Copeland®

REFRIGERATION MANUAL

Part 1

FUNDAMENTALS OF REFRIGERATION

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Section 1

BASIC REFRIGERATION PRINCIPLES

Most users normally associate refrigeration with cold and cooling, yet the practice of refrigeration engineering deals almost entirely with the transfer of heat. This seeming paradox is one of the most fundamental concepts that must be grasped to understand the workings of a refrigeration 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 which are basic in the study of refrigeration.

The first and most important of these laws is the fact that energy can neither be created or destroyed, but can 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.

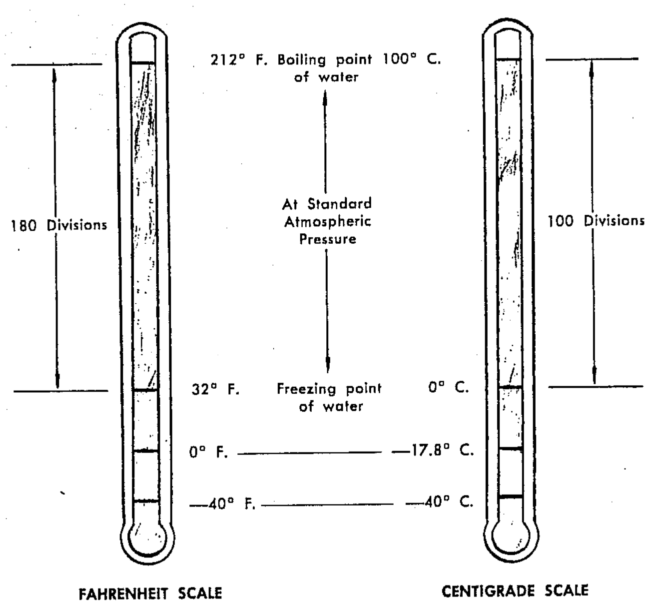
HEAT

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 which creates heat.

Heat is often defined as energy in transfer, for it is never content to stand still, but 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. A spoon in ice water loses its heat to the water and becomes cold; a spoon in hot coffee absorbs heat from the coffee and becomes warm. But 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, which is approximately 460 degrees below zero Fahrenheit. 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, but the Centigrade scale (sometimes termed Celsius) is widely used in other parts of the world. Both scales have two basic points in common, the freezing point of



COMPARISON OF TEMPERATURE SCALES

water, and the boiling point of water at sea level. Water freezes at 32° F. and O° C., and water boils at sea level at 212° F. and 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 Centigrade scale the temperature difference is divided into 100 equal increments or degrees C. The relation between Fahrenheit and Centigrade scales can always be established by the following formulas:

Fahrenheit = 9/5 Centigrade + 32°
Centigrade = 5/9 (Fahrenheit — 32°)

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 necessary to raise 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 BTU's.

$$8.3 \times (80 - 70) = 83$$

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.

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 tremendously greater than the small one, but because of the temperature difference, heat will travel from the small ball to the large one 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. 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. There is little radiation at low temperatures, and at small temperature differences, so 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.

Convection is the flow of heat by means of a fluid medium, either gas 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.

In a typical refrigeration application, heat normally will travel by a combination of processes, and 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, while asbestos has so much resistance to heat flow it can be 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 also can 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 when the vapor changes back to a liquid, and when the liquid is changed to a solid.

The most common example of this process is water, which exists as a liquid, can exist in solid form as ice, and exists as a gas when it becomes steam. As ice it is a usable form of refrigeration, absorbing heat as it melts at a constant temperature of 32° F. If placed on a hot stove in an open pan, its temperature will rise to the boiling point (212° F. at sea level). Regardless of the amount of heat applied, the temperature cannot be raised above 212° F. because the water will completely vaporize into steam. If this steam could be enclosed in a container and more heat applied, then the temperature could again be raised. Obviously 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 evaporating. (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 is 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 BTU's 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, but 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 and .22 respectively.

LATENT HEAT OF FUSION

A change of 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 BTU's at a constant temperature of 32° F., and if one pound of water is to be frozen into ice, 144 BTU's 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, and 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 BTU's at a constant temperature of 212° F. (at sea level) and to condense one pound of steam to water 970 BTU's must be extracted from it.

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 apply 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, and the movement of the latent heat involved is the basic means of refrigeration.

LATENT HEAT OF SUBLIMATION

A change in state directly from a solid to a vapor without going through the liquid phase can occur in some substances. The most common example is the use of "dry ice" or solid carbon dioxide for cooling. The same process can occur with ice below the freezing point, and is also utilized in some freeze-drying processes at extremely low temperatures and high 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, and for water at sea level, the saturation temperature is 212° F. At higher pressures, the saturation temperature increases, and with a decrease in pressure, the saturation temperature decreases.

SUPERHEATED VAPOR

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

SUBCOOLED LIQUID

Any liquid which has a temperature lower than the saturation temperature corresponding to its pressure is said to be subcooled. Water at any temperature less than its boiling temperature (212° F. at sea level) is 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, and this has been established at 14.7 pounds per square inch, which 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 pounds per square inch.

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 pounds per square inch when not connected to a pressure producing source. Therefore the absolute pressure of a closed system will always be gauge pressure plus atmospheric pressure. Pressures below 0 psig are actually negative readings on the gauge, and are referred to as inches of 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.

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

Table 1

PRESSURE RELATIONSHIPS AT VARYING ALTITUDES

Altitude	Psig	Psia	Pressure in Inches Hg	Boiling Point of Water
0 ft.	0	14.7	29.92	212° F.
1000 ft.	0	14.2	28.85	210° F.
2000 ft.	0	13.7	27.82	208° F.
3000 ft.	0	13.2	26.81	206° F.
4000 ft.	0	12.7	25.84	205° F.
5000 ft.	0	12.2	24.89	203° F.

The absolute pressure in inches of mercury indicates the inches of mercury vacuum that a perfect vacuum pump would be able to reach. Therefore, at 5,000 feet elevation under standard atmospheric conditions, a perfect vacuum would be 24.89 inches of mercury, as compared 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 are too large for accurate reading. The micron, a metric unit of length, is used for this purpose, and 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, or at standard conditions, 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, which is the pressure being exerted by the tiny molecules seeking to escape the liquid and become vapor, increases with an increase in temperature until at the point where 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. If some means, a compressor for example, were used to vary the pressure on the surface of the water in a closed container, the boiling point could be changed at will. At 100 psig, the boiling point is 327.8°, F., and at 1 psig, the boiling point is 102° F.

Since all liquids react in the same fashion, although at different temperatures and pressure, pressure provides a means of regulating a refrigerating temperature. If a cooling coil is part of a closed system isolated from the atmosphere and a pressure can be maintained in the coil equivalent to the saturation temperature (boiling point) of the liquid at the cooling temperature desired, then the liquid will boil at that temperature as long as it is absorbing heat — and refrigeration has been accomplished.

PRESSURE—TEMPERATURE RELATIONSHIPS, GASES

One of the basic fundamentals of thermodynamics is called the "perfect gas law." This describes the relationship of the three basic factors controlling the behavior of a gas — pressure, volume, and temperature. For all practical purposes, air and highly superheated refrigerant gases may be considered perfect gases, and their behavior follows the following relation:

$$\frac{\text{Pressure}_1 \times \text{Volume}_1}{\text{Temperature}_1} = \frac{\text{Pressure}_2 \times \text{Voume}_2}{\text{Temperature}_2}$$

Although the "perfect gas" relationship is not exact, it provides a basis for approximating the effect on a gas of a change in one of the three factors. In this relation, 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 gas.

One of the problems of refrigeration is disposing of the heat which has been absorbed during the cooling process, and a practical solution is achieved by raising the pressure of the gas so that the saturation or condensing temperature will be sufficiently above the temperature of the available cooling medium (air or water) to insure efficient heat transfer. When the low pressure gas 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, and the vapor is discharged as a high pressure gas, readily condensed because of its high saturation temperature.

SPECIFIC VOLUME

Specific volume of a substance is defined as the number of cubic feet occupied by one pound, and 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, but within the limits of practical refrigeration practice, it may be regarded as incompressible.

DENSITY

The density of a substance is defined as weight per unit volume, and in the United States is normally expressed in pounds per cubic foot. Since by definition density is directly related to specific volume, the density of a gas may vary greatly with changes in pressure and temperature, although it still remains a gas, 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.

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, since water weighs approximately 62.4 pounds per cubic foot, the pressure exerted by 1 foot of water is $62.4 \div 144 = .433$ pounds per square inch. Ten feet of water would exert a pressure of $10 \times .433 = 4.33$ pounds per square inch. The same relation of height to pressure holds true, no matter what the area of vertical liquid column. The pressure exerted by other liquids can be calculated in exactly the same manner if the density is known.

Fluid head is a general term used to designate any kind of pressure exerted by a fluid which 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. ($1 \text{ psi} \div .433$ 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 $= .433 \div 12 = .036$ psi.

Table 2

PRESSURE EQUIVALENTS IN FLUID HEAD

Pounds per Square Inch	Inches Mercury	Inches Water	Feet Water
.036	.07	1.0	.083
.433	.80	12	1.0
.491	1.0	13.6	1.13
1.0	2.03	27.7	2.31
14.7	29.92	408	34.0

FLUID FLOW

In order for a fluid to flow from one point to another, there must be a difference in pressure between the two points to cause the flow. With no pressure difference, no flow will occur. Fluids may be either liquids or gases, 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 temperature, 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, and therefore resistance to flow. Sharp bends in the tubing, valves and fittings, and other obstructions also create resistance to flow, so the basic design of the piping system 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, and large pressure drops must be avoided.

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 general 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 on the other hand, tends to allow an insulating film to form on the metal surface which resists heat flow, and reduces the rate of heat transfer.

Section 2

REFRIGERANTS

Large quantities of heat can be absorbed by a substance through an increase in sensible heat involving either a big temperature difference or a large weight. In a change of state involving latent heat, however, a fraction of the weight will absorb an equivalent amount of heat.

