

The Optimal Match of Streams for Maximum Heat Transfer from a Gas Fired Absorption Refrigeration Unit

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ABSTRACT

In this study, a 5 tons of refrigeration (60,000 BTU h^{-1}) commercial absorption refrigeration unit was characterized and instrumented, and a simplified thermal and exergetic analysis of the system was performed, aiming the optimization of external operating parameters for maximum thermodynamic performance. The first and second law of thermodynamics were used to evaluate the energy (first law) and the exergy (second law) efficiencies of the system. The experimental results showed the existence of a double maximum for the thermal and exergetic efficiencies for the optimized unit with respect to the water mass flow rates of the cold and hot sides of the absorption refrigerator. Maximum variations of 30 % and 44 % in the first and second law efficiencies, respectively, were observed according to the mass flow rate range used, which stresses the importance of the optima found for maximum thermodynamic performance, and therefore minimum energy consumption in actual engineering applications.

NOMENCLATURE

В	bias limit	
c	specific heat, $kJ kg^{-1} K^{-1} (BTU lb^{-1} \circ R^{-1})$	
ī	average specific heat of the water/ethylene-glycol	mixture,
	$kJ kg^{-1} K^{-1} (BTU lb^{-1} R^{-1})$	
ex	specific flow exergy, $kJ kg^{-1} (BTU lb^{-1})$	
Ė	exergy rate, $kW(BTU h^{-1})$	
h	specific enthalpy, $kJ kg^{-1} (BTU lb^{-1})$	
LHV	lower heating value, $kJ kg^{-1} (BTU lb^{-1})$	

ṁ	mass flow rate, kg s ^{-1} (lb s ^{-1})
Р	pressure, Pa (psi)
Pl	precision limit
Ż	heat transfer rate, $kW(BTU h^{-1})$
S	specific entropy, kJ kg ⁻¹ K ⁻¹ (BTU lb ⁻¹ °R ⁻¹)
Т	temperature, $K(^{\circ}R)$
U	experimental uncertainty of measurements
v	specific volume, $m^3 kg^{-1}(ft^3 lb^{-1})$
$\dot{\forall}$	volumetric flow rate, $m^3 h^{-1}(ft^3 min^{-1} \text{ or CFM})$

Greek symbols

$\dot{\forall}$	volumetric flow rate, $m^3 h^{-1}(ft^3 min^{-1} \text{ or CFM})$
Greek symbols	
Δh	specific enthalpy variation, $kJ kg^{-1} (BTU lb^{-1})$
Δs	specific entropy variation, kJ kg ⁻¹ K ⁻¹ (BTU lb ⁻¹ °R ⁻¹)
ΔΤ	temperature variation, $K(^{\circ}R)$
η	efficiency
ρ	gas density, kg m ⁻³ (lb ft ⁻³)
ξ_{ch}	chemical exergy, $kJ kg^{-1} (BTU lb^{-1})$

Subscripts

0	reference state
ch	chemical
comb	combined hot and cold systems
С	cold fluid
eg	ethylene-glycol
f	a given fluid

Н	hot fluid
i	inlet
Ι	related to the first law of thermodynamics
Π	related to the second law of thermodynamics
LPG	liquid petroleum gas
m	maximum
m,m	double maximum
0	outlet
opt	optimum
S	hot or cold systems
W	water

INTRODUCTION

In the last few decades, due to the increasing level of pollution worldwide and the cost of energy, the search for maximum exploitation of the available energy has lead to the development and use of cogeneration or trigeneration systems. Heating, ventilation, air-conditioning, and refrigeration systems (HVAC-R) play a major role in modern society energy consumption. These systems are mostly based on the vapor compression cycle, due to high efficiency, but the vapor compression cycle needs work input, and high energy consumption is still observed, therefore research efforts have been made to develop intelligent refrigeration systems in order to reduce energy consumption (Vargas and Parise 1995; Buzelin et al. 2005). Hence, alternative HVAC-R systems have been the subject of much recent scientific research. Among these systems, absorption refrigeration is receiving great attention since it may produce energy, heat and cold, using, as energy source, waste heat from industrial processes or, for instance, exhaust gases in automobiles (Temir and Bilge 2004).

The major companies working on this area focus on large capacity absorption systems, i.e. above 100 TR. However, since most refrigeration and air-cooling units are of small capacity and operate based on vapor compression cycle systems, there is still a vast field in which absorption systems could be employed.

An absorption system also allows the direct use of primary energy, particularly solar energy and natural gas, for refrigeration purposes (Ezzine et al. 2004). Although this system is less costly and simpler than vapor compression systems, its comparatively low coefficient of performance has limited its use to few and specific applications. Nevertheless, the absorption refrigeration system may reach a refrigeration capacity higher than that of a vapor compression system when energy sources such as waste (residual) heat from industrial processes, gas or vapor turbines, sunlight or biomass are used instead of electricity (Adewusi and Zubair 2004).

The performance of absorption systems is dependent on an adequate choice of the refrigerant/sorbent working pair, and ammonia-water has been receiving great attention since these fluids do not contribute to the greenhouse effect (Bruno et al. 1999; Lazzarin et al. 1996).

