WHITEPAPER

Design Strategies for Maximizing Efficiency and Performance of Waterside Economizers

By JMP Equipment Company

WHITEPAPER

According to Energy Star, *waterside economizers*, the focus of this whitepaper, are best suited in climates where the wet bulb temperature is below 55°F for 3000 hours or more. Since that covers all but the extreme Southwest and lower Southeast, it is important that HVAC engineers have waterside economizing design in their repertoire.

The general application guidelines for waterside economizers per ASHRAE 90.1-2010 and 2013 are:

• Water economizer systems shall be capable of cooling supply air by indirect evaporation and providing up to 100% of the expected system cooling load at outdoor air temperatures of 50°F dry bulb/45°F wet bulb and below.

• Economizer systems shall be *integrated* with the mechanical cooling system and be capable of providing partial cooling even when additional mechanical cooling is required to meet the remainder of the cooling load.

• Pre-cooling coils and water-to-water heat exchangers used as part of water economizer systems must either have a water-side pressure drop of less than 15 ft. or a secondary loop must be created so that the pressure drop will not impact the circulating pumps when the system is operating in non-economizer mode.

This whitepaper explores how the above changes impact the way engineers design waterside economizers and what strategies can and should be used for best success.

Integrated Economizer Design

We know that ASHRAE says that waterside economizer must be "integrated" with mechanical cooling, but what exactly does that mean?

It means that chillers and economizers have to be able to operate *simultaneously* whenever outdoor conditions are suitable for free cooling. And since either/or operation is not an option, engineers can no longer design economizers and chillers in parallel. A single pipe design that simply redirects flow from the chiller to the economizer for part of the year is *not* an integrated design.

Control is also more complicated. Keep in mind that economizers need cold water to operate efficiently. Chillers, on the other hand, have a minimum condensing temperature, so the water temperature can only drop so low or else the chiller will not operate properly. This means that you may require two different cooling tower cold water supply temperatures.

How Wet Bulb Impacts Cooling Tower Performance

It may seem counterintuitive but a larger cooling tower may be required to perform the same amount of cooling when the outdoor air is cold than when it is hot. Remember, cooling towers cool condenser water through the process of evaporation and water simply doesn't evaporate very much when it is cold outside because cold air holds less water. So when the wet bulb temperature drops outside, a cooling tower is essentially handicapped.

