Passive Indirect Evaporative Cooler

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Abstract: Evaporative coolers are viable alternative to air conditioners because of their low power consumption. However, in tropical humid environments the increased humidity that accompanies the cool air in the evaporative coolers causes an uncomfortable wet feeling, since it hinders the evaporation of our natural transpiration. Here we test an evaporative cooler with wet and dry ducts. Then, the air from the dry ducts is conducted indoor providing a truly refreshing air, while the wet air is either discharged or conducted to ambients without people, like attics, repositories, etc. This system is so called indirect cycle. We also analyze the viability of a passive cooler, which does not use any electrical power, relying on the convection to drive the air. We do an optimization analysis to obtain the separation between the evaporating plates that maximize the heat sequestration per unit volume of the cooler.

Keywords: Evaporative cooler, indirect cycle, evaporation, convection-diffusion.

1. Introduction

Figure 1 shows the representation of an ordinary evaporative cooler. The air is forced through a wet honeycomb structure and the evaporation cools the air. This is a so called direct cycle, i.e. the same air that causes the evaporation is also used to cool the room. The disadvantage of the direct cycle is that the cool air is also humid and can cause discomfort. In the indirect cycle the humid cold air chills a secondary air duct, which remains dry and is breezed in the room. The primary wet air can be discharged. In most applications the indirect cycle is more pleasant, however, there are drawbacks that will discussed ahead. Our objective is to determine the configuration of the cooler that can extract the greatest amount of heat per second per unit volume of the device. This analysis is potentially interesting to cooler manufacturers.
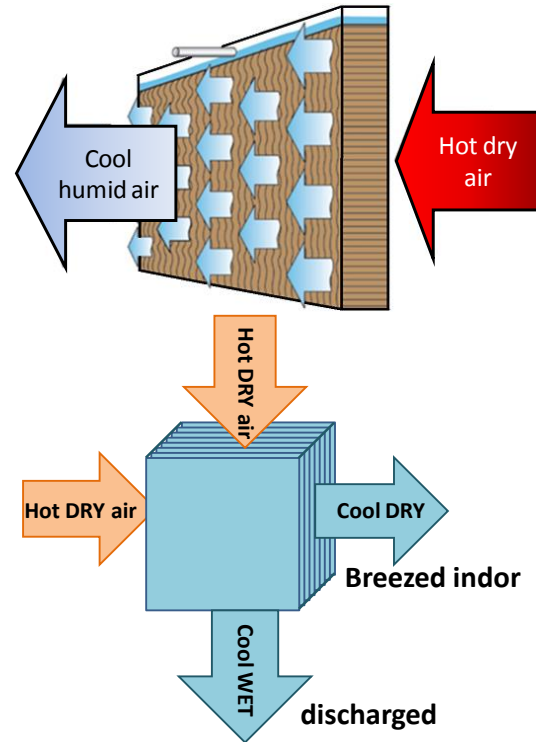

Figure 1. In the top and bottom, respectively, a representation of the direct and indirect evaporative cooler concepts.

2. Use of COMSOL Multiphysics

2.1 Geometry

The indirect cooler is represented in 2D as shown in fig. 2. There are two domains separated by a thin plate. The right side of this plate is kept wet. The evaporation in the right side will cool the plate and consequently the duct at the left. Therefore, the right domain will carry the humid cool air flux, while the left will take the cool dry air flux. The plate simulated here has thickness Ls=1 mm. The air domains have length Le and height Hi. In some simulations the ratio Hi/Le was the order of 100. To deal with the large aspect ratio of the domain we disabled the "keep aspect ratio" check box, so the geometry can be observed as a whole in a square region shown in fig. 2.
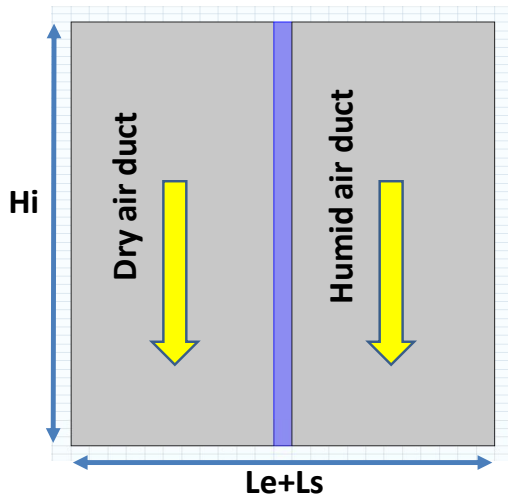
Figure 2. 2D geometry analyzed.

