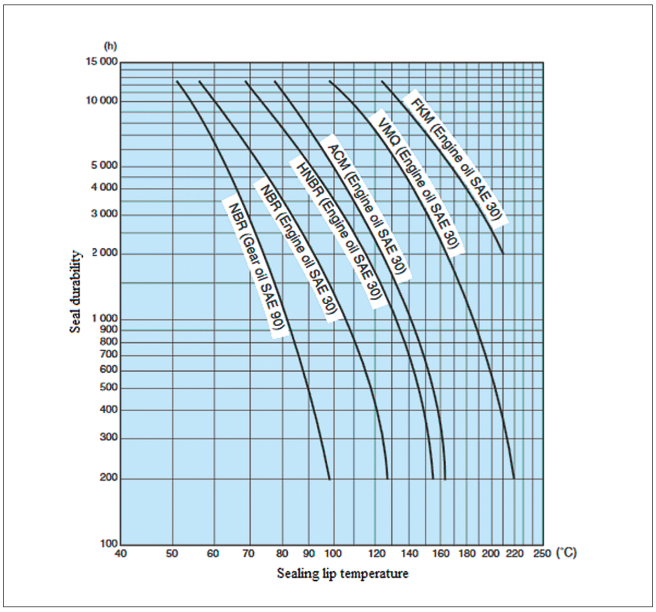


Fig. 6. Lip seal durability estimation curves: FKM - fluorocarbon rubber, VMQ – silicone rubber, ACM– acrylic rubber, HNBR – hydrogenated nitrile rubber, NBR – nitrile rubber. Source: Oil Seals & O-rings. KOYO Sealing Techno CO.,LTD

The results of these studies were taken into account in the author's studies, which were carried out on a laboratory stand enabling, among others, the study of energy losses for various variants of shaft sealings, as tare nodes. This stand and the obtained research results are presented in point 4. of the article (Test stand and results of the seal durability test).

TEST STAND AND RESULTS OF THE SEAL DURABILITY TEST

The test stand was slightly modified in comparison to the stand described in the paper [7]. A ball of bearing steel with a hardness of 62.7 HRC was placed in the upper grip and treated as a rotating crankshaft. Three rubber balls of hardness in the range 15–80°sh were placed in the lower handle-cartridge in the shape of a bowl and treated as a sealing lip because they are immobilized and pressed against the upper ball with a force of up to 7200 N. This modification made it possible to obtain similar conditions of operation of the stationary friction node (stationary tribological system) to the conditions of operation of the seal, whose elements are the lip seal pressed by a ring spring against the crankshaft of the engine.

In the next part of the article, the values of parameters proving the correctness of the seal and its sealing lip were presented.

PARAMETERS FOR THE CORRECT FUNCTIONING OF THE SEALING LIP

While testing the correctness of the sealing lip on the modified friction node of the four-ball apparatus T-02, a constant oil temperature of 20°C and a minimum possible constant load of 61.2 N were assumed. As a multitude having a significant influence on the changes in the value of the correct functioning of the sealing lip, a variable speed of rotation n of the handle of the upper ball, treated as the crankshaft of the engine, was assumed.

Three model variants of rotary shaft seals were tested:

1. Dry friction node consisting of three rubber balls (seal lip) and one steel ball (crankshaft) without lubricant – symbol A.
2. A friction node consisting of three rubber balls immersed in pure SAE30 lubricating oil and one steel ball – symbol B.
3. A friction node consisting of three rubber balls immersed in SAE30 lubricating oil with additives to improve its quality and one steel ball – symbol C.

For these three variants the following has been determined: P_S power loss and the coefficient of correctness of operation W_{PD} of sealing lips on the basis of the algorithm [7], the shortened verse of which is presented in the algorithm in Fig. 7.

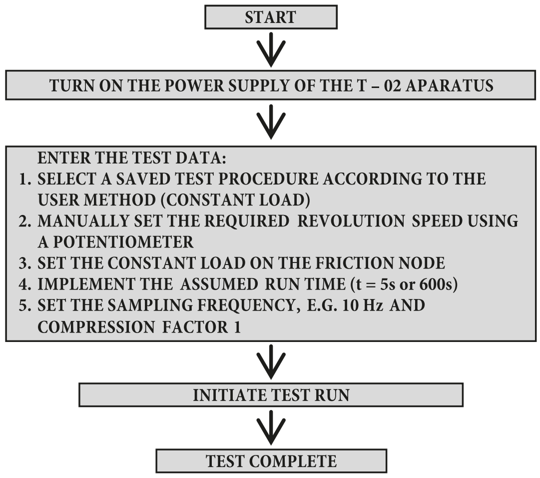


Fig. 7. Algorithm of tests on the measuring stand

The measurements were repeated four times and their mean value was checked on the basis of the t-Student's distribution with a probability of 0.68.

