Evaporative Condenser Engineering Manual

Introduction

The objective of a mechanical refrigeration system is to remove heat from a space or product, and to reject that heat to the environment in some acceptable manner. Evaporative condensers are frequently used to reject heat from mechanical refrigeration systems. The evaporative condenser is essentially a combination of a water-cooled condenser and an air-cooled condenser, utilizing the principle of heat rejection by the evaporation of water into an air stream traveling across the condensing coil.

Evaporative condensers offer important cost-saving benefits for most refrigeration and air-conditioning systems. They eliminate the problems of pumping and treating large quantities of water associated with water-cooled systems. They require substantially less fan horsepower than air-cooled condensers of comparable capacity and cost. And most importantly, systems utilizing evaporative condensers can be designed for a lower condensing temperature and subsequently lower compressor energy input, at lower first cost, than systems utilizing conventional air-cooled or water-cooled condensers.

The Refrigeration System

A schematic of a basic vapor compression system is shown in **Figure 1**. The corresponding heat transfer processes can be represented on a plot of pressure versus enthalpy as shown in **Figure 2**.

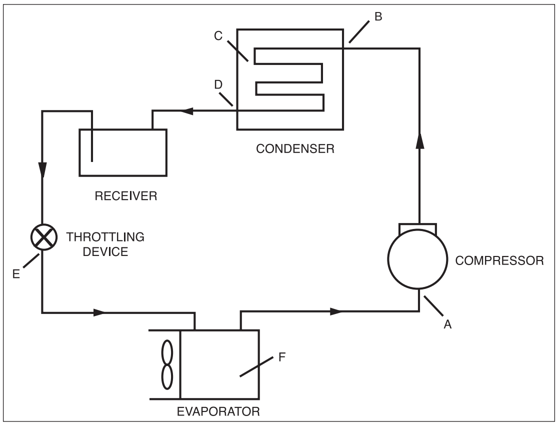

Figure 1. Vapor Compression Refrigeration System

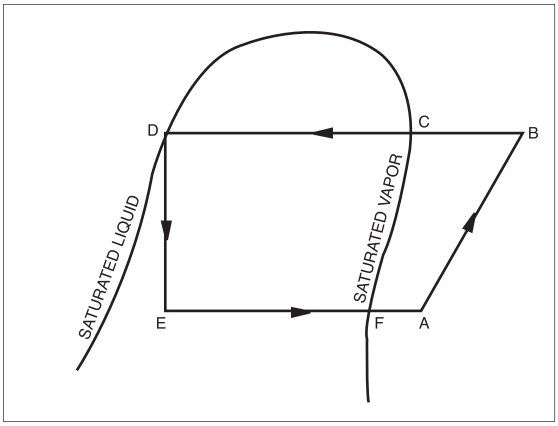

Figure 2. Pressure-Enthalpy Diagram for Compression Refrigeration System

Refrigerant vapor enters the compressor from the evaporator at a slightly superheated condition (A) and is compressed to the condensing pressure (B). The amount of suction gas superheat (F-A) is a function of the type of evaporator and the heat absorbed from the atmosphere as the gas travels along the suction line from evaporator to the compressor.

The compressed and further superheated vapor enters the heat rejection device (condenser) at Point B, where the superheat is quickly removed and the saturated vapor state (Point C) is reached. From Point C to Point D, condensation of the refrigerant occurs at constant pressure until the refrigerant reaches a saturated liquid state at Point D.

There may be some subcooling of the liquid refrigerant near the outlet of the evaporative condenser, but this is quickly dissipated in the drain line from the condenser to the receiver, and in the receiver itself. The drain line and the receiver contain both refrigerant liquid and vapor, and where these two phases coexist, it is impossible for the liquid temperature to remain below the saturation temperature. Therefore, the lower heat content of the subcooled liquid condenses some of the refrigerant vapor until an equilibrium condition is reached at a saturated temperature corresponding to the condensing pressure. So, from a practical standpoint, the refrigerant liquid going to the evaporator should be saturated as represented by Point D. The only exception to this is when a separate subcooling device is used to subcool the liquid after it leaves the receiver.

The refrigerant liquid at Point D is passed through a throttling device (orifice, capillary, or valve) where the pressure is reduced at constant enthalpy to the system suction pressure at Point E. The refrigerant at Point E consists of liquid and vapor, the vapor resulting from the "flashing" of some of the liquid in order to cool the remaining liquid from condensing temperature (Point D) to the evaporating temperature (Point E). The evaporation of the remaining liquid from Point E to Point F represents the useful work of heat pickup in the evaporator.

Refrigeration Heat Rejection Systems

"Once-Through" Condensing System

Water, because of its availability and heat transfer characteristics, has long been the principal medium used for heat rejection from refrigeration and air conditioning systems.

The simplest heat rejection system is one using city, well or surface water directly through a refrigerant condenser and then dumping that water into the sewer, to the ground, or back to the surface water source. The heat removed in the condenser is dependent upon the temperature rise and the flow rate of the water. For an average heat rejection of 15,000 BTUH/TR of refrigeration and a water temperature rise of 20°F in the condenser, approximately 1.5 USGPM of water per ton must be supplied to and wasted from the refrigerant condenser.

This "once-through" type of system at one time was used almost universally for refrigerant condensing. However, the increasing cost of water, high sewerage charges, and restrictions on thermal pollution have made this type of system uneconomical and obsolete.

Refrigerant Condenser and Cooling Tower

One of the early modifications to the "once-through" system was the addition of a cooling tower to permit recirculation of the cooling water and thus conserve water. In a cooling tower, the heated water from the condenser is brought in contact with air, and a small portion of the water is evaporated into the airstream. For each pound of water evaporated, approximately 1,000 BTU are removed from the remainder of the recirculated water. Therefore, only 15 Ib/hr, or 0.03 USGPM of water is used per ton of refrigeration, a theoretical savings of 98% of the water required by the "once-through" system. In actual practice, however, the savings is approximately 95%, because a small amount of water must be "bled off" from the system in order to control the concentration of impurities in the recirculated water.

Evaporative Condenser Engineering Manual

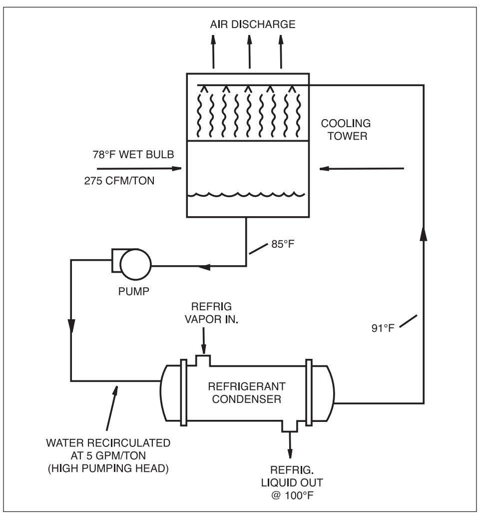

Figure 3. Refrigerant Condenser with Cooling Tower

The temperature of the water leaving the cooling tower is determined by the ambient air wet-bulb temperature. In most areas, design wet-bulb temperatures are such that the temperature of the water leaving the cooling tower is substantially higher than well or surface water temperatures (see **page J8** for geographical wet-bulb data). Therefore, to compensate for the higher cooling water temperature and the additional step of heat exchange introduced by the cooling tower, the condenser water circulation rate and the design condensing temperature often must be increased in comparison to a "once-through" system.

