บรรยายพิเศษในงานสัมมนาของ Master Refrigeration

เรื่อง

# แนวทางการออกแบบระบบ Industrial Refrigeration ที่ดีในโรงฟู้ดส์และห้องเย็น

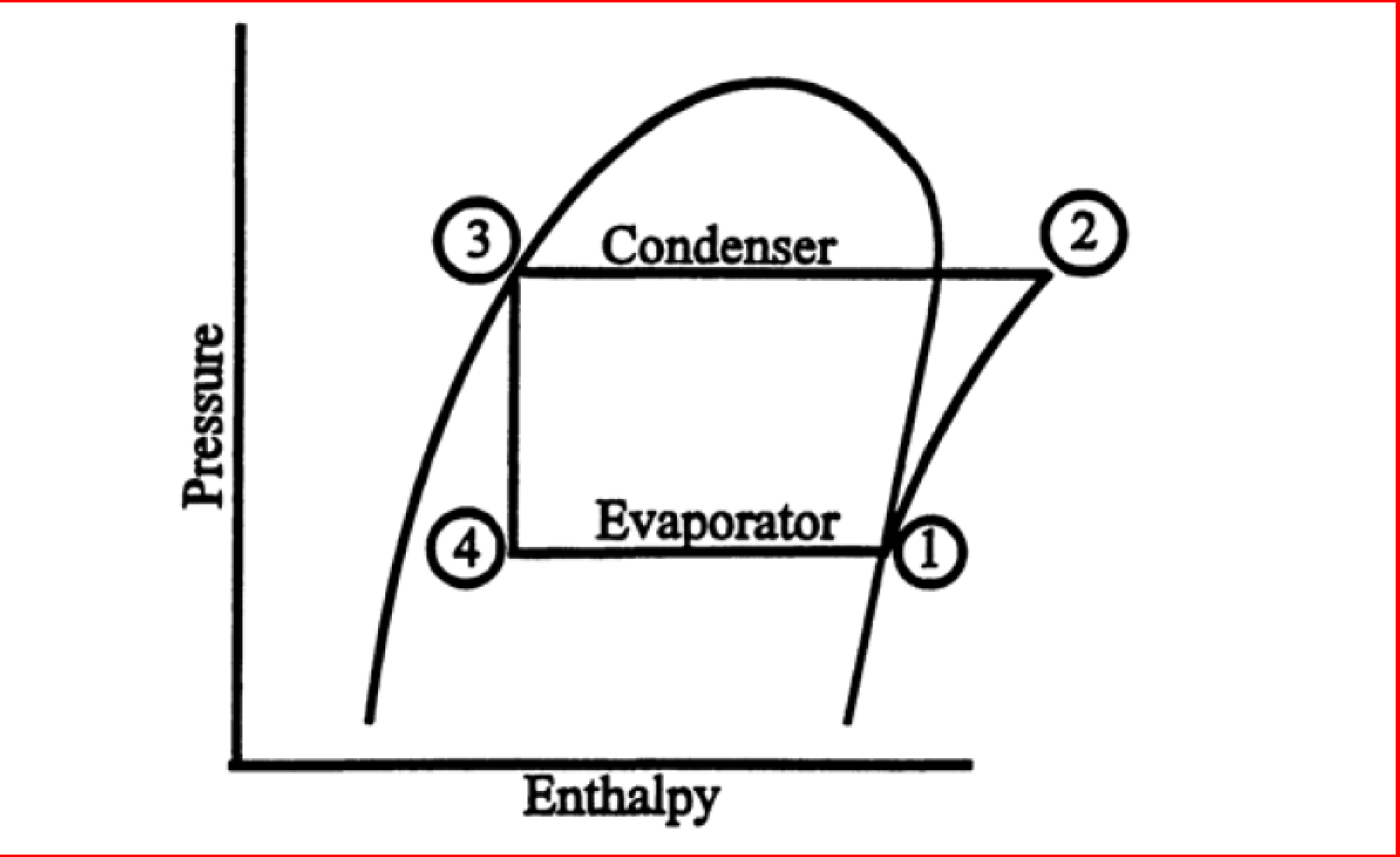
โดย

ผศ.ดร.ตุลย์ มณีวัฒนา
ภาควิชาวิศวกรรมเครื่องกล คณะวิศวกรรมศาสตร์ จุฬาลงกรณ์มหาวิทยาลัย
เขตปทุมวัน กรุงเทพฯ 10330
โทรศัพท์: 081-836-9582 E-mail: tul.m@chula.ac.th

# ประวัติวิทยากร

ผู้ช่วยศาสตราจารย์ ดร.ตุลย์ มณีวัฒนา เป็นวุฒิวิศวกรในสาขาวิศวกรรมเครื่องกลที่มีประสบการณ์ในการออกแบบระบบปรับอากาศ ระบายอากาศ และระบบเครื่องทำความเย็นขนาดใหญ่ ในอาคารและโรงงานต่างๆ มากว่า 30 ปี ท่านเป็นอาจารย์ที่ภาควิชาวิศวกรรมเครื่องกล คณะวิศวกรรมศาสตร์ จุฬาลงกรณ์มหาวิทยาลัย มาตั้งแต่ปี พศ. 2523 รายวิชาที่ท่านสอนอยู่ส่วนมากคือรายวิชาการทำความเย็นและการปรับอากาศสำหรับนักศึกษาในภาควิชาวิศวกรรมเครื่องกลชั้นปีที่ 4 และรายวิชาการทำความเย็นและการปรับอากาศขั้นสูงสำหรับนักศึกษาในระดับปริญญาโท/เอก ตำแหน่งต่างๆที่ท่านเคยดำรงมาในอดีตที่สำคัญก็มี อาทิเช่น หัวหน้าภาควิชาวิศวกรรมเครื่องกล คณะวิศวกรรมศาสตร์ จุฬาลงกรณ์มหาวิทยาลัย นายกสมาคมวิศวกรรมปรับอากาศแห่งประเทศไทย (ACAT) กรรมการบริหาร ASHRAE Thailand Chapter และ ที่ปรึกษาของสมาคมเครื่องทำความเย็นไทย (TRA) นอกจากนั้นแล้ว ในปีที่ผ่านมา ท่านก็ยังได้รับเลือกจากสมาคมวิศวกรรมปรับอากาศแห่งประเทศไทย ให้เป็น วิศวกรปรับอากาศดีเด่นประจำปี 2560 อีกด้วย

# Refrigeration Cycle



# Generic Refrigeration System

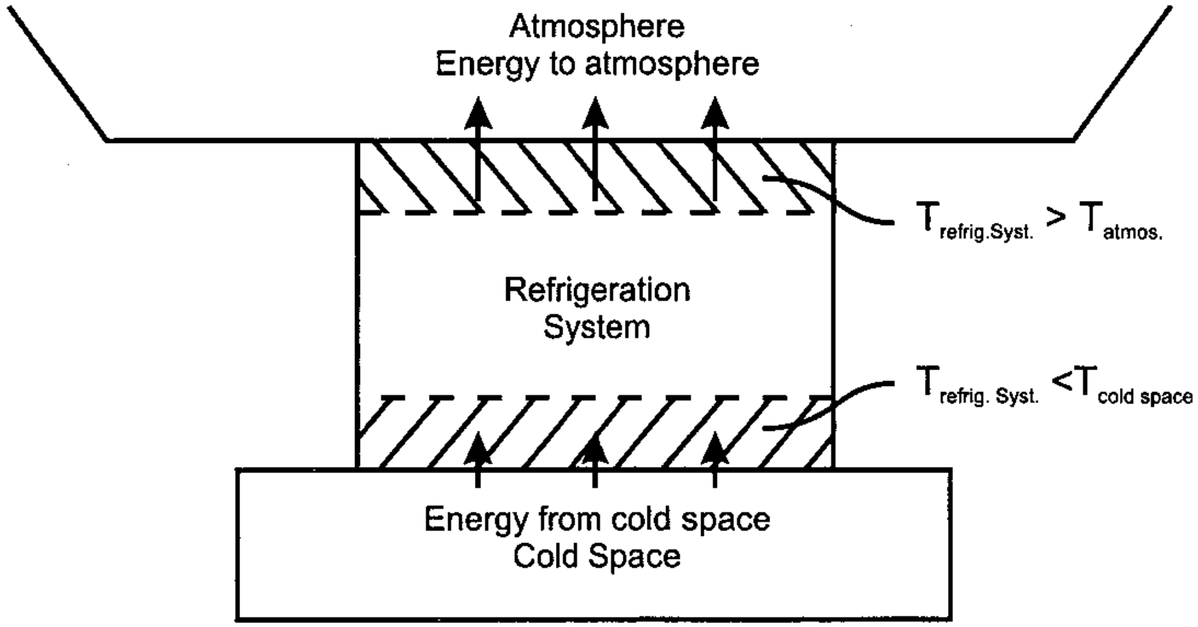
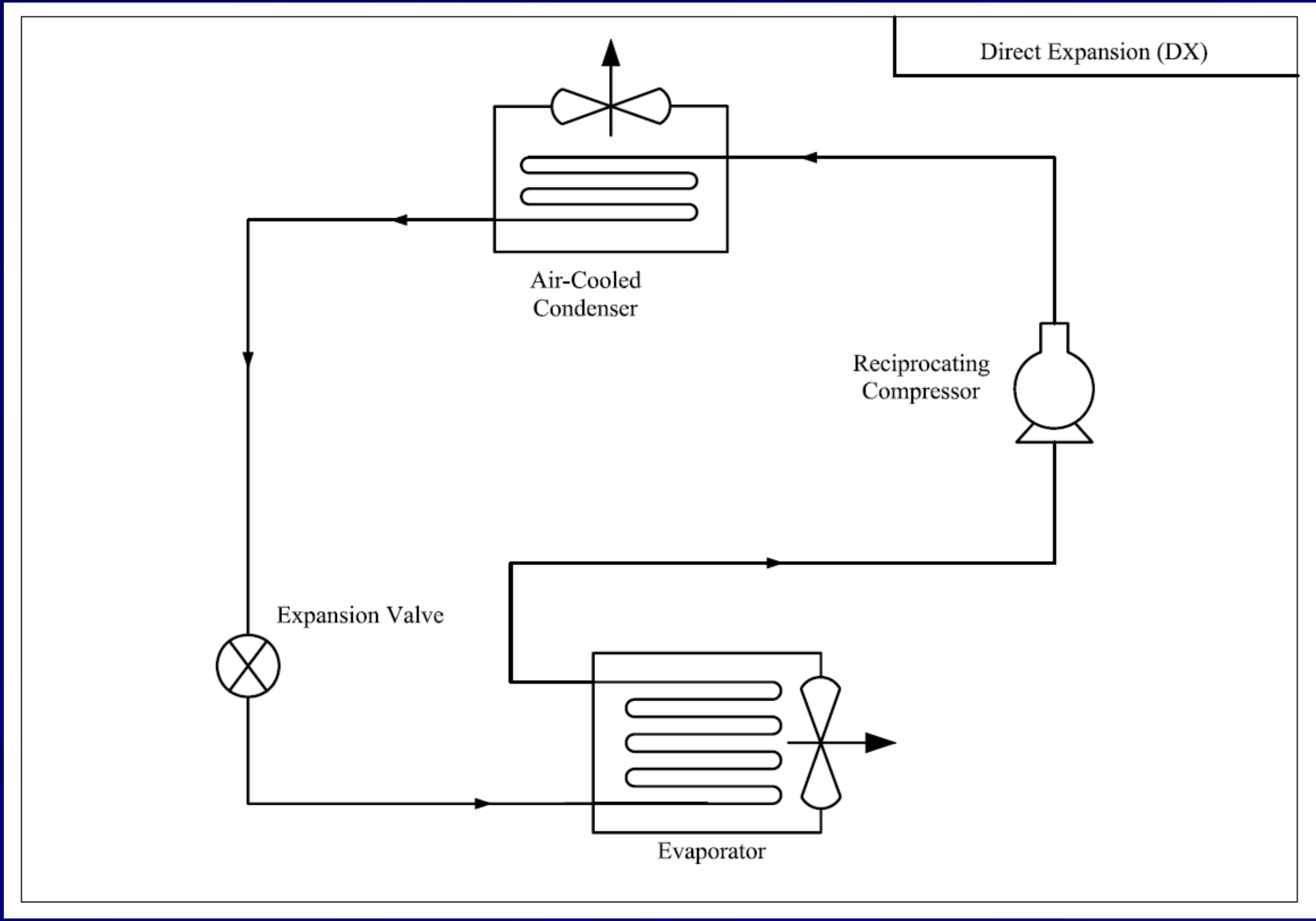
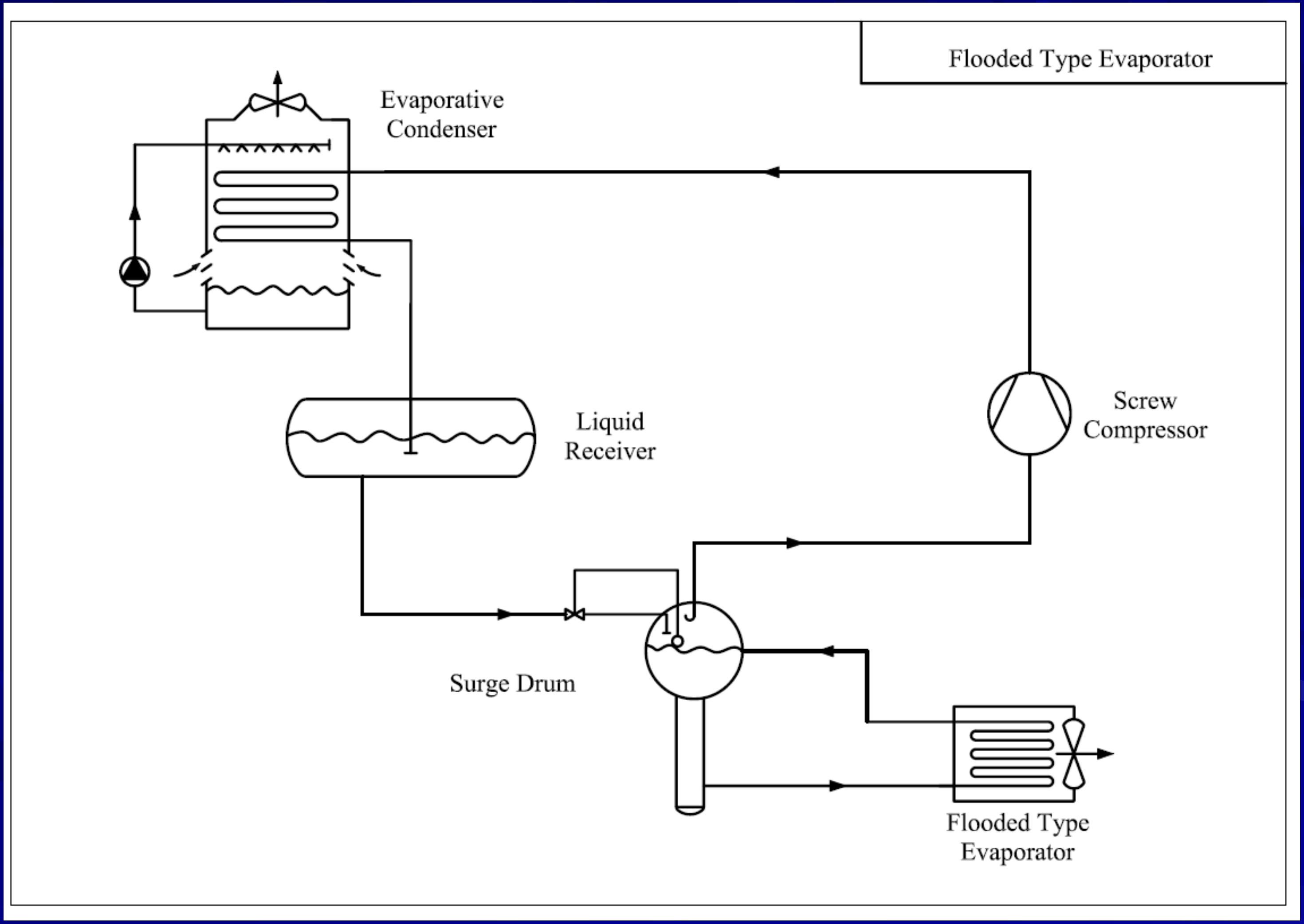



**Figure 1-1. Generic Refrigeration System**

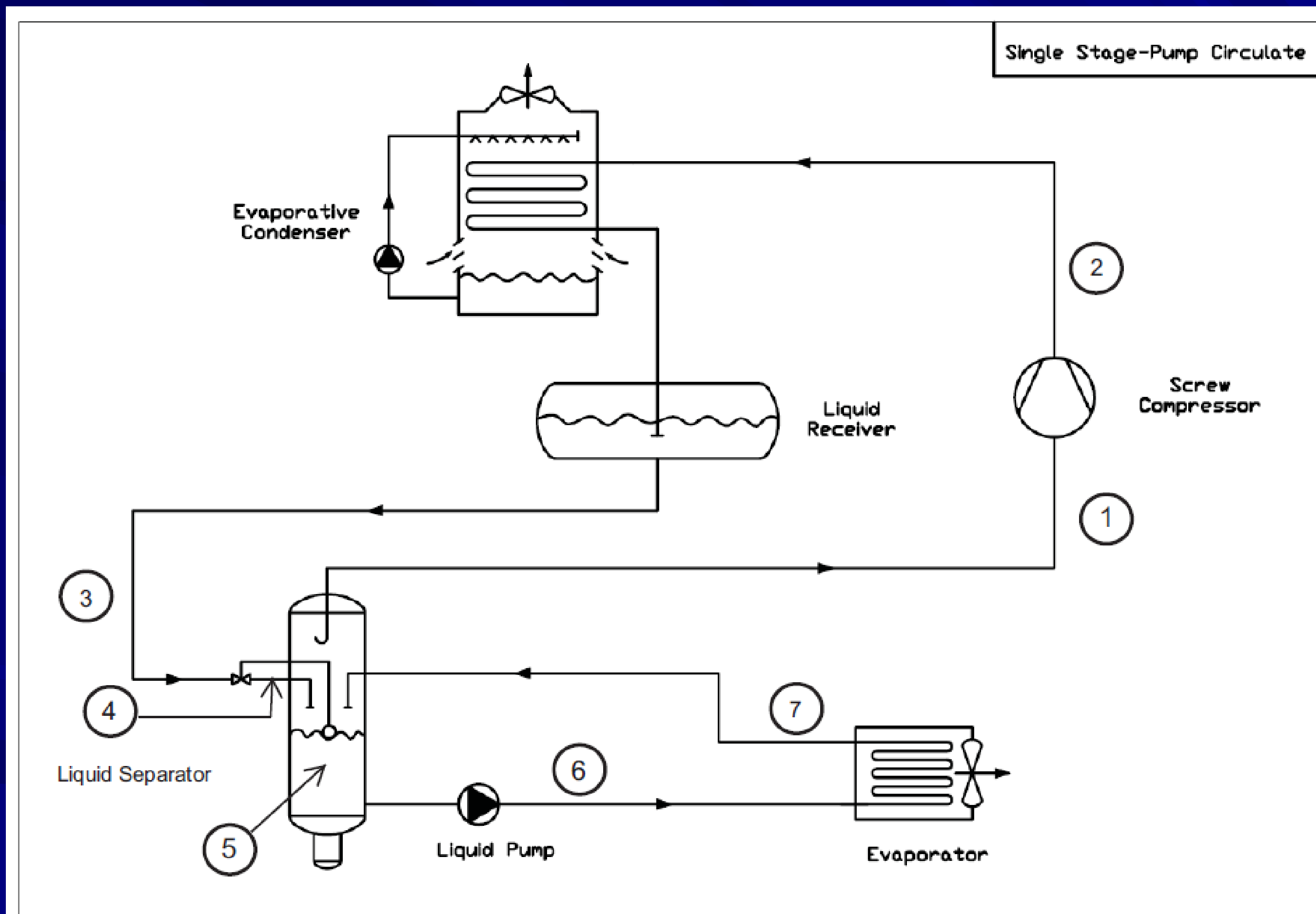
# DX Evaporator



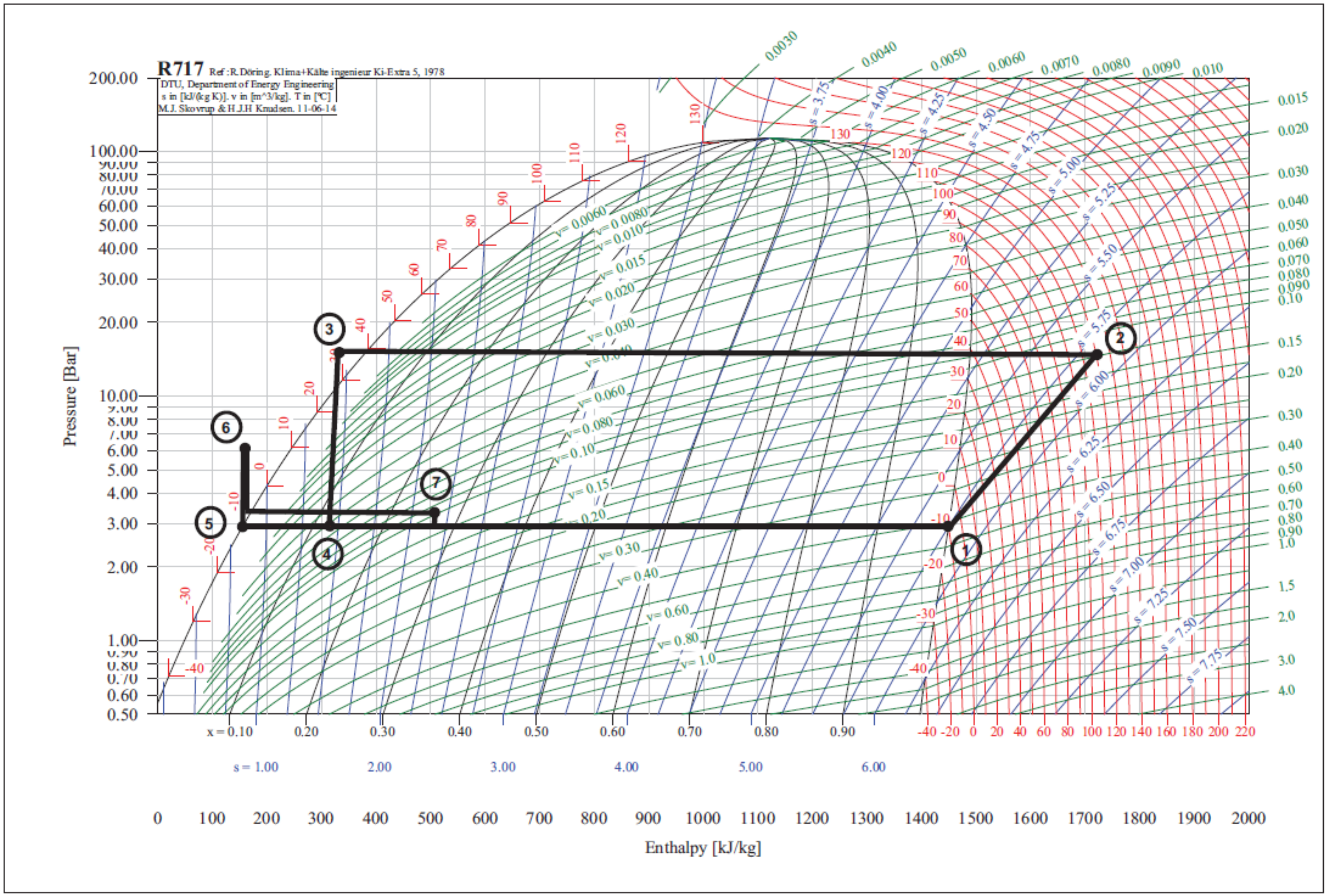
# Flooded Evaporator



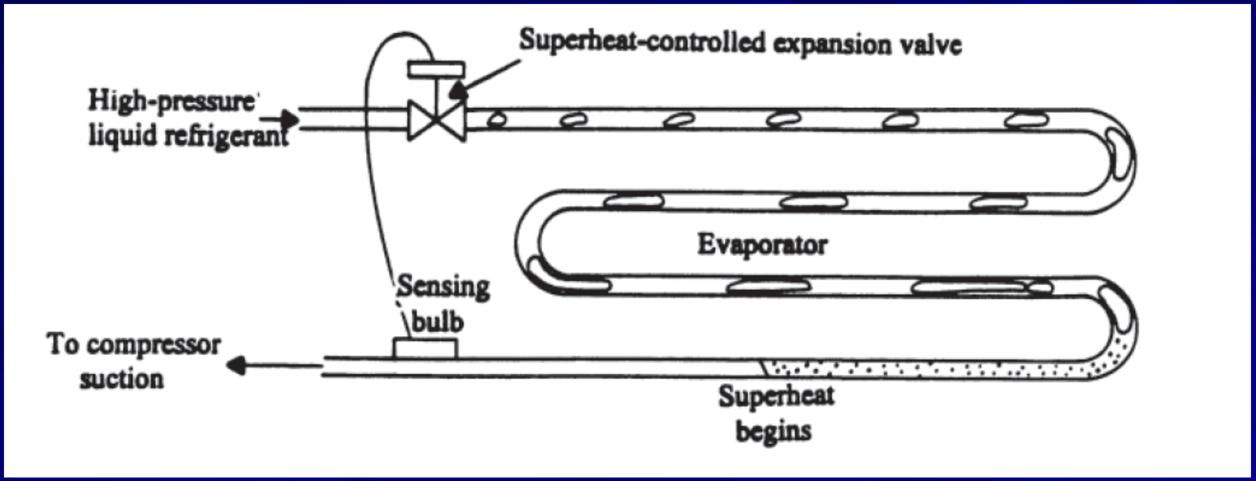
# Schematic Diagram of Pump-circulate Evaporator

