ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1136849

THE PROPERTIES OF DAMPING VIBRATIONS BY A LAYER OF CARBON NANOTUBES ON THE LATERAL SURFACE OF A PISTON

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Abstract

As a result of applying the hydrodynamic lubrication theory in the piston-cylinder group of a combustion engine, the authors have obtained the opportunity for practical elimination of abrasion of this unit. However, it turns out that in the case of the engine start, especially after a longer standstill and before the oil film is formed, the piston lateral surface's micro-roughness comes into direct contact with the cylinder bearing surface. For this reason, manufacturers more and more commonly apply a special layer on the lateral surface of a piston, which decreases the friction force and eliminates very damaging effects of the so-called semi-dry friction. It turns out that applying an enriching layer on the lateral surface of the piston may result in an additional effect, which is not usually associated with the piston-cylinder unit. This effect is the damping of torsional vibrations. Thus, despite the seemingly insignificant properties of materials which the elements of the kinematic pair are made of, as a result of the occurrence of the layer with specific properties on these elements, one may find that these layers – even when they are separated by the oil film – can change the parameters of collaboration of the elements of the kinematic pair. In particular, it concerns generating and damping vibrations, and in such circumstances frictional losses may also be subject to change, which is the main domain of the conducted research presented in this paper. The article analyzes various effects which are caused by applying a special layer formed of carbon nanotubes (NCT in short) on the lateral surface of the piston.

Keywords: *carbon nanotubes, combustion engine, friction losses, vibrations*

1. Introduction

Damping torsional vibrations of the engine shaft is affected by diverse properties of materials such as, for instance, the shaft deformation hysteresis, the viscosity of the oil lubricating the shaft bearings, and the viscosity of the oil lubricating the cylinder bearing surface. According to the results of various research, the share of viscous force on the cylinder bearing surface in damping torsional vibrations of the shaft is usually the largest. It turns out that within this unit one can also obtain an additional damping effect by applying a composite layer with special properties resulting from viscous-elastic characteristics of each material on the lateral surface of the piston. Usually, however, for the majority of materials it is either viscosity or elasticity that becomes dominant. These properties are interrelated mathematically in the Maxwell model as the following formula (1):

$$\dot{\gamma} = \frac{\tau}{\eta} + \frac{\dot{\tau}}{G}, \tag{1}$$

where:
$\dot{\gamma}$ – movement angular velocity of laminar layers of material,
τ – unit tangential force,
η – coefficient of dynamic viscosity,
$\dot{\tau}$ – rate of change in unit tangential force,
G – modulus of tangential elasticity.

It turns out that in the nanoscale composite structures can exhibit both viscosity and elasticity, wherein both properties being expressed by the coefficient of viscosity and elasticity are comparable. It gives the possibility of obtaining an additional damping effect in the piston-sleeve kinematic pair [1, 3, 4]. The authors have performed a comparative study of an engine equipped with pistons without the additional layer on their lateral surface, and an engine whose pistons were enriched with a layer of carbon nanotubes. The results of these studies unambiguously show a decrease in the amplitude of torsional vibrations of the engine shaft in the case of the pistons coated with the layer of carbon nanotubes. This article presents the results, which show a noticeable impact of a layer of CNTs on the parameters of the co-operation between the piston and the cylinder bearing surface, including the change of friction forces in particular, despite the existence of a permanent oil film between these components.

2. Test bed

The station used in the study consisted of the main elements such as a piston combustion engine coupled with an electric machine. A general view of the station is shown in Fig. 1.

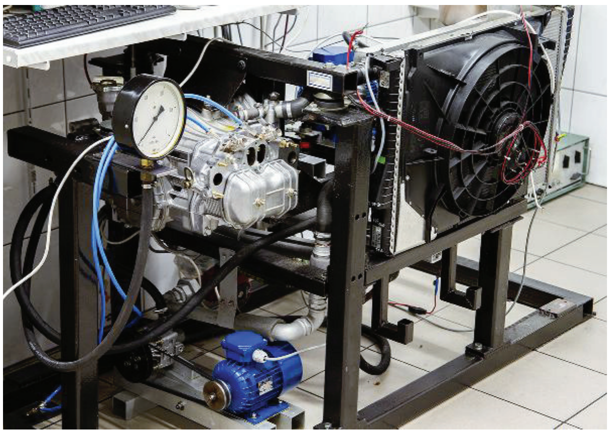

Fig. 1. A general view of the test station; in the foreground there is the tested 2-cylinder combustion engine, in the top-right corner there is the outer cooling fan of the temperature stabilizing system, the electric machine is partly overshadowed by the fan

Torsional vibrations in the combustion engine – power receiver unit result in the generation of a torsional moment between them, which is substantially different from the moment resulting from the ongoing cycles of a piston combustion engine. Fig. 2 shows the course of the unit tangential force generated by a 2-cylinder combustion engine obtained by a computer simulation under the conditions similar to engine idle running.

