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Experimental Study on Ammonia Finned Air-Coolers Working under Frosting Conditions

Rodica Dumitrescu, Anica Ilie, Alina Girip, Madalina Teodora Nichita, Alina Filip

[a]Technical University of Civil Engineering, Bucharest, Bd. Lacul Tei 122-124, Bucharest, 020396, Romania

Abstract: *This paper reports experimental data obtained for an ammonia air-cooling evaporator working under conditions of frost formation. The experimental study was carried out on a specially designed set-up that allowed measuring, control and acquisition of data. Experiments were conducted for ammonia evaporation temperature of -18℃ and -10℃ and cooling capacity from 7.6 kW to 18.3 kW, during steady state operating conditions. The experimental investigation aimed to determine the effect of frost formation and growth on thermal performances of the ammonia finned air-cooler, in time, under different operating conditions. Influence of air dry bulb temperature, air humidity ratio and evaporating temperature has been studied. Air side pressure drop, for the same dry bulb temperature and different humidity air ratio has also been studied. This experimental investigation on an ammonia air-cooling evaporator addresses the current issue of energy savings by assisting improved design of this type of heat exchangers in order to achieve higher thermal performances of evaporators and refrigerating systems. By using ammonia as a refrigerant, this experimental investigation addresses also current ecological issues linked to refrigerants. It is well known that apart from its attractive thermo-physical properties, as compared to HFCs, ammonia is also an environment friendly natural refrigerant, known for its zero ozone depleting potential and zero global warming effect.*

Key words: finned air-cooler, ammonia evaporator, frost formation, heat transfer.

Rezumat: *Această lucrare prezintă rezultatele experimentale obţinute în urma testării unui răcitor de aer cu suprafaţă aripată, care reprezintă vaporizatorul unei instalaţii frigorifice cu amoniac. Studiul experimental s-a desfăşurat pe un stand special proiectat pentru a asigura măsurarea, controlul şi achiziţia de date. Experimentele s-au efectuat în regim staţionar, pentru temperaturi de vaporizare ale amoniacului de -18°C şi -10°C şi puteri de răcire ale aerului cuprinse între 7.6kW si 18.3 kW. S-a urmărit influenţa formării şi depunerii de brumă asupra performanţelor termice ale răcitorului de aer aripat, în funcţie de timp, în diferite condiţii de funcţionare. Acestea din urmă au fost reprezentate de: temperatura aerului după termometrul uscat şi conţinutul de umiditate al aerului la intrarea pe suprafaţa de răcire, ca şi de temperatura de vaporizare a amoniacului. De asemenea, s-a studiat variaţia căderii de presiune pe partea aerului, în diferite condiţii de funcţionare ale instalaţiei. Acest studiu experimental vizează problema* *permanent actuală a economiei de energie, prin baza de date pe care o oferă activităţii de proiectare, având ca obiectiv creşterea performanţei termice a vaporizatorului şi a instalaţiei frigorifice. În plus, prezenta lucrare abordează aspecte legate de protecţia mediului înconjurător, prin aceea că utilizează un agent frigorific natural. Amoniacul are impact nul asupra stratului de ozon şi asupra încălzirii globale şi este bine cunoscut pentru proprietăţile sale termofizice superioare celor ale agenţilor frigorifici de tip HFC.*

Cuvinte cheie: răcitor de aer aripat, vaporizator cu amoniac, formare şi depunere de brumă, transfer de căldură.

1. Introduction

Air-cooling evaporators are refrigerant to-air heat exchangers widely used in industrial and technological refrigeration, as well as commercial refrigeration for conservation of perishable products. They consist of cooling coil and fan(s), so they are forced-circulation air coolers. Their role is to remove heat and water vapor from the humid air, immediately surrounding the refrigerated/ freezing/stored products. These heat exchangers use tubes to carry refrigerant with fins applied to the tube exterior, in order to increase the area available for heat and mass transfer. Refrigerant evaporates inside the tubes as it absorbs heat from air flowing over the outside surface of the finned tubes. When air-cooling evaporators operate with both coil surface temperatures and entering air dew-point temperatures above the coil surface temperature, below 0°C, which is the case in both refrigeration/freezing and cold storage of food product, moisture from the air being cooled accumulates on the fins and tubes of the coil in the form of frost. The growth of frost is a complex transient process in which both heat and mass transfer occur simultaneously. The formation and growth of frost on the finned-type evaporator leads to a decrease in its cooling capacity, [1], [2], [3], [4]. This is due to increased air-side pressure drop that causes airflow decrease through the coil and to increased resistance to heat transfer between the air and the refrigerant caused by the insulating effects of the frost, [5]. With decreased heat transfer, the evaporator temperature drops, causing a decrease in efficiency. If frost is allowed to accumulate further, even liquid flood back to the compressor can occur due to reduced evaporator capacity.

