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Engineers Newsletter

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The Impact of VSDs on Chiller Plant Performance

The efficiency of various chiller plant designs and operation strategies is a hot industry topic. A recent five-part series in the *ASHRAE Journal* provided an excellent process for designing an efficient modern chiller plant.

Also reverberating through the industry is the concept of the all-variable-speed chiller plant. With the popularity and falling prices of variable-speed drives (VSDs), the sentiment of SOAV (Slap On A VSD) has ramped up. While investing in a VSD on chiller plant components typically results in energy savings, the magnitude of savings and the payback can vary significantly.

The purpose of this *Engineers Newsletter* is to compare the impact of the addition of VSDs to various chiller plant components under a few different design and control conditions. It is our hope that it will provoke plant designers to explore the range of plant design and control possibilities on future projects.

The Analysis

To provide enough diversity to make this a useful analysis, the following examples will be analyzed.

Building Types:

- Chicago office with economizer
- Memphis hospital no economizer
- Miami office no economizer

Base Chiller Plant Configurations:

Chilled-water conditions	56°F–42°F (1.7gpm/ton)
Condenser water flow conditions	85°F–94.4°F (3 gpm/ton)
Cooling tower cell per chiller	(38.2[1] gpm/hp)
Condenser water pump per chiller	(19 W/hp)
1, 2, and 3 constant-speed chillers	(0.567 kW/ton)
Fixed tower setpoint control	85°F
ASHRAE 90.1-2010 Path A compliant	

Alternatives: From these base conditions the analysis will consider:

- optimized control sequences,
- the addition of VSDs to various components, and
- near-optimum system design conditions.

Because several of the optimized control strategies considered are difficult to analyze in commercially available energy modeling software, a custom program was created to perform the analysis. It utilizes multivariable quadratic chiller modeling algorithms and the ASHRAE cooling tower performance model, deviating from design setpoints only where specified to evaluate optimized control. The modeling program performs an 8760 hour analysis using TMY3 weather files.

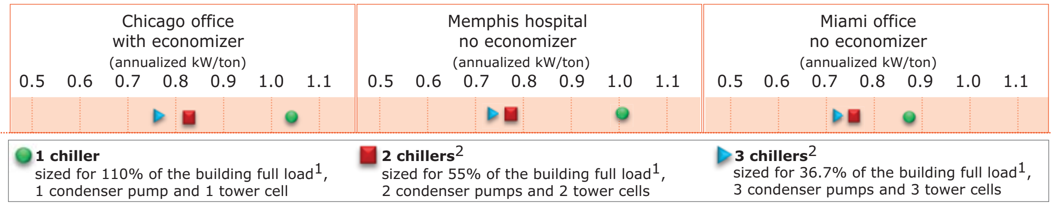
The resulting energy performance is reported as *annualized kW/ton*. This value is calculated by dividing total annual chiller plant kWh by total annual system ton-hrs. It represents a year-long average of the chiller plant's performance.

Finally, it is important to note that in order to maintain a reasonable scope for this analysis, we considered the energy consumption of only chiller and heat rejection equipment (condenser pump and tower fan).

[1] Per ASHRAE 90.1 2007 - Appendix G Baseline Building

Figure 1. Base case system performance in annualized kW/ton

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base	CS	1 spd	3	CF	85°F



[1] *To represent a reasonable safety factor*

[2] *Equally sized chillers, sequenced to keep the constant-speed chillers as fully loaded as possible*

The Base Case. Figure 1 represents our base case for this EN comparison—performance of an all-constant-speed system operating with a cooling tower setpoint of 85°F. The left side of the table shows the plant configuration and operating conditions. Table abbreviations represent the following:

CS	constant speed
VS	variable speed
1 spd	single speed
3 gpm/ton	high flow rate
2 gpm/ton	near optimal flow rate
CF	constant flow
VF	variable flow
85°F	constant leaving water setpoint
Opt	real-time optimized tower water temp. control

The energy performance results for each location and building type are shown on the right in terms of annualized performance of kW/ton.