In mechanical refrigeration a process is required that can transfer large quantities of heat economically and efficiently, and can be repeated continuously. The processes of evaporation and condensation of a liquid are, therefore, the logical steps in the refrigeration process.

Practically any liquid could be used for absorbing heat by evaporation. Water is ideal in many respects, but it boils at temperatures too high for ordinary cooling purposes, and freezes at temperatures too high for low temperature conditions. A refrigerant must satisfy two main requirements:

1. It must readily absorb heat at the temperature required by the product load.

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

There is no perfect refrigerant, and there are varying opinions as to which may be best for specific applications.

TYPES OF REFRIGERANT

There are many different types of refrigerant available, several of which are in common use. In early refrigeration applications, ammonia, sulfur dioxide, methyl chloride, propane, and ethane were widely used, and still are used in many applications. However, due to the fact that they are either toxic, dangerous, or have other undesirable characteristics, they have been largely replaced in most applications by compounds developed especially for refrigeration use. Specialized refrigerants are used for ultra-low temperature work, or for large centrifugal compressors, but for the normal commercial refrigeration and air conditioning applications utilizing reciprocating compressors, refrigerants 12, 22, and 502 are now used almost exclusively. These are frequently referred

Table 3

COMPARATIVE PROPERTIES OF R-12, R-22, and R-502

	R-12	R-22	R-502
Saturation Pressure, 70° F., psig	70.2	121.4	136.6
Boiling Point at 14.7 psia, °F. (Standard Atmospheric Pressure)	—21.6°	—41.4°	—50.1°
Liquid Density at 70° F., lb./cu. ft.	82.7	75.5	78.6
Solubility of Water at 78° F., ppm	93	1,300	560
Solubility of Water at —40° F., ppm	1.7	120	40

Comparative Refrigeration Effect

At operating conditions of
—20° F. evaporating temperature
110° F. condensing temperature
0° F. liquid subcooling
65° F. return gas temperature

	R-12	R-22	R-502
Evaporating pressure, psig	0.6	10.2	15.5
Condensing pressure, psig	136	226	246
Compression Ratio	9.9	9.7	8.6
Specific Volume of return gas, cu.ft./lb.	3.03	2.53	1.66
Refrigeration Effect BTU/lb.	53.7	73.03	48.72
Refrigeration Effect BTU/cu. ft.	17.8	28.9	29.3

to as R-12, R-22, and R-502, and although these were developed originally by Dupont as Freon refrigerants, the numerical designations are now standard with all manufacturers.

REFRIGERANT 12

Refrigerant 12 is 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 almost odorless, is not toxic or irritating, and is suitable for high, medium, and low temperature applications.

REFRIGERANT 22

Refrigerant 22 in most physical characteristics is similar to R-12. However, it has much higher saturation pressures than R-12 for equivalent temperatures, has a much larger latent heat of evaporation, and a lower specific volume, and as a result for a given volume of saturated refrigerant vapor, R-22 has a much greater refrigerating capacity. This allows the use of lower compressor displacement, sometimes 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.

Because of its characteristics at low evaporating temperatures and high compression ratios, the temperature of the compressed R-22 vapor becomes so high it frequently causes damage to the compressor. Copeland recommends R-22 in single stage systems for high and medium temperature applications only, although it is suitable for low temperature applications in multi-stage systems where the vapor temperature can be adequately controlled.

REFRIGERANT 502

Refrigerant 502 is an azeotropic mixture of R-22 and R-115. An azeotrope is the scientific name given to a specific mixture of different compounds in which the resulting mixture has different characteristics than either of its components, and which can evaporate and condense without a change in composition. 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. Therefore for a given compressor displacement, its refrigerating capacity is comparable to that of R-22, and at low temperatures usually will be greater. As with R-22, a compressor with lower compressor displacement may be used for performance equivalent to R-12. Because of its excellent low temperature characteristics, R-502 is well suited for low temperature refrigeration applications, and is recommended for all single stage applications where the evaporating temperature is 0° F. or below. It also is very satisfactory for use in two stage systems for extra low temperature applications, and is becoming popular for use in the medium temperature range.

REFRIGERANT SATURATION TEMPERATURE

At normal room temperatures, all three refrigerants can exist only as a gas unless under pressure, since their boiling points at atmospheric pressure are far below 0° F. Therefore refrigerants are always stored and transported in special pressure resistant drums. So long as both liquid and vapor are present in a closed system, with no external pressure influence, the refrigerant will either evaporate or condense depending on the outside temperature until the saturation pressure and temperature corresponding to the outside temperature is reached and heat transfer can no longer take place. A decrease in outside temperature will allow heat to flow out of the refrigerant, cause condensation, and lower the pressure; and an increase in outside temperature will cause heat to flow into the refrigerant, cause evaporation, and raise the pressure.

REFRIGERANT EVAPORATION

Now, presume the refrigerant is enclosed in a refrigeration system, with its temperature equalized with the outside temperature. Instead of changing the outside temperature, the pressure in the refrigeration system is lowered. Since this lowers the saturation point, the temperature of the liquid refrigerant is now above its boiling point. It will immediately start boiling violently, absorbing heat in the process and thus reducing the temperature of the remaining liquid and changing into gas as the change of state takes place. Heat will now flow into the system from the outside due to the decreased temperature of the refrigerant, and boiling will continue until the outside 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 outside temperature. If a means is provided, such as a compressor, to remove the refrigerant vapor so the pressure does not increase, while at the same time liquid refrigerant is fed back into the system, continuous refrigeration will be taking place. This is basically the process taking place in a refrigeration system evaporator.

REFRIGERANT CONDENSATION

Again, presume the refrigerant is enclosed in a refrigeration system, with its temperature equalized with the outside temperature. If hot refrigerant vapor is pumped into the system the pressure in the refrigeration system is increased, raising the saturation point.

As heat is transferred from the incoming hot vapor to the refrigerant liquid and the walls of the system, the temperature of the refrigerant vapor falls to its condensing temperature, and condensation starts. Heat from the latent heat of condensation flows from the system to the outside until the pressure in the system is lowered to the equivalent of the saturation pressure at the outside temperature. If a means is provided, such as a compressor, to maintain a supply of hot, high pressure refrigerant gas, while at the same time liquid refrigerant is drawn off, continuous condensation will take place. This is basically the process taking place in a refrigeration system condenser.

REFRIGERANT-OIL RELATIONSHIPS

In reciprocating compressors, oil and refrigerant mix continuously. Refrigeration oils are soluble in liquid refrigerant, and at normal room temperatures they will mix completely. The ability of a liquid refrigerant to mix with oil is termed miscibility, and the refrigerant is described as being miscible with oil.

Oil circulating in a refrigeration system may be exposed to both very high and very low temperatures. Because of the critical nature of lubrication under these conditions, and the damage that can be done to the system by wax or other impurities in the oil, only highly refined oils specifically prepared for refrigeration usage can be used.

In general naphthenic oils are more soluble in refrigerants than paraffinic oils. Separation of the oil and refrigerant into separate layers can take place with both types of oil, although at somewhat lower temperatures with naphthenic oils. Separation does not necessarily affect the lubricating ability of the oil but it may create problems in properly supplying oil to the working parts.

Since oil must pass through the compressor cylinders to provide lubrication, a small amount of oil is always circulating with the refrigerant. Oil and refrigerant gas do not mix readily, and the oil can be properly circulated through the system only if gas velocities are high enough to sweep the oil along. If velocities are not sufficiently high, oil will tend to lie on the bottom of refrigeration tubing, decreasing heat transfer and possibly causing a shortage of oil in the compressor. As evaporating temperatures are lowered, this problem becomes more critical since the viscosity of the oil increases with a decrease in temperature. For these reasons, proper design of piping is essential for satisfactory oil return.

One of the basic characteristics of a refrigerant and oil mixture in a sealed system is the fact that refrigerant is attracted by oil and will vaporize and migrate through the system to the compressor crankcase even though no pressure difference exists to cause the movement. On reaching the crankcase the refrigerant will condense into liquid, and this migration will continue until the oil is saturated with liqud refrigerant.

Excess refrigerant in the compressor crankcase can result in violent foaming and boiling action, driving all of the oil from the crankcase and causing lubrication problems. Therefore, provisions must be made to prevent the accumulation of excess liquid refrigerant in the compressor.

R-22 and R-502 are much less soluble in oil than R-12, and for that reason proper piping and system design for these two refrigerants is more critical with regard to oil return.

REFRIGERANT TABLES

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

Figure 1 is an excerpt from an R-12 saturation table, which lists five major saturation properties of R-12, both liquid and vapor, at various temperatures. Pressure, volume, and density have been discussed previously.

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, and 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 at temperatures below $-40°$ F. is considered to have a negative enthalpy, while at all temperatures above $-40°$ F. the refrigerant 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. If the heat content per pound of the refrigerant entering and leaving a cooling coil can be determined, then the cooling ability of that coil can be calculated if the refrigerant flow rate is known.

Entropy can best be described as a mathematical ratio used in thermodynamics which is of use in solving complex refrigeration engineering problems. It is not easily defined or explained, is seldom used in commercial refrigeration applications and a discussion of it is beyond the scope of this manual.

Figure 2 is an excerpt from a R-502 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.

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. Figure 3 is a typical example of a pocket sized chart.

A saturation chart for ready reference is an invaluable tool for the refrigeration serviceman or for anyone checking the performance of a refrigeration system. Suction and discharge pressures can be readily checked by means of gauges, and from these pressures the evaporating and condensing temperatures can be determined.

