Part 4
SYSTEM DESIGN

Section 17. BASIC APPLICATION RECOMMENDATIONS

Section 18. REFRIGERATION PIPING

Section 19. LOW TEMPERATURE SYSTEMS

Section 20. TRANSPORT REFRIGERATION

Section 21. CAPACITY CONTROL

Section 22. LIQUID REFRIGERANT CONTROL IN REFRIGERATION AND AIR CONDITIONING SYSTEMS

Section 23. ELECTRICAL CONTROL CIRCUITS

INDEX OF TABLES

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 five 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 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 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 motor-compressors should never be operated beyond published operating limits without prior approval of the Copeland 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 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. All applications with refrigerants other than R-12, R-22, and R-502 must be approved by the Copeland Application Engineering Department.

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 R-502, 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 are 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 applications 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 against the compressor body. On compressors with multiple heads such as the Copeland 4R and 6R models, auxiliary horizontal air flow 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 . . . $\text{CFM} = \frac{\text{BTU/HR}}{{}^\circ\text{TD}}$

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

$$\text{CFM} = \frac{48{,}543 \text{ BTU/HR}}{10^\circ \text{ TD}} = 4{,}854 \text{ CFM}$$

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.

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

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 Copeland factory are charged with Suniso 3G, 150 viscosity refrigeration oil, and the use of any other oil must be specifically cleared with the Copeland 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 Suniso 3G oil, the possibility of two-phasing is greatly reduced.

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 Copeland 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 minutes 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 Copeland 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 compressors must be specifically approved by the Copeland 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 Copeland limits, 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 compressors should not be operated at suction pressures in excess of the published limits on compressor specification sheets without approval of the Copeland 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 dessicant 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 higher than the saturated 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 a 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 wider, 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 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. Copeland 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 Copelaweld compressors 1¾ 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.

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

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 Copeland 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 oversized 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 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.

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							
3/8	.054	4.25	.171	.011	.018	.044	.065
1/2	.100	7.88	.317	.021	.033	.081	.120
5/8	.162	12.72	.514	.033	.054	.131	.195
7/8	.336	26.4	1.065	.069	.112	.262	.405
1 1/8	.573	45.0	1.82	.118	.191	.464	.690
1 3/8	.872	68.6	2.76	.179	.291	.708	1.05
1 5/8	1.237	97.0	3.92	.254	.412	1.01	1.49
2 1/8	2.147	169.0	6.80	.441	.715	1.74	2.58
2 5/8	3.312	260.0	10.5	.680	1.10	2.68	3.98
3 1/8	4.728	371.0	15.0	.97	1.57	3.82	5.69
3 5/8	6.398	503.0	20.3	1.32	2.13	5.18	7.70
4 1/8	8.313	652.0	26.4	1.71	2.77	6.73	10.0
R-22							
3/8	.054	3.84	.202	.013	.021	.052	.077
1/2	.100	7.12	.374	.024	.04	.096	.143
5/8	.162	11.52	.605	.038	.064	.156	.232
7/8	.336	24.0	1.26	.079	.134	.323	.480
1 1/8	.573	40.8	2.14	.136	.228	.550	.820
1 3/8	.872	62.1	3.26	.207	.348	.839	1.25
1 5/8	1.237	88.0	4.62	.294	.493	1.19	1.77
2 1/8	2.147	153.0	8.04	.51	.858	2.06	3.06
2 5/8	3.312	236.0	12.4	.78	1.32	3.18	4.72
3 1/8	4.728	336.0	17.7	1.12	1.88	4.55	6.75
3 5/8	6.398	456.0	24.0	1.51	2.55	6.15	9.14
4 1/8	8.313	592.0	31.1	1.97	3.31	8.0	11.19
R-502							
3/8	.054	3.98	.284	.020	.033	.077	.112
1/2	.100	7.38	.525	.037	.061	.143	.208
5/8	.162	11.95	.852	.061	.098	.232	.337
7/8	.336	24.8	1.77	.126	.204	.481	.700
1 1/8	.573	42.3	3.01	.215	.347	.820	1.19
1 3/8	.872	64.4	4.60	.327	.527	1.25	1.81
1 5/8	1.237	91.2	6.5	.465	.750	1.77	2.57
2 1/8	2.147	159.0	11.3	.806	1.30	3.08	4.48
2 5/8	3.312	244.0	17.4	1.24	2.0	4.74	6.90
3 1/8	4.728	349.0	24.8	1.77	2.87	6.76	9.84
3 5/8	6.398	471.0	33.6	2.40	3.87	9.15	13.32
4 1/8	8.313	612.0	43.8	3.12	5.03	11.90	17.30

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.

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

Figures 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 2⅝" 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.

"FREON" 12 REFRIGERANT
PRESSURE DROP IN LINES (65°F Evap. Outlet)

C-34 (65)

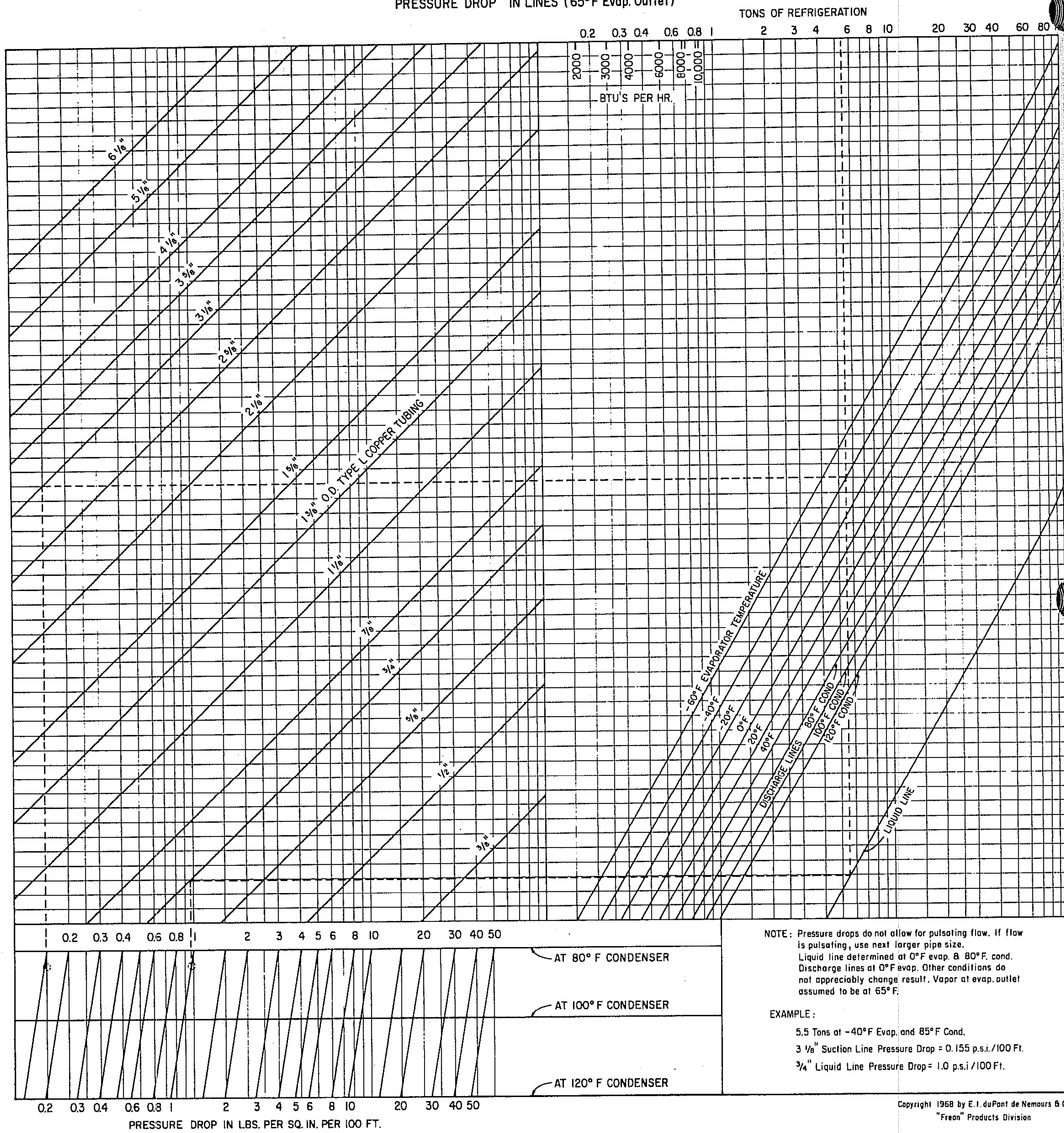


Figure 76

"FREON" 22 REFRIGERANT

PRESSURE DROP IN LINES (65° F Evap. Outlet)

C-35(65)

