PDHonline Course M328 (5 PDH)

HVAC Chilled Water Distribution

Instructor: A. Bhatia, B.E.

2020

PDH Online | PDH Center

An Approved Continuing Education Provider

HVAC Chilled Water Distribution

A. Bhatia, B.E.

A chilled water system is a cooling system in which chilled water is circulated throughout the building or through cooling coils in an HVAC system in order to provide space cooling. The principal objectives of chilled water pumping system selection and design are to provide the required cooling capacity to each load, to promote the efficient use of refrigeration capacity in the plant, and to minimize pump energy consumption subject to whatever budgetary constraints may apply. In the typical design process, such decisions are made on the basis of economic calculations.

In chilled water systems, pumping normally draws from around 6 to 12 % of the total annual plant energy consumption. Conventional chilled water plants distribute water at constant flow rate, regardless of the actual cooling demand. Since most air-conditioning systems only reach peak load a few hours a year, energy is wasted by continually running the pumps at constant flow (speed). An efficient distribution system use variable flow that tracks the variable thermal load.

This course provides a description of common chilled water system types from the most basic constant flow to all variable flow systems along with advantages and limitations. For purposes of discussion, chilled water pumping systems are divided into three categories.

1. Constant Volume Chilled Water Systems

2. Primary/Secondary Chilled Water Systems

3. Variable Primary Flow Chilled Water Systems

Before we move further, let's briefly refresh the basic hydronic principles:

In chilled water systems, water is distributed throughout the entire hydraulic network. The chilled water flow rate required for a given heat load is determined as follows:

$$CWFR = \frac{HL}{D \times 60\frac{min}{hr} \times SG \times SH \times \Delta T}$$

Where:

- CWFR: Chilled water flow rate (gal/min)

- HL: Heat load (BTU/hr)……or chiller capacity

- D: Liquid density (water density = 8.34 lb/gal)

- SG: Liquid specific gravity (water specific gravity = 1)

- SH: Liquid specific heat (water specific heat = 1 BTU/lb-°F)

- ΔT: Chilled Water temperature differential [i.e. T return water - T supply water] in °F

Simply stated the water flow rate is given by equation:

$$GPM = \frac{BTU/hr}{500 \times \Delta T\,(°F)}$$

Since, the HVAC designer's converse more in terms of tons of refrigeration, the equation can be simplified as:

$$GPM/Ton = \frac{12000}{500 \times \Delta T\,(°F)}$$

OR

$$GPM/Ton = \frac{24}{\Delta T\,(°F)}$$

Note that 1 ton of refrigeration = Heat extraction rate of 12000 BTU/hr

Example

How many tons of cooling is served by a 6,000 GPM operating at 42°F supply water temperature and 54°F return water?

GPM = 6,000

Supply water temperature = 42°F

Return water temperature = 54°F

ΔT in distribution system = 12 degrees ----- [54°F – 42°F]

LFR = 24/ ΔT = 24/12 = 2 GPM/Ton

Tons Cooling = GPM / (24/Delta T) = 6,000/2 = 3,000 tons

The example illustrates a very important aspect. For a given design load, the chiller is selected for fixed temperature range ΔT and the associated chilled water pumps are selected for calculated flow rate. The chilled water supply temperature usually has a fixed set point and the only variable that affects the output capacity of chiller is the return water temperature. We will learn more in the following sections.

Pump Affinity Laws

The affinity laws are rules that govern the performance of a pump under varying conditions of flow and pressure. The laws can be roughly stated as follows:

- Flow varies with rotating speed

- Head varies with the square of the rotating speed

- Brake horsepower varies as the cube of the rotating speed

Example

A chilled water pumping system delivers 10000 GPM @ 240 BHP. What is the approximate required pumping horsepower the flow rate is cut to half by variable speed drive on the pumps?

10,000 GPM requires 240 BHP

5,000 GPM requires what BHP?

Q_1 = 10,000 GPM, Q_2 = 5,000 GPM, BHP_1 = 240

$BHP_2 = BHP_1 (Q2/Q1)^3$ or $BHP_2 = 240(5,000/10,000)^3 = 240/8 \approx 30$ BHP

The example illustrates another important aspect. If the flow rate were 50% of the design flow, half the higher flow rate then the power necessary to provide the slower flow rate would be $(0.5)^3 = 0.125$ or 12.5 % of the power required for the higher flow rate. It means an energy saving of 87.5 %. Such savings justify the applications of variable speed pumping in chilled water distribution schemes, which we will study in following sections.

SCHEME #1   CONSTANT FLOW CHILLED WATER SYSTEM

A constant flow system is the simplest chilled water distribution scheme. Here, a set of constant speed pumps distributes fixed quantity of water at all times and the temperature varies to meet the load. The system uses 3-way control valves at air handler coils that allow some water to bypass the cooling coil during part load conditions. At low loads, the chilled water flow through the cooling coil is restricted (in response to supply air temperatures to the space) but the total quantity returned to the chiller remains constant.

Figure below shows the schematic of the constant-flow rate primary system.

PRODUCTION LOAD — DISTRIBUTION LOAD

(CHILLER PLANT)   (BUILDING COOLING LOAD)
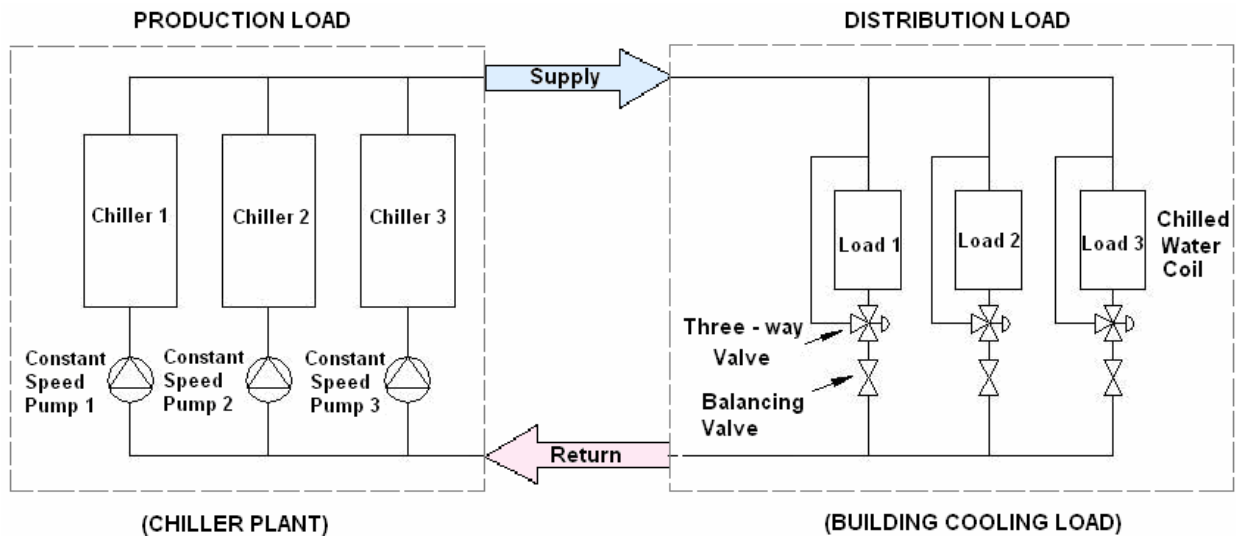
Looking at the equation that governs heat transfer, the capacity of a chiller is proportional to the product of flow rate and the temperature difference of entering and leaving chilled water, or chiller capacity (BTU/hr) = GPM x 500 x ΔT

In constant-flow systems, flow rate (GPM) is established for peak design condition and Delta-T (the difference between the chiller supply and return) varies in reponse to the load (BTU/hr).

In air-conditioning applications, since the design conditions occur only during 1% of the operational hours in a year [i.e. 99% of the time the system runs on part load], some of the chilled water will always bypass through the three-way valve for most of coil's operational life. Higher quantities of bypassed chilled water mixes with leaving water from the cooling coil yielding lower chilled water return temperature to the plant. The lower return water temperature reduces the operating temperature differential (ΔT) across the chiller as the supply water temperature is fixed to a setpoint. This phenomenon is termed "Low delta – T syndrome" in HVAC industry.

Remember that the chillers are able to control only the temperature at which chilled water is sent out, while cooling loads control the temperature at which it is returned and, ultimately, Delta-T.

Higher return water temperature leads to higher loading of chillers.

Impact of Low delta-T (∆T) syndrome

Low ΔT can be a serious problem for constant flow system because it robs chiller of its useful capacity. For example, if chiller is receiving return water temperature of 52°F rather than the design return water temperature of 55°F, the chiller will be loaded to:

$$CHL(\%) = \left[\frac{CWRTR - CWSTD}{CWRTD - CWSTD} \right] \times 100$$

Where:

• CHL (%): Percent chiller loading

• CWRTR: Real chilled water return temperature

• CWSTD: Design chilled water supply temperature

• CWRTD: Design chilled water return temperature

In this case, ΔT_{Design} = 55- 45 = 10°F and ΔT_{Actual} = 52 – 45 = 7°F:

$$CHL(\%) = \left[\frac{52 - 45}{55 - 45} \right] \times 100 = 70$$

What does this mean?

The chiller can be loaded only to 70% of its capacity. To overcome the capacity deficiency, the plant operators turn on extra chillers and associated pumps to provide the needed capacity. This means, not only the delivered capacity of the chillers is reduced, but also the operation becomes inefficient as the chillers must remain "on-line" to satisfy the building load. This increases plant's energy usage and compounding the problem, when chillers are not optimally loaded, these are less efficient consuming higher energy rate per ton of cooling (i.e., kW per ton). The system may be keeping the building cool but it is inefficient and a lot of chiller capacity is wasted.

Let's check this out again with an example.

Example

- Building type = Office

- Structure = 6 floors

- Design load of building = 600 TR (100 tons of refrigeration each floor)
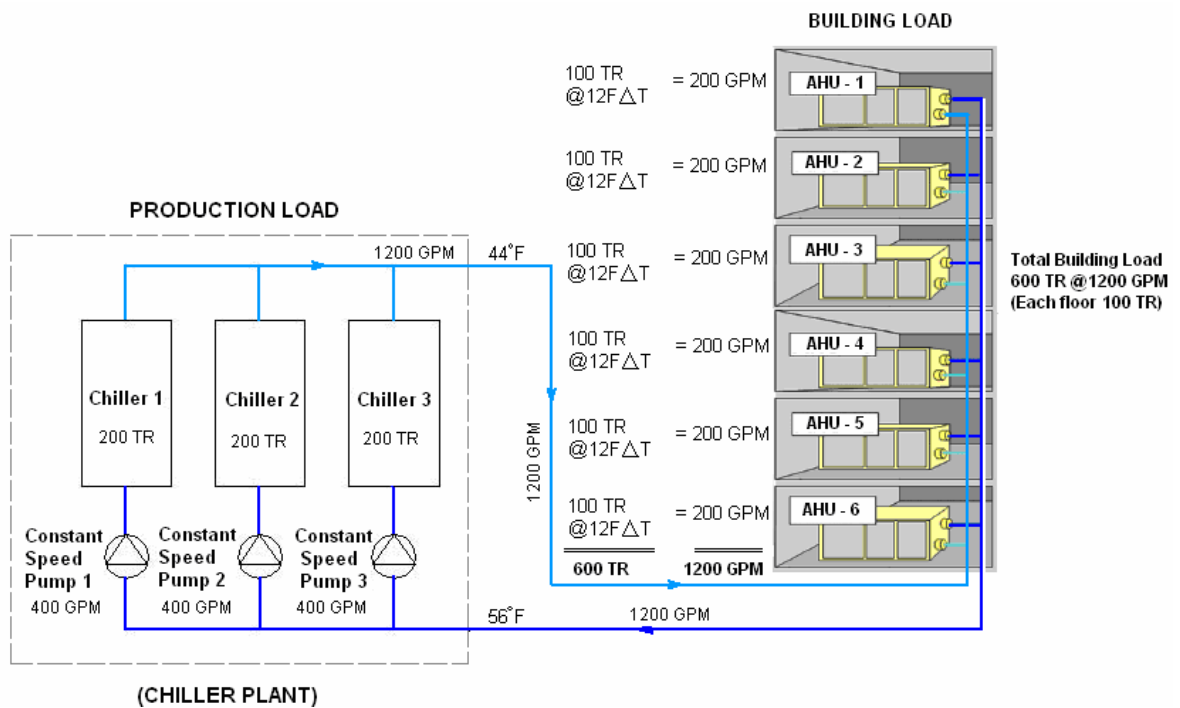
- Each floor (coil) is designed for 100 tons of refrigeration with a cumulative load of the 600 TR.

- Three x 200 ton capacity chillers serve to meet this load with chilled water supply (CHWS) temperature of 44°F.

- Both the chillers and the cooling coils of AHUs are designed for ΔT of 12°F yielding flow rate of 400 GPM [24/ΔT x 200 TR chiller capacity] thru chillers and 200 GPM [24/ΔT x 100 TR AHU cooling coil capacity] thru AHU cooling coils.

