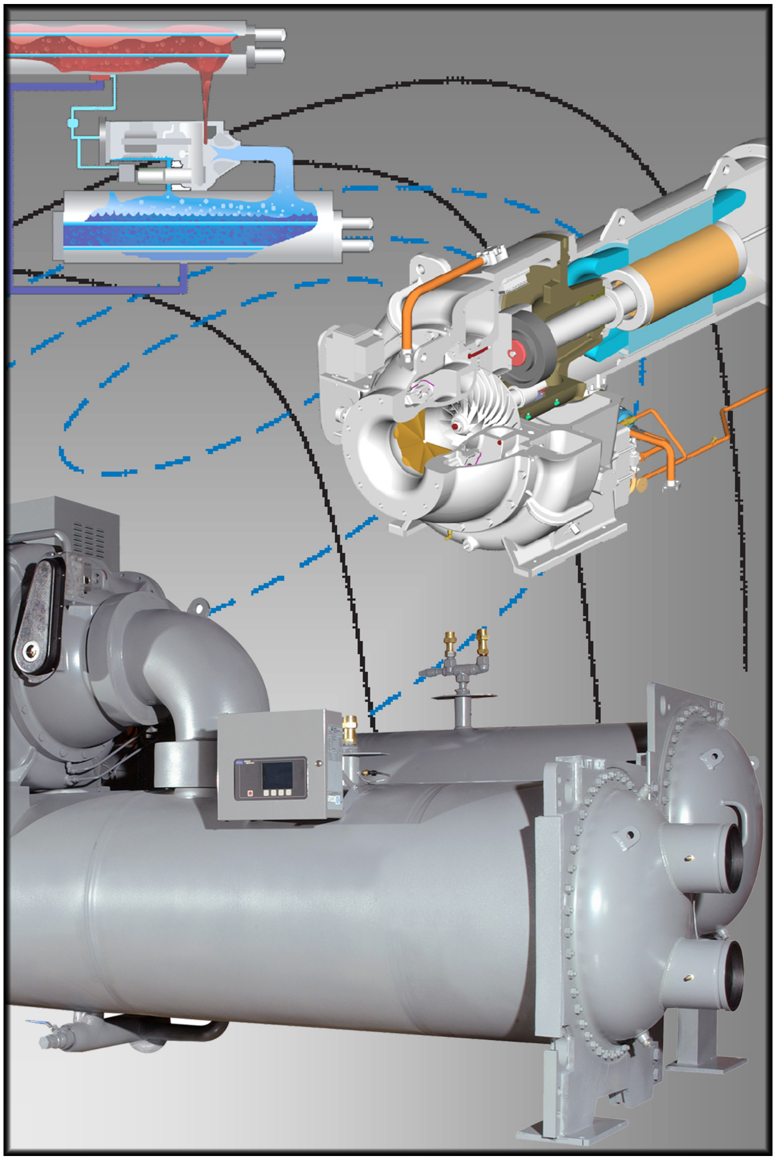


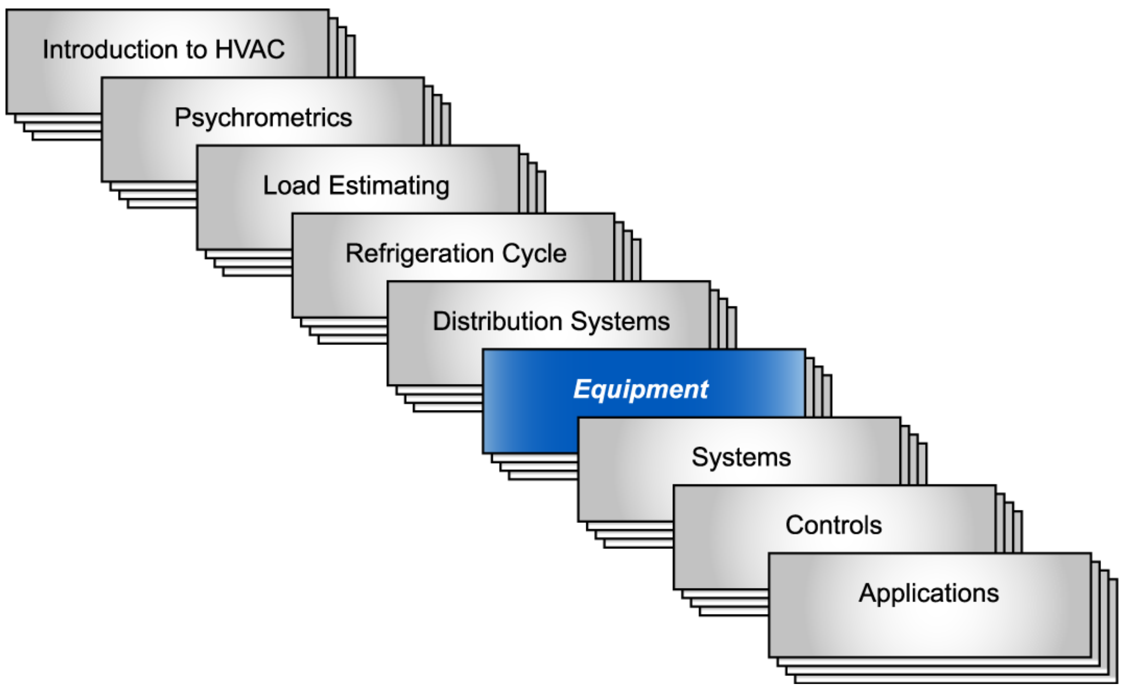
COMMERCIAL HVAC CHILLER EQUIPMENT

Water-Cooled Chillers

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



Water-cooled chillers range in size from small 20-ton capacity models that can fit in an elevator to several thousand-ton models that cool the world's largest facilities such as airports, shopping centers, skyscrapers, and other facilities. This TDP module will review all sizes of water-cooled chillers, but will contain more information on the larger chillers in the range of 200-ton and upward. Screw and centrifugal compressor water-cooled chillers tend to be the most popular designs for larger commercial applications, while scroll and reciprocating compressor chillers are used on the smaller ones. Air-cooled chillers are covered in a companion module, TDP-622.

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Table of Contents

Introduction

This TDP module on water-cooled chillers starts with a history of the first centrifugal chiller and describes the first applications for early water-cooled chillers. After a discussion of the relative merits of water-cooled chillers, the refrigeration cycle for a water-cooled centrifugal chiller is explained using pressure-enthalpy diagrams.

We will examine the major components used in water-cooled chillers such as evaporators, condensers, compressors, and metering devices. The types of chiller starters and their applications are also discussed.

A greater emphasis is placed on the larger screw and centrifugal types of water-cooled chillers in this TDP module. A more detailed discussion of the screw and centrifugal compression process and its characteristics is included. Current refrigerant issues, phase out dates, and applicable codes and standards for water-cooled chillers are also examined.

Finally, computerized selection software for a centrifugal chiller is used to demonstrate the required inputs and the selection process for a typical application.

The First Centrifugal

The art of building centrifugal air compressors was already 75 years old in 1916 when Dr. Willis H. Carrier recognized their potential use in the then infant air-conditioning industry.

By 1922, the Carrier Company had purchased a German-manufactured centrifugal air compressor and modified it for use with dielene refrigerant ($C_2H_2Cl_2$). After two years of test and development, this first centrifugal refrigeration machine was sold in 1924 to the Onondaga Pottery Company in Syracuse, New York. The machine ran for 26 years, providing air conditioning until 1950. The compressor of that first machine was then retired to the Smithsonian Institute in Washington, D.C. where it remains today on exhibit as one of the major technical developments in the United States.

Dielene

was used in the dry cleaning industry as a cleaning agent.

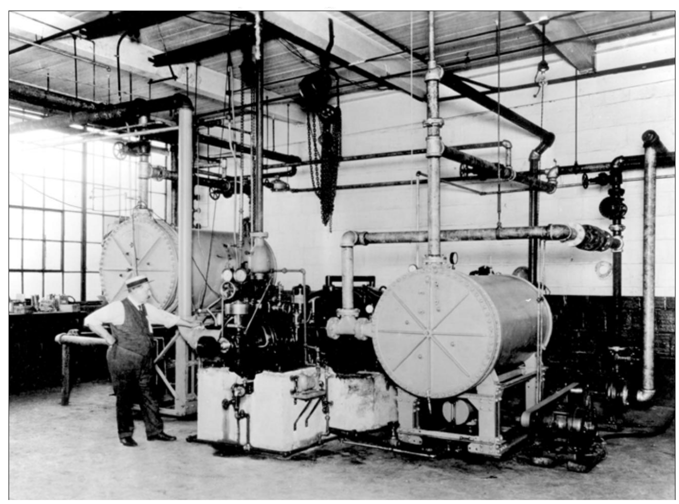


Figure 1

Early Centrifugal Chiller

Carrier's second centrifugal machine was installed in 1923 in Cambridge, Massachusetts, at the candy manufacturing plant of the W. F. Schraft and Sons Company. The second one built ended up being installed prior to the first one.

The first centrifugal machine

was tested for two years at Carrier. It was sold to the Onondaga Pottery Company in Syracuse, New York.

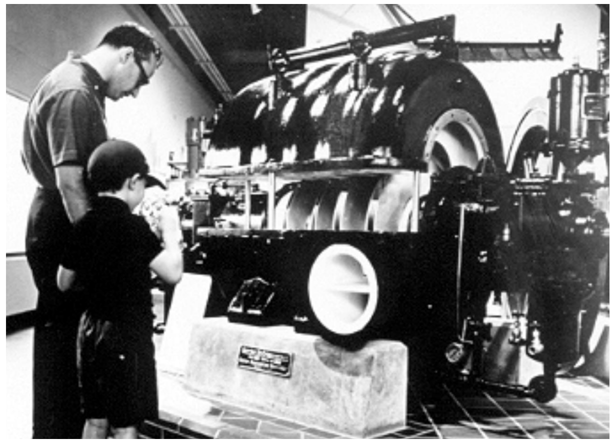


Figure 2

A Carrier chiller is on display at the Smithsonian Institute.

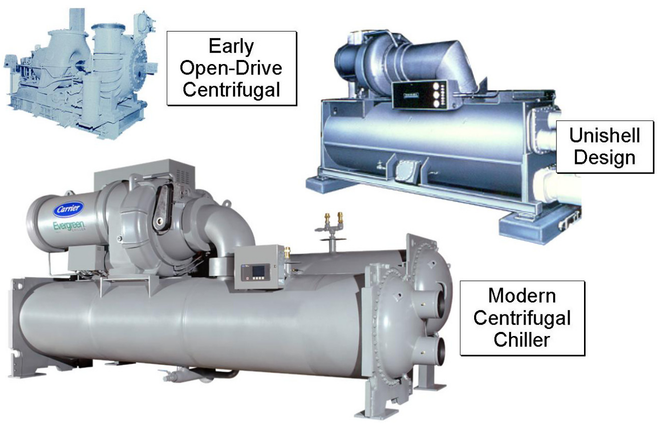


Figure 3

Evolution of Centrifugal Chillers

Because of those early efforts in the 1920s, water-cooled chillers have gained widespread acceptance in both large and medium systems. Technology has resulted in the evolution of water-cooled chillers, which are characterized by their excellent reliability, high efficiency, and compact, cost-effective construction.

Water-Cooled versus Air-Cooled Chillers

Two methods are used to condense the refrigerant in chillers. The condensers can be air-cooled or water-cooled. A typical air-cooled chiller uses propeller fans to draw ambient air over a finned coil to condense the refrigerant. It may contain multiple or single compressors. For a complete discussion on air-cooled chillers, refer to TDP-622, Air-Cooled Chillers.

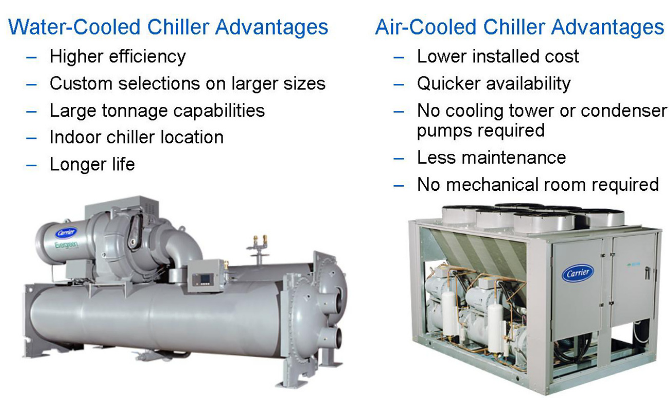


Figure 4

Air-Cooled and Water-Cooled Chiller Benefits

A typical water-cooled chiller uses recirculating condenser water from a cooling tower to condense the refrigerant. For a complete discussion on cooling towers, refer to TDP-641, Condensers and Cooling Towers.

Absorption Chillers

This TDP will cover water-cooled chillers using the vapor-compression refrigeration cycle. Absorption chillers that use water as the refrigerant will not be covered in this TDP.

Cost and efficiency are the important factors when considering air or water-cooled chillers. Chilled-water systems with air-cooled chillers typically have lower installed and maintenance costs than water-cooled because a condenser water system using a cooling tower is not required. A condenser water pump and chemical treatment for the condenser water loop adds to the maintenance required with a water-cooled system. However, water-cooled chillers have higher efficiency and therefore lower operational costs. Air-cooled chillers are chosen when it is impractical to use a cooling tower, such as when little water is available or water is highly corrosive.

The refrigerant condensing temperature in an air-cooled chiller is dependent on the ambient dry-bulb temperature. In a water-cooled chiller, refrigerant condensing temperature is dependent upon the entering condenser water temperature (and flow rate), which is a function of the ambient wet-bulb temperature. Since the wet-bulb temperature is always lower than the dry-bulb temperature, the refrigerant condensing temperature (and pressure) in a water-cooled chiller is often significantly lower than in an air-cooled chiller. This is why water-cooled chillers are more efficient.

In terms of capacity, air-cooled chillers are available in packaged sizes ranging up to approximately 500 tons, while water-cooled chillers are typically available up to 3,000 tons, with limited custom designs available up to 10,000 tons.

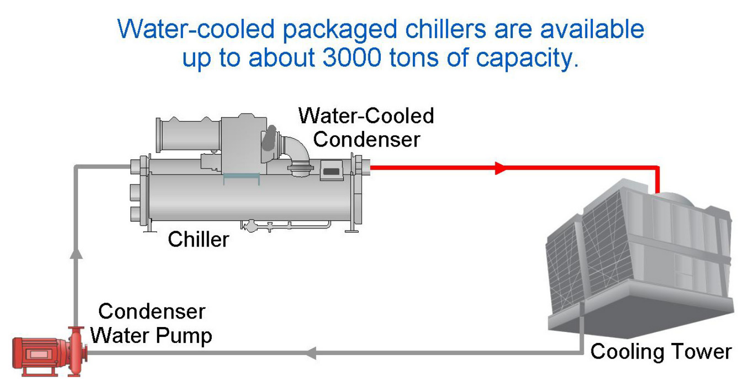


Figure 5

Typical Water-Cooled Chiller System

Water-cooled chillers typically last longer than air-cooled chillers. Air-cooled chillers may last 20 years while water-cooled chillers may last 23 years or more. This may be attributable to the fact that water-cooled chillers are installed indoors, and most air-cooled chiller configurations spend their lives outdoors in the elements. Also, some of the larger water-cooled chillers are constructed with heavy duty, industrial-grade components.

Basic Refrigeration Cycle for Water-Cooled Chillers

In this TDP, we will explain the refrigeration cycle using components from a centrifugal chiller since that type of chiller is water-cooled. The following temperatures are typical of the standard refrigeration cycle for comfort cooling applications. In the evaporator of a water-cooled chiller, liquid refrigerant at approximately 42º F takes on heat from building return water (whose entering temperature may be represented at 54º F) flowing through the evaporator, and changes to a vapor. The refrigerant vapor is drawn into the compressor and its temperature and pressure are elevated. The compressor provides the work necessary to compress the gas to a temperature and pressure required by the condenser, typically 97º F. The gas is then discharged into the condenser where it condenses on tubes through which water flows, typically at 85º F. This is the entering condenser water from the cooling tower. The condensed droplets of liquid refrigerant then fall to the bottom of the condenser, flow through a pressure reducing device such as a float valve or an orifice, and return to the bottom of the cooler where the process repeats itself.

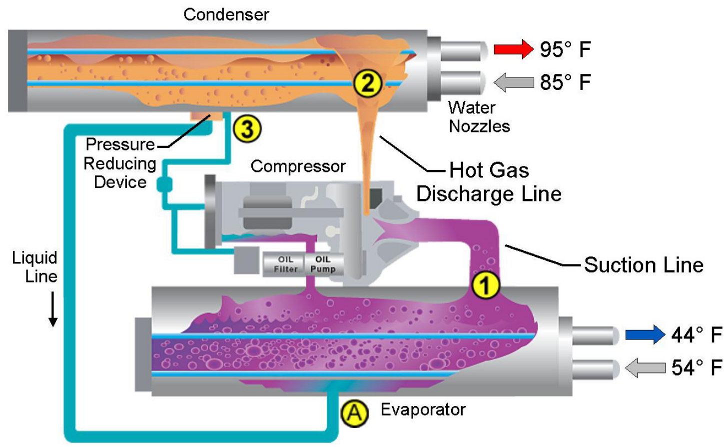


Figure 6

Components of a Centrifugal Chiller Refrigeration Cycle

The cycle can be shown on a pressure-enthalpy (p-h) diagram. Pressure is the force exerted per unit area, while enthalpy is the total heat content expressed in Btu per pound of the substance. When the compressor is close-coupled to the evaporator, there is negligible pressure loss in the suction line, and gas enters the compressor at approximately the saturated conditions that exist in the evaporator, Point 1.

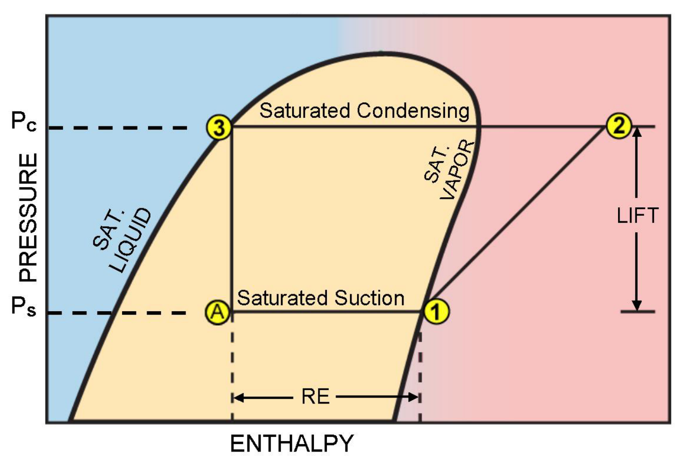


Figure 7

Pressure-Enthalpy (p-h) Diagram Showing Lift

If we follow the steps shown, you can see that from A to 1 is the refrigeration effect. In this step, building heat from the chilled water is absorbed by the refrigerant, and the refrigerant transitions from a liquid to a saturated vapor, at Step 1. From Step 1 to 2 is the compression stage. This stage raises the temperature and pressure of the saturated refrigerant vapor to the saturated condensing temperature, so that heat can be rejected to the condensing fluid. This compression is also called the “lift” of the compressor, which will be discussed later in the TDP.

In step 2 to 3, heat rejection takes place and the refrigerant transitions from superheated vapor to saturated liquid. Step 3 to A is the refrigerant passing through the expansion device to reduce pressure and temperature to the necessary conditions in the evaporator.

Subcooling Cycle

To increase the cycle efficiency, and reduce power input in a water-cooled chiller, a subcooling circuit, also called a thermal economizer, may be built into the condenser. The condensed liquid refrigerant is circuited to a special section in the bottom of the condenser called the subcooler. The subcooler cools the refrigerant below the saturated liquid temperature, Point 3. As a result of this, the refrigeration effect will be increased, as shown as A – 1, thus increasing cycle efficiency.

