THE EFFECT OF THE SPEED ON THE BLOW-BY OF AN INTERNAL COMBUSTION ENGINE

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Abstract - Internal Combustion Engines are the generators of energy for many transportation applications, but they still have an overall low efficiency due to mechanical and thermal losses. The combustion chamber is the core element of the engine and it ought to be perfectly sealed, however some of the gas leaks toward the crankcase due to imperfect sealing of the rings. This leakage is known as blow-by and affects efficiency, correct lubrication and emissions.The aim of this paper was to understand in a more detailed way, how the engine speed could affect the sealing efficiency of a ring-pack.The problem, referred to a turbo diesel engine, was formulated in terms of motion equations for the ringsand gas equations for the inter-ring crevices and solved in ©Ricardo RINGPAK solver. The results were discussed and compared with the reference literature and show that inertia and inter-ring gas pressures drive the sealing behavior of the rings. The blow-by trend resulted to decrease with the engine speed and increase with the load, exception made for the idle condition where the values were totally different to the other cases and didn't follow a trend. Nevertheless, these findings can already be taken into account for further studies or experimental investigations.

Keywords - Blow-by, Internal Combustion Engines, Speed and Load, Piston Ring Dynamics.

I. INTRODUCTION

The power obtained in the engine's combustion chamber is transferred to the user by means of the piston and connecting rod. Piston rings main duty is to seal off the combustion chamber, however, their tightness is not perfect and part of the intake gas mixture is lost toward the crankcase, a phenomenon which is commonly known as blow-by gas. In addition, part of these gasses canreturn to the combustion chamber (blow-back) and be ejected as unburned hydrocarbons.Blow-by has an overall detrimental effect on performances due to the reduction of peak pressure [1], and on lubrication and emissions due to oil transport [2-5].Several studieshave addressed the phenomenon of blow-by. Furuhama et.al. [6] and Namazian et.al. [7], werethe first who formulated the problem equations and provide experimental evidence ofthe influence ofring axial motion (also called axial flutter) on the tightness capacity. However, their analysis did not provide the motion of the rings in other directions.Keribar et.al.[8] and Wannatong et.al. [9], providedmore complete formulations, including also aspects of lubrification and surface properties. In these works it was given evidence to the unstable motion of the rings in the axial direction, called axial flutter, and in the radial direction which was called radial collapse. Lijima et.al. [11], and Przesmitzki et.al. [12],observed experimentally the ring radial motion and defined the phenomenon of the ring radial collapse as a contact lift-off between ring face and cylinder liner. Both references [11-12], states that increasing thespeed and load of the engine, increases the possibility for the rings to undergo a radial collapse.Tian in [13, 14], Rabutè et.al. [15] and Cheng et.al. [16]has moreover included the ring twist in their models. The ring-liner and ring-piston interface contact depends on many parameters. Some are related to the design, some others to the material and surface properties, and some others on the engine operation regime. The most important operational parameters of the engine are speed and load, which affect directly the piston and ring dynamics. In the reference literature Curtis [22] states that the engine speed is a major parameter affecting the ring dynamics and gas flow. Kim et. al. in [23] states that the blow-by tends to increase with load but decrease with speed. Tamminen in[24] made some experiments and saw that the blow-by and the inter-ring pressures increased with the engine load. Zottin et.al. in [25] saw that the blow-by tended to increase with the engine load, but its peak value was obtained at the medium load. They explained it with the ring instability. Arnault in [5] referring to a diesel engine saw that the blow-by increased with speed and load, and the maximum value was seen at full load and medium speed condition. According to the results of Irimescu et.al.[26], the blow-by decreased with the engine speed and increased with the compression ratio. Rabuté et.al. in [15] saw that the blow by had peak values at high speed and zero load.

This study was focused in understanding the behavior of the ring-pack and the trend of the blow-by for the complete range of engine speed and load of a reference engine. In literature, similar analyses have mainly considered one of the two parameters, or are in opposition with other's results. The present work features a crossed analysis of working conditions of the engine, ranging from idle to maximum load and for two engine speeds, 2000 and 4000 r.p.m..The mathematical model of the problem was formulated in terms of ring and gas dynamics and solved in ©Ricardo RINGPAK. The analysis was applied to a

turbo diesel engine and the performance of the ring-packwas evaluated in terms ofinter-ring pressures, ringaxial and radial dynamics and blow-by gasses.
The results are then discussed and compared between them and with literature. The work ends with the conclusions stemming from the discussions.

