Condensation-Evaporation Modeling of drift from a mechanical cooling towertower

K. Han, PhD, PE
[ASHRAE Affiliation]

A. Mistreanu, PhD
[ASHRAE Affiliation]

C. Sobecki, PhD
[ASHRAE Affiliation]

B. d'Entremont, PhD
[ASHRAE Affiliation]

ABSTRACT

Numerical study for investigating the phase change behavior of a drift from a mechanical draft cooling tower is presented. A k-ε turbulence model, discrete phase model, species transport equations, and a heat-mass exchange model are solved simultaneously to capture the heat and mass transfer effects of a drift condensing and evaporating effects. The simulation result is compared to an available real scale test data and its limitation is also discussed. A parametric study is carried out by varying the ambient dry bulb temperature, ambient relative humidity, wind speed, liquid droplet diameter, and the drift amount. It is found that by combining the evaporation and condensation models, a drift phase change phenomena is accurately depicted. A parametric study reveals that wind speed impact is affected by the ambient temperature and the number of liquid droplets and their size play a critical role in the phase change phenomenon.

INTRODUCTION

Drift is small water droplets ranging from 10 to 2000 microns coming out of cooling tower that contain identical chemicals and solids as the circulating water such as salt, bacteria, dirt, and debris. Even though drift eliminators can reduce cooling water loss rate down to 5×10^{-5}%, the escaped tiny droplets still cause health concerns such as Legionella, pollutes nearby water sources, erosion of mechanical system, corrosion of steel structure, deposition on nearby equipment, and water loss. Therefore, a drift trajectory prediction is one of many concerns for mechanical draft cooling tower operation and design. Escaped drift from the cooling tower grows or diminishes by condensation and evaporation and condensation. Moreover, its dynamic behavior is affected by various factors like fan, buoyancy, ambient conditions, drift amount, liquid droplet size, as well as nearby topology such as buildings. These upper mentioned complex physics are difficult barriers for analytical or theoretical approaches.

Liquid droplet evaporation modeling has been reported in various applications. For the defense application, it is being used for lowering infrared signature (Han, 2014). It is also being applied to dry cooling systems in power industry to compensate the power loss by spraying water (Han, 2016). Regarding condensation modeling work in a mechanical wet cooling tower, many analytical approaches have been made (Hanna, 1972) to estimate plume rise which is not suitable to a condition having the recirculation near the ground and in complex environmental

K. Han is a fellow engineer at Savannah River National Laboratory, Aiken, S.C. **A. Mistreanu** and **B. d'Entrement** are senior advisory engineers at Savannah River National Laboratory, Aiken, S.C. **C. Sobecki** is a postdoctoral research associate at Savannah River National Laboratory, Aiken, S.C.

conditions. A CFD approach was recently reported (Ershov et al., 2021) using the phase change mass fluxes predicting equation derived from statistical mechanics as well as the thermodynamic states of the liquid and vapor without considering heat transfer. More recently, a simple condensation prediction model using CFD (Han et al., 2023) was developed based on psychrometric equations.

In this study, drift behavior from a mechanical cooling tower is studied with the assistance of a general purpose CFD software. The pre-established evaporation (Han, 2016) and condensation (Han et al., 2023) models track the growth and decay of drift by considering relative humidity, wind speed, liquid droplet size and amount, and ambient temperature. A subroutine is developed based on psychrometric equations and incorporates the phase change phenomena.

NUMERICAL METHOD

The numerical analyses for a two-dimensional domain were carried out by simultaneously solving species transport equations along with turbulence models, discrete phase models, the continuity equation, momentum equations, energy equations, and heat/mass exchange models to track the phase change of liquid droplets. The standard k-ε model is adopted as the turbulence model. The mixing and transport of species are modeled by solving conservation equations describing convection, diffusion, and reaction sources for each component species. The fluid phase is treated as a continuum by solving the Navier-Stokes equations, while the dispersed phase is solved by tracking a large number of liquid droplet particles throughout the calculated flow field. The dispersed phase exchanges momentum, mass, and energy with the fluid phase. Considering the lower volume fraction of the dispersed second phase, particle-particle interactions are neglected. The droplet trajectories are computed individually at specified intervals (every 10 iterations) during the fluid phase calculation. The trajectory of a discrete phase particle is predicted by integrating the force balance on the particle. Heat and mass exchanges are modeled using heat and mass transfer relationships. The moisture amount in the bulk gas is acquired from solution of the transport equation. Depending on the humidity ratio, condensation and evaporation modes are switched. When the humidity ratio exceeds the saturated value, droplet condensation is initiated. The exceeding moisture forms spherical liquid droplets, and the corresponding latent energy is used as a source term in the energy equation. All psychrometric parameters like saturation pressure and humidity ratio are calculated using the equations in ASHRAE fundamentals (2017). The second order upwind scheme is used for density, momentum, species, and energy discretization. The coupled scheme is introduced to pressure-velocity coupling. For turbulent kinetic energy and turbulent dissipation rate discretization, the first order upwind scheme is applied. The set of coupled governing equations are solved using the commercial CFD software. The detailed equations and modeling descriptions can be found in the liquid droplet evaporation modeling work (Han, 2016).

