




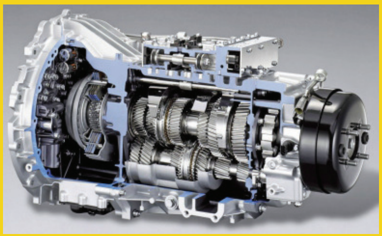
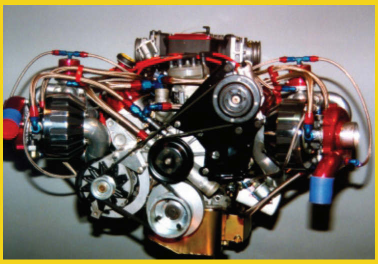

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ENERGY AND EXERGY ANALYSIS OF AN AUTOMOTIVE DIRECT INJECTION DIESEL ENGINE

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Abstract:
The article presents a numerical analysis of energy balance of an automotive diesel engine and exergy analysis of exhaust gas and cooling systems. A model of the engine was built in advanced simulation code AVL Boost. In order to validate the model a comparison between estimated and real engine effective power was conducted at full load. Energy balance revealed a maximum engine efficiency of 42.1% at full load and 2000rpm. The highest quantity of lost energy contains the exhaust gas. The maximum estimated exhaust gas enthalpy is 108kW at 4000rpm. At the same operating point the cooling enthalpy is twice lower – 55kW. At the engine speed lower than 2000rpm the lost energy in exhaust gas and cooling system has the same quantity. The exergy analysis revealed that waste heat recovery potential in exhaust gas is much higher than cooling system. The results obtained in this study will be further used in a Rankine-Hirn waste heat recovery system development due to increase overall engine efficiency.

Keywords: *energy, exergy, internal combustion engine, numerical simulation, waste heat recovery.*

1. Introduction

To meet the future restriction of CO_2 emissions from passenger vehicles it will be necessary to improve overall engine efficiency by development of next level technologies. For European Union the target level of CO_2 must be reduced by 41% at 2020 in comparison with 2006. The new target level measured by NEDC will be 95g/km [1].

Although the number of modern techniques such as: turbocharging, high pressure direct injection, variable valve timing and etc., the overall engine efficiency is lower than 40%. Only in special engine such as high boosted diesel engine for stationary application and for the ships the efficiency could meet 50%.

It means that at most commonly used operating points more than 60% of the fuel energy is lost. Some part of this energy is rejected by exhaust gas, other part is lost in the cooling system and the rest is lost to drive the auxiliaries [2,3]. The recovery of this lost energy seems to be a good prospective for further improvement of engine overall efficiency.

Research of other authors revealed that the energy contains in the exhaust gas and cooling system are almost the same [4,5]. Usually the energy in cooling system is higher at low engine speed while the energy of exhaust gas is higher at high engine speed and load. To evaluate the recovery potential of the lost energy it is not enough to estimate only the quantity of this energy [4,6]. According the thermodynamics laws not all of that energy can be transformed in mechanical power. For that reason it is necessary to estimate the exergy contained in the exhaust gas and the cooling system. The exergy reveals the quantity of energy which can theoretically be transformed in mechanical work by closed loop thermodynamical cycle. With other words it presents the available energy in the system. A number of research on exergy analysis of lost energy in internal combustion engines have been reported [4, 6-9].

Although almost the same quantity of energy in exhaust gas and cooling system the exergy is much higher in exhaust gas [4]. Punov *et all*. [10] revealed in a study on a tractor engine that the exergy is the highest at high load and high speed and slightly decreases with load and speed.

The aim of the article is to study the energy balance and exergy analysis of a modern direct injection diesel engine intended for passenger car.

2. Simulation model

A simulation model of the engine was built in advanced simulation software AVL Boost. This model provides opportunities to simulate the engine performance, fuel consumption as well as the energy rejected by exhaust gases, the energy in cooling system and mechanical losses.

2.1 Engine parameters

The engine under study is 2.0liter four cylinders direct injection diesel engine developed by PSA. The maximum output power is 101kW at 4000rpm as the maximum torque is 320Nm at 2000rpm. The engine is equipped with variable geometry turbocharger. The boost pressure is limited to 1.4 bar. Common rail system of the engine is produced by Delphi. The maximum injection pressure is 1600 bar. The engine is also equipped with EGR system and post treatment system including catalytic converter and DPF. The cylinder is equipped with four valves per cylinder. The main geometrical parameters of the engine are listed in Table 1.

