Model Based Building Chilled Water Loop Delta-T Fault Diagnosis

Lei Wang, Ph.D., P.E. James Watt, P.E. Juan Zhao, Ph.D., P.E
Energy Systems Laboratory, Texas A&M University
College Station, Texas

Abstract
Improving chilled water delta-T, which is the temperature difference of chilled water supply and return temperature, in campus buildings that are connected to a central distribution loop will not only improve the power consumption of the building through reduced tertiary (building) pumping power but the impact on the central distribution system and chiller efficiencies will be even greater. A degraded delta-T is almost inevitable and it can be expected to fall to about one-half to two-thirds of design at low loads (Taylor, 2002) due to various causes, such as air entering and leaving temperatures, chilled water supply temperature, type and effectiveness of flow control valves, tertiary connection configuration types and operation, coil cooling loads, air economizers, etc. However, in most variable-flow chilled water with 2-way control valve systems, the root cause of low delta-T is at coil side (Zhang, 2012), for example the geometric configuration of coil. This paper firstly discusses chilled water coil heat exchanger model results to help define methods for detecting opportunities for improved delta-T when analyzing campus building systems for performance optimization measures. Meanwhile, the author developed an effectiveness-NTU cooling coil models for a case study building containing chilled water coils with a range of design configurations to study cooling coil delta-T characteristics under various conditions in order to diagnose the low delta-T imposed on the chilled water distribution loop by the building's chilled water system under various loading conditions. The results show model-based building chilled water Loop delta-T fault diagnosis is an effective way to evaluate existing building chilled water loop delta-T performance and identify avoidable or resoluble causes for improving chilled water loop delta-T.

Key Words: chilled water loop Delta-T, chilled water coil heat exchanger model, central distribution system, variable-flow Chilled Water System, energy efficiency

1. Introduction

Improving chilled water delta-T, which is the temperature difference of chilled water supply and return temperature, in campus buildings that are connected to a central distribution loop will not only improve the power consumption of the building through reducing tertiary (building) pumping power but the impact on the central distribution system and chiller efficiencies will be even greater. However, almost every real chilled water system is plagued by low delta-T syndrome, particularly at low cooling loads. Delta-T degrading is almost inevitable can it can be expected to fall to about one-half to two thirds of design at low loads (Taylor, 2002). Many factors contribute to the loop chilled water delta-T, such as chilled water supply temperature, cooling coil air entering and leaving temperature, type of flow control valves, tertiary connection types coil cooling loads, air economizers, etc. However, in most variable-flow chilled water with 2-way control valve systems, the root cause of low delta-T is at coil side (Zhang, 2012).

Various studies have discussed how to keep a higher delta-T for chilled water systems. Taylor (2002) addressed the causes of degrading delta-T along with mitigation measures. The causes of low delta-T syndrome are broken into three categories: causes that can be avoided, causes that can be resolved but may not result in energy savings, and causes that cannot be avoided. The water laminar flow in the cooling coil is introduced in the second category. He addressed why delta-T degradation would usually occur and how to design around that eventuality to maintain chiller plant efficiency, despite a degrading delta-T. The focus was to improve chiller low load performance and try to fully load the chiller. Wang et al. (2006) studied the factors, such as cooling coil size, chilled water supply temperature, outside air flow, space cooling load, coil fouling condition, and so on, which may cause low delta-T syndrome in a district cooling system. The influences for the delta-T of these factors are compared in the simulation with the conclusion that the main cause for the low delta-T syndrome for the system in the simulation is the improper use of 3-way control valves. Fiorino(1996) recommended 25 “best practices” to achieve high chilled water delta-T ranged from component selection criteria to distribution system configuration guidelines. An example was provided that low delta-T will prevent the building’s cooling from being satisfied at peak cooling load conditions. It was pointed out the chilled water delta-T in a variable flow hydronic cooling system should be equal to design at full load and greater than design at part load. Moe (2005) proposed to apply pressure independent control valves to achieve high delta-T across coils. Conventional 2-way control valves were replaced with pressure independent control valves at coils and used to control the process. This valve could eliminate the effect of sudden pressure difference variations on the coil flow rate control or authority distortion. These studies proposed various qualitative analyses to delta-T degradation based on practical observation or simplified engineering calculations. However, there is still shortage of detailed quantitative analyses on coil performance change at various operating conditions.

In this paper, the author develops a cooling coil model with design geometric configurations to study cooling coil delta-T characteristics under various conditions. The simulation results will support CC® engineers to evaluate existing building chilled water loop delta-T performance and identify avoidable or resoluble causes.

2. Energy impact of degrading Delta-T

Figure 1 shows a schema of chiller plant serving several building in a larger facility, such as a university campus. The system is piped in a typical primary-secondary manner with some tertiary pumps at remote buildings.

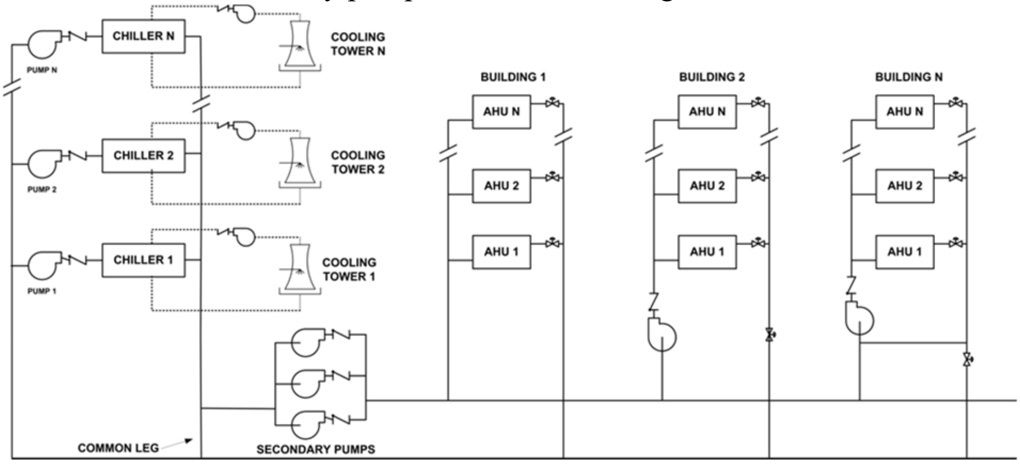

Fig. 1 Typical chilled water plant and distribution system

$$Q = m_w C_{pw}(CHWRT - CHWST) \quad (1)$$

If the delta-T in a system is low, at least two problems results: increased pump energy usage and either an increase in chiller and cooling tower energy usage or a failure to meet cooling loads. The increase in chilled water pump energy is obvious. According to Equation 1, any reduction in delta-T must cause a proportional increase in chilled water flow rate. Pump energy, theoretically, is proportional to the cube of the flow rate, so any increase in flow will have a much higher increase in pump energy. In real systems, actual pump energy impact will be less than this theoretical relationship suggests, but the impact is significant.

