Engineers Newsletter

The Impact of Variable-Speed Drives on HVAC Components

Variable-speed drives (VSDs) have been applied to various HVAC components for many years in different capacities. In addition to being able to precisely control motor speed, variable-speed drives can be used to trim or balance systems and provide soft start capability. They can also spin a motor beyond synchronous speed.

When a variable-speed drive is attached to a component a short-hand rule of thumb is "power varies with the cube of the speed." This is not always true. This EN explains the different effects that variable-speed drives have on fans, pumps and refrigeration equipment.

Variable-speed drives (VSD) are devices that vary the rotational speed of single- and three-phase induction or permanent magnet motors. These devices, sometimes called adjustable-frequency drives (AFD), adjustable-speed drives (ASD), inverters, or variable-frequency drives (VFD) take incoming power from the utility and change the frequency before it's applied to the motor.

A change in alternating current frequency results in a change of motor speed. Components attached to a variable-speed motor, such as fans, pumps, and compressors, behave differently as their speed changes.

It's important for the system designer to understand the impact before applying a variable-speed drive to a piece of equipment.

Affinity (Fan and Pump) Laws

What is the effect of a changing speed? Engineers use a 'shorthand' for estimating the effect of speed (rpm) on power. This method is referred to as the affinity laws or fan and pump laws.

When a fan operates in an unchanging system, the fan laws tell us that the *airflow varies proportionally to fan rpm, static pressure varies by the square of the rpm, and horsepower varies by the cube of the rpm.* These relationships are written as an expression of a current condition (subscript 1) and a new condition (subscript 2):

$$\frac{cfm_1}{cfm_2} \alpha \frac{rpm_1}{rpm_2}$$

$$\frac{SP_1}{SP_2} \alpha \left(\frac{rpm_1}{rpm_2}\right)^2$$

$$\frac{hp_1}{hp_2} \alpha \left(\frac{rpm_1}{rpm_2}\right)^3$$

As an example, assume a fan's design operating point is 8000 cfm running at 884 rpm, 1.5 inches of static pressure, and 6.4 horsepower. At a given time, assume half the airflow is needed. Using the fan law relationships, a new motor speed, static pressure, and power can be calculated.

$$\frac{8000}{4000} \alpha \frac{884}{rpm_2}$$

Solving for rpm_2 yields 442 revolutions per minute. Using this value, the new static pressure and horsepower value can be determined:

$$\frac{1.5}{SP_2} \alpha \left(\frac{884}{442}\right)^2, SP_2 = 0.375\,in$$

$$\frac{6.4}{hp_2} \alpha \left(\frac{884}{442}\right)^3, hp_2 = 0.8\,hp$$

At the new airflow and motor speed, less static pressure is generated and significantly less fan power is needed.

A variable-speed drive could be used to reduce the motor speed. As a result, consumed energy is dramatically lower. In this instance, only 12.5 percent of the design power is needed to deliver half the design airflow.

The affinity laws hold true for the component by itself—how does the type of *system* and *fan* affect this relationship?

Fan type and system effects

To explore fan type and system effects on savings available from VSDs, we'll consider two types of fans: free discharge and ducted fans.

Free discharge fans. Fans that discharge to the atmosphere or an open environment are typically called free discharge fans. These fans do not have a fixed pressure component that might be found within a ducted airside system. Examples of free discharge fans include cooling tower fans

and small fans inside terminal units, such as fan-coils, water-source heat pumps, and VRF terminals.

Historically, large free discharge fans on cooling towers used constant-speed motors with on/off control or two-speed motors. Due to reduced cost and the power savings, VSDs are most often applied today. Figure 1 shows three motor control strategies applied to a free discharge fan on a cooling tower.

Two-speed motor control certainly improves upon on/off motor control for a single-speed fan; however, additional savings can be realized when using a VSD.

Because free discharge fans do not have a fixed pressure component, they follow the affinity laws more closely as illustrated. If less airflow is needed, the fan is able to reduce its speed. The reduction in speed results in a power consumption decrease. With a free discharge fan, the rule "*power varies with the cube of the speed*" holds true. This, combined with minimal system effects, make these fans excellent candidates for VSDs.

Supply fan capacity control methods in a multiple-zone VAV system have evolved over the years from discharge dampers, inlet guide vanes, and vane-axial variable pitch controls to variable-speed drives. All of these technologies are designed to vary and control the static pressure generated by the fan to ensure there is enough conditioned air to meet the space load. Each method controls to a static pressure setpoint. As we'll discuss, there are methods for adjusting the setpoint in response to system load, as required by energy codes.

