International Journal of Technology (2015) 3: 365-379
ISSN 2086-9614

EXPERIMENTAL AND SIMULATION STUDY ON THE PERFORMANCE OF COUNTER FLOW CLOSED COOLING TOWER SYSTEMS

Budihardjo[1*], Nasruddin[1], Mohammad Hafil Nugraha[1]

[1] *Department of Mechanical Engineering, Faculty of Engineering, Universitas Indonesia Kampus Baru UI Depok, Depok 16424, Indonesia*

(Received: April 2015 / Revised: July 2015 / Accepted: July 2015)

ABSTRACT

Cooling towers are required in building HVAC systems that use water as the cooling condenser fluid. Cooling towers used in this study are of the forced draft, counter flow, indirect/closed evaporative type. This study sought to demonstrate the performance characteristics of a closed system cooling tower by its effectiveness value, Number of Transfer Units (NTU), cooling capacity, and overall heat transfer and mass coefficient of the cooling tower. Experiments were performed on a heat exchanger coil intercrossed with ⅜ inch diameter intersections on parallel lines. Results of the experiment were then compared with the heat and mass transfer correlations taken from previous studies, and also combined with Computational Fluid Dynamics (CFD) simulations to examine the physical processes that occur in the cooling towers. All the experimental results, theoretical calculations and CFD simulations used variations of warm water mass, cold air, and water spray to present a clear description of the performance characteristics of a closed system cooling tower. The results of this study have shown that an increase in the amount of water spray mass flow causes an increase in the effectiveness value, heat transfer and overall mass transfer, as well as the cooling capacity of the cooling tower. The waste heat typically utilizes up to 80% of latent evaporation heat, and 20% of sensible air heat; however, waste heat in the closed system cooling tower utilizes 100% of latent evaporation heat. The mass transfer coefficient rate tends to be stable for a small mass of water spray.

Keywords: CFD; Cooling tower; Evaporative cooling; NTU; Spray water

1. INTRODUCTION

There are several types of cooling towers used in the applications. Wet cooling towers work using natural flow or mechanical flow. Mechanical cooling towers can either be of the flow pressure or the flow induced type. Air flow and water flow can be in opposing directions (counter flow), cross-flow, or both. Each type of cooling tower has its own characteristics. Based on the type of contact between the hot fluids with the cooling air, the cooling tower can either be a direct contact or an indirect contact type.

In the direct contact cooling tower or open type cooling tower, the direct contact of water and air causes water evaporation and consequently, a simultaneous reduction in temperature. The result is that water which evaporates in the form of vapor (water in gas phase) is added to the air and causes humid air on the outlet side. The water that evaporated must be replaced with new water to maintain the condenser cooling water discharge circulation; this water is called make-

* Corresponding author's email: budihardjo@eng.ui.ac.id, Tel. +62-21-7270032, Fax. +62-21-7270033
Permalink/DOI: http://dx.doi.org/10.14716/ijtech.v6i3.986

up water. The volume of make-up water in a direct contact cooling tower is quite large,because a great flow of water and air contact is needed to maintain the effectiveness of the cooling tower.

In urban areas, water consumption is limited. Buildings which consume large amounts of water can be classified as wasteful energy buildings. A series of fundamental research on heat and mass transfer in a closed type cooling tower which used both cross flow and counter flow within plain or finned tubes has been carried out. Various kinds of experimental test rigs to support such research also have been developed.

The first theory for thermal performance of cooling towers has been introduced by Merkel (1925). Some critical assumptions have been made by Merkel to solve the calculation. Therefore, the Merkel method could not be represented in the heat and mass transfer phenomena in the counter flow cooling tower packing.

In the early nineties, Poppe and Rögener (1991) developed a mathematical model which required iterative calculations and also did not include the Merkel assumptions. Kloppers and Kröger (2005) explained in detail the critical differences between both methods. Khan and Yaqub (2003) developed a more detailed version of the e-NTU method. Gan et al. (2001) also carried out research and experimentally measured the performance of the closed wet cooling tower. It has been shown that the efficiency of the cooling tower increases with the air flow rate and decreases with the increasing water flow rate. The efficiency increases slightly with the increasing wet bulb temperature of the supply air. It also elaborated on the use of CFD for performance evaluation of cooling towers in terms of cooling capacity and pressure loss or for optimum design of cooling towers, according to tube pitches and flow rates of supply air and spray water (Riffat et al., 2000).

Hasan (2005) and Hasan and Siren (2002) presented a theoretical analysis and computational modeling of Closed Wet Cooling Towers (CWCTs). It was shown in simplified analytical models with an assumption of a constant spray water temperature. The results of the simplified CWCT model are integrated with CFD to assess the effects of air flow distribution inside the tower on its performance.

Stabat and Marchio (2003) developed a behavioural model of a closed cooling tower using building energy simulation programs. The effectiveness of the model was used by simplification of heat and mass balance in the transfer equations. Facão and Oliveira (2004) presented the analogy between heat and mass transfer in an indirect contact cooling tower to estimate the mass transfer coefficient and the result reported a deviation in experimental results of 53 and 90% (in a Sherwood number coefficient), depending on the use of the heat transfer coefficient from the Zhukauskas correlation or from CFD simulations. Kaiser et al. (2005) developed a new type of cooling tower and numerically studied it. This showed that a strong influence occurred at the average water drop size on efficiency and reveals the effect of other variables like wet bulb temperature, water mass flow to air mass flow ratio and temperature gap between water inlet temperature and wet bulb temperature. Sarker et al. (2008) has carried out experimental measurements using bare and fin tubes at the heat exchanger to enhance the cooling capacity of the hybrid closed circuit cooling tower. Their results for cooling capacity and pressure drop concerning the variable air inlet velocities, wet-bulb temperatures, cooling water inlet temperatures and the air to cooling water volume flow rate ratio (G/W ratio) were presented.

The study of mass transfer coefficients for variable air velocity of CWCT has been conducted by Shim et al. (2008). It was reported that the cooling capacity per unit volume of the CWCT using 15.88 mm tubes with two paths has the highest values and therefore, it is effective to use smaller diameter tubes in the heat exchanger in order to increase surface area per unit volume.

