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Steady state measurements and dynamic behaviour of an absorption heat transformer operating in an industrial environment

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Abstract

An Absorption Heat Transformer or Type II Absorption Heat Pump is a thermally driven heat pump able to upgrade approximately half of its driving heat up to a higher temperature level with an almost neglectable consumption of electric power. In the frame of a European project a 200 kW industrial pilot heat pump has been designed and installed in the power plant of a petrochemical facility. This work presents operation data of this AHT prototype operating at industrial scale. The presented results include steady state as well as dynamic performance measurements. The pilot AHT had continuously delivered heat flow rates between 220 and 270 kW with a thermal COP of around 0.5. The comparison of the monitoring results with those of a laboratory prototype present a deviation in the performance of around 20 % in terms of usable heat flow rates, but some explanation for this deviation are presented based on the operating conditions. The use of a dedicated adiabatic absorption chamber on the other side, partially compensate this deviation in performance, confirming the advantages of using an absorber with this configuration.

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1. Introduction

An absorption heat transformer (AHT), also called type II or booster absorption heat pump is a device able to split a heat flow at an intermediate temperature level in two heat flows, one at a higher temperature level (upgraded heat flow) and another one at a lower temperature level (rejection heat flow).

These kind of thermal heat pumps are especially suitable for the recovery and upgrade of waste heat from industrial processes. For the European industry, which accounts for the fourth part of the energy consumption in Europe, approximately 70 % of its energy demand is used for thermal purposes, and a third part of this is directly rejected, being the largest part of this waste heat rejected at low-temperatures below 100 ºC.   The Absorption Heat Transformers (AHT) is a ready-to-market technology that can upgrade these large quantities of available low-temperature waste-heat into a usable temperature level with a very reduced electrical consumption.

Applications reported in the literature concentrate in the pulp and paper (during and lamination processes), chemical and petrochemical (distillation and purification processes) and food and beverage sectors (drying, sterilization, fermentation and mold and bacteria processes) [1–4]. The literature reports present installations with capacities between 100 kW and 5 MW, and agree that AHTs presents attractive payback periods, between 2 and 5 years, depending mainly on their capacity. Furthermore, over the past years some market installations by the manufacturers Thermax and Johnson Controls – Hitachi, offering AHTs between 0.5 and 10 MW, have been documented in their respective websites. Despite these recent advancements, there remains a lack of

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knowledge about the integration and performance of such systems in the industry. Furthermore, there is a noticeable lack of open experimental data available regarding the performance of these systems.

Within the Indus3Es Project (Horizon 2020 Project No. 680738) a single stage AHT has been developed, focusing on its design and on its integration into existing plants and its adaptability into various industrial processes. The developed system is being under demonstration in a real environment at the Izmit facilities of Tüpras, the main petrochemical industry in Turkey, enabling to analyze integration aspects, as well as operational and business-related issues. This work presents the monitored experimental performance data of the pilot scale 200 kW AHT and its comparison with previously presented operational results under laboratory conditions.

2. Description of the Heat Upgrade System based on an AHT

A heat upgrade system including a 200 kW nominal capacity AHT has been installed at the power plant of the petrochemical facilities of Tüpras in Izmit (Turkey) in the final phase of the Indus3Es project. The waste-heat that drives the absorption cycle in the AHT system is the condensation heat obtained from an oily steam drum tank, that was released to the ambient before the installation of the Indus3Es system.

The heat upgrade system combines direct recovery of part of the condensation heat of the oily steam in a plate heat exchanger (HX E-7 in Figure 1) with recovery and upgrade of the remaining part of the condensation heat of the oily steam by means of the AHT.

The demineralized water feeding the boilers of the steam cycle of the power plant enters the heat upgrade system with a temperature that is relatively constant throughout the year at 65 °C. The demineralized water is first preheated to a temperature of around 95 ºC at HX E-7, and then further heated up at the heat exchanger HX E-5 by the hot glycol-water stream in the absorber circuit at the temperature level T2.

The heat released by the condensation of part of the oily steam at the heat exchanger HX E-6 heats up the water-glycol stream in the generator/evaporator circuit up to 95 ºC, that enters the generator and evaporator of the AHT driving the cycle. The AHT splits this driving heat flow into a heat flow at the heat rejection circuit at temperature level T0 connected to the condenser, and an upgraded heat flow at the temperature level T2 that is transferred by the absorber circuit between the AHT and the heat exchanger HX E-6. The water-glycol in this absorber circuit is heated up to 140 ºC by the AHT and heats up the demin-water circuit up to 135 ºC before it is sent to the boilers. The temperature of the cooling water entering the condenser at the heat rejection circuit is obtained from the cooling water network of the plant and have a nominal temperature of 25 ºC.