R-12

SATURATION PROPERTIES—TEMPERATURE TABLE

TEMP. °F	PRESSURE PSIA	PSIG	VOLUME cu ft/lb LIQUID v_f	VAPOR v_g	DENSITY lb/cu ft LIQUID $1/v_f$	VAPOR $1/v_g$	ENTHALPY Btu/lb LIQUID h_f	LATENT h_{fg}	VAPOR h_g	ENTROPY Btu/(lb)(° R) LIQUID s_f	VAPOR s_g	TEMP. °F
−40	9.3076	10.9709*	0.010564	3.8750	94.661	0.25806	0	72.913	72.913	0	0.17373	−40
−39	9.5530	10.4712*	0.010575	3.7823	94.565	0.26439	0.2107	72.812	73.023	0.000500	0.17357	−39
−38	9.8035	9.9611*	0.010586	3.6922	94.469	0.27084	0.4215	72.712	73.134	0.001000	0.17343	−38
−37	10.059	9.441*	0.010596	3.6047	94.372	0.27741	0.6324	72.611	73.243	0.001498	0.17328	−37
−36	10.320	8.909*	0.010607	3.5198	94.275	0.28411	0.8434	72.511	73.354	0.001995	0.17313	−36
−35	10.586	8.367*	0.010618	3.4373	94.178	0.29093	1.0546	72.409	73.464	0.002492	0.17299	−35
−34	10.858	7.814*	0.010629	3.3571	94.081	0.29788	1.2659	72.309	73.575	0.002988	0.17285	−3[illegible]
−33	11.135	7.250*	0.010640	3.2792	93.983	0.30495	1.4772	72.208	73.685	0.003482	0.17271	
−32	11.417	6.675*	0.010651	3.2035	93.886	0.31216	1.6887	72.106	73.795	0.003976	[illegible]	
−31	11.706	6.088*	0.010662	3.1300	93.788	0.31949	1.9003	72.004	73.904	[illegible]		
												95
−30	11.999	5.490*	0.010674	3.0585	93.690						[illegible]16330	96
	12.299	4.880*	0.010685	2.9890						0.061959	0.16326	97
						3.1566	30.619	56.242	86.861	0.062381	0.16323	98
					[illegible].923	3.2017	30.859	56.086	86.945	0.062804	0.16319	99
100	131.86	117.16	0.012693	0.30794	78.785	3.2474	31.100	55.929	87.029	0.063227	0.16315	100
101	133.70	119.00	0.012715	0.30362	78.647	3.2936	31.341	55.772	87.113	0.063649	0.16312	101
102	135.56	120.86	0.012738	0.29937	78.508	3.3404	31.583	55.613	87.196	0.064072	0.16308	102
103	137.44	122.74	0.012760	0.29518	78.368	3.3877	31.824	55.454	87.278	0.064494	0.16304	103
104	139.33	124.63	0.012783	0.29106	78.228	3.4357	32.067	55.293	87.360	0.064916	0.16301	104
105	141.25	126.55	0.012806	0.28701	78.088	3.4842	32.310	55.132	87.442	0.065339	0.16297	105
106	143.18	128.48	0.012829	0.28303	77.946	3.5333	32.553	54.970	87.523	0.065761	0.16293	106
107	145.13	130.43	0.012853	0.27910	77.804	3.5829	32.797	54.807	87.604	0.066184	0.16290	107
108	147.11	132.41	0.012876	0.27524	77.662	3.6332	33.041	54.643	87.684	0.066606	0.16286	108
109	149.10	134.40	0.012900	0.27143	77.519	3.6841	33.286	54.478	87.764	0.067028	0.16282	109
110	151.11	136.41	0.012924	0.26769	77.376	3.7357	33.531	54.313	87.844	0.067451	0.16279	110
111	153.14	138.44	0.012948	0.26400	77.231	3.7878	33.777	54.146	87.923	0.067873	0.16275	111
112	155.19	140.49	0.012972	0.26037	77.087	3.8406	34.023	53.978	88.001	0.068296	0.16271	112
113	157.27	142.57	0.012997	0.25680	76.941	3.8941	34.270	53.809	88.079	0.068719	0.16268	113
114	159.36	144.66	0.013022	0.25328	76.795	3.9482	34.517	53.639	88.156	0.069141	0.16264	114
115	161.47	146.77	0.013047	0.24982	76.649	4.0029	34.765	53.468	88.233	0.069564	0.16260	115
116	163.61	148.91	0.013072	0.24641	76.501	4.0584	35.014	53.296	88.310	0.069987	0.16256	116
117	165.76	151.06	0.013097	0.24304	76.353	4.1145	35.263	53.123	88.386	0.070410	0.16253	117
118	167.94	153.24	0.013123	0.23974	76.205	4.1713	35.512	52.949	88.461	0.070833	0.16249	118
119	170.13	155.43	0.013148	0.23647	76.056	4.2288	35.762	52.774	88.536	0.071257	0.16245	119
120	172.35	157.65	0.013174	0.23326	75.906	4.2870	36.013	52.597	88.610	0.071680	0.16241	120
121	174.59	159.89	0.013200	0.23010	75.755	4.3459	36.264	52.420	88.684	0.072104	0.16237	121
122	176.85	162.15	0.013227	0.22698	75.604	4.4056	36.516	52.241	88.757	0.072528	0.16234	122
123	179.13	164.43	0.013254	0.22391	75.452	4.4660	36.768	52.062	88.830	0.072952	0.16230	123
124	181.43	166.73	0.013280	0.22089	75.299	4.5272	37.021	51.881	88.902	0.073376	0.16226	124
125	183.76	169.06	0.013308	0.21791	75.145	4.5891	37.275	51.698	88.973	0.073800	0.16222	125

EXCERPT FROM TYPICAL MANUFACTURER'S R-12 SATURATION PROPERTIES TABLE

Figure 1

"FREON - 502" — SUPERHEATED VAPOR

V=volume in cu ft/lb; H=enthalpy in Btu/lb; S=entropy in Btu/ (lb) (°R) (saturation properties in parentheses)

TEMP. °F	ABSOLUTE PRESSURE, lb/sq in												TEMP. °F
	25			26			27			28			
	10.30 psig			11.30 psig			12.30 psig			13.30 psig			
	(—28.41° F)			(—26.70° F)			(—25.04° F)			(—23.43° F)			
	V	H	S	V	H	S	V	H	S	V	H	S	
	(1.612)	(76.64)	(0.1777)	(1.554)	(76.86)	(0.1776)	(1.499)	(77.06)	(0.1774)	(1.449)	(77.26)	(0.1772)	
—25	1.627	77.15	0.1789	1.561	77.11	0.1782	1.500	77.07	0.1774	—	—	—	—25
—20	1.648	77.91	0.1807	1.581	77.87	0.1799	1.519	77.83	0.1792	1.462	77.79	0.1784	—20
—15	1.669	78.67	0.1824	1.602	78.63	0.1816	1.539	78.59	0.1809	1.481	78.55	0.1802	—15
—10	1.690	79.44	0.1841	1.622	79.40	0.1834	1.559	79.36	0.1826	1.500	79.32	0.1819	—10
— 5	1.711	80.21	0.1858	1.643	80.17	0.1851	1.579	80.13	0.1843	1.519	80.09	0.1836	— 5
0	1.732	80.98	0.1875	1.663	80.95	0.1868	1.598	80.91	0.1860	1.539	80.87	0.1853	0
5	1.754	81.76	0.1892	1.683	81.72	0.1884	1.618	81.69	0.1877	1.558	81.65	0.1870	5
10	1.775	82.54	0.1909	1.703	82.51	0.1901	1.638	82.47	0.1894	1.576	82.43	0.1887	10
15	1.796	83.33	0.1925	1.724	83.29	0.1918	1.657	83.26	0.1910	1.595	83.22	0.1903	15
20	1.817	84.12	0.1942	1.744	84.08	0.1934	1.677	84.05	0.1927	1.614	84.01	0.1920	20
25	1.838	84.91	0.1958	1.764	84.88	0.1951	1.696	84.84	0.1943	1.633	84.81	0.1936	25
30	1.858	85.71	0.1975	1.784	85.68	0.1967	1.716	85.64	0.1960	1.652	85.61	0.1953	30
35	1.879	86.51	0.1991	1.804	86.48	0.1983	1.735	86.44	0.1976	1.671	86.41	0.1969	35
40	1.900	87.32	0.2007	1.825	87.28	0.2000	1.755	87.25	0.1992	1.689	87.21	0.1985	40
45	1.921	88.13	0.2023	1.845	88.09	0.2016	1.774	88.06	0.2008	1.708	88.03	0.2001	45
50	1.942	88.94	0.2039	1.865	88.91	0.2032	1.793	88.87	0.2025	1.727	88.84	0.2017	50
55	1.963	89.75	0.2055	1.885	89.72	0.2048	1.813	89.69	0.2040	1.746	89.66	0.2033	55
60	1.983	90.58	0.2071	1.905	90.54	0.2064	1.832	90.51	0.2056	1.764	90.48	0.2049	60
65	2.004	91.40	0.2087	1.925	91.37	0.2079	1.851	91.34	0.2072	1.783	91.30	0.2065	65
70	2.025	92.23	0.2103	1.945	92.20	0.2095	1.870	92.16	0.2088	1.802	92.13	0.2081	70
75	2.045	93.06	0.2118	1.965	93.03	0.2111	1.890	93.00	0.2104	1.820	92.97	0.2097	75
80	2.066	93.89	0.2134	1.984	93.86	0.2126	1.909	93.83	0.2119	1.839	93.80	0.2112	80
85	2.087	94.73	0.2149	2.004	94.70	0.2142	1.928	94.67	0.2135	1.857	94.64	0.2128	85
90	2.107	95.58	0.2165	2.024	95.55	0.2157	1.947	95.52	0.2150	1.876	95.49	0.2143	90
95	2.128	96.42	0.2180	2.044	96.39	0.2173	1.966	96.36	0.2165	1.894	96.34	0.2158	95
100	2.149	97.27	0.2195	2.064	97.24	0.2188	1.986	97.22	0.2181	1.913	97.19	0.2174	100
105	2.169	98.13	0.2210	2.084	98.10	0.2203	2.005	98.07	0.2196	1931	98.04	0.2189	105
110	2.190	98.98	0.2226	2.104	98.96	0.2218	2.024	98.93	0.2211	1.950	98.90	0.2204	110
115	2.210	99.85	0.2241	2.123	99.82	0.2233	2.043	99.79	0.2226	1.968	99.76	0.2219	115
120	2.231	100.71	0.2256	2.143	100.68	0.2248	2.062	100.66	0.2241	1.987	100.63	0.2234	120
125	2.251	101.58	0.2271	2.163	101.55	0.2263	2.081	101.52	0.2256	2.005	101.50	0.2249	125
130	—	—	—	—	—	—	—	—	—	2.023	102.37	0.2264	130