II. MODELING APPROACH FOR GAS DYNAMICS

Piston rings arecurved beams with an end gap, Fig.1.This allows the mounting operation and ensures a radial force toward the cylinder liner. This force is an important element of the sealing capability of a ring, while the gap is the main escape route for the gas. The standard solution for automotive applications is a three-ring pack,composed of two compression rings and one oil ring,although in the literature other solutions have been proposed [17]. In order to account for thermal expansion and groove distortion, the ring axial width is smaller than the respective groove height, Fig.2[28].This clearance is responsible for an additional route for the gas to escape by when the rings are in motion.

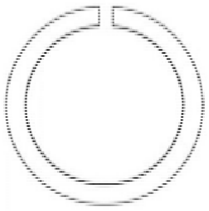


Fig. 1Piston ring

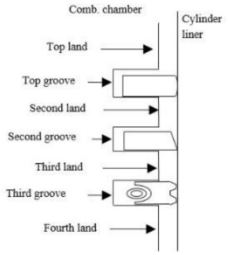


Fig. 2*Schematic of piston, ring-pack and cylinder liner*

To compute the pressure distribution in each zone, the ideal gas equation(1.1)is coupled with the continuity equation (1.2)[9].

$$\left(\frac{dp}{dt}v + \frac{dv}{dt}p\right) = \left(\frac{dm}{dt}\right)RT + mR\left(\frac{dT}{dt}\right) \quad (1.1)$$

$$\frac{dm}{dt} = Q_{in} - Q_{out} \quad (1.2)$$

The flow through the gap can be modeled using the orifice-flow modelfor a laminar and compressible flow[9].

$$Q_{gap} = \frac{C_D A_{gap} P_U}{\sqrt{RT_U}} f_m \quad (1.3)$$

Where $f_m$has the following expression

$$f_m = \left(\frac{P_D}{P_U}\right)^{\frac{1}{\gamma}} \left\{\left(\frac{2\gamma}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma+1)}} \left[1 - \left(\frac{P_D}{P_U}\right)^{\frac{\gamma-1}{\gamma}}\right]\right\}^{\frac{1}{2}} \quad \text{for} \quad \frac{P_D}{P_U} > \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (1.4)$$

Or when the flow is chocked

$$f_m = \gamma^{\frac{1}{2}} \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{2(\gamma+1)}} \quad \text{for} \quad \frac{P_D}{P_U} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad (1.5)$$

The discharge coefficient can be calculated using the following equation[9].

$$C_D = 0.85 - 0.25\left(\frac{P_D}{P_U}\right)^2 (1.6)$$

Often equation (1.1) is simplified with respect to the differentials of volume and temperature. These simplifications are justified by the fact that volume variation in the different inter-ring crevices is negligible compared to the combustion chamber volume, in addition our model will be considered rigid and not include thedeformations of the components. Furthermore, two different assumptions are usually made for the gas temperature[6, 7]. It can be considered to have an adiabatic distribution and thus be calculated for each step of the solution, or have an isothermal distribution and thus be equal to the temperature of the surrounding walls. Experimental results in [6, 7], showed that the isothermal assumption is closer to the reality,this assumption was also made in the present paper.

III. MODELLING APPROACH FOR RING DYNAMICS

Inside the groove, the piston ring has three degrees of freedom, axial, radial and torsion, hence it is necessary to write the equilibrium equations for each direction of motion. Fig. 3 and 4 gives all the forces assumed to act on the ring cross section in axial and radial direction, while Fig.5 shows the moments acting on the ring section.

