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Thermodynamic investigation and design of a multi-generation ORC-ejector cooling cycle heat pump for vessel waste heat recovery

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Abstract:
The present study presents the thermodynamic design of a vessel waste heat recovery prototype based on a cascade Organic Rankine Cycle (ORC) and an ejector cooling-vapor compression cycle (EVCC) in the context of the ZHENIT project. The system can be driven by high-temperature (>120 °C) and a low-temperature (<80 °C) waste heat and features 3 main operating modes: 1) electricity-only, 2) combined heat and power (CHP) via heat recovery from the ORC condenser and 3) combined cooling and power (CCP) from the simultaneous operation of the ORC and EVCC. Waste heat is supplied to the ORC (topping cycle), while heat recovered during the desuperheating is used for driving the EVCC (bottoming cycle). A series of parametric analyses were carried out to investigate the influence of key design parameters (working fluids and maximum ORC and EVCC temperatures, recuperator temperature rise, condensation temperature, compressor pressure ratio) on the performance of the prototype and select its design point. R1233zd(E) and R1234ze were found to be the best-performing fluids in the ORC and EVCC, respectively. Furthermore, the selected design ORC evaporation temperature is 120 °C, the condensation temperature (CHP mode) is 50 °C, while the EVCC generator evaporation temperature and compressor pressure ratio (CCP mode) are equal to 52 °C and 0.84, respectively. Under the above conditions, the prototype ORC electric efficiency is up to 12.62% (electricity-only mode), while the overall electrical and CCP efficiency (CCP) are 10.29% and 14.96%, respectively. Finally, the electrical COP of the EVCC is 7.62.

Keywords: Thermodynamics; ORC; ejector cooling; waste heat; CHP; CCHP; trigeneration.

1. Introduction

The Organic Rankine Cycle (ORC) is a widely considered technology for electricity production from low-temperature heat sources such as solar, biomass, geothermal and waste heat from industrial processes as well as exhaust gases and hot jacket cooling water of internal combustion engines of stationary and mobile applications, such as maritime vessels [1, 2]. Especially in the maritime industry, large amounts of fossil fuels are annually consumed for ship propulsion, cargo operations and electricity production. Therefore, there is an abundance in waste heat that could be exploited by ORC systems [3] in the form of engine and genset exhaust gases, compressed air intercoolers and jacket water cooler with typical temperatures from 60°C to 250°C [4].

Vessel energy demand is not limited to electrical energy but also includes heating and cooling. The combination of ORCs with thermally activated cooling systems can also contribute to covering these demands. In the literature, cascade configurations of ORCs and sorption cooling technologies have been considered to this end, especially, absorption [5] and adsorption [6] chillers. In such configurations, low-temperature heat is recovered from the ORC desuperheater or condenser and is used for driving the chiller. An alternative thermally activated cooling technology is based on the ejector cooling cycle (ECC), which is typically considered for use in solar thermal applications (solar cooling) [7]. The operating principle of ECCs is similar to that of conventional vapor compression cycles (VCCs), however, it involves the substitution of the mechanical compressor with an ejector device that thermally compresses the low-pressure vapor exiting the cooling evaporator utilizing a high-pressure stream generated through the exploitation of waste heat.

Notably, ejectors of ECCs can be easily integrated with conventional electricity-driven vapor compression cycles (VCCs) to improve their performance and improve their reliability and flexibility. The aforementioned concept is the basis of the prototype to be developed in the ZHENIT project, which is based on an innovative ejector-vapor compression cooling cycle (EVCC). In particular, the prototype will be based on a cascade layout, with an ORC as a topping cycle and an EVCC as a bottoming cycle. The ORC will be driven by waste heat, while the EVCC will be driven by heat recovered from the ORC desuperheater.

The present work presents the preliminary thermodynamic design process and results of the prototype based on three operating modes: 1) electricity-only, 2) CHP and 3) CCP. For the design, a series of parametric investigations were carried out to select the most appropriate working fluids and investigate the impact of different key design parameters on its performance.

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2. Methodology

2.1. System description

The ZHENIT ORC-EVCC prototype is designed for vessel engine waste heat recovery at temperature levels ranging from 60°C (jacket engine cooling water) to 150°C (engine exhaust gases) for electricity, heating and cooling production. Within the project, a micro-scale (10 kW_e) prototype will be developed and demonstrated at lab-scale, along with the alternative vessel waste heat recovery solutions that are part of the ZHENIT solutions.

In the present study, a high-T waste heat source consisting of Therminol VP1 oil [8] at a mass flow rate of 2.5 kg/s with inlet/outlet temperatures of 150 °C /130°C, is considered. Meanwhile, a low-T waste heat source is additionally considered, consisting of hot water at inlet/outlet temperatures of 60°C /52°C with a mass flow rate of 0.4 kg/s.

