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An Experimental Study on Frictional Losses of Coated Piston Rings with Symmetric and Asymmetric Geometry

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Abstract

An increase in the efficiency of internal combustion engines is a key challenge for engineers today. Mechanical losses contribute significantly to engine inefficiency, and the piston assembly has the largest share in these losses. Various measures are therefore taken to reduce friction between the piston and the rings against the cylinder. However, the undertaken changes most frequently generate new challenges. For instance, lowering the viscosity of the engine oil or increasing the engine load may lead to accelerated wear of the mating surfaces. In order to resolve this problem, more and more complex materials and anti-wear coatings have to be used. Furthermore, under these conditions, the shape of the ring's sliding surface becomes more important.

This article presents the results of experimental research on the influence of the geometry of the sliding surface and the use of various anti-wear coatings. Motoring tests were carried out on a specially adapted engine over a wide range of rotational speeds and temperatures. Nine sets of rings with three different sliding surface profiles, along with chrome, molybdenum (Mo), and titanium-based coatings, were tested. The obtained results confirmed that the friction of compression rings can be significantly reduced through the appropriate geometry of the sliding surface and the use of anti-wear coatings. Of all the tested variants, the best results were achieved for the rings with moderate asymmetry and with titanium nitride (TiN) coatings.

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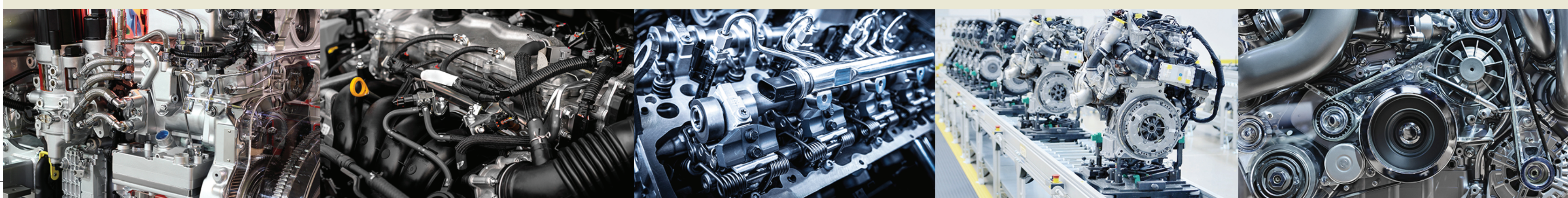


1. Introduction

Currently, most internal combustion engines in the transport sector operate on average at a small or medium load. At such conditions, a significant proportion of engine inefficiency can be attributed to mechanical losses. It is estimated that 11.5% of fuel energy is used to subdue the engine's own friction [1]. The operation of the piston assembly is the largest source of mechanical losses. It amounts to 38-68% of the total mechanical losses [2, 3]. The friction of the rings against the cylinder accounts for up to 50% of the piston assembly losses [4]; therefore, reducing the friction of the rings can significantly improve mechanical efficiency and fuel economy.

Numerous measures are taken to reduce the friction of the rings. However, due to the different functions of the rings, any improvement in one aspect often worsens the other. For example, the changes that reduce friction may lead to an increased blowby, deterioration in the heat flow from the piston to the cylinder, or increased oil consumption. In order to reduce friction, either the axial heights or tension of the rings are reduced, or lower viscosity oils are used. Most of these measures reduce the thickness of the oil film and increase the proportion of mixed friction. Therefore, it is becoming more and more important to optimize the shape of the sliding surface of the rings in order to preserve the conditions of fluid friction as well as the material and structure of this surface to reduce the friction in the conditions of mixed and boundary lubrication and to minimize wear.

Simulation tests are very helpful in assessing the influence of the shape of the sliding surface on the interaction between the rings and the cylinder. Many models of both hydrodynamic and mixed lubrication have been developed. The influences of various factors, such as bore distortion and ring conformability, the piston secondary motion and the non-axisymmetric position of the rings in the cylinder, and the circumferential variation of the ring face profile are considered, in particular, models [5, 6, 7, 8]. Some of these models allow for taking into account the impact of surface texturing on friction losses [9, 10, 11]. The effect of the sliding surface geometry on the thickness of the oil film and the frictional forces were analyzed in detail in [12]. This work also points out the consequences of changes in the shape of this surface due to wear. The influence of the sliding surface geometry, especially its asymmetry, was also analyzed in [13, 14].

Experimental works on the influence of the shape of the sliding surface are very rare. The work [15] presents the results of engine tests, which showed a positive effect of the asymmetry of the top ring face on oil consumption.

There are enormous capabilities of lubrication models to analyze the influence of the geometrical features of the mating surfaces and working conditions, i.e., speed, loads, and temperature. However, the assessment of the adhesion of the coatings made of various materials to the substrate, or the wear and scuffing resistance, is mainly based on experimental tests usually carried out on tribometers.

The work in [16] investigated the effect of Cr2AlC additives on the tribological properties of the piston ring coated with the Ni-Mo-Al alloy. It was found that the coatings showed good adhesion properties to the substrate even when the rings were heavily loaded. In the work [17], ionically deposited chrome coatings were tested with the use of tribological machines. It was also found that such coatings should improve engine life as well as fuel efficiency. Many studies have also confirmed good tribological properties of molybdenum (Mo)-based layers [18, 19, 20].

Very promising results have been obtained for various titanium-based coatings. Physical vapor deposition (PVD) multilayer coatings were tested by Lyubimov et al. [21]. It was stated that the coating should consist of two layers: the wear-resistant internal layer made of titanium nitride (TiN) and a running-in liable external one made of titanium. The application of such coating resulted in the reduced wear by 40%, even compared to the rings having chromium (Cr)-electroplated coatings. The high wear resistance and low friction coefficient of plasma-deposited TiN coatings under nonlubricated conditions and at high loads are shown in [22, 23, 24, 25, 26]. Reduced friction and wear and no tendency to seize were also proved for titanium suboxide coatings [27, 28]. These studies were conducted on tribometers under dry or mixed friction conditions. The excellent tribological properties of the TiN coating have been demonstrated in comprehensive tests carried out both on a tribotester and a running engine [29].

The results of the research confirm that coatings, especially TiN coatings, can reduce friction and wear. However, despite many studies on their impact on mechanical losses, as well as on their durability and methods of their deposition, there still remain difficulties and problems to be solved [30].

It should be remembered that in the case of boundary friction, the effects of the coating depend on the engine oil composition. The applied oil additives can greatly reduce friction and wear with some coatings, but not with others. The existence of the synergy between coatings and some oil additives was demonstrated in [31]. The potential of using advanced coatings for energy efficiency and emissions from internal combustion engines was assessed on the basis of a comprehensive study in [32].

Most research on the ring's sliding profile has been made using numerical methods. On the other hand, most research on the coating has been made by means of wear testers. Tribotesters allow for very precise measurements, but the conditions of cooperation of the tested surfaces, even in reciprocating testers, significantly differ from those in the real engine. Besides, usually, either the profile or the coating is investigated. In this work, combined tests on coatings and the geometry of the sliding surface of compression rings were conducted. The tests were carried out on a dynamometric stand using a specially adapted two-cylinder engine.