The impact on chiller energy usage is more complex to determine and will be a function of how the chillers are controlled. There are two basic chiller start/stop control strategies, one based on system flow rate and the other based on thermal load. Ideally, the two strategies would be effectively the same since flow and load should track in a variable-flow system. However, when flow and load do not track, when delta-T falls, neither strategy can work ideally.

The flow-based chillers strategies stage chillers and primary chilled water pumps in an attempt to keep the primary system flow larger than the secondary system flow. In this way, the secondary supply water temperature is equal to the primary water temperature leaving the chillers. When flow in the secondary exceeds the primary, another primary water pump and chiller associated cooling tower and condenser pump are started. A pump and chiller are shut off when flow in the common leg exceeds that one pump.

The load-based strategy measures system load or indirect indication of load such as return water temperature. Chillers are started when the operating chillers are operating at their maximum capacity. Chillers are stopped when the measured load is less than the operating capacity by the capacity of one chiller.

The flow-based control system will always make sure loads are met by starting additional chillers and pumps to keep the primary system flow larger than the secondary flow. But this means that chillers are not fully loaded when delta-T is below design. For example, assume the system was sized a 14°F (7.8 °C) delta-T on both the primary and secondary sides. If the system were at 50% load but the actual delta-T was only 7 °F (3.9°C), all the chillers, cooling towers, condensing pumps and primary pumps in the plant would have to operate to keep the primary flow up. This wastes pumps, chillers and other auxiliary equipment energy since the chillers would all be operating at 50% of capacity, less than the 65% to 85% range where efficiency is typically maximized for fixed speed chillers, and the other auxiliary equipment, cooling towers and condensing pumps, would all also be operating.

The load-based control system would not start a new chiller until the operating chillers were loaded. As delta-T degrades, secondary flow increases, causing water in the common leg to flow from the secondary return back into secondary pumps. This causes the secondary supply water temperature to rise, which in turn causes coil performance to degrade, which in turn causes control valves to open more to demand more flow, which in turn causes ever increasing flow in the secondary and ever warmer supply water temperatures. Eventually, coils will starve, their control valves will be wide open, and temperature control is lost. The system controlling chiller stating would be obvious to these problems; it would not start more pumps and chillers since the operating chillers were not fully loaded.

3. Cooling coil Model

Chilled water cooling coils are often fin and tube heat exchangers, which consist of rows of tubes that pass through sheets of formed fins. As the air passes through the coil and contacts the cold fin surfaces, heat transfers from the air to the chilled water flowing through the tubes. A wide range of models for heat exchangers is currently available. The effectiveness-NTU model (Braun,1989) is used in simulating the cooling coil performance. This model simulates the performance of cooling coils utilizing the effectiveness model for counter-flow geometries. The performance of multi-pass cross flow heat exchangers approaches that of counter-flow devices when the number of rows is greater than four. The minimum possible of the exit air through a cooling coil is that the exit air was saturated at a temperature equal to that of the incoming water stream. The air-side heat transfer effectiveness is defined as the ratio of the air enthalpy difference to the maximum possible air enthalpy difference if the exit air was at the minimum possible enthalpy. Assuming that the Lewis number equals one, Braun (1989) has shown that the air effectiveness can be determined using the relationships for sensible heat exchanges with modified definitions for the number of transfer units and the capacitance rate ratios. Fin efficiencies are required in order to calculate heat transfer coefficients between air stream and coil. Threlkeld(1970) notes that the performance of rectangular-plate fins of uniform thickness can be approximated by defining equivalent annulus fins. Efficiencies are calculated for annulus fins of uniform thickness ignoring end effects. Polynomial approximations are used to evaluate the

Bessel function used in calculating the efficiencies.

If the coil surface temperature at the air outlet is greater than the dew point of the incoming air, then the coil is completely dry throughout and standard heat exchanger effectiveness relationships apply.

In terms of the air-side heat transfer effectiveness, the dry coil heat transfer is

$$\dot{Q}_{dry} = \varepsilon_{dry,a}\dot{m}_a C_{pm}\,(T_{a,i} - T_{w,i}) \quad (2)$$

Where,

$$\varepsilon_{dry} = \frac{1 - \exp(-Ntu_{dry}\,(1 - C^*))}{1 - C^* \exp(-Ntu_{dry}\,(1 - C^*))}$$

$$C^* = \frac{\dot{m}_a C_{pm}}{\dot{m}_w C_{pw}}$$

$$Ntu_{dry} = \frac{UA_{dry}}{\dot{m}_a C_{pm}}$$

The airside convection coefficient is calculated using the correlations developed by Elmahdy and biggs (1979). The average heat transfer Colburn J-factor is:

$$J = C_1 {Re_a}^{C_2} \quad (3)$$

The quantities C_1 and C_2 are constant for a particular coil over the airside Reynolds number (Re_a) range of 200 to 2000.

If the coil surface temperature at the air inlet is less than the dew point of the incoming air, then the coil is completely wet and dehumidification occurs throughout the coil. For a completely wet coil, the heat transfer is

$$\dot{Q}_{wet} = \varepsilon_{wet,a}\dot{m}_a(h_{a,i} - h_{s,w,i}) \quad (4)$$

Where,

$$\varepsilon_{wet,a} = \frac{1-\exp(-Ntu_{wet}(1-m^*))}{1- m^* \exp(-Ntu_{wet}(1-m^*))}$$

$$m^* = \frac{\dot{m}_a C_s}{\dot{m}_{w,i} C_{pw}}$$

$$\mathrm{Ntu_{wet}} = \frac{\mathrm{UA_{wet}}}{\dot{\mathrm{m}}_\mathrm{a}}$$

UA's are normally given in terms of a temperature difference, but in this case UAwet is the heat conductance in terms of an enthalpy difference. Threlkeld (1970) gives a method for computing fin efficiencies for wet coils using the relationships available for dry coils.

