Experimental investigation of the valorization of the waste heat of a gasoline engine based on a Rankine cycle power system

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Abstract:

Europe is currently the leader on the reduction of CO_2 emissions and its target for 2020 is 95 g of CO_2 per km. Cars roughly represent 12% of the CO_2 emissions in EU. Passenger car gasoline engines present the major losses in thermal energy (60% in engine coolant and exhaust gases). This paper focusses on the waste heat available in the exhaust gases only because of their high exergy content. A Rankine cycle test-rig has been built and coupled to the exhaust gases of a passenger car gasoline engine. The performance of each component (evaporator, scroll expander and gear pump) are described and discussed. The net efficiency of the Rankine cycle reaches 6% with a net power of 885 W. Models are validated based on experimentation to predict the performance of the Rankine cycle. This model is coupled with a car model to obtain the gain in performance on a real driving cycle.

Keywords:
Waste heat recovery, Rankine cycle, Experimental investigation, Gasoline engine, Passenger car

1. Introduction

According to the European directive, cars are responsible for around twelve percent of the total EU emissions of CO2 [1]. Some regulations on the emissions of CO2 of cars have then been proposed. Among others, Europe wants to reach a goal of 95 g/km of CO2 on average on a car manufacturer fleet in 2020. Roughly one fourth of the combustion energy is useful. Generally, the major losses during the combustion are known to be heat losses in the engine coolant and heat losses in the exhaust gas. One solution to reduce the car fuel consumption is to reuse the waste heat released in the exhaust gas or in the coolant fluid. The present work only focuses on the recovery of the energy lost in the exhaust gases using a Rankine cycle power system.

The use of a Rankine cycle to recover waste heat from a thermal engine is not new and can be dated back to the 1920s with a locomotive application [2]. This solution has rapidly been abandoned due to the low price of coal compared to diesel but the interest in Rankine cycle in waste heat recovery rose again during the oil crisis in the seventies and some systems have been developed for trucks [3] or boats [4]. After the oil crisis, that technology fades away again until recent years. Most of the car and truck manufacturers [5-7] are putting large efforts in the development of a Rankine cycle power system based on waste heat recovery technology. Indeed the 2020 EU directive requires more and more efforts for the car manufacturers to meet the goals. A recent report [8] also states that the market is expected to grow strongly in the next few years but it will be mostly around Rankine cycle or turbo-compound technology. Rankine cycle waste heat recovery technology is expected to help decrease fuel consumption by 5 to 10 percent according to various sources [6, 9]. A recent study [10] has shown the interest for the Rankine cycle compared to other technologies in terms of efficiency, costs, maturity and packaging (Fig. 1).

