

Selecting Lubricant for the Ammonia Refrigeration System

By George C. Briley, P.E., Fellow/Life Member ASHRAE

When selecting a lubricant for your ammonia refrigeration system, several requirements should be considered.

1. **Is the lubricant approved by the manufacturer of the compressors?** Most refrigeration compressor manufacturers and packagers provide a list of lubricants that they recommend. Some do not consider the application of the compressor or the type of system in which it is used. For ammonia compressors, the usual viscosity recommendations are 68 centistokes (300 saybolt seconds universal, SSU). The normal question might be: "Would my compressor operate more efficiently using a less viscous lubricant?" I have asked the question of several screw compressor manufacturers, and their usual answer is "no."

Always follow the manufacturers' recommendations and never use any additive that is advertised to increase your system capacity by 15% or 20%.

2. **Is it readily available?** The lubricant supplier should maintain a stock near your plant or should keep it in your plant. Note that lubricant should not be kept in the equipment room.

3. **What is its vapor pressure?** Besides the viscosity requirement the lubricant vapor pressure is most important. All lubricants have a volatile component that becomes a vapor at elevated temperatures and pressures. These vapors cannot be separated in a coalescing element, which is the normal lubricant separation device used in screw compressors. These coalescers are usually about 99% effective while separating lubricant aerosols (a fine mist resembling tobacco smoke) from the ammonia discharge gas. To keep lubricant carryover to a minimum, the selection for mineral lubricants should have a vapor pressure of at least 7.7×10^{-5} mm of mercury at 150°F. This ensures around 2 ppm carryover provided the manufacturer of the separator supplied an efficient coalescer system.

Coalescing separators are almost this efficient for reciprocating compressors. However, the carryover will be slightly higher because of the much higher discharge temperature, which will increase the vapor pressure.

4. **Is it hygroscopic?** Some lubricants such as esters are hygroscopic. They attract moisture from the atmosphere. Normally this lubricant type is not used in ammonia refrigeration systems. Hygroscopic lubricants should not be considered.

5. **Is it soluble with ammonia at all operating temperatures and pressures?** Mineral lubricants are only slightly soluble in ammonia. At 95°F (35°C) the solubility is 170 ppm and at -30°F (-34°C) it is only 10 ppm. This accounts for the lubricant accumulating at the lowest temperature component of the refrigera-

tion system. Polyalphaolefin (PAO) lubricant, another lubricant that is excellent with ammonia refrigeration systems, has similar solubility characteristics. This property of ammonia refrigerants dictates the piping and equipment locations of the system.

6. **What is the pour point of the lubricant?** The pour point is important if the system operates at temperatures below -20°F (-29°F). It is best to have a lubricant with a pour point below the operating suction temperature. This pour point is controlled by the amount of wax left in the lubricant after it is dewaxed at the refinery. Many lubricants that indicate a -30°F (-34°C) pour

point do not meet these criteria when subjected to the ASTM test, so ask for ASTM test results. Mineral lubricants are available that have undergone rigid processing (hydrotreated), have minimal wax content, and low pour points. The PAO lubricants have a pour point that approaches -100°F (-73°C).

7. **Does it have the proper additives to make it compatible with the system?** A reliable ammonia compressor mineral lubricant will have several approved additives, which provide better lubrication and system performance.

a. **Defoaming.** The screw compressor lubricant separator is designed to separate the lubricant that is injected into the compressor during operation from the ammonia gas. Even with a change of direction when the lubricant falls into the sump, some ammonia will remain in the lubricant. It will gradually separate. However, this tends to cause foaming. A defoaming agent specifically designed for this duty is a must.

b. **Pour point extension.** If the system is to operate at low temperatures, the lubricant should have a pour point depressant additive. Without the additive, wax will separate from the lubricant and "foul" the evaporators and the control valves. The evaporators then lose efficiency as does the entire system. A mere 1.5 mils of wax and lubricant coating an evaporator surface can cause a 23% loss in the heat transfer rate.

c. **Oxidation inhibitor.** An oxidation inhibitor also is helpful in ammonia refrigeration systems. Where the system operates in a vacuum, air often is drawn into the system along with the moisture it contains. Maintenance procedures that require opening the system often introduce air into the system. Moisture air and any gases that might be generated by the breakdown of the ammonia or be a part of the plant charge can cause corrosion. This antioxidant is an excellent additive.



Briley

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●



Blast Freezing

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Blast freezers are available in two types: static and dynamic. Static freezers are arranged to freeze boxed product on pallets or on a rack with wheels. The dynamic freezer conveys racks or pallets through the freezing tunnel on hydraulically driven conveyors at a predetermined rate. Each type has similar design criteria. ♦ Many boxed products are frozen in static blast freezers (sometimes called batch freezers). When considering installed cost only, blast

freezers are the least expensive way to provide a quality product. Unfortunately, many so-called blast freezers are improperly designed and operated. Most of the operation problems are related to improper positioning of the pallet or cart in the freezer.

Static blast freezers (*Figure 1*) must be loaded one row at a time to ensure proper airflow over the product. In general, pallets/carts should not be placed more than four deep in the direction of the airflow because the temperature rise is too high to freeze the fifth row in the allotted time.

The most efficient blast freezers have three rows. *Figure 2* shows a typical blast freezer layout.

When designing a blast freezer, the freezing time must be known. This is usually determined through a test that also should determine the required air velocity. Air velocities around 1,500 fpm (8 m/s) are common. The design should ensure that no air bypasses the air unit(s). Pallets/carts should be arranged so that the air passes over the packaged product along the long side of the box.

The pallet should be stripped. Several dividers made of wood (1 in. × 1 in. × length of pallet [25 mm × 25 mm × length]) are placed between the layers of product. This allows air to be directed over each layer. Dividers made of aluminum decrease freezing time. Separating the packages using egg crate plastic sheets increases the freezing time.

Static blast freezers are manual and require properly trained operators. These versatile freezers can accommodate various sizes of product. However, they require a large amount of space, are labor intensive and difficult to keep clean.

The dynamic blast freezer (sometimes called a tunnel freezer) is semi-automatic. The carts/pallets are loaded on conveyors that transport them through a controlled air blast at temperatures that can approach -40°F (-40°C). Multiple products can be frozen or cooled when two or three conveyors are used because each conveyor can be operated at a different speed.

The dynamic blast freezer requires less labor and floor space than the static blast freezer. For the best performance, refrigerated anterooms are required at the inlet and outlet to prevent excess coil frosting and defrosting.

Blast freezing has been used to freeze "open" product on racks. However, this is not recommended because of sanitation problems and excessive product shrinkage. Whole fish often

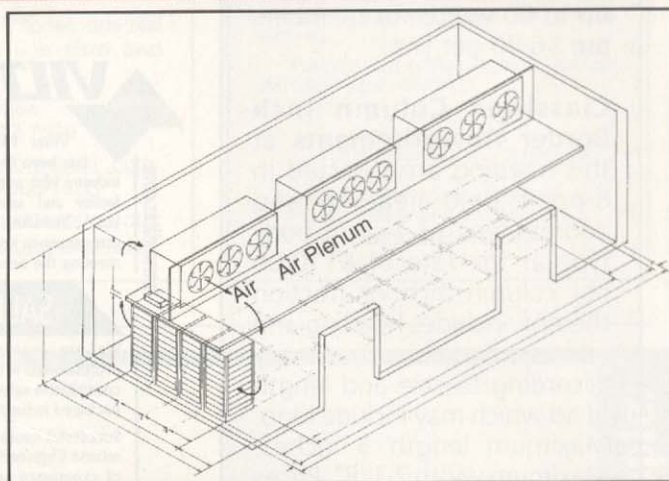


Figure 1: Static blast freezer.

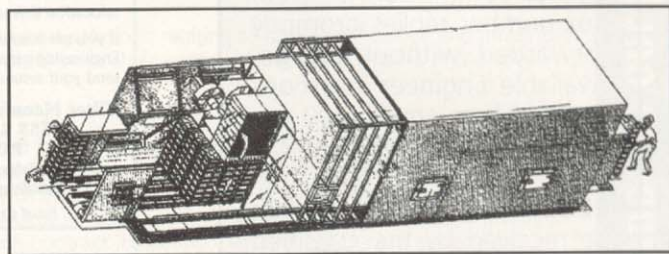


Figure 2: Dynamic blast freezer.

are frozen in blast freezers — particularly tuna and similar fish.

Operating blast freezers for large quantities of product requires good planning and using several blast cells. Cells are loaded one at a time and the freezing cycle started. Most blast systems operate on a 24- or 48-hour cycle (except for three-shift operations).

Freezing time for any product should be verified before designing a blast freezer. The dynamic system usually provides faster and continuous freezing, which is highly desirable in a multi-shift operation.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ♦

CO₂ Making Comeback

By George C. Briley, P.E., Fellow/Life Member ASHRAE

“What goes around comes around” is particularly true for carbon dioxide (CO₂) as a refrigerant. In the late 1800s, CO₂ was used as a refrigerant in installations including chilling water and brines. It also was used for freezing and maintaining frozen products such as fish and meat. Manufacturers converted standard ammonia compressors to the high pressures required to condense CO₂ (ranging from 800 to 1,000 psi [5516 to 6895 kPa]). Of course, this limits the CO₂ system condensing temperature to 80°F or 85°F (27°C or 29°C), which is difficult to accomplish.

Although CO₂ was still used as a refrigerant into the late 1940s, it was generally replaced by ammonia and halocarbons, which first appeared in the 1930s. Carbon dioxide's efficiency was not as good as other refrigerants, such as ammonia or the halocarbons.

The Frick Company designed and installed a system in the mid-1930s that used CO₂ as a cascade refrigerant, which kept the system pressures within the norm of standard ammonia refrigeration equipment. This system was designed to freeze and store frozen food. Using CO₂ as the cascade refrigerant produces suction temperatures down to -65°F (-54°C). Normally, the CO₂ would be condensed between -5°F and -10°F (-21°C and -23°C) around 250 psi (1724 kPa).

At the time of this installation, the frozen food industry was in its infancy. The concept did not survive even though it was equal in efficiency to a two-stage ammonia refrigeration system. Other factors in this system's demise include halocarbon refrigerants that were storming the market, and the beginning of World War II.

After the end of the war, the frozen food industry began to move forward rapidly and a variety of freezing systems appeared. Vegetables and fruits were frozen in small boxes that resulted in a frozen block of product. A type of crude blast freezer usually was used at 0°F to 5°F (-18°C to -15°C). This was accomplished at suction temperatures from -25°F to -35°F (-32°C to -37°C), most systems being two-stage ammonia. At the time, this satisfied the housewife and provided good quality as it was measured then.

In the mid-1960s individually quick frozen (IQF) freezers were introduced by the Lewis Refrigeration Co. This increased the quality of frozen vegetables and fruit. For the sake of system efficiency, suction temperatures were kept at -30°F (-34°C) or higher (usually employing two-stage ammonia systems).

