Innovative architecture to valorize the waste heat of a passenger car through the use of a Rankine cycle

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ABSTRACT

It is well known that passenger car gasoline engines have large thermal energy losses (roughly 60% between engine coolant and exhaust gases). This paper presents a new architecture for an organic Rankine cycle that uses the engine coolant as the cycle working fluid and exploits waste heat from both the coolant and exhaust gases. This configuration has the potential for lower cost, complexity, and environmental impact compared to conventional approaches. The main losses of the proposed system are taken into account in the analysis: under and over-expansion of the expander, by-pass flow, and finite heat exchange. Two working fluids are considered: a mixture of water and ethanol and a mixture of glycol and water. The cooling engine loop maximum pressures and temperatures are limited to 20 bars and 200°C based on new improvements for passenger vehicles. A parametric analysis is performed using the simulation that to identify the main losses of the proposed architecture and determine optimal control variables (fraction of by-pass flow and pressure) that lead to the best performance. Performance is presented as a function of the speed of the vehicle and results show that greater benefits in terms of fuel consumption are realized for speeds higher than 75 km/h. In addition, the consumption improvements are analyzed for two standard driving cycles (WLTP and NEDC) and are compared with other classical architectures. The maximum reduction in energy consumption for the WLTP cycle is estimated to be 6% for optimal control.

1. INTRODUCTION

According to the European directive, cars are responsible for around twelve percent of the total EU emissions of CO_2 (European Union, 2014). Approximately one fourth of the combustion energy is useful. Generally, the major losses during combustion are known to be heat losses to the engine coolant and exhaust gas. One solution to reduce the car fuel consumption is to reuse the waste heat released in the exhaust gas and/or in the coolant fluid. The present work focuses on use of waste heat to drive a Rankine cycle power system, which has been widely demonstrated to be the most suitable technology (Legros et al. (2014)).

The most investigated heat sources for the Rankine cycle are the exhaust gas (EG) and the engine coolant (CE). Table 1 compares the advantages and disadvantages of both heat sources. The exhaust gas benefits from high temperature (high theoretical Carnot efficiency) and good performance at cold start (low inertia). However, the disadvantage of the EG versus the CE are: (i) a decrease of the available energy because of the limitation on the exhaust gas temperature to 120°C (condensation issues); (ii) large exergy destruction due to expander temperature constraints; (iii) an additional power consumption due to the pumping losses because of the addition of a heat exchanger in the exhaust gases (see Section 3.1), and low availability of waste heat at part-load conditions.

Table 1: Comparison between two waste heat sources in an ICE.

Architecture	Exhaust gas (EG)	Cooling Engine (CE)
Energy on a driving cycle	low	high
Part load performance	low	+
Pumping losses produced with the additional heat exchanger in the exhaust gases	yes	none
High temperature (exergy/efficiency)	yes	no
Cold start	favourable	unfavourable

Recent works (Ziviani et al., 2017; Arnold et al., 2017) have investigated the feasibility of employing the cooling engine fluid as the working fluid in the Rankine cycle. This solution has the potential to decrease the cost, complexity, and environmental impact. Ziviani et al. (2017) obtained a theoretical brake power improvement of up to 4.5% by using the Exhaust Gas recirculation (EGR) and tail pipe as heat sources, as shown in Figure 1 (left). The authors considered a very low Rankine mass flow rate (m_2) (<5%) compared to the mass flow rate of the cooling engine (m_{tot}). Arnold et al. (2017) achieved a 4.5% brake specific fuel consumption decrease by using an ethanol mixture as a working fluid (and cooling engine fluid) for a heavy-duty long-haul truck application.

Combining both ideas (cooling engine fluid as working fluid and utilization of EG and CE as heat sources for the Rankine cycle) leads to an innovative architecture shown in Figure 1 (right). In particular, a pump imposes the working fluid flow circulating in the engine (m_{tot}). Once pre-heated in the engine, a given fraction of the working fluid (m_2) can recover heat from the exhaust gases and evaporator before generating work through the expander. Afterwards, this fraction of working fluid (m_2) is mixed with the expanded by-pass flow (m_1) before being condensed in the radiator of the car. The by-pass flow is required for ensuring adequate engine cooling and also to optimize the state of the fluid at the inlet of the expander.

