Engine Speed and Load on the Sealing Capacity of a Piston Ring-Pack: A Theoretical Analysis and Comparison with Literature

Erjon Selmani, and Arian Bisha

Abstract—The combustion chamber is ought to be perfectly sealed, however, part of the air and fuel mixture can escape from it. Among the several losses there is the gas flow from the inter-ring crevices, which is always present. This leakage is known as blow-by, and affects efficiency, correct lubrication and emissions. The amount of leakage is dependent on many factors, and among the most important are the engine speed and load, which are able to affect the system through the forces applied on it. The aim of this paper was to understand in a more detailed way how the engine speed and load could affect the sealing efficiency of a ring-pack. For this purpose, a complete range of speeds and loads were used in the simulations. The equations of the ring motions and gas dynamics has been implemented and solved in ©Ricardo RINGPAK solver. The results showed that inertia and inter-ring gas pressures drives the sealing behavior of the rings. The blow-by trend showed to decrease with the speed and increase with the load, exception made for the idle condition where the values were different to the other cases, especially at higher speeds. Among the two parameters, the engine speed resulted to affect more significantly the blow-by trend.

Index Terms—Blow-By, Internal Combustion Engines, Speed, Load, Ring Dynamics.

I. INTRODUCTION

The 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. This phenomenon is commonly known as blow-by gas and was recognized to brink negative effects on performances, lubrication and emissions [1-5], in particular Kim et al in [19] found the blow-by gasses to be responsible for 10 to 30 % of the UHC emissions of a SI engine. The initial studies [6-7] have analysed the ring motions in the axial direction, further studies [8-10] have also included the motions in the radial direction. In references [11-14], the ringpack was accounted also with the third degree of freedom, the twist around the axis of the cross section. There are several factors affecting the ringpack behavior and the resulting blow-by, among the most important are the engine speed and load. Curtis in [18] states that the engine speed is a parameter which may affect the ring dynamics and gas flow. Kim et al. in [19] states that the blow-by tends to decrease with speed, with its peak values reached in the medium speed and load conditions. Tamminen in [20] made some experiments and saw that the blow-by and the inter-ring pressures increased with the engine load. Zottin et al. in [21] saw that the blow-by tended to increase with the engine load, but its peak value was obtained at the medium load, explaining 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. Furthermore, according to the results of Irimescu et al. [22], the blow-by decreased with the engine speed and increased with the compression ratio. Rabuté et al. in [13] saw that the blow-by had peak values at high speed and zero load. However, in any of the references wasn't indicated how the blow-by varies within the entire engine map, or which of the two parameters may be more important to the blow-by rates.

This study was focused in understanding in detail the behavior of the ring-pack and the ongoing of the blow-by for the complete range of speed and load and the combination of the two parameters. In literature, similar analyses have mainly considered one of the two parameters, or has had few concern to investigate the combined effect between them, limiting the anlyses to few values. Furthermore, the models in the literature often does not include all the ring motions, or neglects other details, in addition, in some of the references, the results are discontinuous or even opposite between them. The present work features a crossed analysis of working conditions of the engine, ranging from idle to full load and from minimum to maximum speed. The theoretical 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-pack was evaluated in terms of inter-ring pressures, ring axial and radial dynamics and blow-by gases. The results are then discussed and compared with the reference literature. The work ends with the conclusions stemming from the discussions.

II. APPROACH FOR RING AND GAS DYNAMICS

Piston rings are curved beams with an end gap, Fig. 1, the gap 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 a main escape route for the gas. Typically there are three rings, two compression rings and one oil ring as described in Fig. 2.

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III. SIMULATION

The number of parameters affecting the sealing efficiency of a ring-pack is high. In previous works [15-17], 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 has been proposed to study the effect of the speed and load of the engine on the behavior of the ring-pack, and on the blow-by. The range of speeds and loads are reported in Table 1, and covers the entire map of values for the engine under exam. The speed varies with steps of 500 rpm, while the load with steps of 20% increment nearly. The load for a turbo diesel engine is reflected in an increase of the combustion pressure in the chamber. In literature the effect of those parameters has usually been considered separately, or for a limited range of values. In the present work, the analysis for the entire engine map will allow to understand which of the two parameters is more significant for the blow-by control.

Simulations are performed considering a direct injection turbo diesel engine with 0.120 m bore and 0.26 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. In addition, the liner distortion has been considered as given in previous work [16]. The input data are implemented and solved in ©Ricardo RINGPAK solver.