EXCERPT FROM TYPICAL MANUFACTURER'S R-502 SUPERHEAT TABLE

Figure 2

Vapor Pressure, Psig

°F.	"Freon-113"	"Freon-114"	"Freon-13"	R-500**
−50		27.2*	57.0	
−48		27.0*	60.0	
−46		26.8*	63.0	
−44		26.6*	66.2	
−42		26.3*	69.4	
−40		26.1*	72.7	7.9*
−38		25.9*	76.2	6.7*
−36		25.6*	79.7	5.4*
−34		25.3*	83.3	4.2*
−32		25.0*	87.1	2.8*
−30	29.3*	24.7*	90.9	1.4*
−28	29.3*	24.4*	94.9	0.0
−26	29.2*	24.0*	98.9	0.8
−24	29.2*	23.7*	103.0	1.5
−22	29.1*	23.3*	107.3	2.3
−20	29.1*	22.9*	111.7	3.1
−18	29.0*	22.5*	116.2	4.0
−16	28.9*	22.1*	120.8	4.9
−14	28.9*	21.6*	125.7	5.8
−12	28.8*	21.1*	130.5	6.8
−10	28.7*	20.6*	135.4	7.8
−8	28.6*	20.1*	140.5	8.8
−6	28.5*	19.6*	145.7	9.9
−4	28.4*	19.0*	151.1	11.0
−2	28.3*	18.4*	156.5	12.1
0	28.2*	17.8*	162.2	13.3
2	28.1*	17.2*	167.9	14.5
4	28.0*	16.5*	173.7	15.7
6	27.9*	15.8*	179.8	17.0
8	27.7*	15.1*	185.9	18.4
10	27.6*	14.3*	192.2	19.8
12	27.5*	13.5*	198.6	21.2
14	27.3*	12.7*	205.2	22.7
16	27.1*	11.9*	211.9	24.2
18	27.0*	11.0*	218.8	25.7
20	26.8*	10.1*	225.8	27.3
22	26.6*	9.1*	233.0	29.0
24	26.4*	8.1*	240.3	30.7
26	26.2*	7.1*	247.8	32.5
28	26.0*	6.1*	255.5	34.3
30	25.8*	5.0*	263.3	36.1
32	25.6*	3.9*	271.3	38.0
34	25.3*	2.7*	279.5	40.0
36	25.1*	1.5*	287.8	42.0
38	24.8*	0.2*	296.3	44.1
40	24.5*	0.5	305.0	46.2
42	24.2*	1.2	313.9	48.4
44	23.9*	1.9	322.9	50.7
46	23.6*	2.6	332.2	53.0
48	23.3*	3.3	341.6	55.4
50	22.9*	4.0	351.2	57.8
52	22.6*	4.8	361.1	60.3
54	22.2*	5.6	371.1	62.9
56	21.8*	6.4	381.3	65.5
58	21.4*	7.3	391.7	68.2
60	21.0*	8.1	402.4	71.0
62	20.6*	9.0	413.3	73.8
64	20.1*	9.9	424.2	76.7
66	19.7*	10.9	435.6	79.7
68	19.2*	11.9	447.0	82.8
70	18.7*	12.9	458.8	85.8
72	18.2*	13.9	470.7	89.0
74	17.6*	15.0	482.9	92.3
76	17.1*	16.1	495.3	95.6
78	16.5*	17.2	508.1	99.0
80	15.9*	18.3	521.0	102.5
82	15.3*	19.5	534.1	106.1
84	14.6*	20.7	547.5	109.7
86	13.9*	22.0	ABOVE CRITICAL TEMPERATURE	113.4
88	13.2*	23.3		117.3
90	12.5*	24.6		121.2
92	11.8*	25.9		125.1
94	11.0*	27.3		129.2
96	10.2*	28.7		133.3
98	9.4*	30.2		137.6
100	8.6*	31.7		141.9
102	7.7*	33.2		146.3
104	6.8*	34.8		150.9
106	5.9*	36.4		155.4
108	4.9*	38.0		160.1
110	4.0*	39.7		164.9
112	3.0*	41.4		169.8
114	1.9*	43.2		174.8
116	0.8*	45.0		179.9
118	0.1	46.9		185.0
120	0.7	48.7		190.3
122	1.3	50.7		195.7
124	1.9	52.7		201.2
126	2.5	54.7		206.7
128	3.1	56.7		212.4
130	3.7	58.8		218.2
132	4.4	61.0		224.1
134	5.1	63.2		230.1
136	5.8	65.5		236.3
138	6.5	67.7		242.5
140	7.2	70.1		248.8
142	8.0	72.5		
144	8.8	74.9		
146	9.6	77.4		
148	10.4	80.0		
150	11.2	82.6		
152	12.1	85.2		
154	13.0	87.9		
156	13.9	90.7		
158	14.8	93.5		
160	15.7	96.4		

*Inches mercury below one atmosphere.
**Patented by Carrier Corporation.

Vapor Pressure, Psig

°F.	"Freon-11"	"Freon-12"	"Freon-502"	"Freon-22"
−50		15.4*	0.0	6.0*
−48		14.6*	0.8	4.7*
−46		13.8*	1.6	3.3*
−44		12.9*	2.5	1.8*
−42		11.9*	3.4	0.3*
−40	28.4*	11.0*	4.3	0.6
−38	28.3*	10.0*	5.2	1.4
−36	28.2*	8.9*	6.2	2.3
−34	28.1*	7.8*	7.2	3.2
−32	28.0*	6.7*	8.3	4.1
−30	27.8*	5.5*	9.4	5.0
−28	27.7*	4.3*	10.5	6.0
−26	27.5*	3.0*	11.7	7.0
−24	27.4*	1.6*	12.9	8.1
−22	27.2*	0.3*	14.2	9.2
−20	27.0*	0.6	15.5	10.3
−18	26.9*	1.3	16.9	11.5
−16	26.7*	2.1	18.3	12.7
−14	26.5*	2.8	19.7	13.9
−12	26.2*	3.7	20.2	15.2
−10	26.0*	4.5	22.8	16.6
−8	25.8*	5.4	24.4	18.0
−6	25.5*	6.3	26.0	19.4
−4	25.3*	7.2	27.7	20.9
−2	25.0*	8.2	29.4	22.5
0	24.7*	9.2	31.2	24.1
2	24.4*	10.2	33.1	25.7
4	24.1*	11.2	35.0	27.4
6	23.8*	12.3	37.0	29.2
8	23.5*	13.5	39.0	31.0
10	23.1*	14.6	41.1	32.9
12	22.7*	15.8	43.2	34.9
14	22.3*	17.1	45.4	36.9
16	21.9*	18.4	47.7	39.0
18	21.5*	19.7	50.1	41.1
20	21.1*	21.0	52.4	43.3
22	20.6*	22.4	54.9	45.5
24	20.2*	23.9	57.4	47.9
26	19.7*	25.4	60.0	50.2
28	19.1*	26.9	62.7	52.7
30	18.6*	28.5	65.4	55.2
32	18.1*	30.1	68.2	57.8
34	17.5*	31.7	71.1	60.5
36	16.9*	33.4	74.1	63.3
38	16.3*	35.2	77.1	66.1
40	15.6*	37.0	80.2	69.0
42	14.9*	38.8	83.4	72.0
44	14.2*	40.7	86.6	75.0
46	13.5*	42.7	90.0	78.2
48	12.8*	44.7	93.4	81.4
50	12.0*	46.7	96.9	84.7
52	11.2*	48.8	100.5	88.1
54	10.4*	51.0	104.1	91.5
56	9.5*	53.2	107.9	95.1
58	8.7*	55.4	111.7	98.8
60	7.7*	57.7	115.6	102.5
62	6.8*	60.1	119.6	106.3
64	5.8*	62.5	123.7	110.2
66	4.8*	65.0	127.9	114.2
68	3.7*	67.6	132.2	118.3
70	2.6*	70.2	136.6	122.5
72	1.5*	72.9	141.1	126.8
74	0.4*	75.6	145.6	131.2
76	0.4	78.4	150.3	135.7
78	1.0	81.3	155.1	140.3
80	1.6	84.2	159.9	145.0
82	2.2	87.2	164.9	149.8
84	2.9	90.2	170.0	154.7
86	3.6	93.3	175.1	159.8
88	4.3	96.5	180.4	164.9
90	5.0	99.8	185.8	170.1
92	5.7	103.1	191.3	175.4
94	6.5	106.5	196.9	180.9
96	7.3	110.0	202.6	186.5
98	8.1	113.5	208.4	192.1
100	8.9	117.2	214.4	197.9
102	9.8	120.9	220.4	203.8
104	10.6	124.6	226.6	209.9
106	11.5	128.5	232.9	216.0
108	12.5	132.4	239.3	222.3
110	13.4	136.4	245.8	228.7
112	14.4	140.5	252.5	235.2
114	15.3	144.7	259.2	241.9
116	16.4	148.9	266.1	248.7
118	17.4	153.2	273.1	255.6
120	18.5	157.7	280.3	262.6
122	19.6	162.2	287.6	269.7
124	20.7	166.7	295.0	277.0
126	21.9	171.4	302.5	284.4
128	23.0	176.2	310.2	291.8
130	24.3	181.0	318.0	299.3
132	25.5	185.9	326.0	307.1
134	26.8	191.0	334.1	315.2
136	28.1	196.1	342.3	323.6
138	29.4	201.3	350.7	332.3
140	30.8	206.6	359.2	341.3
142	32.2	212.0	367.8	350.3
144	33.7	217.5	376.7	359.4
146	35.1	223.1	385.6	368.6
148	36.6	228.8	394.7	377.9
150	38.2	234.6	404.0	387.2
152	39.7	240.5	413.4	396.6
154	41.3	246.5	423.0	406.1
156	43.0	252.6	432.7	415.6
158	44.6	258.8	442.6	425.1
160	46.3	265.1	452.6	434.6

*Inches mercury below one atmosphere.