As seen in Figure 1, a numerical domain with a 4000 ft (1200 m) length and 2000 ft (600 m) height is constructed with 16.40 ft (5 m) squared meshes majorly, whereas 3.28 ft (1 m) size square cells are placed in the vicinity of a cooling tower. The cooling tower width and height are both 60 ft (18 m) and is located 1300 ft (400 m) from the left velocity inlet. A fan boundary is placed inside the cooling tower to depict the air motion near the cooling tower and predetermined air-water droplet mixtures are discharged from the top coolingtowe ropening having 31ft (9.5m) diameter.

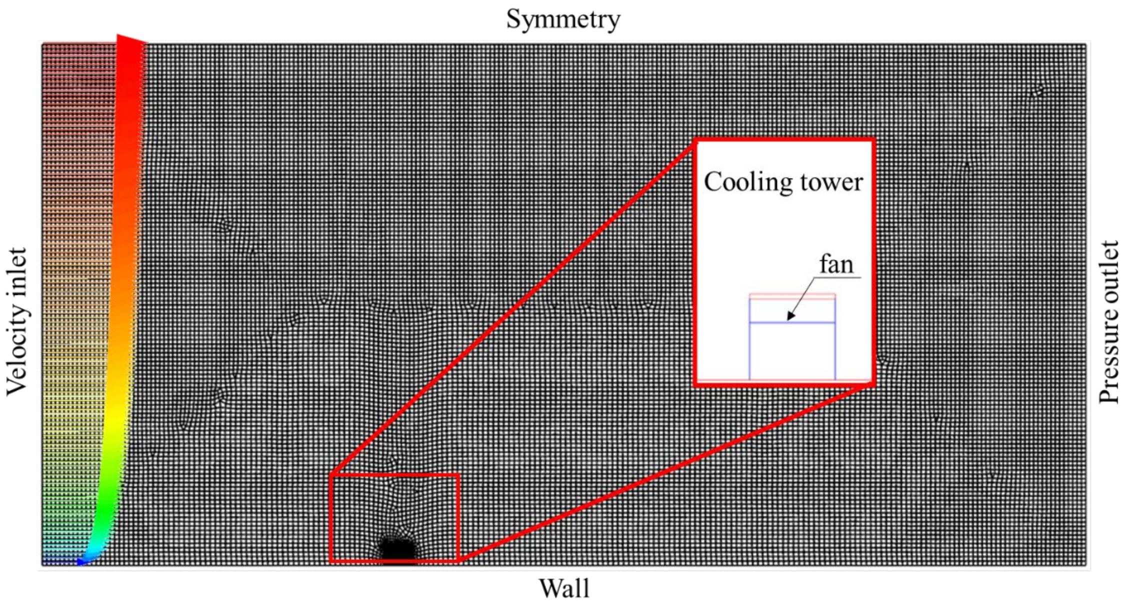

Figure 1 Numerical domain and boundary settings

The grid dependency is checked using three different meshes having 27,802, 44,728, and 79,160 cells. The averaged relative humidity on the ground level and air sides varies within -1.7% and 1% as meshes. The averaged temperature change as grid density is within -0.07% and 0.36%. These small variations as grid density are ignorable when it considers the result fluctuation in tables 1 and 2. Hence, a computational domain having 27,802 cells is selected for this analysis.

