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The influence of cylinder disengaging on the friction resistance in the piston-cylinder assembly

Abstract: On the combustion engine specific fuel consumption characteristic the high efficiency area is limited to the high engine loads. But most of the time the necessary engine power used for driving is between 20 to 50 % of the nominal engine power, which means that the engine works beyond the high efficiency range. This situation can be changed with disengaging part of the cylinders by cutting off the air and fuel supply to these cylinders, which means that the other cylinders will work with higher load to compensate power loss of disengaged cylinders. Such idea is known for long time but its practical realization meats series of problems. The long lasting idle running of piston–rings–cylinder group may have impact on engine durability and friction losses generated by switched off cylinders. In the paper friction losses of the piston-cylinder assembly of four cylinder in-line engine are presented. Three cases are considered: maximum efficiency working range, idle run and 50 % of the engine rated power within maximum efficiency range. The simulations were made for the classic three piston rings packet and piston with carbon nanotubes layer on its skirt. On the basis of carried out simulations conclusions about purposefulness of cylinder disengaging were formulated.

Keywords: *piston-cylinder assembly, friction losses cylinder disengage*

Wpływ wyłączania cylindrów na opory tarcia w grupie tłokowo-cylindrowej

Streszczenie: Z charakterystyki ogólnej silnika spalinowego wynika, że pole pracy charakteryzujące się dużą sprawnością cieplną jest bardzo ograniczone. Najczęściej jednak moc wykorzystywana na jazdę stanowi 20 do 50% mocy nominalnej silnika a to oznacza, że silnik pracuje poza obszarem najwyższych sprawności. Sytuację można zmienić wyłączając z zasilania w paliwo część cylindrów silnika wielocylindrowego. O ile koncepcja taka znana jest od dziesięcioleci to jej praktyczna realizacja napotyka na szereg trudności. Długotrwała praca zespołu tłok–pierścienie–cylinder na biegu jałowym może wpłynąć na trwałość silnika a straty tarcia generowane w wyłączonych cylindrach obniżają sprawność ogólną tak użytkowanego silnika. W artykule przedstawiono straty tarcia w grupie tłokowo – cylindrowej silnika czterocylindrowego dla trzech przypadków: silnika pracującego w polu maksymalnej sprawności ogólnej, silnika na biegu jałowym oraz silnika generującego połowę mocy odpowiadającej pracy w polu maksymalnej sprawności. Symulacje przeprowadzono z uwzględnieniem pakietu trzech pierścieni na tłoku oraz powierzchni bocznej tłoka uszlachetnionej warstwą propoślizgową wykonaną z nanorurek węglowych. Na podstawie przeprowadzonych symulacji sformułowano wnioski określające celowość stosowania systemu wyłączania zasilania paliwem części cylindrów, przez co pracujące cylindry generują moc w obszarze maksymalnej sprawności ogólnej.

Słowa kluczowe: *grupa tłokowo-cylindrowa, straty tarcia, wyłączanie cylindrów*

1. Introduction

The idea of increasing engine total efficiency when working with partial loads by disengaging air and fuel supply of the part of total cylinder number has been known for many years. This method was so far applied only to the engines with the cylinder number more than eight, for example Mercedes-Benz V8 spark ignition engines. But it was recently also applied to the four cylinder in-line engine [1]. Side effects of cylinder disengaging in the four cylinder engine are still not recognized. There is a question how the crank mechanism will be affected by long term idle cylinders run without fuel supply. The piston with rings packet design should be optimized in aspect of friction losses during idle cylinder run so the positive effect of optimal working cylinders load will be not lessen. Therefore there is a necessity to determine friction losses for the piston and for each piston ring in maximum total efficiency working range of the engine, for idle engine run and for the engine working with 50 % of its power in maximum efficiency range. Following the other working parameters of the crank mechanism will be analysed to estimate the possible premature wear of the crank mechanism elements.

