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VIABILITY STUDY OF RETRIEVING THE EVAPORATED WATER IN A MECHANICAL DRAFT CROSS FLOW COOLING TOWER

Rafael Zimmermann
Federal University of Santa Catarina
Florianopolis, SC, Brazil

Marcia B. H. Mantelli
Federal University of Santa Catarina
Florianopolis, SC, Brazil

Thomaz P. F. Borges
Federal University of Santa Catarina
Florianopolis, SC, Brazil

Camilo A. S. Costa
Federal University of Santa Catarina
Florianopolis, SC, Brazil

ABSTRACT

Cooling towers use evaporative heat transfer to reduce process water temperature. In mostly cases, the amount of water lost to evaporation is irrelevant, but in the present case of study, the amount of makeup water due to evaporation, is around 700 m^3/h. That alone is a considerable amount of fresh water that needs to be removed from the local rivers and is currently having a negative impact in the fresh water availability for consumption. This study explores viable solutions to recover some of the evaporated water, but not compromising the effectiveness of the cooling tower.

The plant is located in the southeast region of Brazil where temperature and humidity vary considerably along the year. The present study focuses on retrieving the evaporated water through condensation and capturing fog without compromising the cooling tower performance. Data from the actual cooling tower was used in the thermodynamic analysis. Computational fluid dynamics (CFD) was used to simulate the mixing of air leaving the cooling tower with outdoor air. An experimental setup is also being developed to reproduce the condensation process.

INTRODUCTION

Cooling towers can be simply taken as equipment that delivers cold water and warm humid air from warm water and cold air using a certain amount of power. An extensive number of studies evaluating the cooling tower design and performance are available in literature. The air leaving the tower is the case of study in the present work, where the viability in retrieving water from the humid air is investigated. At first, other processes and devices with similar objectives were reviewed and are presented here.

Studies involving the use of condensed water captured from humid air are not too common in literature. A review in Atmospheric Water Vapor Processing (AWVP) was presented in [1]. AWVP are devices where potable water is removed from atmospheric air. The author proposed a division of equipment which includes: surface cooling below the dew point of the ambient air to condensate water vapor, concentrate water vapor through use of solid or liquid desiccants, or induce and control convection in a tower structure.

Temperature reduction in the surface cooling processes was obtained by thermal irradiation and refrigeration cycles. Surface cooling by thermal irradiation is a passive method to keep the surface at a temperature bellow the dew point and condensate water vapor. This is possible because of the radiative heat transfer between the surface and the sky. Its observed that surfaces with high emissivity are needed in order to condensate water usually only at night time because of solar irradiation during daylight [2]. Several factors have influence over surface cooling by thermal irradiation. Among them are the cloud occurrence, wind velocity, and outdoor air properties like dew point and wet-bulb temperature.

In [3] and [4], the authors attest that wind is necessary to bring humid air to replace the air that was already dehumidified, but wind speeds higher than 1 m/s surpass any surface cooling due to thermal irradiation, because of the increase in convective heat transfer. In mechanical draft cooling towers, the wind speed atop the tower is much higher than 1 m/s because of the high powered fans used in this kind of equipment, making the application of thermal irradiation for surface cooling not possible in these conditions.

Other studies including [5], [6], [7], [8], [9], and [10], investigated this method of obtaining water, with results indicating that a very large area is needed to condensate a considerable amount of water.

Surface cooling by refrigeration cycles use an evaporating refrigerant to reduce surface temperature bellow the dew point. These applications are common in cooling and dehumidification processes. According to [11], the process consists of air flowing through duct coils (banks of bare tubes or banks of tubes which have finned or extended surfaces), while the refrigerant flows through the duct, evaporating at very low temperatures, gaining heat from the external air flow. Figure 1 shows a schematic of the cooling coil (top) and the process represented in a psychrometric chart (bottom). A psychrometric chart allows reading of thermodynamic properties of moist air and is commonly used for graphical solutions of processes utilizing moist air.

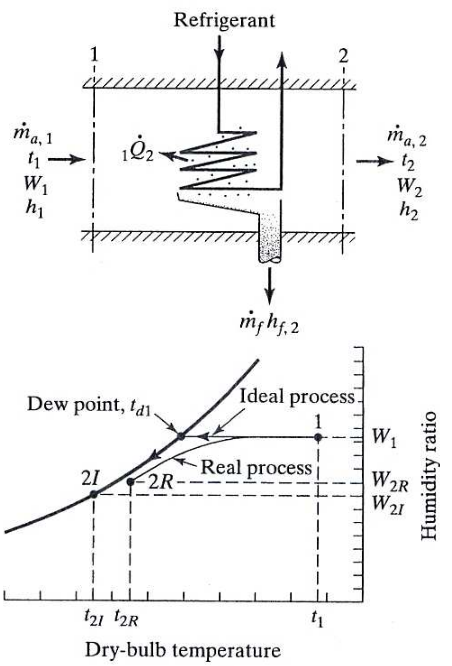

Figure 1 - Dehumidification by cooling coil [11].

Most applications of surface cooling by refrigeration are not designed to obtain water, but cold dry air. The condensed water is considered a leftover from the process and is generally discarded [12]. In [13], fresh water was obtained using evaporators in a refrigeration cycle as cold surfaces. The water production reported for a hot and humid climate was between 14.4 kg/m².h and 16.97 kg/m².h during a year period.