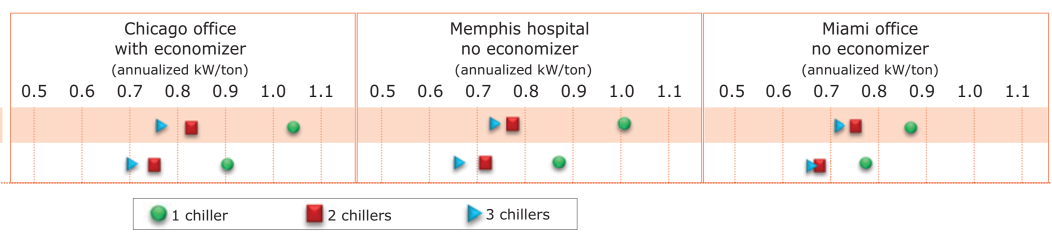
For the two- and three-chiller examples, the lag chillers are cycled off as soon as the plant load allows. In an all-constant-speed system, if the lag chillers are left on at lower loads, the annualized plant performance will be worse, approaching or equaling the energy use of the single-chiller system.

Observations. From this base case analysis we can make two observations.

- First, the use of multiple chillers significantly decreases the energy use of the plant, with the greatest impact seen in going from one chiller to two. This occurs because at many part-load hours, half or more of the pump and fan energy can be cycled off. This results in a much better balance of chiller, pump and fan power relative to the cooling load. At many part-load hours, one or more chillers also can be cycled off, allowing the remaining chillers to operate at a more efficient load point.
- Second, the annual plant efficiency for the Chicago location looks worse than the others. As chillers are added, the difference becomes less. There are two significant reasons.
  - Even with airside economizer operation, the Chicago office has a higher percentage of hours operating at lower loading on the chillers. With the entering condenser water being controlled to 85°F, the increased low load kW/ton of the constant-speed chiller(s) and high relative condenser pump power results in worse system efficiency at low-load hours.
  - At low loads there are fewer tons across which to distribute the high flow/high level of condenser pump energy, resulting in a more pronounced negative effect on the system annualized performance.

Figure 2. Alternative 1 and base case comparison of constant-speed versus variable-speed cooling-tower fan control

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	**VS**	3	CF	85°F



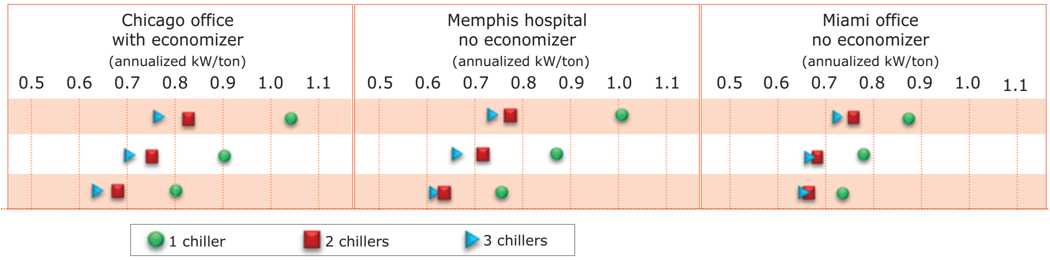
Alternative 1. The first alternative (Figure 2) applies variable-speed control to the cooling-tower fan, again with a cooling-tower leaving-water temperature setpoint of 85°F.

Observations:

- Adding VSDs to the cooling-tower fans improves plant efficiency by 8 to 13 percent. As might be expected, the least improvement is on the three-chiller Miami plant and the greatest percentage improvement is on the single-chiller Chicago plant.
- Cycling operation of a single fan on a cooling tower is a very inefficient method of tower capacity control.
- Taking advantage of the affinity laws on a free discharge variable-speed device, even without optimized setpoint control, results in substantial savings.
- While not obvious from the data, the stable temperature control enabled by the tower variable-speed capacity control also enhances system efficiency.

