Hybrid Compression-Absorption Chiller Driven by low Temperature Waste Heat: Modelling and Analysis

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Abstract: The aim of this study is to model and investigate the concept of a hybrid ammonia-water absorption chiller. Its particularity is having a compressor that creates an intermediate pressure at the desorber. This modification should increase the flexibility of the machine, allowing the recovery of lower temperature waste heat. A preliminary simulation was based on a single stage commercial 10 kW ammonia-water absorption chiller. An experimental study was completed to validate the numerical model of a traditional single stage absorption chiller. The comparative energy and exergy analysis of the hybrid and traditional absorption chillers was undertaken. Due to the reduction of the waste heat temperature for the hybrid chiller, the exergy efficiency increases by 34%. A 38% reduction of the UA coefficient (the product of overall heat transfer coefficient and heat exchanger area), for the same waste heat temperature and refrigeration power, is shown for the hybrid system compared with the traditional one. The theoretical lower bound of the waste heat temperature to be used to drive a hybrid cycle is 42 °C. An economic analysis shows that a hybrid cycle is initially more expensive but quickly becomes profitable compared to a gas-fired absorption machine.

Keywords: Ammonia-water Refrigeration, Hybrid Cycle, Modelling, Experimental Validation, Exergy Analysis, Economic Analysis.

1. INTRODUCTION

Refrigeration applications consume almost 15% of all electricity generated and air conditioning applications are responsible for 45% of housing and commercial building energy consumption [1]. This electricity is mainly produced by non-renewable sources harmful to the environment [2], while population growth increases world electrical energy demand. Alternative energy sources are therefore explored for refrigeration systems.

The use of renewable and abundant solar radiation is proposed as a suitable alternative, particularly for remote areas or areas suffering from electricity shortage [3]. Another source of energy that could be considered is to reuse industrial waste heat to drive refrigeration cycles. For instance, in some parts of the world waste heat produced by industrial processes are generally considered as a contaminant and simply rejected to the environment [4]. The waste heat will be considered as a free energy source to drive absorption chillers [5]. The traditional absorption machine is divided into a refrigerant and a driven part. The refrigerant part is composed of an evaporator, a condenser and a refrigerant expansion valve in which circulates the refrigerant (see Figure 1). The driven part is composed of an absorber, a desorber, a Solution Heat Exchanger (SHX) for improving the overall efficiency, a pump and a solution expansion valve. These components compress the low-pressure refrigerant from the evaporator. In the absorber, an exothermic absorption reaction occurs at low pressure between the liquid absorbent and the more volatile refrigerant. The liquid mixture rich in refrigerant (rich solution) is then pumped to the desorber. The inverse reaction occurs inside the desorber at high pressure where heat is transferred from the external heat source. A vapour stream continues to the refrigerant part while the remaining liquid, poor in refrigerant (poor solution), is throttled back to the absorber. The more commonly used absorbent-refrigerant mixtures are water-lithium bromide and ammonia-water, where water is the refrigerant in the former and the absorbent is the later. Ammonia-water will be used in this article as an example [6,7].

Nowadays, waste heat sources under 70-80 °C are difficult to use for driving ammonia-water chillers [8,9]. However valorization of lower waste heat temperature would be profitable because of their abundance and the actual few end uses of these sources [10]. For instance, breweries, bakeries [11] and applications of Combined Heat and Power (CHP) [12] produce such waste heat and combine the need for refrigeration. The reduction in a source temperature leads to a considerable drop in the Coefficient of Performance (COP) [13].

The combination of absorption and compression machines diminishes their individual drawbacks and allows to reduce the required heat temperature [14].
Only a few publications on these hybrid cycles can be found. A high-pressure compressor between the desorber and condenser increases refrigerant production by lowering the desorber pressure and its corresponding temperature [15]. Alternatively, a compressor between the evaporator and absorber can also operate at a lower desorber temperature and with a better COP than a traditional absorption configuration [16]. Both of these hybrid cycles have respectively an electricity consumption of a third and a half that of vapour compression cycles [17]. However, the three publications mentioned supposed that the refrigerant is pure ammonia without traces of water, whereas practical experience with Absorption Refrigeration Ammonia-Water (ARWA) shows that the ammonia rectification process is a crucial issue in order to obtain an efficient and reliable system [18]. The present study will take into account the purity of ammonia and the resulting conditions.

