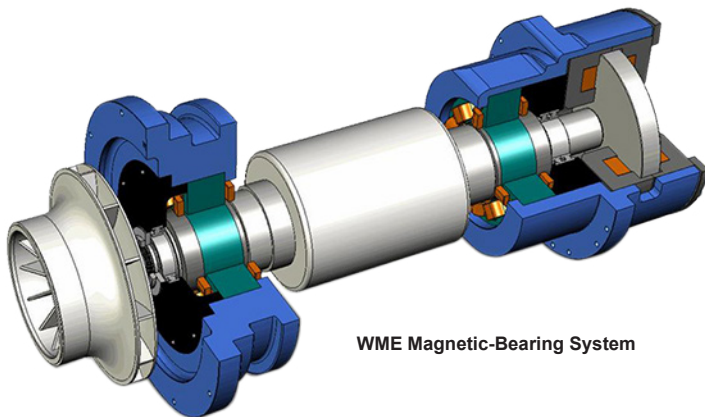
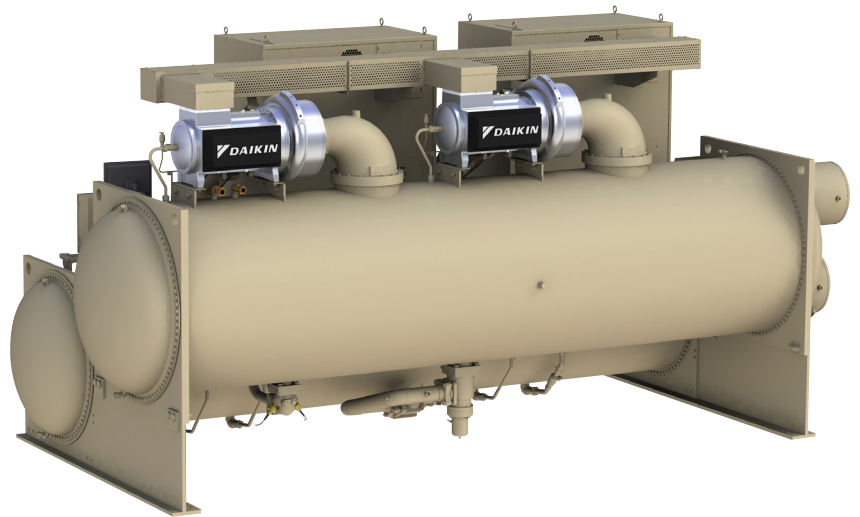
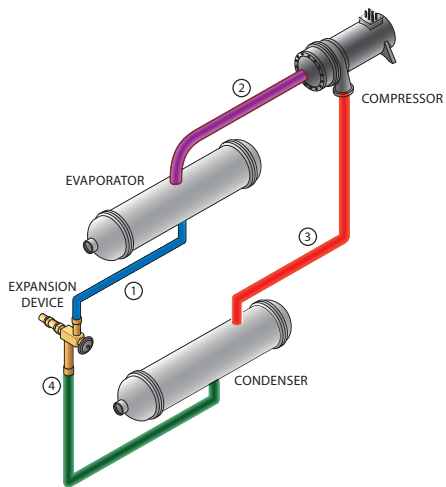


Application Guide

Revision: February 2021

Centrifugal Chiller Fundamentals



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Introduction

There are over 80,000 centrifugal chillers in operation in North America. They are usually the most economical means to cool large buildings. Most design engineers will sooner or later use centrifugal chillers to meet their design needs.

A general understanding of their design and operating characteristics will assist in applying the product properly and avoiding major pitfalls.

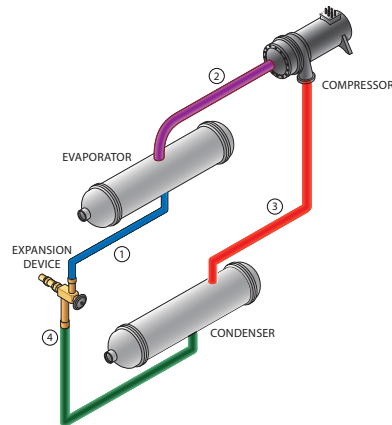
The purpose of this manual is to provide design engineers with a basic knowledge of how centrifugal chillers operate so that they will better understand the interdependency of the chiller and the other components of the chilled water plant.

Although this guide is generally non-proprietary, examples and some descriptions of chiller fundamentals and features use Daikin Applied nomenclature and model names and numbers.

Basic Refrigeration Cycle

A centrifugal chiller utilizes the vapor compression cycle to chill water and reject this heat collected from the chilled water plus the heat from the compressor to a water loop cooled by a cooling tower. [Figure 1](#) shows the basic refrigeration circuit. It consists of the following four main components;

Figure 1: Basic Refrigeration Cycle



Evaporator

The evaporator in a centrifugal water cooled chiller is usually a shell and tube heat exchanger that removes heat from the entering chilled water lowering its temperature in the process. The heat is used to boil the refrigerant changing it from a liquid to a gas. Daikin chillers use a flooded type evaporator, which is very energy efficient. Flooded evaporators place the chilled water in the tubes and refrigerant in the shell completely submerging the tubes in refrigerant. Large chillers can have over five miles of tubing in their heat exchangers.

Compressor

The compressor assembly is made up of a prime mover and a centrifugal compressor. Daikin chillers use liquid refrigerant cooled hermetic electric motors. The centrifugal compressor is a dynamic device similar to a centrifugal water pump. It raises the pressure and temperature of the refrigerant by converting kinetic energy into pressure. In this document we will refer only to centrifugal compressors.

Condenser

Like the evaporator, the condenser is usually a shell and tube heat exchanger. In this case, it removes heat from the refrigerant gas causing it to condense to a liquid. The heat raises the temperature of the cooling water often referred to as condenser water. The condenser water then carries the heat to the cooling tower where the heat is rejected to atmosphere.

Expansion Device

After the refrigerant condenses to a liquid, it passes through a pressure reducing device. This can be as simple as an orifice plate or as complicated as an electronic modulating expansion valve. Daikin chillers use either a thermal expansion valve for chillers over 600 tons or an electronic modulating expansion valve for chillers under 600 tons to give excellent modulation with a wide range of capacity and temperature conditions.

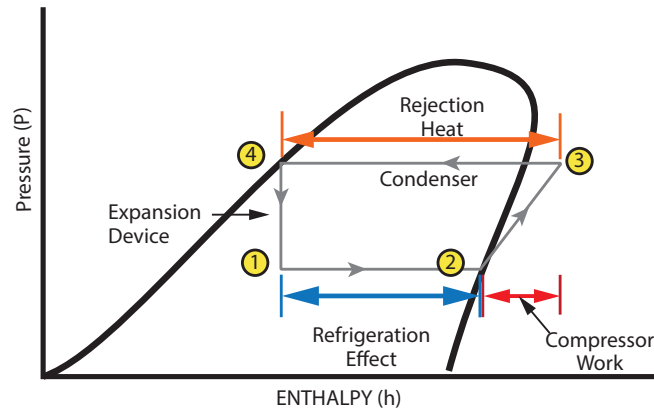
Pressure-Enthalpy Diagram

The Pressure-Enthalpy (P-H) diagram is another way of looking at the refrigeration cycle. It has the advantage of graphically showing the process, the cooling effect and the work required to make it happen.

Figure 2 shows the Pressure-Enthalpy (P-H) diagram for the same refrigeration circuit shown in Figure 1. The process for each of the components is indicated. The evaporator process is from point 1 to point 2. As the refrigerant changes from a liquid to gas, the pressure stays constant. The heat is being absorbed as a phase change (latent energy). The refrigeration effect is the change in enthalpy from 1 to 2, simply expressed as BTU/lb. of refrigerant circulated.

The line from 2 to 3 represents the compression process. The work is the change in enthalpy from point 2 to point 3. The BTU/lb. times the lb./min equals compressor power input. Work of compression ends up as heat in the refrigerant. The vertical aspect of the curve shows the rise in refrigerant pressure from 2 to 3.

Figure 2: Refrigeration Circuit, P-H Diagram

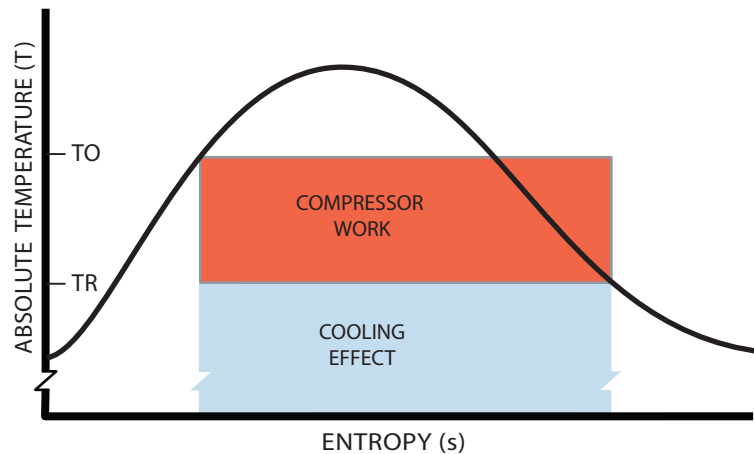


The next process takes place in the condenser. (from 3 to 4) The first section (outside the refrigerant dome) is the desuperheating process. Once the refrigerant is saturated, condensation occurs and the refrigerant changes from a gas to a liquid. Like the evaporator, the line is horizontal indicating constant pressure. Some condensers are capable of providing liquid subcooling to the left of the dome. It is easy to see on the P-H diagram, how subcooling would increase the total cooling effect. It increases the refrigeration effect per pound of refrigerant (a larger Δh) so that more cooling is done without an increase in compressor power input.

The final process is the expansion device. This shows as vertical line from point 4 to point 1, indicating the pressure drop that occurs as the refrigerant passes through the expansion valve.

Chillers provide chilled water at a constant temperature, as desired. Typically they reject heat into the atmosphere, but this could also be a river or ocean. Chillers consume work (in the compressor) to move the heat against the natural gradient. The larger the temperature difference (lift) between the chilled-water temperature and the heat sink, the more energy the chiller consumes. This can be seen by reviewing the Carnot cycle, which is the ideal heat cycle.

Figure 3: Carnot Cycle



Carnot efficiency is defined as;

$$\text{COP}_{\text{carnot}} = \text{TR} / (\text{TO} - \text{TR})$$

Where:

COP is Coefficient of Performance

TR is the temperature of the region to be refrigerated in °R

TO is the temperature of the region where the heat is to be rejected in °R

It can be seen from the definition of Carnot efficiency, that increasing the lift ($\text{TO} - \text{TR}$) lowers the COP. This single issue affects chiller efficiency more than any other.

