The use of the gas flow model to improve the design of the piston-rings-cylinder system of a diesel engine

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Abstract. The article presents the results of simulation research aimed at assessing the impact of selected dimensions of the rings and piston on the ring-pack operation and developing proposals of changes that will improve its performance. The object of the research was a diesel engine for delivery vehicles. A mathematical model of the ring-pack developed by the author of this work was used for the research. The results have shown that with restrictions imposed by the engine manufacturer regarding the scope of modernization, it is possible to significantly reduce or even eliminate the reverse gas flow from the inter-ring space to the combustion chamber, but this must be done at the expense of increased blow-by. The effect of proposed changes on the axial displacement of the rings in the grooves has also been discussed.

1. Introduction

The main task of a ring-pack is to prevent leakage of the working gas from the combustion chamber to the crankcase. These leaks, called blow-by, worsen the performance and efficiency of the engine. However, minimizing the blow-by is not the only function of the piston-rings-cylinder system. In addition to the good sealing of the combustion chamber, this system should provide low resistance to movement (friction in the PRC system could amount up to 50% of mechanical losses of the entire engine [1]) and high durability, and low lubricating oil consumption at the same time. Increased engine oil consumption is undesirable because it requires refilling and increases service costs, but above all increases the emissions of hydrocarbons and particulate matters [2-6].

Improving the design of the system is not easy. Firstly because of the few functions that it performs. It turns out that changes in design that reduce blow-by often increase oil consumption. Measures to reduce oil consumption can lead to increased resistance to movement and wear intensity [4]. This forces the designers to look for compromise solutions. Secondly, because a large number of design parameters decide about the operation of the ring-pack. Due to the large number of these parameters, there are a huge number of possible design variants. Preparing many variants and their empirical testing means a long time and enormous costs of research. These costs are so high because this system can practically be tested only when mounted in an engine and when the engine is running on a dynamometer.

To reduce the time and cost of testing, simulations with the use of mathematical ring-pack models are carried out. Advanced models allow to study the effect of various design parameters on the operation of the ring-pack [7-19].

This paper discusses such simulation research carried out using the mathematical model of the ring-pack developed by the author of this work. These research was part of the wider work aimed at
meeting the Euro 6 standard by the engine. The engine was produced by an independent manufacturer and was used in delivery vehicles.

2. Objective and assumptions
The purpose of this work was to assess the possibility of improving the performance of the ring-pack of the engine by modifying the selected dimensions of the compression rings and piston.

In setting the assumptions to the simulations, the manufacturer was guided by changes introduced by other large manufacturers of similar engines. There was a tendency to increase some clearances in the piston-rings-cylinder assembly, including a large increase in the joint gap of the second ring, and thus reduce the blow-back (blow-back is a reverse gas flow from inter-ring spaces in direction of the combustion chamber, what happens when pressure in the inter-ring space is higher than in the combustion chamber) and axial displacements of the rings in the grooves at the expense of increased blow-by. Such modifications were done because results of many research indicated that blow-back causes increased oil consumption and emissions of hydrocarbons and particulates [20–21]. The axial displacements of the rings are unfavorable, because in addition to increasing the gas flow and pumping lubrication oil, increase the wear of the side surfaces of the rings and grooves, thus reducing the durability of the engine.

Dimensions, which the manufacturer allowed to change, are the joint gaps of the top and second compression ring \( G_1 \) and \( G_2 \) and the diameters of the piston between the top and second ring \( D_{2u} \) and \( D_{2d} \) and below the second ring \( D_{3u} \) (figure 1). However, these dimensions could only be changed in the direction of increasing the clearances, i.e. the joint gaps could be increased and the piston diameters reduced. Differentiation of diameters in the upper and lower part of the second land was allowed – in the initial piston these diameters were equal, i.e. \( D_{2u} = D_{2d} \). The engine manufacturer did not allow the possibility of reducing the clearances for fear of seizing and damaging the engine. Particularly, as it was also planned to increase the maximum power of the engine.

The changes in the above dimensions that would reduce the blow-back and lower the pressure in the inter-ring spaces, and minimize the number of axial displacements of the rings in the grooves were to be sought. Since the modifications could only consist in increasing the clearances, the increase in the blow-by was counted. However, it was expected that this increase would be possibly small.

