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ScienceDirect

Energy Procedia 129 (2017) 551–558

Energy Procedia

www.elsevier.com/locate/procedia

IV International Seminar on ORC Power Systems, ORC2017
13-15 September 2017, Milano, Italy

Design and Off-Design Analysis of an ORC Coupled with a Micro-Gas Turbine

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Abstract

In the recent years, the possibility of recovering heat from gas turbine (GT) exhaust gases using Organic Rankine Cycles (ORC) have been largely explored. However, it is difficult to identify working fluids properly matching with micro-GT exhaust gases. For this reason, in the present work, the fluid selection and the plant layout optimization of an ORC which recovers the exhaust gases heat content of a 65 kW micro-gas turbine is presented. During the optimization process different plant configurations are considered: simple or regenerative and subcritical or transcritical. Exergy and economic analyses are also performed to estimate the exergy destruction rate and evaluate the economic feasibility of the optimized solutions. In order to find out the most suitable ORC unit and its behaviour, an off-design analysis is also performed using the commercial software Aspen Plus. Adopting a management strategy that maintains the turbine inlet temperature constant the best off-design performance is reached with Cyclopentane as working fluid.

Keywords: Waste heat recovery, Organic Rankine Cycle, Fluid selection, Design analysis, Off-design analysis

1. Introduction

In the last decades, the consumption of energy from all sources has increased due to the rapid growth of population. But, concerns about energy security and effects of fossil fuel emissions on the environment supported the use of renewable energy sources and natural gas (which is the least carbon-intensive fossil fuel) as well as the development of waste heat recovery units able to convert medium and low temperature heat sources into electricity.

The Organic Rankine Cycle (ORC) technology is one of the most promising methods to convert medium and low temperature heat into electricity because cycles using water as working fluid fail for technical and economic reasons in this temperature range [1,2].

The design of an ORC is a complicated task because the type and temperature of the heat source significantly influences the choice of the working fluid which in turn determines the configuration, the performance and the economy of the plant [2].

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1876-6102
10.1016/j.egypro.2017.09.192

For these reasons, several researchers worked on the ORC fluid selection and plant layout optimization using high, medium and low temperature heat sources. As an example, Delgado-Torres and Garcia-Rodriguez[3] performed an analysis and optimization of a low-temperature solar Organic Rankine Cycle while Astolfi et al.[4] performed a thermodynamic and techno-economic optimization of an ORC which recovers the geothermal fluid heat at a temperature of 120–180 °C. Tańczuk and Ulbrich[5] implemented a biomass-fired cogeneration plant in which the ORC is used as heat source for a small scale heat distribution system while investigations on the use of ORCs which recover waste hear from industrial processes, medium size gas turbine and biogas engine have been presented in, e.g., [6–9].

Despite the large variety of available works, only few of them are focused on recovering micro-gas turbine waste heat. As an example, Invernizzi et al.[10] and Clemente et al.[11] designed ORCs which recover the heat from the flue gases of a 100 kW_{el} commercially available small gas turbine fed by natural gas.

In the above mentioned researches, the Authors focused their attention on the ORC fluid selection and plant layout optimization at design point condition but, during the real operation, the temperature of the heat source may be different from the value assumed in the design phase. This aspect greatly affects the ORC output power. For this reason, an off-design analysis needs to be performed in order to select the ORC working medium, the plant layout and the management strategy which guarantees the highest ORC performance during the entire year of operation. As for fluid selection and plant configuration design, several works are available in the scientific literature. For example, Hu et al.[12], Cao and Dai[13] and Song et al.[14] analysed the ORC off-design behaviour under different operating conditions and management strategies.

However, despite the large number of published works, as far as the Authors know, no one has investigated the possibility of improving the performance of a 65 kW_{el} micro gas turbine (m-GT) by adding an ORC which recovers the exhaust gases heat considering both the design and off-design operation. For these reasons, in the present work, the Authors propose the ORC fluid selection and plant layout optimization at design point condition using an "in-house" optimization tool called "ORC-Plant Designer" while the ORC off-design behaviour is predicted using the software ASPEN Plus. The rest of the paper is organised as follow: in Section 2 the case study and the method adopted to perform the design and off-design analysis are described while, in Section 3, the optimization results and the plant off-design behaviour are presented and discussed. Finally, conclusions remarks are given in Section 4.