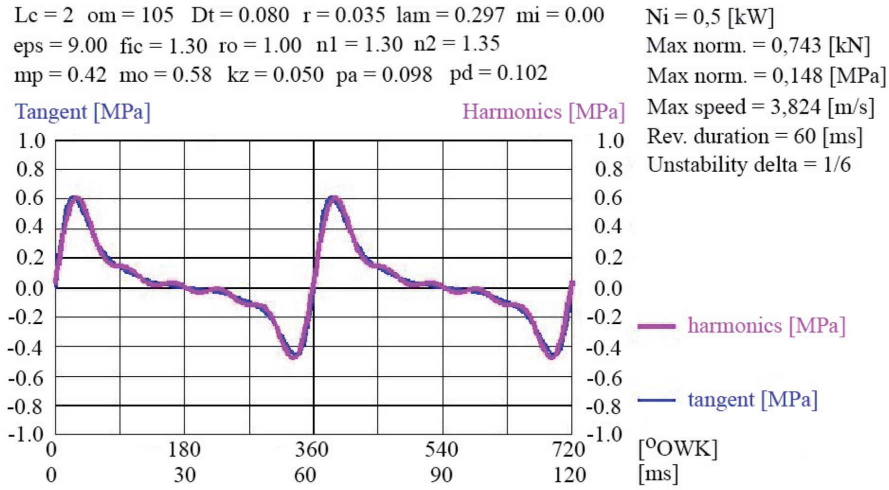

Fig. 2. The computational course of the unit tangential force generated by a 2-cylinder engine – the course marked with the blue line and the course of the sum of 10 harmonics – marked with the pink line

Here, the unit tangential force is equivalent to the moment, since when multiplied by the cross-sectional area of the cylinder and the radius of the crank it just gives the moment transmitted to the power receiver. In addition, Fig. 2 shows the sum of 10 harmonics, which comprise the analysed running (the pink line). The accuracy of mapping the actual course by the sum of the 10 harmonics indicated the number of harmonics that was to be taken into account during laboratory tests described in the further part of this article.

Due to the adopted low angular velocity, it is the gaseous forces that most of all determine the course of the graph of the unit tangential force. Bench testing was performed for the conditions adopted in the computational model used while constructing the graph in Fig. 2. In effect, the obtained results are shown in Fig. 3.

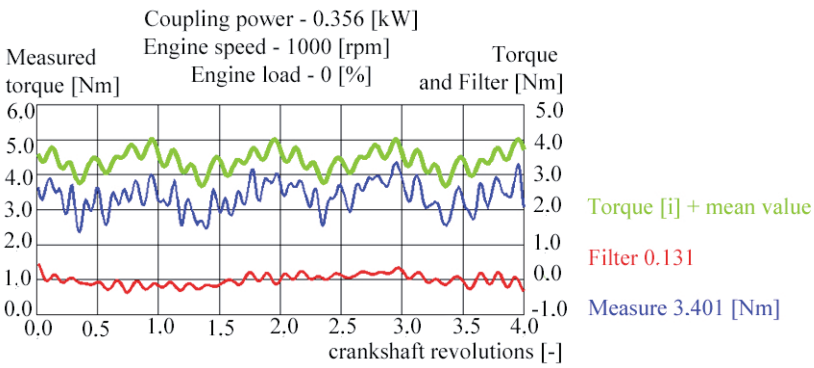

Fig. 3. The course of the coupling moment of the combustion engine with the electric machine for the conditions adopted in the computer simulation in Fig. 2 – the course is plotted with the blue line, the course of the moment after subtracting the harmonics which are not the multiple of the rotational frequency of the crankshaft – the green line, the course plotting the sum of the subtracted harmonics – the red line, angular velocity of the shaft – 1000 rpm

