

Evaluation of ORC modules performance adopting commercial plastic heat exchangers

S.L. Gómez Aláez ^{a,*}, P. Bombarda ^a, C.M. Invernizzi ^b, P. Iora ^b, P. Silva ^a

^a Politecnico di Milano, Energy Department, via Lambruschini 4, 20156 Milano, Italy

^b Università di Brescia, Department of Mechanical and Industrial Engineering, via Branze 38, 25123 Brescia, Italy

ABSTRACT

In this paper the possible replacement of conventional metallic heat exchangers with plastic components is investigated with reference to low size Organic Rankine Cycles, aiming at a reduction of the plant investment cost.

A thermodynamic optimization of a 20 kW regenerative ORC plant, representative of a low temperature (<140 °C) heat recovery application, has been carried out according to the presently available data for plastic shell and tubes heat exchangers offered on the market.

N-heptane was selected as the working fluid, thanks to the capability to operate within the pressure limits for evaporation and condensation processes imposed by the adoption of plastic components.

Finally, the potential economic benefit of the plastic solution in comparison with conventional heat exchangers made of carbon steel was evaluated for the whole plant; the case of enhanced materials adoption, which is mandatory for the evaporator in presence of corrosive heat source media, was also considered.

It turns out that advantages of the proposed solution become appreciable whenever the presence of corrosive heat source media requires the use of materials other than carbon steel. For instance, for a plant availability of 5000 h/year and discount rate of 10%, we obtain a cost of the produced electricity of 94.8 \$/MW h, 95.4 \$/MW h, 101.5 \$/MW h, and 118.9 \$/MW h respectively for plastic, carbon steel, stain-less steel and titanium solutions.

* Corresponding author. Tel.: +39 0223993817, mobile: +34 649447754.

E-mail address: sgomezalaez@gmail.com (S.L. Gómez Aláez).

Nomenclature

U	global heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
k	thermal conductivity (W m K^{-1})
Q	thermal power (W)
A	area (m^2)
ΔT_{lm}	mean logarithmic difference in temperature (K)
Re	Reynolds number (dimensionless)
Nu	Nusselt number (dimensionless)
Pr	Prandtl number (dimensionless)
C_p	specific heat ($\text{J kg}^{-1} \text{K}^{-1}$)
h	enthalpy (J kg^{-1})
D	diameter (m)
L	length of the heat exchanger tubes (m)
m	mass flow (kg/s)

Greek symbols

α	convective coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
μ	dynamic viscosity (Pa s)
ρ	density (kg m^{-3})
δ	heat exchanger tubes thickness (m)

Subscripts

1–6	state points
i	in
o	out
cond	condenser
evap	evaporator

Abbreviations:

BOP, balance of plant;
ECTFE, polyethylene chlorotrifluoroethylene;
ETFE, polyethylenete tetrafluoroethylene;
FEP, fluorinated ethylene propylene;
GWP, global warming potential;
N/A, non available;
ODP, ozone depletion potential;
ORC, Organic Rankine Cycle;
O&M, operation and maintenance;
PCTFE, polychlorotrifluoroethylene;
PE, polyethylene;
PFA, perfluoroalkoxy poly-mer;
PP, polypropylene;
PTFE, polytetrafluoro ethylene;
PVC, polyvinylchloride;
PVDF, polyvinylidene fluoride.

1. Introduction

The main purpose of this study is to investigate whether it is possible to use plastic heat exchangers as an alternative to conventional carbon steel heat exchangers in an Organic Rankine Cycle.

Due to intrinsic potential advantages, plastic heat exchangers application has already been considered for many years (one of the first possible field of application considered was desalination, see for example [1]) and a comprehensive review on plastic heat exchanger development can be found in [2]. However, to the authors' knowledge, no application of plastic heat exchangers for an Organic Rankine Cycle was yet realized.

The Organic Rankine Cycle represents a well established technology adopted in order to produce electric power or combined heat and power from low temperature sources, as waste heat, biomass, geothermal sources, and possibly others. This technology is suitable for application in a very large power range, starting from a few kW up to tens of MW. As of today, due to plant economics, which strongly depends on the plant size, it is mostly used for plants with a minimum size on the order of a few hundred kW. However, several studies (see for example [3–7]) and many research projects for very small plants are currently ongoing, for the most different applications: heat recovery, renewable sources, buildings and transport [8], exploiting waste heat recovery from Diesel engines [9].

In case of very small plants, the possible adoption of plastic heat exchanger could be advantageous because of the lower cost with respect to conventional heat exchanger; moreover, in case of heat recovery from corrosive or dirty fluids, a further advantage would come from the material intrinsic corrosion resistance and scale deposition inhibition; or eventually, damaged heat exchangers, if low cost, could be considered as disposable. Unluckily, the operating range (temperature and pressure) of plastic heat exchanger is at the moment narrower than that of conventional metallic heat exchanger: this limit prevents their adoption in supercritical ORC cycles, and may restrict their application even in subcritical cycles.

When feasible, application of plastic heat exchanger would be interesting for distributed power generation, and above all in case of low-temperature geothermal sources, whereby a binary cycle is employed and the geothermal fluid is usually corrosive, with also a high scaling potential.