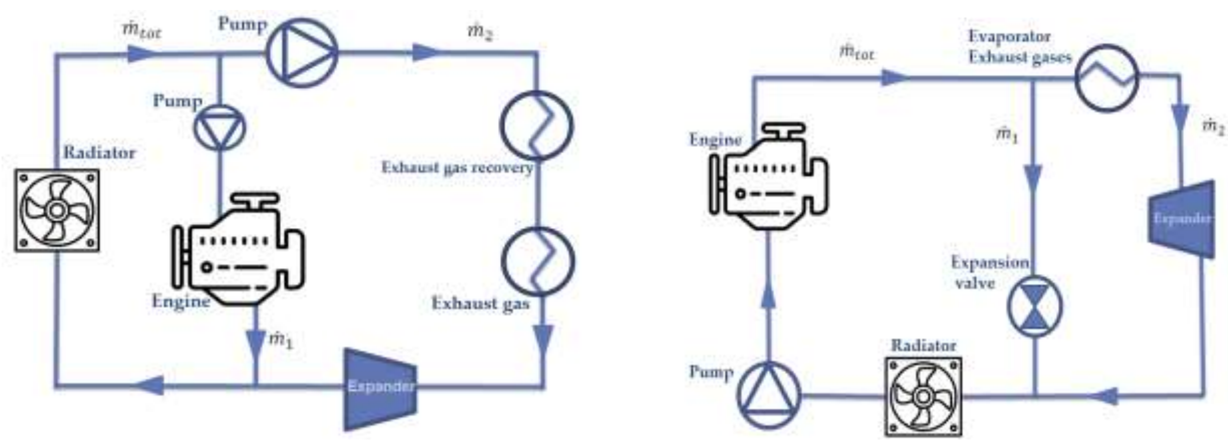

Figure 1: (left) Layout of the architecture proposed by Ziviani et al. (2017); (right) layout of the innovative architecture.

This innovative architecture is investigated in the case of a passenger car with a 157 kW gasoline engine (displacement of 1598 cm^3) without exhaust gas recirculation (EGR). The exhaust gas temperature varies between 400°C and 800°C, whereas the cooling engine temperature can vary between 70°C and 200°C (at maximum 20 bar).

Section 2 presents an overview of the overall modeling approach and the performance criteria, whereas Section 3 presents results showing the influence of the two main parameters, an analysis of the losses, and the performance of the cycle for two driving cycles.

2. METHODOLOGY

2.1 Hypothesis and constraints

The considered architecture is a classical Rankine system (Figure 2). In this figure, the pump (1) delivers the mass flow rate to the evaporator where the water is vaporized and overheated (2). Following this, the vapor is expended in the expander to produce power and enters into the condenser (3). Finally, the condenser transforms the vapor coming from the expander into a sub-cooled fluid which comes back to the pump (1).

The hypotheses and constraints are listed in Table 2. The considered engine presents three limits for the working fluid: pressure below 20 bar, temperature below 200°C and working fluid in liquid state at the exhaust of the cooling engine. The engine efficiency is supposed independent of temperature of the coolant. The condensation temperature in the radiator is fixed to the saturation temperature at 1 bar, the performance of the evaporator is evaluated through a calibrated semi-empirical model and the pump isentropic efficiency (Equation (1)) is assumed to be equal to 70%. The expander model takes into account over and under-expansion losses based on the volume ratio of the machine (maximal isentropic efficiency of 60% - Equation (2)). Two mixtures are considered: water ethylene glycol (50% mass) and water ethanol (50%).

$$\varepsilon_{pp,is} = \frac{\dot{V}_{wf}(P_{ex} - P_{su})}{\dot{W}_{el}} \tag{1}$$

$$\varepsilon_{exp,is} = \frac{\dot{W}_{el}}{\dot{m}_{wf}(h_{su} - h_{ex,is})} \tag{2}$$

The two main parameters to optimize are the working fluid pressure (P_{ev}) and the mass flow rate in the Rankine cycle (m_2). REFPROP (SUN et al., 2003) is used to retrieve the thermo-physical properties of the mixtures.