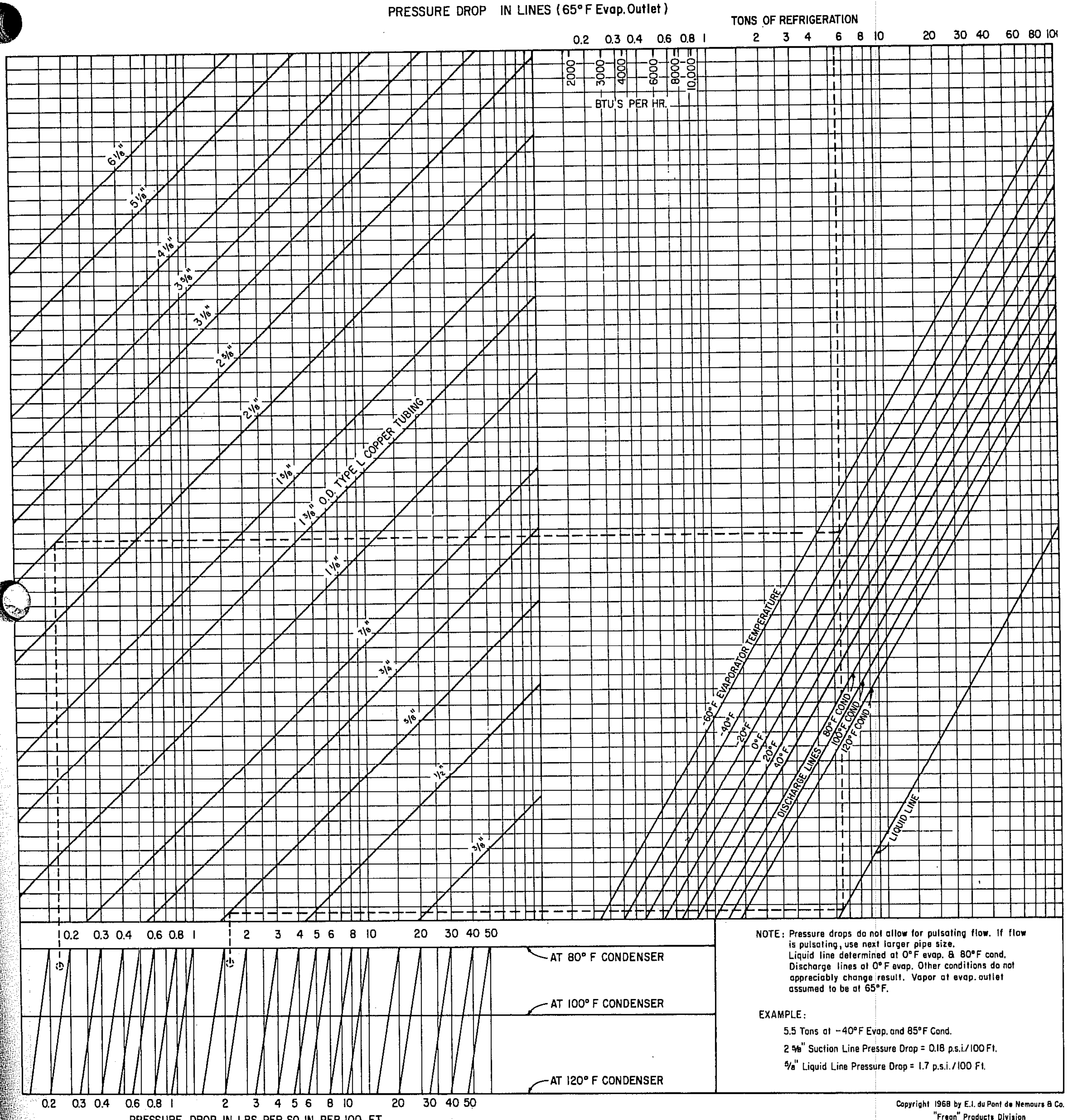


Figure 77

"FREON" 502 REFRIGERANT
PRESSURE DROP IN LINES (65° F Evap. Outlet)

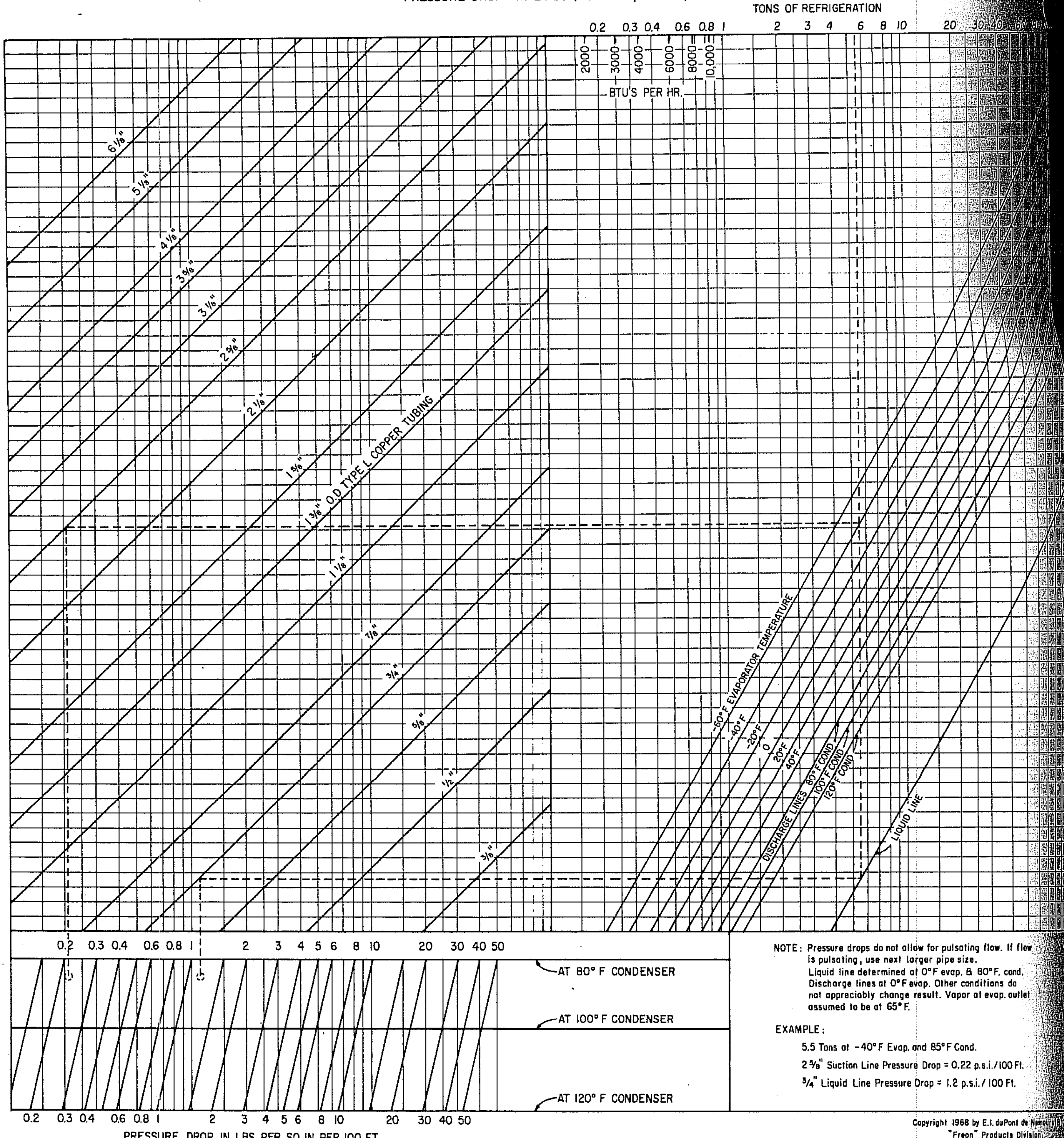


Figure 78

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 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-12 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 79 indicate the design condition. Since this is below the 2⅛″ O.D. line, the maximum size that can be used to insure oil return up a vertical riser is 1⅝″ 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