Though the power size of geothermal plants is commonly higher than 1 MW, R&D activity related to very small power plants was performed already in the past: the geothermal pilot plant of Mulka, (Australia) [10] built in 1986, generated 15 kW electric power from hot water at low temperature (86 °C) and an application is found in a hotel in Ischia (Italy), using a 6 kWe ORC fed by solar and geothermal energy [11]; both applications would be compatible with plastic heat exchanger adoption. The generating potential of geothermal “slim holes” was thoroughly investigated [12], assessing that off-grid geothermal rural electrification projects down to 100 kW range using slim holes is theoretically feasible, but cost and reliability were still to be proved.

Presently, a German-Indonesian collaboration project [13], aimed at sustainable geothermal energy development, considers 50 kW as the minimum range of power for exploitation with a binary cycle; several geothermal plants, moreover, with power in the range of a few tens of kW, were recently built, mainly in Japan [14].

Some progress is still needed for a broad application of plastic heat exchangers, but significant improvements are soon expected: it is therefore worthwhile to be concerned with plastic heat exchanger application.

2. Plastic heat exchangers

There are at present all over the world several companies aimed at the production of plastic heat exchangers; among those companies we can find TMW (France), Aetna plastic (United States of America), Polytetra (Germany), HeatMatrix Group B.V. (Netherlands), Fluorotherm (United States of America), Ametek (United States of America) and Kansetu (Japan). The type of heat exchangers produced is widely varied, shell and tube and plate heat exchangers are available, as well as particular heat exchangers (e.g. immersion, hollow plates, tube plate). The materials most commonly used are PVDF and PFA, for high temperatures, and PE or PP for low temperatures, but, depending on the company, other materials may be selected, usually fluoropolymer materials (ECTFE, PTFE, FEP, ETFE, PCTFE), or PVC [15–21].

There are several advantages related to plastic heat exchangers:

- they resist to chemical attacks at fairly elevated temperatures
- the fouling by scale deposition is minimized, so a constant operating efficiency can be assumed
- maintenance cost is low.

Thanks to these advantages, the applications where plastic heat exchangers are commonly used are related to the heating or cooling of corrosive media (acids, sea water...), e.g. in the galvanizing industry, desalination, desulphurization of flue gases, food, pharmaceutical or textile industry. A widespread utilization of plastic heat exchangers is nevertheless limited by the modest mechanical properties of polymers (low tensile strength), low thermal conductivity and somewhat limited maximum operating temperature.

The main purpose of this section is to collect, summarize and compare information regarding maximum operating conditions for plastic heat exchangers. Despite the presence of several companies on the market, just a few technical data are publicly available: most of the data herein presented are derived from Calorplast (Aetna group dedicated to the production of plastic heat exchangers) data sheet [22].

Firstly, carbon steel shell and tube heat exchangers costs have been collected for low sizes in order to investigate the variation of cost with the heat transfer surface. The results are shown in Fig. 1. The cost values were obtained from [23]. As the values obtained for carbon steel heat exchangers were given at the year 2002,

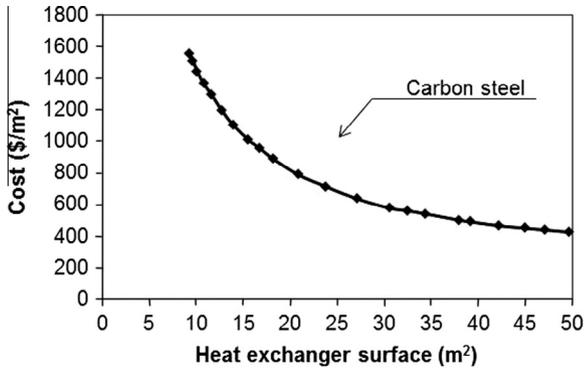


Fig. 1. Specific cost for carbon steel shell and tube heat exchanger.

inflation rate has been used in order to update the values to 2014 [24]. As it can be observed from Fig. 1, if the heat transfer area needed is small, the specific cost increases significantly as the size decreases, which could lead to the opportunity of using plastic heat exchangers.

Secondly, once the appropriate case is found, it is of paramount importance to consider the maximum operating conditions that can be admissible for plastic heat exchangers, which are much more restrictive with respect to steel adoption.

As shown in Fig. 2, PE heat exchangers imply more restrictive maximum working conditions than PVDF heat exchangers [25]. In addition, it has to be taken into account that these maximum conditions represent an absolute limit that cannot be definitely exceeded, but a lower limit may exist depending on the selected fluid flowing through the heat exchanger [26]; this must be considered when selecting the ORC working fluid.

As an outcome, it can be appraised from Figs. 1 and 2, that plastic heat exchanger could be proposed only for small scale, low temperature ORC; if the plant scale is large, it is in fact more cost effective to adopt conventional steel heat exchanger, while, if the operation temperature is above 140 °C, there is no chance at present to adopt plastic heat exchangers.

3. Working fluid selection

Adoption of plastic heat exchangers requires that the operating parameters of the utilized fluids must comply with the temperature and pressure restrictions shown in Fig. 2.

