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Field Performance Evaluation of ORC Geothermal Power Plants Using Radial Outflow Turbines

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Abstract

This paper, after a brief description of the radial outflow turbine and of its main features, discloses the field performances evaluation of two operating geothermal ORC (Organic Rankine Cycle) plants installed by Exergy Spa in Turkey. The work describes the test procedure, the measurements and calculation methods used to obtain the turbine efficiency as well as overall power cycle performance from the set of available experimental data.

Keywords: Geotermal Energy, double level ORC, experimental campaign, turbine efficiency

1. Introduction

In recent years the ORC market expanded rapidly in terms of installed plants, power and number of ORC manufactures that can provide nowadays a wide range of plant size from small (few kW) devices up to large (tens of MW) plants. New markets, like the waste heat recovery from industrial processes or endothermic engines, are growing very fast but the biomass combustion and the exploitation of geothermal brines have been confirmed as the two main fields of application of ORC technology [1]. Regarding the latter, the presence of monetary subsidies is making attractive the exploitation of geothermal brines with temperatures as low as 100°C, but it is mandatory to maximize the plant efficiency in order to payback the relevant cost of exploration and drilling. Efficiency of low-

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medium temperature geothermal plants can be enhanced: (i) by realizing a two evaporation level cycle able to reduce the irreversibilities in the heat introduction process, (ii) by lowering the condensation temperature and (iii) by increasing the turbine efficiency. Exergy Spa has recently installed a number of geothermal plants in Turkey where the three aforementioned precautions are considered in the system design. In addition, Exergy has introduced in the ORC market an interesting innovation by developing a radial outflow turbine that has several unique characteristics qualifying this unconventional configuration as advantageous for many ORC applications [1, 2, 3].
Compared to axial and centripetal turbines [1,3,4,5,6,7] radial outflow turbines can accommodate many stages on the same disk allowing for a lower stage loading and a high expansion efficiency without incurring in rotordynamics issues. Moreover, their particular design allows to easily handling very high volume flow ratio thanks to the increase of stage radius along the expansion: blade height of first stages increases while last stages height is limited with beneficial effects on expansion efficiency. Last interesting feature is a constant peripheral speed along the blade span thus allowing for the use of prismatic blades.

In spite of the rising interest in this configuration, nowadays only few experimental data about it have been presented [8]. This paper provides some experimental results about the performance of the radial outflow turbine in two different geothermal plants that are in operation from several thousands of hours (> 10.000 h).

Nomenclature and acronyms

η	efficiency
PH	pre-heater; it can be on High Pressure Cycle (PH_{HP}) or on Low Pressure Cycle (PH_{LP})
EVA	evaporator; it can be on High Pressure Cycle (EVA_{HP}) or on Low Pressure Cycle (EVA_{LP})
TUR	turbine; it can be on High Pressure Cycle (TUR_{HP}) or on Low Pressure Cycle (TUR_{LP})
REC	recuperator
COND	condenser
G	generator
P	pump

2. Plants description

Both the plants investigated in this paper have double evaporation levels cycles. Figure 1 depicts their layouts and the instrumentation installed. The first plant (named GREENECO) exploits a 140°C geothermal brine and produces around 13 MW of electrical power. It is designed with two independent saturated cycles working with isopentane at two different evaporation pressures. The high pressure cycle is recuperative and has two preheaters in series ($PH1_{HP}$ and $PH2_{HP}$) where the working fluid is heated cooling down the geothermal brine. On the contrary the low pressure cycle is non recuperative and has a single preheater (PH_{LP}). The geothermal brine firstly flows through the high pressure evaporator (EVA_{HP}), the high temperature preheater of the high pressure cycle ($PH2_{HP}$) and the low pressure evaporator (EVA_{LP}); after that it is split in two parallel streams flowing in the low temperature preheater of the high pressure cycle ($PH1_{HP}$) and the preheater of the low pressure cycle (PH_{LP}). Saturated vapor is expanded in both cycles by a radial outflow turbine (TUR_{HP} and TUR_{LP}), having 4 and 2 stages respectively. Both turbines are mounted on the same shaft connected to a single generator in a double-ended configuration. Both cycles are condensed by a loop of cooling water connected to wet cooling towers for the heat rejection to the ambient.