POWER LOSSES

The test results obtained for speeds n = 312,500 and 1450 rpm are shown in Table 1, which contains the power loss recorded at the start of the applied test. The power loss was determined on the basis of the formula:

$$P_S = \frac{L_S}{\tau} = \frac{2 \cdot \pi \cdot P_r \cdot r \cdot \mu}{\tau} \tag{5}$$

where:

$L_S = 2 \cdot \pi \cdot P_r \cdot r \cdot \mu$ – "harmful work" during the first rotation of the steel ball fixture;
P_r – radial load [N];
μ – coefficient of friction [–];
r – shaft radius [m];
τ – duration of one rotation of the steel ball fixture.

Tab. 1. Power losses

Rotational speed during test n[rpm]	Power loss [Nm/s]		
	Symbol A	Symbol B	Symbol C
312	1.62	0.84	0.87
500	1.88	1.54	1.40
1450	9.77	5.16	5.02

The presented calculations of power losses show that in a dry friction node the largest dissipation of energy takes place, i.e. for this combination the largest increase of the dissipated "harmful work" takes place. For this reason, in further empirical research, as a measure of the permissible "harmful action" of a seal, the action of a dry friction node is assumed and the coefficient of lawfulness of action of any friction node determines the formula:

$$W_{PD} = 1 - \frac{L_S \cdot \tau}{L_{S\,(\mathrm{dop})} \cdot \tau} \tag{6}$$

where:

$L_{S(\mathrm{dop})}$ – "harmful work" of the seal during the first rotation of the steel ball holder at the set speed of the dry friction node;
L_S – "harmful work" when rotating the steel ball fixture at a set speed of the friction node under test;
τ – duration of one rotation of the steel ball fixture.

COEFFICIENT OF CORRECT OPERATION OF SEALING LIPS

Results of tests lasting 600 seconds, obtained according to the concept presented in the paper [7] and the shortened algorithm Fig. 7 and formulas (5, 6), for rotational speed n = 312,500 and 1450 rpm are shown in Table 2, 3, 4 and Figures 7, 8, 9.

Tab. 2. W_{PD} seal performance factors for rotational speed 312 rpm

Mesurement time [s]	W_{PD} [–]		
	Symbol A	Symbol B	Symbol C
Start	0.00	0.00	0.00
100	failure	0.44	0.41
200		0.43	0.44
300		0.39	0.44
400		0.37	0.44
500		0.37	0.43
600		–0.06	0.46

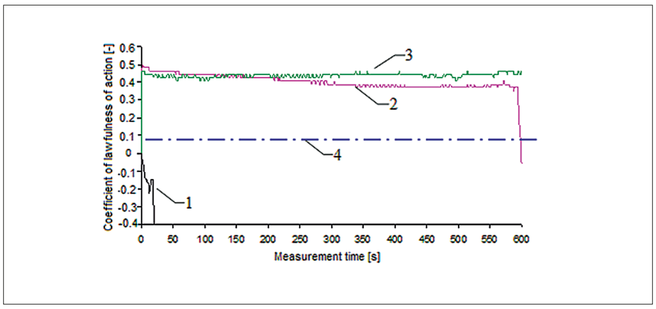


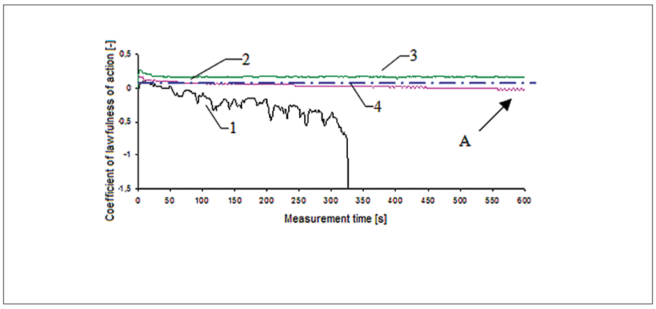
Fig. 8. Comparison of the W_{PD} seal performance coefficients at rotational speed of 312 rpm: 1 – symbol A; 2 – symbol B; 3 – symbol C; 4 – permissible 'harmful' effect