Figure 1. Free discharge fan performance, motor power versus airflow
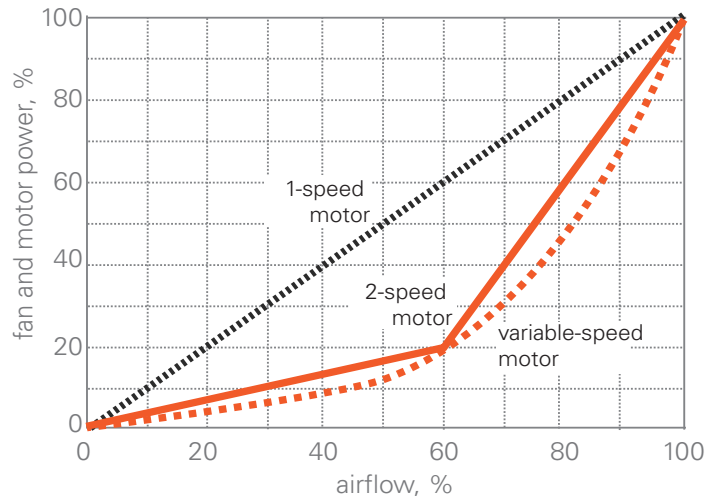
Variable-speed requirements from ANSI/ASHRAE/IESNA Standard 90.1-2010.
Requirements for variable-speed control can be found throughout ANSI/ASHRAE/IESNA Standard 90.1, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. In specific instances for common systems, the standard includes mandatory and prescriptive requirements for speed reduction control. The most common method to satisfy these requirements is to use a variable-speed drive.

Mandatory Requirements (paraphrased—consult your energy code for variation)	
6.4.3.10	**Single-Zone Variable-Air-Volume Controls.** Air-handling and fan-coil units with chilled-water cooling coils and supply fans 5 hp or greater, must use a fan capacity modulation device for part-load operation that reduces airflow to ½ of the full speed fan's airflow or to the Standard 62.1 (ventilation) requirement. Direct expansion systems with a cooling capacity greater than or equal to 110,000 Btu/h (9.17 tons) that serve single zones must also use a fan-speed modulation device that reduces airflow to 2/3 of the full speed fan's airflow or to the ventilation requirement for part-flow operation.
Prescriptive Requirements	
6.5.3.2	**VAV Fan Control (Including Systems Using Series Fan Power Boxes).** Individual VAV fans with motors greater than or equal to 10 hp shall have a fan capacity modulation device that provides VFD-like energy performance (30% wattage and 50% of design air volume at 1/3 of the total design static pressure).
6.5.3.2.3	**Setpoint Reset. VAV Systems That Employ Direct Digital Control (DDC).** VAV systems that employ DDC must use a central control system that employs static pressure reset, as an exception to setting the duct static pressure setpoint to 1/3 of the total design static pressure. Static pressure reset adjusts fan volume by requiring only enough pressure to satisfy the worst-case zone.
6.5.4.1	**Hydronic Variable Flow Systems.** Hydronic systems that have total pump power exceeding 5 hp with more than three control valves must have pumps with VFD-like performance at part-flow operation.
6.5.4.4.2	**Hydronic Heat Pumps and Water-Cooled Unitary (Equipment).** Units with total system pump power exceeding 5 hp must use variable-speed pumps or have VFD-like performance.
6.5.5.2	**(Heat Rejection Equipment) Fan Speed Control.** Heat rejection equipment fans with motors 7.5 hp or larger are required to have the capability to operate at 2/3 of full fan speed or less. There are numerous exceptions that should be investigated.

Fans in larger, ducted VAV systems. A ducted system imposes additional pressure requirements on the fan. The purpose of a fan is to generate pressure and airflow. Zone dampers in a ducted VAV system impede airflow in response to the zone cooling or heating requirements. Closing one or more dampers creates an increased pressure effect at the same fan speed, but how much they impact system pressure depends on which dampers—those close to the fan or those farther down the duct. To determine how much a VSD can slow down a ducted fan (and save energy), we need to consider these system effects and the control method.

The 2010 Trane *Engineer's Newsletter Live!* program "Fans in Air-handling Systems" explains the relative system effects of various strategies in much more detail. The following section is simplified for brevity.

Some VAV systems are operated to maintain a fixed static pressure at some location in the supply duct—the setpoint of the fan capacity control modulation to ensure proper airflow at all system conditions. As airflow increases, so does system resistance and static pressure. Figure 2 shows the system resistance curve for a system with a fixed static pressure setpoint. The variable static pressure component of the fan's work is also shown. It is this component that is reduced at part load and lower airflows. As we'll discuss later, more of the fixed component can be made variable, through the proper use of a variable static pressure setpoint.

This minimum static pressure setpoint reduces the potential energy savings from the theoretical cubic savings implied by the affinity laws.

Impact of part load on the system resistance curve. A further consideration when calculating potential VSD savings is that the system resistance curve changes at part load when the airflow and pressure requirement is reduced.

Zone dampers in VAV terminals regulate the airflow to a zone by adjusting the damper position. When damper positions change, the system resistance and static pressure upstream of it change. A multiple-zone VAV system may have many zone dampers. As

