NEW ASPECTS OF THE STATE-OF-THE-ART ON LUBRICATED COUPLES IN HIGH PERFORMANCE ENGINES. ANALYSIS OF TWO SIGNIFICANT COUPLINGS

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SUMMARY
In high performance engines mass economy and increasing demands are design targets; as follow-up, stiffness reduction and strain-stress state increment of the components are originated. Direct consequence of this trend is the major change in shape of the lubricated couples, with cyclic variation of the oil film thickness, inducing redistribution of the oil pressure pattern and, sometimes, serious oil losses, leading to seizure. It is necessary to develop simultaneously a structural and tribological analysis, which allows to integrate the whole procedure between oil film profile and oil pressure pattern, to obtain the mutual forces between the lubricated couples, until convergence.

Starting from the observation of the emerging troubles in high performance engines components, our team is developing a focused research, trying to simulate and explain these phenomena. A previous paper [1] reported case histories of lubrication problems in high performance engines. In the present paper, two significant couplings in a high performance engine (liner - rings and crankshaft - main bearings) are investigated and the main results reported; a following paper [2] concerns the case of con rod - pin - piston system.

Keywords: high performance engine, crank mechanism, tribology, failure analysis, structural-tribological analysis

1 STATE-OF-THE-ART
The high performance internal combustion engines, unlike those motorising the large series cars, are quite different since the concept and design phase. Missions strongly critical but time-limited, prototypical productions and costs included in generous budget, make it possible to use over-refined methodologies and more valuable materials. In such framework, the mass is highly significant and its strong reduction, joined to a short time of engine development, involves complicated analysis concerning stiffness distribution and strain-stress state. Hardly robust designs originate, critical to solve within a short time: the performance to be achieved is always extreme.

The failure of components, generally not unusual during the phase of development, demands a careful investigation (failure analysis) to explain the structural behaviour, associated to high number of revolutions. Just starting from the examination of drawbacks and damages occurred in the past, during several evolutions of high performance engines, we have noticed initial difficulties in finding references; this is synonymous of substantial empiricism in the utilised solutions. Also for reasons of natural privacy, state-of-the-art description is unapproachable.

In this framework, being not urgent a research on prompt solutions of technical problems, we proposed to ourselves as a target to examine the available material (some failed pieces), so rich of information, never carefully examined, owing to lack of dedicated time. So, the idea to canalise the investigation in trends, concerning the coupling of lightened and strongly stressed structures, in the presence of lubricant, has materialised.

A previous paper [1], presented at the 2000 AIMETA International Tribology Conference (L'Aquila - Italy), reported case histories of lubrication problems in high performance engines. Here that previous investigation has been developed, trying to evidence the typical phenomena in operating conditions. It is quite difficult to prevent the defects in these engines by means of suitable test campaigns, during the initial phase of development.

All the main couplings of the crank mechanism are subject to deformation or can fall in resonance conditions, which increase criticalities and may cause irregularities in the functioning, owing to wear phenomena, that, if uncontrolled, may produce seizure. Ticking points on the border of cavities machined on the piston surface, pitting or flaking on the camshafts, scuffing on the rings sliding in cylinder liners, heavily distorted or vibrating con rods and main bearings with evident local areas of wear or with superficial overheatings, external surfaces of cylinder liners with small
or great craters of cavitation, relative sliding between con rod body and cap with vibrations on the interface plane and still other phenomena ... are evident marks of incipient decay of the engine reliability.

Knowledge of these phenomena, generally not evident in mass produced engines, is instructive for the implementation of new designs. Besides, the better knowledge of these phenomena gives the designer a much wider view to choose different technical solutions and to evaluate solutions that, apparently brilliant for their lightness and for using more valuable materials, are not always successful.

In the high performance field, considering components as massive structures or beam is not realistic, because thin wall structures must be analysed by more appropriate and complex methods, as shell theory [3].

The assurance that an oil film profile may continue its lifting action, so to separate the two surfaces and take away the generated heat, is impaired if the oil film geometry changes its shape, as a result of non-perfectly stiff surfaces, subjected to high loads.

This is the scenario: new, alluring and complex, in which we will try to introduce a few rules and to deduce some valuable information.

In this paper some phenomena referring to couplings (piston - ring - liner and crankshaft - main bearings) on high performance engines are analysed; another paper [2], presented at this same Congress, concerns a further coupling (con rod - pin - piston system).