Tab. 3. W_{PD} seal performance factors for 500 rpm

Mesurement time [s]	W_{PD} [–]		
	Symbol A	Symbol B	Symbol C
Start	0.00	0.00	0.00
100	–0.10	0.08	0.15
200	–0.18	0.05	0.18
300	–0.38	0.03	0.18
400	failure	0.03	0.15
500		0.00	0.18
600		0.00	0.15

Tab. 4. W_{PD} seal performance factors for 1450 rpm

Mesurement time [s]	W_{PD} [–]		
	Symbol A	Symbol B	Symbol C
Start	0.00	0.00	0.00
100	failure	0.03	0.05
200		0.03	0.05
300		–0.03	0.05
400		–0.03	0.05
500		–0.05	0.05
600		–0.05	0.05



Detail "A"

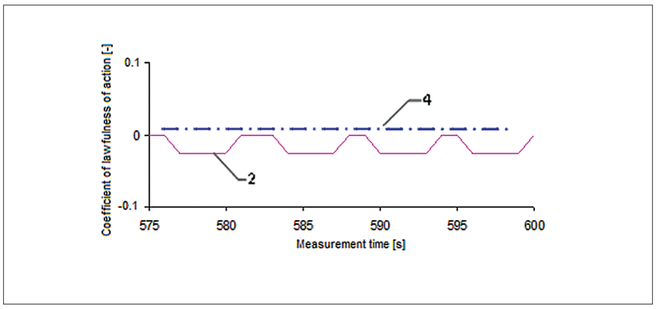


Fig. 9. Comparison of the WPD seal performance coefficients at rotational speed of 500 rpm: 1 – symbol A; 2 – symbol B; 3 – symbol C; 4 – permissible 'harmful' effect

At a preset load and speed of 312 rpm, for a dry friction node immediately after starting, the coefficient of performance of $W_{PD} < 0$, i.e. it is in a state of failure S_3, which means that the version of the model with symbol A is worthless. On the other hand, the model marked with symbol B lost its full capacity after about 600 seconds and went to the state of incomplete capacity S_2. The life span of this version of the combination model is approximately 600 seconds. The C-model remains at full capacity throughout the entire test, meaning that the tribologic system lasts more than 600 seconds.

In the second series of tests at a given load and speed of 500 rpm on the basis of the obtained test results, it can be stated that:

- in the case of testing a system with symbol A: the system changed from S_1 to S_2 partial capacity after approx. 10 seconds and in the case of partial capacity the seal lasted approx. 250 seconds, after which it went into zero capacity state.
- in the case of testing a system with symbol B: After the system has passed from the full capacity state S_1 to the partial capacity state S_2, an oil pumping effect may be observed, resulting in a specific cooling and reduction of the friction coefficient (detail "A") and a random transition from the full capacity state S_1 to S_2 and vice versa.
- in the case of testing a system with symbol C: as in the previous test, the system remains in full S_1 compliance and has a life expectancy of more than 600 seconds.

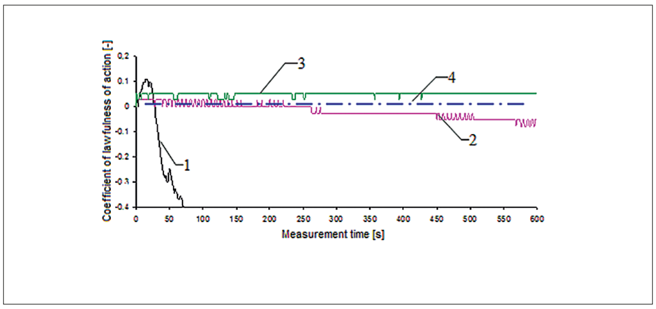


Fig. 10. Comparison of the performance factors at a speed of 1450 rpm: 1 – symbol A; 2 – symbol B; 3 – symbol C; 4 – permissible 'harmful' effect

In the final series of tests at a given load and speed of 1450 rpm, based on the test results, it can be concluded that:

- for testing a system with symbol A: after approximately 40 seconds, the system has changed from S_1 to S_3 without a regeneration 'test' in the S_2 partial capacity condition,
- in the case of testing a system with symbol B: already after about 80 seconds, the system changed from the S_1 to the S_2 partial capacity status, where, however, due to too high speed, the pumping effect did not prevent the system from going to state of zero capacity after about 400 seconds of durability,
- for testing the C system: as in the previous two tests, the system remains in S_1 full capacity and has a life expectancy of more than 600 seconds.