Figure 1 also shows the grids along with boundary conditions. Velocity inlet and pressure outlet boundaries are placed on the vertical boundary lines whereas a symmetry boundary is assigned on the top line to reduce backflow and improve convergence. Additionally, a no-slip boundary condition, with a zero-roughness height, is applied to the ground and walls of the cooling tower. For an open terrain with scattered obstructions having heights generally less than 30 ft (9 m), a velocity profile is proposed per ASHRAE handbook (ASHRAE Fundamentals, 2017).

$$U_H=1.58632U_{met}(H/10)^{0.14} \tag{1}$$

A typical meterological tower height is assumed to be 33 ft (10 m). The default wind velocity at meteorological tower is the half of the averaged fan air velocity 30 ft (9.12 m/s). The atmospheric temperature is height dependent and has a gradient of -0.003567°F/ft (-0.0065 K/m) as per the ASHRAE handbook (ASHRAE Fundaments, 2017). Moisture amount is initialized with the identical value for the entire numerical domain because water vapor amount does not vary with altitude up to 9900 ft (3 km) according to ASHRAE handbook (ASHRAE HVAC Applications, 2019).

RESULTS

To validate the numerical model, an actual field measurement (Meyer, 1974) is compared with the CFD results. Velocity and temperature profile from the actual test is introduced to the validation model. The selected ambient condition is 30°F (-1°C) dry bulb temperature, 50% relative humidity, and 16.73 ft/s (5.1 m/s) wind speed. Air flow rate of the fan is 23,200 ft^3/s (657 m^3/s). Dry bulb temperature and relative humidity of air coming out of the cooling tower are 81.5°F (27.5°C) and 98%. The liquid droplet diameter and the drift flow rate are assumed as 0.004 inch (0.1 mm) and 0.0008 lbm/hr (1.0×10^{-7} kg/s) considering typical droplet size and modern drift eliminator performance.

The actual temperature profile measurement at 1200 ft (365 m) altitude is compared to CFD in Figure 2 where a temperature difference is noticeable near the cooling tower location. This temperature inversion in the field test trial was caused by the mixing of two air flows having different wind directions at 1000 ft (300 m) altitude. This illustrates the difficulty of the atmospheric modeling in CFD, especially for two-dimensional geometry.

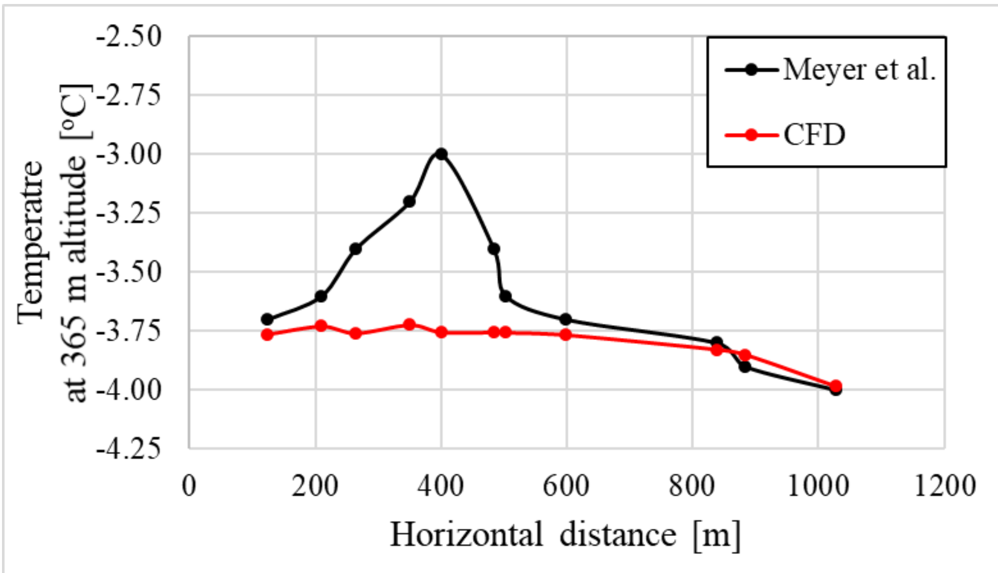

Figure 2 Comparison of temperature between the actual field test trial and CFD.

A simple comparison illustrates the simultaneous condensation/evaporation effect compared to evaporation or condensation only scenarios. When only condensation is allowed for Case 5, condensation flux (kg/s) increases 0.12% on the ground level while 0.1% decrease in air compared to a case including both phase change phenomena. In case only evaporation is activated, condensation does not occur at all as it should be. This illustrates that both evaporation and condensation should be considered in a model.

