ENHANCED CASCADED CYCLE

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ABSTRACT

Two level pressure ORC cycles (also known as cascaded cycles) has been realized for 30 years as a solution to increase the conversion efficiency of large geothermal plants or, in other words, to increase the power output from a certain geothermal resource, by realizing a thermodynamic cycle that better couples with the cooling of the resource.

An improvement of this well-known cycle is proposed which allows to further increase the conversion efficiency by introducing an additional intermediate pressure level.

Two alternatives are presented to exploit the additionally produced vapor flow, showing the efficiency increase achievable for a given typical application and also reporting some considerations about the cost effectiveness of this solution.

1. INTRODUCTION

In heat recovery and geothermal applications, the adoption of an organic Rankine cycle has proven to be a feasible, efficient and economical solution compared to the traditional steam cycle, particularly when the temperature of the heat source is from medium to low (i.e. lower than 250 ° C) and with sources mainly in liquid or mixed liquid - vapor phase (Rossi di Schio *et al*. 2018).
In the case of heat source mainly in liquid phase (as often is the case of low-medium enthalpy geothermal applications), the introduction of heat into the thermodynamic cycle from the source occurs at a strongly variable temperature.
On the contrary, the transfer of condensation heat to the cooling source is mainly at a nearly constant (or slightly variable) temperature since the technical-economic optimization of the flow rate of the cooling fluid (both air and water) leads to the use of large flow rates and therefore small temperature differences.
A great interest has been devoted in optimizing the conversion efficiency of thermodynamic cycles suitable to exploit this kind of sources. In particular, the so-called 'two level pressure cycle' also often referred as 'cascaded cycle' has been proposed and realized for more than 30 years.
The reason for the advantage of this solution will be more evident by a comparison with other thermodynamic cycles.
The thermodynamics of the ORC cycles that better couples with variable temperature heat sources has been described in several books and papers, for example recently well summarized in Macchi and Astolfi (2017a) and in Macchi and Astolfi (2017b)

In Figure 1a to 1e, thermodynamic cycles associated with variable temperature heat sources are represented in the Temperature-Entropy diagram.
TH_in and TH_out indicate the input and output temperature of the hot source, respectively, while TC_in and TC_out indicate the input and output temperature of the cold source, respectively.

The cycles in Figure1a and 1b are ideal cycles since:
- the heat exchange with the sources occurs with zero temperature difference (corresponding to having an infinite surface of the heat exchanger).

- adiabatic compression and expansion transformations are ideal and therefore represented by 2 vertical segments (no increase in entropy)

Note:
1) for simplicity and for graphic requirements the dotted line that represents the working fluid heating is drawn slightly separated from the line representing the hot source cooling and not overlapped.
2) The hot and cold sources curves are represented with straight segments although in reality in the T-s plane said lines should be slightly curved

The ideal thermodynamic cycle that maximizes conversion efficiency is a trapezoidal cycle (Figure 1a) because it matches, better than the Carnot cycle (rectangular, Figure1b), with the variable temperature source and maximizes the achievable Work L (corresponding to the area of the cycle itself).
A particular kind of this trapezoidal cycle is the triangular cycle also called 'Lorentz cycle' (where TH_out approaches TC_out).
Both trapezoidal and triangular cycles are described, for example, in Bertani (2017).
Practically, the triangular cycle is rarely considered as a reference for real geothermal applications because of a frequent limitation in the minimum geothermal outlet temperature TH-OUT to avoid salts (present in the geothermal brine) to precipitate and scale the heat exchangers surfaces (Gunnarsson I., Arnórsson S., 2003).

A real organic cycle (Figure 1c) has a more or less favorable heat introduction curve depending on the critical temperature T_CR of the fluid adopted in relation to the temperature of the source. A super-critical cycle (Figure 1d) has potentially thermodynamic advantages with respect to subcritical cycles, as it better approaches the ideal trapezoidal cycle of Figure 1_a (Valdimarsson, 2014).
However, for matters related to a correct sizing of the machines, to avoid high pressures, or in any case to exploit other favorable characteristics of organic fluids, it is often preferred to adopt a scheme with multiple pressure levels (for example like the one in Figure 1e with two pressure levels) instead of a super-critical cycle.