The technical literature is rich in publications on the absorption refrigeration field. Particularly, Abreu (1999) and Villela and Silveira (2005) used as heat source for absorption systems, the combustion of liquid petroleum gas (LPG) and biogas, respectively, studying the design and performing a thermoeconomic analysis of the analized systems. Other studies focused on the exergy analysis of absorption refrigeration systems, including Sedighi et al. (2007), Hasabnis and Bhagwat (2007), Khaliq and Kumar (2007), Arivazhagan et al. (2006), and Sencan et al. (2005). Simulation and optimization studies have also been published analyzing the absorption refrigeration system in isolation (Vargas et al. 1996; Vargas et al. 2000a; Vargas et al. 2001). However, the exergy analysis and optimization of an absorption refrigerator to produce cooling and heating, based on a theoretical-experimental model, could not be found in the open literature.

The aim of this work is two-fold: i) to formulate theoretically the absorption system heat transfer interactions using a simplified mathematical model for the energy and exergy analysis of an existing LPG (gas fired) driven absorption refrigeration unit, and ii) based on experimental measurements, to characterize system pull-down times and to carry out an energetic and exergetic optimization for maximum thermodynamic performance of the system, i.e., minimum energy consumption.

THEORY

As illustrated in the introduction, many studies have been published in the literature that dealt with simple and complex internal component-wise energetic and exergetic analyses of absorption refrigerator systems. So, it is not within the scope of this study to investigate the absorption cycle itself, but to address how to extract the most from existing absorption units (e.g., single-effect, double-effect, generator-absorber heat exchange-GAX cycle) by investigating the possibility of optimally tuning some of their external operating parameters. Such study can be quite challenging, since the thermodynamic fundamentals are far from obvious when the objective is to identify a general optimization opportunity and to pinpoint the candidate parameters to be optimized. For that, simple models that include the basic thermodynamic phenomena of the system to be analyzed are recommended, so that the optimization opportunities are visible and expected to be present in actual systems, no matter how complex they may be (Bejan 1988). Therefore, the simplest possible mathematical formulation is proposed in this section to be applied together with experimental measurements to achieve the objectives of this study.

Energy analysis of the absorption system deliverables

As the absorption system may simultaneously operate for water cooling and heating, both cold and hot water reservoirs were considered part of the system, each requiring an energy balance. The cooling and heating capacity rates are functions of the design and operating parameters, as well as the water mass flow rates. For the purpose of modeling the existing absorption unit utilized in this work, that uses shell and tube heat exchangers, with the refrigerant circulating in the inner tubes and the cooled fluid around them, the heat transfer rate actually extracted by the refrigerant in the evaporator from the water/ethylene-glycol mixture in the system cold side (\dot{Q}_{c}) is calculated as follows:

$$\dot{Q}_{\rm C} = \dot{m}_{\rm C} \bar{c}_{\rm C} \Delta T_{\rm C} \tag{1}$$

where: \dot{m}_{C} and \bar{c}_{C} are the cold mass flow rate and the average specific heat of the water/ethylene-glycol mixture (75/25 in weight), respectively. The latter may be calculated by $\bar{c}_{C} = 0.25 c_{eg} + 0.75 c_{w}$, for $c_{w} = 4.186 \text{ kJ kg}^{-1} \text{ K}^{-1} (1 \text{ BTU lb}^{-1} \text{ °R}^{-1})$ and $c_{eg} = 2.391 \text{ kJ kg}^{-1} \text{ K}^{-1} (0.571 \text{ BTU lb}^{-1} \text{ °R}^{-1})$.

The temperature variation of the water/ethylene-glycol mixture, ΔT_C , is defined as $\Delta T_C = T_{C,o} - T_{C,i}$,

where $T_{C,o}$ and $T_{C,i}$ are, respectively, the outlet and inlet temperature of the mixture at the evaporator. As mentioned earlier in the text, the existing absorption unit utilized in this work uses shell and tube heat exchangers, with the refrigerant/absorbent solution circulating in the inner tubes and water (condenser and absorber) or water/ethylene-glycol mixture (evaporator) around them. Therefore, the heat transfer rate actually captured by the water that circulates in series in the condenser and absorber in the system hot side, \dot{Q}_{H} , is calculated by:

$$\dot{Q}_{\rm H} = \dot{m}_{\rm H} c_{\rm H} \Delta T_{\rm H}$$
⁽²⁾

where: $\dot{m}_{\rm H}$ is the hot water mass flow rate, $\Delta T_{\rm H}$ is the temperature variation of the water in the system hot side (absorber and condenser), i.e. $\Delta T_{\rm H} = T_{\rm H,o} - T_{\rm H,i}$, and $c_{\rm H} = c_{\rm w}$, since in this case the fluid is pure water.

In this work, combustion of liquid petroleum gas (LPG) was used as the heat source and the fuel heat transfer input rate, \dot{Q}_{LPG} , was calculated as follows:

$$\dot{Q}_{LPG} = \dot{m}_{LPG} LHV_{LPG}$$
(3)

where: \dot{m}_{LPG} is the fuel mass flow rate and LHV_{LPG} , the fuel lower heating value, which was taken at 25 °C (77 °F), 1 atm (14.7 psi).