Heyns and Kröger (2010) investigated the thermal-flow performance characteristics of an evaporative cooler from the experimental results. It established also correlations for the water film heat transfer coefficient, air–water mass transfer coefficient and air-side pressure drop.

A novel, closed wet cooling tower has been designed and studied numerically by Xia et al. (2011). The result was the mass transfer area of the spray water and the airflow is about three times larger than the heat transfer area of the process water and the spray water in the Heat and Mass Transfer Unit (HMTU). Zheng et al. (2012) carried out experimentally the performance characteristics of a new oval tube, closed wet cooling tower and also developed a mathematical model to predict the outlet cooling water temperature. Papaefthimiou et al. (2012) has developed a new model to simulate the processes taking place inside a closed wet cooling tower and also to investigate the effect of ambient air conditions on its thermal behaviour.

The review of indirect evaporative cooling technology concerning current status and research is reported by Duan et al. (2012). This study will portend to be of an enormous implication in promoting the application of the indirect evaporative cooling technology in buildings and to contribute to realization of low carbon air conditioning for buildings and associated energy saving and carbon emission measures. Jiang et al. (2013) studied about the effect of the process water temperature and flow rates of the air, spray water and process water on the cooling capacity, wet bulb efficiency, heat and mass transfer coefficients in a cross-flow CWCT based on the fin-tube arrangement. The flow arrangement of the air and the process water was counter flow, while that of the air and the spray water was cross flow. The results are empirical correlations between the heat and mass transfer coefficients based on the influencing factors.

In this study, the performance of counter flow, closed wet cooling tower will be analyzed. Experimental tests will be carried out to investigate the behaviour and influencing factors of the closed wet cooling tower.

2. METHODS AND TEST EQUIPMENT

The testing equipment used is the Mass and Heat Transfer Experimental Apparatus that is installed in the Refrigeration and Air Cooling Technical Laboratory of Department of Mechanical Engineering, Faculty of Engineering, Universitas Indonesia. An experimental closed wet cooling tower model is shown schematically in Figure 1. The tower is comprised of a 0.28×0.28 m^2 cross sectional area about 1.0 m height. A bare copper tube type bundle, having an outer diameter of 10.5 mm, has been installed inside the tower using a staggered arrangement, which has 10 rows and 8 columns with a transversal pitch of 25 mm and lateral pitch of 23 mm and the total transfer area of 0.50 m^2. The total number of tubes is 80.

A thermostatic bath (1) contains a water temperature that can be adjusted within a 3 kW capacity electric water heater. With the help of pump P1, which has a capacity of 24, l/min and head of 6 m, the water from the thermostatic bath is then distributed to the tube bundle. Water flow rate into the tube bundle can be adjusted with a manual valve and rotameter FM1. The water outlet from of the tube bundle and is then distributed back to thermostatic bath.

The spray water is distributed at the top of tube bundles. Using the water electric Pump P2, which has a capacity of 30 l/min at 30 m head, a manual valve and a rotameter FM2 the flow rate of the spray water to outer surface of the tube can be varied. And then, spray water is distributed back into the water tank (3) and then recirculated to the upper part of tube bundles. The rotameter FM1 has an accuracy of 5 L/h and FM2 has an accuracy of 0.5 L/h.

A centrifugal blower (5) with the capacity of 16 m3/min, a static pressure 180 mmH_2O, with 0.75 kW power consumption has a function to deliver the outside air into the test chamber.

The air temperature can be variated with an electric air heater (6) which has a capacity of 3 kW. The air flow pattern can be established by the flow straightener (7). The amount of air flow rate can be varied with the suction air shutter at the blower and indicated by orifice plate (8) within the accuracy of a 1% reading.

The air condition in the inlet and outlet of the test chamber can be seen from reading the psychrometers (11) and (10) which consist of calibrated dry and wet bulb temperature readings from resistance type thermometer detectors PT100 within an accuracy of 0.1^{o}C. The temperatures of the cooling water, spray water and air were measured by K-type thermocouples within an accuracy of 0.5^{o}C.

