Copeland®
brand products

Refrigeration Manual

Part 4 - System Design

EMERSON™
Climate Technologies

This is the fourth of a series of publications comprising the Emerson Climate Technologies, Inc. Refrigeration Manual. Although each separate part covers a specific area of refrigeration theory and practice, each successive publication presumes a basic understanding of the material presented in the previous sections.

Part 1 Fundamentals of Refrigeration
Part 2 Refrigeration System Components
Part 3 The Refrigeration Load
Part 4 System Design

The application and design recommendations are intended only as a general guide. The exact requirements of a given installation can only be determined after the specific design criteria and desired operating conditions are known.

Part 4
SYSTEM DESIGN

Section 17. BASIC APPLICATION RECOMMENDATIONS

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SECTION 17
BASIC APPLICATION RECOMMENDATIONS

FUNDAMENTAL DESIGN PRINCIPLES

There are certain fundamental refrigeration design principles which are vital to the proper functioning of any system.

1. The system must be clean, dry, and free from all contaminants.

2. The compressor must be operated within safe temperature, pressure, and electrical limits.

3. The system must be designed and operated so that proper lubrication is maintained in the compressor at all times.

4. The system must be designed and operated so that excessive liquid refrigerant does not enter the compressor. Refrigeration compressors are designed to pump refrigerant vapor, and will tolerate only a limited quantity of liquid refrigerant.

5. Proper refrigerant feed to the evaporator must be maintained, and excessive pressure drop in the refrigerant piping must be avoided.

If these give steps are accomplished, then operation of the system is reasonably certain to be trouble free. If any one is neglected, then eventual operating problems are almost certain to occur. These basic fundamentals are closely inter-related, and must always be kept in mind with regard to the application of any component, or whenever any change in system operation is contemplated.

COMPRESSOR SELECTION

The compressor must be selected for the capacity required at the desired operating conditions in accordance with the manufacturer's recommendations for the refrigerant to be used. Standard Copeland® brand single stage compressors are approved for operation with a given refrigerant in one of the following operating ranges.

	Evaporating Temperature
High Temperature	45° F. to 0° F. or 55° F. to 0° F.
Medium Temperature	25° F. to -5° F.
Low Temperature	0° F. to -40° F.
Extra Low Temperature	-20° F. to -40° F.

Operation at evaporating temperatures above the approved operating range may overload the compressor motor. Operation at evaporating temperatures below the approved operating range is normally not a problem if the compressor motor can be adequately cooled, and discharge temperatures can be kept within allowable limits. Evaporating temperatures below -40° F. are normally beyond the practical lower limit of single stage operation because of compressor inefficiencies and excessive discharge gas temperatures. Because of problems of motor cooling or overloading, some motor-compressors may have approval for operation at limited condensing or evaporating temperatures within a given range, and if so, these limitations will be shown by limited performance curves on the specification sheet.

A given compressor may be approved in two different operating ranges with different refrigerants, for example, high temperature R-12 and low temperature R-502. Since the power requirements for a given displacement with both R-22 and R-502 are somewhat similar, in some cases a compressor may be approved in the same operating range for either of these refrigerants.

Two stage compressors may be approved for evaporating temperatures as low as -80° F., but individual compressor specifications should be consulted for the approved operating range.

Operation at temperatures below -80° F. is normally beyond the practical efficiency range of Copeland® brand two stage compressors, and for lower evaporating temperatures, cascade systems should be employed.

Compressors with unloaders have individually established minimum operating evaporating temperatures since motor cooling is more critical with these compressors. As the compressor is unloaded, less refrigerant is circulated through the system, and consequently less return gas is available for motor cooling purposes.

Copeland® brand motor-compressors should never be operated beyond published operating limits without prior approval of the Emerson Climate Technologies, Inc. Application Engineering Department.

SYSTEM BALANCE

If the compressor or condensing unit selected for a given application is to satisfactorily handle the refrigeration load, it must have sufficient capacity. However, over capacity can be equally as unsatisfactory as under capacity, and care must be taken to see that the compressor and evaporator balance at the desired operating conditions. Checking the proposed system operation by means of a compressor-evaporator-condenser balance chart as described in Section 16 is recommended.

If fluctuations in the refrigeration load are to be expected, which could result in compressor operation at excessively low suction pressures, then some means of capacity control must be provided to maintain acceptable evaporating temperatures. If compressors with unloaders are not available or suitable, and if the load cannot be adequately handled by cycling the compressor, a hot gas bypass circuit may be required.

REFRIGERANT

Copeland® brand compressors are primarily designed for operation with Refrigerants 12, 22, and 502. Operation with other refrigerants in cascade systems may be satisfactory if the proper motor and displacement combination is selected, adequate lubrication can be maintained, and if adequate compressor protection is provided.

R-502 is highly recommended for all single stage low temperature applications, and particularly where evaporating temperatures of -20° F. and below may be encountered. Because of the undesirable high discharge temperatures of R-22 when operated at high compression ratios, R-22 should not be used in single stage low temperature compressors 5 HP and larger.

Different expansion valves are required for each refrigerant, so the refrigerants are not interchangeable in a given system, and should never be mixed. If for some reason it is desirable to change from one refrigerant to another in an existing system, it is usually possible to convert the system by changing expansion valves and control settings providing the existing piping sizes and component working pressures are compatible. In some cases the existing motor-compressor may be satisfactory—for example, in converting from R-22 to R-502. If the conversion will result in higher power requirements as is the case in changing from R-12 to R502, then it may also be necessary to change the motor-compressor.

The refrigerant charge should be held to the minimum required for satisfactory operation, since an abnormally high charge will create potential problems of liquid refrigerant control.

COMPRESSOR COOLING

Refrigerant cooled motor-compressors are dependent on return suction gas for motor cooling, and to a considerable extent, on both air and refrigerant cooled motor-compressors, the discharge gas temperature is directly related to the temperature of the return suction gas. Discharge temperatures above 325° F. to 350° F. contribute to oil breakdown and valve plate damage, and to avoid compressor damage, operating temperatures must be kept below this level. Peak temperatures occur at the discharge valves, and normally the temperature of the discharge line will be from 50° F. to 100°F. below the temperature at the valve plate. Therefore the maximum allowable discharge line temperatures from 225°F. to 250°F.

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, and must never exceed 100°F. On some abnormally critical low temperature applications it may be desirable to insulate the suction lines and return the suction gas to the compressor at lower than normal temperatures to prevent the discharge temperatures from exceeding safe limits, but this is not normally necessary on commercial application where the saturated evaporating temperature is -40°F. or above. The low discharge temperature characteristics of R-502 have made possible much more trouble free operation in single stage low temperature applications.

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

On air cooled condensing units, adequate cooling can normally be accomplished by locating the compressor in the discharge air blast from the condenser fan. For proper cooling, the fan must discharge air directly against the compressor, since the compressor usually 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, cooling must be provided by means of an auxiliary fan discharging air directly again the compressor body. On Copeland® brand compressors with multiple heads such as the 4R and 6R models, auxiliary horizontal airflow may not provide satisfactory cooling, and vertical cooling fans are required.

Water cooled compressors are provided with a water jacket or are wrapped with a copper water coil, and water must be circulated through the compressor cooling circuit before entering the condenser.

Two-stage compressors are equipped with a desuperheating expansion valve for interstage cooling, and no auxiliary cooling is required.

If compressors or condensing units are located in a machine room, adequate ventilation air must be provided to avoid an excessive temperature rise in the room. To allow for peak summer temperatures a 10°F. temperature rise is recommended, although a 15°F. rise in cooler ambients might be acceptable.

The most accurate calculation is to determine the total heat to be rejected by adding the compressor refrigerating capacity at the design operating condition to the heat equivalent of the motor input. The CFM can then be calculated by the formula…

$$\text{CMF} = \frac{\text{BTU/HR}}{{}^{o}\text{TD}}$$

For example, determine the machine room ventilation for an air cooled condensing unit operating at -25°F. evaporator, 120°F. condensing with a net refrigeration capacity of 23,000 BTU/HR, 6,400 watts input to the compressor motor, and a 1 H.P. condenser fan motor.

Compressor capacity	23,000	BTU/HR
Heat equivalent 6400 watts x 3.413	21,843	BTU/HR
Heat equivalent 1 H.P. fan motor	3,700	BTU/HR
Total Heat to be Rejected	48,543	BTU/HR

$$\text{CMF} = \frac{48{,}543 \text{ BTU/HR}}{10^{o}\text{ TD}} = 4.854 \text{ CFM}$$

With remote condensers, approximately 10% of the heat rejected is given off by the compressor casting and the discharge tubing, and the ventilation can be calculated accordingly.

For convenience, table 20A gives a quick estimate of the ventilation air requirement if only the compressor capacity is known.

COMPRESSOR LUBRICATION

An adequate supply of oil must be maintained in the crankcase at all times to insure continuous lubrication. The normal oil level should be maintained at or slightly above the center of the sight glass while operating. An excessive amount of oil must not be allowed in the system as it may result in slugging and possible damage to the compressor valves.

Compressors leaving the factory are charged with naphthenic refrigerant oils. A complete list of acceptable refrigerants and lubricants are listed on form #93-11. The use of any other oil must be specifically cleared with the Emerson Climate Technologies, Inc. Application Engineering Department. The naphthenic base oil has definite advantages over paraffinic base oils because separation of refrigerant from paraffinic oils occurs at substantially higher temperatures with the same oil-refrigerant concentration. When this separation or two phase condition exists the oil floats on top of the refrigerant and the oil pump inlet at the bottom of the sump is fed almost pure refrigerant at start up. The resulting improper lubrication can result in bearing failure. Because of the lower separating temperature of naphthenic oil, the possibility of two-phasing is greatly reduced.

TABLE 20A
Ventilation Air Requirements For Machine Rooms CFM/1000 BTU/HR at 10° F. Air Temperature Rise

	Low Temp.	Medium Temp.	High Temp.
Air cooled condensing unit.	200 CFM	165 CFM	145 CFM
Compressor with remote condenser.	20 CFM	15 CFM	15 CFM

Copelametic® compressors are shipped with a generous supply of oil in the crankcase. However the system may require more or less oil depending on the refrigerant charge and the system design. On field installed systems, after the system stabilizes at its normal operating conditions, it may be necessary to add or remove oil to maintain the desired level.

OIL PRESSURE SAFETY CONTROL

A major percentage of all compressor failures are caused by lack of proper lubrication. Improper lubrication or the loss of lubrication can be due to a shortage of oil in the system, logging of oil in the evaporator or suction line due to insufficient refrigerant velocities, shortage of refrigerant, refrigerant migration or floodback to the compressor crankcase, failure of the oil pump, or improper operation of the refrigerant control devices.

Regardless of the initial source of the difficulty, the great majority of compressor failures due to loss of lubrication could have been prevented. Although proper system design, good preventive maintenance, and operation within the system's design limitations are the only cure for most of these problems, actual compressor damage usually can be averted by the use of an oil pressure safety control.

An oil pressure safety control with a time delay of 120 seconds is a mandatory requirement of the Emerson Climate Technologies, Inc. warranty on all Copelametic® compressors having an oil pump. The control operates on the differential between oil pump pressure and crankcase pressure, and the two minute delay serves to avoid shut down during short fluctuations in oil pressure during start-up.

A trip of the oil pressure safety switch is a warning that the system has been without proper lubrication for a period of two minutes. Repeated trips of the oil pressure safety control are a clear indication that something in the system design or operation requires immediate remedial action. On a well designed system, there should be no trips of the oil pressure safety control, and repeated trips should never be accepted as a normal part of the system operation.

The oil pressure safety control will not protect against all lubrication problems. It cannot detect whether the compressor is pumping oil or a combination of refrigerant and oil. If bearing trouble is encountered on systems where the oil pressure safety control has not tripped, even though inspection proves it to be properly wired, with the proper pressure setting, and in good operating condition, marginal lubrication is occurring which probably is due to liquid refrigerant floodback.

OIL SEPARATORS

Proper refrigerant piping design and operation of the system within its design limits so that adequate refrigerant velocities can be maintained are the only cure for oil logging problems, but an oil separator may be a definite aid in maintaining lubrication where oil return problems are particularly acute.

For example, consider a compressor having an oil charge of 150 ounces, with the normal oil circulation rate being 2 ounces per minute. This means that on a normal system with proper oil return at stabilized conditions, two ounces of oil leave the compressor through the discharge line every minute, and two ounces return through the suction line. If a minimum of 30 ounces of oil in the crankcase is necessary to properly lubricate the compressor, and for some reason oil logged in the system and failed to return to the compressor, the compressor would run out of oil in 60 minutes. Under the same conditions with an oil separator having an efficiency of 80%, the compressor could operate 300 minute or 5 hours before running out of oil.

As a practical matter, there seldom are conditions in a system when no oil will be returned to the compressor, and even with low gas velocities, some fraction of the oil leaving the compressor will be returned. If there are regular intervals of full load conditions or defrost periods when oil can be returned normally, an oil separator can help to bridge long operating periods at light load conditions. Oil separators are mandatory on systems with flooded evaporators controlled by a float valve, on all two stage and cascade ultra-low temperature systems, and on any system where oil return is critical.

Oil separators should be considered as a system aid but not a cure-all or a substitute for good system design. They are never 100% efficient, and in fact may have efficiencies as low as 50% depending on system operating conditions. On systems where piping design encourages oil logging in the evaporator, an oil separator can compensate for system oil return deficiencies only on a temporary basis, and may only serve to delay lubrication difficulties.

If a system is equipped with a suction accumulator, it is recommended that the oil return from the separator be connected to the suction line just ahead of the accumulator. This will provide maximum protection against returning liquid refrigerant to the crankcase. If the system is not equipped with a suction accumulator, the oil return line on suction cooled compressors may be connected to the suction line if more convenient than the crankcase, but on air cooled compressors, oil return must be made directly to the crankcase to avoid damage to the compressor valves.

If the separator is exposed to outside ambient temperatures, it must be insulated to prevent refrigerant condensation during off periods, resulting in return of liquid to the compressor crankcase. Small low wattage strap-on heaters are available for oil separators, and if any problem from liquid condensation in the separator is anticipated, a continuously energized heater is highly recommended.

SUCTION LINE ACCUMULATORS

If liquid refrigerant is allowed to flood through a refrigeration or air conditioning system and return to the compressor before being evaporated, it may cause damage to the compressor due to liquid slugging, loss of oil from the crankcase, or bearing washout. To protect against this condition on systems vulnerable to liquid damage a suction accumulator may be necessary.

The accumulator's function is to intercept liquid refrigerant before it can reach the compressor valves or crankcase. It should be located in the suction line near the compressor, and if a reversing valve is used in the system, the accumulator must be located between the reversing valve and the compressor. Provisions for positive oil return to the crankcase must be provided, but a direct gravity flow which will allow liquid refrigerant to drain to the crankcase during shut-down periods must be avoided. The liquid refrigerant must be metered back to the compressor during operation at a controlled rate to avoid damage to the compressor.

Some systems, because of their design, will periodically flood the compressor with liquid refrigerant. Typically, this can occur on heat pumps at the time the cycle is switched from cooling to heating, or from heating to cooling. The coil which has been serving as the condenser is partially filled with liquid refrigerant, and when suddenly exposed to suction pressure, the liquid is dumped into the suction line. On heat pumps equipped with expansion valves, there may be further flooding due to the inability of the expansion valve to effectively control refrigerant feed for a short period after the cycle change until the system operation is again stabilized.

A similar situation can occur during defrost cycles. With hot gas defrost, when the defrost cycle is initiated, the sudden introduction of high pressure gas into the evaporator may force the liquid refrigerant in the evaporator into the suction line. If the defrost cycle is such that the evaporator can fill with condensed liquid during defrost, or on systems utilizing electric defrost without a pumpdown cycle, an equally dangerous situation may exist at the termination of the defrost cycle.

On systems with a large refrigerant charge, or on any system where liquid floodback is likely to occur, a suction line accumulator is strongly recommended. On heat pumps, truck applications, and on any system where liquid slugging can occur during operation, a suction line accumulator is mandatory for compressor protection unless otherwise approved by the Emerson Climate Technologies, Inc. Application Engineering Department. The actual refrigerant holding capacity needed for a given accumulator is governed by the requirements of the particular application, and the accumulator should be selected to hold the maximum liquid floodback anticipated.

PUMPDOWN SYSTEM CONTROL

Refrigerant vapor will always migrate to the coldest part of the system, and if the compressor crankcase can become colder than other parts of the system, refrigerant in the condenser, receiver, and evaporator will vaporize, travel through the system, and condense in the compressor crankcase.

Because of the difference in vapor pressures of oil and refrigerant, refrigerant vapor is attracted to refrigeration oil, and even though no pressure or temperature difference exists to cause a flow, refrigerant vapor will migrate through the system and condense in the oil until the oil is saturated. During off cycles extending several hours or more, it is possible for liquid refrigerant to almost completely fill the compressor crankcase due to the oil attraction. For example, in a system using R-12 refrigerant which is allowed to equalize at an ambient temperature of 70° F., the oil-refrigerant mixture in the crankcase will end up about 70% refrigerant before equilibrium is reached.

The most positive and dependable means of keeping refrigerant out of the compressor crankcase is the use 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 of refrigerant to the compressor crankcase is prevented.

Pumpdown control can be used on all thermostatic expansion valve systems with the addition of a liquid line solenoid valve, provided adequate receiver capacity is available. Slight refrigerant leakage may occur through the solenoid valve, causing the suction pressure to rise gradually, and a recycling type control is recommended to repeat the pumpdown cycle as required. The occasional short cycle usually is not objectionable.

A pumpdown cycle is highly recommended whenever it can be used. If a non-recycling pumpdown circuit is required, then consideration should be given to the use of a crankcase heater in addition to the pumpdown for more dependable compressor protection.

CRANKCASE HEATERS

On some systems operating requirements, noise considerations, or customer preference may make the use of a pumpdown system undesirable, and crankcase heaters are frequently used to control migration.

By warming the oil, the absorption of refrigerant by the oil is minimized, and under mild weather conditions, any liquid refrigerant in the crankcase can be vaporized and forced out of the compressor. For effective protection, heaters must be energized several hours before starting the compressor. It is recommended that they be energized continuously, independent of compressor operation. Improperly sized heaters can overheat the oil, and heaters used on Copeland® brand compressors must be specifically approved by the Emerson Climate Technologies, Inc. Application Engineering Department.

It would be a mistake to assume that crankcase heaters are a dependable cure for all migration problems. As the ambient conditions contributing to migration worsen, the ability of the crankcase heater to keep refrigerant out of the crankcase decreases. If the suction line slopes toward the compressor, and the temperature to which the suction line is exposed is sufficiently lower than the temperature of the oil, refrigerant may condense in the suction line and flow back to the compressor by gravity at a rate sufficient to offset the heat introduced by the heater. Heaters will not protect against liquid slugs or excessive liquid flooding. However, where operating conditions are not too severe, crankcase heaters can provide satisfactory protection against migration.

Where a pumpdown cycle is not used, crankcase heaters are mandatory on heat pumps, and on other air conditioning applications if the refrigerant charge exceeds the established limits for Copeland® brand compressors, unless tests prove the compressor is adequately protected by other means.

To prevent possible damage in shipment, crankcase heaters are not installed on compressors at the factory.

CRANKCASE PRESSURE REGULATING VALVES

In order to limit the power requirement of the compressor to the allowable operating limit, a crankcase pressure regulating valve may be necessary. This most frequently occurs on low temperature compressors where the power requirement during pulldown periods or after defrost may be greatly in excess of the compressor motor's capabilities. Copeland® brand compressors should not be operated at suction pressures in excess of the published limits on compressor specification sheets without approval of the Emerson Climate Technologies, Inc. Application Engineering Department.

Since any pressure drop in the compressor suction line lowers the system capacity, the CPR valve should be sized for a minimum pressure drop. In order to restrict pull down capacity as little as possible, the valve setting should be as high as the motor power requirement will allow.

Thermal expansion valves of the pressure limiting type are not recommended when a CPR valve is used, particularly if the pressure settings are fairly close, because of the possibility of the action of the two valves coming in conflict in their response to system pressures.

LOW AMBIENT HEAD PRESSURE CONTROL

Within the operating limitations of the system, it is desirable to take advantage of lower condensing temperatures whenever possible for increased capacity, lower discharge temperatures, and lower power requirements. However, too low a discharge pressure can produce serious malfunctions. Since the capacity of capillary tubes and expansion valves is proportional to the differential pressure across the capillary tube or valve, a reduction in discharge pressure will reduce its capacity and produce a drop in evaporating pressure.

Low discharge pressures can result in starving the evaporator coil with resulting oil logging, short cycling on low pressure controls, reduction of system capacity, or erratic expansion valve operation.

Systems with water cooled condensers and cooling towers require water regulating valves, or some other means of controlling the temperature or the quantity of water passing through the condenser.

If air cooled air conditioning systems are required to operate in ambient temperatures below 60° F., a suitable means of controlling head pressures must be provided. Refrigeration systems are also vulnerable to damage from low head pressure conditions, and adequate head pressure controls should be provided for operation in ambient temperatures below 50°F.

Several proprietary control systems are available for low ambient operation, most of which maintain head pressure above a preset minimum by partially flooding the condenser and thus reducing the effective surface area. Methods of this type can control pressures effectively, but do require a considerable increase in refrigerant charge and adequate receiver capacity must be provided.

Air volume dampers on the condenser operated from refrigerant discharge pressure provide a simple, economical, and effective means of control which is widely used.

Adequate protection at lowest cost can often be provided by a reverse acting high pressure control which senses discharge pressure, and acts to disconnect the condenser fan circuit when the head pressure falls below the control's minimum setting. The proper adjustment of the off-on differential is particularly important to avoid excessive fan motor cycling, and the resulting fluctuations in discharge pressure may contribute to uneven expansion valve feeding. In cold ambient temperatures the condenser must be shielded from the wind.

LIQUID LINE FILTER-DRIER

A liquid line filter-drier must be used on all field installed systems, and on all systems opened in the field for service. Filter-driers are highly recommended for all systems, but are not mandatory on factory assembled and charged units where careful dehydration and evacuation is possible during manufacture. Precharged systems with quick connect fittings having a rupture disc are considered to be the equivalent of factory charged systems.

Moisture can be a factor in many forms of system damage, and the reduction of moisture to an acceptable level can greatly extend compressor life and slow down harmful reactions. The desiccant used must be capable of removing moisture to a low end point and further should be of a type which can remove 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.

SIGHT GLASS AND MOISTURE INDICATOR

A combination sight glass and moisture indicator is essential for easy field maintenance on any system, and is required on any field installed system unless some other means of checking the refrigerant charge is provided.

A sight glass is a convenient means of determining the refrigerant charge, showing bubbles when there is insufficient charge, and a solid clear glass when there is sufficient charge. However, the operator should bear in mind that under some circumstances even when the receiver outlet has a liquid seal, bubbles or flash gas may show in the sight glass. This may be due to a restriction or excessive pressure drop in the receiver outlet valve, a partially plugged drier or strainer, or other restriction in the liquid line ahead of the sight glass. If the expansion valve feed is erratic or surging, the increased flow when the expansion valve is wide open can create sufficient pressure drop to cause flashing at the receiver outlet.

Another source of flashing in the sight glass may be rapid fluctuations in compressor discharge pressure. For example, in a temperature controlled room, the sudden opening of shutters or the cycling of a fan can easily cause a reduction in the condensing temperature of 10°F. to 15°F. Any liquid in the receiver may then be at a temperature equivalent to the lower condensing pressure, and flashing will continue until the system has stabilized at the new condensing temperature.

While the sight glass can be a valuable aid in servicing a refrigeration or air conditioning system, a more positive liquid indicator is desirable, and the system performance must be carefully analyzed before placing full reliance on the sight glass as a positive indicator of the system charge.

LIQUID LINE SOLENOID VALVE

A liquid line solenoid valve is recommended on all field installed systems with large refrigerant charges, particularly when the system has a charge in excess of three pounds of refrigerant per motor HP. The solenoid valve will prevent continued feed to the evaporator through the expansion valve or capillary tube when the compressor is not operating, and will control migration of liquid refrigerant from the receiver and condenser to the evaporator and compressor crankcase.

If a pumpdown cycle is not used, the liquid line solenoid valve should be wired to the compressor motor terminals so that the valve will be de-energized when the motor is not operating.

HEAT EXCHANGER

A liquid to suction heat exchanger is highly recommended on all refrigeration systems, and is required on package water chillers and water to water heat pumps because of the low operating superheat. On medium and low temperature applications, a heat exchanger increases system capacity, helps to eliminate flashing of liquid refrigerant ahead of the expansion valve, and aids both in preventing condensation on suction lines and in evaporating any liquid flooding through the evaporator.

On small systems, soldering the liquid and suction lines together for several feet makes an effective heat exchanger.

THERMOSTATIC EXPANSION VALVES

Thermostatic expansion valves must be selected and applied in accordance with the manufacturer's instructions. Either internally equalized or externally equalized valves will feed properly if applied correctly. If the thermal expansion valve is of the externally equalized type, the external equalizer line must be connected, preferably at a point beyond the expansion valve thermal bulb. Do not cap or plug the external equalizer connection as the valve will not operate without this connection.

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

A minimum of 15°F. superheat at the compressor must be maintained at all times to insure the return of dry gas to the compressor suction chamber, and a minimum of 20°F. superheat is recommended. Note that this is not superheat at the expansion valve, but should be calculated from pressure measured at the suction service valve and the temperature measured 18" from the compressor on the bottom of the horizontal run of suction line tubing. Lower superheat can result in liquid refrigerant flooding back to the compressor during variations in the evaporator feed with possible compressor damage as a result. Excessively wet refrigerant vapor continually returning to the compressor can reduce the lubricating qualities of the oil and greatly increase compressor wear, as well as resulting in a loss of capacity.

It is important that users realize that flash gas in the liquid line can seriously affect expansion valve control. So long as a head of pure liquid refrigerant is maintained at the expansion valve, its performance is relatively stable. But if flash gas is mixed with liquid refrigerant fed to the valve, a larger orifice opening is required to feed the same weight of liquid refrigerant. The only way the orifice opening can be increased is by an increase in superheat, and as the percentage of flash gas increases, the superheat increases, the valve opens wide, and the evaporator is progressively more starved.

If the valve has been operating with a large percentage of flash gas entering the expansion valve, and a head of pure liquid refrigerant is suddenly restored, the orifice opening will be larger than required for the load, and liquid will flood through the system to the compressor until the valve again regains control. Conventional expansion valves with the thermal bulb strapped to the suction line may be somewhat sluggish in response, and it may be several minutes before control can be restored to normal.

Typically, changes in the quality of liquid refrigerant feeding the expansion valve can occur quickly and frequently because of the action of head pressure control devices, sudden changes in the refrigeration load, hunting of the expansion valve, action of an unloading valve, or rapid changes in condensing pressure.

On systems with short suction lines and low superheat requirements, quick response thermal bulbs or wells in the suction line may be essential to avoid periodic floodback to the compressor.

Temperatures and pressure alone may not give a true picture of the actual liquid refrigerant control in a system. Excessive oil circulation has the effect of increasing the evaporating temperature of the refrigerant. The response of the expansion valve is based on the saturation pressure and temperature of pure refrigerant. In an operating system, the changed pressure-temperature characteristics of the oil rich refrigerant will give the expansion valve a false reading of the actual superheat, and can result in a somewhat lower actual superheat than apparently exists, causing excessive liquid refrigerant floodback to the compressor. The only real cure for this condition is to reduce oil circulation to a minimum. Normally excessive oil in the evaporator can only result from an excessive system oil charge or other factors which could cause excessive oil circulation, or from low velocities in the evaporator which result in oil logging. In low temperature applications where proper oil circulation cannot be maintained, an oil separator may be required.

Vapor charged valves are satisfactory for air conditioning usage, and are desirable in many cases because of their inherent pressure limiting characteristics. For all refrigeration applications, liquid charged valves should be used to prevent condensation of the charge in the head of the valve and the resulting loss of control in the event the head becomes colder than the thermal bulb.

