Simulation of the cylinder bore distortion and effect on the sealing capacity of the ringpack

Erjon Selmani¹ · Cristiana Delprete² · Arian Bisha¹

© Springer Nature Switzerland AG 2019

Abstract

One of the several losses of a combustion chamber is the gas leakage toward the crankcase due to imperfect sealing of the rings, it is commonly known as blow-by and it affects efficiency and emissions. One important factor affecting it is the distorted cylinder bore. The paper describes the equations that rule the blow-by phenomenon in terms of ring and gas dynamics, subsequently, the bore distortion of the cylinder was obtained and its orders calculated through Fourier series. The ringpack capacity in the distorted bore cylinder was then analysed using ©Ricardo RINGPAK Solver. Order zero distortion resulted to be the most important order due to the highest amount of gas lost in the crankcase. Orders one and two didn't showed significant differences from the non-distorted bore case, while orders three and four resulted in high blow-by values, even if their magnitude of distortion was lower in comparison to other orders. The ring pack behavior showed to be affected by bore distortion and its magnitude, however, the blow by phenomenon showed to depend also on other factors in a complex manner.

Keywords Blow by · Internal combustion engines · Piston rings · Cylinder bore distortion

1 Introduction

In 20th and 21th century, the problems of air pollution and fuel shortage have gained large importance due to stringent regulations, and due to the increasing awareness of the pollution effect on the climate change. Moreover, electric propulsion technologies have made car manufacturers to make exhaustive research on increasing the efficiency or reducing the losses of internal combustion engines. Finally, only part of the car manufacturers have invested on the new propulsion technologies and resources. In order to continue to maintain this propulsion system, until the electric systems will be a valid alternative everywhere, internal combustion engines will continue to play an important role.

Part of the power obtained from the fuel combustion is lost away due to thermodynamic and mechanic losses. Piston rings are the mechanical elements interposed between the piston and the liner to guarantee air tightness, oil distribution and oil scrapping. However, their tightness is not perfect and part of the intake gas mixture is lost toward the crankcase, known as blow-by gas. In addition, some trapped gas may go back to the combustion chamber during the exhaust stroke and ejected as unburned hydrocarbons, also known as blow-back. Piston rings perform also secondary duties such as heat transfer, oil control and distribution, and piston support. On one inner and lateral side they make contact with the piston grooves and on the outer side with the cylinder liner. The contact with the piston is highly influenced by tolerances, by ring cross-section and by the dynamics of the motion. On the outer side, the contact is

Handling Editor: Prof. Karabegovi.

✉ Erjon Selmani, erjon.s@hotmail.it; Cristiana Delprete, Cristiana.delprete@polito.it; Arian Bisha, arian_bisha@yahoo.com | ¹Universiteti Politeknik i Tiranës, Tirana, Albania. ²Politecnico di Torino, Turin, Italy.

SN Applied Sciences (2019) 1:314 | <https://doi.org/10.1007/s42452-019-0303-0>

Received: 30 October 2018 / Accepted: 25 February 2019 / Published online: 7 March 2019

influenced by the lateral force exerted by the ring and by the distortion of the cylinder liner. The cylinder bore is subjected to high forces, pressures and temperatures. Additionally, thermal dilatation causes cylinder block expansions and compressions, and this causes axial distortion of the cylinders and adds distortion orders.

2 Ring dynamics and gas flow

Piston rings are curved beams with an end gap. Their main function is to seal the combustion chamber against the crankcase and guarantee an acceptable tightness of the system. Due to the gap, the ring acts as a spring and exerts a radial force toward the cylinder liner, this force is an important element of the sealing capacity of a ring. In Ref. [1] is given a historic review of piston ring technology. The standard solution for the majority of the applications is a three-ring pack, made by two-compression rings and one oil-ring [2], however in literature different solutions have also been proposed [3]. In order to compensate the thermal expansion in axial direction, ring axial width is smaller than the respective groove height. This clearance is responsible for an additional way for the gas to escape when rings are in motion. Inter ring dynamics is highly affected by inter ring fluid dynamics and thus, both phenomenon must be studied together. The ring motion with respect to its groove can be subdivided in three directions: axial, radial and twist.

