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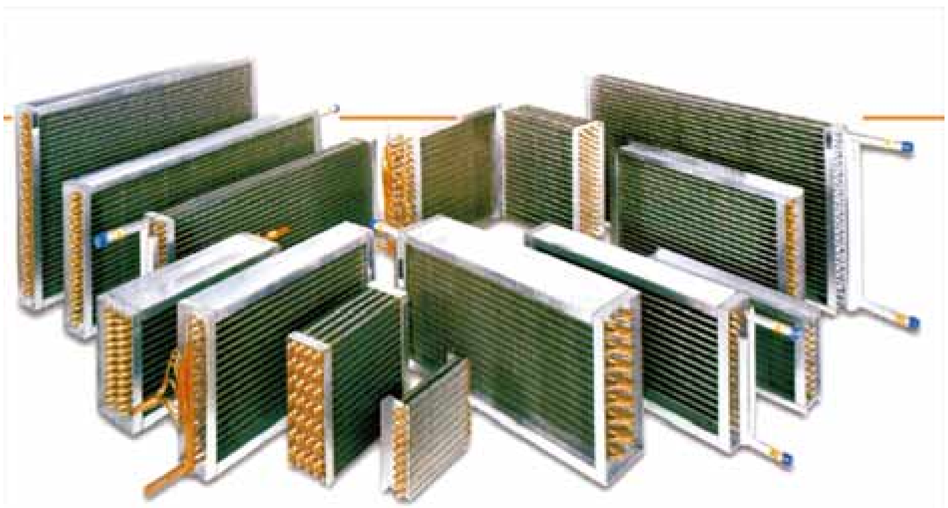

High-Delta T Chilled Water Systems

Conventional chilled water systems work on a Δ T of about 5°C. This is raised to 9-10°C in high-Δ T systems. The big advantage is, of course, the reduction in water flow rate/ton, which can only be achieved by careful selection of cooling coils and control valves. How to go about addressing these issues is covered in this article.

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What is a High-ΔT System ?

In conventional air conditioning systems, the difference between entering and leaving temperatures across the chillers (ΔT) is about 5°C. It will, however be higher – at 9 to 10 °C – in the case of high-ΔT chilled water systems. This high ΔT needs to be achieved not only at chillers but also on low side terminals like air handling units and fan coil units.

For the normal ΔT of 5°C (9°F), the flow rate will be 0.16 l/s (2.5 gpm/TR). In a high-ΔT system, on the other hand, the flow rate comes down to 0.09 l/s (1.5 gpm/ TR). The reduction in flow will therefore be in the ratio of 5/9 i.e., about 0.55.

High-ΔT Benefits - its Future

The most important benefits are, of course, reduction in sizes of pipes, valves, fittings, control valves, balancing valves and pumps. Obviously, the insulation cost will also be lower; likewise, connected pumps power requirements will also be lower. It follows therefore that one of the main themes in today's air conditioning systems viz., conservation of energy, is well addressed in this system. It is needless to say that first cost of the entire piping system will also be lower in this system.

The saving in connected power due to smaller pumps employed will be greater than any increase in power requirements of the chillers due to the larger range through which the chillers will be required to work.

These advantages become increasingly significant as the plant capacities go up. For large plants, therefore high-ΔT systems are the first choice. This is typified by the District Cooling Plants. Capacities of such plants range from about 10,000 TR to over 200,000 TR. Using normal ΔT systems for plants of such size is unthinkable.

