Evaporative Cooling in Solar Absorption Chiller

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Introduction

Solar cooling systems based on evaporation and absorption processes have been actively investigated since 1970s, and systems based on different mechanisms and various refrigerants have been designed and developed [1]. Amongst these systems, a single-stage close cycle design utilizing lithium bromide salt and water (LiBr-H_2O) solution as the working fluid has been proved to be one of the most appropriate and low cost systems for solar based applications as it requires relatively low as well as wide ranges of operation heat source temperatures [2]. Figure 1 shows the schematic of a typical single-stage absorption chiller. In this design, water in the solution is vaporized by solar heated media and then condensed by cooling media typically cooled by a chilling tower in a closed chamber at reduced pressure. The separated water is then sprayed into another chamber at a much lower pressure causing the water to evaporate and thus releasing latent heat to cool media running through the chamber. The water vapor is subsequently condensed and then mixed with the high concentration salt solution resulted in the first step returning it back to initial concentration ready for next cycle.

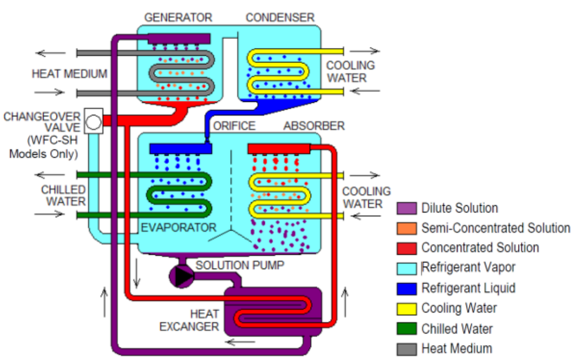

Figure 1. Schematic of a single-stage absorption chiller [3]

This study focuses on the modeling and simulation of the evaporation process when the water is sprayed into the second chamber as it directly affects the cooling of the media running through it. The objective is to understand how humidity and flow of the air-water vapor mixture affect the cooling of the media running through the tubes. The ultimate goal is to be able to optimize these parameters for higher efficiency.

Governing Equations

In the evaporative cooling process, there are three physics involved including water evaporation and transport of water vapor, the flow of the air-water vapor mixture, and heat transfer due to conduction as well as convection, and water evaporation of the water.

Turbulent Flow

For the evaporator coil modeled in this study, given air density ρ = 1.29 *kg/m*^{3}, highest modeled flow speed U = 5.0 *m/s*, a representative evaporator coil dimension L = 0.2 *m*, air viscosity μ =18.1×10^{-5} *kg/(m.s)*, the *Re* number at these conditions is approximately 7.13×10^4 indicating turbulent conditions. Therefore, the airflow is molded as turbulent flow using low *Re* number k-ε interface, in which two additional parameters are considered: the turbulent kinetic energy k and the turbulent dissipation rate ε. The turbulent viscosity is modeled as:

$$\mu_T = \rho C_\mu \frac{k^2}{\varepsilon}$$

where C_μ is a model constant. The transport equation for turbulent kinetic energy k is represented as follows:

$$\rho\frac{\partial k}{\partial t} + \rho u \cdot \nabla k = \nabla \cdot \left(\left(\mu + \frac{\mu_T}{\sigma_k}\right)\nabla k\right) + P_k - \rho\varepsilon$$

The product term in the above equation is expressed as:

$$P_k = \mu_T\left(\nabla \boldsymbol{u} : (\nabla \boldsymbol{u} + (\nabla \boldsymbol{u})^T) - \frac{2}{3}(\nabla \cdot \boldsymbol{u})^2\right) - \frac{2}{3}\rho k \nabla \cdot \boldsymbol{u}$$

The transport equation for the turbulent dissipation rate ε is represented as the following.

$$\rho\frac{\partial \varepsilon}{\partial t} + \rho \boldsymbol{u} \cdot \nabla \varepsilon = \nabla \cdot \left(\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\nabla \varepsilon\right)$$
$$+ C_{\varepsilon 1}\frac{\varepsilon}{k}P_k - C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k}$$

Constant	Value
C_μ	0.09
$C_{\varepsilon 1}$	1.44
$C_{\varepsilon 2}$	1.92
σ_k	1.0
σ_ε	1.3

Table 1. Constants in the k-ε turbulent flow model

The constants in these equations are determined from experimental results and the values are listed in the Table 1 shown above.

