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POSSIBILITIES OF WASTE HEAT RECOVERY ON TRACTOR ENGINES

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Abstract:

Tractor engines present significant possibilities of waste heat recovery due to their high power and their operation cycles. The waste heat recovery system based on Rankine cycle seems to be the most effective way of reducing the fuel consumption, respectively the emission of CO_2. Experimental studies of the real working cycles of a tractor engine reveal that such type of engines operate at high load close to the maximum most of the time when the tractors run on the fields. The aim of the article is to evaluate the energy and exergy available at different location points in the exhaust system of the tractor engine. A combination of 1D and 0D approaches is used to build the engine model in the software AVL BOOST. The experimental results such as: fuel consumption, effective power, mass flow, wasted energy in the cooling system, temperature in the exhaust system etc. have been used for calibration of the model. The energy balance of the engine shows that more than 35% of the fuel energy is lost by exhaust gases on the most typical operating points. Finally, the energetic and exergetic analysis at a certain point of the exhaust system is presented. The results show good prospects for further research on the Rankine system mounted on the tractor engine studied here.

Keywords: *waste heat recovery, tractor, engine, energy, exergy, engine modeling.*

1. Introduction

Reducing the fuel consumption and emission of CO_2 will be the major challenge for all types of internal combustion engines in the years to come in the future. For road vehicles the target level of CO_2 in 2020 will be lower by 41% in comparison with that for the year 2006 for the European Union [1]. The reduction of CO_2 emissions depend on the overall engine efficiency. It is well known that the modern engines have great thermodynamic and mechanical efficiency which make future improvement very difficult. All of the current techniques such as high pressure direct injection, VVT, downsizing, variable geometry turbine etc. will not be enough to meet the future restrictions. More complicated systems for improving the engine efficiency should be developed over the next few years.

Despite of the high level of development of the engine systems and control the maximum efficiency of 40% is reached in certain operation points. Most of the time engines run with efficiency of 15% to 35%. It means that more than 60% of the fuel energy is lost [2]. The lost energy is in form of heat in the cooling and the exhaust system. The literature review [3] shows that highest potential of recuperation is in the heat of the exhaust gases. In some engines the exhaust gases heat is used for heating the cooling system during the warm up period. It accelerates the warming up of the engine and reduces the fuel consumption. It does not have significant effect on fuel consumption because of the warm up period is relatively short. For further recuperation of waste energy more complicated systems are known:

- ✓ Turbocompound;
- ✓ Recovery based on termodynamics cycles (Rankine cycle, Stirling cycle, et.);
- ✓ Termoelectric generators;

Turbocompound is currently used in truck engines. The problem is the back pressure which affects the gas exchange processes of the engine. It seems the electrical turbocompound is more a perspective way for recuperation but still there is the problem in developing a high speed electrical machine coupled to the shaft of the turbocompressor. Both the thermoelectric generators and the Rankine cycle do not produce back pressure. While the thermoelectric generators need future development the recuperation based on Rankine cycle seems to be the most perspective way.

The Rankine cycle is well known in thermal power stations. The principle of operation is presented in Fig. 1. The heat is used to evaporate the working fluid within the heat exchanger 2 (boiler). Before entering the boiler the cold fluid is pressurized to the working pressure by the pump. The vapor goes out from the boiler and expands within the expander 3 (turbine). Then the fluid is condensed to liquid within the condenser 4. The efficiency depends on the differences in the temperatures of the hot and cold side of the cycle as well as the efficiency of the elements.

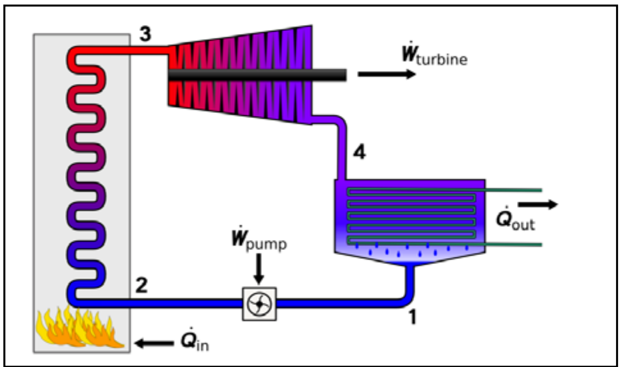

Fig.1. Principle of operation of Rankine cycle

Two sources of waste energy can be used for heating the working fluid: the heat of the exhaust gases and the heat of the cooling system. In engines with water cooled exhaust gases recirculation system the heat from the gases passing through the system could be used too.