Depending upon the entering conditions and flow rates, only part of the coil may be wet. A detail analysis involves determining the point in the coil at which the surface temperature equals the dew point of the entering air. In order to calculate the heat transfer through the cooling coil, the relative areas associated with the wet and dry portions of the coil must be determined Braun (1989) presents the following method for calculating the heat transfer in a partially wet coil. The fraction of the coil surface area that is dry is

$$\mathrm{f_{dry}} = \frac{-1}{\mathrm{K}} \ln\left[\frac{(\mathrm{T_{dp}} - \mathrm{T_{w,o}}) + \mathrm{C^*}\,(\mathrm{T_{a,i}} - \mathrm{T_{dp}})}{\left(1 - \frac{\mathrm{K}}{\mathrm{Ntu_0}}\right)(\mathrm{T_{a,i}} - \mathrm{T_{w,o}})}\right] \quad (5)$$

Where,

$$\mathrm{K} = \mathrm{Ntu_{dry}}\ (1 - \mathrm{C^*})$$

The effectiveness for the wet and dry portions of the coil is

$$\varepsilon_{\mathrm{wet,a}} = \frac{1 - \exp(-(1 - \mathrm{f_{dry}})\mathrm{Ntu_{wet}}(1 - \mathrm{m^*}))}{1 - \mathrm{m^*}\exp(-(1 - \mathrm{f_{dry}})\mathrm{Ntu_{wet}}(1 - \mathrm{m^*}))} \quad (6)$$

$$\varepsilon_{\mathrm{dry,a}} = \frac{1 - \exp(-\mathrm{f_{dry}}\mathrm{Ntu_{dry}}(1 - \mathrm{C^*}))}{1 - \mathrm{C^*}\exp(-\mathrm{f_{dry}}\mathrm{Ntu_{dry}}(1 - \mathrm{C^*}))} \quad (7)$$

The water temperature at the point where condensation begins is

$$\mathrm{T_{w,x}} = \frac{\mathrm{T_{w,i}} + \frac{\mathrm{C^*}\varepsilon_{\mathrm{wet,a}}\left(\mathrm{h_{a,i}} - \mathrm{h_{s,w,i}}\right)}{\mathrm{Cpm}} - \mathrm{C^*}\varepsilon_{\mathrm{wet,a}}\varepsilon_{\mathrm{dry,a}}\mathrm{T_{a,i}}}{(1 - \mathrm{C^*}\,\varepsilon_{\mathrm{wet,a}}\varepsilon_{\mathrm{dry,a}})} \quad (8)$$

The exit water temperature is

$$T_{w,o} = C^*\varepsilon_{dry,a}T_{a,i} + (1 - C^*\varepsilon_{dry,a})T_{w,x} \quad (9)$$

The water side heat transfer coefficient is determined using standard turbulent flow relations in effectiveness-NTU model (Braun,1989). The most commonly used one for fully developed turbulent flow inside smooth round tubes is the Dittus-Boelter correlation (Dittus and Boelter 1930).

The Effectiveness-NTU coil model is a forward cooling coil model calculates the coil cooling capacity from the entering air and water flow rates and temperatures. However, the real control logic is to determine the sole chilled water flow rate at give air discharging dry-bulb temperature set point. The corresponding chilled water leaving temperature will be calculated from the energy conservation principle.

4. Case study building

The case study building, pictured below in Figure 1, was constructed in 1990 and is located on the west campus of Texas A&M University in College Station, Texas, US (see Figure 1 below). It consists primarily of laboratories and offices, with a few classrooms, a dining center, a computer lab, and other miscellaneous spaces. The building has four floors for a total area of 166,079 square feet (14,947m^2).

Fig. 2. Case study building

The chilled water system in the building utilizes two 20 hp, 840 gpm (190 m^3/hr) chilled water pumps, with VFDs under EMCS control, and operates on a lead/lag schedule. The chilled water pumps and the related building return valve were controlled to maintain the minimum of three end loop DPs at its set point.

The HVAC system in the building consists of eight single-duct, variable air volume (VAV) air handling units (AHUs) and two small constant volume air handling units. All air handling units, pumps and terminal boxes are operated by Siemens DDC controls system. The total design maximum supply flow in the building is 201,670 cfm (95,179 L/s), of which by design a minimum of 161,400 cfm (76,409 L/s) is outside air. The design information of chilled water coils are presented in table 1

Table 1 Chilled water coils design information (IP and SI)

Unit	Service	Supply cfm	Min Outside Air cfm	Max Outside Air cfm	Design Area SQFT	ENT. Air D.B °F	ENT. Air W.B °F	LVG. Air D.B °F	LVG. Air W.B °F	FIN /IN
AHU L1	LABS	44,500	44,500	44,500	90	96	76	50.7	50.7	14
AHU L2	LABS	45,000	45,000	45,000	90	96	76	50.9	50.9	14
AHU L3	LABS	45,000	45,000	45,000	90	96	76	50.7	50.7	14
AHU L4	ANIMAL ROOM	11,760	11,760	11,760	29.4	96	76	50.4	50.4	14
AHU LS	SEMINAR	4,500	1,310	4,500	11.5	83.8	69	50.7	50.7	8
AHU LB	BOOKSTORE	4,650	460	4,650	11.5	79.8	63.8	50.8	50.5	8
AHU LC	COPYSTORE	4,300	430	4,300	11.5	79.8	63.8	50.7	50.5	8
AHU LD	DINING	14,160	7,840	14,160	29.4	89.4	71.5	50.5	50.5	14
AHU LO	OFFICES	19,000	5,600	19,000	42.8	83.3	66.8	51.8	51.5	8
AHU SG	SWITCHGEAR	8,800	0	8,800	20.4	90	72	52.5	52.5	14

Unit	Service	Supply L/s	Min Outside Air L/s	Max Outside Air L/s	Design Area m^2	ENT. Air D.B °C	ENT. Air W.B °C	LVG. Air D.B °C	LVG. Air W.B °C	FIN /cm
AHU L1	LABS	21,002	21,002	21,002	8.36	35.6	24.4	10.4	10.4	5.5
AHU L2	LABS	21,238	21,238	21,238	8.36	35.6	24.4	10.5	10.5	5.5
AHU L3	LABS	21,238	21,238	21,238	8.36	35.6	24.4	10.4	10.4	5.5
AHU L4	ANIMAL ROOM	5,550	5,550	5,550	2.73	35.6	24.4	10.2	10.2	5.5
AHU LS	SEMINAR	2,124	618	2,124	1.07	28.8	20.6	10.4	10.4	3.1
AHU LB	BOOKSTORE	2,195	217	2,195	1.07	26.6	17.7	10.4	10.3	3.1
AHU LC	COPYSTORE	2,029	203	2,029	1.07	26.6	17.7	10.4	10.3	3.1
AHU LD	DINING	6,683	3,700	6,683	2.73	31.9	21.9	10.3	10.3	5.5
AHU LO	OFFICES	8,967	2,643	8,967	3.98	28.5	19.3	11.0	10.8	3.1
AHU SG	SWITCHGEAR	4,153	-	4,153	1.90	32.2	22.2	11.4	11.4	5.5