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CONDENSED PRESSURE - TEMPERATURE CHART

Figure 3

Section 3

THE REFRIGERATION CYCLE

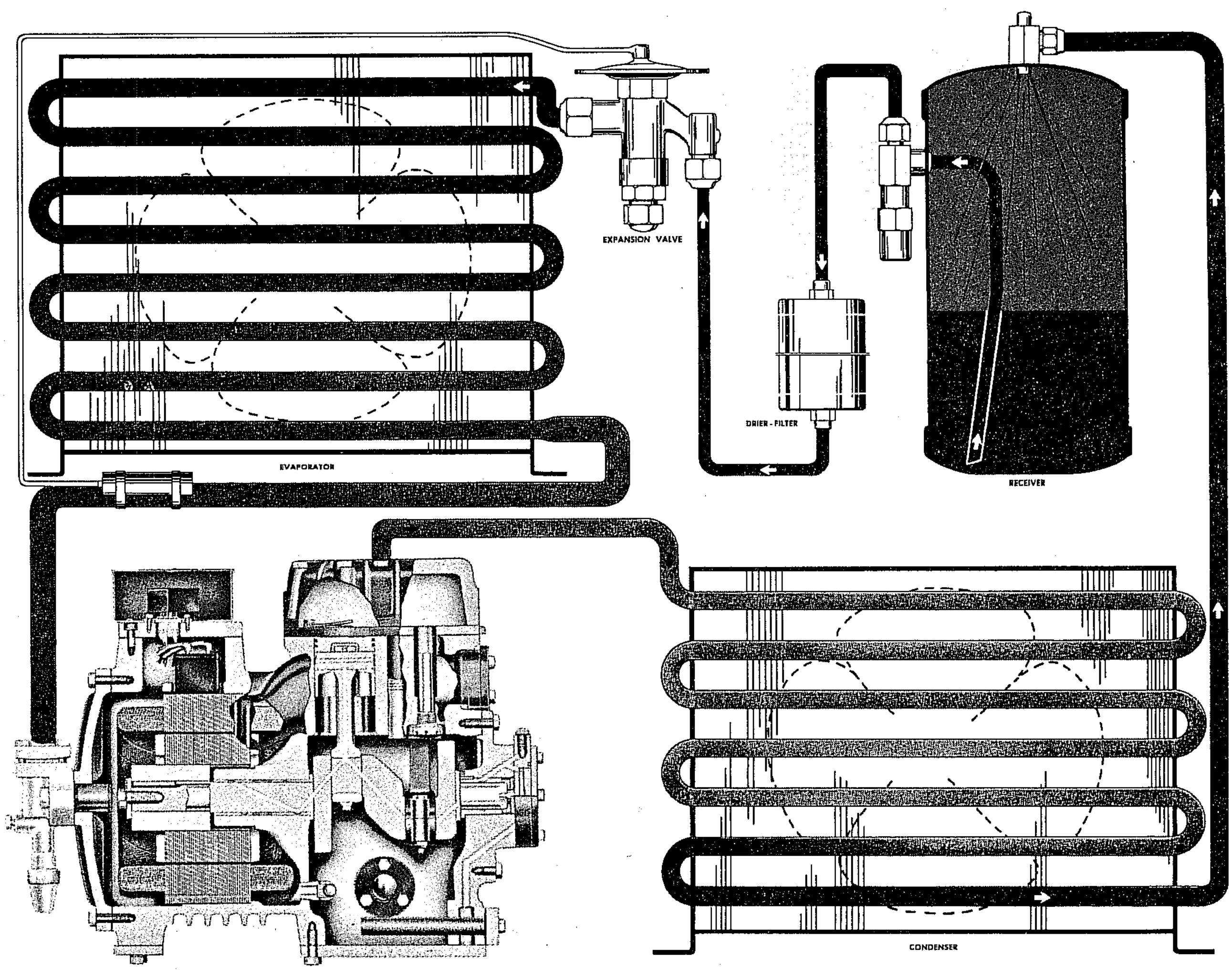
Continuous refrigeration can be accomplished by several different processes. In the great majority of applications, and almost exclusively in the smaller horsepower range, the vapor compression system, commonly termed the simple compression cycle, is used for the refrigeration process. However, absorption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

Copeland compressors, as their name implies, are designed for use with the compression cycle, and 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 from the evaporator to the condenser where it is given off to the am-



TYPICAL COMPRESSION REFRIGERATION SYSTEM

Figure 4

bient 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 discharge large quantities of heat efficiently.

The basic cycle operates as follows:

High pressure liquid refrigerant is fed from the receiver through the liquid line, and through the filter-drier to the metering device separating the high pressure side of the system from the low pressure evaporator. Various types of control devices may be used, but for purposes of this illustration, only the thermostatic expansion valve will be considered.

The thermostatic expansion valve controls the feed of liquid refrigerant to the evaporator, and by means of an orifice reduces the pressure of the refrigerant to the evaporating or low side pressure.

The reduction of pressure on the liquid refrigerant causes it to boil or vaporize 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 to the refrigerant, causing the boiling action to continue until the refrigerant is completely vaporized.

The expansion valve regulates the flow through the evaporator as necessary 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 its temperature, and acts to modulate the feed through the expansion valve as required.

The 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 both the pressure and the temperature. The hot, high pressure gas is forced out the compressor discharge valve and into the condenser.

As the high pressure gas passes through the condenser, it is cooled by some external means. On air cooled systems, a fan, and fin-type condenser surface, is normally used. On water cooled systems, a refrigerant-to-water heat exchanger is usually employed. As the temperature of the refrigerant vapor reaches the saturation temperature corresponding to the high pressure in the condenser, the vapor condenses into a liquid and flows back to the receiver to repeat the cycle.

The refrigeration process is continuous as long as the compressor operates.

HEAT OF COMPRESSION

When the refrigerant gas is compressed in the compressor cylinder, the 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 gas. But in order to compress the refrigerant gas, 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 gas 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 gas. This causes a sharp increase in the temperature of the compressed gas, so that the compressor discharge valves are always subjected to the highest temperature existing in the refrigeration system.

The heat of compression is defined as the heat added to the refrigerant gas as a result of the work energy used in compression.

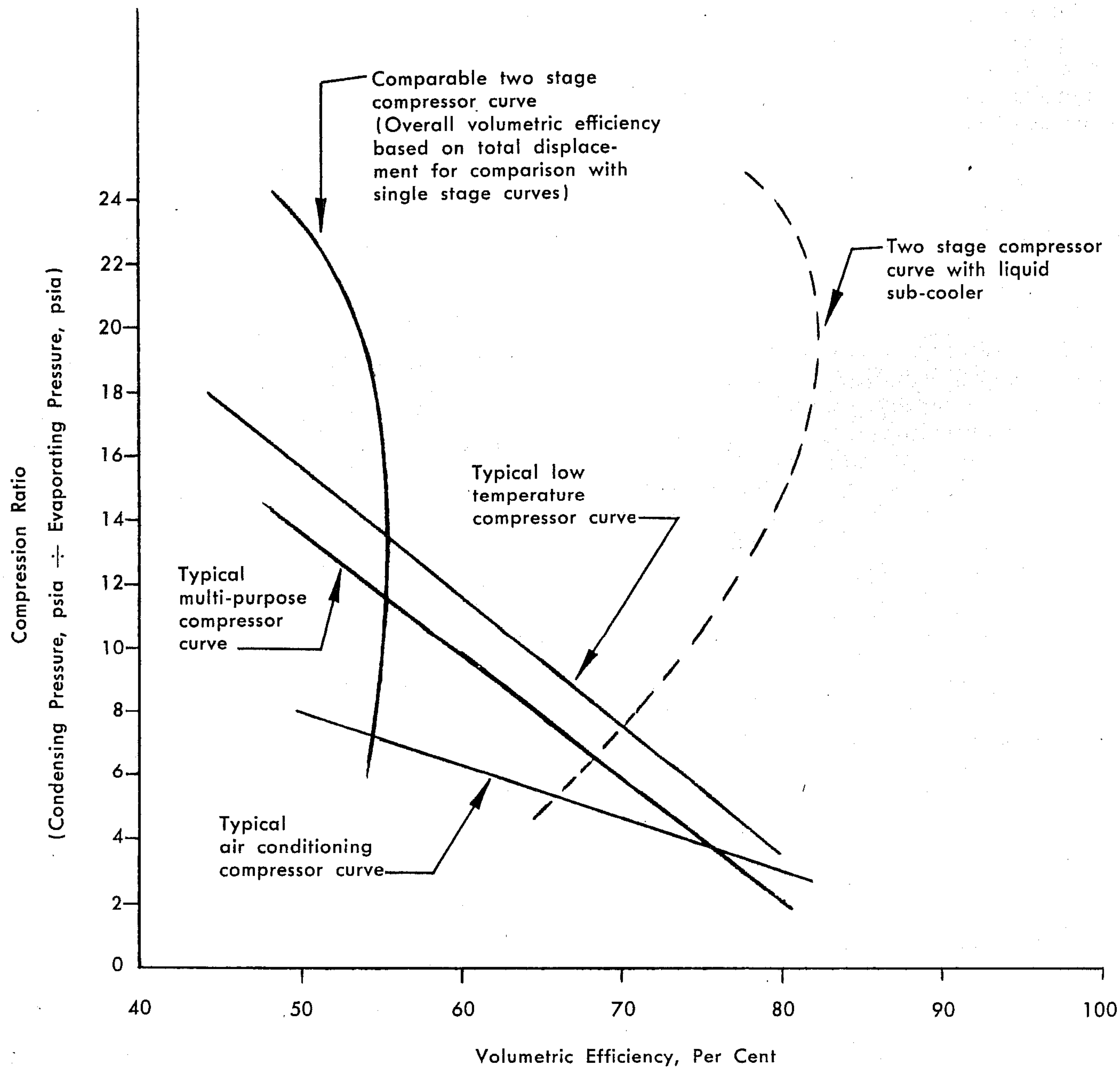
The heat which must be discharged by the condenser, termed the heat of rejection, is the total of the heat absorbed by the refrigerant in the evaporator, the heat of compression, and any heat added to the system due to motor inefficiency. For hermetic and accessible hermetic motor-compressors, the heat 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.

