SIMULATION OF A LARGE CENTRAL COOLING AND HEATING PLANT USING TRNSYS AND CALIBRATION WITH MONITORED DATA

Danielle Monfet and Radu Zmeureanu
Centre for Building Studies
Department of Building, Civil and Environmental Engineering
Faculty of Engineering and Computer Science
Concordia University, Montréal, Québec, CANADA

ABSTRACT

This paper presents the development of a model, using TRNSYS, of a large central cooling and heating plant. The model includes the chillers, cooling towers and pumps of the chilled water loop, and the heat recovery system from the chillers. The model is calibrated with monitored data of June 23rd to June 29th, 2008, and then tested with data over the summer season, from June 23 to September 21, 2008. The simulated water temperature at several key locations in the system and the electric input to the equipment such as chillers, cooling towers and circulating pumps are compared with the measured values. The simulated Coefficient of Performance (COP) of the chillers and the overall central cooling plant are compared with the measured values.

INTRODUCTION

The calibration of computer models of the whole building energy performance, for instance using the utility bills or more detailed monitoring of major end-uses, or of the secondary HVAC systems was in the past the topic of several publications. Less frequent were, however, the publications about the central plants calibration with detailed measured data. Bourdouxhe and André (1997) simulated different control strategies for a centralized cooling plant using TRNSYS. The model was calibrated using measured data and used to evaluate the thermal storage discharge mode of the central plant. Troncoso (1997) evaluated the performance of a large central cooling plant using the electrical demand and consumption as the main input parameter. Monitored data were used to calibrate the model and evaluate the performance of the plant. In most cases, simplified components model were developed using manufacturer data and monitored data to simulate the performance of central plant with focus on the control strategies rather than on detailed modeling and calibration of the central plant (Ahn and Mitchell 1997, Ono et al. 2007, Wang, F. et al. 2007, Wang, S. et al. 2007).

Monfet et al. (2007) presented the calibration of a computer model, using the EnergyPlus program, of the air-side air-handling units installed in a large new university building, the Concordia Sciences Building (CSB). In this paper, the work is extended, using the TRNSYS program (2007), to the central plant that provides chilled water, heating water and steam to two large buildings, the CSB and the administration building (AD).

DESCRIPTION OF THE CENTRAL PLANT

Figure 1 presents a schematic of the central plant.

Designed operation for the chilled water loop

During the summer, two centrifugal chillers, CH-1 and CH-2, provide chilled water to the air handling units. The chillers are CenTraVac CVHF910 Trane model that use R-123 refrigerant, have a cooling capacity of 3165 kW (900 tons) each, and the rated power input is 549 kW with a coefficient of performance (COP) of 5.76 at design conditions. The design leaving chilled water temperature is 5.6°C (T_{CHWS}) and the return is 13.3ºC (T_{CHWR}). The chillers are water-cooled by two perpendicular flow cooling towers, CT-1 and CT-2, having a capacity of 4750 kW (1350 tons) each at design conditions. The cooling towers are Baltimor Aircoil model 3676A. At design conditions, the condenser water temperature enters the cooling tower at 36.3°C (T_{CNDS}) and leaves at 29.4ºC (T_{CNDR}). During the summer, only one of the two chillers can operate under heat recovery mode, i.e., the condenser water is first directed to a heat exchanger (HX-3) to pre-warm the heating water return, and is then sent to the cooling tower.

Designed operation for the heating water loop

During the summer, the heating water loop, which is used for re-heat purposes only, operates on 35ºC water supply (T_{HWS}) and 29.4ºC water return (T_{HWR}) temperatures. At most, two pumps are turned on simultaneously. In the summer, if chillers CH-1 or CH-2 are in operation, the SOFAME system (S-1) is turned off, and heat is recovered only from the cooling towers using the heat exchanger HX-3.

If the supply heating water is below 35ºC when leaving the heat exchanger HX-3 (T_8 in Figure 1), the steam heat exchanger (HX-2) provides additional heating to reach the setpoint temperature (T_6). Steam is provided by a high efficiency natural gas boiler having a capacity of 815 kW. Steam is also used to produce low-, mid- and high-temperature heating water for various purposes as well as being directly used to feed the humidifiers, if required.