Figure 3 shows a typical arrangement for a cooling tower/refrigerant condenser system. The recirculated water flow rate of 5 USGPM/TR of refrigeration and the 6°F water temperature increase are representative of those existing in an ammonia refrigeration system. The 100°F condensing temperature is about the practical minimum that could be obtained at a 78°F design wet-bulb temperature. Since the pump must circulate water through the refrigerant condenser, cooling tower and interconnecting piping, relatively high pumping head is required.

Halocarbon refrigerant systems may be and usually are designed for somewhat higher condensing temperatures than ammonia systems. This permits a higher water temperature rise through the condenser, but increases the compressor horsepower. Water circulation is normally 3 USGPM/TR versus 5 to 6 USGPM/TR required for an ammonia system.

Air-Cooled Condensers

The air-cooled condenser is another type of heat rejection device used for refrigeration and air conditioning systems.

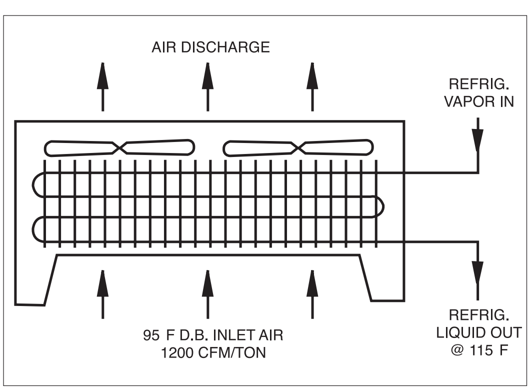

Figure 4. Air-Cooled Condenser

Figure 4 shows a typical air-cooled condenser. Since it does not utilize the evaporative principle, the amount of cooling in the air-cooled condenser is a function of the ambient dry-bulb temperature. Design dry-bulb temperatures are normally 15°F to 25°F higher than design wet-bulb temperatures, so condensing temperatures using air-cooled equipment will be at least that much higher than condensing temperatures using evaporative cooling, resulting in increased compressor horsepower.

Air-cooled condensers reject heat from the refrigerant by sensible heating of the ambient air that flows through them. The low specific heat of air results in a large volume flow rate of air required (approximately four times that of evaporative cooling equipment), with corresponding high fan horsepower and large condenser plan area. The net result of the use of an air-cooled condenser is a savings of water, but at the expense of increased power consumption by the compressor and the condenser.

Evaporative Condensers

Evaporative condensers reject heat from refrigeration and air conditioning systems while using minimum quantities of energy and water. As shown in **Figure 5**, water is pumped from the basin section and is distributed over the exterior of the condensing coil by a series of distribution troughs or spray nozzles. The flow rate of water need only be enough to thoroughly wet the condensing coil to provide uniform water distribution and prevent accumulation of scale. Therefore, minimum pumping horsepower is required.

A fan system forces air through the falling water and over the coil surface. A small portion of the water is evaporated, removing heat from the refrigerant, and condensing it inside the coil. Therefore, like the cooling tower, all of the heat rejection is by evaporation, thus saving about 95% of the water normally required by a "once-through" system.

The evaporative condenser essentially combines a cooling tower and a refrigerant condenser in one piece of equipment. It eliminates the sensible heat transfer step of the condenser water which is required in the cooling tower/refrigerant condenser system. This permits a condensing temperature substantially closer to design wet-bulb temperature, and consequently, minimum compressor energy input.

The temperatures and water flow rate shown in F**igure 5** are typical of an evaporative condenser applied to a refrigeration or air conditioning system at the designated design wet-bulb temperature with either ammonia or a halocarbon refrigerant. These conditions result in an economical evaporative condenser selection. However, a lower condensing temperature and lower compressor energy input could be obtained with a larger condenser at this same wet-bulb temperature.

The evaporative condenser offers a number of important advantages over other condensing systems:

1. **Low System Operating Costs –** Condensing temperatures within 15°F of design wet-bulb are practical and economical, resulting in compressor horsepower savings of 10% or more over cooling tower/condenser systems and more than 30% over air-cooled systems. Fan horsepower is comparable to cooling tower/condenser systems and is about one-third that of an equivalent air-cooled unit. Because of the low pumping head and reduced water flow, water pumping horsepower is approximately 25% of that required for the normal cooling tower/condenser installation.

2. **Initial Cost Savings –**The evaporative condenser combines the cooling tower, condenser surface, water circulating pump, and water piping one assembled piece of equipment. This reduces the cost of handling installing separate components of the cooling tower/condenser system.

   Since the evaporative condenser utilizes the efficiency of evaporative cooling, less heat transfer surface, fewer fans, and fewer fan motors required resulting in an initial material cost savings of 30 to 50% over comparable air-cooled condenser.

3. **Space Saving –** The evaporative condenser saves valuable space by combining the condensing coil and cooling tower into one piece equipment, and eliminating the need for large water pumps piping associated with the cooling tower/condenser system.

   Evaporative condensers require only about 50% of the plan area of a comparable sized air-cooled installation.

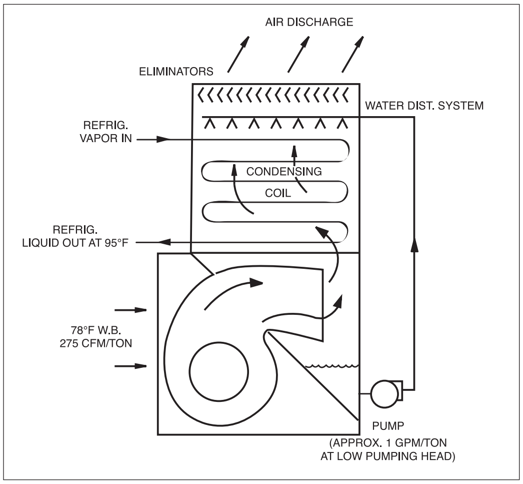

Figure 5. Evaporative Condenser

Evaporative Condenser Engineering Manual

Evaporative Condenser Operation and Installation Recommendations

Winterization

Most evaporative condenser installations operate year-round so consideration must be given to protect against freezing of the recirculated water in locations where the ambient temperature falls below 32°F. There are several protection methods that can be used.

Remote Sump

One method involves the use of an auxiliary sump tank with a spray water recirculating pump located within a heated space. **Figure 6** shows a typical arrangement of an evaporative condenser with a remote sump tank. All of the water in the condenser basin drains to the indoor sump whenever the recirculating pump is not operating. The indoor remote sump must be sized to provide an operating suction head for the pump and a surge volume above this operating level to hold all the water that will drain back when the pump is shut down. This includes water in suspension in the condenser and the water in the condenser basin during normal operation, plus that in the pipe lines between the condenser and sump. The amount of water in suspension plus the amount of water in the condenser basin during remote sump operation for BAC condensers is available on **page J226** or from your local BAC Representative.