The blue line presents the measured coupling moment for four revolutions of the crankshaft. A comparison of the nature of the course of the tangential force in Fig. 2 and the moment in Fig. 3 shows substantial differences. First of all, one can see a less stable course obtained on the basis of the measurements. A mathematical analysis of this course proves that the course is disturbed by the harmonics, which are not the multiple of the rotational frequency of the crankshaft. On the other hand, it should be noted that the engine work in cycles makes most phenomena in this engine change cyclically with the frequency of the revolutions of the shaft. This note refers to; inter alia, gaseous forces, inertia, friction, etc. There is however, a phenomenon, whose frequency is not related to the revolutions of the shaft, and this is the shaft torsional vibration. These vibrations are determined by solid moments of inertia of the crank throws and the rigidity of the shaft segments, which connect the crank, throws. Of course, depending on the number of cylinders, there is the possibility of generating various forms of vibrations, which differ in frequency. It is unlikely, however, that the frequency of either of the forms of proper vibrations was exactly the multiple of the engine angular velocity.

One can therefore support the thesis that if a particular course harmonic shown in Fig. 3 is not a multiple of the rotational frequency, it is then the proper vibrations harmonic. Such a claim allows for determining quite accurately the harmonics, which lead to the disturbances in the repeatability of the course measured in Fig. 3, every two revolutions of the crankshaft.

A simple test that allows us to evaluate the extent to which the frequency of the harmonic of the parameter related to the engine work can vary from the multiple of the frequency of the shaft angular velocity involves the introduction of a slightly different angular velocity from its real value into the formula defining the period of the harmonic. If the real angular velocity of the shaft equalled 1000 rpm, then inserting the angular velocity of 1010 rpm leads to obtaining the course of the sum of filtered moments presented in Fig. 4, whereas if the adopted $n = 990$ rpm, then the course shown in Fig. 5 is obtained.

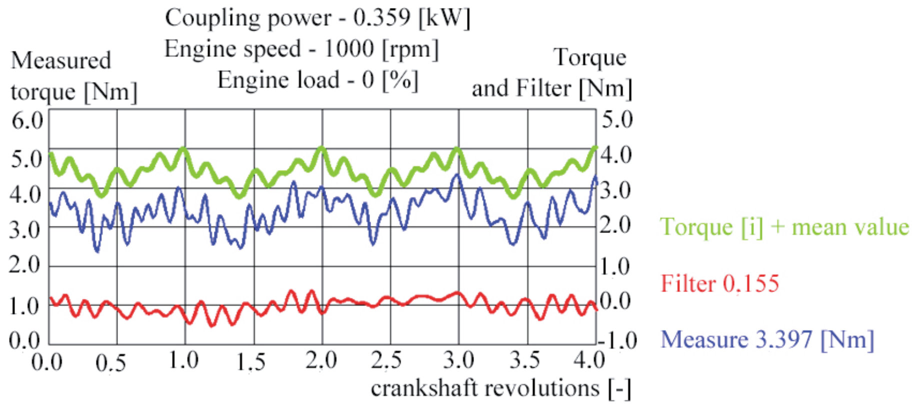

Fig. 4. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2 – the course plotted with the blue line, the course after subtracting the harmonics which are not the multiple of the rotational frequency of the crankshaft – the green line, the course plotting the sum of the subtracted harmonics – the red line, angular velocity of the shaft – 1010 rpm

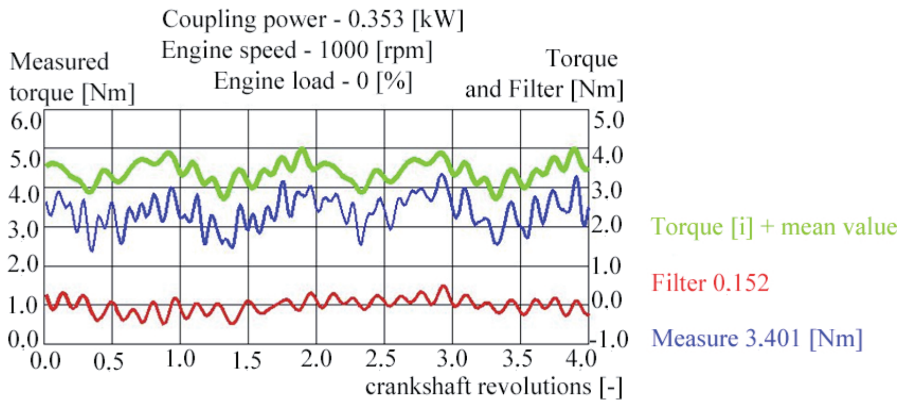

Fig. 5. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2 – the course plotted with the blue line, the course after subtracting the harmonics which are not the multiple of the rotational frequency of the crankshaft – the green line, the course plotting the sum of the subtracted harmonics – the red line; the angular velocity of the shaft – 990 rpm