The performance of the system may be evaluated by considering the first law (energetic) efficiencies for both systems ($\eta_{I,C}$ and $\eta_{I,H}$), and the herein defined combined system first law (energetic) efficiency ($\eta_{I,comb}$) that recognizes the cooled water/ethylene glycol mixture in the evaporator and the heated water in the absorber and condenser as valuable products to be used in practical applications (e.g., refrigeration, industrial processes), which are calculated as follows:

$$\eta_{I,C} = \frac{Q_C}{\dot{Q}_{LPG}}$$
(4)

$$\eta_{I,H} = \frac{\dot{Q}_H}{\dot{Q}_{LPG}}$$
(5)

$$\eta_{I,\text{comb}} = \frac{\dot{Q}_{C} + \dot{Q}_{H}}{\dot{Q}_{LPG}}$$
(6)

Exergy analysis of the absorption system deliverables

The exergy content of the heat transfer rates of the system cold and hot sides were calculated as follows:

$$\dot{E}_{C} = \dot{m}_{C} \left(ex_{C,o} - ex_{C,i} \right) \qquad \dot{E}_{C} = \dot{m}_{C} \left[\Delta h_{C} - T_{0} \left(\Delta s_{C} \right) \right] \qquad (7)$$

$$\dot{E}_{H} = \dot{m}_{H} \left(ex_{H,o} - ex_{H,i} \right) \qquad \dot{E}_{H} = \dot{m}_{H} \left[\Delta h_{H} - T_{0} \left(\Delta s_{H} \right) \right] \qquad (8)$$

where $ex_{f,o}$ and $ex_{f,i}$ are, respectively, the specific flow exergy of a particular fluid f at the outlet and at the inlet. The specific flow exergy of a particular fluid is given by (Bejan 1988):

$$ex_{f} = (h_{f} - h_{0}) - T_{0}(s_{f} - s_{0})$$
(9)

The reference temperature T_0 used in this study was 298.15 K and the variations of enthalpy and entropy were calculated using the incompressible fluid model, i.e. $dh_f = (cdT + vdP)_f \cong c_f dT_f$, assuming

a constant pressure process. Therefore
$$\Delta h_f = \int_i^o dh_f = c_f \Delta T_f$$
, and $ds_f = \left(\frac{c}{T} dT\right)_f$, thus

$$\Delta s_{f} = \int_{i}^{o} ds_{f} = c_{f} \ln \frac{T_{f,o}}{T_{f,i}}.$$

The fuel exergy input rate (\dot{E}_{LPG}) was calculated using the fuel mass flow rate (\dot{m}_{LPG}) and its chemical exergy ($\xi_{ch,LPG}$) as $\dot{E}_{LPG} = \dot{m}_{LPG} \xi_{ch,LPG}$. The second law (exergetic) efficiencies for both sides ($\eta_{II,C}$ and $\eta_{II,H}$), and the combined system second law (exergetic) efficiency ($\eta_{II,comb}$) are given by:

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$$\eta_{\rm II,C} = \frac{\dot{\rm E}_{\rm C}}{\dot{\rm E}_{\rm LPG}} \tag{10}$$

$$\eta_{\mathrm{II},\mathrm{H}} = \frac{\mathrm{E}_{\mathrm{H}}}{\mathrm{\dot{E}}_{\mathrm{LPG}}} \tag{11}$$

$$\eta_{\rm II,comb} = \frac{E_{\rm C} + E_{\rm H}}{\dot{E}_{\rm LPG}}$$
(12)

The combined system second law (exergetic) efficiency ($\eta_{II,comb}$) defined by Eq. (12) recognizes the cooled water/ethylene glycol mixture in the evaporator and the heated water in the absorber and condenser as valuable products to be used in practical applications (e.g., refrigeration, industrial processes).

EXPERIMENTS

In this work, a single-effect commercial water-ammonia absorption system was used, having nominal cooling and heating capacity rates of 17.5 and 35 kW (60,000 and 120,000 BTU h^{-1}). The system is capable of operating with a minimum chilled water/ethylene glycol mixture temperature of -5 °C (23 °F) and a maximum heating water temperature of 60 °C (140 °F).

Fig. 1 shows a partial view of the experimental unit. Since the system delivers two products, cooling and heating, two water reservoirs were used. The hot side reservoir was a large capacity tank (50 m³ – 1765.75 ft³) located underneath the experimental setup, so that a temperature reservoir was provided, i.e., the reservoir water temperature remained constant during the experiments, whereas the cold side container was a 300-liter (79.252-gal) tank, filled with a water/ethylene-glycol mixture, 75/25 in weight, in which a 4000 W (13,652 BTU h⁻¹) thermostat controlled electrical resistance was installed. The electrical resistance was used only to establish the same chilled water/ethylene glycol mixture initial temperature for all runs, and then switched off, for the purpose of pull-down time comparisons in different operating conditions. So, the electrical resistance was not turned on together with the absorption system in this study. A 0.5 HP pump was used to drive the flow in the cold and hot circuits.