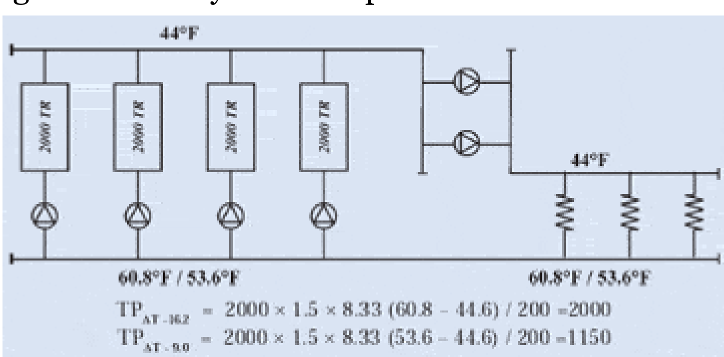

Figure 1 : Chillers cannot be loaded to their full capacity if ΔT is low.

Low-ΔT Syndrome

The high-ΔT systems are not new, though they have come into vogue more recently than primary and secondary pumping for chilled water systems. High-ΔT systems are now accepted all over the world; nevertheless, there are, in fact, concerns about the difficulties /

inability to achieve the high design ΔTs, so much so that the term "Low ΔT Syndrome" has gained widespread currency.

It is well to recall in this context, the history of primary - secondary systems of chilled water piping. The concept when it was introduced was readily accepted, – because of its obvious attractions. There were, many aspects of its design and performance that caused concerns; nevertheless, they did not deter the industry from accepting the system – the difficulties were merely addressed and solutions evolved – as indeed is being done even at the present time. Likewise, not withstanding the "Low ΔT Syndrome", high-ΔT systems have come to stay. And we need only to find solutions to tackle it.

What then are the several symptoms that constitute the "Low ΔT Syndrome"?

Figure 1 shows four chillers each of 2000 TR capacity, so that the total plant capacity is 8000 TR at full load. A plant of this type may have to operate at loads as low as say 1500 TR (about 18% of full load). One would expect that it will suffice to run one chiller at partial load, to meet this load of 1500 TR. But, then such expectations are often belied.

Ideally speaking (in a high-ΔT system), the temperature of return water from the load should be around 16°C (61°F) for a supply water temperature of 7°C (44.6°F). The chiller in turn, will therefore cool from 16°C to 7°C. In practice, the return water temperature could be much lower – say about 12°C. It will deliver only 12–7 / 16–7 = 5/9 of 2000 TR i.e., 1100 TR. The chiller will continue to cool water to 7°C, but no lower. Its capacity will therefore be reduced to 1100 / 2000 × 100 i.e., 55%. Accordingly, two chillers will have to be on line (each delivering 750 TR approx) in order to deliver a capacity of 1500 TR. This is the situation at part load.

If the system had been designed for high-ΔT and it operates at lower ΔTs, the entire piping system including pumps, pipes, coils etc., will turn out to be undersized to handle the higher flow rates that the lower ΔTs will call for to meet the full load requirements. In effect, this is equivalent to under sizing of the plant.

There is another important effect. If, as the load on a individual terminal falls and the flow is not reduced appropriately, the terminal will be over cooling the area it is serving. At the same time, another terminal which is fully loaded, may not get adequate water for effective cooling. Moreover, when the flow is not throttled appropriately, there will be no increase in the differential pressure across supply and return mains. Accordingly, the pump speed is not reduced and therefore, neither is the pump flow reduced. Thus the opportunity for energy saving is lost – which infact, is the primary reason for going for variable flow systems.

	Electrical Distribution	Hydronic Cooling
Symptoms	• High current demand • High voltage losses in conductors and transformers	• High flow demand • High pressure in pipes and heat exchangers
Problems	• Low power factor	• Low ΔT
Root Cause	• Large, under-loaded induction motors	• Improperly designed, operated and/or maintained cooling coils
Solution	• Power factor correction at induction motors	• ΔT correction at cooling coils
Benefits	• Increased power distribution and lower current demand and voltage losses	• Increased cooling distribution and lower flow demand and lower pressure losses

Low-ΔT Syndrome Compared to Low Power Factor Scenario

If then the high-ΔT systems pose such serious problems how can they be tackled? It is perhaps easiest to start with an electrical analogy to gain a better understanding of the problem. That is because the “Low- ΔT Syndrome” is the equivalent of low power factor in an electrical distribution system. Please see the comparisons on the previous page.

