




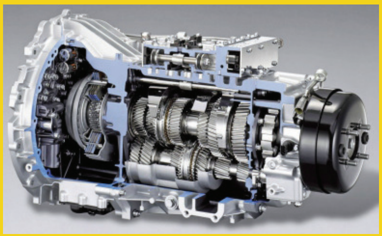
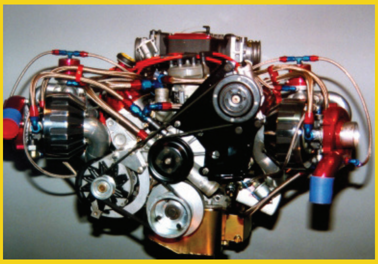

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DEVELOPMENT OF 0D SIMULATIONAL MODEL FOR RANKINE-HIRN CYCLE HEAT EXCHANGER OPTIMISATION

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Abstract:
The article presents a 0D model development of the heat exchanger intended for a Rankine-Hirn cycle waste heat recovery system in internal combustion engines. The heat transfer in the exchanger was estimated as the volume was separated by small elements. For each of the elements the heat transfer was calculated depends on the temperature variation, surface of the transfer and total heat transfer coefficient. An estimation code based on the model was developed in Python. CoolProp simulation code was used for working fluid parameters determination. Numerical estimation of the heat transfer was presented with different working fluids at the most commonly used operating point of a tractor engine. Finally, a study of the working fluid mass flow rate effect on heat transfer effectiveness was conducted as water was used into the Rankine-Hirn cycle.

Keywords: *heat exchanger, Rankin-Hirn, waste heat recovery, simulation*

1. Introduction

In order to meet future restriction of CO_2 emission it is necessary to reduce the engine fuel consumption which means to increase the engine overall efficiency. Despite of the advance technologies, the engine efficiency of more than 35% is achieved only at certain type of engines. In fact more than 60% of fuel energy is lost in form of heat in exhaust gas and cooling system.

Waste heat recovery seems to be the most prospective way to increase engine efficiency and reduce fuel consumption [1-5]. A study [1] revealed that waste heat recovery by exhaust gas is the most effective due to the highest exergy contains into. Several techniques for exhaust heat recovery are known [3, 4, 6-8]:

- Conversion of exhaust gas energy into mechanical energy by means of supplementary turbine: turbo-compounding;
- Conversion of exhaust gas energy into mechanical energy by means of heat machine: Rankine-Hirn cycle, Ericson engine, Stirling engine, etc.;
- Conversion of exhaust gas energy directly into electrical energy by means of thermogenerators;
- Thermoacoustic heat engines.

The Rankine-Hirn cycle indicates the highest potential in energy recovery due to a small effect on the backpressure and the highest efficiency which can reach more than 10% [9-15].

2. The Rankine cycle operation

The system consists – tank, pump, heat exchanger-evaporator, turbine and condenser (Figure 1). The pump increases the fluid pressure during the phase ***a-b***. The pressure value at point ***b*** depends on the type of fluid. In the heat exchanger ***b-c*** the fluid is heated by exhaust gas on three different stages: preheating,

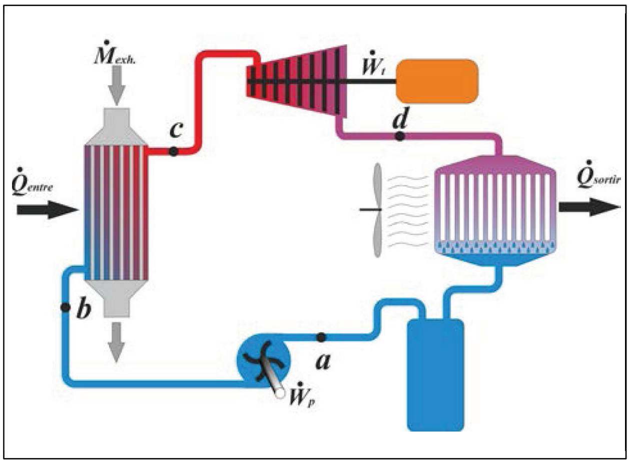

Figure 1. Rankine cycle scheme

evaporation and overheating. The process ***c-d*** is expansion of the preheated vapor. Different types of expander can be used such as turbine or piston machine. The mechanical power on the output shaft of the expander can be added to the crankshaft of the engine or can be transformed into electrical power by a generator. At the end of the circuit the fluid condensation is carried out in the condenser: process ***d-a***.

