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How to Design & Control Waterside Economizers

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ASHRAE/IES Standard 90.1[1] requires either airside economizers or waterside economizers for most cooling system applications. Air economizers are more common and almost always more energy efficient. But there are some applications where water economizers are preferred, such as:

- High-rise office buildings with floor-by-floor air handlers where access to outdoor air is architecturally limited.
- Large data centers, where air economizers are not always physically practical because airflow rates are so high, or air economizers are precluded by concerns (valid or not) that they can increase particle contamination risk and cause low humidity levels. Water economizers can also provide energy savings on par with air economizers on data centers that have hot aisle containment and warm supply air temperatures, e.g., "cold" aisle supply air temperatures over 80°F (27°C) and hot aisles over 100°F (38°C). These warm air temperatures, in turn, allow chilled water supply temperatures over 65°F (18°C), readily attainable by cooling towers for most of the year in most climates.
- Small fan coil systems where airside economizers are physically impractical or overly expensive.
- Chilled beams, radiant cooling, and other hydronic systems for which airside economizers do not apply.

Here are some tips for designing cost effective and energy efficient waterside economizers:

System Design & Equipment Selection

- To maximize performance (and also to meet ASHRAE Standard 90.1 prescriptive requirements), the economizer must be "integrated" with the chillers, meaning the economizer has to be able to reduce the load on the chillers even if it cannot handle the entire load. This means the heat exchanger must be in series with, and upstream of, the chillers as shown in *Figure 2,* rather than in parallel with the chillers *Figure 1*. In the series position the economizer can pre-cool the return chilled water to reduce the load on the chillers even when the chillers must run to deliver the desired chilled water temperature. The first costs are basically the same as a nonintegrated economizer, but the number of hours the economizer can be operational is vastly extended, significantly improving energy performance and cost effectiveness. Integrated economizers are also easier to control, as described in more detail later. There are few, if any, advantages to nonintegrated economizers.
- As shown in *Figure 2,* the same cooling towers and condenser water pumps should be used to serve both the economizer heat exchanger and the chiller condensers. Some designers provide separate towers and pumps, at considerable expense because of concerns about low chiller head pressure due to low condenser water supply temperatures. But head pressure control is easily and inexpensively addressed by making the condenser isolation valves modulating and controlling them off the head pressure control signal output that is standard on most chiller controllers. The valves throttle flow through the condenser as needed to maintain chiller minimum lift regardless of how cold the condenser water supply temperature is in economizer mode.
- The capacity and quantity of cooling towers and condenser water pumps remains the same as they would be without the economizer. For office building applications, this is intuitively clear: we know that when the economizer is on, weather will be cold so loads will be well below design loads; hence only one of the two chillers (in the example shown in *Figure 2*) will be needed, freeing the other to sup-

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FIGURE 1 Nonintegrated economizer.