3. Object and method

3.1. Engine specification
The engine was a 4-cylinder, direct injection, turbocharged diesel engine with a displacement of 2.6 dm\(^3\) and compression ratio 17.5, with intercooler and exhaust gas recirculation. The engine had a single overhead camshaft and 2 valves per cylinder and a Common Rail injection system. The engine block was made of cast iron and nominal diameter of cylinders was 94 mm. The stroke of the piston was 95 mm. The aluminum piston had a cast iron insert under the top compression ring. According to technical specification, the maximum power of the engine was 85 kW obtained at 3600 rpm and the maximum torque was 250 Nm in the range of 1800-2200 rpm.

The ring-pack of the engine consists of three rings (figure 1): a keystone, barrel faced compression ring made of heat treated spheroidal graphite cast iron with plasma plating, taper faced compression ring made of heat treated spheroidal graphite cast iron, and a double-beveled, chrome plated oil control ring made of standard grey cast iron equipped with a spiral spring. The nominal dimensions of the rings are presented in table 1.

3.2. Model of the ring pack
An integrated model of gas flow through the crevices between the piston, rings and cylinder and ring displacements in the ring grooves was used in this study. In the gas flow model the piston-rings-cylinder seal was treated as a labyrinth seal composed of a series of stages connected by throttling passages. The labyrinth stages are formed by the inter-ring and behind ring spaces, and the throttling
passages correspond to the ring-end gaps and the clearances between the side surfaces of the rings and grooves (figure 1). Thermal deformations and positions of the rings in the grooves are taken into account when determining the instantaneous volumes of the stages and cross-sections of the throttling passages. The axial positions of the rings in the grooves are determined allowing for the following forces acting on them: gas pressure, inertia and friction against the cylinder. The gas pressure in individual labyrinth stages are determined applying the principles of preservation of mass and energy and the ideal gas law. The gas flow rates through the individual throttling passages are calculated assuming that the flow is isentropic, including cases of subcritical and critical flows, and taking into consideration the empirical flow coefficients. A detailed description of the model has been presented in previous works [22-23].

**Table 1.** Nominal dimensions of the rings.

|                | Height [mm] | Thickness [mm] | Joint gap [mm] |
|----------------|-------------|----------------|----------------|
| Top ring       | 3.00        | 3.90           | 0.35           |
| Second ring    | 1.75        | 3.90           | 0.35           |
| Oil ring       | 3.00        | 4.05           | 0.35           |

**Figure 1.** Drawing of the engine ring pack and corresponding orifice-volume representation used in the model (1 – top land, 2 – region behind the top ring, 3 – second land, 4 – region behind the second ring, 5 – third land, 6 and 7 – regions behind and below the oil ring, \( p_i \), \( T_i \) and \( V_i \) – pressure, temperature and volume of gas in the \( i \)-th region, respectively, \( p_{ind} \) – pressure in the combustion chamber, \( p_{cc} \) and \( T_{cc} \) – pressure and temperature in the crankcase, \( m_{ij} \) and \( A_{ij} \) – mass flow rate and area of flow from region \( i \) to region \( j \), \( x_I, x_{II}, x_{III} \) – axial position of the top, second and oil ring in the groove; \( D2u, D2d \) and \( D3u \) – examined diameters of the piston: second land under the top ring and over the second ring and third land under the second ring, respectively).
3.3. Simulations

In the first stage, calculations for the initial, existing version of the piston were made. The dimensions, temperatures and thermal deformations of the parts, as well as the course of pressure in the combustion chamber necessary for the calculations were provided by the engine manufacturer. The pressure in the combustion chamber came from measurements made while the engine was running at rated conditions on the engine dynamometer. The blow-by determined in the calculations was similar to the actual one measured on the engine test stand, so it was assumed that the results of simulations are reliable.

In the next stage, the influence of particular dimensions ($G_1$, $G_2$, $D_{2u}$, $D_{2d}$ and $D_{3u}$) on the operation of the ring-pack, i.e.: the blow-by and blow-back, pressure in the inter-ring spaces and axial displacement of the rings in the grooves, was examined. At this stage, only one dimension was altered, and the others remain unchanged and equal to the initial ones.

In the last stage, all examined dimensions were simultaneously changed and the best from the point of view of the research goals combination of dimensions was sought.

4. Results and discussion

4.1. The initial version

The results of the simulations for the initial, not modified version of the ring pack are presented in figures 2-4. Pressure in the second land achieves relatively high value and drops very slowly. This pressure is higher than pressure in the top land for almost half of the working stroke, the entire exhaust and intake strokes and half of the compression stroke (figure 2). This means that gas flows from the second land to the top land for over two thirds of the engine cycle (figure 3). The top ring changes its position in the groove only once, what is good. However, the disadvantage is that it lifts from the bottom flank of the groove already in the working stroke. The second ring does not change its axial position in the groove at all. The third ring moves up and downward three times during the engine cycle.

