Modeling of an Organic Rankine Cycle Waste Heat Recovery system for automotive engine applications

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Abstract. The remarkable investments made by manufacturers over the last few decades have contributed to improving the performance of internal combustion engines in every aspect: lower polluting emissions, greater specific power and thermal efficiency. Despite this, on an average, about 40% of the thermal power theoretically available from the combustion of the fuel is still stored in the exhaust gases and therefore dispersed in the environment. In this work the modeling and validation of a waste heat recovery (WHR) plant will be described, combining the engine with a low temperature Organic Rankine Cycle (ORC) system, in order to investigate the feasibility of this system on board of a vehicle, analyzing the quantity of thermal power recovered and made available in the form of electrical power. The ORC plant is modeled using a 0D/1D thermo-fluid dynamic approach. Starting from experimental tests, a map-based model for the piston pump and the scroll expander has been developed. The model has been validated through the use of a vector optimization technique, exploiting a genetic algorithm (MOGA). Subsequently, this system has been coupled to a spark ignition engine for automotive applications, adapting its speed range to comply with the ORC experimental tests. To have an accurate control over the expander inlet temperature, a bypass circuit and two throttles actuated by a PI controller have been implemented. The simulations were performed by considering 18 engine points at maximum load and different rpm. An average thermal efficiency increase of the system of 2.6% was obtained by introducing the recovery plant, and wide improvement chance can be foreseen in the case of ORC full-power use.

Keywords: Waste Heat Recovery (WHR); Organic Rankine Cycle (ORC); Internal Combustion Engine; 1D simulation.

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1 Introduction

An Internal Combustion Engine (ICE) is capable to convert only a part of the chemical power contained in the fuel. The remaining portion is wasted into the environment through the exhaust gases or the cooling liquid.

Figure 1 shows the energy balance of a typical 1.4 litre 4 cylinder spark ignition internal combustion. The global efficiency can be evaluated as:

$$\eta_g = \frac{E_u}{m_f\, H_i} \tag{1}$$

where E_u is the useful energy, m_f is the mass of fuel and H_i is the lower heating value. The efficiency ranges from 15 to 32% under normal operation. The energy released through the radiator can be 18 to 42% whereas the energy released through the exhaust gases is from 22 to 46%. The peak temperatures of exhaust gases can reach values close to 900 °C where the coolant temperature usually ranges around 100 °C [1].

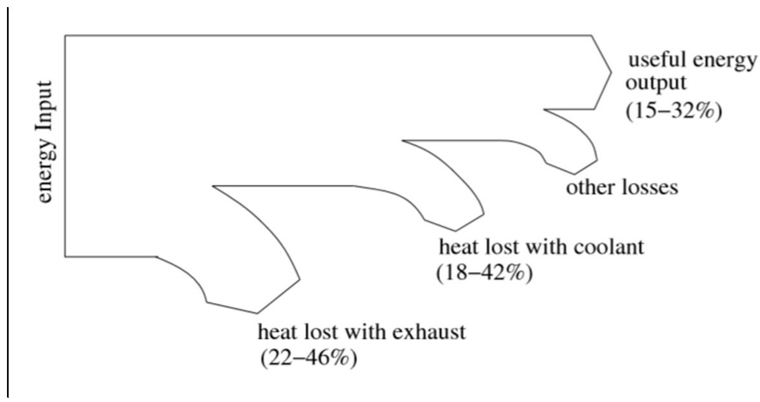

Figure 1 - Energy balance of a 4-stroke spark ignition engine

The use of recovery systems of energy contained in the exhaust gases or in the refrigerant system is a common practice for large engines for marine applications [2] or for electric generators [3] using cogeneration techniques, resulting in a significant improvement of the global efficiency.

Currently, almost all the systems dedicated to the recovery of the exhaust gases of internal combustion engines used for powertrain application are implemented on board of heavy-duty vehicles [4]. This is due to the higher space available, which allows an easy installation of the additional system. Furthermore, large-displacement engines offer significant quantities of exhaust energy and therefore recovery is also facilitated [5].

Indeed, due to the significantly more compact engines and the reduced space on board of passenger cars, there are very few applications of exhaust gas energy recovery systems. The most important implementation attempt was that of the German company BMW, with the project called "Turbosteamer", coupling a 1.8-liter spark ignition engine with a Rankine system. According to the tests carried out, the following were declared: 13 HP and 20 Nm peak returned to the shaft (+ 10%), with increases in efficiency of the order of 15% over long distances (the reference guide cycle was not specified). Although the first results were encouraging, it unfortunately did not find space in series production [6].

