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Chiller Performance Monitoring and Troubleshooting Guideline

Technical Report

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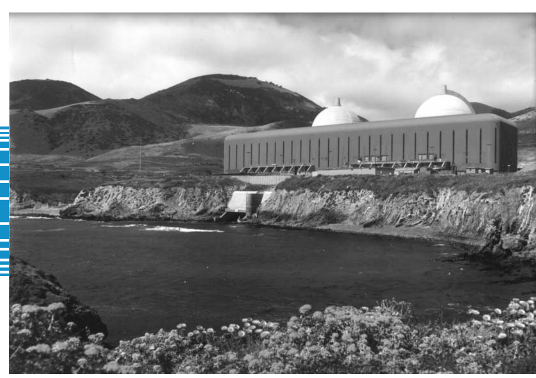

NMAC

Chiller Performance Monitoring and Troubleshooting Guideline

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REPORT SUMMARY

Background

Refrigerant systems, often referred to as chillers, serve a vital function in nuclear power plants. They provide a means of removing unwanted heat from locations within the plant during normal operation and in emergencies. To ensure that these components are operating properly, plant personnel must be able to effectively and accurately monitor the performance of this equipment. When problems are encountered, the ability to quickly diagnose and correct them to ensure continued plant operation and availability is extremely important.

Objectives

- To provide methods of monitoring the thermal performance of certain chiller units, specifically, low-, medium-, and high-pressure chillers, direct expansion chillers, and direct expansion air conditioning units
- To provide the means for troubleshooting thermal performance problems of these chillers
- To provide general preventive and predictive maintenance recommendations if thermal performance is deemed unacceptable

Approach

In cooperation with the Nuclear Heating, Ventilation, and Air Conditioning Utility Group and interested Nuclear Maintenance Applications Center members, a task group of utility engineers and industry experts was formed. They met several times over a year and a half to generate this guideline by identifying and preparing the guidance set forth in this document. The task group identified and discussed experience-proven practices and techniques, which are collected and summarized in this guideline for use by all power plant personnel.

Results

This report provides a practical approach that can be used by power plant personnel to monitor chiller performance and to diagnose and troubleshoot chiller systems and component performance problems. In support of these activities, the guideline provides for plant maintenance and engineering personnel a basic understanding of refrigeration fundamentals and a solid understanding of the thermal performance aspects of refrigeration equipment. Preventive and predictive maintenance practices are identified to help avoid unexpected component failures or degradation.

EPRI Perspective

This guideline represents a significant collection of technical information on chiller performance monitoring and troubleshooting. Assemblage of this information provides a single reference for

power plant personnel. The intended audience of this guideline includes component, maintenance, and system engineers involved in chiller monitoring, testing, troubleshooting, and maintenance.

Keywords
Design engineers
Plant support engineering
Plant maintenance
Plant operations
Chillers
Refrigerant systems

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1 INTRODUCTION

1.1 Purpose

This guideline provides the following information to plant maintenance and operations personnel:

- Methods to monitor the thermal performance of certain chiller units. Specifically, the guidance focuses on low-, medium-, and high-pressure chillers, direct expansion (DX) chillers, and DX air conditioning (A/C) units.
- Means for troubleshooting thermal performance problems of these chillers.
- General preventive and predictive maintenance recommendations if thermal performance is deemed unacceptable.

The guideline provides for plant maintenance and operations personnel a basic understanding of refrigeration fundamentals and a solid understanding of the thermal performance aspects of refrigeration equipment.

1.2 Scope

This guideline applies specifically to all pressure ranges, as defined by operating pressures of the refrigerant type, of refrigerant systems (low, medium, and high) and DX chillers and A/C units. The main goals of any licensee should be to ensure the performance of the refrigerant system and that the A/C unit is maintained as near to design conditions as practical. Even though design conditions vary by plant, this guideline provides a success path for each user to properly assess system performance. It is not the intent of this guideline to address adsorption-type equipment.

The refrigerant systems for nuclear power plants consist of several components that must function together as a dynamic system to ensure safe, reliable, and dependable operation. This guideline provides an overview of the requirements to develop and monitor the thermal performance of several types of refrigeration systems.

Section 2 provides terminology and definitions, acronyms, and conversion factors used in this guideline.

Section 3 contains fundamental information about refrigerant systems in addition to the terminology, theories, operational principles, and components that typically comprise these systems. Plant personnel must have a thorough understanding of the components that comprise these systems before knowledge of how the system functions as a whole.

Section 4 provides parameters used to monitor refrigerant system performance. This includes monitoring critical parameters to provide an assessment of continued safe, reliable operation. In addition to parameter monitoring and trending, testing can provide quantitative results of system performance. Many of these systems are subject to variations in performance that frequently function differently than the design of the original equipment manufacturer (OEM). Some of these variations in performance are the result of intrusive changes (for example, new tubes in the condenser), and some are caused by gradual component changes (such as component wear and instrument drift). Performance monitoring is crucial to ensure the maintenance of optimum system performance.

Refrigerant systems found to have unacceptable key parameters or test data cannot be relied on to operate as required. Section 5 addresses troubleshooting techniques that assist personnel in the restoration of these systems. This section provides the end user with techniques to develop an action plan to ensure the timely restoration of the refrigerant system to a satisfactory performance level.

Section 6 contains preventive maintenance guidance to maximize reliability and avert unplanned unavailability of refrigerant systems.

Section 7 lists the references used in the development of this guideline.

Appendix A provides key engineering fundamentals of refrigerant system analysis. These fundamentals may be most beneficial to personnel assigned to these areas of responsibility who have not been exposed to the field of thermodynamics and heat transfer or who might need a refresher of these concepts.

Appendix B summarizes the various types of portable and installed instrumentation and data acquisition equipment to assist personnel in the assessment of refrigerant system performance.

Appendix C summarizes applicable operating experiences.

Appendix D provides a description of testing techniques that provide quantitative assessments of the refrigerant system performance.

Appendix E lists all of the key points in the guideline, arranged by category, and includes the location in the guideline for each key point. This appendix can help plant personnel to determine if they have taken advantage of the key information in this guideline.

1.3 Key Points

Throughout this report, vital information is summarized in *key points*, defined as bold-lettered boxes that succinctly restate information covered in detail in the surrounding text, making it easier to locate.

By emphasizing vital information, key points enable personnel to take action for the benefit of their plant. The information included in these key points was selected by Nuclear Maintenance

Applications Center (NMAC) personnel, consultants, and utility personnel who prepared and reviewed this report.

The three categories of key points are operations and maintenance (O&M) cost, technical, and human performance. Each category has an identifying icon, as shown below, that draws attention to it, making it easy for personnel to quickly locate vital information.

	Key O&M Cost Point **Emphasizes information that will reduce purchase, operating, or maintenance costs.**

	Key Technical Point **Targets information that will lead to improved equipment reliability.**

	Key Human Performance Point **Denotes information that requires personnel action or consideration in order to prevent injury or damage or to ease completion of the task.**

Appendix E contains a listing of all of the key points in each category. The listing restates each key point and provides reference to its location in the body of the report. By reviewing this listing, users of this guideline can determine if they have taken advantage of key information that can benefit their plants.

2
GLOSSARY OF TERMS AND ACRONYMS

2.1 Terminology and Key Definitions

Air conditioning unit	An assembly of refrigeration equipment for the treatment of air to simultaneously control its temperature, humidity, cleanliness, and distribution to meet the comfort requirements of a conditioned space.
Approach temperature	The difference between the saturated refrigerant temperature and the leaving chilled water temperature (for the evaporator), or the difference between the saturated refrigerant temperature and the condenser water temperature (for the condenser).
Chilled water system	A closed-loop system that circulates water cooled by a chiller to provide cooling to various components or areas via heat exchangers.
Chiller	An assembly of refrigeration equipment designed to cool a liquid to meet the heat rejection requirements to a variety of components or areas via heat exchangers.
Coefficient of performance	The system output (the amount of heat removed) divided by the amount of required energy input to operate the cycle.
Cycle	A combination of all processes in which the initial and final states of the system are identical.
Direct expansion evaporator	An evaporator is classified as direct expansion when the refrigerant is allowed to change state within the tubes of the evaporator to provide cooling to the cooled media.
Efficiency	A ratio of the actual amount of heat transferred to the total amount of heat transfer possible.
Flooded evaporator	An evaporator is classified as flooded when the refrigerant is in the shell and the liquid to be cooled is in the tubes.
Heat sink	The medium that absorbs the rejected heat from a refrigerant system.
Heat source	The medium from which the heat is removed.

Isenthalpic	A process that takes place at constant enthalpy.
Isentropic	A process that takes place at constant entropy.
Isobaric	A process that takes place at a constant pressure.
Isothermal	A process that takes place at a constant temperature.
Latent heat	The heat added to or taken from a substance to change its state but not its temperature or pressure.
Low-, medium-, and high-pressure chillers	These are determined by the type of refrigerant used in the system.
Monitoring	A technique used to evaluate the performance of equipment in a qualitative method.
Process	A change in state that can be defined as any change in the properties of a system.
Receiver	An optional internal refrigerant system storage container located between the condenser and the metering device.
Refrigerant	A refrigerant is any substance that can readily absorb heat, usually resulting in a change of state, which can readily dissipate heat again, usually resulting in a second change in state.
Refrigerant system	A system of components specifically designed to circulate a refrigerant throughout a system to transfer heat from the heat source to the heat sink.
Resistance	This is the opposition to the flow of electrical current. The ohm (Ω) is the unit of measure for resistance.
Reversible cycle	The state of working fluid and system's surroundings can be restored to the original parameters or properties.
Sensible heat	The heat that produces a temperature change, but does not induce a change of state. For example, when water is heated from 60 to 90°F, there is a 30°F change in temperature, but the state of the water remains the same, that is, in liquid form.
State	The state of a system is defined by a listing of its properties.

Subcooling	The difference between the temperature of a pure condensable fluid below saturation and the temperature at the liquid saturated state, at the same pressure.
Superheat	The difference between the temperature of a pure condensable fluid above saturation and the temperature at the dry saturated state, at the same pressure.
Ton of refrigeration	A measure of heat flow equal to 12,000 Btu/hr derived from absorbing 288,000 Btu in the process of melting 1 ton (2000 lbs) of ice in a 24-hour period.
Total heat	As applied to refrigerant systems, the sum of the sensible heat and the latent heat of a mixture of air and moisture. For example, in cooling air over a coil, the weight of air-cooled times the range is the measure of sensible heat; and if moisture collects on the coil, this is the result of actual latent heat extraction of moisture from the air. Total heat is also known as enthalpy.
Wet bulb temperature	The temperature at which water vapor changes state from a gas to a liquid.

2.2 Acronyms Used in This Report

A/C	Air conditioning
AEV	Automatic expansion valve
ANSI	American Nuclear Standards Institute
ARI	Air-Conditioning Refrigeration Institute
ASHRAE	American Society of Heating, Refrigeration, and Air Conditioning Engineers
ASME	American Society of Mechanical Engineers
CFM	Cubic feet per minute
CFR	Code of Federal Regulations
COP	Coefficient of performance
CR	Control room
CREACS	Control room emergency air conditioning system

CRZ	Control room zone
CSW	Chilled service water
D/P	Differential pressure
DAS	Data acquisition system
DMM	Digital multi-meter
DX	Direct expansion
EPA	Environmental Protection Agency
EPRI	Electric Power Research Institute
FPM	Feet per minute
GPM	Gallons per minute
HP	High pressure
HVAC	Heating, ventilating, and air conditioning
IEEE	Institute of Electrical and Electronics Engineers, Inc.
LCO	Limiting condition for operations
LED	Light-emitting diode
LP	Low pressure
NDE	Nondestructive examination
NMAC	Nuclear Maintenance Applications Center
NRC	Nuclear Regulatory Commission
O&M	Operations and maintenance
OEM	Original equipment manufacturer
OSHA	Occupational Safety and Health Administration
P-h	Pressure-enthalpy
PM	Preventive or predictive maintenance

PMT	Postmaintenance test
RTD	Resistance temperature detector
RTF	Run-to-failure
SOV	Solenoid-operated valve
SR	Safety-related
SW	Service water
TCV	Temperature control valve
TEV	Thermal expansion valve
TS	Technical specification
TXV	Thermostatic expansion valve
WC	Chilled water

2.3 Conversion Factors for Units Used in This Report

SCFM: Standard cubic feet per minute

Note: SCFM is standard cubic feet per minute at 60°F (16°C) at 14.7 psia (101 kPa). Because not all countries convert SCFM to SI units in the same way, these units are not converted to SI units in this report.

1 gallon = 3.78 liters

1 inch = 2.54 cm

1 Btu = 1055 joules

1 lb = 0.45 kg

1°F = (°C x 9/5) + 32

1 gpm = 0.063 liters/second

1 psi = 6.89 kP

1 inch w.g. = 250 pascals

1 ft^3 = 28.3 liters

1 ft^2 = 929 cm^2

1 fpm = 0.51 cm/sec

3 OVERVIEW OF REFRIGERANT SYSTEMS

Plant personnel must have a basic understanding of refrigerant systems in order to properly monitor chiller performance or to troubleshoot problems that can arise.

This section describes the basic components of typical mechanical refrigerant systems. Examples of these systems as a combination of various components follow. Appendix A describes key engineering fundamentals, and both refrigerant processes and cycles are described in more detail.

Mechanical refrigerant systems are primarily identified by the condensing medium in use, that is, water-cooled or air-cooled. Additionally, they are classified by the method in which the system evaporator removes the heat from the heat source (DX coil or shell and tube heat exchanger). Further identification breakdown can be accomplished by specifying the construction of the compressor or the metering device or both.

3.1 Refrigerant System Fundamentals

Refrigerant systems provide cooling to equipment, processes, and personnel spaces. The systems do this with refrigerants that remove heat from the heat source and discharge it to a heat sink. The basic refrigerant system consists of five primary components: a compressor, a condenser, a metering/expansion device, an evaporator, and the connecting piping. Figure 3-1 displays a simplified mechanical refrigeration system.

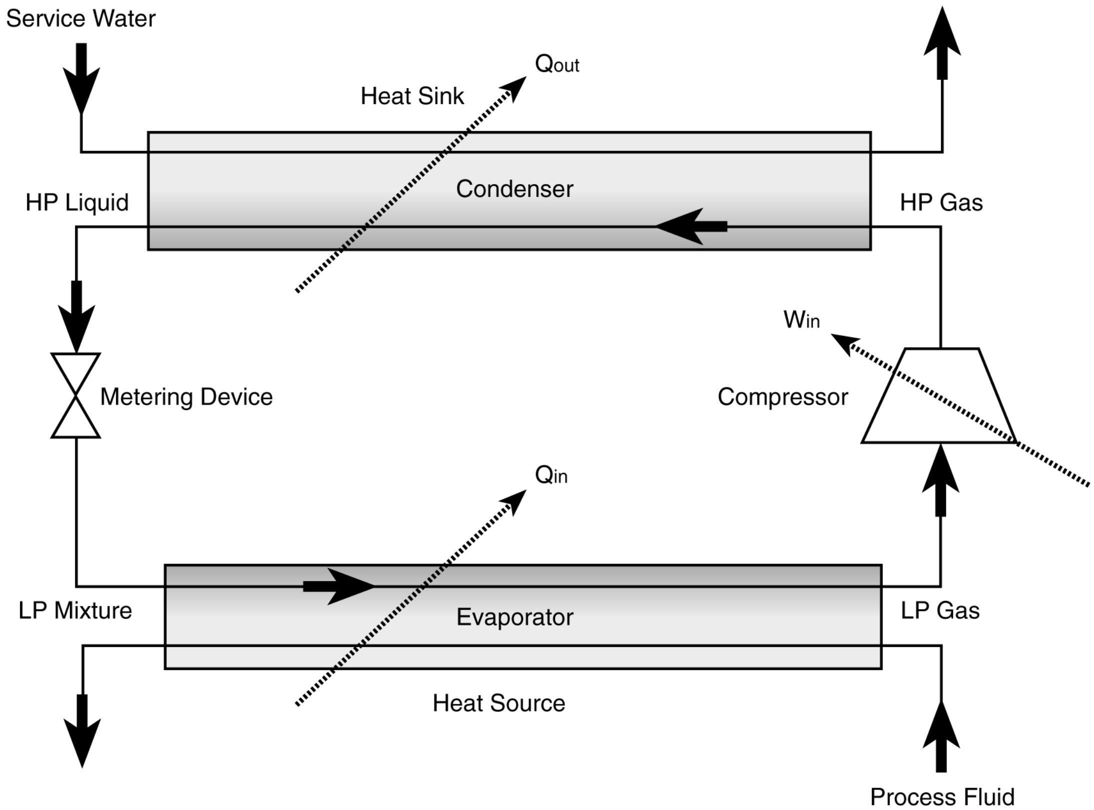


Figure 3-1
Basic Refrigerant Cycle

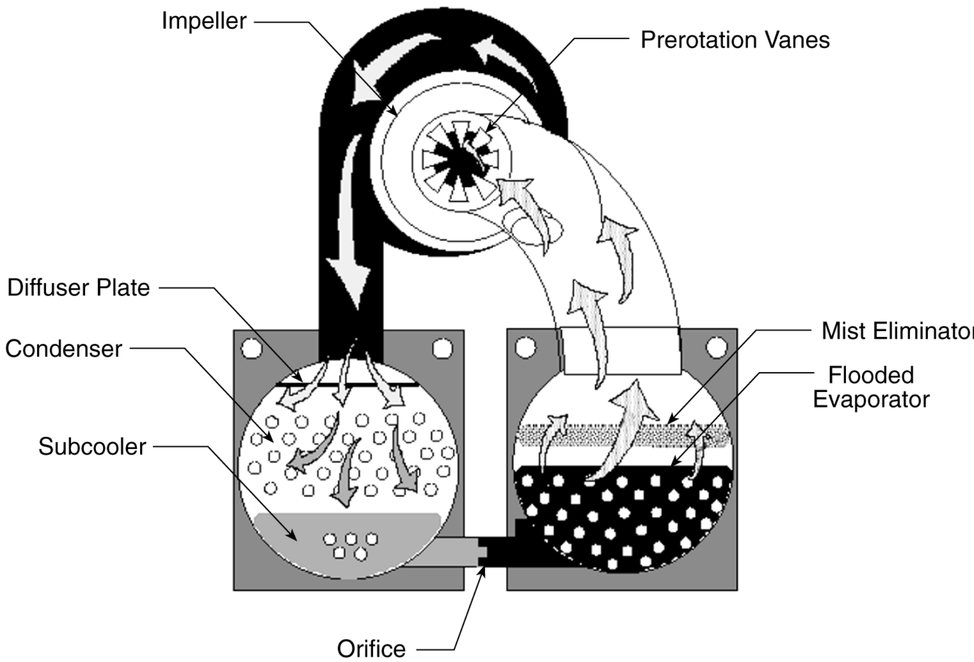
The compressor draws suction on the refrigerant gas exiting the evaporator. The compressor creates a pressure difference between the evaporator and the condenser and changes the refrigerant from a low-temperature, low-pressure (LP) gas to a high-temperature, high-pressure (HP) gas (temperature and pressure increase from the compression).

The high-temperature, HP gas moves to the condenser where it transfers its heat to the environment (air or water) and the gas is changed to a low-temperature, HP liquid.

The liquid refrigerant flows to the metering device/expansion valve that separates the HP portion of the system from the LP portion. The metering device/expansion valve acts as a reducing valve to lower the refrigerant pressure as it passes through the valve. (The refrigerant must be delivered to the evaporator in a liquid state to permit it to be vaporized at the desired temperature by reducing the pressure.) The refrigerant is vaporized by the heat flowing across the evaporator, resulting in a low-temperature, LP gas. The low-temperature, LP gas is returned to the compressor, thus completing the cycle.

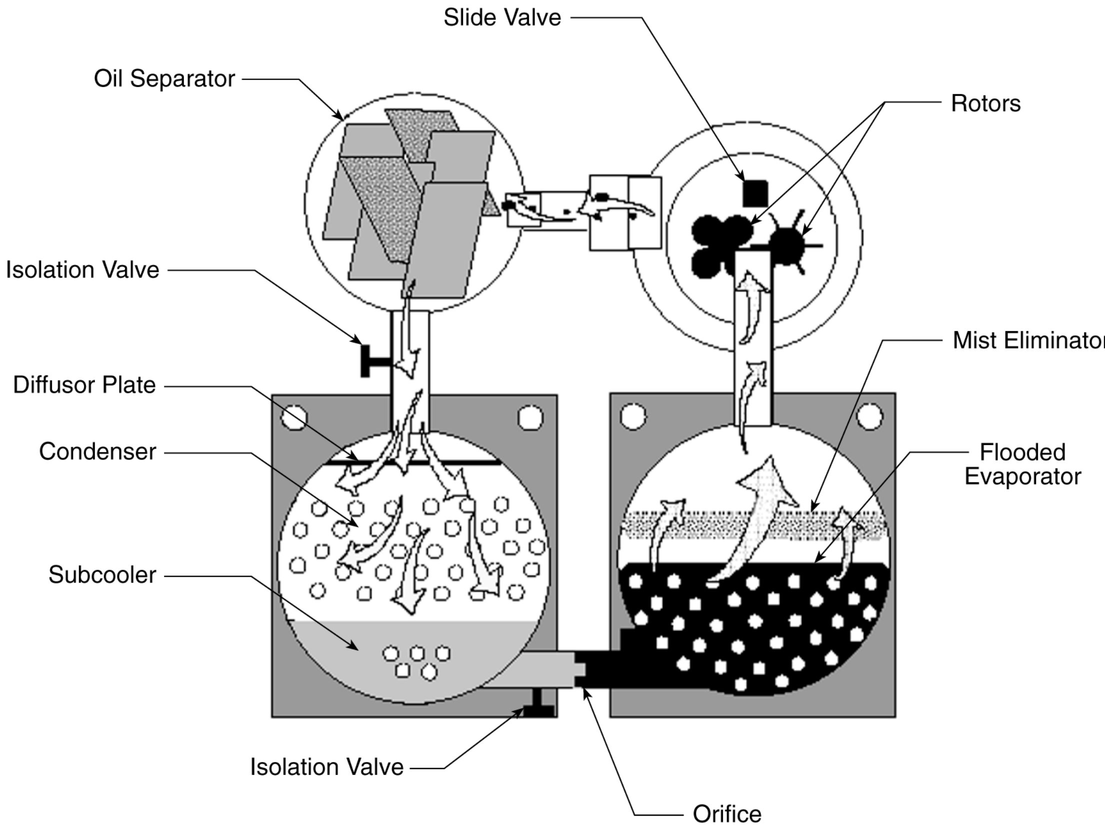
Typical refrigerant systems use three loops to perform their function: an evaporator loop, a refrigerant loop, and a condenser loop. As shown in Figure 3-1, the evaporator loop provides the heat source to the refrigerant system via the evaporator. The refrigerant loop was previously described and functions to transfer the heat from the evaporator to the condenser. The condensing loop accepts the heat from the refrigerant loop via the condenser and transfers it to the heat sink (condensing water or air).

The refrigerant cycle can be accomplished in different ways. Figure 3-2 shows a typical chiller with a centrifugal compressor. Figure 3-3 shows a typical chiller with a screw compressor. Figure 3-4 shows a typical DX chiller.



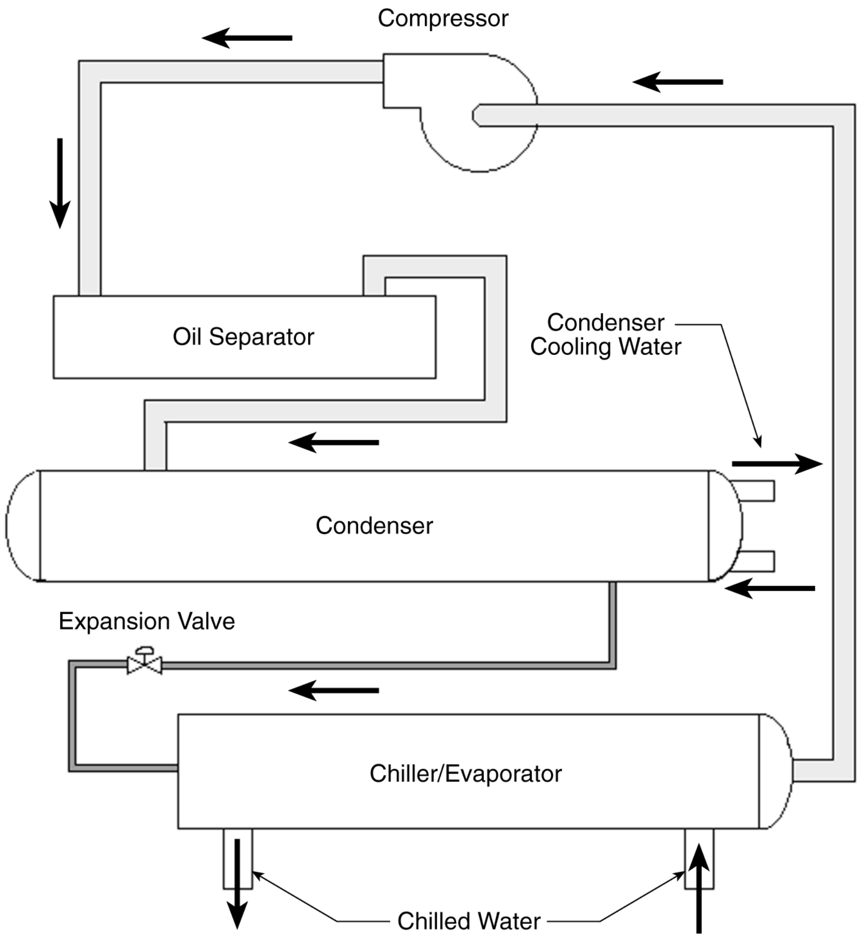
Courtesy of York

Figure 3-2
Typical Chiller with a Centrifugal Compressor



Courtesy of York

Figure 3-3
Typical Chiller with a Screw Compressor



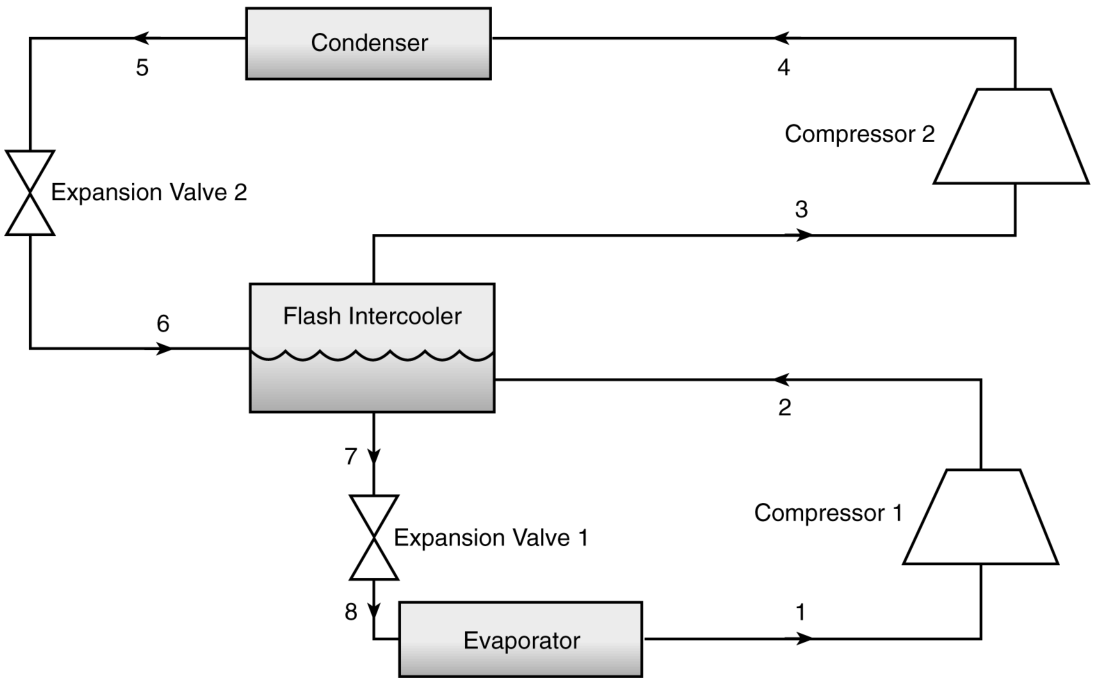
Direct Expanding (DX) Chiller

Liquid refrigerant inventory maintained in condenser shell side and metered through an expansion valve into the tube side of the evaporator.

Courtesy of Dunham-Bush

Figure 3-4
Typical Direct Expanding Chiller

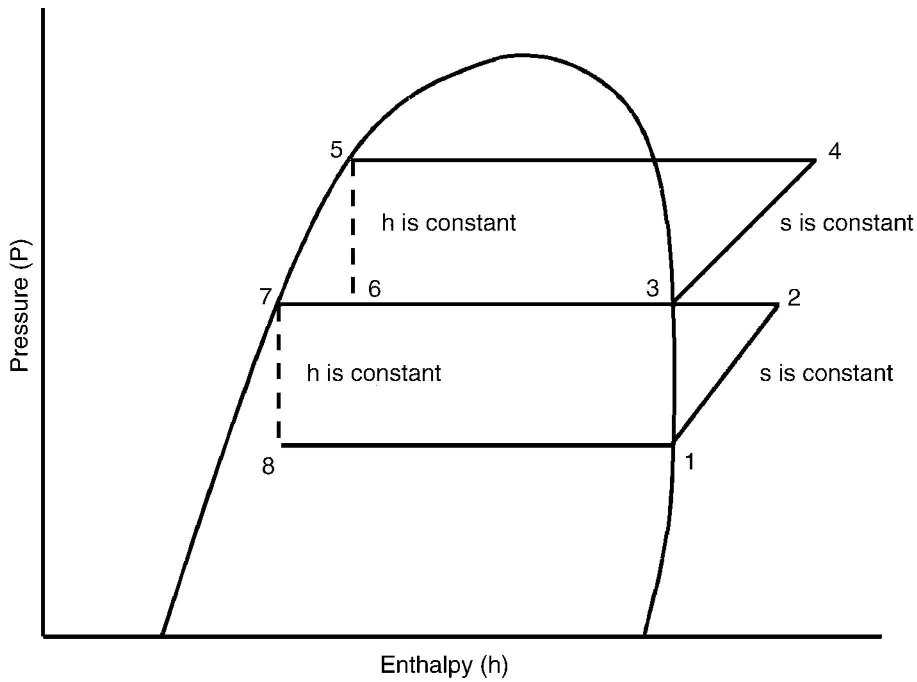
Some manufacturers equip their chillers with a flash economizer (intercooler) and two-stage centrifugal compressor. The flash economizer is located between the return side of the condenser and the evaporator, and it ultimately increases the unit's overall efficiency. Figure 3-5 shows a typical chiller with a flash economizer and two-stage centrifugal compressor.



Courtesy of ASHRAE

Figure 3-5
Typical Chiller with a Flash Economizer and Two-Stage Centrifugal Compressor

Refrigerant returning from the condenser to the evaporator enters the flash economizer and is directed into a chamber that is connected to the inner stage of the compressor. This lower pressure results in a portion of the refrigerant flashing to gas and cooling the remaining refrigerant. The flashed gas is returned to the second stage of the compressor where it mixes with the gas already compressed by the first compressor stage. The cooled refrigerant is then metered and enters the evaporator. The use of the flash economizer is depicted on the pressure-enthalpy (P-h) diagram in Figure 3-6.



Courtesy of ASHRAE

Figure 3-6
Pressure-Enthalpy Chart for a Chiller with a Flash Economizer and Two-Stage Centrifugal Compressor

This provides savings in operating cost and it increases cycle efficiency. Only a percentage of the total refrigerant flow is pumped through the first stage of the compressor while the refrigerant entering the evaporator has been additionally cooled from the flashing in the economizer.

3.2 Major Refrigerant System Components

3.2.1 Compressor

3.2.1.1 Function

A compressor provides the pressure increase necessary to facilitate the transfer of heat. It functions as the system driver by taking the LP gaseous refrigerant from the evaporator and increasing the pressure before delivering it to the condenser. No matter how complex the compressor, the purpose is simply to compress refrigerant gas.

3.2.1.2 Types and Configurations

Compressors are manufactured in different types of compression devices and housing configurations as well as in varying sizes. Types of compression devices include:

- Reciprocating piston
- Orbiting scroll
- Rotary vane
- Centrifugal
- Helical rotary screw

Configurations include hermetic, semi-hermetic, and open-drive.

Several methods transfer the power to the compressor. An electric motor, steam or gas turbine, or other prime mover can transfer power to the compressor shaft, either directly or indirectly. The motor can be linked directly to the compressor shaft via a coupling. The motor can be mounted inside the compressor so that the motor shaft is used as the compressor shaft.

Compressors are designed based on cooling requirements. As such, a compressor is often the single most expensive and mechanically complex component in a system.

The most common compressor is the reciprocating compressor, which acts as a positive displacement pump. It has a piston(s) within which it provides a relatively constant volume flow rate of refrigerant over a wide range of pressure ratios.

The screw compressor has two rotors with mating lobes. As the two rotors turn, the space between the lobes decreases. The confinement of the gas causes compression that increases the gas pressure.

The orbiting scroll compressor has two identical spirals in which one side is the involute of the other. One spiral is fixed and the other, an orbiting spiral, is rotated through 180 degrees relative to the first, so that they touch at a number of points, separating the rotating chambers. As the spiral orbits, the chambers move toward the center and become smaller, thus compressing the refrigerant gas taken in at the outside.

A centrifugal compressor has one or more rotating impellers that spin the refrigerant vapor in a specially designed housing. The refrigerant vapor is forced against the housing walls using centrifugal force, and the velocity energy gained is converted to pressure.

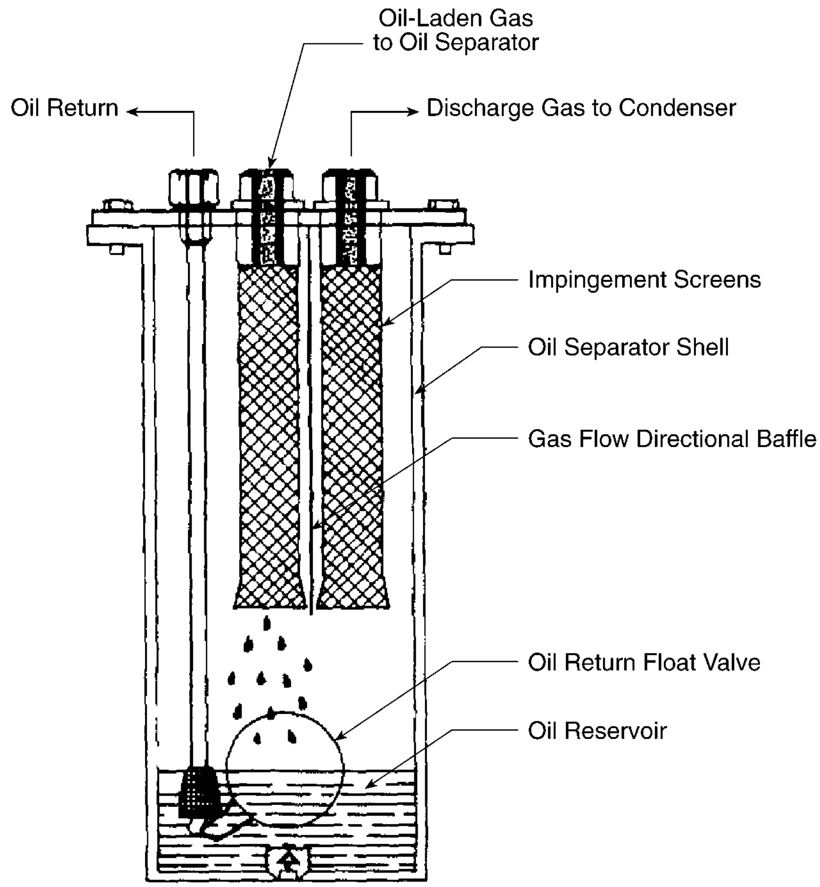
Rotary compressors, the least common type of compressors used, are divided into two categories: rolling piston and rotary vane.

3.2.1.3 Compressor Lubrication Systems

The two primary methods of ensuring compressor lubrication are the splash system and the pressurized system. In a splash system, “flingers” are attached to the rotating crankshaft. As the crankshaft turns, oil is picked up by these flingers and thrown onto bearing surfaces, pistons, and shafts. The preferred and predominant method is the pressurized system. In this system, an oil pump, often attached to the end of the crankshaft, maintains the oil flow through channels that keep a steady flow of oil on all surfaces that require lubrication. In a reciprocating compressor, a piston slides in a cylinder. A seal must exist between the side of the piston and the wall of the cylinder. This seal is created by oil in the refrigerant suction gas as it travels through the compressor crankcase. Without this seal, the cylinder is not able to reach maximum compression, which results in a loss of efficiency. A pressurized system allows oil to continuously maintain a coating on the cylinder walls from the crankcase underneath the piston. A portion of the oil travels as a mist with the refrigerant through the system. Because some of the oil travels with the refrigerant, sufficient refrigerant gas velocity must be maintained in the system to prevent oil from collecting in other parts of the system and to maintain oil flow back to the compressor. This factor is taken into account by design engineers and/or manufacturers so that subsequent provisions are made during pipe sizing and configuration layout.

In the refrigeration circuit, oil and refrigerant are in all parts of the system. Oil does not chemically mix with refrigerant but does travel with it. Oil is less dense than refrigerant and it will float on the liquid refrigerant. If liquid refrigerant enters the compressor, the refrigerant will wash out the oil and the oil will then enter the compression chamber. Positive displacement, reciprocating, rotary, and screw compressors cannot compress liquids. The phenomenon of liquid refrigerant entering a compressor is called *flood back*. The phenomenon of liquid oil (or refrigerant, for that matter) entering the compression chamber is called *slugging*. In both cases, irreparable damage can be done to a compressor. In the case of oil washout, the bearing surfaces can be severely damaged. In the case of slugging, connecting rods, pistons, and/or valves can be broken. In any event, the compressor can be destroyed internally very quickly. The damage would occur before there was an opportunity to shut the unit down.

Oil that is in the system but outside of the compressor serves no lubricating purpose. Therefore, in larger systems, the refrigerant and oil mixture is not allowed to progress far from the compressor. To limit the oil flow to the remainder of the system, an oil separator is installed. Figure 3-7 illustrates a typical oil separator.



Courtesy of ASHRAE

Figure 3-7
Oil Separator

When enough oil has entered the separator, the buoyancy of the oil raises the float and the valve opens. Because the discharge refrigerant pressure is greater than the crankcase pressure, the greater pressure forces the oil back to the compressor crankcase. When the oil level drops, the float closes the valve. Separators are often insulated and heated to keep them warm, which ensures that excess refrigerant is not mixed with the oil. Manufacturers typically provide guidelines on which type of oil to use.

