Section 3

THE REFRIGERATION CYCLE

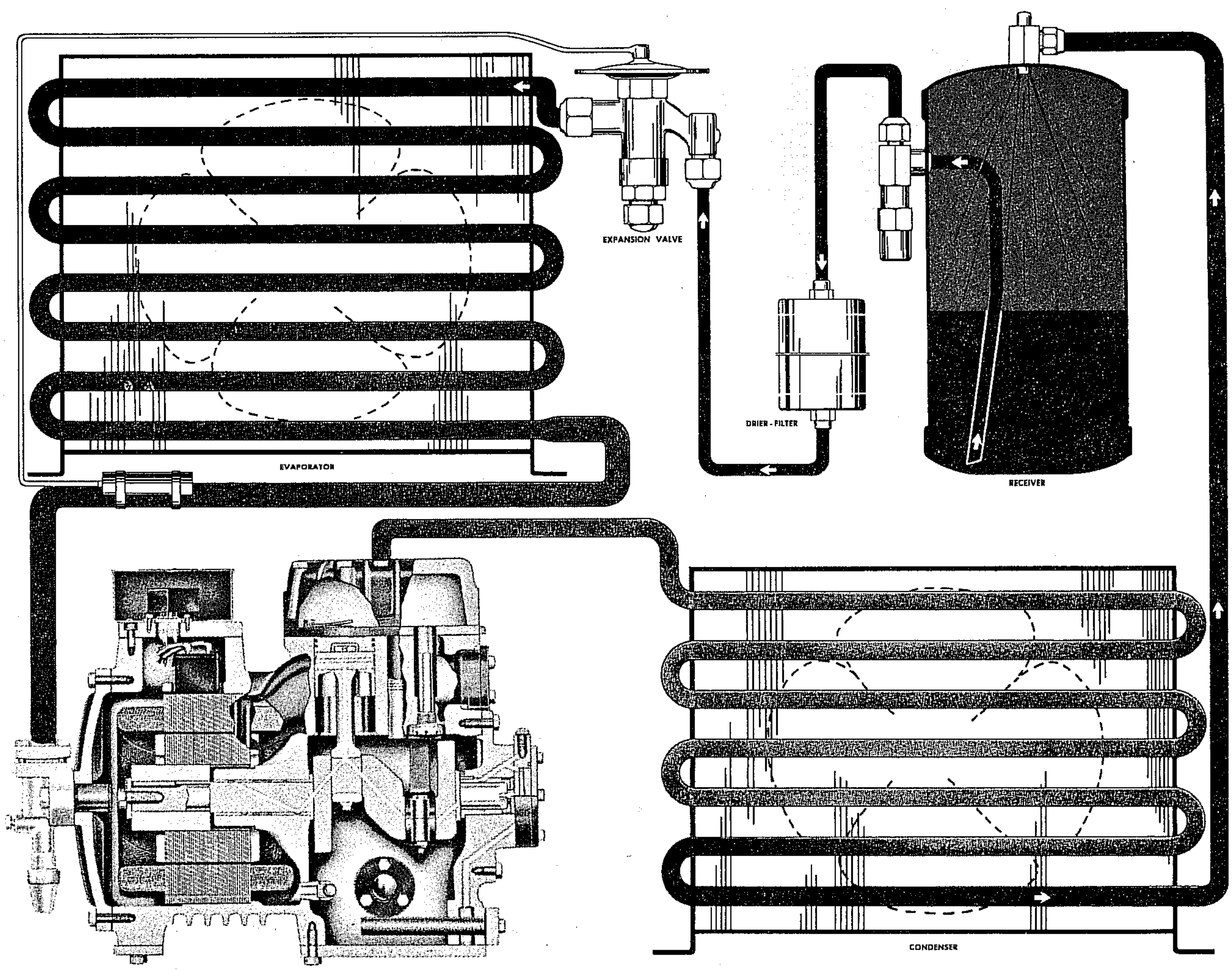
Continuous refrigeration can be accomplished by several different processes. In the great majority of applications, and almost exclusively in the smaller horsepower range, the vapor compression system, commonly termed the simple compression cycle, is used for the refrigeration process. However, absorption 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.