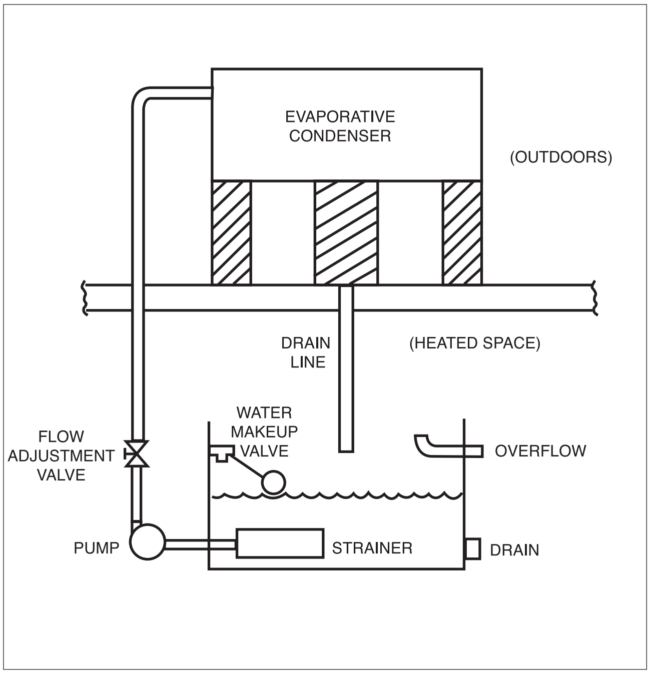

Figure 6. Evaporative Condenser With Remote Sump Tank

Recirculating water pumps for remote sump applications must be selected for the required flow at a total head which includes the vertical lift, pipe friction (in supply and suction lines) plus the specified pressure required at the inlet header of the water distribution system (should not exceed 2.0 psig for all evaporative condensers). A balancing valve must always be installed in the discharge line from the pump to permit adjusting flow to the condenser.

Basin Heaters

Occasionally, because of the condenser location or space limitations, a remote sump application may be impractical. In such cases, electric heaters or steam coils can be installed in the condenser basin to prevent freezing at low ambient temperatures when the condenser is completely idle. In addition, the pump suction line, pump, and pump discharge pipe (up to overflow connection) should be traced with heating tape and insulated.

Capacity Control

Most refrigeration and air conditioning systems are subject to wide load variations and substantial changes in ambient temperature conditions. Where refrigerant control requires a reasonably constant condensing pressure, some form of capacity control is required.

Fan Cycling

Fan cycling is the simplest method of capacity control on evaporative condensers. This method can result in relatively large fluctuations in condensing pressures, however. On ammonia systems, most evaporators are fed by high pressure or low pressure float valves or float switches which are less sensitive to variations in head pressure. On this type of system, fan cycling of the evaporative condenser will usually provide satisfactory capacity control on the high side of the system. This is particularly true on larger ammonia systems, where the evaporative condenser may have several fan motors which can be cycled in steps.

Halocarbon systems generally utilize evaporators controlled by thermal expansion valves. A reasonably constant pressure differential across the thermal expansion valve is required for its proper operation. Therefore, this type of system requires a closer degree of evaporative condenser capacity control than can be obtained with fan cycling.

Two-Speed Fan Motors

The number of steps of capacity control can be doubled by using two-speed fan motors in conjunction with fan cycling. This is particularly useful on single fan motor units which normally have only one step of capacity control using simple fan cycling.

Normally the two-speed fan motor will be selected so that the low speed is half of the full speed, such as 1800/900 rpm. An evaporative condenser will deliver approximately 58% of its rated capacity at half speed operation.

An additional benefit of two-speed fan motors is reduced fan horsepower at low speed. Brake horsepower varies as the cube of the fan speed, so the unit will use only about one eighth of the full load brake horsepower when operating at low speed. Maximum load and maximum wet-bulb temperature occur infrequently, so the unit will be operating at half speed and hence sharply reduced brake horsepower much of the time.

Another benefit of two speed motors is that when an evaporative condenser is operating at low speed it will have substantially lower operating sound levels. The sound pressure levels of both centrifugal and propeller fan evaporative condensers will be reduced by four to ten decibels, depending on the sound frequency.

BALTIGUARD™ Fan System

The BALTIGUARD™ Fan System consists of two standard single-speed fan motor and drive assemblies. One drive assembly is sized for full speed and load, and the other is sized approximately 2/3 speed and consumes only 1/3 the design horsepower. This configuration allows the system to be operated like a two-speed motor, but with the reserve capacity of a standby motor in the event of failure. As a minimum, approximately 70% capacity will be available from the low horsepower motor, even on a design wet-bulb day. Controls and wiring are the same as those required for a two-speed, two-winding motor. Significant energy savings are achieved when operating at low speed during periods of reduced load and/or low wet-bulb temperatures.

BALTIGUARD PLUS™ Fan System

The BALTIGUARD PLUS™ Fan System builds on the advantages of the BALTIGUARD™ Fan System by adding a VFD on one motor.

Evaporative Condenser Engineering Manual

Independent Fan Operation

The independent fan option consists of one fan motor and drive assembly for each fan to allow independent operation, adding redundancy and an additional step of fan cycling and capacity control to models with more than one fan.

Variable Frequency Drives

Precise capacity control and energy savings are achieved with the BAC variable frequency drive (VFD) option. VFDs offer a more efficient and durable way to reduce fan speed compared to fan cycling, fan discharge dampers, or mechanical speed changers. The inherent ability for VFDs to provide soft starts, stops, and smooth accelerations prolongs the mechanical system life (fans, motors, belts, bearings, etc.). Sound levels are also reduced at lower fan speeds, and start-up noise is eliminated with the soft start feature. See section E for information on BAC's enclosed control and variable frequency drive offerings.

NOTE: An inverter duty motor is required for all models operating with a variable frequency drive.

Modulating Fan Discharge Dampers

Modulating fan dampers, located in the fan discharge of centrifugal fan units, provide an infinite number of capacity control steps. Modulating dampers also affect a reduction in fan motor horsepower which is approximately proportional to the reduction in CFM as the dampers move toward the closed position.

- **Single Coil Circuit Units –** On a single circuit condenser, a condensing pressure sensing element is located in the compressor discharge line or in the receiver (see **Figure 7**). The pressure controller is electrically connected to a damper motor, and when condensing pressure changes, a signal is sent to the damper motors to reposition the dampers and provide more or less airflow as required.
- **Multiple Coil Circuit Units –** On multiple circuit condensers where it is necessary to control condensing pressures for two or more circuits, a spray water temperature sensing controller, located in the basin, is substituted for the condensing pressure controller. Maintaining spray water at approximately summertime temperatures will indirectly provide control of condensing pressures on the multiple condenser circuits. Even with a very light load on one circuit, the condensing temperature in that circuit cannot fall below the spray water temperature.