In this paper the authors propose the modeling and validation of a waste heat recovery (WHR) plant, combining a 2 litres spark ignition engine with a low temperature Organic Rankine Cycle (ORC) system with R245fa as working fluid, in order to investigate the feasibility of this system on board of a vehicle, analyzing the additional electrical or mechanical power supplied by the system and the increase in global efficiency. Both the engine and the ORC plant are modeled using a 0D/1D thermo-fluid dynamic approach. Moreover, authors highlight the potentiality of this application for automotive, considering the intrinsic non-stationarity of the engine behavior through the use of the non-stationary flow equations. This analysis highlights both the potential (efficiency improvement without a significant increase in size and weight) and the criticality (importance of control and the need to integrate the WHR System into the engine architecture) of the proposed configuration.

Numerous researchers [7] have analyzed different working fluids to convert low-grade heat, focusing on the thermal efficiency [8] and power delivered by the turbine/expander, but also on the economic, safety and environmental aspects [9,10]. For the latter, two parameters are considered: Ozone Depletion Potential (ODP) which measures the degradation of the ozone layer and the Global Warming Potential (GWP) which quantifies the greenhouse effect caused by the working fluid [11]. For heat source temperatures below 120 °C mainly alkanes and traditional refrigerants were investigated, where the latter often contain fluorine and chlorine, so they have unfavorable ODP and GWP values [12]. Currently, hydrofluorocarbons (HFCs) are a popular choice of working fluid for ORCs [13] and the experimental campaign conducted by Desideri et. al [14] in 2016 proved that R245fa is a promising working fluid for a low temperature ORC.

2 System model

Figure 2 shows a schematic representation of the model, whose main characteristics are shown in Table 1.

Table 1 – Main characteristics of the engine model

Type	4 cylinders, 4 stroke
Total Displacement	1998.8 cm^3
Maximum Brake Power (Torque)	96 kW @ 5500 rpm (183 Nm @ 4250 rpm
Ignition	Spark ignition

The engine features the performance of a typical B-segment application, and thus represents a significant case-study.

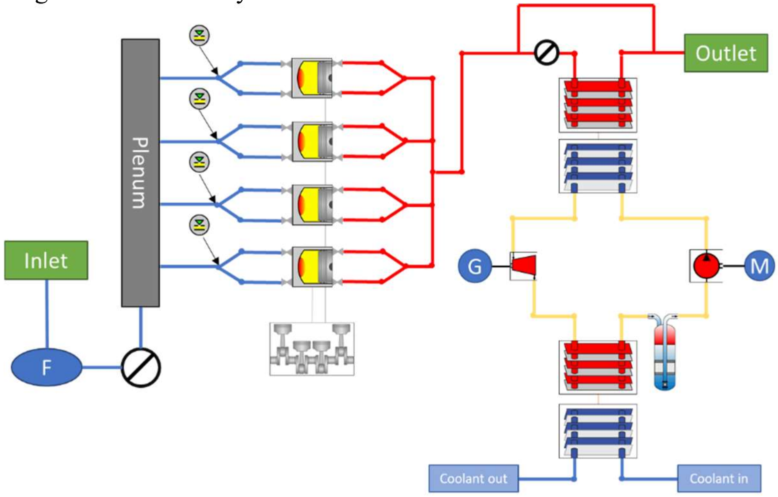

Figure 2 - Schematic representation of the model

For this system, the mass and energy conservation equations are solved as a function of time in the combustion chamber volume [15]:

$$\frac{dm}{dt} = \dot{m}_{in} - \dot{m}_{out} + \dot{m}_{inj} \quad \textbf{(2)}$$

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} - d\frac{Q_w}{dt} + \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{m}_{inj}h_{inj} \quad \textbf{(3)}$$

Equation 2 is the mass balance, in which $\dot{m}_{in}$ is the input mass flow rate, $\dot{m}_{out}$ is the output mass flow rate and $\dot{m}_{inj}$ is the input fuel flow rate.