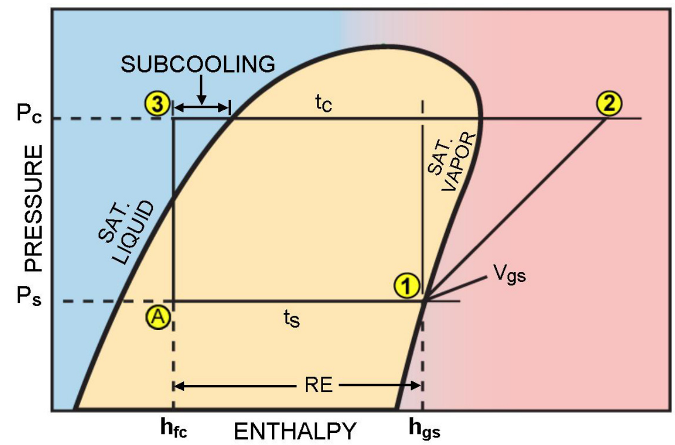


Figure 8

Pressure Enthalpy Diagram Showing Subcooling

RE = refrigeration effect
P_s = suction pressure
P_c = condensing pressure
t_s = suction temperature
t_c = condensing temperature
h_{fc} = enthalpy of liquid
h_{gs} = enthalpy of gas
v_{gs} = volume of gas at compressor inlet

In the subcooler, the refrigerant is subjected to the coldest entering condenser water. The liquid refrigerant is subcooled by 10º F to 15º F below the saturated condensing temperature. A subcooler is an economical way to increase refrigeration cycle efficiency without adding work on the compressor. It is ideal for use with positive pressure refrigerants such as HFC-134a and single-stage compressors, which we will discuss later.

The typical method of subcooling as described earlier simply cools the liquid below its saturation temperature in a tube bundle that cold condenser water circulates through. This might also be described as sensible subcooling, because no refrigerant has flashed or changed state (latent heat transfer).

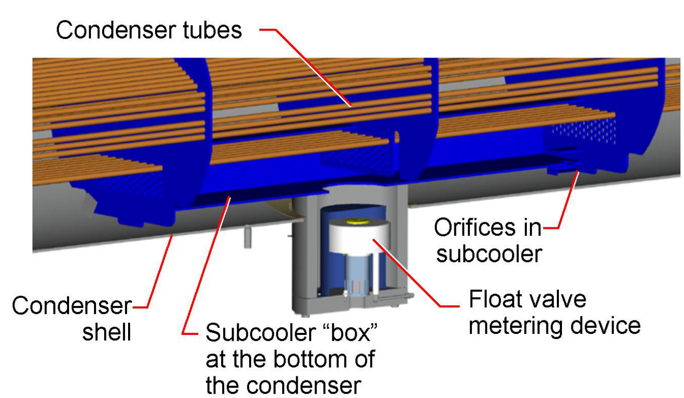


Figure 9

Condenser Subcooler

Another method is called “flash subcooling.” In the flash subcooler a portion of the refrigerant is allowed to flash back into vapor further cooling the remaining refrigerant and effectively lowering the saturated condensing temperature. The flash subcooler is not really a simple subcooler but a lower saturation temperature condenser. Thermodynamically, the net effect of a properly sized sensible subcooler can be the same as that of a flash subcooler in that the amount of heat being rejected by the refrigerant to the condenser water is the same.

The “flash subcooler” has two cost advantages over a sensible subcooler:

1. It requires less refrigerant charge, since vapor is less dense than refrigerant and most of the refrigerant side volume in a flash subcooler is vapor while it is liquid in a sensible subcooler.
2. It has a very good refrigerant-side heat transfer coefficient (two-phase flow has a better heat transfer coefficient than a pure liquid).

For example, it takes 71.22 Btu/lb to convert each pound of liquid R-134a to a gaseous form. If the liquid refrigerant did not undergo a phase change, yet underwent 5° F temperature change, the change in enthalpy is only 1.78 Btu/lb.

Economizer Cycle

Multiple-stage centrifugal and some screw chillers improve the cycle efficiency by using an economizer, also known as a flash economizer. An economizer is a separate vessel after the condenser that improves the refrigeration cycle by allowing a small amount of gas to flash into vapor after the condensing stage. This phase change decreases the temperature of the remaining liquid refrigerant. The remaining vapor is then drawn back into the compressor at some secondary stage of compression to be sent back through the cycle. Shown in Figure 10 is the increase in refrigeration effect produced by the addition of an economizer to the centrifugal cycle.

$\boxed{\text{This increase in refrigeration effect due to an economizer is shown by } h_{fc} - h_{fe}.}$

P_c = condensing pressure
P_e = pressure at economizer
P_2 = pressure at stage 2
P_s = pressure at stage 1
h_{fe} = enthalpy of liquid with economizer
h_{fc} = enthalpy of liquid without economizer
h_3 = enthalpy of mixture
h_{ge} = enthalpy of gas with economizer
h_{gs} = enthalpy of gas suction inlet

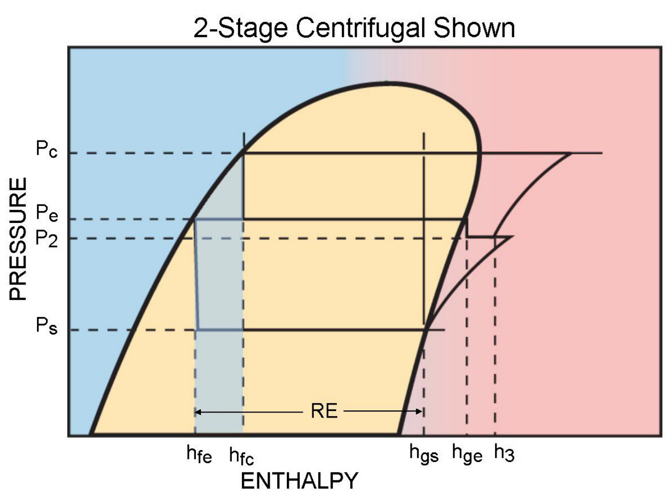


Figure 10

Refrigeration Cycle with Economizer

Water-Cooled Chiller Components

Evaporator

The evaporator, or cooler, as it is often called, is the vessel in which the water is cooled. There are several designs of evaporator, and their use is typically based on the size of the chillers.

Brazed-Plate

For smaller capacity chillers (15-60 tons), the brazed-plate evaporator is often used. These evaporators consist of a series of plates brazed together with every second plate turned 180 degrees. This design creates two highly turbulent fluid channels that flow in opposite directions that result in a high heat transfer coefficient over a small surface area. The plates are stacked so they form a multi-layered design for two independent paths of fluid to travel. Each layer or path is linked to an inlet and outlet via a manifold at either end. The design of a brazed-plate evaporator maximizes heat transfer at an attractive first cost for smaller chiller designs. The waterside of a brazed-plate evaporator must be kept free of sediment and debris. Strainers are required. The brazed-plate evaporator requires chemical cleaning since the waterside is not mechanically cleanable.

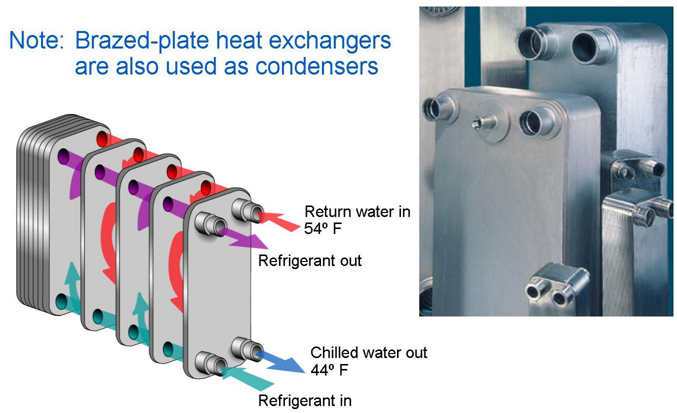


Figure 11

Brazed-Plate Evaporator
Photo Courtesy of API Heat Transfer

DX Shell-and-Tube

The direct expansion (DX) evaporator is a shell-and-tube heat exchanger design in which the refrigerant flows through the tubes and the water flows around the tubes inside the shell section. As heat is transferred to the colder refrigerant from the warmer water, the refrigerant in the tubes boils while the water is cooled. Baffles within the shell direct the water flow path over the tubes to create turbulence, resulting in improved heat transfer.

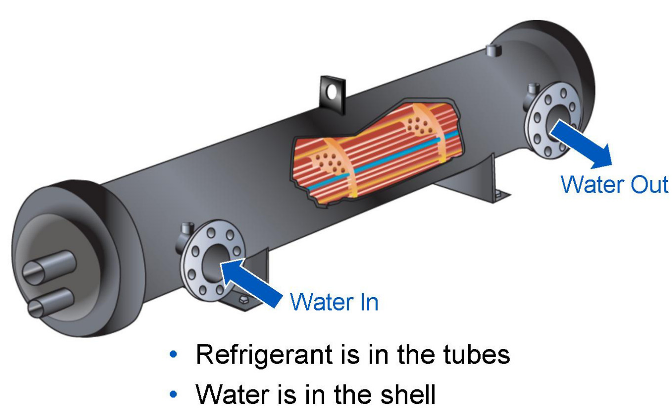


Figure 12

Direct Expansion (DX) Evaporator
Photo Courtesy of Standard Refrigeration

However, the shell of a direct expansion evaporator must be chemically cleaned. This type of heat exchanger is typically used in the 40-60 ton chiller range. Internal baffling changes are made on DX evaporators to accommodate flow variations and maintain acceptable pressure drops.

Cleaning DX Evaporators

When it is necessary to clean the water circuit on a direct expansion-type evaporator, a chemical cleaning process is required. This is because water is on the shell side and mechanical cleaning is not possible due to the baffling inside the shell.

Flooded Shell-and-Tube

In larger capacity water-cooled chillers, the cooler is normally a shell-and-tube heat exchanger with the refrigerant on the shell side and the heat transfer fluid passing through the tubes. This fluid can be fresh water, an antifreeze mixture, or any process liquid that requires cooling. The advantage to this type of heat exchanger is that the tubes can be mechanically cleaned. The fluid in a chilled water loop is chemically treated and closed to the environment to protect the equipment and piping system from corrosion and fouling.

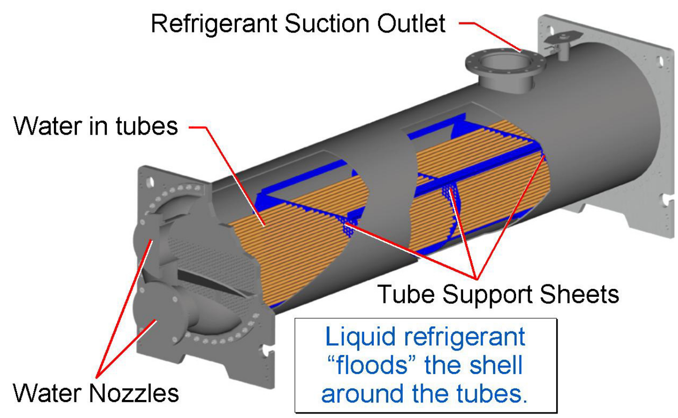


Figure 13

Flooded Shell-and-Tube Evaporator

Waterbox nozzle connections are used to connect the water distribution piping to the chiller. A typical evaporator has a suction pipe (also known as a vapor outlet) and an inlet pipe for the liquid refrigerant. The inlet pipe enters at the bottom of the cooler and empties into a distribution system, which spreads the flow of refrigerant evenly over the entire length of the cooler tube bundle. This provides optimal use of the cooler tube surface by producing an even refrigerant level throughout the cooler.

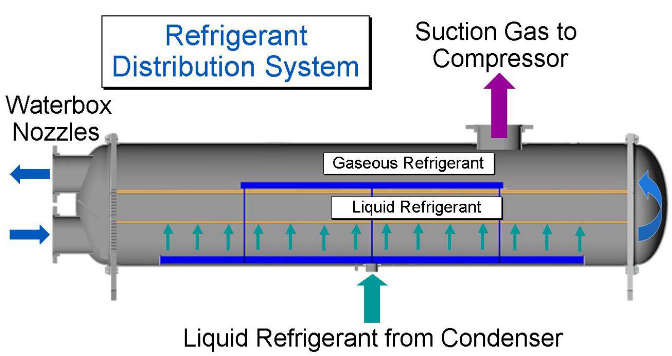


Figure 14

Evaporator Refrigerant Path

Because refrigerant goes through a change of state from liquid to gas within the shell (refrigerant) side, some means of separating the vapor from the boiling liquid must be provided. This is because liquid refrigerant droplets tend to become entrained in the flow of suction gas that is going to the compressor inlet. These liquid droplets that are "carried over" can be corrosive or destructive to the centrifugal compressor. Space elimination is the preferred design method. The cooler is designed with enough space between the top row of tubes and the compressor inlet to ensure that the velocity of the refrigerant vapor is low enough to prevent entrainment of liquid refrigerant droplets.

Other designs use eliminators to trap the liquid droplets before they reach the inlet of the compressor. Eliminators are either a series of parallel plates bent in a Z-shape, or a wire mesh. The disadvantages to eliminators are that they can eventually fail, they introduce a pressure drop, and they have an upper limit on face velocity. Eliminators work by converting mist into large drops that fall back against the oncoming vapor flow. If the velocity of the refrigerant vapor is too high, the liquid droplets that were eliminated are re-entrained and carried through the eliminator. The allowable face velocity is a function of the ratio of liquid to vapor densities.

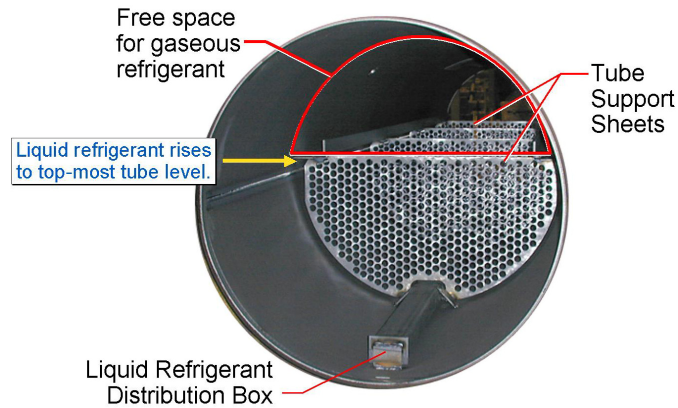


Figure 15

Space Elimination of Refrigerant Droplets

Mechanical elimination

requires approximately the same area as space elimination.

Tube support sheets at intermediary points are critical to ensure that the tubes do not deflect due to the boiling action that takes place around the tube bundle. They also aid in the replacement of tubes, should that be necessary. The end tube sheets form the ends of the heat exchanger, and tubes are sealed at the end on these tube sheets to create the refrigerant to water boundary. Welding, chemically sealing, or rolling the tubes into the grooved tube sheets can accomplish this sealing. Among these, grooved tube sheets provide the best seal, and with a double groove, the seal is even better. To increase the stability of the tubes, they can also be swaged (expanded) at the intermediate tube support sheets, a process that expands the tube just enough to fill the machined holes. This process enables the tubes to be replaced by use of a pulling tool that extracts the tube from the tube support sheet.

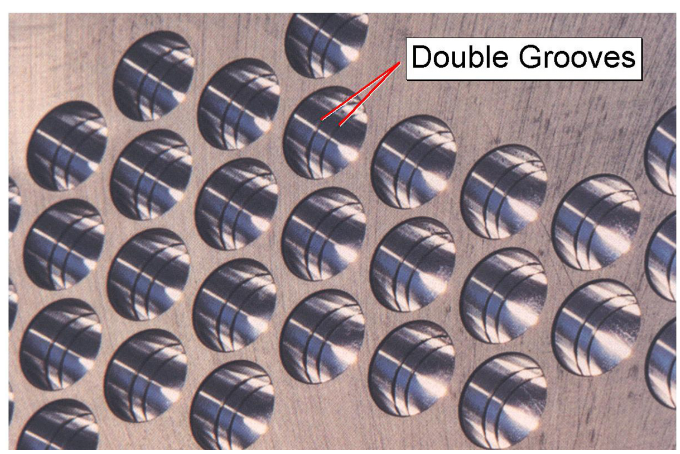


Figure 16

Double Grooved Tube Sheet for Tight Seal

Evaporator Pros and Cons

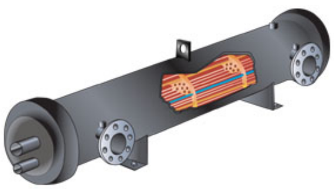
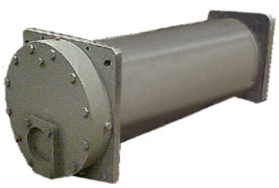
Brazed-Plate Evaporator** 	Pro	Smaller footprint
	Pro	Lightweight
	Pro	Cost effective for smaller sizes
	Con	Sensitive to contamination of the water circuit
	Con	Not mechanically cleanable
DX Shell-and-Tube Evaporator* 	Pro	Cost effective for small and medium sizes
	Pro	Familiarity with customers/engineers
	Pro	Design can accommodate heater cable on the shell to prevent freezing of water
	Con	Larger footprint
	Con	Not mechanically cleanable
Flooded Shell-and-Tube Evaporator 	Pro	High efficiency
	Pro	Mechanically cleanable
	Con	Large footprint
	Con	Design cannot accommodate heater cable on the shell to prevent freezing of water in tubes

**Photo courtesy of Standard Refrigeration*
***Photo courtesy of API Heat Transfer*

Parallel and Series Chiller Evaporators

Where chiller capacities greater than can be supplied by a single chiller are required, or where stand-by capability is desired, chillers may be installed in parallel. Units may be of the same or different sizes. Usually, equally sized chillers are utilized to accomplish commonality of parts and maintain simplicity. If unequal sized chillers are used, cooler flow rates must be balanced to ensure proper flow to each chiller. Software is available from the chiller manufacturer that automatically stages multiple chillers of equal or unequal size.

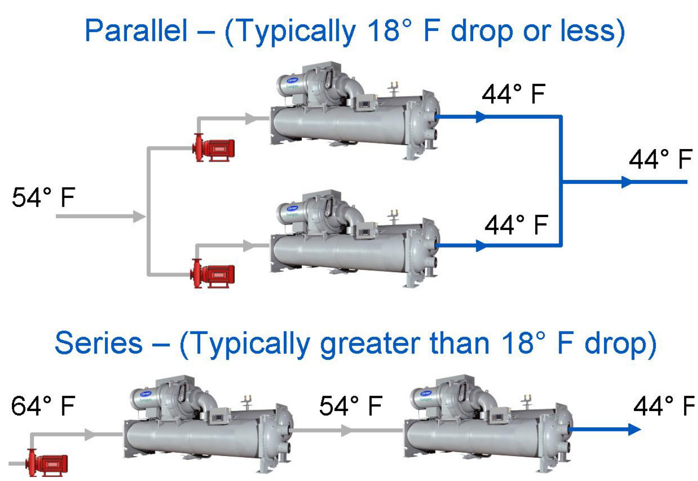


Figure 17

Parallel and Series Chillers

Where a large temperature drop (greater than about 18º F) is desired, chillers may have to be installed in series. The 18º F value is a rule of thumb. Actual chiller selections should be made to determine when a series chiller configuration is necessary. The evaporator minimum entering fluid temperature limitations should be considered for the downstream chiller. Use of a reduced pass configuration may be required to keep waterside pressure drop at an acceptable level. This is covered in detail later in the TDP. Additional piping configurations can be found in TDP-705, Chilled Water Systems.

Future Expansion Arrangements

Parallel chiller arrangement lends itself to future expansion because another chiller can simply be added to the "ladder." Two chillers in a series cannot accommodate future expansion since pressure drop in the evaporator circuit would become too great with three chillers.

Condenser

Water-cooled chillers either use brazed-plate or shell-and-tube heat exchangers for the condenser. Since the focus of this TDP is on larger chillers, the shell-and-tube design will be discussed further. As in the evaporator, the refrigerant is on the shell side, and the fluid is on the tube side. The fluid is generally fresh water, used with an open cooling tower.

Other condenser water sources include lakes, rivers, or cooling ponds. Use of these water sources constitutes a once-thru system. Once-thru condenser applications are very limited compared to recirculating cooling tower applications. That is because they are often considered as a source of thermal pollution. In coastal areas, seawater may also be used; however, appropriate materials, like titanium, should be used to minimize tube and tube sheet corrosion.

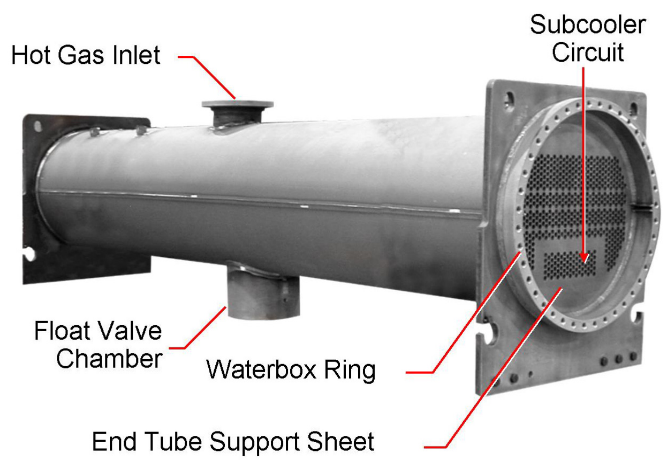


Figure 18

Large Chiller Shell-and-Tube Condenser

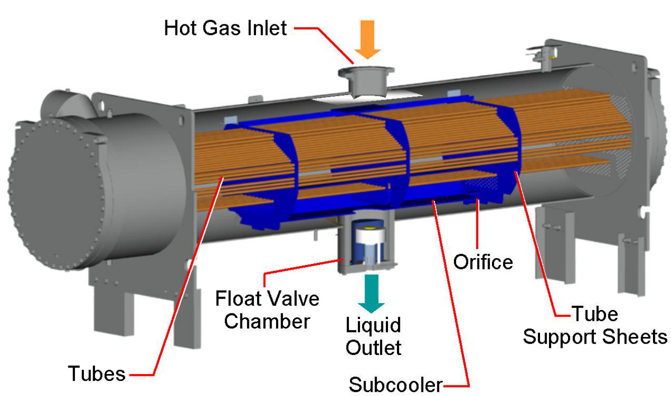


Figure 19

Large Chiller Condenser Cutaway

See the tubing discussion later in this TDP.

