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Characterization of various coatings in terms of friction and wear for internal combustion engine piston rings

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1. Introduction

In general, the terrestrial transports and particularly automobile transport is the most important source of gas emissions detrimental to the environment.

On the one hand, the cars are responsible for most of the CO_2 emissions in the atmosphere, thus it is important to reduce these emissions. In the other hand, the manufacturing of cars is an industry which uses a large variety of materials and process, and some of those constitute an ecological or toxicological threat [1].

Indeed, the greenhouse effect of the gas emissions is mainly caused by fuel combustion in the car engines. The best method to reduce the consumption of the thermal engines is to decrease the emissions at their source. It is well-known today that one of the most effective ways to reduce fuel consumption of the vehicles is to improve efficiency of the engines. This improvement, for example, can be obtained by the reduction of friction losses [2].

Most of the mechanical friction power loss in the internal combustion engines occurs at the contact liner/piston rings and liner/skirt [3]. The losses due to friction of the piston ring/cylinder liner assembly account for approximately 40% of the total losses [4].

As a matter of fact, top piston ring of the engines is very often covered with a hard chromium coating to ensure its longevity; this coating is obtained via an electrolytic process which requires the use of hexavalent chromium ions. This product is classified as toxic [1].

Thus, the interest of this choice to work on the reduction of friction in the contact between the top piston ring and the cylinder liner is double since we will be able through this study to also contribute to reduce the ecological and toxicological impact of the manufacturing process of these parts of the engine.

This study consists in making an approach to comprehension of the problem and to knowledge of the behavior of various coatings. In this sense, some alternative solutions, which can give a reduction of the friction in the piston-ring/cylinder liner contact and a contribution towards a better respect of the environment, were tested.

2. Importance of the selected parameters

The principal parameters of operation affecting the friction force are speed of the engine and its load [5].

Studies showed that unit piston ring/piston/liner has various modes of lubrication [6, 7], with a mixed lubrication behaviour around the top dead center (TDC) and the bottom dead center (BDC), and with a hydrodynamic lubrication when speed of the piston is sufficient [5]. Therefore increase in the engine speed involves an increase in hydrodynamic friction and reduction in mixed friction. In term of power loss, mixed friction localised at the TDC and BDC have a small effect due to the low speed of the piston but overall, friction level tends to increase with the speed of the engine [5].

The impact of friction reductions, following the changes of the operating conditions, is important in fuel consumption at weak load, and is less important at full load. At full load, the friction losses are reduced to less than 10%. Also, 50% of the friction reduction in piston ring/piston/liner will reduce the fuel consumption up to 40% at the idle and only to 2% in full load [5].

The influence of load is large at the ends of race in particular near the TDC. The speed of the engine at the same point does not have an influence [6]. To the neighbourhoods of the medium, the force of friction tends to increase considerably with the increase of the engine speed while the influence of load is relatively small [6].

About the aspect material, the surface treatments, in particular the unconventional processes developed recently, are the largest allies of metals since they contribute to mitigate their defects by deposit of a protective material.

The application of hard and thin deposits on engine parts would make it possible on the one hand to increase wear resistance of the components and on the other hand to decrease the losses of energy by friction, which involves savings in fuel and limitation of the rejections.

For many years it has been possible to deposit a wide range of hard wear-resistant coatings on to steel substrates using different techniques [8].

Varieties of coatings were developed and employed such as coatings based on MoS2 or carbon (graphite) [8] or nitriding [9]. The last process is extensively used in automotive industry due to excellent wear resistance of the nitrided layer and the superior fatigue life of nitrided parts [9].

Coatings Diamond Like Carbon (DLC) are also used. They are hard and produce a lower friction compared to the nitrided hard coatings [10]. The PVD Me-C:H is a type of DLC coating which was tested on a great number of automobile parts during the last decades [11].

Exhaust gas recirculation (EGR-system) technique gives a decreased burning temperature that leads to decreased emissions of nitrogen oxides but it creates problems of soot, which always increases the request for piston rings and liners materials resistant to wear [11].

Actually, High velocity oxy-fuel (HVOF) flame spraying technique is a competing technology to several other surface modification technologies. The HVOF process is still under intensive discussion [12].

In the present work, one will limit oneself to study only the influence of the factor "coating" on tribology of the contact top piston ring/liner under "different operating conditions".

3. Experimental description

3.1. Experimental equipment

The comparative studies of influence of the contact materials between the top piston ring and cylinder liner are obviously very difficult to realise in their real environment. The tribological interactions encountered on a piston ring/cylinder liner contact in an ignited engine are difficult to reproduce in a laboratory system (chemistries of the lubricants, combustion gases, etc.). So, the use of simpler laboratory test rigs is required.