The heat source of the system was the direct combustion of LPG, fed by a system of 4 LPG cylinders. LPG may be considered a mixture of propane (C_3H_8) and butane (C_4H_{10}), 50/50 in weight, and its

properties may be found in Bejan (1988). Considering that the lower heating value (LHV) of propane and butane are 2044.0 kJ mol⁻¹ (1937.3 BTU mol⁻¹) and 2658.5 kJ mol⁻¹ (2519.73 BTU mol⁻¹), at 25 °C (77 °F), 1 atm (14.7 psi), respectively, and that their chemical exergy are 2149.0 kJ mol⁻¹ (2036.82 BTU mol⁻¹) and 2802.5 kJ mol⁻¹ (2656.2 BTU mol⁻¹), respectively, the calculated lower heating value of LPG (LHV_{LPG}) is 46008.2 kJ kg⁻¹ (19783.53 BTU lb⁻¹) and its chemical exergy ($\xi_{ch,LPG}$) is 48444.4 kJ kg⁻¹ (20830.92 BTU lb⁻¹).



Figure 1. Water-ammonia absorption experimental unit.

The experimental work involved data acquisition in real time. This task was performed through the utilization of a computational data acquisition system which consisted of a digital multimeter board, an analog output board, a 32-channel high voltage multiplexer, and accessories, which allows for the sequential data acquisition from 32 channels at interval times of 0.1 s. All the data were processed by a home made platform application to convert the sensors signals in readable temperatures. In this way, the temperature measurements originated from the thermistors utilized in the experiments were read almost simultaneously, i.e., in less than a second. With the equipment described in this section, it was possible to measure all required temperatures almost simultaneously.

For the temperature measurements, high precision thermistors were used after calibration. The calculated bias limit of the sensor was 0.001°C (Dally et al. 1993; Howle et al. 1992). Two flowmeters, were used to measure flow rates, with a bias limit of ± 2 % and 1 % repeatability. The LPG consumption was monitored with a gas volume meter, with a bias limit of ± 0.016 m³ (± 0.565 ft³). The water mass flow rate and LPG consumption measurements were manually recorded. Figure 2 shows the positioning of all temperature and flow rate readers used for monitoring the system, along with the water and LPG valves, named V_w and V_{LPG}, respectively.

Pull-down characterization

The performance of the refrigeration system was evaluated using pull-down tests, which consisted of measuring the time required for the temperature of the water/ethylene-glycol mixture to fall from a reference (initial) temperature to a desired setpoint band steady state. These tests were carried out using the cooling circuit only, and the water tank initial temperature and the outlet cold temperature setpoint, i.e. at the evaporator outlet, were selected as 30 ± 1 °C and 10 ± 1 °C, respectively. The thermostat controlled electrical resistance that was installed in the 300-litre tank established the water tank initial temperature and the turned off during the entire test, and the refrigerator control system controlled the desired evaporator outlet temperature setpoint band.

Energetic and exergetic analyses

In all tests, the absorption refrigeration system was conducted to the established setpoint band steady state detailed in the previous paragraph. The experiments were carried out using various cooling ($\dot{m}_{\rm C} = 0.37, 0.50, 0.67$ and 0.83 kg s⁻¹ or $\dot{m}_{\rm C} = 0.81, 1.1, 1.47$ and 1.8 lb s⁻¹) and heating ($\dot{m}_{\rm H} = 0.40, 0.50, 0.67, 0.83, 0.90$ kg s⁻¹ or $\dot{m}_{\rm H} = 0.88, 1.1, 1.47, 1.8$ and 2 lb s⁻¹) fluids mass flow rates with the objective of investigating the existence of an optimal pair ($\dot{m}_{\rm C}, \dot{m}_{\rm H}$)_{opt} for maximum system first and second law efficiencies within those mass flow rates tested ranges. Three experiments were carried out for each set of experimental parameters.



Figure 2. Schematics of the absorption system and positioning of the temperature and flow rate

readers.

Calculation of uncertainties

In this work, the experimental measurement of temperatures, flow rates and LPG volume were used to calculate heat transfer and exergy rates, first and second law efficiencies. For each experimental condition, 3 measurements were taken at steady state. The precision limit $(Pl_{\Delta T_s})$ for every temperature variation (ΔT_s) was estimated as twice the standard deviation of these measurements, with a confidence level of 95 %. The related uncertainties were then calculated as follows:

$$\frac{\mathbf{U}_{\Delta \mathrm{T}_{\mathrm{s}}}}{\Delta \mathrm{T}_{\mathrm{s}}} = \left[\left(\frac{\mathrm{Pl}_{\Delta \mathrm{T}_{\mathrm{s}}}}{\Delta \mathrm{T}_{\mathrm{s}}} \right)^{2} + \left(\frac{\mathrm{B}_{\Delta \mathrm{T}_{\mathrm{s}}}}{\Delta \mathrm{T}_{\mathrm{s}}} \right)^{2} \right]^{\frac{1}{2}} \cong \frac{\mathrm{Pl}_{\Delta \mathrm{T}_{\mathrm{s}}}}{\Delta \mathrm{T}_{\mathrm{s}}}$$
(13)

where $B_{\Delta T_s}$, the bias limit of ΔT_s , was found to be much smaller than $Pl_{\Delta T_s}$, due to the fact that a high precision thermistor was used (resistance 2250 Ω at 25 °C), with reported bias limits as low as ± 0.001 K (Dally et al. 1993; Howle et al. 1992).