Heat Transfer

Convection due to the turbulent flow of the air-water vapor mixture determines the heat transfer in this domain. Thermal properties of this mixture are determined by the moist air theory [4]. The heat transfer within the evaporator coil and the thin layer of water on top of it is due to conduction only.

Water Evaporation and Vapor Transport

As the thin layer of water on the surface of the evaporator coil evaporates, latent heat is released to further cool down the water and the evaporator coil. This additional heat released to air-water vapor mixture is the main energy source of cooling in the absorption chiller.

The amount of this heat flux Q_{evap} is determined as $Q_{evap} = H_{vap} * M_{evap}$ where H_{vap} is the latent heat of water and M_{evap} is the amount of water evaporated. Considering the air-water vapor mixture as an ideal gas, the saturation concentration can be expressed as [5]:

$$c_{sat} = \frac{P_{sat}(T)}{RT}$$

where c_{sat} is the saturation concentration, r is the ideal gas constant, T is the temperature, and $P_{sat}(T)$ is the saturation pressure which is represented as [5]:

$$P_{sat} = 610.7Pa \cdot 10^{7.5\frac{T-273.15K}{T-35.85K}}$$

The amount of evaporated water can then be expressed as [5]:

$$M_{evap} = k(c_{sat} - c_V)M_V$$

where k is the evaporation rate, M_V is the molar mass of water vapor, and C_V is the vapor concentration. And, the evaporative heat flux can be represented as [5]:

$$-\boldsymbol{n} \cdot (-k\nabla T) = H_{vap}\boldsymbol{n} \cdot (-D\nabla c + \boldsymbol{u}c)$$

where $\boldsymbol{n}$ is the normal vector, k is the coefficient of thermal conduction, D is the diffusivity, c is the convective heat flux, and $\boldsymbol{u}$ is the speed vector. Since the turbulent flow also affects the diffusion process, the following turbulent diffusivity vector D_T should also be added to the diffusion vector:

$$D_T = \frac{V_T}{S_{cT}}\boldsymbol{I}$$

where V_T is the turbulent kinematic viscosity, S_{CT} is the turbulent Schmidt number and $\boldsymbol{I}$ is the unit vector.

Numerical Model

The geometry consists of two evaporator aluminum foils (each has a dimension of 20.00cm × 5.00cm × 0.25cm with 2.00cm gap in between) and two copper tubes in which the cooling media running through (each tube has a dimension of Φ0.50cm × 3.25cm and are placed symmetrically 10.00cm apart center to center and going perpendicularly through the two foils) are modeled as a representative unit of a evaporator as shown in Figure **1**. A layer of water with 0.05cm thickness covers the two foils on top and the representative unit is placed in a 40.00cm × 10.00cm × 15.00cm volume representing air as show in Figure 2. Symmetric plane is also introduced to reduce the model complexity and size.

