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Bottoming Organic Rankine Cycles for Passenger Cars

Antonio Mariani, Biagio Morrone, Andrea Unich*

Department of Engineering - Università della Campania "L. Vanvitelli" Via Roma 29, Aversa (CE) 81031, Italy

Corresponding Author Email: andrea.unich@unicampania.it

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ABSTRACT

Organic Rankine Cycles (ORC) are very efficient and flexible conversion systems with a high degree of maturity. They can be used with different heat sources, mainly using exhaust heat from different processing also with low temperature level sources. They have been mainly considered suitable for stationary power plants. Furthermore, the limitations of layout and size are less stringent if compared with road transportation propulsion systems, in particular when passenger cars are considered.
In this paper, the authors numerically investigate an ORC system as a bottoming solution for energy recovery from exhaust gases for internal combustion engine (ICE) passenger car. A passenger car was tested over a Real Driving Emission (RDE) cycle. Exhaust gas mass flow rate and temperature have been sampled allowing calculation of the thermal power available for the ORC plant at realistic driving conditions. The engine operational range was discretized using engine torque and speed values. As a result, a grid of 10 points was set up in the operational plane and the running conditions assigned to the closest discretized point in the grid, each one characterized by a residence time. The ORC recovered power resulted between 0.5 and 2.5 kW, the Rankine cycle efficiency ranged from 11 to 12% while engine efficiency increase varied from 2.5 to 12%.
By considering the permanence time in each discretized operating condition the engine efficiency increment resulted slightly higher than 6%.

1. INTRODUCTION

Fossil fuels for power generation, heating and transportation must be reduced to cope with their shortage and climate changes.

Renewable energy source share is without doubts going to increase all over the world due to the shortage in fossil fuel availability and to more stringent pollutant emission limits. Thus, fossil fuel usage for power generation, heating and transportation must be reduced. Anyway, it is not possible to predict, at least not accurately, how long the replacement process will take and how the energy mix of the future will be composed of.

In the meanwhile, fossil fuels must not be depleted too fast, and the efficiency of the conversion systems should be as high as possible in order to reduce both carbon dioxide and pollutant emissions.

A growing attention is focused on improving efficiency in energy conversion systems. Internal combustion engines are the main power sources for road transportation, with their contribution to total oil consumption in Europe around 50% and the carbon dioxide emissions accounting for around 25% of the whole emissions in 2016 (Eurostat statistics).

Passenger car diesel engines peak efficiency can attain values slightly higher than 40% when operating in optimal conditions, while lower values are attained at part loads. The fuel energy which is not converted in shaft work is rejected as heat from the exhaust gases and the cooling system. Such heat can be usefully recovered and converted to work by means of some useful systems. Organic Rankine Cycles (ORC) are particularly suited for heat recovery, even at low temperatures.

ORC applications to internal combustion engines is recent and still in a pre-commercial phase [1]. These systems appear convenient for marine propulsion, characterized by almost constant operating conditions without relevant size constraints, and for heavy duty compression ignition engines for trucks and buses. On the contrary passenger cars have strict mass and size requirements, consequently the installation of an ORC is more difficult.

The present paper analyzes an ORC for a passenger car equipped with a compression ignition engine with the aim at estimating the recoverable mechanical power. As a consequence, the increase of engine efficiency due to the additional power can be evaluated. The thermal power available for the ORC system has been measured when the vehicle was performing an RDE test, in order to consider realistic values. Mass flow rate and temperature of exhaust gas have been recorded together with engine speed, while engine torque has been calculated taking into account vehicle resistances and inertia forces. Engine operating conditions obtained in this way have been discretized into 10 points of a torque - engine speed grid, each one characterized by a residence time. Thus, a set of significant conditions has been identified in terms of exhaust mass flow rate and temperature, engine torque and speed. The actual Rankine cycle has been calculated in these conditions to evaluate the power recovered and the engine efficiency increase in real driving conditions. The working fluid selected for ORC calculations was n-pentane (R 601) and, only for comparison purposes, R134a has been chosen as reference.

2. MODELLING AND ANALYSIS

ORC systems are able to convert low temperature heat into shaft work. Usually they are used when waste heat is available at medium and low temperatures. The ORC system considered in the paper is reported in Figure 1. The main components are a pump, a heat exchanger fed by exhaust gases from the internal combustion engine, an expander and a condenser. This simple plant solution was preferred to the more complex one considering multiple heat sources, such as Exhaust Gas Recirculation (EGR), compressed air cooler and engine cooling system, mainly to limit the additional mass and volume due to application constraints.