2.2 Meshing

The solution in the y-direction varies little as will be seen in the results. Thus, not only the geometry can be scaled down in the y-direction, but also the mesh can be hugely asymmetric in x and y directions without compromising the accuracy of the solution. In the setting of the "*free triangular*" we set the asymmetric scale of the mesh as shown in fig. 3. Note that the y-direction scale is the inverse of the aspect ratio of the geometry. This generates elements that are stretched the y-direction.
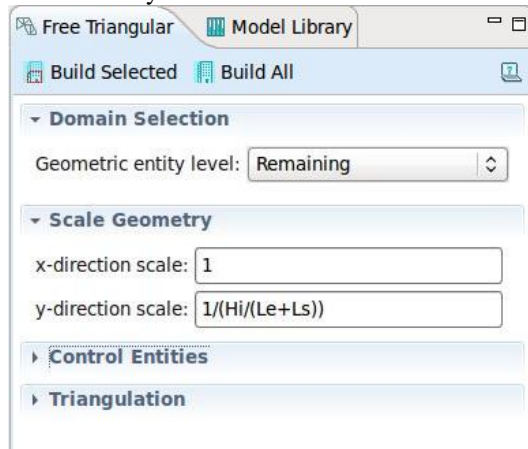

Figure 3. Feature in Free Triangular settings to enter the asymmetric mesh scale: In this example the scale in the y-direction is much larger.

These mesh settings largely reduce the number of elements and save processing time. The resulting mesh is shown in fig 4. The mesh is finer at the top where the air is incoming. In this region the solution has steep variations in the y-direction. We make the mesh finer closer to the air intake, where the solution has larger variations.
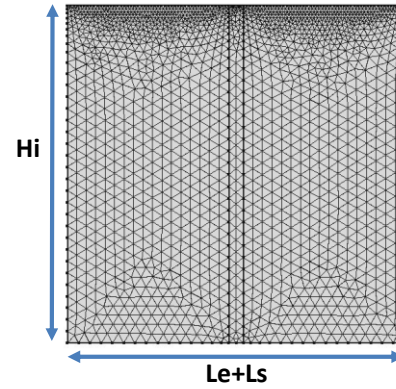

Figure 4. Final mesh using asymmetric scaling.

2.3 Physics

We couple a *Non-Isothermal Flow* and a *Diffusion-Convection Equation* to model this system. The latter evaluates the concentration of water vapor (*rho*) in the air according to:

$$- Dif \nabla^2 rho + \beta.\nabla rho = 0, \qquad (2)$$

where *Dif* is the diffusion of water vapor in air, β is the convection velocity field. *Dif* is available in the internet [1] as a function of the temperature. The $\beta = (u,v)$ are the fluid velocity components calculated in the *Non-isothermal flow* module. The boundary shown in fig 5 (c) is set as a Dirichlet boundary condition in the *Diffusion-Convection Equation* module. The *rhos* is defined as:

$$rhos = rhoair(T)\frac{Ps(T)}{Patm}, \qquad (3)$$

where *Ps* is the vapor pressure of water, given as an interpolating function [2], *Patm* is the atmospheric pressure.

The heat source in fig. 5(c) has to be defined according to the evaporation rate of water in that surface, which in turn, will depend on the local temperature and vapor saturation, all calculated iteratively. Then, the heat absorbed *Q* in W/m^2 is calculated from:

$$Q = \frac{\partial rho}{\partial x} Dif(T)*Ql(T) \qquad (4)$$

where, $Ql = 2.27$MJ/kg is the latent heat of evaporation [3].