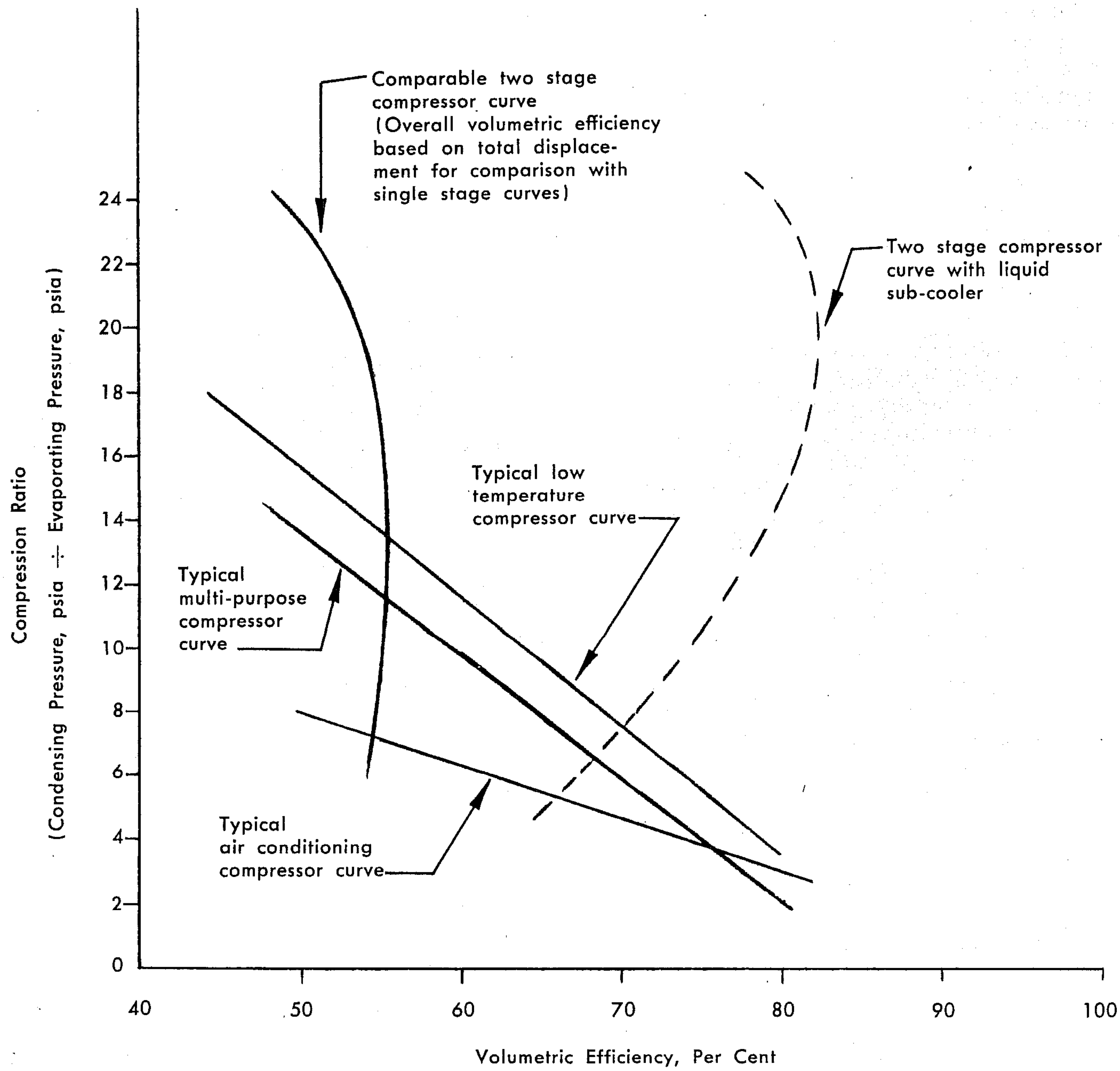
The heat which must be discharged by the condenser, termed the heat of rejection, is the total of the heat absorbed by the refrigerant in the evaporator, the heat of compression, and any heat added to the system due