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Engineers Newsletter

volume 44–3

Dual-Temperature Chiller Plants

This Engineers Newsletter describes several dual-temperature chiller plant configurations that can provide two temperatures—one cool enough to provide sensible cooling but warm enough to avoid condensation, and the other cold enough to dehumidify.

Some types of space cooling equipment require a supply of chilled water which is 1) cool enough to provide sensible cooling for the space, but also 2) warm enough to avoid condensation on the cold surfaces of the equipment.

Typical examples include radiant cooling panels (or tubing embedded in the building structure), chilled beams, fan-powered VAV boxes with sensible-only cooling coils, or fan-coils/blower-coils that have been selected for sensible cooling only. (These types of space cooling equipment will be collectively referred to as "terminal units" for this EN.)

The common characteristic of these types of terminal units is that they provide sensible cooling only, and are not intended to provide dehumidification (no means to manage condensation).

They are typically supplied with water in the range of 57°F to 60°F, which is cool enough to provide space sensible cooling, but warmer than the dew point in the space, thereby avoiding condensation on sensible coils in the terminal units. However, since these terminal units only provide sensible cooling, a separate dehumidification system is usually needed.

This separate system uses a chilled-water coil to dehumidify 100-percent outdoor air, or a mixture of outdoor and recirculated air. This dehumidifying coil needs to be supplied with water cold enough (typically 40°F to 45°F) to dry this air well below the space dew point (usually about 55°F).[1]

This Engineers Newsletter describes several dual-temperature chiller plant configurations that can deliver these two different temperatures.

Plant with a Single Chiller

Many small chilled-water systems include only one water chiller. Often this is an air-cooled chiller that may have independent refrigeration circuits to provide some degree of redundancy.

In this case, the chiller will need to produce the colder of the two water temperatures (40°F in this example), and use either an intermediate heat exchanger or a blending valve to provide the warmer water temperature (57°F).

When dehumidification is required, this precludes any efficiency benefit of operating a chiller at the warmer leaving-water temperature. During drier weather, when the dehumidifying coils are no longer needed, the leaving-water temperature setpoint for the chiller can be reset up from 40°F to 57°F.

What about using a standalone dehumidifier?

Some systems are designed with a chiller plant that provides 57°F water to the terminal units, but then uses a standalone, direct-expansion (DX) unit for the dehumidification system. While this approach benefits from operating the water chiller at the warmer temperature, there is no redundancy if either the chiller or DX dehumidification unit needs to be repaired, replaced, or serviced.

In addition to providing this redundancy, designing a chiller plant to serve both the space sensible cooling load and the ventilation/dehumidification load can increase system efficiency, and provides opportunities to incorporate other strategies like waterside heat recovery, thermal storage, or water economizing.

For example, the full-load cooling efficiency (not including supply fan power) of a standalone, air-cooled DX dehumidifier is likely to be between 1.1 and 1.2 kW/ton. The overall full-load efficiency (including pumps) of a plant that uses a single, shared air-cooled chiller (Figure 2) is likely to be between 1.0 and 1.1 kW/ton. And the overall full-load efficiency (including pumps and tower fans) of a dedicated water-cooled chiller serving the dehumidifying coils (Chiller 2 in Figure 8 or 9) is likely to be between 0.7 and 0.9 kW/ton.

Plus, the performance of water chillers is certified through AHRI; whereas there is no existing certification program currently in place for standalone DX dehumidification units.[2]

Intermediate heat exchanger. In the first configuration (Figure 1), the water chiller produces 40°F fluid (water or brine). Some of this fluid is distributed to the dehumidifying coils in the air-handling units; while the rest passes through a plate-and-frame heat exchanger that is used to produce 57°F supply water for the sensible-only terminal units.

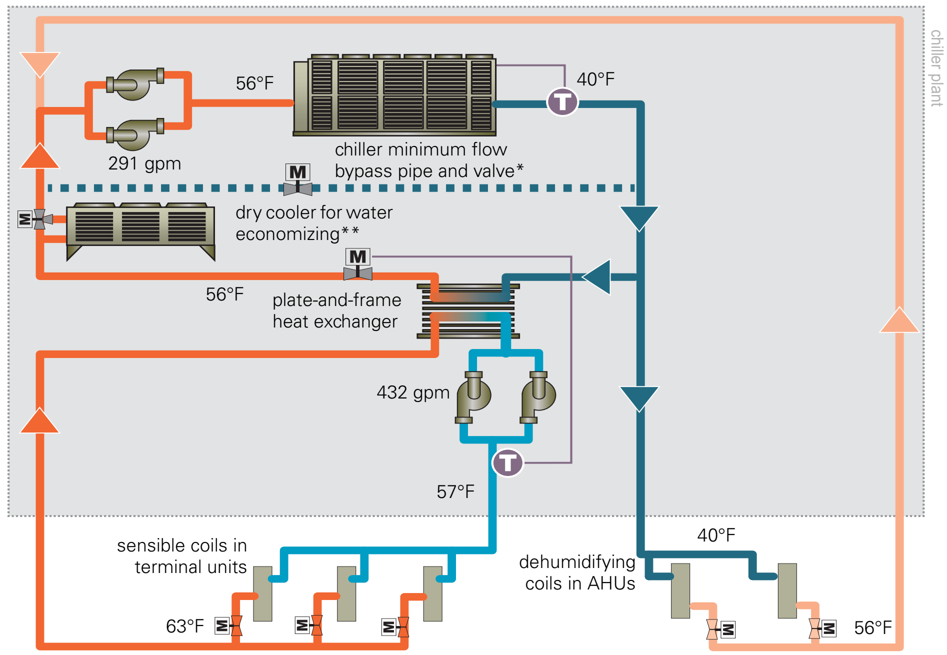
The benefit of this configuration is simplified hydronics and control. The control valve for the heat exchanger is selected and controlled similar to the valves on the dehumidifying coils, and the delta T (inlet-to-outlet temperature difference) selected for the chiller and dehumidifying coils are more-or-less independent of the delta T dictated by the terminal units.