Table 1

Type of engine	HDI
Number of cylinders	4
Total volume	2L
Cylinder bore	85 mm
Cylinder stroke	88 mm
Compression ratio	17,6
Valves per cylinder	4

2.2 Engine model

In order to build a realistic engine model it was necessary to input geometrical parameters of intake and exhaust system as well as valves diameters and valves lift curves. The parameters (diameters and length of the pipes) of intake and exhaust system were measured on a real engine mounted on a test bed. The valves lift curves were calculated as the cams lifts had been measured by means of special equipment and the kinematic scheme of valve train mechanism was used. The valves seats diameters were taken from the technical documentation.

The engine model was built in advanced simulation code AVL Boost by means of elements available into (Fig. 1). The main elements which were used are: cylinder, plenum, pipe, intercooler, turbocomressor and general engine element (E1).

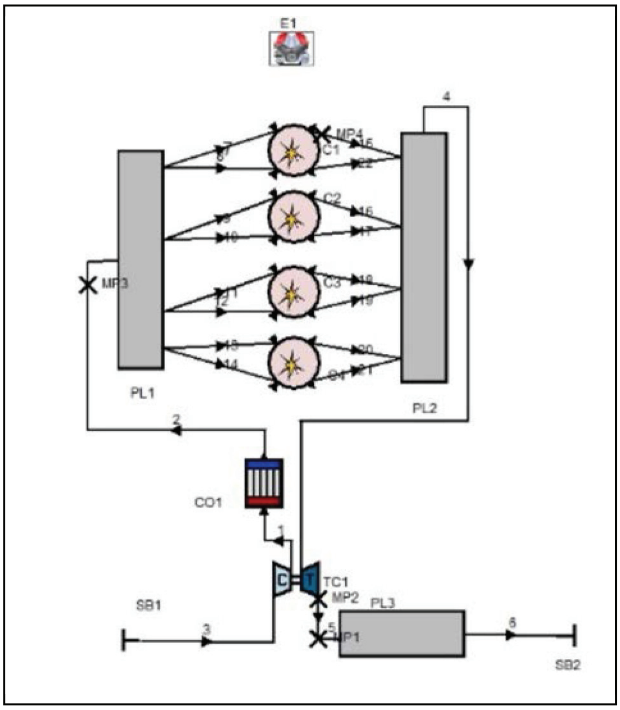

Figure.1. Engine model built in AVL Boost simulation software

Into the element E1 was input main control parameters such as: engine speed, number of cylinders, type of engine, simulating cycles and etc. The parameters for calculation the mechanical losses into the engine are also defined in this element.

The element “Cylinder” was used to define the geometrical parameters of the cylinder, injection and combustion model as well as heat transfer to cylinder wall. Mixing-controlled combustion model was used in the simulation. For this model it was necessary to define the main injection parameters such as: nozzle geometrical parameters, injection pressure, injection timing and duration. Air-excess ratio was used as a control parameter of the injection process. The most commonly used model of Woschni was used for heat transfer estimation.

The intake system includes air filter, compressor, intercooler, plenum and pipes. The elements by means of which exhaust system was developed are: plenum, turbine, silencer and pipes. The compressor and turbine turns with same speed due to the common shaft between them.

3. Model calibration

It was not possible to measure the engine performance on our test bed because the engine is still not ready to be run. Due to that the model calibration

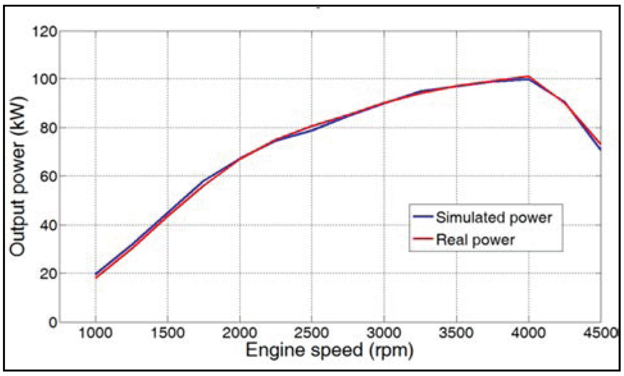

Figure 2. Engine model calibration by output power comparison at full load

was conducted by comparison between estimated engine output power and the engine power obtained by the manufacturer at full load. The comparison is presented in Fig. 2.

Calculated output power is higher than that presented by the manufacturer within the speed range from 1000rpm to 2000rpm. A maximum deviation of 2.5kW was observed at 4500rpm. In relative values it presents 3.4% from calculated value. The deviation can be explained as that we had not information about factory settings of injection parameters. The small deviation obtained in the model calibration allowed us to use the simulation model in our numerical study.

