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Comparison of Enhanced Organic Rankine Cycles for Geothermal Power Units

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ABSTRACT

Binary cycles have drawn the attention as a technical solution for the geothermal power production. This attention is mainly due to the huge potential of medium-low temperature geothermal sources, typically exploited by means of a binary cycle, and the relevance of the environmental concern, which can be conveniently dealt with by means of a closed cycle. The binary cycle has been therefore the object of an extended research activity, in order to attain higher plant performance. A crucial matter is the improvement of the heat introduction process. For a given geothermal fluid in liquid state, i.e. for a variable temperature heat source, in a conventional ORC the working fluid evaporation process is responsible for an important second law loss: removal of this loss allows greater power and possibly higher cycle efficiency to be attained. Aim of the present paper is to investigate and compare recently proposed technical solutions based on the current technology, which do not entail considerable operating risk or relevant investment; they can however lead to an improvement in plant performance and economics. The selected cycle options were dealt with in the open literature, and try to reduce the heat introduction second law loss: in the first one, the so called OFC, this loss is strongly reduced, because heat is introduced in the cycle when the working fluid is in liquid phase, but a dissipative flash process is then required. In the second one, the so called Pinch Point Smoother, this loss is reduced because the working fluid heating curve is smoothed by means of a flow split, which allows a fraction of the working fluid flow to evaporate at a pressure lower than the pressure of the main flow, but mechanical recompression is then required to inject the separated flow fraction into the turbine. The result of comparison may depend both on the temperature level of thermal sources involved and on the working fluid selected: the present paper will discuss several examples, representative of geothermal applications, and try to assess whether the adoption of these solutions can be convenient for geothermal exploitation.

1. INTRODUCTION

The selection of the plant scheme and cycle parameters for a geothermal plant is a complex item, involving many variables. In the present paper liquid-phase geothermal fluid only will be considered. This focus on liquid-phase is due to the general trend towards the exploitation of lower temperature and lower enthalpy geothermal heat sources, associated with the basic requirement to avoid any emission of pollutants. An effective way to reduce emissions consists in keeping the geothermal fluid liquid by proper pressurization, all along its pattern through the wells and at the surface in the power block, and implementing a full reinjection of the geothermal fluid and its gas and salt content. Also, the heat exchanger scaling abducted during maintenance, a waste product which has to be disposed, is normally present in smaller quantity if the heat source fluid is pressurized and phase change is avoided.

The huge potential of medium-low temperature geothermal sources is typically exploited by means of a binary cycle. The binary cycle has been therefore the object of an extended research activity, in order to attain higher plant performance. A crucial matter is the improvement of the heat introduction process. For a given geothermal fluid in liquid state , i.e. for a variable temperature heat source, the working fluid evaporation process is responsible for an important second law loss in a conventional subcritical ORC: removal of this loss allows greater power and possibly higher cycle efficiency to be attained.

The multilevel evaporation cycle, the Kalina cycle, the supercritical cycle, the zeotropic mixture cycle are just some of the most known binary cycle options, all aimed at reducing the second law heat introduction loss; contextually to these cycle configurations, the trilateral cycle was also proposed.

Aim of the present paper is to investigate and compare recently proposed technical solutions based on the current technology, which do not entail considerable operating risk or relevant investment and which can however lead to an improvement in plant performance and economics. The Kalina cycle and the zeotropic mixture cycle were therefore intentionally disregarded. The former, in fact, though already implemented in a few power plants (Husavik, Hjartarson et al. (2005)), Unteraching (Richter (2010)), Bruchsal (Herzberger (2010))) and possibly others) entails however high operation pressures, corrosion problems, and a complex plant scheme; the latter, only exceptionally adopted, entails disadvantages because of possible working fluid composition change and low heat transfer coefficient (it is known that the heat transfer coefficient for mixtures is lower than that of pure fluids). Recently proposed cycles which adopt a two-phase expander, were as well disregarded, as the two–phase expander is not yet a proven technology.

Moreover, in the present paper focus will be placed on subcritical cycles, though some supercritical cycles will be considered in the comparison too. It is in fact well known (Angelino and Casci (1969)) that supercritical cycles have a definite advantage concerning the good matching of the heat release curve of the geothermal fluid and the working fluid heat input curve. In a supercritical cycle, also, the pressure in the whole organic fluid circuit is often much higher than in the subcritical plants. This fact yields much lower volume flow rates on the vapour side components and piping: hence the cross section area for the flow of working fluid is lower, components are compact and the cost of both components and connecting piping can possibly be lower.

As a matter of fact, however, for most of the existing successful geothermal ORC plants Di Pippo (2005)/ a subcritical cycle was adopted; nevertheless, very effective supercritical systems can be obtained (Astolfi et al. (2013)). Both technical solution have their own strength points.

