LUBRICATING OIL CONSUMPTION AT LOW TEMPERATURES

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Abstract

The problem of lubricating oil consumption still seems to be important one at least because of the use of DPFs in exhaust systems of diesels. Even a minor consumption can be a source of particulate matter that blocks filters. On the other hand formation of oil film parameters that guarantee minimum oil consumption could lead to the increased friction losses. All these remarks relate to the nominal temperature of operation. Situation can differ at temperature of cold start.

Presented paper deals the problem of oil consumption at lower temperatures. The observation concerning friction losses under these conditions are mentioned as well. Parameters influencing the lubricating oil consumption, oil consumption at nominal temperature of engine run, oil consumption at low temperature are presented in the paper. As well course of oil film thickness, layer thickness and friction force for upper compression ring at engine nominal temperature of operation, course of oil film thickness, layer thickness and friction force for upper compression ring at lower engine run temperature, ring face wetting level at nominal temperature of operation, ring face wetting level at lower temperature of operation, lubricating oil consumption and friction losses generated by ring pack of single cylinder are subject of the paper

Keywords: piston rings, oil film, lubricating oil consumption, friction losses

1. Introduction

Consumption of lubricating oil does not make a considerable component of operational costs on modern engines. However, even a superficial analysis of values presented by car manufacturers shows that the range of admissible oil consumption is pretty wide from quite negligible ones up to even 1 litre per every 1000 km. One could wonder how justify so wide range of oil consumption considered as admissible. Taking into account prices of high quality synthetic oils, consumption of

1 litre per every 1000 km increases the average cost of car run by almost 1000 zł. One of the causes of high but still admissible oil consumption could be short distances when engine can not reach a required operational temperature. Usually such situation happens in cities on short travels to and from workplace. Following paper will present oil consumption defined in such conditions, i.e. during engine warm-up or idle run when waiting on red lights.

The viscosity of lubricating oil is high at temperatures lower than that provided by the producer for conditions of stable operation. At low temperatures one can expect high thickness of oil film generated by the pack of piston rings. On the other hand high oil layer thickness could lead to the increased oil consumption resulting from its unintentional scrapping towards the combustion chamber as well as from its evaporation or combustion being a consequence of contact with a flame in combustion chamber. The paper will deal with the first of phenomena mentioned.

2. Parameters influencing the lubricating oil consumption

In the region of piston sealing labyrinth the lubricating oil consumption mainly depends on:

- pressure of individual rings against the liner,
- geometric parameters of ring face,
- mutual distances between ring grooves and distance between oil control ring faces in particular,
- operational parameters of combustion engine,
- lubricating oil viscosity.

Precise definition of the mentioned parameters is not a simple task while some of them significantly affect the oil penetration towards the combustion chamber which directly corresponds with the oil consumption. The ring specific pressure against the liner definitely is the most important parameter. Unfortunately, determination of this parameters is very difficult because a direct measurement similarly to its estimation according to a mathematical model is burdened with a considerable error. Ring free gap could be a measure of compression ring pressure. Measurement of the free gap should be performed exclusively by the scanning because even the lightest pressure e.g. of the dial gauge could lead to the ring deformation. Fig. 1 presents a picture of scanned free gap of both compression rings.

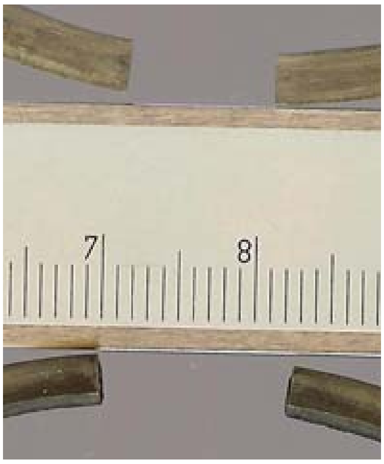

Fig. 1. Free gap of the AXD 2.5l R5 TDI engine upper and lower ring

The free gap of upper compression ring is 9.5 mm, while of the lower one is 11.8 mm. Assuming the even pressure distribution the ring pressure equals 0.12 MPa and 0.15 MPa for the upper and lower ring, respectively. The new rings reveal an increased pressure close to the gap but after the running-in period this pressure becomes even. However, on the grounds of relation between the free gap and the greatest dimension of free ring contour one can evaluate the level of pressure evenness. Fig. 2 presents scans of the analyzed rings.