Many working fluids such as: water, alcohols, n-pentane, toluene, refrigerator agent HFC-245fa, etc. can be used. The choice of the working fluid depends on the temperature of the heat source. The water is suitable for high temperature heat sources while organic fluids are used in case of lower heat source temperature. A number of additional requirements to the fluids must be taken into consideration in order to make proper choice [3, 4].

The evaporator is a heat exchanger which is mounted within the exhaust system of the engines. Different type of exchangers can be used depending on the way of the fluids flow in them. The most effective ones are exchangers with counter flow in which the exhaust gases and the working fluid go in opposite directions. The heat exchanger must be strong enough because of the high pressure of the fluid (up to 30bars) and the higher temperature of the exhaust gases (up to 900K) [5]. The mounting position of the exchanger is quite important. To increase the efficiency of the cycle it is essential to mount it as close as possible to the exhaust valves ports. In the other hand the exchanger decreases the exhaust gases temperature which could have a negative effect to the catalytic converters and DPF efficiency. Due to the emission restrictions in modern engines the exchanger must be placed down to the aftertreatment system.

The expander converts the heat of the steam into the mechanical power. In most application the mechanical power then is converted to electrical by generator. Two types of machines are used: turbines and piston machines [6]. Turbines are most suitable for high power engines due to their small size. Piston machines are preferred in small engines application. Their low shaft speed provides opportunities to use the electrical machines directly mounted the shaft. The piston machines have some disadvantages as higher friction losses and inability to work for long time with a fluid in which there is liquid. The above requirements mean that at the end of expansion no liquid phase must appear in the fluid.

Within the condenser the working fluid changes its state from steam to liquid. For this process intensive cooling is necessary. Often in automotive engine application the conventional condenser from air-conditioning automotive systems are used in Rankine prototypes [7].

2.Operation cycle of a tractor engine

A tractor engine has been chosen for further development of waste heat recuperation system based on Rankine cycle. Research of a Massey Ferguson tractor (Fig. 2) shows that the tractor engine is suitable for waste heat recuperation due to: long time operation at high load, high fuel consumption and enough space for mounting the components of Rankine system.

Fig.2 Tractor Massey Ferguson, tested by Irstea

Tractors have many applications but their operation could be divided into two regimes:

transportation and work in the fields. The fuel consumption and emissions of CO_2 are higher when the tractor runs in the fields because a higher engine power is needed and the time of operation is longer. In the fields the tractor runs with constant speed from one to other side of the field then it takes some time to turn at the end of the field. This cycle is repeat many times. Typical operation cycle is shown in Fig. 3.

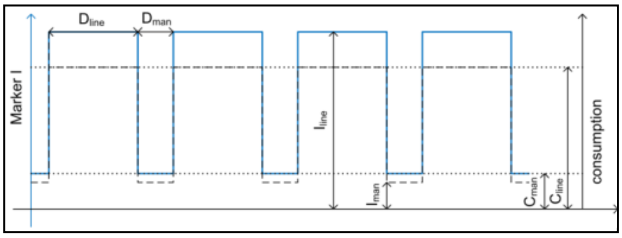

Fig.3. Typical operation cycle of a tractor working in field.

The constant speed of the tractor means that the engine runs at constant speed and almost constant load. The fuel consumption, engine power, engine efficiency and the enthalpy of the exhaust gases are constant. According to the experimental results the constant quantity of waste energy is produced during 80% of the time when the tractor works in the fields. It makes prediction the energy of the exhaust gases and development of high efficient Rankine system quite easy.

The values of the engine speed and load depend on the type of operation. In Fig. 4 the experimental results of engine speed distribution as a function of time is presented.

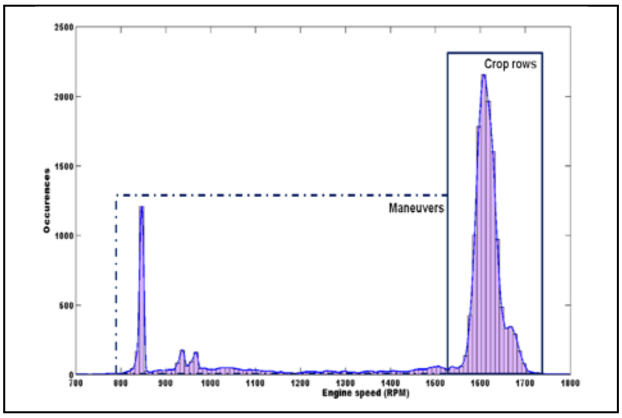

Fig.4. Engine speed distribution (during plowing).