Besides the waterbox connections, a typical condenser has a refrigerant hot gas inlet pipe and a liquid outlet pipe. The inlet pipe enters the top of the condenser where hot discharge gas from the compressor impacts a refrigerant distribution baffle.

The baffle redirects the discharge gas along the length of the condenser, lowering its velocity and preventing direct impact of the high velocity discharge flow on the tubes. This prevents tube failure due to vibration. The same tube sealing and support methods as in the evaporator discussed previously are used in the condenser.

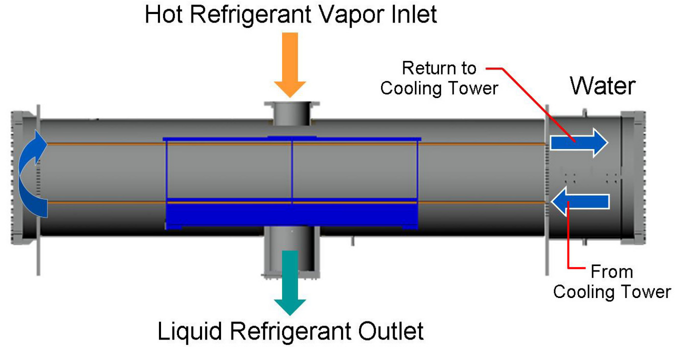


Figure 20

Refrigerant Path in a Shell-and-Tube Condenser

Compressors

There are several widely used compressors for water-cooled chillers. They are grouped into two categories: positive displacement or dynamic compression (also called non-positive displacement). Centrifugal compressors are the only type of non-positive displacement compressor.

Reciprocating

Reciprocating compressors, like a reciprocating engine, have pistons, rods, discharge, and intake valves. The valves operate on suction and discharge pressure. Compression is achieved by trapping a fixed amount of refrigerant gas into a chamber. For this reason, reciprocating compressors are positive displacement type compressors as are scrolls and screws.

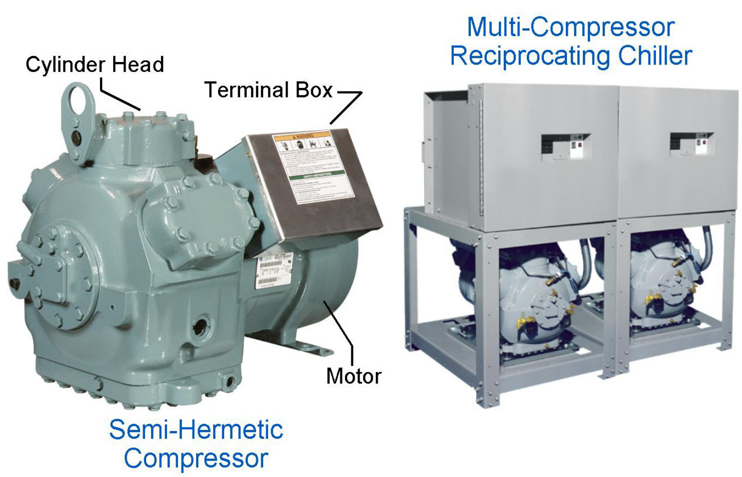


Figure 21

Reciprocating Compressor

Scroll

Scroll compressors are a popular alternative to reciprocating compressors. Multiple scroll compressors are often used in a single chiller design to meet larger capacities. Scroll compressors are 10-15 percent more efficient than reciprocating and are proven very reliable because they have approximately 60 percent less moving parts.

Scroll compressors have a unique compression process. A fixed scroll coupled with the movement of an orbiting scroll ingests the suction gas into several pockets. As the orbiting scroll moves, the pockets of gas are compressed to an intermediate pressure. In a final orbit, the pockets reach discharge pressure and exit through the discharge port.

Reciprocating and scroll compressors are currently used in the 10 to 400-ton chiller range, and are used in both water-cooled and air-cooled chillers.

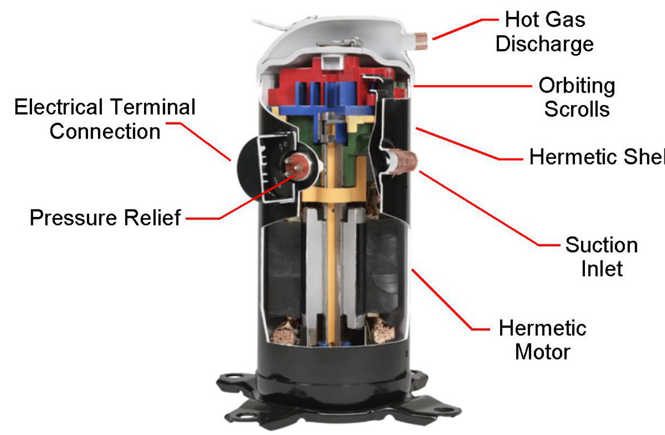


Figure 22

Scroll Compressor

Screw

Screw compressors are another version of positive displacement compression. As the screw rotors turn, the gas is compressed. Screw compressors have one or more rotors to accomplish the compression. Due to this characteristic, positive displacement compressors are best suited to handle smaller volumes of refrigerant gas over high compression ratios. The HVAC industry is trending to phase out reciprocating compressors, favoring screw and scroll compressors. Screw compressors are used in the 70 to 500-ton chiller range in multiples, and are also used in both water-cooled and air-cooled chillers. For an in-depth discussion of screw compressor operation, see the Screw Compressor Operational Details section (page 26).

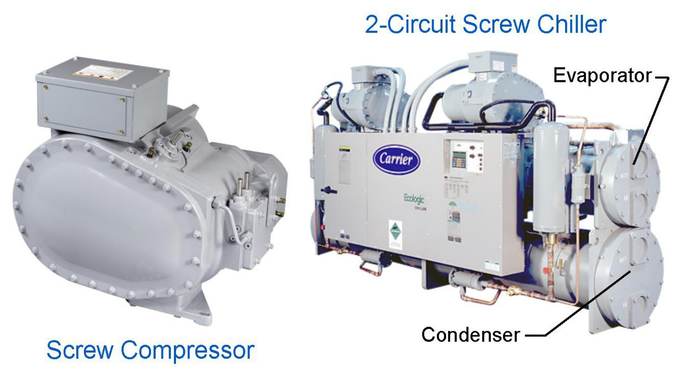


Figure 23

Screw Compressor

Centrifugal

Centrifugal compressors are dynamic compression, and they are water-cooled because their efficiency is optimized at the lower condensing temperatures and pressures. As refrigerant molecules are flung outward by centrifugal force, new ones are drawn into the compressor to replace them. The overall effect is one of continuously compressing the stream of refrigerant gas. This process of compression allows large volumes of refrigerant gas to be compressed resulting in a relatively compact size chiller. Thus, centrifugal compressors are dominant in larger capacities.

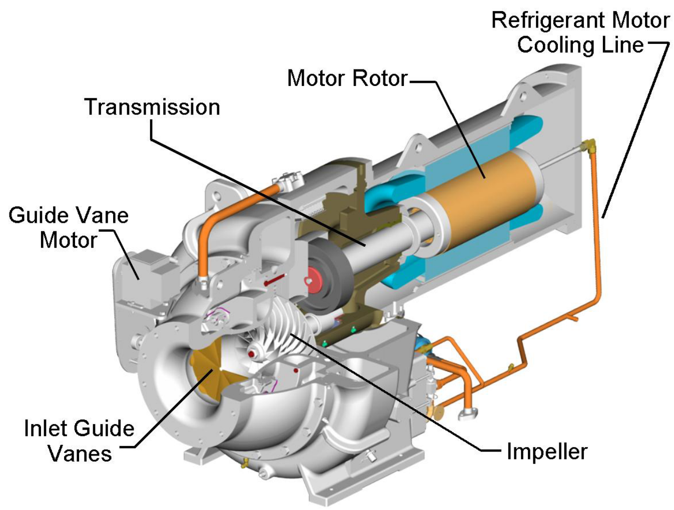


Figure 24

Centrifugal Compressor

> ***Note:***
>
> ***The break point from positive displacement to centrifugal type compression in air-conditioning chillers generally occurs between 300 to 500 tons due to design and capacity related reasons.***

As an alternative for larger capacity requirements, multiple chillers with positive displacement compressors can be used to match the capacity of a single centrifugal chiller.

There are several possible drivers for refrigeration compressors. These include motors, steam turbines, gas engines, and gas turbines. As electric motors are the most commonly used method for driving a compressor, we will focus on these.

Centrifugal compressors fall into two broad categories: hermetic and open. The compressor driven by these motors may be direct-drive, where the motor rotor and the impeller are on the same shaft, or gear driven, where the rotor shaft and impeller shaft are coupled by a transmission. In a direct-drive compressor, the impeller spins at the same speed as the motor (slightly below 3600 rpm for 60 Hz). In a gear-driven compressor, the gearing in the transmission is changed by the manufacturer to meet different power frequency requirements or compression requirements.

Figure 25 shows a cut away view of a hermetic transmission-drive centrifugal compressor. The motor is attached to the transmission and is hermetically sealed from the machine's ambient environment. Motor cooling is accomplished by spraying liquid refrigerant on the motor windings. Hermetic motors reject their heat directly into the refrigerant that is cooling them, which is then rejected by the condenser. The highly efficient motor cooling results in the use of smaller motors than could be utilized with an open-drive motor of the same type. Thus, hermetic motors are extremely reliable, run cooler, and are lighter than comparable open drive, air-cooled motors. They also are protected from environmental contamination (dirt). Hermetic motors are squirrel cage induction type, which are well suited for sealed environments.

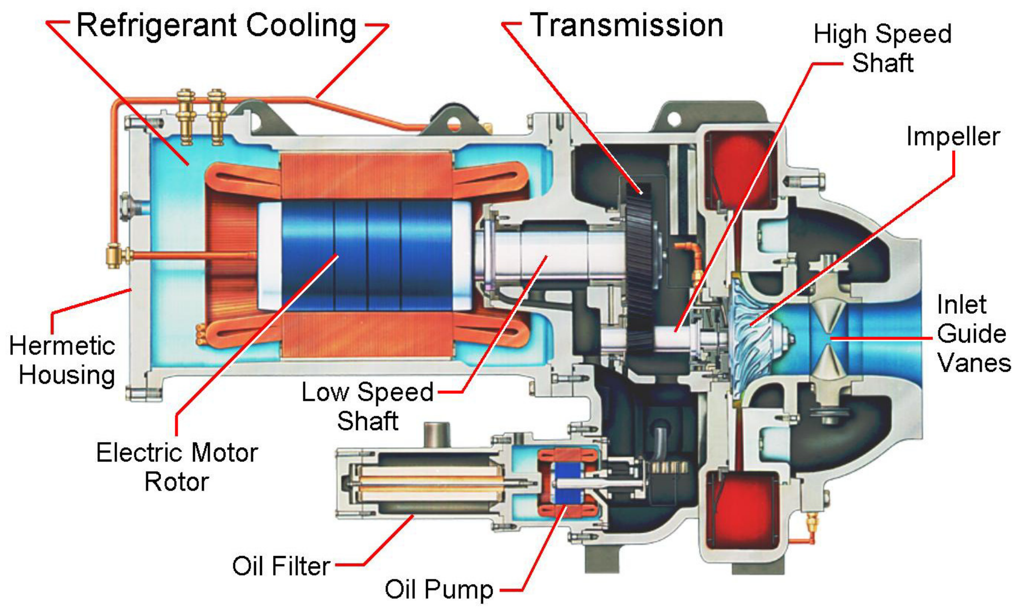


Figure 25

Hermetic Centrifugal Compressor – Cutaway

Open-drive compressors have the compressor shaft extending through the casing to facilitate connection to a number of different types of drives as discussed earlier. However, these machines require shaft seals, which must be replaced on a regular basis. By design, shaft seals leak oil and refrigerant to keep contaminants out of the chiller. This oil and refrigerant must be reclaimed, typically in a separate container outside the chiller. Additionally, when routine preventive maintenance is done, the compressor and motor must be realigned. An open drive motor also rejects its heat directly into the

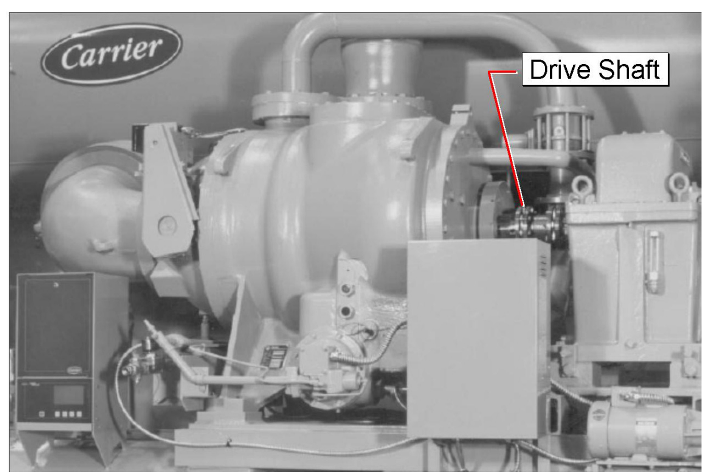


Figure 26

Open-Drive Centrifugal Compressor

mechanical room, requiring additional cooling or ventilation to reject the heat from the building. This also subjects the motor to the hot, and typically dirty, atmosphere in a mechanical room.

Shaft seals

in open-drive compressors pass oil to lubricate themselves. The seal chamber area is flooded with oil and the oil holds in the refrigerant. However, a small amount gets by the seal.

Refrigerant Metering – Expansion Device

To maintain the necessary pressure difference between the evaporator and condenser, some form of metering device must be used. There are several types used on the larger water-cooled chillers such as screws and centrifugals. They are mainly fixed or variable orifices and expansion or float valves. Fixed orifices are simply perforated plates that cause a pressure drop in the fluid. These orifices are designed for the maximum flow of refrigerant, which hinders their performance at lower flows/loads. Fixed orifices are used in some chiller designs. Variable geometry orifices can respond better to the changes in loads by changing area, and some require external inputs to make this change. Expansion valves, for instance, use saturated suction temperature to regulate the amount of refrigerant that passes through them, and are typically controlled electronically. Float valves are used on larger screw and centrifugal chillers and can automatically vary the flow of refrigerant in a system by responding to changes in liquid refrigerant levels. These level changes are the result of changes in load conditions and pressure differences in the heat exchangers. Float valves are ideal on larger water-cooled chillers because they are simple, always optimize the refrigerant flow, and do not require any external inputs to vary the flow rates of the refrigerant.

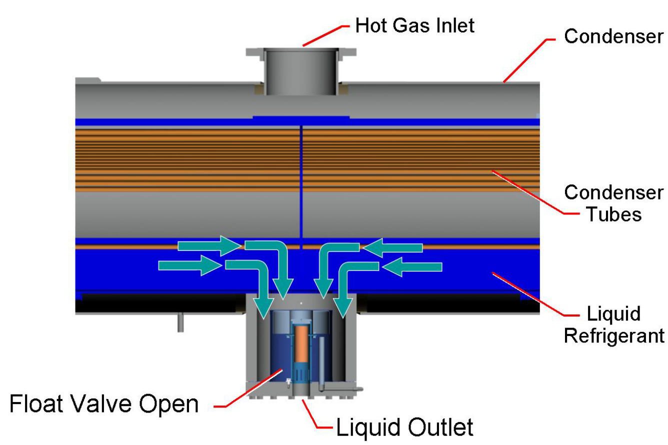


Figure 27

Refrigerant Metering – Large Chiller

Another benefit of the variable geometry orifice (float valve) on larger screw and centrifugal chillers is that it can pass increased mass flow of refrigerant during periods of lower entering condenser water temperature by increasing the area of the opening into the evaporator. More refrigerant flow will result in an approximate 15 percent increase in chiller capacity above the design full load tons. This increase in capacity is called "maximum capacity." Fixed orifice designs cannot accomplish this feature.

Why might this be important? On multiple chiller projects, each chiller that can accomplish maximum capacity will handle a larger portion of the total building load than a fixed orifice design chiller. This means the second (or third) chiller in a multi-chiller plant can remain inactive as long as possible saving energy over designs that must run more than one chiller.

For a discussion on refrigerant metering devices used on the smaller chillers with reciprocating and scroll compressors (and some smaller screw compressors), refer to TDP-622, Air-Cooled Chillers, and TDP-403, Expansion Devices & Refrigeration Specialties.

Waterboxes

Waterboxes are used to connect the chiller waterside to the chilled or condenser water loop piping. There are two types of waterbox construction in common use on larger screw and centrifugal chillers: "nozzle-in-head" and "marine" type.

The most popular and economic design is the "nozzle-in-head" type. This design is a result of the industry trend in air conditioning to make machines more compact and less costly. The only drawback to nozzle in head type waterboxes is that the piping connection must be undone, and the "heads" must be completely removed to perform any tube maintenance.

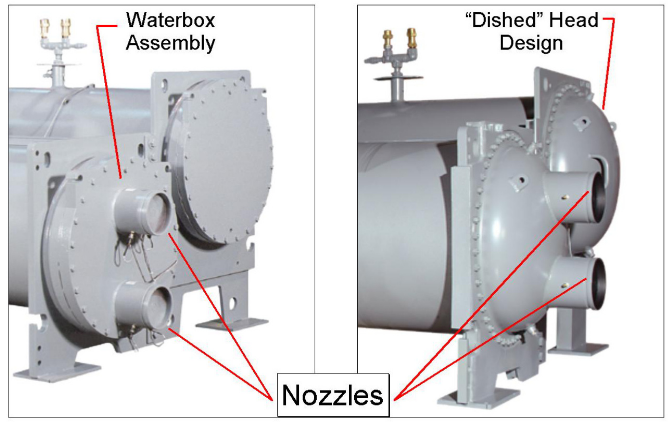


Figure 28

Nozzle-in-Head Waterboxes

The "marine" type waterbox is larger and more costly. However, it provides the ability to clean the tubes without disturbing the piping external to the machine. For this reason, the marine waterboxes are generally used in chillers above 700 tons. Moving the piping is time consuming and expensive, so making use of a marine type waterbox offers owners an attractive feature and reduced maintenance costs. When available, marine waterboxes with hinged waterbox covers eliminate rigging of heavy waterbox covers.

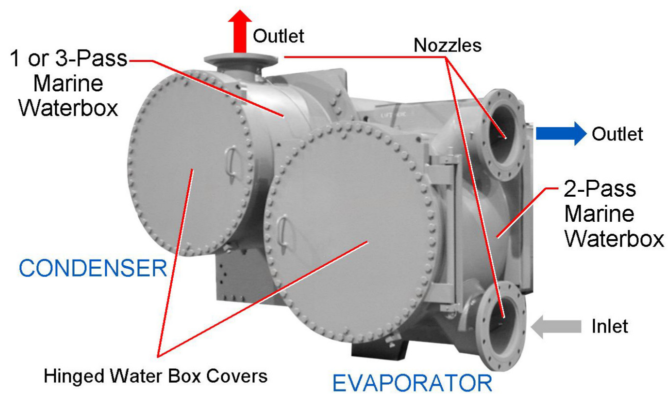


Figure 29

Marine Waterboxes

Waterboxes are typically designed for 150 or 300 psig, the latter used for taller buildings, where there may be a large water column in the water distribution piping system.

In a high-rise building, the static head imposes pressure on the chiller waterside components if the chiller is located in the basement. Example: A 50-story building with 12 ft between floors would be 600 ft high. Since 2.31 ft equals 1 psi, the pressure on the components at the lowest level would be 600 ft/2.31 ft/psi = 260 psi. The hydronic system components in the basement MER (mechanical equipment room) must be designed for this pressure. In this example, the chiller "waterboxes" (area where piping connections are made to the chiller) would have to be constructed to accommodate 260 psi. That means optional 300 psi waterboxes would have to be specified.