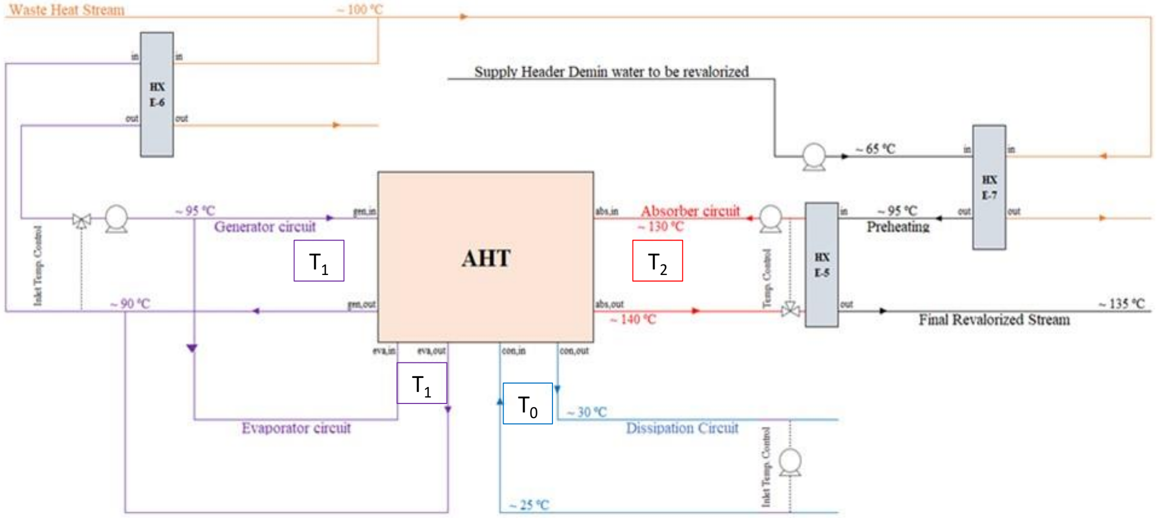

Fig. 1. Scheme of the AHT heat upgrade installation

3. Steady State Measurements of the 250 kW Industrial AHT

3.1. Operational Results

From November 2019 to Mai 2020 (end of the Indus3Es Project), the heat upgrade system has been in operation and monitoring data of the system under different sets of conditions have been analyzed, with the objective of the characterization of the AHT.

The heat upgrade system has been controlled with a dedicated control system developed within the project based on the characteristic equation method with the goal of maintaining the temperature of the demineralized water sent to the boilers at a constant value, that has been changed along the monitoring phase.

A set of valves controlled by the system act on the three circuits connected to the AHT at the temperature levels T_0, T_1 and T_2 according to an internal logic that reduced the use of cooling water from the cooling water network by recirculation at the heat upgrade circuit. The temperature at the circuit T_1 is kept as high as possible to make the best possible use of the waste heat. The pumps at the three circuits T_0, T_1 and T_2 are equipped with frequency variable drivers to control the flow rate.

Table 1 presents the nominal, minimal and maximum temperature and flow rate values registered during the first operation phase. The temperature at the condenser inlet was lower than the nominal one because the tests of this first phase took place in wintertime. Lower generator/evaporator temperature than those used for planning were a consequence of the lower-than-expected flow rate of oily steam. Therefore, lower than planned absorber outlet temperatures were measured in this first phase.

Table 1. Operating conditions of the heat upgrade system for the monitored steady state points

Prototype	design	nominal	minimum	maximum
Flow rate at circuit T_2 (Absorber circuit) in m3/h	24	27	1	5
Flow rate at circuit T_1 (G-E) in m3/h	74	74,7	1,1	5
Flow rate at circuit T_0 (C) in m3/h	36	36	14,8	24,4
Solution Mass Flow Rate in kg/h	3000	2600		
Temperature at Condenser inlet, $T_{0,in}$ in ºC	25	18	15.5	28.4
Temperature at Generator/Evaporator inlet $T_{1,in}$ in ºC	95	90	86.1	95.2
Temperature at Absorber Inlet, $T_{2,in}$ in ºC	130	115	113.1	126.2
Temperature at Absorber Outlet, $T_{2,out}$ in ºC	140	125	121.2	133.9
Gross Temperature Lift, GTL in K	45	35		
Upgraded heat flow rate, $Q_{A,}$ in kW	200	230	218.8	248.5
COP = Q_A / (Q_G + Q_E)	0.48	0.50	0.42	0.51

The table also includes the range of values registered for the upgraded heat flow rate, Q_A, and the thermal coefficient of performance COP, that is the ratio between the upgraded heat and the driving heat transferred at the evaporator Q_E and generator Q_G.

Although the COP value remains practically constant at around 0.5, the heat flow rate varies between 218 and 248 kW. As shown in Table 1, the design value of the upgraded heat flow rate, 200 kW, is smaller than the measured values, but the operating conditions in the monitored period are different than the design ones.