3. The engine model description

A precise model of the studied engine was necessary for an energetic and exergetic analysis. The advanced engine simulation code AVL Boost has been chosen because of its complexity in simulating the working cycle by means of both 1D and 0D approaches. The model provides opportunities for analing the distribution of energy as well as the available energy into certain points of the exhaust system. The results of the model are based on the calculation of the parameters into the cylinders taking into consideration the specifics of the combustion process (the rate of heat release or the rate of injection), heat exchange with the cylinders walls, blow by and the gas exchange processes. The gas exchange in both intake and exhaust ports is modeled by means of quasi stationary flow calculation. The mass flow rate through the ports depends on the pressure differences in the port and the cylinder and the effective cross section of the valves.

The 1D model in the pipes of the intake and exhaust system provides calculation of the pressure in the valves ports taking into consideration the unsteady flow. This approach is accuracy enough and reduces the calculation time under a minute for a cycle simulation.

The model of the friction losses exist in the software by means of this model the friction power is calculated. The losses depend on construction of the engine, oil characteristics and available accessories which consume the engine power. There is multiplier to control the friction losses by the user.

The energy balance can be made through a comparison between: the energy put in with the fuel, the energy converted in the effective power, the energy lost through friction, the energy lost in the cooling system and the energy lost with exhaust gases. The energy of the exhaust gases is calculated by the enthalpy which goes out from the cylinders trough the exhaust valves.

3.1. The parameters of the engine under study

The engine is produced by Perkins. It is 6,6l six cylinders diesel engine with a turbocompressor and an intercooler. The boost pressure is limited to 1,6bar. The engine is equipped with direct injection common rail system. A Valve train mechanism with four valves per cylinder driven by a single camshaft and push roods is used in it. There is no aftertreatment system which makes the exhaust system of the engine suitable for the application of heat exchanger of Rankine system. The main technical parameters of the engine are shown in the table below.

Table 1

Type of the engine	Perkins 1106D
Number of cylinders	6 inline
Volume	6.6 L
Bore	105 mm
Stroke	127 mm
Compression ratio	16.2
Number of valves per cylinder	4

The geometries of the intake and the exhaust system were measured on the tractor engine while the geometry of the cylinders and valves was taken from the technical documentation. There were some missing parameters such as the characteristic of the turbocompressor and the valves lifts curves. In this case the simpler model of the turbocompressor was input into the engine model due to difficulties in receiving the maps of the turbocompressor by the manufacturer. The compressor is represented by constant efficiency and pressure ratio while the turbine is represented in more complicated way. Valves lift curves were represented by the curves of similar engine which are available within AVL Boost tutorials.

3.2. AVL Boost model

The model of the engine has been made by means of available elements within AVL Boost. The main elements which were used are: cylinder, junction, plenum, cooler, turbocompressor, air cleaner and boundary. The connections between the elements are made through pipes. Each element is defined by

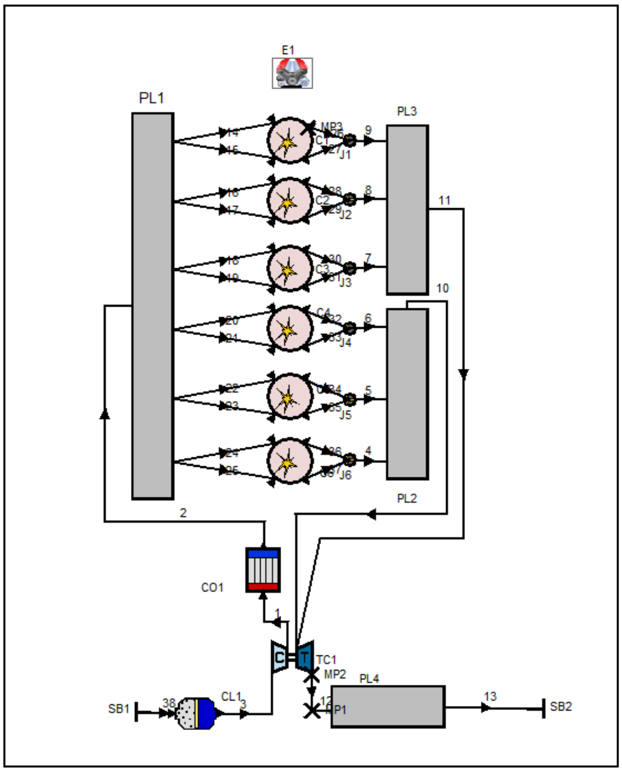

Fig.5. AVL Boost model of the engine

different parameters which are necessary to calculate the physical parameters in it. To define the main parameters of the simulation the element type “engine” E1 is used. This element is used also to define the friction loses. The model of the whole engine is represented in Fig. 5.

3.3 Modeling the intake system

The intake system of the engine includes air filter, compressor, air-air intercooler and intake manifold. All of the elements are connected to each other by pipes. A photo of the air filter and the intercooler can be seen in Fig.6.

