Preliminary Analysis of Pressure-Independent Flow Control for Cooling Coils: Heat Exchanger, Controller, and Resulting System Performance.

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Introduction.

The flow of chilled water to cooling coils is typically controlled to maintain a supply air temperature setpoint. The supply air setpoint is, in turn, controlled to satisfy the building cooling load such that the maximum flow rate available at each conditioned zone is sufficient to maintain the temperature setpoint of said zone.

Savings Mechanisms.

There are several mechanisms by which improved control of cooling coil flow rate can translate into energy savings

1. Smaller zone temperature fluctuations result in slightly higher zone setpoints for equivalent comfort. Higher setpoints translate directly to lower cooling loads.

2. Reduced temperature droop (tracking the supply air setpoint more closely), translates (by, in turn, reducing zone thermostat droop) to slightly higher average zone temperatures as well as slightly higher average chiller evaporator temperatures.

3. Improved HX effectiveness (Gordon, 2000) results in slightly higher average chiller evaporator temperature.

4. To the extent that chilled water flow rates and return temperature trajectories are less noisy, improved coil control enables better tracking of optimal chiller operating point. Note, however, that optimal chiller control (Braun, Henze) is not currently in common use.

5. Reduced reheat energy.

Note that, except in the case of systems that use a lot of reheat (e.g., to achieve tight humidity control), the savings mechanisms all involve second-order effects. In mechanisms 1 and 2 the cooling load has been reduced only slightly and in mechanisms 3-5 the basic cooling load has not changed at all. Instead we see that a small change in acceptable set point, a small change in droop, or a small shift in a given control point result in slightly reduced load or slightly improved plant or distribution efficiency.

Additional (and possibly more significant) savings may accrue over time in the form of reduced maintenance. The potential to significantly reduce average frequency and/or magnitude of valve/actuator position changes is obtained with pressure independent control.

Lastly, eliminating hunting in the supply air temperature control loop reduces the likelihood that an operator will try to make an inappropriate “quick fix.” Such actions are rampant and

known to lead to significant energy waste—i.e. more than any of the above-listed mechanisms—in many cases.

Note that any method that reduces the effects of disturbances on the supply air temperature-chilled water flow control loop will lead to some or all of the savings enumerated above. Such methods include:

1. Pressure independent flow-control valves (the subject of this discussion) at each coil;

2. Three way valve at each coil;

3. Variable-speed primary pumps with a conventional valve at each coil;

4. Variable speed primary pump(s) at the chiller(s) and a variable speed secondary pump at each coil.

Method 2 is undesirable because the magnitude of chiller delta-T is proportional to aggregate load and this means lower average evaporator temperatures. Method 4 is only cost effective for large systems. Methods 1 and 3 are probably best for most retrofits and for new installations when variable speed secondary pumping is not an option.

It is important, when variable speed primary pumping is used, to tune the flow control loop properly and make sure the pressure transducer is properly located and responsive. There should be no need for balancing valves in a properly designed distribution network. However, first-cost considerations often result in distribution piping that is smaller than optimal (life-cycle cost basis). Undersized piping can result in large pressure variations from one coil to another.

Verification by Field Measurement.
The savings obtained by the mechanisms described above can by estimated by simulation. However there are many assumptions that must be made to implement a simulation model. Assumptions about boundary conditions, such as the pressure fluctuations presented to a given coil in the chilled water distribution network, are very difficult to justify without actual measurements in real systems. Measuring the savings is equally difficult but, if properly planned and executed, avoids many of the credibility problems inherent in assessment by simulation. With accurate field data for typical, and perhaps a few wide-ranging, sets of operating conditions, one can more credibly apply simulation to generalize the results (see Appendix A and B). Here we outline some measurements needed to document the savings mechanisms.

1. Smaller zone temperature fluctuations result in slightly higher zone setpoints for equivalent comfort. The time history of zone temperatures should be measured at one minute or better intervals (average, max and min over each interval) with and without the supply-air temperature control improvements (alternate weeks for 4-50 weeks). Concurrent measurements of outdoor and supply air temperature and RH, and of chilled water temperatures should also be recorded if possible. Zones should be selected to have cooling loads that are skin and internal

gain dominated, not solar dominated. Core zones are preferred. If perimeter zones or monitored the reheat and/or radiator inputs will have to be measured or the monitoring will have to be accomplished when heat is off for the summer.