to motor inefficiency. For hermetic and accessible hermetic motor-compressors, the heat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

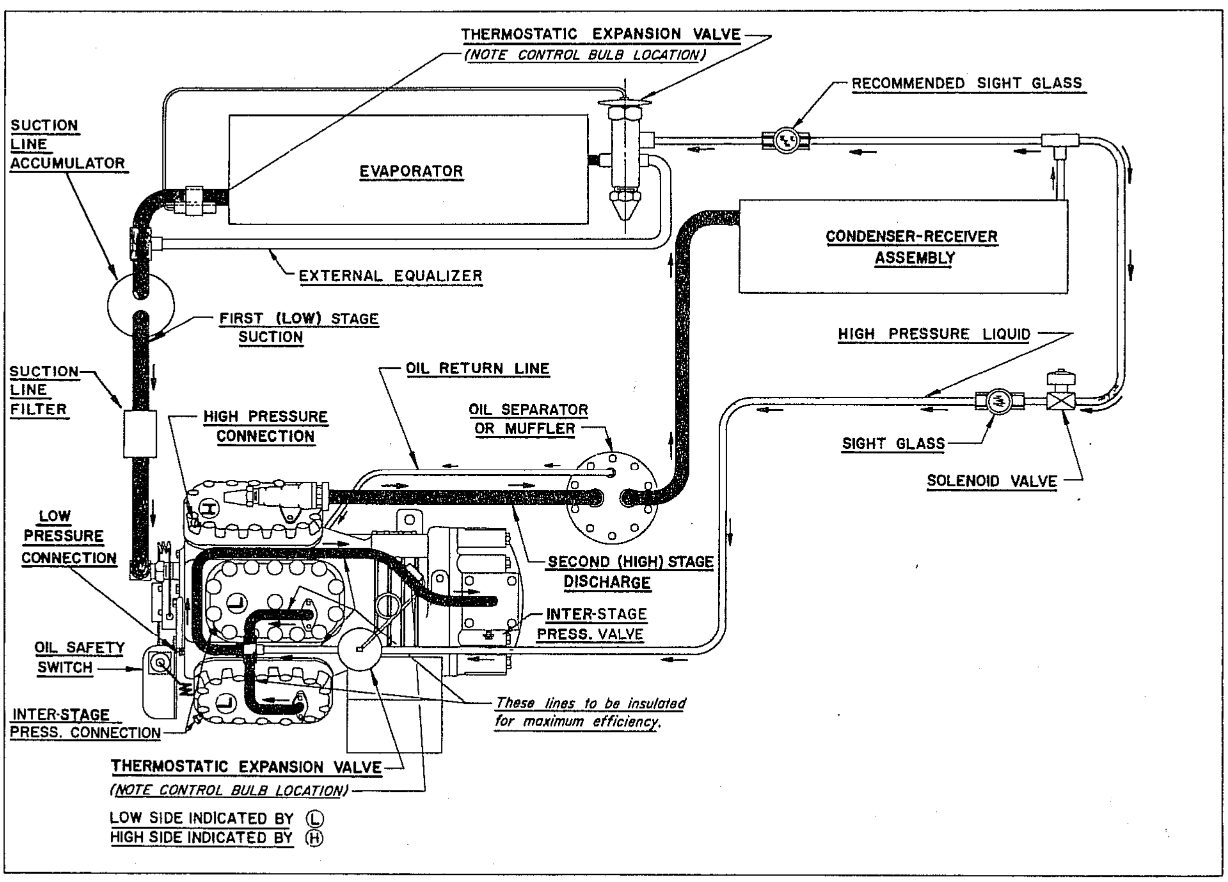
TWO-STAGE SYSTEMS

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.