Next to the description of the curve labelled 'the filter', there is the average absolute value of the filtered torque expressed in Nm. The higher the value of the filtered torque, the greater the uncertainty of the obtained result. A comparison of the values of the filtered torque for the angular velocities of 990, 1000 and 1010 rpm (amounting to 152, 131 and 155 Nm respectively) clearly showed how important it is to precisely determine the true value of the angular velocity, as well as how accurate the proposed method of defining the coupling moment of a combustion engine and an electric machine is.

Because the coupling moment determined at the test station is the sum of friction losses in many units, the friction losses in the piston-cylinder unit, which are within the scope of interest of the authors, are only a minor part of the total losses. Their change caused by coating the lateral surface of the piston with a layer of nanotubes constitutes – in the extreme case – a dozen or so per cent of the total value of friction losses. Hence, the necessity for a very precise determination of changes of friction losses resulting from the improvements made.

In order to estimate the accuracy of measurements at the test station, the repeatability of the obtained results under constant experimental conditions was tested. Therefore, the average coupling power was determined at time intervals of 0.1 s. Sample results are shown in Fig. 6-8.

In the recording time period chosen at random, similar values of the coupling moment of the combustion engine with the electric machine were obtained. It means high repeatability of the measured coupling power with the differences that do not exceed 2% of the value of this physical quantity. It can therefore be assumed that for two types of the cylinder bearing surface finish, the differences in the coupling power of the value higher than 2% indicate a substantial impact of this finish on friction losses being the derivative of the coupling moment.

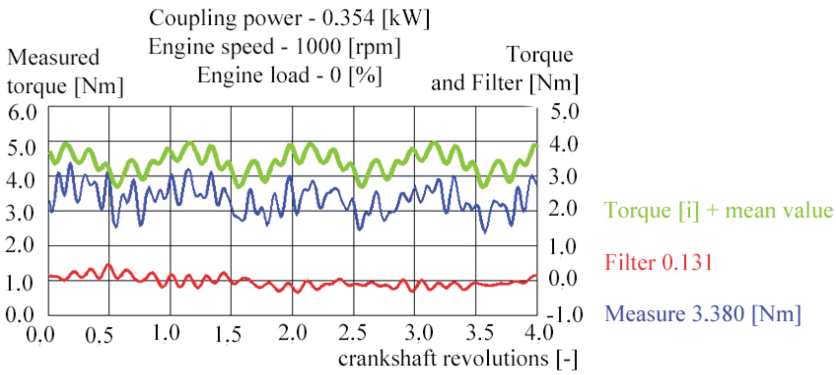

Fig. 6. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2 – the course plotted with the blue line, the course after subtracting the harmonics which are not the multiple of the rotational frequency of the crankshaft – the green line, the course plotting the sum of the subtracted harmonics – the red line, the angular velocity of the shaft – 1000 rpm, the measurement made 0.1 s after the start of recording

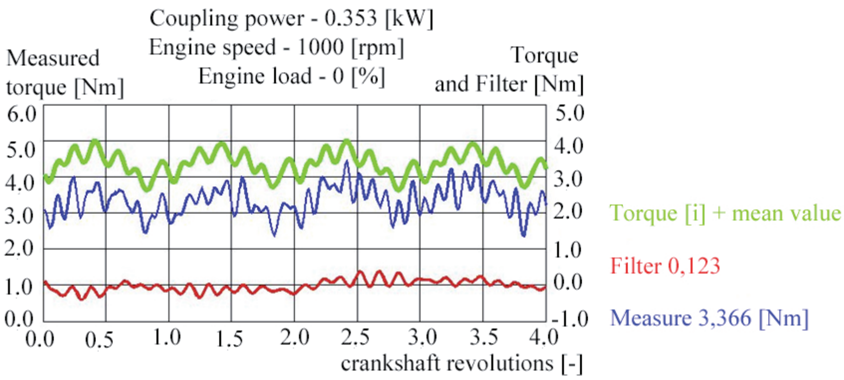

Fig. 7. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2 – the course plotted with the blue line, the course after subtracting the harmonics which are not the multiple of the rotational frequency of the crankshaft – the green line, the course plotting the sum of the subtracted harmonics – the red line, the angular velocity of the shaft – 1000 rpm, the measurement made 0.2 s after the start of recording