The precision limit of all variables used to determine heat transfer and exergy rates were negligible in comparison with the precision limit of the temperature variations (namely, ΔT_H and ΔT_C). Therefore, the uncertainties associated with the heat transfer and exergy rates were estimated as follows:

$$\frac{\mathbf{U}_{\dot{\mathbf{Q}}_{s}}}{\dot{\mathbf{Q}}_{s}} = \left[\left(\frac{\mathbf{Pl}_{\dot{\mathbf{m}}_{s}}}{\dot{\mathbf{m}}_{s}} \right)^{2} + \left(\frac{\mathbf{Pl}_{c_{s}}}{c_{s}} \right)^{2} + \left(\frac{\mathbf{Pl}_{\Delta T_{s}}}{\Delta T_{s}} \right)^{2} \right]^{\frac{1}{2}} \cong \frac{\mathbf{Pl}_{\Delta T_{s}}}{\Delta T_{s}}$$
(14)

$$\frac{\mathbf{U}_{\dot{\mathbf{E}}_{s}}}{\dot{\mathbf{E}}_{s}} = \left[\left(\frac{\mathbf{Pl}_{\dot{\mathbf{m}}_{s}}}{\dot{\mathbf{m}}_{s}} \right)^{2} + \left(\frac{\mathbf{Pl}_{c_{s}}}{c_{s}} \right)^{2} + \left(\frac{\mathbf{Pl}_{\Delta T_{s}}}{\Delta T_{s}} \right)^{2} + \left(\frac{\mathbf{Pl}_{\Delta \ln T}}{\Delta \ln T} \right)^{2} \right]^{\frac{1}{2}} \cong \frac{\mathbf{Pl}_{\Delta T_{s}}}{\Delta T_{s}}$$
(15)

where: $\left(\frac{Pl_{\Delta \ln T}}{\Delta \ln T}\right)$ is due to the calculation of entropy variation, being negligible in comparison with $\frac{Pl_{\Delta T_s}}{\Delta T_s}$,

and s represents either H or C, depending on what side of the system is being analyzed, i.e., the hot or cold side, respectively.

Analysis of Eqs. (13-15) shows that the calculation of uncertainties for all studied parameters is basically dependent on the temperature precision limit $Pl_{\Delta T_s}$. The largest calculated uncertainty in all tests was ± 0.059 (= 5.9 %) and therefore adopted for all error bars presented in the results of this work.

RESULTS AND DISCUSSION

In order to find the optimum operating conditions of the absorption refrigeration system, a series of tests was carried out varying the flow rates of the cold and hot systems. On these tests, two constraints of the equipments were known, the absorption system and the flow meter could only operate on the $0.30-0.90 \text{ kg s}^{-1}$ ($0.66-2 \text{ lb s}^{-1}$) and $0.17-0.93 \text{ kg s}^{-1}$ ($0.37-2.05 \text{ lb s}^{-1}$) ranges, respectively. Furthermore, although the refrigerator may operate in cooling and heating modes, it prioritizes the cooling mode.

For the first and second law analyses, the following considerations were used:

(i) The data were taken at steady-state;

(ii) The effect of the potential and kinetic energy variations were neglected with respect to the variation of the internal energy of the fluid;

(iii) For all tests performed, the initial condition was defined as $T_0 = 298.15 \text{ K} (537 \text{ }^{\circ}\text{R})$ and $P_0 = 101325 \text{ Pa} (14.7 \text{ psi});$

(iv) The loss and/or gain of heat, pressure and exergy on the cold and hot water pipes were neglected considering the magnitude of the variations on these quantities within the heat exchangers, and

(v) The LPG consumption (\dot{m}_{LPG}) was kept constant and equal to 0.000727 kg s⁻¹ (0.0016 lb s⁻¹). Thus,

for a gas density (ρ_{LPG}) of 2.5 kg m⁻³ (0.155 lb ft⁻³), the LPG volumetric flow rate used was

 $\dot{\forall}_{LPG} = 1.0475 \text{ m}^3 \text{ h}^{-1} (0.61654 \text{ ft}^3 \text{ min}^{-1}).$

Pull-down experiments

Pull-down tests are interesting to characterize the behavior of any refrigeration equipment, and in many engineering applications the refrigerator pull-down time is crucial for proper operation (e.g., food conservation). In Figure 3a, for a typical pull-down time determination experiment, the variation of $T_{C,o}$ and $T_{C,i}$ with respect to time is shown, defining the pull-down time as the time taken for the system to achieve the established setpoint band, i.e., $T_{c,o} = 10 \pm 1$ °C. In this case, for $\dot{m}_C = 0.83 \text{ kg s}^{-1}(1.8 \text{ lb s}^{-1})$ and $\dot{m}_H = 0.90 \text{ kg s}^{-1}(2 \text{ lb s}^{-1})$, the estimated pull-down time was approximately 24 min.