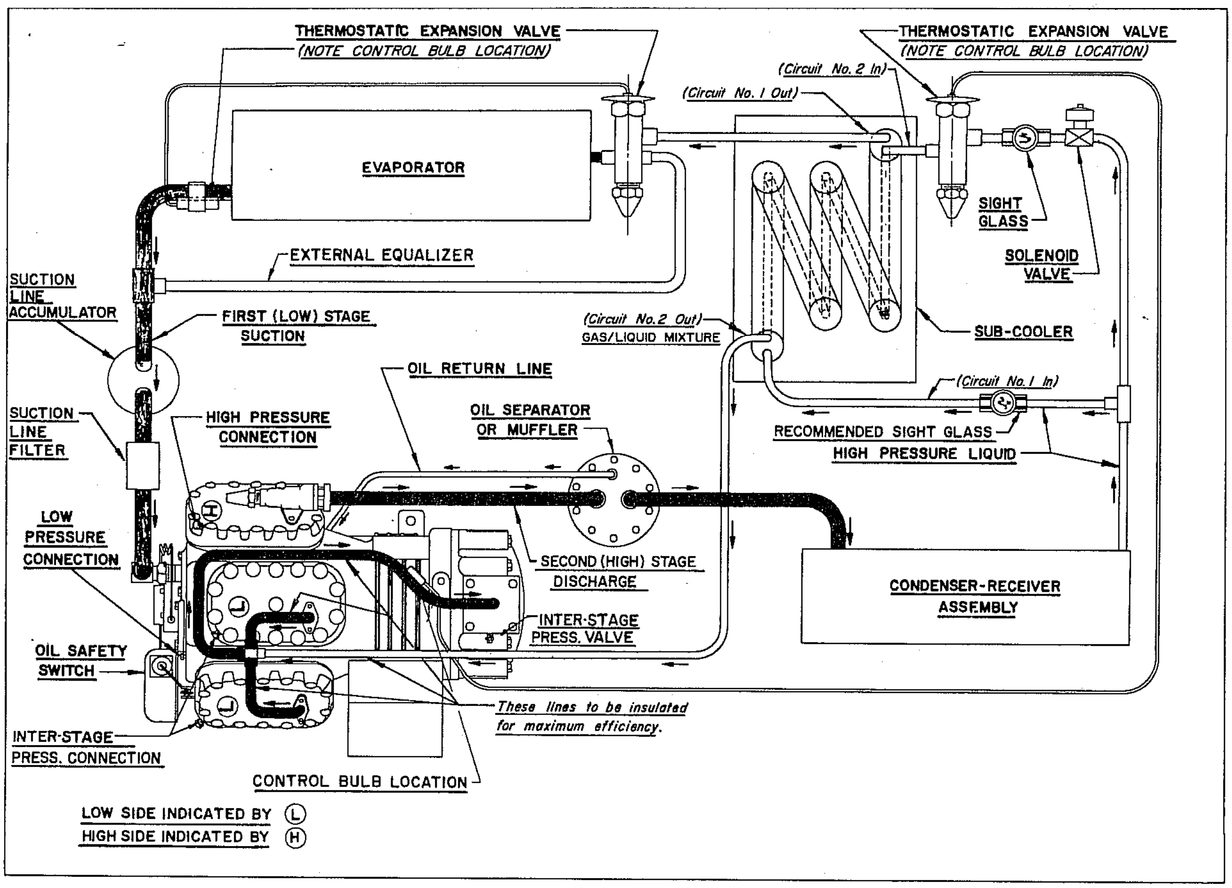
Two-stage compression may be accomplished with the use of two compressors with the discharge of one pumping into the suction inlet of the second, but because of the difficulty of maintaining proper oil levels in the two crankcases, it is