2. Reduction in supply air temperature droop (tracks set point more closely), translates (via thermostat droop) to slightly higher zone temperatures. Field measurements to verify this mechanism are the same as for mechanism 1.

3. Improved HX effectiveness (exergy) results in higher mean chiller evaporator temperature. Measure entering and leaving coil air and water temperatures; also measure entering air humidity and condensate flow rate. In addition to point temperature measurements, thermopiles should be used to accurately measure the air and water side temperature *differences*. The sensing junctions should be distributed uniformly across the entering and leaving air flows. On the water side, the probes should be uniformly distributed across the leaving water or positioned far enough from the coil exit to ensure a uniform mixed water temperature.

4. Improved tracking of optimal chiller operating point. This effect will only be observed in buildings that have advanced chiller, chilled and condenser water pump, and cooling tower and supply air fan speed controls. The first step is to establish baseline performance of the chilled water plant (kw/Ton as a function of conditions where kW includes chiller, pump, and fan power) and ascertain that the existing plant control is near optimal. The accuracy of controller inputs (supply air setpoint or total building cooling load, outdoor conditions, chiller, pump, and fan power) and outputs (chiller, pump and fan speeds) must be verified. In addition, measure chilled water supply and return temperature and flow rate and supply air temperature at each coil.

5. Reduced reheat energy. Measure cooling and reheat coil inlet and discharge conditions: temperatures and flow rates on the water side, temperatures and RH on the air side. Systems with full air flow through both coils (cooling and heating) are desirable so that only one airflow measurement is needed. The key variable to be measured is static pressure at the throat of the fan inlet with respect to fan inlet plenum. For data reduction formulas, see Appendix C. The time history of zone temperatures should be measured at one minute or better intervals (average, max and min over each interval) with and without the supply-air temperature control improvements (alternate weeks for 4-50 weeks).

Analysis of Field Measurements and Model Verification.

To obtain a credible field demonstration of control-loop performance it is essential to have a model of the coil and control elements (sensor, actuator, compensator) that can be validated by the field measurements.

A static coil model is described in Appendix A. A preliminary analysis of the effect of chilled water flow rate on heat exchanger effectiveness is also given.

A transient coil model is described in Appendix B. The open-loop response to a step change in chilled water supply-return pressure is shown. The model is extended to include the chilled

water flow control elements. Performance of controls in the face of pressure disturbances can be studied using the resulting closed-loop model.

Next Steps.
The energy savings that can result by improved control of water flow rate to cooling coils are relatively small and application specific. For example, the magnitude of pressure fluctuations varies considerably from one building to another. To obtain a credible assessment of savings, a combination of analysis and measurement is needed. Buildings with significant reheat have the greatest potential for savings. One or more test sites with reheat and known pressure fluctuation problems in the chilled water loop should therefore be sought. For a given instrumented site a dynamic model should be prepared and validated against measured data. The resulting model can then be used with reasonable confidence to estimate savings in other climates, buildings, and system configurations. Repeating this process at several sites will improve our confidence in model validity as well as providing a better picture of the typical boundary conditions, particularly the pressure disturbances, encountered.

PNNL can provide part or all of the foregoing monitoring and analysis as a follow-on contract. We would expect the client to recruit sites and assist in assessing suitability of candidate sites. Latent load needs to be addressed. Client will research the comfort aspect and obtain typical pressure disturbance data; PNNL will model the chiller plant, coils, valves, savings mechanisms 1-4.

Other related research questions:
1. Chiller plant optimization based on NG model
2. Energy cost of poor coil control performance in reheat systems

Appendix A: Heat Exchanger Effectiveness

Pressure-independent control has an impact on heat exchanger effectiveness with changes in flow rates of the two fluids. The details of this theory are given below.