MAXIMUM RECOMMENDED VERTICAL DISCHARGE LINE SIZES FOR PROPER OIL RETURN

R-12 and R-502

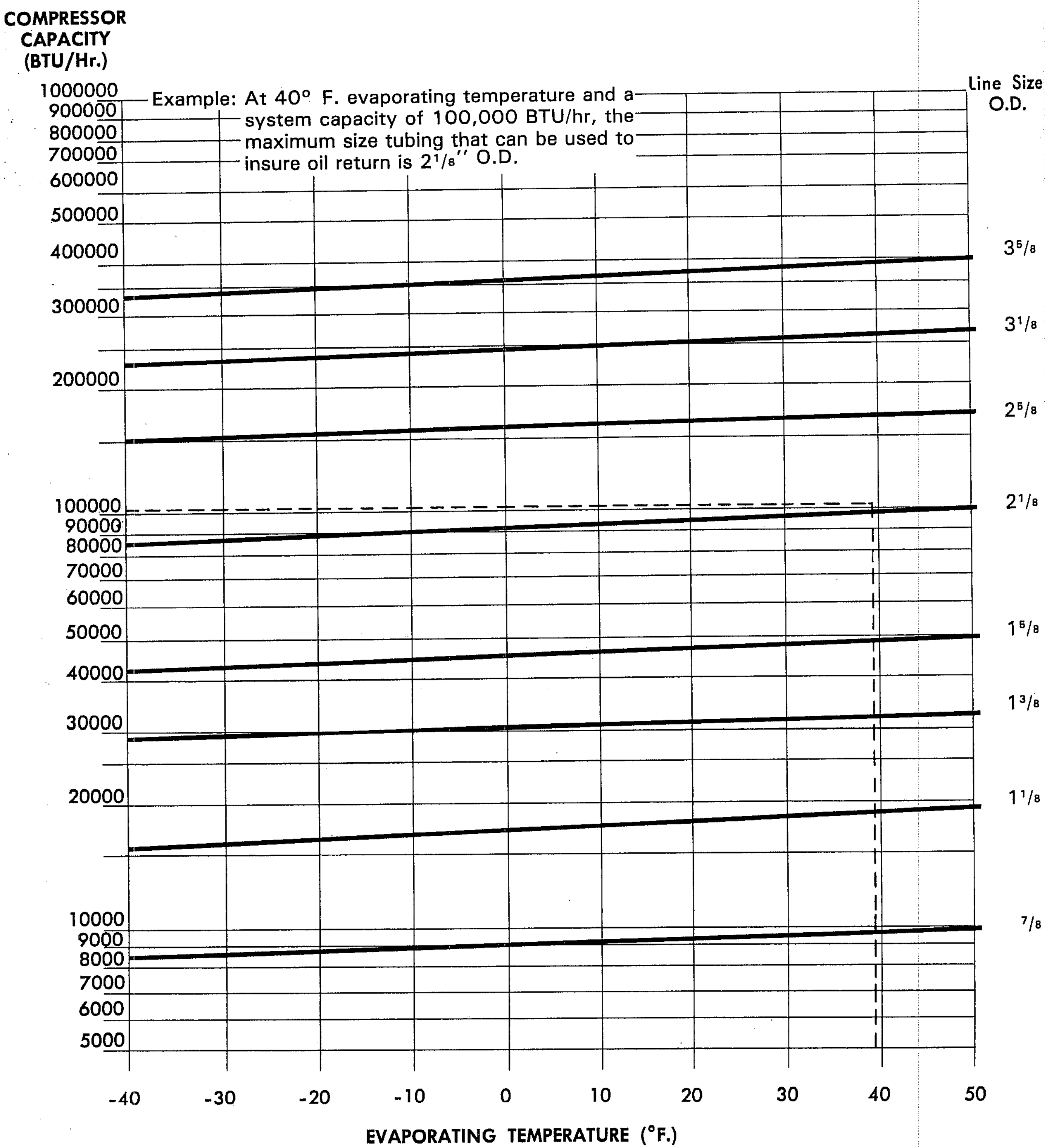


Figure 79

MAXIMUM RECOMMENDED VERTICAL DISCHARGE LINE SIZES FOR PROPER OIL RETURN

R-22

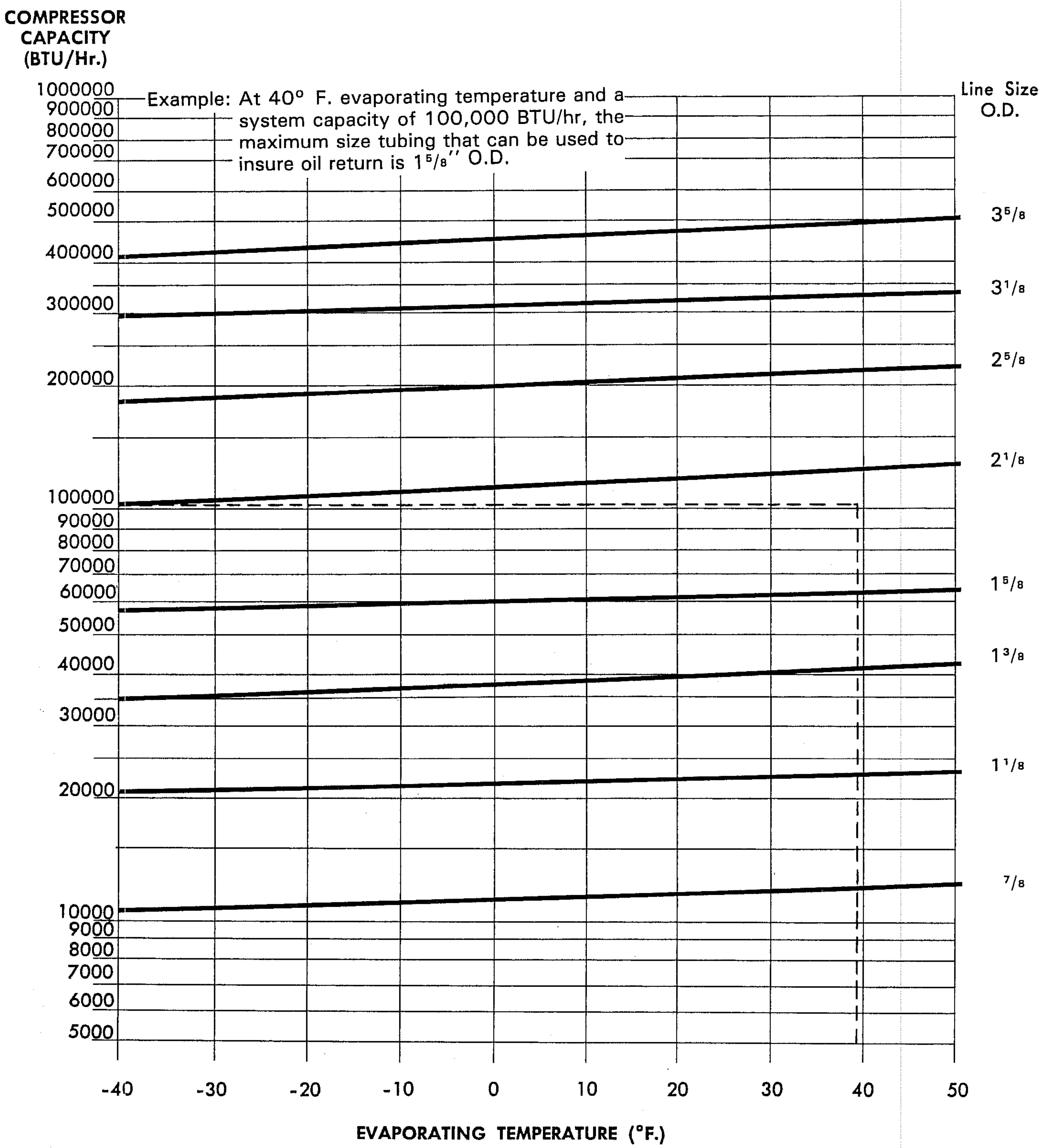


Figure 80

DISCHARGE LINE VELOCITIES FOR VARIOUS BTU/Hr. CAPACITIES

R-12

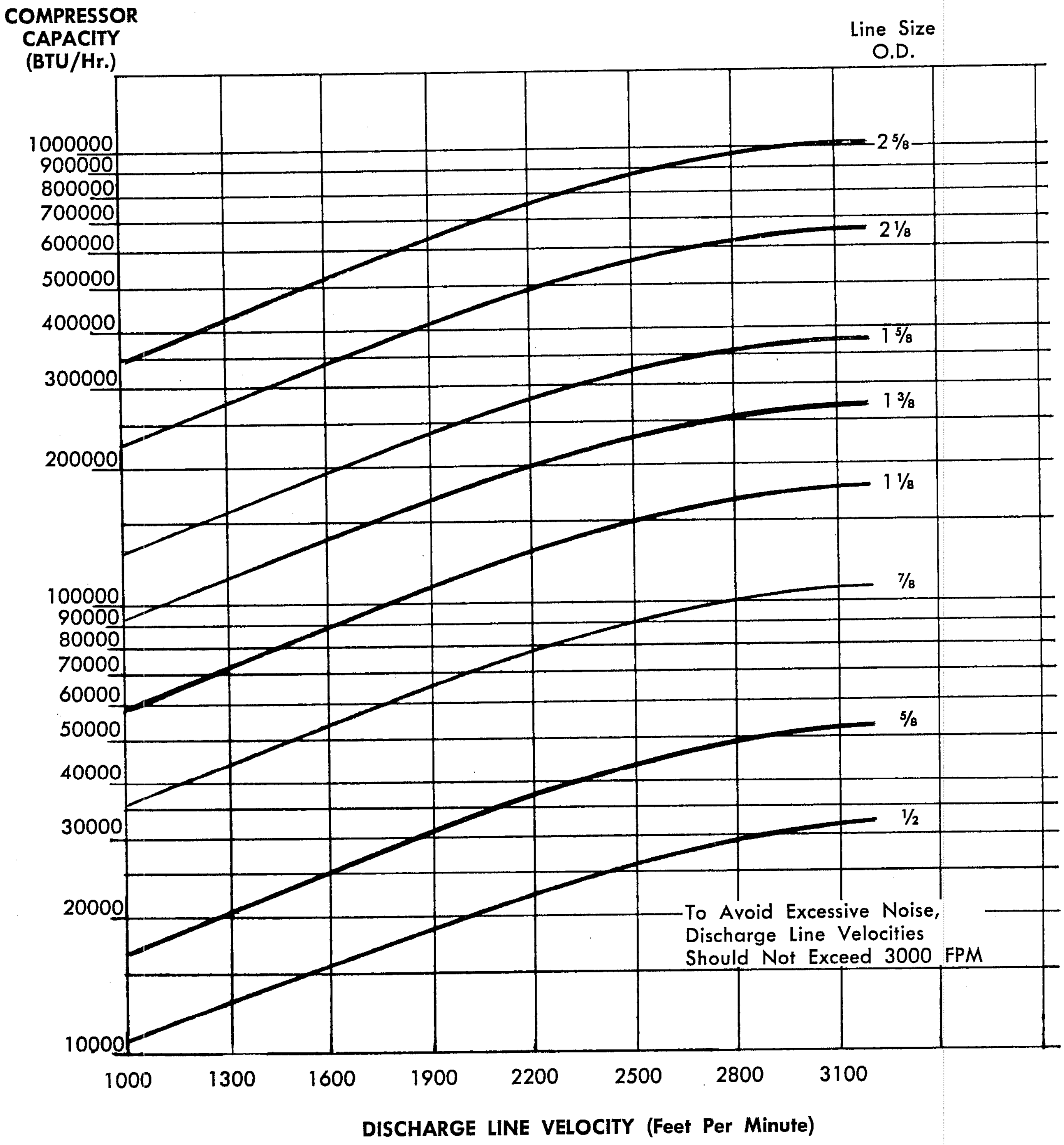


Figure 81

DISCHARGE LINE VELOCITIES FOR VARIOUS BTU/Hr. CAPACITIES

R-22 and R-502

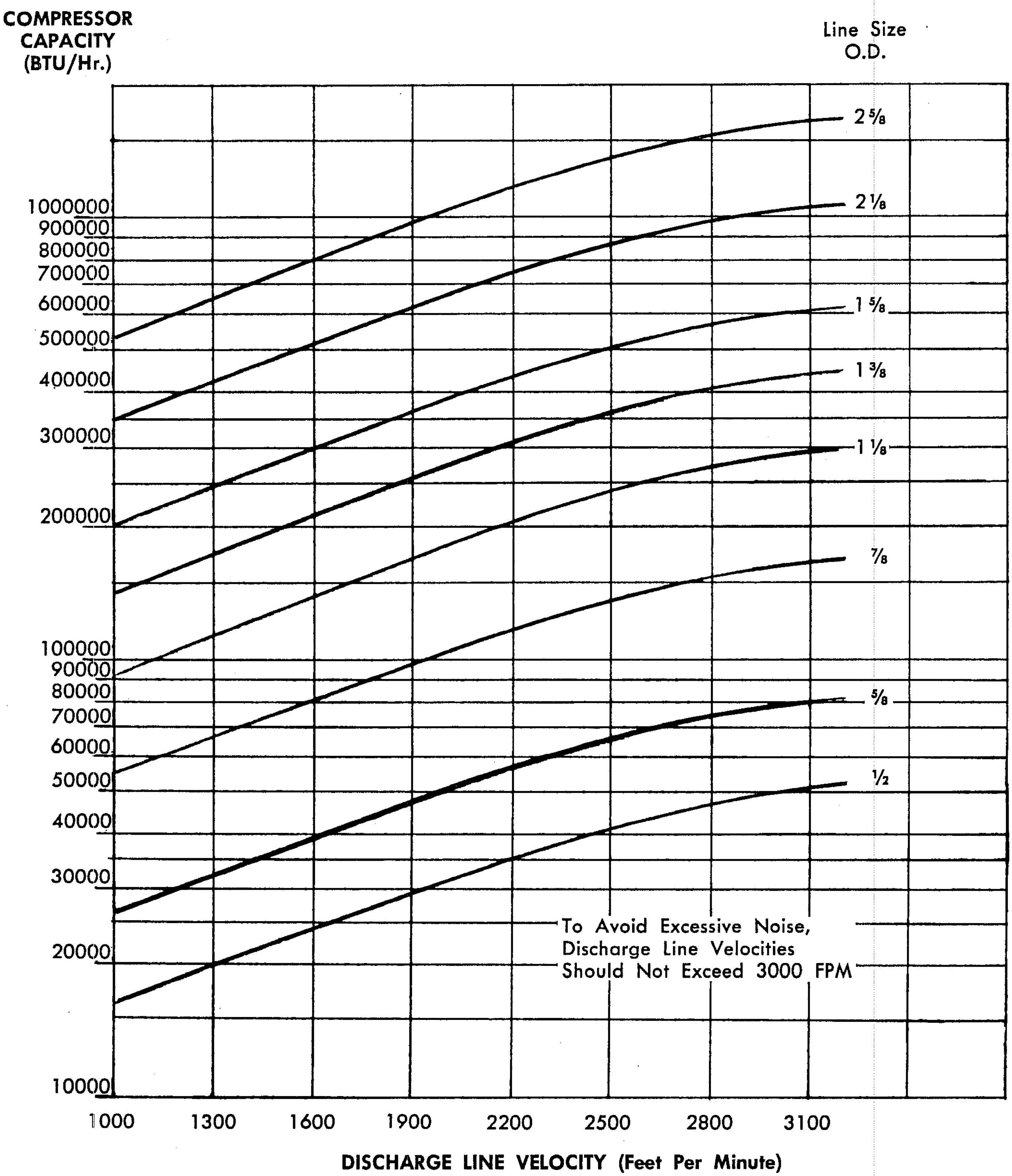


Figure 82

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 2⅝″ O.D. riser, at light load conditions, the system would have only 100,000 BTU/hr capacity, so a 1⅝″ O.D. riser must be used. In checking the pressure drop chart, Figure 76, at maximum load conditions, a 1⅝″ O.D. pipe