International Journal of Engineering Trends and Technology
ISSN: 2231 – 5381 /doi:10.14445/22315381/IJETT-V69I12P204
Volume 69 Issue 12, 21-29, December, 2021

Theoretical Investigation of The Optimal Spray-Condenser Configuration of A Sprayed Finned-Tube Air-Cooled Condenser: Analysis of the Intensification Factor And Water Loss Spots

Ibra BOP, Biram Dieng, Seydou Nourou DIOP, and Amadou WARORE

Renewable Energy Research Team, Materials and Laser, Department of Physic, UFR SATIC, University Alioune DIOP of Bambey (UADB), Bambey 21400, BP30, Senegal

Abstract *- Spraying water upstream of an air-cooled condenser is an efficient way to improve the energetic performance of a refrigeration system. In this work, the optimal spray-condenser configuration, which allows covering the maximum surface to be cooled while avoiding as much water loss as possible, is investigated. The target area is represented by the front surface of the condenser ($S_f = 0{,}2\ X\ 0{,}2\ m^2$). The calculation model, applied to a finned-tube air-cooled condenser, is developed using Matlab software in order to visualize changes in lost water flow rates and the thermal intensification factor. It is found that the best cooling and the minimum water loss are recorded when the spray impact surface is just inscribed on the front surface of the condenser and when the misting regime is without an excess of water. In this configuration, the ratio of the spray impact area to the condenser frontal area is equal to 0,9398. But when the misting regime with an excess of water is reached, water loss through drainage is added to the total lost water flow. On the other hand, when the spray impact area is greater than the front surface of the condenser, only a portion of the water flow enters the condenser, and the intensification factor decreases. The results also show that the outer wall temperature from which one works in the misting regime without an excess of water depends strongly on the collected water flowrate (T_{wall} > 30 °C for 0,0231 mg.s^{-1} and T_{wall} > 40 °C for 0,0462 mg.s^{-1}).*

Keywords *- Optimal configuration, air-cooled condenser, thermal intensification factor, frontal area, spray and water flow rate*

Specification	Symbole	Unit
	NOMENCLATURE	
Specific heat	C_p	J. kg^{-1}. K^{-1}
Diameter	D	m
Tubes outer diameter	D_o	m
Fin thickness	E	m
Thermal transfer coefficient	h	W. m^{-2}. K^{-1}
Matter transfer coefficient	h_w	m.s^{-1}
Height of the condenser	H_{cond}	m
Latent heat of vaporization of water at the condenser outer wall temperature	ΔH_v	J. kg^{-1}
Length of the condenser	L_{cond}	m
Spray water flow rate	$\dot{m}$	m^3.s^{-1} or kg.s^{-1}
Number of tubes	N_{tube}	
Fin pitch	p_{fin}	
Exchanged heat flux	Q	W
Spray full cone angle	θ	rad
Spray half-angle	α	rad
Radius	R	m
Surface	S	m^2
The ratio of the spray impact area to the condenser frontal area		m^2
Temperature	T	K ou °C
The temperature difference between the condenser outer wall and the water misted air	ΔT	k

Speed	V	$m.s^{-1}$
Spray-condenser distance	x	m
Absolute humidity	y	$\frac{\text{kg water}}{\text{kg air}}$
NOM DIMENSIONNELS NUMBERS		
Thermal intensification factor	I	$\frac{h}{h_0}$
Colburn factor	j	$\frac{N_u}{R_e D P_r^{\frac{1}{3}}}$
Nusselt number	N_u	$\frac{hD}{\lambda}$
Prandtl number	P_r	$\frac{\mu C_p}{\lambda}$
Reynolds number	R_e	$\frac{\rho V D}{\mu}$
INDICES		
relative to the airflow	a	
relative to the captured water	capt	
GREEK LETTERS		
Thermal conductivity	λ	$W.\ m^{-1}.\ K^{-1}$
Dynamic viscosity	μ	Pa.s
Mass volume	ρ	$Kg.\ m^{-3}$
Collection rate	τ	
condenser	cond	
relative to the convective heat flux	conv	
relative to the evaporated water flow	ev	
relative to the condenser frontal area	f	
relative to the spray impact area	imp	
maximal	max	
relative to the outer surface of the condenser	o	
relative to the optimal configuration	opt	
relative to the saturated air	sat	
relative to the sensible heat flux	sen	
total	tot	
relative to the tubes of the condenser	t	
relative to the water flow rate	w	

I. INTRODUCTION

The study of the efficiency of refrigeration systems, especially those with mechanical compression, can greatly contribute to energy savings by improving their coefficient of performance, reducing greenhouse gas emissions, and preserving the environment [**1; 2; 3**]. Among their main components, the condenser plays a very important role by evacuating the heat absorbed by the refrigerant to the outside environment [**4; 5**]. In general, due to technological, economic, and environmental constraints, the use of ambient air as a means of cooling for condensers is more widespread than the use of water [**6; 7**]. In fact, the air is available in unlimited quantity, natural, free, and presents no danger for its evacuation, unlike water, which is scarce and whose use requires sewers or cooling towers. However, the air has a very low specific heat and overall heat transfer coefficient, and heat removal will require large volumes of air and large exchange surfaces [**8**]. Thus, much research on techniques for improving the efficiency of heat transfer in air-cooled condensers, especially water misting, has been developed to improve heat transfer between the air and the refrigerant. Another method for obtaining the most effective cooling of a heated surface is by jet impingement; it is applied in general where high rates of heat transfer are required thanks to its easiness to be implemented [**9**]

Misting is a technique that has long been used in the form of water curtains for firefighting and to limit the progression of dust[**10; 11**]. It involves producing and dispersing fine droplets of water into the air using injection nozzles called atomizers. It is also applied in many areas such as air conditioning, washing, and food preservation [**12; 13**].

