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OPTIMISATION OF WASTE HEAT RECOVERY SYSTEM OPERATING PARAMETERS FOR DIESEL ENGINE BASED ON RANKINE CYCLE

NIKOLAY MILKOV
Departement of Combustion Engines, Automobiles and Transport,
Techmical University, Sofia, Bulgaria
nikolay.milkov@tu-sofia.bg

PLAMEN PUNOV
Departement of Combustion Engines, Automobiles and Transport,
Techmical University, Sofia, Bulgaria
plamen_punov@tu-sofia.bg

QUENTIN DANEL
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
quentin.danel@outlook.fr

CHRISTELLE PÉRILHON
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
christelle.perilhon@cnam.fr

PIERRE PODEVIN
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
pierre.podevin@cnam.fr

Abstract:

This article presents an optimization study of the Rankine cycle operating parameters as a function of diesel engine operating mode. The Rankine cycle here is studied as a waste heat recovery system which uses the engine exhaust gases as a heat source. The engine exhaust gases parameters (temperature, mass flow rate and composition) were defined on the engine test bed at constant engine speed and variable load. It was found that the exhaust gases temperature is within the range of 167°C to 557°C and the mass flow rate varies from 88,5kg/h to 281,3kg/h. An engine speed of 2000 rpm was chosen in this study due to the fact that this speed provides higher engine thermal efficiency. The Rankine cycle thermal power and efficiency was numerically estimated by means of a simulation code in Python(x,y). This code includes discretized heat exchanger model and simplified model of the pump and the expander based on their isentropic efficiency. The Rankine cycle working fluid mass flow rate was varied within the range of 1g/s to 10g/s and the working pressure was within the range of 5bars to 25 bars. The numerical results revealed that the cycle thermal power increases slightly as a function of engine load. Thus, the optimal operating parameters of the Rankine cycle depend on the heat source.

Keywords: *diesel engine, waste heat recovery, Rankine cycle, numerical and experimental simulation.*

1. Introduction

The future CO_2 emissions restriction imposed to the internal combustion engines is difficult to be met by means of available technologies. The waste heat recovery system based on Rankine cycle seems to be a promising method for increasing the engines thermal efficiency.

The engine waste heat recovery system based on Rankine cycle was recently largely studied [1-11]. In a previous study of our team [1] the parametric analysis of the Rankine cycle based waste heat recovery from exhaust gases of a diesel engine was conducted. The results in this study revealed the improving in the engine output power from 1,7% to 3% in the studied engine operating points. The estimated Rankine cycle output power was ranged from 0,25 kW to 1,0kW. In [2], the effect of waste heat recovery system by means of Organic Rankine cycle (ORC) on the passenger car engine fuel consumption was studied. This study revealed reduction in fuel consumption by 5,96%. Punov et al. [3] presented a study on the impact of a waste heat recovery system based on Rankine cycle on the fuel consumption of an diesel engine for passenger car. The results revealed lower fuel consumption up to 7,57%.

Domingues at al. [4] studied different types of heat exchangers. For an ideal heat exchanger the simulations revealed higher engine thermal efficiency from 1,4% to 3,52%, while for a shell and tube heat exchanger, the simulations showed an increase of 0,85% to 1,2% on the thermal efficiency. The work of Peris at al. [5] is focused on waste heat recovery from the engine cooling system. Results show that the Double Regenerative ORC using SES36 has the maximum net efficiency of 7,15%, increasing the engine thermal efficiency by up to 5,3%. In the paper reported by Zhu at al. [6], a theoretical study of a bottoming Rankine cycle for engine waste heat recovery is conducted from energy and exergy point of view. The results revealed that the global recovery efficiency does not exceed 14% under typical operating conditions.

Wang et al. [12] reported a dual-loop ORC optimization. Their study represented that ORC output power increased with increasing the evaporation pressure.

Thus, the aim of this study is a parametric analysis of the steam Rankine cycle implemented as waste heat recovery technic in a diesel engine exhaust system at various engine operating points.