2. Case study and methodology

The case study is the power generation system of a small manufacturing industry. At the time of writing, a boiler is used to produce both the process heat and that for space heating while the electric load is partially covered with a photovoltaic system installed on the industry roofs. The photovoltaic plant and the boiler design power are 50 kW_{el} and 250 kW_{th}, respectively. Based on an electricity and heat consumption analysis, the industry owner has decided to install a micro gas turbine with a design power of 65 kW to self produce electrical energy. The m-GT is a Capstone C65 non-cogenerative turbine fed by natural gas with the design characteristics listed in Table 1.

Table 1. Micro-gas turbine characteristics.

Parameter	
Ambient Temperature [°C]	15
Relative Humidity [%]	60
Ambient Pressure [bar]	1
Net Power [kW]	65
Net Efficiency [%]	29
Exhaust Gases Mass Flow Rate [kg/s]	0.49
Exhaust Gases Temperature [°C]	309

Being the exhaust gases temperature relatively high, the industry owner wants to explore the possibility of adding a waste heat recovery unit to additionally improve the self production. To this purpose, the ORC is, firstly, designed and, then, its off-design behaviour analysed. The methodology adopted to perform the design and off-design analysis is briefly summarized in the following.

As widely known, the design of an ORC is a complicated task because the type and temperature of the heat source significantly influences the choice of the working fluid which in turn determines the configuration, the performance and the economy of the plant [15]. Therefore, the Authors have developed the ORC Plant Designer (ORC-PD tool) which is an "in-house" optimization code able to select the appropriate working fluid and plant architecture starting from different type of heat sources [16]. The tool is implemented in MATLAB environment and uses the MATLAB genetic algorithm available in the optimization toolbox.

The ORC-PD tool has been built in such a way that, with a single mathematical model, it is possible to design ORC modules operating with low-temperature (i.e. geothermal, solar, etc.) or medium-high temperature heat sources, including the exhaust gases released by, e.g., a gas turbine or an internal combustion engine.

Obviously, to cover this wide range of heat sources, appropriate working fluids and plant configurations need to be implemented into the code. So, a set of 115 pure fluids (including HydroCarbons, HydroFluoroCarbons, PerFluoroCarbons, Siloxanes, etc.) that can be good candidates for several types of ORC units have been added. These fluids are not available in a unique database, therefore the ORC-PD tool is linked with REFPROP [17] and CoolProp [18] databases. The use of two databases guarantees a larger number of fluids and compensates the lack of fluids that can occur using a single database.

Pure fluids are good working medium candidates but, as presented in several works (see, e.g. [19–21]), also zeotropic mixtures are suitable candidates due to their non-isothermal phase transitions during vaporization and condensation. For this reason, the possibility of adopting mixture is also included into the tool. Obviously, it is inefficient to test all the possible combinations among the implemented pure fluids. Hence, the method proposed in [22] is used to select the suitable components of the mixtures.

Different types of heat sources require different types of working fluids but also different plant configurations [1,2,23]. At the time of writing, in the "ORC-PD tool", the following plant architectures are implemented: basic scheme and recuperative configuration, two-stage plant [24], regenerative and recuperative architecture [25], dual pressure levels and dual fluid schemes [26].

In the above-mentioned configurations, the heat source medium and the ORC fluid streams exchange heat directly but, if the ORC medium is a flammable substance and the heat source temperature is really high, an intermediate thermal oil loop needs to be incorporated to avoid risky contacts between the heat source medium and the organic fluid. In this way, also flammable organic media can be adopted. In the ORC-PD tool, the schemes with the intermediate oil loop are also implemented. All the proposed cycles can be subcritical or transcritical. Note that the adoption of a sub- or transcritical cycle and the convenience of superheating the working fluid are optimization process results.

Several objective functions, such as net electric power, thermal or exergetic efficiency, net present value, etc., can be selected and optimized adopting a single or multi-objective optimization approach.