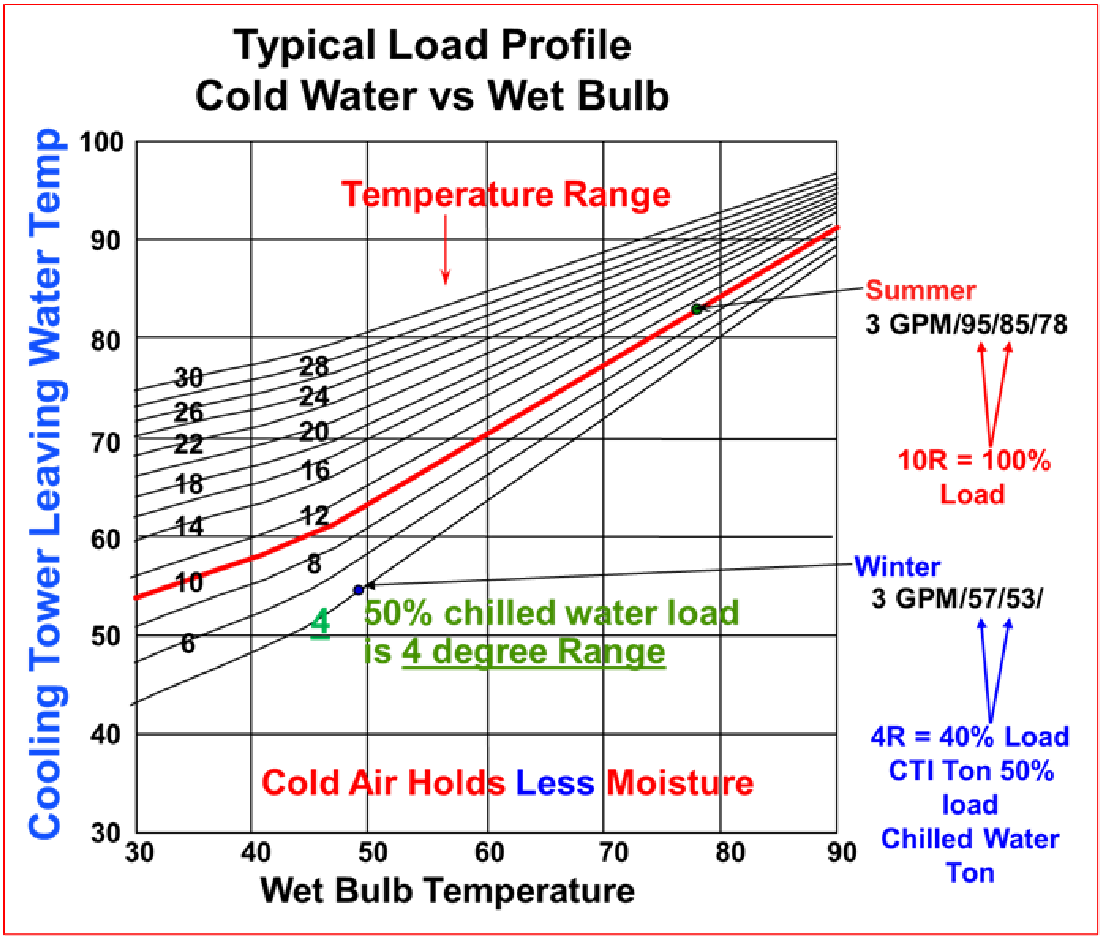


Figure 1

Figure 1 illustrates the degree to which cold air impacts the performance of a cooling tower.

Let's say we have a cooling tower that is selected for peak a summertime condition of 78°F wet bulb. Looking at the load profile above we see that we can cool the water from 95°F to 85°F running at 100% load and get a 7 degree approach. In other words, we can get within 7°F of the wet bulb temperature. Not bad! But what happens in the winter when the wet bulb temperature drops to 50°F?

If we maintain 100% load on the cooling tower, the coldest water we can make is about 63°F. Now our approach has gone from 7 degrees to 13 degrees. The tower's ability to cool water has almost been cut in half. And it gets worse as the wet bulb drops.

That makes things a little challenging when trying to meet ASHRAE's 90.1 – 2010, which requires economizing whenever the outdoor temperature is 50°F dry bulb/45°F wet bulb or below. Does that mean cooling towers should be selected based on the coldest days of winter instead of the hottest days of summer? While it might work theoretically, it's neither practical nor economical.

Consider a cross flow tower selected for 79°F wet bulb, with 95°F water going to the cooling tower and leaving at 85°F. That's a 10-degree range and a 6-degree approach. But what happens if we want to operate that same cooling tower in the winter with 60°F entering water and maintain a 10-degree range? The wet bulb would have to be 31.85°F and our approach would be 18.18 degrees! Practically speaking, how often is that going to happen? Not very often, so your opportunities for free cooling/ economizing during the winter would be drastically reduced.

A better solution would be to size the tower for peak summer hours but operate it at part load during colder conditions. The chiller and the tower/economizer work *together* to maintain system supply temperatures, while still maintaining a reasonable approach. This, however, requires that the cooling tower(s) be piped *in series* with the economizer.

Cooling Tower Piping Strategies

Waterside economizing relies on cold water from the cooling tower (condenser water) to absorb some or all of the heat from inside the building. In typical waterside economizing applications, plate and frame heat exchangers are used to transfer heat from the return chilled water to the cooling tower water.

It is a pretty simple process compared to what occurs in a chiller, where a refrigerant loop sits between the supply/return cold water and the condenser water. The refrigerant loop includes an expansion valve and compressor which are there to facilitate and maximize the heat transfer from the return cold water to the refrigerant, and then from the refrigerant to the condenser water.

A chiller's condenser water temperature can vary greatly depending on whether it is a hermetic or open chiller. This is all because of head pressure control. The thresholds vary slightly from manufacturer to manufacturer, but generally speaking hermetic chillers operate best when the entering condenser water is 75°F or above and open chillers operate best when the entering condenser water is at 55°F or above. This is because of the direct proportional relationship between pressure and temperature. As one increases or decreases, so does the other. So if the condenser water is cold (e.g. 55°F) it will be at a lower pressure.

