Refrigeration Manual

Part 1 - Fundamentals of Refrigeration

Copeland ®
brand products

EMERSON ™
Climate Technologies

FOREWORD

The practice of refrigeration undoubtedly goes back as far as the history of mankind, but for thousands of years the only cooling mediums were water and ice. Today refrigeration in the home, in the supermarket, and in commercial and industrial usage is so closely woven into our everyday existence it is difficult to imagine life without it. But because of this rapid growth, countless people who must use and work with refrigeration equipment do not fully understand the basic fundamentals of refrigeration system operation.

This manual is designed to fill a need which exists for a concise, elementary text to aid servicemen, salesman, students, and others interested in refrigeration. It is intended to cover only the fundamentals of refrigeration theory and practice. Detailed information as to specific products is available from manufacturers of complete units and accessories. Used to supplement such literature—and to improve general knowledge of refrigeration—this manual should prove to be very helpful.

Table of Contents

Section 1
BASIC REFRIGERATION PRINCIPLES

Most users of refrigeration products normally associate refrigeration or air conditioning with cold and cooling, yet the practice of refrigeration engineering deals almost entirely with the transfer of heat. This seeming contradiction is one of the most fundamental concepts that must be grasped to understand the workings of a refrigeration or air conditioning system. Cold is really only the absence of heat, just as darkness is the absence of light, and dryness is the absence of moisture.

THERMODYNAMICS

Thermodynamics is that branch of science dealing with the mechanical action of heat. There are certain fundamental principles of nature, often called laws of thermodynamics, which govern our existence here on Earth. Several of these laws are basic to the study of refrigeration.

The first and most important of these laws is the fact that energy can neither be created or destroyed. It can only be converted from one type to another. A study of thermodynamic theory is beyond the scope of this manual, but the examples that follow will illustrate the practical application of the energy law.

HEAT

Heat is a form of energy, primarily created by the transformation of other types of energy into heat energy. For example, mechanical energy turning a wheel causes friction and is transformed into heat energy. When a vapor such as air or refrigerant is compressed, the compression process is transformed into heat energy and heat is added to the air or refrigerant.

Heat is often defined as energy in motion, for it is never content to stand still. It is always moving from a warm body to a colder body. Much of the heat on the Earth is derived from radiation from the sun. The heat is being transferred from the hot sun to the colder earth. A spoon in ice water loses its heat to the water and becomes cold. Heat is transferred from the hot spoon to the colder ice water. A spoon in hot coffee absorbs heat from the coffee and becomes warm. The hot coffee transfers heat to the colder spoon. The terms warmer and colder are only comparative. Heat exists at any temperature above absolute zero even though it may be in extremely small quantities.

Absolute zero is the term used by scientists to describe the lowest theoretical temperature possible, the temperature at which no heat exists. This occurs at approximately 460° below zero Fahrenheit, 273° below

zero Celsius. By comparison with this standard, the coldest weather we might ever experience on Earth is much warmer.

TEMPERATURE

Temperature is the scale used to measure the intensity of heat, the indicator that determines which way the heat energy will move. In the United States, temperature is normally measured in degrees Fahrenheit. The Celsius scale (previously termed Centigrade) is widely used in most other parts of the world. Both scales have several basic points in common, (See Figure 1-1) the freezing point of water, and the boiling point of water at sea level. At sea level, water freezes at 32°F (0°C) and water boils at 212°F (100°C). On the Fahrenheit scale, the temperature difference between these two points is divided into 180 equal increments or degrees F, while on the Celsius scale the temperature difference is divided into 100 equal increments or degrees C. The relation between Fahrenheit and Celsius scales can always be established by the following formulas:

Fahrenheit = 9/5 Celsius + 32°
Celsius = 5/9 (Fahrenheit -32°)
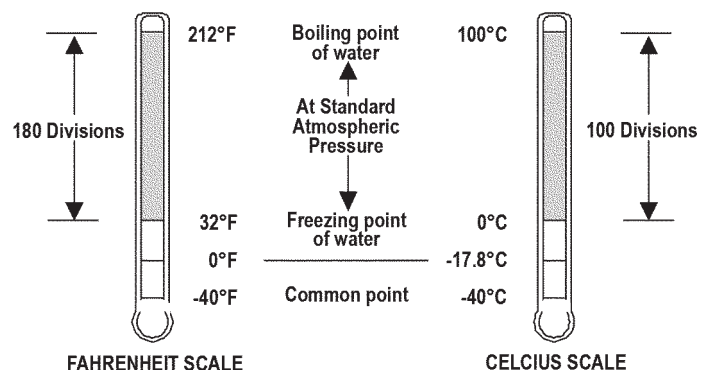
COMPARISON OF TEMPERATURE SCALES
Figure 1-1

Further observing the two scales, note that at -40°, both the Fahrenheit and Celsius thermometers are at the same point. This is the only point where the two scales are identical. Using this information, the following formulas can be used to determine the equivalent Fahrenheit or Celsius values.

Fahrenheit = ((Celsius + 40) x 9/5) - 40
Celsius = ((Fahrenheit + 40) x 5/9) - 40

HEAT MEASUREMENT

The measurement of temperature has no relation to the quantity of heat. A match flame may have the same temperature as a bonfire, but obviously the quantity of heat given off is vastly different.

The basic unit of heat measurement used today in the United States is the British Thermal Unit, commonly expressed as a BTU. A BTU is defined as the amount of heat added or removed to change one pound of water one degree Fahrenheit. For example, to raise the temperature of one gallon of water (approximately 8.3 pounds) from 70°F to 80°F will require 83 BTUs.

1 gallon (8.3 pounds) x (80°F - 70°F)ΔT = 83 BTUs heat added

8.3 pounds x 10°ΔT = 83 BTUs

In the metric system, the basic unit of heat measurement is the Calorie. A Calorie is defined as the amount of heat added or removed to change one gram of water one degree Celsius. For example, to lower one liter of water (1000 grams) from 30°C to 20°C will require 10,000 Calories of heat to be removed.

1000 grams X (30°C - 20°C)ΔT = 10,000 Calories of heat removed.

HEAT TRANSFER

The second important law of thermodynamics is that heat **always** travels from a warm object to a colder one. The rate of heat travel is in direct proportion to the temperature difference between the two bodies.

390°F ← **400°F**

Heat Flow
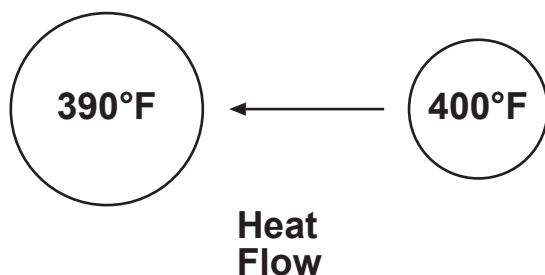
Figure 1-2

Assume that two steel balls are side by side in a perfectly insulated box. One ball weighs one pound and has a temperature of 400°F, while the second ball weighs 1,000 pounds and has a temperature of 390°F. The heat content of the larger ball is much greater than the small one, but because of the temperature difference, heat will travel from the small ball to the large one (See Figure 1-2) until the temperatures equalize. Heat can

travel in any of three ways; radiation, conduction, or convection.

Radiation is the transfer of heat by waves similar to light waves or radio waves. For example, the sun's energy is transferred to the Earth by radiation.
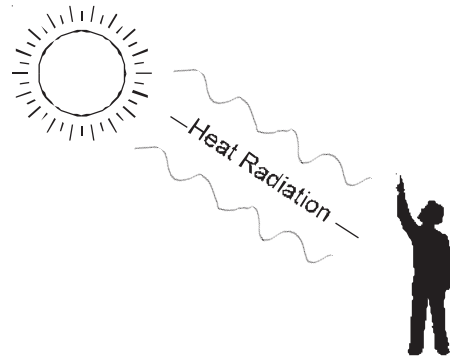
— Heat Radiation —

Figure 1-3

One need only step from the shade into direct sunlight to feel the impact of the heat waves even though the temperature of the surrounding air is identical in both places. Another example of radiation is standing in front of a bonfire. The side of you facing the bon fire is receiving radiant heat and that side is hot. The side away from the fire may feel cool. There is little radiation at low temperatures and at small temperature differences. As a result, radiation is of little importance in the actual refrigeration process. However, radiation to the refrigerated space or product from the outside environment, particularly the sun, may be a major factor in the refrigeration load.

Conduction is the flow of heat through a substance. Actual physical contact is required for heat transfer to take place between two bodies by this means. Conduction is a highly efficient means of heat transfer as any serviceman who has touched a piece of hot metal can testify.

HOT   WARM   COOL
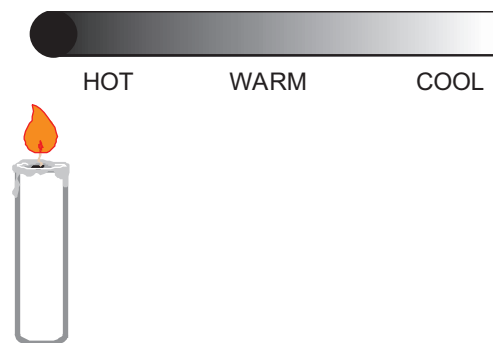
Figure 1-4

Figure 1-4 shows a flame heating one end of a metal rod. Heat is conducted to the other end by the process of conduction.

Convection is the flow of heat by means of a fluid medium, either vapor or liquid, normally air or water. Air may be heated by a furnace, and then discharged into a room to heat objects in the room by convection.
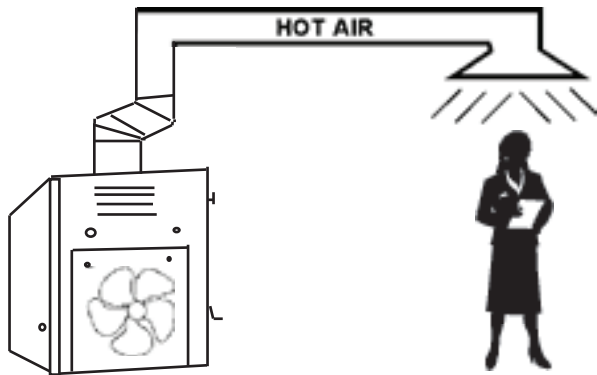
Figure 1-5

In a typical air conditioning/refrigeration application, heat normally will travel by a combination of processes. The ability of a piece of equipment to transfer heat is referred to as the overall rate of heat transfer. While heat transfer cannot take place without a temperature difference, different materials vary in their ability to conduct heat. Metal is a very good heat conductor. Fiberglass has a lot of resistance to heat flow and is used as insulation.

CHANGE OF STATE

Most common substances can exist as a solid, a liquid, or a vapor, depending on their temperature and the pressure to which they are exposed. Heat can change their temperature, and can also change their state. Heat is absorbed even though no temperature change takes place when a solid changes to a liquid, or when a liquid changes to a vapor. The same amount of heat is given off, rejected, even though there is no temperature change when the vapor changes back to a liquid, and when the liquid is changed back to a solid.

The most common example of this process is water. It generally exists as a liquid, but can exist in solid form as ice, and as a vapor when it becomes steam. As ice it is a usable form for refrigeration, absorbing heat as it melts at a constant temperature of 32°F (0°C). As water, when placed on a hot stove in an open pan, its temperature will rise to the boiling point, 212°F (100°C) at sea level. Regardless of the amount of heat applied, the waters temperature cannot be raised above 212°F (100°C) because the water will vaporize into steam. If

this steam could be enclosed in a container and more heat applied, then the water vapor, steam, temperature could again be raised. Obviously the fluid during the boiling or evaporating process was absorbing heat.

When steam condenses back into water it gives off exactly the same amount of heat that it absorbed during evaporation. (The steam radiator is a common usage of this source of heat.) If the water is to be frozen into ice, the same amount of heat that was absorbed in melting must be extracted by some refrigeration process to cause the freezing action.

The question arises, just where did those heat units go? Scientists have found that all matter is made up of molecules, infinitesimally small building blocks which are arranged in certain patterns to form different substances. In a solid or liquid, the molecules are very close together. In a vapor the molecules are much farther apart and move about much more freely. The heat energy that was absorbed by the water became molecular energy, and as a result the molecules rearranged themselves, changing the ice into water, and the water into steam. When the steam condenses back into water, that same molecular energy is again converted into heat energy.

SENSIBLE HEAT

Sensible heat is defined as the heat involved in a change of temperature of a substance. When the temperature of water is raised from 32°F to 212°F, an increase in sensible heat content is taking place. The BTUs required to raise the temperature of one pound of a substance 1°F is termed its specific heat. By definition, the specific heat of water is 1.0 BTU/lb. The amount of heat required to raise the temperature of different substances through a given temperature range will vary. It requires only .64 BTU to raise the temperature of one pound of butter 1°F, and only .22 BTU is required to raise the temperature of one pound of aluminum 1°F. Therefore the specific heats of these two substances are .64 BTU/lb. and .22 BTU/lb. respectively. To raise the temperature of one pound of liquid refrigerant R-22, 1°F from 45° to 46°, requires .29 BTUs, therefore its specific heat is .29 BTU/lb.

LATENT HEAT OF FUSION

A change of state for a substance from a solid to a liquid, or from a liquid to a solid involves the latent heat of fusion. It might also be termed the latent heat of melting, or the latent heat of freezing.

When one pound of ice melts, it absorbs 144 BTUs at a constant temperature of 32°F. If one pound of water is to be frozen into ice, 144 BTUs must be removed from the water at a constant temperature of 32°F. In the freezing of food products, it is only the water content

for which the latent heat of freezing must be taken into account. Normally this is calculated by determining the percentage of water content in a given product.

LATENT HEAT OF EVAPORATION

A change of a substance from a liquid to a vapor, or from a vapor back to a liquid involves the latent heat of evaporation. Since boiling is only a rapid evaporating process, it might also be called the latent heat of boiling, the latent heat of vaporization, or for the reverse process, the latent heat of condensation.

When one pound of water boils or evaporates, it absorbs 970 BTUs at a constant temperature of 212°F (at sea level). To condense one pound of steam to water, 970 BTUs must be extracted from the steam.

Because of the large amount of latent heat involved in evaporation and condensation, heat transfer can be very efficient during the process. The same changes of state affecting water applies to any liquid, although at different temperatures and pressures.

The absorption of heat by changing a liquid to a vapor, and the discharge of that heat by condensing the vapor is the keystone to the whole mechanical refrigeration process. The movement of the latent heat involved is the basic means of refrigeration.

When one pound of refrigerant R-22 boils, evaporates, it absorbs 85.9 BTUs at 76 psig. To condense one pound of R-22, 85.9 BTUs must be extracted from the refrigerant vapor.

LATENT HEAT OF SUBLIMATION

A change in state directly from a solid to a vapor without going through the liquid phase can occur with some substances. The most common example is the use of "dry ice" or solid carbon dioxide when used for cooling. The same process can occur with ice below the freezing point. This process is utilized in some freeze-drying processes at extremely low temperatures and deep vacuums. The latent heat of sublimation is equal to the sum of the latent heat of fusion and the latent heat of evaporation.

SATURATION TEMPERATURE

The condition of temperature and pressure at which both liquid and vapor can exist simultaneously is termed **saturation**. A saturated liquid or vapor is one at its boiling point. For water at sea level, the saturation temperature is 212°F. At higher pressures, the saturation temperature increases. With a decrease in pressure, the saturation temperature decreases.

The same condition exists for refrigerants. At the refrigerants boiling point, both liquid and vapor exist simultaneously. For example, refrigerant R-22 has a boiling point of 45°F at a pressure of 76 psig. It's boiling point changes only as it pressure changes.

SUPERHEATED VAPOR

After a liquid has changed to a vapor, any further heat added to the vapor raises its temperature. As long as the pressure to which it is exposed remains constant, the resulting vapor is said to be superheated. Since a temperature rise results, sensible heat has been added to the vapor. The term superheated vapor is used to describe a vapor whose temperature is above it's boiling or saturation point. The air around us is composed of superheated vapor.

Refrigerant 22 at 76 psig has a boiling point of 45°F. At 76 psig, if the refrigerants temperature is above 45°F, it is said to be superheated.

SUBCOOLED LIQUID

Any liquid that has a temperature lower than the saturation temperature corresponding to its saturation pressure is said to be subcooled. Water at any temperature less than its boiling temperature (212°F at sea level) is subcooled.

The boiling point of Refrigerant 22 is 45°F at 76 psig. If the actual temperature of the refrigerant is below 45°F at 76 psig, it is said to be subcooled.

ATMOSPHERIC PRESSURE

The atmosphere surrounding the Earth is composed of gases, primarily oxygen and nitrogen, extending many miles above the surface of the Earth. The weight of that atmosphere pressing down on the Earth creates the atmospheric pressure in which we live. At a given point, the atmospheric pressure is relatively constant except for minor changes due to changing weather conditions. For purposes of standardization and as a basic reference for comparison, the atmospheric pressure at sea level has been universally accepted. It has been established at 14.7 pounds per square inch, (psi). This is equivalent to the pressure exerted by a column of mercury 29.92 inches high.

At altitudes above sea level, the depth of the atmospheric blanket surrounding the Earth is less, therefore the atmospheric pressure is less. At 5,000 feet elevation, the atmospheric pressure is only 12.2 psi., 28.84 inches of mercury.

ABSOLUTE PRESSURE

Absolute pressure, normally expressed in terms of pounds per square inch absolute (psia), is defined as the pressure existing above a perfect vacuum. Therefore in the air around us, absolute pressure and atmospheric pressure are the same.

GAUGE PRESSURE

A pressure gauge is calibrated to read 0 psi regardless of elevation when not connected to a pressure producing source. The absolute pressure of a closed system will always be gauge pressure plus atmospheric pressure. At sea level, atmospheric pressure is 14.7 psi, therefore, at sea level, absolute pressure will be gauge pressure plus 14.7. Pressures below 0 psig are actually negative readings on the gauge, and are usually referred to as inches of mercury vacuum. A refrigeration compound gauge is calibrated in the equivalent of inches of mercury for negative readings. Since 14.7 psi is equivalent to 29.92 inches of mercury, 1 psi is approximately equal to 2 inches of mercury on the gauge dial. In the vacuum range, below 0 psig, 2 inches of mercury vacuum is approximately equal to a -1 psig.

It is important to remember that gauge pressure is only relative to absolute pressure. Table 1-1 shows relationships existing at various elevations assuming that standard atmospheric conditions prevail.