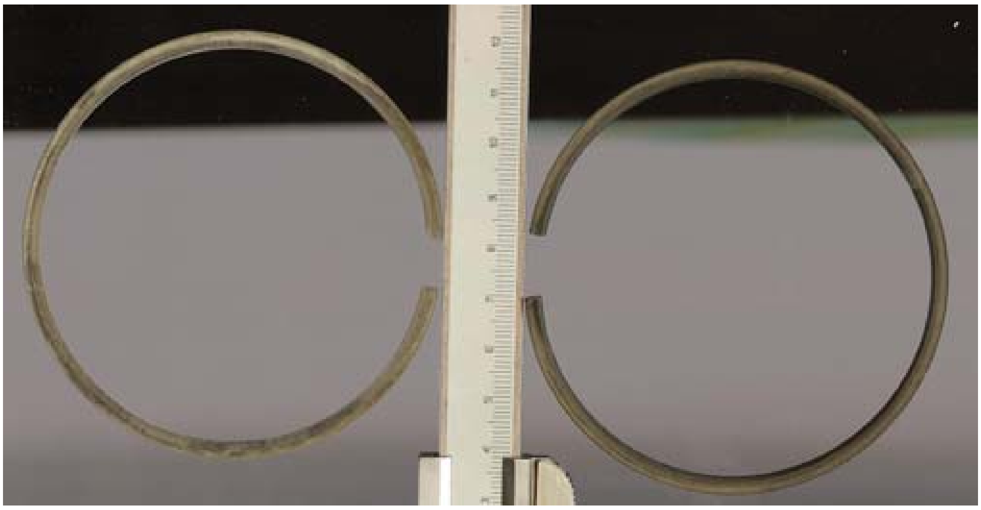

Fig. 2. Upper (left) and lower compression ring of the AXD 2,5l R5 TDI engine

The presented scans allow to determine precisely the maximum dimensions of free ring contour as 84.47 and 85.58 mm for A1 and A2, respectively. For rings of even pressure distribution these dimensions – marked in Fig. 3 with letter A – should be 84.64 and 85.78 mm, respectively.

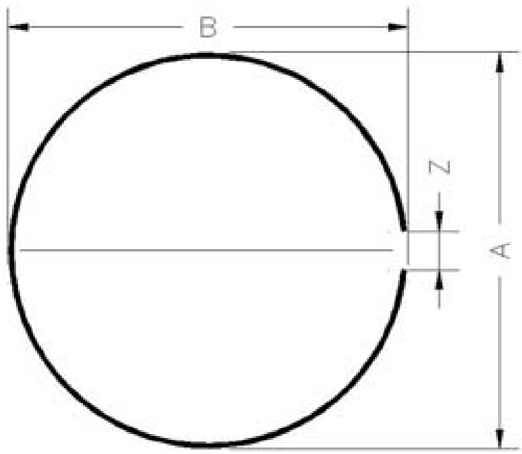

Fig. 3. Overall dimensions of a free ring

The ring overall dimensions have been defined on the basis of free gap assuming the pressure distribution described with a harmonic array. A substitutional model in a form of rigid sections array connected with joints has been assumed for the ring [1]. The deformability of joint equals the real deformability of ring section adjacent to the joint. The substitutional model used for computations consisted of 360 sections. In order to minimize the error resulting from ring discretization a correction has been utilized consisting in estimation of coordinate change tendency when sections of substitutional model are doubled [1].

Earlier proved apparently insignificant difference of order of 0.02 mm in ring overall dimensions A practically means a disappearance of ring pressure against the liner in gap vicinity. In order to prove this a ring pressure distribution has been assumed as presented in Fig. 4. The obtained overall dimensions of free ring are A1 = 84.52 and A2 = 85.64 mm.

Therefore the difference between measured and calculated overall dimensions is 0.05 and 0.06 mm respectively, which makes that the obtained results are close to the error of real ring overall dimensions measurement. This proposition is also confirmed by the B dimension (see Fig. 3), which is 83.00 and 83.40 mm for real rings, and 82.96 and 83.34 respectively for the pressure distribution as in Fig. 4. So differences are just 0.04 and 0.06 mm.