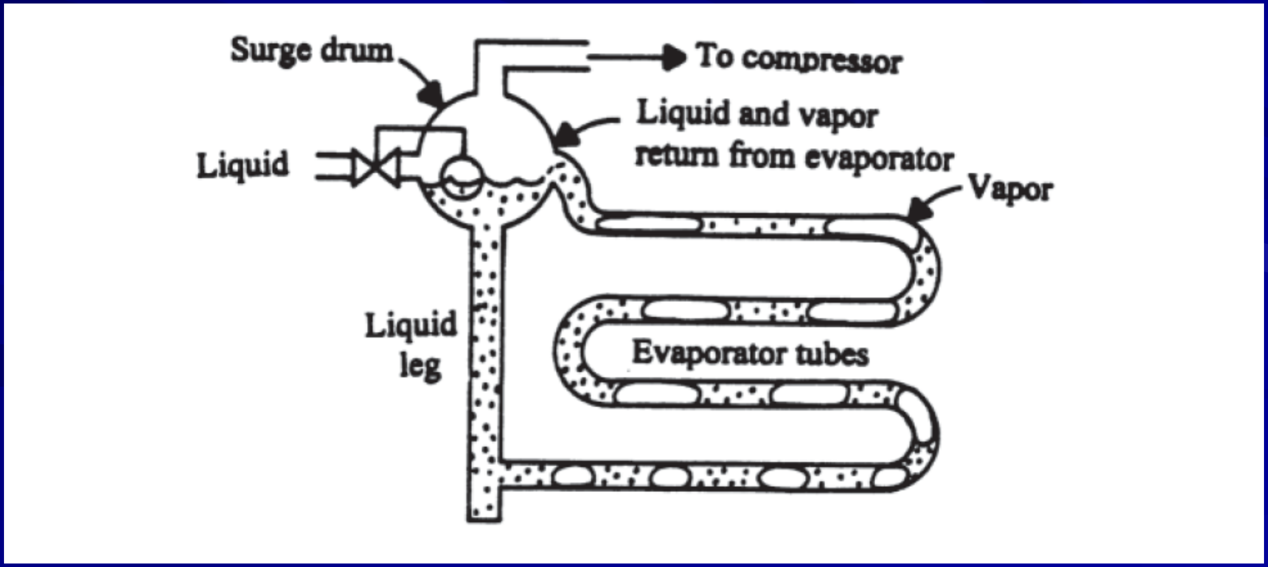


# P-h Diagram of Pump-circulate Evaporator



# DX VS Flooded Coil



# Liquid Subcooling

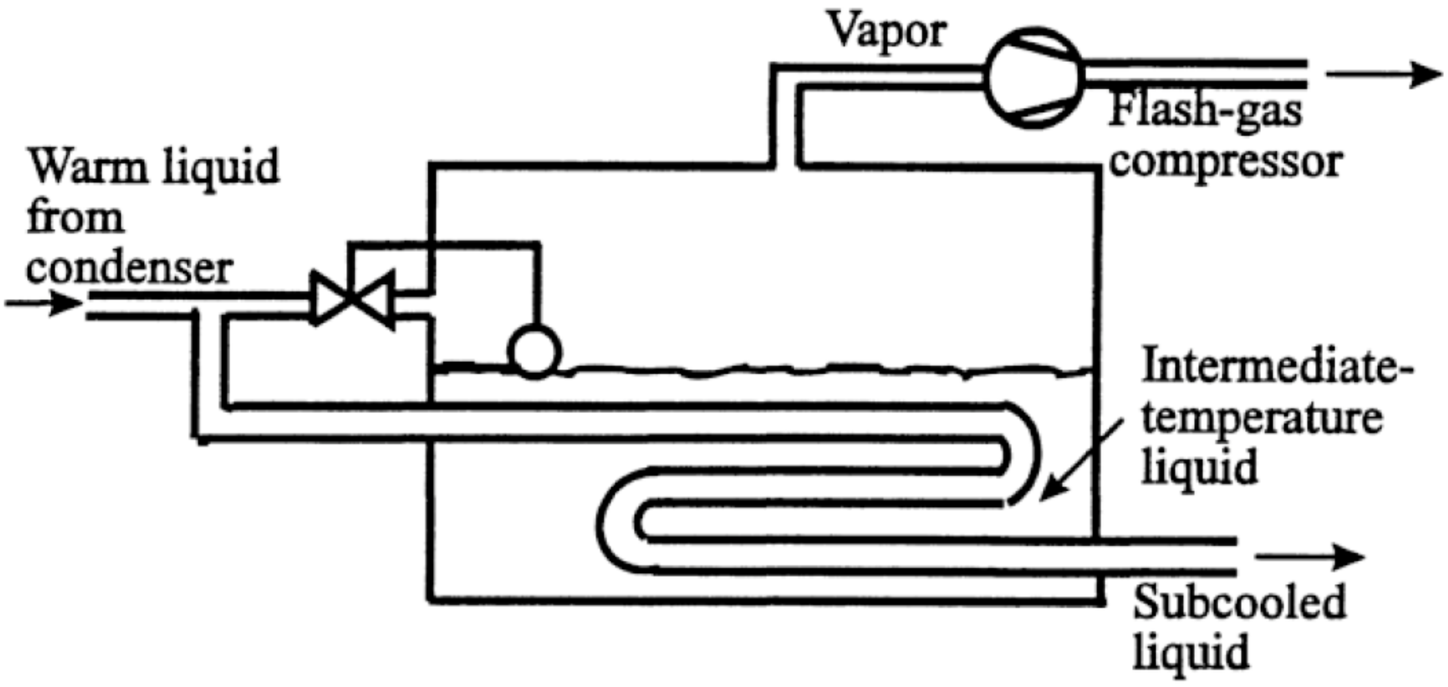
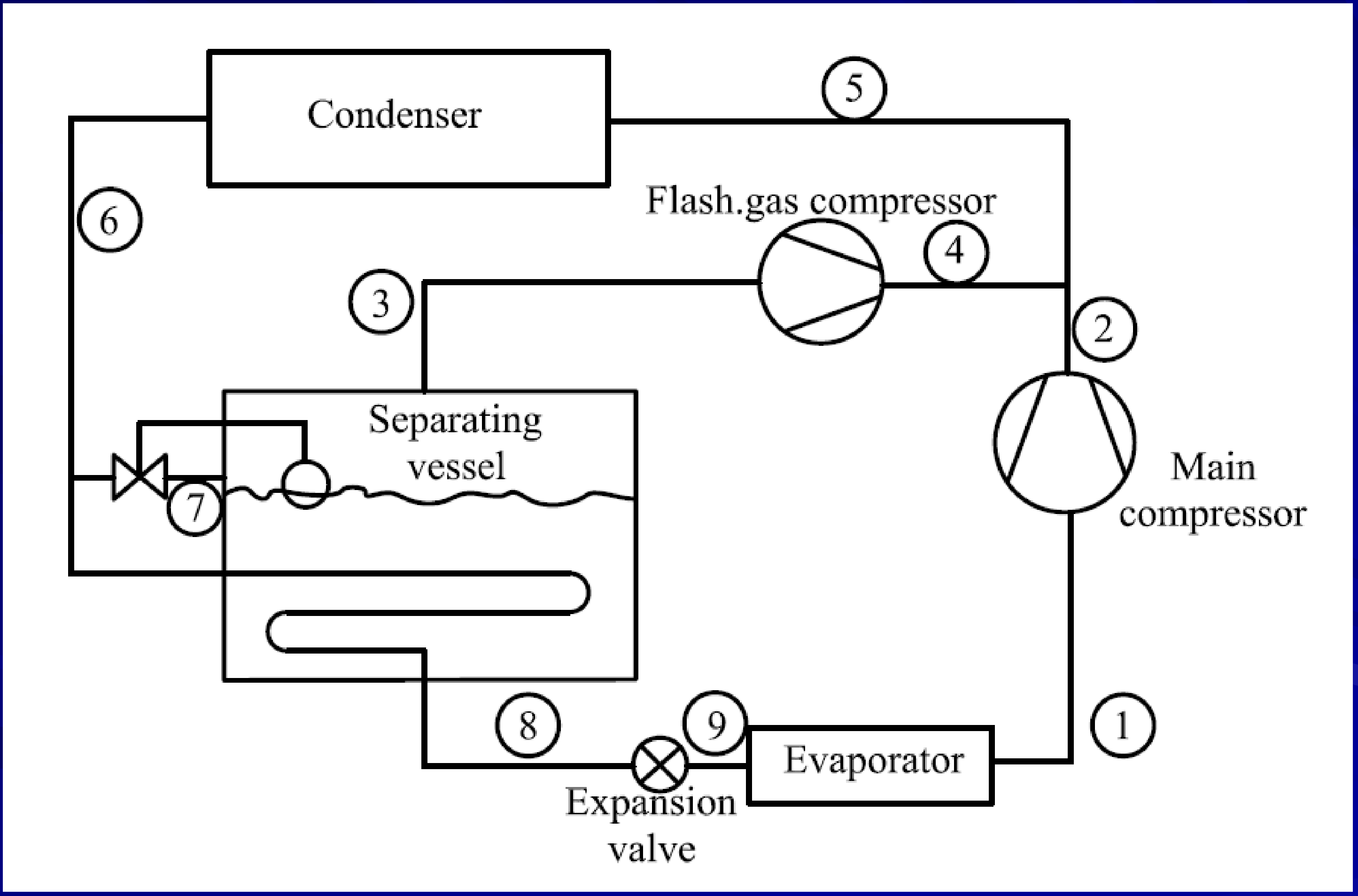
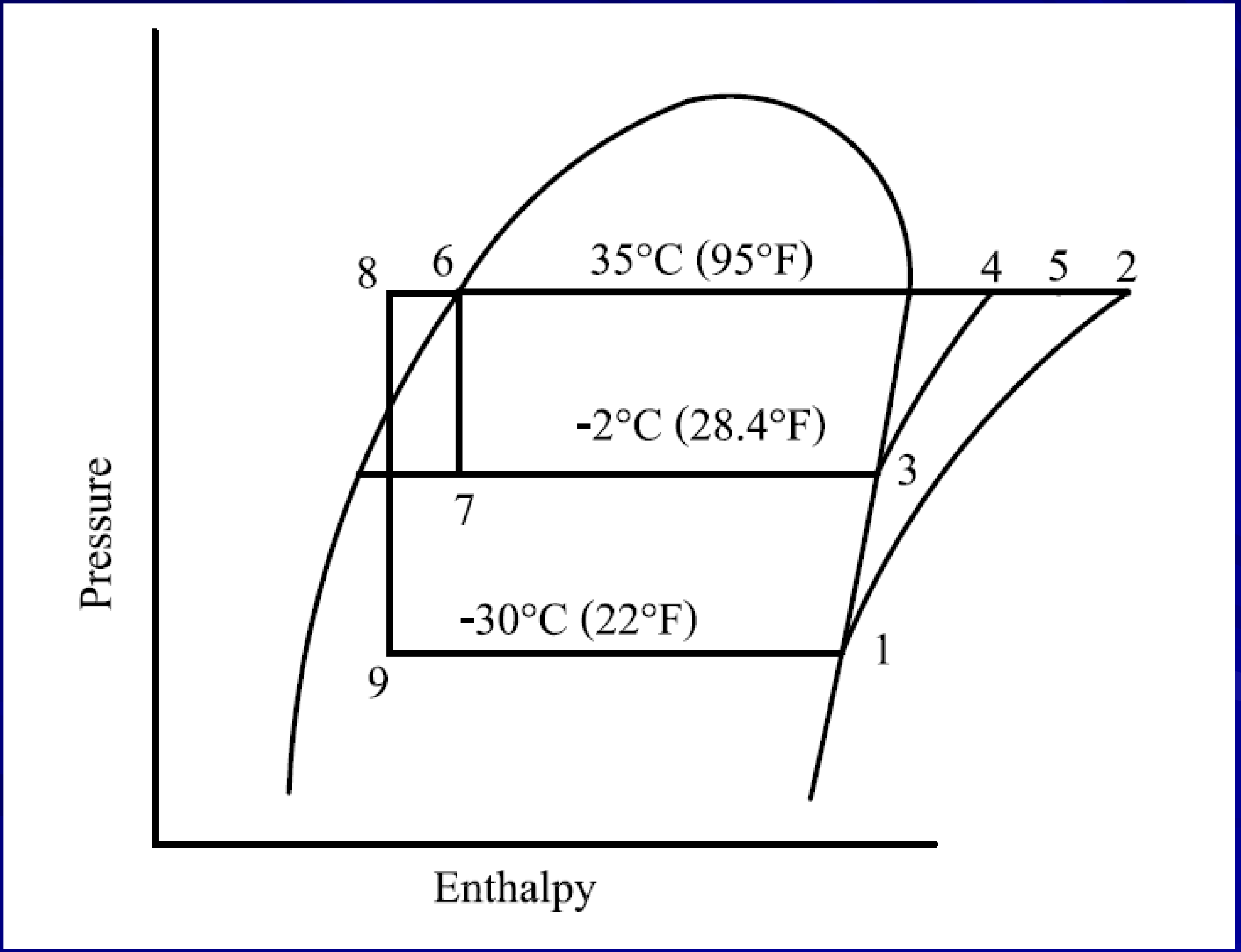



**FIGURE 3.5**
A liquid subcooler using a coil immersed in the liquid of an intermediate-pressure vessel.

# Liquid Subcooling



# P-h Diagram of Liquid Subcooling



# Standard Two-stage System with One Evaporating Temperature

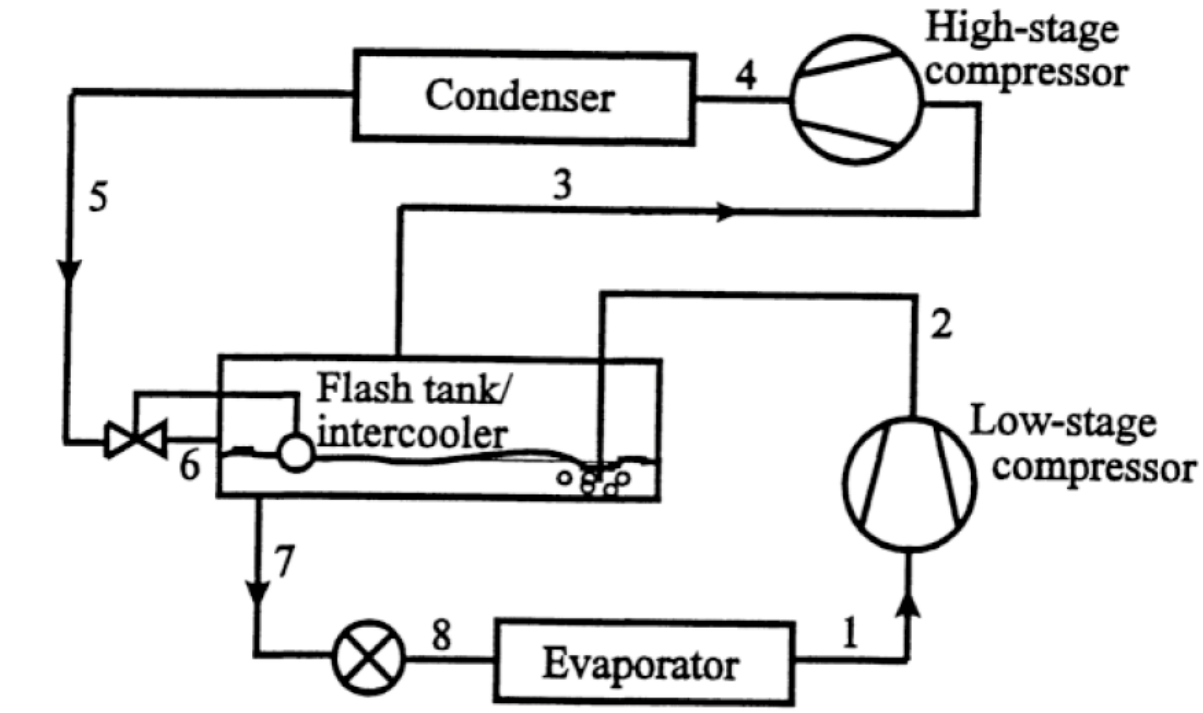



**FIGURE 3.15**
A standard two-stage system with flash-gas removal and intercooling.

# P-h Diagram for Standard Two-stage System

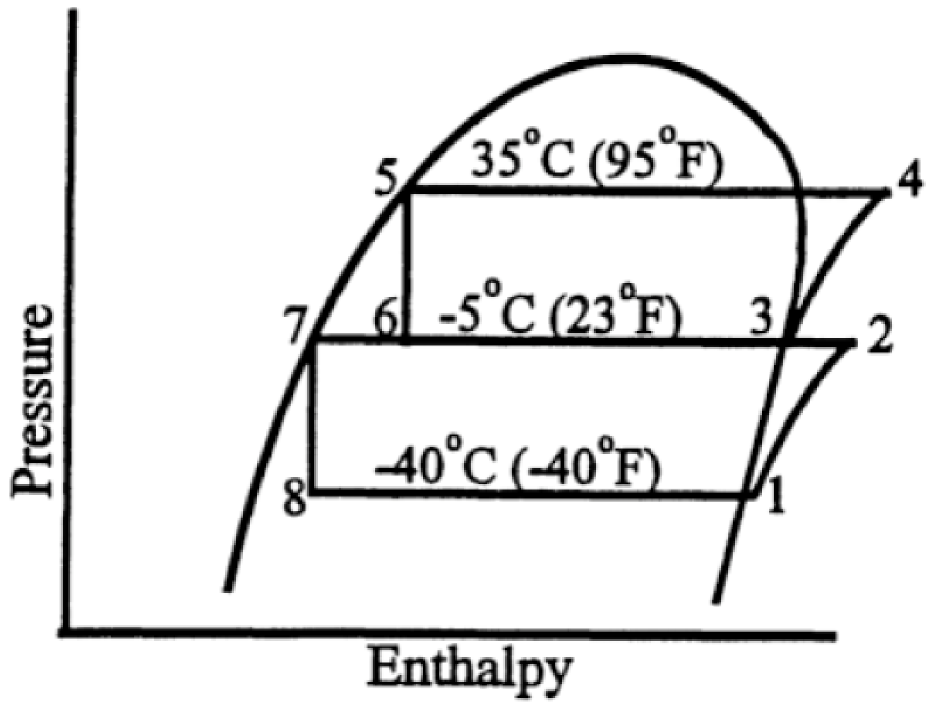
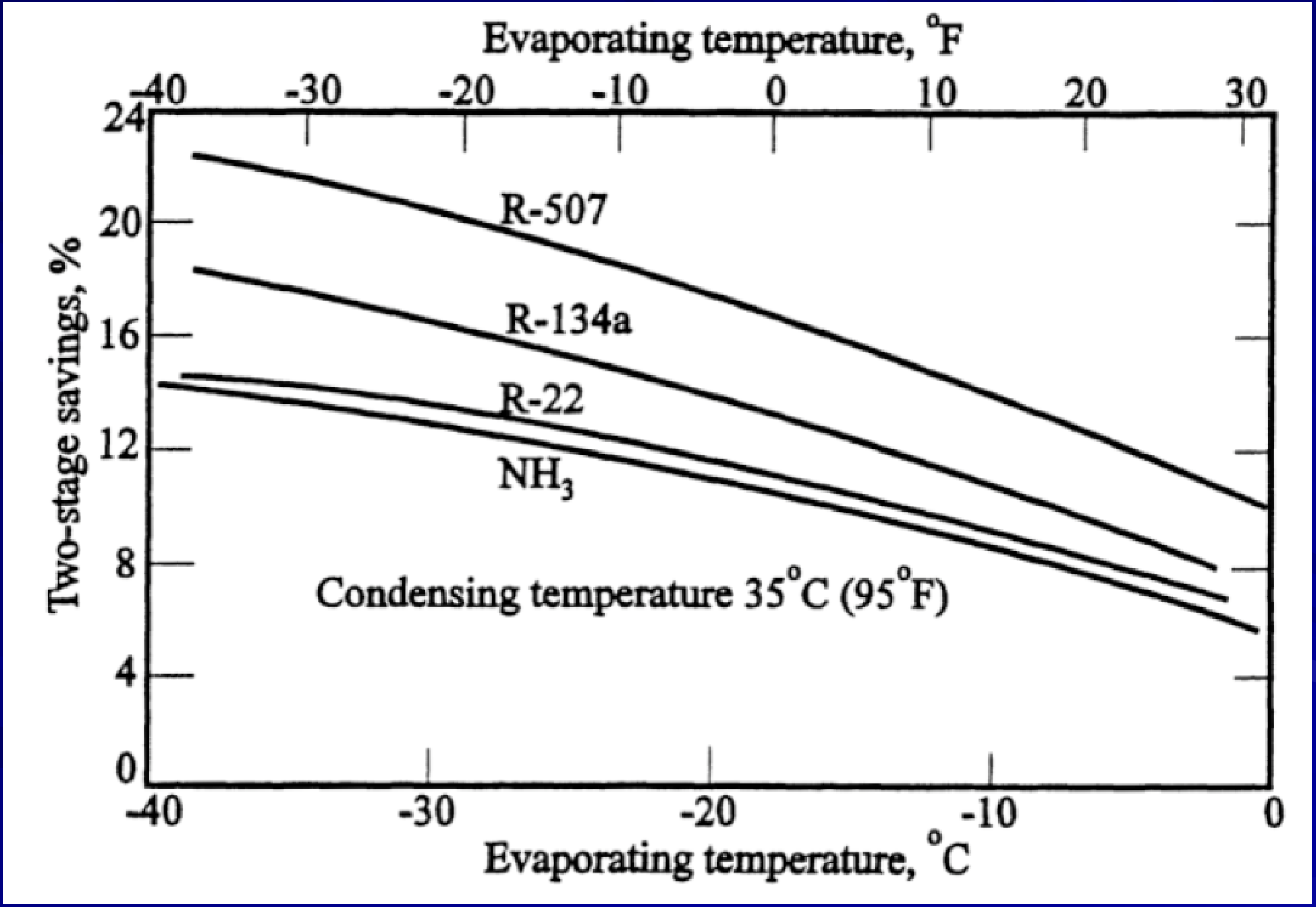