We selected therefore the commercial oil Therminol VP-1 [27] for the hot source, which feeds the evaporator, assuming the following values: mass flow 5 kg/s, inlet temperature 140 °C (i.e. the current limit value for PVDF) and pressure 2 bar; for the cold

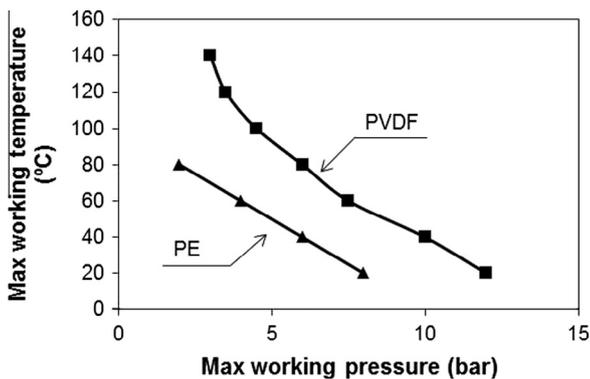


Fig. 2. Maximum operating conditions for plastic shell and tube heat exchangers.

source, supplying the condenser, we selected water, assuming that it is available at 20 °C and atmospheric pressure.

The selection of the ORC working fluid has to fulfill the general technological and environmental criteria, widely discussed in literature (among others: [4,28–33]), such as suitable fluid thermodynamic properties, no toxicity, no or low flammability, low cost, low GWP and no ODP impact.

Moreover, in the present case, the adoption of plastic heat exchangers results in the following additional constraints:

- Boiling temperature at atmospheric pressure within 80 and 110 °C.
- Saturation pressure at 35 °C higher than 0.1 bar, in order to keep a reasonably high condensing pressure inside the condenser.

Under these conditions, from a search in both Aspen Properties Database and the database provided by [34] an initial list of 65 eligible fluids was obtained, to be reduced to 26 when taking into account environmental concerns (ODP and GWP) [35]. The final list of eligible fluids is presented in Fig. 3. All the fluids are hydrocarbons, which, given the criteria adopted in the selection, show similar thermodynamics properties. It is worth noting that, although flammable, hydrocarbons are commonly used as working fluids in ORC plants [36,37].

As an additional criterion for the fluid selection, the isentropic volumetric flow rate ratio of the turbine was calculated for different working fluids at fixed evaporation and condensation temperatures. The fluids with the lowest volume flow ratio, to be preferred in terms of the design of the expander [37], are listed in Table 1.

Among these, n-heptane, although it is a highly flammable and mildly toxic fluid, shows the best compatibility with the PVDF adopted for the evaporator according to the information available from the heat exchanger's manufacturer [26], allowing an operating temperature up to 100 °C and was therefore selected as the cycle working fluid.

As already pointed out, the choice of a hydrocarbon as working fluid in a low temperature ORC is rather common, although the selection of n-heptane is rather unusual (even though, studies using it as working fluid in ORC can be found [38]); the reason is to be found in the low operating pressure requirement, which forces to adopt a working fluid with high critical temperature. Given a family of fluids (in this case hydrocarbons), the critical temperature increases with the molecular complexity: as explained in [37], this entails a positive slope of the saturated vapor in the T-s plane, which will then result in a superheated end expansion point.

4. ORC cycle configuration and thermodynamic results

The proposed configuration is a regenerative Organic Rankine Cycle, which therefore exploits an evaporator, a condenser and a regenerator; the simplified plant scheme is shown in Fig. 4. In addition to efficiency improvement, the adoption of a regenerative cycle is reasonable considering the slope of the saturated vapor in the T-s plane, see Fig. 5, and moreover it allows a lower temperature at the condenser inlet and a lower thermal power to be discharged by the condenser.

Performance evaluations were conducted assuming to adopt commercial plastic heat exchangers, if applicable.

For the evaporator's material, PVDF was selected; this is in fact the hottest heat exchanger of the cycle, and the adoption of PVDF is mandatory.

The selection of the regenerator's material will be tackled in the following, while its sizing was performed so that the condenser