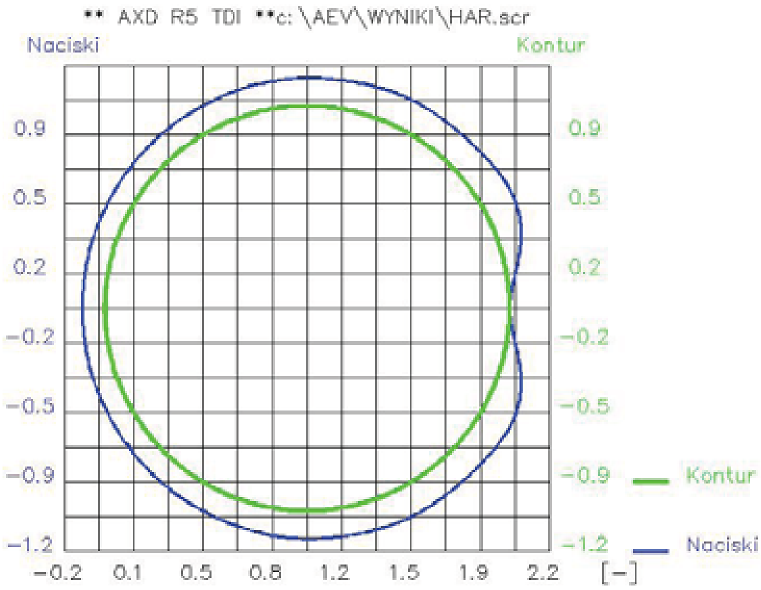

Fig. 4. Distribution of ring pressure against liner described with harmonic array for a case of pressure absence in gap vicinity

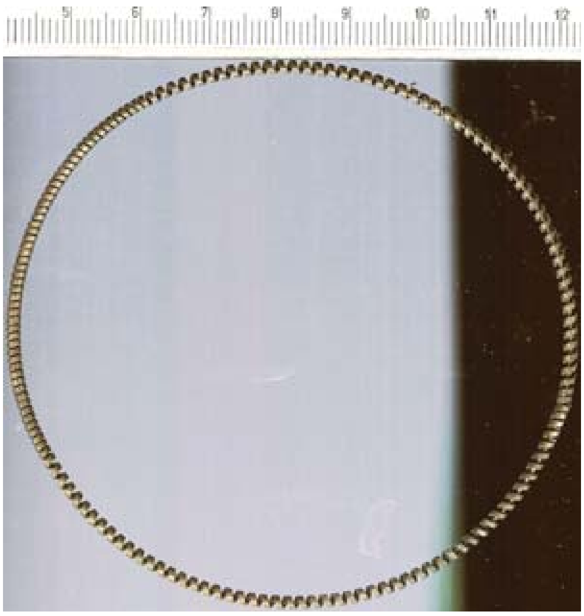

Fig. 5. Coil spring pressing on the oil control ring to the cylinder liner

A specific pressure of oil control ring plays a key part in calculations of lubricating oil consumption. In this case definition of this pressure is relatively simple because it is generated by a spring squeezed with a circumferential force of 70 N. Without going into details on transformation of circumferential force into radial specific pressure of oil control ring face this pressure has been defined as 2.2 MPa. This value has been accomplished for ring face axial height of 0.4 mm and nominal diameter of 51 mm. Fig. 5 presents the type of coil spring pressing on the oil ring against cylinder liner.

Beside pressure of individual rings their mutual distance plays very considerable part in the simulations of lube oil consumption. Fig. 6 presents the ring carrying part of piston which allows to realize about rings location.

Fig. 6. View of the piston ring zone after engine running-in

Fig. 6 presents also clearly noticeable increasing height of upper ring face shining part close to the gap. This means that radial wear of this area is high and specific pressure vanishes.

3. Oil consumption at nominal temperature of engine run

Taking into consideration the earlier analyzed parameters of piston-cylinder group a computer simulation of specific oil consumption resulting from its sweep towards combustion chamber has been carried out. The courses of oil film thickness, oil layer thickness and friction force of upper ring against the liner for steady temperature of 110°C are shown in Fig. 7.

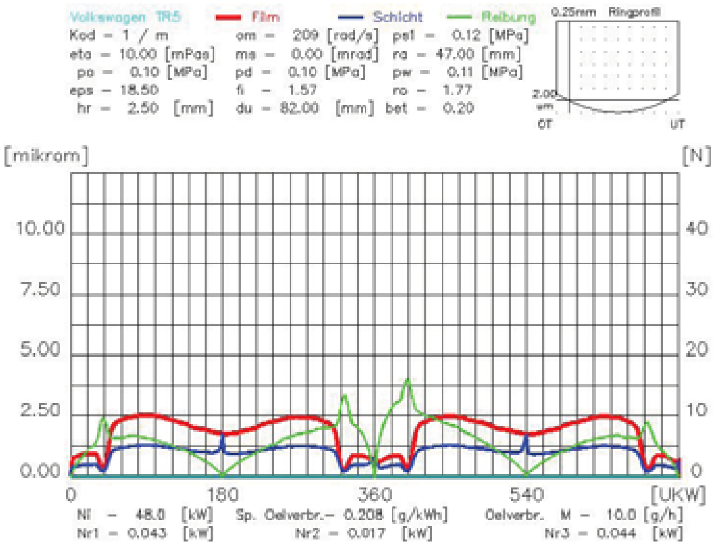

Fig. 7. Course of oil film thickness, layer thickness and friction force for upper compression ring at engine nominal temperature of operation

Above the graph there is data taken for model representation of oil consumption while below the graph are results including the oil consumption of 0.208 g/kWh for the oil viscosity η = 0.010 Pas.