This presents a problem because the condenser needs that extra head pressure to force the refrigerant through the

Cross flow tower at **100% load selected for** 79 degrees wetbulb, 95° hot water and 85° cold water with a 10 degree range and a **6 degree approach**.

Typical cooling tower cold water vs wet bulb approach temperatures with a constant 60° supply hot water temperature base on % load.

60 degree supply hot water to the tower

60	degree hot water
10	degree range
100%	**load**
18.18	**degree approach required**
31.85	degree web bulb (not practical)

60.0	degree hot water
7.5	degree range
75%	**load**
52.5	degree cold water
12.43	**degree approach**
40.07	degree web bulb

60.0	degree hot water
5	degree range
50%	**load**
55.0	degree cold water
7.54	**degree approach**
47.46	degree web bulb

60.0	degree hot water
2.5	degree range
25%	**load**
57.5	degree cold water
3.49	degree approach
54.01	degree web bulb

expansion valve along with oil during the refrigeration cycle. That's why it is always important to verify the minimum condenser water temperature that a chiller can handle.

Why You May Need Two Different Condenser Water Supply Temperatures

When it comes to waterside economizing, the colder the condenser water the better. Chillers, on the other hand, have their limitations when it comes to cold condenser water. This isn't a problem if a system always operates in either chilled water mode or economizing mode. But remember, ASHRAE 90.1-2010 requires that waterside economizing occur anytime the outdoor temperatures are 50°F dry bulb/45°F wet bulb or below--even when 50°F dry bulb/45°F wet bulb is not sufficient to cool the chilled water low enough to handle the load. More often than not, the chiller(s) and the waterside economizer must operate simultaneously. That means the design might need to provide two different supply temperatures from the cooling towers – one for the economizer and one for the condenser.

In most cases, one of the following approaches will work.

In Figure 2 notice that there is a dedicated cooling tower for the waterside economizer and one chiller, while the other two chillers are manifolded to the other two cooling towers. The waterside economizer loop is separated from the chillers, but the functionality is integrated thanks to the three-way valve. In this arrangement, the condenser water from the dedicated cooling tower will bypass its associated chiller and flow through the plate & frame heat exchanger when the system is in economizing mode. That cooling tower can be controlled to supply the coldest water possible, while the other cooling towers will supply warmer water to the two operating chillers that simultaneously share the cooling load with the economizer.

Another way to integrate economizer and chiller operation is to use one common supply and return pipe between the towers and the chillers (Figure 3).

This will likely require a 2-way modulating valve on the condenser water piping to maintain head pressure on the chillers. However, you should check with the chiller manufacturer to make sure the chiller will function properly with modulating condenser water flow.

Chilled Water Piping Requirements

Now let's consider some options for piping the chilled water side in an economizing application. Remember, we want to pipe the waterside economizer heat exchanger so that we can run both the chiller(s) and the economizer at the same time.

In the past most systems were installed with the heat exchanger piped in parallel with the chillers. (Figure 4)

However, this piping arrangement allows for only the chiller(s) or the economizer to operate at one time to independently to satisfy the demand. Not only does this not meet ASHRAE's latest requirements, it limits waterside economizing to periods