The main difference found in the operating conditions is on the T_0 circuit: although the system was designed for cooling water temperature of 25 ºC, the temperature from the cooling water network of the plant was mainly in the range between 16 and 22 ºC. Additionally, the system was designed for temperatures at the absorber circuit T_2 higher than those at which it has operated for most of the time. The reason for this is that the recirculation valve adjusting the absorber temperature in circuit T_2 was not configurated for the operation range required by the new cooling water conditions, and thus the set point value could only be achieved when the cooling water temperature aproached its design value. The necessary adjustments of the control parameters of the absorption recirculation valve could not be made because the visit planned for parameter adjustment in March 2020 was cancelled due the irruption of the COVID-19 pandemic.

A set of 28 steady state points however have been obtained from the first operation phase. Each steady state correspond to a monitored operation period of the system of at least 60 minutes with no relevant change in temperature or flow rate at the three external circuits T_0, T_1 and T_2.

The upgraded heat flow rate and coefficient of performance are presented in the Figure 2 against the cooling water inlet temperatures $T_{c,in} = T_{0,in}$. As the Figure 2 shows, most of the monitored point correspond to points with driving temperature $T_{1,in}$ in the range between 87 and 89 ºC and condenser inlet temperatures $T_{0,in}$ between 16 and 22 ºC. As it was presented in [6] for the results of the first laboratory prototype, for constant values for $T_{1,in}$ and $T_{2,in}$ a decrease of the absorber capacity or heat flow rate Q_2 is expected with increasing $T_{0,in}$. Although a tendency trend can be observed for this results, no conclussions can be obtained since neither T_{1in} nor $T_{2,in}$ are constant for these measurements.

On the lower section of the figures the COP values are presented. The values remain constant at a value around 0.50 with the exception of one point with $T_{1,in}$=95 ºC, $T_{0,in}$=16 ºC and $T_{2,out}$=134 ºC (COP=0.42).

For another point with $T_{1,in}$=95 ºC, $T_{0,in}$=18 ºC and $T_{2,out}$=132 ºC however, the monitored COP value was 0.51. For most of the cases, the COP measured for the system is larger than the expected one for the design conditions at around 0.48.