will have a pressure drop of approximately 4 psi per 100 feet at a condensing temperature of 120° F. If the total equivalent length of piping exceeds 150 feet, in order to keep the total pressure drop within reasonable limits, the horizontal lines should be the next larger size or 2⅛″ O.D., which would result in a pressure drop of only slightly over 1 psi per 100 feet.

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.

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.

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, page 18-30.

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.

LIQUID LINE VELOCITIES FOR VARIOUS PRESSURE DROPS

R-12, R-22, R-502

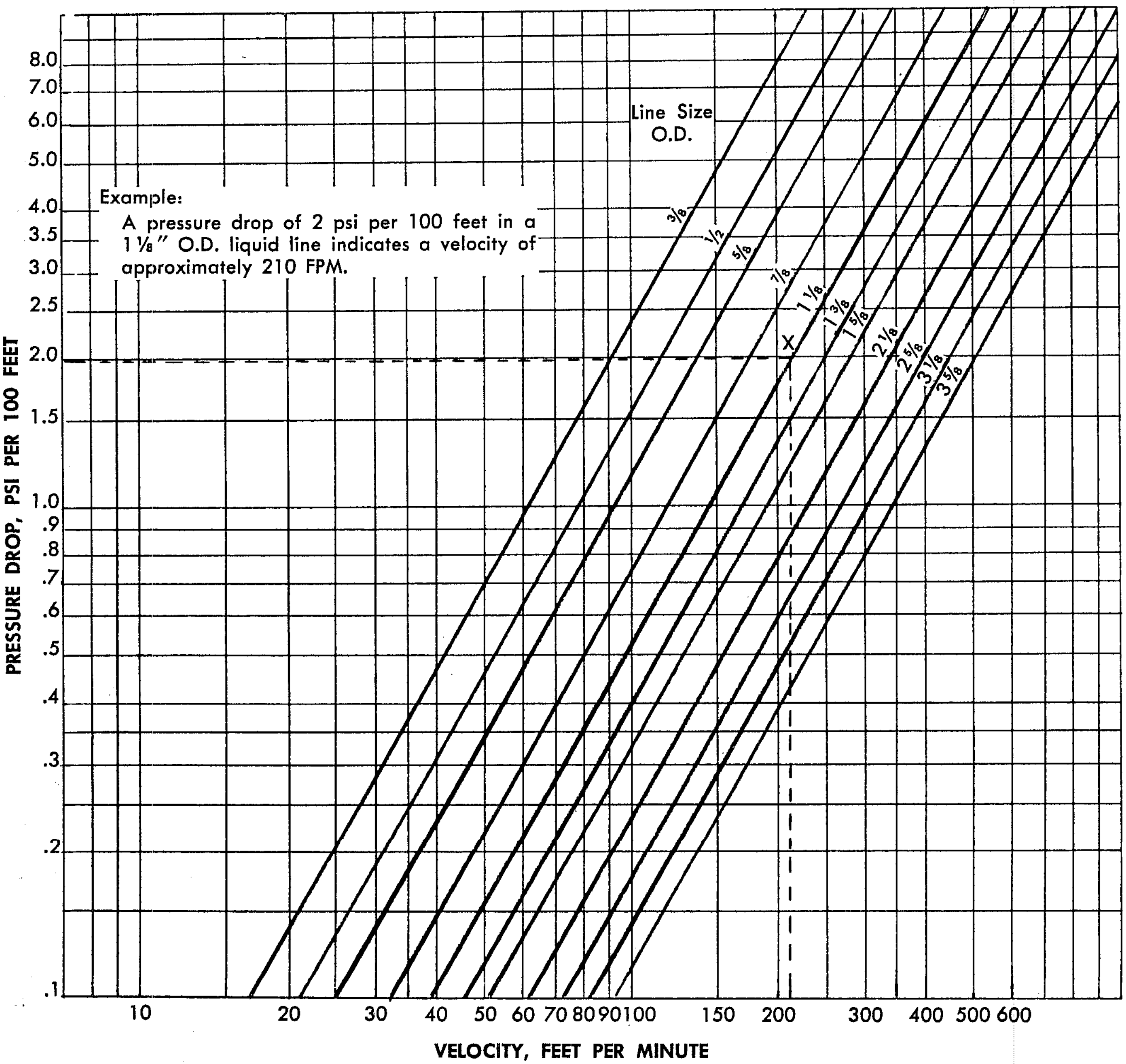


Figure 83

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, page 18-29.