FINAL REMARKS AND CONCLUSIONS

The presented diagnostic model enables identification of rotary crankshaft seals of piston combustion engines on the basis of correctness of operation of the friction node of the T-02 four-ball apparatus. The analysis of durability of seals (i.e. friction nodes being tribological systems) was performed by determining the coefficient of correctness of operation of W_{PD}, which enabled comparison of three versions of operation. The best D_S action was found in the case of the tribological system with symbol C, in which the lubricating oil was of the best quality. Therefore, it can be concluded that the hypothesis presented in the paper [7] that the quality of oil affects the correctness of operation of the tested tribological system was finally confirmed, which justifies the need for good research on sealing lips, carried out under the conditions of the actual exploitation of internal combustion engines.

Additionally, the presented research results show that:

1. To interpret the results of the study, a model of the friction phenomenon in the contact area of the sealing lip (rubber balls in the oil sump of the T-02 apparatus friction node) and the crankshaft of the engine (steel ball in the upper grip) were used.
2. After the start-up and first rotation of the friction node, the greatest loss of power per revolution occurred at technically dry friction and then the greatest permissible distortion of the contact lips, i.e. rubber balls, took place.
3. The W_{PD} sealing lip performance factor entered allows for the separation of capacity and zero capacity states.
4. The observed pumping effect is the result of the tribological system that occurs on both sides of the lip, so that heat exchange takes place.
5. The process, due to the pumping effect, of a system switching from partial S_2 capacity to full S_1 capacity or S_3 zero capacity, is random and therefore requires a probabilistic analysis to determine the probability of *P(S/X)* and *P(X/S)* as determined by formulae (1) and (2).

The research shows that the diagnostic model proposed by the author to identify the correctness of operation of each tribologic system may be useful for determining the durability of lip seals and for compatibility tests between the sealing lip, crankshaft and sealed lubricating oil.

REFERENCES

1. Kragelsky V.: *Friction Wear Lubrication*. English translation, Mir Publishers, 1981, Vol. 3, pp. 82–111.

2. Pieter Baart, Bas van der Vorst, Piet M. Lugt & Ron A.J. van Ostayen: *Oil Bleeding Model for Lubricating Grease Based on Viscous Flow Through a Porous Microstructure*, Tribology Transactions, Volume 53, 2010, pp. 340–348.

3. Paszota Z.: *Energy losses in hydrostatic drive*. LABERT Academic Publishing, 2016.

4. Girtler J.: *A method for evaluating theoretical and real operation of diesel engines in energy conversion formulation taking into account their operating indices*. Polish Maritime Research. Vol. 18, No. 3(70), 2011, pp. 31–36.

5. Girtler J.: *The method for determining the theoretical operation of ship diesel engines in terms of energy and assessment of the real operation of such engines, including indiators of their performance*. Journal of Polish Cimac. Vol 6 no. 1, 2011, pp. 79–88.

6. Girtler J.: *Quantitative interpretation of energy-based systems and index of their reliability*. Journal of Polish Cimac. Vol 3 no. 1, 2008, pp. 87–94.

7. Bzura P.: *Influence of lubricating oil improvers on performance of crankshaft seals*, Polish Maritime Research. Special Issue S1 (97) 2018 Vol. 25, pp. 172 177.

8. Johnston D.E.: *Design aspects of modern rotary shaft seals*, Proc. I. Mech. E. 213 (Part J) (1999), pp. 203–214.

9. Flitney R. K.: *Seals and Sealing Handbook (Sixth Edition)*. 2014 Elsevier Ltd., Chapter 3.

10. Salant R.F.: *Soft elastohydrodynamic analysis of rotary lip seals*, Proc. I. Mech. E. Part C J. Mech. Eng. Sci. 224 (12) (2010), pp. 2637–2647.

11. Kammüller M.: *Zur abdichtwirkung von radial wellendichtringen (Dr-Ing thesis)*, Universität Stuttgart, 1986.

12. Baart, P., Lugt, P.M. and Prakash, B.: *Review of the lubrication, sealing and pumping mechanisms in oil and grease lubricated radial lip seals*. Proc. Inst. Mech. Engineers., part J: J. Engineering Tribology, 2009; pp. 347–358.

13. FST Technical Manual 2015. Freudenberg.

14. Jagger E.T. *Rotary shaft seals: the sealing mechanism of synthetic rubber lip seals running at atmospheric pressure*, Proc. I. Mech. E. 171 (1957).

15. Koyo Oil Seals& O-Rings. KOYO Sealing Techno CO., LTD.

CONTACT WITH THE AUTHORS

Piotr Bzura
e-mail: pbzura@pg.edu.pl

Gdańsk University of Technology
Narutowicza 11/12
80-233 Gdansk
Poland