As the food freezing industry progressed, processors found that lower suction temperatures provided a higher quality product. By lowering the evaporator temperature and, thus, the freezing time, there was an initial cost increase. However, the additional gain in quality allowed processors to charge a bit more. Evaporator temperatures were lowered to around -50°F (-46°C). Then, the ozone depletion theory was put forward along with rules and regulations that banned the chlorine in refrigerants and other gasses. This caused the price of refrigerants used in low temperature refrigeration systems to increase to \$20 to \$30 a pound (\$44.09 to \$66.14 per kilogram).

Two-stage ammonia refrigeration systems can produce -50°F (-46°C) evaporator temperatures. However, the ammonia suction volume is very large. During this time, many industrial and chemical plant users had existing systems with evaporator temperatures in the -65°F to -35°F (-54°C to -37°C) range that were using halocarbons in a cascade system. Many of the systems were converted from halocarbons to CO₂. In general, the compression equipment was quite a bit smaller.

CO₂ is gradually becoming the accepted cascade refrigerant in food freezing plants, freeze dry plants and others that require temperatures around -65°F (-54°C).

Why? Because it is inexpensive (around \$0.05 per pound [\$0.11 per kilogram]), the systems are efficient, and compressors are usually about 16% to 20% smaller than an ammonia compressor.

CO₂ is a global warming gas. However, CO₂ suppliers obtain the gas from chemical processes that normally vent the gas to the atmosphere. Thus, the CO₂ that is used as a refrigerant does not add to the CO₂ reaching the atmosphere. It is contained and used in a productive manner.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●



Controlled Atmosphere Storage Of Fruits and Vegetables

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Fruits and vegetables have individual requirements for the amount of oxygen (O_2) and carbon dioxide (CO_2) necessary to stay fresh. Controlled atmosphere (CA) storage provides the precise environmental conditions needed for quality preservation. Air is comprised of about 21% oxygen, 78% nitrogen and other gases. This atmosphere, even when chilled to just above freezing, is not conducive to long-term storage of fruit and vegetables.

Fruits and vegetables are living, breathing organisms that will mature and die within a few days at temperatures above 40°F (4°C). Thus, a process was developed to extend the "life" of high quality fruits and vegetables.

Fruits and vegetables breathe much like humans—in a process called respiration. The produce consumes O_2 from the surrounding air and gives off CO_2 , along with water vapor and some heat (Btus). By lowering the amount of oxygen in a sealed enclosure, the rate of respiration of the produce can be retarded. Thus, the product is held in a state of suspended animation.

Each type of fruit or vegetable has a specific amount of O_2 and CO_2 that will keep it in suspension. An O_2 range of 1% to 5% is normal. Too little O_2 may cause problems. CO_2 quantities can range up to 12% CO_2 , but usually are 3%. The key to quality CA storage is maintaining the proper percentages of both gases. Most state agricultural departments can provide data regarding a specific fruit and a specific variety of that fruit. For example, almost every apple variety requires a different mix of O_2 and CO_2 . The owner must understand, however, that CA storage cannot *improve* product quality. Rather, it ensures that the product will retain most of its original quality while in storage.

Three factors govern quality storage: a tightly sealed room, a well-engineered refrigeration system and an atmosphere generator.

First, the room must be tightly sealed so there is no infiltration of ambient air, and no exfiltration except that generated by ambient conditions or defrosting of the evaporators. When defrosting occurs, the air surrounding the evaporator is warmer, which increases air volume and room pressure. Special seals are necessary (usually static water traps) for exhausting excessive room pressure.

For the storage room, galvanized or aluminum cladding with joints caulked and taped works well. Doors should have double seals. Portholes are helpful for observing the product. Insulated panel construction is a must.

The second factor is a properly sized refrigeration system that uses R-22 or R-717. Although some CA operators dislike

ammonia, it is used safely in many operations (provided adequate ammonia detection is installed).

Refrigeration systems should be engineered to rapidly cool the product from the field temperature to 29°F (−2°C), depending on the product. The control system should be designed to reduce the coil split to 2°F to 3°F (1.1°C to 1.7°C) during the holding period to ensure high room humidity. This helps to ensure that the product does not lose moisture and weight, as well as quality and value. Good air circulation also is a must. In many cases, humidity levels of 95% to 98% are suggested.

The system should also include an ORSAT analysis to verify the CO_2 and O_2 content of the storage room. Today, automatic gas measurement equipment is available. A CO_2 scrubber is necessary to quickly establish the desired room CO_2 conditions. Automatic CO_2 scrubbers are available that use various processes including activated carbon, molecular sieves and caustic soda. Automatic O_2 generators also are available. Dual-purpose generators that can control CO_2 and O_2 in a CA room are called atmosphere generators.

The atmosphere generator is the third factor for quality storage. It provides the desired CA room conditions in a short time—not depending upon the product to create its own atmosphere. This generator allows the CA storage to store some short-time products, such as strawberries, corn and tomatoes. For long-term storage of apples, pears, etc., the atmosphere generator makes it possible to vent a room and remove some of the fruit for marketing or maintenance and then promptly returns the CA room to the desired atmosphere.

Some other considerations for a well-operated CA storage room include:

- The temperature must be monitored continuously in several areas to ensure proper air circulation.
- A refrigerant leak detector (R-717 or R-22) should be designed to alarm at several places, including the manager's office and home.
- Continuous monitoring of CO_2 and O_2 is essential.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

Efficiency for R-717 and R-22 Systems

By George C. Briley, P.E., Fellow/Life Member ASHRAE

This article offers suggestions to increase the operating efficiency of R-717 (ammonia) and R-22 industrial refrigeration systems. Not all of these suggestions apply to every plant. Thus, the owner must analyze each suggestion to decide which are suitable for his plant.

Raise the System Suction Pressure

As ambient weather conditions are lowered, the refrigeration plant's load rapidly decreases. If this occurs, one way to save energy is to gradually raise the suction pressure. For example, if the suction pressure is raised from -20°F to -10°F (-29°C to -23°C), the operating cost savings of a two-stage system is approximately 8%. If the plant has blast freezers, raising the suction pressure from -40°F to -30°F (-40°C to -34°C) results in 12% in operating cost savings.

In new plants, system components should be sized so that the plant can operate at the optimum suction pressure at all times. Raising and/or lowering suction pressure should be done one degree at a time. I suggest raising the suction pressure until the temperature in the cold space starts to increase, and then lower the temperature one degree.

Raise the System Intermediate Pressure

Many older refrigeration systems have had rotary compressors as low-stage or boosters with the intermediate system pressure dictated by the compressor's compression ratio limitations. Many older R-717 plants were operated at 20 psig (5°F) (138 kPa [-15°C]) because of this limitation. When the rotary compressors were replaced (and many of them have been), the refrigeration plant operators continued to operate as before—at an intermediate pressure. This is inefficient and in most cases unnecessary.

With reciprocating or screw compressors as low-stage or booster compressors, the intermediate pressure should be adjusted to approach 25°F (39 psig) (-4°C [269 kPa]) where refrigeration requirements are in the 35°F to 45°F (2°C to 7°C). If there are no intermediate refrigeration requirements at these temperatures, then the intermediate pressure can be raised even further. This will reduce the total kW per TR. The optimum intermediate pressure equalizes the absolute compression ratio between the two stages. For optimum efficiency in a two-stage refrigeration system, the low-stage discharge gas should be cooled to the intermediate temperature and the high-pressure liquid cooled to the same or nearly same temperature. One way to decrease the desuperheating load from the low stage is to cool the discharge gas in a heat exchanger cooled with thermosyphon liquid, thus removing heat from the high-stage system and exhausting it to the atmosphere then the condenser.

Use Indirect Lubricant Coolers

Injection cooling of screw compressors is inefficient and can decrease system efficiency in single-stage applications by as much as 10% or 12%. If the screw compressor is operated economized with injection cooling, the system efficiency is reduced even further to around 25% to 30% depending upon the suction pressure. Therefore, it benefits the owner to use some type of indirect cooling. Some packagers also have found a direct relationship between injection cooling and excessive bearing failures and a shortening in the time between overhauls. These failures occur over a rather long time (usually outside the warranty).

When a low-stage screw compressor is injection-cooled, it will prove to be highly inefficient because all of the motor heat is transferred to the high-stage compressors. With indirect lubricant cooling, a large percentage of the motor horsepower can be removed from the gas stream—sometimes up to about 40% to 50%. As an example, a 200 hp (150 kW) low-stage compressor that is injection cooled adds up to 20 tons (70 kW) more duty and more than 35 hp (26 kW) to the high-stage system. That is a lot of extra energy consumption.

An indirect screw compressor lubricant cooling system consists of a shell-and-tube or a plate-and-frame exchanger that is cooled by cooling tower water, glycol cooled in a fluid cooler or a section of an evaporative condenser or by a thermosyphon system that uses the refrigerant at condensing temperature. Cooling tower water is not recommended because of the fouling and trash collection that can occur with an oil cooler near the floor and at the low point of the water system. A fluid cooler to cool the glycol is a good way to cool screw compressor lubricant. However, it requires a pump and the accompanying maintenance. By using a thermosyphon system where refrigerant at condensing pressure is directed to the heat exchanger, there is no need for a pump and the heat exchanger will not require cleaning or much maintenance.

With thermosyphon cooling, the liquid refrigerant at condensing pressure/temperature boils in the tubes or circuits of the heat exchanger and the resulting vapor is directed to the condenser, where it is condensed.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

Efficiency for R-717 and R-22 Systems, Part 2

By George C. Briley, P.E., Fellow/Life Member ASHRAE

This article offers more suggestions to increase the operating efficiency of R-717 (ammonia) and R-22 industrial refrigeration systems. Not every suggestion applies to every plant. The owner must decide which suggestions best suit the plant.

Mechanically Pumped Liquid Recirculation System

Most industrial refrigeration systems use a liquid recirculation system to distribute refrigerant to the evaporators. These systems are referred to as liquid overfeed systems. The two systems used for distributing refrigerant throughout the plant are a mechanically pumped system and a refrigerant gas pumped system.

Some years ago, Gustav Lorentzen of Norway published an astute paper regarding these two methods. The study describes tests he performed in his laboratory under ideal conditions. His findings indicated that gas pumped systems were 4% to 15% less efficient than mechanically pumped systems. Unfortunately, field applications of gas pumped systems are far from ideal. Systems must be properly adjusted and maintained to even approach these efficiencies.

Most evaporators (air units) are designed for an overfeed ratio of 3 to 1. The gas pumped systems must be kept at no more than 2 to 1 for the system to operate at its most efficient point. Therefore, the evaporators must be circuited properly. Mechanically pumped systems can provide much higher pumping ratios resulting in more efficient evaporators. When an R-717 system with a capacity of 500 tons (1760 kW) operates at 0°F (-18°C) (15 psig [103 kPa]) and 96°F (36°C) (185 psig [1275 kPa]) and then a gas pumper is used, the losses approach 33 bhp (25 kW) (I believe that it would be more). This same system would use, at the maximum, a 5 hp (3.7 kW) liquid ammonia pump. Besides being more efficient and more readily maintained, there exists a sizeable savings in electrical energy. I have been involved in revising several refrigeration systems where replacing a gas pumping system with a mechanical pumping system resulted in energy savings approaching 25%.

