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DESIGN, CONSTRUCTION AND EXPERIMENTAL PERFORMANCE OF A CHEMISORPTION CHILLER DRIVEN BY HOT WATER

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Abstract. *We designed, constructed and tested the performance of a chemisorption chiller in which hot water was the main source of energy. It had one evaporator, one condenser, two reactors, and used ammonia as refrigerant. Each reactor contained 9.4 kg of composite consolidated adsorbent with 60 % (w/w) of NaBr and 40 % (w/w) of expanded graphite. The mass flow and temperature of water at the inlet and outlet of the condenser, evaporator and reactors were measured to allow the calculation of the global heat transfer coefficient of the condenser and evaporator, and the chiller cooling power and coefficient of performance (COP). The values of the calculated global coefficient of performance were compared to those obtained by Nusselt correlations available in the literature. The experiments to obtain the chiller's performance were conducted using hot water at 70 °C as main heat source, tap water with temperature of 28 °C as heat sink, while the inlet temperature of water in the evaporator was being reduced from 23 °C to 4 °C. With the reduction of the water temperature that flowed through the evaporator, the mean cooling power and COP also decreased, and from the end of the first cycle to the end of the last one, their values dropped respectively, from 1690 to 710 W, and from 0.41 to 0.19.*

Keywords: Ammonia, Adsorption, Chiller, Heat-powered, Refrigeration.

1. INTRODUCTION

Mechanical compression systems driven by electricity are the most popular type of equipment for air conditioning, and the market for this type of equipment in Brazil rose 10 % between 2013 and 2014 (ASBRAV, 2014). Meanwhile, since the beginning of 2014, Brazil had faced two regional black-outs when the the Electric System National Operator (Operador Nacional do Sistema Elétrico - ONS) registered the highest historical levels of electricity consumption, which were caused by the intensive use of air conditioners during the summer (Borba *et al.*, 2015). One of the alternatives to reduce the impact on the electricity grid caused by air conditioning systems is the utilization of machines driven mainly by thermal energy. This alternative becomes more attractive when the thermal source is waste heat or solar energy.

Sorption chillers, either of the absorption type or adsorption type, are heat-powered machines that have been focused in much research (Boudehenn *et al.*, 2012; Castro *et al.*, 2007; Erickson, 2007; Kwak and Choi, 2007; Pongtornkulpanich *et al.*, 2008; Sarbu and Sebarchievici, 2013; Wang and Oliveira, 2006; Zaltash *et al.*, 2008; Zhai and Wang, 2009), due to their capability to produce cooling power directly from waste heat or solar energy.

To effectively use solar energy or low grade waste heat as main source of energy, the temperature level of the heat source should be as low as possible, and preferably, below 80 °C. Saha *et al.*(2001) developed a two-stage silica gel adsorption chiller that could be driven with hot water at 55 °C and produce 3.2 kW of cooling power with COP of 0.36 when rejected heat to a sink at 30 °C and cooled water from 14 °C. The silica-gel-water adsorption chiller studied by Xia *et al.* (2009) also achieved a COP of 0.36, but cooled water from 16 °C, and used water at 75 °C and 31 °C, as rescpectively, heat source and heat sink. A higher COP of 0.43 was achieved by the silica-gel adsorption chiller developed by Chen *et al.* (2010) when the chiller was also used to cool water at 16 °C and rejected heat to a sink at 31 °C, but was driven by a heat source at 73 °C. Vasta *et al.* (2012) used zeolite as adsorbent in their adsorption chiller and achieved COP of 0.41 to cool water at 13 °C when the heat source was at 90 °C and the heat sink was 28 °C. Both silica-gel and zeolite are physical adsorbents, and their use in adsorption chillers is more common than the use of chemical adsorbents like metallic salts, which is the case in this work. The latter type of adsorbent usually has the advantage of higher adsorption capacity but the disadvantage of lower adsorption rate when compared to physical adsorbents.