With this aim, the use of the Cameron – Plint TE77 Tribometer (Fig. 1) is very well suited because it allows samples taken from real parts to be used, and its kinematics is close to the operating conditions of a piston ring in an engine. The normal load W on this machine can be varied between 10 and 500 N and the rotational frequency f from 0 to 20 Hz, corresponding to linear velocities between 0 and almost 1 m/sec.

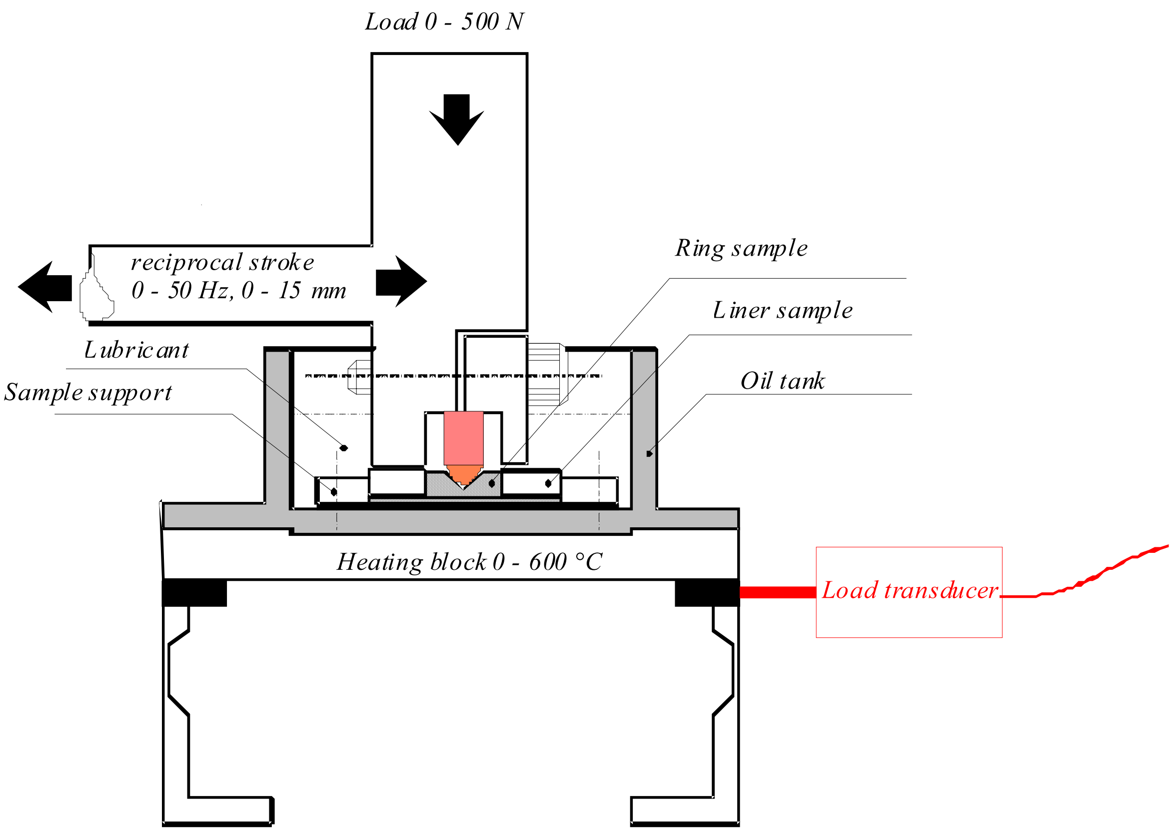

Fig. 1 Tribometer PLINT

3.2. Specimens

Table 1 gives the list of five coatings of the top piston ring to be tested the configuration of which is typical of a European 2.0 liters turbocharged diesel engine PSA DW10.

Cylinder liner test samples are cut in real cylinder liners (Fig. 2) and the test samples of piston rings are also cut in a compression ring (Fig. 3) of an engine. The diameter of each part is measured and suitable combinations

Table 1

Coatings used for the tests

Coatings	Specifications
A	Cast iron + electrolytic chromium coating reinforced by alumina (reference coating) (CKS36)
B	Cast iron + electrolytic chromium coating reinforced by diamond (GDC50)
C	Cast iron + HVOF with coating (WC-CrC) (MKJet502)
D	Lamellar Cast iron without coating
E	Spheroidal Cast iron + chromium nitride PVD coating (CrN PVD)
F	Ion nitrided alloyed steel

(couples) of the piston rings and liners samples for each test should be found so that all the tests will be done in the same configuration. Good couples of liners and piston-rings have been obtained.