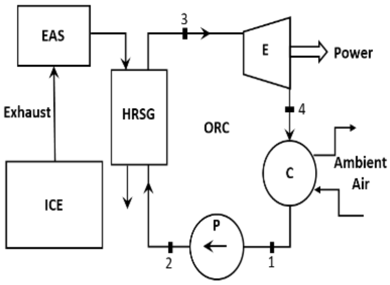

Figure 1. Sketch of the ORC system

The waste heat available for the system has been estimated considering a passenger car equipped by a compression ignition engine. This choice is motivated by a large share of diesel cars in Europe [2] and for considering less favourable exhaust gas conditions for the ORC. In fact, Diesel engines show higher efficiency than spark ignition engines, consequently the heat rejected to the ambient and the exhaust temperature is lower. Realistic engine operating conditions are accounted for by considering data registered during an RDE test. The speed profile is reported in Figure 2.

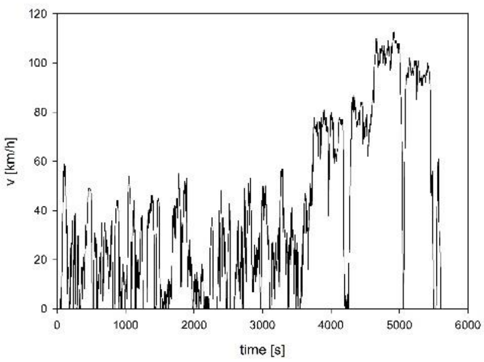

Figure 2. Vehicle speed over the RDE test

The driving cycle characteristics determine the power the engine delivers and, consequently, the exhaust mass flow rate and temperature, which represent the inputs to the ORC performance calculations.

The main vehicle and engine data are reported in Table 1.

Inlet air and fuel mass flow rates were sampled from the engine ECU together with engine speed, while exhaust gas temperature was measured by a thermocouple downstream the Exhaust After-treatment System (EAS), composed by a Diesel Particle Filter (DPF), a Diesel Oxidation Catalyst (DOC) and a Selective Catalytic Reduction (SCR). Engine torque has been calculated by estimating vehicle resistances and inertia forces. The torque-speed values obtained have been discretized by means of a 10 points grid, as displayed in Figure 3. Each calculated point has been attributed to the closest grid points, with a permanence time inversely proportional to the distance between calculated point and grid points and considering the calculated point work and the grid point work constant. As a consequence the transient effects have been considered negligible Discretization permitted to simulate a driving cycle by defining a limited number of exhaust gas conditions, being the input for the ORC analysis. In Table 2 the percent residence time is also displayed referring to the total elapsed time during the RDE.

Table 1. Engine and vehicle main characteristics

Engine type	**4-stroke turbocharged Diesel**
Rated power	202 kW @4000 rpm
Rated torque	600 Nm @2200 rpm
Vehicle mass	2800 kg
Aerodynamic drag coefficient	0.36
Frontal area	4.17 m^2
Rolling resistance coefficient	0.01

The results have been obtained with n-pentane, R 601, as the working fluid for ORC calculations, and they have been compared with those obtained using R134a as a reference fluid, which is a refrigerant largely used in car air conditioning but characterized by a high GWP; for this reason this fluid is to be phased out in the next future.

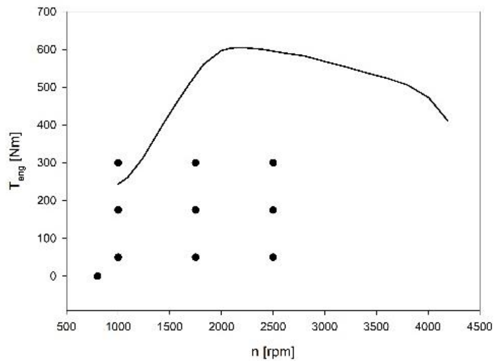

Figure 3. Discretization grid and engine torque curve. The grid covers the engine operating range over the RDE test

In Figure 4 the comparison between R 601 and R134a is displayed showing the different thermodynamic characteristics: the first fluid, a hydrocarbon, belongs to the dry class, whereas the latter to the wet fluid class. In terms of thermodynamic performance for direct energy conversion cycle, the dry fluid class is to be preferred since an isentropic expansion, starting at saturated vapor condition, brings the fluid to superheated vapor [3]. In Table 3 some meaningful thermodynamic data are reported. It can be observed that R601 shows higher critical temperature and lower critical pressure than R 134a [4]. More important the enthalpy of evaporation at ambient pressure: R601 shows larger value than R134a, implying better thermal performance. Also, the saturation temperature of R601 at ambient pressure shows a more favorable value for heat transfer process.

