

CHAPTER 4

**SECONDARY COOLANTS IN REFRIGERATION SYSTEMS**

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**S**ECONDARY coolants are liquids used as heat transfer fluids that change temperature as they gain or lose heat energy without changing into another phase. For lower refrigeration temperatures, this requires a coolant with a freezing point below that of water. This chapter discusses design considerations for components, system performance requirements, and applications for secondary coolants. Related information can be found in Chapters 2, 3, 20, 21, and 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

**COOLANT SELECTION**

A secondary coolant must be compatible with other materials in the system at the pressures and temperatures encountered for maximum component reliability and operating life. The coolant should also be compatible with the environment and the applicable safety regulations, and should be economical to use and replace.

The coolant should have a minimum freezing point of 5°F below and preferably 15°F below the lowest temperature to which it will be exposed. When subjected to the lowest temperature in the system, coolant viscosity should be low enough to allow satisfactory heat transfer and reasonable pressure drop.

Coolant vapor pressure should not exceed that allowed at the maximum temperature encountered. To avoid a vacuum in a low-vapor-pressure secondary coolant system, the coolant can be pressurized with pressure-regulated dry nitrogen in the expansion tank. However, some special secondary coolants such as those used for computer circuit cooling have a high solubility for nitrogen and must therefore be isolated from the nitrogen with a suitable diaphragm.

**Load Versus Flow Rate**

The secondary coolant pump is usually in the return line upstream of the chiller. Therefore, the pumping rate is based on the density at the return temperature. The mass flow rate for a given heat load is based on the desired temperature range and required coefficient of heat transfer at the average bulk temperature.

To determine heat transfer and pressure drop, the specific gravity, specific heat, viscosity, and thermal conductivity are based on the average bulk temperature of coolant in the heat exchanger, noting that film temperature corrections are based on the average film temperature. Trial solutions of the secondary coolant-side coefficient compared to the overall coefficient and total log mean temperature difference (LMTD) determine the average film temperature. Where the secondary coolant is cooled, the more viscous film reduces the heat transfer rate and raises the pressure drop compared to what can be expected at the bulk temperature. Where the secondary coolant is heated, the less viscous film approaches the heat transfer rate and pressure drop expected at the bulk temperature.

The more turbulence and mixing of the bulk and film, the better the heat transfer and higher the pressure drop. Where secondary

coolant velocity in the tubes of a heat transfer device results in laminar flow, heat transfer can be improved by inserting spiral tapes or spring turbulators that promote mixing the bulk and film. This usually increases pressure drop. The inside surface can also be spirally grooved or augmented by other devices. Because the state of the art of heat transfer is constantly improving, use the most cost-effective heat exchanger to provide optimum heat transfer and pressure drop. Energy costs for pumping secondary coolant must be considered when selecting the fluid to be used and the heat exchangers to be installed.

**Pumping Cost**

Pumping costs are a function of the secondary coolant selected, load and temperature range where energy is transferred, pump pressure required by the system pressure drop (including that of the chiller), mechanical efficiencies of the pump and driver, and electrical efficiency and power factor where the driver is an electric motor. Small centrifugal pumps, operating in the range of approximately 50 gpm at 80 ft of head to 150 gpm at 70 ft of head, for 60 Hz applications, typically have 45 to 65% efficiency, respectively. Larger pumps, operating in the range of 500 gpm at 80 ft of head to 1500 gpm at 70 ft of head, for 60 Hz applications, typically have 75 to 85% efficiency, respectively.

A pump should operate near its peak operating efficiency for the flow rate and pressure that usually exist. Secondary coolant temperature increases slightly from energy expended at the pump shaft. If a semihermetic electric motor is used as the driver, motor inefficiency is added as heat to the secondary coolant, and the total kilowatt input to the motor must be considered in establishing load and temperatures.