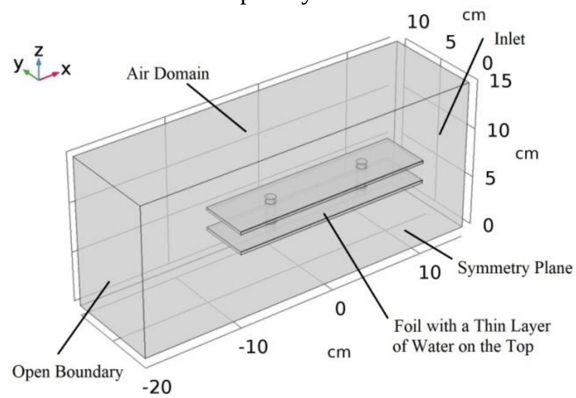

Figure 2 Model geometry with boundary conditions

The COMSOL CFD, Heat Transfer, and Transport of Diluted Species modules are used to model the air-water vapor mixture flow, heat exchanges, and evaporation of water as well as coupling of the three physics in this process. The heat transfer includes the latent heat released during evaporation of the thin layer of water on the surface of the evaporator foil, and the conduction within the water, the evaporator foils, and the tubes in which the cooling media flow. As the water continuous to evaporate, water vapor pressure increases causing it to flow into the lower pressure absorber in which the water vapor is condensed by the absorber. This pressure difference results in continuous flow of water vapor from the evaporator to the absorber, and it is modeled using the low Reynolds k-ε turbulent flow model in the CFD module. In this study, this flow is simplified by introducing an inlet with constant air flow speed. The Transport of Diluted Species interface is used in the low pressure air domain surrounding the water surface to obtain correct amount of water evaporating from it since the evaporation depends on the relative humidity of the air domain. The initial temperature of both the thin layer of water and the evaporator foil are set as 448.15[k] (75^0C).

To solve the problem, a stationary low Re number k-ε turbulent flow is modeled first to find the velocity field using the mesh shown in Figure 3 that is optimized for the wall boundary conditions. As the density of the air-water vapor mixture in the domain could vary greatly, this mixture is considered compressible. The flow velocity filed is subsequently used in a transit study where the heat transfer coupled with the transport of the water vapor using the mesh shown in Figure 4 that is optimized for near the thin layer of the water on the evaporator foil as it is where the evaporation occurs.

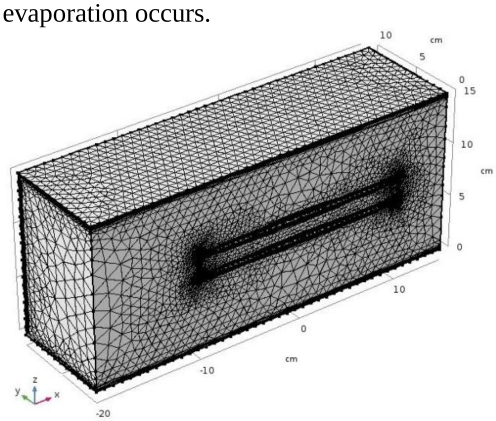

Figure 3 Mesh for low *Re* number *k*-*ε* turbulent flow

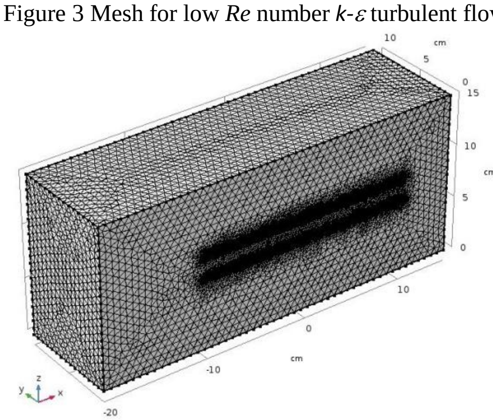

Figure 4 Mesh for heat transfer/transport of water vapor

Results and Discussion

A set of models representing varying inlet air flow speed from 0.0m/s up to 2.0m/s with an increment of 0.5 m/s and initial air humidity of 20% are modeled. Initial air humidity ranging from 0% to 40% with an increment of 10% and inlet air flow speed of 1.0m/s are also investigated. The goal is to find out effects on the cooling due to air humidity and air-water vapor mixture flow speed which in practice is determined by the evaporation and condensation of water in the evaporator and absorber of the absorption chiller respectively. Selected results are presented below.

