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NUMERICAL COMPARISON OF AN ULTRASONIC MIST GENERATOR AND AN ULTRASONIC SPRAY ATOMIZER USED AS EVAPORATIVE COOLERS

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1. Abstract

One of the techniques that is currently being developed to improve the efficiency of air conditioning equipment is using evaporative cooling to pre-cool the air that reaches the condenser of a refrigeration machine, this reduces the condensing pressure and improves performance. Previous works studied the use of an ultrasonic mist generator and obtained promising results. However, the mist generator has some limitations, as not being able to control the distribution of the droplets. The main objective of this work has been to develop a numerical model of an ultrasonic spray atomizer that overcomes these limitations.

The spray injections have been modified depending on the different arrangements of the spray atomizer studied. The commercial code ANSYS FLUENT will be used to numerically solve the governing equations. It has been interesting to explore the influence of the number of injections and their location. Experimentally this is complex, but from the CFD it can be configured easily. In view of the results, the evaporative performance is similar to that obtained with the mist generator, $\overline{\eta^{L}}$ = 0.416, however, the spray atomization system has two great strengths. With this system, the wet length is considerably reduced (L_w/L= 0.308) for the most unfavorable case. With the mist generator it was always higher than unity. The performance remains constant when increasing the mass flow rate of water, unlike the mist generator where it was reduced. The maximum COP for all the simulations studied was 6.587, for 144 atomizers.

2. Introduction

The energy consumption used in buildings around the world today in air conditioning systems or electric fans to stay cool accounts almost 20% of their total electricity consumption. This further development will have implications for countries' overall energy demand, putting pressure on electricity grids and increasing local and global emissions [1]. An answer to achieve a more sustainable path necessarily involves the use of renewable energy and the use of very efficient technologies.

Evaporative cooling techniques applied to the condenser of a refrigerating machine represent one of the most effective and immediately applicable solutions for improving the efficiency of domestic and commercial air conditioning systems worldwide.

Applications of ultrasonic energy to enhance a wide variety of processes or to improve system efficiency have been explored in recent years. Yao [2] made an overview of studies about the applications of ultrasound as a new technology in the field of Heating, Ventilation and Air-

Conditioning (HVAC). He claimed that, from a general point of view, all the effects produced by ultrasound could be interesting in applications involving heat or mass transport, decreasing both the external and internal resistance to transport. Recently, Yao et al. [3] presented a review of the state-of-the-art of high-intensity ultrasound and its applications. However, the authors did not specifically cite evaporative cooling as an application of ultrasound, denoting the little attention they have received to date.

Computational Fluid Dynamics (CFD) techniques have been used extensively in recent decades as an effective and powerful tool capable of characterise spray cooling in different applications. CFD enables scientists and designers to understand the physical principles of spraying processes saving time and effort in tedious experimental work. Literature review shows that several authors face the analysis of spray cooling for different applications using CFD models. For example, Sadafi et al. [4] showed the possibility of using saline water for pre-cooling the inlet air to a heat exchanger. They develop a CFD model to predict the wet length of saline water droplets in spray cooling for different ambient conditions and spray characteristics. However, although it has been possible to study several cases of CFD simulations of droplet spraying for pre-cooling, few references have been found in the bibliographic review related to the numerical simulation of ultrasonic spray atomizers, and none of them was related to the evaporative pre-cooling of the inlet air of the condenser in air conditioning applications. Recently, Kim et al. [5] numerically simulated an ultrasonic gas atomizer. Through numerical simulation, they decided generation position and operating conditions of ultrasound atomizer. As last references, there would be the previous works of this research group. Martínez et al. [6] shows the design and the first tests in a ultrasonic mist generation. Its thermal performance and its water mist production capacity were assessed in terms of the mass flow rate of water and size distribution of the droplets generated. For the design of the prototype and based on the experimental data collected, a numerical model was developed in CFD. Which made it possible to carry out parametric studies to know the effect that certain operating variables had on the cooling capacity and the distribution of the generated mist, Ruiz et al. [7]. Although the tests were satisfactory and their scientific interest has been revealed by the recent publications achieved, there are still many aspects related to the search for the optimal design of the pre-cooling system based on ultrasound techniques and its integration into commercial equipment. Therefore, this work seeks to optimize the design of the ultrasonic mist generation system from an energy perspective along with minimize the impact of droplets on the condenser heat exchanger. Another issue that has to be solved is the difficulty of achieving homogeneous cooling in the whole air stream. To do that, other ultrasonic methods of water mist or spraying are proposed. The main modification proposed is the change in the droplet generation system, if in the cited works a mist generator was used, in the current study the use of ultrasonic spray atomizers is proposed.

