INTERDISCIPLINARY STRUCTURAL AND TRIBOLOGICAL ANALYSIS IN HIGH PERFORMANCE ENGINES: THE CASE OF CON ROD - PIN - PISTON SYSTEM

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SUMMARY
 In another paper the state-of-the-art on structural and tribological couplings in high performance engines has been dealt with [1]. In this paper the con rod - pin - piston system is analyzed, including lubrication, in order to investigate the cause of undesired phenomena, both in static and dynamic conditions. The design target of these components is mainly represented by mass reduction and high performance. This is synonymous of large strains; since the eigen-frequencies can easily fall in the operating range, bending and torsional vibrations of the whole system also appear. Owing to these phenomena, the oil film profile changes abruptly; if the oil thickness becomes lower than a safety value, the oil pressure peaks can increase so much as to cause local wear. In 3D space, rather than in 2D, further criticalities appear. A larger misalignment between the lubricated couplings of the con rod, together with the high operating speed, can originate lateral oil losses. Besides, con rod big and small ends show different behaviours: hydrodynamic lubrication in the big end and squeeze lubrication in the small end (the latter, not analyzed here), requiring different approaches for the analysis. Our research started from visual evaluation of component failures [2]. An iterative procedure has been set up, allowing to model the relationship between the oil film geometry and the oil pressure pattern to be used for the structural analysis of the con rod big end. Concluding, an interdisciplinary analysis, both structural and tribological, is strictly recommended in high performance engines design.

Keywords: high performance engine, crank mechanism, tribology, stiffness, vibration

1 INTRODUCTION
 The design of a connecting rod needs particular attention to guarantee reliability in missions, which are especially critical in high performance engines.

Several design methodologies of the past considered a few mandatory parameters (length between big- and small-end centres, diameters of small and big end, bolt axes …) to define the geometry of the component, suitable to guarantee an homogeneous strain and stress distribution in the cross sections of the body, particularly in the connection between big end and cap, analysing only the dead points. So, the stress state was defined by classical relations, especially of the beam theory. The finite element numerical analysis allows to simulate complex geometric shapes and to consider materials with different characteristics; higher computation capacities make strain and stress maps of a whole component easy to draw [3].

In nearly all references, numerical structural analyses consider monolithic structures: the result is one piece rectilinear beam, with hollow circular beams, at both ends. Two equal tensile loads, applied to both ends of this structure, ovalize the big and small ends, creating a tensile strain-stress peak on a plane, orthogonal to the load direction, close to the inner side of the circular portion, which contacts the crank pin. Actually, since the beam is interrupted, it is physically impossible the presence of tensile strain and stresses at the interface between con rod body and cap. Besides, in a monolithic model, any relative sliding between con rod body and cap is hindered, due to different local stiffness. The monolithic model is over-simplified and is useful especially to visualise strain and stress trends a bit far from the interfacing areas; such monolithic models are accepted for dynamic study, but modify uncontrollably the strain-stress state, because they neglect the physical feature of the considered structure.

This is why a more realistic structural simulation has been sought through two pieces con rod models and considering the two components, body and cap, as two different elements, connected by bolts. In this way, stresses between body and cap can be evaluated and elasto-hydrodynamic lubrication (EHL) of the con rod big end analysed, considering more accurately the stiffness of the bearing and crank pin (Fig 1a and 1b).
Figure 1a: Model of con rod (monolithic) - pin - piston system usable for dynamic analysis

Figure 1B: Con rod model in two pieces to be used for EHL analysis

Being the con rod body more stiff than the cap, relative sliding is inevitable: operating conditions may be more critical for vibrations on the interface plane, producing non-negligible shear stresses in the bolts and larger deformation on the cap.

So far, the con rod stress analysis was limited to the effects of gas pressures, inertial forces, both axial and cross. In high performance engines, at very high revolution speed, for reducing moving masses, materials with remarkable ultimate tensile stress but low density and Young's modulus are used. Consequently, higher deformations are obtained, likely to jeopardise correct operation of lubricated couplings. This is another not negligible difference between mass production and high performance engines.

The crank motion, even considered bi-dimensionally (2D), causes significant deformations of the con rod cap and body, altering the oil film profile between crank pin (more stiff) and con rod big end bearing (less stiff). So, for studying crank mechanism, from the structural point of view tribological analysis as well should be considered [4].

The combined effect of mass reduction and high rotational speed may cause another phenomenon: torsional and bending vibrations, not of the con rod only [5], [6], but of the whole system (con rod - pin - piston), close or within the operating range [7].