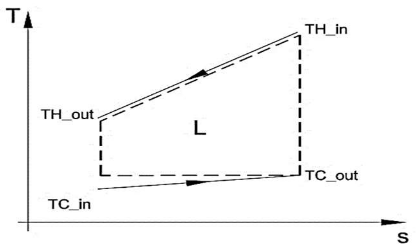

Figure 1_a: Ideal trapezoidal cycle

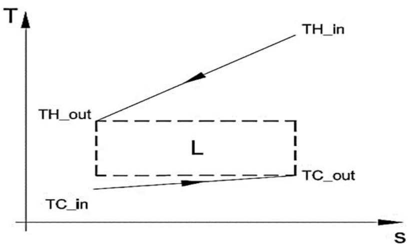

Figure 1_b: Ideal Carnot cycle

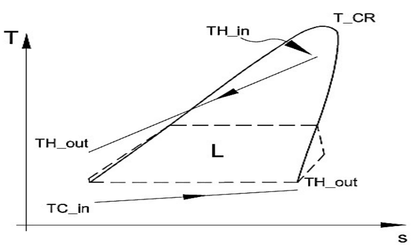

Figure 1_c: ORC sub-critical cycle

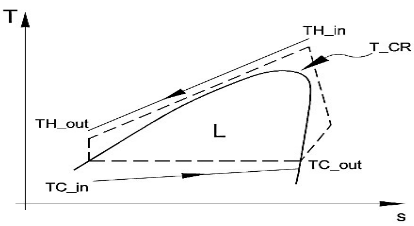

Figure 1_d: ORC super-critical cycle

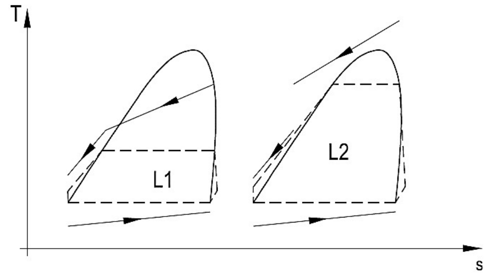

Figure 1_e: 'Traditional' two level pressure cycle

The proposed enhanced scheme is an improvement of the well-known two level pressure cycle.
The improvement allows to further increase the conversion efficiency with small increase of plant complexity compared to the adoption of, for example, a three level cycle.

2. DESCRIPTION OF THE ENHANCED SCHEME

The two level pressure or cascaded cycle allows to better exploit variable temperature sources compared to a simple, single level pressure cycle. In other words, the cascaded cycle uses a plurality of Rankine cycle modules, the source fluid being applied in series to the heat exchangers of each module in order to maximize the power produced by the system. Typically, in the case of two modules, they will be referred to as the "high temperature - HT" and "low temperature cycle - LT".
One of these schemes has been patented by the company Ormat (patent GB2162583A) in 1986. The T-s diagram is represented in Figure 1e while the circuit arrangement of both hot source and working fluid is reported in Figure2 (in both figures the hot source is represented by a solid line while the working fluid by a dotted line).