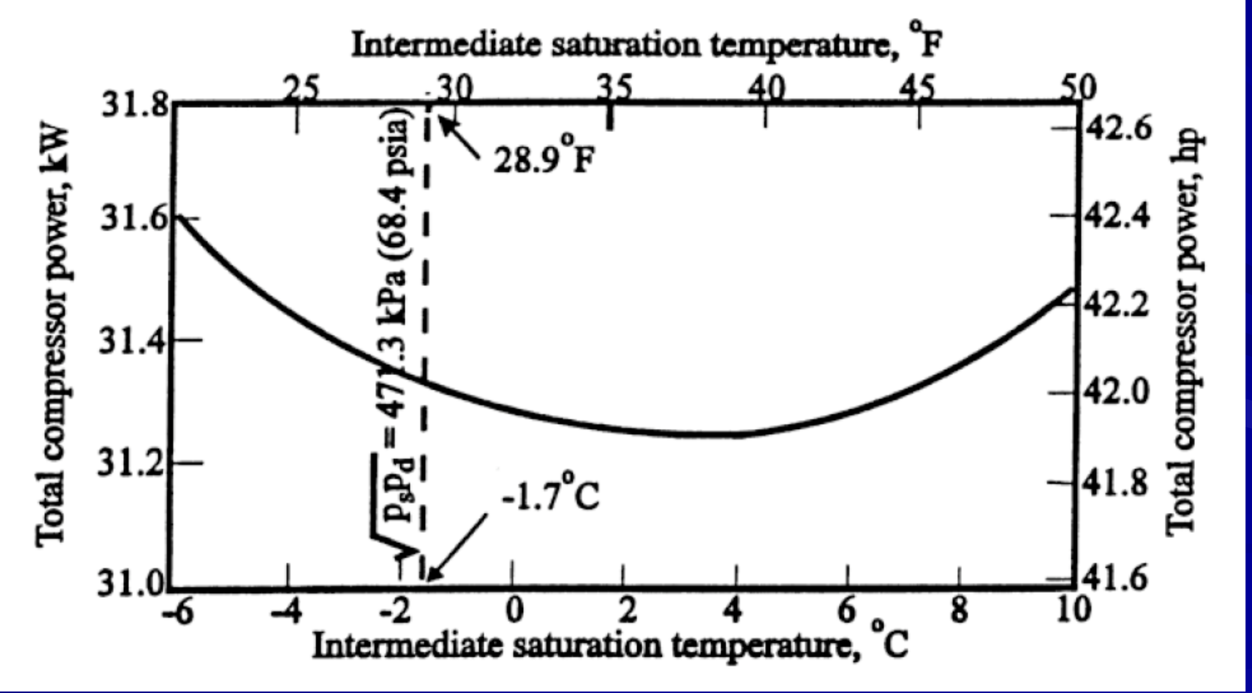

**FIGURE 3.16**
Pressure-enthalpy diagram for the standard two-stage system of Figure 3.15.

# Saving of a Standard Two-stage

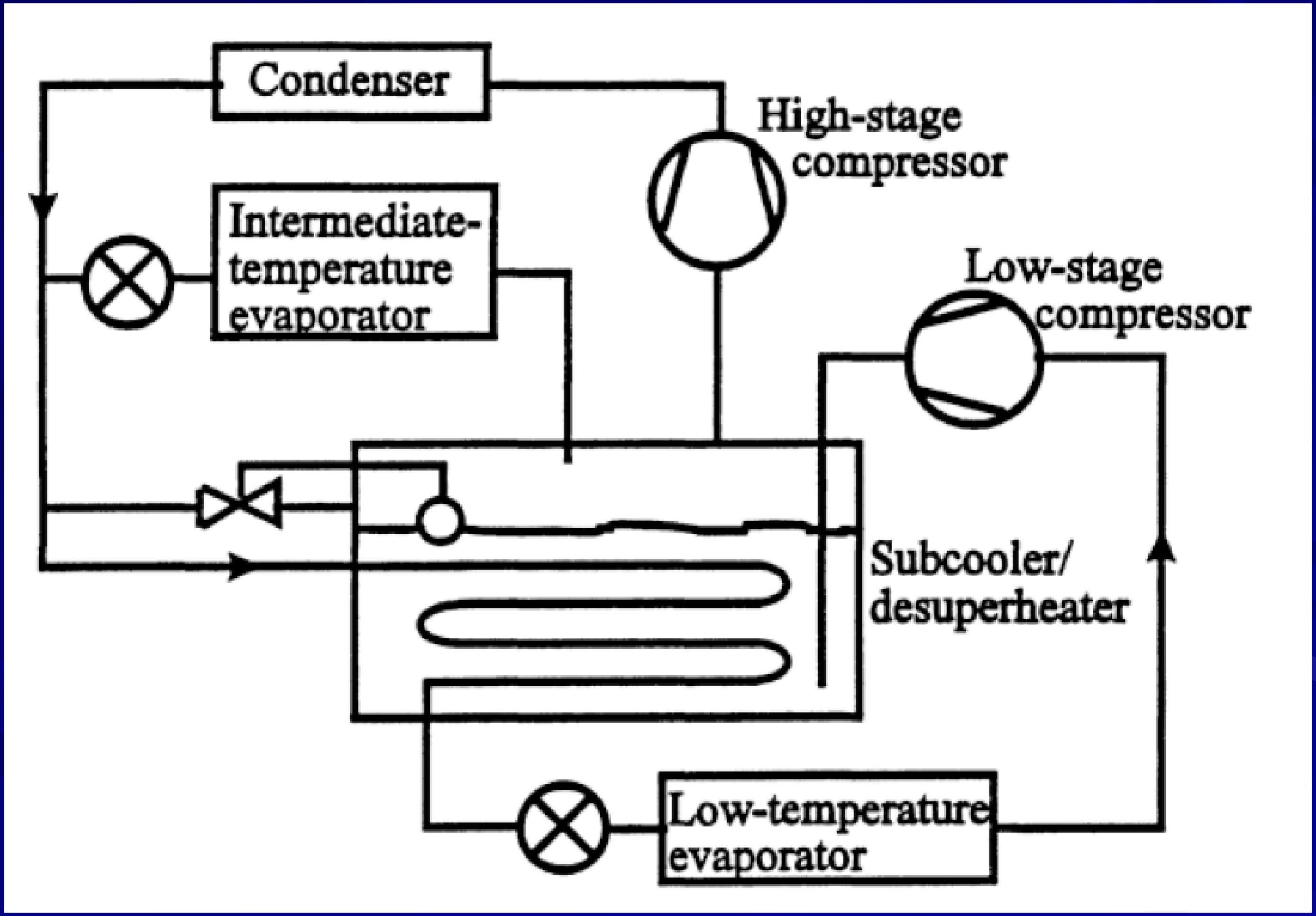


# Optimum Intermediate Pressure

$$p_{intermediate} = \sqrt{(p_{suction})(p_{discharge})} \tag{3.6}$$



# Two-stage with Both Low and Medium-Temp. Evaporators



# Side-port Economizer

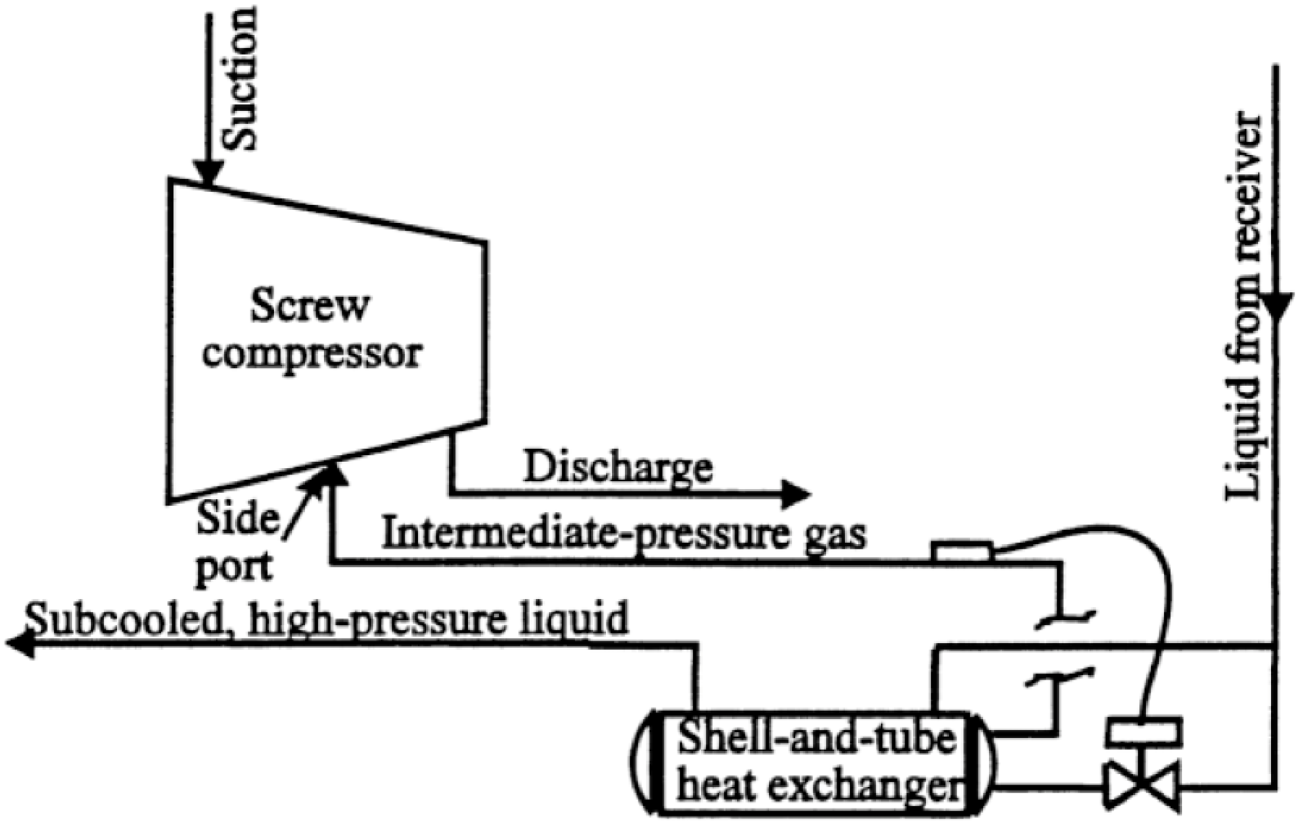
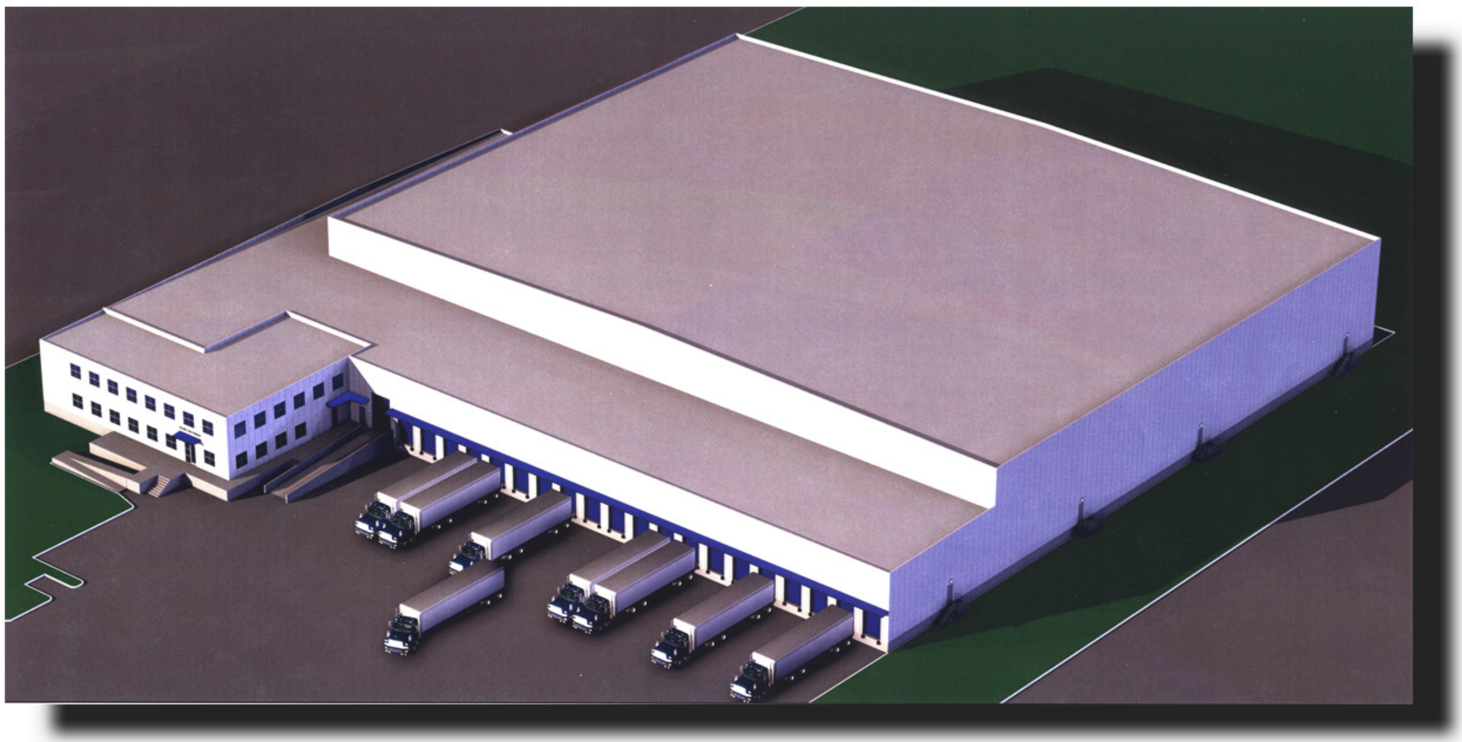



**FIGURE 3.24**
Using the side port of a screw compressor as an economizer to subcool liquid.

# When to Use Two-Stage

1. When Evaporator Temperature drop below -10C
2. When the Pressure Ratio is more than 8 for Reciprocating Compressor with $NH_3$

# Refrigeration Load Calculations



# Load Components

1) Transmission Load
2) Product Load
3) Internal Load
4) Infiltration Air Load
5) Equipment-related Load

# Insulation Thickness

ตาราง ความหนาของฉนวนสำหรับห้องเย็น

| สำหรับอุณหภูมิห้อง, °C | ฉนวนสไตโรโฟม K = 0.038 W/(m.K) | | ฉนวนโปลียูริเทน K = 0.025 W/(m.K) | |
|---|---|---|---|---|
| | mm | นิ้ว | mm | นิ้ว |
| -45 ถึง -36 | 250 | 10 | 150 | 6 |
| -35 ถึง -26 | 250 | 10 | 125 | 5 |
| -25 ถึง -18 | 200 | 8 | 100 | 4 |
| -17 ถึง -9 | 175 | 7 | 100 | 4 |
| -8 ถึง -4 | 150 | 6 | 75 | 3 |
| -3 ถึง 4 | 125 | 5 | 75 | 3 |
| 5 ถึง 10 | 100 | 4 | 50 | 2 |
| 11 ถึง 16 | 75 | 3 | 50 | 2 |

# Infiltration by Air Exchange

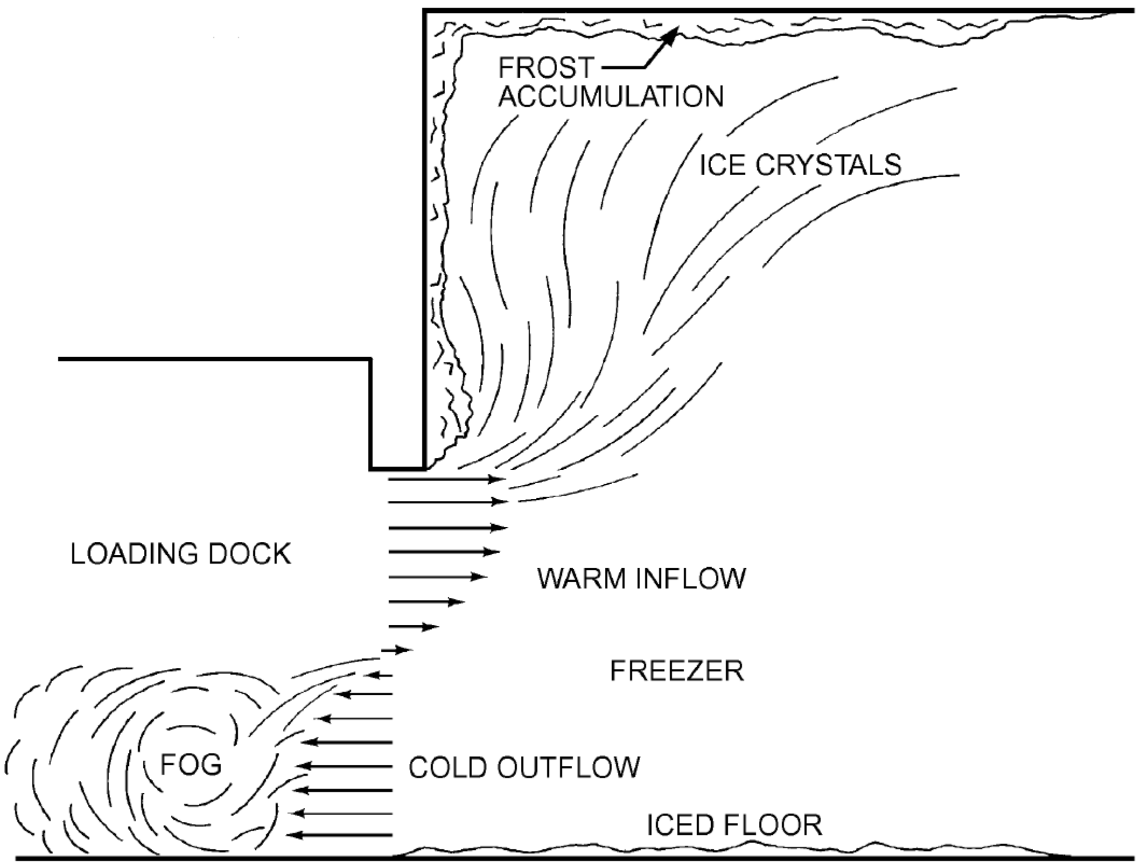



**Fig. 3 Flowing Cold and Warm Air Masses for Typical Open Freezer Doors**

# Air Curtain

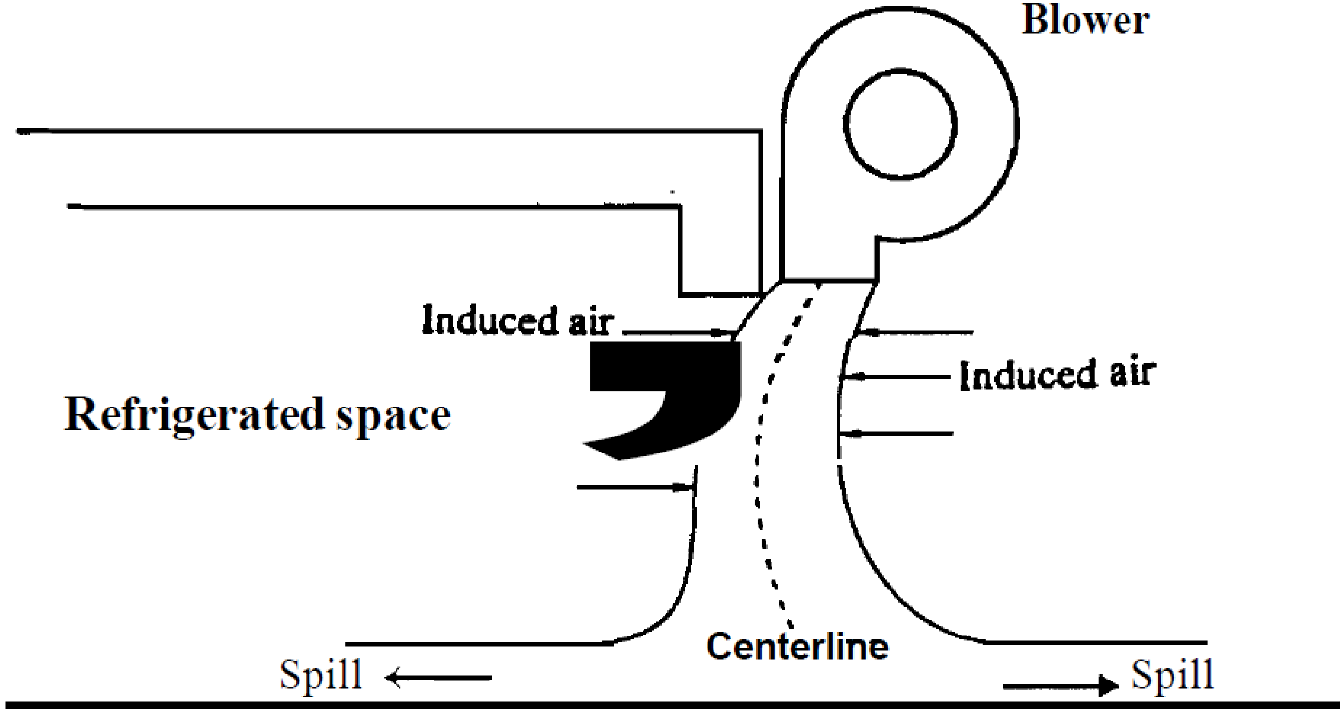



**FIGURE 18.11**
An air curtain resisting infiltration through an open door.

# Door Heat Gain

Heat gain through doorways from air exchange is as follows:

$$q_t = qD_tD_f(1 - E) \quad (11)$$

*where*

$q_t$ = average heat gain for the 24 h or other period, kW
$q$ = sensible and latent refrigeration load for fully established flow, kW
$D_t$ = doorway open-time factor
$D_f$ = doorway flow factor
$E$ = effectiveness of doorway protective device

# Door Heat Gain

Gosney and Olama (1975) developed the following air exchange equation for fully established flow:

$$q = 0.221A(h_i - h_r)\rho_r(1 - \rho_i/\rho_r)^{0.5}(gH)^{0.5}F_m \tag{12}$$

*where*

$q$ = sensible and latent refrigeration load, kW
$A$ = doorway area, $m^2$
$h_i$ = enthalpy of infiltration air, kJ/kg
$h_r$ = enthalpy of refrigerated air, kJ/kg
$\rho_i$ = density of infiltration air, $kg/m^3$
$\rho_r$ = density of refrigerated air, $kg/m^3$
$g$ = gravitational constant = 9.81 $m/s^2$
$H$ = doorway height, m
$F_m$ = density factor

# Door Heat Gain

$$F_m = \left[ \frac{2}{1 + (\rho_r / \rho_i)^{1/3}} \right]^{1.5} \qquad (13)$$

For cyclical, irregular, and constant door usage, alone or in combination, the doorway open-time factor can be calculated as

$$D_t = \frac{\left( P\theta_p + 60\theta_o \right)}{3600\theta_d} \qquad (15)$$

*where*

$D_t$ = decimal portion of time doorway is open
$P$ = number of doorway passages
$\theta_p$ = door open-close time, seconds per passage
$\theta_o$ = time door simply stands open, min
$\theta_d$ = daily (or other) time period, h

# Door Open-Close Time

The typical time $\theta_p$ for conventional pull-cord-operated doors ranges from 15 to 25 s per passage. The time for high-speed doors ranges from 5 to 10 s, although it can be as low as 3 s. The time for $\theta_o$ and $\theta_d$ should be provided by the user. Hendrix et al. (1989) found that steady-state flow becomes established 3 s after the cold-room door is opened. This fact may be used as a basis to reduce $\theta_p$ in Equation (15), particularly for high-speed doors, which may significantly reduce infiltration.

# Doorway Flow Factor

The doorway flow factor $D_f$ is the ratio of actual air exchange to fully established flow. Fully established flow occurs only in the unusual case of an unused doorway standing open to a large room or to the outdoors, and where cold outflow is not impeded by obstructions (e.g., stacked pallets in or adjacent to the flow path in or outside the cold room). Under these conditions, $D_f$ is 1.0.

Hendrix et al. (1989) found that a flow factor $D_f$ of 0.8 is conservative for a 16 K temperature difference when traffic flow equals one entry and exit per minute through fast-operating doors. Downing and Meffert (1993) found a flow factor of 1.1 at temperature differences of 7 and 10 K. Based on these results, the recommended flow factor for cyclically operated doors with temperature differentials less than 11°C is 1.1, and the recommended flow factor for higher differentials is 0.8.

# Effectiveness of Protective Devices

The effectiveness $E$ of open-doorway protective devices is 0.95 or higher for newly installed strip, fast-fold, and other non-tight-closing doors. However, depending on the traffic level and door maintenance, $E$ may quickly drop to 0.8 on freezer doorways and to about 0.85 for other doorways. Airlock vestibules with strip or push-through doors have an effectiveness ranging between 0.95 and 0.85 for freezers and between 0.95 and 0.90 for other doorways. The effectiveness of air curtains ranges from very poor to more than 0.7. For a wide-open door with no devices, $E = 0$ in Equation (11).

# Loading Dock

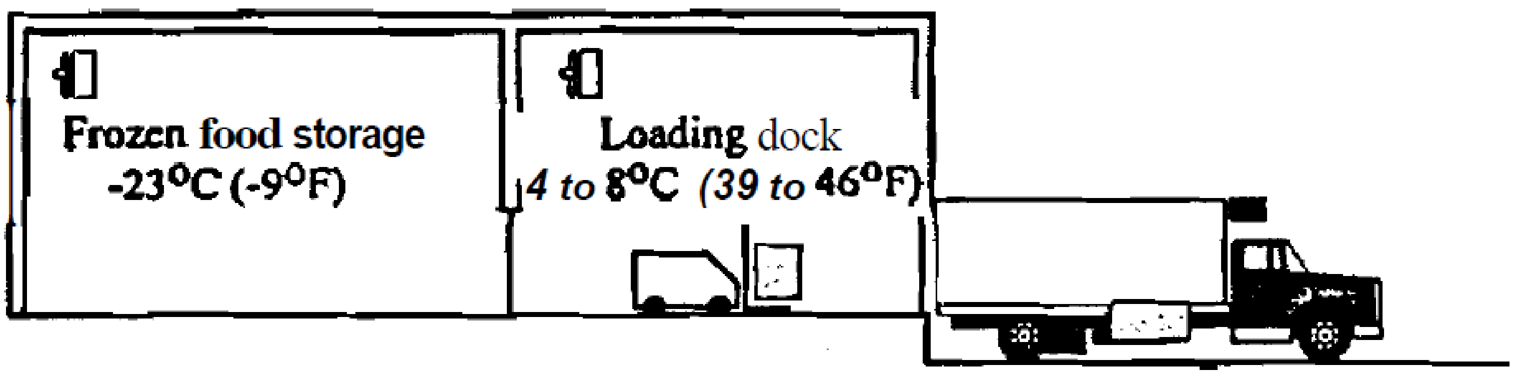
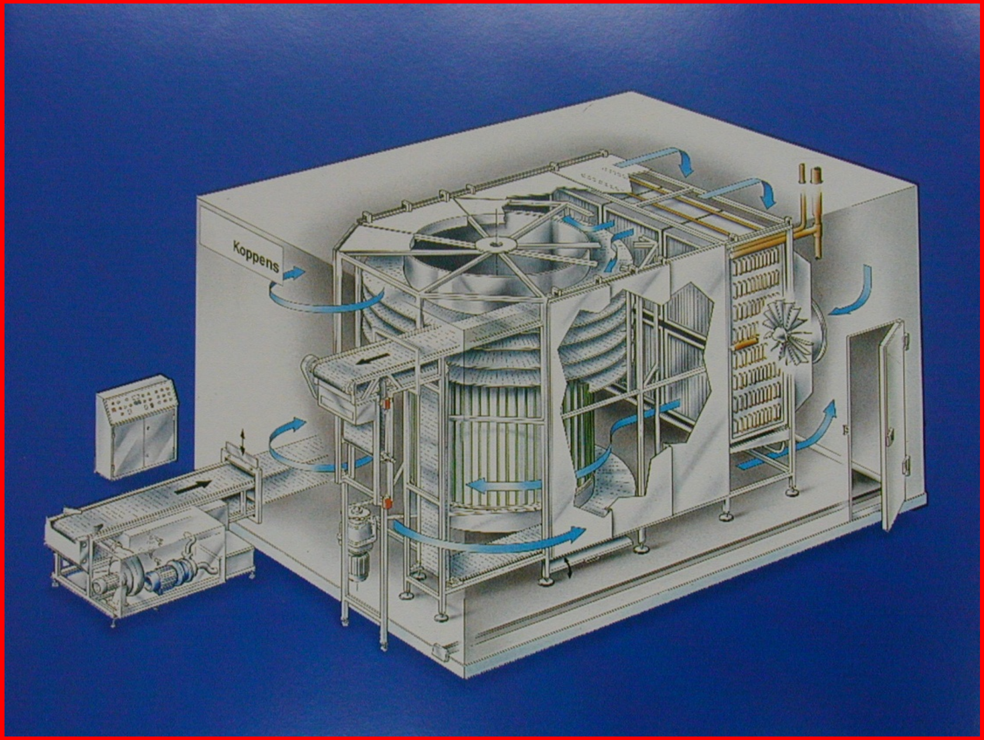
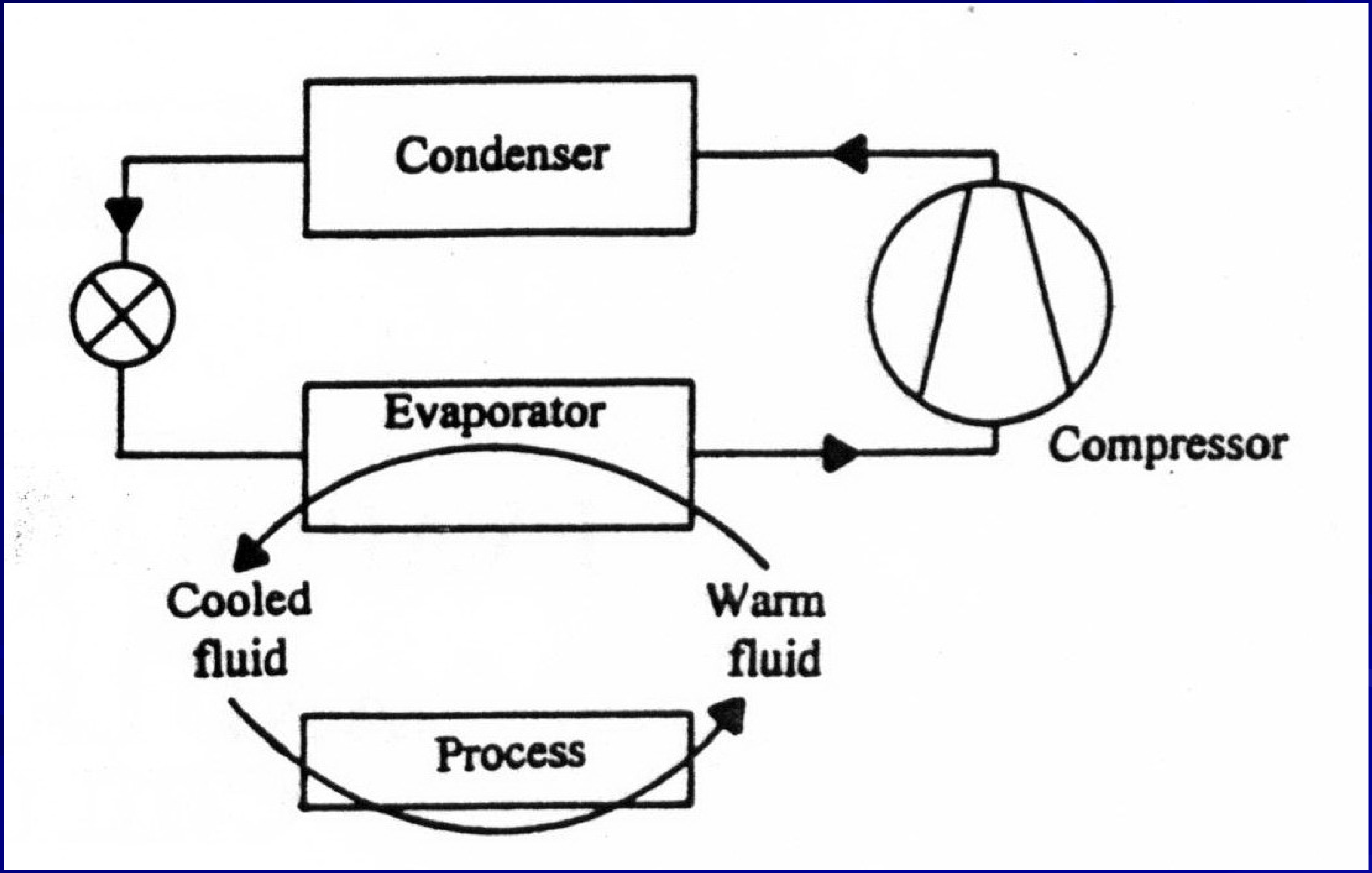



**FIGURE 18.13**
Refrigerated loading dock to reduce refrigeration load in the low-temperature space.

# Chapter 6
# EVAPORATOR




# Cooling at Evaporator



# Flooded Air Coil

**TABLE 6.3**
**Penalties in evaporating temperature due to static head of liquid in the evaporator.**

| Evaporating temperature | Increase in evaporating temperature, °C per m (°F per ft) | |
|---|---|---|
| | Refrigerant-22 | Ammonia |
| 0°C (32°F) | 0.774 (0.425) | 0.392 (0.215) |
| -40°C (-40°F) | 2.81 (1.54) | 1.77 (0.97) |

Several advantages of flooded evaporators in comparison to direct expansion are:

- the evaporator surfaces are used more effectively because they are completely wetted
- problems in distributing refrigerant in parallel-circuit evaporators are less severe
- saturated vapor rather than superheated vapor enters the suction line, so the temperature of suction gas entering the compressor is likely to be lower, which also reduces the discharge temperature from the compressor.

Several disadvantages of the flooded evaporator in comparison to the direct-expansion evaporator are:

- the first cost is higher
- more refrigerant is needed to fill the evaporator and surge drum
- oil is likely to accumulate in the surge drum and evaporator and must be periodically or continuously removed.