In this case, a 2D study is no more sufficient, because misalignment (between the rod big end and crank pin axes) and torsional motion of the con rod (with rotation of the piston surface, influencing the valves relative motion) arise. Extensive wear of con rod bearings and ticking points on the border of cavities, machined on the piston surface, evidence these phenomena.

The particular anomalies of functioning, more and more evident, even during the testing phase, suggested more accurate analyses on the behaviour of the system (con rod - pin - piston). Moreover, a study of the crank mechanism, limited to the dead points, is insufficient, but the whole engine cycle has to be investigated for evaluating the oil film profile evolution.

These phenomena may be the cause of component failures, as observed several times on field.

2 NUMERICAL INVESTIGATIONS

The lubricated couplings of the crank mechanism are: crankshaft - main bearing, crank pin - con rod big end, pin - con rod small end, piston - rings - liner. In the present work, the crank pin - con rod big end is analysed.

Kinematics - dynamic behaviour of the crank mechanism is universally known; so, only these analyses are considered here:

- purely structural, with the rod aligned to Top Dead Centre (TDC): dynamic of the whole con rod - pin - piston system [7]
- structural - tribological: EHL of the crank pin - con rod big end [8].

2.1 Dynamic analysis of the con rod - pin - piston system

The dynamic behaviour of the crank mechanism was never studied deeply, unlike the crankshaft, because undesired vibrations appear very seldom. Actually, in mass produced vehicles, vibrational phenomena of the crank mechanism are insignificant, due to the limited revolution speed.

Amazingly, such phenomena (especially torsional vibrations) were familiar, instead, to steam engine designers over a century ago, due to large masses and dimensions and slenderness of those con rods. Nowadays, these phenomena, as we have demonstrated, have a totally different cause: the very high revolution speed. Let us consider some drawbacks produced.
In several cases, evident marks of collision near the cavities, machined on the piston surface (to avoid any contact against the open valves) have been noticed in high performance engines. A torsional vibration of the whole crank mechanism explains such marks. The first eigen-frequency (torsional rather than bending) may be in the higher part of high performance engines revolution range.

To avoid this phenomenon, the con rod profile can be modified to increase stiffness and so to shift upwards the first torsional eigen-frequency or, else, cavities in the piston surface may be widened (to allow a more extended vibration), even if this is detrimental to the combustion process.

The finite element numerical simulation overcomes the difficulty to determine experimentally the eigen-frequencies. The finite element model, here examined, starts from a 3D model, produced by CATIA CAD system. First of all, models of the con rod and piston, monolithic and both assumed linear, have been built and used in this analysis; then, to make more realistic the con rod 3D model, elements of not linear characteristics have been built in the interface plane (con rod big end - cap); pre-loaded bolts have been also considered [9].

Due to the complex geometry of the components, a mesh with hexahedron elements was impossible to generate automatically; as a consequence, tetrahedron elements with 10 grids and homogeneous mesh density on the whole component have been built. This solution is acceptable since only a dynamic analysis has to be performed; evaluations of strains or stresses would require, on the contrary, a more refined mesh. The whole model has about 64,000 3D elements. The pin has been simulated with beam elements, having a rectilinear geometry. The characteristics of the materials of the three components are reported in table 1.

The dynamic analysis has been performed both with the con rod aligned with the crank, simulating the TDC, and in slightly bent position, to evaluate the system stiffening.

The distribution of the mass inside the finite element model has been compared with that of the actual system: a difference of about 10%, considered acceptable, has been obtained [7].

### 2.2 Lubrication analysis of the con rod big end - crank pin

The theory of hydrodynamic lubrication, used for the con rod big end, is generally applied to rigid bodies, where the oil film thickness is determined by the value of the clearance between the two parts and by the bearing - journal eccentricity.

The non-deformability assumption can be considered correct for the crank pin, but not for the con rod big end; the latter can have a remarkable ovalisation during operating conditions, showing, moreover, vibrations or separation at the interface between body and cap. The ovalisation changes the lubrication conditions, altering oil film thickness and its pressure pattern; so, recirculation and cavitation phenomena increase [10], [11].

In some areas, oil film thickness may be so reduced as to cause contact between the two coupled components: over-heating of the metal may be produced and even seizure.

Cavitation, in some areas, may cause mechanical losses in the lubricant circuit, damaging the coupled surfaces. To solve this problem, the designer tries to use high pressure areas of the oil film profile to feed the lubrication circuit; so, a numerical procedure for obtaining results approximately accurate is fundamental: this simulation is one of the most important phase of the design.

#### 2.2.1 EHL of con rod big end - crank pin

The EHL theory takes into account the deformability of the bodies and affects significantly the profile of the oil film.