Cooling Coil Design and Selection

Focusing again on the hydronic systems, it is obvious that the chilled water flow rate in a variable flow system should decrease as the load decreases – for otherwise, there would be nothing to commend it. This decrease does indeed occur, but how much should be the decrease?

First, we shall note that in a (cross flow) cooling coil, the standard conditions are 7°C entering chilled water temperature, 27°C mixed air entering temperature, 12°C leaving air temperature and a water flow rate of 0.16 L/s (2.5 gpm/TR). One would assume – simplistically – that when the load falls to say 50%, so would the flow. The performance of such a coil would be "linear". Unfortunately, cross flow coils do not have such characteristics, but instead, their performance is “non linear”; thus, at 50% flow, the coil capacity is still 80% (and not 50%). Further, the flow required to produce 50% capacity is less than 30%). This can be seen from **Figure 2** – which is taken from ASHRAE Hand Book Application Volume.

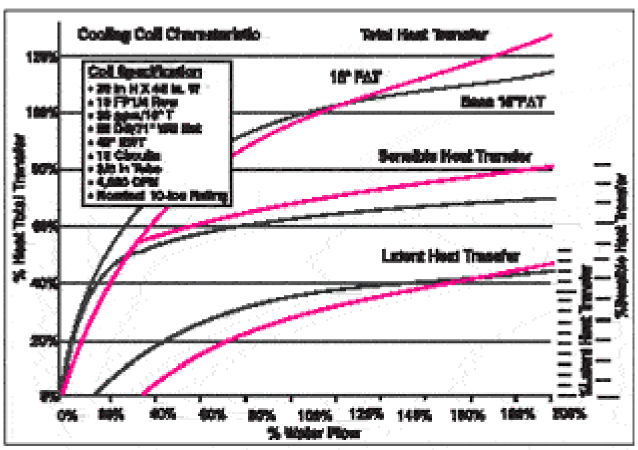

Figure 2 : Characteristics of cooling coil.

Saying that the coil capacity at 50% flow is 80% is the same thing as saying that the ΔT at 50% capacity should be 80 / 50 = 1.6 times the ΔT at 100% capacity. Thus if ΔT at full load is 9°C, it should be 1.6×9 = 14.4°C (26°F approx) at 50% capacity. (Incidentally this statement holds for the normal 5°C ΔT systems also; in this case, the ΔT at 50% load should be 1.6 × 5 = 8°C).

Non-linear Characteristics of Cooling Coil

Why is the coil characteristic non-linear ?

We all know that a project engineer is invariably required to check a cooling coil selection, when the manufacturer puts forward his submittal. It is usual for him to fall back upon the following three equations:

qa = qw = q = coil capacity .... (1)

qa = Qw × Δh × C1 .... (2)

qw = Qa × ΔT × C2 .... (3)

Where,

q = Coil capacity (given)

qa = Coil capacity from air side calculations.

qw = Coil capacity from water side calculations.

Qa = Air flow rate

Qw = Water flow rate

Δh = Entering air enthalpy minus leaving air enthalpy.

ΔT = Leaving water temperature minus entering water temperature.

C1, C2 – are constants to take care of factors required to obtain results in desired units.

q = qw = qa

These checks are necessary and essential, but there is another equation and it is that which plays the vital role in understanding the low-ΔT problem.

q = qa = qw = A × U × LMTD .... (4)

Where,

A = Coil surface area.

U = Overall heat transfer co-efficient.

LMTD = Logarithmic Mean Temperature Difference.

In this equation, A is obviously constant for a given coil.

As the capacity falls at say 50% part load across the coil, if the chilled water also falls to 50% of full load flow, the heat balance between air and water would be neatly maintained. This would be simple. Unfortunately this is not how things work.