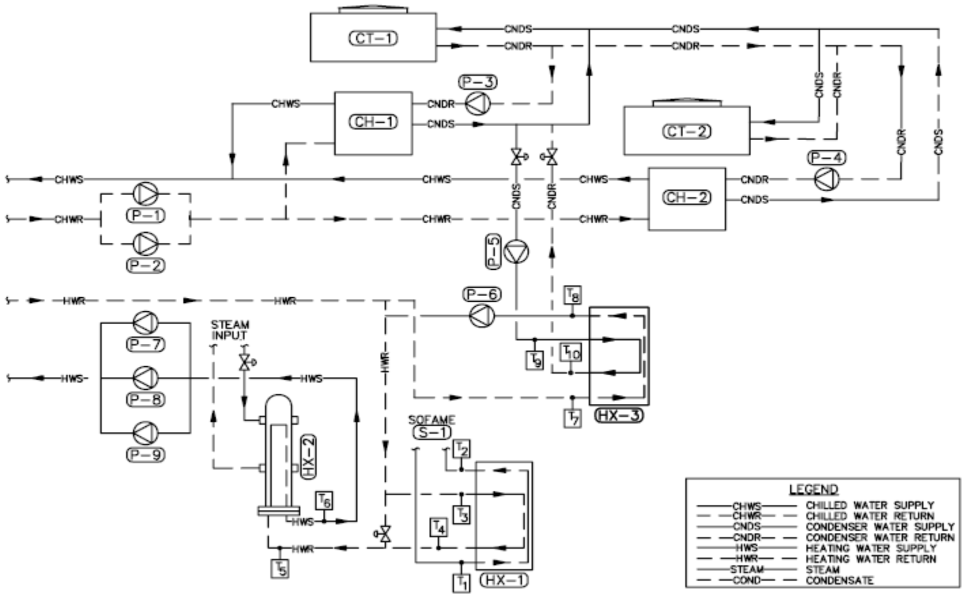

Figure 1 Central plant schematic

Sequence of control

Two smaller chillers, chillers CH-3 and CH-4, are installed in the CSB to provide chilled water to the building during the winter and part of the shoulder season. When the CH-3 and CH-4 cannot meet the CSB cooling demand, the first large chiller (CH-1 or CH-2) is turned on, and chillers CH-3 and CH-4 are turned off. If the first chiller operates at 85% of its capacity for more than 30 minutes and the chilled water supply temperature is above its setpoint, the second chiller is started. The corresponding chilled water and condenser water pumps are started simultaneously. The fans of the cooling towers are started when the condenser pumps are started, if required. The fans rotation speed is varied to maintain the condenser water temperature (T_{CNDR}) at its setpoint. The fans can however be turned off whenever the outdoor air temperatures (dry- and wet-bulb) allow for it.

MONITORED DATA AT THE CENTRAL PLANT

A total of fifty-eight points were monitored every fifteen minutes by the Monitoring and Data Acquisition System (MDAS), and the information was made available through the collaboration of the Physical Plant of Concordia University. The system uses Siemens Insight version 3.7. The monitored points include the supply and return water temperatures for the chilled, condenser and heating water loops, and the equipment operation (e.g. the chillers' partial load). Since the water flow rate of all constant flow pumps is not continuously monitored; the water flows of most constant speed pumps (P-1, P-2, P-5 and P-6) were measured using an ultrasonic water flow meter in collaboration with CanmetENERGY Varennes research center (Table 1). For the condenser pumps P-3 and P-4, the turbulence and/or the presence of air in the pipe did not allow a conclusive measurement. For this reason, the pressure difference at the pump was measured, and the water flow rate was evaluated using the manufacturer pump curve. The average water temperatures and standard deviations, monitored in the central plant between June 23 and September 21, 2008, and the average outdoor conditions are presented in Table 2.

Table 1
Measured water flow of constant water flow pumps

CIRCUIT		FLOW, L/s	POWER, kW
Chilled water	P-1 or P-2	86.75	100
Condenser water	P-3 or P-4 P-5 at HX-3	110.0 60.00	75 30
Heating water	P-6 at HX-3	107.25	55

Table 2
Average monitored data during the system operation, June 23-September 21, 2008