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Fig. 5 shows the "volume force" condition applied in the air domains. This force is originated due to the cool air between the plates, which is always denser than surroundings. The volume force is vertical, point downward and is has units of N/m^3. It is a propriety that depends on the local temperature evaluated from Comsol. It acts in the y-direction given by the equation:

$$F_y = g\left[rhoair(T) - rhoair(Tamb)\right], \qquad (1)$$

where g is the acceleration of gravity, *rhoair* is the density of air given as an interpolating function, T is the local temperature in the domain and *Tamb* is the ambient temperature (300K/27°C). The term in the brackets is always negative so the force is downward.
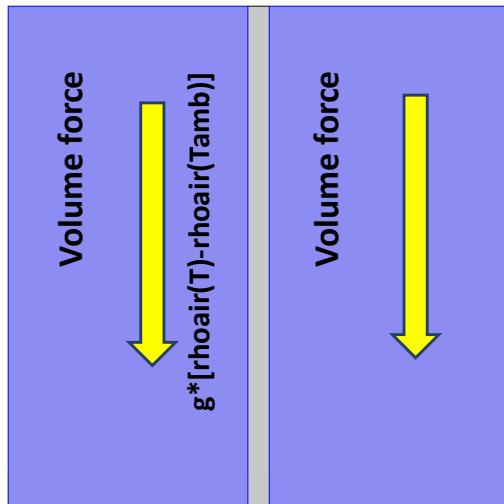

Figure 5. Volume force that drives the system is caused by the difference in density of the cool and warm air.

2.4 Boundary conditions

Fig. 6 shows the main boundary conditions used. In (a), the symmetry boundary is set at the sides, so the simulation account for a system twice larger than the figure. In (b), the air will enter from the top and exit from the bottom, so these boundaries are set as open boundaries. The driving force to move the air in the domains is the cool and denser air between the plates, which is the scent of the passive mechanism. In the *Diffusion-Convection Equation* module the top boundaries has a constant concentration defined as a Dirichlet boundary condition given as:

$$rho_{boundary} = rhos(Tamb)*he, \qquad (5)$$

where, *he* is the relative humidity of the ambient air. In (c), the right boundary of the plate is a wet surface, which will evaporate and suck heat in the process. At this boundary the humidity is always 100%, so the density of water vapor is the saturation density (*rho=rhos*). In (d), the walls between the two ducts preventing the dry and wet airs form mixing. they resist to the flux so there is an optimal separation, which sequestrate maximum heat for a given cooler volume.
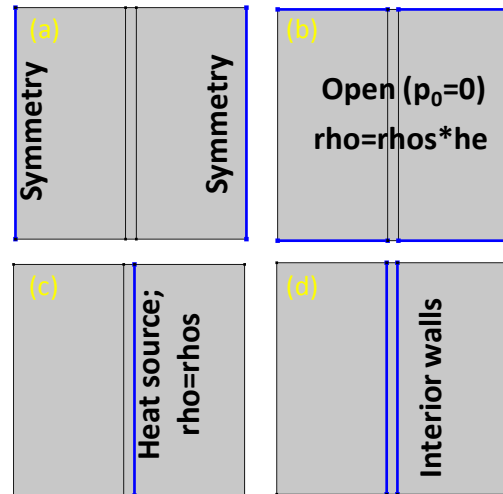

Figure 6. Main boundary conditions: In (a) the symmetry planes accounts for an infinite alternating dry and wet air conducting ducts. In (b) the open to air boundary condition (normal stress=0). In (c) the heat sink boundary, where water evaporates and cools. In (d) the plates, which keep the dry and humid airs apart;

2.5 Materials

The air in the ducts and the plate are the only two materials in this system. The proprieties Comsol requires to evaluate the dependent variables in the *Non-Isothermal Flow* are the density (*rhoair*), the thermal conductivity (*kair*) and the heat capacity at constant pressure *Cair*. The *Diffusion-Convection Equation* requires the diffusivity and the convection velocity, which we already defined in section 2.3. We set the region between the interior walls as having the density, conductivity and heat capacity of aluminum. However, this region is treated as a fluid with no volume force applied. Nevertheless the plate works as a solid because the interior walls. Leaving the plate a a fluid is much simpler than coupling another Physics to account for the solid plate. All parameters of the air are temperature dependent given as an interpolating

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function. These functions are built from tables available in the internet [3].