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 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-22	R-502
45° F.	2.0	3.0	3.3
20° F.	1.35	2.2	2.4
0° F.	1.0	1.65	1.85
-20° F.	.75	1.15	1.35
-40° F.	.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 1967 ASHRAE Guide and Data

MAXIMUM RECOMMENDED SUCTION LINE SIZES FOR PROPER OIL RETURN

VERTICAL RISERS R-12 and R-502

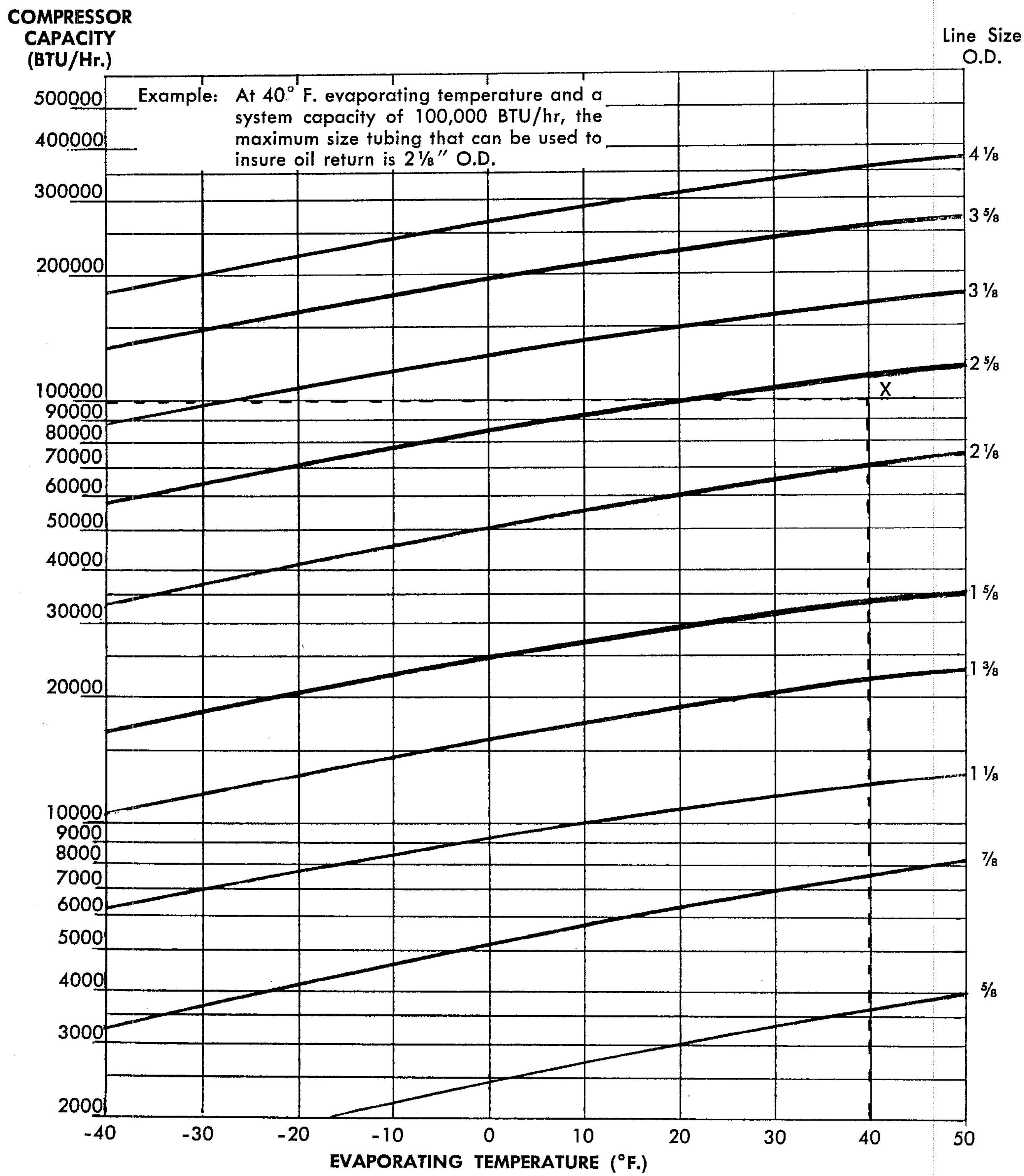


Figure 84

MAXIMUM RECOMMENDED HORIZONTAL SUCTION LINE SIZES FOR PROPER OIL RETURN

R-12

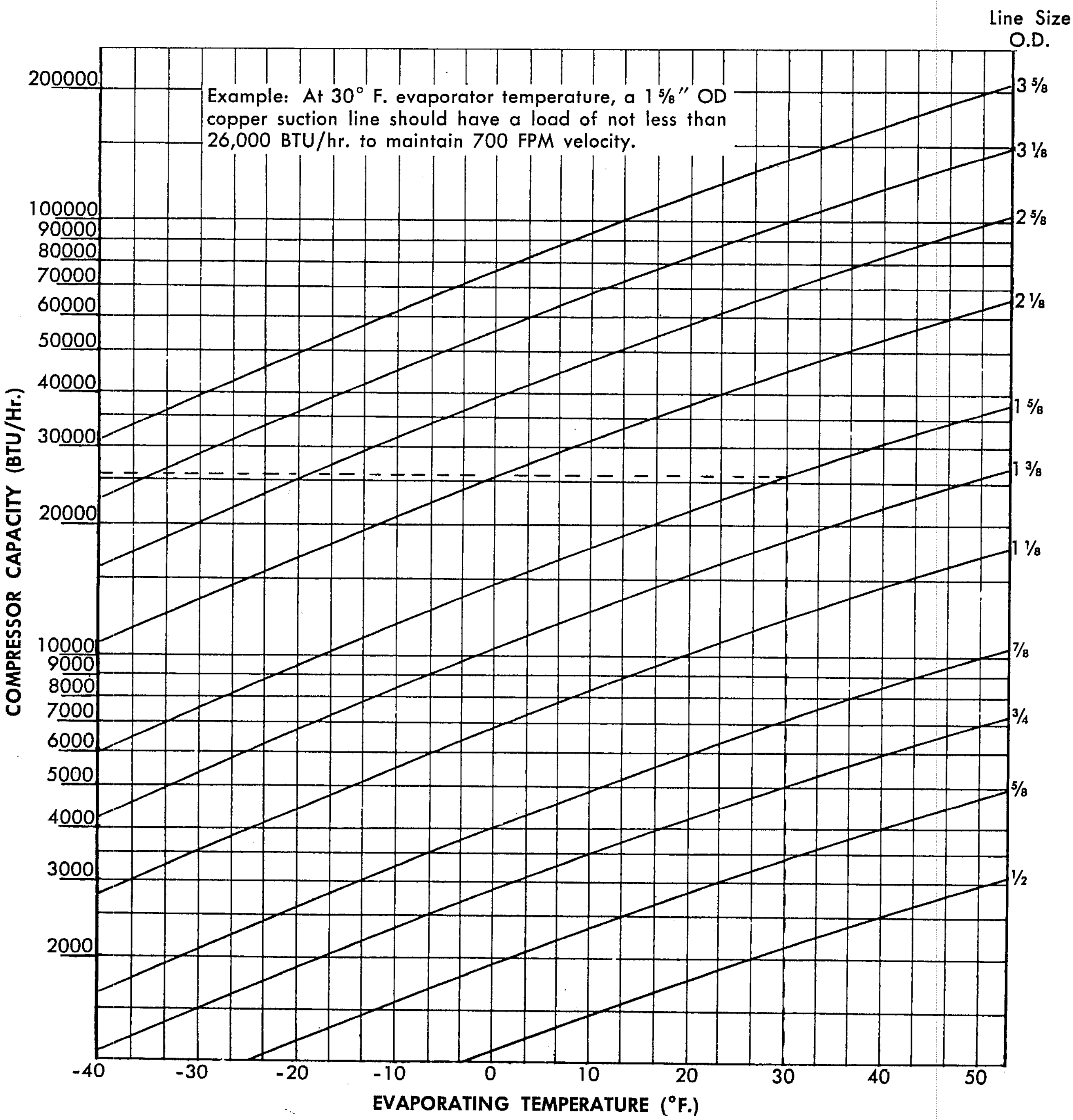


Figure 85

MAXIMUM RECOMMENDED SUCTION LINE SIZES FOR PROPER OIL RETURN

VERTICAL RISERS R-22

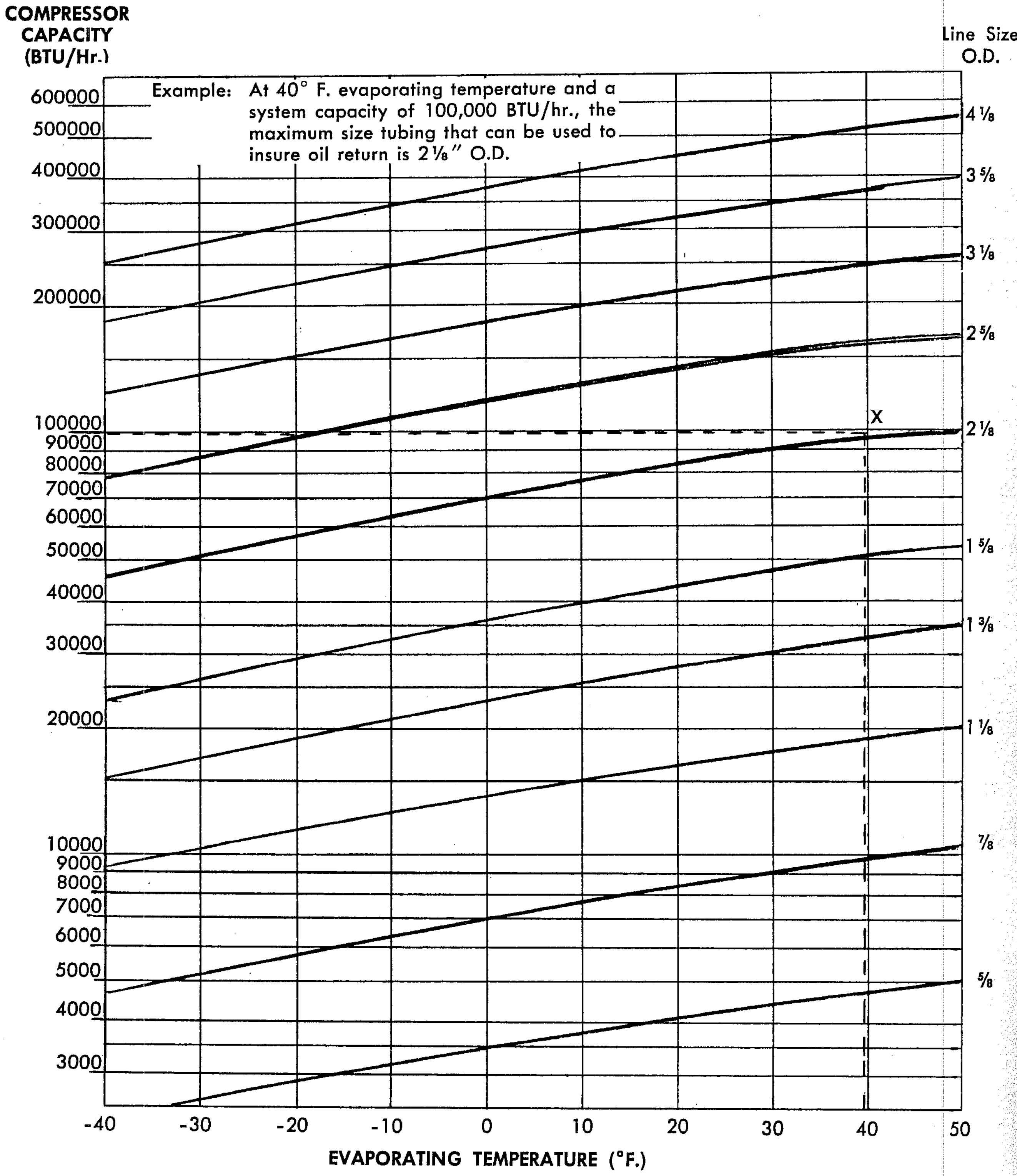


Figure 86

MAXIMUM RECOMMENDED HORIZONTAL SUCTION LINE SIZES FOR PROPER OIL RETURN

R-22 and R-502

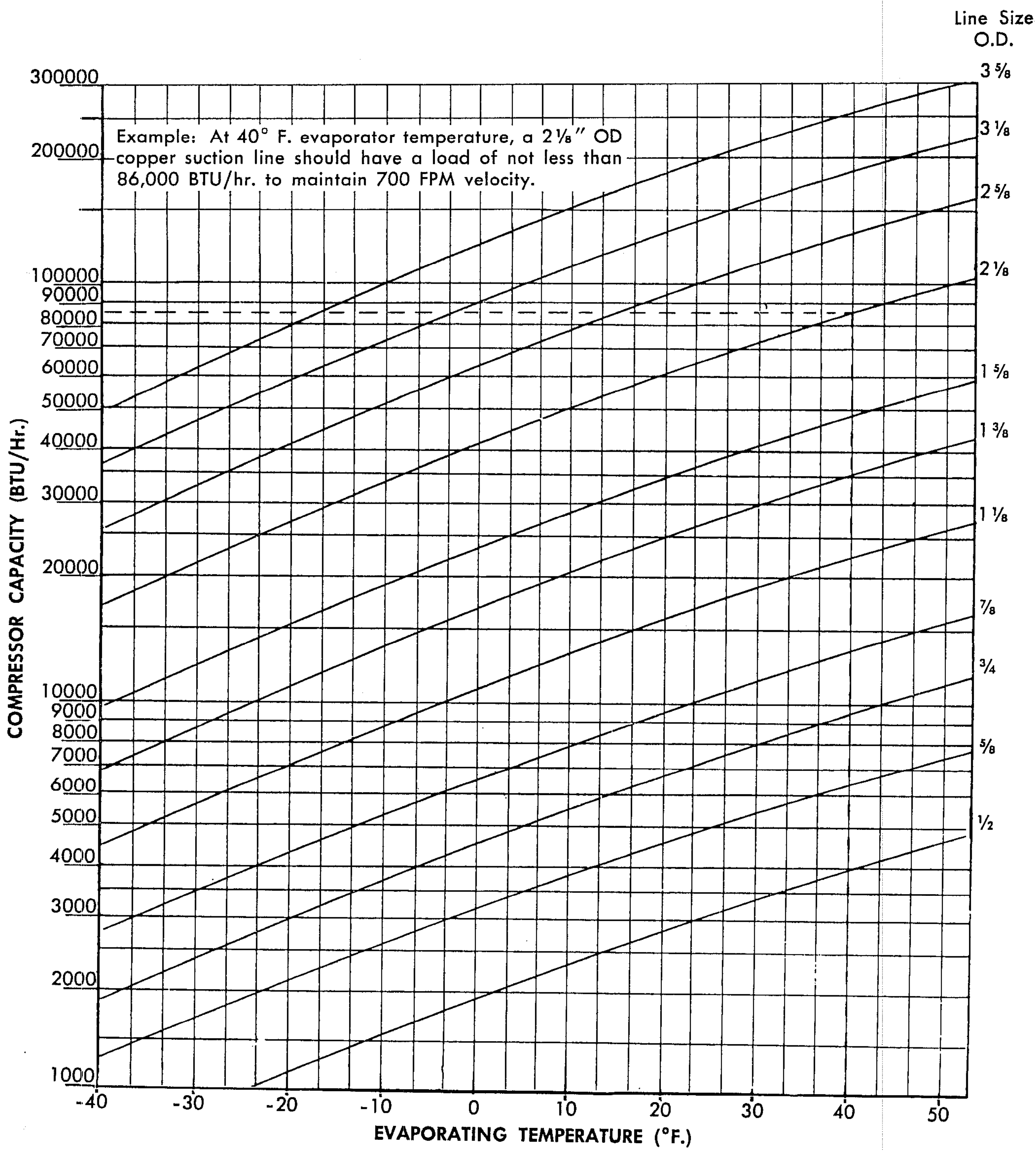


Figure 87

Book have been calculated and are plotted in chart form for easy usage in Figures 84 and 86. These revised recommendations supersede previous Copeland 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.

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 2⅛" 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.

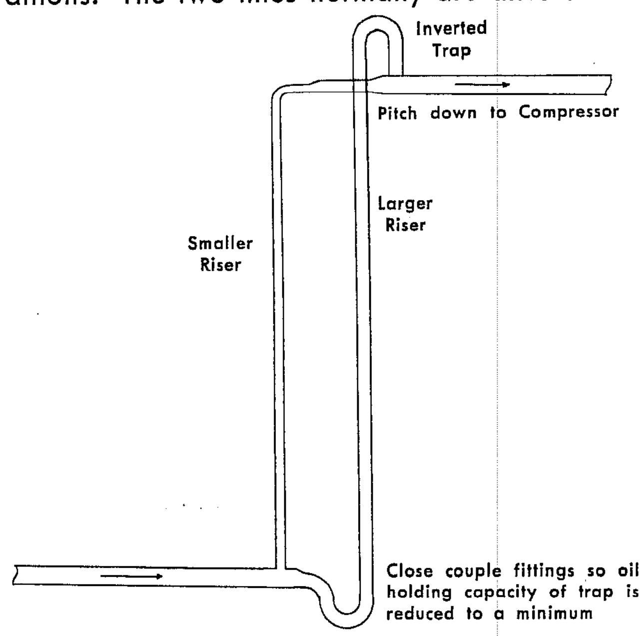