Figure 2. System resistance curve and fixed pressure region
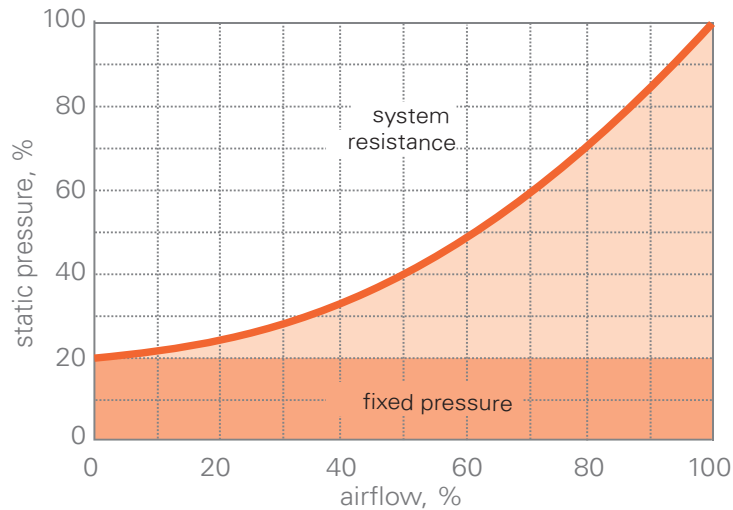
Figure 3. Variable-air-volume fan unloading and the VAV system modulation
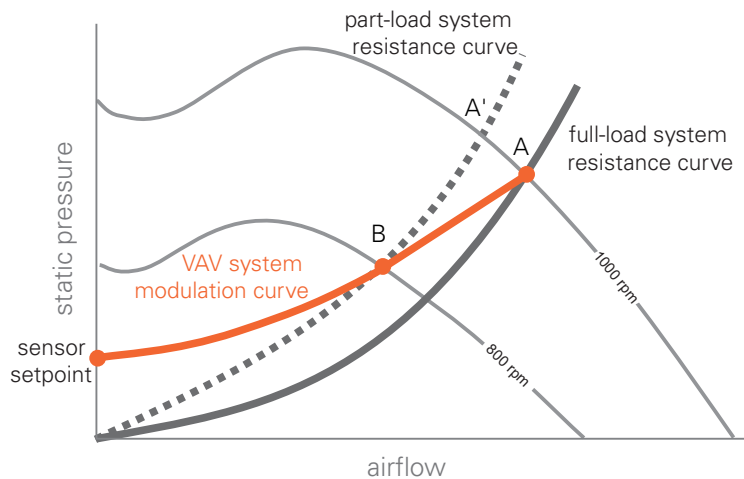
some of the dampers modulate toward closed, the system resistance curve shifts upward (from point A toward point A' in Figure 3). Unless the fan speed is reduced, the fan "rides" its constant-speed (rpm) curve (hence the phrase "riding the curve"). Additional static pressure is generated by this combination of the fan and the dampers, and less airflow is delivered at this new operating point.

The resulting higher static pressure is sensed by the static pressure sensor, which then relays a signal to reduce fan speed. With the reduction in speed, the fan generates less static pressure and the duct

static pressure is returned to its setpoint (from point A' to point B). The system controller continues to modulate fan speed to maintain the static pressure setpoint, and the zone dampers modulate to maintain the airflow necessary to meet their space temperature setpoint. Mapping additional fan unloading conditions creates the VAV system modulation curve in Figure 3. This curve depicts how the fan and the system balance one another, and create the static pressure and airflow that meet the zones' cooling and heating demands.

Static pressure sensor location. The position of the static pressure sensor can vary from being factory-mounted at the fan discharge to near the end of the duct run. The controller is set to maintain static pressure at design airflow. Mounting the duct static pressure sensor at the fan discharge requires that all the dampers partially close to increase static pressure above design. Sensing the excess pressure, the fan controller reduces the fan speed to reduce generated static pressure. All the dampers must be throttled and energy is wasted.

Another common approach is to field-mount the sensor some distance from the supply fan (e.g., two-thirds of the longest duct run). The dampers upstream of the sensor affect the operation of the fan, while dampers downstream of the sensor do not. Moving the sensor farther down the ductwork increases the number of dampers that are able to affect fan operation. Less fan energy may be wasted; however, in rare cases, one or more of the upstream dampers may become starved for air, as the sensed static pressure reflects the aggregate airflow requirements of the upstream dampers. One solution to this problem is to increase the sensor static pressure setpoint, although doing so increases fan energy consumption.

With proper control of the fan speed to maintain duct static pressure, significant energy savings can be realized. However, the VAV system modulation curve, the static pressure sensor location, and static pressure setpoint together determine how much fan energy savings can be achieved with variable-speed modulation.

As discussed previously, "power varies with the cube of the speed" doesn't hold true for systems with a fixed static pressure setpoint. However, system controls can increase the energy savings. How much additional energy can be saved with system controls?

Figure 4. Optimized static pressure control
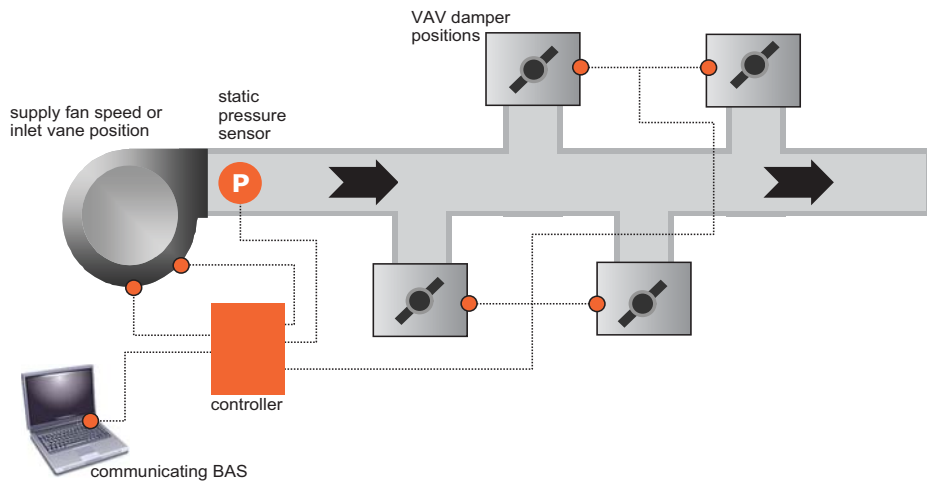
Duct static pressure reset. Additional energy savings can be realized if the static pressure setpoint is reset during operation. This strategy, sometimes called *critical zone reset* or *fan pressure optimization*, uses a building automation system (BAS) to monitor and control the air delivery system and all terminals on the system as shown in Figure 4.The duct static pressure sensor is connected to the BAS system. The location of the static pressure sensor can vary—it can be mounted as close as several duct diameters from the fan or along the duct run.