In conclusion, the works referring to the precooling of the inlet air with ultrasonic atomization techniques have been reviewed and it has been observed that this technique has received limited attention for this application, with few references found in the literature to date. Despite this, it has been proven that ultrasound is a promising method to improve the design of evaporative pre-cooling systems. Therefore, the main objective of this study will be to develop a numerical model of an ultrasonic spray atomizers system and compare the results with those obtained with an ultrasonic mist generator. The objective is the optimization of the pre-cooling system taking into account its key operating parameters: maximize evaporative efficiency, increasing the useful area and reducing the wet length.

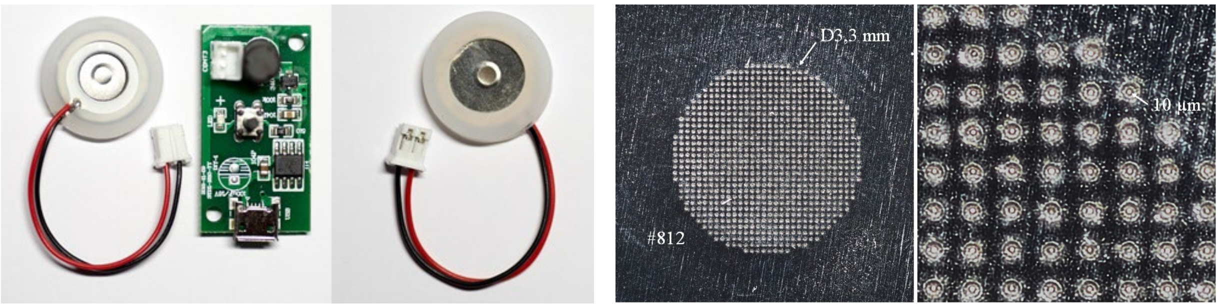
3. Materials and method

3.1. Experimental test facility

To obtain key parameters that will be used in the simulations, a series of experimental tests have been carried out with the ultrasonic spray atomizer that will be used in the future in the experimental prototype. The test equipment mainly consists of two components: an ultrasonic spray atomizer system and a wind tunnel.

The device used in the system is a hybrid mesh and ultrasound spray atomizer (Figure 1 (a)). This is composed of a ceramic piezoelectric, which surrounds a porous membrane, and a PCB controller board that generates a pulse signal at a frequency of 108 kHz for the spray atomizer. This component is characterised by having a low cost and consumption (2W).

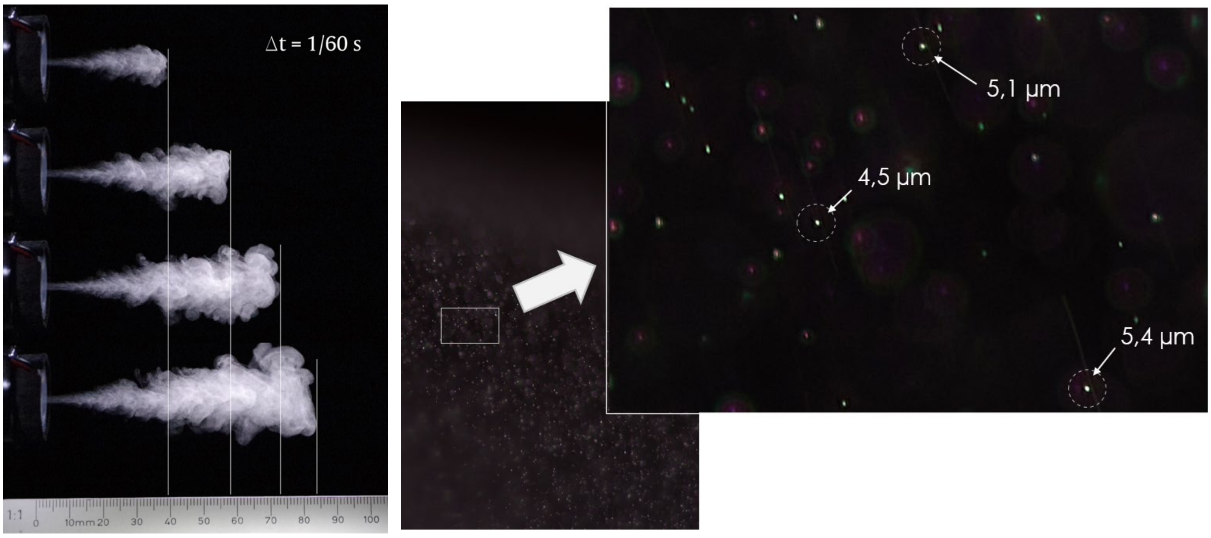
When the current is supplied, the piezoelectric initiates an expansion/contraction cycle, this oscillation allows the water to pass through its microscopic holes (around an opening of 10 µm, Figure 1 (b)), and pushes the drops forming a column of water mist. The mass flow rate is approximately 8.33 x 10^{-6} kg s^{-1}.



