Section 19

LOW TEMPERATURE SYSTEMS

SINGLE STAGE LOW TEMPERATURE SYSTEMS

Low temperature single stage systems become increasingly critical from a design and application standpoint as the desired evaporating temperature is decreased. The combination of high compression ratios, low operating temperatures, and rarified return gas can cause lubrication and overheating problems, and make the compressor more vulnerable to damage from moisture and contaminants in the system.

The compressor selection, suction temperature, and application must be such that the temperature of the discharge line measured within 1″ to 6″ of the discharge service valve does not exceed 230° F. for Refrigerants 12, 22, and 502. Under these conditions, the estimated average temperature at the discharge port (measured at the valve retainer on the valve plate) will be approximately 310° F. for R-12 and R-502, and 320° F. for R-22.

The compressor displacement, pressure limiting devices, and quantity of cooling air or water must be selected to prevent the motor temperature from exceeding the limits stated below:

A. 210° F. when protected by inherent protectors affected by line current and motor temperature.

B. 190° F. when protected by motor starters.

The temperature of the motor should be determined by the resistance method and should be determined when the compressor is tested in the highest ambient in which it is expected to operate, at 90 per cent of rated voltage, with 90° F. return suction gas temperature. For longer motor life, operating temperatures of 170° F. to 190° F. are highly recommended.

In order to prevent the discharge and motor temperatures from exceeding recommended limits, it is very desirable, and in some instances absolutely necessary, to insulate the suction lines and return the suction gas to the compressor at a lower than normal temperature. This is particularly important with suction-cooled compressors when R-22 is used. (Approximately 30° F. superheat suggested.)

Suction cooled compressors require auxiliary cooling by means of an air blast on the compressor for operation below 0° F. evaporator temperature.

Either the evaporator must be properly designed, or a pressure limiting device such as a pressure limiting expansion valve or crankcase pressure regulating valve must be provided to prevent motor overloading during pulldown periods, or after defrost.

Copeland now recommends R-502 for all single stage low temperature applications where evaporating temperatures of -20° F. and below may be encountered. Now that R-502 is readily available, R-22 should not be used in single stage low temperature compressors, 5 H.P. and larger. The lower discharge temperatures of R-502 have resulted in much more trouble-free operation.

An adequate supply of oil must be maintained in the crankcase at all times to insure continuous lubrication. If the refrigerant velocity in the system is so low that rapid return of the oil is not assured, an adequate oil separator must be used. The normal oil level should be maintained at or slightly above the center of the sight glass. An excessive amount of refrigerant or oil must not be allowed in the system as it may result in excessive liquid slugging and damage to the compressor valves, pistons, or cylinders.

The formation or make up of the lines must be so designed that oil trapping will not exist. The highest velocity possible without encountering excessive pressure drop is recommended.

Care must be taken to prevent the evaporating temperature from dropping so far below the normal system operating point that refrigerant velocity becomes too low to return oil to the compressor. The low pressure control cut-out setting should not be below the lowest published rating point for the compressor, without prior approval of the Copeland Application Engineering department.

The smallest practical size tubing should be used in condensers and evaporators in order to hold the system charge to a minimum. When large refrigerant charges are unavoidable, re-cycling pumpdown control should be used.

If air cooled condensing units are required to operate in low ambient temperatures, the use of some means of head pressure control to prevent the condensing pressure from falling too low is highly recommended to maintain normal refrigerant velocities. Several commonly used types of control are described in Section 17.

An adequate filter-drier of generous size must be installed in the liquid line, preferably in the cold zone. The desiccant used must be capable of removing moisture to a low end point and be capable of removing a reasonable quantity of acid. It is most important that the filter-drier be equipped with an excellent filter to prevent circulation of carbon and foreign particles. A permanent suction line filter is highly recommended to protect the compressor from contaminants which may be left in the system during installation.

A combination liquid sight glass and moisture indicator should be installed for easy field maintenance.

After complete assembly, all systems should be thoroughly evacuated with a high grade vacuum pump and dehydrated to assure that no air or moisture remains in the system. The triple evacuation method should be used, breaking the vacuum each time with dry refrigerant through a dehydrator. The compressor motor must not be operated while the high vacuum pump is in operation, otherwise motor damage is very likely to occur.

The system should be charged with clean dry refrigerant only through a dehydrator. Other substances such as liquid dehydrants or alcohol must not be used.

TWO STAGE LOW TEMPERATURE SYSTEMS

Two stage systems because of their basic design and operation are inherently more efficient and encounter fewer operating hazards at low operating temperatures than single stage equipment. The two stage compressor has its limitations. At evaporating temperatures below -80° F. it loses efficiency and motor heating becomes an increasing problem. Copeland's lowest approved operating range is -80° F. and at lower evaporating temperatures a cascade system is recommended. But for applications with evaporating temperatures in the -20° F. to -80° F. range, the two stage compressor efficiency is high, the discharge temperatures are low, and field experience with properly applied two stage compressors has been excellent.

The two stage system is somewhat more complex and sophisticated than a simple single stage system, and many of the operating problems encountered on two stage systems stem from the fact that too often they have been applied without sufficient appreciation of the safeguards which must be taken in system design.

VOLUMETRIC EFFICIENCY

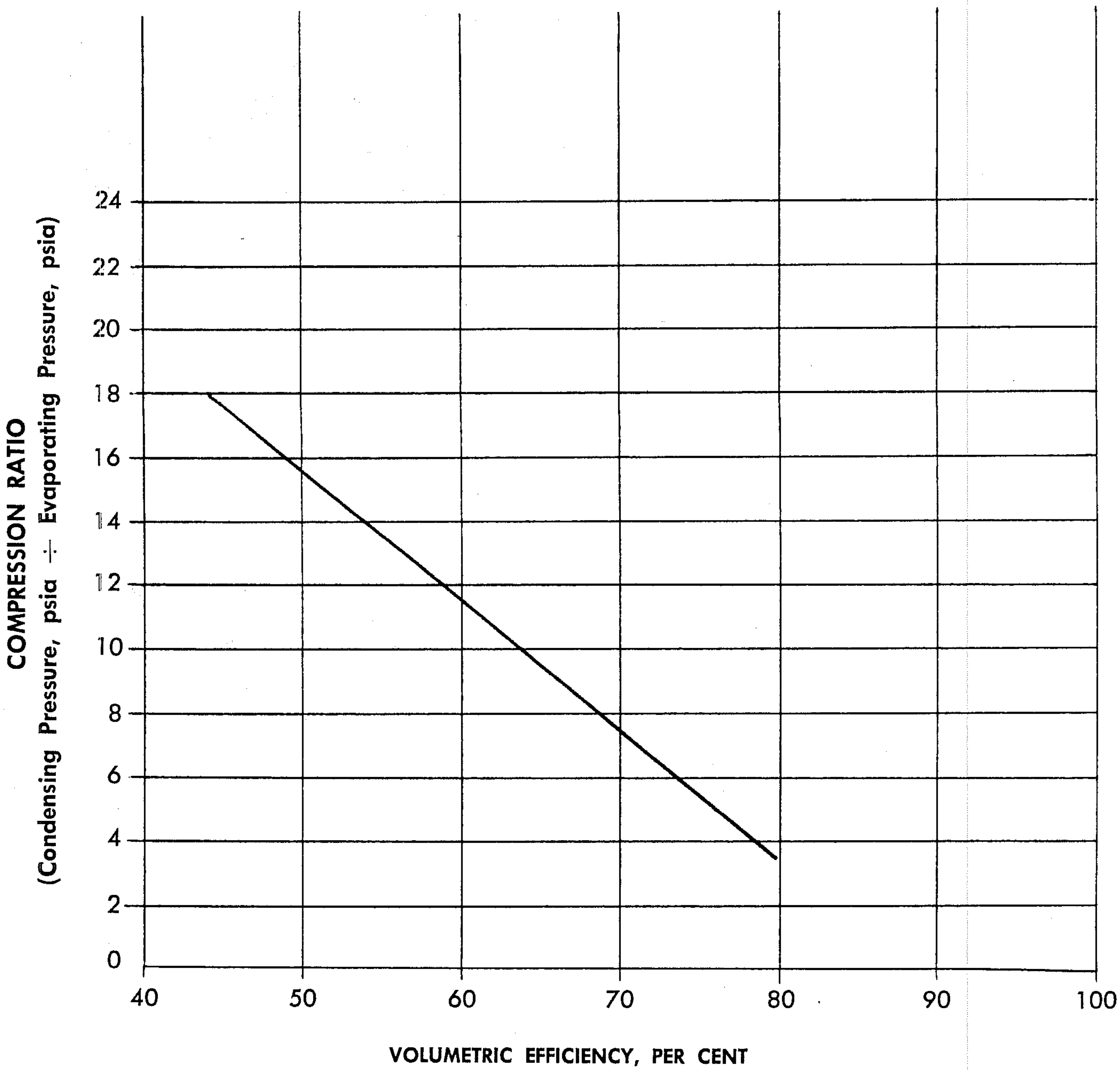
Three definitions given previously are of importance in analyzing two stage systems.

The compression ratio is the ratio of the absolute discharge pressure (psia) to the absolute suction pressure (psia).

The absolute pressure is gauge pressure plus atmospheric pressure, which at sea level is standardized at 14.7 pounds per square inch.

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

Figure 92 illustrates a typical single stage volumetric efficiency curve. Note that as the compression ratio increases, the volumetric efficiency decreases.



TYPICAL SINGLE STAGE LOW TEMPERATURE COMPRESSOR EFFICIENCY CURVE

Figure 92

Two factors cause a loss of efficiency with an increase in compression ratio. The density of the residual gas remaining in the cylinder clearance space after the compression stroke is determined by the discharge pressure — the greater the discharge pressure the greater the density. Since this gas 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 vapor from the suction line. As the compression ratio increases, the more space in the cylinder on the intake stroke is filled by the residual gas.

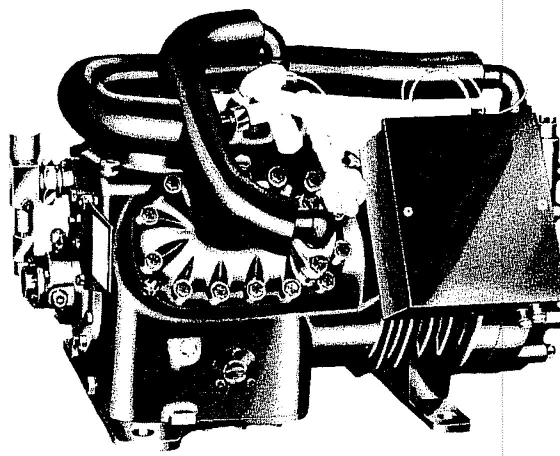
The second factor in the loss of efficiency is the high temperature of the cylinder walls resulting from the heat of compression. As the compression ratio increases, the heat of compression increases, and the cylinders and head of the compressor become very hot. Suction gas entering the cylinder on the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

Obviously, a single stage compressor has its limitations as compression ratios increase. The effective low limit of even the most efficient single stage system is approximately -40° F. evaporating temperature. At lower evaporating temperatures, the compression ratio becomes so high that capacity falls rapidly, the compressor may no longer be handling a sufficient weight of return gas for proper motor cooling, and because of decreased gas density, oil may no longer be properly circulated through the system.

TWO STAGE COMPRESSION AND COMPRESSOR EFFICIENCY

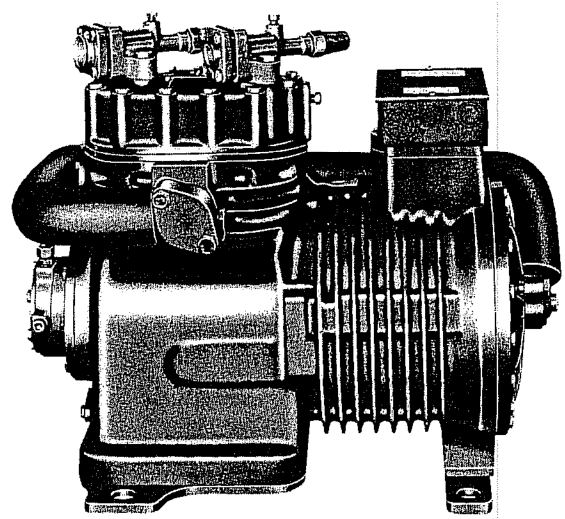
In order to increase operating efficiency at low evaporating temperatures, the compression can be done in two steps or stages. For two stage operation, the total compression ratio is the product of the compression ratio of each stage. In other words, for a total compression ratio of 16 to 1, the compression ratio of each stage might be 4 to 1; or compression ratios of 4 to 1 and 5 to 1 in separate stages will result in a total compression ratio of 20 to 1.

Two stage compression may be accomplished with the use of two compressors with the discharge of one pumping into the suction 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. On Copeland 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 refrigerant vapor at low and interstage pressures. While the compression ratios of the two stages are seldom exactly equal, they will be approximately the same. A typical 6 cylinder two stage compressor with its external manifold and desuperheating expansion valve is shown in Figure 93, and a typical 3 cylinder two stage compressor with external manifold is shown in Figure 94.



TYPICAL SIX CYLINDER
TWO STAGE COMPRESSOR

Figure 93

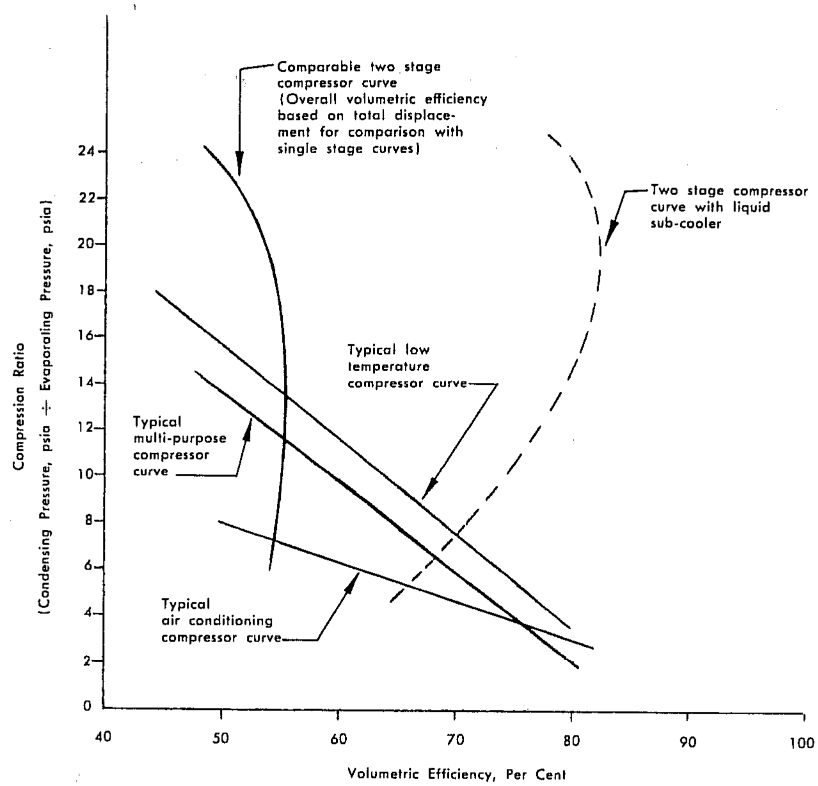


TYPICAL THREE CYLINDER
TWO STAGE COMPRESSOR

Figure 94

Figure 95 shows a comparison of five different volumetric efficiency curves. The three straight lines are typical single stage curves—one for an air conditioning compressor, one for a typical multi-purpose compressor, and one for a low temperature compressor. There are some variations in compressor design involved, but the primary difference in characteristics is due to clearance volume.

The two vertical curved lines represent the comparative efficiency of a two stage compressor. Actually each separate stage would have



TYPICAL COMPRESSOR VOLUMETRIC EFFICIENCY CURVES

Figure 95

a straight line characteristic similar to the single stage curves, but to enable comparison with single stage compressors, the overall volumetric efficiency has been computed on the basis of the total displacement of the compressor, not just the low stage displacement.

The solid black curve represents the efficiency of a two stage compressor without a liquid subcooler. Note that the efficiency is relatively constant over a wide range of total compression ratios, and that the crossover in efficiency with the best low temperature single stage compressor is at a compression ratio of approximately 13 to 1. In other words, at compression ratios lower than 13 to 1, a single stage compressor will have more capacity than a two stage compressor of equal displacement without liquid subcooling.

The dotted curve represents the efficiency of the same two stage compressor with a liquid subcooler. In the subcooler, the liquid refrigerant being fed to the evaporator is first subcooled by liquid refrigerant fed through the interstage desuperheating expansion valve, and a much greater share of the refrigeration load has been transferred to the high stage cylinders. Since the high stage cylinders operate at a much higher suction pressure, the refrigeration capacity there is far greater per cubic foot of displacement than in the low stage cylinders. In effect the capacity of the compressor has been greatly increased without having to handle any additional suction gas returning from the evaporator. Note that with the liquid subcooler, the crossover point in efficiency as compared with a single stage compressor is at a compression ratio of approximately 7 to 1. In other words, at compression ratios lower than 7 to 1, the single stage compressor will have more capacity for equal displacement, but at compression ratios higher than 7 to 1, the two stage compressor will have more capacity.

Table 42 lists comparative operating data at varying evaporating temperatures for a Copeland compressor available either as a single stage or two stage compressor. Although the displacement, refrigerant, and motor are the same, the rapidly increasing advantage of two stage operation as the evaporating temperature decreases is plainly shown.

Table 42

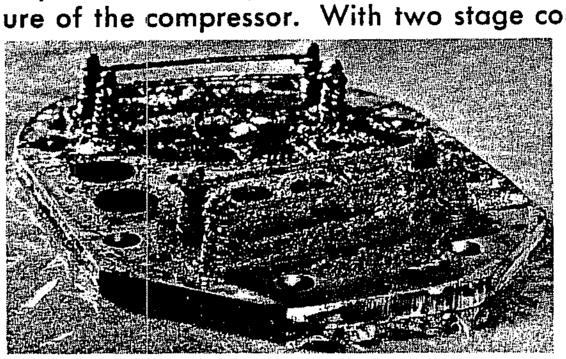
EFFICIENCY COMPARISON OF SINGLE STAGE VS. TWO STAGE COMPRESSION TYPICAL AIR COOLED APPLICATION WITH REFRIGERANT R-502

Capacity Data Based on Equal Displacement

	Evaporating Temperature		
	-30° F.	-40° F.	-50° F.
Condensing Temperature	120° F.	120° F.	120° F.
Condensing Pressure, Psig	280.3	280.3	280.3
Condensing Pressure, Psia	295	295	295
Evaporating Pressure, Psig	9.40	4.28	0.04
Evaporating Pressure, Psia	24.1	18.97	14.74
Single Stage			
Compression Ratio	12.5/1	15.6/1	20/1
Capacity, BTU/hr	46,000	32,000	23,000
BTU/watt	3.42	2.86	2.3
Two Stage with Subcooler			
Compression Ratio—Low Stage	3.74	3.68	4.05
Compression Ratio—High Stage	3.26	4.23	4.95
Capacity, BTU/hr	61,000	50,000	38,500
BTU/watt	4.15	3.84	3.35
Increase in Capacity Two Stage vs. Single Stage	32%	56%	67%

COMPRESSOR OVERHEATING AT EXCESSIVE COMPRESSION RATIOS

In addition to efficiency, the extremely high temperatures created by operation at abnormally high compression ratios makes the use of single stage compressors impractical for ultra-low temperature applications. Figure 96 shows a valve plate with carbon formation due to oil breakdown from excessive heat. Excessive cylinder temperatures can also cause rapid piston and cylinder wear, cylinder scoring, and early failure of the compressor. With two stage compressors, the interstage expansion valve maintains safe operating temperatures and this type of damage is prevented.