Each VAV terminal is also connected to the BAS system. The VAV terminal senses the position of the damper and controls its movement to maintain zone temperature. The BAS system continuously determines the zone with the most-open damper and resets the static pressure sensor setpoint downward so this damper is nearly wide open. The VAV dampers respond by modulating to deliver adequate airflow while the supply fan speed is reduced. The result is that the supply fan uses less power as it generates only enough static pressure to push air through the critical (most-open) damper.

Energy code requirements.

Static pressure sensor location. ASHRAE Standard 90.1-2010 requires the static pressure setpoint to be no greater than one-third the total design fan static pressure. Addendum S to Standard 90.1-2010 (published in 2012) effectively modifies the static pressure sensor location, because it requires that the static pressure sensor setpoint be no greater than 1.2 inches of H_2O (300 Pa). This number conservatively estimates the remaining static pressure for delivering air from the location of the sensor into the zone. If major duct splits exist, multiple sensors may need to be employed. The purpose of this requirement is to better reflect the average zone damper requirement downstream of the sensor. An exception is made for systems with zone damper position reset strategies.

VAV setpoint reset. Duct static pressure setpoint reset is required by ASHRAE Standard 90.1 for VAV systems with direct digital control of terminals connected to a building automation system (BAS) if the fixed static pressure setpoint is higher than one-third of the design pressure (90.1-2010 and earlier) or higher than 1.2 in. w.g (90.1-2013). If choosing the setpoint reset method, 90.1 requires that the BAS system monitor the zone damper position, and reset the setpoint until one zone is nearly wide open. The 90.1-2013 (90.1-2010 addendum S) further requires that the BAS system detect zones that are excessively driving reset logic and generate alarms, and allow system operators to easily remove those "rogue" zones from the reset algorithm.

To visualize how this control turns the VAV system into one that closely follows the affinity laws, consider a 22-horsepower, 24,000 cfm supply fan with 2.7 inches of static pressure. Table 1 shows the potential savings from changing the position of the duct static pressure sensor, or from optimizing the system. Note that mounting the static pressure sensor within the supply duct yields more energy savings than locating it at the fan outlet. Likewise, using an optimized strategy produces the lowest fan input power at part load, allowing the fan system power to approach "power varies with the cube of the speed" savings.

Pumps

Much like system fans, pumps are used to increase pressure within a system. By increasing the pressure, water flows through HVAC components such as chillers, water valves, and coils. Also like fans, as the system resistance is increased (e.g., valves closing to limit water flow through a coil during part-load), and the pump speed (rpm) held constant, the pump operating against greater pressure moves less water. Figure 5 shows this graphically, with pump head on the left-hand vertical axis, power on the right-hand vertical axis, and water flow percent on the horizontal axis. As valves close, we move from right to left on the constant rpm curve, hence the term "riding the pump curve" used to describe constant-speed pump operation at part flow. Flow and power consumption are both reduced.
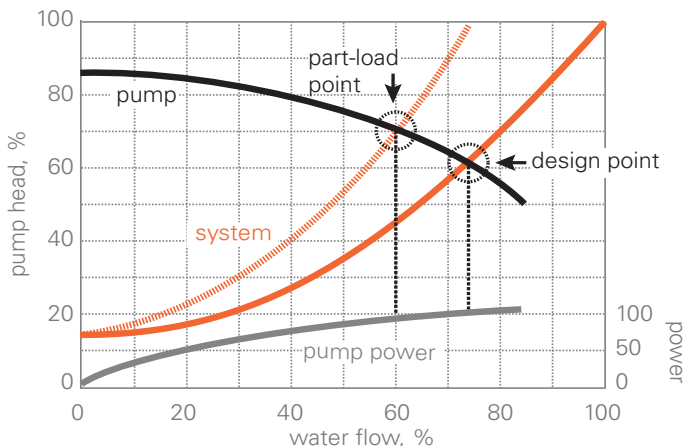
Table 1. Fan power savings of various static pressure sensor locations and optimized systems

Control method	airflow	fan static pressure	fan input power	full-load power
	24,000 cfm (full load)	2.7 in.wg	22 hp	100%
fan outlet	18,000 cfm	2.1 in.wg	13 hp	60%
supply duct	18,000 cfm	1.9 in.wg	12 hp	55%
optimized	18,000 cfm	1.5 in.wg	9.5 hp	43%

The potential energy savings from slowing down and picking a new pump speed are proportional to reduction in flow and inversely proportional to the increase in pump pressure. The energy saved is a function of the pump flow and head characteristics. Selecting a pump with a flat speed curve yields more energy savings. A steeper curve, with larger pressure rises as flow decreases, results in smaller savings.

