CHAPTER 4

# SECONDARY COOLANTS IN REFRIGERATION SYSTEMS



SECONDARY coolants are liquids that are used as heat transfer fluids and that change temperature as they gain or lose heat energy without changing into another phase. For the lower temperatures of refrigeration, this requires a coolant with a freezing point below that of water. In this chapter, the design considerations for components, system performance requirements, and applications for secondary coolants are discussed. Related information can be found in Chapters 2, 3, 20, 21, and 35 of the *ASHRAE Handbook—Fundamentals*.

## COOLANT SELECTION

A secondary coolant must be compatible with the 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 it 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, the viscosity of the coolant should be low enough to allow satisfactory heat transfer and reasonable pressure drop.

The vapor pressure of the coolant 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, to be accurate, the pumping rate in gallons per minute 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 the 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 the 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 greater the amount of turbulence and mixing of the bulk and film, the better the heat transfer and the higher the pressure drop. Where secondary coolant velocity in the tubes of a heat transfer device results in laminar flow, the 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. Since 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 the 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, the load and temperature range where energy is transferred, the pump head required by the system pressure drop (including that of the chiller), the mechanical efficiencies of the pump and driver, and the 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 head that usually exist. The secondary coolant temperature increases slightly from the energy expended at the pump shaft. If a semihermetic electric motor is used as the driver, the 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 (see 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.

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[c] | Pressure Drop,[a] psi | Heat Transfer Coefficient[b] $h_i$, Btu/h·ft$^2$·°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 |

[a]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. $H_D$ ($V^2\rho/2g$) for 7 fps velocity. Evaluations are at a bulk temperature of 20°F and a temperature range of 10°F.

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

[c]Based on inlet secondary coolant temperature at the pump of 25°F.

**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. The actual ID and length vary according to the 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 the same pump head, refrigeration load, 20°F average temperature, 10°F range, and the freezing point (for water-based secondary coolants) 20 to 23°F below the lowest secondary coolant temperature.

Higher pumping rates require larger coolant lines to keep the head and brake horsepower requirement for the pump within reasonable limits. Table 3 lists approximate ratios of pump power for secondary coolants. The 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.

### Other Considerations

Corrosion must be considered when selecting the coolant, an inhibitor, and the 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. The skin temperatures of the hottest elements determine the secondary coolant stability.

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

## DESIGN CONSIDERATIONS

The 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). The 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 would occur 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, Safety Code for Mechanical Refrigeration. The design standards for pressure piping can be found in ASME *Standard* B31.5, and the design standards for pressure vessels can be found 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 35 of the *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 the flow. Control valves that vary the 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 in excess of 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 the 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 the 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, the sizes of the pumps, and the 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 the connections and the 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 specific heat of the coolant $c_p$ is 0.791 Btu/lb·°F. At 28°F, the weight per unit volume of coolant at the pump ($\rho_L$) is 1.183 [(62.4 lb/ft$^3$)/(7.48 gal/ft$^3$)]; at 20°F, the $\rho_L$ is 1.185 [(62.4 lb/ft$^3$)/(7.48 gal/ft$^3$)].

Determine the minimum size storage tank for 90% use, the minimum capacity required for the chiller, and the 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 the low load condition, 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.

