COMPARATIVE ANALYSIS OF RANKINE CYCLE AND ORGANIC RANKINE CYCLE FOR WASTE HEAT RECOVERY IN INTERNAL COMBUSTION ENGINE

СРАВНИТЕЛЬНЫЙ АНАЛИЗ ЦИКЛА РЕНКИНА И ОРГАНИЧЕСКИЙ ЦИКЛ РЕНКИНА ДЛЯ УТИЛИЗАЦИИ ТЕПЛА В ДВИГАТЕЛЕ ВНУТРЕННЕГО СГОРАНИЯ

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Abstract: *This paper presents numerical analysis of waste heat recovery from engine exhaust gases by means of Rankine cycle and Organic Rankine cycle. Both technologies are widely studied in combustion engines but there are still not solid statements which should be chosen. The heat source in this study is the exhaust system of a modern diesel engine, developed for passenger car. Firstly, the engine was experimentally studied at stationary operating mode. Thus, exhaust gas parameters such as: mass flow rate, temperature and enthalpy were obtained at seventeen operating points which correspond to real operating mode of vehicle in NEDC. A simulation model of waste heat recovery system was developed. Based on that model, a numerical code was created in Python as CoolProp open-source platform was used to determine working fluid parameters. Lastly, Rankine cycle and Organic Rankine Cycle output power and efficiency were studied. The results revealed that Organic Rankine cycle using R245fa as working fluid provides better efficiency than steam Rankine cycle. Maximum recovered power was estimated to be 1.69kW while for the steam Rankine cycle it was 1.43kW.*

KEYWORDS*: WASTE HEAT RECOVERY, RANKINE CYCLE, ORGANIC RANKINE CYCLE, DIESEL ENGINE, EXHAUST GASES*

1. Introduction

A number of research [1-3] revealed that, overall efficiency of automobile engines still remains below 40%. Moreover, at partial load it ranges from 15% to 30%. Therefore, minimum 60% of fuel energy is lost as a form of heat in the exhaust and cooling system. Due to that fact, waste heat recovery (WHR) applied to automotive engines, seems to be a prospective way to increase the engine efficiency. Despite approximately the same energy of the exhaust gases and cooling system, WHR from exhaust gases is much more promising due to much higher exergy contain [1].

Rankine cycle (RC) and Organic Rankine cycle (ORC) can be successfully applied as WHR to engine exhaust system. Punov et al. [4], revealed higher potential of RC than ORC at engine operating point typical for tractors during plowing. In this case RC efficiency was observed to be 15.8%. Daccord et al. [5], reported RC efficiency of 10.3% and ORC efficiency of 6.3%. Glavatskaya et al. [6], studied RC with piston expander machine applied to automotive engine. In that study WHR efficiency was estimated to be within the range from 12% to 14%. Katsanos et al. [7] studied effect of RC and ORC on BSFC of heavy-duty diesel engine at partial load (25%). They reported improvement which ranges from 8.5% to 10.2% for ORC (using R245fa) and 6.1% to 7.5% when steam RC is applied. Among organic fluids R245fa is preferred due to quite enough performance and lower impact to the environment.

The aim of this paper is to study the performance of steam RC and ORC using R245fa for waste heat recovery on passenger car diesel engine at operating range typical for NEDC (New European Driving Cycle).

2. Engine experimental study

2.1. Engine test setup

A diesel engine for passenger car was studied at stationary operating mode in our laboratory at Technical university of Sofia. The test bed facility is shown in Figure 1. The engine is mechanically coupled to hydraulic brake D4. The maximum power absorption from the brake is 257kW at 4500rpm. A strain gauge sensor is used in order to measure the brake force. The sensor was produced at Technical university of Sofia for operating range from 0N to 1500N and accuracy of 0,5% at whole operating range.

Figure 1. Engine test bed facility

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. Boost pressure is limited to 1.3 bar. Common rail system of the engine is produced by Delphi. The high pressure pump is DFP 3.1 with integrated transfer pump and inlet metering valve. Solenoid controlled 6-holes injectors of generation DFI 1.3 are used into the system. Maximum injection pressure is 1600 bar. The engine is also equipped with water cooled EGR system and post treatment system including catalytic converter. The valves distribution system with four valves per cylinder is developed. The engine is compliant with EURO 4 emission demands. The main engine data are listed in Table 1.