2.1 Axial dynamics

Main forces acting in the axial direction of the ring cross section are given in Fig. 1 meanwhile Eq. (1) describes the mechanical equilibrium of forces in this direction:

$$m_r \frac{d^2 h}{dt^2} = F_p + I_p + F_F - F_S \quad (1)$$

where m_r = mass of the ring, h = ring-groove clearance, F_p = Force due to gas pressure, I_p = force due to inertia, F_F = Force due to the oil in the ring-liner interface,

Fig. 1 Ring axial movement

F_S = Force generated from the squeeze of the oil in the groove.

2.2 Ring radial dynamics

The radial degree of freedom of a piston ring is imposed by the necessity to compensate thermal expansion of the ring, but also for other reasons, such as piston secondary motion, piston thermal deformation or liner distortion. This motion of the ring gives the so-called “conformability” to the cylinder bore, i.e. the capacity to adapt to the bore surface. In Fig. 2 are given the main forces acting in the radial direction of a ring cross section. When the inward force overcomes the outward one, the ring face lifts-off from the liner, this is called “radial collapse” and suddenly increases the quantity of gas escaped between regions, due to a direct gas flow path. In Fig. 3 is given the contact between the piston ring and the deformed cylinder bore shape under real working conditions.

Equation (2) describes the ring motion in the radial direction and includes the forces depicted in Fig. 2

$$m_r \frac{d^2 x_{cg}}{dt^2} + F_{tension} + (F_{gas_{front}} - F_{gas_{back}}) - F_{oil} = 0 \quad (2)$$

The cylinder bore, which is nominally designed to be perfectly round and straight, in fact is distorted. This distortion is caused by a large number of factors such as the

Fig. 2 Ring radial movement

Fig. 3 Ring contact with the deformed bore

combustion pressure and temperature, thermal expansion and tightening effect of the cylinder head bolts. It introduces new effects on the radial dynamics with respect to the nominal bore shape and according to Fig. 3, we can define the ring center of gravity with reference to relative position, as given in Eq. (3) below:

$$x_{cg} = x + (x_{cg} - x) \quad (3)$$

The ring radial acceleration can be described as:

$$a_r = \frac{d^2 x_{cg}}{dt^2} = \frac{d^2 x}{dt^2} + \frac{d^2 (x_{cg} - x)}{dt^2} \quad (4)$$

The term $(x_{cg} - x)$ refers to the instantaneous twist angle of the ring, when it is small or when its variation is contained, it can be neglected.

Several research were performed on the the radial dynamics of the piston rings. Dukes in [4] experimentally recorded that collapse occurs when ring lifts from lower to upper groove flank. Tian in [5] and Rabutè et al. in [6], discovered that radial collapse is sensitive to operating conditions, to the design of ring cross section and to the extent of lubricated area on ring face. Lijima et al. [7] observed the radial collapse of the ring at high speed and with the ring lifted at the upper groove flank. Przesmitzki et al. [8] described radial collapse with a more complex behavior of the ring dynamics. They observed that ring rotation promoted oil transport and oil transport blocked the gas flow path between rings and grooves, forcing the inward radial collapse.

2.3 Ring twist

The third degree of freedom is the twist of the ring around its circumference axis. Figure 4 shows the positive twist angle α for a ring section. In Fig. 5 is depicted a more realistic case, including also the effect of the bore distortion on the torsional position of the ring with respect to the contact with the liner.

When the twist of the ring is considered, the equation of equilibrium must include all the moments acting on the center of the ring section, according to Eq. (5):

Fig. 5 Twist angle with deformed bore

$$I \frac{d^2 \alpha}{dt^2} + M_s + M_g + M_h = 0 \quad (5)$$

where I = moment of inertia of the ring section [kgm/rad], M_s = Torsional moment [Nm], $M_s = kt * \alpha$, kt = torsional stiffness of the ring [Nm/rad], M_g = moment due to gas pressure [Nm], M_h = moment due to hydrodynamic action of the lubrication oil [Nm].