**Performance Comparisons**

Assuming that the total refrigeration load at the evaporator includes the pump motor input and brine line insulation heat gains, as well as the delivered beneficial cooling, tabulating typical secondary coolant performance values assists in the coolant selection. A 1.06 in. ID smooth steel tube evaluated for pressure drop and internal heat transfer coefficient at the average bulk temperature of 20°F and a temperature range of 10°F for 7 fps tube-side velocity provides comparative data ([Table 1](#)) for some typical coolants. [Table 2](#) ranks the same coolants comparatively, using data from [Table 1](#).

For a given evaporator configuration, load, and temperature range, select a secondary coolant that gives satisfactory velocities, heat transfer, and pressure drop. At the 20°F level, hydrocarbon and halocarbon secondary coolants must be pumped at a rate of 2.3 to 3.0 times the rate of water-based secondary coolants for the same temperature range.

Higher pumping rates require larger coolant lines to keep the pump's pressure and brake horsepower requirement within reasonable limits. [Table 3](#) lists approximate ratios of pump power for secondary coolants. Heat transferred by a given secondary coolant affects the cost and perhaps the configuration and pressure drop of a chiller and other heat exchangers in the system; therefore, [Tables 2](#) and [3](#) are only guides of the relative merits of each coolant.

The preparation of this chapter is assigned to TC 10.1, Custom Engineered Refrigeration Systems.

Table 1 Secondary Coolant Performance Comparisons

Secondary Coolant	Concentration (by Weight), %	Freeze Point, °F	gpm/ton <sup>a</sup>	Pressure Drop, <sup>b</sup> psi	Heat Transfer Coefficient <sup>c</sup> $h_i$ , Btu/h·ft <sup>2</sup> ·°F
Propylene glycol	39	-5.1	2.56	2.91	205
Ethylene glycol	38	-6.9	2.76	2.38	406
Methanol	26	-5.3	2.61	2.05	473
Sodium chloride	23	-5.1	2.56	2.30	558
Calcium chloride	22	-7.8	2.79	2.42	566
Aqua ammonia	14	-7.0	2.48	2.44	541
Trichloroethylene	100	-123	7.44	2.11	432
d-Limonene	100	-142	6.47	1.48	321
Methylene chloride	100	-142	6.39	1.86	585
R-11	100	-168	7.61	2.08	428

<sup>a</sup>Based on inlet secondary coolant temperature at pump of 25°F.

<sup>b</sup>Based on one length of 16 ft tube with 1.06 in. ID and use of Moody Chart (1944) for an average velocity of 7 fps. Input/output losses equal one Vel.  $H_p(V^2p/2g)$  for 7 fps velocity. Evaluations are at a bulk temperature of 20°F and a temperature range of 10°F.

<sup>c</sup>Based on curve fit equation for Kern's (1950) adaptation of Sieder and Tate's (1936) heat transfer equation using 16 ft tube for  $L/D = 181$  and film temperature of 5°F lower than average bulk temperature with 7 fps velocity.

Table 2 Comparative Ranking of Heat Transfer Factors at 7 fps\*

Secondary Coolant	Heat Transfer Factor
Propylene glycol	1.000
d-Limonene	1.566
Ethylene glycol	1.981
R-11	2.088
Trichloroethylene	2.107
Methanol	2.307
Aqua ammonia	2.639
Sodium chloride	2.722
Calcium chloride	2.761
Methylene chloride	2.854

\*Based on Table 1 values using 1.06 in. ID tube 16 ft long. Actual ID and length vary according to specific loading and refrigerant applied with each secondary coolant, tube material, and surface augmentation.

Table 3 Relative Pumping Energy Required\*

Secondary Coolant	Energy Factor
Aqua ammonia	1.000
Methanol	1.078
Propylene glycol	1.142
Ethylene glycol	1.250
Sodium chloride	1.295
Calcium chloride	1.447
d-Limonene	2.406
Methylene chloride	3.735
Trichloroethylene	4.787
R-11	5.022

\*Based on same pump pressure, refrigeration load, 20°F average temperature, 10°F range, and freezing point (for water-based secondary coolants) 20 to 23°F below lowest secondary coolant temperature.

