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TEZĂ DE DOCTORAT

Recuperarea căldurii din gazele de evacuare ale unui motor diesel folosind ciclul Rankine organic

Diesel engine exhaust heat recovery using Organic Rankine Cycle

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CHAPTER 5

Summary

In recent years, there has been a great deal of waste heat energy being released into the environment, such as exhaust gases from turbines and engines and waste heat from industrial plants, which lead to serious environmental pollution. In addition, there are also abundant geothermal resources and solar energy available in the world. These heat sources are classified as low grade heat sources. Therefore, more and more attention has been paid to the utilization of low grade waste heat nowadays for its potential in reducing fossil fuel consumption and alleviating environmental problems.

Since conventional steam power cycles cannot give a better performance to recover low grade waste heat, the organic Rankine cycle (ORC) is proposed to recover low grade waste heat. There are several advantages in using an ORC to recover low grade waste heat, including economical utilization of energy resources, smaller systems and reduced emissions of CO, CO2, NOx and other atmospheric pollutants. The main advantage of the ORC is its superior performance in recovering waste heat with a low temperature.

Besides the ORC, researchers have proposed various thermodynamic cycles, such as Kalina cycle, Stirling cycle, and Ericsson cycle, to convert this low-grade heat sources into electricity. Although there is more power output for the same heat input with Kalina cycles compared to ORCs, the ORC syetem is much less complex and needs less maintenance.

The Organic Rankine Cycle is a cost efficient and proven method of converting low temperature waste heat to mechanical and/or electrical energy. This opens up the possibility to exploit low-grade heat that otherwise would be wasted. It can play an important role to improve the thermal efficiency of internal combustion engines.

A thermal engine converts 30 % of the fuel energy into mechanical shaft work; the rest of energy is wasted through the cooling liquid and the exhaust gases. Thus, it would be possible to convert this wasted heat in order to improve the engine overall efficiency and reduce the fuel consumption of the vehicle.

This paper describes the performance of exhaust heat recovery using an ORC in a passenger car.

In chapter 1, the characteristics of a passenger car-based internal combustion engine are analyzed. From exhaust gas temperatures and exhaust gas mass flows, the characteristic of available waste heat over load and speed is estimated.

To design a reasonable system to utilize various waste heats from the diesel engine with high efficiency, studying the energy distribution in the running process of the diesel engine is necessary. When an engine is running, the energy and exergy quantities of the exhaust and the coolant are significantly different. Because of this, it is very difficult to design a system that can comprehensively recover waste heat from both the exhaust and the coolant of that system. A four-cylinder in-line diesel engine is used as the object of analysis. The main technical performance parameters are listed in Table 1.

When a vehicle is running, the engine speed and load can vary through a wide range. Therefore, the engine performance test was conducted in an engine test cell in order to obtain the thermodynamic parameters of the exhaust and coolant systems overall possible engine operating regions as defined by the engine speed and output torque. For our measurements, the minimal and maximal engine speeds were set to 1000 rot/min and 4500 rot/min, respectively. The intermediate speeds were selected using a step increment of 250 rot/min, starting from the minimum engine speed. At each selected engine speed, different load values were selected, ranging from a 100% load to a minimal stable load value. The values for the output torque, the output power, the engine speed, the mass flow rate of the intake air, the injected fuel quantity, the exhaust gas temperature, and the coolant temperatures at the outlet of the engine's water jacket were all recorded for each load and speed configuration.

Table 1: The main technical performance parameters of the diesel engine [3]

Items	Parameters	Units
Model	Diesel	[-]
Cylinder number	4	[-]
Stroke and cylinder bore	88.3x75	[mm]
Displacement	1560	[cm$^{3]}$
Compression ratio	18:1	[-]
Air intake type	Turbocharged and Intercooled	[-]
Fuel injection system	High pressure common rail	[-]
Rated power	80	[kW]
Rated speed	4000	[rpm]
Maximum torque	240	[Nm]
Speed at maximum torque	1800	[rpm]

The distribution of fuel energy released by combustion under a certain operating condition of the diesel engine is depicted using the first law of thermodynamics, which is:

$$\dot{Q}_{cb} = P + \dot{Q}_r + \dot{Q}_g + \dot{Q}_{rest}$$

(1)

Where: $\dot{Q}_{cb}$ is the heat flux received through fuel combustion; P is the amount of mechanical power produced; $\dot{Q}_r$ is the heat flux rejected through the water cooling system; $\dot{Q}_g$ is the heat rejected through the exhaust gases and $\dot{Q}_{rest}$ is the heat flux rejected through radiation and incomplete combustion that cannot be directly determined in this stage.

The heat flux received through fuel combustion can be computed as:

$$\dot{Q}_{cb} = \dot{m}_{cb} H_{icb} \tag{2}$$

Where $\dot{m}_{cb}[kg/s]$ is the fuel mass flow rate and $H_{icb} = 4200[kJ/kg]$ is the inferior fuel heat value. The inferior fuel heat value is considered from data available in literature [5,6].

Next, the heat flux rejected through the water cooling system can be computed as follows:

$$\dot{Q}_r = \dot{m}_w c_w (t_e - t_i) \tag{3}$$

In eq. (3) $\dot{m}_w[kg/s]$ is the water mass flow rate; $c_w = 4.186[kJ/kgK]$ is the water heat capacity; $t_e[°C]$ and $t_i[°C]$ are water temperatures at engine outlet and inlet, respectively.

The heat flux $\dot{Q}_g [kW]$ rejected through exhaust gases is computed as:

$$\dot{Q}_g = \dot{Q}_{eg} - \dot{Q}_{air} \tag{4}$$

Where, $\dot{Q}_{eg} [kW]$ is the total heat flux available in exhaust gases and $\dot{Q}_{air} [kW]$ is the heat flux due to the fresh load.

The quantity of waste heat contained in exhaust gas is a function of both the temperature and the mass flow rate of the exhaust gas:

$$\dot{Q}_{eg} = \dot{m}_g c_{pg} T_g \tag{5}$$

In eq. (5) $\dot{m}_g [kg/s]$ is the exhaust gasses mass flow rate; $c_{pg} [kJ/kgK]$ is the heat capacity at constant pressure of exhaust gases and $T_g [K]$ is the temperature of exhaust gases. Heat capacity of exhaust gases is considered from available data in literature according to experimental data [5].

The heat flux due to the fresh load can be determined:

$$\dot{Q}_{air} = \dot{m}_{air} c_{pair} T_{air} \tag{6}$$

Where, $\dot{m}_{air} [kg/s]$ is the air mass flow rate $c_{pair} = 1.013 [kJ/kgK]$ is the heat capacity at constant pressure and its value is considered from data available in literature [7] and $T_{air} [K]$ is the measured ambient air temperature.

One common way to present the operating characteristics of an internal combustion engine over its full load and speed range is to plot brake specific fuel consumption contours on a graph of brake mean effective pressure (or engine torque) versus engine speed.

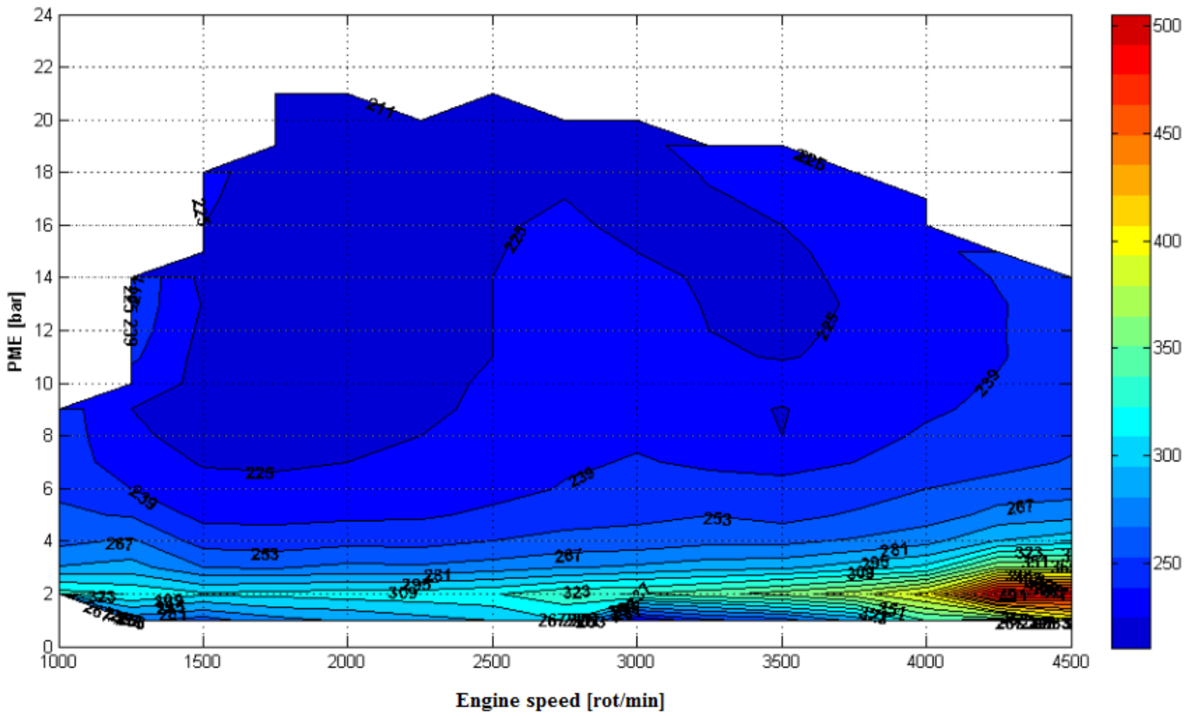

Fig. 1 Brake specific fuel consumption [g/kWh]

The measured engine performance map is displayed in Figure 1. The lowest brake specific fuel consumption (b.s.f.c.) zone is situated at the high duty range between 1500 rot/min and 3000 rot/min and the minimum b.s.f.c. value is less than 210 g/kWh.

The effective thermal efficiency is defined as the ratio of the output torque at the flywheel end to the fuel combustion energy, and the results are given in figure 2. The effective thermal efficiency reaches a peak of greater than 40% in the low b.s.f.c. region.