Table 2. Engine speed (n), torque (T_{eng}), exhaust gas mass flow rate (mfr_{ex}), temperature (T_{ex}), permanence time (t) for each point of the grid

n (rpm)	T_{eng} (N m)	t (%)	T_{ex} (°C)	mfr_{ex} (kg/h)
800	0	17.60	182	40.2
1000	50	19.44	196	91.7
1000	175	12.07	217	111.5
1000	300	3.18	250	144.2
1750	50	16.84	214	128.0
1750	175	20.55	247	159.8
1750	300	7.01	269	191.3
2500	50	0.74	246	217.3
2500	175	1.65	264	254.8
2500	300	0.93	272	263.9

Its flammability is high (4/4) according to NFPA 704 standards, while the health hazard is low (1/4) and the instability is absent (0/4). The fluid does not show an Ozone Depletion Potential and its 100 years Global Warming Potential is 5.

The expander selection is important for ORC systems because its performance strongly affects the actual cycle efficiency. In this case a positive displacement expander was considered and an expansion efficiency of 0.70 was considered, while pump efficiency was fixed at 0.65 [3, 5].

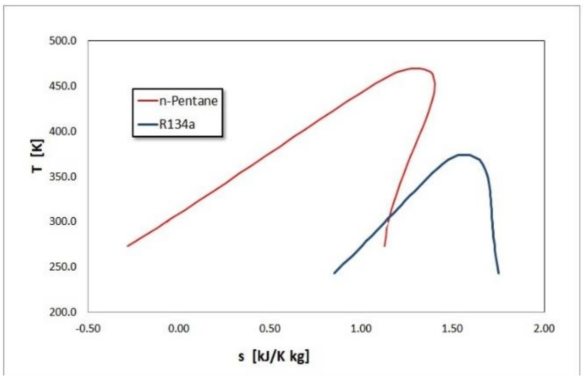

Figure 4. Temperature-Entropy graph for the two investigated fluids

In modeling the thermodynamic cycle pressure drops in pipes and heat exchangers have been neglected,

In solving the energy balance between hot and cold streams inside the recovery heat exchanger, the ORC fluid mass flow rate was varied in order guarantee a fully evaporated stream for the ORC fluid leaving the HRSG. The heat exchanger has been assumed countercurrent with an overall conductance UA equal to 200 W/K.

Simulations of the ORC have been accomplished by using the DWSIM simulation tool [6], which is a process simulator based on thermodynamic calculations, covering a variety of systems. It is able to model phase equilibria between solids, vapor and up to two liquid phase mixtures. Fluid properties are evaluated using the CoolProp package [4] and the Nested Loop flash algorithm is employed for evaluating the phase change of the employed fluids [6]. The tool allows choosing from a variety of components, such as pumps, expanders, heat exchangers and others which can be implemented with different computation properties.

Table 3. Main thermodynamic data for the two investigated fluids

Fluid	T_{cr} [K]	p_{cr} [bar]	h_{evap}* [kJ/(kg K)]	T_{sat}* [K]
R 601	469.7	33.70	357.6	309.1
R 134a	374.2	40.59	216.9	247.2

*At 1.01 bar

3. RESULTS

The effect of maximum cycle pressure on the thermodynamic performance was evaluated, comparing R601, the selected ORC fluid, with the reference fluid R134a. The comparison has been performed at the grid point 1750 rpm - 175 N m, the most representative one, because of its higher permanence time.

The analysis of the actual Rankine cycle has been performed by considering the minimum pressure corresponding to saturate liquid at $T_1 = 314$ K, and the pressure increase in the pump variable from 5 bar to 20 bar, the maximum values being 6.2 to 21.2 bar for R601 and 15.4 to 30.4 bar for R134a.

In Table 4 the temperature and pressure at pump inlet are reported for the two fluids, together with the maximum pressure interval.

Table 4. Main thermodynamic cycle conditions

Fluid	T_1 [K]	P_1 [bar]	P_2 [bar]
R 601	314	1.2	6.2 - 21.2
R 134a	314	10.4	15.4 - 30.4

Figure 5 shows the ORC fluid mass flow rate as a function of the pressure increase in the pump for the two investigated fluids. The trend is opposite because mass flow rate for R601 slightly decrease with increasing pressure, whereas the R134a requires increasing mass flow rate to optimize the extracted power. The opposite behavior is due to the different thermodynamic characteristics of the fluids, which are more favorable for R601, as already discussed and shown in Figure 4.