3. Results

The results were obtained in the stationary state and ambient temperature of 300K/27°C. The depth of the simulation domains are 1 m. Fig. 7 shows a typical simulation. The blue region indicates low values and red indicates high values. The temperature distribution is indicated in "thermal colors" in which the red is for low temperature and white is high temperature.
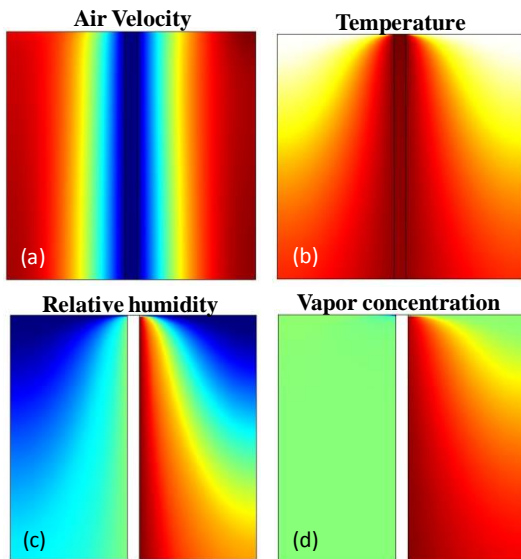

Figure 7. Result in the stationary state: In (a) the heavier cold air in the plates generates the velocity profile. In (b) the air temperature enters warm and cools in the way down as the wet plate evaporates. In (c) the relative humidity of the air increases in the way down due to the drop in temperature and, in the right side duct, also because of the evaporation. In (d) the concentration increases in the right duct due to the evaporation and remains constant in the left.

Fig 8 show the ideal spacing between the plates to obtain maximum heat extraction in the cooler as a function of the height of the cooler and the ambient relative humidity. The plates cannot be adjusted once the cooler is built, so the cooler has to be projected to work ideally in a specific humidity, out of which the performance won't be optimal.
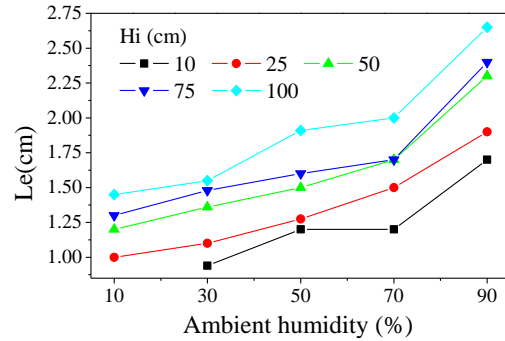

Figure 8. Ideal spacing as a function of the humidity and for several cooler heights.

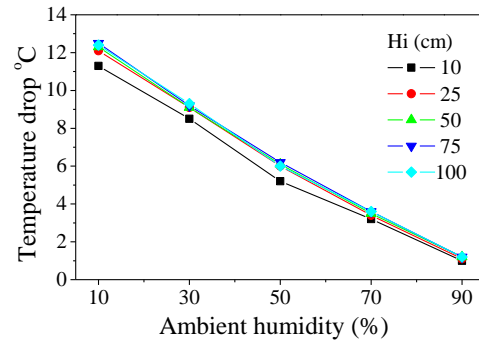Fig. 9 shows the temperature drop as a function of the humidity outdoor.

Figure 9. Temperature drop at the exit of the cooler as a function of the ambient humidity.

Fig 10 show the heat power removed from the dry air as a function of the humidity and for several heights. As a reference, the total power recommended to cool 100 m³ is ~3000W [4]. Naturally, the taller the cooler the more power it sequestrates, however, if we normalize the power by the volume of the cooler, we realize that it is more efficient to have several short coolers than a tall one. Fig. 11 shows the normalized power sequestrated. Note that the order of the curves is inverted in these figures, as the shortest cooler takes the least absolute power and also the most relative power from the air.

