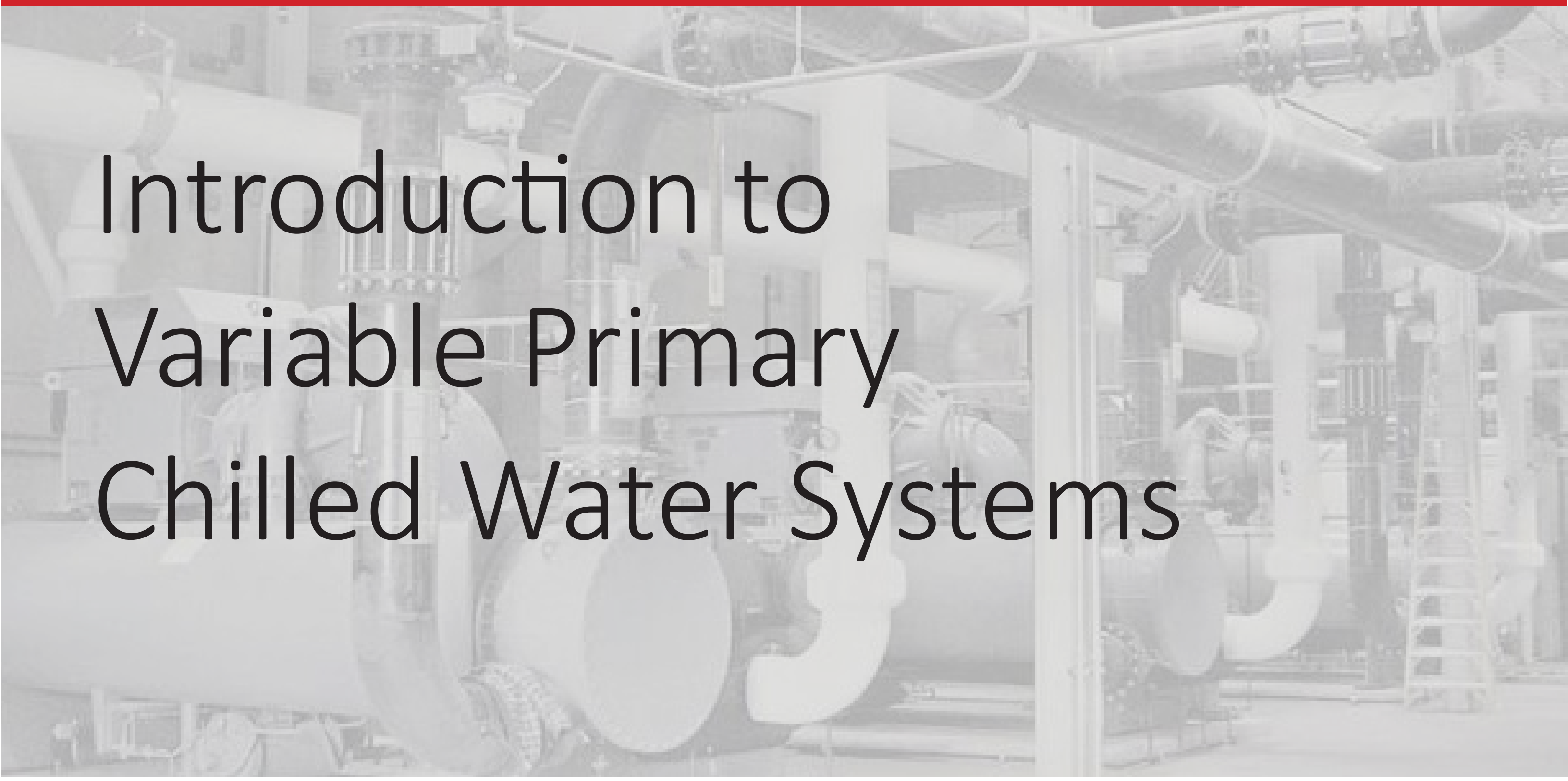
WHITE PAPER



Introduction to Variable Primary Chilled Water Systems

By JMP Equipment Company

What is a variable primary chilled water system? How does it differ operationally from a primary secondary system? And why might you choose one chilled water design over the other? This white paper will answer these very fundamental questions, while also exploring related topics including low delta-T syndrome, chiller staging, and more.

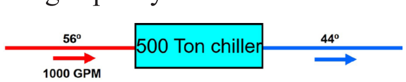
We begin by answering the most fundamental questions of all when it comes chilled water design: What is a chiller (or air conditioning) ton?

In HVAC terms, a ton is a metric for how much heat (e.g. how many BTUs) a chiller or other air conditioning unit can remove from a space in one hour. It takes one ton of air conditioning to remove 12,000 BTUs in an hour. Therefore, a 5-ton chiller can remove 60,000 BTUs (5 x 12,000) in an hour.