If a refrigeration system is short of capacity and has a gas pumping system, consider replacing it with a mechanical pumping system to reduce energy cost and increase system capacity.

Reclaim Heat From the Refrigeration System

It is often difficult to justify adding heat reclaim to most industrial refrigeration systems. Attempting to recover heat from the discharge gas can be a poor investment in terms of capital cost. Many heat recovery systems installed with great expectations resulted in no savings. Many times the system had excessive pressure drop, which penalized the compressors by increasing horsepower requirements. Also, the energy usage for the compressors was increased more than the savings in heating water. There are several methods for refrigeration plants to use heat recovery and save energy.

One way that cold storage facilities save energy is by heating the glycol that is pumped under the floors of freezers to prevent the floors from heaving. This is usually a small load. However, I find that most engineers install the glycol heater on the high-stage compressor discharge instead of on the low-stage (booster) compressor discharge. When installed on the low-stage discharge, the heat used to warm the floor is subtracted from the high-stage compressor duty. This results in an energy savings larger than might be expected when removing heat from the high-stage discharge and the condensers.

Another excellent application for heat recovery is heating boiler feed water. Many plants use water from wells with water in the 50°F to 60°F (10°C to 16°C) range. By using a shell-and-tube condenser in parallel with evaporative condensers, the water can be heated to temperatures approaching 95°F (35°C) in the warm months. However, in winter it is more energy efficient to reduce the condensing temperature/pressure. Therefore, when condensing at 60°F or 70°F (16°C to 21°C), only a small amount of heat is added to the water.

George C. Briley, PE., is president of Technicold Services, San Antonio. ●

Efficiency for R-717 And R-22 Systems, Part 3

By George C. Briley, P.E., Fellow/Life Member ASHRAE

This article offers more suggestions to increase the operating efficiency of R-717 (ammonia) and R-22 industrial refrigeration systems. Not every suggestion applies to every plant. The owner must decide which suggestions best suit the plant.

Helical screw compressors require large lubricant coolers because the lubricant absorbs much of the motor heat when it is injected into the compressor. This motor heat must be removed from the lubricant before it is reinjected into the compressor. Methods for cooling the lubricant include direct injection of refrigerant into the compressor housing, injection of refrigerant using a pump, and indirect cooling by water, air, or refrigerant.

Let us analyze the various ways of cooling by discussing direct injection into the compressor housing. High-pressure refrigerant from the condenser is metered into the compressor housing to maintain a discharge temperature of about 120°F (49°C). This system actually reduces the compressor efficiency by as much as 10% if the system is operating at a high compression ratio. At lower compression ratios the losses are in the 4% to 5% range. This is a high price to pay when an indirect system does not reduce the efficiency at all. Besides being inefficient, several screw compressors manufacturers indicate that injection cooling reduces the mean time between failures of both main and thrust bearings. My personal experience with this type of lubricant cooling confirmed this. This type of system also is difficult to keep adjusted.

Another type of injection cooling uses a pump to inject high-pressure refrigerant into the compressor discharge. This eliminates the losses. However, if the injection rate is not precise, the pump eventually will over inject. Over injection causes the lubricant in the separator to foam violently. This dilutes the lubricant, which causes bearings to fail and causes the lube pump to cavitate and shut down the compressor. The same problem also exists with the direct injection system.

The best way to cool the lubricant is with an indirect system. This consists of a shell-and-tube or plate and frame heat exchanger. The lubricant is pumped through the heat exchanger where it is exchanged with the high-pressure refrigerant. Since the lubricant only needs to be cooled to about 120°F (49°C), and the refrigerant from the condenser is at a boiling point of 95°F to 105°F (35°C to 40°C), the lubricant can be cooled readily. This thermosyphon system does not require pumps and affords maximum screw compressor efficiency.

In a two-stage system where injection cooling is used on the booster (low stage) compressors, the losses are more than for the high stage. By using indirect cooling, some $\pm 50\%$ of the motor horsepower can be removed, thus reducing the high-stage refrigeration requirements. For a 200 hp (149 kW) low-stage compressor fully loaded with indirect lubricant cooling, the savings in high stage brake horsepower is 30 to 35 (22 to 26 kW).

Indirect lubricant cooling using condenser water works fine. However, the tube-cleaning problem can be difficult. Most lube coolers are on or near the floor at the lowest point in the cooling water system. The coolers become a collection point for all the trash in the system.

Another way to cool the lubricant is to use a glycol solution that is cooled by a separate fluid cooler or a section of the evaporative condenser. This system is a good approach in some installations where the injection system is replaced by an indirect system.

An air-cooled lubricant cooler is not recommended.

High-Efficiency Motors

As the price of electricity has increased, engineers have looked to motor drivers as a place to increase efficiency of the electrical system. Some states have passed laws requiring all electric motors to be rated "high efficiency." When analyzing small motor (i.e., NEMA) sizes, many older motors have low efficiencies. A cost analysis normally indicates that high-efficiency motors are a good investment.

However, when analyzing larger motors (i.e., two-pole screw compressor motors), high efficiency may not save money. Most two-pole motors on screw compressors have efficiencies approaching or exceeding 95%. A high-efficiency two-pole motor may have an efficiency of 96.5%, which means that usually the extra investment cannot be justified. To increase efficiency, power factor correction may be a better investment depending on the electrical energy supplier's pricing.

George C. Briley, P.E., is president of Technicold Services, San Antonio.

Efficiency for R-717 And R-22 Systems, Part 4

By George C. Briley, P.E., Fellow/Life Member ASHRAE

This article offers more suggestions to increase the operating efficiency of R-717 (ammonia) and R-22 industrial refrigeration systems. Not every suggestion applies to every plant. The owner must decide which suggestions best suit the plant.

Although demand defrost has been used for many years, many -10°F (-23°C) cold storage warehouses and continuous freezers still use time clocks or the number of liquid solenoid openings to initiate the defrost cycle. Only defrosting when needed requires considerable operator attention. Unfortunately, many operators set the time clock once and hope for the best.

Cold storage warehouses locate air-handling units (AHUs) in various locations, some close to a door that opens often, and some in the back of the room. The operator usually sets the time clock that initiates the defrost for both units to defrost the same number of times each day. In many cases the AHUs in the back of the warehouse only need defrosting about one-third of the number of times that the AHUs at the door require.

When defrosting, the refrigeration system must provide two to three times the refrigeration for which the coil is designed. Thus, a 20 ton (70 kW) coil requires up to 60 tons (210 kW) for the 10 to 15 minutes that the coil is in defrost. If the coil takes more than 15 minutes, problems exist. Demand defrost can solve some of these difficulties. One reason that coils take a long time to defrost is that the condensed liquid formed during defrost is directed into the wet liquid overfeed return line, which is at the plant suction pressure.

A better way to defrost is to use an air pressure switch with a range of 0.5 to 1.5 in. w.g (124 to 373 Pa). The switch should be rated for low temperature operation down to -50°F (-45°C). In most cases the pressure switch can be set at around 1 in. w.g (249 Pa). However, this depends on the coil design and the fan type. This pressure switch actuates the defrost cycle, which means the AHU only defrosts when the coil is sufficiently frosted to actuate the switch.

For more energy savings, arrange the defrost system so that the liquid refrigerant formed during defrost is drained in a separate pipe system through a float valve to the high temperature recirculator or the intercooler. This helps hasten defrost and clears the coil faster than the older method.

Demand defrost has several advantages including that defrost only occurs when necessary, the operator only needs to check periodically that the defrost is working, the variations in summer to winter climates are automated and installing the defrost drainer system decreases the defrost time.

As an example of demand defrost system efficiency, I was involved in designing such a system for a 125,000 ft² (11 600 m²) cold storage warehouse with a time clock defrost system that was not timed correctly. The three coils in each of eight rooms were located on the long wall. The AHU close to the door frosted quite a bit because of its proximity to the often-used door. All AHUs were timed in the same manner as the first unit. The owner installed the demand defrost system using the pressure switch on all units and installed the defrost drain to the intercooler.

The plant electric consumption decreased by about 23% (much to the delight of the owner). Obviously, if the operators set the defrost times exactly right, the demand defrost system is not needed. This additional cost is about the same as the salary of a good operator for one year. This emphasizes the advantages of automation and good engineering practice.

For industrial refrigeration systems that use thermosyphon lubricant cooling, one way to decrease the refrigeration load going to the high-stage compressors is to install a thermosyphon cooled desuperheater for the booster compressors.

Some 50 years ago desuperheaters were used to cool the booster discharge gas to reduce the high-stage load. However, they were cooled with tower water that had to be drained each time the system shut down to prevent freezing as the system equalized. I'm sure that is why ammonia is the coolant. If the high stage in your plant is a bit short, adding a desuperheater may help.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

Efficiency for R-717 And R-22 Systems, Part 5

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Effective routine maintenance increases your refrigeration system's efficiency and minimizes downtime. Preventative or predictive maintenance reduces the overtime hours that technicians spend correcting problems in plants that adhere to the old axiom "if it ain't broke, don't mess with it." So, what are some of the maintenance items that keep plant efficiency at its peak?

In an earlier article we discussed operating the system at the lowest possible condensing pressure. Condensers tend to be fouled by various elements that are in the water in most places in the United States. The only sure way to minimize this fouling is a good water treatment system that is kept in proper working condition. Even with the best treatment system the condenser coils should be inspected thoroughly every month to ensure that the inner portion of the coils are clean. The cleaner the condenser coils, the lower the condensing temperature and the lower the energy costs.

Reciprocating compressors must be kept in good repair. The suction and discharge valves should be inspected at least once a year for breakage. Valves are the most common items that fail in this type of compressor and actually require continuous attention. A broken valve may break into small parts and cause damage to the pistons and rods and bearings in the compressor.

Not many rotary booster compressors are in use today. However, for those users still operating these machines, the blades should be inspected at least once a year. Worn blades decrease compressor efficiency that, at best, is up to 17% less efficient than a rotary screw compressor. Most users have replaced this type of compressor due to its inefficiency.

For screw compressors the capacity reduction system should be monitored continuously. Systems using solenoid-controlled liquid level systems on their recirculators often cause unnecessary variations in suction pressure that cause the screw compressor capacity control system to cycle too often. This causes wear on the screw compressor slide valve piston operator. Thrust bearings and motor/compressor alignment should be checked frequently. If the lubricant coolers use water, the tubes should be cleaned at least once a year. For injection-cooled systems, the control valve system should maintain 120°F (49°C) on a constant basis—never below.

Lubricant that escapes from the compressor system must be removed to keep the heat exchangers operating at their best.

For R-717 systems most plants have oil pots at the low points and the lubricant can be drained as needed. For R-22 systems most have ways to siphon the lubricant into an oil still and separate the lubricant from the R-22. Coalescing lubricant separators have changed the lubricant management situation in all refrigeration systems. They can be engineered to reduce lubricant carryover to less than 2 ppm by weight. To obtain this low carryover a low vapor pressure lubricant must be used. These lubricants cost a bit more than the older ones. However, the small difference in price is more than worth it when you consider the cost to drain the lubricant from the oil pots and oil stills. Lubricant recovery systems should be arranged so they can be drained when the system is in operation.