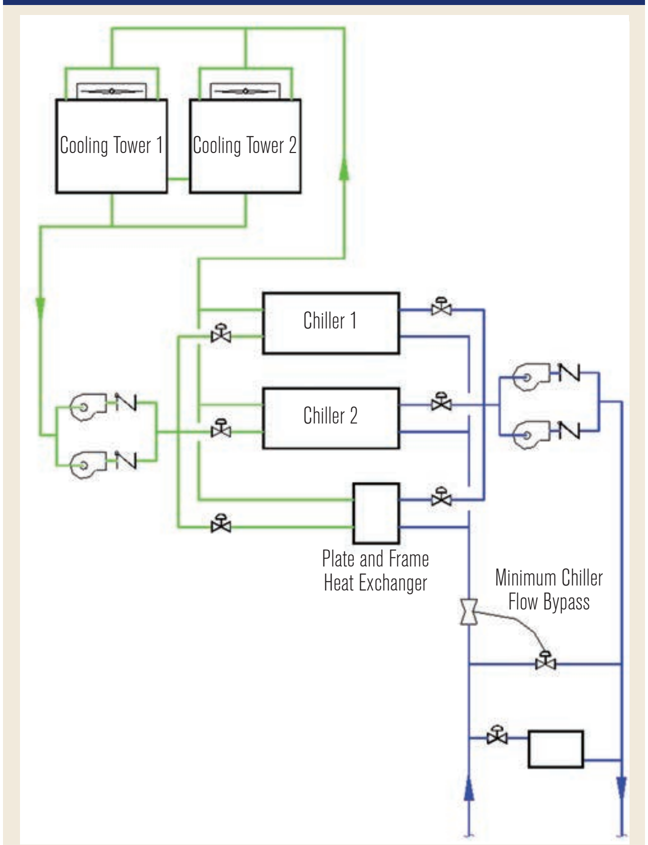
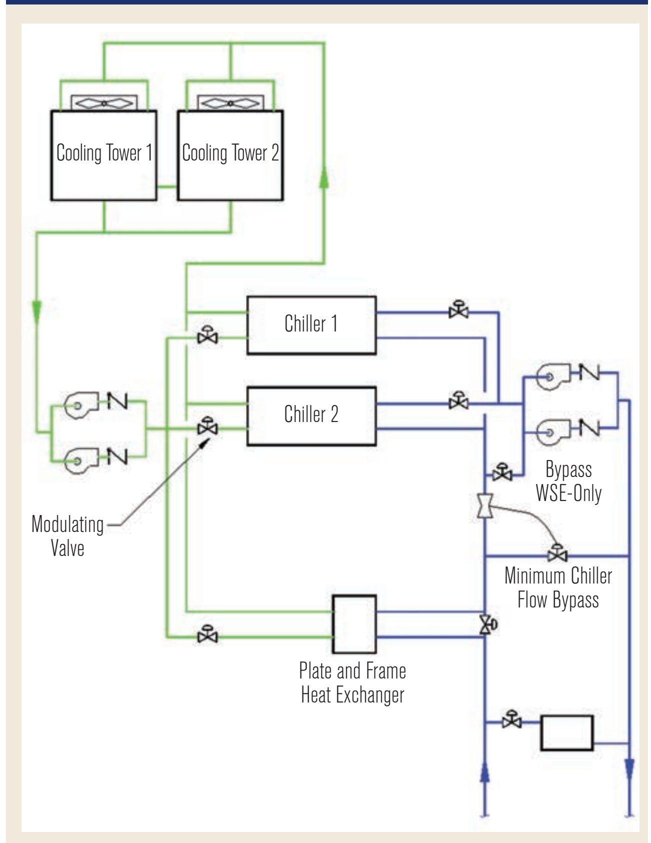


FIGURE 2 Integrated economizer.



ply tower water to the economizer heat exchanger. But this is also true for data centers where the load may require all chillers to run even in cold weather. The reason is that the load on the towers is actually reduced by the economizer since compressor heat is reduced, and condenser water flow to the chiller condensers may be less than design because it is colder (and in fact may be reduced by the throttling of the head pressure control valves discussed earlier), making water available to the economizer heat exchanger without the need to add pumps.

- While cooling tower capacity is not affected by the economizer, it may be necessary to reduce the design approach temperature to meet Standard 90.1 Section 6.5.1.2.1 waterside economizer requirements, particularly for plants with high loads in cold weather. This is discussed further in the sidebar, "Example Design Procedure" (Page 34).
- It is critical that cooling towers be very efficient since they will be running at full speed many hours of the year. A minimum of 90 gpm/hp (7.6 L[s·kW]) at Standard 90.1 conditions (95°F to 85°F at 75°F [35°C to 29°C at 24°C] wet bulb) is recommended for office type applications and 110 gpm/hp (9.3 L[s·kW]) for 24/7 applications such as data centers. These efficiencies are 10% above those shown to be cost effective for non-economizer applications.[2]
- Cooling towers should be selected so that as many tower cells as possible can be enabled when the economizer is enabled to maximize efficiency and capacity while maintaining minimum flow rates required by the tower manufacturer to prevent scaling. Low minimum flow rates can be achieved using weir dams and special nozzles in the hot water distribution pans.
- Chilled water pump head increases due to the pressure drop of the heat exchanger when in economizer mode. However, in applications such as offices where the loads are low when the economizer is on, pump head may not need to increase above design head when the economizer is off; excess head may be available for the heat exchanger when the economizer is active due to the reduced chilled water flow to coils.
- The heat exchanger should be a plate & frame type and selected for an approach of about 3°F (1.7°C) (i.e., the temperature of the chilled water leaving the heat exchanger is equal to 3°F (1.7°C) above the temperature of the condenser