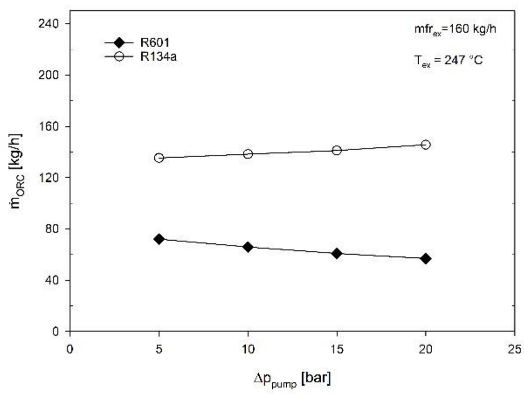

Figure 5. ORC fluid flow rate as a function of pump pressure increase (R601 and R134a)

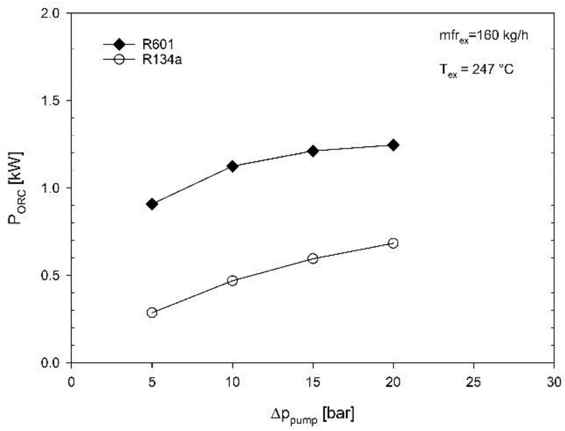

Figure 6. ORC power as a function of pump pressure increase (R601 and R134a)

The ORC recovered power as a function of the pressure increase is displayed in Figure 6 for the same engine and exhaust conditions as before. The power increases with pressure increase for both fluids, but R601 show larger values which are nearly double than those for R134a, also with lower maximum pressure values. In fact, R601 maximum pressure conditions range from 6.2 to 26.2 bar, representing lower values than those obtained by R134a, ranging from 15.4 to 30.4 bar. Of course, the large the values of the maximum cycle pressure the greater the weight of plant for structural and safety reasons.

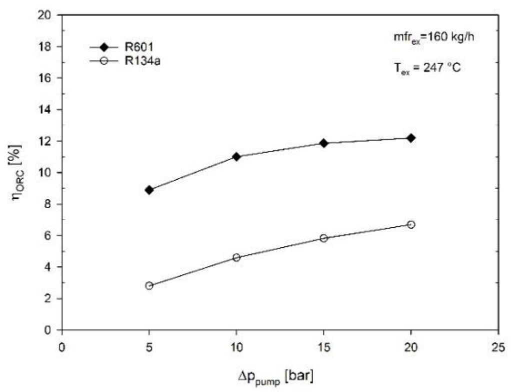

Figure 7. ORC efficiency as a function of pump pressure increase (R601 and R134a)

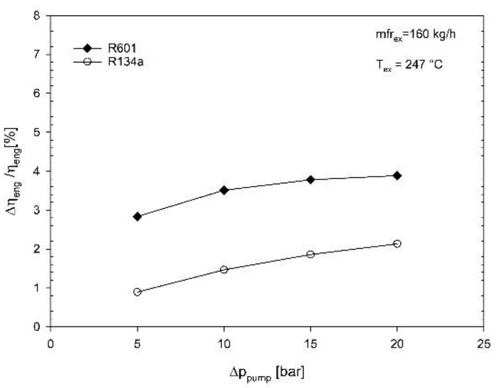

Figure 8. Engine efficiency increment as a function of pump pressure increase (R601 and R134a)

The ORC efficiency comparison between the two fluids is reported in Figure 7 as a function of the maximum pressure increment. It can be observed that the reference fluid, R134a, shows values in the range 3 to 6 %, whereas the values are much higher for R601 displaying values between 9 and 12 %.

Engine efficiency increment is displayed in Figure 8 as a function of the pressure increase in the pump. It shows values from 3 to 4 % for R601 whereas they are lower for the R134a, ranging between 1 and 2 %.

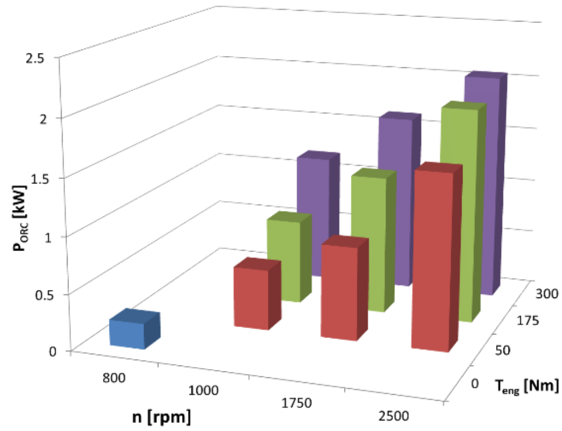

Figure 9. ORC power for the engine operating points defined by the discretization grid, R601

For the other points of the grid the maximum pressure increase, 20 bar, was considered and only R601 was used as working fluid.