Both experimental and numerical studies on heat and mass transfer processes that take place in air-cooling evaporators under conditions of frost formation represent one sine qua non step in assisting designers to make the right decisions related to fin spacing, operating temperature of the coil, defrosting frequency, required defrosting time, in order to achieve maximum efficiency from the evaporator and high refrigerating system COPs.

2. Experimental set-up

The ammonia finned air-cooler under study was placed inside the horizontal section of an closed insulated air loop of rectangular shape (500 x 500 mm) (Figure 1), made of galvanized steel and equipped with multiple regulating and control devices for setting the entering air parameters (dry and wet bulb temperature, humidity ratio, flow rate) and refrigerant parameters, within given ranges [6].

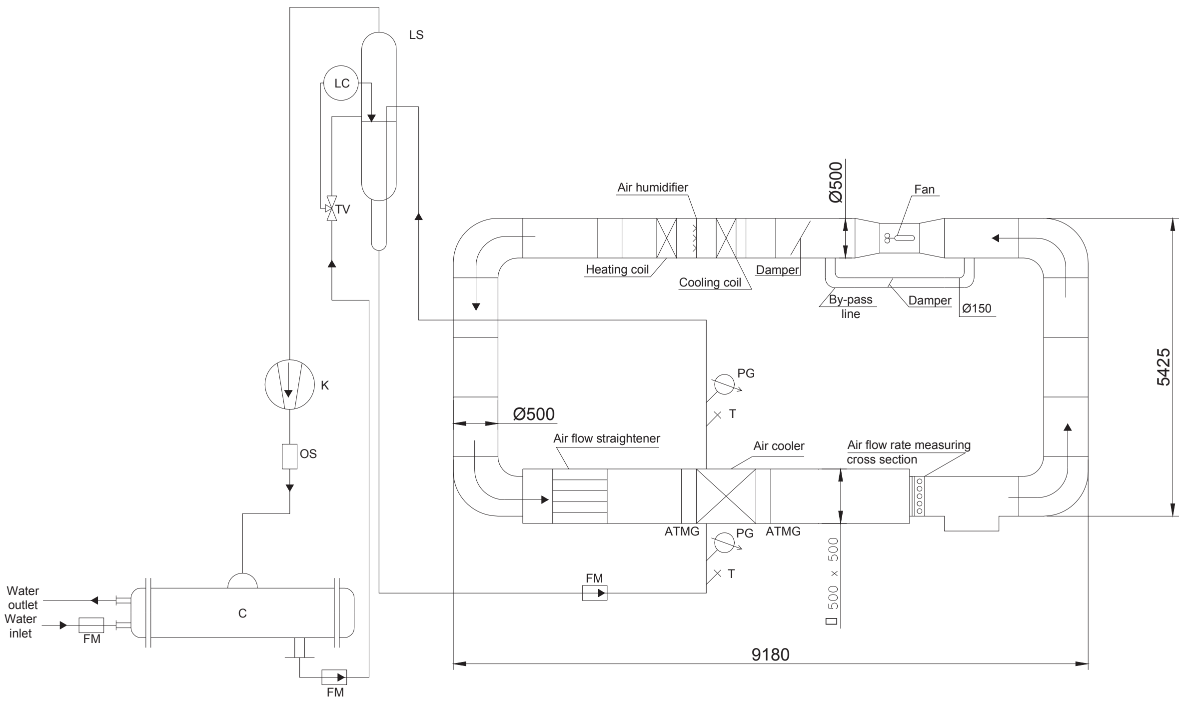


Figure 1. Schematic diagram of the experimental set-up

Legend: LS – liquid separator; OS – oil separator; K- compressor; C – condenser; LC – level control; TV – thermostatic valve; ATMG – air temperature measuring grid; FM – ammonia flow meter; T – temperature sensor; PG pressure gauge.

The transparent top cover of the air loop enables visualization of frost formation and growth on the finned tubes of the ammonia evaporator (Photo 1).