Since the dual goals of low pressure drop and high velocities are in direct conflict, obviously compromises must be made in both areas. As a general approach, in suction line design, velocities should be kept as high as possible by sizing lines on the basis of the maximum pressure drop that can be tolerated, 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, starting on page 18-31.

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 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



SUCTION LINE DOUBLE RISER

Figure 88

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 3⅛" O.D. line is indicated. At the minimum capacity of 50,000 BTU/hr, Figure 87 shows a 3⅝" O.D. horizontal suction line is acceptable, but Figure 86 indicates that the maximum vertical riser size is 2⅛" O.D. Referring again to the pressure drop chart, Figure 78, the pressure drop for 150,000 BTU/hr through 2⅛" 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 3⅛" O.D. riser would maintain adequate velocities, so a combination of line sizes approximating the 3⅛" O.D. line can be selected for the double riser. The cross sectional area of the line sizes to be considered are:

3⅛" O.D.	6.64 sq. in.
2⅝" O.D.	4.77 sq. in.
2⅛" O.D.	3.10 sq. in.
1⅝" O.D.	1.78 sq. in.

At the minimum load condition of 50,000 BTU/hr., the 1⅝" O.D. line will have a pressure drop of approximately .5 psi, and will have acceptable velocities, so a combination of 2⅝" O.D. and 1⅝" 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 rarified 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.

