CHAPTER 3

SYSTEM PRACTICES FOR AMMONIA AND CARBON DIOXIDE REFRIGERANTS

AMMONIA

CUSTOM-ENGINEERED ammonia (R-717) refrigeration systems often have design conditions that span a wide range of evaporating and condensing temperatures. Examples are (1) a food freezing plant operating from +50 to –50°F; (2) a candy storage requiring 60°F db with precise humidity control; (3) a beef chill room at 28 to 30°F with high humidity; (4) a distribution warehouse requiring multiple temperatures for storing ice cream, frozen food, meat, and produce and for docks; and (5) a chemical process requiring multiple temperatures ranging from +60 to –60°F. Ammonia is the refrigerant of choice for many industrial refrigeration systems.

The figures in this chapter are for illustrative purposes only, and may not show all the required elements (e.g., valves). For safety and minimum design criteria for ammonia systems, refer to ASHRAE *Standard* 15, IIAR *Bulletin* 109, IIAR *Standard* 2, and applicable state and local codes.

See Chapter 13 for information on refrigeration load calculations.

Ammonia Refrigerant for HVAC Systems

Using ammonia for HVAC systems has received renewed interest, in part because of the scheduled phaseout and increasing costs of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants. Ammonia secondary systems that circulate chilled water or another secondary refrigerant are a viable alternative to halocarbon systems, although ammonia is inappropriate for direct refrigeration systems (ammonia in the air unit coils) for HVAC applications. Ammonia packaged chilling units are available for HVAC applications. As with the installation of any air-conditioning unit, all applicable codes, standards, and insurance requirements must be followed.

SYSTEM SELECTION

In selecting an engineered ammonia refrigeration system, several design decisions must be considered, including whether to use (1) single-stage compression, (2) economized compression, (3) multistage compression, (4) direct-expansion feed, (5) flooded feed, (6) liquid recirculation feed, and (7) secondary coolants.

Single-Stage Systems

The basic single-stage system consists of evaporator(s), a compressor, a condenser, a refrigerant receiver (if used), and a refrigerant control device (expansion valve, float, etc.). Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals* discusses the compression refrigeration cycle.

The preparation of this chapter is assigned to TC 10.3, Refrigerant Piping.

Economized Systems

Economized systems are frequently used with rotary screw compressors. Figure 1 shows an arrangement of the basic components. Subcooling the liquid refrigerant before it reaches the evaporator reduces its enthalpy, resulting in a higher net refrigerating effect. Economizing is beneficial because the vapor generated during subcooling is injected into the compressor partway through its compression cycle and must be compressed only from the economizer port pressure (which is higher than suction pressure) to the discharge pressure. This produces additional refrigerating capacity with less increase in unit energy input. Economizing is most beneficial at high pressure ratios. Under most conditions, economizing can provide operating efficiencies that approach that of two-stage systems, but with much less complexity and simpler maintenance.

Economized systems for variable loads should be selected carefully. At approximately 75% capacity, most screw compressors revert to single-stage performance as the slide valve moves such that the economizer port is open to the compressor suction area.

A flash economizer, which is somewhat more efficient, may often be used instead of the shell-and-coil economizer (Figure 1). However, ammonia liquid delivery pressure is reduced to economizer pressure.

Multistage Systems

Multistage systems compress gas from the evaporator to the condenser in several stages. They are used to produce temperatures of –15°F and below. This is not economical with single-stage compression.

Single-stage reciprocating compression systems are generally limited to between 5 and 10 psig suction pressure. With lubricant-injected economized rotary screw compressors, where the discharge

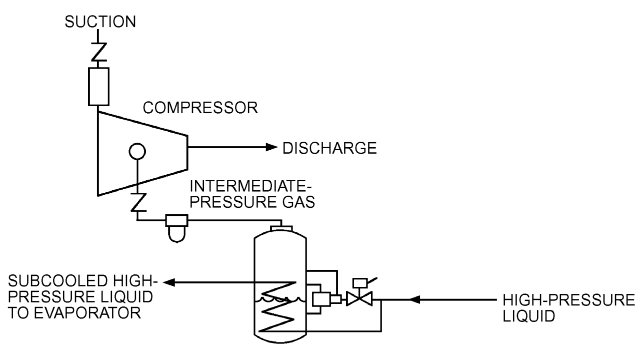


Fig. 1 Shell-and-Coil Economizer Arrangement

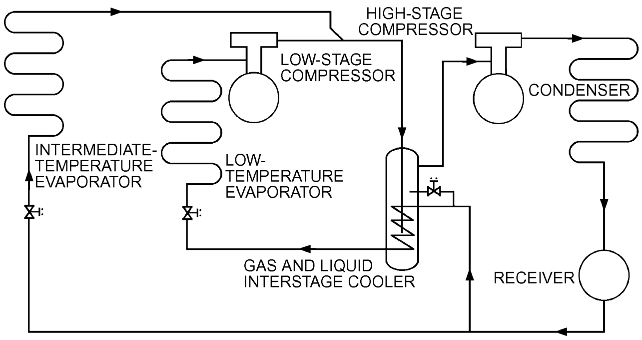


Fig. 2 Two-Stage System with High- and Low-Temperature Loads

temperatures are lower because of the lubricant cooling, the low-suction temperature limit is about –40°F, but efficiency is very low. Two-stage systems are used down to about –70 or –80°F evaporator temperatures. Below this temperature, three-stage systems should be considered.

Two-stage systems consist of one or more compressors that operate at low suction pressure and discharge at intermediate pressure and have one or more compressors that operate at intermediate pressure and discharge to the condenser (Figure 2).

Where either single- or two-stage compression systems can be used, two-stage systems require less power and have lower operating costs, but they can have a higher initial equipment cost.

EQUIPMENT

Compressors

Compressors available for single- and multistage applications include the following:

- Reciprocating
  - Single-stage (low-stage or high-stage)
  - Internally compounded
- Rotary vane
- Rotary screw (low-stage or high-stage, with or without economizing)

The reciprocating compressor is the most common compressor used in small, 100 hp or less, single-stage or multistage systems. The screw compressor is the predominant compressor above 100 hp, in both single- and multistage systems. Various combinations of compressors may be used in multistage systems. Rotary vane and screw compressors are frequently used for the low-pressure stage, where large volumes of gas must be moved. The high-pressure stage may be a reciprocating or screw compressor.

When selecting a compressor, consider the following:

- System size and capacity requirements.
- Location, such as indoor or outdoor installation at ground level or on the roof.
- Equipment noise.
- Part- or full-load operation.
- Winter and summer operation.
- Pulldown time required to reduce the temperature to desired conditions for either initial or normal operation. The temperature must be pulled down frequently for some applications for a process load, whereas a large cold-storage warehouse may require pulldown only once in its lifetime.

Lubricant Cooling. When a reciprocating compressor requires lubricant cooling, an external heat exchanger using a refrigerant or secondary cooling is usually added. Screw compressor lubricant cooling is covered in detail in the section on Screw Compressors.

Compressor Drives. The correct electric motor size(s) for a multistage system is determined by pulldown load. When the final low-stage operating level is –100°F, the pulldown load can be three times the operating load. Positive-displacement reciprocating compressor motors are usually selected for about 150% of operating power requirements for 100% load. The compressor's unloading mechanism can be used to prevent motor overload. Electric motors should not be overloaded, even when a service factor is indicated. For screw compressor applications, motors should be sized by adding 10% to the operating power. Screw compressors have built-in unloading mechanisms to prevent motor overload. The motor should not be oversized, because an oversized motor has a lower power factor and lower efficiency at design and reduced loads.

Steam turbines or gasoline, natural gas, propane, or diesel internal combustion engines are used when electricity is unavailable, or if the selected energy source is cheaper. Sometimes they are used in combination with electricity to reduce peak demands. The power output of a given engine size can vary as much as 15% depending on the fuel selected.

Steam turbine drives for refrigerant compressors are usually limited to very large installations where steam is already available at moderate to high pressure. In all cases, torsional analysis is required to determine what coupling must be used to dampen out any pulsations transmitted from the compressor. For optimum efficiency, a turbine should operate at a high speed that must be geared down for reciprocating and possibly screw compressors. Neither the gear reducer nor the turbine can tolerate a pulsating backlash from the driven end, so torsional analysis and special couplings are essential.

Advantages of turbines include variable speed for capacity control and low operating and maintenance costs. Disadvantages include higher initial costs and possible high noise levels. The turbine must be started manually to bring the turbine housing up to temperature slowly and to prevent excess condensate from entering the turbine.

The standard power rating of an engine is the absolute maximum, not the recommended power available for continuous use. Also, torque characteristics of internal combustion engines and electric motors differ greatly. The proper engine selection is at 75% of its maximum power rating. For longer life, the full-load speed should be at least 10% below maximum engine speed.

Internal combustion engines, in some cases, can reduce operating cost below that for electric motors. Disadvantages include (1) higher initial cost of the engine, (2) additional safety and starting controls, (3) higher noise levels, (4) larger space requirements, (5) air pollution, (6) requirement for heat dissipation, (7) higher maintenance costs, and (8) higher levels of vibration than with electric motors. A torsional analysis must be made to determine the proper coupling if engine drives are chosen.

Condensers

Condensers should be selected on the basis of total heat rejection at maximum load. Often, the heat rejected at the start of pulldown is several times the amount rejected at normal, low-temperature operating conditions. Some means, such as compressor unloading, can be used to limit the maximum amount of heat rejected during pulldown. If the condenser is not sized for pulldown conditions, and compressor capacity cannot be limited during this period, condensing pressure might increase enough to shut down the system.

Evaporators

Several types of evaporators are used in ammonia refrigeration systems. Fan-coil, direct-expansion evaporators can be used, but they are not generally recommended unless the suction temperature is 0°F or higher. This is due in part to the relative inefficiency of the direct-expansion coil, but more importantly, the low mass flow rate of ammonia is difficult to feed uniformly as a liquid to the coil. Instead, ammonia fan-coil units designed for recirculation (overfeed) systems are preferred. Typically, in this type of system, high-pressure ammonia from the system high stage flashes into a large vessel at the evaporator pressure, from which it is pumped to the evaporators at an overfeed rate of 2.5 to 1 to 4 to 1. This type of system is standard and very efficient. See Chapter 1 for more details.

Flooded shell-and-tube evaporators are often used in ammonia systems in which indirect or secondary cooling fluids such as water, brine, or glycol must be cooled.

Some problems that can become more acute at low temperatures include changes in lubricant transport properties, loss of capacity caused by static head from the depth of the pool of liquid refrigerant in the evaporator, deterioration of refrigerant boiling heat transfer coefficients caused by lubricant logging, and higher specific volumes for the vapor.

The effect of pressure losses in the evaporator and suction piping is more acute in low-temperature systems because of the large change in saturation temperatures and specific volume in relation to pressure changes at these conditions. Systems that operate near zero absolute pressure are particularly affected by pressure loss.

The depth of the pool of boiling refrigerant in a flooded evaporator exerts a liquid pressure on the lower part of the heat transfer surface. Therefore, the saturation temperature at this surface is higher than that in the suction line, which is not affected by the liquid pressure. This temperature gradient must be considered when designing the evaporator.

Spray shell-and-tube evaporators, though not commonly used, offer certain advantages. In this design, the evaporator's liquid depth penalty can be eliminated because the pool of liquid is below the heat transfer surface. A refrigerant pump sprays liquid over the surface. Pump energy is an additional heat load to the system, and more refrigerant must be used to provide the net positive suction head (NPSH) required by the pump. The pump is also an additional item that must be maintained. This evaporator design also reduces the refrigerant charge requirement compared to a flooded design (see Chapter 1).

Vessels

High-Pressure Receivers. Industrial systems generally incorporate a central high-pressure refrigerant receiver, which serves as the primary refrigerant storage location in the system. It handles refrigerant volume variations between the condenser and the system's low side during operation and pumpdowns for repairs or defrost. Ideally, the receiver should be large enough to hold the entire system charge, but this is not generally economical. The system should be analyzed to determine the optimum receiver size. Receivers are commonly equalized to the condenser inlet and operate at the same pressure as the condenser. In some systems, the receiver is operated at a pressure between the condensing pressure and the highest suction pressure to allow for variations in condensing pressure without affecting the system's feed pressure.

If additional receiver capacity is needed for normal operation, use extreme caution in the design. Designers usually remove the inadequate receiver and replace it with a larger one rather than install an additional receiver in parallel. This procedure is best because even slight differences in piping pressure or temperature can cause the refrigerant to migrate to one receiver and not to the other.

Smaller auxiliary receivers can be incorporated to serve as sources of high-pressure liquid for compressor injection or thermosiphon, lubricant cooling, high-temperature evaporators, and so forth.

Intercoolers (Gas and Liquid). An intercooler (subcooler/desuperheater) is the intermediate vessel between the high and low stages in a multistage system. One purpose is to cool discharge gas of the low-stage compressor to prevent overheating the high-stage compressor. This can be done by bubbling discharge gas from the low-stage compressor through a bath of liquid refrigerant or by mixing liquid normally entering the intermediate vessel with the discharge gas as it enters above the liquid level. Heat removed from the discharge gas is absorbed by the evaporation of part of the liquid and eventually passes through the high-stage compressor to the condenser. Disbursing the discharge gas below a level of liquid refrigerant separates out any lubricant carryover from the low-stage compressor. If liquid in the intercooler is to be used for other purposes, such as liquid makeup or feed to the low stage, periodic lubricant removal is important.

Another purpose of the intercooler is to lower the temperature of liquid used in the system low stage. Lowering refrigerant temperature increases the refrigeration effect and reduces the low-stage compressor's required displacement, thus reducing its operating cost.

Intercoolers for two-stage compression systems can be shell-and-coil or flash. Figure 3 depicts a shell-and-coil intercooler incorporating an internal pipe coil for subcooling high-pressure liquid before it is fed to the low stage of the system. Typically, the coil subcools liquid to within 10°F of the intermediate temperature.

Vertical **shell-and-coil intercoolers** with float valve feed perform well in many applications using ammonia refrigerant systems. The vessel must be sized properly to separate liquid from vapor that is returning to the high-stage compressor. The superheated gas inlet pipe should extend below the liquid level and have perforations or slots to distribute the gas evenly in small bubbles. Adding a perforated baffle across the area of the vessel slightly below the liquid level protects against violent surging. A float switch that shuts down the high-stage compressor when the liquid level gets too high should always be used with this type of intercooler in case the feed valve fails to control properly.

The **flash intercooler** is similar in design to the shell-and-coil intercooler, except for the coil. The high-pressure liquid is flash-cooled to the intermediate temperature. Use caution in selecting a flash intercooler because all the high-pressure liquid is flashed to intermediate pressure. Though colder than that of the shell-and-coil intercooler, liquid in the flash intercooler is not subcooled and is susceptible to flashing from system pressure drop. Two-phase liquid feed to control valves may cause premature failure because of the wire-drawing effect of the liquid/vapor mixture.

Figure 4 shows a vertical shell-and-coil intercooler as piped into the system. The liquid level is maintained in the intercooler by a float that controls the solenoid valve feeding liquid into the shell side of the intercooler. Gas from the first-stage compressor enters the lower section of the intercooler, is distributed by a perforated

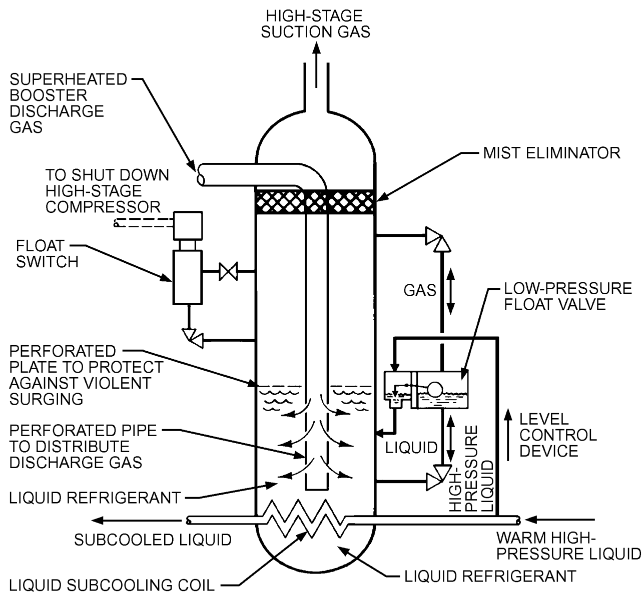


Fig. 3 Intercooler

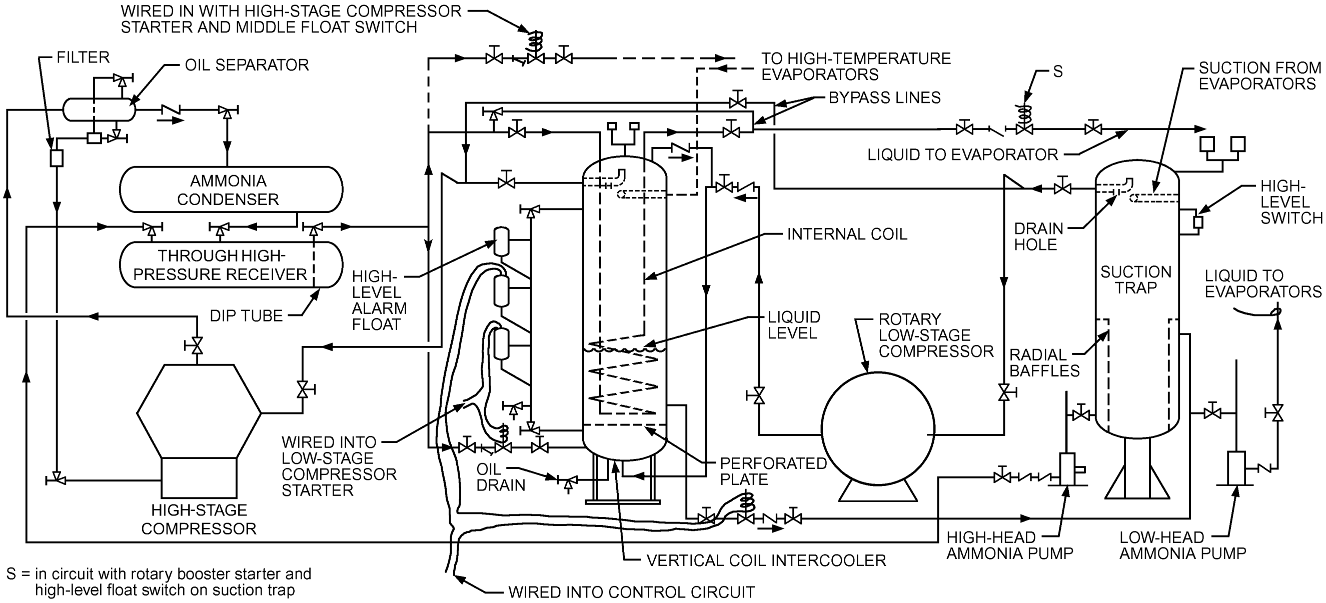


Fig. 4 Arrangement for Compound System with Vertical Intercooler and Suction Trap

plate, and is then cooled to the saturation temperature corresponding to intermediate pressure.

When sizing any intercooler, the designer must consider (1) low-stage compressor capacity; (2) vapor desuperheating, liquid make-up requirements for the subcooling coil load, or vapor cooling load associated with the flash intercooler; and (3) any high-stage side loading. The volume required for normal liquid levels, liquid surging from high-stage evaporators, feed valve malfunctions, and liquid/vapor must also be analyzed.

Necessary accessories are the liquid level control device and high-level float switch. Though not absolutely necessary, an auxiliary oil pot should also be considered.

Suction Accumulator. A suction accumulator (also known as a knockout drum, suction trap, pump receiver, recirculator, etc.) prevents liquid from entering the suction of the compressor, whether on the high stage or low stage of the system. Both vertical and horizontal vessels can be incorporated. Baffling and mist eliminator pads can enhance liquid separation.

Suction accumulators, especially those not intentionally maintaining a level of liquid, should have a means to remove any build-up of ammonia liquid. Gas boil-out coils or electric heating elements are costly and inefficient.

Although it is one of the more common and simplest means of liquid removal, a liquid boil-out coil (Figure 5) has some drawbacks. Generally, warm liquid flowing through the coil is the source of liquid being boiled off. Liquid transfer pumps, gas-powered transfer systems, or basic pressure differentials are a more positive means of removing the liquid (Figures 6 and 7).

Accessories should include a high-level float switch for compressor protection along with additional pump or transfer system controls.

Vertical Suction Trap and Pump. Figure 8 shows the piping of a vertical suction trap that uses a high-head ammonia pump to transfer liquid from the system's low-pressure side to the high-pressure receiver. Float switches piped on a float column on the side of the trap can start and stop the liquid ammonia pump, sound an alarm in case of excess liquid, and sometimes stop the compressors.