If glycol is needed for freeze protection, it can be isolated to flow through the water chiller and dehumidifying coils only. Pure water can be used on the indoor side of the heat exchanger, so the glycol will not affect the capacity of the terminal units.

Blending valve. If glycol is not needed, or if glycol will be circulated through the terminal units also, a blending valve can be used in place of the plate-and-frame heat exchanger (Figure 2). In this configuration, the water chiller again produces 40°F water, some of which is distributed directly to the dehumidifying coils. The remaining cold water is blended with warm water (63°F) returning from the terminal units to produce the required 57°F supply water for the terminals.

This configuration provides a small efficiency advantage, since the intermediate heat exchanger used in the previous configuration is not 100-percent effective. But the hydronics are more challenging because the blending valve must be carefully selected and commissioned to provide accurate, stable control over the expected range of system operating pressures.

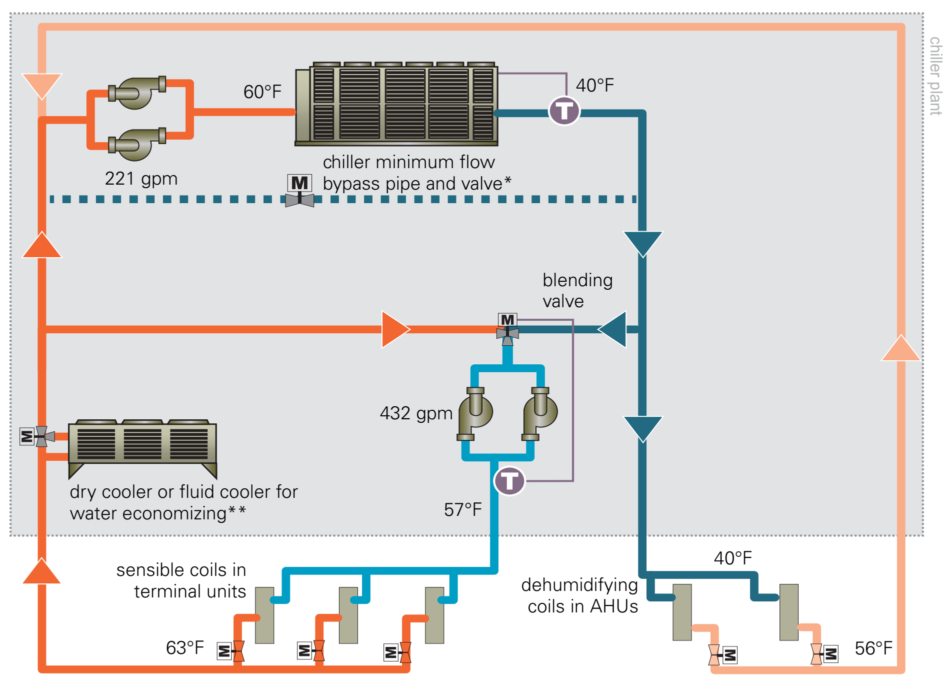
Figure 1. Single-chiller plant (intermediate heat exchanger)



* included if variable-flow chiller pumps are used

** optional (or some systems use a separate air-to-water heat exchanger incorporated into the air-cooled chiller)

Figure 2. Single-chiller plant (blending valve)



* included if variable-flow chiller pumps are used

** optional (or some systems use a separate air-to-water heat exchanger incorporated into the air-cooled chiller)

Plant with Two Chillers

Many chiller plants are designed to include more than one chiller to improve plant efficiency and to provide redundancy if one of the chillers were to fail or require service.

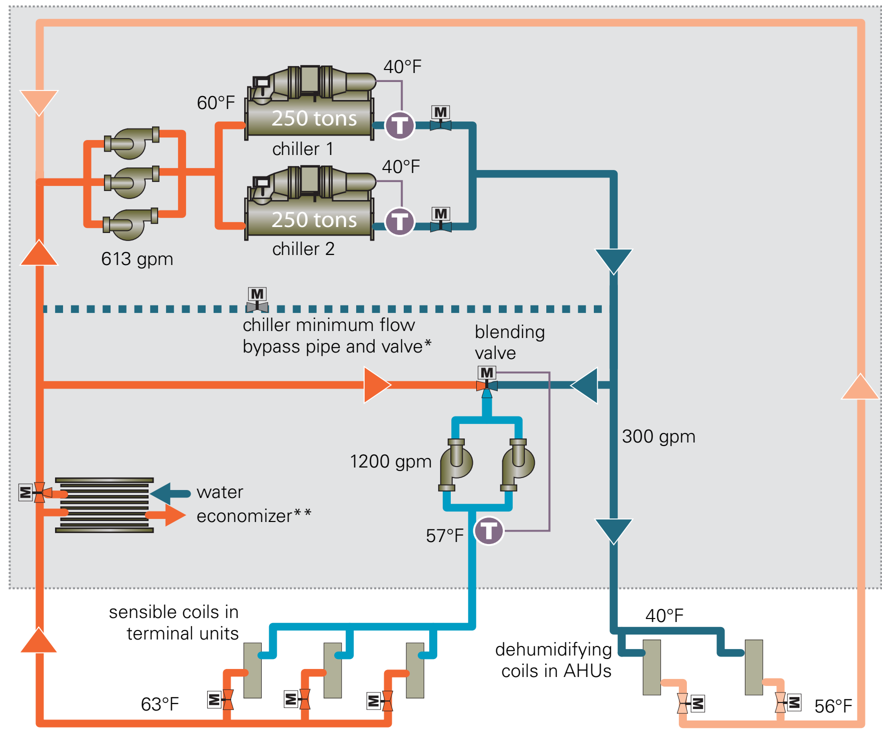
When the plant is designed with two chillers, there are several configurations that might be used to provide the two different water temperatures. While either air- or water-cooled chillers can be used in a two-chiller plant, our example will assume water-cooled chillers.