Using typical chiller operating conditions yields a Carnot efficiency of 0.348 kW/ton. No chiller can ever beat this performance or, due to the second law of thermodynamics, even come close.

Typical Operating Conditions

The design conditions imposed by most water-cooled HVAC systems work very well for centrifugal chillers. The Air-Conditioning Heating and Refrigeration Institute (AHRI) provides test standards and certification for a wide range of HVAC products including centrifugal chillers. The ANSI/AHRI 550/590-2011: *Performance Rating of Water-Chilling and Heat Pump Water-Heating Packages Using the Vapor Compression Cycle* is used to test and rate chillers. Additionally, chillers typically have a certification that provides engineers and owners with a third party validation that the chiller will meet the performance the manufacturer indicates. The AHRI test criteria allows an “apples to apples” comparison of different chillers.

The standard AHRI rating condition is:

- Leaving chilled water temperature 44°F
- Chilled water flow rate 2.4 gpm/ton
- Entering condenser water temperature 85°F
- Condenser water flow rate 3.0 gpm/ton
- 0.0001 evaporator fouling factor and 0.00025 condenser fouling factor

The temperature change in the fluid for either the condenser or the evaporator can be described using the following formula;

$$1. Q = W \times C \times \Delta TF$$

Where

- Q = Quantity of heat exchanged (btu/hr or kw)
- W = mass flow rate of fluid (lb/hr or kg/hr)
- C = specific heat of fluid (btu/lb°F or kJ/(kg•K))
- ΔTF = temperature change of fluid (°F or °C)

Assuming the fluid is water, the formula takes the more common form of;

$$2. \text{Load (btu/hr)} = \text{Flow (USgpm)} \times ({}^{\circ}\text{F}_{\text{in}} - {}^{\circ}\text{F}_{\text{out}}) \times 500$$

Or

$$3. \text{Load (tons)} = \text{Flow (USgpm)} \times ({}^{\circ}\text{F}_{\text{in}} - {}^{\circ}\text{F}_{\text{out}}) / 24$$

Using this equation and the AHRI design conditions, the temperature change in the evaporator is found to be 10°F. The water temperature entering the evaporator is then 54°F.

Recall that the heat that needs to be removed from the condenser is equal to the heat collected in the evaporator plus the work of compression. Assuming the work of compression is 25% of the heat collected in the evaporator, then the heat rejected in the condenser will be 125% of the evaporator heat.

Using the above equation and the AHRI design conditions, the temperature change in the condenser for modern high efficiency chillers is found to be 9.4°F at 3 gpm/ton. The water temperature leaving the evaporator is then 94.4°F. This is often incorrectly rounded off to a 10 degree delta T and a 95.0°F leaving water temperature.

The AHRI design conditions are frequently used as design conditions. Although they represent good “average” conditions to use, they may not represent the best design conditions to use for every project.

Figure 4: Heat Exchanger Performance[PH1]

R-134a LMTD and Lift

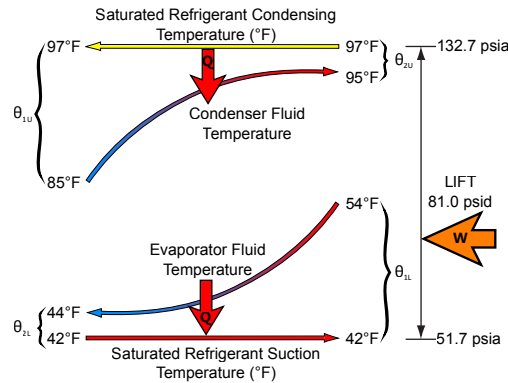


Figure 4 shows the lift requirements for both the condenser and the evaporator. Using the AHRI design conditions, typical temperatures are shown. Looking at the condenser, the refrigerant temperature remains constant at 97°F. The refrigerant is changing from a gas to a liquid (condensing) and is releasing its heat to the tower water which is entering the condenser tubes at 85°F and is gaining heat causing a temperature rise to approximately 95°F.

The evaporator behaves similarly. In this case, the evaporator refrigerant temperature remains constant at 42°F. The refrigerant is changing from a liquid to a gas (boiling) while absorbing heat from the water which is entering the evaporator tubes at 54°F and leaving at approximately 44°F.

The pressure in either the evaporator or condenser will be the saturation pressure for the given temperature. These can be found on temperature-pressure charts. For HFC-134a, the condenser pressure at 97°F is 132.7 psig. The evaporator pressure at 42°F is 51.7 psig.

The rate at which the heat moves from one fluid to the other can be described by equation 4 and 5.

$$4. Q = U \times A \text{ LMTD}$$

And

$$5. \text{LMTD} = \Delta T_F / \text{Log}_e (\theta_{1U} / \theta_{2U})$$

Where; (for the condenser)

- Q = Quantity of heat exchanged (BTU/hr or kW)
- U = overall heat transfer coefficient (BTU/(hr × ft² × °F))
- A = area of heat exchanger tubes (ft²)
- LMTD = Log Mean Temperature Difference between the fluid and the refrigerant (°F or °C)
- ΔT_F = Temperature change of fluid (°F or °C)
- θ_{1U} = Entering temperature difference (°F or °C)
- θ_{2U} = Leaving temperature difference (°F or °C)

Some important relationships can be gleaned from reviewing these equations. Adding tubes (increasing surface area A) will improve heat transfer. It also lowers the fluid pressure drop. The downside to adding tubes is it adds cost.

Increasing the heat transfer coefficient U improves heat transfer. Most chillers utilize copper for tubing. Changing to a material with poorer heat transfer properties in the condenser will also hurt performance. The heat transfer coefficient can be improved by going to internally rifled tubing. The rifling adds surface area and increases turbulence to improve overall heat transfer. Enhancing the outside surface of the tube provides nucleation sites to improve boiling. These are parameters that the chiller manufacturer controls in the design of the chiller.

Decisions made by the chiller manufacturer or the design engineer can affect the Log Mean Temperature Difference (LMTD). Changing the saturated suction temperature or the saturated condensing temperature may change θ_1 and θ_2 . If the saturated suction temperature is lowered from 42°F to 40°F, the LMTD will increase for the same leaving evaporator water temperature. It would then be possible to remove tubes (reducing the tube area A) from the chiller and still maintain the original heat transfer rate. The evaporator will cost less. However, the new saturated pressure is lowered to 35 psig. The compressor lift is increased from 81.7 psig to 83.3 psig. The compressor will have to work harder to do the job.

The design engineer's choices for operating water temperatures will also affect the heat transfer. Changing the return evaporator water temperature from 54°F to 58°F (switching from a 10°F ΔT to a 14°F ΔT) increases ΔT_F and improves the LMTD. However, changing the leaving condenser water temperature from 95°F to 100°F (switching from a 10°F ΔT to a 15°F ΔT) increases ΔT_F and *lowers* the LMTD. The solution to offset the larger condenser penalty ΔT is more tubes (increasing the surface area) or raising the saturated condensing pressure. The later makes the compressor work harder.

Centrifugal Compressor Theory

Figure 4 shows the saturated temperatures for both the evaporator and condenser. As mentioned earlier, the saturated pressures for these temperatures are known. At typical AHRI conditions, the required pressure increase or *lift* is 81.0 psig for R-134a. The purpose of the compressor is to provide this lift.

Centrifugal compressors differ from positive displacement compressors (such as scroll, reciprocating and screw). Centrifugals are aerodynamic or turbine type. They move gas by converting kinetic energy to pressure energy. Positive displacement compressors encase a quantity of refrigerant in a decreasing volume during the compression process. They provide excellent lift characteristics. The advantage of centrifugal compressors is their high flow rates capability and good efficiency characteristics.

The simple way to understand the principle is to imagine a ball on the end of a string. One person swings the ball on the string. A second person is standing on the second floor balcony. If the person swinging the ball releases the ball with enough angular momentum, the ball will fly up to the person on the second floor balcony. The weight of the ball (molecular weight), the length of the string (wheel diameter) and the rotational speed (rpm) affect the angular momentum.

The key parameter to notice is the lift is proportional to the impeller tip speed. Table 1 lists properties for the common refrigerants used with centrifugal compressors. Recall that the chilled and condenser water temperatures and the approach temperatures set the required lift. This is the same for any chiller. Reviewing Table 1, the required tip speed is very close (within 4%) for any of the popular centrifugal refrigerants. The required tip speed is around 650 fps. Whether there is a small wheel spinning fast or large wheel spinning slowly, the tip speed is relatively constant regardless of refrigerant.

Table 1: Refrigerant Properties

Refrigerant	HCFC 123	HFC 134a	HCFC 22
Condenser Press. psig @ 100°F	6.10	124.1	195.9
Evaporator Press. psig @ 40°F (Inches of Mercury Vacuum)	(18.1)	35.0	68.5
Net Refrigerant Effect (BTU/lb)	66.0	68.0	73.0
Refrigerant. Circulated lbs/min./ton	3.08	3.00	2.78
Gas Flow cfm/ton	18.15	3.17	1.83
Head (BTU/lb)	7.73	8.34	9.0
Tip Speed ft./sec.	656	682	707
Ozone Depletion Potential (ODP)	0.02	0.00	0.05

The actual amount of cooling a chiller performs depends on how much refrigerant (cfm) it moves through the compressor. [Table 1](#) shows how many cfm of refrigerant are required per ton of cooling for the popular refrigerants. HFC-134a requires about 3 cfm/ton while HCFC-123 requires about 17 cfm/ton. HFC-134a has a higher density.

A key design parameter for centrifugal compressors is the impeller inlet velocity of the refrigerant gas. It is necessary to stay below Mach 1. Typically, the inlet velocity is limited to about 0.9 Mach. Using a 1000 ton chiller as an example, compare chillers using popular centrifugal refrigerants.