The eight AHUs are grouped into two types: Lab AHUs and office AHUs. The lab AHUs are 100% OA, while office AHUs minimum outside airflow is about 30% of total supply airflow. The AHU L2 and AHU O are selected as representative of lab AHU and office AHU respectively. The cooling coil geometry configuration is the inherent factor determining the coil delta-T characteristics. The geometry parameters of AHU L2 and AHU O are presented in table 2

Table 2 Geometry parameters of AHU L2 and AHU O (IP and SI units)

No	Parameters	AHU L2 IP	AHU L2 SI	AHU O IP	AHU O SI

1	Width	130	inch	330.2	cm	102	inch	259.1	cm
2	Height	90	inch	228.6	cm	55	inch	139.7	cm
3	Number of rows	8				6			
4	Tube outside diameter	0.5	inch	1.3	cm	0.5	inch	1.3	cm
5	Tube inside diameter	0.45	inch	1.1	cm	0.45	inch	1.1	cm
6	Tube material	copper				copper			
7	Fin	14	Fin/Inch	5.5	Fin/cm	10	Fin/Inch	3.9	Fin/cm
8	Fin thickness	0.008	inch	0.02	cm	0.008	inch	0.02	cm
9	Fin material	Aluminum				Aluminum			cm
10	Tubes distance (perpendicular to air flow)	1.25	inch	3.2	cm	1.25	inch	3.2	cm
11	Tube Spacing (Parallel to air flow)	1	inch	2.5	cm	1.25	inch	3.2	cm

5. Model Calibration

The two cooling coil models are calibrated by design performance data and field measure data. The supply air flow, air temperature and relative humidity before and after cooling coils, chilled water supply and return temperatures of AHU L2 were trended for every 15 minutes. Since the AHU O has no flow station, the coil design performance data and field snapshot measured data are used to calibrate AHU O model. The measured and simulated chilled water delta-T for AHU L2 and AHU O are presented in Figure 2 and Figure 3. The cooling coil design performance and field measurement data of AHU O are shown in table 3.

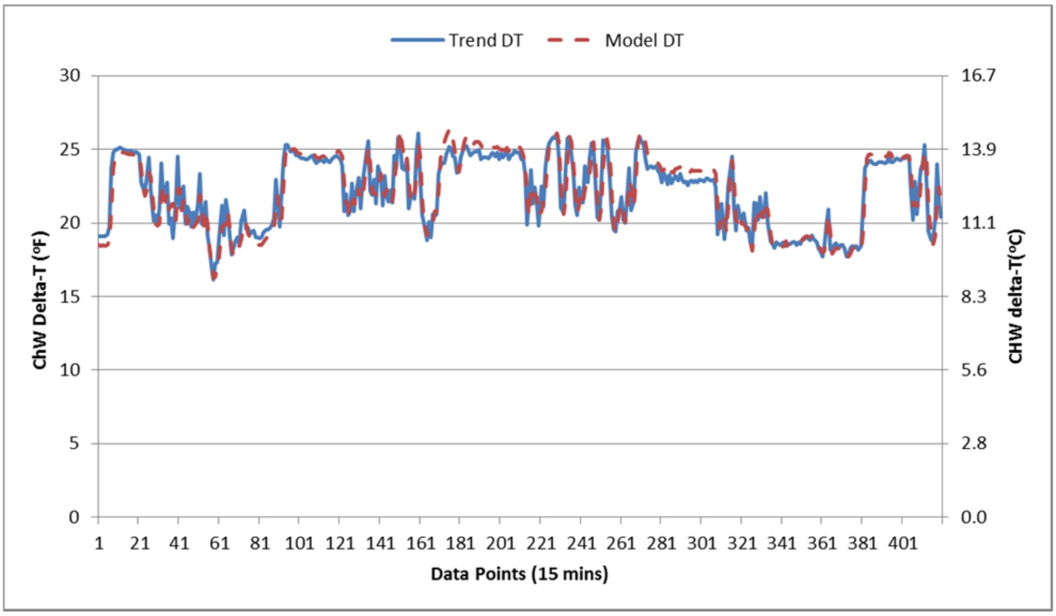

Fig. 3 AHU L2 chilled water delta-T(Model Vs. Trending)

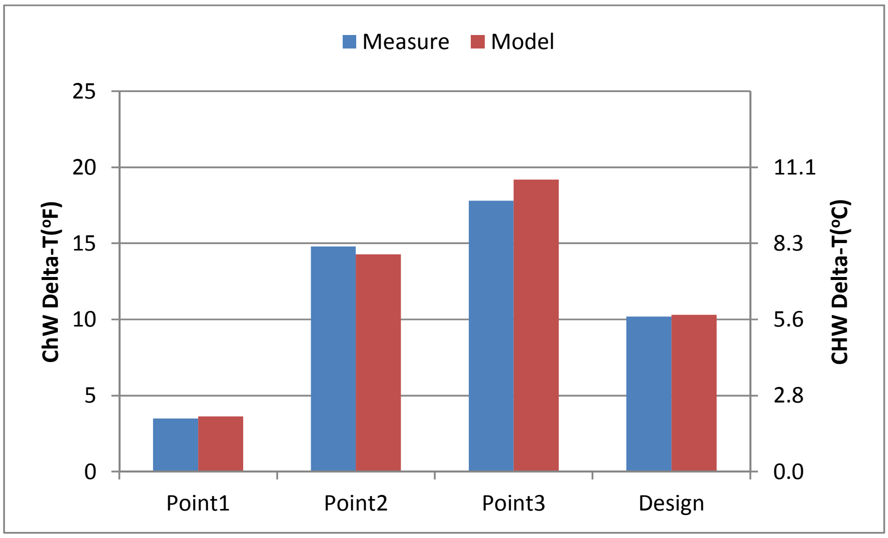

Fig. 4 AHU O chilled water delta-T (IP and SI units)

Table 3 AHU O chilled water calibration results (IP and SI units)