The velocity field and temperature profile on the evaporator foil and the layer of water after 60s when subjected to a 2m/s inlet air speed and 20% initial humidity is shown in Figure 5. Relative humidity of the air domain and near the surface of the evaporator foils at 300s when subjected to 2m/s inlet air flow speed and 20% initial air humidity are also shown in Figure 5Figure 6 and Figure 7 respectively. It can be seen that the water evaporate and travel along the direction of air flow causing a higher humidity on the water-air interface and along down flow area.

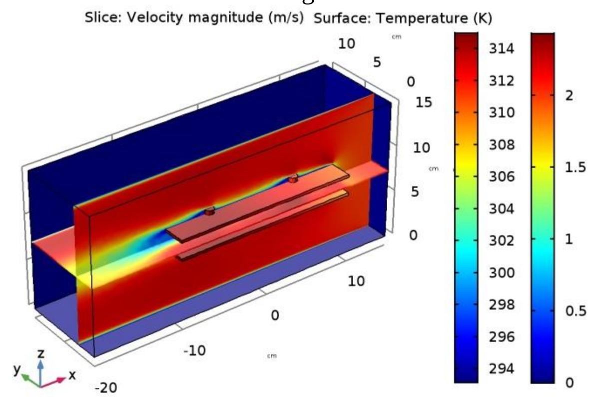

Figure 5 Air velocity and evaporator foil temperature

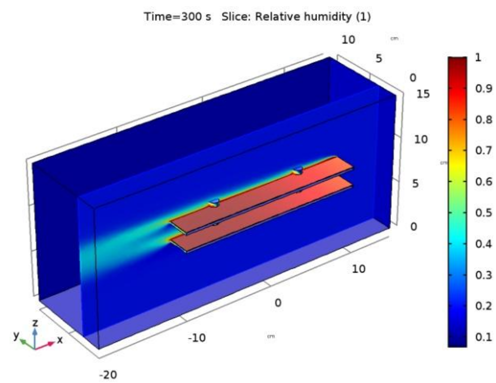

Figure 6 Relative humidity of the air domain

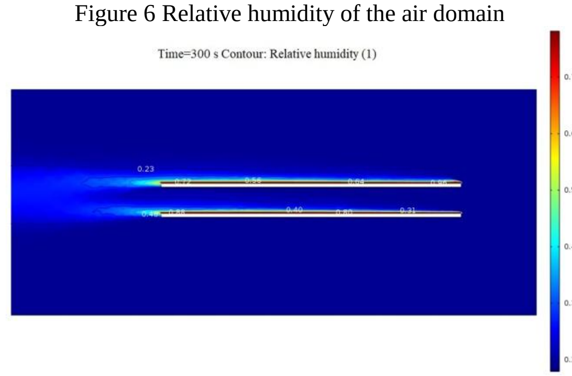

Figure 7 Relative humidity near evaporator foil at 300s

To evaluate the significance of cooling effect due to evaporation and transport of water vapor, two models subjected to the same 2m/s inlet air flow velocity with 20% initial humidity while one model deliberately ignored the heat transfer due to water evaporation and transport are created and the results are compared as shown in Figure 8. The results show that the temperature difference at the end of 900s between the two models are approximately 20^0k which is a significant amount considering that the evaporator coil cools approximately 36^0k if the evaporative cooling is not considered. The effects of inlet air flow velocity are also shown in Figure 9, and it is obvious that how much the evaporator coil will cool greatly depends on the air flow velocity. One fact to notice is that when the inlet air flow velocity equals to 0m/s, there is no obvious cooling which is likely due to the fact that there is not convective heat transfer, and the evaporation is insignificant when there is no air flow. In addition, as the air-water vapor mixture is static, the evaporation of water stops once the air humidity reaches saturation concentration.

