Optimal Chilled Water Delta-T

Technical Assistance to Flow Control Industries (FCI)

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FCI markets a pressure-independent flow control valve for application at chilled water coils in air handlers served by a chilled water loop. Proper control of chilled water to the air handlers affects the staging of upstream pumps and chillers. A common symptom of pump control sequence error is low chilled water temperature difference (*Delta-T*) at part load. In this scenario, the flow rate and head remain high even when the cooling load is a fraction of design load. Because flow rate and delta-T are inversely related, we can use the concepts of pump sequencing, multi-speed control, and variable-speed control more or less interchangeably.

The objective of this Technical Assistance project is a set of analysis tools to help design and operating engineers determine the best control sequence for pumps and fans and to estimate the operating cost penalty for improper chilled water pump sequencing manifest as low *Delta-T*. *Delta-T* is a shorthand term used in the HVAC industry for the chilled water temperature drop across the evaporator, a function of cooling load and chilled-water flow rate.

A *Chiller Plant Control Tool* is needed to generate the control sequences and setpoints or reset schedules that prevent excessive chiller compressor power on the one hand, and excessive fan and pump power on the other. Under a given set of conditions (outdoor temperature and return air or return chilled water temperature) a given cooling load will be met with minimum total power when the relations between chiller, pump and fan power are properly balanced.

In addition to determining the best control strategy for efficient operation of a HVAC plant, the designer should be able to estimate annual energy use and savings with respect to a faulty control sequence. An *Annual-Cost Estimating Tool* is needed to estimate building loads and chiller plant performance for a given control strategy.

It is desirable to structure these tools so that actual characteristics of existing equipment and observed building load information can be plugged in to ensure that the resulting analysis applies to the plant in question—i.e. has specific relevance to the building owner or bill-paying tenants, and is not just a hypothetical paper study.

In this report, preliminary designs of the chiller plant control tool and annual operating cost tool are presented. These preliminary designs illustrate the functions as well as the required input data. However, the component models used at this stage are simplified and the results presented for a simple test case are based on hypothetical plant parameters. Refinement of the models and realistic test cases will be incorporated in the next phase of technical assistance.

Chiller Plant Control Tool

A prototype tool that may be used to find optimal speeds of the compressor, chilled water pump and condenser fan has been developed. Although the use of a single compressor, single pump and single fan is the simplest of possible topologies, it does illustrate the sensitivity of plant performance to delta-T as pump speed is varied. This tool can be readily extended to model a wide variety of plants with multiple chillers and multiple fans and pumps on the condenser and evaporator sides of the compressors.

The plant parameters for an air-cooled chiller must be specified as follows.
Chiller performance maps:
Q_{cmpr} (kBtu/h) = $f_c(\rho_s, P_s, P_d, S_c)$ and
E_{cmpr} (kW) = $f_p(\rho_s, P_s, P_d, S_c)$ where
ρ_s = suction density,
P_s, P_d = suction and discharge pressures, and
S_c = compressor shaft speed.

Evaporator pump and heat exchanger parameters:
C_{eo} (kBtu/h/F) = evaporator chilled water design thermal capacitance rate,
E_{eo} (kW) = pump power at design thermal capacitance rate, and
UA_{eo} (kBtu/h/F) = evaporator heat exchanger conductance.

Condenser fan and heat exchanger parameters:
C_{co} (kBtu/h/F) = condenser fan design thermal capacitance rate,
E_{co} (kW) = fan power at design thermal capacitance rate, and
UA_{co} (kBtu/h/F) = condenser heat exchanger conductance.

In addition, the *conditions* and *cooling load* are specified:
T_{ODB} = outdoor temperature
T_r = chilled water return temperature
Q = total chilled water cooling load.

The combination of pump, fan and compressor speed that satisfies load with the least total plant power are solved for using the relationships developed in Appendix A. A typical solution is shown in Table 1.

Table 1. Motor speed solution for one set of boundary conditions (BC).

