DEVELOPMENT OF DRY AND EVAPORATIVE FLUID COOLER MODELS FOR ENERGYPLUS

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NOMENCLATURE

$\dot{m}$	=	Mass flow rate (kg/s)
$\dot{C}$	=	Capacity flow rate W/K
h	=	Enthalpy (j/kg-K)
T	=	Temperature (°C) or (K)
c_p	=	Specific heat (j/kg-K)
R	=	Resistance (K/W)
W	=	Humidity ratio ((kg_w / kg_{dryair}))
Le	=	Lewis number
Φ	=	Relative humidity
U	=	Fluid cooler overall heat transfer coefficient, W/m^2- ^oC
A	=	Heat transfer surface area, m^2

Subscripts

a	=	Air
w	=	Water
p	=	Process fluid
wb	=	Wet-bulb
in	=	Inlet

out	=	Outlet
fic	=	Fictitious
db	=	Dry-bulb
spray	=	Spray water for evaporative fluid cooler

CHAPTER I

INTRODUCTION

1.1 Background

Dry fluid coolers and evaporative fluid coolers provide a clean and effective method of cooling the process fluid. Fig 1.1 and 1.2 show the schematics of dry and evaporative fluid coolers respectively. The process fluid which is generally water or water/glycol mixture is circulated in the closed loop and the ambient air passes across the coil. In the case of dry fluid coolers only air is used to cool the process fluid while in the case of evaporative fluid coolers spray water is used along with ambient air to enhance the effectiveness of the heat exchanger. Because of the usage of spray water, evaporative fluid cooler can cool fluid up to wet-bulb temperature of air. Dry fluid coolers can cool fluid to ambient air dry-bulb temperature.

Dry fluid coolers are combination of outside fan cooled heat exchanger and a pumping station. The process fluid (water/glycol solution) which is used to cool the equipment is circulated by the pump between the heat exchanger and the equipment. Because of the same fluid circulation, internal scaling and corrosion are virtually eliminated. Unlike cooling towers, dry fluid coolers cool the process fluid without any evaporation loss, water treatment or routine maintenance. Some of the dry fluid coolers switch to adiabatic mode in hot climate where the ambient temperatures are very high to provide sufficient fluid cooling. In the adiabatic mode, a fine mist of water is added in the air before it gets in contact with the coil circulating the fluid.

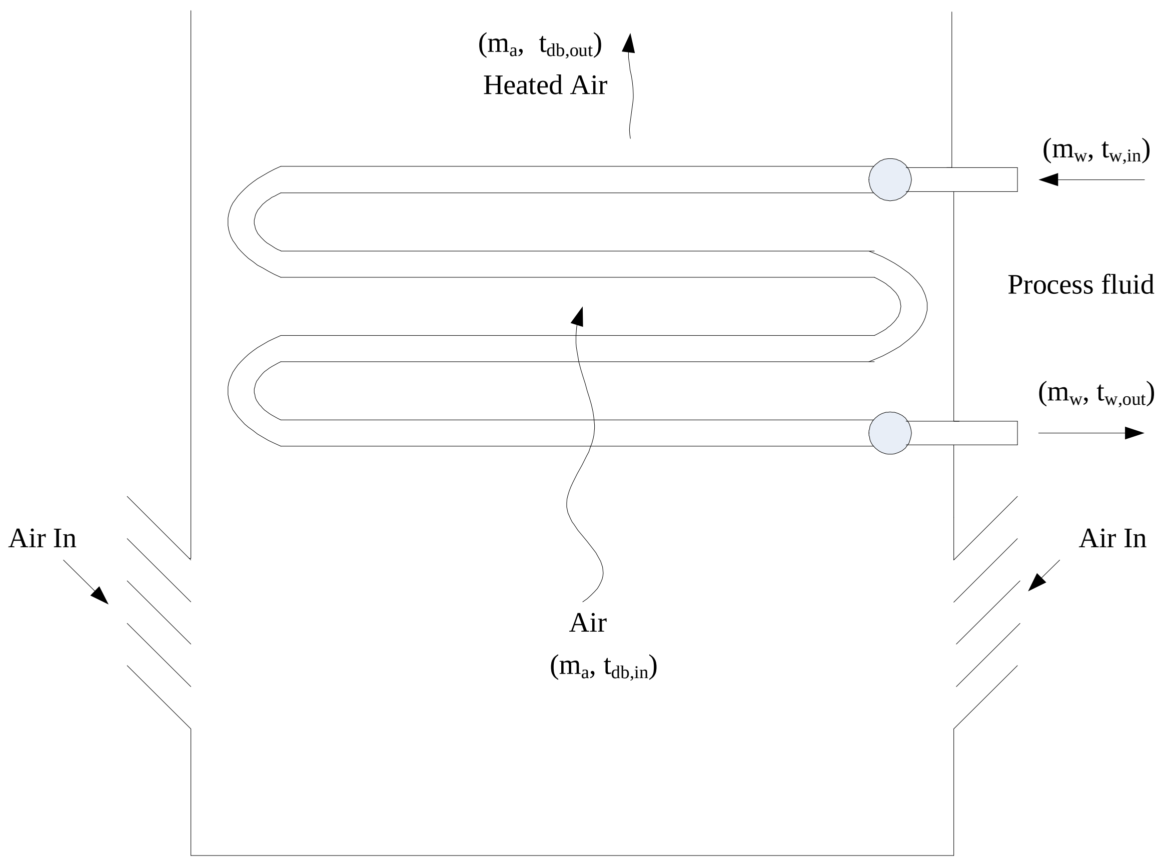


Fig 1.1 Schematic of dry fluid cooler

Water evaporates before coming in contact with the coil and corrosion and scale formation are prevented. The dry fluid cooler has been widely used in both the US and Europe for many years and the installed base of dry fluid coolers is very large.

Another way of cooling the process fluid can be cooling towers. There are two types of cooling towers: open circuit cooling tower and closed cooling tower. Open circuit cooling tower also known as direct contact cooling tower cool the fluid by exposing it to outside air directly. Because of the direct contact with the air, water becomes contaminated. Water treatment, regular heat-exchanger cleaning, difficult cold-weather operation, and large consumption of water are some of the disadvantages of direct contact cooling tower.

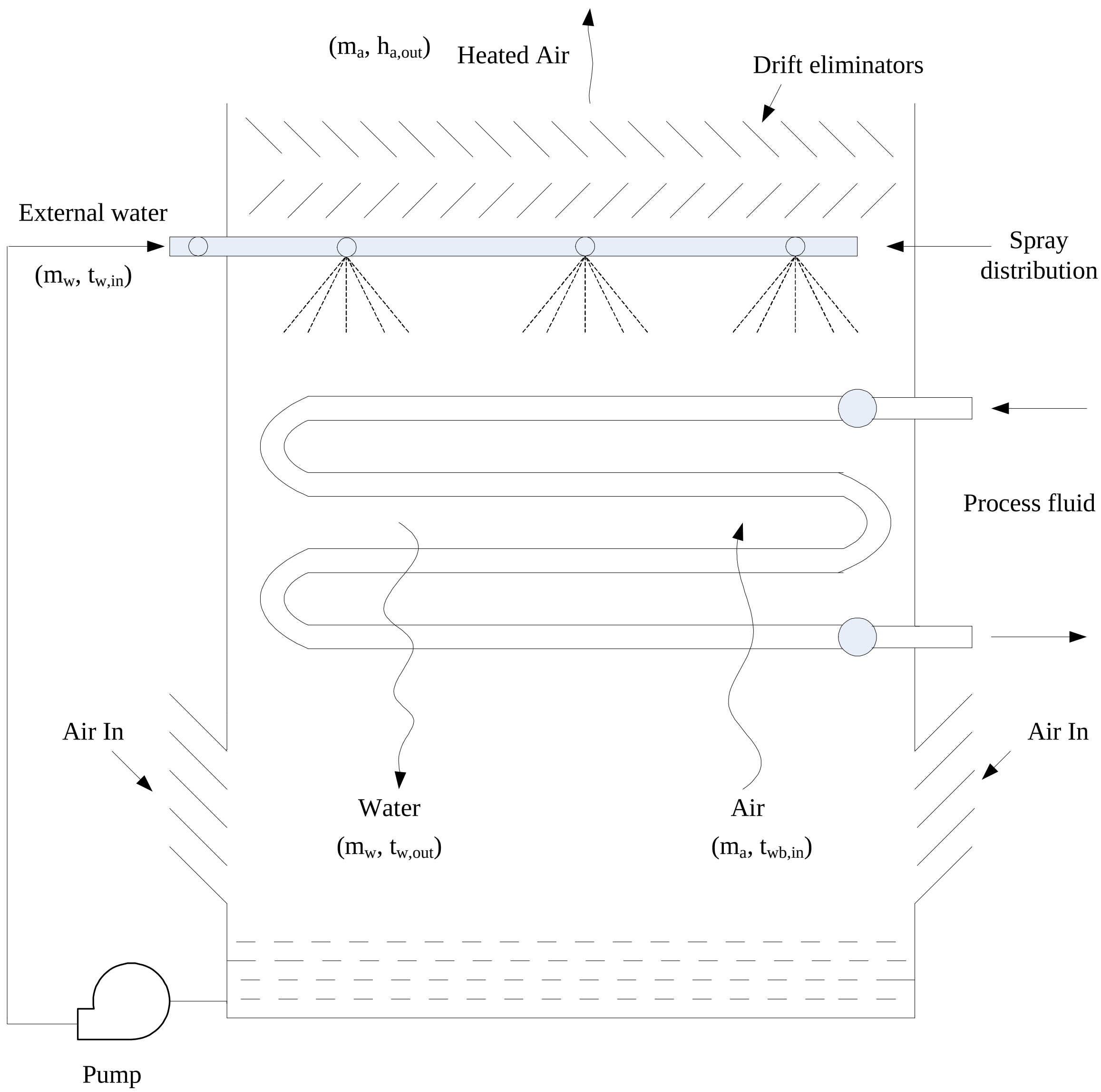
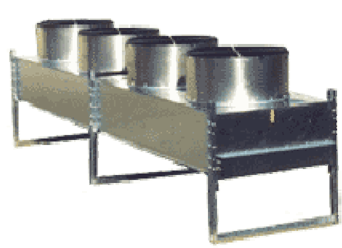


Fig 1.2 Schematic of evaporative fluid cooler

On the other hand closed circuit cooling tower cool the process fluid by circulating them in a closed loop. They require very less maintenance as compare to open circuit cooling towers. Closed circuit cooling towers are also called as indirect contact cooling towers (ICCTs), evaporative fluid/liquid/water coolers, closed wet cooling towers (CWCTs) and closed wet towers. Open circuit cooling towers have been used in the industry for more than eight decades but now dry fluid coolers and closed circuit towers are replacing them more and more. Fig 1.3 shows the dry and evaporative fluid cooler models from colmac coil and general air products respectively.

Evaporative fluid cooler are evaporative enhanced heat exchangers which deliver high efficiency cooling of fluids. Although less common than the dry fluid cooler, it is also an important low energy equipment. This also avoids the health hazards of an open tower. Together these components represent an important addition to the existing plant equipment. The evaporative fluid cooler specifically can be used to target LZEB designs. It requires occasional cleaning but the frequency is much lower as compare to cooling towers. Both dry and evaporative fluid coolers are shown in the figures below.



Dry fluid cooler

(www.colmaccoil.com)

Evaporative Fluid Cooler

(www.generalairproducts.com)

Fig 1.3 Dry fluid cooler and Evaporative fluid cooler

1.2 Objectives

The main objectives of this study are as follows:

1) Obtain information about existing dry and evaporative fluid cooler models through literature review

2) Develop and implement dry and evaporative fluid cooler models in EnergyPlus. The models are added as two new modules in EnergyPlus environment. .

3) Determine sensitivity of the model with respect to various inputs.

4) Provide user documentation for EnergyPlus which states the inputs and outputs of the model. The document also discusses the reference model and how it works in EnergyPlus

5) Verify EnergyPlus model by using other available fluid cooler models and determine the quality of the results.

CHAPTER II

LITERATURE REVIEW

In this chapter, a review of existing models of fluid coolers will be presented. The accuracy, range of applicability and simplicity of the models are discussed. Also the issues pertaining to implementation of models in building simulation programs are discussed e.g. availability of input parameters, convergence problems etc. Finally a summary of the findings is presented in the last section of the chapter.

2.1 Evaporative fluid cooler models

2.1.1 Zalewski and Gryglaszewski (1997) – Mathematical model of heat and mass transfer processes in evaporative fluid coolers

Zalewski and Gryglaszewski (1997) presented the mathematical model of evaporative fluid cooler by using four ordinary differential equations with their associated boundary conditions and some algebraic equations. Fig 2.1 shows the schematic of the evaporative fluid cooling process modeled by them.

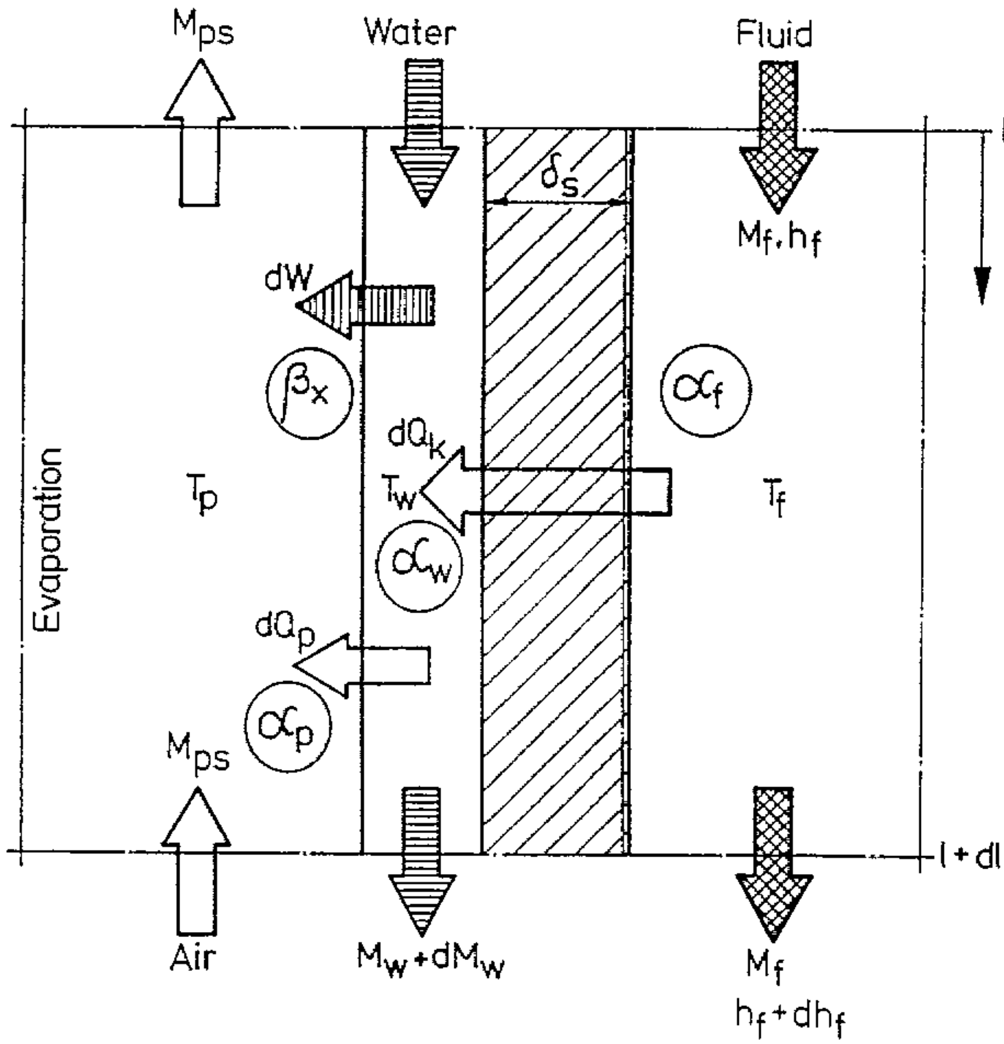


Fig 2.1 Heat exchange scheme of evaporative fluid cooler

The differential equations used by them are described below:

$$\frac{dx}{dl} = -\frac{\beta_x * f_m * B(x''(T_w) - x)}{m_{ps}} \tag{2.1}$$

$$\frac{dT_p}{dl} = -\frac{B(T_w - T_p)}{m_{ps} * c_p} * \{\beta_x * f_m * c_{pw}(x''(T_w) - x) + f_t * \propto_p\} \tag{2.2}$$

$$\frac{dT_w}{dl} = -\frac{B}{m_w * c_w}\{\beta_x * f_m[T_w * (c_w - c_{pw}) - r_0][x''(T_w) - x] \tag{2.3}$$

$$-f_t * \propto_p (T_w - T_p) + k_z(T_f - T_w)\}$$

$$\frac{dT_f}{dl} = -\frac{k_z(T_f - T_w)}{m_f * c_f} \tag{2.4}$$

The boundary conditions for the equations (2.1)-(2.4) are as follows:

$$x(L) = x_1;\ \ T_p(L) = T_p x_1;\ \ \ T_w(0) = T_w(L)\ \ \text{And}\ \ T_f(0) = T_{f1}$$

Where,

x = air humidity ratio ($kg_w\ kg_{ps}^{-1}$)

l = length, linear coordinate, (m)

β_x = mass transfer coefficient, ($kg_{ps}\ m^{-2}\ s^{-1}$)

f, f_t, f_m = area ratio

B = length of wall, length of tubes of exchanger, (m)

T = temperature (°C)

$\propto$ = heat transfer coefficient ($W\ m^{-2}\ K^{-2}$)

r_0 = latent heat of vaporization at 0°C; r_0 = 2500800 ($J\ kg^{-1}$)

h = specific enthalpy ($J\ kg^{-1}$)

W= mass flow rate of water vapor ($kg\ s^{-1}$)

Q_k = heat flux through wall (W)

Q_p = heat flux from water surface into air (W)

δ = thickness, (m)

c = specific heat at constant pressure ($J\ kg^{-1}\ K^{-1}$)

Subscripts

ps = dry air

pw = water vapor

f = cooled liquid

p = moist air

w = spraying water

1= inlet (initial) value

2= final (outlet) value

s = wall

″ = saturation state

The model heavily depends on the specification of the geometry of evaporative fluid cooler which is mostly not available in manufacturer's catalog data. Also, spray water temperature input parameter required by the model is not available in the catalog data. Along with these, determination of heat and mass transfer coefficient is very difficult. Because of the reasons stated above, the model is not suitable for implementing into the building energy simulation programs.

2.1.2 Hasan and Siren (2002) – Theoretical and computational analysis of closed wet cooling towers

Hasan and Siren (2002) presented the theoretical analysis and computation modeling of closed wet cooling towers. They defined tower heat and mass transfer coefficient by using experimental measurements of a prototype of 10 KW tower. They divided the cooling tower tube coils into small elements along the height of the tower. Then heat and mass transfer are considered for each element, starting from first element at cooling water inlet and then proceeding along cooling water flow.

The energy and mass balance equations used in the model are shown below.

The rate of heat lost by the cooling tower dq_c is

$$dq_c = -U_o(T_c - T_s)dA \quad (2.5)$$

Heat transfer rate from water-air interface to air stream is given by

$$dq_a = m_a dh_a = k(h'_s - h_a)dA \quad (2.6)$$

Total energy balance for an element is given by

$$m_c C_w dT_c + m_a dh_a + m_s C_w dT_s = 0 \quad (2.7)$$

The inlet spray water temperature is assumed to be equal to the outlet spray water temperature. So,

$$T_{s1} = T_{s2} \quad (2.8)$$

And finally the mass balance for the element is given by

$$m_e = m_a dW_a = k(W'_s - W_a)dA \quad (2.9)$$

Where,

q = Rate of heat transfer (W)

T = Temperature (°C)

A = Area (m^2)

C = specific heat capacity (kJ/kg-K)

h = Enthalpy (kJ/kg)

W = humidity ratio of moist air (kg water/kg dry air)

m = mass flow rate (kg/s)

k = mass transfer coefficient (kg/s-m^2)

subscripts

c=cooling water

a = air

s= spray water

1 = inlet to tower

2 = outlet to tower

e = evaporation.

superscript

$'$ = saturated condition

There are eight simulation variables which are inlet and outlet values of $\mathrm{T_s}, \mathrm{T_c}, \mathrm{h_a}$ and $\mathrm{W_a}$ and 3 model input parameters which are $\mathrm{T_{c1}}, \mathrm{h_{a1}}$ and $\mathrm{W_{a1}}$. The mass transfer coefficient, which must be specified at the beginning of the simulation, is calculated as follows

$$\mathrm{k} = 0.065\ \mathrm{G_a^{0.773}} \tag{2.10}$$

Where,

$\mathrm{G_a}$ = air mass velocity based on minimum section (Kg s^{-1} m^{-2})

Eq (2.10) is applicable for $0.96 < G_{\mathrm{a}} < 2.76$ (Kg s^{-1} m^{-2}).

The mass transfer coefficient correlation (2.10) is developed for a particular prototype and is not a generally applicable for all the evaporative fluid cooler models. Getting the humidity ratio as the input parameter is very difficult as none of the manufacturers provide it in their catalogs.

2.1.3 Lebrun et al. (2004) - Simplified model

Lebrun et al. (2004) applied a unified theoretical treatment to both evaporative heat exchangers and cooling towers. They regarded these direct and indirect contact cooling towers as classical heat exchangers working in wet regime. The main difference in the model was related to different global heat transfer coefficient for each type.

The mathematical model used by them is described below:

The air side energy balance is

$$\dot{Q} = \dot{m}_a\left(h_{a,out} - h_{a,in}\right) \tag{2.11}$$

Using the fictitious gas assumption, this equation can be expressed as

$$\dot{Q} = \dot{C}_{af}\left(T_{wb,out} - T_{wb,in}\right) \tag{2.12}$$

$$\dot{C}_{afic} = \dot{m}_a c_{p,afic} \quad \text{(Fictitious capacity of humid air)} \tag{2.13}$$

By using equations (2.11), (2.12) and (2.13):

$$c_{p,afic} = \frac{\left(h_{a,out} - h_{a,in}\right)}{\left(T_{wb,out} - T_{wb,in}\right)}$$

The heat flow rate is calculated by:

$$\dot{Q} = \mathcal{E}_{fic}\dot{C}_{min}(T_{w,in} - T_{wb,in}) \tag{2.14}$$

And the process fluid side energy balance is:

$$\dot{Q} = \dot{C}_w(T_{w,in} - T_{wb,in}) \tag{2.15}$$

$$\dot{C}_w = \dot{m}_w c_{p,w} \quad \text{(Capacity of process fluid)} \tag{2.16}$$

The step by step method to calculate effectiveness of the heat exchanger is described below:

$$\dot{C}_{min} = \text{Min}\ (\dot{C}_{afic}, \dot{C}_w)$$

$\dot{C}_{max} = \text{Max}\,(\dot{C}_{afic}, \dot{C}_w)$

$C_r = \frac{\dot{C}_{min}}{\dot{C}_{max}}$

$NTU_{fic} = \frac{AU_{fic}}{\dot{C}_{min}}$

$$\mathcal{E}_{fic} = \frac{1 - e^{(-NTU_{fic}(1-C_r))}}{1 - C_r e^{(-NTU_{fic}(1-C_r))}} \tag{2.17}$$

The global heat transfer coefficient was calculated as follows:

$$AU_{fic} = \frac{1}{R_{fic}} \tag{2.18}$$

$$R_{fic} = R_{afic} + R_w \tag{2.19}$$

$$R_{afic} = R_a \frac{c_{p,a}}{c_{p,afic}} \tag{2.20}$$

$$R_a = R_{a,n} \left[\frac{\dot{m}_a}{\dot{m}_{a,n}}\right]^n \tag{2.21}$$

$$R_w = R_{w,n} \left[\frac{\dot{m}_w}{\dot{m}_{w,n}}\right]^m \tag{2.22}$$

Where,

$\dot{Q}$ = Heat transfer rate (W)

$\dot{m}$ = mass flow rate (kg/s)

m = Water side mass flow rate ratio exponent

n = Air side mass flow rate ratio exponent

c_p= Specific heat capacity (kJ/kg-K)

$\dot{C}$ = Capacity flow rate (W/K)

$\mathcal{E}$ = Effectiveness

AU = Overall heat transfer coefficient (W/K)

NTU = Number of transfer units

R = Resistance (K/W)

subscripts

a = air

w = water

n= nominal

fic = fictitious

in = inlet

out = outlet

min = minimum

max = maximum

r = ratio

wb = wet-bulb

The accuracy of the model is within ± 7.5 % when compared with the manufacturer's data. The model is relatively simple to be implemented in building simulation programs. But the problem associated with the model is the estimation of four parameters i.e. $R_{a,n}$, $R_{w,n}$ and exponents m and$n$. Either the parameters need to be determined separately for a particular fluid cooler model and then used in the building simulation programs or they may be determined in the simulation program itself. This can cause serious convergence problem.

The Lebrun model serves as the basis for EnergyPlus model. But instead of estimating four parameters, an iterative procedure is carried out to estimate AU_{fic} directly by using manufacturer's data. The model, however, is implemented in the Visual Basic for

Application for the verification of EnergyPlus model. All the four parameters are estimated in VBA program.

2.1.4 Stabat and Marchio (2004) - Simplified model

Another simplified model was presented by Stabat and Marchio (2004) for indirect contact evaporative cooling towers. Ɛ-NTU method is used to describe the model. Fig 2.2 shows the heat exchange scheme used in the model. The scheme consisted of two parts, 1) heat transfer between air and water film outside the tube; and 2) heat transfer between water in the tubes and water film outside the tube.

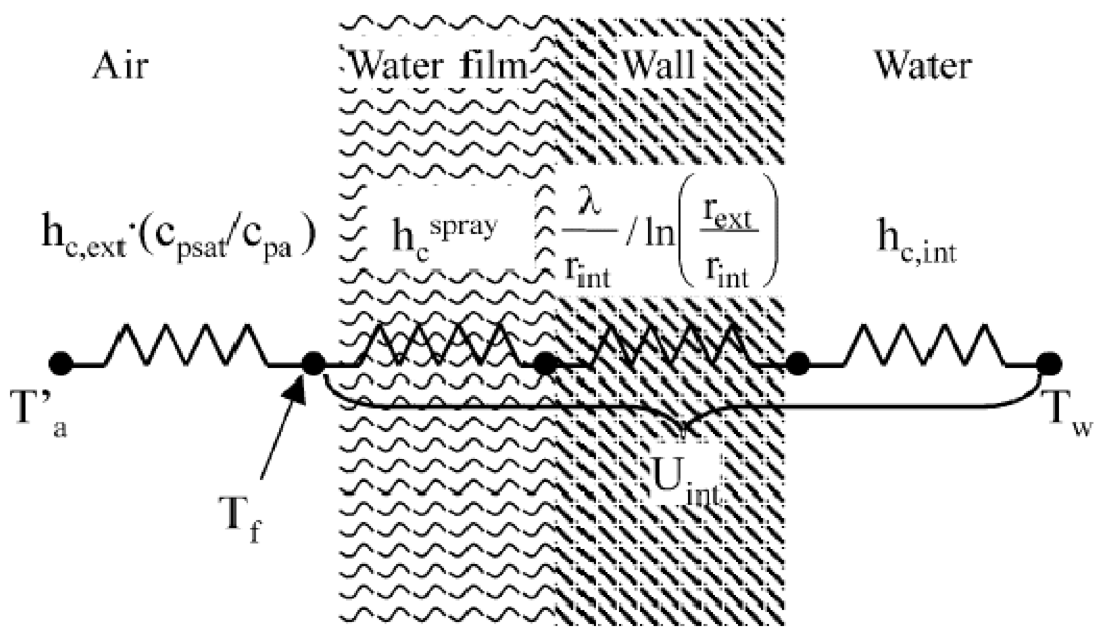


Fig 2.2 Heat transfer scheme of closed circuit cooling tower

These heat transfers are characterized by air and water side heat transfer coefficients respectively. Finally the overall heat transfer coefficient is used to represent heat transfer between water and air. Closed cooling towers are also operated without spray when the atmospheric conditions are favorable. So depending on whether the operation is with or without spray, closed circuit cooling towers operate in wet and dry regimes respectively. Table 2.1 shows the equations used to calculate overall heat transfer coefficient of counter flow single pass heat exchanger for both dry and wet regimes.

