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**Air conditioning using an air-cooled single effect lithium bromide absorption
chiller: results of a trial conducted in Madrid in August 2005**

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Abstract

Trials were conducted to determine the performance of a commercial (Rotartica 045v) 4.5-kW air-cooled, single effect LiBr/H₂O absorption chiller for residential use. The experiments were run at La Poveda, Arganda del Rey, Madrid, in August 2005. Three typical August days, with different outdoor temperatures, were chosen for the study. The hot water inlet temperature in the generator varied throughout the day from 80 °C to 107 °C. Thermal demand was calculated, along with period energy balance and COP. Variations in machine component temperatures were recorded and chilling power and the daily COP calculated for each of the three days. The results for the period as a whole showed that cooling power tended to decline with rising outdoor dry bulb temperatures. At temperatures from 35 to 41.3 °C the chilled water outlet temperature

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in the evaporator climbed to over 15 °C. The average COP for the period, when auxiliary equipment was included into the calculations, was 0.37.

Keywords: absorption, LiBr-water, single effect, air-cooled, Rotartica

Nomenclature

A	area [m^2]
COP	coefficient of performance
C_p	specific heat capacity ($\text{kJkg}^{-1}\text{K}^{-1}$)
f	coincident factor
h_{fg}	latent heat of vaporization (kJkg^{-1})
k_G	overall heat transfer coefficient ($\text{Wm}^{-2}\text{K}^{-1}$)
$\dot{m}$	mass flow rate (kgs^{-1})
N	number
PHE	plate heat exchanger
$\dot{Q}$	heat (kW)
R	solar gain (Wm^{-2})
t	temperature (°C)
V	volume (m^3)
W	electricity (W)
w	specific humidity (kJkg^{-1})

Subscripts

<i>a</i>	air
<i>aux</i>	auxiliary
<i>c</i>	cooling
<i>aver</i>	average
<i>cen</i>	central
<i>cond</i>	condenser
<i>ch</i>	chilled
<i>cryst</i>	crystal
<i>dbo</i>	outdoor dry bulb
<i>e</i>	evaporator
<i>exg</i>	external gain
<i>fc</i>	fan coil
<i>g</i>	generator
<i>h</i>	hot
<i>he</i>	finned tube heat exchanger
<i>i</i>	inlet
<i>in</i>	indoor
<i>int</i>	internal
<i>lab</i>	laboratory
<i>light</i>	lighting
<i>max</i>	maximum
<i>min</i>	minimum
<i>o</i>	outlet
<i>out</i>	outdoor
<i>pers</i>	persons
<i>ren</i>	changes per hour

<i>sec</i>	security
<i>t</i>	thermal
<i>trans</i>	transmission
<i>vent</i>	ventilation
<i>w</i>	water

Greek symbols

	density (kgm^{-3})
η	efficiency

1. Introduction

Absorption and vapour compression technologies for space heating and cooling are based on the same principle: condensation and evaporation of a refrigerant. In compression systems the most common source of energy for the work required to compress the vapour is *electricity*, whereas in absorption systems it is *heat*, fed directly to the generator. The heat required may be provided by low pressure steam, residual heat, solar energy or directly from fossil fuel.

Single effect absorption machines, where the solution circulates in the generator at temperatures ranging from 70 °C to 120 °C, may be air- or water-cooled. Water-cooled facilities have been found to have two drawbacks: the risk of legionnaire's disease and, in residential areas, the difficulty of accommodating the equipment because it includes the cooling tower. In air-cooled systems where heat transfer is lower, higher condensation temperatures are reached, necessitating larger heat exchangers. Such systems are likewise subject to operational limits imposed by the risk of crystallization.

This obstacle has prompted the use of other solutions exhibiting higher resistance to crystallization [1].

The absorber is a key component in absorption chiller design for the impact of its performance on system efficiency, particularly when the ambient temperature is high. Substantial effort has been devoted to improving absorbers in recent years, in an attempt to reduce their size by optimizing heat and mass transfer. Film [2 - 6], drop [6 - 8] and bubble absorbers [9 - 11] as well as rotary absorption [12-13] have all been successfully used to this end.