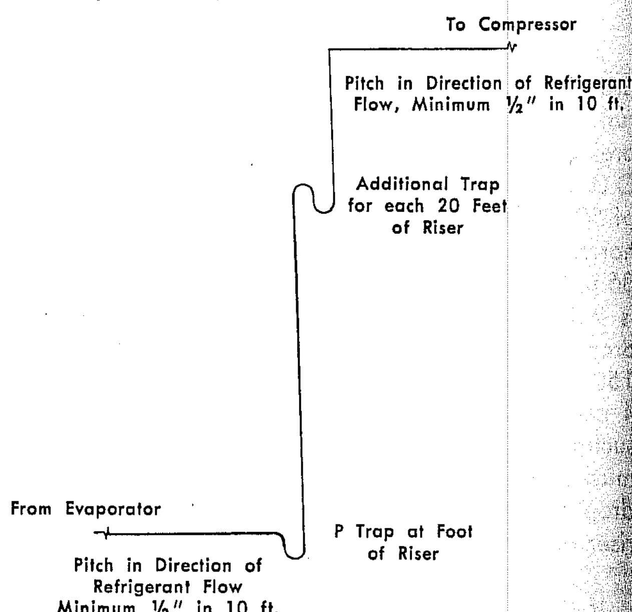
PIPING DESIGN FOR HORIZONTAL AND VERTICAL LINES

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.

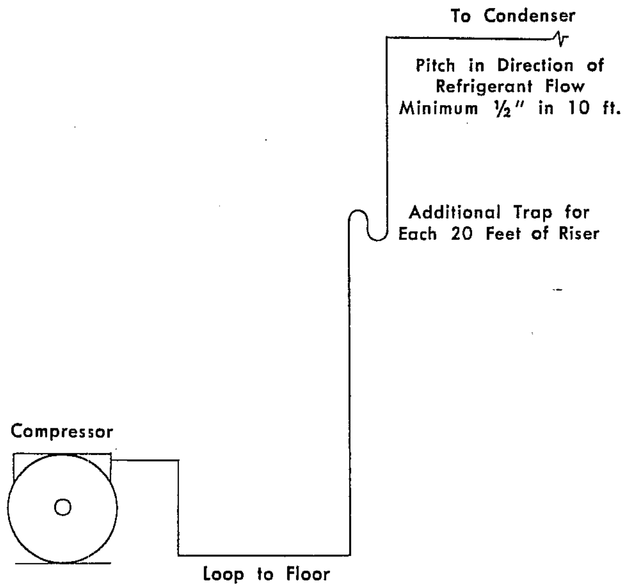
Every vertical suction riser greater than 3 to 4 feet in height should have a "P" trap at the base to facilitate oil return up the riser as shown in Figure 89. To avoid the accumulation of large quantities of oil, the trap should be of minimum depth and the horizontal section should be as short as possible. 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 recommnded 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.



SUCTION LINE RISERS

Figure 89



DISCHARGE LINE RISERS

Figure 90

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 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.

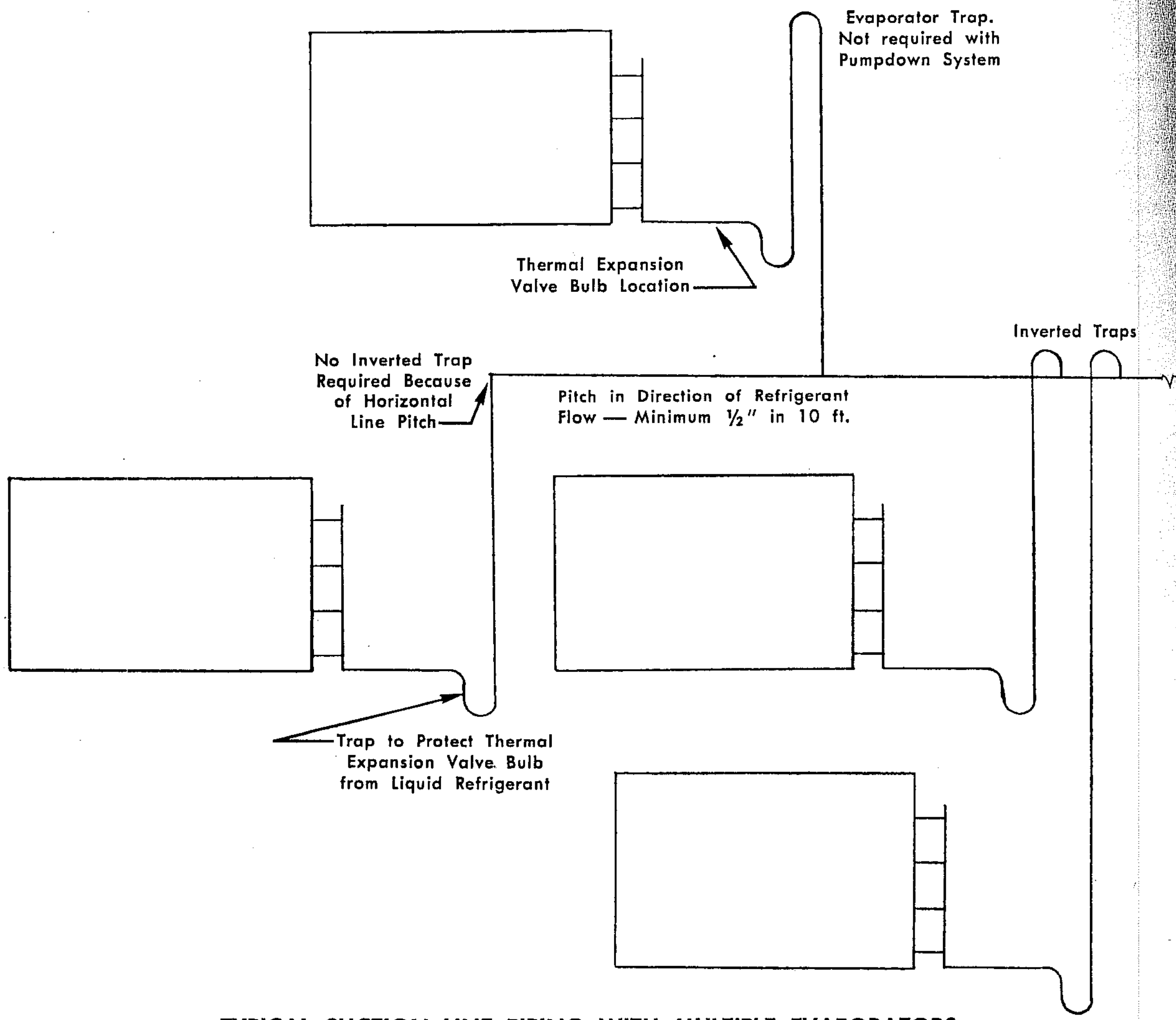
RECEIVER LOCATION

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 higher 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



TYPICAL SUCTION LINE PIPING WITH MULTIPLE EVAPORATORS

Figure 91

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 it 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 o 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.

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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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 2½° 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 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	1/2	1/2	1/2	5/8*	3/8	1/2	1/2	1/2	1/2	1/2	1/2	5/8*
12,000	0	5/8	5/8	5/8	7/8	1/2	1/2	5/8	5/8	5/8	5/8	5/8	7/8
18,000	0	5/8	7/8	7/8	7/8	5/8	5/8	5/8	7/8	5/8	7/8	7/8	7/8
24,000	0	7/8	7/8	7/8	7/8	5/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
36,000	0	7/8	7/8	7/8	1 1/8	7/8	7/8	7/8	7/8	7/8	7/8	1 1/8	1 1/8
48,000	0	7/8	1 1/8	1 1/8	1 1/8	7/8	7/8	7/8	1 1/8	7/8	1 1/8	1 1/8	1 1/8
60,000	0	1 1/8	1 1/8	1 1/8	1 3/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8
	33%	1 1/8	1 1/8	1 1/8	1 3/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8
75,000	0	1 1/8	1 1/8	1 1/8	1 3/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 3/8
	33%	1 1/8	1 1/8	1 1/8	1 3/8	7/8	1 1/8	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 3/8
100,000	0	1 1/8	1 3/8	1 3/8	1 5/8	1 1/8	1 1/8	1 3/8	1 3/8	1 1/8	1 3/8	1 3/8	1 5/8
	33% to 50%	1 1/8	1 3/8	1 3/8	1 5/8	1 1/8	1 1/8	1 3/8	1 3/8	1 1/8	1 3/8	1 3/8	1 5/8
150,000	0	1 3/8	1 5/8	1 5/8	2 1/8	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 5/8
	33% to 50%	1 3/8	1 5/8	1 5/8	2 1/8*	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 5/8
	66%	1 3/8	1 5/8	1 5/8	2 1/8*	1 1/8	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 5/8
200,000	0	1 5/8	1 5/8	2 1/8	2 1/8	1 3/8	1 3/8	1 5/8	1 5/8	1 3/8	1 5/8	1 5/8	2 1/8
	33% to 50%	1 5/8	1 5/8	2 1/8	2 1/8	1 3/8	1 3/8	1 5/8	1 5/8	1 3/8	1 5/8	1 5/8	2 1/8
	66%	1 5/8	1 5/8	2 1/8*	2 1/8*	1 3/8	1 3/8	1 5/8	1 5/8	1 3/8	1 5/8	1 5/8	2 1/8*
300,000	0	2 1/8	2 1/8	2 1/8	2 1/8	1 3/8	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	2 1/8	2 1/8
	33% to 50%	2 1/8	2 1/8	2 1/8	2 1/8	1 3/8	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	2 1/8	2 1/8
	66%	2 1/8	2 1/8	2 1/8	2 1/8	1 3/8	1 5/8	2 1/8*	2 1/8*	1 5/8	2 1/8	2 1/8	2 1/8
400,000	0	2 1/8	2 1/8	2 1/8	2 5/8	1 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8
	33% to 66%	2 1/8	2 1/8	2 1/8	2 5/8	1 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8
500,000	0	2 5/8	2 5/8	2 5/8	2 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 5/8
	33% to 50%	2 5/8	2 5/8	2 5/8	2 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 5/8
	66%	2 5/8	2 5/8	2 5/8	2 5/8	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8
600,000	0	2 5/8	2 5/8	2 5/8	3 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 5/8	3 1/8
	33% to 50%	2 5/8	2 5/8	2 5/8	3 1/8*	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 5/8	3 1/8*
	66%	2 5/8	2 5/8	3 1/8*	3 1/8*	2 1/8	2 1/8	2 1/8	2 5/8*	2 1/8	2 5/8	2 5/8	3 1/8*
750,000	0	3 1/8	3 1/8	3 1/8	3 1/8	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8
	33% to 50%	3 1/8	3 1/8	3 1/8	3 1/8	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8
	66%	3 1/8*	3 1/8*	3 1/8*	3 1/8*	2 1/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8*