The ambient conditions at the inlet of the intake system are defined in boundary element (SB1). The air filter (CL1) is represented by the volume of the

Fig.6 Air filter and intercooler mounted in front of the tractor

core, the volume of both inlet and outlet collector and the length of the core. The pressure drop within the elements is an input parameter for each of them. Due to the missing compressor map a simplified model is used. The compressor of the turbocompressor (TC1) is represented by constant efficiency and target pressure ratio. The target pressure ratio is changed in function of the engine torque. The intercooler (CO1) is defined by the volume and the length of the cooling core. Heat transfer is important for the intercooler. The model based on target output temperature (which is known from the experimental results) and the temperature of the ambient fluid is used in the study. The intake manifold is modeled by the plenum (PL1) and the short pipes between it and the intake valve ports. The main parameter of the plenum is the volume. The connection between the elements of the intake system is made through pipes. The size (length and diameter) of each is measured and input. Then the friction losses on the pipes walls and at the ends of the pipes can be determined. The software provides calculation of the heat transfer in each of the elements presented into the intake system. Although heat transfer model can be neglected in some of the elements of the intake system, especially the elements before the compressor where

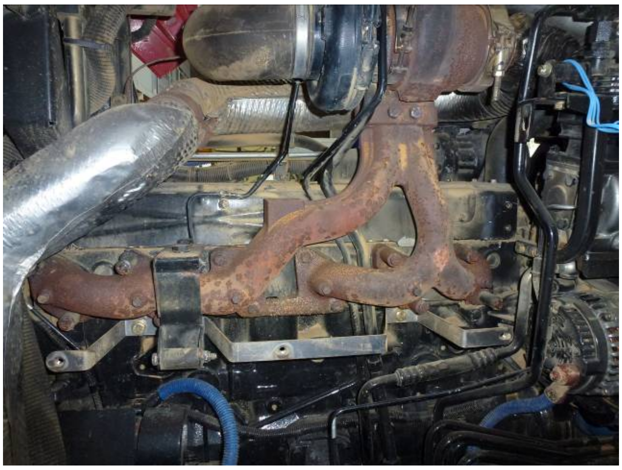

Fig.7 Exhaust manifold and its connection with the turbine housing

the gas temperature inside is quite the same as the environment.

3.4 Modeling the exhaust system

The exhaust system includes: exhaust manifold separated into two banks of cylinders, turbine, silencer, and connection pipes between the elements. The pipe connecting the outlet port of the turbine with the silencer is isolated as can be seen in Fig.7.

The exhaust manifold is modeled by pipes and two plenums (PL2) and (PL3). The pipes number 10 and 11 (Fig. 5) connect each of the plenums with the turbine inlet. The turbine (TC1) is determined by turbine discharge coefficient and turbine efficiency. The mechanical efficiency of the turbocompressor is necessary for power balance. The silencer is simplified by means of the plenum (PL4). The end of the exhaust is represented by boundary element (SB2) with the thermodynamic parameters of the environment.

Within the plenum elements of the exhaust system the heat exchange model is determined taking into consideration the transfer surface and the temperature of the wall. A more complicated model is used in the pipes taking into consideration the heat transfer trough the wall. The thickness of the wall of the pipes, the temperature and the speed of the air outside of the pipe are necessary for this model.

3.5. Modeling the cylinders

The element type "cylinder" provides an assessment of the in-cylinder parameters based on the 0D model including the combustion and heat exchange model. Six equal cylinders are used in the engine model. The estimation of the volume change as a function of the crankshaft rotation is based on the geometrical parameters of the cylinder, compression ratio end kinematic of the crankshaft mechanism.

Different combustion models can be used in AVL Boost. In the current study a simple Vibe function was chosen because of its simplicity. The rate of heat release is defined by the start of combustion, duration of the combustion and the Vibe exponent (Fig. 8). There are no the experimental results for the rate of heat release and the values of the function parameters have been chosen as typical for direct injection diesel engine. The values were changed depending on the engine load and speed.