The layout of the prototype is pictured in ***Figure. 1***. The ORC is the topping cycle and is driven by the high-T and (provisionally) low-T waste heat sources. Heat may be recovered during the desuperheating of the ORC working fluid in a heat exchanger (Generator) and used for driving the EVCC, which is the bottoming cycle. The EVCC is based on integrating an ejector into a conventional VCC. Compared to a VCC, the compression process of the low-pressure working fluid exiting the cooling evaporator is accomplished not only by a mechanical compressor but also by an ejector. More specifically, in the ejector, the working fluid leaving the compressor is further compressed as it is entrained by and mixed with high-pressure vapor produced in the Generator. As a result, the electricity consumption compared to a conventional VCC can be potentially reduced.

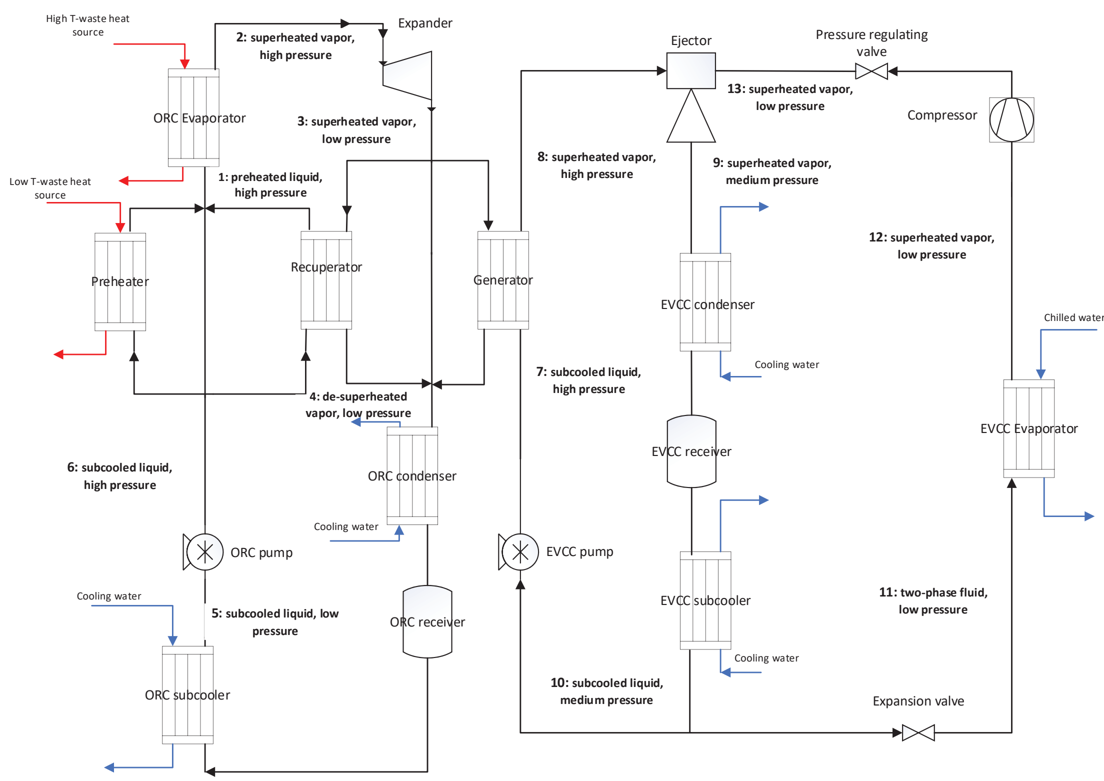

Figure. 1. Waste heat recovery ORC-EVCC layout

2.2. Operating modes

The prototype can operate in three different modes:

Mode 1 – Electricity-only: In this Mode, the EVCC is not operational and no cooling is produced. The ORC is driven exclusively by high-T waste heat. Furthermore, a recuperator is used to increase its efficiency.

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Mode 2 – Combined heat and power (CHP): As in the previous case, the EVCC is not operational and the ORC is driven only by high-T waste heat. However, in this mode, the recuperator is bypassed and the condensation pressure of the ORC is increased to produce hot water at 45 °C.

Mode 3 – Electricity and cooling (CCP): In this Mode, both the ORC and EVCC are operational. The ORC may be driven by both a high-T and a low-T waste heat source. After exiting the expander, the ORC working fluid is desuperheated in the Generator, in which the EVCC working fluid evaporates on the cold side of the heat exchanger. Cooling (chilled water at 10 °C) is produced in the EVCC evaporator. The low-pressure working fluid exiting the cooling EVCC evaporator is compressed in the compressor and enters the low-pressure port of the ejector. Meanwhile, the high-pressure vapor exiting Generator enters the high-pressure port of the ejector. In the ejector, the two flows are mixed. The resulting mixed flow exits the ejector at an intermediate pressure and is driven to the EVCC condenser, where it is condensed by cooling water.