A pressure limiting type valve may be helpful in limiting the compressor load, and also prevents excessive liquid refrigerant floodback on start-up. On systems using hot gas defrost, the defrost load is normally greater than the refrigeration load, and some other means of limiting the compressor power input must be used if required.

The thermostatic expansion valve must be sized properly for the load. Although a given valve normally has a wide operating capacity range, excessively undersized or oversized valves can cause system malfunctions. Undersized valves may starve the evaporator, and the resulting excessive superheat may adversely affect the system performance. Oversized valves can cause hunting, alternately starving and flooding the evaporator, resulting in extreme fluctuations in suction pressure.

The thermal bulb should normally be located on a horizontal section of the suction line, close to the evaporator outlet, on the evaporator side of any suction line trap or heat exchanger. Do not under any circumstances locate the thermal bulb in a location where the suction line is trapped since this can result in erratic feeding. Satisfactory performance can usually be obtained with the bulb strapped to the suction line at the 3 o'clock position. Mounting on the top of the suction line will decrease sensitivity, and may allow possible liquid flooding. Mounting on the bottom of the suction line can cause erratic feeding due to the rapid temperature changes that can result from even small amounts of liquid refrigerant reaching the thermal bulb location. Particular attention should be given to the location of the thermal bulb on multiple evaporator systems to insure that the refrigerant returning from one evaporator does not affect the control of another evaporator.

EVAPORATORS

Evaporators must be properly selected for the refrigeration load. Too large an evaporator might result in low velocities and possible oil logging. Too small an evaporator will have excessive temperature differentials

Table 21

RECOMMENDED MINIMUM LOW PRESSURE CONTROL SETTING

For Single Stage Copelametic Compressors Without Unloaders

Compressor Application	Recommended Operating Limits		Minimum Recommended Low Pressure Control Setting		
	Minimum	Maximum	R-12	R-22	R-502
High Temperature	0° F.	55° F.	5 psig	16 psig	22 psig
Medium Temperature	0° F.	25° F.	5 psig	16 psig	22 psig
Medium Temperature	-5° F.	25° F.	2 psig	13 psig	19 psig
Low Temperature	-40° F.	0° F.	15″ Vac.	6″ Vac.	0 psig
Extra Low Temperature	-40° F.	-20° F.	15″ Vac.	6″ Vac.	0 psig

between the evaporating refrigerant and the medium to be cooled. The allowable TD between the entering air and the evaporating refrigerant may also be dictated by the humidity control required.

Internal volume of the evaporator tubing should be at a minimum to keep the system refrigerant charge as low as possible, so the smallest diameter tubing that will give acceptable performance should be used. Since pressure drop at low evaporating temperatures is critical so far as capacity is concerned, multiple refrigerant circuits with fairly short runs are preferred. At the same time, it is essential that velocities of refrigerant in the evaporator be high enough to avoid oil trapping.

Vertical headers should have a bottom outlet to allow gravity oil drainage.

SUCTION LINE FILTERS

A heavy duty suction line filter is recommended for every field installation. The filter will effectively remove contaminants from the system at the time of installation, and serves to keep the compressor free of impurities during operation. In the event of a motor burn, the filter will prevent contamination from spreading into the system through the suction line.

The suction line filter should be selected for a reasonable pressure drop, and should be equipped with a pressure fitting just ahead of the filter, preferably in the shell, to facilitate checking pressure drop across the filter during operation.

HIGH AND LOW PRESSURE CONTROLS

Both high and low pressure controls are recommended for good system design on all air cooled systems 1 HP and larger, and are essential on all field installed air cooled systems and on all water cooled systems.

When used for low temperature unit operation control, the low pressure control must not be set below the minimum operating limits of the compressor or the system. One of the most frequent causes of motor overheating and inadequate lubrication is operation of the compressor at excessively low suction pressures. Product specification sheets list the approved compressor operating range, and recommended minimum low pressure control settings for various operating ranges are shown in Table 21.

High pressure controls may be either manual or automatic reset as desired by the customer. If of the manual reset type, provision must be made to prevent liquid refrigerant flooding through the system to the compressor in the event of a trip of the high pressure control.

Internal automatic reset pressure relief valves (Copelimit) are provided in most welded compressors 1 ¾ HP and larger. On factory assembled package systems, the internal Copelimit valve may satisfy U.L. and code requirements without the use of an external high pressure control. A similar high side to low side automatic reset pressure relief valve is installed in all Copelametic® compressors with displacements of 3,000 CFH or greater.

On factory assembled and charged package systems, such as room air conditioners, where loss of charge protection is not considered critical, or where the motor protection device can provide loss of charge protection, low pressure controls may not be essential although recommended.

INTERCONNECTED SYSTEMS

When the crankcases of two or more compressors are interconnected for parallel operation on a single refrigeration system, serious problems of oil return and vibration may be encountered unless the system is properly designed. The tandem compressor consisting of two individual compressors with an interconnecting housing replacing the individual stator covers provides a simple, trouble free solution to this problem.

Because of the potential operating problems, interconnection of individual compressors is not approved with the exception of factory designed, tested, and assembled units specifically approved by the Emerson Climate Technologies, Inc. Application Engineering Department.

ELECTRICAL GROUP FUSING

Individual circuit breakers or fuses should be provided for each compressor motor. Group fusing, where two or more compressors are installed on one fused disconnect, is not recommended since an electrical failure in one compressor would not trip the fuse, and extensive electrical damage could result.

SECTION 18
REFRIGERATION PIPING

Probably the first skill that any refrigeration apprentice mechanic learns is to make a soldered joint, and running piping is so common a task that often its critical importance in the proper performance of a system is overlooked. It would seem elementary in any piping system that what goes in one end of a pipe must come out the other, but on a system with improper piping, it is not uncommon for a serviceman to add gallons of oil to a system, and it may seemingly disappear without a trace. It is of course lying on the bottom of the tubing in the system, usually in the evaporator or suction line. When the piping or operating condition is corrected, the oil will return and those same gallons of oil must be removed.

Refrigeration piping involves extremely complex relationships in the flow of refrigerant and oil. Fluid flow is the name given in mechanical engineering to the study of the flow of any fluid, whether it might be a gas or a liquid, and the inter-relationship of velocity, pressure, friction, density, viscosity, and the work required to cause the flow. These relationships evolve into long mathematical equations which form the basis for the fan laws which govern fan performance, and the pressure drop tables for flow through piping. But 99% of the theories in fluid flow textbooks deal with the flow of one homogenous fluid, and there is seldom even a mention of a combination flow of liquid, gas, and oil such as occurs in any refrigeration system. Because of its changing nature, such flow is just too complex to be governed by a simple mathematical equation, and practically the entire working knowledge of refrigeration piping is based on practical experience and test data. As a result, the general type of gas and liquid flow that must be maintained to avoid problems is known, but seldom is there one exact answer to any problem.

BASIC PRINCIPLES OF REFRIGERATION PIPING DESIGN

The design of refrigeration piping systems is a continuous series of compromises. It is desirable to have maximum capacity, minimum cost, proper oil return, minimum power consumption, minimum refrigerant charge, low noise level, proper liquid refrigerant control, and perfect flexibility of system operation from 0 to 100% of system capacity without lubrication problems. Obviously all of these goals cannot be satisfied, since some are in direct conflict. In order to make an intelligent decision as to just what type of compromise is desirable, it is essential that the piping designer clearly understand the basic effects on system performance of the piping design in the different parts of the system.

In general, pressure drop in refrigerant lines tends to decrease capacity and increase power requirements, and excessive pressure drops should be avoided. The magnitude of the pressure drop allowable varies depending on the particular segment of piping involved, and each part of the system must be considered separately. There are probably more tables and charts available covering line pressure drop and refrigerant line capacities at a given pressure drop than on any other single subject in the field of refrigeration.

It is most important, however, that the piping designer realize that pressure drop is not the only criteria that must be considered in sizing refrigerant lines, and that often refrigerant velocities rather than pressure drop must be the determining factor in system design. In addition to the critical nature of oil return, there is no better invitation to system difficulties than an excessive refrigerant charge. A reasonable pressure drop is far more preferable than over-sized lines which can contain refrigerant far in excess of the system's needs. An excessive refrigerant charge can result in serious problems of liquid refrigerant control, and the flywheel effect of large quantities of liquid refrigerant in the low pressure side of the system can result in erratic operation of the refrigerant control devices.

The size of the service valve supplied on a compressor, or the size of the connection on a condenser, evaporator, accumulator, or other accessory does not determine the size of line to be used. Manufacturers select a valve size or connection fitting on the basis of its application to an average system, and such factors as the type of application, length of connecting lines, type of system control, variation in load, and other factors can be major factors in determining the proper line size. It is quite possible the required line size may be either smaller or larger than the fittings on various system components. In such cases, reducing fittings must be used.

Since oil must pass through the compressor cylinders to provide lubrication, a small amount of oil is always circulating with the refrigerant. Refrigeration oils are soluble in liquid refrigerant, and at normal room temperatures they will mix completely. Oil and refrigerant vapor, however, **do not** mix readily, and the oil can be properly circulated through the system only if the mass velocity of the refrigerant vapor is great enough to sweep the oil along. To assure proper oil circulation, adequate refrigerant velocities must be maintained not only in the suction and discharge lines, but in the evaporator circuits as well.

Several factors combine to make oil return most critical at low evaporating temperatures. As the suction pressure decreases and the refrigerant vapor becomes less dense, the more difficult it becomes to sweep the oil along. At the same time as the suction pressure falls, the compression ratio increases, and as a result compressor capacity is reduced, and the weight of refrigerant circulated decreases. Refrigeration oil alone becomes the consistency of molasses at temperatures below 0°F., but so long as it is mixed with sufficient liquid refrigerant, it flows freely. As the percentage of oil in the mixture increases, the viscosity increases.

At low temperature conditions all of these factors start to converge, and can create a critical condition. The density of the gas decreases, the mass velocity flow decreases, and as a result more oil starts accumulating in the evaporator. As the oil and refrigerant mixture becomes more viscous, at some point oil may start logging in the evaporator rather than returning to the compressor, resulting in wide variations in the compressor crankcase oil level in poorly designed systems.

Oil logging can be minimized with adequate velocities and properly designed evaporators even at extremely low evaporating temperatures, but normally oil separators are necessary for operation at evaporating temperatures below -50°F. in order to minimize the amount of the oil in circulation.

COPPER TUBING FOR REFRIGERANT PIPING

For installations using R-12, R-22, and R-502, copper tubing is almost universally used for refrigerant piping. Commercial copper tubing dimensions have been standardized and classified as follows:

Type K	Heavy Wall
Type L	Medium Wall
Type M	Light Wall

Only types K or L should be used for refrigerant piping, since type M does not have sufficient strength for high pressure applications. Type L tubing is most commonly used, and all tables and data in this manual are based on type L dimensions.

It is highly recommended that only refrigeration grade copper tubing be used for refrigeration applications, since it is available cleaned, dehydrated, and capped to avoid contamination prior to installation. Copper tubing commonly used for plumbing usually has oils and grease or other contaminants on the interior wall, and these can cause serious operating problems if not removed prior to installation.

Table 22 lists the dimensions and properties of standard commercial copper tubing in the sizes commonly used in refrigeration systems, and Table 23 lists the weight of various refrigerants per 100 feet of piping in liquid, suction and discharge lines.

FITTINGS FOR COPPER TUBING

For brazed or soldered joints, the required elbows, tees, couplings, reducers, or other miscellaneous fittings may be either forged brass or wrought copper. Cast fittings are not satisfactory since they may be porous and often lack sufficient strength.

EQUIVALENT LENGTH OF PIPE

Each valve, fitting, and bend in a refrigerant line contributes to the friction pressure drop because of its interruption or restriction of smooth flow. Because of the detail and complexity of computing the pressure drop of each individual fitting, normal practice is to establish an equivalent length of straight tubing for each fitting. This allows the consideration of the entire length of line, including fittings, as an equivalent length of straight pipe. Pressure drop and line sizing tables and charts are normally set up on the basis of a pressure drop per 100 feet of straight pipe, so the use of equivalent lengths allows the data to be used directly.

(continued on p. 18-5)

Table 22

DIMENSIONS AND PROPERTIES OF COPPER TUBE

(Based on ASTM B-88)

Line Size O.D.	Type	Diameter		Wall Thickness In.	Surface Area Sq. Ft./Lin. Ft.		Inside Cross-section Area, Sq. In.	Lineal Feet Containing 1 Cu. Ft.	Weight Lb/Lin. Ft.	Working Pressure Psia
		OD In.	ID In.		OD	ID				
⅜	K	0.375	0.305	0.035	0.0982	0.0798	0.0730	1973.0	0.145	918
	L	0.375	0.315	0.030	0.0982	0.0825	0.0779	1848.0	0.126	764
½	K	0.500	0.402	0.049	0.131	0.105	0.127	1135.0	0.269	988
	L	0.500	0.430	0.035	0.131	0.113	0.145	1001.0	0.198	677
⅝	K	0.625	0.527	0.049	0.164	0.138	0.218	660.5	0.344	779
	L	0.625	0.545	0.040	0.164	0.143	0.233	621.0	0.285	625
¾	K	0.750	0.652	0.049	0.193	0.171	0.334	432.5	0.418	643
	L	0.750	0.666	0.042	0.193	0.174	0.348	422.0	0.362	547
⅞	K	0.875	0.745	0.065	0.229	0.195	0.436	331.0	0.641	747
	L	0.875	0.785	0.045	0.229	0.206	0.484	299.0	0.455	497
1⅛	K	1.125	0.995	0.065	0.295	0.260	0.778	186.0	0.839	574
	L	1.125	1.025	0.050	0.295	0.268	0.825	174.7	0.655	432
1⅜	K	1.375	1.245	0.065	0.360	0.326	1.22	118.9	1.04	466
	L	1.375	1.265	0.055	0.360	0.331	1.26	115.0	0.884	387
1⅝	K	1.625	1.481	0.072	0.425	0.388	1.72	83.5	1.36	421
	L	1.625	1.505	0.060	0.425	0.394	1.78	81.4	1.14	359
2⅛	K	2.125	1.959	0.083	0.556	0.513	3.01	48.0	2.06	376
	L	2.125	1.985	0.070	0.556	0.520	3.10	46.6	1.75	316
2⅝	K	2.625	2.435	0.095	0.687	0.638	4.66	31.2	2.93	352
	L	2.625	2.465	0.080	0.687	0.645	4.77	30.2	2.48	295
3⅛	K	3.125	2.907	0.109	0.818	0.761	6.64	21.8	4.00	343
	L	3.125	2.945	0.090	0.818	0.771	6.81	21.1	3.33	278
3⅝	K	3.625	3.385	0.120	0.949	0.886	9.00	16.1	5.12	324
	L	3.625	3.425	0.100	0.949	0.897	9.21	15.6	4.29	268
4⅛	K	4.125	3.857	0.134	1.08	1.01	11.7	12.4	6.51	315
	L	4.125	3.905	0.110	1.08	1.02	12.0	12.1	5.38	256

Table 23

WEIGHT OF REFRIGERANT IN COPPER LINES

Pounds per 100 feet of Type L Tubing

O.D. Line Size	Volume per 100 Ft. in Cu. Ft.	Weight of Refrigerant, Pounds					
		Liquid @ 100° F.	Hot Gas @ 120° F. Condensing	Suction Gas (Superheated to 65°)			
				-40° F.	-20° F.	20° F.	40° F.
R-12							
⅜	.054	4.25	.171	.011	.018	.044	.065
½	.100	7.88	.317	.021	.033	.081	.120
⅝	.162	12.72	.514	.033	.054	.131	.195
⅞	.336	26.4	1.065	.069	.112	.262	.405
1⅛	.573	45.0	1.82	.118	.191	.464	.690
1⅜	.872	68.6	2.76	.179	.291	.708	1.05
1⅝	1.237	97.0	3.92	.254	.412	1.01	1.49
2⅛	2.147	169.0	6.80	.441	.715	1.74	2.58
2⅝	3.312	260.0	10.5	.680	1.10	2.68	3.98
3⅛	4.728	371.0	15.0	.97	1.57	3.82	5.69
3⅝	6.398	503.0	20.3	1.32	2.13	5.18	7.70
4⅛	8.313	652.0	26.4	1.71	2.77	6.73	10.0
R-22							
⅜	.054	3.84	.202	.013	.021	.052	.077
½	.100	7.12	.374	.024	.04	.096	.143
⅝	.162	11.52	.605	.038	.064	.156	.232
⅞	.336	24.0	1.26	.079	.134	.323	.480
1⅛	.573	40.8	2.14	.136	.228	.550	.820
1⅜	.872	62.1	3.26	.207	.348	.839	1.25
1⅝	1.237	88.0	4.62	.294	.493	1.19	1.77
2⅛	2.147	153.0	8.04	.51	.858	2.06	3.06
2⅝	3.312	236.0	12.4	.78	1.32	3.18	4.72
3⅛	4.728	336.0	17.7	1.12	1.88	4.55	6.75
3⅝	6.398	456.0	24.0	1.51	2.55	6.15	9.14
4⅛	8.313	592.0	31.1	1.97	3.31	8.0	11.19
R-502							
⅜	.054	3.98	.284	.020	.033	.077	.112
½	.100	7.38	.525	.037	.061	.143	.208
⅝	.162	11.95	.852	.061	.098	.232	.337
⅞	.336	24.8	1.77	.126	.204	.481	.700
1⅛	.573	42.3	3.01	.215	.347	.820	1.19
1⅜	.872	64.4	4.60	.327	.527	1.25	1.81
1⅝	1.237	91.2	6.5	.465	.750	1.77	2.57
2⅛	2.147	159.0	11.3	.806	1.30	3.08	4.48
2⅝	3.312	244.0	17.4	1.24	2.0	4.74	6.90
3⅛	4.728	349.0	24.8	1.77	2.87	6.76	9.84
3⅝	6.398	471.0	33.6	2.40	3.87	9.15	13.32
4⅛	8.313	612.0	43.8	3.12	5.03	11.90	17.30

The equivalent length of copper tubing for commonly used valves and fittings is shown in Table 24.

Table 24

EQUIVALENT LENGTH IN FEET OF STRAIGHT PIPE FOR VALVES AND FITTINGS

O.D., In. Line Size	Globe Valve	Angle Valve	90° Elbow	45° Elbow	Tee Line	Tee Branch
½	9	5	.9	.4	.6	2.0
⅝	12	6	1.0	.5	.8	2.5
⅞	15	8	1.5	.7	1.0	3.5
1⅛	22	12	1.8	.9	1.5	4.5
1⅜	28	15	2.4	1.2	1.8	6.0
1⅝	35	17	2.8	1.4	2.0	7.0
2⅛	45	22	3.9	1.8	3.0	10.0
2⅝	51	26	4.6	2.2	3.5	12.0
3⅛	65	34	5.5	2.7	4.5	15.0
3⅝	80	40	6.5	3.0	5.0	17.0

From Mueller Brass Co. Data

For accurate calculations of pressure drop, the equivalent length for each fitting should be calculated. As a practical matter, an experienced piping designer may be capable of making an accurate overall percentage allowance unless the piping is extremely complicated. For long runs of piping of 100 feet or greater, an allowance of 20% to 30% of the actual lineal length may be adequate, while for short runs of piping, an allowance as high as 50% to 75% or more of the lineal length may be necessary. Judgment and experience are necessary in making a good estimate, and estimates should be checked frequently with actual calculations to insure reasonable accuracy.

For items such as solenoid valves and pressure regulating valves, where the pressure drop through the valve is relatively large, data is normally available from the manufacturer's catalog so that items of this nature can be considered independently of lineal length calculations.

PRESSURE DROP TABLES

Figure 76, 77, and 78 are combined pressure drop charts for refrigerants R-12, R-22, and R-502. Pressure drops in the discharge line, suction line, and liquid line can be determined from these charts for condensing temperatures ranging from 80°F. to 120°F.

To use the chart, start in the upper right hand corner with the design capacity. Drop vertically downward on the line representing the desired capacity to the intersection with the diagonal line representing the operating condition desired. Then move horizontally to the left. A vertical line dropped from the intersection point with each size of copper tubing to the design condensing temperature line allows the pressure drop in psi per 100 feet of tubing to be read directly from the chart. The diagonal pressure drop lines at the bottom of the chart represent the change in pressure drop due to a change in condensing temperature.

For example, in Figure 78 for R-502, the dotted line represents a pressure drop determination for a suction line in a system having a design capacity of 5.5 tons or 66,000 BTU/hr operating with an evaporating temperature of -40°F. The 2 5/8" O.D. suction line illustrated has a pressure drop of 0.22 psi per 100 feet at 85°F. condensing temperature, but the same line with the same capacity would have a pressure drop of 0.26 psi per 100 feet at 100°F. condensing, and 0.32 psi per 100 feet at 120°F. condensing.

In the same manner, the corresponding pressure drop for any line size and any set of operating conditions within the range of the chart can be determined.

SIZING HOT GAS DISCHARGE LINES

Pressure drop in discharge lines is probably less critical than in any other part of the system. Frequently the effect on capacity of discharge line pressure drop is over-estimated since it is assumed the compressor discharge pressure and the condensing pressure are the same. In fact, there are two different pressures, the compressor discharge pressure being greater than the condensing pressure by the amount of the discharge line pressure drop. An increase in pressure drop in the discharge line might increase the compressor discharge pressure materially, but have little effect on the condensing pressure. Although there is a slight increase in the heat of compression for an increase in head pressure, the volume of gas pumped is decreased slightly due to a decrease in volumetric efficiency of the compressor. Therefore the total heat to be dissipated through the condenser may be relatively unchanged, and the condensing temperature and pressure may be quite stable, even though the discharge line pressure drop and therefore the compressor discharge pressure might vary considerably.

The performance of a typical Copelametic® compressor, operating at air conditioning conditions with R-22 and an air cooled condenser indicates that for each 5 psi pressure drop in the discharge line, the compressor capacity is reduced less than ½ of 1%, while the power required is increased about 1%. On a typical low

(continued on p. 18-9)