The heat exchange model is based on the convective heat transfer between the gasses and the cylinders walls. The Woschni equation [8] was chosen for estimating the heat transfer coefficient. The heat transfer surface of the piston and cylinder head are input as constant parameters while the cylinder wall surface is calculated depending on the position of the piston. The constant temperature of the top of the piston and the cylinder head are used. Their values

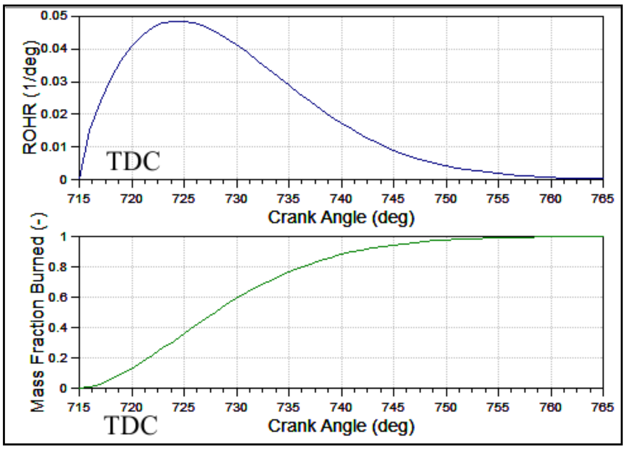

Fig.8 Rate of heat release and mass fraction burned presented by simple Vibe function

were changed according to the engine load. A variable wall temperature of the cylinder liner is used. The change varies linearly with the height of the liner.

The valves ports specifications are included in the element "cylinder". The lift curves both of intake and exhaust valves are input as well as the flow discharge coefficients. The reference valves diameters are taken from the engine specification. The valves lift curves are shown in Fig.9.

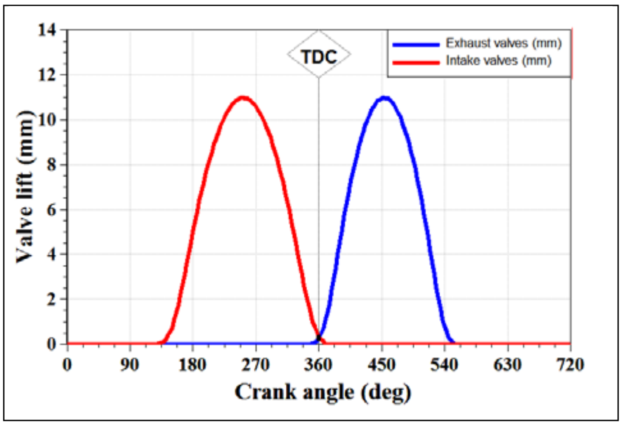

Fig.9. Valves lift curves

4. Model calibration

Calibration of the model has been made before further analysis of the studied engine. The comparison between experimental data and calculated values is made at full engine load. The experimental data are measured on the rolling test bench at Irstea. The data for fuel consumption, engine load, engine speed, engine power and torque, inlet temperature etc. are received. The values of the fuel consumption are input in the model. The intake pressure was calculated depend on engine load. The measure engine power is used for calibration of the model. At those speeds where the differences with experimental data were higher the correction at Vibe function coefficients were made. The optimized values of the combustion duration and the exponent of the function can be seen in Fig. 10. Finally, the comparison shows small difference between experimental and calculated

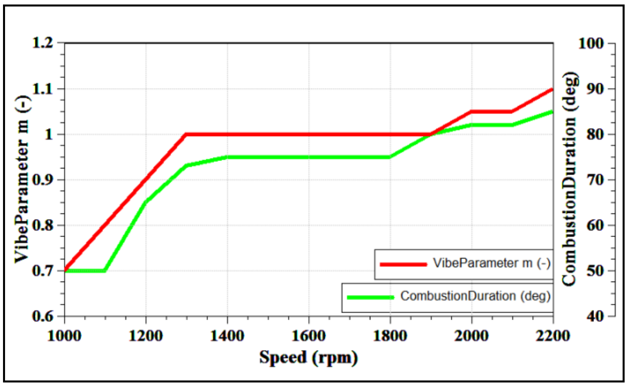

Fig.10. Optimized values for combustion duration and exponent of Vibe function at full engine load

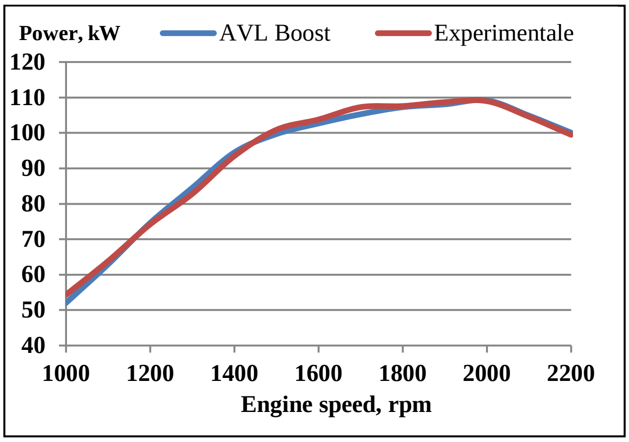

Fig.11. Comparison between measured and calculated engine power

performance curves. The results are shown in Fig. 11. The results of calibration enabled us to use the model for assessing the energy balance of the engine and the exergy contents in the exhaust gases.