2. Method

2.1. Dynamometer Test Stand

Experimental studies on the influence of the shape of sliding surfaces and various anti-wear coatings on the friction of

FIGURE 1 Dynamometer test stand.

FIGURE 2 Test engine on the test stand.

compression rings against the cylinder were carried out on a stand that consisted of an appropriately adapted internal combustion engine driven by a motoring dynamometer (Figure 1). The basic parameters of the engine are given in Table 1, and its view on the test stand is shown in Figure 2. A two-cylinder engine was chosen to increase the share of the ring friction in the total resistance to the motion of the engine. Only the crankshaft with the pistons was driven in the tested engine. All other systems were disconnected. The original valves were removed and replaced with nonreturn valves. One of these valves allowed the cylinder to be filled with air and opened when the pressure in the cylinder dropped below the ambient pressure. In turn, the second one allowed the cylinder to empty the air and was opened when the pressure in the cylinder exceeded the set value. During the presented tests, the value of this pressure was set at 1 MPa. Thanks to this solution, the pressure in the cylinder changed its value during each rotation of the crankshaft from approximately 0 to 1 MPa. The maximum cylinder pressures were lower, and their changes in the function of the crank angle differed from those occurring during normal engine operation, but still, this solution was closer to the conditions occurring during normal engine operation than in the case of no compression at all.

Engine oil and coolant were supplied to the engine from external cooling and lubrication systems. These systems made

TABLE 1 Research engine specifications.

Parameter	Value
Number of cylinders	2
Displaced volume	704 cc
Bore	80 mm
Stroke	70 mm
Compression ratio	8.8:1
Number of valves	2
Top ring axial height	1.5 mm
Top ring thickness	3.35 mm
Second ring axial height	2.0 mm
Second ring thickness	3.35 mm

it possible to precisely control the temperature of the oil and coolant with an accuracy of not less than ±0.2°C. An engine oil of 5W-30 viscosity grade was used to lubricate the engine during the tests.

The control system of the electric motor with an inverter of an adjustable output frequency allowed to adjust the rotational speed in a wide range and precisely maintain its set value. An HBM T5 torque transducer cooperating with the HBM Spider 8 amplifier and an analog-to-digital converter were used to measure the torque (Figure 3). The measuring range of the meter was 50 Nm and the accuracy class 0.1. The HBM catman 5.0 software was used for the acquisition and processing of the recorded signals.

2.2. Experimental Rings

Compression rings with three variants of anti-wear coatings and three variants of the shape of the sliding surface were prepared for the tests. The basic dimensions of the test rings corresponded to those of the standard rings used in the tested engine. This means that the axial and radial clearances of the rings in the piston grooves and the ring end clearances of the tested rings corresponded to the standard ones.

The tests were carried out for the following three variants of anti-wear coatings:

- Cr_Mo: the top ring with a Cr coating, the second ring with a flame-sprayed Mo coating made of Mo wire
- Cr_Mo + P: the top ring with a Cr coating, the second ring with a plasma-sprayed Mo coating and a phosphate coating
- TiN_TiN: top and second rings with TiN coatings made by plasma-assisted PVD

What is more, for each of the three types of coatings mentioned above, the rings with three different sliding surface profiles were prepared (Figure 4):

FIGURE 3 Torque meter.

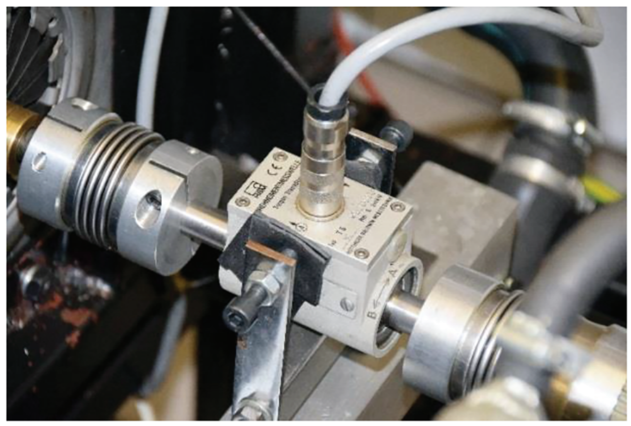
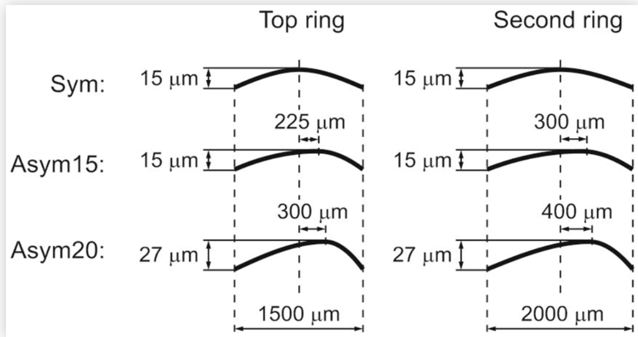


FIGURE 4 The shapes of the sliding surfaces of the tested rings.



- Sym: the rings with symmetrical profiles with the highest point of the profile halfway up the ring height; profile heights were 15 μm for both the top and the second ring.
- Asym15: the rings with asymmetrical profiles where the apex position was shifted by 15% of the height of the ring, i.e., by 225 μm for the top ring and by 300 μm for the second ring; profile heights were 15 μm for both rings.
- Asym20: the rings with asymmetrical profiles where the apex position was shifted by 20% of the height of the ring, i.e., by 300 μm for the top and by 400 μm for the second ring; profile heights for both rings were 27 μm.

All tested compression rings were made of ADI modified cast iron. In the case of Mo and Cr coatings, the shapes of the sliding surfaces were formed using the contour method once the coating was applied. In the case of titanium coatings, the rings were shaped to size prior to the coating. It was due to the application technology and the obtained thickness of the TiN coating equaling 4-5 μm. A detailed description of the coating application technology and its properties are presented in [33].

It should be emphasized that the abovementioned profiles (Figure 4) were to have rings after run-in. This means that when producing the rings, a certain allowance was assumed for the expected wear of the rings during their running-in. The size of this allowance for the individual profile and coating was determined on the basis of preliminary research [33]. For the Mo coatings, barrel allowances from 10 μm to 16 μm were used, whereas for the titanium and Cr coatings: from 1 μm to 3 μm.

In order to ensure the greatest possible repeatability of dimensions and other features important for the friction of the rings, several dozen rings of particular coatings and sliding surface profiles were made and subjected to detailed measurements. From these several dozen rings, test sets were selected so that they differed only in the shape of the face surface or in the coating. The test engine had two cylinders; therefore, two sets of rings with a given coating and profile were selected. In total, nine sets of rings (the two top rings and two second ones), labeled as in Table 2, were selected for testing.

The measurements of the rings' geometry, tangential force, and peripheral contact to the cylinder were made before mounting the rings in the engine and after completing the engine tests.

The shape of the sliding surfaces of the rings was measured using the TOPO 01K version 2D profilograph, calibrated according to the PW-08-L3 procedure [Figure 5(a)]. The measurements were carried out with an accuracy of ±0.001 mm. Profiles were taken at six randomly selected locations around the circumference of each of the rings before and after the bench tests (Figures 6-10).