Misting water upstream of an air condenser helps to intensify heat transfer [**14**]. After injecting the water droplets in the airflow upstream of the condenser, they evaporate, absorbing its energy while generating its adiabatic cooling. When these water droplets impact the outer walls of the condenser, a film of water forms on its surface and can result in increased heat transfer. However, water stagnation resulting from misting is very likely and may be favorable for the growth of bacteria such as Legionella, which can cause disease. Legionella infections are caused by an intracellular bacterium mainly inhabiting aquatic environments and whose contamination is through inhalation of aerosols [**14; 15**]. Thus, to avoid the health risks associated with misting, water stagnation must be avoided as much as possible by minimizing water loss while maintaining good cooling. This can be achieved by good handling of the spray-condenser parameters.

After the numerous research works carried out on the cooling of the air due to a mixture of air and water droplets, the studies were interested in applying misting on heat exchangers in order to improve their efficiency [**16**]. In 2007, Youbi Idrissi et al. [**8**] were the first to use misting upstream of an air-cooled condenser applied to a refrigeration machine. They have shown that there is a threshold water flow rate to be misted beyond which any further addition of water is unnecessary, and the intensification factor hardly changes. This threshold water flow has also been observed by Chen et al. [**17**] and Liehui Xiao et al. [**18**].

Experimental studies by Mudawar and Estes [**19**] have shown that if the spray-condenser distance is very small, only a small fraction of the surface to be cooled is impacted. They also revealed that when this distance is very large, a fraction of the water droplets impacts outside the surface to be cooled and cause water loss. They added that the critical heat flux is maximum when the surface to be cooled is just inscribed on the spray impact area and that the appropriate spray-condenser distance can be determined by knowing the spray angle and the dimensions of the heat exchanger.

In 2018, Pierre Vandé [**20**] studied the impact of the implementation of a water droplet spraying system on the efficiency of an onboard refrigeration unit. By wanting to maximize the impacted surface while maximizing the performance gain of the misting, we sought to find a parameter on which this surface depends, whatever the operating conditions. To do this, he plotted the evolution of the total spray surface as a function of its distance and showed that it strongly depended on it. Recently, in 2019, Fabien Raoult [**16**], by modeling an evaporative two-phase flow along a heated wall, showed that the best cooling of the target surface is obtained when the value of its surface is close to that of the spray impact area. He also showed that for too small angles of the spray, the ratio of these areas becomes very small in front of unity, and only a small part of the target area is cooled.

Despite all these works, the study of the nozzle-condenser configuration must be deepened in order to minimize the lost water flow while covering the maximum surface to be cooled. For the configuration in the work of Mudawar and Estes [**19**], based on the critical heat flux, where the target surface is just inscribed on the spray impact area, although it makes it possible to cover the entire target surface, it also causes water losses outside this surface which do not serve for its cooling. These losses can cause stagnation of water around the exchanger and/or reduce heat transfer. In addition, most of the abovementioned studies are performed for particular applications such as heated cylinders and square blocks. In this work, a model of analyzing the spray-condenser configuration based on the intensification factor and the water losses was proposed for a finned-tube air-cooled condenser; it is based on a mass and thermal balance. The objective is to predict water losses and their impact on the thermal intensification factor in order to better position the nozzle in relation to the condenser. This will minimize these losses while maintaining good cooling with a maximum of surface to be cooled.

II. MODELING

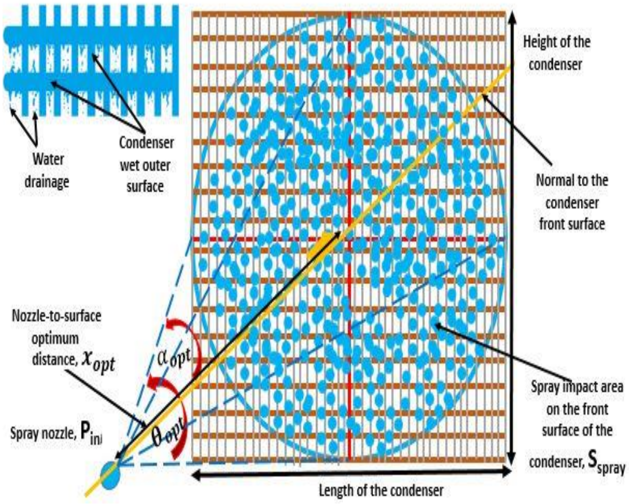

Figure 1: Spray-condenser optimal configuration

A. Masset thermal balance

In this study, a spray refers to the mixture of air and water droplets.

According to Mudawar and Estes [19], the spray volumetric flux distribution along the impact surface $\dot{m}_{w,imp}$($m^3.s^{-1}.m^{-2}$) in the case of a spray normal to the heated surface is given by:

$$\dot{m}_{e,imp} = \frac{1}{2}\left[\frac{\dot{m}_{w,spray}}{\pi(xtan\alpha)^2}\right]\left[\frac{tan\alpha^2}{1-cos\alpha}\right].\frac{1}{\left[1+\left(\frac{r}{x}\right)^2\right]^{\frac{3}{2}}} \qquad \text{Eq.1}$$

In that case, the portion of the water flow received by a surface of radius R_i is given by [**19**]:

$$port = \frac{\dot{m}_{w,f,cond}}{\dot{m}_{w,spray}} = \frac{1}{1-cos\alpha}\left[1-\frac{1}{\sqrt{1+\left(\frac{R_i}{x}\right)^2}}\right] \qquad \text{Eq.2}$$

For $R_i < xtan\alpha$, only a portion of the spray water flow rate enters the condenser ($port < 1$).