The chemisorption chiller developed by Kiplagat *et al.* (2013) employed NaBr impregnated in expanded graphite as composite sorbent to desorb and adsorb ammonia. The machine was utilized to cool water at 13 °C, and reached a COP above 0.4 when the heat source was between 60 and 70 °C and the sink was about 25 °C. Because the chiller studied by these authors had only one reactor, the cooling power halted during the regeneration of the sorbent, and the machine could not operate as the usual adsorption chillers which can produce cooling effect even during the regeneration period of the sorbent. Hence, we designed and constructed a chemisorption chiller with two reactors and filled the reactors with a composite consolidated sorbent of NaBr impregnated in expanded graphite to adsorb ammonia from the evaporator and desorb it to the condenser. The performance of the chiller was assessed when it was used to cool 125 L of water from the initial temperature of 23 °C to 4 °C.

2. MATERIAL AND METHODS

2.1 Description of the chemisorption chiller and the refrigeration sorption cycle.

The chemisorption chiller presented in this work (Fig. 1) had two reactors, one evaporator and one condenser. Information about temperature, pressure and mass flow in different locations of the chiller, as shown in Fig. 1, were measured by sensors and recorded every 6 seconds in a data logger.

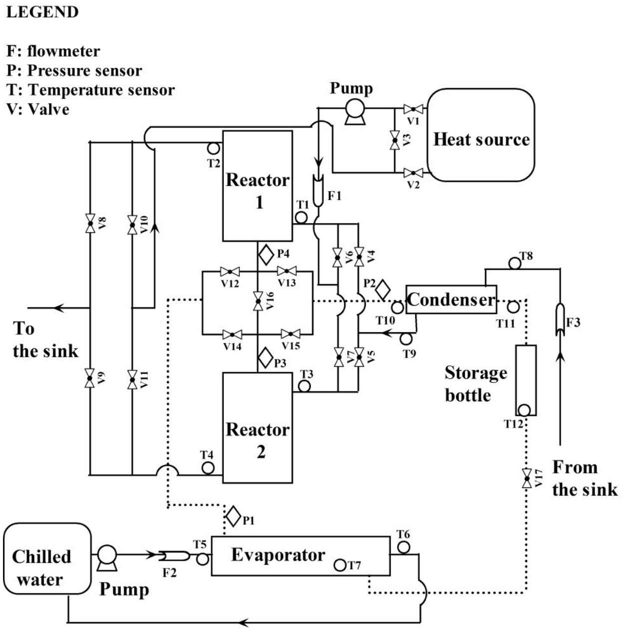

Figure 1. Scheme of the chemisorption chiller.

The reactors contained a consolidated composite sorbent made from NaBr impregnated in expanded graphite. The desorption and adsorption of ammonia by NaBr occured according to the following equation:

$$NaBr + 5.25NH_3 \Leftrightarrow NaBr \cdot 5.25NH_3 + 5.25\Delta H_R \quad (1)$$

The preparation of the composite sorbent was presented elsewhere (Oliveira *et al.*, 2007) and included the expansion of the expandable graphite at temperatures between 700 °C and 750 °C, followed by the mixture of this material to an appropriate amount of 23 % (w/w) salt solution. The slurry formed by these components had the proportion between salt and expanded graphite equal to 3:2. The slurry was dried in an electric oven at 130 °C for a period between 6 and 12 h, depending on the amount of slurry to be dried. The composite powder was put inside the tubes of the reactor and compressed until it reached the bulk density of 500 kg/m^3.

The reactors were tube and shell galvanized steel heat exchangers with the composite sorbent inside the tubes and water flowing in the shell. The photo of the reactor without the shell and with the shell is shown in Fig. 2. Other characteristics of the reactor are shown in Table 1.