Shape deviations measured in different planes around the circumference of the ring for the two mounted rings for each cylinder did not exceed 2 μm for the barrel height and 25 μm for the barrel top position at the axial direction. The deviations in the height of the profiles between all the rings of the same assumed profile did not exceed 5 μm when the tests were finished, while the axial position of the apex was within the range of 75 μm.

It should be emphasized that the significant wear of the sliding surface of the rings with Mo coatings (Figures 8 and 9) occurred mainly in the running-in phase. The wear in the later period, i.e., during the main tests, was much lower [33]. It was the same in the case of Cr and TiN coatings, yet in their case, the absolute wear was several times lower, and in the period after running-in, they were practically immeasurable using the methods available (Figures 6, 7, and 10).

The measurements of the axial height and radial width of the rings were made at six points on the circumference using dedicated dial gauges with an accuracy of 0.002 mm [Figures 5(b) and 5(c)]. The results of the axial height measurements for all tested top rings and in all planes did not differ by more than 10 μm for the measurements carried out both prior to and after the tests. In the case of the second ring, all results were within the range of 15 μm. The radial heights of the rings with the same assumed profiles did not differ by more than 30 μm, both before and after the tests.

TABLE 2 Results of ring measurements made after the tests.

Rings set	Ring no.	Profile height, mm	Profile top position, mm	Tangential force, N	Ring end clearance, mm
Sym Cr_Mo	Top ring	0.015	0.750	13.40	0.38
	Second ring	0.015	1.000	18.20	0.28
Asym15 Cr_Mo	Top ring	0.015	0.975	13.42	0.38
	Second ring	0.015	1.295	17.12	0.31
Asym20 Cr_Mo	Top ring	0.028	1.050	13.48	0.38
	Second ring	0.027	1.410	17.12	0.30
Sym Cr_Mo+P	Top ring	0.015	0.730	13.41	0.39
	Second ring	0.015	1.015	19.13	0.29
Asym15 Cr_Mo+P	Top ring	0.014	0.975	13.42	0.38
	Second ring	0.016	1.270	19.15	0.31
Asym20 Cr_Mo+P	Top ring	0.027	1.050	13.42	0.38
	Second ring	0.027	1.395	19.35	0.32
Sym TiN_TiN	Top ring	0.016	0.750	13.76	0.38
	Second ring	0.015	1.010	18.72	0.25
Asym15 TiN_TiN	Top ring	0.015	0.930	13.75	0.38
	Second ring	0.015	1.285	18.27	0.26
Asym20 TiN_TiN	Top ring	0.027	1.075	13.75	0.38
	Second ring	0.027	1.405	18.19	0.26

The measurements of the ring end gap were made with a clearance gauge once the ring was placed in a reference sleeve with a diameter equal to the nominal cylinder diameter of the tested engine, i.e., 80.005 mm [Figure 5(d)]. The differences between the rings prior to the tests did not exceed 0.01 mm for the top rings and 0.03 mm for the second ones. These differences for the rings after the tests were 0.02 mm in the case of the top rings and 0.08 mm in the case of the second rings.

The tangential force measurements were performed using a special analog-digital device [Figure 5(e)]. The tangential forces of the top rings before the test did not differ by more than 0.4 N and in the case of the second rings by 2.5 N. For the rings after the tests, the differences were 0.5 N and 5 N, respectively.

Moreover, the contact of the ring sliding surface to the bore was checked. A device consisting of a reference sleeve in which the tested ring was placed was used [Figure 2(f)]. The device had a light source behind the sleeve. After turning on the light, the sleeve was rotated on the rollers, and it was observed whether there were no light gaps between the ring and the sleeve. The rings with good conformability were used for the tests, i.e., such ones for which the light was visible only in the place of the ring end gap. The tested rings showed good conformability also after being removed from the engine.

The mean values of the key ring measurements obtained after the tests are presented in Table 2. The differences between the rings with different coatings but of the same assumed profile were very small, and they should not affect the results of resistance to motion measurements. It can therefore be concluded that rings with a given sliding surface profile and different coatings differed only in the coatings.

Standard oil rings for the tested engine were used in the tests. The ring end gaps for both oil rings were 0.25 mm, and the tangential forces stood at 59.22 N and 59.28 N.

2.3. Test Procedure

Preliminary tests revealed that significant wear of the sliding surface of the rings takes place in the first 6-10 hours of the operation of the rings, and during further operation, the wear is minimal [33]. Taking the above into account, after mounting a given set of rings in the engine, they were run in for about 15 hours. During the running-in phase, the engine was operated with the crankshaft rotational speed of 2000 rpm and the engine oil and coolant temperature of 80°C. Then the oil and coolant temperature was set to 50°C, and the motoring torque was measured at successive rotational speeds increased by 200 rpm in the range from 800 rpm to 3000 rpm. Next, the same measurements were taken at temperatures of 80°C and 100°C. The torque was recorded under steady-state conditions when the temperature did not differ from the assumed one by more than 0.1°C and the rotational speed by 1 rpm. This series of torque measurements at three temperatures and different rotational speeds was repeated five times. Having completed the five series, the rings were dismounted and subjected to the measurements described in Section 2.2. The viscosity of the lubricating oil was also measured and the torque meter was calibrated. The dynamic viscosity of the oil was measured with a Bookfield DV-II+Pro rotational viscometer working with Rheocalc32 programs. Then another set of rings was assembled and run in followed by five series of measurements. Such tests were carried out for all nine sets of rings with different coatings and different profiles of the sliding surface.

FIGURE 5 Sliding surface profile measurement (a), ring height measurement (b), ring width measurement (c), ring end-gap measurement (d), tangential force measurement (e), and ring-to-bore peripheral contact checking (f).

