Chapter Four { TC "Chapter Four " \l 1 }

Modeling of components

4.1 Cooling Tower{ TC "4.1 Cooling Tower" \l 2 }

4.1.1 Basic Heat and Mass Transfer in a Cooling Tower{ TC "4.1.1 Basic Heat and Mass Transfer in a Cooling Tower" \l 3 }

Heat and mass transfer in a cooling tower is by evaporation and convection. The cold entering air takes up moisture and heat as it is in contact with the water droplets. Radiation heat transfer is negligible in a cooling tower.

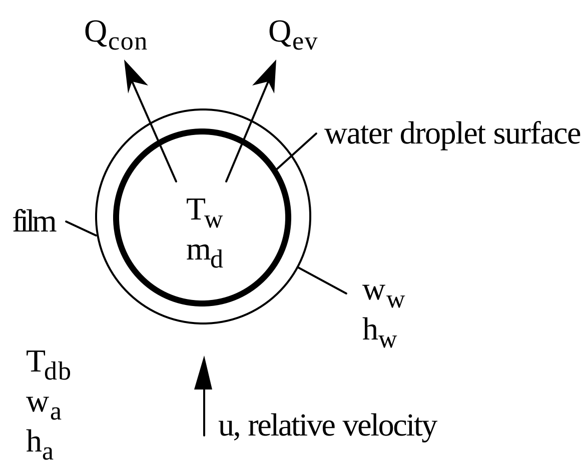

Figure 4.1 Heat and Mass Transfer from a Water Droplet{ TC "Figure 4.1 Heat and Mass Transfer from a Water Droplet" \l 5 }

An energy balance around a single water droplet yields (see figure 4.1) [8]:

$$-(Q_{ev} + Q_{con}) = \frac{dU_i}{dt} \qquad \text{(Equ. 4.1)}$$

U_i is the internal energy of the water droplet and can be computed by

$$dU_i = m_d \cdot c_{pw} \cdot dT_w \qquad \text{(Equ. 4.2)}$$

if the droplet mass and specific heat are constant. The evaporation water loss is considered small and neglected. The evaporation and convection terms can be further specified by basic heat and mass transfer correlations

$$Q_{con} = h_c \cdot a \cdot (T_w - T_{db}) \qquad \text{(Equ. 4.3)}$$

$$Q_{ev} = K \cdot a \cdot (w_w - w) \cdot h_{fg} \qquad \text{(Equ. 4.4)}$$

where w_w is the saturation value of the humidity ratio for air at the local water temperature T_w and w is the humidity ratio of the local air, that surrounds the droplet. T_w and T_{db} describe the local water and air dry bulb temperature, respectively. The surface area of a single droplet is denoted by a, h_c is the heat transfer coefficient and K the mass transfer coefficient. The mass transfer coefficient is related to the heat transfer coefficient via the Lewis number [3]:

$$Le = \frac{h_c}{K \cdot c_p} \qquad \text{(Equ. 4.5)}$$

where c_p is the specific heat of the air vapor mixture. The specific heat of air is assumed constant through the calculation. A Lewis number of unity is commonly assumed for further calculation [4], [5].

If m_d is replaced by $\rho_w \cdot V_d$ where V_d is the volume of a droplet, equation 4.1 can be replaced by combining equations 4.2 through 4.5 by

$$\frac{dT_w}{dt} = -h_c \cdot \frac{\left[(T_w - T_{db}) + \frac{1}{c_p} \cdot (w_w - w) \cdot h_{fg}\right]}{\frac{\rho_w \cdot c_{pw} \cdot V_d}{a}} \qquad \text{(Equ. 4.6)}$$

There are several correlations to calculate the heat transfer coefficient of a droplet moving through a gas, air in this case. One commonly used relation is the Ranz-Marshall correlation that yields the Nusselt number for a droplet [6],[7].

$$Nu_D = 2 + 0.6 \cdot (Re_D)^{\frac{1}{2}} \cdot (Pr)^{\frac{1}{3}} \qquad \text{(Equ. 4.7)}$$

where Re_D is the Reynolds number

$$Re_D = \frac{u \cdot D}{\nu} \qquad \text{(Equ. 4.8)}$$

and Pr is the Prandtl number

$$Pr = \frac{\mu \cdot c_p}{k} \qquad \text{(Equ. 4.9)}$$

The droplet velocity relative to the air is u. Air properties are the dynamic viscosity μ, the kinematic viscosity ν and the conductivity k, evaluated at a mean film temperature. The heat transfer coefficient can be calculated from the Nusselt number

$$Nu_D = \frac{h_c \cdot D}{k} \qquad \text{(Equ. 4.10)}$$

The equilibrium in this process is reached when the amount of energy transfer by evaporation is equal to the negative amount of energy transfer by convection, or

$$Q_{ev} = -Q_{con}$$

Under adiabatic conditions equilibrium is reached when the local water temperature is equal to the local thermodynamic wet bulb temperature. The equilibrium also defines the limiting temperature the water in a cooling tower can be cooled down to. It is not possible to cool the water below the wet bulb temperature of the entering air [8].

The analysis of heat and mass transfer of a single droplet is useful to understand the general heat and mass transfer mechanism, but to examine a whole cooling tower a slightly different approach is necessary. Braun et al. [9] gives a derivation for the heat and mass transfer mechanism in a counterflow cooling tower. As the derivation for a counterflow tower is more evident than for a crossflow tower the derivation is shown for a counterflow tower in the following. The cooling towers at the Columbia Station are of crossflow type, thus a simple adaption will be made to apply the algorithm to a crossflow tower.