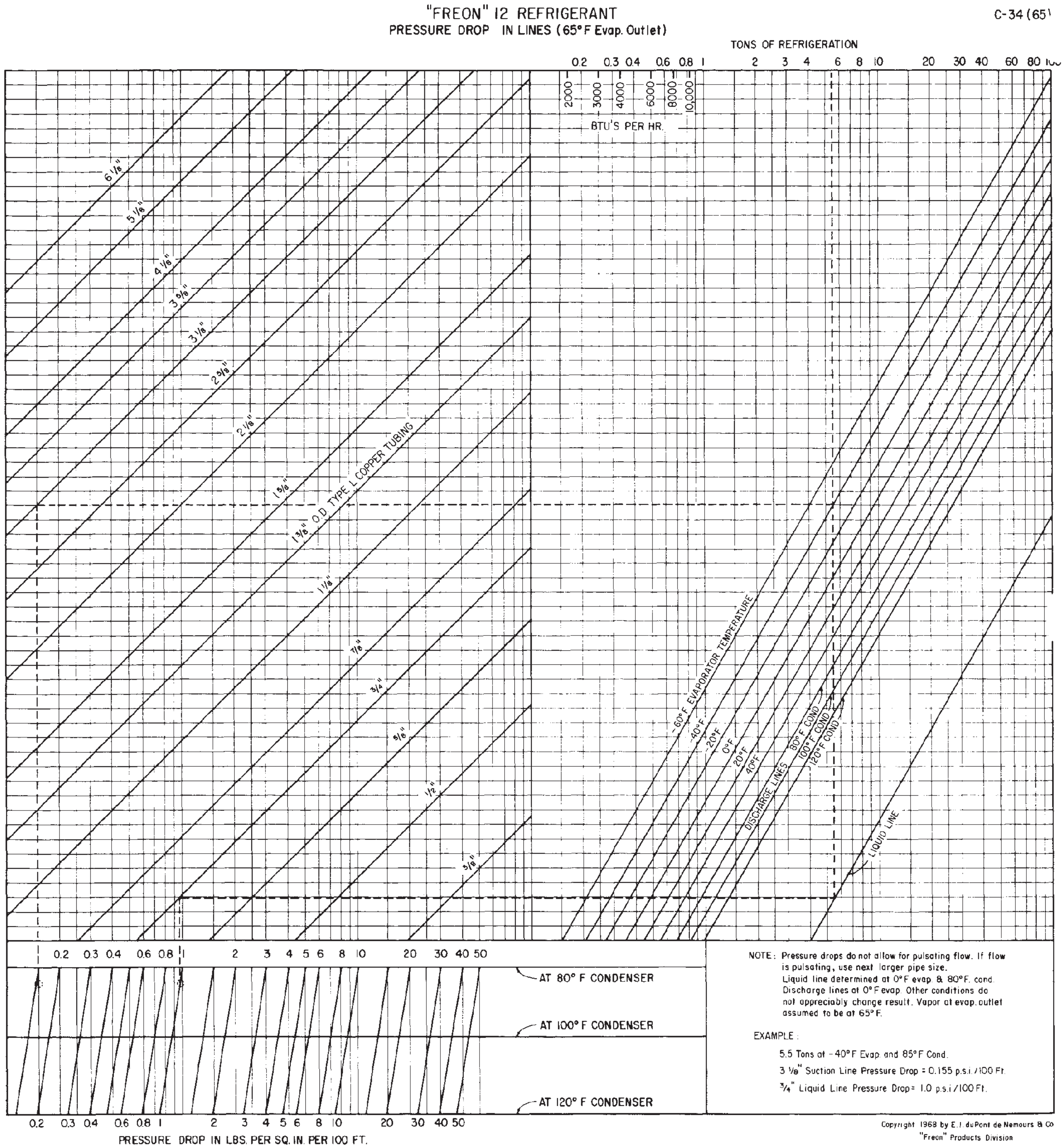

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Figure 76

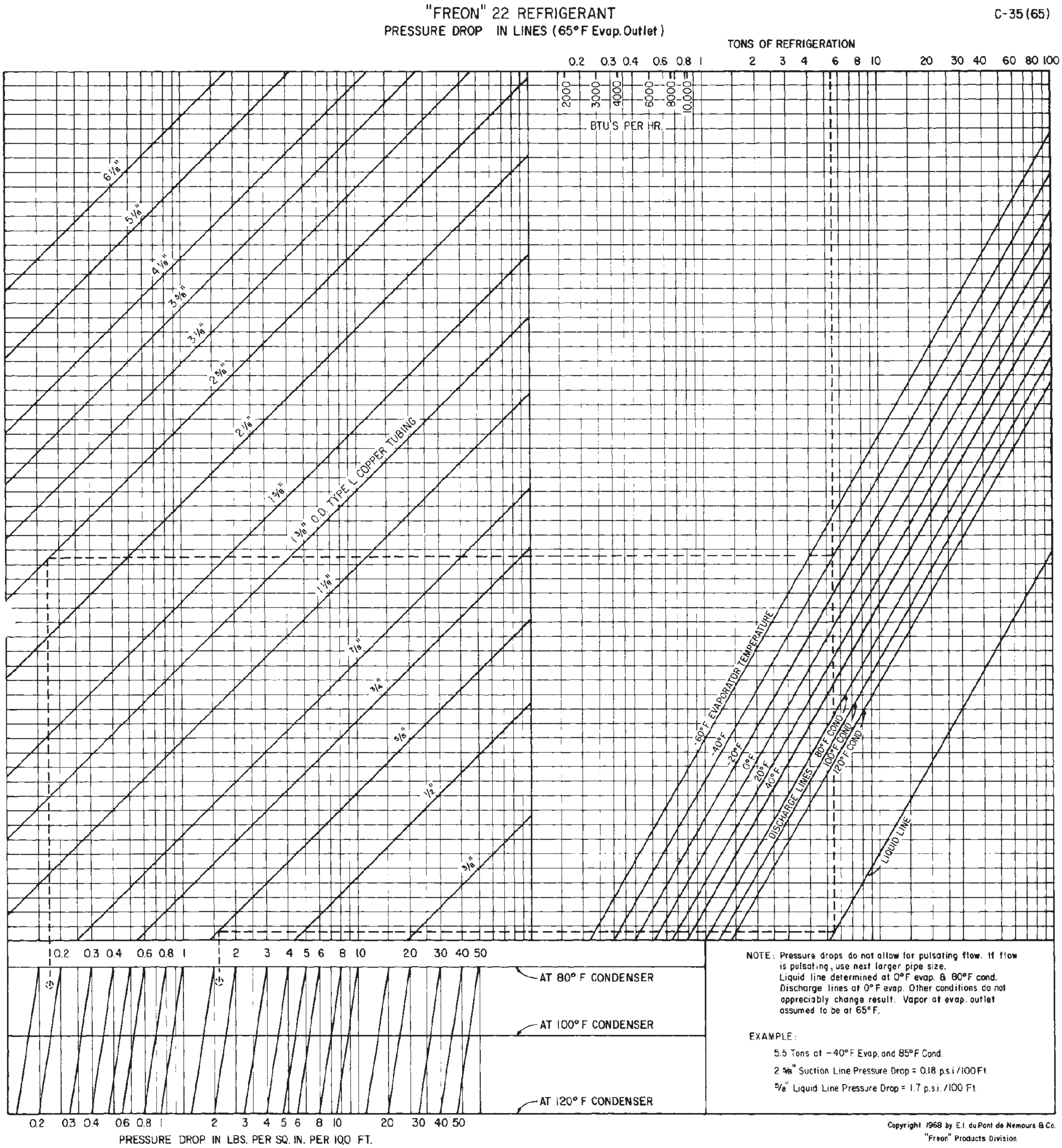

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Figure 77

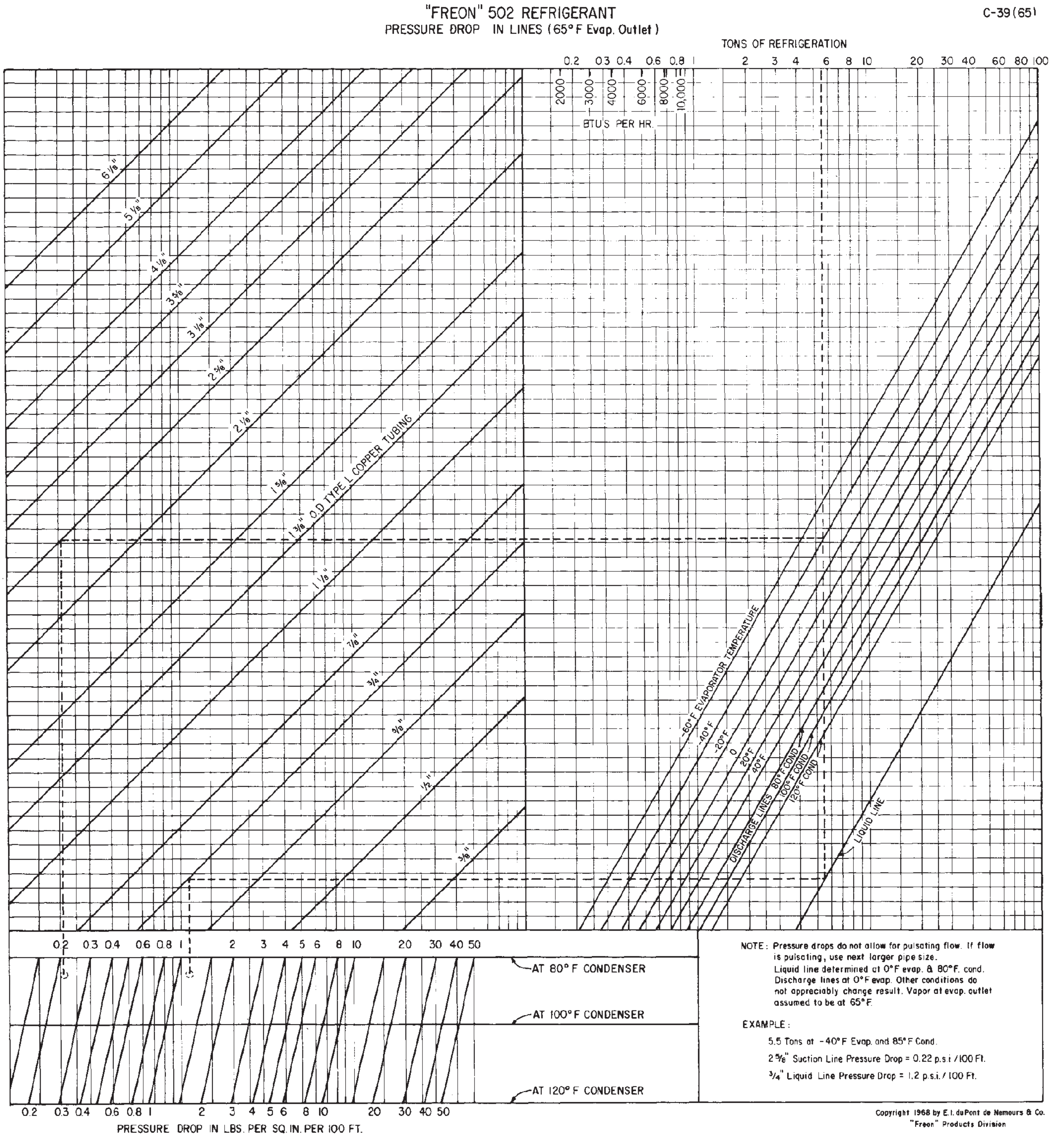
"FREON" 502 REFRIGERANT
PRESSURE DROP IN LINES (65° F Evap. Outlet)
C-39(65)
TONS OF REFRIGERATION
BTU'S PER HR.
2000 3000 4000 6000 8000 10,000
6 1/8"
5 1/8"
4 1/8"
3 5/8"
3 1/8"
2 5/8"
2 1/8"
1 5/8"
1 3/8" O.D. TYPE L COPPER TUBING
1 1/8"
7/8"
3/4"
5/8"
1/2"
3/8"
-60° F EVAPORATOR TEMPERATURE
-40° F
-20° F
0
20° F
40° F
DISCHARGE LINES 80° F COND
100° F COND
120° F COND
LIQUID LINE
AT 80° F CONDENSER
AT 100° F CONDENSER
AT 120° F CONDENSER
PRESSURE DROP IN LBS. PER SQ. IN. PER 100 FT.
NOTE: Pressure drops do not allow for pulsating flow. If flow is pulsating, use next larger pipe size. Liquid line determined at 0°F evap. & 80°F. cond. Discharge lines at 0°F evap. Other conditions do not appreciably change result. Vapor at evap. outlet assumed to be at 65°F.
EXAMPLE:
5.5 Tons at -40°F Evap. and 85°F Cond.
2 5/8" Suction Line Pressure Drop = 0.22 p.s.i. / 100 Ft.
3/4" Liquid Line Pressure Drop = 1.2 p.s.i. / 100 Ft.
Copyright 1968 by E.I. duPont de Nemours & Co. "Freon" Products Division
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Figure 78

temperature Copelametic® compressor operating with R-502 and an air cooled condenser, approximately 1% of compressor capacity will be lost for each 5 psi pressure drop, but there will be little or no change in power consumption.

As a general guide, for discharge line pressure drops up to 5 psi, the effect on system performance would be so small as to be difficult to measure. Pressure drops up to 10 psi would not be greatly detrimental to system performance provided the condenser is sized to maintain reasonable condensing pressures.

Actually a reasonable pressure drop in the discharge line is often desirable to dampen compressor pulsation, and thereby reduce noise and vibration. Some discharge line mufflers actually derive much of their efficiency from pressure drop through the muffler.

Discharge lines on factory built condensing units usually are not a field problem, but on systems installed in the field with remote condensers, line sizes must be selected to provide proper system performance.

Because of the high temperatures existing in the discharge line, oil flows freely, and oil circulation through both horizontal and vertical lines can be maintained satisfactorily with reasonably low velocities. Since oil traveling up a riser usually creeps up the inner surface of the pipe, oil travel in vertical risers is dependent on the velocity of the gas at the tubing wall. The larger the pipe diameter, the greater will be the required velocity at the center of the pipe to maintain a given velocity at the wall surface. Figures 79 and 80 list the maximum recommended discharge line riser sizes for proper oil return for varying capacities. The variation at different condensing temperatures is not great, so the line sizes shown are acceptable on both water cooled and air cooled applications.

If horizontal lines are run with a pitch in the direction of flow of at least ½" in 10 feet, there is normally little problem with oil circulation at lower velocities in horizontal lines. However, because of the relatively low velocities required in vertical discharge lines, it is recommended wherever possible that both horizontal and vertical discharge lines be sized on the same basis.

To illustrate the use of the chart, assume a system operating with R-22 at 40°F. evaporating temperature has a capacity of 100,000 BTU/hr. The intersection of the capacity and evaporating temperature lines at point X on Figure 80 indicate the design condition. Since this is below the 2 1/8" O.D. line, the maximum size that can be used to insure oil return up a vertical riser is 1 5/8" O.D.

Oil circulation in discharge lines is normally a problem only on systems where large variations in system capacity are encountered. For example, an air conditioning system may have steps of capacity control allowing it to operate during periods of light load at capacities possibly as low as 25% or 33% of the design capacity. The same situation may exist on commercial refrigeration systems where compressors connected in parallel are cycled for capacity control. In such cases, vertical discharge lines **must** be sized to maintain velocities above the minimum necessary to properly circulate oil at the minimum load condition.

For example, consider an air conditioning system using R-12 having a maximum design capacity of 300,000 BTU/hr with steps of capacity reduction up to 66%. Although the 300,000 BTU/hr condition could return oil up a 3 1/8" O.D. riser, at light load conditions the system would have only 100,000 BTU/hr capacity, so a 2 1/8" O.D. riser must be used. In checking the pressure drop chart, Figure 76, at maximum load conditions, a 2 1/8" O.D. pipe will have a pressure drop of approximately 3 psi per 100 feet at a condensing temperature of 120°F.

One other limiting factor in discharge line sizing is excessive velocity which can cause noise problems. Velocities of 3,000 FPM or more may result in high noise levels, and it is recommended that maximum velocities be kept well below this level. Figures 81 and 82 give equivalent discharge line gas velocities for varying capacities and line sizes over the normal refrigeration and air conditioning range.

Because of the flexibility in line sizing that the allowable pressure drop makes possible, discharge lines can almost always be sized satisfactorily without the necessity of double risers. If modifications are made to an existing system which result in the existing discharge line being oversized at light load conditions, the addition of an oil separator to minimize oil circulation will normally solve the problem.

To summarize, in sizing discharge lines, it is recommended that a tentative selection of line size be made on the basis of a total pressure drop of approximately 5 psi plus or minus 50%, the actual design pressure drop to a considerable degree being a matter of the designer's judgment. Check Figure 79 or 80 to be sure that velocities at minimum load conditions are adequate to carry oil up vertical risers, and adjust vertical riser size if necessary. Check Figure 81 or 82 to be sure velocities at maximum load are not excessive.

Recommended discharge line sizes for varying capacities and equivalent lengths of line are given in Table 28.

(continued on p. 18-14)

MAXIMUM RECOMMENDED VERTICAL DISCHARGE LINE SIZES FOR PROPER OIL RETURN

R-12 and R-502

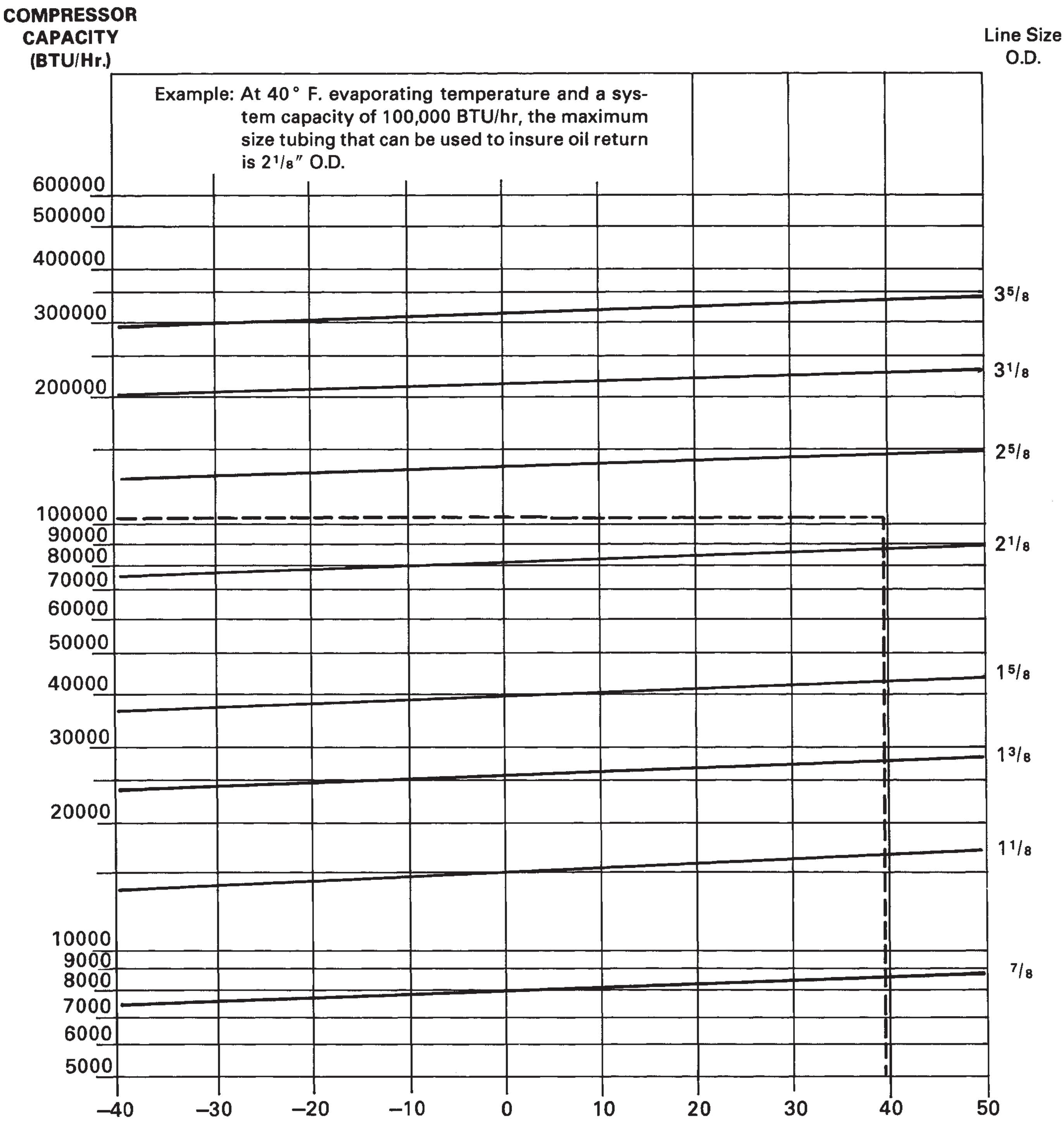


Figure 79

MAXIMUM RECOMMENDED VERTICAL DISCHARGE LINE SIZES FOR PROPER OIL RETURN

R-22

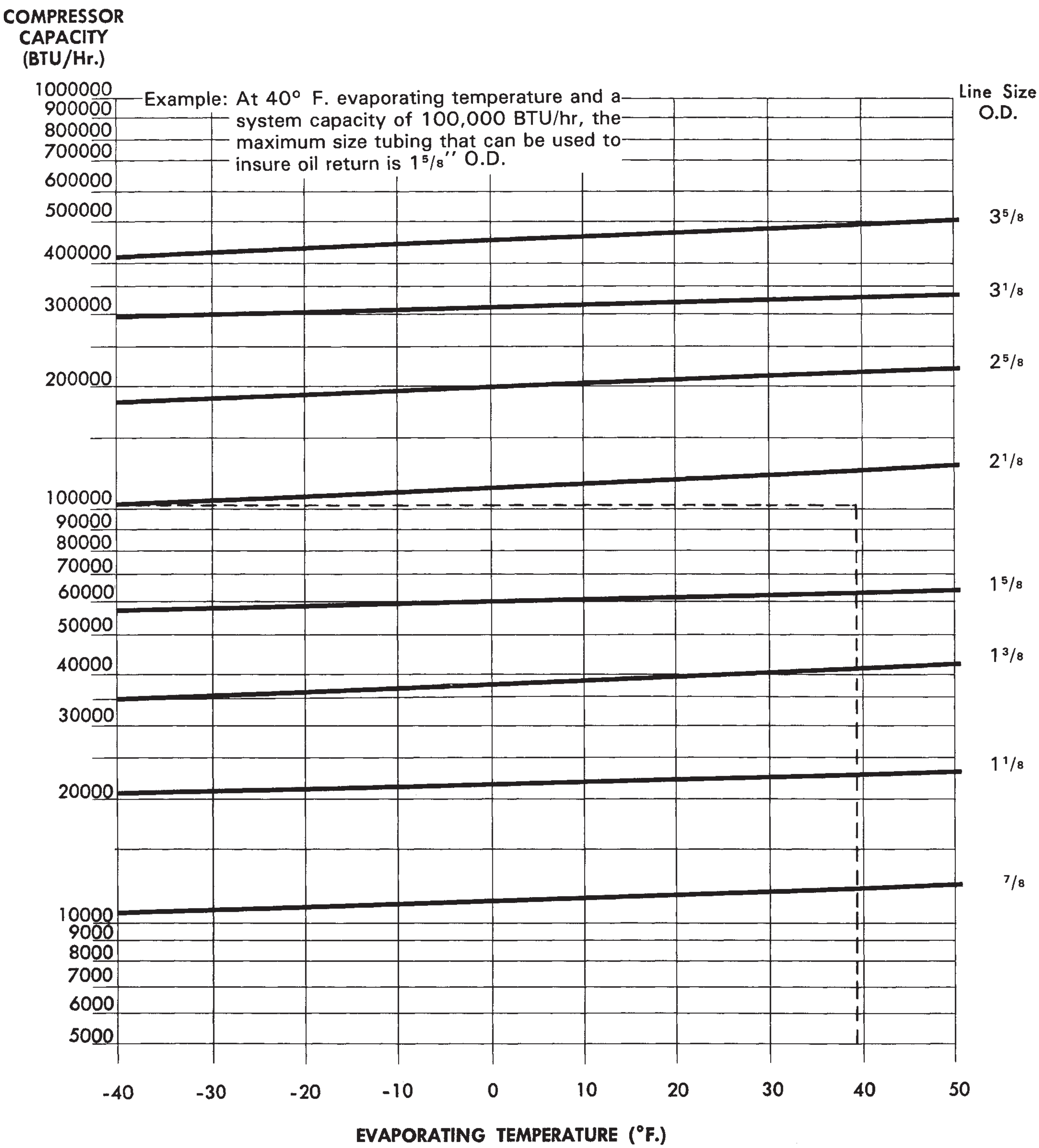


Figure 80

DISCHARGE LINE VELOCITIES FOR VARIOUS BTU/Hr. CAPACITIES

R-12

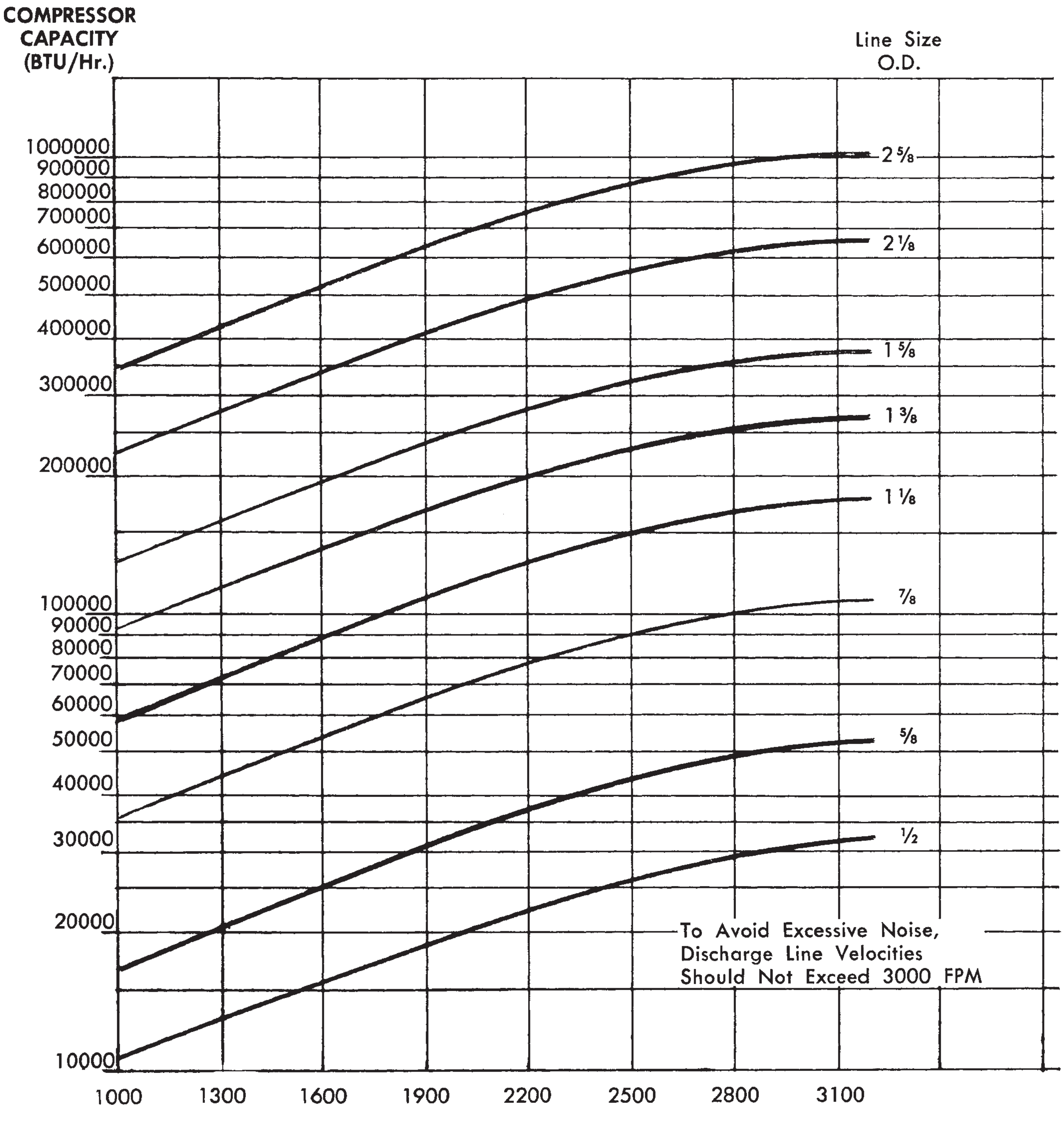


Figure 81

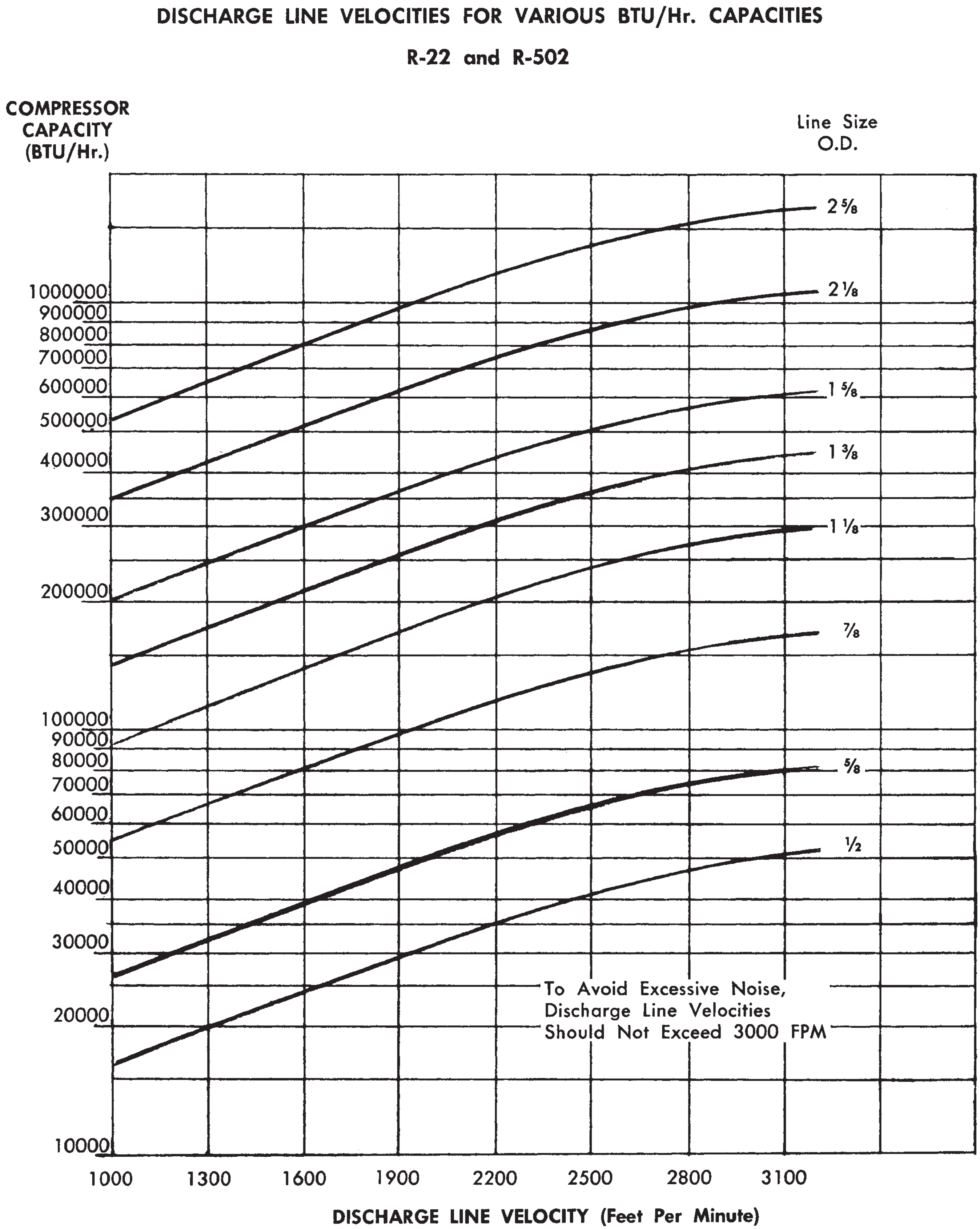

DISCHARGE LINE VELOCITIES FOR VARIOUS BTU/Hr. CAPACITIES
R-22 and R-502
COMPRESSOR CAPACITY (BTU/Hr.)
Line Size O.D.
2 5/8
2 1/8
1 5/8
1 3/8
1 1/8
7/8
5/8
1/2
1000000
900000
800000
700000
600000
500000
400000
300000
200000
100000
90000
80000
70000
60000
50000
40000
30000
20000
10000
1000
1300
1600
1900
2200
2500
2800
3100
To Avoid Excessive Noise, Discharge Line Velocities Should Not Exceed 3000 FPM
DISCHARGE LINE VELOCITY (Feet Per Minute)

Figure 82

SIZING LIQUID LINES

Since liquid refrigerant and oil mix completely, velocity is not essential for oil circulation in the liquid line. The primary concern in liquid line sizing is to insure a solid liquid head of refrigerant at the expansion valve. If the pressure of the liquid refrigerant falls below its saturation temperature, a portion of the liquid will flash into vapor to cool the liquid refrigerant to the new saturation temperature. This can occur in a liquid line if the pressure drops sufficiently due to friction or vertical lift.

Flash gas in the liquid line has a detrimental effect on system performance in several ways. It increases the pressure drop due to friction, reduces the capacity of the expansion device, may erode the expansion valve pin and seat, can cause excessive noise, and may cause erratic feeding of the liquid refrigerant to the evaporator.

For proper system performance, it is essential that liquid refrigerant reaching the expansion device be subcooled slightly below its saturation temperature. On most systems the liquid refrigerant is sufficiently subcooled as it leaves the condenser to provide for normal system pressure drops. The amount of subcooling necessary, however, is dependent on the individual system design.