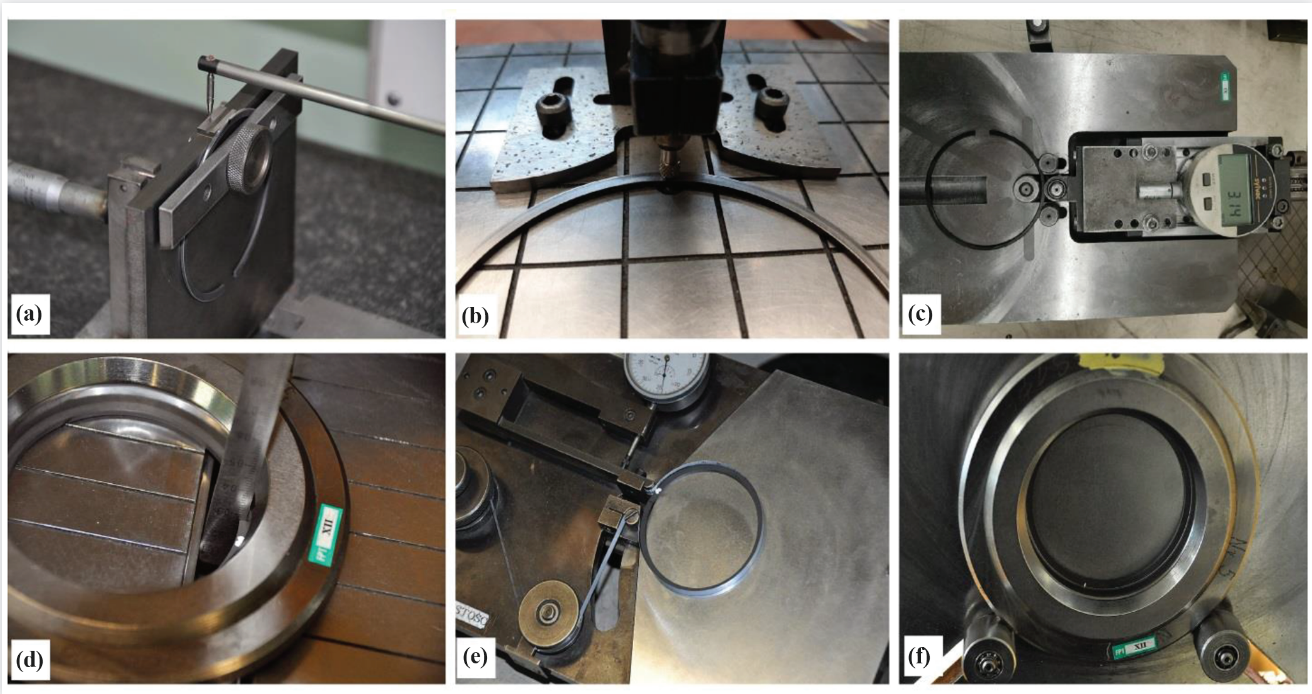


FIGURE 6 Measured profiles of the sliding surfaces of the top rings with the Cr coating.

FIGURE 7 Measured profiles of the sliding surfaces of the top rings with the TiN coating.

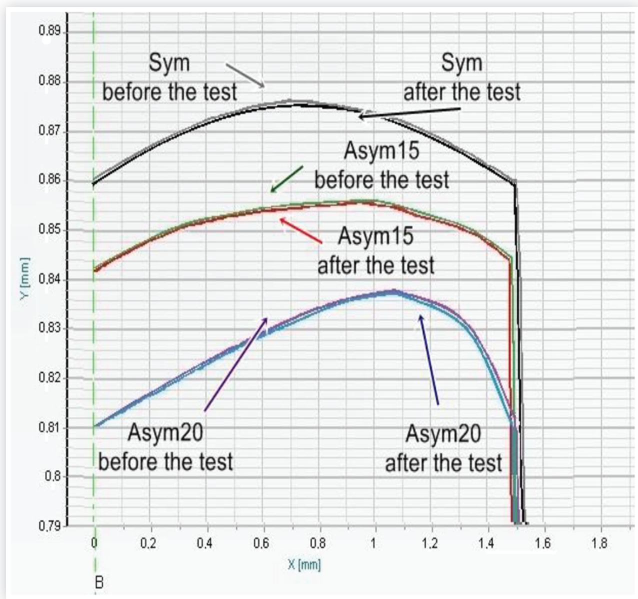


FIGURE 8 Measured profiles of the sliding surfaces of the second rings with the flame-sprayed Mo coating.

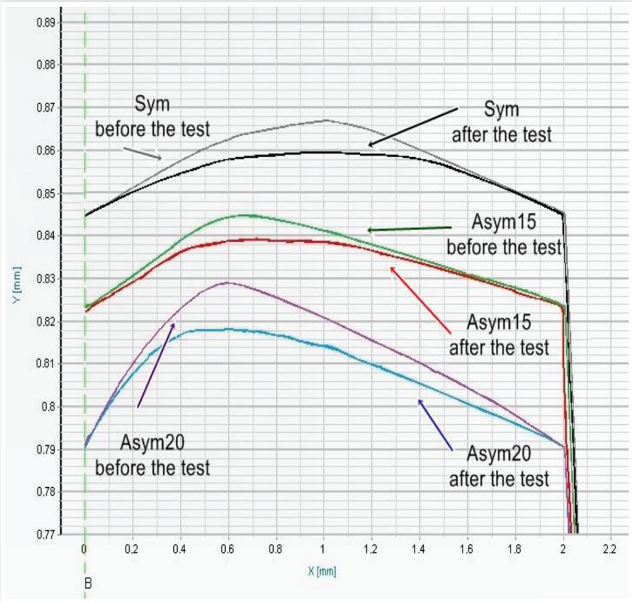
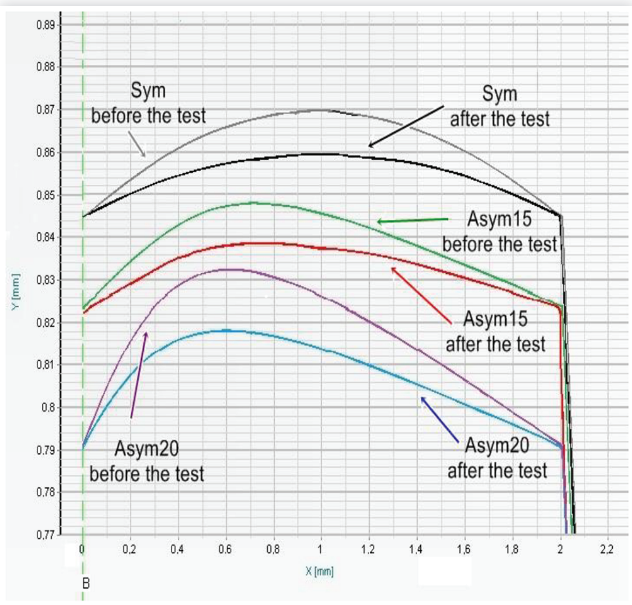


FIGURE 9 Measured profiles of the sliding surfaces of the second rings with the plasma-sprayed Mo and phosphate coatings.



3. Results and Discussion

3.1. Repeatability in Subsequent Measurement Series

The torque required to drive the engine obtained in the successive series of measurements for the given engine operating conditions (the same rotational speed and temperature) was very repeatable. The condition for such high repeatability was the performance of measurements after the rings were run in and the precise maintenance of the thermal state of the engine. Figure 11 shows the results obtained in subsequent measurement series for the Syn Cr_Mo+P ring set, for which the repeatability was the highest (Coefficient of Variation [CV] = 0.24%), and Figure 12 for the Asym15 Cr_Mo set, for which this repeatability was the lowest (CV = 0.55%). The measure of repeatability was the average value of the CVs (the ratio of the standard deviation to the mean value) determined for the given engine operating conditions.

The torque does not tend to increase or decrease for the subsequent measurements. This confirms that the measurements were carried out on the run-in rings. Therefore, in the further part of the work, the influence of engine operating conditions, ring coating, and sliding surface geometry on friction force was analyzed on the basis of average values calculated from measurements obtained in the five series.