2. Experimental and numerical approach

This study was conducted in two parts - experimental and numerical. For our numerical study a simulation code was developed in Python(x,y) while the experimental study was conducted on our engine test bed at Department of combustion engines, automobiles and transport in Technical university of Sofia.

2.1 Experimental equipment

The engine under study is a 2.0 l four-cylinder high-pressure direct injection (HDI) diesel engine developed by PSA Peugeot Citroen. The maximum output power is 101kW at 4000rpm and the maximum torque is 320Nm at 2000rpm. The engine is equipped with a variable geometry turbocharger. The boost pressure is limited to 1.3bar. The Common rail system of the engine is produced by Delphi. The maximum injection pressure is 1600bar. The engine is also equipped with an exhaust gas recirculation (EGR) system and post treatment system including a catalytic converter and diesel particulate filter (DPF). The cylinder is equipped with four valves per cylinder. The main geometrical parameters of the engine are listed in Table 1.

Table 1

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

The experimental study was conducted at Department of combustion engines, automobiles and transport. The test facility includes an engine test bed equipped with hydraulic brake, flexible diesel engine management system and data acquisition system for data analysis. The engine test bed is shown in Fig. 1.

The engine is mechanically coupled to hydraulic brake D4. The maximum power absorption from the brake is 257 kW at 4500 rpm. A strain gauge sensor is

Figure 1. Engine test bed

used in order to measure the brake force. The sensor was produced at Technical university of Sofia for operating range from 0 to 1500 N. Two mechanically controlled valves are used to control the brake.

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. A Lab View project was developed in order to control the engine operation. The entire system provides large functionality such as: injection process control with up to five separate injections per cycle, injection pressure control, boost pressure control, exhaust gas recirculation (EGR) control etc. The front panel of the project host application and the system hardware are shown in Fig. 2.

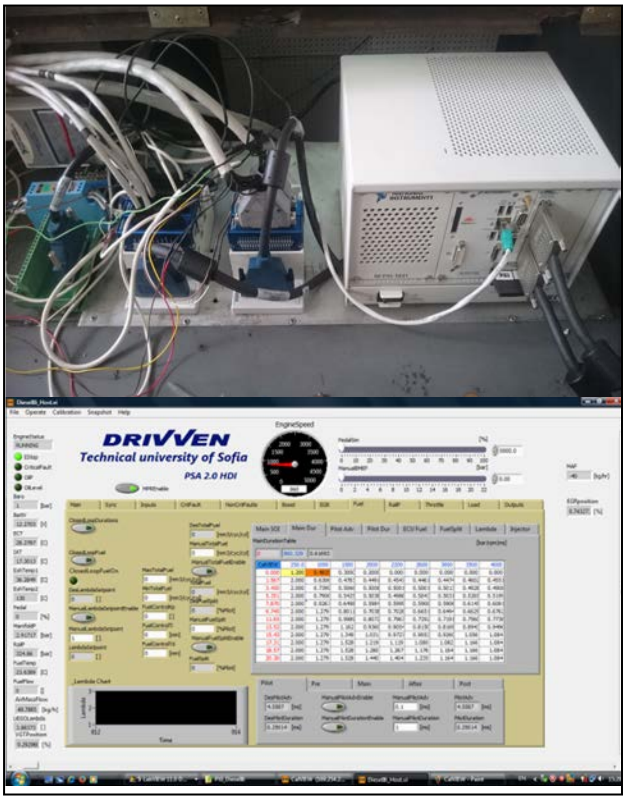

Figure 2. The front panel of the project host application and the system hardware

2.2 Numerical model

A 0D physical model of the Rankine cycle in steady-state operating mode was developed assuming some simplification on the pump and expander machines. Then, the Rankine cycle model was transformed into an simulation code in Python (x,y) [13]. Thermo-physical properties of the working fluid were defined by means of database CoolProp integrated to Python(x,y). The Rankine cycle consists of following elements: working fluid, pump, heat exchanger (evaporator), expander and condenser (Fig. 3).

