冷凍空調基本原理與節能

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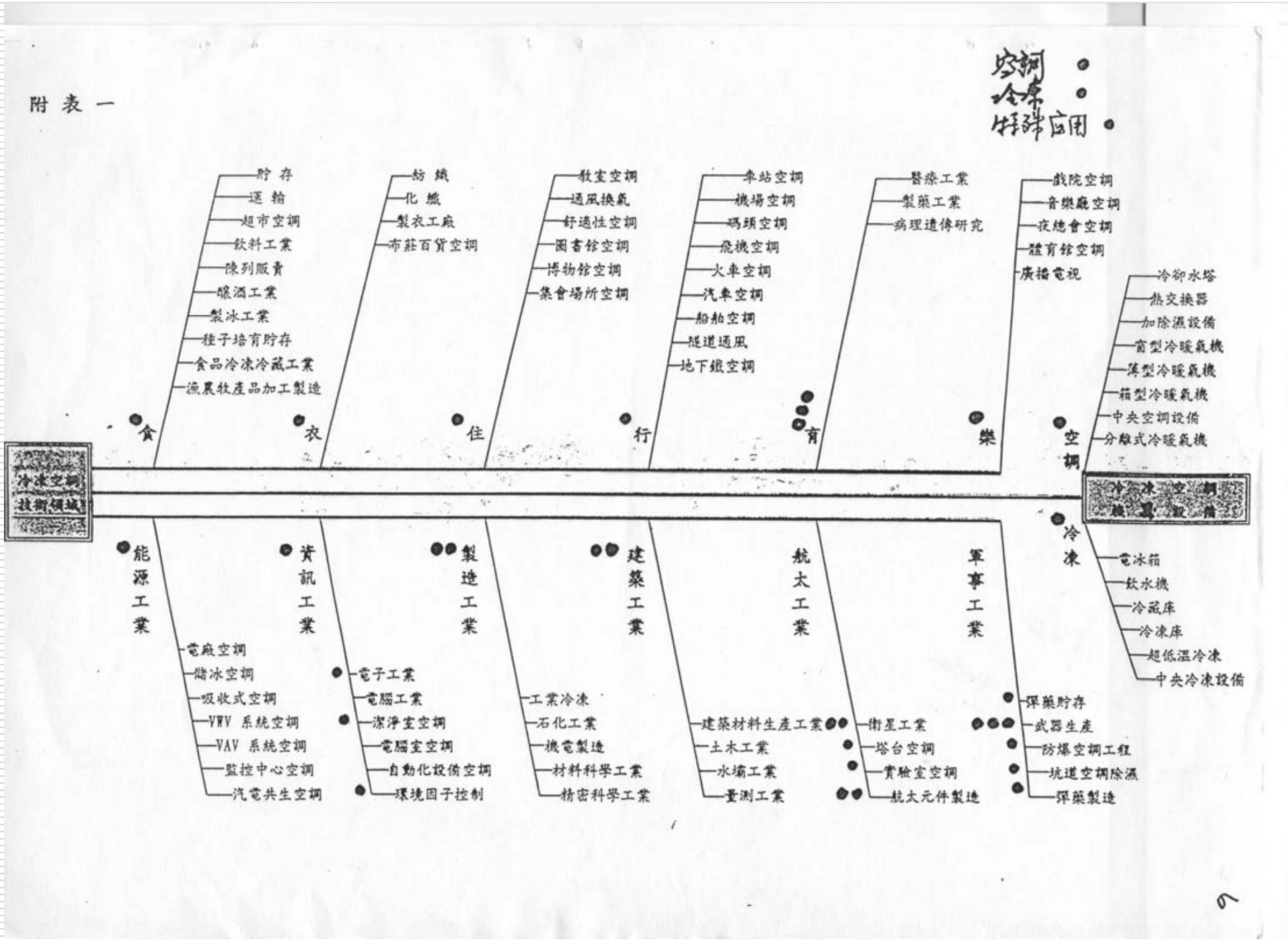
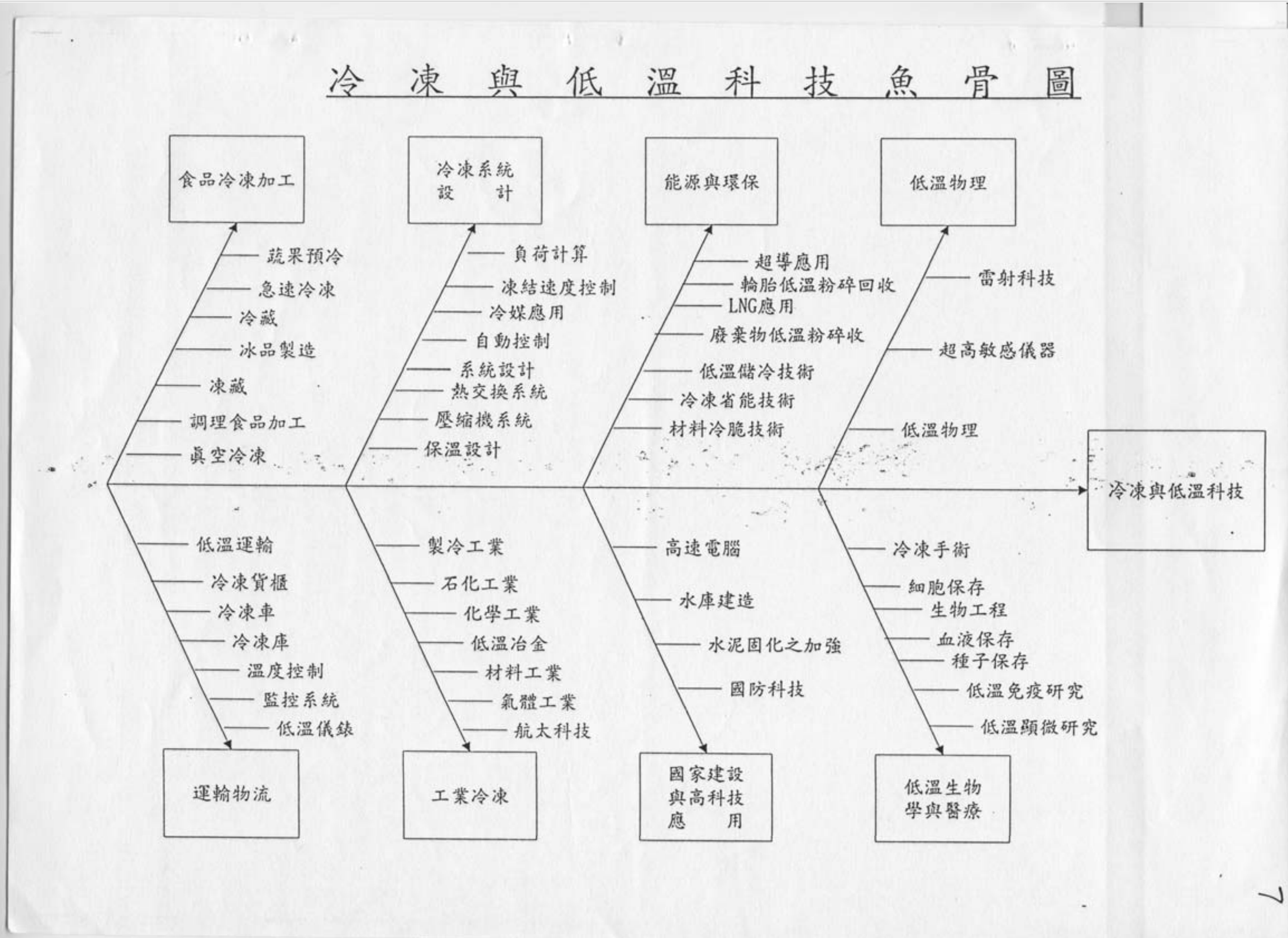
台北科技大學能源與冷凍空調系

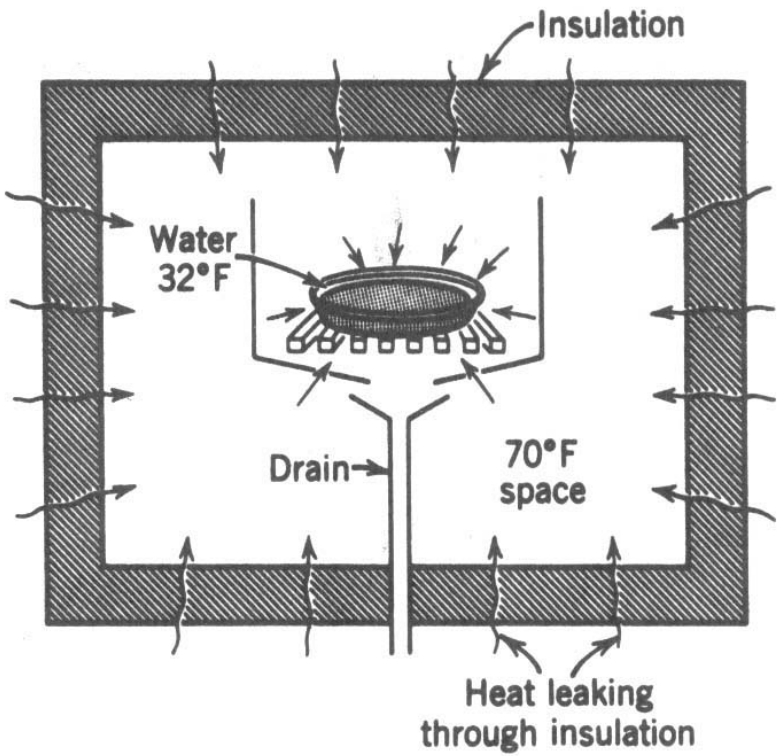
冷凍空調之應用領域


附表一
空調
冷凍
特殊應用
冷凍空調技術領域
冷凍空調機具設備
食
貯存
運輸
超市空調
飲料工業
陳列販賣
釀酒工業
製冰工業
種子培育貯存
食品冷凍冷藏工業
漁農牧產品加工製造
衣
紡織
化纖
製衣工廠
布莊百貨空調
住
教室空調
通風換氣
舒適性空調
圖書館空調
博物館空調
集會場所空調
行
車站空調
機場空調
碼頭空調
飛機空調
火車空調
汽車空調
船舶空調
隧道通風
地下鐵空調
育
醫療工業
製藥工業
病理遺傳研究
樂
戲院空調
音樂廳空調
夜總會空調
體育館空調
廣播電視
空調
冷卻水塔
熱交換器
加除濕設備
窗型冷暖氣機
薄型冷暖氣機
箱型冷暖氣機
中央空調設備
分離式冷暖氣機
能源工業
電廠空調
儲冰空調
吸收式空調
VWV 系統空調
VAV 系統空調
監控中心空調
汽電共生空調
資訊工業
電子工業
電腦工業
潔淨室空調
電腦室空調
自動化設備空調
環境因子控制
製造工業
工業冷凍
石化工業
機電製造
材料科學工業
精密科學工業
建築工業
建築材料生產工業
土木工業
水埔工業
量測工業
航太工業
衛星工業
塔台空調
實驗室空調
航太元件製造
軍事工業
彈藥貯存
武器生產
防爆空調工程
坑道空調除濕
彈藥製造
冷凍
電冰箱
飲水機
冷藏庫
冷凍庫
超低溫冷凍
中央冷凍設備

冷凍之應用領域

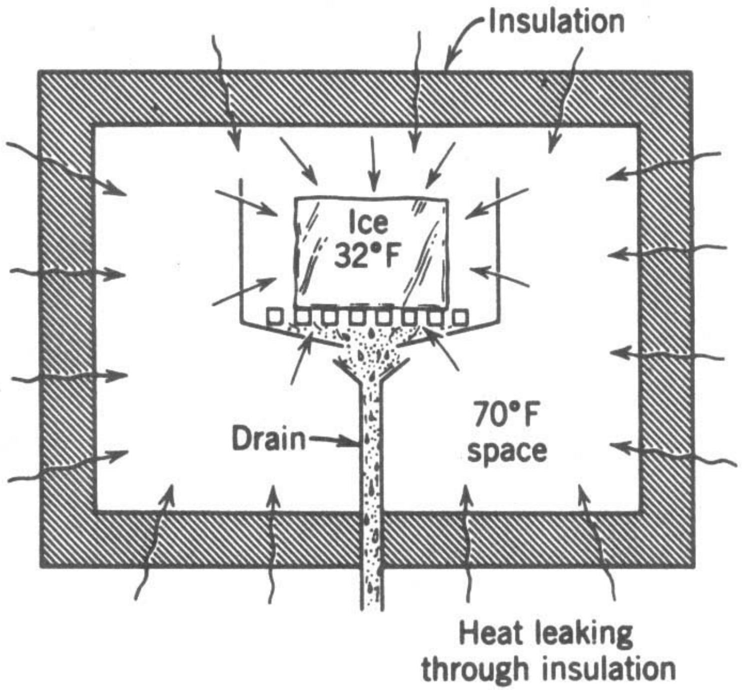

冷 凍 與 低 溫 科 技 魚 骨 圖
食品冷凍加工
冷凍系統設計
能源與環保
低溫物理
蔬果預冷
急速冷凍
冷藏
冰品製造
凍藏
調理食品加工
真空冷凍
負荷計算
凍結速度控制
冷媒應用
自動控制
系統設計
熱交換系統
壓縮機系統
保溫設計
超導應用
輪胎低溫粉碎回收
LNG應用
廢棄物低溫粉碎收
低溫儲冷技術
冷凍省能技術
材料冷脆技術
雷射科技
超高敏感儀器
低溫物理
冷凍與低溫科技
低溫運輸
冷凍貨櫃
冷凍車
冷凍庫
溫度控制
監控系統
低溫儀錶
製冷工業
石化工業
化學工業
低溫冶金
材料工業
氣體工業
航太科技
高速電腦
水庫建造
水泥固化之加強
國防科技
冷凍手術
細胞保存
生物工程
血液保存
種子保存
低溫免疫研究
低溫顯微研究
運輸物流
工業冷凍
國家建設與高科技應用
低溫生物學與醫療

蒸氣壓縮循環製冷原理與系統之進化



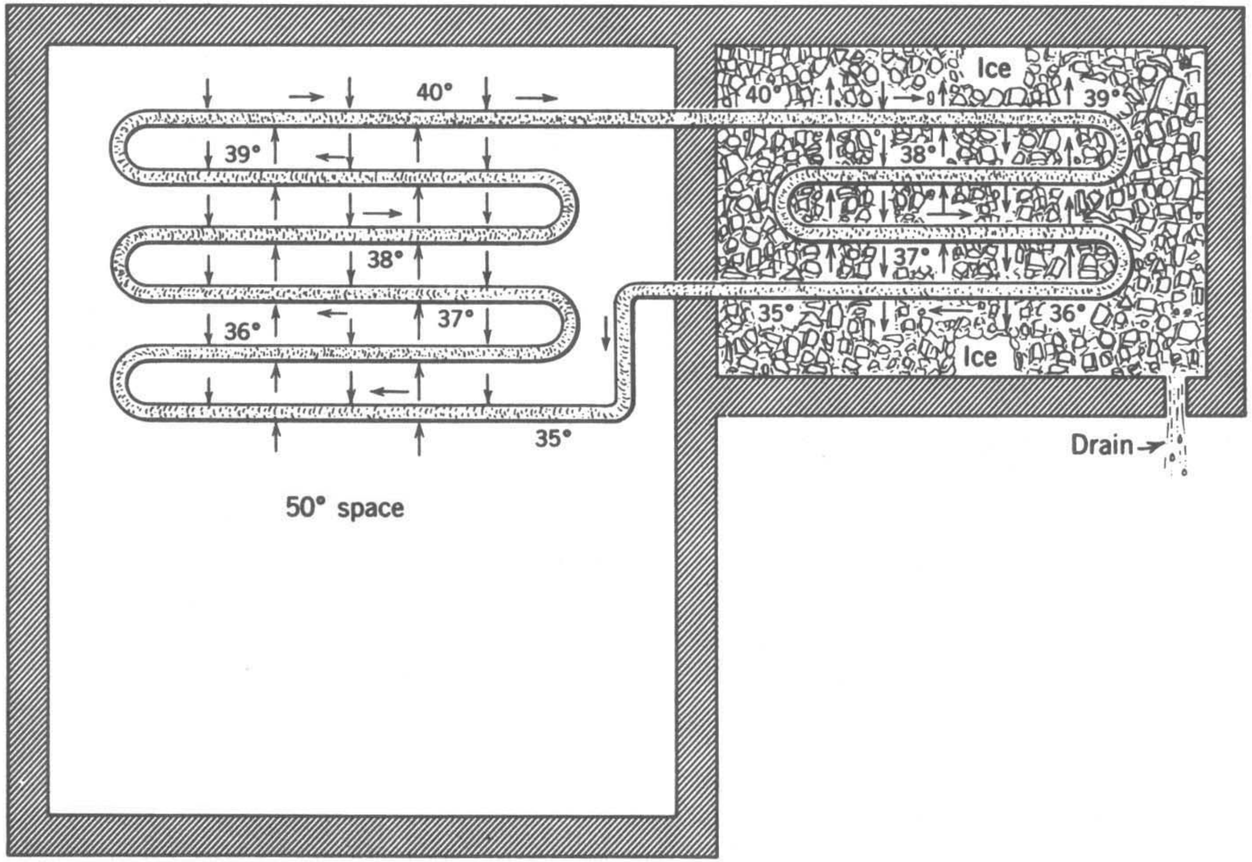
Heat flows from warm space to cold water. Water temperature rises as space temperature decreases. Refrigeration will not be continuous.