(a) (b)

Figure 1. (a) Used ultrasonic spray atomizer. (b) Microscopic image of the water outlet holes in the spray atomizer used.

In order to characterise the real geometry of the spray atomizer discs and the dimensions of the atomised drops, with the aim of making a numerical model as close as possible to reality, it was decided to adopt the photographic techniques described in Martinez et al. [6]. An example of these is shown in Figure 2.



(a) (b)

Figure 2. Experimental test carried out to define: (a) the speed of exit of the drops, and (b) the output size of the drops.

3.2. Physical model

The physical domain considered in the simulations reproduces a portion of the experimental facility where the future tests will be performed. It consists of a 2.5 m long, 0.492 m wide and 0.712 m high (L x b x h) domain (Figure 3 (a)). The water spray injections will be located at certain points of the air inlet section and in the direction of the air outlet section. The reason is that it will be there where the manifold will be located where the spray atomizers will be installed. As an example, in Figure 3 (b) a 9 x 9 collector is shown which are included in a square of 0.35 × 0.35 m^2.

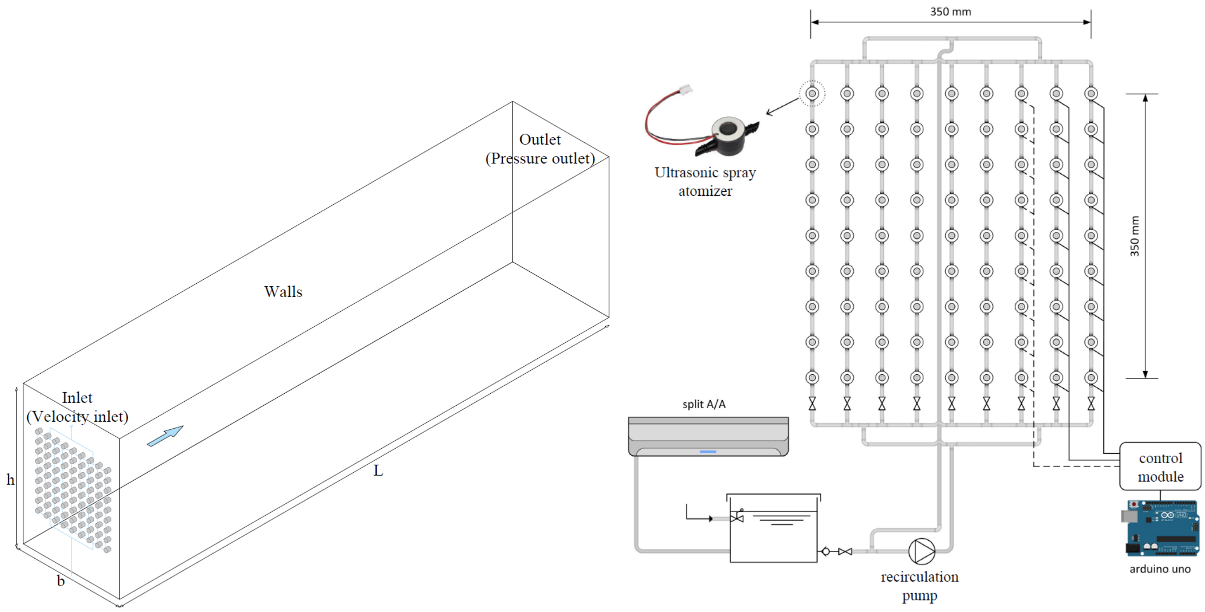


Figure 3. (a) Wind tunnel domain considered in the simulations and boundary conditions. (b) Scheme of the distribution of the spray atomizers in the form of a 9 x 9 manifold.