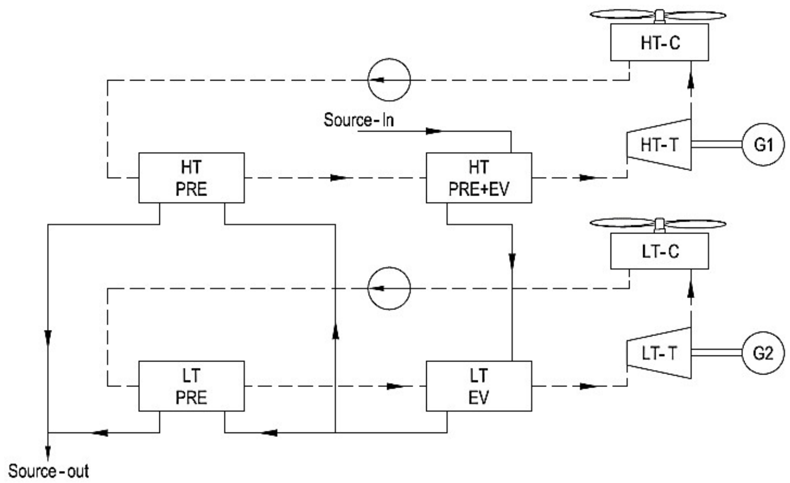

Figure 2: 'Traditional' two level pressure cycle – Circuits arrangement

In a two level pressure cycle, the hot source first supplies the high temperature cycle vaporizer (HT, PRE + EV). The high temperature vaporizer performs both a preheating of the organic fluid and its vaporization (and possibly also its overheating) and can be realized in a single vessel (as in Figure 2) or in two different ones (as in Figure4 heat exchangers #1 and #2). The hot source then passes through the low temperature cycle vaporizer (LT, EV), then it is divided into two flows that feed two partial preheaters of the high temperature (HT, PRE) and low temperature (LT, PRE) cycles.
As commented above, a solution to increase power is to extract more heat from the hot source by increasing the overall temperature drop of the source while at the same time keeping the vapor generation pressure/temperature that feeds the turbine (or the turbines), also reducing as much as possible the gap between the sources and the working fluid curves in the T-s diagram.
A cascaded system already fulfills this task (with respect to a single-level sub-critical cycle like the one shown in Figure 1_a) because it is better approaches to the ideal trapezoidal cycle of Figure 1_a.
The proposed scheme further increases the efficiency of a cascaded organic Rankine cycle with the aim to improve the economic feasibility of geothermal plants, often heavily penalized by high costs for the geothermal side (wells and infrastructures) and for which, therefore, an increase in electricity production is of significant help.
The new scheme consists of a cascade cycle with two evaporation levels where the first high temperature cycle also comprises an additional vaporizer operating at an intermediate pressure between the vaporizer pressure of the high temperature cycle and the pressure of the low pressure cycle

vaporizer. Said further vaporizer is fed by a partial flow of the hot source extracted downstream of the first vaporizer and upstream of a preheater of the same high temperature cycle.

Figure3 represents the thermodynamic cycle of the 'Enhanced' system, while Figure4 the layout scheme of the fluid circuits.

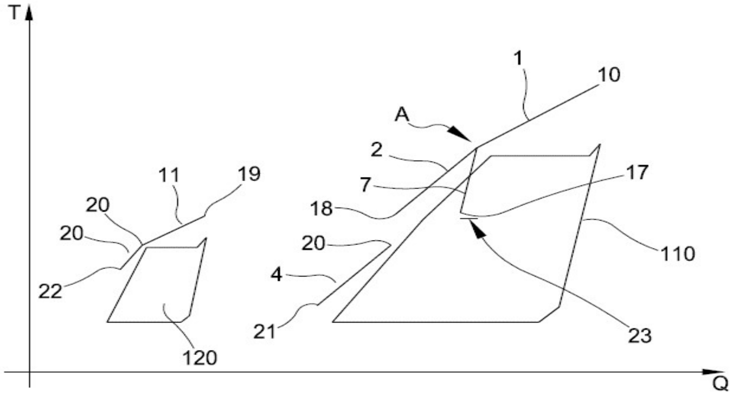

Figure 3: 'Enhanced' two level pressure cycle

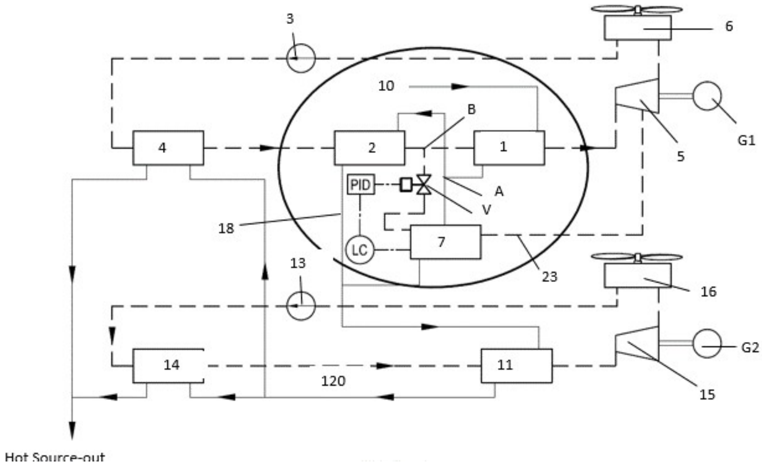

Figure 4: 'Enhanced' two level pressure cycle – circuits arrangement

The hot source 10 supplies the heat exchangers according to the following scheme: First, it passes through the first vaporizer 1 of the high temperature cycle, then by means of a branch identified by the point A in Figure 4, it partially and in parallel feeds the second preheater 2 and the further vaporizer 7. The outputs of the hot source from the preheater 2 and from the vaporizer 7 are then joined and the hot source 10 in its entirety passes through the low temperature vaporizer 11. Finally, the hot source 10 partially and in parallel feeds the first preheater 4 of the high temperature cycle and the preheater 14 of the low temperature cycle.