Table 1-1
Pressure Relationships at Varying Altitudes

Altitude (Feet)	PSIG	PSIA	Inches Hg.	Boiling Point of Water
0	0	14.7	29.92	212°F
1000	0	14.2	28.85	210°F
2000	0	13.7	27.82	208°F
3000	0	13.2	26.81	206°F
4000	0	12.7	25.84	205°F
5000	0	12.2	24.89	203°F

Table 1-1 shows that even though the gauge pressure remains at 0 psig regardless of altitude, the absolute pressure does change. The absolute pressure in inches of mercury indicates the inches of mercury vacuum that a perfect vacuum pump would be able to reach at the stated elevation. At 5,000 feet elevation under standard atmospheric conditions, a perfect vacuum would be 24.89 inches of mercury. This compares to 29.92 inches of mercury at sea level.

At very low pressures, it is necessary to use a smaller unit of measurement since even inches of mercury is too large for accurate reading. The micron, a metric unit of length, is used for this purpose. When we speak of microns in evacuation, we are referring to absolute pressure in units of microns of mercury.

A micron is equal to 1/1000 of a millimeter and there are 25.4 millimeters per inch. One micron, therefore, equals 1/25,400 inch. Evacuation to 500 microns would be evacuating to an absolute pressure of approximately .02 inch of mercury. At standard conditions this is the equivalent of a vacuum reading of 29.90 inches mercury.

PRESSURE-TEMPERATURE RELATIONSHIPS, LIQUIDS

The temperature at which a liquid boils is dependent on the pressure being exerted on it. The vapor pressure of the liquid is the pressure being exerted by the tiny molecules seeking to escape the liquid and become vapor. Vapor pressure increases with an increase in temperature until at the point when the vapor pressure equals the external pressure, boiling occurs.

Water at sea level boils at 212°F, but at 5,000 feet elevation it boils at 203°F due to the decreased atmospheric pressure. (See Table 1-1) If some means, a compressor for example, is used to vary the pressure on the surface of the water in a closed container, the boiling point can be changed at will. At 100 psig, the boiling point is 337.9°F, and at 1 psig, the boiling point is 215.3°F.

Since all liquids react in the same fashion, although at different temperatures and pressure, pressure provides a means of regulating a refrigerant's temperature. The evaporator is a part of a closed system. A pressure can be maintained in the coil equivalent to the saturation temperature (boiling point) of the liquid at the cooling temperature desired. The liquid will boil at that temperature as long as it is absorbing heat and the pressure does not change.

In a system using refrigerant R-22, if the pressure within the evaporator coil is maintained at 76 psig, the refrigerants boiling point will be 45°F (7.2°C). As long as the temperature surrounding the coil is higher than 45°F (7.2°C), the refrigerant will continue to boil absorbing heat.

PRESSURE-TEMPERATURE RELATIONSHIPS, GASES

One of the basic fundamentals of thermodynamics is the "perfect gas law." This describes the relationship of the three basic factors controlling the behavior of a gas: (1) pressure, (2) volume, and (3) temperature. For all

practical purposes, air and highly superheated refrigerant vapors may be considered perfect gases, and their behavior follows this relationship:

$$\frac{\text{Pressure One x Volume One}}{\text{Temperature One}} = \frac{\text{Pressure Two x Volume Two}}{\text{Temperature Two}}$$

This is most commonly stated, $\dfrac{P_1 V_1}{T_1} = \dfrac{P_2 V_2}{T_2}$.

Although the "perfect gas" relationship is not exact, it provides a basis for approximating the effect on a gas with a change in one of the three factors. In this relationship, both pressure and temperature must be expressed in absolute values, pressure in psia, and temperature in degrees Rankine or degrees Fahrenheit above absolute zero (°F plus 460°). Although not used in practical refrigeration work, the perfect gas relation is valuable for scientific calculations and is helpful in understanding the performance of a refrigerant vapor.

One of the problems of refrigeration is disposing of the heat that has been absorbed during the cooling process. A practical solution is achieved by raising the pressure of the vapor so that its saturation or condensing temperature will be sufficiently above the temperature of the available cooling medium (air or water) to assure efficient heat transfer. This will provide the ability of the cooling medium to absorb heat from the refrigerant and cool it below its boiling point (dew point). When the low pressure vapor with its low saturation temperature is drawn into the cylinder of a compressor, the volume of the gas is reduced by the stroke of the compressor piston. The vapor is discharged as a high pressure high temperature vapor and is readily condensed because of its high saturation temperature.

If refrigerant R-22's pressure is raised to 195 psig, its saturation temperature will be 100°F (37.8°C). If the cooling medium's temperature is lower than 100°F, heat will be extracted from the R-22 and it will be condensed, converted back to a liquid.

SPECIFIC VOLUME

Specific volume of a substance is defined as the number of cubic feet occupied by one pound (ft³/lb). In the case of liquids and gases, it varies with the temperature and the pressure to which the fluid is subjected. Following the perfect gas law, the volume of a gas varies with both temperature and pressure. The volume of a liquid varies with temperature. Within the limits of practical refrigeration practice, it is regarded as non-compressible. Specific volume is the reciprocal of density (lb/ft³).

DENSITY

The density of a substance is defined as weight per unit volume. In the United States, density is normally expressed in pounds per cubic foot (lb./ft³). Since by definition, density is directly related to specific volume, the density of a vapor may vary greatly with changes in pressure and temperature, although it still remains a vapor, invisible to the naked eye. Water vapor or steam at 50 psia pressure and 281°F temperature is over 3 times as heavy as steam at 14.7 psia pressure and 212°F.

Refrigerant 22 vapor at 76 psig and at 45°F has a density of 1.66 lb/ft³. At 150 psig and at 83°F, the refrigerants density is 3.02 lb/ft³ or 1.82 times as heavy.

PRESSURE AND FLUID HEAD

It is frequently necessary to know the pressure created by a column of liquid, or possibly the pressure required to force a column of refrigerant to flow a given vertical distance upwards.

Densities are usually available in terms of pounds per cubic foot, and it is convenient to visualize pressure in terms of a cube of liquid one foot high, one foot wide, and one foot deep. Since the base of this cube is 144 square inches, the average pressure in pounds per square inch is the weight of the liquid per cubic foot divided by 144. For example, water weighs approximately 62.4 pounds per cubic foot, the pressure exerted by 1 foot of water is 62.4 ÷ 144 or .433 pounds per square inch. Ten feet of water will exert a pressure of 10 X .433 or 4.33 pounds per square inch. The same relation of height to pressure holds true, no matter what the area of a vertical liquid column. The pressure exerted by other liquids can be calculated in exactly the same manner if the density is known.

The density of liquid refrigerant R-22 at 45°F, 76 psig is 78.8 lb./ft³. The pressure exerted by one foot of liquid R-22 is 78.8 ÷ 144 or .55 psig. A column of liquid R-22 10 feet high would then exert a pressure of 5.5 psig. At 100°F liquid temperature, the density is 71.2 lb./ft³. A one foot column then exerts a pressure of .49 psig. A ten foot column exerts a pressure of 4.9 psig.

Comparing other refrigerants at 45°F, R-404A has a density of 70.1 lb./ft³. It then exerts a pressure of .49 psig per foot of lift. R-134a has a density of 79.3 lb./ft³, therefore it exerts a pressure of .55 psig per foot of lift.

Fluid head is a general term used to designate any kind of pressure exerted by a fluid that can be expressed in terms of the height of a column of the given fluid. Hence a pressure of 1 psi may be expressed as being equivalent to a head of 2.31 feet of water. (1 psi ÷ .433 psi/ft. of water). In air flow through ducts, very small pressures are encountered, and these are commonly

expressed in inches of water. 1 inch of water = .433 ÷ 12 = .036 psi.

Table 1-2
Pressure Equivalents in Fluid Head

Pounds Per Square Inch Absolute	Inches Mercury	Inches Water	Feet Water
.036	.07	1.0	.083
.433	.90	12.0	1.0
.491	1.0	13.6	1.13
1.0	2.03	27.7	2.31
14.7	29.92	408.0	34.0

FLUID FLOW

For a fluid to flow from one point to another, there must be a difference in pressure between the two points. With no pressure difference, no flow will occur. Fluids may be either liquids or vapors, and the flow of each is important in refrigeration.

Fluid flow through pipes or tubing is governed by the pressure exerted on the fluid, the effect of gravity due to the vertical rise or fall of the pipe, restrictions in the pipe resisting flow, and the resistance of the fluid itself to flow. For example, as a faucet is opened, the flow increases even though the pressure in the water main is constant and the outlet of the faucet has no restriction. Obviously the restriction of the valve is affecting the rate of flow. Water flows more freely than molasses due to a property of fluids called viscosity which describes the fluid's resistance to flow. In oils, the viscosity can be affected by temperatures, and as the temperature decreases the viscosity increases.

As fluid flows through tubing, the contact of the fluid and the walls of the tube create friction, therefore resistance to flow. Valves, fittings, sharp bends in the tubing and other obstructions also create resistance to flow. The basic design of the piping system and its installation will determine the pressure required to obtain a given flow rate.

In a closed system containing tubing through which a fluid is flowing, the pressure difference between two given points will be determined by the velocity, viscosity, and the density of fluid flowing. If the flow is increased, the pressure difference will increase since more friction will be created by the increased velocity of the fluid. This pressure difference is termed pressure loss or pressure drop.

Since control of evaporating and condensing pressures is critical in mechanical refrigeration work, pressure drop through connecting lines can greatly affect the performance of the system. Large pressure drops must be avoided. When designing and installing refrigeration and air conditioning system piping, pressure drop and refrigerant velocity must be given serious consideration. Section 18 in this series of manuals discusses piping and proper sizing and installation.

EFFECT OF FLUID FLOW ON HEAT TRANSFER

Heat transfer from a fluid through a tube wall or through metal fins is greatly affected by the action of the fluid in contact with the metal surface. As a rule, the greater the velocity of flow and the more turbulent the flow, the greater will be the rate of heat transfer. Rapid boiling of an evaporating liquid will also increase the rate of heat transfer. Quiet liquid flow (laminar flow) on the other hand, tends to allow an insulating film to form on the metal surface that resists heat flow, and reduces the rate of heat transfer.

Large quantities of heat can be absorbed by a substance through an increase in sensible heat involving either a large temperature difference between the cooling media and the product being cooled or a large quantity (weight) of the cooling media. When the cooling media is involved in a change of state, latent heat, a smaller amount of the cooling media is needed to absorb an equivalent large quantity of heat. (Refer to Section 1.)

In mechanical refrigeration, a process is required that can transfer large quantities of heat economically and efficiently on a continuous basis. The processes of evaporation and condensing of a liquid are the logical steps in the refrigeration process.

Many liquids could be used for absorbing heat through the evaporation process. Water is ideal in many respects. Unfortunately it boils at temperatures too high for ordinary cooling purposes. It freezes at temperatures too high for low temperature conditions. A refrigerant must satisfy three main requirements:

1. It must readily absorb heat and change state to a vapor at the temperature required by the load.

2. It must readily reject heat and be returned to a liquid at a temperature required by the external cooling media, water or air.

3. For economy and continuous cooling, the system must use the same refrigerant over and over again.

There is no perfect refrigerant for all applications. There are varying opinions as to which refrigerant is best for a specific application.

TERMINOLOGY AND EXAMPLES

The following definitions primarily deal with the way the described materials behave as a working fluid in a thermodynamic system. There may be more specific or technically complete definitions which deal with the chemistry, transport properties, or other aspects of these materials' composition or behavior which are unimportant in the present context.

PURE FLUID

A pure fluid is a single component fluid which does not change composition when boiling or condensing. A pure fluid is made up of one type of molecule. Examples: R-11, R-12, R-22, R-134a.

MIXTURE AND BLEND

Technically, there is no difference in the terms mixture and blend. They include any fluids which are composed of more than one component (i.e., more than one type of molecule). Azeotropic Refrigerant Mixtures (ARMs), Near-ARMs, and zeotropes (each is discussed below) are subsets of the larger group consisting of blends and mixtures.

The following definitions apply to dual component mixtures, but three ("ternary") or more component mixtures have similar but more complicated characteristics. From a thermodynamic working fluid point of view, the number of components in the fluid has little or no effect.

AZEOTROPIC REFRIGERANT MIXTURE

An azeotropic refrigerant mixture (ARM) is a multi-component which at the azeotropic point does not change composition when it evaporates or condenses since both components have exactly the same boiling temperature at that composition and pressure. It is made up of two or more types of molecules. In actuality, an ARM only exhibits such behavior at one temperature and pressure. Deviations from this behavior at other pressures are very slight and essentially undetectable.

ARMs are fairly complex mixtures whose properties depend upon molecular interactions which may result from polarity differences. They can be either minimum or maximum boiling point ARM's. Even more complicated behavior can occur with ARMs. However, factors such as these are relatively unimportant when considering their performance in a system. The most important factor is that they essentially behave as a pure substance when changing phase.

Examples of how minimum and maximum boiling point ARMs behave at their azeotropic and zeotropic composition ratios at constant pressure are shown in Figures 2-1 and 2-2. These figures show the "Dew Line" (the temperature at which droplets appears as superheated vapor is cooled) and "Bubble Line" (the temperature at which bubbles first appear as subcooled liquid is heated) for mixtures at various fluid temperature and concentrations for one pressure value. At concentrations values away from the azeotropic value, the components ("A" and "B") have different boiling temperature, and the liquid and vapor phases change percentage composition as the mixture evaporates or condenses.

AZEOTROPIC MIXTURES
(MINIMUM BOILING POINT AZEOTROPE)

Figure 2-1

AZEOTROPIC MIXTURES
(MAXIMUM BOILING POINT AZEOTROPE)
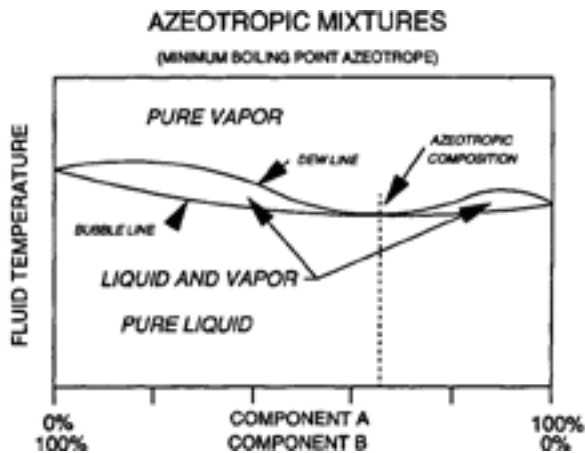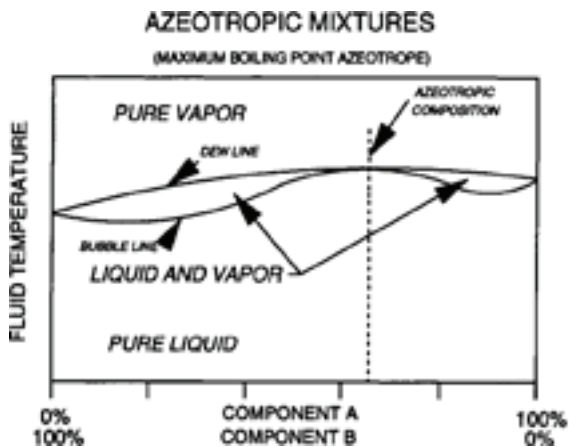
Figure 2-2

Examples:

R-502, an azeotrope of 48.8% R-22 and 51.2% R-115 at +66°F, has lower discharge temperatures than does R-22 for high compression ratio applications.

R-500, an azeotrope of 73.8% R-12 and 26.2% R-152a at +32°F, has approximately 15% more capacity than pure R-12 and was used to compensate for the capacity reduction arising from using a 60 Hz. R-12 system on 50 Hz.

R-507, a non-ozone depleting azeotrope at -40°F is 50% R-125 and 50% R-143a. It is an HFC replacement for R-502.

ZEOTROPIC MIXTURE

A zeotrope is a working fluid with two or more components of different vapor pressure and boiling points whose liquid and vapor components have different com-

position when the fluid evaporates or condenses. It is made of two or more types of molecules. Under constant pressure, the evaporating and condensing temperatures change with composition. (See Figure 2-3.) This change in temperature during constant pressure phase change is called glide, and varies with the components used and their proportions. (See Temperature Glide.) The amount of glide exhibited by a particular zeotrope is a measure of its deviation from being an azeotrope. By definition, azeotropes have zero glide at their azeotropic point. At other conditions, however, they can exhibit glide.

ZEOTROPIC MIXTURES
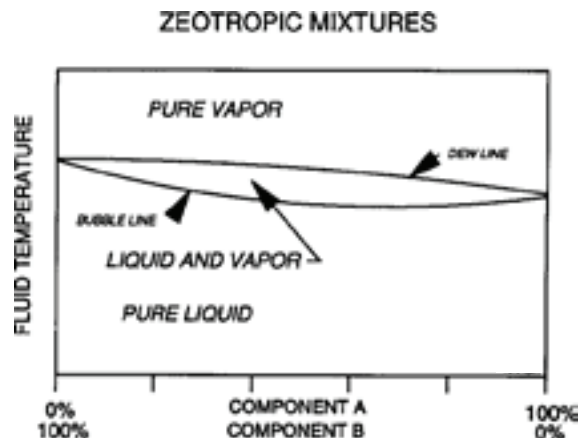
Figure 2-3

Examples:

R-401A is a mixture of R-22, R-152a, and R-124 which closely approximates the vapor pressure and performance of R-12. (R-401A is considered to be a Near Azeotropic mixture.)

NEAR-AZEOTROPIC REFRIGERANT MIXTURE

A Near-ARM is a zeotropic fluid whose composition is such that it exhibits a "small" amount of glide. Thus, "near-azeotropic" is a relative term. (See Figure 2-4.) Some researchers use a maximum glide temperature value of 10°F to distinguish Near-ARMs from zeotropes.

R-404A is a ternary mixture which closely approximates the vapor pressure and performance characteristics of R-502. R-402A is a mixture of R-22, R-125, and R-290 (propane) which closely approximates the vapor pressure and performance characteristics of R-502.

CHARACTERISTICS OF MIXTURES

HOW ARE COMPONENTS CHOSEN?

Components are primarily chosen based on the final characteristics desired in the mixture. These character-

NEAR AZEOTROPIC MIXTURES
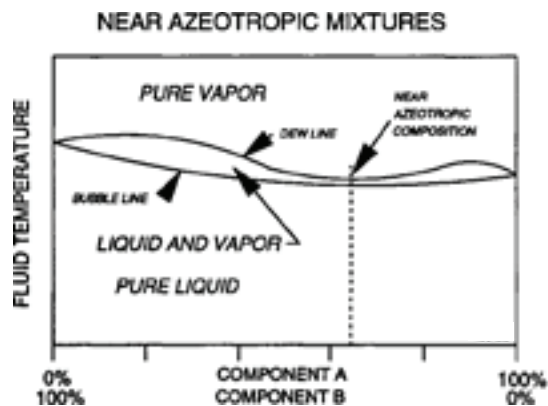
Figure 2-4

istics could include vapor pressure, transport properties, lubricant and materials compatibility, thermodynamic performance, cost, flammability, toxicity, stability, and environmental properties. Availability of components to a particular chemical manufacturer may also be a factor in component selection.