4. Energy balance of engine.

Due to the engines for passenger car run at variable operating point it was necessary to study the energy balance at whole operating range. This approach can reveal to us information which of lost energy is higher depending on engine operating conditions. The energy distribution is presented in Fig. 3 to 6.

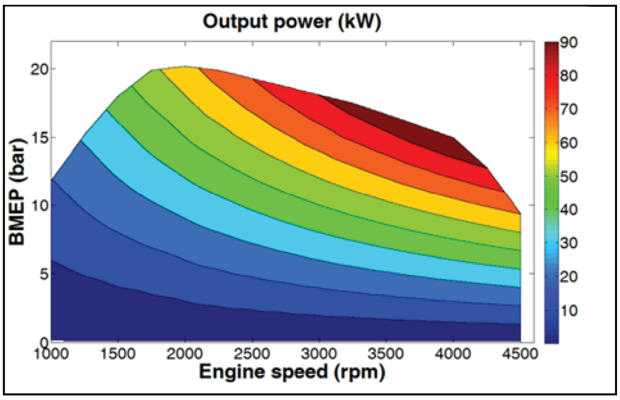

Figure 3. Engine output power

The variation of output power and exhaust gas enthalpy have the same tendency at the whole operating range – Fig. 3 and 4. The maximum exhaust enthalpy was observed at the same operating point as the maximum power was obtained. In absolute value exhaust gas enthalpy is higher than output power at engine speed above 3500rpm and at whole speed range when BMEP is closed to zero. The maximum enthalpy is 108kW which is 8% higher than maximum output

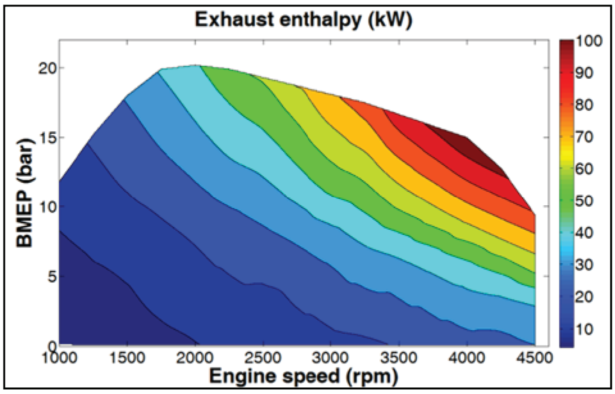

Figure 4. Exhaust gas enthalpy

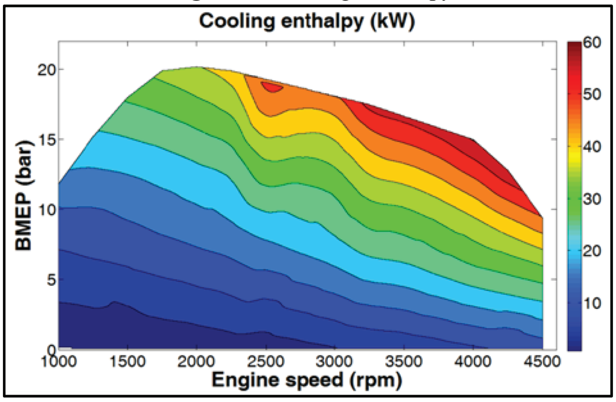

Figure 5. Cooling enthalpy

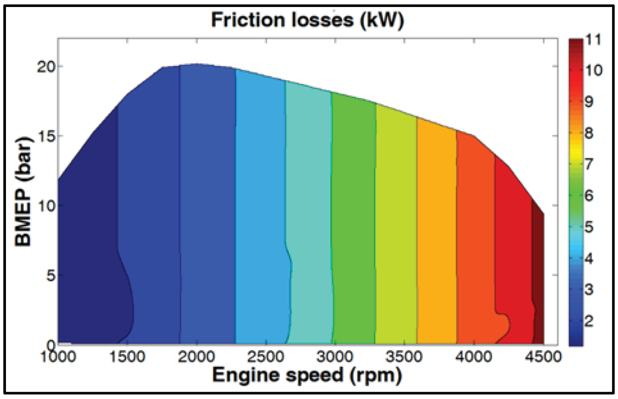

Figure 6. Friction losses

power.