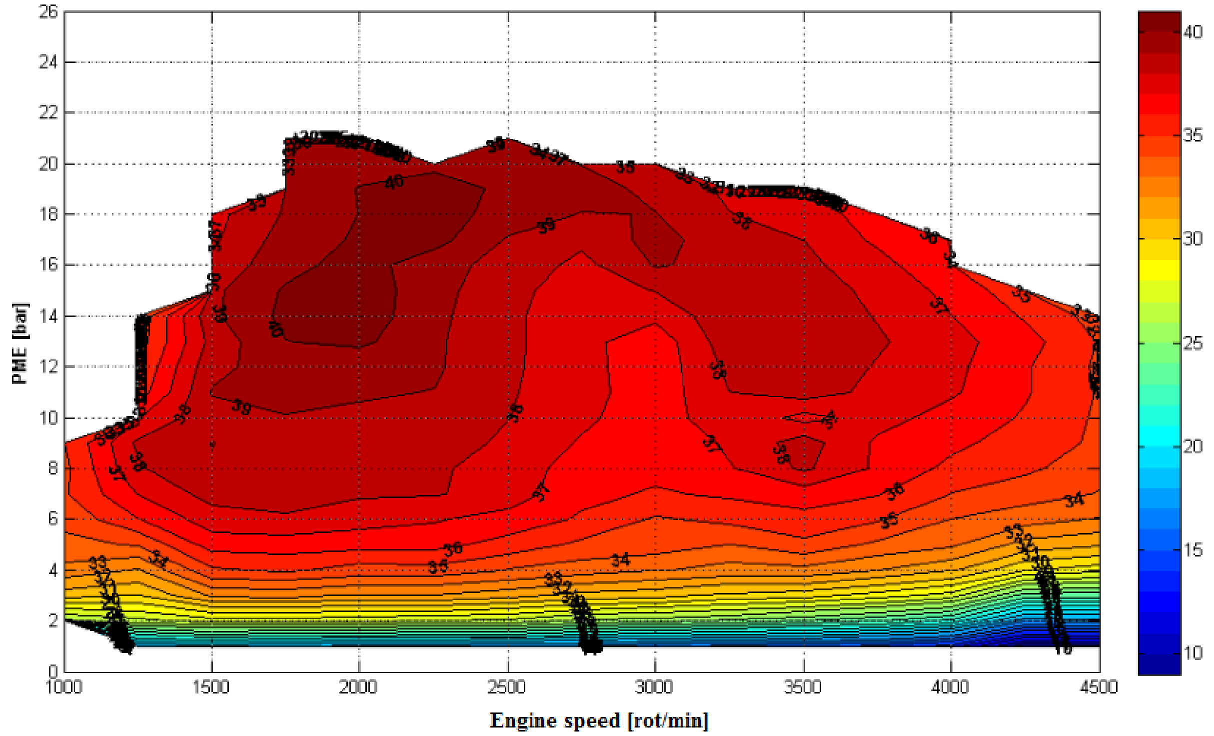

Fig. 2 Engine effective thermal efficiency [%]

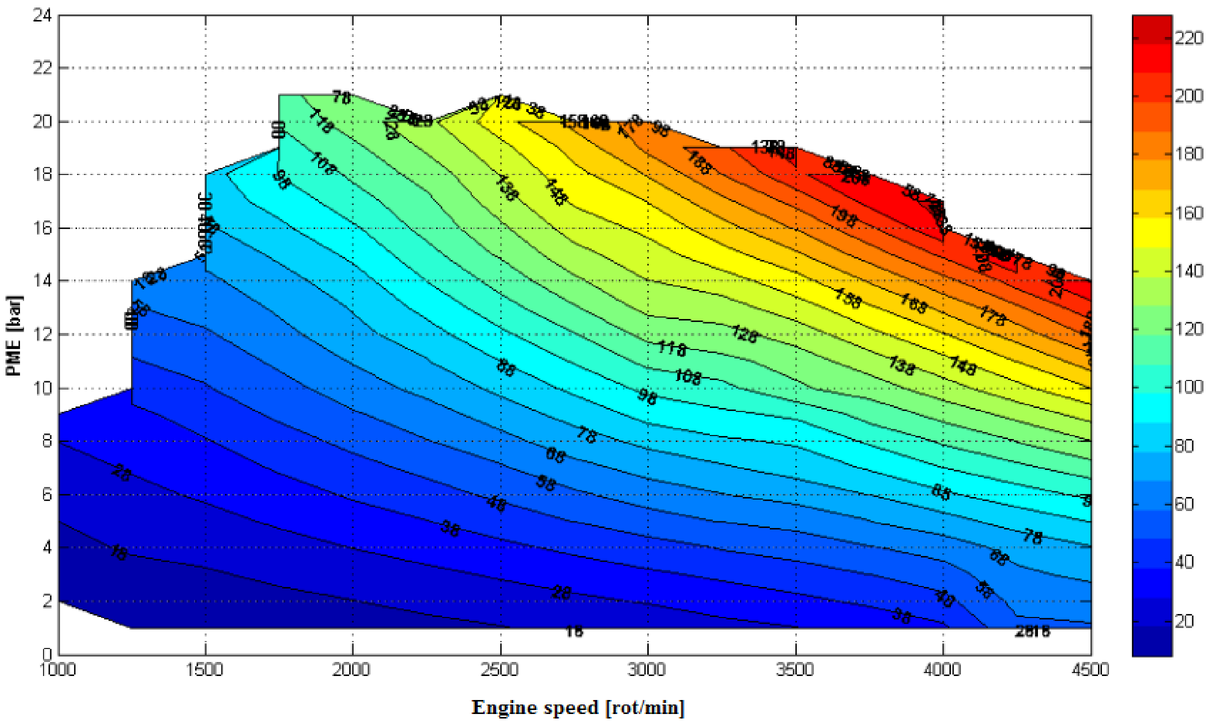

Fig. 3 Fuel combustion energy [kW]

The fuel energy released by combustion is shown in Figure 3. As the engine speed and engine load increases, the fuel energy released by combustion increases gradually. Such phenomenon is primarily caused by the increase in fuel consumption and intake air mass. The combustion energy increases almost linearly with the engine output power, achieving 220 kW at the rated power point. Note that the waste heat quantities of the exhaust and the coolant vary in a similar fashion. The variation of the waste heat quantity carried by the coolant system and exhaust gases over the whole operating range, are shown in figures 4 and 5.

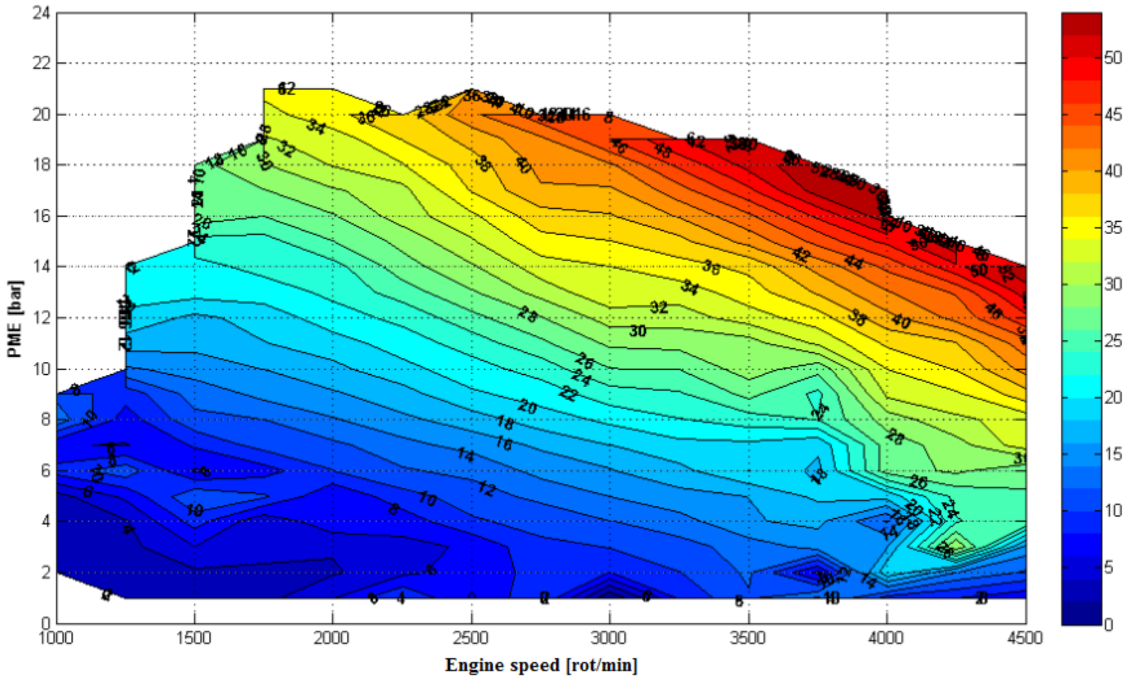

Fig. 4 Energy part of cooling system [kW]

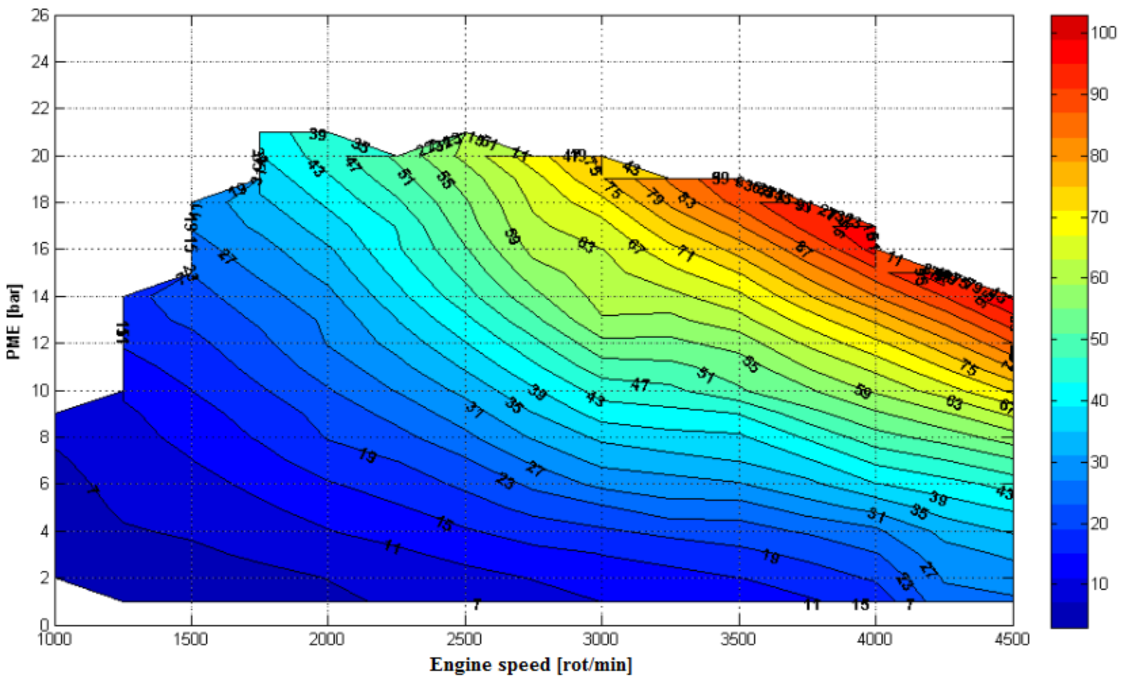

Fig. 5 Energy part of exhaust [kW]

The purpose of second chapter is to compare different waste heat recovery system technologies designed for automotive applications. The exhaust line waste heat energy can be recovered by different means. The use of heat engines describing thermodynamic cycles such as Rankine and Stirling engines is possible. A turbine similar to the one of car turbocharger can also be used (turbocompounding) and coupled to an electric machine or to the transmission line of the vehicle. Thermoelectricity is another alternative in which heat is directly converted into electricity. Turbocompounding and Rankine cycle systems are the most probable technologies to be soon integrated into passenger cars.

The third part reviews the history of internal combustion engine exhaust waste heat recovery focusing on Organic Rankine Cycles since this thermodynamic cycle works well with the medium-grade energy of the exhaust. Selection of the ORC arhitecture, expander design and working fluid are the primary focus of the review, since they are regarded as having the largest impact on system performance.

For each intended application, the additional expenditure and complexity associated with incorporation of preheating with engine coolant waste heat or a recuperator above the traditional Rankine cycle should be weighed against the resulting efficiency gains. No configuration is optimal for every waste heat source; hence, a thermodynamic analysis targeting the specific source must be conducted first.