Purge

Some refrigerants used in centrifugal water-cooled chillers, such as HFC-134a, are positive pressure (operate above atmospheric pressure), and some are negative pressure, such as HCFC-123, operating below atmospheric pressure. Negative pressure centrifugal chillers operate in a vacuum and have gaskets and o-rings that can leak which allows air to enter the chiller. When air leaks in, some water vapor comes with it. This air/water vapor can displace some of the refrigerant in the chiller causing a reduction in efficiency.

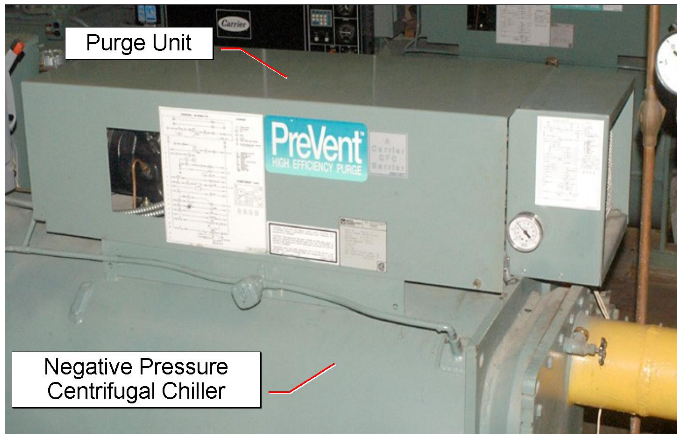


Figure 30

Purge on Negative Pressure Centrifugal Chiller

A device called a purge is required to collect and expel the air and the water vapor from the inside of the chiller. When the air is expelled, some refrigerant goes with it. A modern high efficiency purge reduces the ongoing loss of refrigerant to 0.5 lb for every pound of air removed. Older purges lost from 3 to 5 lbs of refrigerant for every pound of air removed. Air and non-condensables in the chiller causes the head pressure to rise. As a rule of thumb, for every 2.5 psig increase in condenser (head) pressure, the chiller's efficiency falls between 8-10 percent

The purge is continually sampling gas from the chiller condenser and condensing the refrigerant and recirculating it back to the evaporator. Air and other non-condensables cannot be condensed. The purge will collect these non-condensables and eventually discharge them to the atmosphere. Purging to the atmosphere is in compliance with EPA regulations. Water vapor from humid mechanical rooms is also drawn into the chiller when there are leak areas in the chiller and must be condensed and collected in the purge then drained from the purge collection chamber.

> ***Note:***
>
> ***The purge unit adds maintenance and electrical consumption that is not required with a positive pressure chiller design.***

Water mixed with refrigerant can also produce corrosive acids that break down the oil and attack the internal metal parts of the machine.

Storage Tank and Transfer (Pumpout) Unit

In performing certain service procedures on larger screw and centrifugal chillers, it may be necessary to remove or relocate the operating refrigerant charge from the evaporator or condenser. Extreme care must be exercised to prevent releasing the refrigerant to the atmosphere during this procedure.

> ***Note:***
>
> ***A typical example of when the operating charge must be removed from the condenser is when some of the condenser tubes are being replaced.***

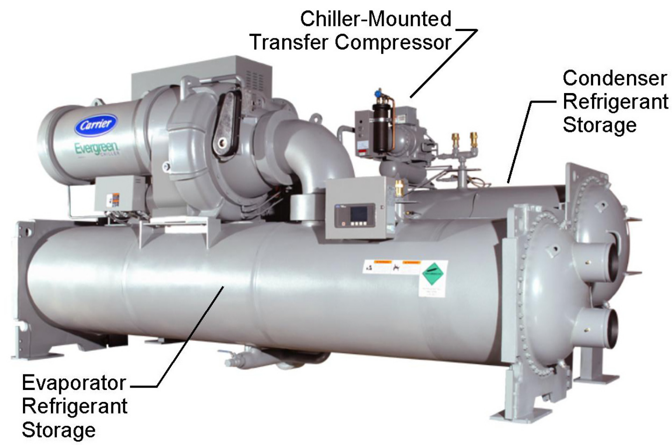


Figure 31

Positive Pressure Chiller/Transfer Compressor

In large positive pressure chillers, such as those utilizing HFC-134a, the construction of the evaporator and the condenser allows either vessel to serve as a refrigerant storage tank. Isolation valves are installed on the chiller to isolate the refrigerant charge in either the evaporator or the condenser.

This means that the operating charge does not have to be completely removed from the chiller to perform service work. The entire chiller operating charge can be stored in the evaporator when working in the condenser, or in the condenser when working in the evaporator. Transfer of charge from one vessel to another is accomplished with a portable or unit-mounted transfer compressor and isolation valve accessory.

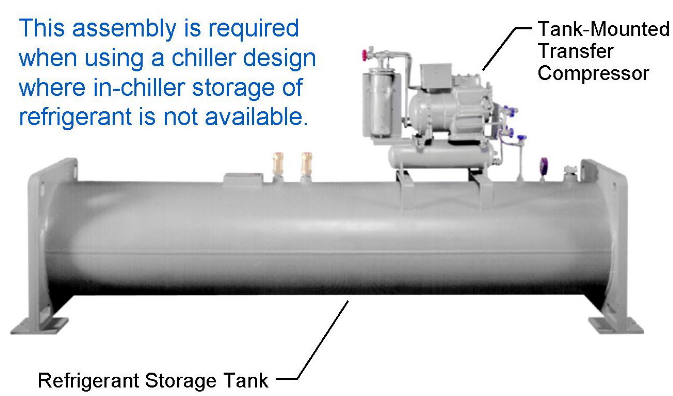


Figure 32

Storage Tank and Transfer Unit

In contrast, negative pressure chillers require a separate refrigerant storage tank and transfer compressor as well as complete removal of the refrigerant from the chiller. The reason is a negative pressure chiller, by virtue of its shell construction, is not rated for in-chiller storage of refrigerant.

If the refrigerant must be completely removed from the chiller, the chiller has to be leak-tested and dehydrated before the refrigerant can be pumped back into the chiller. This increases both the downtime required for maintenance and the potential for refrigerant loss.

Relief Valves

Refrigerant relief valves on either the condenser, the evaporator, or both, are provided as safety devices in the event of overpressurization. Overpressurization might occur during a fire or if hot water were run through the chiller tubes.

The standard pressure relief device on a negative pressure centrifugal chiller is a carbon disk with a membrane that shatters at 15 psig. Overpressurization results in total loss of the refrigerant. Back-up relief valves are available to minimize this loss in the event of overpressurization. These back-up devices contain non-fragmenting disks with a reseating plunger that will relieve the pressure and then reseat. This saves a good portion of the refrigerant that might otherwise be lost.

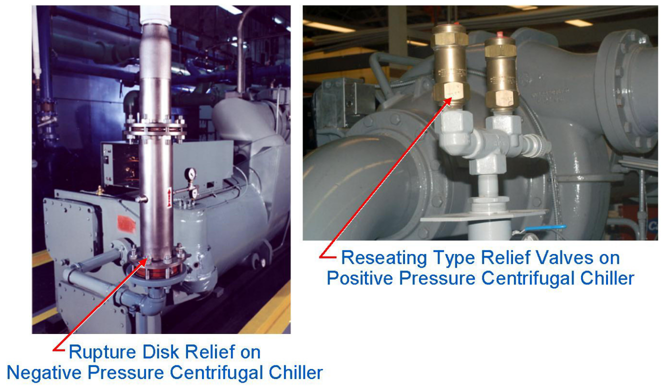


Figure 33

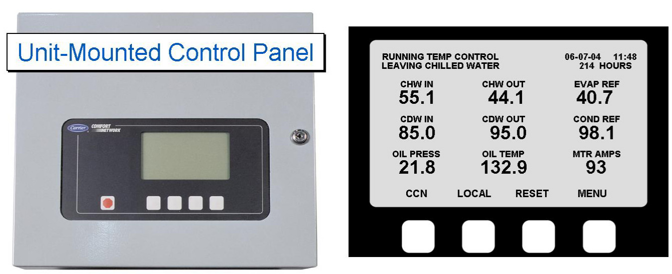
Relief Valves

Positive pressure chiller designs typically use the reseatable valve design as standard. After a release, the valve reseats, thus preventing a total loss of refrigerant.

Chiller Controls

All chillers need to be properly controlled for safe and efficient operation. Chiller designs commonly use microprocessors, electronic sensors, and digital displays that have changed the look of chiller equipment control systems. This technology is referred to a Direct Digital Control (DDC).

DDC systems allow for the use of digital display systems that give the owner-operator and service technician the ability to do a diagnostic evaluation of the operation of the equipment by way of equipment-mounted or handheld user interfaces. In DDC systems, information is digital and can be shared by other equipment to function as a system. Most DDC systems will have a communication terminal that allows for network communications or the attachment of a PC for more detailed diagnostic capabilities. DDC systems are capable of sharing information on a communication bus that can be viewed worldwide.



- Includes visual display/user interface
- Monitors and controls the chiller and auxiliary devices such as pumps
- Provides BAS communication functions

Figure 34

Water-Cooled Chiller Control Panel

Compressor Starting Methods

There are seven types of starters commonly used with water-cooled chillers. Besides cost, the selection of a starter depends on many variables, most of which deal with the electrical system characteristics, voltage, and power company regulations at the installation.

When large horsepower motors on large capacity chillers (approximately 3000 tons and larger) are connected to a power distribution system, the motor inrush current can cause voltage drops that may adversely affect the operation of other equipment (particularly computers) connected to the system. Careful consideration should be given in the selection of the starter type for voltage sensitive applications. Smaller "packaged" chillers typically include an integrated starter as a standard part of the chiller package.

> ***Starters***
>
> ***are also grouped by voltage ranges. Low voltage is any system that utilizes 600 volts and below, medium voltage is considered to be higher than 600 volts.***

For larger chillers, starters may be unit-mounted or remote-mounted from the chiller. Unit mounting saves space and reduces installation costs, and can increase the reliability of the chiller system. Unit-mounted starters are very popular on centrifugal chillers because the entire chiller's electrical requirements can be supplied with power through the starter, thus accomplishing "single point" connection. If you do not use a unit-mounted starter, several electrical connections are required. A separate electrical feed for the compressor, the oil pump, and the unit controls would be necessary. These separate wiring connections must be field-installed between the remote starter and the chiller.

Across-the-Line

Across-the-line, also called full voltage, is the simplest and least expensive method. The motor locked rotor current is drawn directly from the line on starting, and is used when the power system can withstand high inrush currents such as in a large industrial plant. With an across-the-line starter, the inrush can be 6 to 8 times the motor rated full-load amp (FLA) value. Therefore, this method is especially suited to the following motors:

1) low-voltage, single phase motors up to approximately 10 hp
2) low voltage, three phase motors up to approximately 25 hp
3) medium-voltage (over 600 volts) motors of any hp

> ***Across-the-line starters***
>
> ***on large, single compressor chillers are mostly used in the voltages (like 4160v) where the amp draw is low. On smaller multi compressor chillers, across-the-line starters can be used for low voltage applications because the inrush per compressor is manageable.***

Because full voltage is applied directly to the motor terminals, this method also provides the quickest acceleration to running speed. This type of starter is a common one used on smaller tonnage chillers with multiple compressors. This is because the starting amps of the individual compressors are relatively low, and the compressors are staggered on startup.

Auto Transformer

This method reduces inrush line current and motor starting torque by limiting the voltage applied to the motor windings during start-up. It is used typically for medium voltage applications. The torque is accomplished through a transformer within the starter with contacts configured to allow stepped acceleration to full current. Changing incoming line voltage taps within the transformer varies the current drawn from the line, the voltage and current applied to the motor, and the torque developed by the motor at startup. This tap is usually selected and then fixed at installation, and is based on the application. For centrifugal chiller applications, the 65 percent tap is the one that provides the required torque to start the motor compressor.

Auto transformer starters

offer a soft start (relative to across-the-line) but the start is limited to discreet steps.

Primary Reactor

Like an auto transformer, this starter reduces inrush current by limiting the voltage applied to the motor. A reactor, or resistor, is used to reduce the voltage to the compressor motor instead of a transformer. Compared to an auto transformer, a primary reactor is the more economical method for medium voltage. Primary reactor is not the most efficient starter type from an electrical standpoint. A primary reactor starter uses the resistor in the circuit during starting. As the resistor lowers that amp draw, there is some wasted energy. However, primary reactor starters provide a smooth acceleration. This is because the starter adds resistance to reduce the voltages and current at startup. The starter's incoming line voltage taps may be changed at installation to set the level of inrush voltage and current.

Part-Winding

Part-winding starters also produce a soft start. An electrical engineer typically determines if soft starting is required based on an electrical system study that calculates voltage drop in the system when a load is started. If the power supply is weak or the system is heavily loaded, the system is more likely to be sensitive to motor starting, and soft starters are more likely to be required. It is more likely that 208V systems will require soft starters than 460V systems because of the higher amp draw.

Part-winding starters are a cost effective means of providing a reduced current start where smaller horsepower motors are used on water-cooled chiller designs using multiple compressors.

Part-winding starters can only be used with part-winding motors that have two sets of identical windings that are intended to be operated in parallel. During a part-winding start, only one winding is energized, reducing the inrush current to 60-70 percent of normal starting values with both windings energized. With only one winding in the circuit, a typical part-winding motor will not accelerate to more than half of the motor rated speed. Because the motor is operated with one winding only during the initial acceleration period, after the transition from start to run mode, the current draw of the motor may be close to the rated locked rotor amps. A part-winding starter essentially provides a two-step start. Part-winding starters are not used on centrifugal chiller compressors.

A part-winding starter

is commonly used on reciprocating compressors. Availability varies from manufacturer to manufacturer based on the size of the compressor motor used.

Wye-Delta

Wye-delta, also called Star-delta, starters reduce inrush line current and motor starting torque by switching the motor winding connections. Three contactors and a timer change the winding configuration to transition from around 33 percent current in the wye configuration to full current in the delta configuration. Open transition wye-delta starters stop applying voltage to the motor in the short transition between windings. A closed transition wye-delta starter uses shunt resistors. Only closed transition wye-delta starters are now used in large chillers. Due to its relatively low cost when compared to the reduced voltage type starters, wye-delta starting is the most commonly used starter with hermetic centrifugals under 600 volts.

Wye-delta starters

are popular with technicians because of their familiarity with wye-delta trouble shooting procedures.

Wye-delta starters are used only on low-voltage applications.

Solid-State

A solid-state starter's principle of operation utilizes a solid-state device known as a silicon controlled rectifier (SCR). The SCR is an electronic "valve," which allows a volume of current to flow through it in response to a controlling electronic signal. For a solid-state starter, inrush current decreases in a somewhat linear fashion until the operating current of the motor is reached. In this manner, the starter supplies only enough voltage to overcome the motor starting torque requirement. The applied voltage can be controlled to limit inrush at any desired level above that needed to rotate the compressor motor.

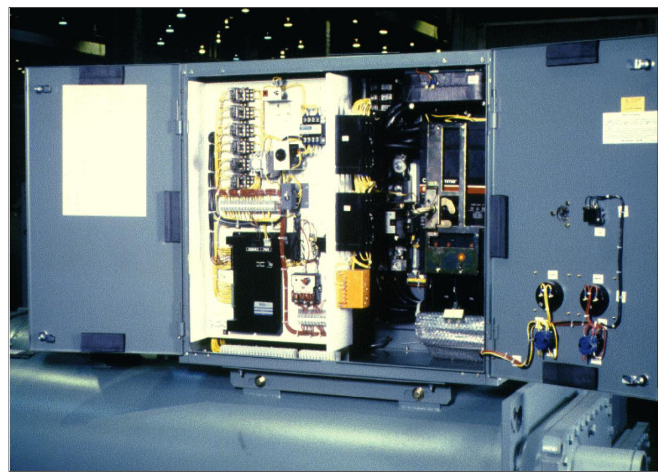


Figure 35

Unit-Mounted Solid-State Starter

Solid-state starters are used for both low-voltage and medium-voltage applications although they are not as common in medium voltage. They are popular for low voltage applications (≤600 volts) because they provide a programmable soft start.

Turn to the Experts.

Variable Frequency Drive

The newest soft-start technology in starting equipment for water-cooled chillers is the variable frequency drive (VFD), also known as variable speed drive (VSD). These drives provide the softest start, because the drive controls start at zero current and ramp up just until the motor inertia can be overcome. This results in inrush current equaling rated load amps (RLA) during start up. RLA is a very low inrush value.

Figure 36

VFD Starter for Unit-Mounting

VFDs have become very popular as a starting method for water-cooled centrifugal chillers. Several reasons are:

- Low inrush at start-up, low mechanical stress
- High power factor (the ratio of active power to apparent power)

  A motor that has a high power factor makes more efficient use of the incoming energy.
- Quiet chiller operation at reduced rpm
- Energy savings at part load
- Best for emergency generator applications due to low inrush
- Utility rebates often available

The advantage of VFDs

is that they can slow down the compressor by varying the frequency applied to the motor. The speed reduction results in reductions in lift. This translates into lower operating costs because a compressor becomes more efficient when the speed is reduced to no more than is required to satisfy the load.

Paybacks of two years or less are not uncommon in VFD applications. This actual power saving is a result of the relationship of power versus speed, which will be discussed later in the TDP.

Starting Method	Motor Starting Current as a % of	
	Locked Rotor Current	Full Load Current
Across-the-Line	100	600
Auto Transformer & Primary Reactor		
80%	80	480
65%	65	390
50%	50	300
Part Winding	65	390
Wye-Delta	33	200
Solid-State	0 - 100	0 - 600
VFD	16.6	100

Figure 37

Motor Starting Current as a Percent of Locked Rotor and Full Load Current

Energy Management

Use of energy management practices can significantly reduce operating costs, especially during off-peak modes of operation. Demand limiting and chilled water temperature reset are two techniques for accomplishing efficient energy management.

Chilled Water Reset

Chilled water reset means to change the chilled-water temperature leaving the chiller based on some parameter. Increasing the leaving chilled-water temperature reduces compressor power usage by reducing lift at both full and part-load conditions. However, at part-load conditions, design chilled-water temperature may not be necessary, especially for comfort cooling applications, so reset is possible. However, increasing the chilled water temperature may result in greater humidity levels in the conditioned spaces. Higher coil temperatures resulting from the increased chilled water temperature will reduce the latent heat capacity of the coils and the ability of the air distribution system to remove space humidity.

> ***Lift and Efficiency***
>
> ***Lift refers to the difference between compressor suction and condensing temperatures. Reducing lift increases efficiency.***

Chilled water temperature can be reset as a function of:

- Ambient air temperature (most common) – used when the ambient temperature is the best indication of load
- Return chilled water temperature – used when return water temperature is the best indication of load. This method is typically used when it is desired to maintain a fixed Δt in the chiller plant.
- Temperature within the building – used when space temperature is the best indication of load on the chiller. This is used when there are critical areas in a building such as labs that might require a specific temperature be maintained.

Reset doesn't always mean an increase in water temperature. For example, space temperature reset could actually lower the chilled water supply temperature, not raise it, in order to maintain conditions.

Demand Limit and Duty Cycling

Demand limit is a feature that limits the unit capacity during periods of peak energy usage. When a utility company's demand for electricity exceeds a certain level, users are charged extra money called demand charges. To avoid these charges, loads in the building are limited to keep demand below a prescribed maximum level. Demand limiting may be desirable on hot days when air conditioning is most needed. Demand may be limited on the chiller by increasing the chilled water temperature or by unloading the chiller regardless of actual load to a predetermined percentage of the current draw which is an indication of load. This may result in temporary increased building or process temperatures if the actual load is greater than the demand limited capacity that the chiller can provide.

> ***Demand is measured***
>
> ***in kW; energy charges are measured in kWh. Together, the two form a typical user's electrical bill.***

Duty cycling will cycle selected electrical equipment in an installation (building, factory, etc.) at regular intervals to limit the electrical demand, thereby lowering demand charges. However, duty cycling is not recommended because constant cycling will cause increased stress and damage to the motor windings, bearing, and controls. If demand must be controlled, the demand limit sequence available from the manufacturer is recommended.

Screw Compressor Operational Details

The screw compressor is classified as a positive displacement compressor, which means that a volume of gas is trapped within an enclosed space whose volume is then reduced. Screw compressors are composed of multiple parallel rotors with external helical profiles fit into a casing. The drive rotor can either be coupled directly to the motor (direct drive) or through a gear transmission. As it turns, it moves the other rotor (driven rotor). It is easy to relate the compression process to a reciprocating compressor, if you consider the drive rotor as the piston and the driven rotors as the cylinder. As the drive rotors and driven rotors un-mesh, an empty cylinder is created, drawing in suction gas through the synchronized opening on the rotor suction face. As rotation continues, the suction and discharge rotor faces are sealed off, trapping the gas in the cylinder. When this happens, the meshing point moves toward the discharge end of the rotors and drives the gas ahead of it. As rotation continues further, the drive rotors rotate into the flutes of the driven rotors and internally reduce the contained volume, resulting in positive displacement compression. The discharge port provided for the gas escape is relatively small, compared to the suction port, reflecting the smaller volumetric flow rate of compressed vapor.