Table 2: Compressor Design Parameters

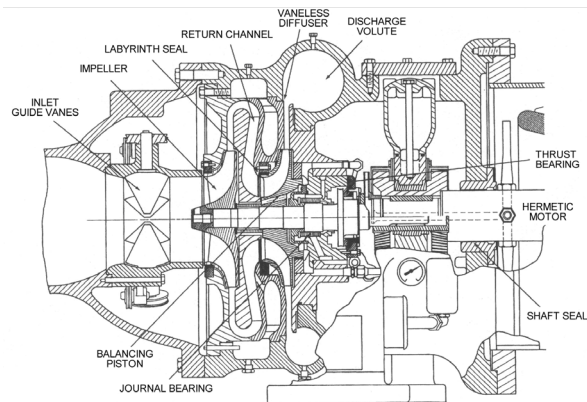
Refrigerant	HFC 134a	HCFC 22	HCFC 123
Chiller size (tons)	1000	1000	1000
Compressor Gas Flow Rate (cfm/ton)	2.68	1.74	17.08
Compressor Gas Flow Rate (cfm)	2680	1,740	17,080
Tip Speed (fps)	653	678	629
Wheel Speed (rpm)	11,884	19,464	3550 @60hz
Wheel diameter (in.)	12.6	8	40.6
Acoustic Velocity @ 50°F (fps)	484	535	417
Minimum. Inlet Diameter. (in.)	4.6	3.5	13.0

The information in [Table 2](#) defines the geometry of the compressor. Compressors based on HCFC-123 typically use direct-coupled motors so that at 60Hz, the compressor speed is fixed at 3550 rpm. The advantage of direct-drive is that no gearbox is required, however, fine tuning of tip speed is not possible without use of a variable frequency drive (VFD). Note that to achieve the correct tip speed, the wheel diameter needs to be 40.6 inches.

Traditional oiled bearing compressors based on either HFC-134a or HCFC-22 normally use a gearbox. Small compressor (5 inch diameter wheels) speeds can reach as much as 30,000 rpm. Again, the tip speed is constant around 650 fpm. It is important to note the stress on the wheel itself is proportional to the square of the tip speed *not* rpm. A large wheel spinning slowly will have the same stress as a small wheel spinning quickly if the tip speeds are the same. Note: Advanced technology magnetic bearing centrifugal chiller compressors are discussed in a later section.

The large wheel diameters required for HCFC-123 put a design constraint on the compressor design. The wheel must be encased in a volute to collect the refrigerant as it leaves the wheel. A common solution to reduce the wheel diameter is to use two or three stage compressors. [Figure 5](#) shows a cut away of a typical two-stage compressor. To improve compressor efficiency, refrigerant economizers are often used with two stage compressors. [Figure 6](#) shows the P-H diagram for a two-stage compressor with economizer.

Figure 5: Two-Stage Compressor Cutaway



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Figure 6: Two-Stage Compressor P-H Diagram

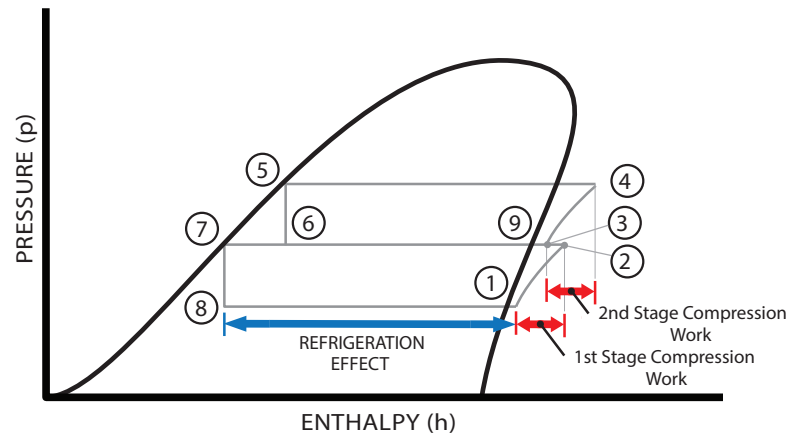
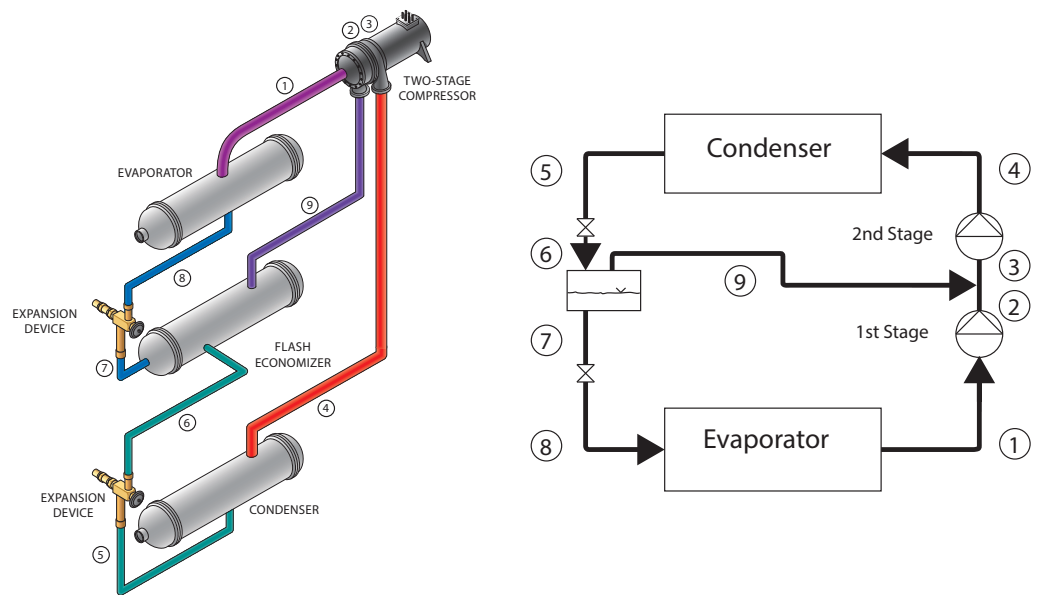


Figure 7: Two-Stage System



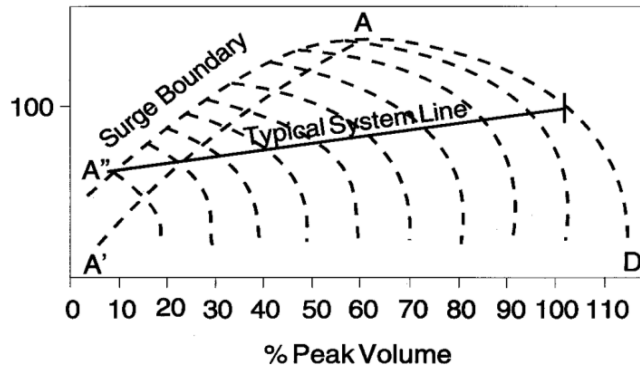
Besides reducing the wheel diameter and reducing casing size, the two-stage economizer compressor has a theoretically more efficient refrigeration cycle. In this case the refrigerant goes through two expansion devices. When the refrigerant goes through the first device, some of the refrigerant flashes, or becomes a gas. The flashed refrigerant is introduced to the compressor between the two stages. It has the effect of “cooling” the superheated refrigerant gas exiting the first stage.

The balance of liquid refrigerant passes through a second expansion device and goes to the evaporator suction with lower enthalpy than if it had been flashed in one step from the condenser pressure. Consequently, there is less mass flow through the evaporator.

Compressor Surge and Stall

Figure 8 shows a typical compressor curve. Like a fan curve, the area to the left represents unstable compressor operation. Returning to the example of the ball and string, if the ball is released toward the person on the second floor balcony but stops short of reaching the person, then a *stall* has occurred. In reality, the refrigerant is no longer moving through the compressor and there is no cooling effect. Worse, all the shaft work is being converted into heat in the compressor that may lead to permanent damage.

Figure 8: Compressor Performance Map



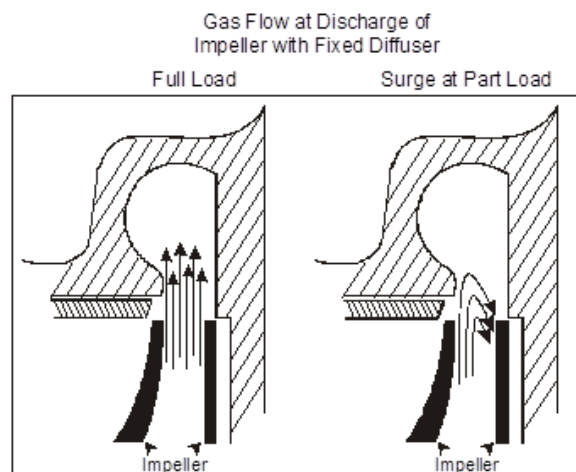
A *surge* occurs when the ball starts to fall back to the person on the ground. In this case, the refrigerant flows backward through the compressor wheel every few seconds until the pressure builds up and the refrigerant moves forward again. This is even more damaging than a stall because it reverse loads the thrust bearings in the compressor shaft.

Daikin chillers have software in their controller which protects the compressor from stalls and surges.

A properly selected chiller will not surge at the conditions it was designed for. A chiller can surge if the operating conditions are changed so that the lift is increased, especially at low load conditions. Raising the tower water supply temperature or lowering the chilled water supply temperature beyond design points can lead to surge.

Centrifugal chillers are vulnerable to surging at part load. Figure 9 shows the refrigerant gas exiting the impeller. At full capacity the gap between the impeller and the volute is sized correctly. As the chiller capacity is reduced, the refrigerant flow rate drops. The refrigerant still exits the impeller at the correct tip speed but the discharge area is too large for the reduced flow and the refrigerant stalls.

Figure 9: Compressor Movable Diffuser Geometry



Hydrodynamic (oil film) Compressor Bearings

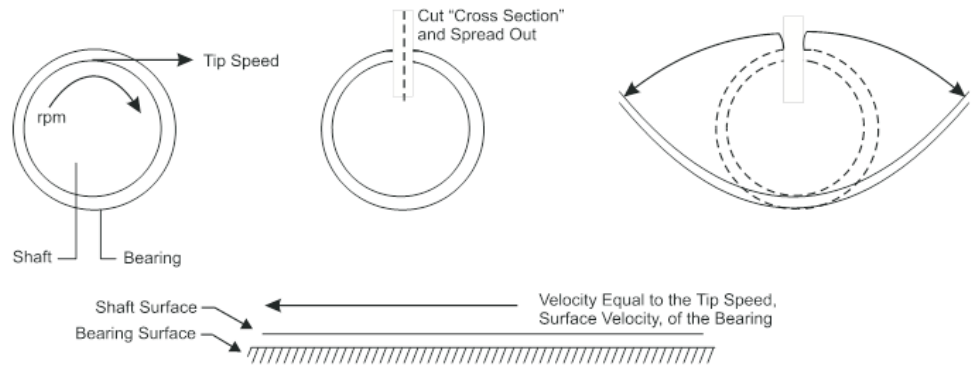
Figure 10 shows how shaft speed (rpm) is related to bearing speed. Small, light, high speed impellers have smaller diameter shafts. The shaft tip speed is low and so is the bearing relative velocity.