2.3. Working fluids

The investigated ORC and EVCC working fluids and their properties are summarized in **Table *1***. A set of hydrofluoroolefins (HFOs), namely R1233zd(E), R1234ze and R1234yf, were considered because of their low environmental impact (low global warming and ozone depletion potential (ODP), being compliant with the regulations imposed by the Montreal [9] and Kyoto Protocols [10], as well as the F-gases regulations [11]. Furthermore, these fluids have either no (R1233zd(E)) or low flammability according to the ASHRAE safety classification. For the ORC, R1233zd(E) and R1234ze were considered, owing to their relatively higher critical temperatures, resulting in improved cycle efficiencies. For the EVCC, R1233zd(E) was not considered because of its significantly lower evaporation pressure, which is below the atmospheric and may cause issues because of air leakage into the refrigerant circuit.

Table 1. Investigated working fluids and their properties

Working fluid	T_{crit} (°C)	p_{crit} (bar)	ODP	GWP	ASHRAE safety group
R1233zd(E) (ORC)	165.5	35.7	0	1	A1
R1234ze (ORC& EVCC)	109.4	33.4	0	6	A2L
R1234yf (EVCC)	94.7	33.8	0	4	A2L

2.4. Design

The modeling of ORC and ECC/VCC systems has been extensively investigated in the relevant literature [7]. Therefore, in the present work, only the most significant modeling aspects which are relevant to the particular investigated prototype are presented.

All modeling assumptions are summarized in ***Table 2***. The thermophysical properties of the heat source stream, working fluids and cooling water are calculated with REFPROP [12]. Two ORC evaporation temperatures are investigated, equal to 90 and 120 °C. Although higher ORC evaporation temperatures would potentially result in improved efficiencies, a maximum upper bound of 120 °C was considered because of technological limitations (material limitations and limitations of built-in volume ratios) of volumetric expansion machines (scroll and screw expanders) at the targeted scale of the prototype (around 10 kW_e) and technical challenges of testing higher temperature heat sources (above 140 °C) at lab scale. A condensation temperature (T_{cond}) of 30÷65 °C is assumed for the ORC (depending on whether only electricity or heating is also produced) and 30 °C for the EVCC. The subcooling degree in both condensers is 5 K to prevent pump cavitation problems, while the superheating degree at the ORC evaporator and EVCC generator is 10 K. Finally, the evaporation temperature in the EVCC evaporator is 5 °C while the working fluid leaves it superheated by 5 K. Additional modeling assumptions are included in the table. For some parameters, ranges are provided instead of particular values. This is because, for these parameters, parametric investigations are carried out (presented in Section 3).

Table 2 General modeling assumptions

Parameter	Value
High-T waste heat source	
HT source mass flow rate	2.5 kg/s
HT source inlet temperature	150°C
HT source outlet temperature	130°C
Low-T waste heat source	
LT source mass flow rate	0.4 kg/s
LT source inlet temperature	60°C
LT source outlet temperature	52°C
ORC	
Expander isentropic efficiency	0.70
ORC pump isentropic efficiency	0.65

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Expander-generator electromechanical efficiency	0.92
ORC pump motor efficiency	0.95
Pinch point in Generator (ORC desuperheater)	5 K
Superheating degree at expander inlet	10 K
Condensation temperature	30 °C (Mode 1), 50÷65 °C (Mode 2)
Subcooling degree at condenser outlet	5 K
Recuperator cold stream temperature rise	10÷20 °C (Mode 1), 0 K (Mode 2 and Mode 3)
Evaporation temperature	70÷120 °C
EVCC	
EVCC pump isentropic efficiency	0.65
EVCC pump motor efficiency	0.95
EVCC compressor isentropic efficiency	0.65
Compressor-motor electromechanical efficiency	0.95
Superheating degree at ejector inlet	10 K
Evaporation temperature (EVCC cooling evaporator)	5 °C
Superheating degree at EVCC cooling evaporator outlet	5 K
Condensation temperature	30 °C
Subcooling degree at condenser outlet	5 K
Compressor pressure ratio (Eq. 9)	0.70÷0.98
Generator evaporation temperature	46÷60 °C
EVCC working fluid mass flow rate in generator	calculated to satisfy pinch point in Generator
Primary flow isentropic efficiency	0.95
Secondary flow isentropic efficiency	0.65
Ejector throat and constant area section cross-flow areas	calculated from ejector model
entrainment ratio	calculated from ejector model

The mass flow of the ORC working fluid ($\dot{m}_{ORC}$) is calculated from the energy balance equation in the ORC evaporator, based on the mass flow rate ($\dot{m}_{wh,HT}$) and enthalpy of the high-T waste heat stream at its inlet ($h_{wh,in}$) and outlet ($h_{wh,out}$):

$$\dot{Q}_{evap,ORC} = \dot{m}_{ORC}(h_2 - h_1) = \dot{m}_{wh,HT}(h_{wh,in} - h_{wh,out}) \tag{1}$$

In Mode 3, the enthalpy of the ORC working fluid at the preheater outlet is determined from the energy balance equation of this heat exchanger based on the mass flow rate ($\dot{m}_{wh,LT}$) and inlet ($h_{wh,in}$) and outlet ($h_{wh,out}$) enthalpy of the low-T waste heat stream in the heat exchanger:

$$\dot{Q}_{pre,ORC} = \dot{m}_{wh,LT}\left(h_{wh,in} - h_{wh,out}\right) = \dot{m}_{ORC}(h_1 - h_6) \tag{2}$$