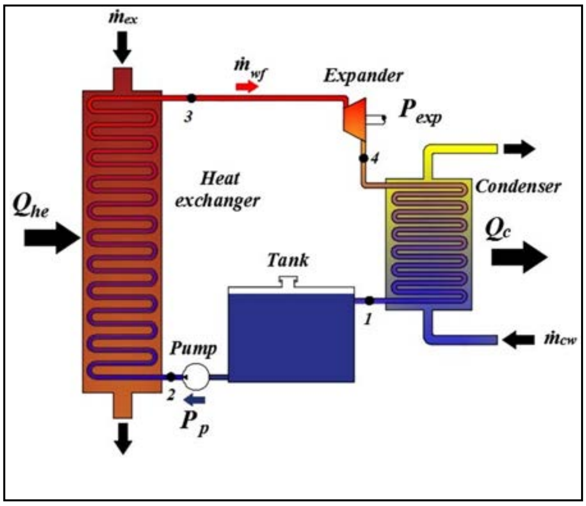

Figure 3. Model of the Rankine cycle

The pump increases the fluid pressure (process *1-2*) at relatively constant temperature. Then, the pump power consumption can be estimated as follows:

$$P_p = \frac{m_{wf}.(h_{2is} - h_1)}{\eta_p} \quad (1)$$

where m_{wf} – working fluid mass flow rate [kg/s]; h_1,h_{2is} – working fluid specific enthalpies at point 1 and 2, respectively [kJ/kg]; η_p – pump isentropic efficiency.

In order to improve the accuracy of the simulation a discrete heat exchanger model was developed Fig. 4.

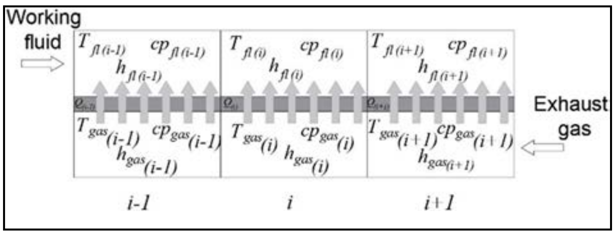

Figure 4. Discrete model of the evaporator

The evaporator heat flow rate was calculated taking into account the temperature of fluids, heat transfer surface and heat transfer coefficient as follows:

$$Q_{he_{[i]}} = A_{[i]}.K_{[i]}.(T_{gas_{[i]}} - T_{wf_{[i]}}) \quad (2)$$

where A – heat transfer surface[m^2]; K – total heat transfer coefficient [W/(m^2.K)]; T_{gas}, T_{wf} – temperature of exhaust gases and working fluid [K], *i*- number of the heat exchanger cell.

Overall heat transfer coefficient was estimated by means of following equation:

$$\frac{1}{K} = \frac{1}{\alpha_1} + \frac{(d_2 - d_1)}{2.k} + \frac{1}{\alpha_2} \quad (3)$$

where α_1 - heat transfer coefficient of the working fluid; α_2 - heat transfer coefficient of the exhaust gas; k – thermal conductivity of the wall material; d_1, d_2 - internal and external diameter of the tube [mm].

Heat transfer coefficient in respect to working fluid was estimated using correlations for single and two-phase flows. In case of single phase fluid Dittus et Boelter equation [14] is used:

$$\alpha_1 = \frac{0{,}023.Re^{0{,}8}Pr^{0{,}4}.\lambda}{D} \quad (4)$$

where λ - thermal conductivity [W/(m.K)]; D - characteristic distance for fluid flow (in our case $D = d_1$).

Within two-phase zone the Kenning-Cooper correlation [15] is used:

$$\alpha_1 = \frac{[1+1{,}8.X^{-0{,}87}].0{,}023.Re^{0{,}8}Pr^{0{,}4}.\lambda}{D} \quad (5)$$

where X is known as the Martinelli factor [14], as given in the following equation:

$$X = \left(\frac{1-x}{x}\right)^{0{,}9}.\left(\frac{\rho_v}{\rho_l}\right)^{0{,}5}.\left(\frac{\nu_l}{\nu_v}\right)^{0{,}1} \quad (6)$$

where x - the vapor quantity; ρ_v, ρ_l- the vapor and liquid densities [kg/m^3]; ν_v, ν_l - the vapor and liquid viscosities [Pa.s].