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water entering the heat exchanger). Heat exchanger cost increases exponentially with approach temperature so very close approaches should be tested for cost effectiveness. The heat exchanger pressure drop on the condenser water side should be similar to that of the condensers so the flow rate will be similar when serving either the condensers or heat exchanger. On the chilled water side, pressure drop is typically limited to about 5 or 6 psi (34 or 41 kPa) to limit the chilled water pump energy impact. The heat exchanger performance must be certified per AHRI Standard 400[3] as required by Standard 90.1.

- To maximize economizer performance, and also performance of the system even when not on economizer, the chilled water system must be designed for a very high temperature rise (ΔT). This maximizes the chilled water return temperature which allows the economizer to operate more hours. The design procedure is simple: all cooling coils should be the largest they can be within the cleanability limitations of Standard 62.1,[4] which requires that dry coil pressure drop at 500 fpm (2.5 m/s) face velocity must not exceed 0.75 in. w.c. (188 Pa).* This will typically be an eight row coil with about 12 fins per inch (5 fins per cm). Using high ΔT coils also reduces first costs, energy demand, and annual energy costs and should be used for all designs[5] but especially those with waterside economizers.

Waterside Economizer Controls

Control systems for waterside economizer are generally the same as for conventional plants except they require one added sensor that is optional on conventional plants: a wet-bulb temperature sensor. This is usually a combination temperature and relative humidity sensor with software or electronics that converts the two signals to wet-bulb temperature. Unfortunately, humidity sensors are notoriously unreliable and require frequent recalibration. To improve reliability it is essential to specify a high quality sensor (see Reference 8 for recommendations) and also provide a quality check by having the control system compare local dew-point temperature to the dew-point temperature data measured at the nearest National Oceanic and Atmospheric Administration (NOAA) site via the Internet. If the local dew-point temperature (calculated from the wet-bulb temperature using psychrometric routines standard in most digital control systems) substantially differs from the NOAA dew-point temperature, alarms can be generated indicating a need for humidity sensor recalibration.

Recommended control sequences for waterside economizers:‡

- Reset chilled water supply temperature setpoint based on valve demand, i.e., raise the water temperature until one chilled water control valve is full wide open.§

*Here is a simple way to test a coil for Standard 62.1 compliance with a manufacturer's coil selection program: Start with the desired coil including desired rows, fin type, and fin density; adjust the airflow rate up or down until the face velocity is 500 fpm (9 km/h); reduce the entering drybulb temperature to 60°F (15°C) to ensure a dry coil; then run the selection. To comply, the pressure drop under these conditions must be 0.75 in. w.c. (188 Pa) or less.

‡Time delays required to prevent rapid mode changes are required but not included here for clarity.

§Many engineers have concerns about the impact of CHWST reset on humidity control. When the waterside economizer is active, outdoor air is cool so humidity should not be a concern. But if the reset logic is based on chilled water valve position, CHWST reset will never have a significant impact on space humidity control regardless of weather. This is because space humidity is a function of the supply air humidity ratio which in turn is a function of the coil leaving dry-bulb temperature setpoint. Regardless of CHWST, the air leaving a wet cooling coil is nearly saturated; lowering CHWST only slightly reduces supply air humidity ratio. So as long as the supply air temperature can be maintained at the desired setpoint, which resetting off valve position ensures, resetting CHWST will not impact space humidity.

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Example Design Procedure[6,7]

1. Calculate the chilled water load at 50°F (10°C) dry-bulb temperature and 45°F (7.2°C) wet-bulb temperature. This is the performance test condition prescribed by Standard 90.1 for buildings other than data centers.† This can be done using standard load calculation software by selecting a spring or fall month and overriding design outdoor air temperatures. All other load assumptions remain the same. Loads should be reduced from design loads due to reduced conduction and outdoor air conditioning loads.