Most of the subcritical cycles adopted are characterized by a working fluid which is liquid at normal ambient temperature, so that it can be easily stored and handled as a liquid without large pressurization. For example iso-pentane, a well-established working fluid, has a 28 °C boiling point, and normally it condenses at a pressure slightly above the atmospheric pressure in an air-condenser. Though geothermal ORC systems are well cared of from the design and construction aspects concerning tightness, a low pressure in the condensing section is certainly an advantage from the point of view of potential leakages; all the more so, if the pressure falls below the atmospheric pressure when the machine is inactive. Another intrinsic advantage is that the low pressure at the evaporator involves lower parasitic power requirements for the liquid feed pump; in ORC systems in fact, due to the low specific work produced in the turbine, the power to drive the feed pumps is much larger than in an equivalent steam system: in the case of supercritical cycles the power of the pump is often of the same order of magnitude as the turbine power. Finally, a further point in favour of low pressure systems, comes from the loads imposed to some critical components by high pressure, namely in the turbine, the sealing systems, the system control valves. As a consequence of the advantages emphasized, the low pressure ORC are really tough and robust.

On these premises, the authors decided to compare the basic simple subcritical Rankine cycle, with Rankine cycles including some cycle enhancements, in order to obtain an increased performance.

In the so called Pinch Point Smoother, introduced by Gaia and Pietra (2013), the working fluid heating curve is “smoothed” by means of a flow split, which allows a fraction of the working fluid flow to evaporate at a pressure lower than the pressure of the main flow, but mechanical recompression is then required to inject the separated flow fraction into the turbine. In the so called Organic Flash Cycle (OFC)(Note that most of the papers found in literature relating to flash cycles, see for example Fischer and Lai (2012), involve a two-phase expander, and are therefore not considered in this work.), see Ho et al, (2012), heat is introduced in the cycle when the working fluid is in liquid phase, but a dissipative flash process is then required.

The result of comparison may depend both on the temperature level of thermal sources involved and on the working fluid selected: the present paper will discuss several examples, representative of geothermal applications, and assess whether the adoption of these solutions can be convenient for geothermal exploitation.

2. SELECTED CYCLE SCHEMES AND WORKING FLUID

The selected cycle schemes are described in the following paragraphs; in all cases the hydrofluorocarbon HFC-245fa (1,1,1,3,3 pentafluoropropane) is selected as working fluid. This working fluid (see tab. 1 for thermophysical properties) is a refrigerant largely used in actual plants, not flammable, featuring low toxicity and easily available, which can be conveniently adopted in subcritical cycles up to a maximum source temperature of about 170 °C; for higher temperature sources a supercritical cycle or a working fluid with a higher critical temperature should be adopted.

Table 1: Thermophysical properties of HFC245fa

Critical temperature (°)	154.05
Critical pressure (bar)	36.4
Normal boiling point (°C)	15.3
Molecular mass (g/mol)	134.05
Ozone Depletion Potential, ODP	0
Atmospheric lifetime (years)	≈ 7
Global Warming Potential, GWP	≈ 1000

Basic single pressure level ORC, two pressure level ORC and in some cases supercritical ORC (adopting a proper working fluid) are here considered for comparison.

These selected cycle schemes aim at the best utilization of the exchange surface. It is well known that the minimum temperature difference between the two streams of an evaporator (the so called Pinch Point Temperature Difference, ΔT_{pp}) is a techno-economic key parameter: reducing this temperature difference requires a higher heat transfer surface, but allows better thermodynamic performance; however, the gain is reduced when the ΔT_{pp} becomes small, because of the constant temperature evaporation process (see fig.1)

All the enhanced cycles considered in this work aim at solving this problem and require a higher heat transfer surface with respect to the base single pressure cycle: the scope is to evaluate which solution has the best ratio between net power gain and extra surface.

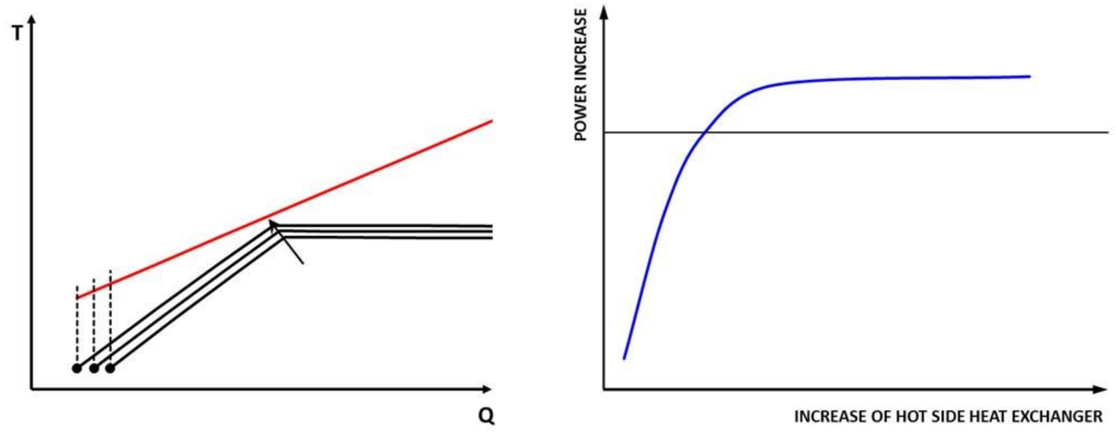

Figure 1: T-Q diagram for single pressure level ORC with different values of ΔTpp (left) and ORC power increase versus hot heat exchanger surface

2.1 Pinch Point Smoother

The plant scheme required for the so called Pinch Point Smoother, is shown in figure 2: the heating of the working fluid up to saturation condition is accomplished by means of two subsequent preheaters: the working fluid flow is split after the first preheater, so that a minor fraction of the flow can be sent to a throttling valve, in order to evaporate at a pressure lower than the pressure of the main flow; mechanical recompression is then required to inject the separated flow fraction into the main flow before the turbine. Such a plant scheme allows a "smoothed", more favourable T-Q diagram with respect to the base case of the single pressure level ORC and can be accomplished with a conventional single admission turbine.