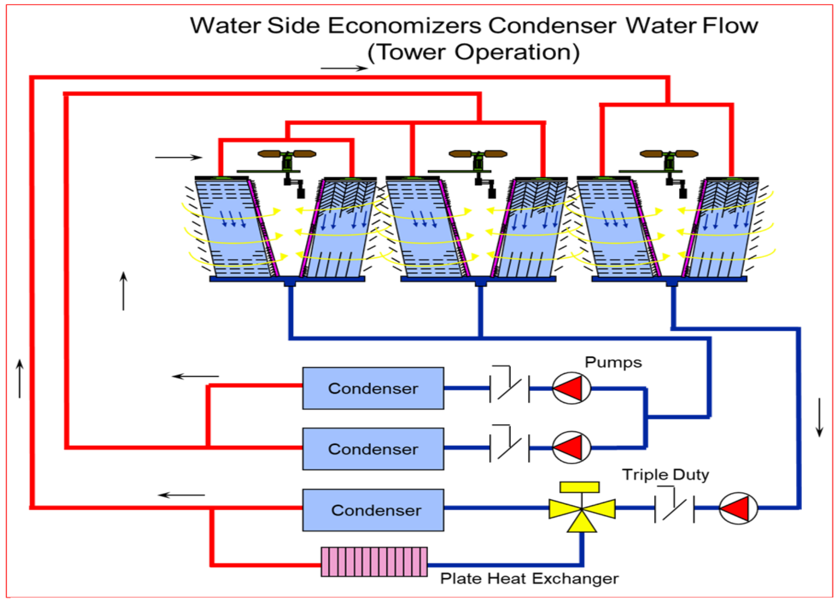


Figure 2

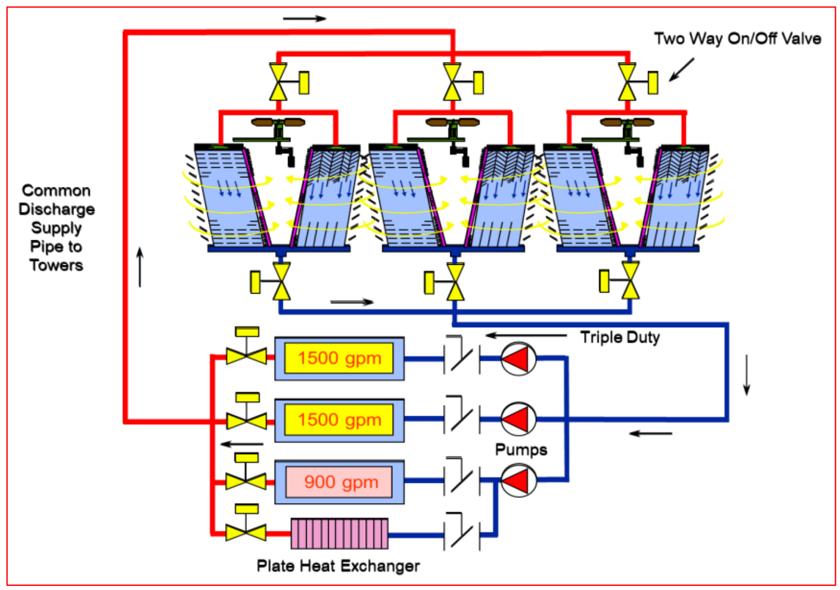


Figure 3

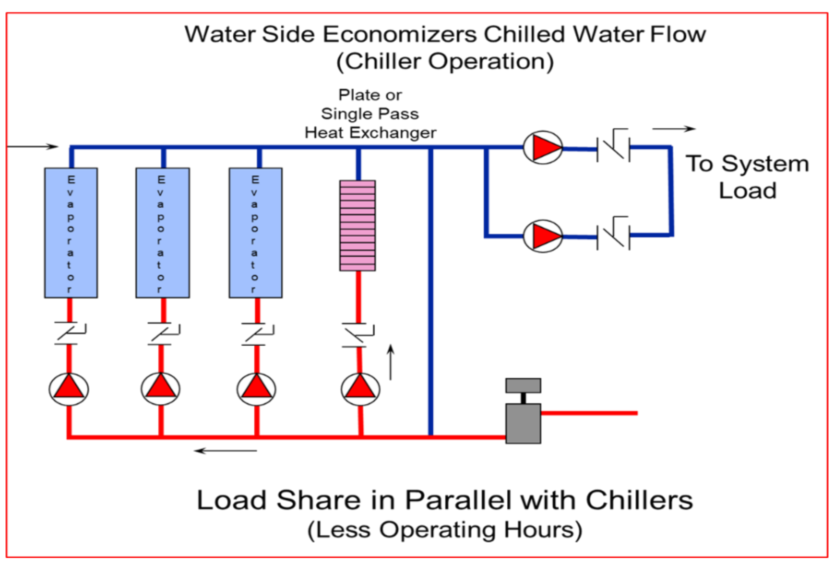


Figure 4

when the wet bulb temperature is low enough to meet the *entire* cooling load.

Piping the exchanger in series with the chillers instead of in parallel will give us many more hours of economizing. This can be achieved by piping the waterside economizer between the return chilled water from the system and the chillers so that you can pre-cool the chilled water *before* it enters the chillers. This simultaneous operation of chillers and waterside economizer allows for many more hours of free-cooling and a much faster payback on the economizer equipment.