Rather than use the Rankine cycle mass flow as an optimization value, an alternative pseudo-quality parameter is defined that leads to improved robustness in the model solution. The pseudo-quality is equal to the working fluid vapor quality leaving the evaporator when it is between 0 and 1. However, it is allowed to be greater than 1 when the evaporator exit condition is superheated. Each 1 K degree of superheat corresponds to 0.04 increment of superheat such that a quality of 1.2 corresponds to a superheating of 50 K.

Table 2: Hypothesis and constraints

Contraints	Value
Temperature at the exhaust of the engine cooling	Lower to the saturation pressure
Working fluid pressure	< 20 bar
Rankine cycle flow (m_2)	[0:m_{tot}]
Working fluid	Water glycol (50%) or water ethanol (50%)
Pump isentropic efficiency	70%
Evaporator	Calibrated semi-empirical model
Expander	Scroll (r_v=3, $\varepsilon_{exp,is}$=0.60) or Piston (r_v=3, $\varepsilon_{exp,is}$=0.60)
Condensation pressure	1 bar

2.2 Performance criteria

Several performance criteria are used to present performance of the new Rankine cycle configuration:

- $ratio_m$ is the ratio of mass flow rate used in the Rankine (m_2) to the total mass flow rate (m_1) (Equation (3)).

$$ratio_m = \frac{\dot{m}_2}{\dot{m}_1 + \dot{m}_2} \tag{3}$$

- The isentropic efficiency of the expander (equation (2)), including under and over-expansion losses (Ziviani et al, 2017).

- $\Delta\dot{Q}$, is the ratio of the thermal power that the condenser has to dissipate because of the addition of the EG thermal power to the dissipated thermal power without the EG thermal power ($\dot{Q}_{ce}$):

$$\Delta\dot{Q} = \frac{(1 - ratio_m)\,\dot{Q}_{ce} + (ratio_m\dot{Q}_{ce} + \dot{Q}_{eg}))(1 - \eta_{ORC})}{\dot{Q}_{ce}} \tag{4}$$

- $ratio_{gas}$, is the ratio between the thermal power used in the Rankine cycle from the exhaust gases and the thermal power available in the exhaust gases (equation (5) - Figure 2). The minimum exhaust gas temperature is set to 120°C to avoid pollutant emissions issues.

$$ratio_{gas} = \frac{\dot{m}_{gas}Cp_{gas}(T_{gas,after\ catalyser} - T_{sat}(P) - PP_{ev})}{\dot{m}_{gas}Cp_{gas}(T_{gas,after\ catalyser} - 120)} \tag{5}$$

- The ORC power production is defined as the expander production minus the pump consumption.
- The cycle efficiency results in the division of the ORC power production by the total thermal power of the considered heat source.
- The percentage of fuel consumption decrease is assumed to be the ORC power production divided by the engine mechanical power.

For each figure featuring the vehicle speed in this paper, the average torque for a given speed in a driving cycle was used to evaluate the mechanical, exhaust gas and cooling engine powers.

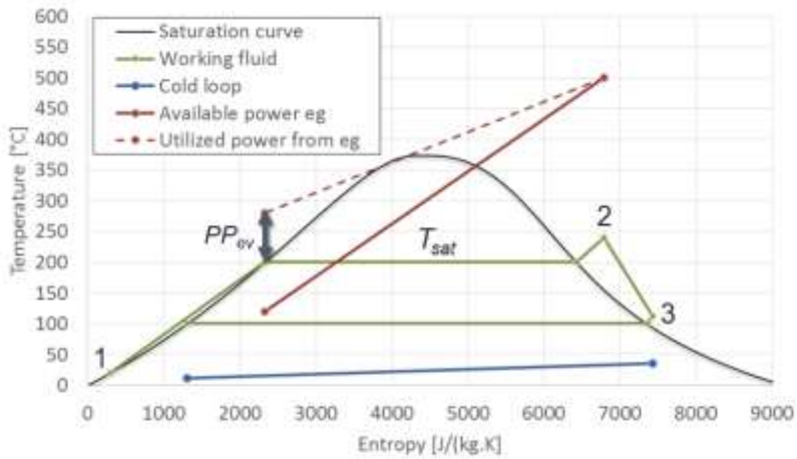

Figure 2: Difference between the thermal power from the Rankine cycle from the exhaust gases and the thermal power available in the exhaust gases.