2. Operating parameters of the crank mechanism elements

The simulations results for the engine crank mechanism processes will be presented in form of graphs which contain complete engine working cycle. The sequence of mating elements relative motion will be as follow. After the oil flowing out of the crank bearings is sprayed at the cylinder liner surface as the first slides on the oil layer the piston

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skirt. Next, some part of the oil is being scraped out of the cylinder liner surface by piston oil ring and at the cylinder liner remains about 1 μm thick oil layer. On such layer moves the lower compression piston ring and finally on the remaining layer moves the upper compression piston ring. During the piston movement from bottom dead centre to top dead centre the sequence of moving elements is reversed.

In carried out researches earlier specified variants of engine working conditions were taken into account. First the most often engine working conditions in passenger car were taken. Statistically this is 3000 rpm and 50 % of maximum load at this engine speed which come from full power engine characteristic.

According to described moving elements sequence as the first the oil parameters for piston skirt were calculated and obtained results are presented on Fig. 1.

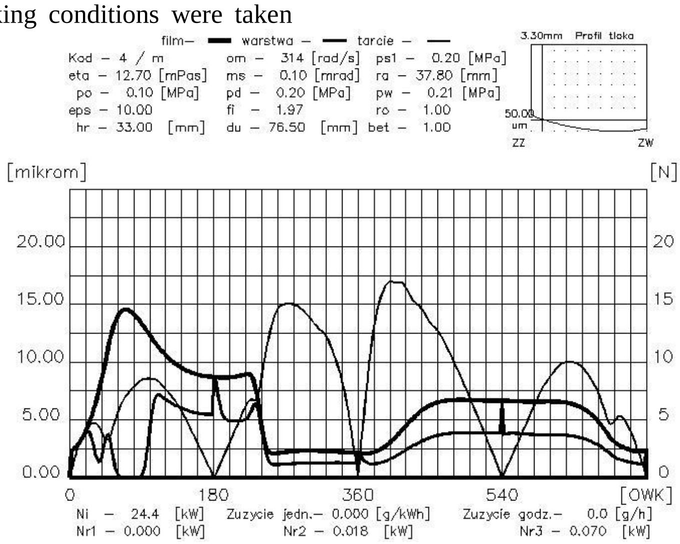

Fig. 1. Oil film thickness – the thickest line, oil layer left on the cylinder liner surface – medium thick line, friction force of the piston lateral surface on the cylinder liner surface – the thinnest line; results for the most often car engine working parameters

When the cylinder fuel supply is disengaged the oil film parameters in the piston-cylinder liner mating will be as shown on Fig. 2.

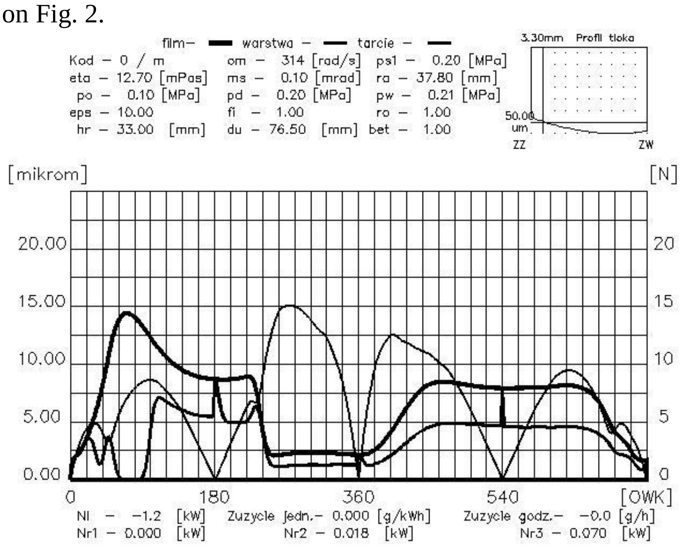

Fig. 2. Oil film thickness – the thickest line, oil layer left on the cylinder liner surface – medium thick line, friction force of the piston lateral surface on the cylinder liner surface – the thinnest line; results for the external engine drive

For full power operating conditions of the engine the analogous oil film parameters will be as shown on Fig. 3.