2. Use coil selection software or other coil models to determine the warmest chilled water supply temperature that can meet 100% of the chilled water load determined above for all coils. To do this, first determine the warmest supply water temperature that can meet the load at the design flow for each coil, then select the coldest of these and use it to determine the chilled water flow through all the other coils. The coil software will also determine the chilled water return temperature from each coil. Typically, the chilled water supply temperature can be reset 5°F (3°C) or more above the design chilled water supply temperature. This can be true even for data centers despite the consistently high load by taking advantage of redundant air handlers to effectively increase available coil area.

3. Select the condenser water flow rate equal to a multiple of design condenser water pump flow rates as required to closely match the chilled water flow rate determined above.

4. Determine the heat exchanger condenser water supply temperature equal to the required reset chilled water supply temperature determined above less the 3°F (1.7°C) approach temperature.

TABLE 1 Chiller design conditions (each).

	Chilled Water Side	Condenser Water Side
CAPACITY, TONS	300	345
FLOW, GPM	290	550
ENTERING WATER, °F	69	75
LEAVING WATER, °F	44	90
ΔP, PSI	6.5	6.7

TABLE 2 Waterside economizer heat exchanger design conditions.

	Chilled Water (Hot) Side	Condenser Water (Cold) Side
TOTAL LOAD, TONS	350	350
FLOW, GPM	560	550
ENTERING WATER, °F	65	47.0
LEAVING WATER, °F	50	62.3
ΔP, PSI	5	4.8

5. Calculate the condenser water return temperature to match the chilled water load based on the condenser water flow and supply temperature determined above.

6. Use cooling tower selection software to verify that the cooling towers can provide the required condenser water supply temperature at 45°F (7°C) wet-bulb temperature. If not, then cooling towers (and/or heat exchangers) will need to be reselected for closer approach temperatures.

Example. Assume the plant in *Figure 2* served an office building with floor-by-floor air handlers. The design conditions of the two chillers at the design cooling peak are shown in *Table 1.* The pump flow rates match the chiller design rates and cooling towers were selected at 68°F (20°C) wet-bulb temperature, a 7°F (3.9°C) approach. The cooling loads were then recalculated at 50°F/45°F (10°C/7.2°C) outdoor air temperature conditions; the load drops from 600 tons to 350 tons (2110 kW to 1231 kW). Coil selection programs were then used to determine the chilled water conditions required at the air handlers, and condenser water conditions were determined using the steps above. The resulting heat exchanger design conditions are shown in *Table 2.*

Then, cooling tower selection software was used to see if the selected cooling towers could cool 550 gpm (35 L/s) (275 gpm (17.5 L/s) across each tower) from 62.3°F (16.8°C) to 47°F (8.3°C), a 2°F (1.1°C) approach to the 45°F (7.2°C) wet bulb temperature. But, the software indicated that the towers were only able to deliver 48°F (8.9°C). So, the towers had to be reselected for a 73°F (22.8°C) leaving water temperature (5°F/2.8°C approach to design wet-bulb temperature) to achieve a 2°F (1.1°C) approach at the conditions above. Note that heat exchanger approach also could have been reduced to deliver the desired chilled water temperature, but it is usually more cost effective to invest in larger cooling towers since they also improve efficiency when the economizer is off.

†Data center performance test conditions vary by climate zone. See Standard 90.1, Table 6.5.1.2.1 for details.

• Enable the economizer if the chilled water return temperature is greater than the predicted heat exchanger leaving water temperature (PHXLWT) plus 2°F (1.1°C). The 2°F (1.1°C) differential is needed to avoid expending a lot of cooling tower fan energy for only minimal economizer load reduction. PHXLWT is estimated using the equation:

$$PHXLWT = T_{WB} + PA_{HX} + PA_{CT}$$

$$PA_{HX} = DA_{HX} \times PLR_{HX}$$

$$PA_{CT} = m \times (DT_{WB} - T_{WB}) + DA_{CT}$$

where

T_{WB} = current wet-bulb temperature
PA_{HX} = predicted heat exchanger approach
PA_{CT} = predicted cooling tower approach
DA_{HX} = design heat exchange approach
PLR_{HX} = predicted heat exchanger part-load ratio (current chilled water flow rate divided by design HX chilled water flow rate)
DT_{WB} = design wet-bulb temperature
DA_{CT} = design cooling tower approach
m = slope developed from the manufacturer's cooling tower selection program or empirically after the plant is operational. Typical values are 0.2 to 0.5 for near constant load applications like data centers. For office

type applications, *m* is typically in the range –0.2 to 0.