Now let's look at two operation scenarios: 1) Full load and 2) Partial load
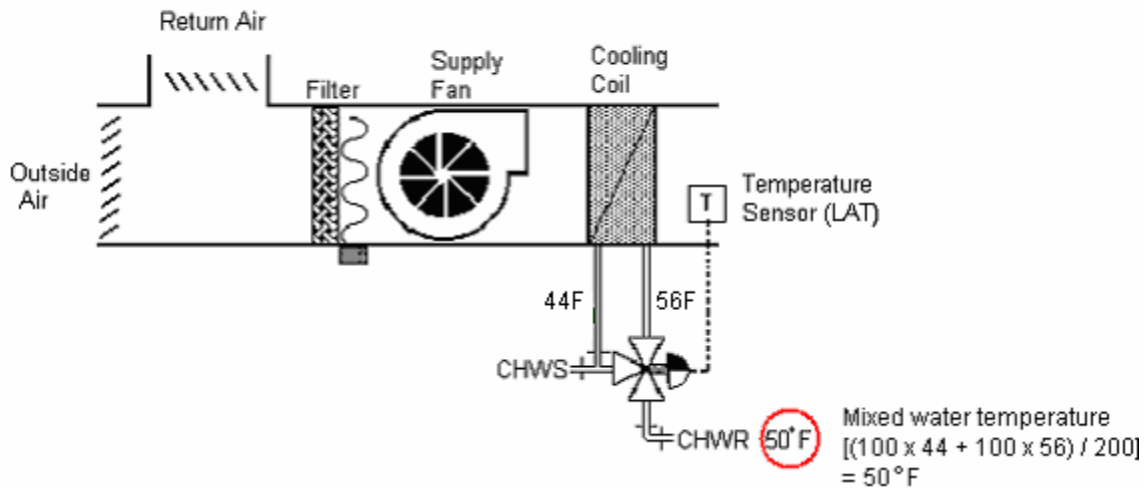
Full Load Conditions

Figure below shows the peak design conditions where chilled water is entering the cooling coil at 44°F and leaving the cooling coil at 56°F. At full load 100% of chilled water @ 200 GPM will flow through the coil and 3-way bypass valve will not allow any water to bypass.

AHU PEAK LOAD CONDITION
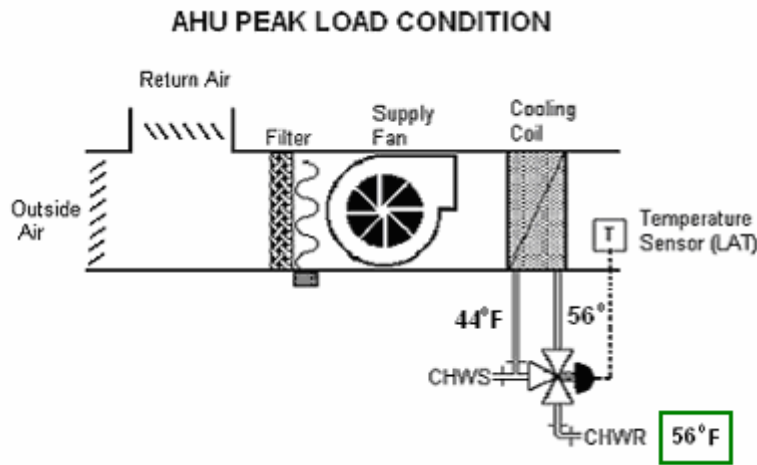
Off Load Conditions

Consider off -peak conditions, when the actual load drops to 50%. As the LOW load is sensed by the leaving air temperature thermostat, the 3-way valves restrict water supply to the cooling coil, allowing some chilled water @ 44°F to bypass directly into the return line water stream temperature of 56°F.

AHU "OFF - LOAD" CONDITION

Return Air

Supply Fan   Cooling Coil

Filter

Outside Air

Temperature Sensor (LAT)

44F   56F

CHWS

CHWR 50°F   Mixed water temperature
[(100 x 44 + 100 x 56) / 200]
= 50°F

Assuming linear* relationship between coil heat transfer and water flow rate – 100 GPM of flow rate will be needed for achieving 50% heat transfer of coils OR in other words 50% (100 GPM of water @ 44°F) will be bypassed resulting in mixed chilled water return (CHWR) temperature of 50°F…. (Refer figure above).

[*Note that the coil characteristics are actually NON-linear i.e. at 50% flow, the coil capacity is still 80% (and not 50%). The flow has to be reduced to a very low percentage of design in order to provide low part-load heat transfer and approximately 30% flow rate is enough for achieving 50% capacity of the coils. Thus actually, 70% of water will bypass @ 44°F and the return water temperature to chiller will lower than 50°F. The delta-T across the chiller will have the worst impact than you may think].

The Delta-T across the chiller will be 6°F and the chiller will be loaded to 50% of its capacity i.e. 100 TR.

$$CHL(\%) = \left[\frac{50 - 44}{56 - 44} \right] \times 100 = 50$$

This leads to three important conclusions:

1. Although the 50% demand of the building (300 TR) can effectively be met through two chillers, the operators will run 3 chillers to meet the capacity. This is because the ΔT has dropped to 6°F limiting the effective delivered capacity of each chiller to 100 TR.

Because evaporator flow rate is constant, full cooling capacity can be achieved only when the chilled water temperature difference across the evaporator is at its design value.

2. Although the 50% demand of the building (300 TR) can effectively be met through 600 GPM of chilled water, all the three chillers pumps must remain on-line to deliver 1200 GPM.

3. So long all the AHUs on the building distribution side are operational, 2 x 300 TR designed chillers must remain on-line irrespective of the load. The only time one chiller can put out of operation is when at least two floors of the building are non-operational.

PUMPING ARRANGEMENT

There are two ways to arrange pumps:

TANDEM PUMPS   HEADERED PUMPS
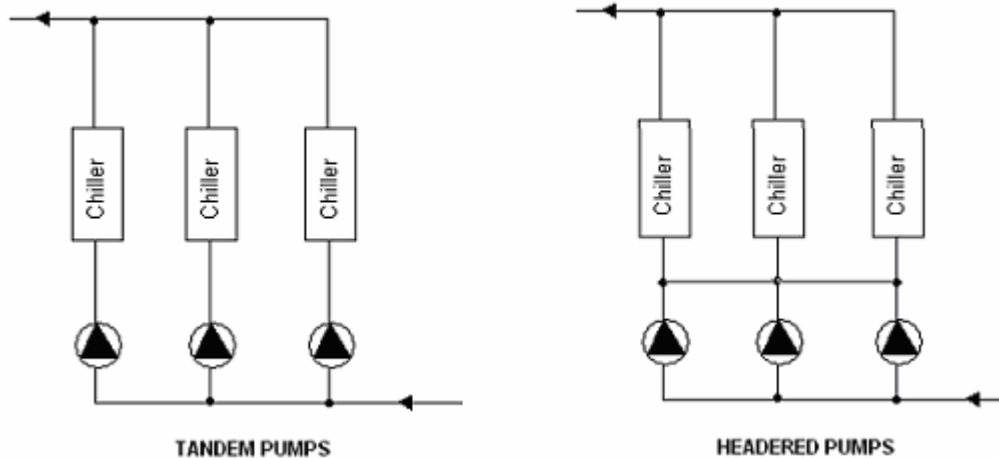
Tandem arrangement – Here each pump is dedicated to its respective chiller i.e. pump 1 is piped directly to chiller 1 and whenever this chiller is operating its dedicated pump should be operating. Building operators find this simplicity beneficial. Another benefit of

dedicated pumps is that they can handle unequally sized chillers without using control valves and flow measurement devices to balance the correct flow to each chiller. The downside is that a standby pump cannot be started automatically by the building control system, but instead needs manual intervention. This can present a problem with chiller plants that do not have standby capacity. The constant flow systems are recommended in distributed arrangement.

Headered arrangement – Here the discharge of multiple pumps is connected to a common manifold before entering the chiller. This arrangement is desirable for a few reasons: First, they give users the ability to operate any chiller with any primary chilled water pump. This helps whenever a single pump is down for maintenance because a pump outage does not correspond to a particular chiller. Also, headered pumps give users the ability to operate more than one pump for a single chiller. This can help solve a low ΔT problem by increasing primary flow and forcing a chiller to a greater load when the return temperature is less than design.

Disadvantages

If the multiple chillers are arranged in headered arrangement, these must be "on-line" all the time irrespective of load. By turning off one chiller, the flow will split into three streams, allowing the flow to continue through the down machine/s. The supply temperature from the plant will thus increase due to the mixing of water from the chiller that has been off, with the water of the chillers that remain "on-line". The only way to counter the supply temperature degradation is to drop the temperature of the water leaving the active chillers, but this complicates the operation and/or controls, making it an unsatisfactory option.

To overcome this deficiency, a control valve on the chiller evaporator is added to balance flow between chillers and keep constant flow to each chiller. The control valve is interlocked to the operation of pump in such a way that if one pump is off, one control valve will remain shut-off in closed position. This works well for balancing flow, but interferes with the benefit of allowing additional flow, if desired, to counteract a system's low ΔT.

HEADERED PUMPS
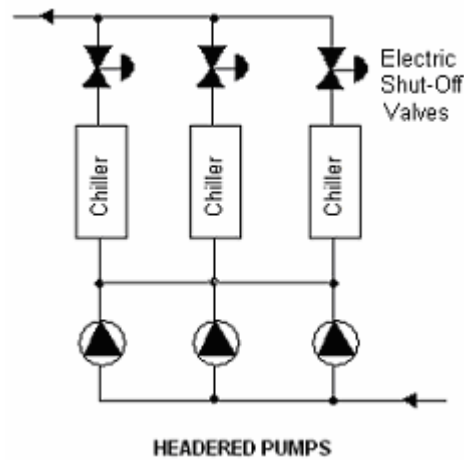
Another problem with manifolded pumps is that an entire chiller plant can fail due to a single pump's failure. Here's how: consider a typical plant with three chillers and three manifolded primary pumps, with two chillers and two pumps operating. If a single pump fails, the flow rate to each chiller drops substantially at the moment of pump failure. This sudden drop in evaporator flow trips each chiller's flow switch and causes failure of both chillers. A misconception exists that manifolded pumps allow for a backup pump to start upon a lead pump failure (thereby maintaining chiller plant operation). On the contrary, the author's experience has been that the evaporator flow switches shut down the chillers before the software of the building control system determines the pump failure and starts the backup pump. There may be a creative solution to this, but if not addressed, this is a strong reason NOT to use a manifolded primary pump arrangement.

_____

**SCHEME # 2   PRIMARY / SECONDAY DISTRIBUTION SCHEME**

ASHRAE/IES Standard 90.1- 1989 - Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings requires "all HVAC hydronic systems having a total pump system power exceeding 10 hp must be capable to flow at 50 % of design value or less." This standard highly recommends use of primary/secondary system for large complexes.

A primary- secondary pumping scheme divides the chilled water system into two distinct circuits (loops) that are hydraulically separated by a neutral bridge (de-coupler).

1. Primary circuit is the place where chilled water is produced and its principal components are the chiller and pumps. The primary pumps are typically constant volume, low head pumps intended to provide a constant flow through the evaporator of the chiller. These are usually placed in tandem with each chiller though can also be arranged in common header.

2. The secondary circuit is responsible for the distribution of the chilled water to the terminal units. Among the components of the secondary circuits are pumps, terminal units such as fan-coils and air handling units (AHU) and control valves. The secondary pumps can be <u>constant speed or variable speed</u> and are sized to move the flow rate and head intended to overcome the pressure drop of secondary circuit only.

3. The neutral bridge consists of two tees that are typically located at the suction header of the secondary pumps and at the suction header of the primary pumps and connected by a de- coupling pipe. This de-coupler separates the primary and secondary loops. This common pipe is designed for negligible pressure drop at design flow. A well designed, low-pressure- drop common pipe is the heart of primary-secondary pumping, allowing the two pumps to operate independently.

CONSTANT PRIMARY & CONSTANT SECONDARY ARRANGEMENT

Refer below for a schematic of constant primary and constant variable system. It consists of one set of constant speed primary pumps and one set of constant speed secondary pumps.
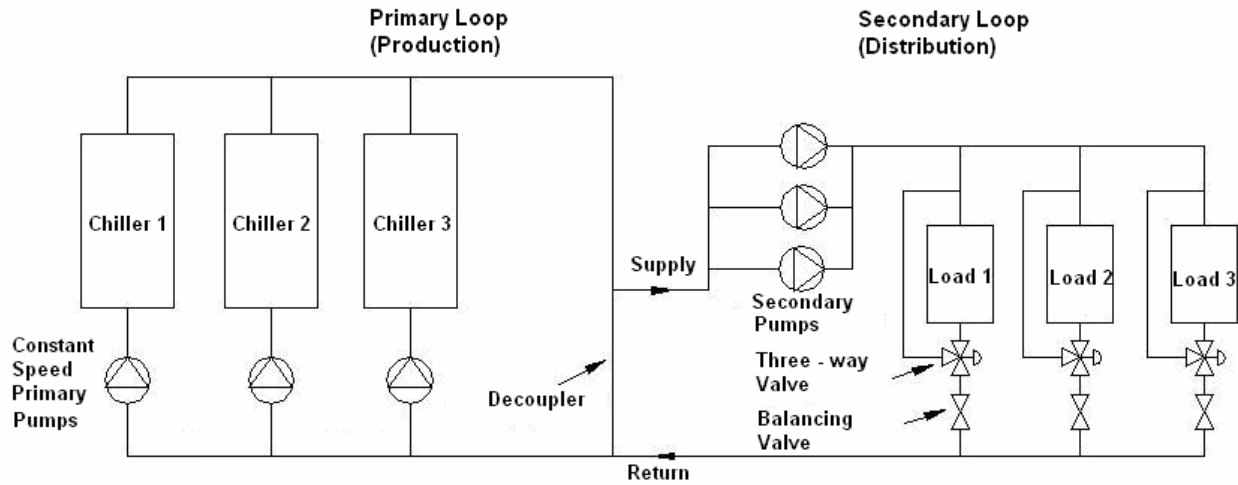
Constant flow primary/secondary chilled water system

Primary pumps are lower horsepower than the secondary pumps because they only have to overcome the friction loss associated with the chiller, pipes, and valves in the primary loop. The secondary pumps, in contrast, are higher horsepower because they must overcome the friction loss associated with the secondary loop: the distribution piping, fittings, valves, coils, etc. The secondary loop contains 3-way valves to vary chilled water quantity through the coil in response to load but the total quantity of flow in secondary loop remains the same.

