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INFLUENCE OF POSITION AND ORIENTATION OF WATER SPRAYING ON THE EFFICIENCY OF A HEAT EXCHANGER

P.E. Vende^{1*}, F. Trinquet¹, S. Lacour¹, A. Delahaye¹, L. Fournaison¹

¹IRSTEA-GPAN

1, rue Pierre-Gilles de Gennes, CS 10030, F-92761 Antony cedex, France
e-mail: pierre-emmanuel.vende@irstea.fr

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Abstract

Spraying water in the upstream air flow of a heat exchanger improves the heat transfer. Two effects are part of this phenomenon: the spray evaporation that decreases the air temperature by adiabatic cooling and the spray impingement on the heat exchanger surface that increases the heat exchanger coefficient. In general, those effects don't apply on the entire heat exchanger surface. So, an experimental study has been carried out in order to compare the influence of several positions and orientations of water spraying on the efficiency of heat transfer. A pilot has been built allowing moving easily the atomizer injecting droplets in air while controlling flow rates, temperatures and humidity.

Among the experimental conditions, it has been chosen a low pressure atomizer that produces a hollow cone spray. This choice has been done to keep in the meantime a good atomization (Sauter mean diameter of 50 μm), a low pressure (5 bar) and low water consumption (1 L/h).

For each configuration tested, a global and a local study of the system have been made. The global study consists in an overall energy balance. Until now, the best overall energy balance reveals an increase of the heat transfer of 10% for a water flow rate injected of 1 L/h with a heat exchanger power of 5 kW.

The local study consists, on one hand, in an energy balance on a pass of the heat exchanger and, on another hand, on the characterization of the impacted surface. The energy balance is based on the microchannels temperatures of the heat exchanger pass. The impacted surface is characterized by the local cooling in the downstream flow of the heat exchanger and a qualitative measure with an infrared camera.

With all these results, an optimization of the system has been done and gave the best position and orientation of the atomizer to obtain the maximal heat transfer.

1 Introduction

According to the Intergovernmental Panel on Climate Change (IPCC), worldwide power consumption for air conditioning alone is forecast to surge 33 times by 2100 as developing world incomes rise and urbanisation advances [1]. Research by the Netherlands Environmental Assessment Agency predicts that the amount of energy used worldwide in cooling will overtake that used in heating by about 2060 [2]. Moreover, the demand for residential air conditioning alone will rise from 300 TWh in 2000, to about 4,000 TWh in 2050 and more than 10,000 TWh in 2100 [1]. As a comparison, it is about all the electricity produced in 2013 by the United States and the Republic of China combined [3]. Solutions must be finding to increase the efficiency of air conditioning and refrigeration systems in order to reduce the cold domain consumption.

For several years now, one solution is studied and already used in industry: the water spraying on the condenser of a refrigeration unit. Indeed, several works [4-7] have shown that there is an improvement of heat exchanges due whether on the spray evaporation that decreases the air temperature upstream the condenser by adiabatic cooling or the spray impingement on the heat exchanger wall surface that increases locally the heat exchanger coefficient. Those two effects have their own advantages and drawbacks. On one hand, total spray evaporation upstream the heat exchanger avoids stagnant water and so any environment where bacteria and microbial organisms can develop. It also can improve the COP up to 50 % [5] and reduce the electric cost by around 18% [7]. However, this solution requires the formation of a very fine spray with droplets diameters less than 20 μm [8]. Thus, it needs a high injection pressure (at least 50 bar) and a high energy demand. On the other hand, the spray impingement doesn't require much energy, but can cause some bacteriological problems with run-off and stagnating water. Several models and experimental measurements [9-11] have been done on wet wall heat transfer coefficients.

Irstea is one of the research institutes that is working on water application on the air flow upstream air condenser. Until now, works focus on maximizing the spray evaporation. Therefore, it has been shown numerically [6] and validated experimentally [12] that the spray should be injected in a counter-flow situation and involves droplets with mean diameter below 25 μm . It has also been assessed that, for a counter flow injection, atomizers can be located at 5 cm away from the heat exchanger and increases the heat exchange by a factor 3. Nevertheless, unlike the actual work, the energy and water consumption were not primordial parameters for industrial applications.