The second plant (named AKCA) is designed to produce 3.6 MW cooling a low temperature geothermal brine from 105°C down to 60°C. This plant fully exploits the capability of a 3 stages radial outflow turbine having a double admission: this configuration allows for a clear advantage compared to an axial configuration because it does not require two separate turbines and all the stages are mounted on the same disk. R245fa is the working fluid. It is firstly pumped up to the maximum cycle pressure, then after a first preheating (PH1) the mass flow rate is split in two streams, one is throttled down to the low pressure and evaporates (EVA_{LP}), the other is further heated in a high temperature preheater (PH2) and then it evaporates at high pressure (EVA_{HP}). A fraction of the geothermal brine is derived just before the reinjection and it is heated up by a stream of a CO_2 rich geothermal steam and then mixed before the low temperature preheater (PH1). The cycle is water cooled condensed.

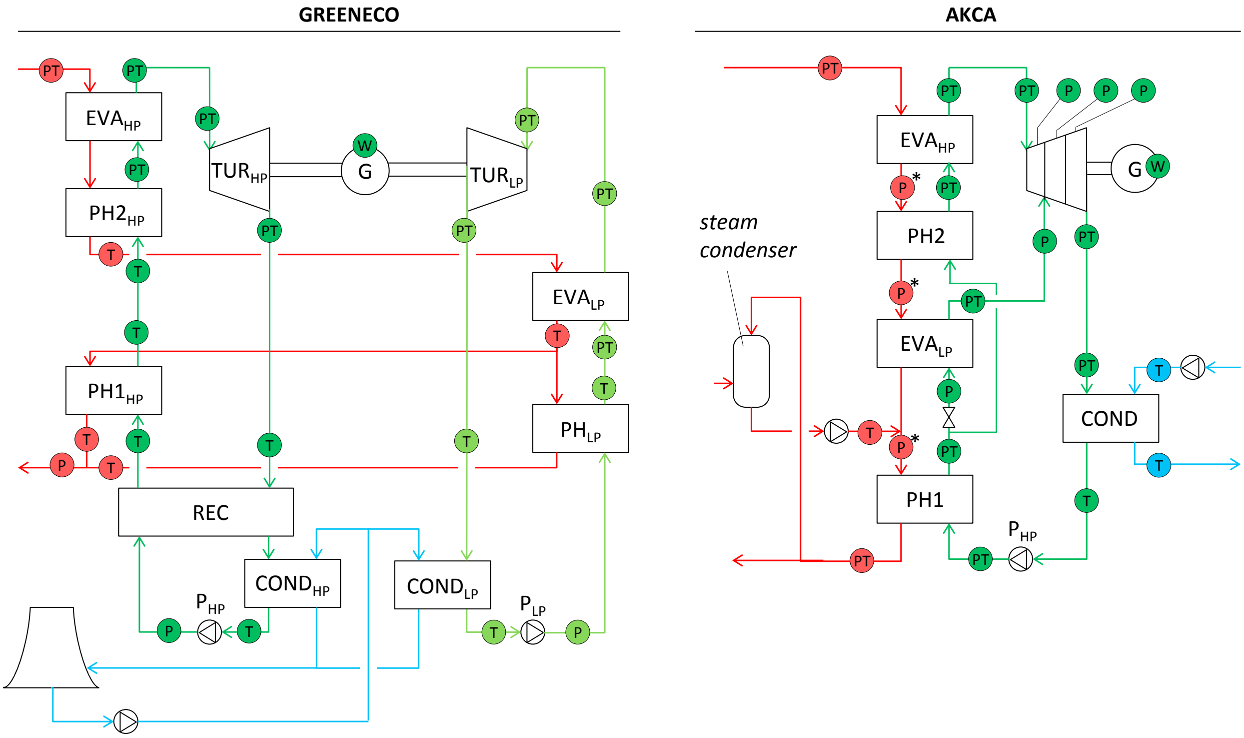

Figure1. Plant schemes of GREENECO plant (left) and of the AKCA plant (right) with information about the field instrumentation. P and T indicate the point of measurements of pressure and temperature respectively. Where symbol (*) is present an additional PT100 temperature instrument has been installed.

Figure 2 depicts the aerial view of the two plants: in both cases preheaters are Shell&Tubes heat exchangers with organic fluid flowing in the shell side, while evaporators are kettle reboilers. For AKCA plant a demister is placed on the top of each evaporator to remove liquid droplets that can be drag by the vapour flow, while in GREENECO plant the demister is included in the kettle reboiler and the tube bundle in the evaporator is not totally submerged possibly leading to a small vapor superheating. Condensers are common Shell&Tubes with cooling water in the tubes while the recuperator is arranged as a bundle of finned tubes enclosed in a vessel.