ITEM		AVERAGE
Outdoor dry-bulb temperature (T_{DB}), ˚C		21.3
Outdoor relative humidity (RH), %		45.7
CH-1	T_{CHWS}, ˚C	6.8±0.7
	T_{CHWR}, ˚C	10.6±1.6
	T_{CNDS}, ˚C	32.6±1.8
	T_{CNDR}, ˚C	28.4±0.4
CH-2	T_{CHWS}, ˚C	6.7±0.5
	T_{CHWR}, ˚C	11.0±1.3
	T_{CNDS}, ˚C	33.4±1.7
	T_{CNDR}, ˚C	28.5±0.4
CT-1	T_{CNDR}, ˚C	29.0±0.6
CT-2	T_{CNDR}, ˚C	29.0±0.5
HX-3	T_{HWR}, ˚C	31.4±1.6
	T_{HWS}, ˚C	32.3±1.7

The instantaneous power input to the chillers (CH) is not currently being monitored; however, the percent relative load amperage (%RLA) is continuously monitored with respect to the maximum amperage. The building operators manually recorded the chiller current (I_{CH}) and corresponding %RLA on a daily basis. Based on this information, a correlation was developed to estimate the intensity of electric current, in amps (Equation 1). A correlation was also developed for the chiller power factor (PF) based on manufacturer information (Equation 2).

$$I_{CH} = 5.884 \cdot (\%RLA) + 0.7978 \qquad (1)$$

$$PF = -0.0365 \cdot \left(\frac{Q_E}{Q_{E,design}} \right)^2 + 0.2483 \cdot \left(\frac{Q_E}{Q_{E,design}} \right) + 0.4915 \qquad (2)$$

where Q_E is the evaporator cooling load, calculated from the water flow rate and temperature difference of the chilled water (Equation 3) and $Q_{E,design}$ is the evaporator cooling load at design conditions, both in kW.

$$Q_E = 4.196 \cdot (\sum_{i=1}^{2} m_{P-i}) \cdot (T_{CHWR} - T_{CHWSPT}) \qquad (3)$$

where m_{P-i} is the measured water flow rate of pumps P-1 and/or P-2; 4.196 is the water specific heat in kJ/(kg·°C) at 9.45°C; T_{CHWR} is the monitored chilled water return temperature; and T_{CHWSPT} is the chilled water supply temperature setpoint (6.8˚C).

The instantaneous electric power for each chiller, in kW, is calculated by assuming that the voltage (V) is constant (Equation 4). Changes to the MDAS have been implemented during the winter season and the instantaneous power will be monitored for the 2009 cooling season. The measured power input will therefore be available, by September 2009, for comparison with the presented correlations (Equations 1 to 4).

$$E_{CH} = \sqrt{3} \cdot V \cdot I_{CH} \cdot PF \qquad (4)$$

The instantaneous electric power of each cooling tower (CT) is calculated based on the monitored variable frequency drive (VFD) level of the two 30 kW fans (Equation 5). The instantaneous electric power of each pump is calculated based on the ON/OFF status at each time step and known electric input (Equation 6).

$$E_{CT} = 2 \cdot \left[\frac{\%VFD}{100} \right]^3 \cdot E_{Fan} \qquad (5)$$

$$E_{p,constant} = E_p \qquad (6)$$

The total cooling electricity use is evaluated at each time step over the summer period by adding the electricity use for chillers, cooling towers and pumps (Table 3). On average, the cooling system operates with one chiller (CH-1 or CH-2), one cooling tower (CT-1 or CT-2), one chilled water pump (P-1 or P-2), one condenser water pump (P-3 or P-4), and the heat recovery pump (P-5). The water pumps operate at constant speed, and for a typical day the total instantaneous power input is 260 kW (Table 1). In a typical day, the chiller operates at 60% of its capacity (330 kW electric input) and the cooling tower %VFD is around 45% (6 kW electric input), for a total of 336 kW. The total seasonal cooling electricity use (Table 3) reflects the high electric demand by pumps and chillers.

Table 3
Cooling electricity use, June 23-September 21, 2008

ITEM	kWh
Chillers	467,220
Cooling towers	12,760
Pumps (P-1 to P-5)	360,940
Total	840,920

Coefficient of Performance (COP)

The COP is defined as the cooling load divided by the electric input (E_{input}) (Equation 7).

$$COP = \frac{Q_E}{E_{input}} \qquad (7)$$

The average COP values of the chillers, chillers plus cooling towers, and the overall cooling plant COP (chillers, cooling towers and pumps) are presented (Table 4). It is interesting to note that the changes made to the system (water temperatures and flows) have improved the performance of the chillers. The COP of the chiller at design conditions is 5.76, while the COP at the current operating conditions is 6.87 ±0.93.