The variation of working fluid thermodynamic parameters is presented at ***p-h*** and ***T-s*** diagrams in Figure 2.

The most critical point of the Rankine-Hirn cycle is the heat exchanger. Inside of it the heat transfer

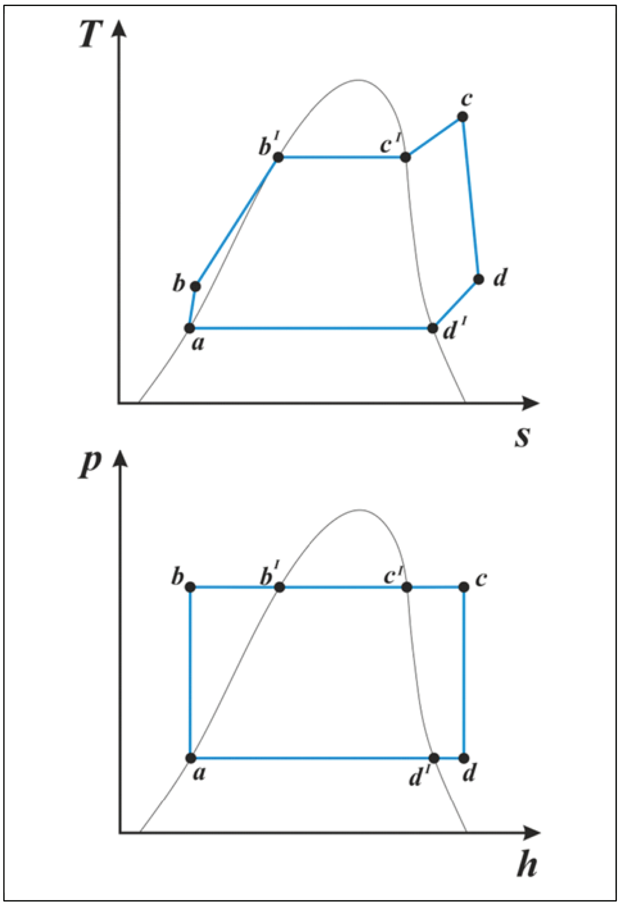

Figure 2. ***T-s*** and ***p-h*** diagram of Rankine-Hirn cycle

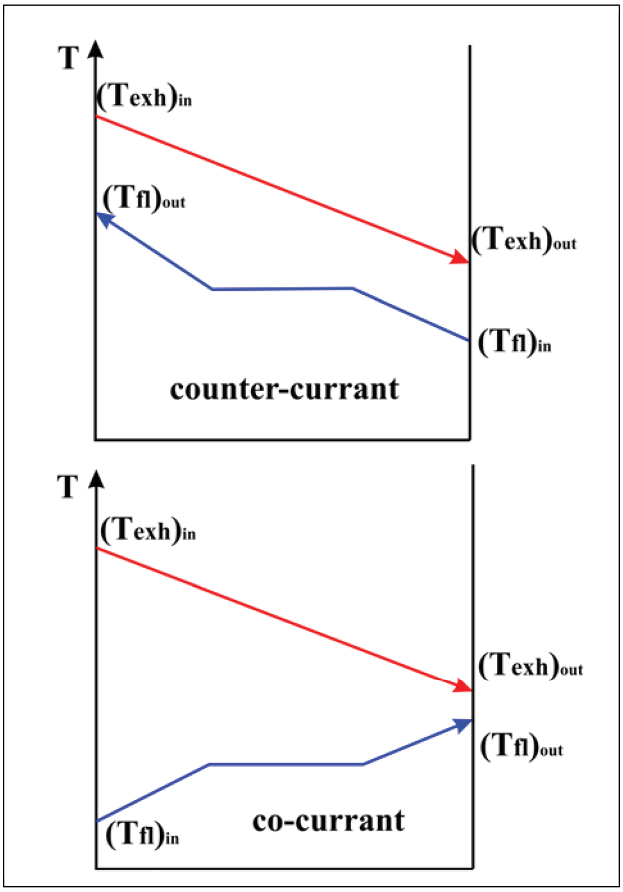

Figure 3. Temperature development in heat exchanger depending on flow direction

between exhaust gas (hot source) and working fluid is occurred. The heating process is conducted with phase changing of the working fluid. Shell and tubes is most commonly used heat exchangers design [16]. Two type of exchanger are known depend on direction of the fluids flow: counter-current and co-current. The temperature variation for both type heat exchangers are presented in Figure 3. Counter-current heat exchanger provides better effectiveness of the heat transfer.

A numbers of simulation models were described in the literature [17-24]. At stationary operating condition ε-NDU [17] and the mean temperature difference method are most commonly used [18, 19]. Two types of models for transient operating condition were reported – boundary-moving model and discretized model [21-24]. The mean temperature difference method is used for heat transfer calculation in case all of the parameters are known. Then following relation is used:

$$Q = A.U.T_{lm} \qquad (1)$$

A – heat transfer surface, m^2;

U – overall heat transfer coefficient, $\frac{W}{m^2K}$;

T_{lm} – logarithmic mean temperature difference, K.

T_{lm} can be estimated as follows:

In case of counter-current flow

$$T_{lm} = \frac{[(T_{exh})_{in} - (T_{fl})_{out}] - [(T_{exh})_{out} - (T_{fl})_{in}]}{ln\frac{[(T_{exh})_{in} - (T_{fl})_{out}]}{[(T_{exh})_{out} - (T_{fl})_{in}]}} \quad (2)$$

In case of co-current flow

$$T_{lm} = \frac{[(T_{gas})_{in} - (T_{fl})_{in}] - [(T_{gas})_{out} - (T_{fl})_{out}]}{ln\frac{[(T_{gas})_{in} - (T_{fl})_{in}]}{[(T_{gas})_{out} - (T_{fl})_{out}]}} \quad (3)$$

Boundary-moving model is based on a heat transfer calculation at each of three zones of the heat exchanger (pre-heating, evaporation and overheating) by means of mean temperature values and transfer surface determined by the boundaries. This adaptive model with moving boundaries provides good accuracy and small computational time.