The net electric power output of the ORC ($P_{e,ORC}$) is equal to the gross electrical power of the expander ($P_{e,exp}$) minus the electrical power consumed by the ORC pump ($P_{e,pump,ORC}$):

$$P_{e,ORC} = P_{e,exp} - P_{e,pump,ORC} = \dot{m}_{ORC}\left(\eta_{em,exp-G}(h_2 - h_3) - \frac{(h_6 - h_5)}{\eta_{M,pump}}\right) \tag{3}$$

These are calculated also considering the ORC expander-generator electromechanical efficiency ($\eta_{em,exp-G}$) and ORC pump motor efficiency ($\eta_{M,pump,ORC}$).

In Mode 2, useful heat is recovered from the ORC condenser ($\dot{Q}_{heat}$), calculated according to the following equation:

$$\dot{Q}_{heat} = \dot{m}_{ORC}(h_4 - h_5) \tag{4}$$

The EVCC is designed considering the thermal input provided to the cycle from the desuperheating of the ORC working fluid in Generator 1 in Mode 3. More specifically, for each EVCC generator evaporation temperature, the mass flow rate of the EVCC primary flow at the ejector inlet ($\dot{m}_p$), along with the state of the ORC working fluid at the Generator outlet are iteratively calculated to achieve the desired 5 K pinch point in this heat exchanger (Table 2), according to the energy balance equation:

$$\dot{Q}_{gen} = \dot{m}_{ORC}(h_3 - h_4) = \dot{m}_p(h_2 - h_1) \tag{5}$$

A critical operational parameter of the EVCC is the mass flow rate of the secondary flow entrained in the ejector ($\dot{m}_s$), which affects the cooling output of the system. For its calculation, a modified ejector design model based on the 1-D model by Huang [13] is applied. The details of the aforementioned model can be found in the

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original publication by Huang [13]. The inputs to this model are: 1) the primary flow thermodynamic state (pressure and temperature), 2) the primary flow mass flow rate, 3) the secondary flow state (pressure and temperature) and 4) the backpressure of the ejector (imposed and equal to the EVCC condensation pressure).

The outputs of the model are the main ejector dimensions, namely the converging-diverging nozzle throat and outlet cross-sectional areas, the mixing section cross-sectional area as well as the diffuser outlet cross-sectional area, as well as the secondary flow mass flow rate of the ejector. The ratio of the secondary and primary mass flow rate of the ejector is defined as the ejector entrainment ratio (ω):

$$\omega = \frac{\dot{m}_s}{\dot{m}_p} \tag{6}$$

Higher entrainment ratios indicate that the ejector operates more efficiently, since more low-pressure vapor is entrained for the same amount of high-pressure vapor supplied to the ejector.

An important design parameter of the EVCC is the compressor pressure ratio (not to be confused with the common compressor pressure ratio definition), which is defined according to the following equation:

$$p_{r,comp} = \frac{p_{13} - p_{12}}{p_9 - p_{12}} \tag{7}$$

The compressor pressure ratio as it is defined in the present study is the ratio of the pressure increase of the secondary flow achieved by the compressor divided by the pressure difference of the EVCC working fluid in the condenser and cooling evaporator. It can theoretically vary from a minimum of 0 (the whole compression is achieved by the ejector and no compressor is used) to a maximum of 1 (the whole compression is achieved by the compressor and no ejector is used).

In addition to the mass flow rate of the secondary flow, an additional output of the ejector model is the state of the mixed flow at the ejector outlet.

The cooling produced in the EVCC evaporator ($\dot{Q}_{cool}$) is calculated from the equation:

$$\dot{Q}_{cool} = \dot{m}_s(h_6 - h_5) \tag{8}$$

The power consumption of the EVCC ($P_{e,EVCC}$) is equal to the power consumption of the compressor ($P_{e,comp}$) plus the power consumed by the EVCC pump ($P_{e,pump,EVCC}$):

$$P_{e,EVCC} = P_{e,comp} + P_{e,pump,EVCC} = \frac{\dot{m}_s(h_{13} - h_{12})}{\eta_{em,comp-M}} + \frac{\dot{m}_p(h_7 - h_{10})}{\eta_{M,pump,EVCC}} \tag{9}$$

For their calculation, the compressor electromechanical efficiency ($\eta_{em,comp-M}$) and the EVCC pump motor efficiency ($\eta_{M,pump,EVCC}$) are considered.