The example in Figure 5 shows a chiller plant with a waterside economizer providing 250 tons of cooling and the evenly loaded chillers providing 750 tons of cooling. We have 3000 GPM of flow in our primary loop with 2000 GPM going out to the system to deliver 1000 tons of cooling. The return chilled water enters the heat exchanger at 57°F and leaves at 54°F. This 54°F return water mixes with 1000 GPM of 45°F water in the chiller loop to make 51°F water entering the chillers with a 6ºF Delta T across the chillers.

Compare that to the exact same system and demand load, minus the waterside economizer (Figure 6):

Notice that each chiller in Figure 6 is 66% loaded with a Delta T of 8°F. By comparison, the system with the waterside economizer (Figure 5) can supply this same amount of cooling by loading each chiller to only 50%, which lowers the Delta T across the chillers to 6°F. We have shifted a significant portion of the load to the waterside economizer and we are still within reasonable operating conditions for our chillers.

Bear in mind that if the load drops any further in our system with the economizer in series, we may have to unload one of the chillers to keep return water temperatures from getting too low or the chillers could short cycle. Look at the

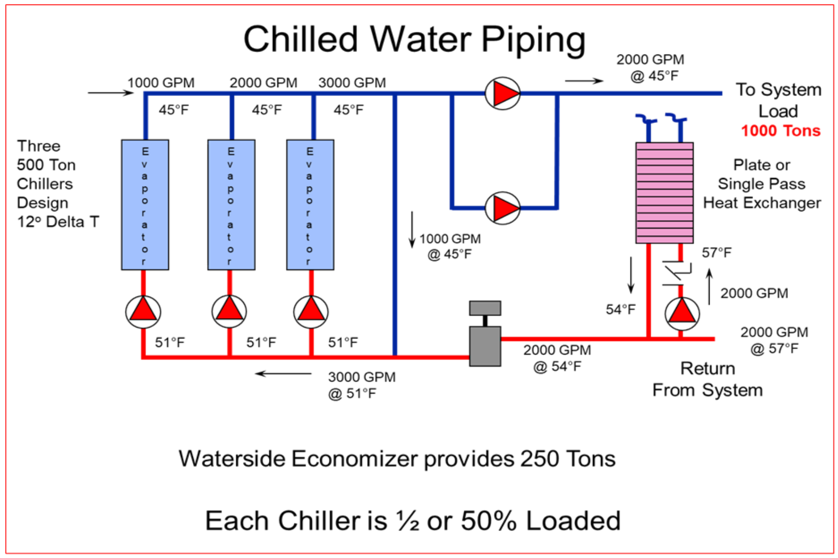


Figure 5

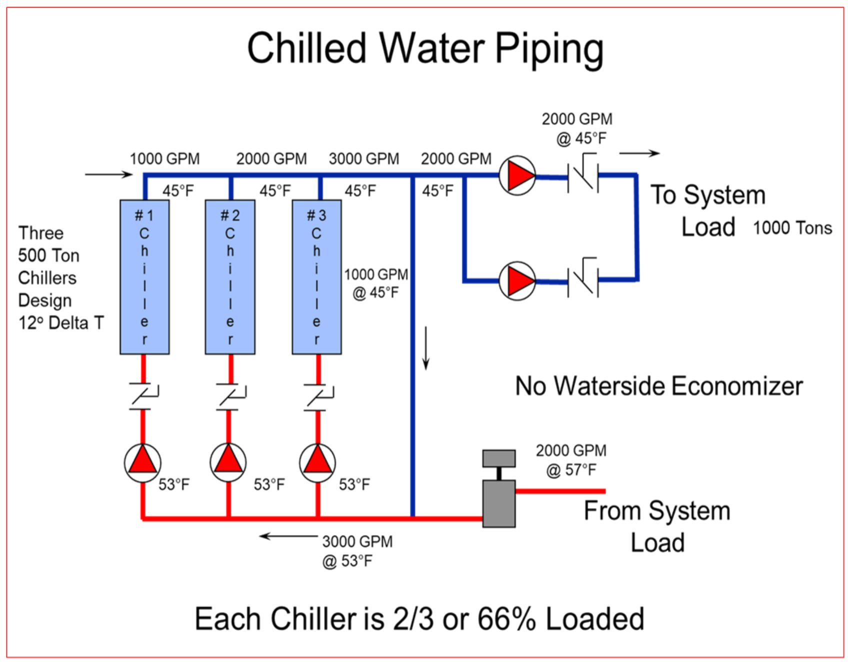


Figure 6

Piping the exchanger *in series* with the chillers instead of in parallel will give us many more hours of economizing.

same example with all three chillers at 17% load (Figure 7).