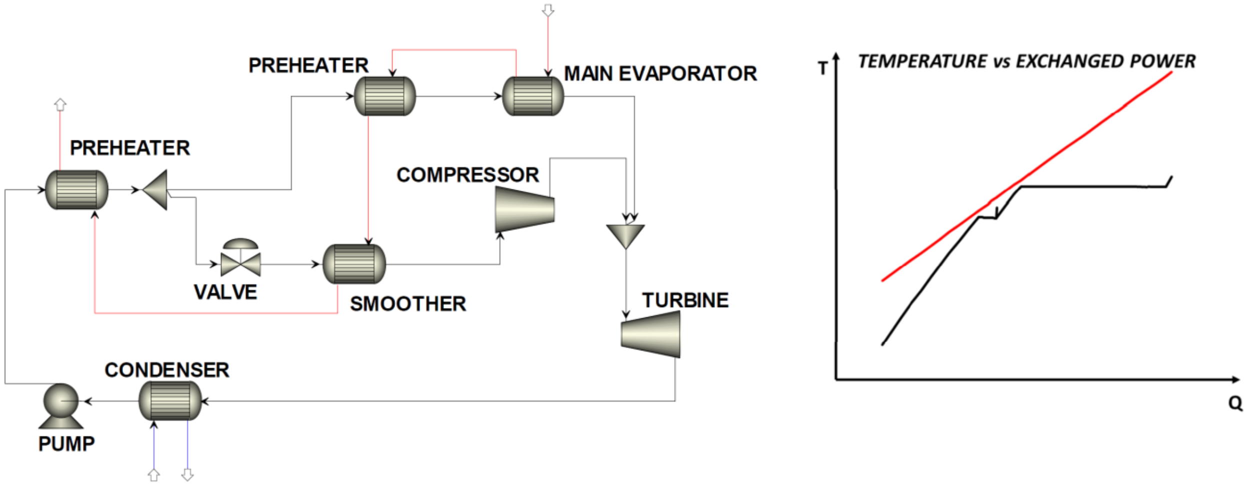

Figure 2: ORC plant scheme with pinch point smoother (left) and corresponding "smoothed" T-Q diagram (right) (derived from a quantitative example)

The main advantage of the Pinch Point Smoother scheme is the proper utilization of the heat exchangers surface.

2.2 OFC cycle schemes

The so called Organic Flash Cycles were recently investigated by Ho, Mao and Greif (2012a): OFC distinctive feature is that the liquid is pumped to a pressure higher than the turbine inlet pressure, so that heat is introduced in the cycle when the working fluid is in liquid phase, and subsequently the working fluid is flashed down to the turbine inlet pressure, in order to generate the vapour to be expanded. Heat is thus introduced in the cycle by means of a single primary heat exchanger placed downstream from the pump, where the liquid is heated up to the saturation point. The basic advantage is clear: the heating and cooling curves in the T-Q diagram are almost ideally matched, the plant scheme is simple, and no new technology is required. However, there are two major drawbacks, the dissipative flash process and the unused working fluid liquid fraction, which is sent to the condenser after a throttling process.

In the preliminary analysis conducted Ho, Mao and Greif found that, for the base OFC, at least for the selected fluids (aromatic hydrocarbons and syloxanes), no real advantage occurs with respect to basic ORC, because the second law loss which is saved during the heat introduction process is than lost in the flash process, thus giving no particular advantage to OFC versus ORC in this case.

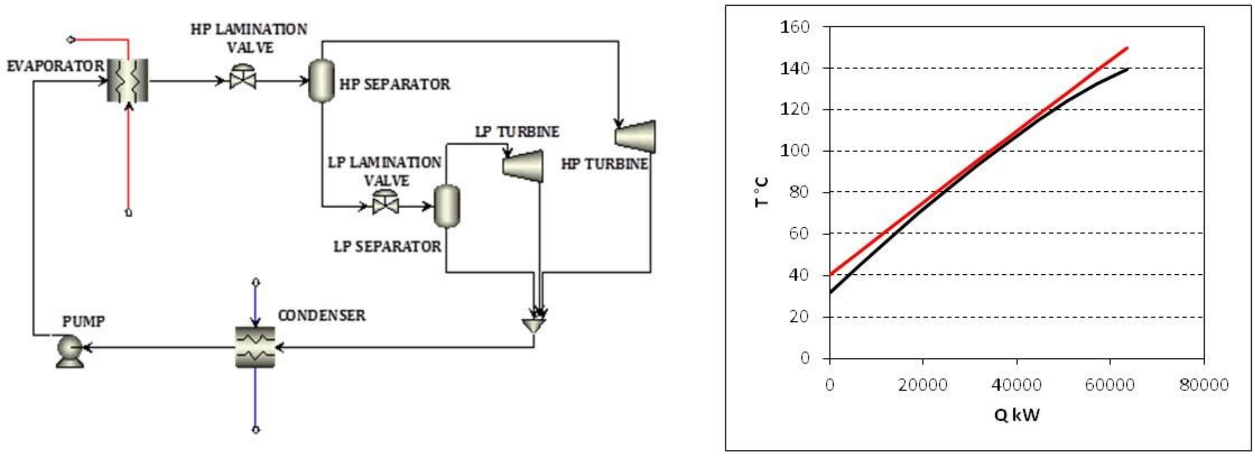

Figure 3: OFC plant scheme with double flash (left) and corresponding T-Q diagram (right), ΔTpp=2 °C