Figure 9 shows the ORC recovered power in all the grid points. The values range between 0.5 kW to 2.5 kW at the higher load and speed, i.e. n=2500 rpm and T_{eng}=300 N m.

In Figure 10 the ORC efficiency is reported. Values are almost constant over the grid points, ranging between 11 and 12 %.

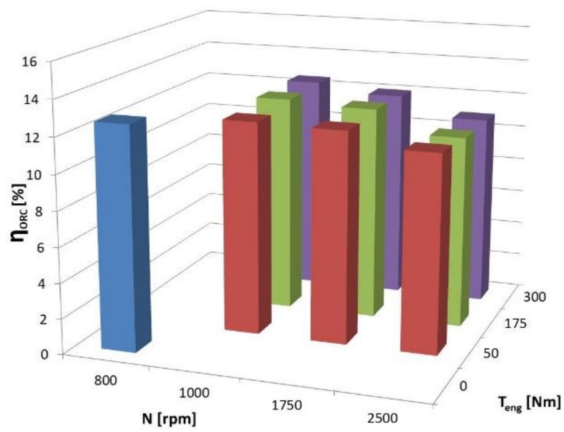

Figure 10. ORC efficiency for the engine operating points defined by the discretization grid, R601

Figure 11 shows the engine efficiency increment in each grid operating condition. Higher values, between 9 and 12 %, have been observed at low engine loads; this result, apparently unexpected, depends on the ORC recovered power values, which are not proportional to engine power, showing a higher incidence at part load. By considering the permanence time in each discretized operating condition the engine efficiency increment over the RDE test resulted slightly higher than 6 %.

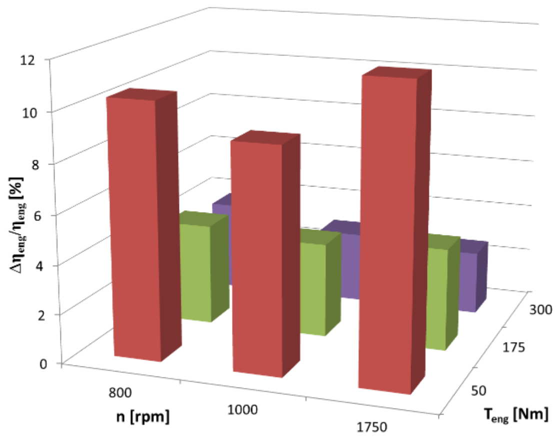

Figure 11. Engine efficiency increment for the engine operating points defined by the discretization grid, R601

4. CONCLUSIONS

An Organic Rankine Cycle bottoming a compression ignition engine installed on a passenger car has been investigated. The thermal power available for the ORC system has been measured when the vehicle was performing an RDE test, in order to consider realistic values. The working fluid selected for ORC calculations was n-pentane (R 601) and, only for comparison purposes, R134a has been chosen as reference.

ORC performance in terms of recovered power, Rankine cycle efficiency and engine efficiency increment have been evaluated in 10 points of a torque – engine speed grid, each one characterized by a residence time. The recovered power resulted between 0.5 and 2.5 kW, the Rankine cycle efficiency ranged between 11 and 12 % while engine efficiency increase varied from 2.5 to 12 %.

By considering the permanence time in each discretized operating condition the engine efficiency increment on the RDE test was around 6 %.

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NOMENCLATURE

A	Area [m^2]
T	Torque [N m]
c_p	Specific Heat [kJ/(kg K)]
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
EAS	Exhaust After-treatment system
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
eng	Engine
HRSG	Heat Recovery Steam Generator
ICE	Internal Combustion Engine
LHV	Lower Heating Value [MJ/kg]
M	Mass [kg]
mfr	Mass flow rate [kg/s]
N	Engine speed [RPM]
P	Pressure [bar]
ORC	Organic Rankine Cycle
P_{ORC}	ORC Recovered Power [kW]
P_{sh}	Shaft Engine Power [kW]
RDE	Real Driving Emission test
SCR	Selective Catalytic Reduction
T	Temperature [K]
U	Overall heat transfer coefficient [W m^{-2} K^{-1}]
V	Speed [m/s]

Greek symbols

η_{eng}	Engine Efficiency [-]
η_{ORC}	ORC Efficiency [-]