We are now at a 2°F Delta T and definitely in danger of short cycling. So while we want to offset as much chiller operation as we can by free cooling, we must be mindful of the minimum load that is required on chillers.

Heat Exchanger Selection

Waterside economizing typically involves the transfer of BTUs from an open loop cooling tower system to a closed loop chilled water system. A heat exchanger is required to prevent mixing of these fluids, but what type of heat exchanger is best?

Plate and frame (P&F) heat exchangers are usually the best choice for waterside economizers. These heat exchangers are comprised of multiple metal plates that are gasketed together and mounted onto a frame. There are inlet and outlet connections at the corners of the plates that allow the two fluids to enter on opposite sides. This allows the fluids to flow counter to each other though alternating channels of the assembly. This counterflow pattern yields a high rate of heat transfer, providing as close as a two degree approach between the condenser water and the chilled water. The counterflow design also allows a temperature cross (the difference between the condenser water and the chilled water temperature at their respective outlets) which is not possible with a typical U-tube heat exchangers.

Figure 8 shows an example of a P&F heat exchanger with a three degree approach and a seven degree temperature cross.

P&F heat exchangers are compact in size and can be disassembled for cleaning. The ability to be cleaned offsets the one disadvantage that P&F heat exchangers bring to an economizing application. The narrow flow channels on the plates are more vulnerable to clogging/fouling so care needs to be taken to try to keep it

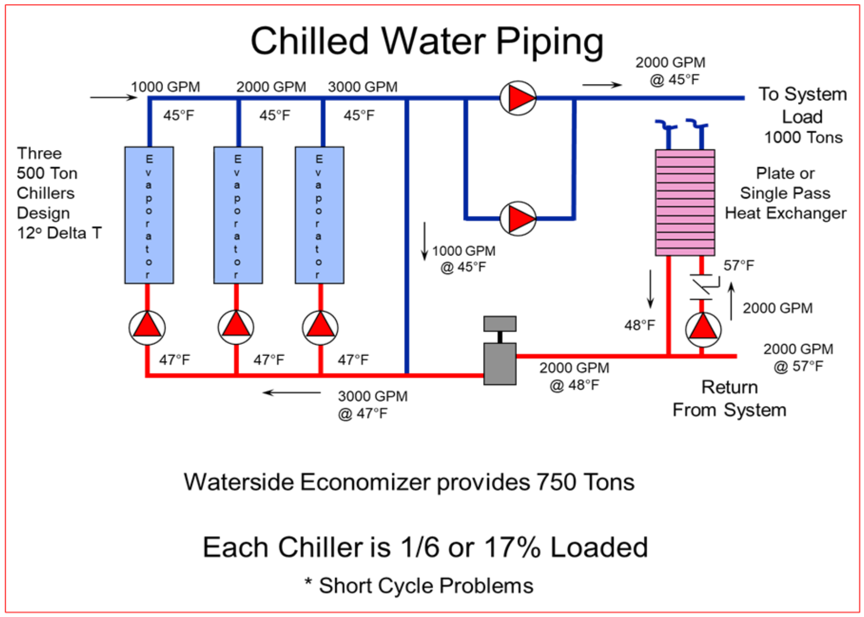
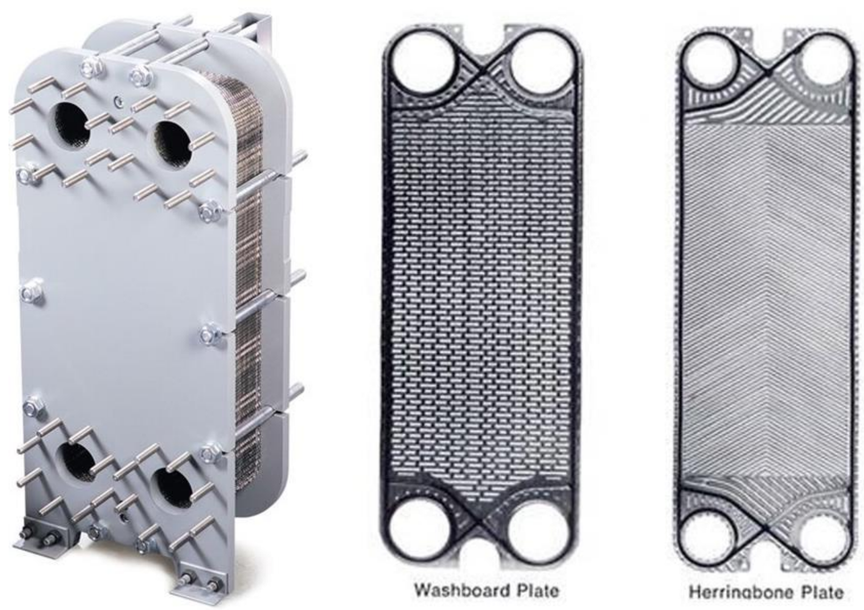