In the present work, the spraying system must be water self-sufficient, thanks to the condensate water of the refrigerator-unit evaporator, in order to work in all atmospheric conditions. A numerical model has established that, in a French climate, condensate water of the evaporator produces an average of 1 L/h. Furthermore, the spraying system must consume as little energy as possible to reduce more the electric cost of air conditioning. For these reasons, the spraying system chosen work under low water flow and pressure conditions. With these parameters, the evaporation rate is limited (around 20% at 10 cm of the heat exchanger in the co-flow configuration) and so the spray impingement on the heat exchanger surface is predominant. Finally, the atomizer orifice has to be placed close to the surface of the heat exchanger (less than 5 cm) because of space constraint. As the surface of the heat exchanger is large (52 \times 40 cm), the water spray will impact only partially the heat exchanger. The aim of this study is to determine the best position and orientation of the atomizer to obtain the maximal heat transfer.

The paper is organized as follows: the experimental setup will be described first with all the measurement devices and configurations tested, then experimental data will be discussed. The system will be first studied globally with an overall energy balance. Then, a discussion will be focused on the local heat exchanges at the pass level and impacted surface.

2 Experimental setup

2.1 Devices

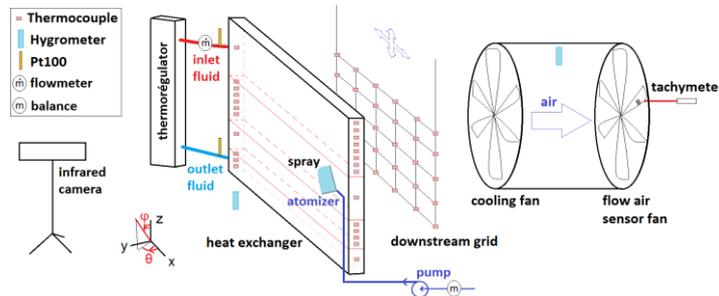


Figure 1 – Schematic view of the experimental setup

The experimental setup is illustrated in Figure 1. It aims to optimize the heat exchange obtained with spraying by varying the air flow, the water flow and the position and orientation of the spraying device beside the heat exchanger wall. The heat exchanger used is a microchannels condenser in aluminium with four passes. As the work is mainly interested in the heat exchanges on the condenser surface, only this component of the refrigeration unit is used. Thus, a water flow is generated inside the condenser by a thermo regulator in a closed loop. As water is a single phase fluid, there is no phase change in the condenser. Inlet water is heated at 80°C much above the condensation temperature of the correspondent refrigerant fluid. In this way, like the real condenser, there is a hot spot at the top of the exchanger, on the first pass. A flowmeter measures the inlet flow rate in the heat exchanger and two Pt100 measure the inlet and outlet temperatures: these three sensors allows doing an overall energy balance of the system. The inlet flow rate is fixed at about 200 L/h. To cool the fluid condenser, an air flow is sucked through the heat exchanger by the downstream cooling fan. This air flow is measured thanks to a flow air sensor fan which frequency rotation is count by a tachymeter. The whole device is located in a room where the temperature and the humidity are controlled. All the experiments are realized at the same atmospheric set point: (25°C, 40%). Two hygrometers are located upstream and downstream the heat exchanger to measure and control air humidity with and without water spraying. Thermocouples are used in two situations. On one hand, they measure microchannels wall temperatures of the heat exchanger. For each pass of the condenser, there is one thermocouple at the inlet and one thermocouple every two microchannels at the outlet (see Figure 1). It allows doing a local energy balance on passes. On the other hand, thermocouples are used to measure air temperature downstream the heat exchanger. These thermocouples are on a grid that can move up and down, and left and right. They are used to know the local cooling downstream the heat exchanger caused by spraying, as well as the heating through the heat exchanger. The spraying system is composed of an atomizer and a pump. The pump brings water from a tank to the atomizer that pulverizes it into a spray. A continuous measurement of the tank weight is done with a precision balance in order to deduce the water flow. An infrared camera is used in order to know the impacted surface by the water spray. This camera takes pictures of the entire heat exchanger surface, with and without spraying. The images have a resolution of 320×240 pixels. Thus, 5 pixels correspond to 1 cm in both directions. The picture without spraying is used as a blank test and subtracted to the pictures with spraying to highlight the impacted surface and do pattern recognition by imaging processing with Python. This pattern recognition is done by binarizing grey-level images with the Otsu method (see Figure 2).