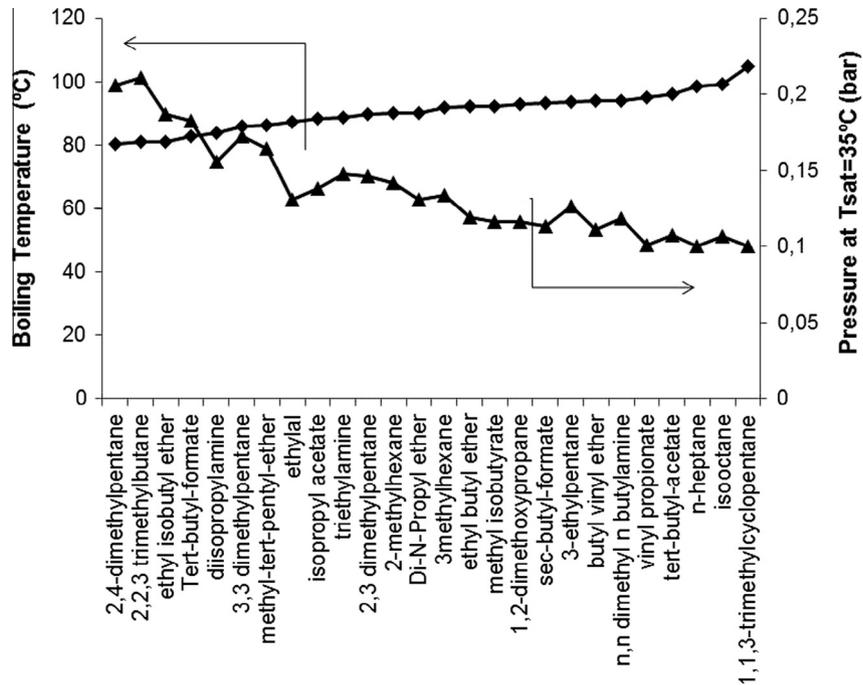


Fig. 3. Fluids complying with the assumed criteria for boiling temperature, condensation pressure, zero ODP value and low GWP value.

Table 1

Volume flow ratio of different working fluids, assuming a mass flow of 1 kg/s.

	V_i (m ³ /s)	Volume flow ratio
2,4-Dimethylpentane	0.16	8.41
2,2,3-Trimethylbutane	0.17	7.97
3,3-Dimethylpentane	0.20	8.44
2,3-Dimethylpentane	0.22	8.96
2-Methylhexane	0.22	9.23
3-Methylhexane	0.23	9.30
3-Ethylpentane	0.24	9.33
n-Heptane	0.28	10.11
Isooctane	0.25	9.47

inlet temperature is 40 °C, a value that can comply with the limit for PE, adopting n-heptane as working fluid. It is advantageous to select PE for the condenser; in fact, the price of a PE heat exchanger is lower than that of a PVDF heat exchanger.

4.1. Cycle assumptions

The following assumptions have been taken into account to carry out the simulations:

- the pressure of condensation is 0.1 bar, corresponding to a condensation temperature of 34.87 °C for the selected working fluid

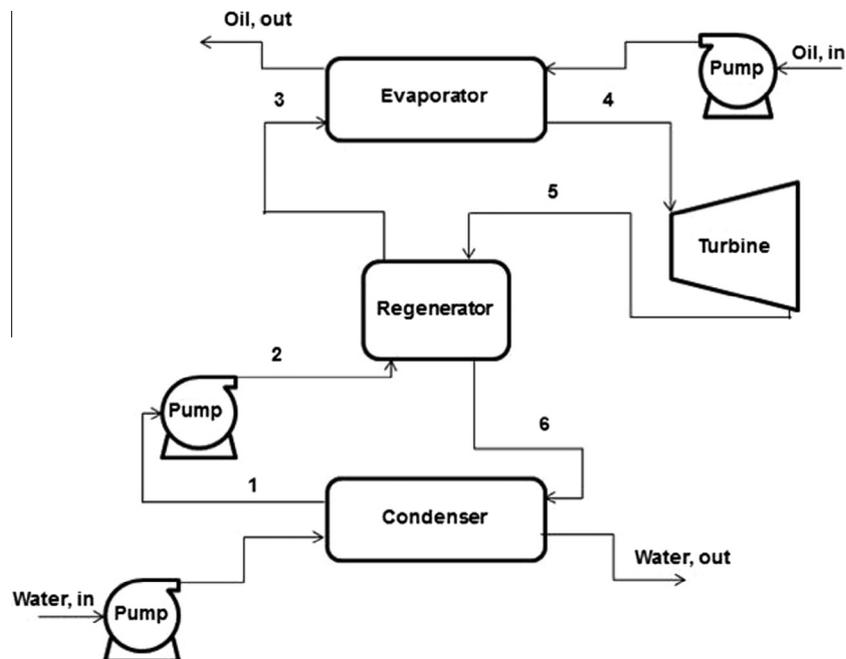


Fig. 4. Simplified regenerative Organic Rankine Cycle scheme.

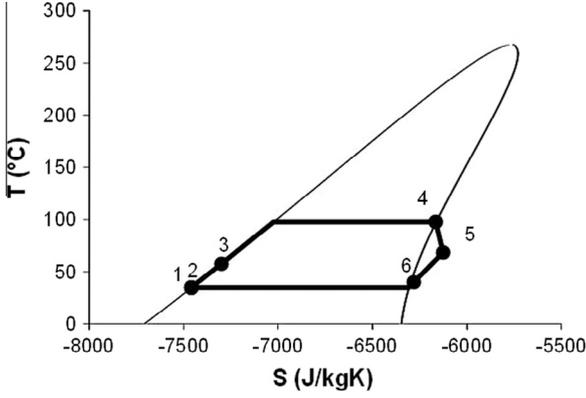


Fig. 5. T-S diagram for the optimized thermodynamic cycle.

- as mentioned before, the hottest temperature in the condenser cannot exceed 40 °C, so that the temperature of the hot flow, at the outlet of the regenerator, (stream 6 in Fig. 5) is fixed at 40 °C
- all the pumps isentropic efficiencies were fixed to a value of 0.7 and the turbine efficiency was fixed to a value of 0.8
- the software Aspen Plus is used to carry out the simulations, and the following equations of state were selected: Peng Robinson for n-heptane, Ideal fluid for Therminol VP-1 and Steam-Table for water.