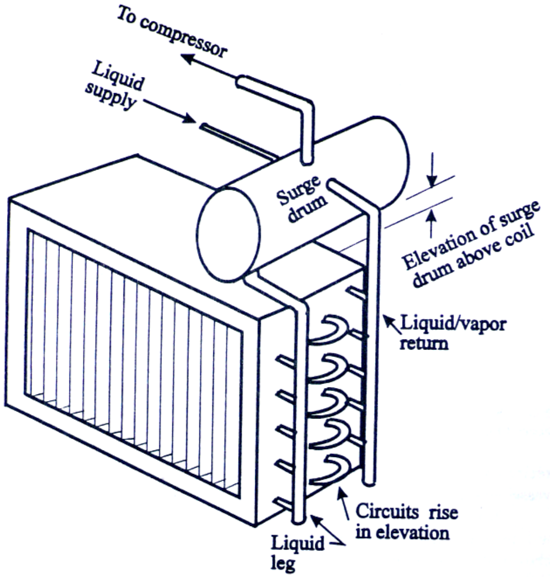



**FIGURE 6.11**
A flooded air-cooling coil for low-temperature application.

# Pump Liquid Recirculation System

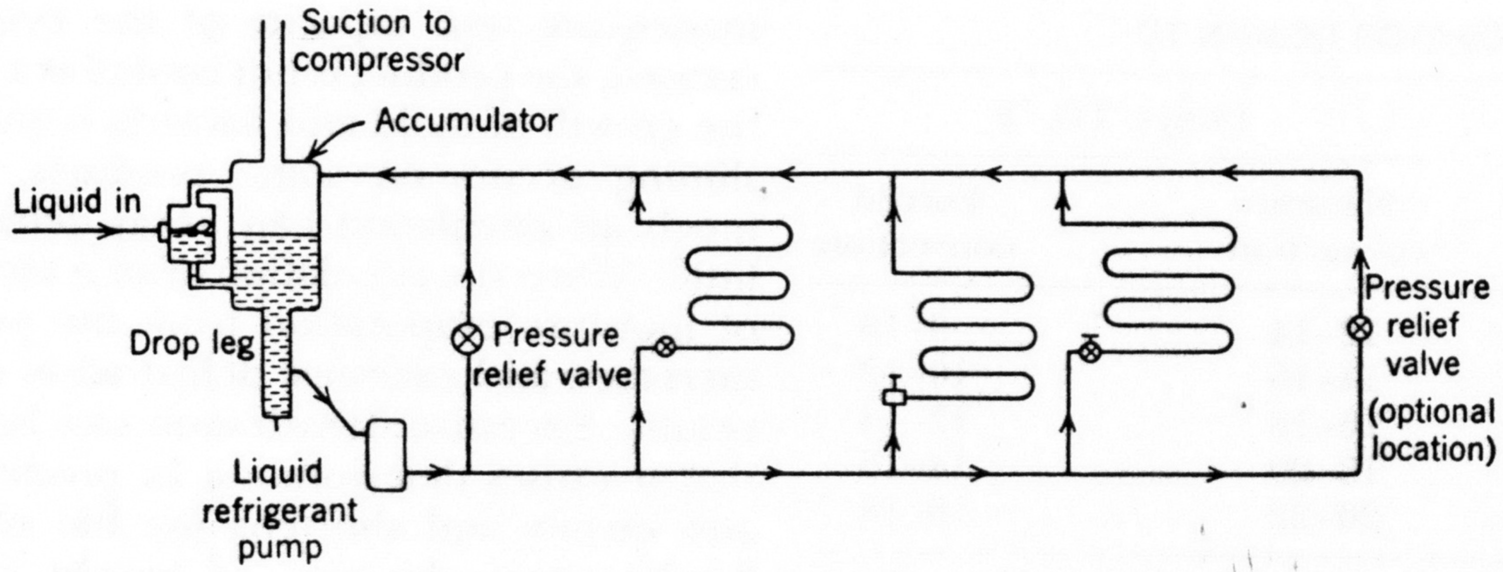
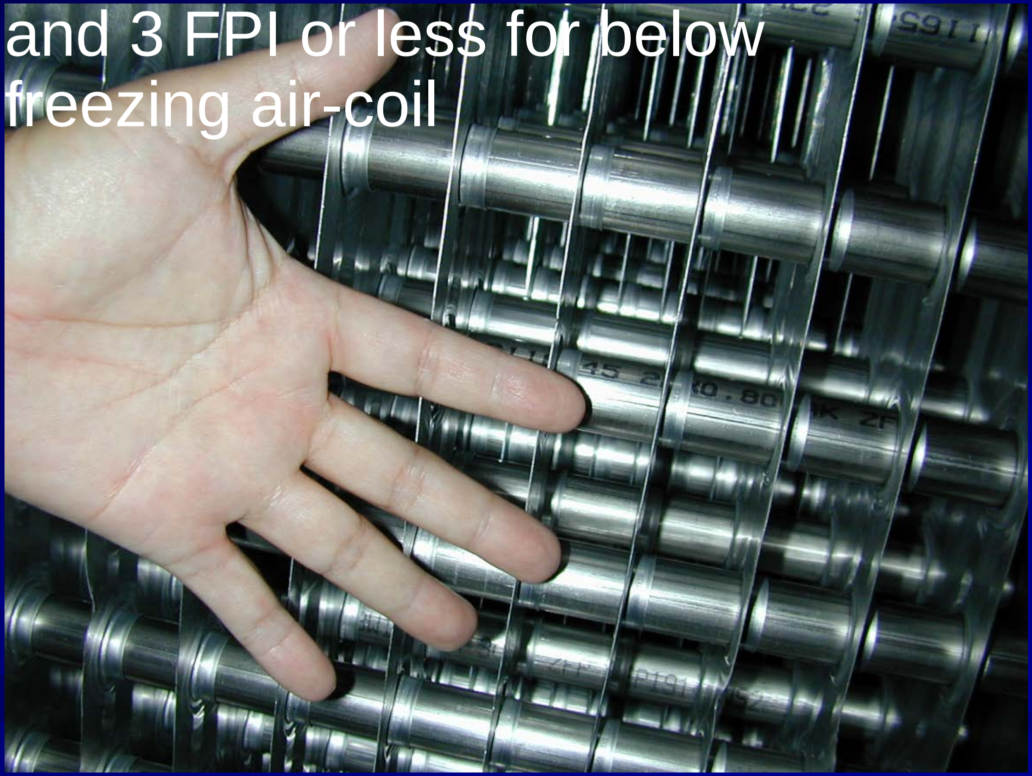



**Fig. 11-19** A multiple evaporator system employing liquid recirculation (overfeed).

# External VS Internal Fin

About 12 FPI for above freezing air-coil

and 3 FPI or less for below freezing air-coil



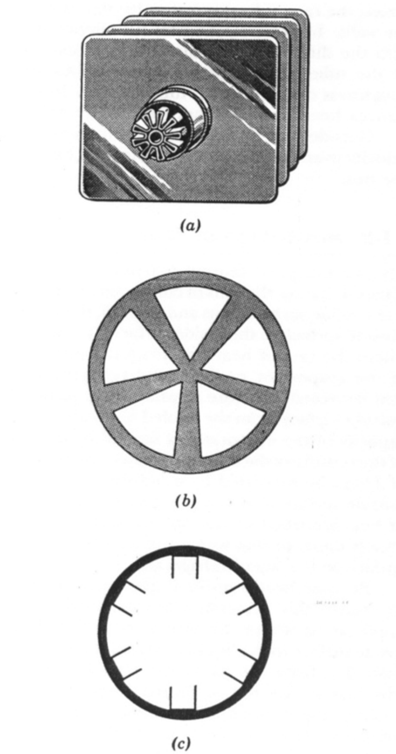
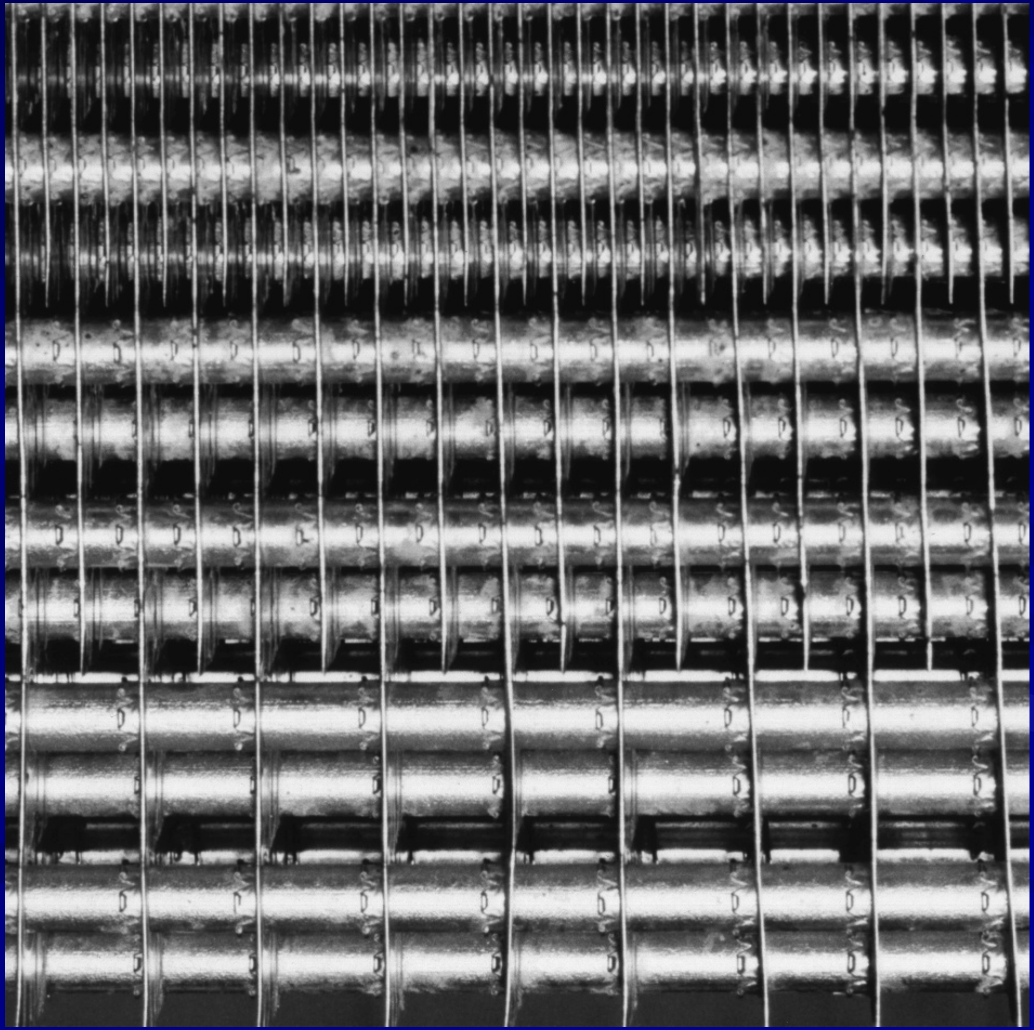



Fig. 11-9 Some methods of inner finning.

# External Variable Pitch Fins



# Evaporator TD

$$TD = T_{AIR,IN} - T_{REFRIG,SAT}$$

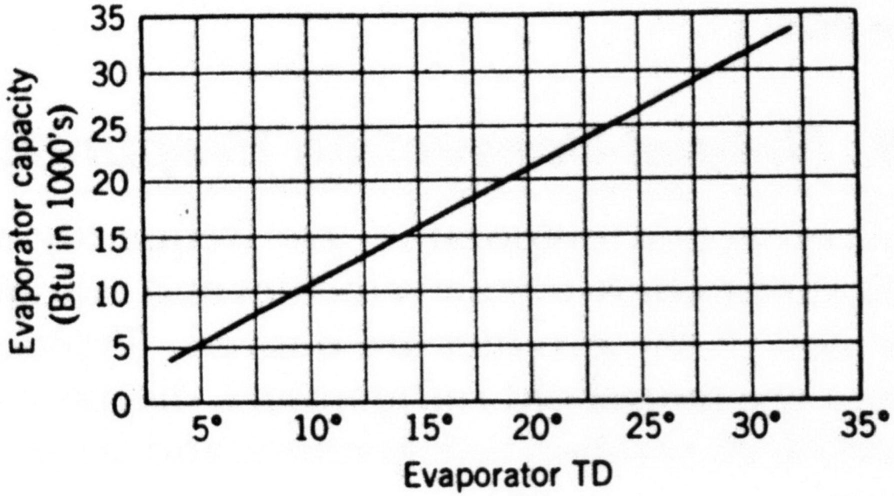



**Fig. 11-20** Variation in evaporator capacity with evaporator TD.

# TD & Humidity

**TABLE 6.7**
**Typical temperature differences—entering air to refrigerant—for several applications.**

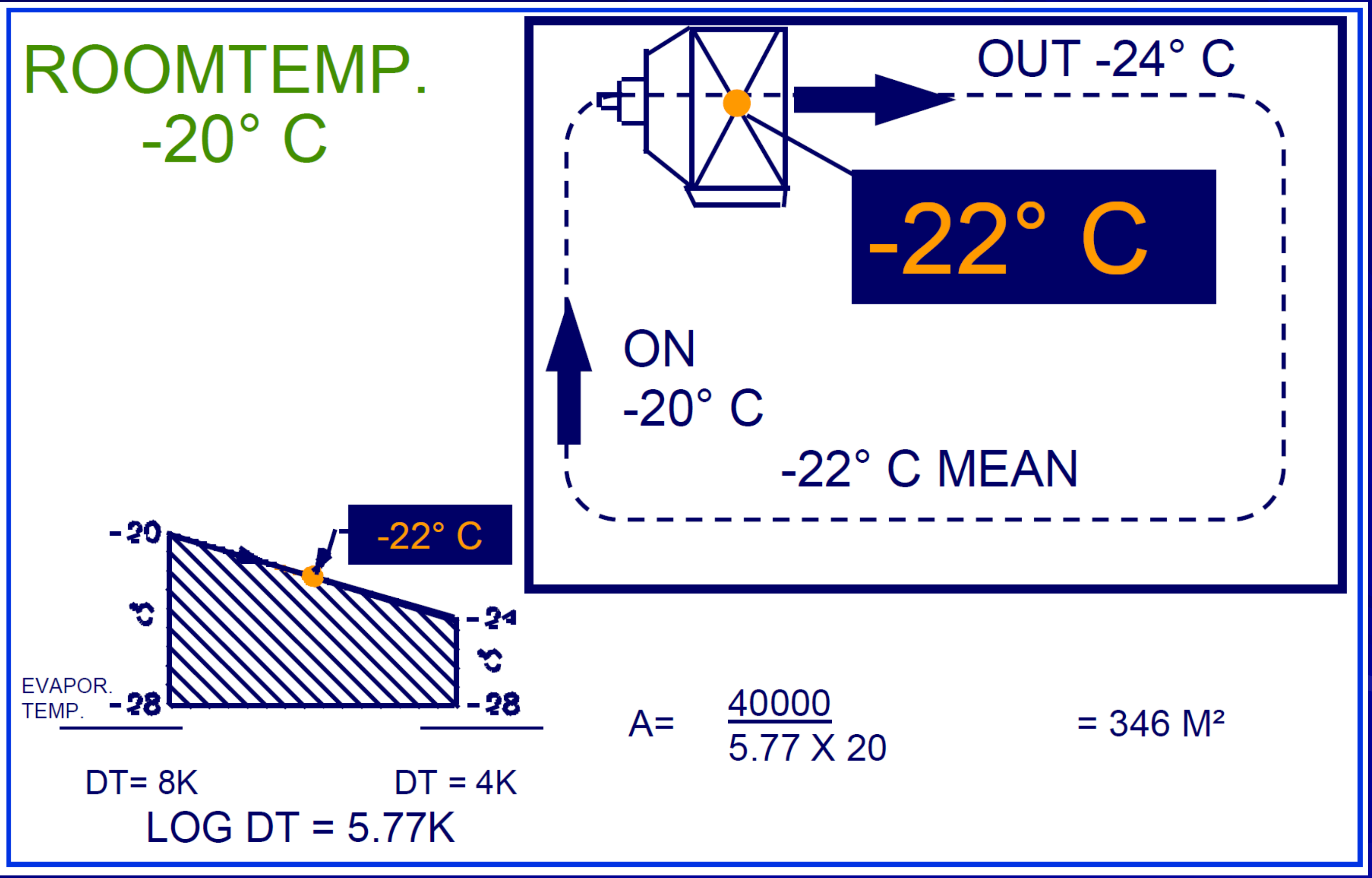
| Application | | $t_{air,in} - t_{refrig}$ |
|---|---|---|
| Below freezing | Storage and blast freezer | 5.5 to 6.5°C (10 to 12°F) |
| Above freezing | Low humidity | 11 to 17°C (20 to 30°F) |
| | High humidity | 2.2 to 4.4°C (4 to 8°F) |

**TABLE 6.8**
**Two strategies for maintaining high humidities in refrigerated spaces.**

| Strategy | Implications |
|---|---|
| Operating with low air-to-refrigerant temperature differences | Large total area of coils, thus, large-size coils and/or a large number of them. Additional coils mean more fans and the sensible loads that their motors impose on the refrigerated space. |
| Higher air-to-refrigerant temperature difference in combination with humidifiers | Moderate total coil area, thus typical size and number of coils. Additional latent load imposed on coils, because the water vapor introduced by the humidifiers must constantly be removed by the coils. |

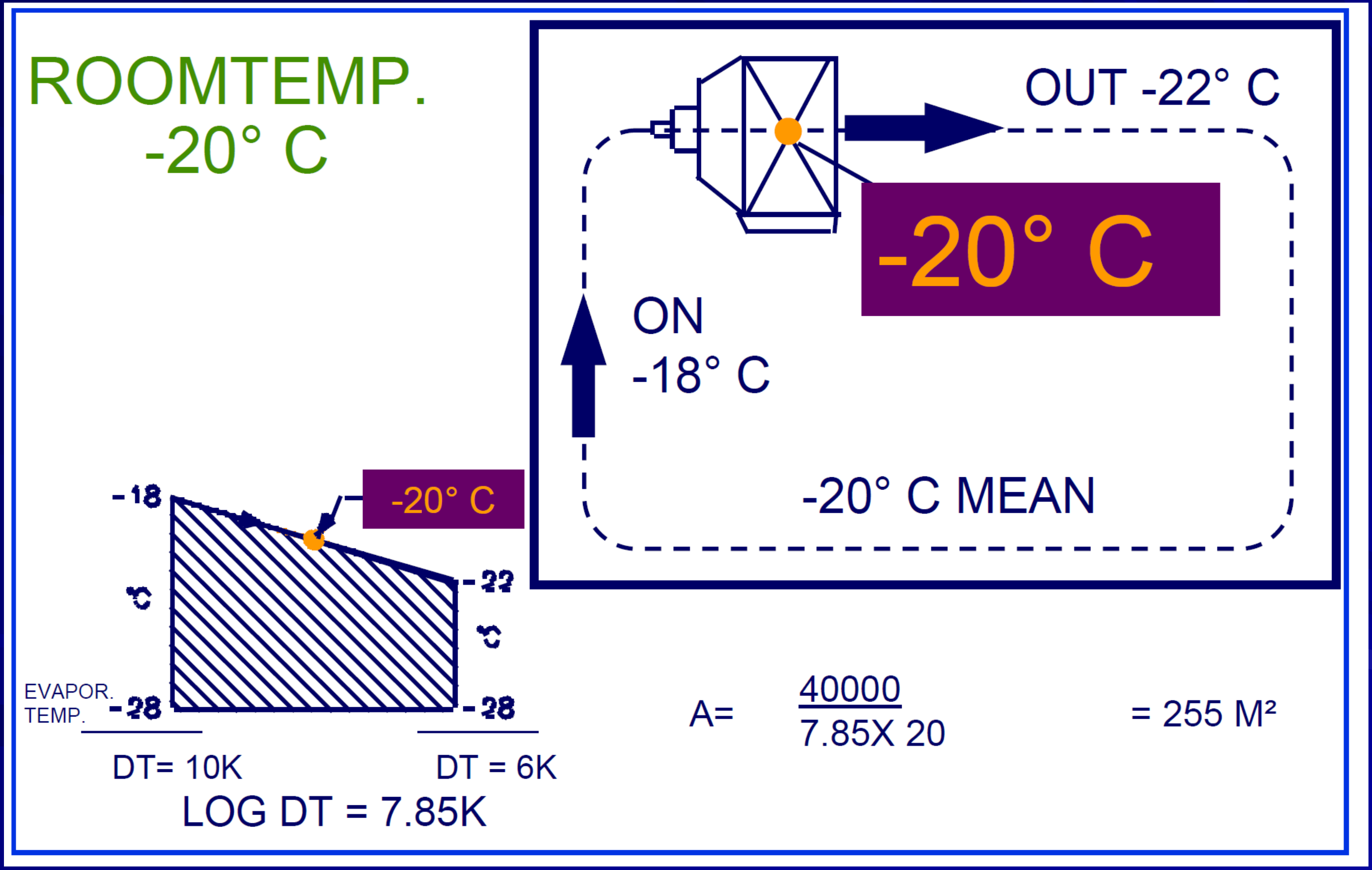
# Effect of TD on Area

Note that TD is sometime called DT



# Effect of TD on Area

Note that TD is sometime called DT



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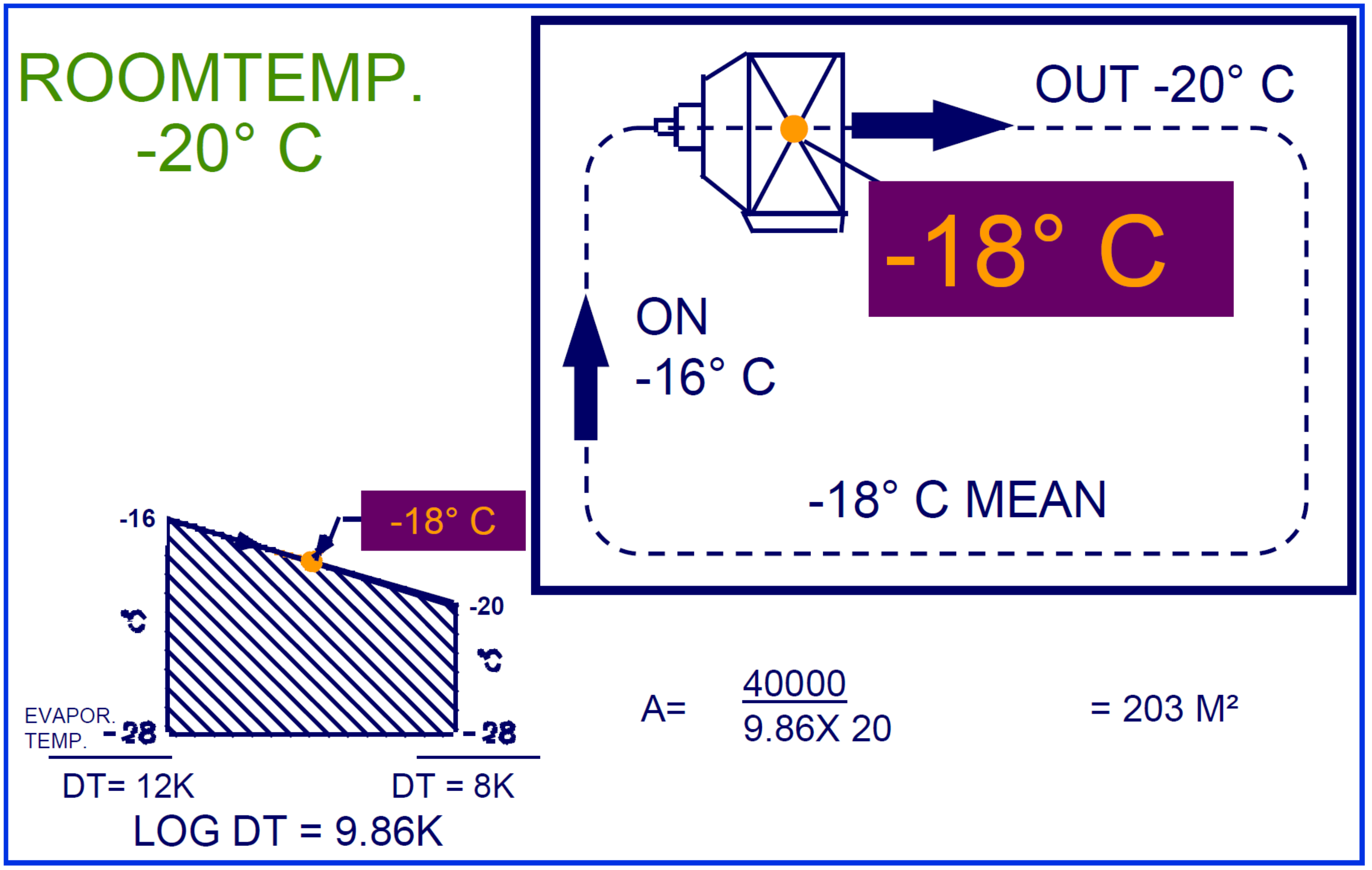


# Optimum Evaporating Temperature

Why the size of the evaporator is dictated by TD?

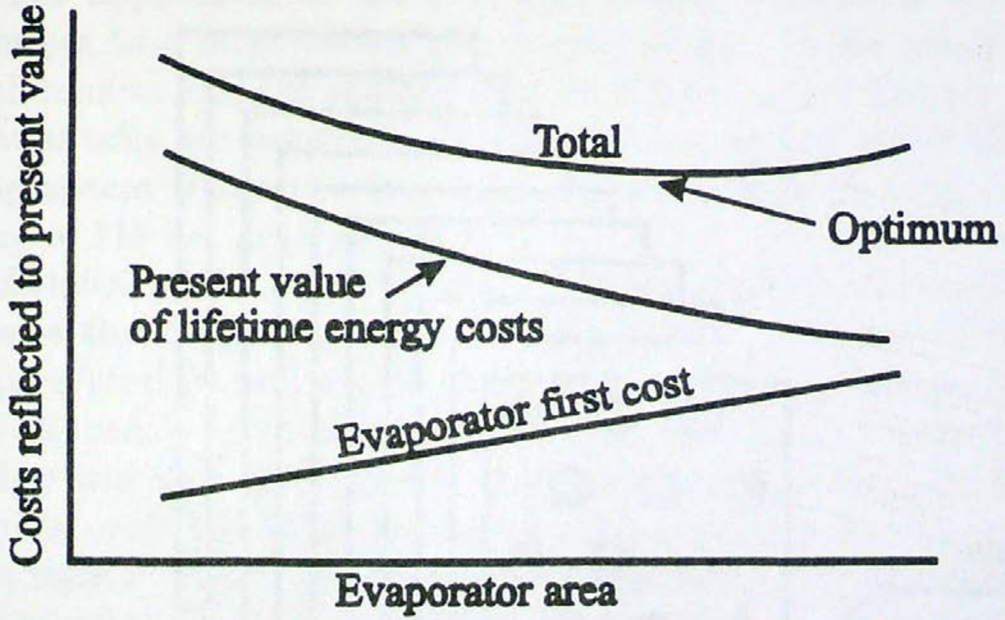



**FIGURE 6.13**
Optimum evaporator area for minimum total of the first cost of the evaporator and the present worth of the lifetime compressor energy cost.