VOLUMETRIC EFFICIENCY OF THE COMPRESSOR

Volumetric efficiency is defined as the ratio of the actual volume of refrigerant gas pumped by the compressor to the volume displaced by the compressor pistons. The efficiency of a compressor can 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).

Several design factors can influence compressor efficiency including the clearance volume



TYPICAL COMPRESSOR VOLUMETRIC EFFICIENCY CURVES

Figure 5

above the piston, the clearance between the piston and the cylinder wall, valve spring tension, and valve leakage. For a given compressor, the effect on compressor efficiency because of design is fairly constant, and volumetric efficiency will vary almost inversely with the compression ratio.

Two factors cause a loss of efficiency with an increase in compression ratio. As the gas is subjected to greater compression, the residual gas remaining in the cylinder clearance space becomes more dense, and since it does not leave the cylinder on the discharge stroke, it re-expands on the suction stroke, thus preventing the intake of a full cylinder of gas from the suction line. The higher the pressure exerted on the residual gas, the more dense it becomes, and the greater volume it occupies on re-expansion.

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 gas entering the cylinder on the intake stroke is heated by the cylinder walls, and expands, resulting in a reduced weight of gas entering the compressor.

Typical volumetric efficiency curves are shown in Figure 5. Air conditioning compressors are usually designed with greater clearance volume, so the efficiency drops off much faster with an increase in compression ratio.

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

EFFECT OF CHANGE IN SUCTION PRESSURE

Other factors remaining equal, as the suction pressure is reduced, the specific volume of the gas returning to the compressor increases. Since a given compressor's pumping capacity is fixed by its speed and displacement, the reduction in density of the suction gas decreases the weight of the refrigerant pumped, with a resulting reduction in compressor capacity. 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, which reflects the performance of the compressor per unit of electrical energy consumed, also decreases rapidly with a drop in suction pressure. Therefore for best capacity performance and operating economy, it is most important that a refrigeration system operate at the highest suction pressure possible.

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, with a consequent loss of volumetric efficiency. While the loss of capacity is not as great as that caused by an equivalent decrease in suction pressure, it is still severe.

For operating economy and maximum capacity, the discharge pressure should be kept as low as practical.

EFFECT OF SUBCOOLING LIQUID REFRIGERANT WITH WATER OR AIR

When the hot high pressure liquid refrigerant is fed into the evaporator through the expansion valve, the refrigerant 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 absorbing the heat from the remaining liquid refrigerant.

The resulting flash gas can produce no further refrigeration, and 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, then the effective capacity of the system could be increased.

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 RERIGERANT BY SUPERHEATING THE VAPOR

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

1. To raise the temperature of the return suction gas so frosting or condensation will not occur on the suction line.

2. To subcool the liquid refrigerant sufficiently to offset any pressure drop that might occur in the liquid line, and prevent the formation of flash gas in the liquid line.

3. 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.

4. To increase total system capacity.

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 (a substantial increase in the case of R-12) 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.

As a practical matter, there may be a substantial capacity increase with all refrigerants. In most systems the suction lines are uninsulated and most of the superheat in the suction gas comes from the ambient air. If the low temperature suction gas can be used to subcool the liquid refrigerant, it is possible that only a small penalty will be paid in the form of higher return gas temperatures, particularly on units with long suction lines. The temperature difference between the suction line and the surrounding air will be smaller, and the heat transfer rate correspondingly less.

EFFECT OF SUPERHEATING THE VAPOR LEAVING THE EVAPORATOR

It is essential that the temperature of the gas returning to the compressor be a minimum of 15° F. above the evaporating temperature to avoid carrying liquid refrigerant back to the compressor. 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. Since such losses may be as high as 10% to 15%, insulation of the suction line can be a worthwhile investment, and may be necessary to prevent the return gas temperature from rising too high.

EFFECT OF PRESSURE DROP IN THE DISCHARGE LINE AND CONDENSER

Pressure drop due to friction as the refrigerant gas flows through the discharge line and condenser reduces compressor capacity due to the resulting 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, and for best operating economy, excessively high pressure drops in the discharge line should be avoided.

EFFECT OF PRESSURE DROP IN LIQUID LINE

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

Subcooling of the liquid refrigerant after condensing by an amount sufficient to offset the pressure drop normally will insure solid liquid refrigerant at the expansion valve. At 120° F. condensing temperature, 10° F. liquid subcooling will protect against flashing for pressure drops as follows:

R-12	21.3 psi
R-22	33.9 psi
R-502	34.5 psi

Refrigerants. 12, 22, and 502 are slightly heavier than water, and 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 must be provided for in system design.

EFFECT OF PRESSURE DROP IN THE EVAPORATOR

Pressure drop occuring 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, pressure drop in the suction line can be extremely detrimental to system capacity, and suction lines must be sized to prevent excessive pressure losses.

For example, on a typical 7½ HP compressor operating on R-12:

Evap. Temp.	Line Pressure Drop	Pressure At Comp.	BTU/hr. Capacity
—10° F.	1 psi	3.5 psig	32,400
—10° F.	2 psi	2.5 psig	30,100
—10° F.	3 psi	1.5 psig	27,800
—10° F.	4 psi	.5 psig	25,600

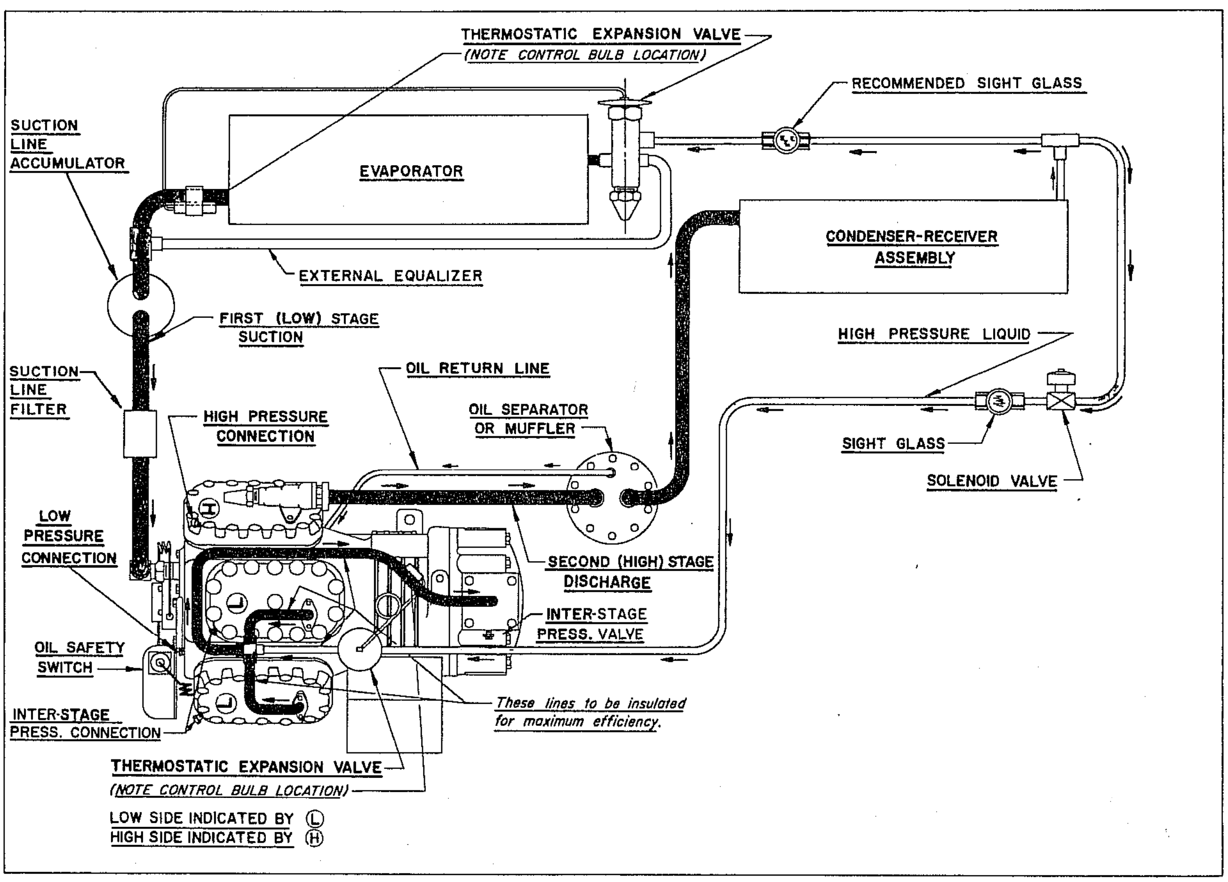
TWO-STAGE SYSTEMS

As the compression ratio increases, the volumetric efficiency of the compressor decreases and the heat of compression increases. For low temperature applications, the decreasing efficiency and excessively high discharge temperatures become increasingly critical and −40° F. is the lowest recommended evaporating temperature for compressors operating on the simple single stage compression cycle.