4.2. Calculations

The evaporation pressure is the parameter to be optimized in order to maximize the net power output, so different cases have been studied with pressure values ranging from 0.9 bar to 1.3 bar.

The same geometry has been selected for the evaporator and condenser: reference has been made to the catalogue of a manufacturer, and the shell and tube heat exchanger with maximum allowable heat transfer area was selected (16.72 m²). This heat exchanger is constituted by 195 tubes, with an internal diameter of 4.8 mm, an outer diameter of 6 mm and a length of 5 m [22].

Preliminary evaluation of the global heat transfer coefficient (U) is required in order to perform cycle calculation: it can be estimated given the heat exchanger geometry and the thermodynamic properties of the fluids. The selected flow scheme considers the condensing or evaporating fluid on the shell side; while the single phase fluid (water or thermal oil respectively), is on the tube side.

The overall heat transfer coefficient depends on the following terms: inside and outside convective heat transfer coefficients (α_i , α_o) and wall thermal resistance (fouling factors are negligible for most applications of plastic heat exchangers [22]):

$$Q = U \cdot A \cdot \Delta T_{lm} \quad (1)$$

$$U = \frac{1}{\frac{1}{\alpha_i} + \frac{\delta}{k} + \frac{1}{\alpha_o}} \quad (2)$$

The convective coefficient α_i is calculated using:

$$\alpha_i = \frac{Nu \cdot k}{D} \quad (3)$$

where the Nusselt number can be calculated using the following correlations, valid for a single phase fluid flowing inside the tubes. Depending on flow conditions, i.e. laminar or turbulent flow regime [39] the proper correlation is selected.

Laminar ($Re < 2300$):

$$Nu = 3.66 + \frac{0.065 \cdot \left(\frac{D}{L}\right) \cdot Re \cdot Pr}{1 + 0.04 \cdot \left(\frac{D}{L}\right) \cdot Re \cdot Pr}^{2/3} \quad (4)$$

Turbulent ($Re > 2300$):

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^n \quad (5)$$

where n takes the value of 0.3 if the fluid is being cooled or 0.4 if it is being heated.

$$Re = \frac{\dot{m} \cdot D}{\mu \cdot A} \quad (6)$$

$$Pr = \frac{Cp \cdot \mu}{k} \quad (7)$$

The heat transfer coefficient α_o is also calculated by means of a proper correlation, selected according to the heat transfer process considered, i.e. respectively the preheating of the liquid until the saturation temperature, the evaporation, and the condensation processes [40]. The values for the thermal conductivity of the materials selected in this study are: 47 W/m K for carbon steel, 0.18 W/m K for PVDF and 0.42–0.51 W/m K for PE [22].

The pressure drop inside the heat exchanger (single-phase fluid side) is calculated using the correlation given by the manufacturer [22], for turbulent conditions:

$$\Delta p(\text{bar}) = 0.02672 \cdot v \left(\frac{m}{s}\right) \cdot L(m) \quad (8)$$

Once the global heat transfer coefficients are known, the heat exchanger equations for the evaporator and the condenser can be immediately solved [41]:

$$Q_{evap} = U \cdot A_{evap} \cdot \Delta T_{lm} \quad (9)$$

$$Q_{evap} = \dot{m}_{oil} \cdot (h_{i,oil} - h_{o,oil}) \quad (10)$$

$$Q_{evap} = \dot{m}_{orc} \cdot (h_4 - h_3) \quad (11)$$

The mass flow of the working fluid can be found from the evaporator equations.

$$Q_{cond} = U \cdot A_{cond} \cdot \Delta T_{lm} \quad (12)$$

$$Q_{cond} = \dot{m}_{water} \cdot (h_{o,water} - h_{i,water}) \quad (13)$$

$$Q_{cond} = \dot{m}_{orc} \cdot (h_6 - h_1) \quad (14)$$

The mass flow of cooling water required can be found from the condenser equations.

All the simulations have been developed using the software Aspen Plus: the maximum power is obtained for a pressure equal to 1 bar, and the results for this optimized case are shown in Table 2. The economic analysis carried out in the next paragraph will be also based on this optimized case.

Prior to proceed to the economic analysis, it may be interesting to discuss the desirable features of the regenerator and the turbine.

In the regenerator, no phase change is involved: the heat transfer coefficient of the gaseous flow, though depending on flow velocities, is definitely lower than the liquid flow coefficient. The adoption of a shell and tube heat exchanger is not recommended, insofar a higher heat transfer surface would be required on the gas side. In a conventional plant, a metallic finned tube heat exchanger would be selected. In our case, the adoption of a plastic shell and tube heat exchanger would not be profitable, but a compact heat exchanger, for example a polymeric extruded microchannel-structured heat exchanger could be proposed [42]; no detailed calculation is however possible at the moment, due to the lack of published data, so that a metallic heat exchanger has to be considered.