## Other Considerations

Corrosion must be considered when selecting coolant, inhibitor, and system components. The effect of secondary coolant and inhibitor toxicity on the health and safety of plant personnel or consumers of food and beverages must be considered. The flash point and explosive limits of secondary coolant vapors must also be evaluated.

Examine the secondary coolant stability for anticipated moisture, air, and contaminants at the temperature limits of materials used in the system. Skin temperatures of the hottest elements determine secondary coolant stability.

If defoaming additives are necessary, their effect on thermal stability and coolant toxicity must be considered for the application.

## DESIGN CONSIDERATIONS

Secondary coolant vapor pressure at the lowest operating temperature determines whether a vacuum could exist in the secondary coolant system. To keep air and moisture out of the system, pressure-controlled dry nitrogen can be applied to the top level of secondary coolant (e.g., in the expansion tank or a storage tank). Gas pressure over the coolant plus the pressure created at the lowest point in the system by the maximum vertical height of coolant determine the minimum internal pressure for design purposes. The coincident highest pressure and lowest secondary coolant temperature dictate the design working pressure (DWP) and material specifications for the components.

To select proper relief valve(s) with settings based on the system DWP, the highest temperatures to which the secondary coolant could be subjected should be considered. This temperature occurs in case of heat radiation from a fire in the area or the normal warming of the valved-off sections. Normally, a valved-off section is relieved to an unconstrained portion of the system and the secondary coolant can expand freely without loss to the environment.

Safety considerations for the system are found in ASHRAE *Standard 15*. Design standards for pressure piping can be found in ASME *Standard B31.5*, and design standards for pressure vessels in Section VIII of the ASME *Boiler and Pressure Vessel Code*.

## Piping and Control Valves

Piping should be sized for reasonable pressure drop using the calculation methods in Chapters 2 and 36 of the 2005 *ASHRAE Handbook—Fundamentals*. Balancing valves or orifices in each of the multiple feed lines help distribute the secondary coolant. A reverse-return piping arrangement balances flow. Control valves that vary flow are sized for 20 to 80% of the total friction pressure drop through the system for proper response and stable operation. Valves sized for pressure drops smaller than 20% may respond too slowly to a control signal for a flow change. Valves sized for pressure drops over 80% can be too sensitive, causing control cycling and instability.

## Storage Tanks

Storage tanks can shave peak loads for brief periods, limit the size of refrigeration equipment, and reduce energy costs. In off-peak hours, a relatively small refrigeration plant cools a secondary coolant stored for later use. A separate circulating pump sized for the maximum flow needed by the peak load is started to satisfy peak load. Energy cost savings are enhanced if the refrigeration equipment is used to cool secondary coolant at night, when the cooling medium for heat rejection is generally at the lowest temperature.

The load profile over 24 h and the temperature range of the secondary coolant determine the minimum net capacity required for the refrigeration plant, pump sizes, and minimum amount of secondary

coolant to be stored. For maximum use of the storage tank volume at the expected temperatures, choose inlet velocities and locate connections and tank for maximum stratification. Note, however, that maximum use will probably never exceed 90% and, in some cases, may equal only 75% of the tank volume.

**Example 1.** Figure 1 depicts the load profile and Figure 2 shows the arrangement of a refrigeration plant with storage of a 23% (by weight) sodium chloride secondary coolant at a nominal 20°F. During the peak load of 50 tons, a range of 8°F is required. At an average temperature of 24°F, with a range of 8°F, the coolant's specific heat  $c_p$  is 0.791 Btu/lb·°F. At 28°F, the weight per unit volume of coolant  $\rho_L$  at the pump =  $[1.183(62.4 \text{ lb/ft}^3)]/(7.48 \text{ gal/ft}^3)$ ; at 20°F,  $\rho_L$  =  $[1.185(62.4 \text{ lb/ft}^3)]/(7.48 \text{ gal/ft}^3)$ .