Review of the literature demonstrates that selection of the working fluid and expander has a significant influence on the efficiency of the WHR system. Most applications achieve the highest ORC efficiencies using nearly isentropic and high critical temperature working fluids. However, these criteria fail to address numerous practical design conditions, such as operating pressures, component sizes, expander rotational speeds, expansion ratios, and environmental concerns. Thus, the space available onboard mobile waste heat sources should be determined prior to cycle design.

Results demonstrate a potential fuel economy improvement around 10% with modern refrigerants and advancements in expander technology.

Chapter 4 describes the performance of exhaust heat recovery using an ORC in a passenger car. The heat transfer properties are evaluated over the engine's entire operating region based on the measured data. Subsequently, a mathematical model of the plate heat exchangers is created based on the specific ORC working conditions. The main aims of this study are 1) the determination of the proper working fluid for ORC system, and 2) the calculation of the heat transfer coefficient and the required surface area for the plate heat exchangers (evaporator and condenser).

The ORC is a vapor power cycle used in numerous applications to generate electrical power. Figure 6 shows a schematic of a simple ORC. It is composed by four main components: a pump, an evaporator, a turbine/generator and a condenser.

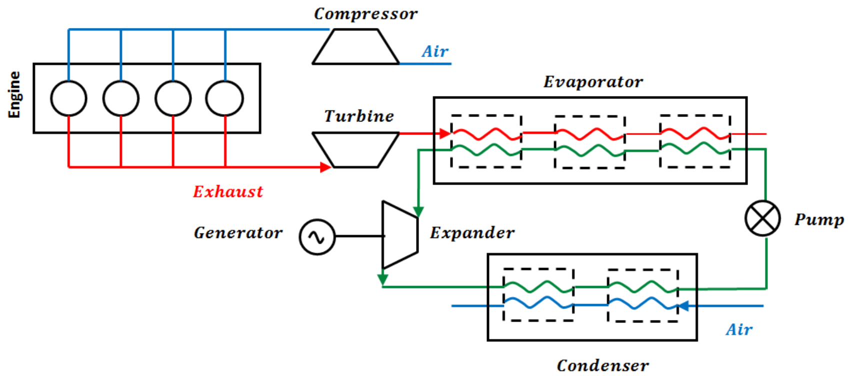

Fig. 6. Schematic of an ORC for engine exhaust heat recovery

The associated T–s diagram of the ORC is described in figure 7. The ideal thermodynamic cycle includes the following processes: an isentropic compression process in a pump (1-2), an isobaric heat transfer process in an evaporator (2-3), an isentropic expansion process through a turbine (or other expansion machine) (3-4), and an isobaric heat transfer process in a condenser (4-5-1).

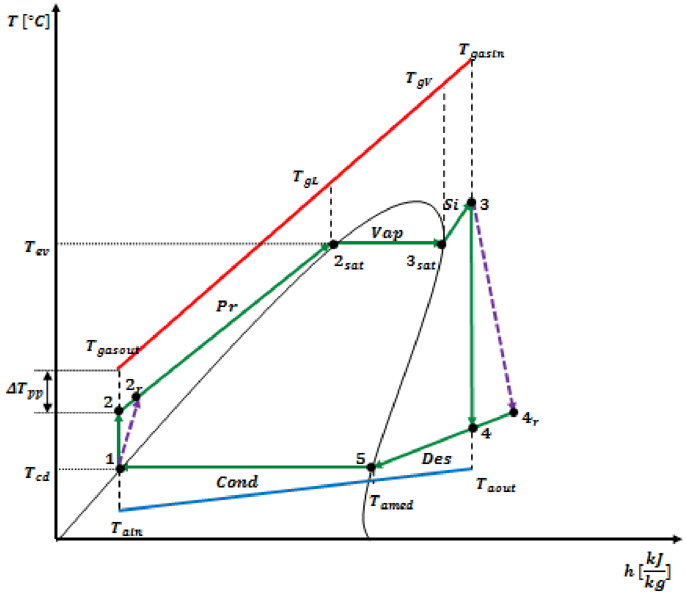

Fig. 7. T–s diagram of the ORC

The performance of the Rankine cycle depends on the heat exchanger's effectiveness as well as the pump and expander selection. The approach used in this work draws inspiration from the work of Vargas et al. [5] where the evaporator is assumed to be divided into three sub-segments "a preheater, a boiler and a superheater" linked in series and the condenser is split into two virtual zones corresponding to the state of the working fluid [6] i.e. two phase and gas phase.

Before create the mathematical model of this system to simplify the analysis, some general assumptions are formulated as follows:

- Steady-state and steady-flow condition;
- No pressure drops in heat exchangers and connecting pipes;
- The condenser temperature is assumed to be 45 °C;
- Exhaust gas temperature at the exit of evaporator is 140 °C due to prevent condensation of components;
- The expander mechanical efficiency, η_D = 70%;
- The efficiency of the pump, η_P = 80%;
- Ambient temperature is 20 °C;
- Heat exchanger effectiveness η_{PHE}=98%.

After choosing the working fluid, the mass flow rate of the working fluid and the heat transfer rates for all zones are computed according to energy equation. Subsequently, the convective heat transfer coefficients of each zone are calculated according to the heat transfer correlations and the thermodynamic properties of the exhaust gas and working fluid on each side. Furthermore, the overall heat transfer coefficient of each zone is obtained. Then, the heat transfer area required for each zone is determined using the logarithmic mean temperature difference (LMTD) method.

The total heat transfer rate between the counter flows in plate heat exchangers can be calculated as follows:

$$\dot{Q}_{pr} = \dot{m}_{ref}\left(h_{2sat} - h_2\right) = \dot{m}_{gas} C_{Ppr}\left(t_{gL} - t_{gasout}\right) \tag{7}$$

$$\dot{Q}_{vap} = \dot{m}_{ref}\left(h_{3sat} - h_{2sat}\right) = \dot{m}_{gas} C_{Pvap}\left(t_{gV} - t_{gL}\right) \tag{8}$$

$$\dot{Q}_{s\hat{\imath}} = \dot{m}_{ref}\left(h_3 - h_{3sat}\right) = \dot{m}_{gas} C_{Ps\hat{\imath}}\left(t_{ga\sin} - t_{gL}\right) \tag{9}$$

$$\dot{Q}_{des} = \dot{m}_{ref}\left(h_4 - h_5\right) = \dot{m}_{aer} C_{Paer}\left(t_{aout} - t_{amed}\right) \tag{10}$$

$$\dot{Q}_{cond} = \dot{m}_{ref}\left(h_5 - h_1\right) = \dot{m}_{aer} C_{Paer}\left(t_{amed} - t_{ain}\right) \tag{11}$$

The logarithmic mean temperature difference can be obtained from the basic counter flow LMTD equation:

$$\Delta T_m = \frac{\Delta T_{\max} - \Delta T_{\min}}{\ln \dfrac{\Delta T_{\max}}{\Delta T_{\min}}} \tag{12}$$

The phase-change heat transfer process generally has three stages in which the heat transfer coefficient abides by different correction equations: liquid phase stage, two phase stage and vapor phase stage. The heat transfer processes for single-phase flow and two-phase flow are respectively discussed below.

In the single phase flow zone the Chisholm and Wanniarachchi correlation is employed to calculate the Nusselt number for both hot fluid and cold fluid, which is a function of the Reynolds, Prandtl numbers and the chevron angle of the plates [11]:

$$Nu = 0.724\left(\frac{6\beta}{\pi}\right)^{0.646} \mathrm{Re}^{0.583}\,\mathrm{Pr}^{1/3} \tag{13}$$

In the two-phase region (condensation or evaporation), the fluid properties such as density, specific heat, viscosity and thermal conductivity are observed to suffer from dramatic variations with the quality variation of organic working fluid. For this reason the heat transfer process in the two-phase region is divided into relatively small sections, with so slight property variations in each section that constant properties can be assumed.

For condensation and evaporation process, in two-phase region, Nusselt number is calculated using Yan and Lin’s correlation.

$$Nu_{R(i)} = 1.926\,\mathrm{Pr}_l^{1/3}\, Bo_{eq(i)}^{0.3}\,\mathrm{Re}^{0.5}\left[1 - x_i + x_i\left(\frac{\rho_l}{\rho_v}\right)^{0.5}\right] \tag{14}$$

$$Nu_{C(i)} = 4.118\,\mathrm{Re}_{eq(i)}^{0.4}\,\mathrm{Pr}_l^{1/3} \tag{15}$$

A program was created to evaluate the evaporator and condenser performance in Engineering Equation Solver (EES) according to the established mathematical model.

The first step in the design procedure of an ORC system is the selection of the organic fluid. To select an appropriate working fluid to achieve the maximum thermal efficiency and exergy efficiency in various working conditions, a preliminary selection was conducted. In addition, material compatibility, flammability, toxicity, global warming potential (GWP), Ozone depletion potential (ODP) and other properties also need to be considered when selecting working fluids. Based on these considerations five working fluids are used in the present study, their basic physical parameters are shown in table 2.

Figures below show the performances of working fluids investigated for the ORC system on following criteria: thermal efficiency, net power output per unit mass flow rate and total heat transfer area per net power output.

Tabel 2: The basic properties of the selected working fluids

No.	Working fluid	Type of fluid	t_{cr} [°C]	p_{cr} [bar]	ODP	GWP_{100}	Safety group classification
1	R245fa	Isentropic	154	36.4	0	1030	A1
2	SES36	Dry	177.5	28.49	0	3126	A1
3	R123	Isentropic	184	36.6	0.06	93	B1
4	R600a	Dry	152	37.96	0	3	A3
5	R141b	Isentropic	204.2	40.6	0.11	630	A2

Comparing the highest thermal efficiency value presented by each fluid, R141b is the highest one of about 15.25% at evaporating pressure 3.5 MPa, followed by R123 (14.75%, 3.66 MPa) > SES36 (13.53%, 2.85 MPa) > R245fa (12.22%, 3.64 MPa) > R600a (10.53%, 3.62MPa).