The most complicated model is discretized. In this model a discretization of heat transfer surface is applied. At each of the finite element (volume) a heat transfer estimates as a basic heat transfer relation is used. A boundary equation is used to transfer the fluids parameters between the elements.

The aim of this study is development of 0D discretized heat exchanger model. This model will be further used for heat exchanger geometrical parameters optimization in real Rankine-Hirn cycle.

3. Mathematical background

The computational scheme of the discretized heat exchanger model is presented in Figure 4. This scheme presents heat transfer estimation in counter-current heat exchanger. The model was developed as it was considered that the mass flow of exhaust gas and working fluid is constant. With other words it means that steady flow was considered. It was supposed that there are not differences in working fluid parameters between the tubes and the parameters varies as a function of tube length. The heat transfer with atmosphere as well as pressure drop at heat exchanger was neglected.

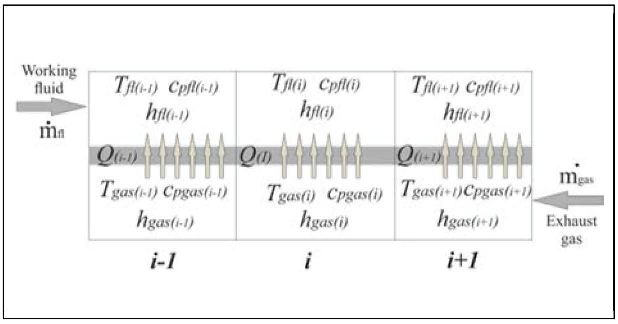

Figure 4. Computational scheme of discretized heat exchanger model

Then the heat transfer rate at the finite volume was calculated as follows:

$$Q_i = A_i . U_i . (T_{gas(i)} - T_{fl(i)}) \quad (4)$$

The heat transfer surface has the same value for each volume:

$$A_i = \frac{A}{n} \quad (5)$$

The heat transfer coefficient was estimated by correlation as follows:

$$U_i = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2}} \quad (6)$$

Heat transfer coefficient in convection usually is calculated by the average Nusselt number *Nu* as follows:

$$\alpha = \frac{Nu.\lambda}{d_h} \quad (7)$$

where λ is thermal conductivity and d_h is hydraulic diameter. The hydraulic diameter depends on the design of heat exchanger. At most commonly used heat exchangers it is inner diameter of pipe for working fluid and outer diameter for exhaust gas. Nusselt number is usually given by a correlation where Reynolds number *Re* and Prandtl number *Pr* are into.