蒸氣壓縮循環製冷原理與系統之進化



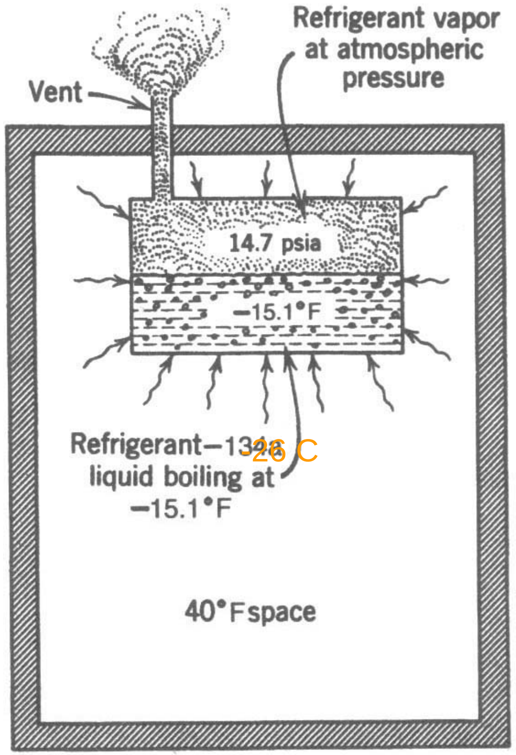
Heat flows from warm space to cold ice. Temperature of space decreases as ice melts. Temperature of ice remains at 32°F. Heat absorbed by ice leaves space in water going out the drain.

蒸氣壓縮循環製冷原理與系統之進化



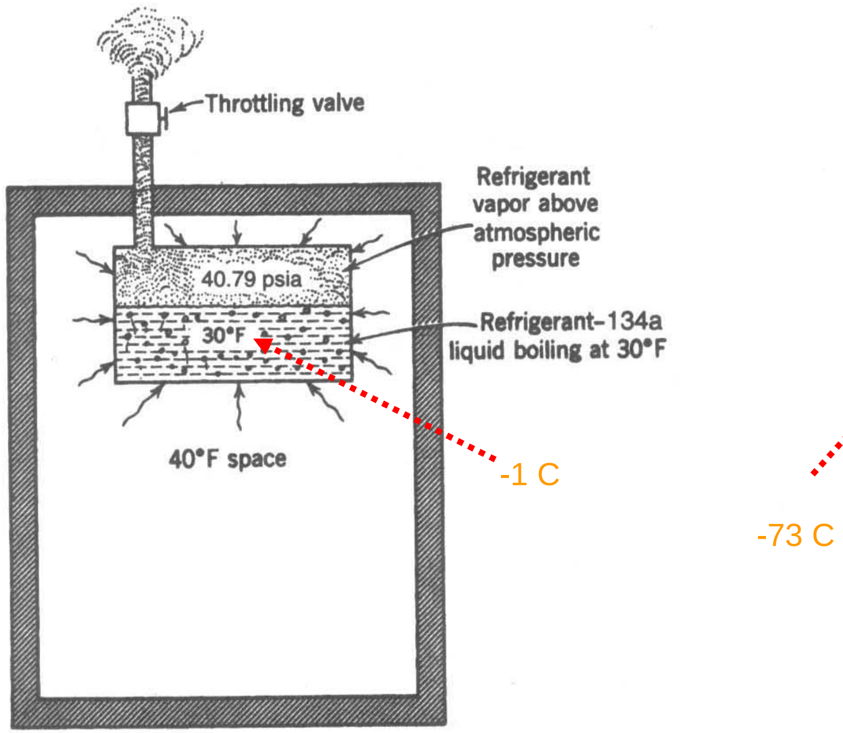
Continuous sensible cooling. Heat taken in by the water in the space is given up to the ice.

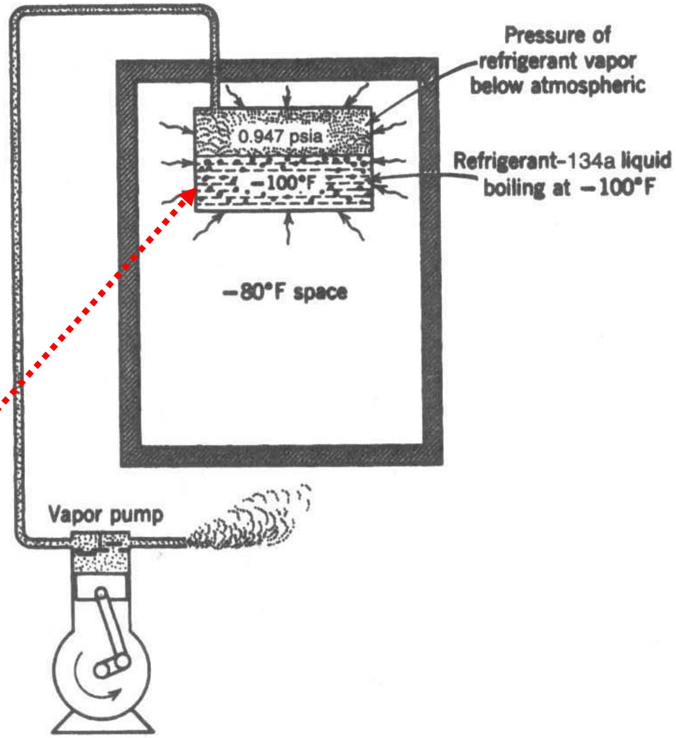
蒸氣壓縮循環製冷原理與系統之進化



The Refrigerant-134a liquid vaporizes as it takes in heat from the 40°F space. The heat taken in by the refrigerant leaves the space in the vapor escaping through the vent.

蒸氣壓縮循環製冷原理與系統之進化

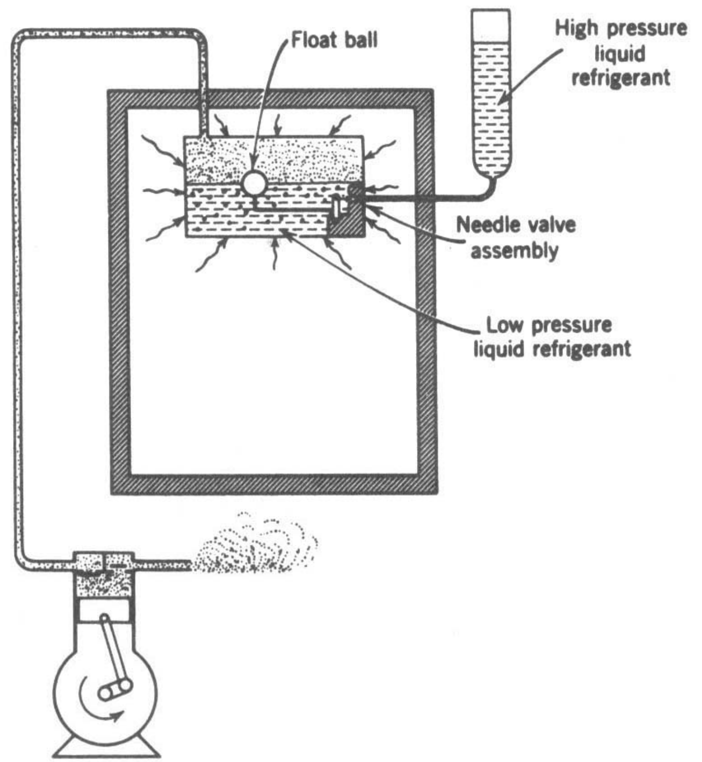




The boiling temperature of the liquid refrigerant in the evaporator is controlled by controlling the pressure of the vapor over the liquid with the throttling valve in the vent.

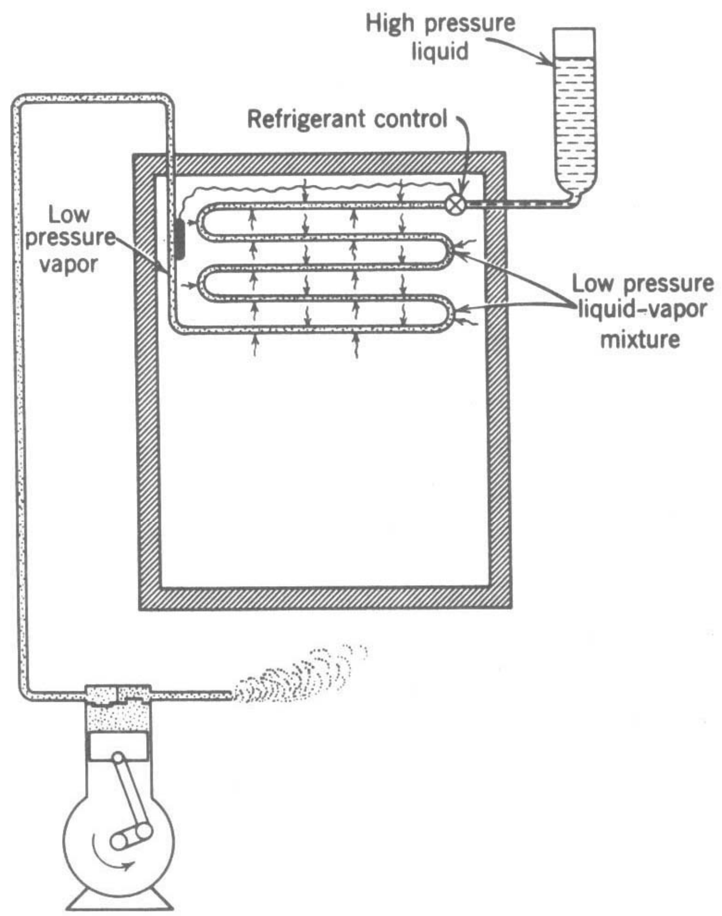
Fig. 6-7 Pressure of refrigerant in evaporator reduced below atmospheric by action of a vapor pump.

蒸氣壓縮循環製冷原理與系統之進化



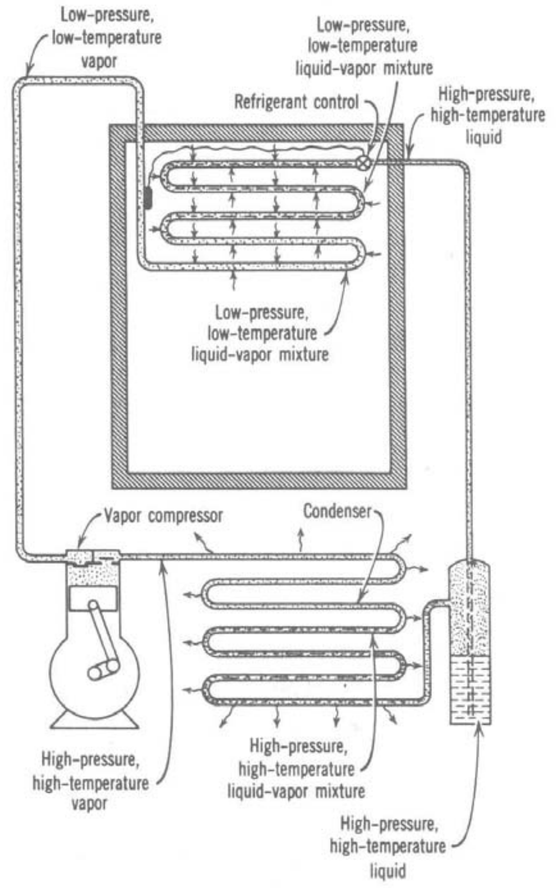
Float valve assembly maintains constant liquid level in evaporator. The pressure of the refrigerant is reduced as the refrigerant passes through the needle valve.

蒸氣壓縮循環製冷原理與系統之進化



Serpentine coil evaporator with thermostatic expansion valve refrigerant control.

蒸氣壓縮循環製冷原理與系統之進化



Collecting and condensing the refrigerant vapor. Refrigerant absorbs heat in evaporator and gives off heat in the condenser.