AHU O	Air Side					Waterside			Measure Delta-T	Model Delta-T	Error
	flow	Before		After coil		Supply	Return	flow			
IP	CFM	T (°F)	RH (%)	T (°F)	RH (%)	T (°F)	T (°F)	GPM	°F	°F	%
1	6,450	74.9	61.6%	45.6	89%	42.8	46.3	224	3.5	3.6	3%
2	6,450	97.1	26%	50.8	95%	42.8	57.6	55	14.8	14.3	-4%
3	6,450	97.9	25%	55.5	97%	42.8	60.6	35	17.8	19.2	8%
Design	19,000	83.3	42%	51.7	94%	44	54.2	177	10.2	10.3	1%
SI	L/s	T (°C)	RH (%)	T (°C)	RH (%)	T (°C)	RH (%)	L/s	°C	°C	%
1	6,450	74.9	61.6%	45.6	89%	42.8	46.3	224	3.5	3.6	3%
2	6,450	97.1	26%	50.8	95%	42.8	57.6	55	14.8	14.3	-4%
3	6,450	97.9	25%	55.5	97%	42.8	60.6	35	17.8	19.2	8%
Design	19,000	83.3	42%	51.7	94%	44	54.2	177	10.2	10.3	1%

Both AHU L2 and AHU O models show a good agreement with measured data. There is 1% error in design condition. The CV(RMSE) is 6% for AHU O model. AHU L2 model underestimates the dynamic character of coil performance in some points between point 25 and 50, but the overall trend of calculation results agree with the measure data. The AHU L2 model CV(RMSE) error is 2.9%.

6. Building chilled water Loop Delat-T fault Diagnosis

The building chilled water loop delta-T versus cooling load tonnage and outside air temperature is presented in figure 4 and figure 5.

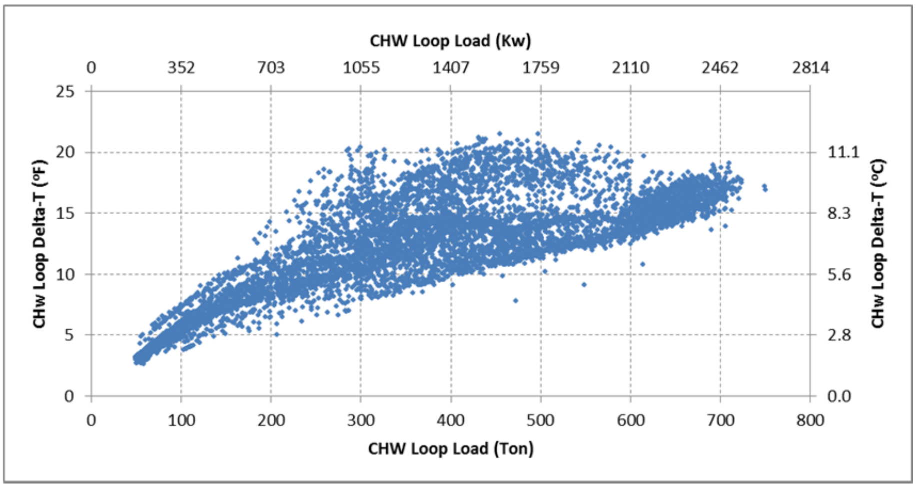

Fig. 5. Chilled water loop Delta-T Vs. cooling load (IP and SI units)

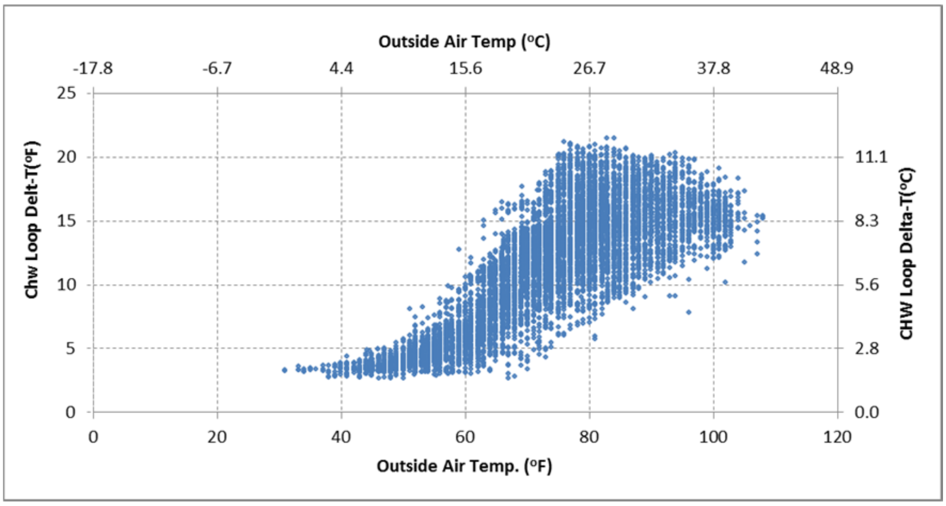

Fig. 6. Chilled water loop Delta-T Vs. outside air temperature (IP and SI units)

The maximum value of existing building chilled water loop delta-T is about 22 °F (12.2°C), while the minimum chilled water loop delta-T is only about 3°F. The design delta-T of cooling coils is 16°F (8.9°C) and 10°F (5.6°C) for Lab AHU and Office AHU respectively. The chilled water loop delta-T can be affected by various causes, such as air entering and leaving temperatures, chilled water supply temperature, type and effectiveness of flow control valves, coil cooling loads and air economizers, etc. Therefore, it is a big challenge for CC engineers to evaluate the chilled water loop delta-T performance and identify opportunities to improve chilled water loop delta-T without a model supported. A calibrated cooling coil model will be a very useful tool to predict the ideal chilled water loop delta pattern versus cooling load or outside air temperature under different scenario.

The airside and water side conditions are the extrinsic factors determining the cooling coil delta-T. In this study, the dry bulb temperature of weather data are divided into 33 bins (40°F ~104°F, 4.4°C ~40°C) and the average dry-bulb and wet bulb temperatures in each dry-blub bin are calculated and used as the outside air temperature profiles. The average chilled water supply temperature in each dry-blub bin is used as the chilled water supply temperature. The space cooling and heating set points are 70°F(21.1°C) and 75°F(23.9°C) respectively. The mixed air temperatures of office AHU are calculated based on outside air temperature, return air temperature and outside air percentage. Each type AHU is calculated under three different supply air flow: minimum, average and maximum. The air flow ratios for office AHU are 30%, 60% and 80% of the design airflow (19,000CFM, 8,967L/s) and for lab AHU are 60%, 70% and 90 % of the design airflow (21,237CFM,).

At the beginning of the fault diagnosis, the measured building chilled water loop delta-T is compared with the simulated cooling coil chilled water delta-T of office and Lab AHU as a function of outside air temperature with different supply airflow, when the cold deck temperature is maintained at normal set point 55°F(12.8°C).

