Binary ORC (Organic Rankine Cycles) power plants for the exploitation of medium–low temperature geothermal sources – Part B: Techno-economic optimization

Marco Astolfi*, Matteo C. Romano, Paola Bombarda, Ennio Macchi

Politecnico di Milano, Energy Department, Via Lambruschini 4, 20156 Milano, Italy

Received 17 May 2013
Received in revised form
14 October 2013
Accepted 20 November 2013
Available online 10 January 2014

1. Introduction

As discussed in Part A, ORC (Organic Rankine Cycles) have been experiencing a great commercial success in recent years and are receiving an increasing interest from the scientific community. Their application for the exploitation of heat sources characterized by low temperatures or small sizes, where important drawbacks limit the application of steam cycles, is subject of a number of papers recently published. Different working fluids and different cycle configurations have been assessed for applications on renewable energy sources (mainly biomass, solar and geothermal energy) and waste heat recovery, with the aim of selecting the optimal fluid and cycle parameters for each application.

Most of the works published in the literature deal with thermodynamic assessments, i.e. they aim at maximizing the plant efficiency or its power output. However, thermodynamic assessment alone cannot provide exhaustive indications on the optimal working fluid, mainly because of the different thermal and volumetric behavior of the fluids, which affect the performance, the size and the cost of the plant components. Therefore, a comprehensive ORC optimization process should include an economic analysis or, at least, an analysis of the size and the technical feasibility of the main components (heat exchangers and turbomachines).

Assuming that heat exchangers are responsible for most of the total plant cost, some authors only accounted for their cost in the economic analysis [1] or used the specific heat exchange area per kW_{el} generated as the function to be minimized [2]. At the authors' knowledge, only Quoilin et al., 2011 [3] report a comprehensive economic analysis to minimize the cost of the electricity from a 100 kW_{th}-scale waste heat recovery ORC, also considering the dependence of the expansion efficiency on the actual operating conditions of the scroll expander considered. The results of their economic optimization show how the optimal key design parameters can change when considering the cost of the components, with different effects for each fluid. For example, as shown in Table 1, they found different optimal fluids, increased optimal evaporation temperatures (+20–30 °C), a rather wide range of optimal minimum ΔT_{pp} in the evaporator (4.0–7.5 °C) and finally

* Corresponding author. Tel.: +39 (0)2 2399 3935.
E-mail address: marco.astolfi@mail.polimi.it (M. Astolfi).

Table 1
Review of the works in the literature on ORC techno-economic optimization.

Reference	Heat source	Types of cycles	Machines efficiencies	Fixed variables	Optimization variables	Optimization function	Component sizing	Optimal cycles
Hettiarachchi, Golubovic, al. (2007) [2]	Geothermal brine @ 70–90 °C	Sub-SA no-rec	$\eta_{is,turb} = 85\%$ $\eta_{wf,\ pump} = 75\%$ $\eta_{mec\text{-}el} = 96\%$ $\eta_{cw,pump} = 80\%$	$T_{cw} = 30$ °C $\Delta p_i =$ calc	T_{eva} T_{cond} u_{geo} u_{cw}	Specific heat exchange area: m^2/kW	Heat exchangers area	*NH_3*: $T_{eva} = 76.9$ °C $T_{cond} = 43.0$ °C $\eta_{cycle} = 8.9\%$ $\eta_{plant} = 8.0$ %[a] $\alpha = 0.34$ m^2/kW
Quoilin, Declaye et al. (2011) [3]	WHR: gas @ 180 °C with HTF.	Sub-SA no-rec	$\eta_{scroll\text{-}exp}$: calc $\eta_{wf,\ pump} = 60\%$ $\eta_{HTF,pump} = 60\%$ $\eta_{mec\text{-}el} = 70\%$	$\Delta T_{pp,PHE} = 10$ °C $\Delta T_{sh} = 5$ °C $\Delta T_{pp,cond} = 10$ °C $\Delta T_{sc,cond} = 5$ °C $T_{cw} = 15$ °C $m_{cw} = 0.5$ kg/s $\Delta p_{eva} = 10$ kPa $\Delta p_{cond} = 20$ kPa	T_{eva}	Net power	Scroll expander (given geometry), plate PHE area, condenser area	*R245fa*: $T_{eva} = 113.5$ °C $\eta_{cycle} = 7.78\%$ $\eta_{plant} = 5.13\%$ *R123*: $T_{eva} = 111.8$ °C $\eta_{cycle} = 8.41\%$ $\eta_{plant} = 5.00\%$
				$\Delta T_{sh} = 5$ °C $\Delta T_{sc,cond} = 5$ °C $T_{cw} = 15$ °C $m_{cw} = 0.5$ kg/s	T_{eva} $\Delta T_{pp,PHE}$ $\Delta T_{pp,cond}$ Δp_{eva} Δp_{cond}	Specific cost: €/kW		*n-butane*: $T_{eva} = 133.2$ °C $\Delta T_{pp,PHE} = 7.5$ °C $\eta_{plant} = 4.47\%$ Cs = 2136 €/kW *n-pentane*: $T_{eva} = 139.9$ °C $\Delta T_{pp,PHE} = 4.0$ °C $\eta_{plant} = 3.88\%$ Cs = 2505 €/kW
Shengjun et al. (2011) [1]	Geothermal brine @ 90 °C	Sub-SA/Sup no-rec	$\eta_{is,turb} = 80\%$ $\eta_{wf,pump} = 75\%$ $\eta_{mec\text{-}el} = 96\%$	$\Delta T_{pp,PHE} = 5$ °C $\Delta T_{pp,cond} = 5$ °C $T_{cw} = 20$ °C $\Delta p_i = 10$ kPa	Sub: T_{eva}, p_{cond} Sup: $T_{in,turb}$ p_{max}, p_{cond}	Specific heat exchange area (α): m^2/kW	Heat exchangers area	*R152a*: $T_{eva} = 74$ °C $T_{cond} = 27.9$ °C $\alpha = 1.64$ m^2/kW
						COE (only heat exchangers cost, function of operating pressure, considered)		*R152a*: $T_{eva} = 60$ °C $T_{cond} = 27.9$ °C COE = 53 €/MWh

[a] Heat available from WHR and geothermal brine calculated by considered cooling of the heat source to ambient temperature.

significantly lower optimal plant efficiencies (from 5.0–5.1% to 3.9–4.5%) with respect to what they obtained in the thermodynamic analysis only.

After the comprehensive thermodynamic assessment performed in Part A, the aim of this part of the work is to present the techno-economic optimization of ORCs featuring different cycle configurations (subcritical/supercritical, saturated/superheated, regenerative/non regenerative) and a variety of working fluids, for the exploitation of low-medium enthalpy geothermal fields (120–180 °C) in the 2–15 MW_{el} power output range.

In order to calculate the cost of the equipment, the size of the heat exchangers and the turbine are estimated. In particular, as far as the turbine is concerned, the number of stages, the optimal rotational speed and the mean diameter were calculated in each case to estimate a realistic cost and efficiency, which both depend on the turbine design. The assumptions and results of the economic analysis are presented in the paper. The cost of the heat source, which may be relevant in geothermal applications due to the high drilling cost, has been included and a sensitivity analysis on its cost has been performed.

2. Model description

In order to perform the economic analysis, cost correlations for each plant component are integrated in the code for the calculation of the thermodynamic performance. The code, described more in detail in Part A, is implemented in Matlab® [4] and is able to perform both thermodynamic and economic optimization of binary systems based on ORC technology. Single pressure level cycles in saturated/superheated, regenerative/non-regenerative, subcritical/supercritical configurations are considered. The developed code is integrated with the Refprop® database [5] that uses accurate equations of state to provide the thermodynamic properties of a large variety of fluids. The list of the 54 pure fluids selected in this study is reported in Table 2 divided by a chemical classification.

Besides thermodynamic considerations and economic results that are treated in this study, many other aspects have to be considered in fluid selection. In particular thermal stability, safety issues such as toxicity and flammability and environmental impact (i.e. GWP (Global warming potential index) and ODP (Ozone

Table 2
Investigated fluids.