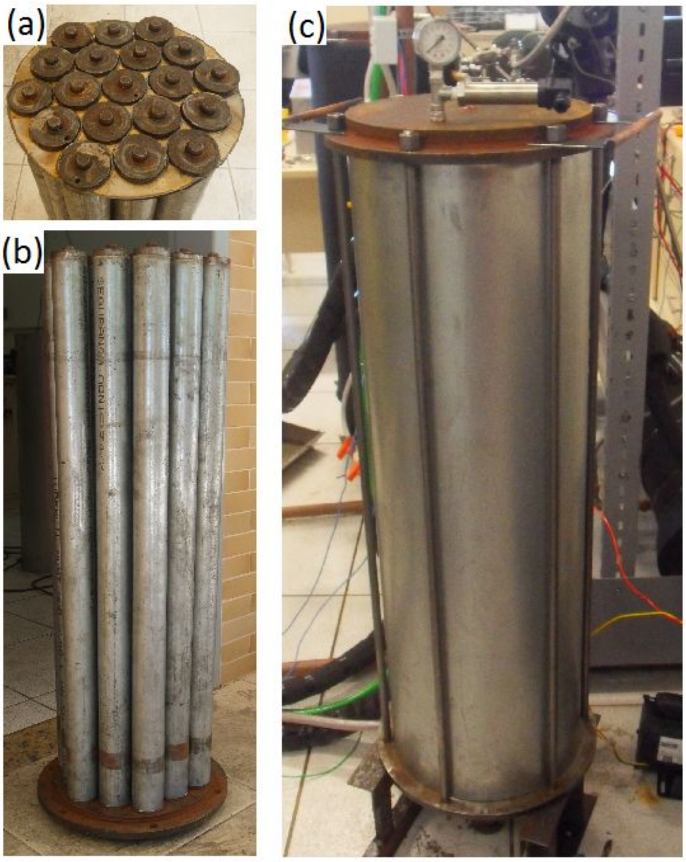

Figure 2. (a) Top view of the reactor tubes. (b) Side view of reactor tubes. (c) Complete reactor with shell.

Table 1. Constructive characteristics of each reactor.

Heat exchange area (m^2)	2.81
Mass of tubes (kg)	82.5
Total metallic mass (tubes, shell, cover, etc.) (kg)	140
Inner diameter of the tubes (m)	0.055
Number of tubes	19
Inner diameter of the shell (m)	0.31
Number of baffles	1
Mass of dry composite adsorbent (kg)	9.4
Thickness of the adsorbent layer (m)	8.4×10^{-3}
Length of the adsorbent inside the tube (m)	0.81

The condenser and the evaporator were also tube and shell reactors. In the condenser, the refrigerant condensed inside the tubes and water flowed through the shell, whereas in the evaporator, the refrigerant evaporated in the shell, around the tubes, and water flowed in multiple passes (11 passes), inside the tubes. The evaporator was only partially filled with tubes to ensure that even with the reduction of refrigerant inside the evaporator during the adsorption period, the tubes remained immersed in the refrigerant. Fig. 3 shows the disposition of the tubes inside the evaporator shell and the tubes cover used to ensure multiple passes. Other characteristics of the evaporator are shown in Table 2.

The chiller operated in a cycle where one reactor was heated and desorbed refrigerant to the condenser while the other reactor was cooled and adsorbed refrigerant from the evaporator. The refrigerant condensed was stored in an insulated bottle. Once the heating and cooling period of the reactors was completed, there was a lag period where the refrigerant stored in the bottle was transferred to the evaporator. During this period, the reactors could exchange heat and mass between them, in a process known as simultaneous heat and mass recovery.