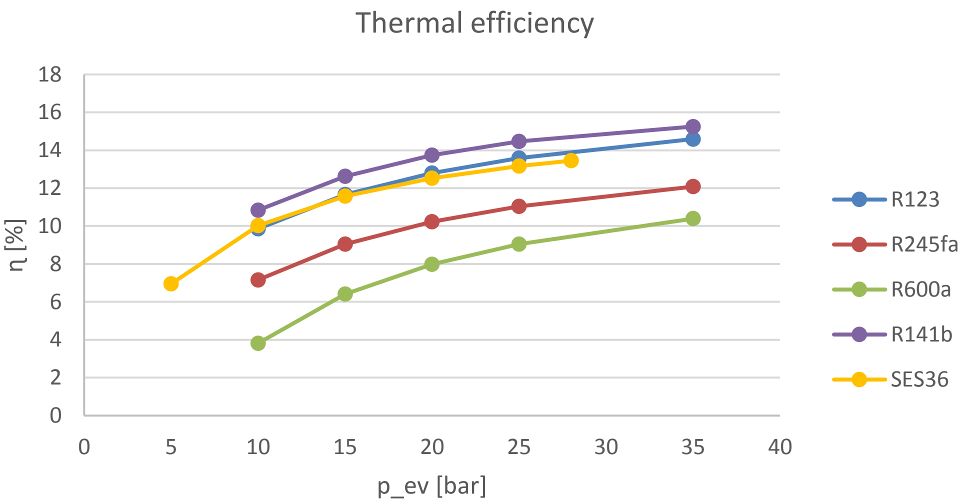

Fig. 8 Variation of thermal efficiency with evaporating pressure

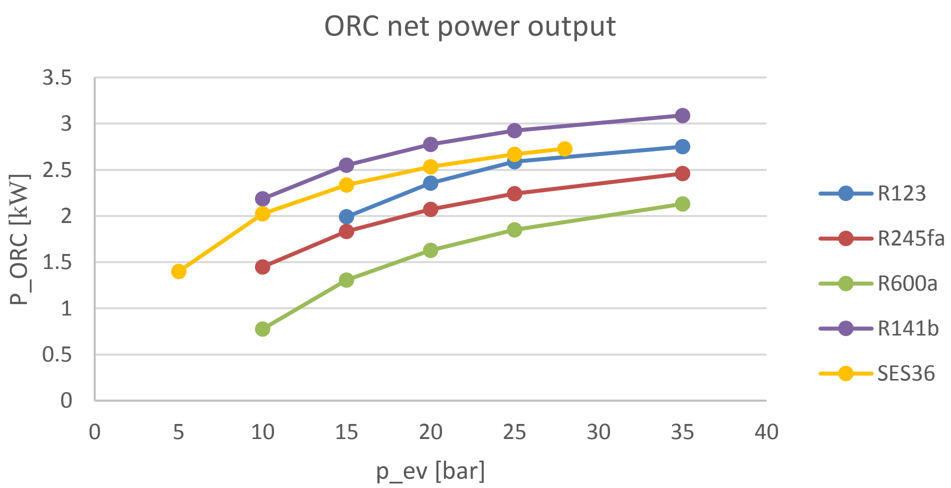

Fig. 9 Variation of net power output per unit mass flow rate with evaporating pressure

As shown in figure 9, ORC net power value is increasing as the increase of evaporating pressures. The increase trend of various working fluids is obvious at low evaporating pressures and becomes smooth near critical pressure. Among all the considered working fluids, R141b presents the highest net power value of about 3.089 kW at evaporating pressure 3.5 MPa.

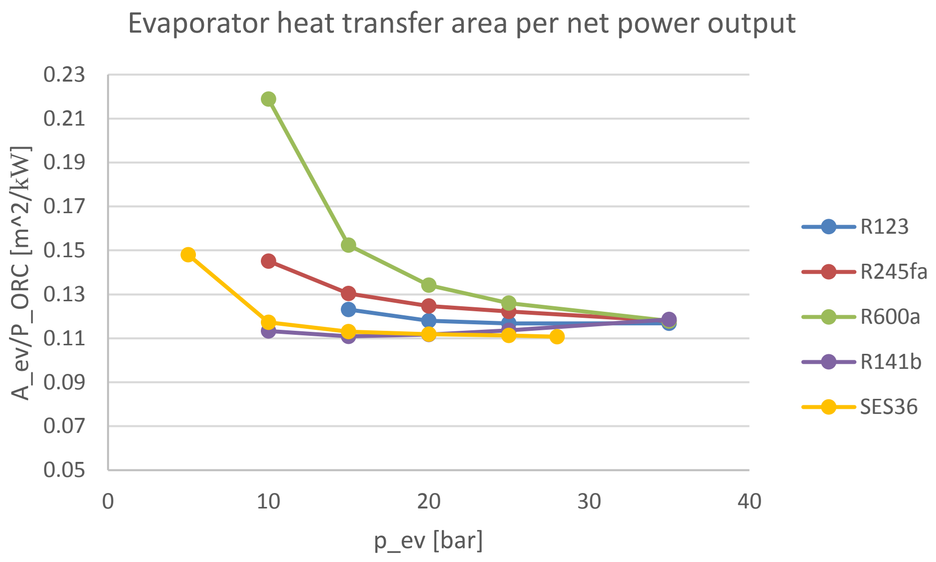

Fig. 10 Variation of evaporator heat transfer area per net power output with evaporating pressure

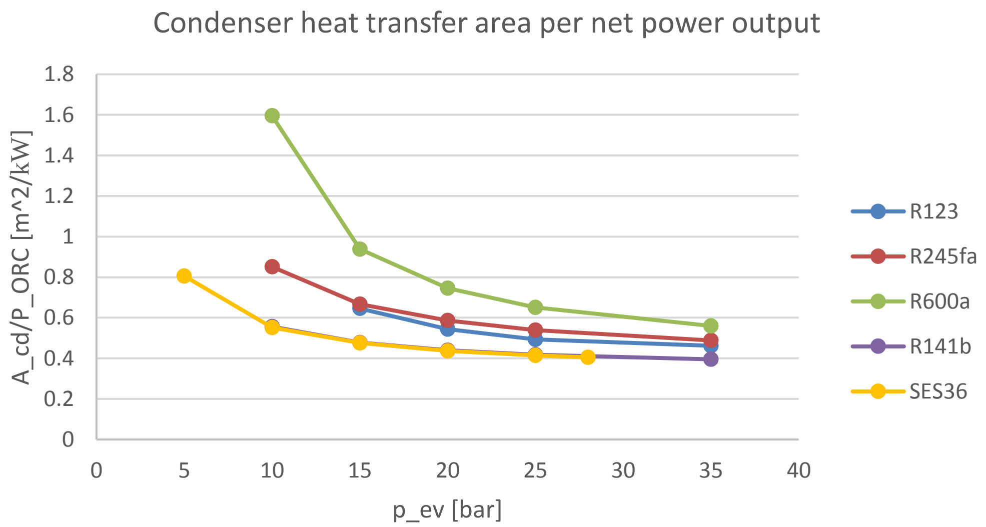

Fig. 11 Variation of condenser heat transfer area per net power output with evaporating pressure

It is noted that lower heat transfer area per net power output value expresses that smaller total heat transfer areas would be needed in order to achieve the same net power output which can indicate the heat transfer performance and reduce the system investment in some aspect [9]. As we can see form figures 10 and 11, SES36 and R141b show the lowest values.

As a results, the R141b show the best performance for the ORC system, but if we are considering the environmental characteristics (ODP value <0.20 and GWP value < 1500), the R245fa will be chosen as working fluid for our study.

To evaluate the plate heat exchangers performance, we first obtain the waste heat quantities of the exhaust of the diesel engine [7]. The variation of the heat transfer rate for each zone accordingly with entire engine's operating region is presented in figures 12-14. This variation characteristic is similar to that of the engine power because the waste heat energy provided by the exhaust gas increases with engine power. At the rated power point, the overall heat transfer rate reaches 60 kW.

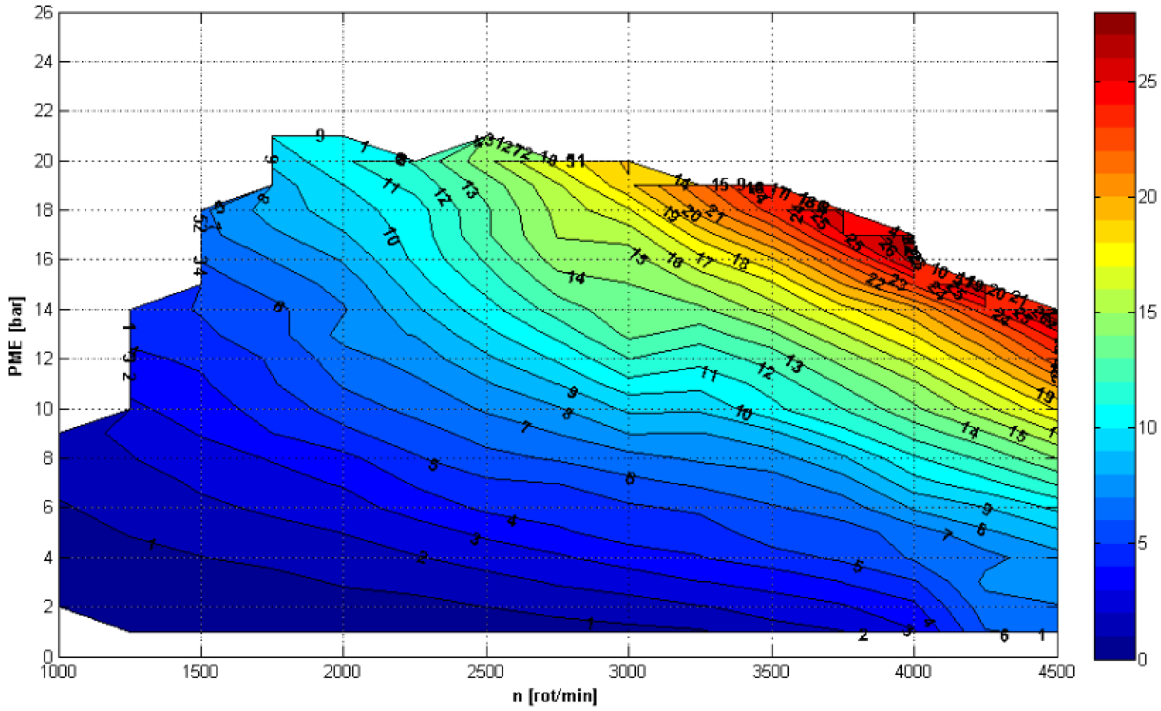

Fig. 12 Qpr

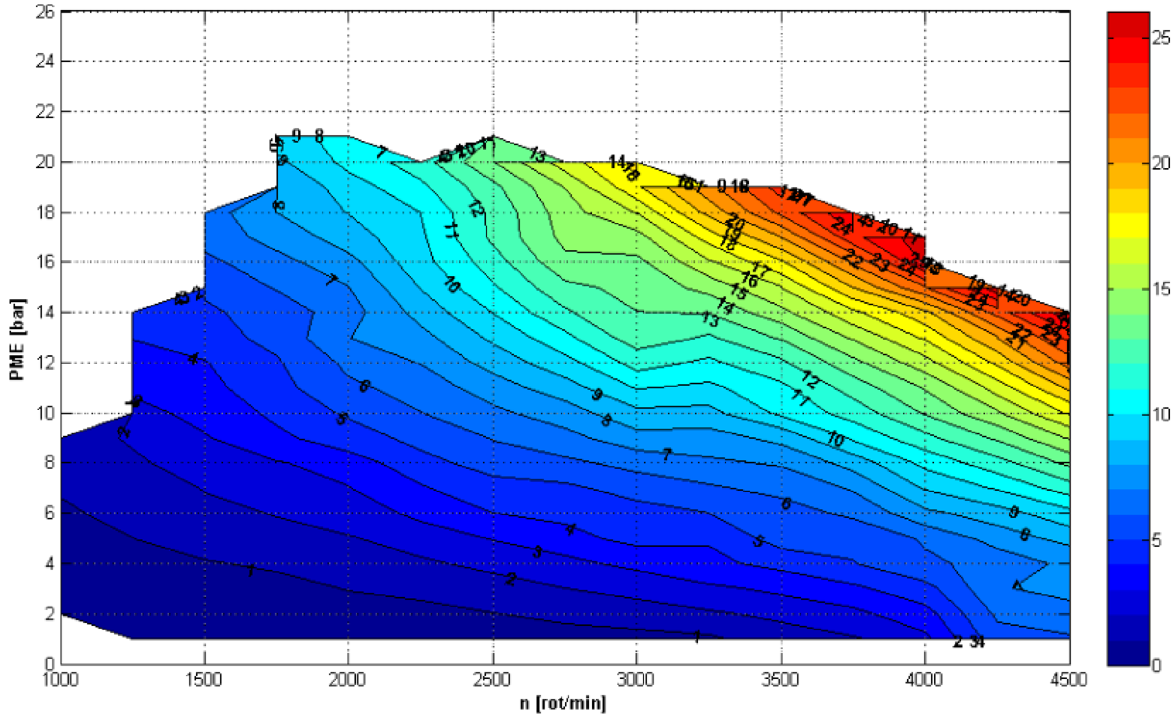

Fig. 13 Qvap

The values of the overall heat transfer coefficient depend on the heat transfer coefficient on the both hot and cold side.