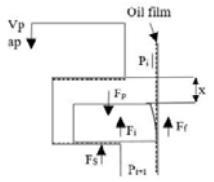


Fig. 3Forces in axial direction

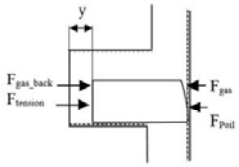


Fig. 4 Forces in radial direction

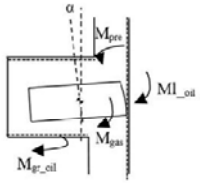


Fig.5 Moments around the central axis

Consequently, the equilibrium equations (Newton second law) for axial, radial and torsional motion can be written according to equations (2.1) to (2.3), [8].

$$m_r \frac{d^2x}{dt^2} = F_P - F_i - F_s - F_f \quad (2.1)$$

$$m_r \frac{d^2y}{dt^2} = F_{gas_{back}} + F_{tension} - F_{gas} - F_{Poil} \quad (2.2)$$

$$I \frac{d^2\alpha}{dt^2} = M_{gas} + M_{gr_oil} + M_{l_oil} - M_{pre} \quad (2.3)$$

In Fig. 5 the tilt angle α can be noticed; this angle is considered positive when the ring is tilted anticlockwise. Inertial force in the radial direction and inertial moment have not been considered in this study due to their low magnitude. Additionally,the asperity forces and moments have not been taken into account in this analysis.

The pressure distribution due to the oil filmcan be determined through the one dimensional Reynolds equation.

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial P_{oil}}{\partial x}\right) = 6\mu U \frac{\partial h}{\partial x} + 12\mu U \frac{\partial h}{\partial t} \quad (2.4)$$

Once the oil pressure has been calculated, the hydrodynamic force, hydrodynamic shear stress and viscous friction exerted on the ring section can be obtained [9]

$$F_{Poil} = P_{oil} * A \quad (2.5)$$

$$\tau = \frac{-\mu U}{h} + \frac{h}{2}\frac{dP_{oil}}{dx} \quad (2.6)$$

$$F_f = \tau * A \quad (2.7)$$

The ringtension and the ring pretwist can be obtained from references [9] and [10].

$$F_{Tension} = p_{tension} * A \quad (2.8)$$

$$p_{tension} = \frac{1}{3}\left(\frac{\sigma''_b * b^2}{D(D-b)}\right) \quad (2.9)$$

$$\sigma''_b = \frac{3}{2}\left(\frac{Ebf''}{L^2}\right) \quad (2.10)$$

$$T_r = \frac{Eb^3 \ln\left(\frac{D_o}{D_I}\right)}{3(D_o + D_I)} \quad (2.11)$$

$$M_{pre} = T_r(\alpha - \alpha_0) \quad (2.12)$$

IV. SIMULATION

According to the equations described above, the inter-ring dynamics, inter-ring pressures and gas flows can be affected by a large number of parameters. These parameters can be operational (speed, load, temperature), mechanical (material, elastic modulus, thermal conductivity), may depend on design (areas, volumes) or may alsodepend on the properties of the lubrication oil and surface textures.

In each of the equations (2.1) to (2.3), the equilibrium is dependent on the applied forces and moments. In the axial direction the main forces are gas pressure and inertia, and when they are comparable in magnitude, the ring position may be unstable and be subject to axial flutter [7]. Under these conditions, the ring loses its sealing capacity and gases canbypass the ring, flowing through the groove channel. In the radial direction, the gas pressure on the ring back is supported by the hydrodynamic pressurein the oil film. The ring tension force is necessary to guarantee a constant static force, since the groove pressure changes through the cycle. In certain conditions, the gas force exerted on the exposed ring facecan overcome the outward forces and push the ring inside the groove.This phenomenon is called radial collapse and when it occurs, gas flows directly between the ring face and cylinder liner [11].

Table 1: values of speed and load used in simulations

Load			
	Idle	***50%***	***100%***
Speed	2000	2000	2000
	4000	4000	4000

The number of parameters affecting the sealing efficiency of a ring-pack is high, so it would be extremely difficult to consider all of them in the space of a scientific paper, hence the decision to study the phenomenon by splitting the parameters. In previous works[19, 20, 21], authors investigated the effect of the ring gaps and elastic forces, the effect of the distortion orders and the effect of the inter-ring volumes on the sealing efficiency of the system. In the present work it is proposed to study the effect of the speed when it is given a certain load of the engine, on the behavior of the ring-pack, and on the total amount of gas lost in the crankcase as blow-by. The range of speeds and loads are reported in the table 1 below, and covers the entire range of values for the engine under exam. Simulations are performed considering a direct injection turbodiesel engine with 0.125 m bore and 0.27 m stroke. The ring pack is composed of three rings; a barrel face top ring, a taper face second ring and a mono-piece and spring-loaded oil ring. The minimum oil film thickness is assumed to be 5.3 μm.The input data are implemented in ©RicardoRINGPAK solver.