The use of desiccants to retain water is also very common in literature. These are materials that have the capacity of adsorbing moisture and can be used in dehumidification. According to [1], desiccants are one type of sorbent which are particularly useful for attracting water molecules, but also retain pollutants and contaminants, and are not attractive processes for reuse of water but for dehumidification only. This process is also known as chemical dehumidification and the water retained by the desiccant need to be forced out prior to reutilization of the desiccant.

Other method for collecting water from air reported in [1] is convection in a tower structure, which expands the air in a very long vertical tower reducing its temperature bellow dew point. No prototypes are known other than the mine shaft analogue in [14], and [15]. Costs and viability of such equipment would be restrictive because of the large structure required, in the range of hundreds to thousands meters high.

Desalination process is used to obtain fresh water from seawater. One desalination method uses a humidification-dehumidification (HD) cycle to obtain freshwater. As described in [16], the HD cycle consists in air absorbing water and heat from a seawater flow and then going through surfaces that are cooled by the same seawater flow, resulting in condensation of a certain amount of freshwater. A review of studies in desalination by HD cycles is presented in [17]. The study points out that in single pass cycles, the amount of freshwater obtained is between 5 to 20 % of the circulating seawater. Also, the HD cycle is applicable in small and large scale.

In the desalination process presented in [18], water is obtained by mechanical compression of humid air. The compression reduces the air capacity of retaining humidity and condensed water is formed as a result. The process occurs inside an evaporation chamber with controlled pressure, and in a very small scale.

Another desalination method is the seawater greenhouse process (SWGH), described in [19] and [20]. Solar irradiation is used to heat the air that absorbs water. The air then flows through surfaces that are cooled by seawater as in a regular HD cycle. This method is used to obtain freshwater for irrigation of cultures in very dry regions. In [21], the authors also investigated irrigation by obtaining water from atmospheric air by an underground system of ducts.

DATA ANALYSIS

The literature review shows known process obtaining water from the moist air. In order to investigate the viability of a dehumidifying process in the cooling tower, it is necessary to know the properties of the air being exhausted by the tower to the atmosphere. To obtain this data, a thermodynamic analysis was made with regular operational parameters. The equipment is a mechanical draft cross flow cooling tower, disposed in two buildings with 8 cells, totaling 16 independent cells. Each cell has an axial fan and a diffuser at the top. The inlet water is pumped to the top of the tower where it flows through the filling. All cells are assumed to be identical with the same volume of water flowing through it, and the total volume of water is collected at the collection basin.

Data from the actual cooling tower was used in the thermodynamic analysis, these including flow rates of circulating and makeup water, inlet and outlet temperatures, and meteorological data. The data is given in an hourly basis and a timeframe of one year was selected to include seasonal variations of weather and tower capacity. The circulating water in the current system amounts to an average of 45,000 m^3/h. The volume of make-up water needed is more than 800 m^3/h, where 100 m^3/h are purged water to reduce concentration of pollutants, and about 0.1 % of the total circulating water is caused by drift losses. About 700 m^3/h are lost to evaporation in the process of cooling the water, and that is a significant volume of water taken from local rivers, from which water is also taken for consumption. Figure 2 shows water flow data for the year long period selected, the total circulating water, the purge water, and the total make-up water.

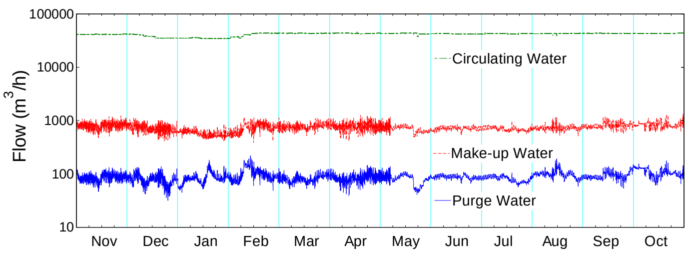

Figure 2 – Data for the water flow in the cooling tower.

Figure 3 shows the water temperature entering and leaving the cooling tower for the same period.

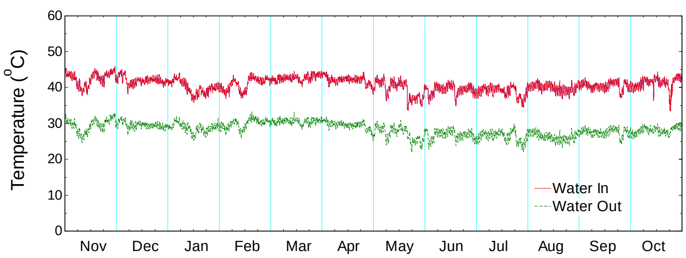

Figure 3 – Water temperatures in the cooling tower.

In Fig. 4, local meteorological data for the region where the plant is located is observed for the same year long period. Among other properties not plotted here are solar irradiation, wind speeds, relative humidity, atmospheric pressure, and precipitation. The ratio between water and air flow was informed in a specific condition, so the total air flow could be determined and was assumed to be constant.