Photo 1. Finned air-cooler inside the air loop – plan view

The ammonia air-cooling evaporator under study uses a staggered tube bundle system of 4 parallel coils, of 8 horizontal tubes each. The coils are made from steel tubing. The outer diameter of the tubes is 25mm. Figure 2 shows the geometrical configuration of the coils and fins. Tube spacing is 70mm on equilateral centers. Fins are made of aluminum and are spaced 5.25 mm apart. The fin thickness is 0.4 mm. As it may be observed from Figure 2 fins are corrugated.

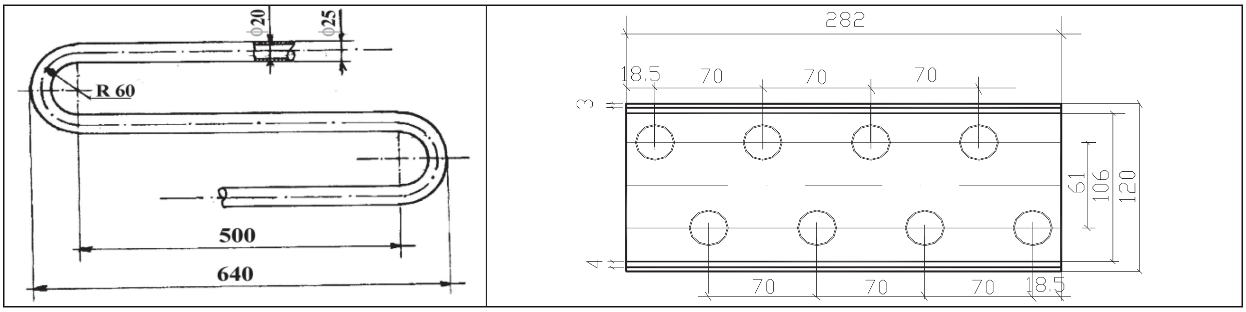


Figure 2. Geometrical configuration of coils and fins

The evaporator is fed by gravity circulation from the vertical liquid separator of an ammonia single stage mechanical vapor compression refrigeration system depicted in Photo 2. Humid air of controlled inlet parameters is simultaneously directed over the finned coil, flowing through the unit.

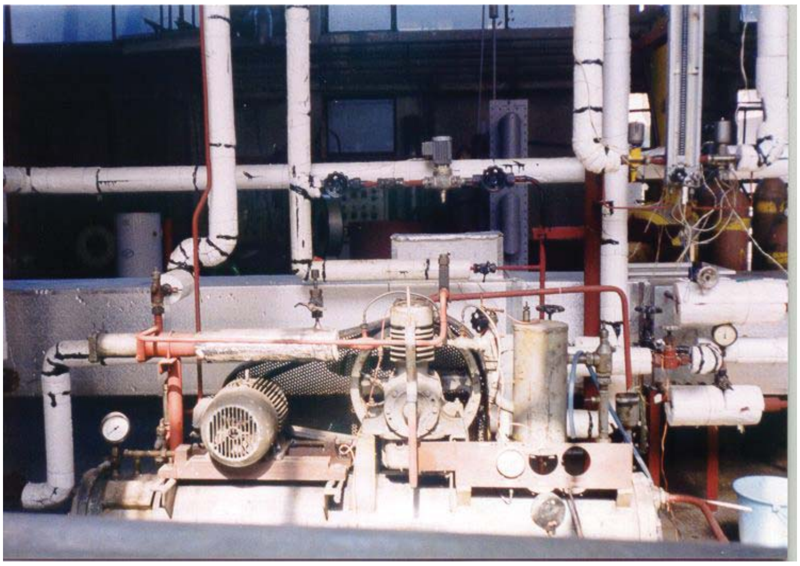


Photo 2. Refrigeration system – overall view

3. Methodology

Experimental investigation has been carried out on the ammonia finned air-cooler described above during steady state conditions. It was considered as such the regime characterized by maximum ±5% variation of the measured parameters, along 10 consecutive readings, at 10 minutes apart. The goal of the experiments was to determine the effect of frost formation and growth on thermal performances of the ammonia finned air-cooler, under different operating conditions. Influence of air dry bulb temperature, air humidity ratio and evaporating temperature has been studied. Air side pressure drop, for the same dry bulb temperature and different humidity air ratio has also been studied.

Experiments were conducted under the following operating conditions:

- cooling capacity from 7.6 kW to 18.3 kW;
- dry bulb temperature of inlet air ranging from +10 °C to +20°C;
- humidity ratio of inlet air ranging from 5 g/kg to 9 g/kg;
- air velocity in the front section of the air cooler with dry surface: 3,3 m/s;
- ammonia evaporating temperature of -18°C and -10°C;
- maximum operation time: (70 …135) min.