Table 2.1 - Ɛ-NTU relations for wet and dry regimes

<table>
<tr><td>Wet regime</td><td>Dry regime</td></tr>
<tr><td>$\varepsilon = \frac{\dot{C}_w(T_{w,in}-T_{w,out})}{\dot{C}_{min}(T_{w,in}-T_{a,in})}$</td><td>$\varepsilon = \frac{\dot{C}_w(T_{w,in}-T_{w,out})}{\dot{C}_{min}(T_{w,in}-T_{a,in})}$</td></tr>
<tr><td colspan="2">$\varepsilon = \frac{1-e^{(-NTU(1-C_r))}}{1-C_r e^{(-NTU(1-C_r))}}$ (if $C_r < 1$) with $NTU = \frac{U_t A_t}{\dot{C}_{min}}$ and $C_r = \frac{\dot{C}_{min}}{\dot{C}_{max}}$
$\varepsilon = \frac{NTU}{1+NTU}$ (if $C_r = 1$)</td></tr>
<tr><td>$\dot{C}_a = \dot{m}_a c_{p,sat}$ and $\dot{C}_w = \dot{m}_w c_{p,w}$</td><td>$\dot{C}_a = \dot{m}_a c_{pa}$ and $\dot{C}_w = \dot{m}_w c_{p,w}$</td></tr>
<tr><td colspan="2">$\dot{C}_{max} = \mathrm{Max}\ (\dot{C}_a, \dot{C}_w)$; $\dot{C}_{min} = \mathrm{Min}\ (\dot{C}_a, \dot{C}_w)$ and $c_{p,sat} = \frac{(h_{a,out}-h_{a,in})}{(T_{wb,out}-T_{wb,in})}$</td></tr>
<tr><td>$\frac{1}{U_t A_t} = \frac{1}{U_{ext}^{wet} A_{ext}} + \frac{1}{U_{int}^{wet} A_{int}}$</td><td>$\frac{1}{U_t A_t} = \frac{1}{U_{ext}^{dry} A_{ext}} + \frac{1}{U_{int}^{dry} A_{int}}$</td></tr>
</table>

Where,

Ɛ = Effectiveness;

T = Temperature (°C)

$\dot{C}$ = Capacity flow rate (W K^{-1})

$c_p =$ Specific heat (J kg^{-1} K^{-1})

$c_{p,sat} =$ Fictitious specific heat (J kg^{-1} K^{-1})

$\dot{m} =$ Mass flow rate (kg s^{-1})

h = Enthalpy (J kg^{-1})

A_{ext}, A_{int} = Surface area at external and internal side (m^2)

$U_{ext}^{wet}, U_{ext}^{dry}$ = Air side heat transfer coefficient in wet and dry regime (W $K^{-1}m^{-2}$)

$U_{int}^{wet}, U_{int}^{dry}$ = Water side heat transfer coefficient in wet and dry regime (W K^{-1} m^{-2})

$U_t A_t$ = Overall heat transfer coefficient (W K^{-1})

NTU = Number of Transfer units

Subscripts

w = water

a = air

in = inlet

out = outlet

t = total

wb = wet-bulb

r = ratio

Determination of air side heat transfer coefficient

The correlations for air side heat transfer coefficients are given by:

$$U_{ext}^{wet} A_{ext} = \beta_{ext}^{wet} c_{p,sat} \dot{m}_a^{0.8} \tag{2.23}$$

$$U_{ext}^{dry} A_{ext} = \beta_{ext}^{dry} c_{pa} \dot{m}_a^{0.8} \tag{2.24}$$

Where,

$\beta_{ext}^{wet}, \beta_{ext}^{dry}$ = constants to be fitted for wet and dry regimes respectively

Determination of water side heat transfer coefficient

For the water side heat transfer, conductive resistance through the tube wall is negligible as compare to convective resistance on the inside and outside of the tube. So the water side heat transfer coefficient can be represented as:

$$\frac{1}{U_{int}^{wet}A_{int}} = \frac{1}{h_c^w A_{int}} + \frac{1}{h_c^{film}A_{ext}} \tag{2.25}$$

Where,

h_c^w = Convective heat transfer coefficient between water and tube (W $K^{-1}m^{-2}$)

h_c^{film} = Heat transfer coefficient between tube surface and water film (W $K^{-1}m^{-2}$)

Using Dittus-Boelter correlation (Incropera and Dewitt 1996) for inside the tube

$$h_c^w = 0.023\frac{k_w}{d_{int}}Re^{0.8}Pr^{0.3} \quad (Re > 10^4 \ \&\ 0.7 \le Pr \le 160) \tag{2.26}$$

Where,

k_w= conductivity of water (W K^{-1} m^{-1})

d_{int} = Inside diameter of tube (m)

Correlation for $h_c^{film}$is described as:

$$h_c^{film} = C\left[\frac{\dot{G}_{spray}}{d_{ext}}\right]^n \tag{2.27}$$

Where,

$\dot{G}_{spray}$ = Flow rate of spray water per unit breadth (kg m^{-1} s^{-1})

d_{ext} = Outside diameter of tube (m)

C, n = Constants to be fitted

So the equation (2.25) can be simplified as follows:

$$U_{int}^{wet}A_{int} \propto Re^{0.8}Pr^{0.3} = \beta_{int}^{wet}\dot{m}_w^{0.8}\mu_w^{-0.5} \quad \text{(Wet-regime)} \tag{2.28}$$

$$U_{int}^{dry}A_{int} \propto Re^{0.8}Pr^{0.3} = \beta_{int}^{dry}\dot{m}_w^{0.8}\mu_w^{-0.5} \quad \text{(Dry-regime)} \tag{2.29}$$

Where,

μ = Dynamic viscosity (kg m^{-1} s^{-1})

$\beta_{int}^{wet}, \beta_{int}^{dry}$ = Constants to be fitted

Overall heat transfer coefficient

The overall heat transfer coefficient for the indirect contact cooling tower can be expressed as:

$$\frac{1}{U_tA_t} = \frac{1}{\beta_{ext}^{wet}c_{p,sat}\dot{m}_a^{0.8}} + \frac{\mu_w^{0.5}}{\beta_{int}^{wet}\dot{m}_w^{0.8}} \quad \text{(Wet regime)} \tag{2.30}$$

$$\frac{1}{U_tA_t} = \frac{1}{\beta_{ext}^{dry}c_{pa}\dot{m}_a^{0.8}} + \frac{\mu_w^{0.5}}{\beta_{int}^{dry}\dot{m}_w^{0.8}} \quad \text{(Dry regime)} \tag{2.31}$$

Determination of β_{int} and β_{ext} requires to two rating points from the catalog data which most of the evaporative fluid cooler manufacturers don't provide. This poses great difficulty in estimation of these parameters. The accuracy of the model is high and computation time is less. The model can also be used under different operational conditions e.g. variable air flow rates and variable wet-bulb temperatures.

2.1.5 Quereshi and Zubair (2005) – Comprehensive design and rating study of evaporative fluid coolers

Quereshi and Zubair (2005) studied effect of fouling on thermal effectiveness of evaporative fluid cooler and evaporative condenser. They took infinitesimal control volume of evaporative heat exchangers consisting of 3 subsystems having air, water and process fluid.

After applying the water mass balance

$$\frac{\partial W}{\partial A} = \frac{1}{m_a} \frac{\partial m_w}{\partial A} \tag{2.32}$$

The mass flow rate of spray water evaporating into the air is given by

$$dm_w = h_D(W_{s,int} - W)dA \tag{2.33}$$

The simplified simultaneous heat and mass transfer equations for Lewis number equal to unity is as follows:

$$dh_a = \frac{h_D}{m_a}(h_{s,int} - h_a)dA \tag{2.34}$$

Energy balance on the process fluid subsystem is given by:

$$dT_p = -\frac{U_{os}}{m_p * c_{p,p}}(T_p - T_{int})dA \tag{2.35}$$

The simplified overall energy balance on the control volume of evaporative fluid cooler is:

$$dT_w = \frac{1}{m_w * c_{p,w}}(m_a dh_a - c_{p,w} T_w dm_w + c_{p,p} m_p dT_p) \tag{2.36}$$

Where,

U = Overall heat transfer coefficient (kW m^{-2} C^{-1})

W = Humidity ratio of mist air (kg_w kg_a^{-1})

h_D = Convective mass transfer coefficient (kg_w m^{-2} s^{-1})

h = Specific enthalpy (kJ kg^{-1})

T = Temperature (°C)

m = Mass flow rate (kg s^{-1})

Subscripts

a = air

p = process fluid

w = water

int = air-water interface

The equations (2.32), (2.33), (2.34), (2.35) and (2.36) describe the evaporative fluid cooler operation. These differential equations are solved by using EES. Different correlations were used to obtain outside tubes heat transfer coefficient and water film mass transfer coefficient.

This model is then integrated with an asymptotic model of fouling growth developed by Qureshi and Zubair (2005) in an earlier work. The results have shown that effectiveness of evaporative heat exchangers are decreased by more than 50% because of fouling. This caused outlet process fluid temperature to increase by 5%. They also did parametric study to evaluate the effects of elevation and mass flow rate ratio in the performance of evaporative heat exchangers. Their experiments have shown that as air gets cooler at high altitudes, less surface area of heat exchanger is required for same amount of process fluid cooling. For different mass flow rate ratios i.e.$\left[\frac{m_{w,spray}}{m_a}\right]$, percentage reduction in surface area with respect to surface area at standard atmospheric pressure is found to be almost the same. It means that increasing mass flow rate ratios $\left[\frac{m_{w,spray}}{m_a}\right]$ does not have significant impact on lowering outlet process fluid temperature.

The evaporative fluid cooler model was experimentally validated by using Jang and Wang's (2001) model. The results were in good agreement. Also the evaporation loss errors were within -0.9 to 6 % when compared with the data provided by Baltimore Aircoil.

The value of h_D is not known for most of the cases. Also the input parameters required by the model are not readily available in manufacturer's catalog data e.g. spray water temperatures.

2.2 Dry fluid cooler models

Dry fluid coolers can be modeled using classic heat exchanger equations. There are 2 methods which are mainly reported in the literature to analyze heat exchangers.

1) Log mean temperature difference method
2) Ɛ-NTU method

2.2.1 Log mean temperature difference (LMTD) method

The heat transfer of classic heat exchanger using LMTD method is given by

$$\dot{Q} = UA * LMTD \tag{2.37}$$

Where,

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \tag{2.38}$$

For parallel heat exchangers

$$\Delta T_1 = T_{h,in} - T_{c,in}$$

$$\Delta T_2 = T_{h,out} - T_{c,out}$$

For counter flow heat exchangers

$$\Delta T_1 = T_{h,in} - T_{c,out}$$

$\Delta T_2 = T_{h,out} - T_{c,in}$

Subscripts

h = hot

c = cold

The main disadvantage of LMTD method is that it requires fluid temperatures as inputs which are typically not known. If only the inlet fluid temperatures are known, a cumbersome iterative procedure can be carried out to implement LMTD method. However, in the same conditions Ɛ-NTU method is much more convenient to use. Because of this reason Ɛ-NTU method is used to model fluid coolers in EnergyPlus.

2.2.2 Ɛ-NTU method

$$\dot{Q} = \varepsilon * \dot{C}_{min} * (t_{h,in} - t_{c,in}) \qquad (2.39)$$

Where,

$\dot{C}_{min} = \text{Min}\ (\dot{C}_h, \dot{C}_c)$

$\dot{C}_{max} = \text{Max}\ (\dot{C}_h, \dot{C}_c)$

$C_r = \frac{\dot{C}_{min}}{\dot{C}_{max}}$ = capacity ratio

Depending on heat exchanger configuration i.e. parallel flow, counter flow or cross flow different correlations can be used to calculate ε (effectiveness). For cross flow configuration when both the streams are mixed, the Ɛ-NTU correlation is given by

$$\varepsilon = 1 - \exp\left[\frac{e^{(-NTU C_r \eta)}}{C_r \eta}\right] \tag{2.40}$$

Where,

$$NTU = UA/\dot{C}_{min} \tag{2.41}$$

$$\eta = NTU^{-0.22} \tag{2.42}$$

Eq. (2.40) is used in EnergyPlus to calculate effectiveness of dry fluid cooler.

In conclusion different fluid cooler models are studied. Their accuracy, range of applicability and relative simplicity are discussed. Lebrun model is used with some modification for the development of EnergyPlus' evaporative fluid cooler model. Dry fluid cooler is modeled as a classical heat exchanger by using Ɛ-NTU correlations from cross flow heat exchanger with both streams unmixed. Chapter 3 elaborates further the fluid cooler models implemented in EnergyPlus.

CHAPTER III

DEVELOPMENT OF FLUIDCOOLER MODELS FOR ENERGYPLUS

In this chapter two new fluid cooler models were developed for EnergyPlus. The chapter also discusses the catalog data provided by the fluid cooler manufacturers. Design input parameters required by the model are presented and finally the actual model algorithms and input specifications are explained.

3.1 Overview of the Models

The fluid cooler models are characterized by a single parameter, the overall heat transfer coefficient-area product, UA. Generally, this parameter is not available and needs to be calculated by using experimental data or manufacturer's catalog data. The catalog data available for fluid coolers are mostly insufficient. Also the manufacturers provide data only for one rating point. There are some standard test conditions which are set by Cooling Technology Institute (CTI) for cooling towers. Standard test conditions are 3 GPM/ton entering water at 35°C (95°F), leaving water at 29.44°C (85°F), entering air at 25.56°C

(78°F) wet-bulb temperature and 35°C (95°F) dry-bulb temperature. The nominal capacity of the cooling tower is the capacity specified at these conditions. Some evaporative fluid cooler manufacturers provide catalog data on these standard test conditions. But a vast majority of them don't follow any standard conditions to publish catalog data. Because of the insufficiency of catalog data, the UA values of the fluid coolers are determined for one rating point only. Fig (3.1) shows the catalog data for evaporative fluid cooler taken from Baltimore Aircoil's website. In the figure, the capacity in U.S. Gallons per minute of water is shown. The hot water/cold water temperatures are (95/85°F, 102/90°F and 115/90°F) and wet bulb temperatures are (72°F, 78°F and 80°F).

Model	95/85°F			102/90°F			115/90°F		
	72°F	78°F	80°F	72°F	78°F	80°F	72°F	78°F	80°F
VF1-009-12G	27	16	12	32	24	20	17	12	11
VF1-009-22G	34	20	15	42	30	26	21	15	13
VF1-009-32G	41	25	18	50	36	31	26	19	16
VF1-009-42G	46	27	20	56	40	34	28	20	18
VF1-018-02H	50	28	20	62	43	37	29	21	18
VF1-018-12H	71	42	31	86	62	53	44	32	27
VF1-018-22J	97	60	45	116	86	75	62	46	40
VF1-018-32J	108	67	51	128	96	83	69	52	46
VF1-018-42J	117	72	55	139	104	90	75	56	49
VF1-027-22H	115	70	53	138	102	88	73	55	48
VF1-027-22J	136	84	64	162	121	105	88	66	58
VF1-027-32K	165	104	80	195	147	128	108	82	72
VF1-027-42K	176	111	85	209	157	137	115	87	76
VF1-036-21L	201	123	93	240	178	154	128	95	83
VF1-036-31L	216	132	100	258	191	166	138	103	90
VF1-036-41L	231	141	107	276	204	177	147	109	95
VF1-036-51L	244	147	111	293	215	186	154	113	99
VF1-048-21M	269	164	124	323	238	206	171	127	110
VF1-048-31N	319	196	149	381	282	245	204	152	133

Fig 3.1 Baltimore Aircoil's catalog data for evaporative fluid coolers

Fig (3.2) shows the catalog data for Motivair Corp. dry fluid cooler. A single rating for different fluid cooler model is shown in the figure.

MFC SPECIFICATIONS

FLUID COOLER MODEL	FLOW RATE (GPM)	HEAT REJECTION CAPACITY* (BTU/HR)	NO. OF FANS	TOTAL AIRFLOW (SCFM)	OVERALL DMENSIONS(IN.) L	W	H	INT VOL. (GAL.)	EMPTY WEIGHT (LBS.)
MFC0200	22	200,000	2	21,000	90	43	50	6	580
MFC0250	28	250,000	2	20,600	90	43	50	9	630
MFC0300	34	300,000	2	19,800	90	43	50	9	650
MFC0350	39	350,000	2	18,500	76	43	50	12	730
MFC0400	45	400,000	3	30,900	130	43	50	17	900
MFC0450	51	450,000	3	29,700	130	43	50	17	930
MFC0500	56	500,000	3	28,600	130	43	50	22	1010
MFC0550	62	550,000	4	41,200	231/121**	88	50	18	1580
MFC0600	68	600,000	4	39,600	231/121**	88	50	18	1620
MFC0700	79	700,000	4	37,000	231/121**	88	50	24	1760
MFC0800	90	800,000	6	61,000	341/176**	88	50	27	1810
MFC0900	101	900,000	6	60,600	341/176**	88	50	27	2390
MFC0950	107	950,000	6	59,400	341/176**	88	50	27	2440
MFC1000	113	1,000,000	6	57,200	341/176**	88	50	36	2550
MFC1100	124	1,100,000	6	55,500	341/176**	88	50	36	2610
MFC1200	135	1,200,000	8	79,200	231	88	50	36	3140
MFC1300	146	1,300,000	8	76,300	231	88	50	48	3200
MFC1400	158	1,400,000	8	74,000	231	88	50	48	3510
MFC1500	169	1,500,000	10	99,000	286	88	50	45	3990
MFC1600	180	1,600,000	10	98,200	286	88	50	60	4000
MFC1700	191	1,700,000	10	95,300	286	88	50	60	4150
MFC1800	203	1,800,000	10	92,500	286	88	50	60	4380
MFC1900	214	1,900,000	12	117,900	341	88	50	72	4710
MFC2000	225	2,000,000	12	114,400	341	88	50	72	4890
MFC2100	236	2,100,000	12	111,000	341	88	50	72	5230

Performance based on 40% glycol entering the cooler @ 125^0 F, leaving the cooler @ 105^0 F, & 95^0 F ambient.
***These coolers are available in eiher a 1-fan or 2-fan-wide configuration.*

Fig 3.2 Motivair corp. catalog data for dry fluid coolers

3.2 EnergyPlus model description

As discussed earlier, UA is single characterizing parameter for the fluid coolers. Two input methods are mainly provided in EnergyPlus to specify fluid cooler performance which are:

1) UA and design water flow rate

2) Design capacity method

Figures (3.3) and (3.4) depict inputs and outputs for both the methods. If the UA value of the fluid cooler and the corresponding water flow rate are known, they can be specified directly in input. This is the first method which doesn't require any iteration to be performed. In the second method i.e. design capacity method, the design parameters obtained from the catalog data are used to estimate UA of the fluid cooler. Fig 3.4 shows the model parameters and simulation variables for this method.

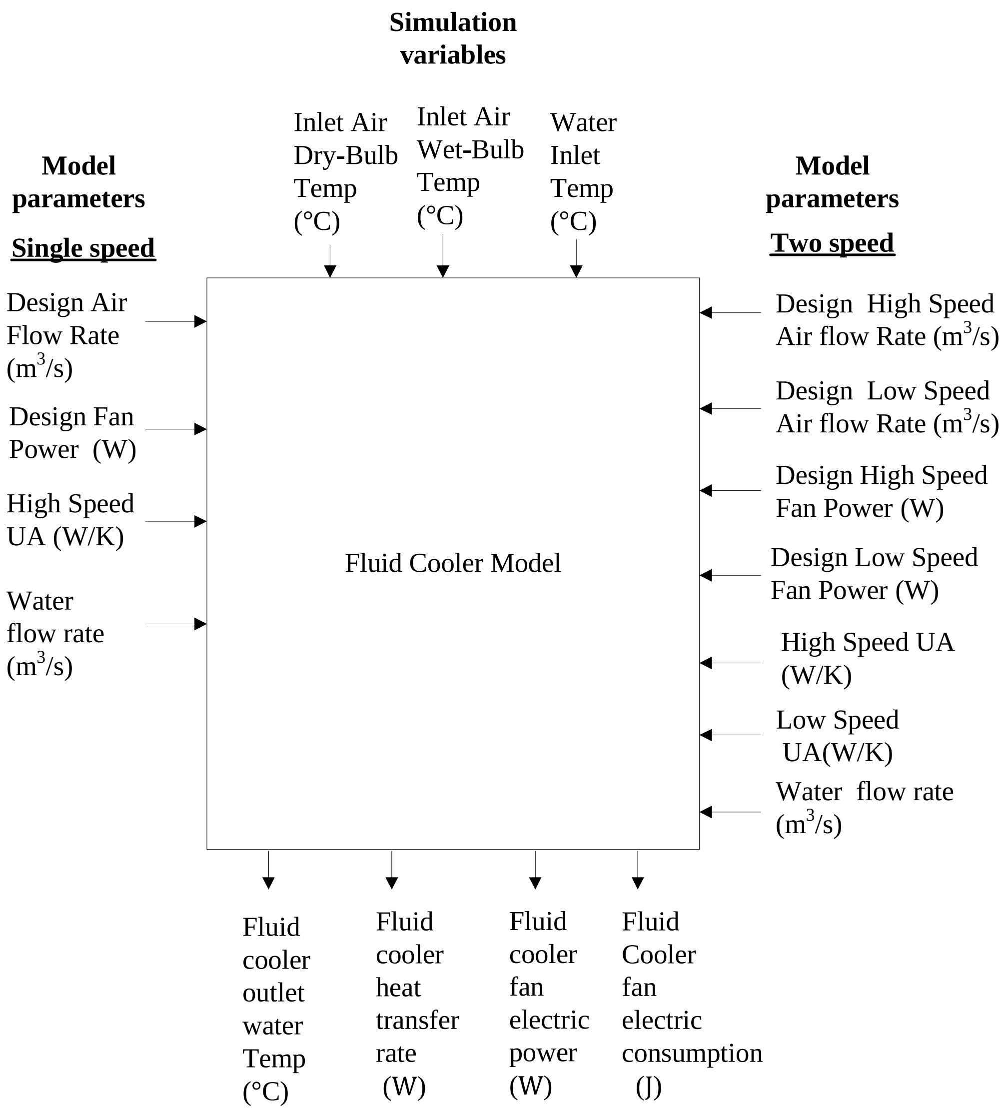


Fig 3.3: Information flow chart for UA and design flow rate method of fluid cooler model in EnergyPlus

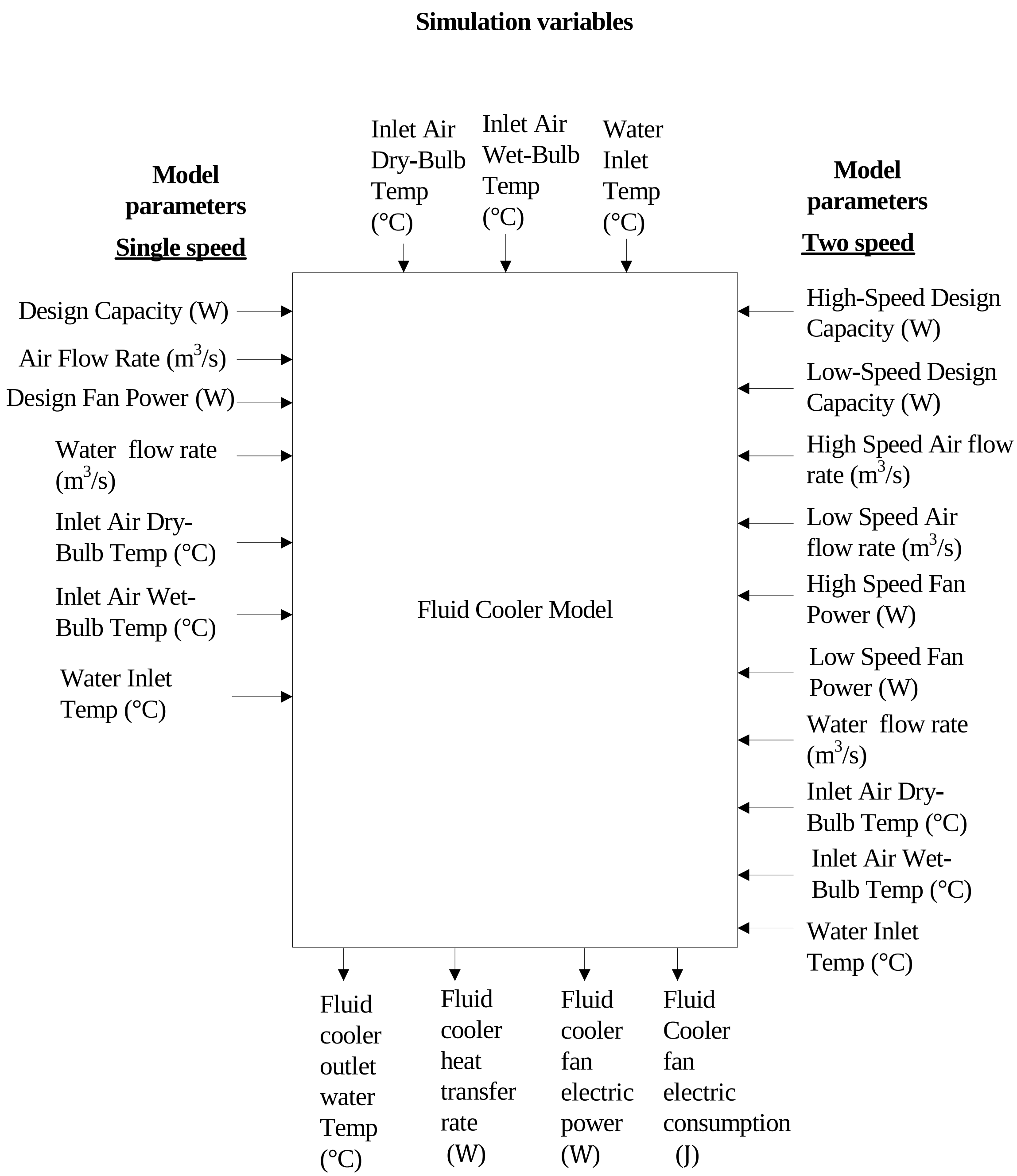


Fig 3.4: Information flow chart for design capacity method of fluid cooler models in EnergyPlus

To estimate UA from design parameters, an iterative procedure, described in Fig (3.5) is used in EnergyPlus. First, the fluid cooler model guesses the UA value and calculates the output conditions. New guesses of the UA value are made by using “regula falsi” method until the iteration converges to a unique solution. Once the UA value is determined, it is used in the subsequent simulation calculations.

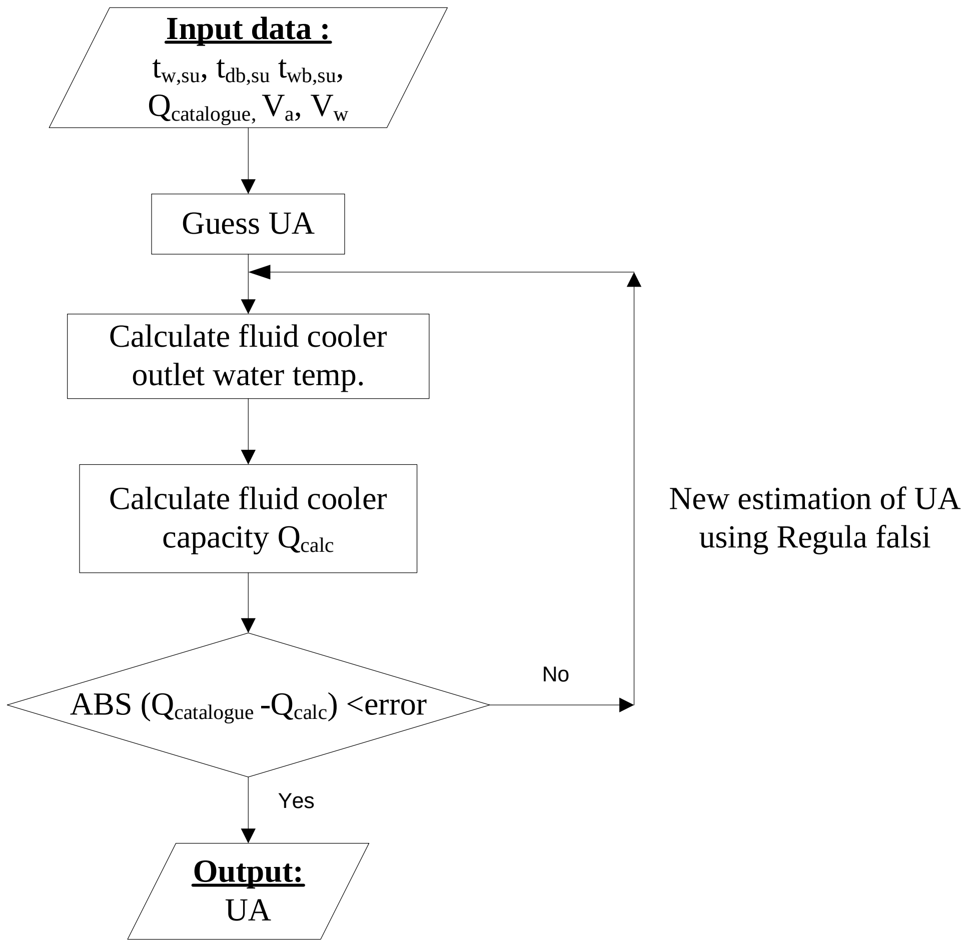


Fig 3.5 Flow chart for UA calculation method used by EnergyPlus

3.3 Implementing the Fluid Cooler Models in EnergyPlus

Since EnergyPlus is a modular simulation program, the dry and evaporative fluid cooler models are implemented as two new modules in EnergyPlus. Ɛ-NTU equations described in section 2.2.2 are used to model dry fluid cooler and Lebrun model (section 2.1.2) is used as the basis to develop evaporative fluid cooler model in EnergyPlus. In the following sections, input specifications and actual algorithms of fluid cooler models implemented in EnergyPlus are discussed.