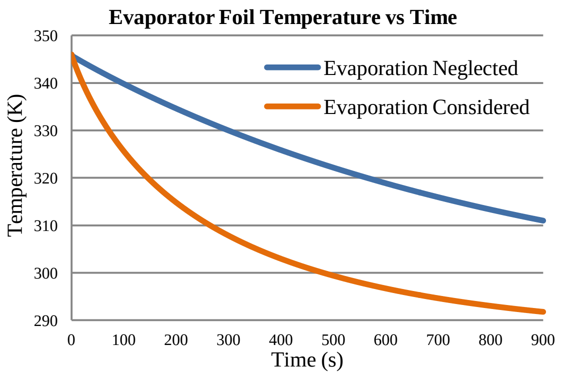

Figure 8 Comparison of cooling due to evaporation

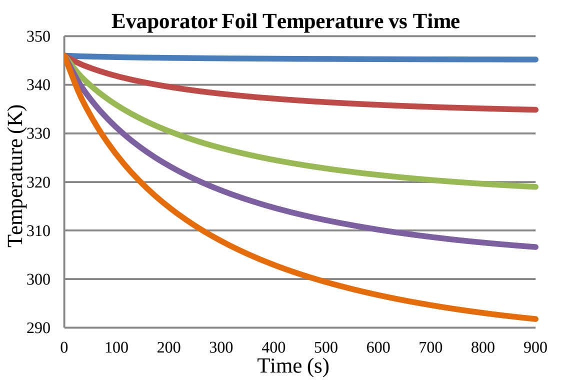

Figure 9 Effect of inlet air flow velocity

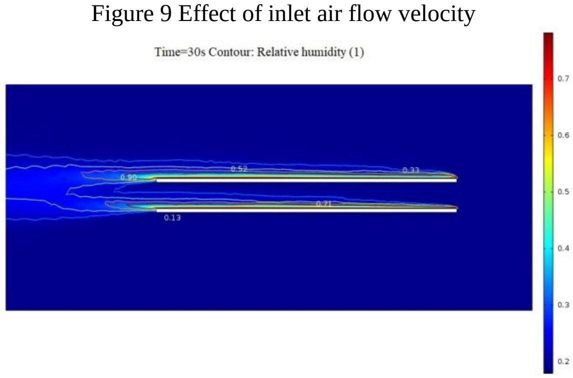

Figure 10 Relative humidity near evaporator foil at 30s

Effects of initial air humidity on cooling of the evaporator coils are also investigated and it is observed that there is no noticeable influence. This is due to the fact that the evaporation of water only occurs at the air-water interface and the air-water vapor mixture flows near the evaporator foil and then the space along the down flow direction. In addition, it is noticed that the evaporation process reaches a relatively steady status in a short time. Figure 10 shows the relative humidity distribution near the evaporator foil at 30s when subjected to 2m/s inlet air velocity and 20% initial air humidity. Comparing it with the results shown in Figure 7, it can be seen that both distribution patterns of the humidity are similar. The actual values of humidity are not identical but are also very close.

Conclusions

From the results presented in the section above with variation of the inlet air flow speed and initial air humidity, the following conclusions are drawn:

- Heat transfer due to evaporation and transfer contribute significantly to cooling of solar evaporative chiller.
- Velocity of Air-water vapor mixture directly affect the heat transfer in the evaporator. The amount of heat released to the air due to convection as well as evaporation and transport of water vapor is higher when subjected to higher air flow velocity.
- Water evaporation rate and thus associated heat transfer are also directly related to the air flow velocity. The higher the flow speed, more water evaporated and cools the evaporator foil more.
- Air initial humidity has insignificant effects on evaporative cooling and can be ignored.

As it can be seen that the air flow plays a key role in this process. As described earlier, the air-water vapor mixture flow is driven by the pressure difference between the evaporator in which water evaporates and thus raise pressure, and the absorber in which the water vapor condenses and thus lower pressure. In this study, this flow is simply modeled by a uniform air inlet, and further study is required to better understand it.

References

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