Table 4
Average COP values, June 23-September 21, 2008

ITEM	COP
Chillers	6.87±0.93
Chillers + cooling towers	6.72±0.88
Cooling plant	3.86±0.81

DEVELOPMENT OF THE TRNSYS MODEL

The flow chart of the model developed in TRNSYS is presented in Figure 2. The interaction between the heating water loop and the chilled water loop is modeled via the heat recovery heat exchanger HX-3. This section presents the inputs to the model, and the TRNSYS types used to simulate the major equipment.

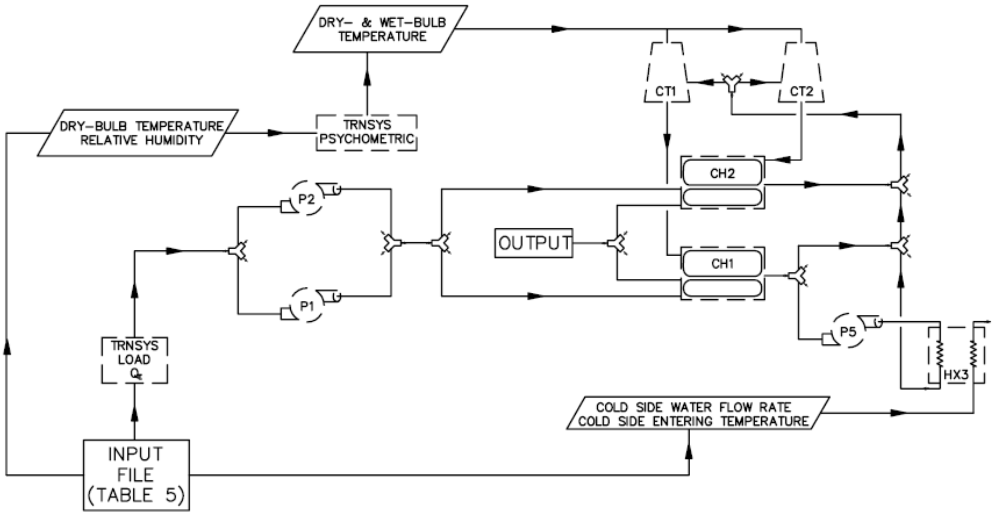

Figure 2 TRNSYS model flow chart

The simulation is run with a time interval of 15 minutes that is equal to the monitoring time step. At this stage of development, some selected measured data or data calculated from measurements are input to the model at each time step (Table 5). This approach ensures proper interaction of the heat recovery components by controlling the chilled and heating water loads entering the central plant. However, in the future, the inputs could be from a building air-side model.

Table 5
Model input data

ITEM	UNIT
Outdoor air dry-bulb temperature (T_{DB})	˚C
Outdoor relative humidity (RH)	%
Chilled water supply temperature (T_{CHWS}) setpoint	˚C
Chilled water return temperature (T_{CHWR})	˚C
Chilled water flow rate	kg/hr
Heating water flow rate in HX-3	kg/hr
Heating water return (HWR) temperature T_7	˚C

The selected TRNSYS types used in the model are presented in Table 6. For each major component, the main input and output variables are presented. Also, additional monitored data of the first week of summer (Monday June 23rd to Sunday June 29th 2008) or manufacturer data were used to determine additional parameters, if required.

Heat exchanger HX-3 (Type5b)

For the heat exchanger, the overall heat transfer coefficient is user-defined (Table 7). Based on the manufacturer information, the overall heat transfer coefficient associated with the heat exchanger surface is equal to 880 kW/K (3,175,430 kJ/hr·K) (Alfa Laval 2002). The cold-side outlet temperature T_8 is calculated by TRNSYS.

Table 6
TRNSYS types used in the central plant model

NAME	TRNSYS TYPE
Counter flow heat exchanger (HX-3)	Type5b
Data reader for generic data files (Input: Table 5)	Type9a
Pipe/duct (to and from CT-1 & CT-2)	Type31
Psychometrics: dry-bulb and relative humidity known	Type33e
Cooling tower: user-supplied performance coefficients (CT-1 & CT-2)	Type51b
Online plotter with file	Type65a
Fluid diverting valve	Type647
Mixing valve for fluids	Type649
Single speed pump (P-1 to P-5)	Type654
Water cooled chiller (CH-1 & CH-2)	Type666
Heating and cooling loads imposed on a flow stream	Type682
Equation	N/A