The cooling enthalpy (Fig. 5) is significant smaller than exhaust enthalpy. However, at small operating range when engine speed is lower than 1750rpm and BMEP is higher than 5bar that cooling enthalpy is similar or higher than exhaust enthalpy. The highest value of 60.4kW was observed at that operating point corresponds to the maximum output power.

The friction losses present a small part of lost energy (Fig. 6). They slightly increase in function of engine speed but they are not influenced from the load. The maximum value is 11.3kW at 4500rpm.

The more interested analysis was made as the energy balance was presented in relative values from lower heat value (LHV) of diesel fuel. These results are shown in Figure 7 to 9. The maximum engine efficiency of 42.1% was observed at operating point

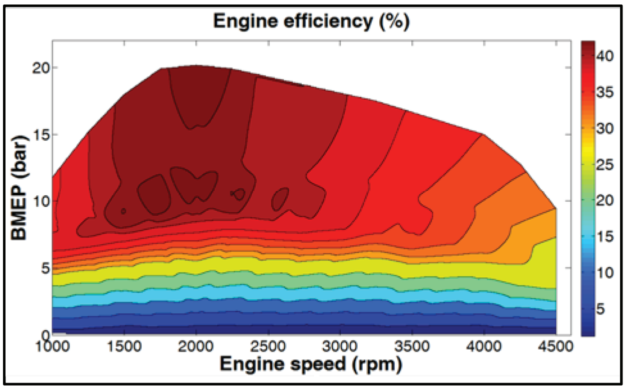

Figure 7. Engine efficiency

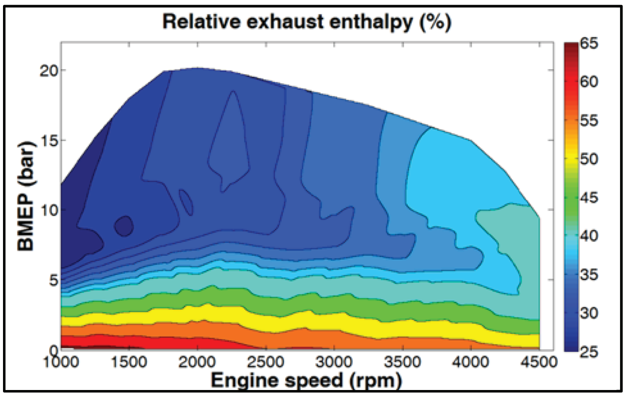

Figure 8. Exhaust enthalpy estimated in percentage from LHV

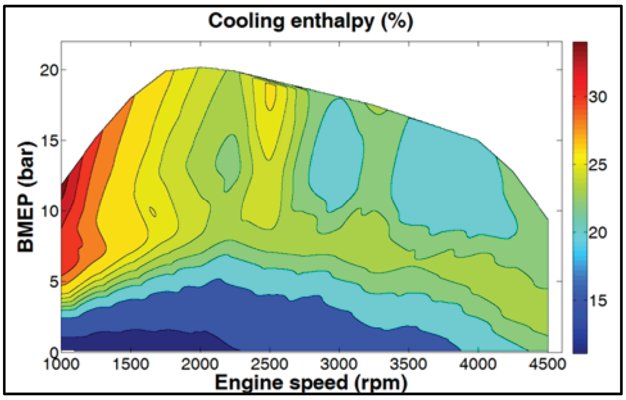

Figure 9 . Cooling enthalpy estimated in percentage from LHV

corresponds to maximum torque. This engine operates with overall efficiency higher than 35% within the operating range defined of BMEP above 6bar and speed bellow 4000rpm. At that operating point corresponding to maximum power the efficiency of 35.9% was estimated.

The lost energy in exhaust gas is within the range from 25.3% to 66% of LHV (Fig. 8). The highest values were observed at engine load close to idle when the quantity of exhaust enthalpy is low and engine power is near zero. At BMEP more than 3 bar exhaust enthalpy is within the range from 25.3% to 40% of LHV. The lower values are observed at the operating range where engine efficiency is the highest.

The lost energy in cooling system is within the range from 11.3% to 34.9% of LHV (Fig. 9). The highest values were observed at minimum engine speed and high load. It can be explained with increasing of the time for heat transfer to cylinder wall. At this operating range the quantity of cooling enthalpy is similar to the quantity of exhaust enthalpy. At engine speed over 2500rpm cooling enthalpy is within the range from 18.5% to 23.7% which is lower than exhaust enthalpy.