The heat transfer area required for each zone is calculated using the LMTD method, figures 15 -18. At the rated power point, the heat transfer areas of evaporator and condenser are 0.35 m^2, and 2.15 m^2, respectively. The area of preheater and boiler zones is increase with engine speed and engine load while the area of superheater is decrease with engine speed and engine load. The percentage area for preheater zone is approximately 67% from total evaporator area, while for boiler zone is 23% and for superheater zone is 10 %. The desuperheater heat transfer area is almost 35% from condenser area.

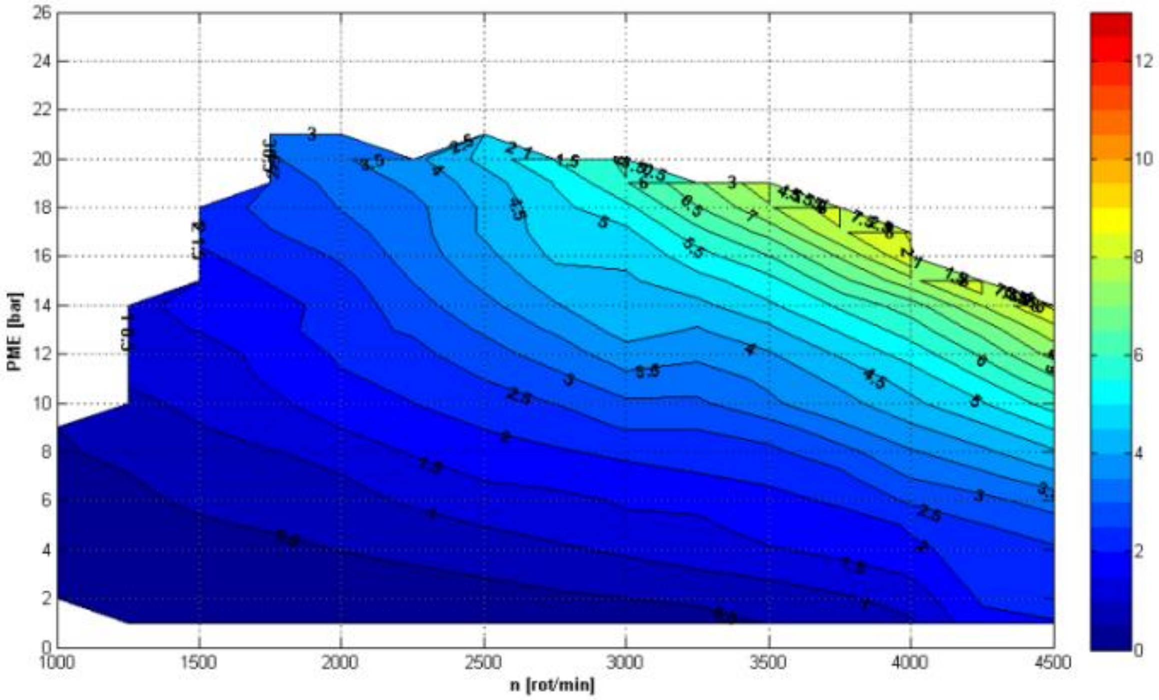

Fig. 14 Qsî [kW]

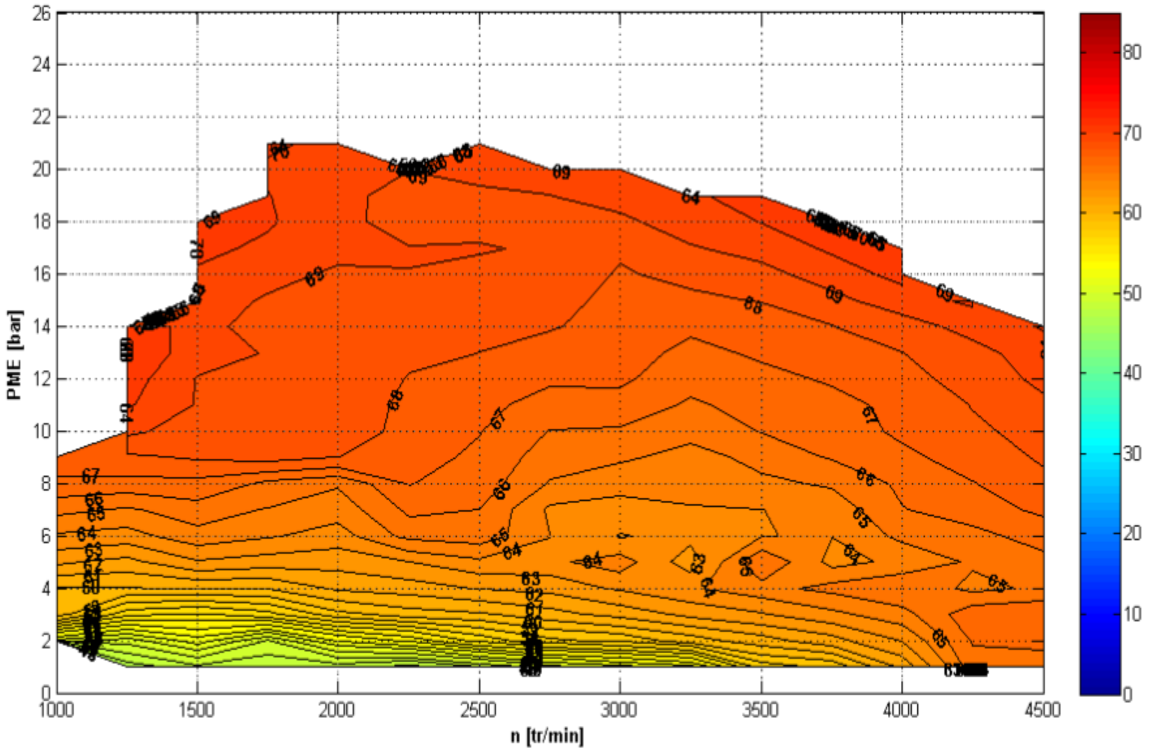

Fig. 15 Heat transfer area for preheated zone [%]

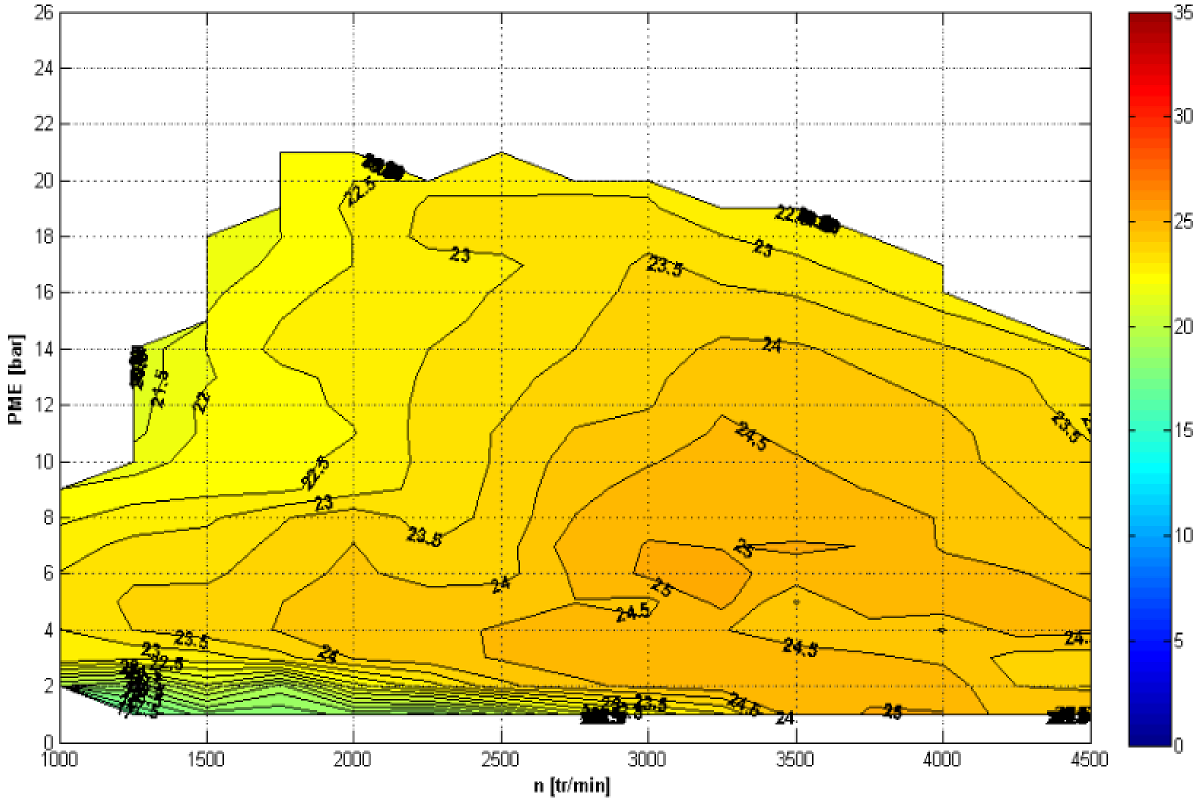

Fig. 16 Heat transfer area for two-phase zone [%]

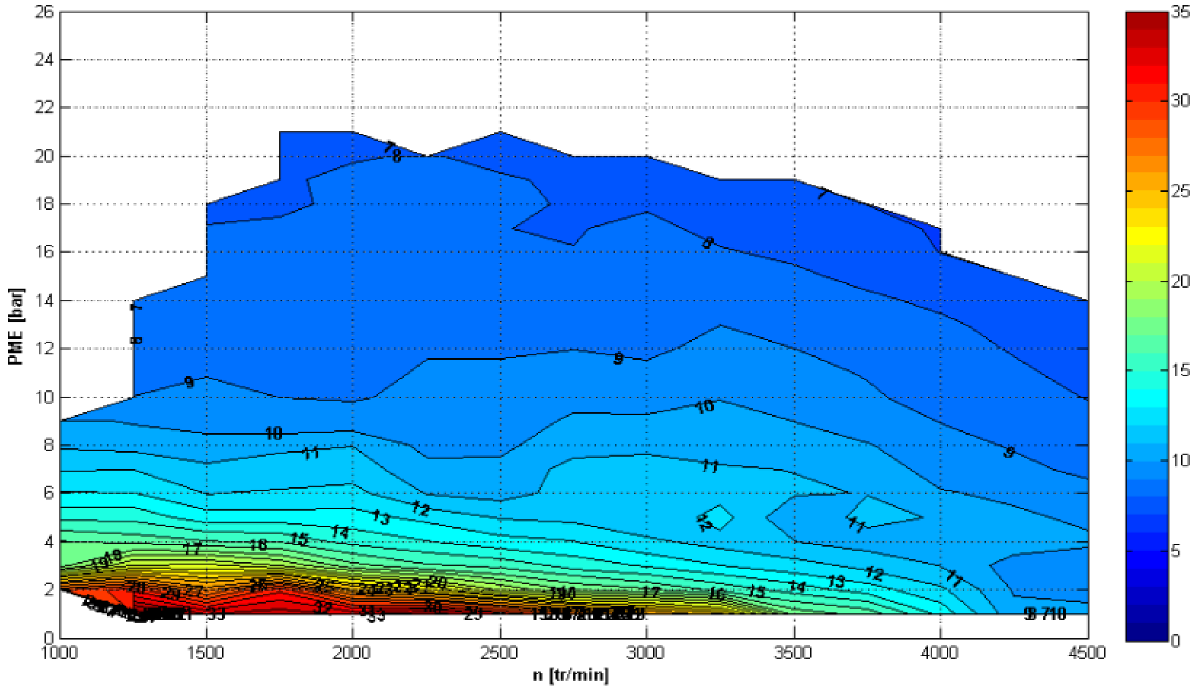

Fig. 17 Heat transfer area for superheated zone [%]

As a conclusion, the overall heat transfer rate increases with engine speed and engine load. Furthermore, the heat transfer rate of each zone is proportional to that of the overall heat transfer rate when the engine operating condition changes. The heat transfer area of the preheated zone is the largest, which is more than half of the total area. The heat transfer area of the two-phase zone is slightly greater than that of the superheated zone primarily caused by the discrepancies of the heat transfer rates.