Figure 7



clean. The cooling tower water loop should be fitted with filtration to help minimize heat exchanger maintenance. This can be done as a full stream sediment separator in front of the heat exchanger or with a sweeping system inside the cooling tower basin.

Maximizing Plate & Frame Efficiency

Most P&F heat exchangers are "single pass, counterflow." In a single pass heat exchanger, water flows in one side of the heat exchanger and out the other side.

Counterflow means that the two fluids (in this case, condenser water and chilled water) flow in a counter direction to each other. This flow pattern reduces the required heat exchanger surface area.

The heat transfer rate of a heat exchanger is defined by the following equation:

Q = **U** x **A** x **LMTD**

Q = Heat Transfer Rate (BTU/hr.)

U = Overall Heat Transfer Coefficient (BTU/hr-ft^2-°F)

A = Cross-sectional Heat Transfer Area (ft^2)

LMTD = Logarithmic Mean Temperature Difference

LMTD is a logarithmic average of the temperature difference between the hot side and cold side fluids. The larger this difference, the greater the heat transfer rate, and the smaller the heat exchanger can be.

P&F heat exchangers have high U factors (1000 units or more) thanks to the turbulence produced by the many tight directional changes in the plated channels. Again, this helps reduce the size of the heat exchanger and thus first cost.

When comparing the performance of P&F heat exchangers for an economizing application, it is important that engineers only consider heat exchangers that have been certified under AHRI 400. This certification ensures that the heat exchanger has been tested to a strictly uniform criteria. It is also an ASHRAE 90.1(Sections 6.4.1.4) requirement for all plate type liquid-to-liquid heat exchangers used in commercial applications. An AHRI 400 certified heat exchanger is critically important in waterside economizing applications because it eliminates the chance that a heat exchanger specified for a 2°F approach temperature might actually have an approach somewhere between 2.0°F and 2.5°F or a higher than published pressure drop.

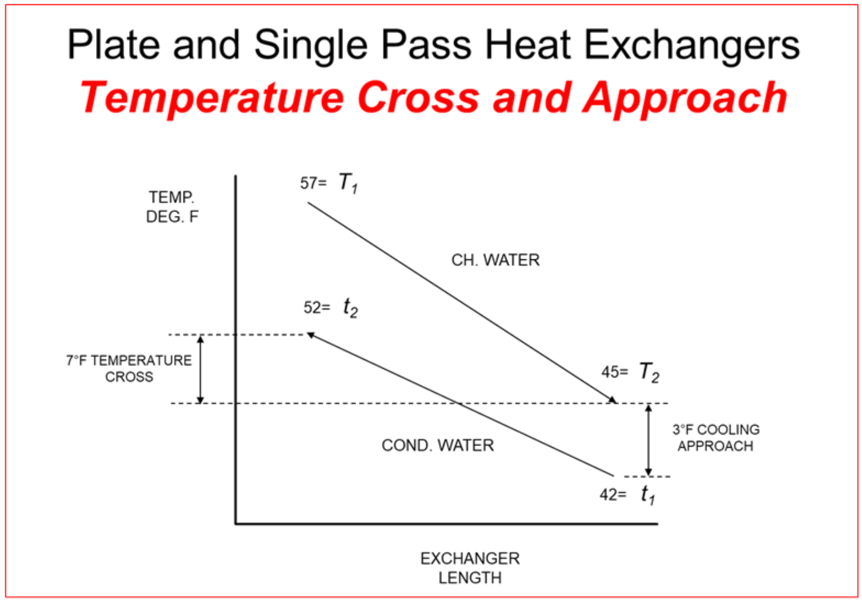


Figure 8

For maximum efficiency, we recommend that engineers select an AHRI 400 certified heat exchanger for a 2°F approach. (Note: In an economizing application, approach is the difference between the chilled water temperature leaving the heat exchanger and the temperature of the condenser water entering the heat exchanger.)