In any system, BTUH = 500 x GPM x ΔT. (Note: 500 is a constant that is based on the weight of one gallon of water (8.33 pounds) times 60 minutes in an hour. So, 8.33 pounds x 60 minutes = 500). To convert this to chiller tons, we simply divide by 12,000 since there are 12,000 BTUs in one cooling ton:

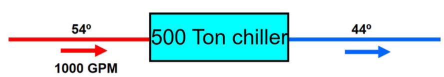
$$Tons = \frac{500 \times GPM \times \Delta t}{12{,}000}$$

Thus, if we have a 1000 GPM chilled water system with a 44°F supply temperature from the chiller and 56°F return temperature to the chiller, giving us a 12 degree delta T, we know we need 500 tons of cooling capacity:



$$\frac{500 \times 1000 \times 12}{12{,}000} = 500 Tons$$

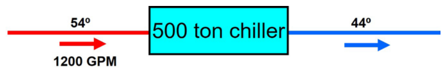
Under these conditions, a 500-ton chiller would be fully loaded and it would be delivering exactly 500 tons of cooling.
But what happens to the chiller output (and by output, we mean its ability to remove BTUs) if we have only a 10 degree delta T? As always, the laws of math and physics prevail, and as you can see by the calculation below, we are no longer able to get 500 tons of cooling capacity from our chiller. In other words the chiller is only 83% loaded.



$$\frac{500 \times 1000 \times 10}{12{,}000} = 417 Tons$$

Is this a problem? Not necessarily. As long as our building load also happens to be at 83%, this is exactly as it should be. However, if we are fully loaded on the building side, a reduced delta T usually means there is a problem out in the system that is preventing the chiller from delivering its full capacity. This holds true no matter what type of chilled water design we have, as does the relationship between all of the values in the above equation.

Now let’s consider another variable in the equation: GPM, or flow. If we had the option of varying flow through the chiller, we might be able to mitigate this problem by temporarily increasing the flow. In this case, if we increase our chiller flow to 1200 GPM, we can re-establish the 500 tons that we need:



$$\frac{500\times1200\times10}{12,000}=500Tons$$

Is there a penalty for increasing the flow? Of course. In this case, increasing the flow to 1200 GPM will increase the pump brake horsepower by 70%. That may or may not be a solution for the given situation, but you begin to see how it could be beneficial to vary flow through the chillers. More importantly you begin to understand the relationship between flow, delta T and tonnage and that is essential when approaching chilled water design.

Primary Secondary Design and Low Delta T

If you are at all familiar with chilled water systems, you are probably aware that most fall into one of two categories: primary secondary or variable primary.

Primary secondary systems utilize two sets of pumps, the primary pump(s) which maintain a constant flow through the chillers/chiller loop, and the secondary pumps which supplies flow to the system. The secondary pumps are equipped with variable speed drives (VFDs) which vary the flow through the secondary loop in response to load conditions.

Variable primary systems, on the other hand, do not have a secondary set of pumps. Rather, a single set of VFD-equipped chiller pumps supply chilled water throughout the entire system, increasing and decreasing flow based on load.

For decades primary secondary dominated chilled water design, mostly because they were simpler to understand and control. However, this simplicity comes at price. Since primary secondary systems are comprised of the primary loop which flows through the chillers and the secondary loop serving the system load, there are three possible flow conditions that can occur:

(1) The primary flow can be greater than the secondary flow.
(2) The primary flow can be less than the secondary flow
(3) The primary flow can be equal to the secondary flow.

Number 3 is the ideal condition because it keeps the two loops operating independently of one another, i.e. there is no flow in the common pipe that is situated between the two loops. Unfortunately, this condition hardly ever occurs since loads are constantly changing. More often than not, there will be unequal flows through the primary and secondary loop in a primary secondary system and it is this difference in flow that creates inefficiency. In some cases, extreme inefficiency.

Most of the time, primary secondary systems operate under conditions in which the primary flow is greater than the secondary flow, as shown in Figure 1. (Note that the design parameters for this system and other examples in this paper are all the same: Two chillers at 500 tons each, 44°F chilled water supply, 56°F chilled water return, 12 degree delta T, and two 1000 GPM chiller pumps.) Here we see that our system is demanding 1500 GPM of flow, so we have to turn on both chillers and primary pumps.

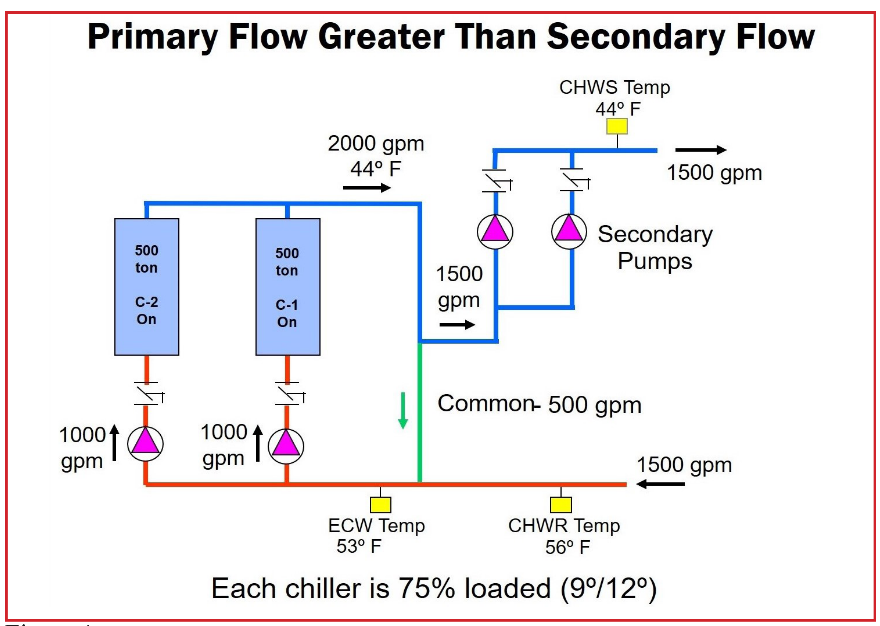


Figure 1

The chillers are partially loaded at 75% each, but because we cannot vary flow through the chillers, we have 2000 GPM in our primary loop even though the secondary loop is only calling for 1500 GPM.