Proportions of components are chosen based on the exact characteristics desired in the final product. It is possible to modify the percentage composition and alter such parameters as capacity, efficiency, discharge temperature, vapor pressure, etc. Of course, changing one parameter will likely change others as well. There must also be a need to balance proportions to guarantee that a given mixture cannot become flammable, toxic, or environmentally undesirable under any foreseeable circumstance, such as leakage.

In many cases, there are computer programs which can use the properties of the individual components and calculate the resulting mixture properties and performance with a fairly high degree of accuracy.

MIXTURE BEHAVIOR

When an azeotropic or zeotropic mixture is entirely in the vapor state (i.e., no liquid is present in the container) the composition is totally mixed and all properties are uniform throughout.

When that same mixture is entirely in the liquid state (i.e., no vapor present in the container), like the 100% vapor state, the composition is totally mixed and all properties are uniform throughout.

In a partially full sealed container of a refrigerant mixture, the composition of the vapor and liquid phases can be different. The degree of difference depends upon whether the mixture is an azeotrope, zeotrope, or near azeotrope.

AZEOTROPE

The percentage of an ARM will be virtually the same in the saturated region where both liquid and vapor are in contact with each other, except at the Azeotropic composition point (See Figures 2-1 and 2-2). At this point its liquid and vapor components will have the same boiling point. At this condition each component has the same boiling point and each vaporizes in proportion to the amount present in the liquid phase. The resulting vapor is the same composition as the liquid. The same is true for the reverse (condensing) process. At other conditions, however, the percentage composition of liquid and vapor phases will be slightly different.

ZEOTROPE

The percentage composition of a zeotropic mixture may be substantially different in the saturated mode when liquid and vapor are in contact with each other. This is because there is no unique boiling point for each component, and they will not vaporize at the rates proportional to their composition in the liquid state. The higher vapor pressure component (with the lower boiling point) will vaporize faster than the lower vapor pressure component (with the higher boiling point), and result in percentage composition changes in both the liquid and vapor phases as vaporization progresses. The higher vapor pressure component will be in higher composition in the vapor phase above the liquid. This process is called **"fractionation."**

NEAR-AZEOTROPIC REFRIGERANT MIXTURES

The percentage composition of the liquid and vapor phases of a Near-ARM will be nearly identical, due to the very similar vapor pressure values of each component. Thus, a Near-ARM behaves essentially the same as an ARM from this standpoint.

WHAT HAPPENS TO MIXTURE COMPOSITION DURING SYSTEM CHARGING?

Depending on how system charging is performed (i.e. with vapor or liquid being removed from the cylinder), the refrigerant may change phase in the cylinder. Since pure fluids and ARMs (except as discussed in a previous section) do not change composition with changes in phase, there is no change in composition with these materials during system charging with vapor or liquid. On the other hand vapor charging with a zeotropic mixture can result in significant composition changes due to fractionation of the components as discussed earlier.

If an entire cylinder of refrigerant is used to charge a system, then the composition change process has no effect since the entire contents of the cylinder will go

into the system. However, if only part of a cylinder of a zeotropic refrigerant is vapor charged into a system, the vapor composition can change substantially during the process. As a result, only liquid charging (i.e., what leaves the cylinder) should be used for zeotropes unless the entire cylinder is to be used for one system. Of course, proper protection against liquid ingestion by the compressor must be provided. This could be in the form of an accumulator-type device which allows the liquid to boil and enter the compressor as a vapor or meters small amounts of liquid into the suction side of the system. Another choice is the "Dial-a-Charge" type of charging system, which takes a measured amount of liquid from the cylinder and puts all of it into the system being sure that it goes into the suction side of the compressor as a vapor.

Since Near-ARMs are actually zeotropes, they also result in composition changes during charging, but to a much smaller extent than occurs with zeotropes. When charging Near-ARM refrigerants, liquid (from the cylinder) should be used to avoid composition changes (unless the entire contents is going into the system). The last few percent of the contents of a cylinder should not be used as this is when composition changes can be the greatest.

Guidelines for charging procedures and how much of cylinder's refrigerant to use during charging will be provided by the refrigerant manufacturers.

Azeotropes, as described above behave as a pure material during boiling and condensing, and do not appreciably change percentage composition.

Zeotropes, on the other hand, do not behave as a pure material during boiling and condensing, and the percentage composition of the liquid and vapor phases can be different. This characteristic can have a significant effect on the composition of the refrigerant left in the system after a leak in the vapor-containing region of a system, and to the subsequent composition after the lost refrigerant had been replaced.

The magnitude of this effect depends strongly upon how much the mixture departs from being an ARM.

Leakage scenarios are discussed later.

TEMPERATURE GLIDE

Figure 2-5 shows how a zeotropic two-component mixture behaves during change of phase at various concentrations for a constant pressure. As subcooled liquid is heated, the higher vapor pressure component eventually reaches its boiling point and begins to form vapor. This condition is called the "Bubble Point," and

is the temperature at which bubbles (flash gas) begin to appear. The vapor is rich with the high vapor pressure component. As the temperature increases, more and more of the high vapor pressure component vaporizes, reducing its component in the liquid phase. At the same time, the lower vapor pressure component eventually reaches its boiling point and begins to vaporize. Finally, the high vapor pressure component is fully evaporated. All that is left is the low vapor pressure component and when the last drop evaporates, the "Dew Point" temperature is established. This is the temperature at which liquid begins to appear when the zeotropic vapor is cooled. The difference between the dew point and the bubble point temperature is known as "**temperature glide**." It varies with percentage composition of the components as well as pressure.

ZEOTROPIC MIXTURES

PURE VAPOR

DEW LINE

BUBBLE LINE

LIQUID AND VAPOR

PURE LIQUID

FLUID TEMPERATURE

0%
100%

COMPONENT A
COMPONENT B

100%
0%
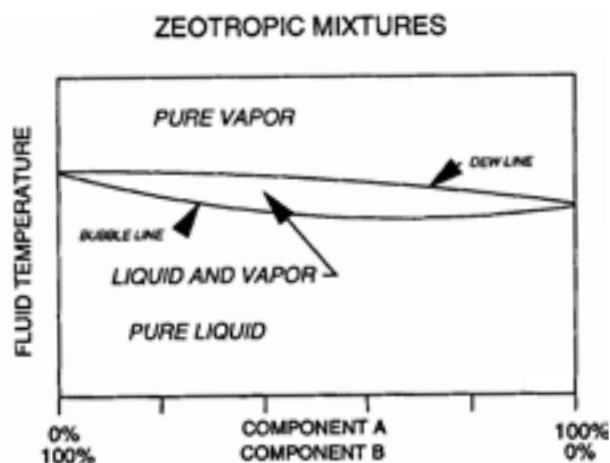
Figure 2-5

The practical effect of glide in heat exchangers is that as the refrigerant mixture flows through the tubing at constant pressure, the evaporating (or boiling) temperature will change as the composition of the liquid and vapor phase change. Thus, a constant evaporating temperature does not occur, even with constant pressure.

The amount of glide varies with the pressure and percentage composition of each component present in the mixture. Glide can vary from an imperceptible amount with a Near-ARM to ten or more degrees F with a zeotrope. Many researchers consider a Near-ARM to have glide less than ten degrees F. The glide of many mixtures is given in Emerson Climate Technologies, Inc. Changeover Guidelines and TIP card/PT chart.

In any heat exchanger, flow of refrigerant through the tubing results in a pressure drop from the entrance to the exit. Consequently, since the pressure at which the phase change is occurring is decreasing along the length of the heat exchanger, the evaporating or con-

densing temperature will decrease as the saturated refrigerant moves through the heat exchanger. Of course, the amount of change in evaporating or condensing temperature depends upon the magnitude of the pressure drop, but it can be several degrees. The change in evaporating and condensing temperature which occurs with today's pure fluids in many systems is similar to that which occurs with Near-ARMs due to glide as they pass through the heat exchanger.
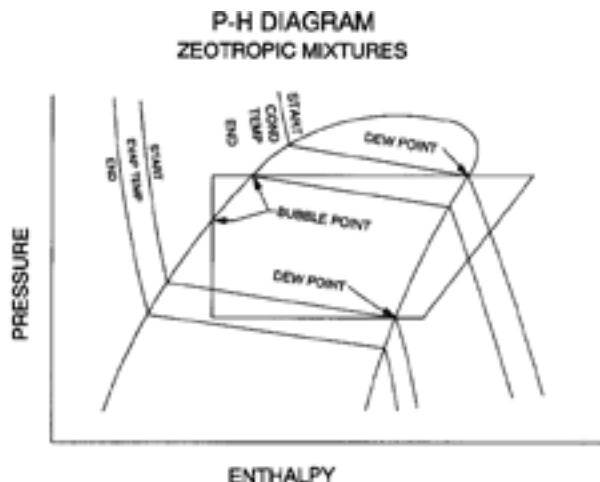
Figure 2-6

A schematic example of how glide with a zeotrope affects temperatures in an evaporator and condenser in a system is shown in Figure 2-6. As a result of the composition change as the refrigerant flows through the heat exchangers, the evaporating and condensing temperatures decrease. Of course, with Near-ARMs the temperature change is very slight and probably undetectable.

As a practical matter, the pressure drop in the evaporator tends to counteract the temperature glide being less than would be expected at constant pressure conditions. The effects are additive in the condenser.

WHAT HAPPENS TO REFRIGERANT COMPOSITION DURING A LEAK?

In a single component refrigerant there is no change in percentage composition of the refrigerant. In an azeotropic mixture there is virtually no change in percentage composition of the refrigerant.

If a leak occurs in a zeotropic mixture in a portion of an operating system where only vapor is present (such as at the compressor discharge or suction line), the system's refrigerant composition will not change since the percentage composition of the vapor is identical to the mixture and each component will leak at the same rate. If a leak occurs in a portion of an operating system where only liquid (such as in the liquid line) is present, the composition will not change since the percentage composition of the liquid is identical to the mixture and each component will leak at the same rate.

However, if a leak occurs in a portion of an operating system where both liquid and vapor exist simultaneously (such as in the evaporator or condenser), "**fractionation**" (unequal evaporation or condensing of the refrigerant in a change in percentage composition between liquid and vapor phase as discussed previously) will occur and there can be a change in percentage composition of the refrigerant left in the system. For example, if a leak occurs in the two-phase potion of the evaporator and only vapor leaks out, the vapor will be richer in the higher vapor pressure component, resulting in a change in the percentage composition of the remaining refrigerant in the system. If the system is recharged with the original composition, the mixture in the system can never get back to the original composition, and system performance (such as capacity or efficiency) may change to some degree. Repeated leak and recharge cycles will result in additional change. However, in most operating systems where two phases are present, turbulent mixing occurs and liquid will leak along with the vapor which minimizes the effect the vapor leakage effect

It is important to keep in mind that to have a change in composition in an operating system, the leak must occur in a portion of the system where both liquid and vapor phases exist simultaneously, and only vapor leaks out.

While a system is off, there will be parts of the system where pure vapor exists and parts where pure liquid exists, and these locations can change with varying environmental conditions. Since the composition of the liquid and vapor phases will be different for a zeotrope in liquid-vapor equilibrium (as discussed previously), a leak in the area where vapor alone is present can result in a composition change in the system. Such a leak could be significant during the long wintertime off cycle of an air conditioning system, or during long periods of non-use for any system.

The effect of such leaks with **Near-Azeotropes** is much less due to the fact that the percentage composition difference between the liquid and vapor phases is very small. Theoretical leakage effect calculations (verified by initial laboratory testing) with Near-ARMs are undetectable, even with several repeated leak and recharge cycles.

TYPES OF REFRIGERANT

There are many different types of refrigerants avail-

able. Several have been in common use for a number of years. In the United States, because of the Clean Air Act of 1990 amended, the common refrigerants are changing. The Montreal Protocol also effects the common usage refrigerants in many parts of the world.

In early refrigeration applications, ammonia, sulfur dioxide, methyl chloride, propane, and ethane were widely used. Some of these are still used today. Because these products are either toxic, dangerous, or have other undesirable characteristics, they have been replaced by compounds developed especially for refrigeration use. Specialized refrigerants are used for ultra-low temperature work, and in large centrifugal compressors. For normal commercial refrigeration and air conditioning applications utilizing reciprocating compressors, refrigerants R-12, R-22, and R-502 have been used almost exclusively. These were developed originally by DuPont as Freon® refrigerants. The ASHRAE numerical designations are now standard with all manufacturers of refrigerants

These refrigerants use chlorine as one of the elements in their composition. The scientific community has determined that chlorine reaching the upper atmosphere causes a reduction in the upper atmospheric Ozone (O_3). R-12 and R-502 are classified as **C**hloro**F**luoro**C**arbons (CFCs) and are being phased out in favor of non Ozone depleting refrigerants. R-22 is classified as a **H**ydro**C**hloro**F**luoro**C**arbon (HCFC). the hydrogen molecule allows the chlorine to break down lower in the atmosphere reducing its Ozone Depletion Potential.

The replacement refrigerants are referred to as **H**ydro**F**luoro**C**arbons (HFCs). These refrigerants do not contain chlorine. R-134a is the HFC replacement for medium temperature R-12 applications. It is not recommended for use in application at saturated suction temperatures below -10°F. R-404A and R-507 are the HFC replacement for R-502 for use in medium & low temperature applications. R-407C and R-410A are the HFC replacements for R-22 for use in high temperature air conditioning applications. R-410A is a high pressure refrigerant and is not a retrofit refrigerant for R-22. These HFC refrigerants are similar to the CFC and HCFC refrigerants, however, they are not identical with respect to pressures, temperatures and enthalpy.

Service Refrigerants

The Clean Air Act of 1990 Amended prohibits the manufacture or import of CFC refrigerants into the United States after December 31, 1995. In order to maintain those systems already in operation using these refrigerants, the chemical companies have developed service replacement refrigerant blends. These blends are an HCFC. They use R-22 as the base refrigerant and blend other refrigerants with it to achieve a desirable property.

It is desirable to have a service replacement that looks very much like the CFC refrigerant in the system. However, they cannot be mixed with the CFC refrigerant. CFC refrigerant in the system must be properly recovered before the replacement HCFC refrigerant is put into the system.

R-401A/B and R-409A are service replacements for R-12. R-402A and R-408A are service replacements for R-502.

NOTE: Not all service replacement refrigerants are approved for use in Copeland® brand compressors.

Refrigerants Solubility of Water

Table 2-1 lists several comparative properties of refrigerants including the Solubility of Water in different refrigerants at two different temperatures. We would consider the 100°F point to be liquid refrigerant entering the TEV

Table 2-1
Comparative Properties of Several Refrigerants

	R-12	R-401A	R-401B	R-409A	R-134a	R-22	R-407C	R-410A	R-502	R-402A	R-408A	R-404A	R-507
Saturation Pressure, psig at 70°F	70.2	85.8*	91.9*	106.1*	71.2	121.4	114.9*	200.6*	137.6	160.4*	135.1*	147.5*	153.6
Boiling Point, °F, at 14.7 psia (Standard Atmospheric Pressure)	-21.6	-27.3**	-30.41**	-29.6**	-14.9	-41.4	-46.4**	-69.9**	-49.8	-56.5**	-46.7**	-51.6**	-52.1
Liquid Density, lb./ft³ at 70°F	82.7	75.4	75.2	77.06	76.2	75.5	71.8	67.6	78.6	72.2	80.9	66.5	66.5
Solubility of Water, PPM at 100°F	165	NA	NA	1600	1900	1800	NA	2850	740	NA	900	970	970
Solubility of Water, PPM at -40°F	1.7	NA	NA	190	NA	120	NA	90	12	NA	100	100	100

NA - Not Available   * Dew Point Pressure   ** Bubble Point Temperatures

Table 2-2
Comparative Refrigeration Effect

	R-12	R-401A	R-401B	R-409A	R-134a	R-22	R-407C	R-410A	R-502	R-402A	R-408A	R-404A	R-507
Evaporating Pressure, psig	0.6	2.9	*0.8	*1.9	*3.7	10.2	6.2	26.3	15.5	18.1	13.4	16.3	17.8
Condensing Pressure psig	136	166.1	176.3	140.6	146.5	226	256.6	365.4	246	289.3	252.1	270.3	280.6
Compression Ratio	9.9	10.3	13.4	11.3	12.5	9.7	13	9.3	8.6	9.3	9.5	9.2	9.1
Specific Volume of Return Gas, ft³/lb	3.03	3.7	4.2	4.1	5.7	2.53	1.6	1.5	1.66	1.4	2.1	1.79	1.69
Refrigeration Effect BTU/lb	53.7	71.7	72.1	68.6	69.5	73.03	70.7	77.3	48.72	51	62	51.8	53.7

* In/Hg

(Data shown at -20°F evaporating temperature, 110°F condensing temperature, 0°F liquid subcooling, 65°F return gas temperature.)
Not all of the refrigerants are recommended at this conditions.

and the -40°F point to be saturated refrigerant in the evaporator. The concern of the technician should be how much water can the refrigerant hold before it becomes free water and causes problems. Ideally there should not be any moisture in the refrigerant in the system. Proper installation and service techniques should assure that the system is clean and dry. This includes the proper use of a vacuum pump and micron gage.

The table lists the solubility in PPM (parts per million by weight). PPM may be a meaningless number to the average technician and installer so let's equate it to something more understandable. Filter dryers are rated in drops of water before they become saturated and can no longer hold any additional water. ARI (The Air Conditioning and Refrigeration Institute) standard is that 20 drops of water equals 1 cc or 1 gram by weight.

Table 2-1 lists R-22 at 100°F having a solubility of 1800 PPM. This simply means that R-22 liquid at 100°F can hold up to 11 drops of water per pound before there is free water. In an Air Conditioning system, when the refrigerants temperature is lowered to +40°F, the solubility drops to 690 PPM (Not shown in Table 2-1.). This equates to 6 drops of water before there is free water. In an Air Conditioning system, the free water will not freeze but may cause other chemical reaction damage.

When the refrigerant goes through the TEV and its temperature is lowered to -40°F, the solubility drops to 120 PPM. This now equates to one drop of water per pound of refrigerant. Once there is more than one drop of water per pound at -40°F, there is free water in the system and the TEV will freeze closed.