On air cooled and most water cooled applications, the temperature of the liquid refrigerant is normally higher than the surrounding ambient temperature, so no heat is transferred into the liquid, and the only concern is the pressure drop in the liquid line. Besides the friction loss caused by flow through the piping, a pressure drop equivalent to the liquid head is involved in forcing liquid to flow up a vertical riser. A head of two feet of liquid refrigerant is approximately equivalent to 1 psi. For example, if a condenser or receiver in the basement of a building is to supply liquid refrigerant to an evaporator three floors above, or approximately 30 feet, then a pressure drop of approximately 15 psi must be provided for in system design for the liquid head alone.

On evaporative or water cooled condensers where the condensing temperature is below the ambient air temperature, or on any application where liquid lines must pass through hot areas such as boiler or furnace rooms, an additional complication may arise because of heat transfer into the liquid. Any subcooling in the condenser may be lost in the receiver or liquid line due to temperature rise alone unless the system is properly designed. On evaporative condensers where a receiver and subcooling coil are used, it is recommended that the refrigerant flow be piped from the condenser to the receiver and then to the subcooling coil. In critical applications it may be necessary to insulate both the receiver and the liquid line.

On the typical air cooled condensing unit with a conventional receiver, it is probable that very little subcooling of liquid is possible unless the receiver is almost completely filled with liquid. Vapor in the receiver in contact with the subcooled liquid will condense, and this effect will tend toward a saturated condition.

At normal condensing temperatures, the following relation between each 1°F. of subcooling and the corresponding change in saturation pressure applies.

Refrigerant	Subcooling	Equivalent Change in Saturation Pressure
R-12	1° F.	1.75 psi
R-22	1° F.	2.75 psi
R-502	1° F.	2.85 psi

To illustrate, 5°F. subcooling will allow a pressure drop of 8.75 psi with R-12, 13.75 psi with R-22, and 14.25 psi with R-502 without flashing in the liquid line. For the previous example of a condensing unit in a basement requiring a vertical lift of 30 feet or approximately 15 psi, the necessary subcooling for the liquid head alone would be 8.5°F. with R-12, 5.5°F. with R-22, and 5.25°F. with R-502.

The necessary subcooling may be provided by the condenser used, but for systems with abnormally high vertical risers, a suction to liquid heat exchanger may be required. Where long refrigerant lines are involved, and the temperature of the suction gas at the condensing unit is approaching room temperatures, a heat exchanger located near the condenser may not have sufficient temperature differential to adequately cool the liquid, and individual heat exchangers at each evaporator may be necessary.

In extreme cases, where a great deal of subcooling is required, there are several alternatives. A special heat exchanger with a separate subcooling expansion valve can provide maximum cooling with no penalty on system performance. It is also possible to reduce the capacity of the condenser so that a higher operating condensing temperature will make greater subcooling possible. Liquid refrigerant pumps may also be used to overcome large pressure drops.

Liquid line pressure drop causes no direct penalty in power consumption, and the decrease in system capacity due to friction losses in the liquid line is negligible. Because of this the only real restriction on the amount of liquid line pressure drop is the amount of subcooling available. Most references on pipe sizing recommend a conservative approach with friction pressure drops in the 3 to 5 psi range, but where adequate subcooling is available, many applications have successfully used much higher design pressure drops. The total friction includes line losses through such accessories as solenoid valves, filter-driers, and hand valves.

In order to minimize the refrigerant charge, liquid lines should be kept as small as practical, and excessively low pressure drops should be avoided. On most systems, a reasonable design criteria is to size liquid lines on the basis of a pressure drop equivalent to 2°F. subcooling.

A limitation on liquid line velocity is possible damage to the piping from pressure surges or liquid hammer caused by the rapid closing of liquid line solenoid valves, and velocities above 300 FPM should be avoided when they are used. If liquid line solenoids are not used, then higher velocities can be employed. Figure 83 gives liquid line velocities corresponding to various pressure drops and line sizes.

To summarize, in sizing liquid lines, it is recommended that the selection of line size be made on the basis of a total friction pressure drop equivalent to 2°F. subcooling. If vertical lifts or valves with large pressure drops are involved, then the designer must make certain that sufficient subcooling is available to allow the necessary pressure drop without approaching a saturation condition at which gas flashing could occur. Check Figure 83 to be sure velocities do not exceed 300 FPM if a liquid line solenoid is used.

Recommended liquid line sizes for varying capacities and equivalent lengths of line are given in Table 27.

SIZING SUCTION LINES

Suction line sizing is the most critical from a design and system standpoint. Any pressure drop occurring due to frictional resistance to flow results in a decrease in the pressure at the compressor suction valve, compared with the pressure at the evaporator outlet. As the suction pressure is decreased, each pound of refrigerant returning to the compressor occupies a greater volume, and the weight of the weight of the refrigerant pumped by the compressor decreases. For example, a typical low temperature R-502 compressor at -40°F. evaporating temperature will lose almost 6% of its rated capacity for each 1 psi suction line pressure drop.

Normally accepted design practice is to use as a design criteria a suction line pressure drop equivalent to a 2°F. change in saturation temperature. Equivalent pressure drops for various operating conditions are shown in Table 25.

TABLE 25
PRESSURE DROP EQUIVALENT FOR 2° F. CHANGE IN SATURATION TEMPERATURE AT VARIOUS EVAPORATING TEMPERATURES

Evaporating Temperature	Pressure Drop, PSI R-12	R-022	R-502
45ºF	2.0	3.0	3.3
20ºF	1.35	2.2	2.4
0ºF	1.0	1.65	1.85
-20ºF	.75	1.15	1.35
-40ºF	.5	.8	1.0

Of equal importance in sizing suction lines is the necessity of maintaining adequate velocities to properly return oil to the compressor. Studies have shown that oil is most viscous in a system after the suction vapor has warmed up a few degrees from the evaporating temperature, so that the oil is no longer saturated with refrigerant, and this condition occurs in the suction line after the refrigerant vapor has left the evaporator. Movement of oil through suction lines is dependent on both the mass and velocity of the suction vapor. As the mass or density decreases, higher velocities are required to force the oil along.

Nominal minimum velocities of 700 FPM in horizontal suction lines and 1500 FPM in vertical suction lines have been recommended and used successfully for many years as suction line sizing design standards. Use of the one nominal velocity provided a simple and convenient means of checking velocities. However, tests have shown that in vertical risers the oil tends to crawl up the inner surface of the tubing, and the larger the tubing, the greater velocity required in the center of the tubing to maintain tube surface velocities which will carry the oil. The exact velocity required in vertical lines is dependent on both the evaporating temperature and the line size, and under varying conditions, the specific velocity required might be either greater or less than 1500 FPM.

For better accuracy in line sizing, revised maximum recommended vertical suction line sizes based on the minimum gas velocities shown in the 1980 ASHRAE Handbook have been calculated and are plotted in chart form for easy usage in Figures 84 and 86. These revised recommendations superseded previous vertical suction riser recommendations. No change has been made in the 700 FPM minimum velocity recommendation for horizontal suction lines, and Figures 85 and 87 cover maximum recommended horizontal line sizes for proper oil return.

(continued on p. 18-21)

LIQUID LINE VELOCITIES FOR VARIOUS PRESSURE DROPS

R-12, R-22, R-502

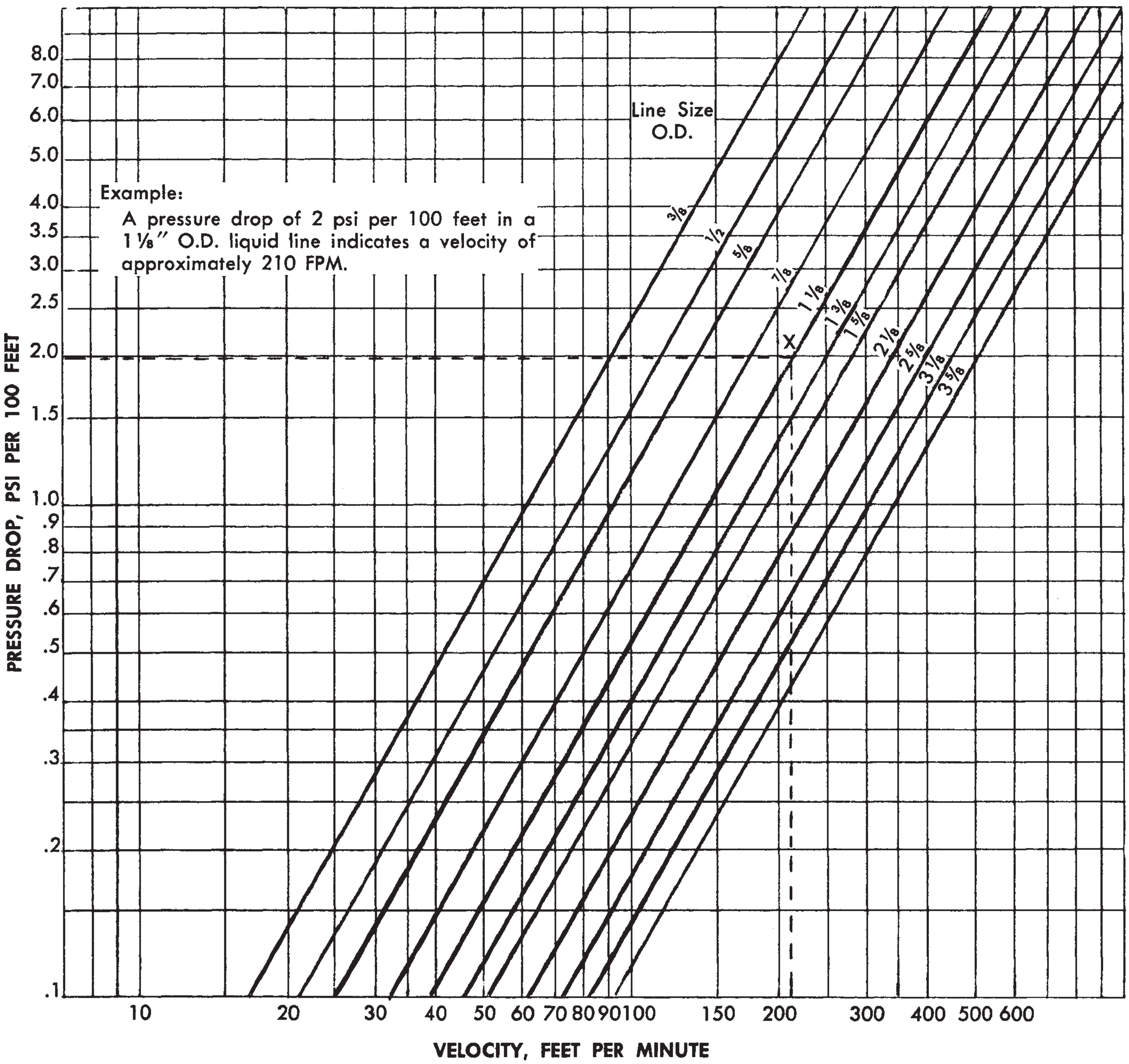


Figure 83

MAXIMUM RECOMMENDED SUCTION LINE SIZES FOR PROPER OIL RETURN

VERTICAL RISERS R-12 and R-502

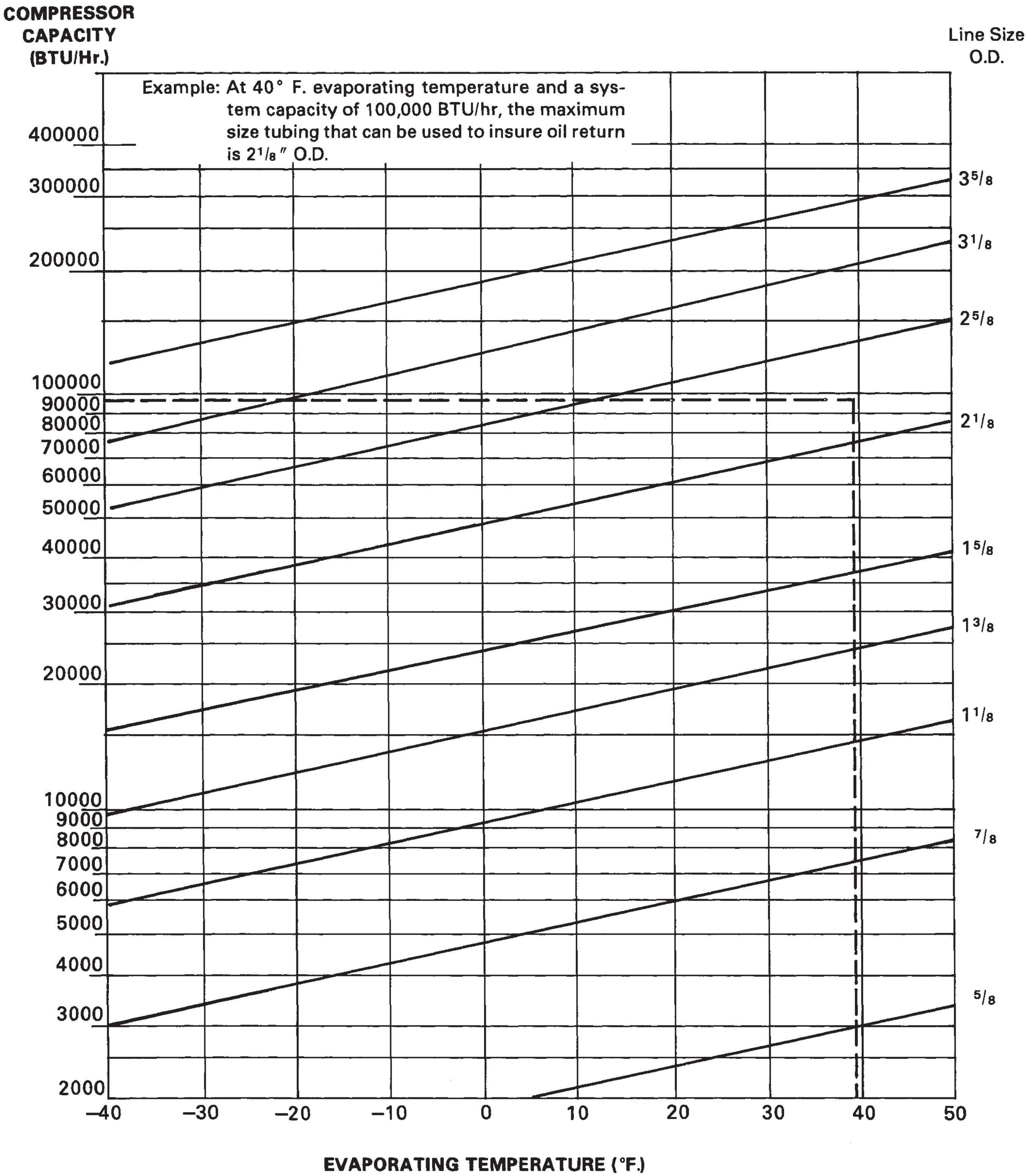


Figure 84

MAXIMUM RECOMMENDED HORIZONTAL SUCTION LINE SIZES FOR PROPER OIL RETURN

R-12

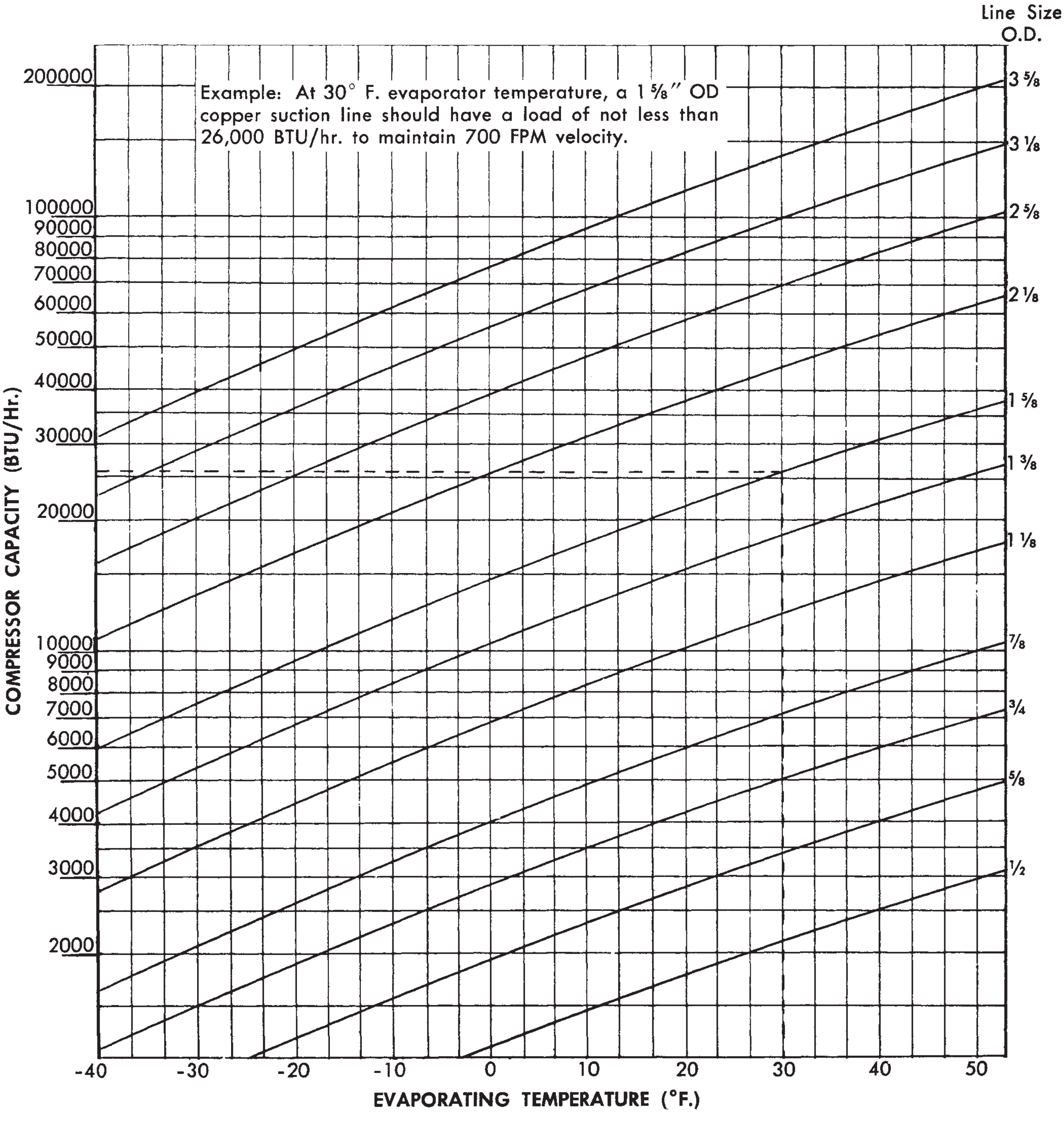


Figure 85

MAXIMUM RECOMMENDED SUCTION LINE SIZES FOR PROPER OIL RETURN

VERTICAL RISERS R-22

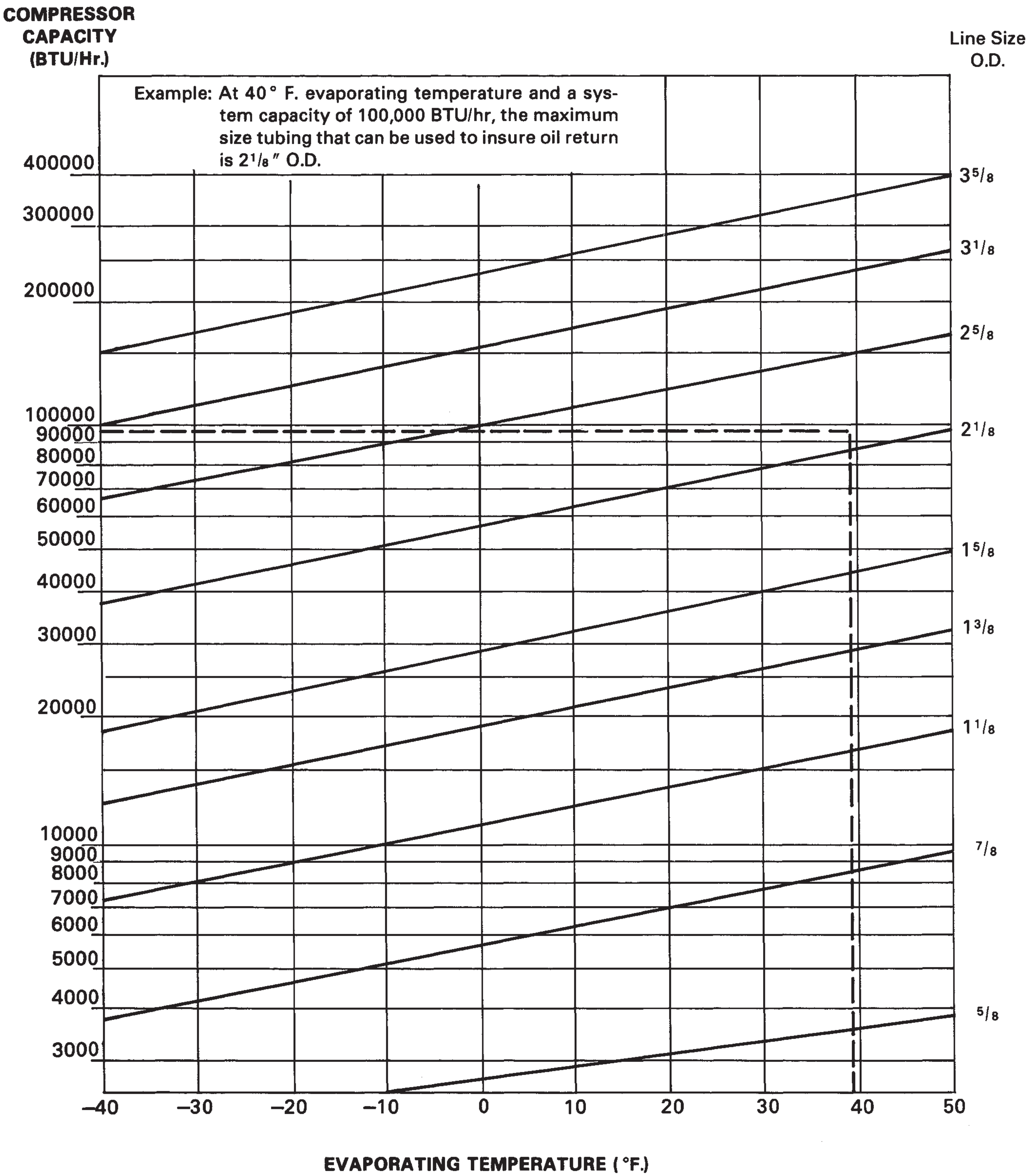


Figure 86

MAXIMUM RECOMMENDED HORIZONTAL SUCTION LINE SIZES FOR PROPER OIL RETURN

R-22 and R-502

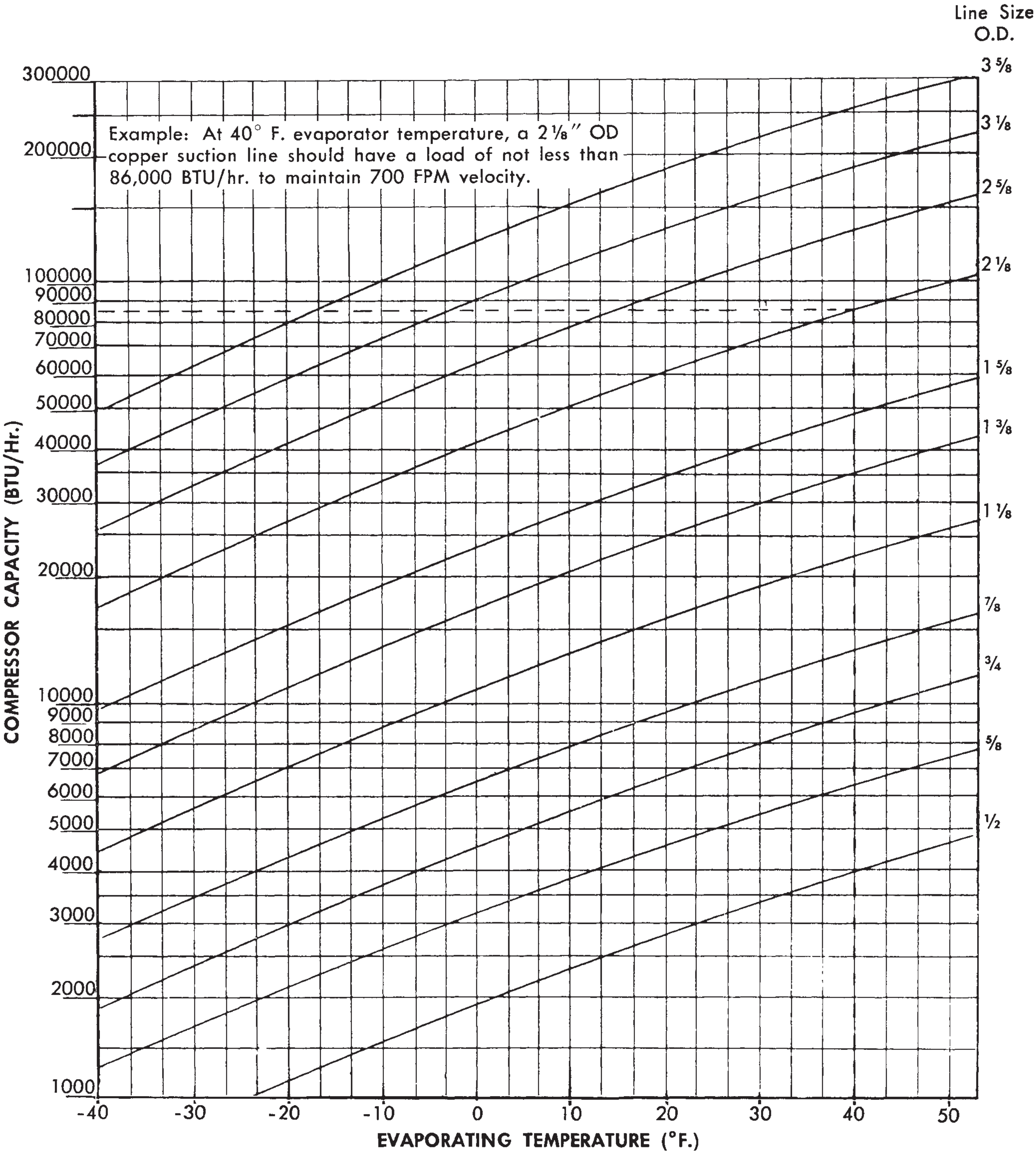


Figure 87

To illustrate, again assume a system operating with R-12 at 40°F. evaporating temperature has a capacity of 100,000 BTU/hr. On Figure 84, the intersection of the evaporating temperature and capacity lines indicate that a 2 1/8” O.D. line will be required for oil return in the vertical suction risers.

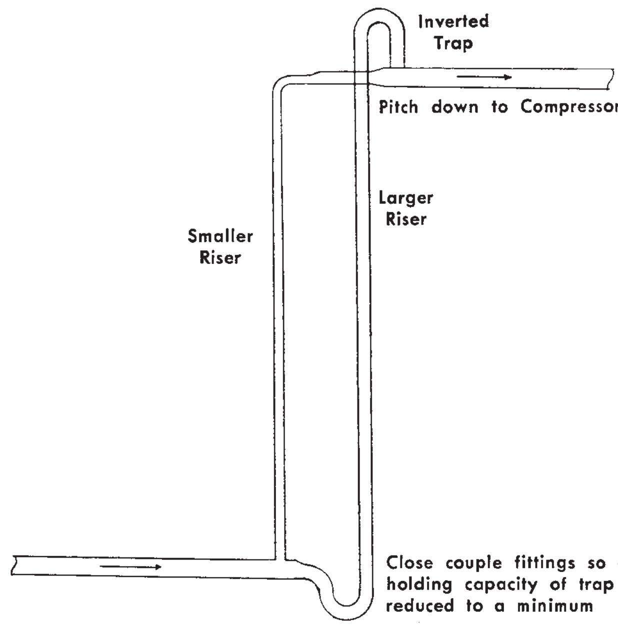
Even though the system might have a much larger design capacity, the suction line sizing must be based on the minimum capacity anticipated in operation under light load conditions after allowing for the maximum reduction in capacity from capacity control if provided.