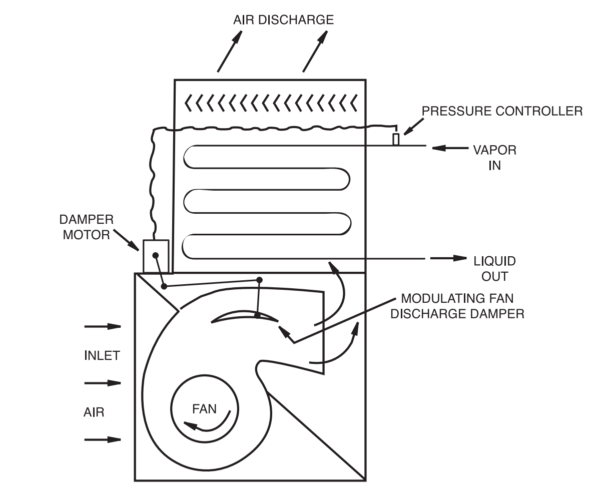

Figure 7. Evaporative Condenser With Modulating Fan Discharge Dampers (Single Coil Circuit Unit)

NOTE: This system will not provide control if the basin is drained for dry condenser operation in winter.

Dry Operation

During winter operation, when the refrigeration load may be reduced and the ambient air temperature is far below the design conditions, the evaporative condenser may be operated dry, i.e., without recirculated water flow. This reduces the capacity of the unit to more nearly match the reduced load.

Dry operation of an evaporative condenser is intended to be a seasonal process. Water pump cycling should not be used for capacity control. Condenser capacity changes greatly with and without spray water, so that this method of control often results in short cycling of the recirculating pump. In addition, alternate wetting and drying of the condenser coil promotes formation of scale on the condensing surface.

Evaporative condensers should not be operated dry in sub-freezing ambient temperatures while the recirculated water is stored in the basin of the unit. The flow of cold air through the unit may freeze the water, even if electric heaters or steam coils have been provided for freeze protection. These heaters are designed to prevent freezing only when the pumps and fans are idle. Furthermore, air turbulence created by the fans will blow water throughout the interior of the unit, and cause icing on the cold surfaces. It is recommended that the evaporative condenser be completely drained of water when dry operation is desired.

Condenser Piping

See **page J181**.

Purging and Purge Piping

Source of Non-Condensables

Air and other non-condensable gases collect in refrigeration systems from several sources:

1. Poor evacuation of a new system prior to charging.
2. A leak into the system low side if operation is at pressures below atmospheric.
3. Failure to evacuate completely after part of a system has been open for repair.
4. Chemical breakdown of oil and/or refrigerant.

If permitted to accumulate, non-condensables in the system cause high condensing pressures and, therefore increased power input to the compressors.

Checking the System for Non-Condensables

To check the system for non-condensable gases, first close the valve in the liquid line running from the receiver to the evaporator (king valve), then pump down the system slightly, enough to assure that if any non-condensables are present they are pumped over to the high side. Immediately after pump-down, close the discharge valve on the compressor. Operate the evaporative condenser for at least two hours or until the water temperature in the basin or remote sump is the same as the entering wet-bulb temperature. Then the temperature corresponding to the pressure in the evaporative condenser should correspond, or nearly so, to the wet-bulb temperature of the entering air. If this temperature is higher than the wet-bulb temperature by more than 2°F, the system has an excessive amount of non-condensables. (Be sure that all gauges are accurate when checking for non-condensables.)

Evaporative Condenser Engineering Manual

Purge Connections

Purging at the high point of the system can only be effective when the system is down. During normal operation the non-condensables are dispersed throughout the high velocity refrigerant vapor and too much refrigerant would be lost when purging from this high point.

However, purging at the condenser coil outlet can be effectively accomplished during system operation. The non-condensables will carry through the condensing coil with the refrigerant liquid and vapor and tend to accumulate in the condensing coil outlet header and connection where the temperature and velocity are relatively low.

In the BAC condenser coil design, the refrigerant outlet connection is tangent to the top of the coil header so non-condensables cannot trap in the header. A 1/2" or 3/4" purge connection should be cut into the top of the liquid outlet along the horizontal run (for a refrigerant connection size less than 4", a purge valve may be provided with the BAC condenser; contact your local BAC Representative for confirmation). Each connection must be valved so that each coil can be purged separately.

Purge Piping

All of the purge connections on the condenser coils plus the purge connection in the receiver may be cross connected to a single purge line, which is connected to an automatic purger. However, only one purge valve should be open at a time. Opening two or more valves tied together equalizes the coil outlet pressures and the effect of the vertical drop legs is lost.

Location

In order to obtain specified performance from an evaporative condenser installation, it is essential that the unit or units be located so as to guarantee design airflow to each unit while minimizing recirculation of the discharge air.

A single condenser located outdoors will seldom pose any layout problem. However, multiple units or a single unit with a fan side facing an adjoining building or wall must be located with reference to the wall (or to each other) to allow ample space for airflow to the fans. **Figure 8** illustrates those dimensions which must be taken into consideration when locating evaporative condensers. BAC Representatives can provide specific location recommendations for the various models of BAC evaporative condensers that are available. Refer to layout guidelines on **page J88**. For PCC layout guidelines refer to **page J108**.

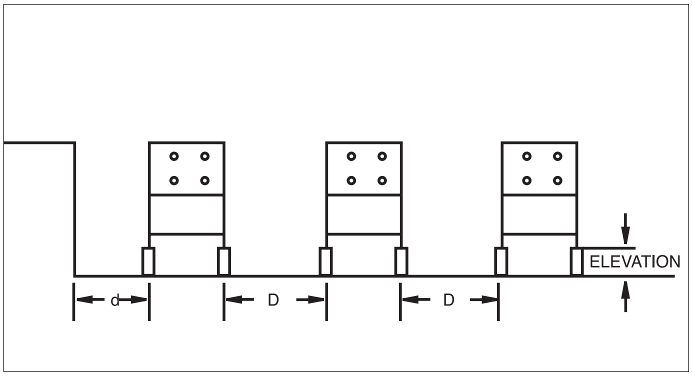

Figure 8. Condenser Spacing Parameters

NOTE: In **Figure 8**, the top (discharge) of the condenser should be at the same or higher level than an adjoining building or wall in order to minimize recirculation caused by down draft between the condenser and wall. Such a down draft might be created by winds blowing across the condenser discharge towards the wall. If for some reason, it is not possible to raise the condenser to the level of the top of an adjoining building or wall, a discharge hood can be used on centrifugal fan condensers (see **Figure 9**). The hood increases the discharge air velocity and elevates the point of discharge to a height where recirculation is minimized. Elevating the condenser increases the area for airflow from beneath the unit and permits placing the condensers closer together or closer to an adjoining wall. However, there is no spacing advantage to elevations greater than 10 feet in this respect.