Another alternative is the use of an ejector to increase the absorber pressure and correspondingly the refrigerant concentration in the rich solution. The minimal waste heat temperature is reduced and the overall COP is increased [6,19]. This principle is similar to the hybrid cycle with a compressor between the evaporator and the absorber but offers reduced electricity consumption. Such a configuration does not change the refrigerant purity.

The aim of this study is to establish a proof of concept based on theoretical analysis of a hybrid cycle, having a compressor between the desorber and the condenser, to determine if the configuration is actually viable. This hybrid configuration has been chosen because it is considered more capable of managing and adapting to heat source temperature variations due to the proximity of the compressor with the desorber. In addition, it might improve the ammonia vapour purity in the desorber. This last point could lead to a reduction in size or possible removal of the rectifier.

The software ASPEN Plus [20] allows user friendly steady-state process modelling and includes interaction properties of ammonia and water based on thermophysical models and data. The Peng-Robinson equation of state [21] to be used has been taken from the Aspen Plus software package. This calculation method has been chosen for its closeness to the Tillner-Roth state equation [22] to model ammonia-water mixture behaviour, showing good results for phase equilibrium calculations [23] and agreement with experimental data [24]. For instance, only a 4% overestimation of the poor solution flow rate was noticed in ASPEN Plus, compared to the experimental results [23].

An experimental study has been completed on a 10 kW commercial ARWA. These experimental results have been used to validate the simulation of a
traditional ARWA and then served as input to the hybrid configuration. After implementing a compressor in the simulation and carrying out some adjustments, a study has been conducted with a lower waste heat temperature than in the traditional configuration. At constant refrigeration load, the minimum heat source temperature and the conditions where it is possible to operate without a rectifier have been investigated and a comparative economic analysis with a gas-fired ARWA has been accomplished.

2. NUMERICAL MODEL OF A TRADITIONAL ARWA

2.1. Simulation Properties and Requirements

The installation studied works with a Solution Heat Exchanger (SHX) shown in Figure 1. The SHX is used to increase the overall installation COP by transferring heat from the relatively hot poor solution (13) leaving the desorber, to the cooler rich solution (8) entering it. This recovery of internal heat in the SHX decreases both the energy that must be evacuated from the absorber and that must be supplied to the desorber.

In the rectifier, the rich solution (9) enters by the top and descends along the trays to the bottom. The vapour generated by the desorber (12) enters the rectifier by the bottom and ascends in countercurrent. This configuration is similar to a direct contact heat exchanger where the rectification process is effected by the rich solution. The rising vapour is cooled and a partial condensation of the water contained inside (12) occurs, determining the refrigerant concentration [18]. The outlet at the top (1) is the purified ammonia vapour which still contains some traces of water. Due to the non-negligible presence of water, the outlet of the evaporator (6) will not be considered as pure ammonia [14].

The desorber is separated from the rectifier and is operated as a thermosiphon, in this way the circulation is driven by passive heat exchange based on natural convection. The density difference between the desorber inlet (11) and outlet (12) is sufficient to make the fluid circulate inside the desorber. The available waste heat is simulated by the stream (26).

This simulation will be validated by the experimental study data and will be followed by a numerical model of the hybrid system. Inputs equivalent to the test bench environmental data are required. For instance, the evaporation temperature is usually an inviolate design parameter set by the process demand and the cooling temperature is set by the environment [18]. The goal is to develop a model in which internal parameters of the machine are calculated automatically as a function of the external loads. A refrigeration heat load is input to the simulation and the other heat duties are then calculated. The model functioning of the absorption chiller is described in detail by Loeb [25] and a similar model has been developed by Somers [26].

The simulation assumptions are:

- Heat losses to the surroundings are negligible [27].
- Pressure changes are taken into account in the expansion valves and the pump only [27].
- Expansion valve outlet properties are calculated assuming isenthalpic expansions [28].
- The absorber outlet stream (7) is set to 3 °C of subcooling and the condenser (29) to 5 °C of subcooling [29].
- Pinch are set in the evaporator (between points 5 and 20) at 5 °C and in the absorber (between points 7 and 22) at 5 °C according to the experimental data [30].
- The pump flow is 15.8 l/min of rich solution [28].
- Isentropic efficiency of the pump is 80% [31]. This value is considered as typical in the literature [32].
- Heat exchanger efficiency of SHX is 80% [28].