In order to increase operating efficiency at low temperatures the compression can be done in two steps or stages. For 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 ¼ 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, which can result in overheating the second stage cylinders and valves. To prevent compressor damage, liquid refrigerant must be injected between stages to properly cool the compressor.

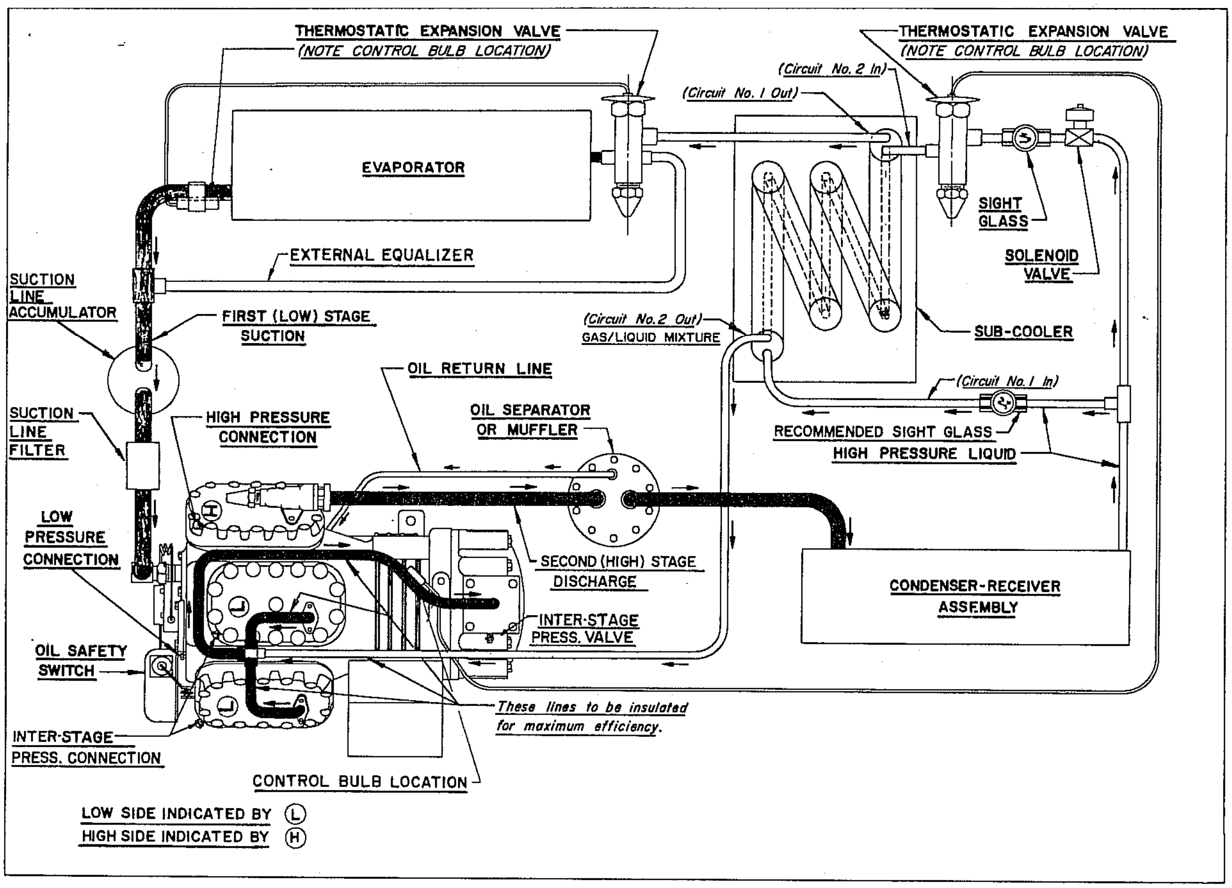
Two-stage compression may be accomplished with the use of two compressors with the discharge of one pumping into the suction inlet of the second, but because of the difficulty of maintaining proper oil levels in the two crankcases, it is more satisfactory to use one compressor with multiple cylinders. 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 6 and 7 illustrate typical two-stage compressors as applied to low temperature systems. Two-stage refrigeration is effective down to evaporator temperatures of −80° F. to −90° F. Below that level, efficiency drops off rapidly.



SYSTEM WITH 6-CYLINDER COMPRESSOR (WITHOUT LIQUID SUB-COOLER)

Figure 6



SYSTEM WITH 6-CYLINDER COMPRESSOR (WITH LIQUID SUB-COOLER)

Figure 7

CASCADE SYSTEMS

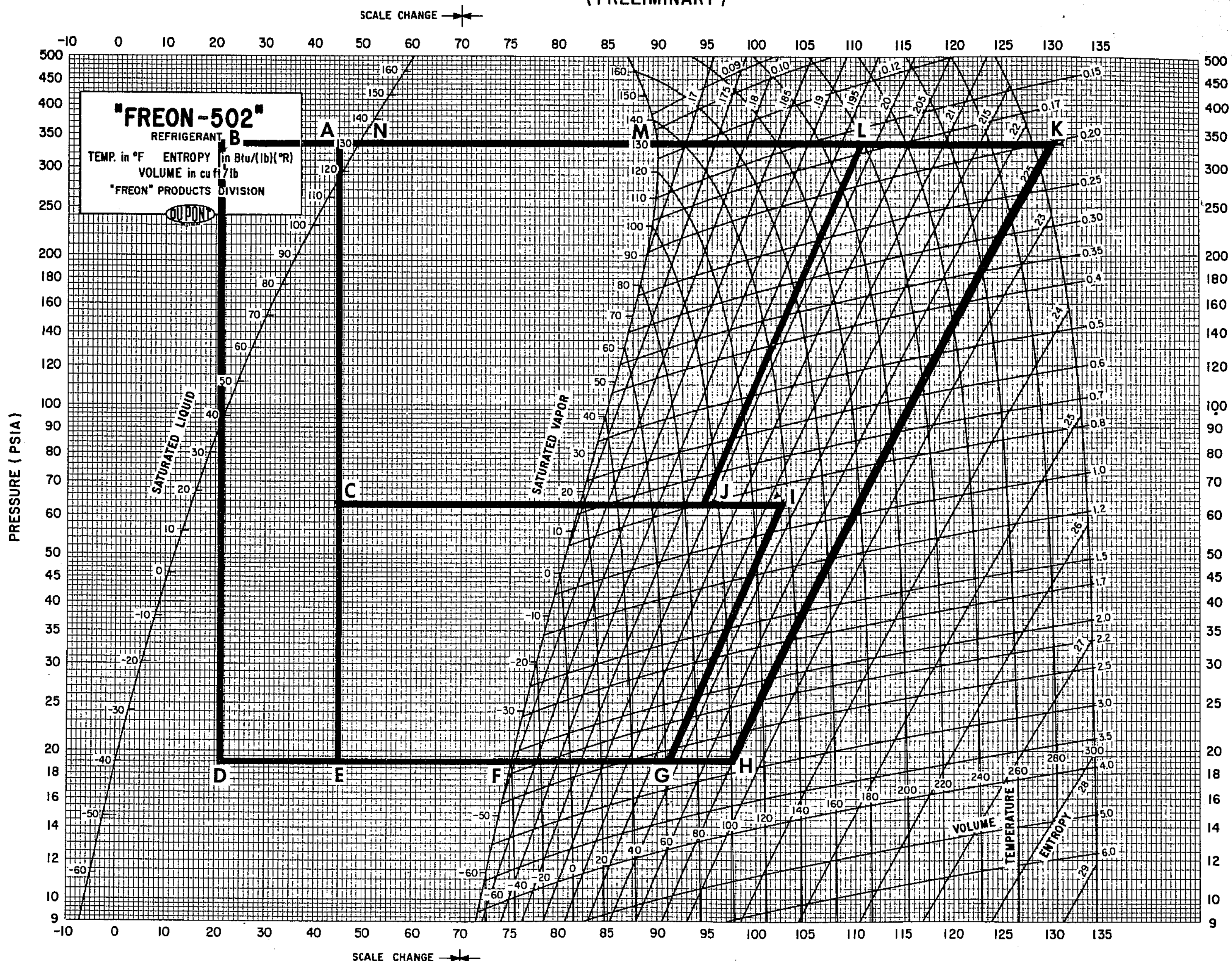
In order to operate satisfactorily at even lower evaporating temperatures, and to increase the flexibility of system design, multiple stage refrigeration can also be accomplished by using separate systems with the evaporator of one serving as the condenser of the second by means of a heat exchanger. 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, and 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.

REFRIGERATION CYCLE DIAGRAMS

Occasionally, in order to analyze system performance, or to visually represent the refrigeration cycle, diagrams and charts are used. The most common of these is the pressure-enthalpy diagram on which the refrigeration cycle is plotted.

Figure 8 is a pressure-enthalpy diagram with R-502 properties plotted. Lines have been drawn on the basic chart for constant enthalpy, constant pressure, constant entropy, constant temperature, constant volume, saturated vapor, and saturated liquid. Charts of typical compression cycles have been superimposed on the R-502 data so that single and two stage cycles can be compared. All cycles are shown with 130° F. condensing temperature and −40° F. evaporating temperature.

PRESSURE-ENTHALPY DIAGRAM
(PRELIMINARY)



ENTHALPY (Btu/lb. above Saturated Liquid at -40°F)

Figure 8

1. Single Stage Cycle

A typical single stage cycle with R-502 refrigerant would follow the processes illustrated by the diagramAEHKA on Figure 8. At point A, the refrigerant is sub-cooled liquid at 332.7 psia pressure and 115° F. temperature.