Experiments have been considered completed as frost thickness reached the maximum acceptable values, in terms of economic operation costs, of 2.2 to 2.3mm, on each side of the fin, of an overall fins spacing of 5.25 mm. It has thus been determined the maximum operating time of refrigeration cycle before defrosting period.

Cooling capacity of the air-cooler with frost formation has been experimentally determined in two different ways, namely: based on the energy balance of the refrigeration system and based on the cooling capacity on the air side [7] and [8].

- Based on the refrigerant energy balance, the cooling capacity on the refrigerant side, $\dot{Q}_{0,r}$, has been evaluated by:

$$\dot{Q}_{0,r} = \dot{Q}_c - P_k \text{ , [W]} \quad (1)$$

where: $\dot{Q}_c$ - condenser heat rejection rate, [W]; P_k - directly measured compressor power input, [W].

Condenser heat rejection rate has also been evaluated by two different methods, namely: by the measured ammonia flow rate and by the energy balance on cooling water side. Based on condensate flow rate, condenser heat rejection rate, $\dot{Q}_{c,r}$, has been evaluated by:

$$\dot{Q}_{c,r} = \dot{m}_r \cdot \left(h_r^1 - h_r^2\right), \text{ [W]} \quad (2)$$

where: - $\dot{m}_r$ - ammonia mass flow rate measured at condenser outlet, [kg/s];

- h_r^1 - superheated vapor enthalpy at condenser inlet, [J/kg];
- h_r^2 - subcooled condensate enthalpy at condenser outlet, [J/kg];

Based on the energy balance on cooling water side, $\dot{Q}_{c,w}$, has been evaluated by:

$$\dot{Q}_{c,w} = \dot{m}_w \cdot c_{pw} \cdot \left(t_w^2 - t_w^1\right), \text{ [W]} \quad (3)$$

where: - $\dot{m}_w$ - directly measured cooling water flow rate at condenser inlet, [kg/s];

- c_{pw} - specific heat at constant pressure of cooling water, at its average temperature, [J/(kg K)];
- t_w^1 - water temperature at condenser inlet, [°C];
- t_w^2 - water temperature at condenser outlet, [°C].

Experimental condenser heat rejection rate values have been considered reliable, if they satisfied the following energy balance with a deviation of less than ±5%:

$$\dot{Q}_c \cong \dot{Q}_{c,r} \cong \dot{Q}_{c,w}\text{, [W]} \tag{4}$$

- Based on measured air flow rate the air cooler cooling capacity, $\dot{Q}_{0a}$, has been evaluated by:

$$\dot{Q}_{0a} = \frac{\dot{V}_a \cdot \rho_a \cdot \Delta h}{3600}\text{, [W]} \tag{5}$$

where:

- $\dot{V}_a$ - air volumetric flow at the inlet of the finned air cooler, [m^3/s];
- ρ_a - air density at the inlet of the finned air cooler, [kg/m^3];
- Δh - enthalpy change of humid air; $\Delta h = h_{a,2} - h_{a,1}$, [J/kg];
- $h_{a,1}$ - enthalpy of air entering the finned air cooler; $h_{a,1} = f\left(t_1^{dry}; t_1^{wet}\right)$, [J/kg];
- $h_{a,2}$ - enthalpy of air leaving the finned air cooler; $h_{a,2} = f\left(t_2^{dry}; t_2^{wet}\right)$, [J/kg].

Based on both methods mentioned above, the cooling capacity of the finned air cooler is given by:

$$\dot{Q}_0 \cong \dot{Q}_{0,a} \cong \dot{Q}_{0,r}\text{, [W]} \tag{6}$$

Experimental $\dot{Q}_0$ values have been considered reliable and considered in further calculations, if they satisfied the energy balance above within a deviation of ± 7%.

Primary measurements in the experiments were:

- on air side: dry bulb temperature (t_1^{dry}, t_2^{dry},), wet bulb temperature (t_1^{wet} , t_2^{wet}) both at the inlet and outlet of the finned air-cooler, volumetric flow rate (Figure 1);
- on ammonia side: inlet and outlet temperature and pressure, liquid volumetric flow rate (Figure 1);
- on water side: temperature inlet and outlet of the condenser and volumetric flow rate (Figure 1).