For $R_i \geq xtan\alpha$, all the spray water flow rate inters the condenser ($port = 1$).

With $\dot{m}_{w,f,cond}$(m^3.s^{-1}.m^{-2})the spray water flow rate entering the condenser, $\dot{m}_{w,spray}$ The total spray water flow was arriving upstream of the condenser and $xtan\alpha$the spray impact radius.

In this work, the total spray water flow rate arriving on the front surface of the condenser $\dot{m}_{w,spray}$ is considered to be equal to the water flow rate remaining after part of the total injected water flow $\dot{m}_{w,inj}$ Is evaporated in the air upstream of the condenser. The frontal area of the condenser is kept constant throughout the study and R_i represents half of the smallest dimension between the length L_{cond} and the height H_{cond}Of the condenser.

$$R_i = \frac{1}{2}\min(L_{,cond}, H_{cond}) \quad \text{Eq.3}$$

For simplicity, the front surface of the condenser is considered as a square($L_{cond} = H_{cond}$).

To cover the maximum surface area upstream of the condenser while confining all the flow of water droplets on its front surface (see figure 1), the following relationships must be satisfied

$$D_{spray} = L_{cond} \quad \text{Eq.4}$$

$$x_{opt} = tan^{-1}\left[\frac{L_{cond}}{2x_{opt}}\right] \quad \text{Eq.5}$$

The maximum surface area of the spray so that its impact zone is just confined to the front surface of the condenser is given as follows:

$$S_{spray,opt} = \pi\left(x_{opt}tan\alpha_{opt}\right)^2 \quad \text{Eq.6}$$

The surface ratio S_r between the spray impact and the condenser frontal area is defined as:

For $xtan\alpha < R_i$:

$$S_r = \frac{S_{spray}}{S_f} \quad \text{Eq.7}$$

For $xtan\alpha > R_i$:

$$S_r = \frac{S_f}{S_{spray}} \quad \text{Eq.8}$$

When the spray impact surface remains inscribed on the front surface of the condenser ($xtan\alpha < R_i$), the surface ratio must tend towards unity to cool the maximum surface area when the spray surface increases and all the spray water flow enters the condenser (see fig. 2aand fig. 2b). On the other hand, when the area covered by the spray becomes greater than the front surface of the condenser $xtan\alpha > R_i$, the latter is cooled, but only a portion of the spray water flow rate enters the condenser, and the surface ratio is reversed (see fig.2cand fig.2d).

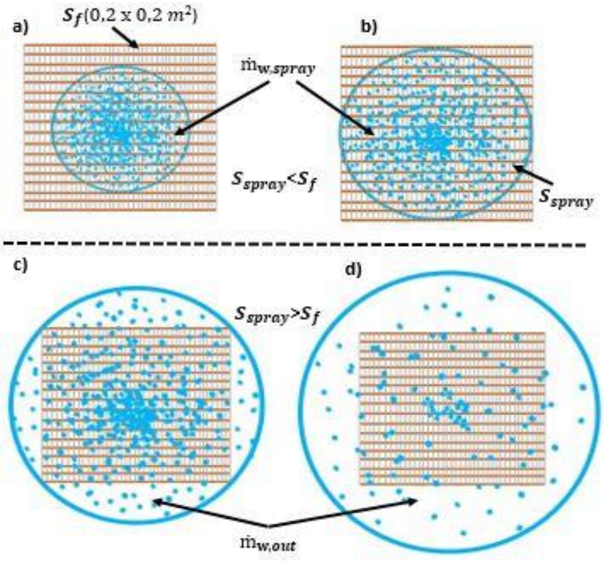

Figure 2: Volumetric flux distribution of the spray water flow rate on the target surface

Let Q_{tot}be the total heat flow supplied by the condenser and Q_0be the convective heat flux between the outer surface of the condenser and the airflow. The intensification factor is given as follows [21]:

$$I = \frac{h}{h_0} = \frac{Q_{tot}}{Q_0} \quad \text{Eq.9}$$

With h andh_0, respectively, the air external exchange coefficients with and without misting.

$$Q_{tot} = Q_0 + Q_{sen} + Q_{ev} \quad \text{Eq.10}$$

- $Q_{sen} = \dot{m}_{w,capt}C_{p,w}(T_{wall} - T_{a,mist})$, the sensible heat to bring the collected water from the temperature of the misted air $T_{a,mist}$ to the outer wall temperatureT_{wall}
- $Q_{ev} = \dot{m}_{w,ev}\Delta H_{ev}$, the heat flow necessary for the evaporation of a given quantity of collected water

The convective heat flow Q_0between the outer surface of the condenser and the misted air flow is given as follows:

$$Q_0 = h_0 S_{o,cond}(T_{wall} - T_{a,mist}) \quad \text{Eq.11}$$

In a regime without excess water, all the collected water flow rate$\dot{m}_{w,capt}$Is evaporated. It is given by:

$$\dot{m}_{w,capt} = \tau_{capt}\dot{m}_{w,f,cond} \quad \text{Eq.12}$$

The evaporated water flow rate $\dot{m}_{w,ev}$ (kg.s^{-1}) on the outer surface of the condenser is often determined from the maximum water flow rate that can be evaporated by [8; 22; 23]:

$$\dot{m}_{w,ev} = S_{w,tot}h_w\rho_a(y_{sat(Twall)} - y_{sat(Ta,mist)}) \quad \text{Eq.13}$$

With $S_{w,tot}$ the condenser wetted the outer area and h_w the matter transfer coefficient obtained by the Chilton-Colburn heat-matter analogy and is given as follows [8]:

$h_w = \frac{h_0}{\rho_a C_{p,a}}$ Eq.14

The intensification factor in a misting regime without an excess of water is given by:

$I = 1 + \frac{\tau_{capt}\dot{m}_{w,f,cond}C_{p,w}}{S_{o,cond}h_0} + \frac{\tau_{capt}\dot{m}_{w,f,cond}}{S_{o,cond}h_0}.\frac{\Delta H_v}{\Delta T}$ Eq.15

In a misting regime with an excess of water, it is given by:

$I = 1 + \frac{\tau_{capt}\dot{m}_{w,f,cond}C_{p,w}}{S_{o,cond}h_0} + \frac{S_{w,tot}}{S_{o,cond}}.\frac{y_{sat(Twall)} - y_{sat(Ta,mist)}}{(T_{wall} - T_{a,mist})}.\frac{\Delta H_v}{C_{p,a}}$ Eq.16

When the evaporation of the water droplets upstream of the condenser reaches saturation, the air temperature after misting ($T_{a,mist}$) is equal to its wet-bulb temperature (T_{wb}).

In this study, we will consider this hypothesis and the minimum flow of water to be misted to saturate the air $\dot{m}_{w,inj,min}$ is given by:

$\dot{m}_{w,inj,min} = \dot{m}_a(y_{sat(Ta,mist)} - y_{(Ta)})$ Eq.17

With $\dot{m}_{w,f,cond}$, $\dot{m}_{w,capt}$ and $\dot{m}_{w,inj,min}$ in kg.s^{-1}

The external air coefficient of convection h_0 is obtained thanks to the Colburn factor j determined experimentally by [**22**].

$j = 0.394 R_{e,D_o}^{-0.392}\left(\frac{E}{D_o}\right)^{-0.0449} N_{tube}^{-0.0897}\left(\frac{p_{fin}}{D_o}\right)^{-0.212}$ Eq.18

We deduce the external heat exchange coefficient on the airside by:

$h_0 = \frac{\lambda_a j R_{e,D_o} P_r^{\frac{1}{3}}}{D_o}$ Eq.19

B. Analysis of water loss spots

When misting water upstream of an air-cooled condenser, the total lost water flow $\dot{m}_{w,lost,tot}$ is the sum of the quantities of water lost through the condenser $\dot{m}_{w,cross}$, outside the condenser $\dot{m}_{w,out}$ and by drainage $\dot{m}_{w,drain}$ When the misting regime with an excess of water is reached. Water losses likely to occur upstream of the condenser before the spray impact are not considered. The total lost water flow is thus given as follows:

$\dot{m}_{w,lost,tot} = \dot{m}_{w,cross} + \dot{m}_{w,out} + \dot{m}_{w,drain}$ Eq.20

For $xtan\alpha < R_i$, $\dot{m}_{w,spray} = \dot{m}_{w,fr,cond}$

In a misting regime without an excess of water, $\dot{m}_{w,ev} = \dot{m}_{w,capt}$ and $\dot{m}_{w,drain} = 0$

$\dot{m}_{w,lost,tot} = \dot{m}_{w,f,cond}.(1 - \tau_{capt})$ Eq.21

In a misting regime with an excess of water, $\dot{m}_{w,ev} < \dot{m}_{w,capt}$

$\dot{m}_{w,lost,tot} = \dot{m}_{w,f,cond} - \dot{m}_{w,ev}$ Eq.22

For $xtan\alpha > R_i$, $\dot{m}_{w,f,cond} = port.\dot{m}_{w,spray}$

In a misting regime without an excess of water, $\dot{m}_{w,ev} = \dot{m}_{w,capt}$ and $\dot{m}_{w,drain} = 0$

$\dot{m}_{w,lost,tot} = \dot{m}_{w,spray}(1 - port.\tau_{capt})$ Eq.23

In a misting regime with an excess of water, $\dot{m}_{w,ev} < \dot{m}_{w,capt}$

$\dot{m}_{w,lost,tot} = \dot{m}_{w,spray} - \dot{m}_{w,ev}$ Eq.24

III. RESULTS AND DISCUSSIONS

To carry out this study, for simplicity, the front surface of the condenser is in the form of a square of surface $S_f = 0.2 \text{ X } 0.2\, m^2$. The initial air temperature is set at 25 °C with a relative humidity of 30%, and the characteristics of the air-cooled condenser are presented in table 1. The horizontal axis of the nozzle coincides with the normal to the front surface of the heat exchanger. The collection rate is set at 60%, and we considered that half of the condenser outer surface is wetted. We used the same air flow rate ($\dot{m}_a$ =4200 m^{3}h^{-1}) as in the work of Youbi Idrissi et al [**8**]. The water droplets' average diameter is taken as equal to 30 µm, and their injection is made co-currents with the airflow. The injection pressure is sufficient so that all the non-evaporated water droplets upstream of the condenser can impact its outer surface.

Table 1: The air-cooled condenser characteristics

Parameters	**Values**
Tube length, $L_{en,t}$ (m)	0.2
Tube outer diameter, D_o (m)	0.016
Tube inner diameter, D_{in} (m)	0.013
Tubethermal conductivity (copper), λ_t ((W. m^{-1}. K^{-1}))	360
Tube vertical pitch, $p_{v,t}$	0.045
Tube horizontal pitch, $p_{h,t}$	0.045
Fin width, l_{fin} (m)	0.1
Fin thickness, E (m)	0.0003
Fin pitch, p_{fin}	0.001
Fin height, H_{fin} (m)	0.2
Fin thermal conductivity (aluminum), λ_{fin} ((W. m^{-1}. K^{-1}))	203