For the humidity and ambient dry bulb temperature impact on condensation and evaporation, total nine cases composing of three different dry bulb temperatures and three relative humidity are established as Table 1. Due to the lack of the mechanical cooling tower operating data, the plume status coming out of the tower is estimated using a simple energy balance.

$$m_{w,i}c_{p,w}T_{w,i}+m_{a,i,wet}h_{a,i}=m_{w,o}c_{p,w}T_{w,o}+m_{a,o,wet}h_{a,o} \tag{2}$$

Dry air mass can be assumed as wet air mass when the small amount of water vapor mass is considered. After dividing the energy equation with the air mass, the cooling tower outlet air enthalpy can be expressed with the circulating water temperature difference and other known values. It is assumed that the fan delivers the constant air flow rate.

$$h_{a,o}=h_{a,i}+(m_{w,i}/m_{a,dry})\times(T_{w,i}-T_{w,o}) \quad (3)$$

If the generated heat from a boiler is constant, water side temperature difference does not change as the ambient condition because water mass flow rate and specific heat of water do not vary much with temperature. In this way, exit air temperature from the cooling tower can be calculated by assuming fully saturated condition and the obtained exit enthalpy from the equation above.

Table 1 presents the ambient air-dry bulb temperature and relative humidity impacts in terms of phase change mass source (kg/s), m_s. Negative number means condensation occurs and positive number indicates evaporation is dominant. Magnitude represents the phase change mass flow rate. For a given dry bulb temperature, condensation is strengthened with the ambient relative humidity. For a given relative humidity, evaporation increases with ambient dry bulb temperature because the increased temperature allows air to hold more moisture. Even though evaporation is dominant in hot air condition (35°C dry bulb temperature), condensation still happens when the relative humidity is high (90%). Interestingly, the condensation and the evaporation are balanced on the ground level for Case 7. However, it should be noted that the condensation modeling parameter needs to be tuned to correlate CFD data to actual condition. In general, phase change intensity of the ground level is stronger than that of air due to the recirculation of plume. The plume coming out of the cooling tower mostly resides the ground level as Figure 3 shows.

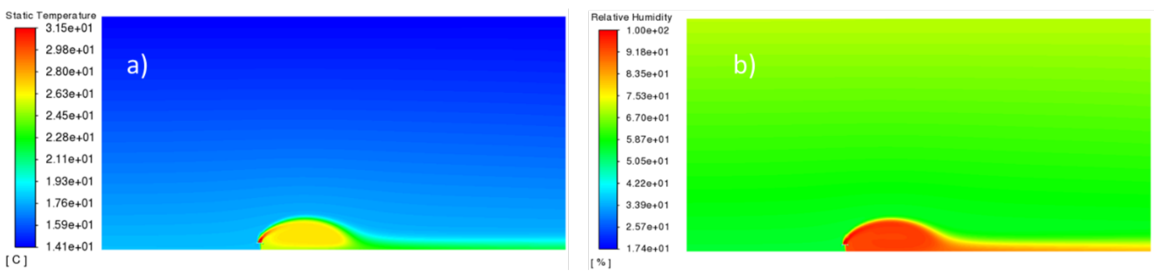

Figure 3 (a) Temperature (b) Relative humidity contours of Case 5.

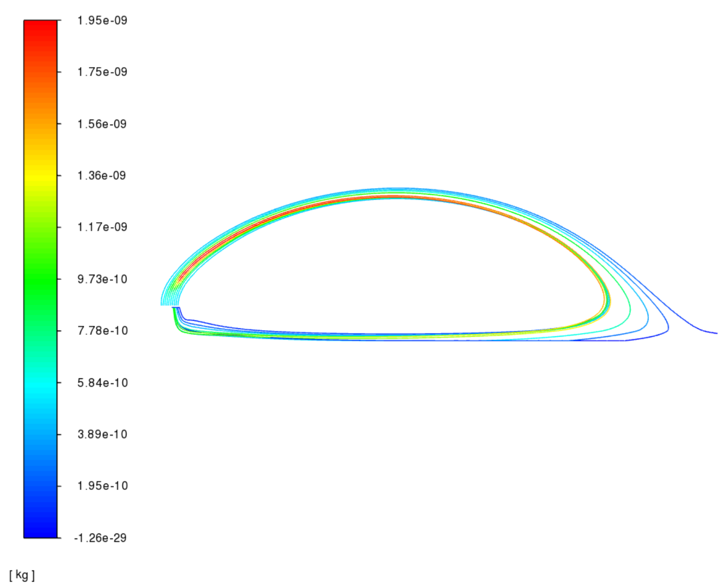

Figure 4 Liquid droplet mass for Case 5.