When the liquid level in the suction trap reaches the setting of the middle float switch, the liquid ammonia pump starts and reduces the liquid level to the setting of the lower float switch, which stops the liquid ammonia pump. A check valve in the discharge line

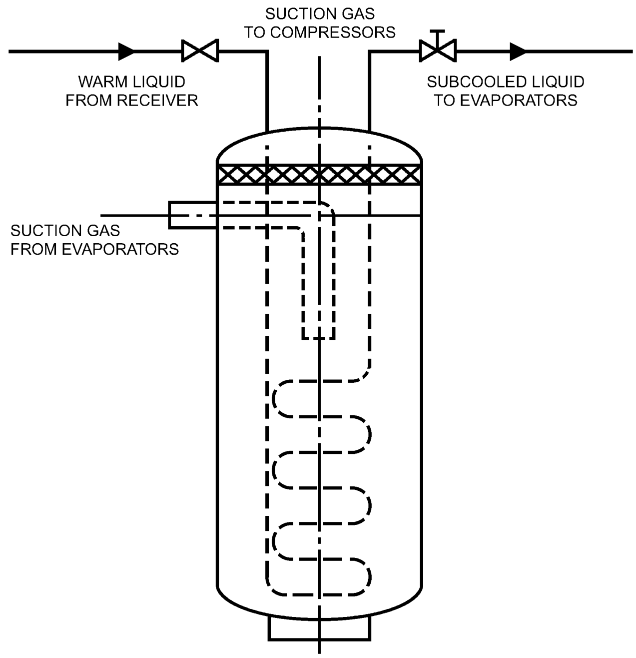


Fig. 5 Suction Accumulator with Warm Liquid Coil

of the ammonia pump prevents gas and liquid from flowing backward through the pump when it is not in operation. Depending on the type of check valve used, some installations have two valves in a series as an extra precaution against pump backspin.

Compressor controls adequately designed for starting, stopping, and capacity reduction result in minimal agitation, which helps separate vapor and liquid in the suction trap. Increasing compressor capacity slowly and in small increments reduces liquid boiling in the trap, which is caused by the refrigeration load of cooling the refrigerant and metal mass of the trap. If another compressor is started when plant suction pressure increases, it should be brought on line slowly to prevent a sudden pressure change in the suction trap.

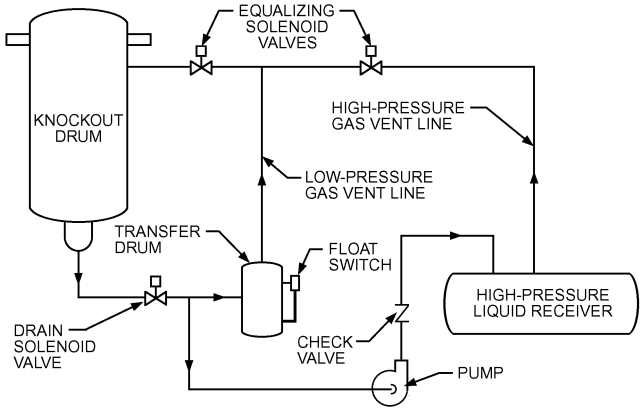


Fig. 6 Equalized Pressure Pump Transfer System

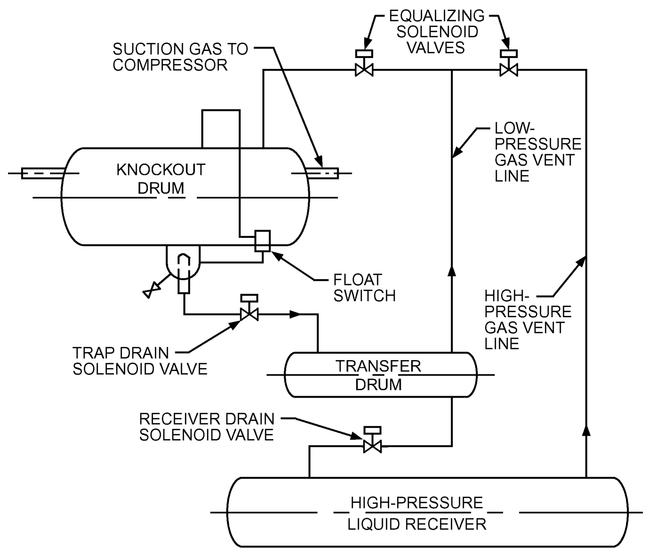


Fig. 7 Gravity Transfer System

A high level of liquid in a suction trap should activate an alarm or stop the compressors. Although eliminating the cause is the most effective way to reduce a high level of excess surging liquid, a more immediate solution is to stop part of the compression system and raise plant suction pressure slightly. Continuing high levels indicate insufficient pump capacity or suction trap volume.

Liquid Level Indicators. Liquid level can be indicated by visual indicators, electronic sensors, or a combination of the two. Visual indicators include individual circular reflex level indicators (bull's-eyes) mounted on a pipe column or stand-alone linear reflex glass assemblies (Figure 9). For operation at temperatures below the frost point, transparent plastic frost shields covering the reflex surfaces are necessary. Also, the pipe column must be insulated, especially when control devices are attached.

Electronic level sensors can continuously monitor liquid level. Digital or graphic displays of liquid level can be locally or remotely monitored (Figure 10).

Level indicators should have adequate isolation valves. High-temperature glass tube indicators should incorporate stop check or excess-flow valves for isolation and safety.

Purge Units. A noncondensable gas separator (purge unit) is useful in most plants, especially when suction pressure is below atmospheric pressure. Purge units on ammonia systems are piped to carry noncondensables (air) from the receiver and condenser to the

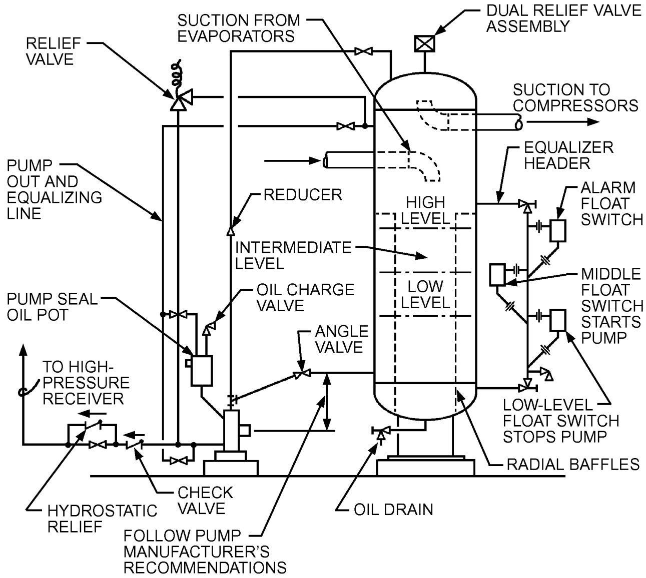


Fig. 8 Piping for Vertical Suction Trap and High-Head Pump

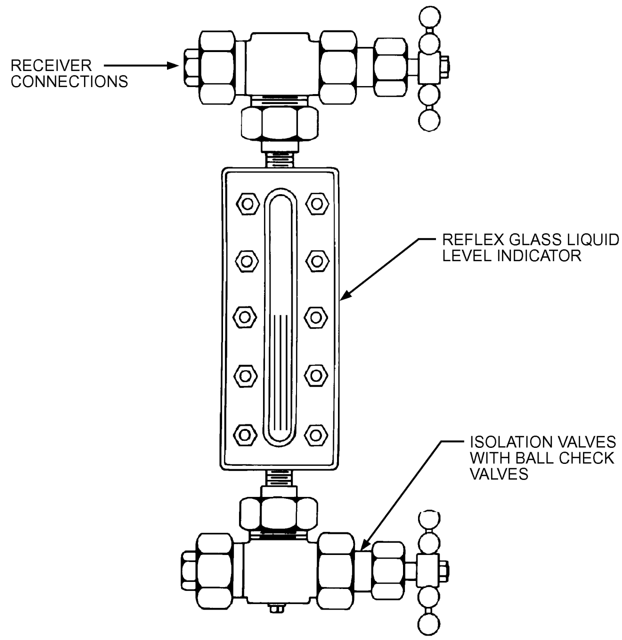


Fig. 9 Gage Glass Assembly for Ammonia

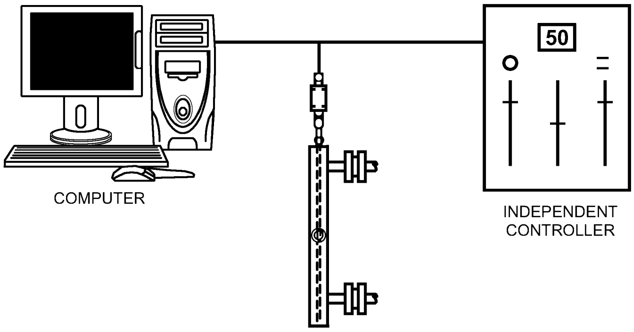


Fig. 10 Electronic Liquid Level Control

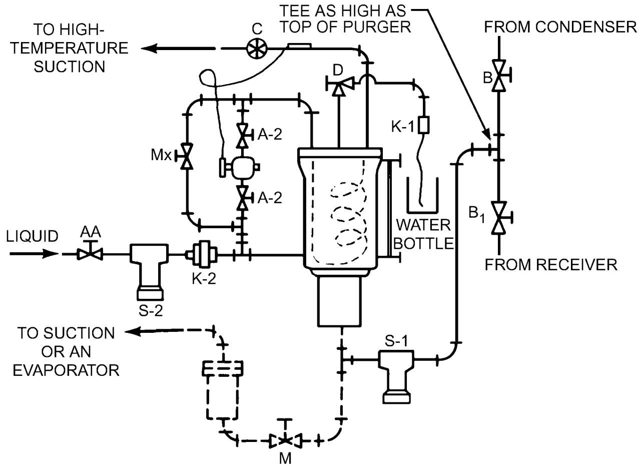


Fig. 11 Purge Unit and Piping for Noncondensable Gas

purger, as shown in Figure 11. High-pressure liquid expands through a coil in the purge unit, providing a cold spot in the purge drum. The suction from the coil should be taken to one of the low-temperature suction mains. Ammonia vapor and noncondensable gas are drawn into the purge drum, and the ammonia condenses on the cold surface. When the drum fills with air and other noncondensables, a float valve in the purger opens and allows them to leave the drum and pass into the open water bottle. Purge units are available for automatic operation.

Lubricant Management

Most lubricants are immiscible in ammonia and separate out of the liquid easily when flow velocity is low or when temperatures are lowered. Normally, lubricants can be easily drained from the system. However, if the temperature is very low and the lubricant is not properly selected, it becomes a gummy mass that prevents refrigerant controls from functioning, blocks flow passages, and fouls heat transfer surfaces. Proper lubricant selection and management is often the key to a properly functioning system.

In two-stage systems, proper design usually calls for lubricant separators on both the high- and low-stage compressors. A properly designed coalescing separator can remove almost all the lubricant that is in droplet or aerosol form. Lubricant that reaches its saturation vapor pressure and becomes a vapor cannot be removed by a separator. Separators with some means of cooling the discharge gas condense much of the vapor for consequent separation. Selection of lubricants that have very low vapor pressures below 180°F can minimize carryover to 2 or 3 ppm. Care must be taken, however, to ensure that refrigerant is not condensed and fed back into the compressor or separator, where it can lower lubricity and cause compressor damage.

In general, direct-expansion and liquid overfeed system evaporators have fewer lubricant return problems than do flooded system evaporators because refrigerant flows continuously at good velocities to sweep lubricant from the evaporator. Low-temperature systems using hot-gas defrost can also be designed to sweep lubricant out of the circuit each time the system defrosts. This reduces the possibility of coating the evaporator surface and hindering heat transfer.

Flooded evaporators can promote lubricant build-up in the evaporator charge because they may only return refrigerant vapor back to the system. In ammonia systems, the lubricant is simply drained from the surge drum. At low temperatures, this procedure is difficult if the lubricant selected has a pour point above the evaporator temperature.

Lubricant Removal from Ammonia Systems. Most lubricants are miscible with liquid ammonia only in very small proportions. The proportion decreases with the temperature, causing lubricant to separate. The evaporation of ammonia increases the lubricant ratio, causing more lubricant to separate. Increased density causes the lubricant (saturated with ammonia at the existing pressure) to form a separate layer below the ammonia liquid.

Unless lubricant is removed periodically or continuously from the point where it collects, it can cover the heat transfer surface in the evaporator, reducing performance. If gage lines or branches to level controls are taken from low points (or lubricant is allowed to accumulate), these lines will contain lubricant. The higher lubricant density is at a lower level than the ammonia liquid. Draining lubricant from a properly located collection point is not difficult unless the temperature is so low that the lubricant does not flow readily. In this case, keeping the receiver at a higher temperature may be beneficial. Alternatively, a lubricant with a lower pour point can be selected.

Lubricant in the system is saturated with ammonia at the existing pressure. When the pressure is reduced, ammonia vapor separates, causing foaming.

Draining lubricant from ammonia systems requires special care. Ammonia in lubricant foam normally starts to evaporate and produces a smell. Operators should be made aware of this. On systems where lubricant is drained from a still, a spring-loaded drain valve, which closes if the valve handle is released, should be installed.

CONTROLS

Refrigerant flow controls are discussed in Chapter 44. The following precautions are necessary in the application of certain controls in low-temperature systems.

Liquid Feed Control

Many controls available for single-stage, high-temperature systems may be used with some discretion on low-temperature systems. If the liquid level is controlled by a low-side float valve (with the float in the chamber where the level is controlled), low pressure and temperature have no appreciable effect on operation. External float chambers, however, must be thoroughly insulated to prevent heat influx that might cause boiling and an unstable level, affecting the float response. Equalizing lines to external float chambers, particularly the upper line, must be sized generously so that liquid can reach the float chamber, and gas resulting from any evaporation may be returned to the vessel without appreciable pressure loss.

The superheat-controlled (thermostatic) expansion valve is generally used in direct-expansion evaporators. This valve operates on the difference between bulb pressure, which is responsive to suction temperature, and pressure below the diaphragm, which is the actual suction pressure.

The thermostatic expansion valve is designed to maintain a preset superheat in suction gas. Although the pressure-sensing part of the system responds almost immediately to a change in conditions, the temperature-sensing bulb must overcome thermal inertia before its effect is felt on the power element of the valve. Thus, when compressor capacity decreases suddenly, the expansion valve may overfeed before the bulb senses the presence of liquid in the suction line and reduces the feed. Therefore, a suction accumulator should be installed on direct-expansion low-temperature systems with multiple expansion valves.

Controlling Load During Pulldown

System transients during pulldown can be managed by controlling compressor capacity. Proper load control reduces compressor capacity so that energy requirements stay within the motor and condenser capacities. On larger systems using screw compressors, a current-sensing device reads motor amperage and adjusts the capacity control device appropriately. Cylinders on reciprocating compressors can be unloaded for similar control.

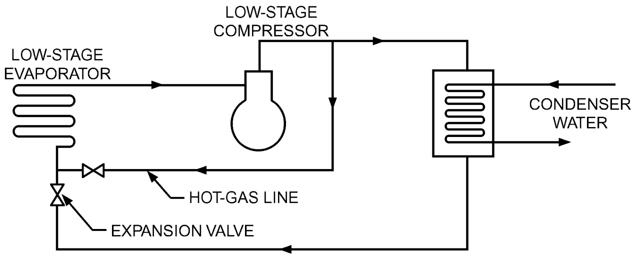


Fig. 12 Hot-Gas Injection Evaporator for Operations at Low Load

Alternatively, a downstream, outlet, or crankcase pressure regulator can be installed in the suction line to throttle suction flow if the pressure exceeds a preset limit. This regulator limits the compressor's suction pressure during pulldown. The disadvantage of this device is the extra pressure drop it causes when the system is at the desired operating conditions. To overcome some of this, the designer can use external forces to drive the valve, causing it to be held fully open when the pressure is below the maximum allowable. Systems using downstream pressure regulators and compressor unloading must be carefully designed so that the two controls complement each other.

Operation at Varying Loads and Temperatures

Compressor and evaporator capacity controls are similar for multi- and single-stage systems. Control methods include compressor capacity control, hot-gas bypass, or evaporator pressure regulators. Low pressure can affect control systems by significantly increasing the specific volume of the refrigerant gas and the pressure drop. A small pressure reduction can cause a large percentage capacity reduction.

System load usually cannot be reduced to near zero, because this results in little or no flow of gas through the compressor and consequent overheating. Additionally, high pressure ratios are detrimental to the compressor if it is required to run at very low loads. If the compressor cannot be allowed to cycle off during low load, an acceptable alternative is a **hot-gas bypass**. High-pressure gas is fed to the low-pressure side of the system through a downstream pressure regulator. The gas should be desuperheated by injecting it at a point in the system where it is in contact with expanding liquid, such as immediately downstream of the liquid feed to the evaporator. Otherwise, extremely high compressor discharge temperatures can result. The artificial load supplied by high-pressure gas can fill the gap between the actual load and the lowest stable compressor operating capacity. Figure 12 shows such an arrangement.

Electronic Control

Microprocessor- and computer-based control systems are becoming the norm for control systems on individual compressors as well as for entire system control. Almost all screw compressors use microprocessor control systems to monitor all safety functions and operating conditions. These machines are frequently linked together with a programmable controller or computer for sequencing multiple compressors so that they load and unload in response to system fluctuations in the most economical manner. Programmable controllers are also used to replace multiple defrost time clocks on larger systems for more accurate and economical defrosting. Communications and data logging allow systems to operate at optimum conditions under transient load conditions even when operators are not in attendance.

PIPING

Local codes or ordinances governing ammonia mains should be followed, in addition to the recommendations here.

Recommended Material

Because copper and copper-bearing materials are attacked by ammonia, they are not used in ammonia piping systems. Steel piping, fittings, and valves of the proper pressure rating are suitable for ammonia gas and liquid.

Ammonia piping should conform to ASME *Standard* B31.5, and to IIAR *Standard* 2, which states the following:

1. Liquid lines 1.5 in. and smaller shall be not less than Schedule 80 carbon steel pipe.
2. Liquid lines 2 through 6 in. shall be not less than Schedule 40 carbon steel pipe.
3. Liquid lines 8 through 12 in. shall be not less than Schedule 20 carbon steel pipe.
4. Vapor lines 6 in. and smaller shall be not less than Schedule 40 carbon steel pipe.
5. Vapor lines 8 through 12 in. shall be not less than Schedule 20 carbon steel pipe.
6. Vapor lines 14 in. and larger shall be not less than Schedule 10 carbon steel pipe.
7. All threaded pipe shall be Schedule 80.
8. Carbon steel pipe shall be ASTM *Standard* A53 Grade A or B, Type E (electric resistance welded) or Type S (seamless); or ASTM *Standard* A106 (seamless), except where temperature-pressure criteria mandate a higher specification material. *Standard* A53 Type F is not permitted for ammonia piping.

Fittings

Couplings, elbows, and tees for threaded pipe are for a minimum of 3000 psi design pressure and constructed of forged steel. Fittings for welded pipe should match the type of pipe used (i.e., standard fittings for standard pipe and extra-heavy fittings for extra-heavy pipe).

Tongue-and-groove or ANSI flanges should be used in ammonia piping. Welded flanges for low-side piping can have a minimum 150 psi design pressure rating. On systems located in high ambients, low-side piping and vessels should be designed for 200 to 225 psig. The high side should be 250 psig if the system uses water-cooled or evaporative cooled condensing. Use 300 psig minimum for air-cooled designs.

Pipe Joints

Joints between lengths of pipe or between pipe and fittings can be threaded if the pipe size is 1.25 in. or smaller. Pipe 1.5 in. or larger should be welded. An all-welded piping system is superior.

Threaded Joints. Many sealants and compounds are available for sealing threaded joints. The manufacturer's instructions cover compatibility and application method. Do not use excessive amounts or apply on female threads because any excess can contaminate the system.

Welded Joints. Pipe should be cut and beveled before welding. Use pipe alignment guides and provide a proper gap between pipe ends so that a full-penetration weld is obtained. The weld should be made by a qualified welder, using proper procedures such as the *Welding Procedure Specifications*, prepared by the National Certified Pipe Welding Bureau (NCPWB).

Gasketed Joints. A compatible fiber gasket should be used with flanges. Before tightening flange bolts to valves, controls, or flange unions, properly align pipe and bolt holes. When flanges are used to straighten pipe, they put stress on adjacent valves, compressors, and controls, causing the operating mechanism to bind. To prevent leaks, flange bolts are drawn up evenly when connecting the flanges. Flanges at compressors and other system components must not move or indicate stress when all bolts are loosened.