		Evap-side	Cond-side
Solution: T_e,T_c	F	35.6968741	115.919812
BC:T_r,T_{ODB}	F	70	90
BC: cooling load, Q	kBtu/h		80
Pump/fan parameters: E_{eo}, E_{co}	kW	2.69	0.81
HX Parameters: C_{eo}, C_{co}	kBtu/h/F	4.32	8.64
HX Parameters: UA_e, UA_c	kBtu/h/F	12	15
Q/dt (input to HX solver)	kBtu/h/F	2.33214898	3.63017512
HX solutions C_e, C_c	kBtu/h/F	2.3462468	3.69383818
Suction density, ρ	Lbm/ft^3		1.41203864
Suction P_e, Discharge P_c	psia	77.095644	260.452879
Refrig't vapor enthalpies h_e,h_c	Btu/lbm	20.2008627	44.3584305
Compressor speed S_c/S_o			1.070
Compressor capacity (Btuh), power (kW)		74790.7801	4.12934608
Pump, fan power E_e,E_c	kW	0.43094527	0.06329616
$J = E_e + E_c + E_{cmpr}$	kW		4.62358752
Delta-T	F	34.32710	22.57433
EER	Btu/Wh		17.3025815

Solutions for a range of loads and conditions are presented in Figure 1. The response surfaces represented in Figure 1 can be evaluated from tabular data (Appendix B) by linear interpolation. The following trends are noted.

Efficiency. With optimal control of fan, pump and compressor, the overall plant efficiency, EER=(cooling effect kBtuh)/(total kW), increases as load decreases. This is primarily the result of reduced flow losses (fan and pump power decrease faster than cooling load) and closer approach temperatures (compressor power decreases slightly faster than load).

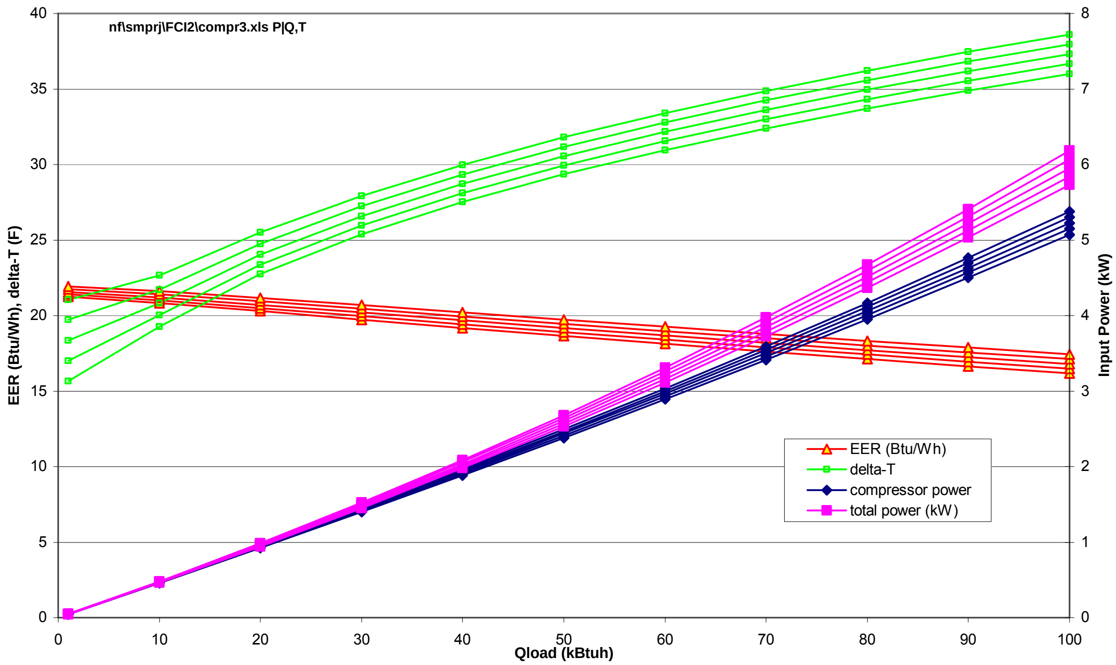

Figure 1. Solutions for range of cooling loads and T_{ODB}=60:10:100°F. Within each family of curves, power and delta-T increase while EER decreases with T_{ODB}.