As far as the turbine sizing is concerned, design values are taken from Table 2, for the optimized case.

Table 2
Simulation results for the optimized thermodynamic cycle.

Oil inlet temperature (°C)	140
Oil outlet temperature (°C)	123.31
Oil pressure (bar)	2
Oil mass flow (kg/s)	5
Evaporation temperature (°C)	97.97
Evaporation pressure (bar)	1
Condensation temperature (°C)	34.87
Condensation pressure (bar)	0.1
Organic fluid mass flow (kg/s)	0.373
Cooling water inlet temperature (°C)	20
Cooling water outlet temperature (°C)	22.95
Cooling water mass flow (kg/s)	10.99
Q, evaporator (kW)	155.21
ΔT_{lm} , evaporator (°C)	37.12
UA evaporator (W/K)	4181
Q, condenser (kW)	135.54
ΔT_{lm} , condenser (°C)	13.40
UA condenser (W/K)	10,117
Q, regenerator (kW)	19.00
ΔT_{lm} , regenerator (°C)	7.70
UA regenerator (W/K)	2467
Turbine power (kW)	19.74
N-heptane pumping power (kW)	0.071
Thermal oil pumping power (kW)	0.047
Cooling water pumping power (kW)	0.227
Net power (kW)	19.40
Efficiency	0.125

The preliminary design of the turbine was carried out assuming ideal stages. The velocity triangles were calculated by the free vortex theory and by a qualitative optimization of the specific speed and of the specific diameter. The sizing shows that a single stage axial turbine would be appropriate. The main turbine features can be summarized as follows: full admission, 30,000 rpm, and a reaction degree of 0.3 at mean diameter; the turbine would have a mean radius of about 7 cm and a rotor blade mean height of 1.6 cm, with a tip peripheral blade velocity of 243 m/s and an absolute Mach number at nozzle discharge of about 2.0.

Finally, for the sake of completeness, temperature profiles inside the evaporator and the condenser are respectively reported in Figs. 6 and 7, for different evaporation pressures.

5. Economic analysis

In this section we evaluate the potential economic benefit in terms of cost of the produced electricity, for the 19.4 kW ORC engine designed with plastic heat exchangers. The comparison is

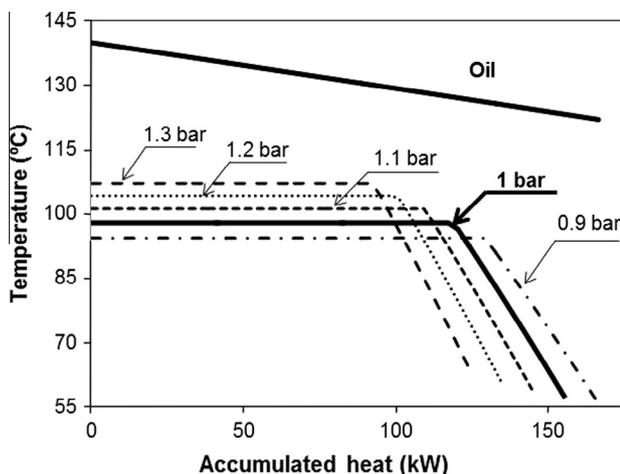


Fig. 6. Temperature profile inside the evaporator for different evaporation pressures (in bold the optimized pressure).

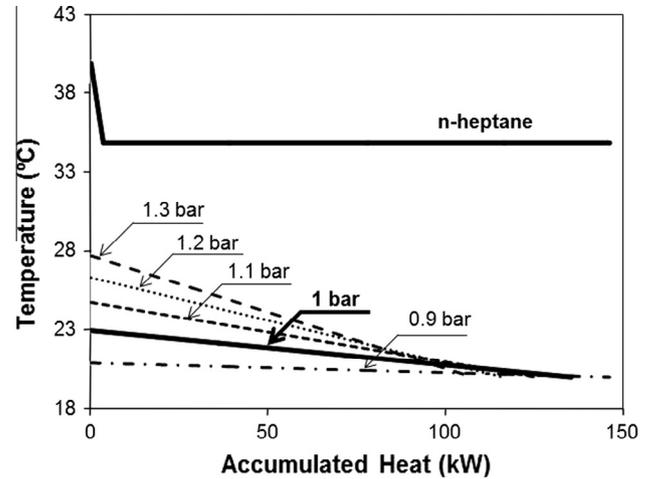


Fig. 7. Temperature profile inside the condenser for the different evaporation pressures (in bold the optimized pressure).

carried out with the conventional solution of heat exchangers made of carbon steel as well as in cases of enhanced materials, namely stainless steel 304, duplex steel and titanium, to be adopted in presence of corrosive heat source media.

The analysis is performed assuming a reference specific overnight cost of 4350 \$/kW, obtained according to data interpolated from [43], for small ORC engine equipped with carbon steel heat exchangers: for a conventional 19.4 kW engine, an overall investment of 84,400 \$ is then estimated. It is also assumed that in this figure a 20% BOP costs are included.