Sleeve bearing systems usually have a forced (oil pump) lubrication system. The oil is heated or cooled to maintain the proper temperature range.

There is no shaft-to-bearing contact in a properly designed and operating sleeve bearing. The bearing life is practically infinite. The shaft rotation creates a lubricant film that the shaft then rides on. Startup is the most critical time. Light impeller/shaft assemblies have the advantage of accelerating quickly and establishing an oil film quickly.

Sleeve bearing material is typically either babbitt, bronze or aluminum. Bearing material must be softer than the shaft material so if foreign material enters the bearing, it embeds in the bearing and not the shaft. Since aluminum is harder than babbitt, the shafts must be harder than when babbitt is used.

Figure 10: Bearing Loading



Vessel Pass Arrangements

The number of passes used is related to the water velocity in the tubes. Higher velocities improve heat transfer on the water side (inside) of the tube. Higher tube velocities raise the fluid pressure drop.

To avoid laminar flow with internally enhanced tubes, the Reynolds number for the fluids must remain above 7500. Rather than calculating the Reynolds number, common practice is to ensure fluid velocities are maintained above 3 fps for water. The maximum tube velocity is limited by tube erosion. To avoid damaging the tubes, 10 fps is typically used as an upper limit.

A balance must be struck between adequate tube velocity and low fluid pressure drop (low tube velocity). While every situation is unique, [Table 3](#) shows some general guidelines for water with no glycol.

Figure 11: Pass Arrangements

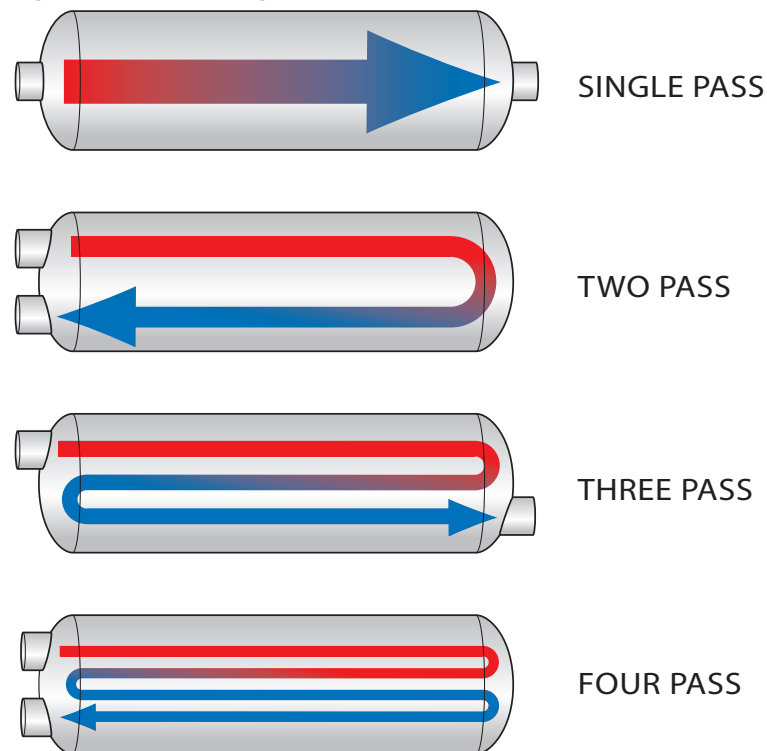


Table 3: Pass Temperature Range (no glycol)

Number of Passes	Temperature Range (°F)
1	0-5
2	6-12
3	13-18
4	18-20

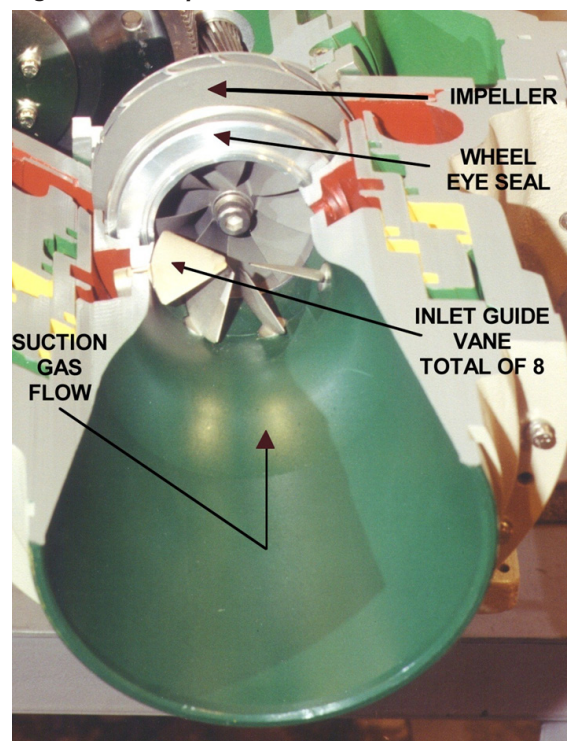
Capacity Control

The volume flow rate of the centrifugal compressor will change in response to changes in head or changes in cooling required. HVAC applications necessitate the compressor to operate over wide range of lift and cooling capacities.

Inlet Guide Vanes

Inlet guide vanes are used to control the capacity of the compressor. Figure 12 shows a cutaway of a Daikin compressor front end and the inlet guide vanes can be seen. As the inlet guide vanes start to close, they change the gas entry angle to the impeller and reduce gas flow and compressor capacity. As the vanes near the closed position, they throttle the refrigerant flow.

Figure 12: Compressor Inlet Guide Vanes



Varying Compressor Speed

Changing the compressor speed can also control compressor capacity. If the prime mover is a turbine or internal combustion engine, the prime mover's speed can easily be changed. Induction motors require a Variable Frequency Drive (VFD) to change their speed.

Changing the compressor speed also changes the tip speed. As the tip speed is lowered, the lift the compressor can produce is lowered. For compressor speed control to work, the required lift *must* be reduced either by raising the supply water temperature or lowering the condenser water temperature. The most common way to reduce the lift is to lower the condenser water temperature. As the ambient wet bulb temperature drops, it is possible to lower the condenser water temperature and realize significant savings. However, it is important to remember that unless the compressor lift is lowered, varying compressor speed cannot be done.

Variable Frequency Drives

The Variable Frequency Drive (VFD) replaces the compressor motor starter. They can be unit or remote mounted. In some cases, the VFDs have to be water-cooled. Chilled water-cooled units add load to the chilled water loop. Condenser water-cooled units do not affect the chilled water loop but are vulnerable to scaling from the open tower water and are not recommended.

Chillers with VFDs still have inlet guide vanes. The chiller controller monitors the operating conditions and uses a combination of inlet guide vanes and speed control. Compressor speed is typically only lowered to about 60% of the design speed. Since VFDs introduce drive losses with induction motors, the chiller will not be as efficient at full load (speed) with a VFD as with a standard starter. Note: Permanent magnet motor technology does not have this loss in efficiency and is discussed in a later section.

VFDs act as a soft starter. They can lower the inrush current for the motor to almost that of the full load running amps. This can be very important where chillers will operate on emergency power generator sets.

The power factor with a VFD is typically around 0.96, which is very good. However, the harmonics from a VFD can be of major concern. The chiller motor is typically the largest single electrical load in the building.

Harmonics

Despite their many benefits, care must be taken when applying VFDs due to the effect of line harmonics on the building electric system. VFDs can cause distortion of the AC line because they are nonlinear loads; that is, they don't draw sinusoidal current from the line. They draw their current from only the peaks of the AC line, thereby flattening the top of the voltage waveform. Some other nonlinear loads are electronic ballasts and uninterruptible power supplies.

Harmonics are an unavoidable issue with variable speed drives. Large VFDs produce harmonics that can be detected in building electrical distribution systems. Some large drives can produce total harmonic current distortion of approximately 30% at the drive terminals. There are measures that can be taken to counteract this inevitable harmonic distortion that occurs in chillers. These measures will result in total harmonic current distortion as low as 10-12%, and a total demand current distortion (TDD) of 8-10%, at the drive terminals.

IEEE-519

IEEE-519 is a standard for the harmonic distortion present at the point of common connection (PCC) to the electric utility in a building. For different transformer impedance levels, IEEE-519 sets different maximum allowable TDD limits. Five percent TDD is the limit set for systems with ISC (maximum short-circuit current at PCC) / I_L (maximum demand load current at PCC) ratios less than 20 A. However, IEEE-519 sets higher maximum allowable TDD percentages for systems with higher impedances.

This standard seeks to avoid distortion from affecting other buildings nearby on the same electrical distribution network. IEEE-519 is intended to be applied at the PCC, not at the drive terminals of a chiller. IEEE 519-1992 defines PCC as *“a point of metering, or any point as long as both the utility and the consumer can either access the point for direct measurement of the harmonic indices meaningful to both or can estimate the harmonic indices at point of interference.”*

If applied at the PCC, the size of the transformer, impedance, and other loads will ultimately determine if the chiller meets IEEE-519. A complete harmonics analysis is required to determine compliance. It has become common practice for specifying engineers to apply IEEE-519 to individual motors and VFDs within buildings in order to protect other sensitive equipment within the building on the same electrical distribution wiring. This is most common for hospital applications.

VFD's are found in many modern control systems and the technology has been evolving for decades. Although harmonics are associated with non-linear loads, it is extremely rare that VFD generated harmonics are an issue in systems with a minimum of 5% internal impedance.

Line harmonics and their distortion can be critical to ac-drives for three reasons:

1. Current harmonics can cause additional heating to transformers, conductors, and switchgear.
2. Voltage harmonics upset the smooth voltage sinusoidal waveform.
3. High-frequency components of voltage distortion can interfere with signals transmitted on the AC line for some control systems.

The harmonics of concern are the 5th, 7th, 11th, and 13th. Even harmonics, harmonics divisible by three, and high magnitude harmonics are usually not a problem.

Harmonic Distortion Analysis

A simple distortion analysis program is available from the local Daikin sales office. It can easily be e-mailed and provides the user with a basic look at voltage and current harmonics or voltage harmonics only.