Plant Electrical Load. Total plant power is almost linear with load because, under optimal control, it is dominated by compressor power--even at the lightest of partial cooling loads.

Delta-T. Chilled water temperature delta-T is relatively constant under optimal control of fan, pump and compressor. There is only a 17% reduction from design delta-T at the 50% part-load mark.

Annual Cost Tool

A bin method used to estimate annual operating cost of package equipment (www.pnl.gov/uac) has been adapted to estimate chiller plant operating costs.

To estimate seasonal performance we must specify, in addition to chiller plant performance, a climate and a load. We must also specify energy prices.

A spreadsheet version of the UAC calculator is shown in Table 2. The load model assumes a sensible cooling load directly proportional to outdoor temperature and a latent load directly proportional to the product of outdoor humidity (mass ratio of water vapor to dry air) and outside air flow rate. An ideal enthalpy control is assumed and a fixed minimum outside air flow rate (10% in this analysis) is specified by the user. The peak load (for sizing purposes) is assumed to occur at the ASHRAE 0.4% dry- and coincident wet-bulb temperatures with the minimum (40 scfm/Ton)[1] outside air-flow setting.

The cooling balance point must also be specified. Here balance point is defined as the thermostat setpoint minus the ratio of average solar and internal gains[2] to envelope UA. For example, with a 75°F setpoint, average gains of 150 kBtu/h and a UA of 60 kBtu/h/°F, the balance point is 75 – 150/60 = 50°F.

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[1] 40 scfm per Ton of cooling capacity corresponds to about 10% outside air, a typical minimum outside air fraction

[2] Average over occupied hours, including metabolic heat output of occupants.

Table 2. Chiller Plant Operation Cost Calculator.

CHILLER PLANT ANNUAL ENERGY, DEMAND, AND COST CALCULATION

Location: Atlanta, GA

ASHRAE 0.4% Design Conditions		
Dry Bulb:	93	°F
Wet Bulb:	75	°F

Design Load	115	kBtuh
Ventilation CFM		scfm
Balance Point	52	°F

Electric rate ($/kWh)	0.08
Summer demand ($/kW)	0
Winter demand ($/kW)	0
Ratchet ($/kW)	0

Weather Conditions						Cooling Load			Economizer		Constant-speed fans & pumps			Sequenced fans & pumps		
A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q
Outdoor Temperature 5°F Increments	Coincident Wet Bulb Temperature, °F	Outdoor Temperature Difference (A–Tbalance)	Seasonal Cooling Hours, TMY	Number of Months with this Peak hour, TMY		Sensible Heat Gain, kBtu/h	Ventilation Heat Gain, kBtu/h	Total Heat Gain, kBtu/h (E+F)	Economizer Cooling Capacity kBtu/h	Remaining Load After Economizer	Plant kW, (f(I,A,B))	Demand Charges	Energy Charges	Plant kW, (f(I,A,B))	Demand Charges	Energy Charges
30	27.27549	-22	276			-46.5										
35	31.64923	-17	424			-35.9										
40	35.31892	-12	473			-25.4										
45	40.87396	-7	629			-14.8	-11.4	-26.2	-113.7	0.0						
50	45.05447	-2	639			-4.2	-7.8	-12.1	-78.5	0.0						
55	49.50995	3	748			6.3	-3.7	2.6	-37.4	0.0	0.000	0.00	0.00	0.000	0.00	0.00
60	54.32564	8	828			16.9	1.0	18.0	0.0	18.0	3.562	0.00	235.96	0.846	0.00	56.04
65	58.80224	13	926			27.5	6.0	33.4	0.0	33.4	3.928	0.00	290.98	1.643	0.00	121.69
70	64.56173	18	1319			38.1	12.9	51.0	0.0	51.0	4.270	0.00	450.62	2.629	0.00	277.37
75	67.23051	23	983			48.6	16.5	65.2	0.0	65.2	4.522	0.00	355.63	3.512	0.00	276.21
80	69.00217	28	696			59.2	19.1	78.3	0.0	78.3	4.747	0.00	264.29	4.399	0.00	244.96
85	71.13729	33	449			69.8	22.2	91.9	0.0	91.9	5.407	0.00	194.22	5.407	0.00	194.22
90	73.31966	38	114			80.3	25.6	105.9	0.0	105.9	6.512	0.00	59.39	6.512	0.00	59.39
95	75.65207	43	16			90.9	29.3	120.2	0.0	120.2	7.671	0.00	9.82	7.671	0.00	9.82
100	0		0													
105																
110																
115																
									Total Annual Costs			$0.00	$1,624.95		$0.00	$1,183.66