12	Alkanes	Propane, isobutane, butane, neopentane, isopentane, pentane, isohexane, hexane, heptane, octane, nonane, decane
16	Other hydrocarbons	Cyclopropane, cyclopentane, cyclohexane, methylcyclohexane, propylcyclohexane, isobutene, 1-butene, trans-butene, cis-butene, benzene, propyne, methanol, ethanol, toluene, acetone, dimethylether
13	HFC	R125, R143a, R32, R1234yf, R134a, R227ea, R161, R1234ze, R152a, R236fa, R236ea, R245fa, R365mfc
3	FC	R218, perfluorobutane (C_4F_{10}), RC318
8	Siloxanes	MM, Mdm, Md2m, Md3m, Md4m, D4, D5, D6
2	Other non-organic fluids	Ammonia, water

depletion potential index)) should be carefully evaluated in order to prevent damage to plant components, to limit operation and maintenance costs, and to avoid hazards for the employees and for the natural environment. For these reasons, hydrochlorocarbons and hydrochlorofluorocarbons, even if they are available in the Refprop® database, are not considered in this study, being banned in Europe due to their high ODP index.

Fig. 1 presents the basic layout of a regenerative, subcritical superheated cycle, useful to define the notation which will be used in following discussions. This cycle is composed by a series of heat exchangers (economizer, evaporator, superheater), a turbine, a regenerator, a condenser and a pump. The regenerator is used to preheat the condensate at the pump outlet by recovering part of the heat released at the turbine outlet during fluid desuperheating. For non-regenerative cycles, points 10 and 3 simply collapse in 9 and 2 respectively, while in saturated cycles point 7 coincides to 6 and in supercritical configuration points 4, 5 and 6 merge to 7.

The plant performance can be evaluated once defined the value of all the model variables: some of these, labeled as fixed variables, are assumed prior to calculation and kept constant in the optimization procedure, independently on fluid and cycle configuration. The others, labeled as design variables, are set up by the optimization routine. All the variables and constrains adopted are reported in Table 3, showing the differences between the thermodynamic and the techno-economic assessments. In particular, the main differences arising in the techno-economic optimization are: (i) the presence of a greater number of optimization variables in order to optimize heat exchangers surface and (ii) the adoption of a model for the prediction of turbine efficiency. In particular, while the turbine isentropic efficiency was assumed constant throughout the thermodynamic optimization in Part A, in the techno-economic analysis the turbine efficiency is estimated on the basis of the effective operating conditions as discussed further on. In addition, a gearbox with a fixed generator efficiency is adopted when decoupling the turbine and the generator rotational speed improves the overall efficiency.

Thermodynamic performance index considered in this work are here resumed, described by the following equations:

$$\eta_{\text{cycle}} = \frac{W_{\text{net}}}{Q_{\text{in}}} \tag{1}$$

$$\eta_{\text{rec}} = \frac{Q_{\text{in}}}{Q_{\text{in,max}}} \tag{2}$$

$$\eta_{\text{plant}} = \eta_{\text{cycle}}\eta_{\text{rec}} = \frac{W_{\text{net}}}{Q_{\text{in,max}}} \tag{3}$$

$$\eta_{\text{II}} = \frac{\eta_{\text{plant}}}{\eta_{\text{lor}}} \tag{4}$$

$$\eta_{\text{lor}} = 1 - \frac{T_{\text{amb}}}{\ln\left(\frac{T_{\text{in,geo}} - T_{\text{lim,geo}}}{T_{\text{in,geo}}/T_{\text{lim,geo}}}\right)} \tag{5}$$

where $T_{\text{lim,geo}}$ is the minimum reinjection temperature for the geothermal brine in order to avoid salt precipitation on heat exchangers surfaces and $Q_{\text{in,max}}$ is the maximum thermal power that can be recovered by cooling the geothermal brine down to $T_{\text{lim,geo}}$ (or ambient temperature, when no reinjection temperature limit is considered). Q_{in} is the thermal power released by the geothermal brine when cooled in the ORC heat exchangers and hence the thermal power received by the ORC divided by the thermal efficiency of the heat exchangers. Lorentz efficiency refers to an ideal trapezoidal cycle if a limit in reinjection temperature is considered or to a trilateral cycle otherwise [6]. Ambient temperature is kept constant and equal to 15 °C.

In our study heat capacities of ambient air and geothermal brine are kept constant neglecting the effects related to temperature, incondensable gases and dissolved salts in the geothermal water, which is hence considered pure and always at liquid state.

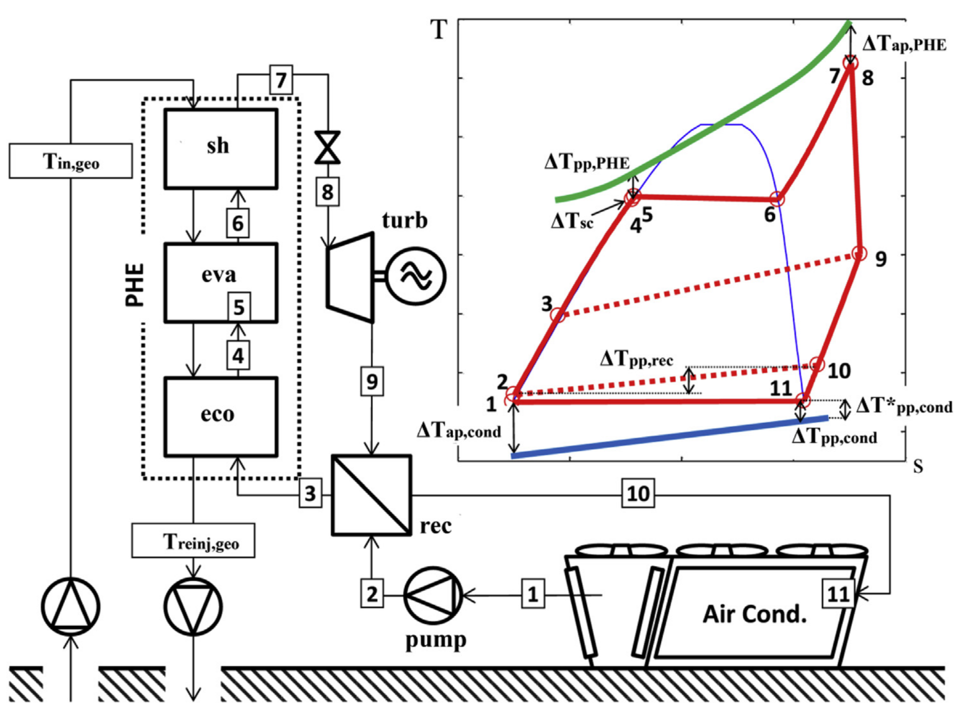

Fig. 1. *T–s* diagram, plant layout and notation adopted to define thermodynamic points.

Table 3
Assumed fixed and variable parameters adopted for the thermodynamic and the techno-economic analyses.

Objective function	Thermodynamic analysis	Techno-economic analysis
	η_{II}	$C_{TOTspec}$
Design variables		
$p_{in,turb}$	Optimized	Optimized
$\Delta T_{ap,PHE}$	Optimized	Optimized
$\Delta T_{pp,PHE}$	3 °C	Optimized
$\Delta T_{pp,rec}$	5 °C	Optimized
$\Delta T_{ap,cond}$	15 °C	Optimized
$\Delta T^{*}_{pp,cond}$	0.5 °C	Optimized
Temperature and pressure drops		
ΔT_{cond}	0.3 °C	
ΔT_{eva}	1 °C	
Δp_{des}	1%	
Δp_{SHE}	5%	
Δp_{sh}	2%	
Δp_{val}	1%	
$\Delta p_{rec,HS}$	2%	
Δp_{eco}	50 kPa	
$\Delta p_{rec,CS}$	50 kPa	
Heat losses from heat exchangers		
Q_{loss}	1%	
Other assumptions		
$\eta_{is,turb}$	85%	Calculated
$\eta_{wf,\ pump}$	70%	
$\eta_{mec\text{-}ele,\ pump}$	95%	
$\eta_{mec\text{-}ele,\ turb}$	95%	
$\eta_{gearbox}$	97%	
ΔT_{sc}	*Max*(1 °C, 0.05(T_5-T_3))	
Constrain		
No droplets along expansion		

From a thermodynamic point of view net power production, plant efficiency or second law efficiency are adopted as term of comparison among the different solutions, while specific cost (€/kW) is selected as objective function in techno-economic optimization.