3.2.2 Condensers: Air Cooled/Water Cooled

The condenser receives the high-temperature, HP gas from the compressor and reduces the temperature to provide a saturated or subcooled HP liquid to the metering/expansion device. The force that drives the refrigerant through the condenser is the differential pressure created by the compressor.

Water-cooled condensers are usually constructed in one of the following configurations:

- Double pipe (pipe-in-pipe)
- Double tube (tube-in-tube)
- Shell and coil
- Shell and tube

For water-cooled condensers, heat is transferred from the condenser to the water and is carried by the water to an acceptable heat sink.

Air-cooled condensers are usually constructed of a number of tubes covered with fins. Air is directed over these tubes either by an external fan or as a result of thermal conditions. The purpose of the fins is to increase the surface area of the condenser and thus increase the heat transfer. Heat is then rejected to the surrounding environment. After enough heat is rejected, the refrigerant vapor changes from a gas to a liquid. At the outlet of the condenser, the refrigerant is a saturated or subcooled liquid. Some designs include additional surface area to permit subcooling. This prevents refrigerant vapor from being introduced to the metering/expansion devices.

3.2.3 Flow Metering/Expansion Devices

The flow metering/expansion device provides the pressure drop necessary to lower the pressure of the refrigerant to provide cooling/dehumidification. Flow metering/expansion devices can be capillary tubes, orifices, automatic expansion valves (AEVs), thermostatic expansion valves (TXV), or float valves (high-side or low-side). The flow metering/expansion device separates the HP side from the LP side of a system and controls the flow of refrigerant into the evaporator.

3.2.3.1 Capillary Tubes

This flow metering/expansion device is the most prevalent in small commercial applications. They can usually be used in systems up to 10 tons. As the liquid refrigerant nears the end of the capillary tube, the liquid starts to change state to a vapor. A capillary tube passes liquid much more readily than vapor due to the increased friction with the vapor. The amount of refrigerant passed through the capillary is a function of the system according to load demand with the compressor running. The HP and LP sides will equalize when the compressor is off, thus permitting an easier start for the compressor.

3.2.3.2 Automatic Expansion Valve

The AEV is a pressure-operated flow metering/expansion device controlled by the pressure in the evaporator. As the evaporator pressure decreases, the pressure in the valve body is reduced, permitting the spring pressure from the adjusting screw assembly to open the valve. When the valve opens, liquid refrigerant flows into the evaporator and vaporizes to maintain the evaporator at a constant pressure.

3.2.3.3 Thermostatic Expansion Valve

The most prominent type of flow metering/expansion device for a direct expansion evaporator is a thermostatic expansion valve, commonly called a TXV, and acts as a constant superheat valve. It is also known as a thermal expansion valve (TEV). Its purposes are to control the flow of refrigerant to the evaporator to keep the evaporator "fully active" and to close when the compressor stops. *Fully active* means that liquid refrigerant evaporates, or boils, throughout the entire evaporator. In this situation, none of the evaporator contains superheated gas. In practice, some of the evaporator must be devoted to superheating the vapor to prevent liquid return to the compressor.

Three pressures affect the opening and closing of this refrigerant control device:

- Bulb pressure
- Evaporator pressure
- Spring pressure

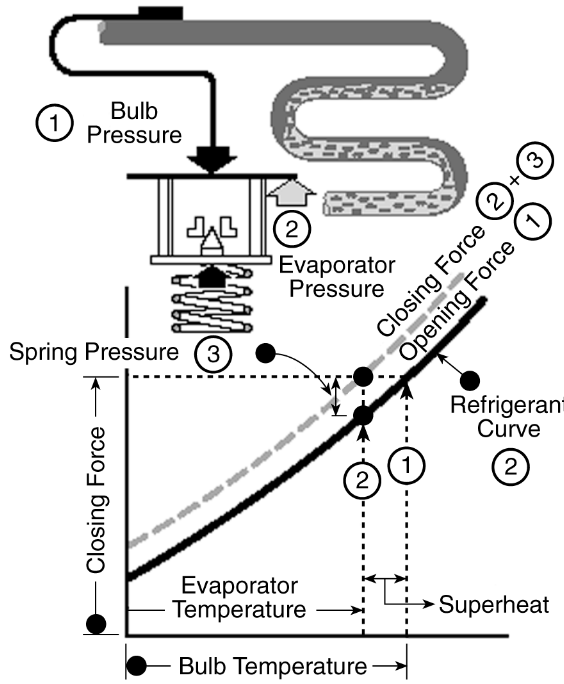
Key Technical Point

The TXV sensing bulb must NOT be located at the bottom of the line. It must have good contact with the line and be insulated so that the temperature sensed is not affected by ambient temperature.

Knowledge of these three pressures is essential to understanding TXV operation. The basic pressure relationship is that bulb pressure must be equal to the sum of evaporator pressure and spring pressure in order for the valve to be in an open position. Evaporator pressure is an essential factor in the operation of a TXV.

Evaporator pressure is applied under the diaphragm in the direction of valve closure, and it is always equal to the pressure that corresponds with the refrigerant saturation temperature. Spring pressure is also applied to the underside of the diaphragm to further implement closing of the valve.

Bulb pressure is applied through a capillary tube to the top of the valve diaphragm. This pressure opposes the sum of evaporator pressure and spring pressure and acts to open the TXV. When a system is in operation, the bulb charge will be part liquid refrigerant and part saturated vapor. Bulb temperature will be equal to suction gas temperature that will be higher than refrigerant saturation temperature, the increase being produced by superheat. Because of the higher bulb temperature, pressure applied to the top of the diaphragm will be higher than the opposing evaporator pressure. To open the valve, however, there must be sufficient pressure difference (increased superheat) to overcome the additional effect of spring pressure.



Courtesy of Sporlan

Figure 3-8
Thermostatic Expansion Valve

The TXV has an uncomplicated function. It meters sufficient liquid refrigerant to the evaporator to satisfy load conditions. It does not directly control temperature, humidity, pressure, or compressor running time. Valve performance cannot effectively be determined by measuring suction pressure or by observing the extent of frost formation on the suction line.

3.2.3.4 Float Valves

High-Side Float Valves:
A high-side float valve controls the mass flow of refrigerant liquid entering the evaporator, and as such, it equals the rate at which the refrigerant gas is pumped from the evaporator by the compressor. The liquid refrigerant flows from the condenser into the high-side float valve body where it raises the float and moves the valve pin in an opening direction, permitting the liquid to pass through the valve port, expand, and flow into the evaporator. Most of the system refrigerant charge is contained in the evaporator at all times. The high-side float system is a flooded system. The float maintains a constant refrigerant level on the HP side.

Low-Side Float Valves:
The low-side float valve performs the same function as the high-side float valve, but it is connected to the LP side of the system. This flow metering/expansion device is used on flooded evaporators to control the level of refrigerant in the evaporator. As the refrigerant level falls, the low-side float also falls, opening the valve or providing an electrical switch output to a solenoid to control level. This admits liquid refrigerant to the evaporator to replace the evaporated liquid to maintain the refrigerant level in the evaporator.

3.2.4 Evaporators

The evaporator absorbs heat from the heat source by the change in state of the refrigerant liquid. The refrigerant's LP at this point permits the refrigerant to boil or evaporate at a lower temperature and results in the absorption of heat. There are many design configurations for evaporators. The evaporator is typically designed based on medium selection (water or air), anticipated loads, the environmental conditions, the type of refrigerant used, and other factors. The force that drives the refrigerant through the evaporator is the differential pressure created by the compressor.

A/C evaporators are usually constructed of a number of tubes covered with fins with system air flow through the evaporator. The purpose of the fins is to increase the surface of the evaporator and thus increase the heat transfer. The fins absorb heat from ambient air by convection, conduction, or radiation depending on the application. The fins transfer heat to the tubes by conduction. Inside of the tube is liquid refrigerant, which absorbs the heat contained in the tube by conduction. As heat is absorbed, the liquid refrigerant begins to change state from a liquid to a vapor. Near the outlet of the evaporator, the refrigerant is a superheated gas. It is imperative that the refrigerant liquid **does not** enter the compressor.

On a DX evaporator, there is usually a fan that drives air across the coil. On the outer surface of the coil, another change of state is usually occurring. The ambient air is a mixture of air and water; thus, as the temperature of the air changes, so does the air's ability to hold water. As the dry bulb temperature (that is, sensible heat) drops, air becomes denser. As the air cools, it reaches its dew point, and water condenses on the evaporator. This change of state, water vapor condensing into liquid water on the evaporator, is an example of latent heat of condensation. The sensible heat in the air becomes latent heat in water vapor providing an evaporative cooling effect.

Chilled water evaporators are generally constructed in one of two configurations: 1) shell and coil or 2) shell and tube.

In a flooded evaporator, water is often used as the cooling medium. The chilled water can then be circulated to the areas where cooling is desired. System water in an evaporator is chilled as its heat is transferred to the refrigerant at low temperature and pressure. As heat is removed from the water, the refrigerant vaporizes and is drawn into the compressor.

3.2.5 Receivers

The liquid receiver is an optional device used as a storage tank for excess refrigerant, and should be used on systems where the condenser does not provide subcooling. As cooling requirements change due to varying system requirements, the quantity of refrigerant necessary for refrigeration to occur also changes. A receiver allows for this variable capacity. The receiver also provides a place to store refrigerant during maintenance operations, that is, refrigerant evacuation.

Receivers are sometimes part of the condenser package, typically located downstream of the condenser, and should be located below the condenser in the same ambient temperature as the condenser.

	Key Technical Point
	Receivers are typically sized to hold the entire system charge when 80% filled with liquid, which allows room for expansion.

If the receiver is located away from the condenser, the line connecting them should slope toward the receiver and be large enough to provide drainage. Thus at full flow, this line should be only about half full so that any vapor generated in the receiver for any reason (for instance, if heated by sun radiation) can find its way back into the condenser. If this line is too small, it fills with liquid and partially fill the condenser until the condensing pressure is high enough to force the liquid out of the condenser into the receiver. This can result in dangerously high discharge pressures, compressor overloading, or shutdown on high head.

3.2.6 Capacity Controls

As the load on the evaporator decreases, the suction pressure and temperature to the compressor also decrease. If the load continued to reduce without some method of capacity control, the compressor might ultimately trip off due to low suction temperature and the resulting LP cutout. Capacity controls include cylinder unloaders, hot gas bypass, inlet guide vanes/slide valves, control switches, and control (solenoid) valves.

3.2.6.1 Cylinder Unloaders

Capacity control for the refrigerant system can be accomplished by a cylinder unloader. The cylinder unloader can function by preventing suction valve closure or by directing the pressurized refrigerant gas to be discharged back to the suction side of the compressor. This is performed by equipment generally designed by the manufacturer for the specific compressor.

3.2.6.2 Hot Gas Bypass

Hot gas bypass provides for capacity modulation and has many variations. Hot gas bypass provides a false load to the compressor by bypassing HP refrigerant gas from the compressor discharge to the system LP side to maintain operation under low load conditions. Proper bypass control is accomplished by a modulating-type pressure regulator, which opens when there is a decrease in valve outlet pressure, as would be created by a low load on the evaporator.

In the most common type of bypass, the bypass line is taken directly from the compressor discharge line, through a bypass regulator, and into the suction line at the compressor. With this system, hot gas should enter the suction line at a location that provides a good mixing of the hot gas being bypassed and the suction gas entering the compressor. Prolonged bypass operation may necessitate the use of a liquid injection expansion valve for the purpose of desuperheating, or quenching. The use of this particular configuration depends on factors such as the system design and compressor design.

An excellent means of capacity control, although more costly than simpler bypass processes, uses a typical balance loader where liquid refrigerant is bypassed from the liquid line through an automatic expansion valve into a heat exchanger in the compressor discharge line. Liquid refrigerant is boiled off in the heat exchanger and superheated gas can then flow into the suction line or suction line accumulator.

For all hot gas bypass applications, the problem of head pressure control must be considered. Piping of the hot gas into the system's low side should be done in such a manner as to provide a homogeneous mixing of the gases. Pipe sizes usually dictate the best method to use. Most major equipment manufacturers have specific instructions concerning these bypass applications. The application manual provided by the compressor or unit manufacturer should be consulted prior to making any configuration changes.

A solenoid valve in the liquid-line to liquid-injection valve is recommended to prevent the flow of liquid into the suction line during periods when the hot gas bypass valve is not operating, such as high load or pull-down periods.

<u>Hot Gas Bypass Line Solenoid:</u>
Just as a solenoid is recommended for the liquid injection valve, a solenoid valve should be installed in the hot gas bypass line ahead of the bypass regulator. This permits automatic pumpdown and unwanted high-side to low-side leakage. It also prevents bypass during unwanted periods. The solenoid can be wired into the system control circuit, parallel with the liquid line solenoid, wired into the cylinder unloader circuit, or wired to a pressure switch. If the bypass valve is piped to a suction line, the bypass line solenoid must be wired in series with a discharge line thermostat set to terminate bypass if the discharge temperature exceeds 300°F.

3.2.6.3 Inlet Guide Vanes and Slide Valves

Inlet guide vanes and slide valves modulate to control the flow of refrigerant gas into the compressor suction. Slide valves are typical for a rotary screw compressor, and inlet guide vanes are typical for a centrifugal compressor.

3.2.6.4 Control Switches

Control switches operate one or more sets of electrical contacts to control system components. Control switches respond to physical changes such as pressure, temperature, liquid level, flow velocity, and proximity.

Pressure and temperature responsive controls have one or more power elements, which can use bellows, diaphragms, snap disks, or bourdon tubes to operate the mechanism.

Level responsive controls can use floats, mercury balance tubes, or electronic probes to operate one or more sets of electrical contacts.

Refrigerant system controls can be categorized into three basic groups: operating, primary, and limit. Operating controls, such as thermostats, turn systems on and off. Primary controls, such as the HP cut-out switch, protect a refrigeration system from unsafe operation.

Pressure Control Switches: The refrigerant pressure is applied directly to the element, which moves against a spring that can be adjusted to control any operation at the desired pressure. The types of pressure control switches are:

- HP cut-out (shut off the compressor when excessive pressure occurs)
- High-side LP (shut off the compressor under low-ambient and/or LP conditions)
- High-side fan cycling (cycle the condenser fan on and off to provide proper condenser pressure)
- Low-side LP (disengage the compressor when low charge or system blockage occurs)
- Low-side cycling (cycle the compressor on and off to provide proper evaporator pressure)

Pressure Transducers: These devices are most often used as a substitute for one or more pressure switches. The pressure transducer is typically an analog device that produces a continuous low-voltage output signal proportional to the pressure applied. It requires a three-pin connection, input and output voltage, and ground.

Temperature Control Switches: An indirect temperature control switch is a pressure control switch in which the pressure-sensing element is replaced by a temperature-sensing element. Operation of the control switch results from these changes in pressure or volume. A direct temperature control switch generally contains bimetallic disks that activate electrical contacts when the temperature increases or decreases. As the temperature of the metallic disk or blade in the switch increases or decreases, the attached electrical contacts engage or disengage.

Fluid Flow Sensing Switches: Both air flow and liquid flow sensing switches are available. The spring-loaded vanes of the air flow switch are installed in a regulated, high-velocity air stream. The pressure of the air causes the vane to deflect and make electrical contact. The liquid flow switches work in a similar manner. Liquid flow sensing switches are normally used to indicate sufficient water flow or refrigerant flow through the evaporator or condenser.

Differential Control Switches: These switches sense a given difference in pressure or temperature between two pipes, lines, or spaces. Instrument differential is the difference in pressure between the low- and high-pressure elements for which the instrument is adjusted. Operating differential is the difference in pressure or temperature required to open or close the switch contacts.

Float Switches: A float switch has a float that operates one or more sets of electrical contacts as the level of a liquid changes.

3.2.6.5 Control Valves

Control valves are used to start, stop, direct, and modulate process flow to satisfy system requirements in accordance with system load.

Evaporator Pressure and Temperature Regulators: The evaporator pressure regulator (backpressure regulator) regulates the evaporator pressure (pressure entering the regulator) at a constant value. It is used in the evaporator pressure or temperature is required.

Suction Pressure Regulators: The suction pressure regulator (holdback valve or crankcase pressure regulator) limits the compressor suction pressure (regulator outlet pressure) to a maximum value.

Condenser Pressure Regulators: Various condenser pressure regulating valves are used to maintain sufficient condensing pressure to allow the air-cooled condensers to operate properly during the winter.

Solenoid Valves: A solenoid valve is closed by gravity, pressure, or spring action and is opened by a plunger actuated by the magnetic action of an electrically energized coil, or vice versa.

Condensing Water Regulators: The condensing water regulator modulates the quantity of the water passing through a water-cooled refrigerant condenser in response to the condensing pressure. The condensing water regulator consists of a valve and an actuator that are linked together. A three-way regulator is similar to the two-way valve except it has an additional port that opens to bypass water around the condenser.

Check Valves: Refrigerant check valves are normally used in refrigerant lines in which pressure reversals can cause undesirable reverse flows. A check valve is usually opened by the portion of the pressure drop. Closing occurs either when a reversal of pressure takes place or when the pressure drop across the check valve is less than the minimum opening pressure drop in the normal flow direction.

Key Technical Point

A safety relief device is designed to relieve positively at its set pressure for one crucial occasion without prior leakage. The relief may be to atmosphere or to the low-pressure side.

4
PERFORMANCE MONITORING (NONINTRUSIVE)

This section provides the methods by which the performance of refrigerant systems and their components can be assessed during normal operation and/or in a standby operating mode. In some instances, measured parameters can be trended to determine whether a system's performance is degrading at an uncharacteristically high rate. Any parameters outside the norm should be addressed and corrected immediately.

4.1 Typical Parameters for Monitoring

Typical recommended parameters that should be monitored and trended are identified in Tables 4-1 through 4-3. Table 4-1 provides recommended daily checks, Table 4-2 provides recommended quarterly checks, and Table 4-3 provides recommended annual checks. The recommended frequencies of the performance monitoring activities can vary depending on plant-specific applications and requirements.

Key O&M Cost Point

Integration of daily checks in routine operator rounds is a cost-effective method for accomplishing chiller performance monitoring.

A performance guide from Trane Company in Section 4.3 addresses performance monitoring of LP chillers with hermetic motors.

Table 4-1
Recommended Chiller Performance Monitoring/Trending: Daily Checks and Trends

Note: These daily checks can be performed at some sites during operator rounds, depending on plant-specific performance monitoring practices.

Daily Checks and Trends
Oil level An adequate amount of oil, measured by checking the oil sump level, is essential for correct pump operation. A low oil level could indicate an equipment leak or migration to another part of the machine, which is caused by operating out of design conditions (inadequate load, cold condensing water). Oil entering the condenser or evaporator affects the unit's efficiency, because oil decreases the thermal properties of the refrigerant resulting in less heat rejected by the condenser and less heat absorbed by the evaporator. A high oil level can be caused by the return of oil to the sump when the unit is properly loaded, or by the presence of an excessive amount of refrigerant in the oil. Refrigerant in the oil can indicate a leaking or broken valve, a condition which lets the refrigerant escape into the oil sump. On LP machines, refrigerant can also enter the sump when the unit is operating with conditions that have sufficiently deviated from design. Oil levels are usually observed through sight glasses in the oil sump.
Oil temperature Refrigerant must be kept out of the oil. When the chiller is in standby condition, oil temperature is maintained by an oil heater. If the oil in the sump is allowed to cool, refrigerant combines with the oil. When the mixture goes through the pump, the LP at the pump suction causes the refrigerant to boil out of the oil, lowering the temperature. This causes foaming and erratic oil pressure and results in a trip of the unit. Oil is often sent through a cooler before it is pumped to its intended destinations. Oil temperature can be obtained by installed instrumentation or by measuring the temperature of the outside of the sump. The thermostatically controlled heaters can also be on when the unit is operating if refrigerant is introduced into the oil sump, lowering the temperature. The heater boils the refrigerant out of the oil and results in foaming on the oil top surface.
Oil heater on when in standby The oil heater keeps the oil at a high enough temperature to prevent refrigerant from mixing with the oil during times when the unit is not running. If the oil temperature is left at ambient temperature, then refrigerant combines with the oil resulting in tripping problems. Oil heaters can be tested with an amp meter to check for proper operation. Oil heaters are thermostatically controlled to keep the temperature in a predetermined range.
Oil pressure Important to the operation of any rotating machinery, oil pressure ensures that bearing surfaces are properly lubricated before the unit is run and after it is shut off. It also serves other functions such as a purge device on LP machines and valve motive force on other machines. Oil pressure is determined by instrumentation on the unit and can be adjusted, usually by regulating a bypass valve in the discharge line.
Oil leaks Signs of oil leaks should be taken seriously because oil provides cooling to bearing surfaces. A lack of such cooling could have serious impact on the longevity of the chiller's operating life. Detection of an oil leak indicates the probability of a refrigerant leak. Oil leaks are more prevalent on the HP applications on the unit, that is, HP oil line to pressure switch or filter connection.

Table 4-1 (cont.)
Recommended Chiller Performance Monitoring/Trending: Daily Checks and Trends

Daily Checks and Trends
Oil filter pressure drop The filter differential pressure (DP) should be monitored to determine filter loading. As the DP increases, so does the fouling of the filter resulting in less oil pressure to the unit. Filter examination is also a valuable tool in machine diagnostics to determine what type of fouling is in the filter (that is, bearing material, rust, gear metal, or seals).
Discharge pressure The discharge pressure is influenced by the mechanical condition of the compressor, refrigerant charge, water or air (condenser heat sink) temperature, and water or airflow through the condenser. Discharge pressures are often maintained at a constant level by a variable speed fan for an air-cooled condenser or modulating valve arrangements for a water-cooled condenser. The compressor discharge pressure is monitored by the installed unit instrumentation or the use of test manifold gauges. If discharge pressure is higher than it should be, the following can be checked: • Air or noncondensables in the condenser, which on an LP machine could mean a malfunctioning purge unit or a leak that exceeds the capacity of the purge unit • Fouled condenser tubes • For multi-pass condensers, leakby in the water boxes caused by defective gaskets
Discharge temperature This is the temperature of the refrigerant gas when it leaves the compressor. As the pressure increases, so does the temperature. Higher than normal discharge gas temperature could be an indication of low-refrigerant charge. The temperature can be measured via installed instrumentation or by obtaining the surface temperature of the discharge gas piping. Some units have thermal wells at the compressor discharge, while others require a contact temperature probe to determine the temperature. The discharge temperature can be converted into pressure for an equipment check. This parameter is a function of the compressor and the same inputs that affect pressure also affect temperature.
Condenser water inlet/outlet temperatures These are the temperatures of the condenser cooling water. This measure is another vital reading to monitor because it is also used in the determination of the real-time performance of the chiller or air conditioner. The difference between these temperatures indicates the existing heat removal capability of the condenser. These temperatures are measured by reading installed instrumentation or placing test instrumentation into thermal wells at the inlet and outlet piping to the condenser.

Table 4-1 (cont.)
Recommended Chiller Performance Monitoring/Trending: Daily Checks and Trends

Daily Checks and Trends
Suction pressure Suction pressure, the refrigerant pressure at the suction of the compressor, depends on the conditions of the evaporator such as the refrigerant charge, the temperature of the air or water passing through the evaporator, and the temperature and flow of air or water passing through the evaporator influence it. The suction pressure is greater if the air or water passing through the evaporator is hot, as is the condition at the start of a unit. As the temperature of the process flow decreases, the lower suction pressure decreases. The installed unit instrumentation or the use of test manifold gauges monitors the compressor suction pressure. Abnormally low suction pressure can indicate several things, such as insufficient refrigerant charge, blocked flow orifice, dirty or restricted evaporator tubes, excess oil in refrigerant charge, or insufficient load for the system capacity.
Suction temperature This is the refrigerant temperature at the suction of the compressor. The suction temperature is directly related to the suction pressure and one depends on the other. As the pressure increases, so does the temperature. Some units have thermal wells at the compressor suction, while others require a contact temperature probe to determine temperature. This can be converted into pressure for an equipment check. The same inputs that affect pressure also affect temperature.
Space temperature This is the temperature of the space(s) that is (are) cooled and typically is (are) monitored and controlled by a thermostat. A higher than normal temperature could indicate an insufficient air flow, an insufficient chilled water flow, a high chilled water supply temperature, a dirty cooling coil, or a fan or fan motor problem. Typically, as the space temperature increases, the thermostat detects the rising air temperature and sends a signal to a controller in the A/C unit. The controller in turn opens additional liquid line solenoid valves, increasing the available coil capacity, which supplies colder air. The room temperature also affects the loading of the compressor. The higher the suction pressure (caused by the warm air passing by the evaporator coils), the more stages of a multi-stage compressor are loaded or in operation. With a chilled water system, the thermostat controls a modulating valve usually located on the return side of the cooling coil. The rising temperature opens the valve to a position where the chilled water flow satisfies the required room temperature.
Supply air temperature The supply air temperature is the air temperature leaving the evaporator coil for an A/C unit or at the cooling coil for a chilled water unit. The return temperature is usually the space temperature. Air filters are used to ensure a source of clean air, which is essential to coil performance. A dirty coil reduces its performance, lowering the refrigerant suction pressure on an A/C unit. On simple units, this has led to freezing the coil as the air flow is restricted by the fouling and the coil temperature drops with the lowered pressure.

Table 4-1 (cont.)
Recommended Chiller Performance Monitoring/Trending: Daily Checks and Trends

Daily Checks and Trends
<u>Condenser pressure</u> This pressure is the saturated refrigerant pressure of the condenser, which is closely related to the compressor discharge pressure. Condenser pressure can be affected by several factors including the condenser media flow rate, condenser media inlet temperature, evaporator heat load, amount of noncondensable gas in the condenser, and condition of the condenser tubes. It is usually available from unit-mounted instrumentation or a test manifold. It is used to verify proper unit performance by comparing it to the liquid temperature in the condenser. Higher than normal readings indicate the possibility of noncondensables in the system. Condenser pressure is limited by the use of relief valves and/or rupture disks. With the advent of digital control panels, it is now possible to prevent tripping of the unit on high-condenser pressure, because the units can now limit the load to keep the condenser below design pressure. For systems with controllable condenser cooling media, the pressure is typically maintained to a preset value.
<u>Evaporator pressure</u> Evaporator pressure, the refrigerant pressure of the evaporator, is closely related to or the same as the compressor suction pressure. A lower than normal evaporator pressure can indicate a capacity control problem or an insufficient refrigerant charge. A higher than normal evaporator pressure can indicate a capacity control problem or a high heat load. The pressure is monitored either from the unit instrumentation or by the use of a test manifold connected to pressure taps. Monitoring the evaporator pressure is a good indication of the refrigerant charge. A low charge results in a low suction pressure. A pressure temperature chart is used to compare the evaporator liquid temperature to the saturation temperature; if a difference in these values exists, the problem of contaminated refrigerant can be identified.
<u>Superheat of vapor refrigerant</u> The calculated difference between the measured refrigerant gas temperature and the saturated refrigerant temperature, superheat of vapor refrigerant is the temperature increase above the saturation temperature or above the boiling point. The superheat should be checked when the TXV on the liquid line of the A/C unit is inspected. Proper superheat ensures that complete evaporation has occurred before the refrigerant enters the suction of the compressor.
<u>Subcooling of liquid refrigerant</u> The calculated difference between the measured liquid refrigerant temperature and the saturated refrigerant temperature, subcooling is the difference between the temperature of the liquid refrigerant in the condenser and the temperature of the liquid refrigerant in the evaporator. The existence of subcooling is important to eliminate the flash gas in the liquid line, thus increasing the efficiency of the unit.
<u>Refrigerant charge (via sight glass)</u> The refrigerant charge, as indicated by a sight glass, is a gross indication of the charge in the machine. In a chiller, a sight glass that reveals the refrigerant level in the evaporator usually measures the refrigerant charge. Most A/C units are charged until the sight glass in the liquid line is free of flashing or bubbles with the unit running at a full load at or close to design conditions.

Table 4-1 (cont.)
Recommended Chiller Performance Monitoring/Trending: Daily Checks and Trends

Daily Checks and Trends
Moisture in refrigerant (via sight glass) Some units are equipped with moisture-indicating sight glasses. If a sight glass indicates the presence of moisture, the unit should be removed from service and the cause determined and repaired. Moisture (water) and refrigerant combine to form an acid that attacks the unprotected interior surfaces of the unit.
Record hour meter reading A log should be maintained as a basis for performing preventive and predictive maintenance.

Table 4-2
Recommended Chiller Performance Monitoring/Trending: Quarterly Checks and Trends

Quarterly Checks and Trends
Vibration checks A periodic vibration check on the refrigerant system components helps to detect potential imbalance, misalignment, or degradation. Even a simple hands-on check can reveal the start of a bearing failure or an imbalance of fan blades or impellers. Vibration instrumentation allows for detection and trending of bearing degradation in advance of equipment failure. Analysis can also differentiate between a bent shaft and misalignment. Electrical wave analysis can monitor equipment vibration.
Refrigerant leaks A periodic leak check on the chiller is important because it helps to determine existence of refrigerant leak. Excessive refrigerant leak can impact the performance of the chiller and could violate EPA law. Leak detectors can identify leaks on units that operate at positive pressure. It may be possible with a modern leak detector to estimate the size of the leak. On an LP machine (one that operates in a partial vacuum), only the HP side can be checked during operation. A leak on the LP side is evident from the excessive amount of purges that the purge unit encounters. The unit must be removed from service, the charge removed, and then pressurized with nitrogen. The pressurized unit can then be leak-checked using a detector. Alternatively, the unit may be pressurized for a leak check by recirculating either the condensing or chilled water without running the chiller. This increases the water temperature and thus increases the machine pressure to permit a leak check using a detector.
Trend oil and refrigerant addition A log should be maintained that tracks the addition and removal of oil and refrigerant from refrigerant systems. Tracking and maintenance of records of refrigerant addition is required by law in accordance with 40CFR82. Systems running at a partial load tend to experience oil migration into the evaporator. If possible, it is preferable to increase the heat load to more fully load the refrigerant system with the low oil level than adding oil to the sump. By increasing the load on the refrigerant system, the oil is returned to the sump as operation returns closer to design conditions. If load increase is not possible, the manufacturer should be contacted for recommendations to alleviate the problem. An excessive amount of oil in the evaporator is detrimental to the efficiency of the unit. Excessive removal of oil could be indication of overcharge of oil during startup or higher than normal rate of refrigerant absorption into the oil. A properly charged unit should not require further additions of refrigerant. If refrigerant levels are low, the unit should be checked for leaks. During major maintenance activities, the charge should be completely recovered and processed. The charge should be weighed for a true determination of the amount of refrigerant loss.
Air filters pressure drop Most cooling units have a filter D/P gauge installed. This allows periodic checking of filter fouling without having to open the unit to inspect the filter media. If no gauge exists, then visual inspection is required. The filter protects the evaporator or cooling coil from external fouling. On a DX air coil with a filter upstream of the evaporator, refrigerant suction pressure decreases as the coils are fouled, which might result in damage. Cleaning is required when a decrease in efficiency begins to occur, and it restores the coils to design performance.

Table 4-2 (cont.)
Recommended Chiller Performance Monitoring/Trending: Quarterly Checks and Trends

Quarterly Checks and Trends
Calibration checks Instrumentation and controls on the units must be kept in calibration to ensure proper operation and response to permissives and safety trip signals. The controls on refrigerant systems are many and varied. Refer to site and/or OEM calibration frequencies. With the advent of digital control panels, instrument calibrations will decrease because more of them are now computer-controlled.
Compressor thrust bearing inspection and bearing temperature Most bearings on smaller A/C units and chillers are internal. Little can be done to check internal bearings. For large units, the bearing high temperature instrumentation must be calibrated and functioning properly. Open drive motor bearings should be greased and monitored on a periodic basis. During walkdowns, the temperature of some bearings can be determined by touching the bearing housing. Overgreasing of motors is often the cause of bearing overheating. Touching the bearing housing also reveals any abnormal vibration in the motors, compressors, or fans. When the bearing temperature is above a preset set point, the unit trips. A persistent high bearing temperature can be caused by a degraded bearing, high oil supply temperature, or low oil flow.
Inspect drain pan on air handling unit Drain pan inspections are important to monitor the fouling that can occur. Fouling can clog the drain resulting in flooding inside the air handling part of the A/C unit or the cooling coils for a chilled water system. Drain pans and drain piping should be periodically treated with a biocide, if allowed, or cleaned out by mechanical means to prevent the fouling of the drain system.
TXV inspection Thermostatic expansion valves are self-contained devices that leave little to inspect apart from the capillary tube. Proper TXV operation can be verified by checking the superheat of the unit. The TXV manufacturer has instructions to accomplish this (see Section 5, “Troubleshooting”).
Check belts Most A/C and chiller units in the nuclear power industry have direct drive compressors. The fans in some of the air handling units, though, may be belt-driven. The belts should be checked for wear, pliability, and proper tightness. The belt sheaves and pulleys should also be properly adjusted.
Water chemistry Periodic water chemistry analysis is used to detect system or tubing leaks.

Table 4-3
Recommended Chiller Performance Monitoring/Trending: Annual Checks and Trends

Annual Checks
Motor meggering Motor meggering is performed to verify the integrity of the compressor motor stator windings. CAUTION: Meggering should not be performed while the system is under deep vacuum.
Eddy current testing Periodic eddy current examination is performed to verify the integrity of the condenser and evaporator tubes.

4.2 Additional Performance Checks

Most compressors on refrigerant systems are precision-built components designed to pump only vapor, and they are designed for high efficiency and long life when properly applied and maintained. In some cases, however, due to defective material or workmanship, normal wear, misapplication, air and moisture, dirt, stress of some extreme condition, or malfunction of some other part of the system, the compressor operates improperly. There are several checks that can be made on a compressor or condensing unit to determine whether or not it is operating properly, and, if not, where the cause of trouble might be. Preventive maintenance (PM) (see Section 6, "Preventive Maintenance") is a tool that, when used hand-in-hand with performance monitoring, can help to ensure the safe, reliable operation of a refrigerant system.

4.2.1 General

Typical performance checks for refrigerant systems include:

- Check for noncondensables in the systems. Pump the entire refrigerant into the condenser (pumpdown) by front-seating the liquid line service valve. Operate the compressor until the low side has been pumped down to approximately 5 psig. Stop the compressor and allow it to sit idle until it has cooled to ambient temperature. Compare the discharge pressure to the pressure indicated on the pressure-temperature chart at the ambient temperature. If the discharge pressure is higher, noncondensables are present and should be vented from the system.
- Check the return air temperature. Adjust the room thermostat or chilled water temperature control valve (TCV) to load and unload the chiller. Verify the operation of unloader solenoids.
- Check the return air or chilled water temperature. The only solution to high return air or water temperature is to allow the unit to operate until the return air or water temperature is lowered to the desired operating temperature. A low return air or water temperature does not place a heavy enough load on the system, and the suction pressure is lower than normal. There can be frost on the evaporator and sometimes extreme sweating of the suction line back to the compressor.

- Check the condition of the contactor contacts. They can prevent the proper flow of current to the motor windings, causing permanent damage. When severely pitted, these contacts do not make good contact, thus causing higher current draw than normal. Damaged contacts should be replaced as soon as possible.
- Check for an inoperative room thermostat. If more than a 10°F calibration is needed, replace the thermostat.
- Check the LP control, HP control, and compressor suction valve according to the manufacturer's recommendations.
- Check for proper refrigerant charge. There are several ways to determine if the system is properly charged, such as weighing the charge, using the sight glass, using the liquid level indicator, using the liquid subcooled method, using the suction superheat method, and using the manufacturer's charging charts.

  For DX Systems

  – The sight glass method: A steady stream of bubbles indicates that the system is low on refrigerant. Admit refrigerant into the system while observing the sight glass and the discharge pressure gauge. When bubbles disappear and do not reappear intermittently, the system is properly charged. A sudden increase in discharge pressure indicates that a system is overcharged. Stop charging the unit and remove some of the charge.
  – The liquid level indicator method: A liquid level indicator may be located in the lower section of the compressor. The system is properly charged if a level of liquid refrigerant can be observed in the liquid level indicator port.
  – The liquid subcooling method: Compare the liquid line temperature to the condensing temperature. The condensing temperature is determined by relating the discharge pressure to the corresponding temperature on a pressure-temperature chart. The liquid line temperature is indicated by strapping a thermometer to the liquid line. If the liquid line temperature is less than 5°F below the condensing temperature, more refrigerant is needed in the system.
  – The suction superheat method: Determine the difference between the suction line temperature and the saturation temperature equivalent to the suction pressure with the unit running. The saturation temperature is determined by relating the suction pressure to the corresponding temperature on a pressure-temperature chart. The suction line temperature is determined by strapping a thermometer on the suction line approximately 6 in. from the compressor. A low superheat can indicate an overcharged unit. Conversely, a high superheat can indicate an undercharged unit.

  For Flooded Evaporators

  – The approach temperature method: Compare the saturation temperature in the evaporator to the leaving chilled water temperature.
  – The superheat method: Compare the compressor discharge temperature to the saturation temperature in the condenser.

4.2.2 Capacity Check

With some systems, it is a simple matter to determine whether a condensing unit is developing its rated capacity. For example, a water refrigerant system can be checked by measuring the water flow and temperature drop of the water passing through the evaporator. The capacity can be calculated by multiplying the gallons of water circulated per hour by 8.33 (to convert to pounds per hour), and then multiplying this by the temperature drop (°F). The result is Btu/hr condensing unit capacity. To get the gallons per hour flow, a flowmeter, either installed or ultrasonic, can be used. If there is a fluctuation in flow, several readings should be taken, and the average determined.

For an A/C unit, reasonably accurate capacity figures can be obtained if the required instruments are available. This involves accurately measuring the wet bulb temperature of the air entering and leaving the air handling unit or evaporator. Knowing the wet bulb temperature drop of the air in passing over the evaporator, one can interpolate from tables or a psychrometric chart the Btu/lbm removed of air circulated. From readings of wet bulb and dry bulb temperatures of the air leaving the coil, the specific volume (cubic feet per pound) of the air leaving the coil can also be calculated from the psychrometric chart. Then with a velometer, prop anemometer, or some other acceptable device for reading air velocity, the average velocity across the face area of the coil or through a discharge duct can be determined.

From this data, the condensing unit capacity can be calculated as follows:

Air circulated (cfm) = average velocity (fpm) x area (ft^2)

Air circulated (lb/hr) = (cfm x 60)/specific volume

Btu/hr capacity = air circulated (lb/hr) x reduction in heat content (Btu/lb)

The specific volume (cubic feet per pound) and heat content of air entering and leaving the coil can be obtained from a psychrometric chart, if the wet and dry bulb temperatures of the air entering and leaving are known.