Union Joints. Steel (3000 psi) ground joint unions are used for gage and pressure control lines with screwed valves and for joints up to 0.75 in. When tightening this type of joint, the two pipes must be axially aligned. To be effective, the two parts of the union

Table 1 Suction Line Capacities in Tons for Ammonia with Pressure Drops of 0.25 and 0.50°F per 100 ft Equivalent

Steel Line Size		Saturated Suction Temperature, °F					
		–60		–40		–20	
IPS	SCH	Δt = 0.25°F Δp = 0.046	Δt = 0.50°F Δp = 0.092	Δt = 0.25°F Δp = 0.077	Δt = 0.50°F Δp = 0.155	Δt = 0.25°F Δp = 0.123	Δt = 0.50°F Δp = 0.245
3/8	80	0.03	0.05	0.06	0.09	0.11	0.16
1/2	80	0.06	0.10	0.12	0.18	0.22	0.32
3/4	80	0.15	0.22	0.28	0.42	0.50	0.73
1	80	0.30	0.45	0.57	0.84	0.99	1.44
1 1/4	40	0.82	1.21	1.53	2.24	2.65	3.84
1 1/2	40	1.25	1.83	2.32	3.38	4.00	5.80
2	40	2.43	3.57	4.54	6.59	7.79	11.26
2 1/2	40	3.94	5.78	7.23	10.56	12.50	18.03
3	40	7.10	10.30	13.00	18.81	22.23	32.09
4	40	14.77	21.21	26.81	38.62	45.66	65.81
5	40	26.66	38.65	48.68	70.07	82.70	119.60
6	40	43.48	62.83	79.18	114.26	134.37	193.44
8	40	90.07	129.79	163.48	235.38	277.80	397.55
10	40	164.26	236.39	297.51	427.71	504.98	721.08
12	ID*	264.07	379.88	477.55	686.10	808.93	1157.59
Steel Line Size		**Saturated Suction Temperature, °F**					
		0		**20**		**40**	
IPS	**SCH**	Δt = 0.25°F Δp = 0.184	Δt = 0.50°F Δp = 0.368	Δt = 0.25°F Δp = 0.265	Δt = 0.50°F Δp = 0.530	Δt = 0.25°F Δp = 0.366	Δt = 0.50°F Δp = 0.582
3/8	80	0.18	0.26	0.28	0.40	0.41	0.53
1/2	80	0.36	0.52	0.55	0.80	0.82	1.05
3/4	80	0.82	1.18	1.26	1.83	1.87	2.38
1	40	1.62	2.34	2.50	3.60	3.68	4.69
1 1/4	40	4.30	6.21	6.63	9.52	9.76	12.42
1 1/2	40	6.49	9.34	9.98	14.34	14.68	18.64
2	40	12.57	18.12	19.35	27.74	28.45	36.08
2 1/2	40	20.19	28.94	30.98	44.30	45.37	57.51
3	40	35.87	51.35	54.98	78.50	80.40	101.93
4	40	73.56	105.17	112.34	160.57	164.44	208.34
5	40	133.12	190.55	203.53	289.97	296.88	376.18
6	40	216.05	308.62	329.59	469.07	480.96	609.57
8	40	444.56	633.82	676.99	962.47	985.55	1250.34
10	40	806.47	1148.72	1226.96	1744.84	1786.55	2263.99
12	ID*	1290.92	1839.28	1964.56	2790.37	2862.23	3613.23

Note: Capacities are in tons of refrigeration resulting in a line friction loss (Δp in psi per 100 ft equivalent pipe length), with corresponding change (Δt in °F per 100 ft) in saturation temperature.

*The inside diameter of the pipe is the same as the nominal pipe size.

must match perfectly. Ground joint unions should be avoided if at all possible.

Pipe Location

Piping should be at least 7.5 ft above the floor. Locate pipes carefully in relation to other piping and structural members, especially when lines are to be insulated. The distance between insulated lines should be at least three times the thickness of the insulation for screwed fittings, and four times for flange fittings. The space between the pipe and adjacent surfaces should be three-fourths of these amounts.

Hangers located close to the vertical risers to and from compressors keep the piping weight off the compressor. Pipe hangers should be placed no more than 8 to 10 ft apart and within 2 ft of a change in direction of the piping. Hangers should be designed to bear on the outside of insulated lines. Sheet metal sleeves on the lower half of the insulation are usually sufficient. Where piping penetrates a wall, a sleeve should be installed, and where the pipe penetrating the wall is insulated, it must be adequately sealed.

Piping to and from compressors and to other components must provide for expansion and contraction. Sufficient flange or union joints should be located in the piping so components can be assembled easily during installation and also disassembled for servicing.

Pipe Sizing

Table 1 presents practical suction line sizing data based on 0.25°F and 0.50°F differential pressure drop equivalent per 100 ft total equivalent length of pipe. For data on equivalent lengths of valves and fittings, refer to Tables 10, 11, and 12 in Chapter 2. Table 2 lists data for sizing suction and discharge lines at 1°F differential pressure drop equivalent per 100 ft equivalent length of pipe, and for sizing liquid lines at 100 fpm. Charts prepared by Wile (1977) present pressure drops in saturation temperature equivalents. For a complete discussion of the basis of these line sizing charts, see Timm (1991). Table 3 presents line sizing information for pumped liquid lines, high-pressure liquid lines, hot-gas defrost lines, equalizing lines, and thermosiphon lubricant cooling ammonia lines.

Valves

Stop Valves. These valves should be placed in the inlet and outlet lines to all condensers, vessels, evaporators, and long lengths of

Table 2 Suction, Discharge, and Liquid Line Capacities in Tons for Ammonia (Single- or High-Stage Applications)

Steel Line Size		Suction Lines (Δt = 1°F) Saturated Suction Temperature, °F					Discharge Lines	Steel Line Size		Liquid Lines	
IPS	SCH	–40 Δp = 0.31	–20 Δp = 0.49	0 Δp = 0.73	20 Δp = 1.06	40 Δp = 1.46	Δt = 1°F Δp = 2.95	IPS	SCH	Velocity = 100 fpm	Δp =2.0 psi Δt = 0.7°F
3/8	80	—	—	—	—	—	—	3/8	80	8.6	12.1
1/2	80	—	—	—	—	—	3.1	1/2	80	14.2	24.0
3/4	80	—	—	—	2.6	3.8	7.1	3/4	80	26.3	54.2
1	80	—	2.1	3.4	5.2	7.6	13.9	1	80	43.8	106.4
1 1/4	40	3.2	5.6	8.9	13.6	19.9	36.5	1 1/4	80	78.1	228.6
1 1/2	40	4.9	8.4	13.4	20.5	29.9	54.8	1 1/2	80	107.5	349.2
2	40	9.5	16.2	26.0	39.6	57.8	105.7	2	40	204.2	811.4
2 1/2	40	15.3	25.9	41.5	63.2	92.1	168.5	2 1/2	40	291.1	1292.6
3	40	27.1	46.1	73.5	111.9	163.0	297.6	3	40	449.6	2287.8
4	40	55.7	94.2	150.1	228.7	333.0	606.2	4	40	774.7	4662.1
5	40	101.1	170.4	271.1	412.4	600.9	1095.2	5	40	—	—
6	40	164.0	276.4	439.2	667.5	971.6	1771.2	6	40	—	—
8	40	337.2	566.8	901.1	1366.6	1989.4	3623.0	8	40	—	—
10	40	611.6	1027.2	1634.3	2474.5	3598.0	—	10	40	—	—
12	ID*	981.6	1644.5	2612.4	3963.5	5764.6	—	12	ID*	—	—

Notes:

1. Table capacities are in tons of refrigeration.
   Δp = pressure drop due to line friction, psi per 100 ft of equivalent line length
   Δt = corresponding change in saturation temperature, °F per 100 ft
2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.55}$$

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

$$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.8}$$

4. Values based on 90°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures:

Condensing Temperature, °F	Suction Lines	Discharge Lines
70	1.05	0.78
80	1.02	0.89
90	1.00	1.00
100	0.98	1.11

5. Discharge and liquid line capacities based on 20°F suction. Evaporator temperature is 0°F. The capacity is affected less than 3% when applied from –40 to +40°F extremes.

*The inside diameter of the pipe is the same as the nominal pipe size.

Table 3 Liquid Ammonia Line Capacities

(Capacity in tons of refrigeration, except as noted)

Nominal Size, in.	Pumped Liquid Overfeed Ratio 3:1	4:1	5:1	High-Pressure Liquid at 3 psi[a]	Hot-Gas Defrost[a]	Equalizer High Side[b]	Thermosiphon Lubricant Cooling Lines Gravity Flow,[c] 1000 Btu/h Supply	Return	Vent
1/2	10	7.5	6	30	—	—	—	—	—
3/4	22	16.5	13	69	4	50	—	—	—
1	43	32.5	26	134	8	100	—	—	—
1 1/4	93.5	70	56	286	20	150	—	—	—
1 1/2	146	110	87.5	439	30	225	200	120	203
2	334	250	200	1016	50	300	470	300	362
2 1/2	533	400	320	1616	92	500	850	530	638
3	768	576	461	2886	162	1000	1312	870	1102
4	1365	1024	819	—	328	2000	2261	1410	2000
5	—	—	—	—	594	—	3550	2214	3624
6	—	—	—	—	970	—	5130	3200	6378
8	—	—	—	—	—	—	8874	5533	11596

Source: Wile (1977).

[a]Hot-gas line sizes are based on 1.5 psi pressure drop per 100 ft of equivalent length at 100 psig discharge pressure and three times evaporator refrigeration capacity.

[b]Line sizes based on experience using total system evaporator tons.

[c]From Frick Co. (1995). Values for line sizes above 4 in. are extrapolated.

pipe so they can be isolated in case of leaks and to facilitate pumping out by evacuation. Sections of liquid piping that can be valved off and isolated must be protected with a relief device.

Installing globe-type stop valves with the valve stems horizontal lessens the chance (1) for dirt or scale to lodge on the valve seat or disk and cause it to leak or (2) for liquid or lubricant to pocket in the area below the seat. Wet suction return lines (recirculation system) should use angle valves to reduce the possibility of liquid pockets and reduce pressure drop.

Welded flanged or weld-in-line valves are desirable for all line sizes; however, screwed valves may be used for 1 1/4 in. and smaller lines. Ammonia globe and angle valves should have the following features:

- Soft seating surfaces for positive shutoff (no copper or copper alloy)
- Back seating to permit repacking the valve stem while in service
- Arrangement that allows packing to be tightened easily

- All-steel construction (preferable)
- Bolted bonnets above 1 in., threaded bonnets for 1 in. and smaller

Consider seal cap valves in refrigerated areas and for all ammonia piping. To keep pressure drop to a minimum, consider angle valves (as opposed to globe valves).

Control Valves. Pressure regulators, solenoid valves, and thermostatic expansion valves should be flanged for easy assembly and removal. Valves 1.5 in. and larger should have welded companion flanges. Smaller valves can have threaded companion flanges.

A strainer should be used in front of self-contained control valves to protect them from pipe construction material and dirt. A ceramic filter installed in the pilot line to the power piston protects the close tolerances from foreign material when pilot-operated control valves are used.

Solenoid Valves. Solenoid valve stems should be upright, with their coils protected from moisture. They should have flexible conduit connections, where allowed by codes, and an electric pilot light wired in parallel to indicate when the coil is energized. A manual opening stem is useful for emergencies.

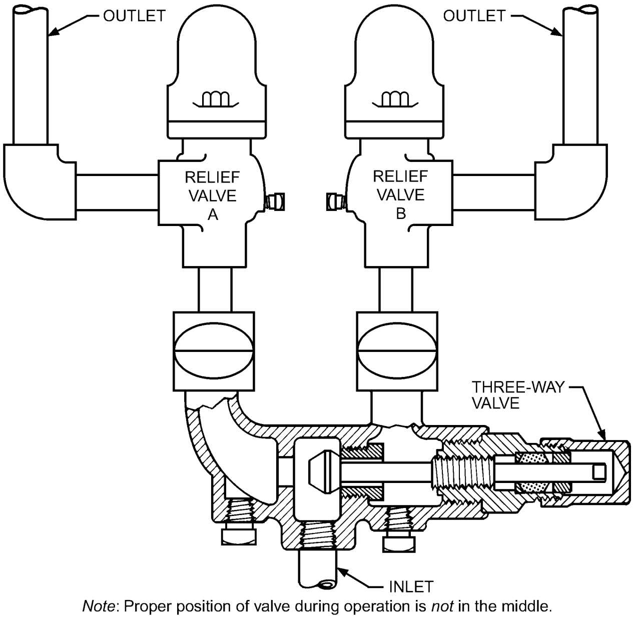
Solenoid valves for high-pressure liquid feed to evaporators should have soft seats for positive shutoff. Solenoid valves for other applications, such as in suction, hot-gas, or gravity feed lines, should be selected for the pressure and temperature of the fluid flowing and for the pressure drop available.

Relief Valves. Safety valves must be provided in conformance with ASHRAE *Standard* 15 and Section VIII, Division 1, of the ASME *Boiler and Pressure Vessel Code*. For ammonia systems, IIAR *Bulletin* 109 also addresses the subject of safety valves.

Dual relief valve arrangements allow testing of the relief valves (Figure 13). The three-way stop valve is constructed so that it is always open to one of the relief valves if the other is removed to be checked or repaired.

Isolated Line Sections

Sections of piping that can be isolated between hand valves or check valves can be subjected to extreme hydraulic pressures if cold liquid refrigerant is trapped in them and subsequently warmed. Additional safety valves for such piping must be provided.



Note: Proper position of valve during operation is *not* in the middle.

Fig. 13 Dual Relief Valve Fitting for Ammonia

Insulation and Vapor Retarders

Chapter 33 covers insulation and vapor retarders. Insulation and effective vapor retarders on low-temperature systems are very important. At low temperatures, the smallest leak in the vapor retarder can allow ice to form inside the insulation, which can totally destroy the integrity of the entire insulation system. The result can significantly increase load and power usage.

RECIPROCATING COMPRESSORS

Piping

Figure 14 shows a typical piping arrangement for two compressors operating in parallel off the same suction main. Suction mains should be laid out with the objective of returning only clean, dry gas to the compressor. This usually requires a suction trap sized adequately for gravity gas and liquid separation based on permissible gas velocities for specific temperatures. A dead-end trap can usually trap only scale and lubricant. As an alternative, a shell-and-coil accumulator with a warm liquid coil may be considered. Suction mains running to and from the suction trap or accumulator should be pitched toward the trap at 1/8 in. per foot for liquid drainage.

In sizing suction mains and takeoffs from mains to compressors, consider how the pressure drop in the selected piping affects the compressor size required. First costs and operating costs for compressor and piping selections should be optimized.

Good suction line systems have a total friction drop of 1 to 3°F pressure drop equivalent. Practical suction line friction losses should not exceed 0.5°F equivalent per 100 ft equivalent length.

A well-designed discharge main has a total friction loss of 1 to 2 psi. Generally, a slightly oversized discharge line is desirable to hold down discharge pressure and, consequently, discharge temperature and energy costs. Where possible, discharge mains should be pitched (1/8 in/ft) toward the condenser, without creating a liquid trap; otherwise, pitch should be toward the discharge line separator.

High- and low-pressure cutouts and gages and lubricant pressure failure cutout are installed on the compressor side of the stop valves to protect the compressor.

Lubricant Separators. Lubricant separators are located in the discharge line of each compressor (Figure 14A). A high-pressure float valve drains lubricant back into the compressor crankcase or lubricant receiver. The separator should be placed as far from the compressor as possible, so the extra pipe length can be used to cool the discharge gas before it enters the separator. This reduces the temperature of the ammonia vapor and makes the separator more effective.

Liquid ammonia must not reach the crankcase. Often, a valve (preferably automatic) is installed in the drain from the lubricant separator, open only when the temperature at the bottom of the separator is higher than the condensing temperature. Some manufacturers install a small electric heater at the bottom of a vertical lubricant trap instead. The heater is actuated when the compressor is not operating. Separators installed in cold conditions must be insulated to prevent ammonia condensation.

A filter is recommended in the drain line on the downstream side of the high-pressure float valve.

Lubricant Receivers. Figure 14B illustrates two compressors on the same suction line with one discharge-line lubricant separator. The separator float drains into a lubricant receiver, which maintains a reserve supply of lubricant for the compressors. Compressors should be equipped with crankcase floats to regulate lubricant flow to the crankcase.

Discharge Check Valves and Discharge Lines. Discharge check valves on the downstream side of each lubricant separator prevent high-pressure gas from flowing into an inactive compressor and causing condensation (Figure 14A).

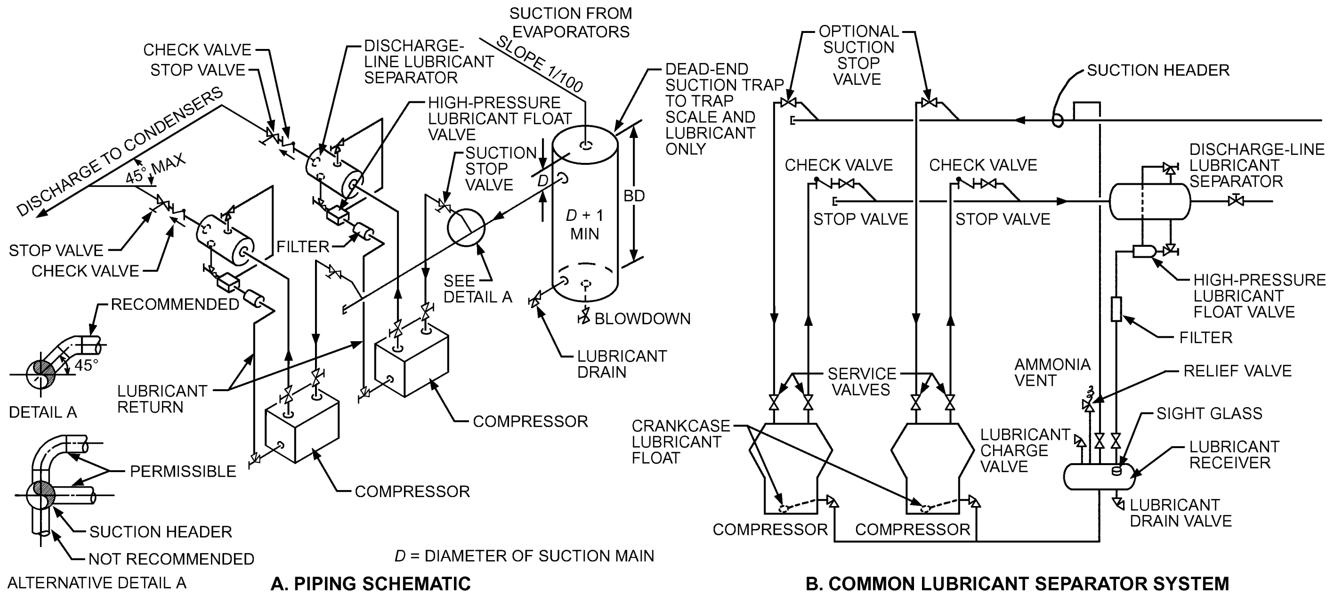


Fig. 14 Schematic of Reciprocating Compressors Operating in Parallel

The discharge line from each compressor should enter the discharge main at a 45° maximum angle in the horizontal plane so the gas flows smoothly.

Unloaded Starting. Unloaded starting is frequently needed to stay within the torque or current limitations of the motor. Most compressors are unloaded either by holding the suction valve open or by external bypassing. Control can be manual or automatic.

Suction Gas Conditioning. Suction main piping should be insulated, complete with vapor retarder to minimize thermal losses, to prevent sweating and/or ice build-up on the piping, and to limit superheat at the compressor. Additional superheat results in increased discharge temperatures and reduces compressor capacity. Low discharge temperatures in ammonia plants are important to reduce lubricant carryover and because compressor lubricant can carbonize at higher temperatures, which can cause cylinder wall scoring and lubricant sludge throughout the system. Discharge temperatures above 250°F should be avoided at all times. Lubricants should have flash-point temperatures above the maximum expected compressor discharge temperature.

Cooling

Generally, ammonia compressors are constructed with internally cast cooling passages along the cylinders and/or in the top heads. These passages provide space for circulating a heat transfer medium, which minimizes heat conduction from the hot discharge gas to the incoming suction gas and lubricant in the compressor's crankcase. An external lubricant cooler is supplied on most reciprocating ammonia compressors. Water is usually the medium circulated through these passages (**water jackets**) and the lubricant cooler at a rate of about 0.1 gpm per ton of refrigeration. Lubricant in the crankcase (depending on type of construction) is about 120°F. Temperatures above this level reduce the lubricant's lubricating properties.

For compressors operating in ambients above 32°F, water flow is sometimes controlled entirely by hand valves, although a solenoid valve in the inlet line is desirable to automate the system. When the compressor stops, water flow must be stopped to keep residual gas from condensing and to conserve water. A water-regulating valve, installed in the water supply line with the sensing bulb in the water return line, is also recommended. This type of cooling is shown in Figure 15.

The thermostat in the water line leaving the jacket serves as a safety cutout to stop the compressor if the temperature becomes too high.

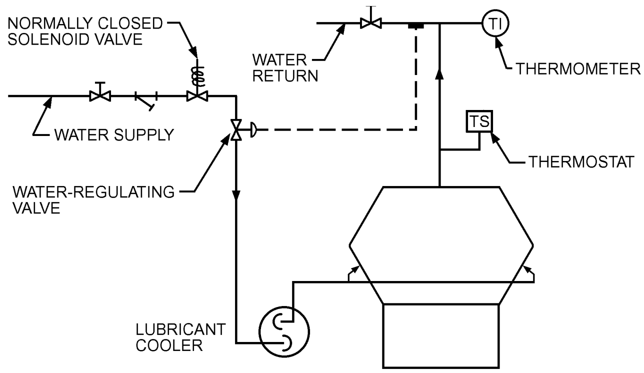


Fig. 15 Jacket Water Cooling for Ambient Temperatures Above Freezing

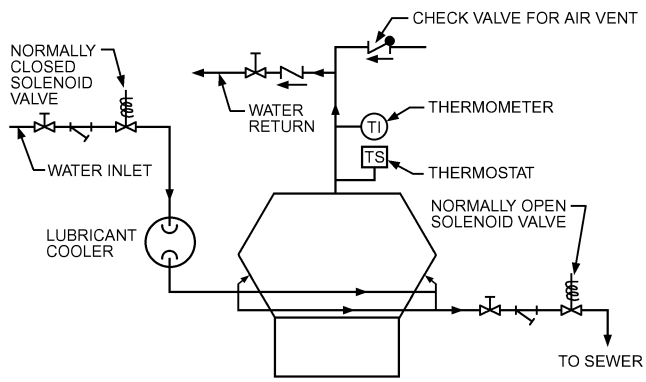


Fig. 16 Jacket Water Cooling for Ambient Temperatures Below Freezing

For compressors where ambient temperatures may be below 32°F, a means for draining the jacket on shutdown to prevent freeze-up must be provided. One method is shown in Figure 16. Water flow is through the normally closed solenoid valve, which is energized when the compressor starts. Water then circulates through the lubricant cooler and the jacket, and out through the water return line. When the compressor stops, the solenoid valve in the water inlet line

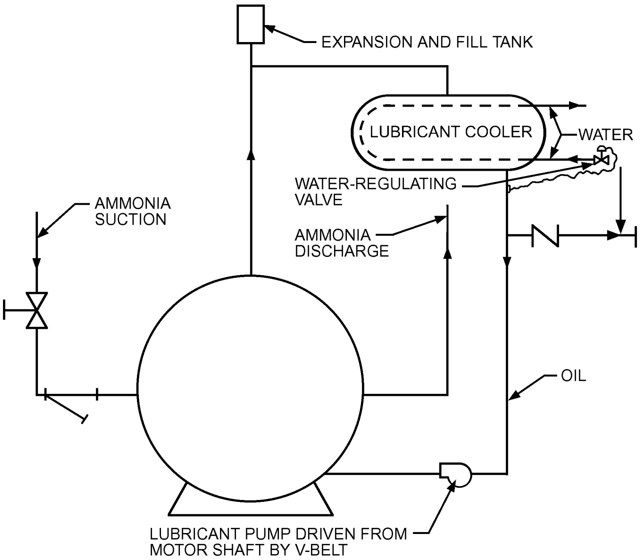


Fig. 17 Rotary Vane Booster Compressor Cooling with Lubricant

is deenergized and stops water flow to the compressor. At the same time, the solenoid valve opens to drain the water out of the low point to wastewater treatment. The check valves in the air vent lines open when pressure is relieved and allow the jacket and cooler to be drained. Each flapper check valve is installed so that water pressure closes it, but absence of water pressure allows it to swing open.