# Draw-Through/Blow-Through

Draw-Through has greater throw/Blow-Through has the thermal advantage

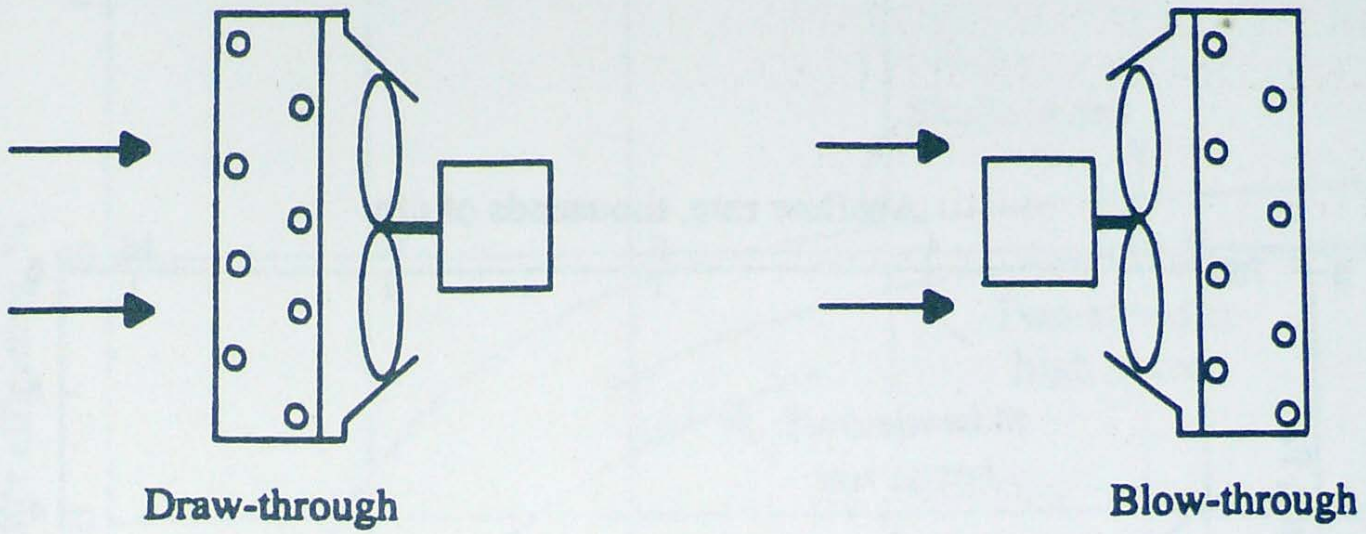



**FIGURE 6.36**
Draw-through and blow-through arrangements of the fan and coil.

# Suggested Location

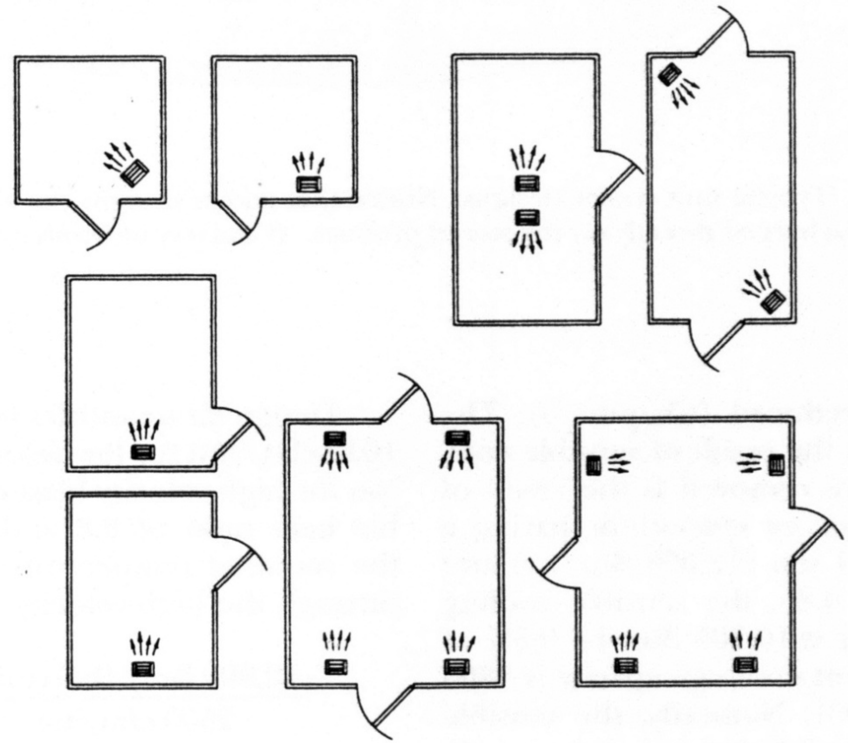


**Fig. 11-25** Suggestions for location of unit coolers in walk-in refrigerators. (From the *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.)

# Coil & Doorway Location

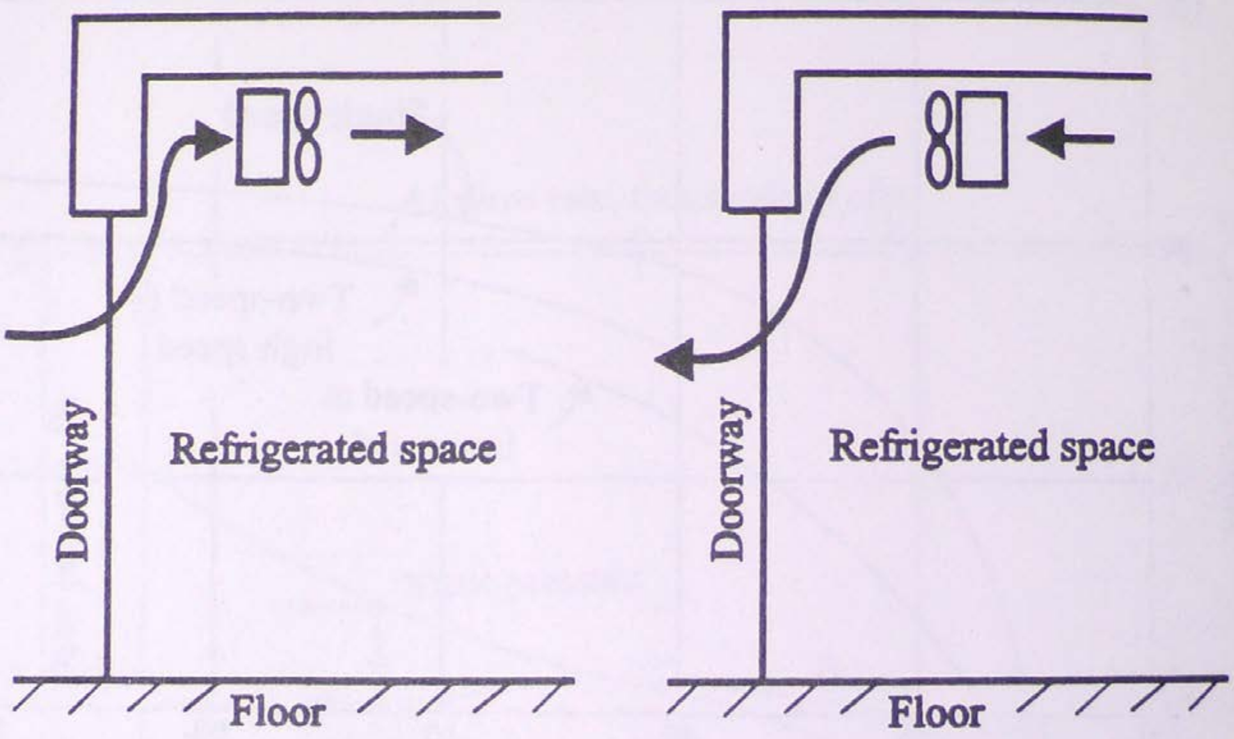
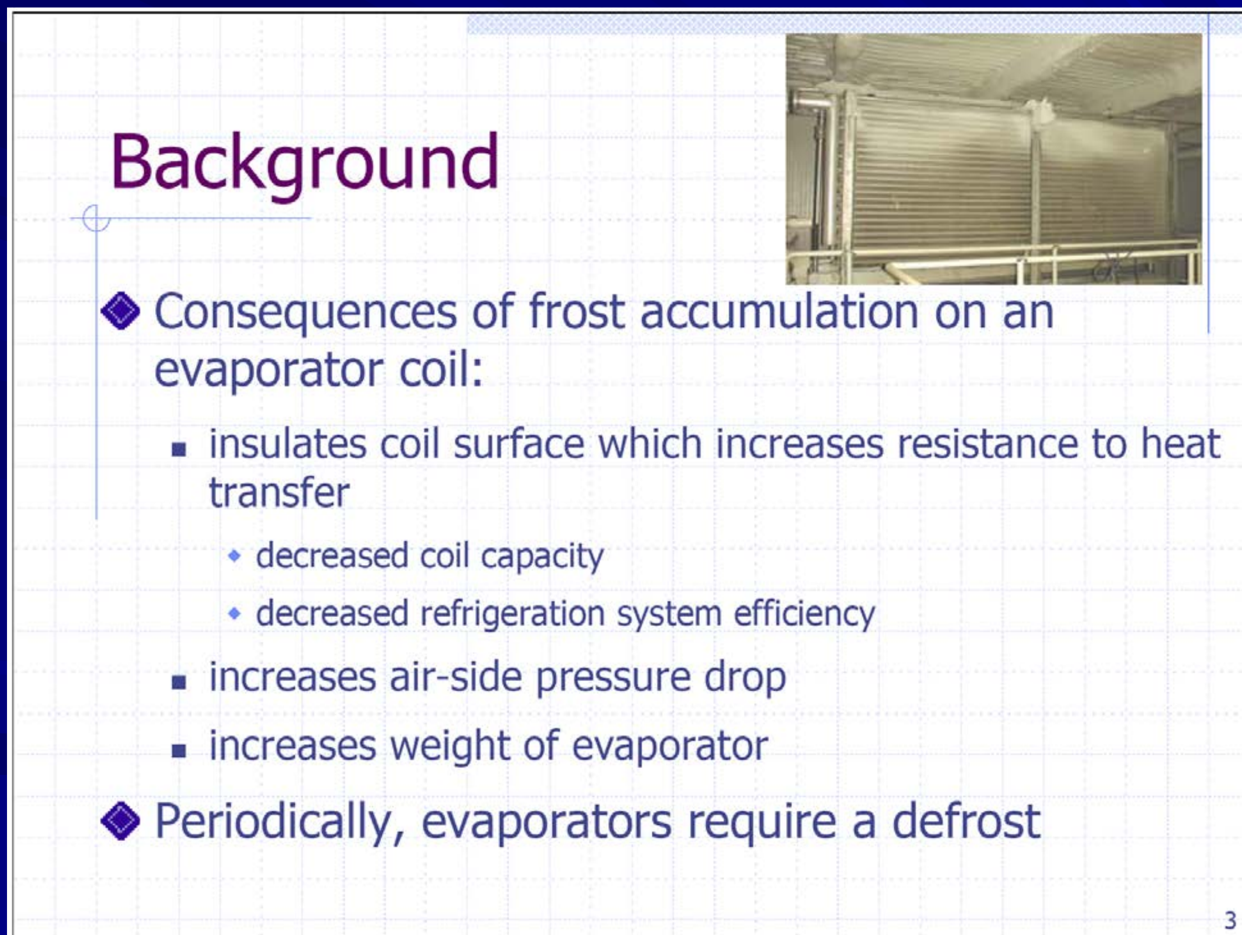



**FIGURE 6.39**
Do not mount coils above a doorway such that they draw warm air in through or discharge cold air out through the doorway. Instead, arrange for the air to flow past the doorway.

# Frost on Coil

*** Undesirable Dehumidification ***



# Effect of Frost on Air Pressure Drop

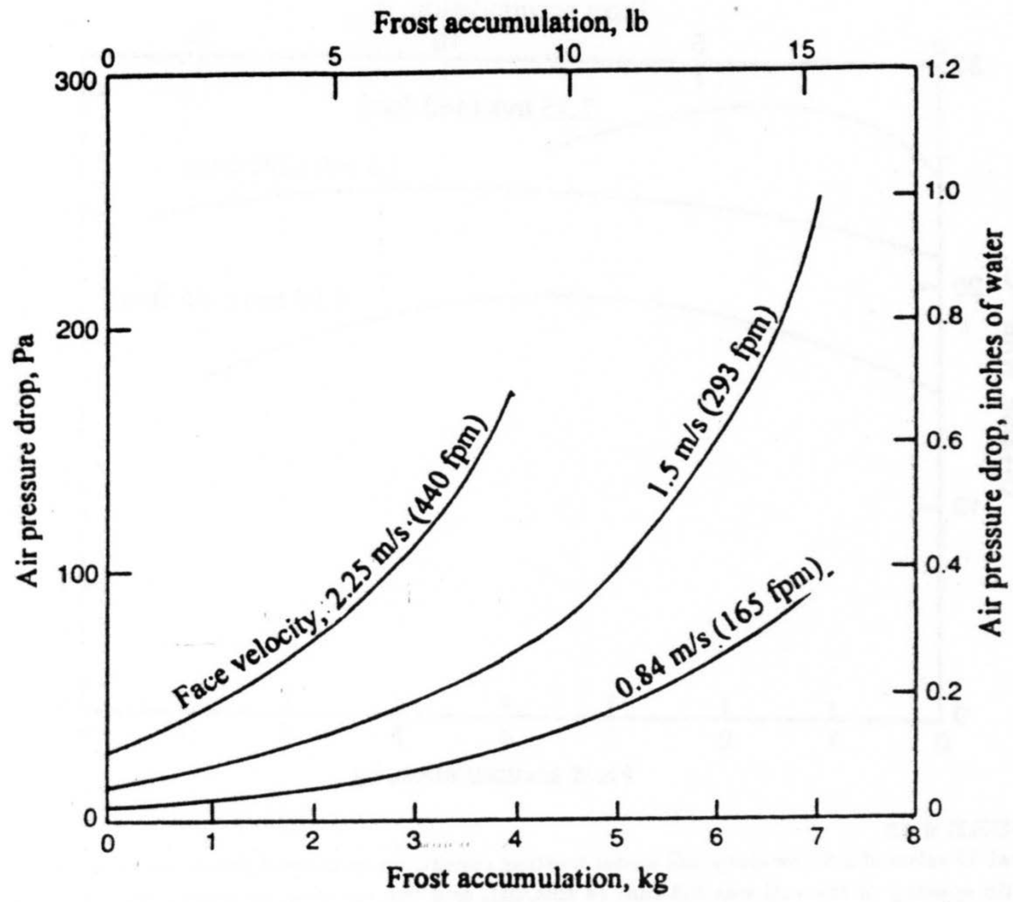



**FIGURE 6.43**
Air-pressure drop as influenced by frost accumulation for the coil described in Fig. 6.42.

# Air Pressure Drop with Time

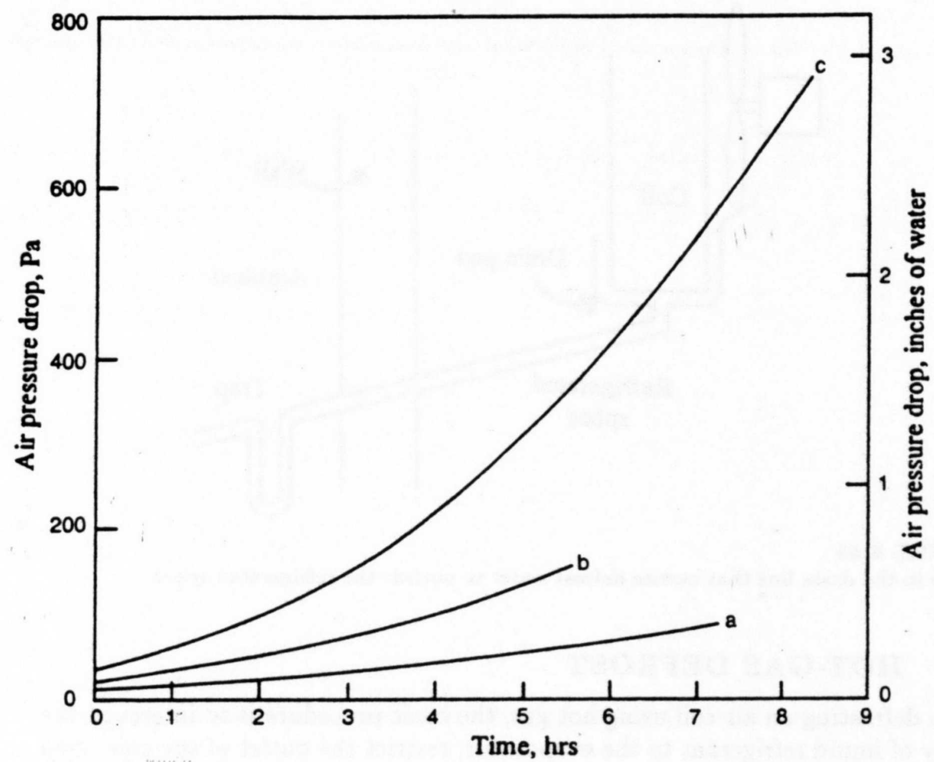
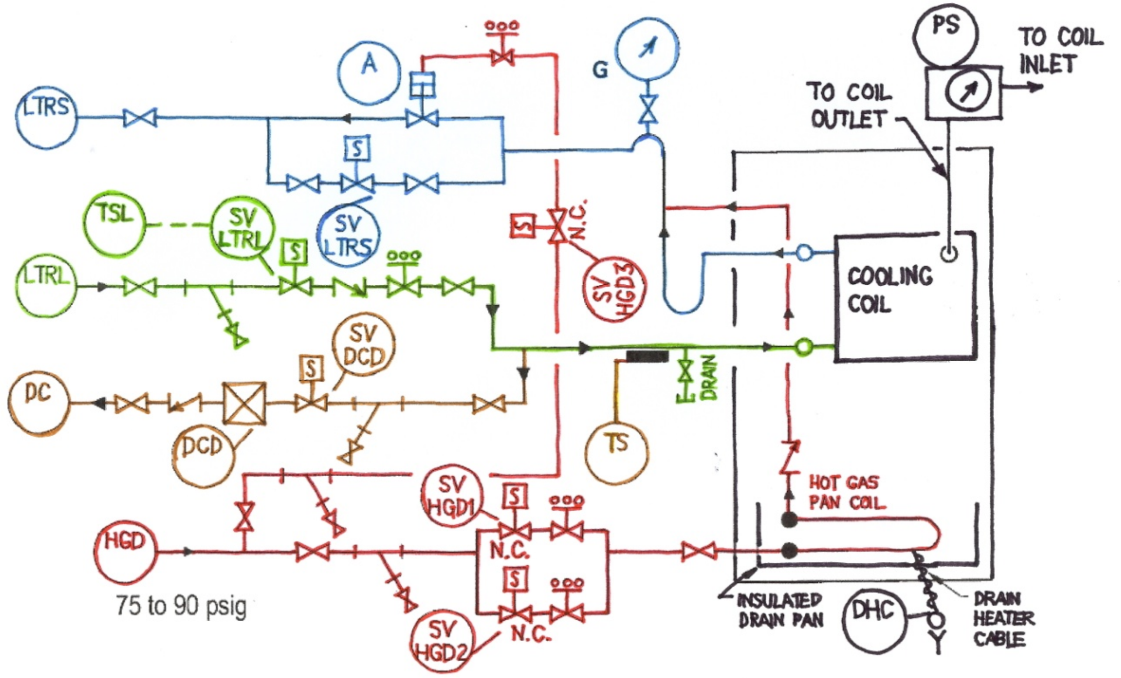



**FIGURE 6.44**
Increase in air-pressure drop with time for coils of different fin spacing. The face velocity was in the range of 3.2 to 3.4 m/s (625 to 675 fpm), and the entering relative humidity was 82%. The fin spacings are: Curve *a*, 15 mm (1.7 fins/in); Curve *b*, 10 mm (2.5 fins/in); and Curve *c*, 7.5 mm (3.4 fins/in)[13].

# Soft Hot-Gas Defrost Cycle




**UPFEED COILS (SINGLE RISER)**

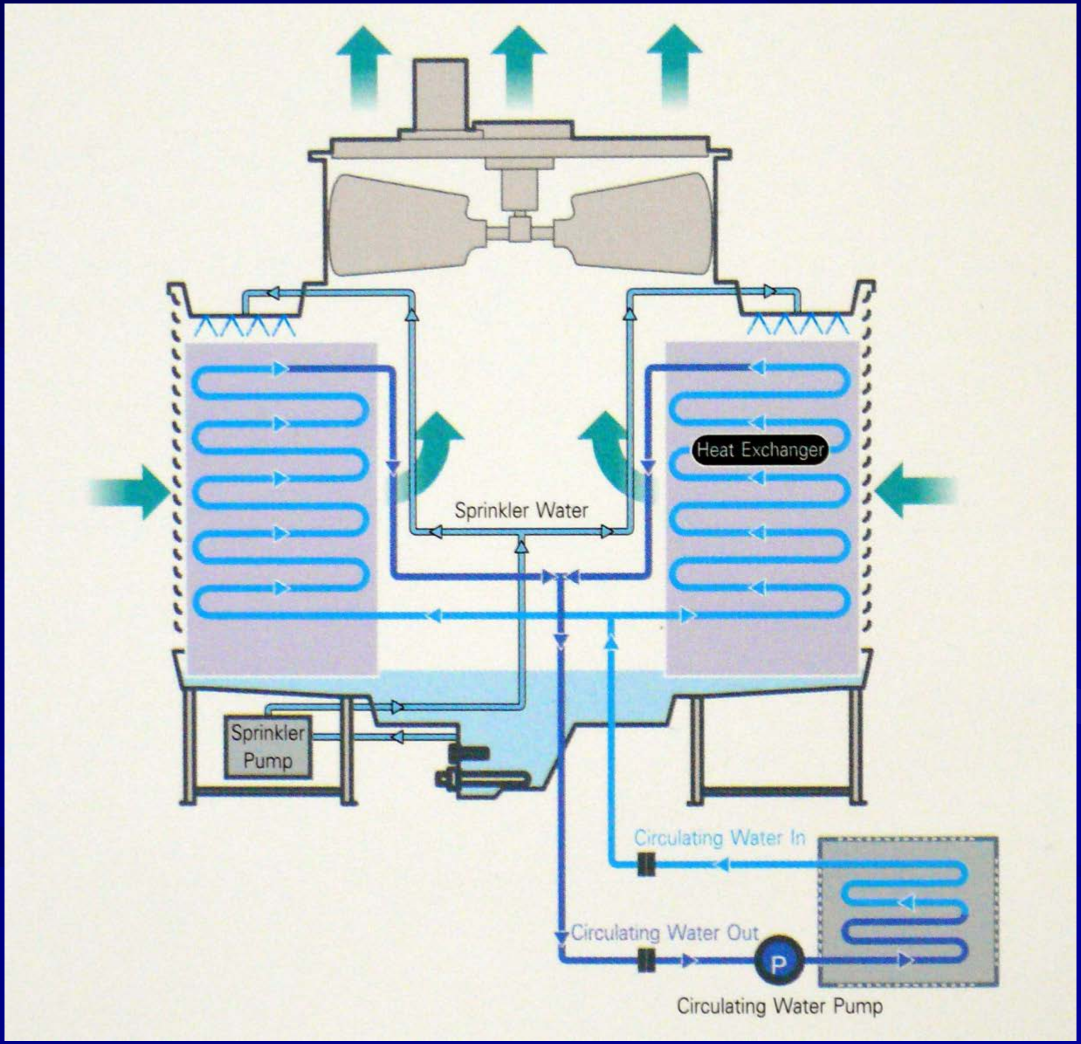
HOT-GAS DEFROST CYCLE ACTUATED BY AIRSIDE PRESSURE SWITCH (PS) AT PREDETERMINED SETTING (approximate 1 in. water gage)

# Hot-Gas Valve Set

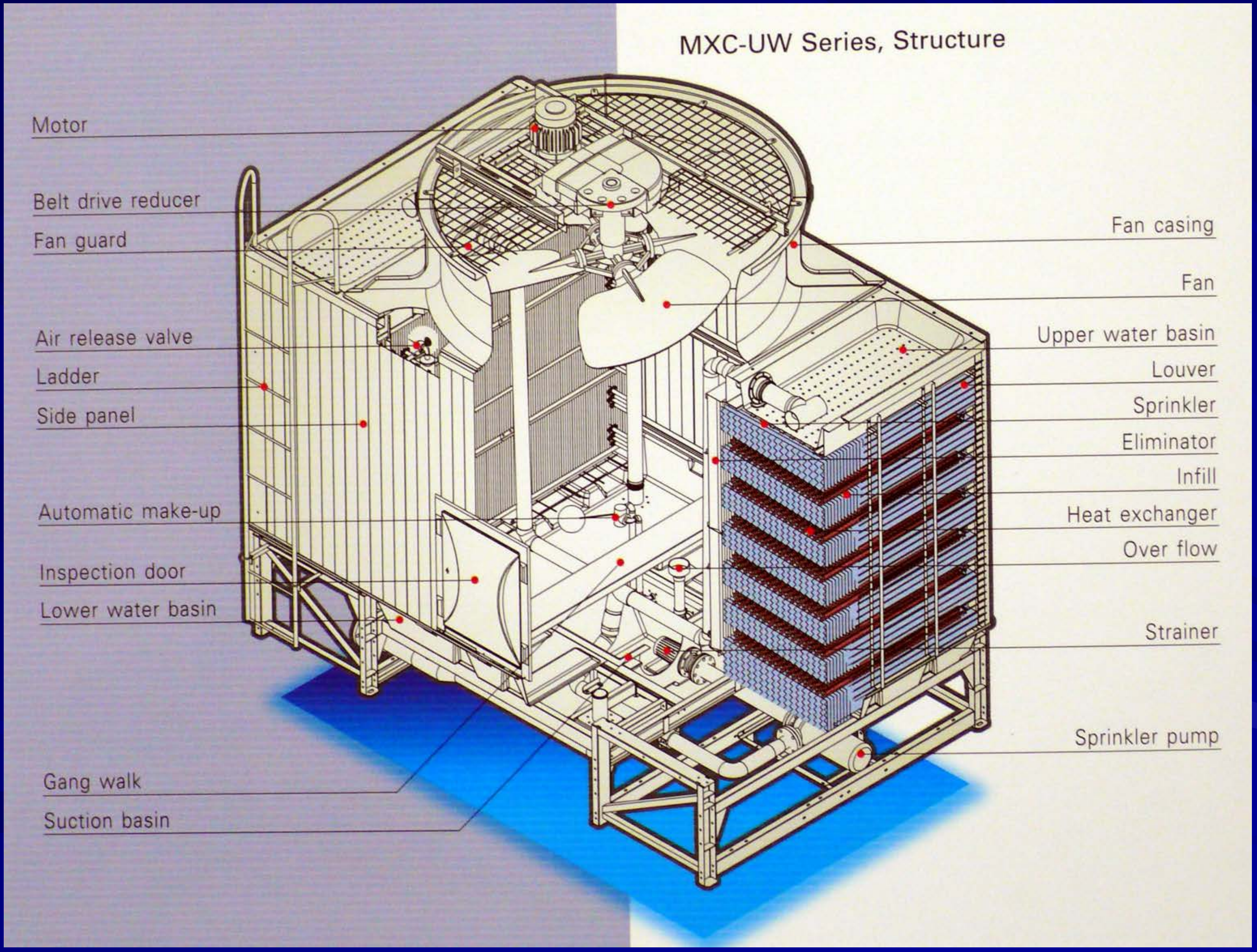
# CONDENSERS & COOLING TOWERS



# Closed Circuit Cooling Tower



# Closed Circuit Cooling Tower



# Evaporative Condenser

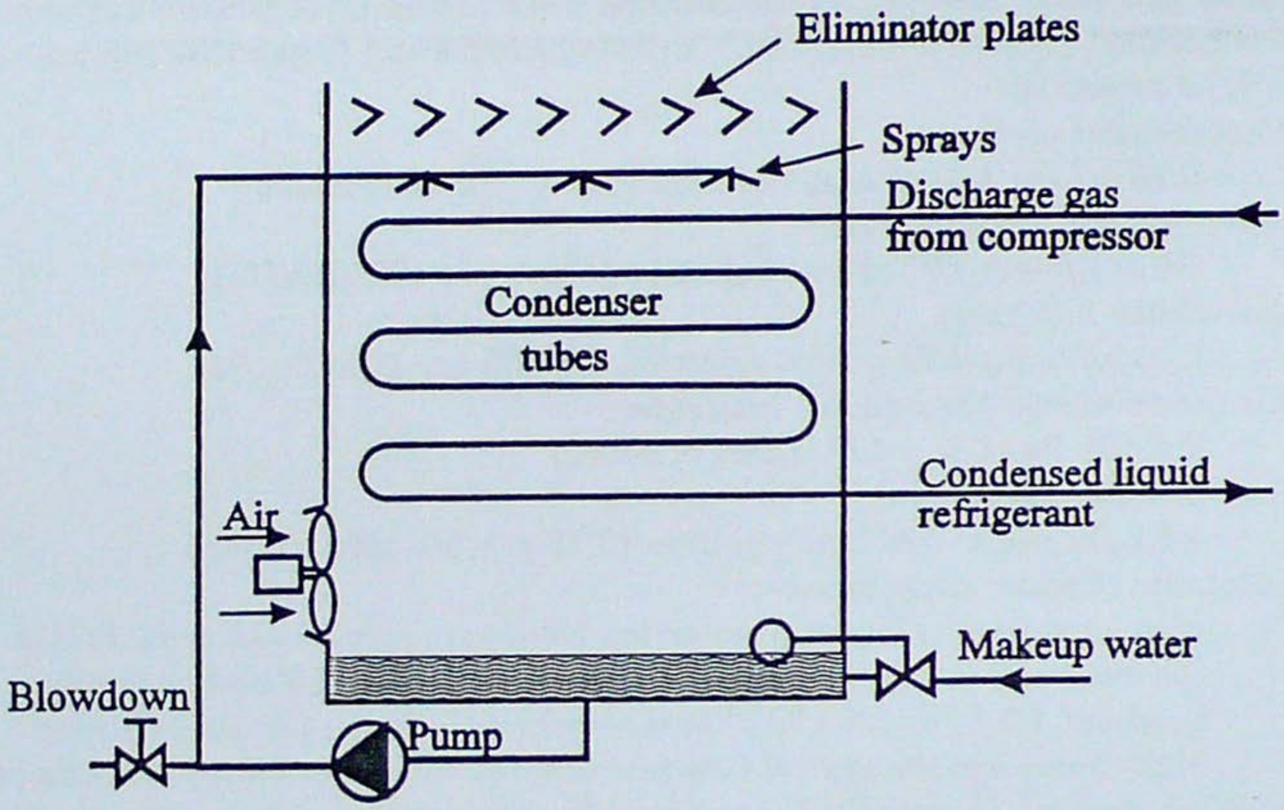



FIGURE 7.11
An evaporative condenser.