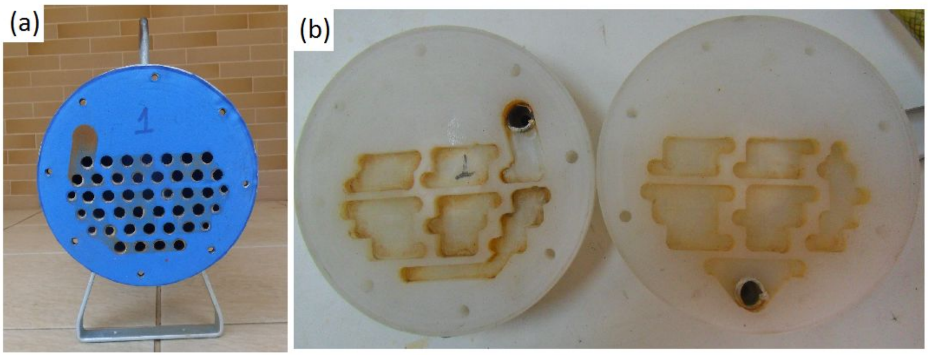

Fig. 3. (a) Front view of the tubes inside the evaporator shell. (b) Cover to ensure multiple passes through the tubes.

Table 2. Constructive characteristics of the evaporator.

Heat exchange area (m^2)	1.57
Total volume of refrigerant (m^3)	1.06×10^{-2}
Volume of refrigerant above the tubes (m^3)	4.89×10^{-3}
Length of the tube (m)	0.93
Number of tubes	43

2.2 Assessment of the global heat transfer coefficient of the condenser and evaporator

The instantaneous heat transfer rate in the condenser and evaporator can be obtained with Eq. (1) or Eq. (2).

$$\dot{Q} = \rho_W \dot{V} Cp_W \left(T_{In} - T_{Out}\right) \quad (1)$$

$$\dot{Q} = \frac{\Delta T_{\ln}}{R} \quad (2)$$

$$R = \frac{1}{UA} \quad (3)$$

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{1}{h_e A_e} + \frac{\ln\left(\frac{r_e}{ri}\right)}{2\pi L \lambda} \quad (4)$$

where ρ_W is the water density at the inlet temperature, $\dot{V}$ is the volumetric flow of water, Cp_W is the specific heat of water, T_{In} is the inlet water temperature, T_{Out} is the outlet water temperature, $\Delta T_{\ln}$ .is the logarithm temperature difference, R is the thermal resistance, which can be calculated by combining Eqs. (1) and (2) or Eqs. (3) and (4), U is the global heat transfer coefficient, A is the reference heat transfer area used to calculate U, h_i and h_e are, respectively, the heat transfer coefficient in the inner and outer surface of a pipe inside the condenser or evaporator, r_e and r_i are respectively, the external and internal radius of a pipe inside the condenser or evaporator, A_i and A_e are the heat transfer area based, respectively, on r_i and r_e, L is the pipe length inside the condenser or evaporator, whereas λ is the pipe thermal conductivity.

The value of the global heat transfer coefficient (U) was calculated based on the outer area of the pipes inside the condenser or evaporator, and we assumed that the experimental value was obtained with the combination of Eqs. (1), (2) and (3), whereas the theoretical value was obtained by with Eq. (4) and the appropriate correlation for h_i and h_e.

In the condenser, h_i for condensation inside pipes was calculated as proposed by Chato (1962 *apud* Osizik, 1990), whereas h_e was calculated as proposed by Taborek (1983) for a flow side the shell side of a tube-and-shell heat exchanger. In the evaporator, h_i was calculated as proposed by Bhatti and Shah (1987 *apud* Shah and Sekulic, 2003), for a turbulent thermally developing flow with Nu_∞ obtained by the correlation proposed by Petukhov and Popov (1963 *apud* Shah and Sekulic, 2003), whereas h_e was calculated as proposed by Taborek (1983, *apud* Spindler, 2010) for nucleate pool boiling.

2.3 Assessment of the chiller cooling power and coefficient of performance

The assessment of the chiller performance was conducted when the chiller operated under a 68 minute cycle with heat and mass recovery and inlet hot water temperature of 70 °C and inlet cooling water of 28 °C. The heat and mass recovery processes lasted four minutes and occurred twice in each cycle. The isostereric heating and cooling periods were short, and usually, around 3 minutes.