Table 1

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

The exhaust gases parameters (temperature and mass flow rate) were measured. We estimated the exhaust mass flow by measuring the intake mass flow and fuel consumption. In order to measure the intake mass flow an intake mass flow meter produced by Bosch was used. It is a flow meter based on thermo anemometry principal. The accuracy is 2% over the whole operating range. The flow meter was previously calibrated in respect to a laboratory mass flow meter.

Fuel consumption was estimated measuring the volumetric flow while the density of the fuel was previously defined to be 0.840g/cm^3. It was used Rotronics RCC101 volumetric fuel flow measurement technic with two flow sensors – one in delivery line and one in return line.

The exhaust temperature was measured by means of K type thermocouple located at exhaust pipe 1.5m downstream the exhaust valves. We considered this location is suitable as inlet section of a Rankine cycle heat exchanger. Measurement range of the thermocouple is up to 1100°C. The probe diameter is 1.5mm. Mounting position of the sensor can be seen in Figure 2.

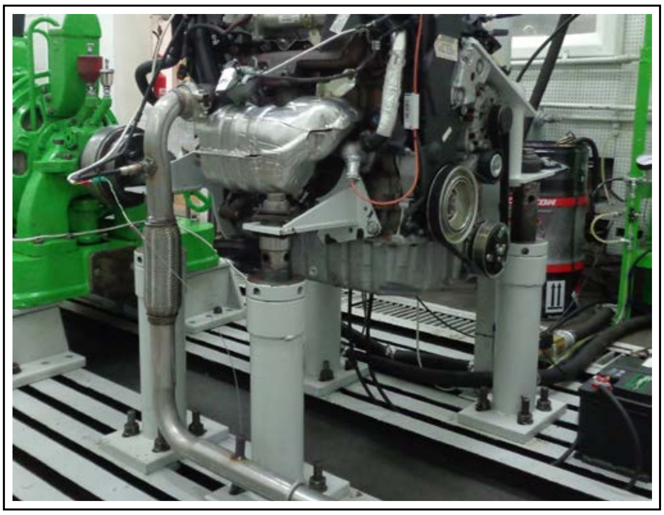

Figure 2. Exhaust temperature sensor position

In our experimental research the engine operation was controlled by means of Real Time controller produced by National Instruments and specialized modules for control and measurement produced by Drivven. It includes: NI Chassis PXI-1031, RT controller NI PXI-8106, FPGA NI PXI-7813R card, two expansion chassis CRIO 9151, two DI driver, AD combo, LS module and O_2 sensor module.

2.2. Engine test points determination

The engine performance has been previously studied numerically by means of 1D model in advanced simulation software AVL BOOST [1, 8]. In this study, it was interesting to determine the engine operating points which correspond to engine operating mode in NEDC test of a passenger car. In order to establish the engine operating points a driving model of a vehicle was used. This model is based on the force balance in longitudinal direction of the vehicle. In this case the traction force should be equal to the sum of the aerodynamic force, rolling resistance force and inertial force. The grade resistance force is zero. NEDC begins with cold start, then it follows four repetitions of UDC mode. It finishes with single repetition of EUDC. The NEDC for 6-speed manual transmission is presented in Figure 3.

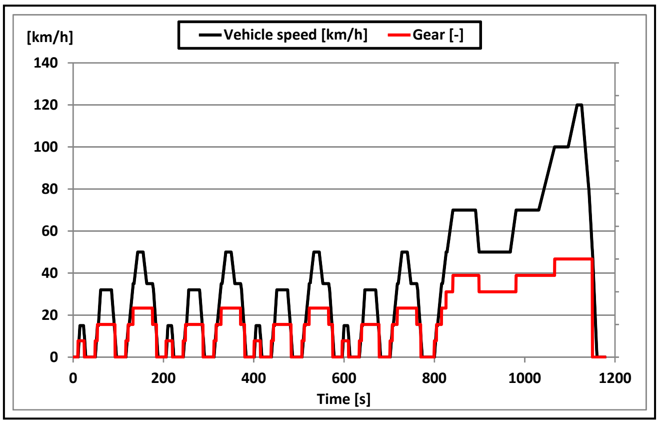

Figure 3. Vehicle speed in NEDC

It was observed seventeen typical engine operating points over the cycle simulation. The engine speed and BMEP for each point is listed in Table 2.