V. RESULTS AND DISCUSSION

In this section the resultsof the simulations are presented andgroupedfor each case.Results will include the ring axial and radial dynamics and the inter-ring gas pressures. The cumulated blow-by for each case will be given at the end of the section in a single graph, with the aim to understand the differences.

5.1 Idle

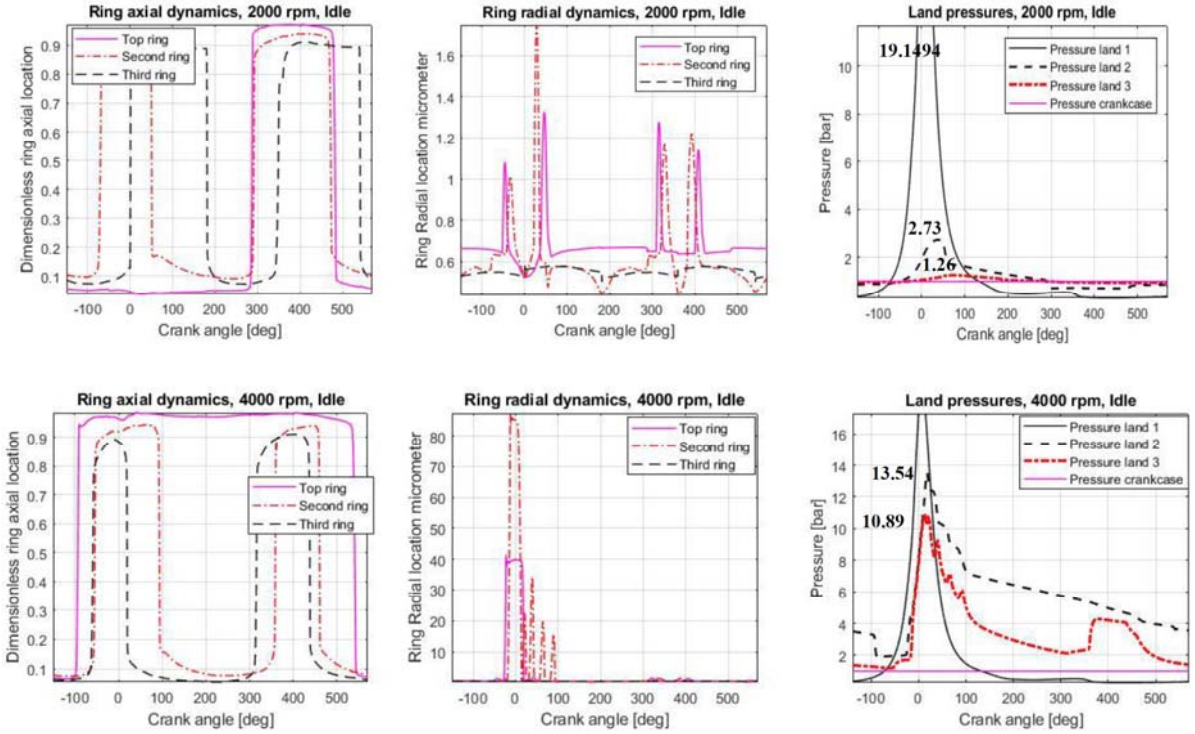


Figure 6 gives the result for the no load condition (idle). At the speed of 2000 r.p.m. the top ring doesn't moves during the first 300 crank angles of the cycle, then after this threshold it moves in synchronous with the second ring, driven faithfully by inertia force. The second ring shows a short lift to the upper groove flank which is related to the inertia in this phase, but as long as the pressure in the second region increases, the ring is constrained to move down until 300 c.a. when it lifts again. The third ring motion is similar to the inertia of the piston and rings, moving between the groove flanks when the inertia force change the direction. The radial motion of all the rings is stable for this level of speed. The third graph shows the inter-ring pressures, in particular the second land pressure is related to the second ring motion because an instant reduction of pressure is seen at the time when the second ring moves from the top to the lower groove flank.The third land pressure is lower in comparison with the second land pressure.