The further vaporizer 7 is fed by a partial flow of working fluid liquid extracted at the exit of the pre-heater 2 (point B in Figure4), throttled, by means of a suitable valve V, at the appropriate intermediate pressure of the vaporizer 7. This lamination will cause partial evaporation of the fluid and full evaporation will be obtained in the vaporizer 7.
The partial flow exits from the vaporizer 7 and feeds the high pressure turbine 5.
This intermediate pressure fluid can be used in the high pressure turbine 5 for two alternative functions:
a. to neutralize a loss of a labyrinth in the turbine
b. supply an intermediate turbine pressure stage

In both cases, the new scheme allows a not negligible increase in the performance of the plant, in terms of mechanical / electrical power, the magnitude of which is related to the actual design of the turbine, the thermodynamic cycle and the hot source characteristics.
It also represents a simple solution because it does not involve the adoption of an additional turbine (but only its modification) and only the addition of a heat exchanger and some controls.

In geothermal applications, in general, the heat exchangers used are of the shell&tube type with the hot geothermal source inside the tubes and the organic fluid on the outside of the tubes (shell side), in order to allow easy cleaning of the tubes (for example by brushing).
This type of heat exchanger can also be used for the further vaporizer 7 and to obtain adequate control of the system, it is possible to control the level of the liquid in the vaporizer 7 with a valve V. Said valve V allows to control the level of organic liquid entering the shell of vaporizer 7 by means of an 'LC' level meter. The 'LC' level meter operates the V valve via, for example, a PID logic control (Proportional Integral Derivative).

3. EVALUATION OF THE BENEFIT IN A REAL GEOTHERMAL CASE

3.1 Thermodynamic analysis

A more detailed analysis is now carried out referring to a real geothermal application having the following data:
Hot source: liquid water
Hot source inlet temperature: 155 C
Hot source flowrate: 317 kg/s

Ambient air temperature (for air cooled condensers): 9 C
Total air flow in the air cooled condensers: 8200 m^3/s

Assumed pump efficiency: 0.80
Assumed motor pump efficiency: 0.937

Assumed turbine efficiency: 0.89
Assumed generator efficiency: 0.975

Working fluid: iso-butane

a) ‘Traditional’ Two level cycle

This cycle has been optimized assuming certain dimensions for the heat exchangers, yielding to the thermodynamic cycle reported in the following Figure5 and the Process & Flow Diagram of Figure6:

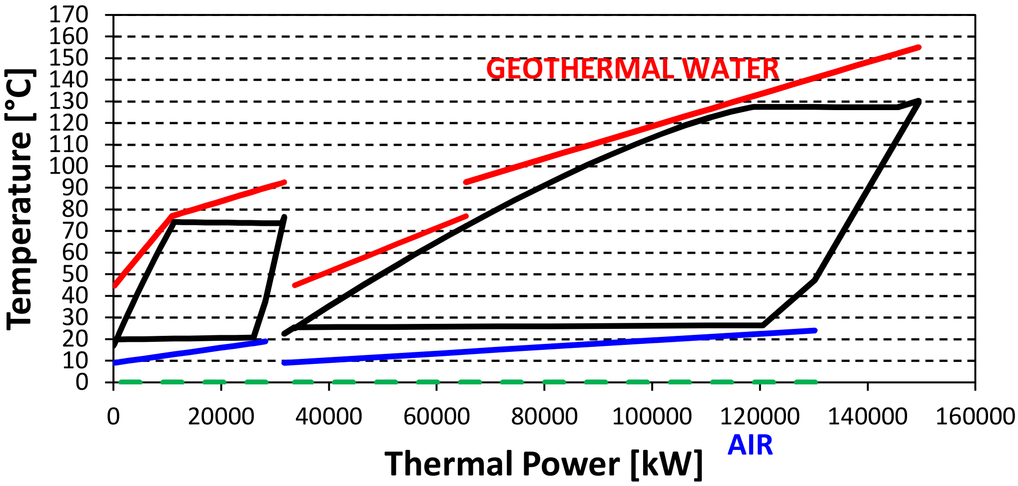

Figure 5: ‘Traditional’ two level cycle optimized for the given source

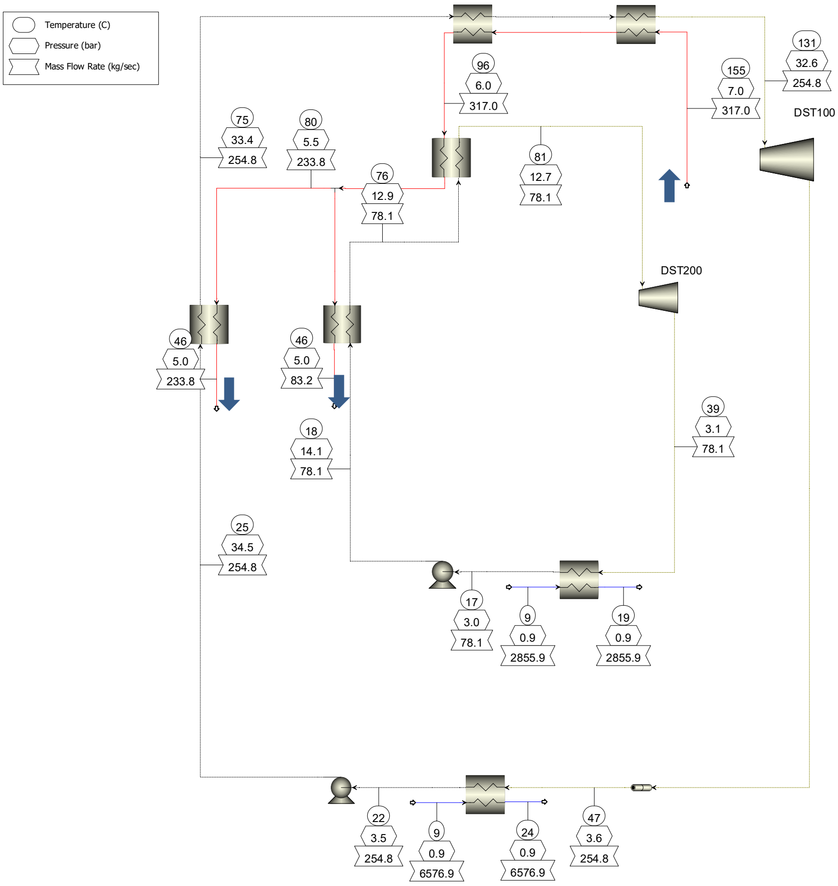

Figure 6: Process & Flow diagram for the 'Traditional' two level cycle optimized for the given source (solid line: water circuit, dotted line: working fluid circuit)

The two evaporation pressure are respectively 32 bar and 11.5 bar (corresponding to saturation temperatures of 127 C and 72.6 C respectively)

In this case the resulting output is:
'Traditional' two level cycle Gross power: 21677 kW
'Traditional' two level cycle Net power (net of feed pump consumption and air cooled condenser): 18671 kW

b) 'Enhanced' Two level cycle

According to the circuit already described in Figure 3 and 4, an additional vaporizer is inserted to produce the following stream:

Vapor pressure (saturated): 20.8 bar @ 102.4 C
Vapor flow rate: 17 kg/s

The heat exchangers in the 'traditional' two level cycle and in the 'enhanced' one have been kept with the same U*S (where U is the overall heat exchange coefficient in W/(m² * Delta_T) and S is the heat exchanger surface in m²). Considering the small variation in high temperature and low temperature heat exchangers operating conditions between the 'Traditional' and the 'Enhanced' schemes, it is correct to conclude that the overall heat exchange coefficient remains constant in the two cases and hence the heat exchanger surfaces are the same in the cases. This allows to compare more correctly the two solutions also from an economic point of view.

The resulting thermodynamic cycle is in Figure7 while the correspondent Process & Flow Diagram is in Figure 8.
For graphic clarity in Figure7, the additional vaporizer and relative expansion are represented at the side of the other diagrams, on the right.