TYPICAL VALVE PLATE WITH CARBON FORMATION FROM OVERHEATING

Figure 96

BASIC TWO STAGE SYSTEM

The basic flow of refrigerant in a six cylinder two stage compressor is shown schematically in Figure 97. The suction gas returning from the evaporator enters the four low stage cylinders directly from the suction line. Since the discharge gas from the first stage cylinders is heated from compression, it must be cooled by the desuperheating expansion valve before entering the motor chamber. The desuperheated refrigerant vapor, now at interstage pressure, enters the high stage cylinders, is compressed, and is then discharged to the condenser.

Figure 98 is a schematic view of a typical two stage system showing the various components necessary for proper operation.

TWO STAGE SYSTEM COMPONENTS

a. Liquid Line Solenoid Valve

To prevent leakage during the off period, a solenoid valve must be placed in the liquid supply line immediately ahead of the desuperheating expansion valve. It should be wired so as to be open when the motor is running and closed when not running. A toggle switch placed in the electric line to the solenoid valve will facilitate service during pumpdown.

A 100 mesh strainer must be installed in the liquid line feeding the desuperheating valve, up stream from the solenoid valve, to protect both valves from contaminants.

b. Oil Separator

At ultra-low temperatures, the decrease in density of the refrigerant suction vapor and the increasing viscosity of the refrigeration oil make oil return during extended periods of operation or during light load periods extremely difficult. In order to minimize oil circulation and safely bridge the operating periods between defrost periods when oil will be returned, oil separators are standard on all Copeland two stage condensing units, and are strongly recommended on all two stage compressors.

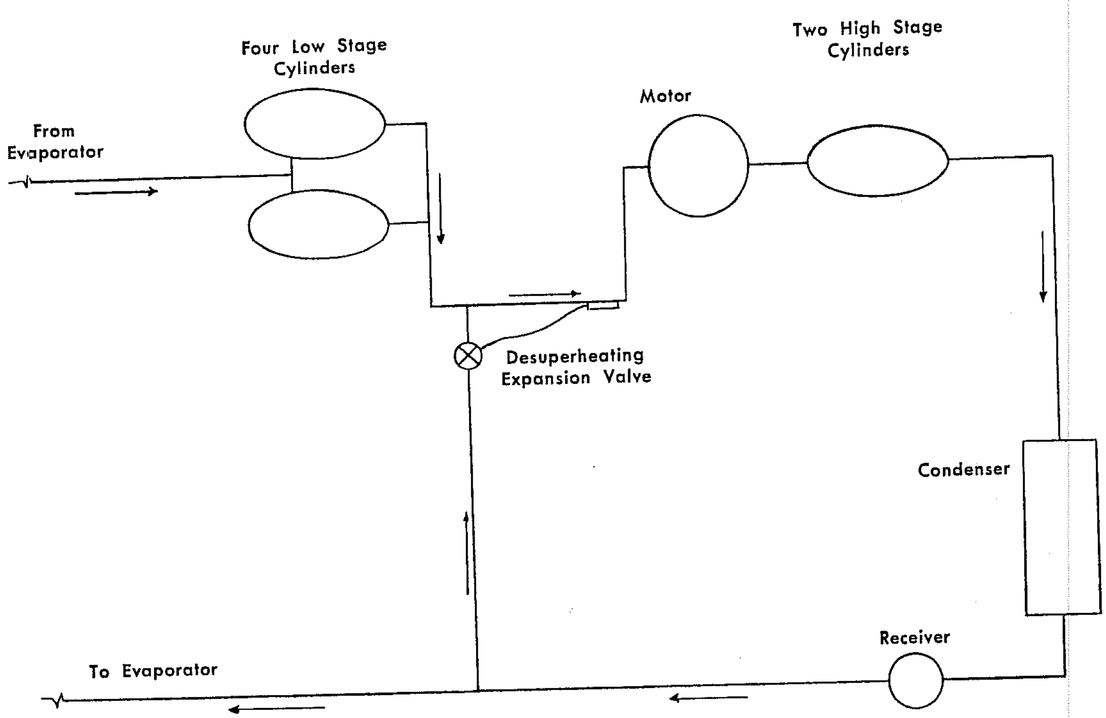
The oil separator will provide some increase in refrigerating capacity due to the increased heat transfer capability of the evaporator surface resulting from the reduced oil in circulation. It will also act as a muffler to reduce discharge pulsation and system noise transmission.

c. Suction Line Accumulator

Since the suction gas is returned directly to the low stage cylinders without going through the motor chamber, the two stage compressor is vulnerable to damage if excessive liquid floods back from the evaporator. To prevent damage from slugging, adequate suction line accumulators are mandatory on any system prone to return slugs of liquid or oil to the compressor. This may be especially critical on systems with hot gas defrost.

d. Suction Line Filter

A good suction line filter is recommended on any field installed system to prevent damage



SCHEMATIC TWO STAGE SYSTEM

Figure 97

from copper filings, solder, flux, bits of steel wool, or other contamination left in the system. The nominal cost of the filter is good insurance for the most vulnerable part of the system.

e. Liquid Sight Glass

A liquid line sight glass should be installed in the liquid line just ahead of the desuperheating expansion valve to provide a positive check for shortage of liquid refrigerant.

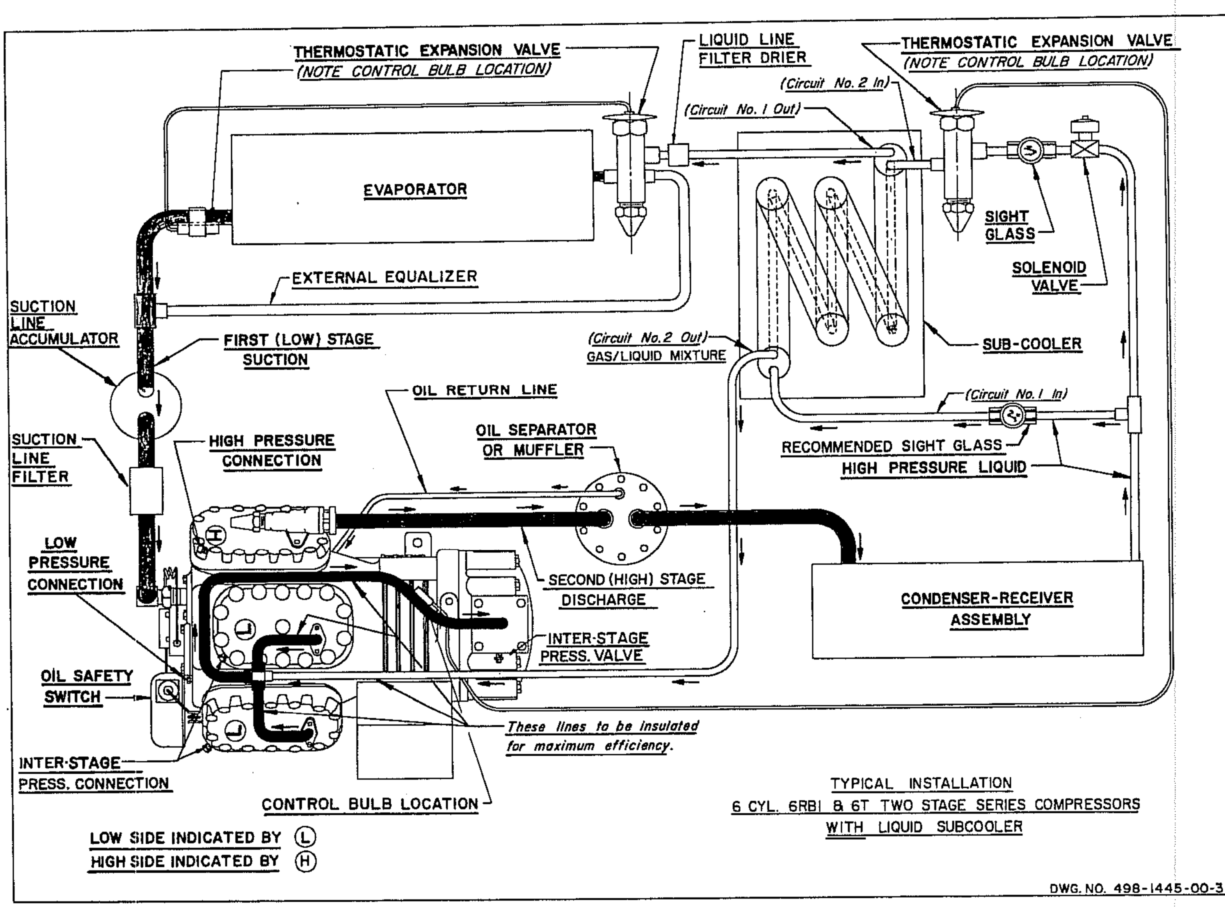
When a liquid subcooler is used, the regular liquid line sight glass (additional to the one ahead of the desuperheating expansion valve) should be installed between the receiver and the subcooler. If installed beyond the subcooler it will not be dependable since it may not show bubbles even when the system is short of refrigerant.

f. Crankcase Pressure Regulating Valve

Two stage compressors will overload if allowed to operate for extended periods with high suction pressures. The suction pressure on some systems can be limited to a satisfactory point by the size and type of evaporator used, or by the use of pressure limiting expansion valves. However the load after defrost is often the most critical, and pressure limiting expansion valves will not protect against an overload at this time. If the system can overload at any time during its operating cycle, a crankcase pressure regulating valve must be used.

g. Desuperheating Expansion Valve

Expansion valves currently supplied as original equipment with Copeland two stage compressors are of the non-adjustable superheat type. In the event of valve failure, standard field replacement valves with adjustable superheat which have been approved by Copeland



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

Figure 98

may be used. Improper valve selection can result in compressor overheating and possible damage due to improper liquid refrigerant control.

It is recommended that the branch liquid line to the desuperheating expansion valve be taken from the bottom of the main liquid line. Never install a tee with a branch off the top of the main liquid line, since this can result in improper refrigerant feed, and possible motor overheating.

h. Defrost Cycle

With electric defrost, the compressor is not running during the defrost cycle, so no special precautions other than those normally required with single stage systems are necessary.

However, motor cooling on two stage compressors is dependent on an adequate feed of liquid refrigerant from the desuperheating expansion valve. If a hot gas defrost system is used, it is imperative that a solid head of liquid is maintained at the desuperheating expansion valve at all times, and that the compressor be adequately protected against liquid refrigerant returning from the evaporator after condensation during the defrost cycle. Since hot gas defrost systems vary widely in design, it is not possible to make a general statement as to what special controls may be required. Most manufacturers have thoroughly pretested their systems, but on field installations, restrictor valves to maintain head pressure, additional refrigerant charge, suction accumulators, or other special controls may be necessary. In general, an electric defrost system is much less complicated and therefore usually more dependable on field installed two stage systems.

i. Condenser Capacity

When two stage compressors were first introduced into commercial usage for supermarkets, some users assumed that a condenser designed for a 10 HP single stage compressor would also be suitable for a 10 HP two stage unit. They failed to take into account the increased efficiency of the compressor, and apparently did not check the relative compressor capacities when selecting condensers. At -40° F. evaporating temperature, a two stage compressor may have almost twice the capacity of a low temperature single stage compressor. As a result some field problems were encountered on early two stage units because of lack of condensing capacity. This is really not a basic problem, but it is a pitfall to be aware of when working with those who are accustomed to thinking of condensing units and condensers in terms of horsepower.

j. Liquid Refrigerant Subcooler

Two stage systems may be operated either with or without liquid subcoolers. The function of the subcooler is to cool the liquid refrigerant being fed to the evaporator by the evaporation of refrigerant fed through the desuperheating expansion valve. This transfers a greater portion of the refrigeration load to the high stage cylinders, and because of the greater compressor capacity at higher suction pressures, the system capacity is greatly increased.

The temperature of the liquid refrigerant being fed to the evaporator is reduced in the subcooler to within approximately 10° F. of the interstage saturated evaporating temperature, and the increase in system capacity can only be realized if the subcooled liquid is maintained at this low temperature, and heat transfer into the liquid line is prevented. Normally this requires insulation of the liquid line.

When selecting expansion valves for two stage systems with liquid subcoolers, the designer must bear in mind that the expansion valve will have greatly increased capacity due to the low temperature of the refrigerant entering the valve. Unless this is taken into consideration, the increased refrigerating effect per pound of refrigerant may result in an oversized expansion valve with resulting erratic operation.

PIPING ON TWO STAGE SYSTEMS

The exact oil return characteristics of any system are difficult to forecast, and there is very little published data available on the design of refrigeration piping for ultra-low temperature systems. It is quite probable that any ultra-low temperature system will trap some oil during operation, even with the most conservative

Table 43

RECOMMENDED DISCHARGE LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications from -20° F. to -80° F.

Capacity BTU/hr	Without Liquid Subcooler				With Liquid Subcooler			
	Equivalent Length, Ft.				Equivalent Length, Ft.			
	25	50	100	150	25	50	100	150
6,000	3/8	1/2	1/2	1/2	3/8	3/8	1/2	1/2
9,000	1/2	5/8	5/8	5/8	1/2	1/2	1/2	5/8 *
12,000	1/2	5/8	5/8	5/8	1/2	5/8	5/8	5/8
18,000	5/8	5/8	7/8	7/8 *	1/2	5/8	5/8	5/8
24,000	5/8	7/8	7/8	7/8	5/8	5/8	7/8 *	7/8 *
36,000	7/8	7/8	7/8	1 1/8 *	5/8	7/8	7/8	7/8
48,000	7/8	7/8	1 1/8	1 1/8	7/8	7/8	7/8	7/8
60,000	7/8	1 1/8	1 1/8	1 1/8	7/8	7/8	7/8	1 1/8
72,000	7/8	1 1/8	1 1/8	1 3/8	7/8	7/8	1 1/8	1 1/8

*** Reduce vertical risers one size**

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

Table 44

RECOMMENDED LIQUID LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications from -20° F. to -80° F.

Capacity BTU/hr	Without Liquid Subcooler				With Liquid Subcooler			
	Equivalent Length, Ft.				Equivalent Length, Ft.			
	25	50	100	150	25	50	100	150
6,000	1/4	1/4	3/8	3/8	1/4	1/4	3/8	3/8
9,000	3/8	3/8	3/8	1/2	1/4	1/4	3/8	3/8
12,000	3/8	3/8	1/2	1/2	3/8	3/8	3/8	1/2
18,000	1/2	1/2	1/2	1/2	3/8	3/8	1/2	1/2
24,000	1/2	1/2	5/8	5/8	1/2	1/2	1/2	1/2
36,000	1/2	1/2	5/8	5/8	1/2	1/2	5/8	5/8
48,000	1/2	5/8	5/8	5/8	1/2	1/2	5/8	5/8
60,000	5/8	5/8	7/8	7/8	1/2	1/2	5/8	5/8
72,000	5/8	5/8	7/8	7/8	1/2	5/8	5/8	5/8

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

Table 45

RECOMMENDED SUCTION LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications

-60° F. Evaporating Temperature
Without Liquid Subcooler

Capacity BTU/hr	Equivalent Length, Ft.							
	25		50		100		150	
	H	V	H	V	H	V	H	V
6,000	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
9,000	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
12,000	1 3/8	1 1/8	1 3/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8
18,000	1 3/8	1 3/8	1 5/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
24,000	1 5/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
36,000	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
48,000	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
60,000	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
72,000	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

H - Horizontal V - Vertical

Table 46

RECOMMENDED SUCTION LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications

-60° F. Evaporating Temperature
With Liquid Subcooler

Capacity BTU/hr	Equivalent Length, Ft.							
	25		50		100		150	
	H	V	H	V	H	V	H	V
6,000	7/8	5/8	1 1/8	5/8	1 1/8	5/8	1 1/8	5/8
9,000	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
12,000	1 1/8	7/8	1 3/8	7/8	1 3/8	7/8	1 3/8	7/8
18,000	1 3/8	1 1/8	1 3/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8
24,000	1 3/8	1 3/8	1 5/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
36,000	1 5/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
48,000	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8
60,000	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
72,000	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

H - Horizontal V - Vertical

Table 47

RECOMMENDED SUCTION LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications

-80° F. Evaporating Temperature
Without Liquid Subcooler

Capacity BTU/hr	Equivalent Length, Ft. 25		50		100		150	
	H	V	H	V	H	V	H	V
6,000	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
9,000	1 3/8	7/8	1 3/8	7/8	1 3/8	7/8	1 3/8	7/8
12,000	1 5/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8
18,000	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
24,000	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
36,000	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
48,000	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
60,000	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
72,000	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

H - Horizontal V - Vertical

Table 48

RECOMMENDED SUCTION LINE SIZES FOR TWO STAGE COMPRESSORS

For R-22 and R-502 Applications

-80° F. Evaporating Temperature
With Liquid Subcooler

Capacity BTU/hr	Equivalent Length, Ft. 25		50		100		150	
	H	V	H	V	H	V	H	V
6,000	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
9,000	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
12,000	1 3/8	7/8	1 3/8	7/8	1 3/8	7/8	1 3/8	7/8
18,000	1 5/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8	1 5/8	1 1/8
24,000	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8	2 1/8	1 3/8
36,000	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
48,000	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
60,000	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
72,000	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8

Recommended sizes are applicable for two stage applications with condensing temperatures from 80° F. to 130° F.

H - Horizontal V - Vertical

piping design, and the evaporator design may have a major influence on oil circulation.

In order to minimize oil circulation and prolong the operating periods between defrost periods or other heavy load conditions which will return oil normally, oil separators are almost invariably required on two stage systems. Even with oil separators (which are never 100% efficient), it may be necessary to increase the number and frequency of defrost periods if oil is lost in the system.

Users frequently fail to realize that two stage systems with and without liquid subcooling present differences in piping design requirements. Two stage systems without liquid subcoolers are similar to single stage systems in that the liquid refrigerant temperature approaches the condensing temperature. On two stage systems with liquid subcoolers the refrigeration effect per pound of refrigerant is greatly increased because of the cold liquid refrigerant entering the evaporator. Therefore the pounds of refrigerant circulated for a given capacity will be greatly reduced, and line velocities correspondingly less. Most standard line sizing and pressure drop tables are based on liquid refrigerant temperatures equivalent to normal air and water cooled condensing temperatures, and do not apply to two stage systems with liquid subcoolers.

Tables 43 through 48 give recommended line sizes for two stage systems. Line sizing has been calculated where possible on the basis of normal single stage pressure drop criteria, but line sizes have been selected to maintain the same mass velocity flow as that which has been found to be acceptable in the normal commercial range. It is essential that the piping designer realize that no piping design on ultra-low stage systems can guarantee proper oil return, and that the requirement for an oil separator and possibly frequent defrost periods still remains, depending on the system characteristics.