Where does that surplus 500 GPM of 44°F water go? It mixes with the return 56°F water from the secondary loop and goes back to the chiller at 53°F. Obviously, there is an energy penalty associated with the extra flow through the primary loop and a lower entering chilled water return temperature to the chillers, but at least the load is satisfied.

A more serious problem occurs when the primary flow is less than the secondary flow. Let's say we have one chiller operating on with 1000 GPM in the primary loop at our design supply of 44°F. Our load starts to increase beyond a single chiller capacity and the system is demanding 1200 GPM in the secondary loop. When that 1200 GPM of return water hits the tee on the return side of the common pipe, 200 GPM of 56°F return water will get pulled back into the supply side of the secondary loop. That means that the 56°F return water will begin mixing with the 44°F supply, increasing the supply temperature to 46°F.

If we continue to operate under these conditions, we will require a second chiller to manage the load (Figure 2).

This doubles the flow in the primary loop, but keep in mind the secondary loop is still only flowing 1200 GPM. Now we have 800 GPM of 44°F water flowing straight through the common pipe and mixing with the 56°F return chilled water. This gives us an entering temperature of 51°F to our chillers and a 7 degree delta T across our chillers. Everyone is still comfortable in the building, but the chillers are not operating as efficiently with the lower delta T because of the mixing in the common pipe.

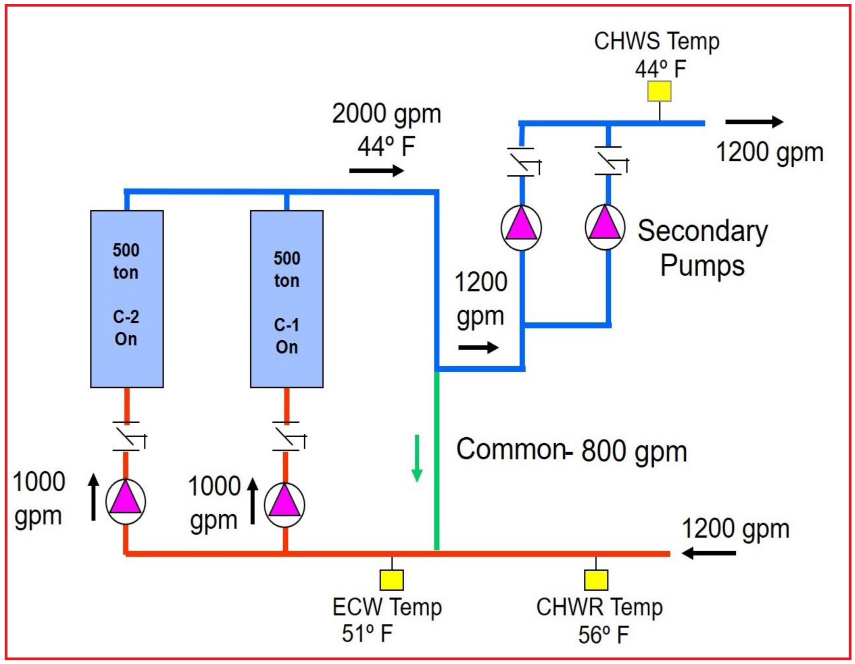


Figure 2. When we turned on primary pump #2, th eprimary flow increased to 2000 GPM, but the secondary flow remained 1200 GPM.1200 GPM.

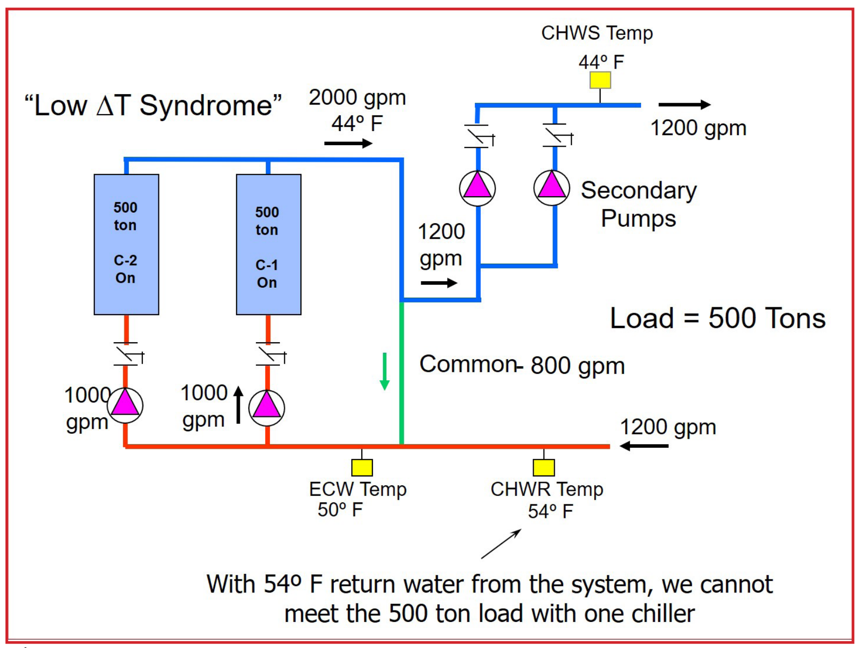


Figure 3

Now let's look at the example in Figure 3. Our return water temperature has dropped to 54°F, the result of any number of common reasons including poor balancing, dirty coils, low control valve authority,

etc. The chillers were designed for a 12 degree delta T but now we are only getting a 10 degree delta T. The system is demanding 500 tons of cooling, so the flow rate increases to 1200 GPM. We have no choice but to bring on a second chiller even though our load is still just 500 tons. Because of our reduced delta T and the mixing of water from our chiller supply water with our return water, we must operate two chillers at a 6 degree delta T instead of a single chiller at a 12 degree delta T. Consequently, we are wasting a lot of energy:

- Our primary pump brake horsepower (BHP) is over twice what it would be with a 12 degree delta T.
- We are pumping 20% more flow on secondary side. Secondary pump BHP suffers an increase of approximately 60%.
- We are running two chillers instead of one; two towers instead of one; and two condenser pumps instead of one.

This is just one example of the problem known as "low delta T syndrome," the most common reason why variable primary systems have become increasing popular. In addition to the energy waste and excessive wear and tear on equipment, we now have no chance whatsoever of meeting a design load of 1000 tons.

There are a few things that can be done to improve the above situations without undertaking a variable primary conversion:

- Fix the low delta T problem out in the system (Remember, low ΔT is not caused by primary secondary), the secondary loop is taking more flow than it should for the load.

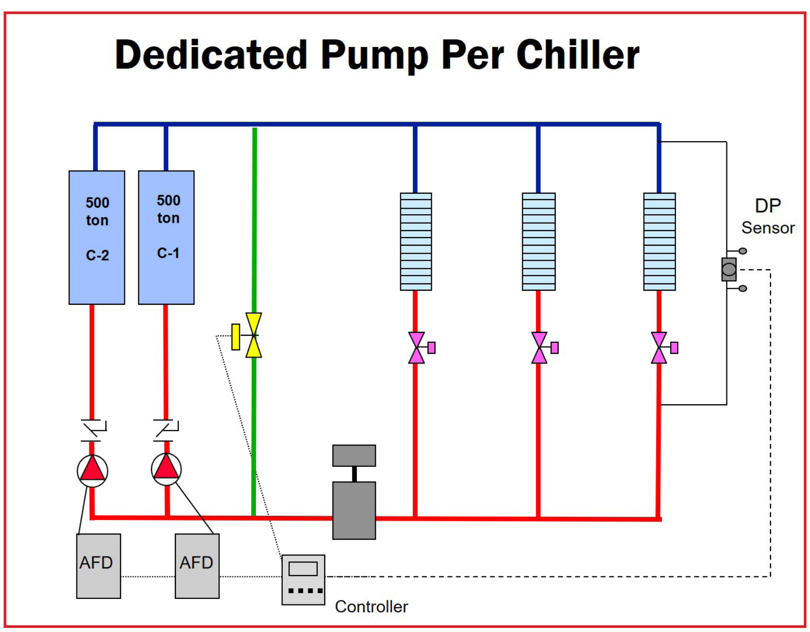


Figure 4

- Delete 3-way valves and bypasses where possible.

- Check control/balance valves for overflow vs load (what's the delta t at the coil?)

- Review the control strategy, the location and setpoints of transducers of the secondary pump vfd to see it this could be the cause of overflowing.

Making the Most of Variable Primary

Now that we've established the real world challenges of traditional constant primary/variable secondary chilled water systems, we can begin to explore the alternative, variable primary, a singular loop system that varies flow through the chillers.

There are two basic ways to pipe the pumps and chillers in a variable primary system. The chillers and pumps can be piped so that each chiller has its own dedicated pump (Figure 4) or they can be piped with the pumps in parallel with a common header between the pumps and a common header in the chiller supply (Figure 5).
If simplicity is a priority then dedicated chiller pumps may be the way to go. But generally speaking parallel pumps with common headers offer more flexibility.

Looking a Figure 5, you can see that we have the ability operate the system in several ways depending on the load and/or current situation. For instance, we can operate two pumps and just one chiller so that we can increase flow out into the system without having to stage on an additional chiller. This actually allows us to slightly exceed the rated tonnage of the one chiller in order to meet load.