Current Harmonics

To mitigate harmonics, increase in reactive impedance in front of the VFD helps reduce the harmonic currents. Reactive impedance can be added in the following ways:

1. Mount the drive far from the source transformer.
2. Add line reactors. They are standard equipment on WMC and WME chillers.
3. Use an isolation transformer.
4. Use a harmonic filter.

Voltage Harmonics

Voltage distortion is caused by the flow of harmonic currents through a source impedance. A reduction in source impedance to the point of common coupling (PCC) will result in a reduction in voltage harmonics. This can be done in the following ways:

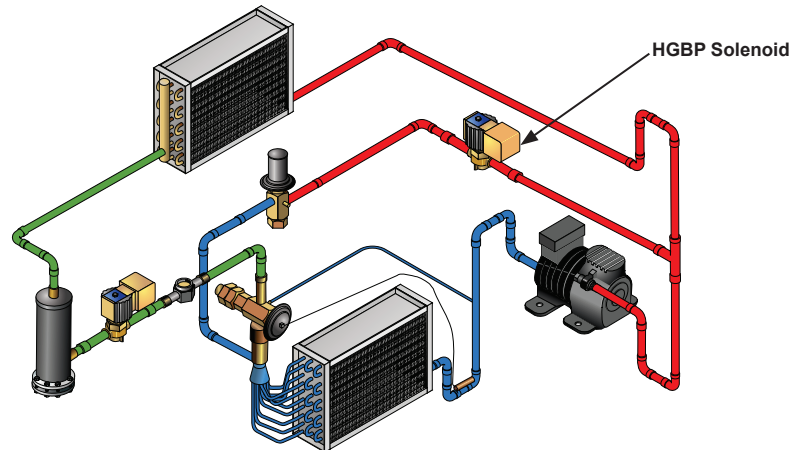
1. Keep the PCC as far from the drives (close to the power source) as possible.
2. Increase the size (decrease the impedance) of the source transformer.
3. Increase the capacity (decrease the impedance) of the busway or cables from the source to the PCC.
4. Make sure that added reactance is "downstream" (closer to the VFD than the source) from the PCC.

Hot Gas Bypass

Hot gas bypass is a means of recirculating hot discharge refrigerant back into the evaporator. The refrigerant must pass through a pressure reducing device (Hot Gas Bypass Valve). The purpose of hot gas bypass is to maintain a minimum gas volume flow rate through the compressor to avoid surging or stalling during low load conditions. A disadvantage is that the work of compression on the recirculated refrigerant does not generate any refrigeration effect.

Hot gas bypass is inefficient and should be avoided whenever possible. Careful selection of equipment size and using compressor that unload to 10 percent of full load capacity can avoid the need for hot gas bypass in most HVAC applications. Many process applications still require hot gas bypass in order to completely eliminate compressor cycling and maintain constant chilled water temperature from zero load to full load.

Figure 13: Hot Gas Bypass



Prime Movers

Most chillers use either open-drive or hermetic induction motors to drive the compressors. The Daikin oiled compressor chillers (WSC, WDC, WCC, and WTC) use semi-hermetic induction motors. Open drive motors are easier to service or replace but being air-cooled means the motors operate at hotter temperatures. The higher operating temperature places additional stress on the motor compared to refrigerant-cooled motors. A major disadvantage to open-drive is that the compressor must have a shaft seal that will leak refrigerant and require a high maintenance effort. Internal combustion (I.C.) engines operating on natural gas, propane and diesel are also used. Steam turbines are sometimes used on large tonnage chillers.

Internal combustion engines offer the opportunity to use a primary energy source and the waste heat from the engine can be collected and used. In such applications COPs around 2 (from a primary energy source) are possible.

For cogeneration applications where waste steam may be available, turbine driven chillers can be a good fit.

For both I.C. engine's and turbine's first cost and maintenance costs are higher than induction motors. Careful economic analysis is required to ensure these more expensive prime movers are viable.

Power Factor

Electrical energy is consumed in varying degrees of three forms;

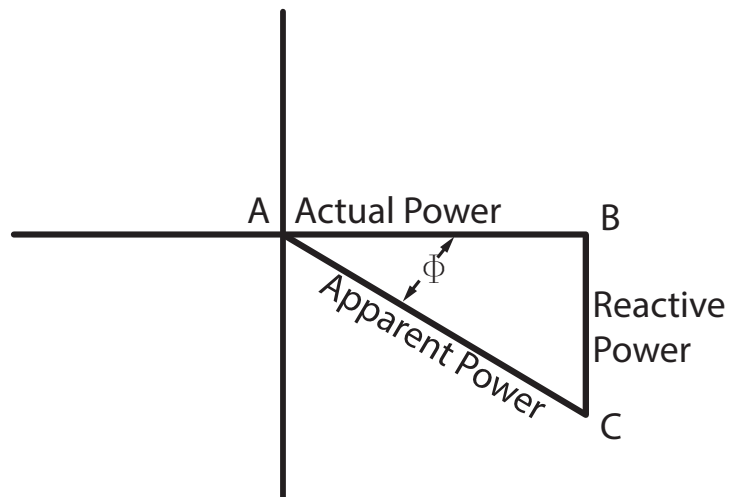
- Resistance
- Inductive Resistance
- Capacitive Resistance

When electrical energy is consumed in the resistance component, real work is done. Examples include lighting and resistance heating. Resistive work is measured in watts.

When electricity is applied to a pure inductor no real work is done. Examples of inductive loads are transformers and lighting ballasts. The inductive reactive power is measured in kilo-volts-amperes-reactive or kVAR.

When electricity is applied to a capacitor, no real work is done. The capacitive reactive power is measured in kVAR. Capacitive reactive power can "cancel" inductive reactive power.

Figure 14: Power Triangle



In reality there are no pure resistance, capacitance or inductance loads. It is always a combination of the three. Figure 14 shows the power triangle. Apparent power is the vector sum of real power and reactive power and is measured in kVA.

Utilities typically bill their customers either 100% of the real power (kW) or 90% of the apparent power (kVA), whichever is larger. If the reactive portion of the power triangle is not controlled, the operating cost will go up.

Power factor (p.f.) is the ratio of actual power (kW) to apparent power (kVA). The large inductive motors associated with centrifugal chillers increase the reactive power portion of the building's power triangle. To minimize this, capacitors are often added.

Most centrifugal motors have a power factor between 0.87 and 0.91. Capacitors can be added to raise the power factor to a practical limit of 0.95. Correction above 0.95 is not recommended since voltage upsets could produce a leading power factor (greater than 1.0) which can damage the motor. Also, adding capacitors can cause reliability issues with solid state starter and VFDs on the site.

The power factor decreases as the motor load is reduced. Below 50% without correction, the power factor drops very quickly. By raising the power factor to 95% at full load, the power factor will remain satisfactory over a wider operating range.

Motor Starters

Large chillers can have motors in excess of 1000 hp. The inrush current must be controlled to minimize the impact on the power grid. Several motor starter types are used for centrifugal chillers. These include Across-the-Line, Star-Delta, Wye-Delta, Autotransformer, Primary Reactor and Solid State. A detailed description of these starters is beyond the scope of this document.

Refrigerant Metering Devices

Daikin chillers use either electronic modulating expansion valves or thermal expansion valves that measure and control refrigerant superheat to the compressor inlet. The valve provides the necessary pressure drop and also ensures the proper flow of refrigerant to the evaporator. As the superheat climbs indicating increasing load, more refrigerant is added. If the superheat drops, the refrigerant flow rate is lowered. This maximizes the efficiency of evaporator.

Another common metering device is an orifice plate that restricts the refrigerant flow from the high pressure side to the low pressure side of the refrigerant circuit. This type of metering device is used on low pressure refrigerant chillers that use HCFC-123.

Chiller Controls

Modern chiller controls are DDC based and employ PID (proportional-integral-derivative) loops for stable control. The controlled variable is the supply chilled water temperature. The controller

measures the supply chilled water temperature and modulates the chiller capacity to maintain the setpoint. Daikin chillers control to $\pm 0.2^{\circ}\text{F}$.

Many other parameters are measured and controlled to ensure smooth and efficient operation of the chiller. This information can be used for energy management, preventive maintenance and service diagnostics. Most manufacturers have some form of gateway to allow the chiller controlled to be connected to a Building Automation System (BAS). Daikin chillers use communication modules to allow the information to be transferred on customer's choice of protocols. (Modbus, LonWorks, BACnet)

Purge Systems

Low pressure Chillers using refrigerants such as CFC-11 or HCFC-123 have evaporators that operate in a vacuum. At AHRI conditions, a chiller using HCFC-123 has an evaporator pressure of 17.6 inches of Mercury. It is very difficult to create a perfectly sealed unit, so some air (including moisture) will leak into the chiller. The air is referred to as a noncondensable. Noncondensables create two problems. The first is the compressor does work moving the noncondensables but they offer no refrigeration effect. They can also blanket tubes with air eliminating them from doing any heat exchange work. Noncondensables [PH4] lower the real efficiency of the chiller from the cataloged or rated performance by as much as 8% at 60 % load and 14% at 100 % load.

The second issue with noncondensables is they are contaminants. Moisture, in particular, is a problem. Moisture allows the formation of acids within the chiller that can cause serious damage to motor windings and bearings.

To minimize the effect of noncondensables with low pressure chillers, an additional component called a purge unit is required. The difficulty with purge units is they are another service item and also release refrigerant to the atmosphere when they remove noncondensables. Today, purge unit efficiency is very good, however older purge systems could lose as much as 25% of the chiller's refrigerant charge per year.

Chillers using positive pressure refrigerants such as HFC-134a do not require purge units. Noncondensables do not enter the chiller during operation to damage it or reduce its efficiency.

Pumpout Systems

Pumpout systems consist of a storage tank large enough to hold the chillers entire refrigerant charge and a refrigerant pump/compressor to move the refrigerant from the chiller to the pumpout tank and back again. Their primary purpose is for servicing the chiller. They allow the charge to be stored while the refrigeration circuit is worked on.

Daikin chillers do not require a separate pumpout system. The condensers a specifically designed to hold the entire refrigerant charge.