M. Izquierdo et al. [14] proved that single effect absorption chillers are less efficient, in terms of primary energy, than air-cooled vapour compression facilities. Consequently, the former are unsatisfactory when fired with conventional fuel. They do constitute an alternative, however, at times of peak electricity demand when powered by solar energy. A. Syed et al. [15] studied the performance of a water-cooled single effect LiBr-H₂O absorption chiller using solar hot water as the source of heat. They found the technology to be more efficient in hot, dry climates with large variations in relative humidity and dry bulb temperature. M. Izquierdo et al. [16] studied an air-cooled double stage absorption system fitted with flat solar collectors to supply heat to the generator and compared the results to the performance of a water-cooled single effect absorption chiller. They found that the condensation temperature reached in the former was 53 °C, whereas in the single effect chiller, in which heat was also supplied by flat solar collectors, condensation temperatures of over 45 °C had to be ruled out due to solution crystallization.

F. Asdrubali et al. [17] conducted an experimental study with a water-cooled single effect absorption chiller in which the required heat was obtained from flat solar collectors. The maximum COP was obtained when the hot water temperature rose to 70 °C, making this an appropriate technology for use with flat solar collectors as the

source of energy. Until 2005, all the single effect absorption chillers on the market, such as those manufactured by Yazaki® and others, were water-cooled. Since that year, Rotartica® has been manufacturing air-cooled machines.

Trials were conducted in August 2005 to determine the highest outdoor dry bulb temperature for which the commercial (Rotartica 045v), air-cooled single effect LiBr/H₂O absorption chiller, is able to produce cool water for air conditioning use. The reader can find more information about the development and history of the machine in the references [13] and [18].

2. Experimental set-up

The chiller used and the experimental procedure followed are described below.

2.1. Air-cooled single effect absorption chiller

The Rotartica model 045v 4.5-kW chiller (Figure 1) is an indirect fired single effect system using LiBr-H₂O as the working fluid. The system is based on rotary absorption technology. Its main components are: the generator, evaporator and condenser-absorber unit, all installed inside a rotary drum. The drum rotates to improve the heat transfer by means of the generation of a turbulent flow in the LiBr solution.

The condenser-absorber unit is cooled by a water-air finned tube heat exchanger and a fan. Absorption takes place in a vacuum-sealed vessel. The nominal electric power is 1200 W and the volume 0.93 m³. It is designed for use in the service sector and residential and office buildings.

The external systems consists of a heat source and three water circuits: the hot and chilled water circuits, and the cooling water circuit for the condenser-absorber unit. A simplified diagram of the facility is given in Figure 2.

2.2. Heat source circuit

The heat source comprises a thermal oil tank with 10-kW electrical resistances, a PHE and an AEG AM880KY4 pump. The thermal oil, heated by the resistances, is pumped to the PHE where it transfers heat to the water subsequently fed to the generator.

2.3. Hot water circuit

The water heated in the plate heat exchanger to a temperature of over 80 °C was delivered to the generator with the aid of a Grundfos UPS40-80R pump. In the generator, the heat from the hot water is used to separate the refrigerant by boiling the solution.

2.4. Chilled water circuit

The water chilled in the evaporator was delivered to the fan coil by a built-in pump. The purpose of the fan coil, a water-air heat exchanger placed inside the laboratory to be cooled, is to cool the indoor air with the chilled water from the evaporator.

2.5. Cooling water circuit

The heat is rejected to the environment through a water pump, finned tube heat exchanger and fan. The cooling water that cools the condenser and absorber is itself cooled in the heat exchanger, from which the absorption and condensation heat is driven by a fan to the outdoor atmosphere.

2.6. *Experimental procedure*

The experimental study was conducted at the Eduardo Torroja Institute Heat Pump Laboratory at La Poveda, Arganda, Madrid, between 1st and 20th of August 2005. Meteodata 285 weather station outdoor dry bulb temperatures (t_{dbo}) were used throughout.

PT100 sensors, with an accuracy of ± 0.1 °C, were positioned at component inlets and outlets to measure air and water temperatures, while three ultrasonic flowmeters were used to measure water flow in the facility. All pipes were insulated with 15-mm thick fibreglass.

According to manufacturer specifications, hot, chilled and cooling water flow rates of over 10 l/min, 20 l/min and 25 l/min, respectively, were required. The chiller began to refrigerate when demand was generated in the laboratory, the hot water temperature was higher than 80 °C and the flow rates in the three water circuits were within manufacturer-specified operating ranges.