3.3. Mathematical method (governing equations)

3.3.1. Continuous phase (moist air)

The airflow was assumed to be steady, incompressible and turbulent flow. The Reynolds-Averaged Navier-Stokes equations (RANS) were used along the standard k-ε turbulence model. No turbulence model can be considered superior than the others [8], but the standard k-ε turbulence model has been reported to accurately predict the flow field and droplet evaporation in several spray cooling applications [9]. The influence of water droplets on the airflow was considered by introducing the source terms of mass, momentum and energy into the air-side governing equations.

In spray cooling applications, heat, mass and momentum is transferred between the moist, unsaturated air stream and the water droplets. Processes related to heat-and-mass transfer between different phases are governed by mass, momentum, energy and species conservation principles. This multiphase flow is described by the set of equations that governs the continuous phase (internal moist air inside the wind tunnel) and the set of equations for the discrete phase (ultrasound-generated water droplets). The continuous and discrete phase equations are coupled by the source terms of the conservation equations. In this study, an Eulerian–Lagrangian approach was used.

$$\frac{\partial(\rho v_i)}{\partial x_j} = S_m \tag{1}$$

$$\frac{\partial(\rho v_i v_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial \tau_{ij}}{\partial x_j} + \rho g_i + S_{mo} \tag{2}$$

$$\rho v_i \frac{\partial e}{\partial x_j} = -p\frac{\partial v_i}{\partial x_j} + \frac{\partial}{\partial x_j}\left(k\frac{\partial T}{\partial x_j}\right) + \frac{\partial}{\partial x_j}\left(\sum_{i'=1}^{n} h_{i'} J_{i'}\right) + \phi_v + S_e \tag{3}$$

$$\rho v_i \frac{\partial m_j}{\partial x_j} = -\frac{\partial J_{i',i}}{\partial x_j} + S_m \tag{4}$$

$$J_{i',i} = -\rho D_{f_{i',m}} \frac{\partial m_j}{\partial x_j} \tag{5}$$

Here, S_m, S_{mo}, S_e represent the mass, momentum and energy source terms introduced by water droplets, respectively, and $J_{i',i}$ is the diffusion flux of species i'.

3.3.2. Discrete phase

The equations for the spherical water droplets are written in a Lagrangian reference frame. The trajectory of a discrete phase particle is obtained by integrating the force balance on the droplet (Newton's second law of motion),

$$\frac{dv_d}{dt} = \frac{18\mu}{\rho_d d_d^2}\frac{C_D \mathrm{Re}}{24}(v - v_d) + g\frac{\rho_d - \rho}{\rho_d} \tag{6}$$

$$\frac{dr_d}{dt} = v_d \tag{7}$$

The drag coefficient (C_D) included in the drag per unit droplet mass term can be calculated as reported by Morsi and Alexander [10].

Due to the existence of a temperature difference and a vapour concentration gradient between the water droplets and the unsaturated air, heat and mass transfer between them occur. The energy conservation equation of a droplet, neglecting the radiation effect, is expressed as,

$$m_d c_p \frac{dT_d}{dt} = h_C A_d (T - T_d) + \frac{dm_d}{dt} h_{fg} \tag{8}$$

This equation relates the temperature change in the droplet to the convective and latent heat transfer between the droplet and the continuous phase. The rate of evaporation of water changes according to,

$$\frac{dm_d}{dt} = h_D A_d M (y_s - y) = h_D A_d (\rho_s - \rho) \tag{9}$$

The convective and mass transfer coefficients (h_C, h_D) are calculated from the Nusselt and Sherwood number correlations reported in [11],

$$Nu = \frac{h_C d_d}{k} = 2 + 0.6\,\mathrm{Re}_d^{1/2}\,\mathrm{Pr}^{1/2} \tag{10}$$

$$Sh = \frac{h_D d_d}{D_f} = 2 + 0.6\,\mathrm{Re}_d^{1/2}\,\mathrm{Sc}^{1/2} \tag{11}$$