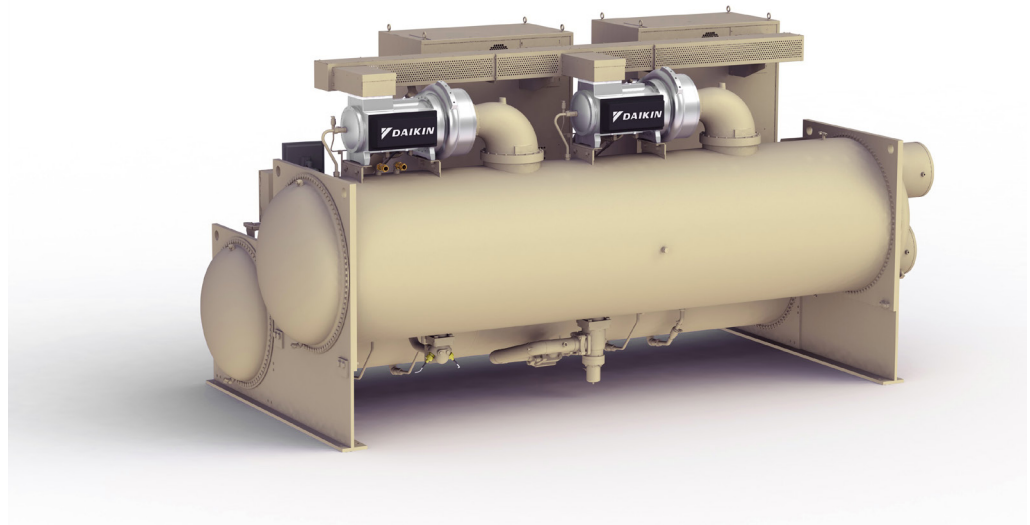
Dual Compressor Chillers

As inlet guide vanes are used to modulate the capacity of a compressor, the compressor efficiency drops off. By utilizing two compressors on a common refrigeration circuit, it is possible to greatly improve the part load efficiency of a centrifugal chiller. This is a major benefit of a dual compressor chiller.

Dual compressor chillers have two compressors operating in parallel between a common evaporator and condenser. For example, a 1000 ton dual will have two nominal 500 ton compressors. As the chiller unloads from 100% load, one compressor shuts off at approximately 600 tons (60% of full load). Only one compressor is required to circulate enough refrigerant to meet the load. That compressor is operating with its inlet guide vanes wide open and at maximum compressor efficiency. Additionally, the compressor is operating with a condenser and evaporator designed for twice the capacity. In effect, the two heat exchangers have twice the required surface area. This lowers the lift and improves the compressor performance. The need for operating only one compressor at mid-range chiller loads accounts for the superior performance of these chillers.

Dual compressor chillers have a unique performance profile. Most oiled single compressor chillers have highest efficiency at or near 100% capacity. Dual compressor chillers are most efficient at 50% to 60% capacity. This matches the typical building load profile very well, offering optimum efficiency where there are the most run hours.

Figure 15: Dual Compressor Chiller



Daikin Magnitude® Magnetic-Bearing Centrifugal Chillers

Some of the earliest patents for active magnetic-bearing technology occurred during World War II for use in ultracentrifuges intended for the enrichment of isotopes of elements needed for the Manhattan Project. These must have been rather crude devices since they came long before modern solid-state electronics allowed the fast switching necessary in shaft positioning. The technology wasn't commercialized to begin with due to very high production costs.

An early successful application was for natural gas compressors for the NOVA Gas Transmission Ltd. Gas pipelines in Alberta, Canada. Avoiding oil in these compressors eliminated one source of possible fire. The success of this application led to the development of digital controls for mag bearings in 1992. The technology grew commercially and from 1996 the Dutch oil and gas company installed 20 large mag bearing gas compressors over a ten year period using bearings produced by American companies.

In 1974 CR Meeks presented a paper entitled "Magnetic-Bearings - Optimum Design and Application", at the International Workshop on Rare Earth Cobalt Permanent Magnets, University of Dayton, Dayton, Ohio which proposed the now familiar hybrid magnetic-bearing design. These bearings use permanent magnets for bias fields and electromagnets for stability and dynamic control.

One of the most successful applications of this technology was by Danfoss Turbocor, a manufacturer of refrigeration compressors.

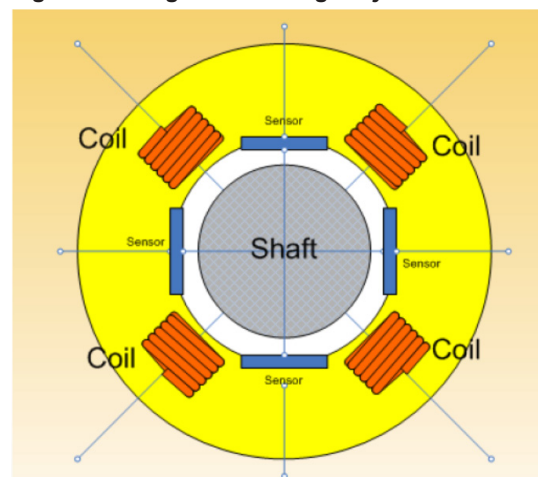
A magnetic-bearing is a bearing that supports a load using magnetic levitation. Magnetic bearings support moving parts without physical contact. For instance, they are able to levitate a rotating shaft and permit relative motion with very low friction and no mechanical wear. Magnetic bearings support the highest speeds of all kinds of bearing and have no maximum relative speed.

An active magnetic-bearing works on the principle of electromagnetic suspension and consists of an electromagnet assembly, a set of power amplifiers which supply current to the electromagnets, a controller, and gap sensors with associated electronics to provide the feedback required to control the position of the rotor within the gap. The power amplifier supplies equal bias current to two pairs of electromagnets on opposite sides of a rotor. This constant tug-of-war is mediated by the controller, which offsets the bias current by equal and opposite perturbations of current as the rotor deviates from its center position.

The gap sensors are usually inductive in nature and sense in a differential mode. The power amplifiers in a modern commercial application are solid state devices which operate in a pulse width modulation configuration. The controller is usually a microprocessor or digital signal processor.

Active bearings have several advantages: they do not suffer from wear, have low friction, and can often accommodate irregularities in the mass distribution automatically, allowing rotors to spin around their center of mass with very low vibration.

Figure 16: Magnetic bearing very limited losses



Magnetic-Bearing Compressors

The frictionless magnetic-bearing compressor was developed to improve performance, reliability and reduce service requirements as compared with conventional centrifugal compressor designs. A digital bearing control system continuously monitors shaft position and adjusts the magnetic-bearing fields in real-time to maintain precise position of the compressor shaft. The result of this technology is outstanding energy efficiency and reliable, long life operation.

The magnetic-bearing compressor is a single rotating component - the compressor shaft - levitated on a magnetic cushion. The magnetic cushion eliminates the metal-to-metal wear inherent in other bearing designs which increases reliability and eliminates the expense of compressor inspections, overhauls, and vibration analysis.

Chiller Protection

The compressor's ability to protect itself from low power quality, and to have controlled response in power loss situations, is a feature that enhances long-term compressor viability and reduces downtime. In extreme or extended power disruptions, these compressors are designed to regenerate power from the spinning motor and feed that power back to the bearings and control system. This regenerative power mode allows the compressor shaft to coast down and gently reseat onto touchdown bearings.

As a secondary system, rolling-element bearings are provided for the remote chance that the magnetic-bearing system should completely fail. These bearings have a small clearance between the shaft and the inner race and thus do not touch the shaft during normal operation. These backup bearings were tested and proven to withstand multiple losses of the magnetic-bearing system and continue to provide a safe backup for a full speed shut down.

Permanent Magnet Synchronous Motors

The magnetic-bearing is an important advancement in chiller technology but is neither the only nor the most important new technology when it comes to energy efficiency. Most of the energy use efficiency improvement in this chiller comes from the use of permanent magnet synchronous motors coupled with variable frequency drives which give the compressor such amazing part load efficiencies.

The permanent magnet synchronous motor (PMSM) can be thought of as a cross between an AC induction motor (ACIM) and a brushless DC motor (BLDC). They have rotor structures similar to BLDC motors which contain permanent magnets. However, their stator structure resembles that of its ACIM cousin, where the windings are constructed in such a way as to produce a sinusoidal flux density in the air gap of the machine. As a result, they perform best when driven by sinusoidal waveforms. However, PMSM motors provide higher power density for their size compared to ACIMs. This is because with an induction machine, part of the stator current is required to "induce" rotor current in order to produce rotor flux. These additional currents generate heat within the motor. However, the rotor flux is already established in a PMSM by the permanent magnets on the rotor.

Most PMSMs utilize permanent magnets which are mounted in the rotor. This makes the motor appear magnetically "round", and the motor torque is the result of the reactive force between the magnets on the rotor and the electromagnets of the stator. These motors are becoming increasingly popular as traction motors in hybrid vehicles, as well as variable speed applications for appliances and HVAC.

As a PMSM unloads, as in the case when the speed drive slows the motor down, its power efficiency remains high, much higher than the traditional induction motor. (See [Figure 18](#)) For as long as this motor operates at these lower speeds it is saving energy through higher efficiency. This is the way that these chillers save so much energy on an annual basis.

Figure 17: Magnetic bearing compressor cutaway

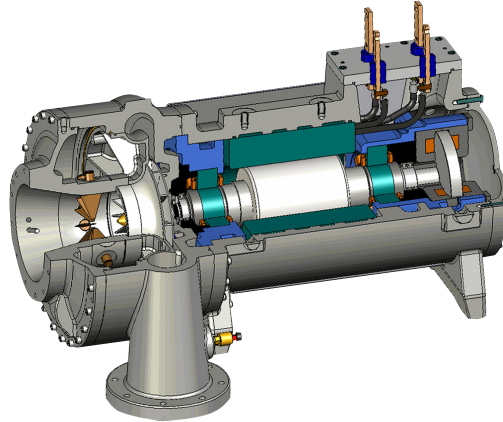
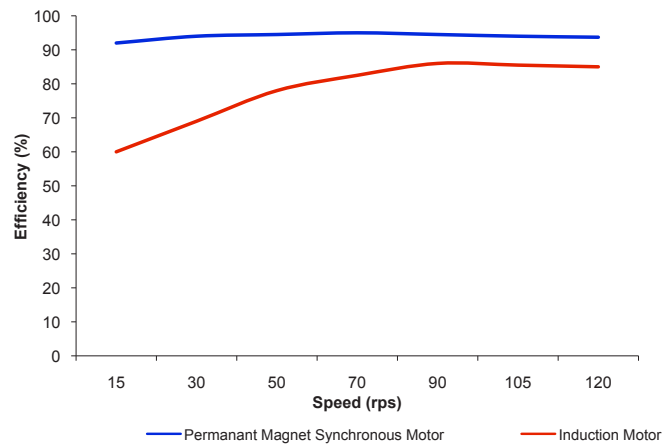


Figure 18: PMSM Efficiencies



Chiller Efficiencies

In general, the magnetic bearing chillers with VFDs are capable of outstanding part-load efficiency and the capability to operate at low-load conditions. In order to accomplish this the entering condenser water temperature (ECWT) must fall in the range of 10°F for every 25% of load as specified in the AHRI 550/590 standard.