References [5, 9–11] addresses the topic in higher detail.

The fluid dynamics is modelled using the ideal gas equation and the continuity law, as described in Eq. (6) for a single region:

$$\frac{1}{RT} \left(P \frac{dV}{dt} + V \frac{dP}{dt} \right) = \left(\frac{dm_{in}}{dt} - \frac{dm_{out}}{dt} \right) \quad (6)$$

where P = pressure of a given volume or piston crevice [bar], V = volume of the crevice [m³], T = temperature or the gas in the crevice [K], R = ideal gas constant [J/kgK], m = mass of the gas [kg].

Furuhami in [12] and Namazian in [13], developed the fluid as fully laminar, isothermal and incompressible. These assumptions will be made also in the following work.

Similar models were also developed by Kuo et al. [14], Keribar et al. [9], Koszalka [15], Tian [16]. The most important forces are gas pressure and inertia, friction remains almost constant and of low magnitude. The most critical situation is at high speed and low load, where inter ring pressures are lower and inertial force are higher. Under this situation the second ring flutters, increasing the amount of gas escaped in the crankcase. Ring lift is not uniform over its circumference and seems to lift initially near the gap. Tian in [5] explains in detail also the phases of the blow back.

Fig. 4 Ring twist angle

3 Cylinder bore distortion

The distortion of the cylinder bore and its effect on the piston-ring-cylinder coupling, has been object of investigation from several authors. Dunaevsky in [17] proposed a conformability criterion of the piston rings on a distorted cylinder bore, highlighting the critical distortion for a correct contact pattern. Hennessy et al. in [18] through a test device, measured the increase of blow by due to the increase of bore distortion caused by bolt tightening, and due to surface roughness. Usman et al. in [19] and Rahmani et al. in [20] investigated the effect of the bore distortion on the lubricating film thickness. Both papers concluded that bore distortion strongly affects the lubricating film thickness and engine performance, and its contribution can be even more significant than the liner surface topography. Aronsson et al. in [21] described the error in the heat release calculation model in presence of a distorted bore. Lombardi et al. in [22] investigated the effect on the bore distortion in Al-Si alloy engine blocks. Ghasemi in [23] proposed CAE simulations, through finite element analysis, to evaluate the bore distortion, while Maassen et al. in [24] proposed analytical and empirical methods to optimize the distortion of the cylinder bore.

The aim of this work is to define the effect of the bore distortion on the inter-ring dynamics and sealing performances. In particular, the effect of each single distortion order on the blow-by has been analysed and discussed in high detail.

Fig. 6 Distortion of the bore

The bore distortion of a transversal section in a cylinder liner is given in Fig. 6, and can be modeled by means of Fourier series according to Eq. (7):

$$y_{\alpha} = A_0 + \sum_{k=1}^n A_k \sin(k_{\alpha} + \varphi_k) \quad (7)$$

where: α = position around the circumference, A_k and φ_k = magnitude and phase of the k th distortion order.

The real deformed bore shape is the combination of different orders. Order 0 corresponds to a uniform expansion of the circumference near the top dead center, due to high temperature and pressure in the combustion chamber. Order 1 distortion represents a rigid shift of the cylinder axis with respect to the nominal axis. This shift is the result of the deformation between top and bottom part of adjacent cylinders. Order 2 distortion shows an elliptical deformation shape and arises due to the expansion of the cylinder top part and compression of the cylinder bottom part. Order 3 distortion shows a three-lobe deformation ($n=3$) and is highly influenced by thermal loads while order 4 distortion ($n=4$) shows a typical clover leaf deformed shape and is highly influenced by the tightening forces exerted by the head bolts on the cylinder block. In general, order zero and order one have larger magnitudes of distortion than orders two to four. Generally speaking, orders ≥ 4 show low magnitude on the distortion of the cylinder and thus are not analysed. Figure 7 shows the first five distortion orders.