We call the theory micro-EHL when deformations are smaller than the mean oil film thickness: in this case "lubricated Hertzian" contacts are considered, as for cam - follower and rings - cylinder liner. We call the theory macro-EHL when deformations are of the same order of magnitude or larger than the mean oil film thickness: that is the case of highly stressed components, as con rod big end - crank pin. Structures, built with materials of high mechanical characteristics and low elastic modulus (for example aluminum and titanium alloys), undergo higher deformations and, therefore, belong to this class.

In literature, studies of the EHL are currently developed by means of suitable algorithms, which utilise essentially the finite element method and enable to solve, at the same time, the equations of elasticity, of hydrodynamic lubrication and of equilibrium to the external loads.

The method more frequently used (Oh - Goenka 1985) [12] provides the simultaneous solution of these equation systems by means of a single algorithm. Extensive information is obtained on oil film profile and pressure pattern at any point of the engine cycle; this method is usually applied to single structural models, ignoring the stiffness distribution.

A more complex but more accurate algorithm can be obtained by a "mixed method", which uses both structural and tribological mesh at the interface between the con rod big end and the lubricant. So, even complicated models, accounting for geometry and stiffness, typical of the con rod big end, can be used. This method is iterative and converges to the oil film profile (Fig 2) [8].
3 RESULTS

3.1 Dynamic analysis of the con rod - pin - piston system

Results are reported in Table 2. The first mode shape is a torsional vibration round the con rod axis; the stiffness of the whole system increases when the crank angle increases (angle at TDC = 0°). In the operating range, the torsional vibration appears when con rod and crank are aligned, i.e. in condition of minimum stiffness.

The critical engine revolution for torsional vibration is about 18,000 rpm and should be outside the operating range. The bearings of the con rod ends allow lateral lubricant leaks; this more actual simulation of the con rod, which can be considered as a mass, supported on two elastic systems, is not included in the present dynamic analysis. As a consequence, the critical value resulting from the analysis of the con rod - pin - piston system may be somewhat lower for the presence of the lubricant in the couplings and along the cylinder liner surface; an additional contribution is due to the friction, owing to the relative motion at the big end surface; the crank pin contributes to the stiffness of the big end. The first mode shapes are reported in figures 3 and 4: con rod aligned with the crank at TDC and at a slight angle, respectively.

3.2 EHL of con rod big end - crank pin

A first assessment of the "mixed method" has been carried out at TDC, at 8,500 rpm; in this position, higher loads and more evident deformation are present.

Setting up the algorithm, the effect of the deformation of the con rod big end on the oil film pressure pattern was evaluated.

Through the structural model, the deformation of the con rod big end is obtained, which, giving the crank pin a reasonable tentative eccentricity, defines the oil film profile [8]. This oil film profile is illustrated in figure 5 and shows two converging and two diverging portions. It must be remembered that, during the engine cycle, this configuration changes cyclically, as a consequence of the con rod big end deformation and of the position of the crank pin relative to the con rod big end.
Figure 5: Pressure pattern at tentative eccentricity (0.035 mm) at TDC

Figure 6: Development of the pressure pattern at TDC

Limiting the analysis to the dead points, we notice:

- at TDC, two areas of higher pressure and two areas of lower pressure than atmospheric (as in fig. 5)
- at Bottom Dead Centre (BDC), only one area of higher pressure and one of lower pressure.

This explains:

- at TDC, a strong ovalisation
- at BDC, a small deformation.

So, the pressure pattern (fig. 6 - TDC position) shows two maxima; in between, the lower pressure, due to the oil film profile, may cause: cavitation (which reduces hydrodynamic lift), lateral hydrodynamic losses of lubricant and gas bubbles which, imploding, can produce local wear. In any other angular crankshaft position, cycle pulses from oval to almost circular shape occur.

A synthetic prospect of the operating condition investigated and of the results, as maximum oil film pressure \( p_{\text{max}} \) and related oil film thickness \( h \), is reported in table 3.

where:

- \( e_z \) = eccentricity component along z axis
- \( \phi \) = angular co-ordinate of the cylindrical reference system related to the con rod big end (fig. 5)
- \( h \) = oil film thickness
- \( p \) = oil film pressure.

4 CONCLUSIONS

Visual observations on field of highly worn components of the con rod - pin - piston system in high performance engines stimulated the start of this research, finalised to structural and tribological analyses.