Unloading Effects on Efficiencies

The unloading capability of a chiller will be determined by the specific application conditions. For instance, at standard AHRI conditions (See section below), a WME500 unit rated at a 550-ton full load capacity will unload to 55 tons or 10% load with the standard reductions in ECWT. However, the same WME500 chiller with a full load rating of 400 tons may still only unload to 55 tons which will be 13% of full load. Higher leaving condenser water temperatures and lower chilled water leaving temperatures will also reduce the percentage of unloading.

The most stringent unloading requirement for centrifugal chillers can be found in a single chiller system that is expected to serve a widely varying load, while maintaining a relatively constant entering condenser water temperature. In this type of application, it is strongly suggested that a chiller be capable of unloading to at least 50% of capacity with constant ECWT to ensure adequate unloading capability.

If the WME will be applied to a multiple chiller system (one where the simultaneous operation of more than one chiller is required to satisfy the design load), or a reliable load profile is available which predicts that low load operation will not be required, chillers capable of a higher percentage of unloading with constant ECWT may be acceptable.

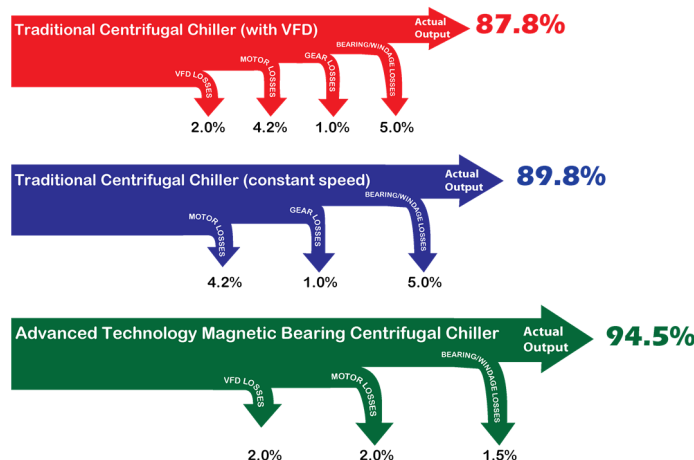
In addition, most applications will have both improved part-load stability and efficiency if unequal capacity chillers are selected to carry the total load. For instance, a 1200-ton load would likely be better served by the combination of a 500-ton and a 700-ton chiller rather than two 600-ton chillers.

Always verify unloading capabilities by running your exact job conditions in the selection software. Unloading capabilities will be limited if operation with constant ECWT is required. If it is determined that a WME chiller is not a good match for the application conditions, it may be advisable to consider a traditional gear-driven Daikin centrifugal chiller which can be configured to provide a larger unloading capability. However, this additional part load stability can be expected to come at the expense of both full and part-load efficiencies.

Unequal sizing effect on efficiency

In addition, most applications will have both improved part-load stability and efficiency if unequal capacity chillers are selected to carry the total load. For instance, a 1200-ton load would likely be better served by the combination of a 500-ton and a 700-ton chiller rather than two 600-ton chillers.

Figure 19: Efficiency by chiller type



Chiller efficiency losses

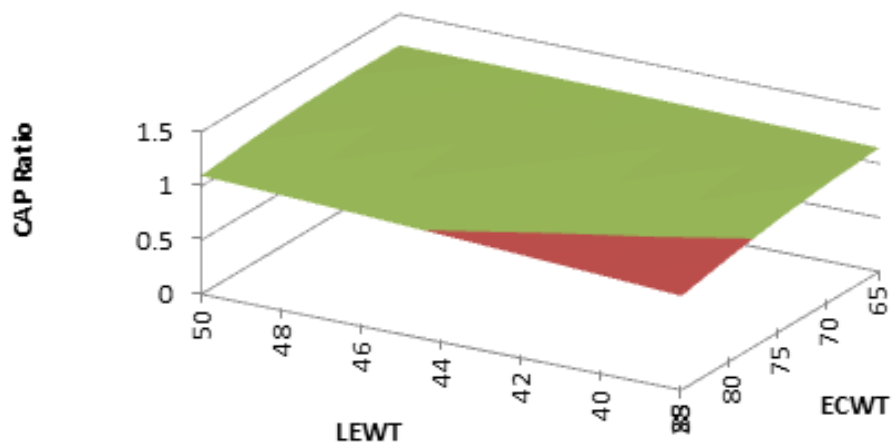
Figure 19 shows the common efficiency for several chiller types. A traditional chiller experiences losses for the VFD, motor, gear set, and bearing and windage resulting in an overall efficiency of 87.8% on average. If the VFD is removed there is a slight gain in full load efficiency to 89.9%. Because the motor is so much more efficient and because the bearings have no losses the magnetic-bearing chiller will reach 94.5% efficiency on average.

Evaporator Water Temperature Limits

Depending on your selected conditions, you may select leaving water temperatures as low as 38°F. Refer to the Applications Considerations section of the current WME catalog for a full explanation of WME operating limits. Use Daikin Selection Tools to verify capacity and performance at your specific job conditions.

Lower leaving chilled water temperatures have a negative effect on the capacity of the chiller. See Figure 20 to see the effect of leaving chilled water temperature (LEWT) and entering condenser water temperature (ECWT) on the rated capacity of a chiller. A CAP ratio of 1 indicates the full rated capacity of the machine. You can see that raising the chilled water setpoint and/or lowering the condenser water set point increases the capacity of the machine. Lowering the leaving chilled water or raising the entering condenser water reduces capacity below rated values. (See the red shaded area in the figure.)

Figure 20: Capacity vs. Temperature



Condenser Water Temperature Limits on Startup

For successful starting with low entering condenser water temperature, the goal is to get sufficient refrigerant flowing into the evaporator during the start period, in order to avoid a nuisance trip for low evaporator pressure or temperature. Starting the chiller under these conditions is a function of several variables, so it is not possible to give a fixed minimum entering condenser water temperature for starting. The following factors will have an effect:

- How much refrigerant is in the evaporator at start,
- How warm the evaporator loop is,
- How cold the condenser loop is,
- How fast the temperature of the condenser loop comes up after a start.

In most situations, the WME should be capable of operating at 55°F entering condenser water temperature. Minimum entering condenser water temperature is a function of leaving chilled water temperature, chilled water delta-T at full load, and the percent chiller load at point of operation.

These factors are dependent on the conditions at shut down, and the amount of refrigerant migration when the chiller is off, making it impossible to know the above conditions with any certainty. It is imperative that a free-cooling loop does not circulate through the condenser while the chiller is off, as all of the refrigerant could migrate to the condenser, giving very little chance of a successful start. The use of electronic expansion valves on centrifugal units blow 600 tons has greatly improved the starting capabilities when coming off of free cooling. This still does not allow for a defined fixed temperature for successful starting. A 3-way valve, used in a tower bypass loop, is recommended to give the chiller the best possible chance of starting under low entering condenser water conditions. The 3-way valve will also allow the lowest condenser water operating temperature to be found, where the chiller can continuously run without nuisance tripping. In extreme upside down conditions (high evaporator loop temperature and low condenser water loop temperature), it will be necessary to warm the condenser water up before starting the chiller. A modulating butterfly type valve could also be used, but this introduces the potential of the flow switch nuisance tripping. It is possible to bypass the flow switch temporarily until the head pressure builds up. See [Daikin Application Guide, AG-31-003, Chiller Plant Design](#), and refer to the section on water-side free cooling for additional discussion. Also refer to the Magnitude Controller Manual for information on providing analog outputs to control tower bypass valves, and tower fans or a combination of both. This control capability is standard in the Magnitude unit controllers. All modulating type valves must be field-supplied.

Flow Variability through evaporator and condenser

The computer selection of the chiller will determine the min/max flow through vessels. This data is available in the submittal documents that came with the chiller. Flow limits must always become part of the commissioning systems manual for a chiller plant.

Daikin recommends minimizing the turn down of the flow rate on the condenser. The control signal from the chiller controller should be used as the signal to control a variable speed drive on the condenser pump. Setpoints in the controller will establish the optimal head pressure which controls the speed drive.

Service Data, which displays water velocities, can aid you in quickly determining your minimum and maximum flow rates. With water, the lower flow rate limit for the evaporator will fall between 2.3 and 3.0 fps. The upper limit on the evaporator flow is 12 fps. Condenser flow limits are between 3.0 and 10.0 fps. These limits will change if there is any glycol in the water.

Varying flow rates on the evaporator will affect the delta-T between entering and leaving chilled water, but it does not change the lift capabilities of the compressor. Your limitation will be the min and max flow rates through the evaporator.

The chiller needs turn-down capacity to work effectively in a variable primary configuration. Therefore, chillers with higher initial pressure drops (smaller tube bundles/more passes) are better suited for these types of applications. The tube velocity is directly proportional to the flow rate. Thus, if a 50% turn down is requested, the initial selection should have a tube velocity of at least 6 fps, since the minimum allowed water velocity through the evaporator is 2.3 fps. When selecting a chiller for a VPF application, be sure to select a unit that allows for adequate turn-

down. If you select a chiller that already has a low fluid velocity at 100% load, you will not have much turn-down capability. To obtain better turn-down, consider selecting a smaller vessel stack and using 3-pass configurations.

As with any chiller used in a variable primary flow (VPF) application, it is important to avoid excessively high and low fluid flow rates as this can damage the chiller and have negative effects on the system. The selection program is programmed to recognize minimum and maximum fluid velocities for the chiller and will not generate selections/ratings outside of the acceptable range.

Maximum allowable evaporator flow rate of change

The chiller is capable of varying the evaporator water flow rate up to 50% per minute. Keep in mind that, while the chiller may be able to accommodate up to a 50% change per minute in evaporator flow, depending on loop design, control of leaving chilled water temperature could be compromised.

Applications

Series-Counterflow

Depending on expected design and operating conditions, a series-counterflow arrangement for 2 WMEs may offer superior efficiency. You can use the chiller selection software to simulate the conditions of each chiller. While the two chillers may be able to operate at the necessary conditions, the controller is not designed to control a “system” of chillers in a series-counterflow configuration. Daikin recommends you engage an experienced controls contractor to ensure the chillers will function as intended.

