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NUMERICAL AND EXPERIMENTAL STUDY OF HEAT EXCHANGER DESIGNED FOR WASTE HEAT RECOVERY SYSTEM FROM EXHAUST GASES BASED ON RANKINE CYCLE

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Abstract:

This article presents a numerical and experimental study of the heat exchanger impact on the overall efficiency of waste heat recovery system based on Rankine cycle. An 0D physical model of the heat exchanger was developed as well as a simulation code in Python(x,y). For experimental study a heat exchanger prototype was constructed. The working fluid circulates inside twenty three helical tubes situated along the length of the evaporator. The heat exchanger is countercurrent with heat exchange surface about 1.8 m^2. Both numerical and experimental results revealed that the evaporator is one of the most important elements of Rankine cycle. The highest overall efficiency of Rankine cycle was observed to be 5.14% and 4.81%. These values were obtained by numerical and experimental study, respectively. On the bases of this study we can conclude that in order to increase the efficiency of the cycle it is needed to improve the heat exchanger design as heat exchanger coefficient has to be increased.

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

1. Introduction

Waste heat recovery by means of Rankine Cycle is a promising approach for achieving significant reductions in fuel consumption and emissions CO_2 of vehicles. This approach is already well established in industrial applications such as gas and steam power plants.

The overall efficiency of modern internal combustion engines is less than 40% [1], only in special engines such as a high boosted diesel engine for stationary application the efficiency can reach 45%.

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

Punov et al. [4] revealed in his study conducted on a tractor engine that not more than 15% of fuel energy can be converted into mechanical work by means of closed loop thermodynamic cycle and the exergy is the highest at high load and high speed and slightly decreases with load and speed.

A number of studies [5-8] revealed that Rankine cycle provides higher potential of waste heat recovery from exhaust gases than cooling system.

Study of the effect of heat exchanger on the Rankine cycle efficiency is the topic of a number of research. Wang et al. [9] studied waste heat recovery potential using Organic Rankine cycle with Multi-coil helical heat exchanger. This study reported that the total fuel saving could be up to 34%. In another study, Zhang et al.[10] presented heat transfer analysis in a finned-tube heat exchanger for engine waste heat recovery. The study revealed that the overall heat transfer rate in the evaporator increases as a function of engine output power.

On the base of these researches the aim of the article is to study the heat exchanger impact on the global efficiency of waste heat recovery system based on Rankine cycle.

2. Description of numerical and experimental approach

This study was conducted in two parts - numerical and experimental. For the numerical study a simulation code was developed in Python(x,y). In order to obtain experimental results a Rankine cycle test equipment was built as a countercurrent flow heat exchanger was used.

2.1 Numerical model

Firstly, a 0D physical model of the Rankine cycle steady-state operation 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) [11]. 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 - Figure1.

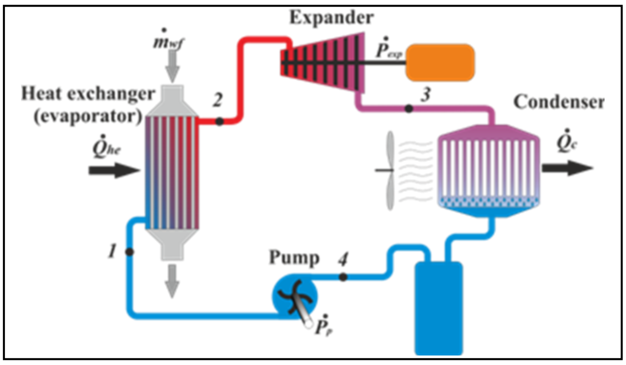

Figure 1. Model of the Rankine cycle

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 enthalpies at point 1 and 2, respectively [kJ/kg]; η_p – pump isentropic efficiency.

The evaporation of the working fluid by exhaust gases (process *2-3*) occurs at constant pressure. The heat transfer rate from the exhaust gases to working fluid in heat exchanger is given by:

$$Q_{he} = m_{wf}.(h_3 - h_2) \quad (2)$$

where h_3 – specific enthalpy of the working fluid at the end of evaporation [kJ/kg].