3. RESULTS AND DISCUSSION

This section is divided into three parts. First, the two optimization parameters (evaporation pressure and quality) are varied within two simple case studies to understand their influence on the cycle performance. Then, the maximized performance of the cycle with optimized parameters is presented. The influence of the working fluid and of the expander is also highlighted. Finally, the performance is compared with other architectures based on a driving cycle analysis.

3.1 Influence of expander inlet pressure and quality

Two parametric studies are conducted to assess the influence of the expander inlet pressure and quality on the cycle performance criteria and net power output. The working fluid considered is the water-ethanol mixture.

At first, the influence of the evaporation pressure is considered. In particular, the pressure was varied between 2 bar and 20 bar by fixing the inlet quality equal to 0.9, the volume ratio of the expander equal to 3 (reference value), the speed of the vehicle constant at 50 km/h. The results are shown in Figure 3. To be noted is that the ratio of flow going through the Rankine ($ratio_m$) increases with the pressure. The increase of the evaporation pressure allows greater thermal energy recovery from the cooling engine. However, increasing the evaporation pressure (above 3.6

bar) leads to under-expansion losses in the expander. The recoverable energy from the exhaust gases decreases significantly with the evaporation pressure, as explained in Figure 2. Thus, the pump, expander, and net power increase monotonically with the evaporation pressure, as shown in Figure 3.

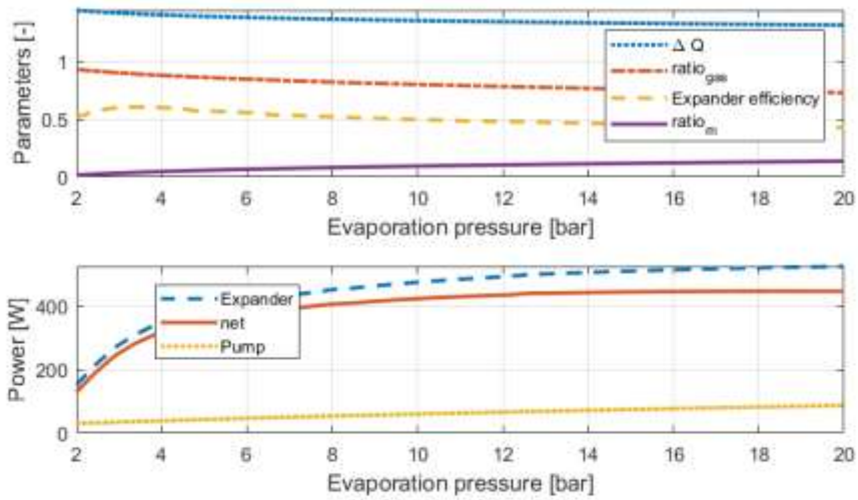

Figure 3: Influence of the evaporation pressure on the performance of the system. The performance criteria are shown in the top figure; whereas, the power output of the expander, input power of the pump, and the net power are plotted in the bottom figure.

The influence of the quality at the inlet of the expander on the performance is reported in Figure 4. In this case, the pressure is fixed to 20 bar, the volume ratio of the expander to 3, and the speed of the vehicle is 90 km/h. The quality at the inlet of the expander does not influence the pump consumption or the recoverable heat in the exhaust gases, as shown in Figure 4. In this example, it appears that the maximum of power occurs at the lowest quality (0.25). More generally, depending on the operating point, when the quality at the inlet of the expander is increased, two opposite effects can be highlighted: the expander efficiency increases, but the recoverable thermal energy from the exhaust gas decreases. Therefore, an optimum quality is found that leads to maximum net Rankine cycle production.