As the capacity changes, only U and LMTD can change (since A is constant for a given coil). U does decline with flow, but less rapidly than q, the capacity. Hence, at – say 50% air flow across the coil, the load will be 50% but U × A would have decreased by less than 50% – say about 40%. Thus the deficit must be met by a decrease in LMTD to balance equation 4.

Of the four temperatures that determine the LMTD, three stay essentially constant; the entering and leaving air temperatures do not change much – and likewise the chilled water supply temperature is nearly constant. It is therefore only the chilled water return temperature that will act to increase (by means of reducing the chilled water flow rate) to reduce the LMTD in response to the reduced load.

The whole point is that ΔT increases at coil part load or alternately, the chilled water flow is reduced to a greater degree as load falls off.

Worked example

A worked example is presented below to support the above concepts.

Coil data :

100% air flow rate	=	29250 cfm
A = face area	=	62.5 sq.ft
face velocity	=	468 fpm (at full load)

Nomenclature :

a.	GTD	=	Greatest Temperature Difference (°F)
		=	(Entering air temp.°F – Leaving water temp.°F)
b.	LTD	=	Least Temperature Difference °F
		=	(Leaving air temp.°F – Entering water temp.°F)
c.	LMTD	=	Log Mean Temperature difference °F
	LMTD	=	$\dfrac{GTD - LTD}{2.3 \log_{10} \dfrac{GTD}{LTD}}$
d.	U	=	Overall heat transfer co-efficient (Btuh/sft/°F)
e.	$\dfrac{Upl}{Ufl}$	=	$\dfrac{\text{U at part load}}{\text{U at full load}}$
f.	TR	=	Tons Refrigeration
g.	gpm	=	Water Flow Rate in gpm
h.	LWT	=	Leaving Water Temperature °F
i.	EWT	=	Entering Water Temperature °F
j.	ΔT	=	LWT – EWT

Procedure :

1. Using 100% values (Row-1) calculate LMTD.
2. Again using 100% values calculate 'U' value.
3. Calculate LMTD for 75% value (Row-2) assuming U = 0.9 times full load value.
4. Enter LMTD value from previous step in the LMTD equation and solve for LWT – by trial & error
5. Determine DT from LWT & Chilled water supply temperature.

6. Determine the flow rate.

Results of Calculations :

The table below furnishes the results of the calculations:

	TR	ΔT - °F	Upl / Ufl	gpm	
				Calculated Values	**Manufacturer's Data**
1	92.8	16.3	1.0	137	137.0
2	74.6	19.7	0.9	91	110.0
3	54.2	22.8	0.8	57	80.4
4	30.3	26.2	0.6	28	45.0

Calculations:

Consider conditions corresponding to Row 2 (i.e., at 74.6 TR load)

a. Calculate LMTD for 100% values (Row-1) :

$$\text{LMTD} = \frac{\text{GTD} - \text{LTD}}{2.3\ \text{Log}_{10} \dfrac{\text{GTD}}{\text{LTD}}}$$

$$\text{GTD} = 78.6 - 60 = 18.6$$

$$\text{LTD} = 54.5 - 42.8 = 11.7$$

$$\text{LMTD} = \frac{18.6 - 11.7}{2.3\ \text{Log}_{10} \dfrac{18.6}{11.7}} = 14.9\ °\text{F}\ \ (9.50°\text{C})$$

b. Calculate U From Row 1 (100%) :

Using 100% load values :

$$\text{TR} \times 12000 = \text{A} \times \text{U} \times \text{LMTD}$$

$$92.8 \times 12000 = 62.5 \times \text{U} \times 14.9$$

$$\text{U} = \frac{92.8 \times 12000}{62.5 \times 14.90} = \frac{11136000}{931.3} = 1196\ \text{Btuh/sft/°F}$$

c. Calculate LMTD for 75% values (Row-2) :

$$74.6 \times 12000 = 62.5 \times 1196 \times 0.9 \times \text{LMTD}$$

$$\text{Hence, LMTD} = \frac{74.6 \times 12000}{62.5 \times 1196 \times 0.9} = \frac{895200}{67259.35} = 13.3$$