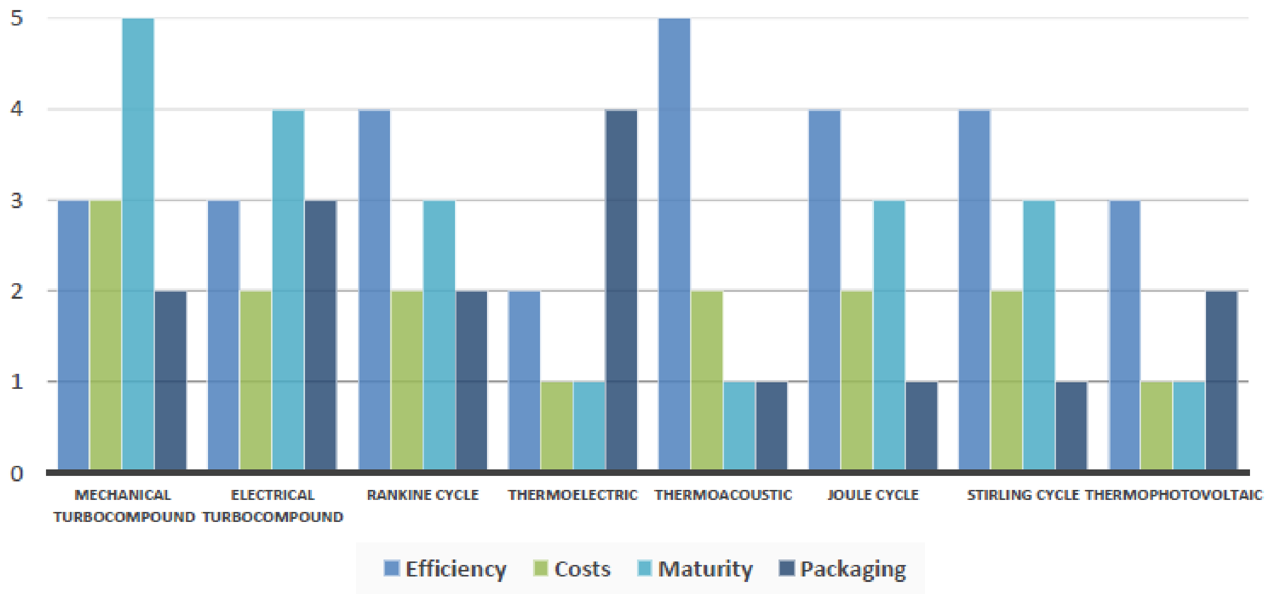

Fig. 1. Comparison of different technologies for waste heat recovery on exhaust gases [10]).

The aim of the present paper is to characterize the performance of the components in order to predict accurately the performance on a driving cycle and the gain in fuel consumption. It is composed of 5 sections: section one is a short introduction with the context, section two describes the components of the test-rig, section three shows the experimental performance of each component and the global performance of a nominal point, section four describes shortly the performance on this Rankine cycle on a normalized driving cycle, section five concludes and discusses the prospects for the project.

1.2. Methodology and description of the test-rig

The test bench includes a gasoline engine (115 kW) and an open Rankine cycle (the inlet of the pump is not connected to the outlet of the heat exchanger ”condenser”) (Fig. 2). The working fluid is pure demineralized water without oil.