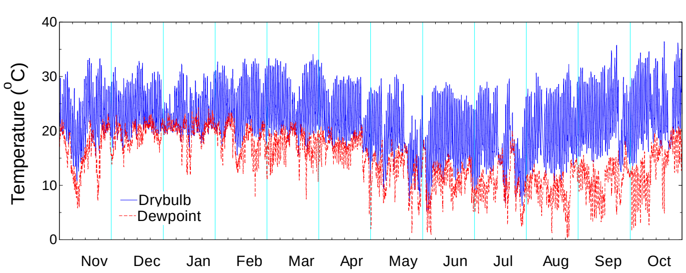

Figure 4 – Local dry-bulb temperature and dew point data.

An energy balance was used to determine the psychrometric properties of the air leaving the cooling tower, according to what is presented in [11], [22], and [23]. The present study focuses on retrieving the evaporated water through condensation and capturing fog without compromising the cooling tower performance.

To solve the energy balance, an algorithm was written using the *Engineering Equation Solver* (EES) software. This software includes in its database an air property library. In the program, the input values are: inlet water temperature, air dry-bulb and wet-bulb temperatures, air and water flows, and the tower characteristic parameter: $\frac{h_c A_v V}{Cp_{air}}$. This parameter is a dynamic function of air flow patterns and water fall along the tower. This can be taken as a constant if air and water flows are also constant. The h_c is the heat transfer coefficient between water and air, A_v is the superficial area of water droplets per volume, and V is the tower volume [11].

The condition of the air and water leaving the tower is obtained by the calculations. The later is then compared with the measured water temperature leaving the tower to determine the tower characteristic.

The tower was divided in various elements for calculations, as shown in Fig. 5, where energy and mass balances were performed for each one. The properties of air and water after each element were calculated with equations (1) to (3), assuming the same amount of air and water flows though each of the divisions. The partial result of an element was then used as input data for the next one, until all properties of the leaving air and water were obtained. Because elements were assumed to have identical air and water flows, the temperature of water leaving the tower is the average temperature of elements at the bottom of Fig. 5. The same is true for the elements at the right of Fig. 5, where the air leaves the cooling tower.

The number of divisions needs to be big enough so large numerical errors are not introduced and small enough to avoid extended periods of computing time. A twenty by twenty net of elements was considered enough because the results did not vary significantly compared to a larger number of divisions.

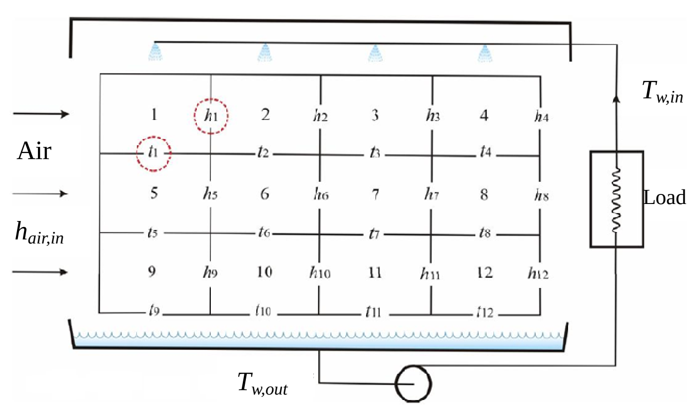

Figure 5 – Cross flow cooling tower segregated in various elements.

The enthalpy of the air leaving the element was obtained by:

$$h_{air,out} = \frac{\dot{m}_{air} \times h_{air,in} + \frac{h_C A_V V}{Cp_{air} \times 2} \times (h_{sat,w,in} + h_{sat,w,out} - h_{air,in})}{\frac{h_C A_V V}{Cp_{air} \times 2} + \dot{m}_{air}} + (W_{air,out} - W_{air,in}) \times h_{l,w,out} \quad (1)$$

The temperature of water leaving the element is given by:

$$T_{w,out} = T_{w,in} - \frac{\dot{m}_{air} \times (h_{air,out} - h_{air,in})}{\dot{m}_w \times Cp_w} \quad (2)$$

The dry-bulb temperature of the air leaving the tower is determined through the equation:

$$T_{air,out} = \frac{T_{air,in} - \frac{h_C A_V V}{Cp_{air} \times 2} \times (T_{air,in} - T_{w,out} - T_{w,in})}{1 + \frac{h_C A_V V}{Cp_{air} \times 2}} \quad (3)$$

This procedure was necessary to predict the cooling tower performance in any climate condition, and also to obtain the properties of the air leaving the tower. The warm humid air leaving the tower will be reproduced in laboratory to evaluate condensation and capture of water in different surfaces.

The ESS program was then executed in order to obtain satisfactory results. As mentioned before, the temperature of the water leaving the cooling tower was used as parameter, comparing the real data with the results from the program. The mean absolute difference between calculated and measured values was 0.7 ºC for all 8,760 hourly sets of data, which comprehends a period of one year. The tower characteristic parameter $\frac{h_C A_V V}{Cp_{air}}$ that was used to obtain this result was equal to 1,019 kW/(kJ/kg).

Figures 6 and 7 show the conditions of air entering and leaving the tower, respectively. These data are plotted in psychrometric charts, showing the humidity ratio against the dry-bulb temperature (horizontal axis). The humidity ratio is the mass of water vapor per kilogram of dry air. The lines crossing the graph represent the relative humidity, varying from zero (dry air) to one (saturated air).

Figure 6 shows all the 8,760 data points that correspond to outdoor air from November of 2006 to October of 2007. It can be observed that this period includes a variety of climate conditions, with temperatures ranging from 5 ºC to 37 ºC and relative humidity starting from below 0.2, up to the saturated condition.