2.2. Experimental study on the test bench

The test bench is equipped with a commercial ARWA, at Hydro-Quebec Energy Technology Laboratory in Shawinigan (Qc, Canada). The cooling unit is made for solar air conditioning (PSC chilli 10, Solarnext) [33]. Among the five heat exchangers, the evaporator and condenser units are shell and tube, while the SHX, desorber and condenser are of the plate and frame style. Two industrial boilers provide the external streams at the requested temperatures. One imitates the waste heat stream at high temperature into the desorber, the other imitates a refrigeration load at the evaporator. The cooling stream at the condenser and the absorber is provided by a compression chiller. Temperature sensors are installed at each location corresponding to a process state point. The installation has already been further detailed; heat duty measurement errors are between ±0.58% and ±4.19% [29,30].
2.3. Comparison between the simulation and the experimental results

In order to compare the results from the simulation and the experiment, both are studied for the same external streams. The cooling fluid (22, 28) and the cold stream (20) are composed of ethylene glycol and water both at a 50/50 mix. The waste heat stream (26) is composed of pure water. These were chosen to simulate a 10 kW application of air conditioning in fan coil unit. The standard operating conditions given by the manufacturer are a waste heat entry temperature (26) of 85 °C and an exit temperature (27) of 78 °C. An entry cooling temperature (22, 28) of 24 °C and an exit (23, 29) of 29 °C. An entry cold stream in the evaporator (20) of 12 °C and an exit temperature of 6 °C [33,34]. A low pressure of 5.1 bar occurs in the simulation at the evaporator and the absorber, with a high pressure of 12.3 bar at the desorber and the condenser.

Table 1 compares the experimental heat duties with those of the simulation study. A heat duty difference of 3% at the desorber, 4% at the condenser and 26% at the absorber are noticed. This can be explained by a higher cooling need in the simulation to counterbalance the experimental heat losses. Indeed, the test heat duty evacuated (22.7 kW) are inferior to the heat duty provided (27.3 kW). The pump electric power of only 433 W has only a small impact on the machine. Table 2 compares the internal temperatures, differences can also be partly explained by measurement error. The results obtained by the simulation are considered as sufficiently similar with the test bench results to suppose that the simulation is accurate.

The ammonia purity at the outlet of the rectifier (1) is calculated to be 99.6 mass percent of ammonia. Around this value, the variation in COP as a function of ammonia concentration is negligible [18]. In close conditions, an ammonia mass fraction between 86.0% and 95.1% at the desorber outlet and between 99.3% and 99.7% at the rectifier outlet has been measured [30]. Thus, a refrigerant with a 99.6% mass of ammonia will be assumed as pure enough to operate in the condenser and the evaporator with good performance. The chosen condition for the possible removal of the rectifier will be a concentration equal or greater than 99.6% mass of ammonia at the desorber outlet in the following hybrid ARWA study. In addition, a circulation of 7.2 l/min of solution is calculated inside the desorber in thermosiphon. This value will be taken as the volumetric flow rate of the pump.

| Test Model |
|---|---|
| Desorber (Waste heat) | | |
| T26 in (°C) | 86.7 | 86.7 |
| T27 out (°C) | 78.7 | 79.2 |
| ΔT in-out (°C) | 8.0 | 7.5 |
| Qdes (kW) | 17.5 | 18.0 |
| ΔT26 (l/min) | 33.8 | 33.8 |

| Evaporator (Cold production) | | |
| T20 in (°C) | 13.2 | 13.2 |
| T21 out (°C) | 9.5 | 9.7 |
| ΔT in-out (°C) | 3.7 | 3.5 |
| Qeva (kW) | 9.4 | 9.4 |
| ΔT20 (l/min) | 45.0 | 45.0 |

| Condenser | | |
| T28 in (°C) | 22.7 | 22.7 |
| T29 out (°C) | 30.7 | 31.0 |
| ΔT in-out (°C) | 7.9 | 8.3 |
| Qcon (kW) | 10.4 | 10.8 |
| ΔT28 (l/min) | 22.3 | 22.3 |

| Pump Wpump (kW) | | |
| Test Model |
|---|---|
| T1 (°C) | 63.5 | 60.5 |
| T2 (°C) | 28.8 | 27.2 |
| T4 (°C) | 5.9 | 5.8 |
| T5 (°C) | 8.8 | 8.2 |
| T7 (°C) | 24.6 | 27.7 |
| T8 (°C) | 24.6 | 27.8 |
| T9 (°C) | 61.8 | 55.9 |
| T10 (°C) | 67.9 | 63.8 |
| T12 (°C) | 77.2 | 73.4 |
| T13 (°C) | 69.7 | 63.8 |
| T14 (°C) | 29.5 | 34.7 |
| T15 (°C) | 29.8 | 33.1 |
| COP | 0.54 | 0.52 |
3. HYBRID CYCLE DESIGN AND ANALYSIS