The line AE represents the throttling process as liquid refrigerant passes through the expansion valve. Since no heat transfer has taken place, the enthalpy has not changed, but the pressure is reduced to 19.0 psia.

Line EF represents the change of state from liquid to gas during the evaporating process at a constant temperature of −40° F. and a constant pressure of 19.0 psia. As heat is absorbed, the enthalpy of the refrigerant increases. Line FH illustrates superheating of the vapor. For this illustration, it was assumed that the gas returning to the compressor was superheated to 60° F., and an additional temperature rise to 100° F. was caused by the compressor prior to entering the compressor cylinders.

Line HK represents the compression process. Since compression is very rapid, very little heat transfer will take place to or from the refrigerant, and the process may be considered adiabatic (a process during which no heat transfer takes place and entropy value is constant). There is an increase in enthalpy because of the conversion of mechanical energy to heat energy in the refrigerant vapor. The process follows a constant entropy line, until the condensing pressure is reached. As a result of the increase in enthalpy, **the temperature of the gas increases to almost 310° F.**

Line KM represents the desuperheating of the hot gas, and at point M condensation begins and continues to point N at a constant temperature of 130° F. and a constant pressure of 332.7 psia. At point N, condensation is complete, and line NA represents liquid subcooling in the condenser to 115° F., where the cycle starts once again.

2. Two Stage Cycle Without Liquid Subcooler

While a pressure-enthalpy diagram can dramatically illustrate the difference in temperatures and pressures resulting from two-stage operation, it does not reflect compressor efficiency because of the lower compression ratios, since all enthalpy values are plotted on the basis of BTU/lb. The difference in capacity between single stage and two stage operation is best shown by a comparison of volumetric efficiency curves. (See Fig. 5).

A typical two stage cycle without liquid subcooling is represented by the diagram AEGIJLA. In addition, the line CJ represents the interstage cooling process.

The cycle again starts with liquid refrigerant at 332.7 psia subcooled to 115° F. The line AE illustrates the throttling process through the expansion valve for liquid refrigerant entering the evaporator. However, on two stage systems, a portion of the liquid refrigerant is used for cooling the refrigerant gas between stages of compression, and the line AC also represents liquid refrigerant passing through the interstage expansion valve.

Evaporation of the liquid refrigerant in the evaporator takes place as before at a constant temperature of −40° F. and a constant pressure of 19.0 psia along the line EF to point F. At point F the liquid is completely vaporized, and superheating of the vapor takes place to point G. In Copelametic two stage compressors, the suction gas returning from the evaporator does not pass over the compressor motor, but enters the cylinders directly, and again it is assumed for this illustration that the gas is superheated to 60° F. as it enters the compressor.

First stage compression takes place along the constant entropy line from G to I. At point I the gas is discharged from the first stage cylinders at the interstage pressure of 62.4 psia at a temperature of 130° F.

At this point the hot gas is mixed with the liquid refrigerant fed through the interstage expansion valve. Line CJ represents the increase in enthalpy for the interstage cooling refrigerant. Line IJ represents the decrease in enthalpy of the first stage discharge gas. The resulting mixture of refrigerant gas passes over the compressor motor and enters the compressor second stage cylinder or cylinders at a temperature of approximately 90° F. and at the interstage pressure of 62.4 psia.

Second stage compression takes place along the line JL to the condensing pressure of 332.7 psia. Note however that the compressor discharge gas temperature is only 220° F. as compared with 310° F. at the same point in the single stage cycle.

Desuperheating of the discharge gas and condensation takes place along line LN as before with liquid subcooling to point A, where the cycle is complete.

3. Two Stage Cycle With Liquid Subcooling

Two stage systems may operate either with or without liquid subcoolers, in which the liquid refrigerant going to the evaporator is cooled in a heat exchanger by means of the same liquid refrigerant feed used for interstage cooling. The effective capacity of a two stage compressor can be greatly increased by means of a liquid subcooler since the liquid entering the evaporator has a much lower enthalpy content, and hence a greater heat absorption capability.

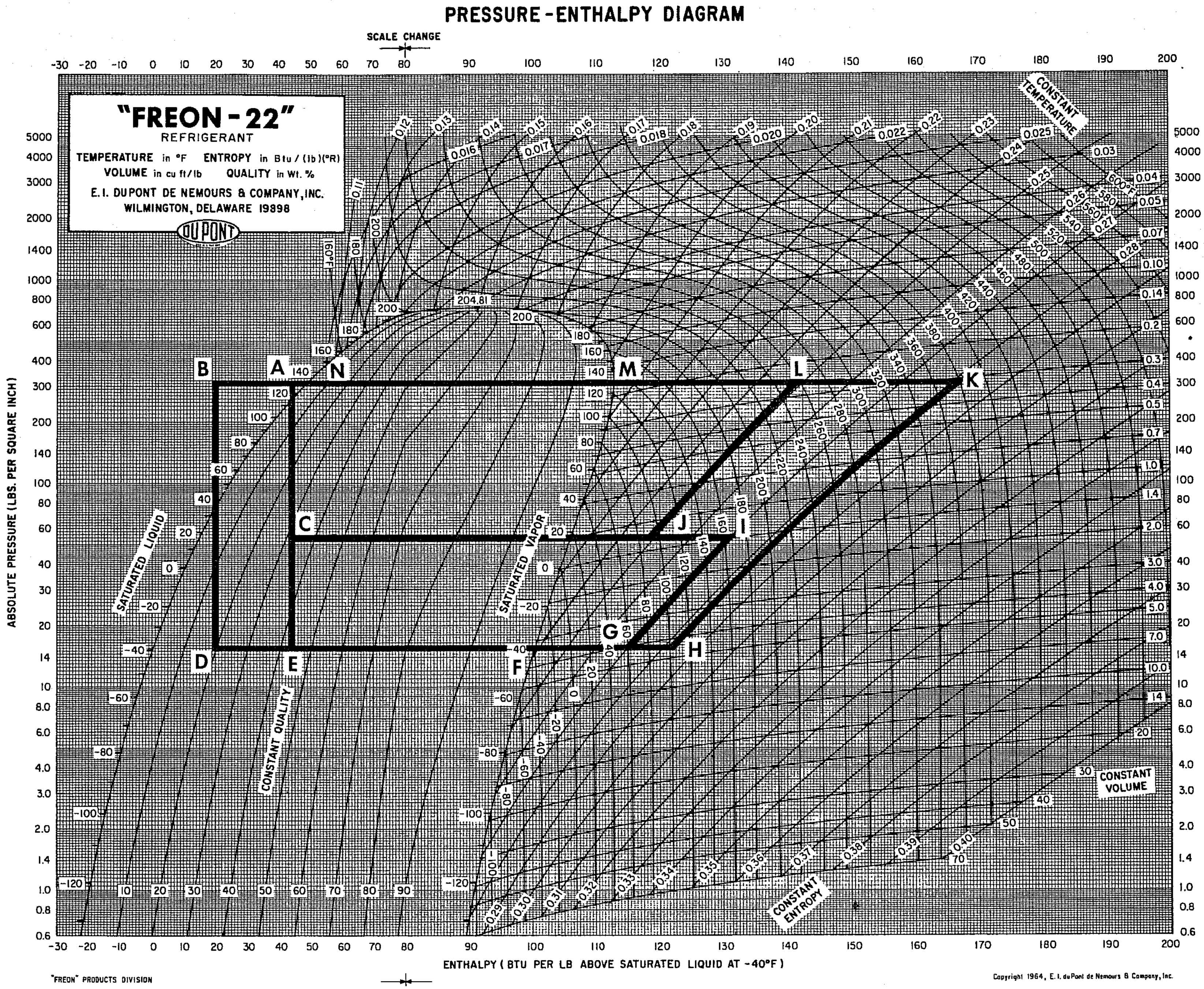
A typical two stage cycle with liquid subcooling is represented by a combination of diagrams BDGIJLB and ACJLA.

As before, the cycle is started at point A with liquid refrigerant at 332.7 psia subcooled to 115° F. Liquid flowing to the evaporator is subcooled along the line AB to 35° F. by liquid fed through the interstage expansion valve along line CJ. Line BD shows the throttling process through the expansion valve for liquid refrigerant entering the evaporator.

Evaporation occurs along line DF, and the balance of the cycle is identical to that of a two stage cycle without a subcooler. A greater quantity of refrigerant must be fed to the interstage expansion valve to accomplish both the liquid subcooling and the interstage cooling.

Figure 9 is a pressure-enthalpy diagram for R-22 similar to the R-502 diagram. Again the single stage cycle, a two stage cycle without liquid subcooling, and a two stage cycle with liquid subcooling are shown. The basic cycles are similar to those described previously for Figure 8.

One very important comparison between R-22 and R-502 is illustrated in these diagrams. The temperature at point K, the single stage discharge temperature, is 390° F. with R-22 as opposed to 310° F. with R-502. It is because of these temperature characteristics that Copeland recommends R-502 in place of R-22 for low temperature single stage systems.



PRESSURE-ENTHALPY DIAGRAM
SCALE CHANGE
"FREON-22"
REFRIGERANT
TEMPERATURE in °F ENTROPY in Btu/(lb)(°R)
VOLUME in cu ft/lb QUALITY in Wt. %
E. I. DUPONT DE NEMOURS & COMPANY, INC.
WILMINGTON, DELAWARE 19898
DU PONT
CONSTANT TEMPERATURE
CONSTANT VOLUME
CONSTANT ENTROPY
CONSTANT QUALITY
SATURATED LIQUID
SATURATED VAPOR
204.81
A
B
C
D
E
F
G
H
I
J
K
L
M
N
ABSOLUTE PRESSURE (LBS. PER SQUARE INCH)
ENTHALPY (BTU PER LB ABOVE SATURATED LIQUID AT -40°F)
"FREON" PRODUCTS DIVISION
Copyright 1964, E. I. du Pont de Nemours & Company, Inc.

Figure 9