By considering separately the heat exchanger costs, it is possible to determine the investment cost of the different ORC plants where the classical carbon steel heat exchangers are replaced by heat exchangers made of the other considered materials [44]. The costs of the metallic heat exchangers are obtained by evaluating in each case the optimized configuration, according to the cycle conditions, keeping in all cases the same value of UA, with Aspen Exchanger Design and Rating [45]. Table 3 shows the optimized design data for the various considered materials, as well as the available information for the plastic heat exchangers used. In Table 4 the global heat transfer data and costs are reported. It can be noted that, despite the differences in the thermophysical properties of plastic and metals, similar values of U are obtained as a consequence of the geometry optimization process carried out by Aspen Heat Exchanger.

The regenerator deserves a separate discussion: as already outlined in the section of calculations, a shell and tube heat exchanger would not be convenient, because the heat transfer coefficient on the gas side is much lower than that on the liquid side. This was confirmed by a quick calculation that has been conducted using typical values of convective coefficients for the organic fluid, 1900 W/m² K for liquid state and 125 W/m² K for vapor, obtained from literature [46]. These values lead to a global heat transfer coefficient of 84 W/m² K for a PVDF shell and tube heat exchanger which implies an area of the regenerator of 29.4 m² which almost doubles the evaporator surface. Therefore, due to lack of published data for a suitable innovative compact polymeric heat exchanger, it is assumed in all cases to adopt a conventional carbon steel regenerator which has therefore no impact on the economic comparison.

Summing up, five possible cases have been studied, that we shall call in the following Plastic, Carbon steel, Stainless steel 304, Duplex and Titanium. The first one is the proposed innovative ORC module, with the condenser made of PE and the evaporator of PVDF. The other cases are named with reference to the material selected for the evaporator, depending on the corrosiveness of

Table 3
Optimized heat exchangers design data.

	PE Cond	PVDF Evap	Carbon steel Cond	Carbon steel Evap	Stainless steel 304 Evap	Duplex Evap	Titanium Evap
Number of tubes	195	195	68	74	81	81	114
Tube ID (mm)	4.8	4.8	14.83	14.83	15.75	15.75	16.56
Tube OD (mm)	6	6	19.05	19.05	19.05	19.05	19.05
Shell ID (mm)	<180	<180	257.45	257.45	266.24	266.24	315.93
Shell OD (mm)	<180	<180	273.05	273.05	273.05	273.05	323.85
Tube length (mm)	5000	5000	3048	3657.6	3657.6	3657.6	1828.8
Number of baffles	N/A	N/A	0	7	7	7	2
Number of passes	1	1	1	1	1	1	2
Effective surface (m ²)	16.7	16.7	12.1	15.9	17.4	17.4	12
Global coefficient (W/m ² K)	605	250	836	263	240	240	348

Table 4
Heat transfer data and costs for the different heat exchangers.

		Plastic	Carbon steel	Stainless steel	Duplex	Titanium
Evaporator	<i>U</i> (W/m ² K)	250	263	240	240	348
	<i>A</i> (m ²)	16.7	15.9	17.4	17.4	12.0
	UA	4181	4181	4181	4181	4181
	Price (\$)	16,700	12,852	17,511	19,046	30,576
Condenser	<i>U</i> (W/m ² K)	605	836			
	<i>A</i> (m ²)	16.7	12.1			
	UA	10,117	10,117			
	Price (\$)	7500	11,761			

Table 5
Investment cost comparison among the different considered solutions.

	Plastic	Carbon steel	Stainless steel	Duplex	Titanium
Total plant investment (\$)	83,904	84,400	89,991	91,833	105,669
Incidence of the evaporator on the cost of the plant components (BOP not included) (%)	23.9	18.3	23.4	24.9	34.7
Incidence of the evaporator on the plant investment (BOP included) (%)	19.9	15.2	19.5	20.7	28.9
Overnight cost (\$/kW)	4325.0	4350.5	4638.7	4733.6	5446.8

the hypothetical heat source, with the condenser always made of carbon steel.

Table 5 reports the resulting investment cost for the various cases considered, based on the heat exchangers costs assumed in Table 4. It can be observed that the investment cost of the Plastic case is roughly equivalent to that of the Carbon steel case (83,904\$ versus 84,400\$), given that the cost advantage due to the PE condenser with respect to carbon steel is counterweighed by the higher cost of the PVDF evaporator. Nonetheless, the effect of reduction of the cost investment becomes evident when the plastic solution is compared to the cases of the enhanced materials, leading to a plant investment reduction up to about 20% when the comparison is made with the Titanium evaporator.

Once the overall investments are determined, the methodology used to compare the profitability of the solutions in terms of the resulting electricity cost, is based on the annual cash flows in accordance with the directives contained in [47]. This calculation scheme is widely used as a reference in economic evaluations of power plants. A brief description of this methodology is given in the following.

It is assumed that the investment needed to implement the system, namely the overnight cost of the technology, (values in the last line in Table 5) represents the overall plant investment thus neglecting allowances for funds used during the construction time. Also, for simplicity, any additional cost, like authorizations, start up, etc., is included in this cost.

The basic assumptions of the economic analysis are reported in Table 6 [48]. The analysis is carried out in correspondence of a

Table 6
Basic assumptions for the economic analysis.