Blending valve with loads in parallel (chillers in parallel or series). The first two-chiller configuration (Figure 3) uses the same blending valve concept described previously. The water chillers, with **evaporators piped in parallel**, both produce 40°F water. Some of this cold water is distributed directly to the dehumidifying coils in the air-handling units, while the rest is blended with warm water (63°F) returning from the terminal units to produce 57°F water.

Because of the large delta T across the chillers, however, this is typically a good application for configuring the chiller **evaporators in series** (Figure 4), rather than in parallel. In this example, the upstream chiller cools the water from 60°F to 50°F, while the downstream chiller cools it the rest of the way to 40°F.

One benefit of configuring chiller evaporators in series is lower overall plant energy use. In this example system (see Table 1), the upstream chiller operates much more efficiently (0.462 kW/ton) since it need only cool the water to 50°F, resulting in less total chiller power. Since the full water flow rate (613 gpm) is pumped through both chiller evaporators, the incremental pump power is higher (4.4 kW compared to 1.1 kW with chillers in parallel). But the total power for both chillers plus incremental pumping power is still 9 percent lower with the chillers configured in series (261.2 kW versus 285.9 kW with chillers in parallel).

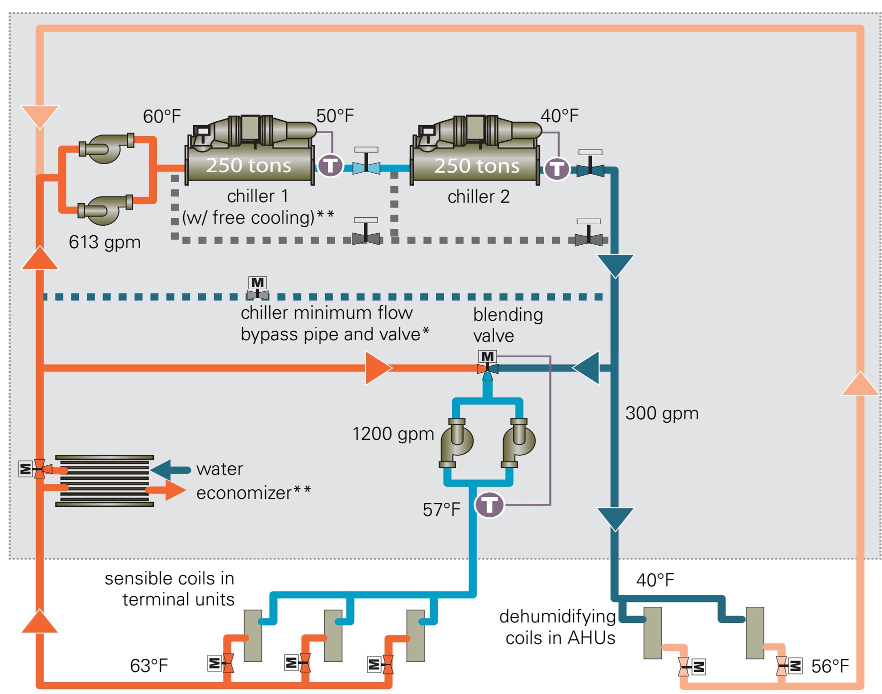
Figure 3. Two-chiller plant (loads in parallel, chillers in parallel)



* included if variable-flow chiller pumps are used

** some systems configure one chiller as a "free-cooling" chiller, while others include a separate plate-and-frame heat exchanger for water economizing

Figure 4. Two-chiller plant (loads in parallel, chillers in series)



* included if variable-flow chiller pumps are used

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Table 1. Comparison on dual-temperature plant configurations that include two chillers

		chiller evaporator flow rate, gpm	evaporator entering water temperature, °F	evaporator leaving water temperature, °F	chiller load, tons	chiller efficiency[1], kW/ton	chiller power, kW	evaporator pressure drop, ft. H_2O	incremental pumping power[2], kW	
loads in parallel, chillers in parallel (Figure 3)	Chiller 1	306.5	60	40	250	0.569	142.4	6.9	0.57	
	Chiller 2	306.5	60	40	250	0.569	142.4	6.9	0.57	
	Totals				**500**		**284.8**	**+**	**1.1**	**= 285.9 kW**
loads in parallel, chillers in series (Figure 4)	Chiller 1 (upstream)	613	60	50	250	0.462	115.6	12.9	2.13	
	Chiller 2 (downstream)	613	50	40	250	0.564	141.2	13.4	2.22	
	Totals				**500**		**256.8**	**+**	**4.4**	**= 261.2 kW**
loads in series, chillers in parallel	Chiller 1	261	63	40	250	0.570	142.6	5.1	0.36	
	Chiller 2	261	63	40	250	0.570	142.6	5.1	0.36	
	Totals				**500**		**285.2**	**+**	**0.7**	**= 285.9 kW**
loads in series chillers in series (Figure 5)	Chiller 1 (upstream)	522	63	51.5	250	0.445	111.3	9.7	1.36	
	Chiller 2 (downstream)	522	51.5	40	250	0.565	141.3	10.1	1.42	
	Totals				**500**		**252.6**	**+**	**2.8**	**= 255.4 kW**
loads in series (split flow) (Figure 7)	Chiller 1 (upstream)	1500	62	57	288	0.399	114.7	16.8	6.79	
	Chiller 2 (downstream)	300	57	40	212	0.586	124.2	6.7	0.54	
	Totals				**500**		**238.9**	**+**	**7.3**	**= 246.2 kW**
dedicated chillers (Figure 8)	Chiller 1 (warm)	1200	63	57	300	0.401	120.3	11.1	3.59	
	Chiller 2 (cold)	300	56	40	200	0.592	118.4	6.6	0.54	
	Totals				**500**		**238.7**	**+**	**4.1**	**= 242.8 kW**