CASCADE REFRIGERATION SYSTEMS

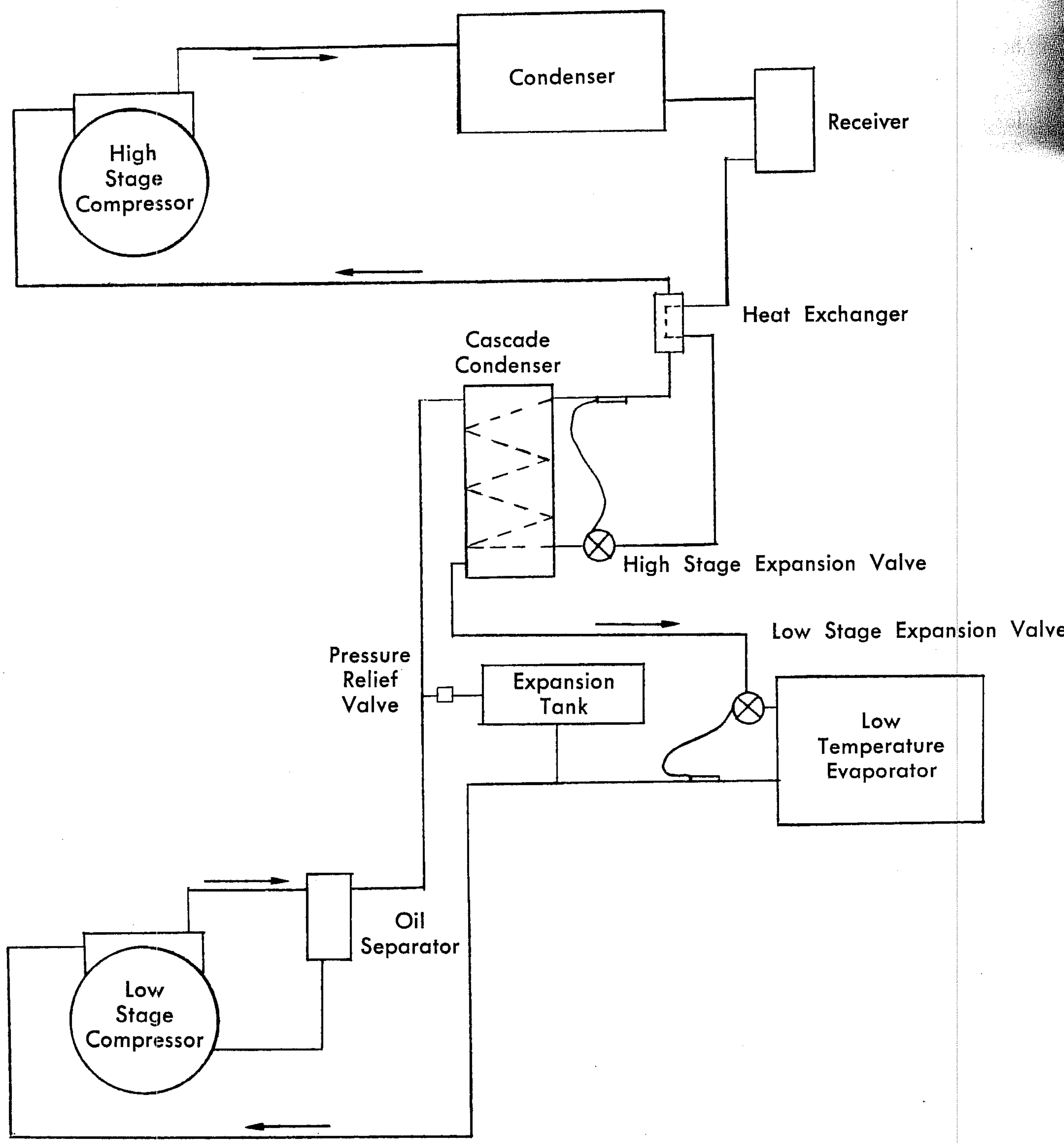
Multiple stage refrigeration can also be accomplished by using separate systems with the evaporator of the high stage serving as the condenser of the low stage by means of a heat exchanger. This type of system, termed a cascade system, is extremely flexible, and is well adapted to extremely low temperature systems, or to any system where the total compression ratio is very large. Since different refrigerants can be used in the separate systems, refrigerants with characteristics suitable for the specific application can be used. Cascade systems in multiples of two, three, or even more separate stages make possible refrigeration at almost any desired evaporating or condensing temperature.

Cascade systems have many hazards and potential problems not normally encountered in single stage refrigeration, and successful system design and application require specialized knowledge and experience.

Figure 99 is a schematic diagram of a typical cascade system consisting of two stages. The cascade condenser is basically a direct expansion heat exchanger, acting as the evaporator of the high stage and the condenser of the low stage.

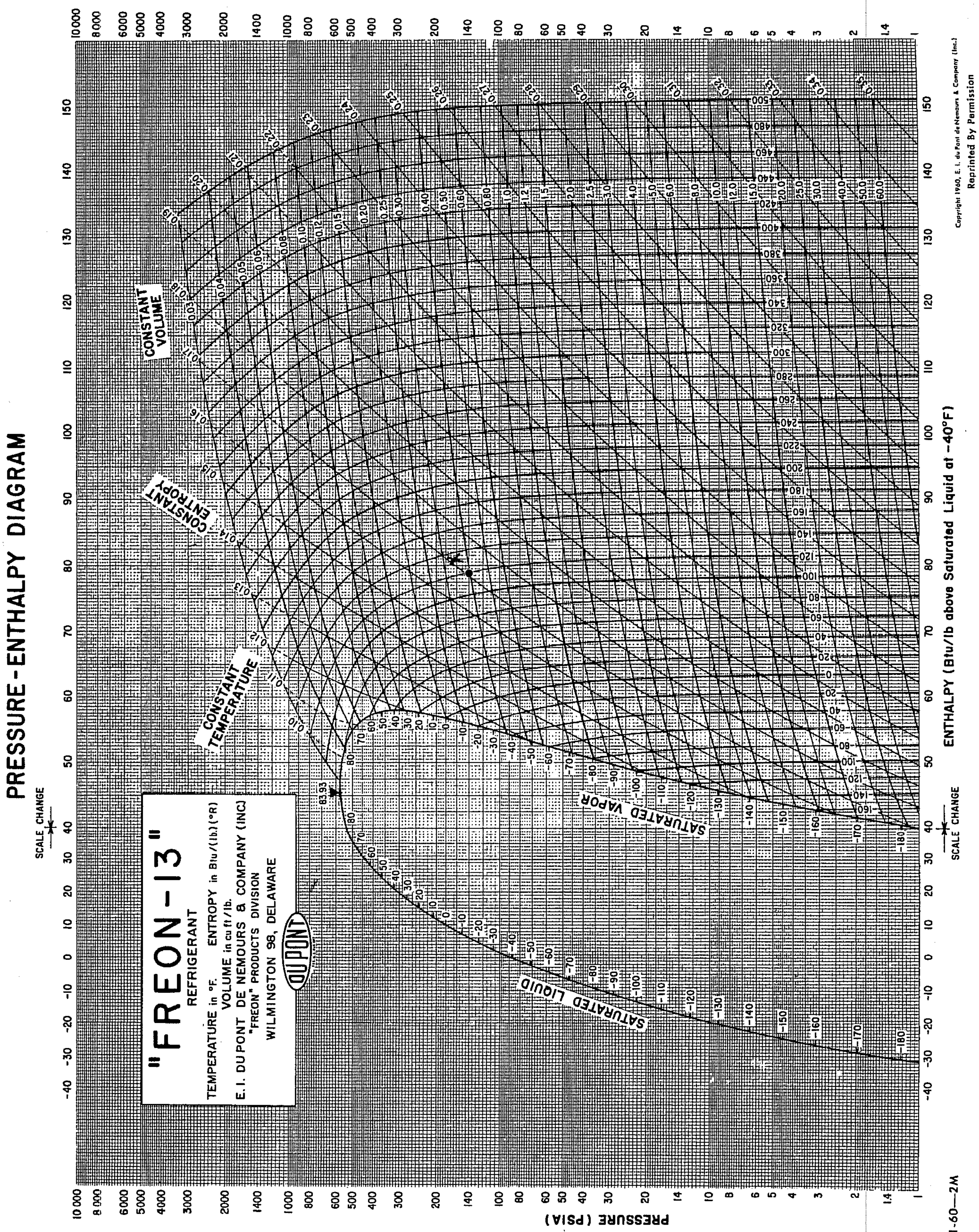
Various refrigerants can be and are used in cascade systems, with R-12, R-22, or R-502 frequently used in the high stage. The absolute pressures necessary to obtain evaporating temperatures below -80° F. with R-12, R-22, and R-502 are so low that the specific volume of the refrigerant becomes very high, and the resulting compressor displacement requirement is so great that the use of these refrigerants in the low stage becomes uneconomical. R-13, ethane, and a new refrigerant, R-23/13 (R-503) are frequently used for low stage applications.

R-13 is commonly used for evaporating temperatures in the -100° F. to -120° F. range since its pressure at those evaporating temperatures is such that its use is practical with commonly available refrigeration compressors. However the critical temperature of R-13 is 84° F. and the critical pressure is 561 psia. This means it cannot be liquified at temperatures above 84° F. regardless of pressure, and the equilibrium pressure of a mixture of gas and liquid at 84° F. is 561 psia. In order to prevent excessive pressures in the system during nonoperating periods, an expansion tank as shown



SCHEMATIC DIAGRAM — TYPICAL CASCADE SYSTEM

Figure 99



PRESSURE - ENTHALPY DIAGRAM
"FREON-13"
REFRIGERANT
TEMPERATURE in °F. ENTROPY in Btu/(lb.)(°R)
VOLUME in cu ft/lb.
E. I. DUPONT DE NEMOURS & COMPANY (INC)
"FREON" PRODUCTS DIVISION
WILMINGTON 98, DELAWARE
DU PONT
CONSTANT VOLUME
CONSTANT ENTROPY
CONSTANT TEMPERATURE
SATURATED VAPOR
SATURATED LIQUID
83.93
SCALE CHANGE
PRESSURE (PSIA)
ENTHALPY (Btu/lb above Saturated Liquid at -40°F)
Copyright 1960, E. I. du Pont de Nemours & Company (Inc.)
Reprinted By Permission
1-60—2M

Figure 100

in Figure 99 must be provided so that the entire refrigerant charge can exist as a vapor during off periods without exposing the compressor crankcase or the piping to excessive pressures. (Normally non-operating system pressures should be held to 150 psig or below).

Normally the expansion tank is located in the low pressure side of the system, with a relief valve from the high pressure side of the system discharging into the tank. The sizing of the tank is determined from the total refrigerant charge, the internal volume of the system, the maximum pressure desired, and the design ambient temperature. The specific volume of the vapor at the design storage conditions can be determined from the pressure enthalpy diagram of the refrigerant, such as shown in Figure 100. For example, at a temperature of 120° F. and a pressure of 140 psia, the specific volume of R-13 is .40 cubic feet per pound. If the system charge is 10 pounds of refrigerant, then the internal volume of the system including the expansion tank must be at least 4 cubic feet.

Because of the pressure relation of the system charge to the internal volume, the refrigerant charge in cascade systems is usually critical. Charging by means of a sight glass is unsafe. Either the exact charge must be measured into the system, or the low stage may be charged with vapor to a stabilized non-operating pressure of 150 to 175 psi in maximum ambient conditions.

The pull down load may be many times the load at design operating conditions, and some means of limiting the compressor loading during the pull down period is normally required, since it is seldom economical to size either the compressor motor or the condenser for the maximum load. Pressure limiting expansion valves or crankcase pressure regulating valves are acceptable if they are sized properly. Frequently a control system is designed to lock out the low stage system until the high stage evaporating temperature is reduced to the operating level so that excessive low stage condensing pressures do not occur on start up.

Various means of capacity control are employed, usually by means of hot gas bypass. Care must be taken to insure proper compressor motor cooling and to avoid liquid floodback to the compressor.

The cascade condensing temperature varies with individual system design. With normal high stage condensing temperatures, either air cooled or water cooled, and evaporating temperatures in the -90° F. to -140° F. range, high stage evaporating temperatures from 30° F. to -30° F. are commonly used. A difference of 10° F. to 20° F. between the high stage evaporating temperature and the low stage condensing temperature results in a reasonably sized cascade condenser. The compression ratios of the high stage and low stage should be approximately equal for maximum efficiency, but small variations will not materially affect system performance.

Copeland compressors are tested only with R-12, R-22, and R-502 in the normal commercial range, and are not tested with the refrigerants normally used in the low stage of cascade systems. Although many models of Copelametic compressors have been successfully applied for many years on cascade systems, the responsibility for the selection and application of the compressor must be that of the system designer, based on his testing, experience, and design approach.

Section 20

TRANSPORT REFRIGERATION

Truck and trailer refrigeration is an increasingly important segment of the refrigeration industry. Despite the fact that transport applications face many operating problems peculiar to their usage, there exists very little application data pertaining to this field.

Many compressor failures in transport refrigeration usage are the result of system malfunction rather than the result of mechanical wear. It is clear that substantial savings in operating cost, and tremendous improvements in unit performance and life would be possible if the causes of compressor failure could be removed. Primarily the problem boils down to one of making sure that the compressor has adequate lubrication at all times.

Part of the problem of identifying the cause of failure stems from the fact that far too few users realize that ultimate failure of a compressor resulting from lack of lubrication frequently takes place at a time when there is an adequate supply of oil in the crankcase. This is due to continued deterioration of the moving parts resulting from the original or repeated damage in the past. It is not uncommon for a damaged compressor to operate satisfactorily all winter and then fail in the spring when subjected to heavier loads.

Another source of field problems is the fact that many units are installed by personnel who may not have adequate training, equipment, or experience. Often units, particularly those in common carrier service, may be serviced in emergencies by servicemen not familiar with the unit, or indeed, with transport refrigeration generally.

Because of the installation and service hazards, it is extremely important that the unit be properly designed and applied to minimize, and if possible, prevent service problems.

COMPRESSOR COOLING

Air-cooled motor-compressors must have a sufficient quantity of air passing over the compressor body for motor cooling. Refrigerant-cooled motor-compressors are cooled adequately by the refrigerant vapor at evaporating temperatures above 0° F. saturation, but at evaporating temperatures below 0° F. additional motor cooling by means of air flow is necessary.

Normally the condenser fan if located so that it discharges on the compressor will provide satisfactory cooling. For proper cooling, the fan must discharge air directly against the compressor. The compressor cannot be adequately cooled by air pulled through a compartment in which the compressor is located. If the compressor is not located in the condenser discharge air stream, adequate air circulation must be provided by an auxiliary fan.

COMPRESSOR SPEED

Open type compressors operating from a truck engine by means of a power take-off or by a belt drive are subject to extreme speed ranges. A typical truck engine may idle at 500 RPM to 700 RPM, run at 1,800 RPM at 30 MPH, and run at 3,600 RPM to 4,000 RPM over the highway at high speeds. Whatever the power take-off or belt ratio, this means the compressor must operate through a speed ratio range of 6 to 1 or greater unless it is disconnected from the power source by some means.

The compressor speed must be kept within safe limits to avoid loss of lubrication and physical damage. Operation within the physical limitations of the compressor may be possible, for example from 400 RPM to 2,400 RPM. It may be possible to use a cut-out switch to disconnect the compressor from the power source at a given speed. The compressor manufacturer should be contacted for minimum and maximum speeds of specific compressors.

If the compressor is of the accessible-hermetic type, there is no problem concerning speed so long as the electrical source is operating at the voltage and frequency for which the motor was designed. If the speed of the generator is varied in order to obtain variable speed operation, the voltage and frequency on the normal alternating current generator will vary proportionally. Since the compressor speed and motor load will vary directly with the frequency, it is often possible to operate over a wide speed range with satisfactory results.

However, it should be born in mind that increasing the frequency and voltage of the generator above the level for which the compressor motor was designed will increase the load on the compressor, may overload the motor, and can result in bearing or other compressor damage. Operation at speeds too low to provide adequate compressor lubrication must also be avoided, although normally lubrication can be maintained on Copelametic compressors down to 600 RPM and possibly lower speeds.

Each new application involving operation of the compressor at a voltage and frequency differing from its nameplate rating should be submitted to the Copeland Application Engineering Department for approval.

One other problem that may arise with operation from a variable speed generator is the operation of electrical contactors, relays, etc. on voltages below or above their nameplate rating. Field tests have shown that the winding design and physical construction of electrical components can cause wide variation in voltage tolerance. The drop-out voltage of various types of commercially available 220 volt contactors may vary from 145 volts to 180 volts depending on construction. If it is planned to operate at variable voltage and frequencies, the electrical components which are to be used should be extensively tested at the electrical extremes in cooperation with the manufacturer to insure proper operation.

COMPRESSOR OPERATING POSITION

Occasionally compressor failures will occur due to loss of lubrication caused by parking the truck on too steep a slope. The resulting tilt of the compressor may cause the oil level to fall below the pick-up point of the oil flinger or oil pump.

Operation of the unit while the truck is parked on steep inclines should be avoided. If this is unavoidable, then consideration should be given to mounting the compressor so that oil will tend to flow to the oil pick-up point. Since this will vary on different model compressors, and the individual parking arrangement will affect the direction of the compressor pitch, each application must be considered individually.

In severe cases, consult with the compressor manufacturer.

COMPRESSOR DRIVE

Direct drive from an engine, either gasoline or diesel, to a compressor requires very careful attention to the coupling design. Alignment between the engine drive shaft and the compressor crankshaft is critical both in parallel and angular planes. Even slight angular misalignment can cause repetitive compressor crankshaft breakage. Because of the sharp impulses from the engine firing, a flexible coupling giving some resiliency is required. The coupling should be capable of compensating for slight parallel or angular misalignment and should also allow some slight endplay movement of the crankshafts. Nylon splines, neoprene bushings, and flexible disc type couplings have all been used successfully.

For a compressor driven from a power take-off by means of a shaft and two universal joints, the crosses in the U-joints must be kept parallel to each other. Where possible, the compressor rotation should be in the same direction whether on electric standby or driven from the engine.

In driving a compressor with V-belts, care must be taken to avoid excessive belt tension and belt slap. A means for easily adjusting belt tension should be provided. It may be necessary to provide an idler pulley to dampen belt movement on long belt drives. Care should be taken to mount the compressor so that the compressor shaft is parallel with the engine crankshaft.

REFRIGERANT CHARGE

Refrigerant R-12 is used in most transport systems at the present time, but R-502 is well suited for low temperature applications, and its use is increasing. Since R-502 creates a greater power requirement for a given compressor displacement than R-12, the motor-compressor must be properly selected for the refrigerant to be used. Different expansion valves are required for each refrigerant, so the refrigerants are not interchangeable in a given system and should never be mixed. Receivers for R-502 require higher maximum working pressures than those used with R-12, so normally it is not feasible to attempt to convert an existing R-12 unit for the use of R-502.

The refrigerant charge should be held to the minimum required for satisfactory operation. An abnormally high refrigerant charge will create potential problems of liquid refrigerant migration, oil slugging, and loss of compressor lubrication due to bearing washout or excessive refrigerant foaming in the crankcase.

Systems should be charged with the **minimum** amount of refrigerant necessary to insure a liquid seal ahead of the expansion valve at normal operating temperatures. For an accurate indication of refrigerant charge, a sight glass is recommended at the expansion valve inlet, and a combination sight glass and moisture indicator is essential for easy field maintenance checking. It should be born in mind that bubbles in the refrigerant sight glass can be caused by pressure drop or restrictions in the liquid line, as well as inadequate liquid subcooling. Manufacturer's published nominal working charge data should be used only as a general guide, since each installation will vary in its charge requirements.

REFRIGERANT MIGRATION

Refrigerant migration is a constant problem on transport units because of the varying temperatures to which the different parts of the system are exposed. On eutectic plate applications, liquid refrigerant will be driven from the condensing unit to the plates during the day's operation, with the threat of floodback on start-up. On both plate and blower units not in operation, the body and evaporator immediately after operation will be colder than the condensing unit, causing migration to the evaporator. During daytime hours the body and evaporator will warm up, and because of body insulation will remain much warmer than the compressor during the night hours when the ambient temperature falls, resulting in a pressure differential sufficient to drive the refrigerant to the compressor crankcase.