For a cross-flow heat exchanger configuration such as this one with both fluids unmixed, the effectiveness (E) is given approximately by the following relation:

$$E = 1 - \exp\left[1/C * N^{0.22} * \{\exp(-CN^{0.78}) - 1\}\right]$$

where

C = Cmin / Cmax, the ratio of the minimum heat capacitance rate to the maximum heat capacitance rate of the two fluid streams, and

N = NTU, the number of (heat) transfer units, given by UA/Cmin

For a given number of transfer units (NTU) for a coil, as the ratio of heat capacitance rates, C, decreases, the effectiveness, E, increases. If we assume constant supply air and supply water temperatures and a constant air flow rate, the leaving air temperature decreases slightly (more cooling achieved) as the water flow rate increases. This is a result of a small increase in effectiveness of heat exchange and a higher magnitude of heat transferred. Table 1 ("Constant Supply Temp") shows the impact on the effectiveness and hence the magnitude of the heat exchange (Q = E * Cmin * ΔTmax) due to specified changes in C for a given value of N. For the same sensible cooling load, higher effectiveness translates into closer approach temperatures. The chiller can therefore operate at a higher evaporator temperature resulting in a higher COP. The effect on heat exchanger performance over a typical cooling season is analyzed in Table A-1, below.

Table A-1. Seasonal analysis of heat exchanger effectiveness.

	Hours	Hours										
	25	1	2	3	4	5	4	3	2	1	25	
Supply Water Temperature	45.0	45.0	45.0	45.0	45.0	45.0	45.0	45.0	45.0	45.0		
Leaving Water Temperature	61.5	57.5	58.3	59.2	60.3	61.5	62.9	64.5	66.4	68.6		
Supply Air Temperature	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0	80.0		
Leaving Air Temperature	51.9	50.2	50.5	50.9	51.3	51.9	52.6	53.4	54.5	55.8		
DT max	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0	35.0		
DT Water	16.5	12.5	13.3	14.2	15.3	16.5	17.9	19.5	21.4	23.6		
Water Flow Gpm [1]	113	158.2	146.9	135.6	124.3	113	101.7	90.4	79.1	67.8	113	
Air Flow Cfm	30000	30000	30000	30000	30000	30000	30000	30000	30000	30000		
		140%	130%	120%	110%	100%	90%	80%	70%	60%		
C_water (Btu/h-F)	56274	78784	73156	67529	61901	56274	50647	45019	39392	33764	56274	
C_air (Btu/h-F)	33000	33000	33000	33000	33000	33000	33000	33000	33000	33000		
Coil NTU	3	3	3	3	3	3	3	3	3	3		
Cmin	33000	33000	33000	33000	33000	33000	33000	33000	33000	33000	33000	
Cmax	56274	78784	73156	67529	61901	56274	50647	45019	39392	33764	56274	
C	0.5864	0.4189	0.4511	0.4887	0.5331	0.5864	0.6516	0.7330	0.8377	0.9774	0.612392	
Effectiveness	0.8033	0.8515	0.8424	0.8317	0.8188	0.8033	0.7842	0.7603	0.7299	0.6904	0.795628	
Heat Transferred Btu/h	927802	983429	972951	960566	945754	927802	905723	878120	842995	797460	912756	
Total Heat * hours	23195061	983429	2E+06	3E+06	4E+06	5E+06	4E+06	3E+06	2E+06	797460	22973759	99.0%
DT Water	16.5	12.483	26.599	42.674	61.114	82.436	71.533	58.516	42.801	23.618	16.9	1.023
		12.483	13.3	14.225	15.278	16.487	17.883	19.505	21.4	23.618	16.87093	

[1] 74 is 50% flow

Appendix B: Transient Response to Pressure Disturbances

Here we formulate a simple multi-node transient model of a crossflow heat exchanger with mixed air-side and unmixed water-side flow configuration. We assume constant air-side flow rate and inlet temperature. We divide the heat exchanger into n equal segments along the water-side flow path. Each water-side node is fully mixed at each time step[1]. The hold-up mass is M_w and the heat exchanger thermal capacitance (empty) is M_cC_e,. The energy balance on a coil segment is:

$$\Delta Q_c = Q_{aconvect} + Q_{wconvect}$$

the temperature of the ith coil segment is thus governed by:

$$\frac{dT_{c,i}}{dt} = \frac{A_c h_c (T_a - T_{c,i}) + A_w h_w (T_{w,i} - T_{c,i})}{M_c c_c}$$