The heat flow at finite volume increases the enthalpy of working fluid and decreases the enthalpy of exhaust gas. Then working fluid enthalpy flow and specified enthalpy at the next finite volume were defined as follows:

$$H_{fl(i+1)} = H_{fl(i)} + Q_i \quad (8)$$

$$h_{fl(i+1)} = h_{fl(i)} + \frac{Q_i}{\dot{m}_{fl}} \quad (9)$$

Exhaust gas enthalpy flow and specified enthalpy were estimated as:

$$H_{gas(i+1)} = H_{gas(i)} - Q_i \quad (10)$$

$$h_{gas(i+1)} = h_{gas(i)} - \frac{Q_i}{\dot{m}_{gas}} \quad (11)$$

In order to estimate working fluid parameters (temperature, density, c_p and etc.) simulation code CoolProp was used.

Exhaust gas temperature was estimated as enthalpy flow equation was used:

$$H_{gas(i+1)} = \dot{m}_{gas} . c_{pgas(i+1)} . (T_{gas(i+1)} - T_0) \quad (12)$$

Total heat transfer at the exchanger was estimated as follows:

$$Q = \sum_{i=1}^{n} Q_i \quad (13)$$

Then enthalpy rate change in working fluid and enthalpy rate destruction in exhaust gas can be estimated as follows:

$$H_{fl(out)} = H_{fl(in)} + Q \quad (14)$$

$$H_{gas(out)} = H_{gas(in)} - Q \quad (15)$$

An estimation code on the basis of mathematical background was developed in Python software. The calculation algorithm is presented in Figure 5.

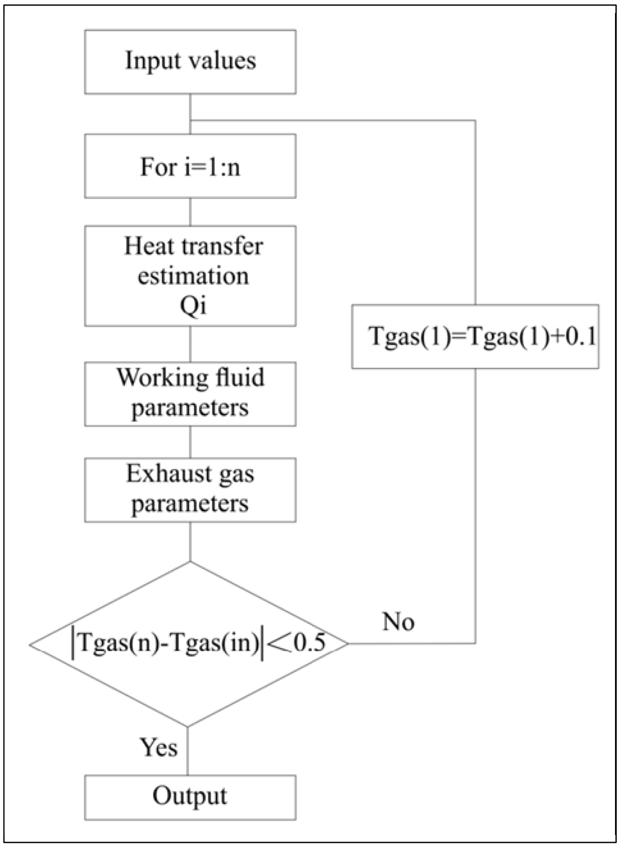

Figure 5. Calculation algorithm of the simulation code

The calculation begins at the finite volume which corresponds to working fluid input and exhaust gas output. In this case exhaust gas temperature as well as other gas parameters were unknown. In order to start the calculation process exhaust gas parameters were input in the algorithm. However at the end of the estimation cycle a comparison between estimated input exhaust gas temperature and real input gas temperature which was known from engine simulation was conducted. If the difference is higher than limited value the main cycle of the code repeats again. The cycle has been repeated until the difference became lower than limited value.