Heat transfer coefficient in respect to exhaust gases was estimated by means of single phase correlation (4), using the values for the thermal conductivity, characteristic distance, Reynolds and Prandtl number correspond to exhaust gases.

The overall heat flow from exhaust gas to the working fluid was estimated as follows:

$$Q_{he} = \sum_{i=1}^{n} Q_{he_{[i]}} \quad (7)$$

The efficiency of the evaporator can be estimated by means of following correlation:

$$\eta_{HE} = \frac{m_{wf}.(h_3 - h_2)}{m_g.c_{pg}(T_{ge} - T_0)}.100 \quad (8)$$

where: T_{ge} – inlet temperature of the exhaust gases [K]; T_0 – ambient temperature [K]; m_g – mass flow rate of the exhaust gas [g/s]; c_{pg} – specific heat capacity at constant pressure of the gas [J/(kg.K)].

The power produced at the expansion machine (process *3-4*) is calculated as follows:

$$P_{exp} = m_{wf}.(h_3 - h_{4is}).\eta_{exp} \tag{9}$$

where h_{4is} – specific enthalpy of the working fluid after an isentropic expansion [kJ/kg]; η_{exp}- expander isentropic efficiency.

The process of condensation in the condenser occurs at constant pressure. Then, the heat transfer rate can be calculated as:

$$Q_c = m_{wf}(h_4 - h_1) \tag{10}$$

$$h_4 = h_3 - \eta_{exp}(h_3 - h_{4is}) \tag{11}$$

Finally, the recovered power and the efficiency of the Rankine cycle were estimated by following correlations:

$$P_{RC} = P_{exp} - P_p \tag{12}$$

$$\eta_{RC} = \frac{P_{RC}}{Q_{he}} \tag{13}$$

The validation of the Rankine cycle model developed in Python (x,y) was presented in a previous study [13]. In that study the results which were presented show maximum deviation of 6,4% in relative values between numerical and experimental results. The small deviation obtained in the model calibration allowed us to use the simulation model in our numerical study.

3. Engine experimental test

The engine test was conducted in order to obtain the experimental data for the engine fuel consumption,

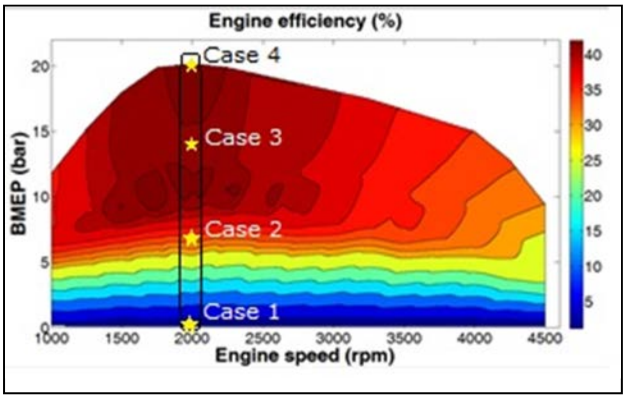

Figure 5. Engine thermal efficiency and studied area.

air mass flow, air-excess ratio, temperature and mass flow of the exhaust gases. An engine speed of 2000 rpm was chosen in this study due to the fact that this speed corresponds to maximum engine torque and maximum engine thermal efficiency (Fig. 5). The experimental results are present in Figures 6 and 7.