Figure 7 shows all the 8,760 data points that correspond to air leaving the top of the tower from the same period. It can be observed that most of the data points are above the 0.8 relative humidity line. This observation of the state of the air close to saturation is relevant, because that is an appropriate condition for and air flux to go through a dehumidification process.

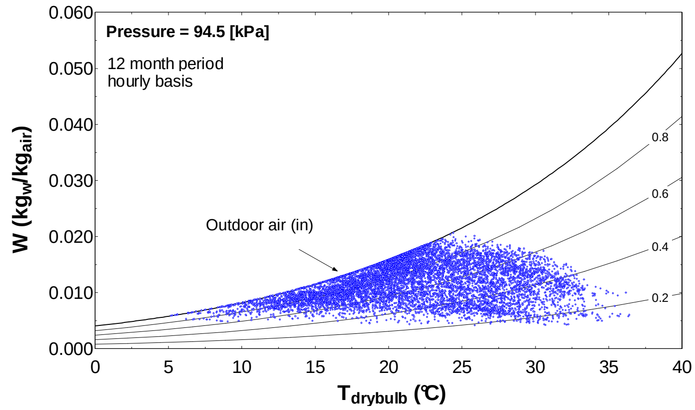

Figure 6 – State of the air entering the cooling tower.

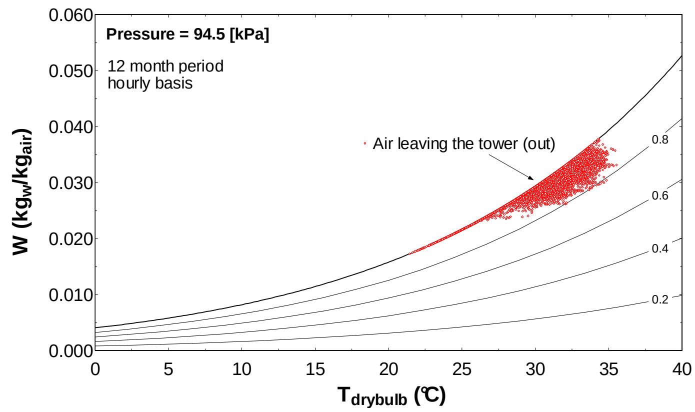

Figure 7 – State of the air leaving the cooling tower.

Observing Figs. 6 and 7, one can notice that the temperatures of air leaving the tower are higher than the local temperatures. This is not necessarily true in cooling towers because the air can gain latent heat and loose sensible heat. This observation assures the possibility of using the local environment as a heat sink in dehumidification process, which will be investigated through future laboratory experiments and simulations.

Figure 8 shows a pair of data corresponding to the inlet and outlet air conditions at a randomly selected time. The straight line does not represent the intermediate states of the air and just connects the inlet with the outlet conditions. There is sensible heat gain and latent heat gain in the air flow through the cooling tower, the later being more noticeable, as expected, because the cooling tower principle is based in latent heat transfer. In a psychrometric chart, latent heat gain corresponds to vertical increase and sensible heat gain to horizontal increase.

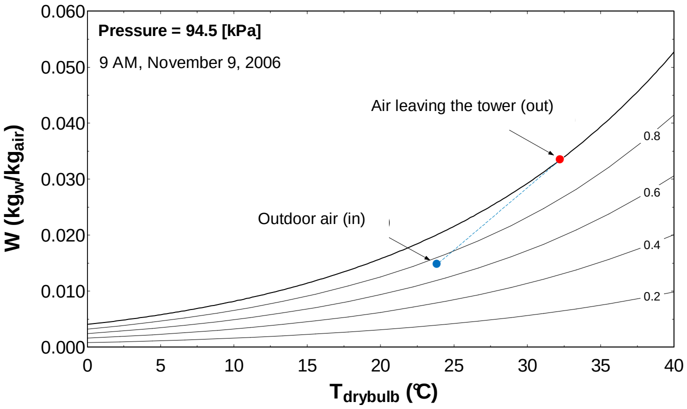

Figure 8 – State of the air entering and leaving the cooling tower for 9 A.M., November 9, 2006.

CFD ANALYSIS

Computational fluid dynamics (CFD) is being used to simulate the mixing of air leaving the cooling tower with outdoor air. Simulations were implemented in ANSYS CFX software. Results try to predict the mixing ratio and formation of fog at certain heights.

At first, a small scale model was simulated in order to determine the working principles of an experimental version of the cooling tower. The simulation results include pressure drop along the experiment and spatial distribution of air properties at the flow outlet, similar to a cooling tower.

Figure 9 shows the domain and boundary conditions used in the small scale model. The simulation domain was modeled as ideal mixture that includes dry air and water vapor, and the mass diffusivity was assumed as 2.092×10^{-5} m^2/s. The turbulence model adopted was the shear stress transport (SST) because of accentuate direction changes in the flow. A prescribed mass flux was used at the inlet with outdoor air conditions, and at a certain point water vapor was added to the air flow to represent the humidification process that occurs in a cooling tower. The simulation was performed in steady state conditions, and the stop criteria adopted was the mean average of iteration differential residual, which should be less than 10^{-4}, with an error smaller than one percent in the energy balance. The thrust and gravitational forces were neglected because of the small density difference between the outdoor air and the humid air flowing at the exit of the duct.