Summarizing, the advantages of the existing version of the ring pack is small blow-by and no axial movements of the second ring in the groove. The main drawbacks are high pressure in the second land and very high reverse gas flowrate equal to half of the blow-by (figures 5-6).

![Figure 2. Pressure in inter-ring spaces for the ring pack before modification (denotation as in figure 1).](image-url)
4.2. The influence of individual dimensions

The results of simulations show that the increase in the top ring end gap causes an increase in the blow-by and a particularly big increase in the blow-back (figure 5). That is because the bigger area of flow in the top ring gap results in a bigger pressure rise in the second land during the second half of the compression stroke and the first half of the working stroke (figure 7). This pressure drops much slower than the pressure in the combustion chamber and remains higher than the pressure over the top ring for the rest of the engine cycle forcing more intense gas flow in direction of the combustion chamber. The higher pressure in the second land also causes the bigger gas flow to the crankcase – higher blow-by.

The increase in the second ring end gap results in an increase in the blow-by and decrease in the blow-back (figure 6). The reason of this is lower pressure in the second land. For a big increase in the second ring end gap pressure in this land does not exceed pressure in the combustion chamber and there are no reverse gas flow (figure 7). However, the lower pressure in the second land also affects the displacements of the rings. The second ring changes its axial position in the groove twice and an additional oil ring displacements appear (figure 8).
Figure 5. Influence of the top ring gap on the blow-by and blow-back.

Figure 6. Influence of the second ring gap on the blow-by and blow-back.

Figure 7. Pressure in inter-ring spaces for the top ring end gap enlarged to 0.55 mm (solid line) and for the second ring end gap enlarged to 1.25 mm (dashed line).

Figure 8. Axial positions of the rings in the grooves for the top ring end gap enlarged to 0.55 mm (solid line) and for the second ring end gap enlarged to 1.25 mm (dashed line).
A reduction of the diameter of the piston below the top ring $D_{2u}$ adversely affects both the blow-by and blow-back (figure 9). The results of increasing clearance in this region is similar to the effect of increasing of the top ring end gap. However, the influence on the blow-back is lower, in comparison to increase in the top ring gap, because the bigger clearance means also the bigger volume of the land and smaller increase in the second land pressure.

The influence of the change in diameter of the second land just above the second ring on the blow-by and blow-back is positive, but relatively small (figure 10). The outcome of reducing of the diameter of the piston below the second ring is favorable to blow-back, however unfavorable to blow-by (figure 11).

4.3. The influence of simultaneous changes of several dimensions

The results of simulations revealed that the increase in the top ring gap $G_1$ and the reduction of the diameter of the upper part of the second land $D_{2u}$ regardless of the other examined dimensions always increase the blow-by and blow-back. Therefore, it was decided that $G_1$ and $D_{2u}$ should not be modified and only three other dimensions were changed in further studies. The impact of different combinations of the three dimensions on the blow-by, blow-back and displacements of the rings were studied. Selected results are shown in figure 12. Generally it can be stated that modifications of the examined dimensions in the assumed range do not allow for significant reduction of the blow-back without the increase in the blow-by. What is more, a given reduction in the blow-back is associated with a similar increase in the blow-by regardless of the combination of dimensions used. However, despite similar blow-by and blow-back, differences in pressure courses and ring displacements can be significant for different combinations of dimensions.
Figure 12. Influence of different combination of dimensions on the blow-by and blow-back (detailed dimensions for particular variant are listed in table 2).

Table 2. Second ring gaps $G_2$, changes in second land diameter over the second groove $\Delta D_{2d}$ and changes in third land diameter under the second ring groove $\Delta D_{3u}$ used in simulations presented in figure 12.