Automate the System

There is no substitute for well-trained technicians for operating a large refrigeration system. However, they can be much more effective when the system is automated. With the availability of microprocessor and programmable devices a refrigeration system can be fully automated (although technicians are necessary to keep the controls in proper repair and adjustment). In fact, an automated system can be much more efficient in maintaining predetermined conditions than a human. Automation has been perfected to the point that many large refrigeration systems have technicians available only on a single shift. If a problem exists, the control system calls a certain person or group of people to report the emergency so that it can be corrected.

Automation is not a cure-all. Technicians are necessary to monitor the system. As an example the defrost system for cold storage rooms must be checked on a daily basis to make sure the program is working, that the coils are free of frost, and that the drain is in order.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

Energy Conservation In Industrial Refrigeration Systems

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Last month, we discussed operating the refrigeration system at the lowest possible condensing pressure/temperature. Here is a way to ensure that the system “automatically” adjusts to the lowest “economical” condensing pressure. Control the condensing pressure in accord with the wet-bulb temperature.

Many ways exist to control the condensing pressure at a fixed minimum. However, this is not the optimum approach. Why should we control the pressure in accordance with the wet bulb?

Envision an equipment room with several large screw compressors that operate during production and a small 75 hp (56 kW) compressor that operates on nights and weekends to maintain a holding freezer or cooler. The condensing pressure is set for 100 psig (690 kPa) (63°F [17°C]) and the wet bulb is 80°F (27°C). Thus, the evaporative condenser's fans and pumps are operating full time—all 125 hp (93 kW) of them. They cannot attain the 63°F (17°C) condition because the condensers cannot attain a condensing temperature much below about 85°F. If a microprocessor was installed to maintain a condensing pressure at, for example, 10°F (5.5°C) above wet bulb, then the system would operate at 90°F (32°C), and the fans and pumps would cycle off, saving energy.

The typical microprocessor-based evaporative condenser control system measures the temperatures and pressures in the condensing system using accurate analog sensors. The measurements made by the controller include dry-bulb temperature, wet-bulb temperature, condensing temperature and condensing pressure. The controller is arranged to cycle the water pumps and the condenser fans to achieve maximum efficiency. The controller will turn on the condenser water pumps and fans until the desired range between the wet bulb and the

condensing temperature are attained. This number can normally be adjusted to between 5°F and 15°F (3°C and 8°C). As the wet bulb changes, so does the condensing temperature.

An additional controller will increase the condensing temperature if air or other non-condensibles are in the system. An ammonia algorithm is installed in the controller to provide the correct pressure/temperature relationship. To identify non-condensibles, the system measures the condensing pressure and the liquid temperature exiting the condensers. If the pressure is greater than it should be by 3 to 5 psi (21 to 34 kPa),

then it turns on the purge unit. If the purge unit does not perform, it alarms. If no purge unit is installed it alarms, so that the system can be purged at the operator's convenience.

Most of these systems record outdoor temperature. Thus, they can be programmed to turn off condenser water pumps at a predetermined ambient dry bulb or turn on condenser pan heaters.

‘Many ways exist to control the condensing pressure at a fixed minimum. However, this is not the optimum approach. Why should we control the pressure in accordance with the wet bulb?’

System Analysis for Two Cities

As an example, a system has three 300 hp (224 kW) compressors, 60 hp (45 kW) of condenser fans and 15 hp (8 kW) of condenser water pumps, plus a 75 hp (56 kW) compressor to be used when the large compressors are off for nighttime and weekend duty. The ammonia condensers are designed for 95°F (35°C) saturated condensing temperature. The large compressors operate at 10°F (–12°C) suction. At 181.1 psig (1250 kPa) (95°F [35°C]) condensing temperature, the compressor bhp is

	Portland, Ore.	Memphis, Tenn.
Design Condensing Temperature	95°F	95°F
Design Wet-Bulb Temperature	69°F	80°F
Bhp Without Wet-Bulb Control (8,700 hrs)	5,752,000	5,752,000
Condenser Pressure Control	175 psig	175 psig
Bhp With WB Control (8,700 hrs)	4,417,000	4,234,000
Savings in Bhp	1,342,000	1,518,000
\$ Savings With \$0.07/kW Electricity	\$93,910	\$106,000

Table 1: Comparing plants in two U.S. cities.

302 (225 kW). At 126 psig (870 kPa) the bhp is 229 (170 kW) and at 103 psig (710 kPa) condensing the bhp is 200 (149 kW). Compare the system operating at a set condensing pressure of 175 psig (1205 kPa) and one with a microprocessor controller programmed to control based on wet-bulb temperature. The differential between wet-bulb temperature and condensing temperature is set at 15°F (8°C), which is high.

Few plants operate 8,700 hours a year. However, the following comparison is made so that the calculations can be modified for fewer operating hours. *Table 1* shows data for two cities that have highly varying wet-bulb temperatures. Only the compressor's energy consumption was considered.

If you assume the plant operates only 2,000 hours at full load, the savings in Memphis are about \$24,000, and the wet-bulb microcontroller can be paid out in less than a year. This analysis does not take into consideration the savings from the reduction in the use of the condenser fans and pumps.

Some evaporative condenser control systems use a thermostat in the condenser water plan in lieu of the wet-bulb instrument. This is fine for one condenser, but it is not too reliable in multiple condenser installations.

In conclusion, wet-bulb evaporator condenser controls are a major item for reducing energy consumption in industrial refrigeration systems.

George C. Briley, PE., is president of Technicold Services, San Antonio. ●



Hydrocooling Fruits and Vegetables

By George C. Briley, P.E., Fellow/Life Member ASHRAE

To prevent recently harvested fruits and vegetables from wilting, shrinking and losing flavor, hydrocooling is used for initial precooling. The boxed product is placed on a conveyor that pushes the boxes through a cooling tunnel.

Large quantities of chilled water are sprayed directly onto the tops of the boxes. Water flows down through the product and is collected in a tank or sump underneath the tunnel. The refrigerated spray often is chlorinated. Sodium

hypochloride may be added to destroy the rot spores on the skins of fruit and vegetables, which improves storage life and market quality.

Ice or a mechanically refrigerated coil is placed on a raceway along the tunnel. Large quantities of water are pumped over the ice/coil. Most designs are based on a water differential of 2°F or 3°F (1°C or 1.5°C) over the product to be cooled.

As much as 15 gpm (0.95 L/s) of spray water is used for every bushel handled per hour. It takes 20 to 25 minutes for the product to travel through the tunnel with a temperature reduction of 35°F to 45°F (1.7°C to 7°C), depending on the product and spray water temperature. Water temperature normally is 33°F to 34°F (0.56°C to 1.1°C).

The water is drawn from the sump or tank underneath the tunnel, passed over the ice (or coils), and then pumped to the spray nozzles or distribution tray at the top of the tunnel (Figure 1).

Hydrocooling is used to remove field heat from sweet corn, spinach, carrots, apricots, green onions, asparagus, and many other fruits and vegetables.

Hydrocooling also is used for frozen food processing. Most vegetables are blanched, i.e., heated to about 180°F (82°C) with hot water (a Birdseye invention) before freezing in an IQF freezer. The product can be cooled to about 100°F (38°C) with plant water. Using hydrocooling with water at 33°F (0.56°C) can lower the product's temperature to about 40°F (4°C). This

has several advantages, including lowering the refrigeration requirement in the individually quick frozen (IQF) tunnel by 20% to 30%, reducing the cost to freeze, and providing a quality frozen product.

The same system often is used to cool fresh processed fruit, vegetables and shellfish to 35°F to 40°F (1.7°C to 4°C). The products are sold in the fresh produce areas of supermarkets.

This type of hydrocooler usually uses a plate-type heat exchanger to cool the recirculated water to 33°F (0.6°C), either placed directly over the belt conveyor or on the process floor near the hydrocooler belt. The plates are refrigerated using R-717 or R-22. Usually, the refrigerant is supplied from the central equipment room (Figure 2).

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

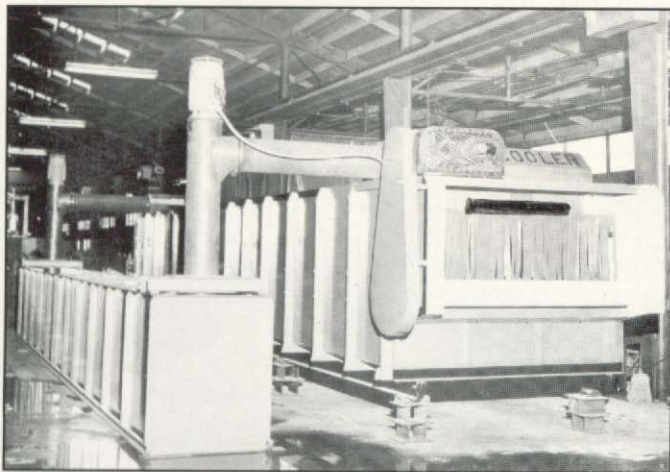


Figure 1: Hydrocooler (ice/coil raceway with pumps at left).

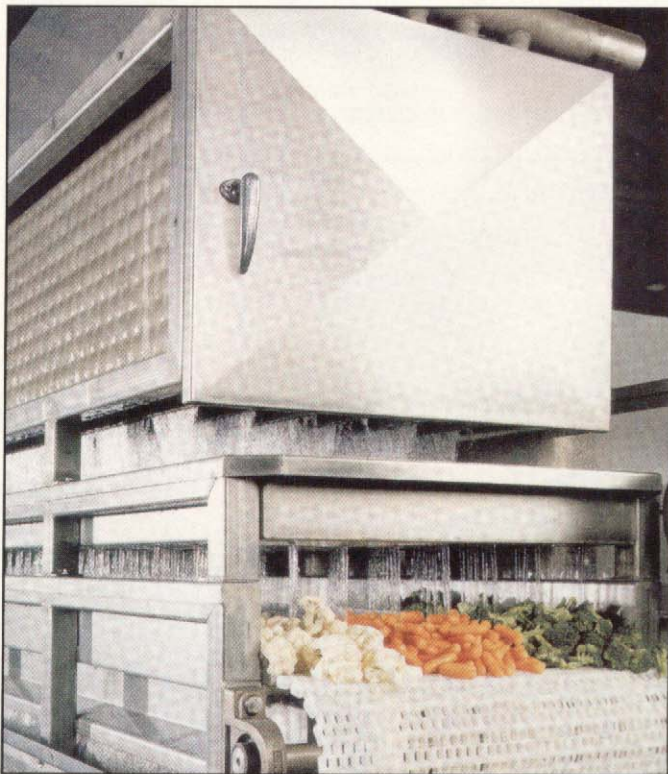


Figure 2: Vegetable processing hydrocooler.