more satisfactory to use one compressor with multiple cylinders. 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

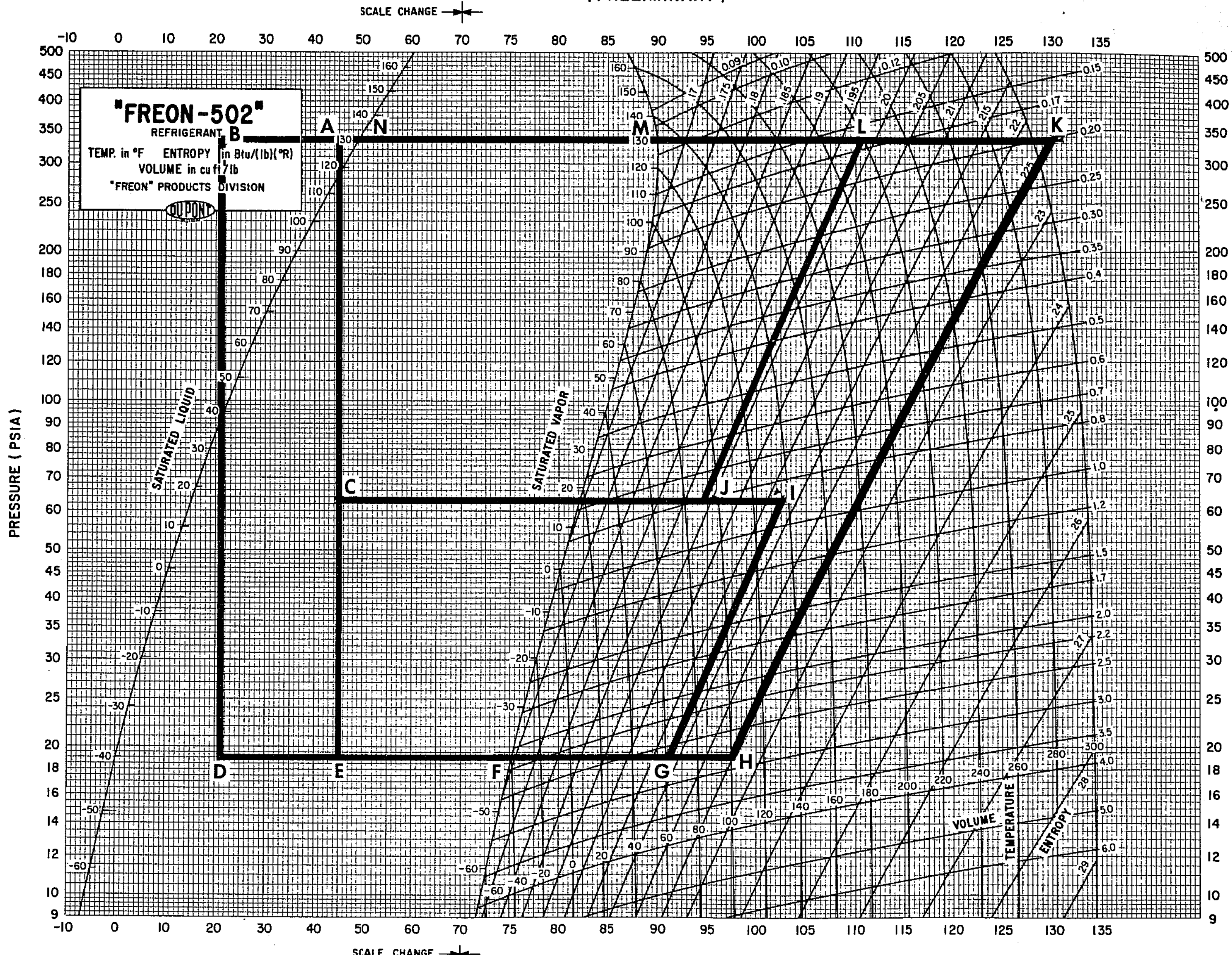
In order to operate satisfactorily at even lower evaporating temperatures, and to increase the flexibility of system