One of the salient features of a primary/secondary pumping schemes is to allow different chilled water flow rates as well delta-T on the two loops.

Example

Consider the same example of scheme 1, with a difference that the scheme is converted to primary/secondary arrangement and 4 x 300 GPM constant volume pumps added in secondary loop. (Refer schematic figure below).
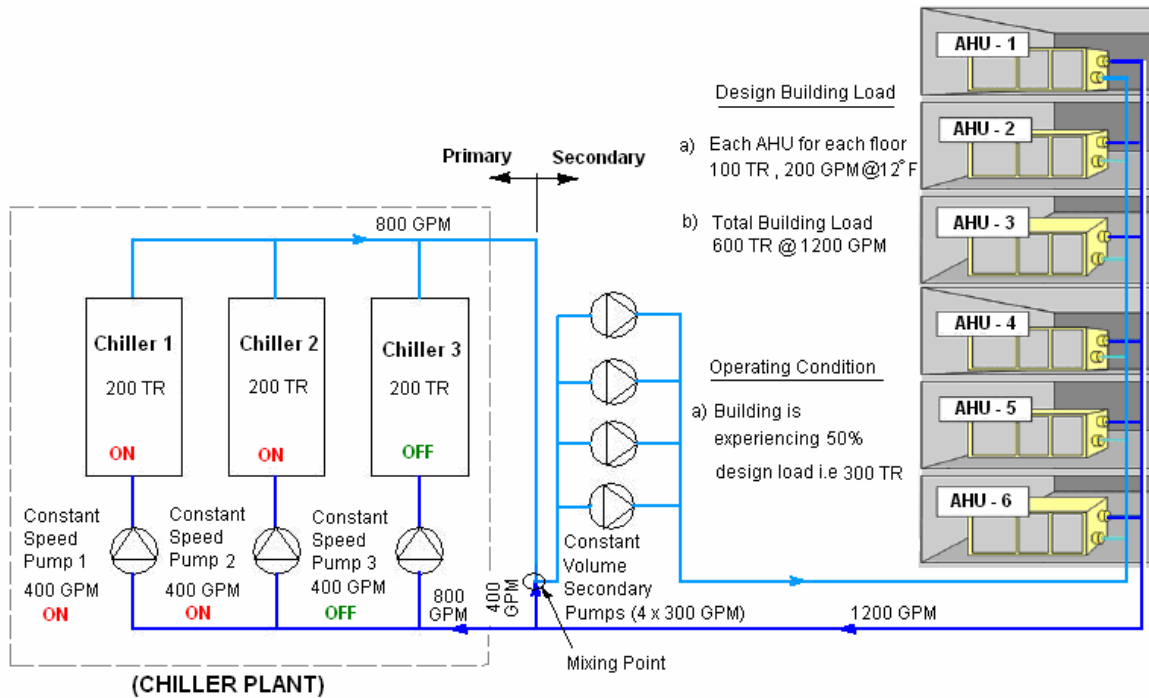
Consider the case when the building is experiencing just the 50% of the load i.e. 300 TR.

The inherent separation of the primary and secondary loops allows two different flows in these circuits because both circuits are hydraulically independent. This way the system face the variable thermal load without having to keep all chillers "on-line" and pumping energy saving are realized during periods of low loads.

Flow in primary circuit

A 50 % drop in load could be faced with two chillers, with around 67 % [33.33% + 33.33%] of the total production flow of the plant. The flow in the primary circuit will be 800 GPM and the secondary loop continues to circulate 1200 GPM. Compared to constant volume systems, it is definitely a more efficient strategy than keeping in service all production units and theirs pumps at all loads.

Flow in secondary circuit

The primary chiller sequencing does not really change the secondary loop flow rate, which remain constant and higher percent than actual load. At 50% off-load conditions, 600 GPM is required for 300 TR load but 1200 GPM delivered through 4 x 300 secondary pumps. This wastes energy at low loads.

It has another disadvantage. The excess flow will simply run through the common pipe in the direction towards the secondary pumps creating a mixing point and further degradation of the supply temperature. This reduces the cooling capacity of the coils, especially latent cooling capacity which could mean a loss of humidity control in the zones.

CONSTANT PRIMARY/ VARIABLE SECONDARY SYSTEM

In primary/secondary systems, water flows through the chiller primary loop at a constant rate, and water flows through the secondary loop, which serves air handlers or fan coils, at a variable rate. The constant speed pumps in secondary circuit are replaced with "variable speed" pumps. The speed of the secondary pumps is determined by a controller measuring differential pressure (DP) across the supply-return mains or across the selected critical zones. The decoupled section isolates the two systems hydraulically.

Also the system uses two-way valves in the air handlers that modulate secondary loop flow rate with load requirements. During light load condition, the 2-way control valves will close (partially or fully) in response to load conditions, resulting in pressure rise in the secondary chilled water loop. A differential pressure sensor measures the pressure rise in the secondary loop and signals variable frequency drive of secondary pumps to alter the speed (flow).

Primary-secondary variable-flow systems are more energy efficient than constant-flow systems, because they allow the secondary variable-speed pump to use only as much energy as necessary to meet the system demand. Refer to the schematic below.
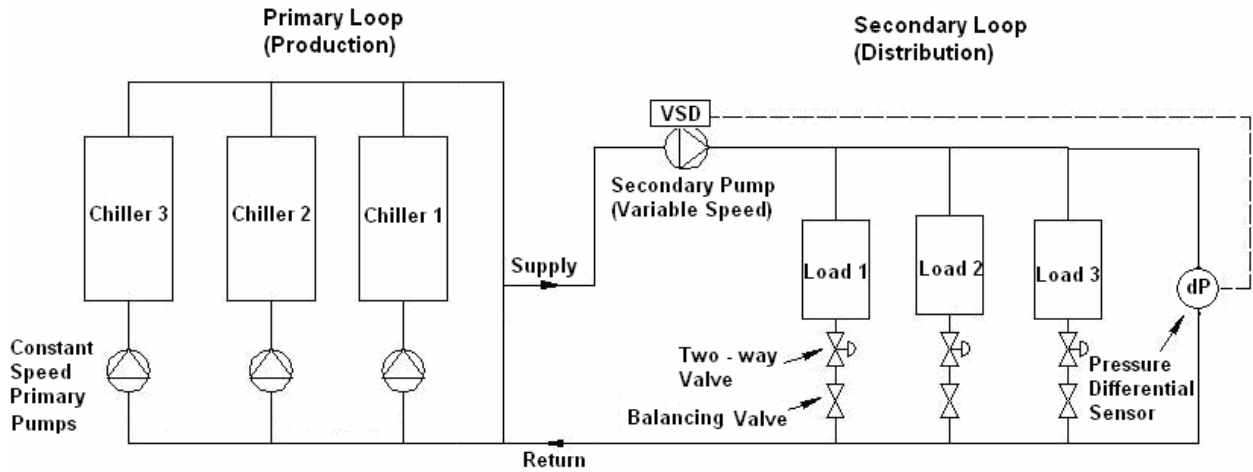
Constant-flow primary/variable-flow secondary chilled water system

While the primary-secondary system takes advantage of the reduced pumping cost offered by variable speed secondary loop pumping, it is still "blinded" by low temperature differential.

Example

Consider the same example of scheme 1, with a difference that the scheme is converted to primary/secondary arrangement and 4 x 300 GPM <u>variable</u> volume pumps added in secondary loop. (Refer schematic figure below).
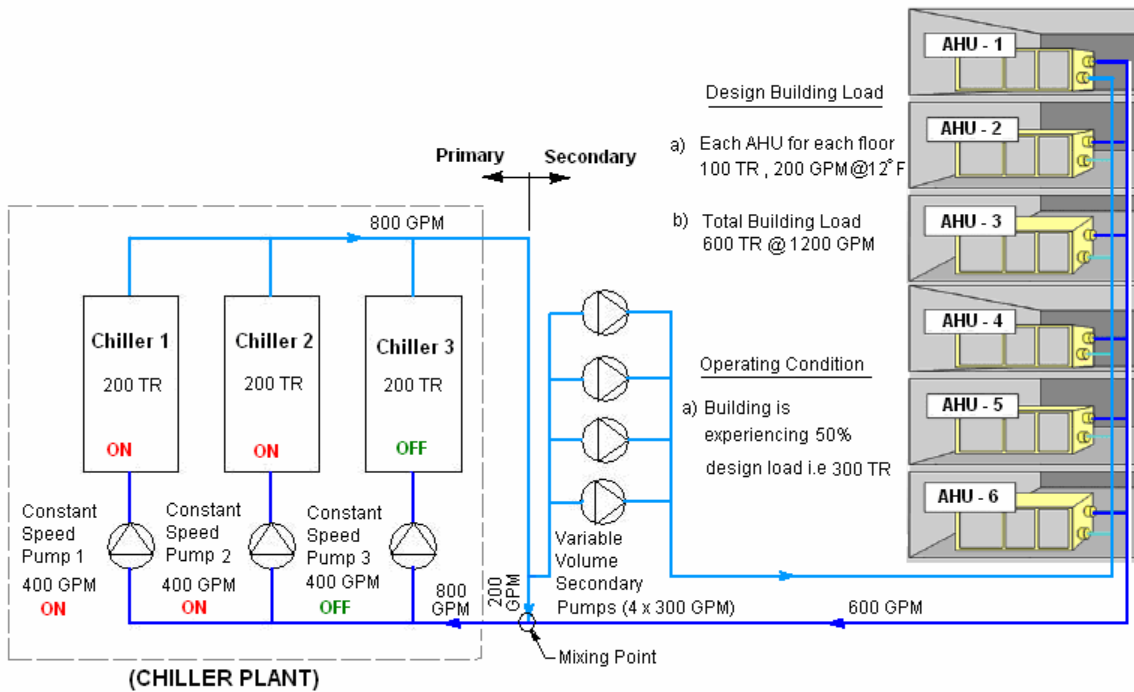
When the building is experiencing just the 50% of the load i.e. 300 TR, the flow requirement in secondary loop is varied in response to load and delivers 600 GPM for 300 TR load. The primary loop is adjusted to 2 x 200 TR or 800 GPM by turning off one chiller.

Energy savings in secondary circuit

As the volume of water circulating in the secondary circuit varies at all times because of the two way control valve on each terminal unit of the system, the variable secondary pumping saves energy at low loads. If the flow rate were half, the power necessary to provide the slower flow rate would be $(0.5)^3$ = 0.125 or 12.5 % of the power required for the higher flow rate. It means an energy saving of 87.5%.

Energy savings in primary circuit

A 50% drop in load could be faced with two chillers. It means facing a 50 % cooling load with around 67 % [33.33% + 33.33%] of the total production flow of the plant. Compared to constant volume systems, it is definitely a more efficient strategy than keeping in service all production units and theirs pumps at all loads. However, chiller sequencing does not

really change the primary loop into a true variable distribution because the flow remains a higher percent than load [600 GPM required for 300 TR and 800 GPM delivered for 2 x 200 TR chillers]. The excess flow will simply run through the common pipe in the direction towards the return loop creating a mixing point and further degradation of return temperature. This reduces the ΔT across chillers.

This "low Delta - T syndrome" has a variety of causes, some of which are easily remedied and some which are not. As per the previous discussion, optimization of the system ΔT is critical for the successful operation of a chilled water system. Let's analyze the causes and mitigation steps for the low ΔT problem.

Cause # 1: Improper setpoint or poor controls calibration:

This problem is created when the supply air set point is lowered from the design set point, often in response to complaints from occupants in the building. The cooling valve will open to increase the flow rate through the coil, which again will lead to a corresponding decrease in ΔT. Table below shows how even a modest drop in supply air temperature setpoint from 54°F to 51°F can cause coil flow rate to more than double and delta-T to drop in half. This is likely the greatest single cause of the low ΔT syndrome.