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

$$P_{exp} = m_{wf}.(h_3 - h_{4is}).\eta_{exp} \quad (3)$$

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 same as the evaporation occurs at constant pressure. Then, the heat transfer rate can be calculated as:

$$Q_c = m_{wf}(h_4 - h_1) \quad (4)$$

Finally, the recovered power and the efficiency of the Rankine cycle can be estimated by following equations:

$$P_{RC} = P_{exp} - P_p \quad (5)$$

$$\eta_{RC} = \frac{P_{RC}}{Q_{he}} \quad (6)$$

For the purpose of this study the determination of heat exchange efficiency in the evaporator is essential for evaluation the heat exchanger impact on the overall efficiency. In the heat exchanger the working fluid can be in three different states, liquid, bi-phasic and vapor. In this case the working fluid heat capacity is not constant and we cannot use the classical heat exchanger relation with the means logarithmic temperature difference. We used a discrete heat exchanger model composed by final number of cells. At each cell we assumed that heat exchange occurs at constant temperature difference. In order to determine the evaporator efficiency heat flow rate was calculated taking into account the temperature of fluids, heat transfer surface and heat transfer coefficient as follows:

$$Q_{he_{[i]}} = A_{[i]}.K_{[i]}.(T_{gas_{[i]}} - T_{wf_{[i]}}) \quad (7)$$

$$Q_{he} = \sum_{i=1}^{n} Q_{he_{[i]}} \quad (8)$$

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

Overall heat transfer coefficient can be estimate by means of following equation:

$$\frac{1}{K} = \frac{1}{\alpha_1} + \frac{(d_2-d_1)}{2.k} + \frac{1}{\alpha_2} \quad (9)$$

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 working fluid flow 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 [12] is used:

$$\alpha_1 = \frac{0{,}023.Re^{0{,}8}Pr^{0{,}4}.\lambda}{D} \quad (10)$$

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 [13] is used:

$$\alpha_1 = \frac{[1+1{,}8.X^{-0{,}87}].0{,}023.Re^{0{,}8}Pr^{0{,}4}.\lambda}{D} \quad (11)$$

where X is known as the Martinelli factor [12], as given in the following equation:

$$X = \left(\frac{1-x}{x}\right)^{0{,}9}.\left(\frac{\rho_v}{\rho_l}\right)^{0{,}5}.\left(\frac{\nu_l}{\nu_v}\right)^{0{,}1} \quad (12)$$

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

Heat transfer coefficient in respect to exhaust gases can be estimated by means of single phase correlation (9), but with the values of thermal conductivity, characteristic distance, Reynolds and Prandtl number correspond to exhaust gases.

$$\alpha_2 = \frac{0{,}023.Re^{0{,}8}Pr^{0{,}4}.\lambda}{D} \quad (13)$$

2.2 Experimental setup

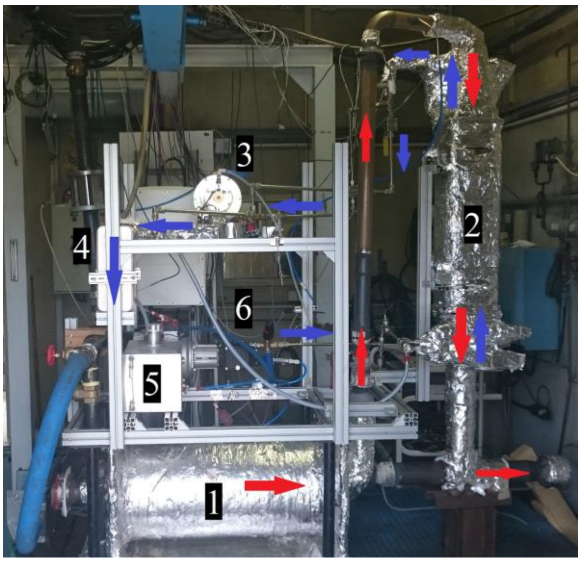

Figure 2. Rankine cycle test bench
1 – Burner; 2 – Heat exchanger (evaporator); 3 – Expander (turbine type); 4 – Cooler (condenser); 5 – Reservoir; 6 – Pump

In order to conduct experimental research a test bench of Rankine cycle waste heat recovery system was constructed by means of following components: Burner; Heat exchanger (evaporator); Expander (turbine type); Cooler (condenser); Reservoir; Pump (Figure 2).