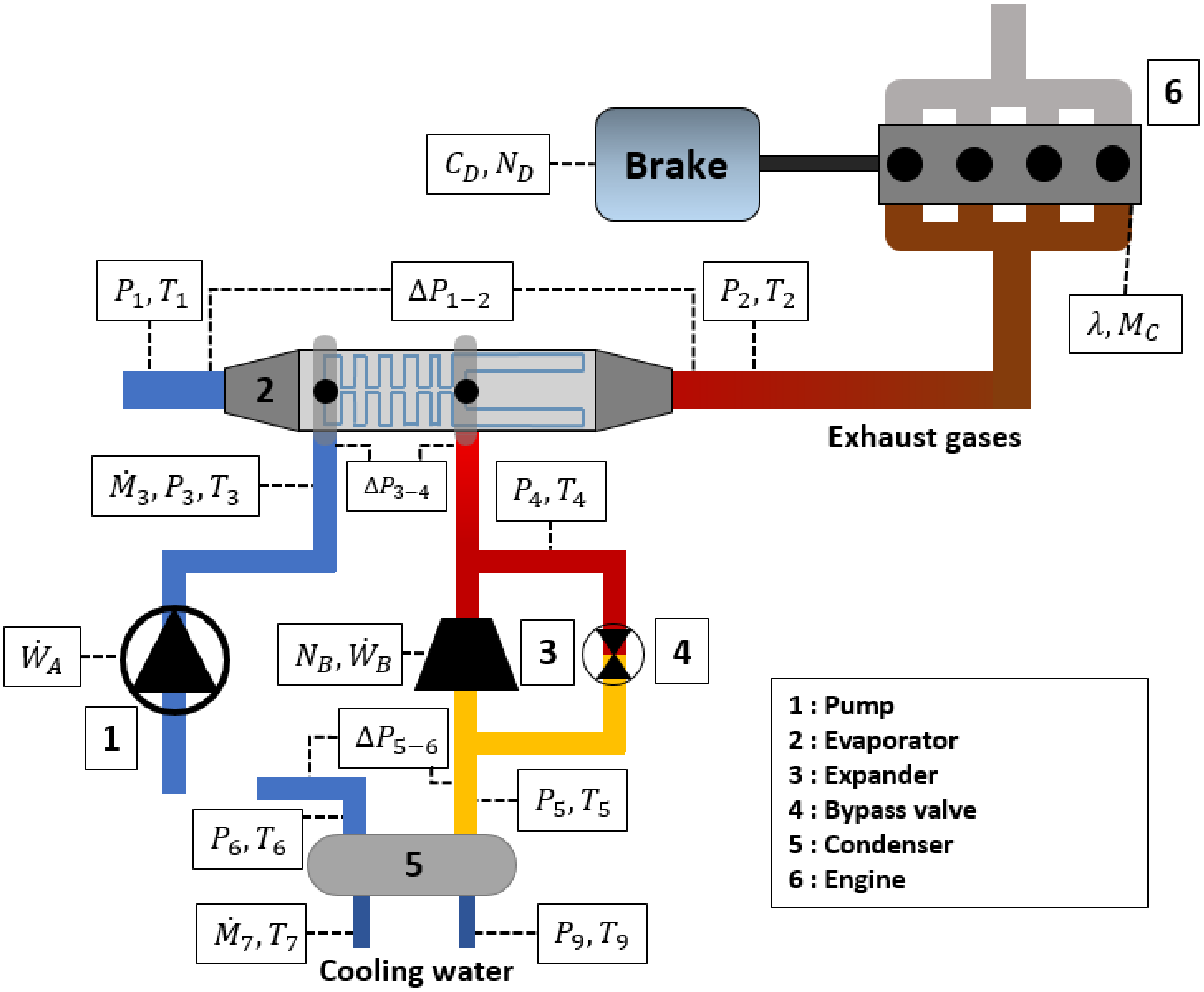

Fig. 2. Hydraulic scheme of the Rankine cycle connected to the exhaust gases of the engine

The Rankine cycle consists of a volumetric gear pump, an evaporator, a scroll expander and a brazed plate condenser. The geometry of the tailor-made hermetic oil-free scroll expander has been optimized in terms of supply port, volume ratio, swept volume and compacity based on a normalized cycle (WLTC) (Legros et al, 2015). The technical specifications for each component are detailed in Table 1. Two different evaporators have been tested: a helicoidal counter current and a hybrid current. Hybrid current heat exchanger allows to decrease the maximum metal wall temperature and therefore its durability. The test-rig is fully instrumented with pressure, temperature, flow, watt, speed meters and internal pressure measurements inside the scroll.

Table 1. Technical data of the components

Component	Parameter	Value	
Gear Pump	Rotational speed [RPM]	3000	
	Max. flow [g/s]	20	
	Swept volume [cm^3]	0.5	
Evaporator	Type of evaporator	Counter Current (CC)	Hybrid Current (HC)
	Mass [kg]	x	1.77.x
	Volume [dm^3]	y	0.83.y
	Water exchange area [m^2]	z	0.36.z
Scroll	Swept volume [cm^3]	8	

expander	Rotational speed [RPM]	15000
	Maximum temperature [°C]	250
	Maximum pressure [bar]	20

2. Experimental results

First, the performance of the pump, the evaporator and the expander are shown individually. This is followed by the description of the global performance of a nominal point.