At 4000 r.p.m. the situation becomes more complicated because the inertia has increased with the square of the engine speed. After the top ring moves from the lower to the upper groove flank it allows a large amount of gas to flow downwards, this pressure, which is depicted in the related graph acts underneath the ring and forces it to stay at that position for the entire cycle. This situation is a clear indication that the top ring located at that position is unable to seal the flow. Due to this large gas pressure the second ring is pushed inwards in its groove and suffers multiple radial collapses, which in combination with the axial motion of the ring allows a large amount of gas to flow in the third land.

5.2 50% load

Figure 8 show the results of the simulations for the 50 % load condition. The case of 2000 r.p.m. in the second row shows how the inertia force has affected the system when compared to the previous case. As we can see, the top ring has only a motion during the exhaust-intake stroke which is related to inertia, the second ring is stable while the third ring motion is constrained to inertia only. In this case, the inter-ring pressures are more close to each other and no one of the rings shows any radial collapse. Increasing the engine speed, the ring axial motions is strictly related to the inertia force and follows exactly its trend. On the other hand, both the compression rings undergo to radial collapse, but this behavior doesn't seems to affect in a marked way the inter-ring pressures, which are lower than the previous cases when the radial collapse didn't occur. The only reasonable explanation for this case is related to the axial ring motions which promoted or prevented the gas flow.