All of the components are connected by means of stainless steel tubes. The test equipment consists of two fluid paths: the first is exhaust gases flow which comes from the burner and passé though the heat exchanger while the second is working fluid flow through the pump, heat exchanger, expander, cooler and reservoir. Water was used as working fluid in this cycle; however the test equipment can work with different fluids such as ethanol et al. The working fluid was cooling as cold water with variable mass flow rate circulates through the condenser.

The purpose of the burner is to produce exhaust gases with the same parameters of that produced by a common rail diesel engine for a tractor with maximum output power of 110 kW. The burner operates with diesel fuel while the fresh air comes from a compressor. The enthalpy of exhaust gases can be varied by adjustment of fuel and air mass flow in the burner. In order to simulate various combination of temperature and mass flow rate a part of fresh air passes outside the burner. Fresh air mass flow is setting by a vane, the main part of this flow goes to the burner and a second part is mix with burned gas to set the temperature. The fuel is injected in the burner by a pump and an internal by-pass is used to set the fuel mass flow.

The experimental setup is equipped with ten thermo-couples type K for measuring the temperature of the working fluid and exhaust gases, four pressure sensors for acquiring the pressure drop in gas flow and working fluid through the heat exchanger as well as the pressure in the condenser, four mass flow sensors are used for measuring the mass flow rate of working fluid, cooling water as well as of the air and fuel through the burner. In order to collect the experimental data a National Instruments DAQ model 6218 and a LabView based program were used.

For experimental setup a countercurrent flow heat exchanger was produced. The design of the exchanger is shown in the Figure 3.

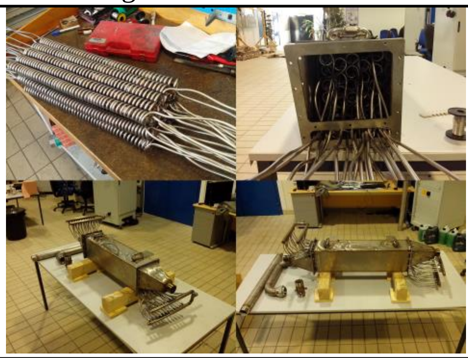

Figure 3. Heat exchanger

The working fluid circulates inside the helical tubes situated along the length of the evaporator. The number of tubes is 23 which provides the heat exchange surface approximately 1.8 m^2. In order to increase the heat transfer efficiency countercurrent flow of fluids is used.

3. Results and discussion

In order to study the heat exchanger efficiency as well as the waste heat recovery overall efficiency of the Rankine cycle numerical and experimental results were obtained with exhaust gases parameters correspond to seven different operating points of the internal combustion engine. These operating points were defined by combination of three different mass flow rates and five different temperatures of exhaust gases.

The Rankine cycle was operated with steam temperature around 220°C and pressure about 5 bars. These values were provided by adjusting the pump speed and the vane cross section area.

The overall waste heat recovery efficiency can be explained with following equation:

$$\eta_G = \frac{P_m}{Q_{gas}}.100 \quad (14)$$

The heat exchanger efficiency can be explained with following equation:

$$\eta_{HE} = \frac{m_{wf}.h_3}{Q_{gas}}.100 \quad (15)$$

where P_m – output power of the cycle [kW]; Q_{gas} – thermal power of the gases at the inlet section of the heat exchanger [kW]

The output power was calculated by means of isentropic expander efficiency as it was assumed that the working fluid is expanded to the atmospheric pressure.

The results obtained by numerical simulation are presented in table 1 and Figures 4 and 5.