The investigation proceeds showing that the pull-down time is sensitive to the variation of the pair $(\dot{m}_{\rm C}, \dot{m}_{\rm H})$. With respect to Fig. 3a, Fig. 3b shows that the pull-down time reduces to approximately 16:40 min:s with the decrease in both mass flow rates $(\dot{m}_{\rm C} = 0.37 \text{ kg s}^{-1} \text{ or } 0.81 \text{ lb s}^{-1}, \dot{m}_{\rm H} = 0.5 \text{ kg s}^{-1} \text{ or } 1.1 \text{ lb s}^{-1})$. Conversely, also with respect to Fig. 3a, and keeping the same cooling water mass flow rate, $(\dot{m}_{\rm C} = 0.83 \text{ kg s}^{-1} \text{ or } 1.8 \text{ lb s}^{-1})$, Fig. 3c shows that the pull-down time increases to



approximately 30 min with the decrease in the heating water mass flow rate $(\dot{m}_{\rm H} = 0.83 \text{ kg s}^{-1} \text{ or } 1.8 \text{ lb s}^{-1}).$



Figure 3. Temperature variation of the water/ethylene-glycol mixture at the inlet and outlet of the evaporator during pull-down experiments carried out using: a) $(\dot{m}_{\rm C}, \dot{m}_{\rm H}) = (0.83, 0.90) \text{ kg s}^{-1}$ or

(1.8, 2) lb s⁻¹; b) (\dot{m}_{C}, \dot{m}_{H}) = (0.37, 0.50) kg s⁻¹ or (0.81, 1.1) lb s⁻¹ and c)

 $(\dot{m}_{\rm C}, \dot{m}_{\rm H}) = (0.83, 0.83) \text{ kg s}^{-1} \text{ or } (1.8, 1.8) \text{ lb s}^{-1}.$

The analysis of Fig. 3 indicates that there might be an optimal pair $(\dot{m}_{C}, \dot{m}_{H})_{opt}$ for minimum system pull-down time if the system thermal load is fixed, which is the present case, i.e., the amount of water/ethylene-glycol mixture to be cooled down in the tank is fixed (300 litres or 79.25 gal). The engineering designer is therefore free to select the project objective function. However, it is not within the scope of the present study to find such optima, since optimal thermodynamic performance is essentially the search for system optimal conditions to deliver maximum exergy output, for a fixed exergy input, which is hereafter pursued. Note that the optima found under such different objective functions will not necessarily coincide. If the thermal load is not fixed (e.g., in the case of delivering a continuous cooling water mass flow rate as in a chiller, with a known high temperature input), the optimum will disappear. This happens because as the mass flow rate decreases pull-down time will decrease as well, since thermal load decreases.

Energetic analysis

The results of the experiments under different operating conditions were used to measure ΔT_{c} , which were then used to estimate efficiencies, as described in the theory section. Figure 4 shows the mean ΔT_{c} results for the various experiments carried out at different \dot{m}_{c} and \dot{m}_{H} conditions. Figure 5 shows the mean steady state cooling heat transfer rate at the evaporator, \dot{Q}_{c} , i.e., the heat transfer rate actually extracted by the refrigerant in the evaporator from the water/ethylene-glycol mixture in the system cold side. Figure 4 shows that ΔT_{c} decreases as \dot{m}_{c} increases, as expected. All curves in Fig. 5 show the same behavior, i.e. there is an optimum cold water mass flow rate for maximum mean steady state cooling heat transfer rate at the evaporator (\dot{Q}_{c}). In Figure 5, for the line where $\dot{m}_{H} = 0.50 \text{ kg s}^{-1}$ or 1.1 lb s^{-1} , probably due to the experimental uncertainty in the measurements, for $\dot{m}_{c} = 0.37 \text{ kg s}^{-1}$ or 0.81 lb s^{-1} and $\dot{m}_{c} = 0.83 \text{ kg s}^{-1}$ or 1.8 lb s^{-1} , \dot{Q}_{c} was slightly higher than expected. However, the main message from Fig. 5 is that a clear maximum exists at the optimal mass flow rate pair, $(\dot{m}_{c}, \dot{m}_{H})_{opt} = (0.67, 0.67) \text{ kg s}^{-1}$ or $(1.47, 1.47) \text{ lb s}^{-1}$. Furthermore, it is observed that the optimum is robust for all tested heating water mass flow rates, and occurs at $\dot{m}_{c} = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s^{-1} , which maximizes the cooling heat transfer rate. This is an important finding for system scalability. The maximum calculated uncertainty on the determination of \dot{Q}_{C} was ± 0.6204 kW or ± 2117.43 BTU h⁻¹ (in absolute values).