3.3.1 Input specifications of fluid coolers

Inputs are specified in EnergyPlus by means of text files. These text files are called IDD (input data dictionary) and IDF (input data file). Different object types and their associated data are described in the IDD while the IDF contains all the input data needed for simulation. The type of the object could be either numeric or alpha. The order of the data in IDF must match the order of data in IDD i.e. each data value in the IDF must go hand in hand with IDD object. Fig (3.6) and (3.7) below show the IDD and IDF examples of dry fluid cooler.

```
Fluidcooler:SingleSpeed,
  A1 , \field Name
       \required-field
       \type alpha
       \note fluidcooler name
  A2 , \field Water Inlet Node Name
       \required-field
       \type alpha
       \note Name of fluidcooler water inlet node
  A3 , \field Water Outlet Node Name
       \required-field
       \type alpha
       \note Name of fluidcooler water outlet node
  A4 , \field Performance Input Method
       \type Choice
       \key UAandDesignWaterFlowRate
       \key NominalCapacity
       \default NominalCapacity
       \note User can define fluidcooler thermal performance by specifying
       \note the fluidcooler UA and the Design Air Flow Rate, or by specifying
       \note the fluidcooler nominal capacity
  N1 , \field U-factor Times Area Value at Design Air Flow Rate
       \type real
       \units W/K
       \minimum> 0.0
       \maximum 2100000.0
       \autosizable
       \note Leave field blank if fluidcooler Performance Input Method is
       \note NOMINAL CAPACITY
```

```
N2 , \field Nominal Capacity
     \type real
     \units W
     \minimum> 0.0
     \note Nominal fluidcooler capacity
N3 , \field Design Entering Water Temperature
     \type real
     \units C
     \minimum> 0.0
     \ip-units F
N4 , \field Design Entering Air Temperature
     \type real
     \units C
     \minimum> 0.0
     \ip-units F
N5 , \field Design Entering Air Wet-bulb Temperature
     \type real
     \units C
     \minimum> 0.0
     \ip-units F
N6 , \field Design Water Flow Rate
     \type real
     \units m3/s
     \minimum> 0.0
     \autosizable
     \ip-units gal/min
N7 , \field Design Air Flow Rate
     \required-field
     \type real
     \units m3/s
     \minimum> 0.0
     \autosizable
N8 , \field Fan Power at Design Air Flow Rate
     \required-field
     \type real
     \units W
     \minimum> 0.0
     \autosizable
     \ip-units W
A5,  \field Fluid Name
      \note (water, ethylene glycol, etc.)
      \type object-list
      \object-list GlycolConcentrations
      \required-field
      \default water
N9,  \field Fluid Glycol Concentration
      \required-field
      \type real
      \units percent
      \minimum 0
      \maximum 100
      \note with the rewrite of fluid properties this parameter
      \note is no longer needed
A6 ; \field Outdoor Air Inlet Node Name
     \type alpha
     \note Enter the name of an outdoor air node
```

Fig 3.6: IDD file for dry fluid cooler

```
Fluidcooler:SingleSpeed,
    Big FluidCooler,                !- FLUIDCOOLER Name
    Condenser FluidCooler Inlet Node,  !- Water Inlet Node Name
    Condenser FluidCooler Outlet Node,  !- Water Outlet Node Name
    NominalCapacity,        !- FluidCooler Performance Input Method
    ,                        !- FluidCooler UA Value at Design Air Flow Rate {W/K}
    58601.,                  !- FluidCooler Nominal Capacity {W}
    51.67,                   !- Design Entering Water tempereture {C}
    35,                      !- Design Entering Air tempereture {C}
    25.6,                    !- Design Entering Air Wet-bulb tempereture {C}
    0.001388,                !- Design Water Flow Rate{m3/s}
    9.911,                   !- Design Air Flow Rate {m3/s}
    500,                !- Fan Power at Design Air Flow Rate {W}
    ethyleneGlycol40Percent, !- Fluid name
    40;                      !- fluid/glycol concentration {percent}
```

Fig 3.7: IDF file for dry fluid cooler

3.3.2 Implementation algorithm of fluid cooler models in EnergyPlus

For each fluid cooler module, there is one main/driver routine which calls other subroutines to provide different services to the main routine. The subroutines called by driver subroutines are discussed below:

- GetFluidCoolerInput

This subroutine obtains input data for fluid coolers and stores it in the data structure. After checking the conformity of inputs between IDD and IDF, this subroutine allocates the arrays and sets up report variables.

- InitFluidCooler

This subroutine initializes fluid cooler components at each environment, day, hour or timestep. Status flags are used to trigger initializations. Also the local simulation variables are updated with the latest node data.

- SingleSpeedFluidCooler and TwoSpeedFluidCooler

These subroutines simulate the operation of single and two speed fluid coolers respectively. The subroutine calculates the period of time required to meet a leaving water temperature set-point. It assumes that part-load operation represents a linear interpolation of two steady-state regimes i.e. fluid cooler ON and OFF. The period of time required to meet the leaving water temperature set-point is used to determine the required fan power and energy.

A RunFlag is passed by the upper level manager to indicate the ON/OFF status, or schedule, of the fluid cooler. If the fluid cooler is OFF, outlet water temperature and flow rate are passed through the model from inlet node to outlet node without intervention. Reports are also updated with fan power and energy being zero.

When the RunFlag indicates an ON condition for the fluid cooler, the mass flow rate and water temperature are read from the inlet node of the fluid cooler (water-side). The outdoor air dry-bulb and wet-bulb temperatures are used as the air-side entering conditions to the dry and evaporative fluid coolers respectively. The fluid cooler

fan is turned on and design parameters are used to calculate the leaving water temperature. If the calculated leaving water temperature is below the set-point, a fan run-time fraction is calculated and used to determine fan power. The fraction of time that the fluid cooler fans must operate is calculated as follows:

$$\omega = \frac{T_{set} - T_{wout,off}}{T_{wout,on} - T_{wout,off}} \qquad (3.1)$$

Where,

ω = Fan run time fraction

T = Temperature (°C)

Subscripts

w = water

out = outlet condition

off = Fluid cooler fan OFF

on = Fluid cooler fan ON

set = set-point

The average fan power is calculated by multiplying ω by the steady-state fan power specified as input. The leaving water temperature set-point is placed on the outlet node.

In the case of two speed fluid coolers, leaving water temperatures are calculated for low speed operation. If the calculated leaving water temperature is at or above the set-point, the fluid cooler fan is turned on 'high speed' and the routine is repeated. If the calculated leaving water temperature is below the set-point, a fan run-time fraction is calculated for the second stage fan and then the fan power is calculated. Eq. (3.2) shows the method of calculating fan run time fraction.

$$\omega = \frac{T_{set} - T_{wout,low}}{T_{wout,high} - T_{wout,low}} \tag{3.2}$$

The subscripts low and high stand for low speed and high speed fan operation respectively. The average fan power for the simulation time step is calculated for the two-speed fluid cooler as follows

$$P_{fan,avg} = \omega(P_{fan,high}) + (1 - \omega)(P_{fan,low}) \tag{3.3}$$

Where,

P_{fan} = Fan power (W)

If the calculated leaving water temperature is above the leaving water temperature set-point, the calculated leaving water temperature is placed on the outlet node and the fan runs at full power (High Speed Fan Power). Water mass flow rate is passed from inlet node to outlet node with no intervention.

```
SUBROUTINE TwoSpeedFluidCooler(FluidCoolerNum,FlowLock, RunFlag)

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

FanModeFrac             = 0.0

UAdesign        = SimpleFluidCooler(FluidCoolerNum)%LowSpeedFluidCoolerUA
AirFlowRate     = SimpleFluidCooler(FluidCoolerNum)%LowSpeedAirFlowRate
FanPowerLow     = SimpleFluidCooler(FluidCoolerNum)%LowSpeedFanPower

Call SimSimpleFluidCooler(FluidCoolerNum,WaterMassFlowRate,AirFlowRate, &
                                              UAdesign,OutletWaterTemp1stStage)

IF(OutletWaterTemp1stStage .LE. TempSetPoint)THEN
  ! Setpoint was met with pump ON and fan ON 1st stage, calculate fan mode
  ! fraction
  FanModeFrac = (TempSetPoint-OutletWaterTempOFF)/(OutletWaterTemp1stStage- &
                                                            OutletWaterTempOFF)
  FanPower        = FanModeFrac * FanPowerLow
  OutletWaterTemp = TempSetPoint
  Qactual         = Qactual * FanModeFrac
ELSE
  ! Setpoint was not met, turn on  FluidCooler 2nd stage fan
   UAdesign        = SimpleFluidCooler(FluidCoolerNum)%HighSpeedFluidCoolerUA
   AirFlowRate       = SimpleFluidCooler(FluidCoolerNum)%HighSpeedAirFlowRate
   FanPowerHigh      = SimpleFluidCooler(FluidCoolerNum)%HighSpeedFanPower

  Call SimSimpleFluidCooler(FluidCoolerNum,WaterMassFlowRate,AirFlowRate, &
                                              UAdesign,OutletWaterTemp2ndStage)

  IF((OutletWaterTemp2ndStage .LE. TempSetPoint).AND. UAdesign .GT. 0.0)THEN
    ! Setpoint was met with pump ON and fan ON 2nd stage, calculate fan mode
    ! fraction
    FanModeFrac  = (TempSetPoint- OutletWaterTemp1stStage)/ &
                              (OutletWaterTemp2ndStage-OutletWaterTemp1stStage)
    FanPower     = MAX((FanModeFrac * FanPowerHigh)  &
                                      + (1.d0- FanModeFrac)*FanPowerLow, 0.0D0)

    OutletWaterTemp = TempSetPoint
  ELSE
    ! Setpoint was not met,  FluidCooler ran at full capacity
    OutletWaterTemp = OutletWaterTemp2ndStage
    FanPower        = FanPowerHigh
  END IF

END IF

CpWater =GetSpecificHeatGlycol('WATER',Node(SimpleFluidCooler(FluidCoolerNum)%&
                 WaterOutletNodeNum)%Temp, SimpleFluidCooler(FluidCoolerNum)% &
                                             FluidIndex,'TwoSpeedFluidCooler')
Qactual = WaterMassFlowRate * CpWater * (Node(WaterInletNode)%Temp – &
                                                               OutletWaterTemp)

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .
. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .
```

```
RETURN
END SUBROUTINE TwoSpeedFluidCooler
```

This subroutine calls SimSimpleFluidCooler and SimSimpleEvapFluidCooler subroutines in dry and evaporative fluid cooler modules respectively to calculate outlet water temperature and heat transfer rate from fluid coolers. The subroutines for dry and evaporative fluid coolers are described below:

```
SUBROUTINE SimSimpleFluidCooler(FluidCoolerNum,WaterMassFlowRate,&
                                          AirFlowRate,UAdesign,OutletWaterTemp)


. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .


MdotCpWater =   WaterMassFlowRate * CpWater
AirCapacity =   AirMassFlowRate * CpAir
! calculate the minimum to maximum capacity ratios of airside and waterside
CapacityRatioMin = MIN(AirCapacity,MdotCpWater)
CapacityRatioMax = MAX(AirCapacity,MdotCpWater)
CapacityRatio    = CapacityRatioMin/CapacityRatioMax
! Calculate heat transfer coefficient and number of transfer units (NTU)
NumTransferUnits = UAdesign/CapacityRatioMin
ETA=NumTransferUnits**0.22d0
A=CapacityRatio*NumTransferUnits/ETA
effectiveness = 1.d0 - Exp((Exp(-A) - 1.d0) / (CapacityRatio / ETA))
! calculate water to air heat transfer
Qactual = effectiveness * CapacityRatioMin * (InletWaterTemp-InletAirTemp)
! calculate new exiting dry bulb temperature of airstream
OutletAirTemp = InletAirTemp + Qactual/AirCapacity
IF(Qactual .GE. 0.0)THEN
  OutletWaterTemp = InletWaterTemp - Qactual/ MdotCpWater
ELSE
  OutletWaterTemp = InletWaterTemp
END IF

RETURN
END SUBROUTINE SimSimpleFluidCooler
```

```
SUBROUTINE SimSimpleEvapFluidCooler(EvapFluidCoolerNum, WaterMassFlowRate,
                                         AirFlowRate,UAdesign,OutletWaterTemp)


INTEGER, PARAMETER  :: IterMax = 50      ! Maximum number of iterations allowed
REAL(r64), PARAMETER :: WetBulbTolerance  = 0.00001d0
! Maximum error for exiting wet-bulb temperature between iterations [delta K/K]
REAL(r64), PARAMETER :: DeltaTwbTolerance = 0.001d0                            !
Maximum error (tolerance) in DeltaTwb for iteration convergence [C]

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .
```

```
. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .



! initialize exiting wet bulb temperature before iterating on final solution
OutletAirWetBulb = InletAirWetBulb + 6.0

! Calcluate mass flow rates
MdotCpWater =   WaterMassFlowRate * CpWater
Iter = 0
DO WHILE ((WetBulbError.GT.WetBulbTolerance) .AND. (Iter.LE.IterMax) .AND. &
                                               (DeltaTwb.GT.DeltaTwbTolerance))
  Iter = Iter + 1
  OutletAirEnthalpy = PsyHFnTdbRhPb(OutletAirWetBulb,1.0d0,  &
                       SimpleEvapFluidCoolerInlet(EvapFluidCoolerNum)%AirPress)

  ! calculate the airside specific heat and capacity
  CpAirside = (OutletAirEnthalpy - InletAirEnthalpy)/(OutletAirWetBulb- &
                                                                InletAirWetBulb)
  AirCapacity = AirMassFlowRate * CpAirside

  ! calculate the minimum to maximum capacity ratios of airside and waterside
  CapacityRatioMin = MIN(AirCapacity,MdotCpWater)
  CapacityRatioMax = MAX(AirCapacity,MdotCpWater)
  CapacityRatio    = CapacityRatioMin/CapacityRatioMax

  ! Calculate heat transfer coefficient and number of transfer units (NTU)
  UAactual = UAdesign*CpAirside/CpAir
  NumTransferUnits = UAactual/CapacityRatioMin
  ! calculate heat exchanger effectiveness
  IF (CapacityRatio.LE.0.995d0)THEN
    effectiveness = (1.d0-EXP(-1.0d0*NumTransferUnits*(1.0d0-CapacityRatio)))/&
     (1.0d0-CapacityRatio*EXP(-1.0d0*NumTransferUnits*(1.0d0-CapacityRatio)))
  ELSE
    effectiveness = NumTransferUnits/(1.d0+NumTransferUnits)
  ENDIF

  ! calculate water to air heat transfer and store last exiting WB temp of air
  Qactual = effectiveness * CapacityRatioMin * (InletWaterTemp-InletAirWetBulb)
  OutletAirWetBulbLast = OutletAirWetBulb
  ! calculate new exiting wet bulb temperature of airstream
  OutletAirWetBulb = InletAirWetBulb + Qactual/AirCapacity
  ! Check error tolerance and exit if satisfied
  DeltaTwb = ABS(OutletAirWetBulb - InletAirWetBulb)
  ! Add KelvinConv to denominator below convert OutletAirWetBulbLast to Kelvin
  ! to avoid divide by zero.
  ! Wet bulb error units are delta K/K
  WetBulbError = ABS((OutletAirWetBulb - OutletAirWetBulbLast)/        &
                                             (OutletAirWetBulbLast+KelvinConv))

END DO

IF(Qactual .GE. 0.0)THEN
  OutletWaterTemp = InletWaterTemp - Qactual/ MdotCpWater
ELSE
  OutletWaterTemp = InletWaterTemp
END IF

RETURN
END SUBROUTINE SimSimpleEvapFluidCooler
```

- SizeFluidCooler

The fluid cooler UA value is calculated in this subroutine. The method used to calculate UA is described in Fig (3.3). First, the UA value is guessed on the basis of design capacity of the fluid cooler and capacity of the fluid cooler which is based on this UA is calculated. If the residual of the capacity is less than the specified accuracy then the desired UA value is obtained. Otherwise new UA value is calculated by using regula falsi and iterations are performed until the solution converges to a UA value for which residual is less than the accuracy.

```
IF (SimpleFluidCooler(FluidCoolerNum)%PerformanceInputMethod == &
                                                       'NOMINALCAPACITY') THEN

  IF (SimpleFluidCooler(FluidCoolerNum)%DesignWaterFlowRate >= &
                                                       SmallWaterVolFlow) THEN

    DesFluidCoolerLoad = SimpleFluidCooler(FluidCoolerNum)% &
                                                     FluidCoolerNominalCapacity

    Par(1) = DesFluidCoolerLoad
    Par(2) = REAL(FluidCoolerNum,r64) ! FluidCooler number
    Par(3) = GetDensityGlycol('WATER',InitConvTemp, &
                     SimpleFluidCooler(FluidCoolerNum)%FluidIndex,CalledFrom) &
                     * SimpleFluidCooler(FluidCoolerNum)%DesignWaterFlowRate
                     ! design water mass flow rate
    Par(4) = SimpleFluidCooler(FluidCoolerNum)%HighSpeedAirFlowRate
                     ! design air volume flow rate
    Par(5) = GetSpecificHeatGlycol('WATER',SimpleFluidCooler(FluidCoolerNum)% &
             DesignEnteringWaterTemp, SimpleFluidCooler(FluidCoolerNum)% &
                      FluidIndex,CalledFrom)
    UA0 = 0.0001d0 * DesFluidCoolerLoad  ! Assume deltaT = 10000K (limit)
    UA1 = DesFluidCoolerLoad             ! Assume deltaT = 1K
    SimpleFluidCoolerInlet(FluidCoolerNum)%WaterTemp =   &
         SimpleFluidCooler(FluidCoolerNum)%DesignEnteringWaterTemp
                     ! design inlet water temperature
    SimpleFluidCoolerInlet(FluidCoolerNum)%AirTemp =   &
         SimpleFluidCooler(FluidCoolerNum)%DesignEnteringAirTemp
                     ! design inlet air dry-bulb temp
    SimpleFluidCoolerInlet(FluidCoolerNum)%AirWetBulb =   &
         SimpleFluidCooler(FluidCoolerNum)%DesignEnteringAirWetbulbTemp
                     ! design inlet air wet-bulb temp
    SimpleFluidCoolerInlet(FluidCoolerNum)%AirPress = StdBaroPress
    SimpleFluidCoolerInlet(FluidCoolerNum)%AirHumRat =   &
         PsyWFnTdbTwbPb(SimpleFluidCoolerInlet(FluidCoolerNum)%AirTemp,  &
                        SimpleFluidCoolerInlet(FluidCoolerNum)%AirWetBulb,  &
                        SimpleFluidCoolerInlet(FluidCoolerNum)%AirPress)
    CALL SolveRegulaFalsi(Acc, MaxIte, SolFla, UA, &
```

```
                                     SimpleFluidCoolerUAResidual,UA0, UA1, Par)
    IF (SolFla == -1) THEN
      CALL ShowSevereError('Iteration limit exceeded in calculating &
                                                               FluidCooler UA')
      CALL ShowFatalError('Autosizing of  FluidCooler UA failed for &
                   FluidCooler '//TRIM(SimpleFluidCooler(FluidCoolerNum)%Name))
    ELSE IF (SolFla == -2) THEN
      CALL ShowSevereError('Bad starting values for UA')
      CALL ShowFatalError('Autosizing of  FluidCooler UA failed for &
                   FluidCooler '//TRIM(SimpleFluidCooler(FluidCoolerNum)%Name))
    ENDIF
    SimpleFluidCooler(FluidCoolerNum)%HighSpeedFluidCoolerUA = UA
  ELSE
    SimpleFluidCooler(FluidCoolerNum)%HighSpeedFluidCoolerUA = 0.0
  ENDIF

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .
. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

ENDIF
```

```
FUNCTION SimpleFluidCoolerUAResidual(UA, Par) RESULT (Residuum)

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

. . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . . .

! SUBROUTINE ARGUMENT DEFINITIONS:
REAL(r64), INTENT(IN)  :: UA                         ! UA of FluidCooler
REAL(r64), INTENT(IN), DIMENSION(:), OPTIONAL :: Par
! par(1) = design FluidCooler load [W]
! par(2) = FluidCooler number
! par(3) = design water mass flow rate [kg/s]
! par(4) = design air volume flow rate [m3/s]
! par(5) = water specific heat [J/(kg*C)]
REAL(r64)         :: Residuum ! residual to be minimized to zero

! FUNCTION LOCAL VARIABLE DECLARATIONS:
INTEGER   :: FluidCoolerIndex                ! index of this FluidCooler
REAL(r64) :: OutWaterTemp                    ! outlet water temperature [C]
REAL(r64) :: Output                          ! FluidCooler  output [W]

FluidCoolerIndex = INT(Par(2))
CALL SimSimpleFluidCooler(FluidCoolerIndex,Par(3),Par(4),UA,OutWaterTemp)
Output = Par(5)*Par(3)*(SimpleFluidCoolerInlet(FluidCoolerIndex)%WaterTemp – &
                                                                OutWaterTemp)
Residuum = (Par(1) - Output) / Par(1)
RETURN
END FUNCTION SimpleFluidCoolerUAResidual
```

- UpdateRecords

This subroutine is used to pass the results to outlet node. Outlet water temperature and water mass flow rates are passed to the outlet node. This subroutine also issues warning in the case of outlet water temperature being lower than the loop temperature, water mass flow rate being greater than loop maximum flow rate or lower than loop minimum flow rate.

CHAPTER IV

PERFORMANCE EVALUATION OF ENERGYPLUS FLUID COOLER MODELS

One major problem which was encountered while modeling fluid coolers in EnergyPlus was the insufficient catalog data. More often than not, manufacturers don't provide enough data. The absence of required design parameters creates problem for modeling. The objective of this chapter is to evaluate the impact of various design parameters in the results of simulation. This will help to recognize the parameters which are really important from simulation point of view. The parameters for which model is very sensitive must be input with least errors while the parameters for which model is very less sensitive can be guessed by using engineering judgment. As discussed in chapter in chapter 3, overall heat transfer coefficient (UA) is the single characterizing parameter for the fluid cooler models. So first the sensitivity of UA with respect to design parameters is discussed and then the sensitivity of simulation result with respect to UA is considered.

4.1 Model sensitivity to input parameters

Spitler et.al (1989) illustrates the use of influence coefficient to determine the impact of simulation input parameters on simulation results. Influence coefficients are partial derivatives of one variable with respect to another variable. In the context of simulation, they are used to quantify the effect of input variables over simulation results. Mathematically influence coefficient is expressed by:

$$\text{Influence coefficient} = \frac{\partial(\text{result})}{\partial(\text{input parameter})} \tag{4.1}$$

The impact of the perturbation of input parameters in the results is quantified by calculating dimensional influence coefficient. This dimensional coefficient is then multiplied with the estimated error in the input to obtain the corresponding error in simulation result. The method of calculating dimensional coefficients is outlined below.

$$\frac{\partial(\mathrm{R}^*)}{\partial(\mathrm{P})} = \frac{\Delta(\mathrm{R}^*)}{\Delta(\mathrm{P})} \tag{4.2}$$

$$\mathrm{R}^* = \mathrm{R}^*{}_{\mathrm{bc}} - \mathrm{R}^*{}_{\Delta} = \frac{\mathrm{R}_{\mathrm{bc}} - \mathrm{R}_{\Delta}}{\mathrm{R}_{\mathrm{bc}}} \tag{4.3}$$

Where,

P = parameter

R = result

* = non-dimensionality

bc = base case

Δ = value for perturbed case

4.1.2 Methodology

As discussed in chapter 3, the fluid coolers are characterized by a single parameter UA i.e. their overall heat transfer coefficient. Fig 3.5 describes the method used by EnergyPlus to calculate UA from design parameters. Inlet air dry-bulb temperature, inlet water temperature, inlet air dry-bulb temperature, water flow rate, design capacity and air flow rate are the design parameters which are used to determine overall heat transfer coefficient at design conditions. So to understand the impact of design parameters in simulation results a three step procedure is used. Fluid cooler fan energy consumption is used as the simulation output variable. The steps are as follows:

1) Sensitivity of UA with respect to change in design parameter is determined i.e. error in different input parameters generate how much error in UA value.
2) A parametric study is performed to understand the impact of set-point at annual fan energy consumption.
3) Finally a location wise parametric study is performed.

The detailed description of the steps is given below. The example building and system description is given in chapter 6.

4.1.3 Sensitivity of UA for change in design parameters

Table (4.1) and (4.2) show the sensitivity of UA with respect to change in design parameters. From the tables it is clear that dry-bulb has negligible influence on the evaporative fluid cooler results and wet-bulb has negligible influence on evaporative fluid cooler results. Also design air flow rate does not seem to play any critical role in causing error in UA value.

Table 4.1 UA sensitivity for dry fluid cooler

Dry Fluid Cooler Design Parameter	Base –case Value	Dimensional I.C.	Est. Error (Parameter)	Est. Error (Result)
Design Inlet air dry-bulb temp.	37.78°C	0.17881292 °C^{-1}	5 °C	±89.4065%
Design Inlet air wet-bulb temp.	30°C	0.000259776 °C^{-1}	5 °C	±0.129888%
Design Air flow rate	9.675(m^3/s)	0.0673012 (m^3/s)$^{-1}$	2 m^3/s	±0.0252 %
Design Water flow rate	4.10E-03 (m^3/s)	11.176952 (m^3/s)$^{-1}$	0.001 m^3/s	±1.1177%
Design Capacity	93753W	1.84109E-05 W^{-1}	9375 W	±17.26019%
Design Inlet Water temp.	54.44 °C	0.084153265 °C^{-1}	5 °C	±42.07663%

Table 4.2 UA sensitivity for evaporative fluid cooler

Evaporative Fluid Cooler Design Parameter	Base –case Value	Dimensional I.C.	Est. Error (Parameter)	Est. Error (Result)
Design Inlet air dry-bulb temp.	35°C	0.000494709°C^{-1}	5 °C	±0.247354%
Design Inlet air wet-bulb temp.	25.6°C	0.25002285 °C^{-1}	5 °C	±125.011%
Design Air flow rate	7.164(m^3/s)	0.0164129 (m^3/s)$^{-1}$	2 m^3/s	±3.28258%
Design Water flow rate	3.98E-03 (m^3/s)	125.17795 (m^3/s)$^{-1}$	0.001 m^3/s	±12.5178%
Design Capacity	73854W	1.94928E-05 W^{-1}	7385	±14.39542%
Design Inlet Water temp.	35 °C	0.1109253 °C^{-1}	5 °C	±55.46267%

4.2 Parametric study with different set-point temperatures

As discussed in chapter 3, set-point for the fluid cooler can either be a fixed set-point temperature or the outdoor air dry/wet-bulb temperature depending upon the requirements for a particular application. If no fixed value of the set-point is provided then outdoor dry-bulb and wet-bulb temperatures can be used as the set-points for dry and evaporative fluid coolers respectively. This section discusses the change in annual fan energy consumption of fluid coolers at different set-point temperatures. Table 4.3 and 4.4 show the results of the study. From the tables it is clear that annual fan energy consumption is more sensitive to UA value at higher set-point temperatures.