3. Techno-economic optimization

Economic optimization entails a greater number of optimization variables with respect to thermodynamic optimization and the necessity to set up a cost correlation for each plant component. Moreover specific considerations on turbine design and efficiency prediction are introduced at this stage, so that all the six independent variables are optimized and turbine efficiency is calculated rather than assumed (see Table 3). The specific plant investment cost (€/kW) represents the objective function to be minimized, since it is representative of the cost of the electricity,[1] assuming negligible variable costs among the considered cases.

3.1. Techno-economic optimization routine

Difficulties in finding the global minimum arise because of the strong non-linearity of the problem and the discontinuities of the objective function. In addition, simultaneous optimization of the six variables entails poor solution accuracy because of the greater influence of the outlet geothermal brine temperature in plant economics with respect to the other parameters.

Therefore, a more efficient two-level optimization procedure has been implemented, tackling the outer level problem with the patter search method of Lewis and Torczon – 2002 [7] as represented in Fig. 2. At the outer level, four optimization variables are considered namely: $p_{in,turb}$, $\Delta T_{ap,PHE}$, $\Delta T_{ap,cond}$ and $\Delta T_{pp,rec}$, while at an inner level the optimization of $T_{reinj,geo}$ and $\Delta T^{*}_{pp,cond}$ is performed. The first step of the optimization procedure is setting lower (LB) and upper bounds (UB) for each optimization variable. These boundaries are set either to obtain realistic results (i.e. feasible temperature differences, dry expansion, etc.) or to limit the search domain. This code structure entails a quite large computational cost but the results obtained are more reliable compared to a simultaneous optimization of six parameters. The main advantage of adopting this methodology is that once the outer optimization level variables are set, it is possible to completely define all the thermodynamic points of the considered cycle. In addition, it is possible to univocally define the lower bound for $T_{reinj,geo}$ and improve the accuracy of the solution.

Lower and upper bounds which are assumed for each variable are here reported:

- $p_{in,turb}$: for subcritical cycles the lower limit is set equal to twice the minimum condensing pressure (which depends on the minimum value of $\Delta T_{ap,cond}$), while the upper limit is chosen equal to the minimum value between the critical pressure and the saturation pressure at the maximum temperature (which depends on the inlet brine temperature and the minimum $\Delta T_{ap,PHE}$). For saturated cycles, the pressure corresponding to the vertical slope of the overhanging saturation curve is also considered as upper limit, to avoid the risk of expansion within the saturation curve. For supercritical cycle, a minimum pressure equal to p_{crit} and an upper bound equal to 100 bar are selected.
- $\Delta T_{ap,PHE}$: lower bound is set to 0.5 °C above the minimum $\Delta T_{pp,PHE}$. In subcritical cycles, the maximum value is the one corresponding to the saturated cycle with the lower allowable $p_{in,\ turb}$. In supercritical cycles, the upper bound is set equal to the difference between the brine inlet temperature and the critical temperature.
- $\Delta T_{ap,cond}$: lower and upper bounds are set equal to 5 °C and 40 °C respectively. If one of these limits is reached at the end of the optimization, the calculation is repeated with a wider range.
- $\Delta T_{pp,rec:}$ lower and upper bounds are set equal to 0.5 °C and 50 °C respectively. The higher limit is chosen by considering the possibility to converge to a solution without the adoption of regenerator.
- $T_{reinj,geo:}$ lower bound is found for a $\Delta T_{pp,PHE}$ equal to 0.1 °C, while the upper bound is set equal to 5 °C less than the brine inlet temperature.
- $\Delta T^{*}_{pp,cond:}$ lower bound and upper bound are set equal to 0.1 °C and 3 °C respectively.

3.2. Considerations on turbine design and efficiency

In geothermal binary ORC power plants different families of turbines can be utilized: some manufactures offer single-stage centripetal turbines [8,9], others are oriented towards axial flow turbines [10,11], which usually adopt more than one stage. A third option is a radial multi-stage outflow turbine [12,13], which permits to achieve higher efficiency when large volumetric flow ratios have to be handled. In this study only axial turbines will be considered, given the availability of accurate correlations for stage efficiency prediction. Turbine design and efficiency are evaluated through the following simplified procedure, which is based on the

[1] The solution yielding minimum COE (cost of electricity) does not necessarily corresponds to the best economic choice, which depends on the specific valorization of the produced electricity. In other words, it could be attractive to produce more electricity, even at higher COEs.

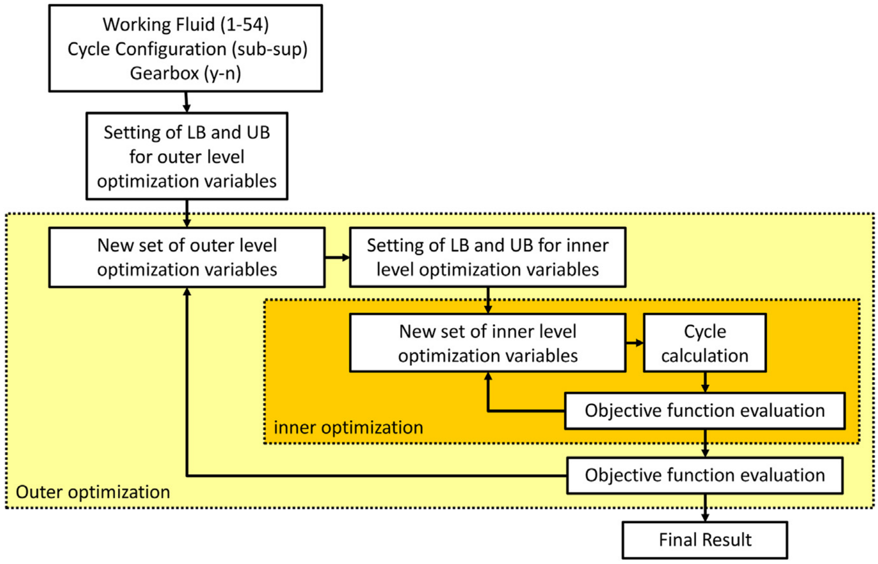

Fig. 2. Code structure in techno-economic optimization.

results presented in Ref. [14]. Starting from the inlet conditions (8) and exhaust pressure (9), the isentropic enthalpy drop and volume flow rates ratio along the expansion are computed. The number of turbine stages is estimated by setting two limits, namely: (i) the stage maximum volume flow ratio and (ii) the maximum allowable stage enthalpy drop.

The first limit is set in order to avoid high Mach numbers and large blade height variations across the rotor blade. This last aspect entails high flaring angles and a consequent losses increase. The maximum V_{ratio} admissible for each stage is assumed equal to 4 in order to limit the above mentioned penalties. If the total V_{ratio} is larger than this limit, the expansion is divided into the minimum number of stages with equal V_{ratio}.

V_{ratio} is strongly dependent on the thermal level of the geothermal brine and on the selected working fluid. Small values can usually be found in low temperature cycles using complex fluids, with values of $T_{\text{crit}}/T_{\text{in,geo}}$ lower than unit. On the contrary, fluids with high critical temperatures have low condensing pressures and volume ratios as high as 200–300. The second limit is set to avoid high mechanical stresses: while fluids with high molecular weight entail relatively small turbine enthalpy drops and give the possibility to design a turbine with moderate peripheral speeds and reduced mechanical stresses, low molecular weight fluids like water or ammonia, show high enthalpy drops along the expansion. For this reason, a limitation related to the maximum enthalpy drop exploitable for each stage is introduced, equal to 65 kJ/kg. This value is derived by assuming a 50% reaction degree stages with a load coefficient (k_{is}) equal to 2 and a mean peripheral speed of 255 m/s according to the following equation:

$$k_{\text{is}} = \frac{2\Delta h_{\text{is}}}{u_{\text{m}}^2} \quad (6)$$

Thus if the second limit is exceeded, the expansion is divided into the minimum number of stages with the same enthalpy drop.