Coil Performance at Low SAT Setpoints

Leaving Air Temperature Setpoint, °F	Flow Rate, gpm	CHW Delta-T, °F	% of Design GPM
54	80	13	100%
53	104	11	130%
52	143	8.5	179%
51	208	6.5	260%
50	327	4.3	409%
49	cannot be attained		

Based on a six-row 100 fpf coil, 78°F entering dry-bulb 63°F entering wet-bulb.

Measure: Use Pressure Independent delta – P Control Valves

Check set points and recalibrate controls regularly. Low setpoints means the control valves are wide open to allow more flow. The problem of exceeding design flow rates when valves are wide open can be resolved by installing "pressure independent delta-P control valves" that maintain a preset maximum flow regardless of the differential pressure across them (within certain limits). These valves, however, add considerable cost and pressure drop to the system, both due to the valves themselves and to the strainers that are often required in front of them to prevent valve clogging. Also, these valves can only prevent flow from exceeding design rates; an improper setpoint can still result in considerable bypass of water above what is actually "used" in the coil at low-load conditions even though flow is limited.

If the control system is digital, "lock" setpoint ranges in software to prevent operators from setting overly low setpoints.

Cause # 2: Using three-way valves:

Valves control the water flow through a coil in order to vary the heat transfer rate. Three-way (diverting) valves, by their nature, bypass supply water into the return to control temperatures at the load. These systems operate with virtually constant flow, which is detrimental to the system ΔT.

Measure: Use two-way valves

All three-way (diverting) valves must be replaced by properly sized two-way, fully modulating valves to achieve variable flow conditions. Plugging the bypass port does not usually offer a good engineering solution since the three-way valve actuator spring range ratings are not adequate to close against the system pressure. In addition, the existing valves may be oversized which can cause unstable operation ("hunting").

Both two-way and three-way valves produce the same result so far as the coil is concerned; they reduce coil flow to reduce heat transfer. But there is a difference …. 3-way valves theoretically maintain constant flow and, hence, Δ-T will vary proportionally

with the load. Compared to 3-way valve systems, 2-way valve provide variable volume and save energy at part load by pumping less water through the system. This is an obvious requirement since operating cost savings for variable primary flow result from flow variation. A fifty percent reduction in system flow has the potential to reduce pumping power by as much as 80%.

In large systems, sometimes, 3-way valves are put at the terminal end (so called end-of-line 3 way valve) to keep water constantly moving through the circuit so that chilled water will be available immediately on demand by any coil in the system. The concept probably comes from domestic water re-circulating systems, which are designed to make sure people do not have to wait to get hot water at a lavatory. But the concept is not applicable to most chilled water systems simply because it seldom matters if coils have to wait a few seconds or a few minutes to get chilled water when they call for it. Unless some process load requires instantaneous chilled water, or if the chilled water plant serves buildings very far away, end-of-line three-way valves are not necessary and should be avoided.

Cause # 3: Wrong selection of two-way valves:

An improperly sized two-way control valve may consume more water flow when open than the design calls for. With full flow through the coil, at partial loads the Δ-T will invariably be lower than design.

Also the oversized or worn two-way valves may act as three-way valves which can lead to "hunting". This means that the valves alternately open and close causing over and under-shooting of the set point. This results in very unstable operation.

Another problem could be undersized actuators with an insufficient close-off pressure rating which can cause the valve to open when that is not intended.

Measure: Don't oversize or undersize 2-way valves

Select the control valves considering the pressure drop of the load served and the available differential across the supply and return mains. *Manufacturers usually recommend that wide-open control valve pressure drop shall be equal to or greater than*

the pressure drop of the coil plus the pipe and fittings connecting them to supply and return mains. (Refer to control valves and actuators description later on this course for detailed description).

Cause # 4: Improper Coil Selection:

Cooling coil selected for a lower Δ-T than the chiller Δ-T is a problem.

Existing coil rows, fin spacing, tube velocity, and circuiting should be investigated to determine any impact (potential reduction of capacity) to the existing HVAC system.

Measure: Select cooling coils for high temperature drops at design conditions

If chiller is selected for ΔT of 12°F, the cooling coils should be selected for ΔT of minimum 12°F or higher say 14°F.

The most effective method of increasing the leaving water temperature (LWT) of a cooling coil is to lower its entering water temperature (EWT). If the EWT to the coil is lowered from the 42°F original design to 38°F, the ΔT should increase from 10°F to 14°F with virtually no impact on the coil capacity.

The good design may be to use "dual-row coils" that is piped with the water entering at two coil locations (last row and middle row), rather a standard "full row" coil where the water enters only into the last row of the coil (the row at the air discharge side of the coil). The dual-row coil has lower tube velocity and, hence, lower water-side pressure drop than the standard coil. It is recommended to use dual-row coils on the hydraulically longest runs to reduce pump head, then compensate by using higher delta-T coils on circuits closer to the pump where excess head is available.

Cause # 5: Reduced coil effectiveness:

The performance (or lack of) of the existing cooling coils and control valves creates the low ΔT problem. Coil transfer effectiveness is reduced over a period of time by water side fouling (slime, scale or corrosion), air side fouling (dirt build-up) and coil bypass air. Any

reduction in coil effectiveness increases the flow rate of water required to deliver the desired leaving water temperature, thus reducing delta-T.

Measure: Improve housekeeping

Waterside fouling is easily controlled by proper water treatment. Since the chilled water piping is most often a closed system, water treatment need not be an ongoing expense.

Reduced airside fouling is a very good reason to consider increasing the filter efficiencies.

Cause # 6: Improperly Piped Coils:

Coils piped in parallel flow arrangement with water entering the coil on the same side as the entering air.

Measure: Coils shall be arranged in counter-flow

Coils must be piped counter-flow. Refer to figure below:
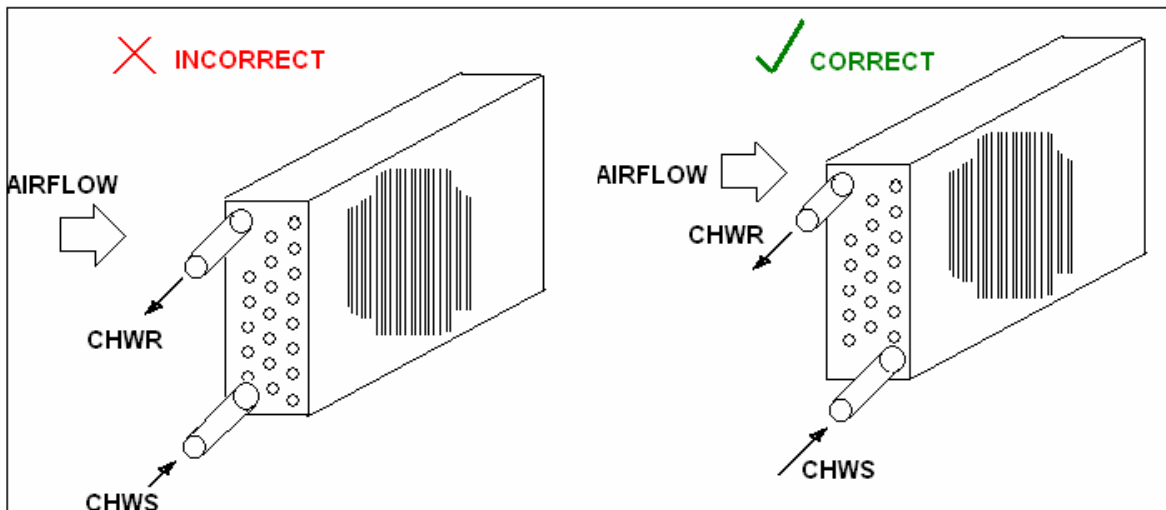
The chilled water supply shall be from bottom and shall be on the opposite side to the direction of airflow.

A coil piped counter-flow can achieve "overlapping" temperature ranges with the supply air, e.g., the leaving water temperature can enter at 44°F and leave at 60°F while the supply air enters at 80°F and leaves at 55°F. With parallel-flow piping, the two ranges cannot overlap and therefore leaving water temperature will always be a few degrees cooler than the leaving supply air temperature.

Cause # 7: Dirty Filters:

Reduction in air flow is caused by dirty filters. This reduces air-side heat transfer coefficients and reduces the overall space cooling capacity of the system. To compensate for the low airflow, the space thermostat causes the chilled water valve to increase water flow rates to deliver colder supply air temperatures, thus reducing delta-T.

Measure: Proper Housekeeping

Air-side fouling is usually minimized by good filtration. Filters should be at least 30% dust-spot efficiency (or MERV 6 or 7 using the new ASHRAE 52.2 rating method).

Although the design should take mitigation measures to prevent low ΔT, experience indicates it is unavoidable primarily due to cause 5 and 7.

PRIMARY/SECONDARY DESIGN PRACTICE

There are three critical design areas that must be considered for any variable volume pumping system:

1. The common pipe (De-coupler)

2. Chiller staging

3. Control valves and actuators

DE-COUPLER

Referring to primary / secondary schematic, it would appear that the primary and secondary pumps are in series. This is not the case however, because of the de-coupler.

Primary-secondary pumping is based on a simple fact that when two circuits are interconnected, flow in one will NOT cause flow in the other if the pressure drop in the piping common to both is eliminated. To ensure proper system performance, the de-coupler sizing criteria is as follows:

De-coupler Sizing

De-couplers should be sized for the flow rate of the largest primary pump. For simplicity of design and installation, the common pipe is often the same diameter as the distribution piping.

<u>The maximum pressure drop in the common pipe shall not exceed 1.5 ft.</u>

By restricting the pressure drop to 1.5 ft maximum, water that is flowing in the primary loop will not flow into the secondary circuit until its circulator turns on. That way the hydraulic isolation between both primary and secondary circuit is achieved and is the fundamental basis of primary-secondary pumping. [Note that the higher friction loss in the common tends to make the primary and secondary pumps act in series, resulting in an induced flow in the system].

<u>Maximum three pipe diameters of separation between the secondary supply tee and the secondary return tee.</u>

A common pipe of 3 pipe diameter apart is more than adequate to eliminate mixing due to excessive return velocity in the secondary return piping. Longer length common pipes may result in an excessive pressure drop greater than 1.5 ft with results as described above.

<u>Keep minimum 10 pipe diameters to the first chiller</u>

When the secondary return flows straight through in the tee to the primary return, there should be at least 10 pipe diameters to the first chiller. This is to help avoid the possibility of having stratification in the primary return line, which can lead to unmixed water to the first chiller. This can lead to chiller cycling.
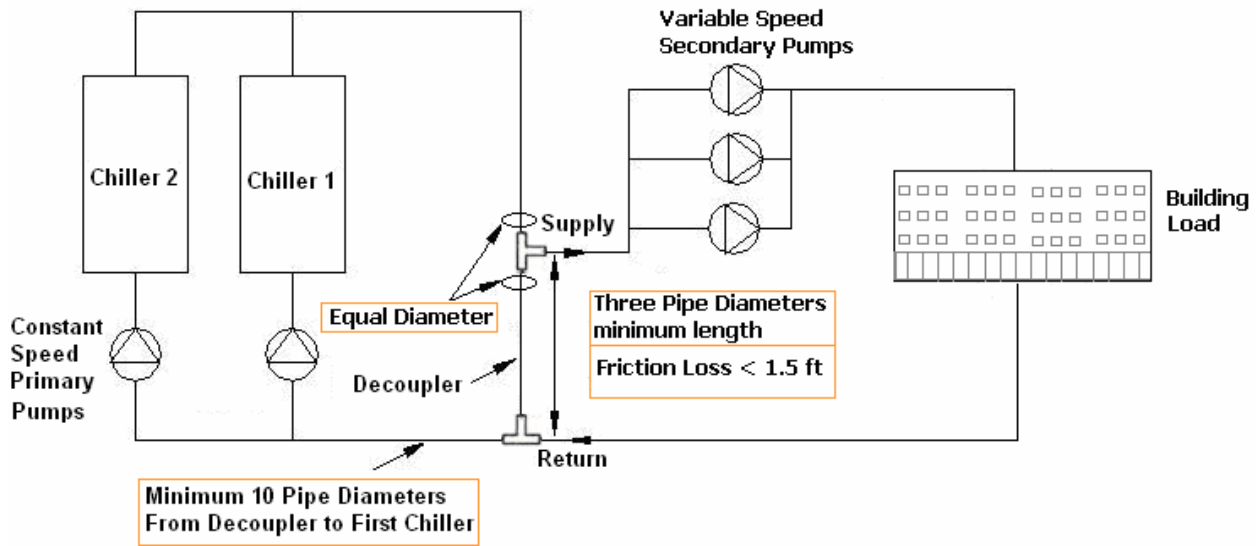
CHILLER SEQUENCING

Proper chiller sequencing plays an important role in the overall performance of a primary-secondary pumping systems. The operator of the system must recognize the three flow conditions that can occur. These are:

1.   Primary flow equals secondary flow

2.   Secondary flow is greater than primary flow

3.   Primary flow is greater than secondary flow

Primary flow equals secondary flow

Only on special occasions will the primary pump flow and the secondary flow be equal. An example would be when the cooling load (and secondary flow) can be met by a fully loaded chiller. Although this flow condition rarely occurs, we will observe the flow pattern of this scenario (Fig. below).
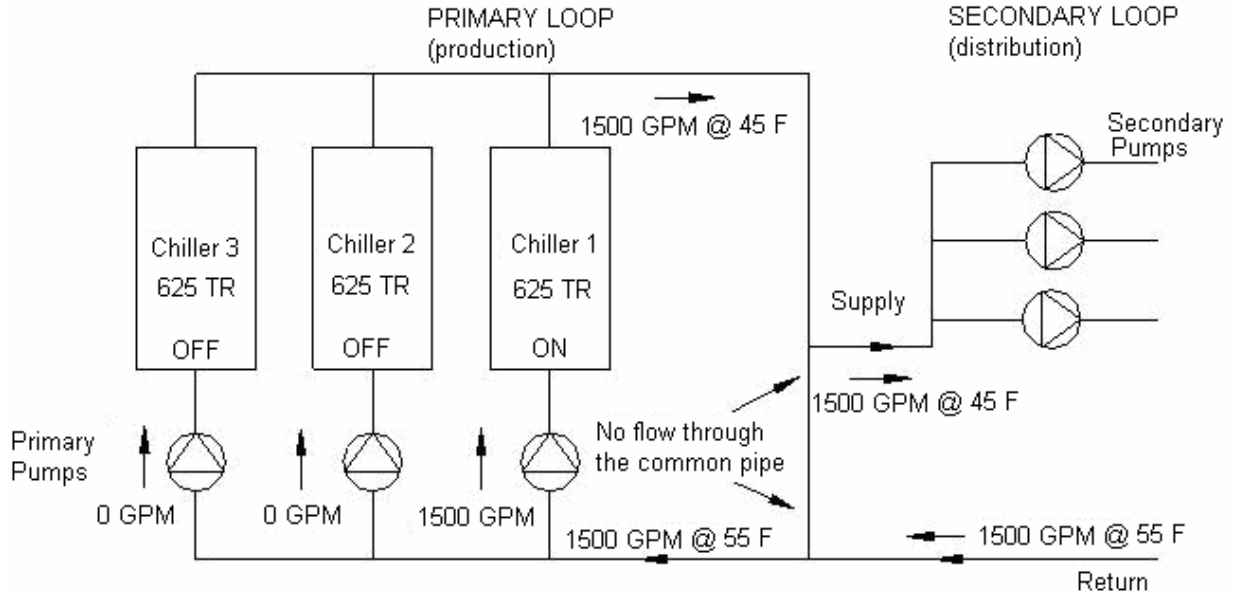
Primary Flow Equals Secondary Flow

The chiller is supplying 1500 GPM of 45°F water to the load systems.

The two way valves in the secondary are responding to conditions equal to 1500 GPM x 10°F ΔT x 500 = 7.5 MBtuh or 625 tons.

Because the load is equal to 625 tons, the return water temperature to the chiller is 55°F at a flow rate of 1500 GPM. The thermal balance is complete. There is no flow in the common pipe.

Secondary flow greater than primary flow

The amount of time the system operates at peak is extremely small compared to the hours per year it operates at part load. Therefore, the primary/secondary loop flow will never be exactly balanced and there will always be some flow in de-coupler line. Two conditions are possible: 1) the secondary flow greater than primary flow or 2) the primary flow greater than secondary.

Continuing from previous example, consider now the distribution side load increases and the two-way valves in secondary circuit modulate open in response, to increase the flow rate to 2000 GPM. The one chiller that is operating will accept only 1500 GPM and to balance the mass flow, the excess 500 GPM must run through the common pipe. (Refer figure below)
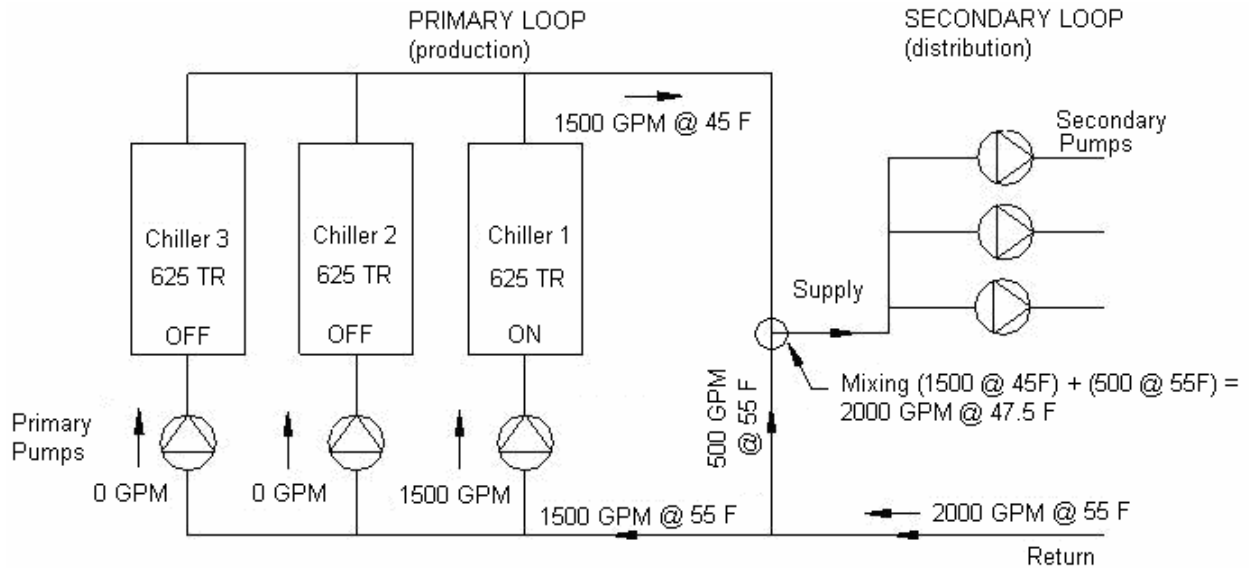
Secondary Flow Greater Than Primary Flow

Look also at the temperature relationships. The temperature of the 500 GPM in the common pipe is 55°F. This blends with the 1500 GPM of 45°F supply water, resulting in 2000 GPM of 47.5°F blended supply water. _A higher secondary supply water temperature to coils hinders its ability to provide dehumidification that could mean a loss of humidity control in the secondary zones._ A common description of this situation from operators is that "the chiller can't handle the load". To deal with this situation, they will immediately start additional chiller that will result in primary flows greater than the secondary.

This requirement leads to a simple design rule: _"primary circuit flow should equal or slightly exceed secondary circuit design flow rate"._ In short, flow in the reverse primary direction in a de-coupler bridge is not good.

Other options include:

Secondary flow is permitted to increase the primary flow if following options are considered:

Option #1 – Select cooling coils on a somewhat higher than design chilled water supply temperature; for example 1°F to 2°F higher inlet temperature to cooling coil is prudent to account for coil fouling over time.

Option #2 – Reset chiller temperature. Within the limits of the type of machine, chiller temperatures can be reset to a lower temperature to compensate for the increased load and secondary flows. In essence, more capacity is provided at a lower operating efficiency. The increase in cost of chiller operation due to the lowering of the chiller supply temperature can range from 1 to 3 percent per degree of reset. This is a very desirable alterative, especially when large chillers are in use. The longer the start of a lag chiller can be delayed, the better it will perform when it is finally brought on line.

Option #3 - Use smaller chiller in series, if a small portion of the load requires a fixed temperature.

Option #4 – Some designers opt for putting a check valve in the de-coupler line that prevents flow in reverse primary direction. Engineers who design chilled water distribution systems have different opinions on this approach. The advantages and disadvantages are highlighted below:

Check Valve Advantages:

- The check valve allows the secondary pumps to "help" pump the primary loop and push additional flow through chiller evaporators above their design rate. This additional flow will force load the chiller to 100% during a situation when the chiller ΔT is less than design, or greater than 100% if this condition coincides with low outdoor wet-bulb temperatures.

- A check valve helps to reduce plant energy consumption by keeping the quantity of active chillers matched to the system load. Additional chiller is not brought on-line simply to provide additional primary water flow.

- A check valve prohibits secondary return water at high temperature to the cooling coils, thus the dehumidification at the secondary coils is not impacted.
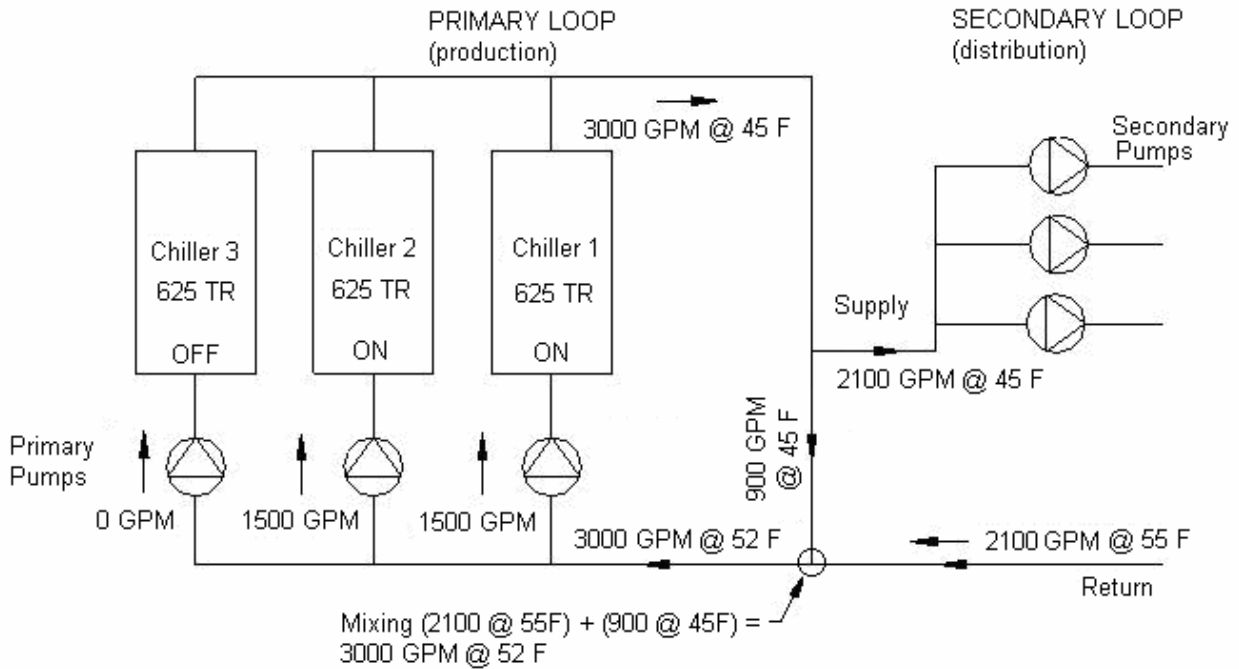
Check Valve Disadvantages:

- A check valve forces the secondary pumps in series with the primary pumps when secondary pump speed and flow is increasing beyond the capacity of the primary pumps. Assisted by the secondary pumps, the constant flow primary pumps move to the right on their characteristic curves to accommodate the increased flow and operate at a head lower than design. This leads to higher chiller discharge temperatures and chiller tube erosion.

- There is possibility of potential dead-head condition of the secondary pumps if no primary pumps are operating. During this condition, the secondary pump speed escalates in response to the differential pressure sensor out in the system, which senses zero differential because no secondary flow exists. One approach to resolving this problem is to require proof of operation of at least one primary pump for any secondary pump to operate. Upon loss of proof of operation of all primary pumps, the secondary pumps stop.

- There is a possibility of ghost flow through inactive chillers with dedicated pumps

The author's experience is to be without a check valve.

Primary flow greater than secondary flow

Continuing from same example, let's say the operator turn-on additional chiller to keep primary flow greater than secondary flow. (Refer Fig below).

PRIMARY LOOP (production)

SECONDARY LOOP (distribution)

3000 GPM @ 45 F

Secondary Pumps

Chiller 3 625 TR OFF

Chiller 2 625 TR ON

Chiller 1 625 TR ON

Supply
2100 GPM @ 45 F

Primary Pumps

0 GPM   1500 GPM   1500 GPM

900 GPM @ 45 F

3000 GPM @ 52 F

2100 GPM @ 55 F

Return

Mixing (2100 @ 55F) + (900 @ 45F) = 3000 GPM @ 52 F

Primary Flow Greater Than Secondary Flow

The flow rate through two chillers is fixed, this time at 3000 GPM. The new secondary load is say 875 tons, which corresponds to demand rate of 2100 GPM @ 10 º F delta -T across the cooling coils. There will be excess flow of 900 GPM in de-coupler as 2100 GPM circulates in the secondary loop and 3000 GPM is being pumped into the primary loop.

Look at temperature relationships. The 900 GPM common flow @45ºF blends with the 2100 GPM @ 55°F to produce 3000 GPM @ 52ºF. This is lower than the desired return water temperature of 55°F, which makes it impossible to fully load the "on-line" chillers and robs the plant of its rated capacity…..low ΔT syndrome.

CHILLER STAGING

Maintaining the leaving-chilled-water-temperature set point is the primary objective of chiller staging. At small part loads, the auxiliary power (cooling tower fans, condenser water pumps, and primary chilled water pumps) can become a very large fraction of the

total energy consumption of the system; consequently, most chiller staging strategies take into consideration the total of chiller and auxiliary power.