Comparative Refrigeration Effect

Table 2-2 lists comparative data for different refrigerants.

Each refrigerant has different suction and discharge pressures for the same operating conditions. This should be expected in that each refrigerant is made up of different chemicals. It is interesting however that the compression ratio for the refrigerants is not that dissimilar. The highest is R-401B, 13.4:1, and the lowest is R-502, 8.6:1.

The Specific Volume (ft³/lb.) of the return gas varies significantly, with the medium pressure refrigerants having the largest Specific Volume, the lowest Density. This equates to a fewer number of pounds of refrigerant being circulated through the compressor per revolution of the compressor motor. The Refrigeration Effect is the pounds of refrigerant in circulation times the refrigerants Enthalpy.

Table 2-3
Refrigerant/Lubricant Chart

Conventional Refrigerants	Service Blends (HCFC)	Non-Ozone Depleting (HFC)
	Refrigerants	
CFC R-12	R-401A R-401B R-409A	R-134a
CFC R-502	R-402A R-048A	R-404A R-507A
HCFC R-22		R-407C R-410A
	Lubricants	
MO	AB POE AB/MO* POE/MO*	POE

*AB or POE must be at least 50% of the system lubricant.

Again note that each refrigerant has its own Refrigeration Effect. The Density of each refrigerant is different as is the Enthalpy of each refrigerant different. Note that the service blends have a higher Refrigeration Effect than their CFC counter parts. It is for this reason that many systems using the service blends do not need as much refrigerant in circulation as they did with the CFCs that were removed.

Each chemical company can provide the information for the refrigerants they manufacture. The form of the information may be in print per the examples shown at the end of this section in Figures 2-4, 2-5 and 2-6 or may be on a computer disc.

Refrigerants and Lubricants

Table 2-3 is a cross reference of the various refrigerants as one goes from the Conventional Refrigerants, to the Service Blends, to the Non Ozone Depleting Refrigerants. It also lists the types of lubricants recommended for use with each category. A description of each refrigerant and its application follows.

Because the mineral oil (MO) used for so many years in air conditioning and refrigeration systems has miscibility issues with the service blends and HFC refrigerants, the proper lubricant approved by Emerson Climate Technologies, Inc. must be used with each type of refrigerant. (Refer to Application Engineering Bulletin 17-1248, Refrigerant Oils.) Later in this section the refrigerant-oil relationship is discussed.

REFRIGERANTS

REFRIGERANT 12

Refrigerant 12 is a pure fluid and is categorized as a ChloroFluoroCarbon (CFC). It has been widely used in household and commercial refrigeration and air conditioning. At temperatures below its boiling point it is a clear, almost colorless liquid. It is not toxic or irritating, and is suitable for high, medium, and low temperature applications. Refrigerant 12 has been determined to cause the depletion of the upper atmosphere Ozone (O_3) layer when it reaches the upper atmosphere. It has been assigned an Ozone Depletion Potential (ODP) of 1. All other refrigerants ODP is measured against R-12. R-12 primary ingredients are chlorine, fluorine, and carbon. The chlorine has been determined to cause the ozone depletion. New R-12 cannot be produced or brought into the United States after December 31, 1995. The result should cause the use of this refrigerant to diminish quickly after that date.

REFRIGERANT R-401A/B

Refrigerants R-401A and R-401B are zeotropic HCFC blends. They are the service replacements for R-12 These refrigerants are a blend of R-22, R-124 and R-152a. The difference between the two is the percentage of each refrigerant in the blend. (R-401A, 53% R-22, 13% R-152a, 34% R-124) (R-401B, 61% R-22, 11% R-152a, 28% R-124) R-401A is the high and medium temperature service replacement for R-12 and R-401B is the low temperature service replacement.

Because these refrigerants are not as miscible with mineral oil in the vapor state, Emerson Climate Technologies, Inc. recommends an approved Alkyl Benzene (AB) lubricant be used with R-401A/B. The AB lubricant must be at least 50% of the lubricant in the system.

The Ozone Depletion Factor of R-401 A/B is 0.030 and 0.035 respectively. The glide is 9.5°F and 8.8°F respectively. The Enthalpy of these blends is approximately 25% greater than R-12, and as such, the system charge may be as much as 15% less than the R-12 charge. The systems pressure will be higher than the R-12 pressures and the discharge temperature will be lower. Even though the system may require less refrigerant than the original refrigerant charge, systems using a TEV must be recharged to insure a full column of liquid at the TEV.

REFRIGERANT R-409A

Like refrigerants R-401A/B, refrigerant R-409A is a zeotropic HCFC blend. It is considered to be a medium/low temperature Service Replacement refrigerant for R-12. It is a blend of R-22, R-124 and R-142b. The glide for R-409A is 14°F. Because it is less miscible in mineral oil, an approved AB lubricant must be used. The AB lubricant must be at least 50% or more of the lubricant in the system. The pressures and temperatures in the system will be different than when using R-12.

REFRIGERANT 134A

Refrigerant R-134a is a pure fluid and is categorized as an HydroFluoroCarbon (HFC). It is the medium and high temperature replacement for R-12. Like R-12, at temperatures below its boiling point, it is a clear colorless liquid. Its basic chemical components are Hydrogen, Fluorine, and Carbon. With the chlorine element removed, its Ozone depletion factor is 0. Unlike R-12, R-134a is not recommended for use in systems where the saturated suction temperature is below -10°F. The saturated suction pressure of R-134a compared to R-12 is similar. The discharge pressure will be higher therefore the compressors compression ratio will be greater. Compressor displacements for R-134a will be similar to

those for R-12. Unlike R-12, it is not miscible in mineral oil. R-134a requires the lubricant in the compressor/system be an approved Polyol Ester (POE). R-134a is a single element compound. (Refer to Emerson Climate Technologies, Inc. Application Engineering Bulletins for the listing of approved lubricants.)

REFRIGERANT 22

Refrigerant 22 in most of its physical characteristics is similar to R-12. However, it has much higher saturation pressures than R-12 for equivalent temperatures. It has a much greater latent heat of evaporation, and a lower specific volume. For a given volume of saturated refrigerant vapor, R-22 has greater refrigerating capacity. This allows the use of smaller compressor displacements, resulting in smaller compressors for performance comparable with R-12. Where size and economy are critical factors, such as package air conditioning units, R-22 is widely used. R-22 is categorized as an HCFC. Its components are hydrogen, chlorine, fluorine, and carbon. Because it contains chlorine, it has an Ozone Depletion Factor of .05 and it will ultimately be phased out under the rules of the Clean Air Act of 1990 Amended.

Because of its characteristics at low evaporating temperatures and it's high compression ratios, the temperature of the compressed R-22 vapor becomes so high it can cause damage to the compressor. In the past, Emerson Climate Technologies, Inc. recommended R-22 in single stage systems for high and medium temperature applications only. It can however be used in low temperature single stage systems only when using Emerson Climate Technologies, Inc. patented Demand Cooling® system or some method of de-superheating before the refrigerant is compressed. It can also be used in low temperature and ultra-low applications in multi-stage systems where the vapor temperature can be adequately controlled. R-22 is a single element compound.

REFRIGERANT 502

Refrigerant 502 is an azeotropic mixture of R-22 and R-115. Its azeotropic rating point is at +66°F. In most physical characteristics, R-502 is similar to R-12 and R-22. While its latent heat of evaporation is not as high as either R-12 or R-22, its vapor is much heavier, or to describe it differently, its specific volume is much less. For a given compressor displacement, its refrigerating capacity is comparable to that of R-22, and at low temperatures is a little greater. As with R-22, a compressor with a smaller displacement may be used for performance equivalent to R-12. Because of its excellent low temperature characteristics, R-502 has been well suited for low temperature refrigeration applications. It

has been the refrigerant of choice for all single stage applications where the evaporating temperature is 0°F or below. It has also been very satisfactory for use in two stage systems for ultra low temperature applications. It gained popularity for use in the medium temperature range.

Like refrigerant R 12, R-502 is considered to be a CFC. The R-115 used to make the azeotropic blend is a CFC. It is this component that makes R-502 a CFC. Like R-12, its production and import into the United States is banned after December 31, 1995. Its Ozone Depletion Factor is 0.3.

REFRIGERANT R-402A

Like refrigerants R-401A/B, R-402A is a Zeotropic HCFC blend. It is considered to be a Service Replacement refrigerant for R-502. It is a blend of R-22, R-125, and R-290. Its enthalpy is similar to R-502 therefore the system charge will be the same. The glide for R-402A is 2.8°F. AB lubricant is recommended for use with this refrigerant and must be at least 50% or more of the lubricant in the system. The system pressures will be higher than with R-502, however the discharge temperature will be lower.

REFRIGERANT R-408A

Refrigerant R-408A is a zeotropic Service Replacement HCFC blend alternative for R-502A. It is a blend of R-22, R-125 and R-143A. Its enthalpy is similar to R-502 therefore the systems charge will be the same. The glide for R-408A is 1.0°F. Because it is less miscible in mineral oil, it is recommended that an AB lubricant be used. The AB lubricant must be at least 50% or more of the lubricant in the system. The systems pressures and temperatures will be different.

REFRIGERANT 404A

Refrigerant R-404A is a zeotropic blend of three refrigerants and is one of two refrigerants considered to be the HFC replacement for R-502. The three refrigerants are R-125, R-143a, and R-134a. R-404A has an Ozone Depletion Factor of 0. Like R-502, it is an excellent refrigerant for low and medium temperature applications. It has performance characteristics similar to R-502 except that it is not miscible with mineral oil. Like R-134a, an approved POE lubricants must be used with R-404A. The system must have less than 5% residual mineral oil.

The saturated suction pressures will be similar to R-502. The discharge pressures will however be higher than R-502. The compressors discharge temperature will be lower when using R-404A as compared to R-502. Because R-404A is a zeotrope, the three refrigerants

that make up it composition do not boil at the same temperature at a specific pressure. Its glide is 0.8°F.

REFRIGERANT 507

Refrigerant R-507 is another HFC replacement for R-502. Like R-404A, it has an Ozone Depletion Factor of 0. This refrigerant is an azeotropic blend made up of R-125 and R-143a. At its azeotropic rating point it is a true azeotrope. Like R-404A, R-507 is not miscible in mineral oil and an approved POE lubricant must be used when the system is using R-507.

R-507 saturated suction pressures will be similar to R-502, however its discharge pressures will be higher than R-502. The discharge temperature will be a little lower than R-502.

REFRIGERANT SATURATION TEMPERATURE

At normal room temperatures, the above refrigerants can exist only as a vapor unless they are pressurized. Their boiling points at atmospheric pressure are below 0°F (See Table 2-1). Therefore, refrigerants are always stored and transported in special pressure resistant drums. As long as both liquid and vapor are present in a closed system, and there is no external pressure influence, the refrigerant will evaporate or condense as a function of the surrounding temperature. Evaporation or condensation will continue until the saturation pressure and temperature corresponding to the surrounding temperature is reached. When this occurs, heat transfer will no longer take place. A decrease in the surrounding temperature will allow heat to flow out of the refrigerant. This will cause the refrigerant to condense and lower the pressure. An increase in the surrounding temperature will cause heat to flow into the refrigerant. This will cause the refrigerant to evaporate, and raise the pressure.

Understanding this principle, and by knowing the surrounding temperature, the refrigerants saturation pressure is known. Conversely, knowing the refrigerants saturation pressure, the refrigerants temperature is known.

REFRIGERANT EVAPORATION

Presume the refrigerant is enclosed in a refrigeration system and it's temperature is equalized with the surrounding temperature. If the pressure in the refrigeration system is lowered, the saturation temperature (the boiling point) will be lowered. The temperature of the liquid refrigerant is now above its boiling point. It will immediately start to boil absorbing heat. In the process, the temperature of the remaining liquid will be reduced. Vapor (flash gas) will occur as the change of state takes place. Heat will now flow into the refrigeration

system from outside of the refrigerant system. This occurs because the refrigerants temperature decreased. Boiling will continue until the surrounding temperature is reduced to the saturation temperature of the refrigerant, or until the pressure in the system again rises to the equivalent saturation pressure of the surrounding temperature. If a means, a compressor, is provided to remove the refrigerant vapor so that the system pressure will not increase, and at the same time liquid refrigerant is fed into the system, continuous refrigeration will take place. This is the process that occurs in a refrigeration or air conditioning system evaporator.

REFRIGERANT CONDENSATION

Presume the refrigerant is enclosed in a refrigeration system and its temperature is equalized with the surrounding temperature. If hot refrigerant vapor is pumped into the system, the pressure in the refrigeration system will be increased and its saturation temperature, boiling point, will be raised.

Heat will be transferred from the incoming hot vapor to the refrigerant liquid and the walls of the system. The temperature of the refrigerant vapor will fall to its condensing, saturation temperature, and condensation will begin. Heat from the refrigerants latent heat of condensation flows from the system to the surrounding temperature until the pressure in the system is lowered to the equivalent of the saturation pressure of the surrounding temperature. If a means, the compressor, is provided to maintain a supply of hot, high pressure refrigerant vapor, while at the same time liquid refrigerant is drawn off, continuous condensation will take place. This is the process taking place in a refrigeration and air conditioning system condenser.

REFRIGERANT-OIL RELATIONSHIPS

In reciprocating and scroll compressors, refrigerant and lubricant mix continuously. Refrigerant gases are soluble in the lubricant at most temperatures and pressures. The liquid refrigerant and the lubricant can be completely miscible, existing as a single phase mixture. Separation of the lubricant and liquid refrigerant into separate layers, two phases, can occur. This generally occurs over a specific range of temperature and composition. This separation occurs at low temperatures, during off-cycles. It occurs in the compressor sump and other places such as accumulators, receivers and oil separators. In the two phase state, the denser liquid refrigerant is underneath the less dense lubricant. This separation does not necessarily affect the lubricating ability of the lubricant but it may create problems in properly supplying lubricant to the working parts. Those compressors with oil pumps have their pickup low in the crankcase.

This will not allow the less dense lubricant to be picked up when the mixture is in the two phase state. Accumulators, receivers and oil separators also pick up liquid low in the component. The location and size of the oil pick up hole is critical. It is imperative that the lubricant return to the compressor is not delayed. The industry has had a successful experience with R-12, R-22 and R-502 refrigerants and mineral oils. The experience with POEs and HFCs has not been as extensive but has been very successful.

The new chlorine-free HFC refrigerants are more polar than the current CFC/HCFC refrigerants. The result is that mineral oils are not miscible with the HFC refrigerants. Polyol Ester (POE) lubricants are more polar than the mineral oils. This polarity of the POE and HFCs make the two miscible and as such, POE is the lubricant to be used with HFC refrigerants. POE lubricants are synthetic, formed by mixing a specific organic acid with a specific alcohol and subjecting them to a reaction. The result is a POE base material and water. The water is driven off and an additive package is mixed with the POE to arrive at a unique approved lubricant. Emerson Climate Technologies, Inc. has approved specific POE lubricants after extensive laboratory and field testing.

POE lubricants are hygroscopic and want to re-absorb water. If care is not taken to keep moisture out of the system, a chemical reaction can occur and produce some weak organic acids. The recommended maximum moisture content in POE is 50 Parts Per Million (PPM). There should be concern when the moisture content is in the 50 to 100 PPM range. Should the moisture content rise above 100 PPM, action must be taken to remove the moisture. It is important that the engineer and the service technician understand the need for "clean and dry" hermetically sealed air-conditioning and refrigeration systems. Proper evacuation techniques and the use of approved filter-driers with adequate moisture removal capacity is crucial to avoid system problems.

HCFC Service Blend refrigerants are a blend of HCFC R-22 and other refrigerants. The other refrigerants are of different types and can be an HFC. This blending is done to achieve a service refrigerant that reacts similarly with respect to temperature, pressure and enthalpy as the CFC refrigerant being removed. These blends are not as miscible/soluble in mineral oil as is the CFC.

Alkyl Benzene (AB) lubricant is a synthetic hydrocarbon. Its composition is more polar than mineral oil. This polar property makes the HCFC service blends more soluble/miscible in the AB lubricant. Like POE, specific AB and Alkyl Benzene Mineral Oil (ABMO) blends have been approved by Emerson Climate Technologies, Inc. after extensive laboratory and field testing.

Since oil must pass through the compressor cylinders to provide lubrication, a small amount of lubricant is always in circulation with the refrigerant. Lubricant and refrigerant vapor do not mix readily. The lubricant can be properly circulated through the system only if vapor velocities are high enough to carry the lubricant along. If velocities are not sufficiently high, lubricant will tend to lie on the bottom of refrigeration tubing, decreasing heat transfer and possibly causing a shortage of lubricant in the compressor. As evaporating temperatures are lowered, this problem increases. For these reasons, proper design of piping is essential for satisfactory lubricant return. (See Section 18, AE-104)

One of the basic characteristics of a refrigerant and lubricant mixture in a sealed system is the fact that refrigerant is attracted to the lubricant. The refrigerant will vaporize and migrate through the system to the compressor crankcase even though no pressure difference exists. On reaching the crankcase the refrigerant will condense into the lubricant. This migration will continue until the lubricant is saturated with liquid refrigerant. Further migration will cause the liquid refrigerant to settle beneath the lubricant.

Excess refrigerant in the compressor crankcase can result in violent foaming and boiling action, driving all the lubricant from the crankcase causing lubrication problems. It can also cause slugging of the compressor at start up. Provisions must be made to prevent the accumulation of excess liquid refrigerant in the compressor.

Proper piping and system design for the refrigerants and lubricants is critical for the lubricant return. The new HFC refrigerants are relatively more soluble in POE lubricants than CFC/HCFC refrigerants and mineral oil. It is important that the engineer and the service technician understand that mineral oil can not be used with the HFC refrigerants and only POE lubricants approved by Emerson Climate Technologies, Inc. are to be used in Copeland® brand compressors.

Refer to Emerson Climate Technologies, Inc. Application Engineering Bulletins for a listing of approved lubricants and refrigerants

REFRIGERANT TABLES

To accurately determine the operating performance of a refrigeration system, precise and accurate information is required. This includes various properties of refrigerants at any temperature and pressure to be considered. Refrigerant manufacturers have calculated and compiled this data in the form of tables of thermodynamic properties. These tables are made available to design and application engineers and others who have a need

for this information.

Table 2-4 is an excerpt from an R-134a saturation table. It lists the five major saturation properties of R-134a, both liquid and vapor, at various temperatures. Pressure, volume, and density have been discussed previously. Table 2-5 is an excerpt from an R-22 saturation table.