The dynamic analysis of the whole system and interdisciplinary structural and tribological analysis of the con rod big end - crank pin have been performed. Due to the particular type of the engines considered, for which mass reduction and the use of materials lighter than steel (i.e. aluminum and titanium alloys) is mandatory, the system studied is less stiff and must endure a higher rotational speed. Such operating conditions may lead to critical situations, near or within the operating range, causing wear and, ultimately, seizure. Actually, the stiffness reduction influences not only the structure, but also the oil film profile; such profiles, which in the classical approach have variable thickness, are in the clearance between two coupled circular sections.

In the case we are investigating, the oil film profile is more evidently variable, but one of the two coupled sections (con rod big end - bearing) is no more circular. The oil film profile varies continuously during the engine cycle and, at TDC, there are two quasi-contact zones, instead of one, between bearing and journal (fig. 5), and, there, the oil film thickness may reduce to nil. Such variations in the oil film thickness influence highly the lubricant pressure pattern; so that this pattern must be considered in the analysis of local stresses of the con rod big end.

In the present work, another phenomenon should be considered, which, as far as we know, has not been published yet: the misalignment between the crank journal axis and the con rod big end axis; such misalignment is caused by dynamic phenomena of the complete con rod - pin - piston system.

The proposed procedure (summarised in fig. 2) calculates the oil film geometry (at TDC by setting a tentative eccentricity value) and draws the related pressure pattern, which allows the definition of a new load state; whence a new oil film geometry results. This step is repeated until an acceptable convergence is reached.

At present, an algorithm is being developed, to perform this procedure (providing also the calculation of the eccentricity between con rod big end bearing and journal) and capable to analyse any position in the engine cycle, not only the TDC.

From this paper, a recommendation arises in studying high performance engines: the need to associate with the structural design (vibrations included) a design taking into account the relative deformations, with the aim to perform a tribological analysis as well.

5 ACKNOWLEDGEMENTS

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 Gianluca Canino and Vincenzo Feola, who have contributed to this research during the development of their degree thesis in Mechanical Engineering at the Politecnico di Torino.
6 REFERENCES


### Table 1: Properties of the utilised materials

<table>
<thead>
<tr>
<th></th>
<th>Piston material:</th>
<th>Con rod material:</th>
<th>Pin material:</th>
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<tbody>
<tr>
<td></td>
<td>Aluminum alloy</td>
<td>Titanium alloy</td>
<td>Steel</td>
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<tr>
<td>Young's modulus $E$ [N/mm$^2$]</td>
<td>71,000</td>
<td>110,000</td>
<td>214,900</td>
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<tr>
<td>Poisson's ratio $\nu$</td>
<td>0.287</td>
<td>0.329</td>
<td>0.334</td>
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<tr>
<td>Density $\delta$ [kg/mm$^3$] $\times 10^{-6}$</td>
<td>3.0</td>
<td>4.4</td>
<td>7.8</td>
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### Table 2: First mode shapes of the system [7]

<table>
<thead>
<tr>
<th>Finite element model</th>
<th>Con rod aligned with the crank</th>
<th>Con rod at a slight angle with the crank</th>
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<tbody>
<tr>
<td>eigen-frequency</td>
<td>mode shape</td>
<td>frequency [Hz]</td>
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<tr>
<td>1$^\text{a}$</td>
<td>torsional</td>
<td>$\approx 303$</td>
</tr>
<tr>
<td>2$^\text{a}$</td>
<td>bending YZ</td>
<td>$\approx 1680$</td>
</tr>
<tr>
<td>3$^\text{a}$</td>
<td>bending XZ and YZ</td>
<td>$\approx 1767$</td>
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<tr>
<td>4$^\text{a}$</td>
<td>bending-torsional</td>
<td>$\approx 2366$</td>
</tr>
<tr>
<td>5$^\text{a}$</td>
<td>bending XZ</td>
<td>$\approx 2527$</td>
</tr>
</tbody>
</table>

### Table 3: Operating parameters and results [8]

<table>
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<tr>
<th>Operating parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>tensile load on the con rod (TDC, 8500 rpm)</td>
<td>20,000 [N]</td>
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<tr>
<td>nominal clearance</td>
<td>0.04 [mm]</td>
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<tr>
<td>tentative eccentricity</td>
<td>$e_z = 0.035$ [mm]</td>
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<table>
<thead>
<tr>
<th>Results</th>
<th>Oil film thickness $h$ [μm]</th>
<th>Maximum pressure $p_{\text{max}}$ [MPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>angular co-ordinate $\phi = 218^\circ$</td>
<td>8</td>
<td>22</td>
</tr>
<tr>
<td>angular co-ordinate $\phi = 337^\circ$</td>
<td>8</td>
<td>23</td>
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