The main conclusions from Figure 3 are that increasing evaporation pressure:

- increases the pump consumption,
- slightly decreases the thermal energy that the radiator needs to dissipate,
- increases the quantity of thermal energy recoverable from the cooling engine,
- has an optimum to avoid under- or over-expansion losses in the expander,
- decreases the thermal energy recoverable from the exhaust gas.

From Figure 4, it appears that increasing quality

- does not influence the pump consumption or the thermal energy recoverable from the exhaust gas, or the thermal energy that the radiator needs to dissipate,
- decreases the thermal energy coming from the exhaust gas,
- increases the isentropic efficiency of the expander.

This analysis shows the influence of the two control variables evaporation pressure and of the quality. The next results presented in this paper are obtained using a genetic algorithm to optimize the net power production of the Rankine by acting on these two variables.

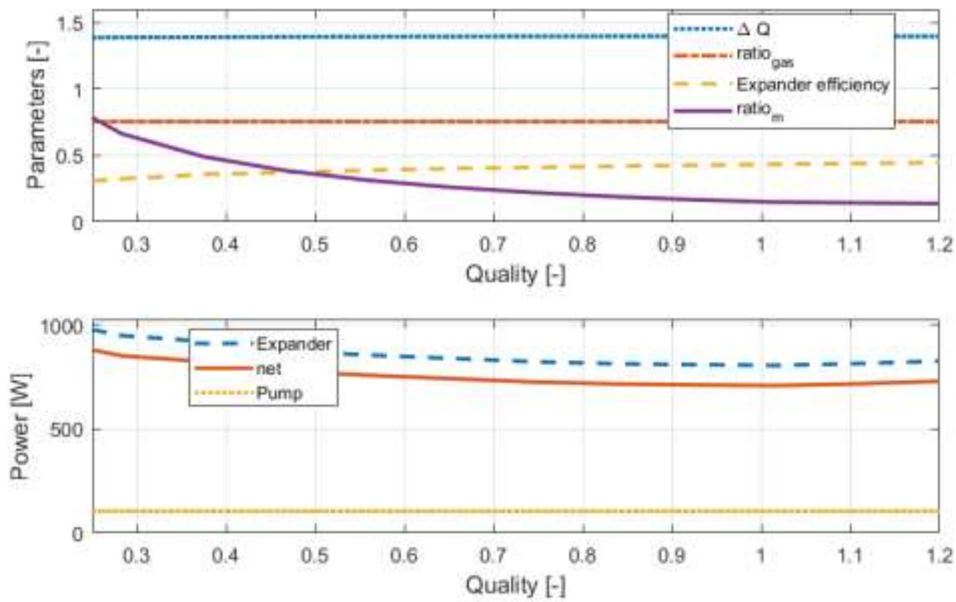

Figure 4: Influence of the quality on the performance of the system. The performance criteria are shown in the top figure; whereas, the power output of the expander, input power of the pump, and the net power are plotted in the bottom figure.

3.2 Losses analysis

The net power output of the system was maximized for different vehicle speeds by optimizing the evaporation pressure and the quality. To understand performance and losses of the Rankine cycle, the following quantities are shown in Figure 5:

- *Qtot* is the sum of the thermal energy content in the exhaust gases and in the cooling engine,
- *-by-pass*, is equal to *Qtot* minus the energy lost in the by-pass (i.e., engine heat rejection to coolant that is not available for recovery by the Rankine cycle),
- *-gas* is equal to *–by-pass* minus the non-recoverable energy in the exhaust gas (equation(5)),
- *exp 100%* is equal to the power produced with a expander having a 100% isentropic efficiency,
- *exp real* is equal to the power produced with an expander having a realistic isentropic efficiency with a volume ratio of 3,
- *net* is equal to the net electrical production with an expander having a realistic isentropic efficiency with a volume ratio of 3 taking into account the pump consumption.