Increasing Operating Efficiency

By George C. Briley, P.E., Fellow/Life Member ASHRAE

In this article, I provide suggestions to increase the operating efficiency of R-717 (ammonia) and R-22 refrigeration systems. They are listed randomly with no intent to make one or the other as the most important. Since industrial refrigeration systems are usually unique, each system requires a thorough analysis to determine which suggestions will provide the most return on investment. Some suggestions do not require any capital expenditure.

1. Operate the system at the lowest possible condensing pressure/temperature.
2. Automatically control condensing pressure in accordance with ambient wet-bulb temperature.
3. Raise the system suction pressure/temperature.
4. Raise the system intermediate or high-stage suction pressure/temperature.
5. If the system uses an overfeed (recirculation) system, use mechanical pumps in lieu of discharge or flash gas pumping systems.
6. Recover as much heat as possible from both the high-stage and the low-stage system—low stage first.
7. Use indirect cooling for helical screw compressors.
8. Use high-efficiency motors.
9. Use *demand* defrost for all evaporators operating below freezing.
10. Automate the refrigeration system.
11. Establish a comprehensive preventative maintenance system.
12. Add a thermosyphon desuper-heater to the low stage of the refrigeration system.

Every industrial refrigeration plant should take full advantage of the ambient conditions. The lower the ambient wet-bulb temperature, the lower the condensing pressure should be. *Figure 1* describes the relationship between compressor brake horsepower (bhp) per ton at various condensing pressures. The savings in operating cost is obvious—the lower the pressure, the lower the energy use. Industrial refrigeration systems may operate at 100 psig condensing pressure if the wet bulb so allows. Operators have many reasons why they cannot operate their systems at low pressures, such as:

- The hot gas piping is too small and I cannot get the defrost system to work properly at low condensing pressures;
- The injection cooled screw compressor will not cool properly if the condensing pressure is too low;
- The plant has ice cream freezers or other evaporators that require high-pressure liquid refrigerant;
- The plant has a hot gas or flash gas pumping system and the recirculation rate cannot be maintained at low pressures; and,
- The plant engineer says refrigeration plants should not

operate at less than 150 psig condensing pressure.

Let's explore each reason. An old saying in the ammonia industry states it is not possible to operate hot gas defrost systems at pressures under 150 psig. This is not correct. Most of the work during the defrost period is accomplished as the warm refrigerant gas is condensed thus if we compare the latent heat of ammonia as an example we find that at 150 psig the latent heat is 493 Btu/lb and at 100 psig it is 515 Btu/lb. The main reason hot gas systems do not work is that the piping is undersized. The designer did not realize that hot gas piping should be sized as if it were a discharge pipe and with a capacity of two to three times the rating of the coil to be defrosted.

Saturated Cond. Temp.	Ammonia		R-22	
	psig	bhp/ton	psig	bhp/ton
105	214	1.58	210	1.76
95	181	1.37	181	1.51
85	151	1.19	155	1.29
75	125	1.00	132	1.08
65	103	0.82	111	0.90

Figure 1: Refrigeration compressor efficiency.

If the piping is too small and the system has several compressors, a way to solve the problem without changing all the piping is to dedicate one compressor to supply hot gas for defrost and maintain the pressure needed for the inadequate piping.

The injection cooled helical screw compressors are not cooled properly when the system condensing pressure is below 135 to 150 psig. Dedicating a compressor will work. *Figure 1* describes the extra energy that the plant requires as the system cannot take advantage of low wet-bulb conditions.

When refrigeration within the plant require high-pressure refrigerant (such as ice cream freezers), often the condensing pressure is held higher than is possible with the ambient wet-bulb temperature. A pump is installed on the high-pressure receiver to supply refrigerant at a specific pressure as required by the plant evaporators.

The plant has a hot gas or flash gas pumping system that will not perform at low discharge or flash gas pressures. Installing a hot gas receiver could solve this problem. However, this type of system can reduce the plant efficiency by 4% to 12%. The best solution is to install a mechanical pump system for the recirculation system. This will allow the condensing pressure to be reduced to 100 psig when taking advantage of the wet-bulb temperature.

I will cover the other suggestions in the next few articles.

George C. Briley, PE., is president of Technicold Services, San Antonio. ●



Industrial Automatic Icemaker

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Industrial icemakers are available in several types. Most are manufactured in sizes from 10 to 50 tons (35 to 176 kW) per day whereas the commercial units vary from 150 lbs to around 6,000 lbs (68 to 2721 kg) per day."

A flake ice machine, introduced in the late 1930s, was probably the first

commercial automatic icemaker.

The ice was frozen on a rotating flexible drum that was immersed in water. The drum was refrigerated with cold brine, and ice was continuously harvested with a cutter bar or a harvesting device. Further development of this system included conversion to a stainless steel drum with R-717 pumped into the drum.

A more recent flake ice design uses a vertical drum with a rotating harvest bar that continuously scrapes the ice off the inner surface of the drum. An annular space outside the freezing surface is flooded with R-717 or HCFC-22 (see Figure 3).

Another flake ice machine design uses a rotating harvest bar on the outside surface of a vertical drum. The annular space inside the drum is refrigerated with R-717 or HCFC-22.

Flake ice is used in the seafood industry to preserve the catch at sea and in the processing plant. It is also used in poultry plants, dye processing plants, and concrete cooling plants.

Tube ice appeared about 40 years ago. It is produced in a vertical shell and tube heat exchanger. The shell side is flooded with refrigerant. Water is constantly circulated in the tubes, and a water distributor at the top of each tube directs a film of water on the inside of the tubes. When the ice is harvested, it normally has a hole in the cylinder of ice. Ice is harvested by injecting hot refrigerant gas into the shell side of the icemaker. Cycle time is usually about 10 to 15 minutes (see Figure 1).

Applications that use tube ice include

packaged ice, food processing, and thermal energy storage systems.

Shell ice is another industrial ice-making system. The machine has vertical stainless steel tubes welded into a large header. Water is distributed on the outside of the tubes. The inner portion of the tubes is usually flooded with R-717. For harvesting, hot gas is introduced into the tubes. This causes the accumulated ice to shatter into small segments. Shell ice is used in the poultry processing industry, seafood industry, mine cooling, among others.

Plate ice or fragment ice is frozen on vertical plates. Water in a thin film is circulated over the plates, which are refrigerated by flooding, pumping or expanding R-717 or HCFC-22 inside the plates. The ice is harvested by introducing hot refrigerant gas inside the plates. As the ice falls from the plates in a sheet it is crushed into small segments by a high speed rotating device. Fragment ice is used for packaged ice, food processing and chemical processing (see Figure 2).

The energy costs to produce ice automatically varies little from one type of ice to another because each system must remove 144 Btus (152 kW) to freeze a pound (0.45 kg) of ice. Deciding which system is needed for a particular application is based on:

- How fast must the ice melt to perform its task? When cooling a dye process or in a concrete cooling plant, the ice needs to melt promptly, so flake ice is the usual choice.

- What type of ice will the public buy for that big party? Since people don't

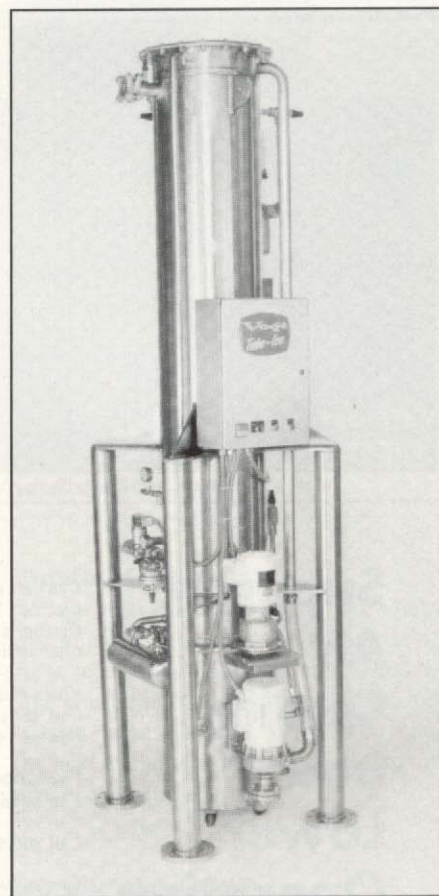


Figure 1: Tube ice maker.



Figure 2: Fragment ice maker.

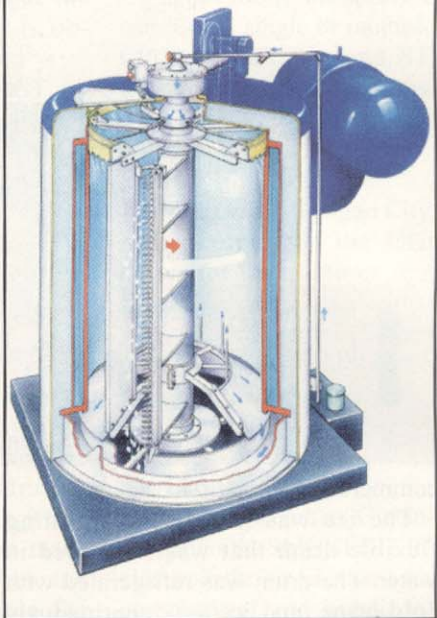


Figure 3: Flake ice machine.

want the ice in their drink to melt too fast, the obvious choices are tube or fragment ice.

- How can ice be stored and transported throughout a processing plant that has multiple users with varying demands? An ice storage bin is needed that is of a size that will provide adequate ice for the varying demands. Several types of ice storage bins are available for long- and short-term storage. Each has its own method of automatically loading and unloading the ice.

Ice storages need the refrigeration to keep the ice subcooled and in a uncongealed state so that it can be conveyed about the plant in screw conveyors or in a pneumatic conveying system. Where ice is stored more than 9 to 10 ft high (2.7 to 3 m) it tends to congeal into a massive block at the bottom of the storage. Thus, most ice bins must be completely emptied on a periodic basis.

When buying an icemaker and/or an ice storage and distribution system be sure to visit an installation similar to your needs. Making and handling ice is not a simple task — it requires sound, experienced engineering.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

*Individually Quick Frozen*

Fluidized Freezers

By George C. Briley, P.E.,

Fellow/Life Member ASHRAE

This article is the second in a series discussing mechanical food chilling and freezing systems.

IQF means "individually quick frozen," and that is the purpose of fluidized freezers. For many years, vegetables were frozen in boxes in plate freezers or blast freezers. As the demand for frozen vegetables increased and the pursuit of quality began to impact the marketplace, various types of belt freezers were installed.

These belt freezers used a stainless steel mesh belt that was 6 to 8 ft wide (2 to 2.5 m). The vegetable might be peas, cut corn, lima beans or even french fries. Cold air at -20°F to -30°F (-29°C to -34°C) was directed down onto the belt.

This was not the best of freezers as the product tended to stick to the belt and clump in large clusters. This motivated the invention of the cluster buster, which reduced the large clusters to smaller clusters that could be metered into bags and small cartons. This belt system did not produce a quality product, or one that was visually appealing.