Determine the minimum size storage tank for 90% use, minimum capacity required for the chiller, and sizes of the two pumps. The chiller and the chiller pump run continuously. The secondary coolant storage pump runs only during the peak load. A control valve to the load source diverts all coolant to the storage tank during a zero-load condition, so that the initial temperature of 20°F is restored in the tank. During low load, only the required flow rate for a range of 8°F at the load source is used; the balance returns to the tank and restores the temperature to 20°F.

**Solution:** If  $x$  is the minimum capacity of the chiller, determine the energy balance in each segment by subtracting the load in each segment from  $x$ . Then multiply the result by the time length of the respective segments, and add as follows:

$$6(x - 0) + 4(x - 50) + 14(x - 9) = 0$$

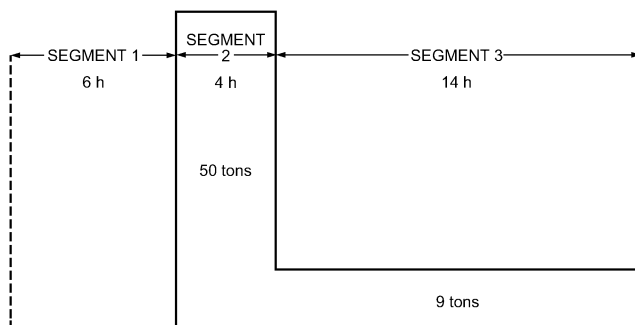
$$6x + 4x - 200 + 14x - 126 = 0$$

$$24x = 326$$

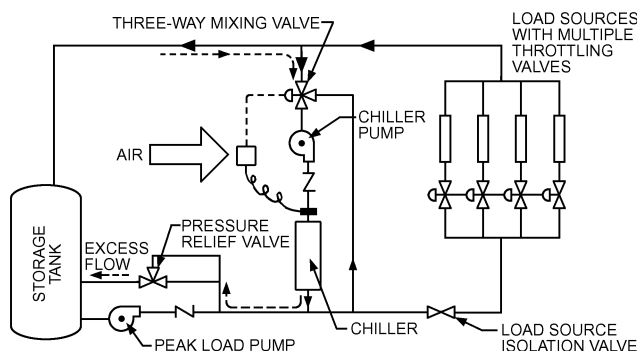
$$x = 13.58 \text{ tons}$$

Calculate the secondary coolant flow rate  $W$  at peak load:

$$W = (50 \times 200)/(0.791 \times 8) = 1580.3 \text{ lb/min}$$



**Fig. 1** Load Profile of Refrigeration Plant Where Secondary Coolant Storage Can Save Energy



**Fig. 2** Arrangement of System with Secondary Coolant Storage

For the chiller at 15 tons, the secondary coolant flow rate is

$$W = (15 \times 200)/(0.791 \times 8) = 474.1 \text{ lb/min}$$

Therefore, the coolant flow rate to the storage tank pump is  $1580.3 - 474.1 = 1106.2 \text{ lb/min}$ . Chiller pump size is determined by

$$474.1/[(1.183 \times 62.4)/7.48] = 48 \text{ gpm}$$

Calculate the storage tank pump size as follows:

$$1106.2/[(1.185 \times 62.4)/7.48] = 112 \text{ gpm}$$

Using the concept of stratification in the storage tank, the interface between warm return and cold stored secondary coolant falls at the rate pumped from the tank. Because the time segments fix the total amount pumped and the storage tank pump operates only in segment 2 (see Figure 1), the minimum tank volume  $V$  at 90% use is determined as follows:

$$\text{Total mass} = [(1106.2 \text{ lb/min})(60 \text{ min/h})(4 \text{ h})]/0.90 = 295,000 \text{ lb}$$

and

$$V = 295,000/[(1.185 \times 62.4)/7.48] = 29,840 \text{ gal}$$

A larger tank (e.g., 50,000 gal) provides flexibility for longer segments at peak load and accommodates potential mixing. It may be desirable to insulate and limit heat gains to 8000 Btu/h for the tank and lines. Energy use for pumping can be limited by designing for 46 ft head. With the smaller pump operating at 51% efficiency and the larger pump at 52.5% efficiency, pump heat added to the secondary coolant is 3300 and 7478 Btu/h, respectively.