Since the dual goals of low pressure drop and high velocities are in direct conflict, obviously compromises must be made in both areas. As a general approach, in suction line design, velocities should be kept as high as possible by sizing lines on the basis of the maximum pressure drop that can be tolerated, but in no case should gas velocity be allowed to fall below the minimum levels necessary to return oil. It is recommended that a tentative selection of suction line sizes be made on the basis of a total pressure drop equivalent to a 2°F. change in the saturated evaporating temperature. Check Figures 84 or 86 to be sure that velocities in vertical risers are satisfactory. Where refrigerant lines are lengthy, it may be desirable to use as large tubing as practical to minimize pressure drop, and Figure 85 or 87 should be checked to determine the maximum permissible horizontal line size. The final consideration must always be to maintain velocities adequate to return oil to the compressor, even if this results in a higher pressure drop than is normally desirable.

Recommended suction line sizes for varying capacities and equivalent lengths of line are given in Tables 29 to 41.

DOUBLE RISERS

On systems equipped with capacity control compressors, or where tandem or multiple compressors are used with one or more compressors cycled off for capacity control, single suction line risers may result in either unacceptably high or low gas velocities. A line properly sized for light load conditions may have too high a pressure drop at maximum load, and if the line is sized on the basis of full load conditions, then velocities may not be adequate at light load conditions to move oil through the tubing. On air conditioning applications where somewhat higher pressure drops at maximum load conditions can be tolerated without any major penalty in overall system performance, it is usually preferable to accept the



SUCTION LINE DOUBLE RISER

Figure 88

additional pressure drop imposed by a single vertical riser. But on medium or low temperature applications where pressure drop is more critical and where separate risers from individual evaporators are not desirable or possible, a double riser may be necessary to avoid an excessive loss of capacity.

A typical double riser configuration is shown in Figure 88. The two lines should be sized so that the total cross-sectional area is equivalent to the cross-section area of a single riser that would have both satisfactory gas velocity and acceptable pressure drop at maximum load conditions. The two lines normally are different in size, with the larger line trapped as shown, and the smaller line must be sized to provide adequate velocities and acceptable pressure drop when the entire minimum load is carried in the smaller riser.

In operation, at maximum load conditions gas and entrained oil will be flowing through both risers. At minimum load conditions, the gas velocity will not be high enough to carry oil up both risers. The entrained oil will drop out of the refrigerant gas flow, and accumulate in the “P” trap, forming a liquid seal. This will force all of the flow up the smaller riser, thereby raising the velocity and assuring oil circulation through the system.

For example, assume a low temperature system as follows:

Maximum capacity	150,000 BTU/hr.
Minimum capacity	50,000 BTU/hr.
Refrigerant	R-502
Evaporating Temperature	-40°F.
Equivalent length of	
piping, horizontal	125 ft.
Vertical Riser	25 ft.
Desired design pressure drop	
(equivalent to 2°F.)	1 psi

A preliminary check of the R-502 pressure drop chart, Figure 78, indicates for a 150 foot run with 150,000 BTU/hr capacity and a total pressure drop of approximately 1 psi, a 3 1/8" O.D. line is indicated. At the minimum capacity of 50,000 BTU/hr, Figure 87 shows a 3 5/8" O.D. horizontal suction line is acceptable, but Figure 84 indicates that the maximum vertical riser size is 2 1/8" O.D. Referring again to the pressure drop chart, Figure 78, the pressure drop for 150,000 BTU/hr through 2 1/8" O.D. tubing is 4 psi per 100 feet, or 1.0 psi for the 25 foot suction riser. Obviously, either a compromise must be made in accepting a greater pressure drop at maximum load conditions, or a double riser must be used.

If the pressure drop must be held to a minimum, then the size of the double riser must be determined. At maximum load conditions, a 3 1/8" O.D. riser would maintain adequate velocities, so a combination of the sizes approximating the 3 1/8" O.D. line can be selected for the double riser. The cross sectional area of the line sizes to be considered are:

3 1/8" O.D.	6.64 sq. in.
2 5/8" O.D.	4.77 sq. in.
2 1/8" O.D.	3.10 sq. in.
1 5/8" O.D.	1.78 sq. in.

At the minimum load condition of 50,000 BTU/hr., the 1 5/8" O.D. line will have a pressure drop of approximately .5 psi, and will have acceptable velocities, so a combination of 2 5/8" O.D. and 1 5/8" O.D. tubing should be used for the double riser.

In a similar fashion, double risers can be calculated for any set of maximum and minimum capacities where single risers may not be satisfactory.

SUCTION PIPING FOR MULTIPLEX SYSTEMS

It is common practice in supermarket applications to operate several fixtures, each with liquid line solenoid valve and expansion valve control, from a single compressor. Temperature control of individual fixtures is normally achieved by means of a thermostat opening and closing the liquid line solenoid valve as necessary. This type of system, commonly called multiplexing, requires careful attention to design to avoid oil return problems and compressor overheating.

Since the fixtures fed by each liquid line solenoid valve may be controlled individually, and since the load on each fixture is relatively constant during operation, individual suction lines and risers are normally run from each fixture or group of fixtures controlled by a liquid line solenoid valve for minimum pressure drop and maximum efficiency in oil return. This provides excellent control so long as the compressor is operating at its design suction pressure, but there may be periods of light load when most or all of the liquid line solenoids are closed. Unless some means of controlling compressor capacity is provided, this can result in compressor short cycling or operation at excessively low suction pressures, which can result not only in overheating the compressor, but in reducing the suction pressure to a level where the gas becomes so rarefied it can no longer return oil properly in lines sized for much greater gas density.

Because of the fluctuations in refrigeration load caused by closing of the individual liquid line solenoid valves, some means of compressor capacity control must be provided. In addition, the means of capacity control must be such that it will not allow extreme variations in the compressor suction pressure.

Where multiple compressors are used, cycling of individual compressors provides satisfactory control. Where multiplexing is done with a single compressor, a hot gas bypass system has proven to be the most satisfactory means of capacity reduction, since this allows the compressor to operate continuously at a reasonably constant suction pressure while compressor cooling can be safely controlled by means of a desuperheating expansion valve.

In all cases, the operation of the system under all possible combinations of heavy load, light load, defrost, and compressor capacity must be studied carefully to be certain that operating conditions will be satisfactory.

Close attention must be paid to piping design on multiplex systems to avoid oil return problems. Lines must be properly sized so that the minimum velocities necessary to return oil are maintained in both horizontal and vertical suction lines under minimum load conditions. Bear in mind that although a hot gas bypass maintains the suction pressure at a proper level, the refrigerant vapor being bypassed is not available in the system to aid in returning oil.

PIPING DESIGN FOR HORIZONTAL AND VERTICAL LINES

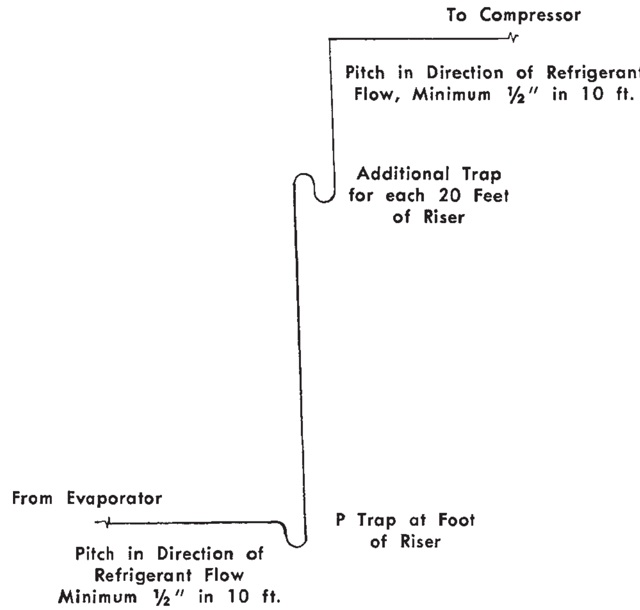
Horizontal suction and discharge lines should be pitched downward in the direction of flow to aid in oil drainage, with a downward pitch of at least ½ inch in 10 feet. Refrigerant lines should always be as short and should run as directly as possible.

Piping should be located so that access to system components is not hindered, and so that any components which could possibly require future maintenance are easily accessible. If piping must be run through boiler rooms or other areas where they will be exposed to abnormally high temperatures, it may be necessary to insulate both the suction and liquid lines to prevent excessive heat transfer into the lines.

Every vertical suction riser greater than 3 to 4 feet in height should have a "P" trap at the base to facilitate oil return up the riser as shown in Figure 89. To avoid the accumulation of large quantities of oil, the trap should be of minimum depth and the horizontal section should be as short as possible. Prefabricated wrought copper traps are available, or a trap can be made by using two street ells and one regular ell. Traps at the foot of hot gas risers are normally not required because of the easier movement of oil at higher temperatures. However, it is recommended that the discharge line from the compressor be looped to the floor prior to being run vertically upwards to prevent the drainage of oil back to the compressor head during shut down periods. See Figure 90.

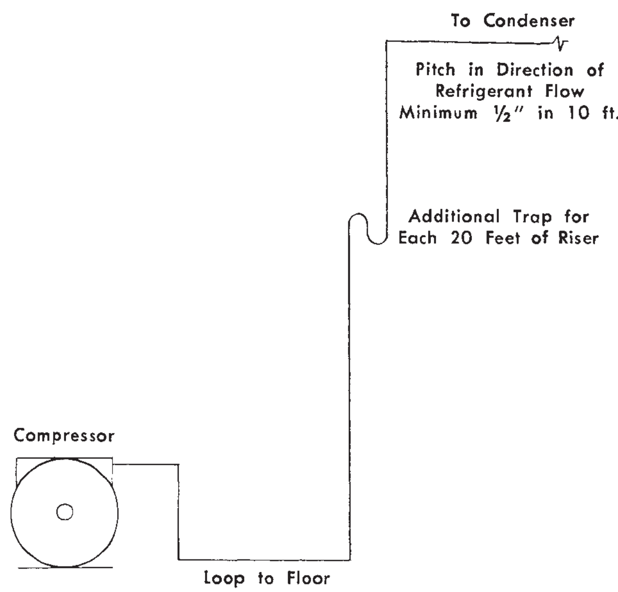
For long vertical risers in both suction and discharge lines, additional traps are recommended for each full length of pipe (approximately 20 feet) to insure proper oil movement.

In general, trapped sections of the suction line should be avoided except where necessary for oil return. Oil or liquid refrigerant accumulating in the suction line during the off cycle can return to the compressor at high velocity as liquid slugs on start up, and can break compressor valves or cause other damage.



SUCTION LINE RISERS

Figure 89



DISCHARGE LINE RISERS

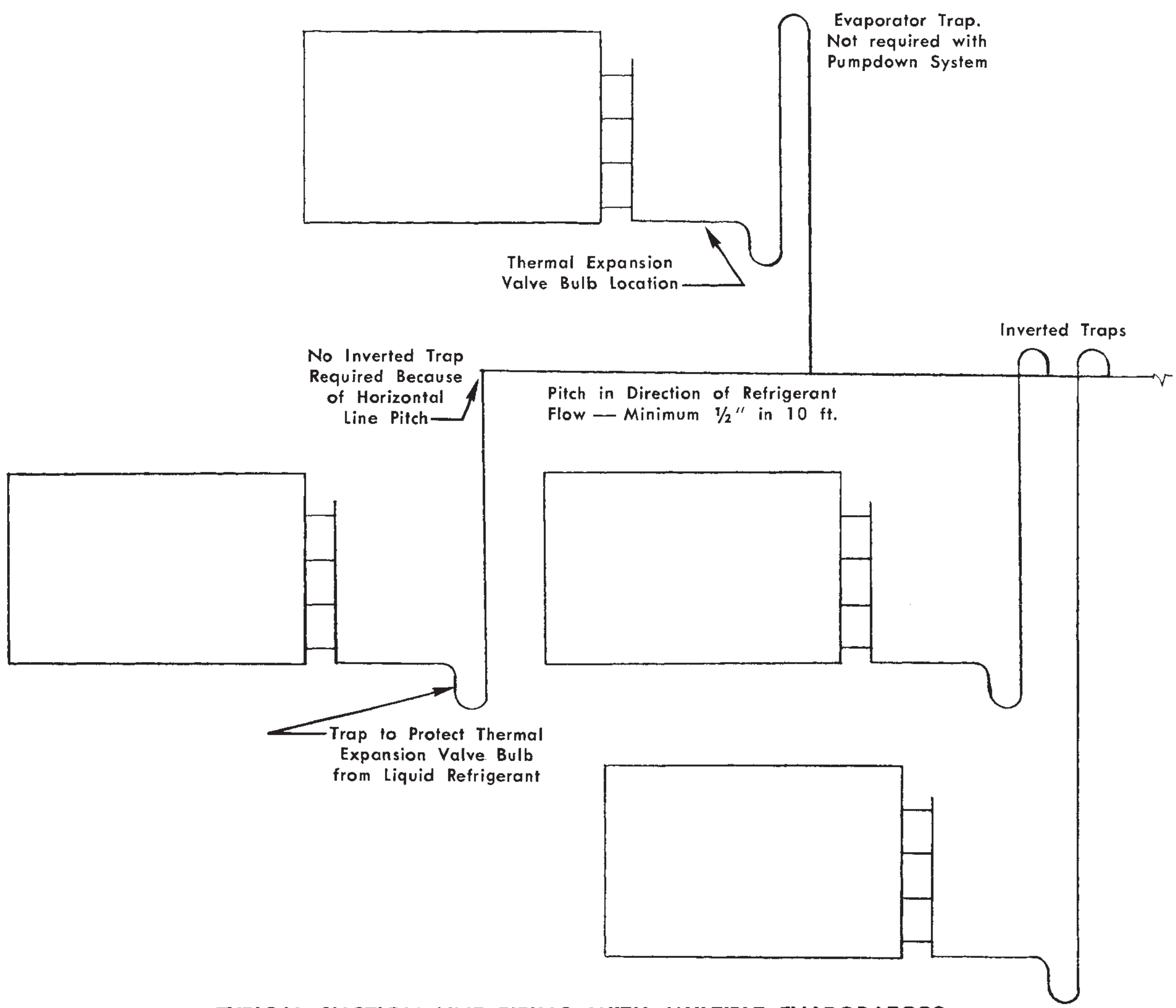
Figure 90

SUCTION LINE PIPING DESIGN AT THE EVAPORATOR

If a pumpdown control system is not used, each evaporator must be trapped to prevent liquid refrigerant from draining back to the compressor by gravity during the off cycle. Where multiple evaporators are connected to a common suction line, the connections to the common suction line must be made with inverted traps to prevent drainage from one evaporator from affecting the expansion valve bulb control of another evaporator.

Where a suction riser is taken directly upward from an evaporator, a short horizontal section of tubing and a trap should be provided ahead of the riser so that a suitable mounting for the thermal expansion valve bulb is available. The trap serves as a drain area, and helps to prevent the accumulation of liquid under the bulb which could cause erratic expansion valve operation. If the suction line leaving the evaporator is free draining or if a reasonable length of horizontal piping precedes the vertical riser, no trap is required unless necessary for oil return.

Typical evaporator connections are illustrated in Figure 91.



TYPICAL SUCTION LINE PIPING WITH MULTIPLE EVAPORATORS

Figure 91

RECEIVER LOCATION

Gas binding at the receiver can occur when the receiver is exposed to an ambient temperature higher than the condensing temperature. Heat transfer through the receiver shell causes some of the liquid in the receiver to evaporate, creating a pressure in the receiver high than in the condenser. This forces liquid refrigerant to back up into the condenser until its efficiency is reduced to the point where the condensing pressure again exceeds the pressure in the receiver.

The best remedy for this problem is to make sure the receiver is always exposed to ambient temperatures lower than the condensing temperature. If this is not possible, the receiver should be insulated to minimize heat transfer. Various types of venting arrangements for receivers have been proposed, but these require extreme care in circuiting to avoid flow problems. When the receiver is vented back to the condenser, the only force causing flow from the condenser to the receiver is gravity. Vented piping arrangements are complicated at best, and should be avoided if possible.

Even though there may not be sufficient heat transfer into a receiver to cause gas binding, on systems where the condensing temperature is lower than the ambient (for example water cooled or evaporative condensers with remote receivers) the liquid refrigerant may be warmed sufficiently in the receiver to lose most and possibly all of its subcooling. As mentioned previously, special subcooling coils or insulation may be required for proper operation under these conditions.

If a difference in temperature exists between two parts of an idle refrigeration system with interconnecting piping, this actually creates a little built-in static refrigeration system. The liquid refrigerant at the high temperature point will slowly vaporize, travel through the system as vapor, and recondense at the lowest temperature point. This most often is a matter of concern with a roof mounted remote condenser when the compressor is located in an inside machine room. If the system is idle, the sun is shining on the condenser, and the machine room is cool, then liquid is going to move out of the condenser and back down the discharge line to the machine room. Occasionally inverted traps are made in the discharge line at the condenser in the belief they will prevent this type of reverse flow. Actually with even a few degrees temperature difference, an inverted loop 20 feet high would be of no value.

However, if the receiver is located either in the machine room, or at some other point where it will not be exposed to the roof heat, reverse flow from the condenser seldom is a source of operating difficulty. The amount of refrigerant actually returning down the discharge line will be minimized and rarely if ever will this cause compressor damage if good piping practice is followed. It is possible to mount a check valve in the discharge line near the condenser as a means of preventing refrigerant backflow of this nature, but check valves in this location are noisy, expensive, and subject to damage, and should be employed only if absolutely essential.

VIBRATION AND NOISE

No matter how well the compressor is isolated, some noise and vibration will be transmitted through the piping, but both can be minimized by proper design and support of the piping.

On small units a coil of tubing at the compressor may provide adequate protection against vibration. On larger units, flexible metallic hose is frequently used. When the compressor is supported by vibration absorbing mounts allowing compressor movement, refrigerant lines should not be anchored solidly at the unit, but at a point beyond the vibration absorber, so the vibration can be isolated and not transmitted into the piping system.

The noise characteristics of a large refrigeration or air conditioning system, particularly when installed with long refrigerant lines and remote condensers, are not predictable. Variations in piping configuration, the pattern of gas flow, line sizes, operating pressures, the compressor and unit mounting, all can affect the noise generated by the system. Occasionally a particular combination of gas flow and piping will result in a resonant frequency which may amplify the sound and vibration to an undesirable level. Gas pulsation from the compressor may also be amplified in a similar manner.

If gas pulsation or resonant frequencies are encountered on a particular application, a discharge line muffler may be helpful in correcting the problem. The purpose of a muffler is to dampen the pulses of gas in the discharge line and to change the frequency to a level which is not objectionable. A muffler normally depends on multiple internal baffles and/or pressure drop to obtain an even flow of gas. In general, the application range of a muffler depends on the mass flow of gas through the muffler, so the volume and density of the refrigerant gas discharged from the compressor are both factors in muffler performance.

A given muffler may work satisfactorily on a fairly wide range of compressor sizes, but is also quite possible that a given system may require a muffler with a particular pressure drop to effectively dampen pulsations. On problem applications, trial and error may be the only final guide. While larger mufflers are often more efficient in reducing the overall level of compressor discharge noise, in order to satisfactorily dampen pulsations, smaller mufflers with a greater pressure drop are usually more effective. Adjustable mufflers are often helpful since they allow tuning of the muffler pressure characteristics to the exact system requirement.

Occasionally, a combination of operating conditions, mounting and piping arrangement may result in a resonant condition, which tends to magnify compressor pulsation and cause a sharp vibration, although noise may not be a problem. For larger Copelametic® compressors, discharge muffler plates have been developed for use when necessary to dampen excessive pulsation. The muffler plate fits between the discharge valve and the compressor body and has a number of muffling holes to provide the proper characteristics for the particular compressor displacement. The muffling holes break up the pattern of gas flow and create sufficient restriction to reduce the gas pulsation to a minimum.

When piping passes through walls or floors, precautions should be taken to see that the piping does not touch any structural members and is properly supported by hangers in order to prevent the transmission of vibration into the building. Failure to do so may result in the building structure becoming a sounding board.

Table 26 gives the maximum recommended spacing for pipe supports.

RECOMMENDED LINE SIZING TABLES

Tables 27 to 41 give recommended line sizes for single stage applications at various capacities and for equivalent lengths of pipe based on the design criteria discussed previously. (For piping recommendations on two stage systems, refer to Section 19).

Vertical suction line sizes have been selected on the basis of a total vertical rise up to 30 feet. For longer risers, individual calculations should be made since the increased pressure drop may require different line sizes and possibly the use of double risers in place of the single riser shown.

Discharge line sizes have been calculated on the basis of a nominal pressure drop of 5 psi. Vertical line sizes have been selected so that minimum velocities necessary to carry oil up the riser will be maintained under the reduced load conditions shown, and velocities have been checked to see that they do not exceed 2,700 FPM at maximum load conditions. Because of the relatively small variation in discharge line velocity over the normal refrigeration and air conditioning range, the line sizes shown may be safely used for evaporating temperatures from -40°F. to 45°F., and condensing temperatures from 80°F. to 130°F.

Liquid line sizes have been calculated on the basis of a nominal pressure drop equivalent to 2 ½° F. subcooling, and velocities have been checked to see that they do not exceed 250 FPM. Liquid lines from the condenser to receiver have been selected on the basis of 100 FPM velocity in accordance with standard industry practice in order to allow a free draining line with gas equalization where piping allows. As in the case with discharge lines, the relatively small variation in liquid line velocities over the normal refrigeration and air conditioning range allows use of the recommended line sizing for evaporating temperatures from -40°F. to 45°F., and condensing temperatures from 80°F. to 130°F.

Suction line sizes have been calculated on the basis of a nominal pressure drop equivalent to a 2°F. change in the saturated evaporating temperature. Both horizontal and vertical line sizes have been checked to see that the necessary minimum velocities are maintained under the reduced load conditions shown. Line sizes have been calculated for various evaporating temperatures, and may be safely applied for condensing temperatures from 80°F. to 130°F.

Table 26

MAXIMUM RECOMMENDED SPACING BETWEEN PIPE SUPPORTS FOR COPPER TUBING

Line Size, O.D., In.	Maximum Span, Feet
5/8	5
1 1/8	7
1 5/8	9
2 1/8	10
3 1/8	12
3 5/8	13
4 1/8	14

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Table 27

RECOMMENDED LIQUID LINE SIZES

Capacity BTU/hr	R-12					R-22					R-502				
	Condenser to Receiver	Receiver to Evaporator Equivalent Length, Ft.				Condenser to Receiver	Receiver to Evaporator Equivalent Length, Ft.				Condenser to Receiver	Receiver to Evaporator Equivalent Length, Ft.			
		50	100	150	200		50	100	150	200		50	100	150	200
6,000	3/8	3/8	3/8	3/8	3/8	3/8	1/4	3/8	3/8	3/8	3/8	1/4	3/8	3/8	3/8
12,000	1/2	3/8	3/8	1/2	1/2	1/2	3/8	3/8	3/8	3/8	1/2	3/8	1/2	1/2	1/2
18,000	1/2	1/2	1/2	1/2	1/2	1/2	3/8	3/8	1/2	1/2	5/8	1/2	1/2	1/2	1/2
24,000	5/8	1/2	1/2	1/2	5/8	5/8	3/8	1/2	1/2	1/2	5/8	1/2	5/8	5/8	5/8
36,000	5/8	1/2	5/8	5/8	5/8	5/8	1/2	1/2	1/2	1/2	7/8	1/2	5/8	5/8	5/8
48,000	7/8	1/2	5/8	5/8	7/8	7/8	1/2	5/8	5/8	5/8	7/8	5/8	5/8	5/8	7/8
60,000	7/8	5/8	5/8	7/8	7/8	7/8	1/2	5/8	5/8	5/8	7/8	5/8	7/8	7/8	7/8
75,000	7/8	5/8	7/8	7/8	7/8	7/8	1/2	5/8	5/8	5/8	7/8	5/8	7/8	7/8	7/8
100,000	1 1/8	7/8	7/8	7/8	7/8	7/8	5/8	7/8	7/8	7/8	1 1/8	7/8	7/8	7/8	7/8
150,000	1 1/8	7/8	7/8	1 1/8	1 1/8	1 1/8	7/8	7/8	7/8	7/8	1 3/8	7/8	7/8	1 1/8	1 1/8
200,000	1 3/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	7/8	7/8	1 1/8	1 1/8	1 3/8	1 1/8	1 1/8	1 1/8	1 1/8
300,000	1 5/8	1 1/8	1 1/8	1 3/8	1 3/8	1 3/8	1 1/8	1 1/8	1 1/8	1 1/8	1 5/8	1 3/8	1 3/8	1 3/8	1 3/8
400,000	1 5/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 1/8	1 1/8	1 3/8	1 3/8	1 5/8	1 3/8	1 3/8	1 3/8	1 5/8
500,000	1 5/8	1 3/8	1 3/8	1 5/8	1 5/8	1 5/8	1 1/8	1 3/8	1 3/8	1 3/8	2 1/8	1 3/8	1 3/8	1 5/8	1 5/8
600,000	2 1/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 3/8	1 3/8	1 3/8	1 5/8	2 1/8	1 5/8	1 5/8	1 5/8	1 5/8
750,000	2 1/8	1 5/8	1 5/8	1 5/8	2 1/8	2 1/8	1 5/8	1 5/8	1 5/8	1 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8

Recommended sizes are applicable with evaporating temperatures from -40° F. to 45° F. and condensing temperatures from 80° F. to 130° F.

Table 28

RECOMMENDED DISCHARGE LINE SIZES

Capacity BTU/hr	Light Load Capacity Reduction	R-12				R-22				R-502			
		Equivalent Length, Ft.				Equivalent Length, Ft.				Equivalent Length, Ft.			
		50	100	150	200	50	100	150	200	50	100	150	200
6,000	0	½	½	½	⅝*	⅜	½	½	½	½	½	½	⅝*
12,000	0	⅝	⅝	⅝	⅞	½	½	⅝	⅝	⅝	⅝	⅝	⅞
18,000	0	⅝	⅞	⅞	⅞	⅝	⅝	⅝	⅞	⅝	⅞	⅞	⅞
24,000	0	⅞	⅞	⅞	⅞	⅝	⅞	⅞	⅞	⅞	⅞	⅞	⅞
36,000	0	⅞	⅞	⅞	1⅛	⅞	⅞	⅞	⅞	⅞	⅞	1⅛	1⅛
48,000	0	⅞	1⅛	1⅛	1⅛	⅞	⅞	⅞	1⅛	⅞	1⅛	1⅛	1⅛
60,000	0	1⅛	1⅛	1⅛	1⅜	⅞	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜
	33%	1⅛	1⅛	1⅛	1⅜	⅞	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜
75,000	0	1⅛	1⅛	1⅛	1⅜	⅞	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜	1⅜
	33%	1⅛	1⅛	1⅛	1⅜	⅞	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜	1⅜
100,000	0	1⅛	1⅜	1⅜	1⅝	1⅛	1⅛	1⅜	1⅜	1⅛	1⅜	1⅜	1⅝
	33% to 50%	1⅛	1⅜	1⅜	1⅝	1⅛	1⅛	1⅜	1⅜	1⅛	1⅜	1⅜	1⅝
150,000	0	1⅜	1⅝	1⅝	2⅛	1⅛	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅝
	33% to 50%	1⅜	1⅝	1⅝	2⅛*	1⅛	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅝
	66%	1⅜	1⅝	1⅝	2⅛*	1⅛	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅝
200,000	0	1⅝	1⅝	2⅛	2⅛	1⅜	1⅜	1⅝	1⅝	1⅜	1⅝	1⅝	2⅛
	33% to 50%	1⅝	1⅝	2⅛	2⅛	1⅜	1⅜	1⅝	1⅝	1⅜	1⅝	1⅝	2⅛
	66%	1⅝	1⅝	2⅛*	2⅛*	1⅜	1⅜	1⅝	1⅝	1⅜	1⅝	1⅝	2⅛*
300,000	0	2⅛	2⅛	2⅛	2⅛	1⅜	1⅝	1⅝	2⅛	1⅝	2⅛	2⅛	2⅛
	33% to 50%	2⅛	2⅛	2⅛	2⅛	1⅜	1⅝	1⅝	2⅛	1⅝	2⅛	2⅛	2⅛
	66%	2⅛	2⅛	2⅛	2⅛	1⅜	1⅝	2⅛*	2⅛*	1⅝	2⅛	2⅛	2⅛
400,000	0	2⅛	2⅛	2⅛	2⅝	1⅝	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅝
	33% to 66%	2⅛	2⅛	2⅛	2⅝	1⅝	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅝
500,000	0	2⅝	2⅝	2⅝	2⅝	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅝	2⅝
	33% to 50%	2⅝	2⅝	2⅝	2⅝	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅝	2⅝
	66%	2⅝	2⅝	2⅝	2⅝	2⅛	2⅛	2⅛	2⅛	2⅝	2⅝	2⅝	2⅝
600,000	0	2⅝	2⅝	2⅝	3⅛	2⅛	2⅛	2⅛	2⅝	2⅛	2⅝	2⅝	3⅛
	33% to 50%	2⅝	2⅝	2⅝	3⅛*	2⅛	2⅛	2⅛	2⅝	2⅛	2⅝	2⅝	3⅛*
	66%	2⅝	2⅝	3⅛*	3⅛*	2⅛	2⅛	2⅛	2⅝*	2⅛	2⅝	2⅝	3⅛*
750,000	0	3⅛	3⅛	3⅛	3⅛	2⅛	2⅝	2⅝	2⅝	2⅝	2⅝	2⅝	3⅛
	33% to 50%	3⅛	3⅛	3⅛	3⅛	2⅛	2⅝	2⅝	2⅝	2⅝	2⅝	2⅝	3⅛
	66%	3⅛*	3⅛*	3⅛*	3⅛*	2⅛	2⅝	2⅝	2⅝	2⅝	2⅝	2⅝	3⅛*

* Use one line size smaller for vertical riser

Recommended sizes are applicable for applications with evaporating temperatures from -40° F. to 45° F. and condensing temperatures from 80° F. to 130° F.