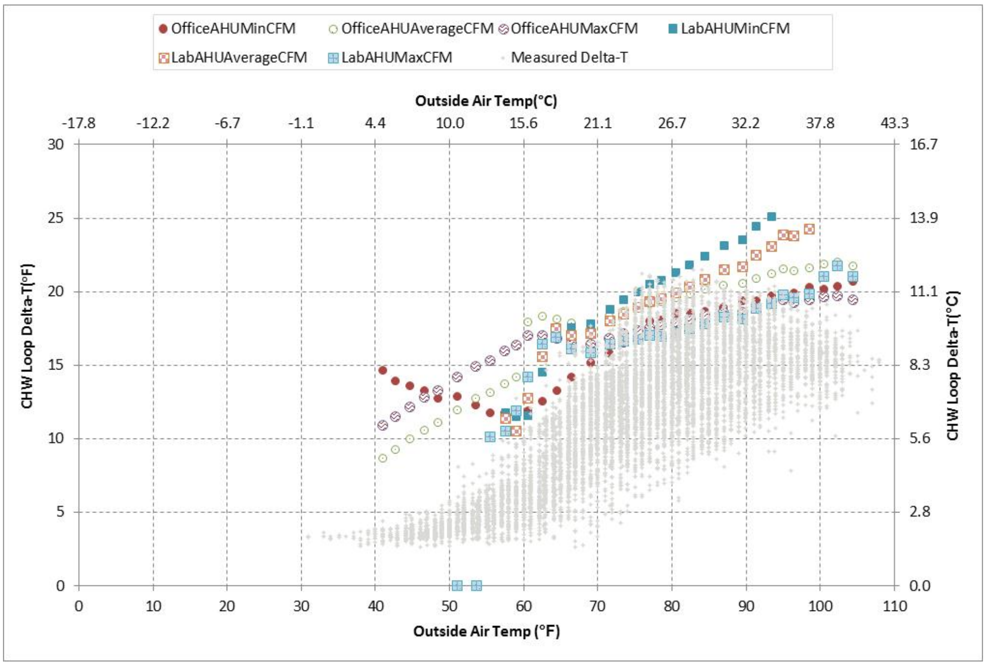

Fig. 7 Measured and simulated chilled water Delta-T

(Cold deck temp. 55°F, 12.8°C, IP and SI units)

It clearly shows the simulated cooling coil chilled water delta-T for both types AHU is higher than measured chilled water loop DT for most of the points. It indicates that there is a good opportunity to improve this building chilled water loop delta-T. It also illustrates that, the chilled water delta-T does not always increase with outside air temperature increase. When the office AHU airflow is minimum, the chilled water delta-T decreases from 14.7°F (8.2°C) to 11.9°F(6.6°C) as the outside air temperature increases from 40°F(4.4°C) to 60°F(15.6°C), then it is rising with outside air temperature increasing. With the load decreasing, the significant increase in water film resistance at low flows would still support the notion that delta-T in the laminar flow region should fall. But there is another factor occurring at the same time that more than offsets this rise in heat transfer resistance: the low flow rate through the coil effectively “sees” an oversized coil, a large amount of heat transfer area relative to the amount of water running through the coil. The water stays in the coil longer and more heat is transferred, which causes the temperature to increase rather than decrease. The chilled water delta-T will be the worst case if chilled water valve is fully open. Figure 7 presents the simulated results when chilled water valve is fully open.

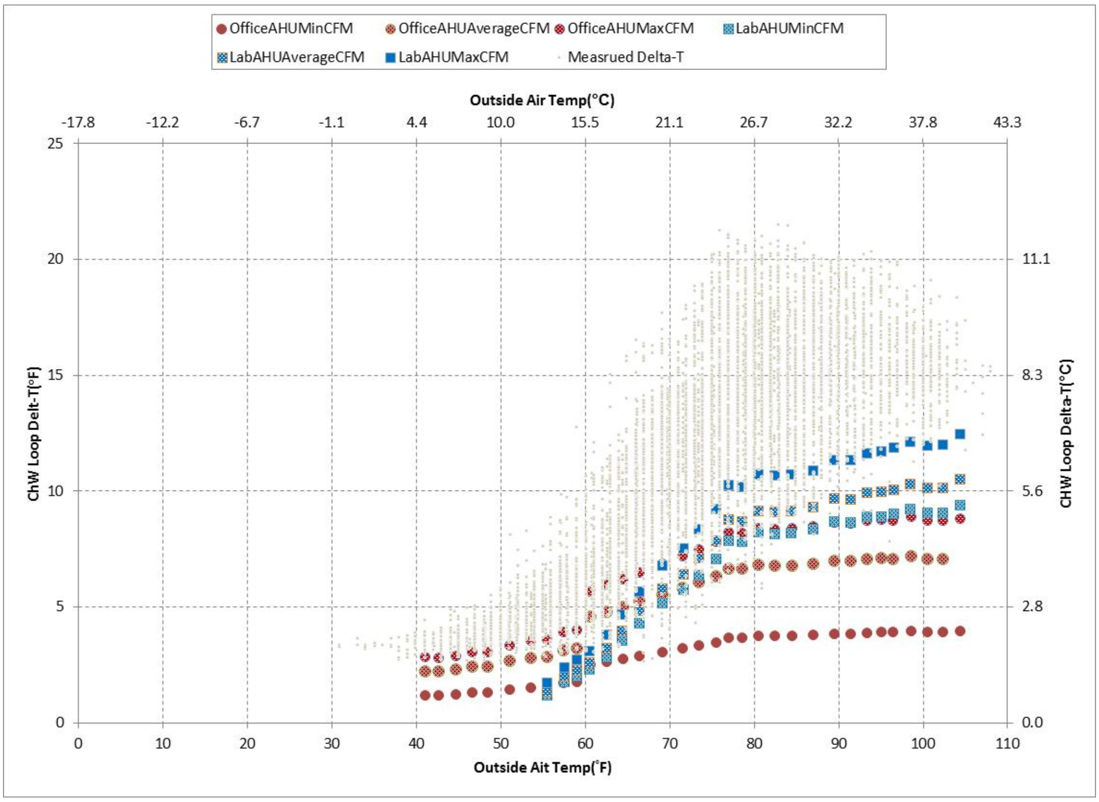

Fig. 8 Measured and simulated chilled water Delta-T

(Chilled water valve fully open, IP and SI units)

The most measured chilled water loop delta-T points are above the worst case points. It illustrates the chilled water loop is under control and the most chilled water values of AHUs are working properly. There is no significant leakage by valves or other mechanical issues on chilled water control valves.