Table 1

Exhaust gas mass flow [g/s]	Exhaust gas temperature [°C]	Heat exchanger efficiency [%]	WHR overall efficiency [%]
60	350	34,40	3,08
60	450	39,55	4,36
80	300	30,97	2,69
80	400	36,41	4,00
80	500	41,82	5,14
100	350	32,47	3,90
100	450	36,38	4,81

Numerical results revealed that the heat exchanger efficiency much highly depends on the temperature of exhaust gases than mass flow rate. The maximum heat

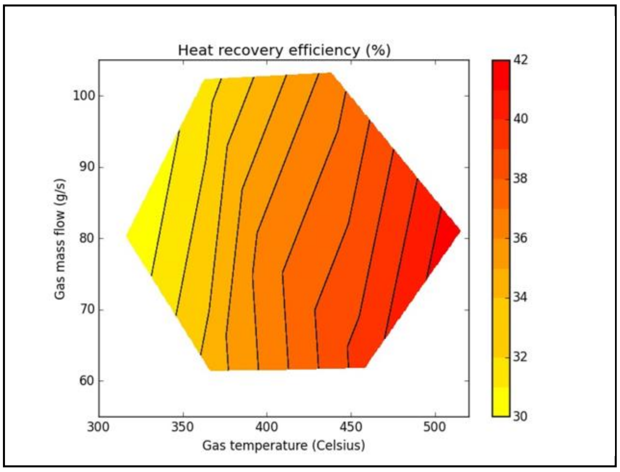

Figure 4. Simulated heat exchanger efficiency.

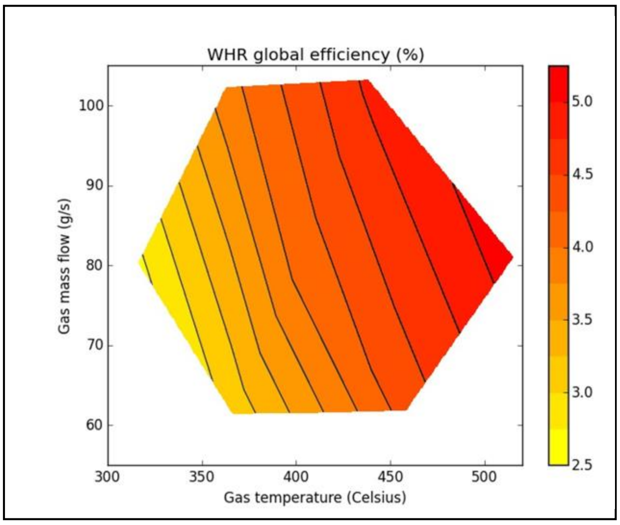

Figure 5. Simulated overall waste heat recovery efficiency

exchanger efficiency is 41.82 % at mass flow rate and temperature of the exhaust gases 80[g/s] and 500 °C, respectively.

Overall Rankine cycle efficiency was estimated to be 5.14 %.

The results from experimental study are presented in table 2 and Figures 6 and 7.

Table 2

Exhaust gas mass flow [g/s]	Exhaust gas temperature [°C]	Heat exchanger efficiency [%]	WHR overall efficiency [%]
60	350	34,69	3,13
60	450	39,64	4,39
80	300	30,64	2,67
80	400	37,56	4,28
80	500	40,69	4,81
100	350	31,65	3,74
100	450	35,95	4,75

Experimental results revealed mostly the same distribution of the heat exchanger efficiency and overall waste heat recovery efficiency. The maximum values are 40.69 % and 4.81%, respectively. These values were obtained at the same engine operating point as simulation results.

The maximum deviation between numerical and experimental data is 6.4% at the maximum overall efficiency operating point.

These results reveal that the prototype of the heat exchanger is not the best construction because the heat exchanger efficiency is not too high. However, the simulation model of the Rankine cycle is quite accurate.

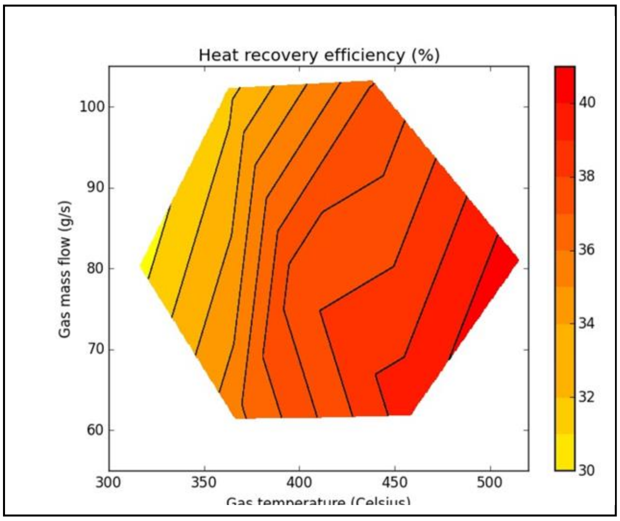

Figure.6. Experimental heat exchanger efficiency.