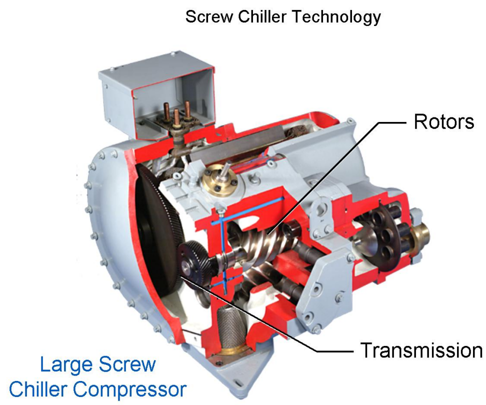


Figure 38

Screw Chiller Technology

Volume Ratio

Volume ratio is defined as the ratio of the volume of the compression chamber when it is open to the suction port divided by the volume of the compression chamber when it is open to the discharge port. This is much like in a reciprocating compressor, where the volume ratio is the volume of the compression chamber when the piston is at the bottom of its travel divided by the volume of the chamber when it is at the top of its travel. Volume ratio (v_r) is also called volume index (v_i).

> ***Most screw compressors***
>
> ***will be badly damaged by reverse rotation so there are built in controls to stop the compressor if reverse rotation occurs.***

Design and Off-Design Performance

The design and size of the rotors and the placement of the discharge ports control the amount of compression for a screw compressor. This design fixes the built-in volume ratio (v_i), which defines the change in volume within the compressor from a larger volume at compressor inlet to a smaller volume at compressor exit. For a given refrigerant, a fixed volume ratio corresponds thermodynamically to a certain pressure ratio. The higher the volume ratio (v_{in}/v_{out}) the larger the corresponding pressure ratio (P_{out}/P_{in}). When the required pressure ratio is smaller than the design pressure ratio, it corresponds to a smaller volume ratio than the built-in volume ratio and over-compression will occur with subsequent sudden expansion when the overcompressed pocket of vapor reached the exit port resulting in the so-called over-compression losses. Conversely, when the required pressure ratio is larger than the design pressure ratio, it corresponds to a larger volume ratio than the built-in volume ratio and under-compression will occur with subsequent sudden compression inside the final section of the compressor when the undercompressed pocket of vapor reached the exit port. The compressor is capable of delivering pressure ratios higher than the design pressure ratio (as opposed to turbocompressors, which have limited additional pressure ratio capability above their design pressure ratio because of surge) but these higher-pressure ratio conditions are suffering from under-compression losses.

Suction volume flow rate defines a screw compressor's capacity. For a given speed, it is more or less constant irrespective of the pressure ratio delivered by the compressor. In order to achieve capacity variation the suction volumetric flow rate has to be changed. There are two ways of doing this in a continuous way. Capacity changes proportional to speed on variable speed machines. The built-in volume ratio is not affected by the speed of the compressor. Fixed speed screw compressors use a slide valve, a mechanical device that tries to change the inlet volumetric flow rate of the compressor while maintaining a required built-in volume ratio at lower flow rates. From this description, it becomes clear that careful consideration must be given by manufacturers in designing the screw compressor's operating envelope to efficiently handle the required compression or work for the HVAC system under all possible operating conditions. It is important to select the optimum built-in volume ratio for a given HVAC system application in order to assure that the compressor operates most of the time close to its design pressure ratio, thus limiting the over-compression and under-compression losses that occur at pressure ratios away from design conditions.

Manufacturer's selection software and printed literature incorporates the important performance criteria described above. When selection data is generated from the software, or is shown in printed literature, that is your assurance that the application points are operationally sound.

Centrifugal Compressor Operational Details

Imagine a ball attached to the end of a string being whirled by a person holding the other end of the string. The ball pulls outward on the string, and if the string is released, the ball flies away on a line perpendicular to its circle of rotation. The heavier the ball, the more it pulls on the string. The longer the string is, the harder the pull of the ball will be. The faster the ball rotates the harder the pull of the ball.

This is analogous to the centrifugal compression of gas. Replace the ball with a molecule of refrigerant gas, replace the string with an impeller, and the effect is the same. The centrifugal force imparted to the refrigerant molecule will fling it outward, compressing it into the impeller passageways. The larger the diameter of the centrifugal impeller is, the larger the force on the molecule. The faster the impeller rotates, the larger the force on the molecule. The above phenomenon describes how the centrifugal compressor increases the pressure of the vapor being compressed in the outer section of the impeller. An additional pressure increase results as the high speed impeller imparts kinetic energy to the gas. The diffuser converts the high kinetic (velocity) energy of the refrigerant gas to static pressure when the gas is released into the stationary discharge or diffuser portion of the compressor.

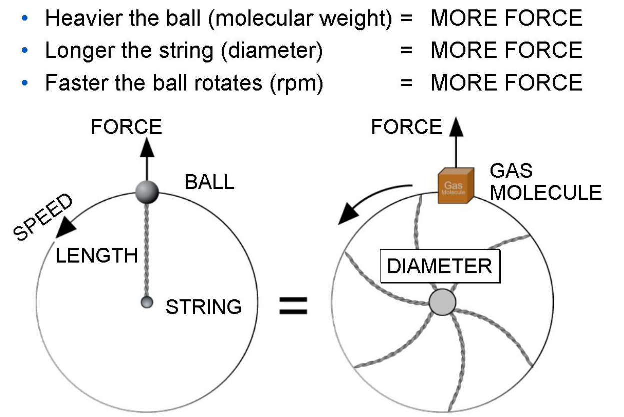


Figure 39

Centrifugal Compressor Theory

Head

Head can be visualized as the height (in feet) of an imaginary column of refrigerant vapor, which, due to its weight, produces the same pressure as that developed across the compressor (pressure at discharge minus pressure at suction). Head has come to be synonymous with a measure of the pressure rise developed by a centrifugal compressor.

> ***Head pressure***
>
> ***is directly affected by entering water temperature from the cooling tower. The entering wet bulb temperature to the tower affects the tower's ability to cool the water.***

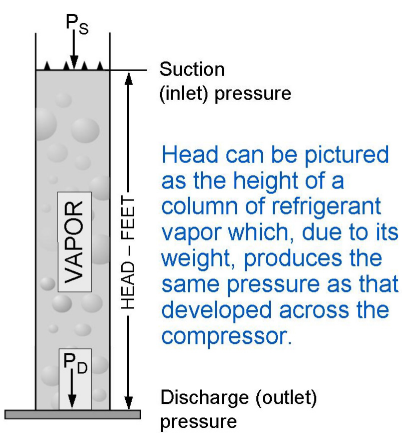


Figure 40

Centrifugal Compressor Theory – Head

Variations in outside conditions result in different entering condenser water temperatures at both full and part load. This term for this is tower water relief and occurs when the outside wet bulb temperature is below the climatic design condition, increasing tower heat rejection and lowering the condenser water temperature entering the chiller.

Lift

Another commonly used term is “lift.” The expression, “55º F of lift” refers to the difference in saturated refrigerant temperature levels between which the compressor operates. To produce 44º F chilled water, a compressor draws in gas at a saturated suction temperature level of approximately 42º F and discharges gas at a saturated discharging temperature of approximately 97º F. This is for typical HVAC duty, with 85 to 95º F condenser water. The conditions described here would produce 55º F of lift (97° F - 42° F). The following pressure enthalpy diagram demonstrates that lift is the difference between saturated condensing and suction temperatures.

Under many circumstances, an increase in lift requires an increase in compressor and/or heat exchanger sizes. Some factors that increase lift are lower leaving chilled water temperatures, lower condenser flow rate, higher condensing temperature, fouling on the waterside of the tubes, and the thermal properties of fluids being used in the heat exchangers.

Any reduction in lift means a reduction in the power needed for the compressor and savings in operating costs, as well as increased capacity for chillers equipped with variable metering devices. Colder condenser water temperatures lower the lift and have the greatest positive impact on the refrigerant saturated condensing temperature.

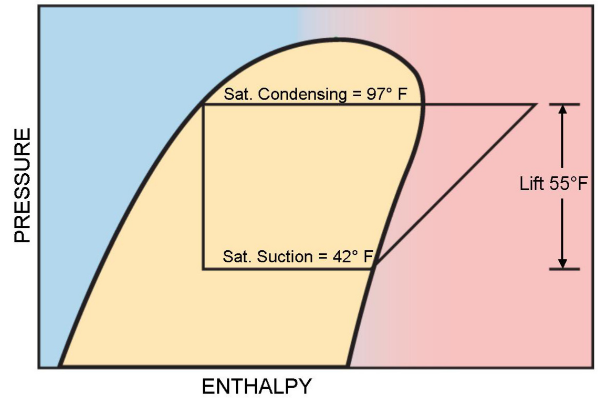


Figure 41

Lift

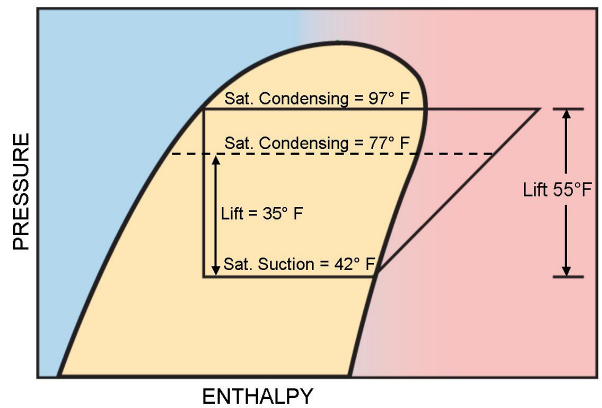


Figure 42

Lift Reduction

> ***Head and Lift***
>
> ***Represent the same thing. Head Is the pressure difference; lift is the temperature difference between compressor suction and discharge.***

Changing weather conditions produce a reduction in outdoor wet bulb temperature that in turn allow the cooling tower to produce colder than design entering condenser water temperature to the water-cooled chiller. Chillers take advantage of this natural occurrence called cooling tower relief in two areas: stable operation down to approximately 55º to 60º F entering condenser water temperature, and an increase in capacity from design tonnage. This is often referred to as “maximum capacity.”

Compressor Boundaries

There are two important factors when discussing the operational envelope of a centrifugal compressor. The first is stonewall, which describes the condition of maximum compression. As the volume of gas increases, its velocity through the compressor also increases. When the velocity of the gas exceeds the local sonic or acoustic velocity at the existing pressure, shock waves develop. When this happens, high head losses occur and the compressor head output drops off sharply. This is the “flow limitation,” and the resultant steep part of the compressor curve showing this effect is referred to as “stonewall.” Centrifugal chiller software automatically keeps selections the proper distance from this line.

The other limitation for the centrifugal compressor is surge. Surge occurs when the refrigerant pressure in the condenser is high enough that the impeller can no longer maintain the required discharge gas flow. Stated another way, the required lift is too high for the compressor. This higher condenser pressure results in flow breakdown in the impeller passageways, and a partial or complete flow reversal through the impeller occurs. A lowering of the system pressure should follow, and this allows the impeller to function normally again, and gas to flow in the correct direction. The system pressure then builds up again until it exceeds the impeller capability, and the reversal of flow repeats itself. This pulsing of pressures is nearly instantaneous, and continues until chiller controls take corrective action to lower the condenser pressure, allowing proper flow of the refrigerant through the impeller, diffuser, and volute. Surging is characterized by an increase in the operating noise level, and by wide fluctuations in discharge pressure and motor current. No detrimental mechanical effects are caused by periodic, short durations of surging. However, prolonged periods of surging may damage the chiller.

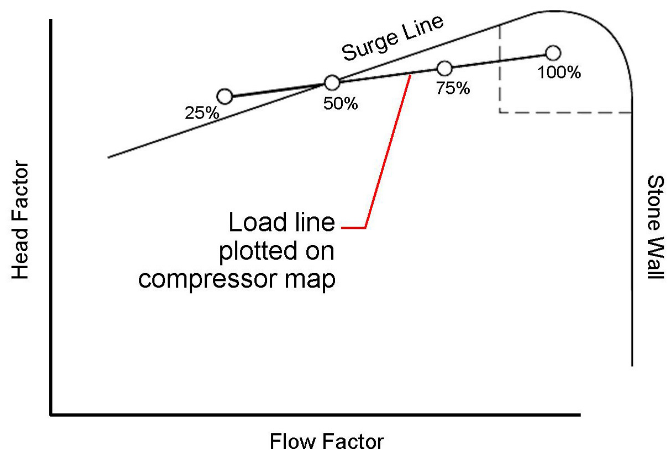


Figure 43

Centrifugal Chiller Compressor Map

Several conditions can cause this pressure imbalance. An example would be a reduction in load, during normal operation, without a reduction in condensing temperature (low load, high lift). The refrigerant flow rate is too low in this case, and so the velocity of the gas, and therefore the pressure generated by the compressor is too low to overcome the pressure in the condenser. A second example would be the operation of a centrifugal chiller at higher entering water temperatures than were used to make the initial selection. This condition may create a lift too great for the compressor.

Most centrifugal chillers

can be selected to run to a certain level of low load and high lift, but these conditions must be planned for at time of selection. The key is to verify the requirements for each project.

These situations can be avoided by properly selecting the chiller for the actual operating conditions for the application.

Compressor Stages

In a centrifugal compressor, each impeller used is referred to as a stage. We have seen from the ball and string example that a single-stage compressor with a large diameter impeller running at a slow speed, or a single-stage compressor with a small diameter impeller running at a fast speed, can produced the same amount of lift. A two-stage compressor with even smaller diameter wheels running at a faster speed can also produce the same amount of lift. Thus, there is a design trade off between speed, staging, and impeller diameter to achieve a desired result. For air-conditioning duty (approximately 55º F of lift) either one or two-stage compressors are used.

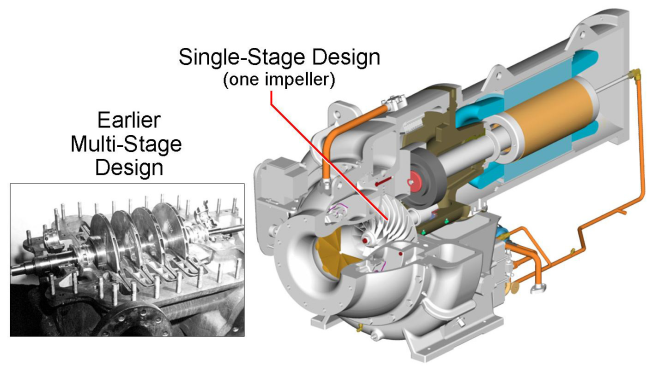


Figure 44

Centrifugal Compressor Stages

Capacity Control Methods for Centrifugals and Screws

In most water-cooled chiller applications, the machine must respond to changes in two basic variables. These are: entering condenser water temperature and refrigeration load. Both parameters affect the lift requirements on the system at any time, and any reduction in either is considered a part load condition. Centrifugal and screw compressors can accommodate these changes, even though they have different means to control refrigerant flow (or capacity).

Inlet Guide Vanes

One method of capacity control involves keeping the speed of the compressor constant and varying the flow of the refrigerant to the compressor. To maintain stable operation in a centrifugal, the most common method of flow control is by using an inlet guide vane assembly (IGV). These vanes are usually located just ahead of the inlet to the impeller and are controlled to the leaving chiller water set point. As the IGVs close, they reduce the mass flow of refrigerant through the compressor.

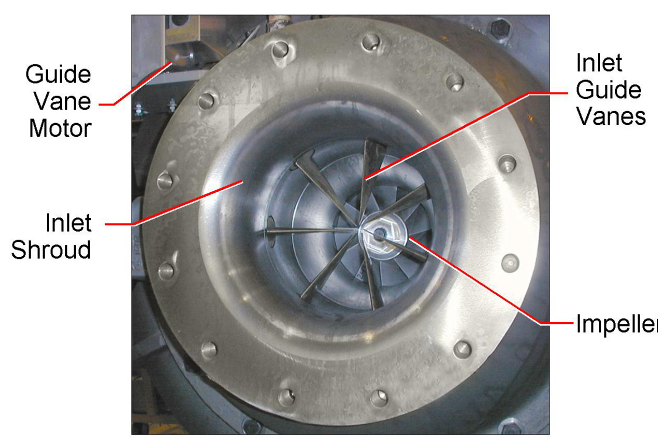


Figure 45

Centrifugal Chiller Capacity Control - IGV

At part load, closing the vanes imparts prewhirl to the gas, minimizing impeller-entry pressure loss.

Inlet guide vanes change the head-volume characteristic of the compressor by changing the angle at which the suction gas enters the compressor. At full load, the gas is directed against the compressor rotation, and the compressor actually "bites into" the suction gas, picking up more gas than if the gas entered radially. At part load, the angle changes such that the gas enters with rotation minimizing any pressure loss through the vanes by imparting a prewhirl to the gas.

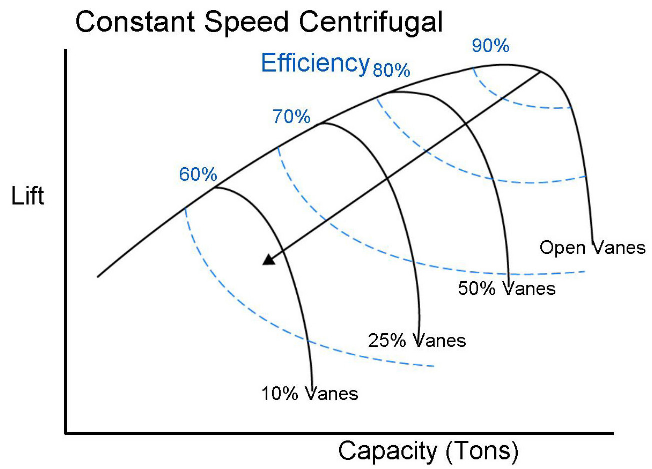


Figure 46

Compressor Efficiency Characteristics with Inlet Guide Vanes

This method offers very good efficiency over a wide range of capacity. At half load condition for example, the input power required is approximately 45 percent of full load input power. Shown here are the compressor head-volume characteristics at part load. The compressor can balance head and volume changes efficiently down to 10 or 15 percent load without the use of hot gas bypass as long as the operating point falls within the surge boundary.

Screw Unloaders

For a screw chiller, either slide valves or solenoid actuated port valves are used to change the volume ratio of the compression chamber.

A common means of capacity control is changing the volume ratio with slide valves. The slide valve opens a passageway along the side of the compression chamber, allowing a portion of the gas to bypass the rotors. This bypassed gas is recycled to the suction cavity and is then compressed. Essentially, a slide valve shortens the length of the compression path, reducing the capacity of the screw compressor. The slide valve offers infinite volume ratio, and therefore infinite capacity control for screw compressors.

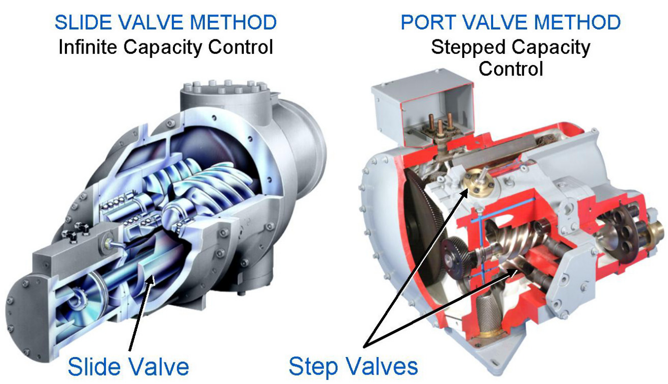


Figure 47

Capacity Control – Slide and Port Valves

Another means of reducing the volume by shortening the compression path is port valves. Port valves are staged along the compression path, and the number of opened valves determines the volume of gas compressed. These valves are less expensive, easier to control, and simpler to maintain than slide valves, but they do not offer infinite control as they open in discrete steps, and therefore reduce the volume of gas compressed in the same discrete steps.

Hot Gas Bypass

Bypassing gas around the condenser has the effect of falsely loading the compressor. Hot gas bypass is used to accomplish stable operation if a load exists beneath the minimum step of normal unloading capability of the chiller compressor. On large screw and centrifugal chillers, hot gas bypass is available either factory-installed or field-installed. Hot gas bypass may also be used as a surge prevention method in centrifugal chillers.