2.1. Pump

(1) gives the isentropic efficiency of the pump, while (2) gives its volumetric efficiency. The theoretical mass flow rate is computed with (3).

$$\varepsilon_{is} = \frac{\dot{V}_W(P_{ex} - P_{su})}{\dot{W}_{el}} \quad (1)$$

$$\varepsilon_{vol} = \frac{\dot{m}_{meas}}{\dot{m}_{th}} \quad (2)$$

$$\dot{m}_{th} = \rho . N . V_{swept} \quad (3)$$

Both isentropic (Figure 3a) and volumetric efficiencies of the pump are detailed (Figure 3b). The performance is relatively high for such a small power range (~100 W) with an isentropic efficiency up to 44% and a high and rather constant volumetric efficiency. However, the volumetric efficiency is dependent on the pressure drop, particularly for low rotational speeds (<1000 RPM). The isentropic efficiency grows monotonically with the pressure drop mainly because of the higher motor efficiency at higher load. This is true except for low rotational speeds where the leakages are predominant.

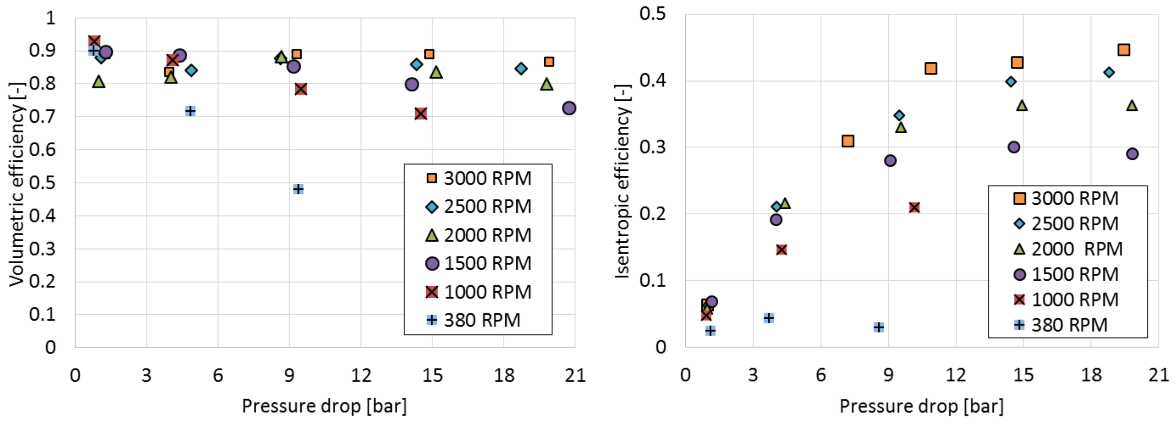

Fig. 3. Performance of the gear pump a) volumetric efficiency b) isentropic efficiency

2.2. Evaporator

In order to compare both evaporators, an efficiency criteria has been defined and is given by the ratio between the transferred power and the maximal transferable power (pinch-point equal to 0 K). This power is defined by (4). This equation is only valid if the pinch point is located on the liquid saturation line which is always the case since the temperature of the exhaust gas is very high compared to the temperature of the working fluid.

$$\dot{Q}_{max} = \dot{M}_{gas} Cp_{gas}\left(T_{gas,su} - T_{w,sat,ex}\right) + \dot{M}_{w}(h_{w,sat,ex} - h_{w,su})) \qquad (4)$$

Fig. 4. introduces the efficiency of each evaporator in function of the kinetic energy per unit of volume of the exhaust gas. For the HC evaporator, kinetic energy becomes bigger with the temperature or the mass flow rate of the exhaust gas and the efficiency is increasing with the kinetic energy. Conditions on the working fluid side are not constant and they play also an important role in the efficiency. The efficiency of the hybrid current evaporator is high (up to 92%). On the contrary, the efficiency of the counter current evaporator is rather constant (around 75%) and slightly decreases with the kinetic energy because of the high pressure drop on the water side (see below).