Table 4.3 Change in dry fluid cooler fan energy consumption at different set-points

Dry Fluid Cooler Set-Point	Annual fan energy consumption (J)		% change in results
	(UA=10674 W/K)	(UA=9674 W/K)	
95°F (35 °C)	3.60E+08	3.78E+08	-5.14673 %
90°F (32.22 °C)	4.75E+08	4.95E+08	-4.20965 %
85°F (29.44 °C)	6.09E+08	6.27E+08	-2.99141 %
80°F (26.67 °C)	7.44E+08	7.61E+08	-2.2531 %
75°F (23.89 °C)	8.70E+08	8.82E+08	-1.36968 %
70°F (21.11°C)	9.59E+08	9.67E+08	-0.80211 %
Out dry-bulb	3.78E+08	3.78E+08	0

Table 4.4 Change in evaporative fluid cooler fan energy consumption at different set-points

Evaporative Fluid Cooler Set-Point	Annual fan energy consumption (J)		% change in results
	(UA=2753 W/K)	(UA=2503 W/K)	
95°F (35 °C)	1.14E+10	1.23E+10	-7.50962 %
90°F (32.22 °C)	1.35E+10	1.46E+10	-7.44074 %
85°F (29.44 °C)	1.67E+10	1.80E+10	-7.36456 %
80°F (26.67 °C)	2.23E+10	2.39E+10	-7.25904 %
75°F (23.89 °C)	3.21E+10	3.40E+10	-5.88973 %
70°F (21.11°C)	4.45E+10	4.62E+10	-3.81421 %
Out wet-bulb	6.98E+10	7.02E+10	-0.55468 %

At higher set-point temperatures, fluid cooler with higher UA value works for lesser time to meet the set-point as compared to fluid cooler with lower UA value. As the set-point reduces the time for which the fluid coolers operates increases. So the change in UA, when the set-point is low, does not cause significant change in results because fluid cooler is not able to meet the set-point and fan runs almost all the time. The tables also show that if the set-point is taken as outdoor dry or wet-bulb temperature the fan energy consumption changes negligibly. The reason for the negligible change is that the outdoor dry/wet-bulb temperatures are the minimum temperatures that the fluid coolers can achieve. Meeting these set-points require very high UA values and thus much higher effectiveness (of the order of 1) which is not the case with the fluid coolers. So the set-point is never met and fluid coolers keep running all the time. For this case, even changing the UA value by a large amount does not cause any significant change in the output as the set-point is still not met.

4.3 Parametric study at different locations

It is clear from tables (4.3) and (4.4), that the fan energy consumption is more sensitive to UA value at higher set-point temperatures. The study of section 4.2 is extended to cover for different locations. Five different locations in USA are chosen and annual fan energy consumption was calculated at each location. The UA values of the fluid coolers are same as shown in tables (4.3) and (4.4). The simulations were carried out for two different set-point temperatures 85°F (29.44°C) and outdoor dry-bulb (for dry fluid cooler) or wet-bulb (for evaporative fluid cooler) temperatures. Fig (4.1) and (4.2) show the results of the parametric study. The figures substantiate the previously drawn conclusion that when the set-point is outdoor dry or wet-bulb temperatures, the change in UA causes negligible change in annual fan energy consumption because the fluid cooler is not able to meet the set-point for both the UA values. So it runs all the time for both UA values. For a fixed set-point, the percentage change in fan energy consumption for different locations is shown below.

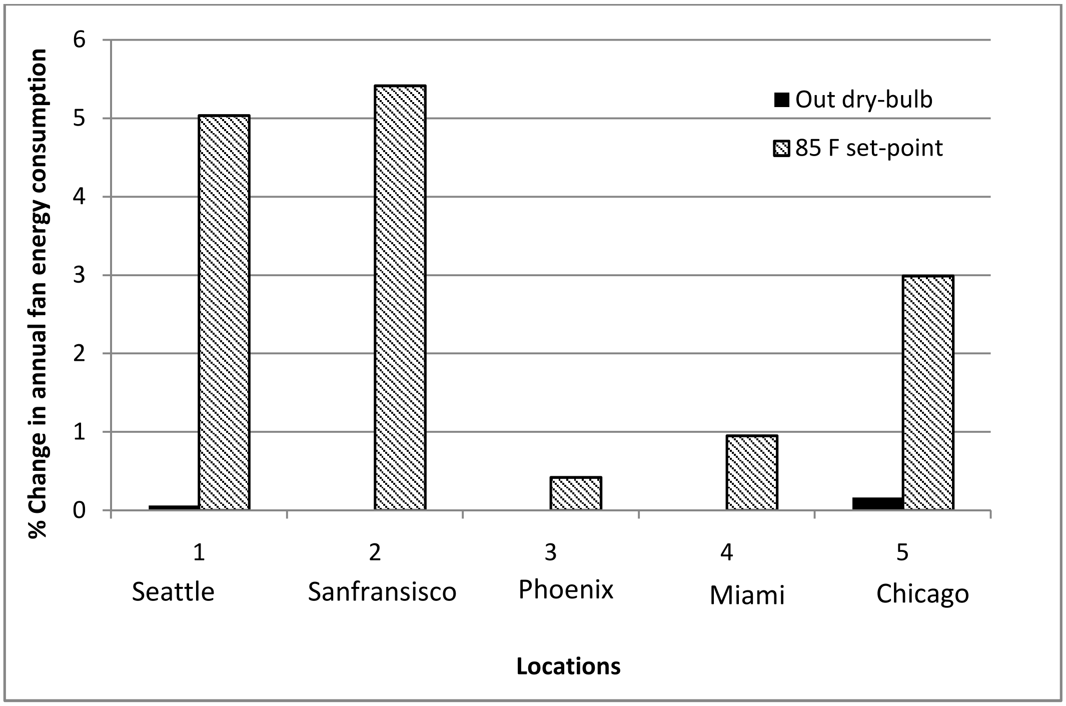


Fig 4.1 Comparison of % change in dry fluid cooler fan energy consumption due to change in UA value at different locations for two different set-points

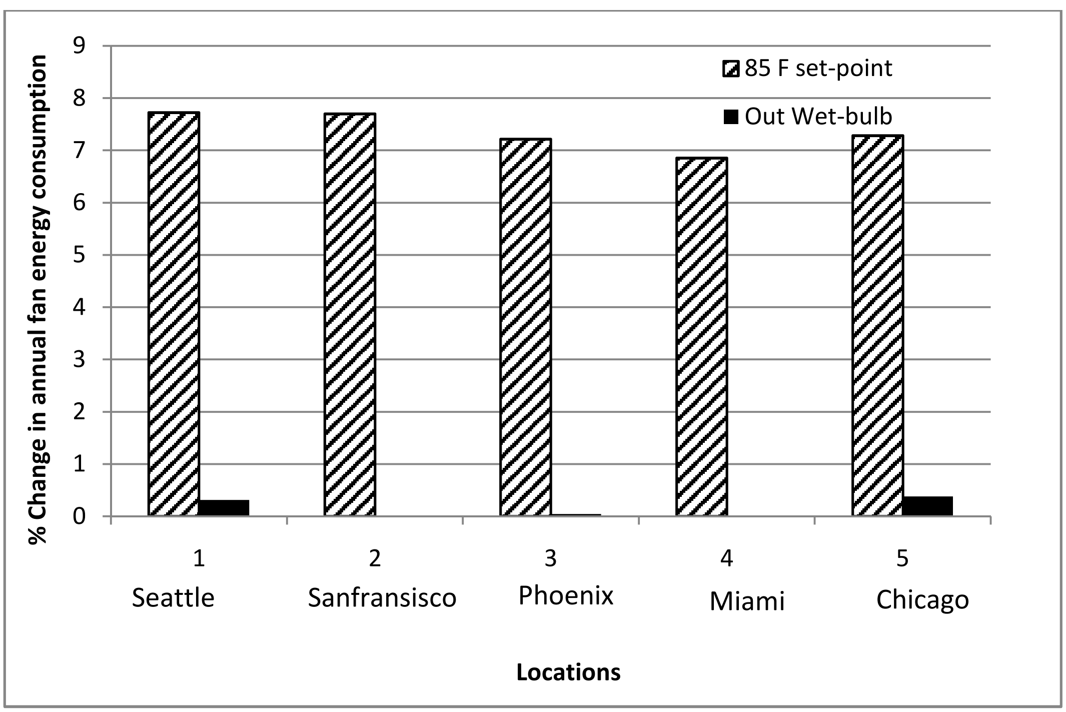


Fig 4.2 Comparison of % change in evaporative fluid cooler fan energy consumption due to change in UA value at different locations for two different set-points

4.4 Summary of results

The model sensitivity analysis can be summarized as follows:

1) In the absence of sufficient manufacturer's data, dry-bulb temperature for evaporative fluid cooler and wet-bulb temperature for dry fluid cooler can be guessed with negligible error in the simulation results.
2) If the set-point temperature is chosen as dry or wet-bulb temperature for dry and evaporative fluid cooler respectively, then it is possible that significant change in UA value will cause negligible change in the simulation results. The reason for this behavior is that the fluid cooler UA value has to be very large (effectiveness equal to one) to reach to the set-point. If the fluid cooler UA value is not that high it will not meet the set-point. In this case, if the UA value of the fluid cooler will be changed it will still not meet the set-point. So the fan which was running at full speed will continue to do so and fan energy consumption will remain unchanged.
3) Change in UA value causes more difference in the simulation results at higher set-point temperatures than at lower set-points. In other words, if the set-point for the fluid cooler is increased, the magnitude of difference in simulation results for the same change in UA will increase.
4) Inlet air wet-bulb temperature, design air flow rate and design water flow rate have very less impact on the fan energy consumption in the case of dry fluid coolers. While in the case of evaporative fluid coolers, inlet air dry-bulb temperature and design air flow rate have negligible effect over the fan energy consumption.

5) Fluid cooler UA value must be carefully chosen when the set-point temperature is defined by the users. Because as the set-point temperature increases, the simulation results become more sensitive to UA.

CHAPTER V

MODEL VERIFICATION

The published catalog data for fluid coolers are mostly insufficient. Moreover, the data is available only for one rating point for a particular fluid cooler model. Very few fluid cooler manufacturers provide the data for part load conditions. Some times, design input parameters e.g. inlet air dry-bulb temperatures, inlet air wet-bulb temperature, air flow rate etc. are missing. In this chapter, evaporative fluid cooler models are verified by using Baltimore Aircoil's catalog data. The data is obtained from their website for multiple rating points. For dry fluid coolers, because of lack of catalog data, the model is verified by using HVACSIM+ dry fluid cooler model (Type 762). Evaporative fluid cooler model is also verified by using Lebrun model, discussed earlier in chapter 2, which is implemented in VBA.

5.1 Evaporative fluid cooler: Comparison with published data sets

Data for evaporative fluid coolers is much more extensively and readily available as compare to dry fluid coolers. Evaporative fluid cooler's data is taken from Baltimore Aircoil

Co. website. The data is shown in table 5.1. Though the data was available for more than one rating point, the dry-bulb temperature, which is needed by EnergyPlus model, was missing.

Table 5.1 Catalog data for Baltimore Aircoil closed circuit cooling tower (VF1-009-12G)

Variable definition	Inlet Air wet-bulb temperature	Air flow rate	Cooling Capacity	Inlet water temperature	Water flow rate
UNITS	(°C)	(m^3/s)	(KW)	(°C)	(m^3/s)
1	26.67	2.69	17.58	35.00	7.57E-04
2	25.56	2.69	23.44	35.00	1.01E-03
3	26.67	2.69	35.16	38.89	1.26E-03
4	22.22	2.69	39.55	35.00	1.70E-03
5	26.67	2.69	40.28	46.11	6.94E-04
6	25.56	2.69	42.19	38.89	1.51E-03
7	25.56	2.69	43.94	46.11	7.57E-04
8	22.22	2.69	56.25	38.89	2.02E-03
9	22.22	2.69	62.26	46.11	1.07E-03

In the last chapter it was shown that dry-bulb temperature has negligible effect on the results of evaporative fluid cooler model. To verify again, a parametric study with dry-bulb temperature ranging from (80 to 95° F) is performed in EnergyPlus to evaluate the significance of the missing dry-bulb temperature input variable. The results of the study are shown in Appendix A. The study has verified that dry-bulb temperature has a negligible effect on the results of the evaporative fluid cooler model i.e. outlet water temperature and the capacity of the evaporative fluid cooler changed slightly for the entire range of the dry-bulb temperatures. This is because of the fact that the EnergyPlus evaporative fluid cooler model uses wet-bulb temperature for the effectiveness calculations. Incoming moist air is assumed as a fictitious perfect gas. The dry-bulb temperature of the fictitious gas is taken as wet-bulb temperature of incoming air. This assumption reduces the model to classic counter flow heat exchanger and the effectiveness of the heat exchanger is then calculated. The dry-bulb temperature is used only to calculate specific heat and density of air which don't vary too much in the dry-bulb range stated above. Since there was negligible difference in the output as a result of changing dry-bulb temperature, 90° F dry-bulb temperature is chosen to make up for the missing dry-bulb data. Each individual rating point is used as input one by one, corresponding UA value is calculated by EnergyPlus and then simulations are performed. The capacity obtained from the model is compared with the published capacity. As fig. 5.1 shows, the error between the cooling capacities published by manufacturer and calculated from EnergyPlus is less than 0.025 %.

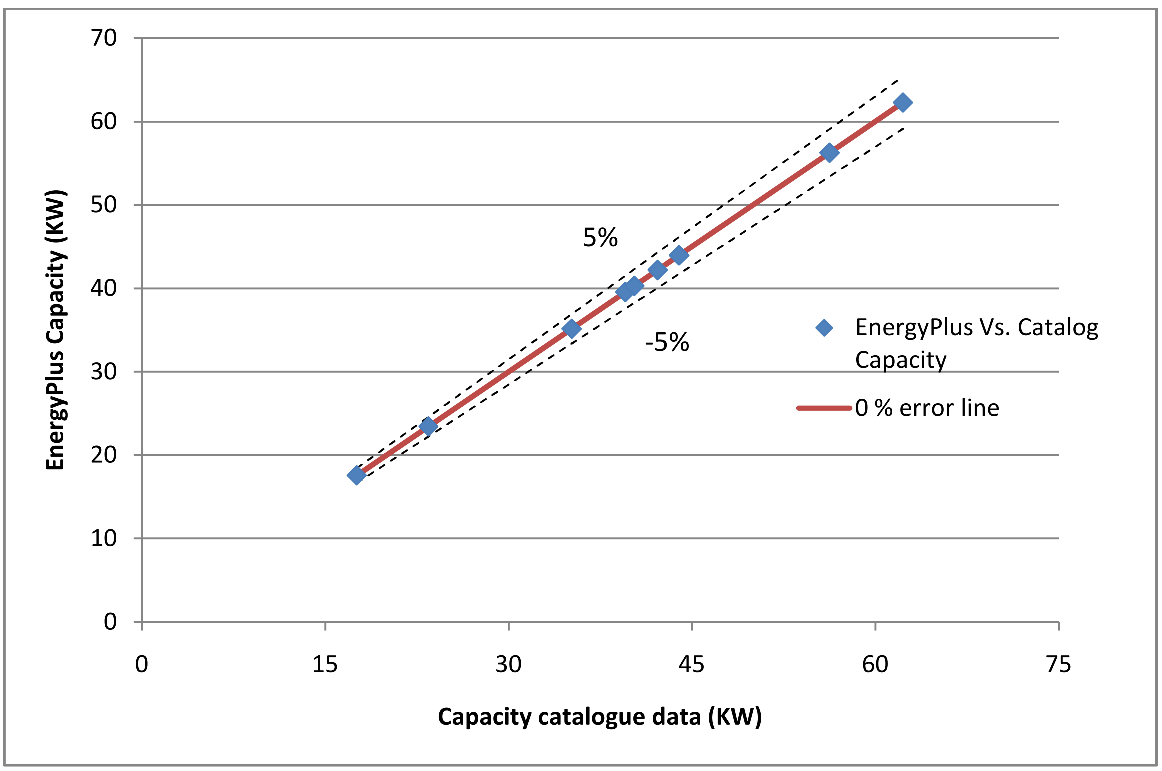


Fig. 5.1 Catalogue data Vs EnergyPlus capacity (KW)

5.2 Dry fluid cooler: Comparison between EnergyPlus and HVACSIM+ models

Xiaowei Xu (2007) developed dry fluid cooler model (Type 762) for HVACSIM+. Both HVACSIM+ and EnergyPlus dry fluid cooler models use classic heat exchanger (Ɛ-NTU) equation to calculate effectiveness of the heat exchanger. Input requirements of both the models are same with the main difference that EnergyPlus model can calculate UA value if not given by the user while HVACSIM+ model essentially needs UA value as input parameter.

For comparison, first EnergyPlus simulation is run for a design day (21 July) using weather data of Chicago. The description of the building and the system are given in detail in chapter 6. Dry fluid cooler's outlet water temperature and capacity are obtained as a result of simulation. Then HVACSIM+ dry fluid model (Type 762) is simulated for the same input conditions which are used for EnergyPlus simulation. The boundary file of HVACSIM+ dry fluid cooler model had the same inlet air and water mass flow rates and same inlet air and water temperatures as in the EnergyPlus inputs. The UA value of the fluid cooler calculated by EnergyPlus is taken as input parameter to HVACSIM+. The simulations were performed and the results were compared. The error between the results was initially high. It was found that there were bugs in the source code of dry fluid cooler model of HVACSIM+. The heat transfer equation and the outlet water temperature calculation equations were based on specific heat (Cp) of the both the fluids while they should be based on capacity flow rates (M*Cp) of the fluids. The erroneous lines of the code are shown below:

Present code (Incorrect)

```
Twex=Twsu-cpMoistAir*(Tdbex-Tdbsu)/C_fluid
```

```
Q=C_fluid*(Twsu-Twex)
```

Correct code

```
Twex=Twsu-CrAir*(Tdbex-Tdbsu)/CrW
Q=CrW*(Twsu-Twex)
```

The internal variables are defined as follows:

cpMoistAir: Moist air specific heat (J/kg/K)

C_fluid: Fluid specific heat (J/kg/K)

CrW: Water heat capacity flow rate (W/K)

CrAir: Air heat capacity flow rate (W/K)

The bugs were fixed but still the results were 3-4% off. Then some intermediate variables are set to investigate the problem further and figure out the cause of this difference. It was found that specific heat capacity (Cp) calculated by HVACSIM+ is not in agreement with specific heat capacity (Cp) calculated by EnergyPlus. For example, the Cp of the water calculated by HVACSIM+ for 39.2°C and 24.5°C entering water temperature was found to be 3444.43 (J/kg-K) and 3617 (J/kg-K) respectively while the values obtained from the EnergyPlus calculations and tables are approximately 4180 (J/kg-K) for both the cases. The correct Cp value (i.e. 4180 (J/kg-K)) is hardwired in HVACSIM+ model and simulation was run again. Table 5.2 shows the input parameters and simulation variables used by both the models. The process fluid is water. As shown in Fig 5.2 the error between the capacities calculated by HVACSIM+ and EnergyPlus is less than 0.3 % error for a very large range of input conditions.

Table 5.2 Input parameters and simulation outputs of HVACSIM+ and EnergyPlus dry fluid cooler models

					EnergyPlus		HVACSIM+	
Inlet Air Temp (°C)	Inlet Water Temp (°C)	Water Mass Flow rate (m^3/s)	Air Mass Flow rate (m^3/s)	UA (W/K)	Q (W)	Outlet Water Temp (°C)	Q (W)	Outlet Water Temp (°C)
21.71	30.78	1.39	11.34	7233.24	32117.3	25.24	32078	25.25
21.71	28.76	1.39	11.34	7233.24	24975.4	24.46	24934.2	24.46
21.71	26.71	1.39	11.34	7233.24	17692.1	23.66	17649	23.66
21.71	26.05	1.39	11.34	7233.24	15377.8	23.40	15361.9	23.4
21.87	24.52	1.39	11.33	7233.24	9366.3	22.90	9356.8	22.9
21.87	25.15	1.39	11.33	7233.24	11600.2	23.15	11588.1	23.15
22.98	27.35	1.39	11.29	7233.24	15460.3	24.68	15443.7	24.7
23.21	27.49	1.39	11.28	7233.24	15165.6	24.88	15194.8	24.88
24.44	28.66	1.39	11.23	7233.24	14939.8	26.09	14901.4	26.1
25.24	29.05	1.39	11.20	7233.24	13476.5	26.73	13447.8	26.75
26.11	29.98	1.39	11.17	7233.24	13640.8	27.63	13618.2	27.62
27.02	31.41	1.39	11.14	7233.24	15489.0	28.74	15483.7	28.75
28.18	32.79	1.39	11.09	7233.24	16243.4	29.99	16199.8	29.99
28.75	33.47	1.39	11.07	7233.24	16634.8	30.60	16628.1	30.6
30.32	35.55	1.39	11.01	7233.24	18429.0	32.38	18398.5	32.39
31.18	37.07	1.39	10.98	7233.24	20747.9	33.49	20721.5	33.5
21.71	39.17	1.39	11.34	7233.24	61821.9	28.52	61770.3	28.51

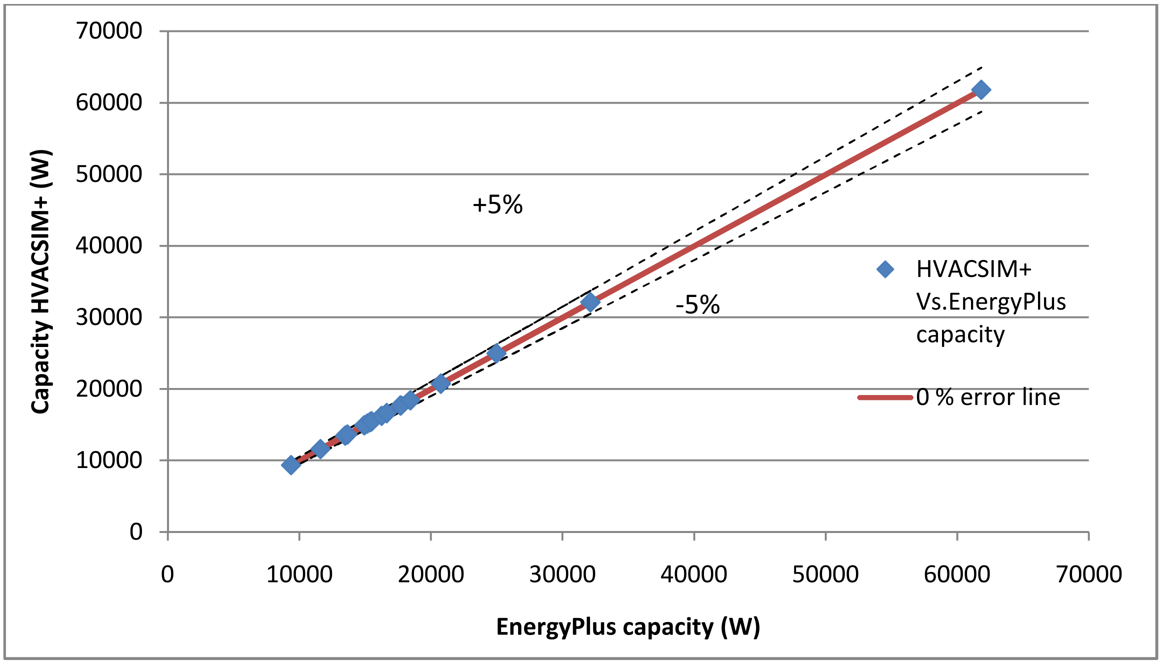


Fig. 5.2 EnergyPlus Vs HVACSIM+ capacity (W)

5.3 Evaporative fluid cooler: comparison between EnergyPlus and Lebrun model

Lebrun model, discussed earlier in chapter 2, is implemented in VBA to verify the EnergyPlus evaporative fluid cooler model. The description of the VBA implementation along with the source code is given in Appendix B. Lebrun model uses the catalog data shown in table 5.1 to estimate the parameters $R_{a,n}$, $R_{w,n}$ and exponents m andn. Table 5.3 shows the estimated parameters by Lebrun model.

Table 5.3 Parameters estimation results of Lebrun model

Parameters	Estimated value by Lebrun model
$R_{a,n}$	0.7562354263
$R_{w,n}$	0.5036001336
m	0.0077285252
n	0.716397443

Comparison between the results of the two models is shown in Fig. 5.3. The models are in reasonably good agreement with the maximum error being less than ± 8% .Since the Lebrun model itself differs by ± 7.5 % with respect to manufacturer's data, ± 8 % error is justified

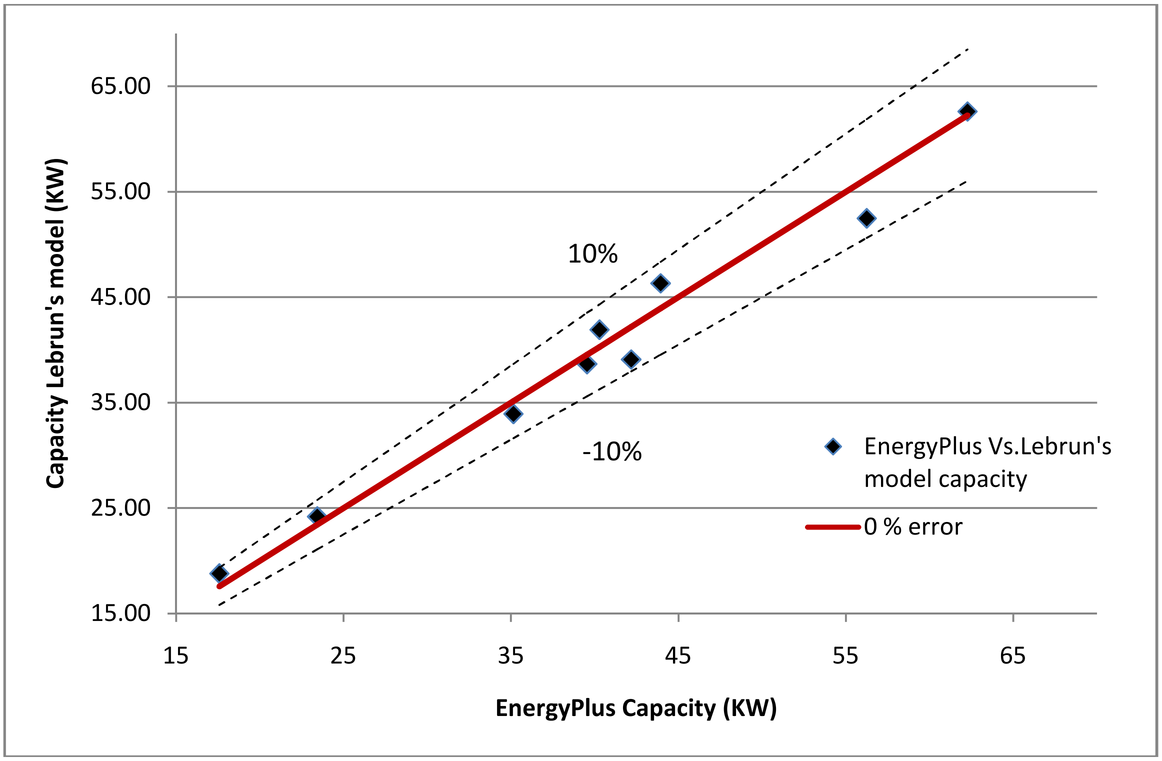


Fig. 5.3 EnergyPlus Vs Lebrun model capacity (KW)

5.4 Summary of results

In this chapter EnergyPlus fluid cooler models were compared with other fluid cooler models and the results were presented. Due to scarcity of manufacturer's data required input variables were not available for complete validation. For evaporative fluid cooler model, Baltimore Aircoil provides data for more than one rating. The dry-bulb temperature was missing in the data set. A parametric study was run in EnergyPlus to understand the impact of dry-bulb temperature in the evaporative fluid cooler results and to make sound engineering judgment about the missing variable. The results of the parametric study are shown in Appendix C. Because of negligible difference in the outputs for the varying range (80 to 95° F), a representative value of 90° F is taken as dry-bulb and the simulation was performed. The error between the catalog data capacity and EnergyPlus capacity was less than 0.025%.

HVACSIM+ dry fluid cooler model had some errors in the source code. After the errors were fixed the model had 3-4% error in the capacities. It was found that specific heat of the fluid calculated by HVACSIM+ model is off than the values obtained from the tables for the same conditions. The correct specific heat value is hardwired in the HVACSIM+ source code and the error reduced to within 0.5%.

Finally, the Lebrun model was also compared with evaporative fluid cooler model. Evaporative fluid cooler model's capacity is off by ± 8 % with respect to Lebrun model capacity. This is reasonably good agreement since the accuracy of Lebrun model outputs as published in the paper is ± 7.5 % when compared with manufacturer's data.

CHAPTER VI

MODEL IMPLEMENTATION VERIFICATION

This chapter discusses the EnergyPlus loop models, example building and the system used for implementation and verification. Different configurations of fluid coolers in the loop are tested and the results are analyzed.

6.1 The EnergyPlus model

In this section a general introduction of the EnergyPlus loop models is presented. Fig 6.1 shows a standard EnergyPlus loop diagram with different configurations. In EnergyPlus, Zones, system and plant are simultaneously solved at each time step. At the beginning of the simulation, the zone heat balance is performed to calculate the heating/cooling load on the HVAC system and plant at every time step. The cooling coil, chiller and fluid cooler interact with each other via the fluid loop. The fluid flow rate must satisfy mass continuity in each loop. The temperature at the condenser loop supply side outlet is updated to the condenser demand inlet, and the temperature at the plant supply outlet is updated to cooling coil inlet node. The chiller works to meet the load on the cooling coil, and the fluid cooler works to meet the condenser side cooling demand of the chiller. The dotted portion in fig 6.1 is replaced by one of the three configurations shown below for the present study. The effect of these different configurations on the condenser loop is discussed.