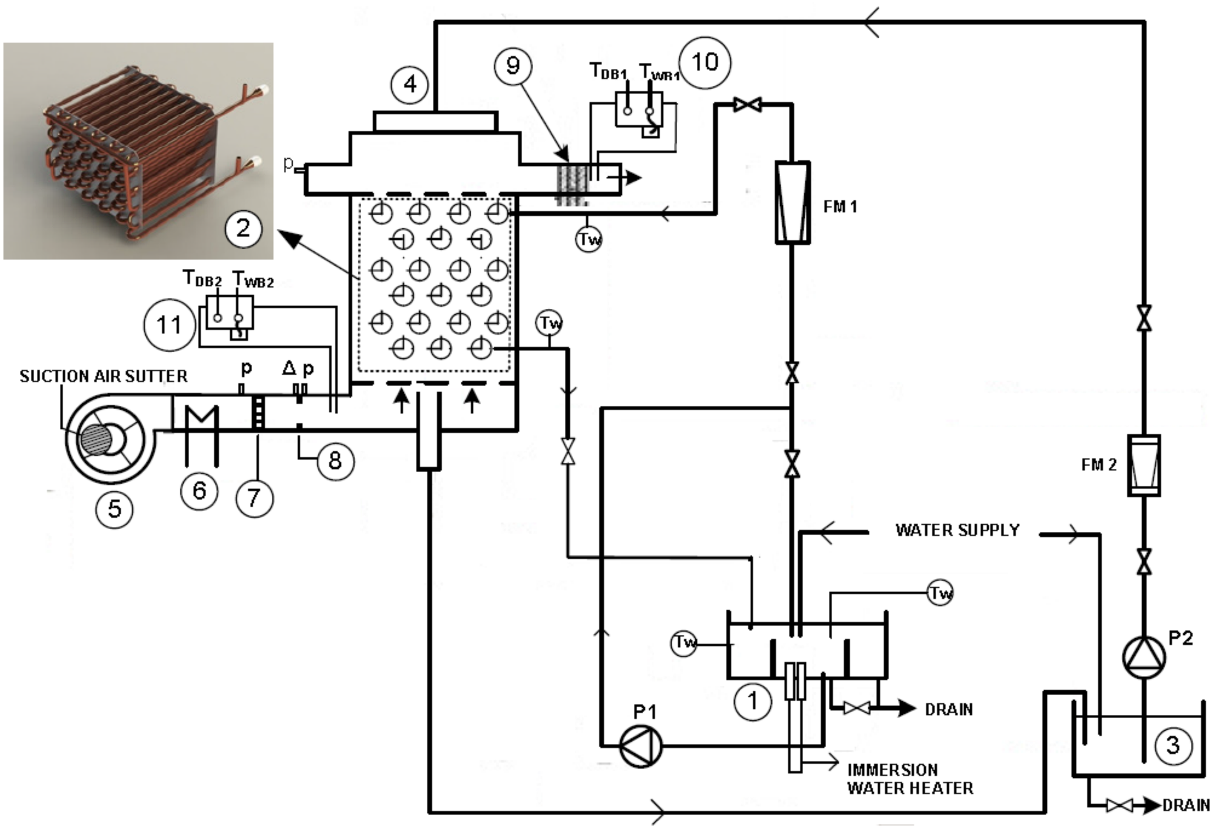

Figure 1 Experimental facility of forced draft, closed wet cooling tower

The experiment was conducted with variations of the cooling water flow rate, cooling water inlet temperature, spray water flow rate, wet-bulb temperature and inlet air velocity. All the data for each experiment were collected and recorded by data acquisition. Table 1 shows the experimental conditions and the tower geometry.

Table 1 Experimental conditions

Cooling water (tube water)	Flow rate	200–500	kg/h
	Inlet temperature	38	^{o}C
Spray water	Flow rate	200–1,100	ml/min
Air	Flow rate	125–275	kg/h
	Inlet dry bulb temperature	34–38	^{o}C

First, the overall heat transfer coefficient is calculated, with correlations associated with the phenomena that occurs in a closed system cooling tower system. Next, a model is constructed with geometric columns and relevant mathematical equations and conditions are entered before meshing is performed. Model dimensions are adjusted using existing equipment to obtain results close to actual conditions.

Then, verification of the model is made to ensure that the initial simulation/dummy shows acceptable results when using a variety of models and equations obtained from the calculations. After seeing the results of the initial simulation, meshing optimization can be conducted. Once more simulations are performed with various input variables.

3. THEORETICAL MODEL

Effectiveness of the cooling tower is calculated by

$$\eta = \frac{range}{range + approach} \tag{1}$$

where *range* = $t_o - t_i$ and *approach* = $t_i - tw_i$. Integrating the energy balance equation, from the inlet to the outlet of the cooling tower with t_s constant gives the equation of

$$\frac{U_o A_t}{m_w C_w} = \ln\left(\frac{t_{w.i} - t_s}{t_{w.o} - t_s}\right) \tag{2}$$

The heat transfer between saturated air-water spray with overall air, represented by enthalpy changes based on Merkel's equation, integrating the energy balance equation of the air inlet to the outlet gives:

$$\frac{KA_t}{m_a} = \ln\left(\frac{h'_s - h_{a.i}}{h'_s - h_{a.o}}\right) \tag{3}$$

taking the log-mean value of the difference in temperature and enthalpy of Equations 2 and 3.

Values *U*o and *K* are the overall heat and mass transfer values. The theoretical value of the *U*o equation is calculated by the equation of:

$$\frac{1}{\mathrm{U}_o\mathrm{A}} = \frac{1}{\alpha' s^{A_o}} + \lambda_{wall} + \frac{1}{\alpha' w^{A_i}} \tag{4}$$

3.1. CFD Simulation

CFD simulation modeling in this study, approaches the actual condition by applying some assumptions.

3.1.1. Simulation assumptions

a) Cooling tower domain
Simulations are carried out in two dimensions. Domains are modeled in accordance with the dimensions of existing tools with a cross-section at the air inlet and outlet sides. It is intended to simplify the simulation domain, because of limited hardware resources, and to speed up simulation time.

b) Coil wall
On the coil wall, the simulation is arranged as a wall boundary condition, by entering a roughness value. Simulations are carried out by inserting a constant temperature value on the wall. For the type of interaction with a discrete phase, splash walls are selected for this type of interaction.

c) Spray water

Spray water is modeled with discrete phases. From the beginning of the simulation it can be seen a tendency that the smaller the droplet diameter, the greater the evaporation will be. This simulation uses a water spray droplet diameter of 1 mm.

The model used in the simulation is activated using an energy equation, a momentum equation, k-epsilon turbulence equations, and equation species.

Domain mesh density in the cooling tower varied according to the gradient of the dummy simulation results. The largest gradient occurs in the area around the coil walls and then it occurs in the density of the area given the denser mesh. Mesh density on the circular coil walls of 1 mm, which is then given a value growth rate of 1.2. Simulations are performed on a nodal number 143488 and a mesh number of 67953 elements, as shown in Figure 2.