In the early 1960s, Lewis Refrigeration Company and Frigoscandia were promoting fluidized freezing systems. Each company had a different approach to the problem. One used a continuous belt; the other used a vibrating tray with perforations in the bottom. Each used cold high-pressure air, introduced beneath the belt or tray in the quantities required to maintain true fluidization.

These systems were major improvements as they produced a high quality individually frozen product that could be stored safely in bulk in less space than packaged products.

Freezing times in IQF freezers were much shorter than the old belt freezers. As an example, the freezing time for peas with -20°F (-29°C) air is less than eight minutes. At a lower temperature the time is even less; some claim four minutes freeze time.

The cost to freeze is most important to the processor. One concern, besides the electrical cost, is the product shrink. Almost any product will lose weight when being frozen. Two major factors in controlling shrink or weight loss are freezing time and freezing temperature. The quicker a product is frozen, the lower the shrink. Most IQF freezers were operated at -20°C (-29°C) air temperature in early years. Some IQF freezers are now operating at temperatures down to -40°F (-50°F) (-40°C to -36°C). At these temperatures, quality increases and shrink decreases. The energy cost to freeze increases slightly; however, the increase in quality offsets this cost and increases the bottom line.

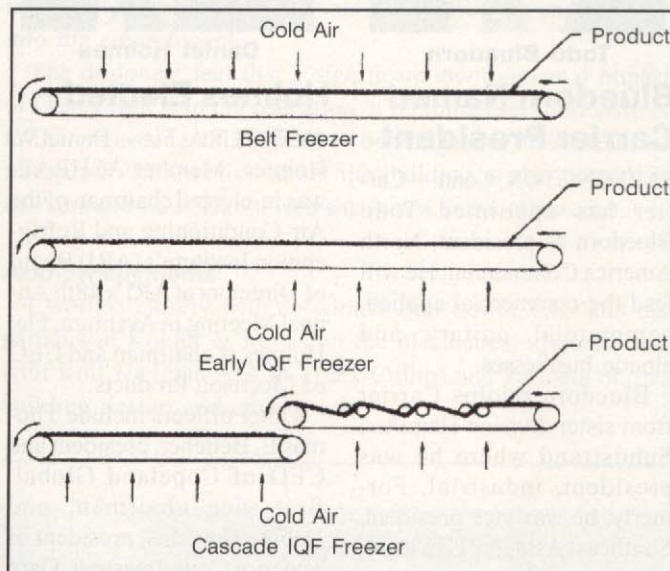


Figure 1: Evolution of IQF freezing.

The early IQF freezers in the 1960s were limited in capability as fluidization was difficult to initiate and to maintain. Product could only be loaded about 1 in. (25 mm) thick on the belt. Several innovations provided a means to start fluidization. A pulsing air jet beneath the belt at a predetermined point in the belt travel, plus a small oval bar, served well.

Most vegetables are covered with a water film as they enter the freezers. The object of the air jet was to break all the vegetables apart just as the water film started to freeze. Then, the fluidization was easy to maintain. These innovations allowed product to be loaded to ± 4 in. (100 mm) deep.

The 1980s brought a rather interesting event. An IQF freezer orientation change led to a cascade IQF freezer design. This design had two belts, one feeding the other. The first belt had several humps in it that caused the product to tumble several times while being blasted with cold air. Thus, as the product was chilled to near freezing, the surface water would freeze but the individual vegetables did not stick together. This conditioning belt would be loaded to about 1 in. (25 mm). It would operate six times faster than the fluidized freezing belt, thus depositing some 6 in. (150 mm) of vegetables on the freezing belt where fluidization was maintained. This allowed engineers to design freezers that used less plant space than older ones.

The trend in the 1980s and 1990s was toward packaging IQF freezers. Figure 2 shows a typical cascade IQF freezer. Freezers



Refrigeration Applications

with capacities up to 15,000 lb/h (1890 g/s) are packaged such that they can be hauled on a normal low-boy trailer.

IQF freezers are manufactured in many sizes, from 1,000 lb/h to 100,000 lb/h (126 to 12 600 g/s). The larger ones are usually used for French fries. Most use recirculated ammonia as the refrigerant.

IQF freezers are installed all over the world, freezing all types of vegetables, including broccoli, cauliflower, cut peppers, okra, carrots, and fruits from strawberries to peach slices.

There are at least 15 manufacturers of IQF freezers. Each has variations and accessories and modifications. Many have several models to meet customer requirements. The basic concepts, developed in the United States, are still being used. Fluidized freezing, though 40 years old, is still with us.

George C. Briley, P.E., is the president of Technicold Services, San Antonio. ●

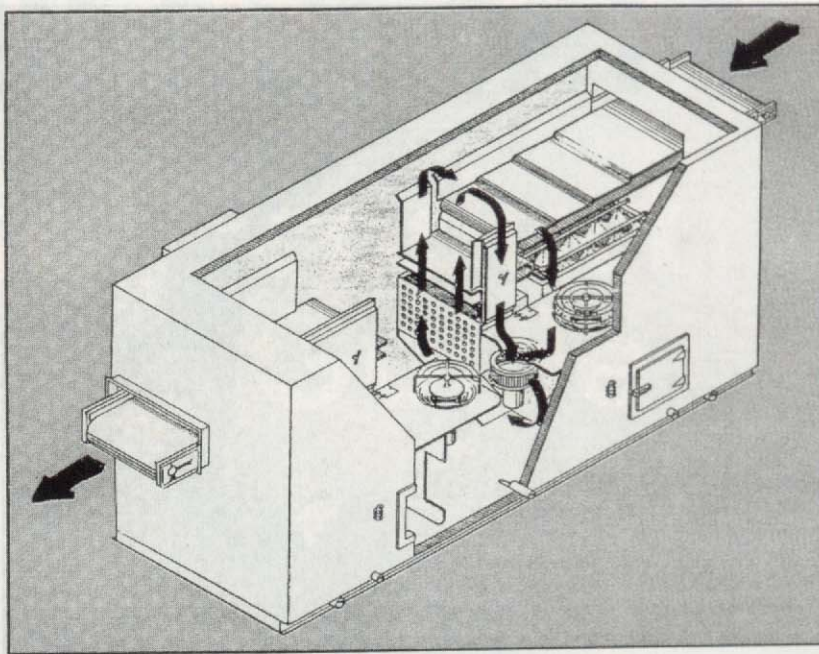


Figure 2: Cutaway view of an IQF freezer shows airflow from evaporators and centrifugal fans. "Humps" on the first-stage conveyor belt keep the product in a thin layer for rapid fluidization.



Liquid Freezing and Chilling Systems

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Liquid freezing systems freeze and chill food products such as poultry, sausage and shrimp. Types of liquid freezant include food-grade sodium chloride (NaCl_2) solution, calcium chloride (CaCl_2) and propylene glycol. Each freezant has specific applications. Mixtures of sodium and calcium chloride are used to lower the freezing point. ♦ Inline spray-type liquid freezers/chillers have a plastic conveyor belt with ample openings to pass the massive flow of sodium chloride feeds into the fiberglass liquid freezer enclosure. The flow of brine is such that the temperature rise is less than 3°F (-1.6°C). Brine is distributed by nozzles with a square pattern located on PVC headers. The enclosure bottom is sloped to the sump where a sanitary (stainless steel) pump circulates the brine through a shell and tube heat exchanger. The brine is chilled to about 2°F (-16.6°C). The etectic point (slush point) is about 6°F (-14°C) if the brine is saturated.

The brine should be dumped daily. A lirtor (brine mixing tank) is used to obtain saturated brine and a storage tank is filled with new brine. The storage tank is always full of brine for makeup during production. The brine density must be continually monitored and the makeup automated.

The spray tunnel has an equilibrating area at the end of the conveyor to allow all of the brine to drip from the product.

The refrigeration system operates at a suction temperature of $-5^\circ\text{F}/-6^\circ\text{F}$ ($-20^\circ\text{C}/-21^\circ\text{C}$), thus the cost to freeze is low compared to blast freezers, spiral freezers, IQF freezers or VRT freezers.

For freezing heads-on or headless shrimp or peeled and deveined shrimp, a saturated solution of sodium chloride (-6°F [-21°C] freezing point) is used with a proprietary solution treatment before freezing. This maltodextrine solution reduces the shrimp's ability to absorb salt from the freezing solution.

Sodium chloride, calcium chloride and/or propylene can be used for freezing shrink-wrapped items. Combinations of calcium chloride and sodium chloride reduce the solution freeze point, providing a frozen product at a lower temperature.

Items that are frozen or chilled in liquid freezing/chilling systems include chubs, Cornish game hens, turkeys and chickens.

A second approach to liquid chilling is a batch-type operation where cages or trees move the product in and out of an insulated enclosure. The cages are moved by a walking beam or a similar conveyor. Brine spray nozzles are strategically located in the enclosure to guarantee total immersion of the product, usually bacon, sausage or hot dogs. The brine is continually circulated over a plate-type chilling unit — usually refrigerated with ammonia. This brine, normally sodium chloride, must be filtered continuously to remove any food particles (Figure 1).

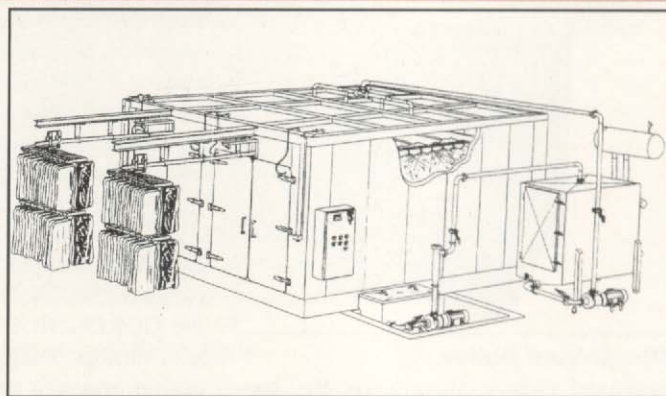


Figure 1: Batch-type liquid freezing system.

When compared to air blast, both systems have the advantages of minimizing product shrink, reducing freezing and/or chilling time, and reducing energy costs. The only downsides are the need for corrosion-resistant materials because brine is corrosive, and the continuous monitoring and maintenance of the brine connections.

Liquid immersion freezing systems freeze packaged poultry products (usually shrink-wrapped). The birds are semi-submerged in a bath of brine. The brine is chilled either by coils submerged in the tank or by an external plate-type chiller, usually refrigerated by ammonia. Brine is circulated in large quantities and this flow is used to move the birds through the tank. Some brine is sprayed on the birds to freeze the portion that is not submerged. After the birds are frozen, they are lifted from the tank and washed to remove any brine from the packaging.

Many turkey operations use immersion systems to partially freeze (crust freeze) their product to control the color, i.e., mostly white or slightly pink. The turkeys are then cartoned and fully frozen in a continuous blast freezing system. If the turkeys are frozen completely in a blast freezer, they tend to show freezer burn or redness.