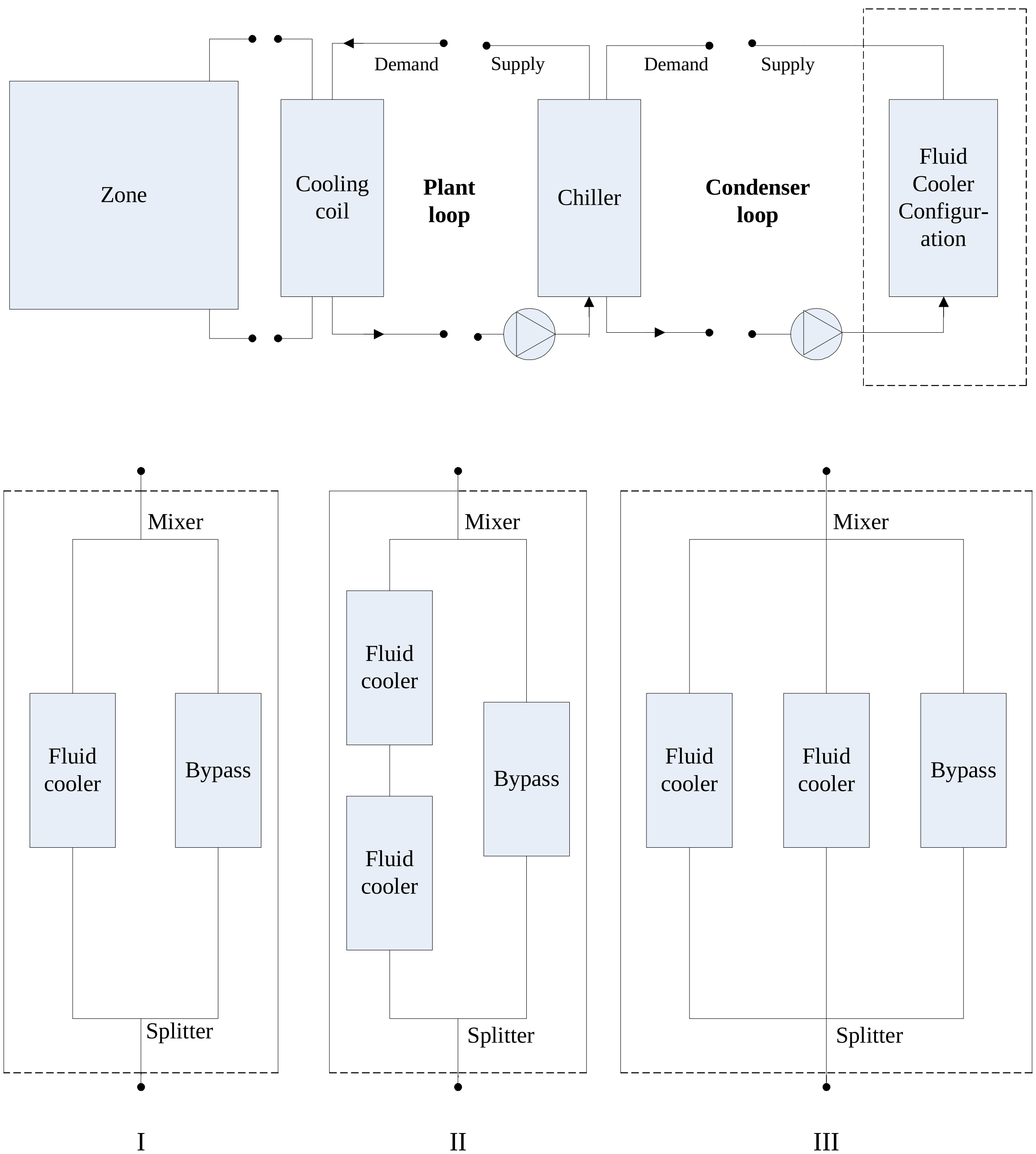


Fig 6.1 EnergyPlus loop diagram

6.2 Example building and system description

Simulations were carried out for a typical office building assumed to be located at Chicago, Illinois. Building loads simulations were run for this region for the summer design day (21st July) and for three different configurations shown in Fig 6.1. The description of the example building (shown in Fig 6.2) and the assumptions are listed below:

1) One story building divided into three interior conditioned zones
2) Roof with no plenum i.e. roofs are exposed to the outdoor environment
3) No ground contact (all floors are adiabatic)
4) Rectangular L-shaped building 40 ft south wall, 40 ft west wall and zone height 10 feet
5) There is a single window in the Resistive zone south wall with the window to wall ratio approximately 0.07
6) The window is single pane 3mm clear
7) The building is oriented due north
8) Floor area is 130.1 m^2 (1403 ft^2)
9) The lighting loads are 18 W/m^2; electrical equipment plug loads are 56.28 W/m^2
10) The office occupancy is assumed to be one person per 13 m^2 with a total heat gain of 131.8 Watts/Person of which 30% is assumed to be radiant heat gain
11) A water cooled chiller is used to meet the building load

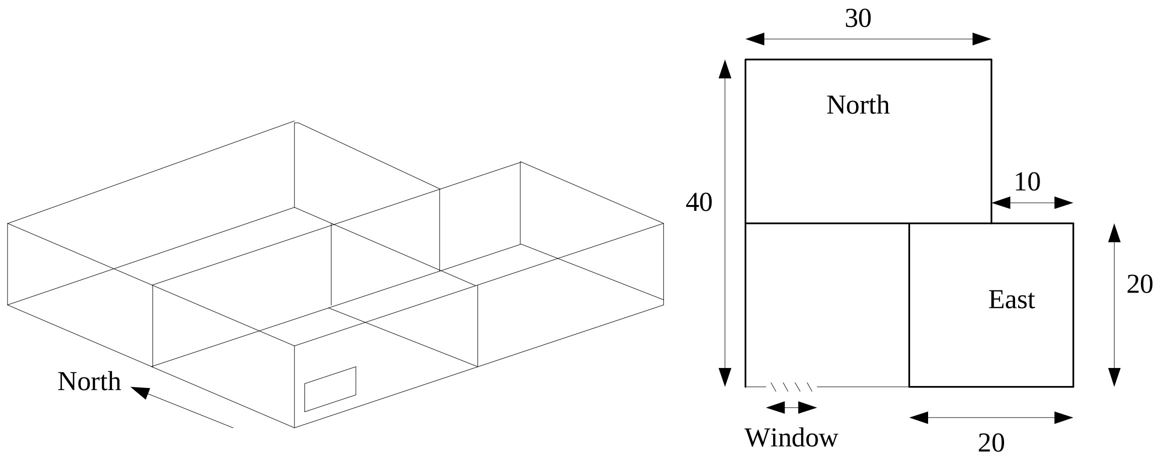


Fig 6.2 Isometric and plan views of the building

6.3 Design day plant loop cooling demand

Fig 6.3 shows summer design day plant loop cooling demand. The peak load at design day is approximately 18.5 KW. Building load is zero till 7 a.m. because all electric equipments are scheduled OFF and the building is unoccupied. Starting at 7a.m. office occupancy, lighting and all electric equipment schedules begin to ramp up. The building cooling demand steadily increases due to the effect of increasing dry bulb temperature, solar heat gains and scheduled loads until the activities come to an end at 5 p.m. That’s why at 5 p.m. the load in the building goes to zero.

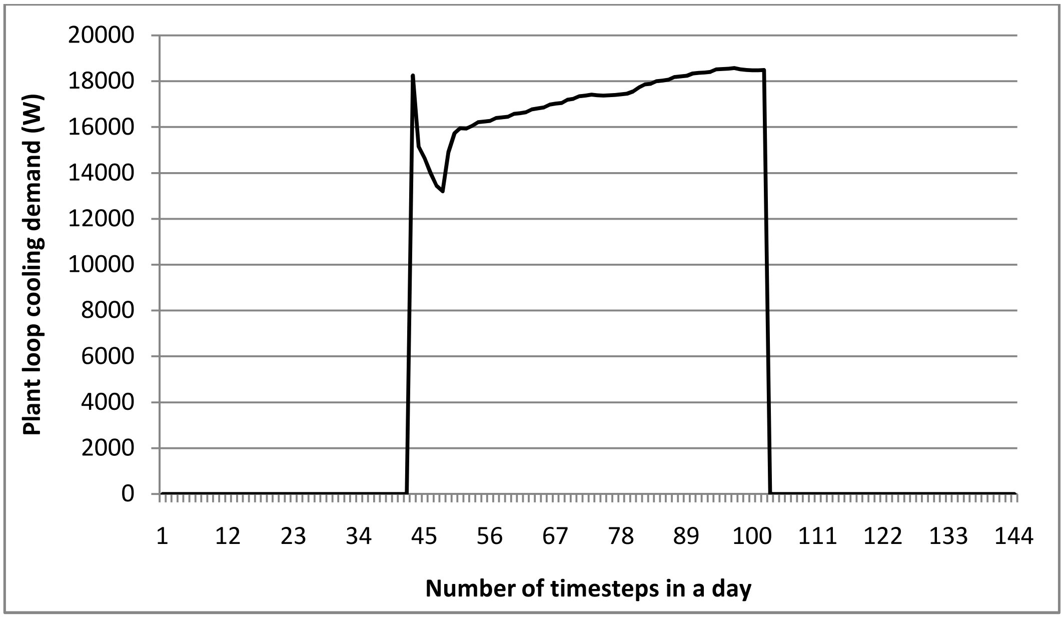


Fig 6.3 Plant loop cooling demand for design day

Dry fluid cooler of 93.75 KW (Heatcraft-Model 21) and Evaporative fluid cooler of 87.9 KW (Motivair Cooling Solutions-MEC0300) design capacity are taken to meet the load requirements of the chiller. The design UA values of the fluid coolers, as calculated by EnergyPlus, are 10674 W/K and 2753 W/K for dry and evaporative fluid coolers respectively. Since evaporative fluid cooler model is a variable UA model; its UA value changes at every time step depending on the location and outdoor air conditions.

6.4 Dry fluid cooler Implementation verification

This section discusses the functioning of different dry fluid cooler configurations. Figure 6.4 shows the fluid cooler heat transfer and combined chiller condenser plus pump heat transfer. The fluid cooler starts at 7 a.m. (43rd time step) because of sudden increase in the building load. This causes a pick up load at the beginning. Once the system stabilizes, the heat transfer rate increases until 5 p.m. because of increasing load on the chiller. The system shuts off at 5 p.m. (102nd time step) because the load in the building and hence chiller comes to zero. The cumulative effect of chiller condenser heat transfer and pump heat addition to the loop fluid is shown in fig 6.4. The fluid cooler rejects the heat to the outside air so that the loop outlet temperature reaches to the set-point which is 95°F (35°C) for this case.

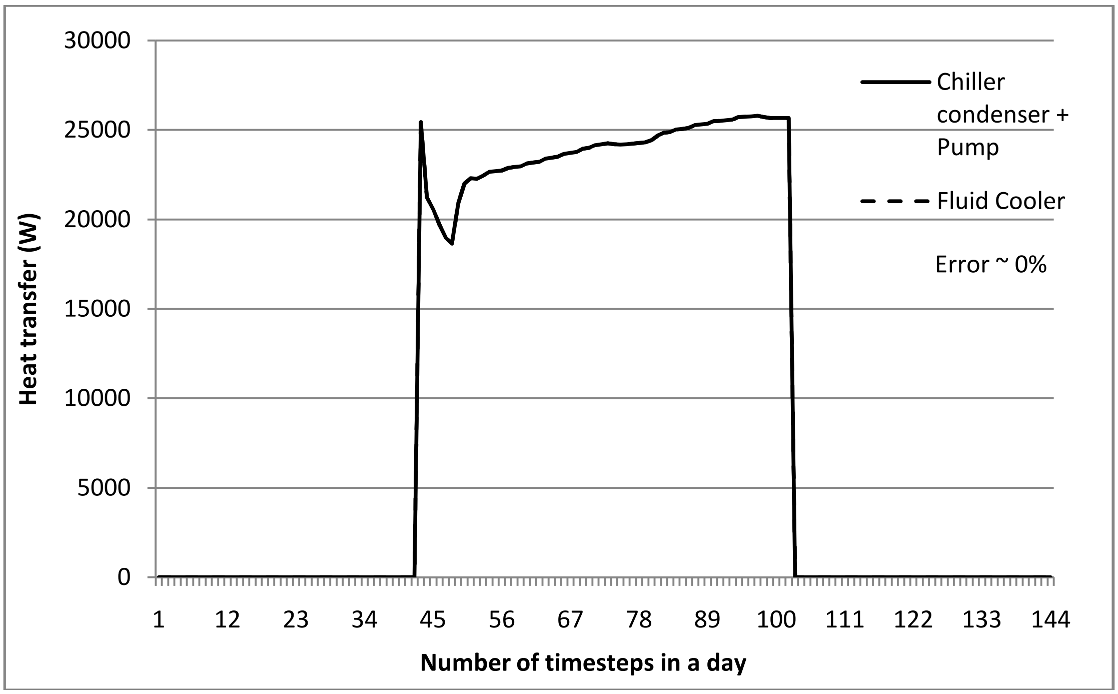
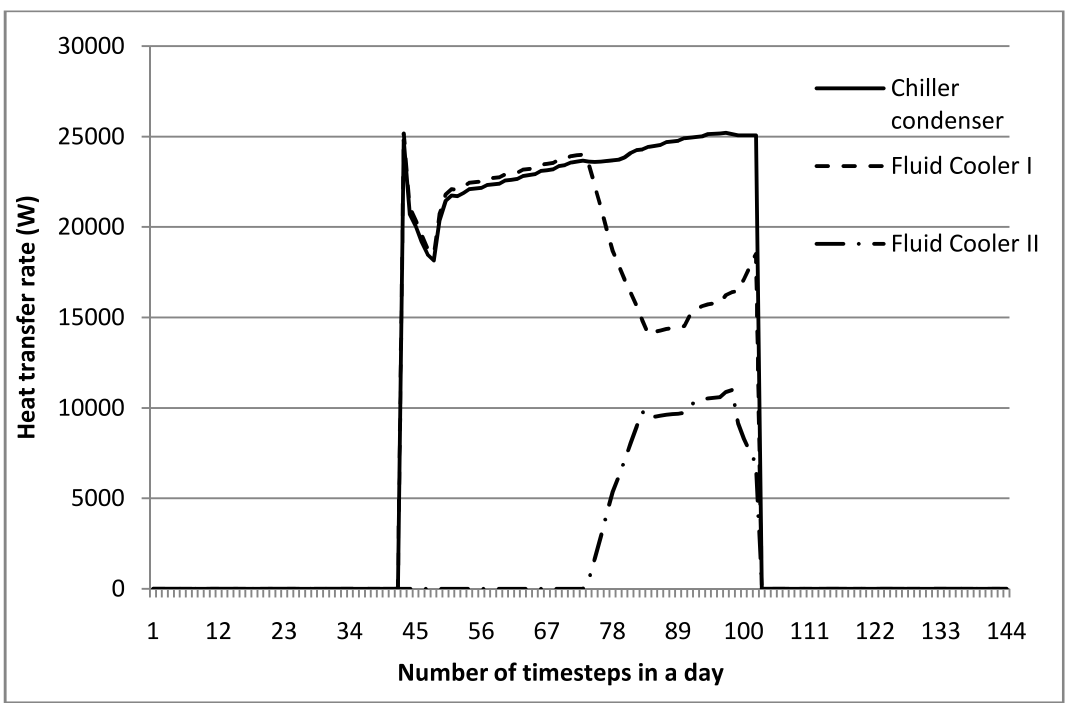


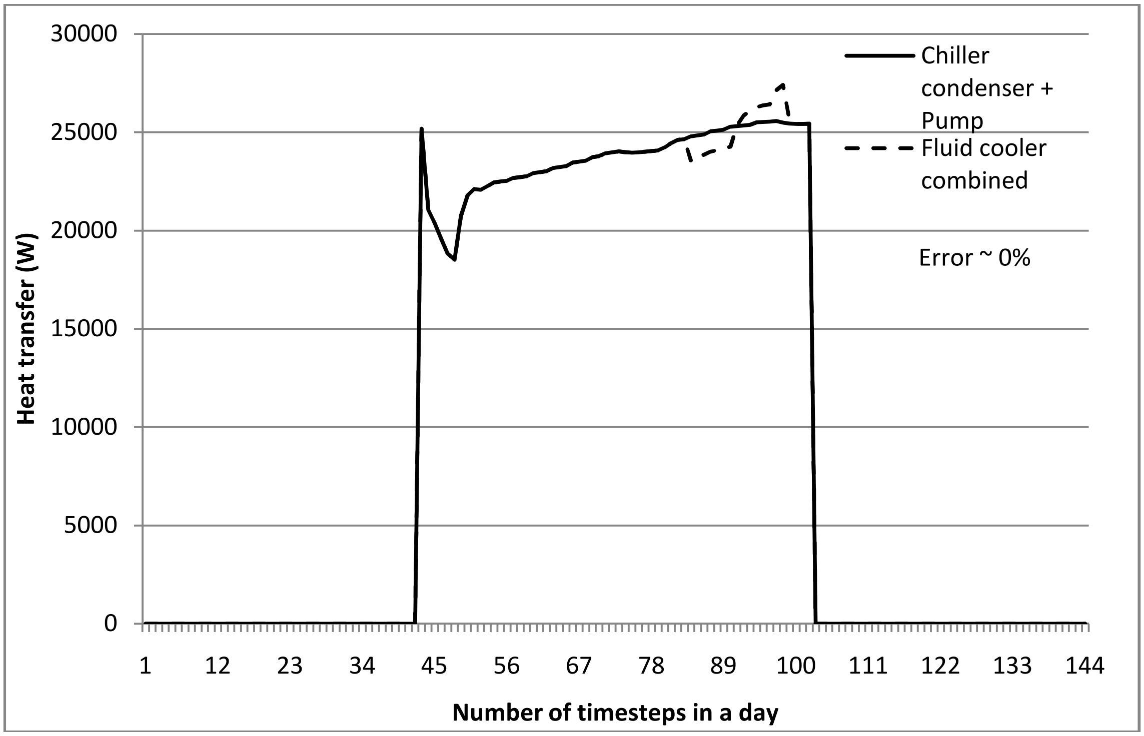
Fig 6.4 Condenser and pump heat addition vs. Fluid cooler heat transfer rate

6.4.1 Series configuration

Figure (6.5) shows the performance of dry fluid coolers in series configuration. The series configuration of the fluid coolers has the set-point of 90°F (32.22°C). From the figures it is clear that until about 12:10 p.m. (73rd time step) the first fluid cooler is able to meet the set-point. After that, as the dry-bulb temperature of the outside air increases the heat transfer rate from the first fluid cooler decreases and loop temperature goes above the set-point. This triggers the second fluid cooler to turn ON which was OFF till now. Because of the combined heat transfer from both the fluid coolers, the configuration meets the set-point till 1:50 p.m. (83rd time step). Between 1:50 to 4 p.m. the dry-bulb temperature increases till 3 p.m. and then starts decreasing but it is above 31°C for the whole time. Fig 6.6 shows the variation of dry and wet-bulb temperature for the design day. Because of the high dry-bulb temperature, the loop temperature goes above the set-point i.e. the combined heat transfer from both the fluid coolers can not reject heat added by chiller condenser and pump. This situation continues until about 4:20 p.m. (98 time step). Between 4:30 to 5 p.m. the loop temperature returns to set-point. The excess heat gained by the loop fluid between 2 to 3 p.m. is rejected by the fluid coolers between 3 to 4:30 p.m. Finally at 5 p.m. the chiller and fluid coolers shut off suddenly because of instant removal of loads.



(a)



(b)

Fig 6.5 Condenser vs. Fluid cooler heat transfer rate

Fig (6.6) and (6.7) show the performance of the configuration when the dry-bulb temperature is taken as set-point. There is a high fluid cooler pick up load in the beginning which causes the loop temperature to drop suddenly. Then the loop temperature increases as the outdoor dry-bulb temperature increases throughout the day. Fig 6.7 shows heat transfer rates from chiller condenser and fluid coolers. A discrepancy is found in EnergyPlus output reporting. The condenser demand inlet temperature was not updated properly. After taking into account the effect of lagging temperature update, the heat balance for the loop is obtained.

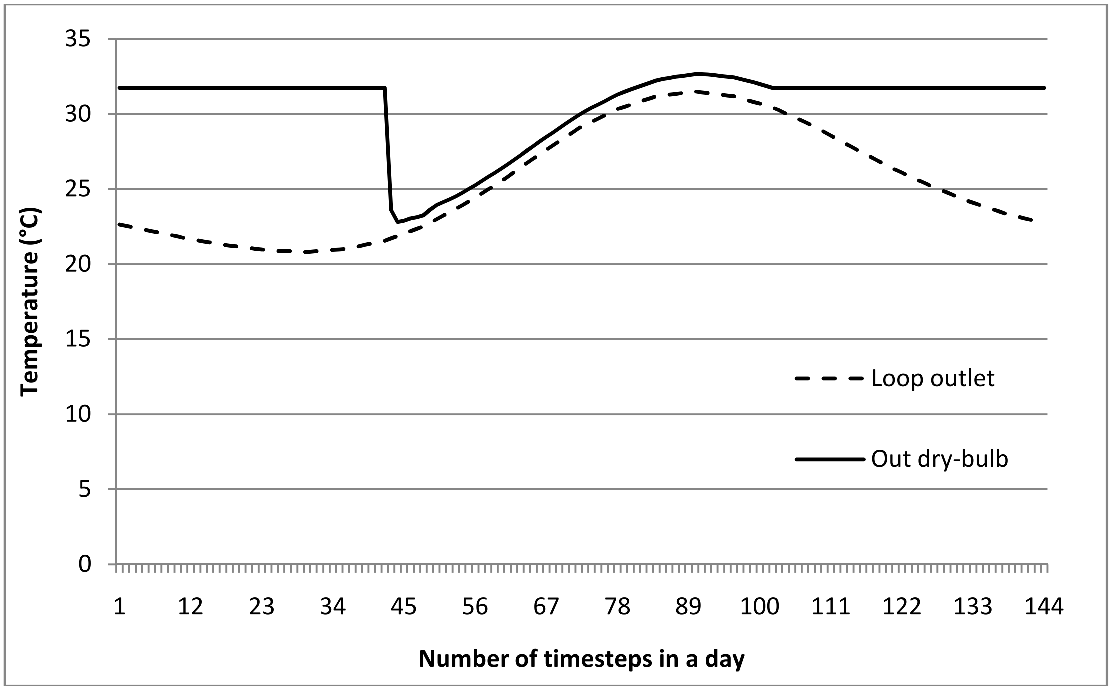
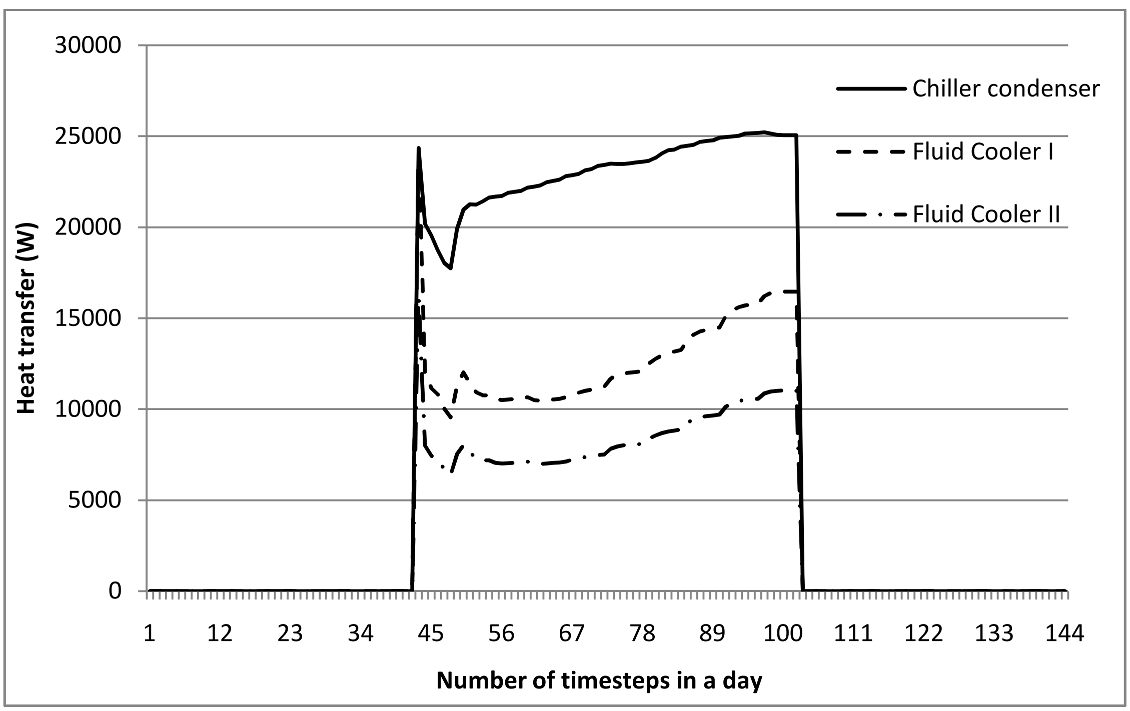


Fig 6.6 Air dry-bulb and loop outlet temperature on the design day



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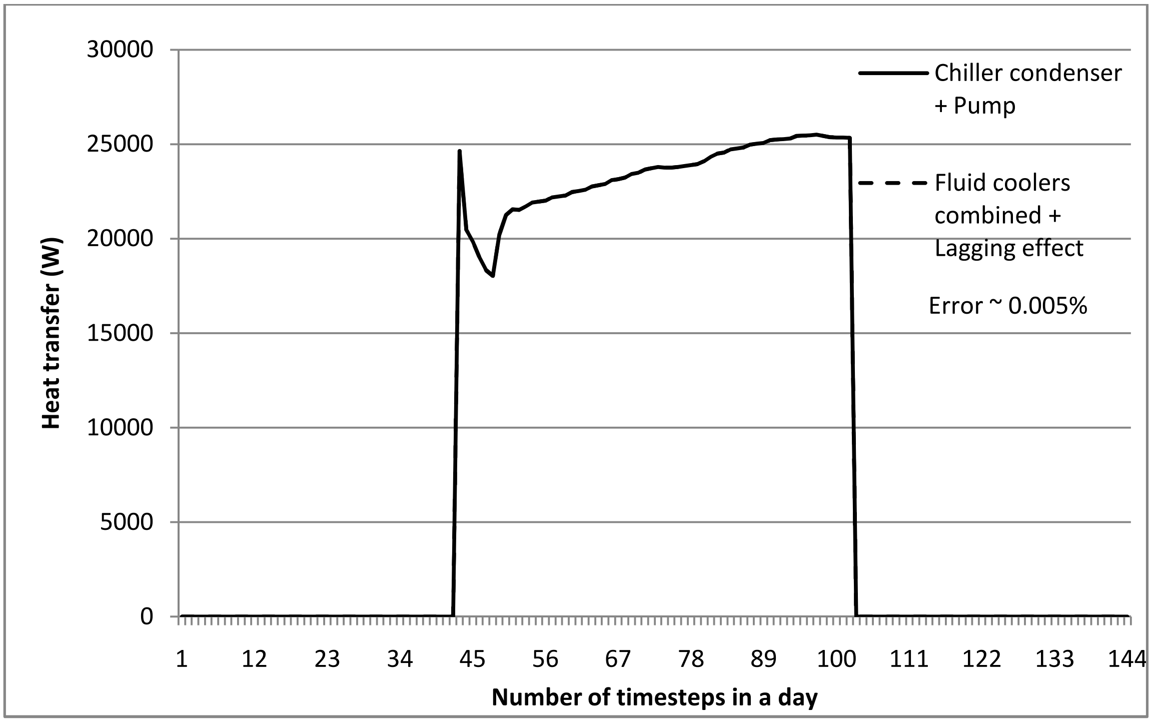
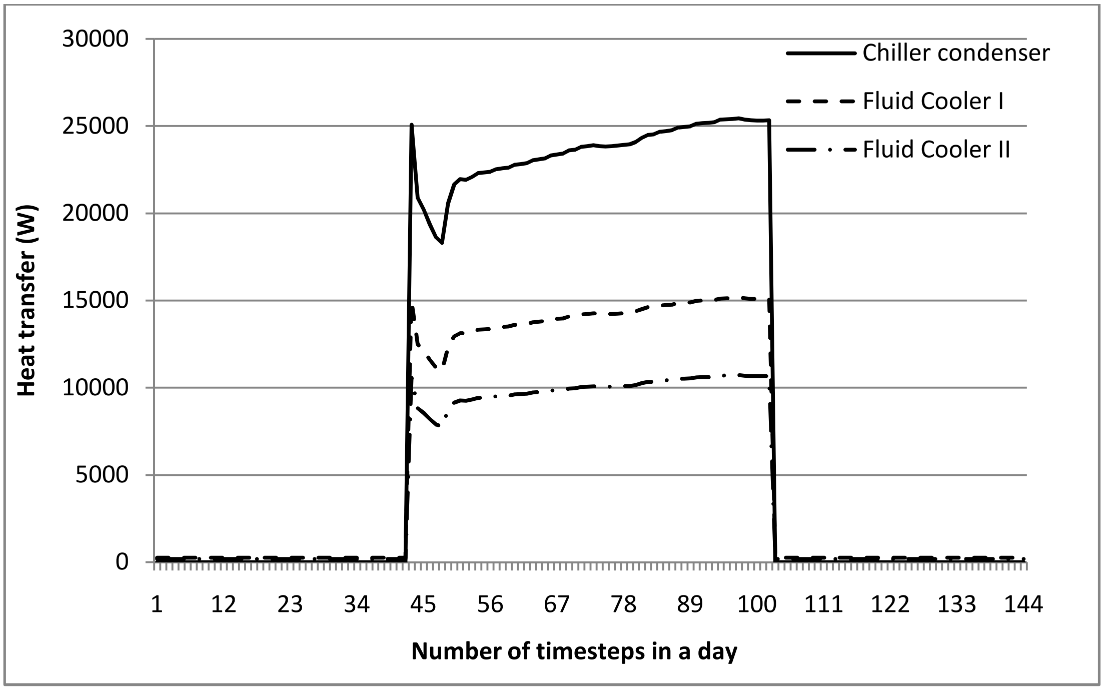