As it can be seen in Figure 5, the net electrical production is rather low compared to the high thermal energy content (Q_{tot}) in the exhaust gas and in the cooling engine. By analyzing Figure 5, it can be seen that the losses are mainly associated with the by-pass losses and the low isentropic efficiency of the expander.

The losses in the by-pass cannot be easily decreased while keeping a high net electrical production because of the inherent constraint of the architecture. However, the expander efficiency can be significantly increased by using a more appropriate volume ratio. Classical scroll expanders typically have a limited volume ratio (up to 4.2 as reported by Lemort and Legros, 2016). Piston expanders can have an internal volume ratio up to 15 which reduces significantly the under-expansion losses in this architecture (Lemort and Legros, 2016). Figure 6 depicts a comparison of net electrical production between an expander with a volume ratio of 3 and 15. The performance improvement with the increased pressure ratio is obvious and, therefore, the next results of this paper will only consider an expansion machine with a volume ratio of 15.
The influence of the working fluid is also compared in Figure 6. The system presents very similar net electrical production with both fluids.

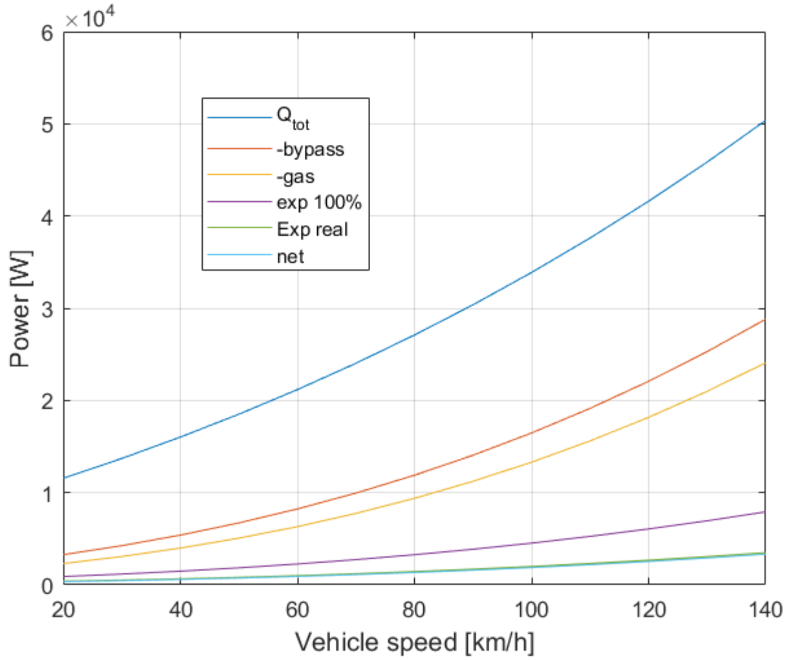

Figure 5: Decomposition of the losses for the new architecture in the case of a volume ratio of 3.

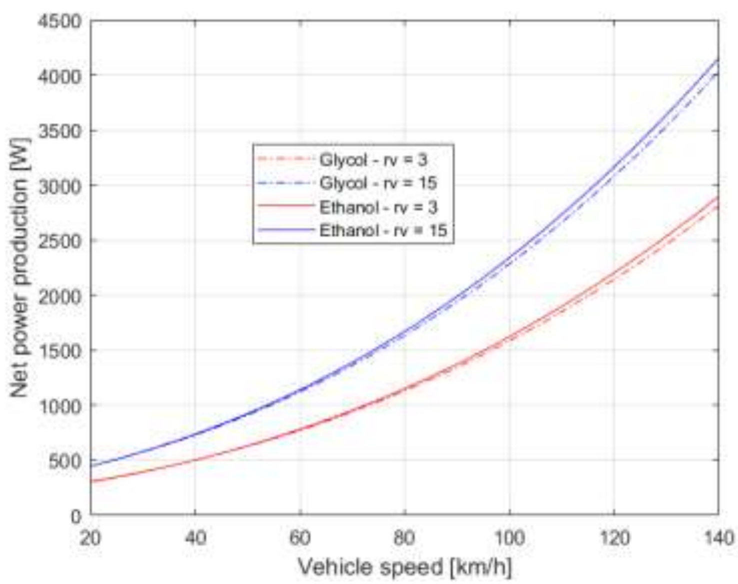

Figure 6: Influence of the volume ratio and the working fluid on the performance for different vehicle speeds.