Table 29

RECOMMENDED SUCTION LINE SIZES

R-12 40° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8
12,000	0	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
18,000	0	7/8	7/8	7/8	7/8	1 1/8	7/8	1 1/8	1 1/8
24,000	0	7/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8
36,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 3/8
48,000	0	1 1/8	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 5/8
60,000	0 to 33%	1 1/8	1 1/8	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 5/8
75,000	0 to 33%	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	1 5/8	1 5/8
100,000	0 to 50%	1 3/8	1 3/8	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
150,000	0 to 33%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	2 1/8
	50% to 66%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
200,000	0	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 5/8
	33% to 50%	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
	66%	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8
300,000	0 to 50%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 5/8
	66%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
400,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	3 1/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
500,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 5/8	3 1/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8
600,000	0 to 66%	3 1/8	2 5/8	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8
750,000	0 to 66%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	4 1/8	3 5/8

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

H - Horizontal
V - Vertical

Table 30
RECOMMENDED SUCTION LINE SIZES

R-12 25° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft. 50 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	0	5/8	5/8	7/8	5/8	7/8	5/8	7/8	5/8
12,000	0	7/8	7/8	7/8	7/8	1 1/8	7/8	1 1/8	7/8
18,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8
24,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
36,000	0	1 3/8	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8
48,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 5/8	1 5/8	1 5/8
60,000	0 to 33%	1 5/8	1 3/8	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
75,000	0 to 33%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
100,000	0 to 33%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	2 1/8	2 1/8	2 1/8
	50%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
150,000	0 to 33%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8
	50% to 66%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	1 5/8 *2 1/8	2 5/8	1 5/8 *2 1/8
200,000	0 to 50%	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
	66%	2 5/8	2 1/8	2 5/8	1 5/8 *2 1/8	2 5/8	1 5/8 *2 1/8	2 5/8	1 5/8 *2 1/8
300,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8	3 5/8	2 5/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	1 5/8 *2 5/8	3 1/8	1 5/8 *2 5/8
400,000	0 to 50%	3 1/8	3 1/8	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8
	66%	3 1/8	2 5/8	3 1/8	1 5/8 *2 5/8	3 1/8	1 5/8 *2 5/8	3 1/8	1 5/8 *2 5/8

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

* Double Riser

H - Horizontal
V - Vertical

Table 31
RECOMMENDED SUCTION LINE SIZES
R-12 15° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft. 50 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	0	⅞	⅞	⅞	⅞	⅞	⅞	⅞	⅞
12,000	0	⅞	⅞	1⅛	⅞	1⅛	⅞	1⅛	⅞
18,000	0	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛
24,000	0	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛	1⅜	1⅜
36,000	0	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜	2⅛	1⅝
48,000	0	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜	2⅛	1⅝
60,000	0 to 33%	1⅝	1⅝	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝
75,000	0 to 33%	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅝	1⅝
100,000	0 to 33%	2⅛	1⅝	2⅛	1⅝	2⅝	1⅝	2⅝	2⅛
	50%	2⅛	1⅝	2⅛	1⅝	2⅝	1⅝	2⅝	1⅝
150,000	0 to 33%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	3⅛	2⅛
	50% to 66%	2⅛	2⅛	2⅝	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
200,000	0 to 50%	2⅝	2⅛	2⅝	2⅝	2⅝	2⅝	3⅛	2⅝
	66%	2⅝	2⅛	3⅛	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
300,000	0 to 50%	3⅛	2⅝	3⅛	2⅝	3⅛	3⅛	3⅝	3⅛
	66%	3⅛	2⅝	3⅛	2⅝	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝
400,000	0 to 50%	3⅛	2⅝	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛
	66%	3⅛	2⅝	3⅛	1⅝*2⅝	3⅝	1⅝*2⅝	3⅝	1⅝*2⅝

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

H - Horizontal
V - Vertical

* Double Riser

Table 32
RECOMMENDED SUCTION LINE SIZES
R-12 -20° F. Evaporating Temperatures

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft. 50 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	0	⅞	⅞	1⅛	⅞	1⅛	⅞	1⅛	⅞
12,000	0	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛	1⅜	1⅛
18,000	0	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜
24,000	0	1⅜	1⅜	1⅝	1⅝	1⅝	1⅝	2⅛	1⅝
36,000	0	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
48,000	0	2⅛	1⅝	2⅛	1⅝	2⅛	2⅛	2⅝	2⅛
60,000	0 to 33%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛
75,000	0 to 33%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	3⅛	2⅛
100,000	0 to 50%	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛	3⅛	2⅛
150,000	0 to 50%	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝	3⅝	2⅝
	66%	2⅝	1⅝*2⅛	3⅛	1⅝*2⅛	3⅛	1⅝*2⅛	3⅝	1⅝*2⅝

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

H - Horizontal
V - Vertical

* Double Riser

Table 33
RECOMMENDED SUCTION LINE SIZES
R-12 -40° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛
12,000	0	1⅜	1⅛	1⅝	1⅜	1⅝	1⅜	1⅝	1⅜
18,000	0	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜	2⅛	1⅝
24,000	0	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅝	1⅝
36,000	0	2⅛	2⅛	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛
48,000	0	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛
60,000	0 to 33%	2⅝	2⅛	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛
75,000	0	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝	3⅛	2⅝
	33%	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛
100,000	0 to 33%	3⅛	2⅝	3⅝	2⅝	3⅝	2⅝	3⅝	2⅝
	50%	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝	3⅝	1⅝*2⅝	3⅝	1⅝*2⅝

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

H - Horizontal
V - Vertical

* Double Riser

Table 34
RECOMMENDED SUCTION LINE SIZES
R-22 40° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	½	½	½	½	⅝	½	⅝	½
12,000	0	⅝	⅝	⅝	⅝	⅞	⅝	⅞	⅝
18,000	0	⅞	⅞	⅞	⅞	⅞	⅞	⅞	⅞
24,000	0	⅞	⅞	⅞	⅞	⅞	⅞	1⅛	⅞
36,000	0	⅞	⅞	1⅛	⅞	1⅛	⅞	1⅛	1⅛
48,000	0	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛
60,000	0 to 33%	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛
75,000	0 to 33%	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛	1⅝	1⅜
100,000	0 to 50%	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜
150,000	0 to 66%	1⅜	1⅜	1⅝	1⅝	1⅝	1⅝	2⅛	1⅝
200,000	0 to 66%	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
300,000	0 to 50%	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅝	2⅛
	66%	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛
400,000	0 to 66%	2⅛	2⅛	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛
500,000	0 to 66%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅝
600,000	0 to 66%	2⅝	2⅝	2⅝	2⅝	2⅝	2⅝	3⅛	2⅝
750,000	0 to 66%	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝	3⅛	2⅝

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.

H - Horizontal
V - Vertical

Table 35

RECOMMENDED SUCTION LINE SIZES

R-22 25° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	½	½	⅝	⅝	⅝	⅝	⅝	⅝
12,000	0	⅝	⅝	⅞	⅝	⅞	⅝	⅞	⅞
18,000	0	⅞	⅞	⅞	⅞	⅞	⅞	1⅛	⅞
24,000	0	⅞	⅞	⅞	⅞	1⅛	⅞	1⅛	⅞
36,000	0	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛
48,000	0	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛
60,000	0 to 33%	1⅛	1⅛	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜
75,000	0 to 33%	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜
100,000	0 to 50%	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜	1⅝	1⅜
150,000	0 to 50%	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
	66%	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝
200,000	0 to 50%	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛	2⅛
	66%	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝
300,000	0 to 50%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅝
	66%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛
400,000	0 to 50%	2⅝	2⅛	2⅝	2⅛	3⅛	2⅛	3⅛	2⅛
	66%	2⅝	2⅛	2⅝	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
500,000	0 to 66%	2⅝	2⅝	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝
600,000	0 to 66%	2⅝	2⅝	3⅛	2⅝	3⅝	2⅝	3⅝	2⅝
750,000	0 to 66%	3⅛	3⅛	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

Table 36
RECOMMENDED SUCTION LINE SIZES

R-22 15° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8
12,000	0	5/8	5/8	7/8	5/8	7/8	5/8	7/8	7/8
18,000	0	7/8	7/8	7/8	7/8	1 1/8	7/8	1 1/8	7/8
24,000	0	7/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
36,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8
48,000	0	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
60,000	0 to 33%	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8
75,000	0 to 33%	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8
100,000	0 to 50%	1 3/8	1 3/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
150,000	0 to 50%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
	66%	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
200,000	0 to 50%	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
	66%	2 1/8	1 3/8*1 5/8	2 1/8	1 3/8*1 5/8	2 1/8	1 3/8*1 5/8	2 1/8	1 3/8*1 5/8
300,000	0 to 50%	2 1/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
	66%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	1 5/8*2 1/8
400,000	0 to 50%	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
	66%	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	1 5/8*2 1/8	3 1/8	1 5/8*2 1/8
500,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
600,000	0 to 66%	3 1/8	2 5/8	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8
750,000	0 to 66%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

Table 37
RECOMMENDED SUCTION LINE SIZES
R-22 -20° F. Evaporating Temperature

Capacity BTU/hr.	Equivalent Length, Ft.							
	50		100		150		200	
	H	V	H	V	H	V	H	V
6,000	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
12,000	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8
18,000	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
24,000	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8
36,000	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
48,000	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.

H - Horizontal
V - Vertical

Table 38
RECOMMENDED SUCTION LINE SIZES
R-502 25° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	5/8	5/8	5/8	5/8	5/8	5/8	5/8	5/8
12,000	0	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
18,000	0	7/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
24,000	0	7/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	1 1/8
36,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
48,000	0	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 5/8	1 3/8
60,000	0 to 33%	1 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 3/8	1 5/8	1 3/8
75,000	0 to 33%	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	1 5/8	1 5/8
100,000	0 to 33%	1 3/8	1 3/8	1 5/8	1 5/8	1 5/8	1 5/8	2 1/8	1 5/8
	50%	1 3/8	1 3/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
150,000	0 to 50%	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
	66%	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
200,000	0 to 50%	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
	66%	2 1/8	1 3/8 * 1 5/8	2 1/8	1 3/8 * 1 5/8	2 1/8	1 3/8 * 1 5/8	2 1/8	1 3/8 * 1 5/8
300,000	0 to 50%	2 1/8	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8
	66%	2 1/8	2 1/8	2 5/8	1 5/8 * 2 1/8	2 5/8	1 5/8 * 2 1/8	2 5/8	1 5/8 * 2 1/8
400,000	0 to 50%	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
	66%	2 5/8	1 5/8 * 2 1/8	2 5/8	1 5/8 * 2 1/8	2 5/8	1 5/8 * 2 1/8	2 5/8	1 5/8 * 2 1/8
500,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	3 1/8
	66%	2 5/8	2 5/8	3 1/8	1 5/8 * 2 1/8	3 1/8	1 5/8 * 2 1/8	3 1/8	1 5/8 * 2 1/8
600,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8	3 5/8	3 1/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8	3 5/8	1 5/8 * 2 5/8
750,000	0 to 50%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	4 1/8	3 1/8
	66%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

Table 39
RECOMMENDED SUCTION LINE SIZES

R-502 15° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	⅝	⅝	⅝	⅝	⅝	⅝	⅝	⅝
12,000	0	⅞	⅞	⅞	⅞	⅞	⅞	⅞	⅞
18,000	0	⅞	⅞	⅞	⅞	1⅛	⅞	1⅛	⅞
24,000	0	1⅛	⅞	1⅛	⅞	1⅛	⅞	1⅛	1⅛
36,000	0	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛	1⅜	1⅛
48,000	0	1⅜	1⅜	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜
60,000	0 to 33%	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜	1⅝	1⅜
75,000	0 to 33%	1⅜	1⅜	1⅝	1⅜	1⅝	1⅜	1⅝	1⅝
100,000	0 to 33%	1⅝	1⅝	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝
	50%	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝
150,000	0 to 50%	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
	66%	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝	1⅝
200,000	0 to 50%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛
	66%	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝	2⅛	1⅜*1⅝
300,000	0 to 50%	2⅛	2⅛	2⅝	2⅛	3⅛	2⅛	3⅛	2⅝
	66%	2⅛	2⅛	2⅝	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
400,000	0 to 50%	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝	3⅛	2⅝
	66%	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝	2⅝	1⅝*2⅛
500,000	0 to 50%	2⅝	2⅝	3⅛	2⅝	3⅝	2⅝	3⅝	3⅛
	66%	2⅝	2⅝	3⅛	2⅝	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝
600,000	0 to 50%	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛	3⅝	3⅝
	66%	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛	3⅝	2⅛*3⅛
750,000	0 to 50%	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛	4⅛	3⅝
	66%	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛	3⅝	2⅛*3⅛

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

Table 40

RECOMMENDED SUCTION LINE SIZES

R-502 -20° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft.							
		50		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
12,000	0	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8
18,000	0	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 5/8	1 1/8
24,000	0	1 1/8	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8
36,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 5/8	2 1/8	1 5/8
48,000	0	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
60,000	0 to 33%	1 5/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
75,000	0 to 33%	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
100,000	0 to 33%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
	50%	2 1/8	1 3/8*1 5/8	2 5/8	1 3/8*1 5/8	2 5/8	1 3/8*1 5/8	2 5/8	1 3/8*1 5/8
150,000	0 to 50%	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
	66%	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	1 5/8*2 1/8	2 5/8	1 5/8*2 1/8
200,000	0 to 50%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8
	66%	2 5/8	1 5/8*2 1/8	3 1/8	1 5/8*2 1/8	3 1/8	1 5/8*2 1/8	3 1/8	1 5/8*2 1/8
300,000	0 to 50%	3 1/8	2 5/8	3 1/8	3 1/8	3 5/8	3 1/8	4 1/8	3 1/8
	66%	3 1/8	2 5/8	3 1/8	1 5/8*2 5/8	3 5/8	1 5/8*2 5/8	3 5/8	1 5/8*2 5/8

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

Table 41

RECOMMENDED SUCTION LINE SIZES

R-502 -40° F. Evaporating Temperature

Capacity BTU/hr.	Light Load Capacity Reduction	Equivalent Length, Ft. 50 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	0	7/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	7/8
12,000	0	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
18,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8
24,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	2 1/8	1 3/8
36,000	0	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
48,000	0	2 1/8	1 5/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
60,000	0	2 5/8	1 5/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
	33%	2 5/8	1 5/8	2 5/8	1 3/8 *1 5/8	2 5/8	1 3/8 *1 5/8	2 5/8	1 3/8 *1 5/8
75,000	0 to 33%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
100,000	0 to 50%	2 1/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
150,000	0 to 33%	2 5/8	2 5/8	3 1/8	2 5/8	3 5/8	2 5/8	3 5/8	2 5/8
	50% to 66%	3 1/8	2 1/8	3 1/8	1 5/8 *2 1/8	3 5/8	1 5/8 *2 1/8	3 5/8	1 5/8 *2 1/8
200,000	0 to 33%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	4 1/8	3 1/8
	50% to 66%	3 1/8	1 5/8 *2 5/8	3 5/8	1 5/8 *2 5/8	3 5/8	1 5/8 *2 5/8	3 5/8	1 5/8 *2 5/8

Recommended sizes are applicable with condensing temperatures from 80° F. to 130° F.
* Double Riser

H - Horizontal
V - Vertical

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 rarefied 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.

Emerson Climate Technologies, Inc. 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 the 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 Emerson Climate Technologies, Inc. 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, recycling 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 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. The 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 increase, the volumetric efficiency decreases.

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.

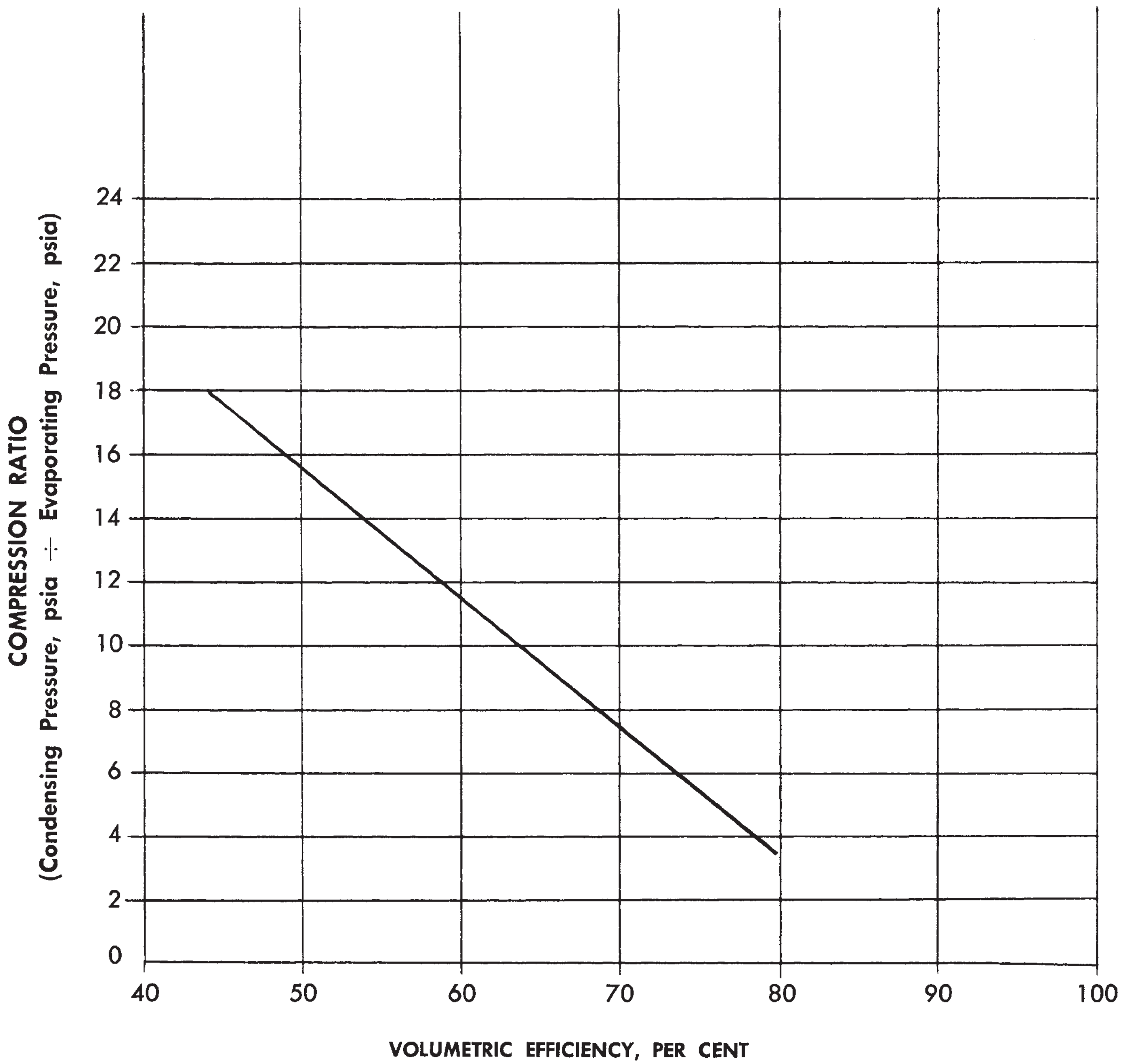
The second factor in the loss of efficiency is the high temperature of the cylinder walls resulting from the heat of compression. As the compression ratio increases, the heat of compression increases, and the cylinders and head of the compressor become very hot. Suction gas entering the cylinder on the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

(continued on p. 19-4)

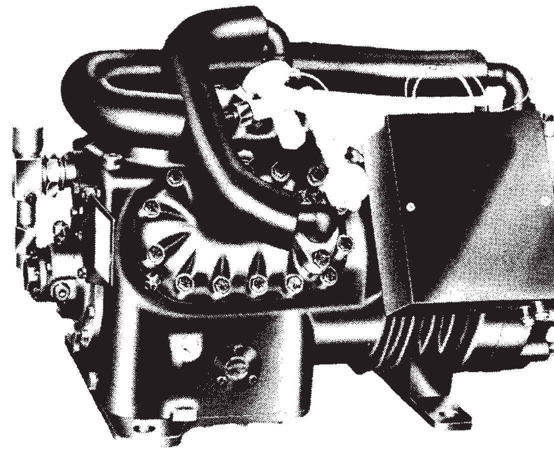


TYPICAL SINGLE STAGE LOW TEMPERATURE COMPRESSOR EFFICIENCY CURVE

Figure 92

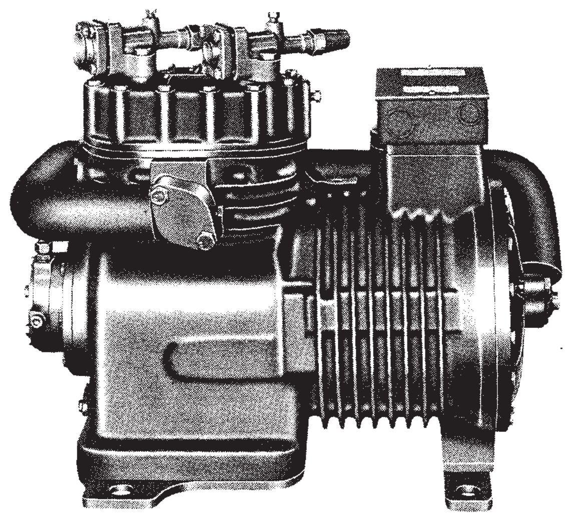
ratio of 16 to 1, the compression ratio of each stage might be 4 to 1; or compression ratios of 4 to 1 and 5 to 1 in separate stages will result in a total compression ratio of 20 to 1.

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® brand 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

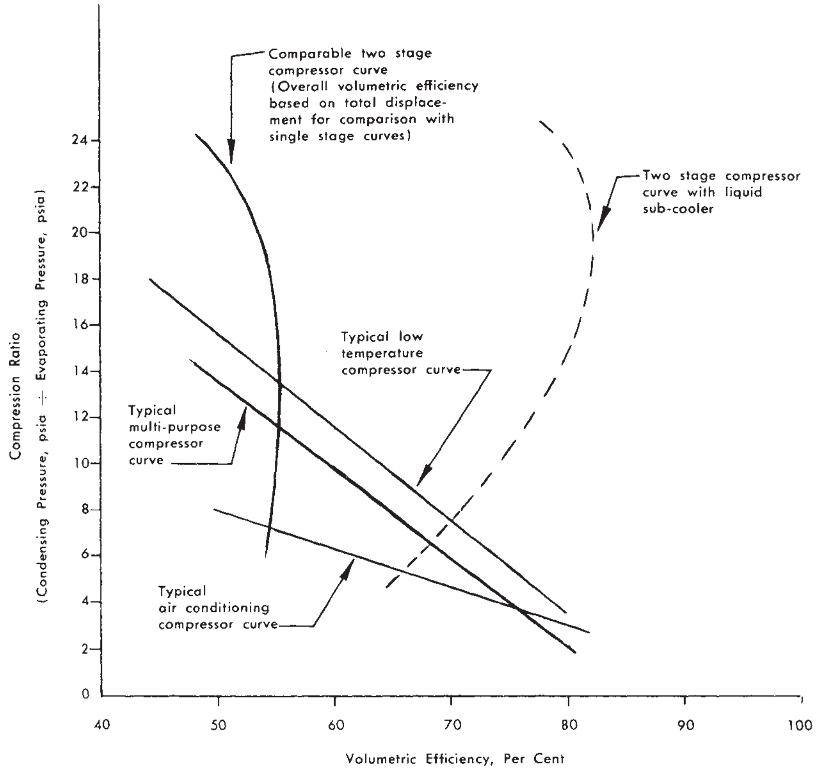
Figure 94

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 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 compres-



TYPICAL COMPRESSOR VOLUMETRIC EFFICIENCY CURVES

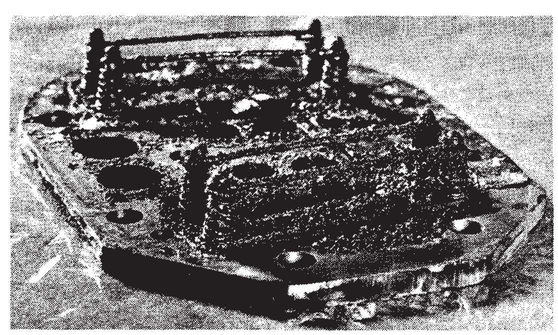
Figure 95

sor 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® brand 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.

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

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%

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

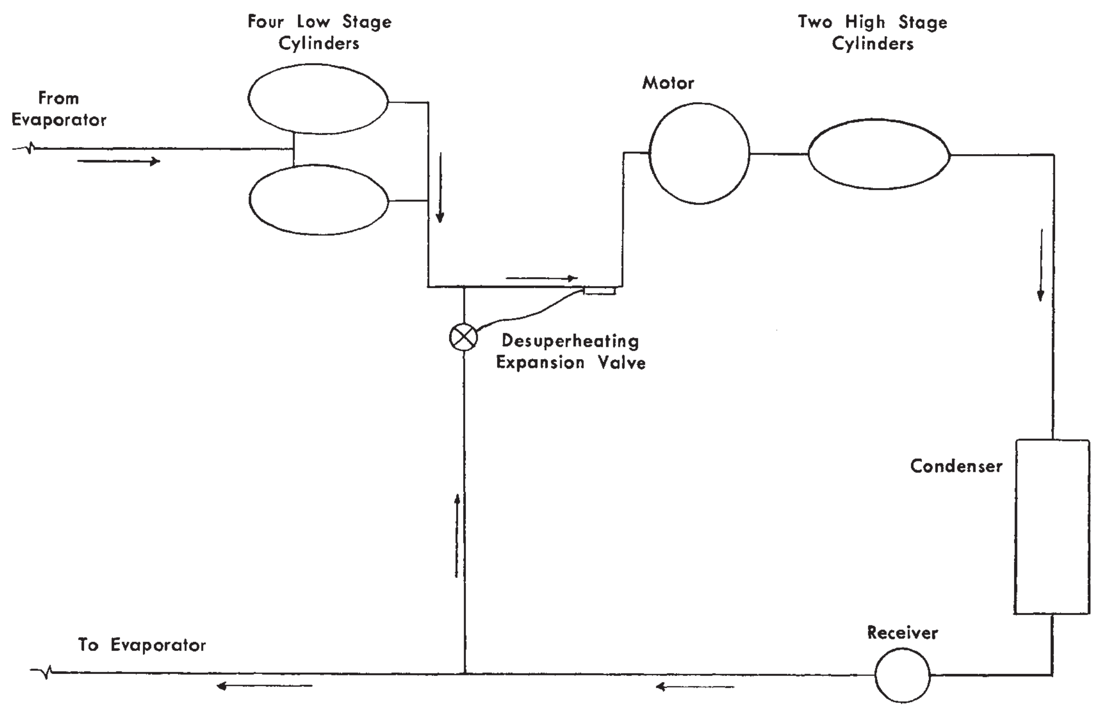
To prevent leakage during the off period, a solenoid valve must be placed in the liquid supply line immediately ahead of the desuperheating expansion valve. It should be wired so as to be open when the motor is running and closed when not running. A toggle switch placed in the electric line to the solenoid 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® brand two stage condensing units, and are strongly recommended on all two stage compressors.