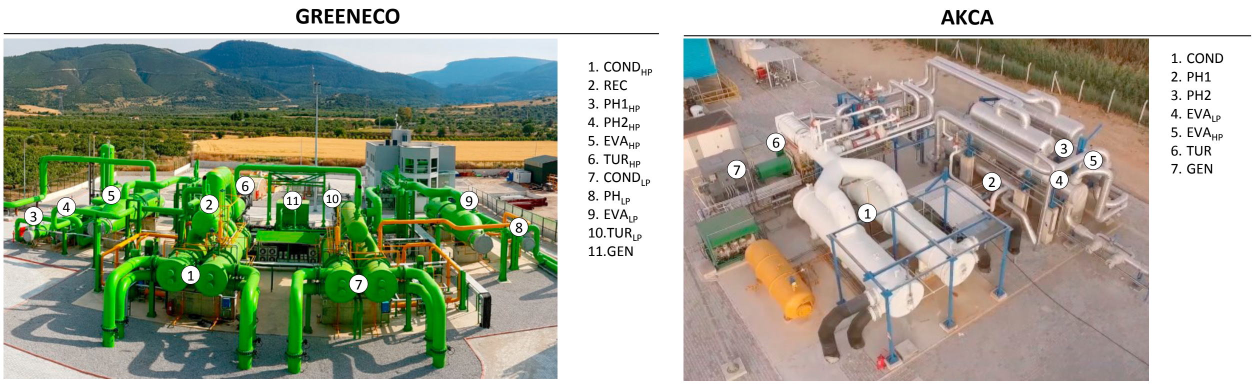

Figure 2. Aerial view of GREENECO (left) and AKCA (right) plants. Component PH2 for AKCA plant is not visible since it is placed below EVA$_{HP}$

3. Experimental campaign and data set definition

As shown by Figure 1, both plants are provided by an adequate field instrumentation for temperature and pressure measurements since several standard precision probes are installed on both brine and organic fluid streams. Thermodynamic state can be evaluated for most of the streams and energy balances can be verified for most of the components. In addition, a high accuracy measure of produced electrical power is available. Unfortunately, both plants lack in reliable measurements of brine, working fluid and cooling water mass flow rate because of poor instruments calibration. Finally, some measures are redundant while other ones are missing as reported in Table 1.

Table 1. Redundant and missing field measurements in GREENECO and AKCA plants: (i) inlet section, (o) outlet section.

<table>
<tr><th colspan="4">GREENECO</th><th colspan="4">AKCA</th></tr>
<tr><td colspan="8">Redundant measurements</td></tr>
<tr><td>stream</td><td>quantity</td><td colspan="2">point of measure</td><td>stream</td><td>quantity</td><td colspan="2">point of measure</td></tr>
<tr><td>HP fluid</td><td>T</td><td>$PH1_{HP}$ (o)</td><td>$PH2_{HP}$ (i)</td><td>fluid</td><td>p, T</td><td>EVA_{HP} (o)</td><td>TUR (i)</td></tr>
<tr><td>HP fluid</td><td>p, T</td><td>EVA_{HP} (o)</td><td>TUR_{HP} (i)</td><td>fluid</td><td>T</td><td>TUR (o)</td><td>COND (i)</td></tr>
<tr><td>HP fluid</td><td>T</td><td>TUR_{HP} (o)</td><td>REC(i)</td><td>fluid</td><td>P</td><td>TUR (o)</td><td>COND (i)</td></tr>
<tr><td>LP fluid</td><td>T</td><td>PH_{LP} (o)</td><td>EVA_{LP} (i)</td><td></td><td></td><td></td><td></td></tr>
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<tr><td colspan="8">Missing field measurements</td></tr>
<tr><td>stream</td><td>quantity</td><td colspan="2">point of measure</td><td>stream</td><td>quantity</td><td colspan="2">point of measure</td></tr>
<tr><td>HP fluid</td><td>p</td><td colspan="2">$PH1_{HP}$ (i), $PH2_{HP}$ (i), $COND_{HP}$ (i)</td><td rowspan="3">Brine</td><td rowspan="3">T</td><td colspan="2" rowspan="3">EVA_{HP} (o)*, PH2 (o)*, EVA_{LP} (o), PH1 (i)*,</td></tr>
<tr><td>HP fluid</td><td>T</td><td colspan="2">$COND_{HP}$ (i), P_{HP} (o)</td></tr>
<tr><td>LP fluid</td><td>T</td><td colspan="2">$COND_{LP}$ (i), P_{LP} (o)</td></tr>
<tr><td>Brine</td><td>p</td><td colspan="2">EVA_{HP} (o), $PH2_{HP}$ (o), EVA_{LP} (o), $PH1_{HP}$ (i), PH_{LP} (i)</td><td colspan="4" rowspan="2">* for this measure a PT100 temperature probe is installed</td></tr>
<tr><td>Brine</td><td>T</td><td colspan="2">EVA_{HP} (o)</td></tr>
</table>