[1] Full-load efficiency based on a 2.0 gpm/ton condenser water flow rate and 85°F water entering the condenser.

[2] This is the incremental power required to pump the water through the chiller's evaporator, assuming 70% pump efficiency [incremental pump kW = (0.746 kW/hp) x (evaporator flow rate, gpm x evaporator pressure drop, ft. H_2O) / (3960 x pump efficiency)].

Another benefit of configuring the chiller evaporators in series is that it simplifies sequencing (turning chillers on and off) in a variable-primary flow (VPF) system.[3]

The drawback of configuring chillers in series is that it is more challenging to provide redundancy if one chiller is not operational. With the chillers in parallel, both chillers are likely to be identical, and can be selected for a little extra capacity so that one chiller operating alone can provide a 60 or 70 percent of the design capacity while the other chiller is serviced.

To provide redundancy with chillers in series, the upstream chiller may need to be selected for "less-than-optimal" performance so that it is capable of producing 40°F water if the downstream chiller is not operational. Unless the upstream chiller is a free-cooling chiller (see p.9), arranging the condensers in a series-counterflow configuration can provide this redundancy without sacrificing performance. [4]

In addition, a set of bypass pipes and shutoff valves need to be added to the plant to enable either of the chillers to operate while the other chiller is serviced (see Figure 4).

Blending valve with loads in series (chillers in parallel or series). When the temperature of the water returning from the dehumidifying coils (56°F in this example) is colder than the water being supplied to the sensible-only terminal units (57°F), some designers consider configuring these **loads in series** (Figure 5). In this configuration, 40°F water from the chiller plant is first distributed to the dehumidifying coils. The 56°F water returning from these coils is then blended with warm water (63°F) returning from the terminal units (and maybe some additional 40°F water from the chiller plant) to produce 57°F water for the terminals.

Configuring the loads in series offers an efficiency benefit compared to the loads in parallel (255.4 kW with chillers in series, versus 261.2 kW with the loads in parallel and chillers in series), since the chiller plant flow rate is lower (522 gpm compared to 613 gpm with loads in parallel) and the larger plant delta T allows the upstream chiller to be selected for a warmer leaving-water temperature.

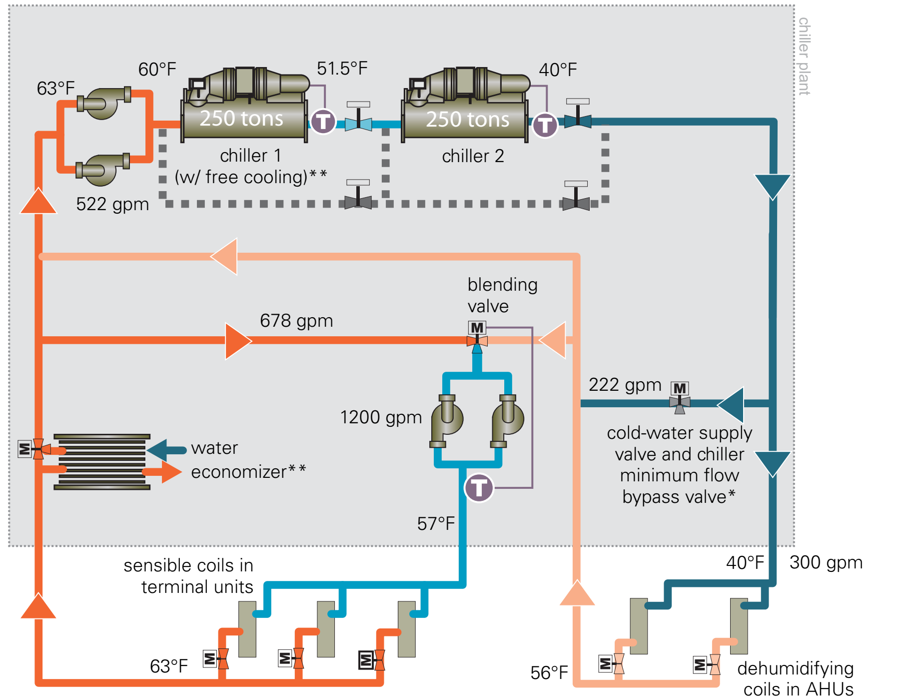
However, this configuration does introduce some added control complexity. The chiller minimum flow bypass valve now has a second purpose. In addition to ensuring minimum evaporator flow through any operating chiller, it must also ensure an adequate supply of cold water at the blending valve.

At design load for this example, 300 gpm of 40°F water is supplied to the dehumidifying coils, which returns at 56°F. Blending only this 300 gpm of 56°F water with 63°F water returning from the terminal units results in a 61°F supply temperature to the terminals (Figure 6, top).