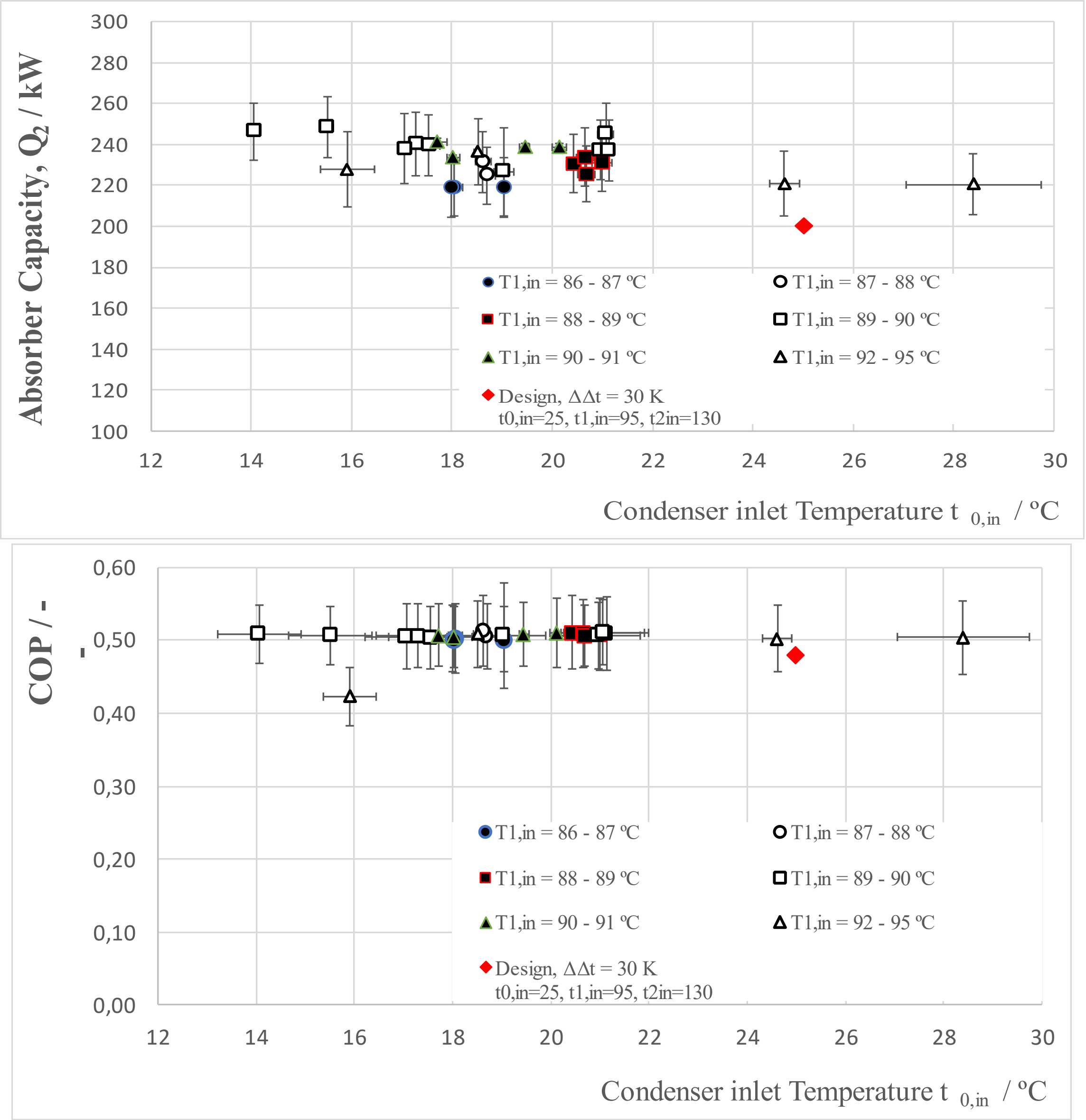

Fig. 2. AHT heat upgrade capacity and COP vs the condenser inlet temperature

In order to have a better overview of the operation map of the AHT during the monitored period, the Absorber capacity Qa and thermal COP values are plotted against the temperature of glycol leaving the absorber $T_{2,out}$ in Figure 3.

The results of Figure 3 reveal that the AHT has been operating in the monitored period for most of the time with temperatures t_{2out} between 122 and 123 ºC. The diagram also shows, that the AHT is able to operate with delivery temperatures of 132 and 134 ºC and heat upgrade capacities between 220 and 240 kW. Due to the difficulties in the control of the valves in the absorber circuit previously mentioned, no measurement values are yet obtained for delivery temperatures t_{2ou} at the design temperature 140 ºC. The COP values presented in the lower part of Figure 3 show that the only point measured with a lower COP correspond to the highest delivery temperature T_{2out}. Although this could be intuitively be interpretated as a correlation of T_{2out} and COP this makes no sense and would contradict the experimental results presented in [6] and [8], that show that COP higher than 0.45 can be obtained with and t_{2out} as high as 138 ºC. More likely, the measurement with the lowest COP correspond to a measurement with NC-gases present in the system [9].