For cases with various time segments and their respective loads, the maximum load for segment 1 or 3 with the smaller pump operating cannot exceed the net capacity of the chiller minus insulation and pump heat gain to the secondary coolant. For various combinations of segment time lengths and cooling loads, the recovery or restoration rate of the storage tank to the lowest temperature required for satisfactory operation should be considered.

As load source circuits shut off, excess flow is bypassed back to the storage tank (Figure 2). The temperature setting of the three-way valve is the normal return temperature for full flow through the load sources.

When only the storage tank requires cooling, flow is as shown by the dashed lines with the load source isolation valve closed. When storage tank temperature is at the desired level, the load isolation valve can be opened to allow cooling of the piping loops to and from the load sources for full restoration of storage cooling capacity.

### Expansion Tanks

Figure 3 shows a typical closed secondary coolant system without a storage tank; it also illustrates different control strategies. The reverse-return piping assists flow balance. Figure 4 shows a secondary coolant strengthening unit for salt brines. Secondary coolant expansion tank volume is determined by considering the total coolant inventory and differences in coolant density at the lowest temperature  $t_1$  of coolant pumped to the load location and the maximum temperature. The expansion tank is sized to accommodate a residual volume with the system coolant at  $t_1$ , plus an expansion volume and vapor space above the coolant. A vapor space equal to 20% of the expansion tank volume should be adequate. A level indicator, used to prevent overcharging, is calibrated at the residual volume level versus lowest system secondary coolant temperature.

**Example 2.** Assume a 50,000 gal charge of 23% sodium chloride secondary coolant at  $t_1$  of 20°F in the system. If 100°F is the maximum temperature, determine the size of the expansion tank required. Assume that the residual volume is 10% of the total tank volume and that the vapor space at the highest temperature is 20% of the total tank volume.

$$ETV = \frac{V_S[(SG_1/SG_2) - 1]}{1 - (R_F + V_F)}$$

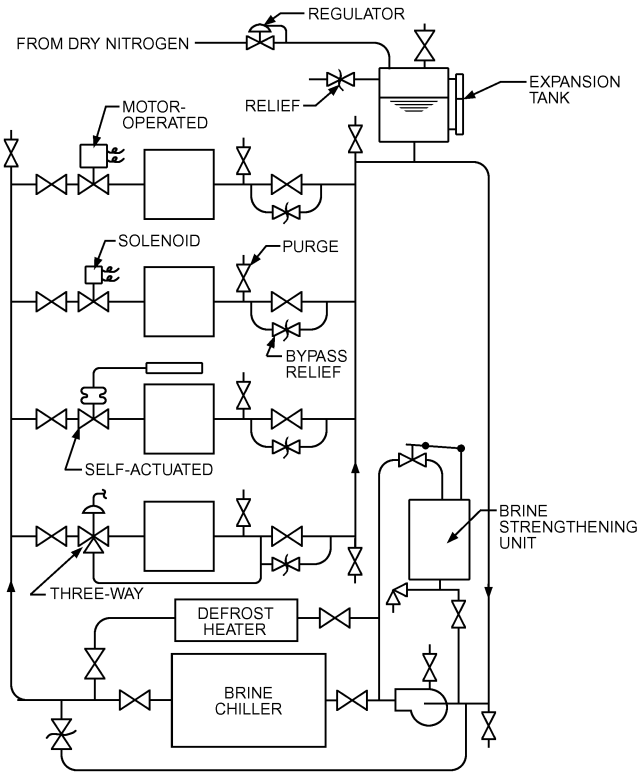


Fig. 3 Typical Closed Salt Brine System

$R_F$  = residual volume of tank liquid (low level) at  $t_1$ , expressed as a fraction  
 $V_F$  = volume of vapor space at highest temperature, expressed as a fraction

If the specific gravity of the secondary coolant is 1.185 at 20°F and 1.155 at 100°F, the tank volume is

$$ETV = \frac{50,000[(1.185/1.155) - 1]}{1 - (0.10 + 0.20)} = 1855 \text{ gal}$$

**Pulldown Time**

Example 1 is based on a static situation of secondary coolant temperature at two different loads: normal and peak. The length of time for pulldown from 100°F to the final 20°F may need to be calculated. For graphical solution, required heat extraction versus secondary coolant temperature is plotted. Then, by iteration, pulldown time is solved by finding the net refrigeration capacity for each increment of coolant temperature change. A mathematical method may also be used.

