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HEAT RECOVERY SYSTEMS FOR PASSENGERS VEHICLES

Yulia GLAVATSKAYA[1,2*]; Gérard OLIVIER[1]; Osoko Fredy SHONDA[1]; Pierre PODEVIN[2]
[1]RENAULT Advanced Electronics and Technologies Division API : TCR AVA 0 58, 1 avenue de Golf 78288 Guyancourt, France
[2]Laboratoire du génie des procédés pour l'environnement, l'énergie et la santé, EA21, Le Cnam (Conservatoire national des arts et métiers), case 2D3R20, 292 rue saint Martin 75003 Paris

Abstract: *Within the scope of new environmental standards (Euro 5, 6 and more), it becomes more difficult to reach the objectives of CO_2 emissions and pollutants by the solely optimization of the thermal engine (HCCI, double supercharging, low-pressure EGR, downsizing). Depending on the driving cycle, the ambient temperature and the technology, a thermal engine converts in average up to 30 % of the fuel energy into mechanical shaft work. A significant proportion of the rest of energy (approximately 60 %) 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 shows the big interest in energy recovery systems. This paper presents the various systems enabling the recovery of this energy. A Rankine cycle, widely used in the industry, is of particular interest. The efficiency of energy conversion varies from 5 to 20 % depending of its conception (working fluid, architecture, coupling…).
Moreover, the layout of the Rankine cycle integrated within a vehicle depends on the choice of the working fluid as well as the technologies of the components i.e. the expander, the evaporator, the condenser and the pump. Within this context, the issue regarding the mass and the size of the system has to be considered.
Finally, we present several solutions of Rankine systems for passenger car application and show each advantage and limits.

Keywords: Heat recovery, heat exchanger, Rankine cycle, thermal engine, exhaust gases

INTRODUCTION

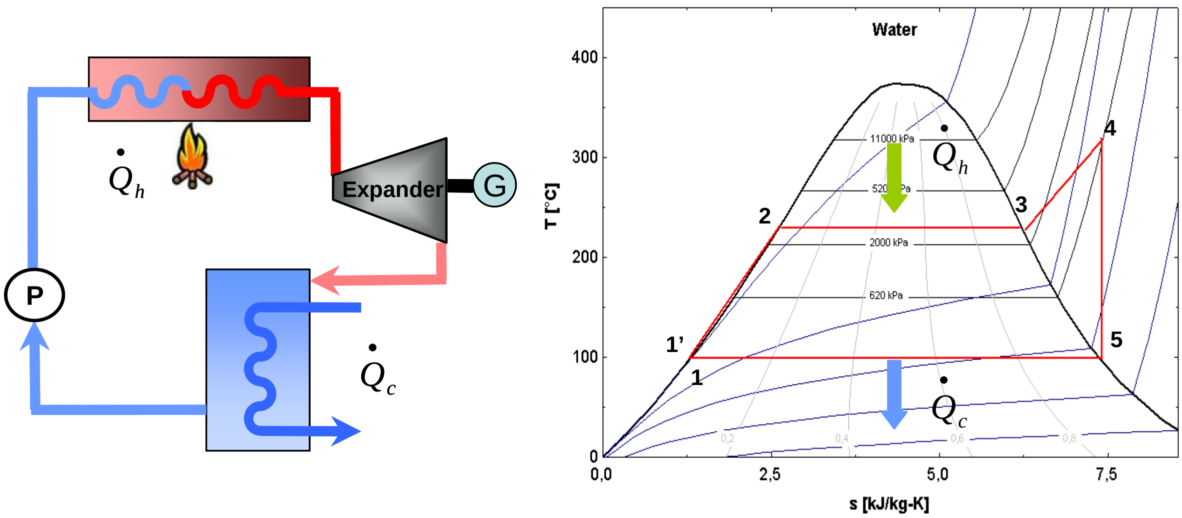
Since the industrial revolution in the 19th century, emissions of greenhouse effect gases have been increasing and consequently caused the climate that we encounter nowadays. In order to reduce the emissions aforementioned, some actions were taken e.g. the Kyoto protocol which aimed to reduce industrial, transport and building emissions. Transport emissions present important part of the GWP. The reduction of transport emission is possible within the improvement of the engine efficiency and its conception (downsizing). However, in average, two thirds of the fuel energy is still wasted throughout the exhaust gases and the cooling liquid. In this context, an improved utilization of wasted energy is of great significance. In order to minimise the fuel consumption and engine emissions, the conversion of wasted energy shows a great potential.
The recovery of the wasted energy is possible within a thermodynamic cycle, thermoelectric generation by Seebeck effect and using a turbo compound placed on the exhaust pipe.
Thermoelectric generator for automotive application has been widely studied for years [1-5]. One of the most important advantages of this device is its compactness. The materials constituting the thermoelectric device affect the recovering efficiency of the system. To improve the recovering capacity of the TE device the study of the different material is very important [2]. Today, Crain et al.