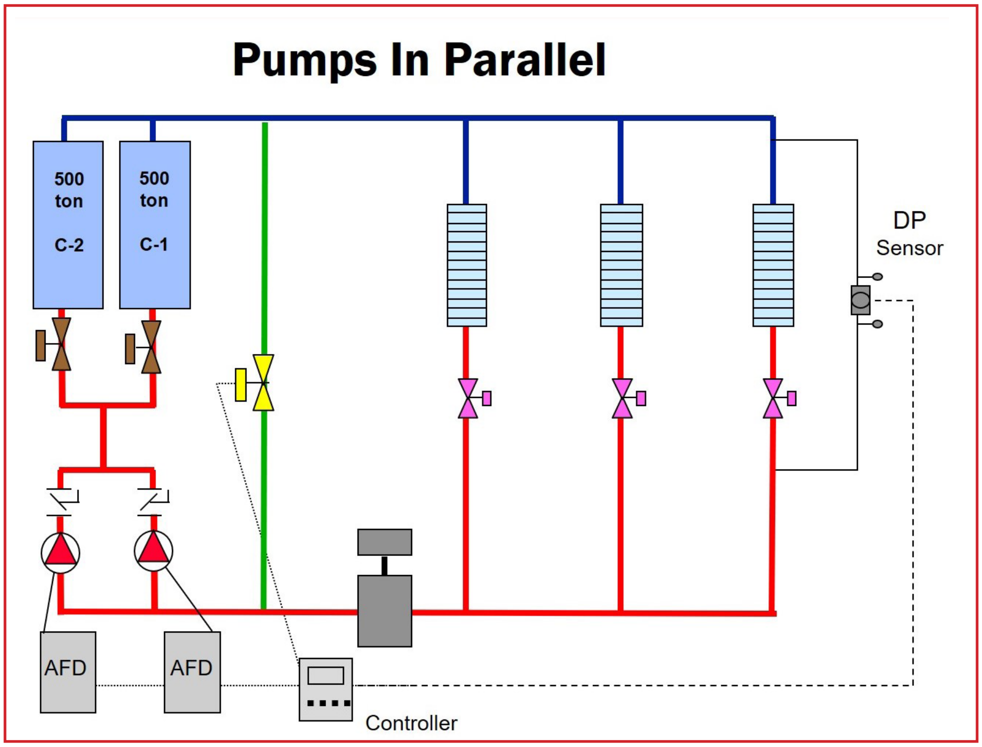


Figure 5

We also have more flexible redundancy with commonly headered pumps and chillers. If a pump goes down, the remaining pump can serve one or both chillers and still meet the required load. If a chiller goes down or needs servicing, we can compensate for the loss in capacity by increasing flow through the remaining chiller while operating both pumps. So, there are lots of operational combinations available to help us increase efficiency and provide redundancy when servicing is necessary.

No matter what piping configuration we choose, we must make sure that we design and operate the system so that we stay within the minimum and maximum required flow of each chiller. This ensures that we maintain good heat transfer and stable operation at lower flows and avoid eroding the tubes at higher flows.

First, it is important to select chillers with a pressure drop that will support (and not limit) your turndown capability. In other words, you don't want the pressure drop through the chiller to be so high or so low that it prevents you from staying within the minimum and maximum flow range of the evaporator as the system stages pumps and chillers to meet the load. Most chiller manufacturers provide delta P sensors to sense flow through the chillers, but other flow measuring devices can also be installed (Figure 6).

A bypass valve is also required to ensure minimum flow through the chillers, especially in situations where the system is demanding little or no flow. If the demand