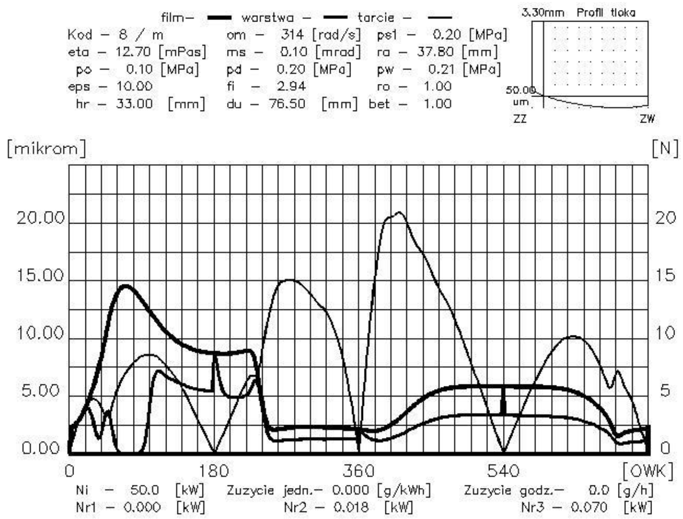

Fig. 3. Oil film thickness – the thickest line, oil layer left on the cylinder liner surface – medium thick line, friction force of the piston lateral surface on the cylinder liner surface – the thinnest line; results for the full load of the engine

Table 1. The most important designations explanation and adopted values from Fig. 1 to 3

	Fig. 1	Fig. 2	Fig. 3
Crankshaft angular velocity om = ω [rad/s]	314		
Lubrication oil viscosity eta = η [mPas]	12.7		
Crankthrow ra [mm]	37.8		
Compression ratio eta = ε [-]	10		
Pressure increase ratio fi = φ	1.97	1.00	2.94
Piston skirt height hr [mm]	33.0		
Cylinder bore du = D [mm]	76.5		

The piston skirt-cylinder liner friction power is the primary part of total inner friction resistance of the internal combustion engine. To calculate this power it is necessary to know the filling degree of the gap between piston and cylinder liner by the lubrication oil. On Fig. 4 the piston skirt coverage degree by the lubrication oil are presented for engine operating parameters analogous to Fig 4.

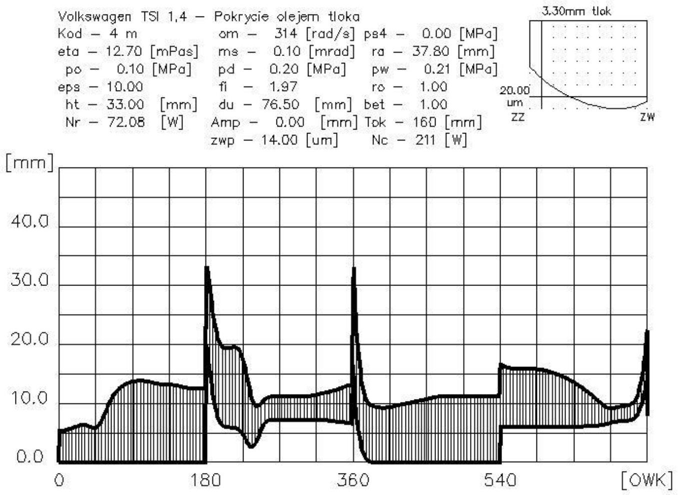

Fig. 4. Piston skirt coverage degree by the lubrication oil for engine operating parameters analogous to Fig. 1

In the right upper corner schematic cross section of the piston lateral surface which was considered during simulations is presented. Mean friction power of the piston is 72.08 W, and it is marked on the figure as Nr. The mean friction power for piston parameters as on Fig. 2 is 65.75 W, and as on Fig. 3 77.36 W. It was found that engine load has moderate influence on friction parameters of the piston – cylinder liner assembly. As a consequence the cylinder deactivation will cause moderate friction power reduction of the engine.