The total net power produced by the whole ORC-EVCC system ($P_{e,tot}$) in Mode 3 is equal to the net power produced by the ORC electrical generator minus the power consumption of the EVCC:

$$P_{e,tot} = P_{e,ORC} - P_{e,EVCC} \tag{10}$$

2.5. Performance assessment indexes

The electrical efficiency of the ORC is defined as follows:

$$\eta_{e,ORC} = \frac{P_{e,net,ORC}}{\dot{Q}_{evap,ORC} + \dot{Q}_{pre,ORC}} \tag{11}$$

The electrical COP of the EVCC is defined according to the following equation:

$$COP_e = \frac{\dot{Q}_{cool}}{P_{e,EVCC}} \tag{12}$$

The overall electrical efficiency of the ORC-EVCC in Mode 3 is defined as follows:

$$\eta_{e,tot} = \frac{P_{e,tot}}{\dot{Q}_{evap,ORC} + \dot{Q}_{pre,ORC}} \tag{13}$$

The combined cooling and power overall efficiency in Mode 3 is defined as follows:

$$\eta_{CCP} = \frac{P_{e,tot} + \dot{Q}_{cool}}{\dot{Q}_{evap,ORC} + \dot{Q}_{pre,ORC}} \tag{14}$$

3. Results and discussion

3.1. Parametric investigations

The design of the prototype is based on the selection of several parameters, some of the most significant being the following: 1) working fluid in ORC, 2) working fluid in EVCC, 3) ORC evaporation temperature, 4) EVCC generator evaporation temperature, 5) compressor pressure ratio (as defined in Eq. 7). The selection of these parameters is subject to thermodynamic (waste heat temperature, efficiencies) and technical constraints (equipment components). As a first step in the design process, a series of parametric investigations are carried out to examine their influence on the performance of the prototype. In the following investigations, while these parameters are varied, all other design parameters are kept constant, equal to the values summarized in ***Table 2***.

The ORC electrical efficiency variation as a function of the recuperator cold stream temperature rise for different ORC evaporation temperatures for Mode 1 (electricity-only) is presented in ***Figure. 2***. As expected, the highest evaporation temperature of 120 °C results in the highest efficiency. Because low evaporation temperatures are considered in the ORC, the possible cold stream temperature rise in the recuperator is relatively low, thus the efficiency improvement potential is very small. A maximum efficiency of 12% is achieved with R1233zd(E) for an evaporation temperature of 120 °C when the temperature rise is 20 K. For R1234ze(E), the maximum electrical efficiency is 8.86% when the evaporation temperature is 120 °C and the temperature rise in the recuperator is 14 K. Note that because of the significantly lower critical temperature of R1234ze(E) compared to R1233zd(E), the cycle's maximum operation temperature is lower, thus less heat can be recovered in the recuperator, and the cold stream temperature rise is limited .

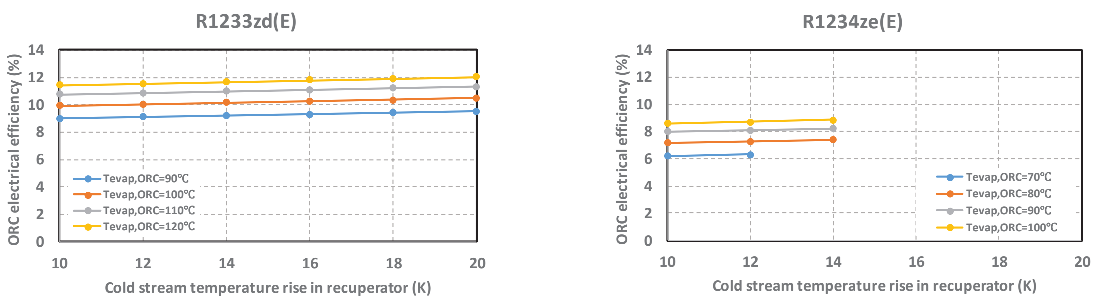

Figure. 2. ORC electrical efficiency as a function of the cold stream temperature rise in the recuperator for different ORC evaporation temperatures in Mode 1 (Electricity-only, T_{cond}=30 °C)

The ORC electrical efficiency as a function of the ORC condensation temperature for different evaporation temperatures in Mode 2 (CHP) is depicted in ***Figure. 3***. As expected, increasing the condensation temperature results in a significant, linear decline in electrical efficiency, since useful heat is produced to the detriment of electrical power. Therefore, the ORC condensation temperature must be selected to have an intermediate value in order to produce hot water at a sufficiently hight temperature as well as securing the highest possible ORC electric efficiency .

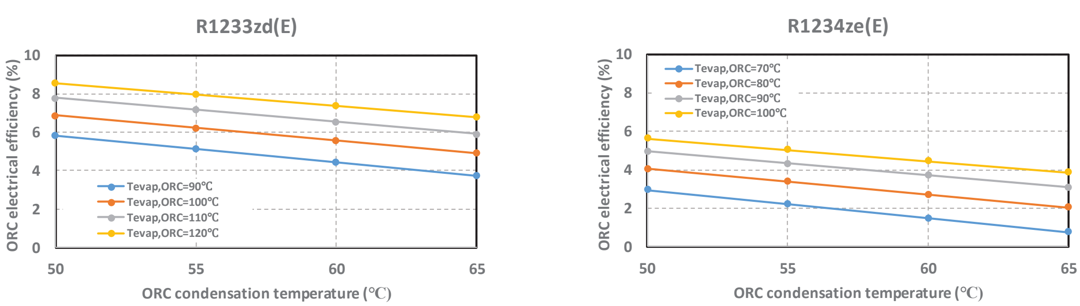

Figure. 3. ORC electrical efficiency as a function of the ORC condensation temperature for different ORC evaporation temperatures in Mode 2 (CHP)

The ejector entrainment ratio as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures is illustrated in ***Figure. 4***.