Outdoor temperature bins represent the median temperature (i.e. 60°F is the range of temperatures between 57.5°F through 62.5°F)

The hourly coil load is evaluated by the load model for a given bin temperature The chiller plant model is then applied to determine average kW for the bin and this is multiplied by the number of annual bin hours to get bin kWh. The resulting kWh numbers are then summed over all bins to get the annual operating energy. The bin energy charge is the product of bin kWh and price. For time-of-use rates one must use a blended rate in which the summer on-peak rate is given extra weight appropriate for the seasonal and time-of-day distribution of chiller plant electrical use.

The monthly peak demands can be estimated from the bins corresponding to the 12 monthly peak cooling load conditions. The bin demand charge is the product of bin demand, its corresponding number of months, and the demand charge. The numbers of months may be decomposed into separate summer and winter columns to allow for different summer and winter demand rates.

Annual energy and demand charges are summed and appear at the foot of each column.

Conclusions and Recommendations

This report contains the preliminary draft versions of a chiller plant control sequence tool and an annual operating cost tool. Following a review of this report by FCI, further refinement of the component models used in the tools and more realistic test cases will be added in the next phase of the technical assistance project.

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Appendix A. Chiller Plant Optimizer.
Boundary conditions are total cooling load and evaporator- and condenser-side temperatures. The condenser-side temperature of interest is ambient dry bulb for air cooled chillers and package equipment or wet bulb temperature for chillers with cooling towers.

The evaporator side temperature of interest depends on the system boundary selected for the analysis. If latent load is satisfied the return-air temperature should be used and the only other evaporator-side boundary condition to satisfy is total cooling load. To satisfy the latent load it may be necessary to specify the chilled water supply temperature as a constraint.

To illustrate the chiller plant analysis we consider an air-cooled chiller with outdoor dry-bulb and returning chilled water temperatures specified. The total cooling load is also specified. The evaporator and condenser saturation temperatures are unknown but we can provide a feasible initial guess because the condensing temperature must be higher than ambient and the evaporating temperature must be below the chilled water return temperature. The boundary conditions and unknowns are thus defined as:

Q = total cooling load,
T_r = chilled water return temperature,
T_{ODB} = ambient dry bulb temperature
T_e = evaporator saturation temperature (initial guess of $T_e < T_r$),
T_c = condenser saturation temperature (initial guess of $T_c > T_{ODB}$).

(Note that, except when a water-side economizer is modeled, the initial guesses must also satisfy $T_e < T_c$ even if $T_r > T_{ODB}$ and must lie within the region modeled by the compressor maps)

The chilled-water thermal capacitance rate, C_e, is determined by solving:

$$Q = \varepsilon_e(C_e, UA_e)UA_e(T_e - T_r),$$

and the chilled water pump power is given by:

$$E_e = E_{eo}\left(\frac{C_e}{C_{eo}}\right)^{x_{Ee}} \quad \text{where}$$

E_{eo}, C_{eo} = pump power and thermal capacitance rate at reference conditions and
$x_{Ee} \approx 3$ = the pump and chilled-water loop subsystem's characteristic load-speed exponent.