3.1. Advantages of the Hybrid Cycle Based on the Thermodynamic Analysis of a Desorber and a Condenser

Given that the condensation temperature must be kept higher than the temperature of the cooling fluid (in this study 22.7 °C), the condensation pressure needs to be sufficiently high, as shown in Figure 3 on the right scale. At the same time the rise in pressure does not allow the sufficient desorption of refrigerant. In a traditional configuration the desorber’s pressure is the same as in the condenser, see Figure 2. Incorporating a compressor enables lowering the desorber pressure while maintaining the same condenser pressure. This lower pressure of desorption allows using lower waste heat temperature while maintaining constant refrigeration load.

3.2. Simulations Properties and Requirements

To date the hybrid cycle has not been prototyped for safety reasons. From the traditional simulation a model of a hybrid configuration is established with the necessary adjustments. In a perspective to determine conditions where the rectifier can be removed, the desorber is no longer in a thermosiphon configuration. This allows, firstly, pumping less volume flow rate from the absorber than in the previous simulation because 100% of the rich solution crosses the desorber and, secondly, to model a configuration where the rectifier could be simply removed. Figure 4 shows the vapour produced in the desorber enters at the bottom of the rectifier (12). A compressor is added into the hybrid cycle on the vapour refrigerant stream, fixing the condensation pressure and allowing the desorber pressure to be set by the pump. A traditional ARWA with the same configuration and assumptions is simulated by simply giving no power to the compressor, in which case the compression process is accomplished solely by the thermal behaviour of the ammonia-water solution.

Figure 2: Principles comparison between a) a traditional ARWA and b) the hybrid cycle configuration.

Figure 3: Influence of the pressure on the refrigerant total condensation temperature and the percentage of rich solution desorbed as refrigerant (nominal fan coil conditions).

The same external stream, high-pressure at the condenser and low pressure at the evaporator and absorber as those of the traditional simulation are employed providing a partial validation of the hybrid numerical model. Both of these pressures are set constant while the desorber pressure varies. The pump outlet pressure (desorber pressure) covers the whole range from the absorber to the condenser pressure. The desorber’s coefficient UA and its heat duty are
calculated in order to provide a requested refrigeration load. The compressor power depends on the desorber pressure and the refrigeration load and is considered as maximal when the desorber pressure is equal to the absorber pressure.

The difference of assumptions in the hybrid ARWA compared to the traditional simulation are:

- The pump flow is 7.2 l/min of rich solution.
- Isentropic efficiency of the compressor is 70% [14].
- Minimal pinch of 5 °C is set in the desorber (between points 26 and 12-13) preventing an oversized desorber. In fact, at constant desorber heat duty a smaller pinch would increase inordinately the desorber UA to reach the target refrigeration load. Thus an arbitrary pinch limit of 5 °C is set, stopping the simulation.

Table 3 and 4 show the results of a hybrid cycle computation with a low waste heat temperature constant of 60 °C and a refrigerant load of 10 kW at different compressor electric power. As shown in Table 3, an increasing compressor power (+0.8 kW) implies a decreasing desorber (-1.4 kW) and absorber (-1.1 kW) heat duty while the condenser heat duty increases (+0.5 kW). The higher condenser heat duty is caused by a hotter vapour from the compressor (1’) needing to be condensed, see Table 4.