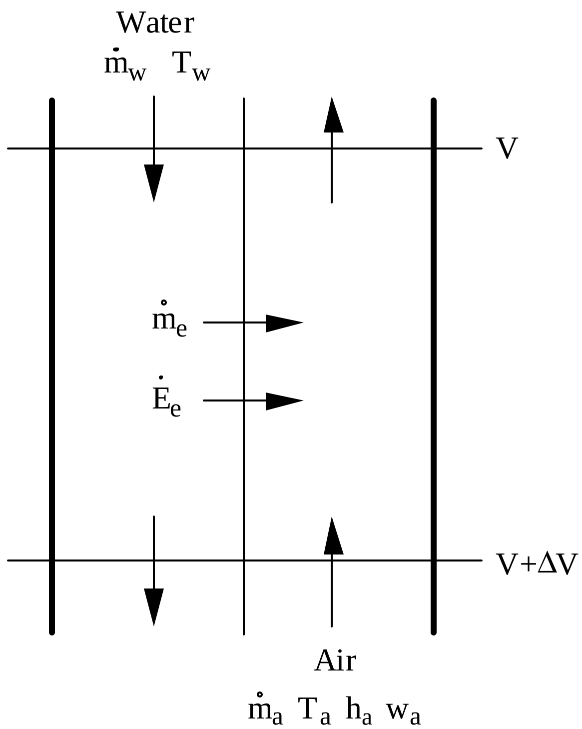

Figure 4.2 Mass and Energy Flows for a Section **{** TC "Figure 4.2 Mass and Energy Flows for a Section " \l 5 **}**;

of a Crossflow Cooling Tower

A control volume is selected that includes a defined volume of the cooling tower fill, see figure 4.2. A mass balance is set up, that relates the mass of water evaporated to the moisture increase of the air stream.

$$\left(\dot{m}_w\right)_V + \left(\dot{m}_a \cdot w\right)_{V+dV} - \left(\dot{m}_w\right)_{V+dV} - \left(\dot{m}_a \cdot w\right)_V = 0 \qquad \text{(Equ. 4.11)}$$

Dividing by dV and passing to the limit yields a differential equation for the water mass flows as functions of distance through the tower, which is represented by volume.

$$\frac{dm_w}{dV} = \dot{m}_a \cdot \frac{dw}{dV} \qquad \text{(Equ. 4.12)}$$

Integrating this expression from the top end of the tower where the water enters to any position in the tower yields an overall mass balance relation. At the top of the tower the water mass flow rate is the entering value and the humidity ratio is the one of exit air. The water mass flow rate at any position in the tower is given by:

$$\dot{m}_w = \dot{m}_{w,in} - \dot{m}_a \cdot (w_{out} - w) \qquad \text{(Equ. 4.13)}$$

Similarly to the mass balance, an energy balance can be established.

$$c_{pw} \cdot \left(\dot{m}_w \cdot T_w\right)_V + \dot{m}_a \cdot (h_a)_{V+dV} - c_{pw} \cdot \left(\dot{m}_w \cdot T_w\right)_{V+dV} - \dot{m}_a \cdot (h_a)_V = 0$$

(Equ. 4.14)

The resulting differential equation relates the energy drop of the water flow to the energy rise of the air flow.

$$c_{pw} \cdot \frac{d\left(\dot{m}_w \cdot T_w\right)}{dV} = \dot{m}_a \cdot \frac{dh_a}{dV} \qquad \text{(Equ. 4.15)}$$

$\dot{m}_w$ varies with position according to equation 4.13. The left side of equation 4.15 is then differentiated by parts. The expression for the local water flow rate, equation 4.13, is substituted into the resulting expression. The resulting relation relates the change in water temperature with distance to the air enthalpy and humidity ratio changes.

$$\frac{dT_w}{dV} = \frac{\dot{m}_a \cdot \left(\frac{dh_a}{dV} - c_{pw} \cdot T_w \cdot \frac{dw}{dV} \right)}{c_{pw} \cdot \left(\dot{m}_{w,in} - \dot{m}_a \cdot (w_{out} - w) \right)} \qquad \text{(Equ. 4.16)}$$

Mass and energy balances for the air and water streams are performed separately to bring in the overall heat and mass transfer coefficients. The conservation of mass for the air stream is applied to a control volume that includes only the air flow. It is important to notice, that the mass flow rate of dry air is constant through the tower. The equation that relates the mass transfer rate of the evaporating water per unit volume to the mass transfer coefficient is

$$\dot{m}_e''' = \frac{h_c}{c_p} \cdot A''(w_w - w) \qquad \text{(Equ. 4.17)}$$

A'' is the heat transfer area per unit volume of the fill. Again the assumption of a Lewis number equal to unity was made to obtain a relation between heat and mass transfer. The mass balance for the water vapor in the air stream is then

$$\dot{m}_a \cdot \frac{dw}{dV} = -\frac{h_c}{c_p} \cdot A''(w_w - w) \qquad \text{(Equ. 4.18)}$$

A variable, the number of transfer units, Ntu, is introduced. In this context the number of transfer units is based on the air flow rate. The Ntu is defined here as

$$Ntu = \frac{h_c \cdot A'' \cdot V}{\dot{m}_a \cdot c_p} \qquad \text{(Equ. 4.19)}$$

Equation 4.18 can then be written as

$$\frac{dw}{dV} = -\frac{Ntu}{V} \cdot (w_w - w) \qquad \text{(Equ. 4.20)}$$

For the same control volume the energy balance can be derived in a similar manner to that for the mass balance. The energy flow per unit volume associated with the evaporating water flow is given in terms of the equivalent mass transfer coefficient and potential for energy transfer (enthalpy difference) as:

$$\dot{E}_e'' = \frac{h_c}{c_p} \cdot A'' \cdot (h_w - h_a) \qquad \text{(Equ. 4.21)}$$

where h_w is the air enthalpy at a saturated state at the local water temperature, as mentioned above and $\dot{E}_e'''$ is the amount of energy transferred per unit volume of the fill. $\dot{E}_e'''$ can be written as

$$\dot{E}_e''' = -\dot{m}_a \cdot \frac{dh_a}{dV} \qquad \text{(Equ. 4.22)}$$

The energy balance can be rewritten using the Ntu definition and equation 4.21:

$$\frac{dh_a}{dV} = -\frac{Ntu}{V} \cdot (h_w - h_a) \qquad \text{(Equ. 4.23)}$$

Equations 4.20 and 4.23 describe the heat and mass transfer processes inside the tower. The set of equations given above is not subject to an analytical mathematical solution. They rather reflect the mass and energy balance at any point in the tower. Different approaches have been employed to obtain relations for the overall performance of a tower. One approach is to numerically integrate the relations [5]. This is useful for determining the performance of a specific tower for a given set of parameters and operating conditions, but provides no generality in the results. For the scope of this work, a cooling tower model is needed, that provides explicit performance data for a given set of operation conditions. A different approach will be shown in the next section that utilizes the analogy between a cooling tower and a sensible heat only heat exchanger [9].