These are just a few examples of how the actual capacity delivered can be checked. Another way of checking capacity, which is only approximate, is by checking the heat rejected by the condenser where a water-cooled condenser is incorporated. Appendix D, “Examples of Testing/Analysis Methods,” provides further methods by which system capacity can be evaluated.

4.2.3 Efficiency Check

If it is suspected or known that a compressor is not pumping efficiently, some checks can determine the cause of trouble. There are several causes of inefficiency. One of the first points to check is the condition of the refrigerant entering the compressor. If any amount of liquid refrigerant is entering the compressor, the efficiency and resulting capacity is very seriously affected. To obtain the rated capacity of a compressor, the suction gas must be superheated to the temperature specified for the rating. To accurately determine the actual degree of superheat, measure the temperature of the suction line and compare this reading with the saturated temperature corresponding to the actual suction pressure at the compressor. If insufficient superheat is indicated (excluding DX units), appropriate steps should be taken to increase the superheat.

Another cause of inefficiency is leaking compressor valves. The discharge valves can easily be checked by installing a suction pressure gauge and simultaneously closing the compressor suction shutoff valve as the compressor is stopped. A leaking discharge valve will cause the crankcase pressure to climb rapidly. Excessive discharge valve leakage warrants the repair or replacement of the valve. However, one should make sure that the leakage is not a leaking cylinder head gasket or oil separator float valve, because the symptoms are the same for both. If an oil separator is incorporated, the possibility of its leaking can be ascertained by closing off or disconnecting the oil return line. A blown or leaking cylinder head gasket can usually be found by close inspection of the cylinder head and gasket after removal of the head. The web and gasket section separating the high-pressure from the LP chambers of the cylinder head should be particularly observed.

Leaking suction valves seriously affect compressor efficiency and capacity, especially on lower temperature applications. A test of suction valve operation can be made by gradually closing the suction shutoff valve and blocking the LP control or by putting a jumper across the control terminals. With suction and discharge pressure gauges installed, one can operate the compressor to determine the degree of vacuum that can be attained.

Key Human Performance Point

Caution should be taken when the suction shutoff valve is throttled closed too quickly due to the possibility of oil slugging the compressor.

One should be prepared to stop the compressor or open the suction shutoff valve if the compressor begins to knock severely. The degree of vacuum that should be attained varies with the make and model of compressor, discharge pressure, and atmospheric pressure. In general, compressors with a high stroke-to-bore ratio should pull slightly greater vacuum. Compressors designed for low-temperature application usually pull a greater vacuum than those designed for A/C and such high-temperature applications. It is important to keep this in mind when checking compressors in high-suction applications. The other variable, atmospheric (barometric) pressure, affects the suction gauge reading, because the suction pressure gauge reads the difference between the actual suction or crankcase pressure and atmospheric pressure. The lower the barometer reads, therefore, the less vacuum the suction gauge indicates.

A compressor with two or more cylinders can sometimes pull a good vacuum using just one cylinder. With a belt-driven compressor, it is possible to determine if both cylinders and all suction valves are operating normally by manually turning over the flywheel. If equal resistance is felt on all cylinders, and the compressor does pull a satisfactory vacuum, normal operation is indicated. With hermetic-type compressors that have two or more cylinders, a good vacuum might be indicated when only one cylinder is operating properly. Therefore, if poor compressor capacity and efficiency is a symptom, the only positive check of a multiple-cylinder hermetic-type compressor is to remove the valve plate and carefully inspect the suction valve reeds or disks and the valve seats. A properly seated valve normally shows a uniform ring marking on the suction reeds or disks. A small particle of dirt, scale, or any foreign matter on a valve seat can seriously affect the capacity of a compressor. Any burrs, pitting, flaws in the metal, or fracture of the valve seat have the same effect.

If a reciprocating compressor is proved to be inefficient, that is, does not pull a satisfactory vacuum, and yet the valve plate and reeds are in perfect condition, the trouble is usually that of loose pistons or worn bearings. Loose pistons cause excessive blowby and lack of compression. This condition can usually be observed by allowable sideways movement of the piston in the cylinder. Loose pistons cause a more pronounced lack of efficiency with ringless pistons. Worn bearings, especially loose connecting rods or wrist pins, prevent the pistons from coming up as far as they should on the compression stroke, which has the effect of increasing the clearance volume, and resulting in excessive re-expansion. It is especially detrimental in low-temperature applications and is usually accompanied by a knocking sound.

4.2.4 Leak Check

Before starting up a new or refilled system, the compressor, as well as all other components and refrigeration lines, should be carefully tested for leaks. The time required is well spent and pays dividends in avoidance of future trouble, especially where the normal operating suction pressure is at a vacuum and a leak resulting in air (with consequent moisture and contaminants) being drawn into the system.

Key O&M Cost Point

Leak checks prior to filling a refrigerant system can prevent costly refrigerant loss, unnecessary repairs, and future problems.

Because the compressor or complete condensing unit in the case of a factory-assembled package has been thoroughly dehydrated previous to shipment, the compressor shutoff and receiver outlet valves should be left closed until the remainder of the system has been leak tested and evacuated.

The compressor or condensing unit can be leak tested separately, introducing refrigerant vapor through the gauge port of the suction shutoff valve if the unit does not already contain sufficient refrigerant vapor pressure to ensure detection of any possible leak. The first step in leak testing is to build up a refrigerant pressure to a high, yet safe, level.

Key Technical Point

Under no conditions should the low-side pressure, or crankcase pressure, be allowed to exceed manufacturer's recommendations for leak testing. If it becomes necessary to mix nitrogen or carbon dioxide with the refrigerant to build up a satisfactory test pressure (such as in low ambient temperature), the gas cylinder must be equipped not only with a pressure regulator, but the charging line must incorporate a relief valve set to lower pressure when the maximum pressure per manufacturer's recommendations is reached.

All joints should be carefully tested with a good leak detector. The common halide detector, when in good operating condition and properly used, should pick up any leak of any consequence. However, for leak testing systems having a small and critical charge, a more modern electronic leak detector is recommended.

Extreme caution should be exercised in the use of any halide torch. The room should be well ventilated to preclude any fire hazard. The fumes discharged from the torch should not be inhaled, because these gases are extremely injurious when even a trace of refrigerant is being picked up by the sampling tube. To perform an accurate leak test, it is necessary to have the surrounding atmosphere free of refrigerant vapors and solvent vapors or any gases that might react, because refrigerant would react and create false indications by the detector.

Key Human Performance Point

Extreme caution should be exercised when using a halide torch. The room should be well ventilated to preclude injury and fire hazard.

All joints of the leak testing compressors should be tested. With open-type compressors, the shaft seal should be tested with special care. If a leak at the shaft seal is indicated, the compressor should be turned several revolutions and tested again. The seal of a compressor that has been idle for a long time may have lost its oil film, which is necessary for a vapor tight seal. If any leaks are found, they should be repaired and the leak testing procedure repeated.

The easiest way to monitor the performance of a condenser or evaporator is to trend the operating parameters over time. This has the advantage of not interfering with the operation of the component. The disadvantage is that the specific cause of component degradation might not be obvious from the results. Parameters that can be trended include:

- Process fluid flows
- Process fluid temperatures
- Chiller approach temperature
- Chiller superheat temperature
- Refrigerant pressure
- Tube-side differential pressure
- Process fluid quality (chemical composition)
- Refrigerant quality (quantity, chemical analysis for contaminants)
- Chiller load
- Thermal performance (thermal performance test)

Any comparison in parameters should be made under consistent operating conditions to provide reliable results. For example, comparing approach temperatures or differential pressures with different chiller loads or flows does not provide useful conclusions.

Monitoring of the process and refrigerant quality is important in identifying the potential for corrosion from leaks or an increase in the rate of fouling. For example, water in-leakage to the refrigerant can cause acidic reactions to the refrigerant side of the process or corrosion. Excess oil in the refrigerant can also result in increased degradation of the refrigerant side for the component. Poor process water quality can increase the fouling mechanisms on the tube-side of the component. Periodic sampling and analysis of the fluids can identify these conditions.

Inspections and testing provide direct evidence of changes in the component condition, but might require access to the inside of the component. These require the component to be removed from service or operated under conditions other than normal. Examples of testing methodologies are presented in Appendix D, "Examples of Testing and Analysis Methods."

Visual inspection is an easy and powerful tool in assessing the overall condition of the component. Visual inspection can be done with the use of a boroscope or similar device that allows access to most areas of the component. Visual inspection can identify pitting, corrosion, biofouling, macrofouling, and cracking in tubes and can be used for inspecting welds, nozzles, seals, tie rods, baffles, and spacers for abnormal wear or defects.

Nondestructive evaluation (NDE) techniques can be used to supplement visual inspections to better identify problems in the component. Eddy current testing is widely used for examining and assessing the condition of the tubes (for example, pitting, corrosion, cracking, and wall thinning). The results of the eddy current tests can be trended to predict the life of tubes or to identify any drastic change in tube wear. Ultrasonic testing can also be performed to assess the condition of tubes, and it provides results comparable to eddy current testing. Both of these methods require clean tubes to work best and are difficult to use on tubes that are not straight. Radiography can be used to locate defects in thicker parts, such as shell welds, but can be difficult on some locations where the component does not have a uniform shape, such as nozzle attachments. Other NDE techniques for locating surface defects, such as liquid penetrants or magnetic particles, can also be used. Leak testing can also be used to locate leaks on both refrigerant and process sides of the condenser or evaporator.

4.2.5 Motor Load Check

When a compressor is not performing satisfactorily, in addition to checking of capacity (as shown in Section 4.2.2) and efficiency (as shown in Section 4.2.3), checking of the motor load is sometimes revealing. Exceptionally high or low motor load indicates improper operation. Loose pistons, improper suction valve operation, or excessive clearance volume usually lead to a reduction in motor load. Another common cause of poor compressor performance, which might not be indicated by abnormal suction or discharge pressures, is a restricted suction chamber screen. The result is a much lower actual pressure in the cylinders at the end of the suction stroke than the pressure in the suction line as registered on the suction gauge. If such is the cause, an abnormally low motor load also results.

Improper discharge valve operation, partially restricted ports in the valve (which does not show up on the discharge pressure gauge), and tight pistons are usually accompanied by high motor load. It can be difficult or impossible to recognize exceptionally high or low motor loads without exact compressor or condensing unit performance data. With the compressor manufacturer's specifications and performance data, which usually list the wattage input to the motor for various conditions, one can apply a wattmeter and compare the actual wattage of the compressor motor with that listed for the prevailing conditions of suction and discharge pressures. If the manufacturer's performance data are not available, the motor current (amperes) are probably the only comparison. Most motors and some hermetic-type compressors have a nameplate with the amperage stamped on them. This may be indicated as "full load current" in which case the motor is usually good for at least 25% more than this value. Therefore, the fact that a motor is drawing a current greater than that stamped on its nameplate does not necessarily constitute abnormal operation. At low temperature or extra low temperature operation, the motor may be operating at only 75% of its full load rating. Wattage varies with actual suction and discharge pressures for any given model. The motor current (amperes) varies in much the same way. The wattage should be reasonably close to that shown in the manufacturer's published data and corresponding to the prevailing conditions. If the compressor is operating at the lower end of its suction range, the motor current is usually at, or less than, its nameplate rating. If operating at the upper end of its rated suction range, it is usually between its rated nameplate current and 40% above this value. A current in excess of 140% of the nameplate rating indicates a misapplication or something wrong with the motor, compressor, or some part of the system. The only time the current might exceed 140% of the motor nameplate rating, without something being wrong, is in the case of a low-temperature system during the pulldown period.

4.3 Examining the Performance of a Centrifugal Water Chiller

The following performance information on LP chillers with hermetically sealed motors is from Trane Document ST-SP-1.

> It is important to understand the performance of your centrifugal water chiller to obtain optimum machine efficiency. This includes monitoring the chiller operation to compare the characteristics of the unit when operating at a given (optimum) condition with the characteristics as a modified condition.
>
> The first step is to run an operating log for the machine, recording the performance data which include water temperatures and flows for both the evaporator and the condenser, the refrigerant temperatures and pressures, kW, etc.
>
> The next step is to run a heat balance to determine the accuracy of the instrumentation. This is done as follows:
>
> Evaporator Tons + kW = Condenser Tons
>
> (Where all terms are in consistent units)
>
> (kW is converted to Tons-Refrigeration energy units)

$$\frac{(T_{in} - T_{out})_{EVAP} \text{ x } GPM_{EVAP}}{24} + .2845 \text{ x } kW = \frac{(T_{out} - T_{in})_{COND} \text{ x } GPM_{COND}}{24}$$

As an example:	54ºF	water temperature entering the evaporator
	44ºF	water temperature leaving the evaporator
	240	evap GPM (gal/min)
	80	kW
	95ºF	water temperature leaving the condenser
	85ºF	water temperature entering the condenser
	300	cond GPM (gal/min)

Substituting this example data into the heat balance formula:

$$\frac{(54 - 44) \text{ x } 240}{24} + .2845 \text{ x } 80 \approx \frac{(95 - 85) \text{ x } 300}{24}$$

100 + 22.8 < 125

122.8 < 125 The difference in this example is 1.8% based on the condenser tons.

$$\frac{2.2 \quad (\text{Difference})}{125} \text{ x } 100\% = 1.8\%$$

At full load, it should be possible to obtain a heat balance of less than 7.5%. If a heat balance (at full load) is calculated to be in excess of 7.5%, then the instrumentation is suspect.

The next item to examine is the liquid refrigerant temperatures in the evaporator and condenser. The term “Approach Temperature” is used to measure the temperature difference between the liquid refrigerant temperature and the leaving water temperature. For example, if the condenser leaving water temperature is 95ºF and the liquid refrigerant temperature is 102ºF, then the approach temperature is 7ºF. The approach temperature is a function of machine tonnage (as tonnage increases, approach temperature also increases). The approach temperature is also affected by tube fouling, improper refrigerant charge (evaporator only) and other factors.

To determine if the chiller can be operated more efficiently, the operating parameters must be examined.

$$\text{kW} \quad \alpha \quad \frac{\text{Tons x Lift}}{(\text{Motor Efficiency}) \text{ x } (\text{Compressor Efficiency})}$$

Where “Lift” is the temperature difference between the liquid refrigerant temperature in the condenser and the liquid refrigerant temperature in the evaporator. This is an indication of the “head” that the compressor must produce.

Let us examine some things that can be done to possibly improve performance.

1. Reduce kW by chilled water reset. This would directly affect (reduce) the power required by the machine. Reset the temperature controller to increase the leaving chilled water temperature to 45°F instead of 44°F. This also would increase the evaporator refrigerant pressure and temperature, hence decreasing the lift, and resulting in lower compressor power requirements.

2. Reduce lift

   – Make certain that fouling is minimized (this is usually only applicable to condensers, because most evaporator water systems are closed systems, containing treated water). This fouling will act as an insulator to good heat transfer, thus causing the refrigerant condensing temperature to increase. Monitor the condenser approach temperature at the same tons. Clean the condenser tubes as necessary and treat the condenser water if using a tower system.

   – Make certain that no non-condensables are in the unit, as these collect in the condenser, causing the pressure in the condenser to be higher, and hence the compressor has to work harder to produce the higher pressure. Eliminate the leak and purge the non-condensables. (Monitor the amount of time required to purge the chiller when manually purging on a regular basis to see if a leak develops. More frequent requirements for purge operation are an indication of a leak.)

   – Make certain refrigerant charge is optimized. Too much refrigerant will result in evaporator carryover, and too little refrigerant will adversely affect the heat transfer in the evaporator (increasing the approach temperature, and hence increasing the lift).

   – Make certain that oil (from the lubrication system) is not in the refrigerant. This too adversely affects the heat transfer in the evaporator (increasing the approach temperature, and hence increasing the lift).

   – Do not use glycol in the evaporator water system unless necessary, and then only use the minimum percentage of glycol required for freeze protection. Glycol is not as good a heat transfer fluid as water. This increases the approach temperature, hence the evaporator temperature will be lowered, thus affecting lift, and hence power.

   – If the entering condenser water temperature can be reduced by turning on more tower fans, the chiller power will be decreased. Depending on the size of the tower, this can result in a lowering of the total power of the system. Note: Trane CenTraVacs start and operate satisfactorily over a range of load conditions with uncontrolled entering condenser water temperature. However, continuous machine operation with entering condenser water temperature below 65°F is not recommended. When entering condenser water temperature is expected to drop below 65°F, or if the temperature difference (leaving condenser liquid degrees F minus leaving evaporator liquid degrees F) is expected to drop below 35°F, some form of condenser water temperature control is required to insure satisfactory operation.

– Possibly increase the condenser GPM, particularly at full load. This will reduce the refrigerant condensing temperature, hence reducing lift, and therefore resulting in lower kW. Also the pressure drop through the condenser will be increased.

– Examine adding an auxiliary condenser. This will allow more condenser heat transfer surface which will lower the condensing temperature. It will also allow more condenser GPM to be used, resulting in a lowering of the refrigerant condensing temperature, and hence lower power. It may also be possible to use an auxiliary condenser for heat recovery application.

3. The motor efficiency is a function of the particular motor and the load. However, a capacitor can be used which affects the real versus the apparent power through a change in the power factor. This may reduce utility billing for demand charges. Trane has engineering literature which discusses recommended methods of using capacitors for power factor correction.

4. The compressor efficiency is a function of the particular compressor, the selection, and the operating conditions. When a centrifugal chiller selection is made, it is important to make certain that the chiller is selected for the expected duty. In other words, if the requirement is for 1000 tons of duty, make certain that the selection if made for 1000 tons, not 1200 tons. This will result in a wider range of operating capacity control and higher efficiency over the operating range of the chiller.

   When examining the possible purchase of a new chiller, examine the various options for improving performance to include larger heat exchangers, additional economizer, increased condenser GPM, etc. These will result in lower operating costs for the owner. Also examine the anticipated performance at part load, as this is where the chiller will operate much of the time.

In summary, things to be considered for improving the performance of a centrifugal water chiller are as follows:

- Examine chilled water reset
- Maintain clean tubes (clean tubes and treat the water)
- Maintain leak tight machine
- Optimize refrigerant charge
- Check for proper lubrication system operation
- Examine cooling tower performance for affect on chiller performance
- Examine increasing the condenser water GPM
- Examine adding an auxiliary condenser
- Examine power factor correction
- Examine optimum compressor/machine selection (for new machine). Also examine replacing an older machine with a more efficient machine

5
TROUBLESHOOTING

Frequently, 90% of troubleshooting time is spent on locating the malfunction and 10% in correcting it. This section provides an overview of a generic approach to troubleshooting as well as means to address common problems associated with chiller performance.

For more detailed troubleshooting information, the user should reference specific manufacturer's O&M instruction manuals.

5.1 Generic Troubleshooting Issues

5.1.1 Initiating System Troubleshooting

During the development of a troubleshooting plan, it is beneficial to develop a structured, methodical process to eliminate each of the possible influences to the problem. Some of the causes of problems can be eliminated because their effects are typically specific, as can be seen in Tables 5-1 through 5-13. Even considering this, it is worthwhile to develop an all-inclusive checklist of items that can impact system performance. Parameter monitoring is a key to this process; it allows for easier identification of possible impacts on system performance.

Key O&M Cost Point

A structured approach to chiller troubleshooting reduces time and improves the effectiveness of the troubleshooting efforts.

5.1.2 Determining and Validating Operating Conditions

The next step in the troubleshooting process is to determine which operating conditions should be recorded and which parameters must be measured, if any readings other than data taken, and how that information will be validated. Measurements should be taken using calibrated instruments and should be reviewed for consistency against system design basis documents/outputs.

5.1.3 Comparison to Design Requirements/Historical Performance

The next step in the troubleshooting process is to compare the measured parameter(s) against the most recent or historical operating conditions (similar to those items listed in Tables 4-1 through 4-3). The troubleshooter should try to detect trends in performance. To ensure that data obtained are correct, the instrumentation used should have its accuracy validated.

However, if the comparison reveals that a number of parameters have changed, or that the changes are following a trend and degrading over time, then further investigation is warranted.

The measured data should be compared against maintenance history and design requirements that can be found in documents such as:

- Compressor curves
- System configuration drawings
- Vendor baseline data
- Component data sheets
- Vendor technical manuals
- Parts and materials lists
- Component upgrades
- Industry-wide historical operating experiences such as Equipment Performance Information Exchange, 10CFR21 reports (see Appendix C for examples of actual plant operating experience)
- Work maintenance history

These design documents might not provide all of the design information related to the system in which the components are installed. As such, the following sources of design information should also be considered:

- Heating ventilating and air conditioning (HVAC) system design calculations
- HVAC system descriptions
- Design basis documents
- Materials management information system
- Component/system technical specifications
- Component procurement specifications
- Final Safety Analysis Report
- Component assembly drawings

The engineer should consider reviewing the equipment history and should examine the data trends from the performance monitoring and PM logs to determine if the change has been a sudden or gradual one. This includes a review of recent preventive and corrective work orders for conditions and work performed to include filter changes and vibration readings. A review of recent performance testing results on related equipment can also be helpful.

In cases where multiple causes exist for a given system problem, system adjustments should be done methodically and documented in order to provide a clear indication of the effect each parameter has on the overall system performance. The actual determination of the root cause is an iterative process.

5.1.4 Evacuating System Refrigerant

In some instances, it can become necessary to fully evacuate the system of refrigerant. Federal codes and industry documents (for example, 40CFR82, Air-Conditioning Refrigeration Institute [ARI] 740, ARI Guideline K, and ARI Guideline Q) provide guidance for system evacuation, refrigerant recovery and recycling equipment, proper refrigerant containers, and content recovery and recycling of refrigerant containers.

5.1.4.1 Refrigerant Evacuation

After the system is entirely free of leaks, it should be completely evacuated. To thoroughly evacuate a system, connect evacuating lines to the high and low sides of the system. These lines should be properly sized according to the manufacturer or accepted industry guidelines. Run the vacuum pump until an acceptable pressure is reached, then stop the pump and break the vacuum with nitrogen. Repeat this procedure as many times as necessary to ensure that the refrigerant is as near to completely evacuated from the system as possible. If the system contains excessive moisture, the use of an ice-bath dehydrator (freeze trap) should be used to increase the evacuation process. The vacuum can then be broken by charging in vapor of the refrigerant to be used in the system. Remove the vacuum pump and proceed to charge the system to the recommended manufacturer charge.

Key Technical Point

Never draw a deep vacuum on a refrigerant system that contains liquid refrigerant.

Key Human Performance Point

Never operate or megger a hermetic motor while the system is in a deep vacuum.

5.1.4.2 Refrigerant Charging

The recommended procedure for field charging is to connect the charging line between the refrigerant cylinder and gauge port of the suction shutoff valve or process connection, with the suction gauge also connected. If factory-filled cylinders are not being used when charging refrigerant into a system, charge the refrigerant into the system through a dryer. Service cylinders can contain objectionable amounts of moisture. Charge the entire system with gas until the pressure is at the saturated refrigerant temperature above 32°F to prevent freezing. The compressor can then be operated and the valve on the refrigerant cylinder throttled to control the suction pressure at approximately normal operating pressure.

Key Technical Point

To prevent damage, never allow liquid refrigerant to enter the suction line to the compressor.

Key Technical Point

Service or recovery cylinders that are not properly evacuated prior to use can contain objectionable amounts of moisture.

5.1.5 Excess Noise

Excessive noise in the compressor might be due to some abnormal condition outside the compressor or to something defective or badly worn within the compressor. If it is due to some cause outside the compressor, nothing is gained by changing the compressor. External causes should be checked and corrected to dispel the assumption that a compressor is damaged or prior to repairing a damaged compressor. Therefore, before changing or rebuilding a compressor, check for these possible causes of excessive noise:

- Liquid slugging: Make sure that only superheated vapor is entering the compressor.
- Oil slugging: Oil can be trapped in the evaporator or suction line and intermittently return via slugs to the compressor.
- Loose flywheel: On belt-driven units, the flywheel on the compressor shaft might be loose.
- Improperly adjusted compressor mounting: With externally mounted hermetic-type compressors, the feet of the compressor can bump the studs, the holddown nuts may not be backed off sufficiently, or the springs may be too weak, allowing the compressor to bump against the base.
- Broken turning or prerotation vanes.
- Surging in a centrifugal compressor.
- Loose evaporator or condenser baffle plates.
- Rotary screw rotor float.
- Discharge check valve rattle.

Compressor noises from internal sources include the following:

- Insufficient lubrication
- Excessive oil level
- Overtightened piston or bearing
- Defective internal mounting
- Loose bearings

- Damaged valves
- Loose rotor or eccentric
- Vibrating discharge valves
- Gas pulsation
- Improperly charged or contaminated refrigerant
- Loose, misaligned components
- Tight internal components

5.2 Detailed Troubleshooting

5.2.1 Refrigerant System Troubleshooting

Tables 5-1 through 5-13 provide guidance in the troubleshooting of refrigerant systems. They are intended to supplement site-generic troubleshooting guidelines and corrective action programs in order to expedite the resolution of performance concerns. Information similar to that contained in the tables may also be included in the plant vendor manuals specific to the equipment.

Table 5-1: Compressor Short Cycles

Table 5-2: Compressor Runs Continuously

Table 5-3: Compressor Fails to Start

Table 5-4: Compressor Is Noisy

Table 5-5: Compressor Oil Pressure/Level Problems

Table 5-6: Capacity Control Issues

Table 5-7: Discharge Pressure Too Low

Table 5-8: Discharge Pressure Too High

Table 5-9: Suction Pressure Too Low

Table 5-10: Suction Pressure Too High

Table 5-11: System Short of Capacity

Table 5-12: Performance Problems Specific to Centrifugal Chillers

Table 5-13: Miscellaneous Chiller Performance Problems

Table 5-1
Compressor Short Cycles

Trouble	Probable Cause	Recommended Action
Compressor cycles on LP control.	LP control erratic in action.	Raise LP control differential setting.
		Check capillary for restrictions.
		Replace control.
	Compressor suction valve leaking.	Replace valve.
	Compressor suction shutoff valve partially closed.	Open valve.
	Low refrigerant charge.	Repair refrigerant leak(s) and add refrigerant.
	Plugged compressor suction strainer.	Clean or replace strainer.
Compressor cycles on HP control.	HP control erratic in action.	Check capillary tube for restrictions.
	Noncondensables in system.	Remove noncondensables.
	Compressor discharge valve partially closed.	Open valve or replace if defective.
	Condenser scaled.	Clean condenser.
	Condenser water pump or fans not operating.	Start pump or fans, repair or replace if defective.
Compressor will not load or unload.	See Table 5-6, "Capacity Control Issues."	

Table 5-2
Compressor Runs Continuously

Trouble	Probable Cause	Recommended Action
Low process or conditioned area temperature.	Refrigerant system or room temperature control malfunction or set incorrectly.	Reset or replace temperature control, if defective.
		Check surrounding influence on temperature controller.
	Welded contacts on control in motor starter circuit.	Replace contacts.
High process or conditioned area temperature.	Leaky valves in compressor.	Repair or replace compressor valves.
	Low refrigerant charge.	Check and repair leaks and add refrigerant.

Table 5-3
Compressor Fails to Start

Trouble	Probable Cause	Recommended Action
No voltage on line or load side.	Voltage present on line side of fuse. No voltage on motor side.	Check for an open disconnect switch or a blown fuse and correct. Check load on compressor.
	Motor starter nonfunctional.	Check motor starter contacts. Check holding coil to determine if it is burned out. Repair or replace motor starter.
Compressor does not run.	Frozen compressor. Locked up or internally damaged.	Repair or replace compressor.
	Thermostat set too high.	Reset thermostat.
	Thermal overload switch open.	Reset switch.
	Dirty contacts.	Clean all control contacts.
	Motor burned out.	Replace or repair compressor motor.
LP switch open. Suction pressure too low.	Low refrigerant charge.	Check system charge. Repair leak and recharge.
HP switch open.	Discharge pressure too high.	See Table 5-8, “Discharge Pressure Too High.”
Oil pressure switch open. System restarts on reset of oil pressure control.	Oil pressure or oil level too low or oil pressure switch malfunction.	Check oil pressure and level. Test switch. Replace switch if needed.
Flow switch contacts open.	Restricted or no water flow. Flow switch malfunction.	Restore air or water flow. Test flow switch. Repair or replace if needed.

Table 5-4
Compressor Is Noisy

Trouble	Probable Cause	Recommended Action
Compressor stops. Oil pressure switch open.	Insufficient oil charge.	Check oil levels and add oil if needed.
	Intermittent oil pressure switch.	Repair or replace switch.
	Oil migration from oil sump.	Recover oil back to sump.
Compressor knocks, too frequent starting.	Internal compressor damage.	Repair or replace compressor.
Excessively cold suction line due to liquid slugging back to compressor.	TEV set incorrectly.	Check and adjust superheat.
	TEV not properly installed.	Inspect sensing bulb. Clean, tighten, or reinsulate and ensure proper installation bulb.
	TEV stuck open.	Clean, repair, or replace valve.
Driveline/transmission noise.	Loose or misaligned coupling.	Check alignment and tightness.
	Loose or misaligned belts.	Check alignment and tension. Belt slack should be at the top.
Internal compressor noise.	Motor or compressor bearings worn.	Replace bearings.
	Hydraulic knock due to excess oil in circulation.	Remove excess oil. Check expansion valve for floodback.
High vibration.	Loose foundation bolts or holddown bolts.	Tighten bolts.
Pipe rattle.	Inadequately supported piping or loose pipe connections.	Support pipes or check pipe connections.
Hissing.	Insufficient refrigerant flow through expansion valves.	Add refrigerant.
	Clogged liquid line strainer.	Clean.

Table 5-5
Compressor Oil Pressure/Level Problems

Trouble	Probable Cause	Recommended Action
Excessively cold suction line due to liquid slugging back to compressor. (Refrigerant carries oil out of compressor.)	TEV set incorrectly.	Check and adjust superheat.
	TEV not properly installed.	Inspect sensing bulb. Clean, tighten, or reinsulate and ensure proper installation of bulb.
	TEV stuck open.	Clean, repair, or replace valve.
Oil level too low.	Insufficient oil charge.	Check oil levels and add oil if needed.
	Oil return check valve or float valve stuck closed.	Repair or replace check valve.
Oil level gradually drops.	Plugged liquid line filter dryer.	Replace filter dryer core.
	Defective TEV.	Repair or replace valve.
Oil evident on and around compressor.	External fittings or lines leaking.	Repair leaks and recharge system.
	Compressor mechanical seal leak.	Replace seal.
Oil leaves sight glass rapidly.	Defective unloader O-rings.	Repair or replace compressor unloader.
Inadequate or no oil pressure.	Insufficient oil charge.	Add oil to proper system charge.
	Leak in oil line into another part of refrigerant system.	Repair leaks and recover oil to oil sump.
	Faulty oil gauge.	Check and repair or replace.
	Defective oil pressure regulator.	Repair or replace.
	Clogged oil suction strainer.	Clean.
	Broken oil pump tang.	Replace pump assembly.
	Clogged oil line.	Remove obstruction.
	Worn oil pump.	Replace pump assembly.
	Worn compressor bearings.	Replace bearings.

Table 5-6
Capacity Control Issues

Trouble	Probable Cause	Recommended Action
Compressor does not load.	Insufficient oil pressure.	See Table 5-5, "Compressor Oil Pressure/Level Problems."
	Capacity control valve nonfunctional.	Repair or replace.
	Broken or loose pressure line to power element.	Repair.
	Plugged pressure line to power element.	Clean out.
	External adjusting stem damaged.	Replace.
	Control oil strainer blocked.	Clean or replace.
	Temperature sensing device insufficient feedback.	Repair or replace.
	Improper adjustment or attachment of actuator or linkage to prerotation vanes.	Tight or adjust.
	Failure of control module.	Replace.
Compressor does not load or unload.	Nonfunctional compressor unloading system.	Repair or replace faulty unload components.
	Unit in low load-limit mode.	Wait for load to recover.
	Suction valve closed or throttled.	Open valve.
	Thermostat differential too narrow.	Reset thermostat.
	Unit in high load-limit mode.	Check high load-limit setting and raise or correct cause for excessive load.
Rapid unloader cycling.	Oversized TEV causing excessive fluctuation in suction pressure.	Resize TEV.
	Partially plugged control oil strainer.	Clean or replace.
	Insufficient oil pressure.	See Table 5-5, "Compressor Oil Pressure/Level Problems."

Table 5-7
Discharge Pressure Too Low

Trouble	Probable Cause	Recommended Action
Insufficient temperature differential through condenser.	Excessive water flow through condenser.	Adjust flow to obtain design pressure drop.
	Condenser TCV not controlling flow.	Check TCV adjustment. Adjust, repair, or replace valve.
Bubbles in liquid line glass.	Low refrigerant charge.	Check system charge. Repair leak and recharge.
	Leaky compressor suction valve.	Examine valve disks and valve seats. Replace if worn.
	Worn compressor components.	Repair or replace compressor.
Water entering condenser too cold.	Thermostat out of adjustment. Can cause lower condenser pressure that then can cause problems with lower refrigerant flow (for example, surging in compressor).	Readjust thermostat. Decrease water flow. Resolve possible fluctuations in condenser water supply system due to other loads on system. Determine ultimate heat sink temperature. Increase hot-gas bypass. Increase refrigerant flow rate.

Table 5-8
Discharge Pressure Too High

Trouble	Probable Cause	Recommended Action
Excessively hot condenser.	Refrigerant overcharge.	Slowly remove refrigerant to obtain proper subcooling reading.
	Noncondensable gas in system.	Remove contaminant from the system.
	Condenser tubes fouled.	Clean condenser.
	Condenser water bypassing condenser tubes.	Locate components that bypass water around condenser tubes, such as missing gasket on partition plate in channel head, partition plate partly corroded away, and corroded condenser diverter valve. Repair or replace defective components.
Condenser inlet water temperature too high.	Condenser temperature control valve not controlling flow.	Check TCV adjustment. Adjust, repair, or replace valve.

Table 5-9
Suction Pressure Too Low

Trouble	Probable Cause	Recommended Action
Bubbles in liquid line glass.	Low refrigerant charge.	Check system charge. Repair leak and recharge.
No refrigerant flow through TEV.	TEV power element charge lost or liquid line solenoid valve failed closed.	Replace valve.
Loss of system capacity.	Condenser water supply too cold, due to other components in system or fluctuations in condenser water temperature.	Eliminate fluctuations. Check condenser water temperature.

Table 5-10
Suction Pressure Too High

Trouble	Probable Cause	Recommended Action
Excessively cold suction line due to liquid slugging back to compressor.	TEV overfeeding.	Check and adjust superheat. Check valve sensing bulb for good contact.
	TEV nonfunctional (open).	Repair or replace valve.
Compressor runs continuously.	Evaporator overloaded.	Check for excessive makeup air, increased transmission load into room, or increased heat loads within room.
Compressor noisy. Leaving chilled water temperature too high.	Compressor suction valves damaged.	Remove head, inspect valves, and replace damaged valves.
Loss of system capacity.	Condenser water supply too hot, due to other components in system or fluctuations in condenser water temperature.	Eliminate fluctuations. Check condenser water temperature.

Table 5-11
System Short of Capacity

Trouble	Probable Cause	Recommended Action
Superheat reading high. Suction pressure unstable.	Flash gas in liquid line.	Check refrigerant charge and add if needed.
Compressor short-cycling.	TEV sticking or partially blocked.	Repair or replace valve.
Oil foaming in compressor. Leaving chilled water temperature too high or too low.	TEV adjusted incorrectly.	Check and adjust superheat.
Insufficient water pressure drop across chiller.	Reduced water flow. Water supply problem or obstruction in water line.	Restore water supply or remove obstruction from water line.
Superheat reading high.	Excessive superheat.	Reset TEV superheat setting.
Loss of system capacity.	Condenser water supply too hot, due to other components in system or fluctuations in condenser water temperature.	Eliminate fluctuations. Check condenser water temperature.

Table 5-12
Performance Problems Specific to Centrifugal Chillers

Trouble	Probable Cause	Recommended Action
High evaporator pressure.	Temperature controller not controlling.	Check and correct set point.
	Noncondensables in refrigerant.	Check for in-leakage. Repair leaks. Remove contaminant from the system.
	High refrigerant charge.	Check refrigerant charge and correct if needed.
	Evaporator tube leak.	Check for presence of water in refrigerant and repair tube leak.
Low evaporator pressure.	Temperature controller fails low.	Adjust controller.
	Low refrigerant charge.	Check and repair leaks and add refrigerant.
	Insufficient refrigerant flow.	Investigate cause and correct low refrigerant flow.
	Excessive oil in evaporator.	Check oil return system and low oil sump level. Remove excessive oil.
		Increase load on chiller.
		Check oil addition log for amount excess oil and remove from system.
Chilled water temperature fluctuates.	Prerotation vanes hunt/temperature load controller throttling range too narrow.	Increase throttling range.
	Defective temperature load controller.	Repair or replace temperature load controller.
Chilled water temperature higher than set point.	Prerotation vanes fail to open.	Check vane motor and control circuit. Repair or replace.
Oil reservoir/sump temperature too low.	Oil cooler coolant flow too high.	Adjust oil cooler water flow.

Table 5-12 (cont.)
Performance Problems Specific to Centrifugal Chillers

Trouble	Probable Cause	Recommended Action
Oil reservoir/sump temperature too low. (cont.)	Oil heater not maintaining temperature (when chiller in standby).	Check oil heater circuit. Repair or replace oil heater or control.
	Excessive refrigerant in oil.	Increase chiller load or return chiller to design operating conditions.
Oil reservoir/sump temperature too high.	Oil cooler coolant flow too low.	Adjust oil cooler water flow.
	Oil heater not maintaining temperature (when chiller in standby).	Check oil heater circuit. Repair or replace oil heater or control.
	Oil cooler tubes fouled.	Clean tubes or replace cooler.
	Oil level too high.	Remove excess oil.
	Bearing failure.	Check bearing by vibration testing and replace bearing.
	Low oil flow.	Check oil pressure control, oil filter, or oil pump performance. Repair or replace component.
Compressor surging (refrigerant flow stalled due to compressor unable to develop discharge pressures greater that the pressures in the condenser).	Noncondensables in condenser.	Check purge unit operation and check for refrigerant boundary leaks.
	High condenser pressure.	Check for condenser water temperature too high or flow inadequate.