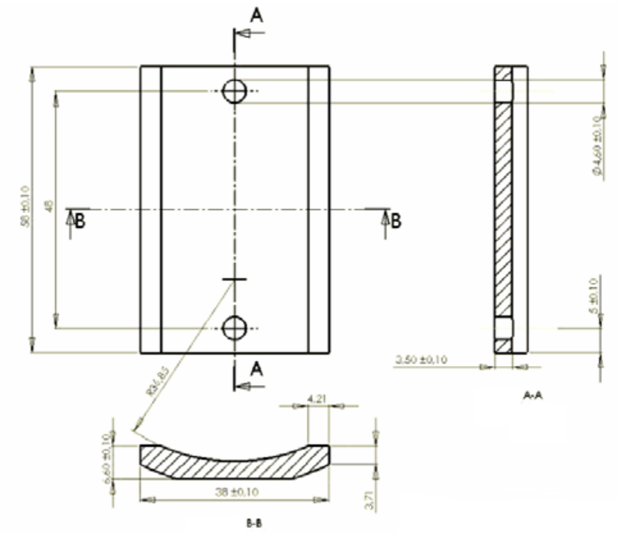

Fig. 2 Specimen of cylinder liner

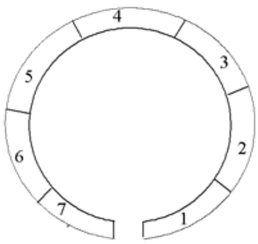

Fig. 3 Ring samples

3.3. Tests procedure

The procedure of the study is divided into two. First, one wants to characterize these coatings in terms of friction, then in terms of wear.

For the study of friction, the selected conditions are determined for the construction of Stribeck curves for each pair of the piston ring and cylinder liner to. Therefore, in order to simulate the conditions encountered from the boundary regime to the hydrodynamic regime, a certain number of load and speed combinations is necessary.

The loads W used were fixed at 20, 40 and 80 N. For each of these loads, the frequencies f were fixed as follows: 0.2; 1; 4; 7 and 9 Hz.

For the tests of wear, the applied load W selected is about 90 N under a limiting operation with an oscillation frequency f of 0.4 Hz. The couple is left in friction during eight hours. The objective in this case is to know the evolution of the contact (wear) during these eight hours of friction for each couple piston-ring/liner.

The sliding stroke of the piston ring is set at 15 mm (± 7.5 mm) and is sinusoidal.

The lubricant used is a basic mineral oil (N 175) without additives (kinematic viscosity of 5 cST at 100°C). The reason to use basic oil is to eliminate the influence of special ingredients added to oil and to see only the effect of coatings on friction and wear in the same and simple configuration.This lubricant can be reasonably considered as Newtonian.

4. Results

4.1. Study of friction

4.1.1. Construction of Stribeck curve

It is possible to vary the speed and normal load over sufficiently wide ranges in order to reach, on the test rig, the boundary and the hydrodynamic regimes.

These combinations of variation of the load and/or the speed give rise to the two curves (Figs. 4 and 5) where it appears clearly that the system running conditions move gradually from a mixed lubrication regime (Fig. 4), to a lubrication regime close to a hydrodynamic one (Fig. 5).

The tests with higher loads and lower frequencies would give a rectangular shape to the tangential effort vs. displacement curve and lead to the same values of the maximum friction force as the one measured at the edges of the stroke where the sliding speed falls to zero.

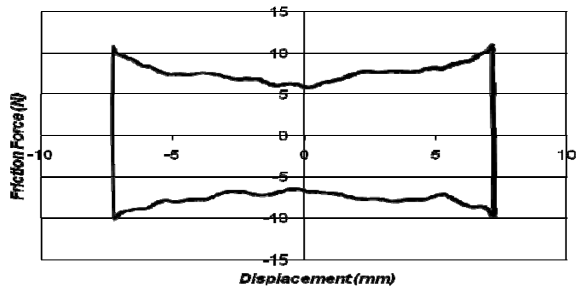

Fig. 4 Example of friction force with respect to displacement (W = 20 N, f = 0.2 Hz) for the coating C, boundary regime

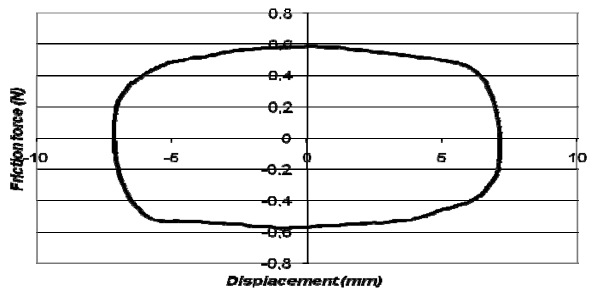

Fig. 5 Example of friction force with respect to displacement (W = 40 N, f = 9 Hz) for the coating C, hydrodynamic regime