Temperatures were measured using type K thermocouples with reading accuracy within ± 0.1°C. Air volumetric flow rate was measured with a hot wire anemometer placed into air stream that provided a direct reading of air velocity with an accuracy of 2 to 5% of reading over the entire velocity range. Water volumetric flow rate was measured with a Danfoss electronic flow meter that had a reading accuracy of ±3% and liquid ammonia volumetric flow rates were measured by Coriolis type flow meter that had a reading accuracy of ±3%. The compressor power input was measured with three-phase active energy meter (3 x 380 / 220) that had a reading accuracy of ± 0.005 kWh. All sensors were calibrated prior to testing.

4. Experimental results

4.1. Validation of measured heat transfer rate [6]

Figure 3 shows deviations of condenser heat rejection rate measured, as previously detailed, by two different methods, namely using measured ammonia flow rate and energy balance on cooling water side.

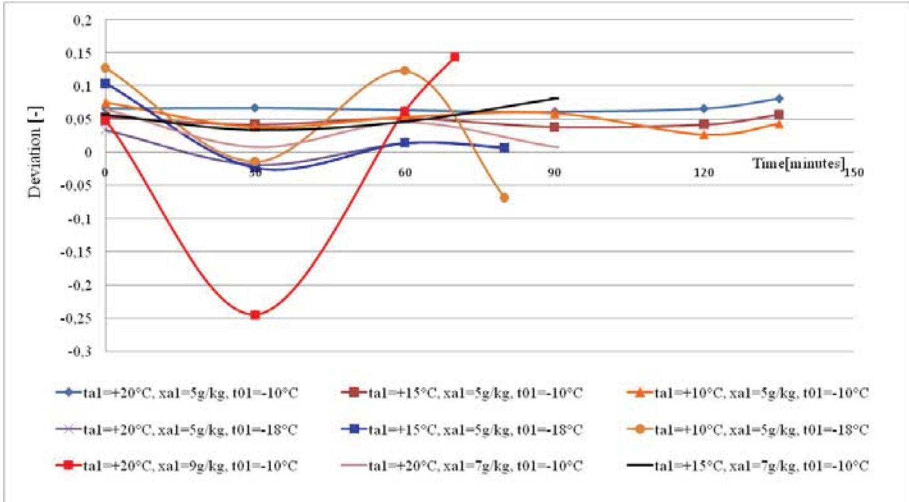


Figure 3. Deviations of condenser heat rejection rate measured by two different methods

It may be observed from Figure 3 that deviation of measured condenser heat rejection rate ranges from +14.4% to – 6.9%; there is a single experiment, in which the deviation reaches, at some point, an unacceptable value of approx. -25%. This particular experiment was excluded from further calculations. The error analysis shows that 55% of the experimental data fall within an error band of ±5%, admitted as acceptable. As a consequence, just these measured condenser heat rejection rate represent reliable data.

Figure 4 shows deviations of cooling capacity measured, as previously mentioned, by two different methods, namely: based on the energy balance of the refrigeration system and based on the cooling capacity on the air side.