So the bypass valve must open to mix 222 gpm of 40°F water with the 300 gpm of 56°F water returning from the dehumidifying coils, to ensure that the blending valve has enough cold water to deliver the required 57°F water to the terminals (Figure 6, bottom). This amount of bypass varies as the control valves on the dehumidifying coils modulate or close, and may require the bypass pipe to be larger than if it were only needed to ensure minimum evaporator flow.

For this loads-in-series configuration, the chiller evaporators can be piped either in parallel or series. Similar to the previous example, configuring the chiller evaporators in series results in 11 percent less total power (255.4 kW versus 285.9 kW with the chiller evaporators in parallel, see Table 1). But as mentioned, configuring the chillers in series makes it more challenging to provide redundancy in case one chiller is not operational.

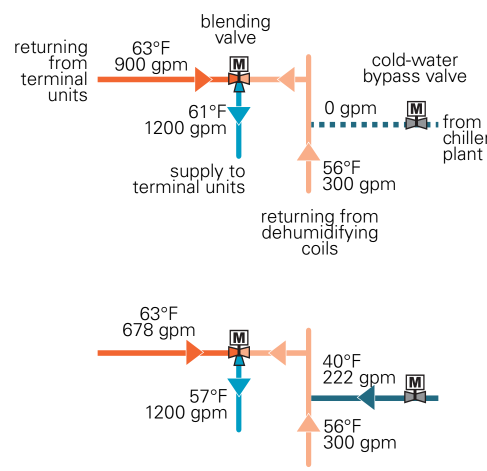
Figure 5. Two-chiller plant (loads in series, chillers in series)



* used to ensure adequate supply of cold water for sensible coils when dehumidification AHU coil control valves are modulating or closed AND also used to ensure minimum evaporator flow if variable-flow chiller pumps are used

** some systems configure one chiller as a "free-cooling" chiller, while others include a separate plate-and-frame heat exchanger for water economizing

Figure 6. Cold-water bypass when loads in series



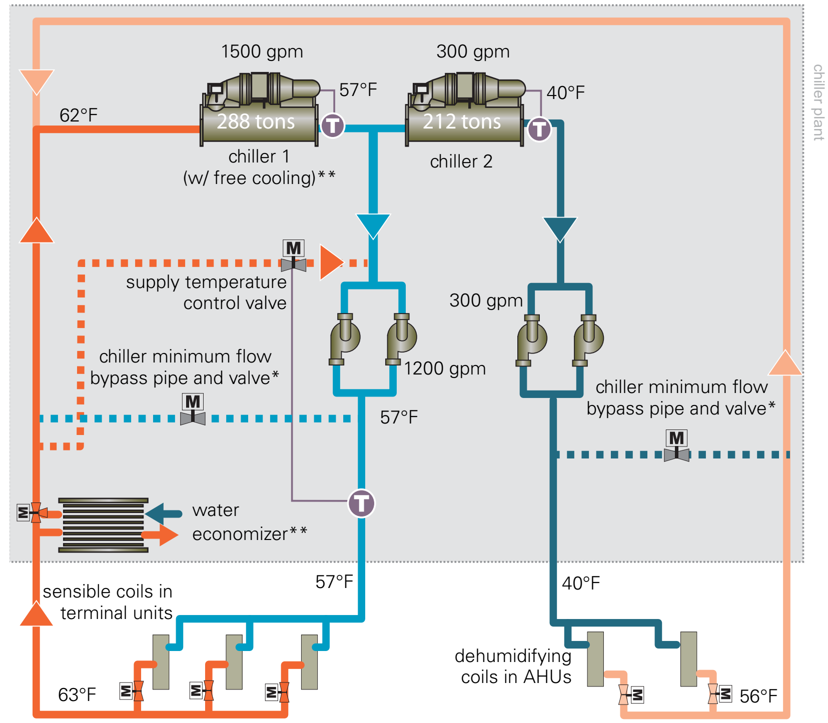
Chillers in series with split flow. With the chiller evaporators configured in series, some designers consider splitting the flow between the two chillers (Figure 7). In this example, the upstream chiller cools all the water to 57°F. Some of this water is then distributed directly to the terminal units for space sensible cooling, while the rest passes through the downstream chiller and is cooled to 40°F for the dehumidifying coils.

Since the upstream chiller operates at an even warmer leaving-water temperature (57°F), it is more efficient (0.399 kW/ton). However, because there is no blending, the full flow rate required by the terminal units (1200 gpm) plus the flow rate required by the dehumidifying coils (300 gpm) passes through the upstream chiller, resulting in a higher evaporator pressure drop and pump energy penalty.

This configuration is an attempt to maximize the efficiency of the upstream chiller, since it does not cool the water intended for the terminal units any more than needed. However, if the water returning from the dehumidifying coils (at 56°F in this example) is cooler than the water leaving the upstream chiller (57°F), there is energy wasted. The 56°F return water is mixed with 63°F water returning from the terminal units and passes through the upstream chiller. Rather, it would be more efficient to simply direct that 56°F return water right into the downstream chiller.

So splitting the flows provides an efficiency benefit for the upstream chiller, but directing all of the flow through the upstream chiller is inefficient unless the water returning from the dehumidifying coils is significantly warmer than the water leaving the upstream chiller…which is unlikely.