Second significant source of the friction losses in the crank mechanism is the upper compression ring. When the pressure in the cylinder is high the upper compression ring is pressed against the cylinder bearing surface much more intensive that it results from elastic thrust of average 0.2 MPa. On Fig. 5 to 7 the main operating parameters of the upper compression ring are presented.

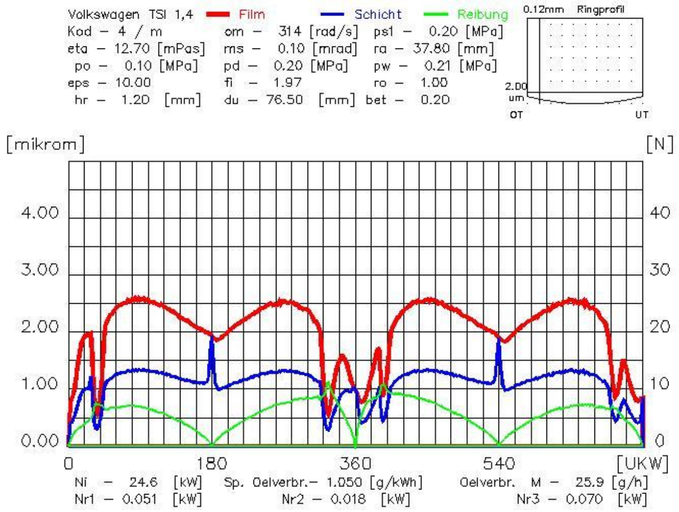

Fig. 5. Oil film thickness – red line, thickness, thickness of the oil layer that is left at the cylinder bearing surface – blue line, friction force of the upper compression ring – green line; results correspond to the most common operating conditions of the engine in the passenger car

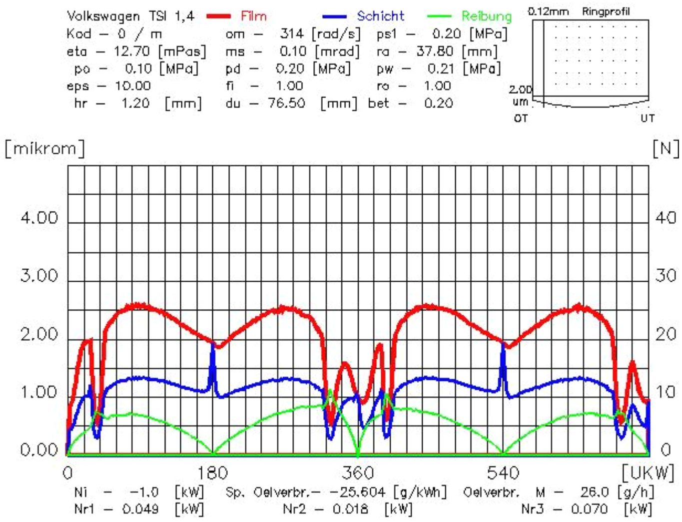

Fig. 6. Oil film thickness – red line, thickness, thickness of the oil layer that is left at the cylinder bearing surface – blue line, friction force of the upper compression ring – green line; results correspond to the external drive of the combustion engine