Figure 2 - Thermal images without and with spraying, difference in grey scale levels and binarized

2.2 Protocol

Eleven position and orientations of atomizer have been chosen: they are illustrated in Figure .

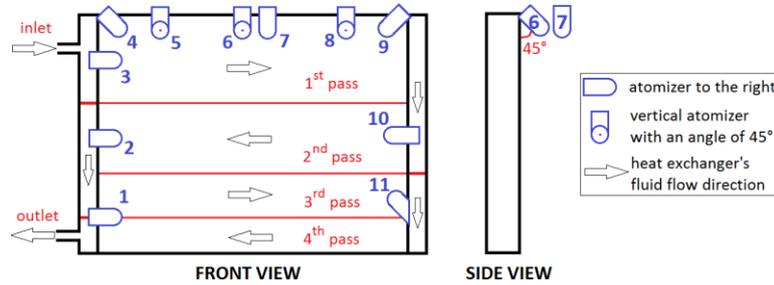


Figure 3 – Atomizer positions and orientations on the heat exchanger

In all these configurations, atomizer is parallel or with an angle of 45° to the surface of the heat exchanger. These orientations have been chosen as a compromise between counter-flow and co-flow configurations. Indeed, even if studies show that counter-flow configuration is the best way to inject water in the air, the spray takes too much space and goes back too far in the air flow for the application. The co-flow configuration is not a better way, because the spray angle is narrowed by the air flow and impacted a small surface. Atomizer must be closed to the surface of the heat exchanger. As a consequence, all the configurations are located on the edges of the heat exchanger: thus, the atomizer doesn't disturb air flow and the surface impacted by the spray can be easily seen by the infrared camera. For each of these eleven configurations, the influence of water flow and air flow rate on the heat exchange is studied. Two air flow speed and two water flow values are tested. For air flow, the two set points are named $v1$ and $v2$ in the rest of the paper and have values of respectively $420 \text{ m}^3/\text{h}$ and $775 \text{ m}^3/\text{h}$. These set points correspond to the extreme operating conditions of the cooling fan. The water spraying is tested with two pump pressures: at 5 bar and 10 bar. These pressures correspond to water flow rate of respectively 1 L/h and 1.4 L/h and are named $p5$ and $p10$ in the rest of the paper.

2.3 Treatment

In the overall energy balance, a heat gain coefficient evaluates the global gain of the system due to spraying device. This spraying heat gain η_{exp} (in %) is the ratio of the additional spraying heat flux and the heat exchanger heat flux without spraying. These quantities are given by relations:

$$Q_{hex}^{off} = \dot{m}_f C p_f (T_{in,f} - T_{out,f}^{off}) \quad (1)$$

$$\Delta Q_{hex}^{on} = \dot{m}_f C p_f (T_{out,f}^{off} - T_{out,f}^{on}) \quad (2)$$

So the spraying heat gain η_{exp} is:

$$\eta_{exp} = \frac{T_{out,f}^{off} - T_{out,f}^{on}}{T_{in,f} - T_{out,f}^{off}} \quad (3)$$

This experimental gain is compared, on one hand, to the total evaporation gain η_{evap} obtained if all water is evaporated (and so all the latent energy is exploited) and on the other hand, to the saturation gain η_{sat} obtained if the wall temperature of the heat exchanger on the impacted surface is equal to the air saturation temperature. These gains can be expressed as follow:

$$\eta_{evap} = \frac{\dot{m}_w L_v}{Q_{hex}^{off}} \quad (4)$$

$$\eta_{sat} = \dot{m}_a C p_a \frac{S_{imp}}{S_{hex}} \cdot \frac{T_{a,sat} - T_a}{Q_{hex}^{off}} \quad (5)$$