The tool input parameters are: heat source medium, inlet temperature ($T_{Hot,in}$), pressure ($p_{Hot,in}$) and mass flow rate ($\dot{m}_{Hot,in}$) of the heat source, pump isentropic efficiency, pump mechanical efficiency, electric motor efficiency, expander mechanical efficiency and electric generator efficiency while the variables that are optimized, for each working fluid, by the ORC-PD tool are: the heat source outlet temperature ($T_{Hot,out}$), the evaporation pressure of the organic medium (p_{ev}), the turbine inlet temperature (TIT), the ORC medium concentration if the fluid is a mixture (X_1), the minimum temperature difference in the evaporator, condenser and recuperator if the component is included into the cycle ($\Delta T_{pp,eva}$, $\Delta T_{pp,cond}$, $\Delta T_{pp,rec}$), the recuperator efficiency, again, if the component is present (E) and the condensation temperature (T_{cond}). The isentropic efficiency of the expander is estimated by the axial and radial flow turbines efficiency charts developed by Macchi and Perdichizzi[27] and Perdichizzi and Lozza[28], respectively. During the optimization process, the optimizer computes the size parameter and the volumetric flow rate, as defined in [27,28], then interpolates the efficiency charts and computes the turbine isentropic efficiency for both the axial and radial configuration, respectively. The higher isentropic efficiency is selected by the tool to perform the rest of the optimization. In future ORC-PD tool update, piston and scroll expander will be included because these kind of expanders are an efficient and cost effective alternative to radial turbine for mini and micro ORCs.

Each heat exchanger is discretized in "n" elements and, for each element, the tool computes the thermodynamic states of the two streams and performs the pinch point violation check. To better match the hot fluid profile with the cold one, the pinch-point position is variable. A high discretization of the heat exchangers is therefore needed to identify the pinch-point position and avoid pinch-point violations. The non-predefined pinch-point position and the non-fixed pinch value in all the ORC heat exchangers are innovative features implemented in ORC-PD tool. The Bell and Ghaly[29] method is used to correct the heat transfer coefficient obtained by Shah[30] in the case of mixtures.

Several checks are implemented into the code to avoid pinch-point violation in the heat exchangers, the presence of liquid at the turbine inlet, a low value of steam quality at the turbine outlet and a start of the evaporation process into the recuperator (if the component is present into the cycle). In addition, other checks are introduced to detect numerical issues caused by bad evaluation of the fluid thermodynamic properties acquired from fluids databases.

To sum up, after the inputs definition, for each working fluid candidate and each plant layout the design-point analysis is performed in order to determine the thermodynamic cycle that maximizes the objective function. Then, exergetic and economic analyses are also performed.

Note that, in the developed tool, the user can select to perform or not the exergetic and economic analysis and the cooling system method: cooling tower or water based architecture. In the first scheme, the condenser is fed with water that is then cooled down in a cooling tower while in the case of a water cooling system it is assumed to get the water from a river or a sea. In both cases, in the economic analysis, the devices investment and operating cost is taken into account. For the sake of compactness, refer to [16], for the energy, exergy and economic equations implemented into the ORC-PD tool and for the validation process results.

After the design point optimization process, the off-design analysis has be performed with the aim of selecting the organic medium and the management strategy which guarantee the highest part-load performance.

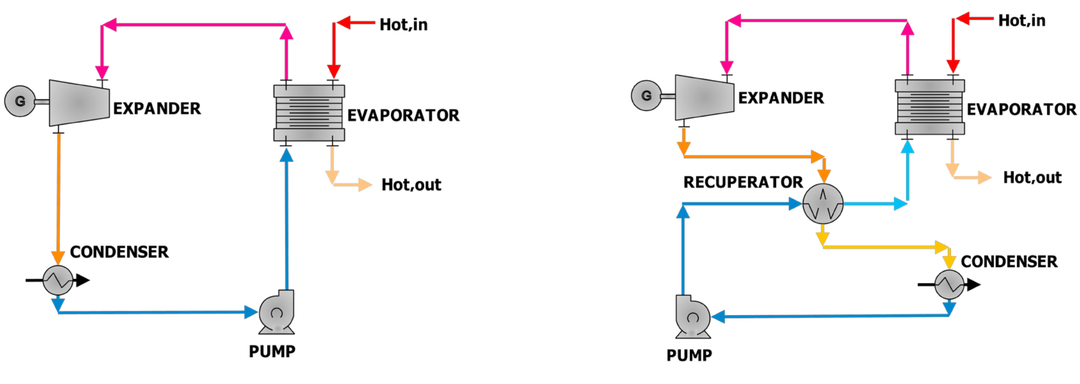
Firstly, using the commercial software ASPEN Exchanger Design and Rating (EDR), the ORC heat exchangers (HXs) are designed in detail. Then, the ORC mathematical model, able to predict the part-load behaviour, has been built in ASPEN Plus environment. The geometry of the heat exchangers, obtained with ASPEN EDR, are uploaded into the ASPEN Plus HX component in order to better predict the device behaviour. The turbine is modelled using the Stodola's equation [31] and setting nominal operating conditions (such as mass flow rate, inlet and outlet temperature and pressure), mechanical and isentropic efficiency. The pump model is built using a module which describes a centrifugal pump. The machine map, retrieved from a commercially available pump, is also included into the model.