Table 7
Type5b: Input variables for the heat exchanger

ITEM	INPUT
Hot side inlet temperature T_9, ˚C	From simulation
Hot side water flow rate, kg/hr	Measured
Cold side inlet temperature (T_7), ˚C	From input file (Table 5)
Heating water flow rate, kg/hr	From input file (Table 5)
Overall heat transfer coefficient, kJ/hr·K	3,175,430 (from manufacturer)

Cooling towers (Type51b)

For the cooling towers, two coefficients are user-defined: the mass transfer constant (L/G) and the mass transfer exponent (n). The mass transfer constant is defined by the inlet water mass flow rate (kg/s) over the air mass flow rate (kg/s). Based on the measured water flow rate and the manufacturer air mass flow rate, the mass transfer constant is evaluated at 1.2. For the mass transfer exponent, ASHRAE (2004) recommends a value of -0.65 (Table 8).

Table 8
Type51b: Input variables for cooling towers

ITEM	INPUT
Water inlet temperature (T_{CNDS}), ˚C	From simulation
Inlet water flow rate, kg/hr	Measured
Dry-bulb temperature T_{DB}, ˚C	From input file (Table 5)
Wet-bulb temperature, ˚C	From type33e and input file (Table 5)
Sump make-up temperature, ˚C	25
Relative fan speed for cell-1	Calculated (Eqns 8 or 9)
Relative fan speed for cell-2	Calculated (Eqns 8 or 9)

The control of the cooling towers is performed by varying the fans speed to maintain a cooling tower outlet water temperature (T_{CNDR}). Since the cooling tower outlet temperature is relatively constant at 29°C; emphasis is put on properly simulating the cooling tower electricity demand. Therefore, correlations that estimate the variable frequency drive (VFD) level for CT-1 and CT-2 (Equations 8 and 9) were developed using measured data of the week of June 23-29 2008 (dry-bulb temperature and relative humidity), and using the calculated cooling tower load (Equation 10). For the training data set, the correlation gives a R^2 value of 0.957 for CT-1. The week of July 21st to 27th 2008 was used as a validation set. The calculated fans power input is in good agreement with measured data (Figure 3), with a R^2 value of 0.718 for CT-1. The correlations developed for the first week of summer (June 23-29, 2008) are not necessarily representative of the average summer operating conditions. A more representative summer week would provide better correlations between simulated and measured data for the entire summer season.

The water temperature leaving the cooling tower (T_{CNDR}) is controlled at about 29°C during the summer (Table 2). This value is used as the water temperature entering the condensers.

$$VFD_{CT-1} = -28.4302 + 0.0186 \cdot Q_{CT-1} + 1.1813 \cdot T_{DB} + 0.2061 \cdot RH \qquad (8)$$

$$VFD_{CT-2} = 21.9880 + 0.0197 \cdot Q_{CT-1} - 0.180 \cdot T_{DB} - 0.0376 \cdot RH \qquad (9)$$

where,

$$Q_{CT} = 4.17 \cdot \dot{m}_{CNDS} \cdot (T_{CNDS} - 29) \qquad (10)$$

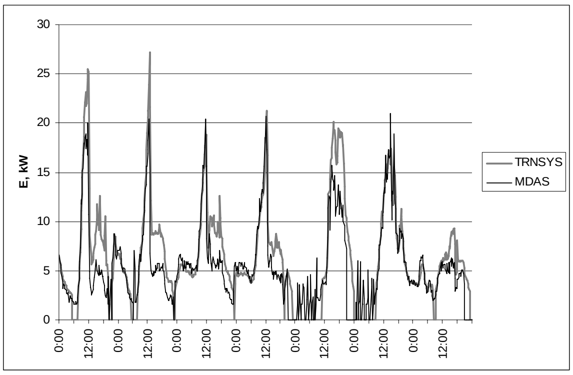

Figure 3 Simulated versus measured instantaneous electric power for CT-1, July 21-27 2008

Chillers (Type666)

Table 9 shows the input variables to the chillers. Two external files are also used to define: (1) the chiller performance data and (2) the electric input part-load ratio (PWR) in terms of cooling part-load ratio (PLR). For the chiller performance data file, the default file, which defines the capacity and COP ratios for a combination of leaving chilled water temperatures and entering condenser water temperatures, is modified using manufacturer data for leaving chilled water temperature between 6 and 8°C. Figures 4 and 5 present data for an entering condenser water temperature of 30°C.