3.3. Simulation Results

Assuming that the pump electric power is negligible [28], the COP value of the hybrid system based on the first thermodynamic principle is:

$$\text{COP} = \frac{Q_{eva}}{Q_{des} + W_{comp}}$$  \hspace{1cm} (1)

Given that the quality of mechanical and thermal energies are different [14], the efficiency based on the second thermodynamic principle should be evaluated. It is assumed that the environment temperature \( T_{0} \), is the temperature of the cooling streams (considered as free) entering at the condenser (28) and at the absorber (22) [35]. In order to evaluate the exergy flows of a system, equation (2) is used to calculate the equivalent temperatures \( T_{eq} \) of the external circuits at the evaporator and the desorber [36].

$$T_{eq} = \frac{h_{out} - h_{in}}{s_{out} - s_{in}}$$  \hspace{1cm} (2)
Carnot temperature coefficients are calculated by using the equivalent temperatures:

\[ \theta_{\text{eva}} = 1 - \frac{T_0}{T_{\text{eq,eva}}} \]  
(3)

\[ \theta_{\text{des}} = 1 - \frac{T_0}{T_{\text{eq,des}}} \]  
(4)

The exergy efficiency is defined in an absorption machine as the ratio between the exergy obtained from the evaporator and the exergy provided to the desorber [5]. In a hybrid absorption machine the exergy efficiency also includes the compressor work:

\[ \eta_{\text{ex}} = \frac{\theta_{\text{eva}} \times Q_{\text{eva}}}{\theta_{\text{des}} \times Q_{\text{des}} + W_{\text{comp}}} \]  
(5)

Table 3: Comparison of Hybrid Cycle Thermodynamic Properties for a 0.8 and 1.6 kW Compressor Power with a Waste Heat Temperature of 60 °C and a Refrigeration Load of 10 kW

| Test Model | 0.8 kW | 1.6 kW |
|------------|--------|--------|
| Compressor | \( W_{\text{comp}} \) (kW) | 0.8 | 1.6 |
| Pump       | \( W_{\text{pump}} \) (kW)  | 0.2 | 0.2 |
| Desorber (Waste heat) | \( T_{26} \) (°C) | 60.0 | 60.0 |
|            | \( T_{27} \) out (°C)    | 54.2 | 54.8 |
|            | \( \Delta T \) in-out (°C) | 5.8 | 5.2 |
|            | \( \dot{V}_{26} \) (l/min) | 33.8 | 33.8 |
|            | \( Q_{\text{des}} \) (kW) | 14.1 | 12.7 |
| Evaporator (Cold production) | \( T_{20} \) in (°C) | 13.2 | 13.2 |
|            | \( T_{21} \) out (°C)    | 9.4 | 9.4 |
|            | \( \Delta T \) in-out (°C) | 3.8 | 3.8 |
|            | \( \dot{V}_{20} \) (l/min) | 45.0 | 45.0 |
|            | \( Q_{\text{eva}} \) (kW) | 10.0 | 10.0 |
| Absorber | \( T_{22} \) in (°C) | 22.7 | 22.7 |
|            | \( T_{23} \) out (°C)    | 28.8 | 28.3 |
|            | \( \Delta T \) in-out (°C) | 6.1 | 5.6 |
|            | \( \dot{V}_{22} \) (l/min) | 37.2 | 37.2 |
|            | \( Q_{\text{abs}} \) (kW) | 13.3 | 12.2 |
| Condenser | \( T_{28} \) in (°C) | 22.7 | 22.7 |
|            | \( T_{29} \) out (°C)    | 31.7 | 32.1 |
|            | \( \Delta T \) in-out (°C) | 9.0 | 9.4 |
|            | \( \dot{V}_{28} \) (l/min) | 22.3 | 22.3 |
|            | \( Q_{\text{cond}} \) (kW) | 11.8 | 12.3 |

Table 4: Comparison of Hybrid Cycle Internal Properties for a 0.8 (Left Column) and 1.6 kW (Right Column) Compressor Power with a Waste Heat Temperature of 60 °C and a Refrigeration Load of 10 kW

| Test Model | 0.8 kW | 1.6 kW |
|------------|--------|--------|
| Test       | 46.2   | 32.9   |
| \( T_{1} \) (°C) | 91.5  | 121.8 |
| \( T_{1}' \) (°C) | 91.5  | 121.8 |
| \( T_{2} \) (°C) | 27.2  | 27.1  |
| \( T_{4} \) (°C) | 5.6   | 5.6   |
| \( T_{5} \) (°C) | 8.2   | 8.2   |
| \( T_{7} \) (°C) | 27.7  | 27.7  |
| \( T_{8} \) (°C) | 27.7  | 27.7  |
| \( T_{9} \) (°C) | 45.5  | 32.9  |
| \( T_{11} \) (°C) | 46.3  | 33.0  |
| \( T_{12} \) (°C) | 52.2  | 37.9  |
| \( T_{14} \) (°C) | 32.6  | 29.8  |
| \( T_{15} \) (°C) | 32.7  | 29.8  |
| \( P_{\text{des}} \) (bar) | 8.1   | 5.4   |