4.1.2 Algorithm for tower performance calculation, the cooling tower - { TC "4.1.2 Algorithm for tower performance calculation, the cooling tower - " \l 3 }

heat exchanger analogy approach

For the development of a simplified method for evaluating cooling tower performance the assumption is made, that the water mass flow rate is constant throughout the tower. This assumption is reasonable, as the amount of evaporated water is usually in the range of 2 to 5% of the total water flow [4]. To mitigate the error associated with this assumption, at the end of the analysis the evaporation of water will be partially included. If the water loss is neglected the overall energy balance equation 4.16 has the following form

$$\frac{dT_w}{dV} = \frac{\dot{m}_a}{\dot{m}_w \cdot c_{pw}} \cdot \frac{dh_a}{dV} \qquad \text{(Equ. 4.24)}$$

To eliminate the temperature T_w from equation 4.23, an effective specific heat is introduced. This allows the water temperature T_w to be replaced with the saturated air enthalpy h_w at the water temperature T_w. The effective specific heat is then

$$c_s = \left(\frac{dh_w}{dT_w}\right)_{sat} = \left(\frac{h_{w,in} - h_{w,out}}{T_{w,in} - T_{w,out}}\right) \qquad \text{(Equ. 4.25)}$$

The effective specific heat c_s is the change in enthalpy with temperature along the saturated air line, see figure 4.3. The enthalpies $h_{w,in}$ and $h_{w,out}$ represent the air enthalpies at saturation at the water in- and outlet temperature, respectively. Evaluating the effective specific heat requires iteration as the outlet water condition is not known at the beginning of the calculation. Combining equations 4.24 and 4.25 the energy balance can be written in terms of enthalpies

$$\frac{dh_w}{dV} = \frac{\dot{m}_a \cdot c_s}{\dot{m}_w \cdot c_{pw}} \cdot \frac{dh_a}{dV} \qquad \text{(Equ. 4.26)}$$

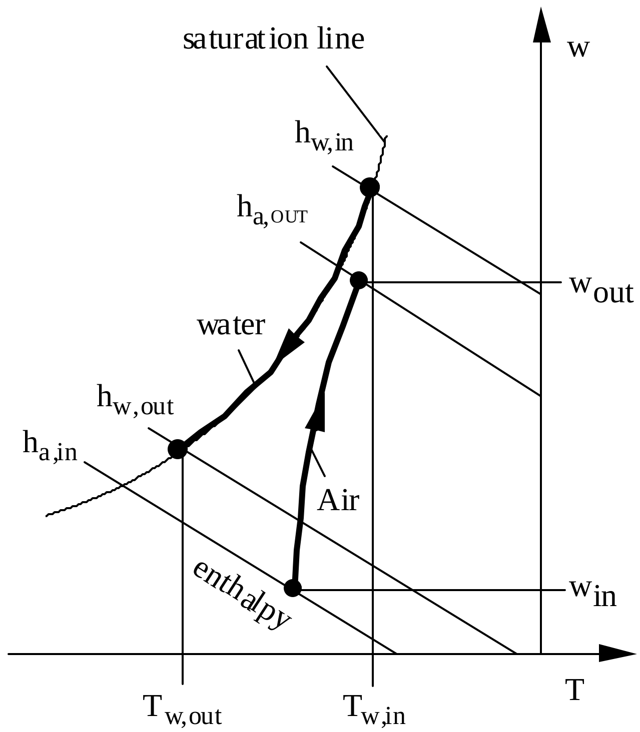

Figure 4.3 Process Diagram for a Cooling Tower{ TC "Figure 4.3 Process Diagram for a Cooling Tower" \l 5 }

Similar to a sensible heat only heat exchanger an equivalent capacitance rate is defined

$$m^* = \frac{\dot{m}_a \cdot c_s}{\dot{m}_w \cdot c_{pw}} \qquad \text{(Equ. 4.27)}$$

The two equations that describe the heat and mass transfer in a cooling tower are now equations 4.23 that describes the air side energy balance in terms of the overall heat transfer coefficients and equation 4.26 that gives the overall energy balance for the cooling tower. Equation 4.26 can be rewritten as

$$\frac{dh_w}{dV} = m^* \cdot \frac{dh_a}{dV} \qquad \text{(Equ. 4.28)}$$

and equation 4.23 is (from above)

$$\frac{dh_a}{dV} = -\frac{Ntu}{V} \cdot (h_w - h_a) \qquad \text{(Equ. 4.23).}$$

These two equations are analogous to those for a sensible heat only exchanger [9]. The only difference is that the temperatures in the heat exchanger equation are replaced by enthalpies. Due to this analogy, the effectiveness - Ntu relations that are developed originally for a heat exchanger are applied for a cooling tower. The total energy transfer in a cooling tower can be described in terms of the effectiveness. The effectiveness is defined with respect to the air inlet enthalpy and the equivalent saturated air enthalpy at the water inlet temperature. The overall energy transfer in the cooling tower then becomes

$$\dot{Q} = \varepsilon \cdot \dot{m}_a \cdot (h_{w,in} - h_{a,in}) \qquad \text{(Equ. 4.29)}$$

where ε is the effectiveness of the tower. Effectiveness is always the air side effectiveness. From the effectiveness - Ntu model the effectiveness for a crossflow cooling tower is derived as [3]

$$\varepsilon = \left(\frac{1}{m^*}\right) \cdot \left(1 - \exp\left\{-m^* \cdot [1 - \exp(-Ntu)]\right\}\right) \qquad \text{(Equ. 4.30)}$$

The air outlet condition, defined by its enthalpy, can be calculated by

$$h_{a,out} = h_{a,in} + \frac{\dot{Q}}{\dot{m}_a} \qquad \text{(Equ. 4.31)}$$

The total energy transfer is also the heat transfer from the water flow. An energy balance only on the water flow yields, if evaporation is included

$$\dot{Q} = c_{pw} \cdot \dot{m}_{w,in} \cdot T_{w,in} - c_{pw} \cdot \dot{m}_{w,out} \cdot T_{w,out} \qquad \text{(Equ. 4.32)}$$

Equation 4.31 and 4.32 together give the water outlet state. The same result can be accomplished by integrating equation 4.15. Equation 4.15 can be integrated directly without the assumption that the water flow rate is constant. The result of the integration is

$$c_{pw} \cdot \left(\dot{m}_{w,in} \cdot T_{w,in} - \dot{m}_{w,out} \cdot T_{w,out} \right) = \dot{m}_a \cdot \left(h_{a,out} - h_{a,in} \right) \qquad \text{(Equ. 4.33)}$$