3.4. Numerical method

The boundary conditions used in the present analysis are shown in Figure 3 (a). Air enters the computational domain at the leftmost surface shown in the figure. It is in this section that the water injections are located. These have been individually inserted into each of the spray atomizer locations. In Figure 2 (a) it can be seen as the flow that there is a lot of turbulence in the dispersion of the particles, this has been introduced in the RANS model using a stochastic tracking model referred as Discrete Random Walk in ANSYS FLUENT. The turbulent dispersion of droplets is predicted by integrating the trajectory equations for individual particles, using the instantaneous fluid velocity along the particle path during the integration. By computing the trajectory for a sufficient number of representative particles (tries), the random effects of turbulence on the particle dispersion may be simulated. Therefore, the number of particles that we introduce will be multiplied by 5.

The inlet air speed conditions have been considered uniform and an air speed of 1.5 m s^{-1} has been set. A turbulence intensity of 5% was assumed for the inflows [12]. The incoming spray water was adjusted to match what was measured. Therefore, to establish the injection speed of the continuous phase, experimental measurements were made, as shown in Figure 2 (a). It is around 2.5 m s^{-1}. On the other hand, the droplet temperature was set equal to the air temperature used. The pressure was stated to be equal to the ambient pressure at the outlet section (pressure outlet). Wall boundary condition was selected for the tunnel walls. The zero heat flux and no-slip condition were imposed on them. The 'escape' boundary condition was set, meaning that the droplets exit the computational domain when they reach the walls or the tunnel exit [9]. The droplet size distribution used was 10 μm. This correspond to the size of the drops generated by the mist generator used in previous works [7].

The commercial code ANSYS FLUENT (version 21 R1) [13] was utilised to numerically solve the governing equations. The selected grid is an unstructured grid with 770.317 polyhedral cells. This will be the same for all analysis. The SIMPLE algorithm was employed to solve the coupling between continuity and momentum equations through pressure. All calculations were performed using discretization providing second order accuracy. The convergence criterion in each case was $|\varphi(i+1)-\varphi(i)|/\varphi(i)<10^{-4}$, where i denotes the iteration number and φ can stand for any of the dependent variables.

4. Results and discussion

4.1. Parametric analysis

A comparative study was carried out between the mist generator and the ultrasonic spray atomizer system. The objective is to analyze the difference between both systems and to know if this novel system manages to solve the limitations of the mist generator reported in [7]. Therefore, the conditions used, and the configuration described in section 2 and in the mentioned work are the same. More information about the model validation process can be found here [7].

The conditions used in the simulations are summarized in Table 1. The key variable in the simulations is the mass flow rate of water. As mentioned in section 2, the mass flow rate of water of each atomizer is fixed, therefore, to modify it the only way is to modify the number of atomizers. In this work, different distributions of the atomizers along the inlet section have been studied. The total number of simulations carried out was 6. The rest of the variables involved in the parametric analysis were considered constant.

Table 1. Conditions and distributions used in the simulations.

Tests	$\dot{m}_a$ (kg s^{-1})	T∞ (ºC)	$\phi\infty$ (-)	Distribution (Injections)	h x b (m)	$\dot{m}_w$ (x10^4 kg s^{-1})
1	0.619	25	0.5	2 x 3 (6)	0.05 x 0.10	0.500
2	0.619	25	0.5	3 x 4 (12)	0.10 x 0.15	1.000
3	0.619	25	0.5	6 x 6 (26)	0.22 x 0.22	3.000
4	0.619	25	0.5	7 x 7 (49)	0.26 x 0.26	4.082
5	0.619	25	0.5	9 x9 (81)	0.35 x0.35	6.750
6	0.619	25	0.5	12 x12 (144)	0.44 x 0.44	12.000

As an example, Figure 4 shows the temperature contours (a, c) and evolution of the particles (b, d) along the domain for similar cases using the different systems: (a, b) spray atomizers and (c, d) mist generator. As can be seen, with the spray atomizers the injections have been evenly distributed in the inlet section. This ensures that the evolution of the temperature contours is quite uniform throughout the entire domain, once all the drops have evaporated, unlike what happened with the mist generator. In the same way that happens with the evaporation of the drops, since, as they are much more distributed, they evaporate much faster.