Speed Control

Variable speed control provides the most efficient method to vary capacity, but in the past, it has been expensive. Recent improvements in technology and the widespread acceptance of variable frequency drives (VFD) have lowered the cost such that today's VFD technology is often in the 1-2 year payback period. Any time the lift requirements of a chilled water system are reduced, a VFD applied to the compressor can realize power consumption savings that come from the lower speeds required at the reduced lift. For a centrifugal compressor, a reduction in speed is represented by the reduction in gas flow rate, lift capability, and input power as shown.

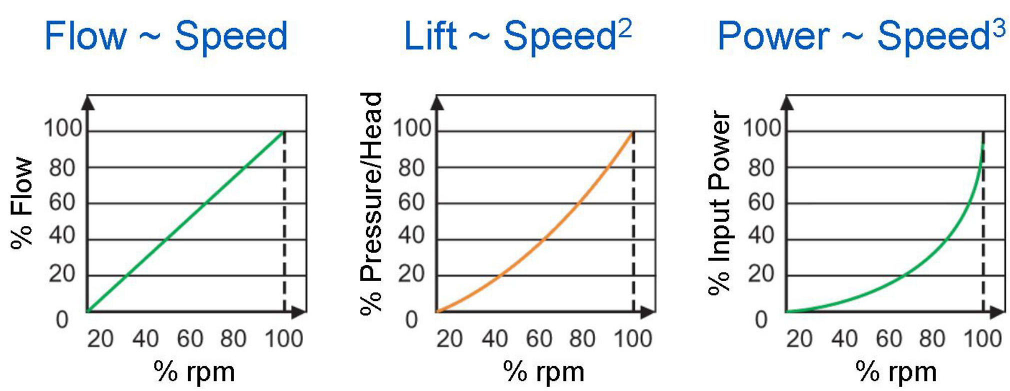


Figure 48

Laws for Centrifugal Loads

1. Flow is proportional to speed
2. Lift is related to speed squared
3. Power is related to speed cubed

All of these centrifugal laws are based on how fast the impeller is spinning, and anytime there is a reduction in lift and flow, the compressor speed can be reduced by a VFD, resulting in energy savings.

VFD retrofit

is an increasingly popular energy saving measure on existing large chillers.

VFD control provides a much more efficient means to control capacity for a centrifugal compressor, because as inlet guide vanes close, the efficiency of the compressor rapidly declines.

The example graph shows a plot of the kW per ton of a partly loaded centrifugal chiller at reduced lift conditions. The reduced lift conditions are the direct result of a reduction in entering condenser water temperature at part load. Besides the energy savings, there are lower sound levels that result from running the chiller at reduced speeds.

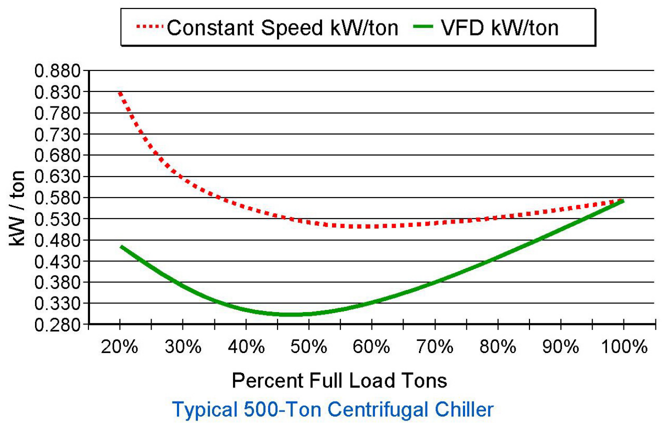


Figure 49

VFD Versus Constant Speed Centrifugal

The effect of VFD control is to continuously "shrink" the compressor, keeping the operation in the maximum efficiency region over a much broader range of operation. Essentially, the VFD adjusts the speed of the compressor to keep the IGVs as open as possible to meet the system lift requirements, with the lowest power consumption. Combined with condenser water temperature reduction that occurs naturally in an air-conditioning system, the variable speed centrifugal compressor more efficiently meets the flow and lift condition or state point required by the system.

A centrifugal compressor's dynamic design needs sufficient rotational speed to overcome system lift requirements. As a result, centrifugal chillers cannot be controlled by speed variation *alone*. Centrifugal chillers must be controlled by a combination of speed variation and inlet guide vane actuation in variable speed applications. Thus, centrifugal compressors are limited in the amount that they reduce compressor speed under high lift conditions.

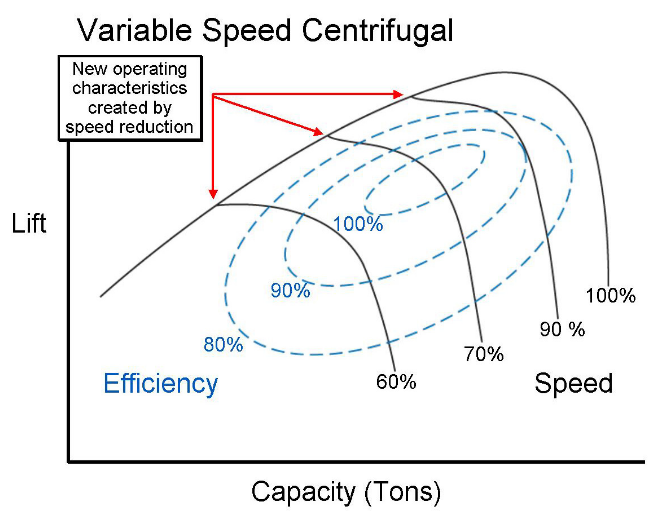


Figure 50

Variable Speed Centrifugal Operating Characteristics

Refrigerant Related Topics

Design pressures, cycle efficiencies, toxicity, chemical stability, and environmental impact have limited the refrigerants in common use to a group of fluorinated hydrocarbons. Members of this group include Refrigerants 22, 123 and 134a. There are also several refrigerants that are blends of other refrigerants. These blends are classified as either azeotropic or zeotropic.

New refrigerant choices…

Include blends, like R-410a (Puron™) and R-407c that are being used to replace R-22 in applications other than in centrifugal chillers.

The former means that the blend behaves as if it were a single refrigerant; the latter describes a blend where the different refrigerants in the blend evaporate at different temperatures. The disadvantage of a zeotropic refrigerant is that if there is a leak, a change in the composition of the blend may reduce the system capacity and/or efficiency. In some cases, the entire charge might be unusable if a leak were to occur.

Regulations

Recently, refrigerant selection has been driven by two international environmental agreements, the Montreal Protocol on Substances that Deplete the Ozone Layer and the Kyoto Protocol, which is focused in minimizing the impact of climate change. The Montreal Protocol mandates the phase out of ozone depleting substances. The CFC's, R-11, R-12, R-113, R-114 and R-115, were phased out on January 1, 1996. The HCFCs (e.g. R-22 and R-123) will be phased out on January 1, 2030 with interim reductions taking place on January 1, 2004, January 1, 2010, January 1, 2015, and January 1, 2030. From January 1, 2020 to January 1, 2030, consumption will be reduced to 0.5 percent of the baseline and new HCFCs may only be used to service existing equipment. The US and Europe, as well as a majority of developed countries, have adopted the Montreal Protocol and in some cases have passed regulations that accelerate the phase out of some or all HCFCs. In Europe, most uses of HCFCs have already been banned.

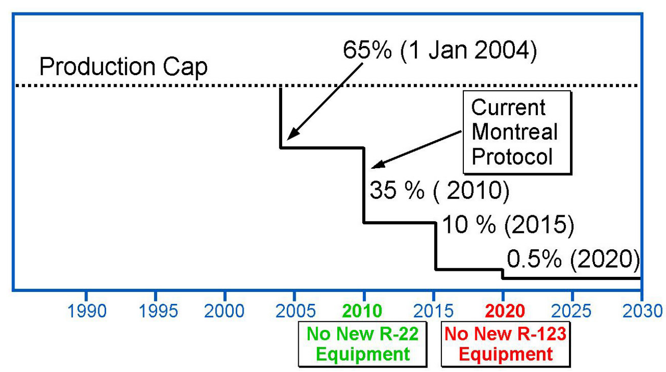


Figure 51

Montreal Protocol Showing EPA Equipment Phaseout Dates

Note:

The Montreal Protocol treats HCFC-22 and HCFC-123 the same, however, the U.S. has adopted chemical-specific phase-out dates. HCFCs with higher ODP (R-22)will be phased out first followed by HCFCs with lower ODP. Therefore, in the U.S., HCFC-22 Is being phased out ahead of HCFC-123. For example, as of 2010, no new R-22 equipment can be produced, and as of 2020 no new R-123 equipment can be produced.

The Kyoto Protocol mandates the reduction of emissions of greenhouse gases (i.e. carbon dioxide, methane, nitrous oxide, HFCs). It does not call for the phase out of the chemicals. The Kyoto Protocol has affected chiller design by influencing manufacturers to find more efficient refrigerants, and therefore designing more efficient chillers to reduce the electrical consumption of HVAC systems.

It is also important to note that any refrigerant for use in chillers must be on the EPA's Significant New Alternative Policy approved list, and in ASHRAE Standard 34 to ensure that it has an "R" designation and is assigned a safety classification based on toxicity and flammability data. If it is not, using it is a violation of the Clean Air Act, and is potentially hazardous. Several refrigerants that are described as drop-in replacements for HCFC chillers are considered dangerous and are not approved by the EPA. In addition to this, the equipment that the substitute refrigerant will go into must undergo design changes to ensure safe operation of the approved refrigerant alternative. One significant reason for this is the different operating pressures of refrigerants. As an example, R-22 replacement R-410A operates at about 30 percent higher pressure, and if the original equipment is not designed to operate at that pressure, it would not be safe to apply R-410A into the system without modifications. For complete discussion on this topic, refer to TDP-402, Refrigerants.

Chiller Construction

The choice of refrigerant affects the construction of the chiller as well. Different refrigerants have different operating pressures, so various codes dictate their construction. The cooler and condenser may need to be constructed to ASME pressure vessel standards as defined in Section VIII, Division I of the current ASME code. ASME stands for the American Society of Mechanical Engineers. A data report for each positive pressure centrifugal chiller is on file with the National Board of Boiler and Pressure Vessel Inspectors and is available for public review. In this report, an authorized independent inspector will have approved the welds and verified the pressure tests. These inspectors, working on behalf of ASME, not the chiller manufacturer, ensure that the positive pressure chillers are manufactured and tested in accordance with the strict standards of ASME and will bear the "U" stamp indicating compliance with ASME. They provide third party certification during the manufacturing process.

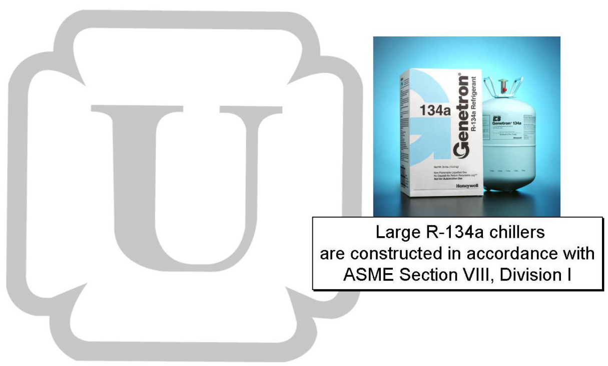


Figure 52

Chiller Construction ASME "U" Stamp
Photo Courtesy of Honeywell Genetron Refrigerants

For large screw and centrifugal chillers, the quality of the construction enables hermetic positive pressure chillers to have approximately 0.1 percent annual refrigerant leak rate. ASME currently requires that the refrigerant side (shells) of the R-134a chiller be pneumatically tested to 1.1 times the design pressure, a value typically around 200 psig. Negative pressure chillers are not inspected by an ASME authority, and are tested to 30 psig as required by ANSI the American National Standards Institute. Waterside testing of the tubes is typically 1.3 times the design pressure of the water. This design pressure is usually 150 or 300 psig.

ASME stamp requirements

dependent on the physical size and the refrigerant pressure rating on the chiller. For example, cooler and condenser vessels less than 6 inches inside diameter may be constructed to comply with another nationally recognized testing laboratory other than ASME. This may vary from manufacturer to manufacturer.

On larger chiller designs like centrifugals and screws using positive pressure refrigerants such as R-134a, the evaporator and condensers meet or exceed the minimum size requirement and they must bear the symbol signifying compliance with ASME. On negative pressure centrifugal chillers, the refrigerant pressures involve exclude the chiller construction from consideration by ASME.

Chiller construction is also affected by the physical size of the refrigerant molecules. The chart shows the relative molecular size of the refrigerants currently in use. This means that for the same capacity, an HFC-134a chiller can often be approximately 30 percent smaller than a negative pressure chiller of the same cooling capacity. The smaller footprint results in savings in installed costs and mechanical room size.

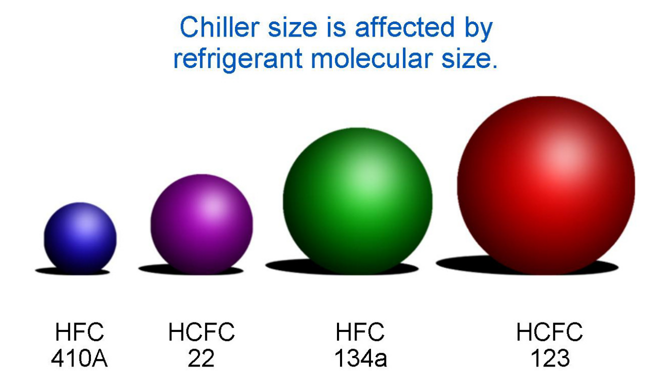


Figure 53

Refrigerant Impact on Chiller Size

Safety

ASHRAE Standard 34 divides refrigerants into two groups, lower and higher toxicity, designated by the letters A and B, respectively. Refrigerants with allowable exposure limits (AEL) of 400 ppm or higher are classified as group A (lower toxicity) refrigerants, and those with AELs of less than 400 ppm are group B (higher toxicity). R-134a with an exposure limit of 1000 ppm is classified as Group A and R-123 with an exposure limit of 50 ppm is classified as Group B.

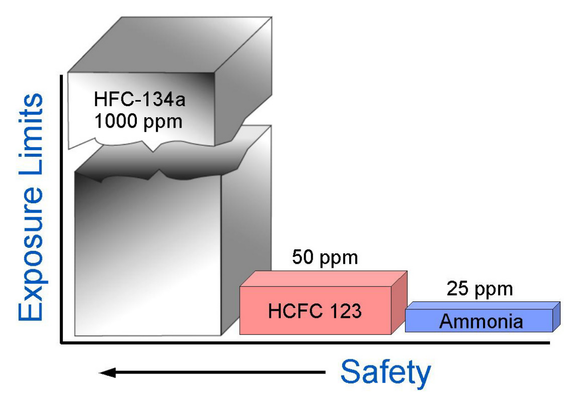


Figure 54

Allowable Exposure Limits

Refrigerants are further classified by their degree of flammability: (1) no flame propagation, (2) lower flammability, and (3) higher flammability. HFC-134a is a safe choice, because it has a low toxicity and no flame propagation properties

HFC-134a has become the leading choice for a long-term refrigerant solution because it is chlorine-free and therefore has zero ozone depletion potential and is not subject to phaseout under the Montreal Protocol. Today, HFC-134a is being applied in foam blowing, asthma inhalers, auto air conditioning, residential air conditioning, water fountains, and domestic refrigerators.

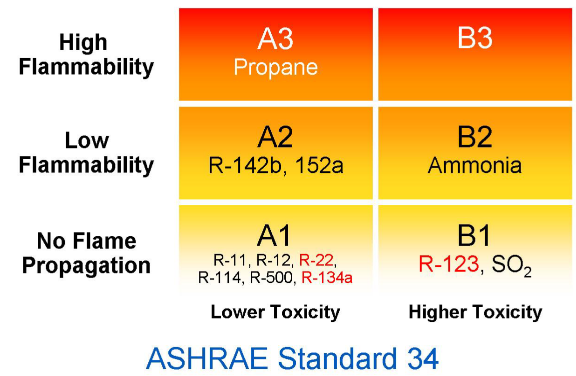


Figure 55

Refrigerant Safety Groups

Heat Transfer

This section will look at the basic heat transfer relationships of the heat exchangers for a water-cooled chiller. The information provided is general in nature and should provide a basic understanding of the heat transfer principles associated with shell-and-tube heat exchangers.

Shown is the longitudinal cross section of a typical condenser and evaporator together with their respective temperature profiles. The heat transfer relationships of the evaporator and condenser are similar.

In looking at the figure, it can be observed that there are two basic heat balances occurring in each heat exchanger. They are:

1. Heat given off or gained by the fluid as it passed through the tubes.
2. The overall transfer of heat between the fluid in the tubes and the refrigerant in the shell.

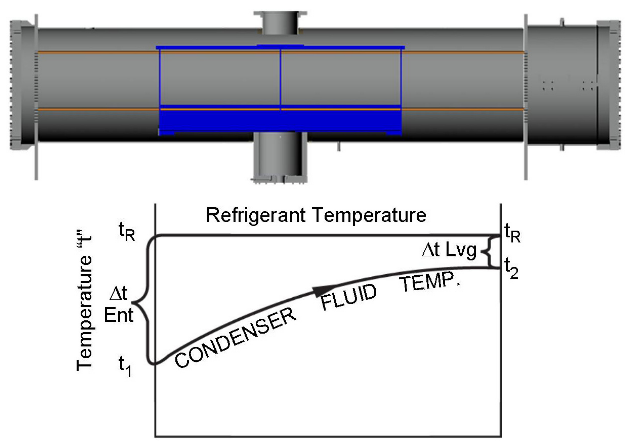


Figure 56

Heat Transfer – Condenser, Fluid Heated t_1–t_2

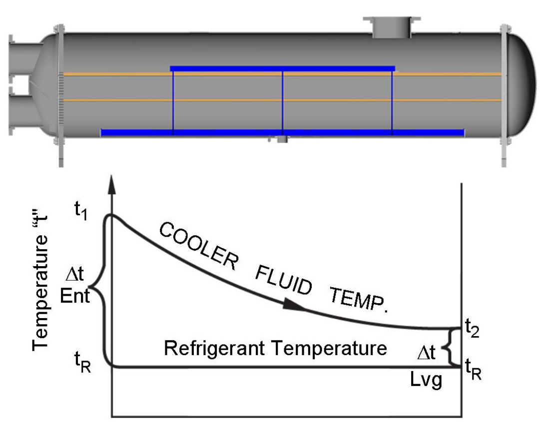


Figure 57

Heat Transfer – Evaporator, Fluid Cooled t_1–t_2

Heat transfer technology has improved to the point where the difference between the refrigerant temperature in a heat exchanger (evaporator or condenser) and the leaving fluid temperature can be very low…in the range of 2-3 ºF. That means, for instance, if the saturated condensing temperature is 97º F, the water returning to the cooling tower may be as high as 95º F. If the saturated suction temperature is 42º F, the leaving chilled water may be as low as 44º F. Later in this TDP module we will examine the components affecting heat transfer between the refrigerant and water in the tubes as well as tubing technology that promotes increased heat transfer.

Heat Balance of Fluid

The fluid in the cooler tubes is cooled from t_1 to t_2 as it gives off heat to the colder refrigerant. Similarly, the fluid in the condenser tubes is heated from t_1 to t_2 as it takes on heat from the hotter refrigerant. The total amount of heat exchanged in either vessel can be expressed as follows:

$$q = W * C_P * \Delta t_F$$

Where:

q = quantity of heat exchanged (Btuh)

W = flow rate of fluid in tube (lb/h)

C_P = specific heat of fluid [Btu/lb(ºF)]

Δt_F = temperature rise or drop of fluid in tubes

$sp\ gr$ = specific gravity

The above equation can be rearranged in the following form, and it applies to both evaporator and condenser.

$$tons = \frac{gpm * \Delta t_F * sp\ gr * C_P}{24}$$

For the evaporator, the tons equals cooling capacity. For the condenser, the tons equal cooling capacity plus the heat equivalent (in tons) of the work of compression.

The above equation for tons is commonly used to determine flow rate, temperature rise, or load when any two of the factors are known. When the fluid is fresh water, the specific gravity (sp gr) and specific heat (C_P) have values of approximately 1.0, and the equation reduces to:

$$tons = \frac{gpm * \Delta t_F}{24}$$ for the evaporator and the condenser.

Where:

gpm = flow rate for evaporator or condenser

Δt_F = difference in fluid entering and leaving

24 = conversion to tons for fresh water

In converting

the tons equation into a more recognizable form involving tons, some standard units of conversion were used.

Heat balance for a water-cooled chiller means that the evaporator capacity in Btuh or tons when added to the compressor work, numerically equals the total heat rejected by the condenser water flow. A few simple equations allow checking the heat balance when hermetic motor type chillers are being evaluated. There is a small degree of latitude in the heat balance permitted by the industry to account for measurement and manufacturing tolerances.