* Corresponding author. Email: yulia.glavatskaya@renault.com

[3] announce the power output of a thermoelectric generator of up to 125W at 600°C hot air test bench.
Turbocompounding, introduced in the late 1940s, shows a good potential in the waste heat recovery especially for truck application [6-7]. Improvement of the truck fuel consumption of 11% is possible for road application. It was found that within a transient driving mechanical turbo compound device could cause the increasing of the fuel consumption due to backpressure.
Different thermodynamic cycles are used for an energy production. The Stirling engine is commonly uses for an electricity production by solar power plant [8], in marine and cryogenic estates. In the automobile industry, the Stirling engine was used as well. Toyota patented a system enables to recovery the heat from the exhaust gases for mechanical work production [9]. Bonnet et al. [10] studied the application of an Ericsson engine for a heat recovery. Two technologies have a good potential in the waste heat recovery for a stationary application. The application of the Stirling and Ericsson engine is not practical due to weight, cost and size.
Stobart et al. [11] discussed different thermodynamic cycles in order to point out the best for car applications. Bottoming cycles, commonly used in the industry, can be used for the heat recovery. A Rankine cycle presents a good potential for a waste heat recovery light-duty track as well as passenger vehicles [12-14].
This paper summarises several solutions of Rankine systems for passenger car application and shows its advantages and its limits.

Rankine cycle principles

The Rankine cycle is a thermodynamic cycle dedicated to generate mechanical work from heat. This mechanical work can be converted into electrical energy.



(a) (b)

Figure 1. Scheme (a) and Ts diagram (b) of the Rankine cycle

The superheated Rankine cycle (or Hirn cycle) consists of the following processes:
1-1': pumping of the working fluid
1'-3: evaporation of the fluid into the boiler
3-4: superheating the working fluid
4-5: expansion through an expander
5-1: condensation of the fluid into the condenser.
In order to ensure a good efficiency of the Rankine cycle, the choice of the working fluid is very important. The environmental, corrosion, frost and safety aspects have to be considered as well. Teng et al.[16] compared three types of the working fluids determined by the slope of the saturated vapour line (Figure 2): dry, wet and isentropic. The isentropic and dry fluids are preferable for a Rankine cycle because they remain in the gas phase after an isentropic expansion. This prevents the destruction of the expansion device.

El Chammas et al. [24], Ringler at al.[25] and Aoun B. [16] compared organic fluids and water. Its researches show that to use water becomes inefficient for Rankine cycle at low boiling temperature. In this context, organic fluids should be used.