* 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		100		150		200	
		H	V	H	V	H	V	H	V
6,000	0	⅝	⅝	⅞	⅝	⅞	⅝	⅞	⅝
12,000	0	⅞	⅞	⅞	⅞	1⅛	⅞	1⅛	⅞
18,000	0	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛	1⅛
24,000	0	1⅛	1⅛	1⅛	1⅛	1⅜	1⅛	1⅜	1⅛
36,000	0	1⅜	1⅛	1⅜	1⅜	1⅜	1⅜	1⅝	1⅜
48,000	0	1⅜	1⅜	1⅝	1⅜	1⅝	1⅝	1⅝	1⅝
60,000	0 to 33%	1⅝	1⅜	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝
75,000	0 to 33%	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
100,000	0 to 33%	1⅝	1⅝	2⅛	1⅝	2⅛	2⅛	2⅛	2⅛
	50%	1⅝	1⅝	2⅛	1⅝	2⅛	1⅝	2⅛	1⅝
150,000	0 to 33%	2⅛	2⅛	2⅝	2⅛	2⅝	2⅝	2⅝	2⅝
	50% to 66%	2⅛	2⅛	2⅝	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
200,000	0 to 50%	2⅝	2⅝	2⅝	2⅝	3⅛	2⅝	3⅛	2⅝
	66%	2⅝	2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛	2⅝	1⅝*2⅛
300,000	0 to 50%	2⅝	2⅝	3⅛	2⅝	3⅝	2⅝	3⅝	2⅝
	66%	2⅝	2⅝	3⅛	2⅝	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝
400,000	0 to 50%	3⅛	3⅛	3⅛	3⅛	3⅝	3⅛	3⅝	3⅛
	66%	3⅛	2⅝	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝	3⅛	1⅝*2⅝

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	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
12,000	0	7/8	7/8	1 1/8	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 3/8	1 1/8	1 3/8	1 1/8
24,000	0	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 3/8
36,000	0	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8	2 1/8	1 5/8
48,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	2 1/8	1 5/8
60,000	0 to 33%	1 5/8	1 5/8	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 5/8	1 5/8
100,000	0 to 33%	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	2 1/8
	50%	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8	2 5/8	1 5/8
150,000	0 to 33%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8
	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 1/8	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	2 5/8
	66%	2 5/8	2 1/8	3 1/8	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%	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	3 1/8	3 5/8	3 1/8
	66%	3 1/8	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	2 5/8	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 5/8	1 5/8 *2 5/8	3 5/8	1 5/8 *2 5/8

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	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 3/8	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 5/8	1 5/8	1 5/8	2 1/8	1 5/8
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	1 5/8	2 1/8	2 1/8	2 5/8	2 1/8
60,000	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
75,000	0 to 33%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8
100,000	0 to 50%	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	3 1/8	2 1/8
150,000	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 5/8	1 5/8 *2 5/8

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 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	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
12,000	0	1 3/8	1 1/8	1 5/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8
18,000	0	1 3/8	1 3/8	1 5/8	1 3/8	1 5/8	1 3/8	2 1/8	1 5/8
24,000	0	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8	2 5/8	1 5/8
36,000	0	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
48,000	0	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	0 to 33%	2 5/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
75,000	0	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8
	33%	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8	3 1/8	2 1/8
100,000	0 to 33%	3 1/8	2 5/8	3 5/8	2 5/8	3 5/8	2 5/8	3 5/8	2 5/8
	50%	3 1/8	1 5/8*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 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 H	50 V	100 H	100 V	150 H	150 V	200 H	200 V
6,000	0	1/2	1/2	1/2	1/2	5/8	1/2	5/8	1/2
12,000	0	5/8	5/8	5/8	5/8	7/8	5/8	7/8	5/8
18,000	0	7/8	7/8	7/8	7/8	7/8	7/8	7/8	7/8
24,000	0	7/8	7/8	7/8	7/8	7/8	7/8	1 1/8	7/8
36,000	0	7/8	7/8	1 1/8	7/8	1 1/8	7/8	1 1/8	1 1/8
48,000	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
60,000	0 to 33%	1 1/8	1 1/8	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
75,000	0 to 33%	1 1/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8	1 5/8	1 3/8
100,000	0 to 50%	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8
150,000	0 to 66%	1 3/8	1 3/8	1 5/8	1 5/8	1 5/8	1 5/8	2 1/8	1 5/8
200,000	0 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
300,000	0 to 50%	2 1/8	2 1/8	2 1/8	2 1/8	2 1/8	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
400,000	0 to 66%	2 1/8	2 1/8	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8
500,000	0 to 66%	2 1/8	2 1/8	2 5/8	2 1/8	2 5/8	2 1/8	2 5/8	2 5/8
600,000	0 to 66%	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	2 5/8	3 1/8	2 5/8
750,000	0 to 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

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 H	50 V	100 H	100 V	150 H	150 V	200 H	200 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	1 ⅛	⅞
24,000	0	⅞	⅞	⅞	⅞	1 ⅛	⅞	1 ⅛	⅞
36,000	0	1 ⅛	1 ⅛	1 ⅛	1 ⅛	1 ⅛	1 ⅛	1 ⅜	1 ⅛
48,000	0	1 ⅛	1 ⅛	1 ⅛	1 ⅛	1 ⅜	1 ⅛	1 ⅜	1 ⅛
60,000	0 to 33%	1 ⅛	1 ⅛	1 ⅜	1 ⅜	1 ⅜	1 ⅜	1 ⅜	1 ⅜
75,000	0 to 33%	1 ⅜	1 ⅜	1 ⅜	1 ⅜	1 ⅝	1 ⅜	1 ⅝	1 ⅜
100,000	0 to 50%	1 ⅜	1 ⅜	1 ⅝	1 ⅜	1 ⅝	1 ⅜	1 ⅝	1 ⅜
150,000	0 to 50%	1 ⅝	1 ⅝	2 ⅛	1 ⅝	2 ⅛	1 ⅝	2 ⅛	1 ⅝
	66%	1 ⅝	1 ⅝	1 ⅝	1 ⅝	1 ⅝	1 ⅝	1 ⅝	1 ⅝
200,000	0 to 50%	2 ⅛	2 ⅛	2 ⅛	2 ⅛	2 ⅛	2 ⅛	2 ⅛	2 ⅛
	66%	2 ⅛	1 ⅜ * 1 ⅝	2 ⅛	1 ⅜ * 1 ⅝	2 ⅛	1 ⅜ * 1 ⅝	2 ⅛	1 ⅜ * 1 ⅝
300,000	0 to 50%	2 ⅛	2 ⅛	2 ⅝	2 ⅛	2 ⅝	2 ⅛	2 ⅝	2 ⅝
	66%	2 ⅛	2 ⅛	2 ⅝	2 ⅛	2 ⅝	2 ⅛	2 ⅝	2 ⅛
400,000	0 to 50%	2 ⅝	2 ⅛	2 ⅝	2 ⅛	3 ⅛	2 ⅛	3 ⅛	2 ⅛
	66%	2 ⅝	2 ⅛	2 ⅝	2 ⅛	2 ⅝	1 ⅝ * 2 ⅛	2 ⅝	1 ⅝ * 2 ⅛
500,000	0 to 66%	2 ⅝	2 ⅝	2 ⅝	2 ⅝	3 ⅛	2 ⅝	3 ⅛	2 ⅝
600,000	0 to 66%	2 ⅝	2 ⅝	3 ⅛	2 ⅝	3 ⅝	2 ⅝	3 ⅝	2 ⅝
750,000	0 to 66%	3 ⅛	3 ⅛	3 ⅛	3 ⅛	3 ⅝	3 ⅛	3 ⅝	3 ⅛

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	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	7/8
24,000	0	1 1/8	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 3/8	1 1/8	1 3/8	1 1/8	1 3/8	1 1/8
48,000	0	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 3/8	1 5/8	1 3/8
60,000	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
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 5/8	1 5/8	1 5/8	1 5/8	2 1/8	1 5/8	2 1/8	1 5/8
	50%	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8	1 5/8
150,000	0 to 50%	2 1/8	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 5/8	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 5/8
	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
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	2 5/8
	66%	2 5/8	2 5/8	3 1/8	2 5/8	3 1/8	2 5/8	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 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 1/8	1 5/8 *2 5/8	3 1/8	1 5/8 *2 5/8
600,000	0 to 50%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	3 5/8	3 5/8
	66%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	3 5/8	2 1/8 *3 1/8
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 5/8
	66%	3 1/8	3 1/8	3 5/8	3 1/8	3 5/8	3 1/8	3 5/8	2 1/8 *3 1/8

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