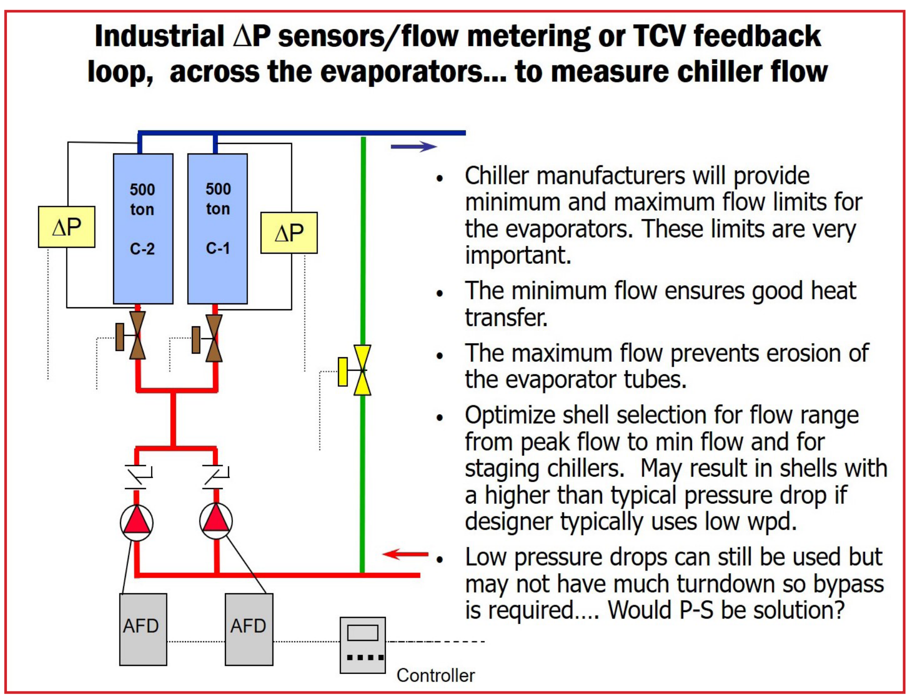


Figure 6

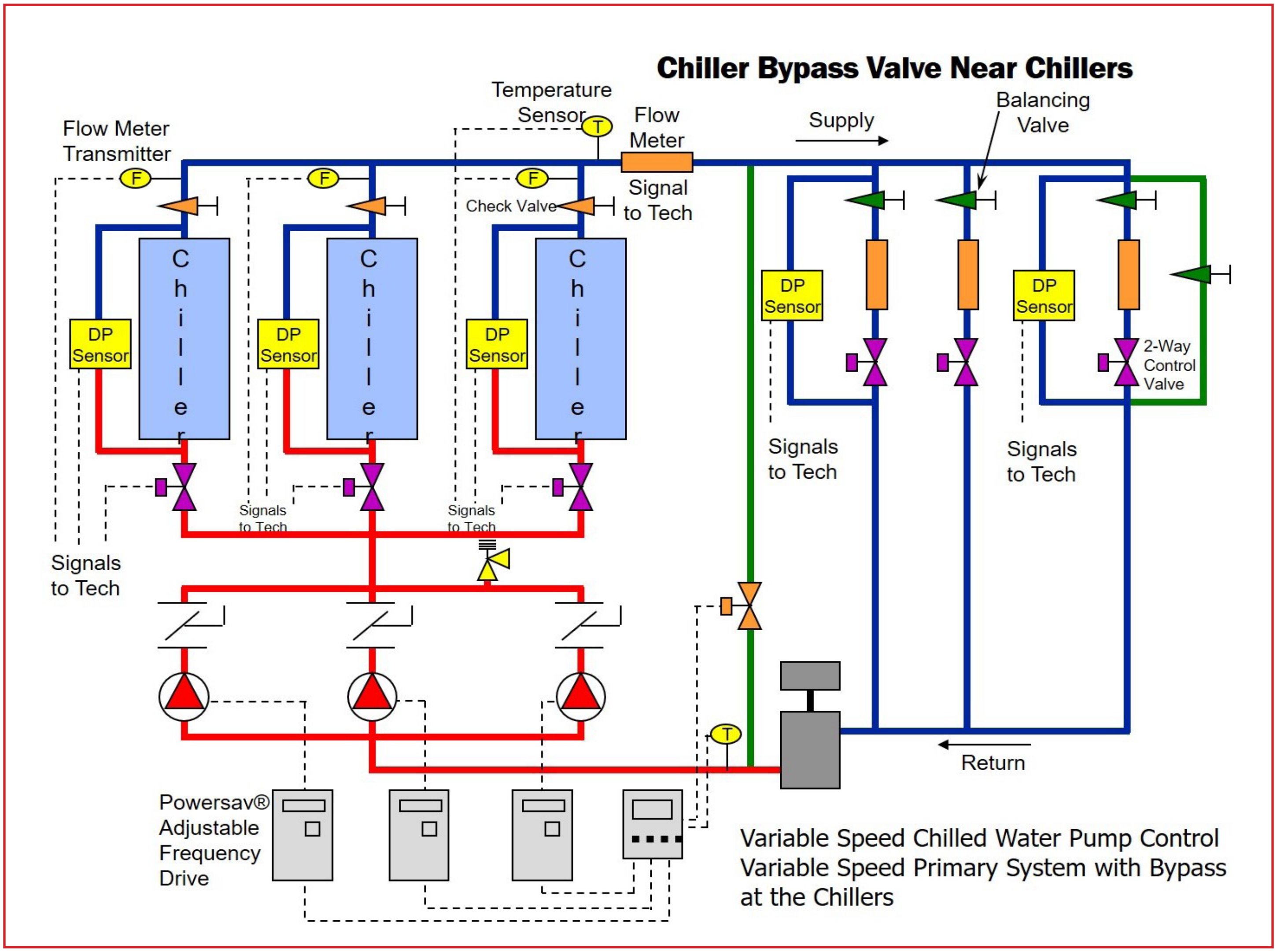


Figure 7

of the building is less than the minimum chiller flow, the bypass valve should open to maintain this flow. The bypass valve also ensures that there is a sufficient flow thru the pump if all the 2-way valves out in the system are closed, otherwise the pumps could deadhead.

The bypass valve should be sized so that when it is wide open it delivers the highest minimum flow that is required of any single chiller. The bypass valve differential pressure should be the same as the differential pressure setpoint out in the system so we can supply this flow. It should not be a fast opening valve like a butterfly valve because this will cause unstable flow changes during staging, which is not good for the chillers.

The bypass can be located in the mechanical room or at the end of the system. However, we recommend installing it in the mechanical room in close proximity to the chillers. By installing the bypass between the chiller and system, we don't have to size our entire piping system for the additional flow. This also helps minimize pump operating cost and facilitates communication between differential pressure sensors and controls. Figure 7 shows a more complex system with the bypass installed between the chillers and the system.

How Variable Primary Helps Mitigate Low Delta T

How exactly does variable primary reduce the potential for low delta T?

Going back to our earlier explanation of chiller tonnage, we know that BTUH = 500 x GPM x ΔT. If we want to convert this equation to chiller tons, we simply divide by 12,000 because there are 12,000 BTUs in one cooling ton:

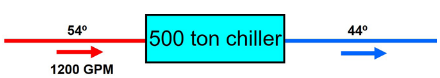
$$Tons = \frac{500 \times GPM \times \Delta t}{12{,}000}$$

Hold that thought as we take a look at the variable primary system in Figure 8.

In this example we are operating at a lower delta T than design. Our system is demanding 1200 GPM of flow at a 10 degree delta T, whereas our design was for a 12 degree delta T. Our variable speed primary pumps are sized for 1000 GPM each. If this were a standard primary secondary pump, we would have no choice but to operate both pumps, chillers, condenser water pumps and cooling tower; otherwise we would not be able to satisfy the load. That a lot of equipment and a lot of energy.

However, with variable primary, we can operate both our primary pumps at 600 GPM each and meet our flow demand while still achieving our rated 500 tons of cooling through just one chiller. Consequently, we avoid having to operate an extra cooling tower and condenser water pump.

Our chiller is fully loaded but we are operating at a 10 degree delta T instead of our design 12 degree delta T. That's okay because if we go back to our equation and plug those numbers in, we can see that we are still able to get 500 tons of cooling from our chiller:



$$\frac{500 \times 1200 \times 10}{12{,}000} = 500 Tons$$

We also have the option of sizing our pumps up to 1200 GPM and operating only one pump under this same condition. Either way, we are saving a tremendous amount of energy and wear and tear on chilled and condenser water equipment!