蒸氣壓縮循環製冷原理與系統之進化

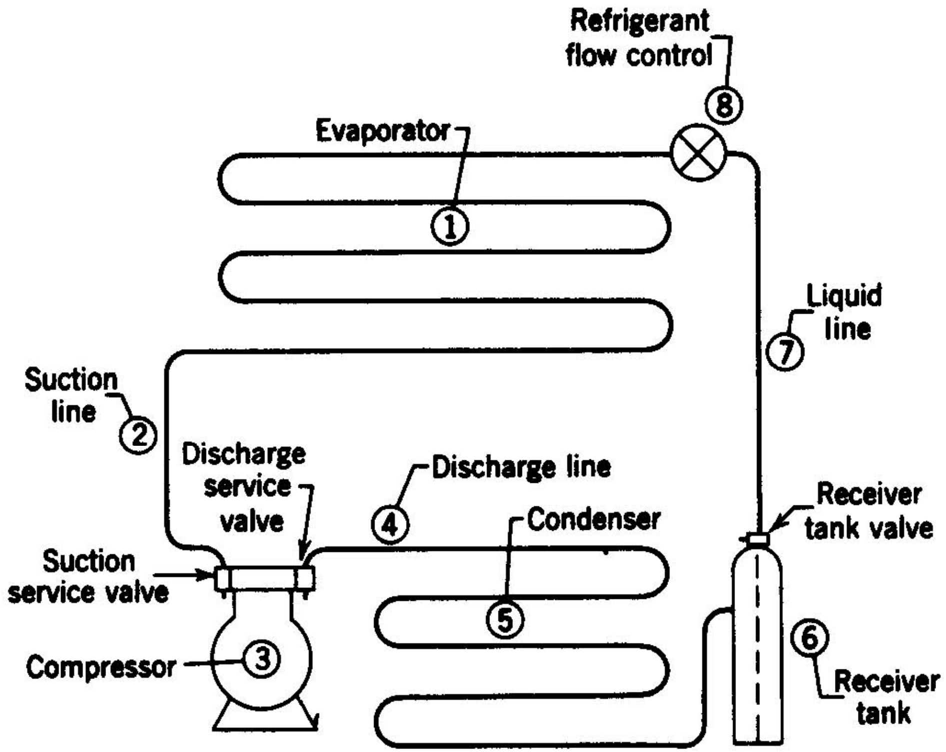
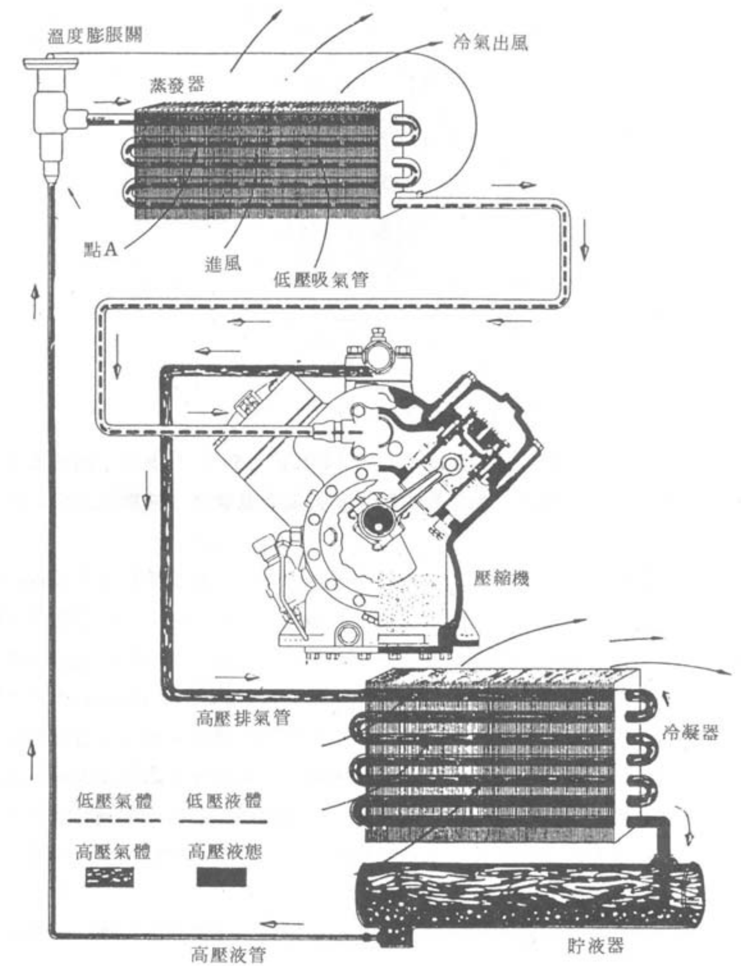


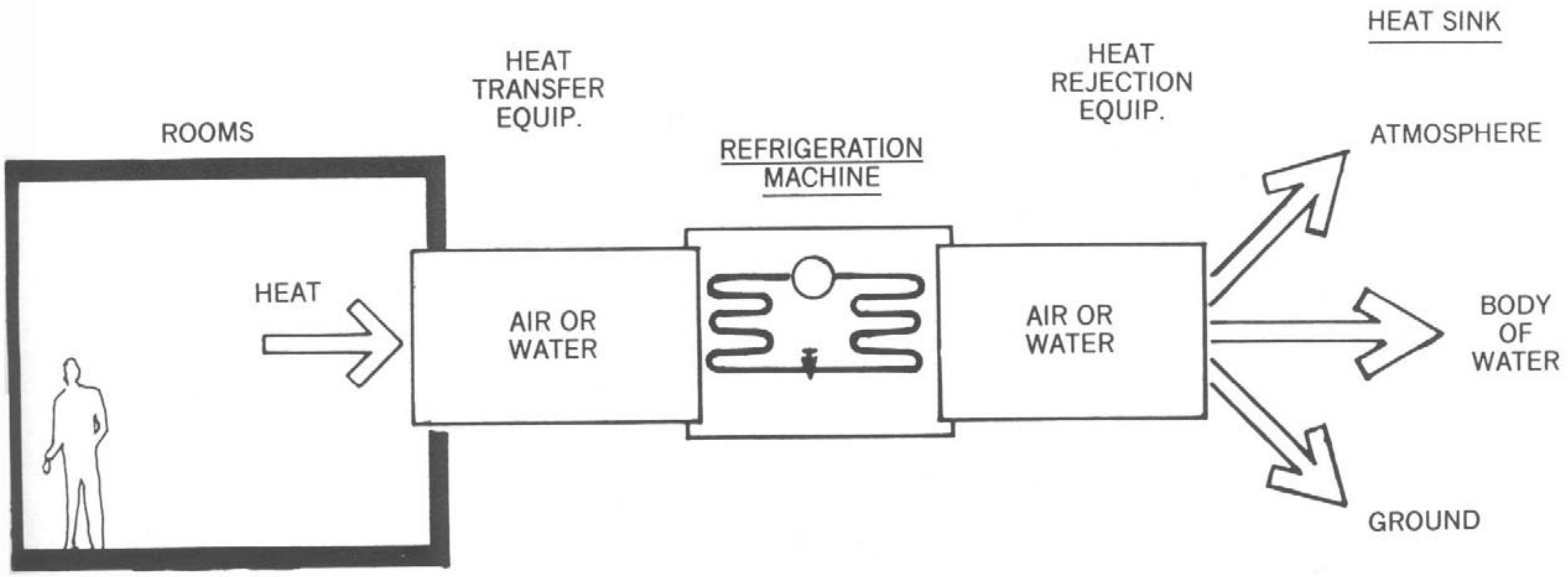
Fig. Flow diagram of simple vapor compression system showing the principal parts.

蒸氣壓縮循環製冷原理與系統之進化



典型之壓縮循環系統應用之結構圖

蒸氣壓縮循環製冷原理與系統之進化



Cooling systems vary mainly in how heat is transferred to and from the refrigeration machine.

蒸氣壓縮循環製冷原理與系統之進化

蒸氣壓縮循環製冷原理動畫

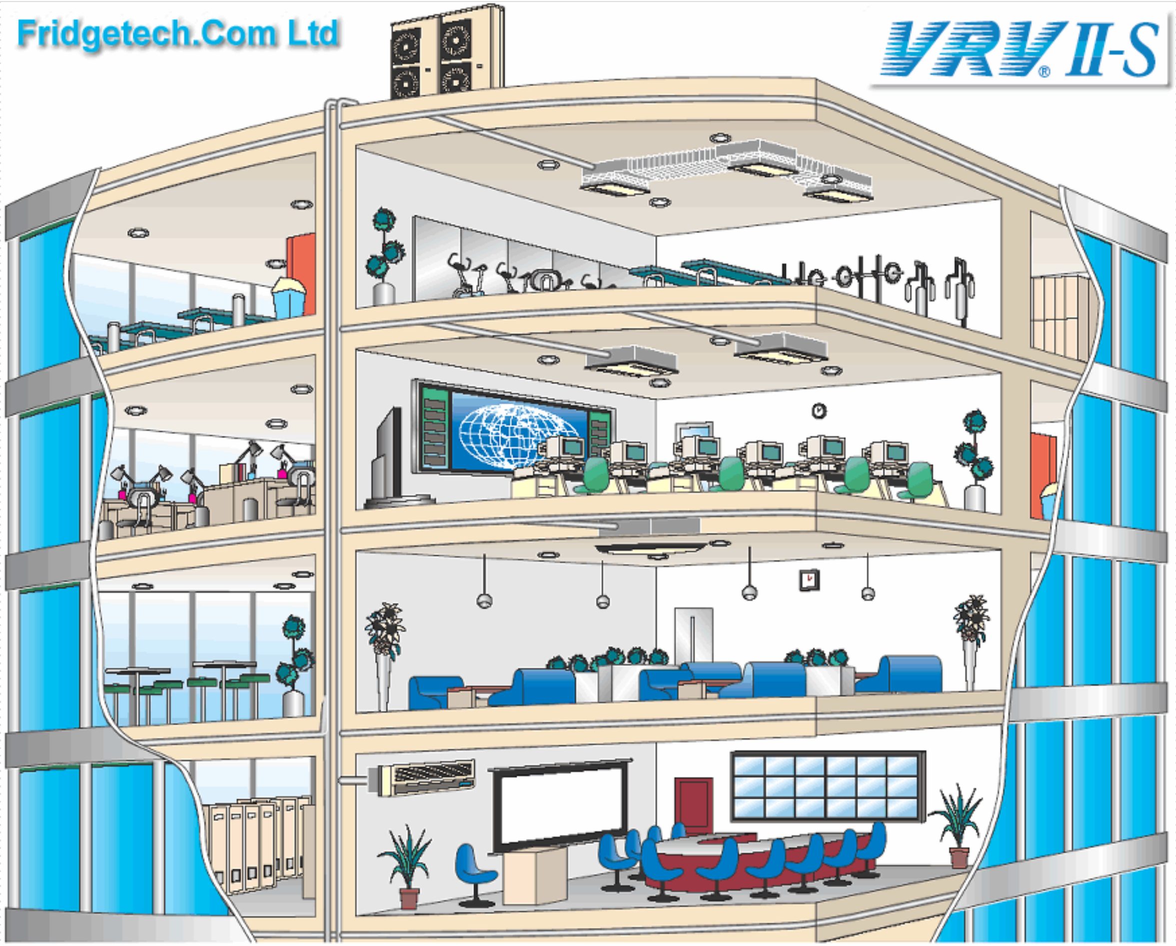
- 冷凍壓縮循環
  - cycle_download.swf
- 往復式壓縮機
  - reciprocating_download.swf
- 渦卷式壓縮機
  - scroll_download.swf

中央空調系統簡介

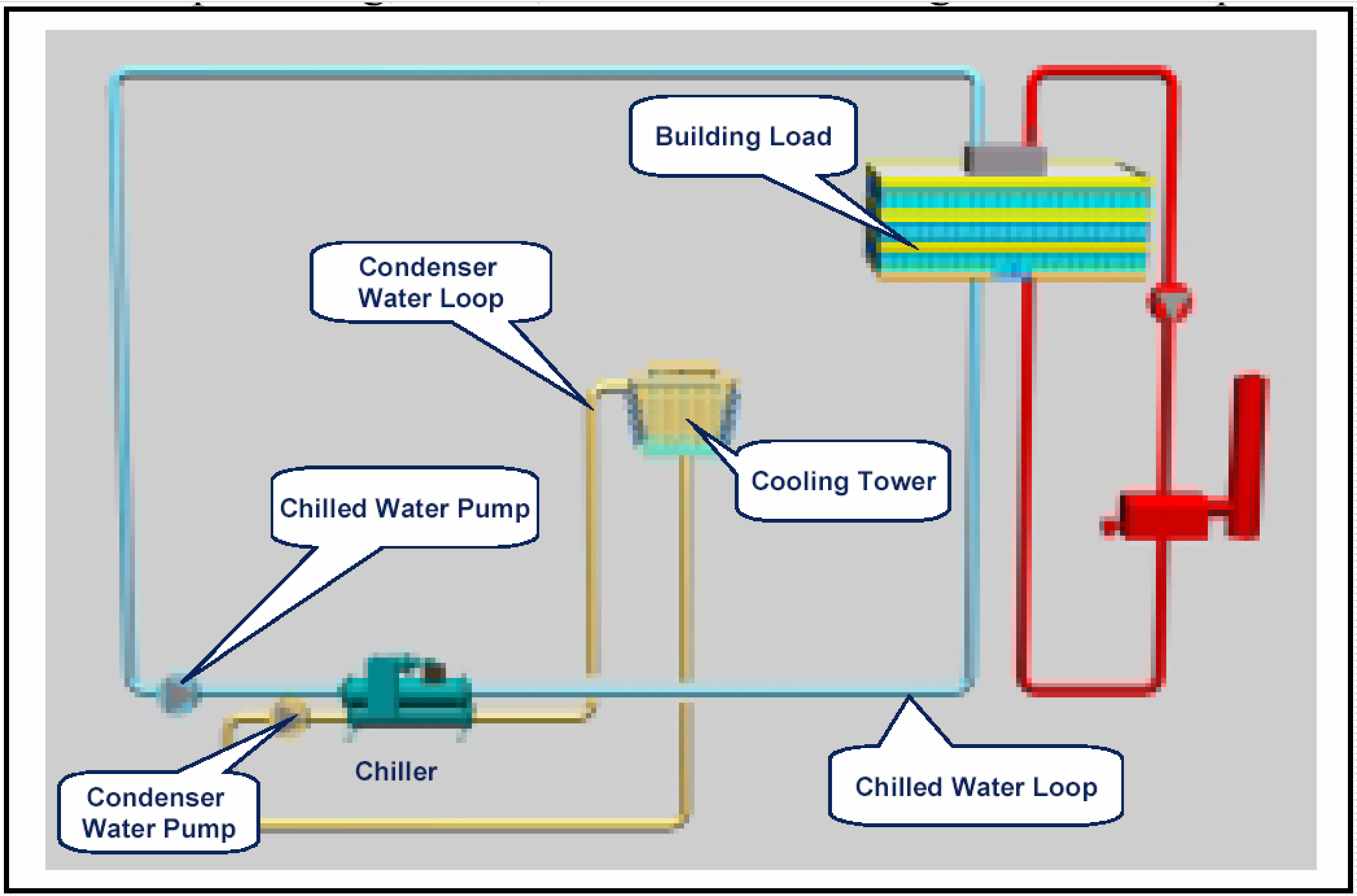
中央空調系統類型

- 冰水主機系統
  - FCU (Fan Coil Unit/小型送風機空調)
  - CAV (Constant Air Volume/定風量全氣式風管空調)
  - VAV (Variable Air Volume/變風量全氣式風管空調)
- 多聯變頻冷媒系統
  - VRF (Variable Refrigerant Flow)