Once the number of stages is determined, it is possible to compute the SP (size parameter) (Eq. (7)) and the specific speed Ns (Eq. (8)) of the first and the last stage. According to the similarity rules, SP is proportional to stage diameter and it is used in the turbine cost correlation. Ns and V_{ratio} allow estimating the turbine efficiency. Using the results obtained in Macchi, Perdichizzi – 1981 [14] for the optimization of different turbine stages in a wide range of Ns and V_{ratio}, the isentropic efficiency of the first and the last stage are computed by fitting the curves in Fig. 3 and afterward with a linear interpolation on V_{ratio}.

$$\text{SP} = \frac{\sqrt{\dot{V}}}{\Delta h_{\text{is,Stage}}^{1/4}} \quad (7)$$

$$\text{Ns} = \frac{\text{RPM}}{60}\frac{\sqrt{\dot{V}}}{\Delta h_{\text{is,Stage}}^{3/4}} \quad (8)$$

Stage efficiency is defined as the ratio between the actual enthalpy drop and the available isentropic enthalpy drop while V_{ratio} is the ratio between the specific volume in outlet and inlet conditions for an isentropic expansion.

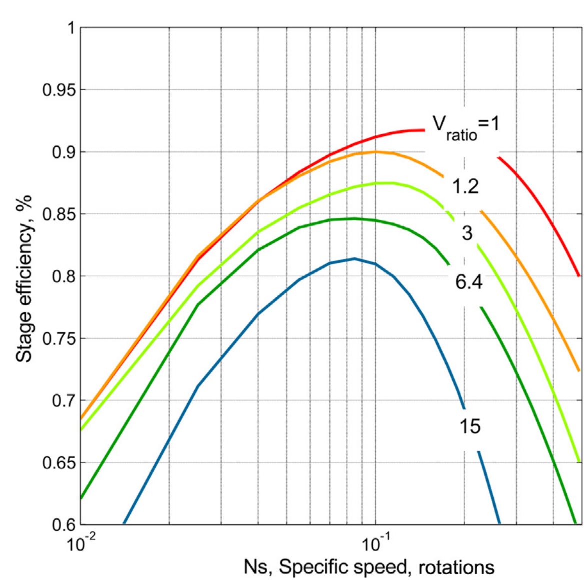

Fig. 3. Parametric curves adopted for turbine stage efficiency prediction [14].

Fluid expansion has been calculated by considering a turbine isentropic efficiency calculated as the average of the first and the last stage efficiencies. While a complete stage-by-stage approach would lead to higher accuracies, this methodology is considered a good compromise between computational time, code stability and accuracy, also considering the limited number of stages which typically characterizes ORC turbines and the limited variation in stage enthalpy drop. Finally, the convenience of adopting a gearbox is also checked. This option could be really profitable for turbines with Ns at the last stage far below the optimum value ($\sim$0.1) and hence low isentropic efficiencies when directly coupled with the generator. This usually occurs with fluids with relatively low molecular complexity and large Δh_{is}, which benefit from high rotational speeds, or for high complexity fluids expanding to very low pressures where lower speed of revolution allows increasing the turbine efficiency. For the first class a gearbox is adopted, the rotational speed is computed by imposing Ns of the last stage equal to the optimal value, and considering agearbox efficiency of 97%, while for the others a generator with a greater number of poles is assumed.

3.3. Cost correlations

Cost correlations have been defined for each component of the ORC plant, based on the exponential scaling law method. Due to the wide variety of operating conditions, cycle configurations and fluids considered in this work, correlations with high generality needs to be used, capable of taking into account the different design requirements of the components, resulting from each simulated case. However, such correlations are often lacking in the open literature. This is particularly evident for the turbine, which represents an important share of the total investment cost and whose cost cannot be estimated simply as function of its power output (as in the correlations typically proposed in the literature [15]). As a matter of fact, its design (i.e. number of stages and their size) and hence its cost can vary significantly for cases with the same power output but different enthalpy drops, volume ratio and volume flow rate.

Parallel considerations can be made for the condenser, representing another important contribution to the total investment cost, whose heat exchange surface greatly depends on the heat transfer coefficient on the air side. Therefore, dependence of both heat transfer coefficient and air side pressure drops (influencing the operating cost) with the air velocity should be correctly predicted when calculating its cost and optimizing its operating conditions.

For these reasons, based on the experience gained by the authors in the cooperation with manufacturers of ORC plants and turbines [11–13] and heat exchangers [16], new correlations are proposed in this paper for these key components. Also for more conventional components (pump and generator), data from manufacturers has been used to define the reference cost.

The complete list of the cost correlations used is here summarized:

- PHE (Primary Heat Exchanger) and regenerator costs are calculated according to Eqs. (9)–(10) and Table 4. The effect of the maximum cycle pressure is taken into account by means of the correction factor proposed in Ref. [17] and considering that the high pressure organic fluid flows on the shell side in the PHE[2] and on the tube side in the regenerator.

$$C_{\mathrm{PHE,rec}} = C_0\left(\frac{\mathrm{UA}}{\mathrm{UA}_0}\right)^{0.9} a \tag{9}$$

$$a = 10^{\left(a_1 + a_2\,\log(p) + a_3\,\log^2(p)\right)} \tag{10}$$

where C_0 is referred to an heat exchanger with a maximum pressure of 5 bar.

- The cost of the air condenser (including electric fans) is calculated according to Eq. (11) with $C_0 = 530$ k€ and $A_0 = 3563$ m^2. The heat exchange area is calculated by estimating the overall heat transfer coefficient with the correlation shown in Fig. 4:

$$C_{\mathrm{cond}} = C_0\left(\frac{A}{A_0}\right)^{0.9} \tag{11}$$

- The cost of the turbine is calculated as a function of the number of stages n and the last stage size parameter SP and the power as scaling factors, according to Eq. (12):

$$C_{\mathrm{turb}} = C_0\left(\frac{n}{n_0}\right)^{0.5}\left(\frac{\mathrm{SP}}{\mathrm{SP}_0}\right)^{1.1} \tag{12}$$

with $C_0 = 1230$ k€, $n_0 = 2$, $\mathrm{SP}_0 = 0.18$ m

- The cost of the electric generator is calculated according to Eq. (13):

$$C_{\mathrm{gen}} = C_0 \cdot \left.\frac{W_{\mathrm{el}}}{W_{\mathrm{el},0}}\right)^{0.67} \tag{13}$$

where $W_{\mathrm{el},0} = 5000$ kW and $C_0 = 200$ k€. In case a gearbox is present, its cost is considered equal to 40% of the electric generator cost.

- The cost of the pump is calculated according to Eq. (14):

$$C_{\mathrm{pump}} = C_0 \cdot \left.\frac{W_{\mathrm{el}}}{W_{\mathrm{el},0}}\right)^{0.67} \tag{14}$$

where $W_{\mathrm{el},0} = 200$ kW and $C_0 = 14$ k€

- The balance of plant costs are estimated as 40% of the total ORC components cost.

Due to the lack of information on the chemical aggressiveness of all the fluids considered in our analysis, no correction factor has been included in the economic model to take into account extra cost for the use of advanced materials. However, for the optimal fluids found in this work, the possibility of using carbon steel materials has been verified.

Specific cost for some of the investigated working fluids is really hard to obtain, especially for perfluoro alkanes and siloxanes. Anyway, experience on real plants operation indicates that the fluid inventory contribution on the total plant cost is at least one order of magnitude lower than the equipment cost.

Finally, the cost of the geothermal wells is included in the optimization procedure by adding a constant value of 12 M€. This value results by considering two production and two reinjection wells in order to supply a brine flow of 200 kg/s and could be

[2] As required by the necessity of mechanically cleaning the tubes internal surface to limit fouling effects of geothermal brine.

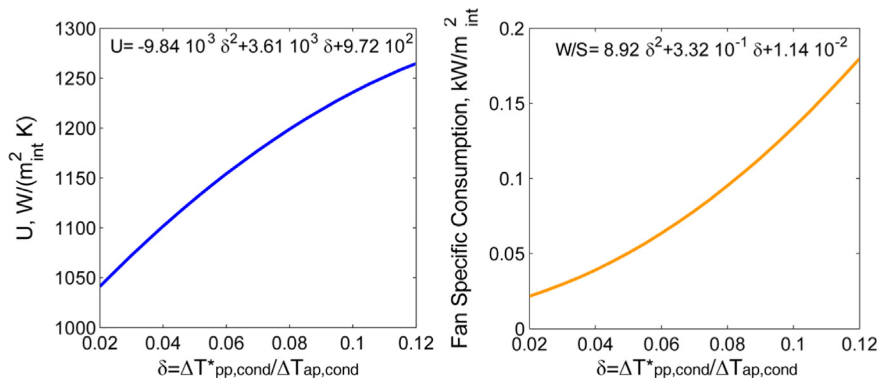

Fig. 4. Correlation for U and W/S for the air condenser unit with ambient temperature between 15 and 25 °C [16]. The range of fan velocity investigated corresponds to frontal air velocities between 1.0 and 2.5 m/s. Organic fluid at inlet assumed at saturated vapor conditions. The effect of de-superheating in the range experienced in this work has been tested on some fluids considered in this work, leading to maximum errors of the thermal power of 2% with respect to a condition of saturated vapor condensation.

representative of an area with a favorable geothermal gradient of 50 °C/km [18]. Due to the large uncertainty and the strong site dependency, a sensitivity analysis has been performed for this assumption.