# Positioning the Condenser

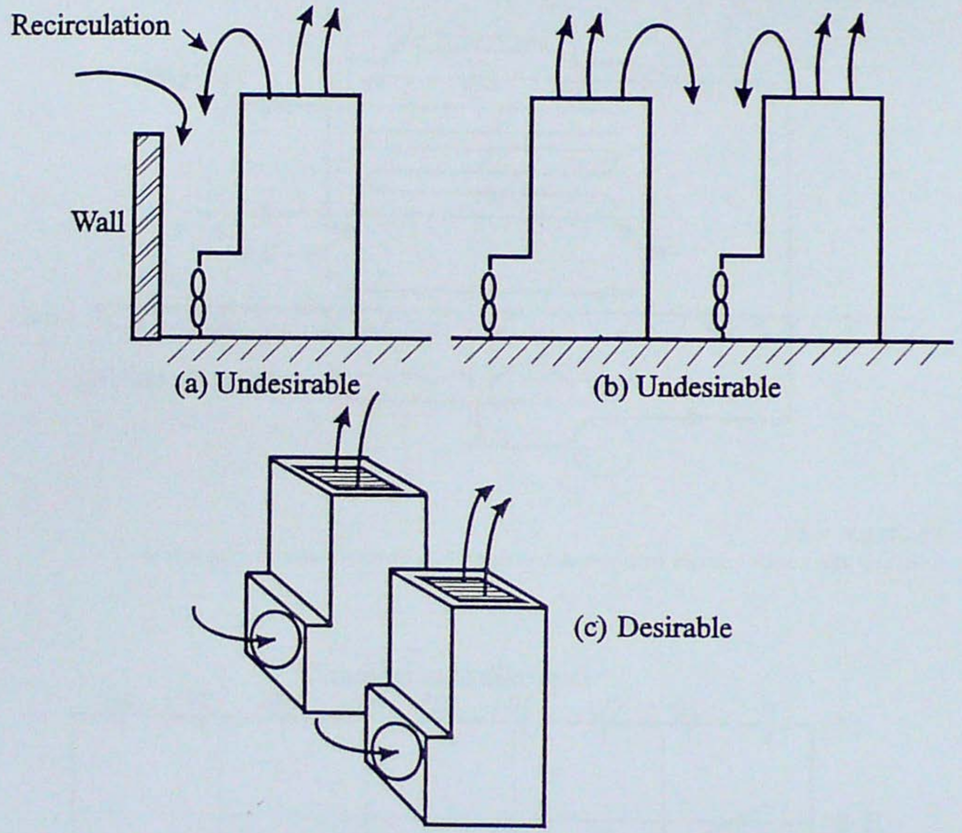



**FIGURE 7.19**
The condensers should be positioned so that the flow of entering air is not obstructed and as little as possible discharge air recirculates to the entrance.

# Evaporative Condenser Piping

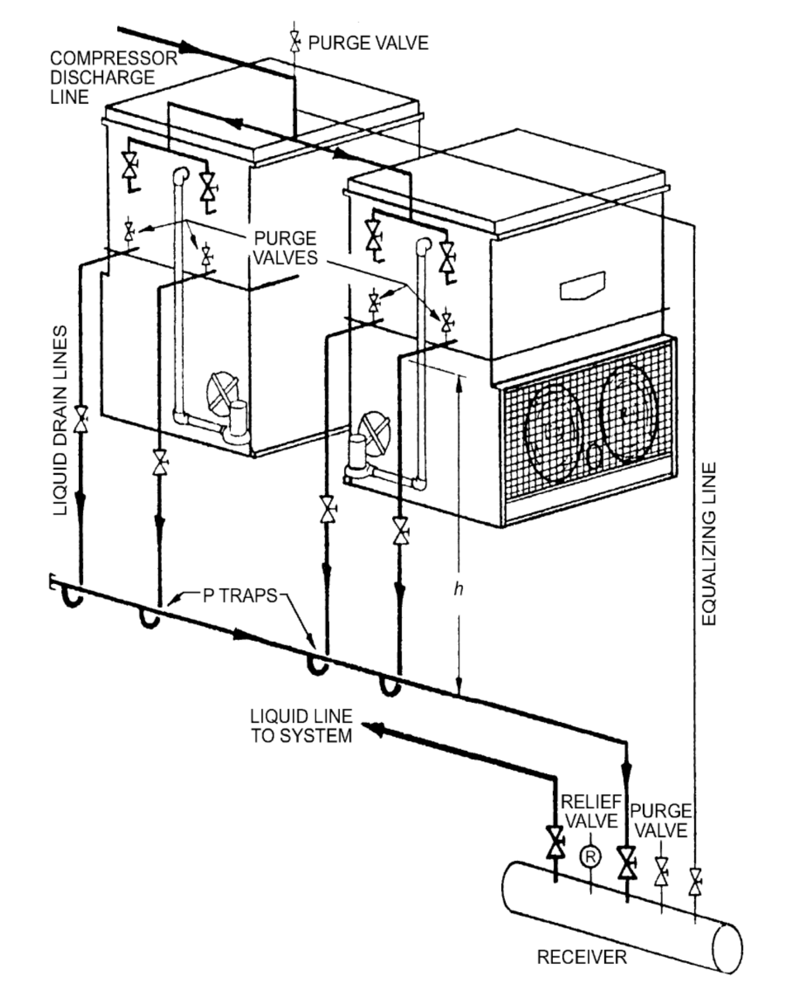
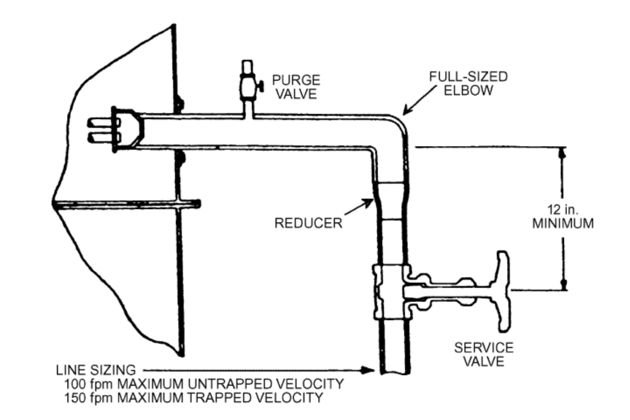



**Fig. 35 Piping for Parallel Condensers with Top Inlet Receiver**



# Installation on Roof Deck

# Evaporative Condenser vs Water Cooled Condenser + CT

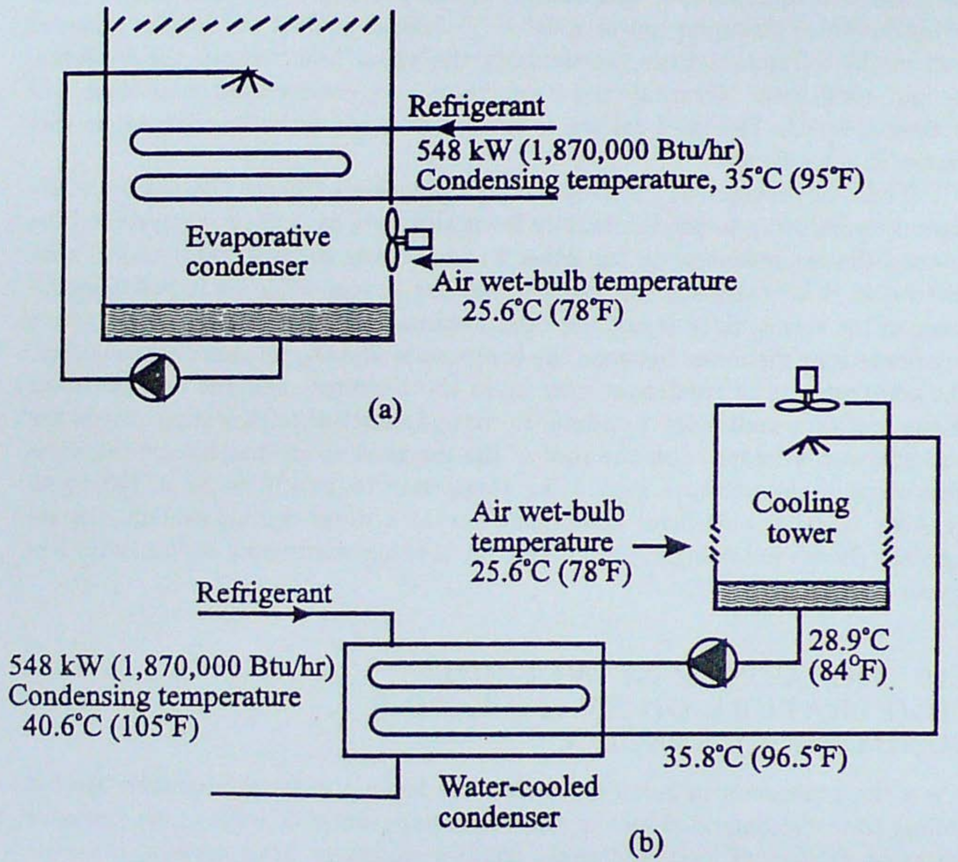



FIGURE 7.12
Achieving a lower condensing temperature with an evaporative condenser in comparison to the combination of a water-cooled condenser and cooling tower.

# Effect of WB on Performance of EC

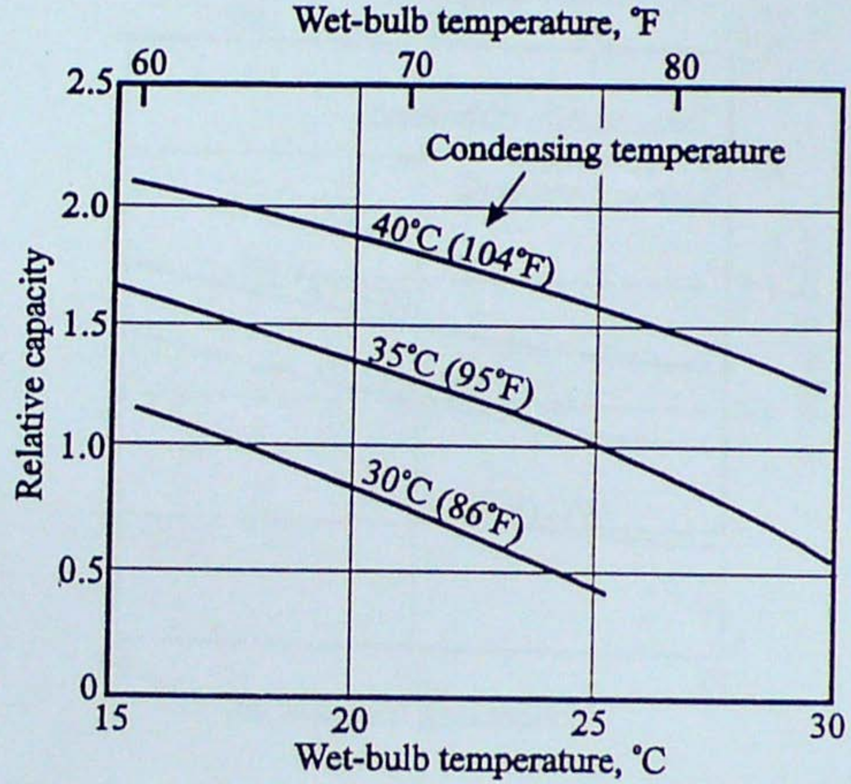



**FIGURE 7.13**

Relative heat-rejection capacity of an ammonia evaporative condenser as a function of the condensing and wet-bulb temperatures. The reference point is a 35°C (95°F) condensing temperature and a wet-bulb temperature of 25°C (77°F).

# Air in Condenser

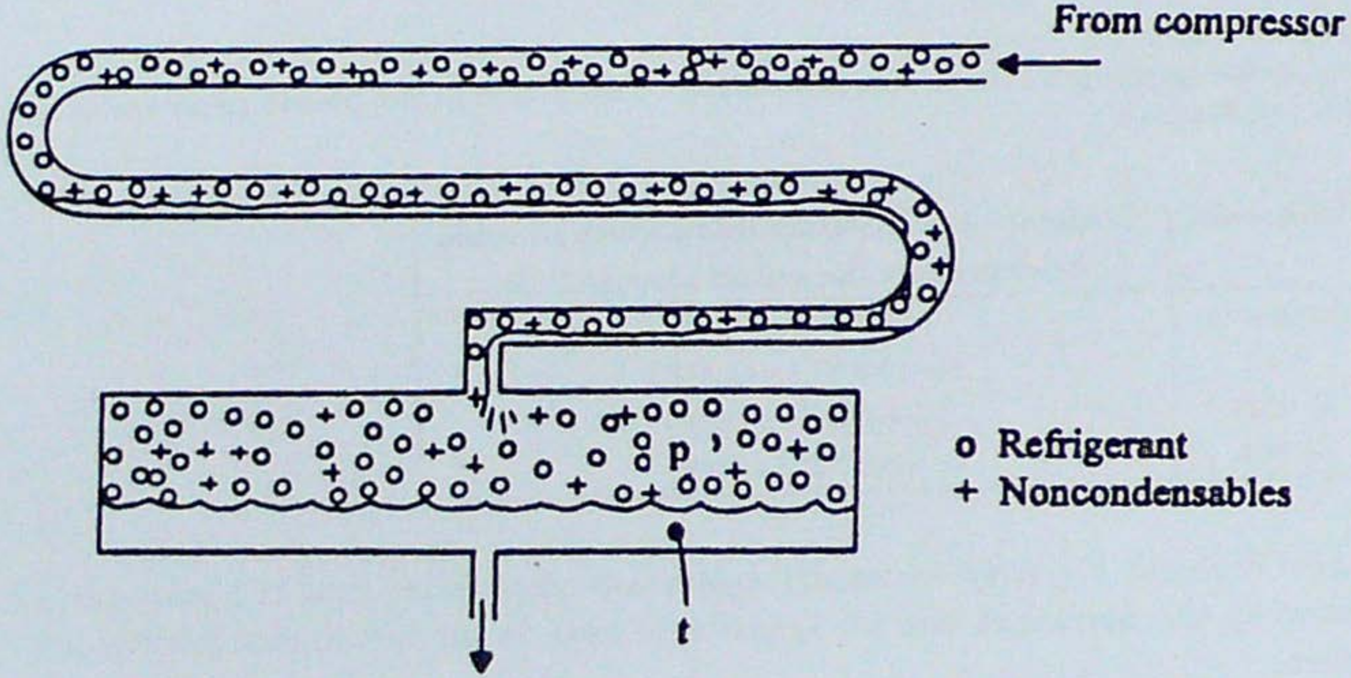



FIGURE 7.22
Noncondensables in a condenser.

# Condenser Sizing

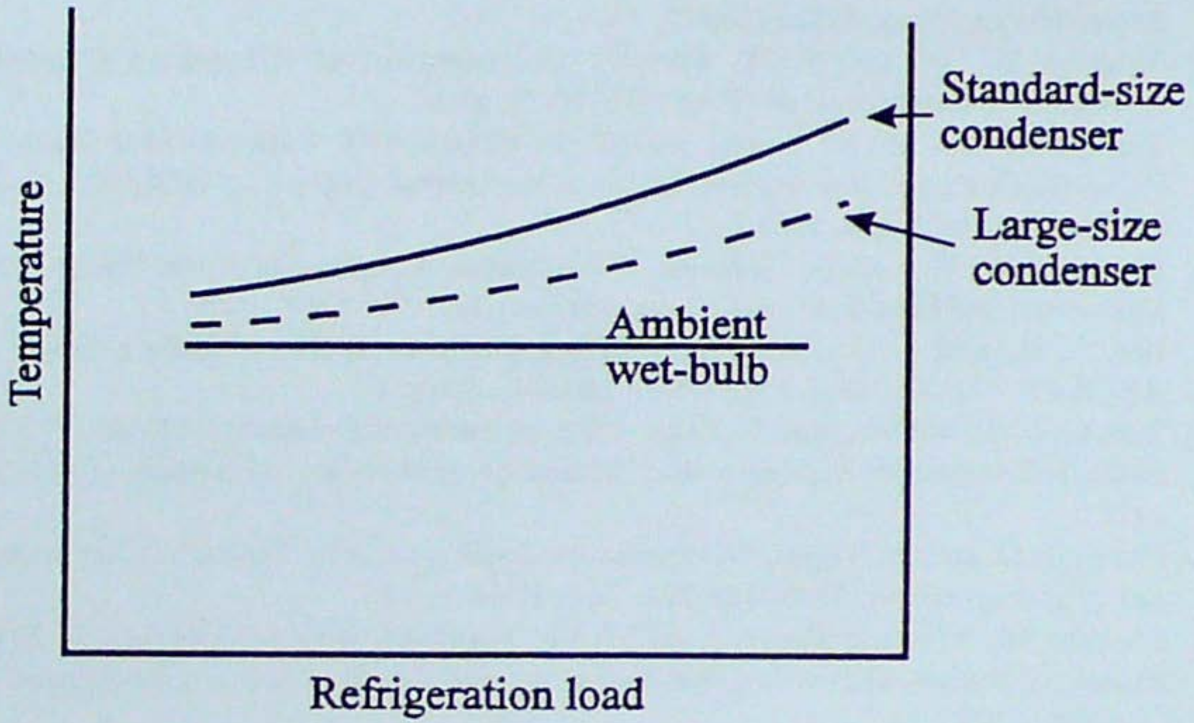



**FIGURE 7.35**
Condensing temperatures as affected by the refrigeration load and the condenser size.

# Condenser Heat Recovery

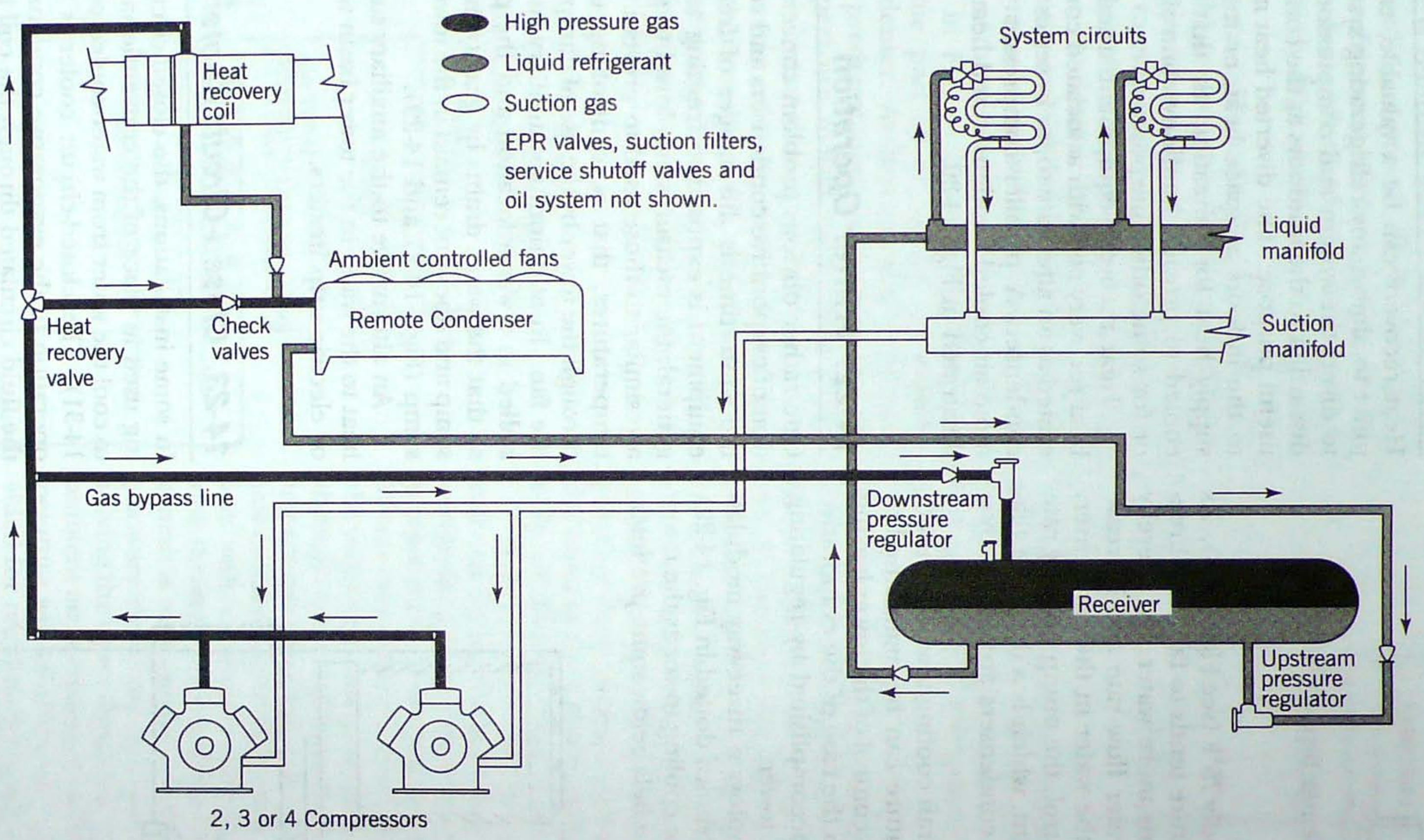
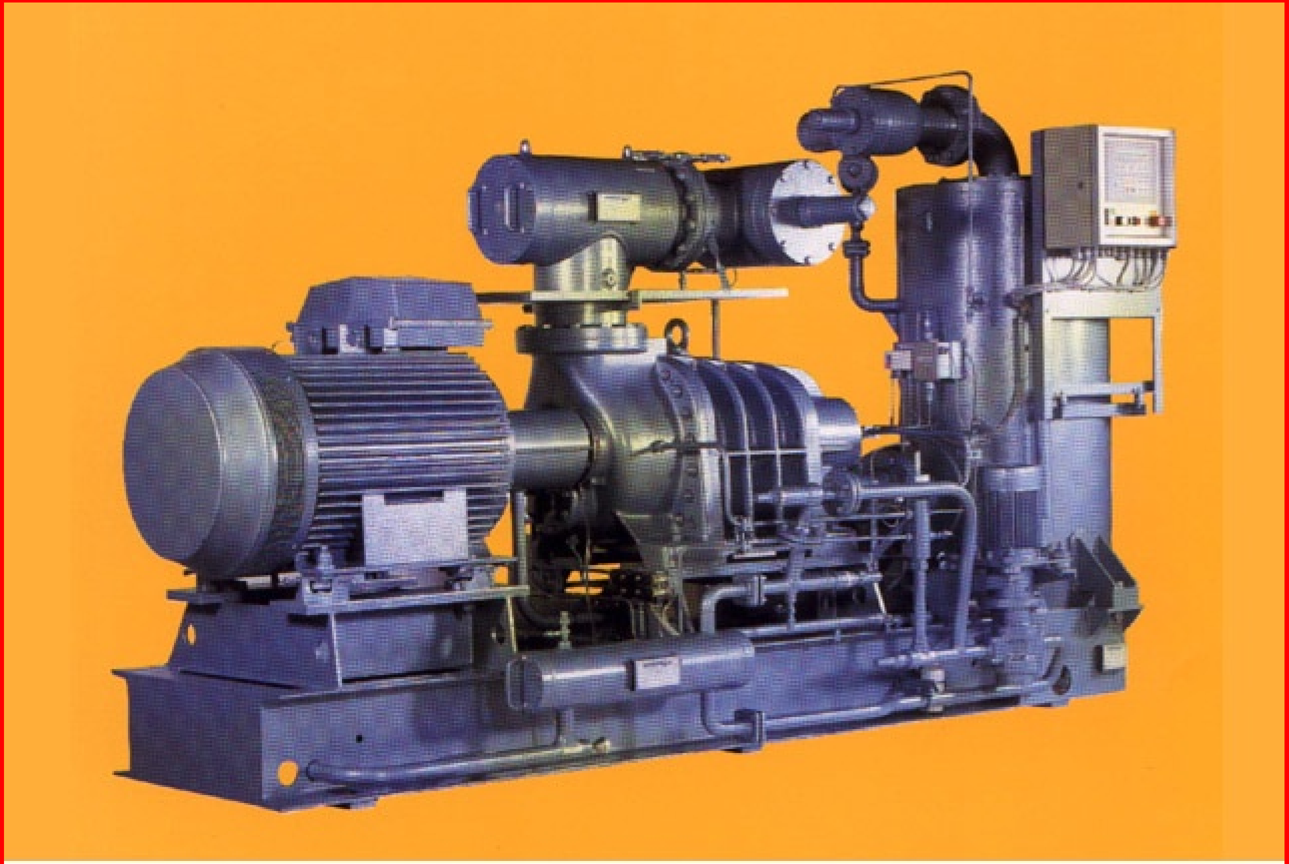



**Fig. 14-30** Basic parallel system with remote air-cooled condenser and heat recovery. (Reprinted by permission of Tyler Refrigeration.)