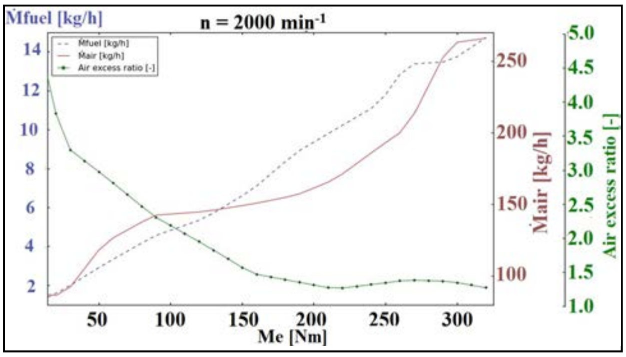

Figure 6. Fuel consumption, air excess ratio and air mass flow.

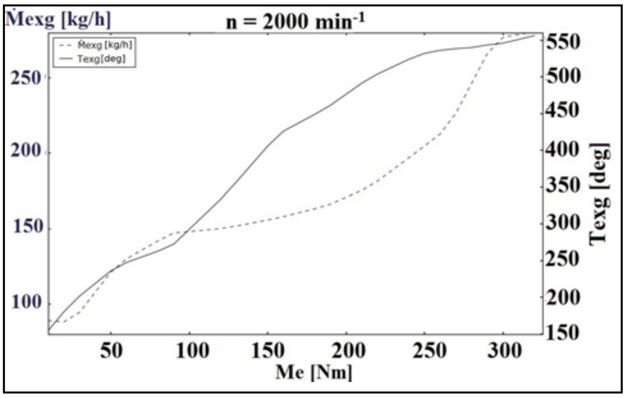

Figure 7. Exhaust gases temperature and mass flow.

It was found that the engine fuel consumption varies from 1,41kg/h to 14,87kg/h, the engine air mass flow is in the range of 85,9 kg/h to 266,4 kg/h and the air-excess ratio varies from 4,27 to 1,23.

The temperature of the exhaust gases was measured 1,5m downstream the engine exhausts ports. We considered this section of the exhaust system suitable for the Rankine cycle heat exchanger. The temperature of the exhaust gases is in range of 167,3°C to 557,5°C and the exhaust mass flow was calculated to be in the range of 87,6kg/h to 281,3kg/h.

The parametric study of the Rankine cycle was conducted in four cases using the measured exhaust gases parameters. The values of the exhaust gases parameters are presented in Table 2.

Table 2

	n	Me	Tex	$\dot{M}ex$
	[rpm]	[N.m]	°C	[kg/h]
Case 1	2000	24.9	195	87.6
Case 2	2000	123.1	340	150.4
Case 3	2000	245.7	529.8	201.4
Case 4	2000	322.4	557.5	281.3

The first and the fourth cases are chosen in order to study the Rankine cycle operation at maximum and minimum thermal energy of the exhaust gases while the other cases are chosen at partial engine load in order to observe the whole engine map at 2000 rpm.

4. Rankine cycle optimization

The Rankine cycle numerical study was conducted using water as working fluid. In this model the parameters of the working fluid are calculated by means of CoolProp database. The isentropic efficiency of the expander machine was considered to be 60%.

The numerical study was performed by variation of the mass flow rate and pressure of the working fluid at each case. Here, the pressure values between 5 bars and 25 bars are used while the mass flow rate was varied from 1 g/s to 10 g/s. The results of this study are presented in Figures 8 to 11.

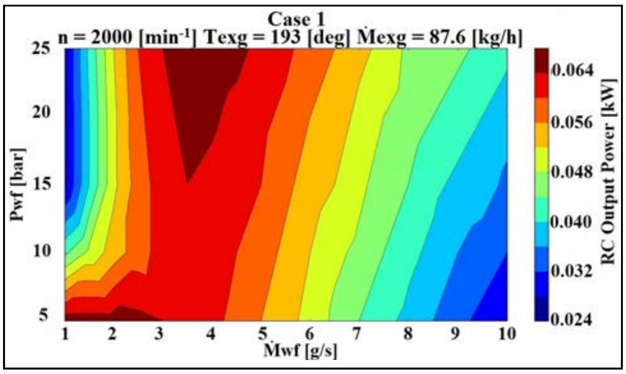

Figure 8. Rankine cycle output power map

In case 1 (Fig. 8), the estimated Rankine cycle output power is in the range of 0,003kW to 0,066kW. In this case the output power is close zero, because the temperature and mass flow rate of the exhaust gases are low. Normally, in this case it is better to use the organic fluids which provide higher efficincy in low temperatures at high pressure. Regarding to the engine output power in this case the Rankine cycle leads to maximum improvement by 0,5%.