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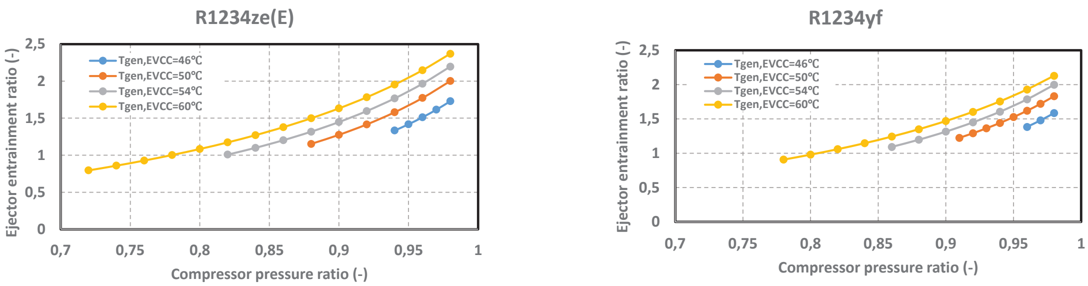

Figure. 4. Ejector entrainment ratio as a function of the compressor ratio for different EVCC generator evaporation temperatures

Higher compressor pressure ratios result in higher ejector entrainment ratios since the secondary flow enters the low-pressure port at a higher pressure and is more easily entrained by the primary flow. Furthermore, higher EVCC generator evaporation temperatures also result in increased entrainment ratios. This is because as the pressure of the primary flow is increased, it can more easily entrain low-pressure vapor. It can be observed that for each EVCC generator evaporation temperature, there is a minimum compressor pressure ratio which is necessary for the ejector to be able to operate. This is because, for lower pressure ratios, the primary flow can't entrain any low-pressure vapor exiting the cooling evaporator. The minimum required pressure ratio is positively correlated with the primary flow pressure (and hence the EVCC generator pressure). Notably, for an EVCC generator evaporation temperature of 46 °C, the compressor pressure ratio must be at least equal to approximately 0.94 (R1234ze) and 0.96 (R1234yf) for the ejector to be able to operate. On the other hand, if the EVCC generator evaporation temperature is increased to 60 °C, the compressor pressure ratio may be reduced to as low as 0.72 (R1234ze) and 0.76 (R1234yf).

If the compressor pressure ratio is higher than 0.9, essentially the whole compression is carried out by the compressor, which greatly assists the primary flow in entraining low-pressure vapor, reading to entrainment ratios above 1.5. Note that the ejector entrainment ratio is independent of the ORC evaporation temperature.

The EVCC electrical COP as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures is illustrated in ***Figure. 5***. On the same diagrams, the electrical COP of a conventional VCC operating within the same cooling evaporation and condensation temperatures (see ***Table 2***) is illustrated (straight line).

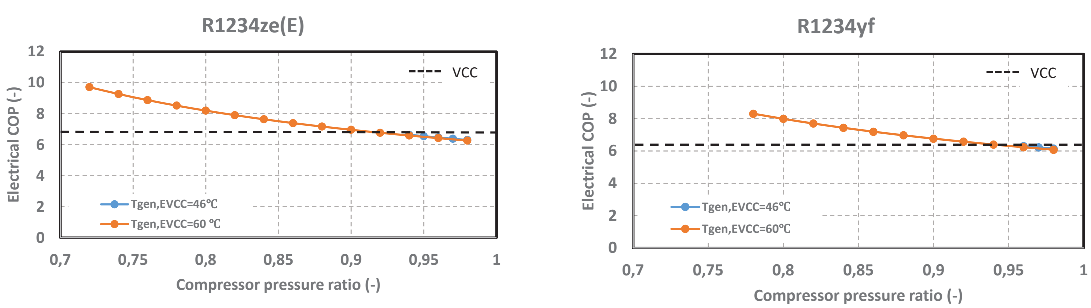

Figure. 5. Electrical COP of EVCC as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures in Mode 3 (CCP)

As the pressure ratio increases, both the cooling output and the electrical consumption of the compressor increase. However, the increase in the compressor electricity consumption is more significant, and thus the electrical COP ultimately declines. Notably, for pressure ratios above approximately 0.95, the electrical COP of the EVCC is lower than that of a conventional VCC. In this case, the secondary's stream pressure rise through the ejector is significantly low, thus the whole cycle is operating practically as a VCC. However, due to the existence of EVCC pump the electrical power required is slightly higher compared to VCC, thus the electrical COP takes lower values. Although lower compressor pressure ratios result in higher electrical COP values, they are also associated with lower ejector entrainment ratios and cooling output. Therefore, the compressor ratio should be selected to ensure 1) a higher electrical COP than that of a conventional VCC and 2) a sufficiently high cooling production for a particular application, and 3) it should be sufficiently high to enable the ejector operation.