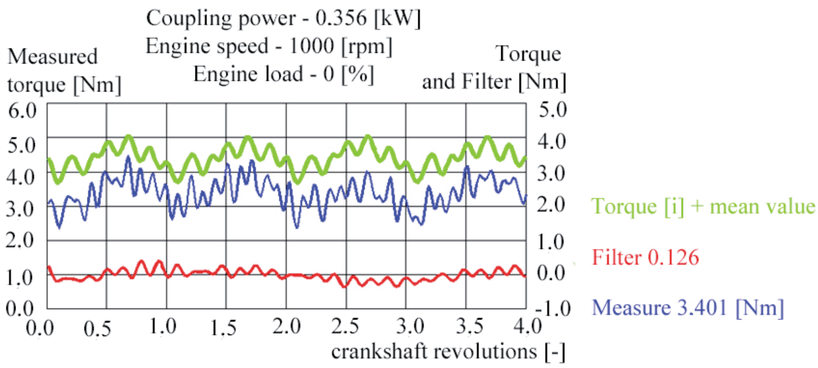

Fig. 8. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2, the angular velocity of the shaft – 1000 rpm, the measurement made 0.3 s after the start of recording

3. Preliminary research results of the impact of the piston lateral bearing surface finish on friction loss

Two sets of factory pistons were selected for comparative studies. Measurements were performed for the pistons without the layer of nanotubes on their lateral surface and the pistons on which such a 5 μm-thick layer was synthesized. As a result of coating the lateral surface of the pistons with a layer of nanotubes, the piston assembly clearance was reduced by 10μm in diameter. The tests of the coupling moment generated by the pistons coated with the layer of nanotubes were performed after 10, 30 and 50 min from the starting position. The results obtained in the presented time intervals differed significantly, wherein initially the changes in the values of the coupling moment generated by friction were quite intensive, and after 50 minutes' working, the results stabilized. Fig. 9–11 show the course of the coupling moment in 20-minute time intervals.

In the case of the pistons without the covering layer of nanotubes, the course of the coupling moment presented in Fig. 3 was obtained, whose first line of the description includes friction power resulting from the obtained course of the coupling moment. The resulting friction power amounts to 359 W. On the other hand, the friction power obtained for the pistons coated with a layer of nanotubes was 420 W initially, and after 20 min, the power decreased to the value of 393 W, and after additional 20 min to the value of 374 W. The obtained values of friction power show that covering the lateral surface of pistons with a layer of 5 μm-thick nanotubes does not lead to a reduction in friction losses. Once the pistons coated with nanotubes are run in, the friction power remains slightly higher than in the case of the pistons without the nanotubes coating; this power amounts to 374 W and is 15 W higher than for the pistons without the nanotubes coating.

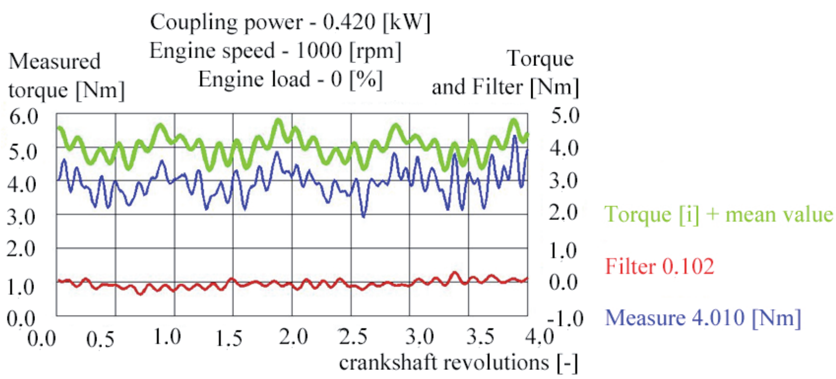

Fig. 9. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2, the angular velocity of the shaft – 1000 rpm, the measurement made 10 min after starting the test station