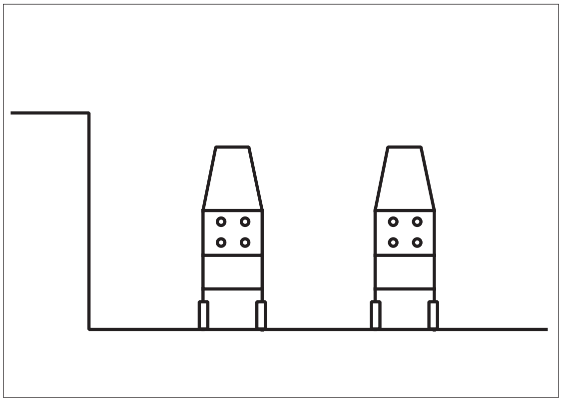

Figure 9. Discharge Hoods to Increase Discharge Air Velocity

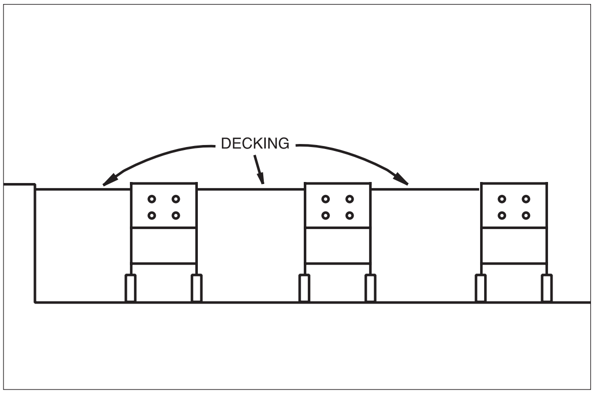

Figure 10. Decking Between Condenser and Wall or Between Condensers

Occasionally, the minimum spacing cannot be provided. By "decking over" between the condensers or between a condenser and an adjoining wall (providing a solid surface between the air discharge and air intakes, **Figure 10**), the condenser spacing can be decreased accordingly.

Condenser installations involving large capacities and/or multiple units do not lend themselves to the application of rigid layout guidelines. Some such installations virtually create their own environment and all potential problems of airflow and recirculation are magnified. In some cases, it may be necessary to increase the design wet-bulb temperature for which the condensers are selected. It is recommended that the layout parameters of any installation other than a single unit on an open roof be reviewed by the local BAC Representative.

Evaporative Condenser Engineering Manual

Recirculated Water System

An evaporative condenser obtains its ability to condense the refrigerant by evaporating a portion of the water recirculated over the condensing coil. As the water evaporates, any impurities present in the supply water remain in the unit. The concentration of impurities increases rapidly, and continues as long as the unit is in operation.

In addition, any impurities in the air (such as chemical fumes in an industrial area or salt air near the coastline) will be absorbed by the recirculated water, resulting in a corrosive solution.

To prevent an excessive build-up of impurities in the recirculated water, it is recommended that water be removed or “bled” from the unit at a rate at least equal to the amount of water being evaporated. In many localities this constant bleed and replacement with fresh water will keep the concentration of impurities in the system at an acceptable level. Note: In addition to any bleed or chemical treatment, all systems must be treated for biological contaminants.

An evaporative condenser will evaporate approximately 3 USGPM of water per 100 tons of refrigeration. Allowing an equal quantity for bleed, total water consumption is approximately 6 USGPM per 100 tons of refrigeration.

Most evaporative condensers that are furnished with a factory-installed recirculating pump (or pumps) are also furnished with a water bleed line and flow adjusting valve. Units furnished for remote sump application must have a bleed line and valve installed at the remote sump. It is important to keep the bleed lines operative and properly adjusted through periodic inspection. The water removed through the bleed line will more than pay for itself through increased unit life.

If the condition of the water and/or the air is such that continuous bleed will not control scaling or corrosion, the recirculated water must be treated. A reputable local water treatment company should be consulted to analyze the system water and recommend proper treatment. See the appropriate *Operation and Maintenance Manual* available at www.BaltimoreAircoil.com.

Most evaporative condensers are constructed of galvanized (zinc-coated) steel, and any chemical treatment must be compatible with this material. Chemicals should be fed into the recirculated water on a continuous metered basis to avoid localized high concentration which may cause corrosion. Batch feeding of chemicals does not afford adequate control of water quality, and is not recommended.

When acid treatment is required, it is essential that the acid be accurately metered into the recirculated water, and the concentration properly controlled. Acid should not be fed directly into the cold water basin; it must be fed into the recirculated water piping so it will mix thoroughly before reaching the basin.

Special Applications

Desuperheaters

A desuperheater is an air-cooled finned coil usually installed in the discharge airstream of an evaporative condenser. **Figure 11** shows a typical arrangement. Its primary function is to increase the condenser capacity by removing some of the superheat from the discharge vapor before the vapor enters the wetted condensing coil. The amount of superheat removed is a function of the desuperheater surface, condenser airflow and the temperature difference between refrigerant temperature and the temperature of the air leaving the condenser. Practically, the application of a desuperheater is limited to reciprocating compressor ammonia installations where discharge temperatures are relatively high (250°F to 300°F).

It is economically impractical to provide a desuperheater on an evaporative condenser with enough heat transfer surface to remove all of the superheat in the ammonia refrigerant. Therefore, complete superheat removal is never attained under design conditions of load and ambient wet-bulb temperature with the standard desuperheater coils furnished by evaporative condenser manufacturers. The anticipated capacity increase on an ammonia condenser with a standard desuperheater is in the area of 10% rather than the 16% theoretically possible.

Occasionally, where condenser space is limited, the addition of a desuperheater may permit a smaller plan area unit. However, with the numerous size increments available in today's evaporative condensers, such instances are rare. The air-cooled desuperheater is not as efficient as wetted condenser surface, so it is more economical to select a condenser with additional wetted surface to achieve greater capacity.

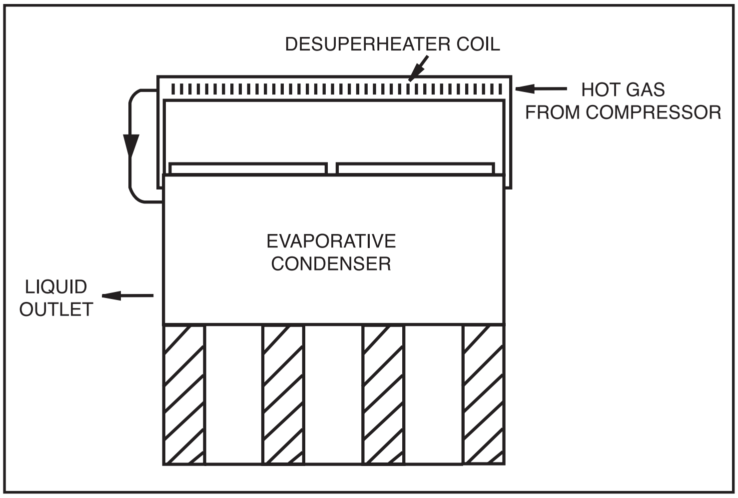

Figure 11. Evaporative Condenser with Desuperheater Coil

Desuperheaters have been recommended by some manufacturers to assist in oil removal from the ammonia vapor and also to minimize scaling of the upper tubes of the wetted condensing coil by reducing entering refrigerant gas temperatures to the wetted coil.