Enthalpy is a term used in thermodynamics to describe the heat content of a substance. In refrigeration practice, enthalpy is expressed in terms of BTU per pound. An arbitrary base of saturated liquid at -40°F. has been accepted as the standard zero value. In other words, the enthalpy of any refrigerant is zero for liquid at -40°F. Liquid refrigerant at temperatures below -40°F. is considered to have a negative enthalpy. Refrigerant at all temperatures above -40°F. has a positive enthalpy value.

The difference in enthalpy values at different parts of the system are commonly used to determine the performance of a refrigeration unit. When the heat content per pound of the refrigerant entering and leaving a cooling coil is determined, the cooling ability of that coil can be calculated provided the refrigerant flow rate is known.

Entropy can best be described as a mathematical ratio used in thermodynamics. It is used in solving complex refrigeration engineering problems. It is not easily defined or explained. It is seldom used in commercial refrigeration applications and a discussion of it is beyond the scope of this manual. For our purpose, the compression process within the compressor is an Isentropic process.

Figure 2-6 is an excerpt from an R-404A superheat table. Superheat tables list saturation evaporating temperature and pressure in increments of 1 psi, and tabulate changes in specific volume, enthalpy, and entropy for various increases in temperature of the refrigerant vapor or superheat. Since superheat tables are quite lengthy and are available separately in bound volumes, complete superheat tables have not been included in this manual.

SATURATION PROPERTIES

Temperature/Pressure tables are specific to a refrigerant. The temperature and pressure columns of these tables are most usable to a service technician/engineer. These tables are cumbersome for the average service person because they generally are multiple pages per refrigerant. Table 2-7 is an example of the basic pressures and saturated temperatures for refrigerant R-507 taken from the basic tables. This consolidates the data into one single table.

POCKET TEMPERATURE-PRESSURE CHARTS

Small pocket sized folders listing the saturation temperatures and pressures of common refrigerants are readily available from expansion valve and refrigerant manufacturers. Table 2-8 is a typical example of a pocket sized chart for refrigerants approved for use in a Copeland® brand compressor.

A saturation chart for ready reference is an invaluable tool for the refrigeration and air conditioning technician or for anyone checking the performance of a refrigeration or air conditioning system. Suction and discharge pressures can be readily measured by means of gauges. From these pressures, the saturated evaporating and condensing temperatures can be determined. Knowing the saturated temperatures makes it easy for the technician to determine the amount of superheat or sub-cooling.

Table 2-4
R-134a
Saturation Properties – Temperature Table

TEMP. °F	PRESSURE psia	VOLUME lb/ft³		DENSITY lb/ft³		ENTHALPY Btu/lb			ENTROPY Btu/(lb)(°R)		TEMP. °F
		LIQUID V_f	VAPOR V_g	LIQUID $1/V_f$	VAPOR $1/V_g$	LIQUID h_f	LATENT h_{fg}	VAPOR h_g	LIQUID S_f	VAPOR S_g	
90	119.138	0.0136	0.3999	73.54	2.5009	41.6	73.6	115.2	0.0855	0.2194	90
91	121.024	0.0136	0.3935	73.40	2.5416	41.9	73.4	115.3	0.0861	0.2194	91
92	122.930	0.0137	0.3872	73.26	2.5829	42.3	73.1	115.4	0.0868	0.2193	92
93	124.858	0.0137	0.3810	73.12	2.6247	42.6	72.9	115.5	0.0874	0.2193	93
94	126.809	0.0137	0.3749	72.98	2.6672	43.0	72.7	115.7	0.0880	0.2193	94
95	128.782	0.0137	0.3690	72.84	2.7102	43.4	72.4	115.8	0.0886	0.2192	95
96	130.778	0.0138	0.3631	72.70	2.7539	43.7	72.2	115.9	0.0893	0.2192	96
97	132.798	0.0138	0.3574	72.56	2.7981	44.1	71.9	116.0	0.0899	0.2191	97
98	134.840	0.0138	0.3517	72.42	2.8430	44.4	71.7	116.1	0.0905	0.2191	98
99	136.906	0.0138	0.3462	72.27	2.8885	44.8	71.4	116.2	0.0912	0.2190	99
100	138.996	0.0139	0.3408	72.13	2.9347	45.1	71.2	116.3	0.0918	0.2190	100
101	141.109	0.0139	0.3354	71.99	2.9815	45.5	70.9	116.4	0.0924	0.2190	101
102	143.247	0.0139	0.3302	71.84	3.0289	45.8	70.7	116.5	0.0930	0.2189	102
103	145.408	0.0139	0.3250	71.70	3.0771	46.2	70.4	116.6	0.0937	0.2189	103
104	147.594	0.0140	0.3199	71.55	3.1259	46.6	70.2	116.7	0.0943	0.2188	104
105	149.804	0.0140	0.3149	71.40	3.1754	46.9	69.9	116.9	0.0949	0.2188	105
106	152.039	0.0140	0.3100	71.25	3.2256	47.3	69.7	117.0	0.0955	0.2187	106
107	154.298	0.0141	0.3052	71.11	3.2765	47.6	69.4	117.1	0.0962	0.2187	107
108	156.583	0.0141	0.3005	70.96	3.3282	48.0	69.2	117.2	0.0968	0.2186	108
109	158.893	0.0141	0.2958	70.81	3.3806	48.4	68.9	117.3	0.0974	0.2186	109
110	161.227	0.0142	0.2912	70.66	3.4337	48.7	68.6	117.4	0.0981	0.2185	110
111	163.588	0.0142	0.2867	70.51	3.4876	49.1	68.4	117.5	0.0987	0.2185	111
112	165.974	0.0142	0.2823	70.35	3.5423	49.5	68.1	117.6	0.0993	0.2185	112
113	168.393	0.0142	0.2780	70.20	3.5977	49.8	67.8	117.7	0.0999	0.2184	113
114	170.833	0.0143	0.2737	70.05	3.6539	50.2	67.6	117.8	0.1006	0.2184	114
115	173.298	0.0143	0.2695	69.89	3.7110	50.5	67.3	117.9	0.1012	0.2183	115
116	175.790	0.0143	0.2653	69.74	3.7689	50.9	67.0	117.9	0.1018	0.2183	116
117	178.297	0.0144	0.2613	69.58	3.8276	51.3	66.8	118.0	0.1024	0.2182	117
118	180.846	0.0144	0.2573	69.42	3.8872	51.7	66.5	118.1	0.1031	0.2182	118
119	183.421	0.0144	0.2533	69.26	3.9476	52.0	66.2	118.2	0.1037	0.2181	119
120	186.023	0.0145	0.2494	69.10	4.0089	52.4	65.9	118.3	0.1043	0.2181	120
121	188.652	0.0145	0.2456	68.94	4.0712	52.8	65.6	118.4	0.1050	0.2180	121
122	191.308	0.0145	0.2419	68.78	4.1343	53.1	65.4	118.5	0.1056	0.2180	122
123	193.992	0.0146	0.2382	68.62	4.1984	53.5	65.1	118.6	0.1062	0.2179	123
124	196.703	0.0146	0.2346	68.46	4.2634	53.9	64.8	118.7	0.1068	0.2178	124
125	199.443	0.0146	0.2310	68.29	4.3294	54.3	64.5	118.8	0.1075	0.2178	125
126	202.211	0.0147	0.2275	68.13	4.3964	54.6	64.2	118.8	0.1081	0.2177	126
127	205.008	0.0147	0.2240	67.96	4.4644	55.0	63.9	118.9	0.1087	0.2177	127
128	207.834	0.0147	0.2206	67.80	4.5334	55.4	63.6	119.0	0.1094	0.2176	128
129	210.688	0.0148	0.2172	67.63	4.6034	55.8	63.3	119.1	0.1100	0.2176	129
130	213.572	0.0148	0.2139	67.46	4.6745	56.2	63.0	119.2	0.1106	0.2175	130
131	216.485	0.0149	0.2107	67.29	4.7467	56.5	62.7	119.2	0.1113	0.2174	131
132	219.429	0.0149	0.2075	67.12	4.8200	56.9	62.4	119.3	0.1119	0.2174	132
133	222.402	0.0149	0.2043	66.95	4.8945	57.3	62.1	119.4	0.1125	0.2173	133
134	225.405	0.0150	0.2012	66.77	4.9700	57.7	61.8	119.5	0.1132	0.2173	134
135	228.438	0.0150	0.1981	66.60	5.0468	58.1	61.5	119.6	0.1138	0.2172	135
136	231.502	0.0151	0.1951	66.42	5.1248	58.5	61.2	119.6	0.1144	0.2171	136
137	234.597	0.0151	0.1922	66.24	5.2040	58.8	60.8	119.7	0.1151	0.2171	137
138	237.723	0.0151	0.1892	66.06	5.2844	59.2	60.5	119.8	0.1157	0.2170	138
139	240.880	0.0152	0.1864	65.88	5.3661	59.6	60.2	119.8	0.1163	0.2169	139
140	244.068	0.0152	0.1835	65.70	5.4491	60.0	59.9	119.9	0.1170	0.2168	140
141	247.288	0.0153	0.1807	65.52	5.5335	60.4	59.6	120.0	0.1176	0.2168	141
142	250.540	0.0153	0.1780	65.34	5.6192	60.8	59.2	120.0	0.1183	0.2167	142
143	253.824	0.0153	0.1752	65.15	5.7064	61.2	58.9	120.1	0.1189	0.2166	143
144	257.140	0.0154	0.1726	64.96	5.7949	61.6	58.6	120.1	0.1195	0.2165	144
145	260.489	0.0154	0.1699	64.78	5.8849	62.0	58.2	120.2	0.1202	0.2165	145
146	263.871	0.0155	0.1673	64.59	5.9765	62.4	57.9	120.3	0.1208	0.2164	146
147	267.270	0.0155	0.1648	64.39	6.0695	62.8	57.5	120.3	0.1215	0.2163	147
148	270.721	0.0156	0.1622	64.20	6.1642	63.2	57.2	120.4	0.1221	0.2162	148
149	274.204	0.0156	0.1597	64.01	6.2604	63.6	56.8	120.4	0.1228	0.2161	149

Reprinted with permission from E.I. DuPont

Table 2-5
"FREON" 22
Saturation Properties – Temperature Table

TEMP. °F	PRESSURE		VOLUME cu ft/lb		DENSITY lb/cu ft		ENTHALPY Btu/lb			ENTROPY Btu/(lb)(°R)		TEMP. °F
	psia	psig	LIQUID V_f	VAPOR V_g	LIQUID $1/V_f$	VAPOR $1/V_g$	LIQUID h_f	LATENT h_{fg}	VAPOR h_g	LIQUID S_f	VAPOR S_g	
10	47.464	32.768	0.012088	1.1290	82.724	0.88571	13.104	92.338	105.442	0.02932	0.22592	10
11	48.423	33.727	0.012105	1.1077	82.612	0.90275	13.376	92.162	105.538	0.02990	0.22570	11
12	49.396	34.700	0.012121	1.0869	82.501	0.92005	13.648	91.986	105.633	0.03047	0.22548	12
13	50.384	35.688	0.012138	1.0665	82.389	0.93761	13.920	91.808	105.728	0.03104	0.22527	13
14	51.387	36.691	0.012154	1.0466	82.276	0.95544	14.193	91.630	105.823	0.03161	0.22505	14
15	52.405	37.709	0.012171	1.0272	82.164	0.97352	14.466	91.451	105.917	0.03218	0.22484	15
16	53.438	38.742	0.012188	1.0082	82.051	0.99188	14.739	91.272	106.011	0.03275	0.22463	16
17	54.487	39.791	0.012204	0.98961	81.938	1.0105	15.013	91.091	106.105	0.03332	0.22442	17
18	55.551	40.855	0.012221	0.97144	81.825	1.0294	15.288	90.910	106.198	0.03389	0.22421	18
19	56.631	41.935	0.012238	0.95368	81.711	1.0486	15.562	90.728	106.290	0.03446	0.22400	19
20	57.727	43.031	0.012255	0.93631	81.597	1.0680	15.837	90.545	106.383	0.03503	0.22379	20
21	58.839	44.143	0.012273	0.91932	81.483	1.0878	16.113	90.362	106.475	0.03560	0.22358	21
22	59.967	45.271	0.012290	0.90270	81.368	1.1078	16.389	90.178	106.566	0.03617	0.22338	22
23	61.111	46.415	0.012307	0.88645	81.253	1.1281	16.665	89.993	106.657	0.03674	0.22318	23
24	62.272	47.576	0.012325	0.87055	81.138	1.1487	16.942	89.807	106.748	0.03730	0.22297	24
25	63.450	48.754	0.012342	0.85500	81.023	1.1696	17.219	89.620	106.839	0.03787	0.22277	25
26	64.644	49.948	0.012360	0.83978	80.907	1.1908	17.496	89.433	106.928	0.03844	0.22257	26
27	65.855	51.159	0.012378	0.82488	80.791	1.2123	17.774	89.244	107.018	0.03900	0.22237	27
28	67.083	52.387	0.012395	0.81031	80.675	1.2341	18.052	89.055	107.107	0.03958	0.22217	28
29	68.328	53.632	0.012413	0.79604	80.558	1.2562	18.330	88.865	107.196	0.04013	0.22198	29
30	69.591	54.895	0.012431	0.78208	80.441	1.2786	18.609	88.674	107.284	0.04070	0.22178	30
31	70.871	56.175	0.012450	0.76842	80.324	1.3014	18.889	88.483	107.372	0.04126	0.22158	31
32	72.169	57.473	0.012468	0.75503	80.207	1.3244	19.169	88.290	107.459	0.04182	0.22139	32
33	73.485	58.789	0.012486	0.74194	80.089	1.3478	19.449	88.097	107.546	0.04239	0.22119	33
34	74.818	60.122	0.012505	0.72911	79.971	1.3715	19.729	87.903	107.632	0.04295	0.22100	34
35	76.170	61.474	0.012523	0.71655	79.852	1.3956	20.010	87.708	107.719	0.04351	0.22081	35
36	77.540	62.844	0.012542	0.70425	79.733	1.4199	20.292	87.512	107.804	0.04407	0.22062	36
37	78.929	64.233	0.012561	0.69221	79.614	1.4447	20.574	87.316	107.889	0.04464	0.22043	37
38	80.336	65.640	0.012579	0.68041	79.495	1.4697	20.856	87.118	107.974	0.04520	0.22024	38
39	81.761	67.065	0.012598	0.66885	79.375	1.4951	21.138	86.920	108.058	0.04576	0.22005	39
40	83.206	68.510	0.012618	0.65753	79.255	1.5208	21.422	86.720	108.142	0.04632	0.21986	40
41	84.670	69.974	0.012637	0.64643	79.134	1.5469	21.705	86.520	108.225	0.04688	0.21968	41
42	86.153	71.457	0.012656	0.63557	79.013	1.5734	21.989	86.319	108.308	0.04744	0.21949	42
43	87.655	72.959	0.012676	0.62492	78.892	1.6002	22.273	86.117	108.390	0.04800	0.21931	43
44	89.177	74.481	0.012695	0.61448	78.770	1.6274	22.558	85.914	108.472	0.04855	0.21912	44
45	90.719	76.023	0.012715	0.60425	78.648	1.6549	22.843	85.710	108.553	0.04911	0.21894	45
46	92.280	77.584	0.012735	0.59422	78.526	1.6829	23.129	85.506	108.634	0.04967	0.21876	46
47	93.861	79.165	0.012755	0.58440	78.403	1.7112	23.415	85.300	108.715	0.05023	0.21858	47
48	95.463	80.767	0.012775	0.57476	78.280	1.7398	23.701	85.094	108.795	0.05079	0.21839	48
49	97.085	82.389	0.012795	0.56532	78.157	1.7689	23.988	84.886	108.874	0.05134	0.21821	49
50	98.727	84.031	0.012815	0.55606	78.033	1.7984	24.275	84.678	108.953	0.05190	0.21803	50
51	100.39	85.69	0.012836	0.54698	77.909	1.8282	24.563	84.468	109.031	0.05245	0.21785	51
52	102.07	87.38	0.012856	0.53808	77.784	1.8585	24.851	84.258	109.109	0.05301	0.21768	52
53	103.78	89.08	0.012877	0.52934	77.659	1.8891	25.139	84.047	109.186	0.05357	0.21750	53
54	105.50	90.81	0.012898	0.52078	77.534	1.9202	25.429	83.834	109.263	0.05412	0.21732	54
55	107.25	92.56	0.012919	0.51238	77.408	1.9517	25.718	83.621	109.339	0.05468	0.21714	55
56	109.02	94.32	0.012940	0.50414	77.282	1.9836	26.008	83.407	109.415	0.05523	0.21697	56
57	110.81	96.11	0.012961	0.49606	77.155	2.0159	26.298	83.191	109.490	0.05579	0.21679	57
58	112.62	97.93	0.012982	0.48813	77.028	2.0486	26.589	82.975	109.564	0.05634	0.21662	58
59	114.46	99.76	0.013004	0.48035	76.900	2.0818	26.880	82.758	109.638	0.05689	0.21644	59
60	116.31	101.62	0.013025	0.47272	76.773	2.1154	27.172	82.540	109.712	0.05745	0.21627	60
61	118.19	103.49	0.013047	0.46523	76.644	2.1495	27.464	82.320	109.785	0.05800	0.21610	61
62	120.09	105.39	0.013069	0.45788	76.515	2.1840	27.757	82.100	109.857	0.05855	0.21592	62
63	122.01	107.32	0.013091	0.45066	76.386	2.2190	28.050	81.878	109.929	0.05910	0.21575	63
64	123.96	109.26	0.013114	0.44358	76.257	2.2544	28.344	81.656	110.000	0.05966	0.21558	64