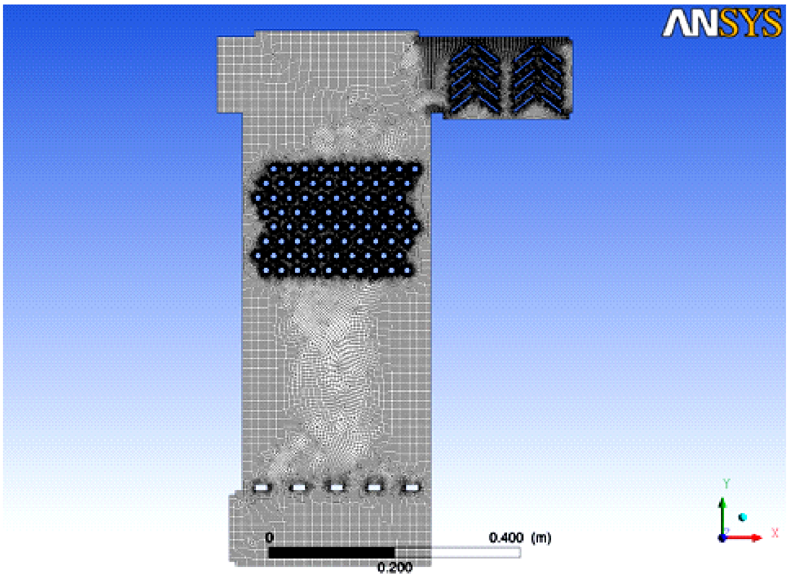

Figure 2 Meshing on the geometry of the closed wet cooling tower

4. RESULTS AND DISCUSSION

4.1. CFD Simulation Results

Simulations were performed with the energy equation convergence criteria at $1e^{-06}$, and $1e^{-03}$ for other equations (continuity, momentum, *k*, epsilon, H_2O), and a simulation convergence at a 130–150 iteration.

The velocity vector shows that the speed of cold air as a medium of displacement increases as it passes through the heat exchanger coil arrangement. An increase in air velocity also increases the Reynolds number, Nusselt, and the heat transfer convection coefficient. The velocity vector also indicates turbulence in the region before passing the coil arrangement, as shown in Figure 3. This is due to the fact that the preexisting air is not evenly distributed before it enters the column. The highest speed is in the drift eliminator region due to a significant change in space.

4.2. Energy Equilibrium

A margin of error for the equilibrium value comparisons in the experiment did not exceed 30% of the equilibrium line, as shown in Figure 3, which means that this experiment is quite acceptable. From the graphic chart it can be noted that the change in energy values on the air side tends to be greater than on the water side.

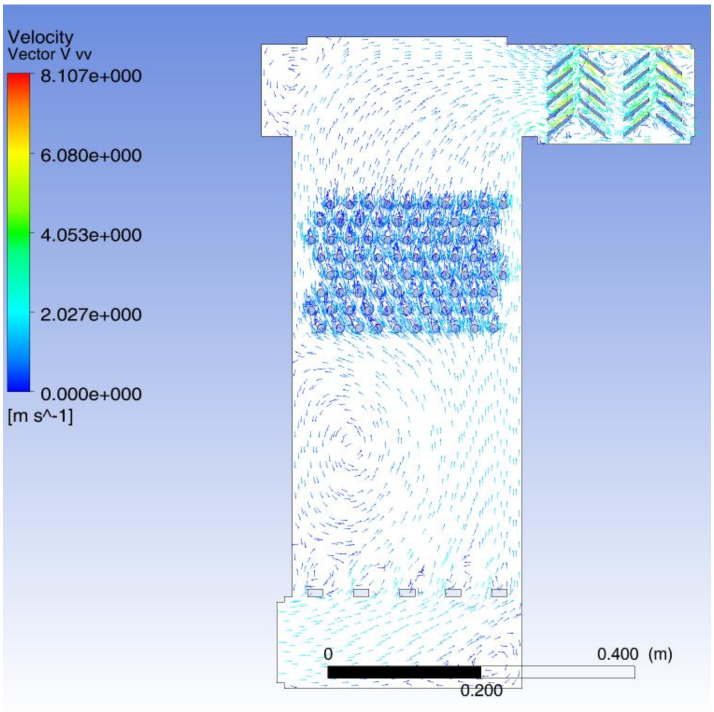

Figure 3 Velocity vector at the cooling tower

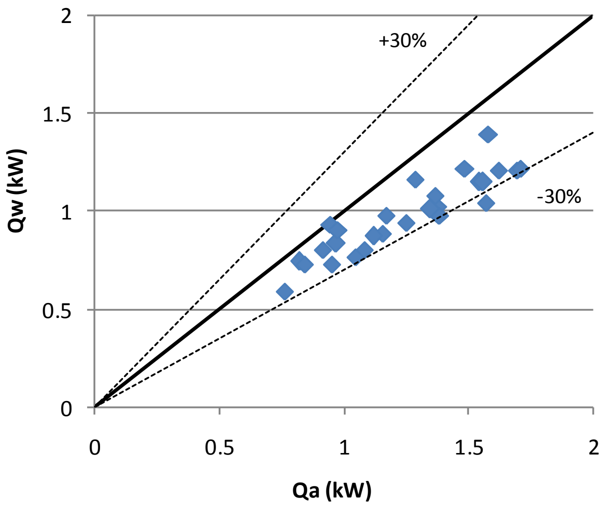

Figure 4 Energy equilibrium between the air side and the water side

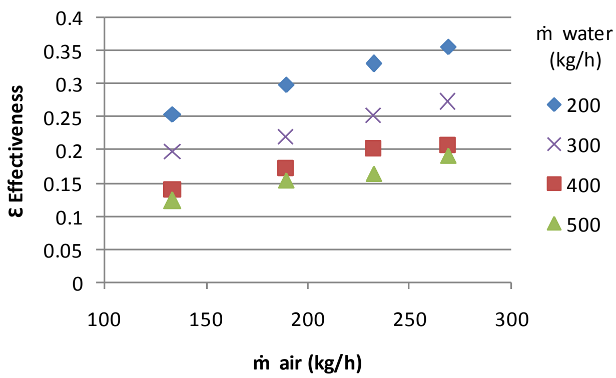

Figure 5 Effectiveness with air mass flow variation

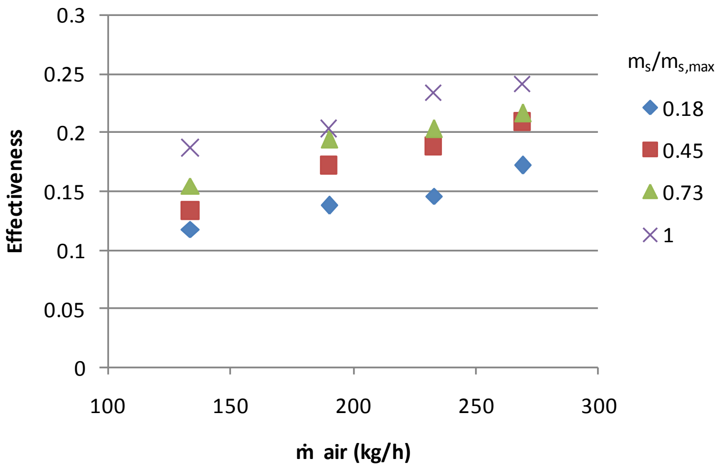

Figure 6 Effectiveness vs. air mass flow (water spray mass variation)