Excessive refrigerant in the compressor crankcase on start-up can cause slugging, bearing washout, and loss of oil from the crankcase due to foaming. Dilution of oil with excessive refrigerant results in a drastic reduction of the lubricating ability of the oil. Adequate protective measures must be taken to keep migration difficulties at a minimum. Consideration should be given to keeping the refrigerant charge as low as possible, using a pump down cycle, use of a suction accumulator, and the use of a liquid line solenoid valve.

OIL CHARGE

Compressors leaving the Copeland factory are charged with Suniso-3G, 150 viscosity refrigeration oil and no other oil should be used without specific authorization from the Copeland Application Engineering Department. The napthenic base of the Suniso-3G oil has definite advantages over paraffinic oils because of less tendency to separate from the refrigerant at reduced temperatures.

Compressors are shipped with a generous supply of oil. However, the system may require additional oil depending on the refrigerant charge and system design. After the unit stabilizes at its normal operating conditions on the initial run-in, additional oil should be added if necessary to maintain the oil level at the ¾ full level of the sight glass in the compressor crankcase. The high oil level will provide a reserve for periods of erratic oil return.

OIL PRESSURE SAFETY CONTROL

A major percentage of all compressor failures are caused by lack of proper lubrication. Only rarely is the lack of lubrication actually due to a shortage of oil in the system or failure of the oiling system. More often the source of the lubrication failure may be refrigerant floodback, oil trapping in the coils, or excessive slugging on start up.

To prevent failures from all these causes, the Copeland warranty requires that an approved manual reset type oil pressure safety control with a time delay of 120 seconds be used on all Copelametic compressors having an oil pump. The control operates on the differential between oil pump pressure and crankcase pressure, and the time delay serves to avoid shut down during short fluctuations in oil pressure during start up. A non-adjustable control is strongly recommended, but if an adjustable type control is used, it must be set to cut out at a net differential pressure of 9 psig. Oil pressure safety controls are available with alarm circuits which are energized should the oil pressure safety control open the compressor control circuit.

OIL SEPARATORS

Proper refrigerant velocities and good system design are the only cure for oil trapping problems. Oil separators are vulnerable to damage from float valve vibration, and for that reason are not commonly used on transport units. Oil separators are not normally recommended for over-the-road use on trailers, but they have been used successfully in some city operations on ice cream truck applications.

The oil separator traps a major part of the oil leaving the compressor, and since the oil is returned directly to the crankcase by means of a float valve, oil circulation in the system is minimized. On low temperature systems, oil separators may be of value in holding the amount of oil in circulation to a level which can be adequately returned to the compressor by the refrigerant in the system. However, on systems where piping design encourages oil logging in the evaporator circuit, an oil separator may only serve to delay lubrication difficulties.

The oil separator should be insulated to prevent refrigerant condensation and return of liquid to the compressor crankcase. A convenient means of returning oil to the compressor, and still providing maximum protection against liquid return is to connect the oil return line to the suction line just before the suction accumulator.

CRANKCASE PRESSURE REGULATING VALVE

In order to limit the load on the compressor, a crankcase pressure regulating valve may be necessary. During periods when the valve is throttling, it acts as a restrictor, and on start-up or during a hot gas defrost cycle, it acts as an expansion valve in the line. The preferred location for the CPR valve is ahead of the suction line accumulator. The accumulator will trap liquid refrigerant feeding back and allow it to boil off or feed the compressor at a metered rate to avoid compressor damage. However, location of the accumulator ahead of the CPR valve is acceptable if the accumulator has adequate capacity to prevent liquid floodback to the compressor.

The CPR valve should be sized for a minimum pressure drop to avoid loss of capacity, and should never be set above the published operating range of the compressor.

CONDENSER

Condenser construction must be rigid and rugged, and the fin surface should be treated for corrosion resistance unless the metal is corrosion resistant. The area in which the condenser is mounted affects its design. Condensers mounted on the skirt of a truck or beneath a trailer receive a great deal of road splash, while those mounted high on the nose of a truck or trailer are in a somewhat cleaner atmosphere. If the condenser is mounted beneath a trailer facing in the direction of travel, a mud guard should be provided. The type of tube and fin construction affects the allowable fin spacing, but in general, fin spacing of no more than 8 fins to the inch is recommended, although some manufacturers are now using fin spacing as high as 10 and 12 per inch.

Since the unit will operate for extended periods when the vehicle is parked, ram air from the movement of the vehicle cannot be considered in designing for adequate air flow, but the condenser fan should be located so that the ram air effect aids rather than opposes condenser air flow. It also should be born in mind that often many trucks or trailers will be operating side by side at a loading dock, and the air flow pattern should be such that one unit will not discharge hot air directly into the intake of the unit on the next vehicle.

Since the space available for condenser face area is limited in transport refrigeration applications, the condenser tube circuiting should be designed for maximum efficiency.

Low head pressure during cold weather can result in lubrication failure of compressors. With trucks operating or parked outside or in unheated garages in the winter months, this condition can frequently occur. A decreased pressure differential across the expansion valve will reduce the refrigerant flow, resulting in decreased refrigerant velocity and lower evaporator pressures, permitting oil to trap in the evaporator. Frequently the feed will be decreased to the point that short-cycling of the compressor results. The use of a reverse acting pressure control for cycling the condenser fan, or some other type of pressure stabilizing device to maintain reasonable head pressure is highly recommended.

RECEIVER

Because of field installation and repair, all units should be equipped either with a receiver or an adequately sized condenser so that the refrigerant charge is not critical. Valves should be provided so that the system can be pumped down. A positive liquid level indicator on the receiver will aid in preventing over-charging, and high and low test cocks have been used satisfactorily for this purpose. The size of the receiver should be held to the minimum required for safe pump down.

It is recommended that a charging valve be provided in the liquid line. While not essential, it is a fact that most servicemen will charge liquid rather than vapor into a system, and a charging valve makes this possible without damage to the compressor.

On units in operation over-the-road, powered either from the truck engine or a separate engine power source, the receiver may be subjected to temperatures higher than the condensing temperature because of heat given off by the engine. This can result in abnormally high condensing pressures because of liquid refrigerant being forced back into the condenser, excessive refrigerant charge requirements, and flashing of liquid refrigerant in the liquid line. If excessive heating of the receiver can occur, provisions should be made for ventilation of the receiver compartment with ambient air, or the receiver should be insulated.

PURGING OF AIR FROM SYSTEM

Occasionally due to improper installation or maintenance procedures, a unit will not be completely evacuated, or air will be allowed to enter the system after evacuation. The noncondensable gases will exert their own pressures in addition to refrigerant pressure, and will result in head pressures considerably above the normal condensing pressure.

Aside from the loss of capacity resulting from the higher head pressure, the presence of air in the system will greatly increase the rate of corrosion and can lead to possible carbon formation, copper plating, and/or motor failure.

As a temporary measure, it may be possible to purge refrigerant from the top of the condenser while the unit is not operating, and blow out any air trapped in the condenser. However, it is almost impossible to purge all of the air out of the compressor crankcase, and air may also trap in the receiver. If it is discovered that air has been allowed to contaminate the system, the refrigerant should be removed, and the entire unit completely evacuated with an efficient vacuum pump.

LIQUID LINE FILTER-DRIER

On all transport refrigeration systems, because of the uncertainties of installation and service, a liquid line filter-drier is essential. It is recommended that the filter-drier be oversized by at least 50% for the refrigerant charge because of the many opportunities during field maintenance for moisture to enter the system. It should have flare connections for easy replacement.

HEAT EXCHANGER

A heat exchanger should be considered mandatory on all units. It improves the performance, insures liquid refrigerant at the expansion valve, and helps assure the return of dry gas. Normally it should be located inside the refrigerated space to avoid loss of capacity, but it can be located externally if insulated.

LIQUID LINE SOLENOID VALVE

When, because of the design of the system, the refrigerant charge cannot be held to a level which can be safely handled by the compressor should refrigerant migration occur, a normally closed liquid line solenoid may be required. On 3 HP systems with refrigerant charges exceeding 15 pounds, and on 5 HP systems with refrigerant charges exceeding 20 pounds, a liquid line solenoid is recommended, and some manufacturers make liquid line solenoids mandatory on all units 1 ½ HP and larger.

The valve should be wired in parallel with the compressor so that it will be closed when the system is not in operation. It should be installed between the receiver and the expansion valve, and should have a filter-drier or strainer mounted just upstream from it in the liquid line. A soft-seated valve, of teflon or similar material, is preferred for better control during over-the-road operation.

SUCTION LINE ACCUMULATOR

A suction line accumulator is considered mandatory on all systems 2 HP and larger in size, and is recommended for all units. The purpose of the accumulator is to intercept any liquid refrigerant which might flood through the system before it reaches the compressor, particularly on start-up or on hot gas defrost cycles. Because crankcase heaters or a pumpdown cycle are not always operative on transport units, the accumulator is the best protection that can be provided for the compressor.

Provisions for positive oil return to the crankcase must be provided, but a direct gravity flow is not acceptable since this would allow liquid refrigerant to drain to the crankcase during shutdown periods. Capacity of the accumulator usually should be a minimum of 50% of the system charge, but the required size will vary with system design. Tests are recommended during the design phase of any new unit to determine the minimum capacity for proper compressor protection.

An external source of heat is desirable to accelerate the boiling of the liquid refrigerant in the accumulator so that it may return to the compressor as gas. Mounting in the condenser air stream or near the compressor will normally be satisfactory.

CRANKCASE HEATERS

Because of the interruptable power source inherent in transport refrigeration, it is difficult to insure continuous operation of the heaters. A continuous drain on the truck battery would not be acceptable.

Crankcase heaters will help when connected to a continuous power source, but cannot be relied on for complete protection against damage from liquid migration.

PUMPDOWN CYCLE

A pumpdown cycle is the best means of protecting the compressor from refrigerant damage, particularly if an excessively large charge cannot be avoided. As in the case of crankcase heaters, the fact that power may not always be available makes a pumpdown system unreliable. It is quite possible that the power to the unit might be shut off at any moment with the unit in operation and refrigerant in the coils. If pumpdown control is used, special operating precautions should be taken to insure complete pumpdown before the electric power is disconnected.

FORCED AIR EVAPORATOR COILS

Air velocities across the coil should not exceed 500-600 FPM in order to avoid blowing water from the coil onto the load. Care should be taken to insure even air distribution across the coil, since uneven air flow can cause uneven loading of the refrigerant circuits. Fin spacing exceeding 6 per inch is not recommended because of the rapid build-up of frost on the fins. However, some users and manufacturers recommend spacing as low as 3 or 4 fins per inch, while others report satisfactory experience with spacings as high as 8 per inch provided proper defrost controls are used.

Delivered air velocity should be adequate to insure good air circulation in the vehicle. Noise level is not a design limitation in a van, so velocities up to 1,500 FPM or higher can be used.

Internal volume of the refrigerant tubes should be kept to a minimum to keep the refrigerant volume as low as possible. Since pressure drop at low temperatures is critical so far as capacity is concerned, multiple refrigerant circuits with fairly short runs are preferred. Pressure drop in the evaporator should be no more than 1 to 2 psig. At the same time, it is essential that velocities of refrigerant in the evaporator be high enough to avoid oil trapping. ⅝″ evaporator tubes are acceptable, but ½″ are preferred, and ⅜″ tubing has been used successfully. Vertical headers should have a bottom outlet to allow gravity oil draining.

An evaporator face guard should be provided to protect the fins and tubing from cargo damage. Ample air inlet area should be provided, with access from both sides and the bottom if possible, to prevent blocking of air to the evaporator by cargo stacked in the vehicle.

THERMOSTATIC EXPANSION VALVES

Because of the wide range of load conditions and the premium on pulldown time in the transport field, it has been common practice for some manufacturers to oversize expansion valves used on transport units, particularly on units equipped with blower evaporator coils. If the expansion valve is oversized too greatly, surging of the refrigerant feed will result with possible floodback and erratic operation. If this occurs, a smaller valve must be used.

A liquid charged type valve is essential to retain control, since the head may frequently be colder than the sensing bulb. Vapor charged expansion valves should not be used on transport refrigeration systems.

Valve superheat should be preset by the valve manufacturer and field adjustment should be discouraged. However, valves in need of adjustment should be set to provide 5° F. to 10° F. superheat at the evaporator. Too high a superheat setting will result in starving the evaporator and poor oil return. Too low a superheat setting will permit liquid floodback to the compressor.

Pressure limiting type valves are sometimes used to limit the compressor load according to the allowable suction pressure. Since oil return to the compressor is extremely slow during the pulldown period due to the throttling action of this type of valve, MOP valves are generally not recommended for transport applications, and a crankcase pressure regulating valve is recommended if the compressor load must be limited.

It should be born in mind that the pressure across the valve affects its maximum capacity and its rate of feed. Therefore, the valve operation and the amount of superheat may be materially affected by changes of head pressure caused by changes in the ambient temperature. Some means of stabilizing head pressure is desirable to provide a uniform expansion valve feed.

DEFROST SYSTEMS

A defrost system, either electrical, reverse cycle, or hot gas, is essential for satisfactory operation of any low temperature transport unit equipped with forced air evaporators. If trucks are to be used as weekend storage containers at temperatures close to 32° F., return air as a defrosting medium may result in load temperature fluctuations.

An electrical defrost system is feasible when the unit is operating from an engine generator set or from a stationary electrical supply. The reverse cycle defrost using a four-way valve is exceedingly fast and effective, but may be sensitive to any foreign material in the system. Hot gas defrost using the heat of compression is effective only if some means of maintaining head pressure on the compressor is available, or if refrigerant condensing in the evaporator can be re-evaporated. Partial flooding of the condenser has been used, but this results in carrying a very large charge of refrigerant in the system. Some proprietary systems using heat from the engine cooling water or heat from the engine exhaust have been used with success.

Drain pan heaters are required on low temperature installations to prevent the build up of ice in the drain pan. To prevent the defrost heat from entering the cargo space, the evaporator fan should be stopped during defrost, or a damper installed in the air outlet.

Automatic start of the defrost cycle is recommended to avoid excessive accumulation of frost on the evaporator, and automatic termination should be provided to avoid returning overheated gas to the compressor. Since vibration will cause maintenance problems on time clocks, a control responsive to fan air pressure is frequently used for defrost initiation, and a temperature responsive control for defrost termination. Another method of automatic defrost control that has been used satisfactorily is a two element control sensing return air and coil temperatures, and operating on the differential between the two temperatures.

A suction accumulator is considered mandatory with any system using a hot gas or reverse cycle defrost system. The use of steam or hot water for cleaning or defrost purposes should be avoided unless a suction accumulator of adequate size is used to intercept the liquid driven out of the plates or evaporator by the heat.

THERMOSTAT

If the unit is controlled by a thermostat, a snap action type is essential to prevent chattering of the contacts. It is recommended that enclosed type switches be sealed against moisture. A calibrated adjustment with a set temperature indicator is highly desirable. The construction of the control should be such that it will withstand road shock and vibration. A liquid charged sensing bulb is desirable for fast response and accuracy of control.

HIGH-LOW PRESSURE CONTROL

A combination high and low pressure control is recommended for all systems. If a thermostat is used for unit control, and a pumpdown system is not used, a low pressure control of the manual reset type should be wired in series with the thermostat to serve as a safety cut-off in the event of loss of refrigerant charge or other abnormal conditions resulting in low suction pressures.

When used for low temperature unit operational control, the low pressure control should be provided with a low differential for accurate control. For accuracy, refrigeration gauges must be used in setting cut-in and cut-out points, since the indicator on the face of the control is not sufficiently accurate for control purposes.

Motor-compressors with single phase motors having inherent protection, 2 HP and smaller, can be operated directly on a pressure control, but larger HP compressors usually require a contactor since oil pressure safety controls require a pilot circuit, as they cannot carry the running current.

EUTECTIC PLATE APPLICATIONS

Eutectic plate applications are subject to both oil logging in the evaporator and liquid floodback to the compressor on start-up unless care is taken in system layout and installation. Since either of these conditions can result in compressor failure, adequate steps must be taken to protect the compressor.

In order to avoid trapping oil, high refrigerant velocity must be maintained through the evaporator tubing. Since the velocity is dependent on the volume of refrigerant in circulation, plates should be connected in series as required to provide an adequate refrigeration load for each expansion valve circuit.

The following table may be used as a guide in determining the minimum eutectic plate surface that must be connected to one expansion valve to insure velocities sufficient to return oil to the compressor. The recommendations are based on refrigerant evaporating temperatures 15° F. below the plate eutectic temperature, plate manufacturers' catalog data and recommendations, and a leaving gas velocity of 1,500 FPM. For easy field calculation, the eutectic plate surface shown is for one side of the plate only, e.g. a 24" x 60" plate would have 10 square feet of surface.

Recommended Plate Surface For Each Expansion Valve Circuit

Tubing Diameter	Low Temperature Plates Below 0° F. Eutectic		Medium Temperature Plates Above 0° F. Eutectic	
	Minimum	Maximum	Minimum	Maximum
⅝" O.D.	12 sq. ft.	32 sq. ft.	15 sq. ft.	32 sq. ft.
¾" O.D.	17 sq. ft.	40 sq. ft.	22 sq. ft.	40 sq. ft.
⅞" O.D.	35 sq. ft.	50 sq. ft.	40 sq. ft.	50 sq. ft.

Basically the circuiting and valving of a truck plate system should be designed so that velocities in each refrigeration circuit will be above a given minimum (for adequate oil return) and below a given maximum (for a pressure drop that does not cause excessive capacity penalty). It is recommended that circuits approaching the maximum should be used whenever possible.

For example, if in a given truck for low temperature use, plates with below 0° F. eutectic solution were used, circuits might be selected as follows:

Given:

2 - 24" x 120" plates @ 20 sq. ft. each
2 - 24" x 60" plates @ 10 sq. ft. each
1 - 30" x 60" plate @ 12.5 sq. ft.

⅝" O.D. Tubing

Circuit		
A	1 - 24" x 120" plate	20 sq. ft.
B	1 - 24" x 120" plate	20 sq. ft.
C-Series	(1 - 30" x 60" plate (2 - 24" x 60" plates	12.5 sq. ft. 20 sq. ft.

¾" O.D. Tubing

Circuit		
A-Series	(1 - 30" x 60" plate (2 - 24" x 60" plates	12.5 sq. ft. 20 sq. ft.
B-Series	2 - 24" x 120" plates	40 sq. ft.

⅞" O.D. Tubing

Circuit		
A-Series	(1 - 30" x 60" plate (2 - 24" x 60" plates	12.5 sq. ft. 20 sq. ft.
B-Series	2 - 24" x 120" plates	40 sq. ft.

Normally the eutectic plates are selected by the system designer for the particular truck and application requirement. In order to keep the refrigerant charge within acceptable limits, it is important that both the total number of plates and the plate internal refrigerant volume be kept to an absolute minimum required to accomplish the desired refrigeration.

Because of the large refrigerant charge required for plates, and the variable nature of the load imposed on the compressor, plate circuits are subject to extreme variations in refrigerant velocity. It has been our experience that proper velocities are of much greater importance than low pressure drop in determining the heat transfer rate between the refrigerant and the eutectic solution. Many users, following normal commercial refrigeration practice where it is assumed that refrigerant charges are low and velocities are consistently high, have placed an undue importance on low pressure drop in selecting and circuiting plates, and as a result have unknowingly created lubrication problems in their systems while gaining little or nothing in capacity performance. In many instances capacity has actually been reduced due to loss of proper refrigerant control.

A common misconception is that the use of separate expansion valves on each plate will give increased capacity and more rapid pulldown. This is not necessarily so. The use of more expansion valves will result in a lower pressure drop through the refrigerant circuit which might aid capacity slightly, but in most cases the resulting improper control actually decreases capacity.