Table 5-13
Miscellaneous Chiller Performance Problems

Trouble	Probable Cause	Recommended Action
Frosted liquid line.	Receiver shutoff valve partially closed or restricted.	Open valve or remove restriction.
	Restricted filter dryer.	Remove restriction or replace filter dryer.
Freeze-up.	Improperly set safety thermostat.	Check safety thermostat for proper setting at beginning of each season.
	Operating with safety thermostat bypassed.	If thermostat was bypassed for checking, be sure it is back in the circuit before starting unit.
	Improper circulation of chilled water.	Correct low flow conditions. Remove blockages. Chemically treat the water to prevent formation of deposits.
Low chilled water flow.	Flow obstruction.	Check for condenser tube blockage.
	Throttle valve closed.	Open valve.
	Pump flow inadequate.	Check pump for proper flow.
High chilled water flow.	Throttle valve open.	Turn throttle valve in the closed direction to obtain desired results.
High chilled water differential pressure.	Throttle valve open.	Turn throttle valve in the closed direction to obtain desired results.
	Possible tube blockage.	Check for condenser tube blockage.
High condenser pressure.	Chiller load too high for condenser water temperature.	Adjust load or condenser water temperature. Increase process flow.
	Incorrect refrigerant charge.	Check refrigerant charge and correct if needed.
	In-leakage from process. Noncondensables in refrigerant.	Check for in-leakage.

Table 5-13 (cont.)
Miscellaneous Chiller Performance Problems

Trouble	Probable Cause	Recommended Action
Low condenser pressure.	Incorrect refrigerant charge.	Check refrigerant charge and correct if needed.
	Leak in condenser.	Check for in-leakage.
High chilled water temperature.	High evaporator pressure.	Check refrigerant charge and correct if needed.
	Temperature control loop fails high.	Test/repair control loop.
	Condenser water temperature too high.	Adjust controller. Adjust load or condenser water temperature.
Low chilled water temperature.	Low evaporator pressure.	Check refrigerant charge and correct if needed.
	Temperature control loop fails low.	Test/repair control loop.
	Low chilled water flow.	Adjust controller. Adjust load or condenser water temperature.
	Defective or improperly set TEV.	Increase superheat. Valve operation must be stable (no hunting).

5.2.2 Thermostatic Expansion Valve Troubleshooting

A superheat measurement is the initial step in correctly determining whether or not a TXV is performing its function.

A superheat calculation is accomplished in four simple steps:

1. Check temperature of suction gas at bulb location.
2. Measure suction pressure.
3. Convert suction pressure to saturation temperature.
4. Subtract saturation temperature from suction gas temperature.

Suction gas temperature is determined by cleaning an area of the suction line close to the remote bulb and taping a thermocouple sensing unit, a resistance temperature detector (RTD), or an accurate thermometer to the line at the cleaned area. The temperature measurement device is insulated to prevent its being influenced by ambient temperature. The reading obtained is the suction gas temperature.

Suction pressure can be measured directly at the evaporator outlet if a gauge fitting has been provided, or an accurate gauge can be connected into the external equalizer line with a line-piercing valve. Suction pressure can also be measured at the compressor suction valve port on package or other close-connected installations. When the measurement is taken in this manner and the compressor is more remote from the evaporator, a normal pressure drop factor must be added to the pressure at the compressor.

Determination of actual superheat is the important first step of troubleshooting a refrigerant flow problem that might involve the thermostatic expansion valve itself. The next step is to check valve data to learn how much superheat is normal. If design or trend data for the superheat value are not available, use vendor-supplied information. If actual superheat is too high, the problem can be analyzed as insufficient refrigerant flow into evaporator. If actual superheat is too low, the problem is too much refrigerant being fed to the evaporator.

While both of these conditions could be the fault of the TXV, the real trouble is frequently found elsewhere in the system. Unfortunately, many TXVs are needlessly replaced before further troubleshooting reveals the actual source of the problem.

If the TXV is operating in an erratic manner, the valve’s temperature sensing bulb should be checked for proper location. The specific valve manufacturer’s installation recommendations should be followed.

If this information is not available, the following guidelines can be used:

- If the bulb is attached to a suction line in a vertical orientation, the bulb should be below the capillary tubing attached to the bulb.
- If the bulb is attached a suction line in the horizontal orientation, the bulb should not be in a position that it will sense the oil flowing in the line, normally in the 6 o’clock position.
- For suction lines less than 7/8 in., the bulb should be placed in the 12 o’clock position.
- For suction lines 7/8 to 1-5/8 in., the bulb should be located in the 10 or 2 o’clock position.
- For suction lines larger than 1-5/8 in., the bulb should be located in the 4 or 8 o’clock position.

Key Technical Point

Ensure that uniform air or chilled water flow across or through the evaporator and that adequate load on the evaporator is available when reviewing causes of erratic thermostatic expansion valve operation.

6
PREVENTIVE MAINTENANCE

6.1 Template for Preventive Maintenance

Preventive maintenance is an exceptional method by which problems with refrigerant systems can be averted. This section provides supplemental guidance for the responsible site personnel to develop or enhance a PM program. Table 6-1 is a template for the parameters and components that should be included in a PM program for refrigerant systems.

Table 6-1
Refrigerant System Preventive Maintenance Template

			1	2	3	4	5	6	7	8
	Critical	**Yes**	X	X	X	X				
		*** No**					X	X	X	X
	Duty Cycle	**High**	X		X		X		X	
		Low		X		X		X		X
	Service Condition	**Severe**	X	X			X	X		
PM Task		**Mild**			X	X			X	X
Condition Monitoring										
Refrigerant analysis	See PM Application Note 6.4.1		1Y	1Y	1Y	1Y	1Y	1Y	1Y	1Y
Glycol analysis	See PM Application Note 6.4.2		6M	6M	6M	6M	6M	6M	6M	6M
Oil analysis	See PM Application Note 6.4.3		1Y	1Y	1Y	1Y	1Y	1Y	1Y	1Y
Vibration analysis	See PM Application Note 6.4.4		1Y	1Y	1Y	1Y	1Y	1Y	1Y	1Y
Calibration	See PM Application Note 6.4.5		2Y	2Y	2Y	2Y	AR	AR	AR	AR
Time Directed										
External visual inspection	See PM Application Note 6.4.6		3M	3M	3M	3M	6M	6M	6M	6M
Internal inspection: Heat exchangers	See PM Application Note 6.4.7		2Y	2Y	2Y	2Y	2Y	2Y	2Y	2Y
Internal inspection: Compressors	See PM Application Note 6.4.8		10Y	10Y	10Y	10Y	10Y	10Y	10Y	10Y
Failure Finding										
System testing: Performance tests	See PM Application Note 6.4.9		2Y	2Y	2Y	2Y	AR	AR	AR	AR
System Testing: Functional tests	See PM Application Note 6.4.10		2Y	2Y	2Y	2Y	AR	AR	AR	AR

* This template does not apply to the run-to-failure (RTF) components; noncritical here means not critical but important enough to require some PM tasks.

6.1.1 Template Notes and Definitions

Refer to specific application notes for each PM task.

AR: As required
M: Months
Y: Years

6.1.2 General Notes

Note 1:
Tasks with intervals stated in years or months should be performed as close as possible to the intervals indicated in the template unless specific means are used to add confidence that a more extended interval can be used (for example, visual inspection, the use of condition monitoring, maintenance history, and as-found conditions).

Deferral of any task requires an evaluation. One evaluation method is to compare, using a sampling process, the candidate equipment's maintenance history and as-found conditions to those of other components with similar specifications and operating conditions.

Note 2:
If the component operates in severe service conditions, the plant-specific conditions must be considered to select appropriate intervals.

Note 3:
If there are plant-specific conditions (that is, one or more columns) for which no PM task is appropriate, this is considered to be RTF, a maintenance option for only those noncritical components that meet all of the following conditions:

- The component is not required for vital system redundancy.
- The component's failure does not promote failure of other components.
- There is no increased personnel radiation exposure if the component is RTF.
- It is more cost-effective to repair or replace the component than to perform PM.
- There is no simple cost-effective task to maintain the component.

Note 4:
After completion of any action that could affect component function, it is prudent to verify that the component operates in an acceptable manner, whether or not this is required by applicable regulations.

Note 5:
Existing technical specifications and other regulatory requirements, for example, American Society of Mechanical Engineers (ASME) Section XI and other licensing commitments, should always be followed. Moreover, there are parts of ASME Section XI that can be used to trend condition monitoring data such as vibration levels and bearing temperatures. If the recommendations in Table 6-1 differ from these regulations, the more conservative approach should be followed. Appropriate task interval determination could lead to recommendations for intervals that differ from the existing regulations. In such cases, there might be a basis for seeking changes to the regulations.

Note 6:
Overhaul is a task performed for large compressors according to the manufacturer's recommendations. Overhaul principally addresses failures of:

- Gaskets
- Aging of the oil pump
- Worn bearings
- Worn couplings

In addition, it covers:

- Damaged rotary screw elements for rotary screw compressors
- Worn bearing journals
- A failed shaft seal
- A cracked or scored shaft
- A worn or failed unloader valve
- Loose mounting bolts
- A cracked frame on the reciprocating compressors

6.2 Definitions of Template Application Conditions

Critical:

Yes Functionally important, for example, risk significant, required for power production, safety related, or other regulatory requirements.

No Functionally not important, but economically important for any of the following reasons: high frequency of resulting corrective maintenance, more expensive to replace or repair than to perform PM, or high potential to cause the failure of other critical or economically important equipment.

Duty Cycle:

High Continuous operation.

Low Alternated between periods of standby and continuous operation.

Service Condition:

Severe High or excessive humidity, excessive temperatures (high/low) or temperature variations, excessive environmental conditions (for example, salt, corrosive, high radiation, spray, and steam), and high vibration.

Mild Clean area (not necessarily air-conditioned), temperatures within OEM specifications, and normal environmental and process conditions.

6.3 Examples of Components That Satisfy Template Conditions

1.

Critical High duty cycle Severe service condition	Refueling water storage tank chillers

2.

Critical Low duty cycle Severe service condition	No example given

3.

Critical High duty cycle Mild service condition	Control room chillers Switchgear room chillers Dry well chillers

4.

Critical Low duty cycle Mild service condition	Alternate shutdown panel room chillers

5.

Noncritical High duty cycle Severe service condition	Main station chillers

6.

Noncritical Low duty cycle Severe service condition	No example given

7.

Noncritical High duty cycle Mild service condition	Auxiliary building chillers Turbine building chillers

8.

Noncritical Low duty cycle Mild service condition	Office building chillers

Note: Range of service conditions (for example, inside or outside) influences these assignments.

6.4 Preventive Maintenance Application Notes

6.4.1 Refrigerant Analysis

Failure Locations and Causes:
Refrigerant analysis tests the refrigerant for the presence of moisture, acid, and oil. The task is important to detect conditions that lead to oil return system failures on centrifugal compressors, whether from a plugged eductor, inadequate system differential pressure, or other component failure. Refrigerant analysis also addresses gasket failures, excessive or ineffective purging of noncondensable gases, and plugged refrigerant metering devices.

Progression of Degradation to Failure:
All of the failure causes for the oil return system contribute to making this a dominant failure mechanism for chiller units with centrifugal compressors, but only the plugged eductor and inadequate system differential pressure have random occurrences that can affect the interval for this task. Leaking gaskets, and the several other causes of failure in the oil return system, are expected on time scales significantly longer than 1 year. The plugged refrigerant metering device and excessive or ineffective purging also have random occurrences and contribute to the chance of a failure on a time scale of 1or 2 years.

Fault Discovery and Intervention:
Refrigerant analysis is recommended at an interval of 1 year to control failures from the above causes. The only practical backup to this task for the oil return system degradations is in the form of observations during operator rounds.

6.4.2 Glycol Analysis

Failure Locations and Causes:
This task is only applicable when a glycol cooling fluid is used, and it addresses the change in fluid composition that results from fluid leaks. If this fluid leak goes unchecked, glycol fouling of the condenser heat exchanger is the result.

Progression of Degradation to Failure:
Leakage that is the precursor to this failure mechanism has a random occurrence pattern on a scale of 1 to 2 years.

Fault Discovery and Intervention:
Glycol analysis at a 6-month interval provides assurance of system integrity and is expected to detect degradation before a system failure.

6.4.3 Oil Analysis

Failure Locations and Causes:
Oil analysis is performed on compressor oil regardless of the type of compressor, and it measures degradation of the oil, presence of particulates, moisture content, and acid content. Oil analysis reveals incorrect or degraded oil as well as wear particles from worn bearings in all types of

compressors. Scored shafts and worn journals can also be detected, most likely in reciprocating compressors subject to impulsive mechanical stresses.

Progression of Degradation to Failure:
None of the above are dominant degradation mechanisms, although many of them have random failure occurrence characteristics that provide a significant likelihood of failure after a few years, and which therefore require detection relatively frequently.

Fault Discovery and Intervention:
An interval of approximately 1 year provides an adequate degree of protection, although larger compressors used for service air and instrument air have oil sampling more frequently than this. During the oil analysis for chiller compressors, efforts must be made to protect the oil samples from contamination by atmospheric moisture. In addition to altering the moisture content, exposure to the atmosphere results in the formation of additional acid. The analysis laboratory must be specifically experienced in chiller oil analysis and in providing the treatment and storage of chiller oil samples.

6.4.4 Vibration Analysis

Failure Locations and Causes:
Vibration analysis is focused on the detection of wear and misalignment in bearings and couplings on all types of compressors, and of worn pulleys, belts, and sheaves, and loose mounting bolts or a cracked frame on reciprocating compressors. Worn gears on centrifugal compressors and worn screw elements on rotary screw compressors can also be detected.

Progression of Degradation to Failure:
None of these are dominant degradation mechanisms, although many of them have random failure occurrence characteristics that provide a significant likelihood of failure after a few years.

Fault Discovery and Intervention:
An interval of approximately one year should provide an adequate degree of protection for all compressors.

6.4.5 Calibration

Failure Locations and Causes:
Calibration addresses drift, misadjustment, leaks, loose connections, and component failures of chiller controls such as refrigerant freeze protection and high and low refrigerant pressure, oil level and pressure, and capacity controls. Individual instruments or controls include pressure switches, relays, electrical or electronic transmitters, TCVs, programmable logic controls, level and flow switches, RTDs, I-to-P transducers, solenoid valves, and pneumatic devices.

Progression of Degradation to Failure:
Many of these causes of drift in instrumentation and controls are random on relatively short time scales of 1 to 2 years. There are many failure causes and a relatively high proportion of control

devices in HVAC systems. It is not surprising that control failures are dominant degradation mechanisms for this equipment.

Fault Discovery and Intervention:
An interval of 2 years was thought to provide sufficient protection from instrument drift. For the majority of the instruments and controls, calibration is the only means available of performing PM. Failure to calibrate is usually equivalent to RTF for this equipment, except for capacity, oil, and refrigerant controls that are also subject to performance testing.

6.4.6 External Visual Inspection

Failure Locations and Causes:
The external visual inspection is focused on the detection of leaks of water, oil, air, or refrigerant, and the verification of operating parameters, including system capacity and control in the form of local area temperatures and pressures. Important parameters are condenser water-side pressure and flow rate, TCV position, and refrigerant temperature, which all assist in detecting water-side fouling of condenser tubes. External leaks are most likely from failed shaft seals, failed gaskets, pneumatic control device tubing and fittings, and refrigerant actuated control valves. Additional significant items problems are sticking of the refrigerant actuated control valves and binding of prerotation vanes on centrifugal compressors.

Progression of Degradation to Failure:
Water-side fouling of condenser tubes, leaking pneumatic fittings, and leaking gaskets all occur randomly. Shaft seal leakage appears to have a failure-free period of 5 to 10 years, although the period is stated to be much shorter for rotary screw compressors (2 years). Control malfunctions, often detectable by observation of area temperatures and pressures, are dominant failure mechanisms that occur randomly and relatively frequently.

Fault Discovery and Intervention:
Fouling of the condenser tubes, various forms of leaks, and malfunction of control devices are all additional dominant degradation mechanisms. Several of these are also detectable during operator rounds, but they are not readily detectable by any other PM tasks. These include stuck or leaking refrigerant-actuated control valves, leaking shaft seals, clogged oil lines or oil filters, and a failed separator oil heater. Therefore, external visual inspection is of significant importance for chiller equipment, both for the compressors and for other components. Inspection every 3 to 6 months is recommended, depending on the criticality of the equipment. Operator rounds and external visual inspection are especially important for detecting system malfunctions in noncritical equipment because these will generally not also be covered by calibration or system testing of any kind. For critical equipment, operator rounds and external visual inspection also provide additional opportunities to detect control malfunctions beyond those provided by system testing and calibration tasks.

External visual inspection should include the following activities:

- Verify the general condition and performance of the unit.
- Inspect for loose, missing, or damaged hardware, parts, and components.

- Verify that the superheat and subcooling are within specifications.
- Verify that the refrigerant control valve is within its expected operating range.
- Ensure that the refrigerant level is correct and that there is no evidence of leakage.
- Ensure the cooling water flow rate to the heat exchanger is correct and that the inlet and outlet temperatures and pressures are within specification.
- Verify that the pressure relief device is closed with no evidence of leakage.
- Verify that the purge unit is seated and not venting.
- If present, verify that the refrigerant-operated control valve is operating properly.
- Inspect all pneumatic devices for evidence of leaks and loose or damaged air supply tubing.
- Verify that all lubrication levels are correct and that there is no evidence of leakage.
- Inspect all compressor/motor couplings for wear and proper alignment.

On centrifugal compressors:

- Verify that the oil pump is operating correctly and providing specified pressures.
- Inspect the oil lines for damage.
- If applicable, verify that the oil level is correct.
- Ensure that the capacity control (prerotation vanes) are free and not bound.
- Verify that the shaft seal and all gaskets show no evidence of leakage.
- Verify proper operation of the separator oil heater.

On reciprocating compressors:

- Inspect cylinders and heads for evidence of cracking.
- Verify that the shaft seal and all gaskets show no evidence of leakage.
- Verify that the unloader and relief valves are operating correctly.
- Inspect belts and sheaves for evidence of wear and misalignment.
- Verify that the oil level is correct.
- Verify that the oil pressure is correct.
- Verify that the oil filter or screen is not clogged and is free of debris, and that the oil lines are not damaged and show no evidence of leakage.

On rotary screw compressors:

- Verify that the oil level in the oil separator is correct.
- Verify that the oil pressure is correct.

- Verify that the oil filter or screen is not clogged and is free of debris, and that the oil lines are not damaged and show no evidence of leakage.
- Ensure that the cooling water flow to the compressor is at the specified temperature and pressure.
- Verify that the shaft seal and all gaskets show no evidence of leakage.
- Verify that the separator oil heater is operating properly.

6.4.7 Internal Inspection: Heat Exchangers

Failure Locations and Causes:
The internal inspection is focused on fouling of heat exchangers, improper water-side flow rate, leaking gaskets, and corrosion and erosion in general.

Progression of Degradation to Failure:
Erosion of gaskets and water-side fouling of condenser heat exchangers appear to be randomly occurring degradations that are dominant failure causes for chiller equipment. The time scale on these degradations is uncertain, but they are expected to require attention in just a few (2 to 5) years. The frequency of PM on heat exchangers should be based on operating experience and water system quality.

Fault Discovery and Intervention:
Internal inspection should include the following activities:

- Complete all task items from the external visual inspection.
- Inspect for loose, damaged, or missing parts, fasteners, and components.
- Inspect the condenser heat exchanger for evidence of fouling, corrosion, tube sheet degradation, and the condition of the sacrificial anode; clean and recoat the tube sheet as necessary.
- If required, eddy current test the heat exchanger looking for tube wall thinning, and tube sheet and baffle damage or wear.
- Replace the condenser heat exchanger seals and gaskets.

6.4.8 Internal Inspection: Compressors

Failure Locations and Causes:
This inspection provides an opportunity to address worn, eroded, or cracked impellers and binding prerotation vanes in centrifugal compressors, and to inspect the inlet and outlet head valves and check valves on reciprocating compressors. Other important internal tasks on reciprocating compressors include inspection of the piston rings and liners.

Progression of Degradation to Failure:
Internal inspections of compressors should be based on run-time hours. Follow the recommended actions of the manufacturer.

<u>Fault Discovery and Intervention:</u>
Inspection of compressors would be performed at the initial indication of a loss of refrigerant system capacity.

Internal Inspection should include the following activities:

- Complete all task items from the external visual inspection.
- Inspect for loose, damaged, or missing parts, fasteners, and components.
- Inspect the metering orifice for cleanliness.
- Check the integrity of the pressure rupture disk and for the presence of moisture in the vent line.
- Check the pressure control tubing for integrity and damage. Note: Replacement of plastic pressure control tubing on chillers is not to be performed at every internal inspection and may only require replacement approximately every 3 years.
- Change the oil filters and refrigerant dryers.

On centrifugal compressors:

- Megger the compressor motor if it is the hermetically sealed type.
- Inspect the float valve for proper operation, and check for evidence of wear, sticking, or a leaking float.
- Inspect the capacity control device for binding.
- Perform an electrical insulation test on the oil pump motor, if internal to the refrigerant system.
- Inspect the gaskets and seals (including the seal of motor leads on hermetic units).
- Inspect the impeller for wear and damage. Note: This does not need to be done at every inspection interval.

6.4.9 System Testing: Performance Tests

<u>Failure Locations and Causes:</u>
These tests address the integrated system performance and are therefore not directly separable from performance testing for air handling equipment, ducting, and dampers. A major focus is the performance of controls and instrumentation. Drift, component failures, and misadjustment of these components form one of the dominant failure causes for HVAC systems. Water-side fouling of condenser heat exchangers is another dominant failure cause and can be detected by performance testing. Other failure causes such as improper heat exchanger water flow rates caused by other component failures, or a plugged refrigerant metering device or failed refrigerant float valve, are also of potential significance.

Progression of Degradation to Failure:
Condenser heat exchanger fouling, drift of instrumentation and controls, and other control system failures or misadjustment are random occurrences on a time scale of 1 to 2 years. Their occurrence characteristics suggest that they can happen at any time; their relatively short time scales and the many possible control malfunctions are responsible for the fact that they are dominant failure mechanisms.

Fault Discovery and Intervention:
The recommended task interval of 2 years depends on backup from calibration, functional testing, operator rounds, and external visual inspection to provide adequate coverage for critical equipment. Operator rounds and external visual inspection are especially important for detecting system malfunctions in noncritical equipment, because these will generally not also be covered by calibration or system testing of any kind. Performance testing in many plants might not be recognized as a formal PM task. Nevertheless, this task has the function of a PM task, usually focused on failure finding.

System Testing: Performance tests should include the following activities:

- Verify that the cooling capacity is within specification.
- Measure, track, and trend the ΔT, ΔP, and flow of the water-side of condensers and evaporator heat exchangers.
- Measure and verify that the refrigerant is within normal temperature and pressure ranges.
- Measure, track, and trend the compressor motor amperage as a function of chiller load.

6.4.10 System Testing: Functional Tests

The functional test consists of a start and load sequence on large reciprocating and rotary screw compressors. It verifies that capacity control devices respond correctly, that the flow switch operate correctly, and that key parameters are in the normal operating range. It can be conducted as a postmaintenance test to verify operability and readiness to return to service, and as an in-service test on standby equipment, or on swapping over trains.

System Testing: Functional tests should include the following activities:

- Verify that the capacity controls devices respond as expected.
- Verify that the flow switch operates correctly.
- Verify that key operational parameters are in the normal operating range.

7
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A
SUPPLEMENTAL ENGINEERING FUNDAMENTALS

This appendix supplements the information presented in Section 3, "Overview of Refrigerant Systems." Proper evaluation of refrigerant systems requires a working knowledge of thermodynamics and heat transfer. The use of thermodynamic variables (measurable properties such as temperature, pressure, and volume) assist in this evaluation. The refrigeration cycle uses a fluid, called a refrigerant, to move heat from one place to another. Many other variables (such as density, specific heat, compressibility, and the coefficient of performance) can be identified and correlated to produce a more complete description of a refrigerant system and its relationship to its environment.

A.1 Refrigerant Cycles

Figure A-1 represents a thermodynamic boundary that has been placed on a refrigerant system to assist in evaluating its performance.

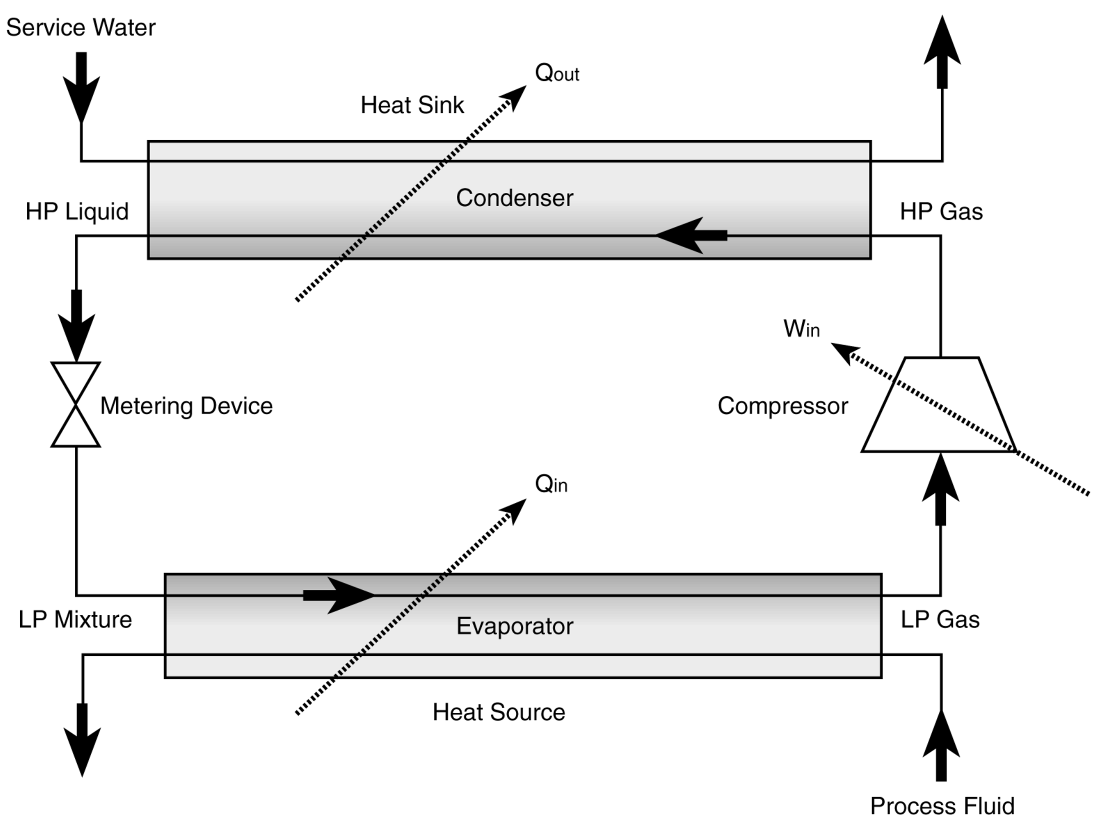


Figure A-1
Thermodynamic Boundary of a Simple Refrigerant System

The entire system in Figure A-1 represents a system within a thermodynamic boundary chosen to assist in the analysis. Based on this boundary, the following energy balance equation can be derived, based on the First Law of Thermodynamics:

$$Q_{out} = Q_{in} + W_{in}$$

where:

Q_{out} = heat removed by the system (specifically via the condenser) (Btu)

Q_{in} = heat added to the system (specifically via the evaporator) (Btu)

W_{in} = work added to the system (specifically via the compressor) (Btu)

Heat transfer describes what occurs during the operation of a refrigerant system, and it is the refrigerant system's primary purpose. The First Law of Thermodynamics states that energy cannot be created or destroyed, only changed from one form to another. Heat can be classified in three categories: latent, sensible, and specific. It is important to understand the differences between these to be able to evaluate a refrigerant system's performance.

Latent, or hidden, heat is energy added to a substance that causes a change in physical state, also referred to as a phase change (that is, when a substance changes from a solid to a liquid, a liquid to a gas, or vice versa). Latent heat does not change the substance's temperature, and is therefore not measurable by a temperature reading alone.

Sensible heat energy does not cause a change in state, but does cause a change of the substance's temperature that is measurable via temperature reading alone. The measurement of sensible heat is also commonly referred to as the dry bulb temperature. Temperature can be measured in a number of ways as is illustrated in Appendix B, "Types of Instrumentation and Data Acquisition Equipment."

Figure A-2 illustrates the difference between latent and sensible heat in water.

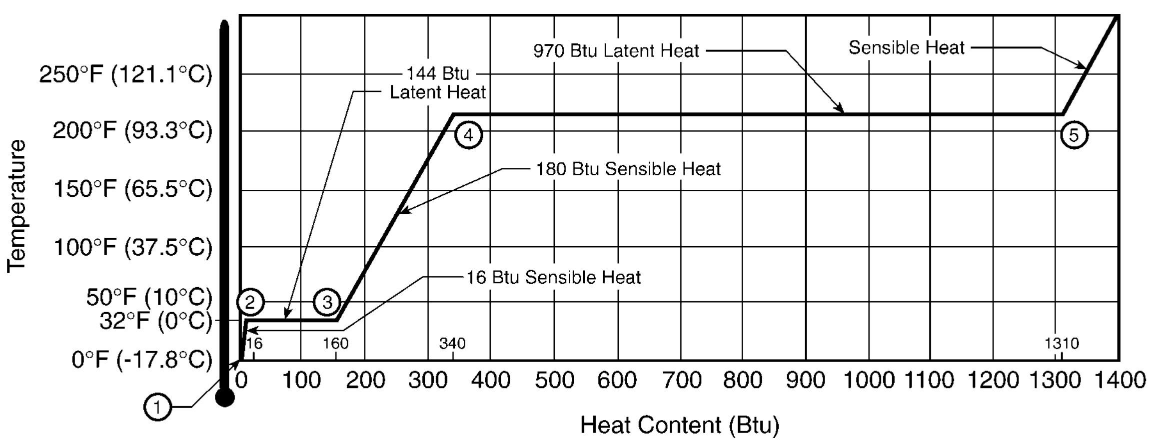


Figure A-2
Heat Content of H_2O

The substance H_2O is in the form of ice at Point 1, 0°F. Heat added from Point 1 to Point 2 is sensible heat. This is registered as a rise in temperature on a thermometer. When Point 2, 32°F, is reached, the ice is saturated with sensible heat. If any more heat is added, the ice will turn into a liquid. This change in state requires latent heat to be added. Notice that the temperature does not change, just the physical state. When Point 3 is reached, the substance is saturated water. Heat added after this point is reached is sensible heat, and this is measurable by a thermometer because the temperature rises. Point 4 is yet another saturation point. At this point, 212°F, also known as the boiling point, the temperature of the water will not rise; any heat added is latent heat, which creates steam. When this heat is added, the temperature doesn't rise until Point 5, where there is another saturation point. At this point, the steam is referred to as saturated. Any heat added after this point is surpassed is once again sensible, and the steam is referred to as superheated.

It is important to note that the amount of heat H_2O holds from Points 4 to 5. This displays that water, in this case, holds more heat as a steam than at any other state.

To apply this concept to other ideal gases and incompressible fluids, the terms subcooling, saturated, and superheated must be fully understood. These terms help to define the state, and therefore the heat content, of a substance. They will be used in combination with other parameters to allow for complete evaluation of a process.

In tables of properties for H_2O and refrigerants, it is assumed that the liquid is saturated, that is, in equilibrium with its vapor. In most cases this is true, but in some processes the liquid can become subcooled. The temperature of the liquid can be reduced below that associated with pressure in tables of properties. In this case, pressure on the liquid is higher than vapor pressure. Properties of liquids in this region are not usually well known. However, the change in enthalpy or heat content with pressure is very slight over a reasonably small range of temperature so that the enthalpy of a subcooled liquid can be assumed to be the same as that of saturated liquid at a subcooled temperature.

Saturated vapor is vapor that is in equilibrium with or associated with liquid. Tables of H_2O and refrigerant properties include data for saturated vapor. Liquid is changed into vapor by adding the latent heat of vaporization. If more heat is added to saturated vapor, it becomes superheated, that is, it has a higher heat content than when it is in a saturated condition. The addition of more heat has caused the gas to move into a region where liquid can no longer exist. The addition of heat causes temperature to rise above that for saturated vapor assuming that pressure remains the same.

Another important concept related to incompressible fluids and ideal gases is boiling and evaporation. These terms describe virtually the same process. They both refer to change from liquid to vapor. The difference is the rate at which the change of state occurs. In boiling, the rate is rapid and formation of bubbles and agitation of liquid can be observed. In evaporation, the rate is typically slower and there might be no visible liquid agitation. In addition, boiling occurs at a constant temperature as long as pressure remains constant. On the other hand, evaporation can take place at almost any temperature below the boiling point.

The specific heat of a substance is the amount of heat in British thermal units (Btu) required to change the temperature of 1 pound of a substance 1 degree Fahrenheit. Every substance has a specific heating value. Tables of these values are available in technical publications. All specific heat values are shown as a comparison to the specific heat of water; therefore, specific heat is the ratio that exists between the heat energy required to raise the temperature of a substance 1 degree and the energy required to raise the temperature of an equal volume of water 1 degree.

The change in heat within a substance can be calculated by the following equation:

$$Q = mc_p \Delta T$$

where:

Q = change in heat content (Btu)

m = mass (lbm)

c_p = specific heat of a substance ($\frac{Btu}{lbm°F}$)

ΔT = temperature difference ($°F$)

In summary, sensible heat gain is through convection, conduction, and/or radiation. Latent heat gain is through the introduction of moisture. It is necessary to calculate both sensible and latent heat loads to perform a complete energy analysis. Temperature and heat quantity are not the same thing. Temperature is a measure of sensible heat whereas heat quantity is a measure of the heat content of a substance. This concept introduces enthalpy, which is a combination of latent heat and sensible heat.

Enthalpy is not measured directly, rather, it addresses the heat added or lost by the system, which is the change in enthalpy, or ΔH, which is expressed in units of Btu. ΔH represents the difference between the enthalpy of the system at the beginning of the reaction compared to what it is at the end of the reaction:

$$\Delta H = H_{final} - H_{initial}$$

Considering the enthalpic state of the system:

If the system has **higher** enthalpy at the **end** of the reaction, then it absorbed heat from the surroundings (endothermic reaction).

If the system has a **lower** enthalpy at the **end** of the reaction, then it gave off heat during the reaction (exothermic reaction).

Therefore:

For endothermic reactions, $H_{final} > H_{initial}$ and ΔH is positive ($+\Delta H$)

For exothermic reactions, $H_{final} < H_{initial}$ and ΔH is negative ($-\Delta H$)

Enthalpy can be stated in terms of specific enthalpy, h, which is expressed in units of *Btu/lbm*. Because most applications in a refrigerant system deal with incompressible fluids and ideal gases, enthalpy can be correlated to the specific heat of a substance. This is shown by the following equation:

$$\Delta h = c_p \Delta T$$

where:

Δh = change in specific enthalpy (*Btu/lbm*)

c_p = specific heat of a substance (*Btu/lbm°F)*

ΔT = temperature difference ($°F$)

This allows for the computation of the change in heat content within a substance as a result of a process, which can be calculated by the following equation:

$$Q = m\Delta h$$

It is important to note that these equations are applicable only to constant mass systems.

Latent heat does not change the substance's temperature; therefore other measurable parameters are necessary to assist in the evaluation of the refrigerant system. One of these parameters is pressure. The simplest definition for pressure is the amount of force an object exerts on a unit of area. Pressure, for our purposes, will be expressed as pounds per square inch (psi). An example of this is a cube measuring 1 in. on each side and weighing 1 lb being placed on a flat surface. The cube is exerting 1 psi on the surface. Fluid pressure is the force exerted by a gas or liquid. Fluid pressure exists on all refrigerant piping/tubing, coolant lines, and duct work. It is this type of pressure that this section concentrates on, and it can be measured in a number of ways as is illustrated in Appendix B, "Types of Instrumentation and Data Acquisition Equipment."

There is a relationship between temperature and pressure. A liquid boils and changes to a saturated vapor for one of two reasons: 1) its temperature is raised, or 2) its pressure is reduced.

In either case, heat must be added to the liquid. Conversely, a saturated vapor changes to a liquid, for one of two reasons: 1) its temperature is lowered, or 2) its pressure is raised.

In either case, heat must be removed from the vapor. The temperature at which a saturated vapor condenses to a liquid is exactly the same temperature as that at which that liquid boils to a saturated vapor, provided that the pressure is the same in both cases. Moreover, the amount of heat required to change the liquid to a vapor (boil) is exactly the same as that which the vapor releases when it changes to a liquid (condenses), provided that the temperatures and pressures are the same in both cases.

The best way to illustrate the relationship between temperature and pressure for ideal gases is Boyle's Law, Charles' Law, and the combination of these that is known as the Perfect Gas Law.

Boyle's Law states that the volume of an ideal gas varies inversely as its pressure if the temperature of the gas remains constant. In other words, the product of the pressure times the volume remains constant, or that if the pressure of a gas doubles the new volume will be one half of the original volume. Boyle's Law uses absolute pressure for expression.

Mathematically,

$$p_1V_1 = p_2V_2$$

where:

p_1 = original pressure

V_1 = original volume

p_2 = new pressure

V_2 = new volume

Charles' Law states that the volume of a gas is in direct proportion to its absolute temperature, provided that the pressure is kept constant; and the absolute pressure of a gas is in direct proportion to its absolute temperature, provided that the volume is kept constant.

Mathematically,

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \text{ and } \frac{p_1}{p_2} = \frac{T_1}{T_2}$$

where:

V_1 = original volume

V_2 = new volume

T_1 = absolute temperature of original gas

T_2 = absolute temperature of gas after change

Charles' Law and Boyle's Law are combined to form the Perfect Gas Law. Mathematically, the Perfect Gas Law is found by simultaneously solving the above equations to result in the following equation:

$$\frac{p_1 V_1}{T_1} = \frac{p_2 V_2}{T_2}$$

It is important to note the difference between gauge and absolute pressure. The difference between the two is atmospheric pressure. To convert from gauge pressure to absolute pressure, add the atmospheric pressure to the gauge pressure. Gauge pressure is stated in units of psig, whereas absolute pressure is stated in units of psia.

CAUTION: When determining differential pressure, be sure to use the same unit of measurement between each of the pressure measurements (that is, p_1[psia] – p_2[psia]). Differential pressure is stated in units of psid.

Unfortunately, refrigerants can become contaminated, thus affecting their heat content. As a result, it is important to be able to determine the properties of the combined constituents, known as a mixture. According to Dalton's Law, the properties of each constituent are considered as though each existed separately at the volume and temperature of the mixture. In other words, in a mixture of gases, the pressure of the mixture is the sum of the partial pressures of the components of the gas. As long as the gases do not chemically react, the total pressure is the sum of the pressures created by the gases in the mixture. In equation form:

$$p_{total} = p_1 + p_2 + p_3 + ... + p_n$$

where:

n is the total number of gases in the mixture

There are other measurements necessary to evaluate the performance of a refrigerant system. The Btu can also be used to express amounts of electrical energy. It can be converted to units of horsepower. The units of measurement most often mentioned for electrical measurements are volts (V), amperes (A or I), resistance (R), British thermal unit (Btu), and horsepower (hp). All of these terms are used to describe units of energy measurement, more specifically, electrical energy. These types of measurement will be necessary to determine the amount of work in the form of power that has been added to the refrigerant system. These parameters can be measured in a number of ways as is illustrated in Appendix B, "Types of Instrumentation and Data Acquisition Equipment."