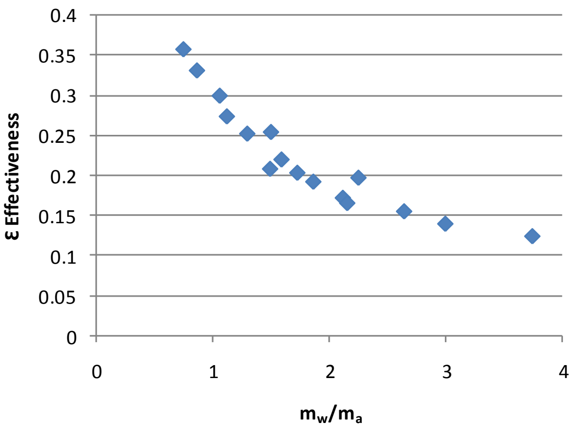

Figure 7 Effectiveness vs. ratio m_w/m_a

4.3. Wet Bulb Effectiveness

The effectiveness value of the cooling tower is obtained from Equation 1 and in Figures 5 and 6, which show the influence of air and water mass flow rates on the effectiveness value of the cooling tower. Figure 5 depicts a scenario where a higher effectiveness value equates to a higher air mass flow. In contrast, value effectiveness decreases with increasing volumes of cooled warm water and the effectiveness value increases with an increase in water spray mass, as illustrated in Figure 6. The greatest effectiveness value is with the maximum flow of air mass and water spray mass.

Figure 7 shows the effect of the warm water mass flow ratio compared to the air mass rate. From the graph in Figure 7, it is apparent that effectiveness values have a tendency to decrease with the increasing ratio mw/ma. Effectiveness values are influenced by the magnitude of warm water mass flow and cold air. The less warm water that is cooled, the lower is the effectiveness value. Conversely, the more cold air there is flowing, the greater is the effectiveness value.

4.4. Number of Transfer Unit (NTU)

NTU values represent the performance of the cooling tower. This value will tend to be constant with variations of water flow and the amount of air that occurs in the cooling tower system, as formulated in Equatioin 5:

$$K(og) = NTU = \int_{H2}^{H1} \frac{dH}{H' - H} = \sum \frac{\Delta H}{H' - H} \quad (5)$$

NTU values are solved numerically, and the generated values are plotted in Figures 8 and 9. Unlike the effectiveness values shown in Figures 5 and 6, the NTU values, which are also used as a benchmark in cooling tower performance, give different results. The NTU value does not seem to vary in response to changes in air and warm water. With warm water mass flow of 300 kg/h, and with a large water spray mass, the NTU value stands at 0.5 and decreases with an increase in air mass rate.

The characteristics of the NTU value can be seen by plotting a NTU comparison graph with water spray mass flow ratio (m_s) and with air mass flow ratio (m_a). The graph shows that the higher the value of NTU is, the greater the m_s/m_a ratio is, Figure 10.

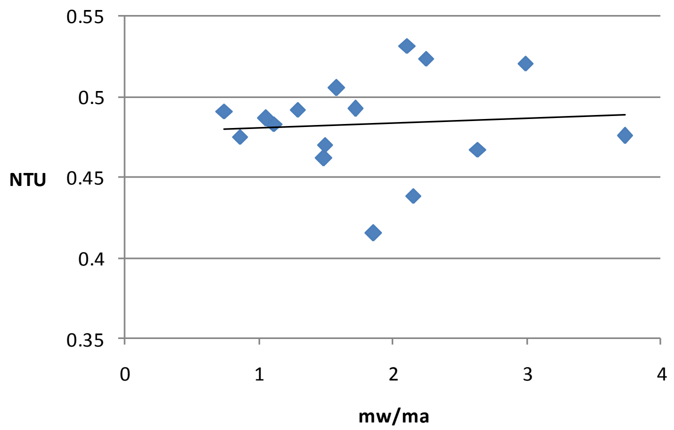

Figure 8 NTU value influenced by ratio m_w / m_a

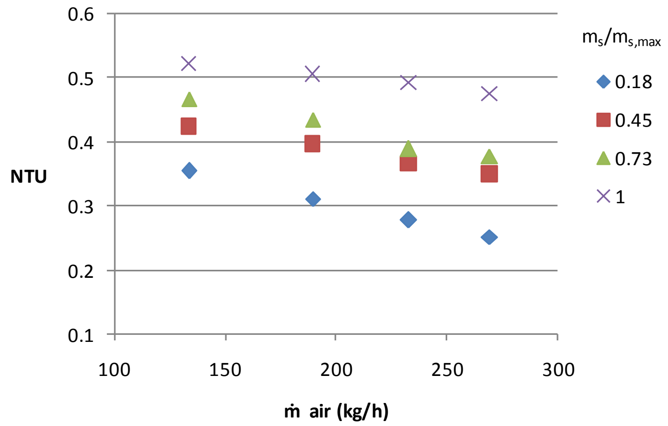

Figure 9 NTU vs. air mass flow (water spray mass variation)