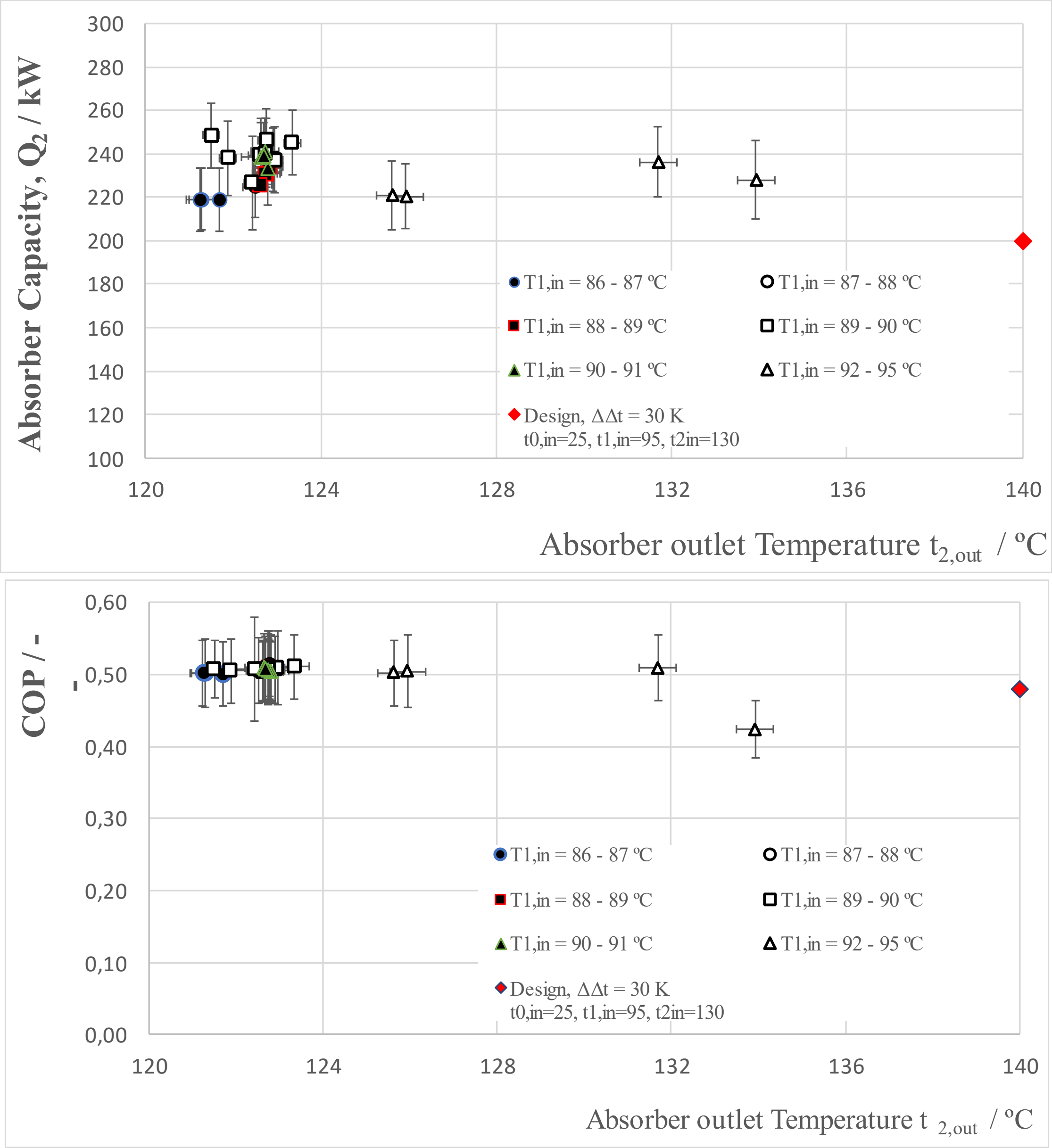

Fig. 3. AHT heat upgrade capacity and COP vs the absorber outlet temperature

3.2. Comparison with laboratory prototype

The results presented in Figures 2 and 3 have been compared with the results previously measured at the laboratory under controlled conditions and presented in [6] and [8]. The measurement results of the first prototype developed within the Indus3Es project have been previously presented in the 12th HPC Conference [6] showing the dependances of absorber capacity and COP with the external circuit temperatures. The measurement results of the second prototype (pr II) developed in the project has been presented in [7] analyzing its differences with those of prototype I. The measurement results of this prototype pr II have been filtered and analyzed in detail in [8], being used also to validate and update the characteristic equation model of an absorption heat transformer.

Table 2. Comparison of prototype characteristics

Prototype	A_A (m^2)	A_G/A_A (-)	A_E/A_A (-)	A_C/A_A (-)	V_A (m^3/h)	$V_G = V_E$ (m^3/h)	V_C (m^3/h)	m_{sol} (kg/h)
Laboratory Prototype pr I [6]	7.6	1	1	1	5	5	5	700
Laboratory Prototype pr II [7]	8.2	1,1	0,8	0,8	5	5	5	700
Pilot Scale Prototype (This work)	43.3	1,1	0,8	0,7	24	37	36	3000

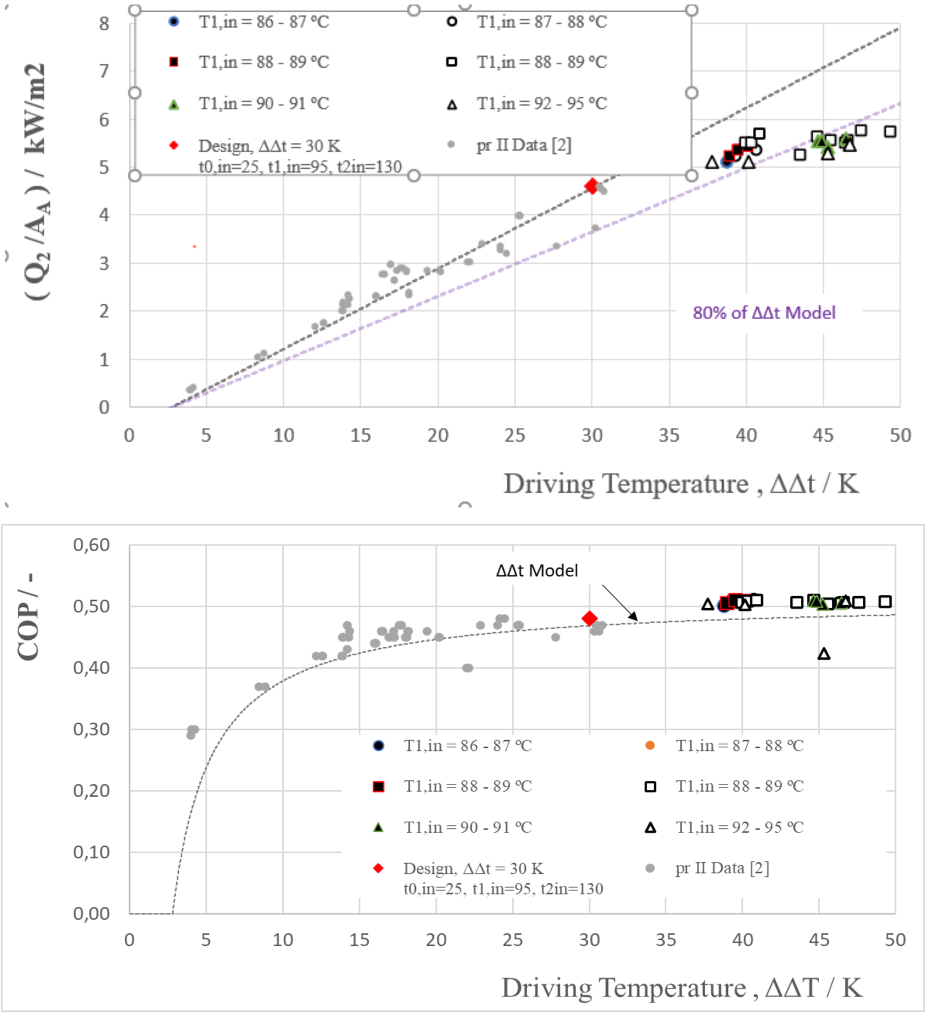

Fig. 4. AHT Absorber capacity and COP compared to that of the laboratory prototype pr II