The energy balance on a water segment is:

$$\Delta Q_w = Q_{wconvect} + Q_{advect}$$

the temperature of the ith water segment is thus governed by:

$$\frac{dT_{w,i}}{dt} = \frac{\dot{m}_w c_w (T_{w,i-1} - T_{w,i}) + A_w h_w (T_{c,i} - T_{w,i})}{M_w c_w}$$

where

m_w = water mass flow rate
T_c = coil segment temperature state vector
T_w = water segment temperature state vector (T0 = inlet temperature)
M_cC_c = coil thermal capacitance
M_wC_w = thermal capacitance of resident water
A_ah_a = air-side conductance
A_wh_w = water-side conductance
n = number of segments

The commanded and actual valve positions, V_{cmd} and V_{act}, are governed by:

$$V_{cmd} = \mathrm{PID}(T_{sa}-T_{saSet})$$

$$\Delta V_{act} = \mathrm{sign}(V_{cmd} - V_{act})*\min(\Delta t/t_{stroke}, \mathrm{abs}(V_{cmd} - V_{act}))$$

where

$T_{sa} = T_{ma} + (Q_{aconvect} - Q_{latent})/(m_a c_a)$
T_{ma} = entering (mixed air) temperature,
$Q_{aconvect}$ = sum of air to coil node heat rates,
Q_{latent} = latent heat coil load (here assumed zero), and
PID(ΔT) = compensator output given history of temperature deviations, ΔT,
t_{stroke} = time required for full valve stroke (fully open to fully closed).

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[1]A plug flow model will return a more realistic simulation of leaving air spatial temperature variation and leaving water time history. However the mixed-node model presented here is usually considered adequate for transient response of the leaving bulk air temperature. Plug flow models require careful coordination of node and timestep size, i.e., variable node size and/or variable timestep size.

The desired chilled water flow rate is not generally achieved because of valve nonlinearity and pressure disturbances. The actual chilled water flow rate is given by:

$$m_w = f(Valve, \Delta P)$$

where

ΔP = chilled water supply-return pressure difference at the coil ports.

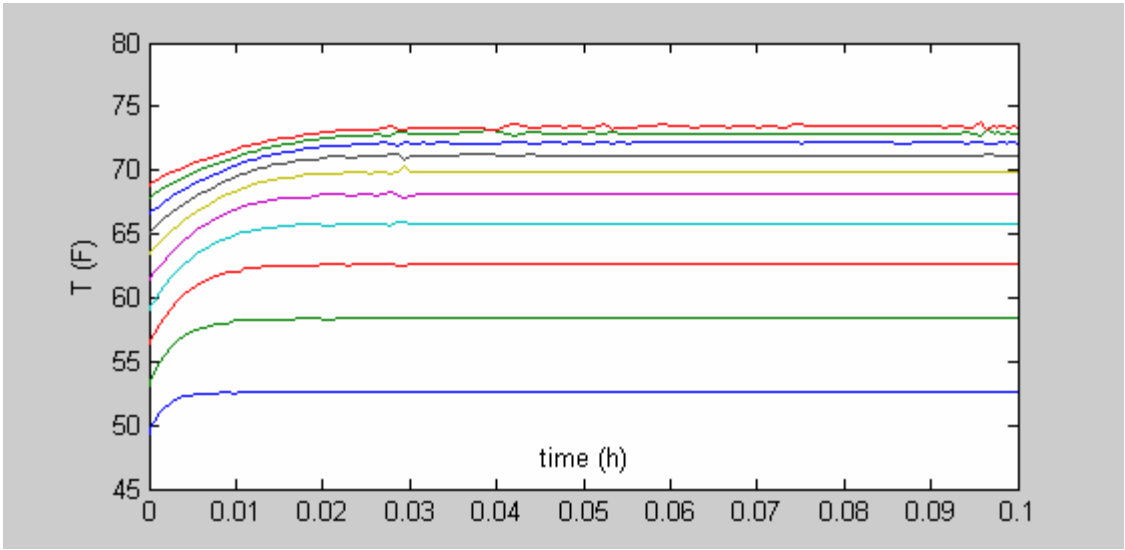

Figure B-1. Response of water node temperatures to a step change (drop) in chilled water mass flow rate. The response to a change in static pressure difference, ΔP, that is not compensated for (i.e. open-loop response) by a change in valve position will have the same shape.

Figure B-2. Closed-loop response of water node temperatures to a step change (drop) in chilled water static pressure difference.

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