A. Influence of the spray-condenser distance

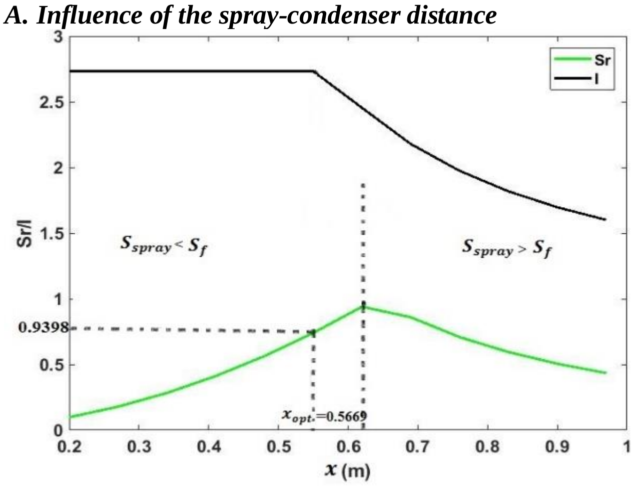

Figure 3: **Surface ratio and the intensification factor as a function of the nozzle-wall distance**

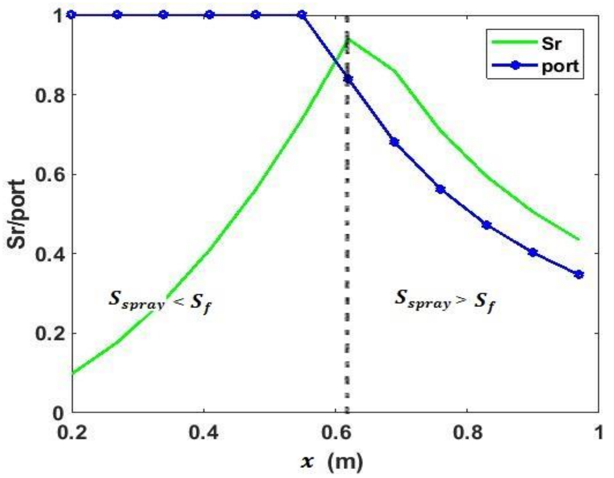

Figure 4: Condenser inlet water portion and the surface ration as a function of the nozzle-condenser distance

Figure 3 shows that when the spray-condenser distance increases for a given angle of the spray (20 °), the spray impact area increases, tends to equalize to the condenser front surface, and the surface ratio remains less than unity. In this zone, the condenser front area is greater than the spray impact surface ($S_{spray} < S_f$). From the equality between these two surfaces, when the spray-condenser distance increases, the spray surface becomes greater than the front surface of the condenser ($S_{spray} > S_f$) and the surface ratio is reversed. As for the intensification factor, it is constant when the surface of the spray is smaller than the front surface of the condenser and begins to decrease when the values of the two surfaces approach for spray-wall distances beyond 0.5669 m. This distance beyond which the intensification factor begins to decrease is called the optimum nozzle-condenser distance. It can be seen that as long as the spray-condenser distance is less than it or equal to it, the intensification factor is a constant. This evolution of the intensification factor can be explained from **figure 4,** which clearly shows that for values of the spray-wall distance less than or equal to the optimal distance, the entire spray water flow rate enters the condenser ($port = 1$). This is what justifies the constancy of the intensification factor, which depends on the spray water flow rate. On the other hand, when the spray-condenser distance becomes greater than its optimum value, the portion of the water flow rate that impacts the condenser outer surface decreases ($port < 1$), and so is the intensification factor. Therefore, the cooling is maximum when all the spray water flow is confined to the front surface of the condenser. However, when the spray-condenser distance is very small, the ratio of the area of the spray to the front area of the condenser is very small, and only a small portion of the outer surface of the condenser will be cooled. On the other hand, when this distance is optimized so that the spray impact area is very close to the front surface of the condenser and is just inscribed on it, the surface ratio is close to unity ($S_r = 0.9398$) and a large part of the condenser outer surface is cooled. **Figure 4** also shows that the portion of the condenser inlet spray water flow rate begins to decrease long before the impact surface of the spray can equalize with the front surface of the condenser. After this equality, the portion of the spray water flow rate and the surface ratio drop roughly at the same rate, keeping a constant gap.

B. Influence of the spray angle

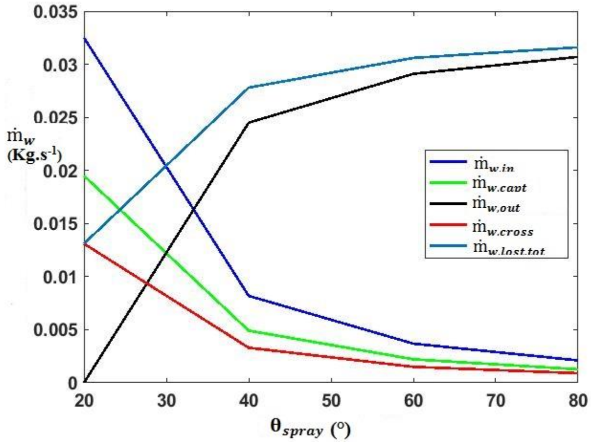

Figure 5: The water flow rates as a function of the spray angle in a misting regime without an excess water