5. Exergetic analysis

As it was mention above, engine energetic balance can revealed engine effectiveness and quantity of lost energy in exhaust gas and cooling system but it cannot revealed recovery potential of each of lost energy. For this reason an exergetic analysis of exhaust gas and cooling system was conducted. The exergy was estimated at whole engine operating range as follows:

$$Ex = (H - H_0) - T_0(S - S_0),$$

where H and S are enthalpy and entropy of heat source as H_0, S_0 and T_0 are the reference parameters. In the calculation T_0 was considered to 293K. The exergy contains in exhaust gas and cooling system is presented in Fig. 10 and 11.

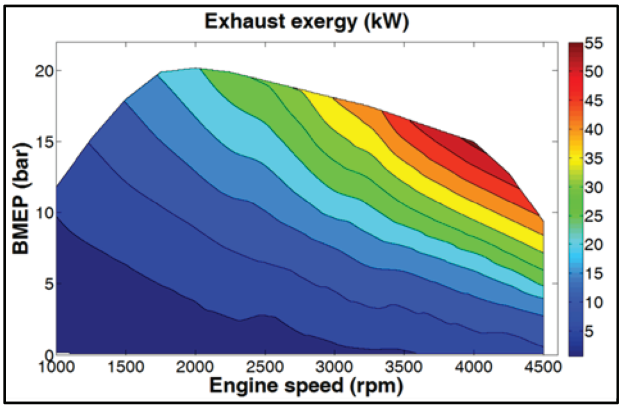

Figure 10. Exergy contains in exhaust gas

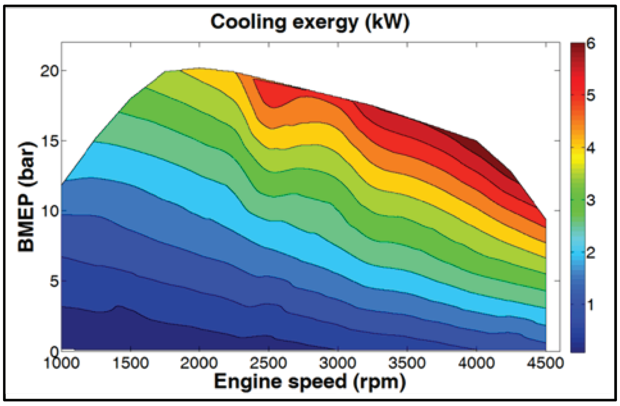

Figure 11. Exergy contains in cooling system

The exergy diagram for both exhaust gas and cooling system copies the enthalpy diagrams at whole operating range. The exhaust gas exergy is within the range of 0.6kW to 56.2kW which means that no more than a half of exhaust enthalpy can be transformed by thermodynamics cycle in mechanical work. The cooling system exergy is within the range from 0.2kW to 6.4kW which means that more than 85% of cooling enthalpy would be destructed in case of energy recovery. This result revealed that recovery potential of exhaust gas is much higher than that of cooling system. The main reason for that is the much higher temperature of exhaust gas than in the cooling system of the engine. The energy recovery from cooling system by means of Rankine-Hirn cycle is possible only in case of organic working fluid with low condensing temperature is used.

An estimation of exhaust gas and cooling system exergy was conducted at percentage from LHV of the diesel fuel. The results are presented in Fig. 12 and 13. These results revealed theoretical possibility for improvement of engine efficiency. On the basses of exergetic analysis of exhaust gas engine efficiency can be improved in within the range from 10.4% to 20.2%. The maximum effective could be exhaust heat recovery at operating point close to engine maximum power. It is necessary to mention that this exergetic analysis was made as exhaust gas parameters was taken at exhaust valves ports. In the practice it is not possible to develop a recovery system which can use the whole available exhaust gas enthalpy. There are two reasons for that: a part of exhaust enthalpy is used in the turbine of the turbocompressor and secondly the heat transfer to the atmosphere. In a modern engine with post treatment systems it is not preferable to install a heat exchanger of Rankine-Hirn cycle at exhaust gas path before the catalytic convertors and DPF due to extremely decreasing of gas temperature.