A heat balance is very useful in checking chiller selections for careless mistakes or verifying chiller performance of an existing chiller. In using a heat balance on an existing chiller, it may be difficult to predict the exact water flow.

Another equation that is useful in converting water flows into heat transfer is:

Btuh = 500 * (gpm) * Δt, where 500 is a constant for fresh water.

Example: Heat balance for a centrifugal chiller selection.

Shown in the capacity printout:

Capacity	640 tons
Compressor kW	402
Condenser gpm	1393
Condenser rise °F	13

Let's calculate the heat balance for the chiller:

Heat In = Heat Out

Chiller capacity + compressor kW = condenser heat rejection

$$640\ tons * 12{,}000\ Btuh/ton + (402\ kW * 3413 Btuh/kW) = 500 * 1393\ gpm * 13°\ F$$

3413 converts Btuh to kW

9052 MBtuh is the evaporator and compressor heat

9054 MBtuh is the condenser total heat of rejection

This heat balance is basically equal at less than 1 percent difference. General industry practice is a balance within 5 percent is considered acceptable.

Overall Heat Transfer

Heat added to or subtracted from the refrigerant at a constant pressure results in a change of state of the refrigerant. The heat required to change the refrigerant state in the evaporator from a liquid to a vapor is known as the "latent heat of vaporization." In the condenser, the heat required to change the refrigerant state from a vapor to a liquid is known as "the latent heat of condensation."

In either the evaporator or the condenser, this process takes place at a constant refrigerant saturation temperature corresponding to the saturation pressure in the vessel. The rate at which heat transfer takes place is directly related to the entering temperature difference, and the leaving temperature difference.

The overall heat exchanged between the fluid and the refrigerant for the cooler or the condenser can be expressed as:

$$q = U * A * LMTD$$

Where:

q	=	overall heat exchange rate (Btuh)
U	=	overall heat transfer coefficient
A	=	area of heat exchanger tubes (ft^2)
$LMTD$	=	log mean temperature difference between the fluid and the refrigerant (ºF)

The LMTD of either heat exchanger can be calculated from an equation that takes into account the entering and leaving differences between fluid and refrigerant temperatures.

$$LMTD = \frac{\Delta t_F}{Log_e \dfrac{\Delta t_{ent}}{\Delta t_{lvg}}}$$

Where:

Δt_F = rise or drop in temperature of the fluid flowing through the tubes

Δt_{ent} = temperature difference (ºF) between the entering condenser fluid temperature and the saturated condensing temperature, or the entering evaporator fluid temperature and the evaporator suction temperature

Δt_{lvg} = temperature difference (ºF) between the leaving condenser fluid temperature and the saturated condensing temperature, or the leaving evaporator fluid temperature and the evaporator suction temperature

This equation for overall heat transfer looks familiar to us. The heat transfer through a wall assembly when figuring heating and cooling loads is quite similar ($U * A * \Delta t$). The heat transfer in a chiller involves both a sensible heat transfer along with a phase change of the refrigerant; the Δt is not a simple linear function, it is a logarithmic function.

> ***Note***
>
> ***This section on overall heat transfer is included to establish a foundation on the factors involved in heat transfer so that the reader may appreciate the impact of LMTD and U on chiller efficiency and performance. These terms will be utilized in the next section.***

Heat Transfer Coefficient

The mean temperature difference (LMTD) between the refrigerant and the fluid in the tubes is the driving force that overcomes the resistance to heat transfer. The resistance to heat transfer is made up of four components, which are related to the overall heat transfer coefficient (U) by the following equation:

$$\frac{1}{U} = R_W + R_F + R_M + R_R$$

Where:

R_W = fluid film resistance

R_F = fouling deposits resistance

R_M = tube wall metal resistance

R_R = refrigerant film resistance

HEAT TRANSFER RESISTANCES BETWEEN FLUID IN THE TUBES AND THE REFRIGERANT

R_W	0.000469	46%	Fluid Film
R_F	0.000250	24%	Fouling
R_M	0.000029	3%	Tube Material
R_R	0.000277	27%	Refrigerant Film
R_T	0.001025	100%	Total

Figure 58

Typical Resistances to Heat Transfer- Baseline

Shown are the typical magnitudes for each component of the total resistance. The refrigerant film resistance R_R comes from manufacturer test data and is a function of the heat exchanger design, the properties of the refrigerant used, and the geometry of the outside tube surface.

The other three, R_F, R_W, and $R_{M,}$ are all variables (representing 75 to 80 percent of the total resistance) and are within the control of those applying or selecting the heat exchangers. The impact of these variables on the resultant refrigerant temperature (t_R) is discussed below.

Impact of Fouling Factor on (U)

Fouling is the build up of scale deposits on the inside surface of the heat exchanger tubes as minerals or suspended materials in the fluid precipitate out. Fouling begins as soon as the heat exchanger is placed in operation.

Shown in Figure 59 is the resultant impact on the overall resistance when fouling is allowed to build up in the tubes of a typical condenser. The impact of the R_F factor has increased from 24 percent at 0.00025 to 56 percent of the total resistance at 0.001 – a dramatic change making R_F the dominant factor in the resistance to heat flow.

R_W	0.000469	26%	Fluid Film
R_F	0.001000	56%	Fouling
R_M	0.000029	2%	Tube Material
R_R	0.000277	16%	Refrigerant Film
R_T	0.001775	100%	Total

Figure 59

Resistances with Increase in Fouling

With a rise in R_F the overall heat transfer coefficient U will decrease. Thus, the temperature driving force LMTD must increase in order for the heat exchanger to transfer the same amount of heat q. The result is an increase in the saturated condensing temperature.

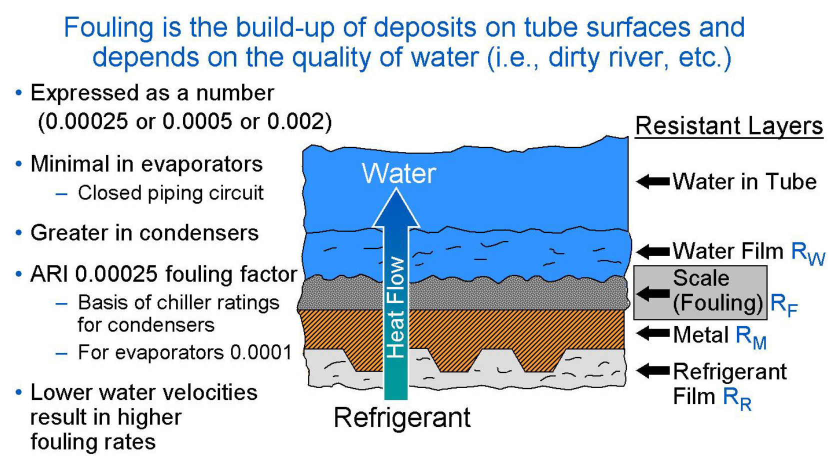


Figure 60

Fouling

The degree of fouling from buildup of scale deposits depends upon the quality of the fluid circulating though the tubes, and the amount and nature of any material it may have in suspension.

The application of the proper fouling factor in the selection of heat exchangers is important and must be evaluated for every job. Larger than standard heat exchangers may be required to accommodate higher fouling factors, which in effect is increasing the area of heat transfer. This will increase the first cost of the machine. If equipment is selected on the basis of small fouling factors, and actual conditions dictate otherwise, it will result in capacity deficiencies and might also affect surge characteristics of a centrifugal chiller.

ARI

(Air Conditioning and Refrigeration Institute) Standard 550/590 details methods for rating chillers and provides guidelines for comfort cooling applications.

Impact of Tube Velocity On (U)

Fluid Film

Fluid flowing through a tube forms a static film or boundary layer that has a zero velocity at the tube wall. This film acts as an insulator and hinders the transfer of heat from the tube wall. At lower fluid velocities, this film can become thicker, and a higher resistance to heat transfer will occur.

Comparing the initial parameters where the tube velocity was 10 fps, we now show the results when the tube velocity is 4 fps. The water resistance R_W increases from 46 percent to 64 percent of the overall resistance to heat transfer.

In addition, because the overall resistance R_T is now higher, both the LMTD and saturated condensing temperature will have to increase in order to transfer the same amount of heat at the lower 4 fps velocity.

R_W	0.000985	64%	Fluid Film (4 fps velocity)
R_F	0.000250	16%	
R_M	0.000029	2%	
R_R	0.000277	18%	
R_T	0.001541	100%	

Figure 61

Resistances with Lower Water Velocity in Tubes

Pass arrangements can be increased or decreased to raise or lower the tube velocity and to circulate more water. The velocity increases with higher passes because the area through which the water flows decreases. In selecting a heat exchanger, the tube velocity should generally be kept between 3 and 12 fps. The higher limit is arbitrary and is based on operation with reasonable pressure drops while simultaneously minimizing any possible tube erosion. Velocities less than 3 fps may result in laminar flow (thick boundary layer), which causes water resistance R_W to increase. Typical HVAC duty requires 1, 2, or 3 passes, based on application conditions such as pressure drop and required capacity. See the pass arrangement section in this TDP.

Tube Erosion

In order for erosion to occur, an agent must penetrate the fluid layer. Agents that cause the tube damage can be chemical, mechanical, or a combination of both. Chemical agents diffuse through the fluid film to attack the tube. Mechanical agents cause damage by the impingement of entrained gas bubbles or suspended material against the tube wall. Excess fluid velocity will accelerate erosion.

Fluid velocity

itself is not the sole cause of tube damage, however, if the fluid does contain attacking agents, increased fluid velocity will increase the rate of attack. A good water treatment program can help prevent premature tube erosion. For a discussion on water treatment, refer to TDP-641, Condensers and Cooling Towers.

Pressure Drop

Pressure drop increases as the square of the velocity. The higher the velocities, the higher the fluid pumping costs. A practical application approach is to specify limits and allow heat exchangers to be selected up to the 12 fps limit, as long as the pressure drop limits are met. This approach would assure reasonable operating costs, minimal erosion problems, and still allow low first cost heat exchangers to be selected.

Impact of Tube Material On (U)

Compared to Figure 59, which was based on copper tubes, Figure 62 shows the impact on the overall resistance to heat flow caused by the use of 70-30 cupro-nickel tubes. Adding CuNi tubes increases the impact of the metal resistance R_M from 3 percent to 26 percent of the overall resistance.

R_W	0.000469	35%	
R_F	0.000250	18%	
R_M	0.000350	26%	70 – 30 CuNi Tube Material
R_R	0.000277	21%	
R_T	0.001346	100%	

Figure 62

Resistances with Another Tube Material

The overall increase in resistance causes an increase in condensing temperature or a decrease in suction temperature, affecting unit capacity. Manufacturers can offer larger water-cooled screw and centrifugal chillers with a selection of tubing materials ranging from the commonly used copper to an exotic type like titanium. Metal resistances of tube materials vary greatly and can significantly impact machine size and power input requirements.

Application	Tube Material	Approximate Cost Multiplier vs. Copper Tubes
Fresh Water	Copper	1.0
Glycols	Copper	1.0
Corrosive Water	Cupro-Nickel	1.3
Special Process	Stainless Steel	2 to 3
Sea Water	Titanium or Cupro-Nickel	3 to 4

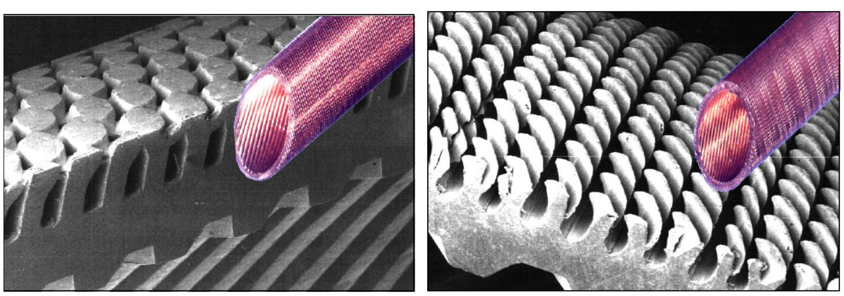
Figure 63

Tubing Materials Chart

The conditions of the fluid used in the application should determine the type of metals used for the tubing.

Evaporator and Condenser Tubing

Having concentrated on the impact of R_W, R_F, and R_M on the overall heat transfer coefficient (U), we will now consider the effects of internally and externally enhanced tubes on R_R and R_W.



Evaporator Tubing Condenser Tubing

Figure 64

Standard tubing – Internally and externally enhanced

The heat transfer coefficient of boiling or condensing refrigerant is several times less than that of water flowing through tubes. Therefore, the outer surfaces of the tubes are usually finned by rolling grooves in the tubes. The net result is to increase the heat transfer surface area. This extra area reduces R_R, and increases heat transfer rate. For that reason, it is standard practice in current large chiller technology to provide both internal and external enhancement to the evaporator and condenser tubes. Enhancements are made to the waterside to increase surface area and to provide a turbulence that reduces the chance of fouling be maintaining a scrubbing action.

Fouling tests indicate

that fouling rates of enhanced tubes is no greater than that of smooth tubes. In fact, test data shows that the R_F (fouling resistance) in the enhanced tube is less than that in the smooth tube.

Freezing of Fluids in Tubes

Whenever the saturated refrigerant temperature is below the freezing temperature of the fluid flowing through the tubes, fluid freeze-up is possible. Freeze-up is most likely to occur at the end of the last pass, because that is where the temperature of the fluid is at its coldest point. Therefore, for safety, the inside tube surface temperature at the end of the last pass should be no lower than 32.4º F when chilling water, and no lower than 1º F above the brine freezing temperature when chilling a brine.

Pass Arrangement

Pass arrangements are normally related to maximum allowable tube velocity or maximum allowable pressure drop requirements. A general rule of thumb is to use as high a pass arrangement as possible, since leaving temperature differences decrease with larger pass arrangements. This has a beneficial effect on compressor input power. For example, based on a 10º F rise in the evaporator, the LTD for a one-pass and two-pass arrangement would be 5º F and 2º F respectively. Using a one-pass arrangement would therefore require the compressor to operate at a saturated suction temperature approximately 3º F lower than would be possible with a two-pass.

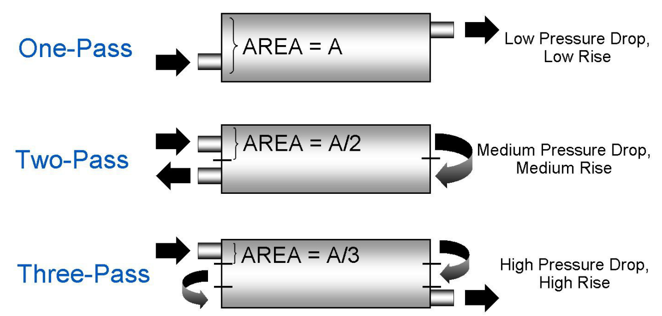


Figure 65

Pass Arrangements for Heat Exchangers with Water in the Tubes

Although higher pass arrangements tend to decrease compressor input power, this trend also increases pumping horsepower for the chilled and condensing water circuits. This is a result of increasing tube velocities caused by increasing the number of passes. Thus, an economic balance must be struck between higher pumping horsepower and lower compressor input power.

The most common method

of selecting of heat exchanger passes is to specify reasonable pressure drop limits and allow the manufacturer to recommend the optimal pass arrangement for the load and application under consideration.

Heat Exchanger Performance

It should be noted that centrifugal compressor performance and horsepower requirements will be directly impacted by the choices made in applying the heat exchangers. The compressor operates between the pressure levels determined by the saturated suction and saturated condensing temperatures. These temperatures are in turn directly related to the LTDs established for the cooler and condenser. The higher the LTD, the greater will be the compressor head and horsepower, and vice versa. High efficiency tubes can provide LTD approaches that are 1.5º F to 2º F.

Loading Rate

Each heat exchanger is designed by the manufacturer to transfer a nominal amount of heat. For a given heat exchanger, the LTD will decrease as the load reduces below the nominal design point. Thus, the LTD will decrease on a machine operating at part load causing the suction temperature to rise or drop and the condensing temperature to drop. Or, for a given load, using larger and more expensive heat exchangers initially, will require smaller LTD's and subsequently lower compressor horsepower. This technique is commonly used today to select higher performance (low kW/Ton) machines using standard heat exchangers and compressors.

Variable Flow Operation

Variable flow operation refers to varying the chilled water flow rate through the chiller evaporator based on the cooling load in order to minimize water-pumping costs. In effect, the chilled water flow rate is reduced when cooling loads are lower and increased when the loads are greater. The maximum gpm would tend to correspond to the gpm at design load.

Modern electronic chiller controls have allowed for variable evaporator flow applications because of the ability of the controls to respond to changing conditions. However, there are both minimum and maximum flow rate restrictions for water flow through the evaporator that apply. If the flow rate through the evaporator becomes too low, the temperature differential between the entering and leaving water becomes too large and may result in leaving water temperatures approaching freezing. Additionally, too low of a flow rate may result in a loss of turbulence that would significantly reduce the heat transfer. Conversely, flow rates that are too high result in higher pressure losses within the evaporator. Additionally, the rate of change of the chilled water flow is very important in variable flow applications. The recommendation is to use a maximum flow rate change of 10 percent of full flow per minute.

Consult the manufacturer's selection software for actual minimum gpm for variable flow systems.

In flooded shell-and-tube evaporators

a typical minimum flow is that which corresponds to 3 fps velocity in the tubes. The minimum evaporator flow is readily available from the chiller selection software.

Codes and Standards

ARI Testing Standards

The Air Conditioning and Refrigeration Institute (ARI) is a trade association for the industry that has established a chiller certification program with defined testing procedures and tolerances. They also certify and label a manufacturer's selection programs and equipment though random testing at calibrated test facilities. The ratings are based on standardized HVAC conditions, and provide an accurate measure of chiller performance.

The ARI testing standard 550/590-98, "Water-Chilling Packages Using the Vapor Compression Cycle," measures two power consumption ratings. The first is full load kW/ton, which measures a chiller's efficiency at full load based upon the rating point. However, ARI recognizes that on average a chiller operates at full load about 1 percent of its run time. Therefore, the second measures of performance are called Integrated Part Load Value (IPLV) and Nonstandard Part Load Value (NPLV) which are weighted kW/ton efficiency indicators that more closely indicate a chiller's overall efficiency than the full load value alone.

IPLV is commonly expressed as either an IkW/ton value or in unites of EER. There is a fixed relationship between IkW/ton and EER.

EER = 12(IkW/ton)

This relationship shows that EER increases as IkW/ton decreases, and vice versa. Therefore, a "better" IPLV is shown as a lower value when the units are IkW/ton, and, due to the relationship just described, a "better" IPLV is a higher value when the units are expressed in terms of EER.

The weightings derived by ARI are at 100, 75, 50, and 25 percent of full load. ARI weights the part load values more, recognizing that chillers operate more at these points. This weighting of part loads is also evaluated with tower relief water, which more accurately reflects the way chilled water systems are operated. IPLV is only appropriate for standard ARI conditions, and NPLV is used when the conditions are not identical to standard ARI conditions.

FOR IPLV CALCULATION

Percent Chiller Load	Entering Condenser Temp	Percent Weighting
100	85	1
75	75	42
50	65	45
25	65	12

Figure 66

ARI Weighting Factors

The standard ARI conditions for IPLV for water-cooled chillers are:

- Leaving chilled water: 44º F
- Chilled water flow rate: 2.4 gpm/ton
- Cooler fouling factor: 0.0001 ($ft^2 \cdot h \cdot$ ºF)/Btu
- Entering condenser water temp: 85º F
- Condenser water flow rate: 3.0 gpm per ton
- Condenser fouling factor: 0.00025 ($ft^2 \cdot h \cdot$ ºF)/Btu
- Entering condenser water temperature: 85º F at 100 percent load, 75º F at 75 percent load, 65º F at 50 percent and 25 percent load

IPLV is then calculated using the following formula:

$$IPLV = \frac{1}{\left(\frac{0.01}{A}\right) + \left(\frac{0.42}{B}\right) + \left(\frac{0.45}{C}\right) + \left(\frac{0.12}{D}\right)}$$

Where:

A = kW/ton at 100% load

B = kW/ton at 75% load

C = kW/ton at 50% load

D = kW/ton at 25% load

Relief

is defined as the natural reduction in entering condenser water temperature made possible by a reduction in ambient wet bulb temperature. For a complete discussion on this topic, refer to TDP-641, Condensers and Cooling Towers.

ARI also allows a tolerance to be applied to the run testing of chillers. These tolerances are applied to power consumption and capacity, and are based on the full load temperature difference across the evaporator. Tolerances increase as evaporator load decreases. These tolerances are used every time a chiller is tested in a manufacturer's ARI-certified test facility. Tolerances are based on aggregate tolerances on instrumentation, as well as engineering and manufacturing processes.