The experiments were used to identify the length of time necessary to cool down 125 L of water from the initial temperature of 23 °C to the minimum value that could be achieved by the chiller, and to assess how the COP and cooling power varied from one successive cycle to another, as the chilled water temperature decreased.

Equation (5) gives the accumulated mean value of the cooling power since the beginning of the cycle.

$$\overline{Q}_{Cool} = \frac{\sum_{i=1}^{n}\left[\Delta t \rho_W \dot{V}_{Evap} Cp_W \left(T_{In,Evap} - T_{Out,Evap}\right)\right]_i}{t_i} \tag{5}$$

where Δt is the interval of time of each data acquisition by the data logger, t_i is the length of time since the beginning of a cycle, n is the number of times that data was acquired by the data logger during a cycle, the subscript *Evap* means that the acquired value was related to the evaporator.

The COP was calculated with Eq. (6) and indicates the amount of heat that need to be provided to the system to obtain a certain amount of cooling load.

$$COP = \frac{Q_{Cool}}{Q_{Hot}} \tag{6}$$

$$Q_{Hot} = \sum_{i=1}^{n}\left[\Delta t \rho_W \dot{V}_{\mathrm{Reac}} Cp_W \left(T_{In,\mathrm{Reac}} - T_{Out,\mathrm{Reac}}\right)\right]_i \tag{7}$$

$$Q_{Cool} = \sum_{i=1}^{n}\left[\Delta t \rho_W \dot{V}_{Evap} Cp_W \left(T_{In,Evap} - T_{Out,Evap}\right)\right]_i \tag{8}$$

where the subscript *Reac* means that the acquired value was related to the reactors.

T_{In} and T_{Out} were measured with resistance temperature detectors (RTD) 1/10 DIN class with uncertainty at 0 °C of 0.03 °C. The uncertainty in the temperature acquisition by the data logger was 0.06 °C. $\dot{V}$ for the evaporator was measure with a flow meter FPR301 (Omega Engineering, Inc.), $\dot{V}$ for the reactor was measure with a flow meter FPR204 (Omega Engineering, Inc.), and both have uncertainty of 3.2×10^{-3} L/s. The uncertainty in the flow acquisition by the data logger was 2.8×10^{-5} L/s for the sensor FPR301 and 8.8×10^{-4} L/s for the sensor FPR204. The data acquisition was done with a data logger 34772a (Agilent Technologies, Inc.). The uncertainty in the values of global heat transfer coefficient, cooling power and COP were obtained, as suggested in the literature (Taylor, 1997).

3. RESULTS

Figure 4(a) shows the experimental and theoretical global heat transfer coefficient (U_{Cond}) for the condenser, whereas Fig. 4(b) shows the saturation temperature and the inlet water temperature. Because the inlet water temperature varied about 4 °C during the experiment, the condenser did not reach a steady state, and this fact influenced the value of the experimental value of U_{Cond}. However, when the mean experimental value of U_{Cond} is compared to the mean theoretical value, the former is only 6 % higher than the latter one, which indicates that the correlations chosen in this work are appropriate to describe the heat transfer process inside the condenser.

The experiments to obtain the global heat transfer coefficient of the evaporator (U_{Evap}) were conducted with constant inlet water temperature of 12 °C and 15 °C. As can be seen in Fig. 5, both the experimental and the theoretical U_{Evap} remain approximately constant during the experiments, and the assumption of steady state operation was valid. The mean experimental value among the twelve experiments conducted was 330±40 W/m²K, whereas the mean theoretical value was 310±40 W/m²K. Due to the small temperature difference between the inlet and outlet of the evaporator, and which was about 2 °C, the mean experimental uncertainty was 120 W/m²K. As the discrepancy between the mean values of U_{Evap} was 6 % and within the range of the experimental uncertainty, it is possible to conclude that the correlations chosen in this work describe the heat transfer process in the evaporator with acceptable accuracy.