Figure 8 presents the comparison result between the measured building chilled water loop delta-T and simulated cooling coil chilled water delta-T of office and Lab AHUs when the cold deck temperature is maintained at 50°F(10°C).

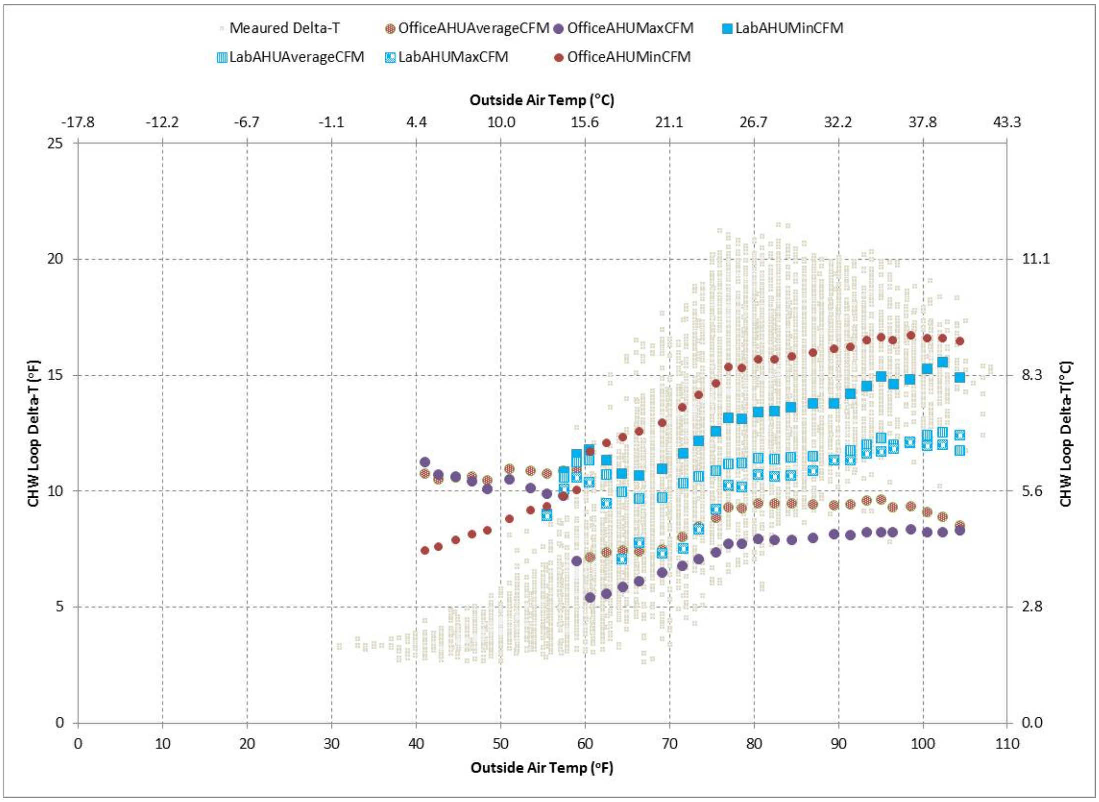

Fig. 9 Measured and simulated chilled water Delta-T

(Cold deck temp. 50°F, IP and SI units)

At this scenario, it is observed that considerable measured chilled water delta-T points are higher than the simulated chilled water delta-T when outside air temperature is higher than 70°F (21.1°C), while measured chilled water delta-T points are lower than simulated chilled water delta-T when outside air temperature is less than 60°F (15.6°C). The results of Figure 6 and Figure 8 indicate that the actual average cold deck air temperature set point is between 50 °F (10°C) and 55°F (12.8°C). The bigger overlap region between measurement and simulated points of figure 8, indicates cold deck air temperature is at cold side. Hence, the building chilled water loop delta-T can be increased by optimizing cold deck air temperature set point.

However, when outside air temperature is less than 55 °F (12.8°C), the measured chilled water delta-T is still significantly lower than simulated value even when the cold deck air temperature is 50°F(10°C). A further analysis is conducted to identify the possible reasons which cause a lower chilled water delta-T at lower outside air temperature condition. When outside air temperature is less than 55°F (12.8°C),, the office AHUs chilled water delta-T will dominate the whole building chilled water loop delta-T.As the lab AHUs are 100% OA, its cooling load will be very low when outside air temperature is cool. Hence, only the office type AHU is simulated to study the low chilled water delta-T issues when outside air temperature is cool. Figure 9 presents office AHU chilled water delta-T pattern versus outside air temperature with and without economizer.