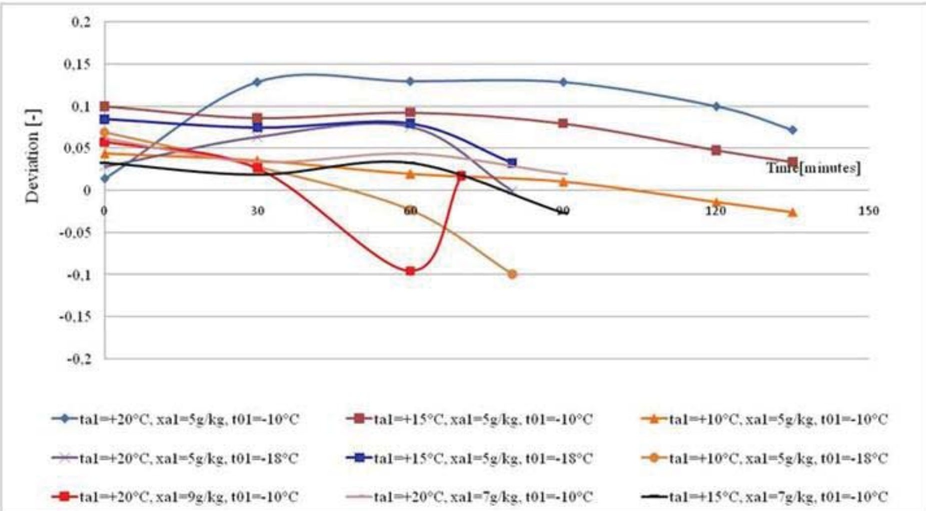
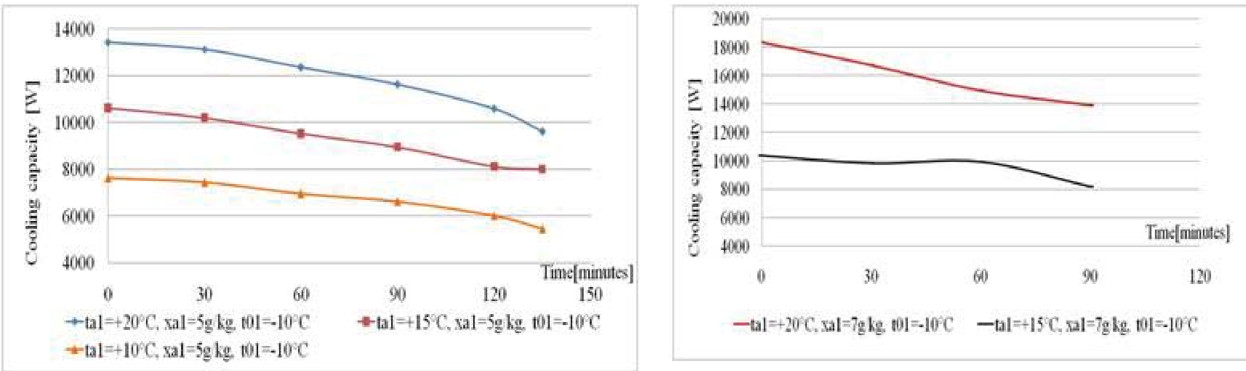


Figure 4. Deviations of finned-coil cooling capacity measured by two different methods

It may be noted from Figure 4 that deviation of measured finned-coil cooling capacity for all experiments ranges from +13.7% to -10%. The error analysis shows that 64% of the experimental data fall within an error band of ±7%, admitted as acceptable. As a consequence, just these measured cooling capacity values represent reliable data.

4.2. Effect of air dry bulb temperature [6]

Figure 5 shows the effect of entering air dry bulb temperature on finned-coil cooling capacity. Experiments have been performed for the same evaporating temperature of -10°C and different humidity ratio of inlet air, namely 5 g/kg and 9 g/kg. Figure 5 illustrates the normal decrease trend of ammonia finned-coil cooling capacity in time, with a decrease in entering air temperature. The explanation lies, on one hand, in decreased mean temperature difference between air and ammonia, and on the other hand, in increased surface temperature of the frost layer, as frost thickness grows in time.



a) b)

Figure 5. Effect of air dry bulb temperature on finned-coil cooling capacity

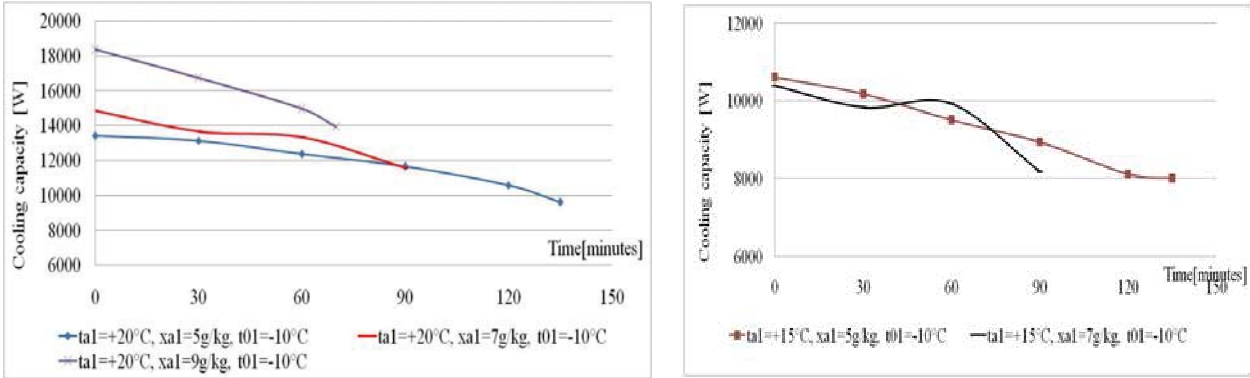
It may be observed from Figure 5a that air dry bulb temperature drop of 10°C (from +20°C to +10°C), for the same inlet air humidity ratio of approximately 5g/kg and the same evaporating temperature of approximately –10°C leads to an average decrease in cooling capacity of 43%. Air dry bulb temperature drop of 5°C (from +20°C to +15°C), for the same humidity ratio of inlet air of approximately 7g/kg and the same evaporating temperature of approximately –10°C leads to an average decrease of cooling capacity of 28% (see Figure 5b).

