I’ve got some bad news—the 20-odd-year experiment with primary-secondary design of chilled water plants hasn’t panned out. If you’ve designed a large distributed chilled water system and monitored the operation of the central plant, you already know about the problems: the ΔT of the chilled water (CHW) returning to the campus plant is below the design value for which the chillers and pumps were selected—in fact, it’s way below; the secondary CHW flow doesn’t vary a hoot; and the expensive variable-speed drive (VSD) purchased to vary the flow of the secondary pumps was great for test and balance but hasn’t done much since (besides heating up the central plant building). The low ΔT at the plant causes the operators to run extra pumps and chillers to meet the load, which, in addition to reducing the plant’s cooling output capacity, wastes energy. The system may be keeping the campus cool, but you know it’s inefficient and idling a lot of chiller capacity.

The problem described above has come to be known as “low ΔT central plant syndrome.” To my knowledge, every large chilled water plant serving distributed loads is afflicted with it to some degree. The article “Troubleshooting Chilled Water Problems at the NASA Johnson Space Center” (HPAC, February 1995) describes a typical situation. A central plant originally designed for a 16 F ΔT between the chilled water return (CHR) and chilled water supply (CHS) could only develop an 8 F ΔT because of low CHR temperature from the campus. This meant not only that twice as much CHW as originally intended had to be pumped around the 5-mile campus piping loop but also that the seven 2000-ton chillers in the central plant couldn’t be loaded much beyond half their capacity. Thus, operators were usually forced to run twice as many chillers to meet the campus load, and the frictional loss in the mains due to the excessive CHW flow made it

1 Archetypal primary-secondary CHW plant design.
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tough to deliver sufficient CHW to hydraulically distant buildings.

The causes of low ΔT syndrome are not mysterious, but they are often pervasive and thus can be hard to remedy. Low ΔT can be caused by dirty cooling coils, throttling valves with insufficient shutoff capability, reset CHS temperature, poorly controlled blending stations, and of course, CHW bypassing out in the system. But most often, low system ΔT is the result of faulty controls and improperly adjusted set points. This article, however, is not about the causes of central plant syndrome. It’s about accepting that the problem exists in virtually every big distributed chilled water system and then recognizing the need to seek design solutions that can cope with it, if not prevent it.

So why can’t a standard primary-secondary chilled water design cope with low CHW ΔT?

Problem #1

The primary-secondary control scheme is “blinded” by low ΔT central plant syndrome. Fig. 1 depicts what I would describe as the archetypal primary-secondary chilled water schematic configuration. The primary feature of the configuration is the decoupled primary and secondary loops, which allow constant flow through the chillers while permitting varying flow in the system to save pumping energy. Chillers are staged on and off based on CHW flow through the crossover bridge (although the sensor may be elsewhere). The sole indicator of system load, upon which control of the chillers and pumps depends, is chilled water flow.

In a plant with low ΔT syndrome, CHW flow is no longer much of an indicator of load. The amplitude of flow variation is just a fraction of the amplitude of load variation. Fundamentally then, a primary-secondary control scheme that depends on system flow to gauge system load is virtually blind to load variation.

Problem #2

The primary loop is constant flow. Constant flow through chillers is a highly desirable feature of primary-secondary chilled water plant design, and most chiller manufacturers still prefer and recommend it. I’ve been convinced, however, that most modern chiller controls no longer require constant flow to keep the chillers out of trouble. Let me explain.

When chiller vanes were controlled by conventional pneumatic proportional controls, response time to changes in load was necessarily slow and gradual to prevent overshoot and hunting as the chiller controls tried to achieve leaving CHS set point. Hence, chiller capacity controls would lag behind a sudden load change. If the change was a drop in load, the chiller would overcool the leaving CHW, dropping it below set point until capacity control vanes could react to reduce chiller refrigerating capacity. If the drop in load was sharp enough, the chiller’s low evaporator temperature safety would knock the chiller off line, requiring a manual reset to restart the chiller. This is a situation to be avoided.

Now consider the response of a chilled water plant designed for constant flow versus one designed for variable flow in the event that load across a fully loaded chiller suddenly dropped in half. (This is a severe upset, but it’s not far-fetched at all. Starting a second chiller in a two-chiller plant, where identical chillers operate in parallel, typically results in the load to the active chiller being halved.) In a constant-flow primary loop designed to chill, say, 55 F CHR to 45 F CHS, a 50 percent drop in load would manifest itself in CHR temperature rising to 50 F. (This might occur because approximately half the primary flow of 45 F CHS is recirculating through the crossover bridge to mix with the 55 F CHR from the system.) The 50 F CHR entering the formerly fully loaded active chiller would initially be subjected to the full cooling capacity of the chiller until its controls could respond to decrease capacity. The chiller would thus tend to drive the entering 50 F CHW down toward 40 F.