Components

A Rankine cycle has four components: an expander, a pump, a boiler and a condenser. The choice of the components is very important in order to maximize the Rankine cycle power output and its efficiency.
Expanders can be classified into two main categories i.e. turbines and volumetric machine (positive displacement machine). Turbines are not suitable for a low power output Rankine cycle (less than 100kW) [16]. Therefore, volumetric machines like screw expanders, rotary vanes, scroll and wobble plate expanders have to be considered [16-18]. The screw, wobble plate and rotary vane expanders require lubrication. Thus, they are more suited for operating with refrigerants [16, 17]. The wobble plate compressors are largely used in automotive application for air conditioning systems, therefore, it has already approved. The scroll machines are also used in automotive application because of its performance and compactness. For a heat recovery Rankine system for a automotive application the choice of the expander bases on the available room inside the vehicle, suitable recovered power, working fluid used into the Rankine loop (lubrication and viscosity of the fluid) and the masse flow rate of the working fluid.
Among all type of the heat exchangers three of them are used for automotive application: plates, shell and tubes and tubes-fin heat exchangers. Plate heat exchangers, compact and efficient, can be used for a heat exchange between two fluids [19]. Shell and tube heat exchanger using for EGR system cooling by water as well as the tubes and fins heat exchangers commonly used for engine cooling present a big interest for exhaust gas recovery. Anyway, the selection of the heat exchanger technology has to be based on the available room into the vehicle and power has to be evacuated or recovered.

Integration of the Rankine cycle into the vehicle

The integration of the Rankine cycle within a vehicle implies the choice of the heat source and the heat sink. There are two main heat sources into the vehicle: exhaust gases and cooling water.

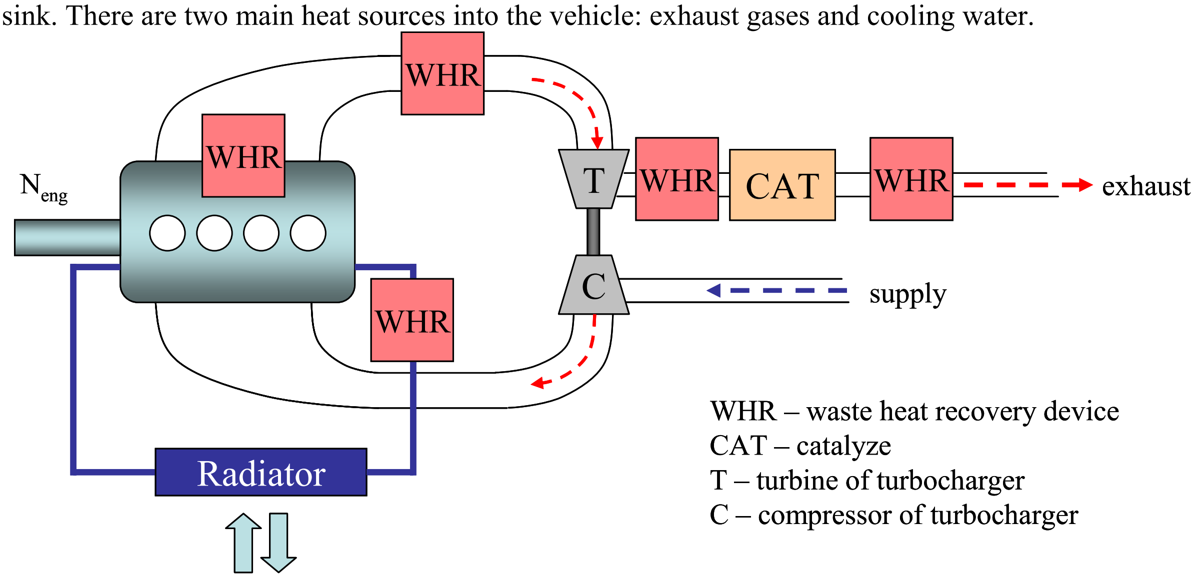


Figure 2. Position of the waste heat exchanger into exhaust line and cooling loop

The amount of the heat used by a Rankine cycle depends on the location of the boiler (Figure2): the closer the heat exchanger to the engine, the greater the temperature of the heat source. As regards the exhaust pipe, the integration of the boiler upstream the catalyser can damage the engine after-treatment efficiency.
To condense the working fluid the cooling water or air could be used. The integration of a condenser into the vehicle cooling system damages the performances of the engine. Thus, the integration of the Rankine cycle into the vehicle implies the study of the overall thermal system of the vehicle.

Heat transfer from heat source should be optimised by means of the choice of the thermal architecture. Different thermal architectures of the Rankine cycle are possible. The architecture depends on heat source and cold sink (Figure3).