Using the exact integration of the energy balance to yield the water outlet temperature corrects, in part, for the assumption made for deriving equation 4.24 that the water flow rate is constant. The water outlet temperature is then calculated to

$$T_{w,out} = \frac{\dot{m}_{w,in} \cdot T_{w,in}}{\dot{m}_{w,out}} - \frac{\dot{m}_a \cdot \left(h_{a,out} - h_{a,in} \right)}{\dot{m}_{w,out} \cdot c_{pw}} \qquad \text{(Equ. 4.34)}$$

The water outlet flow rate can be determined from equation 4.13, if the outlet air humidity ratio is known. The outlet air state is determined in terms of enthalpy. To find the exit air temperature and humidity ratio, the air side energy balance, equation 4.23, is integrated to find an effective surface enthalpy for the water flow. The effective surface enthalpy, h_{eff}, is the equivalent saturation enthalpy of a uniform temperature wet surface that yields the same enthalpy change of the air stream as the actual air stream. Equation 4.23 is modified for this purpose and $h_{s,eff}$ substituted for h_w. The resulting integral form is

$$\int_{out}^{in} \frac{dh_a}{\left(h_{s,eff} - h_a \right)} = \int_0^V \left(-\frac{Ntu}{V} \right) dV \qquad \text{(Equ. 4.35)}$$

The effective surface enthalpy is determined to

$$h_{s,eff} = h_{a,in} + \frac{\left(h_{a,out} - h_{a,in} \right)}{1 - \exp(-Ntu)} \qquad \text{(Equ. 4.36)}$$

Assuming there is the equivalent process of evaporation as in the real tower from this saturated surface, the exit humidity ratio is calculated by integration of equation 4.20. In a similar manner than shown for the effective enthalpy the exit humidity ratio is then calculated to

$$w_{out} = w_{s,eff} + \left(w_{in} - w_{s,eff}\right) \cdot \exp(-Ntu) \qquad \text{(Equ. 4.37)}$$

The effective humidity ratio w_{eff} corresponds to the air state at the effective enthalpy h_{eff}. The outlet water flow rate can then be determined from equation 4.13 if the air inlet and outlet conditions are known. Solution of the equations outlined above is not possible explicitly, but iteration is required, as mentioned for the calculation of c_s. The actual calculation will be performed using an EES or TRNSYS program.

4.1.3 Calibration of the Cooling Tower Model using measured data and { TC "4.1.3 Calibration of the Cooling Tower Model using measured data and " \l 3 } Validation of the Model

Considering the relations outlined in 4.1.2, the performance of a cooling tower can be determined if the mass flow rates, the ambient conditions, the water inlet temperature and the Ntu value are known. The Ntu value is tower specific and is defined by the physical appearance of the tower. The Ntu value is dependent on the heat transfer coefficient that is determined mainly by droplet size and droplet and air velocity (see equ. 4.7 to 4.9). Furthermore the Ntu value depends on the overall heat transfer surface area available per unit volume in the tower, the overall volume of the region in which heat transfer takes place and on the mass flow rate of air (see equ. 4.19).

It is usually not possible to determine the heat and mass transfer correlation from the physical appearance of the tower alone. To obtain a Ntu value from measurements it is necessary to correlate data for specific tower designs. Mass transfer data are usually correlated with the following form [5]

$$\frac{h_c \cdot A'' \cdot V}{\dot{m}_w} = c \cdot \left(\frac{\dot{m}_w}{\dot{m}_a}\right)^n \qquad \text{(Equ. 4.38)}$$

Multiplying both sides of the equation by the mass flow rate ratio $\dot{m}_w / \dot{m}_a$ and utilizing the definition for Ntu gives

$$Ntu = c \cdot \left[\frac{\dot{m}_w}{\dot{m}_a} \right]^{1+n} \qquad \text{(Equ. 4.39)}$$

Using the tower performance test outlined in section 3.2.4, the tower coefficients c and n can be determined. A minimum of two operation data points per tower is required. To determine the tower coefficients, the readily available TRNSYS cooling tower type was used. A more detailed description of the TRNSYS cooling tower model will be given in appendix A. If the tower coefficients are determined, the tower behavior can be simulated for each set of mass flow rates and ambient conditions.

An EES program of a cooling tower, following the relations outlined in sections 4.1.2 and 4.1.3, was written to perform calculations on cooling tower behavior. The EES program is also shown in Appendix A. This program can be used to determine the tower coefficients as well. If only data for two operation points is given, the coefficients can be calculated explicitly. The tower coefficients that are determined from the performance test data are shown in table 4.1.

Tower	c	n
A	2.266	-0.2567
B	3.850	-0.1966

Table 4.1 Tower coefficients{ TC "Table 4.1 Tower coefficients" \l 4 }

The values for parameter c are in the typical range that is between 0.5 and 5 [9]. The values for parameter n are slightly outside the typical range, which is between -0.35 and -1.1. But, as the measurement from section 3.2.4 is considered to be accurate enough for the scope of this work, the obtained tower parameters are used for the further analysis.

Validation of the tower model

For a three day period from July 22 0:00 to 25 12:00 1995 the calculated performance of a cooling tower is compared to measured performance data. As the plant measurement system only yields data for cooling tower A, the comparison could only be done for this tower. Water in- and outlet and drybulb temperatures are taken from plant data, while the wet bulb temperature is taken from weather station measurements. The water flow rates are determined from the number of tower pumps that were operated. The air flow rate was held constant at the average value calculated from performance test data. (see section 3.2). Calculations are performed using EES.

Figure 4.4 shows the calculated and measured water outlet temperatures. The number of pumps is also shown in the chart. In general the calculated and the measured temperature curve have the same shape. That means the general behavior of the towers is correctly predicted. The difference between calculated and measured temperatures is most of the time on the order of 1 to 2 F, which is acceptable considering the measurement errors associated with the data. One main source of deviation in the calculation may be that the local wetbulb temperature differs from the one measured at the weather station. Another possible source of error is that recirculation can occur in the cooling towers. Recirculation means that the exhaust air forms a vortex on the side of the tower, caused by wind blowing over the exhaust hood, and the warm and humid exhaust air enters the tower again. If recirculation occurs, the cooling capacity of a tower is decreased as the actual wetbulb temperature of the inlet air is higher than the ambient wetbulb. Recirculation is hard to predict only by using weather data, so that the predicted tower performance can be better than the actual observed even if the ambient conditions are correctly measured. The overprediction of tower performance occurs because the model uses the ambient wetbulb temperature, which is lower than the actual inlet air wetbulb temperature in the case of recirculation. When the time from hour 70 to 80 in figure 4.4 is considered, it is possible that recirculation occurred in this time period, as the predicted outlet temperature is up to 5 F lower than the measured temperature. Recirculation would be a possible explanation for this difference.