For oil removal, an oil separator is installed between the desuperheater coil and the wetted condenser coil. The theory is that cooling of the hot discharge refrigerant vapor will promote condensation of the oil vapor from the refrigerant-oil mixture and separation of oil from the refrigerant in the oil separator. This claim has merit. However, there is normally no control over the amount of heat removed from the refrigerant vapor in the desuperheater coil. At less than design load or wet-bulb temperature, the desuperheater coil often becomes a condensing coil, and when liquid refrigerant mixes with liquid oil, separation becomes quite difficult.

Evaporative Condenser Engineering Manual

Today there are many oil separators with high efficiencies for removing oil from the hot discharge vapor as it leaves the compressor. The oil separator can be located in the engine room where it can be monitored by the operating engineer and where it is not exposed to the ambient temperatures that would cause refrigerant condensation. From the scaling standpoint, the presence or absence of a desuperheater is immaterial. The primary factor that determines the tendency to form scale on the wetted coil of an evaporative condenser is the external surface temperature of the coil. At the inlet of the wetted coil where only hot refrigerant vapor exists, the internal heat transfer coefficient is quite low. Despite the high vapor temperatures at the inlet (250°F to 300°F), the low internal coefficient reduces the rate of heat transfer through the coil/tubes at that point. The resulting coil surface temperature at the inlet is not appreciably different from the coil surface temperature in the condensing portion of the coil. Therefore, scaling in an evaporative condenser becomes primarily a function of adequate water distribution over the coil, proper bleed-off to prevent concentration of solids, and proper water treatment where water conditions are particularly bad.

The increasing use of screw compressors for industrial refrigeration systems further obsoletes the use of a desuperheater. The screw compressor is an oil seal, oil cooled unit, with the cooled oil injected into the compressor in contact with the refrigerant vapor. Larger, efficient, de-mister type oil traps furnished as part of the screw compressor package minimize problems of oil carryover. Because the cooled oil is in direct contact with the refrigerant vapor, discharge temperatures are relatively low on water-cooled screw compressors (160°F to 190°F), and even lower on refrigerant liquid injected screw compressors (approximately 120°F). Consequently, any capacity gain of a desuperheater used on a screw compressor installation is negligible.

Refrigerant Liquid Subcooling (Halocarbon Systems)

In the case of air conditioning or refrigeration systems, the pressure at the expansion device feeding the evaporator(s) can be substantially lower than the receiver pressure due to liquid line pressure losses. If the evaporator is above the receiver, the static head at the evaporator is less than at the receiver, which further reduces the pressure at the expansion device.

A refrigerant remains in liquid form only as long as the liquid pressure is at or higher than the saturation pressure corresponding to its temperature. Any pressure reduction in the liquid line between the receiver and the expansion device causes flashing or vaporization of some of the liquid. The presence of this flash gas will cause erratic operation of the thermal expansion valve and reduce the valve capacity, sometimes to the point of starving the evaporator.

To avoid liquid line flashing where the above conditions exist, it is necessary to subcool the liquid refrigerant after it leaves the receiver. The minimum amount of subcooling required is the temperature difference between the condensing temperature and the saturation temperature corresponding to the pressure at the expansion valve. To determine the degree of subcooling required, it is necessary to calculate the liquid line pressure drop including valves, ells, tees, strainers, etc., and add to it the pressure drop equivalent to the static head loss between the receiver and the thermal valve at the evaporator, if the evaporator is located above the receiver.

The static head loss due to a vertical rise in the liquid line is a function of the refrigerant density. At normal condensing temperatures, the static head loss is approximately 0.50 psi per foot rise for R-22.

As an example of the calculation to determine the amount of subcooling required, assume an R-22 system designed for 105°F condensing temperature (210.7 psig) with a thermal valve fed evaporator 25 feet above the refrigerant receiver.

Assume that detailed calculations of friction pressure drop indicate a line loss of 8.0 psi. The static head loss for a vertical rise of 25 feet (12.5 psi), plus 8 psi friction pressure drop, results in a total pressure drop of 20.5 psi. So the pressure at the expansion valve is 210.7 - 20.5, or 190.2 psig, and the saturation temperature corresponding to 190.2 psig is 98°F. Therefore, the minimum amount of subcooling to prevent flashing is 105°F (condensing temperature) minus 98°F, or 7°F.

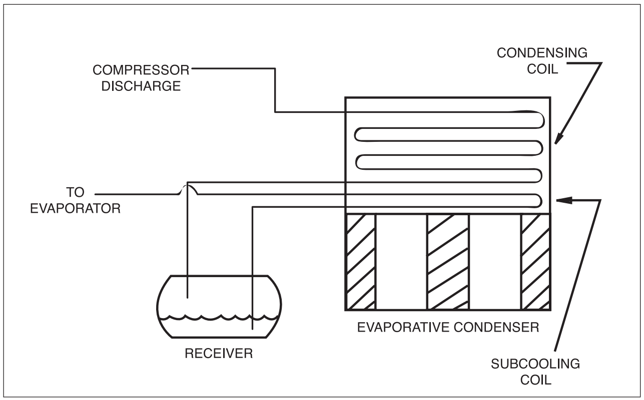

Figure 12. Recommended Piping for Evaporative Condenser with Liquid Subcooling Coil

Some compressor manufacturers publish their compressor ratings based on a fixed amount of subcooling at the thermal expansion valve. Subcooled liquid at the expansion valve of the evaporator does increase system capacity since it increases the refrigeration effect per pound of refrigerant circulated. But the increase is relatively small and seldom justifies the cost of the subcooling device and piping for this reason alone. However, where compressor ratings based on subcooled liquid are used, the specified amount of subcooling must be added to that required for liquid line pressure drop and static head loss.

One method commonly used for supplying subcooled liquid for halocarbon systems is to provide a subcooling coil section in the evaporative condenser, located below the condensing coil (see **Figure 12**). Depending upon the design wet-bulb temperature, condensing temperature, and subcooling coil surface, these sections will normally furnish approximately 10°F of liquid cooling. However, to be effective, the subcooling coil must be piped between the receiver and evaporator as shown in **Figure 12**.

NOTE: Increasing the evaporative condenser size over the capacity required for the system will not produce liquid subcooling. The increased condenser capacity will result only in lower operating condensing temperatures. The same result will occur if the condensing coil is piped directly to the subcooling coil.

Low temperature, multistage ammonia refrigeration systems often use liquid subcooling between stages for more economical operation. However, subcooling coils in an evaporative condenser are seldom, if ever, used with an ammonia refrigeration system for several reasons:

1. Design condensing temperatures are generally lower with ammonia, thus limiting the amount of subcooling that can be obtained.
2. The density of ammonia liquid is approximately 37 LBS/ft^3, less than half that of the normally used halocarbons, and static head losses are proportionately less.
3. The expansion devices and system designs normally used for ammonia systems are less sensitive to small amounts of flash gas.
4. The high latent heat of ammonia (approximately 480 BTU/lb versus 70 BTU/lb for R-22) results in comparatively small amounts of flash gas with a liquid line properly sized for low pressure drop.

Evaporative Condenser Engineering Manual

Multiple Circuit Condenser Coils

The coil in a single condenser can be split in sections to provide a number of individual circuits. A multiple circuit coil is used primarily with the common halocarbon refrigerant (R-134a, R-22, R-404A, R-507) on small air-conditioning or refrigeration systems with two or more reciprocating compressors. The reason for this is that proper oil return to the compressors can be a problem on these systems, and it is good design practice to isolate each compressor.