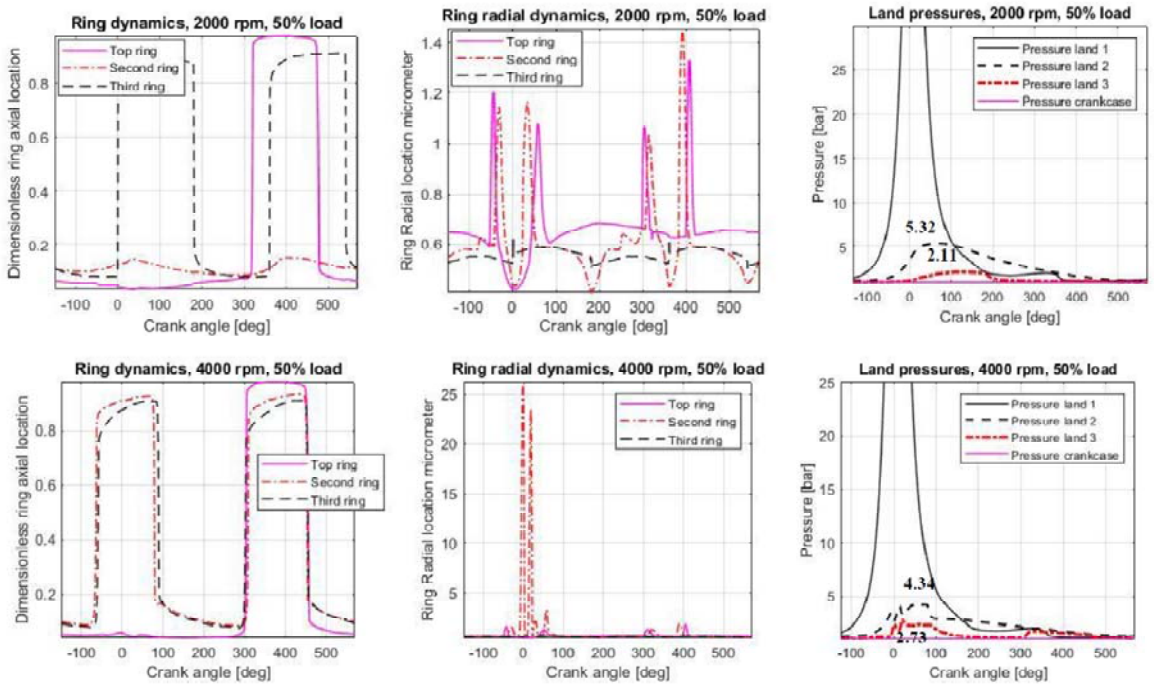
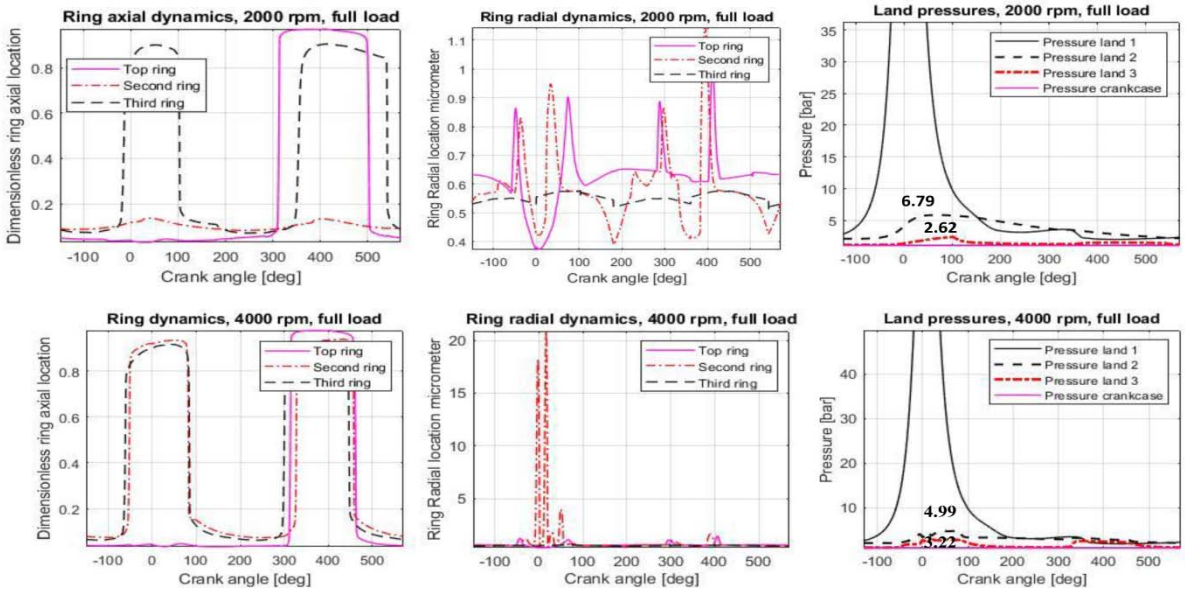


Fig. 8. Results for the 50% load condition.

5.3 100% load

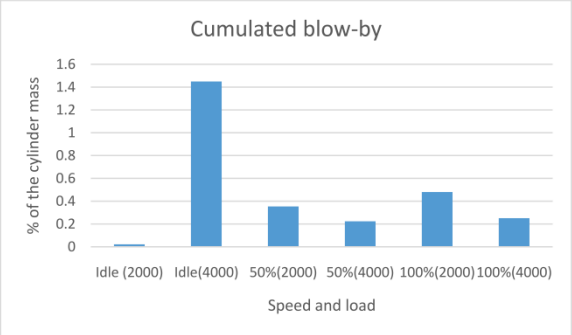
According to figure 9, at 2000 r.p.m. and full load, the ring axial motions is stable: The top ring moves only at the exhaust phase, the second ring is continuously stable while the third ring's motion is related only to the inertia of the piston. No one of the rings did experience a radial collapse at this speed but the second land pressures is relatively high since no one of the rings showed an instability. In this case, the high gas pressure developed inside the combustion chamber has been translated to the second land through the top ring end-gap.

At 4000 r.p.m. the axial motions of the rings are still stable and follow the inertia force but the radial motions aren't anymore, showing a clear radial collapse of the second ring in correspondence to the peak combustion pressure. The stable axial position of the top ring and the reduced cycle time (half the cycle time of the 2000 r.p.m. speed) made the second land pressure to be lower because less gas could reach this area. However, the second ring radial collapse made the third land pressure to be high in relation to the maximum value reached in the second land.