Figure 7. Two-chiller plant (chillers in series with split flow)



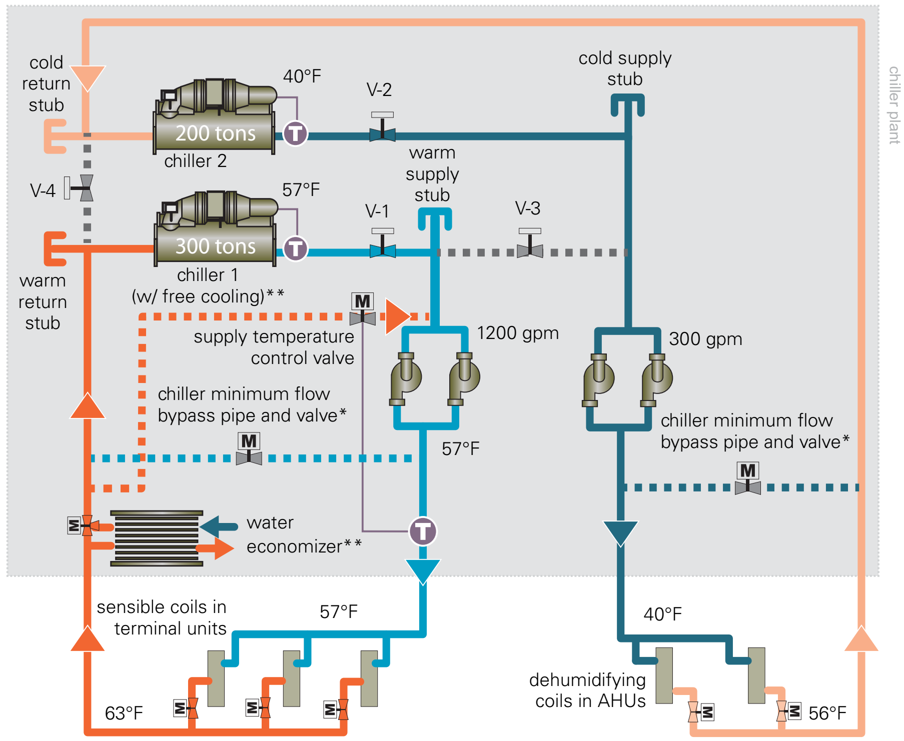
* included if variable-flow chiller pumps are used

** some systems configure one chiller as a "free-cooling" chiller, while others include a separate plate-and-frame heat exchanger for water economizing

For this example, the total power for both chillers plus incremental pumping power is 6 percent lower than the loads-in-parallel/chillers-in-series configuration, and 4 percent lower than the loads-in-series/chillers-in-series configuration (Table 1). Splitting the flows does avoid the complication of controlling the chiller minimum flow bypass valve to also bypass cold water if needed, but the interaction of the two pumping systems may be difficult to balance and control. Plus, it provides very little redundancy since the chiller flow rates and water temperatures are drastically different.

This configuration requires two separate chiller bypass pipes with minimum flow control valves. Figure 7 also shows an additional temperature control valve for the 57°F water loop. Since most sensible-only terminal units cannot allow condensation, the water supplied to them must always remain at a temperature above the space dew point temperature. Depending on how accurate and stably the upstream chiller is able to control its leaving-water temperature, the designer may want to include this additional valve to ensure that too cold of water is never distributed to the terminal units. Under low load conditions, if the chiller has reached its minimum stage of capacity and begins to produce water that is colder than the desired 57°F setpoint, this valve will blend in a small amount of warm return water.

Figure 8. Two-chiller plant (dedicated chillers with hydraulic connections)



* included if variable-flow chiller pumps are used

** some systems configure one chiller as a "free-cooling" chiller, while others include a separate plate-and-frame heat exchanger for water economizing

Dedicated chillers. In this final configuration, one chiller supplies 57°F water directly to the terminal units, while a separate chiller supplies 40°F water directly to the dehumidifying coils (Figure 8).

The benefit of this configuration is that it maximizes the efficiency of the "warm-water" chiller. Only the flow rate required by the terminal units passes through it and this water is only cooled to the required 57°F setpoint (not over-cooled). The "cold-water" chiller handles only the flow required by the dehumidifying coils.

The total power for both chillers plus incremental pumping power is the lowest of all the configurations analyzed—242.8 kW (Table 1).

In order to provide redundancy, two interconnecting pipes and shutoff valves can be added to the plant to enable either of the chillers to operate while the other chiller is being serviced (Figure 8). The plant can still supply both water temperatures with just the remaining chiller operating:

- If the "warm-water" chiller (chiller 1) fails, valve V-1 is closed, valves V-3 and V-4 are opened, the "cold-water" chiller (chiller 2) continues operating to produce 40°F water, and the supply temperature control valve blends in warm return water to deliver 57°F water to the terminal units.
- If the "cold-water" chiller (chiller 2) fails, valve V-2 is closed, valves V-3 and V-4 are opened, the "warm-water" chiller (chiller 1) continues operating with a leaving-water setpoint reset to 40°F, and the supply temperature control valve ensures that 57°F water is delivered to the terminal units.

Another option could be to include pipe stubouts in the chiller plant to enable quick connection of an emergency rental chiller, if needed (see Figure 8). The interconnecting pipes and shutoff valves would allow the chiller plant to "limp along" and provide partial capacity until the emergency chiller arrives (often within 24 hours).

The interconnecting pipes also provide the opportunity to operate only one of the chillers if the combined loads are very low, which may improve overall plant efficiency.

As in the previous configurations with chillers in series, the "warm-water" chiller may need to be selected at "less-than-optimal" performance so that it is capable of producing 40°F water if the "cold-water" chiller is not operational. This might be a good application for a chiller with a positive-displacement compressor (e.g., helical-rotary or scroll) or a centrifugal chiller with a VFD; both of which would be well-suited for operating at either water temperature, if required.

Like with the previous split-flow configuration, this configuration also requires two separate chiller bypass pipes with minimum flow control valves, and the temperature control valve for the 57°F water loop is probably also a good idea. But it does not have the added complication of controlling the chiller minimum flow bypass valve to also bypass cold water (as required in the loads-in-series configuration). While the two sets of pumps are hydraulically separated when both chillers are operational, selection of the pumps should also ensure that the pumps will properly operate in an emergency mode.