3.1 Cylinder analysis

In the present work the liner was analysed by means of F.E.M to obtain the bore distortion. The cylinder has 120 mm bore and 148 mm stroke and was analysed at the working condition of 2000 rpm and full load, with a peak firing pressure of around 128 bar. The thrust force exerted by the piston on the cylinder wall was calculated through a kinematic analysis, while the force exerted by the bolts was taken as advised by the engine manufacturer. The distortion orders and phase (Fourier coefficients) have been calculated using ©Matlab, Fig. 8 reports the results for each order and for various liner locations.

Fig. 7 Bore distortion orders

Fig. 8 Distortion orders

From Fig. 8 we can notice order 3 and in particular order 4 distortion, which are one or two orders of magnitude lower if compared to orders 0 or 1. At the end of the figure is depicted the real distorted bore, where all the orders are combined together.

4 Simulation

In Ref. [25] Authors have proposed a similar analysis where the focus was on other parameters or the engine. In the present work, the problem was extended to the effect of the bore distortion on the ring pack performance. In order to investigate the effect of the bore distortion on the sealing capacity of the ring pack, a set of simulations were performed on the test engine. The simulations were performed considering initially the straight bore (nominal diameter), subsequently was introduced each distortion order as it occurred separately, and at the end the nominal distorted bore. The presented results include the ring dynamics, inter ring gas pressures, ring/liner clearances and the total cumulated blow by for each case under exam.

The engine was a common four cylinder diesel engine, run at the speed of 2000 rpm and full load. ©Ricardo

RINGPAK solver has been employed for the solution of the system. RINGPAK is a simulation suite for the analysis of internal combustion engine components, it allows to insert the user defined data through its interface and offers the possibility for various complexity orders of the problem. For the present analysis we selected a 2D axial symmetric model with rigid components, neglecting features such as piston secondary motion or elastic deformation of the components. The bore distortion was inserted as coordinate points, while the minimum oil film thickness on the liner wall was set equal to 5 μm .

5 Results and discussion

In the next figures are depicted the results for each distortion case, starting from Figs. 9, 10, 11, 12, 13 and 14. Then in Fig. 15 are depicted the values of the blow by gases, expressed as percentage of the total cylinder mass at bottom dead center.

In the case of non-distorted bore, top ring remained stable during the entire cycle while second and second ring experienced a motion inside their grooves. This motion was the result of the inertia force and inter-ring pressures. Due to inertia, the second ring lifted by the

Fig. 9 Results for non-distorted bore

Fig. 10 Results for order 0 distortion

Fig. 11 Results for order 1 distortion

end of the compression and the start of intake strokes, and fluttered axially during the beginning of the expansion stroke. The third ring lifted only when the piston changed its motion (between the strokes), confirming the dependence by inertia force. According to the gas pressure curves, pressures in the second and third land did not overcome the pressure in the first land, and this is the desired behavior of every ring pack, synonymous

of a high sealing capacity. According to the third graph of Fig. 9, the second ring shows a radial collapse for a very short crank angle. This behavior, as confirmed by the pressure rise in the third land, gives evidence that when collapse occur, the gas flow suddenly increases. Furthermore, the pressure curve in the second land never crossed over the pressure curve of the top land.

Fig. 12 Results for order 2 distortion

Fig. 13 Results for order 3 distortion

Fig. 14 Results for order 4 distortion

Figure 10 shows the result of the simulations when the cylinder bore is deformed according to order zero distortion pattern. Order zero distortion makes the cylinder liner to expand near the top dead center and increases almost linearly the bore radius as observed from Fig. 8. In these conditions, the compression rings will also expand and follow the new bore radius, increasing their gaps and allowing larger paths for the gas to flow. This fact is

confirmed by the pressure graph, where the second land pressure resulted to be augmented, and at approximately 150 degree crank angle after combustion, reached and overcome the top land pressure.