On the basis of the work carried out, observing that the single OFC performs roughly the same as the base ORC, Ho, Mao and Greif (2012b) argued that any possible enhancement of the OFC should give to the latter a clear advantage versus the base ORC. They conducted therefore an additional study considering several OFC enhanced schemes: among these, the double flash and the modified OFC were obtainable with current technology, though employing two turbines, while the other enhanced OFC scheme required the two-phase turbine technology (and will be therefore not mentioned in this work). In the double flash cycle (fig. 3), the flash process is repeated two times, so that the unused liquid fraction is reduced. The modified OFC (fig. 4) has a two stage expansion, and employs the throttled liquid fraction to desuperheat the vapour at the end of the first stage expansion: it may be profitable but only with some specific fluids. As a matter of fact, the conclusion of the study was that the modified cycle was profitable for aromatic hydrocarbons (+10%), but it proved useless with siloxanes.

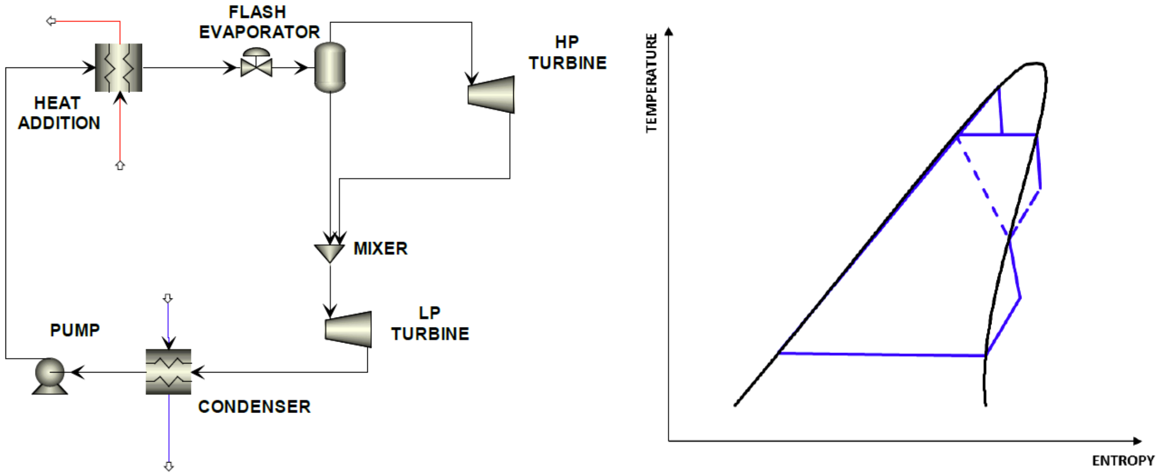

Figure 4: OFC modified

As already mentioned, HFC245fa is considered in this study as working fluid: no information was found in literature regarding OFC performance with refrigerants. Preliminary calculation performed in the frame of this work ((not reported here) showed that, for HFC245fa, with a geothermal source at 150°C the performance of the single level OFC is definitely lower than that of an optimized single level ORC. It was therefore decided to consider in this work only the double flash cycle; the enhanced modified version of the flash cycle with HFC245fa proved also useless.

3. PERFORMANCE EVALUATION

The different cycle schemes are calculated by means of the same commercial process simulator, Aspen plus (2006) which allows to properly evaluate both geothermal fluid and working fluid thermodynamic properties in order to fulfill a correct simulation of the plant behavior and a sound comparison between different plant schemes.

3.1 Calculation model

The thermodynamic properties of the ORC working fluid are calculated by means of a modified Peng-Robinson equation.

3.1.1 Basic assumptions

Basic assumptions for the calculation are reported in table 2. Assumptions regarding the geothermal fluid flow are representative of the Molasse Basin typical operating conditions; moreover no restriction for the discharge temperature was assumed.

Table 2: Basic assumptions

Geothermal fluid temperature	Variable in the range 125-150°C
Geothermal fluid salt content	1 g/l
Geothermal fluid volumetric flow	150 l/s
Ambient temperature	15°C (as per ISO standard)
Condenser cooling medium	Air
Air cooler electric parasitic consumption, percentage of condensation heat	1%
Turbine on-design isoentropic efficiency	0.85
Generator mechanical-electrical efficiency	0.97
Pumps hydraulic efficiency	0.75
Pump mechanical-electrical efficiency	0.95
Compressor isoentropic efficiency	0.8
Compressor motor efficiency	0.96
Hot heat exchangers minimum ΔT_{pp}	2°C
Hot heat exchangers, working fluid site: preheater Δp	1bar
Hot heat exchangers, working fluid site: evaporator Δp	0.2bar
Cold heat exchangers minimum $\Delta T_{pp,}$	2°C
Cold heat exchangers, working fluid site: condenser Δp	0.1bar
Working fluid condensation temperature	30°C

3.1.2 Cycle optimization and performance evaluation parameters

Each cycle has its own operating parameters which must be optimized in order to get the highest performance. The variables which were optimized are: the evaporation temperature for the base ORC case; the two evaporation temperatures for the two pressure level cycle; the fractions of working fluid mass flow and the evaporation temperatures for the Pinch Point Smoother; the flash pressures for the double OFC.