Figure 3. Alternative 2 with optimized variable-speed-drive (VSD) control on cooling tower fan

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	VS	3	CF	85°F
2.	CS	VS	3	CF	**Opt**



Alternative 2. Figure 3 compares performance results of the system with optimized control of the cooling-tower fan speed, properly balancing the fan energy investment relative to the chiller(s) loading.

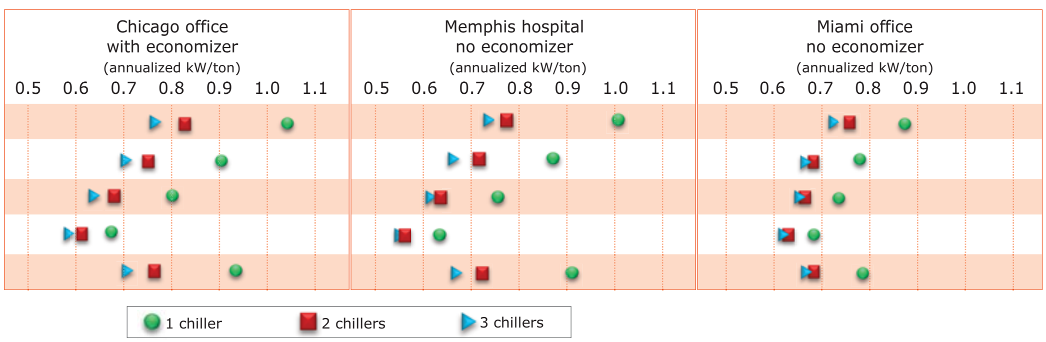
Observations.

- Optimizing the variable-speed cooling tower fan operation significantly improves plant annualized efficiency.
- Compared to the base case, the plant efficiency improves by 11 to 24 percent for the optimally controlled variable-speed cooling tower alternative. Again, the least improvement is on the three-chiller Miami plant. However, this time the greatest percentage improvement is on the single-chiller Memphis plant with the single-chiller Chicago plant not far behind.
- Relative to installed cost, and often on an absolute basis, the application of a VSD with optimized control on a cooling-tower fan results in a greater increase in plant efficiency than any other single optimized application of a VSD in a chiller plant. As we compare more alternatives this will become evident.

Conclusion. *Every* chiller plant should utilize optimized variable-speed control on all cooling-tower fans. There is not a better chiller plant energy-saving investment available.

Figure 4. Alternatives 3 and 4 comparison with the addition of variable-speed drives on the chillers and optimized tower control

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	VS	3	CF	85°F
2.	CS	VS	3	CF	Opt
3.	**VS**	VS	3	CF	**Opt**
4.	**VS**	VS	3	CF	**85°F**



Alternatives 3 and 4. Figure 4 adds two additional alternatives, each with variable-speed centrifugal chillers. Alternative 3 illustrates optimized variable-speed tower fan control. Alternative 4 illustrates the same system with a tower controlled to a design setpoint temperature of 85°F.

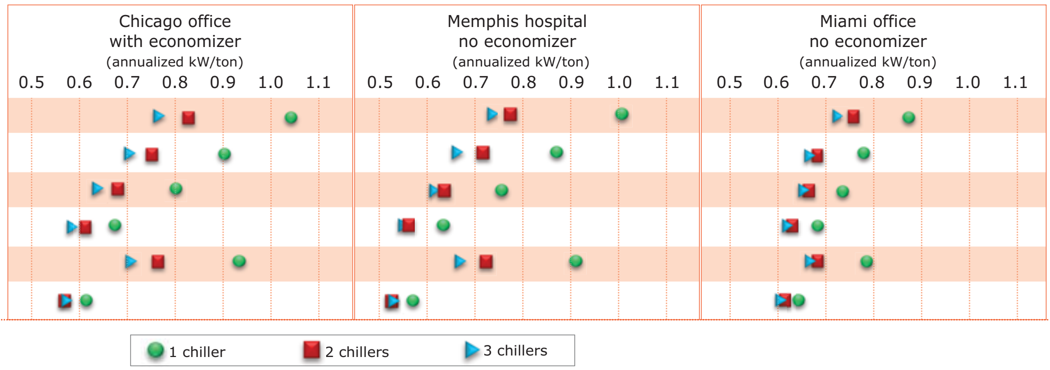
As stated earlier, the full-load efficiency of the constant-speed chiller is modeled at 0.567 kW/ton (ASHRAE 90.1-2010 Path A compliant). These alternatives' variable-speed chillers are modeled at 0.585 kw/ton (ASHRAE 90.1-2010 Path B compliant). This degree of difference is common because the VSD introduces an additional electric efficiency loss. Additionally the increased cost of the VSD may be partially offset by removing chiller condenser or evaporator heat transfer tubes, which negatively impact chiller full- and part-load efficiency.