4. Results

4.1. Heat transfer simulation with different working fluid

Firstly, a simulation of heat transfer into the heat exchanger with four working fluids was conducted. For each of the studied case constant heat source parameters was input in the model. The exhaust gas thermodynamics parameters of an agricultural diesel engine were taken as heat source. The engine was numerically studied before [4]. The exhaust gas parameters which were used in present study were estimated at most commonly used operating point during plowing - $n = 1650rpm$ and BMEP = 10bar. The parameters are listed in Table 1. The exhaust gas output temperature was constant at each of studied case which means a constant effectiveness of heat transfer was obtained. Due to different characteristic of each fluid to provide a constant effectiveness it was necessary to varied heat transfer surface and fluid mass flow rate.

Table 1

Exhaust mass flow $\dot{\boldsymbol{m}}_{\boldsymbol{gas}}$, (kg/s)	0.158
Specific heat capacity $\boldsymbol{c}_{\boldsymbol{pgas}}$, (J/kg.K)	1125
Exhaust gas inlet temperature $\boldsymbol{T}_{\boldsymbol{gas,in}}$, (K)	691
Exhaust gas outlet temperature $\boldsymbol{T}_{\boldsymbol{gas,out}}$, (K)	455
Exhaust gas enthalpy $\boldsymbol{h}_{\boldsymbol{gas}}$, (kW)	70.7

The most commonly used working fluids were studied – water, ethanol, R245fa and R134a. In the Table 2 are listed working fluid parameters and heat exchanger geometrical parameters for each fluid.

Table 2

Fluid	$\dot{m}_{fl}$ (kg/s)	$T_{fl,in}$ (K)	$T_{fl,out}$ (K)	p_{fl} (bar)	A (m^2)
Water	0.0165	373	573	25	23.6
Ethanol	0.033	350	573	20	9
R245fa	0.11	306	473	20	6.4
R134a	0.12	297	473	20	5.3

It is evident that different working fluids required different heat transfer surface as well as different mass flow rate to provide the same effectiveness of heat transfer. The water due to high variation of specific enthalpy during evaporation required of low flow rate. The flow rate is two times lower than ethanol and approximately seven times lower then organic fluids – R245fa and R134a. It means that mechanical energy consumes from the pump in the system is much lower in case water is chosen as a working fluid. However, the water required bigger heat transfer surface than other fluids. The heat transfer surface has to be 2.5 times bigger than ethanol and approximately four times bigger than organic fluids. Organic working fluids own another disadvantage – low condensing temperature which requires big condenser surface.

The variation of both temperatures that is of exhaust gas and working fluid is shown in Figure 6 to 9. By means of this results for each fluid can be determined the boundaries when phases changes occurs in working fluid. The largest evaporating zone was observed in case water was chosen as working fluid. Once the evaporation process of the water is finished the vapor temperature increases rapidly. In this case the variation of exhaust gas