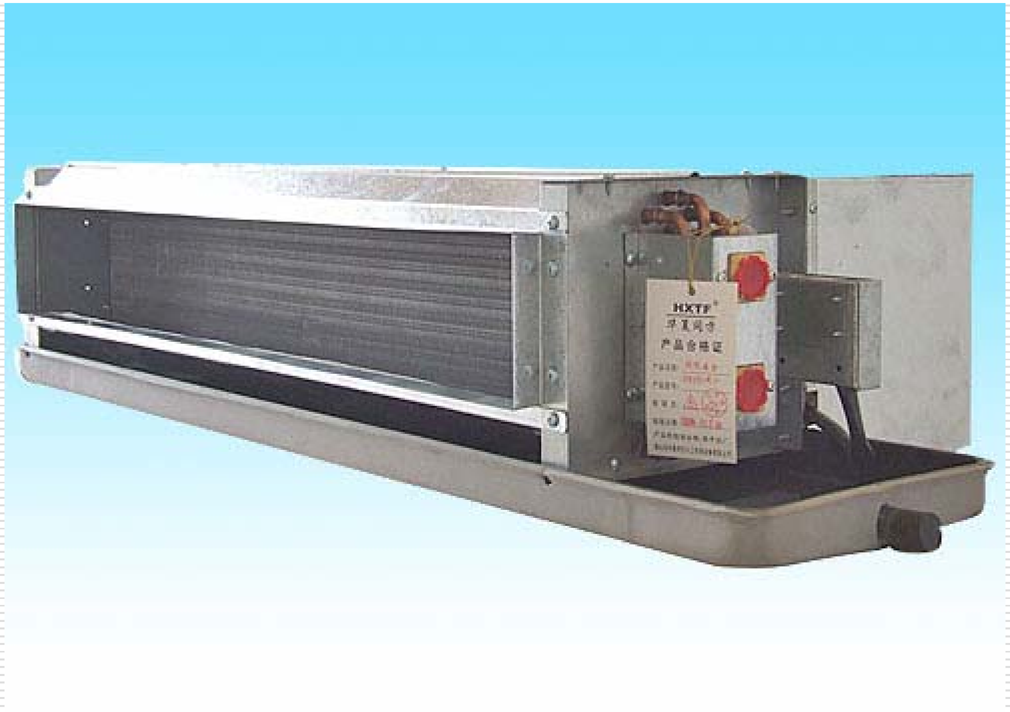
多聯變頻冷媒系統



中央冰水主機空調系統流程圖



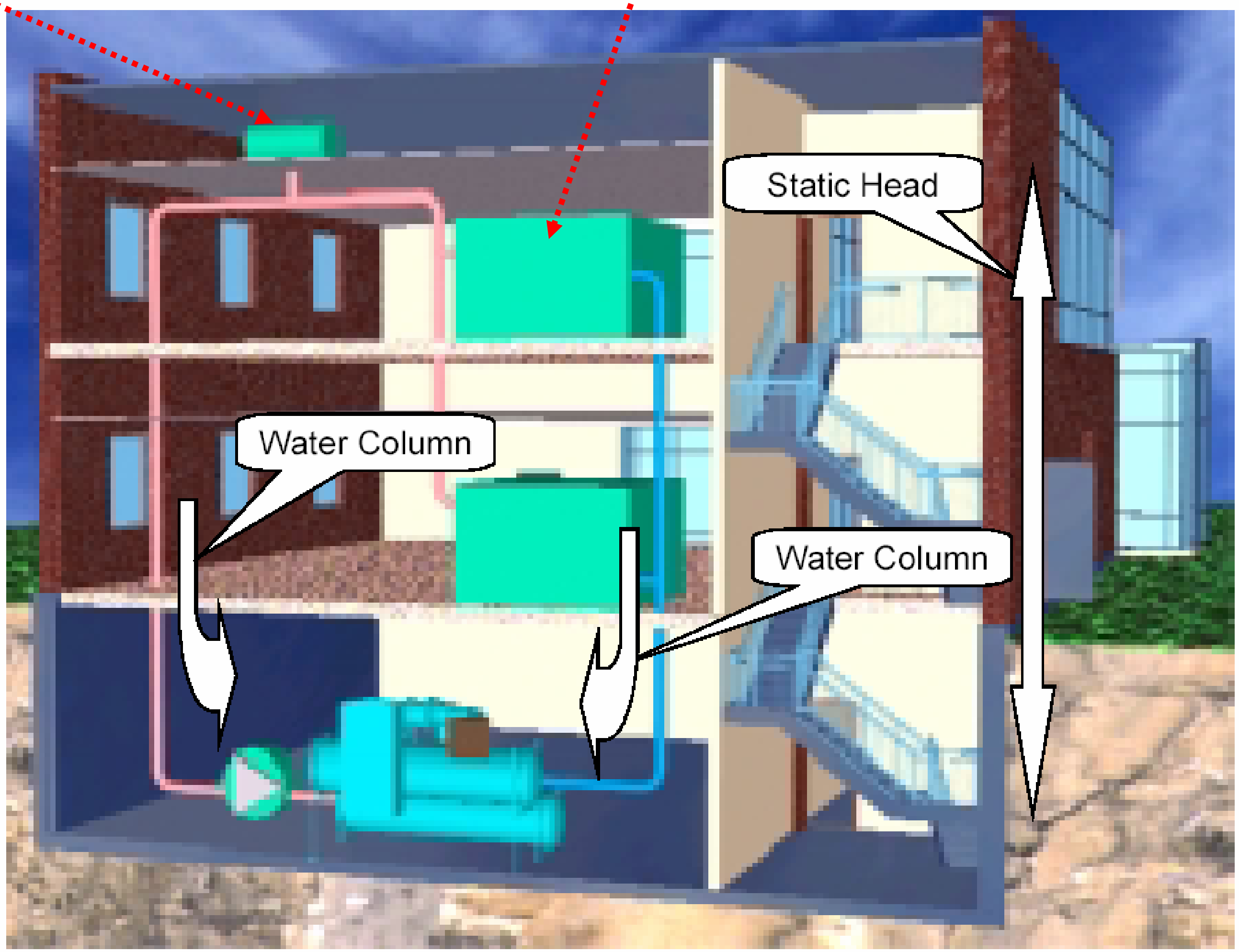
FCU (Fan Coil Unit/小型送風機空調)



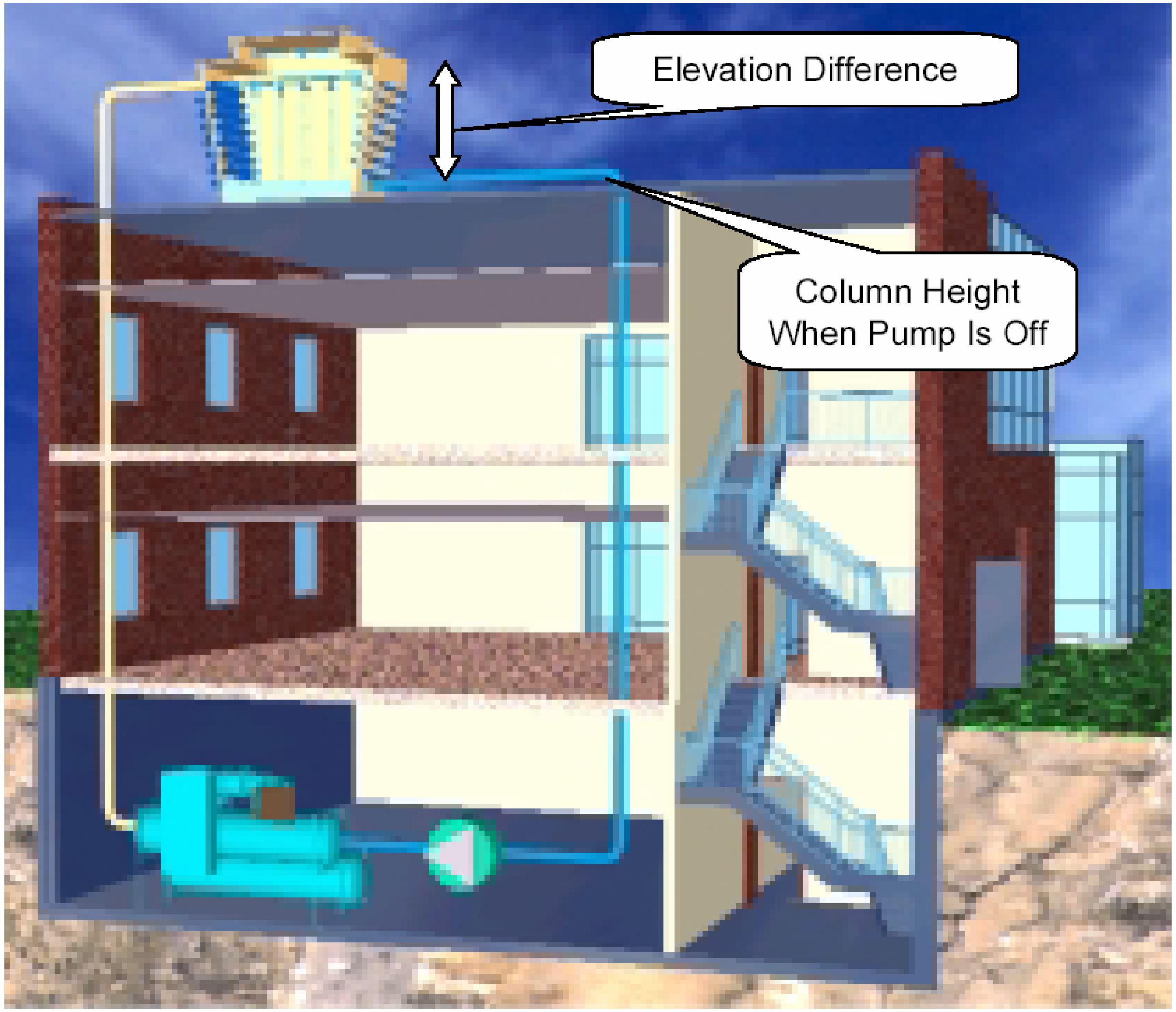
空調系統流程圖—冰水側

膨脹水箱(Expansion Tank)

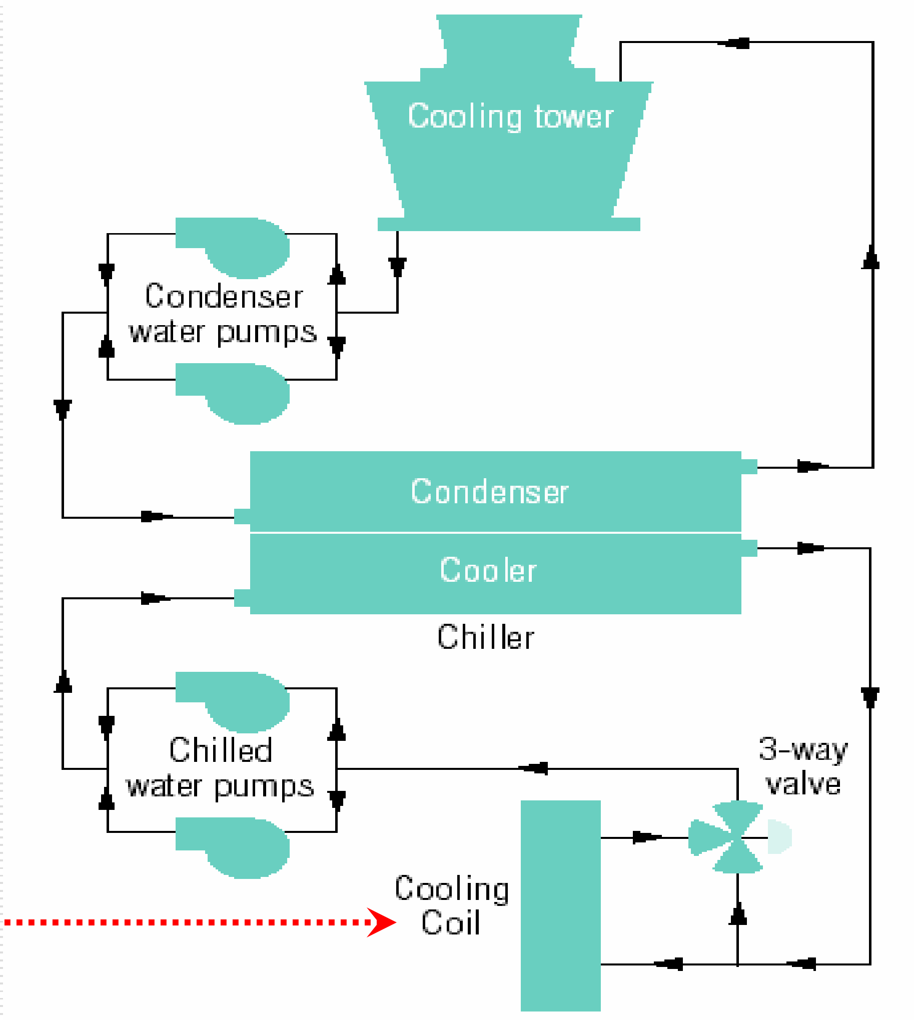
AHU(空調箱)
FCU(小型送風機)



空調系統流程—冷卻水側

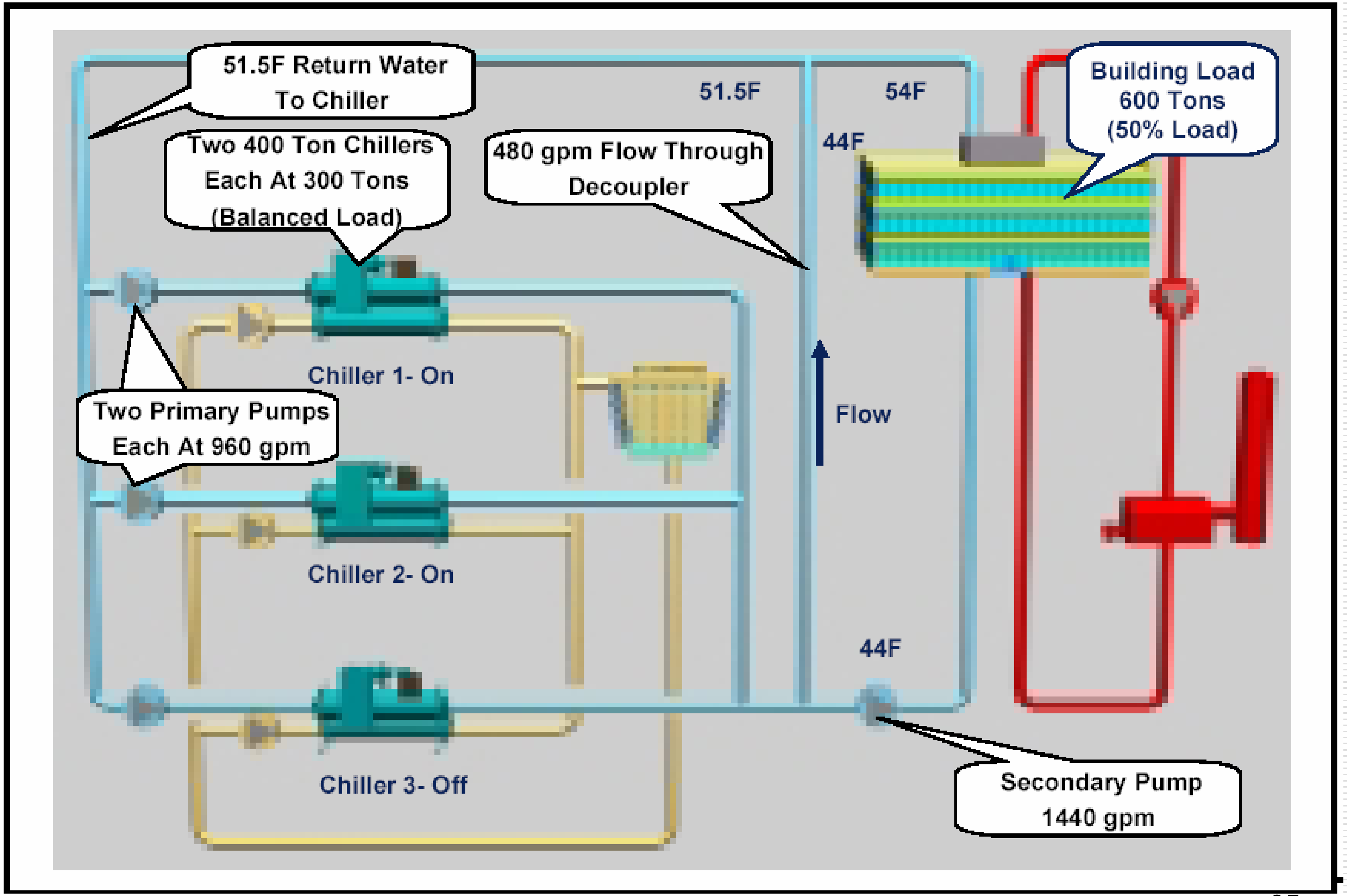


空調系統流程圖



AHU(空調箱)
FCU(小型送風機)