Plant with Three or More Chillers

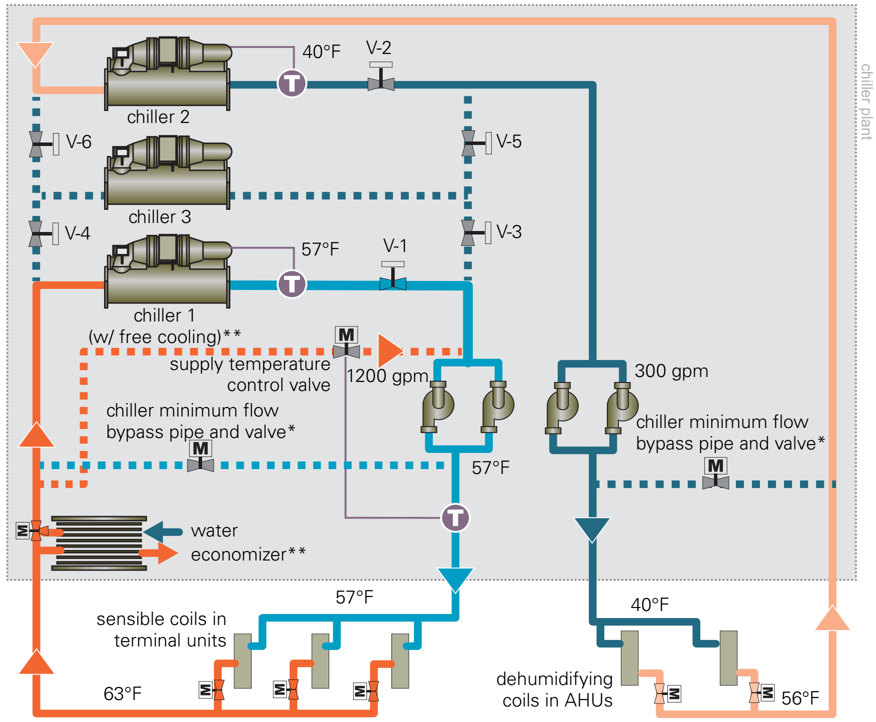
When the plant is designed to include three or more chillers, the most efficient configuration is likely to use the dedicated-chillers approach with interconnecting pipes and shutoff valves to provide redundancy.

In this configuration (Figure 9), one chiller is selected and optimized to supply 57°F water to the terminal units, while a separate chiller is selected and optimized to supply 40°F water to the dehumidifying coils. The third chiller is then selected so that it is capable of providing either 57°F or 40°F water, in the event that one of the other two chillers is in need of service:

- If the "warm-water" chiller (chiller 1) fails, valve V-1 is closed, valves V-3 and V-4 are opened, and chiller 3 is operated to supply 57°F water to the terminal units.
- If the "cold-water" chiller (chiller 2) fails, valve V-2 is closed, valves V-5 and V-6 are opened, and chiller 3 is operated to supply 40°F water to the dehumidifying coils.

Even if the chiller plant was originally conceived to include only two chillers, it may be desirable to design it for three chillers instead, so that the plant provides added redundancy in the event of a chiller failure. This would allow the "warm-water" chiller to be optimized to supply 57°F water, without sacrificing efficiency by having to select it to be capable of making 40°F in an emergency (as would be required in a plant with only two chillers).

Figure 9. Dual-temperature plant with third chiller for redundancy



* included if variable-flow chiller pumps are used

** some systems configure one chiller as a "free-cooling" chiller, while others include a separate plate-and-frame heat exchanger for water economizing

Incorporating a Water Economizer

Most buildings that include sensible-only terminal units use a dedicated outdoor air (OA) system for ventilation, so conventional air economizing is usually not possible. Therefore, a water economizer may be used to provide "free" cooling during mild weather.[5] This can be particularly valuable for systems that use chilled-water terminal units for space cooling, since interior zones may require a supply of water for cooling, even when it's cold outside. In this case, water economizers can allow the chillers to be turned off during the colder months of the year.

Water economizers are most prevalent in systems that use water-cooled chillers, but they can also be used in systems with air-cooled chillers.

Water economizing with water-cooled chillers. If water-cooled chillers are used, a water economizer is typically provided using either 1) a separate plate-and-frame heat exchanger, or 2) by configuring one of the water chillers as a "free-cooling" centrifugal chiller (i.e., a thermosiphon).

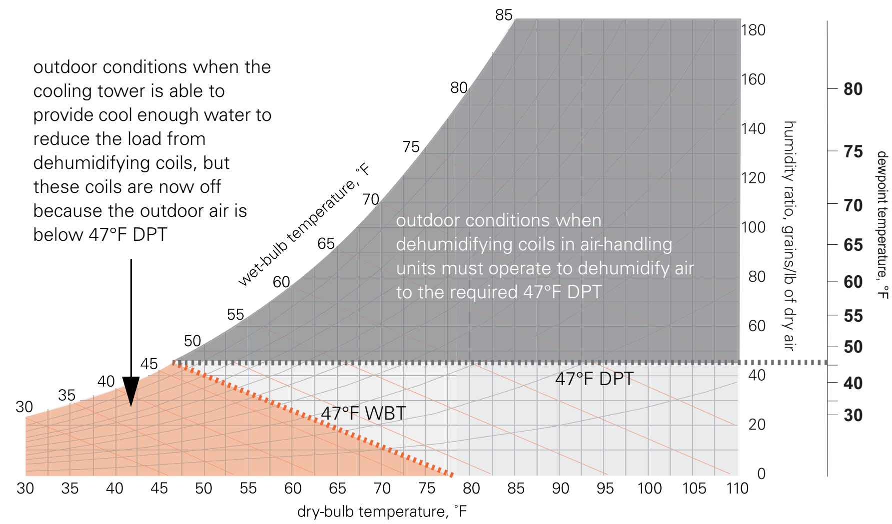
When a **plate-and-frame heat exchanger** is used, cold water from the cooling tower passes through one side of the heat exchanger, which cools the chilled water flowing through the other side. In most applications, locating this heat exchanger to pre-cool the warm water returning from the terminal units provides the greatest benefit (see Figures 3-9). This location is where the chilled-water loop is warmest, so the water economizer is able to reduce the chiller load any time the cooling tower is able to produce water that is colder than about 61°F for this example (63°F return water minus a 2°F heat exchanger approach).