The lack of field measurements about brine temperatures for AKCA plant required the installation of three additional high precision PT100 instruments to better describe the temperature heat introduction process: high precision probes were installed at EVA_{HP} and PH2 outlet and PH1 inlet while it was not possible to measure the temperature at EVA_{LP} outlet because of the absence of the thermowell in the piping. Moreover a set of high precision pressure instruments has been used to check the accuracy of field instruments. Table 2 reports the expanded accuracy of the field and the additional instrumentation used. All the pressure transducers are locally mounted with a small derivation in order to dissipate vapor heat: possible liquid heads due to vapor condensation are avoided positioning the instrument above the measuring point. Effect of ambient temperature on zero signal and span are negligible (ambient temperature is considered constant in the range 18-30 °C). Field instruments are piezo-resistive while higher precision instruments employ inductive transducers. Temperature sensors are located downstream of pressure taps into standard thermowells, which length is equal to pipe radius. Fluid velocity is below 10 m/s and the difference between static and total temperature is almost negligible at all times.

Table 2. Characteristics of the used instruments.

measured quantity	Standard Field Instrumentation		Higher Precision Instrumentation in AKCA	
	Sensor Type	*Expanded uncertainty Confidence level 95.5%*	*Sensor Type*	*Expanded uncertainty Confidence level 95.5%*
Temperature	PT100	0.9°C (at 150°C)	PT100	0.5°C (at 150°C)
Pressure	Absolute Pressure Sensor (0-10 bar)	0.15% f.s.	Absolute Pressure Sensor (0-4 bar)	0.05% f.s.
Pressure	Absolute Pressure Sensor (0-40 bar)	0.15% f.s.	Absolute Pressure Sensor (0-10 bar)	0.07% f.s.

3.1. Experimental data sampling

For both plants, a 20 minutes interval in stable condition has been selected and data are sampled every second for GREENECO plant and every 5 seconds for AKCA plant.

Figure 3 depicts the trend of some quantities during the experimental campaign in GREENECO and AKCA plants. For the latter there is also a comparison between data from field instrumentation and from the instruments with higher accuracy, since during AKCA experimental campaign different instruments having higher precision have been used for measuring the same quantities, in order to expand the statistical samples and to reduce possible systematic errors of measurement. Moreover, for AKCA plant, the energy production in the sampling period has been checked comparing the measurements from a field watt-meter and by an energy-meter at grid input point. The first measure is directly available at the plant control system while the second one is displayed on a dedicated screen. Both measures are collected for approximately one hour and the overall energy is compared. The relative difference is 0.274% showing a high reliability of the power output measured by field instrument.