Staging Variable Primary Systems

There are a few fundamental issues that engineers and operators must be aware of

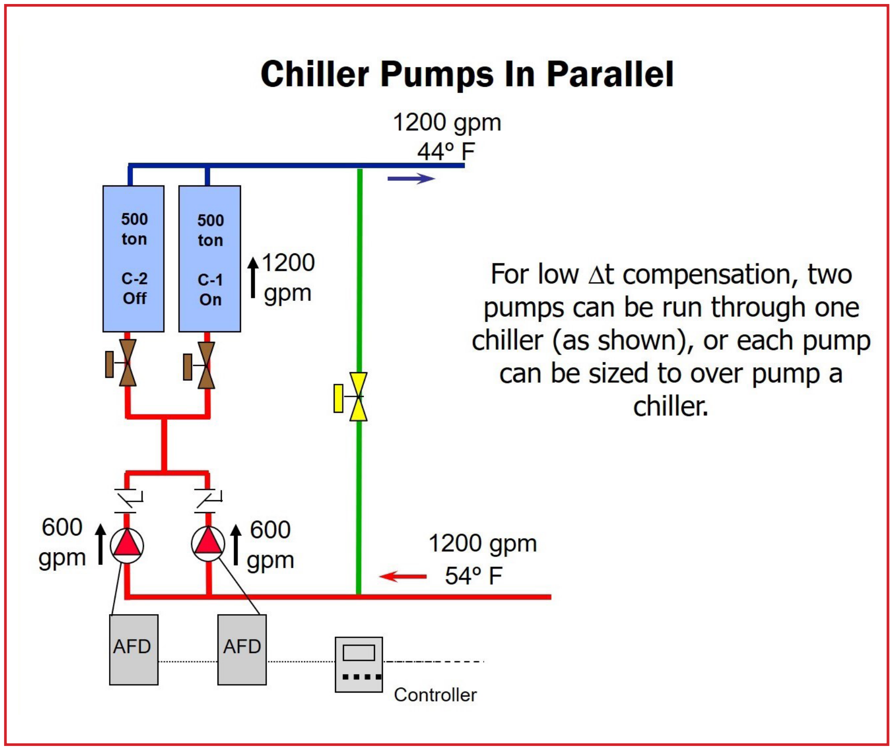


Figure 8

when staging chillers in a variable primary system.

First, it is important to make sure the isolation valves to the chillers are set to open gradually when staging on a new chiller. Otherwise, you run the risk of freezing up the chiller or at least causing it to go off on a FAULT due to extremely low evaporator pressure.

To sketch this out, let's say we are in the first stage of operation in our example variable primary pumping system (Figure 9). Notice that we a have a parallel configurations with commonly headered pumps and chillers, giving us the option to operate the equipment in a variety of combinations. The system is demanding 1000 GPM and we are meeting the load with Chiller 1 and a single pump operating.

If our building demand increases to 1100 GPM, the equivalent of 550 chiller tons, we are going to need that second chiller. But we can't immediately open isolation valve on Chiller 2 to 100%; doing so would cause the flow through Chiller 1 to drop rapidly from 1100 to 550 GPM without sufficient time for the chiller to adjust its output. All of a sudden Chiller 1 would be operating at full

500-ton capacity at 50% flow. As a result, it would be producing 34 degree chilled water and our Delta T would be through the roof at 22. (Figure 10)

To keep this from occurring, we must make sure that we gradually open the isolation valve to Chiller 2. How gradually we open the valve is a function of what a particular chiller can take. When designing a system, we recommend giving manufacturers the worst case scenario in terms of potential flow drop through the chiller and let them provide you with the necessary time lapse for the chiller(s) to adjust.

If our variable primary system has a dedicated pump per chiller (Figure 11), keep in mind that flow through Chiller 2 cannot be established until the pump speed for that chiller is high enough to produce enough pressure to overcome the flow through Chiller 1. As Pump 1 unloads from 1100 GPM to 550, eventually the pressure imposed on the check valve between Chiller 2 and Pump 2 will decrease, allowing it to fully open so that 550 GPM is going through both chillers.

Conclusion: Is Variable Primary Right for Your Project?

When it comes to variable flow and chilled water systems, there is no "one approach fits all" solution. Like virtually every other aspect of mechanical HVAC design, the ideal solution is wholly dependent on the design of the specific building, the building purpose, and the requirements of the owner and the occupants.

That said, there are certain factors that provide solid clues as to whether variable primary design will yield favorable results. Here are the questions that engineers should consider before making that design decision:

Can the system flow be reduced by at least 30%? In other words, if the minimum required flow at any given time is 70% or more of peak design flow, then a variable primary system is unlikely to yield savings that will be worth the added complexity of control.