In general, the halocarbon refrigerant are highly miscible with oil, the degree of miscibility being a function of the refrigerant, the type of oil, the pressure and temperature of the mixture. During normal operation, some oil is lost from the crankcase of the reciprocating compressor and this oil travels around the refrigerant circuit with the refrigerant. It is essential that the oil lost from the compressor be returned to it.

In order to avoid oil return problems, it is common practice on the smaller (200 tons and below) halocarbon refrigeration and air-conditioning systems to design independent refrigerant circuits where two or more reciprocating compressor systems are involved. In order to use a single evaporative condenser, the condenser coils can be split internally to accommodate the capacities of the individual systems.

This practice is not followed with R-717 (ammonia) systems. Oil and ammonia are practically immiscible so that most of the oil carried over from the reciprocating compressors can be removed with discharge line oil separators and returned either directly to the individual compressor crankcase or to an oil receiver and then to the compressor crankcase.

If multiple compressor halocarbon systems are not designed with isolated circuits, an oil return system must be provided to return oil to each compressor crankcase.

Auxiliary Cooling Using Condenser Basin Water

During normal evaporative condenser operation, the recirculated spray water is maintained at a temperature some point higher than the inlet air wet-bulb temperature and lower than the condensing temperature. The exact recirculated water temperature is determined by these two operating parameters. Therefore, this water can be considered as a source of relatively cool fluid for auxiliary cooling requirements on refrigeration plants, such as jacket cooling for reciprocating and rotary compressors, jacket cooling for air compressors and vacuum pumps, and oil cooling for screw compressors.

Water is taken from the basin of the condenser or the remote sump and is pumped to the source of heat, usually by a separate pump (see **Figure 13**). In most cases, only a fraction of the evaporative condenser flow rate is required for cooling purposes. The water flows through the heat source, increases in temperature, and is then returned to the condenser basin or remote sump. The heated water then mixes with the basin water producing a mixture temperature somewhat higher than the normal recirculated water temperature. An increase in temperature of the recirculated water by virtue of an external cooling load has the effect of reducing condensing capacity, but the penalty is relatively small. Consult your local BAC Representative for specific evaporative condenser performance data on systems utilizing basin water for auxiliary cooling.

Using a portion of the recirculated spray water for external cooling purposes is an effective and simple concept. However, there is a significant drawback to this cooling system that does not always make it desirable. An evaporative condenser characteristically behaves as an air washer, stripping dirt and dust particles from the air circulating through it, and holding them in suspension in the recirculated water. Consequently, this can create serious clogging of compressor jackets or heat exchanger tubes. Frequent cleaning of the heat exchanger or sophisticated filtering equipment is usually required.

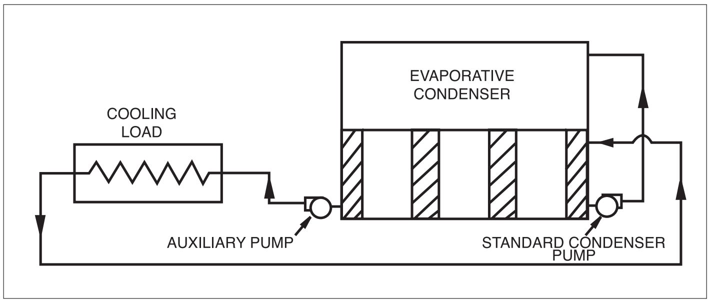

Figure 13. Auxiliary Cooling Using Condenser Basin Water

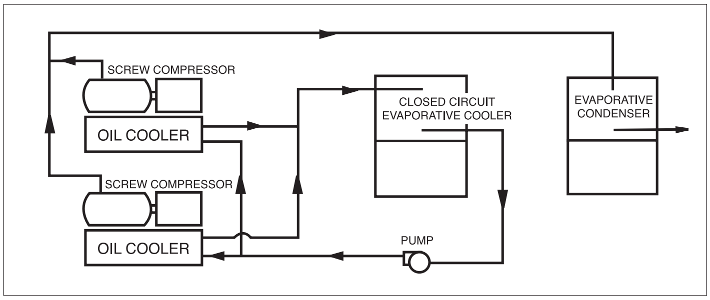

Figure 14. Evaporative Condenser With Closed Circuit Cooling Tower for Fluid Cooling: Cooling Oil Coolers for Refrigeration Screw Compressors

Closed Circuit Fluid Cooling

To eliminate the problem of system contamination associated with using spray water for auxiliary cooling, BAC recommends that a closed system be used for that cooling whenever possible. A separate closed circuit cooling tower, or a split circuit coil in the evaporative condenser, with one circuit for condensing the refrigerant and the other for cooling the liquid, are two good solutions.

As an example, a closed circuit cooling tower could be used to cool water or glycol solution for oil coolers of refrigeration screw compressors. **Figure 14** shows a typical arrangement. This is the ideal cooling system because it provides the following important advantages:

1. Provides closed loop cooling, which precludes the contamination of system fluid.
2. Provides independent control of the condensing and water-cooling systems by separating these two functions into two or more units.
3. Permits the evaporative condenser to be operated as an air-cooled condenser in cold weather, thus minimizing freeze up problems.

It is important to note that if the closed circuit cooling tower is installed in a freezing climate, an antifreeze (glycol) solution must be used instead of water. If a closed circuit cooling tower coil containing water is not provided with a supplementary heat load after shutdown, and is exposed to ambient temperature below 32°F, the water could freeze and rupture the coil. Other winterizing precautions similar to those described earlier in this manual for evaporative condensers apply equally to closed circuit cooling towers.

A separate closed circuit cooling tower for fluid cooling cannot always be justified, particularly on smaller installations. For instance, on refrigerated plants involving only one or two water-cooled screw compressors, it may be more economical to furnish an evaporative condenser with a split circuit coil, with one circuit for condensing refrigerant and the other isolated for fluid cooling. This approach lacks one of the features of the separate unit arrangement, i.e., the fluid cooling and condensing functions cannot be controlled independently. Both functions are handled within the same unit, but the heat rejection capacity of the unit must be controlled by either the condensing pressure or the leaving fluid temperature. Consequently it is necessary to sacrifice close control of one of these parameters, usually the leaving fluid temperature.

Using an evaporative condenser for both condensing and fluid cooling also limits the permissible inlet and outlet fluid temperatures on the fluid cooling circuit. Careful engineering analysis is required to establish satisfactory temperature criteria and properly select the evaporative condenser. Consult your local BAC representative for specific recommendations on split circuit evaporative condensers.

Evaporative Condenser Engineering Manual

Thermosyphon Oil Coolers

Thermosyphon oil coolers (TSOC) operate as unique high-temperature chillers using high-pressure liquid ammonia saturated at 70°F to 95°F (21°C to 35°C), and evaporating in the TSOC at the system condensing pressure 70°F to 95°F (21°C to 35°C). This is made possible by using a gravity feed recirculation refrigerant system based on drawing liquid ammonia from a receiver or auxiliary liquid supply. The liquid source is at condensing pressure and located about 6 to 8 ft (1.8 to 2.4 m) above the TSOC. This source is connected directly via low-pressure drop piping to the tube side of the TSOC shell and tube heat exchanger (see Figure 1).