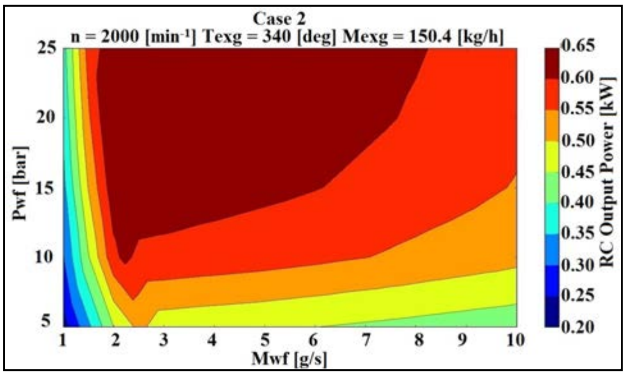

Figure 9. Rankine cycle output power map

In case 2 (Fig. 9), the Rankine cycle output power is in the range of 0,23kW to 0,64kW. In this case the highest Rankine cycle efficiency was observed in the area with the working fluid mass flow variation from 1 g/s to 5,5g/s and the pressure higher than 10 bars. Here, the maximum engine efficiency improvement by 1,98% can be reached.

In case 3 (Fig. 10), the Rankine cycle output power was observed to be in the range of 0,3kW to 2,2kW. Here, the highest output power was estimated when the system pressure is higher than 15 bars and mass flow rate ranged from 4,5g/s to 10g/s. Regarding to the engine output power in this case the Rankine cycle leads to maximum improvement by 4,76%.

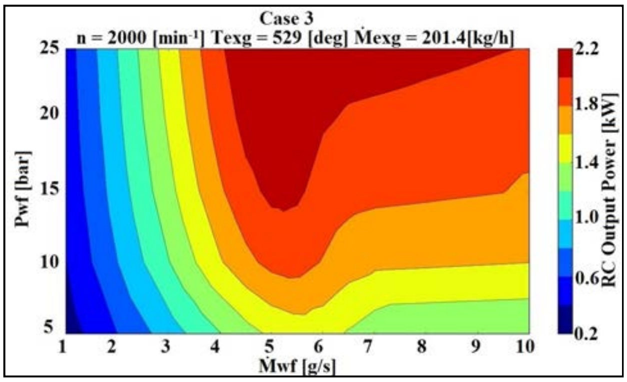

Figure 10. Rankine cycle output power map

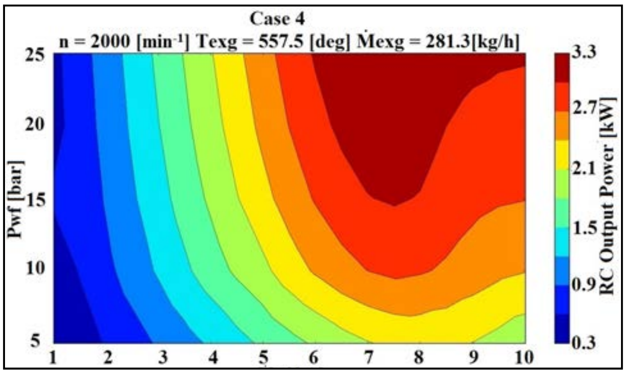

Figure 11. Rankine cycle output power map

In case 4, the Rankine cycle output power is in range of 0,31kW to 3,26 kW (Fig. 11). In this case the highest thermal energy in the exhaust system provides the highest Rankine cycle output power in our study. The maximum engine efficiency improvement by 4,97% can be reached in this case.