For compressors in spaces below 32°F or where water quality is very poor, cooling is best handled by using an inhibited glycol solution or other suitable fluid in the jackets and lubricant cooler and cooling with a secondary heat exchanger. This method for cooling reciprocating ammonia compressors eliminates fouling of the lubricant cooler and jacket normally associated with city water or cooling tower water.

ROTARY VANE, LOW-STAGE COMPRESSORS

Piping

Rotary vane compressors have been used extensively as low-stage compressors in ammonia refrigeration systems. Now, however, the screw compressor has largely replaced the rotary vane compressor for ammonia low-stage compressor applications. Piping requirements for rotary vane compressors are the same as for reciprocating compressors. Most rotary vane compressors are lubricated by injectors because they have no crankcase. In some designs, a lubricant separator, lubricant receiver, and cooler are required on the discharge of these compressors; a pump recirculates lubricant to the compressor for both cooling and lubrication. In other rotary vane compressor designs, a discharge lubricant separator is not used, and lubricant collects in the high-stage suction accumulator or intercooler, from which it may be drained. Lubricant for the injectors must periodically be added to a reservoir.

Cooling

The compressor jacket is cooled by circulating a cooling fluid, such as water or lubricant. Lubricant is recommended, because it will not freeze and can serve both purposes (Figure 17).

SCREW COMPRESSORS

Piping

Helical screw compressors are the choice for most industrial refrigeration systems. All helical screw compressors have a constant-volume (displacement) design. The volume index V_i refers to the internal volume ratio of the compressor. There are three types of screw compressors:

- Fixed V_i with slide valve
- Variable V_i with slide valve and slide stop
- Fixed V_i with bypass ports in lieu of slide valve

When V_i is fixed, the compressor functions most efficiently at a certain absolute compression ratio (CR). In selecting a fixed-V_i compressor, the average CR rather than the maximum CR should be considered. A guide to proper compressor selection is based on the equation V_i^k = CR, where k = 1.4 for ammonia.

For example, for a screw compressor at 10°F (38.5 psia) and 95°F (195.8 psia) with CR = 5.09, $V_i^{1.4}$ = 5.09 and V_i = 3.20. Thus, a compressor with V_i = 3.6 might be the best choice. If the ambient conditions are such that the average condensing temperature is 75°F (140.5 psia), then the CR is 3.65 and the ideal V_i is 2.52. Thus, a compressor with V_i = 2.4 is the proper selection to optimize efficiency.

Fixed-V_i compressors with bypass ports in lieu of a slide valve are often applied as booster compressors, which normally have a V_i requirement of less than 2.9.

A variable-V_i compressor makes compressor selection simpler because it can vary its volume index from 2.0 to 5.0; thus, it can automatically match the internal pressure ratio in the compressor with the external pressure ratio.

Typical flow diagrams for screw compressor packages are shown in Figures 18 (for indirect cooling) and 19 (for direct cooling with refrigerant liquid injection). Figure 20 illustrates a variable-V_i compressor that does not require a full-time lube pump but rather a pump to prelube the bearings. Full-time lube pumps are required when fixed- or variable-V_i compressors are used as low-stage compressors. Lubrication systems require at least a 75 psi pressure differential for proper operation.

Lubricant Cooling

Lubricant in screw compressors may be cooled three ways:

- Liquid refrigerant injection
- Indirect cooling with glycol or water in a heat exchanger
- Indirect cooling with boiling high-pressure refrigerant used as the coolant in a thermosiphon process

Refrigerant injection cooling is shown schematically in Figures 19 and 21. Depending on the application, this cooling method usually decreases compressor efficiency and capacity but lowers equipment cost. Most screw compressor manufacturers publish a derating curve for this type of cooling. Injection cooling for low-stage compression has little or no penalty on compressor efficiency or capacity. However, efficiency can be increased by using an indirectly cooled lubricant cooler. With this configuration, heat from the lubricant cooler is removed by the evaporative condenser or cooling tower and is not transmitted to the high-stage compressors.

Refrigerant liquid for liquid-injection oil cooling must come from a dedicated supply. The source may be the system receiver or a separate receiver; a 5 min uninterrupted supply of refrigerant liquid is usually adequate.

Indirect or thermosiphon lubricant cooling for low-stage screw compressors rejects the lubricant cooling load to the condenser or auxiliary cooling system; this load is not transferred to the high-stage compressor, which improves system efficiency. Indirect lubricant cooling systems using glycol or water reject the lubricant cooling load to a section of an evaporative condenser, a separate evaporative cooler, or a cooling tower. A three-way lubricant control valve should be used to control lubricant temperature.

Thermosiphon lubricant cooling is the industry standard. In this system, high-pressure refrigerant liquid from the condenser, which boils at condensing temperature/pressure (usually 90 to 95°F design), cools lubricant in a tubular heat exchanger. Typical thermosiphon

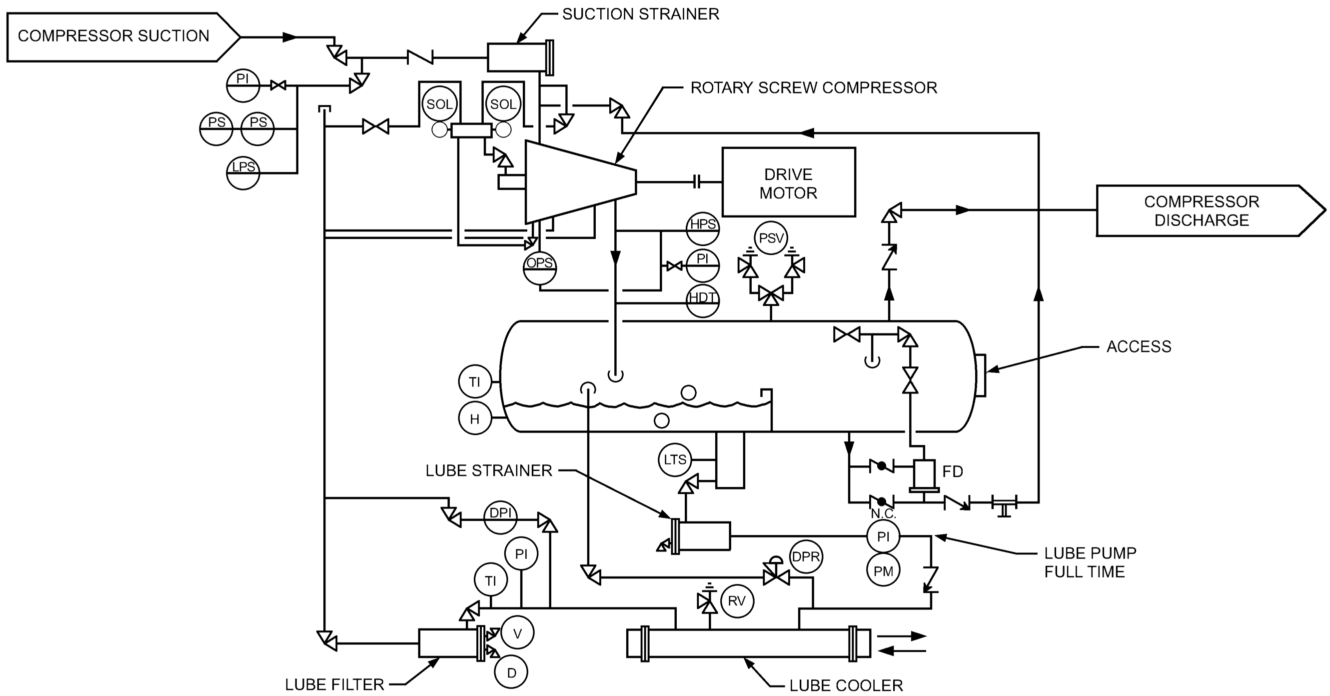


Fig. 18 Fixed-V_i Screw Compressor Flow Diagram with Indirect Lubricant Cooling

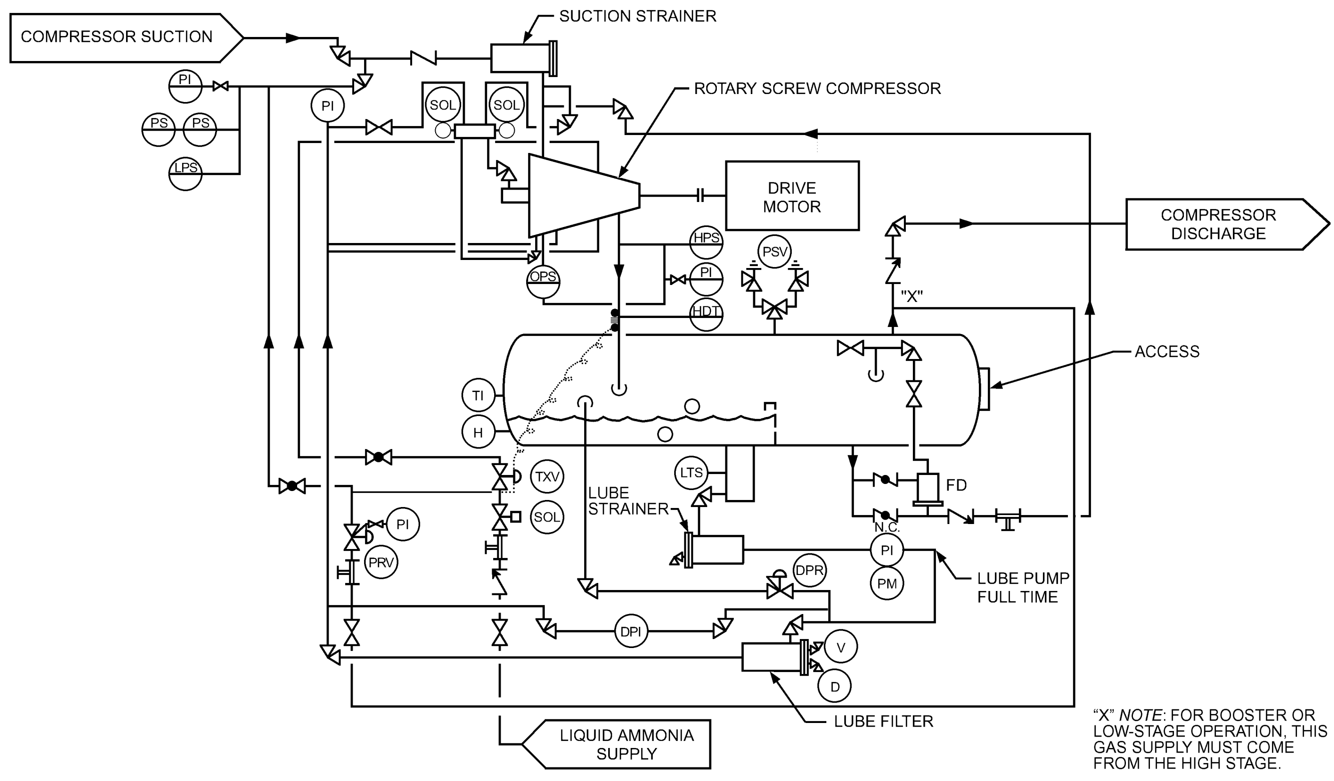


Fig. 19 Fixed-V_i Screw Compressor Flow Diagram with Liquid Injection Cooling

lubricant cooling arrangements are shown in Figures 18, 20, 22, 23, and 24. Note on all figures that the refrigerant liquid supply to the lubricant cooler receives priority over the feed to the system low side. It is important that the gas equalizing line (vent) off the top of the thermosiphon receiver be adequately sized to match the lubricant cooler load to prevent the thermosiphon receiver from becoming gas-bound.

Figure 25 shows a typical capacity control system for a fixed-V_i screw compressor. The four-way valve controls the slide valve position and thus the compressor capacity from typically 100 to 10% with a signal from an electric, electronic, or microprocessor controller. The slide valve unloads the compressor by bypassing vapor back to the suction of the compressor.

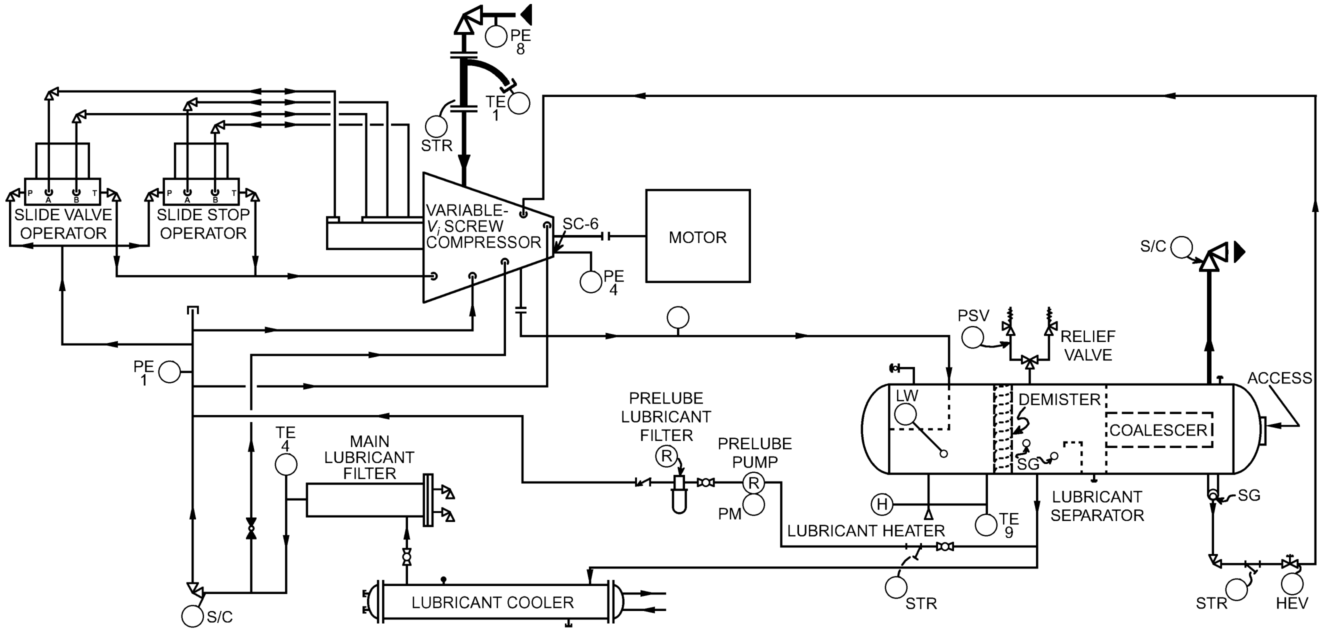


Fig. 20 Flow Diagram for Variable-V_i Screw Compressor High-Stage Only

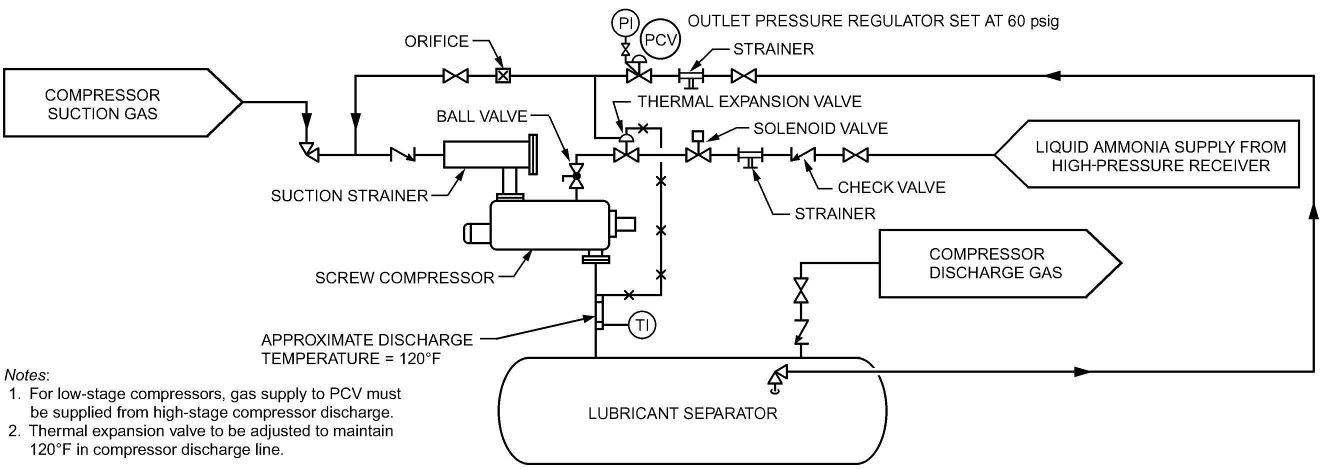


Fig. 21 Flow Diagram for Screw Compressors with Refrigerant Injection Cooling

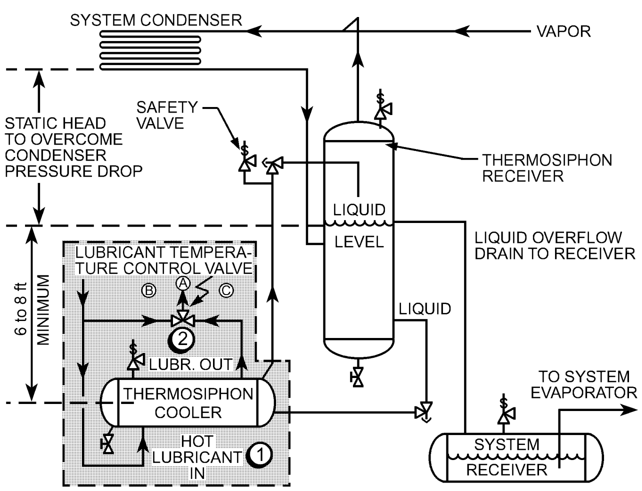


Fig. 22 Typical Thermosiphon Lubricant Cooling System with Thermosiphon Accumulator

Figure 26 shows a typical capacity and volume index control system in which two four-way control valves take their signals from a computer controller. One four-way valve controls capacity by positioning the slide valve in accordance with the load, and the other positions the slide stop to adjust the compressor internal pressure ratio to match system suction and discharge pressure. The slide valve works the same as that on fixed-V_i compressors. Volume index is varied by adjusting the slide stop on the discharge end of the compressor.

Screw compressor piping should generally be installed in the same manner as for reciprocating compressors. Although screw compressors can ingest some liquid refrigerant, they should be protected against liquid carryover. Screw compressors are furnished with both suction and discharge check valves.

CONDENSER AND RECEIVER PIPING

Properly designed piping around the condensers and receivers keeps the condensing surface at its highest efficiency by draining liquid ammonia out of the condenser as soon as it condenses and keeping air and other noncondensables purged.

Horizontal Shell-and-Tube Condenser and Through-Type Receiver

Figure 27 shows a horizontal water-cooled condenser draining into a through (top inlet) receiver. Ammonia plants do not always require controlled water flow to maintain pressure. Usually, pressure is adequate to force the ammonia to the various evaporators without water regulation. Each situation should be evaluated by comparing water costs with input power cost savings at lower condenser pressures.

Water piping should be arranged so that condenser tubes are always filled with water. Air vents should be provided on condenser heads and should have hand valves for manual purging.

Receivers must be below the condenser so that the condensing surface is not flooded with ammonia. The piping should provide (1) free drainage from the condenser and (2) static height of ammonia above the first valve out of the condenser greater than the pressure drop through the valve.

The drain line from condenser to receiver is designed on the basis of 100 fpm maximum velocity to allow gas equalization between condenser and receiver. Refer to Table 2 for sizing criteria.