3. **Outdoor temperatures**

The daily outdoor dry bulb temperatures t_{dbo} for the period studied are shown in Figure 3. The high for the period, 41.3 °C, was reached on 07/08/2005.

Inasmuch as Madrid's summertime t_{dbo} s vary widely, from 25 to 41.3 °C, three representative days were chosen for more detailed analysis. Fig. 4 shows the daily outdoor dry bulb temperatures recorded for the three days chosen, defined as follows:

- Warm, with temperatures under 30 °C (10/08/05)
- Hot, with temperatures between 30 and 36 °C (04/08/05)
- Very hot, with temperatures between 36 and 41.3 °C (07/08/05)

4. Thermal load

The space to be refrigerated, part of the Heat Pump Laboratory, has a net floorspace of 40 m² and a volume of 120 m³. The thermal load Q_t (Eq. (1)) was calculated from the energy balance set out in equations (2)-(5). These equations respectively define the transmission, air change, external heat gain and internal loads.

$$Q_t = Q_{trans} + Q_{vent} + Q_{exg} + Q_{in} \quad (1)$$

$$Q_{trans} = k_G \cdot A_{lab} \cdot (t_{dbo} - t_{ind}) \quad (2)$$

$$Q_{vent} = V \cdot N_{ren} \cdot \rho_a \cdot C_{p_a} \cdot (t_{dbo} - t_{ind}) + V \cdot N_{ren} \cdot \rho_a \cdot h_{fg} \cdot (w_{out} - w_{ind}) \quad (3)$$

$$Q_{exg} = A_{cryst} \cdot R \quad (4)$$

$$Q_{in} = Q_{pers} \cdot N_{pers} \cdot f + q_{light} \cdot A_{light} + Q_{equip} \cdot N_{equip} \quad (5)$$

Thermal load, Q_t , was assumed to be generated only during working hours (9:00 a.m. to 8:00 p.m.) and when the temperature inside the laboratory was over 24 °C. The UA factor for the space to be refrigerated was 252 W/K. The space is occupied by three people for 10 hours a day. The coincident factor was assumed to be $f=0.75$.

The thermal load observed for a comfortable indoor temperature of 24 °C at the maximum t_{db0} recorded (42 °C) was 5.5 kW.

The demand for the three days selected is shown in Figure 5: 10.4 kWh for 10/08/05; 40 kWh for 04/08/05 and 50 kWh for 07/08/05.

Figure 6 shows the demand for August calculated with equations (2) – (5). The total demand for the period was 1.49 MWh.

5. Coefficient of Performance

The coefficient of performance (equation 7) is defined to be the ratio between the heat removed by the evaporator and the energy supplied to the generator:

$$COP = \frac{Q_e}{Q_g} \quad (7)$$

If the electric power used is included in the equation, COP is calculated as:

$$COP = \frac{Q_e}{Q_g + Q_{aux}} \quad (8)$$

$$Q_{aux} = \frac{W}{\eta_{cen}} \quad (9)$$

where W is the power demand corresponding to auxiliary equipment.

6. Results and discussion

The flow in each water circuit was held constant while the chiller was in operation. The values recorded for the hot, chilled and cooling water circuits were 0.28

kg/s, 0.53 kg/s and 0.67 kg/s, respectively, i.e., all within the manufacturer's specified operating range (Figure 7).

6.1. Typical days

The experimental results for the generator, the evaporator, the fan coil and the finned tube heat exchanger on the three days selected for the trial are set out below, together with the daily *COP*.

- Warm day: 10/08/05

The air and water inlet and outlet temperatures recorded on the warm day in the generator, fan coil and finned tube heat exchanger, are given in Figure 8.

The hot water temperature in the generator ranged from 85 °C to 99 °C, with a difference of approximately 5 °C between the inlet and outlet temperatures. These variations in the hot water inlet temperature in the generator, t_{iwg} , affected the chilled water outlet temperature in the evaporator, t_{owe} .

At a dry bulb temperature of 26.7 °C, the chilling power found for the evaporator was 5.6 kW. Figure 5 shows that the thermal load was consistently lower than 3 kW all day. Since the pipe running between the evaporator and the fan coil was insulated, t_{owe} and t_{iwfc} were assumed to be approximately equal. The inlet water temperature in the fan coil, t_{iwfc} , and the inlet water temperature in the generator, t_{iwg} , were found to be inversely related. Moreover, for $t_{dbo} = 26.7$ °C and $t_{iwg} = 99$ °C, t_{iwfc} was 12.7 °C and $t_{owfc} = 14.8$ °C.