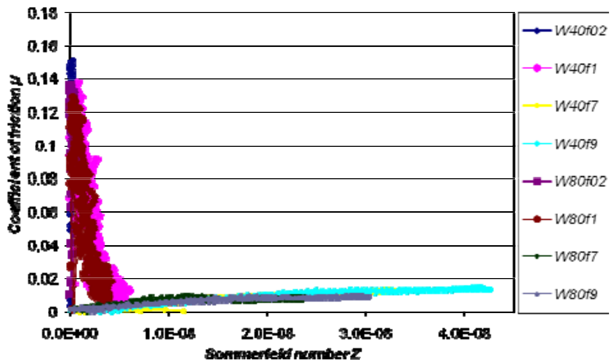

Fig. 6 Construction of Stribeck curve for coating C (load W in Newton, frequency f in Hertz)

Consequently, for each coating, these results can be gathered in a Stribeck curve as illustrated on Fig. 6.

As the load is constant, the friction coefficient μ is obtained directly by the following equation

$$\mu = \frac{F_t}{F_n} \tag{1}$$

where F_t is measured friction force; F_n is normal load ($F_n = W$).

The Sommerfeld number Z can be assessed by the following process

$$Z = \frac{\eta V}{P} \tag{2}$$

where η is dynamic viscosity of the lubricant, V is sliding speed and P is contact pressure.

As the displacement d is measured and supposed to be sinusoidal, the theoretical equation

$$d = d_0 cos\left(\omega t\right) \tag{3}$$

has been verified by fitting the experimental laser displacement data (d_0 is stroke, ω is pulsation and t is time). By a simple differentiation of the displacement vs. time, the sliding speed V can be calculated.

For the calculation of contact pressure between the piston ring and the cylinder liner, the following Hertz equation has been employed [13, 14]

$$P_{max} = \frac{2F_n}{\pi b L} \tag{4}$$

with b is the width of contact and L is the length of contact. The width of contact is calculated with help of the following equation

$$b = \sqrt{\frac{2F_n}{\pi L} \cdot \frac{\left(1-\nu_1^2\right)/E_1 + \left(1-\nu_2^2\right)/E_2}{1/D_2 - 1/D_1}} \tag{5}$$

where ν_1 is liner Poisson's ratio, ν_2 is piston ring Poisson's ratio, E_1 is liner Young modulus, E_2 is piston ring Young modulus, D_1 is diameter of the liner, D_2 is diameter of the piston ring.

By integrating the expression of b in the equation of P_{max}, we obtain

$$P_{max} = 0.798 \sqrt{\frac{F_n\left(D_1 - D_2\right)/L D_1 D_2}{\left[\left(1-\nu_1^2\right)/E_1 + \left(1-\nu_2^2\right)/E_2\right]}} \tag{6}$$

4.1.2. Comparative study of the coatings

- Case of the boundary regime.

In a first step, the study of various coatings can be simplified by observing the levels of friction forces expressed by means of friction coefficients in the only boundary regime. With this aim, the curves representing the tangential effort/displacement relation like the one on Fig. 4 or the curve of Stribeck can be used.

Table 2 summarises the mean values of the friction coefficient in the boundary regime. From these values, one can already notice that the piston ring with the HVOF coating (C) produces the lowest friction force in the boundary regime. It is slightly lower than the reference piston ring covered with the reinforced ceramic-chromium electrolytic coating (A).

Table 2

Values of friction coefficients in the boundary regime

Coating	A	B	C	D	E	F
Mean value of the boundary friction coefficient μ	0.145	0.204	0.142	0.212	0.157	0.147

In fact, one finds two families of results, one whose level of the friction coefficient is located around a maximum value of 0.15 and the second family whose value of the friction coefficient is almost equal to 0.2.

In the first family there are the HVOF coating C and the reference chromium coating A, nitrided steel F and the PVD chromium nitride coating E.

In the second family, one finds the Cast iron without coating D and diamond reinforced chromium coating B.

- Case of the hydrodynamic regime.

In the middle of the stroke where the speed is maximum, and with appropriate test conditions (light load, high frequency), a more or less thick oil film can be generated, leading to a nearly hydrodynamic regime. One can thus study the evolution of friction level in this area in order to evaluate the more or less great aptitude of the considered coatings to generate or maintain such an oil film.

In order to place themselves in an unquestionable way in hydrodynamic regime, one can, for example, establish the evolution curves of the friction force according to the load for a given speed or a graph illustrating the evolution of this same friction force according to the speed for a given load, and this in the middle of the stroke where the sliding speed is maximum.

The curves representing the maximum effort of friction arising in the middle of the stroke can be plotted according to the sliding speed for the various level of load (curves of Figs. 7, 8 and 9).