Chiller staging strategies require comparison of chiller capacity with cooling load. This can be done in a variety of ways.

1. Chilled water temperature leaving the evaporator indicates a capacity shortfall when it remains above set point for a specified length of time, but is not a good indicator of excess capacity.

2. In a primary/secondary system, bypass flow direction (indicated by temperature in the bypass) can indicate a capacity shortfall.

3. Calculation of load from measurements of flow and temperature can be useful both for deciding when to start or stop chillers

4. Measurement of current drawn by the compressor motor is also useful for controlling chiller starts and stops.

5. Models of chiller and chilled water plant operation can, in principle, be used off-line or in real time to support optimal control strategies. This is not currently the norm, but may be the future direction of chilled water plant controls.

Conceptually, the designer needs to determine the best combination of chillers that will meet the flow demand. This is often done by calculating the BTU/h consumed. Instrumentation must be provided to determine:

Secondary supply water temperature, T_2

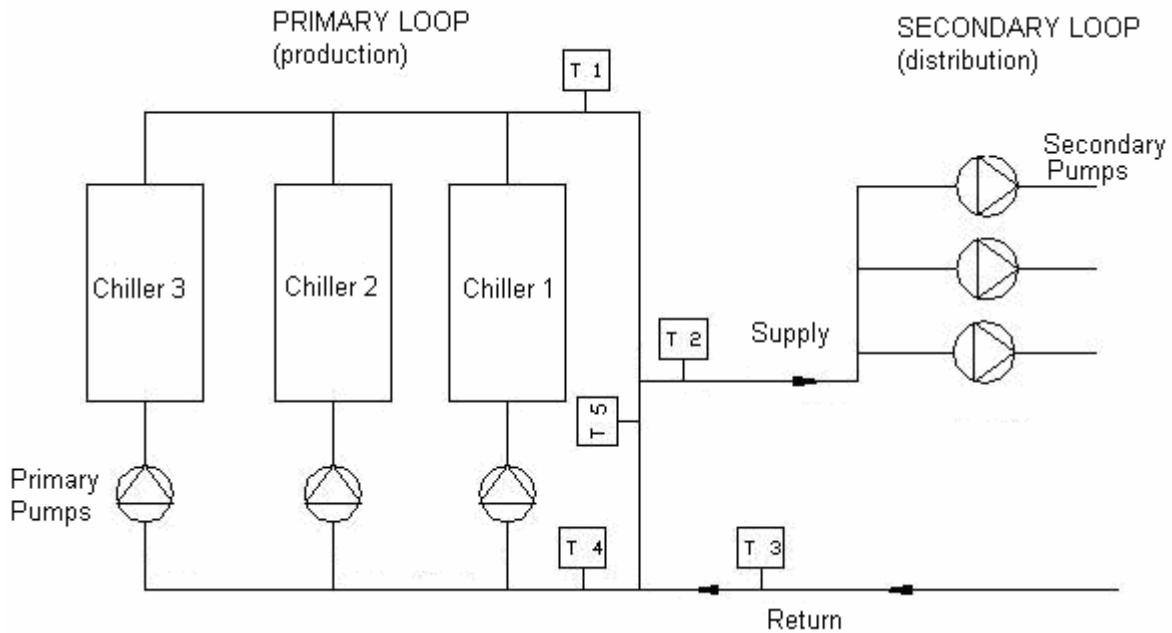Secondary return water temperature, T_3

Secondary flow, F_S

The amount of cooling BTU/h produced is also valuable information. The necessary instrumentation is:

Primary supply water temperature (chiller leaving), $\boxed{\text{T 1}}$

Primary return water temperature (chiller entering), $\boxed{\text{T 4}}$

Primary flow (chiller flow), F_P

How does one know when to take a chiller off-line?

The traditional primary-secondary logic says to measure the flow and when primary flow exceeds secondary flow by the capacity of one primary pump, shut down a pump and chiller. In actuality, systems with low ΔT never see this condition. Shutting down a primary pump will often lead to re-circulating flow through the de- coupler.

How is bypass flow determined?

This can be done relatively simply using temperature measurements at the bypass $\boxed{\text{T 5}}$, the return temperature $\boxed{\text{T 3}}$, and mixed return temperature $\boxed{\text{T 4}}$. Since the chilled-water-flow rate in the primary loop is known to be a good approximation, the flow rate through the

bypass can be calculated using these temperatures. Alternatively, a flow meter in the bypass line can be used to measure the excess primary flow directly.

The simplest indicator of available chiller capacity is percent full load motor amperage, not because it's such a straightforward mea sure of available chiller capacity but because it's generally available on the chiller starter panel.

General Practice

In majority of plants, chiller capacity is staged in response to rising supply chilled-water temperature $\boxed{T\ 1}$ and the chiller is staged off when flow in the bypass exceeds the design flow of one chiller. When the on-line chillers can't hold the chilled water temperature $\boxed{T\ 1}$ set point any longer, the chillers are fully loaded and another chiller should be turned on. With this method, chiller activation is controlled by monitoring the chiller's ability to achieve leaving chilled water set point temperature, not by monitoring chilled water flow through a crossover pipe.

Recommended Practice

Instead of staging in response to rising chilled water temperature $\boxed{T\ 1}$, consider controlling the chiller through $\boxed{T\ 2}$ sensor. As the load increases and more mixing occurs at the supply side of the de-coupler, the secondary temperature will rise further. As soon as this is detected by $\boxed{T\ 2}$ sensor, the chiller supply temperature $\boxed{T\ 1}$ is reset (lowered) automatically via building energy management system (BEMS) to maintain the secondary supply setpoint. This means that during some part of the day the chiller discharge temperature could be lower than the secondary controlled setpoint. When the true maximum capacity (100%) for the chiller is reached, it can do no more work. At this point, the secondary chilled water temperature $\boxed{T\ 2}$ will begin to rise above the desired setpoint. When the secondary actual temperature is 1°F above the secondary setpoint for five minutes, the BEMS will start the next chiller.

[Note that the on-board chiller control panel is responsible for maintaining the temperature leaving the chiller machine (45°F in our example above) and is NOT aware of the temperature in the secondary].

The chiller manufacturer is usually best suited to provide the information and instrumentation to effectively stage and de-stage its product.

CONTROL VALVES & ACTUATORS

In a (cross flow) cooling coil, 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%).
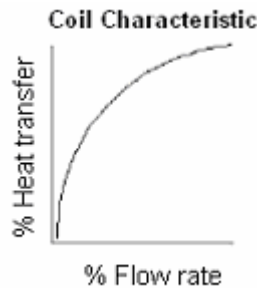
It is clear that the coil requires a flow that falls off steeply to begin with as the load falls from 100% load (valve in fully open position) but a flow that declines slowly as the valve nears its fully closed position. To achieve this, control valves are used, which closes the flow 'fast' to begin with and only 'gradually' later as the capacity approaches zero.

The control valves functions are to vary flow properly through the water coil in response to a variety of building load conditions. Because of their critical nature, great care must be taken when selecting control valves so they perform properly.
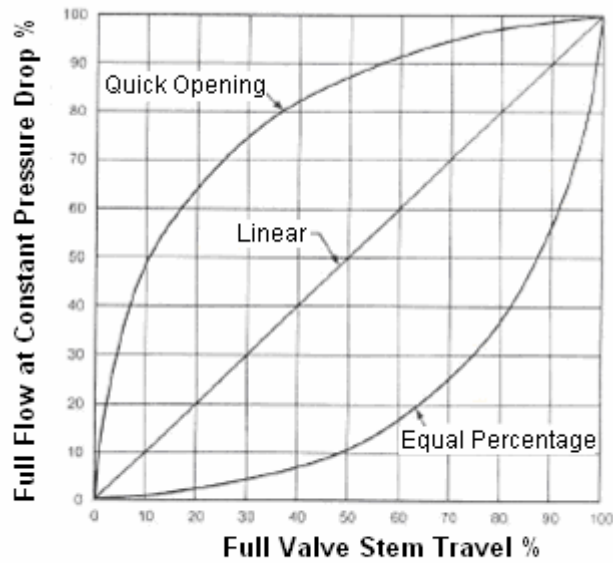
Control Valve Flow Characteristics:

Generally, valves control the flow of fluids by an actuator, which moves a stem with an attached plug. The plug seats within the valve port and against t he valve seat with a composition disk or metal-to-metal seating. Based on the geometry of the plug, three distinct flow conditions can be developed.

Quick Opening: When started from the closed position, a quick opening valve allows a considerable amount of flow to p ass for small stem travel. As the stem moves towards the open position, the rate at which the flow is increased per movement of the stem is reduced in a nonlinear fashion. This characteristic is used in two-position or on/off applications.

Linear: Linear valves produce equal flow increments per equal stem travel throughout the travel range of the stem. T his characteristic is used on steam coil terminals and in the bypass port of three-way valves.

Equal percentage: This type of valve produces an exponential flow increase as the stem moves from the closed position to the open. The term equal percentage means that for equal increments of stem travel, the flow increases by an equal percentage. For example, in Figure below, if the valve is moved from 50 to 70% of full stroke, the percentage of full flow changes from 10 to 25%, an increase of 150%. Then, if the valve is moved from 80 to 100% of full stroke, the percentage of full flow changes from 40 to 100%, again, an increase of 150%. This characteristic is recommended for control on (hot and) chilled water terminals.

CONTROL VALVE FLOW

The combined coil heat transfer characteristics for a cooling coil with a equal percentage valve are shown in figures below:
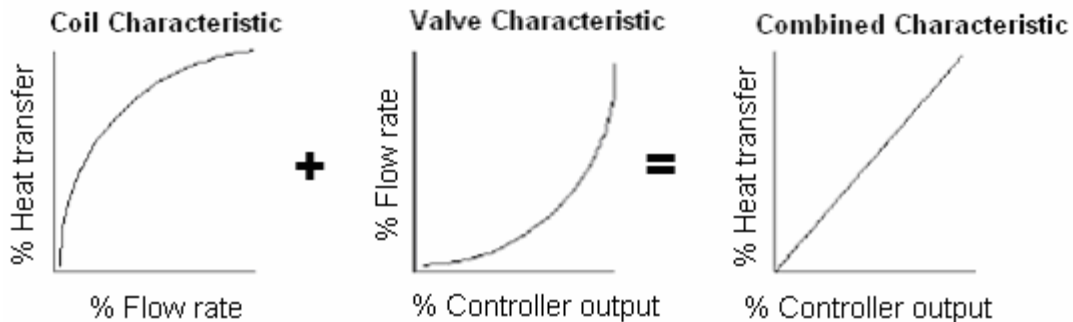
Extreme left shows the coil characteristic and middle shows the valve characteristic. As already discussed, the former depicts the inadequate sensitivity of coil capacity for flow reduction to begin with i.e., as the valve starts closing and an improvement as the valve approaches the fully closed position.

Control Valve Sizing/Selection

A pressure drop must exist across a control valve, if flow is to occur. The greater the drop, the greater the flow at any fixed opening. The pressure drop across a valve also varies

with the disc position – from minimum when fully open, to 100% of the system drop when fully closed. To size a valve properly, it is necessary to know the full flow pressure drop across it. The pressure drop across a valve is the difference in pressure between the inlet and outlet under flow conditions. When it is specified by the engineer and the required flow is known, the selection of a valve is simplified. When this pressure drop is not known, it must be computed or assumed.

If the pressure drop across the valve when fully open is not a large enough percentage of the total system drop, there will be little change in fluid flow until the valve actually closes, forcing the valve's characteristic toward a quick opening form.

It is important to realize that the flow characteristic for any particular valve, such as the linear characteristic is applicable only if the pressure drop remains nearly constant across the valve for full stem travel. In most systems, however, it is impractical to take 100% of the system drop across the valve.

A good working rule is, "at maximum flow, 25 to 50% of the total system pressure drop should be absorbed by the control valve." Although this generally results in larger pump sizes, it should be pointed out that the initial equipment cost is offset by a reduction in control valve size, and results in improved controllability of the system.

Reasonably good control can be accomplished with pressure drops of 15 to 30% of total system pressures. A drop of 15% can be used if the variation in flow is small.

Recommended Pressure Drops for Valve Sizing — Water

With a differential pressure less than 20 psig, use a pressure drop equal to 5 psi.

With a differential pressure greater than 20 psig, use a pressure drop equal to 25% of total system pressure drop (maximum pump head), but not exceeding the maximum rating of the valve.

$$C_v = Q\sqrt{\frac{G}{\Delta P}}$$

Where

- Cv = Valve flow coefficient

- Q = U.S. GPM with ΔP = 1 psi

- S =Specific gravity of fluid relative to water @ 60°F

- ΔP = P1 — P2 at maximum flow, psi

- P1 Inlet pressure at maximum flow, psia (abs.)

- P2 Outlet pressure at maximum flow, psia (abs.)

Valve Authority: 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:

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

This is called "Valve Authority". Its value should be 0.5 minimum.

Rangeability: The "Rangeability" of the valve (the ratio of flow at fully open position and minimum flow of 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.

Two aspects of two-way control valve selection are particularly critical for variable-flow systems: valve size (Cv) and actuator size (shutoff capability).