To approach such analysis, the integration of structural and tribological aspects is necessary: any study, neglecting one of these aspects, could not yield a useful result. The investigations, even if very difficult to perform, try to face the problems in a systematic way.

2 INVESTIGATION ON DIFFERENT COUPLINGS

After observing wear and seizure couplings or failed components in several operating high performance engines, a few research themes were planned, particularly in sliding couplings of crank mechanism [4], [5], [6], [7] and of valve gear mechanism [8], [9].

For each of them, actual operating conditions were investigated; starting from traditional situations of high stiffness and from structural analysis alone, less and less stiff couplings, also taking into account the presence of interposed lubricant, have been studied.

It has been already evinced that the operating range of high performance engines may increment the possibility of resonance in some of their components.

If such components have surfaces coupled with other components, the complexity of the whole system increases, as the configuration, even the geometric domain of the lubricant, varies with time. A definite integration of structural and tribological aspects is then necessary to cope with the complex phenomenology produced during the engine operation.

In the contest of the integration of structural and tribological analysis, a high performance engine is considered as a lean structure, light and highly strain and stressed, according to what can be called a "willow" structure.

For a monitoring of this situation, it is sufficient to think to those aspects:

- peripheral accessories, which might vibrate for stress transmitted by the connection brackets
- pulleys or gears transmitting the vibration of the crankshaft to accessories or to the camshaft
- the crankshaft undergoes deformation and causes a misalignment with the main bearings
- con rods bend and twist with their pins and pistons
- pistons sliding and rotating in various ways
- camshafts with torque vibrations influencing the valve timing
- possibility of interference between valve and pistons ...

All these aspects point out the extreme difficulty of the analysis. In this framework, we confine ourselves to treat two systems:

a) piston - rings - cylinder liner
b) crankshaft - main bearings.

3 DISCUSSION OF TWO COUPLINGS

3.1 Piston - rings - cylinder liner

Investigation on such coupling in high performance engines is a problem of lubricated contacts between elements, which can be considered non-perfectly stiff [10].

The cylinder liner has a dynamic behaviour similar to that of thin wall shells [3]. Besides, due to the strain-stress, caused by the screws between the block and the cylinder head and respectively by the thermal fields, the cylinder liner cannot be studied considering a cylindrical geometry and is better characterised by circumferential waves (3D-lobe).

The elastic rings, with their alternative motion between Top Dead Centre (TDC) and Bottom Dead Centre (BDC) and viceversa, meet continuously varying spaces; the oil film profile may collapse and scuffing on the rings may originate [11], [12].

This study tries to define the lubrication conditions between ring and liner; and, since the study of the lubrication demands to know the oil film geometry for solving the bi-dimensional Reynolds equation, the deformation of the liner must be exactly evaluated.

This problem is very complex; a first approach was to assess the effects of the cooling liquid on the eigenfrequencies of the cylinder liner or, at least, their order of magnitude.

Since the ratio between thickness and radius of the liner is very low (< 0.06) and so is the small bridge between two adjacent liners, the effect of the cooling water was investigated (by a finite element model, significant for this kind of study), both as a mass involved in the vibration and as an absorber [13], [14], [15], [16].
Comparing the results obtained by this simplified model, with and without the cooling fluid inside it, it has been observed that:

- the water mass, participating to the liner vibration, reduces the liner eigen-frequencies by about 10%
- the squeeze effect, due to the fluid interposed between two adjacent liners, is negligible, both as a variation of the eigen-frequencies and as a variation of the system response.

Once the presence of cooling water was found to be non-influential, attention was focused only on the liner, developing a finite element model with a geometry as close as possible to reality.

Modal analysis on such model has produced results consistent with Warburton's theory [17] on thin axial-symmetric cylindrical shells; evaluating the liner eigen-frequencies, the possibility for them to be in the field of the engine frequencies was excluded. This result allows to reject any important resonance conditions regarding the engine component. The liner is then not affected by a significant resonance but its distortion (from which the ring lubrication conditions are depending) is due to the cylinder head bolt load and does not change for any engine rotation field.

By the same model, the deformation of the inner surface of the liner, produced by the gas pressure, has been calculated, which is very important to study the geometry of the oil film between liner and ring. After this analysis, deformations of the liners, produced by tightening the screws of the head, were calculated: data on current production engines [18], [19] and data on high performance engines have been gathered.