Figure 6 shows the temperature variation of the inlet and outlet water that flowed through the evaporator and through the reactors. The hot water used to drive the reactors was kept constant at 70 °C, whereas the tap water used as heat sink varied about 3 °C at the reactor inlet during each half cycle because it passed through the condenser to remove the condensation heat before entered the reactor.

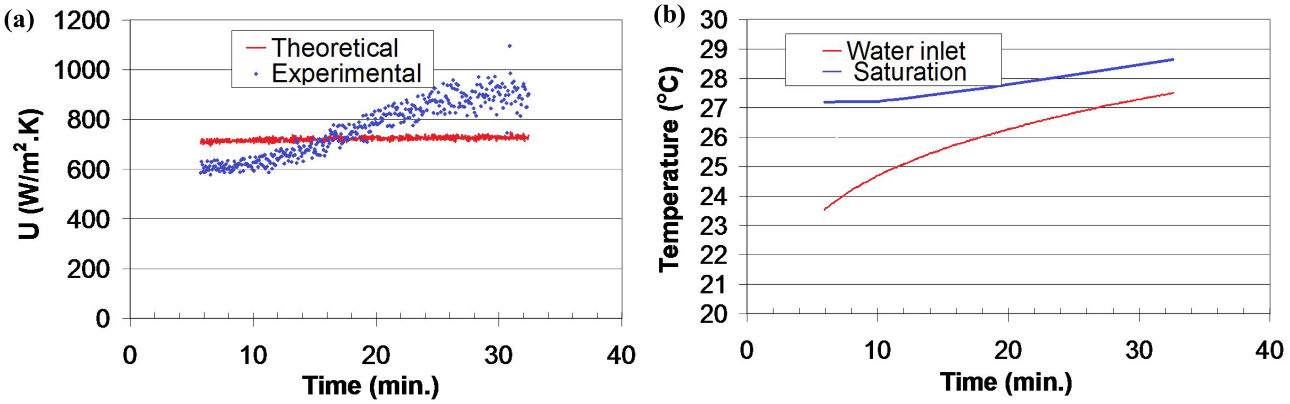

Figure 4. (a) Global heat transfer coefficient of the condenser. (b) Inlet water temperature and saturation temperature in the condenser.

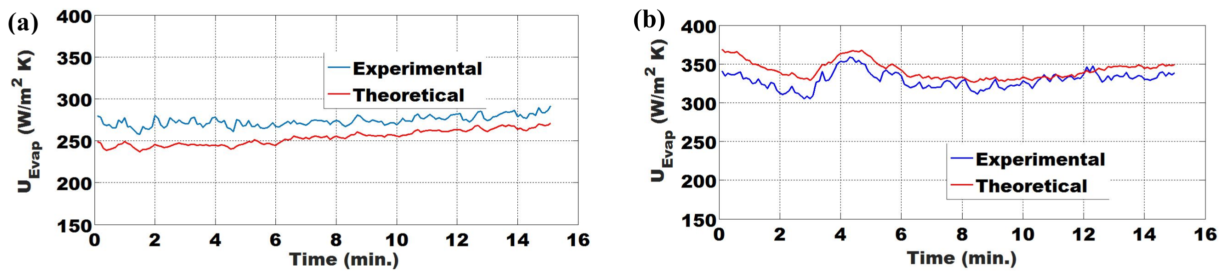

Figure 5. Global heat transfer coefficient of the evaporator. (a) Inlet temperature of 12 °C. (b) Inlet water temperature of 15 °C.