IV. RESULTS AND DISCUSSION

In this section, the results of the simulations are presented and grouped for each case. In order to not overtask the reader with information and figures, the results will include only the ring axial and radial dynamics and the inter-ring gas pressures, as the most important factors to affect the motions. 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 between cases. Due to the fact that there is a good linearity in the behavior of the system, only the speeds 1000, 2000, 3000 and 4000 rpm will be discussed. While at the comparison section, all the cases will be depicted.

A. IDLE

Fig. 5 shows the results of the simulations for the first three idle load conditions, where each row correspond to the results of a case. In the first row we can see the behavior of the system for 750 rpm and no load. All the rings experience axial motion and no one of the rings did show axial flutter. The top ring motion is driven by inertia in the first phase, and by the gas pressure of the second land in the phase after the 100 c.a. The second ring is driven by the pressure of the

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### Table 1: Speed and Load Values Used in the Simulations

| Load   | Idle | 30% | 50% | 70% | 100% |
|--------|------|-----|-----|-----|------|
| Speed  |      |     |     |     |      |
| 750    | 1000 | 100 | 100 | 100 | 100  |
| 1500   | 1500 | 150 | 150 | 150 | 150  |
| 2000   | 2000 | 200 | 200 | 200 | 200  |
| 2500   | 2500 | 250 | 250 | 250 | 250  |
| 3000   | 3000 | 300 | 300 | 300 | 300  |
| 3500   | 3500 | 350 | 350 | 350 | 350  |
| 4000   | 4000 | 400 | 400 | 400 | 400  |

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third land in the first phase (up to 0 degree c.a.) and by inertia from 300 c.a. and on. The third ring is driven only by inertia during the entire cycle. The radial motion of the second shows a normal contact between the ring faces and the liner, where no radial collapse is encountered for this case. In the third figure are given the inter-ring pressures and their maximum values. As we can see, both the second and the third land pressures overcome the top land pressure during the cycle, in addition, the pressure increase in these lands is smooth.

This behavior can be explained with the flow pattern of the gas which can flow mainly through the ring gap. In the second row are given the results for 1000 rpm, the trend is similar to the upper case except for the reduced motion of the top ring in the range -100:100 c.a. which corresponds to the power phase of the cycle. Due to this reason, the second land peak pressure is lower than the previous case.

The third row of Fig. 5 depicts the 2000 rpm case. The top ring axial motion is slightly different from the previous cases, because it is stable at the groove floor for a long time and lifts up only between 300 and 500 c.a. This lift, which occurs at the same time with the second ring, is initially promoted by the gas pressure and then from the inertia force.

This relationship also applies to the first lift of the second ring, while the third ring motion is strictly related to inertia. The second land pressure is related to the second ring motion, because it marks an instant reduction at the time when the second ring moves from the top to the lower groove flank. The radial position of all the rings is stable, except for the second ring which seems to suffer the increase of speed.

In Fig. 6 are given the results for the speeds 3000 and 4000 rpm. In comparison to the previous cases, the axial and radial motions, as long as the inter ring-pressures, shows relevant differences. At 3000 rpm the top ring is almost always lifted up in the groove, except for a short fall after the peak combustion pressure. The second and the third ring axial motions seems to replicate the inertia force, but with different periods, related to the high pressures in their lands.

Both the compression rings undergo to collapse in the radial direction, and both of the times it occurs when the rings are lifted up. The gas pressure curves show a sharp rise when the rings are collapsed.

At 4000 rpm the behavior is similar, exception made for the top ring which is always lifted in the groove ceiling. The reason is related to an increased inertia force, which can overcome the gas pressure force even after the combustion peak. Both the cases shows similar trend between them, where the top ring is almost always lifted up in its groove, while the other rings show some motions during the power phase and during the exhaust-intake phase (300 – 450 c.a.).

The gas pressures in the second and third lands are abnormally high, and are mainly related to the radial collapse of the top and second rings, which occur in all the cases of this figure.

There are also two cases worthy of note in this figure:
In the first row, the top ring motion from top-to-bottom of the groove reduces the amount of radial collapse for this ring, this makes the pressure in the second and third land to be lower than the other graph.

In the second row, the top ring is stable at the upper flank of its groove but the second ring suffers multiple radial collapses, even after the peak firing pressure. As a consequence, the continued radial collapse of the second ring, combined with the axial position of the top ring, proves to be ineffective in the reduction of the gas flow.

B. 30% LOAD

Fig. 7 show the results of the simulations for the 30 % load condition. According to the first row, when the load is 30% and the engine is running at 100 rpm, the top ring lifts to the upper groove flank between 150 and 250 c.a. when the piston is between the expansion and the exhaust stroke. This motion is caused by the gas pressure of the second land, which is high and acts underneath the top ring. After this point the pressure under the ring falls and inertia overcomes it, forcing the ring to change position in the groove. The last stroke of the cycle is the intake stroke, the gas is close to the ambient pressure and the inertia forces the ring to lift up. Due to these motions, the top ring leaves a large amount of gas to flow downwards and this makes the second land pressure to increase and reach a high value.