Considering all these data, it has been concluded that, for any type of engine (aluminium or cast iron block, cast-in or pressed-in liners) the deformations of the liners, observed in planes perpendicular to the axis:

- vary with the distance of the considered plan from the reference plane (usually the head-block plane)
- reach a maximum value of 0.03 mm
- vary with the depth of the screws, which tighten the head

vary in different cylinders, due to the local stiffness of block and head.

Non-uniform thermal expansions, measurable axially on pistons and liners, due to temperature gradients, may be corrected giving the piston a due profile, since the expansions are identical in all cylinders.

So, the analysis of lubricated ring conditions should consider only the deformation generated by tightening the screws between head and block on the inner surface of the liner. Such deformations determine the circumferential shape of the oil film, with areas where there is contact and where there is not; in this condition, the elastic deformation of the ring varies and, likewise, the pressure on the oil film.

Since the liner deformation varies both circumferentially and axially, the minimum oil thickness, between ring and inner surface of the liner, and the distance, between ring and liner where there is no contact, are function of the axial position of the piston. The radial clearance between liner and ring causes high oil consumption (with undesirable effects on emissions) and gas blow-by toward the oil sump (with power losses).

It must be pointed out that the analysis, synthetically presented here, is not correct, because it neglects the ring being mounted in a cavity slightly larger. So, a different location, velocity, acceleration of the piston and different gas pressure will cause different axial location and different rotation of the ring in its cavity:

- the first, because of inertia of the ring and effect of the gas pressure
- the second, because of the torque produced by inertia and friction with the inner surface of the liner.

It may be concluded that the lubrication of the ring can be studied only approximately, without considering the motion of the ring in its cavity. In this study, the variability of the axial position and the rotation of the ring, related to the cavity, have both been ignored.

3.2 Crankshaft - main bearings

Problems concerning the crankshaft - main bearings coupling are considered for obtaining a more robust structural design of the main bearings [20], [21]. Such coupling is strongly influenced by lubricating conditions, which are determined by the mechanical conditions of the coupling elements, namely by their deformation, that changes the lift pattern.

In the interaction crankshaft - main bearings, the lubricant must not be ignored, which, to a concentrated load, reacts with its film lift, distributing the whole load on a more extended surface.

The oil film geometry is made of a convergent portion (where lifting pressures are generated) and of a divergent portion (where pressure field does not reach the lubricant saturation point). An emulsion of oil-gas-vapours may be generated with density and viscosity lower than oil's, producing cavitation phenomena. These phenomena may be increased by the small stiffness of the coupled structures, as it happens in high performance engines. Since sliding velocities are rather high, in the oil film there is turbulent motion of the liquid, which decreases the lifting capacity of the bearing: re-circulation flow is responsible for this phenomenon.

The dynamic viscosity of the lubricant and the clearance, between journal and bearing, control the re-circulation. Besides, the possible deformations of both journal and bearing impair the parallelism of the axes of the two-coupled elements and the bearing is significantly deformed under the action of the oil film pressure. Both these phenomena influence the hydrodynamic lubrication of the bearings and, consequently, the way they work.

The main bearing, stressed by the pressure of the film, causes a further deformation of the oil film profile: pressure distribution varies and, consequently, the
deformation distribution varies as well until equilibrium.

So, to design a main bearing it is necessary to iterate several times the calculation of both the pressure and the deformation distribution, until an acceptable residual error is reached. The block-diagram of this procedure is reported in figure 1.

**Figure 1 - Elasto-hydrodynamic lubrication (EHL) analysis using numerical models**

### 4 RESULTS AND CONCLUSIONS

#### 4.1 Piston - rings - liner

The analysis presented here has only concerned the lubricated coupling, neglecting the thermal field, caused by the operating engine.

This analysis wants to underline an aspect, generally neglected, that is the lubrication of the rings in a bore distorted by tightening the screws of the head, which causes an oil film profile of variable thickness, both axially and circumferentially [10].

Actually, the calculated pressure field and the analysis of the ring deformation demonstrate:

- the ring does not continuously contact circumferentially the liner, but only where the liner is under its nominal diameter
- axially, along a generating line, the non-uniformity of the radial deformation of the liner causes a non-
uniform thickness of the oil film profile and, consequently, non-uniform contact and lubrication conditions of the ring.

Such results evidence a variability, both axial and circumferential, of the interaction between liner and ring, suggesting that, most probably:

- scuffing does not appear on the whole external surface of the ring, but only in contact portions
- portions damaged by scuffing vary with their axial position.