Table 9
Type666: Input variables for chillers

ITEM	INPUT
Chilled water inlet temperature (T_{CHWR}), ˚C	Measured
Chilled water flow rate, kg/hr	Measured
Condenser entering water temperature (T_{CNDR}), ˚C	29.0, average monitored data
Cooling water flow rate, kg/hr	Measured
Set point temperature for chilled water supply (T_{CHWS}), ˚C	6.8, average monitored data
Control signal	ON/OFF, monitored

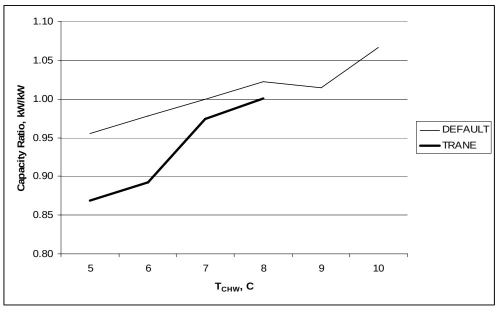

Figure 4 Chiller cooling capacity to cooling capacity at design conditions for a 30°C entering condenser water temperature

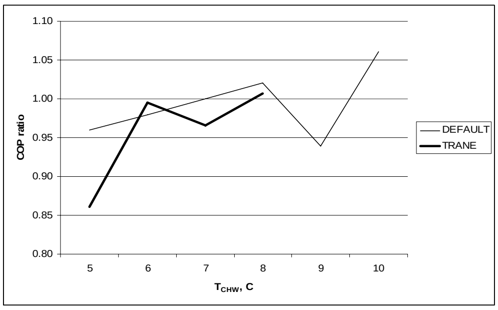

Figure 5 Chiller COP to COP at design conditions for a 30°C entering condenser water temperature

The second external file that defines the chiller power input ratio (PWR) in terms of the part-load cooling load ratio (PLR) at the evaporator is also modified based on manufacturer data (Table 10). The following output variables are calculated by TRNSYS: the supply chilled water temperature (T_{CHWS}), the condenser supply water temperature (T_{CNDS}), the electric input to the chiller, and the chiller COP. At this stage of development, the monitored chilled water pumps (P-1 and P-2) schedule is used to control the chillers operation (ON/OFF). When the first chilled water pump is turned on, the first chiller is started. Similarly, when the second chilled water pump is turn on, the second chiller is started. In the future, however, the operation of the chilled water loop will be controlled by the cooling demand.

Table 10
Electric PWR versus part-load cooling load PLR

PLR	PWR
0.1	0.1380
0.2	0.2047
0.3	0.2659
0.4	0.3258
0.5	0.3847
0.6	0.4737
0.7	0.5724
0.8	0.6882
0.9	0.8276
1.0	1.000

SIMULATION RESULTS

Kaplan and Canner (1992) recommended that the maximum allowable difference between predicted and monitored data be of 15-25% (monthly) and 25-35% (daily) for the simulation of HVAC systems. The annual simulated energy use should be within 10% of collected information, while a difference less than 25% is acceptable on a seasonal basis. For the coefficient of variance (CV), the value should be within ±30% when using hourly data, or 5% to 15% for monthly data (Reddy 2006).

The relative error (R.E.), the root mean square error (RMSE) and the coefficient of variance (CV) are used to evaluate the accuracy of the simulation results (Equations 11 to 13).

$$R.E. = \frac{\sum_{i}^{n} \frac{\left| y_{pred,i} - y_{data,i} \right|}{y_{data,i}}}{n} \cdot 100 \qquad (11)$$

$$RMSE = \sqrt{\frac{\sum_{i=1}^{n} (y_{pred,i} - y_{data,i})^2}{n}} \qquad (12)$$

$$CV = \frac{\sqrt{\frac{\sum_{i=1}^{n} (y_{pred,i} - y_{data,i})^2}{n}}}{\bar{y}_{data}} \cdot 100 \qquad (13)$$

where y_{pred} is the predicted variable (TRNSYS), y_{data} is the measured variable (MDAS), and $\bar{y}_{data}$ .is the average monitored data. Detailed data from the week of July 21st to July 27th 2008 (Monday to Sunday) are compared at key locations.