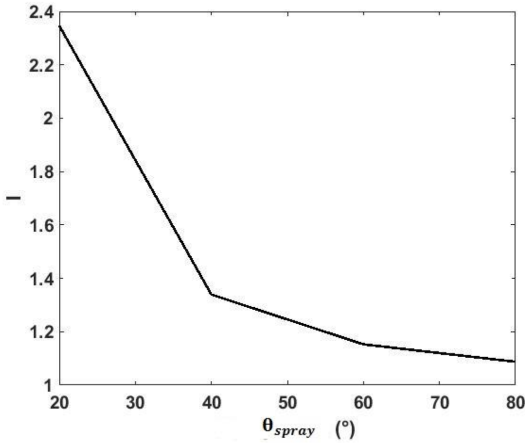

Figure 6: The intensification factor as a function of the spray angle

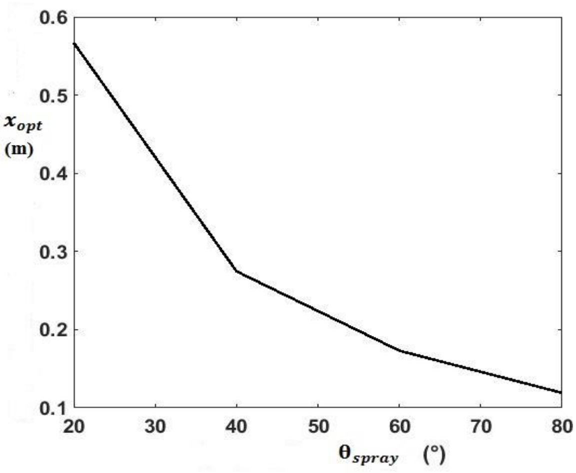

Figure 7: Optimal spray-condenser distance as a function of the spray angle

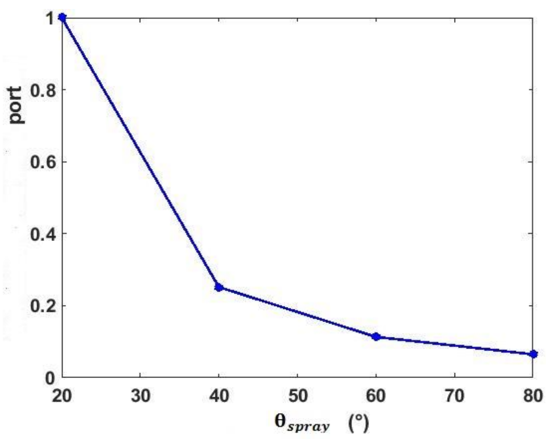

Figure 8: Condenser inlet spray water flow portion as a function of the spray angle

The analysis of the spray water flow rate as a function of the spray angle in **figure 5** shows that when it increases from the optimal value (20 ° for $x_{opt} = 0.5669\ m$ in our study), the total lost water flow increases very quickly. This sudden water loss is due to the increase in the water flow lost outside of the front surface of the condenser, causing the portion of the condenser inlet water flow rate to decrease very quickly, as shown in **figure 8**. Indeed, when the spray angle becomes greater than its optimal value, the spray impact surface tends to be greater than the condenser front surface, and the flux of water droplets impacting outside this surface increases. This implies a decrease in the condenser inlet spray water flow rate resulting in a drop in the collected water flow and in the water flow lost through the condenser, as shown in **figure 5**. In this figure, we also notice that when the condenser is at its optimal position ($x_{opt} = 0.5669\ m$ and $\theta_{opt} = 20$ °), there is no lost water flow outside the condenser frontal surface, and the only spots of water loss are reduced to the water flow lost through the condenser, which depends on the collection rate. The reduction in the condenser inlet water flow rate noted in this figure justifies the decrease in the intensification factor from its maximum value, as indicated in **figure 6**.

Figure 8 shows the evolution of the optimal spray-condenser distance according to the optimal spray angle. We note that when this angle increases, the optimal nozzle-condenser distance decreases in order to keep the spray impact surface just inscribed on the front surface of the condenser and to avoid water losses outside this surface (see **figure 7**).

C. Influence of the condenser inlet spray water flow rate

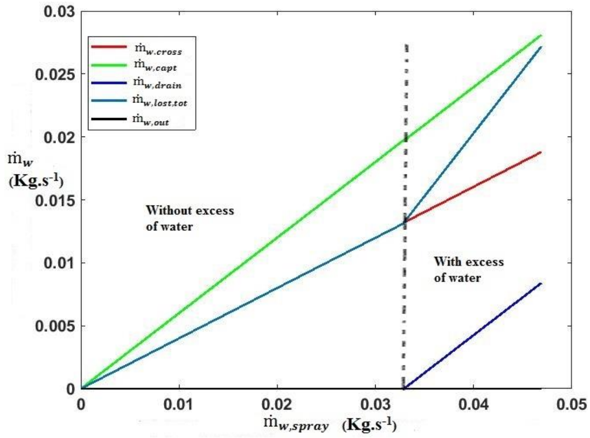

Figure 9: Water flows as a function of the condenser inlet spray water flow rate

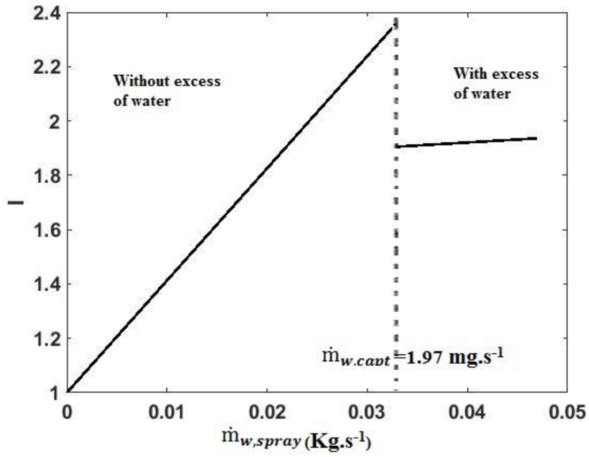

Figure 10: Theoretical evolution of the intensification factor as a function of the condenser inlet spray water flow rate

In the optimal configuration, when the condenser inlet water flow increases in a misting regime without an excess of water, as shown in **figure 9**, the collected water flow and the total lost water flow increase while the drained water flow and the lost water flow outside the front surface of the condenser are zero. The total lost water flow is reduced to the lost water flow through the condenser. However, when the condenser inlet water flow becomes greater than a threshold value of 0.0329 mg.s^{-1}, the drained water flow is different from zero and increases very quickly with the condenser inlet water flow. This drainage is due to the fact that the collected water flow becomes very large that it can no longer be completely evaporated; we are in the misting regime with an excess of water. From this threshold water flow, the total lost water flow increases sharply due to the drained water flow, which is added to the lost water flow through the condenser.