Due to this pressure, a high force is generated above the second ring and its motion is prevented until the last stroke of the cycle, when the pressure has fallen and inertia can overcome against it.

The third ring has an axial flutter during the first stroke (compression), this is a result of the effectiveness of the second ring to prevent the gas flow due to its stable position. No one of the rings shows problems with the radial position.

The second row gives the results of the 2000 rpm case.

The engine speed has doubled but the inertia has increased with the square of the speed. Due to this, the ring positions are now more similar to the inertia graph. The top ring lifts only at the last part of the cycle while the second ring shows a short lift before the peak firing pressure.

The lift is caused by the inertia, but while the pressure increases, the gas flowing from above forces the ring to sit down. The third ring lifts only when the inertia force changes direction. Due to the late lift of the top ring, the gas pressure on the second land is low in comparison to the previous case, which indicates that the top ring axial position is crucial on this matter.
Fig. 5. Results for the first three idle conditions

Fig. 6. Results for the remaining idle conditions
In the remaining rows of Fig. 7 are given the 3000 and 4000 speed cases, which are both similar in their trends. The axial motion of the rings is entirely driven by the inertia force acting on them, exception made for the first ring, which is subjected by the gas pressure force until 300 c.a. The main changes with respect to the previous cases are encountered in the radial direction where the second ring suffers a radial collapse, which occurs several times few crank angles after the peak combustion pressures. As can be seen from the graphs, the second land pressure is lower than the previous cases but the second ring encounters problems with the radial collapse.

This behavior is related to the axial position of the second ring, which cannot move from the top to the bottom of the groove due to the very high inertia force, and so, the gas pressure can only push the ring inwards into the groove and flow into the third land.

C. 50% LOAD

In Fig. 8 are given the results for the 50% load case. The results are very similar to the results of Fig. 7. The ring axial
and radial positions are almost the same and the main difference lays on the higher inter-ring pressures in the third graph, but this fact is normal because of the higher peak pressure into the combustion chamber in comparison to the previous case. Similarly to the previous case, the inter-ring pressures are higher at the lower speeds and tend to reduce when the speed increases. Furthermore, when the second ring collapses, the pressure curves for the second and third land shows a stepped rise.

D. 70% AND 100% LOAD

In Fig. 9 given only the results for the 100% load case, since the graphs of the 70% case are very similar to those depicted in this figure. The behavior of the system for each level of speed and for both levels of load are similar to the 50% load case, however, there are also differences between the two. The main difference can be found in the peak pressure of the second land, for the 70% load and 1000 rpm, which is higher than the respective pressure of the 100% load. This difference is related to the time at which the top ring lifts from the bottom of the groove, which in the 70% case occurs earlier. In addition, at the speed of 2000 rpm, the top ring lift occurs earlier for the 70% load case, and this makes the second land pressure to be lower than the 100% load case.
V. COMPARATIVE ANALYSIS

In Fig. 10 are given all the results obtained from the simulations in terms of cumulated blow-by. The cumulated blow-by is calculated making the integral of the mass flow of gas through the third ring and land-groove, which ends up into the crankcase.

According to the theese graphs, except for the idle case, the highest values of blow-by is obtained at the lowest speeds. In addition, the obtained trend shows that the value decreases as long as speed increases. At idle, the values are close to zero until 2000 rpm and then sharply increase. According to discussions, at these speeds, the second ring radial collapse reaches a value which is double in comparison to all the other cases, furthermore, the top ring is continuously located at the upper groove flank. In addition to this, at speeds higher than 2000 rpm, also the top ring suffers the radial collapse. Another trend, which can be seen from figures, is related to the smoothness of the curves and the values between the lowest and the highest speed points. While load increases, the curve smoothness decreases and the extreme point values increases.