It is important that you are aware of the following restrictions:

- a. Circuit and compressor unloading characteristics are optimized in the selection software to display maximum efficiency at each requested load point.
- b. Daikin does not purport to offer any device that would control loading and cycling of the compressors to achieve this performance.
- c. AHRI does not certify *system* performance ratings for this configuration. The individual chillers selected by the selection software are AHRI certified when rated separately.
- d. Compliance with ASHRAE 90.1 applies only to individual chiller ratings.
- e. Selection of chillers in series counterflow application should be done by factory application engineers.

Oil Fouling Heat Exchangers.

In 1967 RC Downing wrote in his book *Oil in Refrigeration Systems* that a good estimate of system performance degradation would be slightly less than 1% for each 1% increase in oil concentration. In 1984, when analyzing the effect on heat pumps, JT McMullan and JW Hughes reported that, due to heat transfer degradation, increased pressure drop, elevation of the boiling point, and reduction of the latent heat capacity, system performance can degrade by as much as 30%. Later on in 1988, McMullan reported system COPs fell by about 15% as oil concentrations increased to 6.4%. Some researchers have reported increased system performance by the use of oil scrubbers in systems using ammonia as a refrigerant.

Payvar reported in ASHRAE research project RP-751 that the presence of oil in a flooded tube bundle has a significant effect on the boiling process for plain, finned, or porous enhanced tubes. Many studies indicate increased degradation of performance at higher heat fluxes in the exchanger. Other researchers have shown some increases in heat transfer with modest increased oil concentrations for systems with R-113 and R-114 as a refrigerant. These contradictory results have been attributed to foaming at the tube surface.

Small grooves or dimples in the tubes create more surface area and better heat transfer. However, they also hold onto oil.

Figure 21: Copper tubing detail

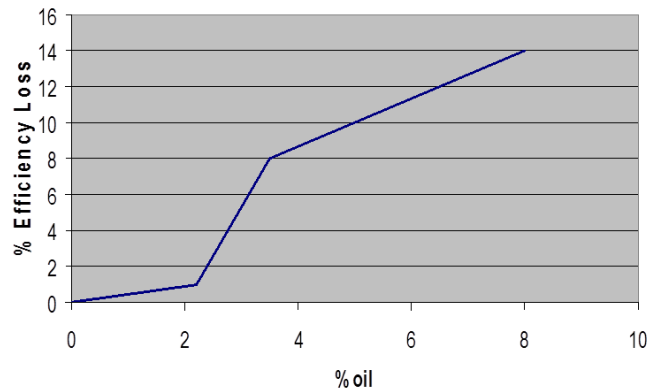


Table 4: Oil Contamination

Oil in Evaporator	Performance Loss
1 – 2%	2 – 4%
3 – 4%	5 – 8%
4 – 6%	9 – 11%
7 – 8%	13 – 15%

Source: Air Conditioning, Refrigeration News June 6, 2006

Figure 22: Efficiency Loss Due to Oil

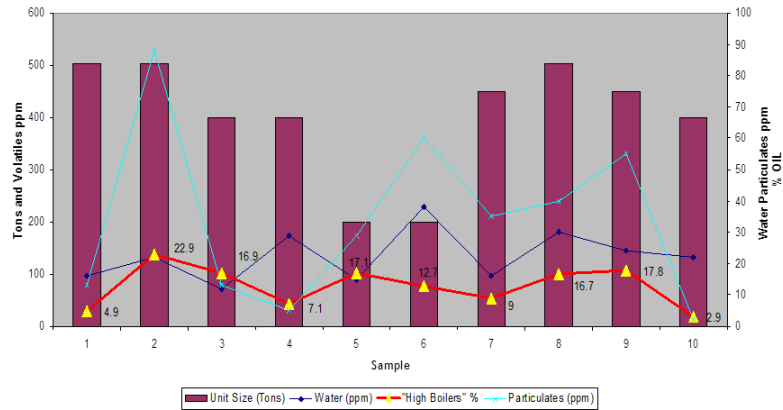


ASHRAE research project RP601 studied the refrigerant on existing chillers in the field which had low pressure refrigerant and had some maintenance done for one reason or another such as burned out motor windings. The ten machines tested ranged in age from three years to twenty seven years. They varied in size from 200 to 504 tons. The refrigerant was tested for contaminants in several categories. The results are summarized here.

Range of findings from testing the refrigerant:

- Particulates (ppm)
 - 2 to 88 ppm
 - Fresh refrigerant <1 ppm
- Water (ppm)
 - 12 to 16 ppm for refrigerants < 6 years old
 - 22 to 38 ppm for refrigerants 13<years<27
 - fresh refrigerant about 9 ppm
- Oil (%)
 - 2.9% to 22.9%
 - Fresh refrigerant <0.01%

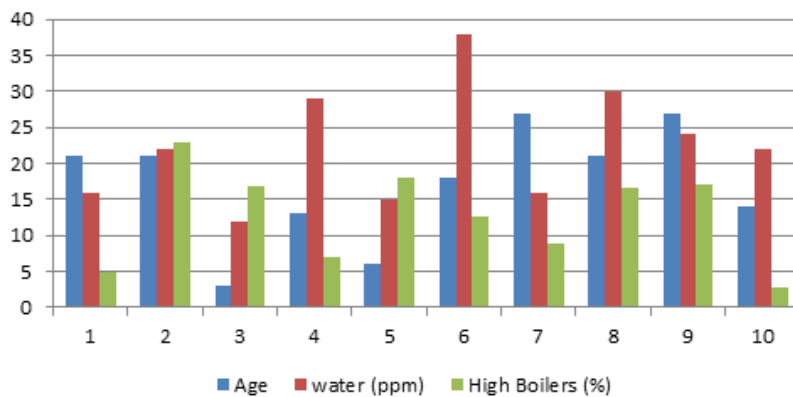
Figure 23: ASHRAE research project RP601 results



From these results shown in Figure 24, it is clear to see that there is no correlation between age and contamination nor size and contamination.

The best of these chillers, having oil contamination of only 2.9% would experience an efficiency drop of around 7% while the worst of them would be more than 15% below their new efficiency level. Since these chiller were all similar in design and construction the differences could probably only be attributed to maintenance.

Figure 24: Refrigerant contaminants RP601



AHRI Standard 550/590-2011

AHRI Certification

On-going performance verification of chiller capacity and power input plus AHRI certified computerized selection output assure the owner of specified performance in accordance with the latest version of AHRI Standard 550/590.

All chillers that fall within the scope of the certification program have an AHRI certification label at no cost to the owner. Equipment covered by the AHRI certification program include all water-cooled chilling packages rated up to 3,000 tons (10,550 kW) for 60 hertz service at AHRI standard rating conditions, with electric driven motor not exceeding 15,000 volts, and cooling water (not glycol). For 50 hertz applications, the certification program capacity range covered is 200 to 3,000 tons (703 to 10,550 kW). AHRI allows manufacturers to optionally certify 50 hertz chilling packages below 200 tons.

Published certified ratings verified through testing by AHRI include:

- Capacity, tons (kW)
- Energy efficiency, kW/ton (COP)
- Pressure drops, ft. of water (kPa)
- Integrated Part Load Value (IPLV) or Non-Standard Part Load Value (NPLV)

As part of the AHRI certification program, AHRI has the Daikin computer selection program used to select and rate chillers. The certified computer program version number and issue date for all manufacturers is listed in the AHRI Directory of Certified Applied Air-Conditioning Products available on www.ahrinet.org.

AHRI Standard 550/590-2020 for Performance Rating of Water-chilling and Heat Pump Water-heating Packages Using the Vapor Compression Cycle and associated manuals define certification and testing procedures and tolerances of all units that fall within the application rating conditions.

Leaving chilled water temperature 36°F to 78°F
 Entering condenser water temperature 55°F to 115°F

Rating outside the range of the certification program may be listed or published but must include a statement describing such. The standard rating conditions are:

Leaving chilled water temperature 44°F
 Entering chilled water temperature 54°F
 Evaporator waterside field fouling allowance. 0.0001
 Entering condenser water temperature 85°F
 Leaving condenser water temperature 94.3°F
 Condenser waterside field fouling allowance. 0.00025

IPLV/NPLV Defined

Part load performance can be presented in terms of Integrated Part Load Value (IPLV), which is based on AHRI standard rating conditions (listed above), or Non-Standard Part Load Values (NPLV), which is based on specified or job site conditions. IPLV and NPLV are based on the following equation from AHRI 550/590.

$$\begin{aligned} \text{IPLV} &= \frac{1}{\frac{0.01}{A} + \frac{0.42}{B} + \frac{0.45}{C} + \frac{0.12}{D}} \quad \text{or} \quad 0.01A + 0.42B + 0.45C + 0.12D \\ \text{NPLV} & \end{aligned}$$

Where: A = kW/ton at 100%	Where: A = COP at 100%
B = kW/ton at 75%	B = COP at 75%
C = kW/ton at 50%	C = COP at 50%
D = kW/ton at 25%	D = COP at 25%

Weighting

The percent of annual hours of operation are weighted as follows:

100% Load at 1%, 75% Load at 42%, 50% Load at 45%, 25% Load at 12%

Tolerances

The AHRI test tolerance that accounts for instrument error, per AHRI Standard 550/590-2011, for capacity (tons), power input per ton (kW/ton), and heat balance is:

$$\%Tolerance = 10.5 - (0.07 * \%FL) + \left[\frac{1500}{\frac{DT_{FL}}{FL} * \%FL} \right]$$

Where: FL = Full Load

DT_{FL} = Chilled Water Delta-T at Full Load

Summary

Centrifugal chillers are a key building block for many HVAC systems. As the market place demands more and more performance from their HVAC systems, chillers will be applied in new and innovative ways. Variable flow and chillers with VFDs are two such applications. While this manual does not go into detail on applications, it does give the engineer the basics to understand the impact such design considerations will have on a centrifugal chiller.

We have also shown that even the most efficient chiller can suffer from poor maintenance and quickly lose its startup performance. Both refrigerant and oil testing on a regular basis will assure the plant operator that their chillers are running at peak efficiency.

Further recommended reading includes Daikin's Chiller Plant Design Application Guide AG 31-003. Your local Daikin Sales Representative can also help with specific applications, selections and further training.



Daikin Applied Training and Development

Now that you have made an investment in modern, efficient Daikin equipment, its care should be a high priority. For training information on all Daikin HVAC products, please visit us at www.DaikinApplied.com and click on Training, or call 540-248-9646 and ask for the Training Department.