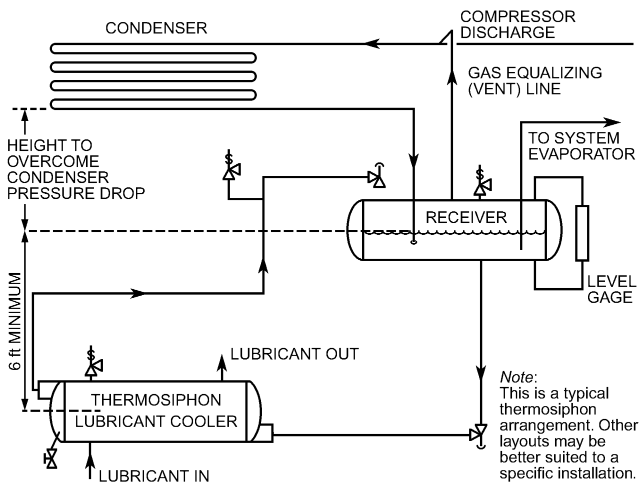


Fig. 23 Thermosiphon Lubricant Cooling System with Receiver Mounted Above Thermosiphon Lubricant Cooler

Parallel Horizontal Shell-and-Tube Condensers

Figure 28 shows two condensers operating in parallel with one through-type (top inlet) receiver. The length of horizontal liquid drain lines to the receiver should be minimized, with no traps permitted. Equalization between the shells is achieved by keeping liquid velocity in the drain line less than 100 fpm. The drain line can be sized from Table 2.

EVAPORATIVE CONDENSERS

Evaporative condensers are selected based on the wet-bulb temperature in which they operate. The 1% design wet bulb is that wet-bulb temperature that will be equalled or exceeded 1% of the months of June through September, or 29.3 h. Thus, for the majority of industrial plants that operate at least at part load all year, the wet-bulb temperature is below design 99.6% of the operating time. The resultant condensing pressure will only equal or exceed the design condition during 0.4% of the time if the design wet-bulb temperature and peak design refrigeration load occur coincidentally. This peak condition is more a function of how the load is calculated, what load diversity factor exists or is used in the calculation, and what safety factor is used in the calculations, than of the size of the condenser.

Location

If an evaporative condenser is located with insufficient space for air movement, the effect is the same as that imposed by an inlet damper, and the fan may not deliver enough air. In addition, evaporative condenser discharge air may recirculate, which adds to the problem. The high inlet velocity causes a low-pressure region to develop around the fan inlet, inducing flow of discharge air into that region. If the obstruction is from a second condenser, the problem can be even more severe because discharge air from the second condenser flows into the air intake of the first.

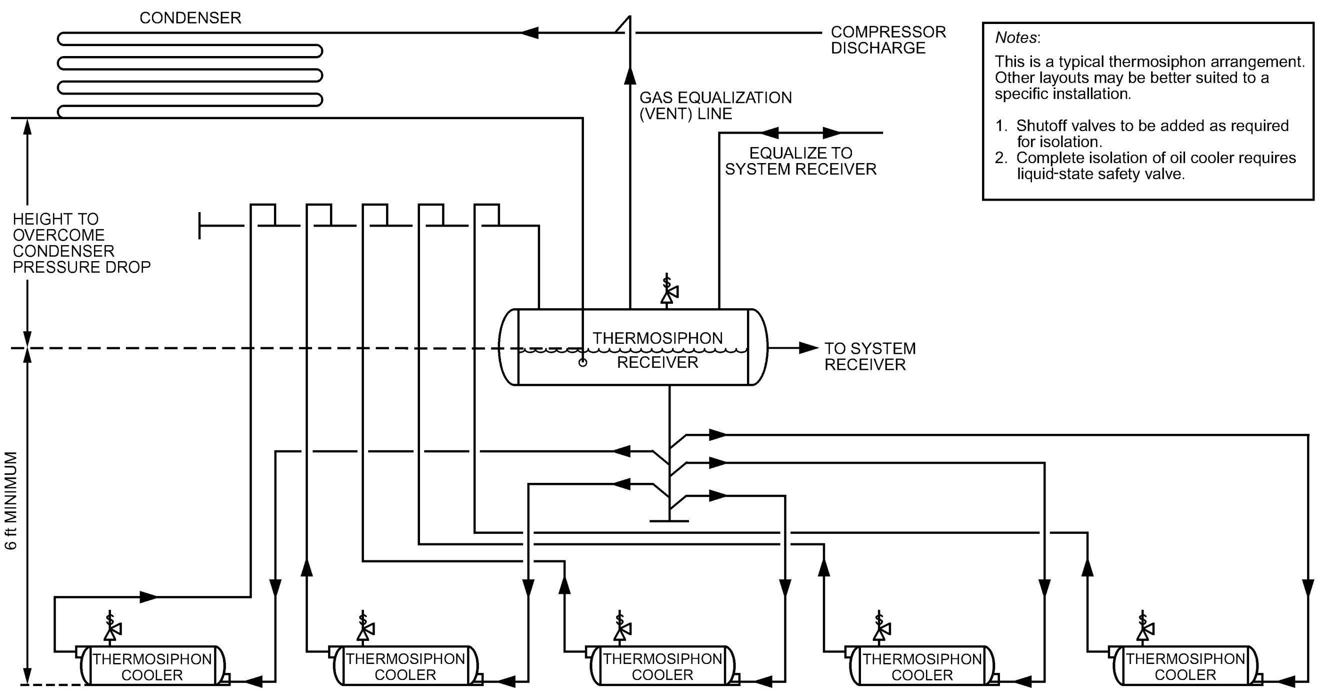


Fig. 24 Typical Thermosiphon System with Multiple Oil Coolers

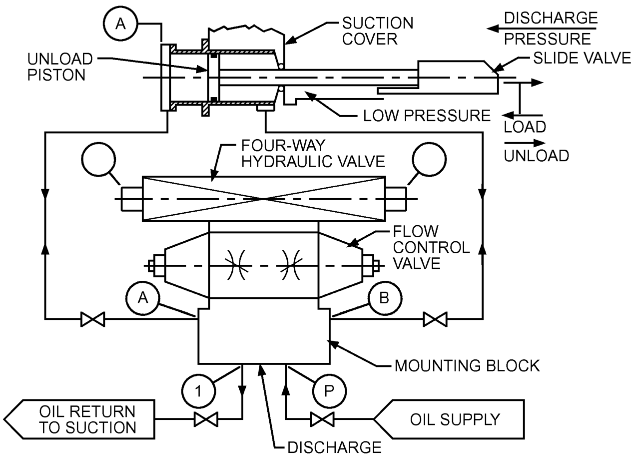


Fig. 25 Typical Hydraulic System for Slide Valve Capacity Control for Screw Compressor with Fixed V_i

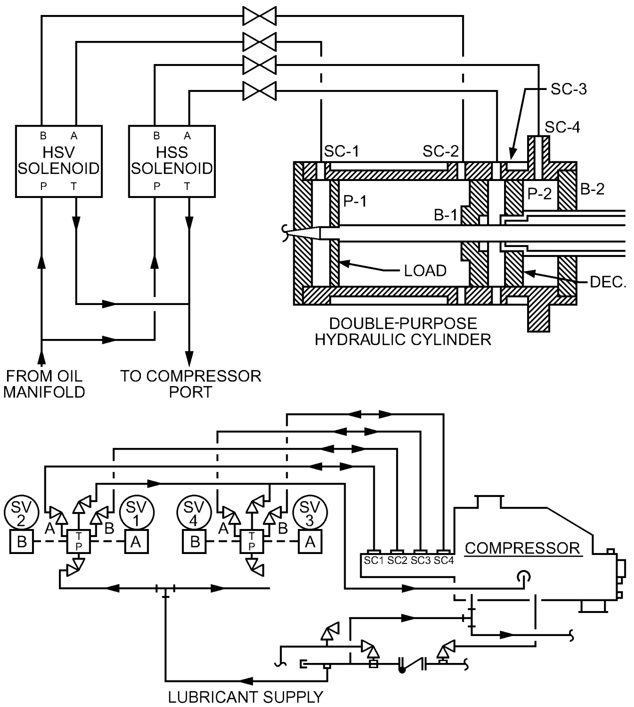


Fig. 26 Typical Positioning System for Slide Valve and Slide Stop for Variable-V_i Screw Compressor

Prevailing winds can also contribute to recirculation. In many areas, the winds shift with the seasons; wind direction during the peak high-humidity season is the most important consideration.

The tops of condensers should always be higher than any adjacent structure to eliminate downdrafts that might induce recirculation. Where this is impractical, discharge hoods can be used to discharge air far enough away from the fan intakes to avoid recirculation. However, the additional static pressure imposed by a discharge hood must be added to the fan system. Fan speed can be increased slightly to obtain proper air volume.

Installation

A single evaporative condenser used with a through-type (top inlet) receiver can be connected as shown in Figure 29. The receiver must always be at a lower pressure than the condensing

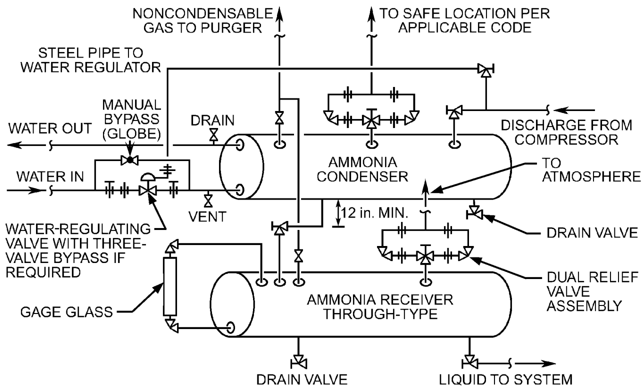


Fig. 27 Horizontal Condenser and Top Inlet Receiver Piping

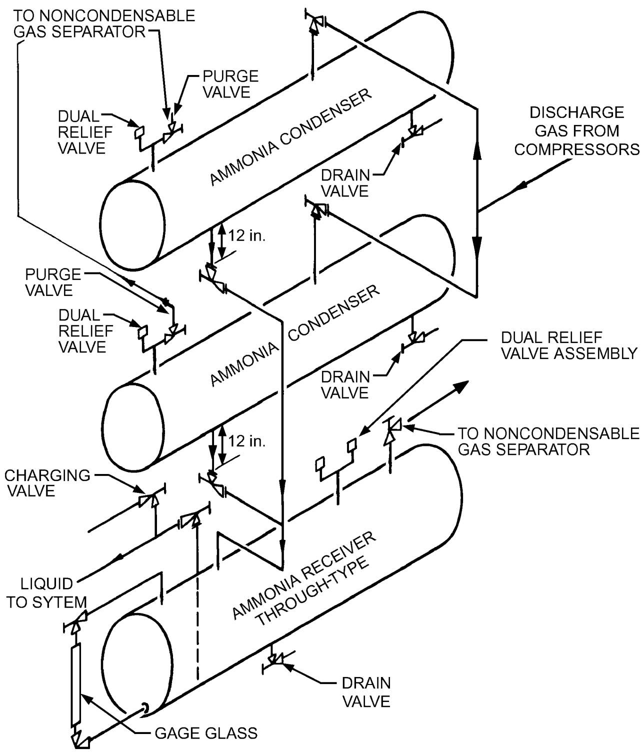


Fig. 28 Parallel Condensers with Top Inlet Receiver

pressure. Design ensures that the receiver is cooler than the condensing temperature.

Installation in Freezing Areas. In areas having ambient temperatures below 32°F, water in the evaporative condenser drain pan and water circuit must be kept from freezing at light plant loads. When the temperature is at freezing, the evaporative condenser can operate as a dry-coil unit, and the water pump(s) and piping can be drained and secured for the season.

Another method of keeping water from freezing is to place the water tank inside and install it as illustrated in Figure 30. When outdoor temperature drops, the condensing pressure drops, and a pressure switch with its sensing element in the discharge pressure line stops the water pump; the water is then drained into the tank. An alternative is to use a thermostat that senses water or outdoor ambient temperature and stops the pump at low temperatures. Exposed piping and any trapped water headers in the evaporative condenser should be drained into the indoor water tank.

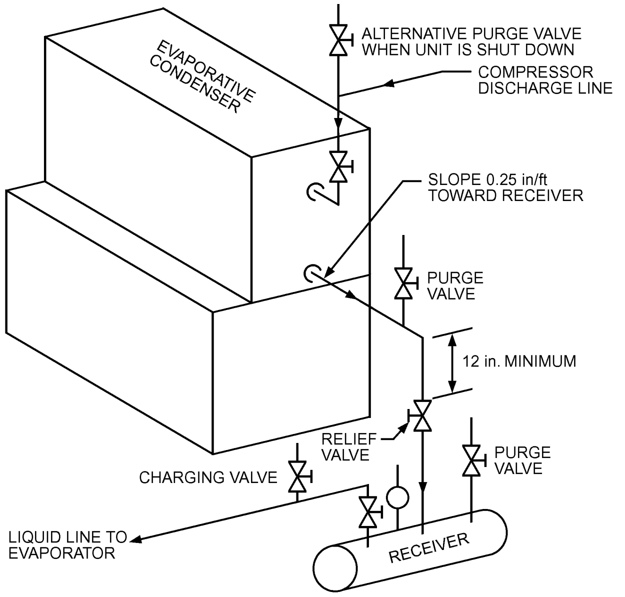


Fig. 29 Single Evaporative Condenser with Top Inlet Receiver

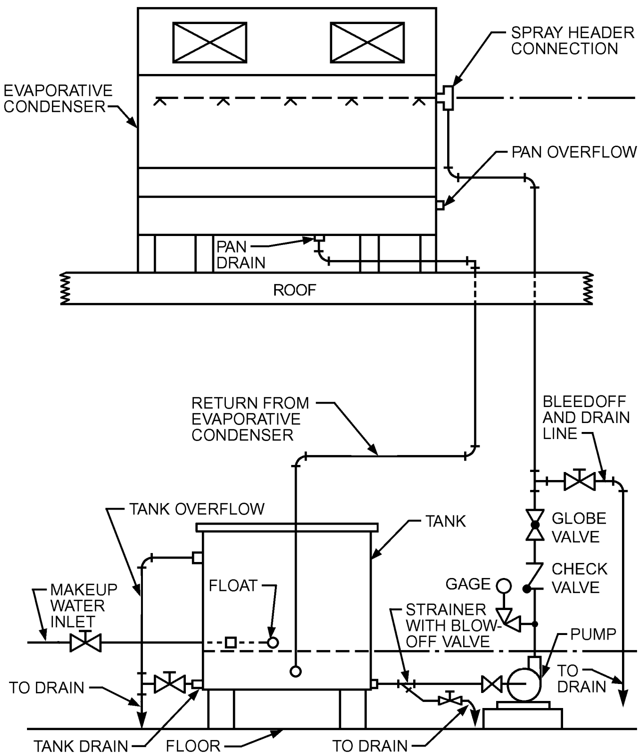


Fig. 30 Evaporative Condenser with Inside Water Tank

Air volume capacity control methods include inlet, outlet, or bypass dampers; two-speed fan motors; or fan cycling in response to pressure controls.

Liquid Traps. Because all evaporative condensers have substantial pressure drop in the ammonia circuit, liquid traps are needed at the outlets when two or more condensers or condenser coils are installed (Figure 31). Also, an equalizer line is necessary to maintain stable pressure in the receiver to ensure free drainage from condensers. For example, assume a 1 psi pressure drop in the operating condenser in Figure 31, which is producing a lower pressure

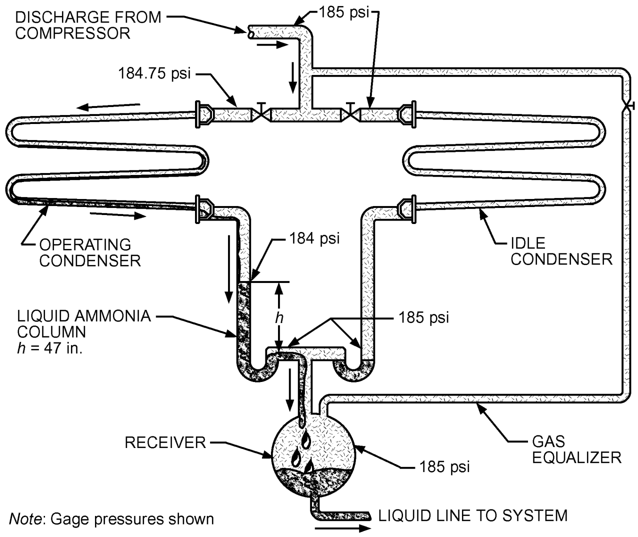


Fig. 31 Two Evaporative Condensers with Trapped Piping to Receiver

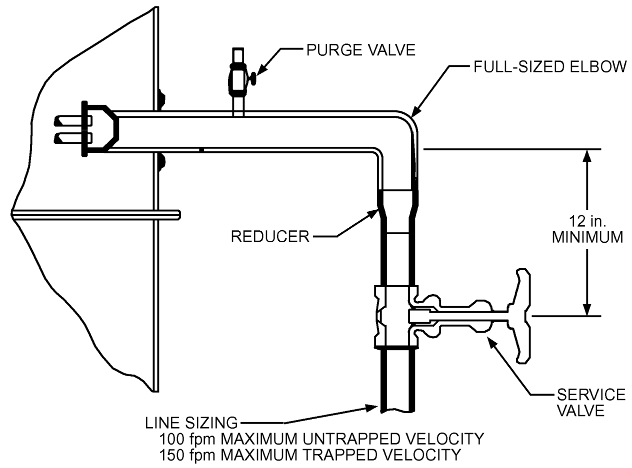


Fig. 32 Method of Reducing Condenser Outlet Sizes

(184 psig) at its outlet compared to the idle condenser (185 psig) and the receiver (185 psig). The trap creates a liquid seal so that a liquid height *h* of 47 in. (equivalent to 1 psi) builds up in the vertical drop leg and not in the condenser coil.

The trap must have enough height above the vertical liquid leg to accommodate a liquid height equal to the maximum pressure drop encountered in the condenser. The example illustrates the extreme case of one unit on and one off; however, the same phenomenon occurs to a lesser degree with two condensers of differing pressure drops when both are in full operation. Substantial differences in pressure drop can also occur between two different brands of the same size condenser or even different models produced by the same manufacturer.

The minimum recommended height of the vertical leg is 5 ft for ammonia. This vertical dimension *h* is shown in all evaporative condenser piping diagrams. This height is satisfactory for operation within reasonable ranges around normal design conditions and is based on the maximum condensing pressure drop of the coil. If service valves are installed at the coil inlets and/or outlets, the pressure drops imposed by these valves must be accounted for by increasing the minimum 5 ft drop-leg height by an amount equal to the valve pressure drop in height of liquid refrigerant (Figure 32).

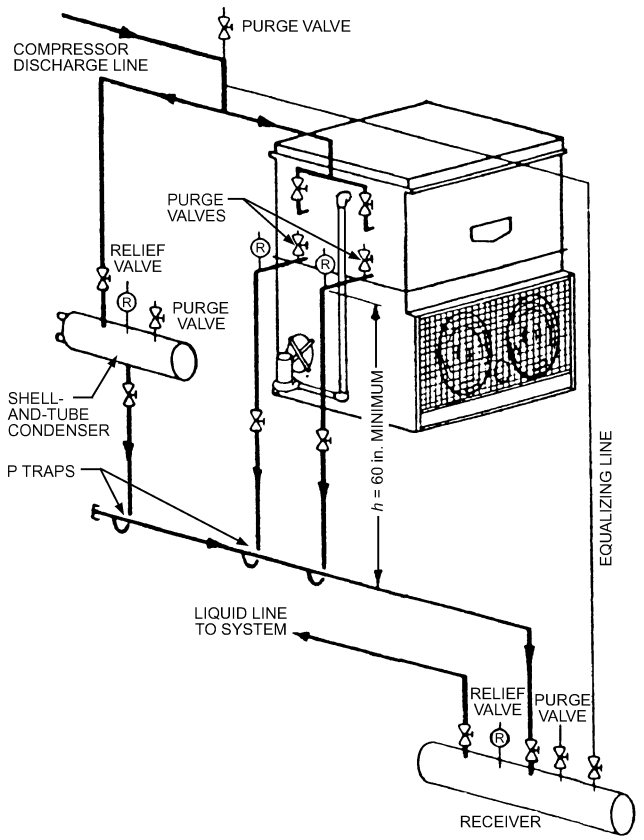


Fig. 33 Piping for Shell-and-Tube and Evaporative Condensers with Top Inlet Receiver

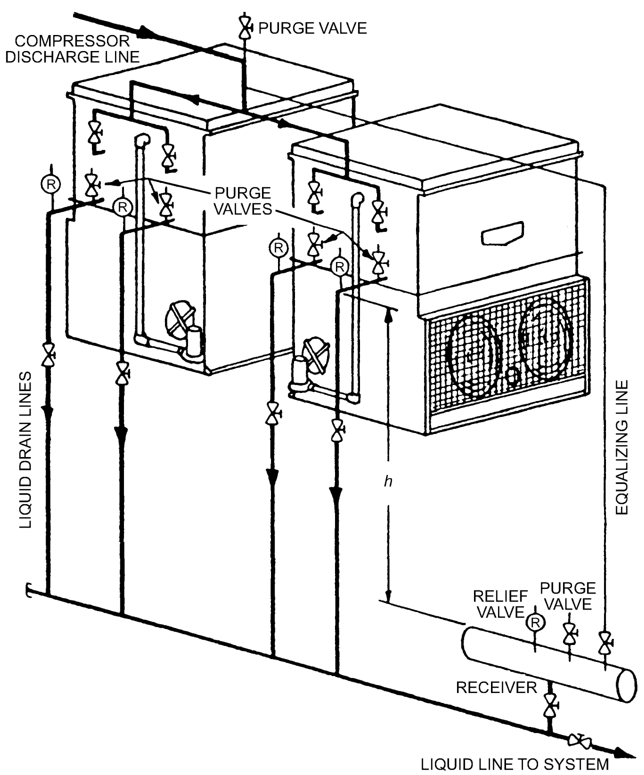


Fig. 34 Piping for Parallel Condensers with Surge-Type Receiver

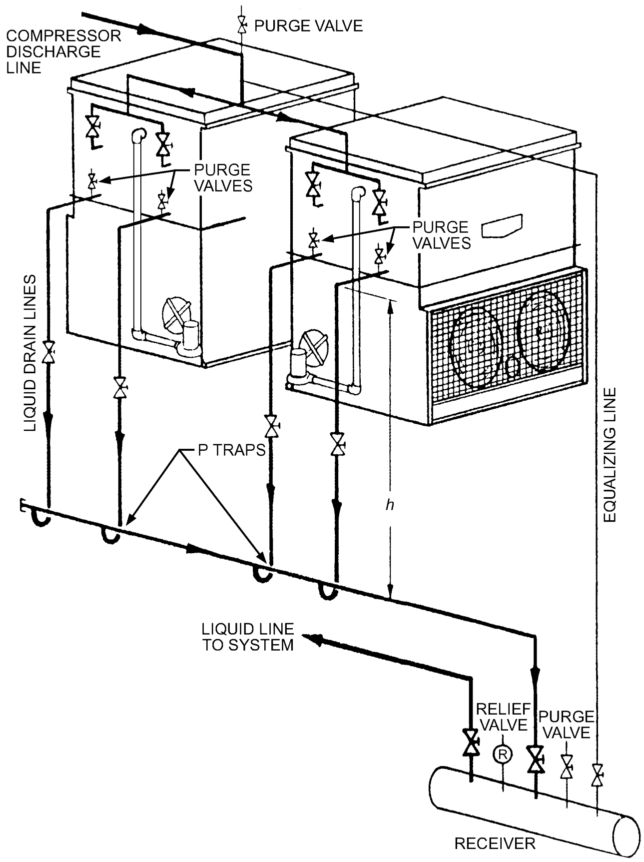


Fig. 35 Piping for Parallel Condensers with Top Inlet Receiver

Figures 33, 34, and 35 illustrate various piping arrangements for evaporative condensers.