3. Results and Discussion

Being the aim of the present work the design of an ORC which recovers the m-GT exhaust heat, the selected optimization goal is the maximization of the net electric power (P_{el}) produced by the ORC.

The hot source fluid is modelled as a mixture of nitrogen, oxygen, water, carbon dioxide and argon. The mass flow rate and temperature of the exhaust gases considered in the analysis are 0.49 kg/s and 309°C, respectively.

Regarding the ORC architectures, the Authors have considered only the basic and the recuperative configurations in order to reduce the plant cost and complexity. Being the recuperator efficiency a variable that is optimized by the tool, in practice, the ORC configuration is also a result of the optimization process. In Figure 1, the possible ORC architectures are depicted.



(a) (b)

Fig. 1. (a) ORC basic scheme and (b) recuperative ORC architecture.

The assumed parameters, based on both data available in the scientific literature and plant manufacturers' knowledge, are listed in Table 2 while the upper and lower bound (UB and LB) fixed for the variables that are optimized are listed in Table 3.

Table 2. Assumed machines and components efficiency for small-scale ORC units.

Parameter	
Pump isentropic efficiency [-]	0.80
Pump mechanical efficiency [-]	0.92
Pump electric motor efficiency [-]	0.90
Turbine mechanical efficiency [-]	0.90
Electric generator efficiency [-]	0.92

Table 3. Upper and lower bounds (UB and LB) fixed for the variables that are optimized.

Parameter	LB	UB
Heat source outlet temperature, $T_{Hot,out}$ (°C)	90	$T_{Hot,in}$
Evaporation pressure of the organic medium, p_{ev} (bar)	p_{cond}	$1.3 \cdot p_{crit}$
First mixture component concentration, X_1 (-)	0	1
Recuperator efficiency, E (-)	0	0.8
Turbine Inlet Temperature, TIT (°C)	TIT_{min}	TIT_{max}
Condensation temperature, T_{cond} (°C)	30	90
Minimum temperature difference in the evaporator, $\Delta T_{pp,eva}$ (°C)	25	100
Minimum temperature difference in the recuperator, $\Delta T_{pp,rec}$ (°C)	20	100
Minimum temperature difference in the condenser, $\Delta T_{pp,cond}$ (°C)	10	100

p_{cond} and p_{crit} are the condensation and critical temperature of the working medium, respectively while TIT is the Turbine Inlet Temperature. TIT_{min} assumes the value of T_{cond}+10°C while TIT_{max} assumes the lower value among the ones computed with the following equations:

$$TIT_{max} = T_{Hot,in} - \Delta T_{pp,evaporator} \quad or \quad TIT_{max} = T_{fluid} \quad or \quad TIT_{max} = T_{decomposition} \tag{1}$$

where T_{fluid} is the maximum temperature at which the fluid is defined in REFPROP/CoolProp database while $T_{decomposition}$ is the maximum temperature at which the fluid thermochemical stability is guaranteed. As pointed out in [32], the thermal stability of the organic fluid is a key parameter, therefore, if the decomposition temperature is available in the scientific literature, TIT_{max} is set equal to this value. As an example, the Cyclopentane decomposition temperature is set equal to 300°C as suggested in [32,33].

The minimum temperature difference in the heat exchangers strongly affects the design and, consequently, the cost of the devices. For this reason, different values for the lower bound have been selected in the case of evaporator, recuperator and condenser. The highest value has been assumed in the evaporator in order to design a heat exchanger with an acceptable volume, while the smallest value has been assumed for the condenser. In this way, it is possible to reduce the condensation temperature despite the adoption of a cooling tower. Obviously, solutions that violate the minimum acceptable temperature difference or other constrains are automatically erased.

The minimum admissible steam quality at the turbine outlet is set equal to 0.85 while it is assumed that the condenser is fed by water coming from a cooling tower which operates with air at 30°C. In order to take into account the purchasing cost of the cooling system, the optimizer computes also the water and air mass flow rates.