A preliminary economic optimization procedure was carried out with four fluids with very different thermodynamic properties in order to validate the correlations adopted in a wide range of conditions and to verify the influence of the various parameters which mainly affect the final optimal solution. A geothermal brine with a temperature at PHE inlet equal to 150 °C and a reinjection temperature equal to 70 °C is taken as reference case.

Fluids here considered are R134a, ammonia, water and octane. R134a, despite its GWP, is commonly used as refrigerant fluid and as working fluid for supercritical regenerative cycles [19]. Ammonia, thanks to its thermodynamic properties, gives the possibility to obtain a compact design of the components and may be profitable from an economic point of view. However, it is rarely used in power plants due to its toxicity.

Water is the most common fluid used for power generation, but it is not competitive for medium-low temperature heat sources and small sizes, mostly because of the uneconomical turbine design. For this case, since a wet expansion cannot be avoided with reasonable cycle parameters, the constraint which keeps the expansion in superheated vapor region is suppressed. The selected configuration for ammonia and water consists of a subcritical superheated cycle with the adoption of a gearbox.

Finally, octane is considered in order to investigate the difficulties which can occur when using complex fluids with a high critical temperature. A subcritical superheated cycle is adopted, allowing the solver to converge to a saturated configuration acting on $\Delta T_{ap,PHE}$ variable.

In Table 5, the calculated efficiencies and values of the optimized parameters resulting from the thermodynamic and from the economic optimization are reported.

In Fig. 5 a comparison between thermodynamic and economic optimizations is presented for the four investigated fluids. Wide stacked bars are referred to absolute component cost repartition while tight bars show the change in total plant specific cost and power block specific cost.

Final results for R134a show a lower $\Delta T_{ap,cond}$ (i.e. a lower T_{cond}) than the value adopted in the thermodynamic optimization with a consequent increase of both air condenser area and fans consumption. On the hot side, optimal maximum pressure decreases while $\Delta T_{ap,PHE}$ increases entailing a higher value of $\Delta T_{ml,PHE}$, which allows reducing the cost of the primary heat exchanger, accepting lower cycle efficiency. Thanks to the optimization of $\Delta T_{pp,rec}$, geothermal brine is completely exploited in both cases. The main effect of economic optimization is a reduction of absolute total plant cost but also a lower value of both total plant (−7%) and power block (−16%) specific costs, even if net power production decreases.

The optimization procedure applied to ammonia entails a reduction of the condensing temperature (almost identical to the one found for R134a) with an almost constant evaporating pressure and the removal of the regenerator in order to achieve a higher recovery efficiency. Superheating temperature is higher and, together with a lower T_{cond}, it leads to a greater cycle efficiency. Power block cost increases because of the lower ΔT_{ml} in both condenser and PHE but power block specific cost is slightly reduced, thanks to the higher power output. It is also important to note that the absolute total plant cost is lower for ammonia than for R134a because subcritical cycles always lead to higher ΔT_{ml} and lower PHE costs. In addition also the ammonia turbine is cheaper than the R134a one, because of its small SP, even if it is designed with a higher number of stages (4 vs. 2). The total plant specific cost finally decreases according to the higher plant efficiency. In both cases component cost repartitions show that primary heat exchanger, condenser and turbine cover almost the 75% of total power block cost. The final specific cost is lower for R134a, thanks to its higher efficiency.

As expected the results obtained for water and octane are very different. For these fluids, the turbine is largely the most expensive component and economic optimization leads to an increase in both evaporation and condensing pressure, with the main goal of reducing the turbine size. Cycle efficiency slightly increases but recovery rate drops because of the higher evaporation temperature, entailing a lower plant efficiency and power production. The overall power block and total plant specific costs decrease, but reaching values which are more than twice than the R134a and ammonia ones.

Octane is selected as representative of high complexity fluids. It features an overhanging saturation line and the optimization numerically converges to a saturated cycle, which is the most

Table 4
Assumption for PHE and recuperator costs.

	PHE	Rec
C_0, k€	1500	260
UA_0, kW/K	4000	650
a_1	0.03881	−0.00164
a_2	−0.11272	−0.00627
a_3	0.08183	0.0123

Table 5
Comparison between thermodynamic and economic optimization results for four test fluids. ΔT_{SH} for supercritical cycles (i.e. R134a) is defined as the difference between maximum fluid temperature (point 7) and critical temperature. The maximum geothermal brine temperature is 150 °C while the m_{geo} is equal to 200 kg/s

		R134a		Ammonia		Water		Octane	
		Th	Eco	Th	Eco	Th	Eco	Th	Eco
Cycle configuration		Sup, rec		Sub, SH, rec		Sub, SH no-rec		Sub, SA rec	
$T_{crit}/T_{in,geo}$		0.884		0.958		1.529		1.345	
η_{cycle}	%	12.43	12.36	10.64	11.47	10.86	11.76	11.74	12.58
η_{rec}	%	100	100	92.74	97.03	83.29	57.78	84.71	71.46
η_{plant}	%	12.43	12.36	9.86	11.13	9.05	6.79	9.94	8.99
η_{II}	%	50.69	50.39	40.23	45.39	36.90	27.7	40.55	36.66
W_{gross}	MW	10.51	10.21	7.364	8.367	6.343	4.600	6.973	6.162
W_{pump}	MW	1.870	1.467	0.444	0.454	0.005	0.005	0.039	0.037
W_{fan}	MW	0.4	0.55	0.38	0.53	0.34	0.09	0.34	0.17
$C_{TOT,\ Specific}$	€/kW	2697	2510	2869	2673	8259	7252	7430	6828
$C_{PB,\ Specific}$	€/kW	1241	1046	1034	1047	6258	4588	5610	4815
$p_{in,turb}$	bar	52.1	44.7	46.8	46.4	0.57	1.25	0.38	0.58
$\Delta T_{ap,PHE}$	°C	9.93	13.49	5.6	2.0	4.5	2.00	54.35	41.75
$\Delta T_{pp,PHE}$	°C	3	6.44	3	2.0	3	0.73	3	0.68
$\Delta T_{pp,cond}$	°C	0.5	0.428	0.5	0.405	0.5	0.62	0.5	0.42
$\Delta T_{ap,cond}$	°C	15	11.66	15	11.43	15	30.57	15	20.57
$\Delta T_{pp,rec}$	°C	5	15.97	5	9.588	–	–	5	12.88
ΔT_{sh}	°C	39.1	35.51	57.77	61.73	60.32	41.36	1.10	0.64
T_{eva}	°C	–	–	86.63	86.27	85.18	106.6	95.65	107.62
T_{cond}	°C	30	26.66	30	26.43	30	45.57	30	35.57
p_{cond}	bar	7.7002	6.99	11.67	10.48	0.043	0.099	0.025	0.033
η_{turb}	%	85	87.06	85	88.7	85	86.35	85	85.56
n_{stages}		–	2	–	4	–	6	–	2
Ns		–	0.084	–	0.1	–	0.1	–	0.1
SP	m	–	0.252	–	0.159	–	0.913	–	2.01
Rotational speed	RPM	–	3000	–	8385	–	1500	–	600

advantageous configuration also from the economic point of view due to the higher $\Delta T_{ml,PHE}$ as for the water. The main difficulty found in using high critical temperature fluids is the very low condensing pressure and the consequently high specific volume at turbine exhaust, causing a high cost of this component. In conclusion, even if water and octane are competitive from a thermodynamic point of view, they are economically unsuitable for medium low temperature geothermal sources. In particular, water is more indicated for high temperature heat sources with net power of at least 10 MW, in order to reduce criticalities (small volume flow rates) in the design of the turbine first stage. On the other hand, octane and other complex hydrocarbons and siloxanes are attractive for high temperature small-scale cogenerative applications, where the higher condensing temperature and pressure lead to a favorable size of the last turbine stage.