It can be seen on these curves that the HVOF coating C (in particular), the reference coating A and the nitrided steel F exhibit a hydrodynamic behaviour in the

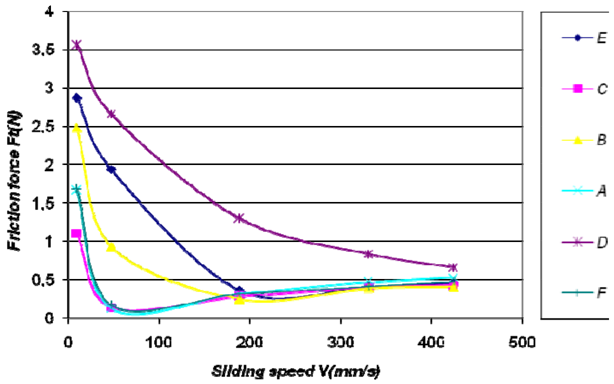

Fig. 7 Variation of the friction force with respect to sliding speed, load W of 20 N

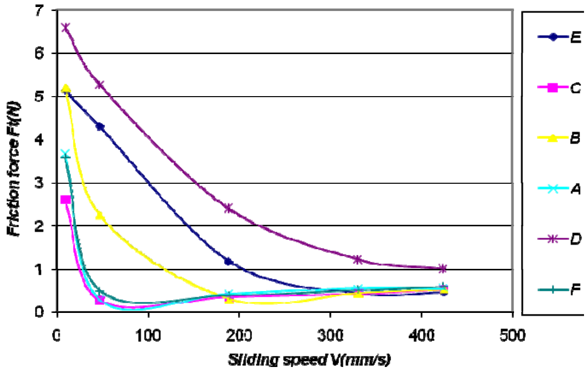

Fig. 8 Variation of the friction force with respect to sliding speed, load W of 40 N

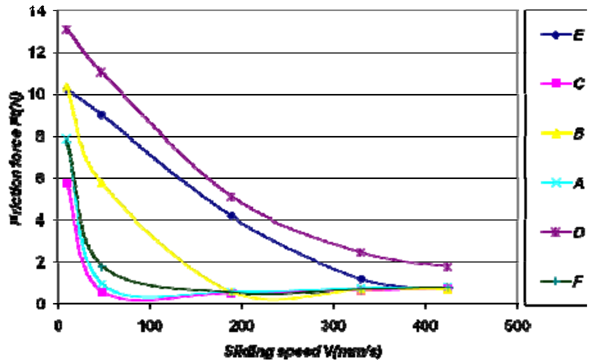

Fig. 9 Variation of the friction force with respect to sliding speed, load W of 80 N

middle of the stroke for the speed of 75 mm/sec for almost all loads. The diamond reinforced coating B pass in hydrodynamic regime only for the higher speed running conditions (200 mm/sec); the PVD CrN coating E reaches hydrodynamic regime from the speed of 250 mm/sec and the spheroidal cast iron D does not seem to reach such a regime even for a weak load of 20 N.

A first classification of various coatings according to the measured tangential force F_t can be made as follows: the HVOF coating C, the nitrided steel F and the referenced coating A, the diamond reinforced coating B, the PVD CrN coating E and the cast iron D.

4.2. Study of wear

In this study case of wear, the load W was fixed at 90 N and the oscillation speed f of ring at 0.4 Hz for a work period of the tribological couple equivalent to 8 hours and this for the whole of ring/liner combinations.

Acquisition and recording of the parameters (displacement and effort of friction) were made after two minutes of the test beginning and at the end.

At the end of continuous friction for each couple, the signs of wear were noted, and the changes of the curves shape of friction were observed and recorded (Fig. 10).

The comparison is carried in this situation on the values of friction force in the middle of stroke, by taking the values of F_t after 2 min of the test beginning and after 8 hours of service for each rubbing couple. The results are gathered in Table 3.

As for the case of the friction tests, the coating C presents during 8 hours of tests the most stable behavior compared to the other coatings with lower friction force.

Table 3

Values of Ft after 2min and 8h from the test

Coating		C	A	F	E	D	B
F_t (Mean)	After 2 min	4.543	8.017	7.189	12.122	12.741	9.957
	After 8 h	2.074	1.841	1.233	8.277	12.679	0.296

For the naked Cast iron D the situation is practically unchanged during 8 hours and with the values of Ft reaching 12 N. For the other coatings, there is CrN PVD (E) which relatively changed form passing from 12 N to 8 N. The coatings A and F have approximately the same behavior during 8 hours of permanent contact. The couple which much changed form is coating B (from 10 N until 0.5 N).