It is also evident from Figure 5 that the economic operating time of the ammonia air-cooler decreases by 33% as the latent heat transfer potential of the enetring air increases from 5g/kg to 7g/kg, for the same air dry bulb temperature of +20°C and the same evaporating temperature of approximately –10°C.

4.3. Effect of air humidity ratio [6]

Figure 6 shows the effect of entering air humidity ratio on finned-coil cooling capacity. Experiments have been performed for the same evaporating temperature of -10°C, different humidity ratio of inlet air, namely 5g/kg, 7g/kg and 9g/kg and different entering air dry bulb temperature of +20°C and +15°C.

It may be observed from Figure 6 that finned-coil cooling capacity increases with an increase in entering air humidity ratio, for the same entering air dry bulb temperature and the same evaporating temperature. Cooling capacity increases on average by 13% over the economic operating time with an increase in air humidity ratio from 5g/kg to 7g/kg, for a fixed entering air dry bulb temperature of +20°C (Figure 6a).



a) b)

Figure 6. Effect of air humidity ratio

Cooling capacity increases on average by only 4% over the economic operating time with an increase in air humidity ratio from 5g/kg to 7g/kg, for a fixed entering air dry bulb temperature of +15°C (Figure 6b). This normal trend of cooling capacity increase is explained by an increase in latent heat transfer component.

It may also be noted that the maximum cooling capacity, for given entering air dry bulb temperature of +20°C and evaporation temperature of -10°C corresponds to maximum inlet air humidity ratio of 9g/kg.

It is also evident from Figure 6a that the economic operating time of the ammonia air-cooler decreases by 48% as the latent heat transfer potential of the enetring air increases from 5g/kg to 9g/kg, for the same air dry bulb temperature of +20°C and the same evaporating temperature of approximately –10°C. The economic operating time of the ammonia air-cooler decreases by only 33% as the latent heat transfer potential of the enetring air increases from 5g/kg to 7g/kg, for the same air dry bulb temperature of +15°C and the same evaporating temperature of-10°C (Figure 6b).

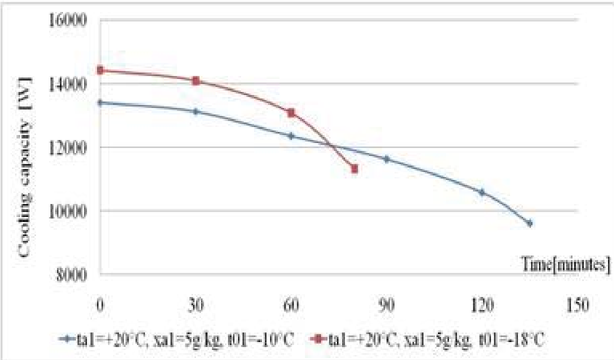
4.4. Effect of evaporating temperature [6]

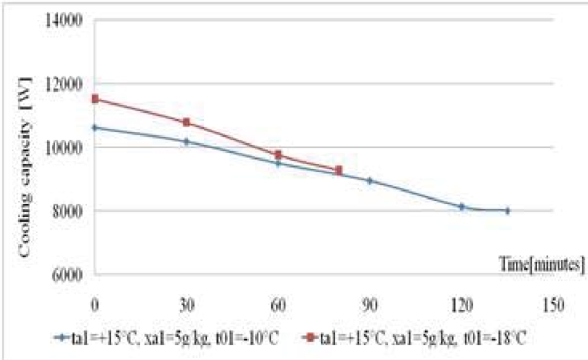
Finned-coil cooling capacity is plotted against time for two different evaporating temperatures of -10°C and -18°C in Figures 7a and 7b, respectively.

Experiments have been conducted for entering air dry bulb temperature of +20°C, +15°C and +10°C and entering air humidity ratio of 5g/kg. Figure 7 shows that regardless of the entering air dry bulb temperature level, along the common operating time, the finned-coil cooling capacity has higher values for the lower evaporating temperature (-18°C).