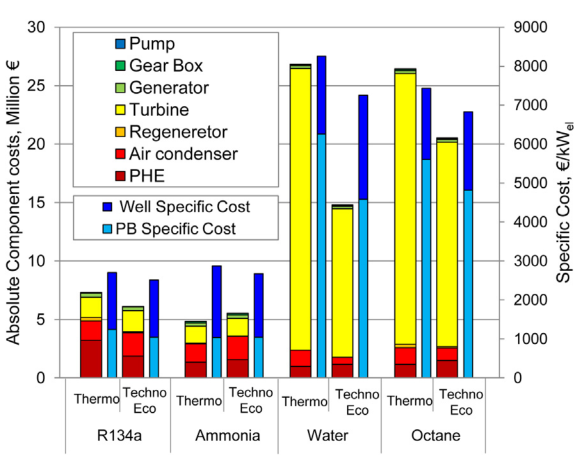

Fig. 5. Absolute component cost repartition (left axis) and specific costs (right axis) achievable for four test fluids for a geothermal source at 150 °C and mass flow rate is equal to 200 kg/s; cost are representative of a large scale, standard plant, located in an area with a favorable geothermal gradient.

3.4. Geothermal well cost sensitivity analysis

In this paragraph the influence of the geothermal well cost on the optimal economic cycle is shortly discussed. The total drilling cost is a site dependent variable, strongly related to well depth and to the geothermal gradient. It also depends on the type of rocks to be drilled and the commodity price. In some cases, a geothermal well could be already present and only the costs of repowering have to be considered. All these aspects reflect in a wide variability for geothermal well cost. Therefore, a sensitivity analysis on the supercritical R134a cycle previously considered is carried out by reducing the field cost by 50%, 75% and 100%. The main results are listed in Table 6 and a graphical comparison between economic optimizations obtained with maximum and zero well costs are reported in the T–s and T–Q diagrams in Fig. 6.

All the reported variables show relatively small changes for the first three simulations but a strong change when a null cost of geothermal well is assumed. By reducing the geothermal well cost, economic optimization moves towards cycles with a lower efficiency, that is mainly obtained by increasing ΔT_{ml} in both PHE and condenser. Variation of ΔT_{ap} of these two components entails a reduction of the cycle efficiency and in the last two cases also a reduction of the heat recovery efficiency that strongly penalize power production.

In particular, for a well cost equal to zero the $\Delta T_{pp,PHE}$ does not occur internally in the heat exchanger but at the hot end, because of the high value of the optimal $T_{reinj,geo}$. In addition, regenerator is almost inhibited as shown in Fig. 6. On the contrary, the maximum pressure remains almost unchanged among all the investigate cases. In conclusion, geothermal field cost strongly affects the total plant specific cost. However, for this specific case investigated, it is also shown that a quite high underestimation (even −50%) has just a little influence on cycle design. This aspect is crucial because it entails a more general validity of the economic optimization results obtained.

Table 6
Results for well cost sensitivity analysis applied to an R134a supercritical regenerative cycle.

		Well cost reduction multiplier			
		1	0.5	0.25	0
$W_{specific}$	kW/(kg/s)	40.962	39.121	35.917	18.348
η_{cycle}	%	12.36	11.8	11.56	8.35
η_{rec}	%	100	100	93.75	66.25
η_{plant}	%	12.36	11.8	10.83	5.53
η_{II}	%	50.39	48.13	44.18	22.57
W_{gross}	MW	10.21	9.97	9.14	4.972
W_{pump}	MW	1.467	1.533	1.422	1.008
W_{fan}	MW	0.55	0.61	0.53	0.30
C_{TOT}	M€	20.57	13.86	9.81	3.03
C_{PB}	M€	8.57	7.86	6.81	3.03
$C_{TOT,\ Specific}$	€/kW	2510	1772	1366	825.7
$C_{PB,\ Specific}$	€/kW	1046	1005	948	825.7
$\Delta T_{ap,PHE}$	°C	13.48	14.92	16.12	16.27
$\Delta T_{pp,PHE}$	°C	6.44	7.54	9.14	16.27
$\Delta T_{ml,PHE}$	°C	12.70	13.87	15.65	24.32
$\Delta T_{ap,cond}$	°C	11.66	12.66	15.41	30.31
$\Delta T_{pp,cond}$	°C	0.43	0.60	0.85	2.50
$\Delta T_{ml,\ cond}$	°C	5.15	5.86	6.88	15.8
$V_{ratio,\ stage}$		2.56	2.60	2.45	3.99
η_{turb}	%	87.06	86.97	87.22	83.92
n_{stages}		2	2	2	1
Ns		0.084	0.083	0.083	0.045
SP	m	0.252	0.247	0.238	0.155

3.5. Techno-economic optimization results

The economic optimization routine is applied to all the considered fluids for the three selected geothermal brine inlet temperatures 120, 150 and 180 °C. Both supercritical and subcritical cycles with or without gearbox are considered.

In Fig. 7, the detailed results of the 150 °C brine case, with 70 °C reinjection temperature limit are reported. Firstly, it is possible to note a rather regular trend of the total plant specific cost on $T_{crit}/T_{in,geo}$ parameter. The minimum is found in the range between 0.8 and 0.9, where several cycles with different fluids can achieve good economics. For values of $T_{crit}/T_{in,geo}$ lower than 0.8, the cost of optimized solutions rapidly increases due to the reduction of plant performances as already found from the thermodynamic analysis. For values above 0.9, costs rise as well, due to the lower efficiency of subcritical saturated cycles and to the strong increase of turbine cost which mainly affects the final solution. In this case, the optimal solution is an R134a supercritical regenerative cycle with no gearbox. For this fluid, a well-designed two-stage turbine directly jointed with the generator can be used. For ammonia, results also deserve to be discussed: the optimal cycle is subcritical with no regeneration and a high level of superheating. In this case, the adoption of a speed reducer is advantageous and the mechanical losses associated to this component are more than compensated by the increased turbine efficiency. Optimal rotational speed is around 8350 RPM (rotation per minute) and allows increasing the specific speed Ns of the last stage, which would be far from the optimum value without the adoption of a speed reducer.

From these calculations, it appears that the use of a gearbox is generally not profitable. However, this result is strongly affected by the thermal level and the flow rate of the geothermal brine. An increase in energy input modifies the working fluid mass flow and, for a given set of design parameters, the volume flow rate discharged by the turbine. This variable have a direct influence on Ns and finally on the turbine efficiency.

In all the investigated applications, fluids with high critical temperatures show extreme values of outlet volume flow rates, which entail optimized rotational speed far below 3000 RPM. In these cases, the adoption of multi-pole generator is certainly advantageous, but turbine size and cost increase a lot, leading to non-competitive costs. As already mentioned, this kind of fluids are more indicated for back pressure co-generative power plants where an increase of the condensing pressure allows keeping a reasonable diameter of the last turbine stage.