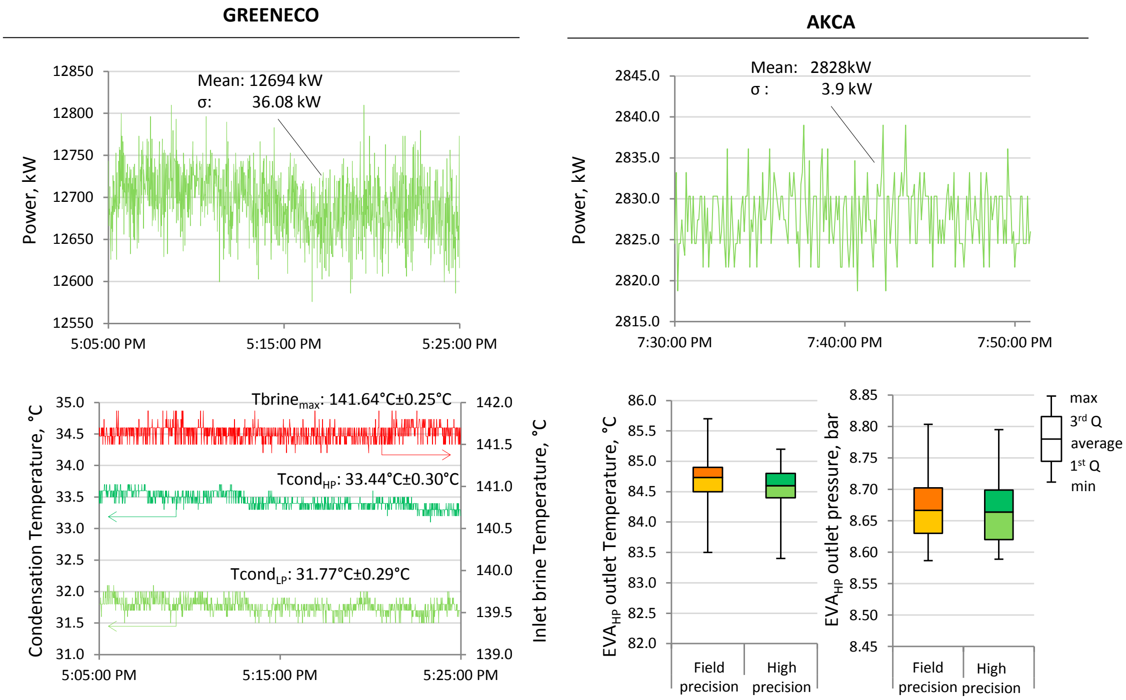

Figure 3. Trend of data set collected in the experimental campaign. GREENECO is close to nominal point while for AKCA plant power production is lower than the nominal because of a lower brine mass flow rate and temperature.

3.2. Data set definition and preliminary check

For the calculations the average value of each measured quantity is used. Moreover for some redundant measurements the mean value between the two reference ones is used: this is the case of the discharge temperature from the turbines in GREENECO plant and the temperatures between $PH1_{HP}$ and $PH2_{HP}$ and PH_{LP} and EVA_{LP}.

First, a consistency test is performed between the experimental data and the results of the state-of-the-art Equation of State implemented in Refprop 9.1 for the working fluids. The thermodynamic state of the fluid at evaporators outlet can be checked calculating the evaporation temperature from the value of measured pressure and comparing it with the measured value of temperature. In GREENECO plant it is found a small superheating at both high pressure and low pressure evaporators outlet, while for AKCA plant no superheating is observed at both levels, in accordance with the use of a demister. Same check is done at condenser hotwell comparing the measured temperature with the saturation temperature at condenser inlet pressure (in case of the GREENECO high pressure plant we assumed a pressure drop on recuperator hot side equal to 5%). In all the cases small subcooling temperatures were found. Since the liquid level in the condensers has been measured below the tube bundle we concluded that the “apparent” subcooling is due to the presence of non-condensable gas in the condenser. However, this fact t does not affect the methodology and the results that are obtained.

4. Methodology

Both for GREENECO and AKCA case, the lack of some point of measurement and the absence of direct measures of mass flow rate do not allow to close completely the energy balance for whole plant. For this reason, a number of assumptions and the use of additional data related to turbine and heat exchangers design are required in order to check the consistency of the experimental dataset and to calculate the performance of the components. The methodology adopted depends on the available measures and the plant layout thus it is not possible to define a unique approach. All the calculations are carried out assuming pure fluids and using Refprop 9.1 [9] for the calculation of fluid thermodynamic properties.

4.1. GREENECO plant

For GREENECO plant the following procedure is used:

1. Missing values of brine pressure are evaluated assuming an homogeneous repartition of the overall pressure drop among the heat exchangers, allowing to calculate brine enthalpy without affecting the results since brine is a subcooled liquid. Same approach is followed for HP fluid where the total pressure drop between pump outlet and evaporator inlet is assumed equally divided between REC, $PH1_{HP}$ and $PH2_{HP}$. Organic fluid pump efficiency is set equal to 0.75 and we verified that the effect of different assumed values is negligible on the overall energy balance. We verified that varying the value of pump efficiency by ±10% leads to variation lower than 0.15% on the overall heat balance with a small effect on final results.
2. HP fluid mass flow rate is estimated from the TUR_{HP} inlet conditions and considering chocked flow in the outlet section of the 1st stator vanes. Geometrical data of turbine (row radius, number of blades, gauging ratio, fillet factor) were measured by Exergy during turbine manufacturing. A coefficient equal to 0.95 is assumed to take into account stator efficiency and boundary layer blockage. The calculated value of mass flow rate is in good agreement with the result of an independent 3D CFD calculation made by Exergy on the same turbine.
3. The same approach is followed for TUR_{LP} component finding low pressure mass flow rate.
4. The overall energy balance of EVA_{HP} and $PH2_{HP}$ components is imposed since the measure of brine temperature between these two components is not available: neglecting heat losses a first value of the brine mass flow rate is calculated. Successively energy balance is evaluated for only EVA_{HP} finding the brine temperature at evaporator pinch point.
5. Energy balance of EVA_{LP} is imposed finding a second value of brine mass flow rate.
6. Energy balances of $PH1_{HP}$ e PH_{LP} are imposed and a third value of brine mass flow rate is obtained.
7. Turbine shaft power is computed by two alternate methods: (i) from electrical power output of generator considering shaft mechanical losses measured by Exergy in test bench and generator efficiency provided by generator manufacturer as function of power output and (ii) from the product of mass flow rate and enthalpy drop across turbine.

In order to obtain the verification of the energy balance of each component, the thermodynamic values are slightly modified with the aim to minimize the differences among the three calculated mass flow rates and the difference between the calculated and the measured power output. It is found that changing brine and isopentane temperatures within a range of ±0.25°C (a value lower than the instrument expanded uncertainty) it is possible to reduce all the above mentioned differences down to less than 0.1%.

The high pressure turbine mass flow rate is 145.5 kg/s with a volume ratio equal to 6.54, while for the low pressure turbine these quantities are respectively 129.2 kg/s and 3.03. Turbine total-to-static efficiencies are equal to 85.5% and 88.4% for high and low pressure machine respectively.

A sensitivity analysis is finally carried out for GREENECO plant with the aim at giving a confidence interval on the mean value for the calculated turbine efficiencies. The procedure is repeated assuming the turbine discharge temperature equal alternatively to the higher and to the lower measured value. The energy balance of the system is closed with a lower accuracy or it entails a more marked modification of brine temperatures but errors are below 1.5%. The maximum efficiency of TUR_{HP} and TUR_{LP} are respectively equal to 86.4% and 88.65% while the lower ones are 84.3% and 88.1% respectively. GREENECO plant efficiency results equal to 11% that corresponds to a 76% second law efficiency calculated respect to the trapezoidal Lorenz cycle working between brine temperatures and condensation temperature.

4.2. AKCA plant

For AKCA plant, it is not possible to calculate the turbine efficiency directly from measures since the machine has two inlets and the temperature between high pressure and low pressure stages is not measured. We decide to assume a single value for both high pressure and low pressure stages as representative of the whole machine in order to close the energy balances and to calculate the turbine outlet conditions. The following procedure is used:

1. The high pressure mass flow rate is calculated as previously explained for GREENECO plant (point 2) knowing the inlet conditions and the geometrical data of the first stator.
2. Thanks to the use of additional high precision temperature probes, the energy balance of EVA_{HP} component is imposed and the brine mass flow rate is calculated.
3. Energy balance of PH2 is evaluated and the error for the heat balance is calculated.
4. A tentative value of the efficiency of the first turbine stage is assumed to calculate the thermodynamic state of the discharged flow from first stage. Since brine temperature at EVA_{LP} outlet is not measured because the lack of the thermowell in the piping a pinch point temperature difference (ΔT_{pp}) is assumed and a first value of low pressure mass flow rate is calculated through the energy balance of EVA_{LP}. The mixing between the low pressure vapor from EVA_{LP} and the fluid expanded from turbine first stage allows calculating the second stage inlet thermodynamic conditions. Total mass flow rate expanded by low pressure stages is calculated as in the previous cases allowing to calculate a second value of low pressure mass flow rate. The two values are compared and the error is set to zero by varying the assumption on ΔT_{pp}.
5. The mass and energy balance of brine mixing process after EVA_{LP} is imposed allowing to calculate the mass flow rate of the brine coming from the steam condenser and the mass flow rate of brine entering in the PH1 component.
6. Energy balance of PH1 is evaluated and the error for the heat balance is calculated
7. The two measured values of turbine outlet temperature differ by around 2°C leading to two very different turbine isentropic efficiencies. For this reason the turbine outlet temperature is calculated using a value of isentropic efficiency equal the tentative value assumed for the high pressure stage and eventually compared the calculated outlet temperature with the measured values. Generator electrical power output is then computed considering mechanical losses measured by Exergy in test bench and generator efficiency provided by generator manufacturer. The error between the calculated and the measured values of power output is finally calculated.