3.2. Effect of Engine Rotational Speed

The motoring torques averaged for the five series for all the tested sets of rings are presented in Figures 13 to 15. The changes in the torque as a function of the rotational speed at a given temperature were very similar for all of the tested sets. At a temperature of 50°C, the torque increased almost linearly with the rotational speed. At temperatures of 80°C and 100°C, the greatest torque also occurred at the highest rotational speed and decreased with its drop, but only to the speed of approximately 1400-1500 rpm. At these speeds, the torque reached the minimum value, and a further decrease in speed increased the torque.

At temperatures of 80°C and 100°C, the course of the torque as a function of rotational speed is similar to the Stribeck curve. Such a course indicates that at lower speeds and higher temperatures, the share of mixed or boundary friction was significant, and this share increased with the lowering of the rotational speed. The increases in torque with the decreasing speed at the temperature of 100°C were faster than at the temperature of 80°C. This is justified because, at a higher temperature, and thus the lower oil viscosity, the expected thickness of the oil film is less, and the share of mixed and boundary friction should be greater.

FIGURE 10 Measured profiles of the sliding surfaces of the second rings with the TiN coating.

3.3. Effect of Oil Temperature

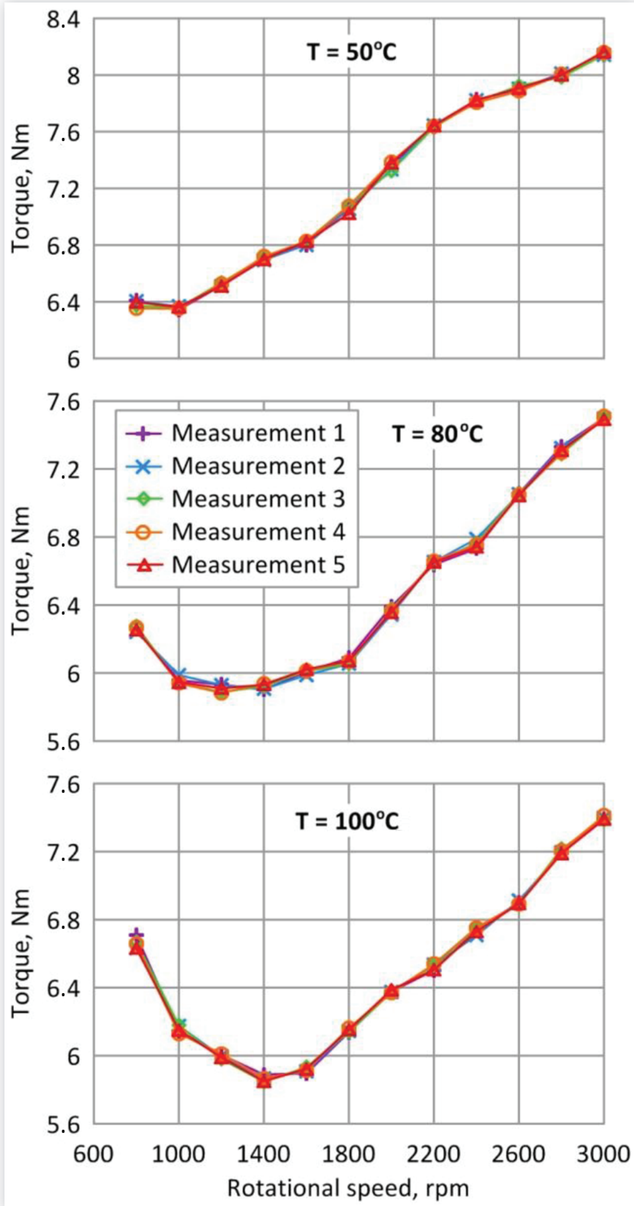
In the range of high speeds, the torque required to drive the engine decreased with the rising temperature. However, that decrease was much greater when the temperature changed from 50°C to 80°C than when it changed from 80°C to 100°C. Much the same occurred at medium speeds for the rings with symmetrical and Asym20 profiles. On the other hand, for the rings with Asym15 profile, the lowest torques were recorded at the temperature of 80°C. In the range of low engine speeds, the highest torques were obtained at the temperature of 100°C for all sets of rings. Within this speed range, the motoring torques at temperatures of 50°C and 80°C were comparable.

Increasing the temperature means reducing the viscosity of the oil. Therefore, if the friction were fluid, the torque should decrease with increasing temperature. This is the case only in the high and partly medium speed range. In the range of low speeds, the opposite is true, which indicates a large increase with temperature share of mixed or boundary friction in the total friction.

3.4. Effect of Sliding Surface Profile

Regardless of the coating, the biggest torque required to drive the engine was needed for rings with the symmetrical profile and the smallest for rings with the Asym15 one (Figures 13-15). At low engine speeds, the Asym15 profile was by far the most advantageous. In this speed range, the torques for the Sym and Asym20 profiles were similar. The higher the speed was, the advantages of the Asym15 profile in relation to the Asym20 profile decreased, and at the highest speeds, the torques for both profiles were similar. At high speeds and a temperature of 50°C, the Asym20 profile was even better than the Asym15 one.

FIGURE 11 The results for the ring set Sym Cr_Mo+P for which the repeatability of the measurements was the highest (Cv = 0.24%).



According to the hydrodynamic theory of lubrication, the thickness of the oil film and the viscous friction force depend both on the shape of the sliding surface and the dynamic viscosity of the oil, the relative velocity of the ring in relation to the cylinder, and the ring radial load. In the conducted tests, the rotational speed of the crankshaft and the oil temperature were kept very precisely, so for the given engine operating conditions, the relative velocity of the rings and the oil viscosity did not depend on the tested set of rings.

FIGURE 12 The results for the ring set Asym15 Cr_Mo for which the repeatability of the measurements was the lowest (Cv = 0.55%).