LMTD required = 13.3 °F.

d. Calculate Leaving Water Temperature :

$$\text{LMTD} = \frac{(78.6 - \text{LWT}) - (53.3 - 42.8)}{2.3\ \text{Log}_{10} \dfrac{78.6 - \text{LWT}}{53.3 - 42.8}}$$

$$= \frac{(78.6 - \text{LWT}) - 10.5}{2.3\ \text{Log}_{10} \dfrac{78.6 - \text{LWT}}{10.5}} = 13.3°\text{F}$$

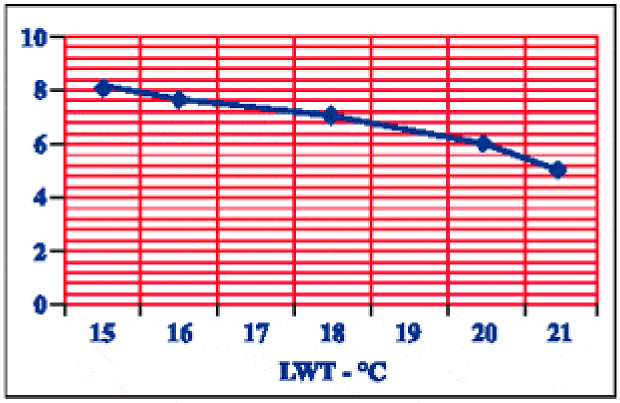

Figure 3a : LMTD vs LWT

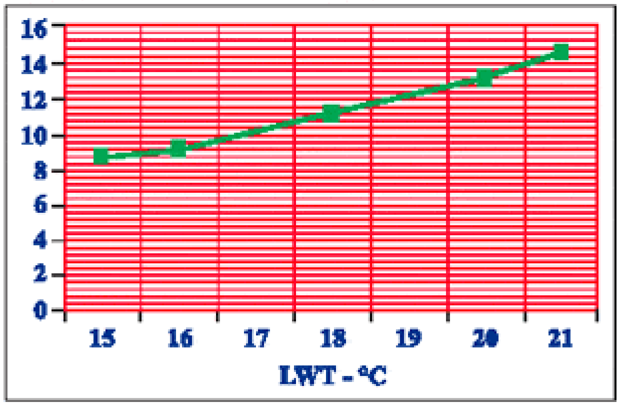

Figure 3b : Delta T vs LWT

Calculate LWT by trial and error from the above equation :

$$\text{Try LWT} = 65$$

$$= \frac{\Delta\ (78.6 - 65) - 10.5}{2.3 \text{ Log}_{10}\ \dfrac{78.6 - 65}{10.5}}$$

$$= \frac{13.6 - 10.5}{2.3 \text{ Log}_{10}\ \dfrac{13.6}{10.5}}$$

$$= \frac{3.1}{2.3 \text{ Log}_{10}\ 1.295238} = \frac{3.1}{0.258} = 12.0°\text{F}$$

Since the discrepancy between the required value (13.3°F) and the calculated value (12.0°F) is not acceptably small, repeat the calculation with different value of LWT. This procedure leads to a LWT value of 62.5 °F (for which LMTD is 13.1°F).

e. Determine $\Delta T = 62.5 - 42.5 = 20.00$

f. Determine flow rate $= \dfrac{24 \times \text{TR}}{\Delta T}$

$$= \frac{24 \times 74.6}{20} = \frac{1790.4}{20.00} = 89.52 \text{ gpm}$$

Table - 1a : Data of Manufacturer - 1 dated 23.08.2004

Capacity %	Total TR	Flow		DT		LPS/TR
		USGPM	LPS	Deg C	Deg F	
100	92.8	137.5	8.56	9.0	16.26	0.092