On two plates, for example, the use of two expansion valves would result in two sections of tubing being used as drier area in order to obtain the necessary superheat for proper operation of the expansion valves. If only one expansion valve were used, only one length of tubing for this superheating function would be required, and the effective refrigeration area would be increased. The use of one expansion valve on multiple plates results in a much higher velocity, and as a result the scrubbing action of the refrigerant on the walls of the tube causes a much higher rate of heat transfer. Our experience would indicate, particularly at low evaporating temperatures, that very possibly multiple plates operating on one expansion valve will have more capacity and a better pulldown than the same plates operating with individual expansion valves.

A similar misconception is that the use of larger O.D. tubing in plates will result in a lower pressure drop and therefore increase capacity. As in the case of expansion valves, the use of smaller tubing, although possibly resulting in a slightly higher pressure drop, will greatly increase refrigerant velocity, increase the heat transfer rate as a result, and again our experience indicates on low temperature plates that capacity may actually be increased because of the smaller tubing. The smaller tubing requires a smaller refrigerant charge, and therefore also decreases the problem of refrigerant migration.

Expansion valves on plate circuits should be no larger than 1 ton size, and ½ ton valves will give better control on smaller circuits in the medium temperature range. The piping and thermal sensing bulbs should be located so that each valve operates independently and is not influenced by the return line controlled by another valve.

Field experience indicates that due to the throttling action of an MOP valve after shutdown or defrost periods, oil may not be returned to the crankcase at a fast enough rate to maintain compressor lubrication in the event oil is lost from the compressor on start-up due to liquid refrigerant foaming in the crankcase. Therefore, pressure limiting type expansion valves are not recommended for plate circuits.

Because of the amount of oil trapped in the plates during operation, additional oil normally must be added to the compressor during the initial pulldown cycle, or after the unit reaches its normal operating conditions. Sufficient oil should be added to maintain the oil level at approximately the ¾ full level of the compressor oil sight glass.

As the eutectic solution becomes frozen, the boiling action of the refrigerant slows, and a higher percentage of liquid refrigerant lies in the bottom of the evaporator tubing. When the unit cycles off, or the power is disconnected, the plates may be partially filled with liquid refrigerant and oil. At some later time when the compressor is again started, the liquid will flood back to the compressor.

To protect against liquid floodback, a suction accumulator is mandatory on units of 2 HP and larger, and is recommended on all transport units. If a crankcase pressure regulating valve is used, the accumulator should be located if possible between the CPR valve and the compressor in order to provide the maximum protection.

A liquid line solenoid valve can be helpful in minimizing migration from the condenser and receiver to the evaporator and compressor during periods when the unit is not in operation. If the system refrigerant charge is not excessive, a liquid line solenoid may not be required, but some manufacturers feel they should be mandatory on all plate systems 1½ HP in size and larger.

All plate applications should be equipped with the following:

a. Properly sized expansion valves.

b. A liquid to suction heat exchanger for maximum efficiency.

c. A liquid line filter-drier.

d. A combination sight glass and moisture indicator for easy maintenance.

e. An oil pressure safety control on all compressors having oil pumps.

f. A reverse acting pressure control to stop the condenser fan in order to maintain satisfactory compressor head pressure during cold weather operation.

g. Suction line accumulator (2 HP and larger).

One of the major problems in low temperature eutectic plate applications is the practice of the operator or serviceman of reducing the low pressure cut-out below the operating limits of the refrigeration system, possibly to such a low setting that the resulting refrigerant velocities are too low to return oil to the compressor. This practice has been stimulated by the demand for lower and lower ice cream temperatures, and the serviceman often fails to realize the hazard he is creating. The increased compression ratio is not a problem in a properly designed compressor so long as adequate lubrication is maintained. But once the eutectic solution is frozen, the decrease in evaporator load causes the compressor suction pressure to drop rapidly, and at extremely low suction pressures, compressor capacity falls off rapidly. From -25° F. to -40° F. the capacity may decrease by 50% in the best R-12 low temperature compressor, and from -25° F. to -50° F. the reduction in capacity may be as high as 75%. As a result, there may no longer be adequate refrigerant velocity in the evaporator circuit to return oil to the crankcase. At such low capacities, the expansion valve may no longer be able to properly control the liquid refrigerant feed.

Repeated extended periods of operation below the operating range of the system are almost certain to result in eventual compressor failure. There is no cure for this situation except adequate education of the user. To provide proper protection for the compressor the low pressure control should be set to cut out at approximately 10° F. below the normal evaporating temperature.

For example, a system equipped with plates containing -8° F. or -9° F. eutectic solution will normally operate with a refrigerant evaporating temperature of approximately -25° F., and the low pressure cut-out should be set at the equivalent of -35° F. or 8″ of vacuum on R-12 refrigerant, and 5 psig on R-502. If the system is controlled by a thermostat sensing the truck air temperature, the thermostat should be set no lower than the plate eutectic temperature.

The user must realize that a compressor's application is limited by the rest of the system. Because of the inherent problems of oil return presented by the shape and mounting characteristics necessitated in truck applications, and the large amounts of tubing which must be used in plate construction, the minimum satisfactory evaporating temperature for both R-12 and R-502 is approximately -40° F. **Low pressure controls on all plate systems must be set to cut out at or above the equivalent pressure setting; for R-12, 11″ of vacuum, and for R-502, 5 psig.**

In order to maintain the evaporating temperature within acceptable limits, it is essential that the combination of condensing unit and plates be properly balanced. The selection of too small a condensing unit may result in a freezing rate that is too slow. But of equal and possibly greater importance, the selection of too large a condensing unit may result in an excessively large temperature difference between the plate eutectic temperature and the refrigerant evaporating temperature. This condition most frequently occurs when a large condensing unit is selected in order to achieve a quick pulldown, or to shorten the time necessary to freeze the eutectic solution. Since the minimum satisfactory evaporating temperature is approximately -40° F., the condensing unit should be selected so that the normal operating evaporating temperature on low temperature plates is not below -30° F. to -35° F.

REFRIGERANT PIPING

Normal good piping practice should be followed in installing refrigerant lines for split systems. A silver solder alloy should be used for making connections to the compressor and for long runs of tubing where vibration may be a problem, and a high temperature silver solder alloy only must be used on compressor discharge lines. For other connections, 95/5 solder is acceptable, and makes possible easier field repair. 50/50 solder should not be used since it does not have sufficient strength for transport usage. Acid core type solder should not be used.

The suction line should be sized to keep refrigerant velocities above 700 FPM for horizontal runs, and 1,500 FPM for vertical risers at the lowest expected capacity.

VIBRATION

The greatest single hazard of transport refrigeration usage is damage from vibration and shock. Although shock tests on the nose of trailers have recorded very high shock levels, the great majority of all failures from this source are due to the cumulative effect of small vibrations. Any line, capillary tube, or structural member that is subjected to continuous sharp vibration, or that rattles against a neighboring member in operation is almost certain to fail within a fairly short period of time. It cannot be stressed too strongly that normal commercial construction of condensing units and evaporators for the usual commercial application is not adequate for over the road usage.

Copeland manufactures a line of condensing units specially designed for transport usage. The frames are ruggedly constructed, and all components are mounted to minimize vibration.

When compressors are installed in a system manufacturer's condensing unit, care must be taken to see that the compressor is bolted down firmly. Neoprene or other resilient shock mounts may be used, but spring mounting is not acceptable. Internally spring mounted compressors are not suitable for transport applications due to the danger of internal damage from severe shocks, and continuous spring movement.

Vibration eliminators should be mounted in the compressor discharge and suction lines. A very common fault is the installation of a vibration absorber between two sections of rigid piping, in which case the vibration absorber may be as rigid as the piping. Metal vibration eliminators should never be mounted in such a fashion that they are subjected to stress in either compression or extension. An improperly installed vibration eliminator can actually cause line failure. Flexible refrigerant lines such as Aeroquip, Stratoflex, or Anchor which are specifically designed for use with the appropriate refrigerant may be used in place of metallic vibration absorbers. Metallic vibration absorbers should have joints adequately sealed to prevent condensation from freezing and damaging the joints.

Welding is preferable to bolting in fastening structural members. Sheet metal screws, and other metal fasteners not securely held by lock washers or lock nuts are not dependable. All wiring and piping should be protected with grommets where passing through sheet metal holes.

Evaporator and condenser tube sheets, when used for mounting, should be of solid, one piece construction, and may require heavier gauge construction than used in normal commercial practice for strength purposes. Coil tube sheets should be manufactured with collars, as raw edge holes can cut the tubing due to vibration.

ELECTRICAL PRECAUTIONS

Electrical failures are a common field maintenance problem due to the wet environment, shock and vibration, and the possibility of improper power from an engine generator set.

For the safety of operating and maintenance personnel, **the electrical system should be grounded to the frame, and the frame in turn grounded by means of a chain or metal link to the ground if a generator set is mounted on the vehicle.** All component's should be grounded from one to the other, such as the generator set to condensing section to evaporator section. **Cables to remote sources of power should carry an extra wire for grounding purposes at the supply plug.**

At the time of manufacture, each system should be given a high potential test to insure against electrical flaws in the wiring. All relays and terminals should be protected against the weather, and all wiring should be covered with protective loom to guard against abrasion. All switches should be of the sealed type, recommended by the manufacturer for use in wet environments. Plug type line connectors should be of the waterproof type. Electrical cables connecting split units should have a watertight cable cover, or should be run in conduit. All wiring should be fastened securely to prevent chafing, and should be clearly identified by wire marking and/or following the color code specified by the National Electrical Code.

Adequately sized extension cords, plugs, and receptacles must be used to avoid excessive voltage drop. Voltage at the compressor terminals must be within 10% of the nameplate rating, even under starting conditions. Many single phase starting problems on small delivery trucks can be traced to the fact that power is supplied to the compressor from household type wiring circuits through long extension cords, neither of which are sized properly for the electrical load. Single phase open type motors which are used for belt driving a compressor during over-the-road operation must be equipped with a relay to break the capacitor circuit, rather than a centrifugal switch. The variable speed operation experienced during truck operation may cause a centrifugal switch to fail because of excessive wear at low operating speeds. All start capacitors must be equipped with bleed resistors to permit the capacitor charge to bleed off rapidly, preventing arcing and overheating of the relay contacts.

When units are operated from several power sources, be sure all plugs and receptacles are wired in the same sequence, so that the compressor rotation will not be reversed.

INSTALLATION

A large number of field failures that now occur could be prevented by proper installation practice. To assure trouble free operation, every effort should be made to carry out the following minimum procedures.

1. Read the manufacturer's instructions.

2. Be sure that structural or reinforced members are provided to mount the units.

3. Thoroughly clean all copper lines before assembling. Do not use steel wool for cleaning since the metal slivers may cause electrical problems in the compressor. If the tubing is not precleaned and capped, pull a rag saturated with refrigerant oil through the tube and blow out with nitrogen prior to connecting lines to the evaporator and condenser.

4. Use only a suitable silver solder alloy or 95/5 solder in making soldered joints.

5. When brazing lines, circulate inert gas such as dry nitrogen through the line to prevent oxidation.

6. Install piping in the wall or floor of the vehicle, or provide an adequate guard.

7. After the lines are installed, pressurize to 150 psig, and leak test. The use of an

electronic leak detector is recommended for greater sensitivity. As a final check, the system should be sealed for 12 hours after pulling a deep vacuum. If the vacuum will not hold, the system should be rechecked for leaks, repaired, and retested to insure that it is ready for evacuation and charging.

8. Use a good high vacuum pump to evacuate the system and leave the pump on the system for a minimum of 4 hours. Evacuate to less than 1,500 microns, and break the vacuum with refrigerant to 5 psig. Repeat the evacuation process, and break with refrigerant as before. Evacuate a final time to 500 microns or less and the system is ready for charging.

   WARNING: To prevent motor damage do not use the motor-compressor to evacuate the system. A motor-compressor should never be started or operated while the system is under a deep vacuum, or serious damage may result because of the reduced dielectric strength of the atmosphere within the motor chamber.

9. Charge the unit with refrigerant, either vapor through the suction valve, or preferably liquid through a liquid line charging valve if provided. **The compressor must never be charged with liquid refrigerant through the suction side.**

10. If using an engine-generator as a power source, start the engine and check the generator output voltage to be sure it is correct.

11. Check the voltage at the compressor terminals, start the unit, check the amperage draw of the compressor, and the rotation of the fans to be sure the unit is phased properly.

12. Observe the discharge and suction pressures. If an abnormal pressure develops, stop the unit immediately and check to see what is causing the difficulty. Take corrective action if required.

13. Observe the refrigerant oil level and check the oil pressure, if the compressor is equipped with a positive displacement oil pump. If the oil level becomes dangerously low during the pulldown period, add oil to the compressor. After the unit reaches normal operating conditions, add oil if necessary to bring the level to a point ¾ full in the crankcase sight glass.

14. Check all manual and automatic controls.

15. After a minimum of two hours of operation, make another leak test.

16. After the unit has reached the proper operating conditions, and all controls have been checked, run the unit overnight on automatic control to be sure operation is satisfactory. Check oil level in the compressor, and add oil if necessary.

17. When unit is delivered to the customer, be sure that operating personnel have proper written instructions on operating and maintenance procedures. The responsible sales personnel should verbally explain the operation of the unit to the user, and wiring diagrams and operating instructions should be permanently carried on the vehicle, either by means of a decal or in an envelope properly protected from loss or damage.

FIELD TROUBLESHOOTING ON TRANSPORT UNITS

The great majority of all low temperature compressor failures in transport refrigeration can be traced to lubrication problems. No compressor can operate satisfactorily unless oil logging and liquid floodback can either be prevented or safely controlled by safeguard devices in the refrigeration system.

The following check-off list covers possible corrective action on units experiencing field difficulties. For a more detailed discussion of each item, refer to the appropriate section in this manual. The need for any particular modification would of course depend on the individual application.

1. **Eutectic plate circuiting for high refrigerant velocity**

   It is essential for proper oil return to the compressor that high refrigerant velocities be maintained through the evaporator circuits. For a rough rule of thumb on plates with ⅞″ or ¾″ O.D. tubing, there should be no less than 3 small or 2 large plates in series on one expansion valve. On plates with ⅝″ O.D. tubing, there should be no less than 2 small or one large plate on one expansion valve.

2. **Expansion valves on eutectic plates**

   Expansion valves should be no larger than 1 ton capacity in size, liquid or cross charged, and internally equalized.

3. **Refrigerant Charge**

   The refrigerant charge must be held to a minimum to avoid refrigerant migration problems. Use a sight glass to check for a liquid seal at the expansion valve at low temperature operating conditions.

4. **Liquid Line Solenoid Valve**

   If excessive refrigerant migration to eutectic plates is occurring during over-the-road operation, a liquid line solenoid valve may be required to properly control large refrigerant charges.

5. **Suction Line Accumulator**

   A suction line accumulator is the best protection that can be provided to guard against liquid floodback. **It should be mandatory on all truck applications 2 HP and larger.** For maximum efficiency, it should be installed close to the compressor, and if a CPR valve is used, between the compressor and the CPR valve. The accumulator must have provisions for positive oil return.

6. **Head Pressure Control**

   In winter operation, head pressures may drop so low that inadequate feeding of the expansion valve may result, and the evaporator may be starved. A reverse acting high pressure control should be used to cycle the condenser fan if head pressures drop below 80 psig on R-12 operation, or 125 psig on R-502 operation, unless other acceptable means of controlling head pressure are provided.

7. **Oil Level in Crankcase**

   When compressors without oil pumps are used on truck applications, the oil level should be maintained high in the compressor sight glass to assure a reserve of lubricating oil for periods of erratic oil return. The user should be warned that the compressor may not be getting adequate lubrication if the oil level drops below the bottom of the sight glass. Only Suniso 3G oil should be used which has a viscosity of 150, a pour point of -35° F. and a floc point of -70° F. This oil has proven satisfactory for all low temperature applications.

8. **Oil Pressure Safety Control**

   On all compressors having positive displacement type oil pumps, an oil pressure safety control is required.

9. **High Pressure Cut-Out**

   Several manufacturers have produced units with no high pressure control. Failure of the condenser fan motor may result in excessive head pressures, and subsequent compressor failure. A high pressure control is essential.

10. **Liquid Line Filter-Drier and Heat Exchanger**

    These should be standard on all units.

11. **Low Pressure Control Setting**

    A major educational effort is required to point out to the user the dangers of bypassing the low pressure cut-out, or setting it at dangerously low levels.

    When eutectic plates are completely frozen, the compressor suction pressure falls very rapidly, with a consequent sharp drop in compressor capacity, and resulting lubrication difficulties, since velocity in the plates may no longer be sufficient to return oil to the compressor. Users, particularly ice cream distributors, frequently try to reduce the van body temperature to the lowest temperature possible as an added safety factor for the day's operation.

Since the system is normally not designed for the extremely low evaporating temperatures at which the compressor can operate under such conditions, the compressor pays the penalty. The user must realize that a compressor's application is limited by the rest of the system. Because of the inherent problems of oil return presented by the shape and mounting characteristics necessitated in truck applications, and the large amounts of tubing which must be used in plate construction, the minimum satisfactory evaporating temperature for both R-12 and R-502 is approximately -40° F. **Low pressure controls on transport systems must be set to cut out at or above 11″ of vacuum on R-12, and 5 psig on R-502.**

12. **Location of Truck while System is Operating**

Trucks must be parked on a reasonably level surface while the refrigeration unit is in operation. Short periods of operation on an incline such as experienced in over-the-road operation are not a problem, but long periods of operation while the truck is parked on a steep incline or on the side of a hill may rob the compressor of lubrication if the oil level flows away from the pick up point of the oil flinger or oil pump.

Section 21

CAPACITY CONTROL

On many refrigeration and air conditioning systems, the refrigeration load will vary over a wide range. This may be due to differences in product load, ambient temperature, usage, occupancy, or other factors. In such cases compressor capacity control is a necessity for satisfactory system performance.

The simplest form of capacity control is "on-off" operation of the compressor. This works acceptably with small compressors, but for larger compressors, it is seldom satisfactory, because of fluctuations in the controlled temperature. Under light load conditions it can result in compressor short cycling. On refrigeration applications where ice formation is not a problem, users frequently reduce the low pressure cut-out setting to a point beyond the design limits of the system in order to prevent short cycling. As a result, the compressor may operate for long periods at extremely low evaporating temperatures. Since the compressor capacity decreases rapidly with a reduction in suction pressure, the reduced refrigerant density and velocity frequently is inadequate to return oil to the compressor. Operation of the system at temperatures below those for which it was designed may also lead to overheating of the motor-compressor. Both of these conditions can cause compressor damage and ultimate failure.

In order to provide a means of changing compressor capacity under fluctuating load conditions, larger compressors are frequently equipped with cylinder unloaders. If compressors with unloaders are not available or cannot provide the capacity modulation required, hot gas bypass may be utilized.