The 15 ton system in the examples has a 30.03 ton capacity at a maximum of 50°F saturated suction temperature (STP). For pulldown, a compressor suction pressure regulator (holdback valve) is sometimes used. The maximum secondary coolant temperature must be determined when the holdback valve is wide open and the STP is at 50°F. For Example 1, this is at 70°F coolant temperature. As coolant temperature is further reduced with a constant 48.1 gpm, refrigeration system capacity gradually reduces until a 15 ton capacity is reached with 26°F coolant in the tank. Further cooling to 20°F is at reduced capacity.

Temperatures of the secondary coolant mass, storage tanks, piping, cooler, pump, and insulation must all be reduced. In Example 1, as the coolant drops from 100 to 20°F, the total heat removed from these items is as follows:

Brine Temperature, °F	Total Heat Removed, Million Btu
100	31.54
80	23.62
70	19.67
60	15.73
40	7.85
20	0

From a secondary coolant temperature of 100 to 70°F, the refrigeration system capacity is fixed at 30.03 tons, and the time for pulldown is essentially linear (system net tons for pulldown is less than the compressor capacity because of heat gain through insulation and added pump heat). In Example 1, pump heat was not considered. When recognizing the variable heat gain for a 95°F ambient, and the pump heat as the secondary coolant temperature is reduced, the following net capacity is available for pulldown at various secondary coolant temperatures:

Brine Temperature, °F	Net Capacity, Tons
100	29.86
80	29.58
70	29.44
60	25.28
40	17.80
26	14.10
20	12.70

A curve fit shows capacity is a straight line between the values for 100 and 70°F. Therefore, the pulldown time for this interval is

$$\theta = \frac{[(31.54 \times 10^6) - (19.67 \times 10^6)]}{12,000[(29.86 + 29.44)/2]} = 33.4 \text{ h}$$

From 70 to 20°F, the capacity curve fits a second-degree polynomial equation as follows:

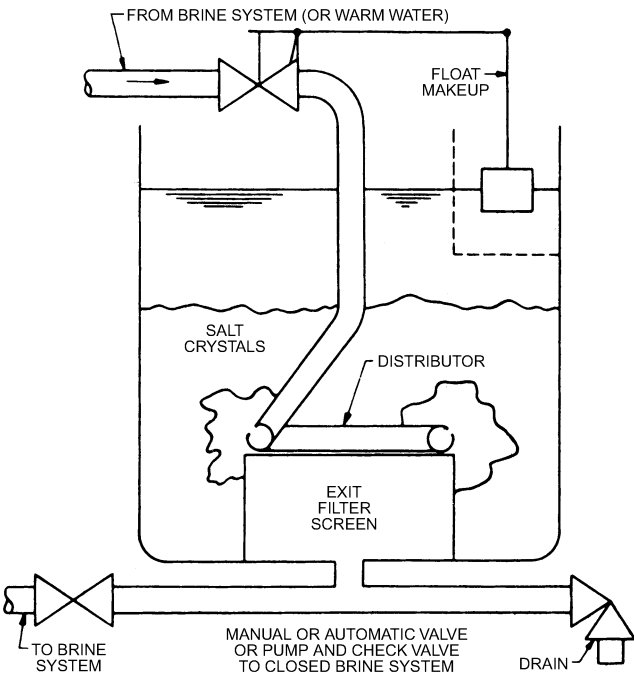


Fig. 4 Brine Strengthening Unit for Salt Brines Used as Secondary Coolants

where

- ETV = expansion tank volume
- $V_S$  = system secondary coolant volume at temperature  $t_1$
- $SG_1$  = specific gravity at  $t_1$
- $SG_2$  = specific gravity at maximum temperature