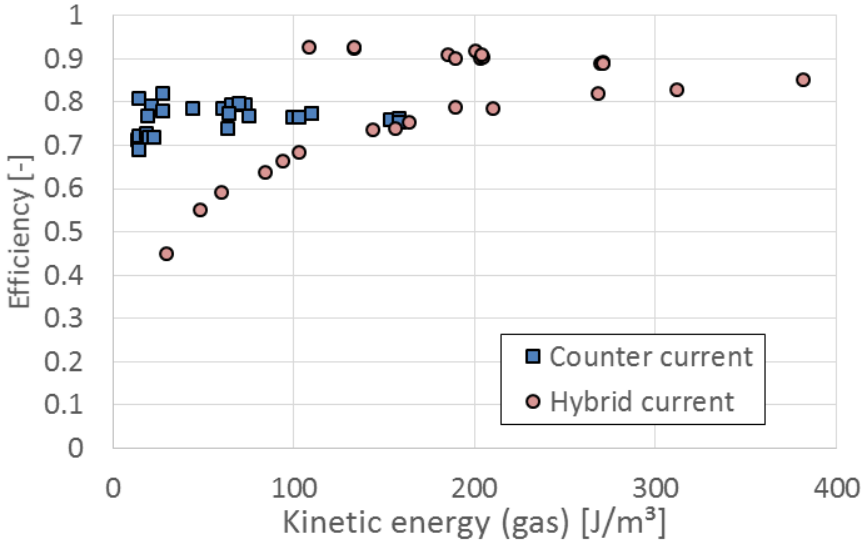

Figure 4: Evaporator efficiency

A second criteria used for comparison is the pressure drop on the gas side. Indeed, the engine is quite sensible to the counter pressure at the exhaust of the engine. The evolution of the pressure drop with the kinetic energy of the gas at the inlet of the evaporator is depicted in Fig. 5(a). Evaporator *HC* shows lower pressure drops due to the high free flow area. However, the pressure drop of the evaporator *CC* is not very big neither.

Pressure drop on the working fluid side is presented for both evaporators in Fig. 5(b). Pressure drop in evaporator *CC* increases very quickly with the mass flow rate. It seems to be majorly due to the internal construction of the heat exchanger. The cross section area is constant all along the pass of the working fluid whereas evaporator *CC* has a variable cross sectional area. This high pressure drop leads to higher working fluid pump consumption, lower evaporator efficiency and higher operating pressure in the cycle.

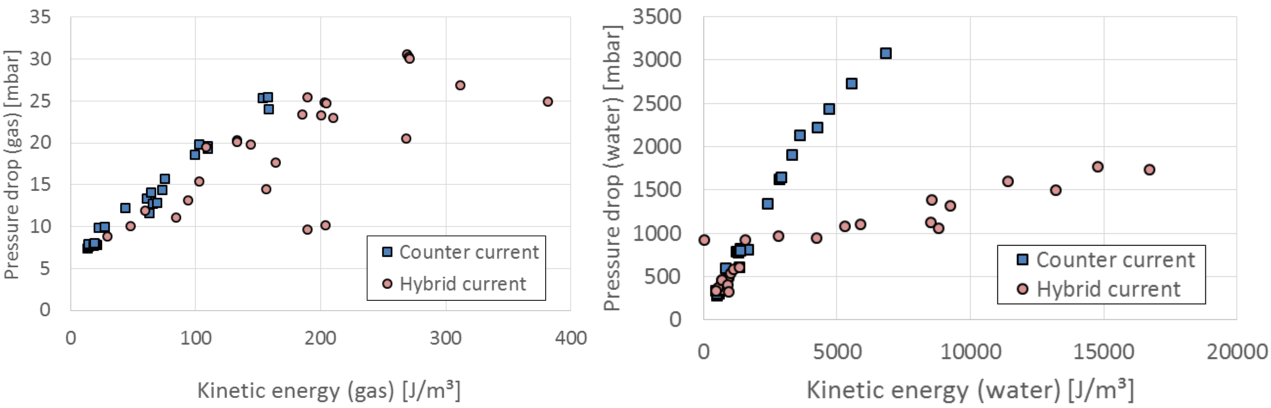

Figure 5: Pressure drop of the evaporator: a) Exhaust gas side b) Water side

2.3. Expander

The performances of two generations (V1 and V2) of scroll expanders prototypes are compared: the second generation (V2) differs from the first generation with a reduced lateral clearance (20%). The isentropic efficiency of the scroll is given by (5) where $h_{ex,s}$ is the enthalpy computed with the exhaust pressure and the supply entropy.