Equation 3 is the energy balance, in which $-p\frac{dV}{dt}$ is the mechanical energy exchanged between the piston and the flow, $-d\frac{Q_w}{dt}$ is the thermal power exchanged with the cylinder walls, h_{in} is the specific enthalpy of the input mass flow rate, h_{out} is the specific enthalpy of the output mass flow rate and h_{inj} is the specific enthalpy of the input fuel mass flow rate.

The pipes are modeled applying 1-D equations. In particular, the mass, energy and momentum balance are solved:

$$\frac{\partial\rho}{\partial t} + \frac{\partial\rho}{\partial x} + \rho u\left(\frac{1}{\Omega}\frac{\Omega}{dx}\right) = \frac{\partial\rho}{\partial t} + \frac{\partial\rho}{\partial x} + \rho u\alpha_A \quad \textbf{(4)}$$

$$\frac{\partial(\rho E)}{\partial t} + \frac{\partial\rho uH}{\partial x} + \rho uH\left(\frac{1}{\Omega}\frac{\Omega}{dx}\right) = \frac{\partial(\rho E)}{\partial t} + \frac{\partial\rho uH}{\partial x} + \rho uH\alpha_A \quad \textbf{(5)}$$

$$\frac{\partial(\rho u)}{\partial t} + \frac{\partial(\rho u^2 + p)}{\partial x} + \rho u^2\left(\frac{1}{\Omega}\frac{\Omega}{dx}\right) = \frac{\partial(\rho u)}{\partial t} + \frac{\partial(\rho u^2 + p)}{\partial x} + \rho u^2\alpha_A \quad \textbf{(6)}$$

Where ρ, u, p, E and H are the density, speed, pressure, total internal energy and total enthalpy per unit of mass. Furthermore $\alpha_A = \left(\frac{1}{\Omega}\frac{\Omega}{dx}\right)$ represents the section variation of the pipes. The ORC system consists of a scroll expander, a pump and two heat exchangers (Fig.3). Also in this case, the equations (6-8) are solved in the pipes, while a map-based approach was used for the individual components starting from experimental data). The bypass circuit was necessary to avoid reaching too high fluid temperatures for two reasons: the first is for safety reasons, keeping the temperature below the flammability and thermal stability temperatures of r245fa [16]; the second is for plant engineering reasons, given that as the temperature increases, the heat to be removed during the condensation phase also increases, resulting in an increase in the surface of the heat exchanger and the difficulty of integrating it on board the vehicle.

The experimental plant chosen for the modeling of the recovery system has been assembled at the Purdue University, Indiana. The geometric data of the plant and the experimental tests conducted on the individual components and on the complete plant are described in detail in the work [17]. In particular, the geometric data of the heat exchangers and the experimental data of the heat exchangers, pump and expander are described, in addition to the 61 experimental measurements with reference to the complete ORC system.

For the first assembly of the ORC system, in the absence of some information, such as pipe diameters and length, and fluid charge, values taken from the literature were first considered. The model of the ORC plant was initially simulated stand-alone, using hot water as a hot fluid, in order to compare the quantities obtained from the model with the respective experimental quantities in 61 different experimental conditions (Figure 3).

Table 2 shows the mean % error evaluated as:

$$err_{i,\%} = \sum_{j=1}^{n} \frac{|Q_{mod,j} - Q_{exp,j}|}{Q_{exp,j}} \cdot 100 \quad \textbf{(7)}$$

Where Q_{mod} is the value obtained from the simulations of the mathematical model and Q_{exp} is the experimental value in the j-th experimental condition.

The mean percentual error ranges from the 3.49% of the pump mass flow rate and the 6.72 % of the condenser thermal power. Although the average percentage error ranges around5%, the model behaviour is not good enough given that the maximum punctual error is higher than 40%

Table 2 – Mean and maximum error of some quantities

Data	Mean % Error	Maximum % Error
Condenser thermal power	6.72	23.42
Evaporator thermal power	6.96	25.82
Expander power	6.73	40.15
Expander mass flow rate	4.14	22.94
Pump mass flow rate	3.49	33.72