A prescribed air flow of 1200 kg/s with relative humidity of 70 % and temperature of 21.7 ºC was used at the inlet. The outdoor air was modeled at the same conditions with atmospheric pressure equal to 101.35 kPa.

For the sub-domain region shown in Fig. 9, a water vapor source was inserted at a rate of 0.05 $kg/m^3.s$, according conservation of energy, mass, and momentum equations, for a counter flow cooling tower [25]. Thus the air leaving the experiment would have a relative humidity close to 100 %. These equations where programmed in the ANSYS CFX software and adiabatic walls in all ducts were used as boundary conditions.

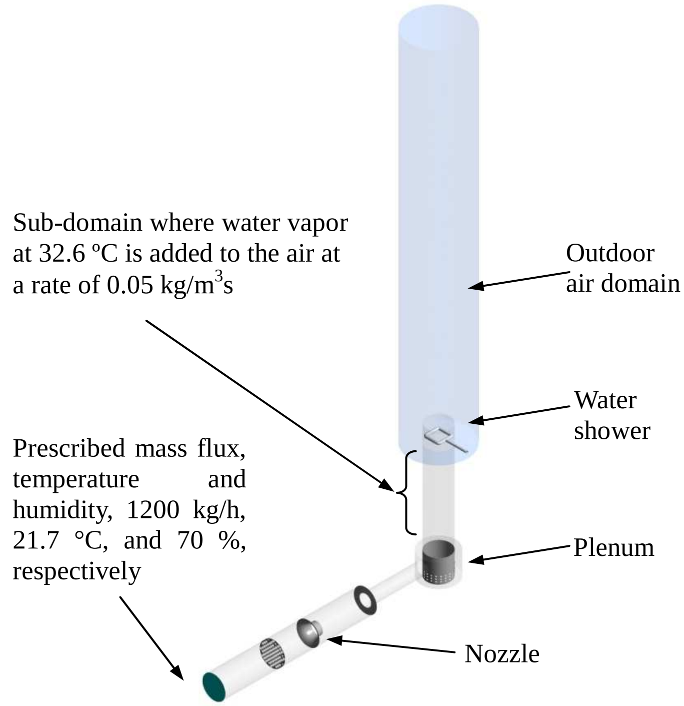

Figure 9 – Simulation domain and boundary conditions for the small scale model.

Figure 10 shows the plane view distributions of relative humidity on the left, and the velocity distribution on the right. Observing the relative humidity distribution is noticeable that air close to 100 % humidity leaves the experiment and this value is gradually reduced. After a length of approximately 0.2 m the humid air mixes with the outdoor air reducing the relative humidity. Through the observation of velocity distribution on the right side of Fig. 12, is noticeable a very similar trend with the humidity on the left, with a cone shape at the outlet.

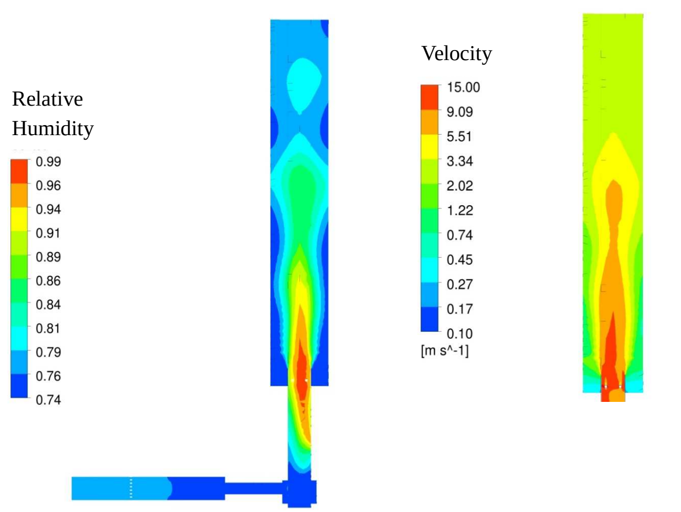

Figure 10 – Simulation results for relative humidity distribution on the left, and velocity distribution on the right.

The flow streamlines presented in Fig. 11 show uniformity in the flow leaving the experiment. This is a good observation because this flow will be directed to a surface to condensate and collect water in a future experiment.

Through simulation, the total pressure drop was reduced to approximately 2 kPa in the simulation experiment, thus reducing the power demand from the fan. This was achieved by redesigning the plenum and ducts manually, and after every change in the design, the simulation was then repeated. This proposed experimental setup was tested virtually with this simulation, avoiding future problems that would only be noticeable after its conception.

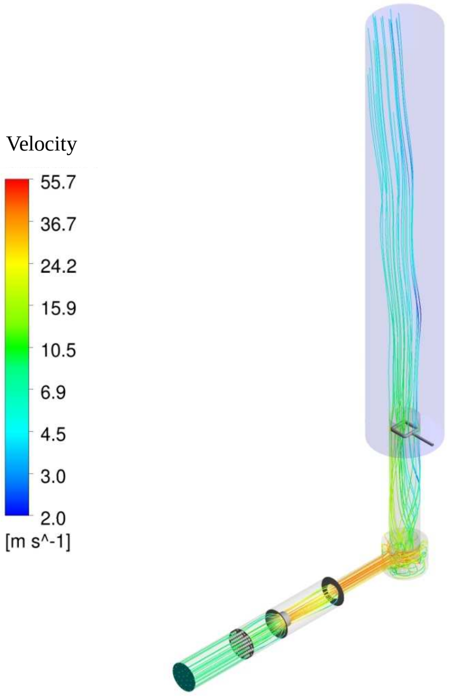

Figure 11 – Simulation results for air flow streamlines.