Note that the EVCC generator evaporation temperature does not affect the electrical COP. As the generator evaporation temperature is increased, the primary flow mass flow rate in the generator is decreased because the heat recovered from the ORC is reduced (this happens because the pinch point in the EVCC generator is located at the end of preheating and the start of boiling for the EVCC working fluid, thus increasing the EVCC generator saturation temperature results in less heat being recovered from the ORC working fluid), while the enthalpy difference of the EVCC working fluid in the heat exchanger is increased (Eq. 5). Although the entrainment ratio of the ejector is increased, the secondary flow mass flow rate is reduced, because the reduction of the primary flow mass flow rate is more significant. Consequently, increasing the EVCC generator evaporation temperature only leads to lower secondary flow mass flow rates, equally resulting in both lower compressor power consumption and cooling output for a given thermal input to the EVCC, affecting in the same way both the numerator and denominator of the electrical COP (Eq. 12). Considering the above, the EVCC generator evaporation temperature should be kept as low as possible so that more heat is recovered from the ORC, but at the same time the ejector is capable of operating at low compressor pressure ratios (that also lead to better electrical COP).

The overall electrical efficiency of the ORC-EVCC as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures and ORC evaporation temperatures in Mode 3 is illustrated in ***Figure. 6***. For the results presented in both of these diagrams, the ORC working fluid is assumed to be R1233zd(E).

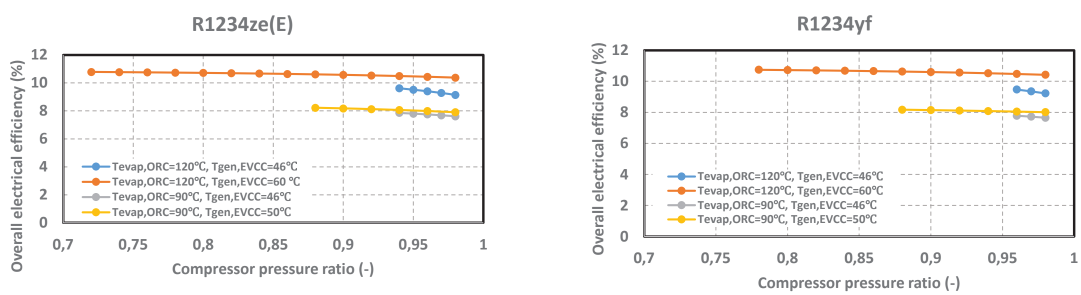

Figure. 6. Overall electrical efficiency of ORC-EVCC as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures and ORC evaporation temperatures in Mode 3 (CCP)

Higher overall efficiencies are observed for the highest ORC evaporation temperature of 120 °C because of the significantly higher power output of the ORC. Furthermore, the overall electrical efficiency is increased for higher EVCC generator evaporation temperatures, since these lead to lower compressor power consumption (as described previously). Finally, for higher compressor ratios, more electric power is required to drive the compressor, thus the net electric power output (and overall electrical efficiency) is decreased. Regardless, the impact of the compressor pressure ratio on the overall electric efficiency is not significant. Therefore, to maximize the overall electric efficiency, both the ORC evaporation and EVCC generator evaporation temperatures must be as high as necessary.

The overall combined electrical and cooling efficiency of the ORC-EVCC as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures and ORC evaporation temperatures is illustrated in ***Figure. 7***.

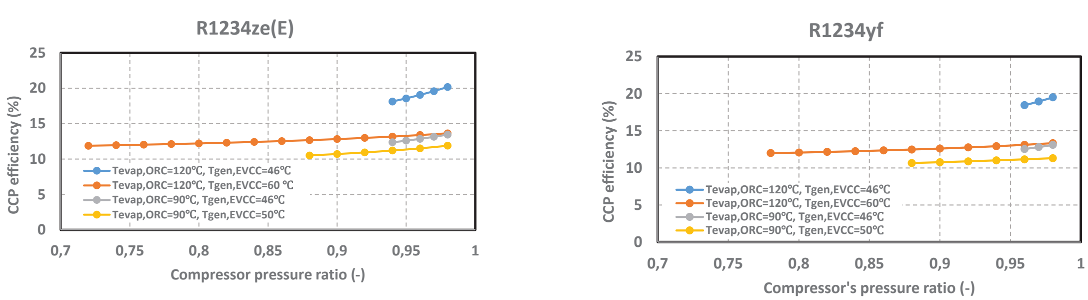

Figure. 7. *C*ombined cooling and power efficiency of ORC-EVCC as a function of the compressor pressure ratio for different EVCC generator evaporation temperatures and ORC evaporation temperatures in Mode 3 (CCP)

The variation of the CCP efficiency depends on the combined variation of the net power output and the cooling output of the system. The highest CCP efficiency is observed for the ORC evaporation temperature of 120 °C,

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for which the net power output is significantly high. The influence of the EVCC generator evaporation temperature on the CCP efficiency is not completely straightforward, since, as was mentioned previously, increasing the EVCC generator evaporation temperature results in both lower cooling output and also compressor power consumption. However, because the cooling output reduction is more significant, the CCP efficiency is reduced.