In order to summarize the results, the Rankine cycle output power depends on the thermal energy content at the exhaust gases (related to engine operating point) as well as the pressure of the working fluid and the working fluid mass flow rate. The steam Rankine cycle studied here as a waste heat recovery system from the exhaust gases of a modern diesel engine reveals promising results in the engine operating range with high exhaust gases thermal energy. However, in the operating area with low exhaust energy the Rankine cycle cannot improve the engine efficiency significantly.

5. Conclusions

This article presents an optimization study of the Rankine cycle operating parameters as a function of diesel engine operating mode. The chosen engine speed was 2000 rpm while the load varies from idle to maximum torque. That operating mode provides highest engine thermal efficiency and it is commonly used in the studied diesel engine designed for light-duty vehicle.

On the bases of the results it can be concluded:

- The working fluid parameters (pressure, and mass flow rate) of the Rankine cycle have to be chosen precisely depending on the exhaust gases parameters in order to optimize the efficiency of the cycle;
- The water as working fluid at the Rankine cycle provides the best efficiency of the cycle in cases of high value of the exhaust gases thermal energy. In case of low exhaust gases thermal energy the steam Rankine cycle efficiency is not significant;
- The maximum Rankine cycle output power for each case studied here is 0,066kW; 0,66kW; 2,2kW and 3,26kW. Taking into account the engine output power it leads to engine efficiency improvement by 0,5%; 1,98%; 4,76% and 4,97%, respectively.

Our future research will be focused on the impact of waste heat recovery system based on Rankine cycle over the engine efficiency improvement over the whole engine operating map. It includes the Rankine cycle parameters optimization as a function of exhaust gases parameters.

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References

[1] N. Milkov, P. Punov, Q. Danel, C. Périlhon, P. Podevin, Etude parametrique d'un cycle de Rankine pour la recuperation d'énergie des gaz d'échappement d'un moteur d'automobile, Scientific Conference COFRET'16, Bucarest, Romania, 2016.

[2] P.Punov, N. Milkov, Q. Danel, C. Périlhon, A Study of Waste Heat Recovery Impact on a Passenger Car Fuel Consumption in New European Driving Cycle, Proceedings of the European Automotive Congress EAEC-ESFA 2015, Bucarest, Romania, Springer, 2015, 151-163, DOI 10.1007/978-3-319-27276-4_14.

[3] P.Punov, T. Evtimov, N. Milkov, G. Descombes, P. Podevin, Impact of Rankine cycle WHR on passenger car engine fuel consumption under various operating conditions, Proceedings of ECOS 2015 - the 28th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems June 30-July 3, 2015, Pau, France.

[4] A. Domingues, H. Santos, M. Costa, Analysis of vehicle exhaust waste heat recovery potential using a Rankine cycle, Energy, 2012; 49: 71-85.

[5] B. Peris, J. Navarro-Esbrí, F. Molés, Bottoming organic Rankine cycle configurations to increase Internal Combustion Engines power output from cooling water waste heat recovery, Applied Thermal Engineering, 2013; 61: 364 – 371.

[6] S. Zhu, K. Deng, S. Qu, Energy and exergy analyses of a bottoming Rankine cycle for engine exhaust heat recovery, Energy, 2013; 58: 448 – 457.

[7] F. Yang, H. Zhang, C. Bei, S. Song, E. Wang, Parametric optimization and performance analysis of ORC (organic Rankine cycle) for diesel engine waste heat recovery with a fin-and-tube evaporator, Energy, 2015; 91: 128 – 141.

[8] Y. Kim, D. Shin, C. Kim, G. Cho Single-loop organic Rankine cycles for engine waste heat recovery using both low- and high-temperature heat sources, Energy, 2016; 96: 482 – 494.

[9] B. Kölsch, J. Radulovic, Utilisation of diesel engine waste heat by Organic Rankine Cycle, Applied Thermal Engineering, 2015; 78: 437 – 448.