design, multiple stage refrigeration can also be accomplished by using separate systems with the evaporator of one serving as the condenser of the second by means of a heat exchanger.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

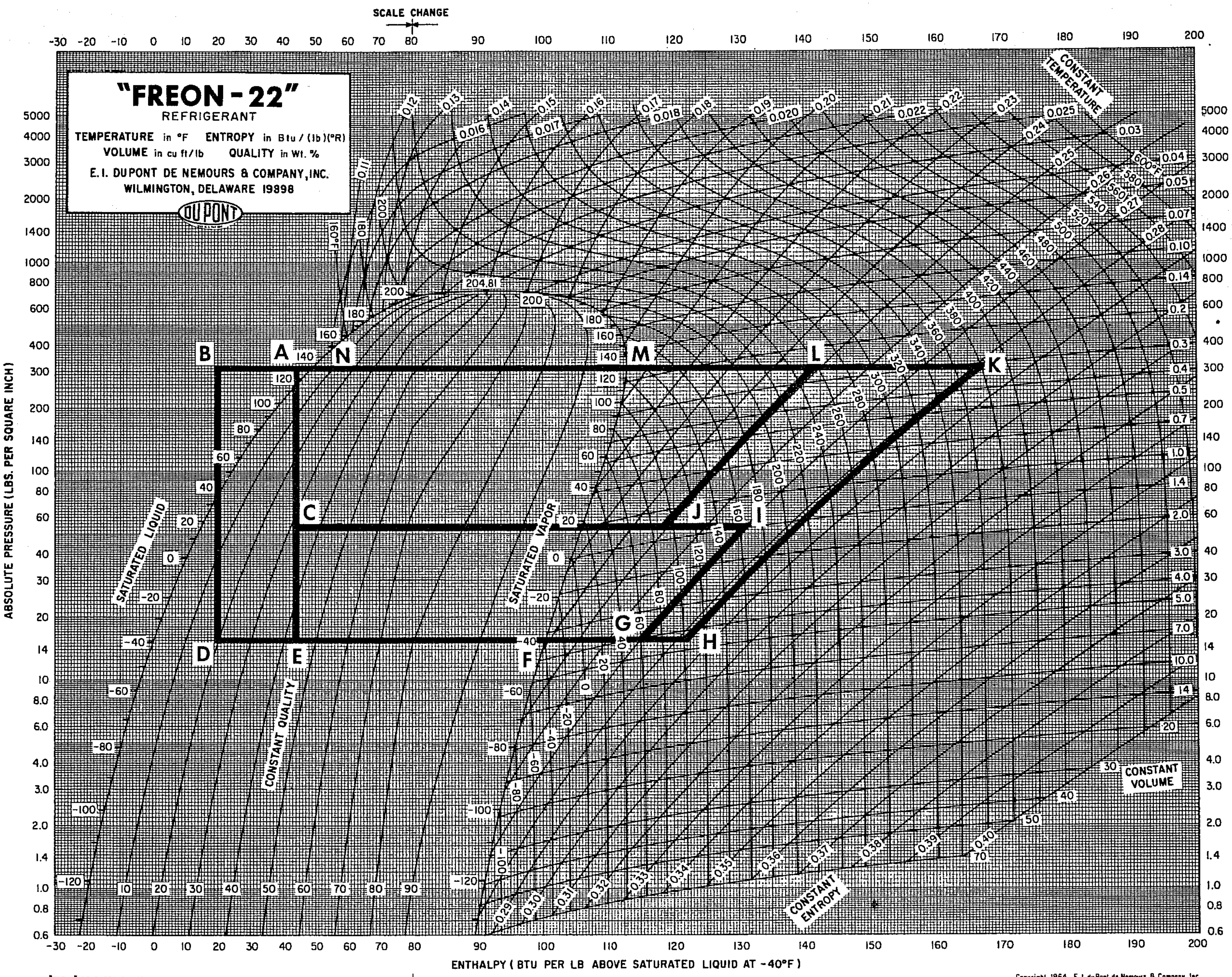


Figure 9