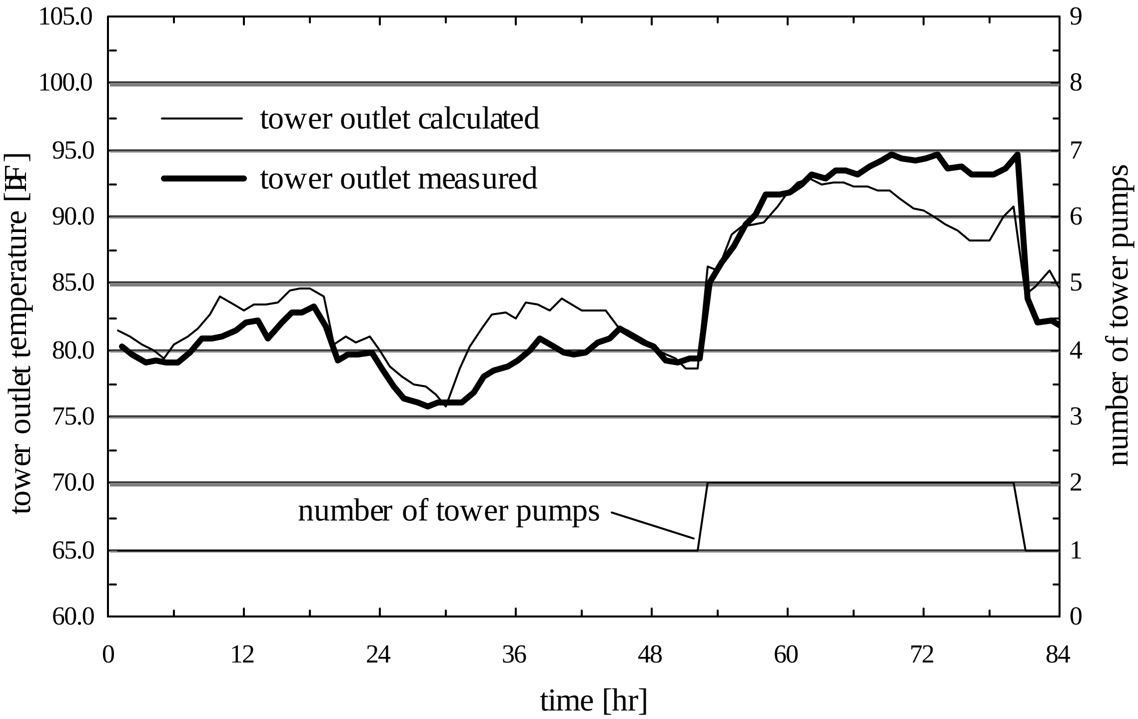

Figure 4.4 Comparison of calculated vs. measured tower outlet temperatures **{** TC "Figure 4.4 Comparison of calculated vs. measured tower outlet temperatures " \l 5 **}**;

for the Period of July 22 0:00 to July 25 12:00 1995;

In general the tower model predicts the tower performance in relatively good agreement with measurements, in the boundaries of measurement accuracy, that is associated with the on-line plant measurements as well as with the cooling tower performance test that was used to calibrate the model. Thus the tower coefficients that were determined in section 4.1.2 will be used in further calculations. It is assumed that the tower coefficients fit equally well for tower B, too.

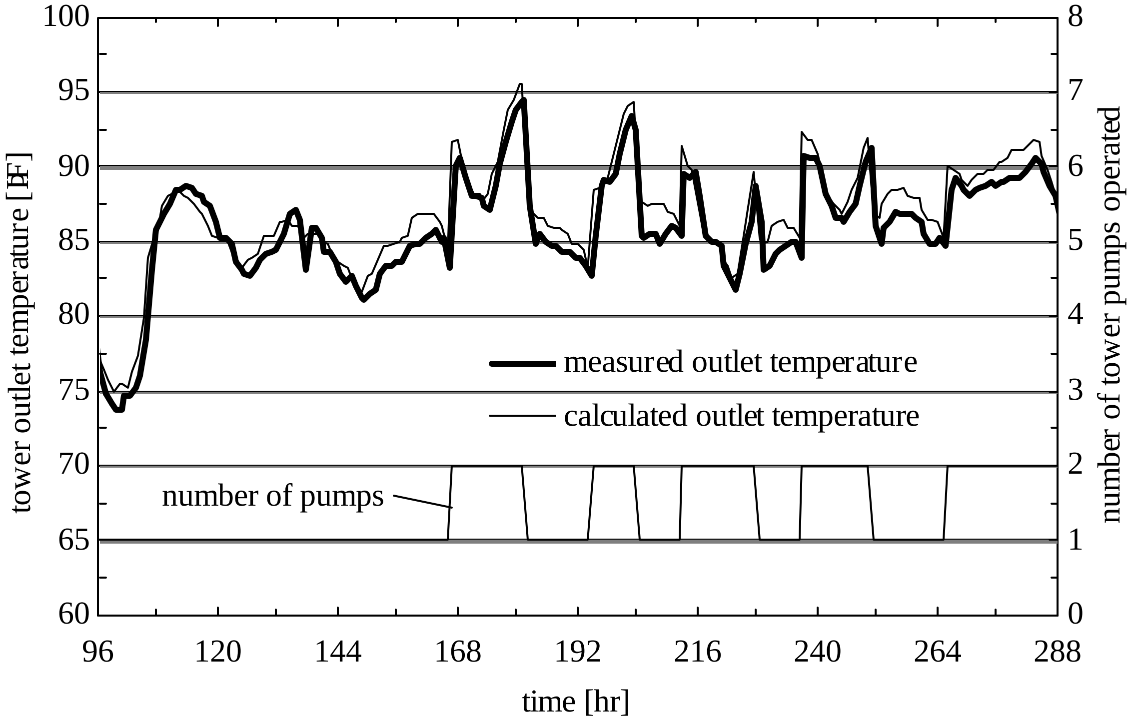

Figure 4.5 Measured versus calculated cooling tower outlet temperature and { TC "Figure 4.5 Measured versus calculated cooling tower outlet temperature and " \l 5 }; number of cooling tower pumps operated