The oil separator will provide some increase in refrigerat-



SCHEMATIC TWO STAGE SYSTEM

Figure 97

ing 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 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

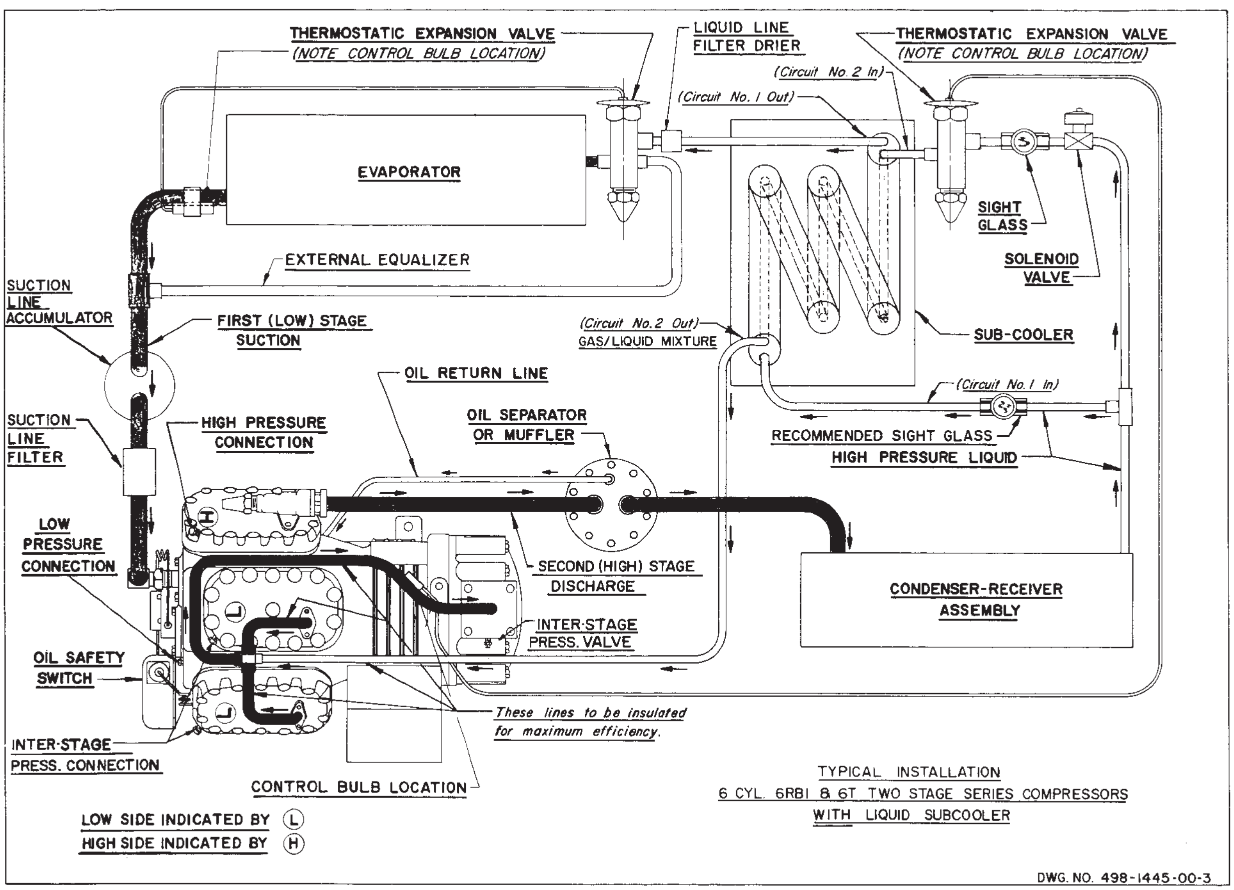
Expansion valves currently supplied as original equipment with Copeland® brand two stage compressors are of the non-adjustable superheat type. In the event of valve failure, standard field replacement valves with adjustable superheat which have been approved by Emerson Climate Technologies, Inc. may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



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

Figure 98

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 state 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 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.

(continued on p. 19-13)

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 Equivalent Length, Ft.				With Liquid Subcooler 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 Equivalent Length, Ft.				With Liquid Subcooler 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	1⅛	⅞	1⅛	⅞	1⅛	⅞	1⅛	⅞
9,000	1⅜	⅞	1⅜	⅞	1⅜	⅞	1⅜	⅞
12,000	1⅝	1⅛	1⅝	1⅛	1⅝	1⅛	1⅝	1⅛
18,000	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜
24,000	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜
36,000	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝
48,000	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛	2⅝	2⅛
60,000	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛
72,000	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛

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	1⅛	⅞	1⅛	⅞	1⅛	⅞	1⅛	⅞
9,000	1⅛	⅞	1⅛	⅞	1⅛	⅞	1⅛	⅞
12,000	1⅜	⅞	1⅜	⅞	1⅜	⅞	1⅜	⅞
18,000	1⅝	1⅛	1⅝	1⅛	1⅝	1⅛	1⅝	1⅛
24,000	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜	2⅛	1⅜
36,000	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
48,000	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝
60,000	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝	2⅝	1⅝
72,000	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛	3⅛	2⅛

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

H - Horizontal V - Vertical

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 liquefied 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 non-operating periods, an expansion tank as shown 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).

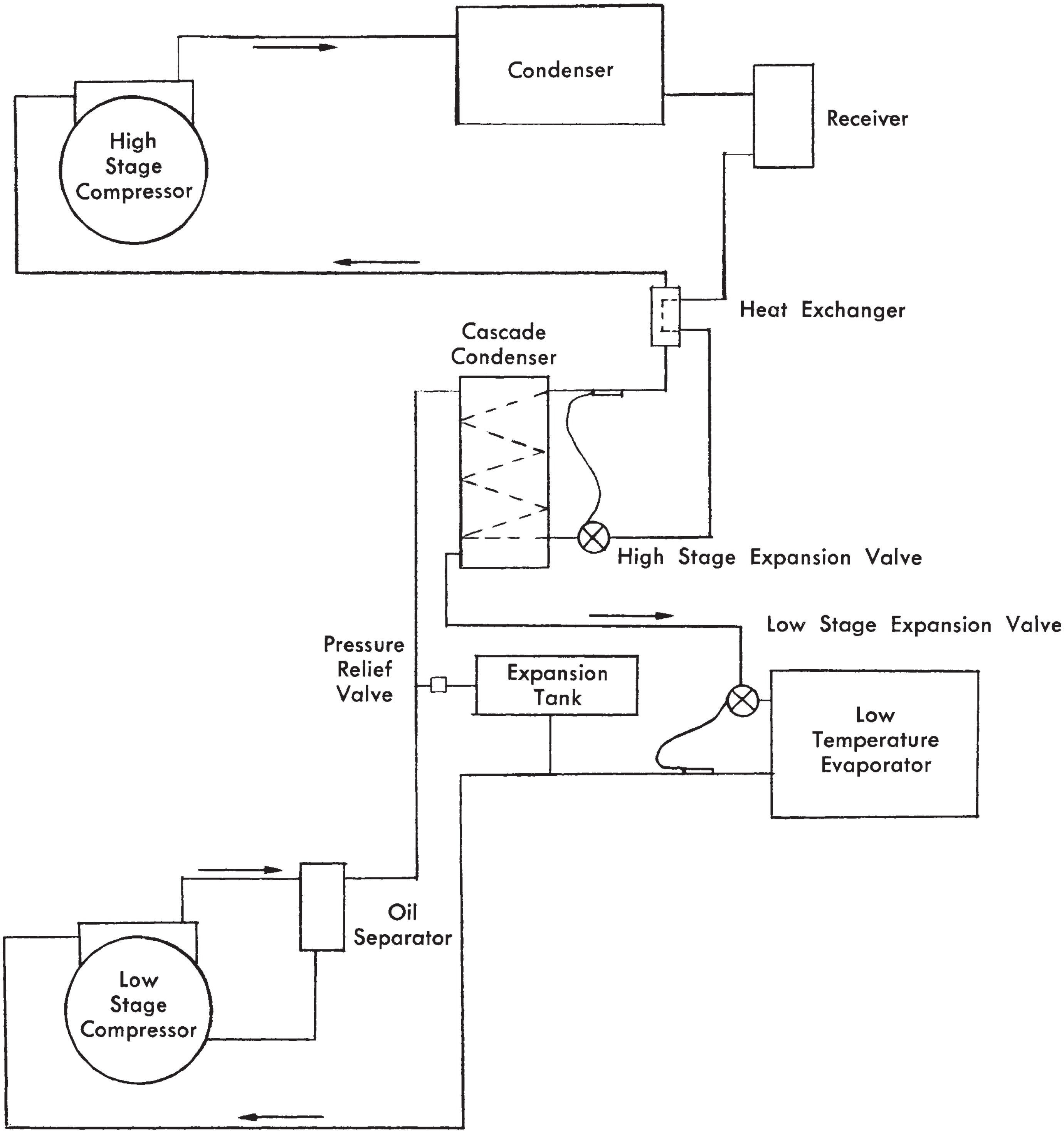
Normally the expansion tank is located in the low pressure side of the system, with a relief valve from the high pressure side of the system discharging into the tank. The sizing of the tank is determined from the total refrigerant charge, the internal volume of the system, the maximum pressure desired, 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.

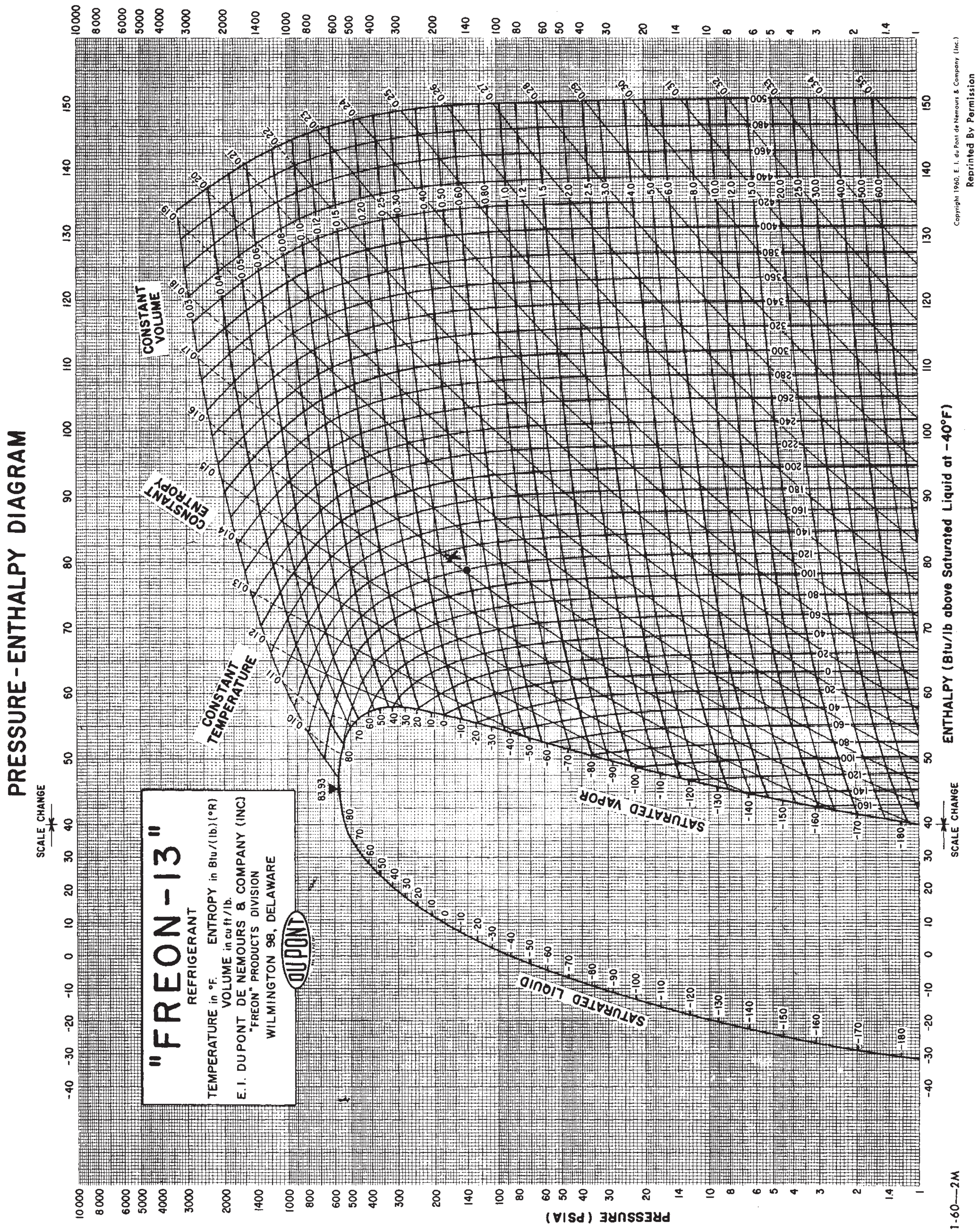
Various means of capacity control are employed, usually by means of hot gas bypass. Care must be taken to insure proper compressor motor cooling and to avoid liquid floodback to the compressor.

(continued on p. 19-16)



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. DU PONT 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.)
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1-60—2M

Figure 100

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 state 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® brand compressors are not tested with the refrigerants normally used in the low stage of cascade systems. Although many models of Copeland® brand 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 Emerson

Climate Technologies, Inc. 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 test 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 form 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 factory are charged with naphthenic 150 viscosity refrigeration oil. A complete list of acceptable refrigerants and oils is available on form #93-11. The naphthenic 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 Emerson Climate Technologies, Inc. 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 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 affect 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 the 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 AIR IN A 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.

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 non-stick coating 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 interruptible 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 airflow 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. $^5/_8$" evaporator tubes are acceptable, but ½" are preferred, and $^3/_8$" 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, place 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
5/8" O.D.	12 sq. ft.	32 sq. ft.	15 sq. ft.	32 sq. ft.
3/4" O.D.	17 sq. ft.	40 sq. ft.	22 sq. ft.	40 sq. ft.
7/8" 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.

5/8" 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	12.5	sq. ft.
	(2 - 24" x 60" plates	20	sq. ft.

3/4" O.D. Tubing

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

7/8" O.D. Tubing

Circuit			
A-Series	(1 - 30" x 60" plate	12.5	sq. ft.
	(2 - 24" x 60" plates	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 a 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 shut-down 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 1 ½ 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 is 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.

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.

Emerson Climate Technologies, Inc. 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 than 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 components 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 manufacturer, 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 form 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 $^{7}/_{8}$" 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 $^{5}/_{8}$" 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 pressure 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 the 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 a naphthenic 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 by-passing 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 many 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. Both of these conditions can cause compressor damage and ultimate failure.

Two different types of unloading are used on Copeland® brand compressors, internal and external.

INTERNAL CAPACITY CONTROL VALVES

A schematic illustration of the Copeland® brand internal unloading valve is shown in Figure 101.

In the normal operating position with the solenoid valve de-energized, the needle valve is seated on the lower port, and the unloading plunger chamber is exposed to suction pressure through the suction pressure port. Since the face of the plunger is open to the suction chamber, the gas pressures across the plunger are equalized, and the plunger is held in the open position by the spring.

When the solenoid valve is energized, the needle valve is seated on the upper port, and the unloading plunger chamber is exposed to discharge pressure through the discharge pressure port. The differential between discharge and suction pressure forces the plunger down, sealing the suction port in the valve plate, thus preventing the entrance of suction vapor into the unloaded cylinders.

With the suction port sealed, the cylinder pumps down into a vacuum until it reaches a point where no pumping action occurs.

EXTERNAL CAPACITY CONTROL VALVES

For Copeland® brand three cylinder compressors, a solenoid operated external bypass valve is used for unloading, as shown in Figure 101.

Copelametic® compressors with external capacity control have a bypass valve so arranged that the unloaded cylinder is isolated from the discharge pressure created by the unloaded cylinders. The bypass valve connects the discharge ports of the unloaded cylinder to the compressor suction chamber. Since the piston and cylinder do not work 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.

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® brand 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.

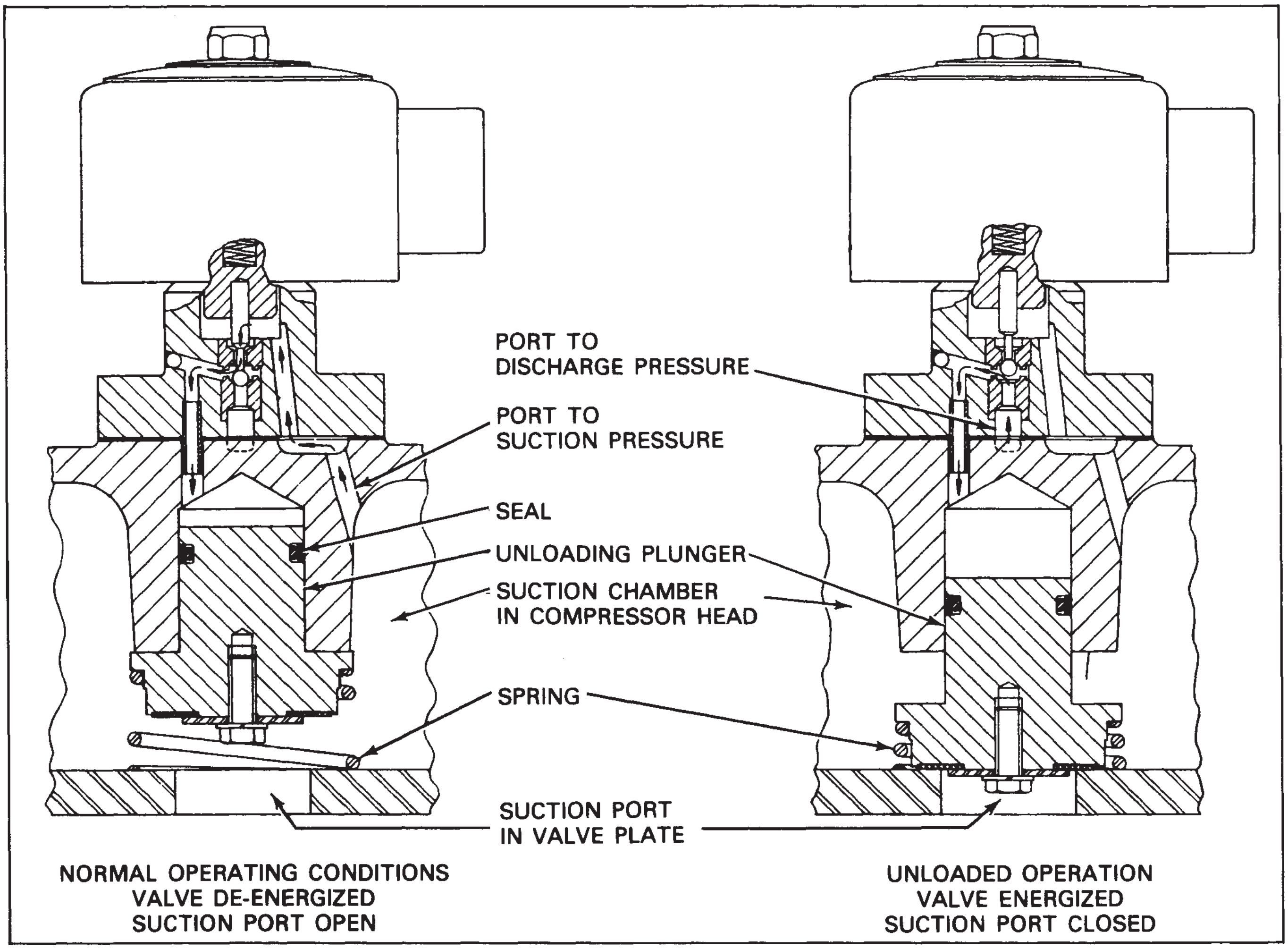
HOT GAS BYPASS

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.

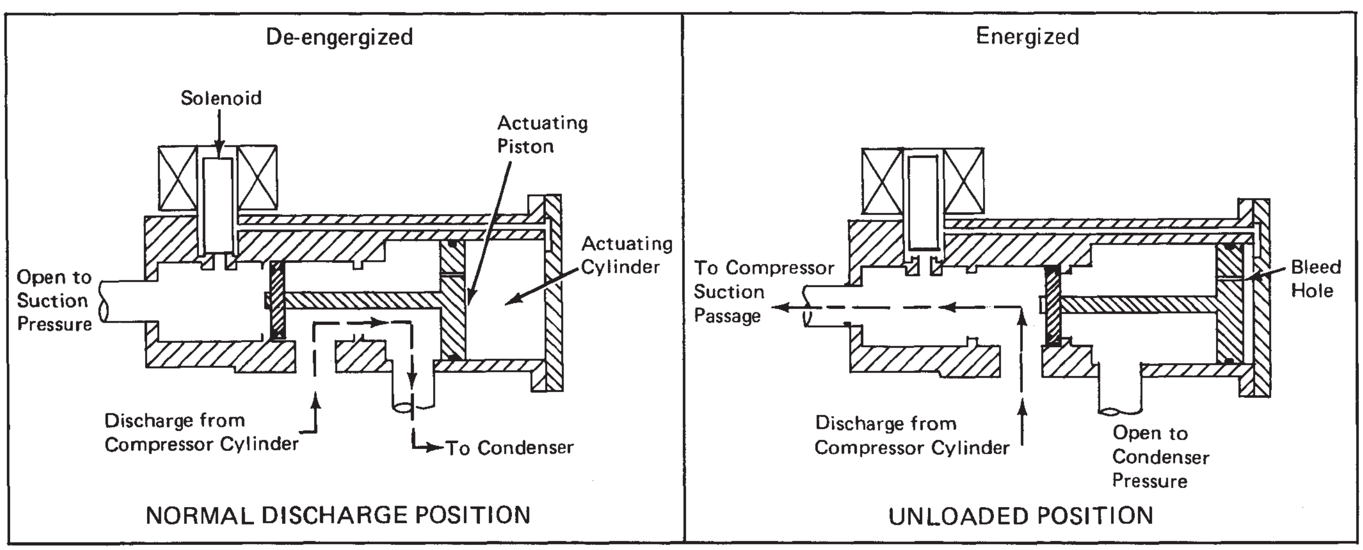
All hot gas bypass valves operate on a similar principle. They open in response to a decrease in downstream pressure, and modulate from fully open to fully closed over a given range. Introduction of the hot, high pressure gas into the low pressure side of the system at a metered rate prevents the compressor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

(continued on p. 21-3)



SCHEMATIC OPERATION OF
INTERNAL UNLOADER VALVE



SCHEMATIC OPERATION OF
EXTERNAL UNLOADER VALVE

Figure 101

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

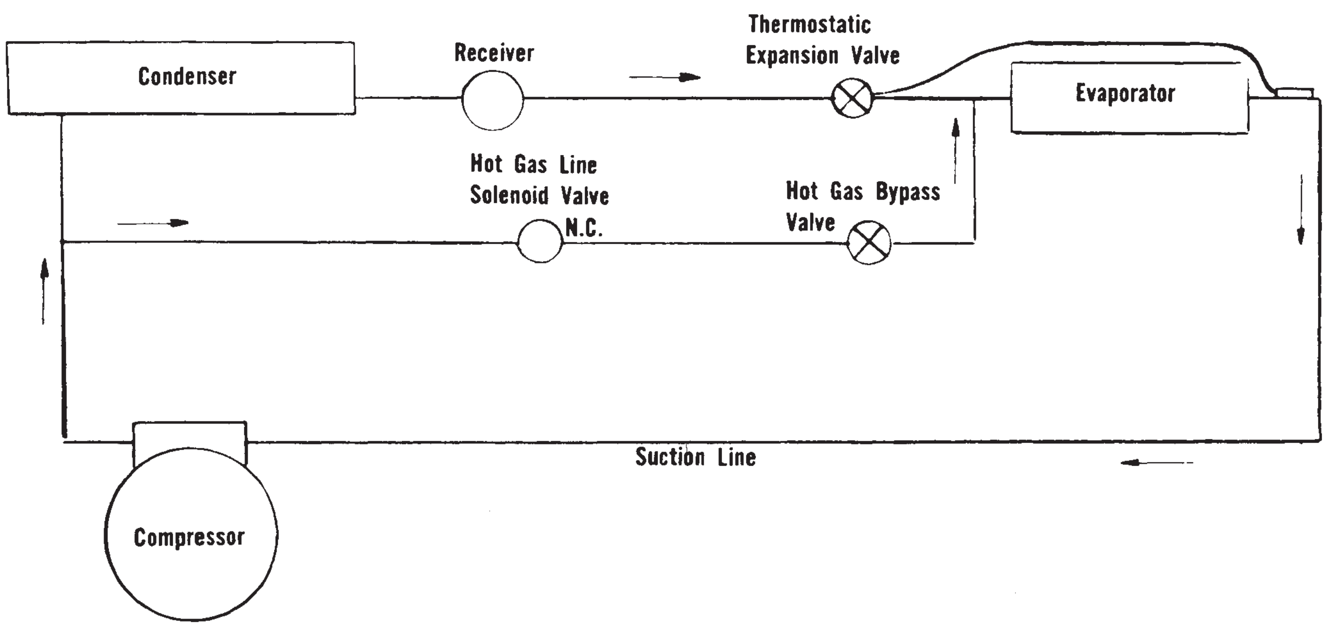
On single evaporator, close connected systems, it is frequently possible to introduce the hot gas into the evaporator inlet immediately after the expansion valve. Distributors are available with side openings for hot gas inlet. Bypassing at the evaporator inlet has the effect of creating an artificial 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.

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 floodback. See Figure 103 for typical installation.