Figure A-3 is a graphic display that connects all of the information presented in this appendix as it relates to a refrigerant system.

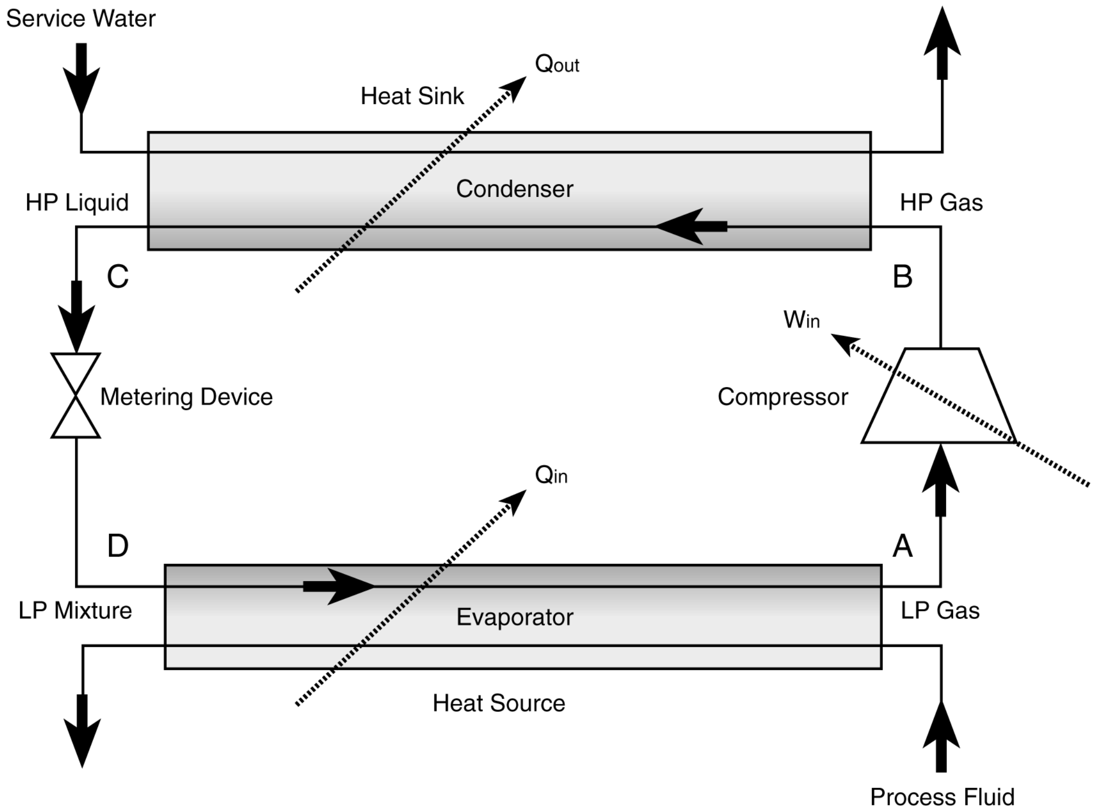


Figure A-3
Thermodynamic Boundary of a Simple Refrigerant System

A pressure-enthalpy chart specific to the refrigerant used in the system can be used to graphically represent the process. Figure A-4 shows an example that correlates to the system shown in Figure A-3.

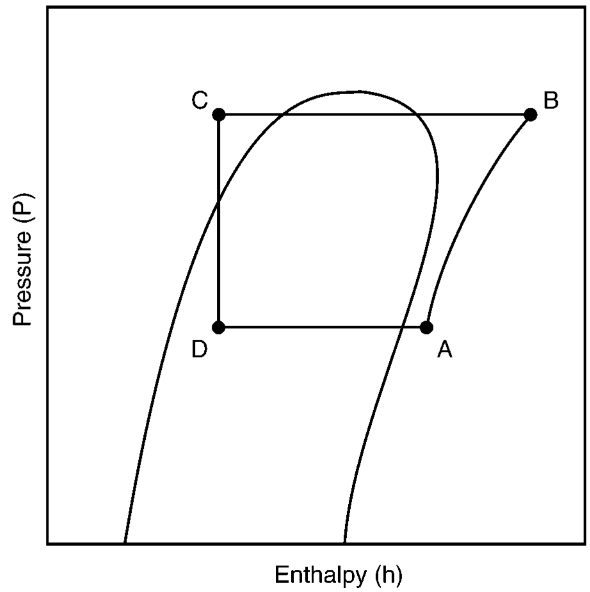


Figure A-4
Pressure-Enthalpy Diagram for a Simple Refrigerant Cycle

The process that occurs between each of these points is described as follows:

- A to B: The refrigerant changes from a low-pressure (LP), hot gas to a high-pressure (HP), hot gas by passing through the compressor.
- B to C: The refrigerant changes from an HP, hot gas to an HP, cool liquid by passing through the condenser.
- C to D: The refrigerant changes from an HP, cool liquid to an LP, cool mixture by passing through the metering device.
- D to A: The refrigerant changes from an LP, cool mixture to an LP, hot gas by passing through the evaporator.

The refrigerant flow rate depends on the total heat load on the refrigerant system and the amount of that heat load that the refrigerant can absorb, and is expressed in the following equation:

$$\dot{m} = \frac{q}{(h_A - h_D)}$$

where:

$\dot{m}$ = mass flow rate of refrigerant (*lbm/hr*)

q = heat load the refrigerant absorbs (*Btu/hr*)

h_A = heat content (enthalpy) at Point A (*Btu/lbm*)

h_D = heat content (enthalpy) at Point D (*Btu/lbm*)

The work done on the refrigerant during the compression is the product of the enthalpy increase in the refrigerant inside the compressor and the refrigerant flow rate, expressed in the following equation:

$$w = \dot{m}(h_B - h_A)$$

where:

w = rate of work done on the refrigerant by the compressor (*Btu/hr*)

h_B = heat content (enthalpy) at Point B (*Btu/lbm*)

The heat rejected to an acceptable heat sink via the condenser depends on the refrigerant flow rate and the latent heat of condensation of the refrigerant, expressed in the following equation:

$$q = \dot{m}(h_B - h_C)$$

where:

q = heat load the condenser rejects (*Btu/hr*)

h_C = heat content (enthalpy) at Point C (*Btu/lbm*)

There is no change in pressure between Points B and C on Figure A-3. This is said to be an isobaric process. In actuality there is a slight drop in refrigerant pressure as it passes through the condenser.

The work done on the refrigerant by the metering device is the product of the enthalpy change in the refrigerant across the metering device and the refrigerant flow rate, expressed in the following equation:

$$w = \dot{m}(h_C - h_D)$$

where:

w = rate of work done on the refrigerant by the metering device (*Btu/hr*)

In this example, no work is performed by the metering device because it is an isenthalpic process (that means no change in enthalpy occurs during the said process) across the metering device. The only function of the metering device in this example is to change the refrigerant from an HP, cool liquid to an LP, cool mixture.

The heat absorbed by the evaporator depends on the refrigerant flow rate and the latent heat of evaporation of the refrigerant, expressed in the following equation:

$$q = \dot{m}(h_A - h_D)$$

where:

q = heat load the evaporator absorbs (*Btu/hr*)

Similar to the condenser, there is no change in pressure between Points D and A on Figure A-3. This is also an isobaric process. Again, in actuality there will be a slight drop in refrigerant pressure as it passes through the condenser.

Consider again the first equation in this appendix:

$$Q_{out} = Q_{in} + W_{in}$$

By using electrical measurement devices the power, or work, input to the compressor from the compressor motor can be determined. Electrical power is rated in hp or watts. A hp is a unit of power equal to 746 watts or 33,000 lb-ft per minute (550 lb-ft per second). A watt is a unit of measure equal to the power produced by a current of 1 amp across the potential difference of 1 volt. It is 1/746 of 1 hp. The watt is the base unit of electrical power. Motor power is rated in hp and watts.

Hp is used to measure the energy produced by an electric motor while doing work.

To calculate the hp of a single-phase motor when current, efficiency, and voltage are known, apply this formula:

$$hp = \frac{V(I)(\eta)}{746}$$

where:

hp = horsepower
V = voltage (volts)
I = current (amps)
η = efficiency

To calculate the hp of a three-phase motor when current and efficiency, and voltage are known, apply this formula:

$$hp = \frac{(\sqrt{3})V(I)(\eta)}{746}$$

By using the conversion factor presented in Table A-1, compressor work can be converted from hp to *Btu/min*, which can then be converted to units similar to heat content, thus allowing the equation to be solved with a variety of parameters measured. These parameters should be carefully evaluated to determine which will provide for the lowest uncertainty when performing a performance evaluation.

Table A-1
Equations and Conversion Factors
Courtesy of E. I. Du Pont de Nemours & Co., Inc.

1. Net Refrigerating Effect, Btu/lb	$= \left[\text{Heat Content of Vapor Leaving Evaporator, Btu/lb}\right] - \left[\text{Heat Content of Liquid Entering Evaporator, Btu/lb}\right]$
2. Net Refrigerating Effect, Btu/lb	$= \left[\text{Latent Heat of Vaporization, Btu/lb}\right] - \left[\text{Change in Heat Content of Liquid from Condensing to Evaporating Temperature, Btu/lb}\right]$
3. Net Refrigerating Effect, Btu/lb	$= \dfrac{(\text{Capacity, Btu/min})}{(\text{Refrigerant Circulated, lb/min})}$
4. Refrigerant Circulated, lb/min	$= \dfrac{(\text{Load or Capacity, Btu/min})}{(\text{Net Refrigerating Effect, Btu/lb})}$
5. Compressor Displacement, cu ft/min	$= \left[\text{Refrigerant Circulated, lb/min}\right] \times \left[\text{Volume of Gas Entering Compressor, cu ft/lb}\right]$
6. Compressor Displacement, cu ft/min	$= \dfrac{\left[\text{Capacity Btu/min}\right] \times \left[\text{Volume of Gas Entering Compressor, cu ft/lb}\right]}{(\text{Net Refrigerating Effect, Btu/lb})}$
7. Heat of Compression, Btu/lb	$= \left[\text{Heat Content of Vapor Leaving Compressor, Btu/lb}\right] - \left[\text{Heat Content of Vapor Entering Compressor, Btu/lb}\right]$
8. Heat of Compression, Btu/lb	$= \dfrac{(42.418\ \text{Btu/min})\ (\text{Compression Horsepower})}{(\text{Refrigerant Circulated, lb/min})}$
9. Compression Work Btu/min	$= \left[\text{Heat of Compression Btu/lb}\right] \times \left[\text{Refrigerant Circulated lb/min}\right]$
10. Compression Horsepower	$= \dfrac{(\text{Compression Work, Btu/min})}{(\text{Conversion Factor, 42.418 Btu/min})}$
11. Compression Horsepower	$= \dfrac{\left[\text{Heat of Compression Btu/lb}\right] \times \left[\text{Capacity, Btu/min}\right]}{\left[42.418\ \text{Btu/min}\right] \times \left[\text{Net Refrigerating Effect, Btu/lb}\right]}$
12. Compression Horsepower	$= \dfrac{(\text{Capacity, Btu/min})}{\left[42.418\ \text{Btu/min}\right] \times \left[\text{Coefficient of Performance}\right]}$
13. Compression Horsepower per Ton	$= \dfrac{(4.715)}{\text{Coefficient of Performance}}$
14. Power, watts	$= \left[\text{Compression Horsepower per Ton}\right] \times 745.7$
15. Coefficient of Performance	$= \dfrac{(\text{Net Refrigerating Effect, Btu/lb})}{(\text{Heat of Compression, Btu/lb})}$
16. Capacity, Btu/min	$= \left[\text{Refrigerant Circulated, lb/min}\right] \times \left[\text{Net Refrigerating Effect, Btu/lb}\right]$
17. Capacity, Btu/min	$= \dfrac{\left[\text{Compressor Displacement, cu ft/min}\right] \times \left[\text{Net Refrigerating Effect, Btu/lb}\right]}{\left[\text{Volume of Gas Entering Compressor, cu ft/lb}\right]}$
18. Capacity, Btu/min	$= \dfrac{\left[\text{Compression Horsepower}\right] \times \left[42.418\ \text{Btu/min}\right] \times \left[\text{Net Refrigerating Effect, Btu/lb}\right]}{(\text{Heat of Compression, Btu/lb})}$

Efficiency for a refrigerant system is usually described by a *Coefficient of Performance* (COP), which is defined as the benefit of the cycle (the amount of heat removed) divided by the amount of required energy input to operate the cycle. Stated in equation form:

$$COP = \frac{\text{Useful refrigeration effect}}{\text{Net energy required from external sources}}$$

For the type of refrigerant cycle addressed in this guideline, the net energy supplied is usually in the form of work, mechanical or electrical, and includes any work to the compressor or fans or pumps or both. Thus:

$$COP = \frac{Q_i}{W_{net}}$$

where:

Q_i = useful refrigeration effect

W_{net} = net energy required from external sources

A.2 Ideal Refrigerant Cycles

Before describing examples of ideal refrigerant cycles, additional thermodynamic terms must be introduced. These, combined with the description of an ideal refrigerant cycle, allow for the calculation of a refrigerant system's efficiency.

Entropy measures the molecular disorder of a system. The more mixed a system, the greater its entropy; conversely, an orderly or unmixed configuration is one of low entropy. Because this is a difficult term to fully grasp, the applicability of entropy can be further understood by the information provided in the American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) Fundamentals Handbook.

The most basic description of the Second Law of Thermodynamics is how it can differentiate and quantify processes that proceed only in a certain direction (irreversible) from those that are reversible. Examples of irreversible processes include pressure drop, either in piping or heat exchangers, heat transfer between fluids of different temperatures, mechanical friction, and the conversion of electrical power to a mechanical function. By reducing the total irreversibilities within a refrigerant cycle, the efficiency of the cycle is improved.

Application of the Second Law to an entire refrigerant cycle shows that a completely reversible cycle, also known as an ideal cycle because it is conceptual only and impossible to achieve in real-world applications operating under the same conditions as the real-world system, has the maximum possible COP. A measure of the departure of the actual system from an ideal, reversible cycle is given by the system's efficiency. In equation form:

$$\eta = \frac{COP}{(COP)_{rev}}$$

The Carnot Cycle is typically used to reflect an ideal reversible refrigerant cycle.

The Carnot Cycle is an ideal heat engine in which the sequence of operations forming the working cycle consists of isothermal expansion (expansion of the refrigerant at a constant temperature), adiabatic expansion (expansion of the refrigerant at a constant heat capacity), isothermal compression (compression of the refrigerant at a constant temperature), and adiabatic compression back to its initial state (compression of the refrigerant at a constant heat capacity). Stated another way, the Carnot Cycle operates between two fixed temperatures or between two fluids at different temperatures, each with infinite heat capacity. No refrigeration cycle may have a COP higher than that for a reversible cycle operated between the same temperature limits; and all reversible cycles, when operated between the same temperature limits, have the same coefficient of performance.

Figure A-5 can be used to represent the Carnot Cycle for Figure A-3. W_{net} is the net energy required from external sources, and Q_i (represented by the patterned area) is the useful refrigeration effect. The refrigerant, a saturated liquid at State C expands isentropically (constant entropy) to the low temperature and pressure of the cycle at State D. Heat is added isothermally and isobarically by evaporating the liquid phase refrigerant from State D to State A. The cold saturated vapor at State A is compressed isentropically to the high temperature in the cycle at State B. However, the pressure at State B is below the saturation pressure corresponding to the high temperature in the cycle. The compression process is completed by an isothermal compression process from State B to State B_F. The cycle is completed by an isothermal and isobaric heat rejection or condensing process from State B_F to State C. Therefore, the COP_{rev} can be expressed as:

$$COP_{rev} = \frac{Q_i}{W_{net}}$$

In a reversible process, the state of the working fluid and system's surroundings can be restored to the original ones. This requires that the working fluid goes through a continuous series of equilibrium states. A reversible process must satisfy the following criteria:

- No internal or mechanical friction is allowed.
- The temperature and pressure difference between the working fluid and its surroundings should be infinitely small.

There are no truly reversible processes in practice. The real processes are called irreversible. However, there are some processes, such as processes in cylinders with reciprocating piston, that can be assumed to be internally reversible with good approximation. The working fluid is always in an equilibrium state in internally reversible process. However, the surroundings undergo a state change that can never be restored. Some processes, such as processes in turbo machinery, may not be assumed to be internally reversible. The irreversibility of these processes is due to the high degree of turbulence of the working fluid. A reversible process between two states can be shown by a continuous curve on any diagram of properties. Points on the curve represent the intermediate states.

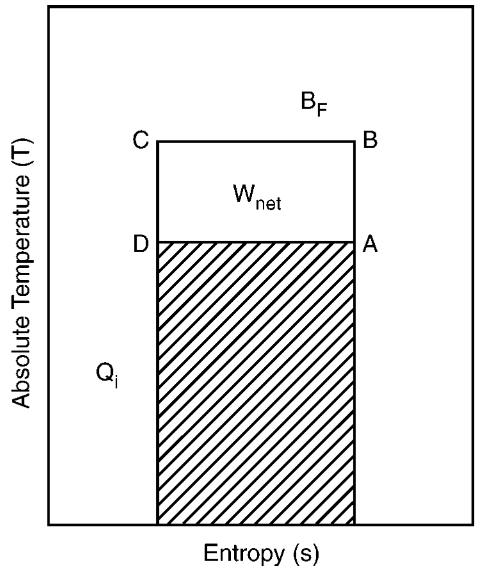


Figure A-5
Carnot Refrigerant Cycle

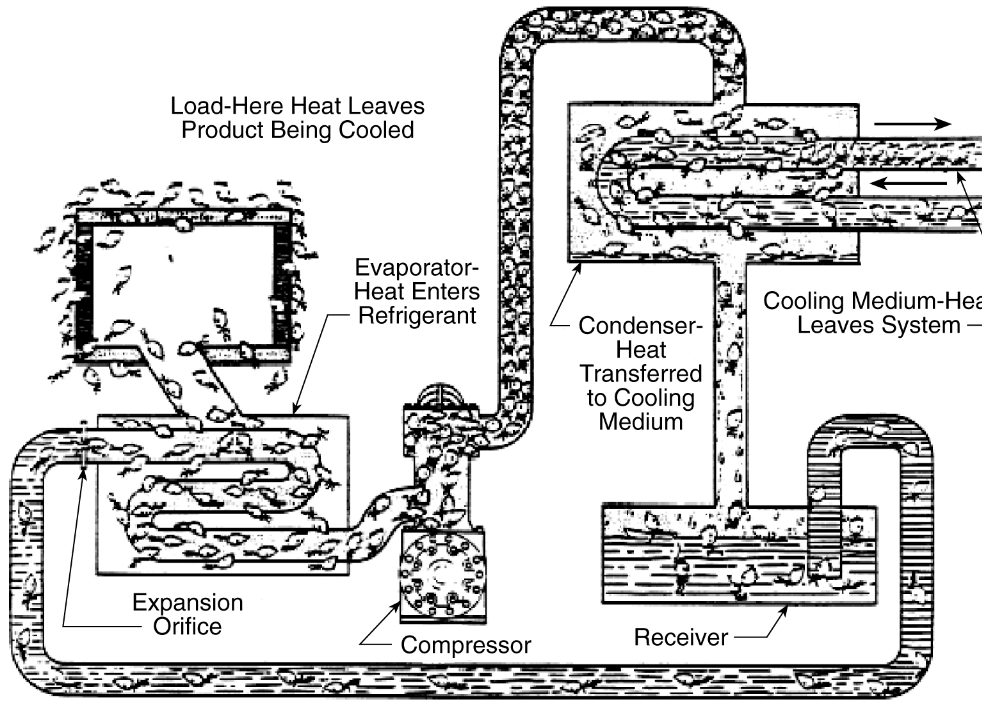
A.3 Centrifugal Liquid Chilling Systems and Components

The following is an excerpt from York Form 55.50-NM3, *Centrifugal Liquid Chilling Units*.

A.3.1 Refrigerant

A refrigerant is a compound whose physical and thermodynamic properties are such that it is economically possible to use it as a medium through which heat can flow or be made to run "down-hill" from a warmer to a cooler substance. It functions as the working medium by means of which the necessary transformations of thermal energy are accomplished during the refrigerating cycle.

Heat from the load (usually air to be cooled) is transferred to the expanded refrigerant within the EVAPORATOR or COOLER. After being raised in level or intensity by the COMPRESSOR, the original heat plus the heat of compression, see COOLING CYCLE, is transferred within the CONDENSER to the cooling medium (usually water). The refrigerant, at the same time, is liquefied or condensed and is ready to repeat the cycle. During each of these transfers, the heat has flowed from a higher to a lower temperature level. The heat originally absorbed from the load is thus carried away from the system by the condenser cooling medium (See Figure A-6).



Courtesy of York

Figure A-6
Refrigerant Function

Refrigerants are divided into two classes, primary and secondary (chilled liquid) as follows:

a) Primary Refrigerants - are fluids, usually liquids, which absorb heat directly from the substance to be cooled. Primary refrigerants commonly used in centrifugal compressor systems are R-11, R-12, R-500, and R-22.

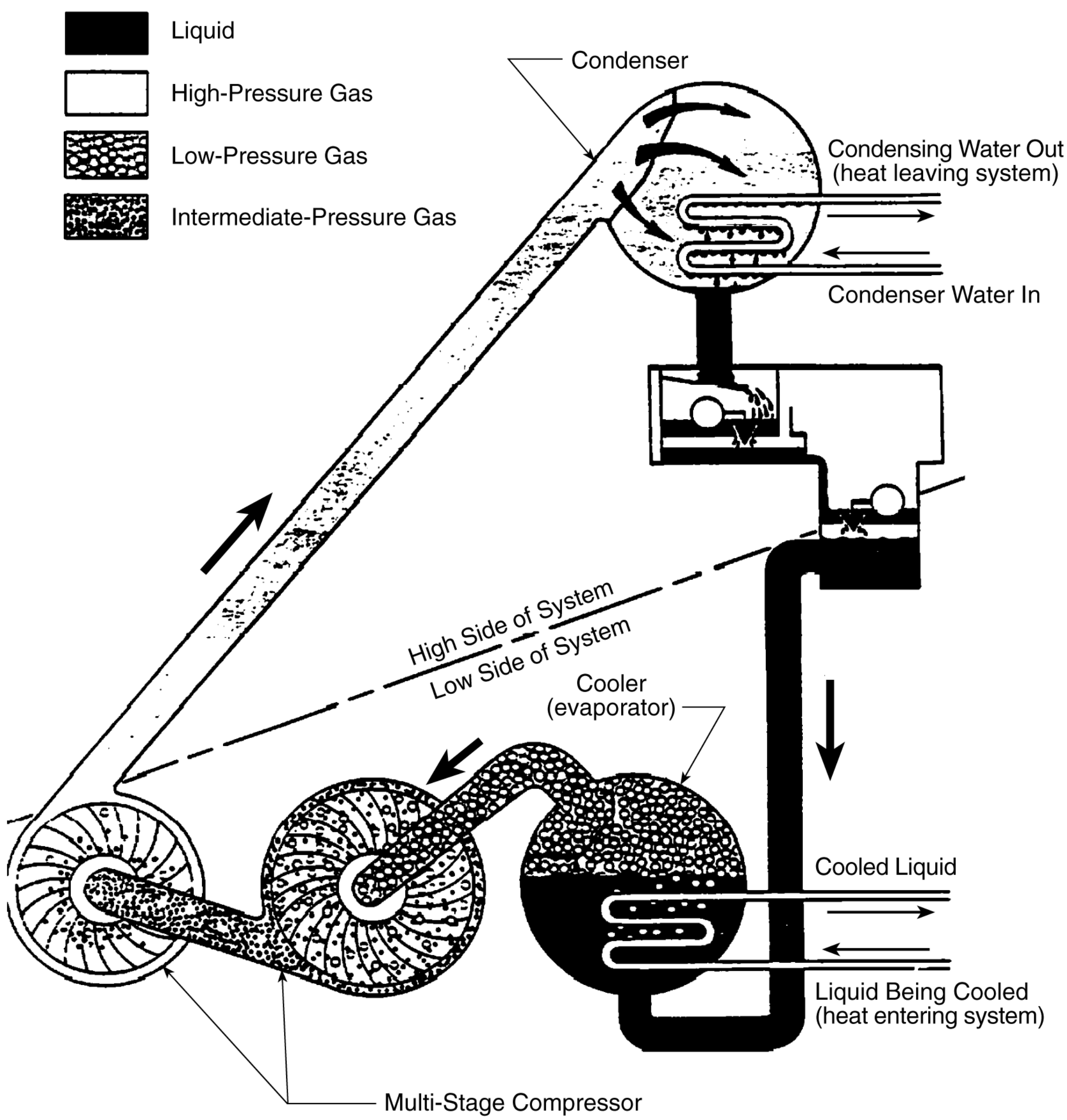
b) Chilled Liquid (secondary refrigerants) - are fluids (or solids) that absorb and transfer heat, remaining usually, in the liquid form.

Examples of chilled liquid (secondary refrigerants) are water or brine that is cooled in the tubes of the evaporator. The cooled liquid is circulated to coils that cool air to be delivered to the spaces being conditioned or to heat exchangers where it cools the product to be refrigerated.

Ice can be classed as a combined primary and secondary refrigerant. Heat is removed from the water to the point where the latent heat of fusion is removed (the water changes to ice). Then the ice is ready to absorb heat from a warmer body, or to cool that body. The ice changes to water at 32°F, the drainage of which carries the absorbed heat to the atmosphere outside. If the process of refrigeration with ice is to be continued, the melted water must be frozen and the ice replaced. This condition with natural ice was not convenient, and the discovery of the properties of volatile liquids made possible a continuous process of refrigeration by mechanical means, hence mechanical refrigeration.

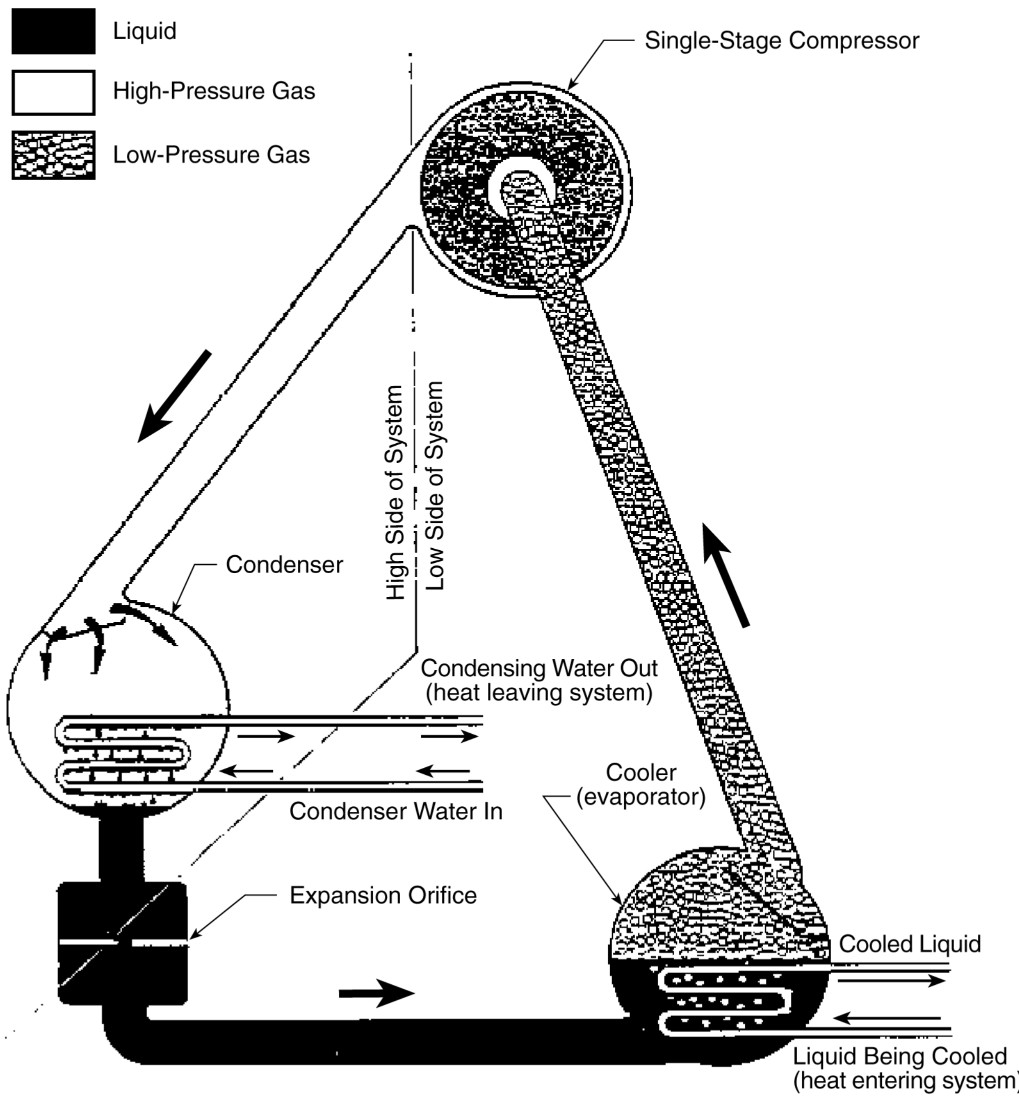
A.3.2 Cooling Cycle

The process of compressing gas within the centrifugal compressor is basically the same as within an ordinary reciprocating compressor except that compound compression occurs in all turbos. Similarly, the cycle through which the refrigerant passes in a centrifugal compressor is basically the same as in any ordinary cooling system. Starting at the outlet of the condenser, the high pressure liquid refrigerant travels through the liquid line to the expansion orifice. (The expansion orifice is located in the intercooler and consists of two float valves - units using multi-stage centrifugal compressors (see Figure A-7) or an expansion orifice with a hole sized to suit the refrigerant tonnage requirements, located in a flow control chamber - units using a single stage centrifugal compressor (see Figure A-8). As the refrigerant passes through the orifice, its pressure is reduced, and part of the refrigerant (known as "flash gas") immediately becomes a vapor by absorbing its latent heat of vaporization from, and therefore lowering the temperature of the low pressure liquid refrigerant in the evaporator or cooler.



Courtesy of York

Figure A-7
Basic Cooling Cycle – Centrifugal Unit Using Multi-Stage Compressor



Courtesy of York

Figure A-8
Basic Cooling Cycle – Centrifugal Unit Using Single-Stage Compressor

The higher temperature brine or water contained inside of the evaporator tubes transfers its heat to the refrigerant contained within the cooler shell outside of the tubes, causing the low pressure liquid refrigerant to boil or evaporate. In doing this, the refrigerant absorbs a quantity of heat equal to its latent heat of vaporization from the brine or water inside the tubes, thereby, lowering the temperature of the brine or water.

The compressor draws off the heat laden refrigerant vapor (suction gas) from the evaporator, which creates space for more refrigerant to enter. The temperature and pressure of the suction gas is increased as it passes through the centrifugal compressor, leaving it in a superheated condition.

The high pressure discharge gas at high temperature is forced into the condenser where an amount of heat equal to the latent heat of condensation plus the heat of compression is transferred from it to the condensing medium (usually water) which is contained on the inside of the condenser tubes with the refrigerant on the outside. This heat removal with no reduction in pressure causes the high pressure refrigerant gas to condense into high pressure liquid, ready to enter the expansion orifice to repeat the cycle.

Refrigerating effect, or removal of heat, is thus accomplished and the process will be continuous as long as the compressor continues to run, the condensing medium is circulated through the condenser tubes, and the liquid to be cooled is circulated through the cooler or evaporator tubes.

From the discharge spaces of the high state impeller wheel of the centrifugal compressor to the expansion orifice, the refrigerant is under discharge pressure. From the expansion orifice to and including the suction inlet of the first stage impeller, the refrigerant is under suction pressure. The portion of the unit under discharge pressure is known as the HIGH SIDE of the unit ad the portion under suction pressure is known as the LOW SIDE. Note that the centrifugal compressor, one or more intermediate pressures exist; namely, the pressure or pressures at the point where a lower stage impeller discharges into the suction of the next higher stage impeller.

A.3.3 Refrigerant Characteristics

A.3.3.1 Pressure - Temperature

The heat absorbing body of refrigerant must be at a temperature lower than that of the substance to be cooled. Heat extracted from the body being cooled by the refrigerant must be carried away and delivered to the atmosphere or earth where it originated. If the lower temperature of the cooled body is to be maintained, the body must be surrounded by insulating material so that the heat cannot return to it.

Every liquid has a different boiling temperature for each different pressure under which it is confined: the boiling point is also the condensation point for that pressure. This temperature-pressure relationship is determined experimentally for each liquid. Water boils at 212°F at atmospheric pressure (zero pounds gauge pressure), at 100°F at 28" vacuum, and at 338°F at 100 psi gauge pressure.

Liquids with low boiling points are used as refrigerants in mechanical refrigeration. When the temperature of a liquid is raised to the boiling point corresponding to its pressure, both liquid and gas can exist together and the condition is said to be saturated. Gas containing particles of liquid is said to be “wet,” but if it is at the boiling temperature with no liquid particles present it is “dry” and saturated.” If the temperature of the gas at existing pressure is raised above its saturation temperature, it is “superheated.” Strictly speaking, saturated gas, either wet or dry, is a vapor and to a gas until the vapor is highly superheated.

Refrigerant-11 (R-11) is known as “low pressure refrigerant.” At atmospheric pressure, the refrigerant boils at a relatively high temperature. At normal evaporating temperature within an air conditioning system, the operating pressure is low. Refrigerants R-12, R-500, and R-22 are all known as high pressure refrigerants. This is due to the fact that at normal operating temperature their respective pressures are high.

A comparison of normal suction and discharge pressures at 40°F evaporator and 100°F condensing temperatures for R-11, R-12, R-22, and R-500 is shown in Table A-2.

Table A-2
Refrigerant Characteristics

REFRIGERANT	USUAL APPLICATION	TEMPERATURE NORMAL SUCTION & DISCHARGE					TEMPERATURE AT ATMOSPHERIC PRESSURE
		RANGE	TEMP.		PRESSURE		
R-11	Air Conditioning	Medium- High	Evap. Cond.	40°F 100°F	15.61” 8.9	Vac. Psig	74.87°F
R-12	Air Conditioning Commercial and Industrial	Low	Evap. Cond.	40°F 100°F	37.0 117	Psig Psig	-21.62°F
R-22	Air Conditioning Commercial and Industrial	Low	Evap. Cond.	40°F 100°F	68.5 196.0	Psig Psig	-41.36°F
R-500	Air Conditioning Commercial and Industrial	Low	Evap. Cond.	40°F 100°F	46.1 141.1	Psig Psig	-28.3°F

In general, the lower pressure refrigerants are used for the higher evaporator temperature applications and the higher pressure refrigerants for the lower evaporator temperature applications. Under normal conditions, due to the specific volume conditions of these refrigerants, a more favorable brake HP per ton of cooling is possible with the low pressure refrigerants. Table A-1 shows a comparison of the boiling or evaporating temperatures of these refrigerants at atmospheric pressure.

Pressure sensitive labels with this information are also shipped in the installation packets for the unit involved. These labels should be put on the unit for reference.

The selection of refrigerants to be used for a centrifugal installation depends largely upon the type of work to be done. The compressor is generally of the minimum size which will operate efficiently from an economic standpoint. There is a limit in centrifugal compressor size, due to required clearance and passages, particularly the suction inlet, or eye of the first stage impeller wheel, below which it is not practical to go. The varying characteristics of R-11, R-12, R-500, and R-22, particularly the densities and latent heats, enter largely therefore, into the determination of the refrigerant and the size turbo compressor to be used for a particular job.

A.3.3.2 Safety

The refrigerants used in centrifugal units, considered with respect to toxicity, inflammability and explosiveness, which are the most important of the safety qualifications, are classed as safe refrigerants.

At normal temperatures, these refrigerants do not have toxic properties, i.e. will not cause injury to human beings. However, it must be remembered, that they are heavier than air, and that in heavy concentrations, a deficiency of oxygen will result, which can cause suffocation of humans or animals.

At higher than normal temperatures, (when exposed to flame or electric heating elements) these refrigerants break down and become highly toxic and even in low concentrations may cause fatal or serious injury in a relatively short time. If flame exists in a room where R-11, R-12, R-22, or R-500 are present, or where their presence is even suspected, it is important that the room or space be adequately ventilated (fresh air introduced immediately).

Since these refrigerants will neither burn nor support combustion, they are safe in regard to inflammability and explosiveness.

A.3.3.3 Leak Detection

Leak detection is readily and accurately accomplished with these refrigerants even in the case of very small leaks. After the usual vacuum and pressure tests, a halide leak detector torch or special electronic leak detector is used. Either of these instruments when used correctly in accordance with the manufacturer’s instructions is capable of extreme precision in the detection of leaks.

A.3.3.4 Miscibility

R-11, R-12, R-500, and R-22 are completely miscible with oil. That is to say, oil and these refrigerants mix completely and in all proportions. This characteristic of these refrigerants results in a condition which is of considerable importance to the operation of centrifugal compressor units.

Oil, which should always remain within the compressor housing, can travel to other parts of the unit when mixed with the refrigerant. This condition is highly undesirable and should be carefully guarded against when operating a centrifugal compressor unit. All YORK centrifugal compressors are equipped with special oil control features to aid in preventing the travel of liquid oil from the compressor to the unit. These features are (1) the Labyrinth Oil Seals, (2) the Oil Heater, (3) the Bearing Oil Seals, and (4) Pressure equalizing tubing and passages built into the compressor and advantageous location of oil sumps. These features are described in detail under their proper headings in other Operating and Maintenance Instructions. Distinction should be made when operating a unit between the oil circulated through the unit in mist (vapor) form, which is normal, and the travel of liquid oil as described above.

A.3.3.5 Washing Effect

Because of the miscibility of these refrigerants, they are excellent oil and grease solvents. Therefore, it is highly important that all materials that make up a centrifugal unit be thoroughly clean before installation since any foreign material such as scale, dirt, silt, rust, chips, or threat compound on the inside of pipe fittings will eventually be loosened from the interior surfaces of the refrigerant-containing portions of the unit. When once loosened, this foreign material will finds its way into the compressor lubricating oil where it can cause damage. The action of these refrigerants on gasket material may lead to difficulty in holding the refrigerant within the system if proper precautions are not observed as outlined under the Installation Instructions.

A.3.3.6 Leak Tendency

The smaller the molecule of a refrigerant, the harder it is to prevent it from leaking from a unit. The refrigerants all possess a rather high leak tendency. However, leaks are very remote in a unit which has been properly installed and tested and is properly maintained.