Observations.

- For alternative 3, adding variable speed to the chillers with optimized VSD cooling-tower fan control results in plant energy savings in all building types and locations.
- In hotter and more humid climates the savings is less, so the return on investment would likely be less attractive.
- Alternative 4 reveals that incorrect tower control can negate the benefit of the variable-speed chillers and make the system work less efficiently than one with constant-speed chillers (e.g., an operator overriding the tower setpoint to 85°F). While this type of operation may seem ludicrous, the author has witnessed similar operation in more than one chiller plant via remote monitoring as well as during personal visits.
- The variable-speed affinity laws can work against the system efficiency too.

Figure 5. Alternative 5 comparison with constant near-optimized condenser flow water (2 gpm/ton)

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	VS	3	CF	85°F
2.	CS	VS	3	CF	Opt
3.	VS	VS	3	CF	Opt
4.	VS	VS	3	CF	85°F
5.	VS	VS	**2**	CF	Opt



Alternative 5. Figure 5 represents the same system configuration and control as alternative 3 but with the chillers, cooling towers and condenser pumps selected for constant flow operation at a near* optimal 2 gpm/ton (15°F delta T).This flow selection is based on the recommendation from a number of industry chiller plant design studies, the latest of which was published in the *ASHRAE Journal* (December 2011).

"Condenser water…life cycle costs were minimized at the largest of the three delta Ts analyzed, about 15°F. This was true for office buildings and datacenters and for both single-stage centrifugal chillers and two-stage centrifugal chillers."

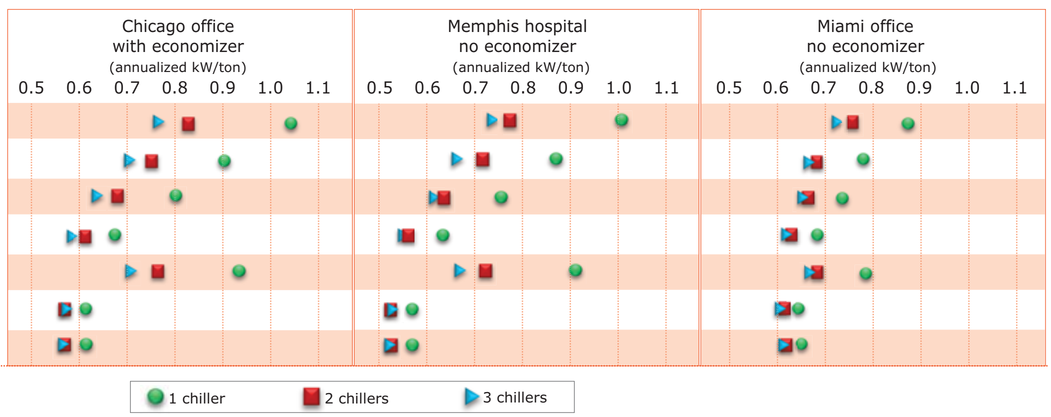
Observations.

- All configurations for alternative 5 show energy savings compared to a system designed with the historically common condenser water flow rate of 3 gpm/ton (9.4°F delta T).
- Although the chiller's design efficiency is decreased, this is offset by decreased condenser pump and tower fan energy use.
- Plants with fewer chillers show greater savings. This is due to the fact that the condenser pumps are not cycled off with load. Also with lower design flow and power draw, the condenser pump energy is less as a percentage of the annualized plant energy use.