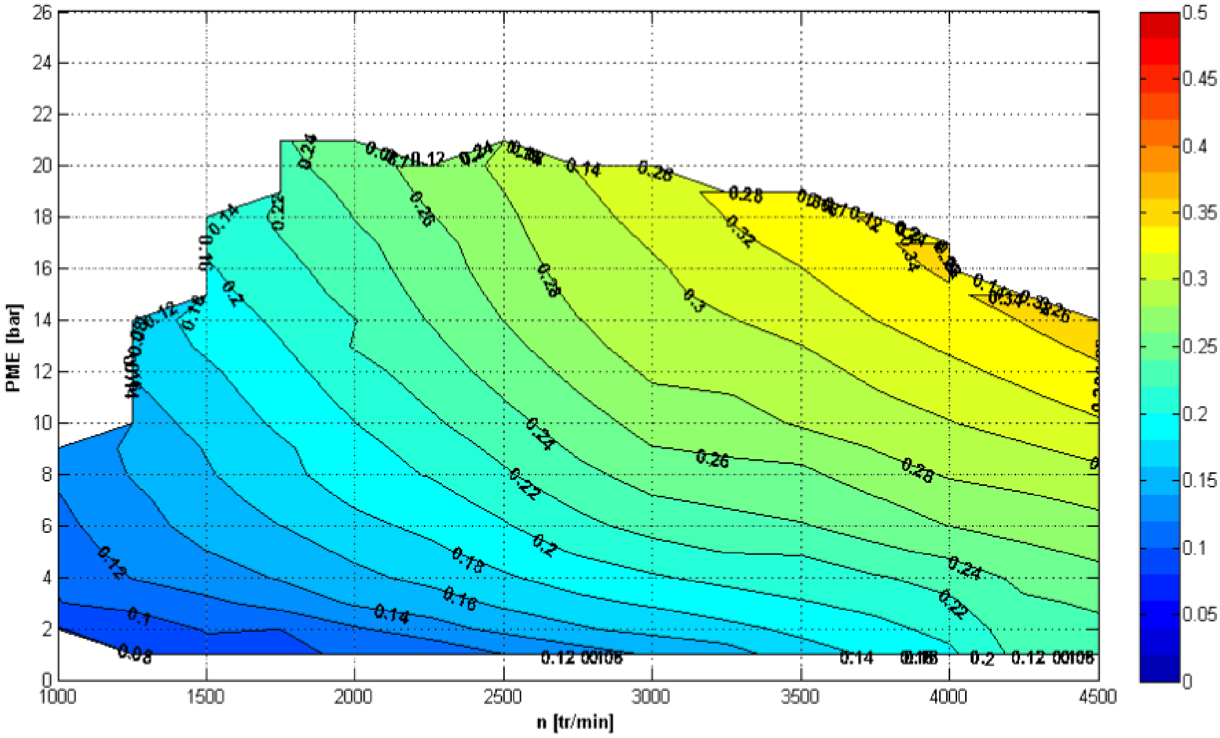

Fig. 18 Overall heat transfer area for evaporator [m2]

After evaluating the heat transfer properties, the performance of the ORC system was analyzed at each measured engine operating point using the established mathematical model.

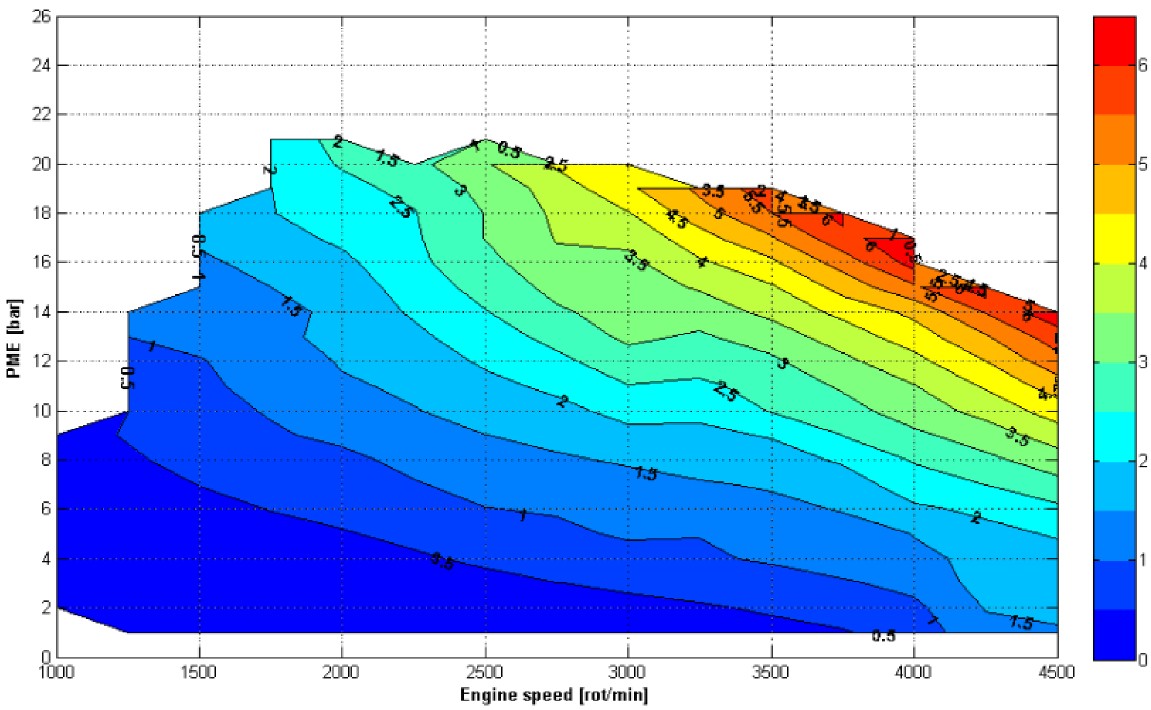

Fig. 19 Net power of ORC system variation over all engine operating regions

Figure 19 shows the variation of the overall net power output of the ORC system over the whole operating range. The net power outputs of the ORC system increase with the engine speed and engine load. At the engine rated condition, the overall net power output of the ORC system reaches the upper limit and is 6.3 kW.

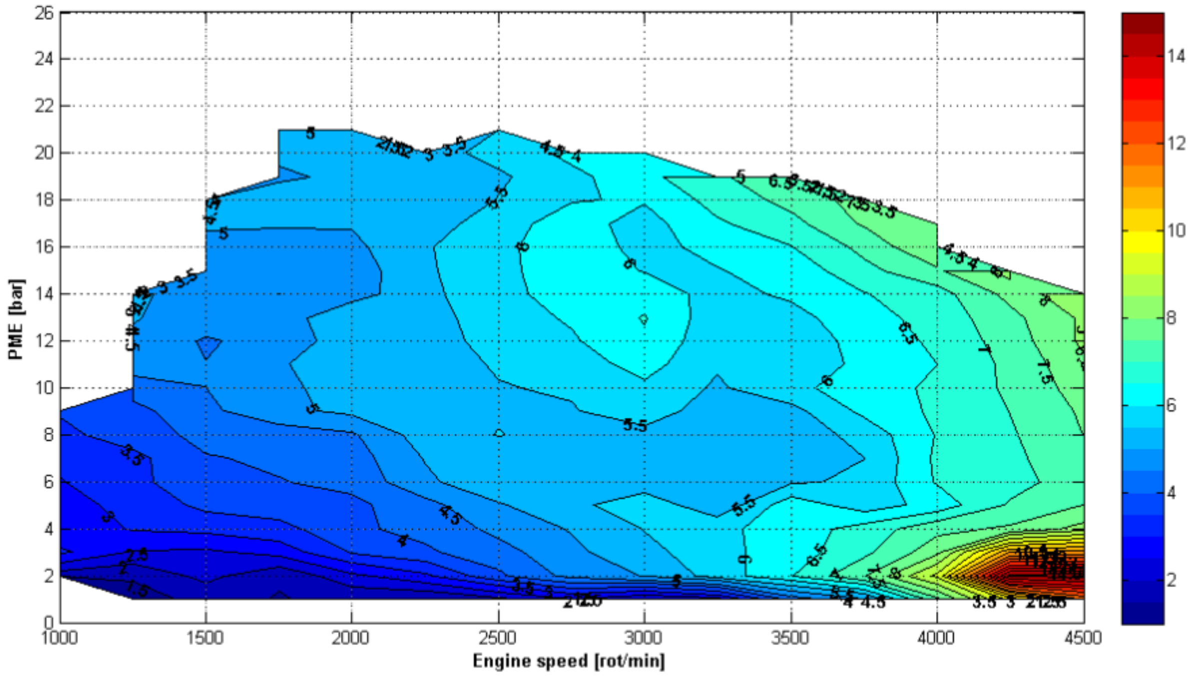

Fig. 20 Improvement of engine effective power over all engine operating regions [%]