Comparison for the chiller CH-1

The following outputs of chiller CH-1 are compared with measured data: the supply chilled water temperature (T_{CHWS}) (Figure 6), the condenser supply water temperature (T_{CNDS}) (Figure 7), and the electric input to the chiller (Figure 8). The predictions made by TRNSYS compare well with monitored data.

Comparison for the heat exchanger HX-3

The predicted cold side leaving water temperature T_8 compares well with measured data of July 21-27 2008 (Figure 9). To complete the water temperature analysis at key locations, the temperatures are compared over the complete summer season. The average temperature difference and the RMSE presented in Table 11 show that the simulated chilled and heating water temperatures are in good agreement with the monitored data over the entire summer, from June 23 to September 21, 2008. The maximum RMSE is 1.0°C, and the average absolute difference does not exceed 0.8°C.

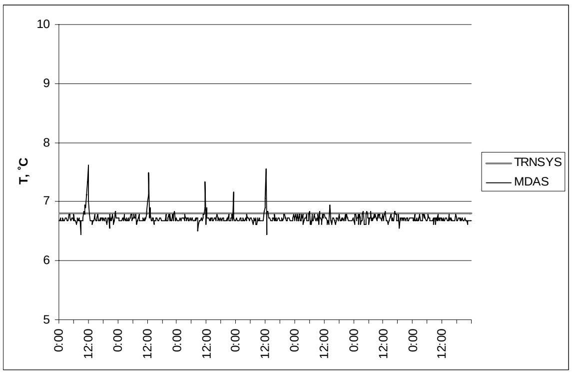

Figure 6 Simulated versus monitored chilled water supply temperature (T_{CHWS}) for CH-1, July 21-27 2008

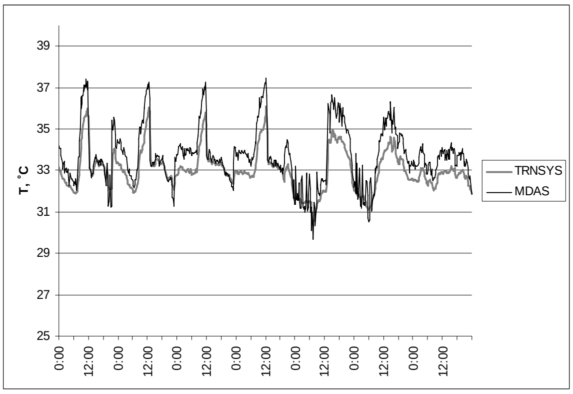

Figure 7 Simulated versus monitored condenser supply water temperature (T_{CNDS}) for CH-1, July 21-27 2008

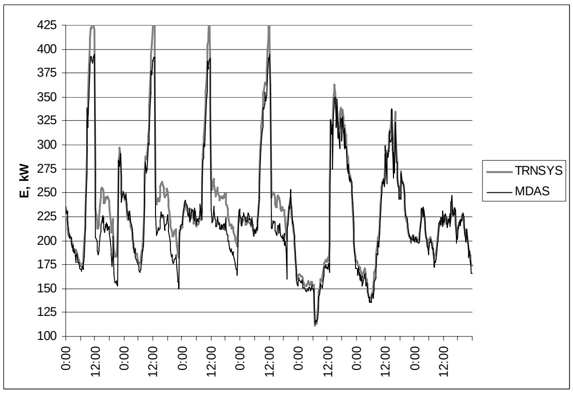

Figure 8 Simulated versus monitored electric input for CH-1, July 21-27 2008

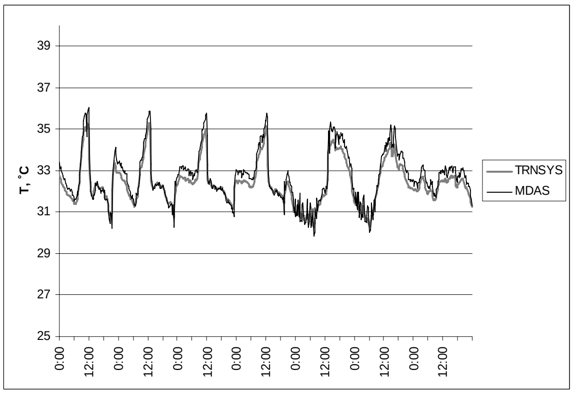

Figure 9 Simulated versus monitored cold-side leaving water temperature (T_8) of HX-3, July 21-27 2008

Table 11
Simulated versus monitored average water temperature during the system operation, June 23-September 21, 2008