Why such a small approach temperature? Besides the obvious fact that you want to transfer as much heat as possible from the chilled water to the condenser water, it's important to understand that increasing the approach by just a *few tenths* of a degree will have a large impact on the size and capability of the heat exchanger. Increasing the approach from 2°F to 2.3°F for example could decrease the size of the heat exchanger by 15%. This reduction in heat exchanger capacity will translate into significantly higher operating cost for the owner.

Let's look at the numbers. The example below (Figure 9) shows the required heat exchanger surface for a P&F heat exchanger with the design parameters in the upper right, including a design approach temperature of 2°F. Using the equation referenced above, we've calculated a required heat exchanger surface of 2083 ft².

Now look what happens if we increase the approach by just few tenths (Figure 10).

As you can see the required area drops significantly with just a slight increase in approach temperature. And while a smaller heat exchanger may *sound* like a good thing, all it really means is that the chiller will have to work harder to compensate for the reduced capacity of the heat exchanger. A 2°F approach will keep you in the sweet spot between upfront cost and performance while also ensuring your heat exchanger can do the job!

Conclusion

Waterside economizing offers significant opportunities for increasing the efficiency of chilled water systems in nearly every part of the United States. With just a few exceptions, ASHRAE 90.1 – 2010 has effectively made water or airside economizing a requirement. Furthermore, the standard now requires that waterside economizers and chilled water systems be integrated to increase yearly economizing hours. The strategies outlined in this whitepaper not only help engineers meet this standard but ensure faster paybacks for owners.

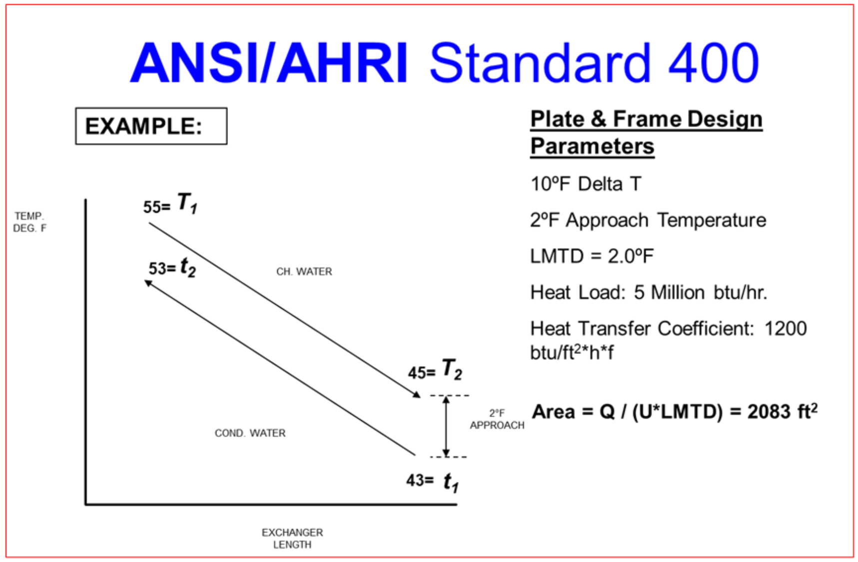


Figure 9

ANSI/AHRI Standard 400

Impact of LMTD temperature alone on Heat Exchanger Area

LMTD (degrees F)	Area Required (Ft^2)	Cost Index
2.0 **(Design Criteria)**	**2083.0**	**1.00**
2.2	1894.0	0.95
2.3	1812.0	0.93
2.5	1667.0	0.87

Results: Increase LMTD **0.2ºF, Lower cost 5%**

0.3ºF, Lower cost 7%

0.5ºF, Lower cost 13%

Figure 10