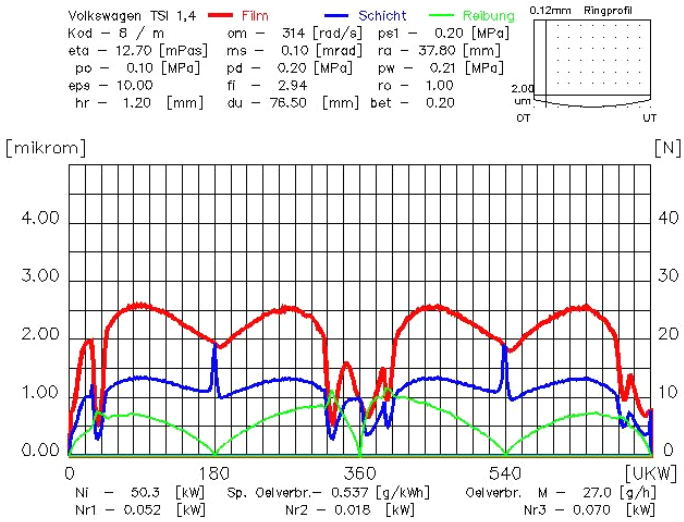

Fig. 7. Oil film thickness – red line, thickness, thickness of the oil layer that is left at the cylinder bearing surface – blue line, friction force of the upper compression ring – green line; results correspond to the full power characteristic of the engine

On Fig. 5 to 7 under the graphs the indicated power and oil specific consumption [g/kWh] values are marked. In the lowest line the following friction losses are marked: Nr1 – friction losses of the upper compression ring, Nr2 – friction losses of the lower compression ring, Nr3 – friction losses of the oil scrapping ring. It is worth noting that friction losses generated by lower compression ring and oil scrapping ring are constant for different engine loads. It results from low gaseous force value that pressures these rings to the cylinder liner. For the upper compression ring friction losses for the three simulated engine loads are 51 W for the most common operating conditions of the passenger car engine, 49 W for cylinder with no fuel supply and 52 W for full power condition. The friction losses changes with engine load changes are not so much significant because maximum difference do not exceed 6 %.

The increased load of the engine cylinders may however cause another adversely effects, that is why the minimal oil film thickness for all piston – cylinder group elements is taken into consideration, as well as the hour oil consumption is. For all of examined operation conditions the oil film thickness in the upper compression ring is never less than 0,5 μm. Such thickness of the oil film guarantees no direct ridges contact of the mating elements so the wear process will not occur.

On the basis of oil consumption model [2], in which the oil is extruded by the upper compression ring into the combustion chamber, the oil hour consumption for three considered cases was estimated.

In the case of the engine where all cylinders are loaded at a uniform rate, the oil consumption is 25.9 g/h – Fig. 5, for the no load running engine 26.0 g/h and for full load running engine 27.0 g/h.

3. Conclusions

The cylinder shut off technique is an effective method of engine fuel consumption reduction in regular driving conditions when the power demand is low or medium. It is achieved by operating the working cylinders closer to the area of maximum efficiency at the specific fuel consumption characteristic. But in the same time it does not give significant advantages in aspects of friction losses reduction, so the mechanical efficiency of the combustion engine remains at the same level. It can be stated that:

1. Cylinder shut off makes noticeable improvement of the engine indicated efficiency [1] but the mechanical efficiency does not significantly change in comparison to the regular engine with all cylinders in operation,
2. If the sliding surfaces of the piston and piston rings are correctly designed the cylinder shut off technique does not result in durability and reliability differences between operating and non-operating cylinders,
3. The oil consumption remains at almost the same level, and for considered case it varies from 25.9 to 27 g/h.

Nomenclature/*Skróty i oznaczenia*

du cylinder bore D

eps compression ratio ε

eta	lubrication oil viscosity μ	Nr3	oil ring friction power
fi	pressure increase ratio φ	om	crankshaft angular velocity ω
hr	piston skirt and piston ring height	M	oil hour consumption
Ni	indicated power	po	ambient pressure
Nr	piston skirt friction power	ra	crankthrow
Nr1	upper compression ring friction power	ro	volume increase ratio ρ
Nr2	lower compression ring friction power		

Bibliography/*Literatura*

[1] Middendorf H. i inni. Der 1,4-l-TSI-Ottomotor mit Zylinderabschaltung. MTZ 03/2012.

[2] Iskra A. Studium konstrukcji i funkcjonalności pierścieni w grupie tłokowo-cylindrowej. Wydawnictwo Politechniki Poznańskiej. Poznań 1996. Str. 1-334.

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