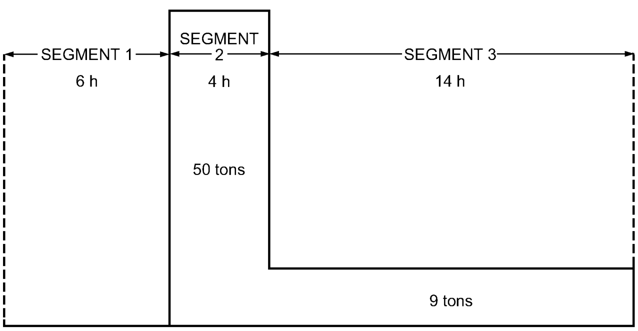


**Fig. 1 Load Profile of Refrigeration Plant Where Secondary Coolant Storage Can Save Energy**

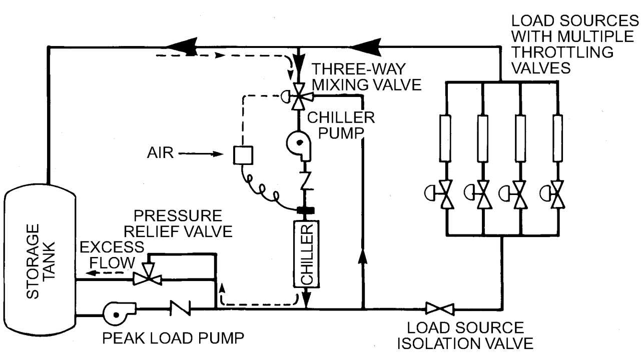


**Fig. 2 Arrangement of System with Secondary Coolant Storage**

**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:

$$\begin{aligned} (x-0)\times 6 + (x-50)\times 4 + (x-9)\times 14 &= 0 \\ 6x + 4x - 200 + 14x - 126 &= 0 \\ 24x &= 326 \\ x &= 13.58 \text{ tons} \end{aligned}$$

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

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

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 lb/min. The 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:

$$1106.2/[(1.185\times 62.4)/7.48] = 111.9 \text{ gpm}$$

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. Since 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

$$V = 294{,}987/[(1.185\times 62.4)/7.48] = 29{,}840 \text{ gal}$$

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, the pump heat added to the secondary coolant would be 3300 Btu/h 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, the 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, the flow is as shown by the dotted lines with the load source isolation valve closed. When the 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. The secondary coolant expansion tank volume is determined by considering the total coolant inventory and the differences in coolant density at the lowest temperature of coolant pumped to the load location ($t_1$) 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.