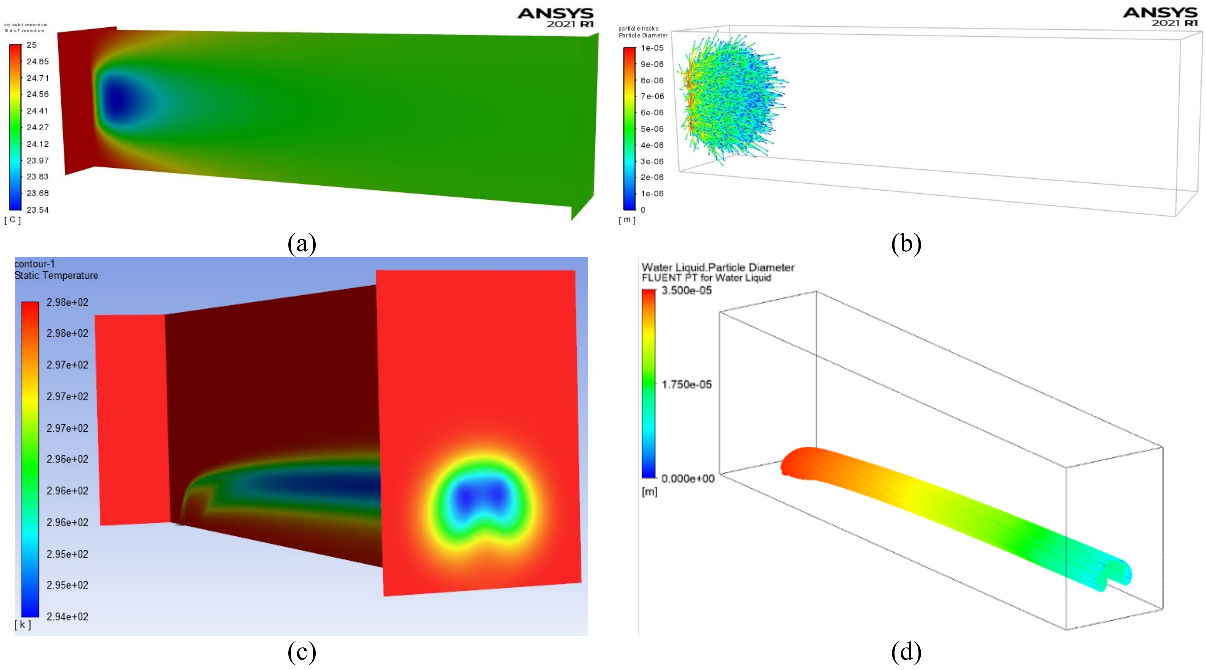


Figure 4. (a, c) Temperature contours in the axial mid plane and the outlet section of the domain and (b, d) evolution of the diameter of the drops from when they leave the spray atomizers until they completely evaporate (a, b) for the spray atomizer with 36 injections and (c, d) for the mist generator with $v_{ai} = 3.0$ m s^{-1} y $m_w = 0.0001$ kg s^{-1}.

Once these data have been presented in a representative way, the results obtained will be analyzed and compared with similar cases studied with the mist generator. At this point it is important to mention that the average temperature in the different planes where measurements have been made hardly varies depending on the length elapsed from the point of injection. Therefore, from now on, only the average temperature in the exit section of the tunnel, $\overline{T^L}$, will be considered.

The first parameter to be analyzed is the evaporative cooling efficiency at the certain section, $\overline{\eta^x}$. It is defined as the ratio of the mean temperature difference calculated at the evaluated section, $T_\infty - \overline{T^x}$, to the wet bulb depression $(T_\infty - T_{wb_\infty})$.

$$\overline{\eta^x} = \frac{T_\infty - \overline{T^x}}{T_\infty - T_{wb_\infty}} \tag{12}$$

Figure 5 shows the evaporative cooling efficiency evaluated in the outlet section of the domain, $\overline{\eta^L}$, as a function of the mass flow rate of water for all the simulations performed and cases of similar conditions studied with a mist generator, where v_{ai} refers to average air flow velocity in the mist discharge section.