In several liners of high performance engines, disassembled after many working hours, numerous non-uniform circumferential streaks (in liner sections orthogonal to the axis) and axially discontinuous streaks were observed; this evidences the validity of the proposed research. Obviously, a more accurate agreement between calculated and observed scuffing shall be possible when all preliminary simplifications will be removed.

In figures 2, 3 and 4, presented hereafter, the circumferential variability, above described, is shown; an idea of the axial variability is achieved by comparing the three different diagrams, corresponding to three different axial position of the ring. They show the parabolic trend of the pressure along the axial thickness of the ring, produced by the rotation of the ring orthogonal section.

![Figure 2 - Pressure field (first ring at position 1)](image1)

![Figure 3 - Pressure field (first ring at position 2)](image2)

![Figure 4 - Pressure field (first ring at position 3)](image3)

The lubrication analysis of the first ring during expansion stroke is represented:

- position 1 at 48\(^\circ\) crank angle after TDC
- position 2 at 82\(^\circ\) crank angle after TDC
- position 3 at 1190\(^\circ\) crank angle after TDC

### 4.2 Crankshaft - main bearings

In this work, the hydrodynamic lubrication in main bearings has been investigated, evidencing several phenomena (cavitation, re-circulation, misalignment of the axes and deformation of the bearing) and suggesting methods for considering such phenomena in structural design [20]. All this is a possible developing algorithm for calculating automatically cavitation, re-circulation and integrating both the tribological and the structural analysis.

It has been evidenced that hydrodynamic lubrication is produced by a pressure distribution in a fluid forced in an oil film profile of a given shape. The capacity to generate pressure in an oil film profile filled with a viscous fluid is ruled by Reynolds differential equation, if starting conditions are correct.

It has been demonstrated that in a main bearing Reynolds conditions are not always valid, both for cavitation and fluid re-circulation. Somebody thought to simulate the cavitation phenomenon by an adequate
The choice of the reological characteristics of the lubricant in the oil film profile. It is necessary, instead, to design the bearing geometry in a way non favourable for the re-circulation flow, choosing accurately the radial clearance, the diameter of the journal and the width of the bearing.

Choosing among several calculations carried out, in figure 5 there is a summarising prospect, where the conditions of functioning of a central main bearing, for some engine revolutions and for several radial clearance values, are reported. The amplitude of the area, where re-circulation flow is established, is also evidenced. On the re-circulation flow phenomenon also other parameters, characterising the geometry of the bearing, have influence, as the journal diameter D and the bearing width L. Increasing D, with L constant, the pressure pattern has a lower peak; the same trend is observed, increasing L, with D constant.

Once an adequate calculation code for the oil film profile has been developed, this analysis allows to design correctly the main bearings and all other hydrodynamically lubricated couplings of an engine. Such analysis may be fruitfully integrated with a finite element simulation for evaluating the actual operating conditions of a journal - bearing coupling.

So, it is possible to perform the design of a block, starting at the initial phases, evaluating the strain-stress state with a model, which may check the agreement on the investigating phenomena.

For high performance engines, effects of high rotating velocities cannot be ignored; it is not negligible the inertia of the lubricating fluid; in such conditions, the Reynolds equation may become useless and it might be necessary to solve the Navier-Stokes equations [22], [23].

Conclusions reached here may be easily extended to other hydrodynamically lubricated couplings, such as: con rod big end - crankpin, con rod small end - pin, piston - liner. We trust that this work may originate useful further developments, mainly for the calculus methodologies and related algorithms, which might enlighten the work to be done, as for interfacing tribological with structural analysis. We believe useful a further development of an algorithm capable of investigating the variation with time-space of the load applied to the lubricant film.

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**Figure 5 - Position and angular amplitude of the bearing re-circulation area**

Shaft radius = 30 mm

\[ \nu = 0.015 \text{Ns/m}^2 \]

Radial clearance between shaft and bearing [m]*10^{-5}

\[ c=1 \quad c=3 \quad c=5 \]

Angular velocity of the shaft, \( n \)

Resultant force, \( F \)

Angular position of load

Recirculation zone

Magnification 100x
ACKNOWLEDGMENT

The authors wish to express their gratitude to management and technical staff of Ferrari, who have supplied documentation and information. Also thanks to the newly-graduate Federico Zaramella and Giovanni Bruni, who have contributed to this research during the development of their degree thesis in Mechanical Engineering at the Politecnico di Torino.

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