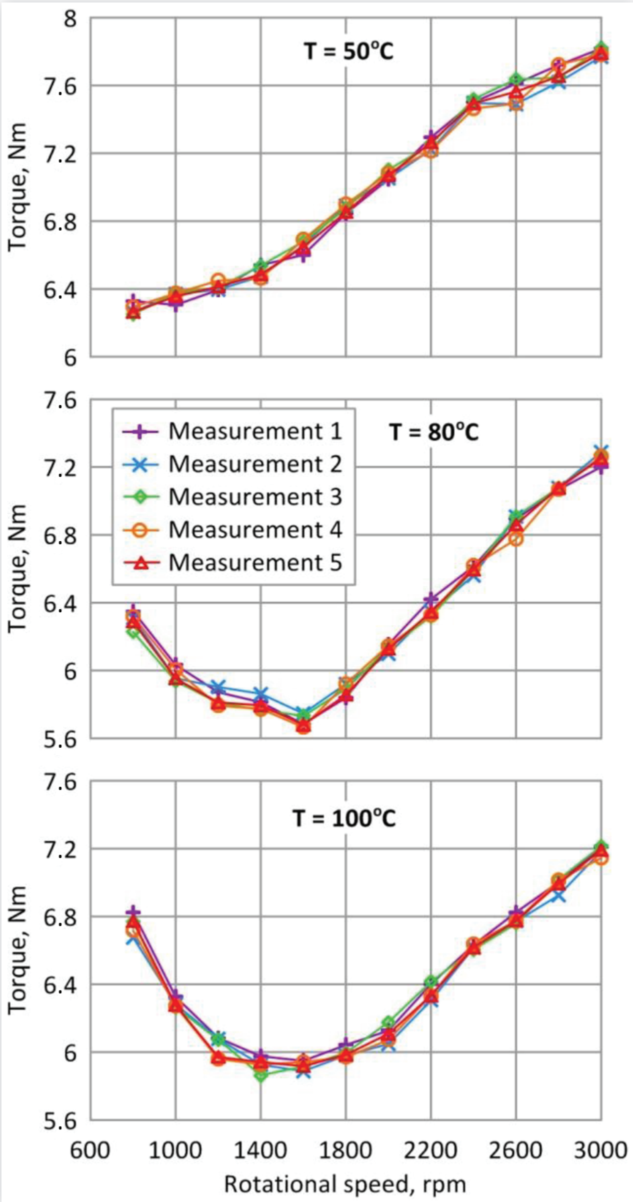
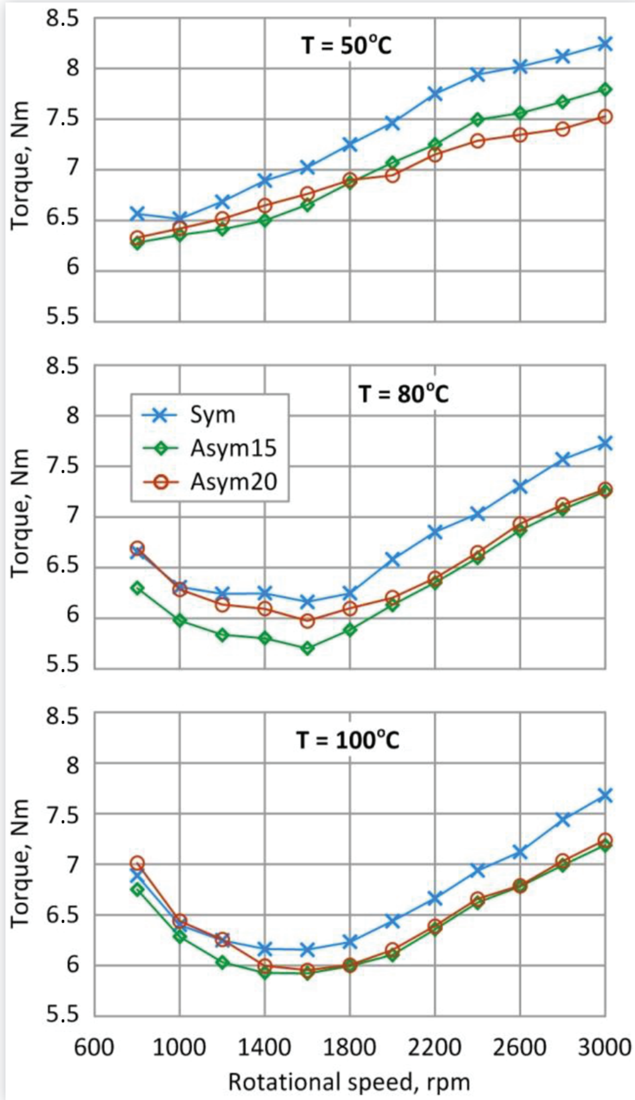


FIGURE 13 The effect of the sliding surface profile on the torque required to drive the engine for rings with Cr_Mo coatings.



The dimensions of the tested rings, and thus the axial and radial clearances of the rings in their grooves, and the ring end gaps were almost identical. Thus, it can be assumed that also the pressures in the inter-ring spaces and the pressure forces acting on the rings were the same. Taking this into account, and the fact that the tangential forces for the tested rings were very similar, it can be concluded that also the load resulting from the ring's own elasticity and the pressure force acting on the ring in the radial direction also did not depend on the tested set of rings.

3.5. The Effect of the Coating

The influence of the coating on the measured torque for individual ring profiles is presented in Figures 16-18. This influence is obvious because for all profiles and all operating conditions the lowest torques were obtained for rings with TiN_Tin coating and the highest for rings with Cr_Mo coating.

With the assumption that the coating affects the friction force only when the surface asperities are in contact, the obtained results indicate that mixed friction occurred in all of the tested engine operating conditions. The share of mixed friction, and hence the differences between the coatings, should increase with the increasing temperature and decreasing speed. Indeed, the smallest differences between the rings with different coatings occurred at the highest speeds

FIGURE 14 The effect of the sliding surface profile on the torque required to drive the engine for rings with Cr_Mo+P coatings.

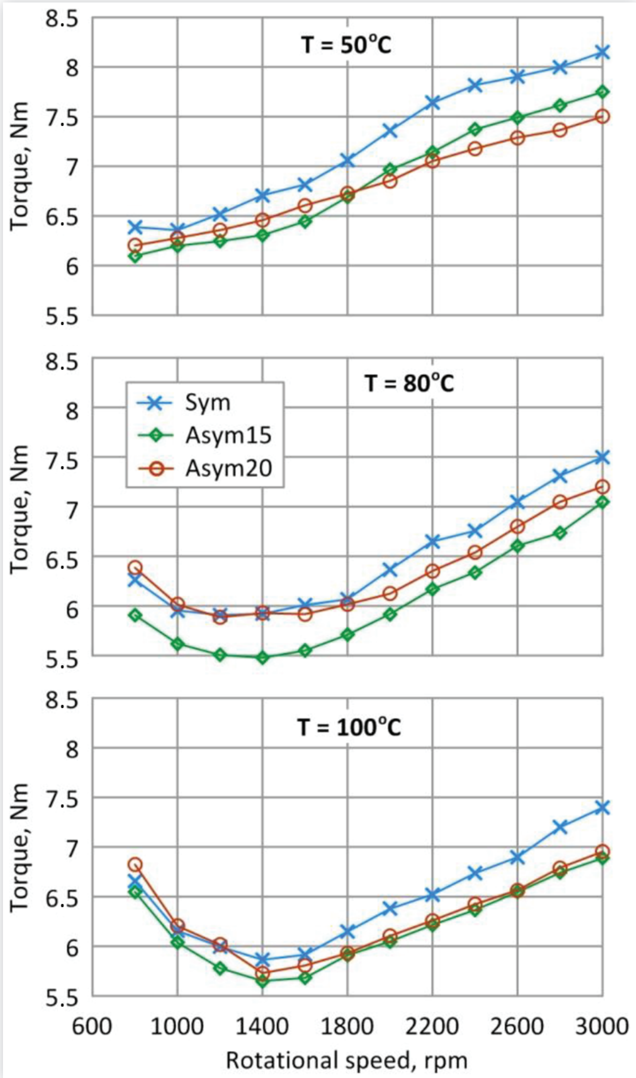
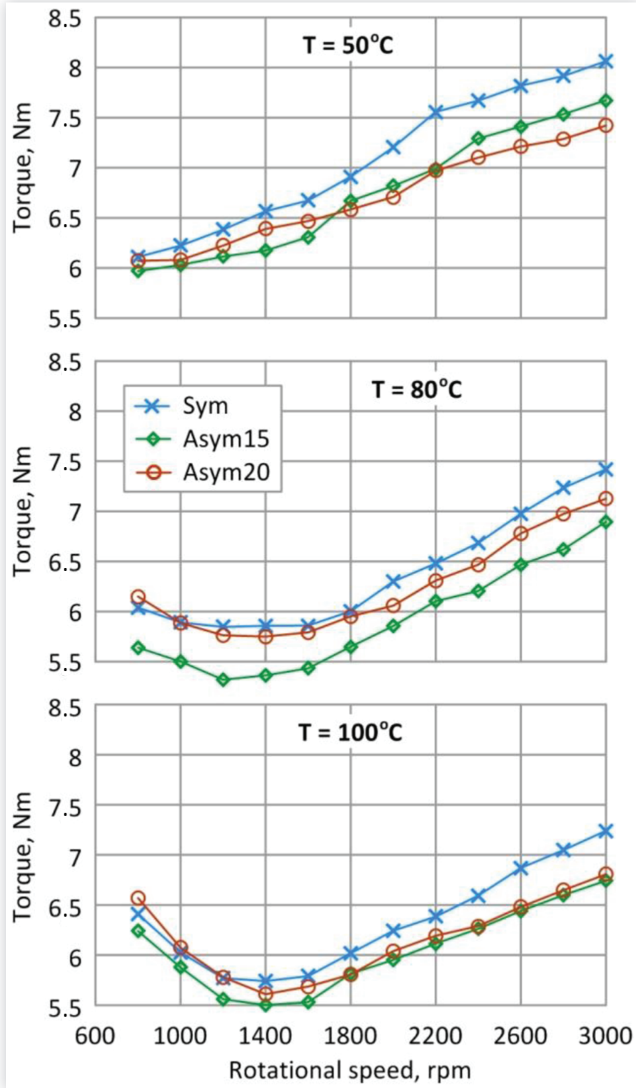