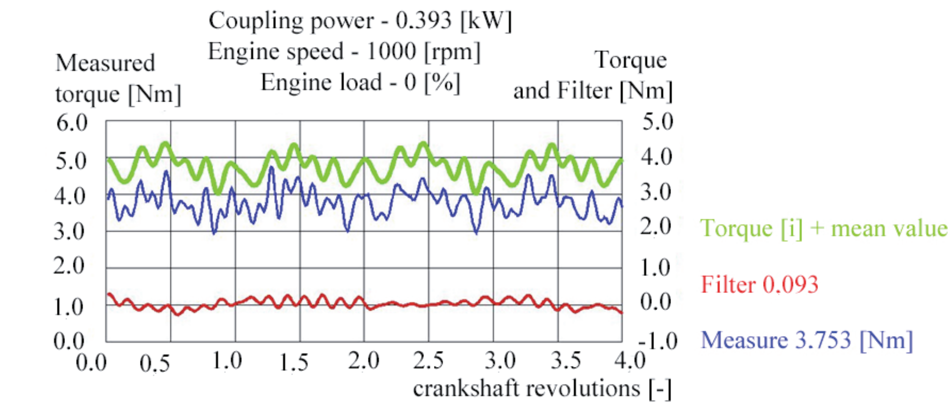

Fig. 10. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2, the angular velocity of the shaft – 1000 rpm, the measurement made 30 min after starting the test station

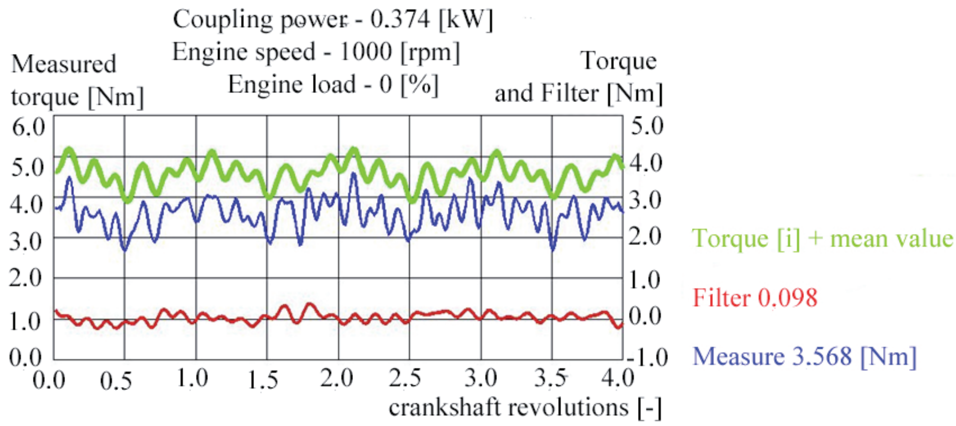

Fig. 11. The graph presenting the torque of coupling a combustion engine and an electric machine for the conditions adopted in the computer simulation in Fig. 2, the angular velocity of the shaft – 1000 rpm, the measurement made 50 min after starting the test station

4. Summary

The obtained results of friction power prove that coating the lateral surface of pistons with a layer of carbon nanotubes which are 5 μm thick does not lead to a reduction of friction losses; on the contrary, it increases the losses slightly. Since the diameter of the pistons coated with a layer of nanotubes was 10 μm larger than the pistons without such layers, it can most likely be assumed that the increase in friction losses results from the decrease in the assembly clearance of the pistons coated with nanotubes. The expressed opinion is due to the fact that even a marginal reduction in the diameter of the pistons caused by the process of running-in results in a decrease in friction power. However, eventually, the remaining layer of nanotubes on the lateral surface of pistons increases the diameter of the piston by a value, which is difficult to assess since the micro roughness on the lateral surface of the pistons does not allow for an unequivocal definition of the thickness of the layer of nanotubes following the running in. The authors intend to continue comparative studies of pistons coated with a layer of nanotubes with pistons without such coating while retaining an identical nominal diameter of all pistons. For this purpose, out of a large batch of pistons, from two selective groups, pistons differing in the nominal diameter by 10 μm were chosen. Because the thickness of the layer of nanotubes can be controlled very precisely, it is not much of a problem to synthesize a layer of a strictly defined thickness, e.g. 5 μm. The results will be presented in subsequent publications. The performed computer simulations of friction power prove that a change in assembly clearance of 10 μm, i.e. from the thickness of 25 μm to 15 μm causes an increase in friction power of about 50 W. The presented situation allows us to assume that while keeping the unchanged assembly clearance, the pistons coated with a layer of carbon nanotubes will cause less friction losses than the pistons without such coating.

Acknowledgements

Project Nr 3374/B/T02/2009/36 u.t. *Redukcja niestabilności momentu generowanego przez silnik spalinowy w zespołach hybrydowych* is financed by Ministry of Science and Higher Education.

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