COMPRESSORS WITH UNLOADERS

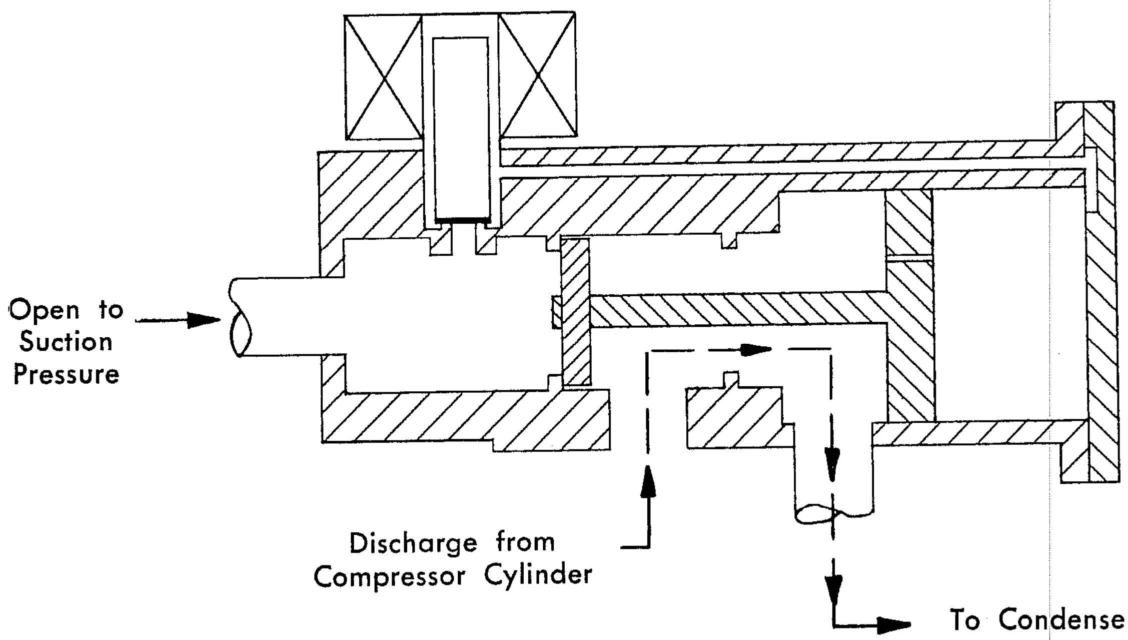
Unloaders on reciprocating compressors are of two general types. On some, suction valves on one or more cylinders are held open by some mechanical means in response to a pressure control device. With the suction valves open, refrigerant vapor is forced back into the suction chamber rather than the discharge chamber during the compression stroke, and the cylinder performs no pumping action.

Copelametic compressors with unloaders have a bypass valve so arranged that the unloaded cylinder or cylinders are isolated from the discharge pressure created by the loaded cylinders. A schematic view of a capacity control valve in the loaded and unloaded positions is shown in Figure 101. The three-way valve connects the discharge ports of the cylinder either to the normal discharge line when loaded, or to the compressor suction chamber when unloaded. Since the piston and cylinder do no work when in the unloaded cycle other than pumping vapor through the bypass circuit, and handle only suction vapor, the problem of cylinder overheating while unloaded is practically eliminated. At the same time, the power consumption of the compressor motor is greatly reduced because of the reduction in work performed. The reduced power consumption and better temperature characteristics of this type of unloading when properly applied are major advantages over external hot gas bypass unloading where all cylinders of the compressor are working against condensing pressure.

Because of the decreased volume of suction vapor returning to the compressor from the system and available for motor cooling, the operating range of unloaded compressors must be restricted. In general, Copeland compressors with capacity control are recommended only for high temperature applications, but in some instances they can be satisfactorily applied in the medium temperature range. Because of the danger of overheating the compressor motor on low temperature systems, either cycling the compressor or hot gas bypass is recommended.

COMPRESSOR UNLOADING VALVE

DE-ENERGIZED



ENERGIZED

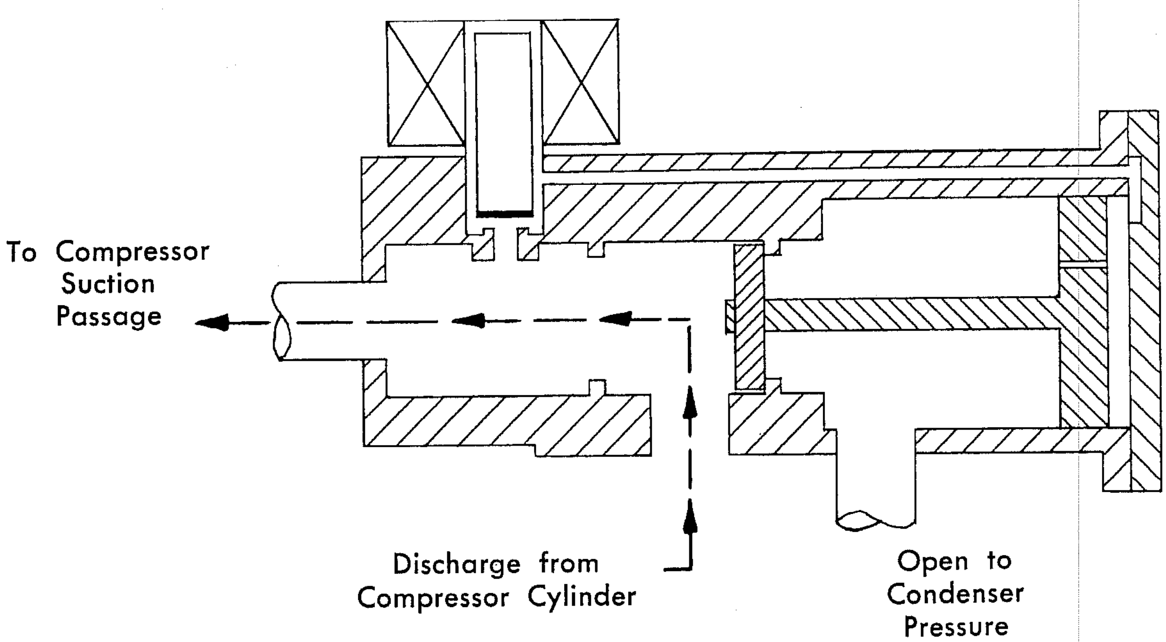


Figure 101

HOT GAS BYPASS

Compressor capacity modulation by means of hot gas bypass is recommended where normal compressor cycling or the use of unloaders may not be satisfactory. Basically this is a system of bypassing the condenser with compressor discharge gas to prevent the compressor suction pressure from falling below a desired setting.

All hot gas bypass valves operate on a similar principle. They open in response to a decrease in downstream pressure, and modulate from fully open to fully closed over a given range. Introduction of the hot, high pressure gas into the low pressure side of the system at a metered rate prevents the compressor from lowering the suction pressure further.

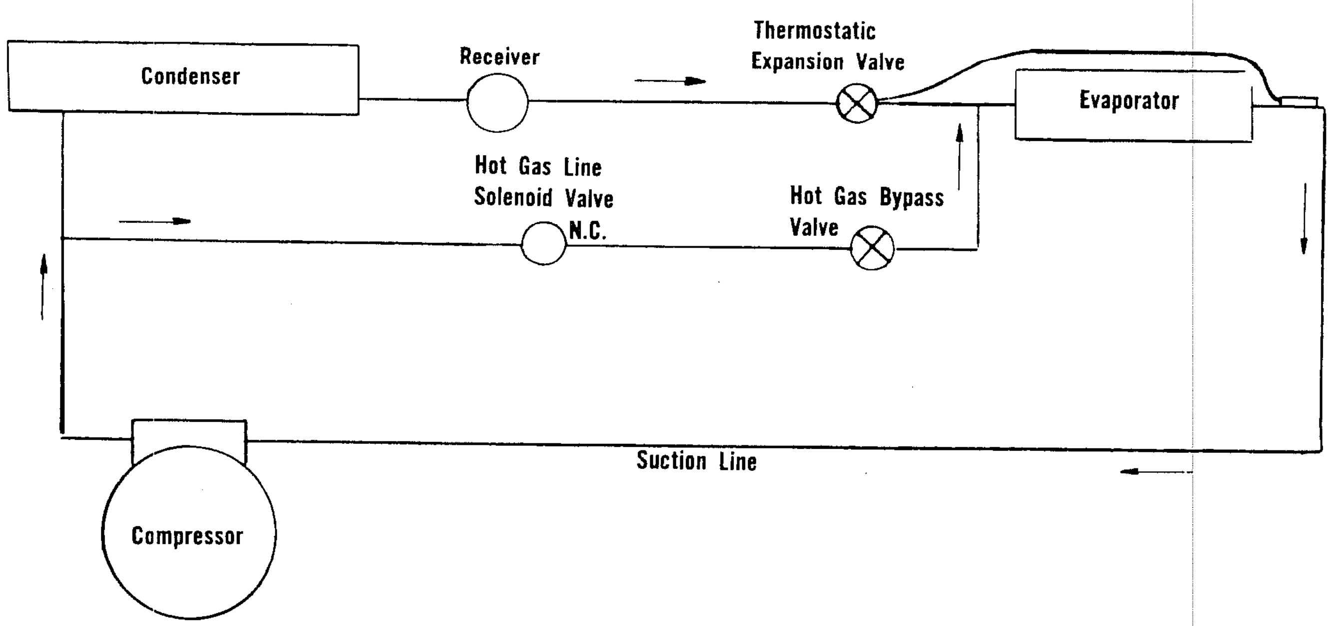
The control setting of the valve can be varied over a wide range by means of an adjusting screw. Because of the reduced power consumption at lower suction pressures, the hot gas valve should be adjusted to bypass at the minimum suction pressure within the compressor's operating limits which will result in acceptable system performance.

If a refrigeration system is properly designed and installed, field experience indicates that maintenance may be greatly reduced if the compressor operates continuously within the system's design limitations as opposed to frequent cycling. Electrical problems are minimized, compressor lubrication is improved, and liquid refrigerant migration is avoided.

Therefore, on systems with multiple evaporators where the refrigeration load is continuous, but may vary over a wide range, hot gas bypass may not only provide a convenient means of capacity control, it may also result in more satisfactory and more economical operation.

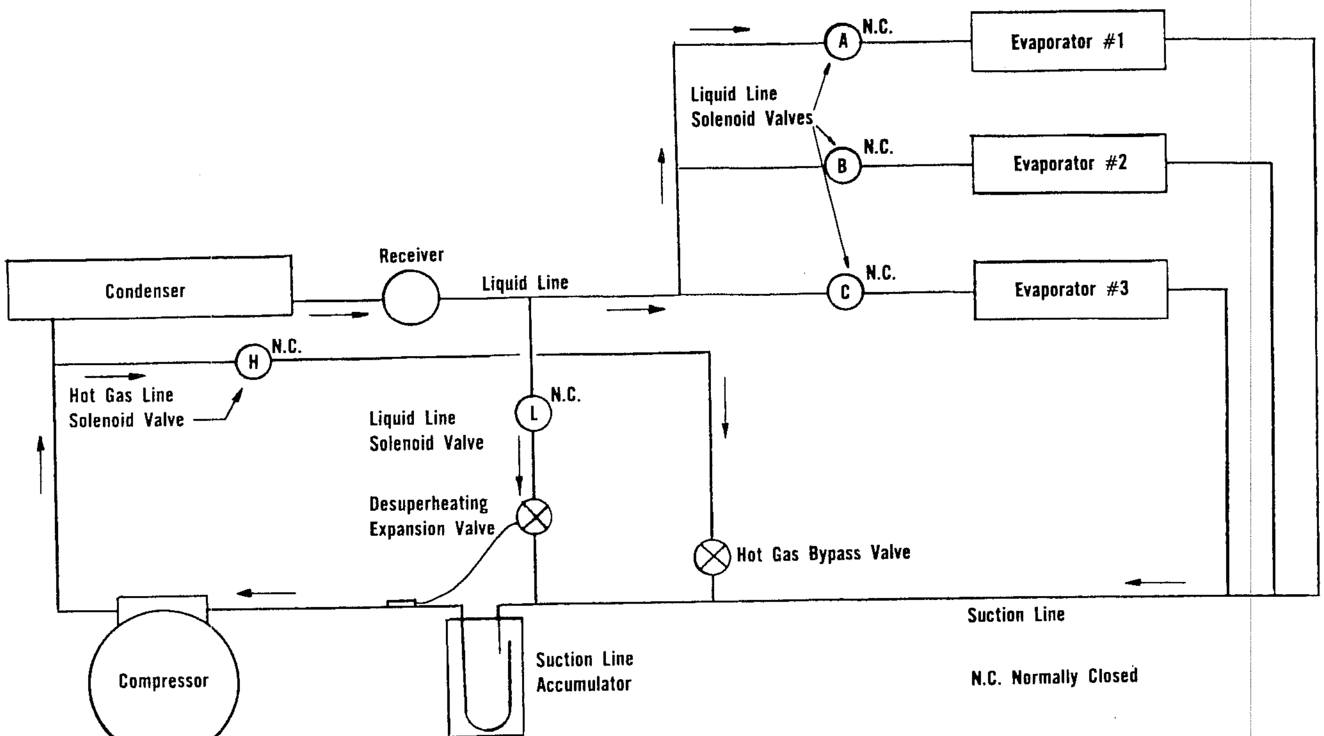
BYPASS INTO EVAPORATOR INLET

On single evaporator, close connected systems, it is frequently possible to introduce the hot gas into the evaporator inlet immediately after the expansion valve. Distributors are available with side openings for hot gas inlet. Bypassing at the evaporator inlet has the effect of creating an artificial cooling load. Since the regular system thermostatic expansion valve will meter its feed as required to maintain its superheat setting, the refrigerant gas returns to the compressor at normal operating temperatures, and no motor heating problem is involved. High velocities are maintained in the evaporator, so oil return is aided. Because of these advantages, this type of control is the simplest, least costly and most satisfactory bypass system. This type of bypass is illustrated in Figure 102.

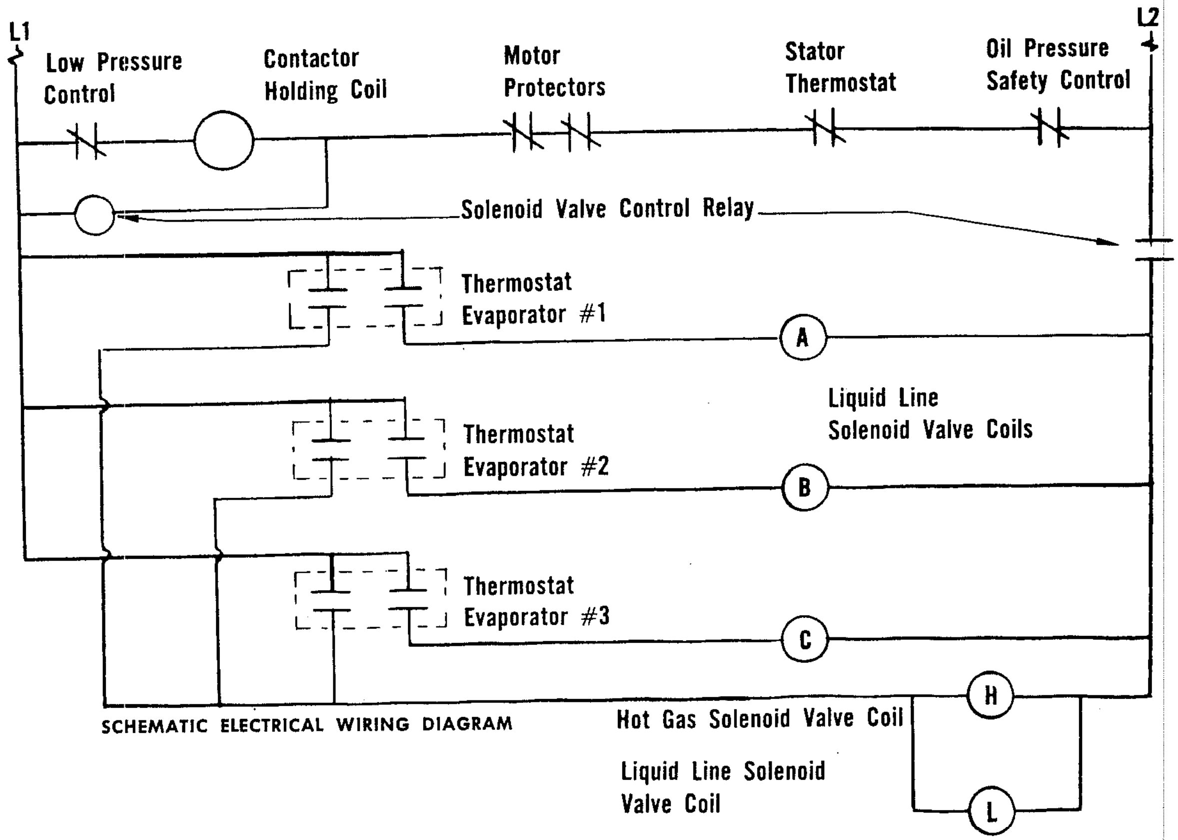


TYPICAL SCHEMATIC CONNECTION — BYPASS INTO EVAPORATOR INLET

Figure 102



SCHEMATIC REFRIGERANT PIPING DIAGRAM



SCHEMATIC ELECTRICAL WIRING DIAGRAM

TYPICAL HOT GAS BYPASS CONTROL SYSTEM
WITH BYPASS INTO SUCTION LINE

Figure 103

BYPASS INTO SUCTION LINE

Where multiple evaporators are connected to one compressor, or where the condensing unit is remote from the evaporator it may be necessary to bypass hot gas into the refrigerant suction line. Suction pressures can be controlled satisfactorily with this method, but a desuperheating expansion valve is required to meter liquid refrigerant into the suction line in order to keep the temperature of the refrigerant gas returning to the compressor within allowable limits. It is necessary to thoroughly mix the bypassed hot gas, the liquid refrigerant, and the return gas from the evaporator so that the mixture entering the compressor is at the correct temperature. A mixing chamber is recommended for this purpose, and a suction line accumulator can serve as an excellent mixing chamber while at the same time protecting the compressor from liquid flood-back. See Figure 103 for typical installation.

Another commonly used method of mixing is to arrange the piping so that a mixture of discharge gas and liquid refrigerant is introduced into the suction line at some distance from the compressor, in a suction header if possible. Figure 104 illustrates this mixing method.

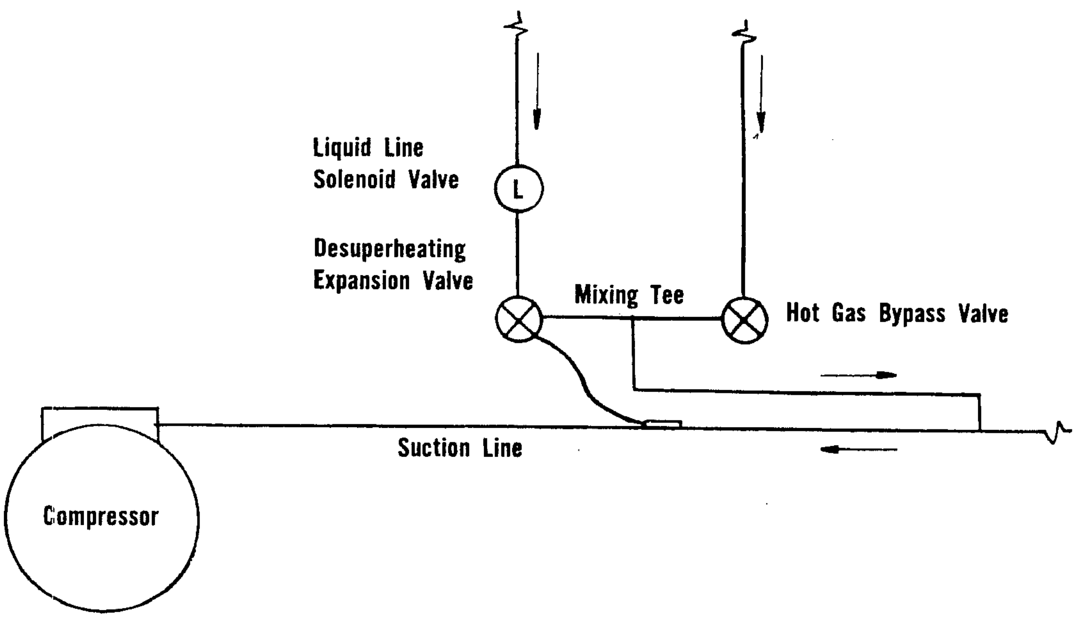
SOLENOID VALVES FOR POSITIVE SHUT-OFF AND PUMPDOWN CYCLE

In order to allow the system to pumpdown, a solenoid valve must be installed ahead of the hot gas bypass valve. Since the hot gas valve opens on a decrease of downstream pressure, it will be open any time the system pressure is reduced below its setting. If the system control is such that this solenoid valve is closed during the normal cooling cycle, it may also prevent possible loss of capacity due to leakage.