Optimization was conducted by means of a sensitivity analysis, continuously varying the parameters to be optimized and evaluating the net electrical power. This procedure, even if not as fast as an optimization algorithm, entails the advantage of permitting to easily follow and understand the dependence of the net electrical power with respect to the operational parameters; it becomes however cumbersome with complex plant schemes. In particular in this work the double flash cycle represents a limit situation, involving a considerable calculation time.

Calculations were carried out considering the net electrical power as the optimization function; the efficiencies were then calculated. For power plants receiving heat from variable temperature heat sources, sometimes called “finite thermal energy reservoirs”, which is the case of geothermal liquid sources, several plant efficiency definitions may be found in literature, above all for the second law efficiency (or exergy efficiency, see Di Pippo (2004)). In the frame of this work the plant second law efficiency is calculated as the ratio of the plant efficiency to the ideal Lorentz cycle efficiency, i.e.:

$$\eta_{I,plant} = \frac{W_{net}}{Q_{input}}$$

$$\eta_{II,plant} = \frac{\eta_{I,plant}}{\eta_{Lorentz}}$$

$$\eta_{Lorentz} = 1 - \frac{T_{amb}}{\left(\dfrac{T_{in,Geo} - T_{discharge,Geo}}{\ln\left(\dfrac{T_{in,Geo}}{T_{discharge,Geo}}\right)}\right)}$$

where $\eta_{I,plant}$ is the first law plant efficiency, W_{net} is the net electrical power, Q_{input} is the thermal power introduced in the cycle, $\eta_{II,plant}$ is the second law plant efficiency, $\eta_{Lorentz}$ is the efficiency of an ideal cycle operating between the same thermal sources of the real cycle, T_{amb} is the ambient temperature, $T_{in,Geo}$ is the geothermal fluid temperature at power plant inlet, and $T_{discharge,Geo}$ is the temperature of the geothermal fluid at the discharge from the power plant.

3.2 Results discussion

Calculation were performed with different constraints: constant minimum ΔT_{pp} and constant "US"; the first constraint is commonly used in power plant design; in the second case, "US", i.e. the product of the heat transfer coefficient multiplied by the heat transfer surface, is an indicator of the heat exchanger cost.

3.2.1 Comparison of plant performance at constant minimum ΔT_{pp}

Following table 2, plant performance was calculated considering the minimum allowed value for ΔT_{pp} (2 °C). Results presented in table 3 and figure 5 and 6 show that, with respect to the base single pressure level ORC, all the enhanced cycles allow a much higher turbine gross power. However, this advantage is partly counterbalanced by the higher ORC parasitic electric consumption: looking at the net power, the two- pressure level cycle, which has the least parasitic consumption among the enhanced cycles, emerges as the best cycle; the single pinch point smoother cycle, has still an advantage versus the base single pressure ORC, but the double flash cycle is only slightly better than the base ORC at 150 °C and performs worst at 125°C. As it can be immediately seen from figure 6, the required heat transfer surfaces are different in the various cases.

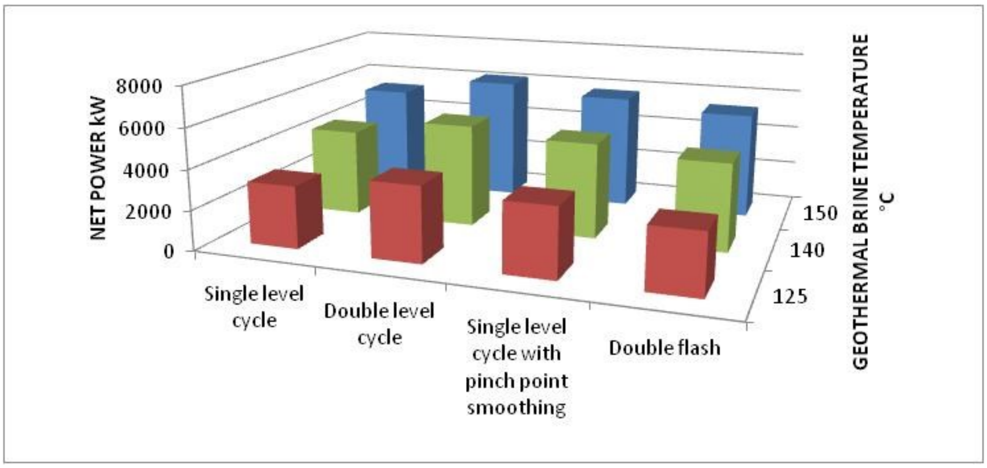

Figure 5: Net power comparison at constant ΔTpp=2°C.