Proper sizing of control valves is critical in a two-way valve system, particularly when pumps are uncontrolled (riding their curves). Many designers feel that oversizing control valves is no longer a concern in modern systems with direct digital controls using PID control loops and variable speed drives to control system pressure. This is certainly partly true, but no amount of control magic can compensate for a grossly oversized valve. Oversized control valves cause the controller to "hunt," alternately opening and closing the

valve, over- and under shooting the setpoint. The overall average flow is higher than desired, and thus delta-T is reduced.

Undersizing the actuator is another common problem. Valve/actuator combinations usually have two ratings: the close-off rating is the maximum differential pressure across the valve against which the valve and actuator can completely close. The dynamic close-off pressure rating is the maximum differential pressure for modulating (as opposed to two-position) applications; above this pressure, control through the entire stroke will no longer be smooth and the design turndown ratio will not be achieved. Often valves are selected only to achieve the required close-off rating, but they cannot provide modulating duty under high differential pressures. The result is hunting and excess water being forced through the coil. Thus the undersized valves may provide insufficient capacity while oversized valves provide poor control.

In variable volume systems, the selected valve actuator must be capable of closing the valve against the maximum pump head pressure. In variable volume closed loop systems, all loads could be reduced causing the pump to ride back on the pump curve; as the pump rides back on the curve, the head being produced by the pump increases. The control valve actuator must be strong enough to continue modulating the valve closed as the pressure increases. In the worst case, this could be the "shutoff pressure" being produced by the pump. Improper actuator sizing may result in some water passing through the valve and coil, which add to low return water temperatures.

ADVANTAGES OF PRIMARY / SECONDARY SYSTEM

The following is a list of the advantages to implementing a primary-secondary system:

1. Constant Flow through Evaporator: The primary-secondary system maintains a constant flow through the evaporator of the chiller even though secondary flow varies. This eliminates the concern for chiller performance and inadvertent shutdowns. Thus, there's no possibility of freezing the chiller's evaporator in an upset condition or allowing evaporator CHW flow to slip into the laminar flow region at low loads. As long as plant primary flow exceeds system flow, chiller pumps and system pumps behave

as if they're decoupled.

2. <u>Simplified Controls</u>: Controls within a primary-secondary system are relatively simple and well established. Typical chiller controls packages do not have difficulty with the staging sequence for the chillers and responding to varying loads.

3. <u>Past Experience</u>: The primary-secondary system is a well established operational philosophy and plant personnel are familiar with its operation. In addition, this pumping scheme has been proven reliable if operated properly.

4. <u>Divided Hydraulic Head</u>: By dividing the total dynamic head between two hydraulically independent loops, the required motor size for each pump type (primary and secondary) will be smaller when compared to a direct-primary system. This also reduces the risk that the system discharge pressure will exceed the design of equipment, piping, and valves in the buildings.

DISADVANTAGES OF PRIMARY / SECONDARY SYSTEM

The following is a list of the disadvantages to implementing a primary-secondary system:

1. <u>Does not resolve Low ΔT Syndrome</u>: The primary-secondary system does not allow an increase in flow through the evaporator above design and; therefore, does not adjust to chilled water return temperatures that are lower than design. In addition, this pumping scheme can further exacerbate the problem during off-peak conditions.

   As the cooling load decreases, the secondary pump VFDs will ramp down to a lower speed, thus allowing these pumps to produce less flow. The constant volume circulation pumps will then over pump the primary loop causing supply water to flow through the neutral bridge and mix with return water. This mixing lowers the return water temperature and deteriorates the system ΔT as described previously.

2. <u>Capital Investment</u>: The greater quantity of pumps and the longer piping runs associated with this pumping scheme can yield a higher capital investment when compared to the direct-primary system.

3. <u>Higher Operating and Energy Costs</u>: The primary-secondary system uses both constant speed and variable speed pumps to circulate chilled water through the plant as well as the distribution system. Because the primary loop will always have a constant flow, energy is wasted within this loop at off-peak loads. In addition, this pumping scheme does not allow adjustable flow through the chillers and is subject to the part load operational inefficiency described above. These features and the need for two sets of pumps will generally yield higher energy and operating costs per annum when compared to the direct-primary system.

4. <u>Requires More Plant Space</u>: Two sets of pumps are needed to circulate chilled water through the chiller evaporator and the chilled water distribution system. This requires more floor space, more spare parts, and results in higher capital costs and pump maintenance costs when compared to the direct-primary system.

SCHEME # 3   VARIABLE PRIMARY FLOW SCHEME

In a variable primary flow (VPF) system, chilled water flow is <u>allowed to vary throughout the loop, including the chiller evaporators.</u> In this system, the secondary pumps are eliminated, the primary pumps provide variable flow to supply system demand to the extent tolerated by the chillers, and the decoupling bypass of the primary/secondary system has been replaced by a bypass with a normally closed control valve that opens only to maintain minimum flow through active chillers.

The function of the bypass line VPF arrangement should not be confused with that of the de-coupler of primary/secondary arrangement. The bypass in this case is a <u>smaller pipe</u> sized for the minimum flow of the largest chiller. It contains a normally closed control valve that modulates open only when the low flow limit is reached.
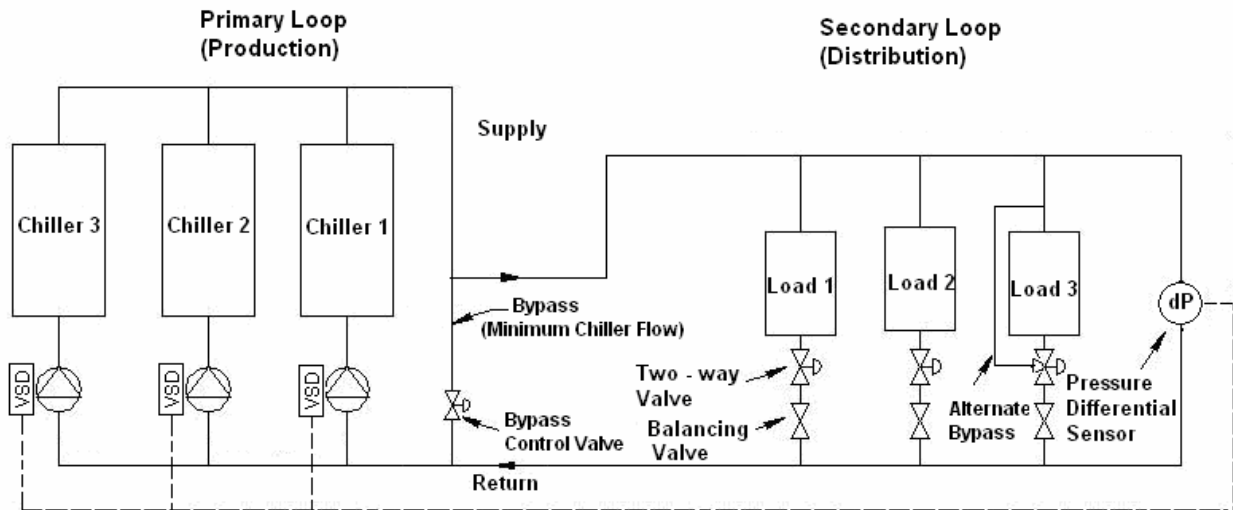
Variable Primary Flow (VPF) System

Like the variable secondary pumping in a primary–secondary system, the pumps in a typical VPF system operate to maintain a target differential pressure (Delta P) at a specific point in the system. This pressure difference tends to decrease when the terminals (air-handlers or fan-coils) two-way control valves open in response to increasing loads.

To restore the Delta P across the system, the pump controller increases the speed of the pump. Conversely, when the terminals control valves close in response to decreased coil loads, the pump controller slows the pump speed to maintain the target Delta P. Meanwhile, the plant controller stages the chillers on and off to match cooling capacity with system load. The pumps are normally sized to deliver 120% of the design flow. If the return water temperature is high (or ΔT is low) the variable speed pump increases the flow rate to the chillers thus increasing the load on the chillers.

Advantages

The following is a list of the advantages of VPF system:

1. VPF systems are not prone to low ΔT syndrome:

    •   *Neutral Bridge:* The VPF systems do not include two hydraulically independent

loops that are separated by a neutral bridge. As a result, the pump flow rate is better matched to the cooling load within the distribution system and the absence of a neutral bridge prevents mixing of supply and return water.

- *Adjustable Chiller Flow:* VPF controls permit flow through evaporators to be increased above design value, making it possible to adjust to less than ideal chilled water return temperature. This maximizes the output of a given chiller and eliminates the need to start additional chillers and pumps prior to reaching nameplate capacity. The combination of using less equipment more efficiently yields savings to the owner/operator.

2. VPF provides enhanced capacity: When the outside wet bulb temperature is below the design value, as it is over 95 percent of the year, then cooler tower water will depress the chiller condensing temperature, giving each chiller additional capacity. At a 65°F entering condenser water temperature, for example, chiller full load capacity will increase at least 7 percent due to refrigerant cycle considerations and up to another 10 percent depending on where the compressor was selected on its compressor curve. If extra chilled water can be put through the chiller, this extra capacity can be tapped.

3. Capital Investment: A VPF design uses fewer pumps and fewer piping connections than primary–secondary systems, which means fewer electrical lines and a smaller footprint for the plant. These factors reduce the initial cost of the chilled water system.

4. Lower Operating and Energy Costs:

- Energy savings are possible because no excess flow recirculates from supply to return through decoupling lines or three-way valves. In theory, every bit of supply water, without any mixing, must pass through a load before returning to the plant.

- A VPF design displaces the small, inefficient, low-head primary pumps used in primary–secondary systems allowing selection of larger and far more efficient pumps.

- Energy savings are also possible when conditions permit flow to one or more chillers to exceed design flow. If outside wet bulb temperature is below the design value, as it is over 95 percent of the year, the condensing temperature will be also lower, giving each chiller additional capacity. If more water can be put through the chiller, this extra capacity can be tapped.

5. Requires Less Plant Space: Constant flow pumps serving a production loop are not needed because the primary pumps circulate the water through the chillers as well as the distribution system. This requires less floor space, fewer spare parts, and can result in lower capital costs and pump maintenance costs.

The Air-Conditioning and Refrigeration Technology Institute (ARTI) has conducted an extensive study that compared VPF system energy use with that of other common system types including: constant flow/primary-only chilled water systems; constant primary flow/variable secondary flow chilled water systems; and primary/secondary chilled water systems with a check valve installed in the de-coupler. According to the ARTI study results, VPF systems reduced the total annual plant energy by 3 to 8 percent, first cost by 4 to 8 percent, and life cycle cost by 3 to 5 percent relative to conventional constant primary flow/variable secondary flow chilled water systems.

DISTRIBUTED OR HEADER ARRANGEMENT OF PUMPS

With distributed (tandem) disposition, flow through operating chillers will always change abruptly even though the pump has a VSD because no flow will go through the starting pump or chiller until the pressure at the pump discharge exceeds the backpressure on its check valve caused by pumps serving the operating chillers. At that point, the check valve will suddenly open and flow will abruptly change through the starting pump and chiller, causing an abrupt change in flow through the operating chillers.

VPF chillers shall be piped to a common chilled water header with provisions to sequence chillers on-line to match the load requirements. The preferred scheme used in VPF systems separates pump control from chiller control (Fig. below). It means the water distribution and circulation functions could be treated apart. A first appealing shape of

such scheme is the prospective of implementing a flow supply control based on only one VSD, which allows some initial budget reductions. However, an isolation valve must be installed on each chiller inlet to avoid the flow circulation through an "off-line" chiller and also to modulate the flow through both during sequencing.
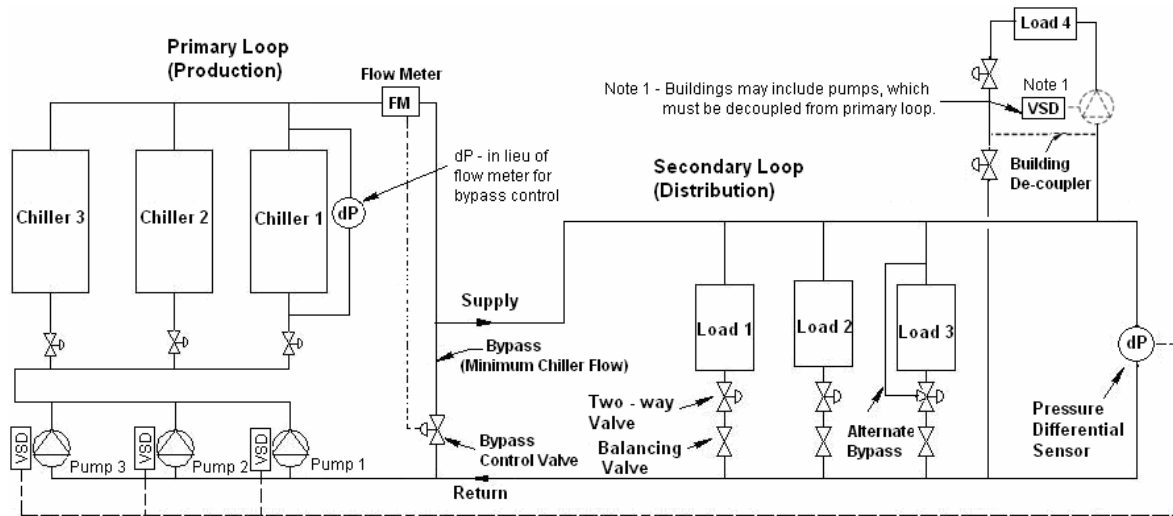
VPF scheme with separated pump control and chiller sequencing

VARIABLE PRIMARY FLOW DESIGN PRACTICE

For many years, chiller manufacturer's encouraged HVAC engineers and cooling-plant operators to keep constant water flow trough the chiller evaporator. The overriding concern was one of protection since reducing water flow faster than the chiller capacity control can accept could result in knocking it off line, requiring a manual reset to restart it.