In the past, multiple pumps of decreasing sizes were manifolded. As less chilled water was needed in the distribution/secondary loop, the controller started another smaller pump and turned the larger pump off. Each pump operated along its own pump-speed curve. Manifolding multiple, constant-speed pumps was complex and could be difficult to control.

Adding a VSD to the pump creates an infinite number of different sized pumps with multiple, individual pump-speed curves. Controlling the pump speed can be done in a number of ways: *constant pressure control at the pump*, constant pressure control at the *end of the system*, and *critical valve reset*.

Constant pressure control at the pump requires a differential pressure sensor to maintain the pressure differential across the pump. The pump pressure rise is controlled by varying the speed to achieve the sensor setpoint, and does not change with load. Control valves "devour" excess system pressure and limit flow through the coils. Because the pump slows down at part load, some savings can be realized.

A better method is to install a **differential pressure sensor at the farthest point of the system**. The sensor is installed across the coil and control valve. As the system operates at part load and control valves close, the sensor detects excess differential pressure in the remote circuit. In response, the control system lowers the pump's differential pressure setpoint to reduce pump speed. The result is constant pressure at the end of the system.

A final strategy requires an integrated control system. Critical-valve pump-pressure control, also called **critical-valve reset**, monitors the system valves in a manner very similar to fan-pressure optimization. The BAS continuously polls the individual valve controllers to determine which valve is farthest open. The pump pressure setpoint is reset so at least one valve is nearly wide open while the remaining valves modulate to maintain desired flow. [This results in the pump generating only enough pressure to deliver design flow to the nearly wide open valve, and significant energy savings—provided that the zone served by that valve can be satisfied.] This strategy is required by Standard 90.1-2010. Depending on the control strategy selected, energy savings can approach the cube of the speed relationship. Using this strategy makes water pumps great applications for VSDs.

Figure 5. Design and part-load operation of a water pump

Refrigeration Compressors

Refrigeration compressors use variable-speed technology to provide capacity control (unloading). It's not a "one size fits all" approach, however. Positive displacement compressors, such as scroll and helical rotary compressors, operate differently than centrifugal compressors.

Scroll compressors. Scroll compressors can be found in many lines of refrigeration equipment including packaged direct expansion equipment, heat pumps, and small chillers. Scroll compressors employ numerous methods and designs for unloading including cycling, hot gas bypass, digital, and variable-speed drives. Some unloading methods maintain compressor rotation but reduce the quantity of refrigerant being compressed (e.g., digital-scroll compressors). Other methods route compressed refrigerant vapor around the expansion device, bypassing the evaporator (e.g., hot gas bypass). These capacity control methods present unloading capabilities, but there isn't a significant reduction in energy usage at part load. Other methods vary the rotational speed and compression ability of the compressor. A VSD can be used to control the speed of the orbiting scroll.

VSDs offer better part-load operation than constant-speed, dual-manifolded compressors. In turn, dual-manifolded compressors are significantly more efficient than digital scroll compressors or those that use hot gas bypass (HGBP) as shown in Figure 6. Systems that use multiple scroll compressors might employ a single variable-speed scroll with several other constant-speed compressors manifolded together. In this design, the constant-speed compressors are staged while the variable-speed scroll compressor is varied to precisely meet the cooling load. In addition to saving energy, this allows the unit to unload and meet the cooling load more efficiently than other unloading methods.

Figure 6. Scroll compressor unloading methods
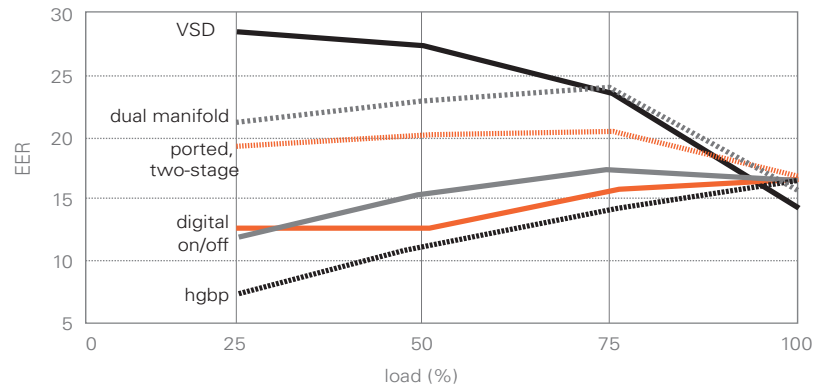
Helical rotary compressors. Helical rotary, or screw, compressors are positive displacement devices that use a combination of male and female rotors (or screws) to compress refrigerant. Only the male rotor is powered by an electric motor and the female rotor is rotated as the male lobes engage it. The rotors counter-rotate within a housing that has tight tolerances. A film of oil is used to provide a seal between the meshing rotors and housing. This oil film prevents refrigerant vapor from escaping.