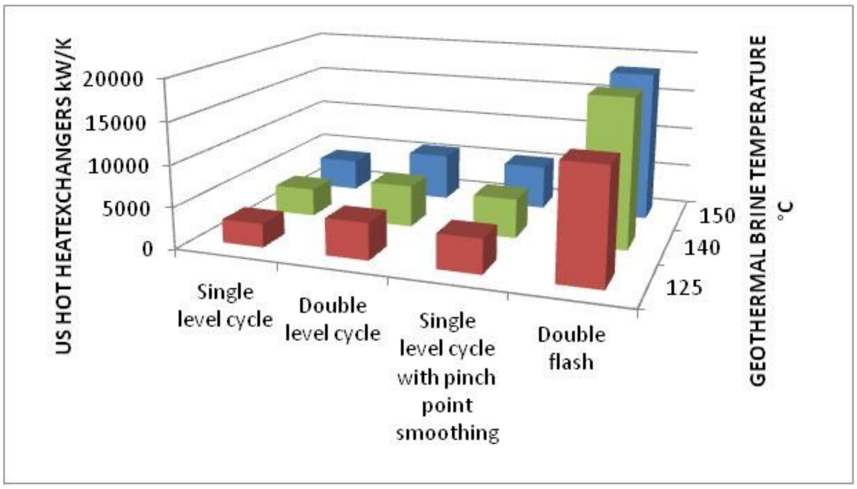

Figure 6: Required "US" for the hot heat exchanger at constant ΔTpp=2°C.

3.2.2 Comparison of plant performance at constant "US"

It is interesting to compare the cycles performance at constant "US", i.e. at the same value of the product between the overall heat transfer coefficient and heat exchanger surface: this comparison shows whether the extra heat transfer surface needed with respect to the base case, is properly used to allow more power. Estimating the cost of the power block in an ORC plant is a tough task: however, it is well known that the cost of heat exchangers represents a major fraction of the overall cost. Comparing the plant performance at the same US gives a roughly idea of performance at comparable heat exchanger cost.

Extended calculations were performed firstly at 150°C; the "US" value of the two pressure level ORC previously calculated with ΔT_{pp}=2°C (5700 kW/K for the hot heat exchangers and 5500 kW/K for the cold heat exchanger) was selected as the common reference value; two supercritical cycles, adopting R134a and propane as working fluid were added for comparison.

Table 3: Calculation results with constant ΔTpp=2°C. ORC auxiliary electric consumption comprise the ORC pump electric consumption and the Pinch Point Smoother compressor electric consumption.

	Base ORC	Two pressure level ORC	Single pinch point smoother	Double flash	Base ORC	Two pressure level ORC	Single pinch point smoother	Double flash
Geothermal source temperature, °C	150	150	150	150	125	125	125	125
Turbine gross electric power, kW	6002	7078	7157	6948	3549	4322	4409	4173
ORC auxiliary electric consumption, kW	254	383	914	1044	126	185	597	705
Air condenser electric consumption, kW	433	492	468	573	320	379	355	446
Net power, kW	5315	6203	5776	5331	3104	3757	3457	3022
Thermal power from geothermal source, MW	49,2	56,1	53,3	63,5	35,5	42,2	39,4	48,3
Plant efficiency, I law	10,8%	11,1%	10,8%	8,4%	8,7%	8,9%	8,7%	6,2%
Geothermal fluid discharge temperature, °C	65,3	53,3	58,2	40,4	65,0	53,6	58,3	43,3

Calculation results (presented in fig. 7) show that, at constant US, the best performance is obtained with the HFC245fa two pressure level cycle and R134a supercritical cycle: these cycle arrangements allow the same performance. A slightly lower performance is obtainable again with a supercritical cycle, but adopting propane as working fluid; when the supercritical cycle is concerned, the adoption of R134a with respect to propane should therefore be preferable, both because of better performance and non–flammability. As already mentioned, attention is drawn in this paper on subcritical cycles, and supercritical cycles are introduced only for comparison: it can be however stressed that, although with similar heat transfer surfaces, supercritical cycle heat exchangers, operating at high pressure, would possibly require higher material thickness than the heat exchanger of subcritical cycles, and cost could be therefore somewhat higher. This occurrence was verified for example by Astolfi et al.(2014), in the frame of techno-economic optimization for ORC power cycles aimed at geothermal sources; it was found that supercritical cycles allow both best thermodynamic and economic performance with respect to subcritical cycles, but from the economic point of view the advantage is less evident.

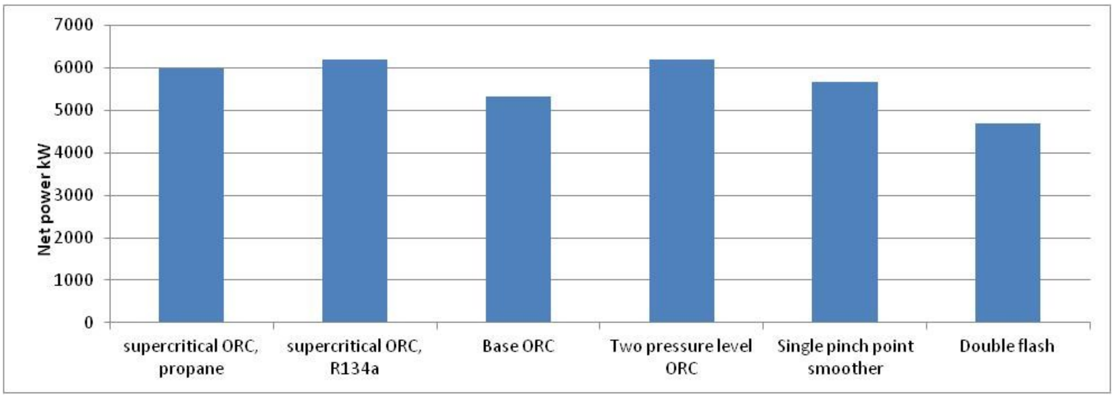

Figure 7: Net power generated at constant US.