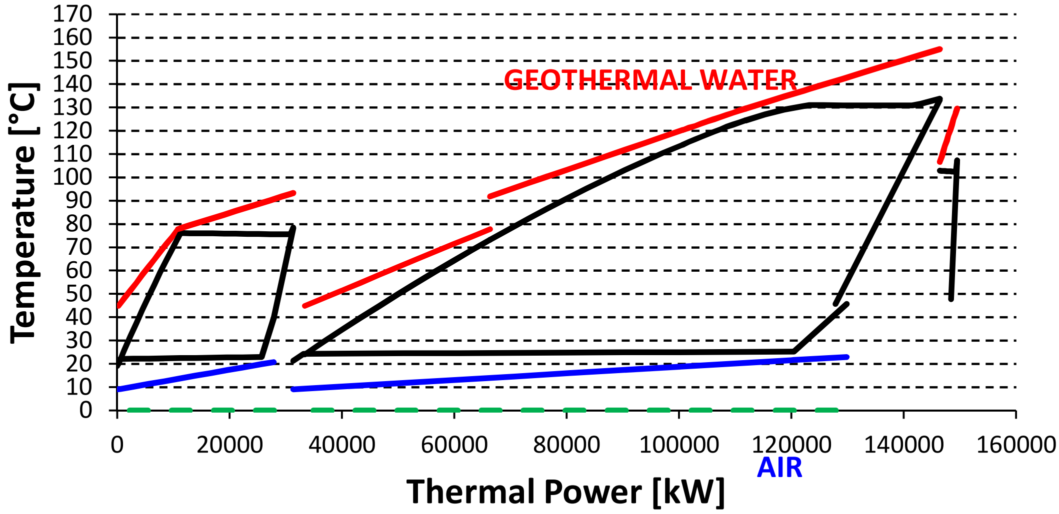

Figure 7: ‘Enhanced’ two level cycle optimized for the given source

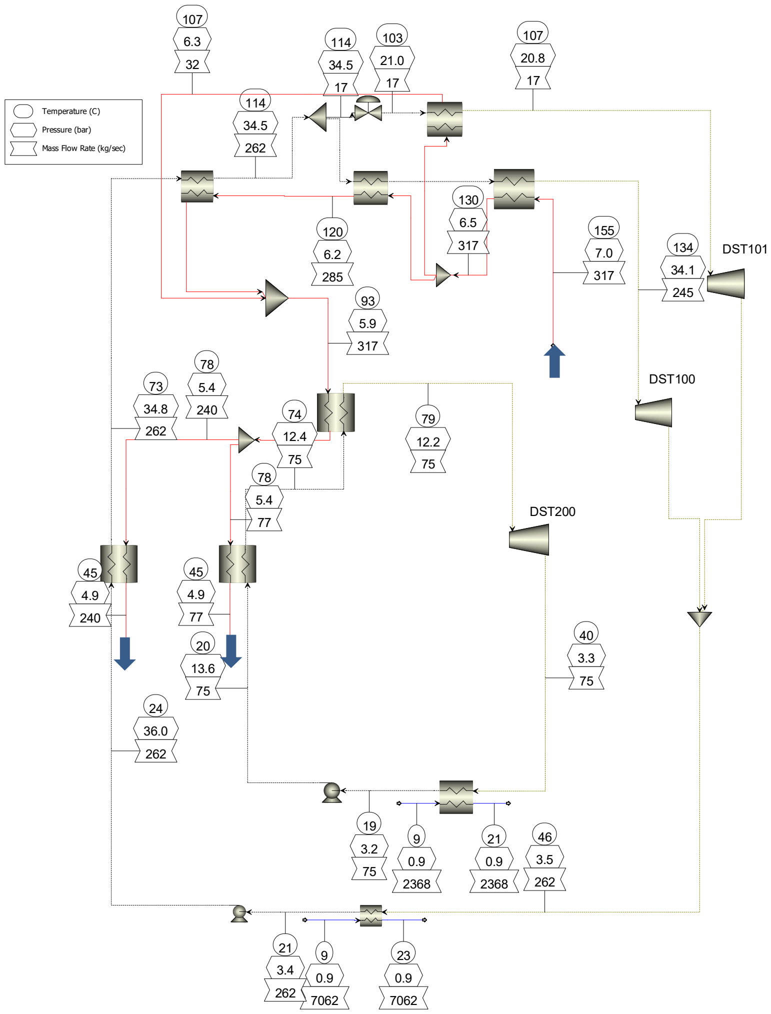

Figure 8: Process & Flow diagram for the 'Enhanced' two level cycle optimized for the given source (solid line: water circuit, dotted line: working fluid circuit)

Assuming to feed this stream in an intermediate turbine stage (see note*) and keeping the same overall turbine efficiency, the new output would be:

Note (*) In the cycle reported above for computational reasons the additional flow rate generated by the additional vaporizer is fed to a separate turbine (DST101), even if it would be more economically advantageous and mechanically feasible to feed a proper intermediate stage of the high pressure turbine instead.