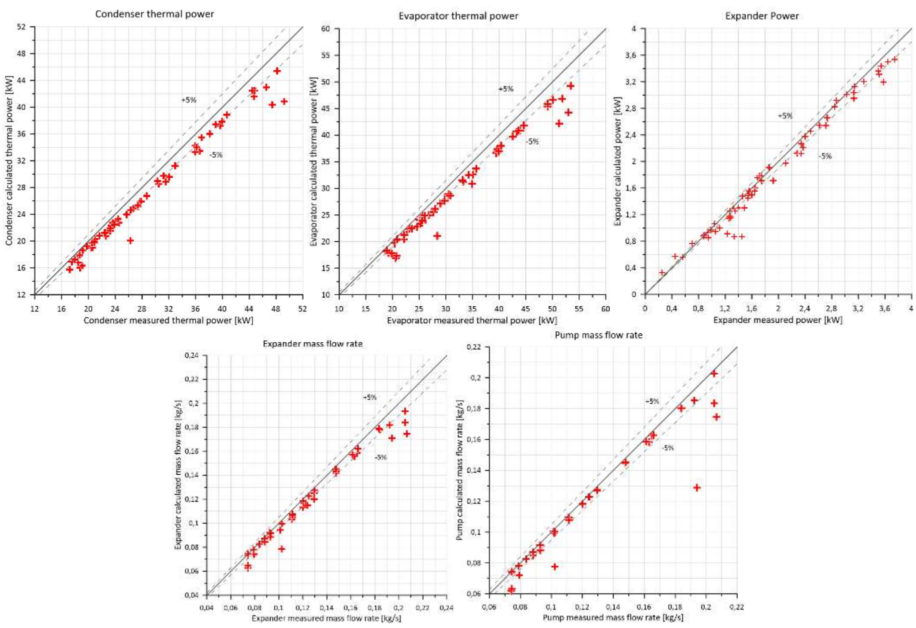

Figure 3 – ORC results before optimization

Therefore, the authors decided to carry out a multi-objective multi-object optimization process [18] in order to minimize the average percentage error by modifying the geometric parameters, which were estimated above.

3 Model Calibration Methodology: vector optimization process

Figure 4 represents the schematization of the multi-objective multi-object validation process. 6 input parameters were chosen as decision variables: the diameters of the 5 pipes of the model and the charge of the fluid R245fa. 5 quantities were chosen as objective parameters: refrigerant mass flow rate, expander power, thermal powers exchanged by evaporator and condenser and isentropic efficiency of the pump. While the minimization of the error between experimental quantity and objective quantity in 13 different experimental conditions, for a total of 65 objective functions, was chosen as the objective function.

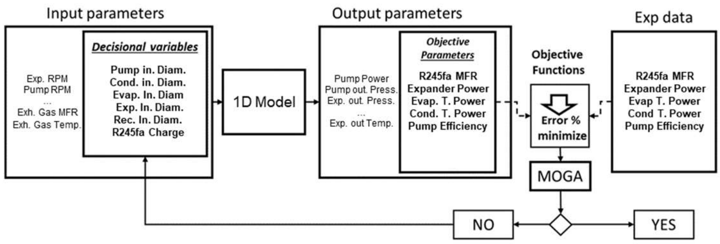

Figure 4 – schematic representation of the calibration process

After about 4000 iterations, the optimal solution was chosen to minimize the distance from the origin of the axes in the infinite solutions that are part of the Pareto front. The optimal solution gave the results shown in Table 3. As can be seen, the average error decreased for all the quantities analyzed, showing a maximum average error of 4.31% of the Expander power and a minimum average error of 0.87% for the Pump mass flow rate. The goodness of the model is also demonstrated by the decrease in the maximum error equal to 20.94%.

Table 3 – Mean and maximum error of the calibrated model

Data	Mean % Error	Maximum % Error
Condenser thermal power	2.74	9.07
Evaporator thermal power	2.71	8.31
Expander power	4.31	20.94
Expander mass flow rate	1.69	6.36
Pump mass flow rate	0.87	4.26

Moreover, the comparison was extended to all 61 operating conditions and the errors of some experimental quantities are shown in Figure 5.