Fig. 10 shows all the blow-by curves plotted on the same graph. From this figure is possible to understand in a concise and clear way, how the blow-by could be related to both speed and load. Exception made for the idle case, the blow-by shows to clearly decrease as the speed increases, and to slightly increase as the load increases. These results are in good correlation with the ones of Kim [19] and Irimescu [22], in partial correlation with the results of Arnault [5] but in opposition with the ones of Rabuté [13].
However, in reference [13] the engine was spark ignited, and thus the comparison must be made for that type of engine. Aghdam et.al. in [23] have used a similar theoretical model to match the results of measurements of gas flow. Their results seem to be more in line with those obtained from this paper. In their case, the gas flows increased with load and decreased with speed. However, they referred to a motored condition for the engine and employed only three level of speeds and loads (namely 750, 1500 and 2000 rpm and 7.6, 10.2 and 12.4 compression ratios). A quite similar analysis was done in reference [24], which used the same software to run the model and did some experimental measurements on the gas pressures. In this paper, which investigated three levels of speed 1100, 1300 and 2100 r.p.m., and only the 110% load condition, authors found that the blow-by tended to increase with speed.

From Fig. 10 it is possible to see another important effect arisen from the analysis, the variation of the blow-by gasses with speed and load is more relevant at the lower speeds. As long as the speed increases, the curves are closer to each other and the difference in value reduces. After 2500 rpm, the 70% and 100% load cases have almost the same values. If we neglect the idle case, which is not a useful condition for the normal drive, the highest blow-by for the engine under exam is obtained at 1000 rpm.

In Fig. 11 has been showed the gas pressures for the second land while in Fig. 12 for the third land. As depicted, these curves are almost similar in trend to the gas flow into the crankcase, or blow-by. It is clear that the amount of blow-by is strictly related to the dynamics of the second land pressure, as long as the top land is linked to the chamber and the third land is linked to the second land on the upper side and on the crankcase on the lower one. The higher will be the gas pressure on the second land and the higher will result the blow-by of the engine. The importance of the second land pressure and its relief has been highlighted also in the work of Dursunkaya et.al. [24].

Fig. 10. Cumulated blow-by

Fig. 11. Gas pressures in the second land.
VI. CONCLUSIONS

The present paper features the tightness analysis of a piston ring-pack for automotive application. The system behavior is analyzed when the entire range of engine speed and load levels are varied. The obtained results are in very good correlation with the literature. However, the following important outcome emerged from the discussions:

- The results proved that the system is highly dependent on the engine speed and load.
- The 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 the main forces in an inter-ring pack and affect the amount of blow-by.
- Apart from the idle condition, the blow-by resulted to decrease with speed and increase with load.
- The highest blow-by values were seen at high speed and idle condition, however, this point is not a usual operating point in everyday use of engines.
- The variation of blow-by between the lowest and the highest speed is higher than the variation between idle and full load.
- Between the two parameters, the engine speed showed to have a more relevant impact on the blow-by compared to the load.
- The inter-ring pressures are responsible for the interring dynamics, in particular at low speed, when the inertia force acting on the ring is of low magnitude. As a result, the highest values of gas escape arise at low speed and high load.
- The inertia force is proportional to the ring mass and the ring acceleration. According to results, the blow-by values decrease as the inertia increase. Hence, heavier rings may produce larger benefits.
- At high engine speed, the phenomenon of radial collapse always occurred for the engine under exam. However, at high loads this phenomenon couldn’t negatively affect the system as at the idle condition.
- The gas pressures on the second land resulted to determine the amount of blow-by. As a result, more attention should be given to this region and the way of relief of the gas present there.
- The paper provided a complete map of the blow-by against the engine speed and load for a diesel engine.
- The positive correlation with some of the reference literature is a validation of the model and the analysis with the software employed. However, experimental tests could allow to validate the model and understand the physical behavior of the rings.

VII. 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. No funds have been granted for this work.

NOMENCLATURES

UHC                  Unburned hydrocarbon emissions
SI                  Spark Ignition
m                  [kg] mass of gas in the i\textsuperscript{th} region
p                  [Pa] pressure in the regions
Q_\text{in}, Q_\text{out}             [kg/s] mass flow entering and exiting a crevice
R                  [J/kgK] ideal gas constant
T                  [K] temperature
m_r                  [kg] mass of the ring
F_{\text{friction}}                 [N] oil friction force
F_{\text{inertia}}                 [N] inertia force
F_{\text{gas,rb}}                  [N] gas pressure in radial direction on the ring section
F_{\text{gas,ax}}                  [N] gas pressure force in axial direction
F_{\text{oil}}                  [N] hydrodynamic oil pressure in radial direction
F_{\text{oil}}                  [N] oil squeeze force in the groove
F_{\text{rel}}                  [N] radial force due to ring installation in the cylinder liner
I                  [kgm\textsuperscript{2}] moment of inertia of ring's cross section
M_\text{twist}                  [Nm] Twist moment of the ring
M_\text{rel}= kt\alpha                 torsional stiffness
M_\text{in}                  [Nm] Moment due to gas pressure
M_\text{oil}                  [Nm] Moment due to oil pressure
c.a.       crank angle

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