空調系統流程圖

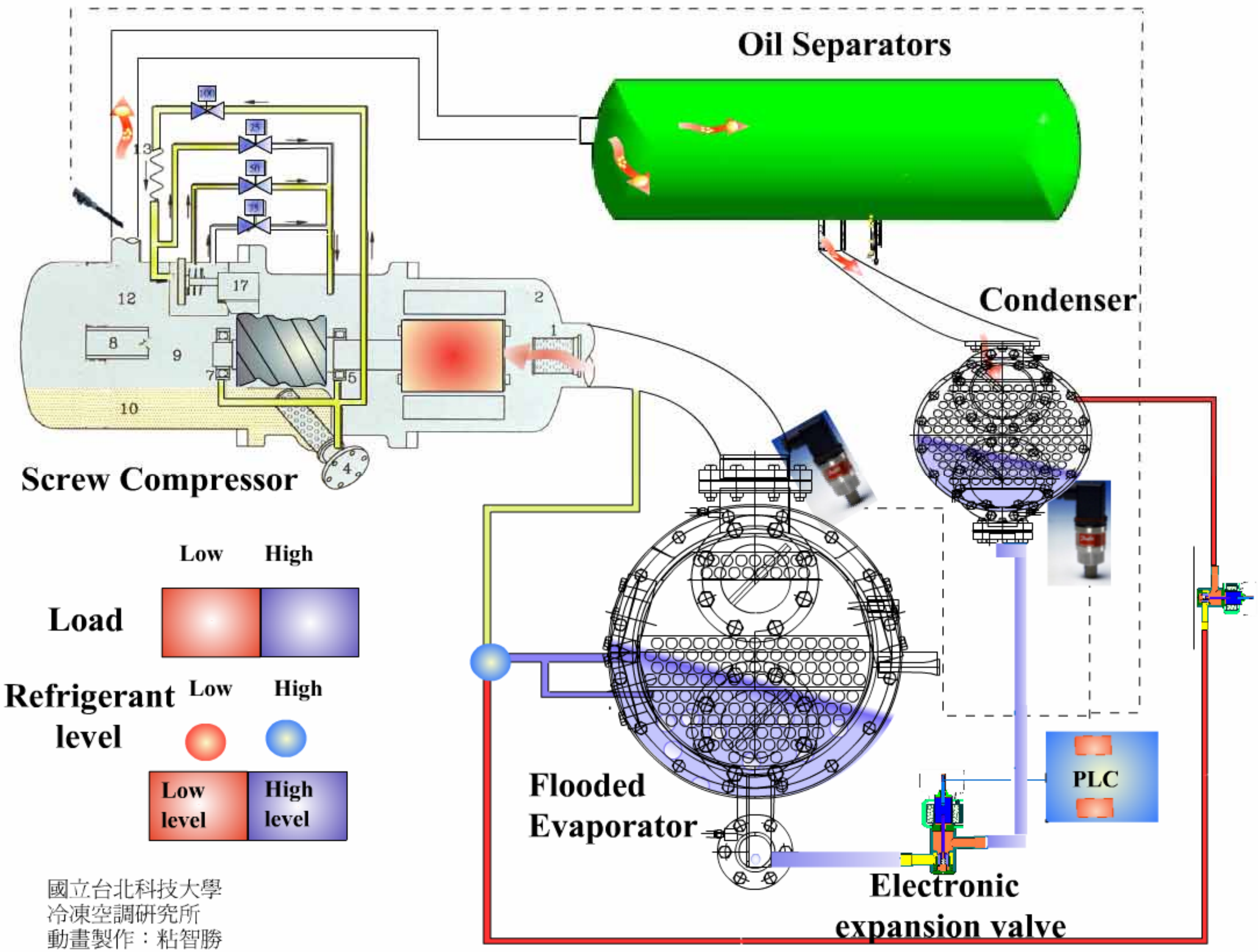


51.5F Return Water To Chiller
Two 400 Ton Chillers Each At 300 Tons (Balanced Load)
480 gpm Flow Through Decoupler
51.5F
54F
44F
Building Load 600 Tons (50% Load)
Chiller 1- On
Two Primary Pumps Each At 960 gpm
Flow
Chiller 2- On
44F
Chiller 3- Off
Secondary Pump 1440 gpm

Refrigeration Engineering

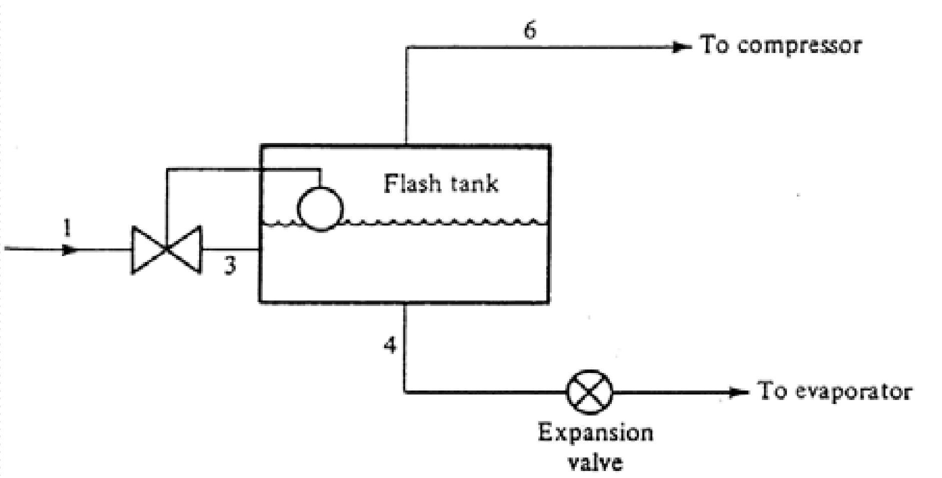
- Multi-Stage Compression

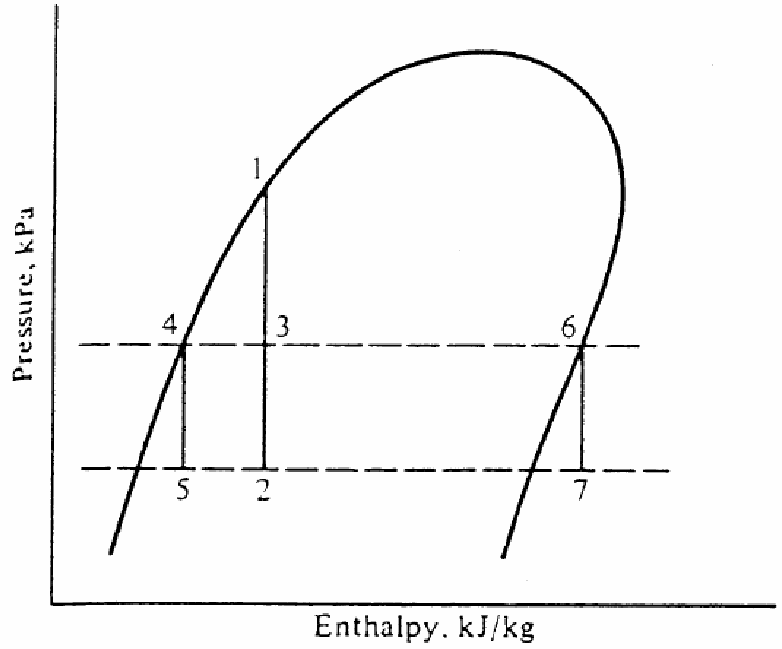
李魁鵬



國立台北科技大學
冷凍空調研究所
動畫製作：粘智勝
設計指導：李魁鵬

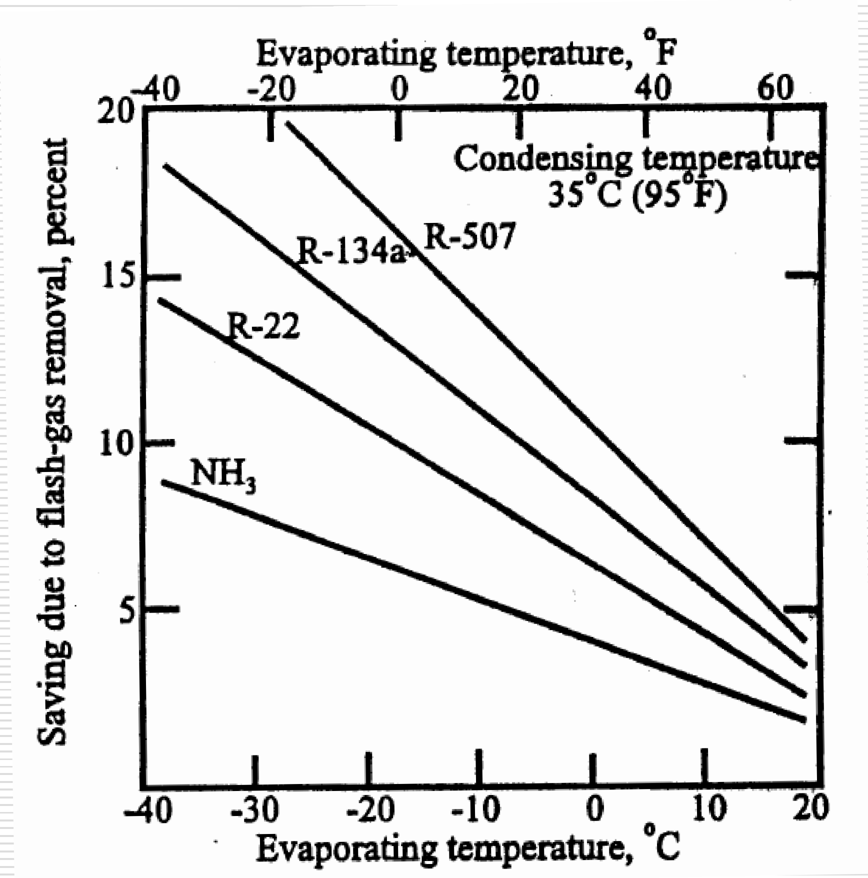
Two-Stage Expansion





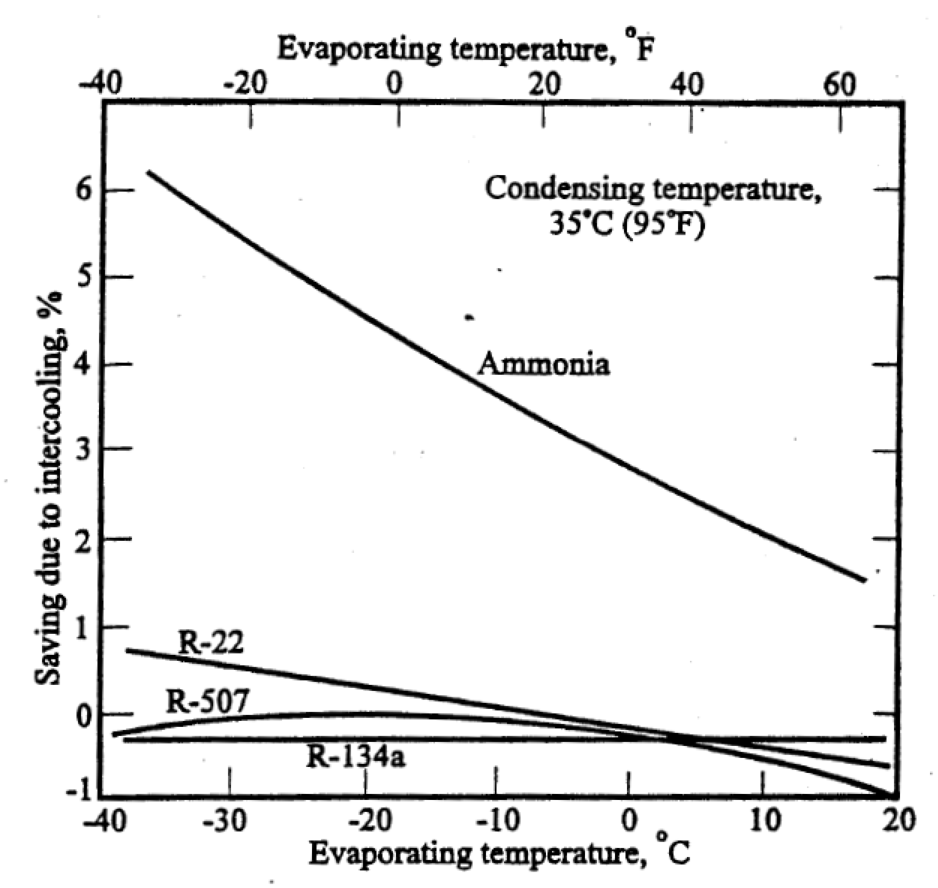
Expansion process showing replacement of process **3 -2** with the combination of 4-5 and 6-7.

Flash-gas removal



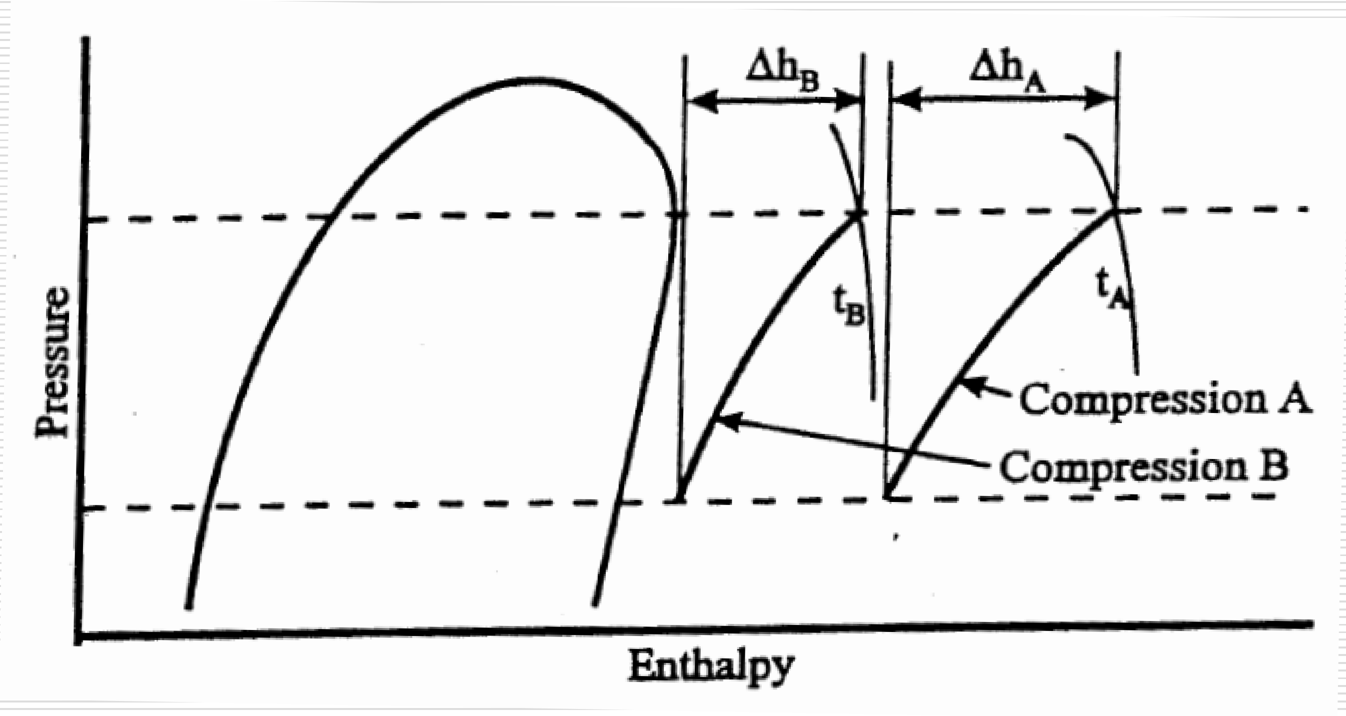
Percent saving in total compressor power resulting from flash-gas removal at the optimum intermediate temperature.