Finally, the water mass efficiency ε_w is defined as the ratio of the spraying heat gain η_{exp} on the water flow rate \dot{m}_w . This ratio allows knowing if it's better to spray a little or a lot of water.

$$\varepsilon_w = \frac{\eta_{exp}}{\dot{m}_w} \quad (6)$$

3 Results

44 configurations have been experimented: indeed, for the eleven positions chosen (see Figure), two air and water flows have been tested. First of all, let's see how the mean impacted surfaces and water efficiencies of these eleven positions are influenced by the air and water flow rates.

Table 1 – Mean impacted surfaces and water efficiencies

	$\overline{S}_{imp} (cm^2)$		$\overline{\epsilon}_w (\% \cdot L^{-1} \cdot h)$	
	v1	v2	v1	v2
p5	141	143	4.9	1.3
p10	219	173	4.7	1.2

In Table 1, the mean impacted surfaces \overline{S}_{imp} and the mean water efficiencies $\overline{\epsilon}_w$ for all atomizer positions are given for the two air and water flow rates. It can be noticed that when the pressure is low (5 bar), the impacted surface is independent of air flow rate. On the contrary, at high pressure (10 bar), the mean impacted surface decreases from v1 to v2 by about 20%. This difference can be explained by the air flow speed that narrows the spray angle. When the water pressure injection varies, the impacted surface increases from low to high pressure. This rise is more important at low air flow (+55%) than at high air flow (+21%). This can be explained by two competitive effects: the augmentation of water pressure injection increases the spray angle but the augmentation of air flow decreases the spray angle. As regards the water efficiency, it can be observed that pressure doesn't change significantly the water efficiency: more water is even better. By contrast, increase air flow causes a drop of water efficiency by about 70%. No obvious correlation is found between impacted surface and water efficiency (or heat gain): it means that another key parameter has to be taken into account.

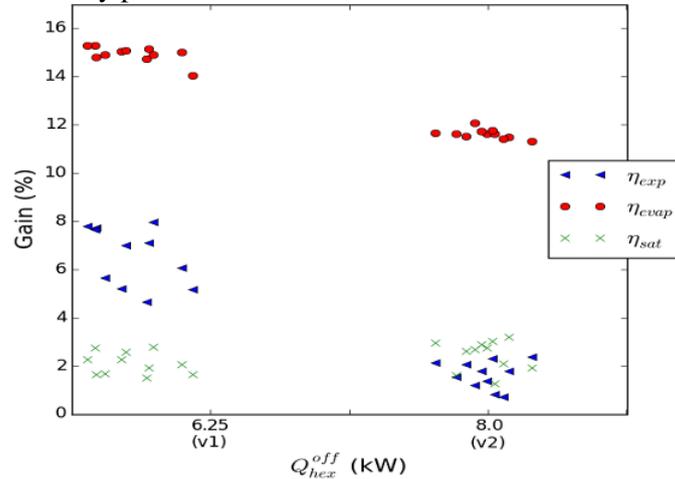


Figure 4 – Experimental, total evaporation and saturation gain at two air flow rates and $p = 10$ bar

The experimental heat gain η_{exp} , the total evaporation gain η_{evap} , and the saturation gain η_{sat} obtained for the two air flow rates tested at $p = 10$ bar are presented in Figure . At v1, η_{exp} varies between 4% and 8% depending on the position of the atomizer. Compared to the ideal gain η_{evap} , η_{exp} is divided by a factor 2 to 3. This is because there is only 10 to 20% of evaporation upstream the heat exchanger. On abscissa there is the heat exchanger heat flux without spraying. This heat flux is on average 6.25 kW at air flow speed v1 compared to 8 kW at v2. It means that the only increase of air flow speed allows increasing heat transfers at about 30%. In comparison, the maximum value of the theoretical evaporation gain at v1 is at about 15%. In theory, it would need about twice more water (~

3 L/h) to evaporate to have the gain corresponding to double the air speed. In practice, as the maximal experimental heat gain is at about 8% at v_1 , it would need at least four times more water. Moreover, η_{exp} is above η_{sat} : it means that, experimentally, the gain is not only due to a decrease of the local air temperature to its saturation value. So, an additional benefit is provided by the formation of a water film: it increases the local heat exchange coefficient and adds to the adiabatic cooling effects. On the contrary, at v_2 , η_{exp} is near η_{sat} : the formation of the water film has quite no impact on the heat exchange.