Finally, for fluids with $T_{crit}/T_{in,geo}$ just above 0.9, supercritical cycles have specific costs higher than subcritical ones, even if they

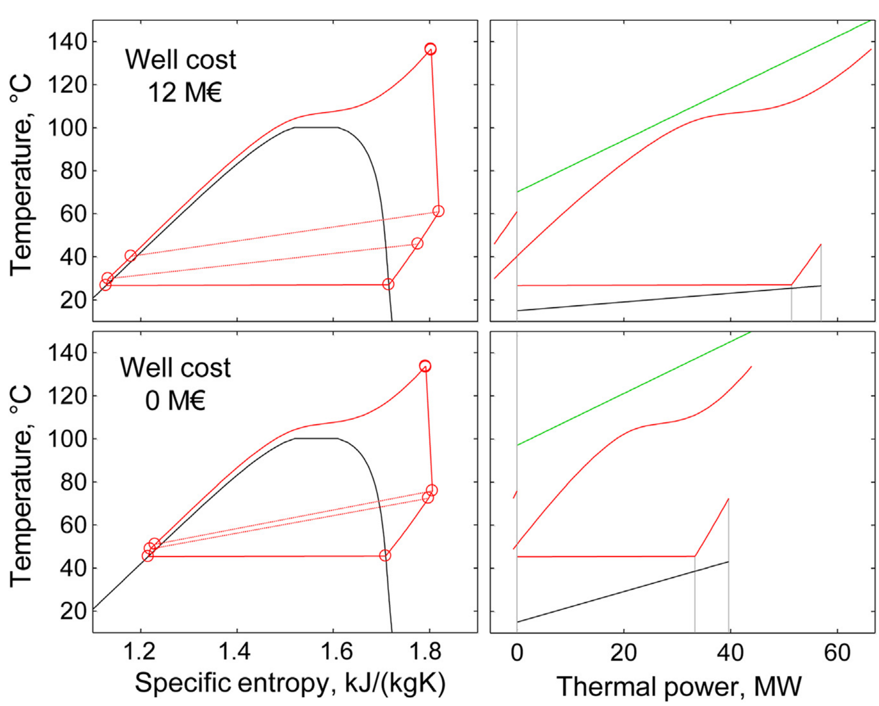

Fig. 6. Optimized power cycles diagrams for the extreme assumed well costs.

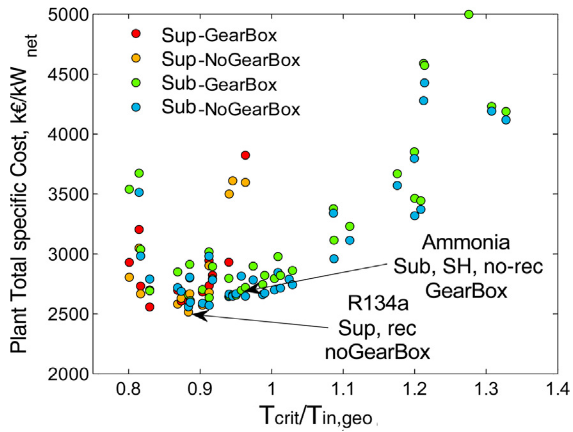

Fig. 7. Economic optimization results for a 150 °C geothermal brine.

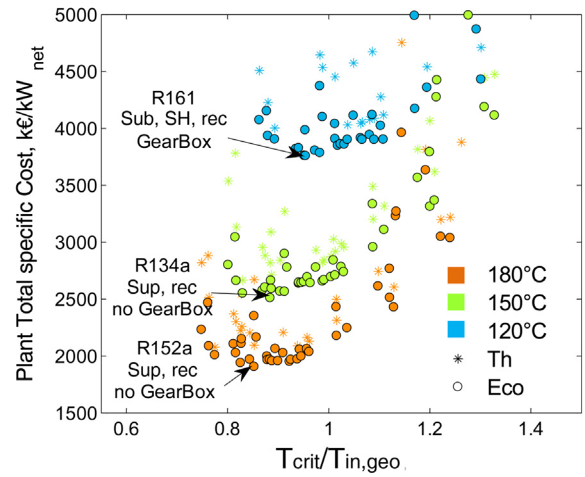

Fig. 8. Economic results for all the investigated brine temperatures.

allow achieving higher performances. Supercritical cycles are kept close to the saturation line and optimal $p_{\text{in, Turbine}}$ and $\Delta T_{\text{ap,PHE}}$ vary in narrow ranges, due to the constraint keeping the expansion in the superheated vapor region. In these cases, the economic optimization cannot effectively increase $\Delta T_{\text{ml,PHE}}$ without changing $T_{\text{reinj,geo}}$ and subcritical cycles can achieve lower specific costs because of the lack of this constraint.

Optimal results for all cycle and for each brine temperature are reported in Fig. 8. Circle markers refer to economic optimization results, while stars are representative of values obtained from thermodynamic optimization. First, it is possible to see the reduction of total plant specific cost as a result of the economic optimization procedure. In many cases this difference seems to be quite small, but it is important to remember that the well cost strongly affects the final solution for geothermal binary power plants. In particular, for the assumed exploration and drilling costs, economic and thermodynamic optimum are quite similar, due to the need of amortizing a large initial investment cost. Moreover, values computed in the thermodynamic optimization are based on a constant turbine efficiency, a hypothesis which is not necessarily warranted for all considered fluids.

For the two extreme temperatures the same trend previously described can be recognized. The optimal solution for the 180 °C brine is a regenerative supercritical cycle with no speed reducer, which entails the same considerations already discussed for 150 °C brine optimal solution. Different aspects can be instead highlighted for the 120 °C brine case, where the optimal solution is a R161 subcritical superheated cycle with regeneration and adoption of a gearbox. This fluid shows a bell shaped saturation line, with no possibility to realize a saturated cycle. The small difference between the critical and the condensing temperature, combined with the low working fluid mass flow, reflects in a small volume flow discharged from the turbine. If the turbine is directly jointed to the generator, too small Ns values are obtained with a strong penalization of turbine performance. On the other hand, with the adoption of a gearbox, the optimized rotational speed results around 6500 RPM with an increase of more than 4% points of turbine isentropic efficiency.

Referring to Fig. 9 it is possible to describe how the optimization routine acts in order to achieve the minimum specific cost. For fluids with low $T_{\text{crit}}/T_{\text{in,geo}}$ optimal techno-economic results are obtained thanks to an efficiency increase and an almost constant plant cost, as can be explained observing optimal values of $\Delta T_{\text{ap,cond}}$ and $\Delta T_{\text{ap,PHE}}$. Optimal $\Delta T_{\text{ap,cond}}$ values are lower than 15 °C (fixed value in thermodynamic optimization) entailing an higher power output but even a more expensive air cooled condenser. On the other hand optimal $\Delta T_{\text{ap,PHE}}$ values are lower than those obtained in previous case involving a smaller PHE and a detrimental effect on plant performances. A Tradeoff between these effects allows to achieve a smaller specific cost respect to thermodynamic optimization. For fluid with high $T_{\text{crit}}/T_{\text{in,geo}}$ values instead the main difficulty is related to turbine sizing and cost. In order to limit the last stage dimension optimal values of $\Delta T_{\text{ap,cond}}$ are higher than in thermodynamic optimization entailing a lower plant efficiency but also a lower cost of both turbine and condenser unit. $\Delta T_{\text{ap,PHE}}$ optimal values are close to the lower bound with a high cost of the PHE and good effects on plant performance. For fluids near the optimum, the techno-economic optimizations acts in different ways depending on fluid properties and a univocal interpretation of results cannot be done.

Finally, it is interesting to highlight that some optimization variables assume values almost constant for each optimized solution around the minimum. Optimal values for $\Delta T_{\text{ap,cond}}$ and $\Delta T^{*}_{\text{pp,Cond}}/\Delta T_{\text{ap,cond}}$ are reported as example in Fig. 10. Lower $\Delta T_{\text{ap,cond}}$ yields lower condensing temperature, while lower $\Delta T^{*}_{\text{pp,cond}}/\Delta T_{\text{ap,cond}}$ requires low speed-low consumption condenser fans and show how a reduction of the geothermal brine temperature entails lower values for both parameters. In this cases

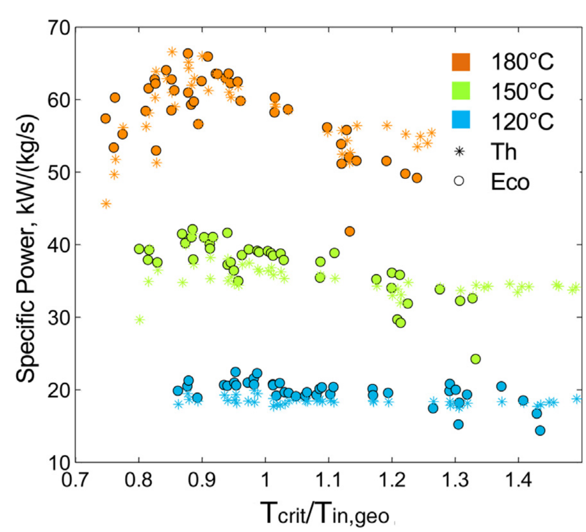

Fig. 9. Specific power index for techno-economic optimizations.