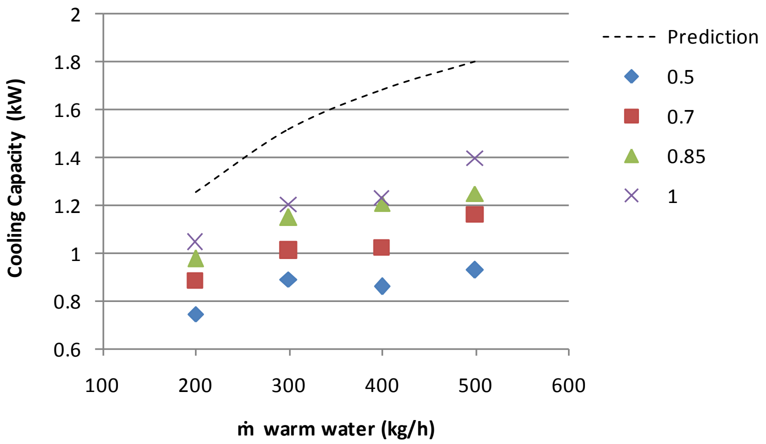

Figure 10 Variation rate of cooling capacity with warm water mass and water spray

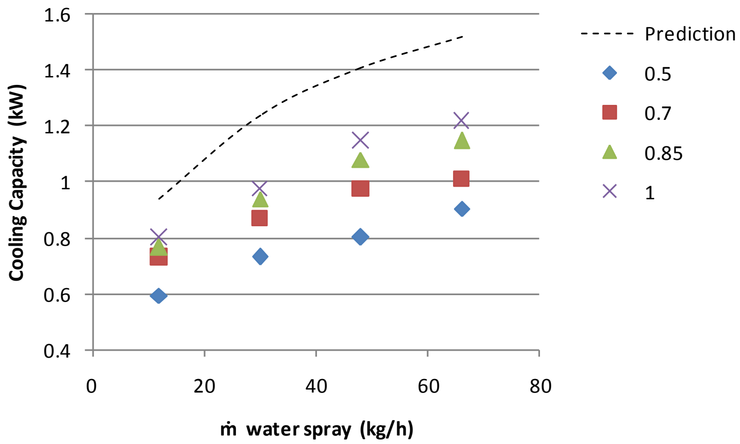

Figure 11 Cooling capacity vs. water spray mass rate (air mass rate variation)

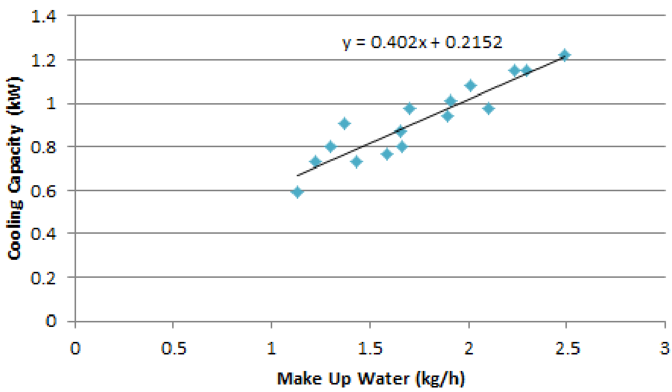

Figure 12 Cooling capacity vs. make-up water

4.5. Cooling Capacity

Figure 11 shows that the value of the cooling tower exhaust heat varies with the amount of hot water flow. Although Figure 7 shows that the less warm water that is cooled, the lower is the effectiveness value. Figure 11 shows that the larger the mass flow of warm water is, the cooling capacity of the cooling tower will be even greater. The graph also plots predictive value calculations using a water spray mass flow ratio of 1. Experiments show the value of cooling capacity is smaller than predicted. However, it can be seen from the graph that the plotted theoretical line (calculations) and the experiment have the same tendency.

The hot exhaust value of the cooling tower varied with the amount of water spray flow. Figure 11 shows that the mass flow of water spray and air mass is directly proportional to the cooling capacity. The prediction also suggests a value greater than the experimental data. Figure 12 shows that the cooling capacity is directly proportional to the volume of make-up water (calculated by $m_e = m_a\ (Xo - Xi\ )$, which is also the evaporated water spray mass. Theoretically, the greater the value of make-up water is, the more it will increase the cooling capacity due to the latent heat of evaporated water.

In Figure 13, the heating capacity value of the cooling tower obtained from the experiments, is evaluated by using the Zukauskas analogy to predict the magnitude of the maximum value of latent heat that occurs on the surface of the heat exchange coil arrangement. The capacity value of waste heat in the experiment is only 50–60% of the theoretical value, which assumes that evaporation occurs all over the coil surface that is coated with a very thin layer of water.

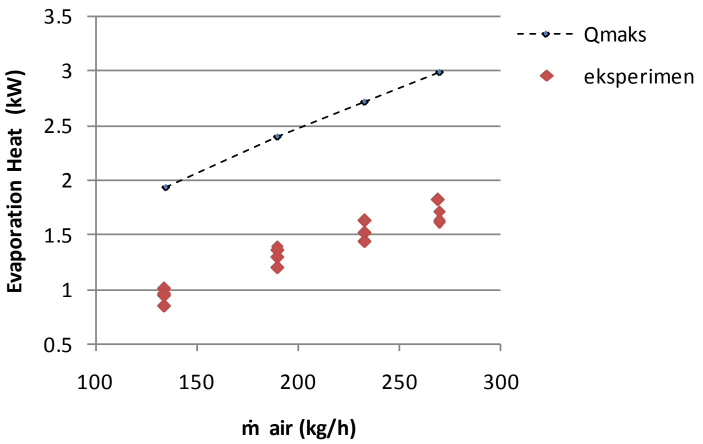

Figure 13 Heat evaporation experiments and predictions based on maximum latent heat

Values that deviate from the theoretical calculations with the experimental data are because they do not meet this assumption. Because of simplifications used to calculate the correlation and because not all of the coil surface was coated by water spray, less evaporation occured. If the heating value of evaporation experiments in Figure 13, compared with the cooling capacity values in Figures 10 and 11, it is seen that the cooling capacity value should be greater. Heat absorption due to the latent heat of evaporation that occurs is not entirely used for discharging heat from warm water, but in fact it can be used as a coolant on the air side.