3.3 Global performance

The proposed cycle architecture (R-CE-EG) is compared with three other architectures reported in the literature:

- ORC-CE: organic Rankine cycle using the cooling engine as the heat source and the radiator as the heat sink.
- R-EG: Rankine cycle using the exhaust gas as the heat source and the cooling engine as the heat sink.
- ORC-EG: organic Rankine cycle using the exhaust gas as the heat source and the radiator as the heat sink.

More details on the layouts, modeling, and performance of these three cycles can be found in (Dumont et al., 2017a; Dumont et al., 2017b, Dumont et al., 2018).

Comparisons of performance between the four architectures are presented in function of the vehicle speed in Figure 7. The top plot shows the engine mechanical power, and the thermal power available in the exhaust gas and in the cooling engine. First, is important to note that the thermal power available from the engine coolant (CE) is higher than the exhaust gas (EG) energy until a vehicle speed of 130 km/h. This leads to better performance of the ORC-CE at lower speeds. It is also important to note that the cycle efficiency of the new architecture (R-CE-EG) is low compared to the other cycles. This is due to the aforementioned constraints inherent in the architecture (by-pass losses mainly). However, because of the two sources for heat recovery, there is greater availability of recovered energy for the R-CE-EG and the net electrical production is higher than the other architectures for vehicle speeds higher than 80 km/h.

In order to evaluate overall performance, driving cycle performance was assessed for the four architectures. The comparisons were performed for two driving cycles (WLTP and NEDC) with two initial conditions:

- Cold start (60 s of inertia for the exhaust gas and 800 s for the cooling engine)
- Hot start (assuming steady state condition from the beginning of the cycle)

The NEDC presents a lower average vehicle speed (33.8 km/h) compared to the WLTP (60.8 km/h). The percentage fuel consumption decrease for each of the energy recovery options relative to the base engine are presented in Figure 8 for the four drive cycle cases. Naturally, the NEDC cycle with low vehicle speeds (i.e. high cooling engine thermal power) is more beneficial for the ORC-CE. However, with this NEDC cycle and a cold start, the new architecture (R-CE-EG) has performance very close to the ORC-CE. For the second driving cycle (WLTP), the R-CE-EG cycle significantly outperforms the other cycles for both the cold start and hot start.