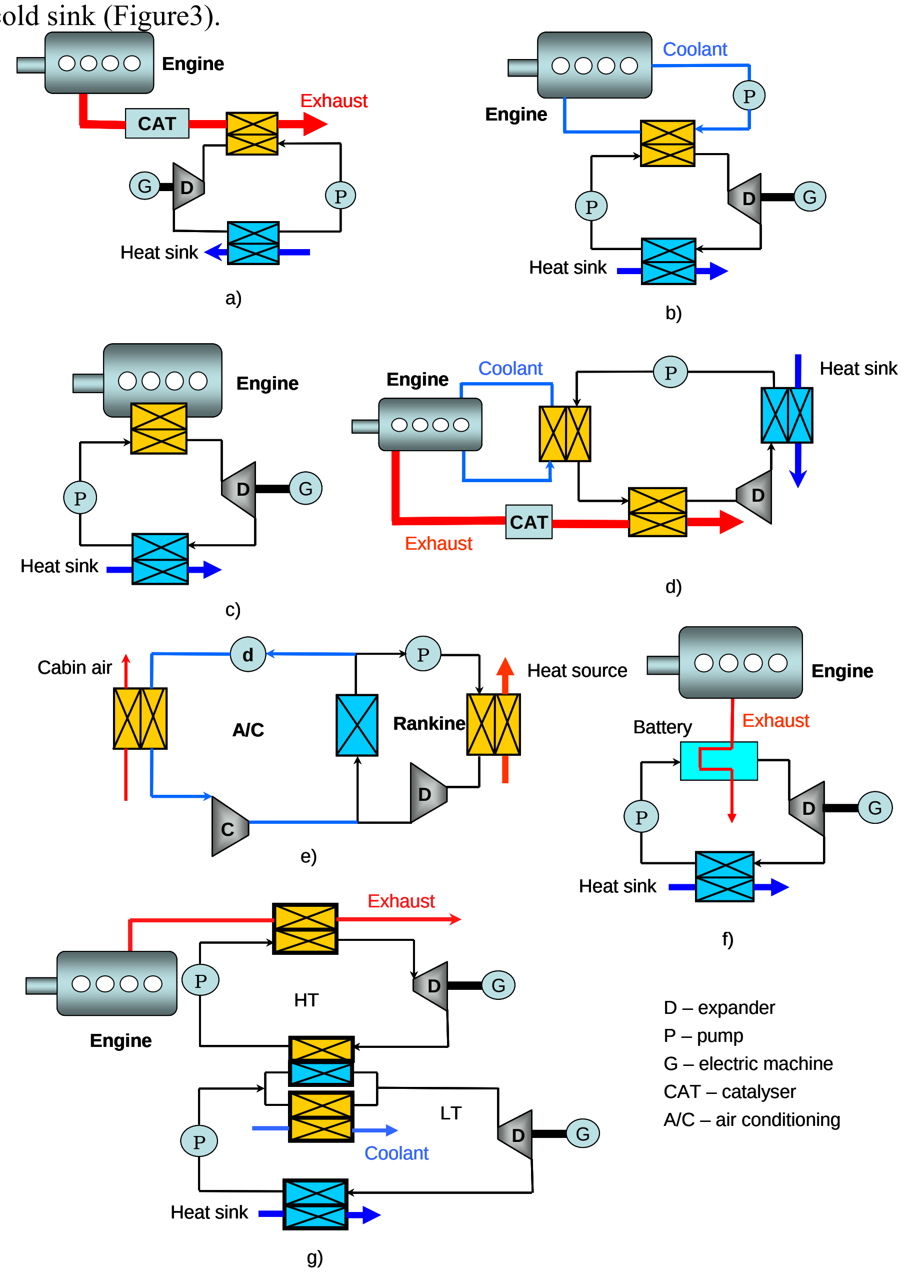


Figure 3. Thermal architectures of the Rankine cycle. a) exhaust only b) coolant only c) evaporator into exhaust manifold d) combined exhaust and coolant system e) coupling A/C and Rankine cycle f) energy stoker boiler g) two loops Rankine cycle

It is important to properly use the produced additional energy. A Rankine cycle can be used for electric and mechanical energy production. Different mechanical architectures are shown in figure 4. The pump and expander can have the same shaft in order to produce mechanical or electrical work (Figure4 a,b). In order to ensure good efficiency of the cycle, the expander and pump should have a different shaft (Figure4 c) enabling different rotary speeds.