Due to the increased pressure in the second land, the top ring has lifted up and the second ring has remained sit down. Part of the gas in the second land has returned back to the top land, and the remaining part has flowed in

Fig. 15 Results for the distorted bore

the third land, increasing also the pressure of this region. According to the pressure graph, the second land pressure crossed over the top land pressure at approximately 120 crank angle degree, in addition none of the rings experienced a radial collapse in this case.

Figure 11 shows the results of the simulations when the cylinder bore is deformed according to order one distortion pattern. Order one distortion represents a rigid shift of the cylinder axis with respect to the cylinder block axis. In this case, the ring dynamics, inter ring pressures and ring/liner clearance are almost similar to the non-distorted case, with exception to the peak pressure values in the lands. The reason for this similarity is linked to the fact that this order represents a more linear distortion and less radial one. In this case, the piston and its rings can follow very well the shape of the cylinder, assuring a high conformability to the liner. According to Fig. 11, the second ring undergoes a radial collapse. This behavior is favored by the missing motion of the top ring (relative lift = 0), which blocks the backwards gas flow to the top land. Due to this reason, the high pressure gas pushes the second ring inwards and opens a direct path to flow downwards. Both in this case and in the non-distorted bore, the radial collapse occurred when the ring is located at the top of the groove, as suggested by experimental results from Ref. [7].

Figure 12 shows the results of the simulations when the bore is deformed according to order 2 distortion. The second order represents an elliptical shape distortion, caused by the expansion of the cylinder block upper part and the compression of the lower part. In this case, the behaviour of the system will be not further explained because of its similarity to the order one case, exception made for a slightly higher second land pressure. The low distortion magnitude for this order, together with the good capacity of the ring pack to adapt over it, has made it to perform like the order one distortion.

Figure 13 shows the results for order three distortion. According to Fig. 8, this distortion causes mainly a

reduction of the cylinder bore, which is more marked at the top of it. This distortion will make the rings to conform in a better way with the liner, as long as the distortion magnitude is equal or lower to the minimum oil film thickness, and at the same time it will cause a reduction of the ring gaps.

According to Fig. 13, the top ring lifts only at the beginning of the intake stroke and remains seated during the rest of the cycle. Due to this behavior the top ring could effectively block the gas flow, as confirmed by the pressure curves, which were of very low values during the entire cycle. Only when the top ring lifted, the second land pressure could rise and cross the top land pressure, maintaining a higher value for a short amount of time. It is possible to claim that the top ring has lifted only due to the inertia force, the same as the second and third rings, which lifted in correspondence to the change of the stroke. No one of the rings experienced radial collapse.

Figure 14 gives the results for the order four distortion, which as seen in Fig. 8, has a magnitude one or two orders lower than previous cases.

According to results, the top ring remains stable in its lower groove flank, but the second and third ring suffered continuous variation of their position in the grooves. This evidence shows that regardless of an optimally located top ring in the groove, some gas can flow downwards. This behavior is confirmed through the pressure graph, where the second land pressure shows a rising trend until it occurs a radial collapse of the second ring. After this point, the pressure curve in the second land exhibits a sharp fall, while the pressure of the third land a slight rise.

Finally, in Fig. 15 are depicted the results for the distorted bore in real conditions, namely when all the distortion orders are included and combined together. According to Fig. 15, the top ring seats at the bottom of the groove until the beginning of the exhaust stroke, then it lifts and stays lifted up for the entire exhaust stroke. After a slight fall, it lifts up and doesn't change its

position for the entire intake stroke. Each of the motions are caused by different reasons: the first lift is due to the high pressure gasses in the second land, the second lift is mainly caused by the inertia force, but the gas pressure can also have its contribution in this case. The second ring remains stable at the groove floor for the entire cycle, while the third ring shows a lift from the combustion until the middle of the expansion stroke, then a light axial flutter and a final lift in correspondence to the intake stroke. Due to a stable second ring motion, the pressure in the second land has risen up until reaching the pressure in the top land at the crossover point, located at 160 crank angle degree approximately. From this point and on, the second land pressure was almost always higher than the top land pressure, affecting also the increase of the third land pressure.