# COMPRESSOR



Screw Compressor

PATKOL

# COP

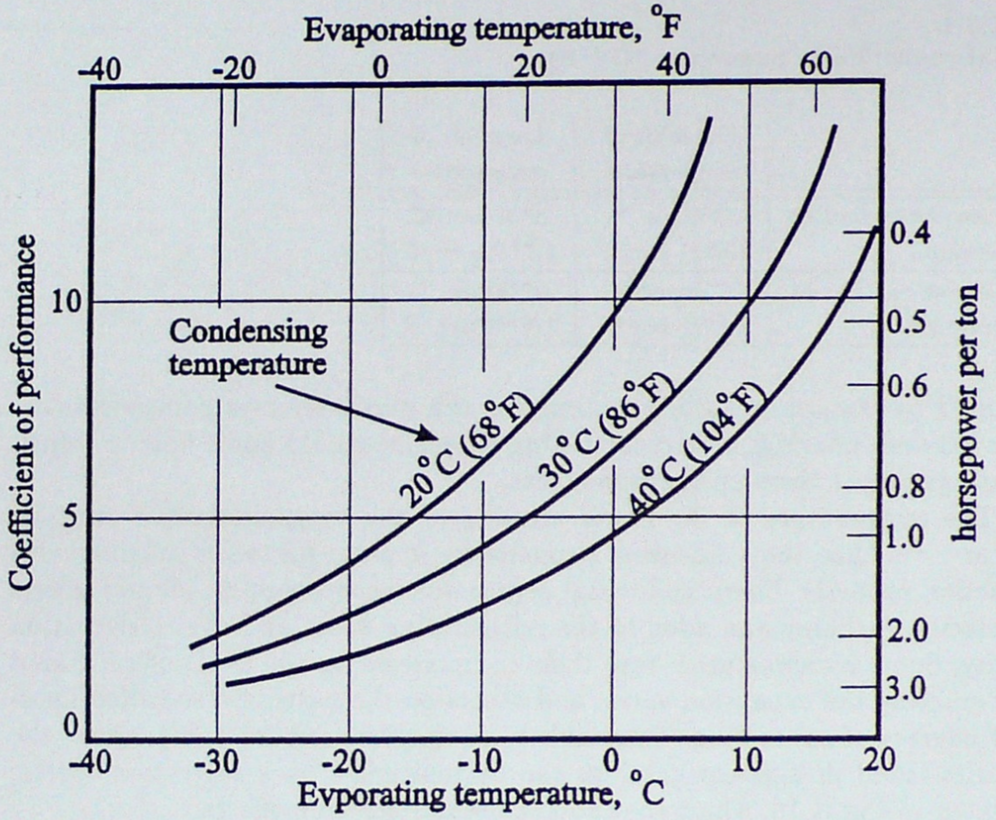



FIGURE 4.15
Coefficient of performance and horsepower per ton as a function of the evaporating and condensing temperatures.

# Variable-volume Ratio

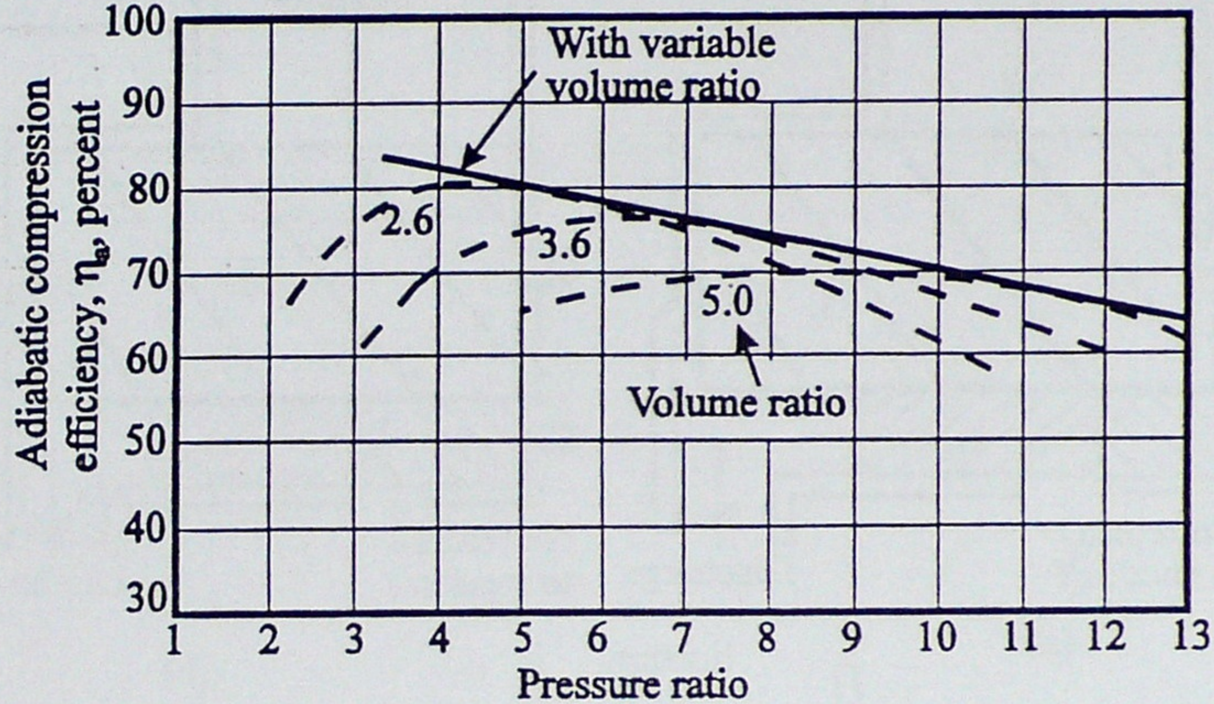



**FIGURE 5.16**
Maintaining peak compression efficiency with a variable-volume ratio device during changes in the pressure ratio.

# Variable-volume Ratio

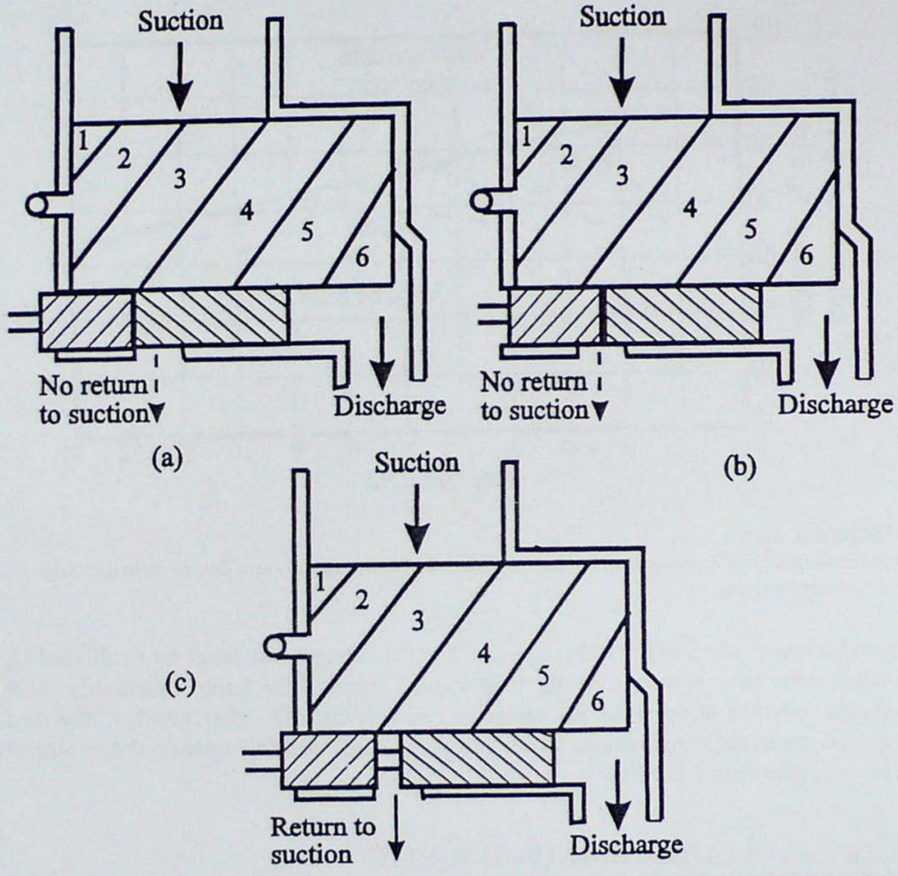



**FIGURE 5.17**
A variable $v_i$ device at the following operating conditions: (a) full load and low $v_i$, (b) full load and high $v_i$, and (c) part load and high $v_i$.

# Oil Cooling

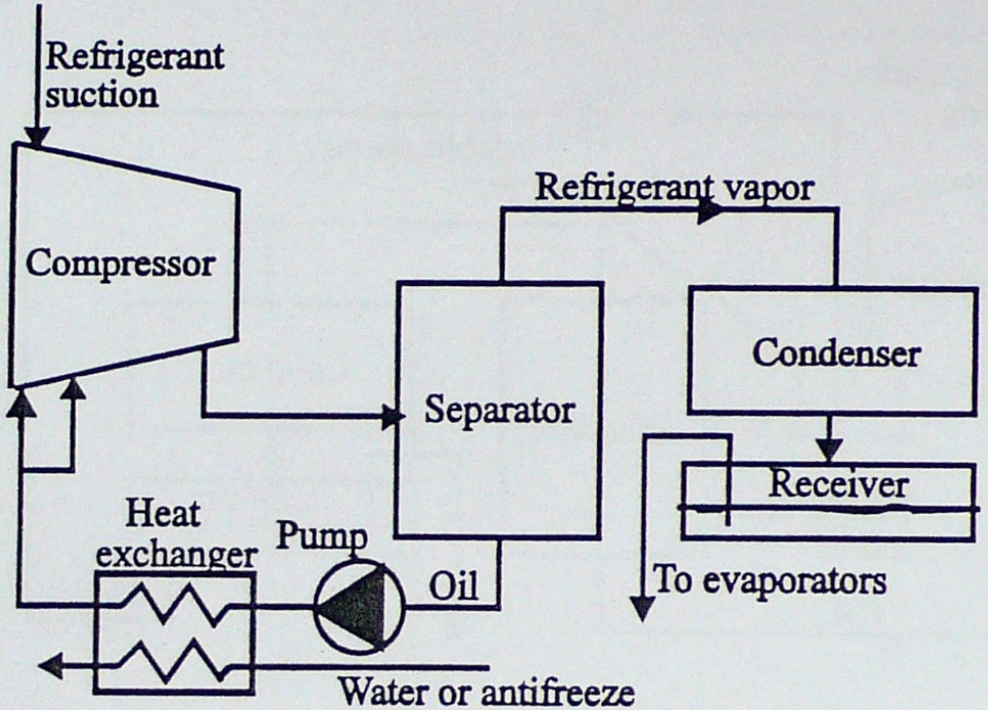



FIGURE 5.19
Oil cooling using an external heat exchanger rejecting heat to water or antifreeze.

# Oil Cooling

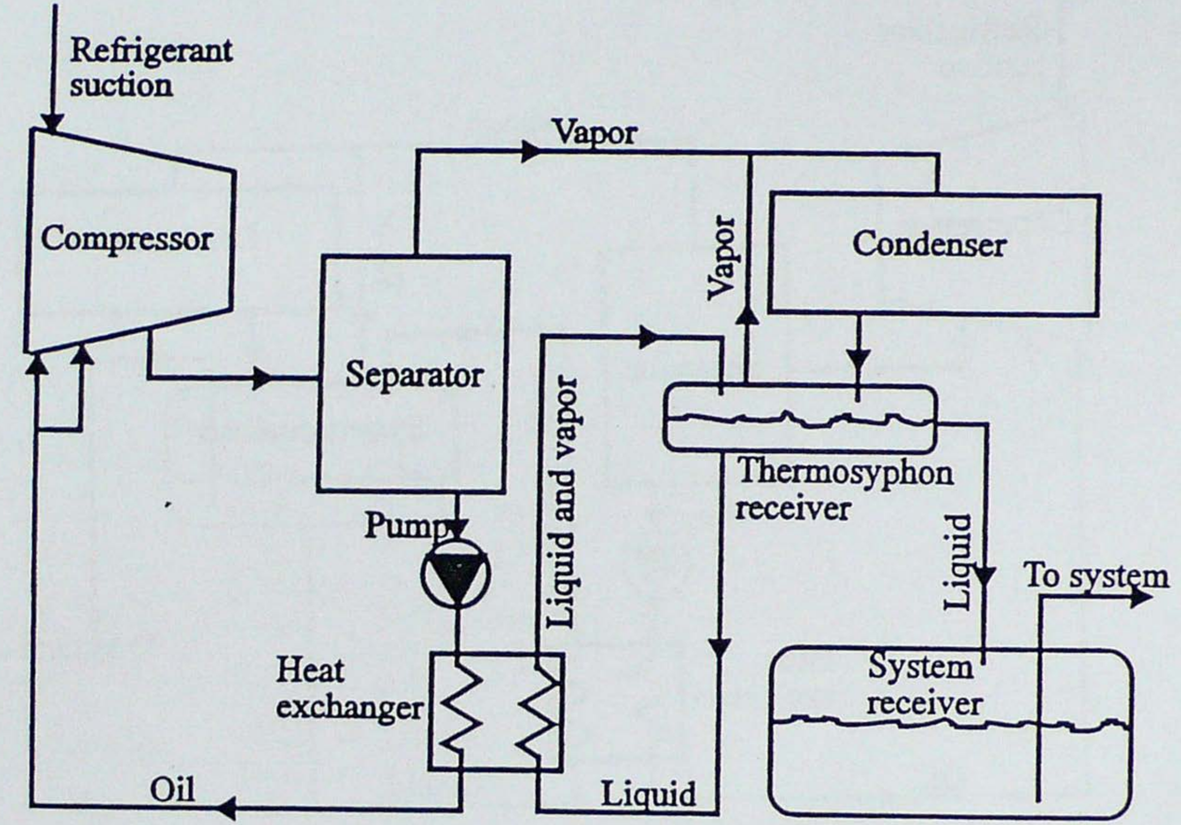



**FIGURE 5.24**
A thermosiphon oil cooling installation where the level of the system receiver is at or below the level of the heat exchanger, requiring an additional receiver.

# Thermosiphon Oil Cooler

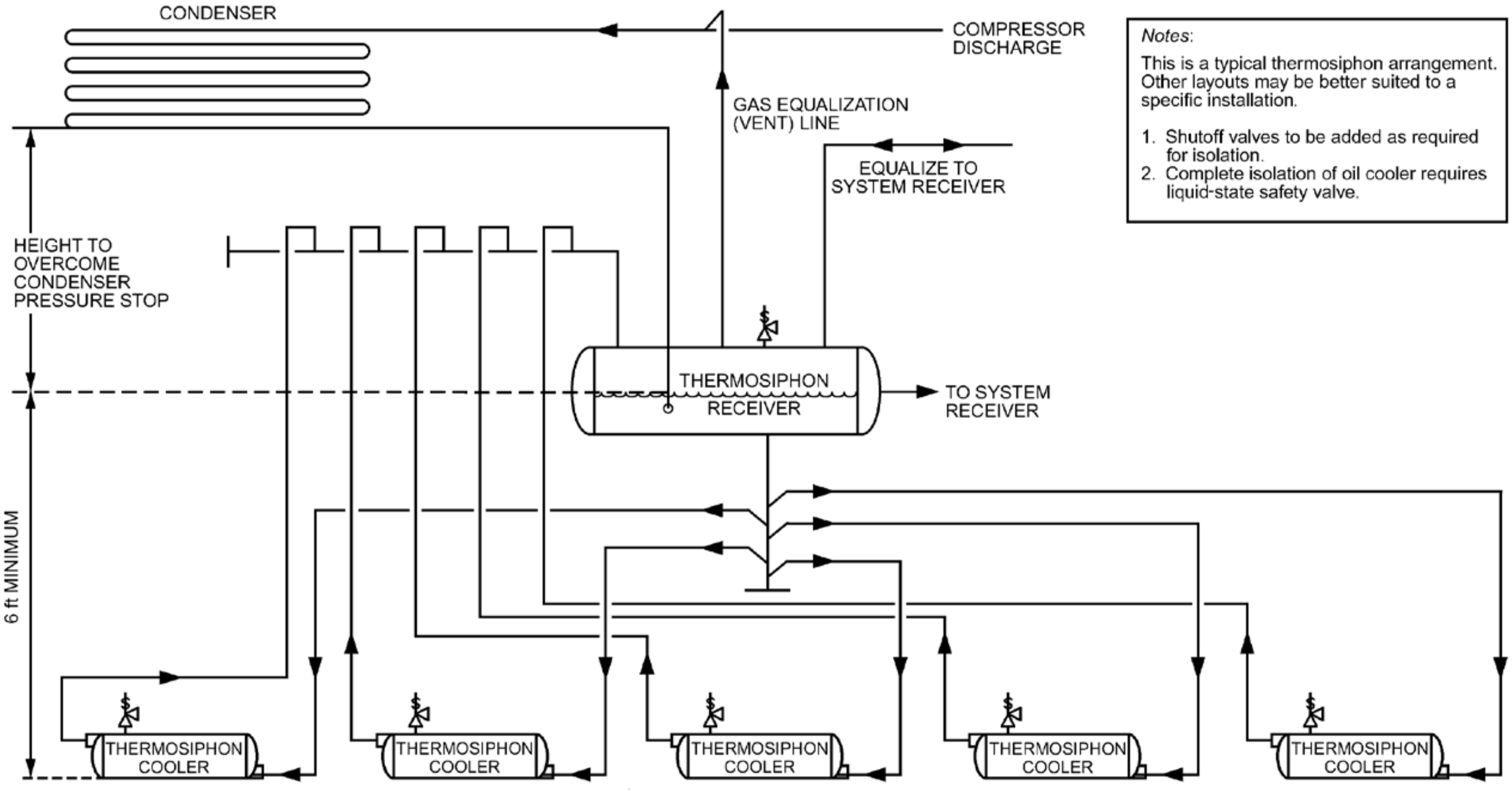



**Fig. 24 Typical Thermosiphon System with Multiple Oil Coolers**

# Heat Rejection by Oil Cooling

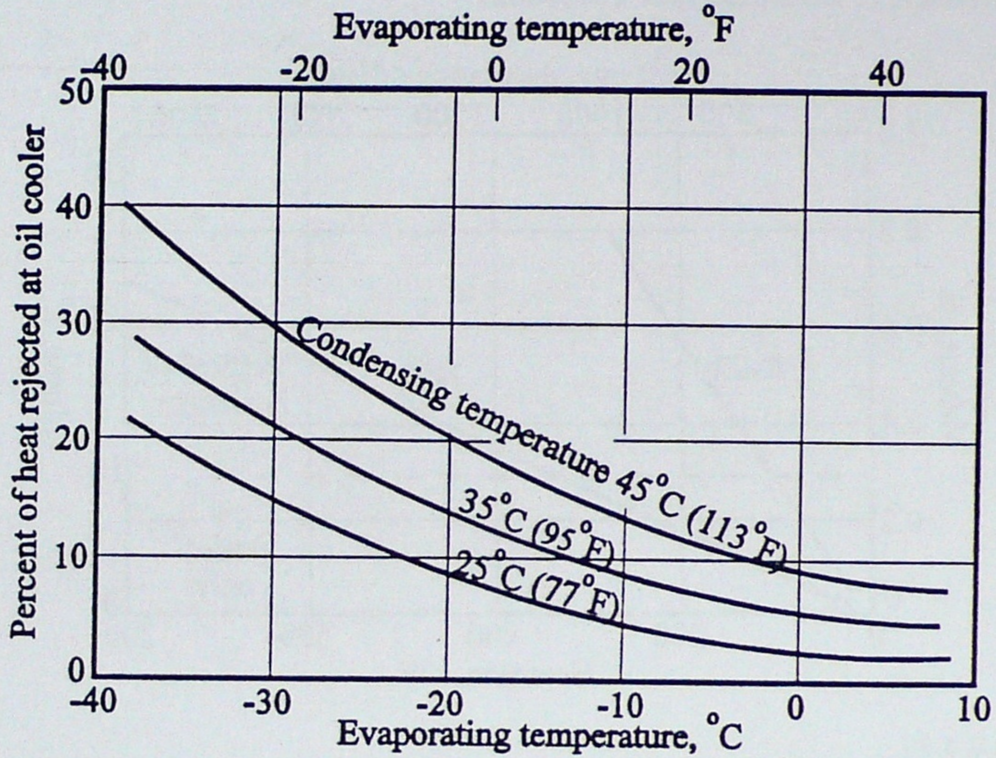
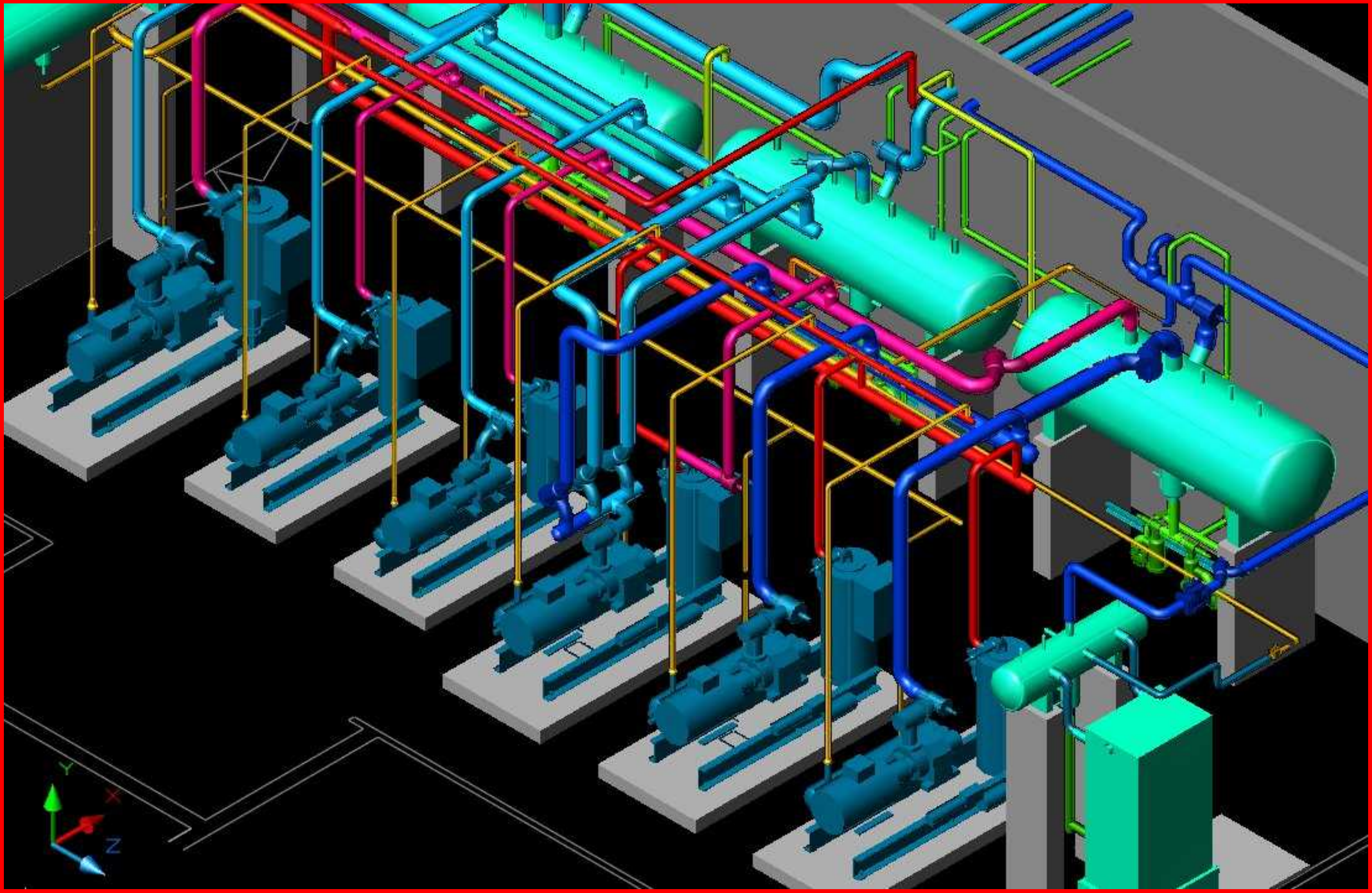



**FIGURE 5.25**
Percentage of heat input (total of refrigeration load and compressor power) that is absorbed by the injected oil in a screw compressor.

# Refrigerant Piping



# Categories of Piping

TABLE 9.1
Categories of piping.

| Role | State of refrigerant | Pressure drop permitted | Geometric requirements |
|---|---|---|---|
| Compressor discharge | Vapor | Moderate | |
| From high-pressure receiver | Liquid | Moderate | Limited increase in elevation |
| In liquid recirculation systems: | | | |
| From pump to evaporators | Liquid | Moderate | |
| Return to low-pressure receiver | Liquid/vapor | Low | Pitch downward |
| Vertical risers | Liquid/vapor | Low | |
| Suction to compressor | Vapor | Low, except for oil return | Trap for oil return in direct-expansion systems |
| Hot-gas lines for defrost | Vapor | Moderate | Trap for liquid removal |

# Effects of Pipe Size

The pipe size exerts a dramatic influence on the velocity and pressure drop. Equation 9.2 can be modifed to show that

$$\frac{V_2}{V_1} = \left(\frac{D_1}{D_2}\right)^2$$

so if the diameter if reduced by 50%, the velocity increases to four times its original value for a given volume flow rate. When the velocity effect is inserted in the pressure drop equation and combined with the appearance of $D$ in the $fL/D$ group, the ratio of the pressure drops varies inversely as the diameter to the 5th power,

$$\frac{\Delta p_2}{\Delta p_1} = \frac{D_1}{D_2}\left(\frac{D_1}{D_2}\right)^4 = \left(\frac{D_1}{D_2}\right)^5$$

Reducing the pipe size by 50% causes a 32-fold increase in the pressure drop, showing that the pressure drop is extremely sensitive to the pipe size.

# Optimum Pipe Size

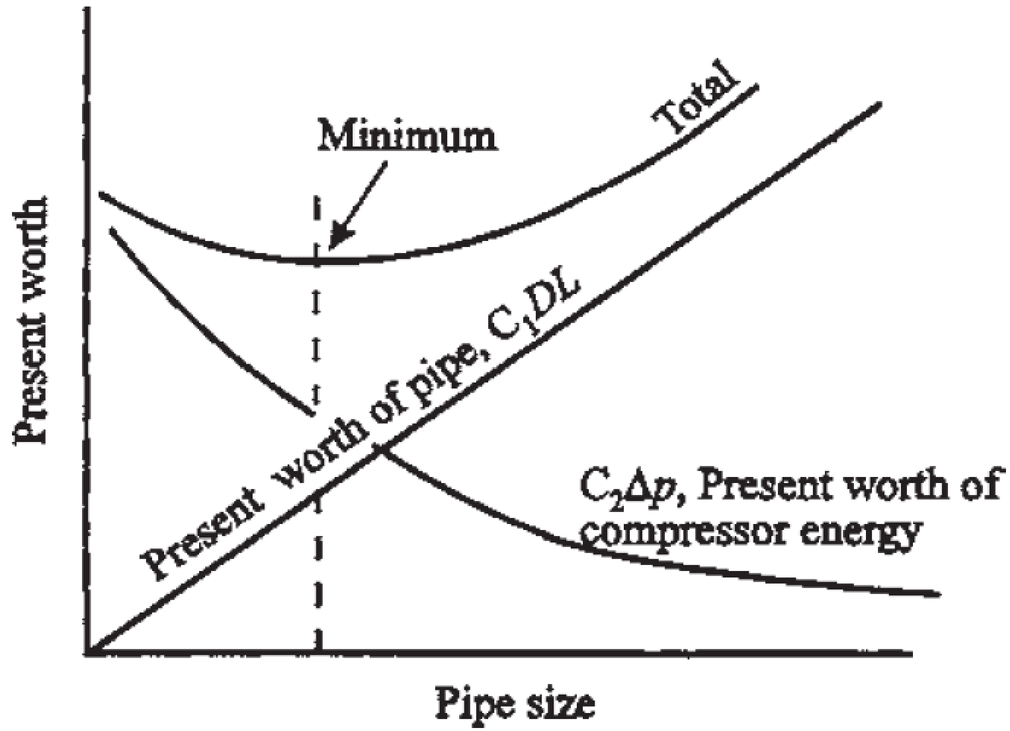



**FIGURE 9.3**
Optimum vapor pipe size at the minimum of the sum of the first cost of the pipe and present worth of the lifetime compressor energy costs.