The characteristic equation model has been previously used for the upscaling process from the lab scale prototype to the 200 kW industrial prototype. The model can also be used to analyze deviations between the expected performance of the AHT and the monitored results

As it is presented in Table 2, the laboratory prototype has been upscaled by enlarging the total heat exchanger area of the absorber (A_A) by a factor of around 5. The relations between the heat exchanger area of the absorber and those of evaporator, absorber and condenser of the pilot scale prototype is the same as that of the laboratory prototype, and the solution flow rate and external heat carrier flow rates have been set in order to have equal heat transfer rates at each exchanger.

With these assumptions, it has been expected that the specific heat upgrade capacity of the AHT expressed in kW/m^2 absorber can be predicted by the characteristic equation line presented in [8].

In the Figure 4, specific absorber capacity and COP of the systems are plotted against the driving temperature difference ΔΔt, that can be defined as presented in equation (1). ΔΔt is the double temperature difference and expresses the driving potential of the AHT for heat upgrade. (R is a constant around 1.15).

$$\Delta\Delta t = \left(t_{gen} - t_{abs}\right) - R \cdot \left(t_{con} - t_{eva}\right) = R \cdot \left(t_1 - t_0\right) - \left(t_2 - t_1\right) \qquad (1)$$

As presented in Figure 4, the driving temperature differences during the monitored operation of the AHT are much larger than expected (between 36 and 48 K instead of 30 K). Therefore, the absorber capacities measured for the AHT are larger than the design value. The Figure 4 reveals however, that the heat flow measured are between 12 and 20 % smaller than those calculated by the characteristic equation model.

The difference between measured and expected capacity could be the consequence of two causes. On the one hand, the dissipation temperature in the condenser was lower than expected in most of the cases. There is scientific evidence that the lower the pressure of the lower pressure vessel, the influence of the NC gases could be increased, which would result in the decrease of the performance [9]. On the other hand, the solution flow rate in these tests was 13 % lower than the one defined from the design phase. During the design phase, the flow rate was set to have the same specific flow rate along the tube´s length. Because of cavitation risk, the solution flow rate was set to about 2600 kg·h^{-1}, representing a decrease of the 13 %.

The lower diagram on Figure 4 on the other side, explains why the monitored COP values are practically constant. The characteristic equation method predicts that as the absorber capacity increases, the impact of the heat losses and other inefficiencies present in the absorption cycle is reduced, and thus for large ΔΔt values the COP remain practically constant. The monitored COP values exceeding expectations may be due to measurement inaccuracies (uncertainties in the COP of 0.05 have been determined from the uncertainties in the determination of Q_A, Q_G and Q_C in the range between 13.9 and 36.0 kW as shown in Figures 2 and 3).

3.3. Improved performance using an adiabatic absorption chamber

One technical innovation included with the AHT Pilot Scale Prototype is the two-chamber absorption design with a separated adiabatic absorption chamber with atomizing spray nozzles distribution system. This design has been adapted to increase the contact area of the solution and vapor in the absorber and optimize the adiabatic absorption process increasing the residence time of the solution. This can be done by atomizing the solution into small droplets, as it happens in the dedicated adiabatic absorption chamber installed before the solution distribution system based on a conventional dropping tray as in the lab scale prototype.

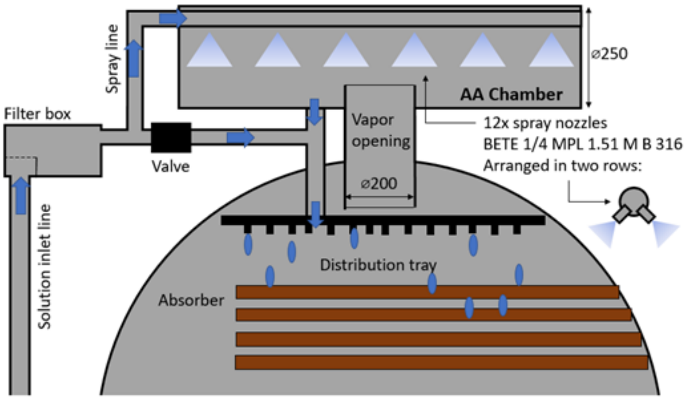

Fig. 5. Schematic view of the adiabatic absorption chamber

This dedicated adiabatic absorption chamber (AA chamber in Fig.5) can be bypassed and cut-off from the system by means of a controlled valve. This valve has been added in order isolate the adiabatic absorption chamber when desired and analyze its contribution separately.