Refrigerant vapor enters the compressor through the intake port and fills the pockets created by the rotor lobes. The refrigerant vapor is confined to the pocket created by the rotors and housing as the rotors spin. During this rotation, the available pocket volume for refrigerant vapor decreases and the refrigerant is compressed. Traditionally, one or more slide valves have been employed to modulate the capacity of the compressor by adjusting the total volume of the compression pocket. The slide valve is integrated into the compressor housing. As the slide valve opens, the compression cavity volume is reduced and less compressed refrigerant gas is discharged out of the compressor to meet the evaporator load. The slide valve allows the machine to vary the rotor length used for compression.

With VSDs, the rotational speed of the screws can be controlled and modulated as load changes. Because capacity can be modulated by controlling the speed, slide valves may be omitted when a VSD is used. Adding a variable-speed drive increases the full load energy rate as shown in Figure 7. At part load, the rotational speed of the screw is reduced, which produces the same effect as a modulating slide valve—a reduced volume of refrigerant is discharged from the compressor. With a reduction in refrigerant vapor, the evaporator capacity is reduced.

Figure 7. Helical rotary (screw) compressor power consumption versus load at constant entering condenser water conditions and AHRI relief conditions
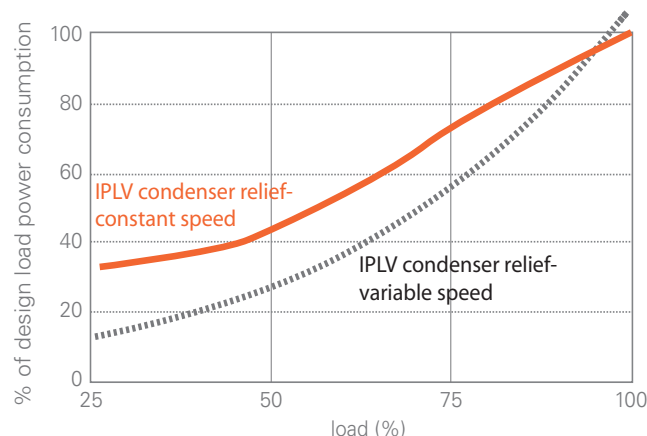
Centrifugal compressors. A centrifugal compressor behaves differently when coupled with a VSD on the motor. Compressing refrigerant with centrifugal technology requires the conversion of velocity pressure into static pressure—essentially pushing the vapor off the end of a rotating impeller into a diffuser that converts the pressure.

The velocity of the refrigerant leaving the impeller can be broken into two components: radial and tangential velocity. The radial velocity is proportional to the refrigerant flow rate and can be modulated by the inlet guide vanes to control capacity. The tangential velocity is proportional to tip speed, which is a function of the impeller's rotational speed (rpm) and diameter (Figure 8.) Centrifugal compressor impellers are typically radial flow. Some centrifugal compressors may incorporate "mixed-flow" designs. Mixed flow impellers also have an axial flow component within the discharge flow. For the following discussion, a radial flow impeller is assumed and for our purposes, a mixed flow impeller will behave similarly.

The resultant velocity pressure is the combination of radial and tangential components. Once converted to static pressure, the outlet pressure must be high enough for the refrigerant to flow out of the compressor, or the compressor will surge (refrigerant flows backward).

The difference between the condenser and evaporator conditions is called *lift* (temperature difference) or head (pressure difference). *Load* (cooling capacity) is independent of pressure and dependent on the evaporator water flow rate and temperature difference.

At part load, the inlet guide vanes close to restrict refrigerant flow and radial velocity decreases. The compressor has to do less work to compress a reduced flow of refrigerant vapor. This lowers the power consumption—all at a constant compressor speed (Figure 9).
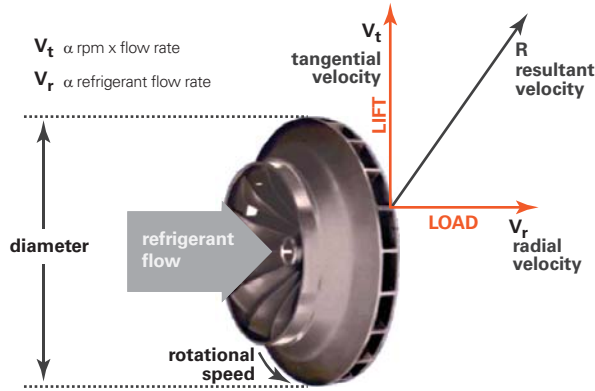
Figure 8. Impeller dynamics