In **figure 10**, there is a rapid and linear increase in the thermal intensification factor as a function of the condenser inlet water flow; we are in the misting regime without an excess of water where all the collected water flow is evaporated. On the other hand, when this water flow reaches a threshold value ($\dot{m}_{w,capt} = 0.0197$ kg.s^{-1}), the intensification factor hardly changes, and any addition of water is unnecessary; we are in the misting regime with an excess of water. The very slow evolution of the intensification factor in this regime could be due to the fact that the temperature of the excess water will tend to equalize the temperature of the outer wall. When we continue to increase the condenser inlet water flow, the new collected water flow at the temperature of the misted air will tend to replace the water excess, and these two fluids mix by exchanging sensible heat to reach a final temperature slightly lower than that of the outer wall. The exchanged heat is thus reduced to the amount of heat necessary to bring this mixture to the temperature of the outer wall.

D. Influence of the condenser outer wall temperature

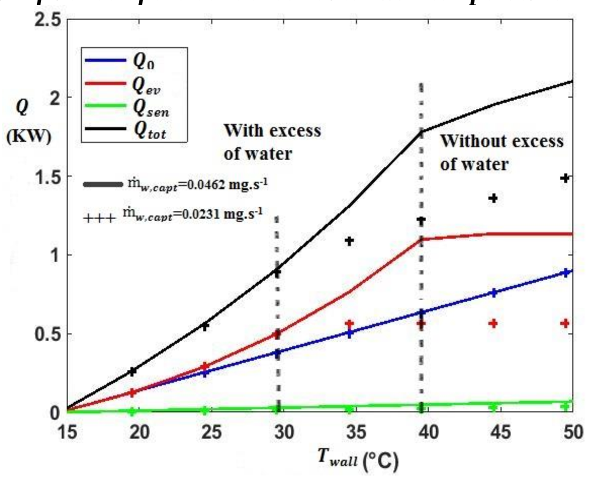

Figure 11: The theoretical heat flux as a function of the outer wall temperature for two collected water flow rates (0,0462 et 0,0231 mg.s^{-1}) and an inlet airflow temperature equal to its wet-bulb temperature at 25 °C and 30% of relative humidity

Figure 11shows the theoretical shape of the exchanged heat flow as a function of the outer temperature. This evolution is divided into two parts. In the first part, for temperatures below 40 °C and for a collected water flow rate of 0.0462 mg.s^{-1}, the heat flux increases very rapidly as a function of the surface temperature, due to the increase of the evaporated water flow, we are in the misting regime with an excess of water where only part of the collected water flow is evaporated. In the second part, for the same collected water flow and for temperatures above 40 °C, the rate of the exchanged heat flux slows down due to the constancy of the heat flow exchanged by evaporation of the collected water, we are in the misting regime without the excess of water where all the collected water flow is evaporated. In this zone, the increase in the exchanged heat flux is only due to the convective heat flux between the outer wall and the misted air and to the sensible heat flux due to the heating of the collected water. This sensible heat, which depends on the collected water flow, is very small and negligible compared to the evaporative and convective heat flows regardless of the outer wall temperature. When the collected water flow rate is reduced to its half (0.0231 mg.s^{-1}), it is observed that the misting regime without an excess of water is reached for surface temperatures above 30 °C. On the other hand, the convective heat flux remains the same for the two collected water flows and depends only on the temperature difference between the misted air and the condenser outer wall. These results are similar to those obtained by Allais et al. [**21**], who found a misting regime without an excess of water for surface temperatures $T_{wall} > 45$ °C for a collected water flow of 7.5 and for temperatures $T_{wall} \geq$ 20 °C for a low flow rate of collected water of 0.75 mg.s^{-1}.

IV. CONCLUSION

In this work, the optimal nozzle-condenser configuration for a finned-tube air-cooled condenser placed in a misted airflow was investigated; the water loss spots were analyzed, and the thermal intensification factor was studied. At the end of this study, we concluded that water losses outside of the condenser could be avoided when the flux of water droplets arriving upstream of the condenser is confined to its front surface. We also concluded that the maximum cooled surface is reached with a maximum intensification factor when the impact surface of the spray is just inscribed on the front surface of the condenser. In this configuration, the only spots of water loss are limited to the water flow lost by the drainage of the excess of water when this regime is reached and to the water flow lost through the condenser, which depends on the collection rate.

REFERENCES

[1] Eli A. Goldstein, Aaswath P. Raman, Shanhui Fan, Sub-ambient non-evaporative fluid cooling with the sky, Nature Energy, 2 (2017) 17143. https://doi:10.1038/nenergy.2017.143;September.

[2] P.E. Vende, F. Trinquet, S. Lacour, A. Delahaye, L. Fournaison, Influence of position and orientation of water spraying on the efficiency of a heat exchanger, 4th International Conference on Contemporary Problems of Thermal Engineering (CPOTE), Sep. 2016, Katowice, Poland. 8 p. hal-01469450, (2016).

[3] Chung-Neng Huang, Ying-Han Ye, Development of a water-mist cooling system: A 12,500 Kcal/h air-cooled chiller, Energy Reports , 1 (2015) 123–128, http://dx.doi.org/10.1016/j.egyr.2015.04.002. May. Elsevier.