[10] D. Di Battista, M. Mauriello, R. Cipollone, Waste heat recovery of an ORC-based power unit in a turbocharged diesel engine propelling a light duty vehicle, Applied Energy, 2015; 152: 109 – 120.

[11] S. Amicabile, J. Lee, D. Kum, A comprehensive design methodology of organic Rankine cycles for the waste heat recovery of automotive heavy-duty diesel engines, Applied Thermal Engineering, 2015; 87: 574 – 585.

[12] Wang, E.H., Zhang, H.G., Fan, B.Y., Ouyang, M.G., Yang, F.Y., Yang, K., Wang, Z., Zhang, J., and Yang, F.B.: 'Parametric analysis of a dual-loop ORC system for waste heat recovery of a diesel engine', Applied Thermal Engineering, 2014, 67, (1–2), 168-178

[13] N. Milkov, Q. Danel, P. Punov, T. Evtimov, C. Périlhon, P. Podevin, Numerical and experimental study of heat exchanger designed for waste heat recovery system from exhaust gases based on Rankine cycle, Presented at Scientific Conference BulTrans-2015, Sozopol, Bulgaria, 2015; 7: 97 - 102, ISSN 1313 - 955X.

[14] C. Herer, D. Gallori, Thermohydraulique des réacteurs à eau sous pression, Techniques de l'Ingénieur, traité Génie nucléaire, 2000.

[15] L. Sun, K. Mishima, An evaluation of prediction methods for saturated flow boiling heat transfer in mini-channels, Sol Energy, 2009; 52: 5323–5329.

ОПТИМИЗИРАНЕ НА РАБОТНИТЕ ПАРАМЕТРИ НА ЦИКЪЛ НА РАНКИН ЗА РЕКУПЕРИРАНЕ НА ЕНЕРГИЯ В ДИЗЕЛОВ ДВИГАТЕЛ

НИКОЛАЙ МИЛКОВ
Departement of Combustion Engines, Automobiles and Transport,
Techmical University, Sofia, Bulgaria
nikolay.milkov@tu-sofia.bg

ПЛАМЕН ПУНОВ
Departement of Combustion Engines, Automobiles and Transport,
Techmical University, Sofia, Bulgaria
plamen_punov@tu-sofia.bg

КУЕНТИН ДАНЕЛ
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
quentin.danel@outlook.fr

КРИСТЕЛ ПЕРИЛЬОН
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
christelle.perilhon@cnam.fr

ПИЕРЕ ПОДВАН
Laboratoire de Chimie Moléculaire, Génie des Procédés Chimiques
et Énergetiques CMGPCE, Cnam, Paris, France
pierre.podevin@cnam.fr

Резюме:

В статията е представено оптимизиране на работните параметри на цикъл на Ранкин във функция от режима на работа на дизелов двигател. Цикълът на Ранкин е изследван като система за рекупериране на топлинна енергия от отработилите газове на двигателя. Параметрите на отработилите газове (температура, дебит и състав) са определени експериментално при постоянна честота на въртене и променливо натоварване (т.е. по товарна характеристика). Измерени са температурата на отработилите газове, която е в диапазона от 167°C to 557°C и техния дебит, който варира от 88,5 кг/ч до 281,3 кг/ч. Честота на въртене от 2000 мин$^{-1}$ е избрана поради факта, че при нея двигателя работи с максимален к.п.д.. Изходящата мощност и ефективността на цикъла на Ранкин е определена числено чрез математичен модел в програмата Python(x,y). Този модел включва дискретизиран модел на топлообменника (изпарител) и опростен модел на помпата и разширителната машина като е използвана тяхната ефективност. Дебита на работния флуид в цикъла на Ранкин е променян в границите от 1г/с до 10г/с, а работно налягане от 5 до 25 бара. Числените резултати показват, че мощността на цикъла се увеличава постоянно във функция от натоварването на двигателя. Следователно, оптималните работни параметри на цикъла на Ранкин зависят от източника на топлина.

Ключови думи: *дизелов двигател, рекупериране на енергия, цикъл на Ранкин, числено и експериментално изследване.*