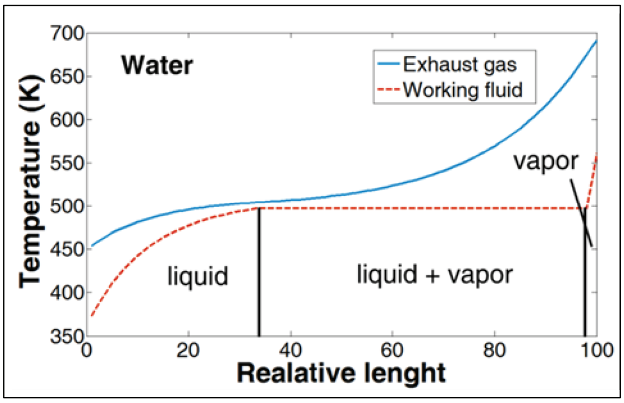

Figure 6. Heat exchanger temperature variation with *Water* as working fluid

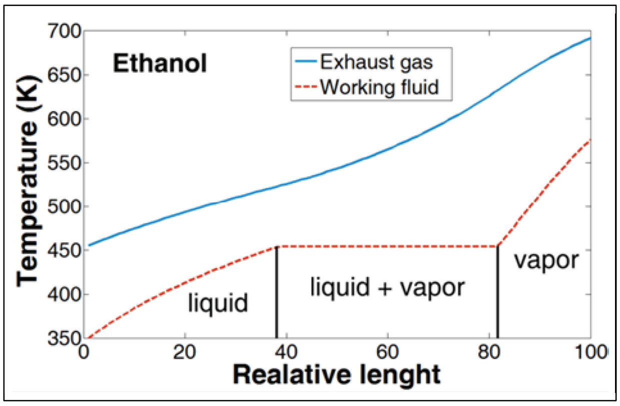

Figure 7. Heat exchanger temperature variation with *Ethanol* as working fluid

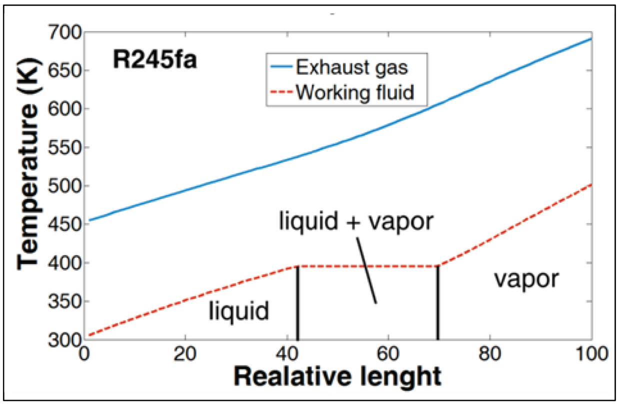

Figure 8. Heat exchanger temperature variation with *R245fa* as working fluid

temperature is strongly nonlinear. At all other simulations the exhaust gas temperature decreases approximately linear from heat exchanger inlet to the outlet. The narrowest evaporating zone was observed in case R245fa was used as working fluid Figure 8.

4.2. Heat transfer simulation as function of the fluid parameters

Secondly, a simulation of heat transfer effectiveness was conducted as a function of working fluid mass flow rate. The simulation was carried out choosing water as a working fluid. The results are presented in Fig. 10.

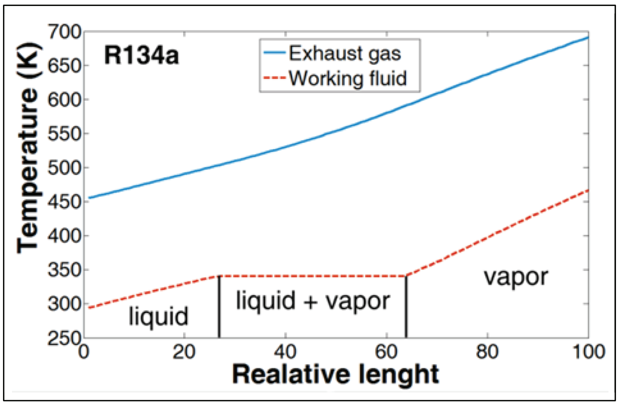

Figure 9. Heat exchanger temperature variation with *R134a* as working fluid

The results revealed that low fluid flow rate significantly decreased effectiveness. At flow rate lower than 0.014kg/s the working fluid temperature at the heat exchanger outlet reached inlet exhaust gas temperature. High fluid flow rate increased heat transfer effectiveness. At flow rate of 0.02kg/s the maximum effectiveness was estimated to 0.62. The diagram revealed that the effectiveness should slightly increases as a function of mass flow at values more than 0.02kg/s. However at flow rate higher than 0.018kg/s outlet fluid temperature is equal to fluid evaporating temperature. It means that not whole quantity of the working fluid is evaporated at heat exchanger outlet. The working fluid droplets have to be avoided into expander machines due to protect expansion machine from mechanical failure especially in case of a turbine is used.