- Disable the waterside economizer if it is not reducing the chilled water return temperature by at least 1°F (0.6°C).
- Disable chillers when HXLWT is at or below the desired chilled water supply temperature setpoint.
- Enable chillers when chilled water supply temperature is greater than desired setpoint. Note that multiple chillers may need to be enabled if the current chilled water flow is well above the design flow of a single chiller.
- Run as many tower cells as tower minimum flow limits will allow.
- Control condenser water flow to roughly match the current chilled water flow but reduce flow (within tower minimum flow constraints) as needed to maintain a minimum 5°F (2.8°C) range. The lower flow and higher range improves tower efficiency and reduces pump power. Flow can be controlled by staging pumps, modulating speed on variable speed pumps, and/or modulating isolation valves on the heat exchanger. Flow rate can be measured directly with a flow meter (full bore magnetic or ultrasonic type are recommended to prevent fouling) or deduced from heat exchanger pressure drop.
- Tower speed control:

  1. When waterside economizer is disabled: Control speed to maintain normal condenser water temperatures which should be reset from load or wet-bulb temperature.[9]
  2. When waterside economizer and chillers are enabled: Run tower fans at 100% speed.
  3. When chillers are disabled: Control speed to maintain HXLWT at desired chilled water supply temperature setpoint.

The only complex sequence above is predicting when the economizer should be enabled. Fortunately, if the prediction calculation is off and the economizer is enabled prematurely, it will shortly be disabled and the plant will see no disruptions in chilled water flow or supply temperature. This contrasts with nonintegrated economizers where switching from economizer to chillers can be disruptive and guessing wrong about economizer performance can result in chiller short cycling and temporary loss of chilled water supply temperature control.

Conclusions

Waterside economizers are preferred to airside economizers in some applications despite being less energy efficient. When waterside economizers are used, they usually require only the addition of a heat exchanger between the condenser water and chilled water; other components remain the same, although cooling tower approach sometimes must be reduced to meet Standard 90.1 performance requirements. Economizers should always be piped in an integrated (series) arrangement. Controls are straightforward except predicting when the economizer should be enabled which requires knowledge of wetbulb temperature and predicted heat exchanger and cooling tower approaches. But with integrated economizer design, this prediction is not critical to reliable plant performance.

References

1. ASHRAE/IES Standard 90.1-2013, *Energy Standard for Buildings Except Low-Rise Residential Buildings*.

2. Taylor, S. 2012. "Optimizing the design and control of chilled water plants, part 4: chiller & cooling tower selection." *ASHRAE Journal*, March.

3. AHRI 400-2001, *Liquid-to-Liquid Heat Exchangers*.

4. ASHRAE Standard 62.1-2010, *Ventilation for Acceptable Indoor Air Quality*.

5. Taylor, S. 2011. "Optimizing the design and control of chilled water plants part 3: pipe sizing and optimizing Δ*T*." *ASHRAE Journal* December.

6. Stein, J. 2009. "Waterside economizing in data centers: design and control considerations." *ASHRAE Transactions* 115(2).

7. Stein, J., S. Taylor. 2013. "Seminar 21: Water Side Economizers, a.k.a. Free Cooling, Integrated Water Side Economizer." ASHRAE Annual Conference.

8. NBCIP. 2005. "Supplement Product Testing Report: Duct-Mounted Relative Humidity Transmitters." http://tinyurl.com/of6leku.

9. Taylor, S. 2012. "Optimizing the design and control of chilled water plants, part 5: optimized control sequences." *ASHRAE Journal* June. ■