Regardless of the high pressures achieved in the several lands, no one of the rings suffered a radial collapse during the cycle.

In Fig. 16 are provided the results of the calculations in terms of blow-by gasses lost in the crankcase, expressed as percentage of the total cylinder mass. The distorted case expresses the nominal blow-by of the given engine, while the other bars are related to the different orders.

Order zero distortion shows to be the worst case in terms of sealing capacity for the ring pack. On the other side, orders one to four presents blow-by values in line with the non-distorted bore profile. The results for order one distortion could be expected, given the shape of the distortion, the but the result achieved by order two distortion is more optimistic than the expectations, due to the elliptical shape of the bore and the increased difficulty of the ring to follow it. Orders three and four distortion shows large values of blow by in comparison to the lower magnitude of distortion with respect to other orders. This result may be caused by an over estimation of the forces and moments causing these distortions, but also may be an indication of new flow paths for the gas flow, probably

between the ring face and the cylinder liner, which were not investigated in this work.

6 Conclusion

The present paper features the tightness analysis of a piston ring-pack for automotive application. The system behavior is simulated evaluating several bore distortion orders. The results, which proved to be in very good correlation with the literature, suggests the following important outcome emerged from the discussions:

- Bore distortion is a very important factor because it affects the sealing capability of the ring pack in different manner for each order.
- Ring axial or radial dynamics are affected from the distortion orders, however, this motion not always explains the amount of gas flowed in the regions.
- The highest values of blow by were recorded in the cases where there was no radial collapse of the rings. On the other side, during these cases the top ring moved between the groove flanks while the second ring remained stable during the entire cycle.
- Apart from the order three distortion, all the cases with the lowest blow by values had in common the top ring stability and the second ring motion.
- The comparison between the blow by for the non-distorted bore (desired result) and the blow by for the distorted bore (real result), can be used as a benchmark for further improvements.
- Order zero distortion considerably reduces the ring sealing capability. The high gas pressure and temperature can't be avoided, but the cylinder block should be designed to reduce the magnitude of distortion for this order.
- Orders one and two are roughly caused from the same factors, in addition, their results are both similar to the

Fig. 16 Blow by for the different distortion orders

non-distorted bore shape. Hence, reducing these factors could double the benefits.

- Orders three and four must be studied more carefully. Although they show a lower distortion magnitude, their blow by values are relatively high.
- Although the speed of 2000 rpm is a frequent speed of operation, a wider range of speeds should be investigated in order to generate a map of the ring pack behavior.

In this paper only part of the parameters affecting inter-ring dynamics, inter-ring pressures and gas flows has been considered. The results can be used for comparison with other cases, or as a tool for the analysis and design of new components. However, the analysis can be further extended to other characteristics of the piston ring-pack such as blow-back, pollution, oil consumption or power loss. In order to account for the complete system characteristics, further parameters such as piston and rings design, ring-liner conformability, piston secondary motion and surface properties should be considered.

Compliance with ethical standards

Conflict of interest The authors declare that they have no conflict of interest.