$$q = 9.514809086 + 0.1089883647t + 0.002524039t^2$$

where

$$t = \text{secondary coolant temperature, } ^\circ\text{F}$$

$$q = \text{capacity for pulldown, tons}$$

Using the arithmetic average pulldown net capacity from 70 to 20°F, the time interval would be

$$\theta = \frac{19.67 \times 10^6}{12,000[(29.44 + 12.7)/2]} = 77.8 \text{ h}$$

If the logarithmic (base  $e$ ) mean average net capacity for this temperature interval is used, the time is

$$\theta = \frac{19.67 \times 10^6}{(19.91 \times 12,000)} = 82.3 \text{ h}$$

This is a difference of 4.5 h, and neither solution is correct. A more exact calculation uses a graphical analysis or calculus. One mathematical approach determines the heat removed per degree of secondary coolant temperature change per ton of capacity. Because the coolant's heat capacity and heat leakage change as the temperature drops, the amount of heat removed is best determined by first fitting a curve to the data for total heat removed versus secondary coolant temperature. Then a series of iterations for secondary coolant temperature  $\pm 1^\circ\text{F}$  is made as the temperature is reduced. The polynomial equations may be solved by computer or calculator with a suitable program or spreadsheet. The time for pulldown is less if supplemental refrigeration is available for pulldown or if less secondary coolant is stored.

The correct answer is 88.1 h, which is 7% greater than the logarithmic mean average capacity and 13% greater than the arithmetic average capacity over the temperature range.

Therefore, total time for temperature pulldown from 100 to 20°F is

$$\theta = 33.4 + 88.1 = 121.5 \text{ h}$$

### System Costs

Various alternatives may be evaluated to justify a new project or system modification. Means (updated annually) lists the installed cost of various projects. NBS (1978) and Park and Jackson (1984) discuss engineering and life-cycle cost analysis. Using various time-value-of-money formulas, payback for storage tank handling of peak loads compared to large refrigeration equipment and higher energy costs can be evaluated. Trade-offs in these costs (initial, maintenance, insurance, increased secondary coolant, loss of space, and energy escalation) all must be considered.

### Corrosion Prevention

Corrosion prevention requires choosing proper materials and inhibitors, routine testing for pH, and eliminating contaminants. Because potentially corrosive calcium chloride and sodium chloride salt brine secondary coolant systems are widely used, test and adjust the brine solution monthly. To replenish salt brines in a system, a concentrated solution may be better than a crystalline form, because it is easier to handle and mix.

A brine should not be allowed to change from alkaline to acidic. Acids rapidly corrode the metals ordinarily used in refrigeration and ice-making systems. Calcium chloride usually contains sufficient alkali to render the freshly prepared brine slightly alkaline. When any brine is exposed to air, it gradually absorbs carbon dioxide and oxygen, which eventually make the brine slightly acid. Dilute brines dissolve oxygen more readily and generally are more corrosive than concentrated brines. One of the best preventive measures is to make a closed rather than open system, using a regulated inert gas over the surface of a closed expansion tank (see [Figure 2](#)). However, many

systems, such as ice-making tanks, brine-spray unit coolers, and brine-spray carcass chill rooms, cannot be closed.

A brine pH of 7.5 for a sodium or calcium chloride system is ideal, because it is safer to have a slightly alkaline rather than a slightly acid brine. Operators should check pH regularly.

If a brine is acid, the pH can be raised by adding caustic soda dissolved in warm water. If a brine is alkaline (indicating ammonia leakage into the brine), carbonic gas or chromic, acetic, or hydrochloric acid should be added. Ammonia leakage must be stopped immediately so that the brine can be neutralized.