The exergetic and economic analyses are performed using the equations listed in [16] while the m-GT purchasing cost, the expected plant life, the plant availability factor, the interest rate, the corporate tax rate, the electricity sell price and the fuel specific cost are assumed equal to 900 $/kW, 15 years, 0.85, 5%, 0.40, 0.25 $/kWh, 0.865 $/kg.

The genetic algorithm population and generation are set both equal to 250 while the crossover fraction and the migration fraction are assumed equal to 0.8 and 0.2, respectively.

The three most promising pure fluids and their main characteristic are listed in Table 4.

Table 4. Most promising pure fluids.

Fluid	P_{el} [kW]	$T_{Hot,out}$ [°C]	p_{ev} [bar]	TIT [°C]	p_{cond} [bar]	E [-]	$\dot{m}_{ORC}$ [kg/s]	η_{ORC} [%]	NPV [M$]	IP [-]	SPB [y]
Cyclopen	12.58	101.8	23.2	190.8	1.85	0	0.210	11.5	0.033	0.168	8.9
R141b	12.42	98.8	34.4	206.8	3.15	0	0.386	11.2	0.016	0.117	9.3
Cyclohex	12.40	102.9	10.3	183.9	0.68	0	0.202	11.4	0.032	0.166	8.9

The ORC layout has a base configuration without the recuperator while the expander is a radial turbine. Cyclopentane guarantees the highest net electric power and Net Present Value (NPV) while Cyclohexane and R141b guarantee, respectively, the lowest evaporating pressure and the highest Simple Pay back (SPB).

Being the net electric power produced by the three fluids more or less the same, the off-design analysis has been performed for the three fluids with the aim of finding the one which reaches the best off-design behaviour. The commercial software ASPEN EDR is used to design the ORCs heat exchangers. As an example, in Table 5, the geometry of the condenser and of the evaporator for the plant which uses Cyclopentane is reported.

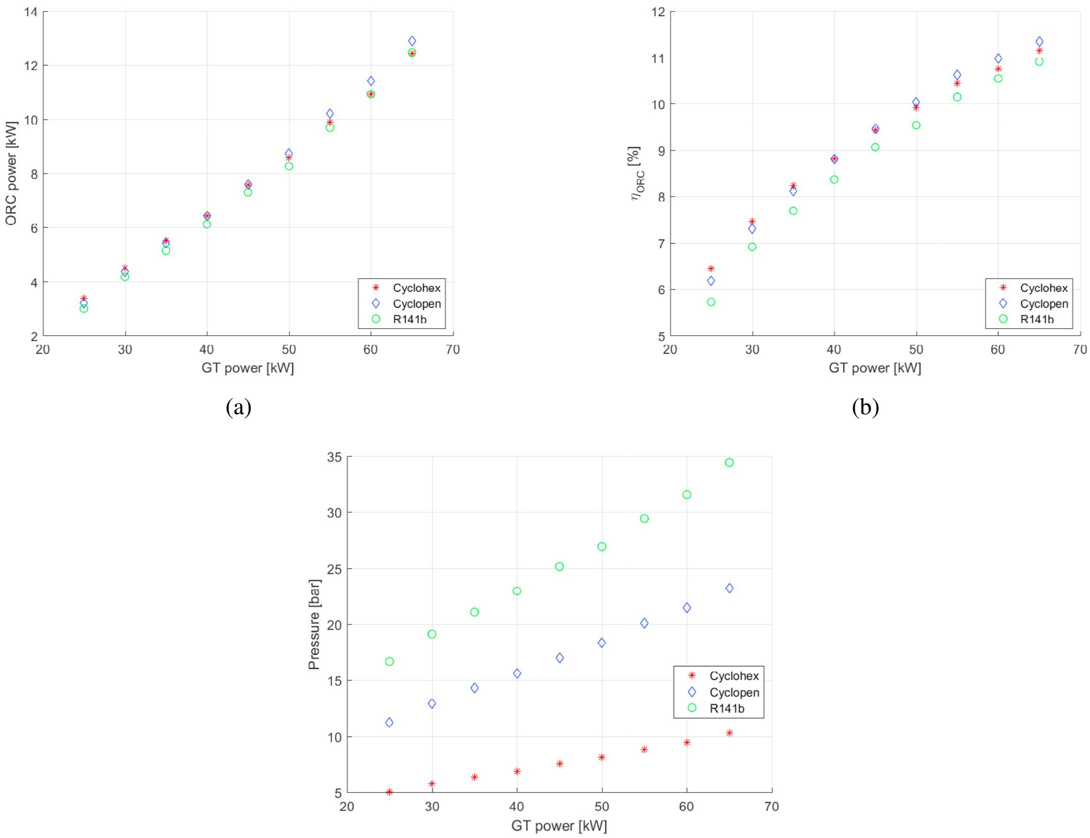
Table 5. Geometry specification of the heat exchangers using Cyclopentane as working medium.