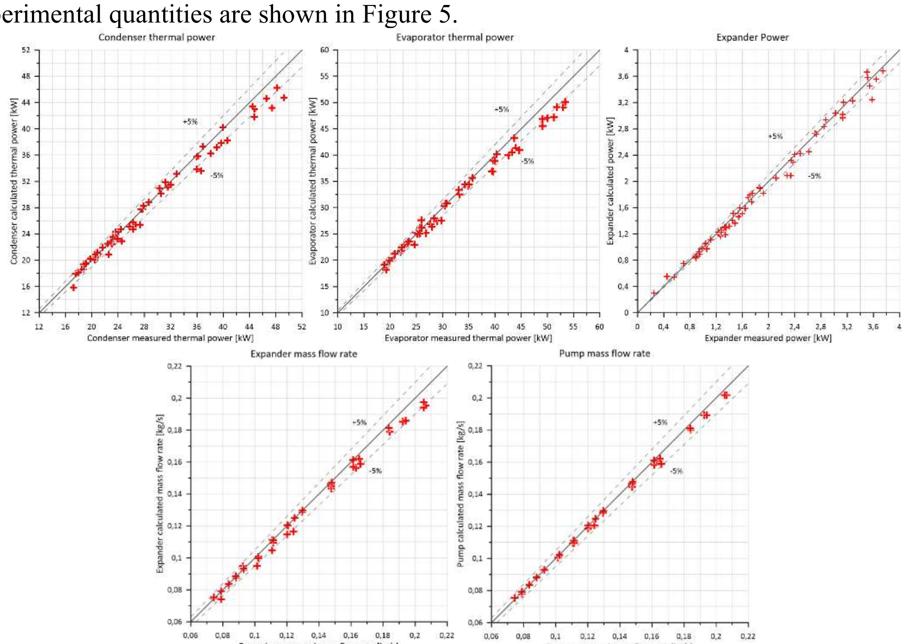

Figure 5 – ORC results of the calibrated model

As can be seen, differently from the results shown in Figure 3, in the optimized model almost all the values (about 85%) obtained from the model fall within the range of + or - 5%, with maximum errors of around 20%. This substantiates the improved robustness of the model.

4 Results and discussion

After the validation of the ORC plant model, it was coupled with the engine as recovery system, using the hot exhaust gas of the engine to power the ORC plant.

A total of 18 different operating conditions were performed. They can be described as follows:

- The engine speed is fixed and increased by steps. The chosen engine speeds are: 2000, 2125, 2250, 2375, 2500, 2625, 2750, 2875, 3000, 3125, 3250, 3375, 3500, 3625, 3750, 3875, 4000, 4125 rpm.
- The throttle valve is completely opened.

Each condition was renamed from case 1 to case 18 for a more practical view in Figures 6-8.

The engine operating conditions have been defined making sure that the thermal power contained in the exhaust gases was sufficient to power the ORC in optimal conditions. An extension of the operating range will be carried out after developing an ORC control system that allows to recover electrical power even in non-optimal conditions.

The performance results of the combined plant are shown in Figure 6, Figure 7 and Figure 8. In particular, Figure 6 shows the net power (in blue) obtained from the ORC system expressed as:

$$P_{net} = P_{exp} - P_{pump} \quad \textbf{(8)}$$

Where P_{exp} (in gray) is the mechanical power obtained from the turbine and P_{pump} (in orange) is the mechanical power absorbed by the pump.

In the 18 cases analyzed, the average net power produced by the ORC plant is 1.4 kW, with peaks of 2.0 kW.

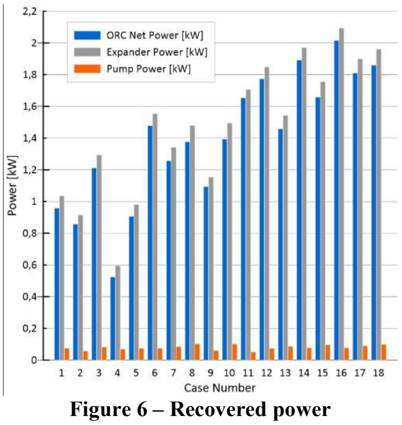

Figure 6 – Recovered power

Figure 7 shows the additional mechanical power of the ORC system (in orange) compared to the engine brake power (in blue) in the analyzed operating conditions. As can be seen, the percentage increase in brake power evaluated as:

$$\Delta P_{\%} = \frac{P_{net}}{P_b} \cdot 100 \qquad \textbf{(9)}$$

Where P_b is the engine brake power and P_{net} is the ORC system net power, ranges from 1.3% at 2375 rpm (minimum value) to 3.5% at 2625 rpm, with a mean value of 2.26%.

