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

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

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

$$h(x,t) = h_m(t) + h_p(x) \tag{8}$$

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

$$W = \int_0^b P(x,t)dx \tag{9}$$

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

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

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].

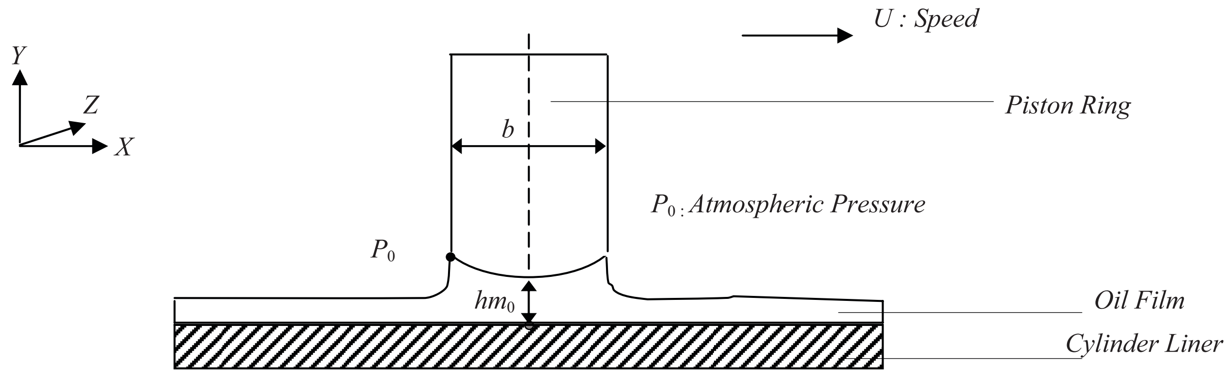

Fig. 11 Diagram of the piston ring/cylinder liner contact

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.

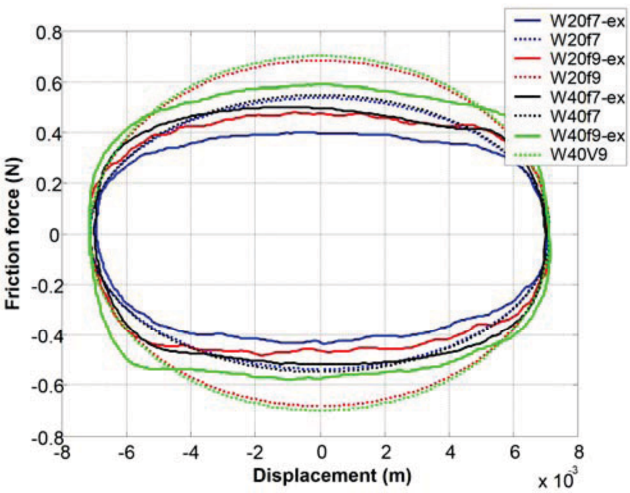

Fig. 12 Friction force vs. displacement: dotted lines - theoretical values: continuous lines - measured values

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-

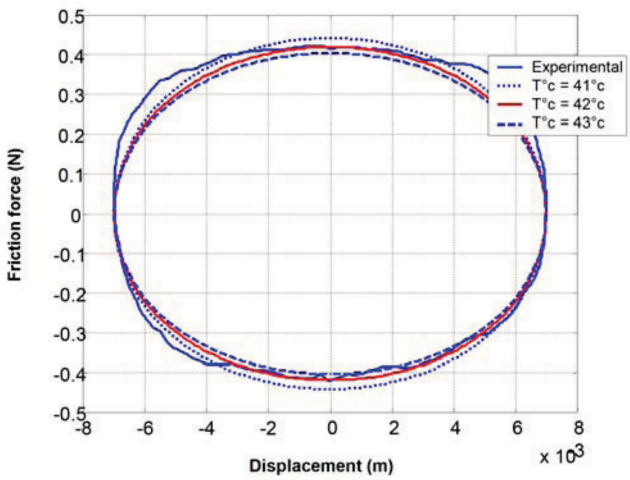

Fig. 13 Friction force vs. displacement: dotted lines - theoretical values: continuous lines - measured values, load W of 20 N and the frequency f of 7 Hz

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.

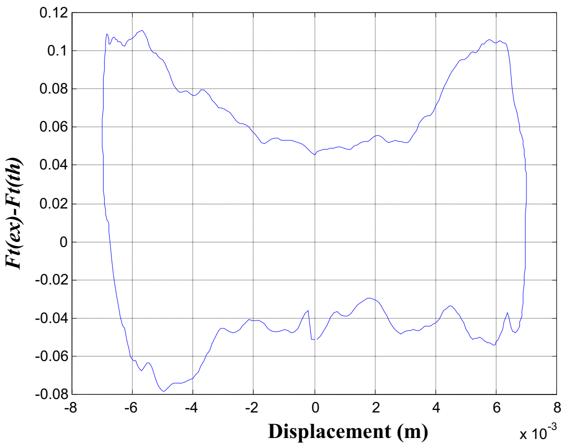

Fig. 14 Difference between the friction forces, experimental $F_t(ex)$ and theoretical $F_t(th)$, with load W of 20 N and a frequency f of 7 Hz

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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A. Guermat, G. Monteil, M. Bouchetara

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.

A. Guermat, G. Monteil, M. Bouchetara

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.

А. Гуермат, Г. Монтэил, М. Боуцхетара

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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