[4] Ahmet Cihan, Kamil Kahveci, Alper Tezcan, Oktay Hacıhafızoğlu, Flow and Heat Transfer Around an Air-Cooled Coil Condenser, Proceedings of the World Congress on Mechanical, Chemical, and Material Engineering(MCM 2015) Barcelona, Spain – July 20-21, (2015), Paper No. 316.

[5] Won-Jong Lee, Ji Hwan Jeong, Heat Transfer Performance Variations of Condensers due to Non-uniform Air Velocity Distributions, IJR (2016),http://dx.doi.org/doi:10.1016/j.ijrefrig.2016.05.009. April.

[6] Ruonan Jin, Xiaoru Yang, Lijun Yang, Xiaoze Du, Yongping Yang, Square array of air-cooled condensers to improve thermo-flow performances under windy conditions, IJHMT 127 (2018)

717–729, https://doi.org/10.1016/j.ijheatmasstransfer.2018.06.135. Elsevier.

[7] R. Al-Waked, M. Behnia, The performance of natural draft dry cooling towers under crosswind: CFD study, International Journal of Energy Research 28 (2004) 147–161.

[8] M. Youbi-Idrissi, H. Macchi-Tejeda, L. Fournaison, J. Guilpart, Numerical model of sprayed air-cooled condenser coupled to the refrigerating system, Energy Conversion and Management, 48 (2007) 1943–1951. January. Elsevier.

[9] Seydou Nourou Diop, Biram Dieng, Izuru Senaha, A study on heat transfer characteristics by impinging jet with several velocities distribution, Case Studies in Thermal Engineering 26 (2021) 101111, https://doi.org/10.1016/j.csite.2021.101111. May. Elsevier.

[10] Jerzy Gałaj, Tomasz Drzymała, Piotr Piątek, Analysis of influence of tilt angle on the distribution of water droplets diameters in a spray generated by the Turbo Master 52 nozzle, Procedia Engineering 172 (2017) 300 – 309, https://doi:10.1016/j.proeng.2017.02.118; Elsevier.

[11] Zhizhong Zhang, Qinggang Li, Experimental Study on a New-type of Water Mist Extinguishing Nozzle, Proceedings of 2012 International Conference on Mechanical Engineering and Material Science (2012). Atlantis Press.

[12] F. Raoult, S. Lacour, F. Trinquet, L. Fournaison, A. Delahaye, Eulerian model and impact of physical parameters on the evaporative cooling of a spray of water droplets, 13th French-Quebec interuniversity conference on the thermal systems (2017).

[13] Santosh Kumar Nayak, Purna Chandra Mishra, Sujay Kumar Singh Parashar, Influence of spray characteristics on heat flux in dual-phase spray impingement cooling of a hot surface, Alex. Eng. Journal, 55 (2016) 1995–2004,http://dx.doi.org/10.1016/j.aej.2016.07.015. July. Elsevier.

[14] S. Cattana, G. Thizy, A. Michon, J.-B. Arlet, F. Lanternier, D. Lebeaux, S. Jarraud, J. Pouchot, E. Lafont, Actualités sur les infections à Legionella, La Revue de médecine interne (2019),https://doi.org/10.1016/j.revmed.2019.08.007. Elsevier.

[15] Kiran Paranjape, Emilie Bédard, Lyle G. Whyte, Jennifer Ronholm, Michèle Prévost, Sébastien P. Faucher, Presence of Legionella spp. in cooling towers : the role of microbial diversity, Pseudomonas, and continuous chlorine application, Water Research 169 (2020) 115252. https://doi.org/10.1016/j.watres.2019.115252; October. Elsevier.

[16] Fabien Raoult, Modélisation d'un écoulement diphasique évaporatif le long d'une paroi chauffée, Génie des procédés, Sorbonne Université, Jan. (2019). Français. NNT : 2019SORUS339. tel-03010497.

[17] C. W. Chen, C. Y. Yang, Y. T. Hu, Heat Transfer Enhancement of Spray Cooling on Flat Aluminum Tube Heat Exchanger, Heat Transfer Engineering, 1(34) (2013) 29-36.

[18] Liehui Xiao, Zhihua Ge, Lijun Yang, Xiaoze Du, Numerical study on performance improvement of the air-cooled condenser by water spray cooling, IJHMT, 125 (2018) 1028–1042. https://doi.org/10.1016/j.ijheatmasstransfer.2018.05.006; May. Elsevier.

[19] I. Mudawar, K. A. Estes, « Optimizing and Predicting CHF in Spray Cooling of a Square Surface », Journal of heat transfer, 118(672) (1996). August.

[20] Vende, P. E., S. O. L. Lacour, F. Trinquet, A. Delahaye and L. Fournaison (in review). « Retention and overlapping in long-term multi-nozzle water spraying » February 2019 International Journal of Thermal Sciences 136(2) (2019) 519-529 DOI:10.1016/j.ijthermalsci.2018.09.019

[21] I. Allais, C. Alvarez et D. Flick, « Analyse du transfert thermique entre un cylindre et un écoulement d'air faiblement chargé en gouttelettes d'eau », Rev Gén Therm 36 (1997) 276-288. Mars. Elsevier.

[22] J. Tissot, P. Boulet, F. Trinquet, L. Fournaison, H. Macchi-Tejeda, Air cooling by evaporating droplets in the upward flow of a condenser, International Journal of Thermal Sciences 50 (2011) 2122-2131, https://doi:10.1016/j.ijthermalsci.2011.06.004

[23] P. Boulet, J. Tissot, F. Trinquet, L. Fournaison, Enhancement of heat exchanges on a condenser using an airflow containing water droplets, Applied Thermal Engineering, 50 (2013) 1164-1173.