EVAPORATOR PIPING

Proper evaporator piping and control are necessary to keep the cooled space at the desired temperature and also to adequately protect the compressor from surges of liquid ammonia out of the evaporator. The evaporators illustrated in this section show some methods used to accomplish these objectives. In some cases, combinations of details on several illustrations have been used.

When using hot gas or electric heat for defrosting, the drain pan and drain line must be heated to prevent the condensate from refreezing. With hot gas, a heating coil is embedded in the drain pan. The hot gas flows first through this coil and then into the evaporator coil. With electric heat, an electric heating coil is used under the drain pan. Wraparound or internal electric heating cables are used on the condensate drain line when the room temperature is below 32°F.

Figure 36 illustrates a thermostatic expansion valve on a unit cooler using hot gas for automatic defrosting. Because this is an automatic defrosting arrangement, hot gas must always be available at the hot-gas solenoid valve near the unit. The system must contain multiple evaporators so the compressor is running when the evaporator to be defrosted is shut down. The hot-gas header must be kept in a space where ammonia does not condense in the pipe. Otherwise, the coil receives liquid ammonia at the start of defrosting and is unable to take full advantage of the latent heat of hot-gas condensation entering the coil. This can also lead to severe hydraulic shock loads. If the header must be in a cold space, the insulated hot-gas main must be drained to the suction line by a high-pressure float.

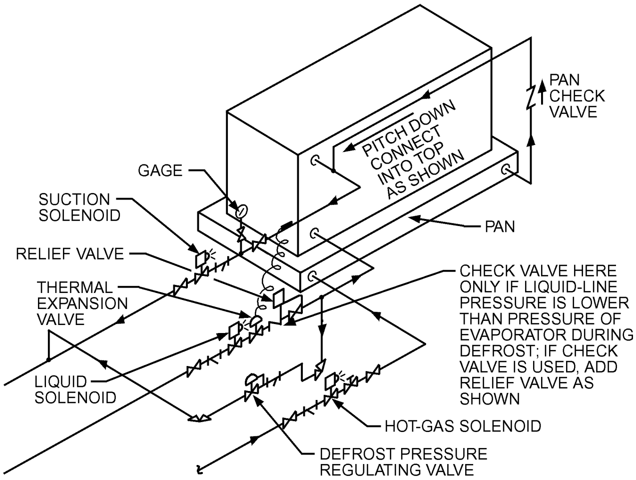


Fig. 36 Piping for Thermostatic Expansion Valve Application for Automatic Defrost on Unit Cooler

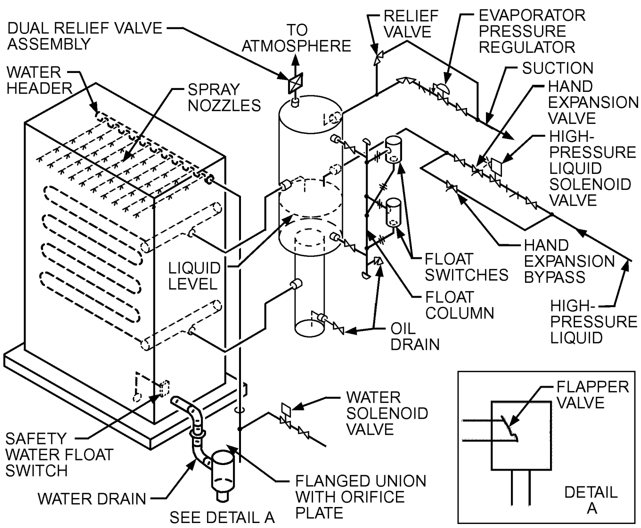


Fig. 37 Arrangement for Automatic Defrost of Air Blower with Flooded Coil

The liquid- and suction-line solenoid valves are open during normal operation only and are closed during the defrost cycle. When defrost starts, the hot-gas solenoid valve is opened. Refer to IIAR *Bulletin* 116 for information on possible hydraulic shock when the hot-gas defrost valve is opened after a defrost.

A defrost pressure regulator maintains a gage pressure of about 70 to 80 psi in the coil.

Unit Cooler—Flooded Operation

Figure 37 shows a flooded evaporator with a close-coupled low-pressure vessel for feeding ammonia into the coil and automatic water defrost.

The lower float switch on the float column at the vessel controls opening and closing of the liquid-line solenoid valve, regulating ammonia feed into the unit to maintain a liquid level. The hand expansion valve downstream of the solenoid valve should be adjusted so that it does not feed ammonia into the vessel more quickly than the vessel can accommodate while raising the suction pressure of gas from the vessel no more than 1 or 2 psi.

The static height of liquid in the vessel should be sufficient to flood the coil with liquid under normal loads. The higher float switch should be wired into an alarm circuit and possibly a compressor shutdown circuit for when the liquid level in the vessel is too high. With flooded coils having horizontal headers, distribution between the multiple circuits is accomplished without distributing orifices.

A combination evaporator pressure regulator and stop valve is used in the suction line from the vessel. During operation, the regulator maintains a nearly constant back pressure in the vessel. A solenoid coil in the regulator mechanism closes it during the defrost cycle. The liquid solenoid valve should also be closed at this time. One of the best means of controlling room temperature is a room thermostat that controls the effective setting of the evaporator pressure regulator.

A spring-loaded relief valve is used around the suction pressure regulator and is set so that the vessel is kept below 125 psig.

A solenoid valve unaffected by downstream pressure is used in the water line to the defrost header. The defrost header is constructed so that it drains at the end of the defrost cycle and the downstream side of the solenoid valve drains through a fixed orifice.

Unless the room is maintained above 32°F, the drain line from the unit should be wrapped with a heater cable or provided with another heat source and then insulated to prevent defrost water from refreezing in the line.

Water line length in the space leading up to the header and the length of the drain line in the cooled space should be kept to a minimum. A flapper or pipe trap on the end of the drain line prevents warm air from flowing up the drain pipe and into the unit.

An air outlet damper may be closed during defrosting to prevent thermal circulation of air through the unit, which affects the temperature of the cooled space. The fan is stopped during defrost.

This type of defrosting requires a drain pan float switch for safety control. If the drain pan fills with water, the switch overrides the time clock to stop flow into the unit by closing the water solenoid valve.

There should be a 5 min delay at the end of the water spray part of the defrosting cycle so water can drain from the coil and pan. This limits ice build-up in the drain pan and on the coils after the cycle is completed.

On completion of the cycle, the low-pressure vessel may be at about 75 psig. When the unit is opened to the much-lower-pressure suction main, some liquid surges out into the main; therefore, it may be necessary to gradually bleed off this pressure before fully opening the suction valve in order to prevent thermal shock. Generally, a suction trap in the engine room removes this liquid before the gas stream enters the compressors.

The type of refrigerant control shown in Figure 37 can be used on brine spray units where brine is sprayed over the coil at all times to pick up the condensed water vapor from the airstream. The brine is reconcentrated continually to remove water absorbed from the airstream.

High-Side Float Control

When a system has only one evaporator, a high-pressure float control can be used to keep the condenser drained and to provide a liquid seal between the high and low sides. Figure 38 illustrates a brine or water cooler with this type of control. The high-side float should be located near the evaporator to avoid insulating the liquid line.

The amount of ammonia in this type of system is critical because the charge must be limited so that liquid will not surge into the suction line under the highest loading in the evaporator. Some type of suction trap should be used. One method is to place a horizontal shell above the cooler, with suction gas piped into the bottom and out the top. The reduction of gas velocity in this shell causes liquid to separate from the gas and draw back into the chiller.

Coolers should include a liquid indicator. A reflex glass lens with a large liquid chamber and vapor connections for boiling liquids and a plastic frost shield to determine the actual level should be used. A refrigeration thermostat measuring chilled-fluid temperature as it exits the cooler should be wired into the compressor starting circuit to prevent freezing.

A flow switch or differential pressure switch should prove flow before the compressor starts. The fluid to be cooled should be piped into the lower portion of the tube bundle and out of the top portion.

Low-Side Float Control

For multiple evaporator systems, low-side float valves are used to control the refrigerant level in flooded evaporators. The low-pressure float in Figure 39 has an equalizer line from the top of the float chamber to the space above the tube bundle and an equalizer line out of the lower side of the float chamber to the lower side of the tube bundle.

For positive shutoff of liquid feed when the system stops, a solenoid valve in the liquid line is wired so that it is only energized when the brine or water pump motor is operating and the compressor is running.

A reflex glass lens with large liquid chamber and vapor connections for boiling liquids should be used with a plastic frost shield to determine the actual level, and with front extensions as required.

Usually a high-level float switch is installed above the operating level of the float to shut the liquid solenoid valve if the float should overfeed.

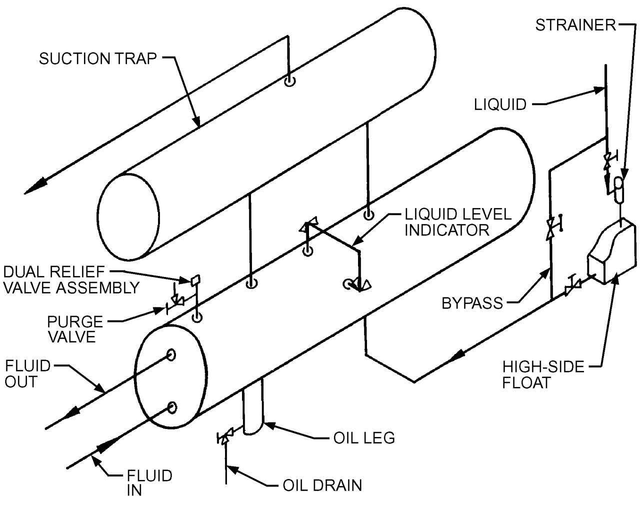


Fig. 38 Arrangement for Horizontal Liquid Cooler and High-Side Float

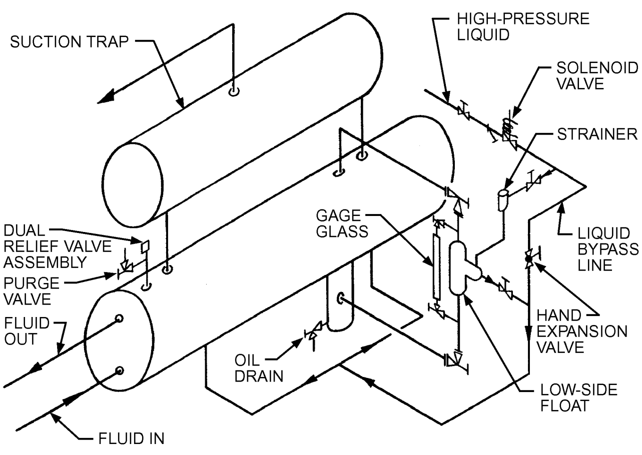


Fig. 39 Piping for Evaporator and Low-Side Float with Horizontal Liquid Cooler

MULTISTAGE SYSTEMS

As pressure ratios increase, single-stage ammonia systems encounter problems such as (1) high discharge temperatures on reciprocating compressors causing lubricant to deteriorate, (2) loss of volumetric efficiency as high pressure leaks back to the low-pressure side through compressor clearances, and (3) excessive stresses on compressor moving parts. Thus, manufacturers usually limit the maximum pressure ratios for multicylinder reciprocating machines to approximately 7 to 9. For screw compressors, which incorporate cooling, compression ratio is not a limitation, but efficiency deteriorates at high ratios.

When the overall system pressure ratio (absolute discharge pressure divided by absolute suction pressure) begins to exceed these limits, the pressure ratio across the compressor must be reduced. This is usually done by using a multistage system. A properly designed two-stage system exposes each of the two compressors to a pressure ratio approximately equal to the square root of the overall pressure ratio. In a three-stage system, each compressor is exposed to a pressure ratio approximately equal to the cube root of the overall ratio. When screw compressors are used, this calculation does not always guarantee the most efficient system.

Another advantage to multistaging is that successively subcooling liquid at each stage of compression increases overall system operating efficiency. Additionally, multistaging can accommodate multiple loads at different suction pressures and temperatures in the same refrigeration system. In some cases, two stages of compression can be contained in a single compressor, such as an internally compounded reciprocating compressor. In these units, one or more cylinders are isolated from the others so they can act as independent stages of compression. Internally compounded compressors are economical for small systems that require low temperature.

Two-Stage Screw Compressor System

A typical two-stage, two-temperature screw compressor system provides refrigeration for high- and low-temperature loads (Figure 40). For example, the high-temperature stage supplies refrigerant to all process areas operating between 28 and 50°F. An 18°F intermediate suction temperature is selected. The low-temperature stage requires a –35°F suction temperature for blast freezers and continuous or spiral freezers.

The system uses a flash intercooler that doubles as a recirculator for the 18°F load. It is the most efficient system available if the screw compressor uses indirect lubricant cooling. If refrigerant injection cooling is used, system efficiency decreases. This system is efficient for several reasons:

1. Approximately 50% of the booster (low-stage) motor heat is removed from the high-stage compressor load by the thermosiphon lubricant cooler.

   Note: In any system, thermosiphon lubricant cooling for booster and high-stage compressors is about 10% more efficient than injection cooling. Also, plants with a piggyback, two-stage screw compressor system without intercooling or injection cooling can be converted to a multistage system with indirect cooling to increase system efficiency approximately 15%.
2. Flash intercoolers are more efficient than shell-and-coil intercoolers by several percent.
3. Thermosiphon lubricant cooling of the high-stage screw compressor provides the highest efficiency available. Installing indirect cooling in plants with liquid injection cooling of screw compressors can increase compressor efficiency by 3 to 4%.
4. Thermosiphon cooling saves 20 to 30% in electric energy during the low-temperature months. When outside air temperature is low, the condensing pressure can be decreased to 90 to 100 psig in most ammonia systems. With liquid injection cooling, the condensing pressure can only be reduced to approximately 125 to 130 psig.

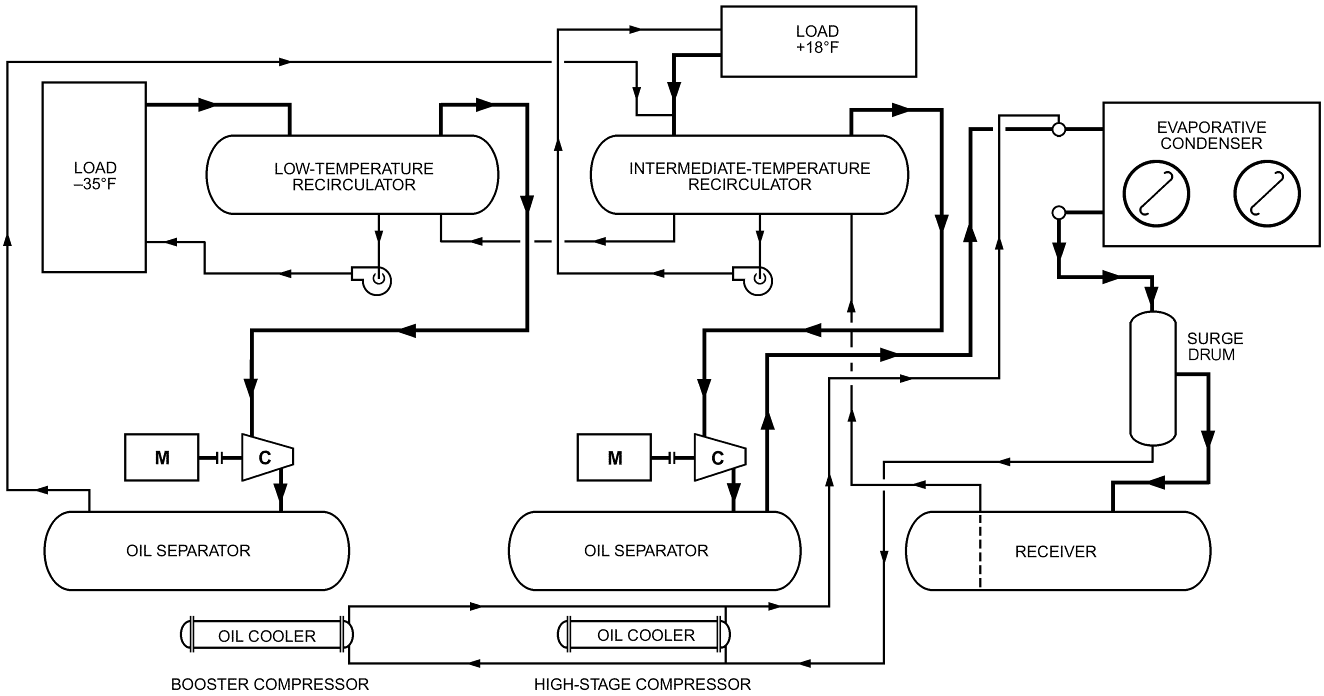


Fig. 40 Compound Ammonia System with Screw Compressor Thermosiphon Cooled

5. Variable-V_i compressors with microprocessor control require less total energy when used as high-stage compressors. The controller tracks compressor operating conditions to take advantage of ambient conditions as well as variations in load.

Converting Single-Stage into Two-Stage Systems

When plant refrigeration capacity must be increased and the system is operating below about 10 psig suction pressure, it is usually more economical to increase capacity by adding a compressor to operate as the low-stage compressor of a two-stage system than to implement a general capacity increase. The existing single-stage compressor then becomes the high-stage compressor of the two-stage system. When converting, consider the following:

- The motor on the existing single-stage compressor may have to be increased in size when used at a higher suction pressure.
- The suction trap should be checked for sizing at the increased gas flow rate.
- An intercooler should be added to cool the low-stage compressor discharge gas and to cool high-pressure liquid.
- A condenser may need to be added to handle the increased condensing load.
- A means of purging air should be added if plant suction gage pressure is below zero.
- A means of automatically reducing compressor capacity should be added so that the system will operate satisfactorily at reduced system capacity points.

LIQUID RECIRCULATION SYSTEMS

The following discussion gives an overview of liquid recirculation (liquid overfeed) systems. See Chapter 1 for more complete information. For additional engineering details on liquid overfeed systems, refer to Stoecker (1988).

In a liquid ammonia recirculation system, a pump circulates ammonia from a low-pressure receiver to the evaporators. The low-pressure receiver is a shell for storing refrigerant at low pressure and is used to supply evaporators with refrigerant, either by gravity or by a low-head pump. It also takes suction from the evaporators and separates gas from the liquid. Because the amount of liquid fed into the evaporator is usually several times the amount that actually evaporates there, liquid is always present in the suction return to the low-pressure receiver. Frequently, three times the evaporated amount is circulated through the evaporator (see Chapter 1).

Generally, the liquid ammonia pump is sized by the flow rate required and a pressure differential of about 25 psi. This is satisfactory for most single-story installations. If there is a static lift on the pump discharge, the differential is increased accordingly.

The low-pressure receiver should be sized by the cross-sectional area required to separate liquid and gas and by the volume between the normal and alarm liquid levels in the low-pressure receiver. This volume should be sufficient to contain the maximum fluctuation in liquid from the various load conditions (see Chapter 1).

Liquid at the pump discharge is in the subcooled region. A total pressure drop of about 5 psi in the piping can be tolerated.

The remaining pressure is expended through the control valve and coil. Pressure drop and heat pickup in the liquid supply line should be low enough to prevent flashing in the liquid supply line.