Table 2

№	Vehicle speed [km/h]	Gear [-]	Engine speed [rpm]	BMEP [bar]
1	15	2	950	0.35
2	16.1	2	1070	2.88
3	16.7	2	1100	3.82
4	50.8	4	1400	4.93
5	50	3	1420	1.27
6	35	3	1430	0.52
7	35.9	3	1470	4.03
8	70.4	5	1600	4.72
9	15	1	1830	0.13
10	49.1	3	1950	4.72
11	69.2	4	1980	6.25
12	100.5	6	1990	8.56
13	32	3	2020	1.01
14	33.9	2	2150	2.89
15	120	5	2280	5.97
16	119.5	6	2300	9.82
17	100	5	2310	4.02

2.3. Exhaust gas parameters

The experimental research was conducted at engine steady-state operating mode. Seventeen operating points were observed corresponding to idling, constant vehicle speed and acceleration. These operating points were defined based on vehicle simulation in NEDC in previous section. During the test the engine was heated up to normal cooling temperature. The ambient temperature was 18°C while the barometric pressure was 955mbar. Then, the values of intake mass flow, fuel flow and exhaust gases temperature were recorded. Exhaust gas enthalpy was calculated as constant value of specific heat capacity was used (c_p=1.15 kJ/kg.K). Exhaust mass flow varies within the range from 53,4kg/h to 188,7kg/h. The highest value was measured at engine operating point corresponds to vehicle speed of 100km/h at 6th gear.

Exhaust gas temperature depends on engine load. At those operating points where output power is high the temperature is much higher than other points. The maximum value was measured at the 16th operating points which corresponds to vehicle speed of 119.5km/h and BMEP = 9.82bar. In this transient operating mode of the vehicle, exhaust gas temperature accounts to 410.3°C. At the same operating point, it was observed the highest exhaust power. The exhaust enthalpy, calculated per time is 23.1kW (Figure 4).

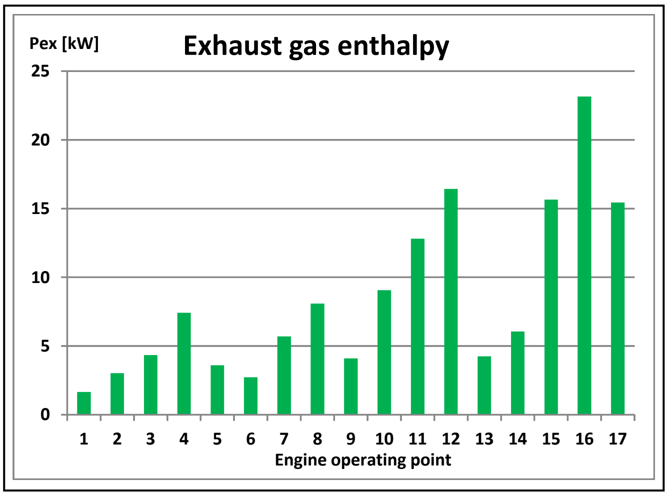

Figure 4. Variation of exhaust gas enthalpy

3. Rankine cycle simulation model

The RC system consists of a tank, pump, heat exchanger (evaporator), expander and condenser (Figure 5). The pump increases the fluid pressure during the phase ***a-b***. The fluid pressure downstream from the pump (point ***b***) depends on the fluid. The fluid is heated by exhaust gases (phase ***b-c***) on three different stages: preheating ($\boldsymbol{b}$-$\boldsymbol{b}^{I}$), evaporation at constant temperature ($\boldsymbol{b}^{I}$-$\boldsymbol{c}^{I}$) and superheating ($\boldsymbol{c}^{I}$-$\boldsymbol{c}$). The process ***c-d*** is the expansion of the super-heated vapor. Two types of expanders can be used, depending on the design: turbines and piston machines. The last stage of the cycle is the condensing process (***d-a***) in the condenser. At the condenser outlet section the fluid reaches the initial parameters. The variation in working fluid thermodynamic parameters is presented in T-s diagrams in Figure 6.