*Our lawyers will not allow us to use absolute terms such as "optimal" without a moderating adjective. In fact it's likely that the true optimal value would depend on the load, location and user's optimization criteria, i.e., life cycle cost, ROI, lowest possible annualized energy use, first cost, etc.

Figure 6. Alternative 6 comparison with addition of variable high condenser-water flow

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	VS	3	CF	85°F
2.	CS	VS	3	CF	Opt
3.	VS	VS	3	CF	Opt
4.	VS	VS	3	CF	85°F
5.	VS	VS	2	CF	Opt
6.	VS	VS	**3**	**VF**	Opt



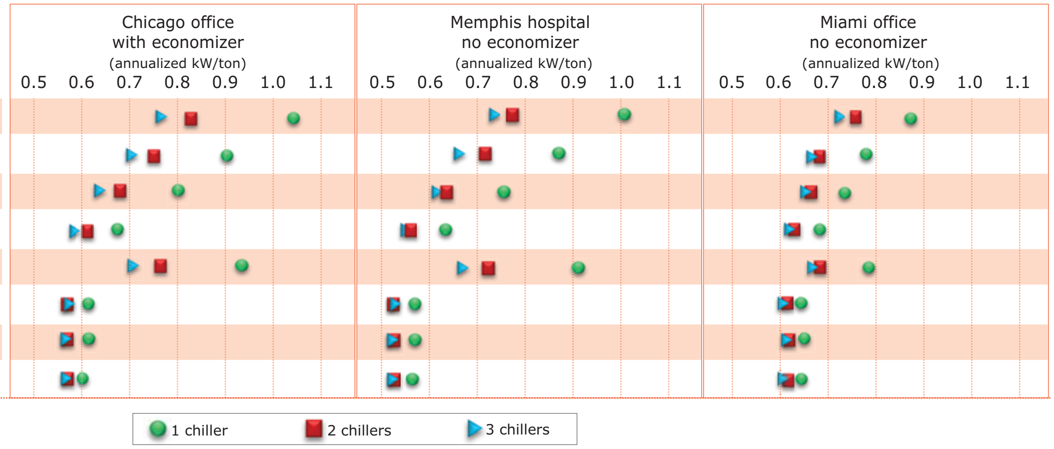
Alternative 6 uses 3 gpm/ton design condenser water flow rate but applies optimized variable condenser water flow to continuously modulate the condenser system flow and pump power use relative to the plant load (Figure 6). The objective of this control is to provide the chiller(s) with higher flow at high loads when it most benefits chiller performance, and reducing flow and pumping power at part load to minimize the excess pump energy consumption.

Observations.

- Energy savings differ by location for this alternative. The Chicago office and the Memphis hospital alternatives show minimal energy savings when compared to the near-optimized constant water flow (alternative 5). Single chiller systems again provide a larger percentage of savings.
- Properly balancing the chiller/pump energy for best life cycle performance (near-optimal constant flow design) leaves little excess pumping energy to be optimized out of the system at part load.
- The Miami office chiller plant energy use is higher for the high design variable flow alternative compared to the near-optimized constant water flow case. This is likely a result of two conditions: The large number of high-wet bulb operating hours which requires high flow to prevent unstable chiller operation results in elevated system pumping power. Secondly, the higher design entering tower water temperature of the low flow system increases the tower heat transfer effectiveness which results in proportionately lower fan power at all loads.
- The efficiency of variable-speed chillers is more negatively impacted by varying condenser water flow. Therefore the expectation is that a system with constant-speed chillers would show slightly greater benefit in annualized efficiency compared to the VSD chiller system.
- As with variable-speed fan control, incorrect control would negate the benefit of the variable-speed condenser water pumping and cause the system energy use to be substantially the same as the 3 gpm/ton constant flow system (alternative 5). This could occur through an operator overriding the VSD to 60 Hz. Again the affinity laws for variable speed can work against the system efficiency.
- Unstable condenser water flow and/or cooling tower fan control would negatively impact system efficiency and may result in unstable chiller operation (surge in centrifugal compressor chillers). The low constant flow alternative eliminates the potential for unstable condenser water flow thus reducing the potential for instability with varying loads.