Provisions for liquid relief from the liquid main back to the low-pressure receiver are required, so when liquid-line solenoid valves at the various evaporators are closed, either for defrosting or for temperature control, the excess liquid can be relieved back to the receiver. Generally, relief valves used for this purpose are set at about 40 psi differential when positive-displacement pumps are used. When centrifugal pumps are used, a hand expansion valve or a minimum flow orifice is acceptable to ensure that the pump is not dead-headed.

The suction header between evaporators and low-pressure receiver should be pitched 1% to allow excess liquid flow back to the low-pressure receiver. The header should be designed to avoid traps.

Liquid Recirculation in Single-Stage System. Figure 41 shows the piping of a typical single-stage system with a low-pressure receiver and liquid ammonia recirculation feed.

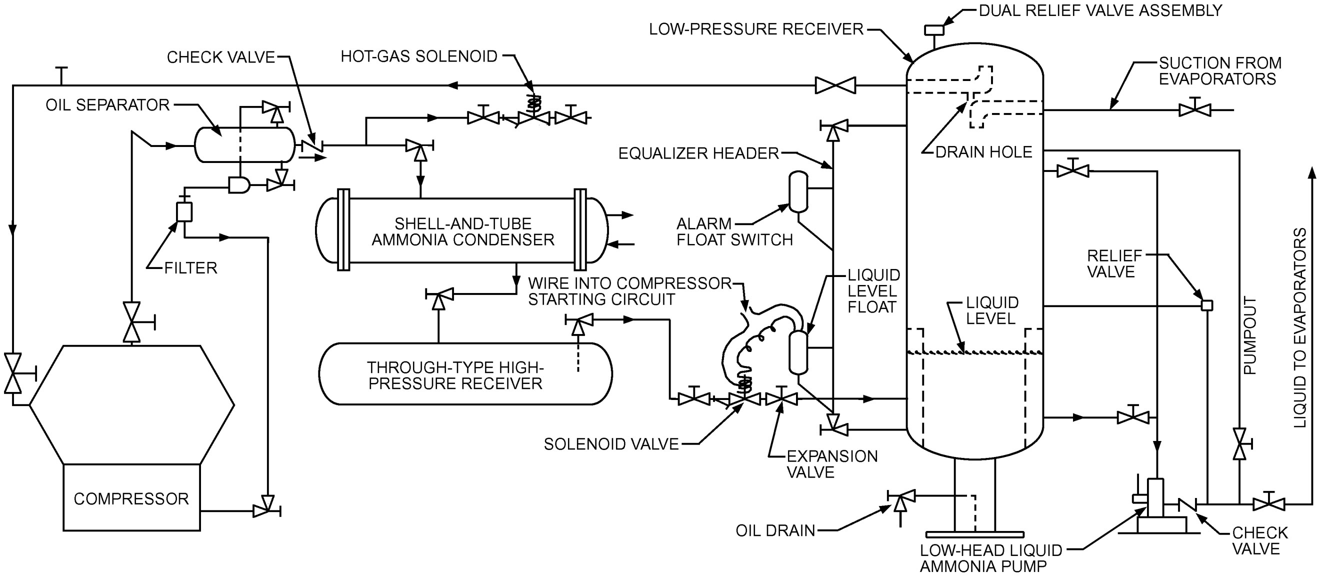


Fig. 41 Piping for Single-Stage System with Low-Pressure Receiver and Liquid Ammonia Recirculation

Hot-Gas Defrost

This section was taken from a technical paper by Briley and Lyons (1992). Several methods are used for defrosting coils in areas below 35°F room temperature:

- Hot refrigerant gas (the predominant method)
- Water
- Air
- Combinations of hot gas, water, and air

The evaporator (air unit) in a liquid recirculation system is circuited so that the refrigerant flow provides maximum cooling efficiency. The evaporator can also work as a condenser if the necessary piping and flow modifications are made. When the evaporator operates as a condenser and the fans are shut down, hot refrigerant vapor raises the surface temperature of the coil enough to melt any ice and/or frost on the surface so that it drains off. Although this method is effective, it can be troublesome and inefficient if the piping system is not properly designed.

Even when fans are not operating, 50% or more of the heat given up by the refrigerant vapor may be lost to the space. Because the heat transfer rate varies with the temperature difference between coil surface and room air, the temperature/pressure of the refrigerant during defrost should be minimized.

Another reason to maintain the lowest possible defrost temperature/pressure, particularly in freezers, is to keep the coil from steaming. Steam increases refrigeration load, and the resulting icicle or frost formation must be dealt with. Icicles increase maintenance during cleanup; ice formed during defrost tends to collect at the fan rings, which sometimes restricts fan operation.

Defrosting takes slightly longer at lower defrost pressures. The shorter the time heat is added to the space, the more efficient the defrost. However, with slightly extended defrost times at lower temperature, overall defrosting efficiency is much greater than at higher temperature/pressure because refrigeration requirements are reduced.

Another loss during defrost can occur when hot, or uncondensed, gas blows through the coil and relief regulator and vents back to the compressor. Some of this gas load cannot be contained and must be vented to the compressor through the wet return line. It is most energy-efficient to vent this hot gas to the highest suction possible; an evaporator defrost relief should be vented to the intermediate or high-stage compressor if the system is two-stage. Figure 42 shows a conventional hot-gas defrost system for evaporator coils of 15 tons of refrigeration and below. Note that the wet return is above the evaporator and that a single riser is used.

Defrost Control. Because defrosting efficiency is low, frequency and duration of defrosting should be kept to the minimum necessary to keep the coils clean. Less defrosting is required during winter than during hotter, more humid periods. An effective energy-saving measure is to reset defrost schedules in the winter.

Several methods are used to initiate the defrost cycle. **Demand defrost**, actuated by a pressure device that measures air pressure drop across the coil, is a good way of minimizing total daily defrost time. The coil is defrosted automatically only when necessary. Demand initiation, together with a float drainer to dump the liquid formed during defrost to an intermediate vessel, is the most efficient defrost system available (Figure 43).

The most common defrost control method, however, is **time-initiated, time-terminated**; it includes adjustable defrost duration and an adjustable number of defrost cycles per 24 h period. This control is commonly provided by a defrost timer.

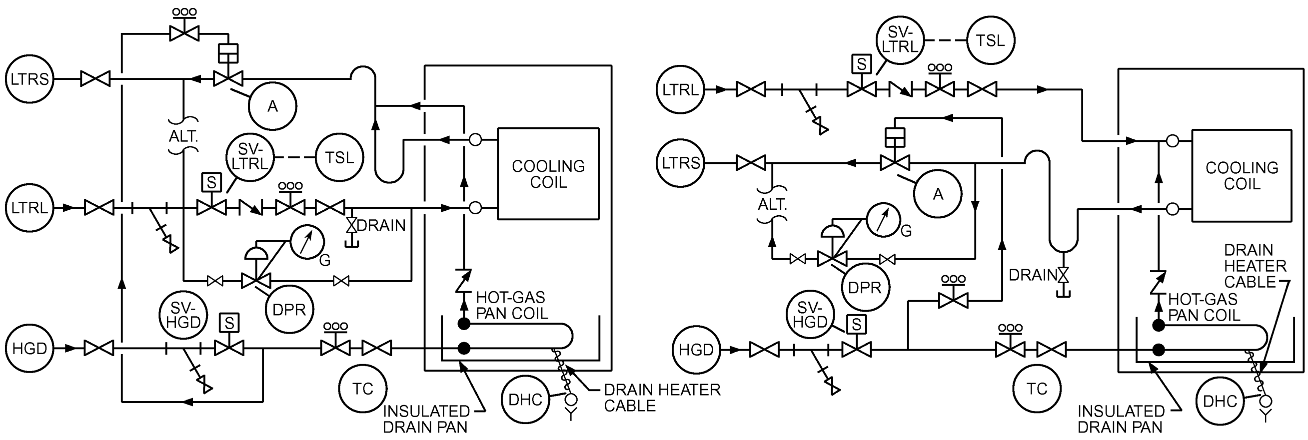
Estimates indicate that the load placed on a refrigeration system by a coil during defrost is up to three times the operating design load. Thus, it is important to properly engineer hot-gas defrost systems.

Designing Hot-Gas Defrost Systems. Several approaches are followed in designing hot-gas defrost systems. Figure 43 shows a typical demand defrost system for both upfeed and downfeed coils. This design returns defrost liquid to the system's intermediate pressure. An alternative is to direct defrost liquid into the wet suction. A float drainer or thermostatic trap with a hot-gas regulator installed at the hot-gas inlet to the coil is much better than the relief regulator (see Figure 43).

Most defrost systems installed today (Figure 42) use a time clock to initiate defrost; the demand defrost system shown in Figure 43 uses a low differential pressure switch to sense the air pressure drop across the coil and actuate the defrost. A thermostat terminates the defrost cycle. A timer is used as a back-up to ensure the defrost terminates.

Sizing and Designing Hot-Gas Piping. Hot gas is supplied to the evaporators in two ways:

1. The preferred method is to install a pressure regulator set at approximately 100 psig in the equipment room at the hot-gas takeoff and size the piping accordingly.



UPFEED COILS **DOWNFEED OR CROSSFEED COILS**

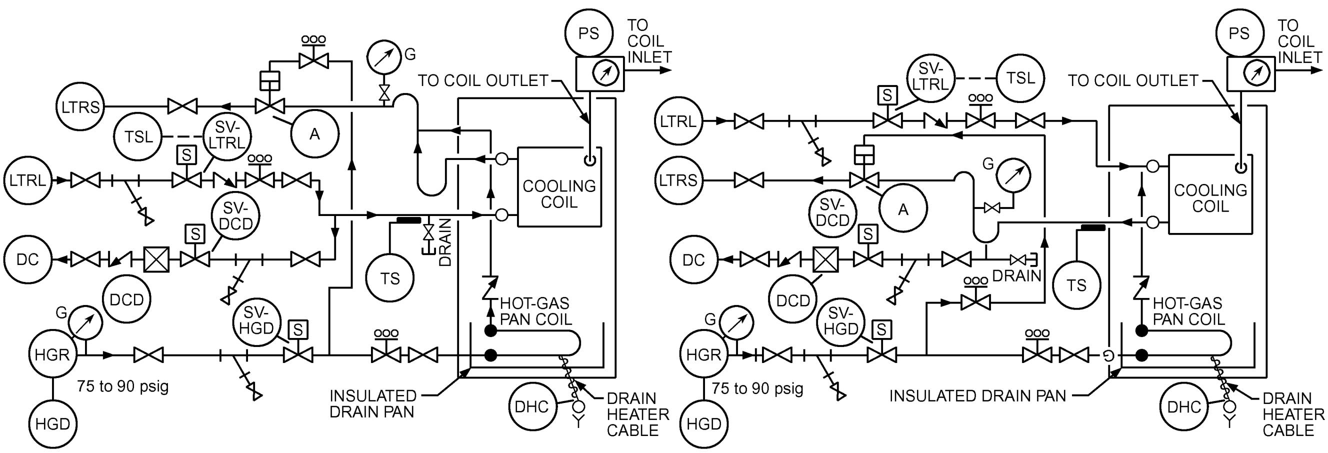
LEGEND		
A	=	Gas-powered automatic shutoff valve
DHC	=	Defrost heater cable
DPR	=	Differential pressure regulator (set at ±90 psig)
G	=	0 to 150 psi, 3 1/2 in. gage with valve
HGD	=	Hot-gas defrost supply (185 to 100 psig)
LTRL	=	Low-temperature recirculated liquid
LTRS	=	Low-temperature recirculated suction
SV-HGD	=	Hot-gas defrost solenoid valve
SV-LTRL	=	Solenoid valve, low-temperature recirculated liquid
TC	=	Time clock
TL	=	Thermostat: operates SV-LTRL on demand
TSL	=	Room thermostat: operates SV-LTRLF on demand

DEFROST CYCLE ACTUATED BY TIME CLOCK

SEQUENCE OF OPERATION

A. Deenergize SV-LTRL and stop fans.
   Delay 2 to 3 min for downfeed and crossfeed coils and 4 to 8 min for upfeed coils.
   Energize SV-HGD, which closes valve A.
B. Coil defrost period terminated by Time Clock. (Keep defrost time to a minimum: just enough to clean and drain pan.)
   TC deenergizes SV-HGD, which opens valve A.
C. Energize SV-LTRL.
   Delay 2 min to freeze any remaining water on coil.
D. Start fans.

Fig. 42 Conventional Hot-Gas Defrost Cycle
(For coils with 15 tons refrigeration capacity and below)



UPFEED COILS

DEFROST CYCLE ACTUATED BY PRESSURE SWITCH AT PREDETERMINED SETTING
(approximately 1 in. water gage)

DOWNFEED OR CROSSFEED COILS

LEGEND		
A	=	Gas-powered automatic shutoff valve
DC	=	Return liquid defrost condensate
DCD	=	Defrost condensate return drainer
DHC	=	Defrost heater cable
G	=	0 to 150 psi, 3 1/2 in. gage with valve
HGD	=	Hot-gas defrost supply
HGR	=	Hot-gas regulator valve
LTRL	=	Low-temperature recirculated liquid
LTRS	=	Low-temperature recirculated suction
PS	=	Air-side pressure switch
SV-DCD	=	Solenoid valve, defrost condensate return drainer
SV-HGD	=	Hot-gas defrost solenoid valve
SV-LTRL	=	Recirculated liquid-line solenoid valve
TS	=	Thermostat: terminates hot-gas defrost cycle
TSL	=	Room thermostat: operates SV-LTRL on demand

SEQUENCE OF OPERATION

A. Deenergize SV-LTRL and stop fans. Delay energize SV-HGD closing A. Coil will defrost.
B. Coil defrost period terminated by TS set at 40°F. Use override timer after 30 min to force SV-HGD to deenergize if TS malfunctions.
C. Deenergize SV-HGD via thermostat TS, opening valve A.
D. Energize SV-LTRL. Delay 2 to 4 min.
E. Start fans.
F. To save energy, return liquid to intercooler of intermediate-temperature recirculator.
   Alternative: Return liquid to wet suction downstream of stop valve.

Note: This defrost control method assumes a hot-gas outlet pressure regulator in equipment room. If high-pressure hot gas is used, an outlet pressure regulator with electric shutoff should be substituted for SV-HGD and set at 75 to 90 psig.

Fig. 43 Demand Defrost Cycle
(For coils with 15 tons refrigeration capacity and below)

2. The alternative is to install a pressure regulator at each evaporator or group of evaporators and size the piping for minimum design condensing pressure, which should be 75 to 85 psig.

A maximum of one-third of the coils in a system should be defrosted at one time. If a system has 300 tons of refrigeration capacity, the main hot-gas supply pipe could be sized for 100 tons of refrigeration. The outlet pressure-regulating valve should be sized in accordance with the manufacturer's data.

Reducing defrost hot-gas pressure in the equipment room has advantages, notably that less liquid condenses in the hot-gas line as the condensing temperature drops to 52 to 64°F. A typical equipment room hot-gas pressure control system is shown in Figure 44. If hot-gas lines in the system are trapped, a condensate drainer must be installed at each trap and at the low point in the hot-gas line (Figure 45). Defrost condensate liquid return piping from coils where a float or thermostatic valve is used should be one size larger than the liquid feed piping to the coil.

Demand Defrost. The following are advantages and features of demand defrost:

- It uses the least energy for defrost.
- It increases total system efficiency because coils are off-line for a minimum amount of time.
- It imposes less stress on the piping system because there are fewer defrost cycles.
- Regulating hot gas to approximately 100 psig in the equipment room gives the gas less chance of condensing in supply piping. Liquid in hot-gas systems may cause problems because of the hydraulic shock created when the liquid is forced into an evaporator (coil). Coils in hot-gas pans may rupture as a result.
- Draining liquid formed during defrost with a float or thermostatic drainer eliminates hot-gas blow-by normally associated with pressure-regulating valves installed around the wet suction return line pilot check valve.
- Returning liquid ammonia to the intercooler or high-stage recirculator saves considerable energy. A 20 ton refrigeration coil defrosting for 12 min can condense up to 24 lb/min of ammonia, or 288 lb total. The enthalpy difference between returning to the low-stage recirculator (–40°F) and the intermediate recirculator (+20°F) is 64 Btu/lb, for 18,432 Btu total or 7.68 tons of refrigeration removed from the –40°F booster for 12 min. This assumes that only liquid is drained and is the saving when liquid is drained to the intermediate point, not the total cost to defrost. If a pressure-reducing valve is used around the pilot check valve, this rate could double or triple because hot gas flows through these valves in unmeasurable quantities.

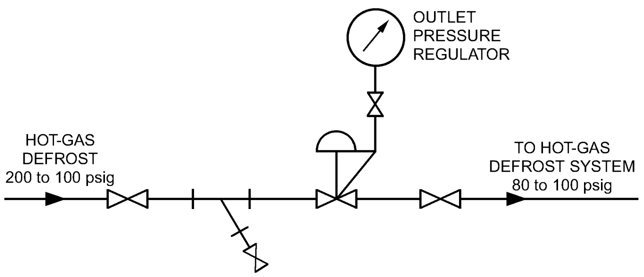


Fig. 44 Equipment Room Hot-Gas Pressure Control System

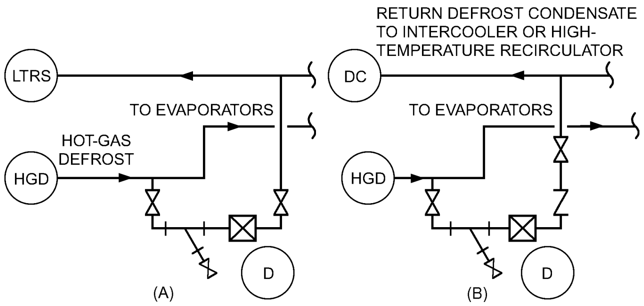


Fig. 45 Hot-Gas Condensate Return Drainer

Soft Hot-Gas Defrost System. This system is particularly well suited to large evaporators and should be used on all coils of 15 tons of refrigeration or over. It eliminates the valve clatter, pipe movements, and noise associated with large coils during hot-gas defrost. Soft hot-gas defrost can be used for upfeed or downfeed coils; however, the piping systems differ (Figure 46). Coils operated in the horizontal plane must be orificed. Vertical coils that usually are crossfed are also orificed.

Soft hot-gas defrost is designed to increase coil pressure gradually as defrost begins. This is accomplished by a small hot-gas feed having a capacity of about 25 to 30% of the estimated duty with a solenoid and a hand expansion valve adjusted to bring the pressure up to about 40 psig in 3 to 5 min. (See Sequence of Operation in Figure 46.) After defrost, a small suction-line solenoid is opened so that the coil can be brought down to operation pressure gradually before liquid is introduced and the fans started. The system can be initiated by a pressure switch; however, for large coils in spiral or individual quick freezing systems, manual initiation is preferred.

This system eliminates check valve chatter and most, if not all, liquid hammer (i.e., hydraulic problems in the piping). In addition, the last three features listed in the section on Demand Defrost apply to soft hot-gas defrost.

Double Riser Designs for Large Evaporator Coils

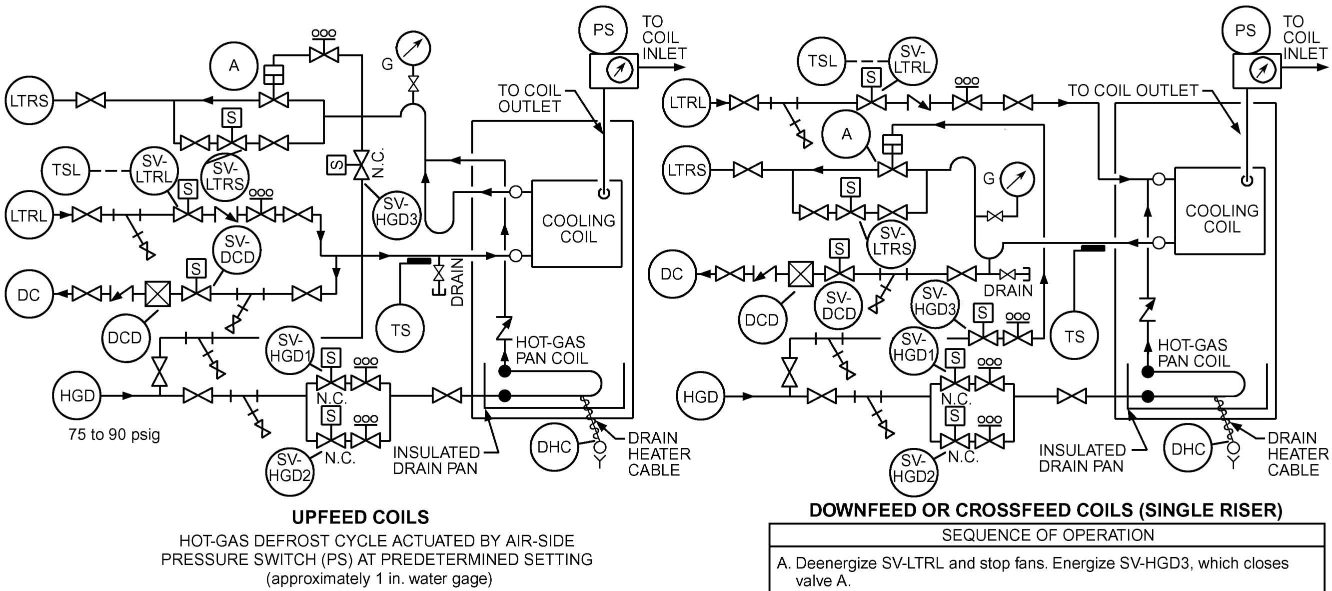
Static pressure penalty is the pressure/temperature loss associated with a refrigerant vapor stream bubbling through a liquid bath. If speed in the riser is high enough, it will carry over a certain amount of liquid, thus reducing the penalty. For example, at –40°F ammonia has a density of 43.07 lb/ft^3, which is equivalent to a pressure of 43.07/144 = 0.30 psi per foot of depth. Thus, a 16 ft riser has a column of liquid that exerts 16 × 0.30 = 4.8 psi. At –40°F, ammonia has a saturation pressure of 10.4 psia. At the bottom of the riser then, the pressure is 4.8 + 10.4 = 15.2 psia, which is the saturation pressure of ammonia at –27°F. This 13°F difference amounts to a 0.81°F penalty per foot of riser. If a riser were oversized to the point that the vapor did not carry liquid to the wet return, the evaporator would be at –27°F instead of –40°F. This problem can be solved in several ways:

- Install the low-temperature recirculated suction (LTRS) line below the evaporator. This method is very effective for downfeed evaporators. Suction from the coil should not be trapped. This arrangement also ensures lubricant return to the recirculator.
- Where the LTRS is above the evaporator, install a liquid return system below the evaporator (Figure 47). This arrangement eliminates static penalty, which is particularly advantageous for plate, individual quick freeze, and spiral freezers.
- Use double risers from the evaporator to the LTRS (Figure 48).