Design Challenges

Varying the water-flow rate through the chiller evaporator poses two control challenges:

Challenge #1:

VPF system design is constrained by the range of flow rate (tube velocities) permitted in the evaporator. While chiller manufacturers have been allowing adjustable flow through the chillers, upper and lower limits have been placed on the flow rates.

The typical limits are:

- Low velocity limits of 3 ft/s to prevent laminar flow from occurring and also to keep tubes clean.

- High velocity limits 10 ft/s to avoid tube erosion.

The design tube velocity determines the range of flow that is permitted as a percentage of design flow rate.

<u>Measure</u>

One benefit of VPF systems is reduced pumping energy. The velocity limitations on chillers pose restrictions on the energy savings the chiller pumps can achieve.

It is important to select chillers that allow operation at low loads without bypass flow. Most of the potential savings are realized by the time that the system flow rate decreases to 50 percent of design. The manufacturer's recommend a minimum evaporator-flow limit of 60 percent for packaged chillers and 40 percent or less for configured chillers.

Challenge #2:

When plant operators desire to bring additional chillers online, the sudden drop in flow through the lead chiller may cause it to trip offline on low flow. Consider a plant comprised of two identical parallel chillers with one chiller on line and fully loaded while the other is on standby with no flow through its evaporator. System load increases slightly causing leaving-chilled-water-temperature to rise and the plant controls bring the second chiller on line. This decreases the flow through the first chiller and if the decrease occurs rapidly due to fast opening of the isolation valve on the second chiller, the first chiller will be placed in a condition of full refrigerant load at half chilled water flow, which effectively doubles the chilled water temperature difference. The chilled water temperature could plunge below the low limit before chiller controls are able to adapt to the new operating conditions forcing the chiller into a protective shutdown.

<u>Measure</u>

- Select chillers and controllers suitable for large rate of chilled water flow variation.

- Select slow acting isolation valves with linear characteristics.

This is discussed further in design considerations below.

Design Considerations

Control logic and sequencing for the pumps and chillers is the heart of this type of system.

Chillers

Multiple chillers are required to maximize the turndown capacity of the chilled water plant. With only one chiller, the low-flow range is limited to the minimum flow of the single chiller. Multiple chillers lower the minimum flow limitation. The designer must know the minimum flow requirements for each chiller.

<u>Select chillers that have (nearly) equal pressure drops at their design flow rates.</u>

Primarily, a VPF system loads and unloads the chiller(s) based on the rate of water flow through the evaporator. If a difference in size or type of evaporator gives one chiller a lower pressure drop than the others in the plant, that chiller will receive a higher rate of water flow and a correspondingly greater load. Dissimilar pressure drops can make it difficult to provide stable plant operation.

Rate of Chilled Water Flow Variation

Rapid flow variations in chiller can cause control instability and compressor flood-back or shutdown. A good chiller must able to withstand large flow variations.

Typically, manufacturers limit this rate of change from 2 percent per minute to less than 30 percent per minute depending on the chiller type, controls, and system turnover time. Vapor compression chillers equipped with continuous capacity control (e.g., water-cooled

centrifugal chiller with inlet guide vanes, air-cooled screw chiller with continuous slide valve, etc.) are generally capable of larger rates of flow variation than equivalent absorption chillers. The centrifugal chillers have superior control systems that enable them to operate with greater stability when evaporator flow varies.

Open the chiller isolation valves slowly to encourage stable operation

If the chiller controller can only handle a flow-rate change of 2 percent per minute, then the isolation valve must take 30 minutes to open … far too long for most applications. To prevent rapid flow rate variations across chiller, it is important to select isolation valves that maintain a linear relationship between valve position and flow rate.

Bypass Flow

Every chiller has a minimum permissible evaporator flow rate fixed by the manufacturer's low water velocity limit. To ensure that minimum flow is always maintained through the chiller, a bypass with a modulating control valve is employed. As loads diminish, system flow reduces. When the system flow approaches the minimum flow requirement of the chiller, the modulating two-way valve in the bypass opens. The sum of the flow in the system plus the flow in the bypass must exceed the minimum flow requirement of the chillers. Flow meters are installed in both the supply and the bypass to calculate the flow sequence.

In order for this to work properly, the control of the system is dependent upon the accurate measurement of chilled water flow and proper selection of the bypass control valve. The inability of the bypass valve to properly open and close may result in the chillers tripping due to low flow.

The following guidelines have been suggested for low-flow bypass design and control:

- Locate the valve close to the plant. If bypassed water has a shorter distance to travel it will have less impact on chilled water pumping energy.

- Size the valve for the minimum flow rate of the largest chiller. The maximum quantity of bypass flow required at any given time is the minimum flow rate of the largest chiller.

- Size the valve to operate properly with a pressure drop less than the set point of the system differential pressure sensor under all operating conditions.

- Flow measured with flow meter or pressure differential across the chiller evaporator.

It should be noted that selecting a single valve to meet all of the criteria above may be difficult in some applications. If controllability over a wide range of differential pressure is of concern, alternatives such as the use of parallel valves may be considered. This, of course, adds to both the cost and complexity of the system. Alternatively, sufficient number of three way valves must be considered in the secondary side to guarantee that the system flow will never fall below this limit.

Bypass Valve

Selecting an appropriate valve actuator is critical. Choose one that maintains a linear relationship between valve position and flow rate; otherwise, the valve may permit too much water flow when it begins to open.

Note: A common butterfly valve won't provide the necessary flow characteristics. Verify the suitability of a particular valve by requesting flow-versus-position data from the supplier.

Bypass Location

In a single-plant configuration, it is preferred to locate the bypass as far from the chiller plant as possible, usually at the point in the distribution system where the pipe mains have reduced down to where they are the same size as the bypass pipe. This location provides a larger reservoir of water in the system that can help buffer changes in the chiller load when flows change and avoid shutdowns due to low evaporator temperature.

(As a side note, the buffering effect is dependent on the dynamics of the chiller compressor control loop and the bypass valve control loop, the shape of the distribution-pump curve, and other factors. The buffering effect of a distant bypass path is not guaranteed, but intuition suggests it is better than a nearby bypass.) As in any VPF system, the safest strategy is to limit the rate of change of the terminal control valves so the chillers do not see a rapid flow change.

Variable Primary Pump Selection

With constant flow chillers, the pumps are selected usually at the design flow of the chiller evaporators. In VPF systems, the pumps are oversized to the take advantage of the chiller's ability to provide additional capacity when the condenser water temperature is lower than the design temperature and second to improve loading when the return water temperature is higher than design (i.e. during low delta-T). The extent of oversizing depends on the maximum allowable velocity and the design velocity across the chillers. The actual allowable percent of overflowing the evaporator is the maximum allowable velocity divided by the design velocity. Assume that the maximum allowable velocity is 10 fps and the design velocity is 7.5 fps; the percent of overflowing allowable is then 10 ÷ 7.5 or 133%.

System Turnover Time

System turnover time is the time required to circulate one system volume at the system flow rate. It is a measure of the system water mass relative to the cooling load and indicates how rapidly temperature disturbances will propagate through the system. Longer turnover times increase stability of chiller control.

Turnover time will be relatively small for small volume, close-coupled systems and large for systems such as district cooling systems. The volume contained in the distribution piping of larger systems is typically more than sufficient to prevent turnover from being a problem.

System turnover time limits are recommended by chiller manufacturers to ensure that capacity controls can react stably to variations in load. Typical recommendation states that the system volume should be at least 6 gallons/ton of installed chiller capacity. This equates to a 3-minute turnover at design flow with a 12°F chilled water temperature difference (i.e. 2 GPM/ton). Turnover time has additional significance for variable primary flow because some manufacturers tie it to their recommended rate of chilled water flow variation through the evaporator.

Chiller Sequencing

Proper sequencing helps to maintain the flow rate through each evaporator within the range recommended by the chiller manufacturer. As the system flow nears the maximum limit for the operating chiller(s), another machine must be brought online. Similarly, as the system load and flow decrease, chillers must be shut down to reduce the need for bypass water flow.

Temporarily unload the operating chillers before starting the next one

Reduce shock resulting from transient flows by unloading the operating chillers *before* opening an isolation valve to bring another chiller online. You can accomplish this by imposing a demand limit of 50-to-60 percent on the operating chillers, or by raising the chilled water setpoint one to three minutes before the isolation valve actuates.

Load chiller optimally before starting another one

The simplest way to control a VPF system is to monitor the leaving-evaporator water temperature and allow the operating chiller(s) to load *almost* fully before bringing the next chiller online. As long as the system can maintain the target temperature, there is no need to activate another chiller.

When the operating chillers no longer provide enough cooling, the plant controller should start the next chiller.

Devise a "stop" strategy that protects the chillers from short-cycling

Knowing when to stop a chiller (to provide sufficient downtime between chiller starts) often is more challenging than knowing when to start it. The most reliable way to do so—assuming that the VPF system is properly installed, calibrated, and maintained—is to *monitor the power draw of the operating chillers*. Most unit controllers measure running load amps (RLA) at regular intervals. The %RLA (actual RLA divided by design RLA) provides a good indication of the present chiller load.

Base the "stop" strategy for a multi-chiller plant with *equally sized machines* on the sum of the present %RLA for all chillers divided by the number of operating chillers *minus one*. If the result is less than the desired capacity for the operating chiller(s), then stop one of the machines.

Flow Detection for VPF Chiller

In variable flow applications, flow detection methods better suited to the application should be used. High-accuracy flowmeter selected for VPF application shall monitor chilled water flow at frequent intervals (recommended 1-second interval) and base control action on multiple readings (recommended 5 consecutive signals).

Alternatively use a differential pressure transmitter selected for variable flow duty, which measures the pressure differential across the evaporator and convert to flow based on the specific manufacturer's published flow-pressure data. Either of these measures will improve the reliability of chillers in variable primary flow applications.

System Head in VPF system

Careful attention must be given to the design pressure of hydronic piping, and valves at the pump shut-off conditions. Because the total dynamic head is produced by a single set of pumps and there is no separation between the plant and distribution systems, VPF scheme will require larger motors. In addition, VPF system will increase the system design pressure and careful attention must be given to the selection of piping and valves to be rated above maximum system pressure.

Summarizing……

Variable primary flow systems are acknowledged to be more difficult to control and maintain than comparable constant primary flow systems even by their proponents. In addition to the controls required in constant primary flow systems, variable primary flow requires an accurate means of measuring chilled water flow rate and a method for adding units of chiller capacity without abruptly reducing the flow rate through active chillers.

COST COMPARISON OF VARIABLE PRIMARY FLOW WITH OTHER PUMPING SCHEMES

Variable primary flow systems are particularly attractive because they have lower equipment costs than primary/secondary systems. Table below shows the first cost comparison of a 500 ton chiller plant.

CHW system type	Primary/Secondary	Variable flow, primary-only
Cooling load, tons	500	500
Total CHW flow rate, gpm	1,000	1,000
Primary pump head, feet	50	120
Secondary pump head, feet	70	NA
CHW system type	Primary/Secondary	Variable flow, primary-only
CHW pump equipment cost, $	10,516	7,358
CHW pump installation cost, $	2,857	1,486
Piping and fittings installed cost, $	19,070	NA
VFD/starter installed cost, $	9,860	14,550
Bypass/decoupler installed cost, $	1,328	929
Bypass valve installed cost, $	NA	1,548
Flow meter installed cost, $	NA	1,800
Total installed cost, $	43,631	27,671
Δ Total installed cost, $	Base	-15,960
Δ Total installed cost, %	Base	-37

*Source - Air-Conditioning and Refrigeration Technology Institute (ARTI) study report on VPF potential benefits.

*The analysis does not take the cost of 500 TR chillers which will be same for both the options.

The above analysis includes just the first cost elements. In general, VPF system offer lower first cost, reduced peak demand, reduced energy sue, less mechanical room space and ability to cope with lower than design ΔT. However, the control must prevent rapid changes in flow, as a sudden drop in flow would result in a rapid drop in leaving chilled water temperature, and could trigger a low-temperature safety alarm. It is also important to maintain a minimum flow rate through the evaporator to ensure proper heat transfer. Chiller manufacturer's chiller management controllers are capable of managing these critical flow and temperature limitations to assure safe chiller operation.

The selection of a determined scheme should be based on a careful analysis of such factors in order to reach the best balance between equipment cost and system performance. In the context of HVAC design, decisions made to save money often involve a tradeoff between acquisition expense and operating cost. If you can realize savings on both fronts, so much the better. Perhaps this explains the increased interest in chilled water systems with a VPF scheme.

<p style="text-align:center">*************</p>