4. Oil consumption at low temperature

Under unsteady conditions when engine temperature is lower than the temperature chosen by manufacturer as the nominal one, the lubricating oil has higher viscosity which affects the friction resistance and oil consumption. Fig. 8 presents a similar situation as in Fig. 7 but the oil viscosity is 0.07 Pas.

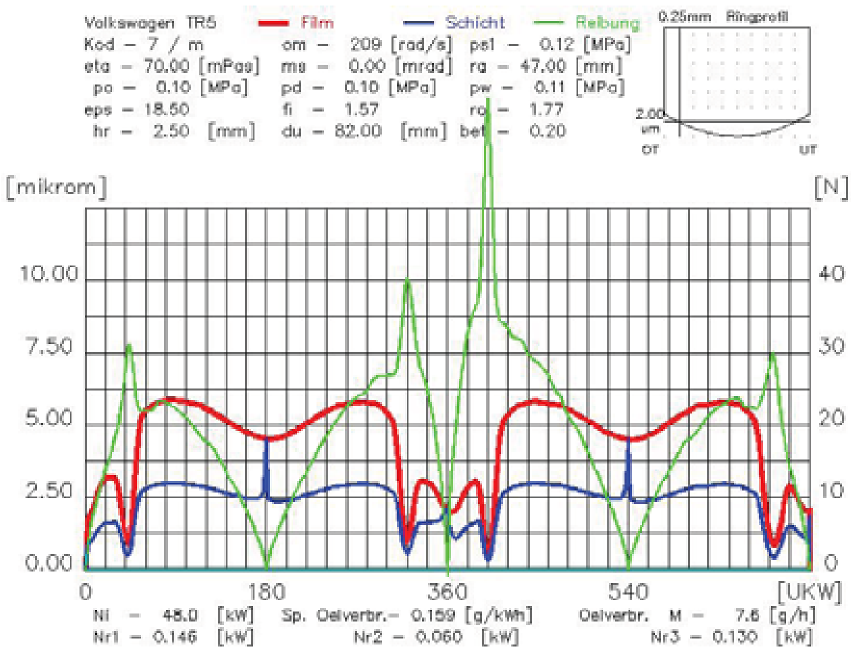

Fig. 8. Course of oil film thickness, layer thickness and friction force for upper compression ring at lower engine run temperature when oil viscosity is 0.070 Pas

Decrease of engine temperature and lubricating oil as well causes an increase in basic parameters of oil film but at the same time the oil consumption falls from 0.208 to 0.159 g/kWh. The results obtained are difficult to explain but one can presume that two phenomena affect the oil consumption. On one hand an increase in oil film thickness at lower temperature promote the increased oil consumption but on the other hand the level of ring face wetting does not increase itself which eventually promote a decrease in oil consumption. The comparison of ring face wetting can be carried out analyzing graphs in Figs. 9 and 10.