Another comparison between measured and calculated tower outlet temperatures was made for a data set from June 14 0:00 to June 23 0:00 1994. That was before the fill and the fans of the towers were upgraded (see section 2). The wetbulb temperature is again taken from weather station measurements. Figure 4.5 shows the calculated and measured cooling tower outlet temperatures. In general the calculated temperature is close to the measured temperature. The difference is always in the order of 1 F to 2 F. The shapes of the measured and calculated curves show good agreement. But the calculated temperature is almost always higher than the measured temperature. This can be due to the change of the fill of the tower and a changed air flow rate caused by the fan upgrade. If the observed temperature difference is due to the tower upgrade, it means that the upgrade actually degraded the tower, as the calculated tower outlet temperature is higher than the measured and the model was calibrated using data from after the upgrade. But some care must be taken in making such a conclusion, as the temperature differences are small and can be due to measurement errors as was discussed for the July 1995 data above or to errors associated with the calculation itself, as the model is based on a mathematical model that was derived by making simplifying assumptions.

The number of pumps operated is shown in figure 4.4 and figure 4.5. It can be seen that in general, if the flow rate is changed at all, two pumps are operated during the night and one pump during the day, corresponding to the control strategy mentioned in chapter 2. Both the time axis of figures 4.4 and 4.5 start at midnight hours. The times for switching the number of pumps varies from day to day, dependent on the weather conditions.

4.1.4 Cooling Tower Characteristics{ TC "4.1.4 Cooling Tower Characteristics" \l 3 }

Figures 4.4 and 4.5 show that the tower outlet temperature drops by 5 to 10 F every time the flow rate is reduced. Thus it becomes obvious that reducing the flow rate leads to colder tower outlet water. But, on the other hand, if the flow rate is reduced the cooling effect of the towers on the overall cooling water circuit decreases as a smaller fraction of the water is cooled down in the towers. It is interesting to look at the overall energy rejection in a cooling tower for different flow rates and ambient conditions.

Table 4.1 shows water outlet temperatures and heat transfer rates for different operation conditions. The inflow temperature is held constant at 110 F, while wet- and drybulb temperature and flow rate are changed. The column containing the water flow rate fraction contains 50 and 100% flow ratios, which denotes one and two pump operation, respectively. Calculations are done using EES with coefficients for tower A.

It can be seen that every time the flow rate is reduced, the water outlet temperature drops considerably. On the other hand the overall heat rejection rate is higher for higher water flow through the tower. This may be explained with an at average higher temperature difference between the water and surrounding air within the tower. A higher heat transfer rate at higher flow rate means that although the water outlet temperature is higher, more energy is rejected to the environment. This is an interesting observation as this behavior suggests that there is an optimum operation point for the towers if the overall performance of the cooling cycle is considered. There has to be a trade off between a higher energy rejection rate that can lead to higher turbine efficiency on the one hand and higher auxiliary power consumption associated with a higher tower water flow rate on the other hand. The effect of changing tower flow rates on the overall performance of the cooling cycle in combination with the cooling pond will be examined in chapter 5.

water in [F]	water out [F]	heat transf. [10^8 Btu/hr]	drybulb [F]	wetbulb [F]	% flow [%]	water loss [%]
110	77.45	1.42	70	60	50	2.6
110	85.88	2.03	70	60	100	1.9
110	77.84	1.41	80	60	50	2.8
110	86.20	2.02	80	60	100	2.1
110	82.66	1.20	80	70	50	2.3
110	89.46	1.74	80	70	100	1.7

Table 4.1 Water outlet temperature and heat transfer rate for **{** TC "Table 4.1 Water outlet temperature and heat transfer rate for " \l 4 **}**

different operation conditions

Table 4.1 gives tower performance for different dry- and wetbulb temperatures. It can be seen, that the drybulb temperature only slightly changes the tower performance. Although the drybulb temperature changes by 10 F, the water outlet temperature and heat transfer rate remain nearly the same. The driving potential for heat and mass transfer in a cooling tower is the enthalpy difference between saturated air at water inlet temperature and the enthalpy of the entering ambient air (equation 4.29). As the air enthalpy depends mainly on the wetbulb temperature and is only a weak function of the drybulb temperature, it is evident that changing the drybulb temperature does not change the tower performance significantly. Thus the sum of evaporation and convection heat transfer remains nearly constant for a constant wet bulb temperature.

The last two rows of table 4.1 show the effect of a changing wetbulb temperature. A higher wetbulb temperature leads to significantly higher water outlet temperatures and lower heat transfer rates. It becomes obvious that the wetbulb temperature has a major impact on cooling tower performance, while the drybulb temperature is not significant if only overall heat transfer rates are considered. Therefore there will not be a big error involved if the drybulb temperature from the airport weather station is used instead of the drybulb temperature from the plant measurement.

The estimated amount of water loss due to evaporation losses is also shown in table 4.1. The amount of losses does not exceed 3 %. It can be seen that at constant wetbulb temperature a higher drybulb temperature leads to higher evaporation losses, although the overall heat transfer is not affected. This is due to a high potential for mass transfer if the humidity ratio of the entering air is low. For higher water flow rates the relative amount of water losses decreases which is due to fact that the air approaches a saturated state faster if more water is pumped through the towers. Therefore the air can not absorb a proportionally higher amount of moisture for higher water flow rates, although the absolute amount of water losses is higher for higher flow rates.

4.1.5 Comparison of performance against manufacturers data and { TC "4.1.5 Comparison of performance against manufacturers data and " \l 3 }

comparison of towers A and B

The manufacturer of the cooling tower, The Marley Company, provided performance curves for the cooling towers that give their original design performance. Figures 4.6 a and 4.6 b show a comparison between manufacturers data and the performance of the actual tower, calculated using the EES program used before. The charts show water outlet temperatures vs. wetbulb temperature for different ranges. The range is the difference between water inlet and outlet temperature. Both curves are calculated for tower A, while the performance data from the manufacturer treats both towers as equal. Figure 4.6 a shows tower performance for 100% design flow, which is 92,500 gpm. Figure 4.6 b shows performance curves for 60% design flow. 60% flow was chosen as this is the lowest flow rate for which manufacturer data is available.