At lower evaporating temperature, for the same temperature of the inlet air, the mean temperature difference between air and ammonia increases, causing an increase in cooling capacity on average by 4% over the common operating time, regardless of the entering air temperature.

In addition, the evaporation temperature drop, from about -10°C to about -18°C, led to a shorter operating time needed for frost to build-up in a layer of 2.2mm ... 2.3 mm height, with about 41%.





a) b)

Figure 7. Effect of evaporating temperature

4.5. Air pressure drop [6]

Figure 8 illustrates comparative air pressure drop recorded at the end of the economic operating time, for different air dry bulb temperatures (+ 20°C and +15°C), different humidity ratio (5g/kg, 7g/kg, and 9g/kg) and the same evaporating temperature (-10°C).

As Figure 8 shows air pressure drop increases by 49.4% over the common operating time with an increase in air humidity ratio from 5g/kg to 9g/kg, for the same air dry bulb temperature of +20°C and the same evaporating temperature (-10°C) ; the increase in air pressure drop reaches only 23.2% over the common operating time as air humidity ratio increases from 5g/kg to 7g/kg, for the same air dry bulb temperature of +15°C.

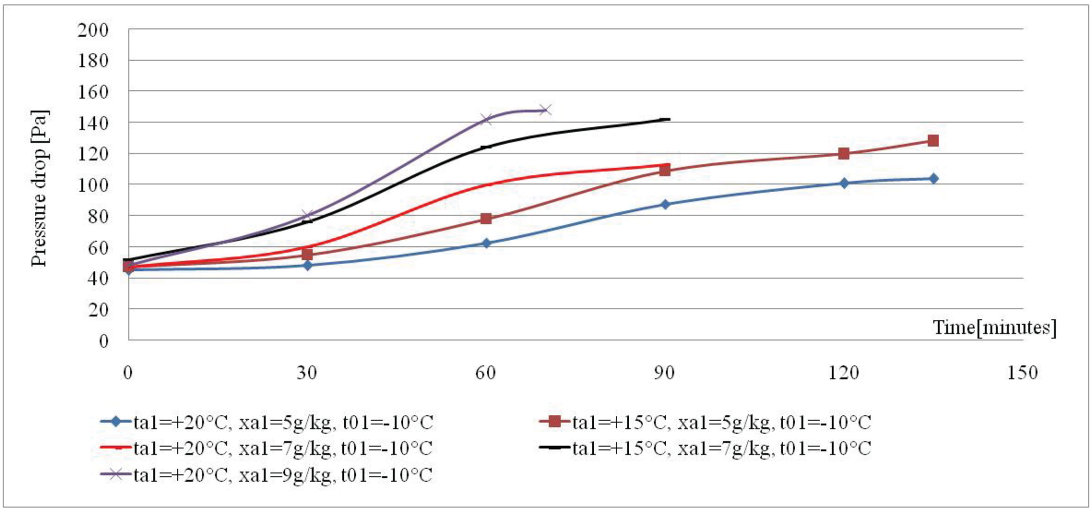


Figure 8. Effect of air pressure drop

5. CONCLUSIONS

In the present work heat transfer performances and air pressure drop of an ammonia finned-coil air-cooler of given geometrical configuration have been investigated experimentally, under different operating conditions. Experimental results led to the following conclusions:

- ammonia finned-coil cooling capacity decreases in time with a decrease in entering air temperature, for the same inlet air humidity ratio and the same evaporating temperature;
- ammonia finned-coil cooling capacity increases with entering air humidity ratio, over the economic operating time, for the same entering air dry bulb temperature and the same evaporating temperature; maximum cooling capacity, for given entering air dry bulb temperature and evaporation temperature corresponds to maximum inlet air humidity ratio;
- the economic operating time of the ammonia air-cooler decreases with an increase in the latent heat transfer potential of the enetring air, for the same air dry bulb temperature and the same evaporating temperature;
- regardless of the entering air dry bulb temperature level, along the common operating time, the finned-coil cooling capacity has higher values for the lower evaporating temperature;
- evaporation temperature drop led to a shorter operating time needed for frost to build-up in a layer of 2.2mm ÷ 2.3 mm height;
- operating time is not significantly influenced by inlet air temperature, but is significantly reduced with an increase in air humidity ratio;
- air pressure drop increases over the common operating time with an increase in air humidity ratio, for the same air dry bulb temperature and the same evaporating temperature.

6. References

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