Figure 6 shows the mean cold system efficiency ($\eta_{I,C}$) values calculated by Eq. (4) for various heating and cooling mass flow rates. The maximum calculated uncertainty on the determination of $\eta_{I,C}$ was ±0.0185 (in absolute values). The findings were similar to those for \dot{Q}_{C} , i.e. within the range of experiments carried out in this study, there is an optimum flow rate at $\dot{m}_{C} = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s⁻¹ for maximum first law efficiency, which is expected, due to the fact that \dot{Q}_{LPG} is fixed.



Figure 4. Variation of the mean ΔT_{C} of the water/ethylene-glycol mixture for the various experiments

under different \dot{m}_{C} and \dot{m}_{H} conditions.



Figure 5. Variation of the mean \dot{Q}_{C} of the water/ethylene-glycol mixture for the various experiments under different \dot{m}_{C} and \dot{m}_{H} conditions.

The opportunity found in the experiments is the heart of the thermodynamic optimization problem conducted in this study: the existence of a maximum heating and cooling rate production with respect to $\dot{m}_{\rm f}$, where f is either the water in the system hot side or the water/ethylene glycol mixture in the system cold side, namely, the thermal fluid. How fast should the thermal fluid be circulated through the heat exchangers? To see the importance of this question, consider the two extremes: (i) when $\dot{m}_{\rm f} \rightarrow 0$, the heat transfer rate collected by the thermal fluid approaches zero, and the flow exergy rate picked up by the thermal fluid stream approaches zero as well, and (ii) when $\dot{m}_{\rm f}$ achieves large and finite values, heat loss to the ambient and the work the fluid has to do to overcome friction increase with consequent higher entropy generation, and the outlet temperature of the thermal fluid stream is almost the same as the inlet temperature in the heat exchanger (condenser, absorber or evaporator), since the heat exchanger length is finite, therefore as flow speed increases, the temperature variation along the thermal fluid path goes to zero and heat transfer as well, and the flow exergy received by the $\dot{m}_{\rm f}$ stream is also small. As a result, in both extremes, the system heating and cooling rate production are negligible. This asymptotic behavior at zero and very high values of $\dot{m}_{\rm f}$ implies the existence of an intermediate (and optimal) flow rate $\dot{m}_{\rm f}$ that

maximizes the system heating and cooling rate production, which is observed experimentally in this study. Another way of understanding the problem is by noting that in a heat exchanger what is sought is the optimal match between the temperature distributions of the two streams, and therefore maximum heat transfer between the two streams. This optimization problem has been addressed in previous studies (Hovsapian et al. 2008; Vargas et al. 2001; Vargas et al. 2000a; Vargas et al. 2000b; Vargas and Bejan 2000; Vargas et al., 1998) for other thermal systems.

Exergetic analysis

Figure 7 shows the variation of the mean exergy rate content (\dot{E}_{C}) of the cooling heat transfer rate, calculated by Eq. (7), for the various experiments under different \dot{m}_{C} and \dot{m}_{H} conditions at steady state. The results show that there is an optimum mass flow rate ($\dot{m}_{C} = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s⁻¹) that maximizes the exergy rate. Considering that the calculated maximum uncertainty in \dot{E}_{C} is low (±0.0307 kW or ±104.78 BTU h⁻¹, in absolute values), the maximum exergy rates confirm the maxima obtained with the first law of thermodynamics.



Figure 6. Variation of the mean $\eta_{I,C}$ of the system for the various experiments under different

 \dot{m}_{C} and \dot{m}_{H} conditions.



Figure 7. Variation of the mean cold stream exergy rate (\dot{E}_{C}) value for the various experiments under different \dot{m}_{C} and \dot{m}_{H} conditions.

Figure 8 shows the variation of the mean efficiency values based on the second law of thermodynamics $(\eta_{II,C})$, calculated by Eq. (11), for the various experiments under different \dot{m}_C and \dot{m}_H conditions at steady state. The findings show the same trend reported for \dot{E}_C , i.e. there is an optimum \dot{m}_C value that maximizes $\eta_{II,C}$, i.e. $\dot{m}_C = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s^{-1} . The maximum calculated uncertainty in the determination of $\eta_{II,C}$ is ±0.00087 (in absolute values).



Figure 8. Variation of the mean $\eta_{II,C}$ of the system for the various experiments under different \dot{m}_C and \dot{m}_H conditions.

The experimental results identified the presence of a first optimum (i.e. maximum) for $\dot{Q}_{C,m}$ at $\dot{m}_{C,opt} = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s^{-1} obtained by varying \dot{m}_C , which is robust for all tested \dot{m}_H . The optimization proceeds by varying simultaneously \dot{m}_H and \dot{m}_C within the ranges presented in the theory section. The results are shown in Figure 9 with error bars, in which another optimum for \dot{Q}_C was found, i.e. there is an optimum pair $(\dot{m}_C, \dot{m}_H)_{opt}$ at $(0.67; 0.67) \text{ kg s}^{-1}$ or $(1.47; 1.47) \text{ lb s}^{-1}$ which characterizes a double maximum ($\dot{Q}_{C,m,m}$) for the system steady state cooling heat transfer rate.