Reprinted with permission from E.I. DuPont

Table 2-6
R-404A
SUVA® HP62 Superheated Vapor – Constant Pressure Tables

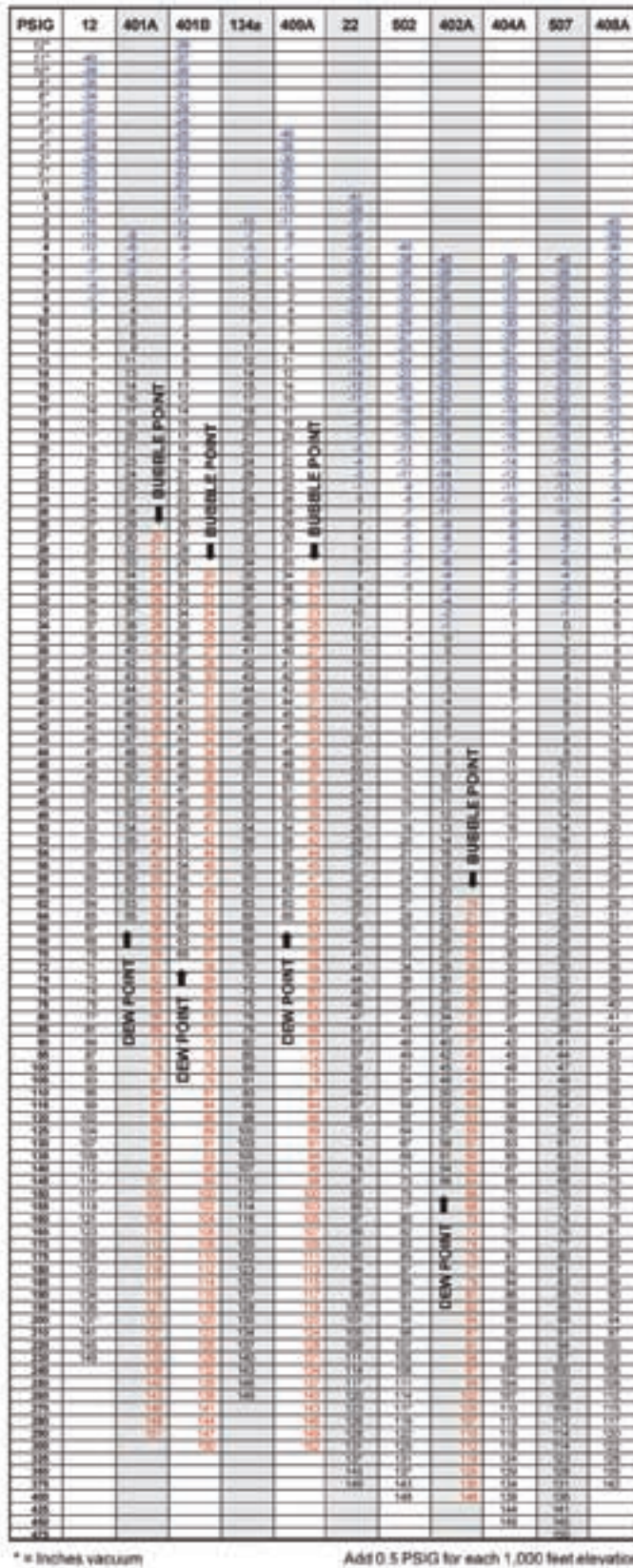V = Volume in ft³/lb H = Enthalpy in Btu/lb S = Entropy in Btu/(lb)(°R) (Saturation properties in parentheses)

ABSOLUTE PRESSURE, psia

TEMP. °F	16.00 (-47.11°F) V	H	S	17.00 (-44.78°F) V	H	S	18.00 (-42.57°F) V	H	S	19.00 (-40.44°F) V	H	S	TEMP. °F
	(2.7271)	(84.2)	(0.2045)	(2.5763)	(84.5)	(0.2042)	(2.4416)	(84.9)	(0.2039)	(2.3206)	(85.2)	(0.2036)	
-40	2.7793	85.5	0.2076	2.6094	85.4	0.2063	2.4584	85.3	0.2050	2.3233	85.3	0.2038	-40
-30	2.8524	87.3	0.2119	2.6786	87.2	0.2106	2.5240	87.2	0.2093	2.3857	87.1	0.2081	-30
-20	2.9253	89.1	0.2161	2.7474	89.1	0.2148	2.5893	89.0	0.2136	2.4478	89.0	0.2124	-20
-10	2.9979	91.0	0.2203	2.8160	90.9	0.2190	2.6543	90.9	0.2178	2.5096	90.9	0.2166	-10
0	3.0703	92.9	0.2245	2.8843	92.8	0.2232	2.7191	92.8	0.2220	2.5712	92.8	0.2208	0
10	3.1425	94.8	0.2285	2.9525	94.8	0.2273	2.7837	94.7	0.2261	2.6326	94.7	0.2250	10
20	3.2145	95.8	0.2327	3.0205	96.7	0.2315	2.8481	96.7	0.2302	2.6938	96.6	0.2291	20
30	3.2864	98.7	0.2368	3.0883	98.7	0.2355	2.9123	98.6	0.2343	2.7548	98.6	0.2331	30
40	3.3581	100.7	0.2408	3.1560	100.7	0.2396	2.9764	100.6	0.2383	2.8157	100.6	0.2372	40
50	3.4296	102.7	0.2448	3.2235	102.7	0.2436	3.0403	102.7	0.2423	2.8764	102.6	0.2412	50
60	3.5010	104.8	0.2488	3.2909	104.8	0.2475	3.1041	104.7	0.2463	2.9370	104.7	0.2452	60
70	3.5723	106.9	0.2528	3.3582	106.8	0.2515	3.1678	106.8	0.2503	2.9975	106.7	0.2491	70
80	3.6435	109.0	0.2567	3.4253	108.9	0.2554	3.2313	108.9	0.2542	3.0578	108.8	0.2530	80
90	3.7146	111.1	0.2606	3.4923	111.0	0.2593	3.2948	111.0	0.2581	3.1180	111.0	0.2569	90
100	3.7855	113.2	0.2644	3.5593	113.2	0.2631	3.3581	113.1	0.2619	3.1782	113.1	0.2608	100
110	3.8564	115.4	0.2682	3.6261	115.3	0.2670	3.4214	115.3	0.2658	3.2382	115.3	0.2646	110
120	3.9272	117.6	0.2721	3.6928	117.5	0.2708	3.4845	117.5	0.2696	3.2981	117.5	0.2684	120
130	3.9978	119.8	0.2758	3.7595	119.7	0.2746	3.5476	119.7	0.2734	3.3580	119.7	0.2722	130
140	4.0685	122.0	0.2796	3.8260	122.0	0.2783	3.6105	121.9	0.2771	3.4177	121.9	0.2760	140
150	4.1390	124.3	0.2833	3.8925	124.2	0.2821	3.6734	124.2	0.2809	3.4774	124.2	0.2797	150
160	4.2094	126.5	0.2870	3.9589	126.5	0.2858	3.7363	126.5	0.2846	3.5370	126.4	0.2834	160
170	4.2798	128.8	0.2907	4.0253	128.8	0.2894	3.7990	128.8	0.2883	3.5966	128.8	0.2871	170
180	4.3502	131.2	0.2944	4.0916	131.1	0.2931	3.8617	131.1	0.2919	3.6561	131.1	0.2908	180
190	4.4204	133.5	0.2980	4.1578	133.5	0.2967	3.9244	133.5	0.2956	3.7155	133.4	0.2944	190
200	4.4906	135.9	0.3016	4.2240	135.8	0.3004	3.9869	135.8	0.2992	3.7748	135.8	0.2980	200
210	4.5608	138.3	0.3052	4.2901	138.2	0.3040	4.0495	138.2	0.3028	3.8342	138.2	0.3016	210
220	4.6309	140.7	0.3088	4.3561	140.6	0.3075	4.1119	140.6	0.3063	3.8934	140.6	0.3052	220
230	4.7009	143.1	0.3123	4.4221	143.1	0.3111	4.1743	143.1	0.3099	3.9526	143.0	0.3088	230
240	4.7709	145.6	0.3159	4.4881	145.5	0.3146	4.2367	145.5	0.3134	4.0118	145.5	0.3123	240
250	4.8409	148.0	0.3194	4.5540	148.0	0.3181	4.2990	148.0	0.3169	4.0709	148.0	0.3158	250
260	4.9108	150.5	0.3229	4.6199	150.5	0.3216	4.3613	150.5	0.3204	4.1299	150.5	0.3193	260

TEMP. °F	20.00 (-38.40°F) V	H	S	21.00 (-36.44°F) V	H	S	22.00 (-34.55°F) V	H	S	23.00 (-32.73°F) V	H	S	TEMP. °F
	(2.2112)	(85.5)	(0.2034)	(2.1119)	(85.8)	(0.2032)	(2.0213)	(86.1)	(0.2029)	(1.9383)	(86.4)	(0.2027)	
-30	2.2612	87.1	0.2070	2.1485	87.0	0.2059	2.0461	87.0	0.2049	1.9525	86.9	0.2039	-30
-20	2.3204	88.9	0.2113	2.2051	88.9	0.2102	2.1003	88.8	0.2092	2.0046	88.8	0.2082	-20
-10	2.3793	90.8	0.2155	2.2615	90.8	0.2145	2.1543	90.7	0.2134	2.0565	90.7	0.2125	-10
0	2.4380	92.7	0.2197	2.3176	92.7	0.2186	2.2081	92.6	0.2176	2.1081	92.6	0.2167	0
10	2.4966	94.6	0.2239	2.3735	94.6	0.2228	2.2616	94.5	0.2218	2.1594	94.5	0.2208	10
20	2.5549	96.6	0.2280	2.4292	96.5	0.2269	2.3150	96.5	0.2259	2.2106	96.5	0.2249	20
30	2.6130	98.6	0.2320	2.4848	98.5	0.2310	2.3681	98.5	0.2300	2.2617	98.4	0.2290	30
40	2.6710	100.6	0.2361	2.5401	100.5	0.2350	2.4212	100.5	0.2340	2.3125	100.4	0.2331	40
50	2.7289	102.6	0.2401	2.5954	102.5	0.2391	2.4740	102.5	0.2381	2.3632	102.5	0.2371	50
60	2.7866	104.6	0.2441	2.6505	104.6	0.2430	2.5267	104.6	0.2420	2.4138	104.5	0.2411	60
70	2.8442	106.7	0.2480	2.7054	106.7	0.2470	2.5793	106.6	0.2460	2.4642	106.6	0.2450	70
80	2.9016	108.8	0.2520	2.7603	108.8	0.2509	2.6318	108.7	0.2499	2.5145	108.7	0.2490	80
90	2.9590	110.9	0.2558	2.8150	110.9	0.2548	2.6842	110.9	0.2538	2.5647	110.8	0.2529	90
100	3.0162	113.1	0.2597	2.8696	113.0	0.2587	2.7364	113.0	0.2577	2.6148	113.0	0.2567	100
110	3.0733	115.2	0.2635	2.9242	115.2	0.2625	2.7886	115.2	0.2615	2.6647	115.1	0.2606	110
120	3.1304	117.4	0.2674	2.9786	117.4	0.2663	2.8406	117.4	0.2653	2.7146	117.3	0.2644	120
130	3.1873	119.6	0.2711	3.0329	119.6	0.2701	2.8926	119.6	0.2691	2.7644	119.5	0.2682	130
140	3.2442	121.9	0.2749	3.0872	121.8	0.2739	2.9445	121.8	0.2729	2.8141	121.8	0.2720	140
150	3.3010	124.1	0.2786	3.1414	124.1	0.2776	2.9963	124.1	0.2766	2.8638	124.0	0.2757	150
160	3.3577	126.4	0.2824	3.1955	126.4	0.2813	3.0480	126.4	0.2804	2.9133	126.3	0.2794	160
170	3.4144	128.7	0.2860	3.2495	128.7	0.2850	3.0997	128.7	0.2840	2.9628	128.6	0.2831	170
180	3.4710	131.1	0.2897	3.3035	131.0	0.2887	3.1513	131.0	0.2877	3.0123	131.0	0.2868	180
190	3.5275	133.4	0.2934	3.3574	133.4	0.2923	3.2028	133.3	0.2914	3.0616	133.3	0.2904	190
200	3.5840	135.8	0.2970	3.4113	135.7	0.2960	3.2543	135.7	0.2950	3.1109	135.7	0.2940	200
210	3.6404	138.2	0.3006	3.4651	138.1	0.2996	3.3057	138.1	0.2986	3.1602	138.1	0.2976	210
220	3.6967	140.6	0.3041	3.5188	140.5	0.3031	3.3571	140.5	0.3022	3.2094	140.5	0.3012	220
230	3.7531	143.0	0.3077	3.5725	143.0	0.3067	3.4084	143.0	0.3057	3.2585	142.9	0.3048	230
240	3.8093	145.5	0.3112	3.6262	145.4	0.3102	3.4596	145.4	0.3092	3.3076	145.4	0.3083	240
250	3.8655	147.9	0.3147	3.6798	147.9	0.3137	3.5109	147.9	0.3128	3.3567	147.9	0.3118	250
260	3.9217	150.4	0.3182	3.7333	150.4	0.3172	3.5620	150.4	0.3163	3.4057	150.4	0.3153	260
270	3.9779	152.9	0.3217	3.7868	152.9	0.3207	3.6132	152.9	0.3197	3.4546	152.9	0.3188	270

Reprinted with permission from E.I. DuPont

R-507
Pressure/Temperature Chart

Vapor Pressure PSIG	Vapor Temperature °F	Vapor Pressure PSIG	Vapor Temperature °F	Vapor Pressure PSIG	Vapor Temperature °F	Vapor Pressure PSIG	Vapor Temperature °F
0	-52.1	45	9	90	40.1	235	97.5
1	-49.7	46	9.9	91	40.7	240	99
2	-47.3	47	10.7	92	41.2	245	100.4
3	-46	48	11.6	93	41.8	250	101.8
4	-43	49	12.3	94	42.4	255	103.2
5	-41	50	13.1	95	42.9	260	104.6
6	-39	51	14	96	43.4	265	105.9
7	-37	52	14.8	97	44	270	107.2
8	-35.3	53	15.6	98	44.5	275	108.6
9	-33.5	54	16.4	99	45.1	280	109.8
10	-31.8	55	17.2	100	45.6	285	111.1
11	-30.2	56	17.9	101	46.1	290	112.4
12	-28.6	57	18.7	102	46.7	295	113.6
13	-27	58	19.4	103	47.2	300	114.8
14	-25.5	59	20.2	104	47.7	305	116
15	-24	60	20.9	105	48.2	310	117.2
16	-22.6	61	21.6	106	48.7	315	118.4
17	-21.2	62	22.3	107	49.2	320	119.6
18	-19.8	63	23.1	108	49.7	325	120.7
19	-18.4	64	23.8	109	50.2	330	121.8
20	-17.1	65	24.5	110	50.7	335	123
21	-15.8	66	25.2	115	53.2	340	124.1
22	-14.6	67	25.8	120	55.6	345	125.2
23	-13.4	68	26.5	125	57.9	350	126.3
24	-12.2	69	27.2	130	60.1	355	127.3
25	-11	70	27.9	135	62.3	360	128.4
26	-9.8	71	28.5	140	64.4	365	129.4
27	-8.7	72	29.2	145	66.5	370	130.5
28	-7.5	73	29.8	150	68.6	375	131.5
29	-6.4	74	30.5	155	70.6	380	132.5
30	-5.4	75	31.5	160	72.5	385	133.5
31	-4.3	76	31.7	165	74.4	390	134.5
32	-3.3	77	32.4	170	76.3	395	135.5
33	-2.2	78	33	175	78.1	400	136.4
34	-1.2	79	33.6	180	79.9		
35	-0.2	80	34.2	185	81.7		
36	0.8	81	34.8	190	83.4		
37	1.7	82	35.4	195	85.1		
38	2.7	83	36	200	86.7		
39	3.6	84	36.6	205	88.3		
40	4.6	85	37.2	210	89.9		
41	5.5	86	37.8	215	91.5		
42	6.4	87	38.4	220	93.1		
43	7.3	88	39	225	94.6		
44	8.2	89	39.5	230	96.1		

Figure 2-7

Typical Pocket Pressure/Temperature Chart
Table 2-8

Section 3
THE REFRIGERATION CYCLE

Continuous refrigeration can be accomplished by several different processes. In the great majority of applications, and almost exclusively in the smaller horsepower range, the vapor compression system is used for the refrigeration process. However, absorption systems are being successfully used in many applications. In larger equipment, centrifugal systems are used which basically is an adaptation of the compression cycle.

Copeland® brand compressors, as their name implies, are designed for use with the compression cycle. This section of this manual will cover only that form of refrigeration.

SIMPLE COMPRESSION REFRIGERATION CYCLE

There are two pressures existing in a compression system, the evaporating or low pressure, and the condensing or high pressure.

The refrigerant acts as a transportation medium to move heat absorbed in the evaporator to the condenser where

it is rejected. The heat rejected may be given off to the ambient air, or in a water cooled system, to the cooling water. A change of state from liquid to vapor and back to liquid allows the refrigerant to absorb and reject large quantities of heat efficiently and repeatedly.

The basic cycle operates as follows:

High pressure liquid refrigerant is fed from the receiver or condenser through the liquid line, and through the filter-drier to the metering device. It is at this point that the high pressure side of the system is separated from the low pressure side. Various types of control devices may be used, but for purposes of this illustration, only the thermostatic expansion valve (TEV) will be considered.

The TEV controls the quantity of liquid refrigerant being fed into the evaporator. The TEV's internal orifice causes the pressure of the refrigerant to the evaporating or low side pressure to be reduced. This reduction of the refrigerant pressure, therefore its boiling point,

Thermostatic Expansion Valve

Evaporator

Filter Dryer

Receiver

Compressor

Condenser
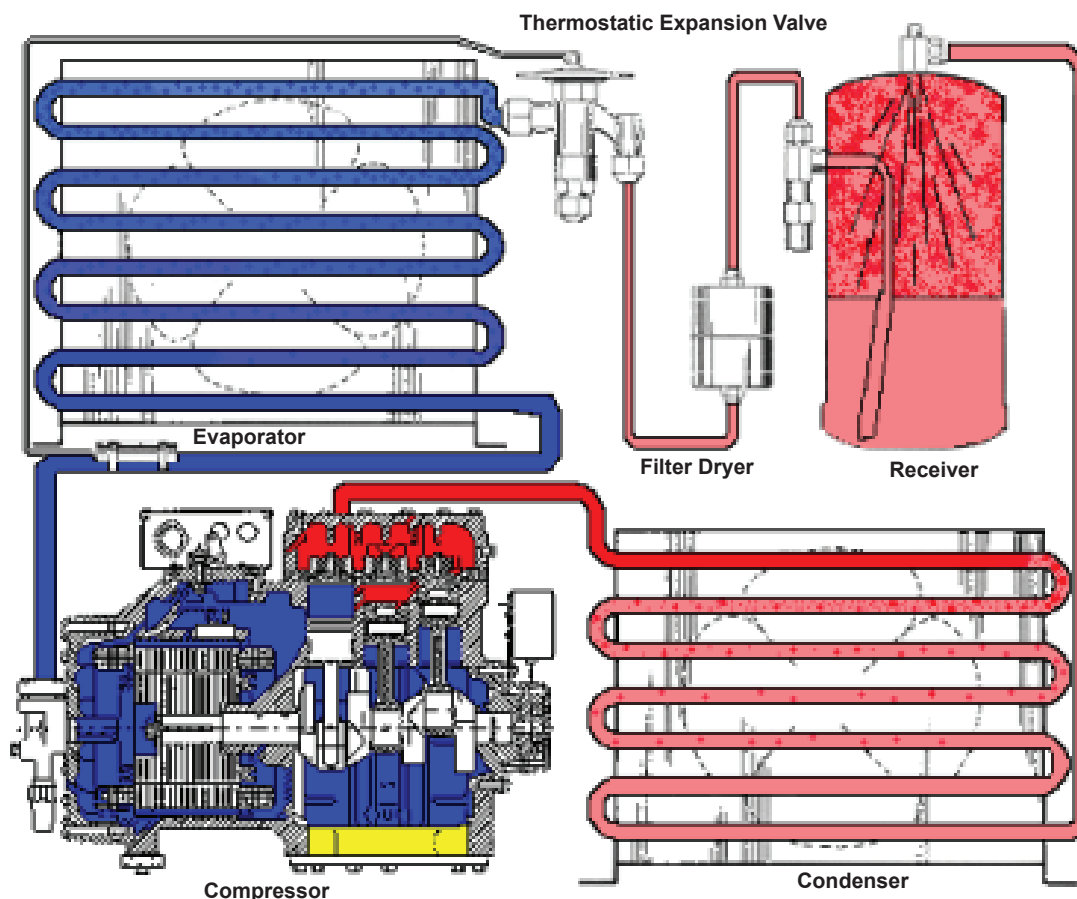
TYPICAL COMPRESSION REFRIGERATION SYSTEM
Figure 3-1

causes it to boil or vaporize, absorbing heat until the refrigerant is at the saturation temperature corresponding to its pressure. As the low temperature refrigerant passes through the evaporator coil, heat flows through the walls of the evaporator tubing into the refrigerant. The boiling action continues until the refrigerant is completely vaporized.