The single Pinch Point Smoother cycle allows a higher performance than base ORC, however lower than the two pressure level ORC. The double flash cycle performance is the lowest: the heat introduced in the cycle is reduced, because of reduction of heat transfer surface, but the intrinsic dissipation process remains, and performance is therefore penalized.

It can be observed from figure 8 that, having fixed the same US value for all the cycles (obviously except the base ORC), the geothermal fluid discharge temperature is similar, i.e. the heat introduced in the cycle is almost the same in all cases: different performance is then obtained because of different cycle arrangement. Observing figure 8 it is also to be stressed that the Pinch Point Smoother cycle allows almost the same heat introduction process of the two pressure level cycle, but requiring a conventional turbine without the two admission ports feature. It can be also observed that, at the operating condition here considered, for the double flash cycle the specific heat is not really constant during all the heat introduction process.

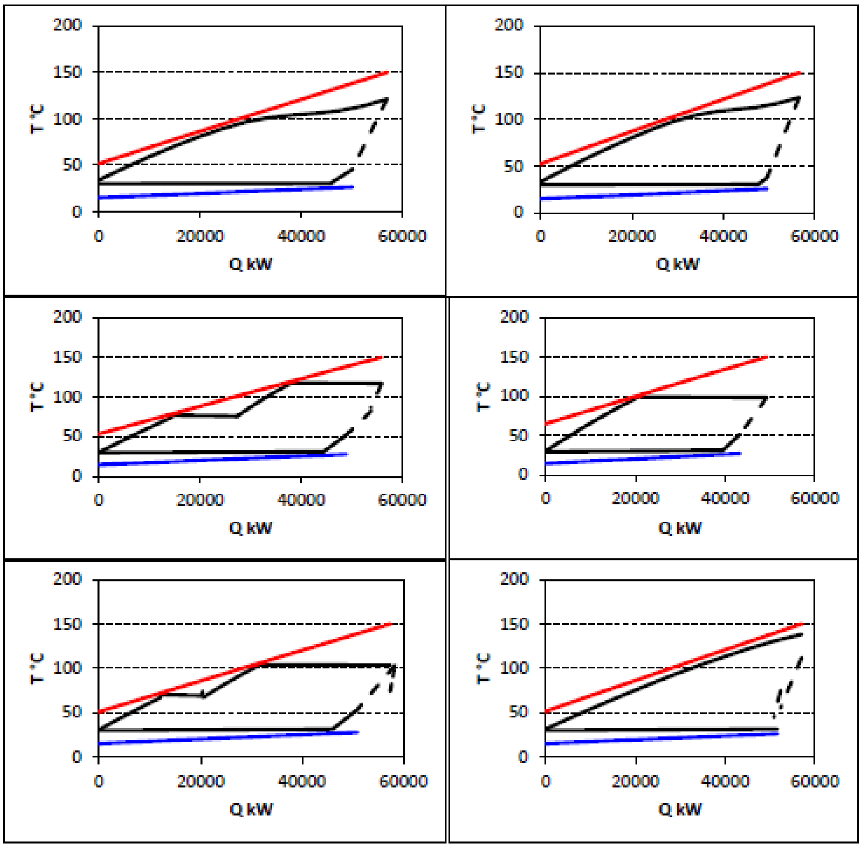

Figure 8: Temperature vs. power during heat introduction and heat rejection process for the cycles considered. Top, left: supercritical, propane. Top, right: supercritical, R134a. Center, left: two pressure level ORC. Center, right: base single pressure level ORC. Bottom, left: single pinch point smoother. Bottom, right: double flash.

Detailed information from calculation discussed in the previous figures is reported in table 4.

Table 4: Calculation results with constant US. ORC auxiliary electric consumption comprise the ORC pump electric consumption and the Pinch Point Smoother compressor electric consumption.

	Base ORC	**2 pressure level ORC**	**Single pinch point smoother**	**Double flash**	**Supercritical cycle, propane**	**Supercritical cycle, R134a**
Geothermal source temperature,°C	150	150	150	150	150	150
Turbine gross electric power, kW	6002	7069	7457	6136	8088	8060
ORC auxiliary electric consumption, kW	254	381	1289	931	1591	1370
Aircondenser electric consumption, kW	433	490	509	516	501	495

Net power, kW	5315	6198	5659	4688	5996	6195
Thermal power from geothermal source, MW	49,2	55,9	57,4	57,0	57,0	56,5
Plant efficiency, I law	10,80%	11,09%	9,87%	8,22%	10,53%	10,96%
Geothermal fluid discharge temperature, °C	65,3	53,6	51,1	51,7	51,8	52,6

Further calculations were conducted varying the temperature of the geothermal fluid, and considering 125°C and 140°C. Calculation results (fig. 9) show that, on the whole, the situation is unchanged: the best performance is obtainable with the two pressure level cycle, though the single pinch point smoother allows better performance than the base ORC; the double flash cycle still performs less than the base ORC.