After successfully simulating the small scale model, a similar approach was used in generating a model for one individual cell of the cross flow cooling tower. To simulate the real cooling tower operating at the plant, a computational domain including two distinct fluids was created. The first is a binary homogeneous mixture that includes dry air and water vapor, where dry air is assumed to have ideal gas behavior. The second fluid is liquid water, which is drifted by the fan in the cooling tower. These two fluids represent the warm humid air leaving the tower.

The homogeneous mixture was modeled as a continuum fluid, and the mass diffusivity was assumed as $2.092\text{x}10^{-5}$ m^2/s between dry air and water vapor. The liquid water was modeled as small drops of diameter $1\text{x}10^{-5}$ m. These hypotheses were the same assumed in [24]. This model is commonly used for cooling towers according to [25].

All the equations and hypothesis adopted in [25] were also used in the model, these including Navier-Stokes equations, energy equation, turbulence, and centrifugal separation forces. For the Navier-Stokes equations, the surface tension between the two fluids was assumed to be equal to 0.0725 N/m^2. The turbulence model adopted was the SST in the ANSYS CFX software. The SST was chosen because of accentuate direction changes in the flow caused by the axial fan.

The simulation was in steady state conditions, and the stop criteria adopted was the same one adopted for the small scale model. The thrust and gravitational forces also were neglected because of the small density difference between the outdoor air and the air leaving the cooling tower.

Figure 12 shows the boundary conditions adopted for the simulation. The adiabatic walls assumption was used at the boundaries of the domain, and the humid air admitted prior to the fan was approximated as a circular shaped domain. That reduces computational time because only a slice of the tower needs to be simulated and the results can be replicate to build a spatial domain. The modeling of the fan blades was necessary because of the turbulent nature of the flow, which is related to the velocity field generated by the fan. If the assumption of homogenous velocity leaving the diffuser located above the fan was considered, considerable errors would be included in the results because of the kinetic energy distribution in the turbulent flow. That would change the diffusion rate from the warm humid air leaving the tower to the outdoor air. For this reason, the blade profile and real dimensions of the axial fan were modeled, thus assuring the simulation can come close to reality.

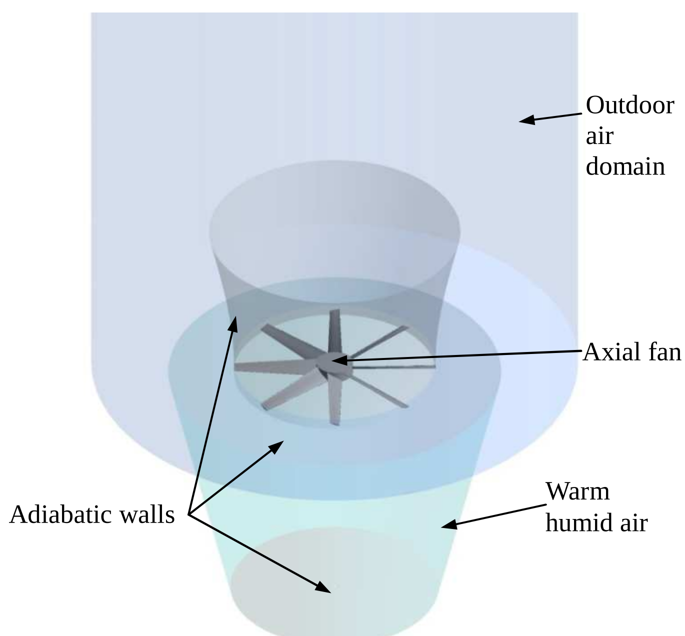

Figure 12 – Simulation domain and boundary conditions for the cooling tower.

In the present case, the air movement generated by the fan was simulated in the ANSYS CFX software through the application of a rotational domain. The grid developed for this case covered the large domain used here, and was refined close to the fan blades to better reproduce flow direction changes.

Figure 13 shows the result for the velocity field in the simulation domain of the cooling tower through streamlines. The streamlines show uniformity in the air flow and no signs of diffusion of the flow towards the laterals.

Figure 14 shows the relative humidity distribution for a selected case. The case presented in Fig. 14 is for dry outdoor conditions (60 % relative humidity and 25 ºC temperature), so the humidity in the air jet leaving the tower diffuses easily to the stationary outdoor air domain. It was observed that the saturated state of the air reached a height of 58 m at these conditions. For other conditions the height with the presence of saturated air was much higher, thus giving the possibility of intervention on the saturated air flow stream at further positions at the top of the cooling tower.