Figure 7. Alternative 7 comparison with addition of variable near-optimized condenser-water flow

	chiller type	clg. twr. fans	cond. wtr. flow rate (gpm/ton)	cond. wtr. flow type	twr. control method
base.	CS	1 spd	3	CF	85°F
1.	CS	VS	3	CF	85°F
2.	CS	VS	3	CF	Opt
3.	VS	VS	3	CF	Opt
4.	VS	VS	3	CF	85°F
5.	VS	VS	2	CF	Opt
6.	VS	VS	3	VF	Opt
7.	VS	VS	**2**	VF	**Opt**



Alternative 7. Figure 7 illustrates a system with 2 gpm/ton design condenser water flow AND optimized variable condenser water flow and cooling tower fan speed control. This configuration leverages low design condenser pump and tower power and reduces it even further when beneficial at part load. This is balanced against a slightly higher chiller power use at full-load design conditions.

Observations.

- This is the most efficient configuration examined, although the efficiency advantage is small in all cases when compared to the near-optimized constant condenser water flow (alternative 5) or variable high condenser water flow system (alternative 6).
- Incorrect control would have a negative, though minimal, impact on the system savings compared to the near-optimized constant water flow system (alternative 5). Again, this could occur through an operator overriding the VSD to 60 Hz.

Summary

While there are many other plant configurations and design conditions that could be examined, these 72 permutations (3 chiller quantities x 3 locations/facilities x 8 plant configurations) provide some clear and important design and control direction.

1. Multiple-chiller systems provide for better annualized chiller plant operating efficiency, particularly for two-chiller versus one-chiller constant-flow systems at historical design conditions.
2. Single-chiller plants benefit most from optimized design conditions and variable-speed components. When properly applied, these plants can approach the efficiency of multiple-chiller plants.
3. Optimally controlled variable-speed cooling-tower fans are fundamental to the efficiency of every chiller plant.
4. Variable-speed chiller technology, with a properly controlled condenser water system, delivers improved annualized efficiency, particularly in mild climate buildings.
5. For a new chiller plant, there is significant potential to raise the annualized plant efficiency without the addition of sophisticated variable-speed condenser water flow control, simply by using near-optimal design flow rates rather than historical AHRI standard rating point flow rates.
6. For existing plants with relatively high design condenser water flow rates, there is significant potential to raise the annualized efficiency by adding proper variable-speed (flow) control on the tower fans and condenser water pumps.
7. The application of VSD technology to both new and existing chiller plant components can provide for significant improvement in annualized plant efficiency and therefore reduction in energy consumption. However, sustained optimized control is critical to realizing the ongoing savings potential.

Two critical questions remain.

The first: *Is there a plant configuration not analyzed here that could provide a significantly improved life cycle cost or ROI compared to the extremes of all-constant-speed chillers or all VSD chillers?* For example, a combination of one or two variable-speed chillers with other constant-speed chillers in a chiller plant. This may be the subject of a future *Engineers Newsletter*.

The second question is actually fundamental to the first: *What is the cost, ROI and life cycle impact of each alternative?* For an excellent treatment of this topic, refer to the five-part series in the *ASHRAE Journal*, "Optimizing Design & Control of Chilled Water Plants" (July, September, December 2011, and March, June 2012 issues).

By Lee Cline and Brian Sullivan, Trane systems engineers. 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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Energy-Saving Strategies for Chilled-Water Terminal Systems.

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(SYS-APM008-EN, updated May 2012)

Water-Source and Ground-Source Heat Pump Systems.

Examines chilled-water-system components, configurations, options, and control strategies. (SYS-APM010-EN, updated November 2013)

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