75	74.6	110.0	6.88	9.0	16.28	0.092
50	54.2	80.4	5.03	9.0	16.19	0.093
25	30.3	45.0	2.81	9.0	16.17	0.093

Table - 1b : Data of Manufacturer - 1 dated 21.01.2005

Capacity %	Total TR	Flow		DT		LPS/TR
		USGPM	LPS	Deg C	Deg F	
100	92.8	137.5	8.59	9.0	16.20	0.093
90	83.3	115.7	7.23	9.6	17.28	0.087
80	74.1	98.8	6.17	10.0	18.00	0.083
70	64.8	83.1	5.19	10.4	18.72	0.080
60	55.6	68.0	4.25	10.9	19.62	0.076
50	46.3	54.2	3.38	11.4	20.52	0.073
40	37.0	41.5	2.59	11.9	21.42	0.070
30	28.0	30.1	1.88	12.4	22.32	0.067
20	18.7	18.8	1.17	13.3	23.94	0.063

Table - 2a : Manufacturer - 1

Capacity %	Total TR	Flow		DT		LPS/TR
		USGPM	LPS	Deg C	Deg F	
100	92.8	137.00	8.56	9.0	16.2	0.092
75	69.45	91.00	5.69	10.2	18.4	0.082
50	46.30	54.00	3.39	11.2	20.1	0.073
25	23.35	24.37	1.53	12.9	23.1	0.093

Table - 2b : Manufacturer - 2

Capacity %	Total TR	Flow		DT		LPS/TR
		USGPM	LPS	Deg C	Deg F	

100	84	125.1	7.81	9.0	16.2	0.093
75	62	78.7	4.91	10.6	19.0	0.079
50	42	45.9	2.86	12.1	21.8	0.069
25	21	20.0	1.25	14.0	25.2	0.060

The variation of LMTD and Delta T with respect to LWT is also depicted graphically in **Figure 3a & 3b**.

Notes:

1. *U values shown at different loads are assumed values. They are not actual values, but the actual values will be proportional to them.*
2. *Calculations have been made on the assumption that the coil is handling only sensible heat load.*
3. *The calculations have been made to illustrate the concept. Actual coil selections should use U value calculations based on approved procedures and duly factoring latent heat loads.*

Example of Cooling Coil Selection

It will be useful to take a look at **Tables 1 & 2** at this point.

This example serves to show how the ΔT at partial load has to increase over ΔT at full load. When this happens, the return water temperature goes up and the LMTD correspondingly goes down. Thus compensation for the inadequate drop in the U factor is achieved.

High return water temperature automatically means a high ΔT. To achieve this high ΔT, it is certain that precautions need to be taken while selecting the coil.

Coil - Control Valve as a Team

It is clear that the coil requires a flow, that falls off steeply to begin with as the load falls from 100% load (valve in fully open position) but a flow that declines slowly as the valve nears its fully closed position. To achieve this, what is required is a valve which closes the flow 'fast' to begin with and only 'gradually' later as the capacity approaches zero. A valve with such a characteristic is a globe type (control) valve with equal percentage plug. Hence an equal percentage valve should be applied for proper flow control.

While **Figure 4** is about the concept, **Figure 5** addresses it in a more detailed and realistic manner.

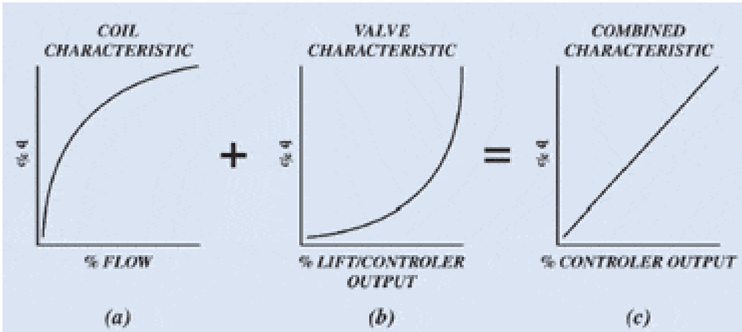

Figure 4 : Typical coil and valve characteristic "marriage"