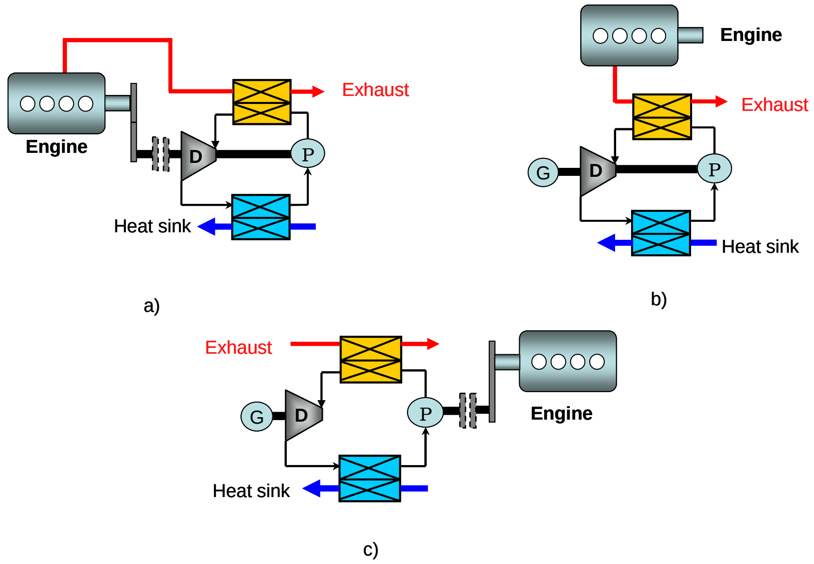


Figure 4. Mechanical architectures of the Rankine cycle: a) mechanical expander b) electric expander c) electric expander/mechanical pump

Integration of the heat recovery Rankine system into a passenger car impacts many aspects of the vehicle operation. The integration of the evaporator within the exhaust pipe can damage the performances of the combustion engine because of the backpressure of the exhaust gases and lesser efficiency of the post-treatment system. Moreover, the Rankine cycle is a thermodynamic system that has to be taken into consideration in global thermal management of the vehicle because its impacts the thermal capacity of the cooling system, engine operation during transient state etc. On the other hand, the Rankine cycle is an additional system that impacts the weight and cost of the vehicle. Thus, the optimum size of the system (components size) has to be carefully chosen. In this context, the modelling of the Rankine system is needed.

Modelling of the Rankine cycle

In order to study the Rankine cycle performances simple architecture was chosen : boiler is integrated into the exhaust gas pipe; the condenser is integrated into a cooling loop. The steady-state model was developed [36]. The model of steam Rankine cycle associates the sub-models of an evaporator, a turbine, a condenser and a pump. The pump supplies the working fluid to the evaporator. After being vaporised and superheated by the evaporator, the high-pressure steam expands through a piston expander. The low-pressure vapour flows towards the condenser using the cooling liquid as the cold source.

Heat exchanger model

The condenser and evaporator are modelled by means of the ε-NTU method. The heat exchangers are split into three virtual zones corresponding to the state of the working fluid [28] i.e. liquid phase, two-phase and gas phase (Figure 5).

The thermal power of the heat exchanger (equation (1)), the heat transfer coefficients and the pressure drops were determined into each of three zones.

$$\dot{Q} = \varepsilon\left(\dot{m} c_p\right)_{\min}\left(T_{h,1} - T_{c,1}\right) \qquad (1)$$