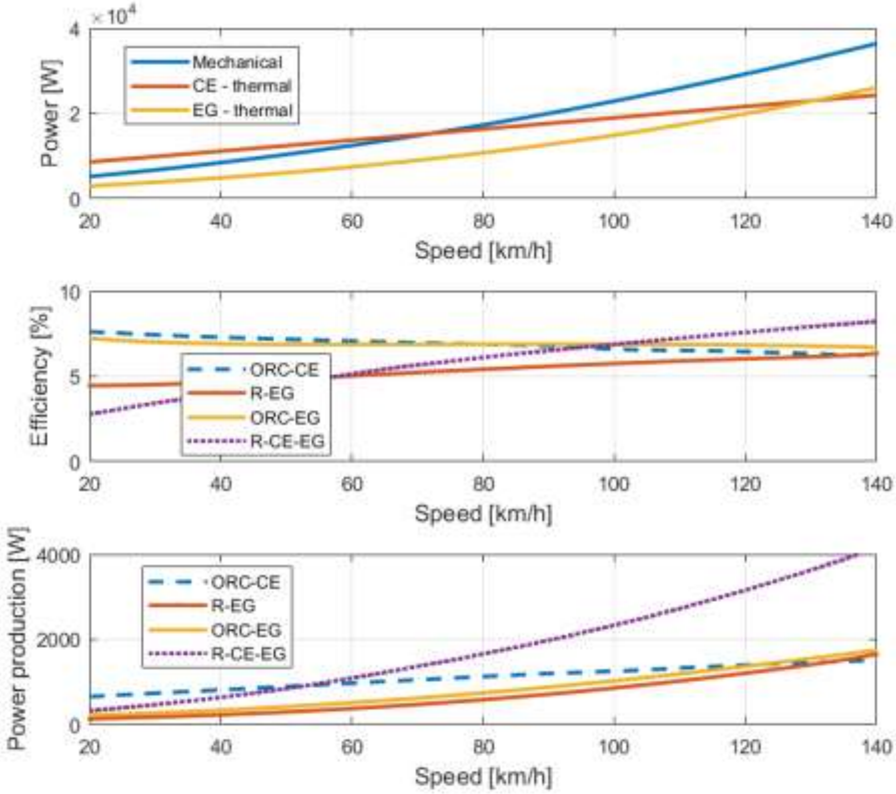

Figure 7: Engine power, net power production of the Rankine cycle and efficiency for 4 different architectures.

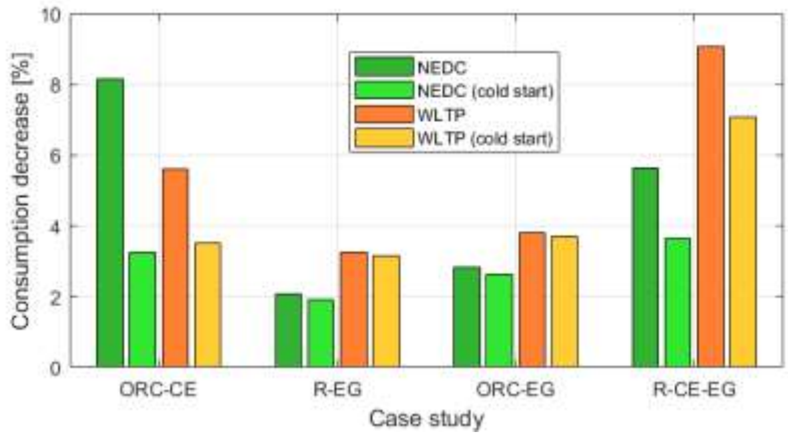

Figure 8: Driving cycle performance for 4 different architectures.

5. CONCLUSION

In this paper, a new architecture for a Rankine cycle that recovers energy from both exhaust gas and engine coolant in engines has been proposed. The cycle is novel in that it employs the cooling engine fluid as the working fluid. The paper presented limitations and constraints of this architecture along with estimates of performance using a simple model. The performances for different driving cycles were assessed and compared with four other classical architectures. The new architecture has better performance than the conventional cycles for speeds larger than 80 km/h. It should be noted that there is room for improvement in the expander design for such a cycle.

It should be note that these results are only preliminary and the new architecture should be investigated in more detail. In particular:

- A test-rig should be built to assess the real performance of the components with different fluids.
- A dynamic model with a realistic controller should be developed to evaluate the real performance for a driving cycle.

NOMENCLATURE

c_p	specific heat	(J/(kg-K))
h	specific enthalpy	(J/kg)
$\dot{m}$	mass flow rate	(kg/s)
P	Pressure	(Pa)
PP	Pinch point	(K)
$\dot{Q}$	Thermal power	(W)
Ratio	Ratio	(-)
r_v	Volume ratio	(-)
$\dot{V}$	Volume flow rate	(m^3/s)
$\dot{W}$	Power	(W)

Acronym

CE	Cooling Engine
EG	Exhaust Gas
EGR	Exhaust gas Recirculation
NEDC	New European Driving Cycle
ORC	Organic Rankine Cycle
R	Rankine cycle
WLTP	Worldwide harmonized Light vehicles Test Procedures

Subscript

ce	cooling engine
Δ	difference
ε,η	efficiency
eg	exhaust gas
el	electrical
ev	evaporator
ex	exhaust
exp	expander
gas	gas
is	isentropic
m	mass flow rate
ORC	organic Rankine cycle
pp	pump
sat	saturation
su	supply
tot	total
wf	working fluid
1	Rankine related flow
2	Bypass related flow

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