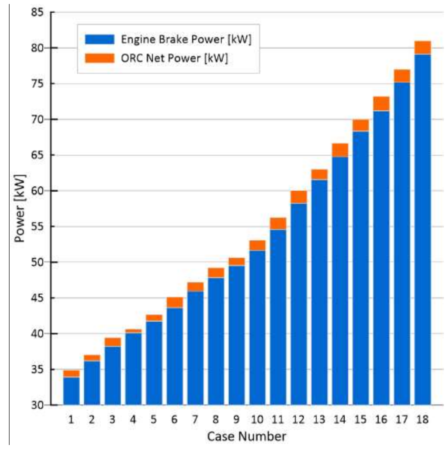

Figure 7– Power production with and without recovery system

Figure 8 shows the increase in global efficiency of the combined plant (in orange) calculated as:

$$\eta_{g,rec} = \frac{P_b + P_{net}}{\dot{m}_f H_i} \qquad \textbf{(10)}$$

The global efficiency the engine calculated with equation (1) is represented in blue. The percentage increase in the global efficiency assessed as:

$$\Delta\eta_{g,\%} = \frac{\eta_{g,rec} - \eta_g}{\eta_g} \cdot 100 \qquad \textbf{(11)}$$

ranges from 1.3% at 2375 rpm to 3.4% at 2625 rpm with a mean value of 2.62%, as shown in Figure 9.

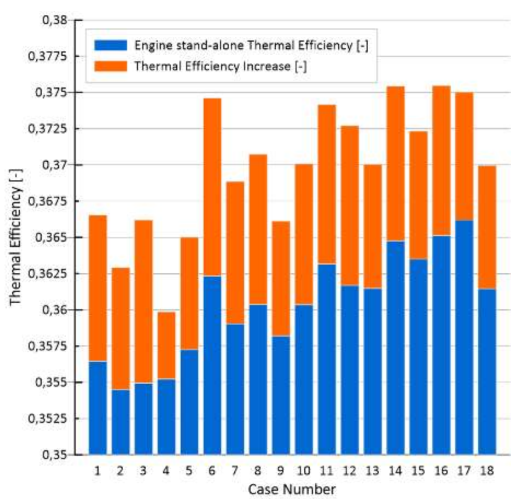

Figure 8- Thermal efficiency with and without recovery system

The results obtained from the model are very promising especially considering the fact that the maximum potential of the ORC system has not been reached, given that a maximum power of 2 kW has been obtained compared to a maximum achievable power greater than 3.5 kW as shown in Figure 5. An ad hoc control system can certainly lead to even better results.

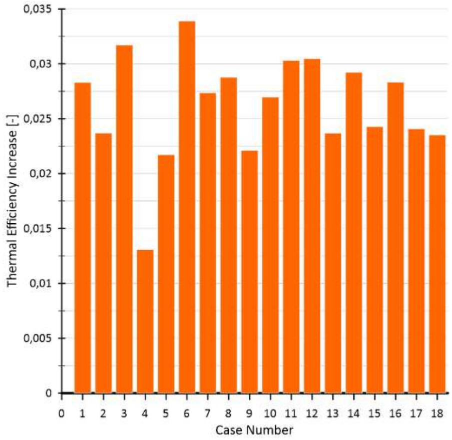

Figure 9 – Global efficiency percentage increase

5 Conclusion

The aim of this work is to investigate the feasibility of a micro-size ORC power plant as recovery system downstream of an engine that will increase the efficiency of the traction system on board the vehicle, applying a 0D-1D model.

Starting from experimental data, the ORC plant was validated with a multi-objective multi-object validation. This approach has proved effective in terms of model reliability and accuracy. The model is capable of recreating all experimental conditions with an average maximum percentage error of 4.31%.

Subsequently, the combined engine-ORC model was simulated in 18 different operating conditions of the engine resulting in an increase in the mechanical power supplied from 1.3% to 3.5%, with an average increase of 2.26%, and a percentage increase in overall efficiency from 1.3% to 3.4 %, with an average increase of 2.62%.

Despite the promising results, the model can be improved both in the control strategies and in the implementation of the individual components of the ORC system. In fact, as regards the first case, the ORC system has never reached the maximum deliverable power, and the development of a control system could allow to recover further energy and extend the operating range of the engine. For the second case, improvements in terms of the validity of the model can be achieved by implementing some components, such as heat exchangers or expander, moving from a map-based approach to a physical one.

Subsequently, the model could be implemented on-board the vehicle, further analyzing the feasibility of this plant by simulating conditions of real driving cycles such as the WLTC homologation cycle.

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