	Evaporator	Condenser
Thermal power [kW]	106.6	94.8
Hot fluid mass flow rate [kg/s]	0.49	0.21
Cold fluid mass flow rate [kg/s]	0.21	2.23
Heat transfer area [m^2]	35.3	4.9
Heat transfer coefficient [$W/(m^2K)$]	95.8	1250.9
Tube external diameter [mm]	8	10
Tube thickness [mm]	1	1
Tube length [mm]	2410	1830
Shell inner diameter [mm]	316	163
Number of tubes	594	88
Number of passes - shell size	1	1
Number of passes - tube side	1	2
Tube pitch [mm]	10.0	12.5
Pressure drop - tube side [bar]	0.03	0.12
Pressure drop - shell size [bar]	0.09	0.05

ASPEN EDR optimizes the ORC heat exchangers geometry using as input the thermodynamic state points computed with the ORC-PD tool. In the present work, it is assumed to adopt a Shell & Tube Exchanger designed following the TEMA style vessels. The code also provides the purchasing cost of the optimized heat exchanger configuration. For the case of Cyclopentane, the evaporator and condenser predicted costs using ASPEN EDR equations are, respectively, 30% and 60% lower than the ones predicted with the ORC-PD tool. This means that the performed economic analysis overestimates the SPB and underestimates the NPV. To this purpose, the Authors are working on the integration between the ORC-PD tool and the ASPEN EDR to improve the reliability of the economic analysis.

After the heat exchangers design phase, the ORC model has been built in ASPEN Plus including the detailed heat exchangers geometry, the pump performance curves and the Stodola equation into the turbine model. Then, the off-design performance of the three ORC units operating with different fluids have been predicted varying the m-GT exhaust gases mass flow and temperature (as provided by the machine manufacture) while the ORC turbine inlet temperature has been maintained constant varying the pump rotational speed with an appropriate controller. Note that different control strategies have been tested but, for the sake of compactness, in this work only the one which prescribes to maintain the TIT constant has been presented.

Based on the results reported in Fig. 2, it is possible to notice that the heat exchangers geometry and the pump and turbine characteristics do not modify the ORC design point performance obtained with the ORC-PD tool. Therefore, Cyclopentane remains the working fluid that guarantees to reach the maximum ORC power and efficiency at full load. Regarding the off-design performance, Fig. 2a shows that Cyclopentane guarantees the highest ones compared to



(a) (b) (c)

Fig. 2. (a) ORC power vs m-GT power (b) ORC efficiency vs m-GT power (c) ORC evaporation pressure vs m-GT power.

R141b and Cyclohexane in the m-GT power range from 40 to 70 kW while Cyclohexane is better in the range 25-35 kW. Similar considerations can be made for the ORC efficiency while Fig. 2c shows that the evaporating pressure of R141b is the highest one. Therefore, to reduce the plant cost, it could be better to use Cyclohexane. But, it is important to notice that the proposed management strategy is not the most suitable one because if reducing the m-GT load and maintaining the TIT fixed, the fluid superheating increases as well as the heat that has to be rejected to the environment via the condenser.

It is possible to conclude that Cyclopentane is the fluid which guarantees the highest performance both at design and off-design conditions using the implemented management strategy. Note that the Authors have tested several management strategies (not reported in the present work) including the one which prescribes to control the condenser hotwell level: a management strategy used by several ORC manufactures.

4. Conclusions

In the present work, the possibility of improving the performance of a micro gas turbine installed in an Italian manufacturing industry has been evaluated. Firstly, using an “in-house” optimization tool, the ORC unit has been designed. Then, using a commercial software, the off-design behaviour of the most promising ORC configurations has been predicted. In the ORC off-design models, the heat exchangers geometry and the pump and turbine characteristics have been taken into account. Results show that Cyclopentane is the fluid which guarantees the best performance both at design and off-design conditions while the non-regenerative ORC plant layout is the configuration which guarantees the highest simplicity and the lowest cost.

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