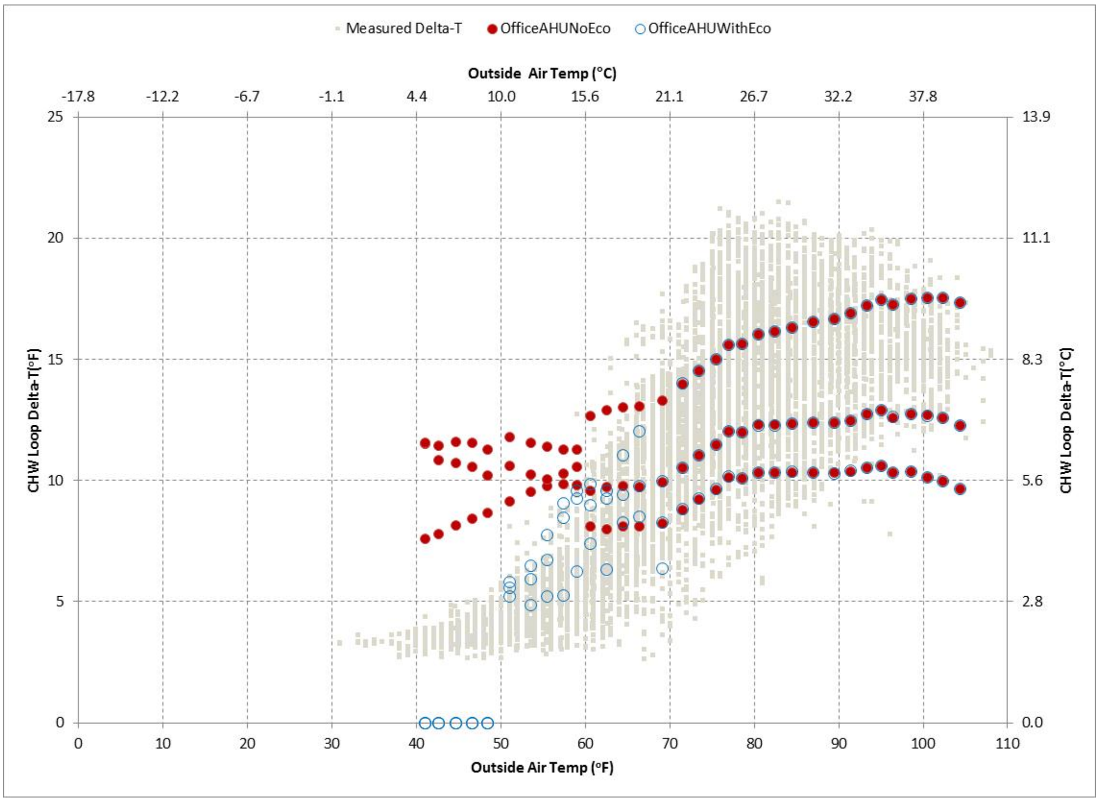

Fig. 10 Office AHU w/o Economizer cold deck air temperature at 50 °F

When an airside economizer is applied, the mixed temperature of outside air and return air or the coil entering temperature is lower than the case without an economizer, which results on a lower chilled water delta-T. When the outside air is cool enough, the mixing temperature drops below the coil discharging set point and there is little cooling on the coil. In real situation, when the coil control valve cannot precisely modulate the chilled water flow, the leaking chilled water will over-cool the air and lead to a lower-than-simulated delta-T. The measured chilled loop delta-T appears this phenomenon that may explain why the measured delta-T is lower than the simulated delta-T when outside air temperature is lower. The water loop pressure fluctuation may also push more water through the control valve and decrease the return water temperature further; however, this issue cannot be identified by simulation model, the field investigation is necessary to further identify these types of issues.

7. Summary

Improving chilled water delta-T in campus buildings that are connected to a central distribution loop will not only improve the power consumption of the building through reducing tertiary (building) pumping power but the impact on the central distribution system and chiller efficiencies may be even greater. However, almost every chilled water system encountered by the authors suffers from low delta-T, particularly at low cooling loads. It is a big challenge for CC engineers to quickly evaluate the chilled water loop delta-T performance and identify potential measures to improve chilled water loop delta-T. Understanding the coil delta-T performance characteristics is a critical step toward identifying possible measures to improve chilled water loop delta-T. This paper demonstrated a building chilled water Loop delta-T fault diagnosis procedure using a case study building as an example. In this procedure, the effectiveness-NTU coil model is employed to model the coil chilled water leaving temperature at given airside and waterside conditions. Both AHU L2 and AHU O models show a good agreement with measured data. The models CV(RMSE) are 6% and 2.9% for AHU O and AHU L2 respectively. According to the above analysis, the following conclusions for the case study building can be drawn:

- Based on simulation results, there is a good potential to improve the case study building's chilled water delta-T.
- The lower discharge air temperature set point is the main avoidable cause of low chilled water delta-T for the case study building. Optimizing cold deck air temperature set point could improve the chilled water loop delta-T.
- Economizer contributes to low chilled water delta-T during cool season.
- The chilled water laminar flow in the cooling coil is not a major cause for cooling coil lower delta-T
- Although the chilled water valves in general appear to be operating properly, the measured chilled water delta-T being lower than simulated delta-T in low load period indicates that a few of the chilled water valves may be leaking by or the coil control valves may not precisely modulate the chilled water flow. Whether leaking or poorly controlling air temperature, a lower-than-expected leaving air temperature will lead to a lower-than-simulated delta-T.

It should be noticed that when a very low cooling load is on the coil, the theoretical chilled water flow is very low and the simulation results may become unreliable. In addition, the possible cause of low chilled water delta-T varies from building to building. Hence, the intent of calibrated cooling model is not to identify all possible causes of low chilled water delta-T, but it can provide a benchmark for cooling coil delta-T performance. In turn, it can help CC engineers to better evaluate current chilled water delta-T performance and provide some clues to find possible solutions to improving chilled water loop delta-T.

NOMENCLATURE

C_{pm} = constant pressure specific heat of moist air
C_{pw} = constant pressure specific heat of liquid water
CHWRT= chilled water return temperature
CHWST= chilled water supply temperature
C_s = average slope of saturation air enthalpy versus temperature
C^* = ratio of air to water capacitance rate for dry analysis (maCpm/mwCpw)
C1, C2 = coefficient
h_a = enthalpy of moist air per mass of dry air

h_s = enthalpy of saturated air per mass of dry air
m_a = mass flow rate of dry air
m_w = mass flow rate of water
NTU = overall number of transfer units
Q = overall heat transfer rate
T_a = air temperature
T_{dp} = air dew point temperature
T_s = surface temperature
T_w =water temperature
UA =overall heat conductance
a =air humidity ratio
ω_s =humidity of saturated air

SUBSCRIPTS
a =air stream conditions
dry =dry surface
e =effective
i =inlet or inside conditions
o =outlet or outside conditions
s =surface conditions
w =water stream conditions
wet =wet surface
x =point on coil where condensation begins

References:

Taylor, S. T. (2002),. Degrading chilled water plant delta-T causes and mitigation, ASHRAE Transactions 108 (1): 641-653

Zhiqin Zhang, Hui Li, Jingjing Liu. (2012) Simulations of chilled water cooling coil delta-T characteristics , ASHRAE 2012 Annulus Conference, San Antonio, Texas

Wang, G., B. Zheng, et al. 2006, Impacts on building return water temperature in district cooling systems. Proceedings of 2006 International Solar Energy Conference, Denver, CO.

Fiorino, D. P. 1996, Twenty-five ways to raise your chilled-water temperature differential, ASHRAE Transactions 102(1)

Braun, J. E. 1989. Effectiveness models for cooling towers and cooling coil. ASHRAE Transactions 96(2): 164-167.

Threlkeld J. L. , Thermal Environment Engineering, Prebtice-Hall, New York, Second Edition, 1970

Dittus, E. J and L. M. K. Boelter 1930, publications on Engineering, Univ. California, Berkeley.

ASHRAE. 2004. ASHRAE Handbook: HVAC Systems and Equipment. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA.

ASHRAE. 2005. ASHRAE Handbook: Fundamentals. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA.

Avery, G. 1998. Controlling chillers in variable flow systems. ASHRAE Journal 40(2):42-45.

Khan, W. A., J. R. Culham, and M. M. Yovanovich. 2006. Convection heat transfer from tube banks in crossflow: analytical approach. International Journal of Heat and Mass Transfer 49(25-26): 4831-4838.

Webb, R. L. 1980. Air-side heat transfer in finned tube heat exchangers. Heat Transfer Engineering 1 (3): 33-49.