The immersion freezing system can freeze any buoyant plastic sealed product in the same way as poultry.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ♦



Moisture Loss During Freezing

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Most freezing/cooling systems are judged by their ability to keep moisture loss (shrink) to a minimum. However, it is basically impossible to cool or freeze an unpackaged food product without some loss of moisture. This is a natural occurrence. Most unpackaged food products are frozen in mechanical systems that use air as the heat transfer medium. Air absorbs moisture in various quantities, depending upon temperature and pressure.

For example, if a product is frozen using 0°F (-18°C) air and the air becomes saturated with moisture, it absorbs 10 times the amount of moisture from a product as does -40°F (-40°C) air. Further, -50°F (-46°C) air would absorb 100% less moisture than -40°F (-40°C) air.

The amount of moisture released from a food product in the freezing process is directly related to the following:

1. The temperature difference and the related vapor pressure between the evaporator and the air used to freeze the product.
2. The freezing efficiency of the system as it relates to "time to freeze." This function is controlled by the pattern of air. In general, the faster a product freezes, the less possible shrink. However, there are practical and physical limitations related to the time to freeze.

3. Besides moisture absorbed by the air, drip loss can be significant. Here, we refer to the losses that occur within the freezer enclosure just prior to, or during, freezing. This drip loss is not to be confused with losses prior to the freezing process or the drip loss that can occur when thawing.

4. The vapor pressure of the moisture in the product being frozen and the diffusion coefficient of water through the structure of the product.

Now, let us examine each criteria.

Moisture in air flows directly from a high vapor pressure area (high temperature) to a low vapor pressure area (lower temperature). Thus, the normal flow of moisture is from the product to the cold air. Higher air temperature means a higher possibility that the product will give up moisture. Of course, this is only true in the initial part of the freezing process, before the outer portions of the product freeze and glaze or seal with ice.

To save on energy costs, most manufacturers use high evaporator temperatures. High evaporator temperatures with high air volumes and air temperatures that are close to the refrigerant temperature provide an optimum approach to food freezing.

Table 1 indicates the moisture content of saturated air. However, air may not be saturated at all times in a freezer. A lower evaporation temperature (the air over the evaporator approaches this temperature) tends to remove more water from the air, which, in turn has the capacity to absorb more water from the product.

Air temperature rises during the process of freezing a product. Thus, maximizing the quantity of air and minimizing air temperature rise reduces shrink.

In a mechanical freezing system, the ideal finned coil selection is one with an infinite amount of surface that has a 0°F temperature difference between the air on the coil and the refrigerant temperature. Of course, this is impossible because the coil size and cost is astronomical. The temperature differential of approximately 10°F (5.5°C) between the air temperature on the coil and the refrigerant evaporating temperature is recommended.

The freezing system efficiency as it relates to time to freeze is the most important factor in determining shrink. More shrink occurs the longer an unpackaged food product is in a poorly designed freezing environment.

For most products, shorter freeze time means that smaller ice crystals form and thus, less cell rupture occurs within the product. Large crystals are formed when products are frozen over long periods of time — say, 24 hours. In most in-line mechanical freezing systems, the freezing time for most products is within the range of cryogenics. For hamburger patties, cryogenics freezing time is about eight to 10 minutes. A mechanical refrigeration spiral freezing time is about 12 to 15 minutes. The mechanical system is within the range of cryogenic

freezing, so crystal formation is nearly the same in either case.

Drip loss varies with every product. Drip loss is affected by the product quality, amount of surface moisture or moisture that can drain/drip from the producer, and by the time to freeze. Quicker glazing of the product's surface means less drip loss. Drip loss can be a factor in shrink when freezing many cooked products.

Moisture diffusion through the product and its tendency to give up this moisture to the freezing medium varies with every product. This component of shrink is difficult to estimate and measure. Some products give up moisture rapidly due to their cellular structure. Others have a cellular structure that will not release the moisture and/or fluids they contain.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

Temperature °F	lb of Water/lb of Dry Air
0	0.0008
-10	0.00046
-20	0.00026
-30	0.00015
-40	0.00008
-50	0.00004

Table 1: Moisture carrying capacity of air (saturated).



Refrigeration for Mine Shafts

By George C. Briley, P.E., Fellow/Life Member

Sinking mineshafts has been a treacherous business, particularly in areas where there was wet soil or water stratas. Cave-ins were common and lives were lost. Europeans began freezing the soil surrounding the mineshaft as early as 1885. The first American application was likely at a sulphur mine in south Louisiana. The project was a major failure. The problem was later solved by Herman Frasch, who developed the Frasch process for sulphur extraction.

The first successful mine shaft freezing process in America was likely implemented by Winston Brothers of Minneapolis, at a potash mine in Carlsbad, N.M. The shaft was 150 ft (46 m) deep. Holes were drilled very carefully on 3.5 ft (1 m) centers in a 31 ft (9 m) circle to a depth of 350 ft (107 m). The drill holes had 6 in. (150 mm) steel pipes capped at the bottom with a 2 in. (50 mm) pipe inserted but open at the bottom (*Figure 1*). Calcium chloride brine was circulated in large quantities to minimize the temperature difference in the 6 in. (150 mm) riser. The brine temperature was 5°F (-15°C).

The object was to freeze the soil so that the ice that formed around each pipe would extend to the ice frozen by the adjacent pipe and form what was identified as closure. Thus, the ice structure would protect the workers digging the shaft and prevent any water from seeping in. To ensure that closure actually took place, various thermocouples were installed both inside and outside the circle.

Drilling the holes in a circle was the most expensive part of freezing a mine shaft, particularly when the shaft was very deep. Also, the drilled holes must be extremely straight and drilling holes to depths exceeding 1,500 ft (460 m) (depths now being frozen) is a difficult task. This is less difficult today with computer control but it was tough to drill holes 350 ft (107 m) deep with requirements that the hole could not vary outside a 3 ft (1 m) circle at any point throughout the length of the drilling. If they were, backup holes were required. Drilling was particularly difficult in alternating rocky soil.

There have been many mine shafts dug using the freezing method. Most have employed ammonia refrigerant to chill the calcium chloride brine (this should be defined as a heat transfer fluid [HTF]) to various temperatures from 5°F to -25°F (-15°C to -32°C). At lower temperatures calcium chloride becomes very viscous and its heat transfer capabilities are reduced. Other heat transfer fluids should be considered for drilling down to 1,500 ft (460 m) and below. Lower temperatures, as low as -60°F (-51°C), also are being considered to reduce freezing times and decrease overall cost to freeze.

Determining the refrigeration capacity required for soil freezing is complex. The load varies with the selected perimeter of the freeze hole circle, the soil thermal conductance, the heat transfer fluid temperature range required, the specific heat of

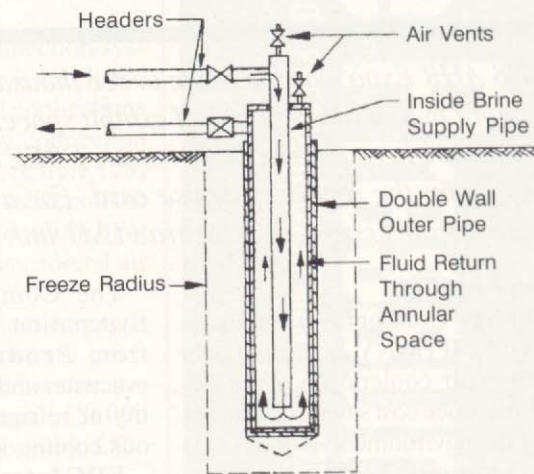


Figure 1: A typical freezing pipe arrangement.¹

the heat transfer fluid, the soil composition, the depth of the holes, the thickness of the ice wall, and the time required to freeze.

The refrigeration system designer must remember that once the freezing process starts it must not stop until the shaft is completed. Thus, the compressors and other components of the system must be spared. The system should have 100% on-line capability. Knowledgeable operators are necessary, even with our microcontrol systems.

Other applications of soil freezing include:

- Deep evacuations for building foundations,
- Deep ditches for laying pipe,
- Stabilizing areas in mines affected by water seepage,
- Tunnels.

References

1. 2002 ASHRAE Handbook—Refrigeration, chap. 35.

George C. Briley, PE., is president of Technicold Services, San Antonio. ●



Refrigeration in Large Breweries

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Breweries are sophisticated chemical plants with specific controlled chemical reactions. The production of lager beer requires refrigeration during the production process. Variations exist in the brewing process due to the brew master's concepts and practices. However, refrigeration is required in almost every step of the process.

The first step in the brewing process is "malting" of the barley. This is a complex and intricate process. The resultant product governs the quality and the taste of the beer. The malt must be cooled during production. This usually is accomplished with packaged water chillers that cool water to around 36°F to 38°F (2°C to 3°C). The steeping process water is usually about 55°F (13°C).

Hops are a significant part of the brewing process. After harvest, it is necessary to maintain hops in proper condition. They must be stored at about 35°F to 36°F (1°C to 2°C) and at a humidity under about 60%. The refrigeration duty is usually a part of the central ammonia refrigeration plant.

After the brewing process, the hot "wort" must be cooled to a temperature of 45°F to 55°F (7°C to 13°C), usually in a plate and frame heat exchanger using water or direct ammonia. After cooling, the "wort" is directed to fermentation tanks. During fermentation, carbon dioxide (CO₂) is produced. Most of this CO₂ is recovered, purified, and condensed. A small part of this CO₂ is used in vapor form to blanket the many beer tanks, piping, etc. This is done to prevent oxygen from contacting the beer. The part that is condensed is normally held in storage tanks at approximately -10°F (-23°C) and 250 psig (1723 kPa). The liquid CO₂ is vaporized, many times with low-pressure steam and used during the bottling, canning, or kegging operation to carbonate the beer. The fermentation tanks are at atmospheric pressure thus the CO₂ must be increased in pressure for plant use and for condensing. The CO₂ condensing system is cascaded into the plant ammonia refrigeration system to prevent any CO₂ from entering the ammonia plant should a tube in the CO₂ condenser leak.

After fermentation the "green" beer is cooled to 32°F to 28°F (0°C to -2°C) usually in a shell and tube spray type chiller with ammonia as the refrigerant. The beer is stored in large stainless steel refrigerated tanks. The tanks have cooling coils internally or attached to the shell. The refrigeration may be either glycol or direct ammonia. This storage area is called a fermenting cellar. Storage times vary from one to six months depending upon the specific brewing process.

The cellars must be maintained at temperatures of between 29°F and 55°F (-2°C and 13°C) and at 40% to 50% relative

humidity to prevent mold growth, etc. The air units normally used to maintain cellar temperatures use a heat transfer fluid that is hygroscopic in direct contact with the recirculated air. The cooling coils usually use ammonia as the refrigerant. The heat transfer fluid requires regeneration as it absorbs moisture from the circulated air. In general, the cooler the recirculated air means the lower the humidity in the cellar.

CO₂ is produced during the final fermentation in the cellar, thus outside air must be used to maintain the CO₂ concentration at acceptable levels. The outside air system uses the same hygroscopic heat transfer fluid as the air units in the cellar. The system usually has twin units, one for supply air, and the other for exhaust air and are arranged for energy recovery. Other types of dry air-to-air heat exchangers are sometimes used. However, they do not remove as much moisture as the hygroscopic fluid system.