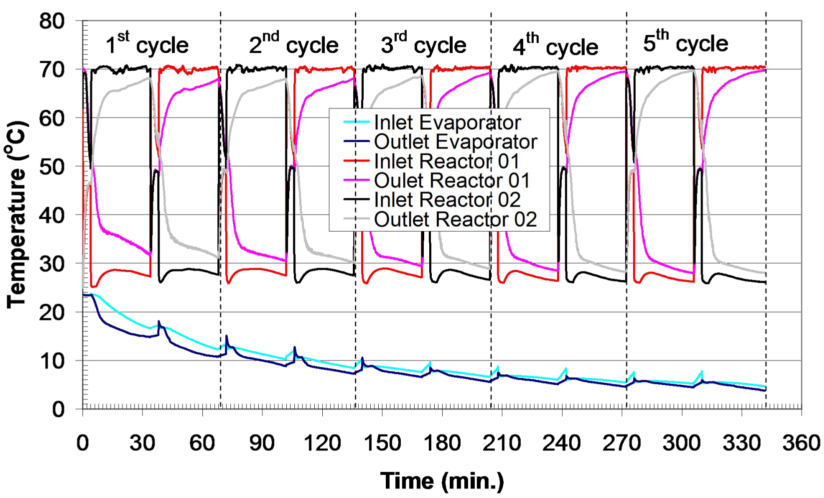

Figure 6. Temperature of the chilled water and water supplied to the reactors as heat source and heat sink.

From the beginning to the end of the first cycle, all water in 125 L tank had its temperature dropped in 11 °C. However, in the following cycles, the temperature drop was smaller, and from the beginning of the forth cycle to the end of the fifth cycle, the temperature dropped only 3 °C, but reached 3.8 °C.

The variation of the chiller cooling power and COP in the successive cycle can be seen in Fig. 7. The red lines in this figure indicate the range of the experimental uncertainty. At the end of the first cycle, the mean cooling power was 1690 W and the COP was 0.41. As the inlet temperature in the evaporator decreased, both cooling power and COP decreased. At the end of the second cycle, when the chilled water in the tank was 8 °C, both the mean cooling power and COP were 71 % of those achieved at the end of the first cycle. As the water in the tank continued to be chilled, both performance indicators continue to drop, and at the end of the forth cycle, they have dropped to about half of their initial value. Hence, although the chemisorption chiller is capable to chill water down to 4 °C, it is preferable to chill water to the minimum of 10 °C to ensure cooling power of about 1400 W and COP around 0.33.

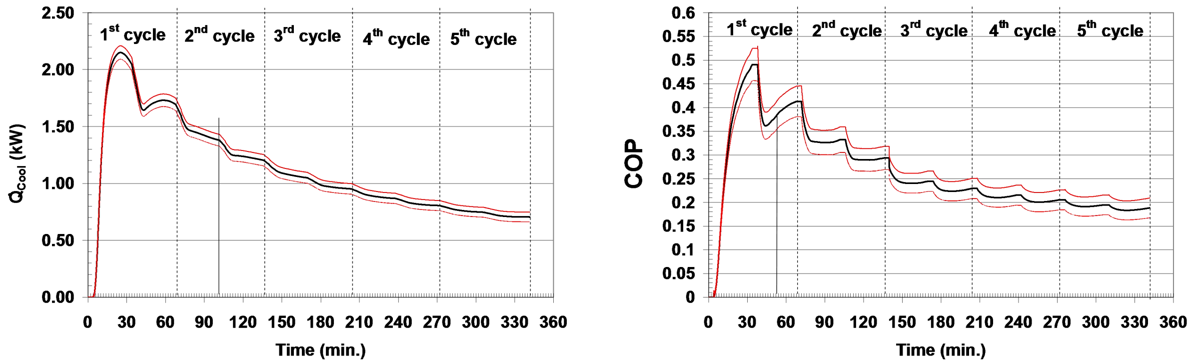

Figure 7. (a) Mean cooling power. (b) Mean COP.

4. ACKNOWLEDGEMENTS

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