FIGURE 15 The effect of the sliding surface profile on the torque required to drive the engine for rings with TiN_TiN coatings.



and the temperature of 50°C. However, in the range of low and medium speeds and temperatures of 80°C and 100°C, the results do not indicate clearly that the effect of the coating increases with the decreasing speed and increasing temperature.

3.6. Overall Effect of Profile and Coating

In order to facilitate the quantitative assessment of the effect of the profile and coating on the rings' frictional losses, the average of the measurements made at all speeds for a given set of rings at a given temperature was calculated. The results are presented in Figure 19.

At 50°C, the Asym20 profile was the most favorable for all of the coatings. On the other hand, at the temperatures of 80°C and 100°C, the Asym15 profile was the most advantageous, and at 80°C this profile was definitely the most beneficial. On average, the torque required to drive the engine for this profile was 5% lower than for the symmetrical profile. For the Asym20 profile, the torque was over 3% lower than for the symmetrical one.

For all the profiles and temperatures, the lowest torque required to drive the engine was needed for the TiN coating, and on average this torque was 4% lower than for the Cr_Mo coating. The torque for the Cr_Mo+P coating was lower by 2.5% in comparison to the Cr_Mo coating.

The smallest friction force was achieved for the Asym15 TiN set of rings, and the motoring torque for this set was 9% lower than for the Sym Cr_Mo set. Very good results were also obtained for the Asym15 Cr_Mo+P set, for which the torque was respectively lower by 7%.

FIGURE 16 The effect of coating on the torque required to drive the engine for rings with the symmetrical profile.

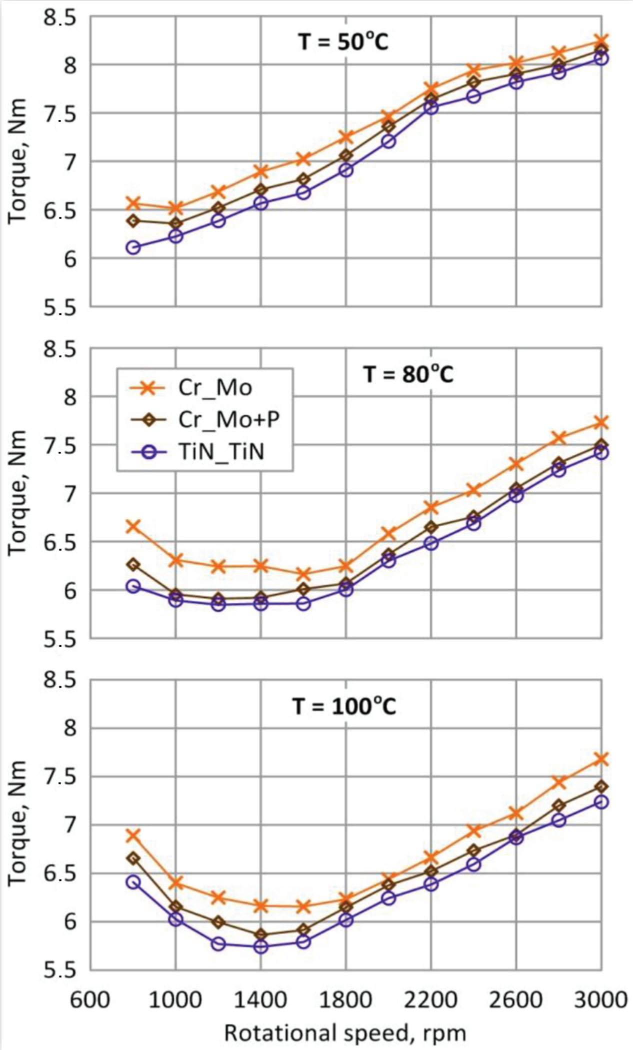
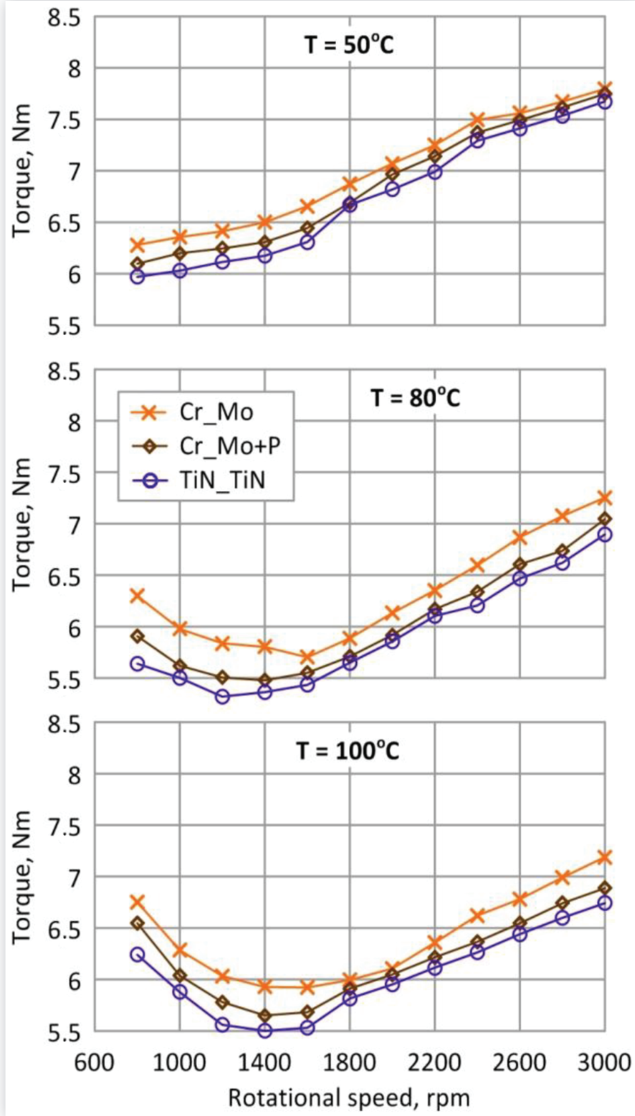


FIGURE 17 The effect of coating on the torque required to drive the engine for rings with the Asym15 profile.