A.3.3.7 Reaction with Moisture

These refrigerants do not absorb water as readily as ammonia (R-707) for instance, but they do unite with it even in small quantities, which causes the formation of acids, particularly hydrochloric acid in a weak form. The latter will readily attack the metals commonly used in fabricating refrigerating equipment as well as some of the materials of which gaskets, and other nonmetallic materials may be composed. It is, therefore, important to see that moisture is carefully eliminated and guarded against.

B
TYPES OF INSTRUMENTATION AND DATA ACQUISITION EQUIPMENT

B.1 Rotating Vane Anemometer

The basic propeller or rotating vane anemometer consists of a lightweight, wind-driven wheel connected through a gear train to a set of recording dials that read the linear feet (meters) of air passing through the wheel in a measured length of time. At low velocities, the friction drag of the mechanism is considerable. To compensate for this, a gear train that overspeeds is commonly used. For this reason, the correction is often additive at the lower range and subtractive at the upper range, with the least correction in the middle of the 200 to 2000 fpm (feet/minute) (1 to 10 meters/second [m/s]) ranges. Most older instruments are not sensitive enough for use below 200 fpm (1 m/s). Newer instruments can read velocities as low as 30 fpm (0.15 m/s).

Other instruments read in feet (meters), and so a timing instrument must be used to determine velocity. Readings are usually timed for 1 minute, in which case the anemometer reading (when corrected according to a calibration curve) will give the result in fpm or meters per minute. For moderate velocities, it may be satisfactory to use a 1/2 minute timed interval, repeated as a check.

Key Technical Point

An uneven airflow is frequently found when measuring airflow across the face of cooling coils because of entrance or exit conditions and/or stratification.

Figure B-1 shows a typical mechanical rotating vane anemometer (see Figure 4-8 from EPRI 1003092, *HVAC Testing, Adjusting, and Balancing Guideline*).

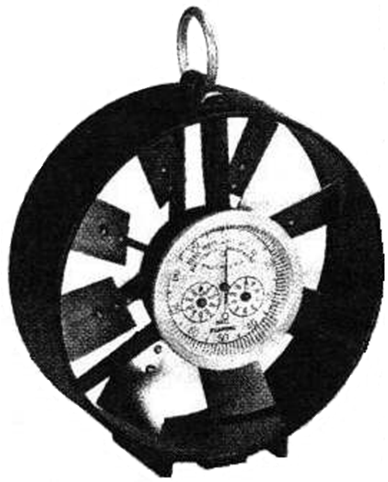


Figure B-1
Typical Mechanical Rotating Vane Anemometer

B.2 Electronic Rotating Vane Anemometer

The electronic rotating vane anemometer is a battery-operated, direct digital or analog readout anemometer. Some have interchangeable remote rotating vane heads. The digital readout of the velocity is automatically averaged for a fixed time period, depending on the measured velocity and the type of instrument. Analog instruments are direct readout with a choice of velocity scales.

Figure B-2 displays a typical electronic rotating vane anemometer (see EPRI 1003092, *HVAC Testing, Adjusting, and Balancing Guideline*).

Figure B-2
Typical Electronic Rotating Vane Anemometer

B.3 Deflecting Vane Anemometer

The deflecting vane anemometer operates by having pressure exerted on a vane that causes a pointer to indicate that measured value. It is not dependent on air density because of the sensing of pressure differential to indicate velocities. The instrument is provided and always used with a dual-hose connection between the meter and the probes, except as noted in the next paragraph.

One type of deflecting vane anemometer uses three interchangeable velocity probes: the low-flow probe, the diffuser probe, and the Pitot probe. The low-flow probe is used in conjunction with the 0–300 fpm scale for measuring terminal air velocities in rooms or open spaces as well as face velocities at ventilating hoods, spray booths, and fume hoods. The low-flow probe is directly mounted to the meter without the use of hoses. The Pitot probe is used for measuring airstream velocities in ducts. The diffuser probe is used for measuring air velocity through supply and return air terminals, using the proper air terminal "K or A_k" factor (effective area) for the airflow calculation. Results will be provided in terms of cubic feet per minute (CFM).

B.4 Pressure Test Gauge

The calibrated pressure test gauge should be of a minimum "Grade A" quality, have a Bourdon tube assembly made of stainless steel, alloy steel, Monel or bronze, and a nonreflecting white face with black letter graduations conforming to ANSI/ASME Specification B40.1. Test gauges are usually 6-1/2 to 6 in. in diameter with bottom or back connections. Dials are available with pressure, vacuum, or compound ranges.

Dial gauges are used primarily for checking pump pressure; coil, chiller, and condenser pressure drops; and pressure drops across orifice plates, valves, and other flow-calibrated devices.

Pressure ranges should be selected so that the pressures to be measured fall in the middle two-thirds of the scale range. The gauge should not be exposed to pressures greater than the maximum dial reading. Similarly, a compound gauge should be used where it is exposed to vacuum. Pressure pulsations can be reduced or eliminated by installing a needle valve between the gauge and the system equipment or piping. Under extreme pulsating conditions, a pulsation dampener or snubber can be installed.

B.5 Differential Pressure Gauge

A differential pressure gauge is a dual-inlet, “Grade A” dual Bourdon tube pressure gauge with a single indicating pointer on the dial face that indicates the pressure differential existing between the two measured pressures. The gauge can be calibrated in psi, inches w.g., or inches mercury. The differential pressure gauge will automatically read the difference between two pressures.

Using a single test gauge, the gauge is alternately valved to the high-pressure side and the low-pressure side to determine the pressure differential. Such an arrangement eliminates any problem concerning gauge elevations and virtually eliminates errors due to gauge calibration.

Section 2 of EPRI TR-109634, *Flow Meter Guideline*, provides a detailed discription of flow measurement principles. The reader is encouraged to refer to this guideline for additional information on any of the following topics:

- Continuity of flow equation
- Bernoulli’s Equation of Energy
- Reynolds Number

Section 2.12 of EPRI TR-109634, *Flow Meter Guideline*, provides a table comparing the relative costs of employing various types of flow meters for measuring water flow.

The following excerpts from EPRI TR-109634 are provided as an overview of the various methods available to measure the flow of incompressible fluids. EPRI TR-109634 should be consulted for additional information and further discussion of the mathematical equations used to calculate flow using each type of metering device.

B.6 Differential Pressure Producers

The principle of measurement is based on the introduction of a differential pressure producer (for example, orifice, nozzle, and Venturi) into a pipe through which a fluid is running full. The introduction of the differential pressure producer creates a dynamic pressure difference between the upstream and downstream sides of the device. The square root of the differential pressure is proportional to the velocity of the fluid. Differential-pressure-producing flow meters determine an area-averaged throat velocity from the measured pressure differential in the following manner:

$$q = C_d * K * A * \sqrt{\frac{\Delta p}{\rho}}$$

where, when using compatible units yields:

q	=	volumetric flow
C_d	=	discharge coefficient
C	=	conversion coefficient
d	=	diameter of threat
A	=	normal, cross-sectional area at measurement plane
Δp	=	pressure differential
ρ	=	density of the flowing fluid

The constant “K” includes values for the ratio of the cross-sectional area of the pipe to the restriction cross-sectional area and dimensional conversion constants. The accuracy of all differential pressure meters depends on a stable, fully developed velocity profile and an accurate measurement of both the pipe and the restriction diameters. Distortion of the velocity profile, due to upstream disturbances or any change in the pipe or restriction dimensions due to scaling or erosion, will negatively impact the accuracy of the flow measurement.

B.7 Ultrasonic Flow Meters

Ultrasonic flow meters operate by transmitting an ultrasonic signal into a flow stream to determine the velocity of the fluid. The velocity is then converted to a volumetric flow measurement, using the flow area dimensions and flow profile coefficient. The following types of ultrasonic flow meters are used for measuring flow in closed conduits:

- Transit-time
- Doppler
- Cross-correlation

Although all three types provide a viable means of measuring fluid flow, the transit-time and cross-correlation methods are the most appropriate for measuring water flow.

B.8 Magnetic Flow Meters

The magnetic flow meter is based on Faraday's Law of (Electro) Magnetic Induction. When a conductive fluid passes through an applied magnetic field, a voltage is generated at right angles to the axis of fluid flow and the applied magnetic field. The generated output voltage is a summation of individual voltages generated by differential volumes moving at discrete velocities across the plane of the pipe. In 1961, Shercliff demonstrated that the voltage output signal represents the average velocity for an asymmetric velocity profile. If the magnetic field is constant and the distance between the electrodes is fixed, the induced voltage is directly proportional to the average velocity of the fluid.

B.9 Turbine Flow Meters

The principle of measurement for a turbine flow meter is based on a rotating element, which is positioned in the flow stream so that the rotational speed of the rotor is proportional to the fluid stream velocity and, therefore, the flow through the measurement plane. A turbine flow meter (the primary element) typically outputs a low-amplitude frequency signal that is input into a signal conditioner (the secondary element), which converts the meter output to an analog signal proportional to the flow. Each meter has a characteristic K-factor that relates output frequency to a volumetric unit (for example, pulses/gallon). These types of flow devices are generically categorized as linear flow meters.

Key Technical Point

Manufacturer recommendations for calibration, inspection, maintenance, and/or installation criteria must be followed to ensure accurate readings.

B.10 Dial Thermometers

Dial thermometers have either a rigid stem or a flexible capillary. They are constructed with various size dial heads, 1-3/4 to 5 in., with a stainless steel encapsulated temperature-sensing element. Hermetically sealed, they are rust-, dust-, and leak-proof and are actuated by sensitive bimetallic helix coils. Some can be field calibrated. Sensing elements range in length from 2-1/2 to 24 in. and are available in many temperature ranges, with and without thermometer wells.

Dial thermometers are more rugged and more easily read than are glass tube thermometers, and they are fairly inexpensive. Small dial thermometers usually use a bimetallic temperature-sensing element in the stem.

The flexible capillary-type dial thermometer has a large temperature-sensing bulb that is connected to the instrument with a capillary tube. The instrument contains a Bourdon tube (the same as in pressure gauges). The temperature-sensing system, consisting of the bulb, capillary tube, and Bourdon tube, is charged with either a liquid or a gas. Temperature changes at the bulb cause the contained liquid or gas to expand or contract, causing a pointer to move over a graduated scale.

In using a dial thermometer, the stem or bulb must be immersed a sufficient distance to allow this part of the thermometer to reach the temperature being measured. Because dial thermometers have a relatively long time lag, enough time must be allowed for the thermometer to reach a steady temperature measurement.

B.11 Electronic Thermometers

There are many types of rugged, lightweight, battery-powered digital electronic thermometers that have precision accuracy with interchangeable probes or sensors or both. Types of electronic thermometers include resistance temperature detectors (RTDs), thermistors, thermocouples, and diode sensors, all of which have either liquid crystal or light-emitting diode (LED) displays. Response time and ease of use vary from model to model and from type to type.

Electronic thermometers have advantages of remote reading, good precision, and flexibility with temperature range. Additionally, some electronic thermometers have multiple connection points on the instrument case as well as a selector switch, which enables the use of a number of temperature sensors that can be placed in different locations and read one at a time.

In piping and duct applications, the surface temperature of the conduit is not equal to the gas or fluid temperature. A relative comparison is more reliable than an absolute reliance on readings at a single circuit or terminal unit.

Electronic thermometers can be used for checking air or liquid temperatures either immersed in the fluid stream or from surfaces. Resistance type thermometers have longer response times than the thermocouple type.

Key Human Performance Point

Extreme caution must be used when working around hot piping.

B.12 Portable Non-Contact Thermometers

These devices are rugged and simple to use. Most are equipped with a laser pointer to facilitate the determination of the location of the temperature-measuring point. These devices work on the principle of infrared energy that all objects above absolute zero radiate. These devices rapidly respond to temperature changes and, at close ranges, are good for determining hot spots. The average effective range depends on the size of the object being measured and the clarity of the air between the object and the detector.

B.13 Color Strip Temperature Indicators

These simple devices employ a temperature sensitive, chemically treated spot on a strip that changes color at certain specified temperatures. There are no moving parts, and employment is usually specified by the manufacturer.

B.14 Volt-Ammeter

The clamp-on type of volt-ammeter, with digital or analog readout, is used for taking field electrical measurements. The clamp-on type of volt-ammeter often has trigger-operated, clamp-on transformer jaws that permit current readings without interrupting electrical service. Most meters have several scale ranges in amperes and volts. Two voltage test leads are furnished that can be quick-connected into the volt-ammeter.

When using the volt-ammeter, the proper range should first be selected. When in doubt, the user should begin with the highest range for both voltage and amperage scales. Readings can be taken at the motor leads or from the load terminals of the starter. To determine the amperages of single-phase motors, the clamp should be placed around one wire after the motor has been started. When involved with three-phase current, readings should be taken on each of the three wires and then averaged. If the average voltage delivered to the motor varies by more than a few volts from the nameplate rating of the motor, several things can occur. A rise in voltage can damage the motor and cause a drop in the amperage reading. A drop in the voltage will cause a rise in the amperage and can cause the overload protectors on the starter to “kick out.” In either case, it is advisable to document and report high- or low-voltage situations.

To measure voltage with portable test instruments, the meter should be set to the most suitable range and the test lead probes connected firmly against the terminals or other surfaces of the line under test.

Key Human Performance Point

Extreme caution must be used when working around energized electrical equipment.

B.15 Airflow Measuring Instruments

Table B-1 provides a listing of commonly used airflow measuring instruments along with their recommended uses and limitations.

Table B-1
Airflow Measuring Instruments

Instrument	Recommended Uses	Limitations
Anemometer rotating vane	Measurement of velocities at air terminals, air inlets, and filters or coil banks	Total inlet area of rotating vane must be in measured airflow. Correction factors may apply.
Anemometer deflecting vane	Measurement of velocities at air terminals and air inlets	Instruments should not be used in extreme temperature or contaminated conditions.

B.16 Hydronic Measuring Instruments

Table B-2 provides a listing of commonly used hydronic measuring instruments along with their recommended uses and limitations.

Table B-2
Hydronic Measuring Instruments

Instrument	Recommended Uses	Limitations
Pressure test gauge (calibrated)	Static pressure measurements of system equipment or piping or both	Pressure gauges should be selected so that the pressures to be measured fall in the middle two-thirds of the scale range. The gauge should not be exposed to pressures greater than or less than the dial range. Pressures should be applied slowly to prevent severe strain and possible loss of gauge accuracy.
Pressure gauge (differential)	Differential pressure measurements of system equipment or piping or both	Same limitations as the pressure test gauge.
Flow meter devices	Highly accurate measurement of volume flow rates in fluid systems	Must be used in accordance with recommendations of the equipment manufacturer.

B.17 Temperature Measuring Instruments

Table B-3 provides a listing of commonly used temperature measuring instruments along with their recommended uses and limitations.

Table B-3
Temperature Measuring Instruments

Instrument	Recommended Uses	Limitations
Dial thermometers	Measurement of temperatures of air and fluids.	Ambient conditions can impact measurement of fluid temperature. Stem or bulb must be immersed a sufficient distance in fluid to record accurate measurement. Time lag of measurement is relatively long.
Electronic thermometers	Measurement of temperatures of air and fluids. Measurement of surface temperatures of pipes and ducts.	Use instrument within the recommended range. Use thermal probes in accordance with recommendations of the manufacturer. Surface temperatures of piping and duct may not equal fluid temperature within due to thermal conductivity of material.
Portable non-contact thermometers	Identification of hot spots on equipment, measurement of general area temperature, and checking of temperature of un-insulated ducts	Does not work effectively in dusty or smoky atmospheres; the size of the measuring spot is a function of distance from object.

B.18 Recommended Accuracy of Instrumentation

Table B-4 provides recommended ranges, accuracy, and calibration schedules for different types of instrumentation. Table B-4 is presented for illustrative purposes only. Licensees typically have plant- or site-specific measuring and test equipment calibration and accuracy requirements, which should always be used in lieu of the values listed in Table B-4.

Table B-4
Instruments' Recommended Calibration Accuracy

Function	Range	Minimum Accuracy	Calibration Schedule
Electrical measuring instrument	0 to 6000 Vac 0 to 100 amperes 0 to 30 Vdc	3% of full scale 3% of full scale 3% of full scale	6 months 6 months 6 months
Air velocity measuring instrument	Minimum range 100 to 3000 fpm	±10% when used in accordance with manufacturers' recommendations	12 months
Temperature measuring instrument (contact)	Minimum range 0 to 240°F	±1% of full scale	12 months
Hydronic pressure measuring instrument	0 to 30 psi 0 to 60 psi 0 to 200 psi 30 in. Hg to 30 psi 30 in. Hg to 60 psi	±1% of full scale ±1% of full scale ±1% of full scale ±1% of full scale ±1% of full scale	12 months
Hydronic differential pressure instrument	Minimum range 0 to 36 in. Hg	±1% of full scale	12 months

B.19 Data Acquisition Hardware

This section provides an overview of various types of data acquisition hardware that can be used in conjunction with the previously mentioned instrumentation to allow for real-time data collection. This is intended to assist the end user with electronic collection of data, expediting the analysis process.

This overview covers:

- Data acquisition system selection
- Data acquisition system examples
- Data acquisition system demonstration

The following considerations must be taken into account in the selection of data acquisition system equipment:

- Input signals (type, range, number)
- Accuracy requirements
- Logging requirements (duration, frequency)
- Measurement locations (remote, local)
- Operating environment
- Ease of use
- Cost

The following are the typical sensors used for a chiller:

- Surface mount RTDs
- Pencil RTDs
- Ultrasonic flow meter
- Differential pressure transmitter

For an air conditioning unit, the following additional sensors are typically used:

- Humidity sensor (or dew point meter)
- Barometer pressure transmitter
- Propeller anemometer
- Pitot tube

As described in Appendix D, “Examples of Testing and Analysis Methods,” an important consideration in monitoring and testing is instrument uncertainty. The uncertainty of the parameter measurement stems from a combination of the sensor and measurement loop uncertainty contributors, which include:

- Instrument bias/calibration accuracy for sensor and data acquisition system (DAS)
- Installation effects
- Environmental effects

In addition, logging requirements must be satisfied. The following three questions help to determine the data storage requirements and how fast the data acquisition system must be:

- How often are data to be recorded?
- How long will the test/monitoring last?
- How many measurement points are required?

The location of measurement points should be carefully considered when determining data acquisition requirements. If sensor locations are in remote areas, a second data logger may be required. Placement of data loggers and any controlling computers should also be considered because local environmental conditions (for example, temperature and radiation dose) could adversely affect personnel or equipment accuracy.

DASs that are self-contained units with signal conditioning and multi-plexing built in are well designed for equipment testing and monitoring. Following are examples of these units, their advantages and disadvantages, and their capabilities.

The four DASs most suited for these applications are:

- Fluke Hydra series
- Fluke NetDAQ
- Keithley 2700 series
- HP/Agilent 34970A

The following models are part of the Fluke Hydra series:

- Hydra 2635A Data Bucket
- Hydra 2625 Data Logger
- Hydra 2620 Data Acquisition Unit

Figure B-3 illustrates the Fluke Hydra series.

Figure B-3
Fluke Hydra Series

Table B-5 presents the specifications for the Fluke Hydra DAS.

Table B-5
Fluke Hydra Data Acquisition System Specifications

Function	Range	Accuracy
DC Volts	90 mV to 150 V	±0.018%
Resistance	300 Ω to 10 MΩ	±0.013%
Frequency	15 Hz to 1 MHz	±0.05%
RTD	-200 to 600°C	±0.05°C
K Type TC	-100 to 1372°C	±0.45°C

The following are the advantages of the Fluke Hydra DAS:

- Stand-alone data acquisition requires no controlling software or PC.
- Onboard data storage or real-time data collection via RS-232 or IEEE 488 interface.
- Universal input module.
- Built-in digital inputs and totalizer.
- Front panel setup and operation.

The following are the disadvantages of the Fluke Hydra DAS:

- Low channel density with a maximum of 20 analog inputs per unit.
- RS-232 and IEEE-488 interfaces limit signal distances.
- Hydra Logger software does not interface with non-hydra components.
- Data manipulations are limited by Hydra Logger channel options.

There are two models within the Fluke NetDAQ series:

- 2640 NetDAQ
- 2645 NetDAQ

Figure B-4 illustrates the Fluke NetDAQ series.

Figure B-4
Fluke NetDAQ Series

Table B-6 presents the specifications for the Fluke NetDAQ series DAS.

Table B-6
Fluke NetDAQ Series Data Acquisition System Specifications

Function	Range	Accuracy
DC Volts	90 mV	0.01% + 7 μV
	300 mV	0.01% + 15 μV
	3 V	0.01% + 1 mV
	30 V	0.01% + 1.5 mV
	150 V/300 V	0.01% + 15 mV
Resistance	300 Ω	0.015% + 20 mΩ
	3 kΩ	0.02% + 0.3 Ω
	30 kΩ	0.03% + 3 Ω
	300 kΩ	0.1% + 40 Ω
	3 MΩ	0.25% + 800 Ω
Frequency	15 to 900 Hz	0.05% + 0.02 Hz
	900 Hz to 9 kHz	0.05% + 0.1 Hz
	9 to 90 kHz	0.05% + 1 Hz
	90 to 900 kHz	0.05% + 10 Hz
	1 MHz	0.05% + 100 Hz
4 Wire RTD	-200 to 600°C	0.06 to 0.19°C
K Type TC	-100°C to 1372°C	0.45°C

The following are the advantages of the Fluke NetDAQ series DAS:

- Ethernet connection allows for placement of units in remote locations.
- Up to 20 units can be scanned simultaneously.
- Universal input module.
- Built-in digital inputs and totalizer.
- Faster scan rates than the Hydra.

The following are the disadvantages of the Fluke NetDAQ series DAS:

- Low channel density with a maximum of 20 analog inputs per unit.
- Requires a computer with a network connection.
- NetDAQ Logger software does not interface with non-NetDAQ components.
- Data manipulations are limited by NetDAQ Logger channel options.
- Not configurable from the front panel.

There are two models within the Keithley 2700 series:

- 2700
- 2750

Figures B-5 and B-6 illustrate the Keithley 2700 series.

Figure B-5
Keithley 2700

Figure B-6
Keithley 2750

Table B-7 presents the specifications for the Keithley 2700 Series DAS.

Table B-7
Keithley 2700 Series DAS Specifications

Function	Range	Accuracy (2640A)
DC Volts	100 mV	0.0025% RDG + 0.0035% RNG
	1 V	0.0025% RDG + 0.0007% RNG
	10 V	0.0020% RDG + 0.0005% RNG
	100 V	0.0035% RDG + 0.0009% RNG
	1000 V	0.0035% RDG + 0.0009% RNG
Resistance	1 Ω	0.008% RDG + 0.008% RNG
	10 and 100 Ω	0.008% RDG + 0.002% RNG
	1 and 10 kΩ	0.008% RDG + 0.0006% RNG
	100 kΩ and 1 MΩ	0.008% RDG + 0.001% RNG
	10 MΩ	0.02% RDG + 0.001% RNG
	100 MΩ	0.2% RDG + 0.003% RNG
Frequency	3 Hz to 500 kHz	0.01% reading + 0.000033% span
RTD	-200 to 630°C	0.06°C
K Type TC	-200 to 1372°C	1.0°C

The following are the advantages of the Keithley 2700 Series DAS:

- Higher channel density with two or five slots.
- Variety of input modules are available.

- Fully functional digital multi-meter (DMM) with front panel inputs.
- Front panel operation.
- RS-232 and IEEE 488 interfaces for remote operation.
- True 6-1/2 digit accuracy.

The following are the disadvantages of the Keithley 2700 Series DAS:

- RS-232 and IEEE-488 interfaces limit signal distances.
- Higher channel density may be overkill, causing unnecessary expense.
- No built-in digital inputs or totalizer.

Figure B-7 illustrates the Agilent (formerly Hewlett-Packard) 34970A.

Figure B-7
Agilent 34970A

Table B-8 presents the specifications for the Agilent 34970A DAS.

Table B-8
Agilent 34970A Data Acquisition System Specifications

Function	Range	Accuracy (2640A)
DC Volts	100 mV	0.0040% RDG + 0.004% RNG
	1 V	0.0030% RDG + 0.0007% RNG
	10 V	0.0020% RDG + 0.0005% RNG
	100 V	0.0035% RDG + 0.0006% RNG
	300 V	0.0035% RDG + 0.003% RNG
Resistance	100 Ω	0.008% RDG + 0.004% RNG
	1, 10, and 100 kΩ	0.008% RDG + 0.001% RNG
	1 MΩ	0.008% RDG + 0.001% RNG
	10 MΩ	0.002% RDG + 0.001% RNG
	10 MΩ	0.8% RDG + 0.01% RNG
	100 MΩ	0.0080% RDG + 0.0080% RNG
Frequency	3 Hz to 5 kHz	0.1% reading
RTD	-200 to 600°C	0.06°C
K Type TC	-100 to 600°C	1.0°C

The following are the advantages of the Agilent 34970A DAS:

- Higher channel density than the Fluke systems with three slots.
- Variety of input modules are available.
- Front panel operation.
- RS-232 and IEEE 488 interfaces for remote operation.
- True 6-1/2 digit accuracy.

The following are the disadvantages of the Agilent 34970A DAS:

- RS-232 and IEEE-488 interfaces limit signal distances.
- No built-in digital inputs or totalizer.
- Data manipulations are limited by Benchlink software channel options.

Power Generation Technologies (PGT) specializes in testing and monitoring in the power plant industry. As a result of this specialization, PGT routinely uses data acquisition equipment. PGT has customized a data acquisition system that meets all of the pertinent criteria for equipment performance testing and monitoring in this field, which is different from the typical design application, in a laboratory for DASs (see Figure B-8).

This system has several key improvements over off-the-shelf DAS units:

- Provides a compact unit centered on an Agilent 34970A data logger/switch unit capable of up to 30 4-wire or 60 2-wire inputs[1]
- Is a rugged, self-contained unit designed for field use without special handling requirements
- Can withstand extreme temperatures and environments without affecting accuracy
- Manufactured to suit a variety of testing and monitoring needs
- Its sensors can be connected to any channel without requiring additional connectors, signal conditioners, or separate input cards
- Scan rate of up to 6 channels per second[2]
- Optional totalizer capability[3]
- Plug-and-play operation with PGT DAS software (see Section B.20)
- Front panel operation allows for manual configuration and monitoring of channels for easier diagnostics

Figure B-8
Power Generation Technologies Data Acquisition System Hardware

B.20 Data Acquisition Software

Typical off-the-shelf DAS hardware is packaged with data-specific software. PGT has developed a DAS software package, PGT-DAS, which allows the use of many types of hardware. This software has been developed under PGT's Quality Assurance Program, which complies with 10CFR50 Appendix B and ANSI/ASME NQA-1: 2000 requirements. It can therefore be used in safety-related applications. PGT-DAS offers the following features:

[1] Assumes three 34901A multi-plexers, each capable of 10 4-wire or 20 2-wire inputs.
[2] Scan rate with 4-wire resistances with 6-1/2 digit resolution and offset compensation.
[3] Each totalizer input replaces one 34901A multi-plexer and ten 4-wire or 20 2-wire inputs.

- Is compatible with Windows® 95/98, NT 4.0, 2000, and XP operating systems
- Is compatible with Microsoft® Excel™ 97, 2000, and XP
- Allows data acquisition from up to five data acquisition units simultaneously via one computer (150 4-wire inputs, 300 2-wire inputs, or any combination thereof)
- Combines reliable data acquisition with spreadsheet functionality for real-time display and manipulation of data (including graphing functionality of Excel™)
- Provides comprehensive data checks to ensure completeness and consistency in the setup and to identify bad data points and errors during data acquisition
- Separate sensor files store calibration information independent of setup and hardware configuration, eliminating the need to enter sensor information for every data acquisition need

Figure B-9 illustrates PGT-DAS during data acquisition.

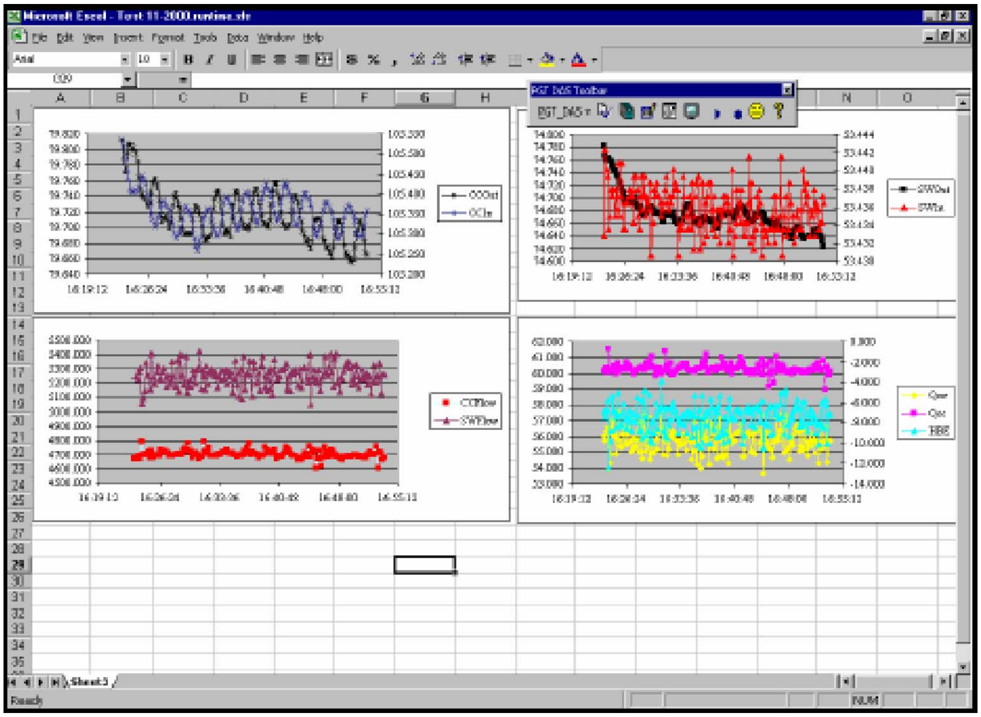


Figure B-9
Example of Power Generation Technologies Data Acquisition System During Data Acquisition

C OPERATING EXPERIENCE AND LESSONS LEARNED

The following cases have been submitted by utility personnel as illustrative examples that provide lessons learned in support of the guidance in this report. Actions described in these cases might not be suitable in all situations and should not be used in lieu of plant-specific procedures.

C.1 Support Systems That Affect Performance

Case #1:
Chiller tripped on low refrigerant pressure due to starting the chiller under abnormal operating conditions of low load coincident with low cooling water temperature. Starting the chiller under these conditions caused an increased rate of condensation, which resulted in an increase in the liquid refrigerant level in the condenser (that is, a stacking condition). The stacking condition in the condenser resulted in low suction pressure in the compressor suction.

Additionally, low load coupled with low cooling water temperature is known to cause stacking of refrigerant in the condenser, which results in reduced refrigerant inventory at the compressor suction. This reduction will cause the refrigerant operating point to drop much faster than the chilled water temperature and result in a low-suction pressure trip.

Lesson Learned: Support system parameters can have a major effect on system performance and operability. When starting a unit for testing, evaluate the conditions of all support systems for their possible effect on the component or system being tested.

Case #2:
The system's design condensing cooling water temperature is 105°F. The design leaving chilled water temperature is 42°F but at the time was operating at 40°F. The condensing cooling water temperature was lowered to 70°F in preparation for a refueling outage. The same cooling water is used for unit coolers in the containment building; the plant quickly lowers containment temperature, thereby supporting early containment maintenance. With the reduced condenser cooling water temperature, the chiller tripped on low suction pressure during startup.

Lesson Learned: Normally, the temperature of cooling water to the condensers is approximately 90°F. The machines can tolerate lower condenser cooling water temperatures during normal running operations. However, if a chiller is started with a hot chilled water loop and cold condenser cooling water temperatures, there is no guarantee it will run unless manual control is taken and it is loaded slowly. The refrigerant backs up in the condenser (that is, stacking) due to a lower condenser pressure brought on by low condenser cooling water temperatures. When the refrigerant is stacking in the condenser, the evaporator will boil dry, and pressure can decrease to the low evaporator trip set point.

Case #3:
On December 19, 2000, the plant was operating in Mode 1 at 100% power, and Unit 1 was in Mode 6. During a low-pressure (LP) service water system flow test, control room zone (CRZ) HVAC Chiller "A" tripped at 13:05. Chiller "B" subsequently tripped at 13:52. Units 2 and 3 entered Technical Specification (TS) 3.0.3 due to a loss of two chilled water trains. The flow test was aborted and air was vented from chiller service water (CSW) piping. Chiller "B" was started at 14:26 but was not immediately declared operable. Units 2 and 3 initiated power reduction at 14:52. At 15:21, Chiller "B" was declared operable, and both Units 2 and 3 exited TS 3.0.3 at 99.5% power.

Lesson Learned: A root cause investigation determined that the discharge piping configuration and flow test conditions allowed air that accumulated at the CSW discharge piping to pass through a recirculation valve to the idle "B" CSW recirculation loop and then into the "A" CSW suction. Air binding of the CSW pumps caused the chillers to trip. Corrective actions include procedural guidance for venting and planned piping changes to preclude air accumulation. During the event, CRZ temperatures remained well within TS 3.7.16 limits. This event is considered of no significance with respect to the health and safety of the public.

C.2 Minor Failure Mechanism Affects System Performance

Case #1:
On March 9, 2000, all three units at a multi-site plant were operating at 100% power. At 09:35, Operations swapped from the "B" control room (CR) air conditioning (A/C) chiller to the "A" chiller for routine monitoring of the "A" chiller. At 10:00, the "A" CR A/C chiller tripped. The "A" chiller was restarted but tripped again. At 10:27, the "B" chiller was restarted, but it tripped at 10:35. Operations declared entry into TS 3.0.3 for all three units. At 11:16, the Nuclear Regulatory Commission (NRC) was notified per 10CFR50.72. At 11:30, Operations began to decrease power.

Refrigerant leaks were found at tubing fittings on both chillers. Refrigerant was added to both chillers. The "B" and "A" chillers were declared operable at 12:46 and 15:36, respectively.

Specifics: York Model YSDCDBS3CNA0 - Rotary Screw Compressor, 380 tons, 875 gpm of chilled water at 45°F, return chilled water at 55°F. Condenser, 860 gpm, 90°F inlet and 102°F outlet.

Lesson Learned: Degraded copper tubing flares at filter/dryer connections on the chillers caused refrigerant leaks. The root cause was insufficient detail in the maintenance procedures to evaluate the flares. A lack of effective monitoring for refrigerant loss contributed to the event. The tubing has been replaced, and enhancements are planned for leakage monitoring.

Case #2:
Loose connection at the "B" terminal connection of the overload block to the motor starter caused pump to trip on motor overloads. Loose connection was found by thermography scan.

Specifics: Carrier 19FA, R12 refrigerant.

Lesson Learned: Periodic monitoring of electrical connections can identify equipment problems prior to failure and avoid potential detrimental system effects.

C.3 Periodic Monitoring Leads to Evaluation of Inadequate Design

Case #1:
During maintenance, the presence of babbited material was found in the refrigerant compressor oil system. Further evaluation determined that there was an inadequate design in the A/C subsystem. It was determined that the A/C subsystem was designed in excess of its expected service conditions. Additionally, the refrigerant system layout was designed in a configuration (for example, loop seal and traps) that was not ideally suited for its intended performance. The refrigerant compressors failed or operated in a degraded condition due to oil and refrigerant slugging or lack of adequate delivery of lubrication or both. The most probable causes for these compressor failures/degraded conditions are attributed to premature oil pump failures and system design deficiencies that allow for oil and liquid refrigerant slugging of the compressors during startup and system operation.

Design changes have been incorporated to improve system operational and monitoring characteristics. New refrigerant compressor suction accumulators have been installed to improve their capability to return oil to the compressor sump during low-load operation. Sections of refrigerant piping have been rerouted and resized to improve system performance characteristics. The thermal expansion valves (TEVs) have been resized to improve system control. New system thermostats have been installed to allow better control of the TEVs. New refrigerant compressor suction transducers and compressor motor current monitoring devices have been added to enhance the monitoring of critical system performance parameters. A new, simpler compressor oil pump design was incorporated to improve pump reliability. Compressor modifications have been implemented to reduce cyclic compressor operation that could lead to premature failure.

Lesson Learned: Periodic monitoring of component lube oil can identify equipment problems prior to failure and avoid potentially detrimental system effects.

C.4 Typical Failure Mechanism for Chiller Guide Vane Control

Case #1:
Chiller became inoperable due to cycle timer failure. Timer stuck in closed position caused the chiller to trip on low refrigerant pressure. The cycle timer is a lever-actuated microswitch operated by a small synchronous motor driving a cam. The cam allows the cycle timer to close for 1 second and open for 3 seconds. This slows down the speed of the guide vanes to prevent tripping the chiller on compressor overloads or low suction pressure or both when starting. In this case, the microswitch failed in the closed position and allowed the guide vanes to open continuously without interruption. The unit has experienced past failures of the cycle timer, but no specific relationship between time in service and failure has been determined. The failures have been attributed to the microswitch, not the synchronous motor. As a result, the cycle timers are replaced at least every 3 years.

Lesson Learned: Failure analysis leads to evaluation of preventive maintenance (PM) activities.

C.5 Chiller Trip As a Result of Less Than Adequate Control of the Chiller Thermostat Setting

Case #1:
The chiller thermostat settings had been lowered three times to lower chilled water temperature to ensure compliance with TS chilled water temperature limits. Adjustments were thought necessary to compensate for instrument drift. During subsequent troubleshooting, a chiller control circuitry wiring problem was identified and corrected. No evaluation was performed to determine if the current lower thermostat setting was applicable, and no adjustment was made to set the thermostat back to the higher setting once the wiring problem was corrected. Subsequent to the correction of the control circuitry, the chiller was placed back in service. The chiller then tripped on low refrigerant pressure due to starting the chiller under abnormal operating conditions of low load coincident with low cooling water temperature. This situation, coupled with a low thermostat set point, resulted in the dropping of the refrigerant operating point much more quickly than the chilled water temperature, which resulted in a low-suction-pressure trip.

Lesson Learned: Postulate system effects prior to proceeding with what appears to be a minor corrective maintenance activity.

C.6 Human Performance Errors

Case #1:
On April 8, 1999, all three units at a nuclear plant site were operating in Mode 1 at 100% power. At 12:00, Operators noticed that the Unit 3 CR area temperature was warmer than desired. The chilled water (WC) makeup tank was found empty, and the makeup line isolation valves were closed, contrary to procedure. The valves were reopened, and the WC system was refilled by 15:00. The TS limit of 80°F was not exceeded. The WC system is shared with the Unit 1and 2 CR, but the Unit 1and 2 CR was not affected. On April 29, 1999, Engineering concluded that WC cooling coils of both trains of the Unit 3 CR A/C system were airbound and inoperable during the event. This would have placed the Unit in TS 3.0.3.