References

- Economou P, Dowson D, Baker A (1982) Piston ring lubrication-part 1. The historical development of piston ring technology. *J Lub Tech* 104:118–126
- Ellermann J, Röhrle MD, Schelling H (1981) Oil consumption and blowby of truck diesel engines-test bench results, No. 810937. SAE Technical Paper
- Yoshida H, Sugihara H, Kusama K (1995) Practical use of two piston ring set for gasoline engine, No. 950817. SAE Technical Paper
- de Dykes PK (1947) Piston ring movement during blow-by in high-speed petrol engines. *Proc Inst Mech Eng Automob Div* 1(1):71–83
- Tian T (2002) Dynamic behaviours of piston rings and their practical impact. Part 1: ring flutter and ring collapse and their effects on gas flow and oil transport. *Proc Inst Mech Eng Part J: J Eng Tribol* 216(4):209–228
- Rabutè R, Tian TIAN (2001) Challenges involved in piston top ring designs for modern SI engines. *Trans Am Soc Mech Eng J Eng Gas Turb Power* 123.2:448–459
- Iijima N, et al (2002) An experimental study on phenomena of piston ring collapse, No. 2002-01-0483. SAE Technical Paper
- Przesmitzki S, Tian T (2008) An experimental study of the time scales and controlling factors affecting drastic blow-by increases during transient load changes in SI engines, SAE Paper 2008-01-0794
- Keribar R, Dursunkaya Z, Flemming MF (1991) An integrated model of ring pack performance. *ASME Trans J Eng Gas Turb Power* 113:382–389
- Ruddy BL, Parsons B, Dowson D, Economou PN (1979) The influence of thermal distortion and wear of piston ring grooves upon the lubrication of piston rings in diesel engines. In: *Proceedings of the 6th Leeds-Lyon Symposium on Tribology*, ASME, NY
- Richardson DE (1996) Comparison of measured and theoretical inter-ring gas pressure on a diesel engine, No. 961909. SAE Technical Paper
- Furuhama S, Tosio TADA (1961) On the flow of gas through the piston-rings: 2nd report, the character of gas leakage. *Bull JSME* 4.16:691–698
- Namazian M, Heywood JB (1982) Flow in the piston-cylinder-ring crevices of a spark-ignition engine: effect on hydrocarbon emissions, efficiency and power, No. 820088. SAE Technical Paper
- Kuo T-W et al. (1989) Calculation of flow in the piston-cylinder-ring crevices of a homogeneous-charge engine and comparison with experiment, No. 890838. SAE Technical Paper
- Kozsárfi G (2004) Modelling the blowby in internal combustion engine. Part I. A mathematical model. *Arch Mech Eng* 51(2):245–257
- Tian T et al (1998) Modeling piston-ring dynamics, blowby, and ring-twist effects. *Trans Am Soc Mech Eng J Eng Gas Turb Power* 120:843–854
- Dunaevsky VV (1990) Analysis of distortions of cylinders and conformability of piston rings. *Tribol Trans* 33(1):33–40. <https://doi.org/10.1080/10402009008981927>
- Hennessy M, Barber GC (1995) The effects of cylinder wall surface roughness and bore distortion on blow-by in automotive engines. *Tribol Trans* 38(4):966–972. <https://doi.org/10.1080/10402009508983494>
- Usman A, Cheema TA, Park CW (2015) Tribological performance evaluation and sensitivity analysis of piston ring lubricating film with deformed cylinder liner. *Proc Inst Mech Eng Part J: J Eng Tribol* 229(12):1455–1468. <https://doi.org/10.1177/1350650115581029>
- Rahmani R, Theodossiades S, Rahnejat H, Fitzsimons B (2012) Transient elastohydrodynamic lubrication of rough new or worn piston compression ring conjunction with an out-of-round cylinder bore. *Proc Inst Mech Eng Part J: J Eng Tribol* 226(4):284–305. <https://doi.org/10.1177/1350650111431028>
- Aronsson U et al (2012) Analysis of errors in heat release calculations due to distortion of the in-cylinder volume trace from mechanical deformation in optical diesel engines. *SAE Int J Eng* 5.4:1561–1570
- Lombardi A, Ravindran C, Sediako D et al (2014) *Metall Mat Trans A* 45:6291. <https://doi.org/10.1007/s11661-014-2580-y>
- Ghasemi A (2012) CAE simulations for engine block bore distortion, SAE Technical Paper 2012-01-1320, <https://doi.org/10.4271/2012-01-1320>
- Maassen F et al. (2001) Analytical and empirical methods for optimization of cylinder liner bore distortion, No. 2001-01-0569. SAE technical paper
- Delprete C, Bisha A, Selmani E (2018) Gas escape from combustion chamber to crankcase, analysis of a set of parameters affecting the blow by. In: *International conference “new technologies, development and applications”*. Springer, Cham

Publisher's Note Springer Nature remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.