Gross power: 21968 kW
Net power (net of feed pump consumption): 18831 kW

Hence, with the enhanced cycle the gross power output is increased of 291 kW i.e. of 1.3%, the net power output is increased of 160 kW i.e. of 0.85 %

Note: of course the same concept could be utilized in additional steps (vaporizers), providing additional evaporators to feed both intermediate pressure levels of the high pressure turbine and of the low pressure turbine

3.2 Cost effectiveness analysis

Though the performance increase is only 1.3% of the gross power and 0.85% of the original net power, the following considerations have to be taken into account for a correct evaluation of the proposed enhancement:

1. The increase of performance is strictly related to the specific case, i.e. with different feeding conditions the amount could change sensibly too
2. Also in the case considered the increase is not negligible at all

Here below some simplified considerations about the cost effectiveness of the solution are reported (a differential cost/income analysis is performed).

Assumptions:
Gross electric energy valorization: 0.25 Euro/kWh (considering, for example, a feed in tariff in Germany valid for geothermal plants and available for 20 years as per the actual EEG2017 Art 45.1 German law)
Electricity price for the auxiliaries: 0.135 Euro/kWh (average value for a typical German geothermal customer)

Number of hours of operation equivalent at full power: 8300 h/year

Gross electric power increase (see above calculations): 291 kW
Auxiliaries electric power increase (see above calculations): 160 kW

Resulting gross extra energy/year produced: 291 kW* 8300 h/year= 2,415,300 kWh/year
Resulting extra income/year from gross electricity production: 2,415,300 * 0.25=603,800 Euro/year

Resulting extra energy/year consumed for auxiliaries: 131 kW* 8300 h/year= 1,087,000 kWh/year
Resulting extra cost/year for auxiliaries consumption: 1,087,000 * 0.135=146,800 Euro/year

Net advantage (income) with the enhanced scheme: 603,800 – 146,800=457,000 Euro/year

Assumed extra cost to implement the solution:
Extra cost for the turbine: 120,000 Euro
Extra cost for the heat exchanger and piping: 190,000 Euro
Extra cost for controls, connection, software: 110,000 Euro
Extra cost to implement the enhancement (total): 420,000 Euro

From the above data, the simple pay back time of the extra investment is about 1 year and the IRR (Internal Rate of Return) over a period of 20 years is about 109%.
Assuming an actualization rate for the future cash-inflows of 3.5%, the Net Present Value considering the initial investment and the extra cash-inflows for a period of 20 years is 6.5 ME.

The following table summarizes the results in the investigated case study and under the assumed hypotheses , as a comparison between the 'Traditional' two level cycle and the 'Enhanced two level cycle':

Datum	*units*	*'Traditional' two level cycle*	*'Enhanced' two level cycle*
Geothermal water inlet temperature	°C	155	
Geothermal water flow	kg/s	317	
Inlet ambient temperature (for Air Cooled Condensers)	°C	9	
Total air flow in ACC	m^3/s	8200	
Turbine efficiency		0.89	
Pump efficiency		0.8	
Electric generator efficiency		0.975	
Electric motor efficiency		0.937	
ORC working fluid		Iso-butane	
Geothermal water outlet temperature	°C	46.0	45.0
Produced Gross Power	kW	21677	21968
Produced Net Power	kW	18671	18831
Pay back time (**)	year		~1
Internal Rate of Return (IRR) (**)	%		109
Net Present value (NPV) (**)	MEuro		6.5

Table 1 – Summary of the obtained results

Note (**): under the assumed economic hypotheses

4. CONCLUSIONS

The proposed two level pressure enhanced cycle, consists in the addition of an intermediate pressure vaporizer which produces and additional flow of vapor to be exploited in the high pressure turbine.

In a specific case referring to a geothermal application in Germany, the advantage has been calculated in 291 kW (i.e. 1.3% of the 'traditional' two level cycle) in terms of gross power, while in terms of net power, the net power output is increased of 160 kW i.e. of 0.85 %.
The additional investment is paid back in about 1 year and gives an IRR of 109% over a period of 20 years.

The additional power produces about 0.46 ME extra income/year, which actualized in 20 years of operation, at an actualization rate of 3.5%, gives a Net Present Value of about 6.5 ME.

Though the performance increase in percentage is small, it cannot considered negligible and surely improves the overall economics of the geothermal plant.

Additional increase could be obtained by adding other vaporizers to feed other intermediate stages both of the high pressure and of the low pressure turbines

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