Specifics: York Model YSDCDBS3CNA0 - Rotary Screw Compressor, 380 tons.

Lesson Learned: An investigation concluded that the most probable root cause was inadequate self-verification by an operator who performed a valve checklist on January 17, 1999. Prior trending had identified a small leak that allowed this system to partially drain while the makeup valves were closed. Corrective actions include revising the procedure to require a second checklist to be run as a verification.

C.7 Environmental Conditions Can Affect Component Life

Case #1:
Several years ago, it was noticed at one nuclear unit that the cooling fins on the outside condensing units (direct expansion [DX] systems) were degrading severely. The aluminum fins on copper coils were disintegrating when touched on “new” units that were only 5 years old. Previous units had lasted for approximately 9 years.

Industry peers, particularly in coastal areas (South Texas Plant is approximately 10 miles from the Gulf of Mexico), were polled regarding life expectancy of their outdoor cooling coils. The local industry was also polled. It was found that local petroleum and chemical plants minimize their maintenance and schedule replacement every 5 years.

Several vendors believed that, based on location, copper coils with copper fins were required. Multiple new units with both copper fins and coils were purchased and installed. Within 3 years, the copper fins are disintegrating.

Vendors currently recommend coated aluminum fins on copper coils.

Lesson Learned: Indeterminate as of June 2002.

C.8 Time Delay Relay Causes Chiller Trips

Case #1:
On June 6, 2000, while Oconee Units 1, 2, and 3 were operating at 100% power, CRZ HVAC Chiller "B" tripped at 14:13 while Chiller "A" was removed from service for maintenance. TS 3.0.3 was entered for all three units due to a loss of required chilled water per TS 3.16.7, Condition E. Chiller "A" was restored to operation per the maintenance contingency plan at 14:33. TS 3.0.3 was exited before the shutdown power reduction was initiated.

During the 20-minute period that both chillers were out of service, CRZ temperatures increased slightly but remained well within TS limits.

Specifics: York Model YSDCDBS3CNA0 - Rotary Screw Compressor, 380 tons.

Lesson Learned: Chiller "B" tripped due to a failure of one of two solenoids that control chiller load. The apparent cause of the solenoid failure was the failure of a time delay relay installed in chiller "B" in 1995. The failed time delay relay was removed, the failed solenoid replaced, and other potentially damaged circuit components replaced. Chiller "B" was tested and determined operable at 13:45 on June 9, 2000.

C.9 Chiller Trip Due to Leaving Chilled Water Temperature Switch

Case #1:
The design chilled water outlet temperature is 42°F. The chiller's load recycle leaving chilled water temperature switch has a set point of 36°F. The chiller tripped on load recycle at 46°F leaving chilled water temperature. The leaving chilled water temperature immediately reset before the machine could go through normal postlube cycle, and the Start signal was lost.

Specifics: Carrier 19FA, R12 refrigerant.

Lesson Learned: Temperature switches can fail and prevent chiller operation.

C.10 Chiller Trip Due to High Bearing/Compressor Temperature

Case #1:
The chiller's high bearing temperature trip is at 200°F and high compressor discharge temperature trip is at 200°F. During normal operation, the machine tripped due to the common high bearing/compressor temperature alarm. Refrigerant inventory was normal. Bearing temperature was normal. Oil inventory and temperature were normal. Condensing cooling water temperature was normal. An electronic module receives temperature resistance inputs from the bearing and compressor discharge temperature sensors. The temperature sensors were tested and found to be good. The electronic module 200°F set point was tested with corresponding resistance and found to trip low.

Specifics: Carrier 19FA, R12 refrigerant.

Lesson Learned: Electronic module set points can drift and should be tested with some frequency.

Case #2:
The chiller's high bearing temperature trip is at 200°F and high compressor discharge temperature trip is at 200°F. During normal operation, the machine tripped due to the common high bearing/compressor temperature alarm. Refrigerant inventory was normal. Bearing temperature was high. Oil inventory was high. Condensing cooling water temperature was normal. An electronic module receives temperature resistance inputs from the bearing and compressor discharge temperature sensors. The temperature sensors were tested and found to be good. The electronic module 200°F set point was tested with corresponding resistance and found to be good. Oil sump level had decreased due to frequent start and stop evolutions during troubleshooting. Oil was subsequently added to compensate for oil loss to the refrigerant.

Lesson Learned: Oil inventory must be closely tracked. Over time, oil will recover from the refrigerant and can overflow the sump. High oil inventory can cause higher than normal loads on the bearing, causing high bearing temperature trips.

C.11 Chiller Trip Due to Compressor Electrical Overloads

Case #1:
During normal startup, the chiller tripped on electrical overloads or high current. Refrigerant inventory was normal. Bearing temperature was high. Oil inventory was high. Condensing cooling water temperature was normal. Carrier provides an electrical component called a *cycle timer* to slow the rate of loading. The component has a small motor, a microswitch, and a cam. The cycle timer allows the compressor to load for 1 second and hold for 3 seconds. The cycle timer had failed, allowing the machine to continuously load, thereby tripping on electrical overloads.

Specifics: Carrier 19FA, R12 refrigerant.

Lesson Learned: A compressor loading too quickly can cause the machine to trip on overloads. The cycle timer should be replaced with some frequency to prevent failures.

C.12 Chiller Replacement and System Modifications Support Better Operation and Maintenance of Chillers

Component: Control Room Chillers.

__Case #1__:
Nuclear power plant personnel replaced their control room/computer room and cable spreading room chillers. Due to licensing basis, Point Beach does not have redundant trains for their chilled water system. The original chillers did not have adequate maintenance capability. When the systems were taken down for maintenance, all mechanical cooling was lost to the space. With the new design, there is the ability to split each refrigeration unit so that 50% at a time could be taken down for maintenance while still supplying cooling. The capability of cross-tying the systems was also added to permit cooling in the event that an entire chiller unit was required to be shut down. This has worked effectively supported operation and maintenance of the chillers. These enhancements are estimated to have added only about 10% to the overall cost of the project.

C.13 Component Oversizing

__Case #1__:
The heat rejection from the CR ventilation to the essential SW system was sized for 50 tons. When doing SW upgrades and system flow modeling, SW design flow could not be met in the model. A modification was proposed to upgrade the SW supply line size to meet the flow. Because of field observations and the generic Btu/sq. ft. approach taken in the original heat load/chiller sizing, it was known that the chiller units were oversized. A heat balance test on the CR was proposed to confirm the system's margin. A test plan was developed that measured all adjacent space temperatures, wall temperatures on each side of the boundary walls, supply duct flows, supply duct temperatures, return duct flows, return duct temperatures, outside air supply flows, and outside air supply temperatures. The data were used to recalculate the heat loads after adjusting the data to conditions expected during the design basis day. The SW loads were reduced by 30–40%, thereby not needing additional SW line modifications. The test and calculation cost approximately $20,000–$30,000.

Lesson Learned: Testing for actual heat load and recalculating actual component requirements should be considered anytime there are flow model deficiencies, chiller replacements, or upgrading to alternative refrigerants.

C.14 Oil Carryover Can Be Misleading

Case #1:
The operator rounds required operators to monitor the oil level to at least half in the sight glass. If the level were low, Operations would write a work order. This was a direct expansion unit that was built up with fairly long liquid supply and gas return lines. The unit had two-stage (two-liquid line) solenoid-operated valves (SOVs) with hot gas bypass on one stage. Hot gas bypass had an SOV of its own. For most of the operational time, the unit ran on one stage and no hot gas bypass.

The oil would hide out in the closed SOV lines and create the appearance of a low oil level condition. When the SOVs opened, a high oil level condition occurred. Maintenance was constantly adjusting oil to maintain proper oil levels.

Lesson Learned: The requirement for operator rounds was changed to consider a low oil level condition only if there were no oil or no oil splashing up into the sight glass.

C.15 Compressor Charge Issues

Case #1:
The operators, during their rounds, used bubbles in the sight glass to indicate if the unit had sufficient charge. Many work orders were written to troubleshoot problems if the bubbles were not present. It was concluded that the bubbles were present only when the compressors were not fully loaded.

Lesson Learned: Evaluating refrigerant level was assigned to the system engineer.

C.16 Consider Dedicated Maintenance Mechanics for Refrigeration Work

Case #1:
The utility was paying considerable amounts of money for contract refrigeration mechanics to perform maintenance on site because dedicated refrigerant mechanics were not on staff. Dedicated refrigerant mechanics were then hired.

Lesson Learned: Fewer equipment problems and expenses were incurred because of more component ownership.

C.17 "Improved" Parts Can Sometimes Be More Trouble Than the Originals

Case #1:
Service water cools the control room chillers. There was a long history of operational problems with the SW temperature/head pressure control valve. During the plant's extended outage, the valves were replaced with a ball-type valve with a "vee" port that had Teflon seats. The valves worked very well for the first several months.

One day, one chiller was out of service for maintenance, and one of the running chillers tripped. Operations was given a list of all of the likely causes, which included the unit tripping on high head pressure. Operations indicated that it checked the manual reset button on high head pressure and that it was satisfactory, but the unit would still not run. Operations had checked all of the manual resets for the unit in accordance with the procedure.

With the chillers down, Operations had started the control room emergency air conditioning system (CREACS) to keep the control room cool. The CREACS had been significantly altered during the outage; however, the new system was working well.

During troubleshooting of the chiller, the Reset button was accidentally pushed, restarting the chiller. The new SW head control valve was observed to be sticking and erratic in its movement. The ball valve seats were deformed and worn, causing the valve to hang up and the compressor to go out on high head pressure.

Lesson Learned: The new and improved valve was actually detrimental to the application's reliability.

C.18 Miscellaneous Direct Expansion Unit Items

Component: Belt Drive Evaporator Fans.

Case #1:
Fan belt adjustment was often made using the 1/2-in. use of the thumb for tension depending on "good mechanical practices" and "skill of the craft." Not until the drive belt manufacturer engineering data were used did the belt slipping and wear issues get resolved. Incorporating the type of belt, with the recommended force, at a specific deflection resolved the issue. The use of industry standard force gauges for this application should be used.

Component: Fan Belt - Match Set vs. Banded Belts.

Case #2:
In many applications, two or more belts are used to drive a fan or compressor. Even when these belts come as a matched set, the stretch and wear are unique to the belt. In some cases, the end user has been known to replace only the belt that broke or not verify the belt tension a day after installation. The result is that only one or two belts in the set carry the belt load. To resolve such issues, use the banded belt design.

Component: Clogged Thermostatic Expansion Valve (TXV) Screen.

Case #3:
Most TXVs will have a downstream screen located directly on the outlet to the valve. Clogging of these screen results in sluggish response by the system to control the system parameters or a lack of change in superheat when adjusting the TXV. Only when the TXV is replaced is this issue resolved. Much time and many materials can be expended in troubleshooting this type of problem.

Component: Sensing Bulb Location.

Case #4:
Much troubleshooting has been expended on a system that fails to properly control the space temperatures and control load on the machine by locating the hot gas bypass or TXV sensing bulb in the wrong location on the pipe or failing to insulate the sensing bulb. The sensing device manufacturer will provide excellent guidelines in locating the bulb. In many cases, locating the sensing bulb in the correct pipe orientation (horizontal vice vertical) along the flooded liquid plane rather than the top of the pipe will help to resolve the issue.

Component: Electrical Switch Failure.

Case #5:
Repeat Maintenance Rule functional failures have been incurred due to a failed switch. In this case, the compressor shut down, immediately restarted, and tripped the breaker. The switch consisted of a contact block assembly inside the panel with the operating mechanism outside the panel, which is used to take the component to run/off/ or other functions. In this case, a switch failure resulted in a disconnection of the contact block from the operating switch. The wires were properly connected, and the contacts were properly made. The switch was later reassembled and operated properly only to fail a few days later. The switch was again disassembled and, this time, the hand operating part consisted of a spring under the handle with a metal ball and a phenolic cam. The metal ball was supposed to be on top of the spring in a recess in the upper part of the switch. The ball was found not on top of the spring but recessed in a cavity under the hand grip. This ball provided the clicking action in a three- or two-position switch. Failure to “click” is a certain indication that the ball is not located on top of the spring.

Component: Load Control Solenoid Failure.

Case #6:
A typical DX load control scheme is to use the compressor-mounted unloaders to fine tune the load control and to use the load control solenoids to control which evaporator sections receive refrigerant for course load control. These load control solenoids interface with the return air temperature sensor to determine how many evaporator sections are used. A failure of any one solenoid or maladjustment of the same would result in the compressor running continuously to offset the space demands. A simple check of which solenoids are energized can be made using any metallic material, such as the end of a pin, without affecting the operation of the equipment. Frequent or routine system walkdowns will result in the system manager knowing how many solenoids are normally energized during certain times of the year.

Component: Fixed Sheave vs. Adjustable Sheaves.

Case #7:
In many applications, the evaporator fans for a DX unit can have adjustable sheaves installed. Adjustable sheaves are used in setting up the system at startup and are normally changed to a fixed sheave after optimum performance is obtained. The adjustable sheave sometimes comes out of adjustment, becomes imbalanced, or continues to add weight and load on the bearings

unnecessarily. Changing the adjustable sheave to a fixed sheave often eliminates future problems and results in extended bearing life.

Component: Facilitation of Oil Sampling.

Case #8:
Oil samples on some DX safety-related (SR) TS units require limiting condition for operations (LCO) time to draw an oil sample because the oil sample is taken from access plugs on the compressor body. This can be expedited by a modification to install oil sample valves that have been properly evaluated.

C.19 Low Oil Pressure Trips

Component: Essential services chilled water chiller.

Case #1:
Two low oil pressure trips of the WC-2A chiller occurred in less than 6 weeks. As a consequence of these events, the chiller was declared inoperable, a 72-hour LCO was entered, and a Maintenance Rule functional failure was recorded. No overtemperature conditions affecting SR areas or equipment occurred as a result of these trips.

Considerations included the following:

- Compressor Oil Transfer Pump Performance. This equipment failure relates to the 5–8 psig difference in output performance between the WC-2A and WC-2B oil pump. This additional margin explains the difference in susceptibility to trips on low oil pressure spikes between the two chiller units. (Contributing Cause)
- Sensing Line Tubing Configuration. This equipment failure relates to the differential pressure switch sensing line configuration, which can allow oil to be displaced by gas during standby conditions, causing pressure fluctuations at startup. (Potential Contributing Cause)
- Foaming of Oil Due to Absorbed Refrigerant. This equipment failure relates to foaming caused by the release of refrigerant from the compressor lube oil due to a change in system pressure at startup, causing significant pressure fluctuations. (Potential Contributing Cause)
- Managerial Methods/Inadequate Communication of Expectations/Goals/Policies. Repetitive chiller failures, which have not been reproduced during testing, have resulted in complacency on the part of site personnel regarding these events. This creates the potential for the inadequate addressing of the safety significance of chiller trips. The prevention of complacency is the goal of the site's initiative for zero tolerance of equipment failures.

Potential failure modes include refrigerant absorption in the oil and the time the unit is in standby. During standby conditions (that is, when the compressor is not running), pressure in the refrigeration unit reaches a steady-state condition (that is, saturation temperature/pressure of the refrigerant gas). This pressure is higher than the operating pressure normally seen at the compressor oil reservoir and increases the amount of refrigerant absorbed in the oil. A 1500-watt heater is provided in the oil reservoir to increase oil temperature in order to minimize oil absorption. Factors that influence gas absorption are the type of oil, the temperature of the oil, the age of the oil, and system standby pressures and temperatures.

When the compressor is started, the refrigeration unit undergoes a dramatic shift in pressures throughout the unit. The oil reservoir is located in an LP zone, which allows the absorbed refrigerant to emerge from solution, creating oil foaming. The oil foaming can create inadvertent fluctuations in the oil pressure sensing lines, leading to an instantaneous trip.

Corrective action taken to address this problem included the oil changeout and modification of the system logic. Refrigerant absorption cannot be eliminated. Factors such as oil temperature, oil age, and standby temperatures and pressures can all impact the absorption process. Oil replacement allows fresh, refrigerant-free oil to be added to the compressor lubrication system to minimize foaming during startup conditions. Modification of the chiller control logic will ensure that oil pump failures, and not spurious or inadvertent spikes, cause low oil pressure trips (due to the foaming) in the system.

Partial contributors include the following:

- Oil Pressure Sensing Line Configuration. The low oil pressure trip switch (PDS-9428) is a differential pressure switch, which senses low side pressure from the compressor housing (gas) and high-side pressure from the oil pump discharge (oil). The current configuration of the high-pressure side of the switch has the potential for some of the oil to be displaced by gas in the high-pressure sensing line during standby conditions, which can lead to instrument fluctuations and potential inadvertent trip on low differential oil pressure. The high-pressure sensing line on the WC-2B unit has a configuration different from that of WC-2A. Additional testing is planned to validate the actual effect that sensing line configuration has on PDS performance and will address the need for future changes.
- Working Differential Oil Pressure Margin Difference Between Units. The oil transfer pump discharge and differential pressure are tracked and trended for system monitoring purposes. The average oil discharge pressure of the WC-2A pump over the past 2 years has been stable (that is, no negative trends emerged) and averages approximately 82 psig. Although actual PDS-9428 values cannot be ascertained by currently installed instrumentation, the oil pump discharge pressure (PG-9428) minus the evaporator pressure (PG-9426) is consistent with the results obtained by EPT-285 (oil pressure signature testing). The calculated oil pump differential is also trended and has remained stable at approximately 43 psid (EPT-285 data indicated 44 psid and 43 psid, respectively).

Similar data from WC-2B indicate average oil pressure of 90 psig and 48-psid differential. The identified oil pump pressure differential is offset slightly by a known instrument inaccuracy of approximately 2.5 psig; however, a 5-psig differential is still apparent and provides additional margin for WC-2B. Additional testing and maintenance is planned to more accurately quantify the actual differences between the two units.

Several potential failure mechanisms were discussed and subsequently concluded to be inapplicable to this event. Of significant importance was the consideration of an electrical failure, similar to the November 1999 trip. This trip, also identified by a low oil pressure trip, was caused by a faulty oil transfer pump motor starter. The current motor starter was inspected, and no problems were identified. Oil analysis was also performed to eliminate the possibility that the wrong oil was used or that the oil was bad.

Lessons Learned: A Kepner-Tregoe change analysis of these events was conducted. This analysis concluded that a momentary low spike in the lube oil pressure during compressor startup was causing the trips; however, the exact cause of this LP spike cannot be pinpointed without additional testing. Further testing will concentrate on three possible causes/contributors:

- Compressor oil transfer pump performance. Lower margins in the lube oil pump discharge pressure on the “A” chiller (compared to the “B” chiller) can allow the pressure fluctuations to fall below the LP set point.
- Foaming of the oil due to absorbed refrigerant.
- Sensing line tubing configuration. Oil sensing line configuration for the “A” chiller can cause a collection of vapor on compressor starts, causing temporary instrument pressure fluctuations.

The chiller control logic will be modified by adding a time delay relay. This will prevent a momentary LP spike in the oil pressure from satisfying the compressor trip logic on low lube oil pressure, regardless of the exact cause.

C.20 Ensure That Preventive Maintenance of Appropriate Frequency Is in Place to Preclude Component Failures

Specifics: York HT Hermetic Turbopak Centrifugal Chiller, 200 tons, R-11.

Case #1:
On February 8, 1997, the 6.9-kV shutdown board chiller train “A” was placed in service, using the standard operating instruction, to perform a postmaintenance test (PMT) following a part replacement. The 6.9-kV shutdown board chillers are part of a SR system that provides cooling to the safe shutdown boards and auxiliary control room. An assistant unit operator and a system engineer were at the chiller to perform the PMT. Soon after startup, the condenser water temperature control valve (TCV) failed to open, causing the chiller discharge pressure to exceed the high trip point. The chiller’s control circuit initiated a chiller shutdown due to high discharge pressure. The breaker failed to open—causing the compressor motor to continue to run–and the chiller control circuit continued to shut down the control functions by placing the lubricating oil circuit in the compressor in shutdown mode. The lubricating oil was then completely isolated. The condenser water temperature control valve was opened by the manual override mechanism. The compressor motor breaker was manually tripped to stop the compressor from continuing to run without oil pressure. The compressor was estimated to have operated for 6 minutes, with no indicated oil pressure or flow for several minutes. The opposite train of the HVAC system was placed in service without any problems.

Troubleshooting found that the actuator of the TCV, NH-91 ITT Hydramotor, hydraulic piston seals had failed due to age. The root cause of the breaker failure to open was not determined, but the apparent cause was that the open mechanism failed. The compressor bearings, impeller, oil pumps, and seal were severely damaged. The hermetic motor received no significant damage.

The compressor and TCV actuator were rebuilt. The breaker had its periodic maintenance and inspections performed. The chiller's control circuit was tested to ensure that all functions would operate to shut down the chiller. All external control cables from the chiller control cabinet to the breaker compartment were inspected and tested. The interface relay between the control circuit and the breaker was tested and replaced.

A design change was completed to install a feedback circuit from the breaker to the chiller controls to prevent the oil system from stopping or isolating if the compressor motor were operating.

The failed chiller compressor and TCV were rebuilt, tested, and returned to service in 10 days.

Lesson Learned: Ensure that PM of appropriate frequency is in place to preclude component failures.

D
EXAMPLES OF TESTING AND ANALYSIS METHODS

The examples in this section represent different test methods used to verify the capacity of a chiller.

D.1 Functional Test Case Study

This case study presents the testing methodology and analysis for a complete functional test of a typical chiller. This test is possible only if all design-limiting condition parameters can be achieved. It test provides a direct measurement of the ability of the chiller to meet the design-limiting condition requirements. This test does not provide a quantitative assessment of the chiller's margin (that is, it does not determine the maximum heat load that could be removed by the chiller under the same conditions, but only if it can remove the applied test heat load).

D.1.1 Test Methodology Overview

The test methodology involves establishing the design-limiting flows and temperatures for the chiller and then demonstrating that the design-limiting condition heat load is removed from the chilled water system, including allowances for test uncertainty. For this test, the following measurements are made:

- Chilled water inlet temperature
- Chilled water outlet temperature
- Chilled water flow
- Cooling water inlet temperature
- Cooling water outlet temperature
- Cooling water flow
- Chiller power input

Not all parameters are required for determining the heat removal capacity of the chiller. Additional measurements can also be made to provide diagnostic or trending information about the chiller.

The target test conditions are for flow rates to be greater then those specified by the design-limiting condition flows and for process temperatures to be within ±2°F of the design-limiting condition temperatures. The applied heat load must also be sufficient to ensure that the heat load minus the uncertainty is greater than 141.4 tons.

This test methodology can require the use of intrusive means such as supplemental piping, in-line heaters, or other means to provide increased cooling water temperatures or increased chiller loads where the design-limiting conditions cannot be met by the existing plant systems.

D.1.2 Assumptions

For this test methodology, the following assumptions apply:

- The design-limiting cooling water flow and inlet temperature can be achieved.
- The design-limiting chilled water flow and outlet temperature can be achieved.
- Sufficient heat load is available to provide at least the design-limiting condition heat load plus any test uncertainty.
- The impact of the tolerances for the established target test condition temperatures (±2°F) is included in the design-limiting condition heat load requirement. This ensures that a test conducted at the target cooling water inlet temperature requirement will produce results that are as valid as a test conducted with a cooling water temperature at the lower end of the target window, which will result in a higher chiller capacity than the test conducted at the target cooling water temperature.
- The test is conducted under steady-state conditions.

D.1.3 Chiller Design-Limiting Conditions

Table D-1 shows the design-limiting conditions for the chiller.

Table D-1
Chiller Design-Limiting Conditions

Parameter	Value
Cooling water inlet temperature	95°F
Cooling water flow	630 gpm
Chilled water outlet temperature	52.5°F
Chilled water flow	303 gpm
Chilled water heat load	141.4 tons

D.1.4 Test Acceptance Criteria

The demonstrated heat removal rate for the chilled water system, including test uncertainty should be greater than or equal to the design-limiting condition value of 141.4 tons.

D.1.5 Nomenclature

Table D-2 describes the nomenclature used for the analysis.

Table D-2
Nomenclature for Functional Test Analysis

Symbol	Description	Units
$c_{p,ch}$	Heat capacity of chilled (CH) water	Btu/lbm/°F
q_{ch}	Volumetric flow rate of chilled water	gpm
Q_{ch}	Heat input from chilled water	Btu/hr
T	Temperature	°F
$T_{ch,I}$	CH temperature inlet to evaporator	°F
$T_{ch,o}$	CH temperature outlet from evaporator	°F
ρ_{ch}	Density of CH water	lbm/ft^3
W_{power}	Work input to compressor	Btu/hr

D.1.6 Example Test Data

Table D-3 describes the test data obtained.

Table D-3
Test Data

Parameter	Value	Uncertainty
Cooling water inlet temperature	95°F	0.2°F
Cooling water outlet temperature	103°F	0.3°F
Cooling water flow	630 gpm	25 gpm
Chilled water inlet temperature	64.5°F	0.2°F
Chilled water outlet temperature	52.5°F	0.3
Chilled water flow	303 gpm	12 gpm
Chiller power input	212 kW	4 kW

The cooling water data are measured for the purposes of calculating a heat balance.

D.1.7 Test Analysis

The object of the functional test is to demonstrate the capability of the chiller to remove the design-limiting heat load at the design-limiting process temperatures and flows, including test uncertainty.

D.1.8 Demonstrated Heat Load

The heat load on the chiller can be expressed by the following equation:

$$Q_{ch} = \rho_{ch} q_{ch} c_{p,ch} (T_{ch,i} - T_{ch,o}) 6.684E - 4$$

Using the test data values, the heat load on the chiller is:

$$Q_{chiller} = 0.999 * 62.3 * 303 * (64.5 - 52.5) * 6.684E - 4 = 151.3 \, tons$$

D.1.9 Demonstrated Heat Load Uncertainty

Evaluation of the test heat load uncertainty is based on the methods in ASME PTC 19.1. Two sources of error are addressed:

- Analytical uncertainty: Uncertainties associated with the calculation of the heat load on the chiller.
- Measurement uncertainty: Uncertainties in the measurement of the flow rate and temperatures. Included in this category are bias and precision uncertainties.

D.1.10 Analytical Uncertainty

Analytical uncertainty approximates the calculation errors associated with the misrepresentation, or lack of fit, by equations or subroutines used in the interim data reduction and analysis. Because the equations used in the heat load calculation are fundamental, well established, and well documented, these errors are assumed to be negligible using engineering judgment.

D.1.11 Measurement Uncertainty

The measurement uncertainty is calculated as shown below using the methods of ASME PTC 19.1 to develop sensitivity coefficients for each contributing parameter and then evaluating the resulting uncertainty using the test data and the associated parameter uncertainties:

$$U_R = \sqrt{\sum_{k=1}^{p} (\theta_k * U_k)^2}$$

where:

$U_R =$ the uncertainty in the test result

$U_k =$ the uncertainty in the measured test parameter k

$\theta_k =$ the sensitivity factor relating the test result to the measured test parameter k

Using the example data, the resulting uncertainty in the demonstrated heat load for the modified functional test is:

$$U_Q = 7.2 \, tons$$

D.1.12 Maximum Demonstrated Heat Load

The maximum demonstrated heat load carried by the chiller unit for the modified functional test is the calculated heat load minus the test uncertainty:

$$Q_{\max} = 151.3 - 7.2 = 144.1 \ tons$$

D.1.13 Test Disposition

Because the maximum demonstrated heat load is greater than the design-limiting condition of 141.1 tons, the chiller meets the requirements imposed by the design-limiting conditions.

D.2 Modified Chiller Functional Test

This case study presents a modified functional test methodology for testing a chiller by establishing design-limiting conditions on part of the chiller with the remainder of the chiller operating under simulated design-limiting conditions and then performing a separate test on the remaining component of the chiller. This test methodology does not require any physical modifications to the chiller or intrusive instrumentation, but does require the ability to provide a load on the chiller greater than the design-limiting condition load.

D.2.1 Test Methodology Overview

For this case study, a control room complex HVAC chiller with a single stage compressor and a single refrigeration loop is shown in Figure D-1. Cooling water is provided by the plant service water system.

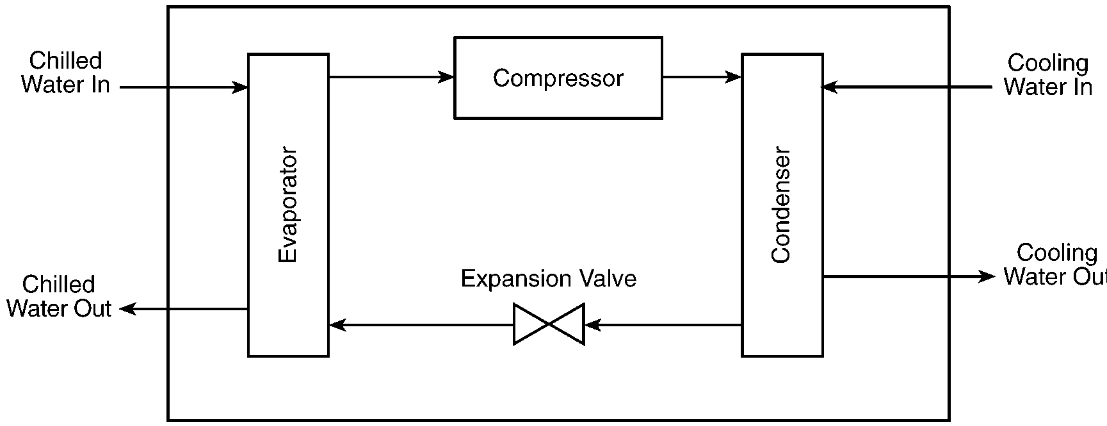


Figure D-1
Control Complex Chiller Diagram

Because all of the design-limiting conditions for the chiller cannot be achieved under normal operating conditions, the system parameters must be manipulated to achieve design-limiting conditions on those components of the chiller that can be operated at design-limiting conditions. For this chiller, the design-limiting condition service water temperature cannot be achieved under normal operating conditions. Therefore, the chiller must be tested in two steps.

D.2.2 Modified Functional Test

The modified functional test involves establishing design-limiting chilled water flow and a heat load equal to or greater than the design-limiting value and then demonstrating that the chiller can maintain the conditions under simulated elevated service water temperatures. By throttling the service water flow, the condenser pressure can be increased until the maximum allowable pressure is reached or the chiller fails to maintain the applied heat load. At this point, the evaporator, expansion valve, and compressor will be operating under their design-limiting conditions for the same condenser pressure and applied chilled water load.

The following parameters are recorded using temporary nonintrusive instrumentation:

- Chilled water inlet temperature
- Chilled water outlet temperature
- Chilled water flow
- Chiller power input
- Service water inlet temperature

- Service water outlet temperature
- Service water flow
- Condenser pressure
- Atmospheric pressure

The service water parameters and compressor power are recorded for the purposes of evaluating a heat balance around the chiller. The condenser pressure and chiller power will be inputs into the condenser test analysis.

The primary difficulty in establishing the test conditions is achieving a condenser pressure near that of the design-limiting condition value. Three variables influence the condenser pressure: chilled water load, service water flow, and service water temperature. As the service water flow is reduced, the flow can reach a regime where turbulent flow conditions are no longer present in the condenser tubes. As a result, it might not be possible to achieve the design-limiting condition condenser pressure before the chiller starts shedding the load due to insufficient condensing of the refrigerant. For this reason, it is desirable to test when the service water temperature is at its maximum achievable and a high chilled water load is applied. This will increase the initial condenser pressure prior to throttling back on the service water flow.

There are no specific requirements on the chilled water temperature. The chilled water temperature will vary with chiller load. Because the chiller is being used to provide chilled water for the HVAC system, the available heat load will be determined by the ambient conditions in the control room complex. The only temperature limitation is to maintain the control room complex temperatures less than the required maximums, which are verified as a part of the test procedure.

D.2.3 Condenser Test

In addition to the modified functional test, the condenser must be analyzed separately to determine its capability to meet the design-limiting conditions. This is done by establishing normal operating conditions on the chiller and acquiring data for the condenser. The same parameters measured for the modified functional test are recorded for the condenser test. The performance of the condenser is then analyzed at the test conditions and projected to the design-limiting conditions.

D.2.4 Assumptions

For this test methodology, the following assumptions apply:

- The chiller is limited by the ability of the condenser to remove the required heat load at the design-limiting condition service water temperature and condenser shell pressure. The ability of the remaining components of the chiller is demonstrated by the modified functional test.
- The correlations used for film heat transfer coefficients are appropriate for the chiller condenser performance calculations. The impact of potential errors in film heat transfer coefficients is included in the uncertainty analysis.

- The uncertainties in the material properties for the water and refrigerant are assumed to be negligible.
- The analysis is based on steady-state operation and does not include effects due to transient conditions.

D.2.5 Chiller Design Data

The design inputs are provided by the chiller manufacturer, chiller operator, or calculated from the design data.

D.2.6 Design-Limiting Conditions

Table D-4 displays the design-limiting conditions that are specified by the design basis documents for the chilled water system.

Table D-4
Chiller Design-Limiting Conditions

Parameter	Value
Service water flow	840 gpm
Service water temperature	102.5°F
Chilled water flow	520 gpm
Chilled water heat load	47.9 tons (574.8 kBtu/hr)
Condenser shell pressure	14.3 psig

D.2.7 Set Points

Table D-5 shows the control set points that limit the operation of the chiller.

Table D-5
Chiller Control Set Points

Parameter	Value
Chilled water outlet low temperature trip	38°F
Maximum chiller power input	192 kW
Compressor discharge high-temperature trip	170°F
Compressor motor current limit	269 Amps
Evaporator low-pressure trip	11.9 in Hg absolute
Condenser high-pressure trip	15 psig

D.2.8 Condenser Design Data

Table D-6 shows the condenser information.

Table D-6
Condenser Design Data

Parameter	Value
Heat transfer area (outside area of finned tubes)	1285 ft^2
Ratio of inside area to outside area	0.294
Number of tubes	184
Length of tubes	14
Outside diameter of finned tube	0.75 in.
Fin height (from base)	0.056 in.
Fin thickness	0.0205 in.
Inside diameter of tube	0.569
Fin spacing	19 fins/in.
Inside heat transfer area	377.8 ft^2
Outside diameter of unfinned tube	0.638 in.
Tube wall thickness	0.0345 in.
Mean heat transfer area	403.5 ft^2
Equivalent diameter of tubes	0.00785 ft
Tube side flow area	0.3249 ft^2
Refrigerant	R-11

D.2.9 Performance Parameters

The performance parameter for each test data set is described in Sections D.2.10 and D.2.11.

D.2.10 Modified Functional Test

For the modified functional test, the heat load on the chilled water system is the performance parameter. The result of the test is a comparison with the maximum demonstrated heat load on the chiller during the functional test with the design-limiting condition heat load of 47.9 tons.

D.2.11 Condenser Test

The chiller condenser performance can be defined in terms of the following parameters:

- Service water inlet temperature
- Service water flow rate
- Condenser pressure
- Heat transfer rate

The performance of the chiller condenser at the design-limiting conditions can be characterized by fixing three of the parameters to their design-limiting values and determining the fourth parameter. For this analysis, the service water inlet temperature is the performance parameter. The result of the test is a comparison of the projected service water inlet temperature with the design-limiting condition service water temperature of 102.5°F.

The projected heat transfer rate at the design-limiting conditions can also be selected as the performance parameter. The resulting analysis follows in a similar manner; however, there are a few modifications required in the resulting equations.

D.2.12 Nomenclature

Table D-7 displays the nomenclature that are used for the analysis.

Table D-7
Nomenclature for Modified Functional Test Analysis

Symbol	Description	Units
a_f	Area of one side of one fin	ft^2
$c_{p,ch}$	Heat capacity of CH water	Btu/lbm/°F
$c_{p,sw}$	Heat capacity of service water	Btu/lbm/°F
d_i	Inside diameter of condenser tubes	In.
d_o	Outside diameter of unfinned condenser tubes	In.
d_{fin}	Outside diameter of finned condenser tube	In.
d_{eq}	Equivalent diameter of finned condenser tube	Ft
f	Friction factor	Unitless
h_{fg}	Enthalpy of vaporization of R-11	Btu/lbm
h_g	Enthalpy of saturated gas R-11	Btu/lbm
h_i	Film heat transfer coefficient inside condenser tubes	Btu/hr ft^2 °F
h_{cond}	Film heat transfer coefficient outside condenser tubes	Btu/hr ft^2 °F
h_l	Enthalpy of saturated liquid R-11	Btu/lbm
k_{tube}	Thermal conductivity of condenser tube	Btu/hr ft^2 °F
k_{sw}	Thermal conductivity of service water	Btu/hr ft °F
$\dot{m}_{sw}$	Service water mass flow	lbm/hr
p_a	Atmospheric pressure	Psia
p_{cond}	Condenser pressure	Psia
$p_{cond,g}$	Condenser gauge pressure	Psig
q_{sw}	Volumetric flow rate of service water	Gpm
q_{ch}	Volumetric flow rate of chilled water	Gpm
r_{inside}	Ratio of inside surface area to outside surface area for condenser tubes	Unitless
A_{eff}	Effective heat transfer area	ft^2

Table D-7 (cont.)
Nomenclature

Symbol	Description	Units
A_{fin}	Heat transfer area corresponding to condenser tube fins	ft^2
A_i	Inside heat transfer area of the condenser	ft^2
A_{mean}	Mean heat transfer area, for tube wall resistance	ft^2
A_o	Total outside area of unfinned condenser tubes	ft^2
A_{prime}	Area of unfinned portion of finned condenser tubes	ft^2
A_{total}	Total outside heat transfer area of finned condenser tubes	ft^2
$A_{tubeflow}$	Tube side flow area	ft^2
F_1	Constant in equation for condensing film heat transfer coefficient	Unitless
K_1	Defined constant, used in calculating maximum service water inlet temperature	Unitless
L_{fin}	Fin height	In.
L_{mf}	Effective fin length	In.
L_{tube}	Length of condenser tubes	Ft
N_{tube}	Number of condenser tubes	Unitless
P_{ch}	Electrical power used by the chiller	Kw
Q_{ch}	Heat input from chilled water	Btu/hr
Q_{cond}	Chiller condenser heat transfer	Btu/hr
R_f	Total fouling resistance	hr ft^2 °F/Btu
$R_{f,I}$	Fouling resistance inside condenser tubes	hr ft^2 °F/Btu
$R_{f,o}$	Fouling resistance outside condenser tubes	hr ft^2 °F/Btu
R_{tube}	Condenser tube wall resistance	hr ft^2 °F/Btu
S_{fin}	Fin spacing	#/in.
T	Temperature	°F
$T_{ch,I}$	CH temperature inlet to evaporator	°F
$T_{ch,o}$	CH temperature outlet from evaporator	°F
$T_{sw,I}$	Service water temperature inlet to condenser	°F
$T_{sw,o}$	Service water temperature outlet from condenser	°F

**Table D-7 (cont.)
Nomenclature**

Symbol	Description	Units
T_{cond}	Condensing refrigerant temperature	°F
T_{surf}	Temperature of the finned surface	°F
U	Overall heat transfer coefficient	Btu/hr ft^2 °F
U_{clean}	Heat transfer coefficient for clean condenser tubes	Btu/hr ft^2 °F
U_{test}	Heat transfer coefficient for tested condenser tubes	Btu/hr ft^2 °F
V_i	Fluid velocity in condenser tubes	ft/hr
δ_{fin}	Fin thickness	In.
δ_{tube}	Tube wall thickness	In.
μ_{sw}	Viscosity of service water	lbm/hr ft
ρ_{ch}	Density of CH water	lbm/ft^3
ρ_{sw}	Density of service water	lbm/ft^3
ϕ	Fin efficiency (accounted for in condenser analytical model)	Unitless
ΔT_{lm}	Log mean temperature difference	°F
ΔT_{surf}	Temperature difference across the condensing refrigerant film	°F
W_{power}	Work input to compressor	Btu/hr
Subscripts		
ch	Chilled Water	N/A
sw	Service Water	N/A
i	Inlet or inside	N/A
o	Outlet	N/A
f	Fin	N/A
Superscripts		
*	At design-limiting conditions	N/A

D.2.13 Example Test Data

The test data in Sections D.2.14 and D.2.15 are provided for the purposes of demonstrating the test analysis methodology. Both the average parameter and the associated measurement uncertainty are presented.