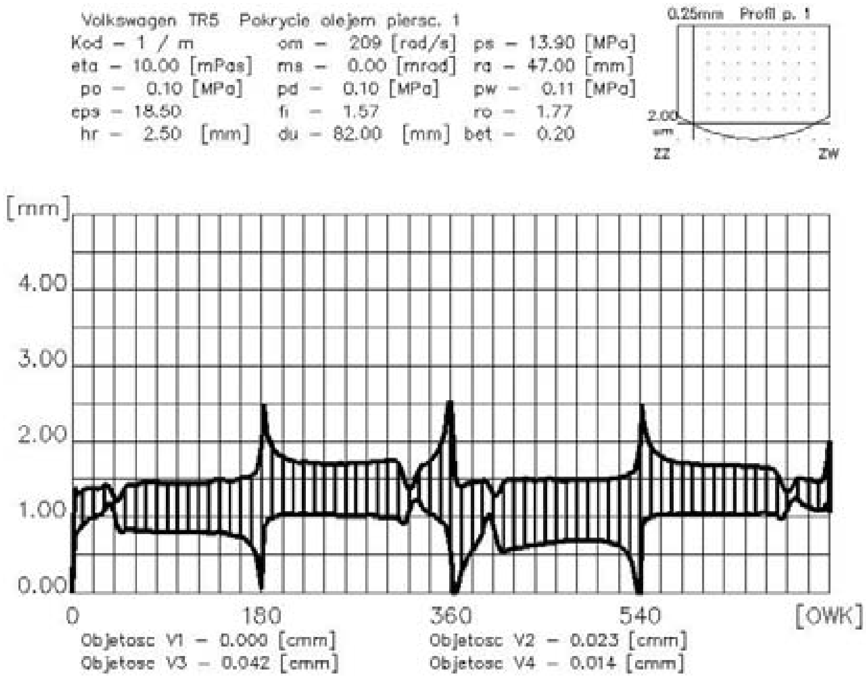

Fig. 9. Ring face wetting level at nominal temperature of operation

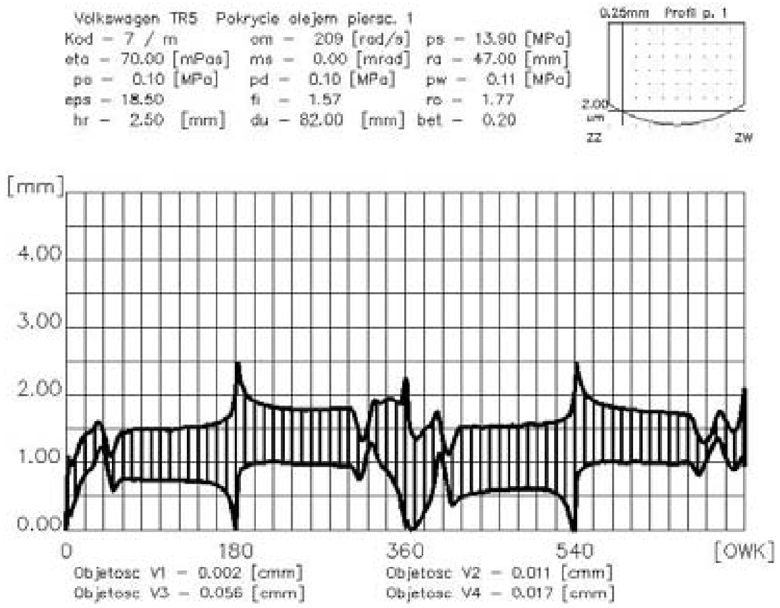

Fig. 10. Ring face wetting level at lower temperature of operation

Within the frames of research program the oil consumption has been analyzed for the viscosity range from 0.010 to 0.070 Pas, which corresponds to the temperature range from 110 to 40°C (10W40 grade oil). Graphs in Fig. 11 illustrate these results.

Fig. 11 additionally presents the friction losses generated by ring pack of a single cylinder of the tested AXD engine.

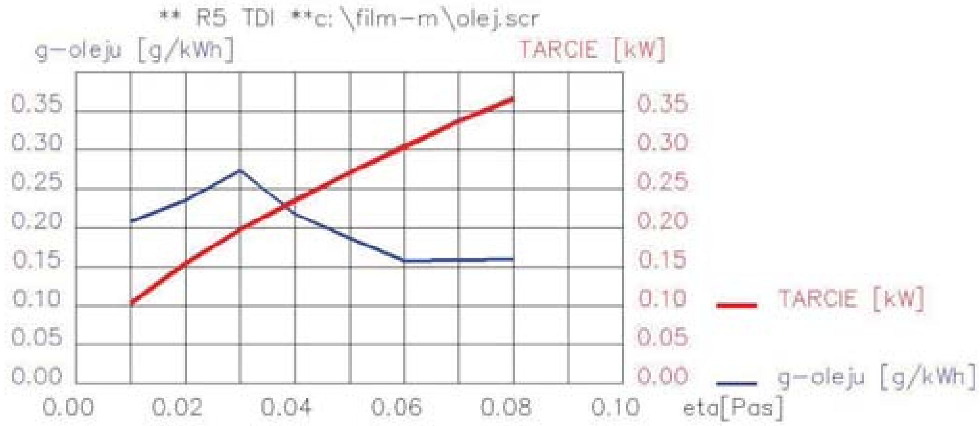

Fig. 11. Lubricating oil consumption and friction losses generated by ring pack of single cylinder for oil viscosity range 0.010 to 0.070 Pas

5. Conclusions

1. Fluctuations of lubricating oil viscosity relative to engine temperature do not affect the oil consumption to a considerable level.
2. The lowest oil consumption is reached within the range of low temperatures, i.e. 40 to 45 °C for 10W40 grade oil, but due to obvious reasons such low temperature of engine run is not advisable at least because of three times higher friction losses (see Fig. 11).

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