Three errors are calculated: two for the energy balance of PH1 and PH2 components and the third between the calculated and the measured electrical power. These errors are minimized by changing the assumed value for turbine efficiency. It is found that with an efficiency of around 92% the all the errors are minimized: the turbine outlet temperature calculated with this value is between the two measured values, confirming the consistency of experimental dataset. The efficiency value in this case is particularly high but the result is affected by a high uncertainty due to the lack of consistency between the two measured values of temperature at turbine discharge. Using these values leads to turbine efficiencies equal to 88.8% and 96.1% but also to a larger errors on energy balances.

5. Results and discussion

For both plants the experimental data set consistency is checked verifying the energy balance of each component. It is found that in both cases the experimental data set is consistent and it allows calculating the missing values by means of the energy balance for each component in the plant and of the geometrical data for the turbines 1st stator. Figure 4 depicts the temperature-heat diagrams for the heat introduction process in both plants with the repartition of the heat introduced between the different components. Very low pinch point temperatures differences are obtained highlighting the choice to adopt very large heat exchanger surface in order to increase system efficiency.

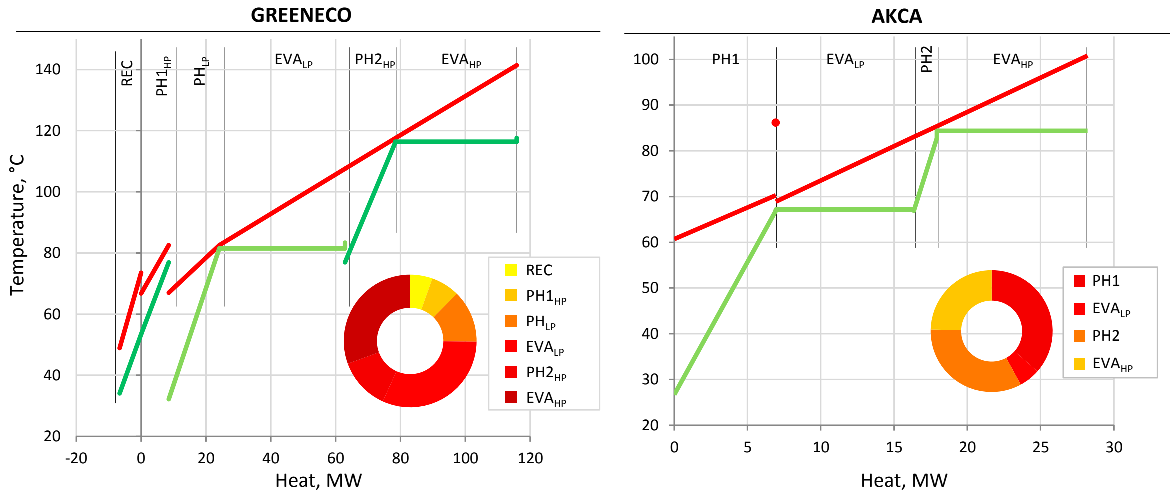

Figure 4. T-Q diagrams for the heat introduction process for GREENECO and AKCA plants. The red dot in Akca plant represent the temperature of the brine stream coming from the steam condenser.

6. Conclusions

The turbine isentropic efficiency could be simply computed from values of inlet and outlet thermodynamic conditions of the working fluid. However, this procedure cannot be applied for steam turbines, since the expansion usually ends inside the saturation line and is not possible to measure the steam quality at outlet conditions. Also in open cycle gas turbines the procedure is not possible, since inlet turbine temperature are too high for temperature probes and the expansion is far from adiabatic, being strongly cooled. In ORC turbines the procedure is possible, but the small temperature drop across the turbine can leave doubts about the precision of the calculated value. With the procedure adopted in this paper, the found values of isentropic efficiency are warranted on the basis of the validation of a large set of experimental data, including direct electrical power measurement.

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