5. Energetic and exergetic analysis of the engine

5.1. Energy balance of the tractor engine

By means of the model of the engine, has been made in AVL Boost, the distribution of fuel energy can be made in the whole working field. As it was mentioned before the tractor engine operates most of the time at constant speed and with load close to the maximum. Because of that first analyze is made at full engine load.

In general the fuel energy is distributed as energy converted in effective power and lost energy. The energy is lost through: exhaust gases, cooling system and friction. Calculation of the exhaust gases energy is important in order to estimate the possibilities for recuperation.

The result of the energy balance (Fig.12), calculated in energy per time i.e. power, shows that energy of the exhaust gases is a significant part of fuel energy at full engine load. At low engine speed the quantity of wasted energy with the exhaust gases is similar to both the energy converted in effective power and the energy lost in the cooling system. At an engine

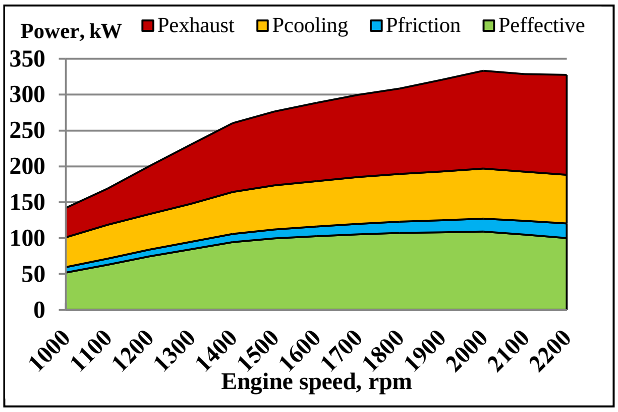

Fig.12. Fuel power distribution in a tractor engine at full engine load

speed above 1400rpm this energy (contented in the exhaust gases) is the biggest part of the fuel energy. In absolute value the waste energy with exhaust gases varies from 41kW to 139,9kW. The maximum value of waste energy in the exhaust system is at maximum speed of 2200min^{-1}. In comparison, the maximum effective power of the engine is 109,2kW at 2000min^{-1}.

In relative values (Fig.13) the waste energy with exhaust gases is more than 28,9% of fuel energy. This value slightly increases depending on engine speed up to 42,5% at 2200rpm. Increase of waste energy depending on engine speed can be described with reducing the time for the working cycle. It extends the combustion process and part of fuel energy is released during expansion stroke. Usually, it increases exhaust gases temperature and reduces engine efficiency.

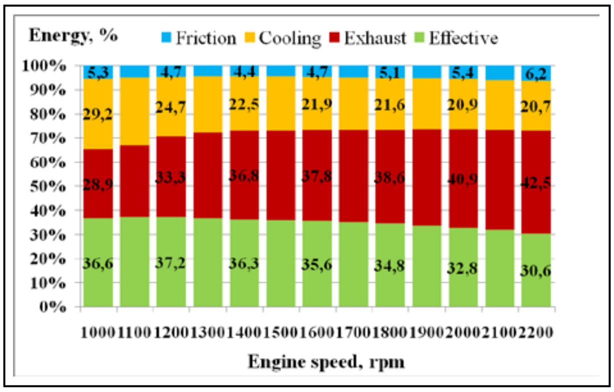

Fig.13. Energy distribution in relative values at full engine load

5.2 Energetic and exergetic analysis in exhaust system.

Calculation of the energy going out of the cylinders is not enough to make an estimation of the recuperation potential of the exhaust gases. Part of this energy is converted into mechanical power in the turbine, another part is lost by heat transfer with the environment in the exhaust pipes.

Analysis of the exhaust system design shows enough space for mounting the heat exchanger of Rankine system at the place located before the silencer. This section of the exhaust system can be seen in Fig.14.

Fig.14. Exhaust system view

The place shown in Fig. 14 is located 1800mm down the turbine outlet. It corresponds to MP1 in Fig.5. The energy in this section (enthalpy of the gases) has been calculated. The comparison between this energy and enthalpy of the gases going out of the cylinders shows the lost energy in the form of power through the gases path. The results are shown in Fig.15. Available energy at the chosen place presents approximately 75% of energy which goes out

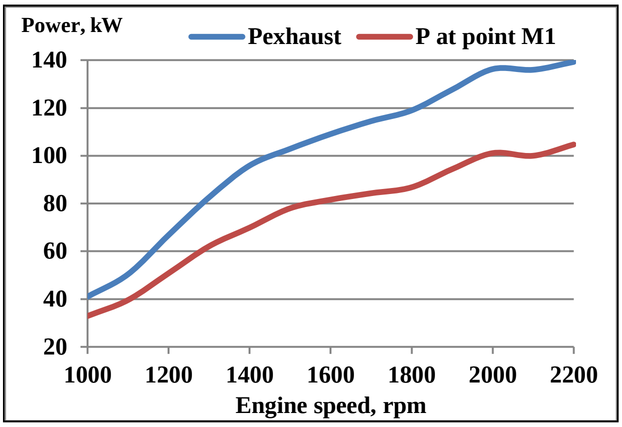

Fig.15. Power of the exhaust gases at exhaust valve port and at place for heat exchanger mounting at full load.

of the cylinders.