This difference is due to a smaller impacted surface at v_2 (Table 1): as the water flow is the same than at v_1 , there is more water deposit by unit surface and so the water film thickness is greater. With this higher thickness, the film is less conductive: it acts like a thermal resistance that clogs the surface of the heat exchanger. This effect of thickness and clogging is also present in Figure .

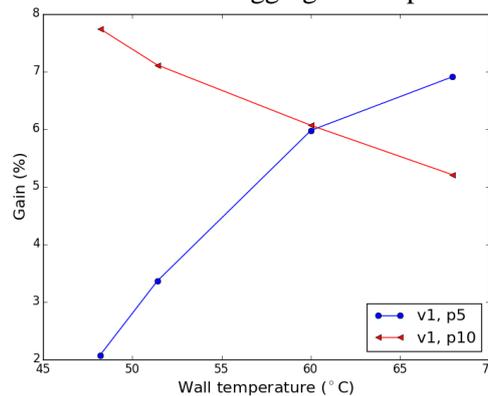


Figure 5 – Experimental gain vs wall temperature at air flow rate $v = v_1$

In Figure , η_{exp} is plotted in function of the wall heat exchanger temperature at air flow rate v_1 for the two pressures tested. The four lateral positions of atomizer (1,2,10,3) have been taken to study the wall temperature impact (see Figure). The wall temperature heat exchanger temperatures considered are the inlet or outlet passes temperatures without spraying at the level of the corresponding atomizer. These positions correspond to different wall temperatures from the bottom of the heat exchanger (the coldest) to the top (the hottest). For $p = 5$ bar, η_{exp} increases with the wall temperature. At low water flow (1 L/h), it's better to point hottest spots. On the contrary, for $p = 10$ bar, when the water flow is increased by 40%, η_{exp} decreases with the wall temperature. So, with high water flow, it's better to point coldest spots. This contradiction is due to a different water deposit. At low pressure, the water film is thin and evaporates quickly. At high pressure the film is thick, and acts like a thermal resistance. So, for the hottest spots, it's better to evaporate as quickly as possible the water deposit contrary to the coldest spots where the presence of a water thermal resistance allows increasing heat exchanges.

In Figure , the temperature profiles are compared with and without spraying, for the positions 3 and 11 (see Figure). They are plotted as function of the curvilinear abscissa. The definition of the curvilinear abscissa is also presented in Figure . On these figures, the curve named “off” correspond to the evolution of wall temperature without spraying T_{wl}^{off} , the one named “on” to the evolution with spraying T_{wl}^{on} . The difference “off – on” ΔT_{wl} is also reported. First of all, it can be noticed that the wall temperature decreasing is high for the two first passes and lower for the latter two: it has an exponential profile typical of microchannels heat exchangers. This is due to a difference between wall temperature and air temperature higher at the top of the heat exchanger: there is more heat exchanges at the top than at the bottom. For the configuration 3, the atomizer sprays water on the first pass: as a consequence, ΔT_{wl} rises on the first pass. Its value maintains on the second pass. But, at the third and four passes, ΔT_{wl} decreases slightly. This is due to the closer temperature approach between wall and

air at the end of the heat exchanger. This phenomenon is also observed for the configuration 11 at the fourth pass. For this last configuration, the spraying zone is on the second and the third passes. The cooling is better than in the configuration 3. This can be explained because the fluid has less distance to travel between the spraying zone and the outlet of the heat exchanger. Thus, in configuration 3, fluid has to cross 3 passes after spraying zone before the outlet. In configuration 11, there is only one pass. So, the fluid warming due to less effective heat exchanges because of a smaller temperature pinch between wall and air is lower. This difference of cooling is reflected in the heat gain: for the configuration 3, it is only 5% against 8% in the 11.