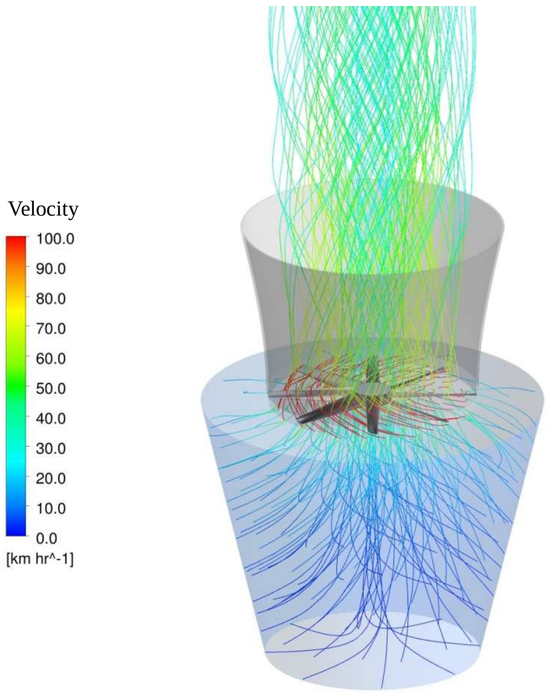

Figure 13 – Streamlines in the simulation domain.

FUTURE RESEARCH

The CFD analysis is ongoing and more expressive results should be obtained soon. A variety of flat plates in different positions are being simulated at the top of the tower with the intention of using these as condensing surfaces. Porous media are also being introduced in the simulation domain to promote condensation. The effect of such structures over the flow and pressure drop will be analized and then will indicate if those are viable solutions for retrieving the water vapor from the air flow.

The phase change phenomena is also beeing implemented in the model, thus the diffusion between the drift water dropplets and air can be simulated, as well as the condensation phenomena.

For comparison and validation of the CFD model, sensors will be placed at the top of the tower to monitor temperature and humidity. The data collected in the tower is necessary because of the large number of hypothesis and parameters that are assumed to run the ANSYS CFX software, most of then taken from theoretical studies in literature. Temperature and humidity sensors will be placed along the flow at the top of the cooling tower and positioned at the external border of the diffuser. The measured data will be compared to simulation results and used as calibration tool for the simulations. The proposed scheme for placement of the sensors is shown in Fig. 15.

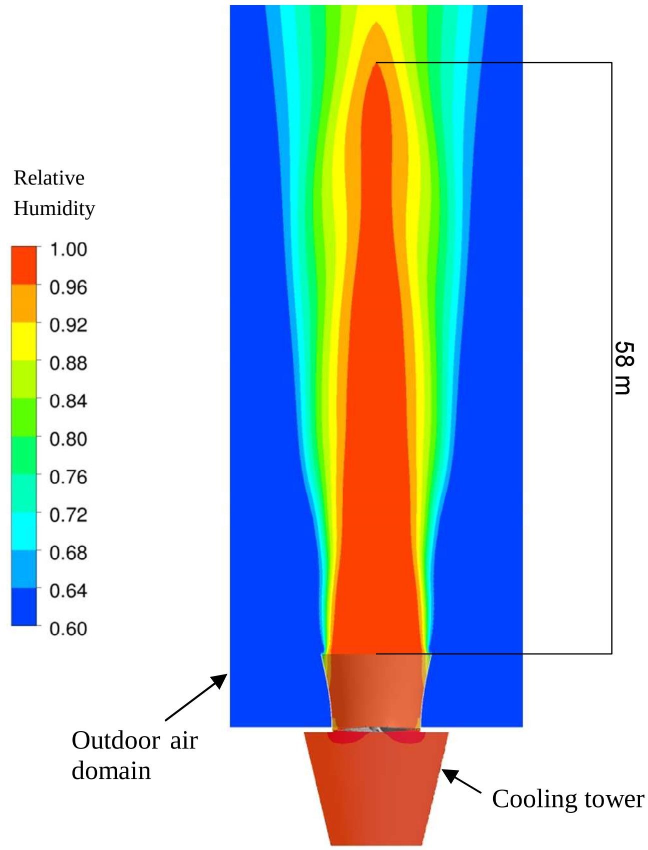

Figure 14 – Relative humidity distribution.

Besides the CFD efforts, an experimental setup is also being developed based on the small scale model presented earlier. The proposed experiment delivers saturated air at a certain temperature, where the flow is directed to surfaces that promote condensation and capture of water that is being removed from the air. The objective is to evaluate different possible shapes and materials of the condensation surfaces and to determine which one is best for condensation and capture of water. The use of heat pipes for cooling the air stream and porous media to collect moisture are being considered. Actually these subjects are some of the mainstream researches at Labtucal-UFSC (Heat Pipes Laboratory), which include heat pipes, wick structures, porous media, and thermosyphons.

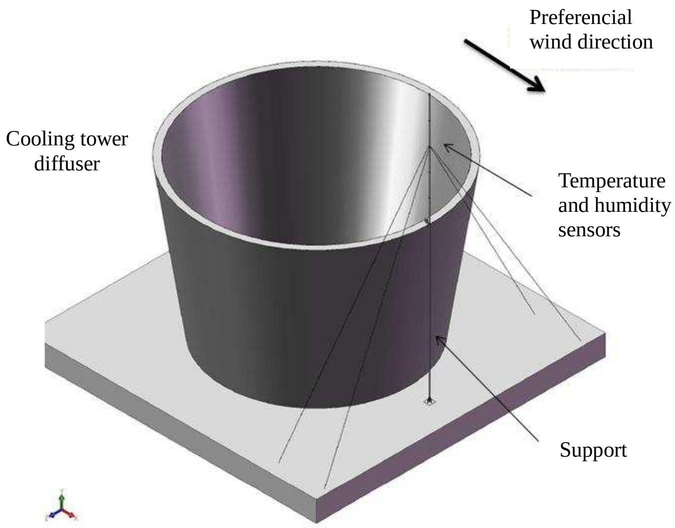

Figure 15 – Scheme of temperature and humidity sensors to be installed at the cooling tower diffuser.