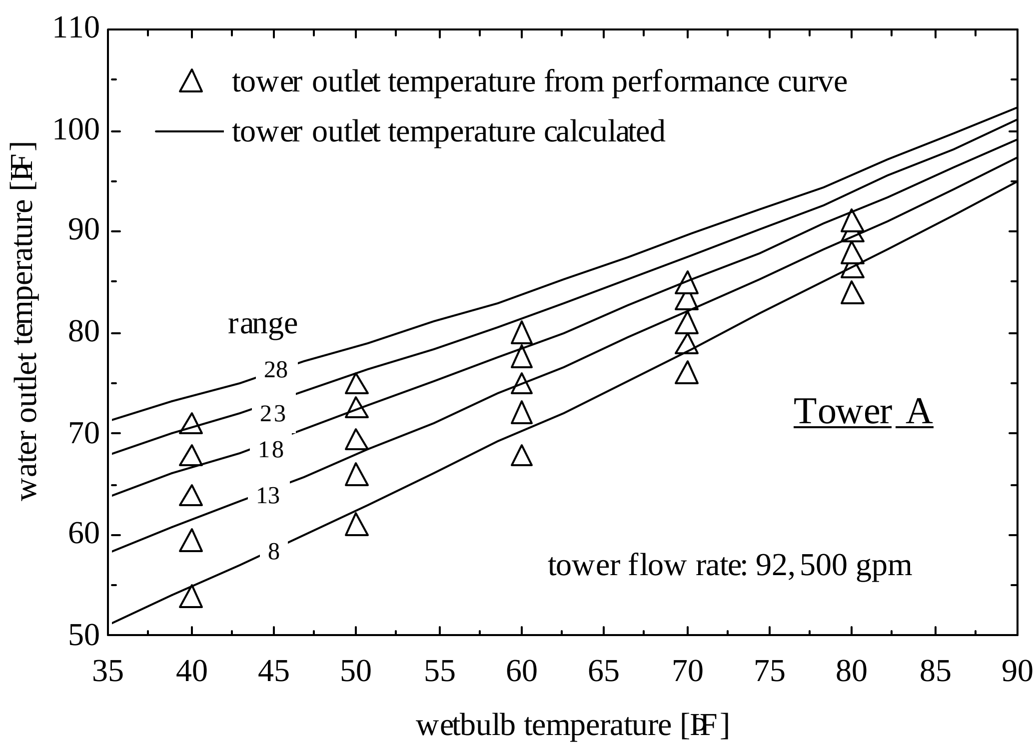

Figure 4.6 a Cooling tower performance data for 100 % design flow rate{ TC "Figure 4.6 a Cooling tower performance data for 100 % design flow rate" \l 5 }

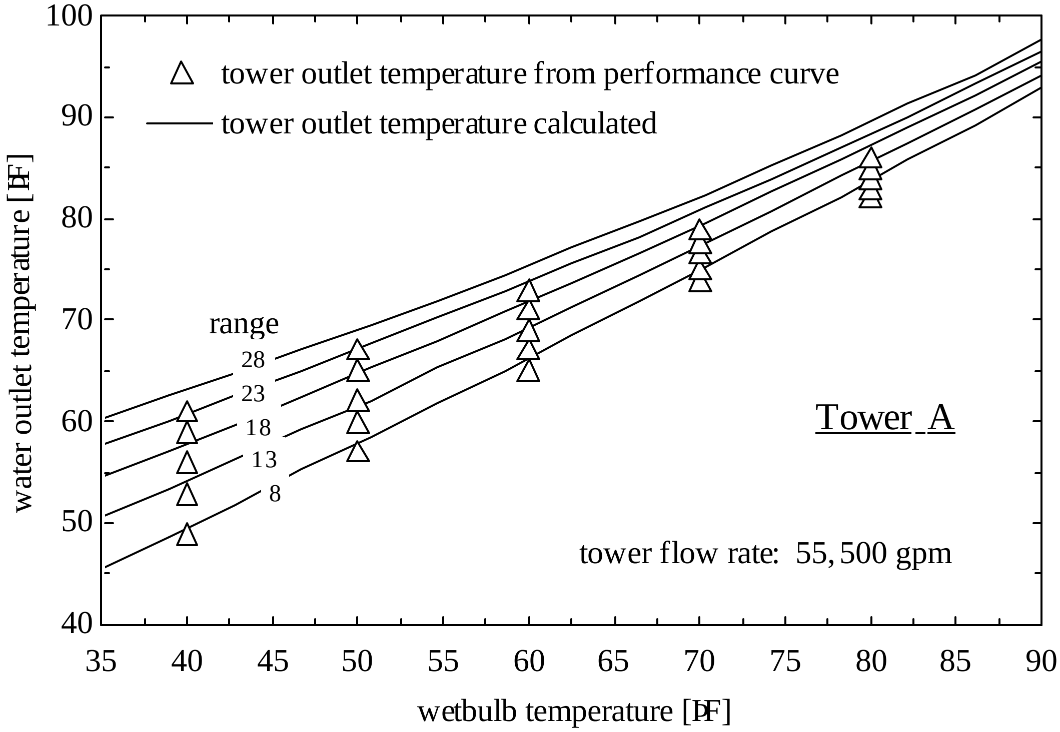

Figure 4.6 b Cooling tower performance data for 60 % design flow rate{ TC "Figure 4.6 b Cooling tower performance data for 60 % design flow rate" \l 5 }

In both charts the calculated tower outlet temperature is higher than the predicted, which was taken from performance curves supplied by the manufacturer. This means that the original design overpredicts the tower performance. Especially for high ranges and high ambient wetbulb temperatures the difference between predicted and actual performance becomes larger. These regions, at high wetbulb and high ranges, are the most crucial for plant operation as they occur at high plant loads during summer months. One reason for the overprediction of performance is the overestimated air flow rate (see section 3.2.4). Another reason can be a wrong estimation of the cooling tower fill heat and mass transfer characteristics.

Comparison of performance of tower A and tower B

Figure 4.7 a. and 4.7 b. show a comparison of performance curves for tower A and tower B. The results shown in the plots are calculated using the tower coefficients shown in table 4.1. In each plot an equal water flow rate is chosen for both towers to be able to compare their performance under the same conditions. The results shown in the charts are calculated for 100 and 50% design water flow. The air flow rate from the tower performance test was used. Thus tower B has a slightly lower volumetric air flow rate than tower A.