$V_t \ \alpha$ rpm x flow rate

$V_r \ \alpha$ refrigerant flow rate

Figure 9. Compressor response to reduced load

full load   part load

Figure 10. Compressor response to lift

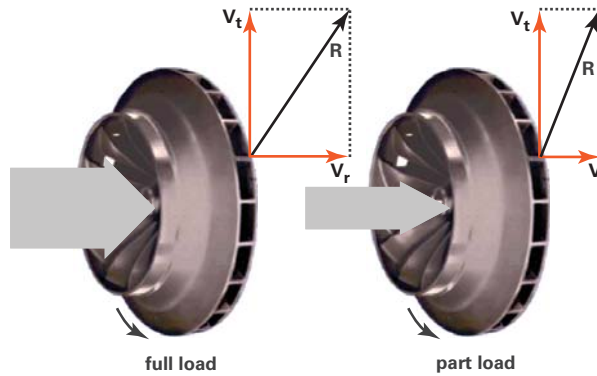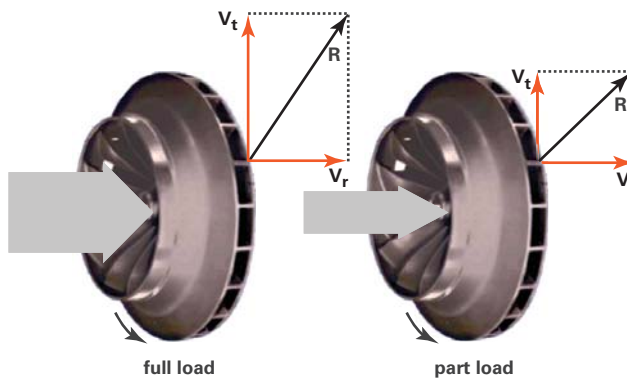full load   part load

In a part lift scenario (Figure 10), the condenser pressure is lower (corresponding to a lower leaving condenser water temperature) or the evaporator pressure is higher (corresponding to an increased evaporator leaving water temperature).

Lower condensing pressures and/or higher evaporating pressures reduce the amount of compressor work needed. Chillers with both inlet guide vanes and VSDs increase the benefit at

part lift conditions especially by coordinating the actions of the two together to meet the operating requirements for load and lift. At or near full load, variable-speed drives have slightly higher energy use due to the inefficiency of the drive.

Centrifugal compressors with VSDs do not experience "power varies with the cube of the speed" relationships. With reduced lift, energy savings can be achieved—the question is: how much?

A VSD should be considered when a substantial number of part-lift operating hours are expected, cool condenser water can be obtained, the utility rates are favorable (low demand charges), when chilled-water reset will be employed or a combination of these conditions exists.

As with scroll chillers, using one variable-speed compressor and one or more constant-speed compressors, chilled-water plants can be designed to blend the benefits of both technologies. Figure 11 shows a comparison of a 600-ton chiller with a VSD and an equivalent cost chiller with a constant-speed motor and more heat transfer surface area. Both chillers represent the same capital investment but operate differently as condenser-water temperatures and loads change. At higher condenser-water temperatures and loads, the constant-speed chiller is more efficient; at lower condenser-water temperatures (lower lift), the variable-speed driven chiller is more efficient. The chiller plant controller can deploy the appropriate sequencing to deliver the best efficiency at all operating conditions.

Summary

Applying a variable-speed drive to an HVAC component does not always yield the same result. System designers should understand the impact before applying a variable-speed drive to a piece of equipment.

Free discharge fans, such as cooling tower fans, are excellent candidates for variable-speed drives given there is no fixed pressure component and experience near cubic energy savings.

Fans in ducted airside systems require dynamic setpoint control strategies for VSDs to achieve near-cubic energy savings. Part-load savings can be realized with variable-speed drives.

Figure 11. Centrifugal chiller power demand versus demand at design and reduced entering water condenser water
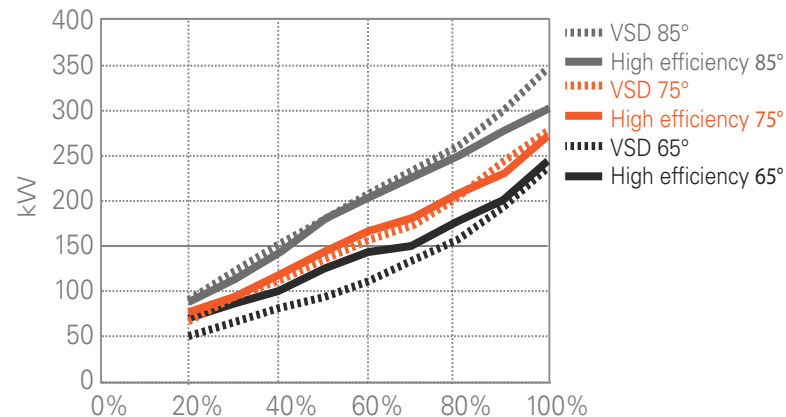
Pumps do not experience cubic energy savings. Part-load savings can be realized with variable-speed drives, especially when controlled optimally through strategies such as critical-valve reset. Optimization may be required by your energy code.

Scroll compressors do not experience cubic energy savings; however, variable-speed drives reduce energy consumption at part load compared to other capacity modulation strategies.

Helical-rotary compressors do not experience cubic energy savings however variable-speed drives do offer potential energy savings over slide valves.

Centrifugal compressors do not experience cubic savings. At part load and, more importantly, part-lift conditions, variable-speed technology has the potential to save enough energy to justify the investment.

- Understand the impact of the technology on the specific component,
- decide if your application benefits from that impact,
- then determine if the economic benefit warrants the investment.

By Eric Sturm, Susanna Hanson and Jeanne Harshaw, 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.

References

[1] American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE). ANSI/ASHRAE/IESNA Standard 90.1-2010: *Energy Standard for Buildings Except Low-Rise Residential Buildings*.

[2] American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). "HVAC Systems and Equipment" Handbook (2012).