Figure 5 illustrates a three dimensional surface at a given refrigeration load. A high compressor power means a low desorber pressure. The sum of the five heat exchangers UA (UA Sum) represents the heat transfer rate of an installation. The upper dotted curve with no compressor power represents the working conditions of a traditional ARWA. This curve breaks up at low temperature at 74 °C because the minimal pinch temperature of 5 °C is reached inside the desorber. The temperature can't be lower because the desorber can't supply sufficient heat duty to meet the refrigeration load. A hybrid machine is necessary to operate at lower waste heat temperatures. This curve also shows a prohibitive increase in the UA sum while decreasing the temperature of the driven source.

Figure 5: Operating range of a hybrid cycle with a 10 kW refrigeration load.
Increasing compressor power decreases the sum of UA, as shown in Figure 6, thus reducing the installation size. It can be observed that the installation of a compressor is less advantageous when the temperature of waste heat is rising. For instance, at 70 °C the compressor’s installation allows a 38% reduction of the UA Sum, but only a 21% reduction at 100 °C. A hybrid installation with a sufficiently powerful compressor would be adaptable for a wide range of waste heat temperatures for given UA values.

The exergetic efficiency increases with the decrease of the waste heat temperature, as shown in Figure 7, becoming more significant at lower temperatures. The exergetic efficiency also decreases with the compression electric power, however the compressor is mainly employed to lower the waste heat temperature at the expense of increasing electricity consumption. For each waste heat temperature, the best exergetic efficiency is at minimum compression power. An optimum exergetic efficiency in this study is found at 62 °C of waste heat, where the value reaches 18.2%. Compared to the exergetic efficiency of the test bench simulation, which has been calculated at 12.1%, a 34% exergy efficiency increased is evaluated.

Figure 8 and 9 illustrate the minimum compressor power which has to be supplied at low temperature to meet the requested refrigeration load. When the desorber pressure reaches the absorber pressure (compression ratio of 2.4), the maximum compression power and ratio are reached; this limit is at 39 °C for a requested evaporator heat duty of 5 kW and 42 °C for 10 kW, see Figure 8. Figure 9 shows that the minimal compression work at 42 °C (1.7 kW of electricity) remains significantly inferior to the desorber heat duty (12.6 kW).

The use of a compressor allows using lower waste heat temperatures and potentially the removal of the rectifier in certain conditions of pressure and temperature, as illustrated in Figure 10. The desorber outlet temperature (12) is not related to the waste heat tem-
temperature, but only to the desorber pressure. The desorber pressure determines the required desorber heat duty, temperature and ammonia purity; Table 4 shows that at high compressor power, the desorber outlet temperature (12) is lower. The hybrid chiller without a rectifier will be restricted to a low desorber pressure under 6.3 bar, see Figure 10.

![Figure 9: Power which has to be supplied, at minimum compression work, with a 10 kW refrigeration load.](image)

![Figure 10: Desorber outlet conditions for a potential rectifier removal.](image)

4. ECONOMIC ANALYSIS

This economic analysis does not aim to give the overall price of a hybrid machine but to evaluate the cost difference between a hybrid machine and a traditional gas-fired ARWA. The gas-fired system is used when gas is available at relatively low price [5,11].

The two machines are assumed to only diverge in the price of the heat exchangers and the compressor. That is to say that the cost of expansion valves, pipes, pump, rectifier and electronics are kept constant, as are the set up and maintenance costs. The price of a direct fired heater is not taken into account in the gas-fired ARWA.

The heat transfer coefficient U is determined by numerous factors such as the material, the thickness, the geometry and the phases. In similar operating conditions an exchanger will operate with similar fluid phases, pressure and flow rates allowing U to be considered constant [31]. In this study, a heat exchanger with a constant UA is considered for a given equipment. In order to obtain the price of a heat exchanger, correlations from the literature use the exchange area A as variable while the hybrid ARWA model provides the heat exchanger UA values. The experimental test bench heat exchanger areas A, in combination with the test bench numerical model UA values, are used to calculate the constant U coefficients (see Table 5). U coefficients are then used to calculate heat exchanger areas from the hybrid model results.