Compare this upset condition to a variable-flow configuration. Starting a second equal CHW pump could cut CHW flow through the active chiller roughly in half. The active chiller would initially continue to try to apply its full output capacity to half the mass flow, thereby doubling the ΔT of CHW passing through it—i.e., it would tend to drive 55 F CHR down to 35 F. This is pretty close to freezing. If the design ΔT was larger, the CHW would be driven down below freezing. In either case, a simple low evaporator temperature sensor would likely cause the chiller to trip off line to protect it from freezing. The constant-flow chiller, in comparison, whose leaving CHS temperature dips only half as far, would probably remain on line. For this reason alone, one can easily understand why chiller manufac-

2 For example, in a variable-flow plant, flow through the active chiller will be cut roughly in two as the second chiller’s pump instantly usurps half the flow. In a primary-secondary plant, approximately half the total primary CHW flow recirculates through the crossover bridge and, once the second chiller’s compressor has begun outputting CHW, mixes with system secondary CHR to halve its ΔT.

3 Assuming immediate system control valve response and the absence of central plant syndrome. If control valve response were slow or the system were afflicted with central plant syndrome, flow would not suddenly fall to half, and thus the upset condition would be far less traumatic.
turers would prefer constant flow
through chillers.

So what’s changed to invalidate
this argument? The low evaporator
temperature control is more so-
plicated, for one. It’s no longer
simply a low-temperature safety
cutout. The Trane Company’s mi-
croprocessor-based control, for ex-
ample, integrates (i.e., sums) the
number of degree-seconds (deg-
sec) below the low evaporator tem-
perature set point. Don Epple-
heimer of Trane tells me that if
this sum remains below 50 deg-sec,
the control logic will not initiate a
safety shutdown. This means:

♦ Evaporator temperature may
drop below freezing momentarily.

♦ The chiller’s capacity controls
are allowed time to catch up with
the load change.

In fact, the sophistication of the
control logic is such that Trane
feels confident in setting its low
evaporator temperature set point
as low as 30 F.

The upshot of this improve-
ment, and to a lesser extent the
capacity control improvements, is
that chillers can survive a severe
upset condition in CHW flow
without tripping off line. In fact,
and this is really the proof of the
pudding, Mr. Eppleheimer says
that Trane routinely tests its
chillers to insure they can with-
stand a 50 percent drop in CHW
flow without tripping the low
evaporator temperature safety
cutout. (Other manufacturers
have different control strategies
for handling upsets in applied
load. York’s chillers, for example,
can accept an input that delays
powering down of the chiller com-
pressor upon a large drop in leav-
ing CHS temperature as long as
the temperature does not fall be-
low 36 F. Carrier’s low evaporator
temperature control overrides the
chiller capacity controller to close
compressor vanes should evapor-
tor temperature approach 33 F. At
33 F, the safety shuts down the
machine.)

There is another benefit to con-
stant flow through chillers. The
possibility of laminar flow
through the evaporator due to low
CHW flow is eliminated. This
condition can easily be avoided, how-
ever, in a variable primary flow
system. Burt Rishel of Systecon
suggests the best way to do this is
the old-fashioned way—with a by-
pass from CHS to CHR opened via
a signal from a flow meter or dif-
fferential pressure controller
across the chiller. This might
seem to replicate the expense of a
primary-secondary crossover
bridge, but the difference is that
the recirculation pipe is sized to
handle no more than about half a
chiller’s design flow, and its func-
tion is less likely to confuse the
operator.

So the rationale for avoiding
variable flow through chillers is,
in my opinion, no longer com-

individual chillers to load them
more fully. But with the primary-
secondary configuration shown in
Fig. 1, this is not possible. Pri-
mary-secondary systems can be
retrofitted, of course, in response
to low ΔT. New pumping capacity
can be added and flow through
chillers can be increased up to the
manufacturer’s recommended
maximum rate. Perhaps evapora-
tors can even be converted from
three-pass to two-pass, reducing
pressure drop through the
chillers. Furthermore, retrofitting
and adding equipment to accommodate a
lower ΔT within the context of the
existing constant-flow design re-

![Diagram of variable-flow CHW plant design.](image)

2 Typical variable-flow CHW plant design.


> The maximum tube velocity recommended in the ASHRAE Equipment Handbook is 7 fps; most manufacturers recommend 11 fps.
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chillers as a target value and then provide for the eventual-
ity that extra flow beyond the design target may be re-
quired. In practice, this means:

- Selecting chiller evapor-
ator tubes for tube velocities not more than about 5.5 fps
at design so that flow can be increased up to twofold if nec-
essary. 