At $t_{dbo} = 26.7$ °C, the cooling water inlet temperature in the finned tube heat exchanger was 36.9 °C while the air temperature rose by approximately 9 °C.

- *Hot day: 04/08/05*

The air and water inlet and outlet temperatures recorded on the hot day in the generator, fan coil and finned tube heat exchanger are given in Figure 9.

The hot water temperature in the generator ranged from 91 °C to 105.4 °C, with a difference of approximately 5 °C between the inlet and outlet temperatures. The generator temperatures rose as a result of the increase in the cooling water (absorber-condenser unit) temperatures.

At dry bulb temperatures of 28.9 °C and 35.6 °C, the chilling power found for the evaporator came to 5.1 and 4.2 kW, respectively. Figure 5 shows that the thermal load increased during the day, peaking at 4.5 kW between 15:00h and 18:00 h solar time. Figure 9 shows that t_{iwc} rose with Q_t and steadied when Q_t stabilized. Moreover, for $t_{dbo} = 28.9$ °C and $t_{iwg} = 97$ °C, t_{iwc} was 14.9 °C and t_{owc} 17.2 °C; and for $t_{dbo} = 35.6$ °C and $t_{iwg} = 105.4$ °C, t_{iwc} was 17.5 °C and t_{owc} 19.4 °C. The indoor temperature in the laboratory remained constant at about 23.5 °C throughout the day.

In the finned tube heat exchanger, at $t_{dbo} = 35.6$ °C, the cooling water inlet temperature was 45.6 °C, while the air temperature rose by approximately 9 °C.

- *Very hot day: 07/08/05*

The air and water inlet and outlet temperatures recorded on the very hot day in the generator, fan coil and finned tube heat exchanger are given in Figure 10. As this was the hottest day in the period studied, the condensation-absorption temperatures recorded were also the highest (Figure 10). The resulting rise in the solution temperature in the generator raised the hot water temperature from 86.5 to 107.2 °C, with a difference of approximately 5 °C between the inlet and outlet temperatures.

At dry bulb temperatures of 28.6 °C and 41.3 °C, the chilling power found for the evaporator came to 5.4 and 3.62 kW, respectively. By 12:00 h solar time, Q_t rose to 4.5 kW, during which time t_{iwc} also increased. Moreover, for $t_{db} = 28.6$ °C and $t_{iwg} = 94.5$ °C, t_{iwc} was 17.3 °C and t_{owc} 19.5 °C. The indoor temperature in the laboratory remained constant at 25.5 °C. Q_t continued upward after 12:00 h solar time, peaking at 5.5 kW. Between 12:00h and 18:00 h solar time, for $t_{db} = 41.3$ °C and $t_{iwg} = 104.1$ °C, t_{iwc} was 25.8 °C and t_{owc} 26 °C. The indoor temperature in the laboratory was 29 °C.

At an outdoor dry bulb temperature of 41.3 °C, the cooling water inlet temperature in the finned tube heat exchanger was 51.3 °C, while the air temperature rose by 8 °C.

- *Daily COP*

The daily COPs on the 10th, the 4th and the 7th of August were respectively 0.64, 0.52 and 0.42.

6.2 . *Period overall*

The results for the entire experimental period are discussed below.

6.2.1. *Chilling power*

The experimental results for the evaporator are represented in Figure 11. For outdoor dry bulb temperatures of from 22.9 to 41.3 °C, the chilling power values obtained ranged from 5.78 to 3.62 kW. This downward trend in chilling temperature with rising dry bulb temperatures is clearly visible in the figure.

6.2.2. Chilled water temperature

The chilled water outlet temperatures in the evaporator are charted against t_{dbo} in Figure 12. The temperature distribution shown in this figure was obtained with t_{iwg} values of from 80 to 107 °C. The following may be deduced from the data set (231 points) plotted in Figure 12:

- For ($25.3\text{ °C} < t_{dbo} \leq 30\text{ °C}$); $t_{owe} > 15\text{ °C}$ for 18.2% of the data
- For ($30\text{ °C} < t_{dbo} \leq 35\text{ °C}$); $t_{owe} > 15\text{ °C}$ for 21.9% of the data.
- For ($35\text{ °C} < t_{dbo} < 41.3\text{ °C}$); $t_{owe} > 15\text{ °C}$ for 81.4% of the data.