$$\varepsilon_{is} = \frac{\dot{W}_{el}}{\dot{m}_w(h_{su} - h_{ex,s})} \qquad (5)$$

The performance of the scroll are given in terms of isentropic efficiency and filling factor, the inverse of volumetric efficiency (2) – (Fig. 6). The second version of the scroll could not be tested extensively because of scroll corrosion.

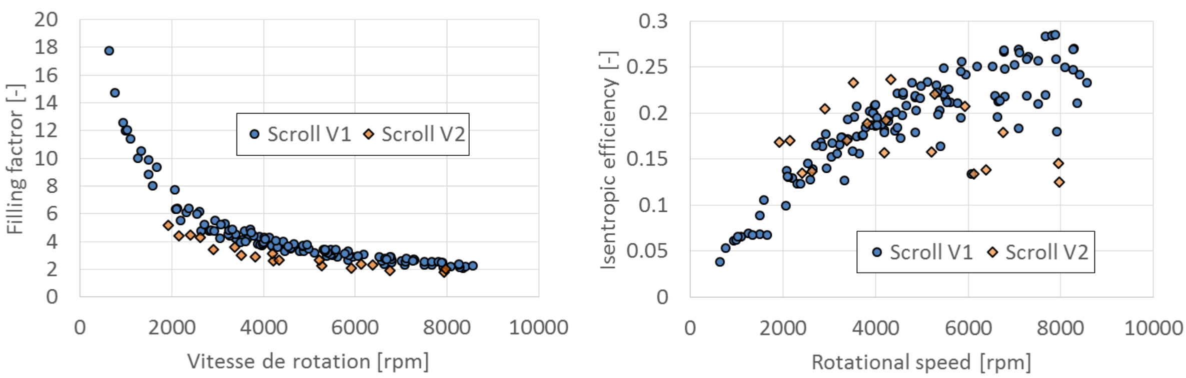

Figure 6: Performance of the scroll expander a) Filling factor and b) isentropic efficiency.

The filling factor is always larger than 1.75 and is very dependent on the rotational speed which gives an idea of the importance of the leakage with this oil free scroll. However, the second version of the scroll (V2) presents a lower filling factor thanks to its lower lateral clearance. The maximum isentropic efficiency reaches 28% for the first version of the scroll and 24% for the second version. A maximum is observed in the isentropic efficiency for the second scroll (V2) around 4000 RPM. At this optimum, the mechanical losses (increasing with the rotational speed) are of the same order of magnitude than the leakage losses (decreasing with the rotational speed).

2.4. Global performance

A typical nominal point is presented through a Ts diagram (Fig. 7.). With exhaust gases at 555°C, 15 kW are transferred to the working fluid through the evaporator. The expander produces 950 W with an isentropic efficiency of 25% while the pump consumes 65 W. It leads to a net electrical power of 885W and a net global efficiency of 5.9%. This electrical production is higher than the maximal electrical consumption of a passenger car (~600 W). However, it is interesting to improve the performance of the components (mainly the expander) for two reasons: to produce decent level of electricity at part load and to fulfil higher energy consumption of hybrid cars.