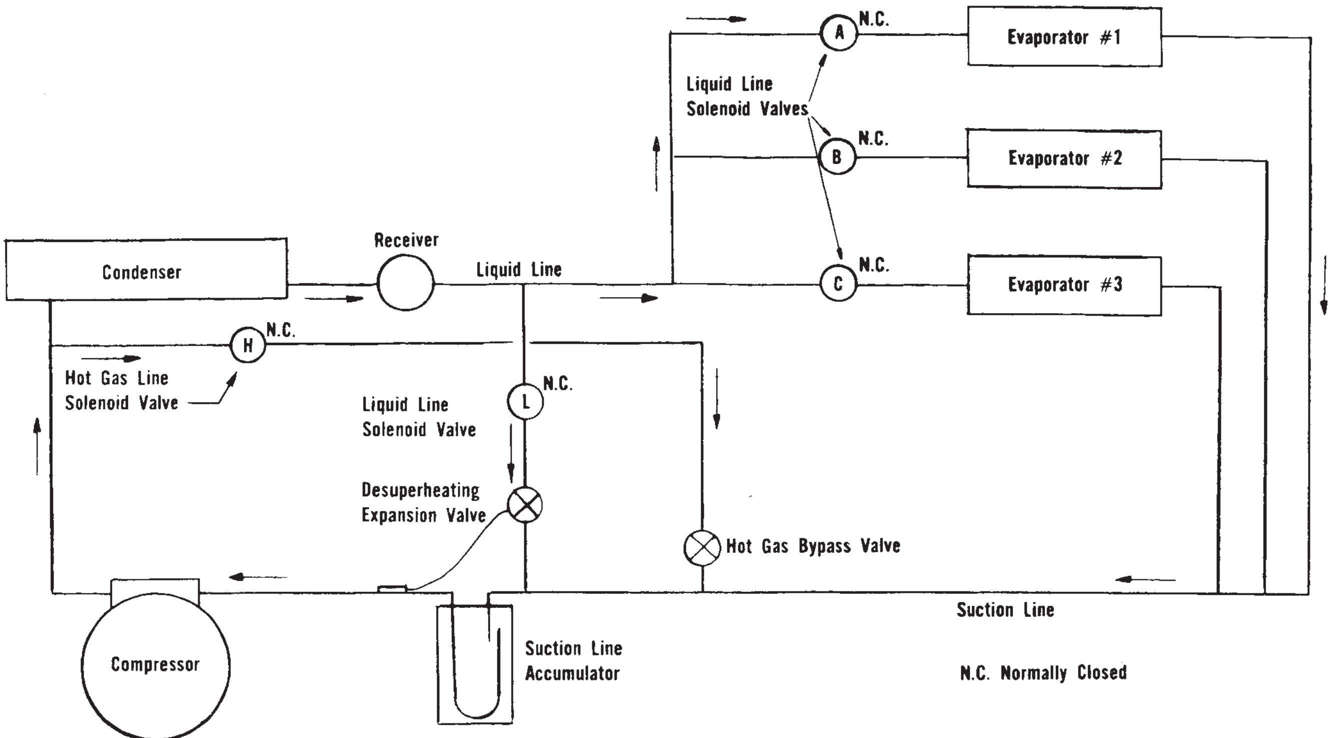
Another commonly used method of mixing is to arrange the piping so that a mixture of discharge gas and liquid refrigerant is introduced into the suction line at some distance form the compressor, in a suction header if possible. Figure 104 illustrates this mixing method.

(continued on p. 21-5)

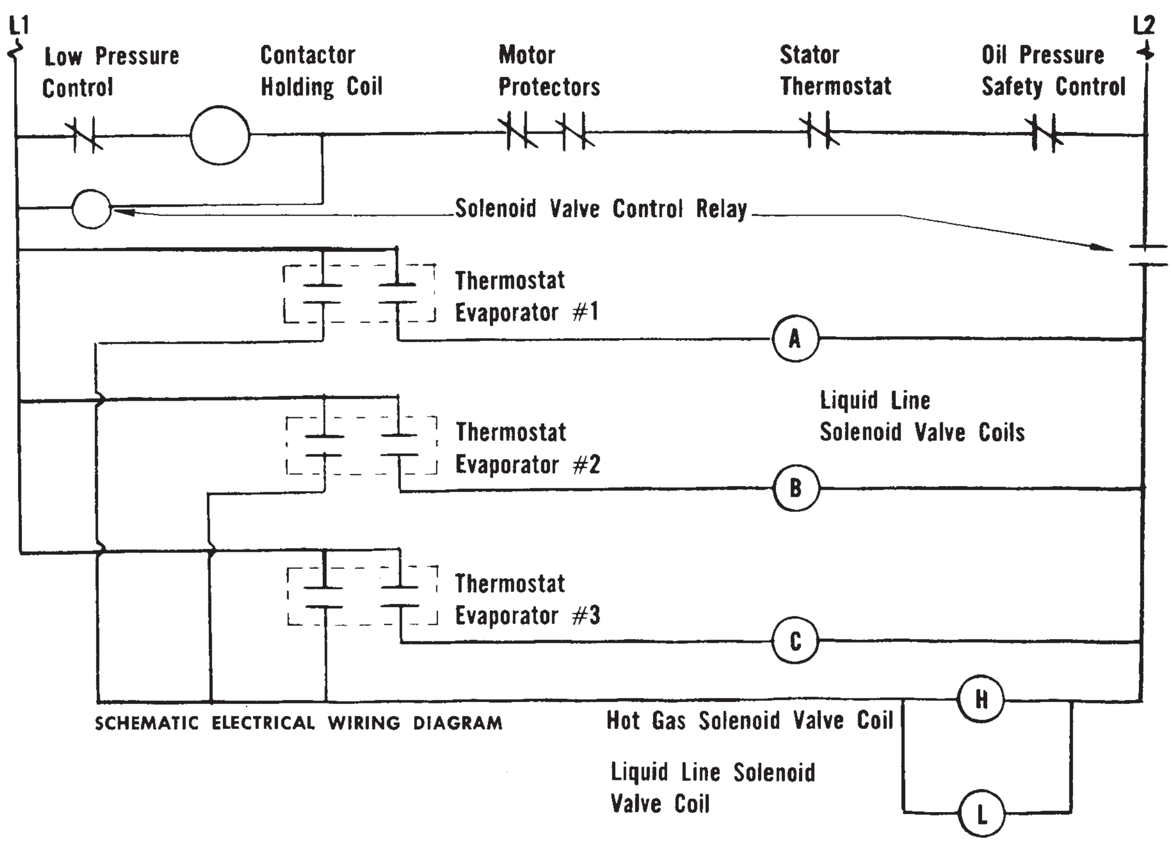


TYPICAL SCHEMATIC CONNECTION — BYPASS INTO EVAPORATOR INLET

Figure 102



SCHEMATIC REFRIGERANT PIPING DIAGRAM



SCHEMATIC ELECTRICAL WIRING DIAGRAM

TYPICAL HOT GAS BYPASS CONTROL SYSTEM
WITH BYPASS INTO SUCTION LINE

Figure 103

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.

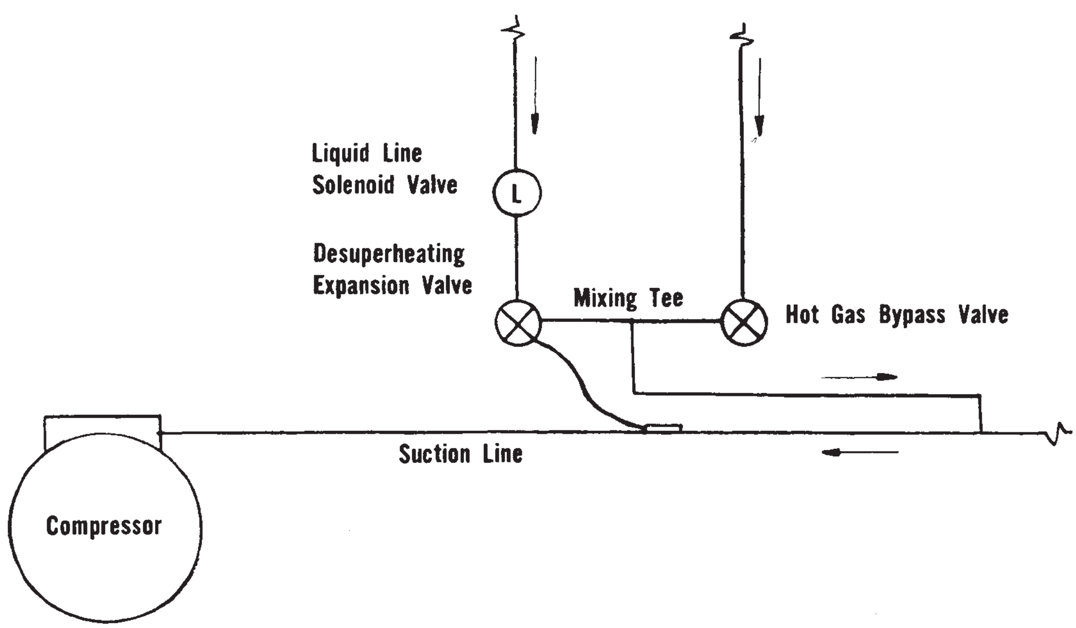
DESUPERHEATING EXPANSION VALVE

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. 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 nonstandard 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



ALTERNATE METHOD OF BYPASS INTO SUCTION LINE

Figure 104

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 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 protection can cycle on the inherent protector independently of the contractor. 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.

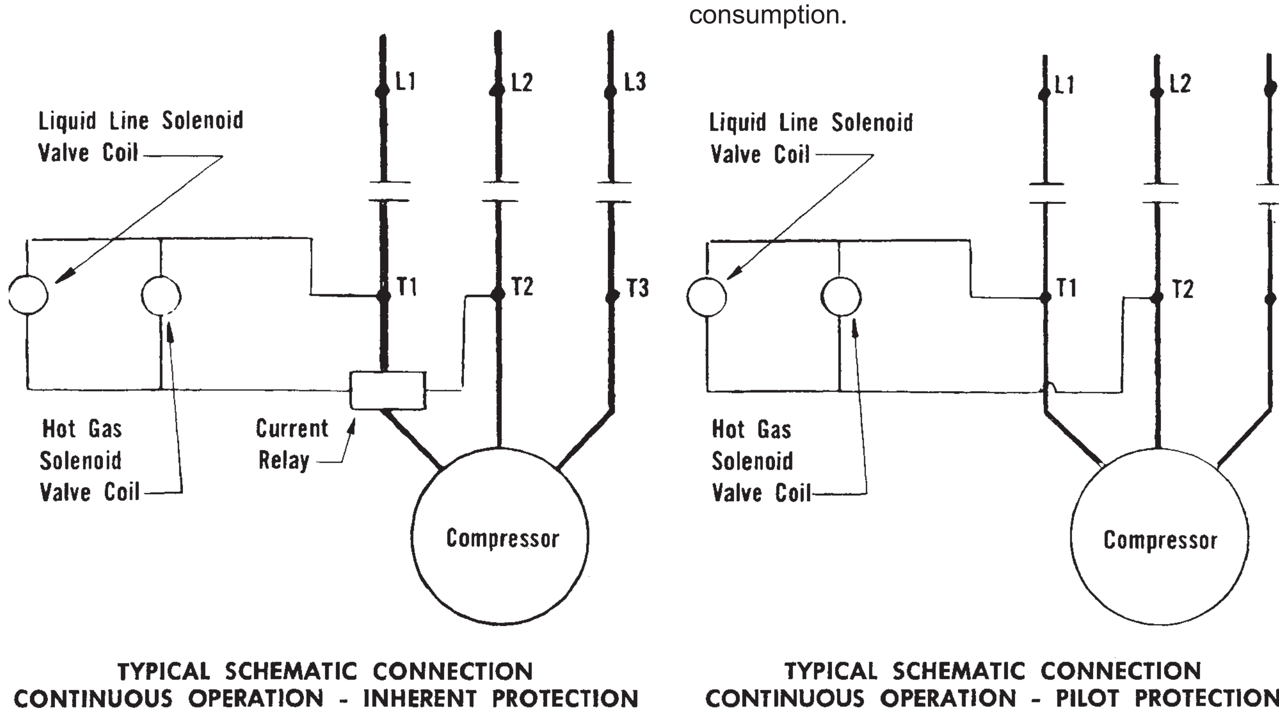


Figure 105

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 the 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 its 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 Emerson Climate Technologies, Inc. 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 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

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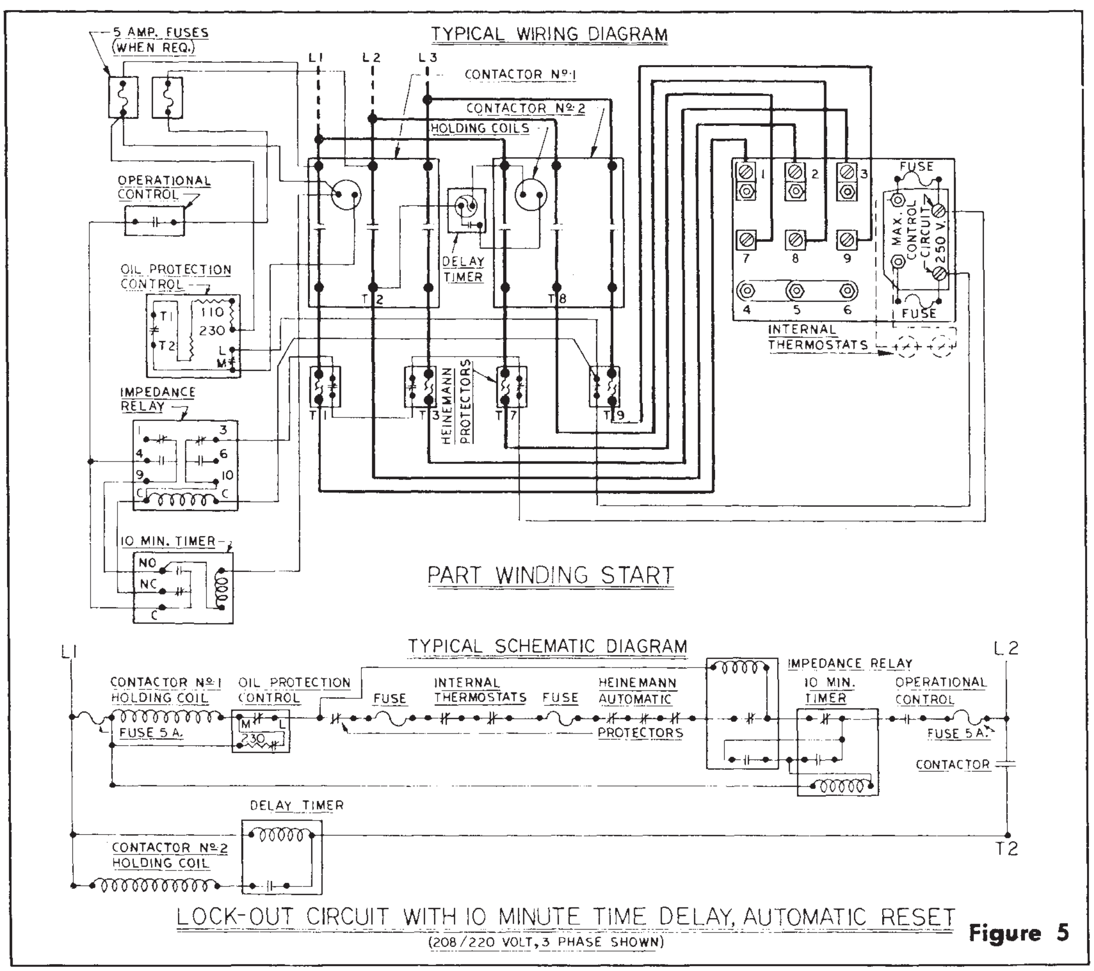
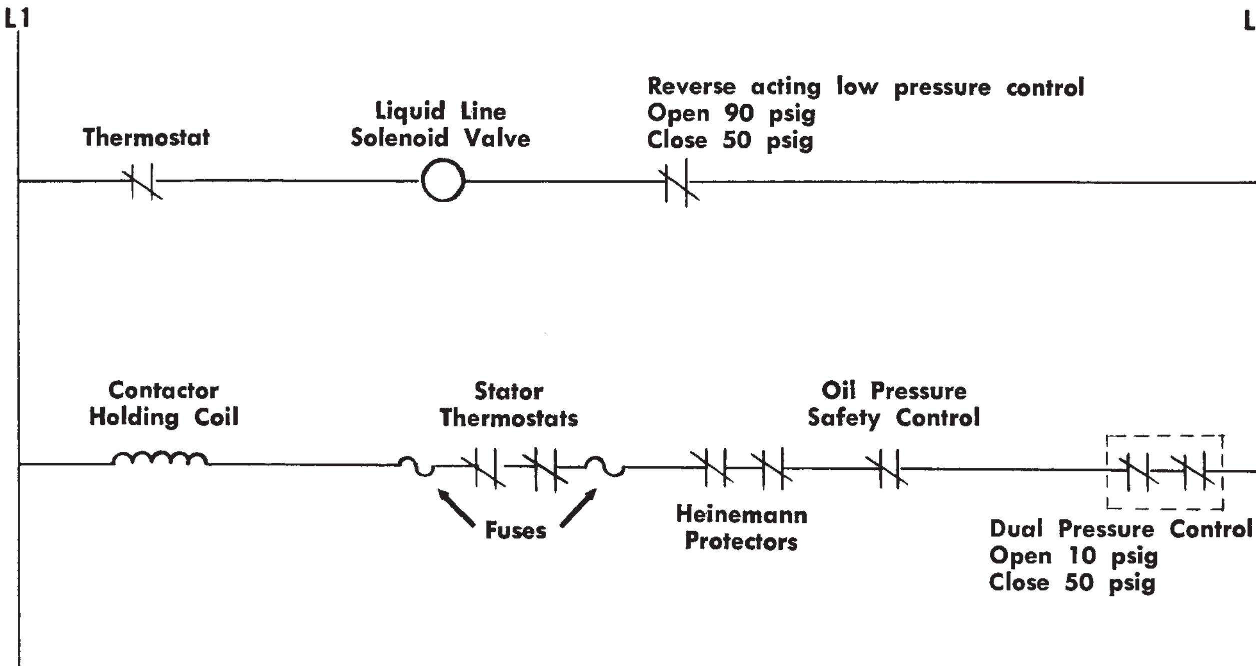
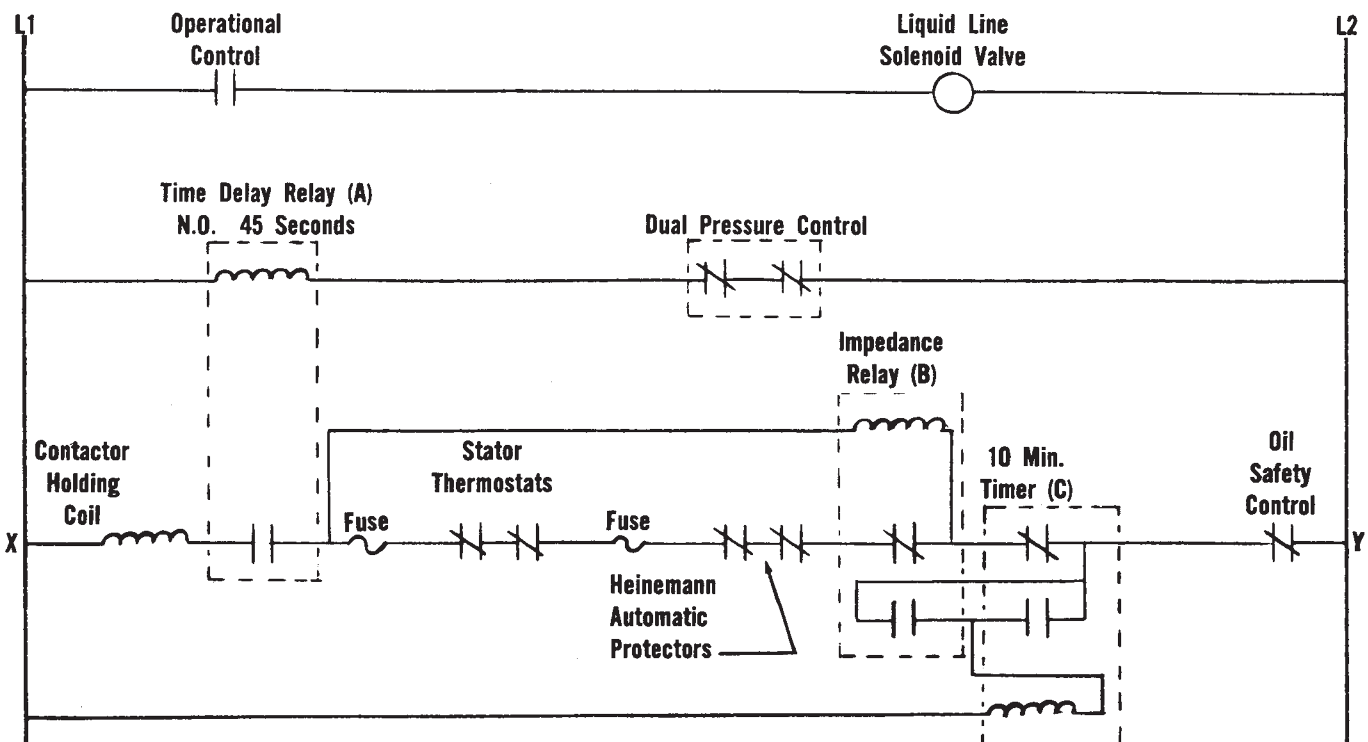


Figure 106



CONTROL CIRCUIT TO PREVENT LIQUID FLOODING

Figure 107



CONTROL CIRCUIT TO PREVENT SHORT CYCLING

Figure 108

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 through 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 re-established 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 internal, 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 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.

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

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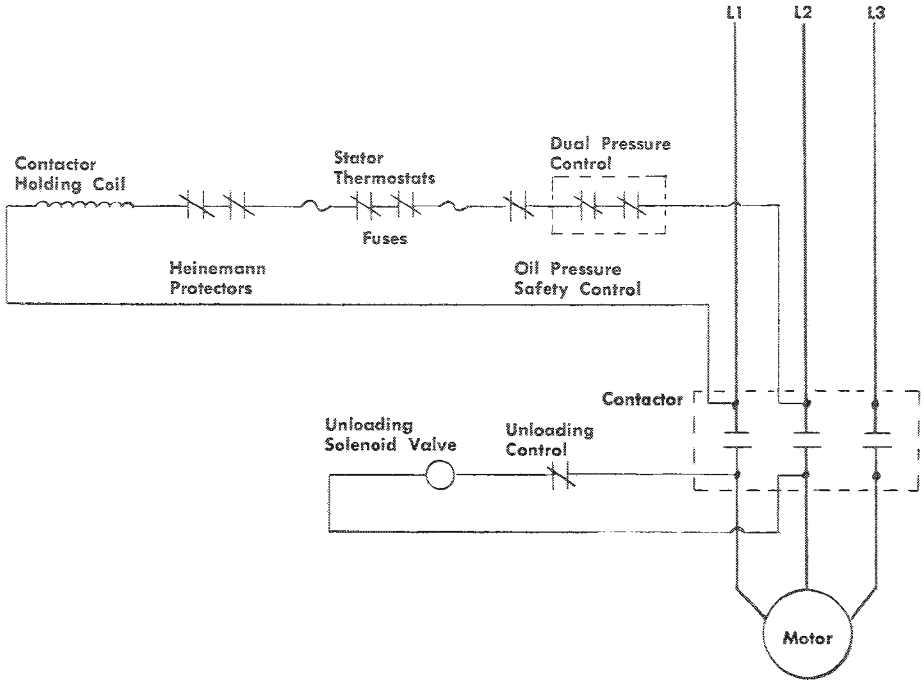
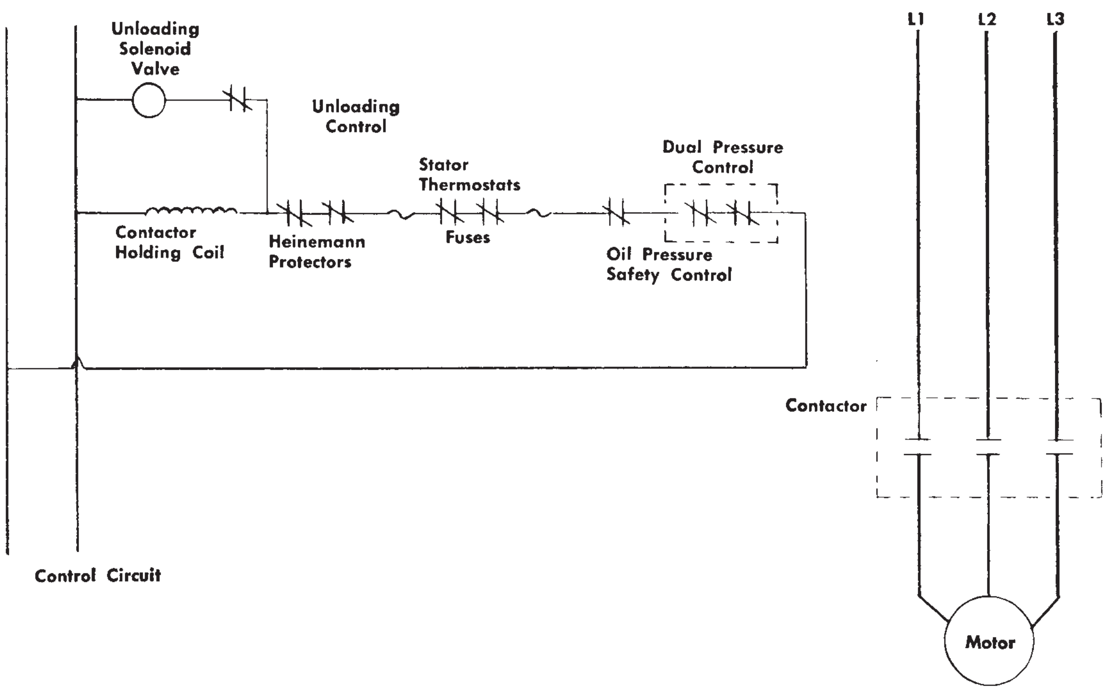


TYPICAL CONTROL CIRCUIT FOR COMPRESSOR WITH UNLOADING VALVE

Figure 109



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

Figure 110

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 operation 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.

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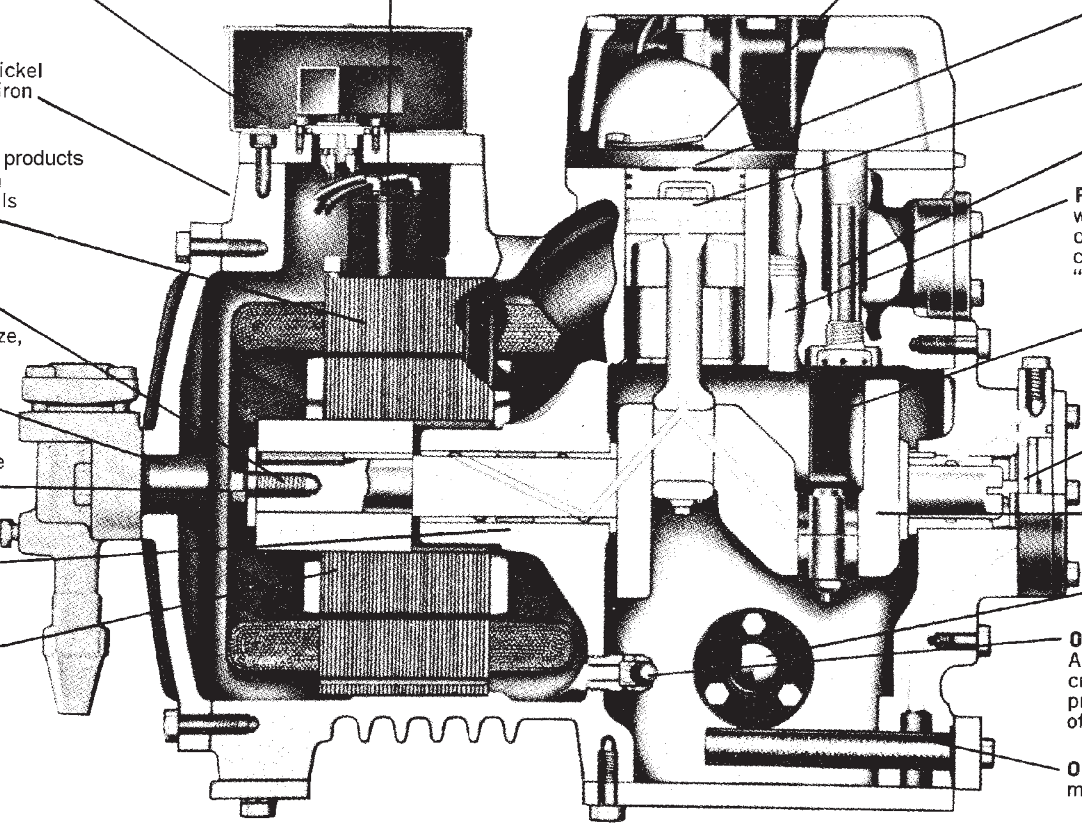

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