(b)

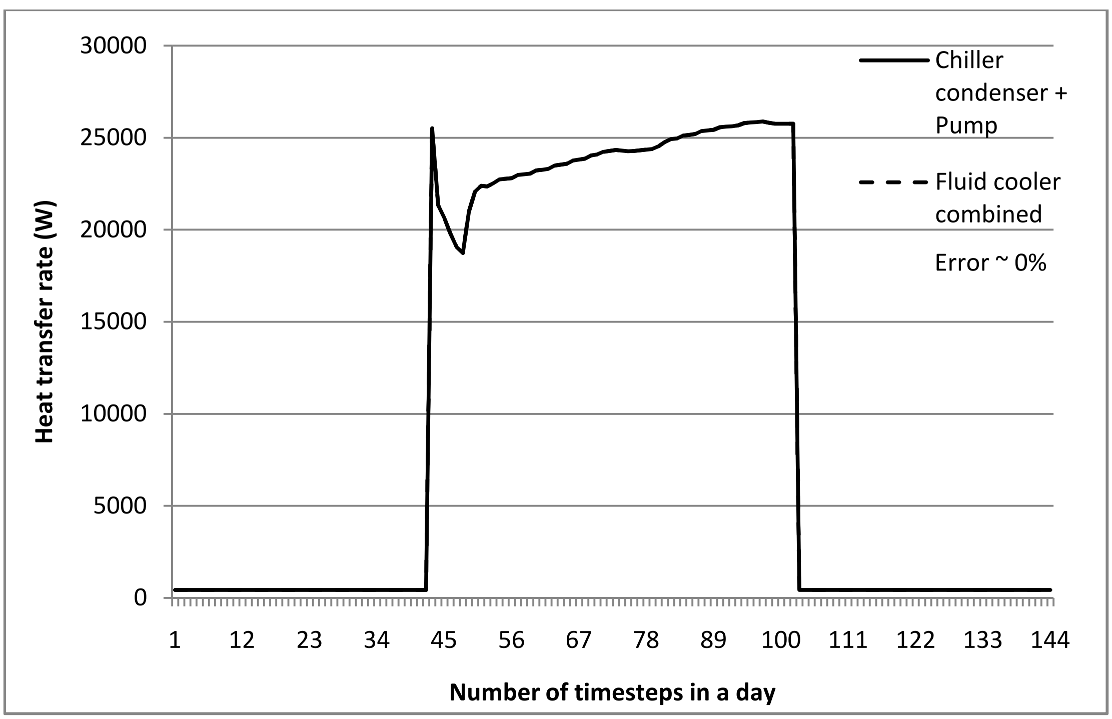
Fig 6.7 Condenser vs. Fluid cooler heat transfer rate for dry-bulb set-point

6.4.2 Parallel configuration

For the present case i.e. in parallel configuration, the total flow rate demand from both the fluid coolers exceeds the maximum available flow rate in the loop. So the second fluid cooler operates undersized. Fig (6.8) shows the results of parallel configuration of fluid coolers. The set-point for the configuration is 95°F (35°C). The continuous operation of the pump adds heat to the loop fluid. So the fluid cooler starts at 12 a.m. to remove pump heat from the fluid. The load on the chiller, and hence on the fluid cooler, increases suddenly at 7 a.m. Fig 6.8 shows the heat balance of the operation. The configuration rejects the total heat added by chiller and pump to the loop fluid.



(a)



(b)

Fig 6.8 Condenser vs. Fluid cooler heat transfer rate

Fig 6.9 and 6.10 show the performance of the parallel dry fluid cooler configuration. Again, as discussed for series configuration, the heat balance is obtained after accounting for the effect of lagging temperature update. Because of pick up load in the beginning, loop outlet temperature drops suddenly. Once the system stabilizes, the loop outlet temperature increases as the outdoor dry-bulb temperature increases throughout the day.

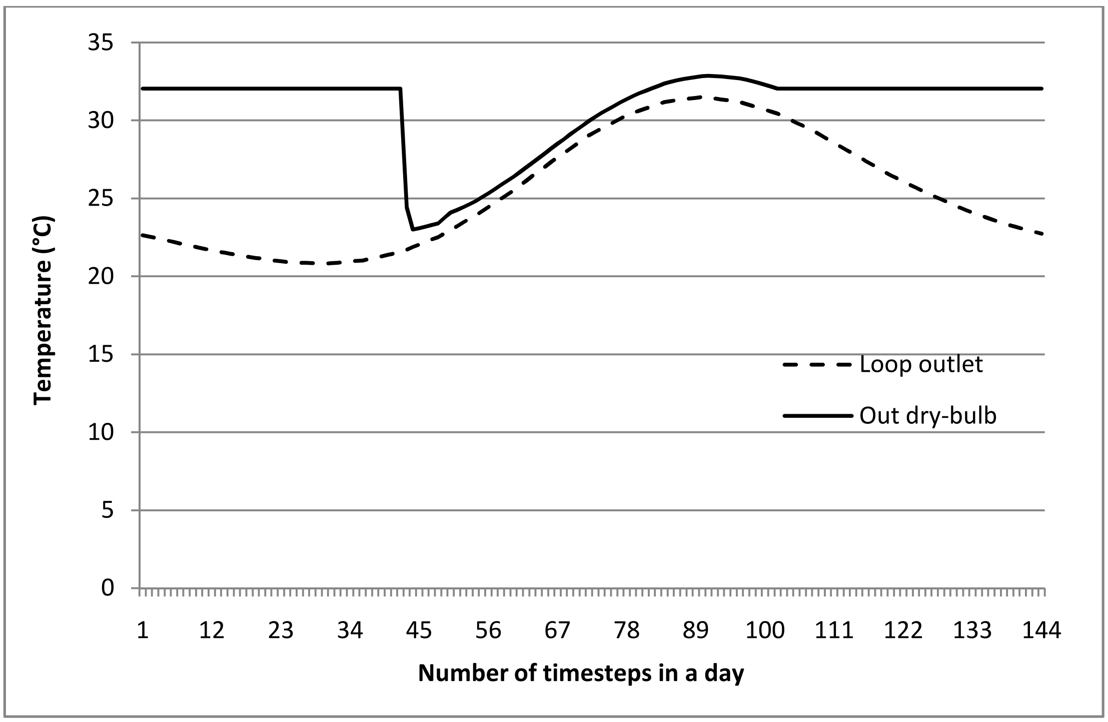
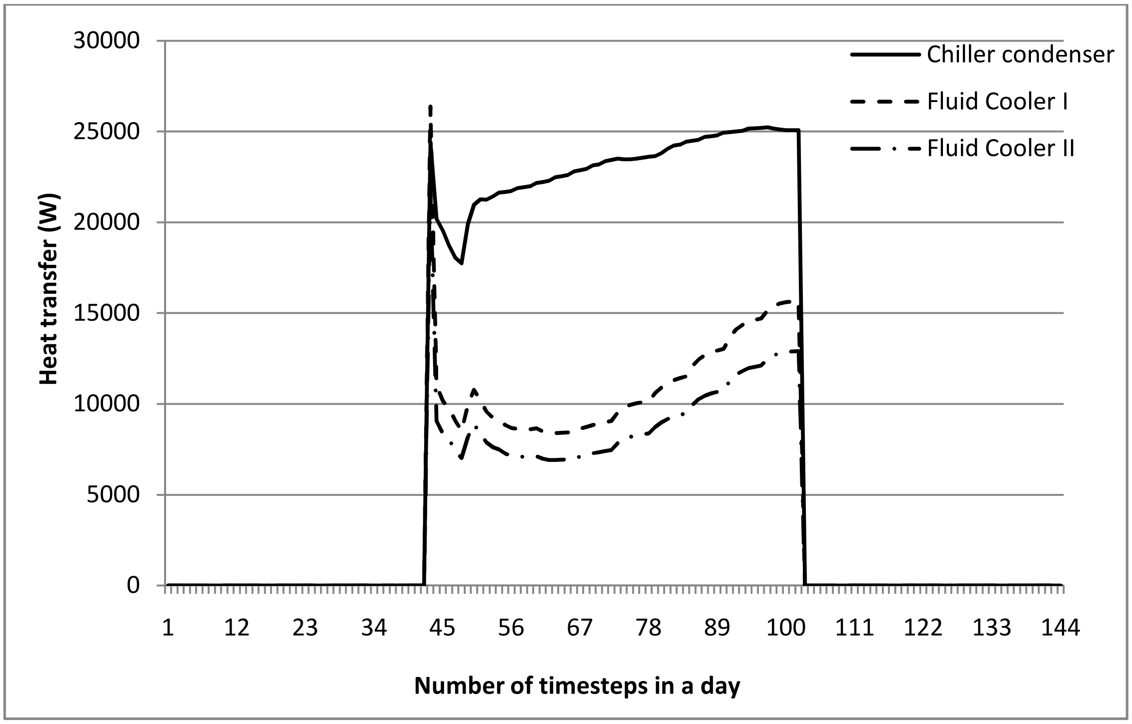
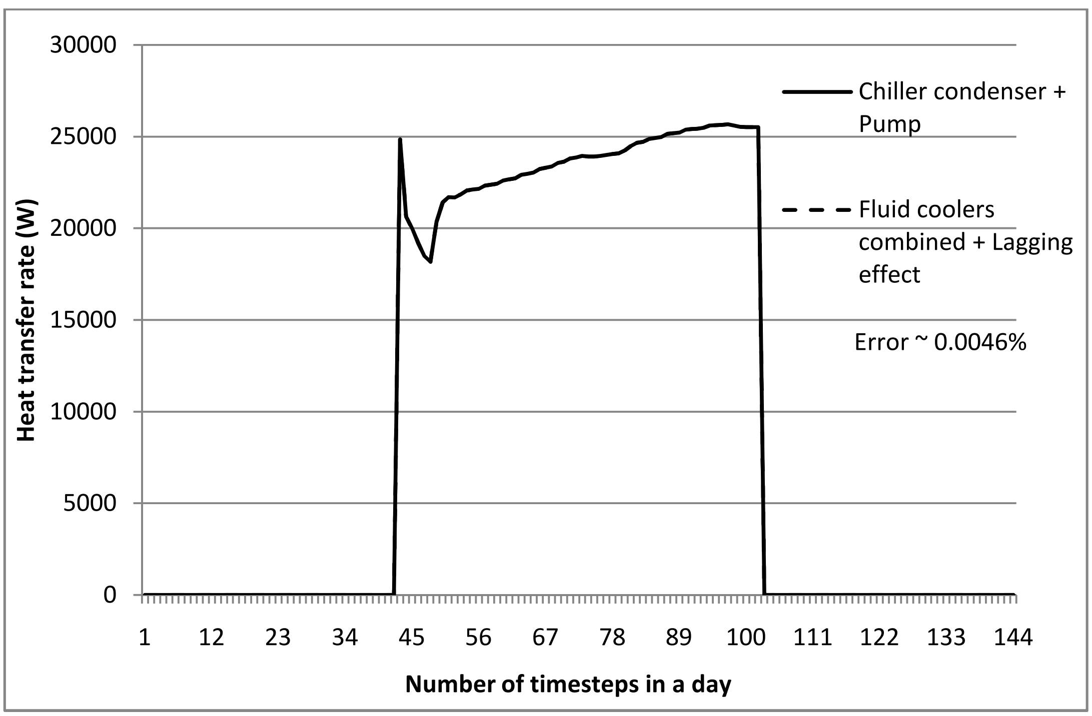


Fig 6.9 Air dry-bulb and loop outlet temperature on the design day



(a)



(b)

Fig 6.10 Condenser vs. Fluid cooler heat transfer rate for dry-bulb set-point

6.5 Evaporative fluid cooler

Evaporative fluid coolers perform similar to dry fluid coolers but the set-point in this case is 85°F (29.44°C). Fig 6.11 shows the pickup load similar to dry fluid cooler and then gradual increase in the heat transfer rate after stabilization. The fluid cooler is capable of rejecting total heat added by the chiller condenser and pump and hence meeting the set-point.

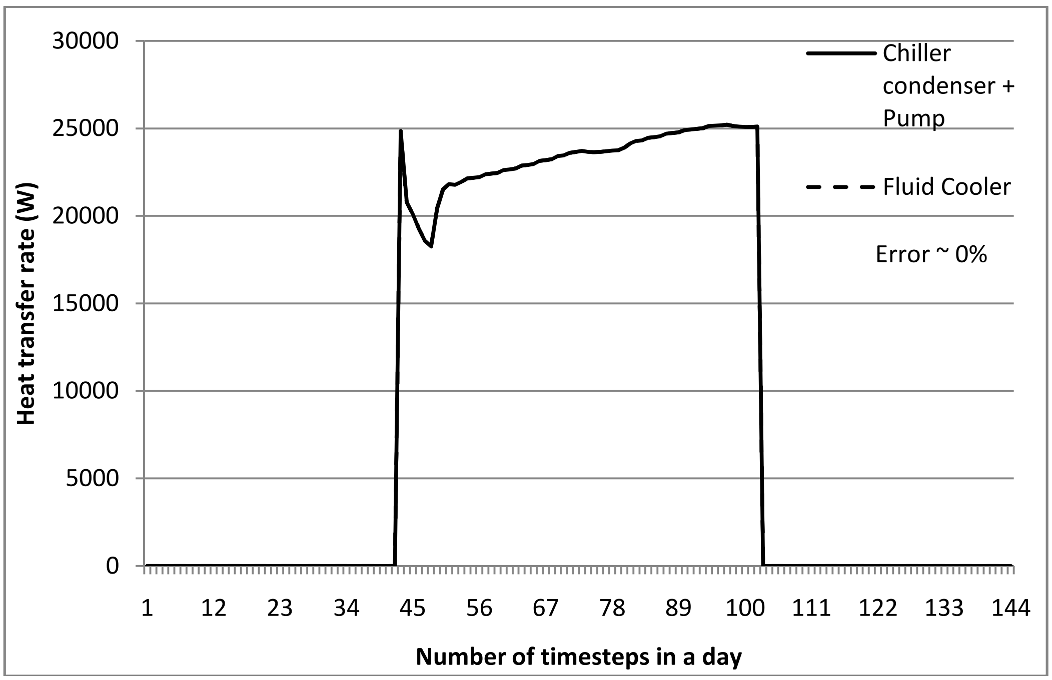
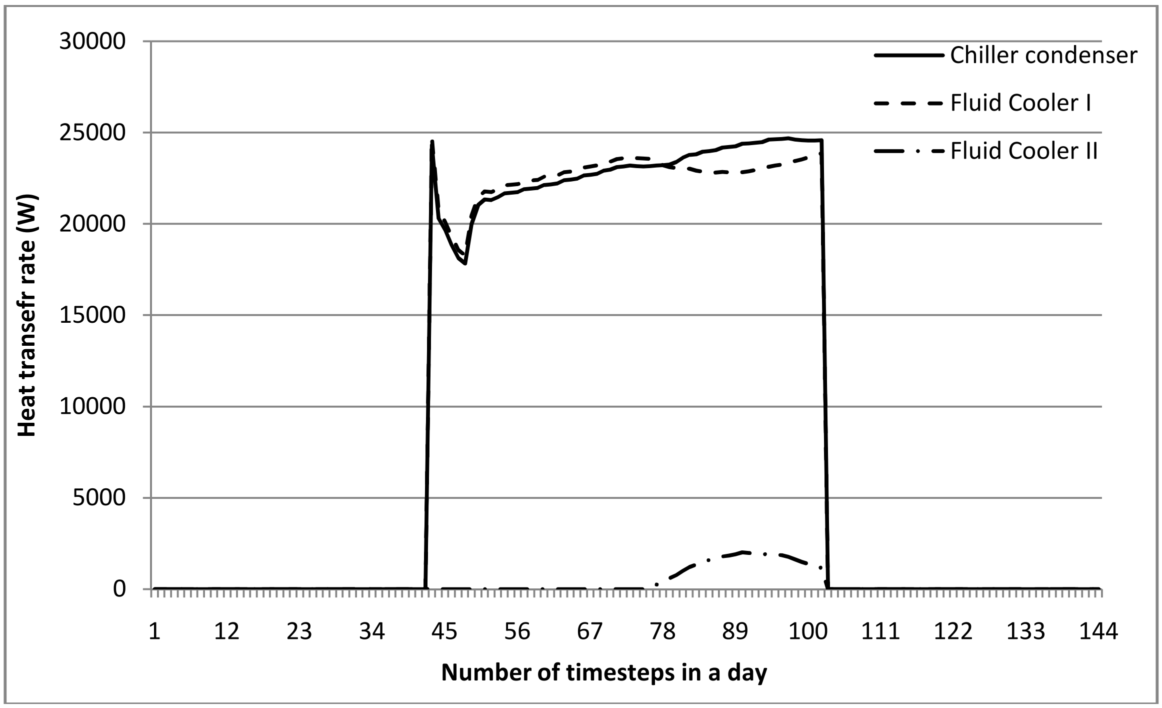


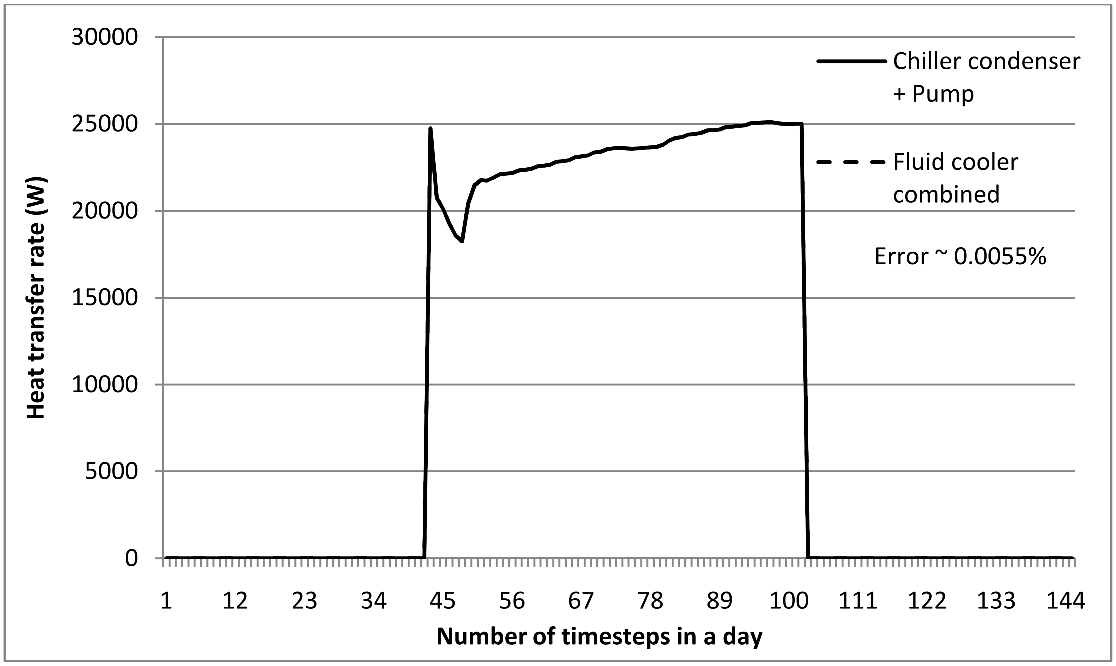
Fig 6.11 Condenser and pump heat addition vs. Fluid cooler heat transfer rate

6.5.1 Series configuration

Fig 6.12 shows the performance of evaporative fluid coolers in series configuration. The set-point at the loop outlet is 78°F (25.56°C). Up until 12:30 p.m. (75^{th} time step), first fluid cooler alone meets the set-point and rejects all the heat added by the chiller and pump. Between 12:30 to 5 p.m. both the fluid coolers operate to lower down the loop fluid temperature though the most of the heat transfer occurs at first fluid cooler only. The cumulative effect of both the fluid coolers maintains the loop outlet temperature at set-point. Part (b) of the figure (6.12) shows the heat balance of the loop.



(a)



(b)

Fig 6.12 Condenser vs. Fluid cooler heat transfer rate

Fig 6.13 and 6.14 show the performance of the series configuration for wet-bulb temperature set-point.

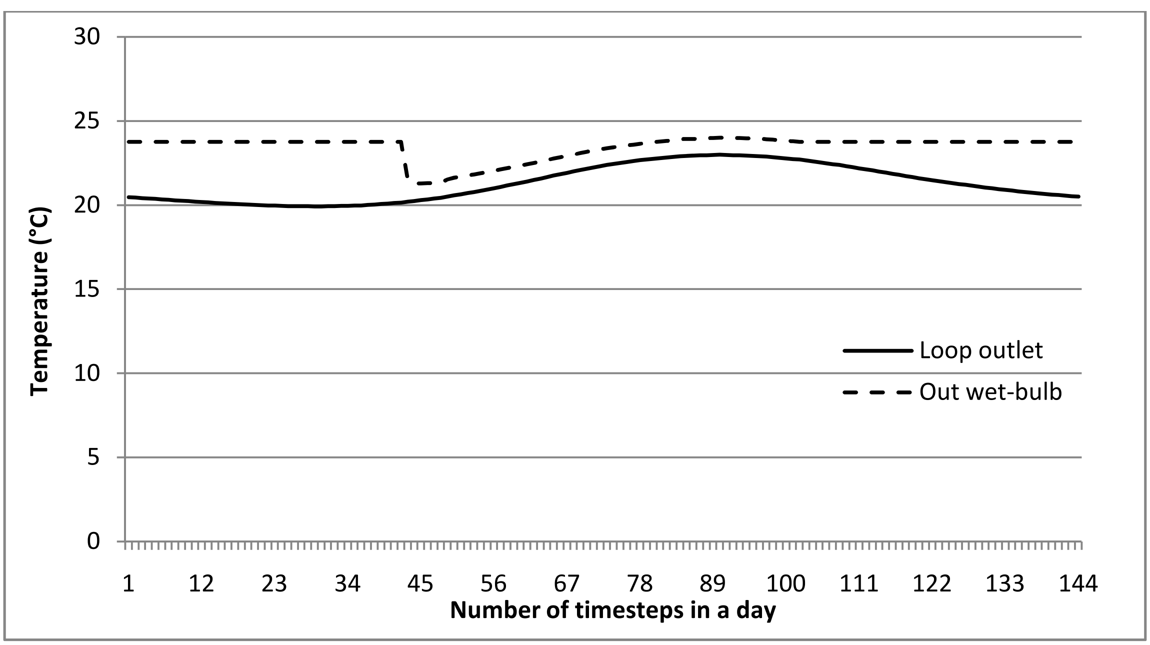
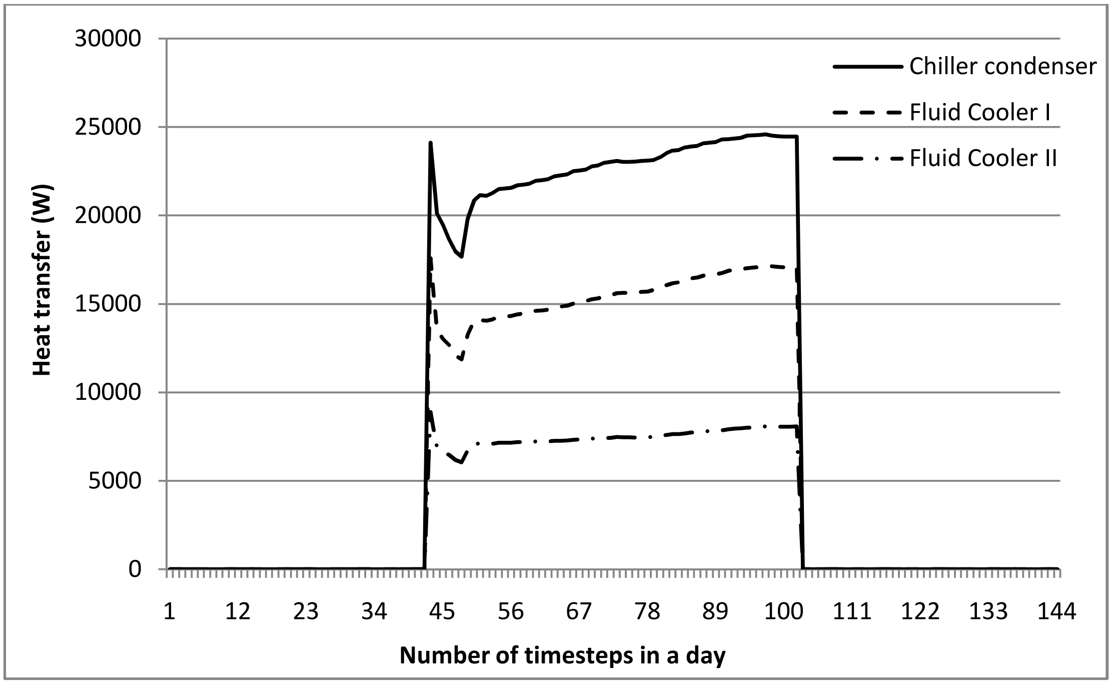
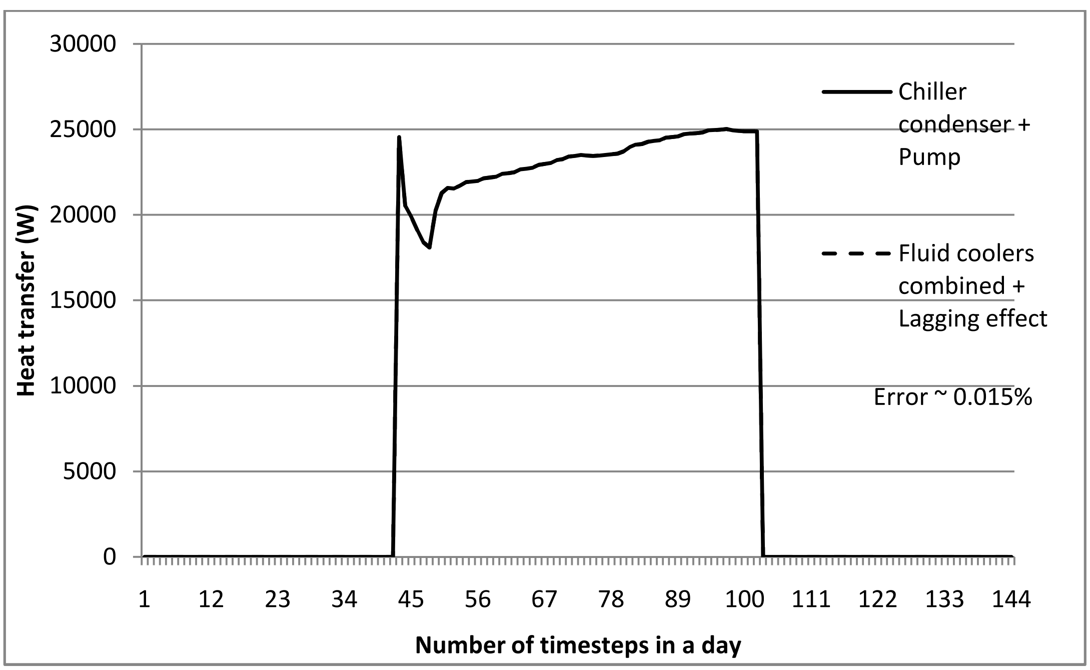


Fig 6.13 Air wet-bulb and loop outlet temperature on the design day



(a)

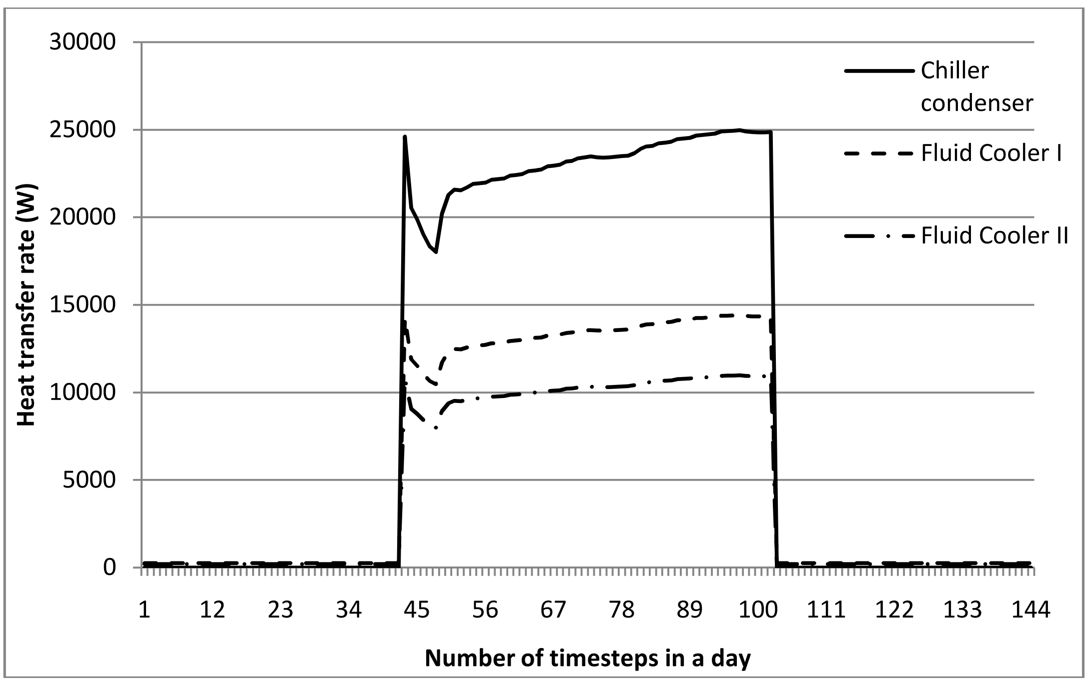


(b)

Fig 6.14 Condenser vs. Fluid cooler heat transfer rate for wet-bulb set-point

6.5.2 Parallel configuration

The performance of evaporative fluid cooler for parallel configuration is shown in figs (6.15). As explained for the dry fluid cooler parallel configuration, this configuration turns ON at 12 a.m. because of continuous operation of the pump. Then pick up load comes at 7 a.m. because of sudden increase in the chiller load. After the system stabilizes, heat transfer continues to increase until 5 p.m. when the activities come to end and load on chiller becomes zero. The combined heat addition by chiller condenser and pump is balanced by fluid coolers as clear from the figure.