Calculation of the exhaust gases enthalpy and temperature at the whole operation field of the engine is necessary for sizing the elements of the Rankine system and the choice of the working fluid. A study of

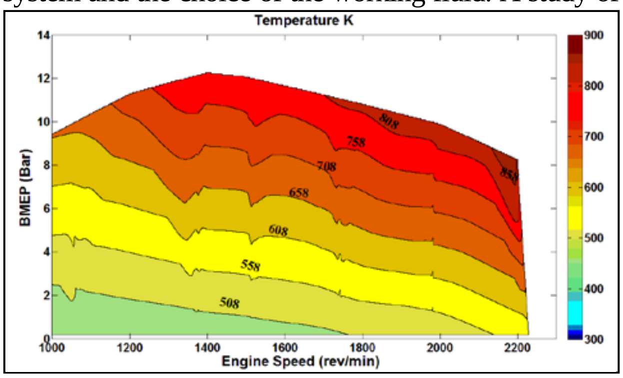

Fig.16. Temperature of the exhaust gases at studied point of exhaust system

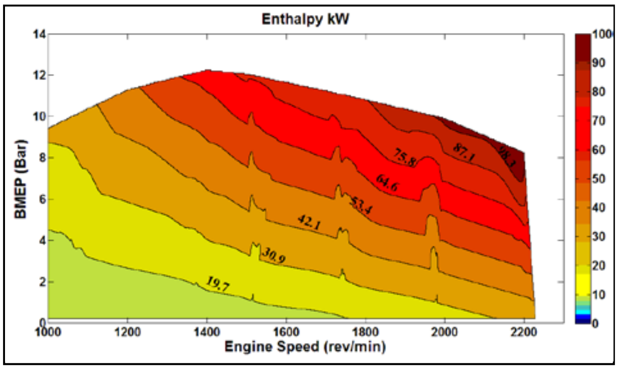

Fig.17.Enthalpy of the exhaust gases at studied point of the exhaust system

the temperature and the enthalpy at point MP1 was made. The results are presented in Fig.16 and Fig.17.

According to the results for engine speed range from 1400rpm to 1800rpm and BMEP about 10 bars (typical operation points of the engine when the tractor runs in the fields) the enthalpy of exhaust gases changes from 50kW to 80kW with temperature in the range of 700K to 800K. Taking into consideration that the engine runs more than 80% of the time between these operation points there is great potential for developing the energy recuperation system.

The numerical results at the studied location together with the experimental data of the engine operation points can give information about the energy which comes to the heat exchanger of the Rankine system as a function of the time.

It is well known from thermodynamics that the whole energy of exhaust gases cannot be converted to useful work. Thus the estimation of waste heat recuperation potential must be made by means of available energy i.e. exergy. The exergy represents the maximum work which can be received by thermodynamic system when it goes to a state of equilibrium. Exergy is defined as:

$$Ex = (H_1 - H_0) - T_0(S_1 - S_0) \qquad (1)$$

The parameters, marked with index 1 in the above formula, are enthalpy and entropy of exhaust gases while these, marked with the index 0, are the parameters of the environment.

The simulation results (Fig.18 and Fig.19) show that exergy is not more than 45% of the exhaust gases enthalpy. Theoretically a maximum of about 45kW can be converted into mechanical power by the recuperation system. In comparison with the maximal effective power of the engine it is 40,9%. This exergy is available only in small part of the operation field at maximum speed and full load. In the most commonly used operation points of the tractor engine the exergy is from 22kW to 35kW.