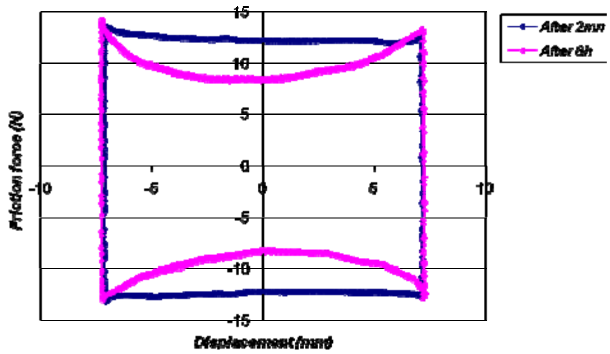

Fig. 10 Tangential effort F_t vs. displacement at 2 min (blue) and at 8 hours (pink) of the test, coating E

5. Modeling of piston ring/liner friction

The results obtained for hydrodynamic lubrication regime show that the shape of the curves approaches the one that we can imagine in theoretical view. So, there is a need to check if this report is valid.

The piston ring/cylinder liner assembly used on the Plint machine for this study of tribological performances of various coatings is assimilated, for the mathematical modelling, with a sample of cylinder liner in contact with a piston ring with a parabolic profile and separated by an oil film (Fig. 11).

The rubbing pair is supposed to work with the following assumptions:

- surfaces of the piston ring and the liner are supposed to be smooth;
- lubricant is incompressible and Newtonian, with a constant viscosity η;
- lubrication regime is hydrodynamic and the flow of fluid is laminar;
- iInertia of the fluid and external volume forces are neglected;
- pressure remains constant along of the oil film thickness;
- offset of the piston ring is null thus the minimal oil film thickness is in the median plane of piston ring;
- effects of lubrication flow in direction Y can be neglected (the film is so fine); also the effects of the lubricant in direction Z are neglected (one considers the piston ring of width unit);
- piston ring is supposed to be in contact with oil film

over all its width b (cavitation is neglected); the hydrodynamic load capacity (bearing pressure) is supposed to be constant along the stroke of the piston ring.

Under these assumptions, the equation of Reynolds can be written in the following form

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial P}{\partial x}\right) = 6\eta u \frac{\partial h}{\partial x} + 12\eta \frac{\partial h}{\partial t} \quad (7)$$

where u is the sliding speed of the lubricating fluid which is equal to U in contact with the piston ring and to zero at the cylinder liner surface.

Thickness of the oil film is given by

$$h(x,t) = h_m(t) + h_p(x) \quad (8)$$

with $h_p(x)$ is a function representing the profile of the piston ring and $h_m(t)$ is the instantaneous minimal film thickness.

Load capacity or the normal hydrodynamic force is equal to the applied normal load W

$$W = \int_0^b P(x,t)dx \quad (9)$$

In our tests, the load W is constant, as well as the viscosity of the lubricant η; the sliding speed of the piston ring U is known; the only unknown parameters in this problem are thus h_m and dh/dt.

A double integration of the Reynolds equation allows the expression of dh/dt to be known.

Euler's method allows the calculation of h_m by using the following equation

$$h_{m_{i+1}} = h_{m_i} + \frac{dh_m}{dt}\left(t_{i+1} - t_i\right) \quad (10)$$

Therefore, on the basis of an initial estimate h_{m0} of h_m, one can calculate in each iteration h_m and dh_m/dt according to the desired tolerance [15].

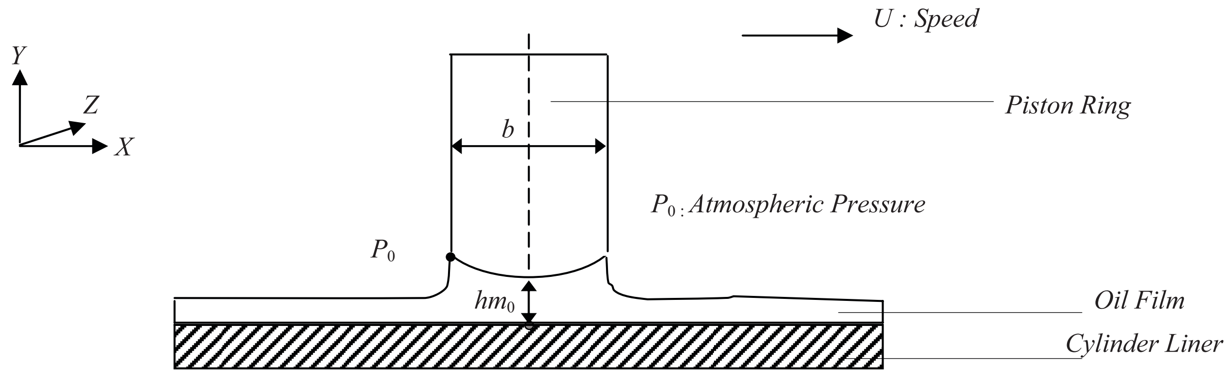

Fig. 11 Diagram of the piston ring/cylinder liner contact

Fig. 12 shows the calculated and measured friction force curves for various loads and frequencies with respect to the displacement and which are superimposed on the same graph.