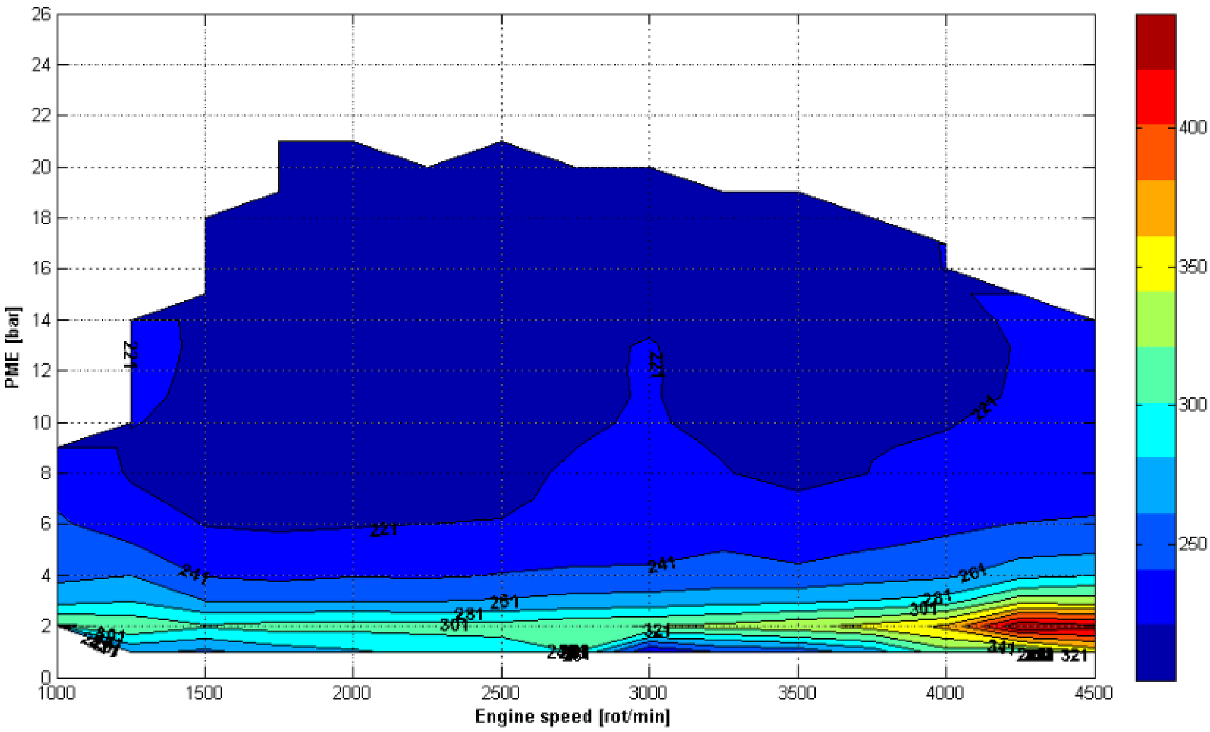

Fig. 21 BSFC of combined system over all engine operating regions [g/kWh]

The improvement in the effective power over the engine's entire working region is displayed in figure 20. In the high effective thermal efficiency region, the augmentation proportion is lowest (4–5%) because the waste heat quantity ratios are lower. The reason for this is better fuel combustion effects, the engine pumping losses are lower, and the ratio of the output power to the combustion energy is higher than in the other regions.

Figure 21 shows the variations of the BSFC of the diesel engine - ORC combined system. Compared with the diesel engine itself, the BSFC can be reduced by 5%. Therefore, the fuel economy of the diesel engine combined with the ORC system is effectively improved, see figure 22.

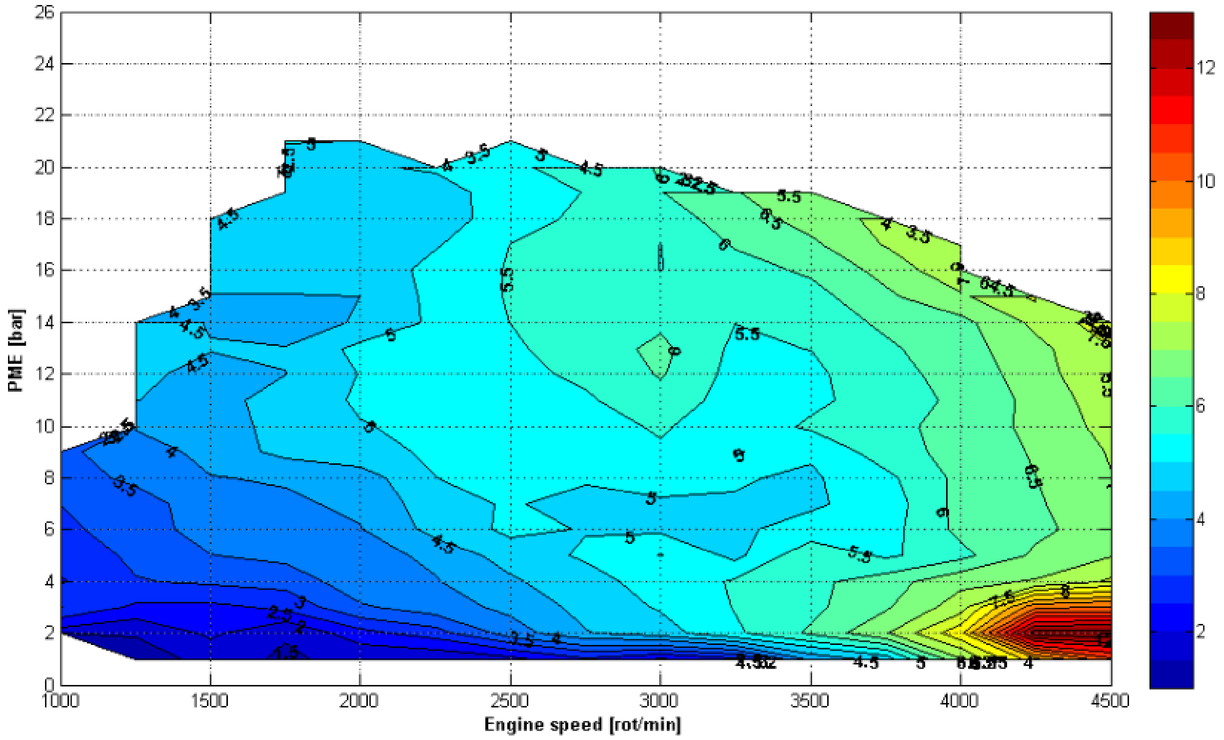

Fig. 22 Improvement of BSFC over all engine operating regions [%]

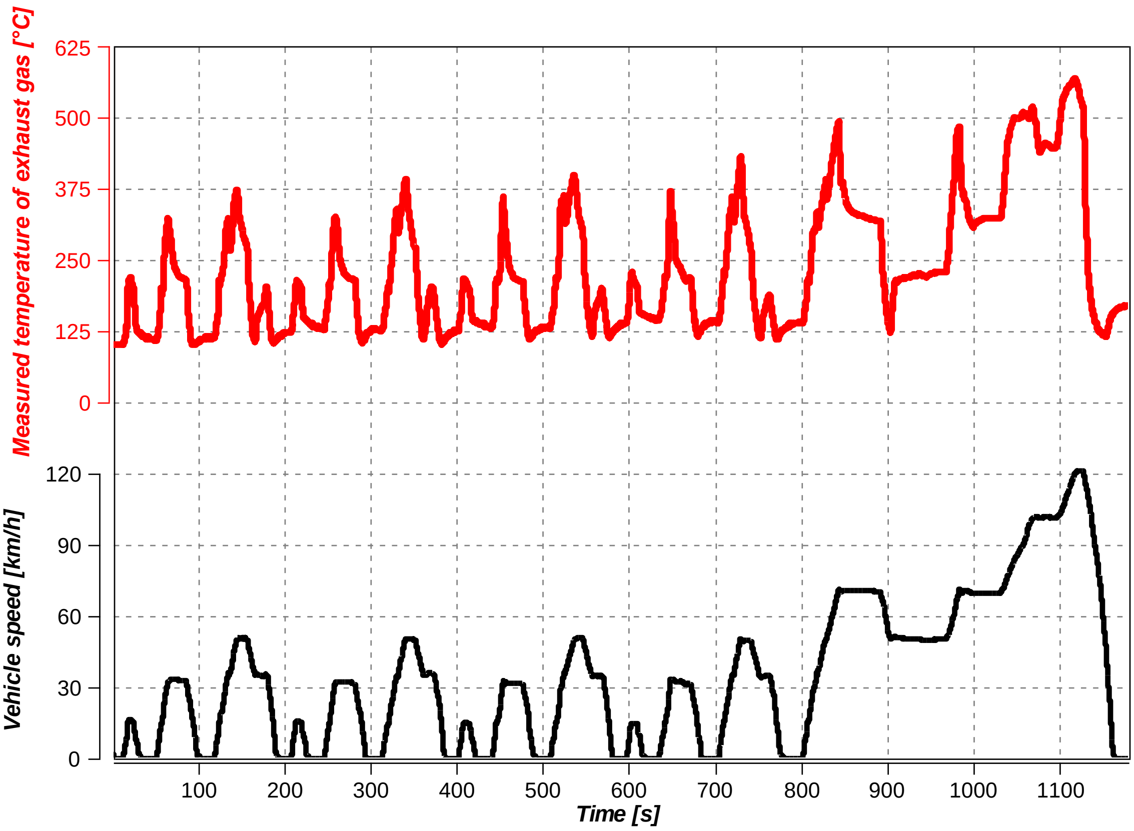

Fig. 23 Measured temperature of exhaust gas under NEDC

Before the research moves to another stage in the work, the experimental setup and all measurement devices of an organic Rankine cycle combined with an internal combustion engine carried out at the Thermal Research Center, Faculty of Mechanical and Mechatronics Engineering, Polytechnic University of Bucharest, are described in chapter 5. This chapter contains a number of tables and figures that describe the experimental setup. The experimental setup is a key element in fulfilling the objectives of the PhD Thesis.