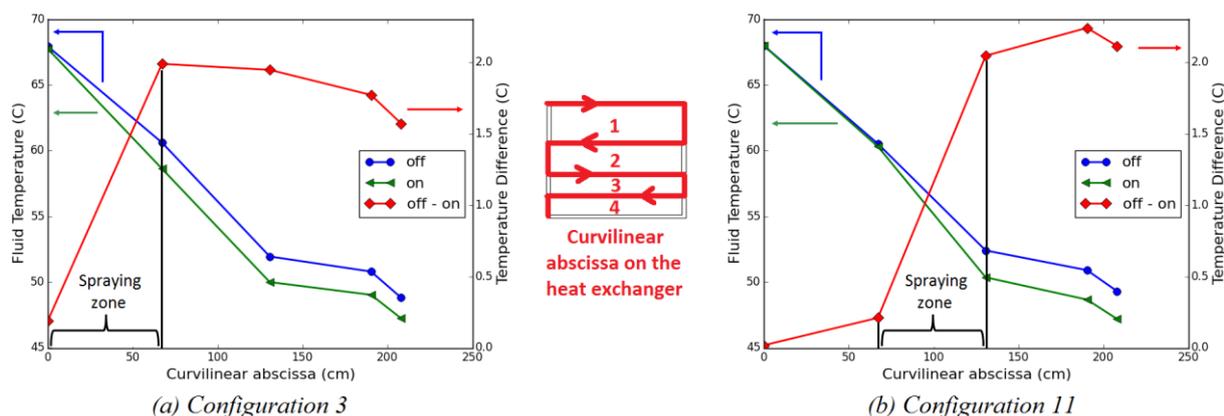


Figure 6 – Evolution of wall temperature of heat exchanger microchannels passes with and without spraying in the configurations 3 (a) and 11 (b) at $v = v1$ and $p = 10$ bar

4 Conclusion

An experimental study has been carried out in order to compare the influence of several positions and orientations of water spraying on the efficiency of heat transfer. A pilot has been built allowing moving easily the atomizer injecting droplets in air while controlling flow rates, temperatures and humidity. With this setup, it has been shown that the impacted surface by the spray depends mainly on the water pressure injection. The water efficiency depends on the air flow. But there is no obvious correlation between impacted surface and water efficiency. The position of the atomizer has an impact on the heat gained by spraying. Thus, with a low water flow, it's better to point the atomizer on the hottest spots, i.e. at the top on the heat exchanger. On the contrary, with a high water flow, it's better to point the atomizer on the coldest spots, at the bottom on the heat exchanger. This contradiction is due to a different water deposit: at low flow, the water film is thin and evaporates quick, unlike at high flow where it is thick, and acts like a thermal resistance. However, at the local scale, it has been observed that the temperature pinch between wall and air increases in the spraying zone, and then decreases along the remaining distance. This decrease is higher when the distance between the spraying zone and the outlet of the heat exchanger is more important. This loss of cooling is reflected in the heat gain with only 5% in the third configuration against 8% in the eleventh.

The next step of this work will be to study the effect of several atomizers on the heat exchange, and also to assess the interest of intermittent spraying for reducing thermal resistance of film.

Acknowledgements

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Nomenclature

Q	Heat flux (W)	ΔT	Difference of temperature ($^{\circ}\text{C}$)
ΔQ	Additional heat flux (W)	S	Exchange surface (cm^2)
C_p	Heat capacity ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)	L_v	Latent heat of vaporization ($\text{J} \cdot \text{kg}^{-1}$)
\dot{m}	Flow rate ($\text{kg} \cdot \text{s}^{-1}$)	η	Heat gain (%)
T	Temperature ($^{\circ}\text{C}$)	ε	Mass efficiency ($\% \cdot \text{L}^{-1} \cdot \text{h}$)
<i>Subscripts</i>			
off	without spraying	$evap$	evaporation
on	with spraying	sat	saturation
hex	heat exchanger	in	inlet
wl	wall	out	outlet
dn	downstream	imp	impacted
		a	air
		w	water
		f	fluid

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