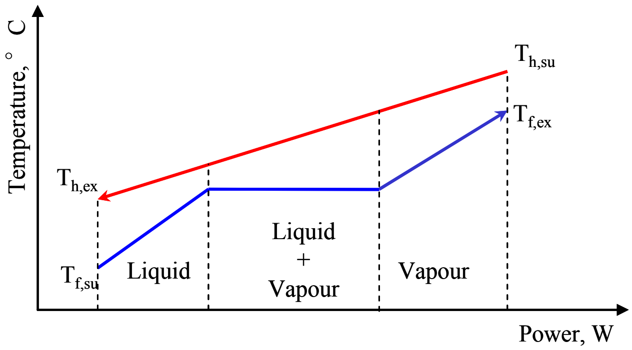


Figure 5. Scheme of the evaporation of the working fluid

The heat transfer coefficients are calculated from the correlations depending on the technology of heat exchangers and the phase of the working fluid (i.e. liquid or two-phase). Thus, Rohsenow correlation [32], Shah correlation [33] were used for evaluate boiling and condensation heat transfer coefficient respectively. Single-phase heat transfer coefficient is evaluated using a non-dimensional relationship (equation (2)):

$$Nu = C\,\mathrm{Re}^m\,\mathrm{Pr}^n \qquad (2)$$

The model enables us to compute one size of the heat exchanger when the two other are given. Thus, the total length of the heat exchanger L is determined as sum of the lengths of each zone.

Single-phase pressure drops are evaluated as the pressure drop through the tube at length L.

$$\Delta P = \frac{2f}{d}\frac{G^2 L}{\rho} \qquad (3)$$

The pressure drop within the two-phase zone depends on the vapour quality. The correlation for frictional multiplier suggested by Friedel [34] for steam flow was used in this case.

Expander model

The expander is a reciprocating machine which model is based on the modelling of a wobble plate compressor developed by Cuevas [35]. The evolution of steam through expander is divided into the following steps (Figure6): suction pressure drop, suction cooling, isentropic expansion, expansion at constant volume, exhaust pressure drop, exhaust heating.

The pressure drops within supply and exhaust ports is computed by reference to the flow through the simply nozzle.

$$\Delta P = \frac{\dot{m}^2}{2\cdot\frac{A_{thr}^2}{v_{su}}} \qquad (4)$$

Supply $\dot{Q}_{su}$ and exhaust $\dot{Q}_{ex}$ heat transfer are described as proposed by Quoilin et al. [28].

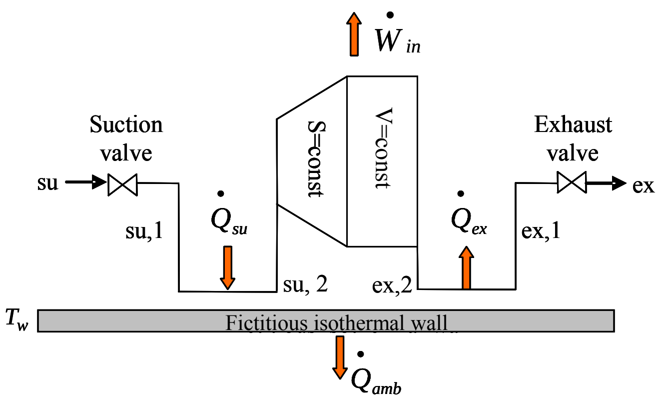


Figure 6. Scheme of expander model

The decreasing of the expander performance is due to the frictional losses. The expander mechanical losses $\dot{W}_{loss}$ are identified based on the information provided by the expander manufacturer. Thus, the expander power output is calculated as following:

$$\dot{W}_{\exp} = \dot{W}_{in} - \dot{W}_{loss} \quad (7)$$

The thermal balance of the expander is evaluated by equation (8).

$$\dot{W}_{loss} - \dot{Q}_{ex} + \dot{Q}_{su} - \dot{Q}_{amb} = 0 \quad (8)$$

Pump model

The pump is characterized by its overall efficiency η_{pp} and swept volume $V_{s,pp}$. The overall efficiency of the pump is product to its mechanical and volumetric efficiencies.

$$\eta_{pp} = \eta_{mec}\eta_{vol} \quad (9)$$

The masse flow of the working fluid trough the pump is calculated as following (equation (10)).

$$\dot{m}_{pp} = \frac{V_{s,pp}}{v_{su,pp}}\frac{N_{pp}}{60} \quad (10)$$

$$\dot{W}_{pp} = \frac{\dot{W}_{pp,in}}{\eta_{pp}} \quad (11)$$

$\dot{W}_{pp,in}$ - internal pump power, W

$\dot{W}_{pp}$ - pump power, W.