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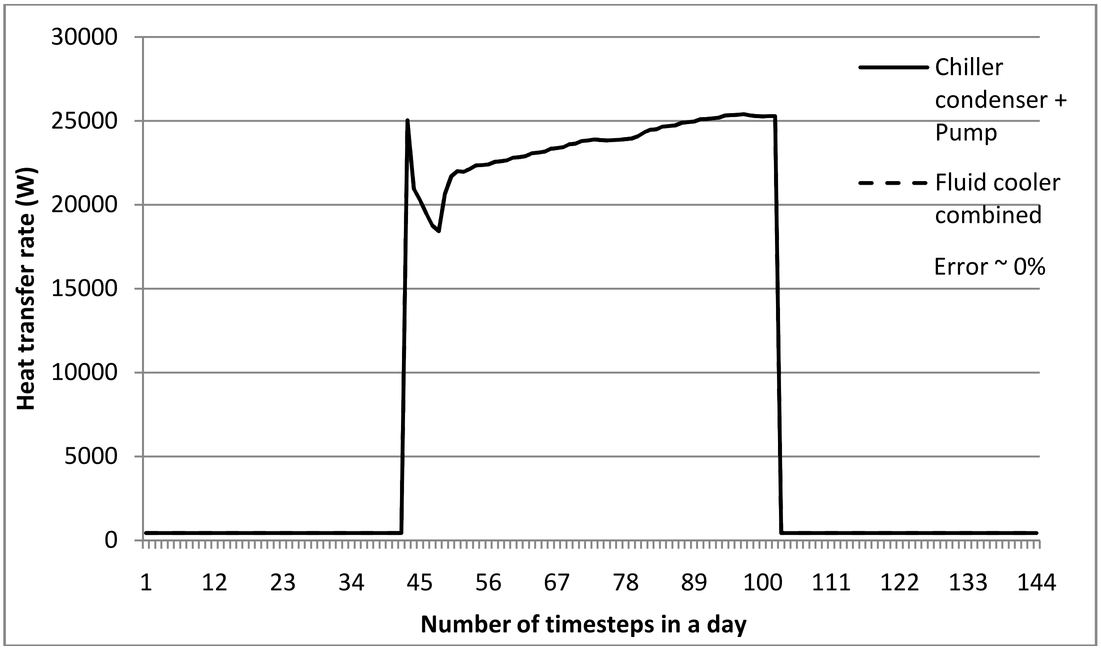


(b)

Fig 6.15 Condenser vs. Fluid cooler heat transfer rate

Finally, the performance of parallel configuration for wet-bulb temperature set-point is shown below.

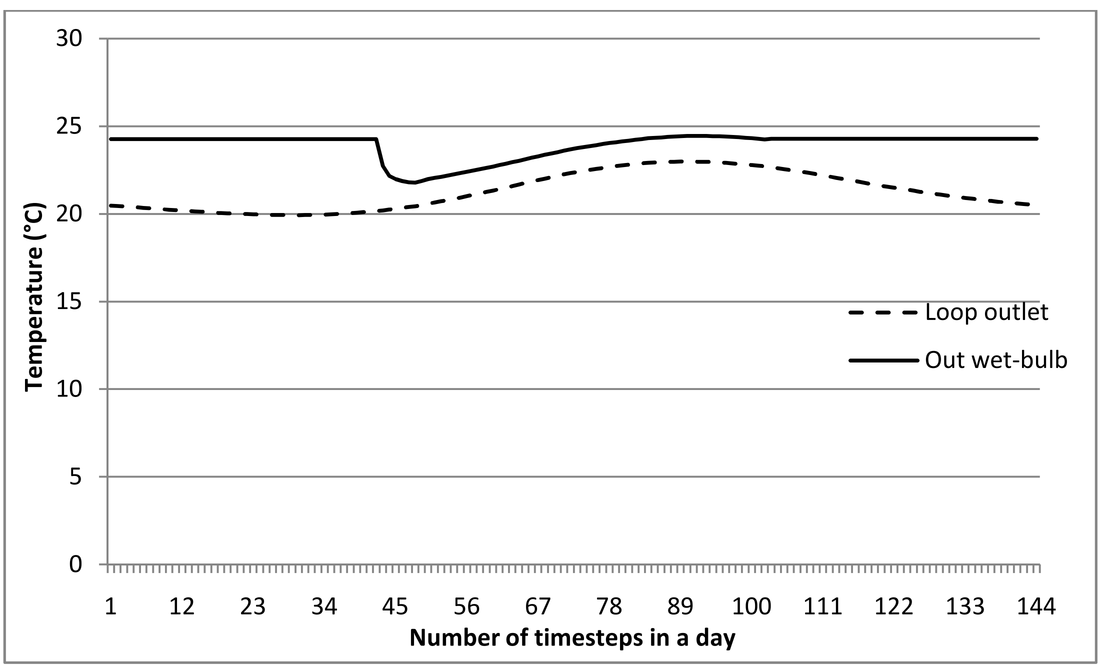
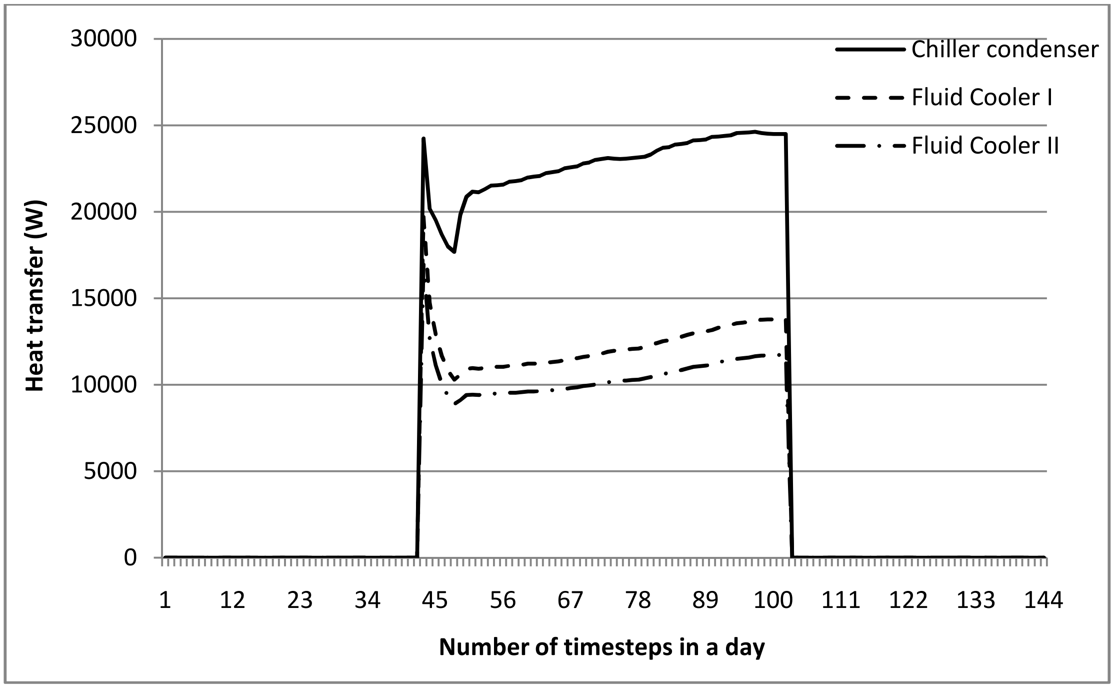
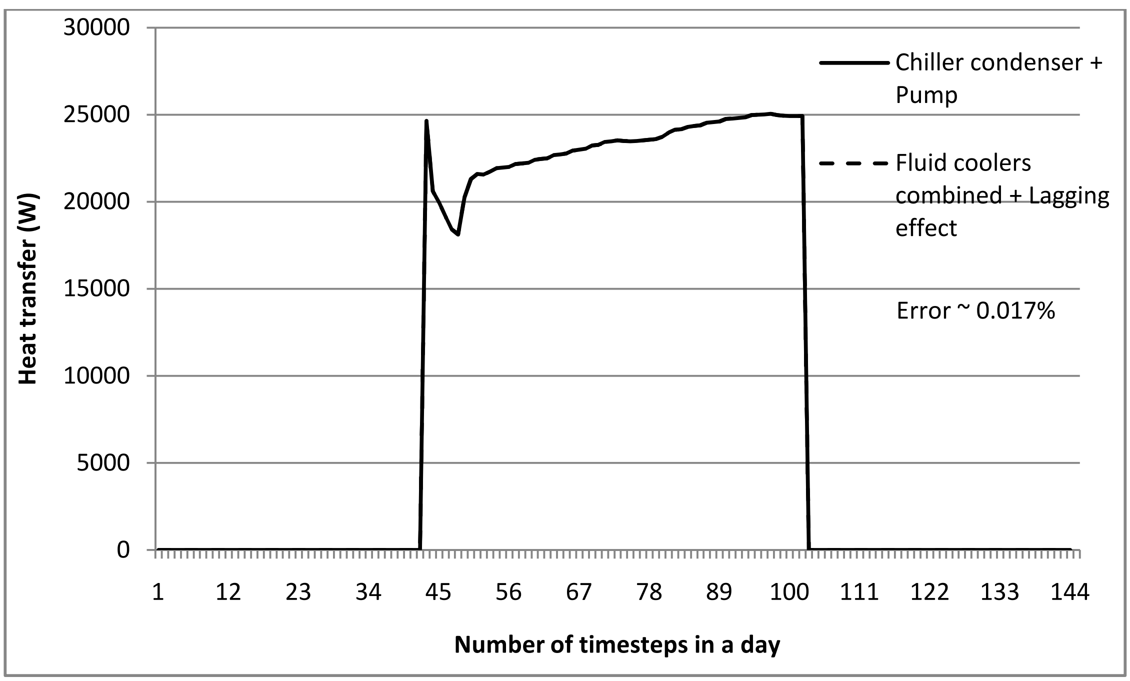


Fig 6.16 Air wet-bulb and loop outlet temperature on the design day



(a)



(b)

Fig 6.17 Condenser vs. Fluid cooler heat transfer rate for wet-bulb set-point

CHAPTER VII

CONCLUSION AND RECOMMENDATION

This thesis presents development, implementation and verification of dry and evaporative fluid cooler models in EnergyPlus. A literature review is performed to collect information about existing fluid cooler models. The dry fluid cooler is modeled as a classis heat exchanger by using (Ɛ-NTU) method. Lebrun model is used as the basis for the evaporative fluid cooler model. The actual model required estimation of four different parameters i.e.$R_{a,n}$, $R_{w,n}$ and exponents m andn to calculate overall heat transfer coefficient (UA). This parameter estimation could cause serious convergence problem in EnergyPlus, so the model is modified to make it simpler to implement. In EnergyPlus, the UA value of evaporative fluid cooler is directly calculated by iteration.

A complete validation of the fluid cooler models for part load conditions could not be performed because of scant manufacturers' data. In order to verify the models Lebrun model was implemented in VBA and the results were compared with EnergyPlus results. HVACSIM+ dry fluid cooler model (Type 762) is used to verify the EnergyPlus model. The obtained results were in good agreements. Baltimore Aircoil provides data for multiple rating points without dry-bulb temperature. A parametric study was performed to estimate the impact of dry-bulb temperature on the output of evaporative fluid cooler models. It was observed that even a large range of temperatures (80 to 95°F) had negligible effect on the evaporative fluid cooler performance. The simulation results are in excellent agreement with the catalog data. The main points of the study can be summarized as follows:

- Results of EnergyPlus dry fluid cooler model are within ±0.3% of HVACSIM+ model.
- Evaporative fluid cooler model compares very well with Lebrun model with maximum error being less than 8%. Lebrun model itself has 7-8 % accuracy range as compare to manufacturer's data.
- The manufacturer's data available for fluid coolers are seriously insufficient. Most of the manufacturer's don't provide catalogue data for multiple rating points. Those who do provide multiple rating points lack some key input parameters. Evaporative fluid cooler model is validated against manufacturer's data published by Baltimore Aircoil. After obtaining the dry-bulb temperature from parametric study, simulation is performed and the results are compared. There was a negligible deviation (0.025% maximum) in the EnergyPlus output with respect to catalogue data.
- The results of the parametric study has shown that dry-bulb temperature affects dry-fluid cooler results and wet-bulb temperature affects evaporative fluid cooler results in a much higher scale than what wet-bulb temperature affects dry fluid cooler results or dry-bulb temperature affects evaporative fluid cooler results.

RECOMMENDATIONS

- Experimental data sets are needed to validate part-load and off-design performance of the models.
- Fan control strategies should be improved as per the actual control and operation strategies of fluid cooler.
- Models to calculate evaporation losses and make up water should also be developed and implemented.
- Dry fluid cooler model should be developed further to model "adiabatic mode" operation.

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APPENDIX A

Parametric study of varying dry-bulb temperature for evaporative fluid cooler model

Most of the times, evaporative fluid cooler manufacturers don't provide dry-bulb temperature in the catalog data. Dry-bulb temperature is required by EnergyPlus evaporative fluid cooler model as design parameter. A parametric study is carried out to understand the impact of dry-bulb temperature on the EnergyPlus evaporative fluid cooler results. Catalog data presented in table 5.1 is used for the present study.

The dry-bulb temperatures are varied from 80 to 95 °F and the corresponding UA, capacity (Q) and outlet water temperatures ($T_{w,out}$) are calculated. Table (A-1) shows the results of the study. Data set 1 to 9 corresponds to the nine rating points shown in table 5.1. From the table (A-1) it is clear that dry-bulb temperature does not have significant impact on the evaporative fluid cooler simulation results.

Table A-1: Parametric study to understand the impact of dry-bulb temperatures on EnergyPlus evaporative fluid cooler model results

Inputs			Outputs		
$T_{db,in}$ (°C)	$T_{wb,in}$ (°C)	$T_{w,in}$ (°C)	UA (W/K)	Q (W)	$T_{w,out}$ (°C)
Data set: 1					
80	22.22	38.89	1238.39	56254.5	32.22
81	22.22	38.89	1238.74	56254.5	32.22
82	22.22	38.89	1239.09	56254.5	32.22
83	22.22	38.89	1239.44	56254.6	32.22
84	22.22	38.89	1239.78	56254.6	32.22
85	22.22	38.89	1240.01	56250.7	32.22
86	22.22	38.89	1240.36	56250.7	32.22
87	22.22	38.89	1240.71	56250.7	32.22
88	22.22	38.89	1241.06	56250.7	32.22
89	22.22	38.89	1241.41	56250.7	32.22
90	22.22	38.89	1241.77	56250.7	32.22
91	22.22	38.89	1242.12	56250.7	32.22
92	22.22	38.89	1242.47	56250.8	32.22
93	22.22	38.89	1242.83	56250.8	32.22
94	22.22	38.89	1243.18	56250.8	32.22
95	22.22	38.89	1243.54	56250.8	32.22
Data set: 2					
80	22.22	46.11	1062.94	62260.7	32.22
81	22.22	46.11	1063.05	62260.7	32.22
82	22.22	46.11	1063.16	62260.8	32.22
83	22.22	46.11	1063.27	62260.8	32.22
84	22.22	46.11	1063.38	62260.8	32.22
85	22.22	46.11	1063.49	62260.8	32.22
86	22.22	46.11	1063.6	62260.8	32.22
87	22.22	46.11	1063.72	62260.8	32.22
88	22.22	46.11	1063.83	62260.8	32.22
89	22.22	46.11	1063.94	62260.8	32.22
90	22.22	46.11	1064.05	62260.9	32.22
91	22.22	46.11	1064.17	62260.9	32.22
92	22.22	46.11	1064.28	62260.9	32.22
93	22.22	46.11	1064.4	62260.9	32.22
94	22.22	46.11	1064.51	62260.9	32.22
95	22.22	46.11	1064.62	62260.9	32.22
Data set: 3					
80	22.22	35.00	1191.67	39552.4	29.44
81	22.22	35.00	1192.07	39552.4	29.44
82	22.22	35.00	1192.47	39552.4	29.44

83	22.22	35.00	1192.88	39552.4	29.44
84	22.22	35.00	1193.28	39552.5	29.44
85	22.22	35.00	1193.69	39552.5	29.44
86	22.22	35.00	1194.09	39552.5	29.44
87	22.22	35.00	1194.5	39552.5	29.44
88	22.22	35.00	1194.91	39552.5	29.44
89	22.22	35.00	1195.32	39552.5	29.44
90	22.22	35.00	1195.72	39552.5	29.44
91	22.22	35.00	1196.13	39552.6	29.44
92	22.22	35.00	1196.54	39552.6	29.44
93	22.22	35.00	1196.96	39552.6	29.44
94	22.22	35.00	1197.37	39552.6	29.44
95	22.22	35.00	1197.78	39552.6	29.44
Data set: 4					
80	25.56	35.00	939.68	23438.1	29.44
81	25.56	35.00	939.94	23438.1	29.44
82	25.56	35.00	940.2	23438.1	29.44
83	25.56	35.00	940.461	23438.1	29.44
84	25.56	35.00	940.722	23438.1	29.44
85	25.56	35.00	940.984	23438.1	29.44
86	25.56	35.00	941.246	23438.1	29.44
87	25.56	35.00	941.508	23438.1	29.44
88	25.56	35.00	941.77	23438.1	29.44
89	25.56	35.00	942.033	23438.1	29.44
90	25.56	35.00	942.296	23438.1	29.44
91	25.56	35.00	942.56	23438.1	29.44
92	25.56	35.00	942.824	23438.1	29.44
93	25.56	35.00	943.088	23438.1	29.44
94	25.56	35.00	943.352	23438.1	29.44
95	25.56	35.00	943.617	23438.2	29.44
Data set: 5					
80	25.56	38.89	1097.82	42188.7	32.22
81	25.56	38.89	1098.13	42188.7	32.22
82	25.56	38.89	1098.44	42188.7	32.22
83	25.56	38.89	1098.75	42188.7	32.22
84	25.56	38.89	1099.05	42188.7	32.22
85	25.56	38.89	1099.36	42188.7	32.22
86	25.56	38.89	1099.67	42188.8	32.22
87	25.56	38.89	1099.98	42188.8	32.22
88	25.56	38.89	1100.29	42188.8	32.22
89	25.56	38.89	1100.6	42188.8	32.22
90	25.56	38.89	1100.91	42188.8	32.22
91	25.56	38.89	1101.22	42188.8	32.22
92	25.56	38.89	1101.54	42188.8	32.22
93	25.56	38.89	1101.85	42188.8	32.22

94	25.56	38.89	1102.16	42188.8	32.22
95	25.56	38.89	1102.48	42188.8	32.22
Data set: 6					
80	25.56	46.11	845.18	43947.5	32.22
81	25.56	46.11	845.269	43947.5	32.22
82	25.56	46.11	845.358	43947.5	32.22
83	25.56	46.11	845.447	43947.5	32.22
84	25.56	46.11	845.536	43947.5	32.22
85	25.56	46.11	845.626	43947.5	32.22
86	25.56	46.11	845.715	43947.5	32.22
87	25.56	46.11	845.805	43947.5	32.22
88	25.56	46.11	845.895	43947.5	32.22
89	25.56	46.11	845.985	43947.5	32.22
90	25.56	46.11	846.075	43947.5	32.22
91	25.56	46.11	846.166	43947.5	32.22
92	25.56	46.11	846.256	43947.5	32.22
93	25.56	46.11	846.347	43947.5	32.22
94	25.56	46.11	846.438	43947.5	32.22
95	25.56	46.11	846.529	43947.5	32.22
Data set: 7					
80	26.67	35.00	823.696	17579.7	29.44
81	26.67	35.00	823.246	17579.7	29.44
82	26.67	35.00	822.797	17579.7	29.44
83	26.67	35.00	822.932	17579.7	29.44
84	26.67	35.00	823.129	17579.7	29.44
85	26.67	35.00	823.326	17579.7	29.44
86	26.67	35.00	823.523	17579.7	29.44
87	26.67	35.00	823.72	17579.7	29.44
88	26.67	35.00	823.918	17579.7	29.44
89	26.67	35.00	824.116	17579.7	29.44
90	26.67	35.00	824.314	17579.7	29.44
91	26.67	35.00	824.512	17579.7	29.44
92	26.67	35.00	824.71	17579.7	29.44
93	26.67	35.00	824.909	17579.7	29.44
94	26.67	35.00	825.108	17579.7	29.44
95	26.67	35.00	825.307	17579.7	29.44
Data set: 8					
80	26.67	38.89	987.458	35157.7	32.22
81	26.67	38.89	986.932	35157.7	32.22
82	26.67	38.89	986.406	35157.7	32.22
83	26.67	38.89	986.586	35157.7	32.22
84	26.67	38.89	986.839	35157.7	32.22
85	26.67	38.89	987.093	35157.7	32.22
86	26.67	38.89	987.346	35157.7	32.22
87	26.67	38.89	987.601	35157.8	32.22

88	26.67	38.89	987.855	35157.8	32.22
89	26.67	38.89	988.11	35157.8	32.22
90	26.67	38.89	988.365	35157.8	32.22
91	26.67	38.89	988.621	35157.8	32.22
92	26.67	38.89	988.877	35157.8	32.22
93	26.67	38.89	989.133	35157.8	32.22
94	26.67	38.89	989.39	35157.8	32.22
95	26.67	38.89	989.647	35157.8	32.22
Data set: 9					
80	26.67	46.11	824.342	40286.3	32.22
81	26.67	46.11	823.81	40286.3	32.22
82	26.67	46.11	823.279	40286.3	32.22
83	26.67	46.11	823.311	40286.3	32.22
84	26.67	46.11	823.403	40286.3	32.22
85	26.67	46.11	823.494	40286.3	32.22
86	26.67	46.11	823.586	40286.3	32.22
87	26.67	46.11	823.677	40286.3	32.22
88	26.67	46.11	823.769	40286.3	32.22
89	26.67	46.11	823.861	40286.3	32.22
90	26.67	46.11	823.953	40286.3	32.22
91	26.67	46.11	824.046	40286.3	32.22
92	26.67	46.11	824.138	40286.3	32.22
93	26.67	46.11	824.231	40286.4	32.22
94	26.67	46.11	824.324	40286.4	32.22
95	26.67	46.11	824.416	40286.4	32.22

APPENDIX B

Description of the VBA implementation of Lebrun model

Lebrun model described in chapter 2 was implemented in VBA to verify EnergyPlus evaporative fluid cooler model. A parameter estimation tool, developed by Kenneth Tang (2005) for water to air heat pumps, was used as the basis to develop this model. All the four parameters, which are $R_{a,n}$, $R_{w,n}$ and exponents m andn required by Lebrun model, are estimated by using a parameter estimation tool. Once the parameters are determined, the capacity of the fluid cooler is calculated. The obtained capacities are in excellent agreement with the publish capacities.

Methodology

Figure B-1 below shows the flow chart of the VBA algorithm of the Lebrun model. To start with, an initial guess of all the four parameters was entered by the user. The model then guesses the outlet fluid cooler wet-bulb temperature and estimates the effectiveness of the fluid cooler. By using this effectiveness, cooling capacity of the fluid cooler is calculated.

This cooling capacity is then used to calculate again the wet-bulb temperature at fluid cooler exit. If the difference between the guess value and calculated of exiting wet-bulb temperature is more than the tolerance limit then iterations are performed until the convergence on the wet-bulb temperature value is obtained.

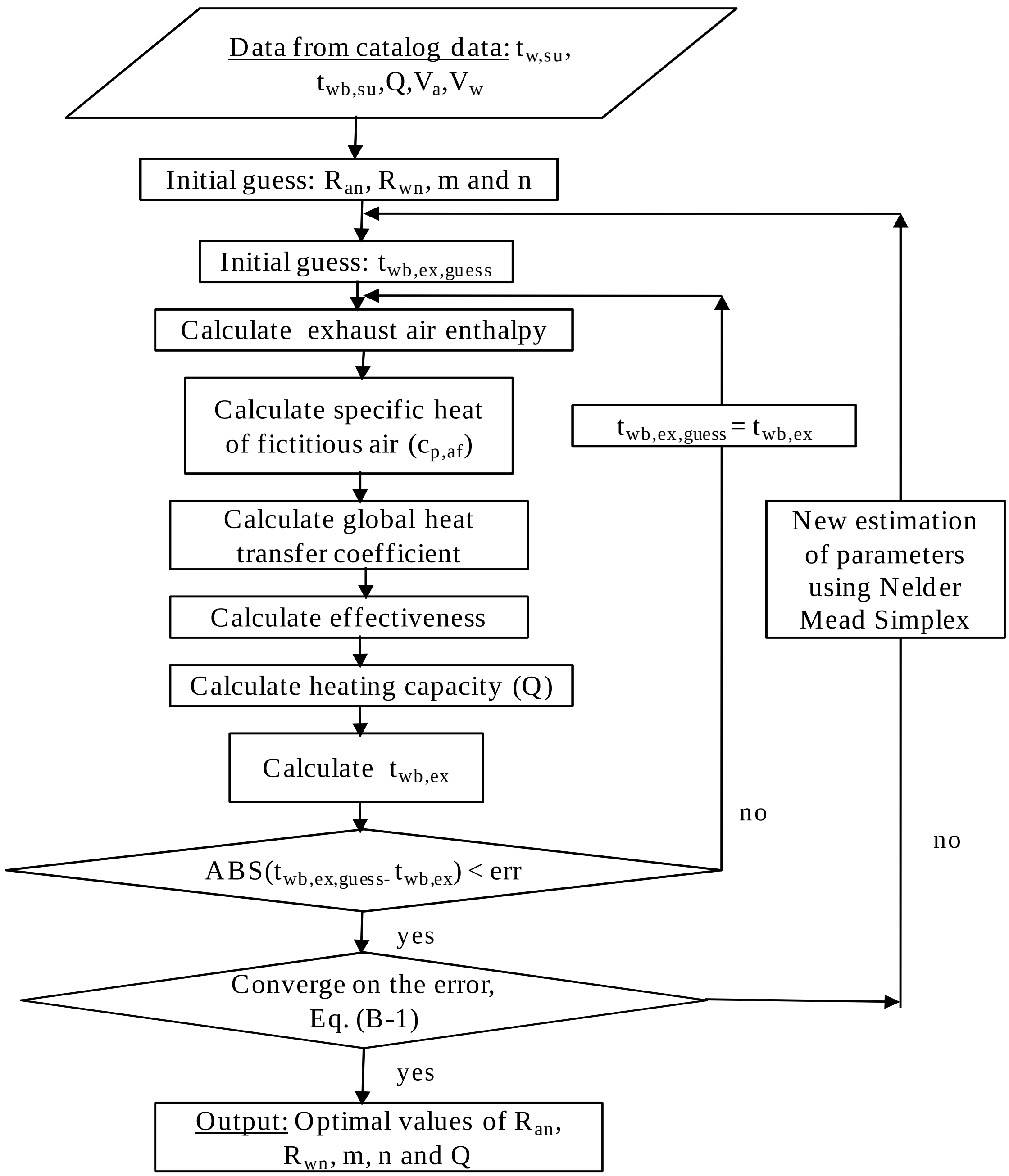

Fig B-1: Flow chart of Lebrun model implemented in VBA

Where,

t = temperature (°C)

V= Volumetric flow rate (m^3/s)

Q = Cooling capacity (W)

c_p = specific heat capacity of water (kJ/kg-K)

err = tolerance of wet-bulb temperature

Subscripts

a = air

ex= exiting

su =supply

w= water

wb= wet-bulb

guess = guessed value

f = fictitious

Once the exiting wet-bulb temperature is fixed, the model uses corresponding cooling capacity to calculate the following objective function:

$$Z = \sum_{i=1}^{N} \left[\frac{Q_{catalog}(i) - Q_{calculated}(i)}{Q_{catalog}(i)}\right]^2 \leq Acc \qquad \text{(B-1)}$$

Where,

N= number of data set from catalog

$Q_{catalog}$ = Catalog cooling capacity (W)

$Q_{calculated}$ = Calculated cooling capacity (W)

Acc = accuracy of the Nelder Mead routine given by the user

Nelder Mead Simplex is used to make new guess of parameters until the value of the above objective function is obtained as less than specified accuracy. Kuester and Mize (1973) implemented the Nelder Mead Simplex into FORTRAN which is the basis of VBA routine of Nelder Mead Simplex used for the present study. For each set of guessed parameters, new exiting wet-bulb temperature is calculated and subsequently cooling capacity and the objective function is calculated. Once the objective function value becomes less than the accuracy, the parameters are reported in the output. The parameters are then used to calculate the cooling capacity for different rating points. The source code of the Lebrun model is shown below.