If a single riser is sized for minimum pressure drop at full load, the static pressure penalty is excessive at part load, and lubricant return could be a problem. If the single riser is sized for minimum load, then riser pressure drop is excessive and counterproductive.

Double risers solve these problems (Miller 1979). Figure 48 shows that, when maximum load occurs, both risers return vapor and liquid to the wet suction. At minimum load, the large riser is sealed by liquid ammonia in the large trap, and refrigerant vapor flows through the small riser. A small trap on the small riser ensures that some lubricant and liquid return to the wet suction.

Risers should be sized so that pressure drop, calculated on a dry-gas basis, is at least 0.3 psi per 100 ft. The larger riser is designed for



LEGEND

A	= Gas-powered automatic shutoff valve LTRS	SV-HGD1	= Hot-gas solenoid valve (main hot-gas defrost supply)
DC	= Defrost condensate return liquid	SV-HGD2	= Hot-gas solenoid valve (25% hot-gas defrost supply)
DCD	= Defrost condensate drainer	SV-HGD3	= Hot-gas solenoid valve (to operate valve A)
DHC	= Defrost heater cable	SV-LTRL	= Liquid-line solenoid valve
G	= 0 to 150 psi, 3 1/2 in. gage with valve	SV-LTRS	= Solenoid valve bypass to LTRS
HGD	= Hot-gas defrost supply 75 to 90 psig	TS	= Thermostat: terminates hot-gas defrost cycle
LTRL	= Low-temperature recirculated liquid	TSL	= Thermostat: operates SV-LTRL on demand
LTRS	= Low-temperature recirculated suction		
PS	= Air-side pressure switch		
PS2	= Pressure switch to operate SV-HGD1		
SV-DCD	= Solenoid valve, defrost condensate return drainer		

SEQUENCE OF OPERATION

A. Deenergize SV-LTRL and stop fans. Energize SV-HGD3, which closes valve A.
B. After 60 s, energize SV-HGD2 to gradually increase coil internal pressure.
C. When coil pressure reaches about 40 psig, pressure switch PS2 energizes SV-HGD1 and defrost continues.
D. When liquid reaches about 40°F, temperature switch (TS) closes SV-HGD1 and SV-HGD2.
E. Wait 1 min and open small solenoid valve SV-LTRS. Then wait 2 to 4 min.
F. Deenergize SV-HGD3, which opens valve A and closes SV-LTRS.
G. Energize SV-LTRL and wait 2 to 4 min.
H. Start fans.
I. To save energy, return defrost condensate (DC) to intercooler or intermediate-temperature recirculator.
Alternative: Return liquid to LTRS downstream of stop valve.

Note 1: This defrost method assumes an outlet pressure regulator in equipment room. If high-pressure hot gas is used, an outlet pressure regulator is installed at the HGD inlet.
Note 2: Soft hot-gas defrost also may be manually initiated.

Fig. 46 Soft Hot-Gas Defrost Cycle
(For coils with 15 tons refrigeration capacity or above)

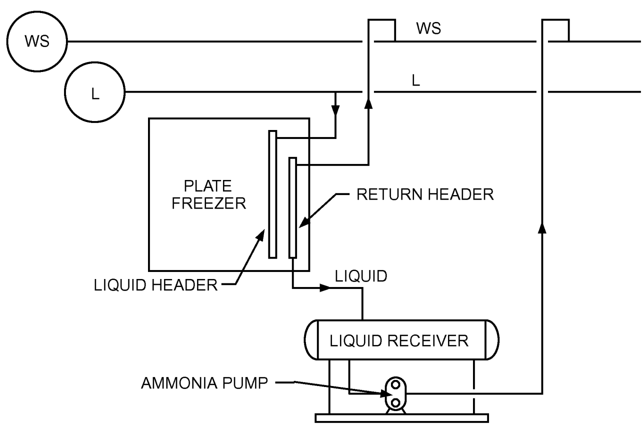


Fig. 47 Recirculated Liquid Return System

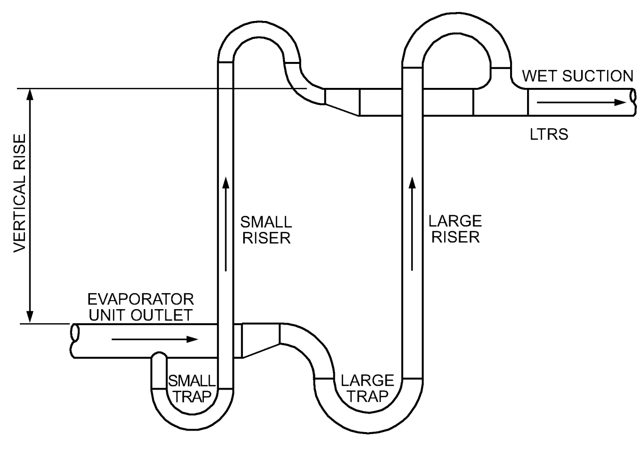


Fig. 48 Double Low-Temperature Suction Risers

approximately 65 to 75% of the flow and the small one for the remainder. This design results in a velocity of approximately 5000 fpm or higher. Some coils may require three risers (large, medium, and small).

Over the years, freezer capacity has grown. As they became larger, so did the evaporators (coils). Where these freezers are in line and the product to be frozen is wet, the defrost cycle can be every 4 or 8 h. Many production lines limit defrost duration to 30 min. If coils are large (some coils have a refrigeration capacity of 200 to 300 tons), it is difficult to design a hot-gas defrost system that can complete a safe defrost in 30 min. Sequential defrost systems, where coils are defrosted alternately during production, are feasible but require special treatment.

SAFETY CONSIDERATIONS

Ammonia is an economical choice for industrial systems. Although ammonia has superior thermodynamic properties, it is considered toxic at low concentration levels of 35 to 50 ppm. Large quantities of ammonia should not be vented to enclosed areas near open flames or heavy sparks. Ammonia at 16 to 25% by volume burns and can explode in air in the presence of an open flame.

The importance of ammonia piping is sometimes minimized when the main emphasis is on selecting major equipment pieces. Mains should be sized carefully to provide low pressure drop and avoid capacity or power penalties caused by inadequate piping.

Rusting pipes and vessels in older systems containing ammonia can create a safety hazard. Oblique x-ray photographs of welded pipe joints and ultrasonic inspection of vessels may be used to disclose defects. Only vendor-certified parts for pipe, valving, and pressure-containing components according to designated assembly drawings should be used to reduce hazards. Cold liquid refrigerant should not be confined between closed valves in a pipe where the liquid can warm and expand to burst piping components. Rapid multiple pulsations of ammonia liquid in piping components (e.g., those developed by cavitation forces or hydraulic hammering from compressor pulsations with massive slugs of liquid carryover to the compressor) must be avoided to prevent equipment and piping damage and injury to personnel.

Hydraulic shock, also known as **hydraulic hammering** (or, in steam and water systems, **water hammer**) can be particularly hazardous. Symptoms in an operating system are loud clattering or banging or pipes moving suddenly and erratically.

The hazard arises out of great, potentially destructive forces that are suddenly generated inside the piping by hydraulic shocks (Lloyko 1992; Shelton and Jacobi 1997a, 1997b) The fundamental flow-phenomenon descriptions of the transients that can cause these shocks are **vapor-propelled**, **liquid slugs**, and **condensation-induced shock**. The phenomenon typically arises when high-pressure warm vapor and cold liquid mix in the same pipe or other refrigerating system element. This most often can occur within, and in the piping around, air-cooling units before, during, and after hot-gas defrosting and in parts of gas-pressure liquid circulating systems, but other parts of the system can also be vulnerable (Glennon and Cole 1998).

Conditions that are most conducive to development of hydraulic transients are pressures lower than approximately 20 psia, with saturated or subcooled liquid present, which are then exposed to high-pressure vapors at pressures equal to or greater than those commonly used for hot-gas defrosting (85 psia). Standing liquid in a horizontal pipe can be excited sufficiently to cause hydraulic shocks when exposed to high-pressure gas flowing over the top of it. In situations where high-pressure gas is introduced regularly, it is necessary that steps be taken as part of the operating protocol to rid the pipe or other system element of liquid before introducing hot gas.

To that end, all valves in horizontal lines should be installed with their stems horizontal, and air-cooling units should be subjected to a complete pumpdown before introducing hot gas for defrosting. Failures occur most often at the pipe or header end closure. These pipe sections, where unavoidable, should be kept very short, less than 24 pipe diameters in length.

Most service problems are caused by inadequate precautions during design, construction, and installation (ASHRAE *Standard* 15; IIAR *Standard* 2). Ammonia is a powerful solvent that removes dirt, scale, sand, or moisture remaining in the pipes, valves, and fittings during installation. These substances are swept along with the suction gas to the compressor, where they are a menace to the bearings, pistons, cylinder walls, valves, and lubricant. Most compressors are equipped with suction strainers and/or additional disposable strainer liners for the large quantity of debris that can be present at initial start-up.

Moving parts are often scored when a compressor is run for the first time. Damage starts with minor scratches, which increase progressively until they seriously affect compressor operation or render it inoperative.

A system that has been carefully and properly installed with no foreign matter or liquid entering the compressor will operate satisfactorily for a long time. As piping is installed, it should be power rotary wire brushed and blown out with compressed air. The piping system should be blown out again with compressed air or nitrogen before evacuation and charging. See ASHRAE *Standard* 15 for system piping test pressure.

CARBON DIOXIDE

Because of its good environmental properties and relative safety, there is renewed and widespread interest in carbon dioxide (CO_2, or R-744) as a refrigerant. Because of its low critical-point temperature (87.8°F) and high pressure, CO_2 presents some unusual technological requirements compared to conventional refrigerants. Another constraint in applying CO_2 is its relatively high triple point at –69.8°F and coincident pressure of 75.1 psia.

Carbon dioxide was used in the early stages of the refrigeration industry, but it lost the competition with halocarbon refrigerants because of its high operating pressure and the loss of capacity and coefficient of performance (COP) when rejecting heat near or above the critical point. In an application with comparable components and heat rejection to ambient air, a CO_2 system's COP is about 40% lower than that of a system using conventional refrigerants. New heat exchanger technology and system components allow CO_2 to reach competitive efficiency levels. The CO_2 efficiency deficit becomes less of a problem in a system when heat is rejected far below the critical point (e.g., in a low-temperature stage of a commercial refrigeration cascade cycle). In this application, the advantageous transport properties of CO_2 and the CO_2-geared design may outweigh CO_2's slight thermodynamic disadvantage.

Table 4 presents selected thermophysical properties of refrigerants used in refrigeration. The high operating pressures (e.g., 490.8 psia at a saturation temperature of 30°F, or 969.6 psia at 80°F) present some unique challenges for containment and safety. However, CO_2 has a number of attractive thermophysical properties and other characteristics. Compared to counterpart halocarbon refrigerants, it has low viscosity, high thermal conductivity, and high vapor density. It is nontoxic, nonflammable, readily available, and of low cost; it has no ozone depletion potential (ODP), and negligible direct global warming potential (GWP). Drawbacks include high operating pressures for medium- and high-temperature refrigeration, and the fact that, during a catastrophic release from the system, CO_2 can adversely affect humans at lower concentrations than HFC refrigerants.

Recently, CO_2 has been intensely studied for application as the primary refrigerant in transcritical mobile air conditioners and vending machines. CO_2 heat pump water heaters are already commercially available in a few countries. In this application, transcritical operation (i.e., rejection of heat above the critical point) is beneficial because it allows good temperature glide matching between the water and supercritical CO_2, which provides COP benefit. In large industrial systems, CO_2 is used as the low-temperature-stage refrigerant in cascade-system arrangements, typically with ammonia as the high-temperature-stage refrigerant. In medium-sized commercial systems, CO_2 is also used as the low-temperature-stage refrigerant in cascade-system arrangements with HFCs and, in

Table 4 Thermophysical Properties for Selected Refrigerants

	CO_2	NH_3	R-134a	R-22	R-404A	R-507A
T_{crit}, °F	87.8	270.1	213.9	205.1	161.6	159.1
P_{crit}, psia	1070.0	1643.7	588.8	723.7	540.8	537.4
P_{sat}, psia	490.8	59.7	40.8	69.7	85.4	87.4
h_{fg},[a] Btu/lb$_m$	100.8	544.7	85.8	88.6	71.9	70.2
ρ_f,[a] lb$_m$/ft^3	58.3	40.0	81.1	80.2	72.1	72.5
ρ_g,[b] lb$_m$/ft^3	5.88	0.21	0.87	1.28	1.84	1.94
c_p,[a] Btu/lb$_m$·°F	0.60	1.10	0.32	0.28	0.33	0.33
k,[a] Btu/h·ft·°F	0.065	0.325	0.053	0.055	0.043	0.043
μ,[a] lb$_m$/ft·s	6.8×10^{-5}	1.2×10^{-4}	8.0×10^{-4}	1.5×10^{-4}	1.2×10^{-4}	1.2×10^{-4}

Source: Lemmon et al. (2002) [a]For saturated liquid at 30°F [b]For saturated vapor at 30°F a few cases, ammonia or hydrocarbons as the high-temperature-stage refrigerant.

In addition, carbon dioxide is used in low-temperature systems as a two-phase working fluid in a typical secondary coolant loop. The primary refrigerant is ammonia, R-22, or R-404A in a two-stage configuration, or R-507 in a single-stage configuration. In one arrangement, CO_2 is circulated through the process cooler as a secondary fluid, without changing phase, thereby experiencing a temperature rise in the process cooler. The process cooler must be sized and circuited to accommodate the nonisothermal flow of the coolant. The necessity of a larger process cooler and the high pressure rating of the secondary loop is offset by CO_2's superior thermophysical properties and low pumping cost. The CO_2 is cooled in a chiller and pumped to the process cooler, just as in a conventional secondary coolant loop.

In another arrangement, the secondary loop is configured as a low-temperature liquid recirculation system with liquid CO_2 pumped to the process cooler. In this case, the CO_2 evaporates in the process, typically in an overfeed arrangement, with theoretically isothermal flow in the process cooler. This may result in a smaller process cooler than in the non-phase-change flow arrangement. The liquid/vapor mixture is returned to the pump receiver/liquid separator, and the vapor is condensed in a chiller/condenser. In this case, the primary refrigerating plant is as described previously and the CO_2 system circulates in a phase-change manner, but operates at a single pressure. Power to move the CO_2 is imparted by the pump and by the vapor pressure difference that results from the temperature difference between the CO_2 in the receiver and the refrigerant in the primary system chiller/CO_2 condenser.

This arrangement is also used in a conventional cascade system arrangement, where the low-temperature-stage compressor operates on CO_2 rather than on the primary refrigerant. Whether this arrangement or the low-stage compressor with the primary refrigerant is used typically depends on the size and nature of the low-temperature load. The size and operating cost of the low-stage compressor dictate whether it should compress CO_2 or the primary refrigerant.

REFERENCES

ASHRAE. 2004. Safety standard for refrigeration systems. ANSI/ASHRAE *Standard* 15-2004.

ASME. 2004. Rules for construction of pressure vessels. *Boiler and pressure vessel code*, Section VIII, Division 1. American Society of Mechanical Engineers, New York.

ASME. 2001. Refrigeration piping and heat transfer components. ANSI/ASME *Standard* B31.5-2001. American Society of Mechanical Engineers, New York.

ASTM. 2005. Specification for pipe, steel, black and hot-dipped, zinc-coated, welded and seamless. ANSI/ASTM *Standard* A53/A53M-05. American Society for Testing and Materials, West Conshohocken, PA.

ASTM. 2004. Specification for seamless carbon steel pipe for high-temperature service. ANSI/ASTM *Standard* A106/A106M-04b. American Society for Testing and Materials, West Conshohocken, PA.

Briley, G.C. and T.A. Lyons. 1992. Hot gas defrost systems for large evaporators in ammonia liquid overfeed systems. *IIAR Technical Paper* 163. International Institute of Ammonia Refrigeration, Arlington, VA.

Frick Co. 1995. Thermosyphon oil cooling. *Bulletin* E70-900Z (August). Frick Company, Waynesboro, PA.

Glennon, C. and R.A. Cole. 1998. Case study of hydraulic shock events in an ammonia refrigerating system. *IIAR Technical Paper.* International Institute of Ammonia Refrigeration, Arlington, VA.

IIAR. 1998. Minimum safety criteria for a safe ammonia refrigeration system. *Bulletin* 109. International Institute of Ammonia Refrigeration, Arlington, VA.

IIAR. 1999. Equipment, design, and installation of ammonia mechanical refrigeration systems. ANSI/IIAR *Standard* 2-1999. International Institute of Ammonia Refrigeration, Arlington, VA.

IIAR. 1992. Avoiding component failure in industrial refrigeration systems caused by abnormal pressure or shock. *Bulletin* 116. International Institute of Ammonia Refrigeration, Arlington, VA.

Lemmon, E.W., M.O. McLinden, and M.L. Huber. 2002. NIST reference fluids thermodynamic and transport properties—REFPROP, v. 7.0. *NIST Standard Reference Database* 23. National Institute of Standards and Technology, Gaithersburg, MD.

Lloyko, L. 1992. Condensation induced hydraulic shock. *IIAR Technical Paper.* International Institute of Ammonia Refrigeration, Arlington, VA.

Miller, D.K. 1979. Sizing dual-suction risers in liquid overfeed refrigeration systems. *Chemical Engineering* (September 24).

NCPWB. *Welding procedure specifications*. National Certified Pipe Welding Bureau, Rockville, MD.

Shelton, J.C. and A.M. Jacobi. 1997a. A fundamental study of refrigerant line transients: Part 1—Description of the problem and survey of relevant literature. *ASHRAE Transactions* 103(1):65-87.

Shelton, J.C. and A.M. Jacobi. 1997b. A fundamental study of refrigerant line transients: Part 2—Pressure excursion estimates and initiation mechanisms. *ASHRAE Transactions* 103(2):32-41.

Stoecker, W.F. 1988. Chapters 8 and 9 in *Industrial refrigeration.* Business News, Troy, MI.

Timm, M.L. 1991. An improved method for calculating refrigerant line pressure drops. *ASHRAE Transactions* 97(1):194-203.

Wile, D.D. 1977. Refrigerant line sizing. *Final Report*, ASHRAE Research Project RP-185.

BIBLIOGRAPHY

BAC. 1983. *Evaporative condenser engineering manual.* Baltimore Aircoil Company, Baltimore, MD.

Bradley, W.E. 1984. Piping evaporative condensers. In *Proceedings of IIAR Meeting*, Chicago. International Institute of Ammonia Refrigeration, Arlington, VA.

Cole, R.A. 1986. Avoiding refrigeration condenser problems. *Heating/Piping/Air-Conditioning*, Parts I and II, 58(7, 8).

IIR. 2000. Carbon dioxide as a refrigerant. *15th Informatory Note on Refrigerants*. International Institute of Refrigeration, Paris. http://www.iifiir.org/2enpubnotes.php.

Gillies, A. 2005. Design consideration when using carbon dioxide in industrial refrigeration systems. *Proceedings of the Natural Working Fluids 2004, 6th IIR-Gustav Lorentzen Conference*. International Institute of Refrigeration, Paris.

Lloyko, L. 1989. Hydraulic shock in ammonia systems. *IIAR Technical Paper* T-125. International Institute of Ammonia Refrigeration, Arlington, VA.

Nuckolls, A.H. The comparative life, fire, and explosion hazards of common refrigerants. *Miscellaneous Hazard* 2375. Underwriters Laboratory, Northbrook, IL.

Schiesaro, P. and H. Kruse. 2002. Development of a two stage CO_2 supermarket system. *Proceedings of New Technologies in Commercial Refrigeration*. International Institute of Refrigeration, Paris.

Strong, A.P. 1984. Hot gas defrost—A-one-a-more-a-time. *IIAR Technical Paper* T-53. International Institute of Ammonia Refrigeration, Arlington, VA.

Swalha, S. 2005. Using CO_2 in supermarket refrigeration. *ASHRAE Journal* 47(8):26-30.