ASHRAE 90.1

The American Society of Heating, Refrigeration and Air Conditioning Engineers developed a standard, ASHRAE 90.1, to require a minimum level of efficiency in water-cooled chillers. This efficiency is based on the ARI 550/590 Standard, which uses 44º F leaving chilled water and 2.4 gpm/ton flow rate, and 85º F entering condenser water and 3 gpm/ton flow rate. As the design conditions move away from this accepted, baseline rating point, the efficiency levels change to account for the corresponding changes in lift conditions for the compressor. If the leaving chilled water temperature is reduced, the lift therefore increases, and the efficiency level decreases. When a lower entering condenser water temperature is used to rate the machine, the lift decreases and the efficiency level is set higher. If the condenser flow rate is lowered, lift increases and the efficiency level decreases.

WATER CHILLING PACKAGES-MINIMUM EFFICIENCY REQUIREMENTS

Equipment Type	Size Category	Min Effic in kW/TON***	Minimum Efficiency	Test Procedure
Water-Cooled, Electrically Operated, Positive Displacement (Reciprocating)	All Capacities	0.837 0.696 IPLV	4.20 COP 5.05 IPLV	ARI 550/590
Water-Cooled, Electrically Operated, Positive Displacement (Rotary Screw and Scroll)	< 150 tons	0.790 0.676 IPLV	4.45 COP 5.20 IPLV	
	≥150 tons and < 300 tons	0.718 0.628 IPLV	4.90 COP 5.60 IPLV	
	≥ 300 tons	0.639 0.572 IPLV	5.50 COP 6.15 IPLV	
Water-Cooled, Electrically Operated Centrifugal	< 150 tons	0.703 0.670 IPLV	5.00 COP 5.25 IPLV	
	≥ 150 tons and < 300 tons	0.634 0.596 IPLV	5.55 COP 5.90 IPLV	
	≥ 300 tons	0.576 0.549 IPLV	6.10 COP 6.40 IPLV	

*****kW/TON = 3.516/COP**

Figure 67

ASHRAE 90.1 Standards, Table 6.2.1C

UL/CSA & ETL

Several safety standards apply to chillers. There are also several widely accepted agencies such as Underwriters Laboratories, Inc., Canadian Standards Association, and ETL Testing Laboratories, which test for compliance to those standards. When a chiller is approved as meeting the required testing, it will bear a mark or label from the certifying agency.

UL (Underwriters' Laboratories, Inc.) is an independent nonprofit organization that tests products for safety and certifies them. The Canadian Standards Association (CSA) is a non-profit association serving business, industry, government, and consumers in Canada. Among many other activities, CSA develops standards that enhance public safety. For heating, ventilation, and air conditioning, UL/CSA Standard UL 1995 / CSA C22, Heating and Cooling Equipment applies.

ETL Testing Laboratories, like UL, conducts electrical performance and reliability testing. OSHA (Occupation Safety and Health Administration) recognizes ETL as a nationally Recognized Testing Laboratory as is Underwriters Laboratories. The ETL Listed Mark and Canadian-ETL Listed Mark are accepted throughout the United States and Canada compliance with nationally recognized standards such as ANSI (American National Standards Institute), UL, and CSA. This certification mark indicates that the product has been tested to and has met the minimum requirements of a widely recognized U.S. product safety standard, that the manufacturing site has been audited, and that the applicant has agreed to a program of periodic factory follow-up inspections to verify continued conformance. If the mark includes a small "US" and/or "C," it follows product safety standards of United States and/or Canada respectively.

ASHRAE Standard 15

ASHRAE Standard 15, "Safety Standard for Refrigeration Systems," has the primary goal of mitigating safety risks to the environment, to mechanical room operators, and ultimately to the public by incorporating specific design requirements for the safe installation and operation of mechanical refrigeration systems. Some of the design requirements of ASHRAE Standard 15 are:

- Location of refrigerant relief piping discharge
- Sizing of refrigerant pressure relief devices and piping per ASHRAE recommendation
- Restricting access to the mechanical room
- Providing sensors and a refrigerant monitor capable of detecting refrigerant leakage
- Installing a mechanical room ventilation system
- If the type of refrigerant in a chiller is changed, or if the chiller is replaced, then ASHRAE 15 will be applied.

Consulting engineers' specifications and drawings incorporate ASHRAE 15 requirements since it is has been adopted by building code authorities.

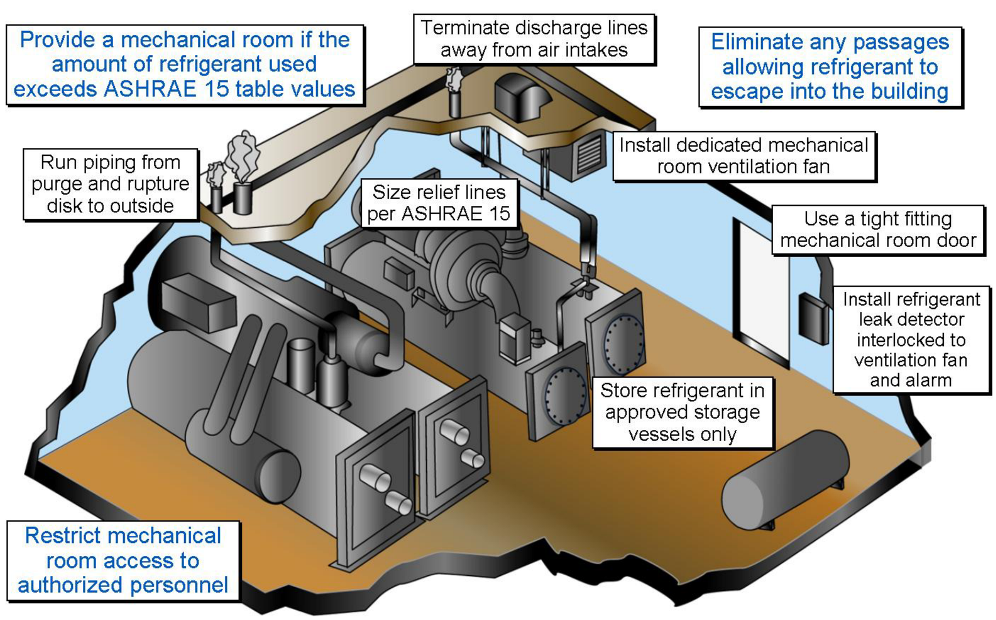


Figure 68

ASHRAE Standard 15 Requirements for a Typical Mechanical Room

Selection Criteria

Now that we have discussed all the items that go into chiller design, construction, and testing/certification, let us look at the parameters that are used to select a centrifugal chiller using a manufacturer's ARI-certified selection program.

The first two pieces of information needed are the required capacity and efficiency. This could be full load, IPLV, or ASHRAE 90.1 minimum efficiency requirements. Next, the stability requirements of the chiller are selected. For humid regions, typical stability is chosen at approximately 50 percent of full load at design entering condenser water, to safeguard against surge conditions.

Stability is a term that is used in evaluating the part load operating condition for a centrifugal chiller. If the head pressure during part load operation is higher than the chiller was selected for, the impeller may not be able to overcome the lift, and the chiller may enter an unstable operational condition causing the compressor to surge.

Centrifugal chillers are typically selected for full load and/or part load kW/ton targets. Then they are checked for part load stability using software provided by the chiller manufacturer. A typical part load stability check may involve running the chiller at part load points at entering condenser water temperatures that follow a relief profile representative of the project geography. For more information on the subject of cooling tower relief, refer to TDP-641, Condensers and Cooling Towers.

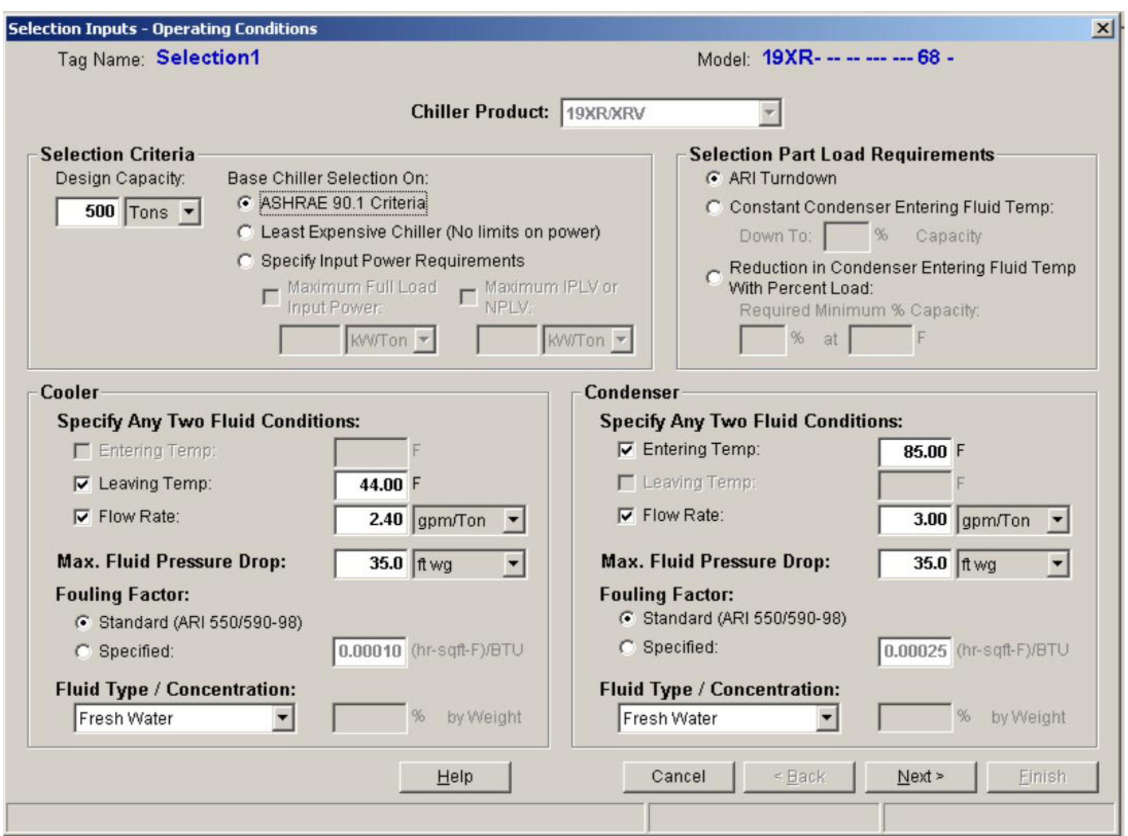


Figure 69

Centrifugal Chiller Input Selection Screen 1

After that, the chilled and condenser fluid conditions are entered. These can be based on the designer's requirements, or at typical ARI conditions. For both the chilled and condenser water, two of three inputs are required. For the chilled water, leaving chilled water temperature is typically set, and then either the flow rate or entering chilled water is entered. In the condenser, the entering condenser water temperature at design is entered, and then either the leaving condenser water temperature or the flow rate is entered.

Other parameters, such as the highest pressure drop allowed for the fluid, and the fouling factors, are also used here to help eventually determine the size of the heat exchangers and the size of the compressor.

The next set of selection criteria is the type of starting equipment and the electrical requirements. These are typically determined by the site and designer or owner preference. The same is true for the number of chiller passes, but this may also be dictated by the requirements of the chiller capacity and water conditions.

Options that affect performance, and therefore the final chiller selection, are entered here. Waterbox-type and waterside working pressure can be entered. Tubing material, wall thickness and tube enhancements can be chosen as well, based on the application and design requirements.

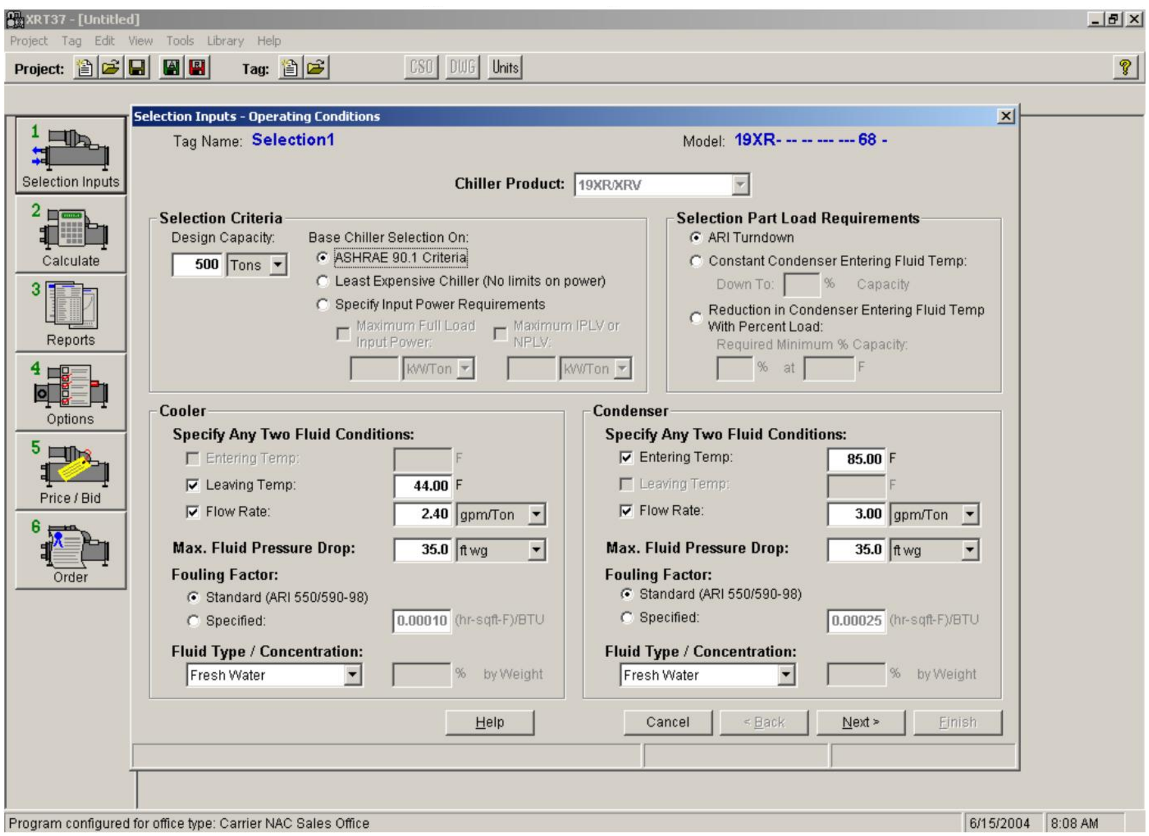


Figure 70

Centrifugal Chiller Input Selection Screen 2

It is best to allow the program to select the cooler, condenser, compressor, and motor for initial selection. The program will pick the best options based on all the other inputs entered thus far. However, these inputs may be entered manually at any time in the selection process. Once the selection program has chosen an initial design, the user can go back and modify these inputs to maximize the performance of the chiller. As an example, the compressor may be increased one impeller size, or the condenser may be increased to improve performance.

These are the main inputs that determine the selected components of a water-cooled chiller from a typical selection program. The format will vary among manufacturers, but the basic inputs remain the same.

Summary

The intent of this TDP was to describe the water-cooled refrigeration cycle, as well as the details of the components, operation, and regulations that determine the chiller's construction. From reading this TDP, the student should be able to understand the main components in the centrifugal refrigeration cycle, and be able to follow the path of water and refrigerant through the chiller. The different types of compressors were described, so the reader should be familiar with the compression theories and have the ability to describe the differences in the compressor operating ranges and chiller construction. The TDP then detailed the construction and uses of the components in a water-cooled refrigeration cycle, and the student should be able to detail the differences and advantages to the various options.

Various starter types and were covered, and the TDP demonstrated the differences in starting current and application for water-cooled chillers. The issue of refrigerants and the regulations governing its usage, safety and chiller construction impacts was covered to allow the reader to make informed decisions on what refrigerant should be used in a water-cooled chiller. The TDP defined different capacity control methods, and the affects on efficiency should be recognized. Heat transfer theory for heat exchangers provided the basic concepts for shell-and-tube heat transfer and the process used to determine the heat exchanger construction and performance. The TDP briefly outlined the different standards that relate to water-cooled chiller performance, construction, and installation. From this section, the student should now understand the agencies and organizations that rate and certify water-cooled chillers. Finally, the TDP covered the necessary selection inputs for a centrifugal chiller.

Work Session

1. List five major components of a large water-cooled chiller.

   ______________________________ ______________________________

   ______________________________ ______________________________

   ____

2. What are two cycles used to improve efficiency in water-cooled chillers?

   ____

   ____

3. List four types of compressors used in water-cooled chillers.

   ____

   ____

   ____

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   Which type is non-positive displacement? _____________________

4. List three types of evaporators used in water-cooled chillers.

   ____

   ____

   ____

5. Describe the two types of compressor motor designs used in centrifugal chillers.

   ____

   ____

6. List some attributes of each motor type described in Question 5.

   ____

   ____

7. List seven types of motor starters

8. Define head and lift.

9. What are two operation envelope limits for the centrifugal compressor?

10. What are three methods of capacity control?

11. Name three important effects refrigerant has on the construction of a chiller?

12. List four components of the heat transfer coefficient.

13. List three testing agencies that certify water-cooled chiller electrical code compliance.

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14. Name 4 specific requirements of ASHRAE 15

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15. List four reasons for the attraction of VFDs on centrifugal chillers.

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Notes

Appendix

References

API Heat Transfer, Buffalo, NY. http://www.apiheattransfer.com/

ARI Standard 550/590, "Water Chilling Packages Using the Vapor Compression Guide." http://www.ari.org/std

ASHRAE Standard 15, "Safety Standard for Refrigeration Systems"

ASHRAE Standard 34, "Designation and Safety Classification of Refrigerants"

ASHRAE Standard 90.1, "Energy Efficient Design of New Buildings Except Low-Rise Residential Buildings

Honeywell International, LLC, Morristown, NJ. http://www.honeywell.com/

Standard Refrigeration, Melrose Park, IL, http://www.stanref.com

Work Session Answers

1. evaporator, compressor-motor, starter, condenser, and pressure reducing device

2. subcooling, economizer

3. a. reciprocating, scroll, screw, centrifugal
   b. centrifugal

4. brazed-plate, D-X shell-and-tube, flooded shell-and-tube

5. hermetic motors, which are sealed to the atmosphere and utilize refrigerant cooling

   open-drive motors, which are coupled to the compressor through a shaft seal and exposed to the mechanical room atmosphere.

6. Hermetic motors are smaller, run cooler, and have lower inrush current. Open drive motors offer flexibility of motor size and types of drivers that can be applied to the compressor.

7. across the line, auto-transformer, primary reactor, part winding, wye-delta, solid state, variable frequency drives

8. Head is the measure of the pressure rise developed in the compressor, and lift is the difference between the saturated suction and condensing temperatures.

9. stonewall, the maximum compression or refrigeration flow

   capacity, and surge, a condition created when condensing pressure is greater than the pressure generated by the compressor

10. refrigerant flow control, hot gas bypass, and speed control combined with flow control

11. design working pressure and construction certification, physical size of chiller, and ability to store refrigerant in chiller

12. refrigerant film resistance, tube wall metal resistance, fouling deposits resistance, fluid film resistance

13. UL, CSA, ETL

14. ASHRAE 15 will help you determine if you need a separate mechanical room. If you do, then you should provide a tight fitting access door, a refrigerant monitor, an alarm system to notify operators if there is a refrigerant leak, a mechanical room ventilation fan, and properly sized relief lines. There are other requirements in addition to these listed in the standard.

15. high power factor, low inrush current, increased part load efficiency, low sound levels at part load

Prerequisites:

This module assumes the participant has an understanding of industry terminology, basic concepts of the air conditioning, and the mechanical refrigeration process. The following TDPs are good reference for this material:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-400	796-037	797-037	Principles of Mechanical Refrigeration, Level 1: Introduction
TDP-401	796-084	797-084	Principles of Mechanical Refrigeration, Level 2: Analysis
TDP-301	796-034	797-034	Load Estimating, Level 2: Fundamentals

Learning Objectives:

After reading this module, participants will be able to:

- Compare the advantages of water-cooled versus air-cooled chillers.
- Identify and diagram the different components of a basic refrigeration cycle as it applies to as water-cooled chiller.
- Compare and describe the differences among scroll, reciprocating, centrifugal, and screw compressors.
- Discuss the differences in construction of water-cooled chillers of various sizes.
- Identify the standards that relate to water-cooled chillers.
- Understand the typical inputs required to select a water-cooled chiller.

Supplemental Material:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-622	796-054	797-054	Air-Cooled Chillers
TDP-705	796-070	797-070	Chilled Water Systems

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.

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