The compressor capacity, power and mass flow rate at reference speed are:

$Q_o = f_Q(\rho, P_c/P_e)$ = chiller capacity at compressor reference speed,
$m_o = f_m(\rho, P_c/P_e)$ = refrigerant mass flow rate at reference speed, and
$E_o = f_E(\rho, m, P_c/P_e)$ = compressor power input at reference speed

where

ρ = suction vapor density,
P_c is the saturation pressure corresponding to T_c and
P_e is the saturation pressure corresponding to T_e.

The compressor speed required to satisfy the load is:

$$S_c = S_o Q / Q_o$$

where S_o is the reference speed. Compressor power is given by:

$$E_{cmpr} = S_c f_E(\rho, m, P_c/P_e) \text{ where } m = (S_c/S_o) f_m(\rho, P_c/P_e)$$

The condenser air flow rate, C_c, is determined by solving:

$$Q + E_{cmpr} = \varepsilon_c(C_c, UA_c)UA_c(T_{ODB} - T_c),$$

and the condenser fan power is given by:

$$E_c = E_{co}\left(\frac{C_c}{C_{co}}\right)^{X_{Ec}} \quad \text{where}$$

E_{co}, C_{co} = condenser fan power and thermal capacitance rate at reference conditions and
$x_{Ec} \approx 3$ = the fan and air-cooling loop subsystem's characteristic load-speed exponent.

There are two unknowns, chilled water delta-T and condenser fan delta-T. When these are specified all other intermediate variables (flow rates, and the speeds and electrical loads of fan, pump and compressor) can be evaluated. The solver, given a feasible initial guess, performs a search to find the values that minimize the total plant electrical load,

$$J = E_e + E_c + E_{cmpr}$$

Simplifying assumptions. To obtain a useful prototype of the *plant control tool* we have made several simplifying assumptions. These simplifications can be addressed with more realistic, complete, and rigorous modeling in the next generation tool:

-superheat is zero (entire cooling effect is delivered at the evaporator saturation temperature),
-the evaporator sensible heat ratio is 1,
-all heat rejection occurs at the condenser saturation temperature,
-no minimum flow requirements for chilled-water or condenser air,
-the chilled-water loop load curve (pressure drop vs. flow rate) is fixed,
-compressor flow and capacity both taken as proportional to shaft speed,
-chilled-water return (or return-air) temperature is fixed,
-plant consists of only one each: supply pump, compressor, and condenser fan.

Appendix B. Test Case Plant Performance Maps

The following plant performance maps were produced for a single load-side condition of T_r=70°F.

Aggregate kW for plant comprised of VSD compressor with VSD pump & VSD condenser fan

	Cooling Load (kBtu/h)										
T_{ODB}	1	10	20	30	40	50	60	70	80	90	100
60	0.046	0.463	0.945	1.449	1.979	2.534	3.116	3.725	4.364	5.033	5.733
70	0.046	0.467	0.958	1.467	2.010	2.570	3.164	3.786	4.439	5.123	5.841
80	0.046	0.472	0.965	1.485	2.032	2.607	3.212	3.848	4.515	5.216	5.952
90	0.047	0.476	0.976	1.503	2.059	2.644	3.261	3.910	4.593	5.310	6.065
100	0.047	0.480	0.986	1.520	2.091	2.681	3.310	3.973	4.671	5.407	6.182

Aggregate kW for plant comprised of VSD compressor with fixed-speed pump & condenser fan

	Cooling Load (kBtu/h)										
T_{ODB}	1	10	20	30	40	50	60	70	80	90	100
60	2.864	3.330	3.622	3.854	4.057	4.242	4.415	4.580	4.739	5.033	5.733
70	2.864	3.334	3.629	3.861	4.068	4.253	4.429	4.596	4.757	5.123	5.841
80	2.865	3.337	3.633	3.869	4.075	4.265	4.442	4.612	4.775	5.216	5.952
90	2.866	3.340	3.638	3.876	4.085	4.276	4.456	4.627	4.793	5.310	6.065
100	2.867	3.343	3.643	3.883	4.096	4.287	4.469	4.643	4.811	5.407	6.182