For example, the first case means that, for $25.3\text{ °C} < t_{dbo} < 30\text{ °C}$, $t_{owe} < 15\text{ °C}$ for 81.8% of the data.

The lowest chilled water temperature recorded, 11.3 °C (22/08/2005), was found for an outdoor dry bulb temperature of 29.9 °C and a t_{iwg} value of 100.5 °C. The chilling power was 4.67 kW. Chilled water outlet temperatures ranged from 11.3 to 24.3 °C. The air flow rate needed to cover thermal demand increased with rising t_{owe} , which had an adverse effect on comfort.

6.2.3. Energy balance

The energy balance was calculated for the evaporator and generator. Heating and chilling power can be expressed as follows:

$$\dot{Q}_g = m_w \cdot C_{p_w} \cdot (t_{iwg} - t_{owg}) \quad (10)$$

$$\dot{Q}_e = m_w \cdot C_{p_w} \cdot (t_{iwe} - t_{owe}) \quad (11)$$

Daily values, which depend on the cooling demand, are shown in Figure 13. The heat supplied to the generator came to $Q_g=1085.5$ kWh and the heat removed in the evaporator to $Q_e= 534.5$ kWh.

6.2.4. Seasonal coefficient of performance

As Figure 14 shows, COP declined with increasing t_{dbo} . The value for the period as a whole was 0.49.

While the chiller tested had 1200W of electrical power at terminals, the value measured during the experiments was 900 W. Water pumps and absorber-condenser fan electric consume during the period came to 112.5 kWh. On the grounds of this electric demand, Q_{aux} , and of the overall generating efficiency of the Spanish electricity system, $\eta_{cen} = 0.33$, COP was computed to be 0.37.

7. Results in other regions

The t_{dbo} values for the 4th, the 7th and the 10th of August used in the present study were recorded in a town a few miles southeast of Madrid. Such values are likewise representative of summer temperatures in other regions of Spain, however. The 10th and the 4th of August outdoor dry bulb temperatures, for instance, can also be found on different dates in cities like Vigo, Mallorca and Cadiz.

Consequently, the results found for typical summer days in Madrid can be extrapolated to other cities with a similar climate.

8. Conclusions

Experimental research was conducted to determine the performance of a 4.5-kW air-cooled chiller, single effect LiBr/H₂O absorption unit. (Rotartica 045v) The experiments were run at the Eduardo Torroja Institute Heat Pump and Absorption Chiller Laboratory at La Poveda, Arganda del Rey, Madrid, in August 2005. Measurements were recorded over a 20-day period. The hot water inlet temperature in the generator varied throughout the day from 80 °C to 107 °C. Daily and period energy flows are given, along with the energy balance for the facility.

The total energy supplied to the generator came to 1085.5 kWh and the heat removed in the evaporator to 534.5 kWh. The average COP for the period as a whole was 0.49. When the electric power used by auxiliary equipment was factored into the equation, primary energy based COP came to 0.37. The results obtained for Madrid can be extrapolated to other regions of Spain with $t_{db,o}$ s lower than 35 °C.

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Fig .1 Rotartica 045v.

Fig. 2. Scheme of heat source and interior of the machine.

Fig. 3. Maximum, average and minimum daily dry bulb outdoor temperatures on August 2005.

Fig. 4. Dry bulb outdoor temperature for three days selected of August

Fig. 5. Thermal load for the three characteristic days.

Fig. 6. Daily demand of August

Fig. 7. Flow rates corresponding to hot, cooling and chilled water loop.

Fig. 8. Left scale: Inlet and outlet temperatures in the generator. Right scale: fancoil and finned tubes heat exchanger (10/08/05).

Fig. 9. Left scale: Inlet and outlet temperatures in the generator. Right scale: fancoil and finned tubes heat exchanger (04/08/05).

Fig. 10. Left scale: Inlet and outlet temperatures in the generator. Right scale: fancoil and fin tubes heat exchanger (07/08/06).

Fig. 11. Chilling power and dry bulb outdoor temperature

Fig. 12. Outlet chilled water temperature in the evaporator

Fig. 13. Heat supplied to the generator and cold produced in the evaporator.

Fig. 14. COP as a function of dry bulb outdoor temperature.