The effective power received from the

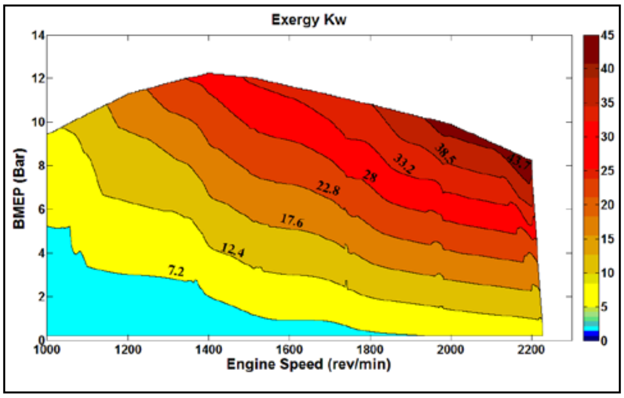

Fig.18.Exergy of the exhaust gases at studied point of the exhaust system.

recuperation system depends on the available energy of

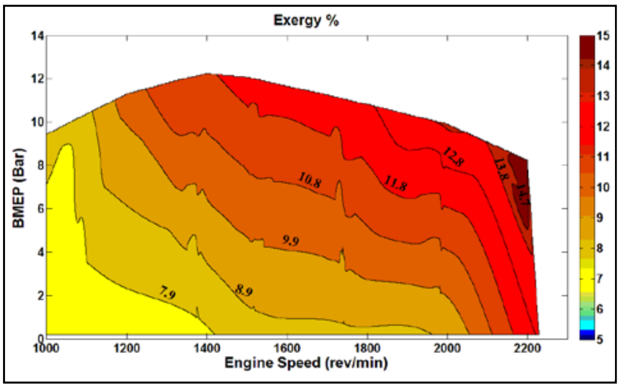

Fig.19.Exergy, calculated in relative values from LHV of the injected fuel

the exhaust gases and the efficiency of the recuperation system. As the tractor engine runs at constant regime most of the time that provides good opportunity to develop a high efficiency Rankine system.

Interesting results for exergy have been calculated in percentage from LHV (lower heat value) of the injected fuel. The results are presented in Fig.19. The values vary from 7% to 15%. Below 1400rpm and above 2000rpm the influence of engine speed is much higher than the engine load while at the middle speed we have almost constant relative exergy as a function of engine speed.

6. Conclusion

The numerical study of energy distribution within a tractor engine shows that a significant part of LHV of the fuel is lost with exhaust gases. At full engine load it varies from 28,9% to 42,5% of LHV of the fuel. As absolute values in form of power it is from 42,5kW to 139,9kW. The energetic analysis of the exhaust system presents only 75% of the energy go out the cylinders can be used as heat source to recuperation system based on Rankine cycle. The rest of the energy is lost in the turbine and due to heat transfer troughs the gases path to the location suitable for the heat exchanger.

Taking into consideration the typical operation points of the tractor engine during plowing (more than 80% of the time the engine runs at high load and almost constant speed) and energy analysis at the location intended for heat exchanger most of the time the temperature of exhaust gases would be varied from 700K to 800K with exhaust gases power from 50kW to 80kW.

The exergetic analysis shows maximum of 15% of LHV can be converted into mechanical work by means of closed loop thermodynamic cycle. At the most typical operation points this value is from 10% to12,8%. However, within the studied tractor engine the absolute value of this exergy varies from 22kW to 35kW.

Acnowledgement

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ВЪЗМОЖНОСТИ ЗА РЕКУПЕРАЦИЯ НА ЕНЕРГИЯ ОТ ОТРАБОТИЛИТЕ ГАЗОВЕ НА ТРАКТОРНИТЕ ДВИГАТЕЛИ

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Резюме:

Тракторните двигатели имат значителен потенциал за рекуперация на енергия от отработилите газове, поради високата им мощност и специфичните особености на тяхната работа. Система за рекуперация на енергия, на базата на цикъл на Ранкин, изгглежда най-перспективния начина за намаляване на разхода на гориво, респективно емисиите на CO_2. Експерименталните изследвания върху работата на тракторен двигател показват, че този тип двигатели работят при високо натоварване, близко до максималното през по-голямата част от времете, през което трактора работи на полето. Целта на публикацията е да се изследва енергията на отработилите газове в различни точки от изпускателната система на тракторен двигател. За тази цел е разработен модел на изследвания двигател в програмата AVL BOOST. Експеримнатлни резултати за разхода на гориво, ефективната мощност, разхода на въздух, отделената топлина в охладителната система, температурата на отработилите газове и др. са използвани за калибриране на модела. Анализа на разпределението на енергията в двигателя показва, че повече от 35% от енергията внесена с горивото се губи чрез отработилите газове при най-често използваните режими на работа. В заключение е представен анализ на наличната топлина в една определена точка по дължината на изпускателната система. Резулатат ите показват добри перспективи за по нататачни излседвания върху възможността за монтиране на система, работеща на принципа на Ранкин на изследвания дизелов двигател.

Ключови думи: *рекуперация на енергия, трактор, дизелов двигател, енергия, моделиране на двигател.*