Source code

```
'INPUT FROM CATALOG
'InletAirWetbulbTemp = inlet air dry bulb temperature (C)
'AirVolFlowRate = Volumetric air flow rate (m^3)
'CoolingCapacity = cooling capacity (KW)
'InletWaterTemp = inlet water temperature (C)
'VS = source side volumetric water flow rate (m^3)

Sub Main()
Dim accuracy As Double, n As Integer,X() As Double, np As Integer, StartTime_ As
Single, FinishTime As Single, TotalTime As Single
Dim i As Integer, guessnum As Integer

StartTime = Timer   'Start Time of the simulation
accuracy   = Worksheets("ParameterEstimator").Cells(1, 2)
                    'Accuracy for Nelder Mead
n = Worksheets ("ParameterEstimator").Cells(2, 2)
                    'Number of Data Set from catalog
fluidtype   = Worksheets("ParameterEstimator").Cells(3, 2) 'Fluid Type 0 or 1
                                             '(Currently 0 is valid fluid type)
guessnum = Worksheets("ParameterEstimator").Cells(10, 3)
                                             'Set of guessed parameters

np = 4     'Numbers of parameters

ReDim InletAirWetbulbTemp(n) As Single, AirVolFlowRate(n) As Single,
CoolingCapacity(n) As Single, InletWaterTemp(n) As Single, WaterVolFlowRate(n)_ As
Single
ReDim X(np + 1, np) As Double
'=============READING INPUT DATA===============
'Read in the catalog data input
 For i = 1 To n
InletAirWetbulbTemp(i) = Worksheets("InputData"). Cells_
                       (T2firstrow + i - 1, T2firstcol)
```

```
AirVolFlowRate(i) = Worksheets("InputData").Cells_
                       (T2firstrow + i - 1, T2firstcol + 1)
CoolingCapacity(i) = Worksheets ("InputData").Cells_
                       (T2firstrow + i - 1, T2firstcol + 2)
InletWaterTemp(i) = Worksheets("InputData").Cells_
                       (T2firstrow + i - 1, T2firstcol + 3)
WaterVolFlowRate(i) = Worksheets("InputData").Cells_
                       (T2firstrow + i - 1, T2firstcol + 4)

Next i

'Check if the input data is entered
If AirVolFlowRate(1) = 0 Then
MsgBox "No input data is entered. Check Worksheet 'INPUT'."
Exit Sub
End If
'===============GENERATING PARAMETERS==============
'Status of the simulation
Application.DisplayStatusBar = True
Application.StatusBar = "SIMULATION STATUS: Reading Input_
                              Data & Generating Parameters"
'Read initial guess of parameters
For i = 1 To np
X(1, i) = Worksheets("ParameterEstimator").Cells(14 + i, guessnum + 1)
Next i

'Check if the initial guess is entered
If X(1, 1) = 0 Then
MsgBox "No initial guess of parameters entered. Check initial guess number."
Exit Sub
End If

Call NelderMead(n, X(), np, accuracy)
Call MI(guessnum, n, np, X())

'Print the parameters generated from Nelder Mead
For i = 1 To np
Worksheets("ParameterEstimator").Cells(20 + i, guessnum + 1) = X(np + 1, i)
Next i

'Analyze the result and calculating errors
Call ResultAnalyst(guessnum, n)

'Status of the simulation
FinishTime = Timer         'End Time of the simulation
TotalTime = FinishTime - StartTime
Application.StatusBar = "SIMULATION STATUS: Ended Succesfully           Simulation
Time=" & TotalTime & " Seconds"
End Sub

Sub ResultAnalyst(guessnum As Integer, n As Integer)
‘Author: Keneth Tang
Dim j As Integer, i As Integer, error As Single, p As_ Single, q As Single

'Calculate %error and write to Worksheet "Result"
For i = 1 To n
error = 100 * (Worksheets("RESULT").Cells(4 + i, 3 + 2 * (guessnum - 1))_       -
Worksheets("RESULT").Cellserror
Next i

Worksheets("ParameterEstimator").Cells(27, guessnum + 1) = RMSError(guessnum,_ n,
                                                                1)
```

```
'Calculate RMS error for CoolingCapacity
Worksheets("ParameterEstimator").Cells(28, guessnum + 1) = _
                         PercentageRMSError(guessnum, n, 1)

 'Calculate %RMS error for W
Worksheets("ParameterEstimator").Cells(29, guessnum + 1) = _
AverageError(guessnum, n, 1)
End Sub

Sub NelderMead(n As Integer, X() As Double, np As Integer, ACC As Double)

‘Author: Keneth Tang

Dim NP1 As Integer, q As Single, p As Single, M As Integer, i As Integer, j As_
Integer, AP As Integer, ITR As Integer,_ Z() As Double,Dim ZHI As Double, ZLO_ As
Double, k As_ Integer, EN As Integer, SUM As Double, EJ As Double, L As_ Integer,
ZCEN As Double, ZREF As Double, ZCON As Double,_ZEX As Double

ReDim XCEN(np + 1, np) As Double, XREF(np + 1, np) As Double, XCON(np + 1, np)_ As
Double, XEX(np + 1, np) As_ Double, Z(np + 1) As Double


Const ITMAX As Single = 1000   'Max number of iteration
Const ALFA As Single = 1   'Reflection coefficient ALFA>0
Const BETA As Single = 0.5'Contraction coefficient 0<BETA<1
Const GAM As Single = 2        'Expansion coefficient
Const A As Single = 0.1



Open "NM_OUPUT.txt" For Output As #1
NP1 = np + 1
q = (A / np * (2 ^ 0.5)) * ((np + 1) ^ 0.5 - 1)
p = (A / np * (2 ^ 0.5)) * ((np + 1) ^ 0.5 + np - 1)
M = np + 1

For i = 2 To M
  AP = 1
  For j = 1 To np
    AP = AP + 1
    If (i = AP) Then
    X(i, j) = X(1, j) + p
    Else
    X(i, j) = X(1, j) + q
    End If
  Next j
Next i

Write #1, Tab(3); "NELDER MEAD OPTIMIZATIION"
Write #1, "N=", n; Tab(10); "ACC="; ACC; Tab(30); "ALFA="; ALFA; Tab(42); "BETA=";
BETA; Tab(56); "GAM="; GAM
Write #1,
Write #1, "Starting Simplex"
Write #1, "----------------------------------"
Write #1,
'-------------BEGIN NELDER MEAD ROUTINE-------------
    ITR = 0
150  For i = 1 To NP1
     Call PE1(n, i, X(), Z(), np, NP1)
     Next i
     ITR = ITR + 1
     If (ITR >= ITMAX) Then GoTo 145
158   Write #1, "Iteration Number", ITR
```

```
ZHI = Max(Z(), NP1)
ZLO = Min(Z(), NP1)
      For i = 1 To NP1
      If (ZHI = Z(i)) Then Exit For
      Next i
      k = i
      EN = np
      For j = 1 To np
        SUM = 0
        For i = 1 To NP1
          If (k = i) Then GoTo 175
          SUM = SUM + X(i, j)
175     Next i
        XCEN(k, j) = SUM / EN
      Next j

      i = k
      Call PE1(n, i, XCEN(), Z(), np, NP1)
      ZCEN = Z(i)
      SUM = 0
For i = 1 To NP1
      If (k = i) Then GoTo 185
      SUM = SUM + (Z(i) - ZCEN) * (Z(i) - ZCEN) / EN
185   Next i
      EJ = (SUM) ^ 0.5
      If (EJ < ACC) Then GoTo 998
      Write #1, "Optimum value of F="; ZLO
      Write #1,
      For j = 1 To np
      XREF(k, j) = XCEN(k, j) + ALFA * (XCEN(k, j) - X(k, j))
      Next j
      i = k
      Call PE1(n, i, XREF(), Z(), np, NP1)
      ZREF = Z(i)

      For i = 1 To NP1
      If (ZLO = Z(i)) Then Exit For
      Next i

      L = i
      If (ZREF <= Z(L)) Then GoTo 240
      For i = 1 To NP1
      If (ZREF < Z(i)) Then GoTo 208
      Next i
      GoTo 215

208   For j = 1 To np
      X(k, j) = XREF(k, j)
      Next j
      GoTo 150

215   For j = 1 To np
      XCON(k, j) = XCEN(k, j) + BETA * (X(k, j) - XCEN(k, j))
      Next j
      i = k
      Call PE1(n, i, XCON(), Z(), np, NP1)
      ZCON = Z(i)
      If (ZCON < Z(k)) Then GoTo 230
For j = 1 To np
      For i = 1 To NP1
      X(i, j) = (X(i, j) + X(L, j)) / 2
      Next i
      Next j
```

```
      GoTo 150

230   For j = 1 To np
      X(k, j) = XCON(k, j)
      Next j

240   For j = 1 To np
      XEX(k, j) = XCEN(k, j) + GAM * (XREF(k, j) - XCEN(k, j))
      Next j

      i = k
      Call PE1(n, i, XEX(), Z(), np, NP1)
      ZEX = Z(i)
      If (ZEX < Z(L)) Then GoTo 255

      For j = 1 To np
      X(k, j) = XREF(k, j)
      Next j
      GoTo 150

255   For j = 1 To np
      X(k, j) = XEX(k, j)
      Next j
      GoTo 150

145   Write #1, "DID NOT CONVERGE IN", ITR
      MsgBox "Fail to converge in 1000 iterations. Change
                          accuracy", , "Interation Problem"
998   Write #1, "FINAL OPTIMUM VALUE OF F=", ZLO
      Write #1, "OPTIMUM VALUES OF VARIABLES "
      Write #1, "-----------------------------------"
      For i = 1 To np
      Write #1, X(NP1, i)
      Next i
Close #1
End Sub

Sub PE1(t As Integer, i As Integer, X() As Double, Z() As Double, np As_ Integer,
NP1 As Integer)
' Author: Chandan Sharma
'INPUT VARIABLES:
'twbsu (C)Supply air wet bulb temperature
'twsu (C) Supply water temperature
'qdota (CFM) Air flow rate from the fan
'mdotw (GPM) Water mass flow rate
'mdotwn (GPM) Water mass flow rate at design conditions
'------------'OUTPUT VARIABLES:
'Q (W) Evaporativwe fluid cooler capacity
'----------------------------------------------------------
'Assumptions:
'The assumptions used in the derivation of the model are as follows:
'1) The humid air is modeles as a fictitios gas whose temperature is the
'   wetbulb temperature of th eair
'2) The air film at the interface is saturated with water vapor.
'3) Lewis number is taken to be one.
'4) The water loss due to evaporation is assumed to be negligible i.e. supply and
exhaust water flow rates are the same.
'---------------'Declare Parameter Variables
Dim Ran As Double, Rwn As Double, M As Single, n As Single

'Declare Subroutine Variables
Dim mdota As Double, mdotw As Double, qdota As Double, qdotan As Double,_ Vdotan As
Single, Vdotwn As Single
```

```
Dim mdotan As Double, mdotwn As Double
Dim twbsu As Single, twbex As Single, twsu As Single, twex_ As Single, haex As_
Double, hasu As Double, twbexlast As_ Double, Dim cpaf As Double, cpr As_ Double,
Caf As Double, Dim Cr As Double, Dim Cmin As Double, Cmax As Double
Dim Rw As Double, Ra As Double, Rfic As Double, Raf As_ Double, Dim AUfic As_
Double, NTUfic As Double
Dim C As Double, q As Double, E As Double
Dim C0 As Single, C1 As Single, C2 As Single, C3 As Single
Dim cpa As Single, Dim iter As Integer, Dim diff As Single
Dim density As Single, Dim WCP As Single, Dim k As Integer

'-----------------------------------------------------------
'mdota : Air mass flow rate (kg/s)
'qdota : Air volumetric flow rate (CFM)
'qdotan: Air volumetric flow rate at nominal conditions(CFM)
'mdotw : Water mass flow rate in the tube i.e. process
                                     'fluid flow rate (GPM)
'mdotan: Air mass flow rate at nominal conditions (kg/s)
'mdotwn: Water mass flow rate at nominal conditions (GPM)
'twbsu : Supply or inlet air wetbulb temperature (C)
'twsu  : Supply or inlet water temperature (C)
'twex  : Exit or outlet water temperature (C)
'hasu  : Supply or inlet air enthalpy (j/kg)
'haex  : Exhaust air enthalpy (j/kg)
'cpaf  : Specific heat for fictitious air (j/kg-k)
'cpr   : Specific heat for water or process fluid (j/kg-k)
'Caf   : Capacity for fictitious air (W/K)
'Cr    : Capacity for water or process fluid (W/K)
'Ra    : Air resistance (m2-K/W)
'Rw    : Water or process fluid resistance (m2-K/W)
'Ran   : Air resistance at nominal conditions (m2-K/W)
Rwn   : Water or process fluid resistance at nominal
                                       'conditions (m2-K/W)
'AUfic : Heat transfer coefficient for fictitious air (W/K)
'e     : Effectiveness of the heat exchanger
'Q     : Capacity/Heat transfer rate of the Evaporative
                                          'fluid cooler (W)
'M=Refrigerant side mass flow rate ratio exponent
'n=Air side mass flow rate ratio exponent
'-----------------------------------------------------------
C0 = 9362.5   ' c0,c1,c2 and c3 are polynomial coefficients for h=f(twb), which
              ' depend on atmospheric pressure
C1 = 1786.1
C2 = 11.35
C3 = 0.98855
Vdotan = Worksheets("ParameterEstimator").Cells(4, 2).VALUE
Vdotwn = Worksheets("ParameterEstimator").Cells(5, 2).VALUE

'The following parameters are calculated for Baltimore 'Aircoil closed circuit
'cooling tower V-series
  Ran = X(i, 1) ^ 2
  Rwn = X(i, 2) ^ 2
  M = X(i, 3) ^ 2
  n = X(i, 4) ^ 2
Z(i) = 0
For k = 1 To t
twbsu = InletAirWetbulbTemp(k)
'Using the polynomial function calculate the supply/inlet 'air enthalpy
hasu = C0 + C1 * twbsu + C2 * twbsu ^ 2 + C3 * twbsu ^ 3
cpa = 1.0057 'Air specific heat (kj/kg-k)
'Guess the exhaust air wetbulb temperature
twbex = twbsu + 7
Call AirDensity(twbsu, density)
```

```
mdota = AirVolFlowRate(k) * density
mdotan = Vdotan * density
mdotwn = Vdotwn * 1000
mdotw = WaterVolFlowRate(k) * 1000
twsu = InletWaterTemp(k)
'Calculate specific heat for entering water
Call WaterSpecificHeat(twsu, WCP)
Cr = WaterVolFlowRate(k) * WCP * 1000
'Add a counter
iter = 0
line1:
'Calculate exhaust air enthalpy
haex = C0 + C1 * twbex + C2 * twbex ^ 2 + C3 * twbex ^ 3
cpaf = (haex - hasu) / (twbex - twbsu)
Caf = mdota * cpaf
If Caf > Cr Then
  Cmin = Cr
  Cmax = Caf
Else
  Cmin = Caf
  Cmax = Cr
End If
C = Cmin / Cmax
'Calculate fictitious resistance AUfic
Ra = Ran * (mdota / mdotan) ^ n
Raf = Ra * cpa / cpaf
Rw = Rwn * (mdotw / mdotwn) ^ M
Rfic = Raf + Rw
AUfic = 1 / Rfic


'Heat exchanger calculation
NTUfic = AUfic / Cmin
If C > 0.995 Then
E = (1 - Exp(-NTUfic * (1 - C))) / (1 - C * Exp(-NTUfic *(1 - C)))
Else
E = NTUfic/ (1+ NTUfic)
Endif
q = E * Cmin * (twsu - twbsu)
'Pass the guess exhaust air wetbulb temp to another place
twbexlast = twbex
'Calculate the exhaust air wetbulb temp from heat balance
twbex = twbsu + q / Caf
iter = iter + 1
diff = Abs(twbex - twbexlast)
If diff > 0.01 And iter <= 10000 Then
  twbex = 0.1 * twbex + twbexlast * 0.9
  GoTo line1
End If

If iter > 10000 Then
  MsgBox "Failed to converge"
End If
twex = twsu - q / Cr


'--------THE OBJECTIVE FUNCTION-------------------

 Z(i) = Z(i) + ((CoolingCapacity(k) - q) / CoolingCapacity(k)) ^ 2
Next k
End Sub

Sub MI(guessnum As Integer, t As Integer, np As Integer, X() As Double)
```

```
'Author: Chandan Sharma
'INPUT VARIABLES:
'twbsu (C)Supply air wet bulb temperature
'twsu (C) Supply water temperature
'qdota (CFM) Air flow rate from the fan
'mdotw (GPM) Water mass flow rate
'mdotwn (GPM) Water mass flow rate at design conditions
'----------------------------------------------------------
'OUTPUT VARIABLES:
'Q (W) Evaporative fluid cooler capacity
'----------------------------------------------------------
'Assumptions:
'The assumptions used in the derivation of the model are as 'follows:
'1) The humid air is models as a fictitious gas whose 'temperature is the wet-
'   bulb temperature of the air
'2) The air film at the interface is saturated with water vapor.
'3) Lewis number is taken to be one.
'4) The water loss due to evaporation is assumed to be negligible i.e. supply
'   and exhaust water flow rates are the same.
'----------------------------------------------------------
'INPUT FROM CATALOG
'InletAirWetbulbTemp = load side inlet air dry bulb (C)
'AirVolFlowRate = load side volumetric air flow rate (m^3)
'CoolingCapacity = cooling capacity (KW)
'HA = source side heat absorbtion (KW)
'InletWaterTemp = source side inlet water temperature (C)
'AMS = source side water mass flow rate (kg/s)
'ER = compressor power input (KW)

'Declare Parameter Variables
Dim Ran As Double, Rwn As Double, M As Single, n As Single

'Declare Subroutine Variables

Dim mdota As Double, mdotw As Double, qdota As Double, qdotan As Double,_ Vdotan As
Single, Vdotwn As Single
Dim mdotan As Double, mdotwn As Double, twbsu As Single, Dim twbex As Single,_ twsu
As Single, twex As Single, haex_ As Double, hasu As Double, twbexlast As_ Double
Dim cpaf As Double, cpr As Double, Caf As Double, Cr As_ Double, Cmin As_ Double,
Cmax As Double, Rw As Double, Ra_ As Double, Rfic As Double, Raf As_ Double, AUfic
As Double
Dim NTUfic As Double,C As Double, q As Double, E As Double
Dim C0 As Single, C1 As Single, C2 As Single, C3 As Single
Dim cpa As Single, iter As Integer, diff As Single
Dim density As Single, WCP As Single, i As Integer, k As _ Integer
'----------------------------------------------------------
'mdota : Air mass flow rate (kg/s)
'qdota : Air volumetric flow rate (CFM)
'qdotan: Air volumetric flow rate at nominal conditions(CFM)
'mdotw : Water mass flow rate in the tube i.e. process fluid flow rate (GPM)
'mdotan: Air mass flow rate at nominal conditions (kg/s)
'mdotwn: Water mass flow rate at nominal conditions (GPM)
'twbsu : Supply or inlet air wetbulb temperature (C)
'twsu  : Supply or inlet water temperature (C)
'twex  : Exit or outlet water temperature (C)
'hasu  : Supply or inlet air enthalpy (j/kg)
'haex  : Exhaust air enthalpy (j/kg)
'cpaf  : Specific heat for fictitious air (j/kg-k)
'cpr   : Specific heat for water or process fluid (j/kg-k)
'Caf   : Capacity for fictitious air (W/K)
'Cr    : Capacity for water or process fluid (W/K)
'Ra    : Air resistance (m2-K/W)
'Rw    : Water or process fluid resistance (m2-K/W)
```

```
'Ran   : Air resistance at nominal conditions (m2-K/W)
'Rwn   : Water or process fluid resistance at nominal
                                       'conditions (m2-K/W)
'AUfic : Heat transfer coefficient for fictitious air (W/K)
'e     : Effectiveness of the heat exchanger
'Q     : Capacity/Heat transfer rate of the Evaporative
                                          'fluid cooler (W)
'M=Refrigerant side mass flow rate ratio exponent
'n=Air side mass flow rate ratio exponent
'----------------------------------------------------------
C0 = 9362.5   ' c0,c1,c2 and c3 are polynomial coefficients for h=f(twb), which
' depend on atmospheric pressure
C1 = 1786.1
C2 = 11.35
C3 = 0.98855
Vdotan = Worksheets("ParameterEstimator").Cells(4, 2).VALUE
Vdotwn = Worksheets("ParameterEstimator").Cells(5, 2).VALUE

'The following parameters are calculated for Baltimore 'Aircoil closed circuit
'cooling tower V-series
  Ran = X(np + 1, 1) ^ 2
  Rwn = X(np + 1, 2) ^ 2
  M = X(np + 1, 3) ^ 2
  n = X(np + 1, 4) ^ 2
For k = 1 To t
twbsu = InletAirWetbulbTemp(k)
'Using the polynomial function calculate the supply/inlet 'air enthalpy
hasu = C0 + C1 * twbsu + C2 * twbsu ^ 2 + C3 * twbsu ^ 3
cpa = 1.0057 'Air specific heat (kj/kg-k)
'Guess the exhaust air wetbulb temperature
twbex = twbsu + 7
Call AirDensity(twbsu, density)
mdota = AirVolFlowRate(k) * density
mdotan = Vdotan * density
mdotwn = Vdotwn * 1000
mdotw = WaterVolFlowRate(k) * 1000
twsu = InletWaterTemp(k)
'Calculate specific heat for entering water
Call WaterSpecificHeat(twsu, WCP)
Cr = WaterVolFlowRate(k) * WCP * 1000

'Add a counter
iter = 0
line1:
'Calculate exhaust air enthalpy
haex = C0 + C1 * twbex + C2 * twbex ^ 2 + C3 * twbex ^ 3
cpaf = (haex - hasu) / (twbex - twbsu)
Caf = mdota * cpaf
If Caf > Cr Then
  Cmin = Cr
  Cmax = Caf
Else
  Cmin = Caf
  Cmax = Cr
End If
C = Cmin / Cmax
'Calculate fictitious resistance AUfic
Ra = Ran * (mdota / mdotan) ^ n
Raf = Ra * cpa / cpaf
Rw = Rwn * (mdotw / mdotwn) ^ M
Rfic = Raf + Rw
AUfic = 1 / Rfic
```

```
'Heat exchanger calculation
NTUfic = AUfic / Cmin
If C > 0.995 Then
E = (1 - Exp(-NTUfic * (1 - C))) / (1 - C * Exp(-NTUfic *(1 - C)))
Else
E = NTUfic/ (1+ NTUfic)
Endif
q = E * Cmin * (twsu - twbsu)
'Pass the guess exhaust air wetbulb temp to another place
twbexlast = twbex
'Calculate the exhaust air wetbulb temp from heat balance
twbex = twbsu + q / Caf

iter = iter + 1
diff = Abs(twbex - twbexlast)
If diff > 0.01 And iter <= 10000 Then
  twbex = 0.1 * twbex + twbexlast * 0.9
  GoTo line1
End If
If iter > 10000 Then
  MsgBox "Failed to converge"
End If
twex = twsu - q / Cr
Worksheets("RESULT").Cells(4 + k, 3 + 2 * (guessnum - 1)) = q
Next k
End Sub

Sub AirDensity(t As Single, density As Single)
‘Author: Jeff D. Spitler
'returns air density in kg/m3 when given
'T in C
Dim p As Single, R As Single, T_K As Single
        p = 101.325 'kpa
        R = 0.28704
        T_K = t + 273.15
        density = p / (R * T_K)
End Sub
Sub WaterSpecificHeat(TW As Single, WCP As Single)
'Author: Jeff D. Spitler
'-------------------------------------------------------
'Specific heat of water at 1 atmosphere, 0 to 100 C.  'Equation from linear
'least-squares regression of data from  'CRC Handbook '(op.cit.)page D-174; in
'J/g-C (or kJ/kg-C).
'For temps > 100, fit to data from Karlekar & Desmond      '(saturated).
Dim ACP0 As Single, ACP1 As Double, ACP2 As Double, ACP3 As Single, ACP4 As_
Double, ACP5 As Single, ACP6 As Single,_ ACP7 As Single, ACP8 As Double
        ACP0 = 4.21534
        ACP1 = -0.00287819
        ACP2 = 0.000074729
        ACP3 = -0.000000779624
        ACP4 = 0.000000003220424
        ACP5 = 2.9735
        ACP6 = 0.023049
        ACP7 = -0.00013953
        ACP8 = 0.000000309247
WCP = ACP0 + TW * ACP1 + (TW ^ 2) * ACP2 + (TW ^ 3) * ACP3 + (TW ^ 4) * ACP4
If (TW > 100) Then WCP = ACP5 + TW * ACP6 + (TW ^ 2) * ACP7 + (TW ^ 3) * ACP8
'Return
End Sub
```

VITA

Chandan Sharma

Candidate for the Degree of

Master of Science

Thesis: DEVELOPMENT OF DRY AND EVAPORATIVE FLUID COOLER MODELS FOR ENERGYPLUS

Major Field: Mechanical Engineering

Biographical:

Education:
Completed the requirements for the Master of Science in Mechanical Engineering at Oklahoma State University, Stillwater, Oklahoma in December, 2009.

Name: Chandan Sharma Date of Degree: December, 2009

Institution: Oklahoma State University Location: Stillwater, Oklahoma

Title of Study: DEVELOPMENT OF DRY AND EVAPORATIVE FLUID COOLER MODELS FOR ENERGYPLUS

Pages in Study: 116 Candidate for the Degree of Master of Science

Major Field: Mechanical Engineering

Dry and evaporative fluid cooler models were developed and implemented in the EnergyPlus simulation environment. First, the operation of the fluid coolers and a comparison with cooling towers were presented. The literature review has shown various models of the fluid coolers. The Lebrun model was chosen as the basis for evaporative fluid cooler model whiles the ε-NTU correlation for cross flow heat exchanger with both streams unmixed was used for the dry fluid cooler model. The model sensitivity for various input parameters has been analyzed. The evaporative fluid cooler model was validated by using manufacturer's catalog data. The HVACSIM+ dry fluid cooler model (type 762) was used to verify the EnergyPlus dry fluid cooler model. Similarly, The Lebrun model was implemented in VBA to verify EnergyPlus evaporative fluid cooler model. The new models produced results that were comparable to previous models. The implementation of the models was also verified by arranging them in series and parallel configurations and analyzing their performance. The models performed as expected for both the configurations.

ADVISER'S APPROVAL: Dr. Daniel E. Fisher