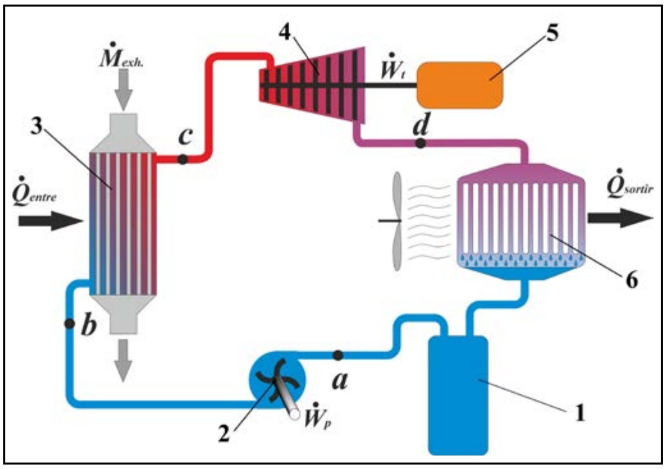

Figure 5. Rankine cycle scheme

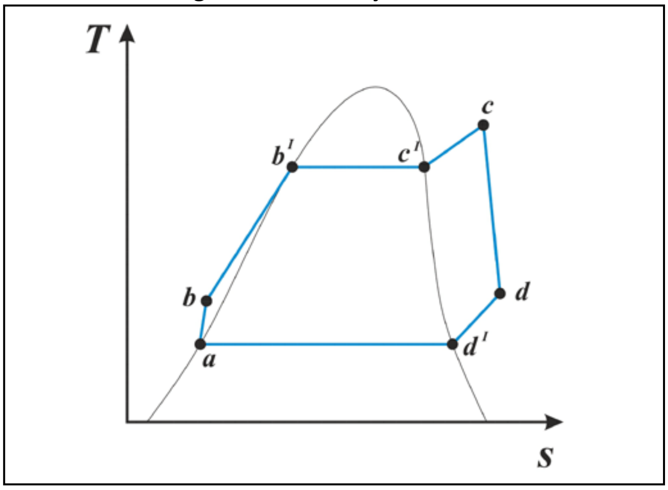

Figure 6. T-S diagram of super-heated RC and ORC

The power consumed by the pump was estimated as follows:

$$P_{pump} = \frac{\dot{m}_{wf}\left(h_{b^s} - h_a\right)}{\eta_{pump}} \tag{1}$$

Heating of the working fluid by exhaust gas occurs at constant pressure – process ***b-c***. At point **c** the fluid is in the form of super-heated vapour. The fluid specific enthalpy at the outlet of the heat exchanger was calculated by means of the following correlation:

$$h_c = h_b + \frac{Q_{he}}{\dot{m}_{wf}} \tag{2}$$

The heat flow rate transferred by the heat exchanger was estimated by means of a discretized heat exchanger model as:

$$Q_{he} = \sum_{i=1}^{n} Q_{he(i)} \tag{3}$$

where heat flow rate at finite volume was estimated as:

$$Q_{he(i)} = A_i.U_i.(T_{gas(i)} - T_{wf(i)}) \tag{4}$$

The output power of the expander is calculated as follows:

$$P_{ex} = \dot{m}_{wf}.(h_c - h_{d^s}).\eta_{ex} \tag{5}$$

Lastly, the power recovered by the RC can be estimated as the difference between the power produced by the expander and the power consumed by the pump:

$$P_{RC} = P_{ex} - P_{pump} \tag{6}$$

The RC efficiency is determined as the recovered power with respect to the heat flow transferred by the heat exchanger as follows:

$$\eta_{RC} = \frac{P_{RC}}{Q_{he}} \tag{7}$$

By means of the model presented above a simulation code was developed in Python. In order to determine working fluid parameters the open-source platform CoolProp was integrated in the code.

4. Results and discussion

Numerical simulation at the same engine operating points (same heat source) with RC and ORC was carried out. In other words with two working fluids: water and organic fluid R245fa. It was imposed some constraints for both cycles. The main parameters of the cycles are listed in Table 3.

Table 3

	RC	ORC
Working fluid	Water	R245fa
Pump pressure, p_b [bar]	5 ÷ 10	5 ÷ 15
Condensation pressure, p_d [bar]	1	
Condensation temperature, T_d [K]	373	310
Exchanger surface, A [m^2]	2.2	
Expander efficiency, η_{ex}	0.7	
Pump efficiency, η_{pump}	0.8	

By means of the numerical simulation RC and ORC output power was calculated. The comparative result between the cycles is presented in Figure 7. The results revealed that ORC power is higher than RC at all operating points. It can be explain with relatively low temperature and energy of exhaust gases. Moreover, the RC output power is zero at those operating points when exhaust gas temperature lower than 450K. Below this temperature very low working fluid mass flow is needed as well as extremely precise control. For that reason it was assumed that RC will be bypassed and no power will be produced. Due to the higher evaporation temperature of the water, the evaporation pressure was reduced to

5bar once the exhaust gas temperature was below 526K. Lower evaporation pressure also reduces RC output power.