A solenoid valve is also recommended ahead of the desuperheating expansion valve to prevent leakage and allow pumpdown. Both of the solenoid valves should be of the normally closed type, and wired so they are de-energized when the compressor is not operating.

DESUPERHEATING EXPANSION VALVE

If a desuperheating expansion valve is required, it should be of adequate size to reduce the temperature of the discharge gas to the proper level under maximum bypass conditions.



ALTERNATE METHOD OF BYPASS INTO SUCTION LINE

Figure 104

The temperature sensing bulb of the expansion valve must be located so that it can sense the temperature of the gas returning to the compressor after the introduction of the hot gas and the desuperheating liquid. Suction gas entering the compressor should be no higher than 65° F. under low temperature load conditions, or 90° F. under high temperature load conditions.

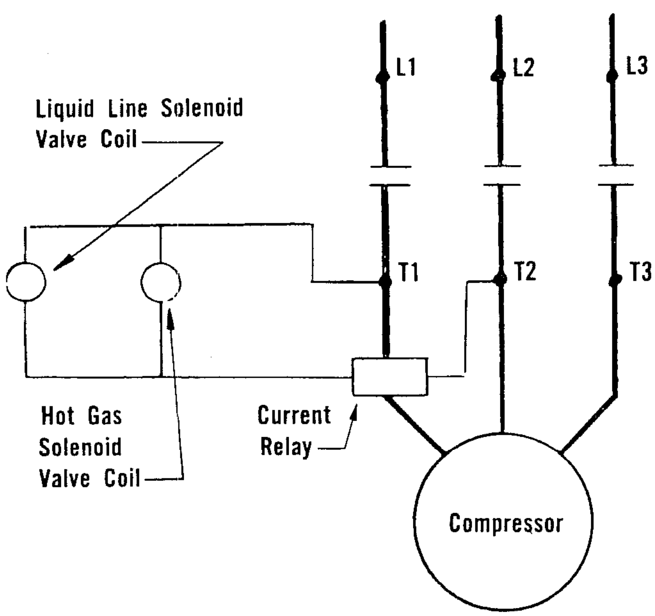
On low temperature applications where hot gas bypass is used to prevent the compressor suction pressure from falling below safe operating levels, valves with unusually high superheat settings may be required. For example, suppose a control was desired to prevent a system using R-502 from operating below -35° F. The temperature of the gas returning to the compressor must be prevented from exceeding 65° F. Therefore, when the desuperheating expansion valve is feeding, it will sense on one side of its diaphragm, the system pressure equivalent to -35° F. or 6.7 psig, and in order to maintain 65° F. return gas, it will require a superheat setting of 65° F. plus 35° F. or 100° F. Expansion valves with special charges are available from expansion valve manufacturers with superheat settings over extremely wide ranges, although these will not normally be available in a local wholesaler's stock. Contact the expansion valve manufacturer's local representative for assistance in selecting valves with non-standard superheat settings.

TYPICAL MULTIPLE-EVAPORATOR CONTROL SYSTEM

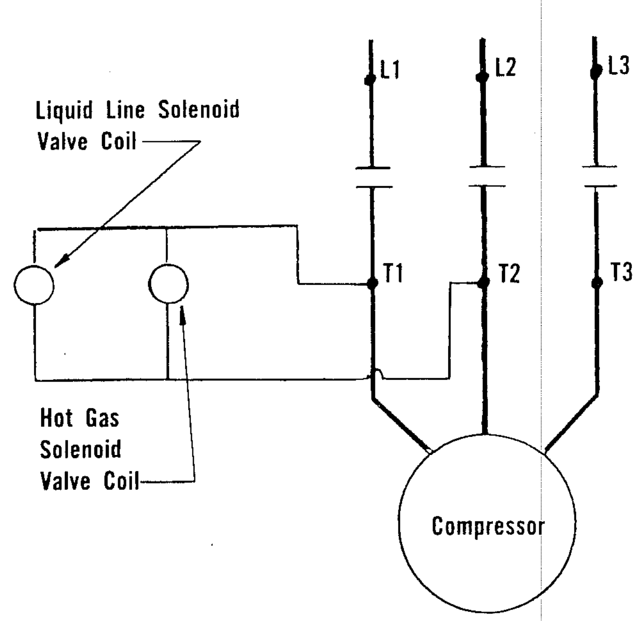
A typical hot gas bypass control system with three evaporators is illustrated in Figure 103 together with a schematic electric control system for cycling control of the compressor. The double pole thermostats close on a demand for refrigeration, and as long as any one evaporator is demanding cooling the compressor operates, and the hot gas bypass valve modulates flow as necessary to prevent the suction pressure from falling below a fixed set point.

If all evaporators are satisfied, all of the thermostats are open, and all liquid line solenoid valves and the hot gas solenoid valve are de-energized, and therefore closed. The compressor will then cycle off on low pressure control until a thermostat again closes.

In order to protect the compressor against danger from liquid flooding in the event of a



TYPICAL SCHEMATIC CONNECTION
CONTINUOUS OPERATION - INHERENT PROTECTION



TYPICAL SCHEMATIC CONNECTION
CONTINUOUS OPERATION - PILOT PROTECTION

Figure 105

trip of a compressor safety device, provision must be made in the wiring circuit to de-energize the hot gas and the desuperheating liquid line solenoid valves if the compressor is inoperative. On a pumpdown system, this can be accomplished by means of a solenoid valve control relay as shown in Figure 103.

If continuous compressor operation is desired, single pole thermostats can be used, and the hot gas and desuperheating liquid line solenoid valves should be connected directly to the load side of the compressor contactor. In the event all three evaporators are satisfied, the compressor will operate on 100% hot gas bypass until cooling is again required.

Compressors equipped with inherent protection can cycle on the inherent protector independently of the contactor. To avoid flooding the compressor with liquid refrigerant in the event the inherent protector should trip, the hot gas solenoid valve and the liquid line solenoid valve should be connected through a current sensing relay such as the Penn R-10A, as shown in Figure 105.

POWER CONSUMPTION WITH HOT GAS BYPASS

Since the power consumption as well as the capacity of a compressor is reduced with a decrease in compressor suction pressure, the control system should be such that the system is allowed to reach its lowest satisfactory operating suction pressure before hot gas is bypassed. Where major reductions in capacity are required, operating economy may be best achieved by handling the load with two compressors. One can be cycled for a 50% reduction in both capacity and power, while the capacity of the compressor remaining on the line is modulated by hot gas control.

It is not necessarily true that continuous compressor operation with hot gas bypass will result in a higher power bill than cycling operation for a given load. Almost all utilities make a monthly demand charge based on peak loads. Since the peak motor demand occurs when locked rotor current is drawn on start-up, the utility demand charge may reflect motor starting requirements rather than the true running load. With continuous operation, once the motors are on the line, starting peaks may be eliminated, and the reduction in the demand charge may offset the increased running power consumption.

Section 22

LIQUID REFRIGERANT CONTROL IN REFRIGERATION AND AIR CONDITIONING SYSTEMS

One of the major causes of compressor failure is damage caused by liquid refrigerant entering the compressor crankcase in excessive quantities. Since improper control of liquid refrigerant can often cause a loss of lubrication in the compressor, most such compressor failures have been classified as lubrication failures, and many people fail to realize that the problem actually originates with the refrigerant.

A well designed, efficient compressor for refrigeration, air conditioning and heat pump duty is primarily a vapor pump designed to handle a reasonable quantity of liquid refrigerant and oil. To design and build a pump to handle more liquid would require a serious compromise in one or more of the following: size, weight, capacity, efficiency, noise, and cost.

Regardless of design there are limits to the amount of liquid a compressor can handle, and these limits depend on factors such as internal volume of the crankcase, oil charge, type of system and controls, and normal operating conditions. Proper control of liquid refrigerant is an application problem, and is largely beyond the control of the compressor manufacturer.

The potential hazard increases with the size of the refrigerant charge and usually the cause of damage can be traced to one or more of the following:

1. Excessive refrigerant charge.
2. Frosted evaporator.
3. Dirty or plugged evaporator filters.
4. Failure of evaporator fan or fan motor.
5. Incorrect capillary tubes.
6. Incorrect selection or adjustment of expansion valves.
7. Refrigerant migration.

REFRIGERANT - OIL RELATIONSHIP

In order to correctly analyze system malfunctions, and to determine if a system is properly protected, a clear understanding of the refrigerant-oil relationship is essential.

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 a liquid, and this migration will continue until the oil is saturated with liquid refrigerant. The amount of refrigerant that the oil will attract is primarily dependent on pressure and temperature, increasing rapidly as the pressure increases and approaching a maximum at saturated pressures and temperatures in the normal room temperature range.

When the pressure on a saturated mixture of refrigerant and oil is suddenly reduced, as happens in the compressor crankcase on start-up, the amount of liquid refrigerant required to saturate the oil is drastically reduced, and the remainder of the liquid refrigerant flashes into vapor, causing violent boiling of the refrigerant and oil mixture. This causes the typical foaming often observed in the compressor crankcase at start-up, which can drive all of the oil out of the crankcase in less than a minute. (Not all foaming is the result of refrigerant in the crankcase - agitation of the oil will also cause some foaming.)

One condition that is somewhat surprising when first encountered by field personnel is the fact that the introduction of excessive liquid refrigerant into the compressor crankcase can cause a loss of oil pressure and a trip of the oil pressure safety control even though the level of the refrigerant and oil mixture may be observed high in the compressor crankcase sight glass. The high percentage of liquid refrigerant entering the crankcase not only reduces the lubricating quality of the oil, but on entering the oil pump intake may flash into vapor, blocking the entrance of adequate oil to maintain oil pump pressure, and this condition can continue until the percentage of refrigerant in the crankcase is reduced to a level which can be tolerated by the oil pump.

Liquid refrigerant problems can take several different forms, each having its own distinct characteristics.

REFRIGERANT MIGRATION

Refrigerant migration is the term used to describe the accumulation of liquid refrigerant in the compressor crankcase during periods when the compressor is not operating. It can occur whenever the compressor becomes colder than the evaporator, since a pressure differential then exists to force refrigerant flow to the colder area. Although this type of migration is most pronounced in colder weather, it can also exist even at relatively high ambient temperatures with remote type condensing units for air conditioning and heat pump applications.

Anytime the system is shut down and is not operative for several hours, migration to the crankcase can occur regardless of pressure due to the attraction of the oil for refrigerant.

If excessive liquid refrigerant has migrated to the compressor crankcase, severe liquid slugging may occur on start-up, and frequently compressor damage such as broken valves, damaged pistons, bearing failures due to loss of oil from the crankcase, and bearing washout (refrigerant washing oil from the bearings) can occur.

LIQUID REFRIGERANT FLOODING

If an expansion valve should malfunction, or in the event of an evaporator fan failure or clogged air filters, liquid refrigerant may flood through the evaporator and return through the suction line to the compressor as liquid rather than vapor. During the running cycle, liquid flooding can cause excessive wear of the moving parts because of dilution of the oil, loss of oil pressure resulting in trips of the oil pressure safety control, and loss of oil from the crankcase. During the "off" cycle after running in this condition, migration of refrigerant to the crankcase can occur rapidly, resulting in liquid slugging when restarting.

LIQUID REFRIGERANT SLUGGING

Liquid slugging is the term used to describe the passage of liquid refrigerant through the compressor suction and discharge valves. It is evidenced by a loud metallic clatter inside the compressor, possibly accompanied by extreme vibration of the compressor.

Slugging can result in broken valves, blown head gaskets, broken connecting rods, broken crankshafts, and other major compressor damage.

Slugging frequently occurs on start-up when liquid refrigerant has migrated to the crankcase. On some units, because of the piping configuration or the location of components, liquid refrigerant can collect in the suction line or evaporator during the off cycle, returning to the compressor as solid liquid with extreme velocity on start-up. The velocity and weight of the liquid slug may be of sufficient magnitude to override any internal anti-slug protective devices of the compressor.

TRIPPING OF OIL PRESSURE SAFETY CONTROL

One of the most common field complaints arising from a liquid flooding condition is that of a trip of the oil pressure safety control after a defrost period on a low temperature unit. The system design on many units allows refrigerant to condense in the evaporator and suction line during the defrost period, and on start-up this refrigerant floods back to the compressor crankcase, causing a loss of oil pressure and recurring trips of the oil pressure safety control.

One trip or a few trips of the oil pressure safety control may not result in serious damage to the compressor, but repeated short periods of operation without proper lubrication are almost certain to result in ultimate compressor failure. Trips of the oil pressure safety control under such circumstances are frequently viewed by the serviceman as nuisance trips, but it cannot be stressed too strongly that they are warning trips, indicating the compressor has been running without oil pressure for 2 minutes, and that prompt remedial action is required.

RECOMMENDED CORRECTIVE ACTION

The potential hazard to a refrigeration or air conditioning system is in almost direct proportion to the size of the refrigerant charge. It is difficult to determine the maximum safe refrigerant charge of any system without actually testing the system with its compressor and other major components. The compressor manufacturer can determine the maximum amount of liquid the compressor will tolerate in the crankcase without endangering the working parts, but has no way of knowing how much of the total system charge will actually be in the compressor under the most extreme conditions. The maximum amount of liquid a compressor can tolerate depends on its design, internal volume, and oil charge. Where liquid migration, flooding, or slugging can occur, corrective action should be taken, the type normally being dictated by the system design and the type of liquid problem.

1. Minimize Refrigerant Charge

The best compressor protection against all forms of liquid refrigerant problems is to keep the charge within the compressor limits. Even if this is not possible, the charge should be kept as low as reasonably possible.

Use the smallest practical size tubing in condensers, evaporators, and connecting lines. Receivers should be as small as possible.

Charge with the minimum amount of refrigerant required for proper operation. Beware of bubbles showing in the sight glass caused by small liquid lines and low head pressures. This can lead to serious overcharging.

2. Pumpdown Cycle

The most positive and dependable means of properly controlling liquid refrigerant, particularly if the charge is large, is by means of a pumpdown cycle. By closing a liquid line solenoid valve, the refrigerant can be pumped into the condenser and receiver, and the compressor operation controlled by means of a low pressure control. The refrigerant can thus be isolated during periods when the compressor is not in operation, and migration to the compressor crankcase is prevented. A recycling type of pumpdown control is recommended to provide protection against possible refrigerant leakage through control devices during the off cycle. With the so-called one time pumpdown, or non-recycling type of control, sufficient leakage may occur during long off periods to endanger the compressor.

Although the pumpdown cycle is the best possible protection against migration, it will not protect against flooding during operation.

3. Crankcase Heaters

On some systems, operating requirements, cost, or customer preference may make the use of a pumpdown cycle undesirable, and crankcase heaters are frequently used to retard migration.

The function of a crankcase heater is to maintain the oil in the compressor at a temperature higher than the coldest part of the system. Refrigerant entering the crankcase will then be vaporized and driven back into the suction line. However in order to avoid overheating and carbonizing of the oil, the wattage input of the crankcase heater must be limited, and in ambient temperatures approaching 0° F., or when exposed suction lines and cold winds impose an added load, the crankcase heater may be overpowered, and migration can still occur.

Crankcase heaters when used are normally energized continuously, since it takes several hours to drive the refrigerant from the crankcase once it has entered and condensed in the oil. They are effective in combating migration if conditions are not too severe, **but they will not protect against liquid floodback.**

4. Suction Accumulators

On systems where liquid flooding is apt to occur, a suction accumulator should be installed in the suction line. Basically the accumulator is a vessel which serves as a temporary storage container for liquid refrigerant which has flooded through the system, with a provision for metered return of the liquid to the compressor at a rate which the compressor can safely tolerate.

Flooding typically can occur on heat pumps at the time the cycle is switched from cooling to heating, or from heating to cooling, and a **suction accumulator is mandatory on all heat pumps unless otherwise approved by the Copeland Application Engineering Department.**

Systems utilizing hot gas defrost are also subject to liquid flooding either at the start or termination of the hot gas cycle. Compressors on low superheat applications such as liquid chillers and low temperature display cases are susceptible to occasional flooding from improper refrigerant control. Truck applications experience extreme flooding conditions at start up after long non-operating periods.

On two stage compressors the suction vapor is returned directly to the low stage cylinders without passing through the motor chamber, **and a suction accumulator should be used to protect the compressor valves from liquid slugging.**

Since each system will vary with respect to the total refrigerant charge and the method of refrigerant control, the actual need for an accumulator and the size required is to a large extent dictated by the individual system requirement. If flooding can occur, an accumulator must be provided with sufficient capacity to hold the maximum amount of refrigerant flooding which can occur at any one time, and this can be well over 50% of the total system charge in some cases. If accurate test data as to the amount of liquid floodback is not available, then 50% of the system charge normally can be used as a conservative design guide.

5. Oil Separators

Oil separators cannot cure oil return problems caused by system design, nor can they remedy liquid refrigerant control problems. However, in the event that system control problems cannot be remedied by other means, oil separators may be helpful in reducing the amount of oil circulated through the system, and can often make possible safe operation through critical periods until such time as system control can be returned to normal conditions. For example, on ultra low temperature applications or on flooded evaporators, oil return may be dependent on defrost periods, and the oil separator can help to maintain the oil level in the compressor during the period between defrosts.

Section 23

ELECTRICAL CONTROL CIRCUITS

Electrical control circuits may be quite simple or extremely complicated, depending on the control requirements of the particular system. Most wiring diagrams furnished with refrigeration equipment are of the pictorial type, and show the wiring as nearly as possible in the manner in which it is installed. Normally the different components are shown, together with terminal designations and wire colors. The pictorial diagram is essential as a guide to proper wiring.

Schematic wiring diagrams are useful in analyzing and explaining the performance of a control circuit, since the schematic diagram shows the various parts of the circuit in a functional manner only, thus reducing the diagram to its simplest form.

Both types of diagram may be used to describe the same control circuit.

TYPICAL LOCKOUT CONTROL CIRCUIT

A typical wiring diagram of a compressor control circuit with part winding motor start, and a 10 minute lockout circuit in the event of a compressor protector trip is shown in Figure 106. The pictorial diagram is shown in the upper half of the illustration, while the schematic diagram is shown at the bottom.

In this circuit, which is designed for fully automatic operation, fast cycling of the compressor from the operation of the motor protectors is eliminated by the use of a 10 minute time delay in conjunction with a double pole impedance relay.

Basically an impedance relay is similar to a normal relay except that the coil has been wound so as to create a high resistance to current passage. If wired in parallel with a circuit having lower resistance, the high impedance (resistance) of the relay will shunt the current to the alternate circuit and the impedance relay will be inoperative. If the alternate circuit is opened and the current must pass through the impedance relay, the relay coil is energized and the relay operates. The voltage drop across the impedance relay is so large that other magnetic coils in series with the relay will not operate because of the resulting low voltage.