As can be seen, the results are scattered over a wide range from, $\overline{\eta^L} = 0.015$ to 0.416. The maximum efficiency of, $\overline{\eta^L} = 0.416$ is given for the case of 144 injections. This is logical since if the rest of the conditions are the same and the $\dot{m}_w$ is increased, a greater thermal jump will be obtained and therefore a greater evaporative efficiency.

On the other hand, if the results obtained with the spray atomizer and mist generator are compared, it is observed that the results are very similar for both equipment. This is justified because the same conditions have been used for both cases.

The evaporative cooling efficiency gives an overall representation of the cooling processes taken place in the domain since is referred to the mean temperature. However, this magnitude fails may not be able to represent local effects. For example, there could be areas where the air cools below the average

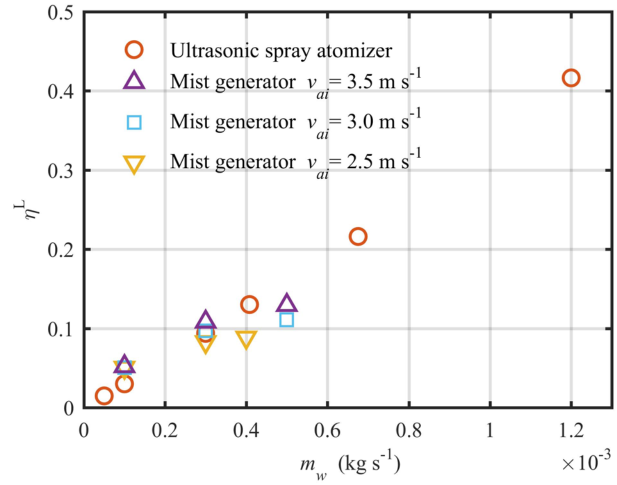


Figure 5. Evaporative cooling efficiency as a function of the mass flow rate of water for all simulations conducted.

temperature set by the evaporative cooling efficiency and others where it is above. As was the case with the mist generator.

To perform a quantitative comparison between the studied cases, the difference between the maximum, mean and minimum temperatures for the 6 tests was studied. With this, it was observed that the temperatures are very close to each other (maximum difference is 0.81%). Therefore, the average temperature is a representative value for all outlet section. This can also be seen in Figure 4 (a).

The last indicator discussed in this section is the wet length, L_w. The wet length is the distance from the droplet injection section until the water droplets completely evaporate. This phenomenon can cause corrosion, scaling and fouling on the heat exchanger bundles if not fully evaporated water droplets are carried out by the airstream to the heat exchanger bundles of the condenser.

The wet length (in its dimensionless form, L_w/L) is shown in Figure 6. As can be seen, in none of the simulated cases is it greater than unity, which means that no drop leaves the domain through the outlet section. In addition, it is observed that the wet length depends on the mass flow rate of water. This is logical, because if the quantity of water is greater, the drops will take longer to evaporate. As an example, Figure 4 (b) shows the prediction of fall trajectories in a representative case.

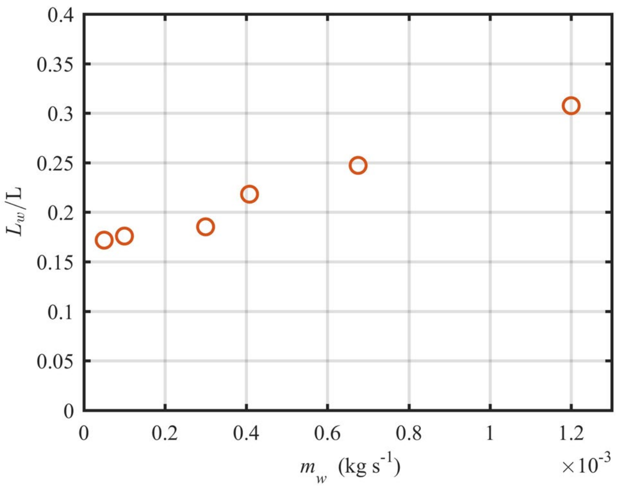


Figure 6. Wet length (in its dimensionless form, L_w/L) as a function of the mass flow rate of water for all simulations performed with ultrasonic spray atomizers.

This allows identifying the minimum length at which to locate the spray atomizers of condenser. For the cases studied, the minimum length is 0.769 m. However, if the installation is expected to work with other conditions than those used (temperature, humidity...), it should study them before taking them to the prototyping phase. This ensures that the installation always works in ranges where complete evaporation occurs.