During the second monitoring phase of the system the AHT was set to work under adiabatic mode. In this second monitoring phase both the installed adiabatic-absorption-vessel and the traditional absorber with distribution tray were in operation, while in the previously presented results.

The performance of the AHT in the second monitoring phase using the adiabatic absorption chamber (AAC) was significantly improved compared to that of the first measurements. It is estimated that the revalorization capacity of the AHT using the AAC is about 12 % higher comparing to the non-adiabatic mode, as it is shown in Fig.6 The black triangles symbols correspond to results using the AAC, and the empty circles to results without using it. The main advantage of the use of the AAV is the increase of the temperature entering to the absorber´s tube bundle. With higher temperature at the first row of the tubes, higher temperature energy could be transferred to the external circuit. During the tests performed, it was estimated that the 13 % of the absorption rate occurs in the AAC, increasing in about 20 K the entering solution temperature.

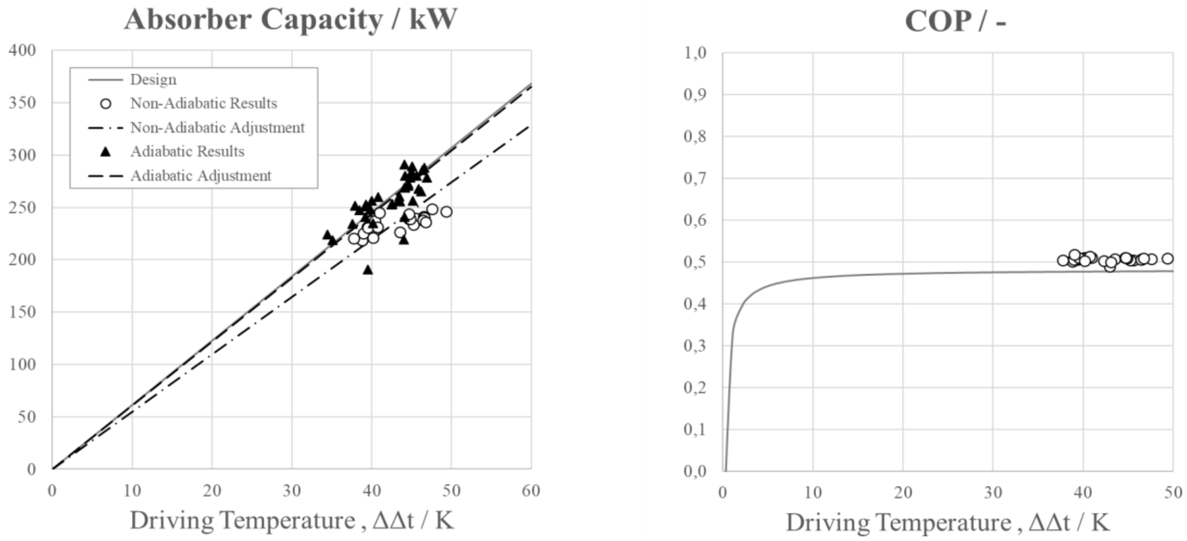

Fig. 6. Pilot Scale AHT absorber capacity and COP under Non-adiabatic(○) and Adiabatic(▲) modes.

4. Dynamic behavior of the absorption heat transformer

Although the parameters of the controller of the heat upgrade system could not be adjusted within the lifetime of the project due to the irruption of the COVID pandemic the control system demonstrated that the heat system could operate autonomously adjusting the outlet temperature at the absorber of the AHT, at the given set point. Figure 7 presents continuous monitored data of the AHT operating between the 19 and 24 of February 2020.

The control system adjusts valves in the circuits in order to maintain T_{2out} at either 120, 125 and 130 ºC. Therefore, the flow rate at the absorber circuit T_2 experiments relatively strong variations. The flow rates at Generator, Evaporator and Demin Water circuits are kept constant but the efforts made in the heat rejection circuit to change the value of T_{0in} are sterile, although the flow rate at the condenser suffer strong variations.
The main oscillations in the driving heat Temperature T_1 are due to oscillations in the flow non-monitored flow rate of oily steam entering the condensing heat exchanger HX-6.