Fig 12 and 13 show the total heat removed including the wet air and the total heat normalized, respectively. This shall be the case if the humid air can also be utilized. Comparing figures 10 and 12 or figures 11 and 13, it's clear that the indirect cycle wastes half the cooling power by discharging the cool wet air, but it can provide more comfort.
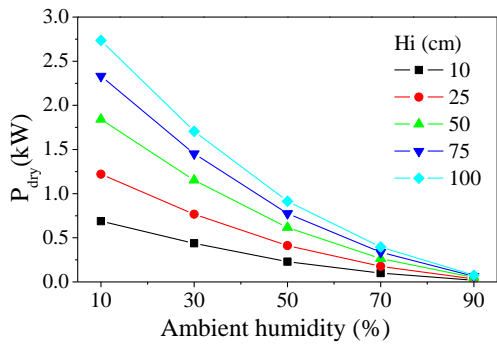
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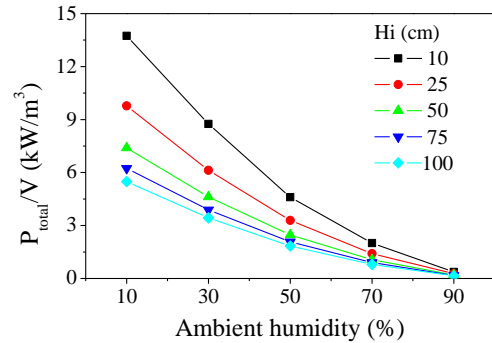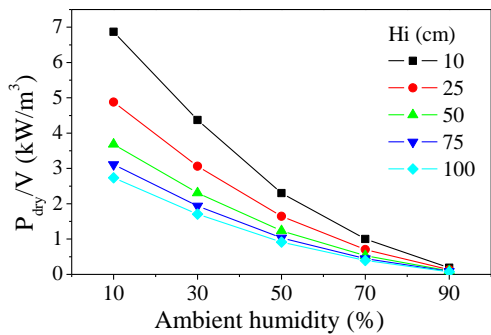**Figure 10**. Power removed from the dry air.

Figure 11. Power removed from the dry air per cubic meter of the cooler.

Figure 12. Total power absorbed in the cooler summing the dry and humid airs.
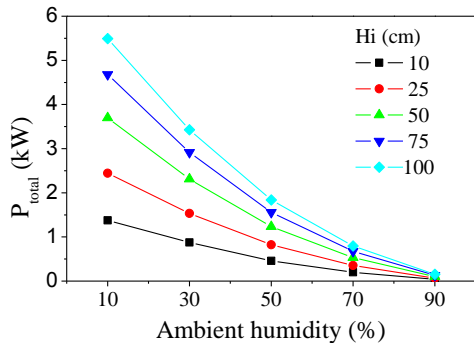

Figure 13. Normalized Total power absorbed in the cooler summing the dry and humid airs.

4. Conclusion

We simulated a passive indirect evaporative cooler and obtained the main characteristics of its operation in a simplified 2D model. We obtained the temperature drop, the heat power sequestration, the ideal spacing between the plates. We also obtained the distributions of temperature, vapor concentration, air velocity, and relative humidity. The indirect cycle has advantages and drawbacks compared to the direct cycle. The power sequestration in the indirect cycle is only half of the direct cycle for the same volume of cooler. In the indirect cycle is best to use only the dry air for ambients where people linger around. This provides more comfort, but also dismisses half the cooled air. The effectiveness of the indirect passive cooler is low in case of high ambient humidity. But this is also a problem for any evaporative cooler.

Acknowledgements

References

[1] http://physics.holsoft.nl/physics/ocmain.htm.
[2] http://en.wikipedia.org/wiki/Vapour_pressure_of_water
[3] http://www.engineeringtoolbox.com/water-thermal-properties-d_162.html
[4] http://www.diy.com/diy/jsp/bq/templates/content_lookup.jsp?content=/content/knowledge/calculators/btu_calculator/btu_calculator.jsp&noCookies=false#result

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