The TEV regulates the quantity of refrigerant, (lb/min) through the evaporator to maintain a preset temperature difference or superheat between the evaporating refrigerant and the vapor leaving the evaporator. As the temperature of the gas leaving the evaporator varies, the expansion valve power element bulb senses this temperature, and acts to modulate the feed of refrigerant through the expansion valve.

The superheated refrigerant vapor leaving the evaporator travels through the suction line to the compressor inlet. The compressor takes the low pressure vapor and compresses it, increasing it's pressure and temperature. The hot, high pressure vapor is forced out of the compressor discharge valve(s), and into the condenser.

As the high pressure high temperature vapor passes through the condenser, it is cooled by an external means. In air cooled systems, a fan, and fin-type condenser surface is normally used. In water cooled systems, a refrigerant-to-water heat exchanger is employed. As the temperature of the refrigerant vapor is lowered to the saturation temperature corresponding to the high pressure in the condenser, the vapor condenses into a liquid and flows back to the receiver or directly to the TEV to repeat the cycle. The refrigerating process is continuous as long as the compressor is operating.

HEAT OF COMPRESSION

Heat of compression is defined as the heat added to the refrigerant vapor as a result of the work energy used in the compression process. When the refrigerant vapor is compressed in a compressors cylinder, its pressure is increased and the volume is decreased. The change in pressure and volume tend to maintain equilibrium in the perfect gas law equation, so this change alone would not greatly affect the temperature of the refrigerant vapor. In order to compress the refrigerant vapor, work or energy is required. Following the first law of thermodynamics, this energy cannot be destroyed, and all of the mechanical energy necessary to compress the vapor is transformed into heat energy. With the exception of a small fraction of the total heat given off to the compressor body, all of this heat energy is transferred to the refrigerant vapor. This causes a sharp increase in the temperature of the compressed gas, therefore, in a reciprocating compressor, the discharge valves are

always subjected to the highest temperature existing in the refrigerating system. In the Copeland Compliant Scroll®, the discharge port or dynamic discharge valve will be subjected to the highest temperature in the system.

The heat which must be discharged by the condenser, termed the heat of rejection, is the total of the heat absorbed by the refrigerant in the evaporator, the heat of compression, and any heat added to the system due to motor inefficiency. Any heat absorbed in the suction and/or discharge lines must also be rejected by the condenser. For hermetic and accessible hermetic motor-compressors, the heat which must be rejected in addition to the refrigeration load can be approximated by the heat equivalent of the electrical power input to the compressor expressed in BTU/hr. (Motor watts X 3.1416 = BTU/hr. of heat to be rejected)

VOLUMETRIC EFFICIENCY OF THE RECIPROCATING COMPRESSOR

Volumetric efficiency is defined as the ratio of the actual volume of refrigerant vapor pumped by the compressor to the volume displaced by the compressor pistons. The volumetric efficiency of a piston compressor will vary over a wide range, depending on the compressor design and the compression ratio.

The compression ratio of a compressor is the ratio of the absolute discharge pressure (psia) to the absolute suction pressure (psia). (Discharge Pressure Absolute ÷ Suction Pressure Absolute)

Several design factors can influence compressor efficiency including the clearance volume above the piston, the clearance between the piston and the cylinder wall, valve spring tension, valve leakage and the volume of the valve plate discharge ports. Reed compressors have from one to three discharge ports per cylinder to allow the compressors refrigerant to exit the cylinder/piston area with a minimum pressure drop. These discharge ports however hold high pressure vapor in them that cannot be sent out to the system. (See Figure 3-2) To improve the volumetric efficiency of low temperature compressors, the number and size of the discharge ports are reduced. (Figure 3-3) Because the volume of refrigerant is less in a low temperature compressor, this can be done with little effect on internal pressure drop but with positive results in increased volumetric efficiency and compressor capacity. Compressor efficiency, because of design, is fairly constant for a given compressor. Volumetric efficiency will vary inversely with the compression ratio.

Two factors cause a loss of volumetric efficiency with an increase in compression ratio. As the vapor is subjected

to greater compression, the residual vapor remaining in the cylinder clearance space and in the valve plate discharge ports becomes more dense. Since it does not leave the cylinder on the discharge stroke, it re-expands on the suction stroke, preventing the intake of a full cylinder of vapor from the suction line. The higher the pressure exerted on the residual vapor, the more dense it becomes, and the greater volume it occupies on re-expansion.

Discus® compressors have less clearance volume and almost no trapped high pressure refrigerant in the discharge ports. The Discus® discharge valve seats at the bottom of the valve plate basically eliminating the trapped high pressure vapor in the valve plate. This reduction in trapped high pressure refrigerant reduces the amount of re-expansion and increases the compressors capacity and efficiency (See Figure 3-4). The
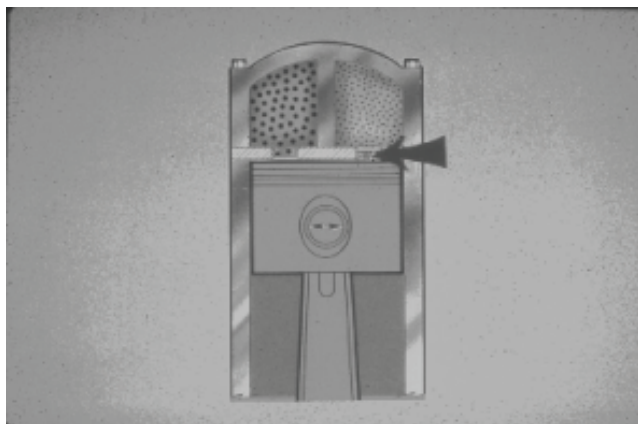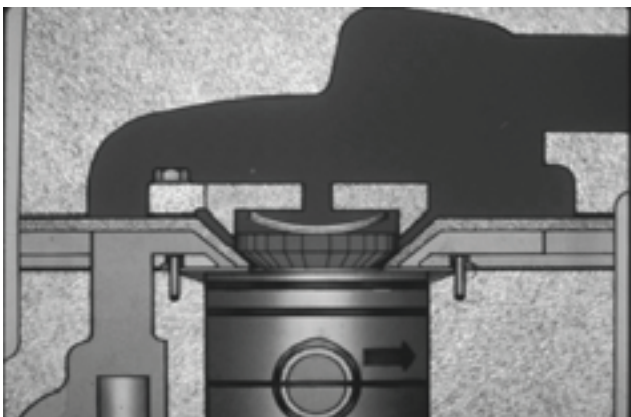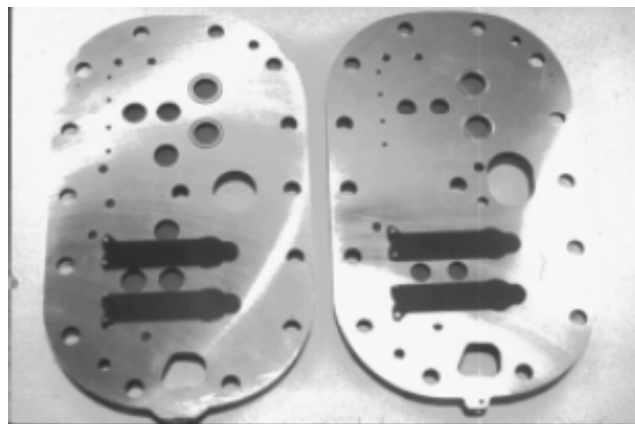
Discus® compressor is more volumetric efficient than the same displacement reed compressor and as such circulates more pounds of refrigerant therefore delivers more BTUs of refrigeration.

The second factor is the high temperature of the cylinder walls resulting from the heat of compression. As the compression ratio increases, the heat of compression increases, and the cylinders and head of the compressor become very hot. Suction vapor entering the cylinder on the intake stroke is heated by the cylinder walls, and expands, resulting in a reduced weight of vapor entering the compressor.

Typical volumetric efficiency curves are shown in Figure 3-5. Air Conditioning and refrigeration compressors are designed with a minimum of clearance volume. As previously stated, clearance volume is a loss in actual capacity versus theoretical capacity. The Discus®

Figure 3-2

High Temperature
Valve Plate

Low Temperature
Valve Plate

Figure 3-3
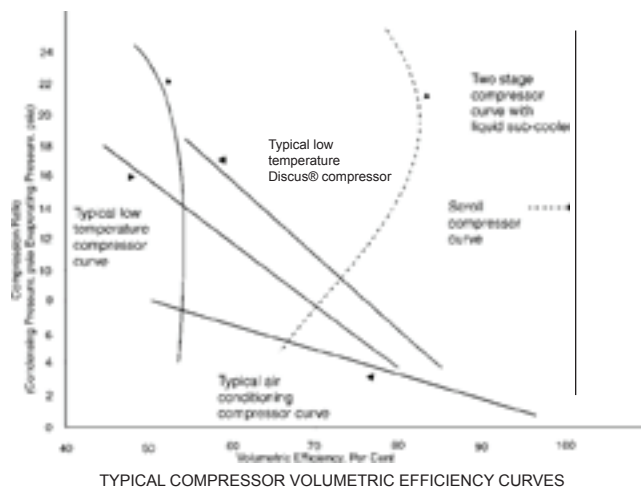
Figure 3-4

TYPICAL COMPRESSOR VOLUMETRIC EFFICIENCY CURVES

Figure 3-5

compressor because of its reduced clearance volume is more volumetric efficient than an equivalent displacement, horse power, reed compressor

While the volumetric efficiency of each stage of a two stage compressor would resemble the typical single stage curves, the overall volumetric efficiency is relatively constant over a wide compression ratio range. Since the use of a liquid subcooler with the two stage compressor can increase the capacity so dramatically. a dotted curve has been added for comparison.

VOLUMETRIC EFFICIENCY OF SCROLL COMPRESSORS

The volumetric efficiency of the Copeland Compliant Scroll® is 100%. When the first pocket of the Scroll closes and captures a volume of refrigerant, all of it will be swept along in the compression process and discharged out of the Scroll members to the system. Unlike the piston compressor, there is no clearance area, clearance volume, to create losses.

EFFECT OF CHANGE IN SUCTION PRESSURE

Other factors remaining equal, as the suction pressure is reduced, the specific volume(ft^3/#) of the vapor returning to the compressor increases. Density (#/ft^3) and specific volume are inversely proportional, therefore the refrigerant density decreases. Since a given compressor's pumping capacity (CFH) is fixed by its speed and displacement, the reduction in density of the suction vapor decreases the weight (#/hr.) of the refrigerant pumped, resulting in a reduction in the compressors capacity (BTU/hr). The loss of capacity with a reduction in suction pressure is extremely rapid. Since the energy input required by the compressor to perform its work does not decrease at the same rate, the BTU/watt ratio decreases rapidly with a drop in suction pressure. This reflects in the performance of the compressors per unit of electrical energy consumed, the Energy Efficiency Ratio (EER).

In addition to the specific volume of the refrigerant being reduced when the suction pressure is reduced, the compression ratio is increased. As stated before, as the compression ratio is increased, the compressors discharge temperature will also be increased. For best capacity performance, operating economy and lowered discharge temperature, it is most important that refrigeration and air-conditioning systems operate at the highest suction pressure possible for the application.

EFFECT OF CHANGE IN DISCHARGE PRESSURE

An increase in the condensing pressure, commonly termed the discharge pressure or head pressure, results in an increase in the compression ratio. This results in a consequent loss of volumetric efficiency except for the Scroll compressor. While the loss of capacity is not as great as that caused by an equivalent decrease in suction pressure, it still is severe.

For operating economy and maximum capacity, the discharge pressure should be kept as low as practical but should not be lower than the equivalent of 70°F saturated discharge pressure.

EFFECT OF SUBCOOLING LIQUID REFRIGERANT WITH WATER OR AIR

When the hot high pressure liquid refrigerant is fed into the evaporator through the TXV, the refrigerants temperature must first be reduced to the evaporating temperature in the evaporator before it can start absorbing heat. This is accomplished by almost instantaneous boiling or "flashing" of a portion of the liquid into vapor. The latent heat of vaporization involved in the change of state absorbs heat from the remaining liquid refrigerant lowering its temperature.

The resulting flash gas will produce little to no further cooling. In effect the refrigerating capacity of the refrigerant has been reduced by the heat absorbed in lowering the liquid temperature. If a portion of this heat could be extracted from the liquid prior to its entry into the evaporator, the effective capacity of the system will be increased. This happens because not as much liquid will flash off to cool the remaining liquid to its desired temperature.

This can be accomplished by subcooling the liquid refrigerant after condensing by means of water or air. If condensing temperatures are relatively high, capacity increases of 5% to 15% are easily obtainable. Since no power is required other than that involved in moving the cooling medium, subcooling the liquid can result in substantial savings in operating cost.

EFFECT OF SUBCOOLING LIQUID REFRIGERANT BY SUPERHEATING THE VAPOR

A suction gas to liquid refrigerant heat exchanger is frequently used for the following reasons:

1. To subcool the liquid refrigerant sufficiently to offset any pressure drop that might occur in the liquid line; to compensate for any heat picked up in the liquid line preventing the formation of flash gas in the liquid line.

2. To provide a source of heat to evaporate any liquid refrigerant which might have flooded through the

evaporator, thus preventing the return of liquid refrigerant to the crankcase.

As pointed out in the previous section, subcooling the liquid refrigerant increases the refrigerating capacity per pound of the refrigerant circulated. In a perfectly insulated system with negligible heat transfer into the suction line outside the refrigerated space, a liquid to suction heat exchanger theoretically will increase system capacity slightly since the heat transferred from the liquid refrigerant to the refrigerant vapor is greater than the capacity reduction at the compressor resulting from the increase in specific volume of the vapor.

EFFECT OF SUPERHEATING THE VAPOR LEAVING THE EVAPORATOR

It is essential that the temperature of the vapor returning to the compressor be superheated to avoid carrying liquid refrigerant back to the compressor. It is generally recommended that the minimum superheat value be 20°F when the system is at low load. If this heat is added to the vapor inside the refrigerated space, the heat absorbed increases the refrigeration capacity, while the increase in specific volume of the gas decreases the compressor capacity. These two factors tend to offset one another, with a negligible effect on capacity.

Heat entering the refrigerant through the suction line from the ambient air outside the refrigerated space results in a net loss of system capacity. These losses may be as high as 10% to 15%. Insulation of the suction line is a worthwhile investment, and may be necessary to prevent the return gas temperature from rising too high. This will also prevent the compressors discharge temperature from rising too high.

EFFECT OF PRESSURE DROP IN THE DISCHARGE LINE AND CONDENSER

Pressure drop due to friction as the refrigerant vapor flows through the discharge line and condenser reduces compressor capacity. This results in higher compressor discharge pressure and lower volumetric efficiency. Since the condensing temperature is not greatly affected, pressure drops of less than 5 psig have very little effect on system capacity.

However, compressor power consumption will increase because of the higher compressor discharge pressure. For best operating economy, excessively high pressure drops in the discharge line should be avoided. A pressure drop in the discharge line between five and ten psig. should be considered normal. Pressure drops over ten psig. should be avoided.

EFFECT OF PRESSURE DROP IN LIQUID LINE

If the pressure of the liquid refrigerant falls below its saturation temperature, a portion of the liquid will flash into vapor. This will cool the liquid refrigerant to a new saturation temperature. This will occur in a liquid line if the pressure drops significantly because of friction or because of vertical lift. If flashing occurs, the feed through the expansion valve will be erratic and inadequate for the evaporator demand.

Subcooling of the liquid refrigerant after condensing by an amount sufficient to offset the pressure drop will insure a solid column of liquid refrigerant at the inlet to the expansion valve. At 120°F saturated condensing temperature, 10°F of liquid subcooling will protect against flash gas forming in the liquid line for pressure drops up to those shown in Table 3-1. These are the maximum allowable that can be tolerated to prevent flashing of the refrigerant in the liquid line.

Table 3-1
Liquid Line Pressure Drop

Refrig.	Press Drop (psig)	Refrig.	Press Drop (psig)	Refrig.	Press Drop (psig)
R-12	21.3	R-22	34.5	R-502	33.9
R-401A	25.9	R-407C	38.5	R-402A	41.1
R-401B	27.2	R-410A	57.7	R-408A	28.5
R-134a	24.8			R-404A	39.4
R-409A	25.4			R-507	41.3

All of the refrigerants listed in Table 3-1 are slightly heavier than water. A head of two feet of liquid refrigerant is approximately equivalent to 1 psi. Therefore if a condenser or receiver in the basement of a building 20 feet tall is to supply liquid refrigerant to an evaporator on the roof, a pressure drop of approximately 10 psi for the vertical head will occur. This must be provided for in system design. (Refer to Section 1 - Pressure & Fluid Head.)

EFFECT OF PRESSURE DROP IN THE EVAPORATOR

Pressure drop occurring in the evaporator due to frictional resistance to flow results in the leaving evaporator pressure being less than the pressure of the refrigerant at the entrance of the evaporator. For a given load and coil, the required average refrigerant temperature is fixed. The greater the pressure drop, the greater the difference between the average evaporator refrigerant pressure and the leaving evaporator refrigerant pressure.