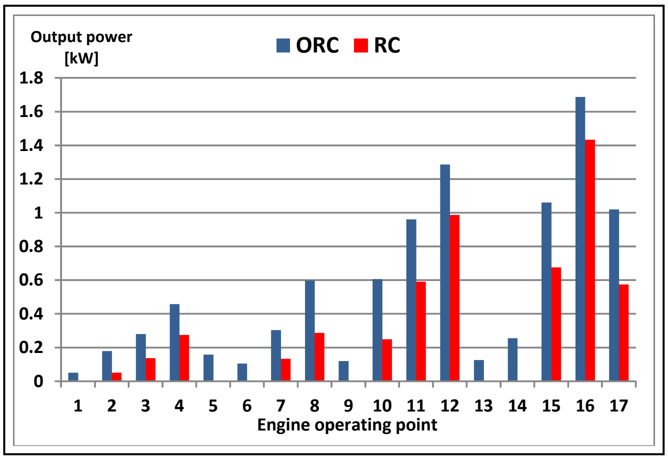

Figure 7. ORC and RC power as a function of engine operating points

The ORC output power varies within the range of 0.05kW to 1.69kW while the RC maximum power is 1.43kW. In percentage the differences decreases when exhaust gas temperature and energy increase. At the 16^{th} operating points (Exhaust gas enthalpy – 23.15kW) RC produces 15% lower output power. WHR efficiency was also studied for both cycles. The results are presented in Figure 8. For RC the maximum recovery efficiency is 3.71% while concerning ORC it accounts to 4.7%.

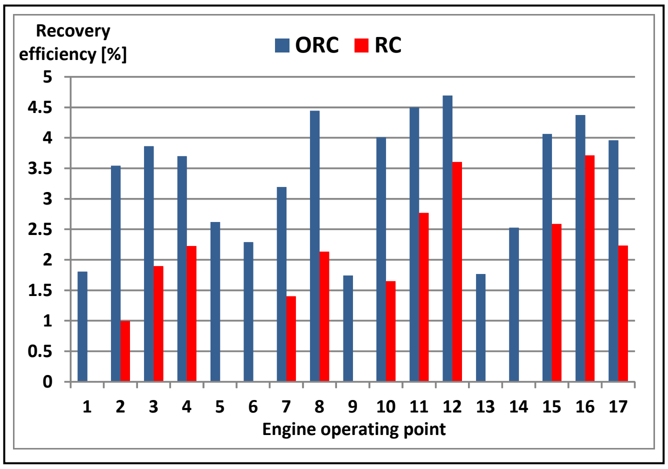

Figure 8. ORC and RC efficiency as a function of engine operating points

5. Conclusions

Exhaust gas parameters of a modern diesel engine for passenger car were measure by means of engine test in stationary operating mode. Studied engine operating points was previously defined by means of vehicle driving model. The vehicle simulation was conducted in NEDC. Thus, engine operating points and exhaust gas parameters respectively, were defined corresponding to NEDC homologation procedure. Seventeen points were studied. Over the test, exhaust mass flow and temperature was measured. It provides opportunity to determine exhaust gas energy. The results revealed that exhaust enthalpy, calculate per time, ranges from 1.66kW to 23.15kW. Exhaust gas mass flow is within the range of 53.39kg/h to 188.66kW while the temperature varies from 388.5K to 683.5K.

RC model was developed at steady-state operating mode. A discretized heat exchanger model was implemented as well. Based on the model a simulation code in Phyton was created. Working fluid parameters were defined by means of open-source platform CoolProp, implemented in Phyton. Therefore, WHR output power and recovery efficiency was estimated as some constrains were applied for both cycles. Two cycles were studied: steam RC and ORC working with R245fa.

In these conditions ORC revealed higher efficiency. Maximum recovered power was estimated to be 1.69kW while at same operating points RC power was 1.43kW. It presents 15% lower output power. In the other operating points RC power is much lower than ORC. It can be explained with low temperature, respectively low energy of exhaust gases over NEDC test procedure of the vehicle. Moreover, lower temperature causes reduction in evaporation pressure of the water. In some operating points when exhaust gas temperature is lower than 450K steam RC power is zero. Both ORC and RC efficiency is not high. The ORC efficiency ranged from 1.74% to 4.7% while RC efficiency is within the range of 0% to 3.71%.

Based on results obtained in this study it can be stated that ORC using R245fa is more efficient way to reduce engine fuel consumption in NEDC than steam RC. However, more complicate analysis should be carried out in the future due to the fact that NEDC is dynamic as well as there is heat accumulated by evaporator which should be taken into consideration in simulations.

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