D.2.14 Functional Test

Table D-8 shows the chiller test data.

Table D-8
Chiller Test Data

Parameter	Average Value	Uncertainty
Cooling water inlet temperature	88.17°F	0.11°F
Cooling water outlet temperature	104.76°F	0.18°F
Chilled water inlet temperature	56.03°F	0.11°F
Chilled water outlet temperature	49.47°F	0.15°F
Condenser pressure	13.76 psig	0.09 psi
Chiller power	151.7 kW	4.2 kW
Chilled water flow	525 gpm	16 gpm
Cooling water flow	263 gpm	17 gpm
Atmospheric pressure	14.75 psia	0.03 psi

D.2.15 Condenser Test

Table D-9 displays the condenser test data.

Table D-9
Condenser Test Data

Parameter	Average Value	Uncertainty
Cooling water inlet temperature	87.90°F	0.11°F
Cooling water outlet temperature	93.01°F	0.12°F
Condenser pressure	8.25 psig	0.09 psi
Chiller power	125.3 kW	3.8 kW
Cooling water flow	841 gpm	28 gpm
Atmospheric pressure	14.75 psia	0.03 psi

D.2.16 Modified Functional Test Analysis

The object of the modified functional test analysis is to calculate the maximum demonstrated heat load and to show that the heat load minus the uncertainty in the heat load is greater than the design-limiting condition heat load.

D.2.17 Demonstrated Heat Load

The heat load on the chiller can be expressed by the following equation:

$$Q_{ch} = \rho_{ch} q_{ch} c_{p,ch} (T_{ch,i} - T_{ch,o}) 6.684E-4$$

Using the test data values, the heat load on the chiller is:

$$Q_{chiller} = 1.00 * 62.4 * 525 * (56.03 - 49.47) * 6.684E-4 = 143.6 \quad tons$$

D.2.18 Demonstrated Heat Load Uncertainty

Evaluation of the test heat load uncertainty is performed in the same manner as described in Section D.1, the functional test case study. Using the example data, the resulting uncertainty in the demonstrated heat load for the modified functional test is:

$$U_Q = 6.5 \, tons$$

D.2.19 Functional Test Results

The maximum demonstrated heat load carried by the chiller unit for the modified functional test is the calculated heat load minus the test uncertainty:

$$Q_{\max} = 143.9 - 6.5 = 137.1 \quad tons$$

Because the maximum demonstrated heat load is greater than the design-limiting condition of 47.9 tons, the chiller meets the requirements imposed by the design-limiting conditions for the compressor, evaporator, and expansion valve at the condenser pressure achieved for the test. For this chiller, the applied heat load was significantly higher than that required by the design-limiting conditions. However, the increased heat load allowed for a higher condenser pressure at the reduced service water flow rate and validates the assumption that the chiller is limited by the condenser's operation at the design-limiting conditions.

D.2.20 Condenser Analysis

The object of the condenser analysis is to determine the maximum service water temperature for which the condenser will achieve the design-limiting condition heat transfer rate and condenser pressure.

D.2.21 Analysis Strategy

The basic analysis strategy is to use the test data to calculate a fouling resistance for the condenser at the test conditions and then to project the performance of the condenser at the design-limiting conditions using the test fouling resistance.

D.2.22 Analysis at Test Conditions

The overall heat transfer coefficient for the chiller condenser is calculated from:

$$Q_{cond} = U_{test} A_{total} \Delta T_{lm}$$

Therefore,

$$U_{test} = \frac{Q_{cond}}{A_{total} \Delta T_{lm}}$$

where:

Q_{cond} = the test heat transfer rate (Btu/hr)

A_{total} = heat transfer area of the chiller condenser based on the finned area on the outside of the condenser tubes (ft^2)

ΔT_{lm} = the log mean temperature difference for the condenser, based on the test data (°F)

The log mean temperature difference is:

$$\Delta T_{lm} = \frac{(T_{cond} - T_{sw,i}) - (T_{cond} - T_{sw,o})}{Ln\dfrac{(T_{cond} - T_{sw,i})}{(T_{cond} - T_{sw,o})}} = \frac{T_{sw,o} - T_{sw,i}}{Ln\dfrac{(T_{cond} - T_{sw,i})}{(T_{cond} - T_{sw,o})}}$$

The fouling resistance for the condenser is defined as:

$$R_f = \frac{1}{U_{test}} - \frac{1}{U_{clean}}$$

where:

R_f = test fouling resistance, based on the total finned area (hr ft^2 °F/Btu)

U_{clean} = predicted heat transfer coefficient for a clean condenser at test conditions (Btu/hr ft^2 °F)

The predicted value of the clean heat transfer coefficient at test conditions is calculated from:

$$\frac{1}{U_{clean}} = \frac{1}{h_{cond}} + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{total}}{A_i h_i}$$

where:

A_i = inside area of the condenser tube bundle (ft^2)

A_{mean} = mean area of the tube wall (ft^2)

R_{tube} = tube wall thermal resistance (hr ft^2 °F/Btu)

h_{cond} = heat transfer coefficient of the condensing refrigerant film (Btu/hr ft^2 °F)

h_i = film heat transfer coefficient inside the tubes (Btu/hr ft^2 °F)

Industry standard correlations are used to calculate the values of the film heat transfer coefficients at the test conditions. These coefficients are a function of service water temperature and flow rate.

D.2.23 Projection of Condenser Performance

To project the service water inlet temperature at the design-limiting conditions:

- The overall heat transfer coefficient at the design-limiting conditions is evaluated by:

$$\frac{1}{U} = \frac{1}{h_{cond}} + R_f + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{total}}{A_i h_i}$$

- The film heat transfer coefficients are evaluated at the design-limiting conditions of flow and temperature.
- The condensing refrigerant temperature is derived from the design-limiting condition of temperature and pressure.
- The log mean temperature difference at the design-limiting conditions is calculated by:

$$\Delta T_{lm} = \frac{Q_{cond}}{UA_{total}}$$

- The maximum service water inlet temperature can then be calculated from the log mean temperature difference

The evaluation of the chiller condenser performance must include consideration of the electrical power required by the chiller compressor. The required chiller condenser heat load at the design-limiting conditions is calculated as follows:

$$Q^*_{cond} = Q^*_{ch} + W^*_{power}$$

where:

Q^*_{ch} = required chilled water heat load at design-limiting conditions

W^*_{power} = electrical power required by the chiller compressor at design-limiting conditions

The power required by the chiller compressor at the design-limiting conditions can be conservatively estimated as the sum of the measured power and the uncertainty in the power measurement during the modified functional test. The required chiller condenser heat load at the design-limiting conditions for the example data, therefore, is:

$$Q^*_{cond} = 47.9 \quad tons \left(12{,}000 \quad \frac{Btu/hr}{ton}\right) + 157.1 \quad kW \left(3412 \quad \frac{Btu/hr}{kW}\right) = 1.111 \quad x \quad 10^6 \quad Btu/hr$$

During the modified functional test, it was not possible to achieve the chiller condenser design-limiting condition pressure of 14.3 psig. Because the value achieved was lower than the design-limiting condition pressure, the value used for the analysis is conservatively set equal to the measured condenser pressure during the modified functional test minus the uncertainty in the measurement. This conservatively predicts the projected service water inlet temperature. This conservative value of the chiller condenser pressure, used for the design-limiting condition analysis is:

$$p^*_{cond} = p_{cond,g} - U_{p_{cond,g}} = 13.76 - 0.09 = 13.67\ psig = 28.37\ psia$$

D.2.24 Analysis of Test Data

The overall thermal resistance for a condensing surface is:

$$\frac{1}{U} = \frac{1}{h_{cond}} + R_{f,o} + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{total}}{A_i} R_{f,i} + \frac{A_{total}}{A_i h_i}$$

where:

$R_{f,o}$ = fouling resistance on the outside of the condenser tubes

$R_{f,I}$ = fouling resistance on the inside of the condenser tubes

The objective of the calculations in this section is to calculate the fouling resistance of the condenser based on the test data. To accomplish this, the heat transfer resistance corresponding to the film heat transfer coefficients and the tube wall resistance is first calculated. The remaining heat transfer resistance is assigned to the fouling resistance.

For clean tubes the fouling resistance is, by definition, zero. Therefore:

$$U_{clean} = \frac{1}{\frac{1}{h_{cond}} + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{total}}{A_i h_i}}$$

The film heat transfer coefficients, h_{cond} and h_i, are evaluated at test conditions using standard correlations while the tube wall resistance is evaluated based on the thermal conductivity of the tube wall material.

The total fouling resistance of the chiller condenser is:

$$R_f = R_{f,o} + \frac{A_{total}}{A_i} R_{f,i} = \frac{1}{U_{test}} - \frac{1}{U_{clean}}$$

D.2.25 Condenser Heat Duty

The condenser heat duty is calculated as:

$$Q_{cond} = \dot{m}_{sw} c_{p,sw} \left(T_{sw,o} - T_{sw,i} \right)$$

where:

$\dot{m}_{sw}$ = mass flow rate of cooling water

$c_{p,sw}$ = heat capacity of cooling water

$$\dot{m}_{sw} = q_{sw} \rho_{sw}$$

q_{sw} = service water volumetric flow rate

ρ_{sw} = density of cooling water

At the test service water inlet temperature of 87.90°F:

ρ_{sw} = 62.14 lbm/ft^3

$c_{p,sw}$ = 0.999 Btu/lbm-°F

For the test service water flow rate of 841 gpm, the test mass flow rate is:

$$\dot{m}_{sw} = 841 \frac{62.14}{7.4805} 60 = 4.191\, x10^5\, \frac{lbm}{hr}$$

The resulting condenser heat duty is:

$$Q_{cond} = 4.191x10^5 \cdot 0.999 \cdot (93.01 - 87.90) = 2.139\, x10^6\, \frac{Btu}{hr}$$

D.2.26 Test Log Mean Temperature Difference

The test absolute condenser pressure is:

$$P_{cond} = P_{cond,g} + P_a$$

where:

$P_{cond,g}$ =test measured condenser pressure (8.25 psig)

P_a = test measured atmospheric pressure (14.75 psia)

Therefore:

$$P_{cond} \quad = \quad 8.25 + 14.75 = 23.00\ psia$$

The condensing refrigerant temperature corresponding to this pressure is 98.59°F.

The log mean temperature difference is:

$$\Delta T_{lm} = \frac{T_{sw,o} - T_{sw,i}}{Ln\dfrac{(T_{cond} - T_{sw,i})}{(T_{cond} - T_{sw,o})}} = \frac{93.01 - 87.90}{Ln\dfrac{(98.59 - 87.90)}{(98.59 - 93.01)}} = 7.86\ {}^{\circ}F$$

D.2.27 Test Overall Heat Transfer Coefficient

The test overall heat transfer coefficient is:

$$U_{test} = \frac{Q_{cond}}{A_{total}\Delta T_{lm}} = \frac{2.139\,x10^6}{1285 \cdot 7.86} = 211.7\frac{Btu}{hr \cdot ft^2 \cdot {}^{\circ}F}$$

D.2.28 Film Heat Transfer Coefficient Inside Tubes

The film heat transfer coefficient is calculated by the Petukhov-Kirillov equation:

$$Nu = \frac{h_i d_i}{k_{sw}} = \frac{\frac{f}{2}\mathrm{Re}\,\mathrm{Pr}}{1.07 + 12.7\sqrt{\frac{f}{2}}\left(\mathrm{Pr}^{2/3} - 1\right)}$$

where:

h_i = the heat transfer coefficient inside the tubes

d_i = the internal tube diameter

k_{sw} = the thermal conductivity of the fluid inside the tubes (water)

Pr = Prandtl number for the fluid inside the tubes

Re = Reynolds number defined as $\mathrm{Re} = \frac{\rho_{sw} V_i d_i}{\mu_{sw}} = \frac{\dot{m}_{sw} d_i}{A_{tubeflow}\mu_{sw}}$

V_i = the velocity of the fluid in the condenser tubes

ρ_{sw} = the density of the fluid in the condenser tubes

μ_{sw} = the viscosity of the fluid in the condenser tubes

The friction factor, f, is calculated by:

$$f = \left[1.58\ln(\mathrm{Re}) - 3.28\right]^{-2}$$

The fluid properties for water at the average service water temperature of 90.46°F are:

μ_{sw} = 1.82 lbm/hr – ft

k_{sw} = 0.358 Btu/hr-ft-°F

Pr = 5.07

The test Reynolds number is:

$$\mathrm{Re} = \frac{4.191\,x10^{5} \cdot 0.569}{0.3249 \cdot 12 \cdot 1.82} = 33{,}583$$

The friction factor is:

$$f = \left[1.58\ln(33{,}583) - 3.28\right]^{-2} = 0.005751$$

The Nusselt number is:

$$Nu = \frac{\frac{0.005751}{2} 33{,}583 \cdot 5.07}{1.07 + 12.7\sqrt{\frac{0.005751}{2}}\left(5.07^{2/3} - 1\right)} = 204.07$$

The resulting film heat transfer coefficient inside the tubes is:

$$h_i = Nu\frac{k_{sw}}{d_i} = 204.07\frac{0.358 \cdot 12}{0.569} = 1540\,Btu / hr \cdot ft^2 \cdot {}^\circ F$$

D.2.29 Tube Wall Resistance

The tube wall resistance is calculated from the thermal conductivity of the tube material (copper) by:

$$R_{tube} = \frac{\delta_{tube}}{k_{tube}}$$

The thermal conductivity of copper is 220 Btu/hr ft °F. Therefore, the thermal resistance of the tube wall is:

$$R_{tube} = \frac{0.0345}{12 \cdot 220} = 1.307\,x10^{5}\ hr \cdot ft^2 \cdot {}^\circ F / Btu$$

D.2.30 Condensing Refrigerant Film Heat Transfer Coefficient

The Beatty and Katz relation defines the heat transfer coefficient for the condensing refrigerant film on low-finned tubes:

$$h_{cond} = 0.689F_1\left(\frac{h_{fg}}{\Delta T_{surf}\,d_{eq}}\right)^{0.25}$$

where:

h_{fg} = the heat of vaporization of the refrigerant at the condensing temperature

d_{eq} = the equivalent diameter of the finned tubes, as previously calculated (0.00785 ft)

F_1 = constant based on refrigerant type, 153 $\left[\frac{Btu^3 lbm}{hr^4 ft^7 {}^\circ F^3}\right]^{0.25}$ for R-11 at Test Conditions (T_{cond} = 98.59°F)

ΔT_{surf} = temperature difference across the condensing refrigerant film,

$$\Delta T_{surf} = T_{surf} - T_{cond}$$

The temperature difference across the condensing refrigerant film and the temperature of the finned surface at the test conditions are not known but can be computed from:

$$\Delta T_{surf} = \frac{Q_{cond}}{A_{total}h_{cond}}$$

The equations for ΔT_{surf} and h_{cond} must be solved iteratively. Once the condensing heat transfer coefficient h_{cond} and temperature difference across the condensing refrigerant film ΔT_{surf} are calculated, T_{surf} can then be determined.

At test conditions (T_{cond} = 98.59°F) the enthalpy of saturated liquid and gas refrigerant are:

h_g = 104.21 Btu/lbm and

h_l = 28.45 Btu/lbm

The enthalpy of vaporization for the refrigerant is:

$$h_{fg} = 104.81 - 29.50 = 75.30\,Btu/lbm$$

The resulting heat transfer coefficient for the condensing refrigerant film, h_{cond}, is calculated to be 895 Btu/hr ft^2 °F with a surface temperature difference, ΔT_{surf}, of 1.86°F, and thus a surface temperature, T_{surf}, of 96.73°F.

D.2.31 Clean Overall Heat Transfer Coefficient

The overall heat transfer coefficient for clean tubes is calculated by:

$$U_{clean} = \frac{1}{\frac{1}{h_{cond}} + \frac{A_{total}}{A_{meaan}} R_{tube} + \frac{A_{total}}{A_i h_i}}$$

Therefore:

$$U_{clean} = \frac{1}{\frac{1}{895} + \frac{1285}{403.5} 1.307\,x10^{-5} + \frac{1285}{377.8 \cdot 1540}} = 296.9 \frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}$$

D.2.32 Test Fouling Resistance

The test fouling resistance is calculated as:

$$R_f = \frac{1}{U_{test}} - \frac{1}{U_{clean}} = \frac{1}{204.4} - \frac{1}{296.3} = 0.001517 \frac{hr \cdot ft^2 \cdot {}^\circ F}{Btu}$$

D.2.33 Projection of Cooling Water Inlet Temperature

The overall heat transfer coefficient at the design-limiting conditions is evaluated using the fouling resistance calculated at the test conditions, the tube wall resistance, and the film heat transfer coefficients at the design-limiting conditions:

$$U^* = \frac{1}{\frac{1}{h^*_{cond}} + R_f + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{tube}}{A_i h^*_i}}$$

From the basic equations for heat transfer, log mean temperature difference, and the condenser heat duty, the following equation for the design-limiting condition inlet service water temperature is derived:

$$T^*_{sw,i} = T^*_{cond} - \frac{K_1 Q^*}{(K_1 - 1)\dot{m}^*_{sw} c^*_{p,sw}}$$

where K_1 is defined as:

$$K_1 = \exp\left(\frac{U^* A_{total}}{\dot{m}_{sw}^* c_{p,sw}^*}\right)$$

D.2.34 Film Heat Transfer Coefficient Inside Tubes at Limiting Conditions

Start by assuming an average service water temperature at the design-limiting conditions. For this example, a value of 106.74°F is assumed. Property values for water at 106.74°F are summarized in Table D-10.

Table D-10
Property Values for Water at 106.74°F

Parameter	Units	Symbol	Value
Average service water temperature	°F	$T_{sw,avg}^*$	106.74
Density	lbm/ft^3	ρ_{sw}^*	61.92
Heat capacity	Btu/lbm °F	$c_{p,sw}^*$	0.999
Viscosity	lbm/ft hr	μ_{sw}^*	1.525
Thermal conductivity	Btu/hr ft °F	k_{sw}^*	0.365
Prandtl number	NA	Pr^*	4.17

The service water mass flow rate is:

$$\dot{m}_{sw}^* = q_{sw}^* \rho_{sw}^* = \frac{840 \cdot 61.92 \cdot 60}{7.4805} = 4.172\,x\,10^5\ lbm/hr$$

The Reynolds number is:

$$\mathrm{Re}^* = \frac{\rho_{ssw}^* V_i^* d_i}{\mu_i^*} = \frac{\dot{m}_{sw}^* d_i}{A_{tubeflow}\,\mu_{sw}^*} = \frac{4.172\,x\,10^5 \cdot 0.569}{0.3249 \cdot 12 \cdot 1.528} = 39{,}846$$

The friction factor is:

$$f^* = [1.58\ln(39{,}846) - 3.28]^{-2} = 0.005522$$

The Nusselt number is:

$$Nu^* = \frac{h_i^* d_i}{k_{sw}^*} = \frac{\frac{f^*}{2}\mathrm{Re}^*\,\mathrm{Pr}^*}{1.07 + 12.7\sqrt{\frac{f^*}{2}}\left(\left(\mathrm{Pr}^*\right)^{2/3} - 1\right)} = \frac{\frac{0.005522}{2}39{,}846 \cdot 4.17}{1.07 + 12.7\sqrt{\frac{0.005522}{2}}\left(\left(4.17\right)^{2/3} - 1\right)} = 215.1$$

The resulting film heat transfer coefficient inside the tubes is:

$$h_i^* = Nu^* \frac{k_{sw}^*}{d_i} = 215.1\frac{0.366 \cdot 12}{0.569} = 1658\,Btu/hr \cdot ft^2 \cdot {}^\circ F$$

D.2.35 Condensing Refrigerant Film Heat Transfer Coefficient

The condensing refrigerant film heat transfer coefficient is:

$$h_{cond}^* = 0.689 F_1 \left(\frac{h_{fg}^*}{\Delta T_{surf}^* d_{eq}} \right)^{0.25}$$

The condensing heat transfer coefficient h_{cond}, must be solved iteratively varying the temperature difference across the condensing refrigerant film ΔT_{surf}^* in the same manner as the test condition analysis.

At the design-limiting conditions (T_{cond}^* = 110.65°F for a condenser pressure of 28.37 psia) the enthalpy of saturated liquid and gas refrigerant are:

$h_g^* = 105.67\,Btu/lbm$ and

$h_l^* = 31.04\,Btu/lbm$

Therefore:

$$h_{fg}^* = h_g^* - h_l^* = 105.67 - 31.04 = 74.63\,Btu/lbm$$

The resulting heat transfer coefficient for the condensing refrigerant film, h_{cond}^* after iteration, is calculated to be 1099 Btu/hr ft^2 °F with a surface temperature difference, ΔT_{surf}^*, of 0.78°F, and therefore a surface temperature, T_{surf}^*, of 109.87°F.

D.2.36 Overall Heat Transfer Coefficient at Design-Limiting Conditions

The tube wall resistance and the fouling resistance are unchanged from the values calculated for the test data. The overall heat transfer coefficient at design-limiting conditions is:

$$U^* = \frac{1}{\frac{1}{h^*_{cond}} + R_f + \frac{A_{total}}{A_{mean}} R_{tube} + \frac{A_{tube}}{A_i h^*_i}}$$

$$U^* = \frac{1}{\frac{1}{1099} + 0.001357 + \frac{1285}{403.5} 1.307\,x10^{-5} + \frac{1285}{377.8 \cdot 1658}} = 229.4 \frac{Btu}{hr \cdot ft^2 \cdot {}^\circ F}$$

D.2.37 Projected Service Water Inlet Temperature at Design-Limiting Conditions

The projected service water inlet temperature at design-limiting conditions is calculated as:

$$T^*_{sw,i} = T^*_{cond} - \frac{K_1 Q^*}{(K_1 - 1)\dot{m}^*_{sw} c^*_{p,sw}}$$

where K_1 is defined as:

$$K_1 = \exp\left(\frac{U^* A_{total}}{\dot{m}^*_{sw} c^*_{p,sw}} \right)$$

$$K_1 = \exp\left(\frac{229.4 \cdot 1285}{4.172\,x10^5 \cdot 0.999} \right) = 2.029$$

$$T^*_{sw,i} = 110.65 - \frac{2.029 \cdot 1.107\,x10^6}{(2.029 - 1)4.172\,x10^5 \cdot 0.999} = 105.41^\circ F$$

The resulting service water outlet temperature and average service water temperature is calculated to verify that the initial assumed value is correct. If the calculated average service water temperature does not match the assumed value, the analysis process must be repeated until the values converge.

D.2.38 Uncertainty Analysis

The purpose of the uncertainty analysis is to evaluate the effect on the performance parameter due to uncertainty in the test measurements and the process of projecting the test results to the design-limiting conditions. Sources of error and uncertainty examined include:

- Measurement uncertainty: That associated with the measurement system. Included in this category are the consideration of instrument bias, spacial variation in the test measurements, and precision uncertainty.
- Process uncertainty: The effect of variations in the process being measured.
- Analytical uncertainty: The uncertainty associated with the ability of the analysis method to accurately characterize the performance of the condenser at test conditions and project its performance to the design-limiting conditions.

D.2.39 Measurement Uncertainty

Errors in the measurement of the test parameters will affect the value of the designated performance parameter. ASME PTC 19.1 Measurement uncertainty specifies a method for projecting and determining the uncertainty of test results based on the uncertainty in the test parameters:

$$U_R = \sqrt{\sum_{k=1,m} \left(\theta_{R,k} U_k\right)^2}$$

where:

U_R	=	uncertainty in the result
U_k	=	uncertainty for test parameter k
$\theta_{R,k}$	=	sensitivity factor relating test parameter k to the result

Sensitivity coefficients are defined as:

$$\theta_{R,k} = \frac{\delta R}{\delta k}$$

Because there is no direct analytical expression for the test result in terms of the individual test parameters, the sensitivity coefficients are numerically evaluated by:

$$\theta_{R,k} \approx \frac{\Delta R}{\Delta k}$$

where:

ΔR = the incremental change in the result

Δk = incremental change in parameter k

For this method, the value of the test parameter is adjusted numerically, and the resulting affect on the performance parameter is documented.

The overall measurement uncertainty in the predicted maximum service water inlet temperature at design-limiting conditions is calculated by combining the individual sensitivity coefficients and the associated parameter measurement uncertainties for each of the contributing measured parameters.

For the example data, the resulting overall measurement uncertainty in the maximum service water inlet temperature at design-limiting conditions is:

$$U_{T^*_{sw,i},meas} = 0.21^\circ F$$

D.2.40 Process Uncertainty

The condenser analysis method assumes steady-state conditions. Criteria are set for variations in test parameters for which the steady-state analysis method is valid. For the temperature measurements, the stability criteria is 2.4°F per hour. The stability criterion for the service water flow is 5% per hour.

Assuming that the stability criteria are met, the steady-state assumption is valid. Therefore, there is no additional uncertainty present due to process variation(s):

$$U_{T^*_{sw,i},process} = 0.00^\circ F$$

D.2.41 Analytical Uncertainty

The estimation of the analytical uncertainty requires the quantification of the probable error involved in the analysis methodology. The major source of the analytical errors is the uncertainty in film heat transfer coefficients. The equations used for these coefficients are general equations used for a wide variety of liquids and heat exchanger configurations. The Petukhov-Kirillov correlation used to calculate film heat transfer coefficients inside the condenser tubes has an accuracy of ±6 percent over a range of Reynolds numbers from 10,000 to 5,000,000 and a range of Prandtl numbers from 0.5 to 200. The Beatty and Katz correlation for the film coefficient for the condensing refrigerant has an uncertainty band +7.2 to -10.2 percent. The effect of errors in the calculation of these coefficients is analyzed by numerically calculating sensitivity coefficients with respect to the values of these coefficients.

This approach assumes that any errors in the heat transfer coefficients at the test conditions will be repeated at the design-limiting condition (that is, the uncertainties are correlated). This reasonable but potentially nonconservative assumption is justified by the proximity of the test conditions to the design-limiting conditions. For the condensing heat transfer coefficient, the calculated value at the design-limiting condition is 1099 Btu/hr-ft^2-°F compared to the test value of 895 Btu/hr-ft^2-°F. The value of the condensing heat transfer coefficient at design-limiting conditions is less than 23% greater than the value at test conditions. It is extremely unlikely that an error of 10% in the condensing film heat transfer coefficient at design-limiting conditions would not be repeated at the test conditions.

A similar argument can be made for the film heat transfer coefficients inside the tubes. The test film heat transfer coefficient inside the tubes is 1540 Btu/hr-ft^2-°F compared to a value of 1658 Btu/hr-ft^2-°F at the design-limiting condition. These differ by less than 2%.

The overall analytical uncertainty for the chiller condenser is calculated by combining the sensitivity coefficients and associated uncertainties for the film heat transfer coefficients:

$$U_{T^*_{sw,i},anal} = \sqrt{\left(\theta_{T^*_{sw,i},h_i} U_{h_i}\right)^2 \left(\theta_{T^*_{sw,i},h_{cond}} U_{h_{cond}}\right)^2}$$

where:

$\theta_{T^*_{sw,i},h_i}$ = the sensitivity coefficient for the film heat transfer coefficient inside the tubes

$\theta_{T^*_{sw,i},h_{cond}}$ = the sensitivity coefficient for the condensing film heat transfer coefficient

U_{h_i} = the uncertainty in the film heat transfer coefficient inside the tubes (assumed to be 10%)

$U_{h_{cond}}$ = the uncertainty in the condensing film heat transfer coefficient (assumed to be 10%)

The resulting analytical uncertainty for the example test data is:

$$U_{T^*_{sw,i},anal} = 0.03^\circ F$$

D.2.42 Overall Test Uncertainty

The overall test uncertainty is calculated as the combination of the measurement, process, and analytical uncertainties:

$$U_{T^*_{sw,i}} = \sqrt{\left(T_{T^*_{sw,i},meas}\right)^2 + \left(T_{T^*_{sw,i},proc}\right)^2 + \left(T_{T^*_{sw,i},anal}\right)^2}$$

$$U_{T^*_{sw,i}} = \sqrt{(0.21)^2 + (0.00)^2 + (0.03)^2} = 0.21^\circ F$$

D.2.43 Test Results

The maximum demonstrated service water inlet temperature to the chiller condenser, which will allow the required heat removal at the design-limiting conditions and under the present fouling conditions, it is conservatively set as the calculated inlet temperature minus the uncertainty in the calculated temperature:

$$\left(T^*_{sw,i}\right)_{\max\ dem} = T^*_{sw,i} - U_{T^*_{sw,i}} = 105.41 - 0.21 = 105.20^o\ F$$

Because this value is greater than the design-limiting condition value of 102.5°F, the results of the test show that the chiller satisfies the requirements imposed by the design-limiting conditions.

D.3 New Chiller Commissioning

This case study presents recommended data to be required from the chiller manufacturer as part of the chiller commissioning and a method for testing the new chiller based on the data provided by the manufacturer.

For new chillers, the performance of the chiller should be provided at both normal operating conditions in addition to design-limiting conditions. The Air-Conditioning Refrigeration Institute (ARI) standard for water chilling packages (ARI 550/590) provides guidance for chiller application ratings at conditions other than the design point. Specifically, the standard recommends providing ratings for the chiller under varying chilled water temperatures in increments of 2°F or less and condensing water temperatures in increments of 5°F or less. Guidelines for other types of chillers (for example, air-cooled or evaporatively cooled) are also provided. The range of temperatures provided for the performance curves should be representative of the expected operating conditions including normal and design-limiting conditions. With the performance of the chiller characterized accordingly, a functional test of the chiller is feasible under normal operating conditions without requiring intrusive instrumentation, piping reconfiguration, or excessive manipulations of chiller controls.

If the application rating for the new chiller is provided per ARI 550/590, testing of the new chiller can be done under normal operating conditions, provided the required heat load is available. Instead of requiring the design-limiting conditions for the test, the chiller can be tested at any point for the corresponding service water temperature and chilled water temperature and the results compared to the manufacturer's stated ratings without requiring extrapolation to design-limiting conditions.

D.3.1 Test Acceptance Criteria

Because the new chiller specification from the vendor should include performance curves at varying operating conditions, the acceptance criteria will be based on the stated chiller performance at the test service water and chiller water temperatures. The test acceptance criteria should reflect the possibility of chiller degradation and test uncertainty with respect to the vendor performance curve.

For example, the vendor chiller specification may indicate that the chiller being tested will carry a load of 125 tons at the design-limiting conditions. The design-limiting heat load requirement may be 110 tons, or 12% less than the stated capacity. Therefore, the acceptance criteria could be set at 12% less than the stated capacity for each point supplied by the vendor.

Alternatively, the vendor can provide the chiller performance based on a degraded condition. In this case, the acceptance criteria will be the degraded performance curves.

D.3.2 Example Chiller Specification

The chiller from the Modified Functional Test case study was required to satisfy the design-limiting conditions in Table D-11.

Table D-11
Chiller Design-Limiting Conditions

Parameter	Value
Service water flow	840 gpm
Service water temperature	102.5°F
Chilled water flow	520 gpm
Chilled water heat load	47.9 tons (574.8 kBtu/hr)

If this chiller were to be replaced with a new one for operation under the same design-limiting conditions, the data specified for the application rating for the new chiller could include ratings at the range of temperatures shown in Table D-12, based on the normal operating conditions for the existing plant systems.

Table D-12
Chiller Temperature Ratings

Parameter	Value
Chilled water outlet temperature	40 through 60°F
Service water inlet temperature	40 through 105°F

The new chiller should provide enough load capacity at the design-limiting conditions to allow for test uncertainty and some degradation in performance over time. This will result in a capacity greater than the required 47.9 tons at the design-limiting conditions for a new and clean chiller. This margin will be available for test uncertainty and performance degradation.

D.3.3 Test Methodology

According to ARI 550/590, the following parameters should be recorded for a chiller test:

- Condenser inlet temperature
- Condenser outlet temperature
- Condenser flow rate
- Condenser water differential pressure
- Evaporator inlet temperature
- Evaporator outlet temperature
- Evaporator flow rate
- Evaporator differential pressure
- Total power input to the chiller

Suggested additional data include the nameplate data for the chiller including make and model, ambient conditions, and actual motor voltages and currents for each phase.

The functional test for the new chiller should include the following steps:

1. Establish design flows for chilled water and service water.

2. Based on the service water temperature and chilled water temperature, establish a load on the chiller equal to or greater than that corresponding to the performance curves. Adjust the load as necessary to provide the maximum possible load on the chiller while maintaining stable conditions.

3. Allow conditions to stabilize.

4. Calculate a heat balance to ensure all instrumentation readings and conditions are stable and valid.

5. Verify that the chiller can maintain the applied load for the duration of the test.

D.3.4 Test Data Analysis

Test data analysis will be the same as that for the Functional Test case study, except that the acceptance criteria will be based on the specified heat load at the test chilled water and service water temperatures rather than a set heat load at one set of conditions.

D.4 Computer Applications

It may be beneficial to use computer analyses to assist in chiller performance evaluations. Computer programs are able to quickly predict expected chiller performance under various operating scenarios. This would be useful in both troubleshooting evolutions and in overall chiller performance analyses.

D.4.1 Troubleshooting

By comparing test output to the computer output, it is possible to determine if the chiller operation has degraded from its original design. An example:

> The chiller is tested under certain cooling water and chilled water inlet conditions. Test results include the outlet temperatures and operating pressures in the condenser and evaporator.
>
> A computer model of the chiller is run using the same inlet conditions as measured during the test.

The results from the computer program can be compared to the recorded values from the test. Differences in these values are an indication that something is not operating to its optimum capacity. The troubleshooting guideline can now be used to pinpoint the area to investigate.

D.4.2 Performance Analyses

A computer program is also useful in determining the expected performance of the chiller under various scenarios. These include:

- Tube plugging conditions
- Tube fouling conditions
- Design heat load evaluations

It is often difficult to use a test to prove the chiller's performance capabilities at all conditions. A computer program can help solve this problem. By first showing that test results match computer output at a given "testable" condition, the program can then be run at other conditions, such as the design heat load, to prove the chiller's performance.

E
LISTING OF KEY INFORMATION

The following list provides the location of "Key Point" information in this report.

Key O&M Cost Point

Emphasizes information that will result in reduced purchase, operating, or maintenance costs.

Referenced Section	Page Number	Key Point
4.1	4-1	Integration of daily checks in routine operator rounds is a cost-effective method for accomplishing chiller performance monitoring.
4.2.4	4-13	Leak checks prior to filling a refrigerant system can prevent costly refrigerant loss, unnecessary repairs, and future problems.
5.1.1	5-1	A structured approach to chiller troubleshooting reduces time and improves the effectiveness of the troubleshooting efforts.

Key Technical Point

Targets information that will lead to improved equipment reliability.

Referenced Section	Page Number	Key Point
3.2.3.3	3-12	The TXV sensing bulb must NOT be located at the bottom of the line. It must have good contact with the line and be insulated so that the temperature sensed is not affected by ambient temperature.
3.2.5	3-15	Receivers are typically sized to hold the entire system charge when 80% filled with liquid, which allows room for expansion.
3.2.6.5	3-18	A safety relief device is designed to relieve positively at its set pressure for one crucial occasion without prior leakage. The relief may be to atmosphere or to the low-pressure side.

Key Technical Point (cont.)

Targets information that will lead to improved equipment reliability.

Referenced Section	Page Number	Key Point
4.2.4	4-13	Under no conditions should the low-side pressure, or crankcase pressure, be allowed to exceed manufacturer's recommendations for leak testing. If it becomes necessary to mix nitrogen or carbon dioxide with the refrigerant to build up a satisfactory test pressure (such as in low ambient temperature), the gas cylinder must be equipped not only with a pressure regulator, but the charging line must incorporate a relief valve set to lower pressure when the maximum pressure per manufacturer's recommendations is reached.
5.1.4.1	5-3	Never draw a deep vacuum on a refrigerant system that contains liquid refrigerant.
5.1.4.2	5-4	To prevent damage, never allow liquid refrigerant to enter the suction line to the compressor.
5.1.4.2	5-4	Service or recovery cylinders that are not properly evacuated prior to use can contain objectionable amounts of moisture.
5.2.2	5-19	Ensure that uniform air or chilled water flow across or through the evaporator and that adequate load on the evaporator is available when reviewing causes of erratic thermostatic expansion valve operation.
B.1	B-1	An uneven airflow is frequently found when measuring airflow across the face of cooling coils because of entrance or exit conditions or stratification or both.
B.9	B-6	Manufacturer recommendations for calibration, inspection, maintenance, and/or installation criteria must be followed to ensure accurate readings.

Key Human Performance Point

Denotes information that requires personnel action or consideration in order to prevent injury or damage or ease completion of the task.

Referenced Section	Page Number	Key Human Performance Points
4.2.3	4-12	Caution should be taken when the suction shutoff valve is throttled closed too quickly due to the possibility of oil slugging the compressor.
4.2.4	4-14	Extreme caution should be exercised when using a halide torch. The room should be well ventilated to preclude injury and fire hazard.
5.1.4.1	5-3	Never operate or megger a hermetic motor while the system is in a deep vacuum.
B.11	B-7	Extreme caution must be used when working around hot piping.
B.14	B-8	Extreme caution must be used when working around energized electrical equipment.

Target:
Nuclear Power

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