In the control circuits shown, the coil of the impedance relay is connected in parallel with the automatic protectors, and in series with the holding coil of the motor starter or contactor. In the event of an overload on the compressor, the stator thermostats or the overload protectors open, and the control circuit current must pass through the impedance relay coil, energizing the impedance relay. Because of the high impedance of this coil, the voltage to the holding coil of the motor starter or contactor falls below the voltage required for operation, and the contactor or starter opens, removing the compressor from the line. When the impedance relay is energized, a single pole double throw 10 minute time delay relay is energized through a set of normally open contacts on the impedance relay. A set of normally closed contacts on the time delay relay break the control circuit and prevent a normal circuit being reestablished through the automatic protectors while the time delay relay is operative. A set of normally open contacts on the time delay relay close when the relay is energized to maintain a circuit to the time delay coil. After a 10 minute interval, a cam on the time delay relay trips, automatically returning the circuit to normal operation. In the event the stator thermostats have not reset, or the overload condition again occurs, the circuit will continue repeating the 10 minute lockout cycle.

CONTROL CIRCUIT FOR COMPRESSOR PROTECTION AGAINST LIQUID REFRIGERANT FLOODING

On systems with large refrigerant charges, compressor damage can occasionally be caused

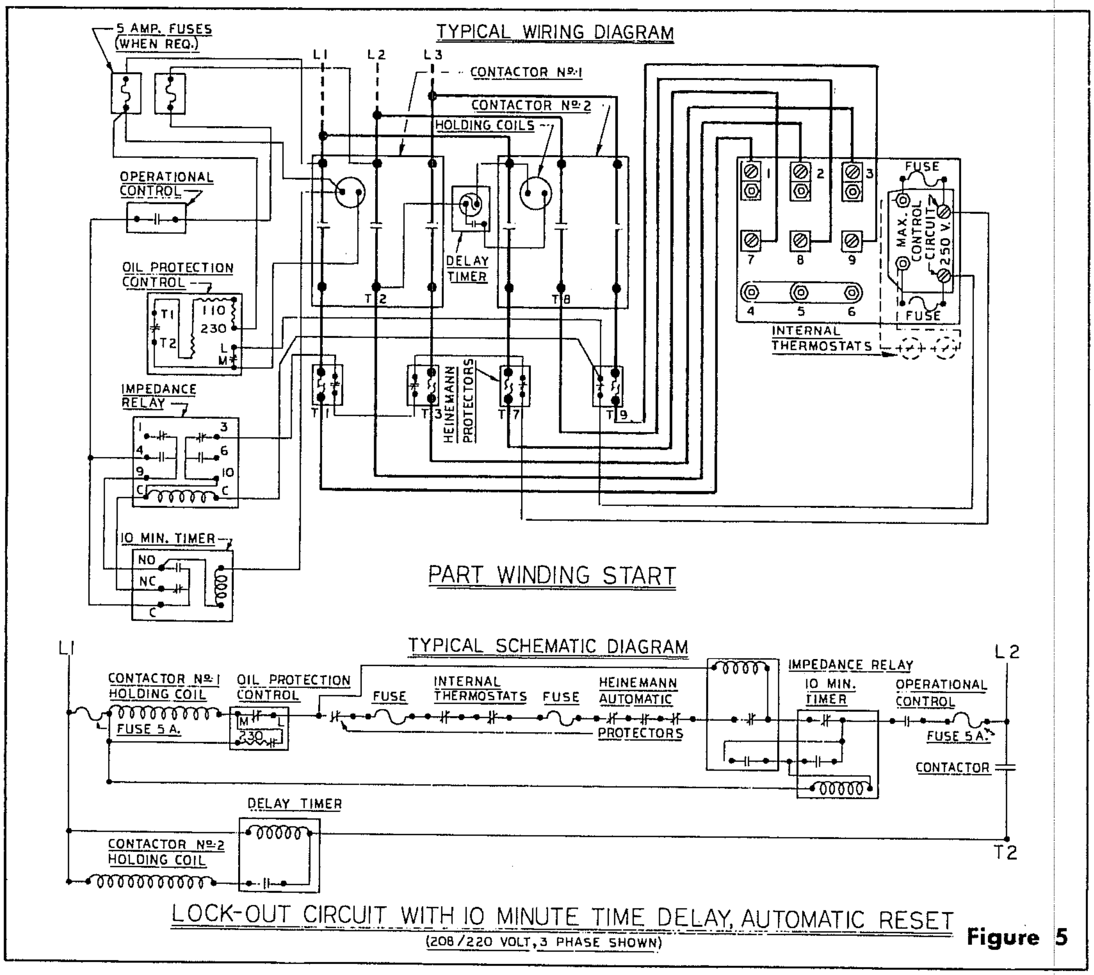


Figure 5

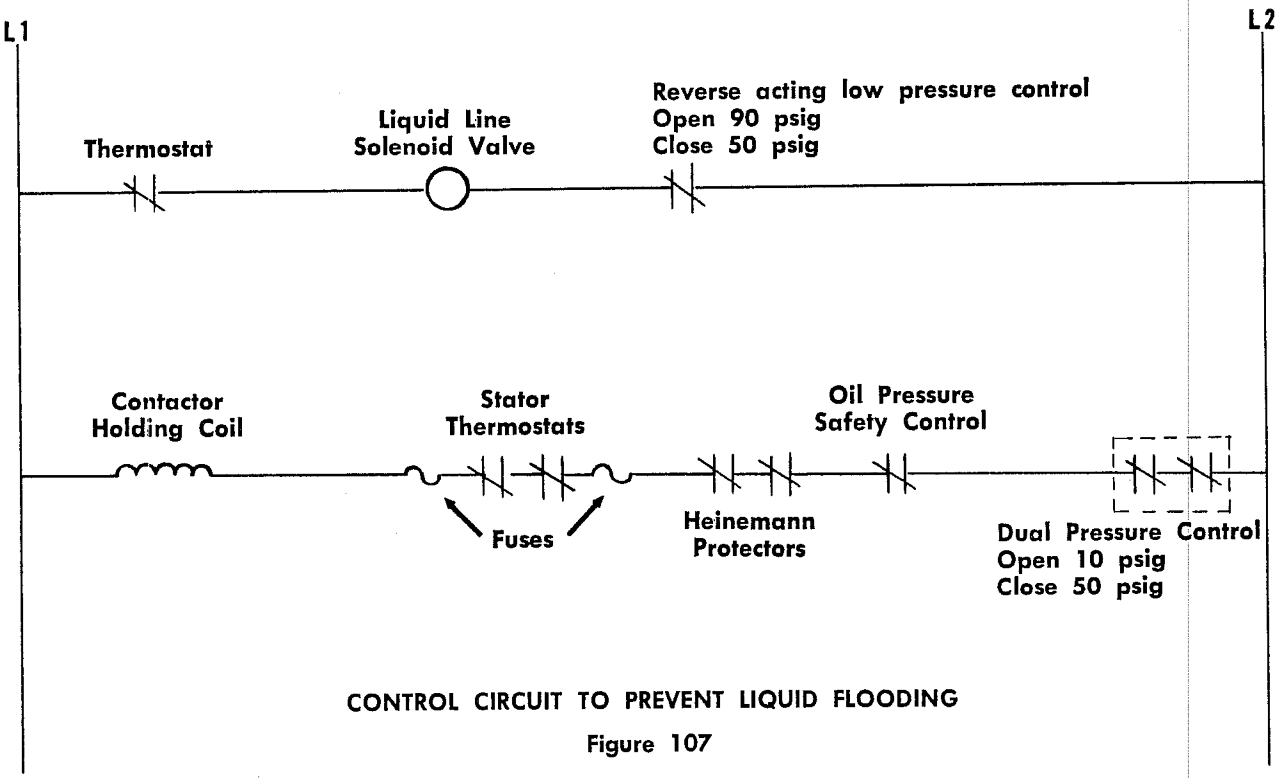
Figure 106

by liquid refrigerant flooding the compressor crankcase should the compressor be non-operative due to a trip of a safety device. This can occur even if the control circuit provides for a continuous pumpdown cycle.

Typically this can happen if the compressor trips either on the motor overload protectors or on the oil pressure safety control. The compressor would then be non-operative, but if the thermostat or other control device is calling for cooling, the liquid line solenoid valve will be open and liquid refrigerant will continue to feed into the evaporator, eventually flooding through to the compressor. When the safety device is either manually or automatically reset and the compressor is restarted, the crankcase will be filled with liquid refrigerant.

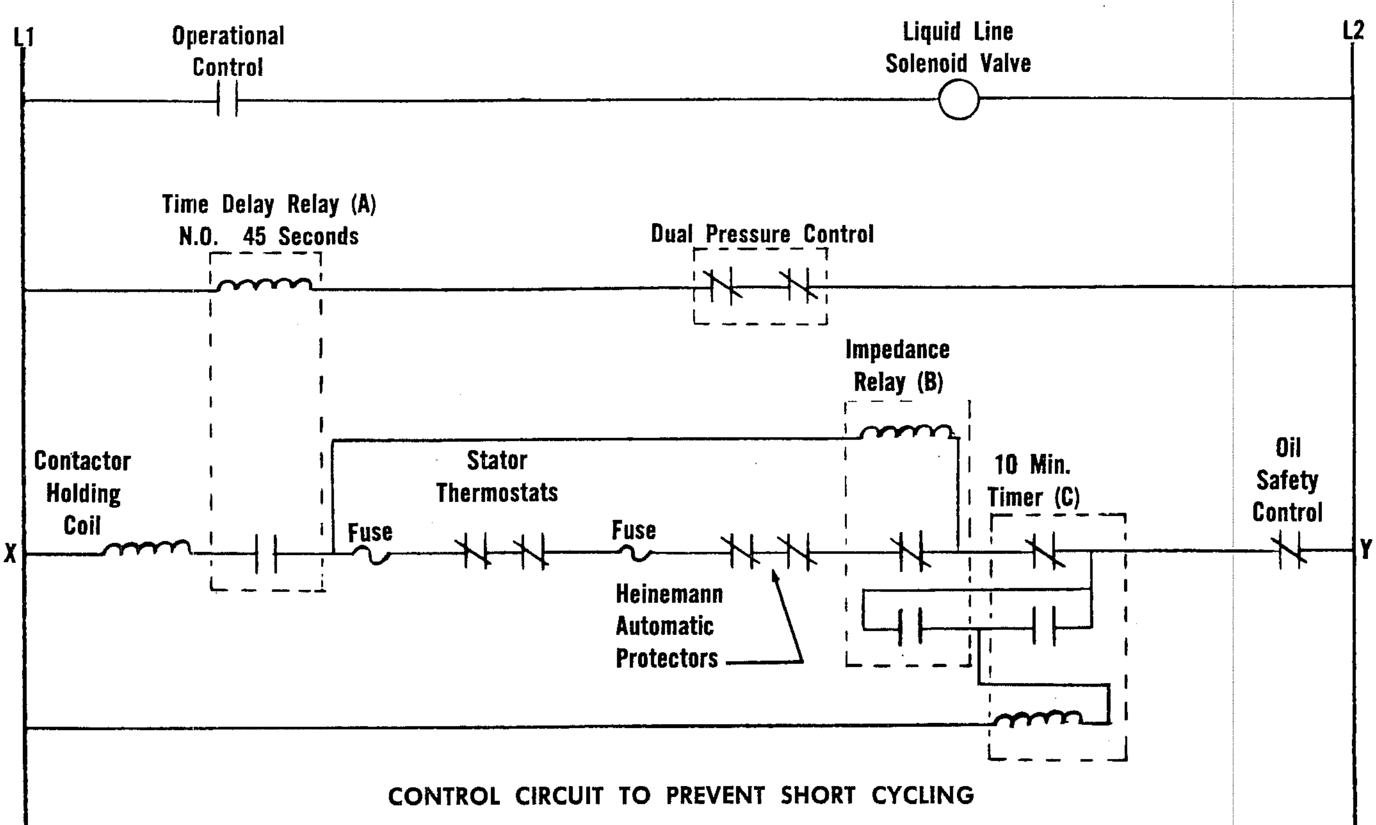
On larger horsepower compressors, this can be a serious problem, both because of the potential cost of the possible damage to the compressor and the amount of refrigerant involved. Flooding of the compressor under non-operative conditions can be prevented by the use of a reverse acting low pressure control as shown in Figure 107.

The liquid line solenoid valve is controlled by the thermostat, but to complete the circuit through the solenoid valve, the contacts in the reverse acting low pressure control must be closed. Since the dual pressure control completes the compressor contactor circuit when the suction pressure reaches 50 psig, the reverse acting control will remain closed during normal system operation. However, should the compressor contactor circuit be broken by any of the safety devices so that the compressor could not start, the reverse acting control will open on a rise in pressure when the evaporator pressure rises above 90 psig. Opening of the reverse acting control de-energizes the liquid line solenoid valve, and stops the liquid refrigerant feed.



CONTROL CIRCUIT TO PREVENT LIQUID FLOODING

Figure 107



CONTROL CIRCUIT TO PREVENT SHORT CYCLING

Figure 108

When the compressor is again restored to operation, the suction pressure is reduced, the reverse acting control again closes, and operation proceeds normally.

The pressure settings shown are tentative settings for an R-22 air conditioning system, and actual settings must be determined after reviewing the system's normal operating range. The important factor is that the reverse acting low pressure control must be set to open well above the setting at which the dual pressure control closes.

CONTROL CIRCUITS TO PREVENT SHORT CYCLING

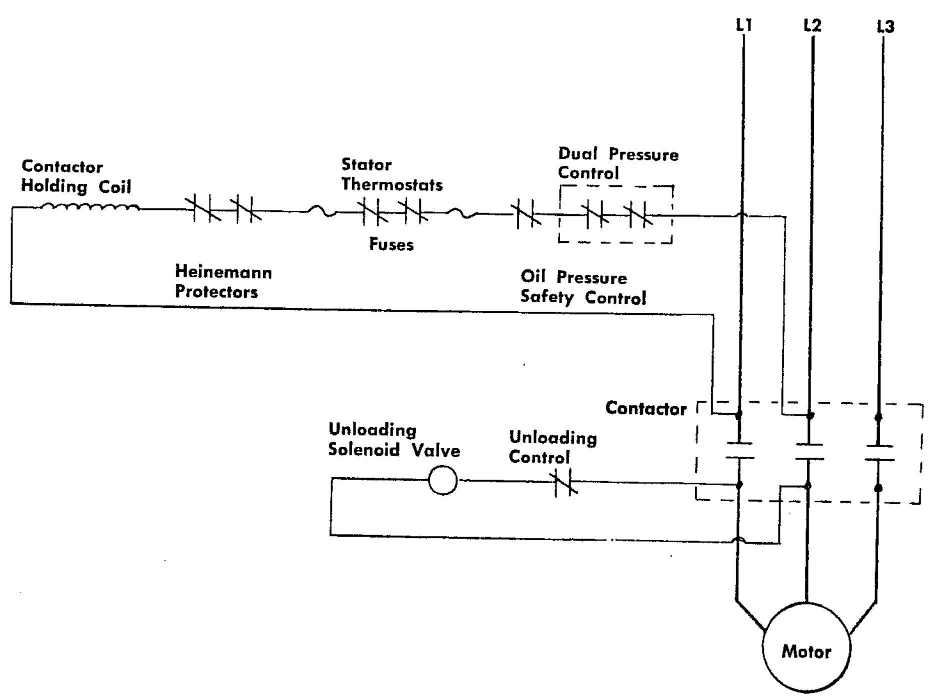
Short cycling often occurs on air conditioning and refrigeration equipment due to a shortage of refrigerant, leaking solenoid valves, incorrect pressure control settings, thermostat chatter, or other causes. Short cycling causes overheating of the compressor and contactor, may cause nuisance tripping of the motor protectors, and in some cases has resulted in welded contactor points and motor failure.

Figure 108 shows a control circuit similar to the lockout circuit discussed previously, with the addition of a pumpdown control circuit with a 45 second time delay to delay starting after closing of the dual pressure control. When the operational control is closed, the normally closed liquid line solenoid valve is energized. The resulting flow of liquid refrigerant into the low pressure side of the system increases the suction pressure, causing the contacts of the low pressure control to make, energizing the time delay relay.

After 45 seconds the time delay relay makes, completing the main control circuit "XY" through the compressor contactor holding coil. In the event of an overload in the compressor circuit, the 10 minute lockout circuit functions as described previously. In the event the operational control or dual pressure control chatter or close immediately after opening, the time delay will prevent re-energizing the circuit for 45 seconds.

CONTROL CIRCUITS FOR COMPRESSORS WITH CAPACITY CONTROL VALVES

To avoid damage to the compressor from refrigerant migration, and to allow proper oper-



TYPICAL CONTROL CIRCUIT FOR COMPRESSOR WITH UNLOADING VALVE

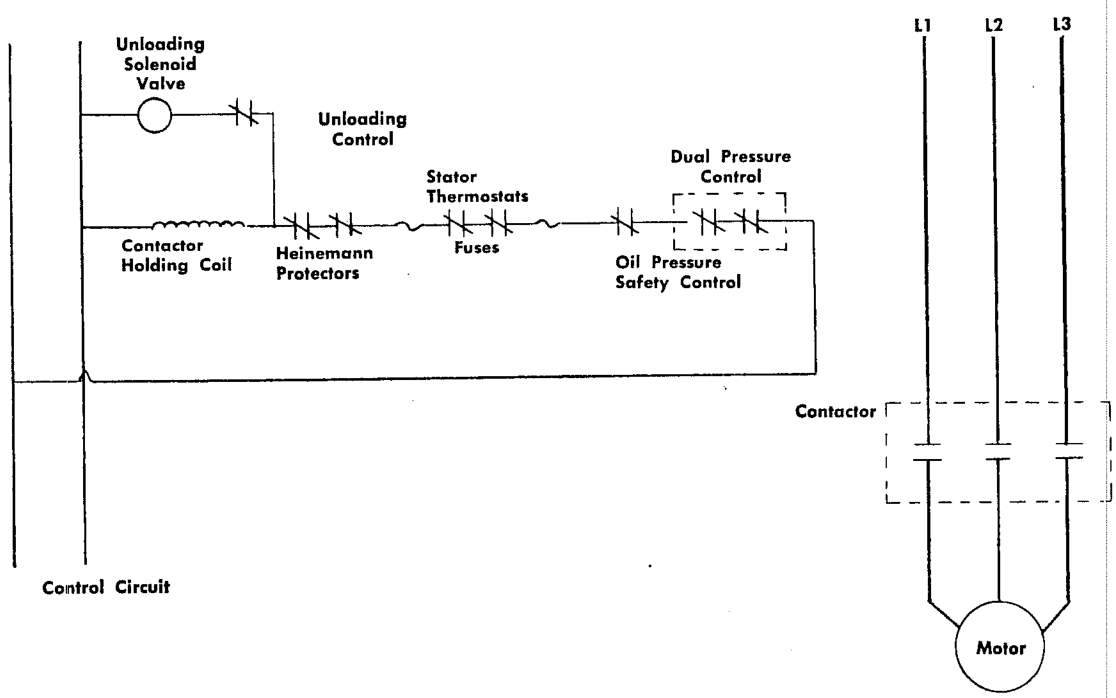
Figure 109

ation on pumpdown systems, it is essential that capacity control solenoid valves be de-energized when the compressor is not operating.

In control circuits operating at line voltage, the solenoid valve and control can be connected to the load side of the contactor as shown in Figure 109.

On large installations, the control circuit may have a power source independent of the compressor power supply. In such cases, the unloading solenoid valve and control may be connected in parallel with the compressor contactor coil as in Figure 110.

There are thousands of variations and types of control circuits, and the above examples are shown merely to illustrate typical circuits frequently encountered in refrigeration work. The basic circuits shown can be adapted as necessary depending on the individual requirement.



TYPICAL CONTROL CIRCUIT FOR COMPRESSOR WITH UNLOADING VALVE AND SEPARATE POWER SOURCE CONTROL CIRCUIT

Figure 110