In **Figure 4a**, the coil characteristic is shown. Figure 4b shows the valve characteristic. As already discussed, the former depicts the inadequate sensitivity of coil capacity for flow reduction sensitivity of coil capacity for flow reduction to begin with i.e., as the valve starts closing and an improvement as the valve approaches the fully closed position. The valve characteristic (for equal percentage valve) shows 'fast' reduction in flow to begin with and a slow down as the valve approaches the fully closed position. The mutually complimentary nature of the characteristics results in a linear characteristic.

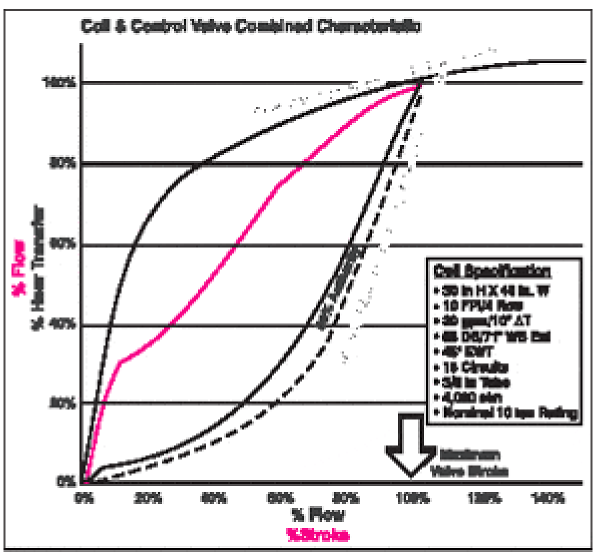

Figure 5 : Typical control characteristic with control valve. The combined coil heat transfer characteristic for a cooling coil with a valve characteristic (valve authority 50%) shown.

Valve Requirements

1. The valve should have a pressure drop that is large enough to ensure that its operation will have necessary impact on flow variation (control). An index of this

aspect of valve performance is:

$$\frac{\text{Valve } \Delta P \text{ at } 100\%}{\Delta P \text{ of the branch circuit (in which the valve is operating)}}$$

2. The rangeability of the valve (the ratio of flow at fully open position and minimum flow that the coil is likely to operate) should be not less than 50 to 1. This will ensure satisfactory modulation down to 1/50 of full flow.
3. The control valve should close tight against the highest differential pressure that the chilled water pumps can generate.
4. Valve actuators, valve cages, trim plugs and seals should be robust. Otherwise high velocities and throttling will cause deterioration of valve seats and permit chilled water leaks in valve closed position. A 50% safety factor is recommended.

Summary

Summarizing, it will be noted that the following points emerge:

1. Chillers should be specified for high ΔT.
2. The chilled-water ΔT should be equal-to-design at full-load and greater-than-design at part-load in a variable -flow hydronic cooling system.
3. Chilled-water ΔTs are determined by a building's various terminal devices.
4. High chilled-water ΔTs result from proper design, installation, operation, and maintenance of cooling coils, control valves, control systems, distribution pumps and distribution piping.
5. More water chillers, larger chilled-water pumps, and/or larger chilled-water piping will not overcome the performance problems resulting from low/belowdesign chilled-water ΔTs.

Conclusion

Inspite of the “Low-ΔT Syndrome”, high-ΔT systems are here to stay. Selection of chillers, coils, control valves and design of the rest of the system particularly piping system, has to be done to meet the demands of high- ΔT systems in a meticulous and professional manner. The designer should make sure that the coil and value selections are carried out specifically to meet the stipulations. The supplier of coil and valves in particular should understand the

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