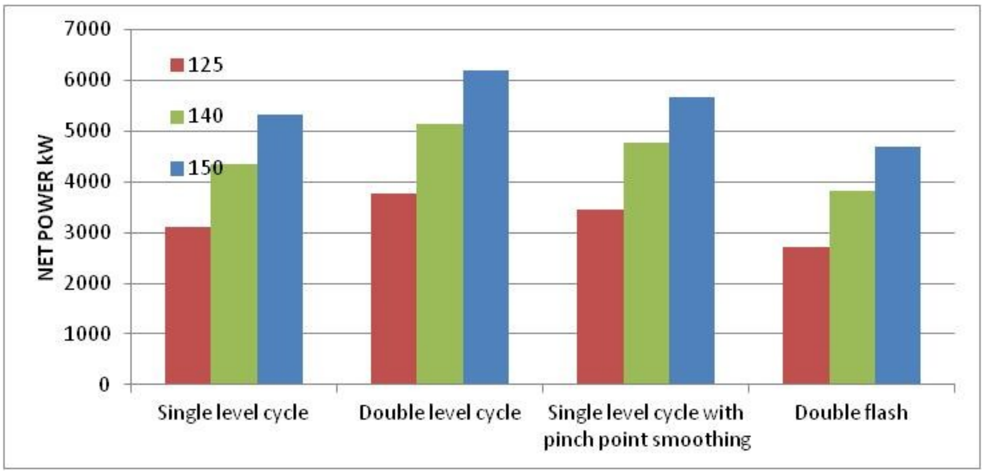

Figure 9: Net power generated at constant US for different values of the geothermal fluid temperature..

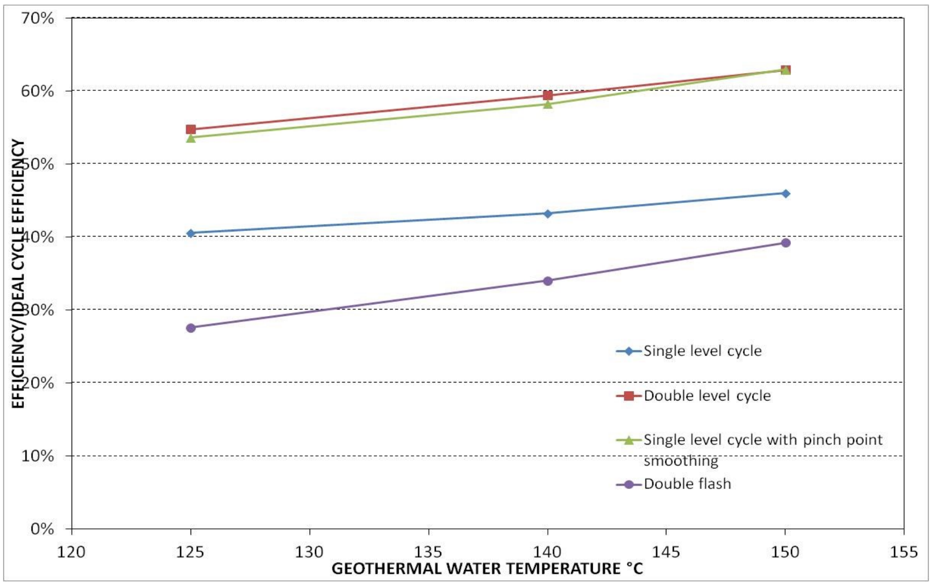

Figure 10 Second law efficiency at constant US for different values of the geothermal fluid temperature.

The second law efficiency is finally reported in figure 10 for the considered cases: both the two pressure level cycle and the Pinch Point Smoother cycle show remarkable efficiencies, definitely higher than the base ORC; on the contrary the flash cycle is strongly penalized by the dissipative flash process.

4. CONCLUSION

The calculation implemented is obviously limited by the fact of considering a single working fluid and a single value for ambient air temperature. However, though limited in scope and conducted on simple assumptions, the reported results allow to draw some conclusion, which should be compared on a larger working fluid selection base and more detailed plant analysis. These conclusions are here summarized:

- The two pressure level solution and the Pinch Point Smoother solution are comparable from the point of view of performance, giving a definite advantage compared to the base cycle, single pressure level ORC and represent therefore a potentially valuable solution for geothermal applications. The decision to adopt one or the other solution will possibly depend on the turbine frame actually available for the specific application (with or without an additional admission port, as required for the two pressure level plant). The Pinch Point Smoother solution could also be adopted for plant retrofit, so as to increase performance maintaining the existing turbine and enhancing the heat introduction process.

- The simple supercritical cycles provide an overall performance similar to the two pressure level cycle, however they are more demanding in terms of maximum cycle pressure (at turbine inlet) and they involve a much higher power for the feed pump.

- Apparently the double flash cycle solution, when restricted to conventional technology, excluding two-phase turbine, is not attractive for applications with HFC245fa; if adopted, it should be implemented with no restriction on the heat transfer surface. It could represent an attractive solution for specific cases, which would take advantage of the single phase primary heat exchanger, e.g. for very high pressure geothermal fluids like in geo-pressurised reservoirs. Also, geo-fluid with highly corrosive characteristics, or high salt content, could be better exploited thanks to a single phase primary heat exchanger.

Only a detailed case by case analysis, taking into account power block investment, time of delivery, reliability, O&M requirements, environmental impact, can ultimately lead to the preferable plant solution.

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