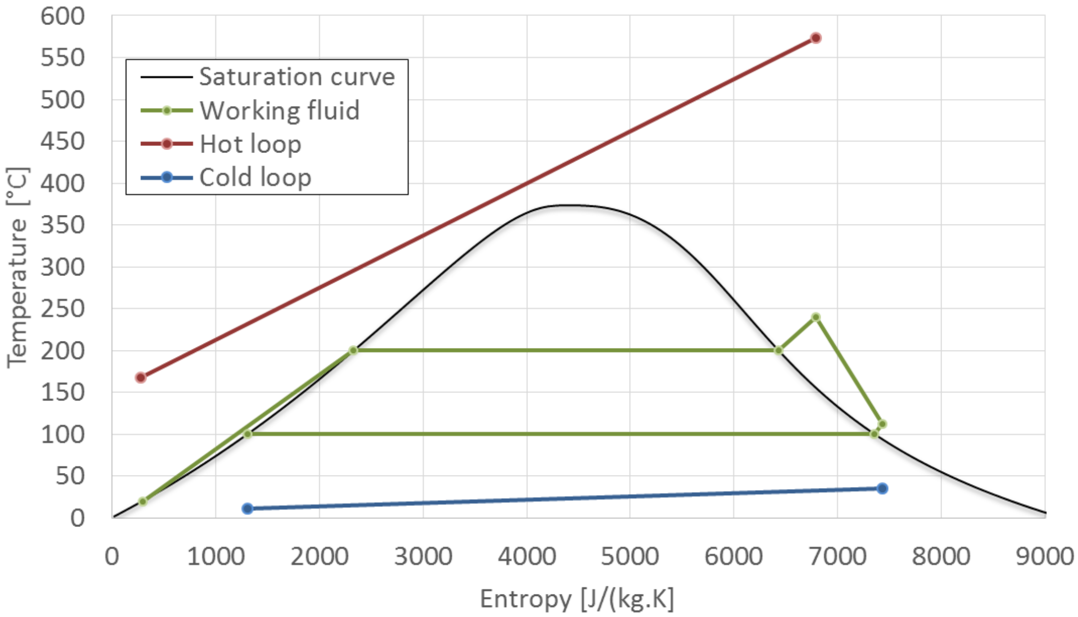

Figure 7. T-s diagram of the Rankine cycle.

3. Model prediction - Calibration cycle

Semi-empirical models are calibrated based on the measurements for each component. These models allow the prediction of the performance with a decent accuracy outside of the calibration dataset. The models for the pump, the exchangers and the expander are described more in details in [11]. The Rankine cycle is modelled by assembling all these components together. It is coupled with a car model that allows predicting the performance of the ORC on a real driving cycle (NEDC). The simulation predicts a possibility of CO2 emissions reduction of 6.7 g/km and reduction of BSFC (Brake Specific Fuel Consumption) up to 5%.

4. Conclusions and prospects

The work presents experimental results of innovative components (prototype of pump, evaporators and tailor-made scroll) for waste heat recovery on the exhaust gas of a passenger car engine with a complete analysis. Based on those experimental results, models can be developed to provide a realistic assessment of the system performance on a driving cycle. In the future, more tests should be investigated:

- A third version of the scroll expander with optimized self-lubricated material for the scroll set. It should allow lower leakages and lower friction losses.
- A mixture of water and oil should be investigated to reduce the filling factor.
- A mixture of ethanol and water instead of pure water to avoid the freezing of the working fluid.

Nomenclature

BFSC Brake Specific Fuel Consumption
CC Counter current
Cp Specific heat [J/(kg.K)]
FF Filling factor
h Enthalpy [J/(kg.K)]
HC Hybrid current
N Rotational speed [Hz]
NEDC New European Driving Cycle
P Pressure [bar]
$\dot{Q}$ Heat flux [W]
$\dot{V}$ Volumetric flow [m^3/s]
$\dot{W}$ Power [W]
$WLTC$ Worldwide Light Test Cycle

Greek symbols

ρ Density [kg/m^3]
ε Efficiency [-]

Subscripts and superscripts

el electrical
ex exhaust
is isentropic
max maximum
meas measured
th theoretical
s isentropic
sat saturation
su supply
vol volumetric
w water

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