Rankine cycle model

The Rankine system has internal (in bold print on Figure7) and external (in italic print on Figure7) operating conditions. The compromise between these operating conditions gives the optimal cycle efficiency and expander power output.

The thermal Rankine cycle efficiency is determined as follow:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{ev}} = \frac{\dot{W}_{\exp} - \dot{W}_{pp}}{\dot{Q}_{ev}} \quad (12)$$

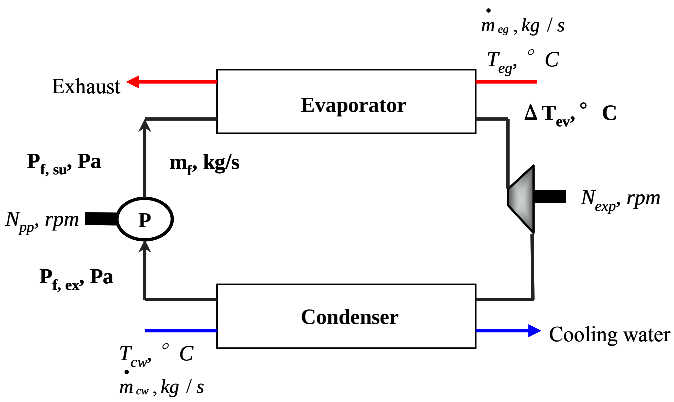


Figure 7. Rankine cycle sketch

DISCUSSION

In this section, we verify if a reasonable performance of the sized Rankine cycle can be achieved on different operating points.

The study was carried out for the high road Rankine system application. Water was chosen as a working fluid. The temperature level of the exhaust gases is set to 718 °C. To study the performance of the cycle the condenser pressure and temperature were kept 1 bar and 90 °C respectively. The pump is coupled to the shaft of the engine through the mechanical belt. The expander speed is set to $N_{\exp} = 4000 rpm$.

Figure 8 shows the variation of the cycle thermal efficiency and power output of the expander with respect to different temperature levels of the exhaust gases. It should be noted that both the energy efficiency and expander power output decrease as the temperature of the exhaust gases decreases. It is obvious that lowest temperature of exhaust gases yields to a lesser evaporator thermal power. Consequently, the high pressure and evaporator superheat also decrease. The expansion ratio of the expander decreases as well as its power output. Despite decreasing the exhaust gas temperature, the expander net power is maintained in a range of 4kW – 5.1kW that corresponds to a cycle efficiency of 0.125-0.145.

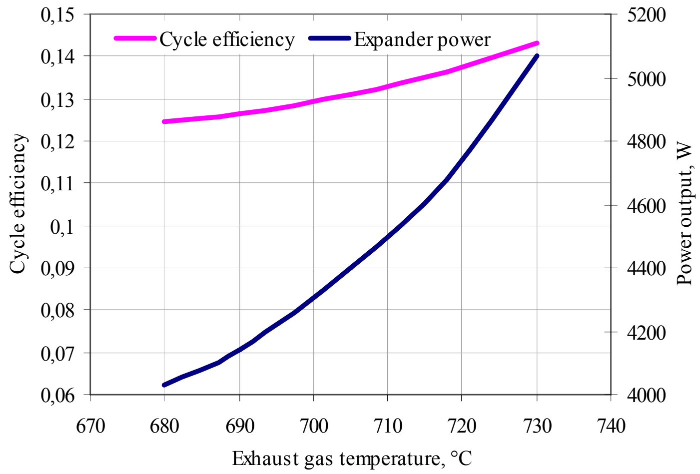


Figure 8. Variation of the efficiency of Rankine cycle and power output of the expander under different operating conditions.

Due to decreasing the exhaust gas temperature lower than 680°C, the working fluid is not superheated into the evaporator because the size of the evaporator is not enough. To continue generating enough power, the control of the Rankine cycle has to be analysed.