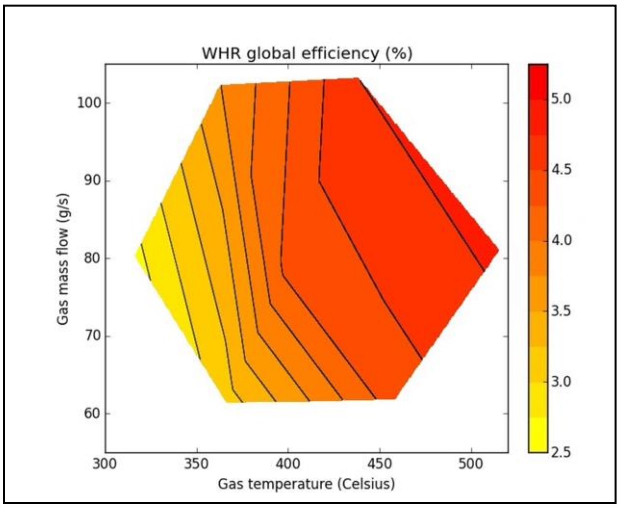

Figure 7. Experimental overall waste heat recovery efficiency.

4. Conclusions

Numerical and experimental analysis of the heat exchanger impact on overall efficiency of a Rankine cycle have been conducted by means of a simulation code in Python(x,y) and Rankine cycle experimental test bench.

Maximum heat exchanger efficiency was estimated to be 41.82% while the experimental research revealed a value of 40.69%. Both values are obtained at the same engine operating point defined by exhaust gases mass flow and temperature of 80 [g/s] and 500 °C, respectively.

However, overall Rankine cycle efficiency is much lower. The maximum value estimated by numerical simulation is 5.14% while the experimental research revealed a value of 4.81%.

On the bases of comparison between numerical and experimental results we can conclude that the maximum deviation is 6.4% at the operating point corresponds to maximum overall efficiency. This deviation is relatively small and allows to us using that 0D simulation model for further analysis on the Rankine cycle.

The results of this study reveal that the evaporator is the milestone in the closed thermodynamic Rankine cycle influencing directly the efficiency of the cycle. On the bases of these results we can conclude that the prototype of the heat exchanger is not optimal solution because the heat exchanger efficiency is not high. In order to increase its efficiency it is necessary to reduce exhaust gas flow cross section area which will increase the heat transfer coefficient from exhaust gas side.

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ЧИСЛЕНО И ЕКСПЕРИМЕНТАЛНО ИЗСЛЕДВАНЕ НА ТОПЛООБМЕННИК ОТ СИСТЕМА ЗА РЕКУПЕРИРАНЕ НА ЕНЕРГИЯ ОТ ОТРАБОТИЛИТЕ ГАЗОВЕ ЧРЕЗ ЦИКЪЛА НА РАНКИН

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Abstract:

В статията е представено числено и експериментално изследване на влиянието на топлообменника върху ефективността на система за рекупериране на енергия базирана на цикъла на Ранкин. За численото изследване е използван 0D физичен модел и на неговата база е създадена изчислителна програма в средата Python(x,y). За провеждането на експерименталното изследване е конструиран прототип на топлообменник. Работният флуид циркулира в двадест и три тръбопровода, огънати спираловидно и разположени по дължината на топлообменника. Топлообменника е с противоположно движение на флуидите и обща топлообменна повърхност от 1.8 m^2. Числените и експерименталните резултати потвърждават, че топлообменникът е един от най-важните елементи в от цикъла на Ранкин. При изследването най-високата ефективността на цикъла на Ранкин е 5.14% и 4.81%. Тези стойности са получени съответно от числените и експерименталните изследвания. На базата на получените резултатите може да се каже, че за повишаване на ефективността на цикъла е необходимо да се усъвършенства конструкцията на топлообменника като се повиши коефициента на топлообмен от страна на отработилите газове.

Keywords: *топлообменник, рекупериране на енергия, цикъл на Ранкин, числено и експериментално изследване.*