Percent saving



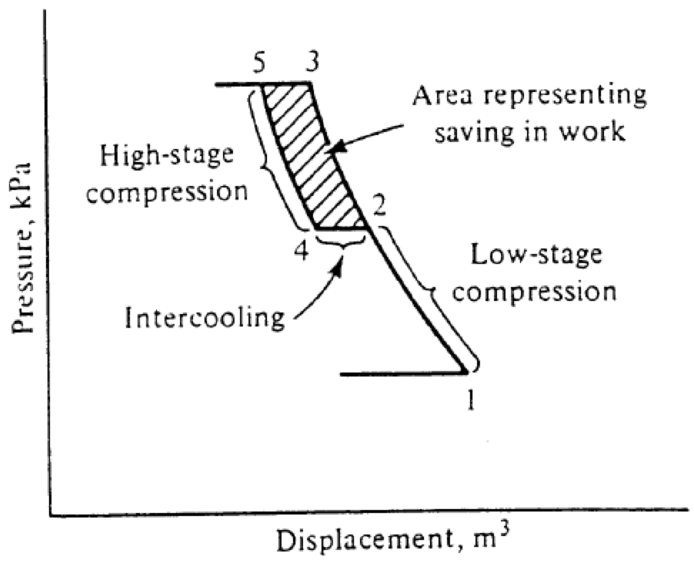
Percent saving in total compressor power resulting from intercooling at the optimum intermediate temperature.

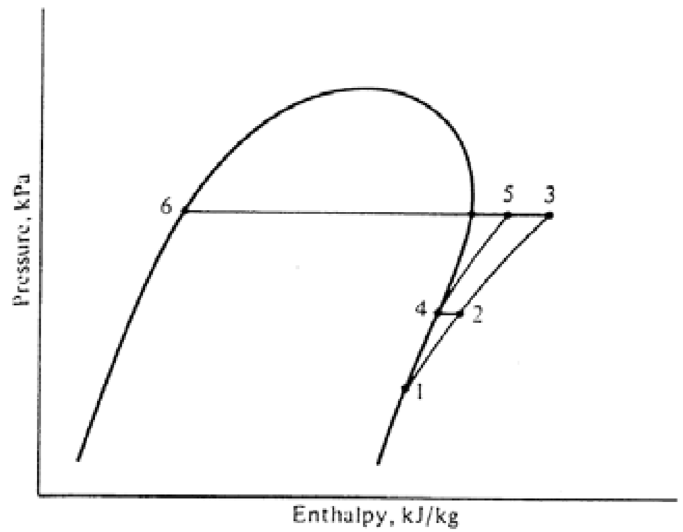
Compression between two given pressures



Comparison of compression between two given pressures with differing initial temperature

Intercooling in two-stage compression





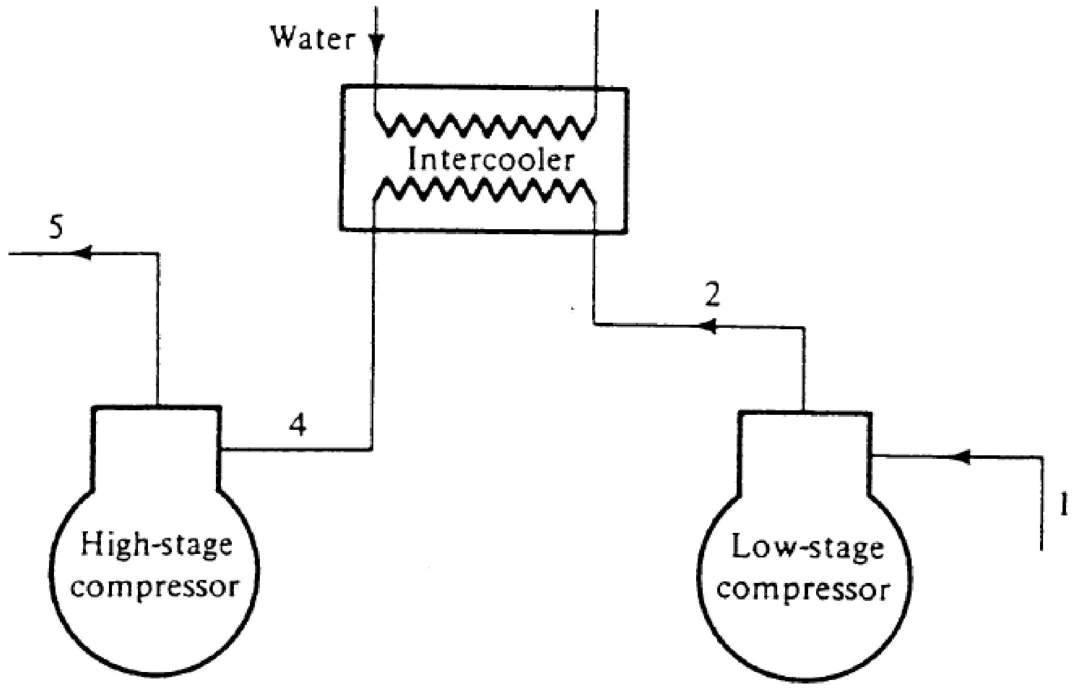
$$W = -\int v\, dp = \frac{n}{n-1} p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{(n-1)/n} \right]$$

where p = pressure, Pa

v = specific volume, m^3/kg

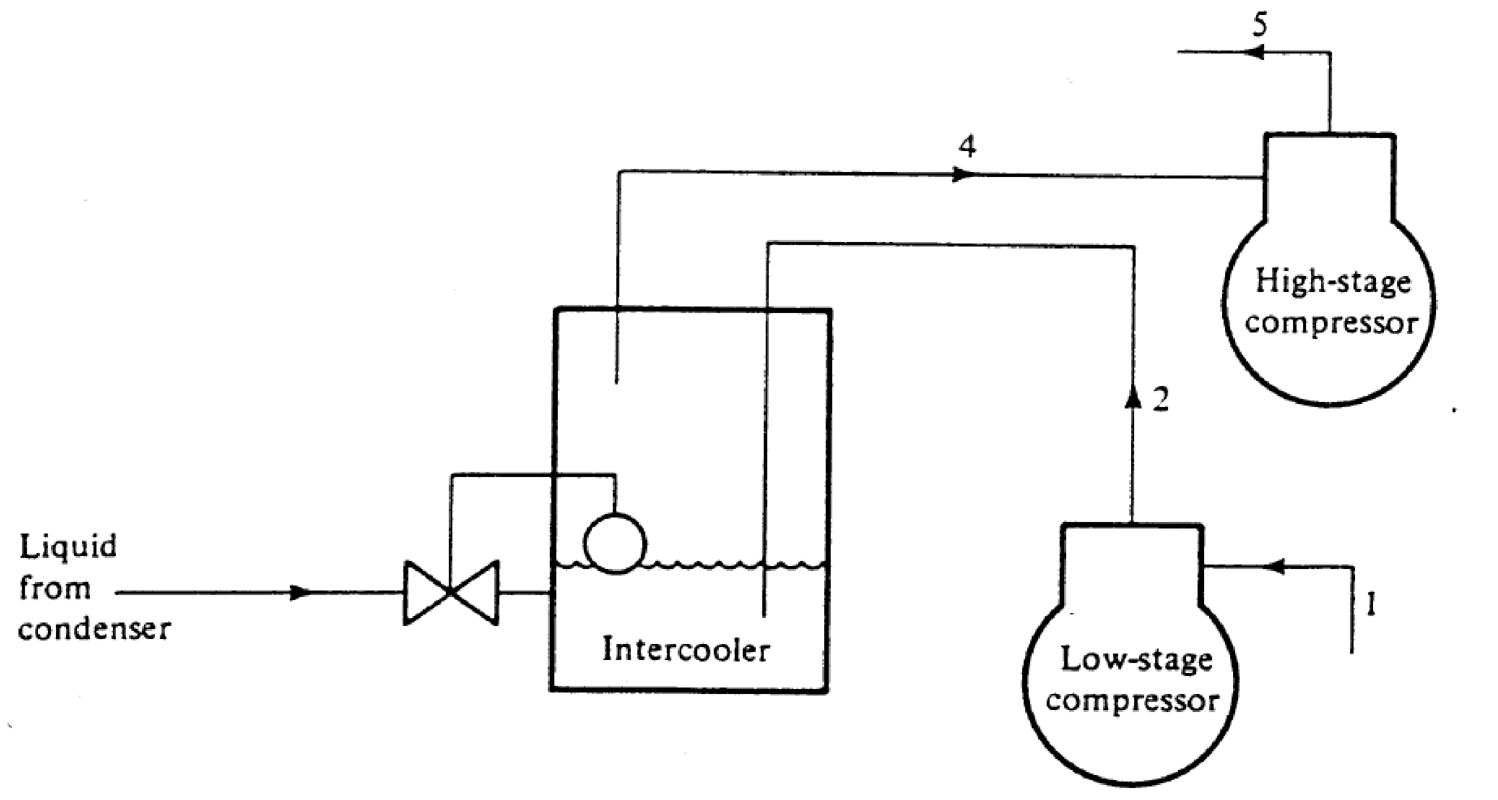
n = polytropic exponent relating the pressure and specific volume during compression, pv^n = const

Intercooling with a water cooled heat exchanger



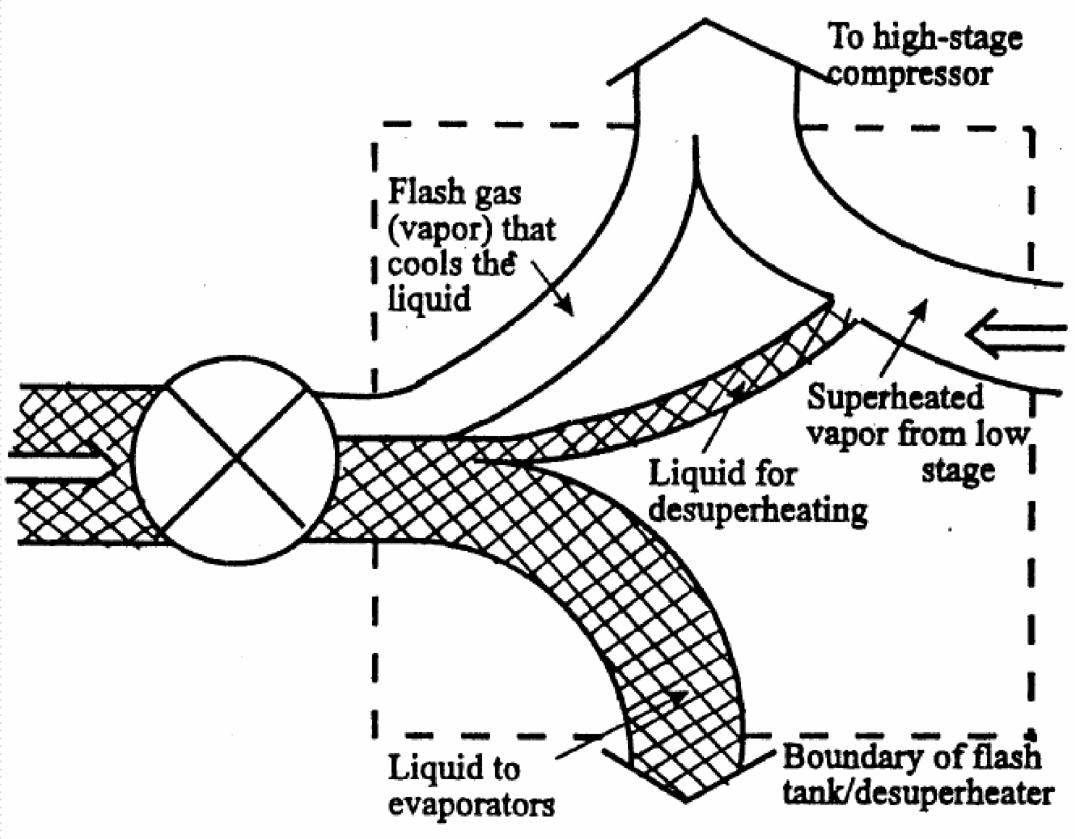
(a)

Intercooling with liquid refrigerant



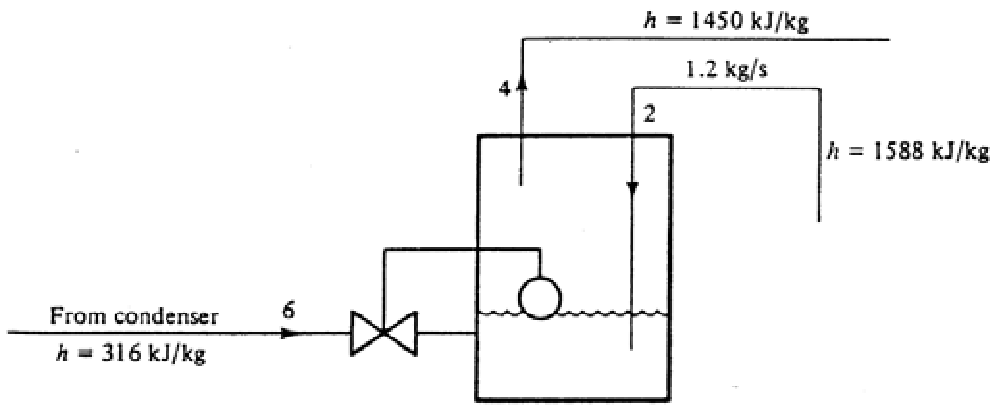
Intercooling **with *(a)*** a watercooled heat exchanger, **and *(b)* liquid** refrigerant.

Refrigerant streams in a flash-tank/desuperheater



Example

Example 16-1 Calculate the power needed to compress 1.2 kg/s of ammonia from saturated vapor at 80 kPa to 1000 kPa (*a*) by single-stage compression and (*b*) by two-stage compression with intercooling by liquid refrigerant at 300 kPa.