Figure 9 also shows that the optimum cooling flow rate is robust with respect to the variation of $\dot{m}_{\rm H}$, which is an important result from the practical point of view, indicating that the optimum cold side mass flow rate, $\dot{m}_{\rm C}$, may be found regardless of the $\dot{m}_{\rm H}$ used. It is reasonable to expect that this phenomenon may also be found in similar refrigeration systems of any size.



Figure 9. Determination of the optimum mass flow rate pair, $(\dot{m}_{\rm C}, \dot{m}_{\rm H})_{\rm opt}$, for maximum $\dot{Q}_{\rm C,m}$ for $0.40 \le \dot{m}_{\rm H} \le 0.90 \text{ kg s}^{-1}$ or $0.88 \le \dot{m}_{\rm H} \le 2 \text{ lb s}^{-1}$.

In an analogous way, considering now the second law of thermodynamics, the above results identified the presence of a first optimum (i.e. maximum) for $\dot{E}_{C,m}$ at $\dot{m}_{C,opt} = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s⁻¹ obtained by varying \dot{m}_{C} . If \dot{m}_{C} and \dot{m}_{H} are varied together (Figure 10), another optimum for \dot{E}_{C} may be found, i.e. there is an optimum pair $(\dot{m}_{C}, \dot{m}_{H})_{opt}$ at $(0.67; 0.67) \text{ kg s}^{-1}$ or $(1.47; 1.47) \text{ lb s}^{-1}$ which also characterizes a double maximum ($\dot{E}_{C,m,m}$) for the system steady state cooling exergy rate.

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Figure 10. Determination of the optimum flow rate pair, $(\dot{m}_{C}, \dot{m}_{H})_{opt}$, for maximum $\dot{E}_{C,m}$ for

 $0.40 \le \dot{m}_{\rm H} \le 0.90 \, \text{kg s}^{-1}$ or $0.88 \le \dot{m}_{\rm H} \le 2 \, \text{lb s}^{-1}$.

CONCLUSIONS

In this work, a thermal and exergetic experimental analysis of a 5-TR (60,000 BTU h⁻¹) commercial water/ammonia absorption system was carried out, based on the first and second laws of thermodynamics, aiming to investigate the existence of optimum external operating parameters (mass flow rates) for maximum system performance.

Considering the range of mass flow rates evaluated in this study $(0.37 \le \dot{m}_C \le 0.83 \text{ kg s}^{-1} \text{ or} 0.81 \le \dot{m}_C \le 1.8 \text{ lb s}^{-1})$, a first optimum was found for $\dot{m}_C = 0.67 \text{ kg s}^{-1}$ or 1.47 lb s^{-1} so that maximum system cooling heat transfer rate $(\dot{Q}_{C,m})$ and cooling exergy rate $(\dot{E}_{C,m})$ were obtained. A second optimum

was found for $\dot{m}_{\rm H}$, thus an optimum pair was determined, i.e. $(\dot{m}_{\rm C}, \dot{m}_{\rm H})_{\rm opt} = (0.67; 0.67) \, \text{kg s}^{-1}$ or $(1.47; 1.47) \, \text{lb s}^{-1}$, for the double maximization of $\dot{Q}_{\rm C}$ and $\dot{E}_{\rm C}$. The determined $(\dot{m}_{\rm C}, \dot{m}_{\rm H})_{\rm opt}$ pair maximized simultaneously cooling heat transfer rate, exergy rate and first and second law efficiencies, with a maximum associated uncertainty of \pm 5.9%, i.e., these are global optima. The variation in $\dot{Q}_{\rm C}$ and $\dot{E}_{\rm C}$ reached 30 and 44 %, respectively, in the range of mass flow rates studied, meaning that the optima found may be clearly noticed, and therefore of practical importance in engineering applications and expected to be found in any absorption refrigeration system (e.g., single-effect, double-effect, GAX cycle), no matter how complex the actual design may be.

The refrigerator response was characterized by pull-down experiments. Pull-down times varied from 13 to 30 minutes in the tests conducted in this study, showing that absorption systems require a higher response time than vapor compression systems, as it is well known. Nevertheless, the pull-down times found here are still considered viable in comparison to vapor compression systems, being 22 min for the $(0.67; 0.67) \text{ kg s}^{-1}$ or $(1.47; 1.47) \text{ lb s}^{-1}$ pair of mass flow rates. Furthermore, the pull-down time could also be investigated in the search for optimum system operating conditions, to address applications in which this characteristic is of greater relevance than energy consumption.

As a suggestion for future studies, the methodology herein shown, and employed to the experimental study and optimization of external parameters of an absorption refrigeration system, may also be applied to other physical systems.

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 $\dot{m}_{H} = 0.90 \ kg.s^{-1}$

 $-\dot{m}_{H} = 0.83 \ kg.s^{-1}$

 $\dot{m}_{H} = 0.67 \ kg.s^{-1}$

 $\dot{m}_{H} = 0.50 \ kg.s^{-1}$

 $\dot{m}_{H} = 0.40 \ kg.s^{-1}$

14

13.5

13

12.5

12

11.5

11







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