5.4Comparative analysis

In order to understand the real effect of the speed at different loads of engine operation on the amount of gas blow-by, in the figure 10 below will be provided the graph for this parameter, where the blow-by will be related to the cylinder mass and therefore the result will be presented as a fraction.



As we can see, the behavior of the system shows a major change where the blow-by seems to differ in great manner from the zero load (idle) condition to the load condition. In particular, at higher loads it results that the system is able to reduce the lost gasses as long the speed increases, while at idle the situation is completely reversed and the highest blow-by is obtained at the highest speed level. The common reason to this non-linear behavior can be searched at the equilibrium between the gas forces and the inertia forces acting on the rings.

CONCLUSION

The present paper features the tightness analysis of a piston ring-pack for automotive application. The system behavior is analyzed whenthe engine speed at different load levels has been changed. The resultsobtained are in very good correlation with the literature. However, the following important outcome emerge from the discussions:

- Inter-ring pressures and inter-ring dynamics showed to influence each other in a direct manner.
- The inter-ring gas pressures and the inertia force are main forces in an inter-ring pack and affect the amount of gas blow-by.
- The system proved to be highly dependent on the engine speed and load.
- Apart from the idle condition, the blow-by resulted to decrease with speed and increase with load.
- The inter-ring pressures are responsible for the inter-ring dynamics, in particular at low speed, when the inertia force acting on the ring is of low magnitude.
- The inertia force is proportional to the ring mass and the ring acceleration. Since the acceleration cannot be reduced because is dependent on the engine speed, piston rings should be as light as possible.
- At high engine speed the phenomenon of radial collapse is likely to occur.However, at high loads this phenomenon couldn't affect negatively the system.
- Apart from the ring end-gaps which always allow the gas escape, the other major factor resulted to be the ring motion between its groove flanks. As a result, the narrower will be the ring-groove axial clearance and the smaller will be the amount of gas to flow during the ring motion.

In this paperonlypart of the engine speed and load rangehas been considered. The results can be used for comparison with other cases,or as a tool for the analysis and design of new components, including also spark ignition engines. However, the complete analysis can be further extended to other characteristics of the piston ring-pack such as blow-back, pollution, oil consumption or power loss.

Conflict of interests

The authors declare that there is no conflict of interest with third parties or institutions and the outcome of the present work is the result of personal research.

Nomenclature

A ring face area subjected to a given pressure, m^2
A_{gap}gap area, m^2
Ap piston acceleration, m/s^2
b ring radial width, m
BDC Bottom Dead Center
D cylinder bore, m
E ring modulus of elasticity, Pa
$F_f$oil friction force, N
$F_i$inertia force, N
F_{gas}, F_{gas_back} gas pressure in radial direction on the ring section, N
$F_P$gas pressure force in axial direction, N
F_{Poil}hydrodynamic oil pressure in radial direction, N
$F_S$oil squeeze force in the grove, N
$F_{Tension}$ radial force due to ring installation in the cylinder liner, N
f "ring gap at free condition, m
$f_m$flow coefficient
h oil film thickness, µm
I moment of inertia for ring section, kgm^2
L ring length, m
m mass of gas in the i^{th} region, kg
m_r mass of the ring, kg
M_{gas}moment due to gas pressure distribution, Nm
M_{gr_oil}moment due to oil pressure in the groove, Nm
M_{l_oil}moment due to oil pressure in the cylinder liner, Nm
M_{pre} moment due to ring pretwist, Nm
od, idring outer and inner diameter, m
U piston velocity, m/s
uhc unburned hydrocarbons
P pressure, Pa

P_U, P_D pressure up and down the gap, Pa
P_{oil}pressure due to oil film thickness, Pa
Q_{in}, Q_{out}mass flow entering and exiting a crevice, kg/s
Q_{gap}mass flow through the ring gap, kg/s
R ideal gas constant, J/kgK
T temperature, K
TDC Top Dead Center
V volume, m^3
Vp piston velocity, m/s
x, y axial and radial position of the ring, m
α, $\alpha_0$ring twist angle in work and in static conditions, deg
γ specific heat ratio
τ hydrodynamic shear stress, N/m^2
μ oil dynamic viscosity, Pa/s
σ"$_b$bending stress, PA

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