- Selecting pumps to over-
pump the chillers. The best scheme is to bank the pumps and provide them with VSDs.

With variable flow pumping through the chillers, the crossover bridge and the secondary pumps can be dispensed with, so a typical schematic layout for a simple building can look like Fig. 2. Chillers are staged based on leaving CHS temperature. When a chiller can’t hold leaving CHS temperature set point, a second chiller is energized. Pump speed is controlled by a differential pressure sensor situated across the hydronically farthest coil. A flow meter and smart controller open a bypass valve should flow through the chillers fall below the manufacturer’s recommended minimum.

The advantages of this system are:

- It automatically responds to low ∆T by increasing flow through chillers.
- There’s only one set of pumps.
- Minimum chilled water flow is pumped.
- The system is simpler; there’s no decoupling bridge.

Problem #3

Secondary pumping is not the most efficient pumping distribution scheme. In Fig. 2, a single set of pumps handles the job of both the primary and secondary pumps. But in a big system, a single set of pumps is not always desirable. If the pressure needed to pump an entire campus is large, it’s advantageous to place a second set of pumps, and perhaps even a third set, out in the system to avoid imposing high pressure on the equipment close to the pumping station. Secondary pumping, as shown in Fig. 1, can achieve this objective, but it’s not the most efficient pumping scheme. That’s because the same head is imparted to all CHW passing through the secondary pumps, whether it’s making the short trip through the closest building or the longest trip through the hydraulically most distant building. The extra head imparted to CHW passing through the closer buildings must be wasted across balancing valves and/or throttling valves at those buildings. Only the small fraction of the total CHW flow going to the most distant building is produced without wasted energy.

A better way to pump distant loads is via distributed pumping, as illustrated in Fig. 3. Burt Rishel of Systecon gives credit to Wilber Shuster of Cincinnati for first proposing this pumping scheme. Distributed building pumps assume the function of the secondary pumps. Each pump is sized to deliver its building’s flow at just the head needed to pump the building hydronic loads and draw the CHW through the mains from the central plant. There are no decoupling loops at the buildings, so no CHW is bypassed. No balancing valves are needed to eat up excess head since there is none. Pump speeds are controlled by VSDs receiving signals from differential pressure switches at the end of the loop in each building. Pumping horsepower saving equals the sum:

\[
\sum_{i=1, N-1} \text{CHW flow}_i \times \frac{(H_{building} - H_{building_i})}{(3960 \times \eta_p)}
\]

The primary pumps are VSD-controlled, as before, and can operate in series with the distributed pumps or be decoupled as shown in Fig. 3. If decoupled, the VSDs would be controlled to maintain slightly positive flow from CHS to CHR in the crossover bridge and not let flow through any chiller go below its minimum recommended value. Chillers are

6 Of course, oversizing the pumps and balancing them down with a throttling valve is not an option unless you routinely wear your shoes on the wrong feet and when you tighten your belt, cut off your windpipe.

7 Assuming equal pump efficiencies for building pumps and hypothetical secondary pumps.
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...staged based on their ability to maintain leaving CHS temperature.

The advantages of this system, besides minimizing pumping power, are:

- It minimizes the potential for low system ΔT by eliminating crossover bridges at the buildings.
- It reduces head pressure imposed on equipment.
- It's simple and, more importantly, looks simple to the operators who run it.

The only unusual aspect of distributed pumping is that it reverses the typical pressure gradient in the system. The CHS main is negative with respect to the pressure in the CHR main. Thus, every load must be pumped.

In conclusion...

The traditional arguments for desiring constant flow through chiller evaporators no longer carry much weight; most modern microprocessor-based chiller controls can effectively deal with upsets due to variable flow. Moreover, constant-flow primary designs cannot respond to the need to put more CHW through chillers in the event that the distribution system returns low CHW ΔT to the central plant.

A variable-flow design with pumps either oversized and controlled by VSDs or banked can respond to low ΔT central plant syndrome. Thus, for the same reason that we as HVAC designers provide freezestats upstream of cooling coils, nonoverloading motors to drive pumps and fans, and tube pull space at chillers, boilers, and air-handling units, we need to design chilled water plants that can anticipate the possibility of low CHW ΔT and respond to it. Therefore, I believe it's time to put primary-secondary pumping back into our tool bag of applications to address specific design situations and adopt a new paradigm for chilled water system design.

Bibliography

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