In a normally running, warm engine, the friction conditions are probably the closest to those at the temperature of 100°C during the conducted tests. For this temperature, the motoring torque for the Asym15 TiN set of rings was by 8%, and the torque for the Asym15 Cr_Mo+P set was 6% lower than for the Sym Cr_Mo set.

4. Conclusions

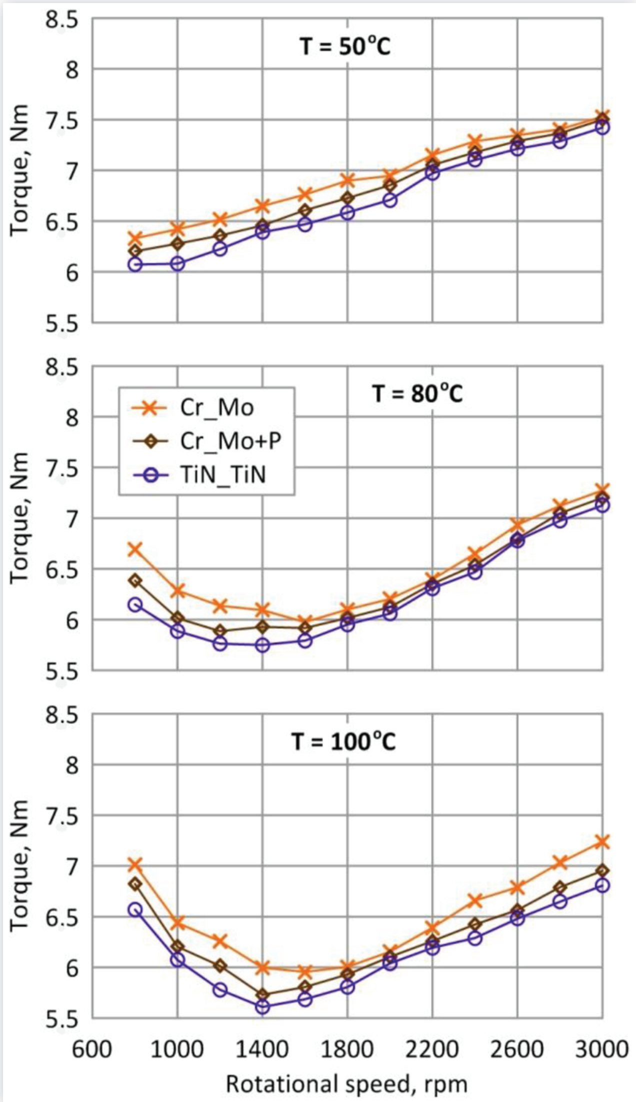
The impact of the shape of the sliding surface and anti-wear coating on the friction of compression rings was experimentally investigated. The tests were carried out using a motoring method on a specially adapted engine in which all unnecessary components were removed.

The tests were carried out on nine sets of compression rings, differing in the profiles of sliding surfaces and coatings. Three profiles were tested: symmetrical, moderately asymmetrical, and strongly asymmetrical. Moreover, the rings of each of these profiles had three types of coatings: the Cr-plated top ring and Mo flame-plated second ring, the Cr-plated top ring and Mo plasma-plated and additionally phosphated second ring, and both rings with TiN coatings.

The lowest friction was provided by the profile with moderate asymmetry and the highest by the symmetrical profile. The torques needed to drive the tested engine with the rings with the abovementioned profiles differed by 5%.

The lowest friction was achieved in the case of TiN coatings. The average motoring torque for these coatings was

FIGURE 18 The effect of coating on the torque required to drive the engine for rings with the Asym20 profile.



4% lower than for the coatings for which the highest torque was required (the Cr-plated top ring and Mo-plated second ring). Thus the effect of the coating on the friction force was only slightly smaller than the effect of the sliding surface profile.

The obtained results indicate that mixed lubrication occurs even at high rotational speeds and low oil temperatures. At lower speeds and higher temperatures, the significance of mixed friction is greater.

The tendency over the years to use oils with increasingly lower viscosity increases the likelihood of boundary friction and certainly increases the share of mixed friction in the total friction of the rings. The introduction of such oils should be accompanied by changes in the geometry of the sliding surface and the use of coatings that reduce both friction and wear. The results of these tests show that the possibilities of reducing the mechanical losses of the engine are significant here. It seems that the available technological possibilities in this area are not effectively utilized.

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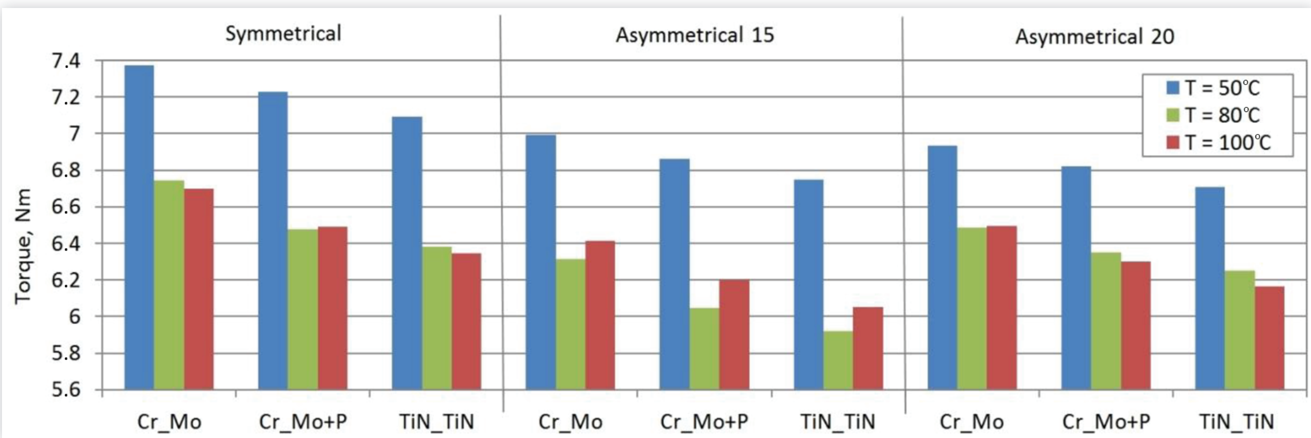
FIGURE 19 Average torques required to drive the engine for all tested sets of rings at different temperatures.



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