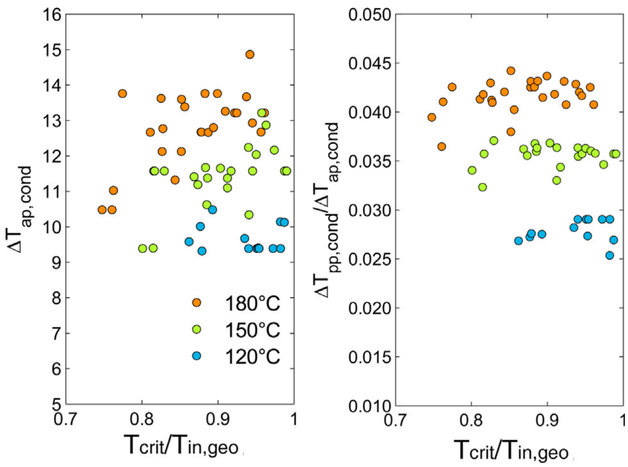

Fig. 10. Optimal values of condenser unit ΔT_{ap} and $\Delta T_{pp}/\Delta T_{ap}$.

the weight of geothermal well cost is so high that the solution is pushed towards a more expensive but also more efficient power plant.

4. Conclusions

In this work a techno-economic optimization of different ORC configurations operating with a number of working fluids is performed, considering equipment cost correlations and a model to estimate the turbine design and efficiency. The multivariable non-linear optimization is carried out with a two level routine, which acts by varying six design parameters ($p_{in,turb}$, $\Delta T_{ap,PHE}$, $\Delta T_{ap,cond}$, $\Delta T_{pp,rec}$, $T_{reinj,geo}$ and $\Delta T^{*}_{pp,cond}$), with the objective of minimizing the total plant specific cost. Results can be summarized in the following statements:

- The choice of fluid and cycle configuration is crucial in order to minimize the total plant specific cost: important performance and cost variations result between optimal and non-optimal cycles.
- The techno-economic optimization provides results different from the ones obtained from the thermodynamic analysis, confirming its primary importance in the optimization of ORC plants. However, it is confirmed that for sufficient maximum geothermal brine ΔTs (in presence of a high brine temperatures or low minimum reinjection temperatures), supercritical cycles perform better also from the economic point of view.
- Optimal fluids have $T_{crit}/T_{in,geo}$ parameter close (slightly lower) to the ones found for the thermodynamic analysis, but advantages related to supercritical cycles over subcritical ones are less evident.
- Adopting fluids with $T_{crit}/T_{in,geo}$ parameter lower than 0.8 in supercritical configuration entails higher plant costs due to the poor cycle efficiency, while for values above 0.9–1 specific costs rise mainly because of the increase of the turbine cost.
- Gearbox is not always profitable; for the investigated cases it can be conveniently adopted for low molecular complexity fluids (e.g. water, ammonia, R161) to obtain acceptable turbine efficiencies.
- High critical temperature fluids require turbines with a low speed of revolution in order to have specific speed near the optimal value and hence the adoption of a multi-pole generator.
- Sensitivity analysis on geothermal well cost shows that the optimal cycle parameters do not change appreciably by changing the geothermal well cost, at least when its contribution on total cost is not lower than half of the cost of the power block.
- An increase of the relative cost of the geothermal well push the economic optimization towards cycles with higher efficiencies and higher power block absolute costs.

Nomenclature

Acronyms

BOP	balance of plant
COE	cost of electricity, €/MWh
Cs	specific cost, €/kW
GWP	global warming potential index
HTF	heat transfer fluid
LB	lower bound
ODP	ozone depletion potential index
ORC	Organic Rankine Cycles
PB	power block
UB	upper bound
WHR	waste heat recovery

Notation

C	cost, €
eco	techno-economic optimization
k	stage load coefficient
n	number of turbine stages, #
Ns	stage specific speed, rotations
p	pressure bar
Q	kW
rec/no-rec	recuperative/non recuperative cycle configuration
RPM	rotation per minute
SA	saturated or slightly superheated cycle
SH	superheated cycle
SP	stage size parameter, m
Sub	subcritical cycle
Sup	supercritical cycle
T	temperature, °C
Th	thermodynamic optimization
u	fluid velocity in heat exchangers, m/s
U	global heat transfer coefficient, kW/(m^2 K)
V	volume flow rate, m^3/s
W	power, kW
α	specific heat exchange area, m^2/kW
Δh	enthalpy drop, kJ/kg K
Δp	pressure difference bar
ΔT	temperature difference, °C
η	efficiency, %

Subscripts

0	reference condition for cost correlations
amb	ambient conditions
ap	approach point
cond	condensation (condition) or condenser (plant component)
crit	critical condition
CS	HE cold side
cw	cooling water
des	desuperheating section of condenser (plant component)
eco	economizer (plant component)
el	electrical
eva	evaporation (conditions) or evaporator (plant component)
gen	generator (plant component)
geo	geothermal brine
HS	HE hot side
II	second law
in	inlet condition

is	isentropic process
lim	limit in reinjection condition
lor	lorentz (efficiency)
loss	dispersion to the environment
m	mean stage diameter
mec	mechanical
ml	mean logarithmic
PHE	primary heat exchanger
pp	pinch point
Pr. Lev.	pressure level (for subcritical cycles)
rec	recovery (efficiency) or regenerator (plant component)
reinj	reinjection condition
sc	sub cooling
scroll-exp	scroll expander
sh	superheater (plant component)
SHE	supercritical PHE
th	thermal turbturbine (plant component)
val	valve at turbine inlet (plant component)
wf	working fluid

References

[1] Shengjun Z, Huaixin W, Tao G. Performance comparison and parametric optimization of subcritical Organic Rankine Cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation. Appl Energy 2011;88:2740–54.

[2] Hettiarachchi HDM, Golubovic M, Worek WM, Ikegami Y. Optimum design criteria for an Organic Rankine cycle using low-temperature geothermal heat sources. Energy 2007;32:1698–706.

[3] Quoilin S, Declaye S, Tchanche BF, Lemort V. Thermo-economic optimization of waste heat recovery Organic Rankine Cycles. Appl Therm Eng 2011;31:2885–93.

[4] MATLAB version 7.8.0. Natick, Massachusetts: The MathWorks Inc; 2009.

[5] Lemmon EW, Huber ML, McLinden MO. NIST standard reference database 23: reference fluid thermodynamic and transport properties-REFPROP, version 9.0. Gaithersburg: National Institute of Standards and Technology, Standard Reference Data Program; 2010.

[6] Di Pippo R. Ideal thermal efficiency for geothermal power plants. Geothermics 2007;36(Issue:3):276–85.

[7] Lewis RM, Torczon V. A globally convergent augmented Lagrangian pattern search algorithm for optimization with general constraints and simple bounds. SIAM J Optim 2002;12(4):1075–89.

[8] Atlas Copco website: www.atlascopco.com

[9] Infinity Turbine website: www.infinityturbine.com

[10] Ormat website: www.ormat.com

[11] Turboden website: www.turboden.eu

[12] Exergy website: www.exergy-orc.com

[13] Spadacini C, Centemeri L, Xodo LG, Astolfi M, Romano MC, Macchi E. A new configuration for Organic Rankine Cycle power systems. In: First Interational Seminar on ORC Power Systems, Delft (NL) 22–23 September 2011.

[14] Macchi E, Perdichizzi A. Efficiency prediction for axial flow turbines operating with non-conventional fluids. ASME J Eng Power 1981;103:712–24.

[15] Holland FA, Wilkinson JK. Process economics. In: Perry's chemical engineers' handbook. 7th ed. McGraw-Hill; 1999.

[16] Filippini S, Merlo U, Romano MC, Lozza G. Potential of water-sprayed condensers in ORC plants. In: First International Seminar on ORC Power Systems, Delft (NL) 22–23 September 2011.

[17] Turton R, Bailie RC, Whiting WB, Shaeiwitz JA. Analysis, synthesis and design of chemical processes. 3rd ed. Prentice Hall; 2009.

[18] Bombarda P. Estimating cost of the geothermal power technologies: main aspects and review. In: EC geothermal electricity workshop – FP7 – DG ENER, Bruxelles (B) 29th April 2011.

[19] Ghasemi H, Paci M, Tizzanini A, Mitsos A. Modeling and optimization of a binary geothermal power plant. Energy 2013;50:412–28.