# Selecting the Pipe Size

- **Suction to compressor.** The total drop in saturation temperature is usually chosen to be 0.5 to 2°C (0.9 to 3.6°F). The exceptions are vertical risers both for halocarbon direct expansion and for ammonia liquid overfeed coils. For halocarbon direct-expansion systems the velocity of the refrigerant vapor must be high enough to convey oil back to the compressor. For ammonia liquid-overfeed coils the vapor velocity in the riser must be high enough to blow the liquid out so that it doesn't fill the riser.
- **Discharge from compressor to condenser.** The total drop in saturation temperature is usually chosen from 1.0 to 3.0°C (1.8 to 5.4°F). A given drop

in saturation temperature in the discharge pipe is slightly less penalizing in compressor power than the same drop in temperature on the suction side.

# Selecting the Pipe Size

- **High-pressure liquid.** A pressure drop in this section may exact no penalty whatsoever on system performance, because the pressure drop that does not occur in the pipe will take place in the expansion device or level-control valve. The expansion device provides the final reduction of pressure to the intermediate pressure (in two-stage compression) or to the low pressure (in single-stage compression). The concern about pressure drop in this line arises more in assuring that the pressure does not drop to the saturation pressure corresponding to the existing refrigerant temperature. Were the pressure to drop to that point, the liquid would flash into vapor, aggravate the pressure gradient, and possibly restrict the flow through the expansion device. Refrigerant velocities chosen for liquid lines range from 1 to 2.5 m/s (3 to 8 ft/s).
- **Liquid/vapor return from evaporators to low-pressure receiver.** The line from the evaporators back to the low-pressure receiver in liquid recirculation systems carries a mixture of liquid and vapor. Calculations of pressure drops in the flow of liquid/vapor mixtures, while possible, are complex. To avoid cumbersome calculatons, yet still make allowances for the presence of liquid, some designers choose the line size, first by determining the appropriate size if the pipe were carrying only vapor, then step up to the next pipe size to allow for the combined flow of liquid.

# Selecting the Pipe Size

- **Hot-gas defrost lines.** To make an intelligent choice of pipe size, the required flow rate of hot gas as a function of the evaporator size should be known. A rough estimate of the hot-gas flow rate is that it is twice the refrigerant flow rate used during refrigeration service. With this assumption, the recommended sizes of ammonia hot-gas branch lines proposed by Hansen[9] use as a basis a velocity of 15 m/s (3000 fpm) with 21°C (70°F) hot gas. This velocity would be appropriate for hot-gas branch lines serving a single evaporator of a cluster of evaporators defrosted at the same time. The hot-gas mains may be sized for carrying half the total of all connected evaporators on the assumption that no more than half the evaporators would be defrosted at one time.

  Recent efforts to operate plants with as low a condensing temperature as possible impacts the desired size of hot-gas lines. The ultimate criterion is the saturation temperature at which the defrost gas can condense in the evaporator being defrosted, so the drop in saturation temperature in the hot-gas line reappears as the most appropriate basis for selecting the pipe size. As the condensing temperature of the plant drops, the defrost gas becomes less dense, and when the condensing temperature of the plant drops from 35°C (95°F) to 15°C (59°F), for example, the drop in saturation temperature for several common refrigerants doubles[10].

# Rising of Liquid Line

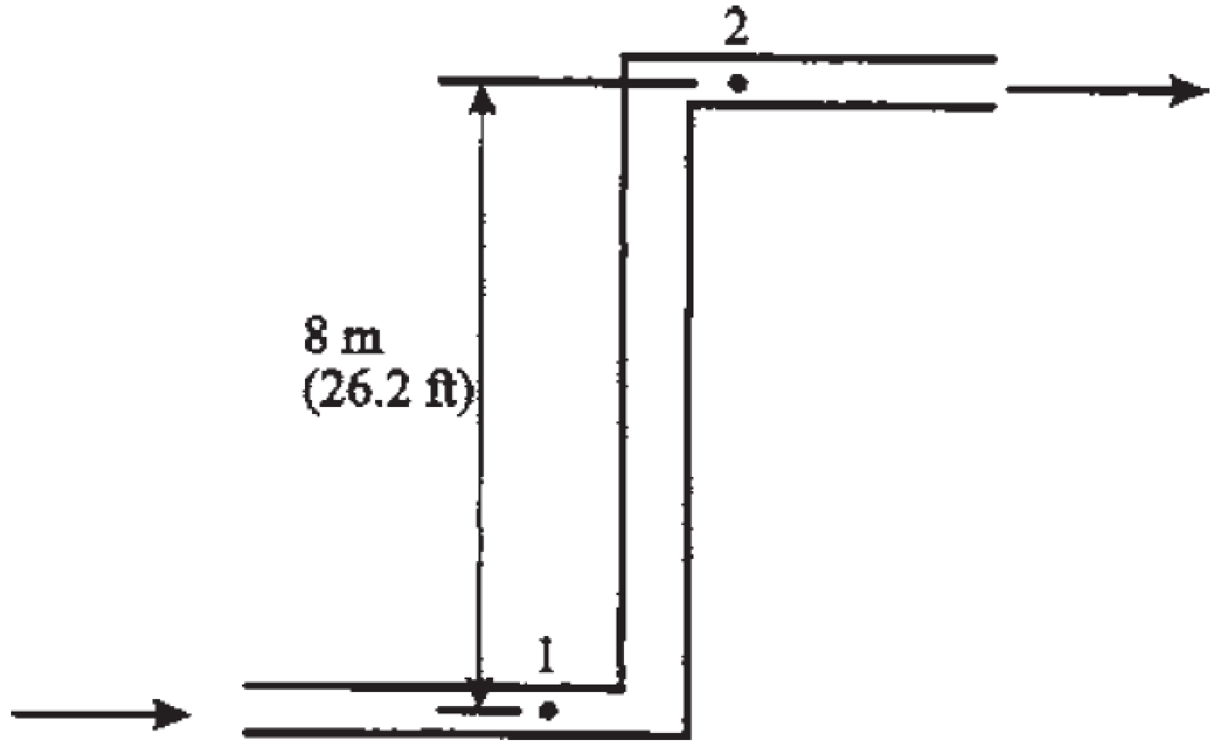



**FIGURE 9.4**
Liquid line riser in Example 9.4.

# Downward Sloping

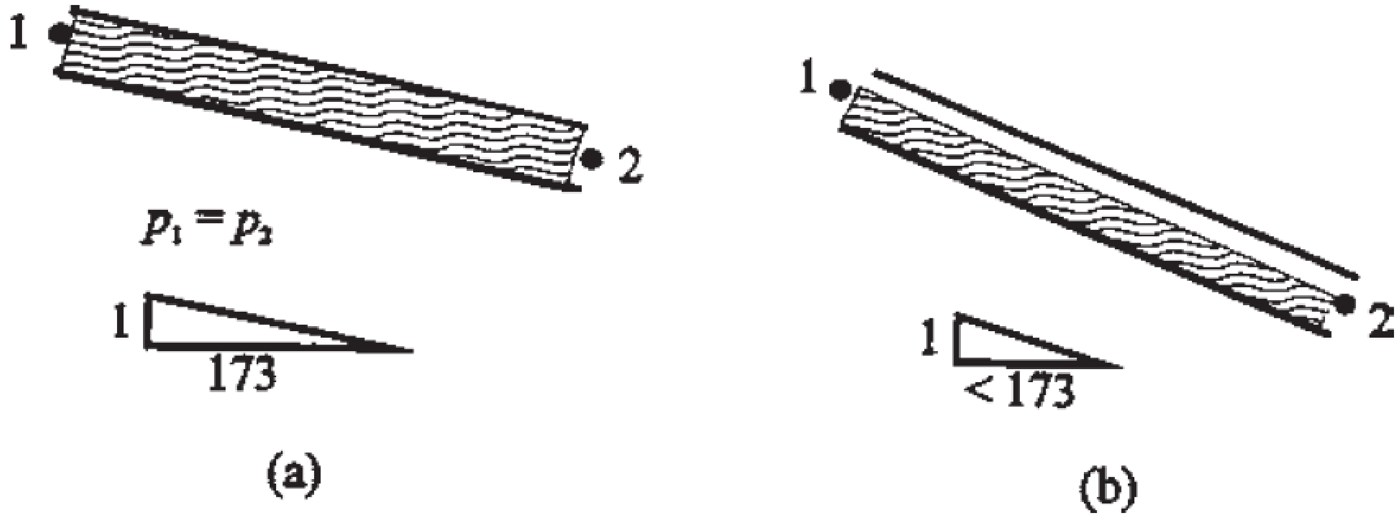



**FIGURE 9.5**
Downward pitch of the liquid ammonia line in Example 9.5: (a) where the pressure drop due to friction is precisely canceled and (b) where the slope is greater and the line runs partially filled with liquid.

# Avoiding Drainage

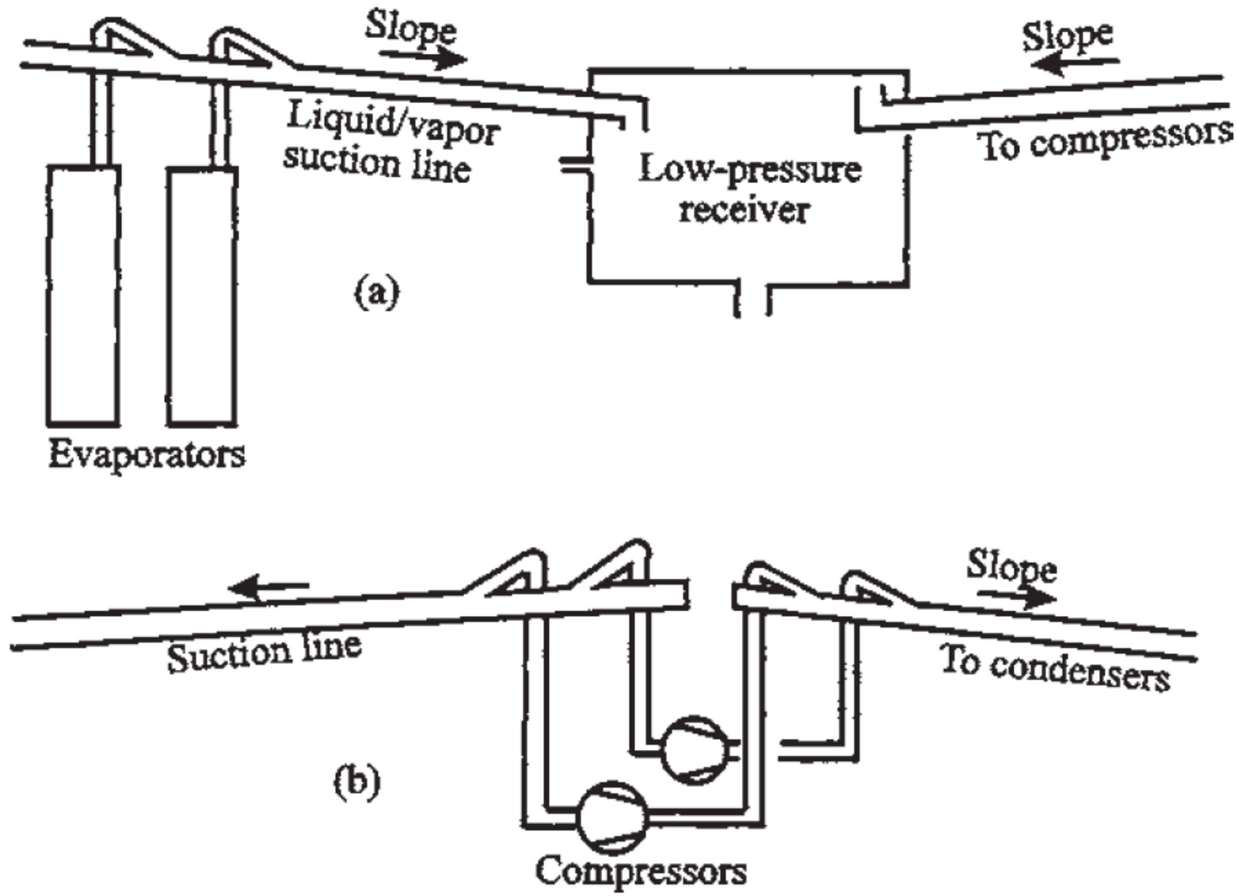



**FIGURE 9.6**
Slope of vapor lines and branch connections from evaporators and compressors.

# Sizing of Wet Return Line


**FIGURE 9.11**
Liquid/vapor return line that is (a) horizontal and (b) sloped slightly in the direction of flow.

The designer must select the size of and compute the pressure drop in the liquid/vapor suction lines shown in Fig. 9.11. The classic method for computing the pressure drop of two-phase, liquid/vapor flow in horizontal lines is the Lockhart and Martinelli correlation[12]. When using this method, the pressure gradient of each phase is computed as though it flowed separately in the pipe. Then, a graph provided by Lockhart and Martinelli combines those two gradients to predict the two-phase gradient. Most designers of refrigeration systems do not expend the effort to make this calculation. Designers often select the pipe

by first determining the size that gives the desired pressure gradient were the vapor alone flowing in the pipe. Then the next larger pipe size is selected with the extra cross-sectional area of the pipe assumed to provide the area for the flow of liquid.

# Lifting Liquid From The Evaporator in Vertical Risers

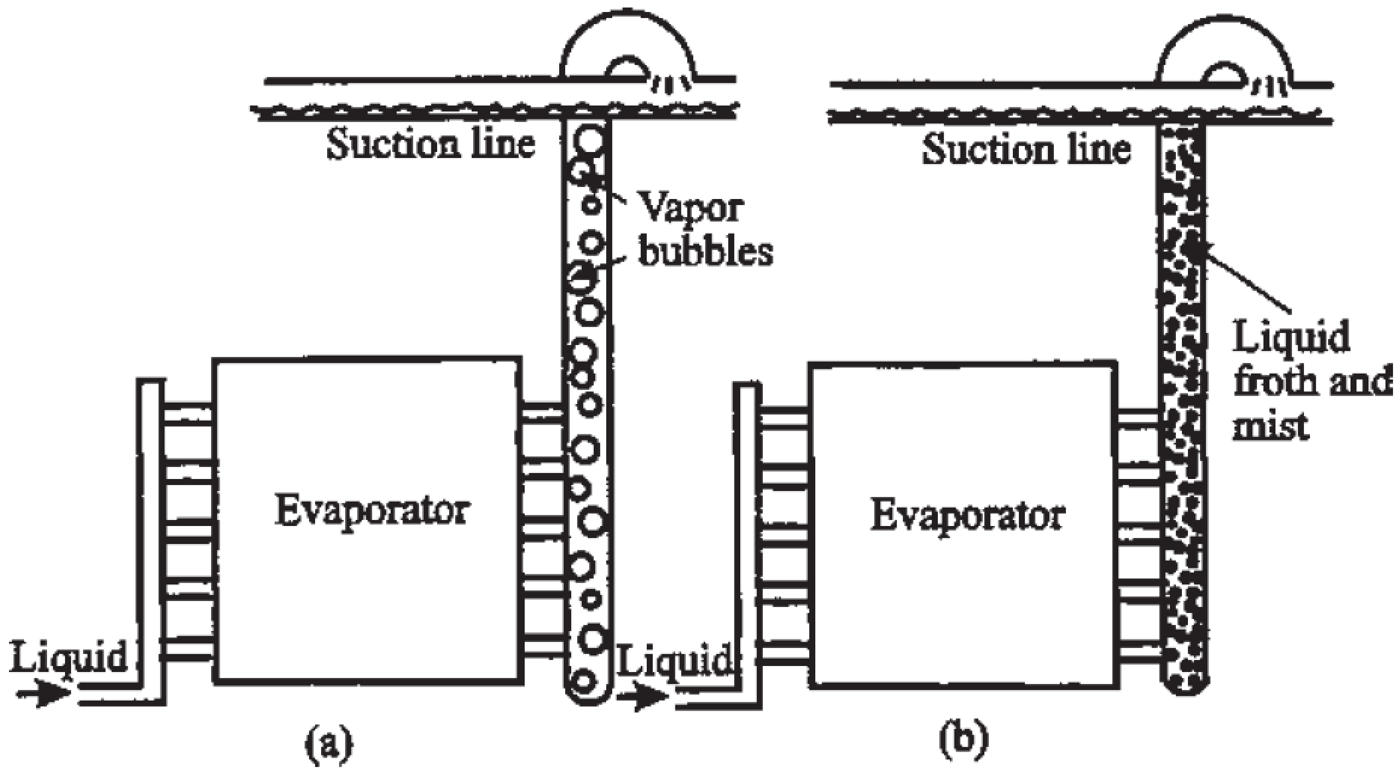



**FIGURE 9.12**
(a) Low vapor velocity in a liquid/vapor riser resulting in a high liquid fraction and (b) vapor velocity high enough to blow most of the liquid out of the riser.

# Optimum Velocity in Riser

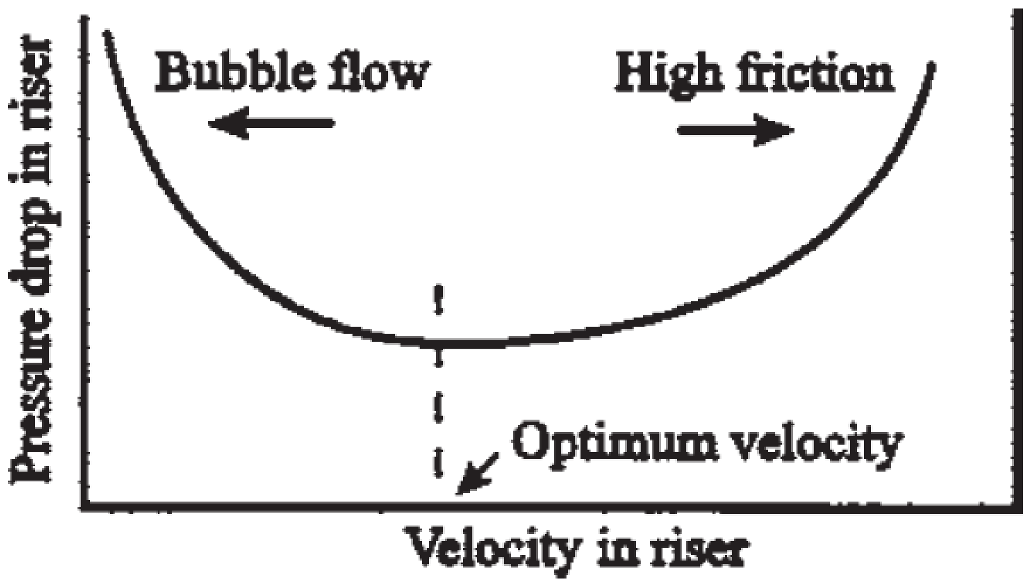



**FIGURE 9.14**
Optimum velocity in riser when lifting liquid with the flow of vapor.

# Optimum Riser Sizing

**TABLE 9.6**

Evaporator capacities resulting in minimum pressure drops for vertical risers with various pipe sizes and circulation ratios. The refrigerant is ammonia at -40°C (-40°F).

| Circulation ratio | Pipe diameter of riser, mm (inch) | | | | | | |
|---|---|---|---|---|---|---|---|
| | 37 (1.5) | 50 (2.0) | 63 (2.5) | 75 (3.0) | 100 (4.0) | 125 (5.0) | 150 (6.0) |
| 2.5 | 18.6 kW (5.3 tons) | 34.8 (9.9) | 53.4 (15.2) | 90.0 (25.6) | 184.2 (52.4) | 324 (92.2) | 513 146) |
| 3.15 | 17.6 kW (5.0 tons) | 32.7 (9.3) | 50.3 (14.3) | 87.9 (25.0) | 173 (49.3) | 305 (86.7) | 482 (137) |
| 4.0 | 16.9 kW (4.8 tons) | 31.3 (8.9) | 48.5 (13.8) | 84.4 (24.0) | 167 (47.4) | 293 (83.4) | 464 (132) |
| 5.0 | 16.2 kW (4.6 tons) | 30.2 (8.6) | 46.8 (13.3) | 81.2 (23.1) | 160 (45.6) | 282 (80.1) | 447 (127) |

# Lifting Liquid From The Evaporator in Vertical Risers

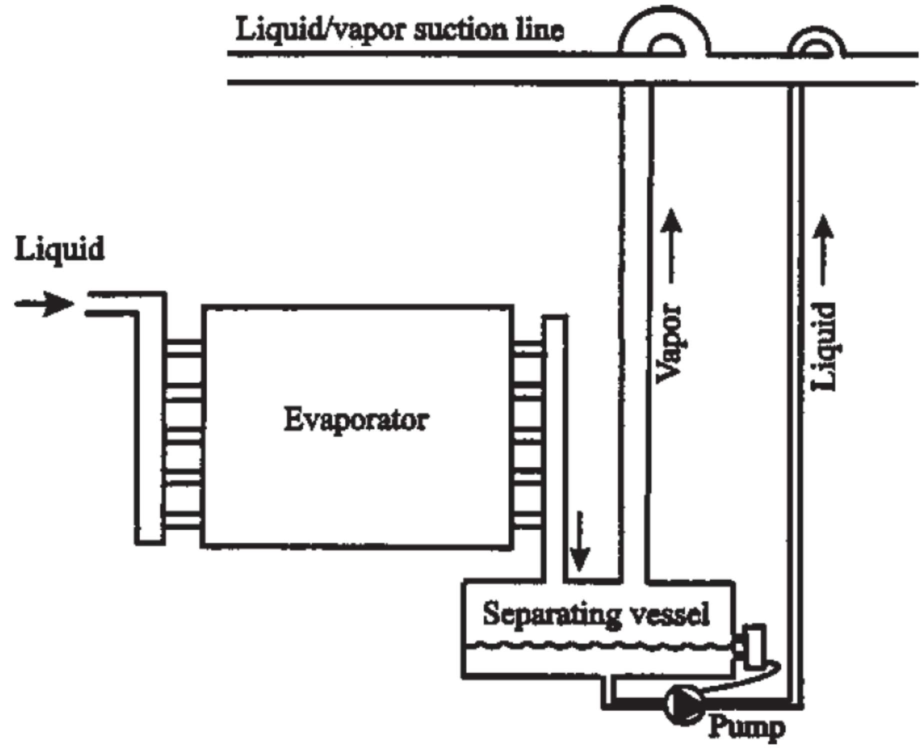



**FIGURE 9.13**
Lifting liquid from the outlet of the evaporator by first separation, then pumping.

# Pipe Safety

- **Choice of pipe and fittings.** Regardless of pressure, the pipe should be Schedule 80 for 1–1/2-in size and smaller. Use Schedule 40 pipe for 2- to 6-in lines and Schedule 20 for 8- to 12-in lines. It may seem illogical to specify the thick Schedule 80 pipe for small diameter, but the reason is not to better contain internal pressure, it is to provide more rigidity should a heavy weight fall on or against the pipe. Joints in pipe of size 1–1/2-in and larger should be welded, not threaded. In fact many contractors weld all joints in 3/4-in pipe and larger.
- **Support and anchoring of pipe.** Pipe hangers should be of the trapeze type and all hanger materials should be galvanized or painted. Hangers should be attached to the building structure or primary supports. Pipe hangers should be placed not more than 3 m (10 ft) apart and should be located not more than 0.7 m (2 ft) from each change of direction. Supports of piping on roofs should be sturdy, painted, or galvanized. The support should rest on a concrete paver or wood sleeper with the roof beneath protected by a membrane material.

# Pipe Safety

- **Location of pipe runs and valves.** When possible, long lines between the machine room and refrigerated space should be run on the roof. When pipes must be inside storage or process areas, they should be in protected locations. Valves should be located in and out of each component of the

  same size as the connecting pipe. Hand valves should close against the flow, and the stems should be oriented horizontally. Screwed connections to valves should not be used for sizes larger than 1 in. Horizontal piping shall have only eccectric reducers with the straight sides on the bottom, except at a pump inlet where the straight side should be on top.
- **Cleanliness of pipe.** For installation, the pipe must be clean, new and free of rust, scale, sand, and dirt. Pipe should be stored inside, out of the weather and the ends should be capped. Some contractors install a liberal number of filter/driers which are replaced periodically.

# Piping Materials

Copper is the material almost universally used for halocarbon piping, with no low-temperature limitations in industrial refrigeration practice. For ammonia systems, since copper is not acceptable, all pipe is steel in one form or another. While aluminum is often used for the tubes of air-cooling coils, it is hardly ever used for liquid and vapor piping in a plant. Cast iron pipe should not be used and wrought iron pipe should not be used for liquid lines.

The most commonly used steel for ammonia systems is carbon steel of type A53 or A106. Type A53 F is butt-welded and should not be used, on the other hand A106 is seamless, so is acceptable. The ANSI/ASME Refrigeration Piping Standard[14] permits the use of A53 or A106 down to temperatures of -29°C (-20°F). One of the steels permitted by the ANSI/ASME standard below this temperature is type A333. Compared to A53 or A106, A333 pipe is approximately three times as expensive, the the cost factor is even higher for valves and fittings. A53 and A106 steels may be used below -29°C (-20°F) provided they pass impact tests at the temperature at which they will be operated. Obviously, impact testing adds to the cost. Another option offered by the standard is to design for the circumferential or longitudinal tensile stress to be 40% of the allowable stress given for the material. Refrigeration interests in recent years have been attempting to get A53 and A106 steels approved for below -29°C (-20°F) operation on the basis that with ammonia the pressure of the refrigerant will be in the range of atmospheric, so that pressure stress on the pipe will be negligible.

At temperatures lower than -46°C (-5°F) many designers uses stainless steel for ammonia and hydrocarbons.

# Piping Insulation

The cost of the insulation of low-temperature pipe is usually of the same order of magnitude as that of the pipe itself. Also, since the diameter of the insulated pipe may be twice or three times that of the pipe itself, the designer must provide sufficient space for the complete assembly.

Vapor barriers are essential for low-temperature pipes. In contrast to the water vapor path through the insulation of a refrigerated building which enters the insulation from the warm outdoors and eventually is extracted by the coils in the refrigerated space, there is no outlet for the water vapor that enters pipe insulation. For this reason, the water vapor retardation should be as effective as possible for prolonging the life of the insulation. Some specifications[15] for insulation on low-temperature pipe include.

- Cover the insulation with vapor-retarding jackets with a perm rating of 0.02 or less.
- The supports must not be in contact with the piping.
- The thickness of insulation must be sufficient to maintain the insulation surface temperature above the dewpoint. Isocyanurate or Styrofoam are recommended.
- Since the cladding, insulation, and the pipe all have different coefficients of thermal expansion, double-layer application is suggested for extremely low temperature insulation with staggered joint construction.
- Pack contraction joints with flexible insulation material.

# Thank You for Your Attention!

Any Question?