As the suction pressure leaving the evaporator is decreased, the specific volume of the gas returning to the compressor increases, and the weight of the refrigerant pumped by the compressor decreases. Therefore pressure drop in the evaporator causes a decrease in system capacity, and it is important that the evaporator be sized so that abnormally high pressure drops do not occur.

EFFECT OF PRESSURE DROP IN SUCTION LINE

The effect of pressure loss in the suction line is similar to pressure drop in the evaporator. Since pressure drop in the suction line does not result in a corresponding decrease in the refrigerant evaporating temperature in the evaporator. Pressure drop in the suction line can be extremely detrimental to system capacity. Suction lines must be sized to prevent excessive pressure losses.

Table 3-2 shows the capacity loss for a typical 7-1/2 HP compressor as the result of suction line pressure drop. The table lists the losses for both R-12 and R-507 refrigerants at a specific saturated suction temperature. The loss in capacity for an R-12 compressor for a change of 3 psig., 1 psig. to 4 psig., is 20%. The loss in the R-507 compressor for the same additional pressure drop is 12%.

Table 3-2
Suction Line Pressure Drop

R-12			
Evap. Temp.	Suction Line Pressure Drop	Pressure at Comp.	BTU/hr. Capacity
-10°F	1 psi	3.5 psig	27,490
-10°F	2 psi	2.5 psig	25,950
-10°F	3 psi	1.5 psig	24,410
-10°F	4 psi	0.5 psig	22,100

R-507			
Evap. Temp.	Suction Line Pressure Drop	Pressure at Comp.	BTU/hr. Capacity
-10°F	1 psi	24 psig	40,400
-10°F	2 psi	23 psig	39,400
-10°F	3 psi	22 psig	37,400
-10°F	4 psi	21 psig	35,500

INTERNALLY COMPOUND TWO-STAGE SYSTEMS

As the compression ratio increases, the volumetric efficiency of the compressor decreases and the heat of compression increases. For low temperature applica-

tions, the decreasing efficiency and excessively high discharge temperatures become increasingly critical. The lowest recommended evaporating temperature for compressors operating on the simple single stage compression cycle, is -40°F.

In order to increase operating efficiency at low temperatures the compression can be done in two steps or stages. For internally compound two stage operation with equal compression ratios, the compression ratio of each stage will be equal to the square root of the total compression ratio (approximately 1/4 of the total compression ratio for the normal two-stage operating range.) Since each stage of compression then is at a much lower compression ratio, the compressor efficiency is greatly increased. The temperature of the refrigerant vapor leaving the first stage and entering the second stage may be high due to the heat of compression. This can result in overheating the second stage cylinders and valves. To prevent compressor damage, saturated refrigerant must be injected between stages to properly cool the compressor.

A two-stage compressor is designed so that suction gas is drawn directly into the low stage cylinders and then discharged into the high stage cylinder or cylinders. On Copelametic® two-stage compressors the ratio of low stage to high stage displacement is 2 to 1. The greater volume of the low stage cylinders is necessary because of the difference in specific volume of the gas at the low and interstage pressures.

Figures 3-6 and 3-7 illustrate typical two-stage compressors as applied to low temperature systems. Two-stage refrigeration is effective down to evaporator temperatures of -80°F. Below that level, efficiency drops off rapidly.
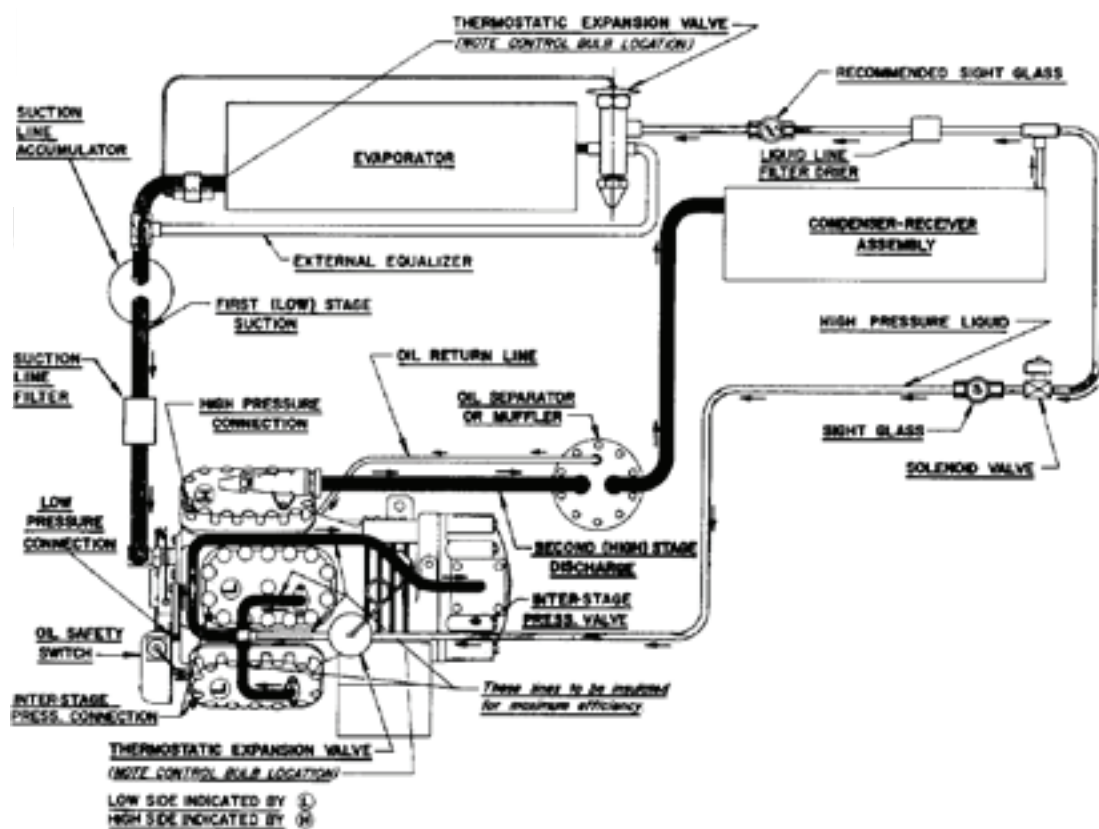
For additional application and service information on internally compound compressors, refer to Application Engineering Bulletin AE 19-1132.

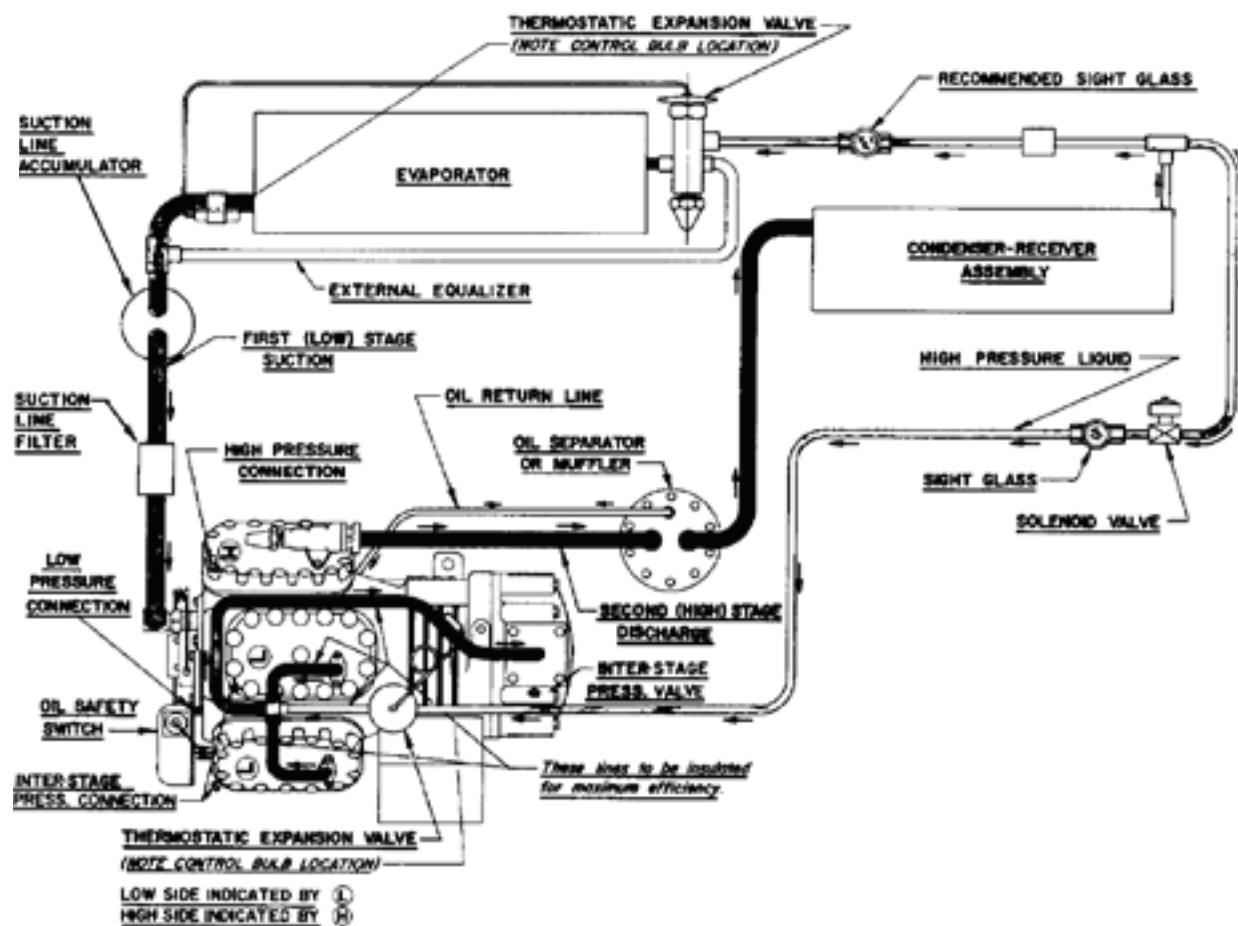### EXTERNALLY COMPOUND SYSTEMS

Two stage compression can be accomplished with the use of two compressors. The discharge of the first compressor becomes the suction of the second compressor. (See Figure 3-8) Like the internally compound compressor, ideally the first compressor will have twice the displacement of the second. However, in the externally compound system, it is not critical.

In the externally compound system, the ideal interstage pressure absolute, can be calculated. It is the square root of the absolute suction pressure times the absolute discharge pressure.
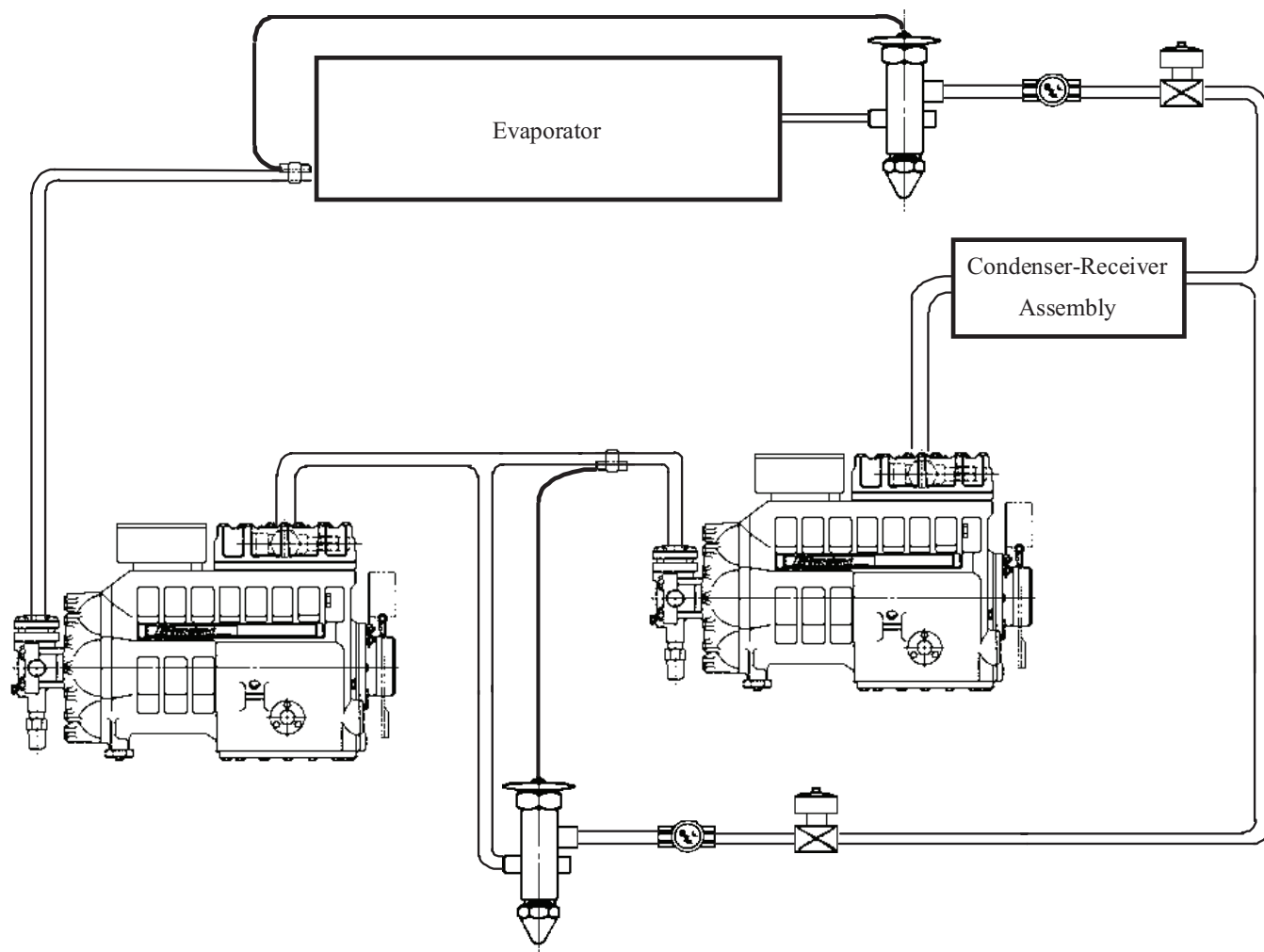
The externally compound system can have compressors in parallel in either or both stages of the system. Compressors can have unloaders. Parallel compressors in both stages can be turned on and off to meet the demands of the low temperature and interstage pressures.
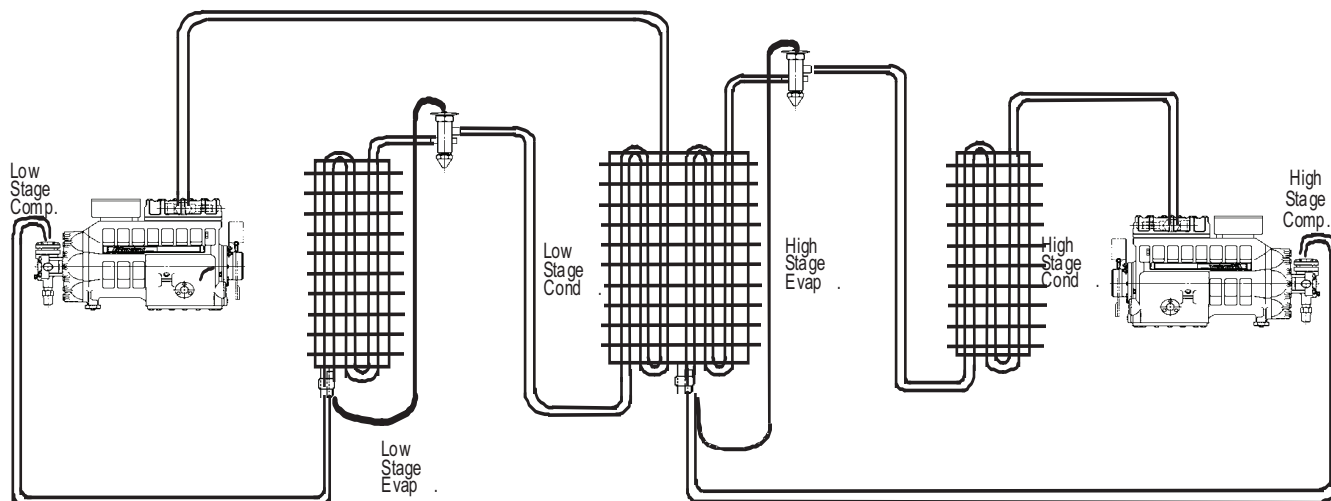

System With 6-Cylinder Compressor (Without Liquid Sub-Cooler)
Figure 3-6

THERMOSTATIC EXPANSION VALVE
(NOTE CONTROL BULB LOCATION)

RECOMMENDED SIGHT GLASS

SUCTION LINE ACCUMULATOR

EVAPORATOR

CONDENSER-RECEIVER ASSEMBLY

EXTERNAL EQUALIZER

FIRST (LOW) STAGE SUCTION

HIGH PRESSURE LIQUID

SUCTION LINE FILTER

HIGH PRESSURE CONNECTION

OIL RETURN LINE

OIL SEPARATOR OR MUFFLER

SIGHT GLASS

SOLENOID VALVE

LOW PRESSURE CONNECTION

SECOND (HIGH) STAGE DISCHARGE

INTER-STAGE PRESS. VALVE

OIL SAFETY SWITCH

These lines to be insulated for maximum efficiency.

INTER-STAGE PRESS. CONNECTION

THERMOSTATIC EXPANSION VALVE
(NOTE CONTROL BULB LOCATION)

LOW SIDE INDICATED BY Ⓛ
HIGH SIDE INDICATED BY Ⓗ

System With 6-Cylinder Compressor (With Sub-Cooler)
Figure 3-7

3-8

Evaporator

Condenser-Receiver
Assembly

Externally Compound System
Figure 3-8

Cascade System
Figure 3-9

CASCADE SYSTEMS

In order to operate satisfactorily at even lower evaporating temperatures, and to increase the flexibility of system design, multiple stage refrigeration can also be accomplished by using separate systems with the evaporator of one serving as the condenser of the second by means of a heat exchanger. (See Figure 3-9) This type of design is termed a cascade system, and allows the use of different refrigerants in the separate systems. Refrigerants with characteristics and pressures suitable for ultra-low temperature refrigeration can be used in the low stage system. Cascade systems in multiples of two, three, or even more separate stages make possible refrigeration at almost any desired evaporating temperature. Cascade systems composed of both single and two-stage compressors can be used very effectively.

Copeland®
brand products

1675 W. Campbell Rd.
Sidney, OH 45365

EmersonClimate.com