In addition to controlling pH, an inhibitor should be used. Generally, sodium dichromate is the most effective and economical for salt brine systems. The granular dichromate is bright orange and readily dissolves in warm water. Because it dissolves very slowly in cold brine, it should be dissolved in warm water and added to the brine far enough ahead of the pump so that only a dilute solution reaches the pump. Recommended quantities are 125 lb/1000 ft<sup>3</sup> of calcium chloride brine, and 200 lb/1000 ft<sup>3</sup> of sodium chloride brine.

Adding sodium dichromate to the salt brine does not make it non-corrosive immediately. The process is affected by many factors, including water quality, specific gravity of the brine, amount of surface and kind of material exposed in the system, age, and temperature. Corrosion stops only when protective chromate film has built up on the surface of the zinc and other electrically positive metals exposed to the brine. No simple test is available to determine chromate concentration. Because the protection afforded by sodium dichromate treatment depends greatly on maintaining the proper chromate concentration in the brine, brine samples should be analyzed annually. The proper concentration for calcium chloride brine is 7.58 gr/gal (as Na<sub>2</sub>Cr<sub>2</sub>O<sub>7</sub>·2H<sub>2</sub>O); for sodium chloride brine, it is 12.128 gr/gal (as Na<sub>2</sub>Cr<sub>2</sub>O<sub>7</sub>·2H<sub>2</sub>O).

Crystals and concentrated solutions of sodium dichromate can cause severe skin rash, so avoid contact. If contact does occur, wash the skin immediately. *Warning: sodium dichromate should not be used for brine spray decks, spray units, or immersion tanks where food or personnel may come in contact with the spray mist or the brine itself.*

Polyphosphate/silicate and orthophosphate/boron mixtures in water-treating compounds are useful for sodium chloride brines in open systems. However, where the rate of spray loss and dilution is very high, any treatment other than density and pH control is not economical. For the best protection of spray unit coolers, housings and fans should be of a high quality, hot-dipped galvanized construction. Stainless steel fan shafts and wheels, scrolls, and eliminators are desirable.

Although nonsalt secondary coolants described in this chapter are generally noncorrosive when used in systems for long periods, recommended inhibitors should be used, and pH should be checked occasionally.

Steel, iron, or copper piping should not be used to carry salt brines. Use copper nickel or suitable plastic. Use all-steel and iron tanks if the pH is not ideal. Similarly, calcium chloride systems usually have all-iron and steel pumps and valves to prevent electrolysis in the presence of acidity. Sodium chloride systems usually have all-iron or all-bronze pumps. When pH can be controlled in a system, brass valves and bronze fitted pumps may be satisfactory. A stainless steel pump shaft is desirable. Consider salt brine composition and temperature to select the proper rotary seal or, for dirtier systems, the proper stuffing box.

### APPLICATIONS

Applications for secondary coolant systems are extensive (see [Chapters 11 to 37](#)). A glycol coolant prevents freezing in solar collectors and outdoor piping. Secondary coolants heated by solar collectors or by other means can be used to heat absorption cooling

equipment, to melt a product such as ice or snow, or to heat a building. Process heat exchangers can use a number of secondary coolants to transfer heat between locations at various temperature levels. Using secondary coolant storage tanks increases the availability of cooling and heating and reduces peak demands for energy.

Each supplier of refrigeration equipment that uses secondary coolant flow has specific ratings. Flooded and direct-expansion coolers, dairy plate heat exchangers, food processing, and other air, liquid, and solid chilling devices come in various shapes and sizes. Refrigerated secondary coolant spray wetted-surface cooling and humidity control equipment has an open system that absorbs moisture while cooling and then continuously regenerates the secondary coolant with a concentrator. Although this assists cooling, dehumidifying, and defrosting, it is not strictly a secondary coolant flow application for refrigeration, unless the secondary coolant also is used in the coil. Heat transfer coefficients can be determined from vendor rating data or by methods described in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* and appropriate texts.

A primary refrigerant may be used as a secondary coolant in a system by being pumped at a flow rate and pressure high enough that the primary heat exchange occurs without evaporation. The refrigerant is then subsequently flashed at low pressure, with the

resulting flash gas drawn off to a compressor in the conventional manner.

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## [Related Commercial Resources](#)