3.2. Design point selection

Based on the results presented in the previous section, R1233zd(E) and R1234ze show the best performance in the ORC and EVCC, respectively, as they result in higher power and cooling output. Therefore, these working fluids were selected to be used in the prototype. Moreover, higher ORC evaporation temperatures are favorable as they result in higher ORC electrical efficiencies and allow for more heat to be recovered by the EVCC, thus also leading to higher electrical COP values. Therefore, the highest possible ORC evaporation temperature (120 °C) is selected for the design of the prototype. Regarding the cold stream temperature rise in the recuperator, the maximum possible value of 20 K is selected, owing to its positive influence on the ORC electric efficiency in Mode 1. Regarding Mode 2, a condensation temperature of 50 °C is selected for the ORC, since it is sufficient for the production of hot water without severely penalizing the electrical efficiency. The final set of variables to be selected include the EVCC generator evaporation temperature and compressor pressure ratio. Based on the parametric investigations that were presented in the previous section, an ECC generator evaporation temperature of 52 °C and compressor pressure ratio of 0.84 represent a good compromise between a sufficiently high electrical COP and cooling output for the EVCC in Mode 3. The system design variables along with the system operating parameters in the three operating modes are summarized in **Table 3**.

Table 3 Design point operating parameters

	Mode 1 (Electricity-only)	**Mode 2 (CHP)**	**Mode 3 (CCP)**
ORC working fluid		R1233zd(E)	
ECC working fluid		R1234ze	
ORC evaporation temperature (°C)		120	
ORC condensation temperature (°C)	30	50	30
Cold stream temp. rise in recuperator (°C)	20	0	0
ECC generator evaporation temperature (°C)	-	-	52
Compressor pressure ratio (-)	-	-	0.84
Electric power output (kW_e)	11.35	8.06	10.57
Cooling output (kW_c)	-	-	4.80
Heating output (kW_{th})	-	85.5	-
ORC electrical efficiency (%)	12.84	10.01	12.62
CHP efficiency (%)	-	-	14.96
Electrical COP (-)	-	-	7.69
Overall electrical efficiency (%)	12.04	8.55	10.29
CCP efficiency (%)	-	-	14.96

4. Conclusions

The present study presented the thermodynamic design of a vessel engine waste heat recovery prototype based on a cascade Organic Rankine Cycle (ORC) and an ejector cooling-vapor compression cycle (EVCC) considering three operating modes: 1) electricity-only, 2) combined heat and power (CHP) via heat recovery from the ORC condenser and 3) combined power and cooling (CCP) from the simultaneous operation of the ORC and EVCC. To determine the preliminary design point, a series of parametric analyses were carried out to investigate the influence of key design parameters on the performance of the prototype and select its design point. R1233zd(E) and R1234ze were selected as the working fluids in the ORC and EVCC, respectively, as they led to the highest electrical efficiency and electrical COP in the two cycles. Furthermore, the selected design ORC evaporation temperature is 120 °C, which can lead to high ORC electrical efficiencies of up to 12.62% in electricity-only mode. The most suitable condensation temperature of the ORC in CHP mode was found to be 50 °C since it can allow the production of hot water at a sufficiently high temperature without penalizing the power output of the ORC. Regarding the design of the EVCC, higher EVCC generator evaporation temperatures do not influence the electrical COP but result in lower cooling outputs. Meanwhile, increasing the compressor pressure ratio has a negative impact on the electrical COP but leads to higher cooling output. Therefore, an EVCC generator evaporation temperature of 52 °C was selected, as it allows the operation of the ejector at a compressor pressure ratio of 0.84, which results in good electrical COP and substantial cooling output. Under the above conditions, the prototype ORC electric efficiency is up to 12.62% (electricity-only mode), while the overall electrical and CCP efficiency (CCP) are 10.29% and 14.96%, respectively. Finally, the electrical COP of the EVCC is 7.62, being superior to that of a conventional vapor compression cycle.

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Nomenclature

Abbreviations

CCP	combined cooling power
EVCC	ejector vapor compression cooling cycle
GWP	global warming potential
HFO	hydrofluoroolefin
ODP	ozone depletion potential
ORC	organic Rankine cycle

Variables

COP	coefficient of performance (-)
h	mass enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
P	power, kW
p	pressure, bar
$\dot{Q}$	heat duty, kW
T	temperature, °C

Greek symbols

η	efficiency (-)
ω	entrainment ratio (ejector)

Subscripts and superscripts

CCP	combined cooling and power
$cond$	condenser
$cool$	cooling
$crit$	critical
e	electric
$evap$	evaporator
exp	expander
G	generator (electric)
gen	generator (heat exchanger)
in	inlet
m	mechanical
M	motor
net	net
out	outlet
p	primary flow (ejector)
pre	preheater

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$pump$ pump

s secondary flow (ejector)

tot total

wh waste heat

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