[3] Cline, L., D. Eppelheimer, R. Geister, M. Schwedler. "Variable-speed Drives and Their Effect on System Components," Engineers Newsletter Live program (2006).

[4] Dieckmann, J., K. McKenney, J. Brodrick, Ph.D., American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc. (ASHRAE). Journal Emerging Technologies Series, Variable Frequency Drives.

[5] Hanson, S., M. Schwedler, B. Spalding. "Upgrading Existing Chilled Water Plants," Engineers Newsletter Live program (2011).

[6] Murphy. J. "Energy Saving Control Strategies for Rooftop VAV Systems." *Engineers Newsletter* 35-4 (2006).

[7] Stanke, D. "VAV System Optimization: Critical Zone Reset" *Engineers Newsletter* 20-2 (1991).

[8] Trane Air-Conditioning and Economics (TRACE™ 700). Download trial software at www.trane.com/TRACE.

Join Trane in 2014 for Engineers Newsletter LIVE!

Mark your calendars!

For event details and registration contact your local Trane office. Visit www.trane.com/ENL for past programs

LEED v4.

LEED v4 will officially launch at Greenbuild 2013. Trane applications engineers will discuss changes in the newest version of LEED and how they impact HVAC practitioners.

Applying Variable Refrigerant Flow.

All HVAC systems have their own set of application challenges. This program will discuss some of the challenges when applying a variable refrigerant flow (VRF) system, such as complying with ASHRAE Standards 15 and 90.1, meeting the ventilation requirements of ASHRAE Standard 62.1, zoning to maximize the benefit of heat recovery and the current state of modeling VRF.

Energy-Saving Strategies for Chilled-Water Terminal Systems.

This ENL will discuss system design and control strategies for reducing energy use in chilled-water terminal systems including variable-speed ECM terminal fan operation, impact of ventilation system design, low-flow chilled-water system design, waterside economizing, waterside heat recovery, and meeting ASHRAE 90.1 requirements.
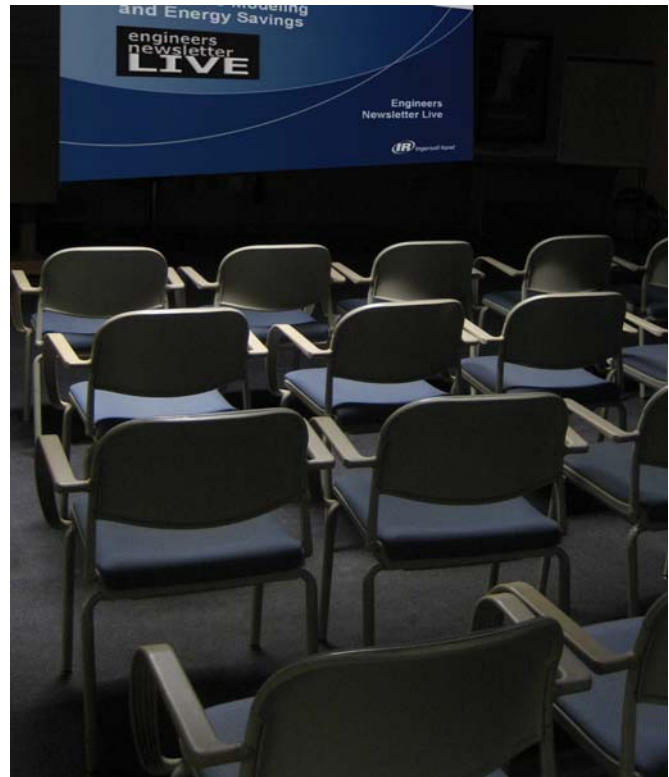
Engineers Newsletter Live. A series of 90-minute programs that provide technical and educational information on specific aspects of HVAC design and control. Topics range from water- and airside system strategies to ASHRAE standards and industry codes. Programs are GBCI and AIA accredited. Contact your local Trane office for a event details. View past programs by visiting **www.trane.com/ENL.**

Learn HVAC design strategies and earn credit

Air conditioning clinics. A series of educational presentations that teach HVAC fundamentals, equipment, and systems. The series includes full-color student workbooks, which can be purchased individually. AIA approved for 1.5 (HSW) learning units. Contact your local Trane office to sign up for training in your area.

On-demand continuing education credit for LEED® and AIA . These 90-minute on-demand programs are available at no charge. For a list of HVAC topics including many LEED-specific courses visit **www.trane.com/continuingeducation.**

Engineers Newsletters. These quarterly articles cover timely topics related to the design, application and/or operation of commercial, applied HVAC systems. Subscribe at **www.trane.com/EN.**

Application manuals. Comprehensive reference guides that can increase your application knowledge of a broad range of commercial HVAC systems. The following are just a few examples. Please visit www.trane.com/bookstore for a complete list.

Central Geothermal Systems . Discusses proper design and control of central geothermal bidirectional cascade systems including system piping, system design considerations, and airside considerations. (SYS-APM009-EN, February 2011)

Chilled-Water VAV Systems . Focuses on chilled-water, variable-air-volume (VAV) systems includes discussion of advantages and drawbacks of the system, review of various system components, solutions to common design challenges, system variations, and system-level control.
(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 October 2013)