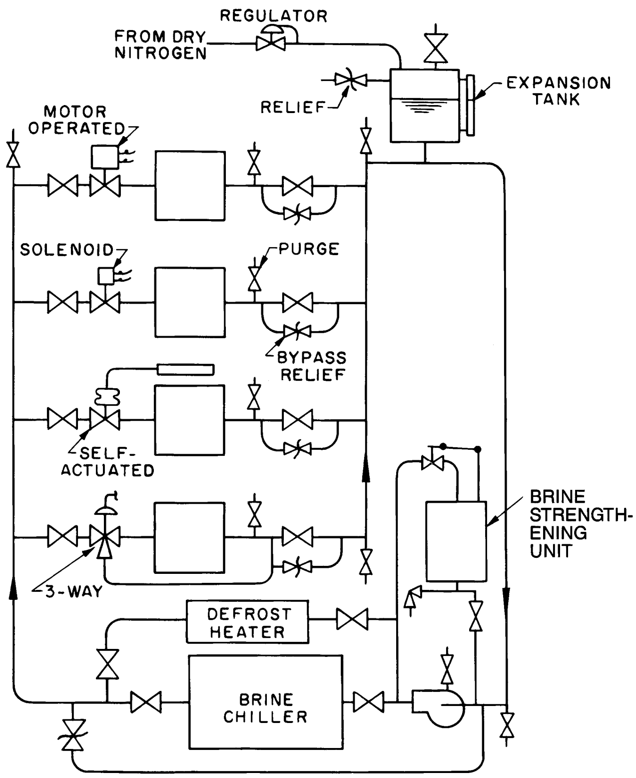


**Fig. 3 Typical Closed Salt Brine System**

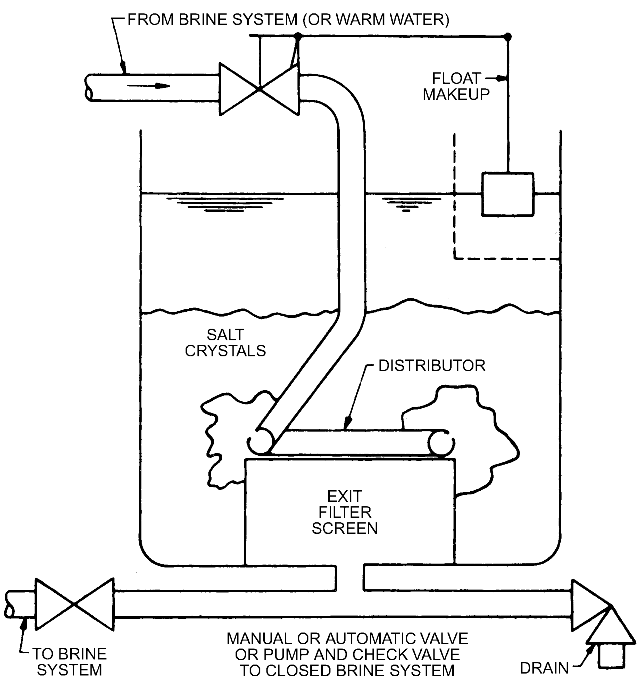


**Fig. 4 Brine Strengthening Unit for Salt Brines Used as Secondary Coolants**

**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.

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

*where*

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

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

## 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 a graphical solution, required heat extraction versus secondary coolant temperature is plotted. Then, by iteration, the 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 refrigeration system cited 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 the coolant temperature is further reduced with a constant 48.1 gpm, the capacity of the refrigeration system is gradually reduced until a 15 ton capacity is reached with 26°F coolant in the tank. Further cooling to 20°F will be 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 is reduced in temperature 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, the 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 the various secondary coolant temperatures:

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

| 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 |

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

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

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

*where*

$t$ = secondary coolant temperature, °F
$q$ = 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

$$\theta = \frac{19.67 \times 10^6}{(19.91 \times 12{,}000)} = 82.4 \text{ h}$$

This is a difference of over 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 the 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 ±1°F is made as the temperature is reduced. The polynomial equations may be solved by computer or hand-held calculator with a suitable program or spreadsheet. The time for pulldown will be 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 (1988) lists the installed cost of various projects. Park and Jackson (1984) and NBS (1978) discuss engineering and life-cycle cost analysis. Using the various time value of money formulas, the payback for storage tank handling of peak loads compared to large refrigeration equipment and higher energy costs can be evaluated. The 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 an alkaline to an acid condition. 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-type carcass chill rooms, cannot be closed.

A brine pH of 7.5 for a sodium or calcium chloride system is ideal, since it is safer to have a slightly alkaline rather than a slightly acid brine. Brine system 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 the pH, an inhibitor should be used. Generally, sodium dichromate is the most effective and economical for salt brine systems. The dichromate has a bright orange color, a granular form, and readily dissolves in warm water. Since 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. The quantities recommended are 125 lb/1000 $ft^3$ of calcium chloride brine, and 200 lb/1000 $ft^3$ of sodium chloride brine.

Adding sodium dichromate to the salt brine does not make it noncorrosive 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 the chromate concentration. Because the protection afforded by the 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 $Na_2Cr_2O_7 \cdot 2H_2O$); for sodium chloride brine, it is 12.128 gr/gal (as $Na_2Cr_2O_7 \cdot 2H_2O$).

Since crystals and concentrated solutions of sodium dichromate can cause severe skin rash, 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 the nonsalt secondary coolants described in this chapter are generally noncorrosive when used in systems for long periods, recommended inhibitors should be used, and a pH check should be performed occasionally.

Steel, iron, or copper piping should not be used to carry the 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 the 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 10 through 36). 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 the cooling, dehumidifying, and defrosting process, 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 *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. But the refrigerant is then subsequently flashed at a low pressure, with the resulting flash gas being drawn off to a compressor in the conventional manner.

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