CONCLUSIONS

This paper proposes summarised solutions of the heat recovery systems for a passenger car application. Having analyzed different solutions, a Rankine cycle was chosen for exhaust heat recovery.

The choice of the working fluid and components for a Rankine cycle were discussed. It should be noted that organic fluids and water could be used for a Rankine cycle depending on the heat source temperature. The different heat source and heat sink were considered. It seems that the organic medium is more adapted for a low-grade Rankine cycle. For a Rankine cycle for exhaust heat recovery, water is better fluid.

Different architectures of the cycle were discussed in the paper. Challenge between thermal architecture and recovered power has to be done. It appears that the complex architecture lead to better heat using but the integration of complex system could be impossible. Using of the recovered energy was discussed as well. Thus, the mechanical or electrical power could be produced by Rankine expansion device.

It is important to point out that it appears many problems due to integration of such system into a car. In particular, the weight and the cost of the vehicle are affected. With integration of the Rankine cycle into a car, the thermal management has to be optimised because such system influences the performances of the cooling and after-treatment system.

Moreover, the size of the components is constrained by available space in vehicle. Small size components are preferable for automotive application. Thus, approved technologies used for Air Conditioning system could match to Rankine cycle application.

To achieve the optimum size of components giving a good power output a numeric study of the Rankine system is needed. The Rankine cycle steady-state model was developed. Detailed model of each component was presented for exhaust heat recovery application. The simulation of the Rankine cycle shown that the efficiency of 12-14% could be achieved (or expander power output of 4-5kW). It should be noted that the efficiency of the cycle depends on the exhaust gas temperature. It is obvious that a lower temperature of exhaust gases yields to a smaller efficiency. To ensure a good power output of the Rankine cycle the choice of the optimal external operating conditions for a different vehicle driving is needed. That implies the elaboration of the cycle control algorithm.

Nomenclature

AU	heat transfer coefficient, W/K
A	area,m^2
C	constant
c_p	specific heat, J/(kg K)
D_e	equivalent diameter of shell, m
D_s	shell diameter, m
D_h	hydraulic diameter, m
d	channel diameter
Fr	Froude number
f	friction factor
G	mass flux, $kg/(m^2s)$
h	heat transfer coefficient, $W/(m^2 K)$
h_{fg}	latent heat of vaporization, J/kg
L	length of heat exchanger, m
M	molar mass
m	constant
$\dot{m}$	mass flow rate, kg/s
n	constant
N	rotation speed, rpm
N_b	number of baffle
Nu	Nusselt number,
P	pressure, Pa
p^*	reduction pressure
P_{crit}	critic pressure, Pa
P_{sat}	saturation pressure, Pa
Pr	Prandtl number
$\dot{Q}$	heat transfer rate, W
$\dot{q}$	heat flux, W/m^2
Re	Reynolds number
R_w	wall resistance, $(m^2K)/W$
T	temperature, K
U	overall heat transfer coefficient, $(m^2 K)/W$
V	volume, m^3
v	specific volume, m^3/kg
$\dot{W}$	power, W
We	Weber number
x	vapor quality

Greek symbols

Δ	difference
δ	wall thickness, m

ε	effectiveness
Φ	frictional multiplier
φ	surface enlargement factor
η	efficiency
λ	thermal conductivity, W/(m K)
μ	viscosity, Pa/s
ρ	density, kg/m^3
$\bar{\rho}$	density of the two-phase fluid, kg/m^3
σ	surface tension, N/m

Subscripts and superscripts

amb	ambient
c	cold/ dead volume
cd	condenser
cw	cooling water
eg	exhaust gas
ev	evaporator
ex	exhaust
exp	expander
f	fluid
g	gaz
Go	gas only fraction
h	hot
is	isentropic
in	internal
l	liquid
Lo	liquid only fraction
loss	mechanical losses
max	maximal
mec	mechanical
min	minimal
net	net power
pp	pump
th	two-phase
thr	throat
tot	total
s	swept
su	supply
v	vapuor/isochoric
vol	volumetric
w	wall

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