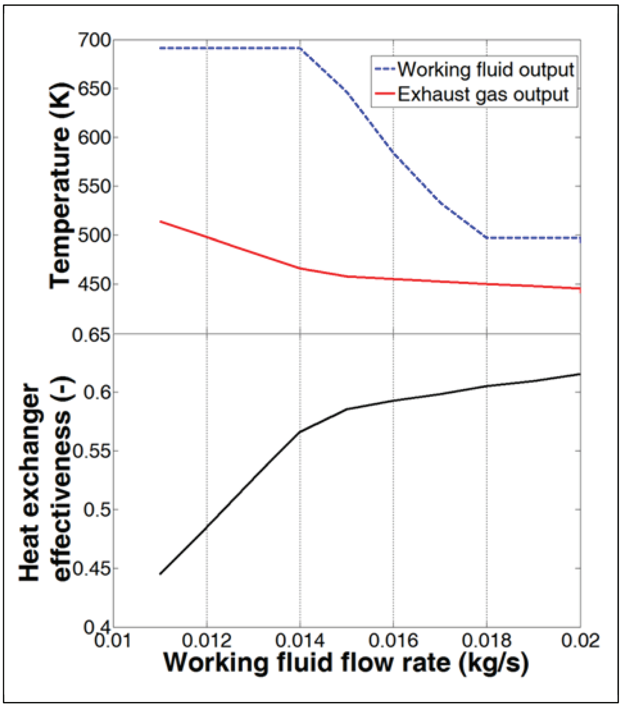

Figure 10. Heat transfer effectiveness as a function of working fluid mass flow rate

5. Conclusions

A 0D discretized model was developed in order to estimate heat transfer into heat exchanger of the Rankine-Hirn cycle based recovery system. The model provides opportunities to study the working fluid parameters as well as the heat exchanger geometrical parameters at stationary operating condition. An estimation code was developed in Python.

A study conducted with four working fluids revealed that a constant heat transfer effectiveness can be obtained as different heat transfer surface and working fluid mass flow rate was applied to each fluid. Water requested low mass flow rate but high heat transfer surface. Whereas organic working fluids R245fa and R134a requested high mass flow rate and low heat transfer surface.

By means of the model the variation of hot source temperature and working fluid temperature along the flow path was presented. For each fluid it revealed the boundaries between the phases at working fluid side. The temperature difference at pinch point can be also determined.

The study of working fluid mass flow rate effect on heat exchanger effectiveness revealed that the effectiveness rapidly decreased at that flow rate where working fluid output temperate reaches the hot source temperature. The effectiveness can be increased by increasing of mass flow rate. However at high flow rate not whole quantity of the working fluid was evaporated at the outlet section. Within mass flow range that provided fluid output temperature within the range between evaporating temperature and inlet hot source temperature the heat transfer effectiveness is within the range of 0.56 to 0.6.

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РАЗРАБОТВАНЕ НА СИМУЛАЦИОНЕН МОДЕЛ ЗА ОПТИМИЗИРАНЕ НА ТОПЛООБМЕНИКА ОТ СИСТЕМА ЗА РЕКУПЕРИРАНЕ НА ЕНЕРГИЯ ЧРЕЗ ЦИКЪЛ НА РАНКИН

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Резюме:
В публикацията е представен стационарен модел на топлообменика от система за рекупериране на енергия от отработилите газове на ДВГ чрез цикъл на Ранкин. Моделирането е извършено като обемът на топлообменика е разделен на елементи с еднакви геометрични размери. За всеки елемент топлообменът е изчислен като са използвани разликата в температурите на двата флуида, топлообменната повърхност и коефициента на топлообмен. На базата на модела е направена изчислителна програма в работната среда за програмиране Python. Софтуерното приложение CoolProp е използвано за определяне на параметрите на работния флуид от цикъла на Ранкин. Представени са резултати, получени с различен работен флуид. Симулациите са извършени с източник на топлина с постоянни параметри, съответстващи на параметрите на отработилите газове на дизелов двигател в най-често използваната работна точка по време на работа. Направено е изследване на влиянието на дебита на работния флуид върху ефективността на топлобмен в случая на използване на вода в цикъла на Ранкин.

Ключови думи: *топлообменик, цикъл на Ранкин, рекупериране на енергия, симулация*