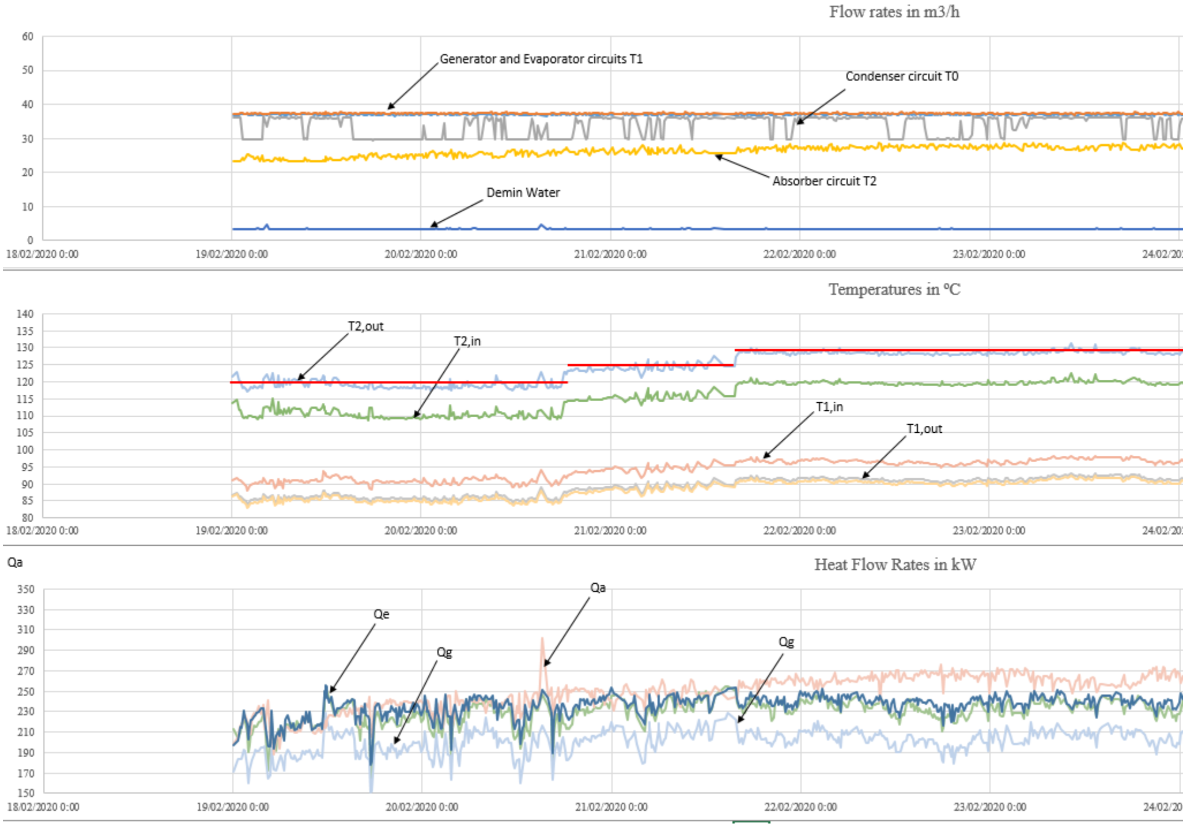

Fig. 7. Dynamic behavior of the AHT in the period between 19 and 24.02.2020

These oscillations are the main difficulty to overcome to find steady state conditions for the analysis of the system and its comparison with the laboratory measurements. Only when this oscillation stops could steady state periods longer than 60 minutes be analyzed as have been presented in the previous figures.

However, although a good comparison between steady state conditions is only possible for some points the results presented in Figure 7 show that the upgraded heat flow rate supplied by the AHT have been continuously above the design value of 200 kW, achieving at moments 270 kW.

5. Final considerations and conclusions

This paper has presented and analyzed the monitored performance of an absorption heat transformer with a design capacity of 200 kW for upgrading heat with up to 140 ºC with a driving heat temperature of 95 ºC. The monitored results presented a very stable thermal coefficient of performance (useful to driving heat ratio) of around 0.50, but with an uncertainty of around 0.05. The measured upgraded heat flow rates are between 218 and 270 kW for heat upgrade temperatures between 115 and 138 ºC, with driving heat temperatures between 88 and 95 ºC. The higher-than-expected measured flow rates are due to more convenient conditions than predicted for the operation. The performance is however somehow lower than expected, although some hypothesis can be formulated for this deviation. The use of the innovative adiabatic absorption chamber, on the other side, increases the performance by around 10%.

The pilot plant prototype also included an innovative motor-less non-condensable (NC) gases purge system solution that due to time limitation within the project and irruption of the COVID Pandemic could not be tested within the project. Additionally, the system a control strategy which prevents the cycle from moving into risky operational conditions. The advanced control system ensures safe operation of the heat transformer minimizing the risk of crystallization while it optimizes the performance characteristics. The control strategies used for the chillers implies the modification of current optimization models. The Characteristic Equation Method has been used for the definition of the control system [5]. The advantages of these technological innovations have not been discussed within this work.

The economical performance of a ready-to-market system with this capacity has been discussed in [10] based on the performance results presented in this work. It concluded that the total cost of the system implementation would be about 420.000 €, i.e. about 1.500 €/kW revalorized by the AHT, and that the investment would be completely recovered within 6 years. If the total considered installation would include a 600 kW prototype, the payback period would be reduced to 3 years, and to 2 years in case 1.2 MW AHT is considered.

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