From the fermenting cellar, the beer is transferred to the stock cellar. This cellar usually is a large insulated structure that encloses the large stainless steel tanks. This cellar is normally held at 32°F (0°C). The tanks of beer usually are blanketed with CO₂ vapor to ensure that no oxygen is in contact with the beer.

Cellars must be washed down from time to time, which requires large quantities of wash water. The water must be cooled to approximately the same temperature as the cellar. Using city water that is much higher in temperature would increase the humidity rapidly. Therefore, most brewery water chiller systems have large water storage tanks for this purpose.

Bottling/canning systems require the beer to be cooled further before the CO₂ vapor injection. Specific beer temperatures and CO₂ pressures are necessary to guarantee good quality control.

Most breweries require their beer to be kept at a specific temperature in the warehouse and in transit. This temperature varies. However, 50°F (10°C) is a good median.

For a detailed description of modern breweries, see Chapter 25 of the 2002 ASHRAE Handbook—Refrigeration.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●



Scraped Surface Heat Exchangers

By George C. Briley, P.E., Fellow/Life Member ASHRAE

Scraped surface heat exchangers (SSHE) are used in the food, chemical, and pharmaceutical industries to heat, cool, cook, slush freeze, emulsify, pasteurize, texturize, sterilize, crystallize, whip and concentrate many types of products. ♦SSHEs consist of a product tube or a product area, and a cylinder that contains the cooling media that may be a secondary refrigerant, a refrigerant such as ammonia, or a halocarbon.

It has a rotating shaft or mutator that normally rotates and carries the scraper blades. The blades vary depending on the type of product being processed, the temperature of the product, and the type of cooling medium. An electric variable speed motor or a hydraulic motor usually drives the shaft.

In food plants most drives are hydraulic, which are less apt to be damaged by the clean-up process. The rotor/blade assembly is housed in a tube constructed from a metal that is compatible with the product being cooled. An outer cylinder contains the cooling media. Most SSHEs are insulated in a thickness compatible with the temperature of the cooling medium. This insulation is covered with a stainless steel jacket.

Many companies manufacture SSHEs. Units are available in vertical and horizontal arrangements. Each has its own advantages.

During processing, the product is normally pumped into the product tube. The product is scraped from the heat transfer tube continuously by the scraper blades. In general, the SSHEs are used to cool or freeze viscous items or items that become viscous when cooled or frozen. The scraped surface of the SSHEs is kept clean and has an exceptionally high heat transfer rate that remains constant as long as the cooling medium continues at design. The scraping action also minimizes product pressure drop. Because of the rapid rate of cooling, SSHEs provide an excellent product. *Figure 1* shows the construction details of a vertical SSHE with the flow of the product to be cooled and the flow of the cooling medium.

Ice cream plants use this type of freezer to harden ice cream to 20°F (-7°C) or so before it is packaged and directed to the hardening process where it is frozen to -20°F (-29°C). The citrus industry uses SSHEs to freeze citrus concentrates to about 20°F (-7°C) before canning. The cans are then directed to a belt freezing system where the product is frozen. Other products that are cooled or partially frozen by SSHEs include: margarine, butter, process cheese, chili, baked beans, pet foods, and marshmallow topping.

The refrigeration system may vary considerably from a liquid overfeed system to a flooded system. Many ice cream freezers use flooded systems, which causes lubricant to accumulate in the SSHE and reduce heat transfer. This type of unit requires a daily cleaning of the refrigerant and processing chambers.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ♦

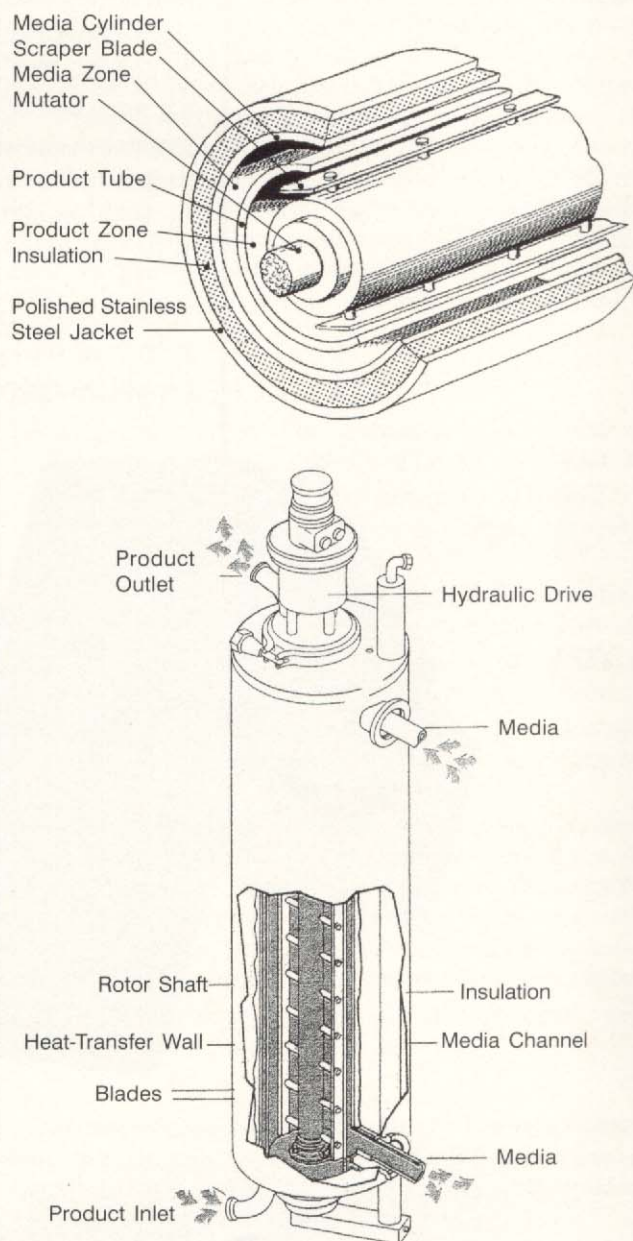


Figure 1: Scraped surface heat exchanger schematics.



Variable Retention Time Freezers

By George C. Briley, P.E., Fellow/Life Member

The variable retention time (VRT) freezer is used to cool or freeze food products in boxes/cartons or trays. Its major advantage is that it can freeze several different products, each with different freezing times and different size packages using a programmable computer that controls its operation. The VRT (*Figure 1*) has a series of independent trays that can be loaded into any tray guide at any level. A programmable industrial computer controls the VRT.

The machine cycle begins when new product is sent from the processing area to the VRT. An accumulating conveyor system receives and holds the boxed product until a sufficient number of boxes are received to fill one freezer tray.

When a full load is ready, the boxes are automatically transferred to the infeed conveyor and loaded onto an empty tray. The vertical loading elevators raise the loaded tray to the designated level. The tray is then inserted into the tray guides. As a new tray is moved into position, it advances all of the trays ahead. The last tray on the loaded level is transferred onto the rear elevator and is lowered. Once the frozen tray reaches the bottom, the product is unloaded and removed by the exit conveyor. Empty trays are returned to the front of the freezer to receive new product.

New trays entering into the freezer are electronically stamped with an entry time. As trays advance through the freezer, their elapsed dwell time is tracked. Periodically, the dwell time of each tray is updated and displayed on the VRT status screen. Through this method, elapsed dwell time of each tray can be reviewed at a glance.

If frozen product needs to be removed from the freezer but no new product needs to be loaded, the unload process is used. Empty trays are returned from the unloaded area and inserted into levels selected for unloading. This cycles empty trays through the freezer and transfers trays with frozen product to the rear elevator for unloading.

In the automatic mode, the computer is used to make the unloading decision. All trays with product meeting or exceeding the required dwell time are automatically cycled out of the freezer.

Most VRTs can be operated in a semi-automatic or manual control mode. Typical industry applications include freezing meat, precooked foods, and fruit/fruit juices in cartons or trays and freezing ice cream in trays or overwraps.

VRT freezers arranged as in *Figure 1* are designed for counterflow heat exchange. Directing the coldest air against the coldest product ensures thorough product freezing. Packages are arranged with minimal spacing to allow for efficient heat rejection from both sides. The perforated steel trays are designed to allow uniform

heat rejection from the package bottom. Packages are loaded on the trays so that the long side is aligned with the airflow. This, along with the perforated trays, exposes the maximum area of the carton to the cold air, thus minimizing freezing time.

After passing over the product, the refrigerated air is returned down the sides of the tunnel and across the evaporators. Careful attention is placed on creating efficient turbulent airflow across the product, while minimizing fan horsepower. This design reduces air quantities by as much as 50% compared to conventional systems using an air-distribution system that distributes air from the sides of the VRT. VRT freezers have controlled airflow with freezing times of 20% to 40% less than blast freezers.

Most VRT freezers use hydraulics for their operation. Water defrost often is used for the freezer coil. Hot gas also is used. Each VRT manufacturer has its own approach to air distribution, fans, safety devices, enclosure, cell configuration, etc. The user should insist on a freezing test of the product under conditions that simulate the proposed system. The test should provide freezing time, which should be specified as part of any purchase.

George C. Briley, P.E., is president of Technicold Services, San Antonio. ●

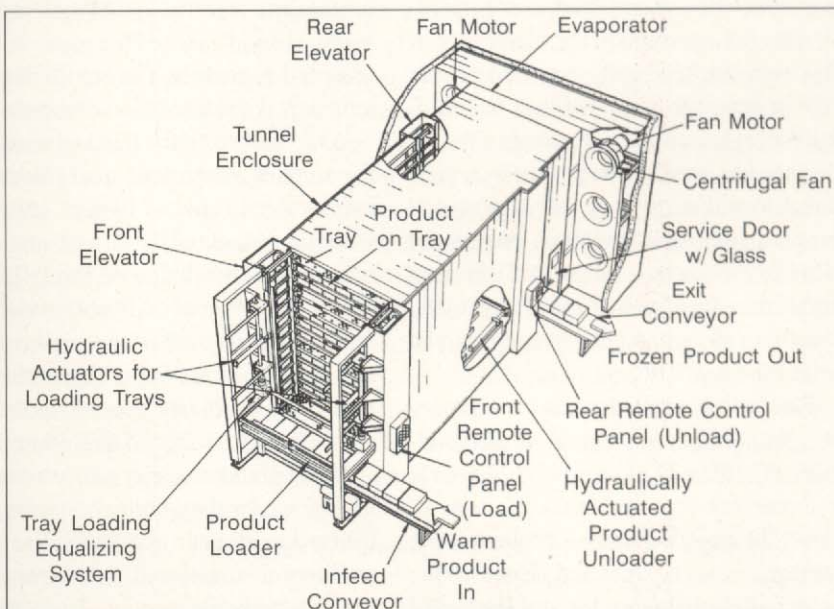


Figure 1: Typical variable retention time freezer.