Table 1. Mass source as ambient conditions

Case	T_{db} [°C]	RH [%]	m_s on the ground level [kg/s]	m_s in air [kg/s]
1	-1	20	-8.11E-07	-4.22E-08
2	-1	55	-1.17E-06	-6.34E-08
3	-1	90	-1.46E-06	-8.24E-08
4	18	20	-1.52E-07	-1.79E-09
5	18	55	-2.37E-07	-5.14E-09
6	18	90	-9.84E-07	-4.38E-08
7	35	20	0	4.54E-12
8	35	55	5.26E-11	4.50E-12
9	35	90	-1.10E-06	-8.74E-09

Case 5 is a good example showing simultaneous condensation and evaporation of drift. The released liquid droplet mass increases initially by condensation and gradually reduces as flow path by evaporation. Hence, this case is used to investigate the impact of other parameters like wind speed, liquid droplet diameter, and drift mass flow rate as summarized in Table 2. 50% reduced wind velocity reduces the recirculation zone and augments condensation which is different from the previous study (Han et al., 2023). In cold environment (Han et al., 2023), strong wind strengthens the condensation by enlarging the mixing zone with lower ambient temperature air. For a mild ambient condition like Case 5, evaporation and condensation balances differently as wind speed. Regarding the particle size impact, the number of droplets increase with reducing droplet diameter because the total drift flow rate is fixed. Smaller droplet enhances phase change by increasing the surface area and having more interacting chance with water vapor molecules. As it can be easily predicted, drift amount increase helps the phase change phenomenon by providing increased number of droplets.

Table 2. Relative change of mass source as wind and drift

Case	Wind speed	Liquid droplet diameter	Drift amount	m_s on the ground level	m_s in air
10	-50%	-	-	75%	242%
11	50%	-	-	-80%	-82%
12	-	10%	-	6563%	4605%
13	-	1000%	-	-100%	-100%
14	-	-	50%	49%	49%
15	-	-	-50%	-50%	-50%

CONCLUSION

A parametric study is carried out to investigate the drift phase change behavior using computational fluid dynamics coupled with a subroutine to model evaporation and condensation at the same time. A set of turbulence models, discrete phase models, heat and mass transfer models, and species transport equations are solved simultaneously. The overall behaviors of drift phase change as ambient temperature and relative humidity agree well with a common physical understanding. The role of wind speed and the recirculation zone varies as ambient condition and further study to correlate their impact on evaporation and condensation is required. Reducing liquid droplet size intensifies phase change phenomenon of drift by enlarging the surface area and increasing the number of particles. The effort to reducing drift amount should be persisted when its linear correlation to phase change is considered. Finally, a comparison with the available experimental data illustrates the atmospheric modeling difficulty.

ACKNOWLEDGMENTS

"This work was produced by Battelle Savannah River Alliance, LLC under Contract No. 89303321CEM000080 with the U.S. Department of Energy. Publisher acknowledges the U.S. Government license to provide public access under the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan)."

NOMENCLATURE

$c_{p,w}$ = specific heat [J/kg/K]
h = enthalpy [J/kg]
H = altitude [m]
m = mass flow rate [kg/s]
T = Temperature [K]
U = velocity [m/s]

Subscripts

a = air
dry = dry air
H = Height
i = inlet
met = meteorological
o = outlet
s = source
w = water
wet = moist air

REFERENCES

Han, D. 2014. Numerical Analysis on Spray Cooling of Gas Turbine Exhaust Gas. *Int. J. Transport Phenomena* 13: 267-276.
Han, D. 2016. Numerical Study on Air Temperature Reduction by Water Spray. *Int. J. Transport Phenomena*. 14:205-217.
Hanna, S. 1972. Rise and Condensation of Large Cooling Tower Plumes. *J. Applied Meteorology* 11:793-799.
Ershov, M.I., Tuponogov, V.G., Abaimov, N.A., and Gorsky, M.A., 2021, CFD modelling of the condensation, evaporation, coagulaton and crystallization processes in the cooling tower emission. *HMTHSF* 2021.
Han D., Mistreanu A., Sobecki C., d'Entremont B. 2023 Condensation modeling of plume from a mechanical draft cooling tower, *ASHRAE Transactions* XX(X):XXXX-XX.
ASHRAE. 2017. *ASHRAE Handbook—Fundamentals*. Atlanta: ASHRAE.
ASHRAE. 2019. *ASHRAE Handbook—HVAC Applications*. Atlanta: ASHRAE.
Meyer, J.H., Eagles, T.W., Kohlenstein, L.C., Kagan, J.A., and Stanbro, W.D., 1974, Mechanical Draft Cooling Tower Visible Plume Behaviors: Measurements, Models, Predictions, Preceedings of the Cooling Tower Environment. *U.S. Atomic Energy Commission Symposium*: 307-352.