This study is intended to provide an overview of the performance characteristics of a closed system cooling tower, but cooling specifications of the cooling tower are presented in the experimental data. In units of refrigeration tons, the maximum waste heat value from the miniature cooling tower is 0.4 RT. Using variations of air mass rates and warm water, the RT value can be seen in Figure 14.

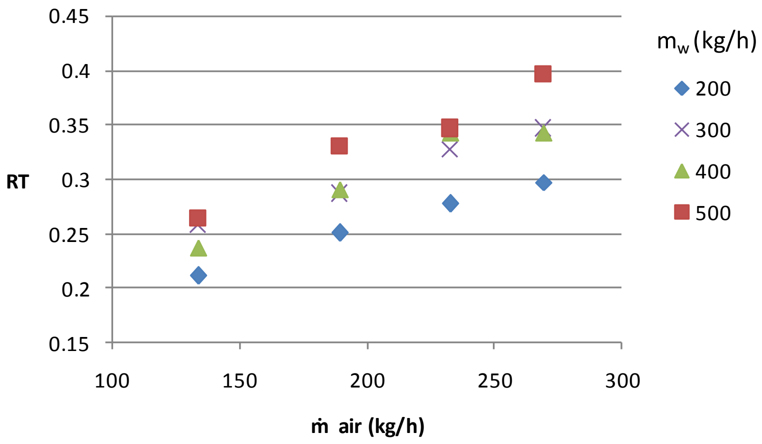

Figure 14 Cooling capacity in RT

4.6. Transfer Phenomena

The heat transfer coefficient value can be plotted on a graph by using Equation 2, while Eq. 3 determines the mass transfer coefficient.

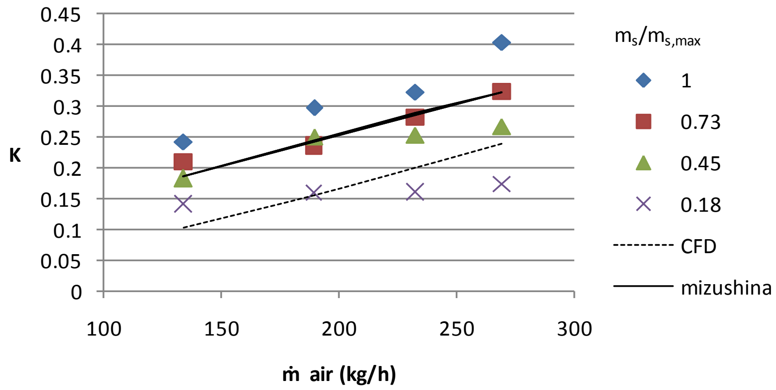

Figure 15 Mass transfer coefficient

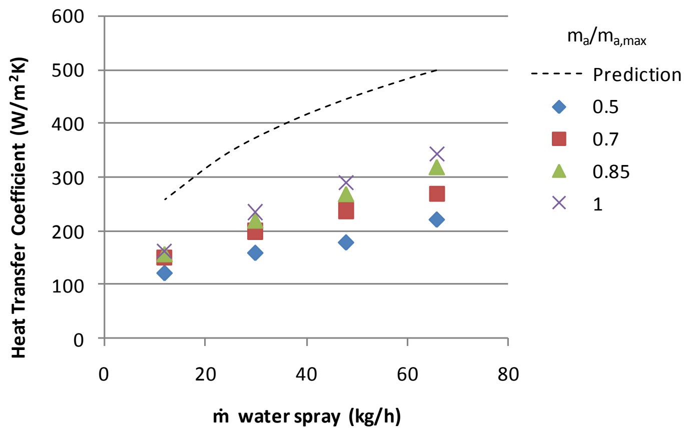

Figure 16 Heat transfer coefficient vs. water spray mass rate

Mizushina et al. (1967; 1968) developed a correlation for the mass transfer coefficient based on their experimental tests of four staggered bundles having a pitch of $2D_o$ with diameters of 12.7, 19.05, and 40 mm, respectively. The data indicate dependence between spray mass flow rate and diameter. Their data spanned 50 < Re_{spray} < 240 and 1.2 < $Re_{air} \times 10^3$ < 14. The correlation may be written in form of:

$$\mathrm{K(og)} = \mathrm{NTU} = \int_{H2}^{H1} \frac{dH}{H' - H} = \sum \frac{\Delta H}{H' - H} \tag{6}$$

In Figure 15, the experimental data results are compared with the correlation shown in Equation 6 and the CFD simulation. The graph shows that there are deviations in the experimental data with related predictions. The *K* value from Equation 6 approaches the experimental value. This differs from the indicated value produced by the CFD simulations which gives a smaller value of *K*, because it is difficult to describe the appropriate boundary conditions for modeling evaporation in the cooling tower. In the CFD simulation, the evaporation is greater and water droplets created for water spray (discrete phase) become smaller. In the simulation, the diameter of the spray droplets was 1 mm.

The value of *K* is directly proportional to the magnitude of air mass rates. For small water spray mass flows, the *K* value gradient increase tends to be small. The increase in the *K* value gradient is also proportional to the magnitude of water spray mass rate. In Figure 16, the data obtained from the experiment is compared with theoretical calculations to compute the value of Uo. The graph shows that there is a deviation of experimental data with related predictions, even though it appears that there is an increase proportional to an increase of water spray mass.

5. CONCLUSION

The conclusion that can be derived from this study is that an increase in the amount of water spray mass flow causes an increase in the effectiveness value, heat transfer and overall mass transfer, as well as the cooling capacity of the cooling tower. Furthermore, there is a minimum value for the water spray mass flow rate, where if it falls below a certain level then the value of heat and mass transfer is not significant.

In direct evaporative cooling, waste heat typically utilizes up to 80% of the latent evaporation heat, and 20% of sensible air heat, however, waste heat in the closed system cooling tower utilizes 100% of the latent heat of evaporation. The mass transfer coefficient rate tends to be stable for a small mass of water spray.