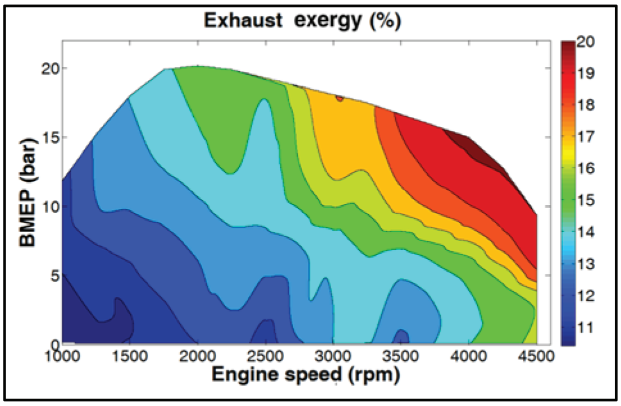

Figure 12. Exhaust gas exergy in percentage from LHV

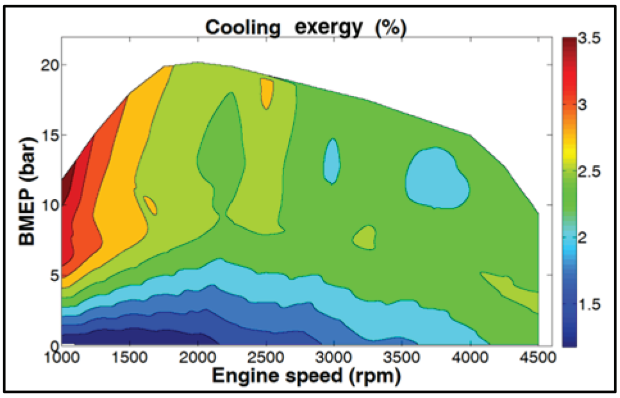

Figure 12. Cooling system exergy in percentage from LHV

Exergetic analysis of the cooling system revealed that heat recovery system can increase engine efficiency by maximum 3.4% at minimal speed and high load. It makes waste recovery from cooling system much lower effective than exhaust recovery. In some application it is also possible a recovery system with two or more heat sources (cooling system, EGR, exhaust system). It can optimize overall recovery efficiency at whole operating range.

6. Conclusions

A comprehensive energetic and exergetic analysis of a modern direct injection diesel engine was made by means of high accurate simulation model developed in advanced simulation software AVL Boost. The maximum deviation between simulated and real output engine power was 3.4%.

The energetic balance of the engine revealed a maximum engine efficiency of 42.1% at operating point corresponds to maximum torque. The engine operates with efficiency higher than 35% in wide operating range including the maximum power operating points.

The highest part of lost energy was rejected by exhaust gas within the range 25.3% to 66% of LHV. The lost energy in cooling system is within the range from 11.3% to 34.9% of LHV as this energy is higher than exhaust energy only at low speed and high load.

Exergetic analysis revealed that much higher exergy is contains in exhaust gas. This exergy is within the range from 10.4% to 20.2% of LHV. The cooling exergy is within the range from 1.2% to 3.4% of LHV.

On the bases of these results our future studies will be concentrated for development of a waste heat recovery system by means of a Rankine-Hirn cycle which uses exhaust gas energy as a heat source. Due to that the engine is equipped with water cooled EGR system a combination of two heat source for the Rankine-Hirn cycle is also possible. It can optimized waste heat recovery efficiency at wide operating range.

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ТОПЛИНЕН БАЛАНС И АНАЛИЗ НА ВЪЗМОЖНОСТТА ЗА РЕКУПЕРИРАНЕ НА ЕНЕРГИЯ В ДИЗЕЛОВ ДВИГАТЕЛ С ДИРЕКТНО ВПРЪСКВАНЕ

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Резюме:

В публикацията е представен топлинен баланс и анализ на възможността за рекупериране на енергия от изпускателната и охладителната система на дизелов двигател, предназначен за лек автомобил. Изследванията са проведени с помощта на модел на двигателя в програмата AVL Boost. За оценка достоверността на модела е направено сравнение между изчислената и реалната ефективна мощност на двигателя в режим на пълно натоварване. Топлинният баланс показа, че максималният ефективен к.п.д. на двигателя е 42,5% при честота на въртене n=2000rpm и пълно натоварване на двигателя. Най-голяма част от загубената топлина се отвежда с отработилите газове. Максималната стойност на енталпията на отработилите газове е 108kW при 4000rpm. В тази работна точка енталпията в охладителната система е около два пъти по-малка - 55kW. При честота на въртене по-малка от 2000rpm топлината отведена с отработилите газове и тази отведена с охладителна система са приблизително по равно. Анализа на възможността за рекупериране на изгубената топлина, показва че потенциала за рекупериране от отработилите газове е значително по-голям. Получените резултати ще бъдат използвани за разработването на система за рекупериране на енергия от двигателя с помощта на цикъл на Ранкин.

Keywords: *енергия, ДВГ, топлинен баланс числено изследване, рекупериране на енергия.*