Some engineers express concern that, in this location, the water economizer can only reduce the cooling load from the terminal units, but not from the dehumidifying coils. In this example, for the water economizer to reduce the load from the dehumidifying coils, the cooling tower must be able to produce water that is colder than 54°F (56°F return water minus a 2°F approach). This would likely require the outdoor wet-bulb temperature to be below 47°F (assuming a 7°F cooling tower approach); meaning that the corresponding outdoor dew point would be no higher than 47°F (Figure 10).

That is, at conditions when the cooling tower is capable of producing water that is cold enough to reduce the load from the dehumidifying coils, the outdoor dew point is likely below the setpoint of the dedicated OA system, so the dehumidifying coils would be off (Figure 10). Therefore, in most applications where sensible-only terminal units are used, there is likely little or no load from the dehumidifying coils when water economizing is available. (The exception is a system with many operating hours when it is hot and dry outside, such that the coils in the dedicated OA units still need to cool the outdoor air.)

In many applications, it may be more desirable to use a **free-cooling chiller**, which would avoid the added cost of the plate-and-frame heat exchanger and added maintenance required to clean it. In a centrifugal chiller, heat can be transferred inside the chiller via refrigerant migration without needing to operate the compressor (i.e., a thermosiphon).

Figure 10. Water economizing likely not useful for dehumidifying coils



When the temperature of the water entering the condenser (from the cooling tower) can be colder than the desired temperature leaving the evaporator, the compressor is turned off and valves inside the chiller refrigeration circuit are opened.[5] Refrigerant vapor from the evaporator migrates directly to the condenser, where the temperature is cooler and, thus, the refrigerant pressure is lower. After the refrigerant condenses, another opened valve allows the liquid refrigerant to flow, by gravity, back into the evaporator. This allows the refrigerant to circulate between the evaporator and condenser without needing to operate the compressor.

A free-cooling chiller may be able to produce up to 45 percent of its design capacity without compressor operation, particularly in systems that use warmer leaving-evaporator temperatures. Therefore, in a plant with dedicated chillers (Figures 8 or 9), configuring the "warm-water" chiller (chiller 1) for free cooling would provide the most benefit. Or in a plant with chillers in series (Figures 4, 5, or 7), configuring the upstream chiller (chiller 1) for free cooling would provide the most benefit.

Water economizing with air-cooled chillers. If air-cooled chillers are used, a water economizer is typically provided using either 1) an air-to-water heat exchanger incorporated into the air-cooled chiller (typically mounted on the outside of the air-cooled condenser coils), or 2) a separate air-to-water heat exchanger ("dry cooler" or closed-circuit cooling tower).

The benefit of incorporating the heat exchanger into the air-cooled chiller is that it usually comes factory-assembled with integrated controls. The drawback is that if the plant includes only one chiller, the water economizer is not able to provide any energy-saving benefit until it is cold and dry enough outside that the dehumidifying coils are shut off and the chiller setpoint is reset up to 57°F.

Using a separate dry cooler, piped in to pre-cool the warm water returning from the terminal units (see Figure 2), typically results in more cooling energy savings since it can reduce the cooling load from the terminal units whenever the outdoor dry-bulb temperature is about 10°F cooler than the water returning from the terminal units (63°F - 10°F = 53°F). Plus, the dry cooler fans only operate when water economizing occurs; whereas if the air-to-water heat exchanger is incorporated into the air-cooled chiller, the condenser fans have to overcome the added pressure drop of those coils any time the chiller is operating.

For a single-chiller plant that uses a glycol-isolation heat exchanger (see Figure 1), a separate dry cooler would likely be piped into the glycol side of the heat exchanger for freeze protection. This would likely result in less cooling energy savings than if it were piped in the warmer, return-water pipe from the terminal units. But once it's dry enough outside that the dehumidifying coils are shut off, the chiller setpoint can be reset up to 55°F (57°F supply water to terminal units minus a 2°F heat exchanger approach) and the water economizer more efficiently reduces the cooling load from the terminal units. In this case, incorporating the water economizer into the air-cooled chiller may be preferred, since there is less benefit from using the separate dry cooler.

Summary

In systems that use sensible-only, chilled-water terminal units, a dual-temperature chiller plant that uses separate water chillers can efficiently provide different water temperatures for both space sensible cooling and dehumidification. Interconnecting pipes and valves can allow the same plant to provide redundant capacity if either of the chillers needs to be repaired, replaced, or serviced. Including more than two chillers allows for more optimal chiller selections and can provide efficient operation at low loads.

By John Murphy applications engineer, Trane. You can find this and previous issues of the Engineers Newsletter at www.trane.com/engineersnewsletter. To comment, send e-mail to ENL@trane.com.

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ADM-APN055-EN (September 2015)