Heat balance:

$$w_6(316\ \text{kJ/kg}) + (1.2\ \text{kg/s})(1588\ \text{kJ/kg}) = w_4(1450\ \text{kJ/kg})$$

Mass balance:

$$w_6 + 1.2 = w_4$$

Solving gives

$$w_4 = 1.346\ \text{kg/s}$$

Intercooling the ammonia with liquid refrigerant reduced the power requirement from 468 to 453.2 kW.

Example

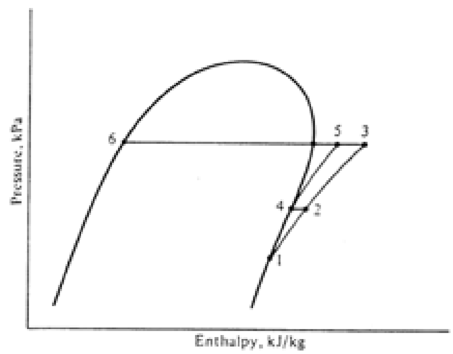


Table 16-1 Comparison of ammonia compression with and without intercooling

	Without intercooling, processes 1–2 and 2–3	With intercooling, processes 1–2, 2–4, and 4–5
$h_2 - h_1$, kJ/kg	1588 – 1410	1588 – 1410
$h_3 - h_2$, kJ/kg	1800 – 1588	
$h_5 - h_4$, kJ/kg	. . . .	1628 – 1450
Flow rate, kg/s, 1 to 2	1.2	1.2
2 to 3	1.2	
4 to 5	. . . .	1.346
Power required, kW, 1 to 2	213.6	213.6
2 to 3	254.4	
4 to 5	. . . .	239.6
Total power, kW	468.0	453.2

Ratio of required capacities

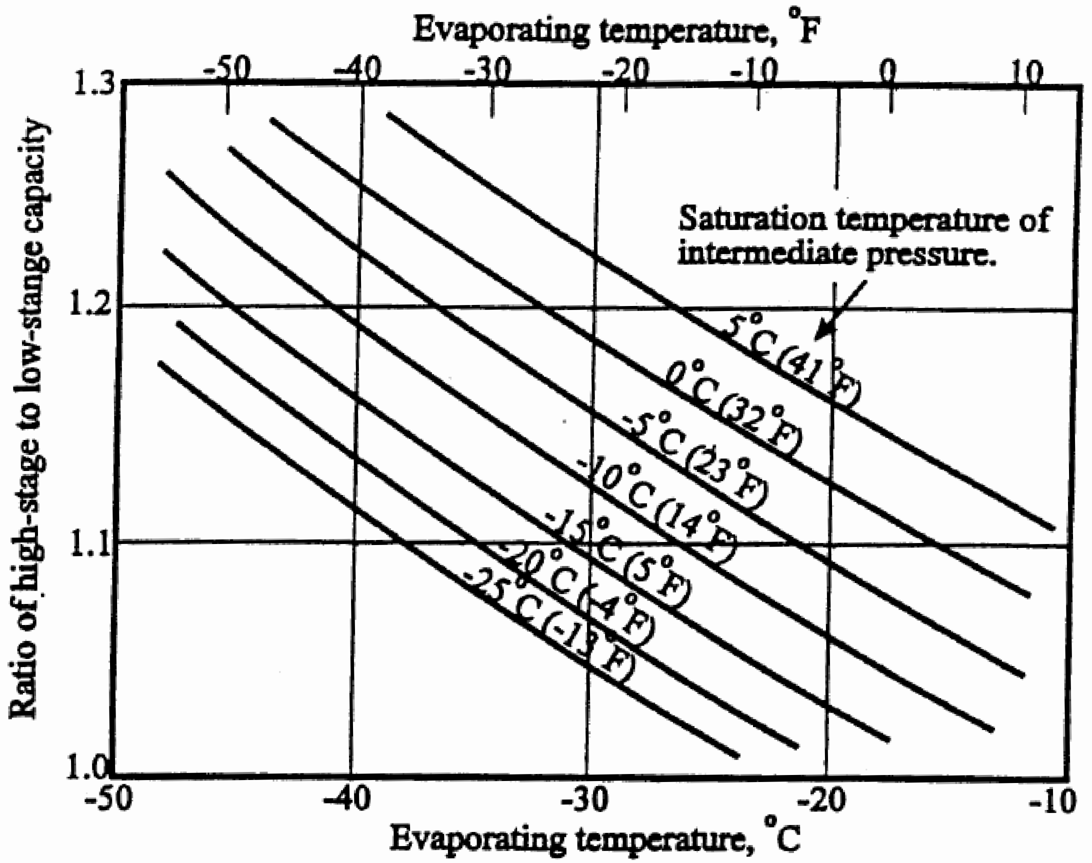


FIGURE 3.20
Ratio of required capacities of the high-stage to the low-stage compressor in a single-evaporator ammonia two-stage system. (*Courtesy of Vilter Manufacturing Corporation*)

Percent savings in power of two-stage systems

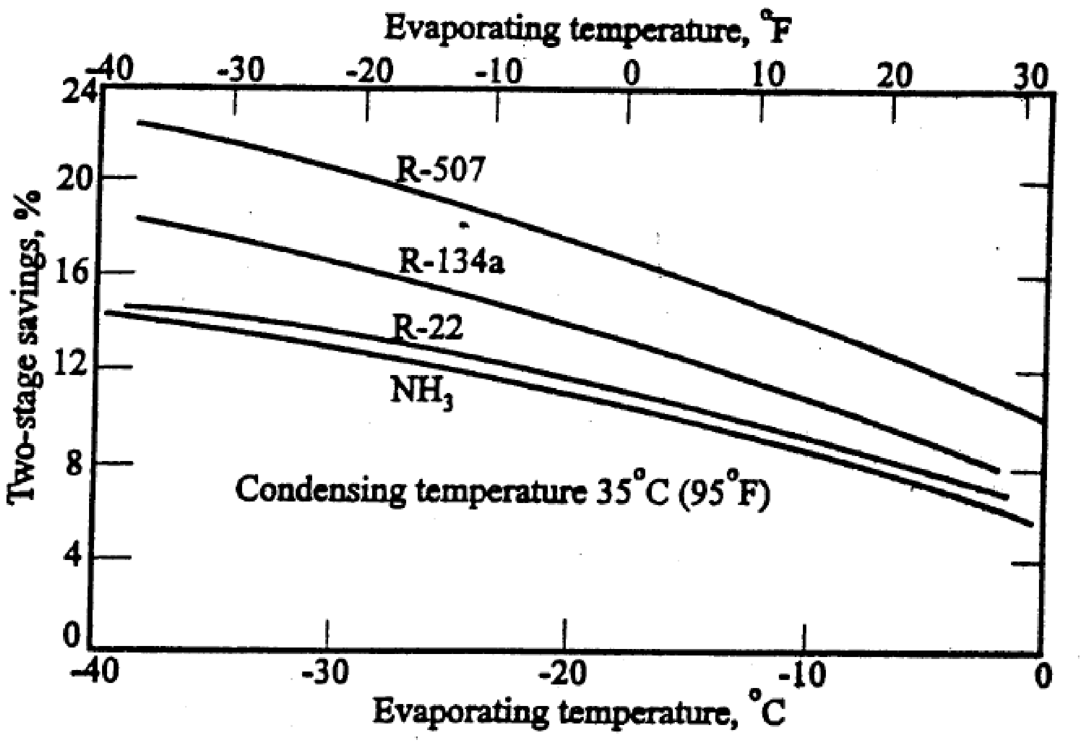


FIGURE 3.25
Percent savings in power of two-stage systems employing flash-gas removal and desuperheating in comparison to single-stage operation.

Liquid Subcooler

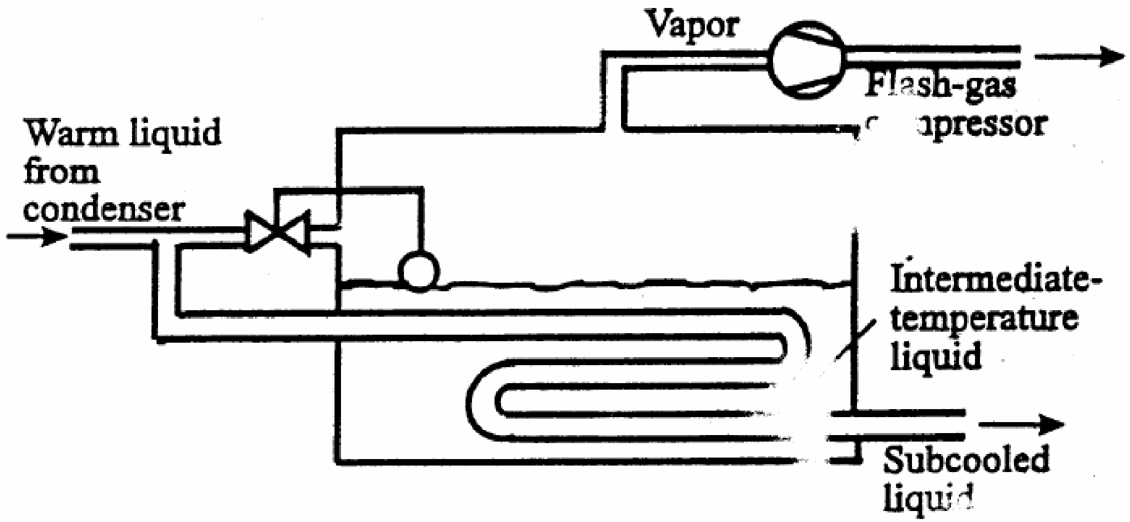


FIGURE 3.5
A liquid subcooler using a coil immersed in the liquid of an intermediate-pressure vessel.

U-values of immersed coil subcoolers, W/m²·K (Btu/hr·ft²·°F)

Refrigerant	Tube-side velocity, m/s (ft/s)	
	0.75 (2.46)	2.0 (6.56)
R-22	75 (13)	125 (22)
Ammonia	180 (32)	275 (49)

Liquid Subcooler

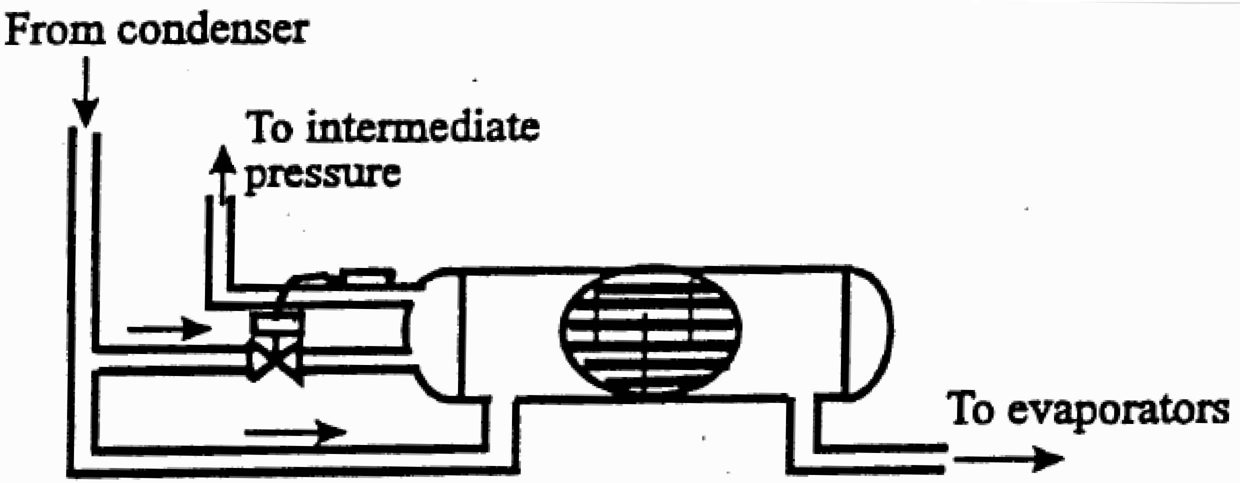


FIGURE 3.6
Liquid subcooling with an external shell-and-tube heat exchanger with boiling refrigerant controlled by an expansion valve.