6. NOMENCLATURE

A	area (m^2)
C	specific heat capacity of moist air (J /(kg.K)
CWCT	closed wet cooling tower
D	tube diameter (m)
d	tube outside diameter (m)
D_{ab}	diffusivity (m2/s)
f	friction factor
G	(kg/m.s)
h	enthalpy (J/kg)
h′	saturated enthalpy (J/kg)
h_{fg}	latent heat of evaporation of water (J/kg)
H	humidity ratio (kg water/kg dry air)
H′	saturated humidity (kg water/kg dry air)
HMTU	heat and mass transfer units
k	thermal conductivity (W/m.K)
λ	thermal conductivity of tube wall (W/mK)
K	mass transfer coefficient (kg/m^2s)
L	tube length (m)

m	mass flow rate (kg/s)
NTU	Number of Transfer Units
Nu	Nusselt number
Nu_D	mean Nusselt number for a tube
p	pressure (Pa)
q	heat flux (W/m^2)
Q	Heat transfer rate (W)
Re	Reynolds Number
Re_D	mean Reynolds number for a tube
RH	Relative Humidity
RT	Refrigeration Tons
t	dry bulb temperature (^oC)
tw	wet bulb temperature (^oC)
α	convective heat transfer coefficient (W/m^2K)
α_m	convective mass transfer coefficient (m/s)
ρ	density (kg/m^3)
ε	effectiveness

Subscript

a	cold air
w	hot water
s	spray water
t	tube
sat	saturation
i	inside, inlet
o	outside, outlet
e	evaporation

7. REFERENCES

Duan, Z., Zhan, C., Zhang, X., 2012. Indirect Evaporative Cooling: Past, Present and Future Potentials. *Renewable and Sustainable Energi Reviews*, Volume 16, pp. 6823–6850

Facão, J., Oliveira, A., 2004. Heat and Mass Transfer Correlations for the Design of Small Indirect Contact Cooling Towers. *Applied Thermal Engineering*, Volume 24, pp. 1969–1978

Gan, G., Riffat, S.B., Shao, L., Doherty, P., 2001. Application of CFD to Closed-wet Cooling Towers. *Applied Thermal Engineering*, Volume 21, pp. 79–92

Hasan, A., 2005. Performance Analysis of Heat Transfer Processes from Wet and Dry Surfaces: Cooling Towers and Heat Exchangers. *Ph.D. Dissertation*, Helsinki University of Technology

Hasan, A., Siren, K., 2002. Theoretical and Computational Analysis of Closed Wet Cooling Towers and its Applications in Cooling Buildings. *Energy and Buildings*, Volume 34, pp. 477–486

Heyns, J.A., Kröger, D.G., 2010. Experimental Investigation into the Thermal-flow Performance Characteristics of an Evaporative Cooler. *Applied Thermal Engineering*, Volume 30, pp. 492–498

Jiang, J-J., Liu, X-H., Jiang, Y., 2013. Experimental and Numerical Analysis of a Cross-flow Closed Wet Cooling Tower. *Applied Thermal Engineering*, Volume 61, pp. 678–689

Kaiser, A.S., Lucas, M., Viedma, A., Zamora, B., 2005. Numerical Model of Evaporative Cooling Processes in a New Type of Cooling Tower. *International Journal of Heat and Mass Transfer*, Volume 48, pp. 986–999

Khan, J.U.R., Yaqub, M., Zubair, S.M., 2003. Performance Characteristics of Counter Flow Wet Cooling Towers. *Energy Convers Manage*, Volume 44, pp. 2073–2091

Kloppers, J.C., Kröger, D.G., 2005. Cooling Tower Performance Evaluation: Merkel Poppe and e-NTU Methods of Analysis. *ASME J Eng Gas Turbin Power*, Volume 127, pp. 1–7

Merkel, F., 1925. Verdunstungskühlung. *VDI-Zeitchrift*, Volume 70, pp. 123–128

Mizushina, T., Ito, R., Miyashita, H., 1967. Experimental Study of an Evaporative Cooler. *International Chemical Engineering*, Volume **7**(4), pp. 727–732

Mizushina, T., Ito, R., Miyashita, H., 1968. Characteristics and Methods of Thermal Design of Evaporative Cooler. *International Chemical Engineering*, Volume 8(3), pp. 532–538

Papaefthimiou, V.D., Rogdakis, E.D., Koronaki, I.P., Zannis, T.C., 2012. Thermodynamic Study of the Effects of Ambient Air Conditions on the Thermal Performance Characteristics of a Closed Wet Cooling Tower. *Applied Thermal Engineering*, Volume 33–34, pp. 199–207

Poppe, M., Rögener, H., 1991. Berechnung von Rückkühlwerken. *VDI-Wärmeatlas*, Mi1-Mi, p. 15

Riffat, S.B., Oliveira, A., Facão, J., Gan, G., Doherty, P., 2000. Thermal Performance of a Closed Wet Cooling Tower for Chilled Ceilings: Measurement and CFD Simulation. *Int. J. Energy Res*, Volume 24, pp. 1171–1179

Sarker, M.M.A., Kim, E., Moon, C.G., Yoon, J.I., 2008. Performance Characteristics of the Hybrid Closed Circuit Cooling Tower. *Energy and Buildings*, Volume 40, pp. 1529–1535

Shim, G.J., Baek, S.M., Moon, C.G., Lee, H.S., 2008. Performance Characteristics of a Closed Circuit Cooling Tower with Multi Path. *Heat Transfer Engineering*, Volume 31, pp. 992–997

Stabat, P., Marchio, D., 2003. Simplified Model for Indirect-contact Evaporative Cooling-tower Behaviour. *Applied Energ*y, Volume 78, pp. 433–451

Xia, Z.Z., Chen, C.J., Wang, R.Z., 2011. Numerical Simulation of a Closed Wet Cooling Tower with Novel Design. *International Journal of Heat and Mass Transfer*, Volume 54, pp. 2367–2374

Zheng, W-Y., Zhu, D-S., Song, J., Zeng, L-D., Zhou, H-j., 2012. Experimental and Computational Analysis of Thermal Performance of the Oval Tube Closed Wet Cooling Tower. *Applied Thermal Engineering*, Volume 35, pp. 233–239