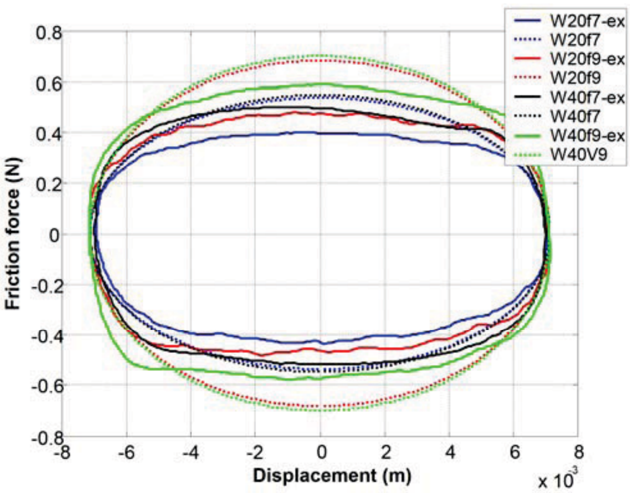

Fig. 12 Friction force vs. displacement: dotted lines - theoretical values: continuous lines - measured values

One can notice immediately that theoretical calculation largely over-estimates the level of friction in all the cases where a hydrodynamic friction is observed. However, one can imagine the assumption that the temperature of the lubricant, sheared in the contact zone, is higher than the one of the oil regulated in the tank, because of the high shear rate of the lubricant during sliding which causes an elevation of its temperature and a decrease of its viscosity inside the oil film.

By a reverse identification we can calculate the temperature of the lubricant which leads to the same fric-

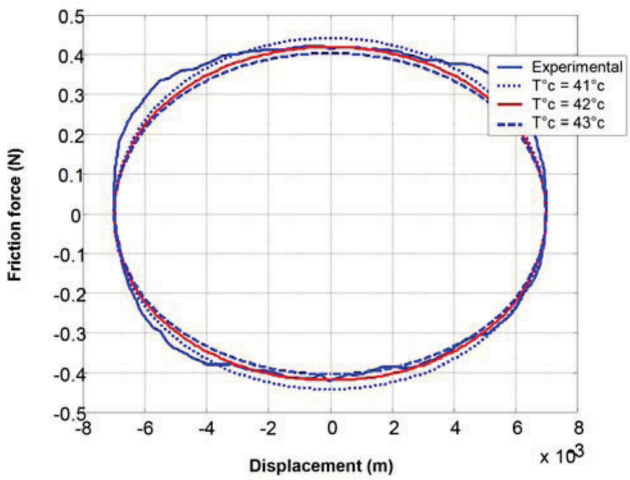

Fig. 13 Friction force vs. displacement: dotted lines - theoretical values: continuous lines - measured values, load W of 20 N and the frequency f of 7 Hz

tion force as to the one measured experimentally (Fig. 13).

One admits that one can allot this variation to the increase in temperature; one can notice that one is able to check the existence of a hydrodynamic behaviour of the contact piston-ring/cylinder liner under the test conditions.

However, when the sliding speed decreases, at the end of the stroke, the experimental curve shows greater forces than the theoretical forecast, sign that one spent in a mixed regime. Moreover, if one makes the difference between these two curves, theoretical and experimental, one can reach the value of the friction force, in excess compared to a hydrodynamic regime, and represent it according to the contact position as shown on the Fig. 14.

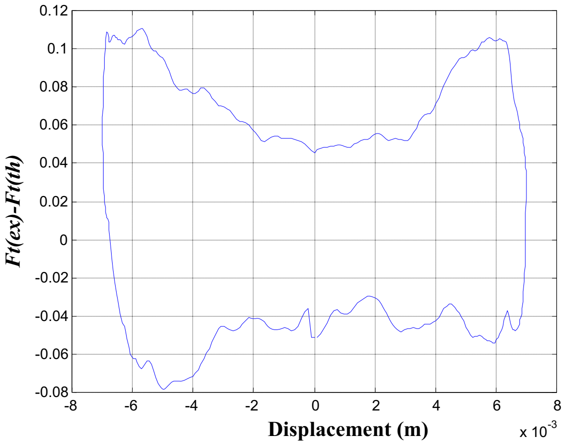

Fig. 14 Difference between the friction forces, experimental $F_t(ex)$ and theoretical $F_t(th)$, with load W of 20 N and a frequency f of 7 Hz

This difference between the two curves (theoretical and experimental) can be exploited in two directions: the prediction of contact under the conditions of mixed or even boundary mode and also the determination of the rate and positions of wear, knowing as the zones of mixed and boundary friction are exposed to the wear and that the rate (percentage) of boundary mode (and mixed) in contact is representative of the wear rate [16].