The experimental apparatus is being developed at the laboratory facilities. The schematic model is show in Fig. 16. It is basically a humidifier with controlled conditions to reproduce the warm humid air leaving the cooling tower.

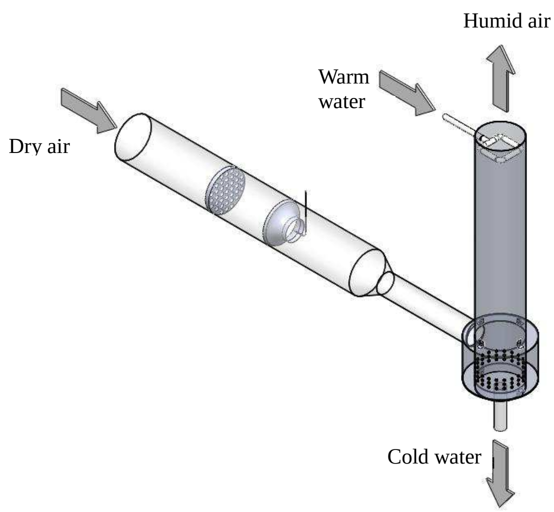

Figure 16 – Scheme of the experimental setup.

The experiment uses the same principle as a cooling tower: outdoor air will be forced by a centrifugal fan through a warm water shower, with the air flow moving upward and the water flow going downward. The heat and mass exchange between air and water will result in humidification of the air and cooling of water.

Sensors will be located at the inlet to monitor pressure, dry-bulb and wet-bulb temperatures of the air. A nozzle will be installed in the duct after a centrifugal fan leading to a plenum, where the air is insufflated. The plenum will distribute the air around the entrance right before it encounters the water flow. A thermal bath with controlled temperature and mass flux will be used to control the water flux.

The air with controlled temperature and humidity will be directed against different kinds of cold surfaces. These surfaces are supposed to be able to condensate water vapor present in the air and collect the liquid formed. The results will be used as evaluation method of viability in developing equipment to obtain water from the air flow leaving a cooling tower.

Because condensation occurs, enhancement of surfaces will be investigated as in [26], and [27]. The knowhow of the laboratory in heat pipes and porous media will be used in developing surfaces with these technologies. The presence of non-condensable gases in condensation, dry air in the present case, will be investigated too, as in [28]. Other studies involving condensation in enhanced surfaces are listed from [29] to [37].

The tests will evaluate mass and heat transfer in the dehumidification process, the condensation rate, and the potential of recovering the water. The apparatus in test should be able to condensate and collect the water for reuse.

CONCLUSIONS

Research about retrieving water from humid air, although scarce, is not new in literature. A limited number of studies where found and are presented here, although none was specifically developed for application in mechanical draft cooling towers.

Using numerical simulation and experimental investigation this work has the objective of evaluating the viability of retrieving water from the humid air in cooling towers. At first, data from an actual cooling tower is analyzed for a year long period in order to include different weather conditions for the tower operation. The data analysis was presented here, showing that the air leaving the tower is seen to be always close to saturation.

CFD is being used to simulate the moist air flow leaving the tower. This simulation will be needed to later determine where to install devices to condensate and collect water, without compromising the cooling tower performance. At first, simulation results of a small scale model were presented, including velocity fields and spatial distribution of the air properties.

Based on the simulation results, an experimental setup is being developed in laboratory to reproduce the air flow and to test surfaces for condensation and capture of water. These enhanced surfaces will be at a temperature bellow the air flow and will promote condensation of water vapor. The Labtucal-UFSC research includes heat pipes, wick structures, and porous media that will be investigated for this application.

NOMENCLATURE

Cp_w Specific heat of water vapor (kJ/kg.K)

Cp_{air} Specific heat of moist air (kJ/kg.K)

$h_{air,in}$ Enthalpy of moist air at inlet (kJ/kg)

$h_{air,out}$ Enthalpy of moist air at outlet (kJ/kg)

$\frac{h_c A_v V}{Cp_{air}}$ Tower characteristic (kW/(kJ/kg))

$h_{l,w,out}$ Enthalpy of liquid at water outlet temperature (kJ/kg)

$h_{sat,w,in}$ Enthalpy of saturated air at water inlet temperature (kJ/kg)

$h_{sat,w,out}$ Enthalpy of saturated air at water outlet temperature (kJ/kg)

$\dot{m}_{air}$ Mass flow rate of dry air (kg/s)

$\dot{m}_{w}$ Mass flow rate of water (kg/s)

$W_{air,in}$ Humidity ratio of moist air at inlet (kg_w/kg_{air})

$W_{air,out}$ Humidity ratio of moist air at outlet (kg_w/kg_{air})

$T_{air,in}$ Dry-bulb temperature of air at inlet (ºC)

$T_{air,out}$ Dry-bulb temperature of air at outlet (ºC)

$T_{w,in}$ Water temperature at inlet (ºC)

$T_{w,out}$ Water temperature at outlet (ºC)

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