Discount rate (%)	5–10
Inflation rate (%)	2
Overall income tax (%)	40
O&M costs (c\$/kW h)	2
Plant book life (years)	15

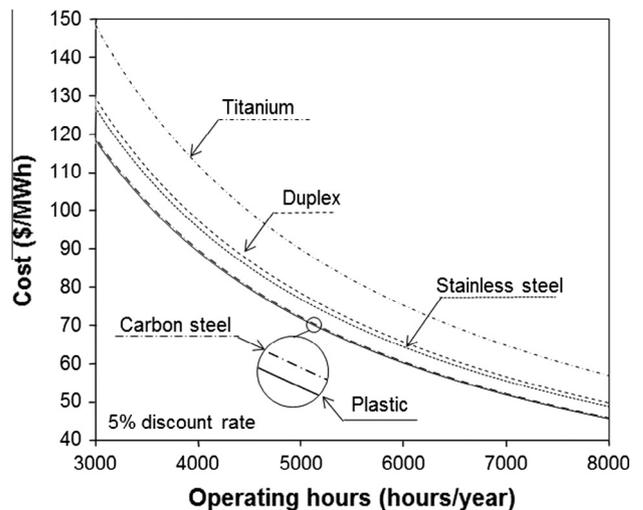


Fig. 8. Cost of electricity as function of plant operating hours in case of 5% discount rate.

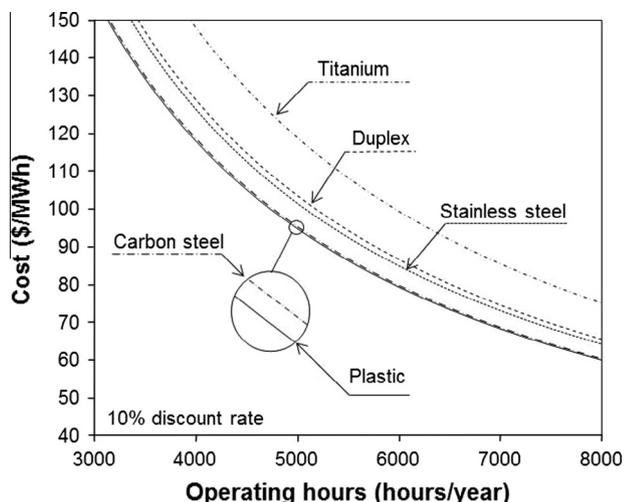


Fig. 9. Cost of electricity as function of plant operating hours in case of 10% discount rate.

plant book life of 15 years, as it is reasonable for such applications, and for two values of the discount rate, namely 5% and 10%.

Results of the economic analysis are reported in Figs. 8 and 9, showing the cost of electricity as function of the plant operating hours. As to be expected, the resulting values reflect the differences in the investment cost assumed for the various solutions, highlighting an appreciable advantage of the plastic case whenever the presence of corrosive heat source media requires the use of materials other than carbon steel. For instance, for a plant availability of 5000 h/year and discount rate of 5%, we obtain 71.8 \$/MW h (Plastic), 72.2 \$/MW h (Carbon steel), 76.8 \$/MW h (Stainless steel), 78.4 \$/MW h (Duplex steel) and 89.9 \$/MW h (Titanium) (Fig. 8), while with the same assumptions in case of a discount rate of 10% we obtain 94.8 \$/MW h, 95.4 \$/MW h, 101.5 \$/MW h, 103.6 \$/MW h and 118.9 \$/MW h, respectively (Fig. 9).

6. Conclusions

ORC plants are widely used for various applications: among others, geothermal application is of particular interest for this work. A conventional low size ORC for geothermal application is in fact characterized by a high specific cost, low temperatures and, usually, corrosive heat source media, so that the cost of heat exchangers is of paramount importance in the overall cost. This aspect could lead to the opportunity of introducing plastic heat exchangers instead of conventional ones made of carbon steel or other materials which resist corrosion. With the aim to study the introduction of plastic heat exchangers, a fluid selection process has been carried out, complying with temperature and pressure limits imposed by the plastic material resistance. N-heptane turned out as the most appropriate fluid for this case: the selection of a non common ORC working fluid is a remarkable aspect of plastic heat exchanger adoption.

Cycle simulations have afterwards been performed in the frame of an optimization procedure, highlighting that, when commercial plastic heat exchangers are adopted for the evaporator and condenser, and n-heptane is selected as working fluid, a regenerative cycle gives the best performance reaching a maximum net power of 19.4 kW.

Finally, the cost of electricity has been evaluated, calculating the overnight cost of an ORC with the evaporator made of PVDF, the condenser of PE and the regenerator of carbon steel. The cost

of electricity has been compared first to a reference case, with all the heat exchangers made of carbon steel and then to the case in which, because of corrosive heat source, enhanced materials are needed for the evaporator. Taking into account a plant book life of 15 years, and two different values of the discount rate, namely 5% and 10%, the proposed solution turned out to be of the same order of magnitude of the reference case, and competitive when the heat source is corrosive and special materials are required: the use of plastic heat exchangers in ORC cycles can therefore be considered nowadays as a promising solution to reduce costs in small size plants.

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