These results are key since this was the main problem with the mist generator. Since all the cases studied exceeded the unit (there were drops that escaped the domain). What supposed a problem to be able to install the system a real installation.

4.2. Optimization analysis

From the above discussion, it has been highlighted that an increase of $\dot{m}_w$, increases the cooling efficiency, at least, until the saturation is reached. However, this will imply having to introduce a greater number of spray atomizers and a higher energy consumption (2W/spray atomizer). Under these conditions, the cooling capacity, $\dot{Q}_{cooling}$, defined as in Equation (13), is also increased due to the temperature difference (via the cooling efficiency).

$$\dot{Q}_{\text{cooling}} = \dot{m}_a c_{p_a}\left(T_\infty - \overline{\overline{T^L}}\right) \tag{13}$$

Considering these, the relative contribution of all these effects can be seen in the coefficient of evaporative performance (COP), Equation (14):

$$COP = \frac{\dot{Q}_{\text{cooling}}}{\dot{W}_{\text{ultrasound}}} \tag{14}$$

Figure 7 presents the variation of the performance coefficient against the air inlet temperature for all the simulations carried out. As can be seen, despite the increase in consumption, the increase in injections always improves the COP, although not very significantly. This is also a differentiating factor with the mist generator. Since, as can be seen, in this case, increasing the mass flow rate of water meant an increase in the energy consumption of the equipment that had a negative impact on its overall performance.

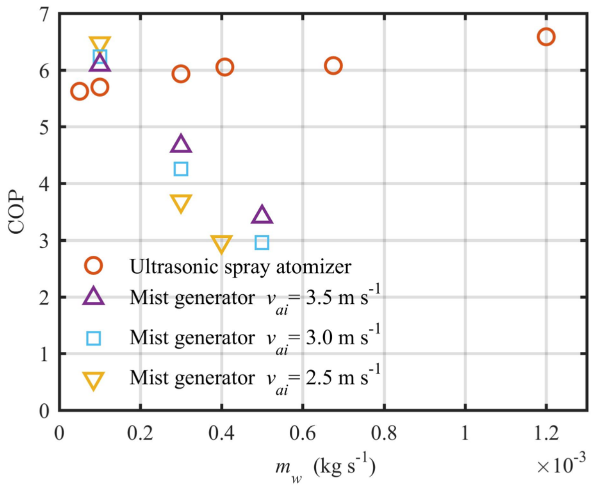


Figure 7. Performance coefficient as a function of the mass flow rate of water for all simulations performed.

In conclusion, it is evident that if more atomizers are added, the performance will increase. However, there is a point where increasing the number of atomizers will lead to an increase in energy consumption that is not compensated by better performance, as with the mist generator. But the range of mass flow rate of water allowed by the spray atomizer system is much greater.

5. Conclusions

In this article, a numerical model of an ultrasonic spray atomizer system for evaporative pre-cooling of condenser inlet air in air conditioning applications has been developed. An analysis was performed that included the number and distribution of injections as variables. Finally, an optimization study regarding the overall cooling performance, for the above mentioned application, was performed and compared to a mist generator. The main conclusions obtained during this investigation can be summarised as follows.

-The average evaporative cooling efficiency evaluated in the output section of the domain, $\overline{\eta^L}$, increases when the number of injections. The maximum value is $\overline{\eta^L}$ = 0.416 is given for the case of 144 injections. The results obtained with the spray atomizers are similar to those of the mist generator.

-The observations regarding the wet length have allowed us to know that the distribution is quite homogeneous. As well as the minimum length at which to locate the spray atomizers of condenser. The most unfavourable moisture length is L_w/L= 0.308 and is given for the case of 144 injections. These results are very promising since they solve the main limitation that existed with the mist generator.

-The optimization analysis based on the coefficient of performance has shown that the operating ranges that increase the number of injections improves performance, despite the fact that this implies greater energy consumption. The best performance obtained was 6.587. However, for these conditions, there is a greater wet length and this, as already mentioned, can be a problem, so both factors must be considered when designing the prototype.

- Increasing the number of atomizers improves the general performance, but up to a certain point, since there is an optimal point from which increasing the number of atomizers used will mean an increase in energy greater than the improvement in performance.

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