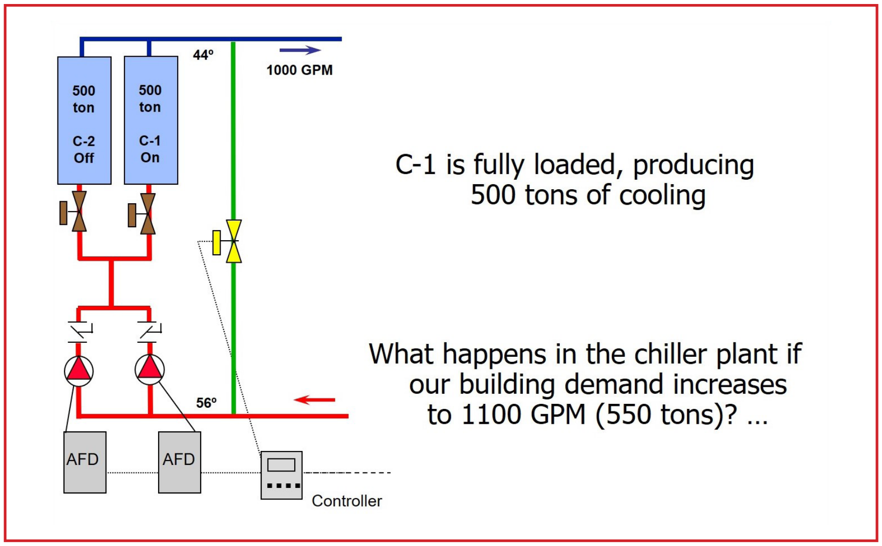


Figure 9

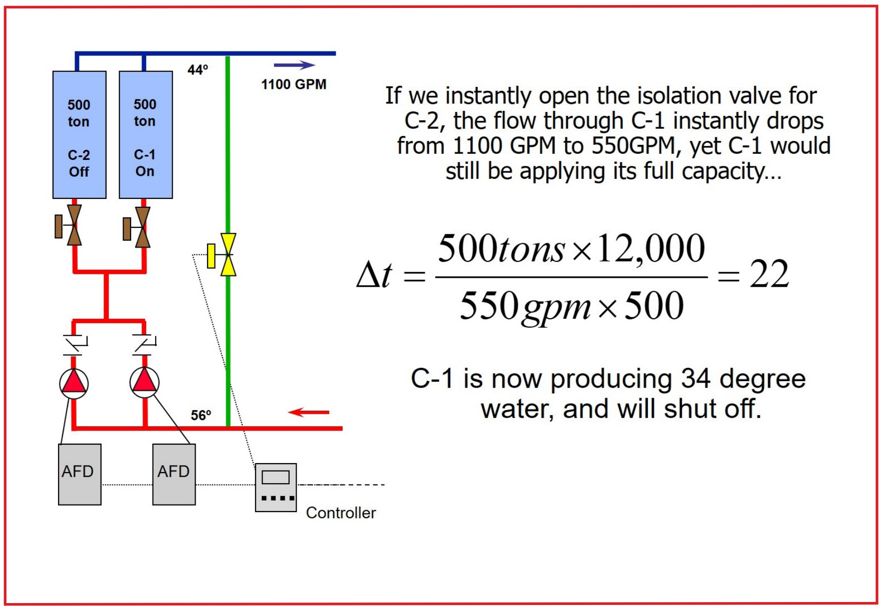


Figure 10

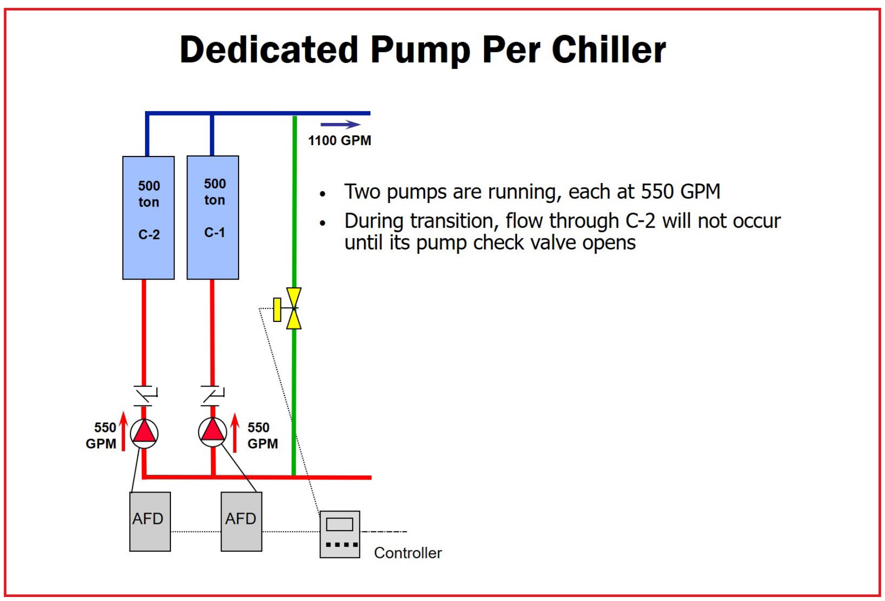


Figure 11

Can the building tolerate a modest change in water supply temperature? When staging up a variable primary system, there will always be a temporary increase in supply water temperatures. Typically this is not for a period greater than 10 or 15 minutes, but we have seen supply temperatures increase to as high as 50°F during this interim period. Obviously, if the leaving supply temperature is critical to the operation of the building, then this could be prohibitive.

Are the building operators well-trained and present? Variable primary systems are more complex and tend to be more successful when the building is supported by onsite mechanical staff who are familiar with the design and control of the chilled water plant.

Does building modeling demonstrate cost savings? Certainly, engineers will want to model and compare the operational cost of primary secondary, variable primary and possibly some combination of both before deciding.

Are the pumps and chillers equally sized? If this is a renovation project with existing equipment, it is important to consider how the current chillers and pumps are sized. Variable primary systems are much simpler to control when all chillers and primary pumps are uniformly sized.

Properly applied, variable primary chilled water design can be a financially strategic option for many types of facilities. Determining whether it is right for your application requires careful evaluation of many factors. **As always, JMP is here to help.**