Figure 4.7 a. shows that at 100% design flow both towers perform equally well under the same operating conditions. If it is recalled that tower B gets a lower water flow rate during operation, it has to be expected that the water outlet temperature of tower B is lower during operation than the outlet of tower A. At 50% design flow rate the effect is amplified. For equal water flow rates tower B yields already lower outlet temperatures, as shown in figure 4.7 b.. Therefore during operation, when tower B gets less water flow than tower A, the outlet temperature of tower B will always be lower than the outlet temperature of tower A for the same water inlet temperatures.

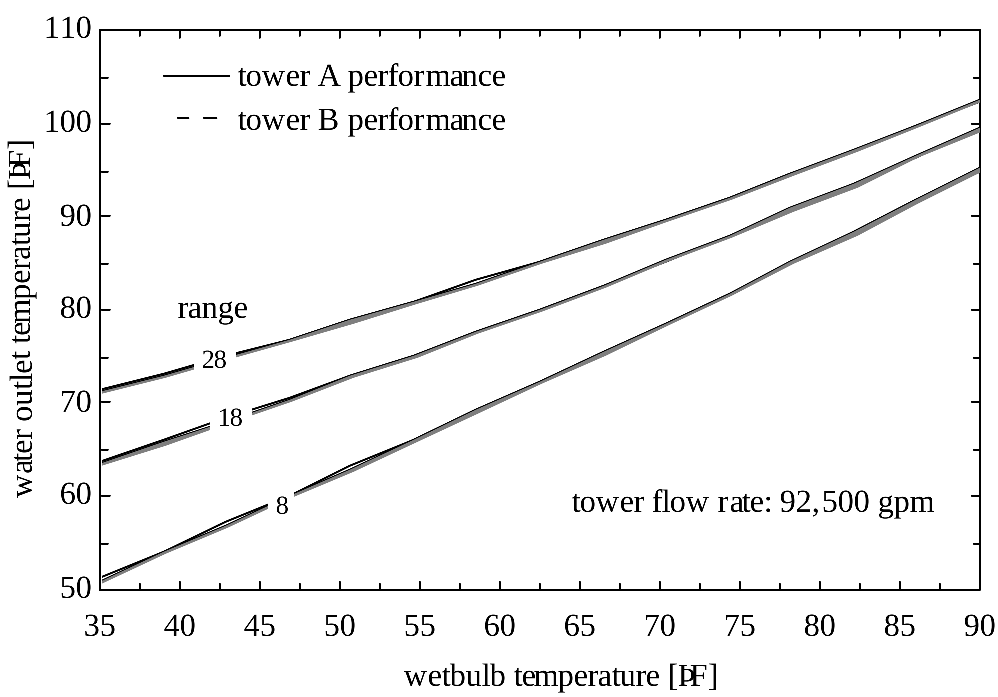

Figure 4.7 a. Comparison of performance of tower A and tower B{ TC "Figure 4.7 a. Comparison of performance of tower A and tower B" \l 5 }

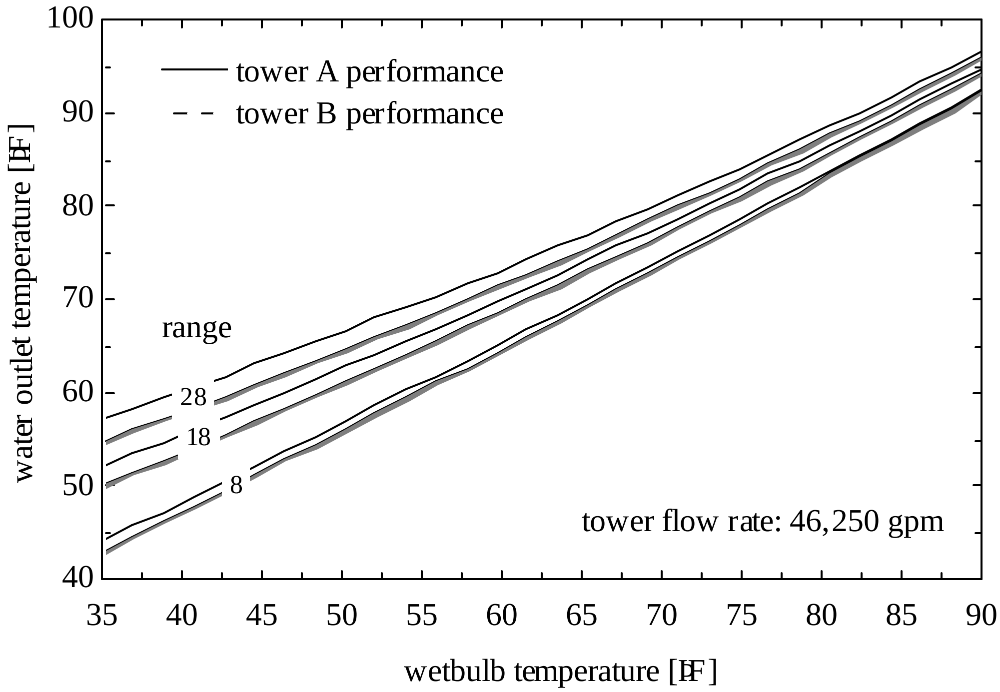

Figure 4.7 b. Comparison of performance of tower A and tower B{ TC "Figure 4.7 b. Comparison of performance of tower A and tower B" \l 5 }

4.1.6 Summary{ TC "4.1.6 Summary" \l 3 }

The tower coefficients that were determined lead to quite accurate predictions of the tower performance, considering the errors associated with measurements. The cooling tower model is therefore considered to be a reliable tool for cooling tower simulation and will be used in the system simulation discussed in chapter 5. The TRNSYS cooling tower routine provides an appropriate tool for this task, as it can be combined with other cooling cycle elements by means of inputs and outputs in a way discussed above. It is also possible to evaluate the effects of a different control strategy, namely the change in water flow rate, or to increase the number of tower cells.

The EES calculations were performed to examine the general operation characteristics of a cooling tower, especially with respect to water flow rates, and help to estimate the influence of different tower operation characteristics on the overall cooling cycle performance. In the further analysis EES will not be used for cooling tower calculations.