6. Conclusion

The work presented here has shown, first of all, that, using a simple laboratory, it is possible to simulate the piston ring/cylinder liner contact and to evaluate the comparative friction efficiency of various alternative coatings to the present electrolytic hard chromium.

Then very interesting alternatives have been found using new deposition techniques such as thermal projection.

As a matter of fact, a HVOF Wc/CrC coating exhibits a much lower friction level than the current reference material and this, in the boundary regime, corresponding to the top and bottom dead centre during an operating cycle, where the HVOF coating friction coefficient is equal to 0.142 comparing to the reference coating which is equal to 0.145.

The same remark is established in the quasi-hydrodynamic regime corresponding to the displacement of the ring in the middle of stroke. A HVOF coating exhibit a best hydrodynamic behaviour in this area and conserve this characteristic for loads between 20 to 80 N at practically the same lower speed of 75 mm/sec and the same lower friction force produced.

Moreover, it preserves, even after long wear test duration, its initial characteristics without generating additional wear on the cylinder liner.

Nitrided steel proves to be an interesting material to lower friction (friction coefficient is about 0.147) but probably not enough to be an interesting alternative to the reference ceramic reinforced hard chromium.

The chromium nitride obtained by PVD technique exhibits also friction performances (friction coefficient is about 0.157) close to the reference coating. As the PVD coating process has one of the lowest experimental impacts, it can be a promising way of improvement for the future.

These conclusions show that, at least, for the present and crucial question of the replacement of hard chromium-electrolytic coatings, some powerful alternatives are available.

In other axe, the results have shown that the hydrodynamic curve is not perfect but it contains a rate of boundary mode which can be exploited in determination of the rate and positions of wear.

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ĮVAIRIOMIS DANGOMIS PADENGTŲ VIDAUS DEGIMO VARIKLIŲ STŪMOKLIŲ ŽIEDŲ TRINTIES IR IŠDILIMO TYRIMAS

R e z i u m ė

Gamybos procese panaudojant CrVI reikalaujama ypač griežtai laikytis aplinkos apsaugos taisyklių. Todėl labai svarbu ieškoti naujų, aplinkai nekenksmingų medžiagų. Artimiausiu laiku reikia rasti, kuo pakeisti chromą elektrolitinio padengimo procese.

Šio darbo tikslas išbandyti kitas alternatyvias dangas, galinčias pakeisti variklių stūmoklių žiedų chromo dangas.

Atliekant šį darbą suprojektuotas specialus tribometras PLINT TE 77 skystajai trinčiai tirti. Jo konstrukcija gerai imitavo kontaktą tarp variklio stūmoklio žiedo ir cilindro sienelės.

Buvo išbandytos įvairių tipų dangos ir pasiūlyta keletas alternatyvių sprendimų.

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CHARACTERIZATION OF VARIOUS COATINGS IN TERMS OF FRICTION AND WEAR FOR INTERNAL COMBUSTION ENGINE PISTON RINGS

S u m m a r y

Nowadays, as the environmental regulations are becoming more pregnant, in particular the end of the use of CrVI in the manufacturing processes, a need for new materials solutions is required. The use of the electrolytic deposition process for the growth of chromium metal coatings requires that, in short term, alternative solutions need to be found.

The purpose of this work is to test several alternative coatings likely to replace the current chromium coating used on the top ring of the engine pistons.

For the realisation of this work, a PLINT TE 77 tribometer specifically designed for this study of friction in lubricated mode has been used. This type of configuration is well appropriate for the simulation of a contact between piston ring and cylinder liner of an engine.

Various types of coatings have been tested and some alternative solutions have been identified.

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ИССЛЕДОВАНИЕ ТРЕНИЯ И ИЗНОСА РАЗЛИЧНЫХ ПОКРЫТИЙ ПОРШНЕВЫХ КОЛЕЦ ДВИГАТЕЛЕЙ ВНУТРЕННЕГО СГОРАНИЯ

Р е з ю м е

В настоящее время правила защиты окружающей среды особенно жесткие для использования CrVI в процессе производства, поэтому особое значение приобретает использование новых материалов. Необходимо в ближайшее время найти новые решения для замены хрома при использовании электролитического процесса покрытия.

Цель настоящей работы – исследовать другие новые покрытия, заменяющие покрытия хромом поршневых колец двигателей.

При проведении этой работы, для исследования жидкостного трения, был спроектирован специальный трибометр PLINT TE 77. Его конструкция хорошо имитировала контакт между поршневым кольцом и стенкой цилиндра.

Были исследованы разные типы покрытий и найдено несколько альтернативных решений.

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