| Variant | 1  | 2  | 3  | 4  | 5  | 6  | 7  | 8  | 9  | 10 | 11 |
|---------|----|----|----|----|----|----|----|----|----|----|----|
| $G_2$ [mm] | 0.35 | 0.35 | 0.35 | 0.35 | 0.35 | 0.35 | 0.45 | 0.45 | 0.45 | 0.45 | 0.45 |
| $\Delta D_{2d}$ [mm] | 0 | -0.8 | -0.8 | -1.2 | -1.2 | -1.2 | -1.6 | -0.8 | -0.8 | -0.8 | -0.8 |
| $\Delta D_{3u}$ [mm] | 0 | -0.4 | -0.8 | -0.8 | -1.2 | -1.6 | -1.6 | -0.4 | -0.6 | -0.8 | -1.2 |

| Variant | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 |
|---------|----|----|----|----|----|----|----|----|----|----|----|
| $G_2$ [mm] | 0.45 | 0.45 | 0.45 | 0.45 | 0.45 | 0.45 | 0.65 | 0.65 | 0.65 | 0.65 | 0.65 |
| $\Delta D_{2d}$ [mm] | -1.2 | -1.2 | -1.2 | -1.2 | -1.2 | -1.0 | -1.0 | -1.0 | -1.0 | -1.0 | -1.0 |
| $\Delta D_{3u}$ [mm] | -0.4 | -0.6 | -0.8 | -1.0 | -1.2 | -1.6 | -0.2 | -0.3 | -0.4 | -0.8 | -1.2 |

Figure 13. Pressure in inter-ring spaces for the selected variant ensuring elimination of the blow-back: $G_2 = 0.6$ mm, $\Delta D_{2d} = -1.0$ mm and $\Delta D_{3u} = -1.0$ mm (solid line) and for the selected variant preventing movements of the second ring: $G_2 = 0.45$ mm, $\Delta D_{2d} = -1.0$ mm and $\Delta D_{3u} = -1.0$ mm (dashed line).
Figure 14. Axial position of the rings in the grooves for the selected variant ensuring elimination of the blow-back: \( G_2 = 0.6 \text{ mm}, \Delta D_2 d = -1.0 \text{ mm} \) and \( \Delta D_3 u = -1.0 \text{ mm} \) (solid line) and for the selected variant preventing movements of the second ring: \( G_2 = 0.45 \text{ mm}, \Delta D_2 d = -1.0 \text{ mm} \) and \( \Delta D_3 u = -1.0 \text{ mm} \) (dashed line).

The following changes: \( G_2 = 0.6 \text{ mm}, \Delta D_2 d = -1.0 \text{ mm} \) and \( \Delta D_3 u = -1.0 \text{ mm} \) were considered the most advantageous variant allowing for the complete elimination of the reverse flow (figure 13). Unfortunately, elimination of the blow-back is not possible without two displacements of the second ring (figure 14). For other variants, which eliminate the blow-back either the blow-by is higher or there are more displacements of the rings.

A combination of dimensions that would allow a possibly large reduction in the reverse flow and a small increase in blow-by, but at the same time would not cause the second ring to move in the groove was also sought. The best solution that fulfills this condition was considered to be: \( G_2 = 0.45 \text{ mm}, \Delta D_2 d = -1.0 \text{ mm} \) and \( \Delta D_3 u = -1.0 \text{ mm} \) (figures 13 and 14).

5. Conclusions
The paper presents the results of research aimed at assessing the possibility of reducing the reverse flow of gases from the inter-ring spaces towards the combustion chamber by increasing the compression rings end gaps and reducing the diameters of the ring part of the piston. An advanced mathematical model of the ring-pack was used for this assessment. This work was a part of a wider work aimed at reducing the emission of toxic components from the diesel engine used in delivery vehicles.

The results of the simulation showed that the top ring end gap should not be increased and the diameter of the shelf under this ring should not be reduced because it leads to a very big increase in the blow-by and blow-back.

Increasing the second ring end gap or reducing the diameter of the piston under this ring leads to increased blow-by, but at the same time causes a large reduction in the blow-back.

Increasing the diameter of the piston over the second ring slightly reduces the blow-by and blow-back, however in combination with changes in other dimensions it can significantly affect the pressure in inter-ring spaces and axial displacement of the rings in the grooves.

As a result of the work, two solutions were proposed: the first one, which allows to completely eliminate the reverse gas flow from the inter-ring space towards the combustion chamber. In this solution, the top ring moves axially in the groove once and the second and oil ring two times during one engine cycle. And the second solution, which ensures no displacement of the second ring and only two displacements of the oil ring (the top ring moves once in all examined versions), and a very large reduction of blow-back – 3.5 times in relation to the initial version. In both cases, the benefits in blow-back come at the cost of increased blow-by, respectively by 65% and 40%.
Summarizing, the application of the model allows to analyze the impact of many design variants on the operation of the ring-pack and to select one or more variants that will be manufactured and tested. This allows for a very large shortening and a huge reduction in the costs of an engine tests.

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