ITEM	TRNSYS	MDAS	AVG. ABS. DIFF. °C	RMSE °C
$T_{CHWS,CH-1}$	6.8±0.1	6.8±0.7	0.13	0.42
$T_{CNDS,CH-1}$	32.7±1.3	32.6±1.8	0.80	1.00
$T_{CHWS,CH-2}$	6.8±0.0	6.7±0.5	0.18	0.52
$T_{CNDS,CH-2}$	33.2±0.7	33.4±1.7	0.38	0.80
$T_{HWS,HX-3}$ (T_8)	31.5±2.5	32.3±1.7	0.52	0.84

Comparison of the cooling energy use

The electricity use of the central plant for cooling purposes is evaluated and compared to the monitored data for the first week of the summer (Table 12) and for the entire summer (Table 13), where CH refers to chillers only, CT to cooling towers, and P to pumps. The accuracy for the simulation of energy consumption, in kWh, is within the recommended values, with a R.E. below 10%. For the instantaneous total electric input, the average absolute difference is 28.1 kW.

Table 12
Simulated versus measured cooling electricity use in kWh, June 23-June 29, 2008

ITEM	TRNSYS	MDAS	R.E., % (over kWh)	AVG. ABS. DIFF., kW
CH	43,800	41,110	6.5	18.8
CT	1,425	1,310	9.1	0.9
P-1 to P-5	29,405	29,300	0.4	0.6
Total	74,630	71,720	4.1	20.3

Table 13
Simulated versus measured cooling electricity use in kWh, June 23-September 21, 2008

ITEM	TRNSYS	MDAS	R.E., % (over kWh)	AVG. ABS. DIFF., kW
CH	497,285	467,220	6.4	20.0
CT	13,875	12,760	8.8	0.9
P-1 to P-5	350,185	360,940	3.0	10.2
Total	861,345	840,920	2.4	28.1

Comparison of the Coefficient of Performance

The Coefficient of Performance is calculated for three cases: 1) only for chillers, (2) for chillers plus cooling towers, and (3) for chillers, cooling towers and pumps (Table 14). The simulation results are in agreement with the monitored data. The maximum relative error (R.E.) is less than 5%, while the coefficient of variance (CV) is about 7.5%.

Table 14
Simulated versus measured average COP of cooling system, June 23-September 21, 2008

ITEM	TRNSYS	MDAS	R.E., %	CV, %
Chillers	6.54±0.64	6.87±0.93	4.8	7.3
CH + CT	6.40±0.60	6.72±0.88	4.7	7.4
Cooling plant	3.68±0.76	3.86±0.81	4.7	7.0

CONCLUSIONS

The paper presented the approach undertaken to develop and calibrate a TRNSYS model of a large central cooling plant. The analysis of the monitored data combined with the manufacturer's information was used to develop the TRNSYS model. The model was calibrated with monitored data of June 23rd to June 29th, 2008, and then tested with data over the summer season, from June 23rd to September 21st, 2008. In many building cooling and heating plants, the power meters for measuring the instantaneous electric input are not installed. This paper shows how the simulation results are validated by using correlation-based models developed from other information currently available. This is a practical problem that building auditors and energy simulators can face quite often. The comparison between water temperatures and instantaneous electricity demand at key locations ensure that the model developed with TRNSYS accurately mimic the operation of the central plant; not only at the central plant level, but also at the component level. The proposed approach was used to demonstrate the importance of component level calibration. This is especially important if the model is later on used to perform optimization at the equipment and central plant levels. Overall the calibration exercise showed good agreement between the simulated and monitored data. The simulated chilled and heating water temperatures, compared at key locations, were in good agreement with the monitored data with a CV value below 8%. For the cooling electricity used, the simulation results were also within the acceptable range recommended by Kaplan and Carner (1992). Future work will focus on improving the model for chillers and cooling towers, developing the model for the heating water loop, including circulating pumps, heat exchangers, and coupling with the chilled water loop.

ACKNOWLEDGEMENT

The authors acknowledge the financial support received from the "Fonds Québécois de la Recherche sur la Nature et les Technologies (FQRNT)", the Faculty of Engineering and Computer Science of Concordia University, through the SRT fund, and CanmetENERGY Varennes research center. The authors would also like to thank Stéphan Drolet and Yves Gilbert from the Physical Plant of Concordia University for their collaboration.

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