The cost of each heat exchanger is calculated based on Equation (6) as the plate and frame correlation [37] and Equation (7) as the shell and tube correlation [38]. Equation (8) is used as the compressor correlation [39].

\[
C = 1600 + 210 \times A^{0.95} \tag{6}
\]

\[
C = 3 \times \left( \frac{A \times 10.76}{100} \right)^{0.024} \tag{7}
\]

\[
C = 61000 \left( \frac{W_{comp}}{100} \right)^{0.79} \tag{8}
\]

These correlations must be adjusted with the cost index of the year when the correlations were edited and with the actual cost index of the current year (2015 is chosen) [40,41]. C is the equipment cost and INDEX is the cost index at year 1 or 2:

\[
\frac{C_1}{C_2} = \frac{INDEX_1}{INDEX_2} \tag{9}
\]

An average electricity cost of 0.10 $/kWh is assumed for medium power sized company, although this
The hybrid system is compared at low waste heat temperature to a gas-fired ARWA with an operational time of 1, 2 and 5 years. A 10 kW heat duty at the evaporator is requested for a waste heat temperature from 50 to 70 °C, as above this temperature range a simple ARWA can be employed, giving less interest to the study. For this range of available waste heat and if the use of an ARWA machine is chosen, these are the two solutions remaining. The gas-fired scenario implies heating the hot flow (26) to the 85 °C nominal test bench heat temperature. The input heat duty to drive the process must be equal to the desorber heat duty, which is 15.8 kW at this temperature, adding a gas consumption cost. The hybrid configuration is determined at minimum compression power, see Figure 9, adding a compressor and electric power consumption cost and giving for each waste heat temperature the lowest initial cost.

Table 6 calculates where the hybrid scenario is profitable. The initial cost is the addition of the component costs considered (heat exchangers and compressor). The lower the waste heat temperature is, the less the hybrid system is profitable compared to a gas-fired ARWA. A lower waste heat temperature (26) increases the consumption of electricity inside a powerful and expensive compressor. However, at high compressor power the hybrid machine could be driven without the use of a rectifier, which could reduce the initial cost.

5. CONCLUSIONS

The hybrid configuration decreases the required UA Sum, allowing the design of a more compact machine. It also allows using lower temperature for desorption purposes, expanding in this way the number of available lower quality heat sources. The lowest waste heat temperature attainable for a 10 kW refrigeration load is 42 °C. These improvements are possible at the expense of increased compression power supplied by electricity.

Exergy efficiency is highly improved at lower waste heat temperature, making the hybrid cycle a thermodynamically efficient machine. In addition, the hybrid machine with a powerful compressor can work without a rectifier. Even though such a machine is more expensive than a traditional ARWA chiller because of the compressor cost, an economic analysis shows that in a
certain scenario this configuration could be profitable in a few years using low waste heat temperature compared to a gas-fired ARWA.

In future work, a modification of the actual test bench to implement the compressor and validate the model would be useful, as well as to validate the operation without a rectifier at low waste heat temperature. This future work will also have to determine how to operate in starting and stopping conditions of the ARWA. It could be interesting to compare the performance of this hybrid cycle with a configuration having a compressor between the evaporator and the absorber. Taking into account the heat losses and studying the control of the compressor in a varying environment are also worth considering.

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NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| A      | heat exchanger area (m²) |
| ARWA   | absorption-refrigeration water-ammonia |
| C      | cost ($US) |
| COP    | coefficient of performance |
| h      | specific enthalpy (kJ/kg) |
| LMTD   | log mean temperature difference (°C) |
| NH₃    | ammonia |
| P      | pressure (bar) |
| Q      | heat duty (kW) |
| s      | specific entropy (kJ/kg-K) |
| T      | temperature (°C) |
| UA     | product of the overall heat transfer coefficient by the area (kW/K) |
| U      | heat transfer coefficient (kW/K-m²) |
| ẋ      | volumetric flow rate (l/min) |
| W      | electric power (kW) |

Greek symbols

| Symbol | Description |
|--------|-------------|
| η      | efficiency |
| θ      | Carnot temperature coefficient |

Subscripts

| Symbol | Description |
|--------|-------------|
| 0      | environmental conditions |
| abs    | absorber |
| comp   | compressor |
| comb   | gas combustion |
| con    | condenser |
| des    | desorber |
| eva    | evaporator |
| eq     | equivalent |
| ex     | exergetic |
| in     | inlet conditions |
| out    | outlet conditions |

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