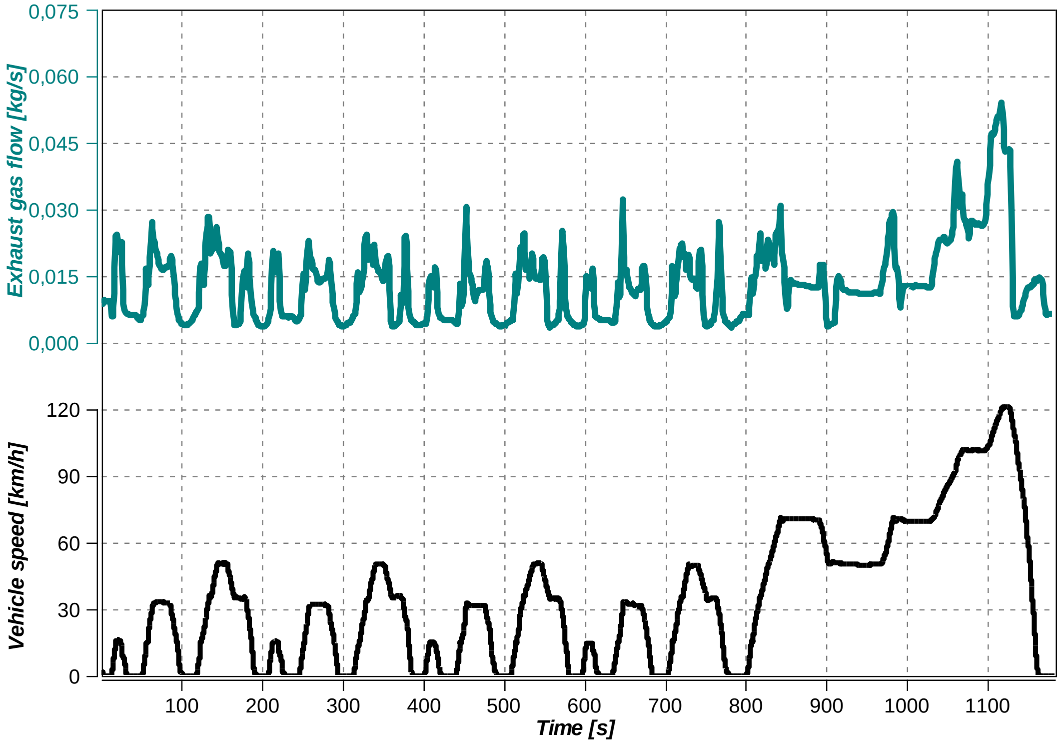

Fig. 24 Exhaust gas flow under NEDC

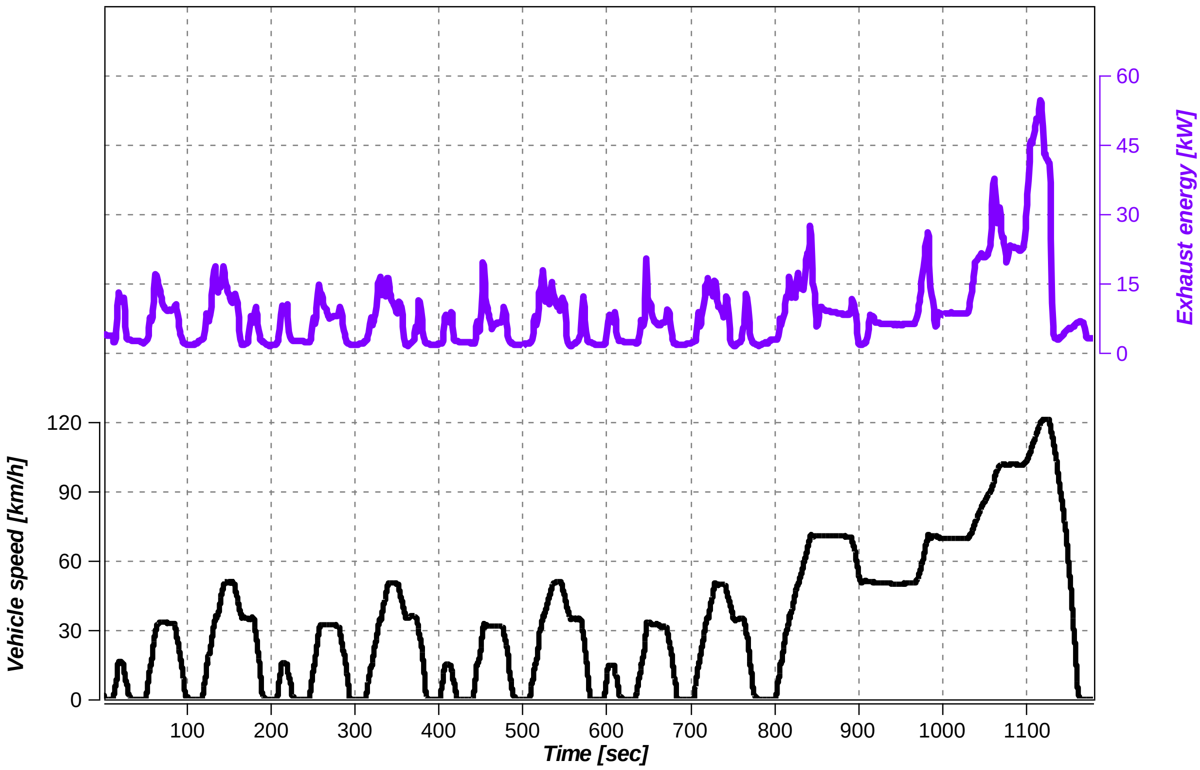

Fig. 25 Engine exhaust waste heat under NEDC

In next chapter the expected waste heat recovery at typical passenger car operation conditions is determined by weighting the waste heat recovery characteristics with the operation conditions of the New European Driving Cycle (NEDC).

As passenger cars are used in transient operation conditions, the variation of engine exhaust waste heat under NEDC cycle must also be taken into account, see figure 25.

Chassis dynamometer measurements were carried out in order to measure the exhaust gas mass flow rate and temperature for the NEDC operating conditions, figures 23 and 24.

Transient analysis is performed using the transient heat exchanger model for evaporator. The model is developed to dynamically predict the temperature of exhaust gases and working fluid at the inlet and outlet of each heat exchanger (preheater, boiler and superheater).

The next step was to calculate the thermal efficiency and the net power output of organic Rankine cycle for the operating points representing the extraurban part of NEDC. Under NEDC the waste heat recovery power output is varying in the range of 0.2 – 2.2 kW, and the thermal efficiency vary between 1 – 14 %, depending on engine load and engine speed, see figures 26 and 27.

For passenger cars that are used mostly inside of cities, and therefore operate mostly at idle speed and low part load conditions, the waste heat recovery benefit will be very low, but if they are used on countryside highways and motorways, and therefore operate mostly at high part load conditions or even full load conditions, the waste heat recovery benefit will be significant.

The ORC system has many effects on the vehicle, such as increased weight, increased engine back pressure, increased cooling demand of the vehicle, and utilization of the ORC power output. These effects should be considered in determining the ultimate feasibility of the system.

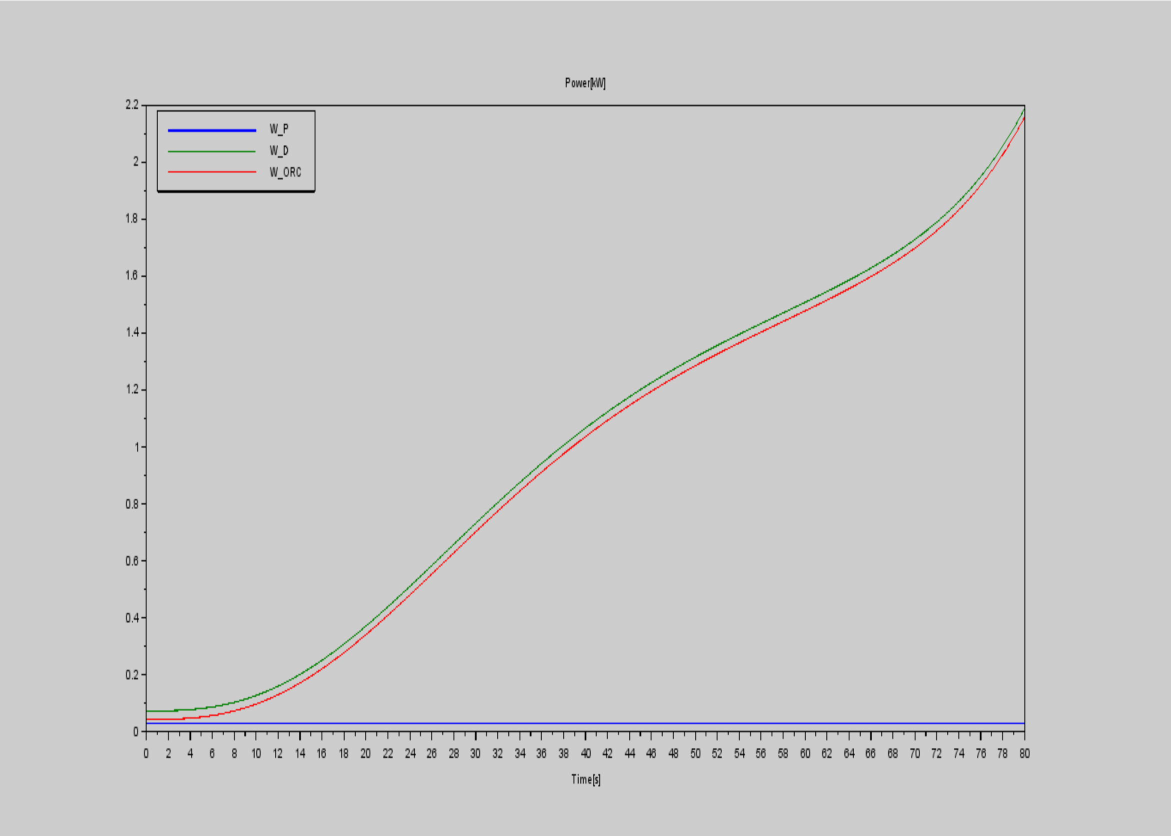

Fig. 26 ORC power under NEDC

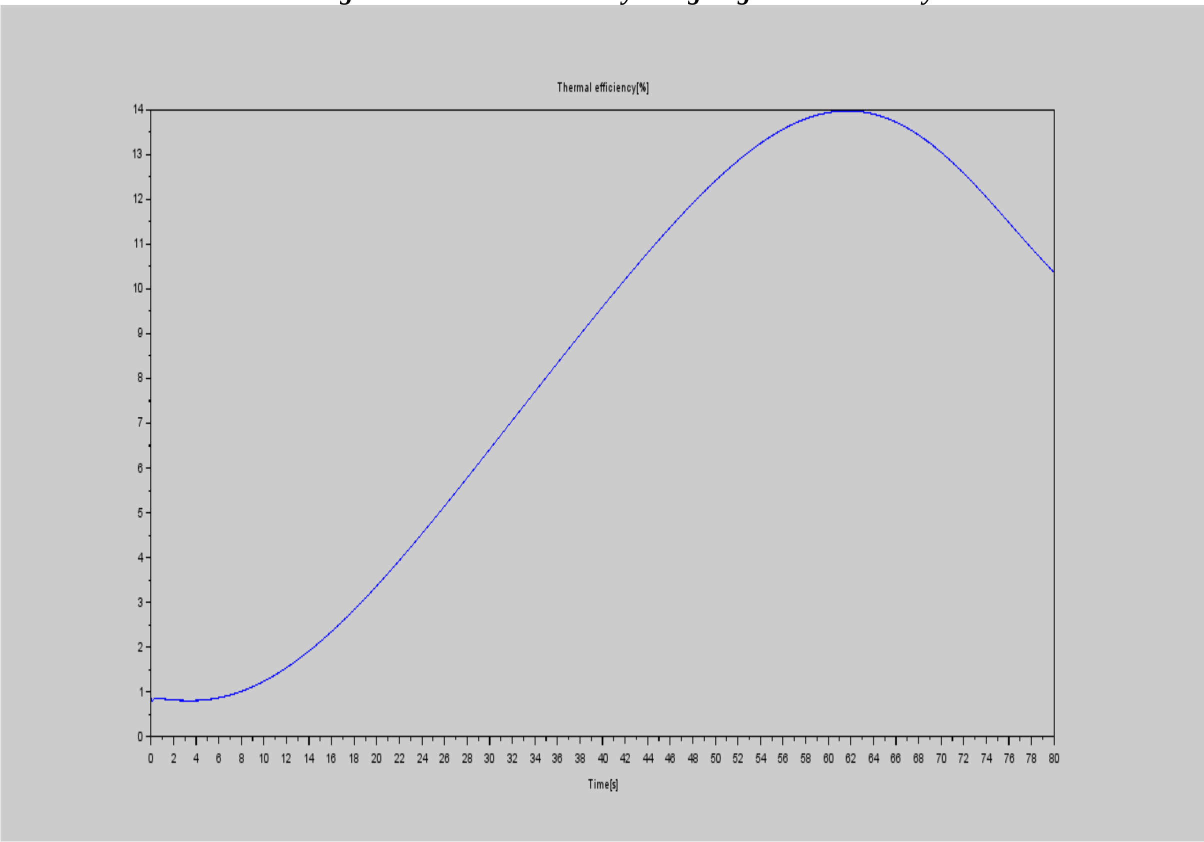

Fig. 27 ORC thermal efficiency under NEDC

Waste heat recovery is an option for passenger cars. However, performance of waste heat recovery depends strongly on operation conditions.

Future work should focus on designing, building up, and testing prototype applications to get comparisons between this study and reality.