The oil to be cooled is piped through the shell-side of the cooler. When the oil entering the cooler is warmer than the saturated liquid temperature, some of the ammonia liquid will boil at the saturated temperature within the tubes, cooling the oil. Vapor generated in the TSOC tubes will rise through the refrigerant return line, which is connected to the liquid receiver above the liquid level.

The vapor bubbles in the return line lower the density of the return liquid/vapor to approximately 3 lb/ft^3 (48 kg/m^3). The supply liquid line, which contains only liquid ammonia, is heavier, weighing about 37 lb/ft^3 (592 kg/m^3).

The weight imbalance between the two legs induces a thermosyphon refrigerant flow that will be in excess of the oil cooler load requirement. The excess liquid returns with the vapor up to the receiver vessel. The liquid drops into the receiver and the vapor is vented to the condenser inlet.

When a TSOC is operating properly, the refrigerant inlet and return lines will be at the same temperature.

Two problems that can cause the TSOC to lose oil-cooling capacity and/or stop cooling entirely.

The first problem, the gradual loss of cooling capacity, may occur on any TSOC application, but is generally found on those ammonia systems that have screw compressors and some older reciprocating compressors with less efficient (non-coalescer) mesh-type oil separators. Coalescing oil separators typically permit minimal oil carryover of 5 to 10 ppm by weight (pound of oil per pound of ammonia pumped). Mesh oil separators will allow more carryover, on the order of 30 to 100 ppm, which may result in 6 to 20 times the oil carryover as with screw compressors.

Oil is virtually immiscible with ammonia. Because it is heavier than liquid ammonia, it will be located at the bottom of any ammonia liquid vessel, including the ammonia in high-pressure receivers and auxiliary TSOC receivers. If the supply of ammonia for TSOCs is taken from the bottom of these vessels, then some oil may be drawn into the TSOC, where it will settle to the bottom, logging the lower tubes and reducing the TSOC capacity by preventing these tubes from participating in the cooling process.

When cooling loss occurs, close the liquid supply to the TSOC, pump out the remaining liquid ammonia, then close the return line. Next, stop the unit and drain the oil from the bottom drain connections on the TSOC heads, but not the shell that contains the oil being cooled. This should clear the problem, but it may require periodic draining every few months or so.

When the problem requires weekly draining, then more serious action is indicated. The oil carryover rate is out of control and the low-side evaporators, level switches and pressure regulators are probably also oil logged. When this occurs, evaluate the oil carryover, track the amount added to reciprocating and screw compressors and the amount drained from the low side of the system. Chances are that oil carry over is extreme and an oil management system is indicated.

An oil management system can include a special “downstream coalescing separator” located between the reciprocating compressors and the condenser. This separator can be designed to remove oil in the range of 5 to 10 ppm carryover, equivalent to that of screw compressors. The oil can be collected in an oil receiver, properly filtered and directed back to the reciprocating compressor crankcases via float level control for automatic handling. This method will rapidly pay for itself considering the savings of the cost of labor, the cost of new oil and the disposal of used oil.

The second problem is that the TSOC units work well for months and then, all at once, one or more coolers in a large plant with many TSOCs connected to a single supply source stop cooling. Obviously, the oil overheats and the screw compressors shut down on high oil temperature. Generally, this occurs with a season change—even mild season changes.

The problem is neither with the TSOC nor with the oil. It is because the TSOC is tied into the same receiver with any number of other TSOCs. This is not bad. It is done all the time. However, the piping for the return lines to the receiver must be respected. The premise is that the thermosyphon principle operates on minimal pressure differences (the 6 to 8 ft [1.8 to 2.4 m] height). One of the primary rules is that the vapor generated in the TSOC must return to the condenser inlet at the same condensing pressure.

Figure 2 shows the proper way to pipe the return on multiple TSOC systems. It is imperative that each TSOC return reaches the receiver return header without influence from the other coolers or they may interfere with each other. This may result in spilling liquid refrigerant down a neighbor's return line, causing the fine pressure balance to be upset and stopping the TSOC refrigerant flow.

If there are multiple TSOCs and one unit quits cooling, look up at the TSOC return lines. It may be that several of the TSOC returns are manifolded into a horizontal line that rises several feet before entering the auxiliary receiver. This is the problem.

The returns will have to be repiped in accordance with the intent of Figure 2. Each TSOC return is individually connected into a return header, located above the receiver liquid level and sloping toward the receiver. Each return must connected to the header by entering from above, so that the liquid return from one TSOC cannot interfere with any other.

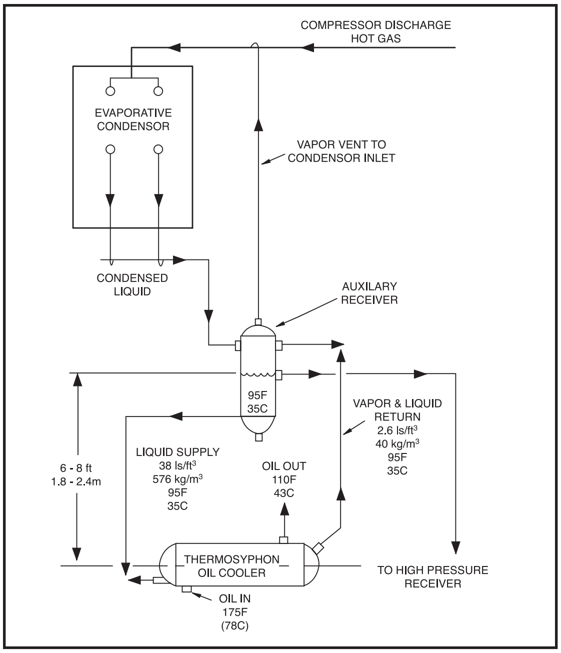

Figure 1. A Basic Flow Diagram of a TSOC

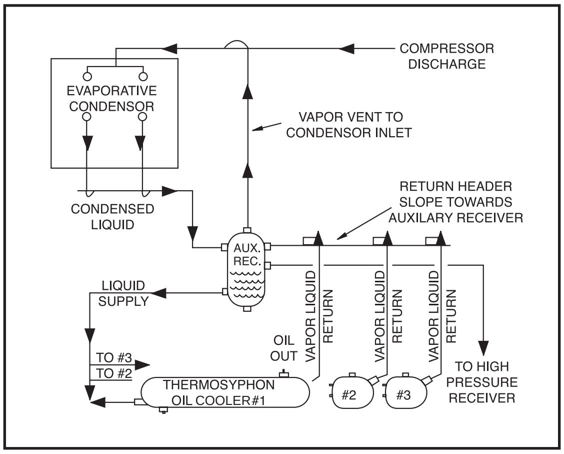

Figure 2. Multiple Thermosyphon Piping

NOTE: Rudy Stegmann, P.E., President of The Enthalpy Exchange. Williamsburg, VA.