Liquid Subcooler

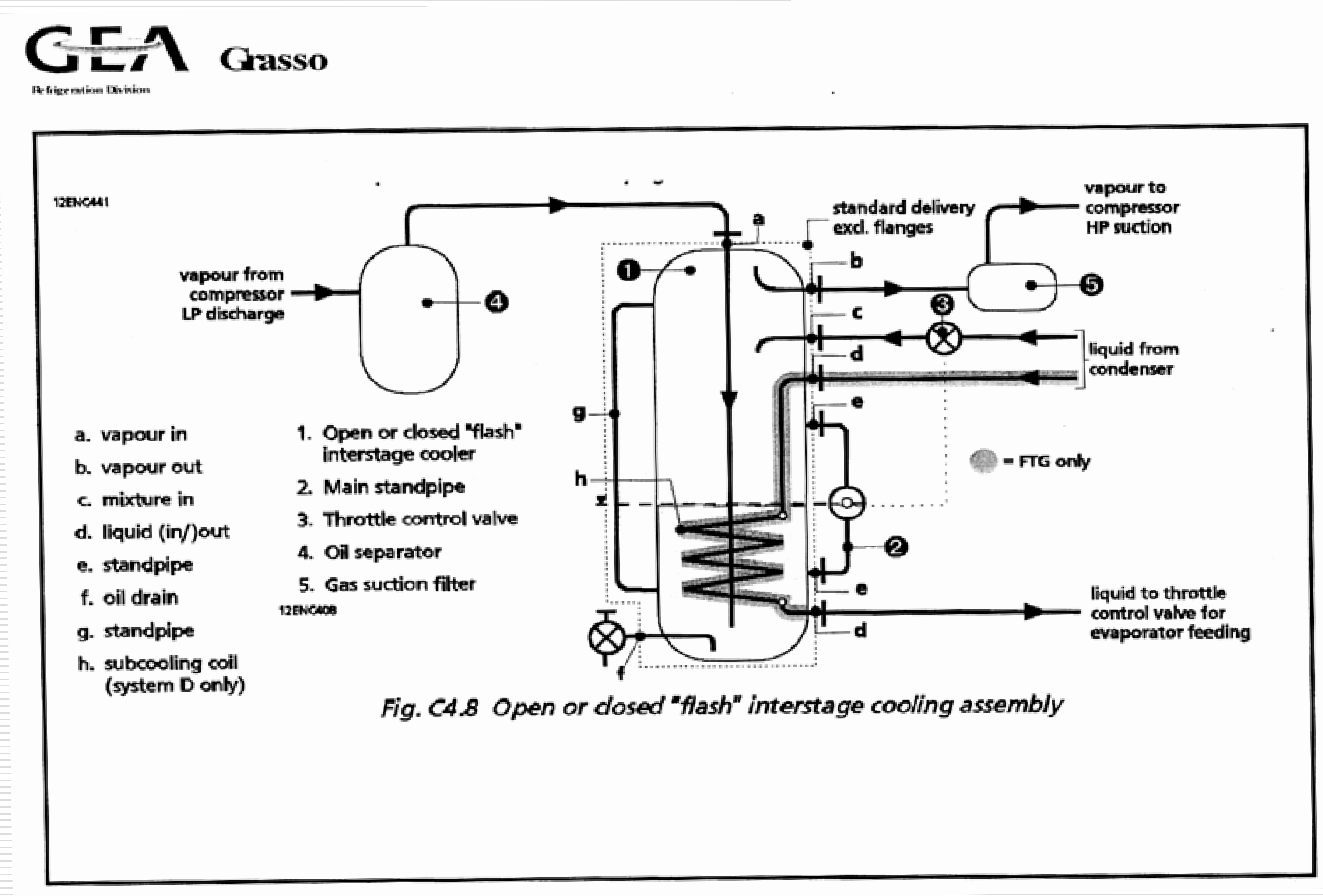


Fig. C4.8 Open or closed "flash" interstage cooling assembly

Liquid Subcooler

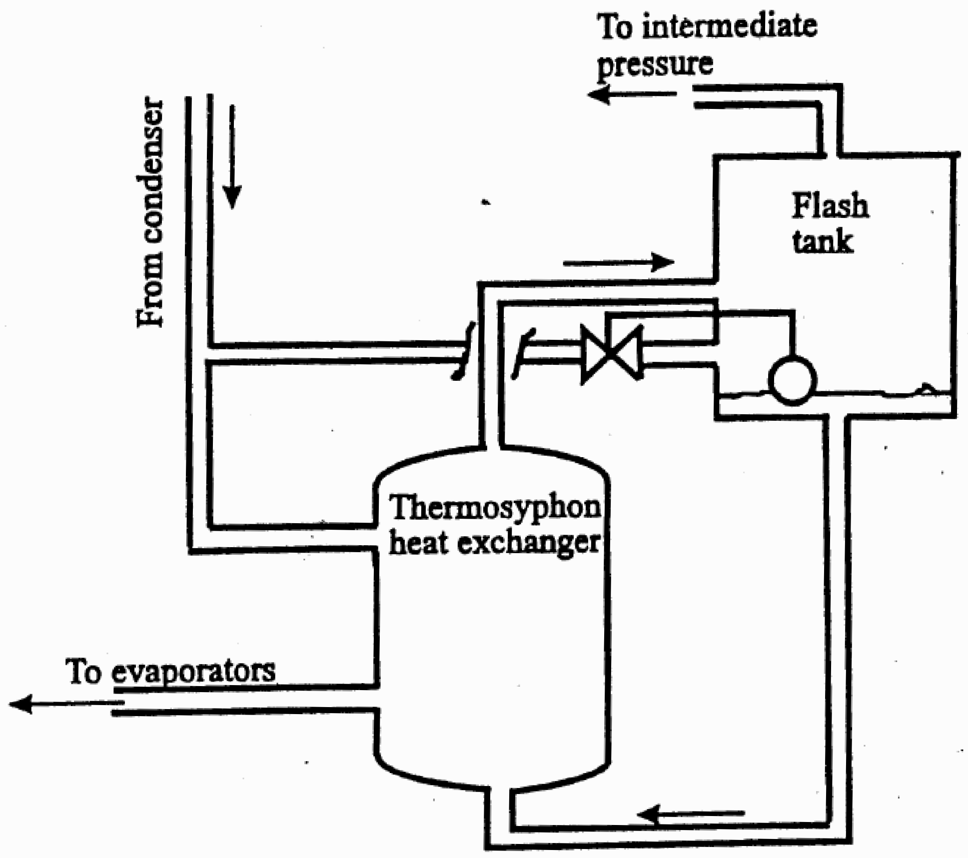


FIGURE 3.7
Liquid subcooling with an external heat exchanger of the thermosyphon type.

Oil injection intercooling

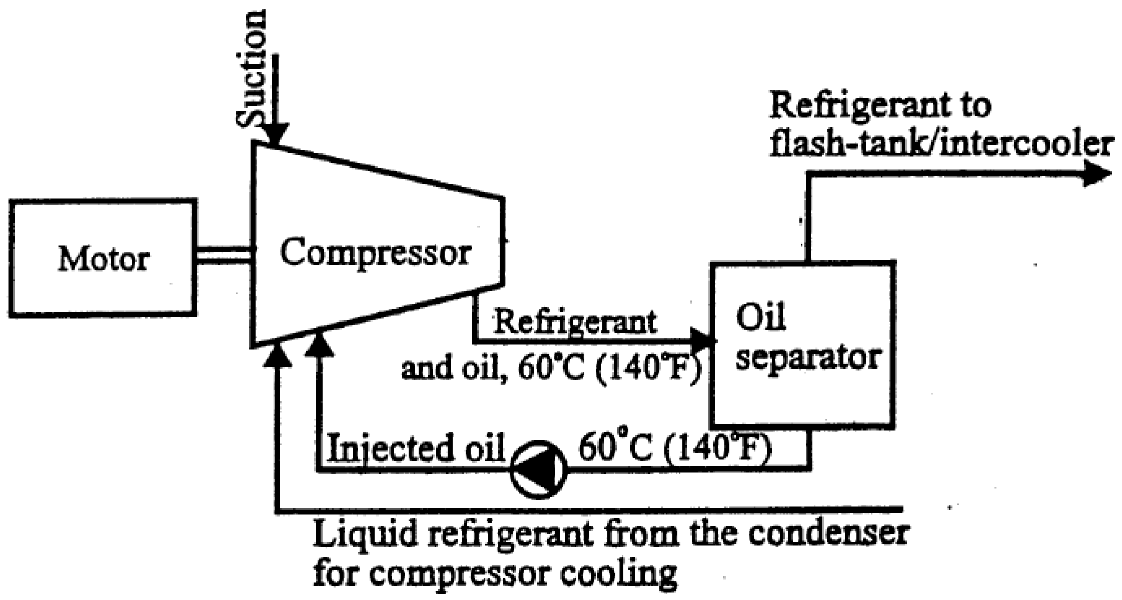


FIGURE 3.21
Cooling the oil injected into a low-stage screw compressor by direct admission of liquid refrigerant.

Oil injection intercooling

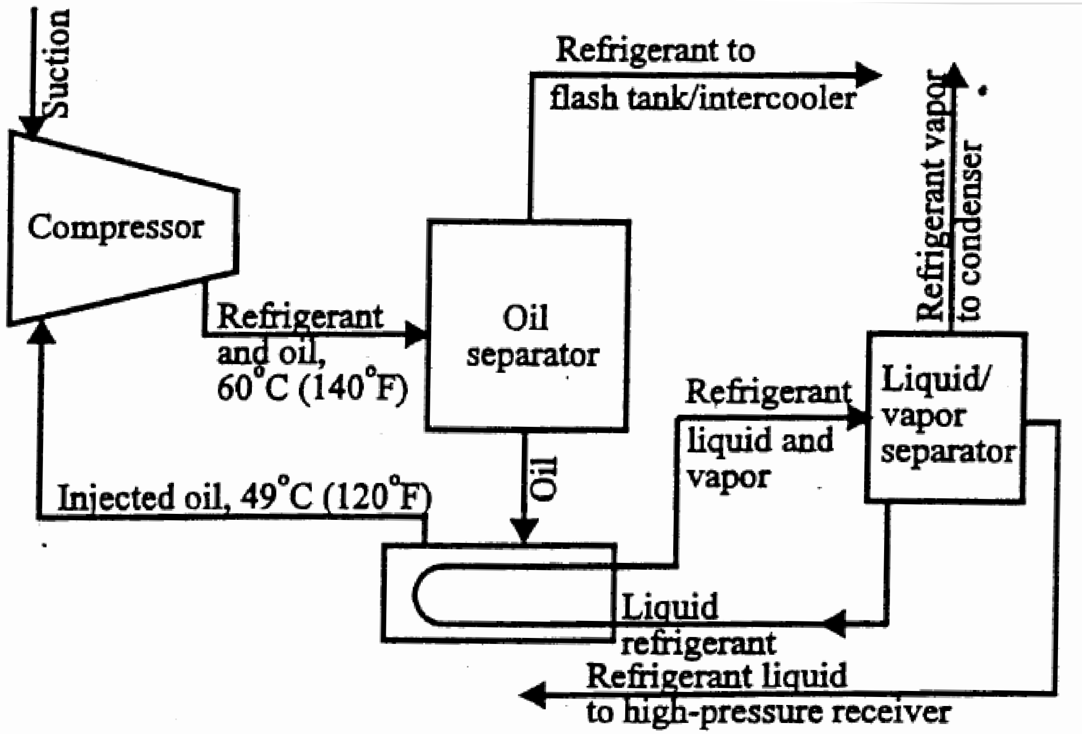


FIGURE 3.22
Cooling the oil injected into a low-stage screw compressor by an external thermosiphon heat exchanger which in turn is cooled by boiling refrigerant.

Partial intercooling

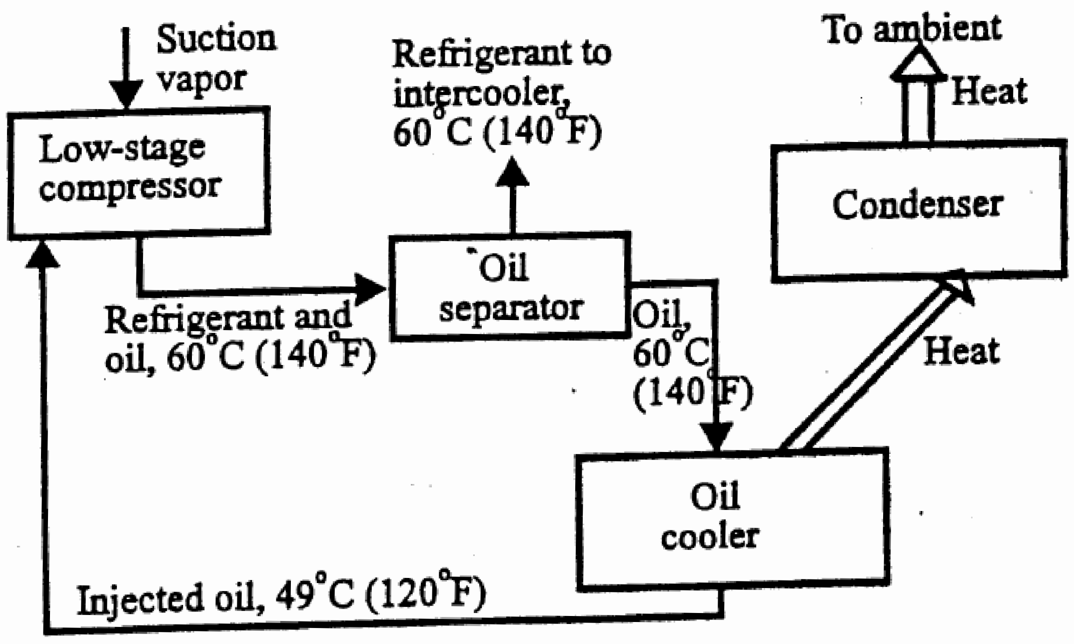


FIGURE 3.13
A partial intercooling provided by oil cooling in a low-stage screw compressor.

Economizer

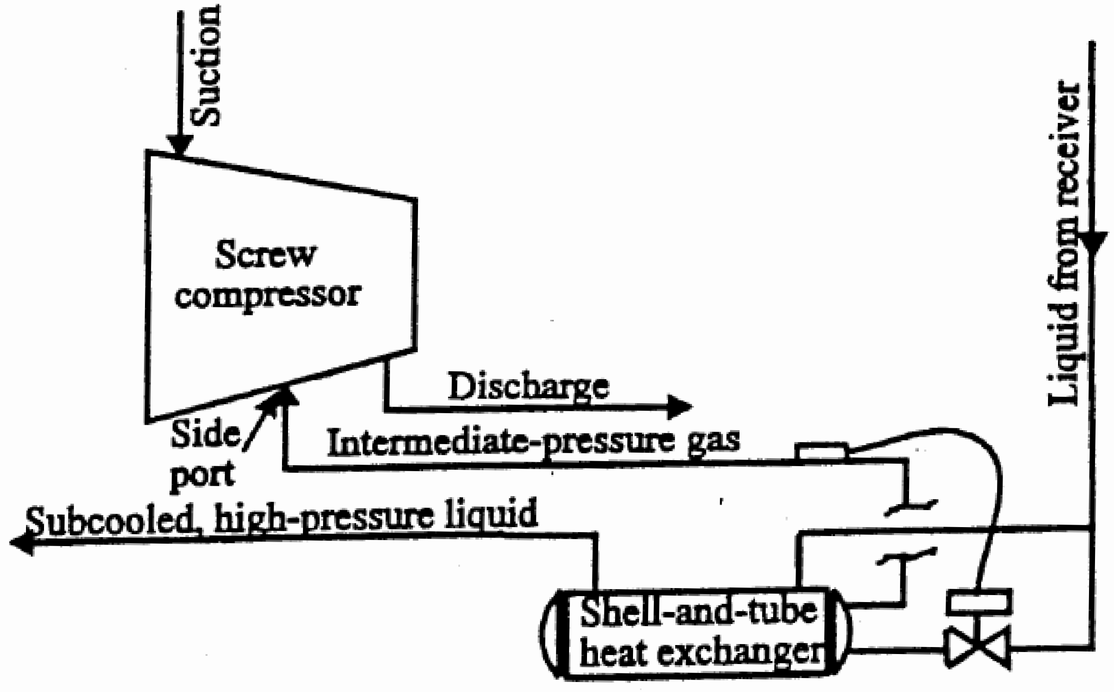


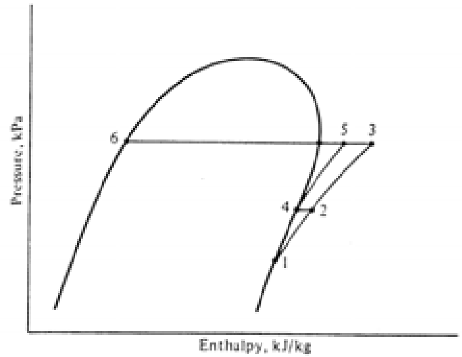
FIGURE 3.24
Using the side port of a screw compressor as an economizer to subcool liquid.

Example

Example 16-2 Compare a compression of 3.5 kg/s of refrigerant 22 from saturated vapor at 100 kPa to a condensing pressure of 1000 kPa (*a*) by single-stage compression and (*b*) by two-stage compression with intercooling at 300 kPa, using liquid refrigerant.

Table 16-2 Comparison of refrigerant 22 compression with and without intercooling

	Without intercooling, processes 1-2 and 2-3	With intercooling, processes 1-2, 2-4, and 4-5
$h_2 - h_1$, kJ/kg	416 - 387	416 - 387
$h_3 - h_2$, kJ/kg	449 - 416	
$h_5 - h_4$, kJ/kg	....	430 - 399
Flow rate, kg/s, 1 to 2	3.5	3.5
2 to 3	3.5	
4 to 5	....	3.74
Power required, kW, 1 to 2	101.5	101.5
2 to 3	115.5	
4 to 5	....	115.9
Total power, kW	217.0	217.4



Optimum intercooler pressure

There is an optimum pressure at which the intercooling should take place in an ammonia system. In the compression of air, where the intercooling is achieved by rejecting heat to the ambient or to cooling water, that intermediate pressure for minimum total power is

$$p_i = \sqrt{p_s p_d} \tag{16-1}$$

where p_i = intercooler pressure, kPa
p_s = suction pressure of low-stage compressor, kPa
p_d = discharge pressure of high-stage compressor, kPa

The development of the equation does not consider the additional refrigerant compressed by the high-stage compressor, but it does provide an approximate guideline for the optimal intermediate pressure.

Optimum intercooler pressure

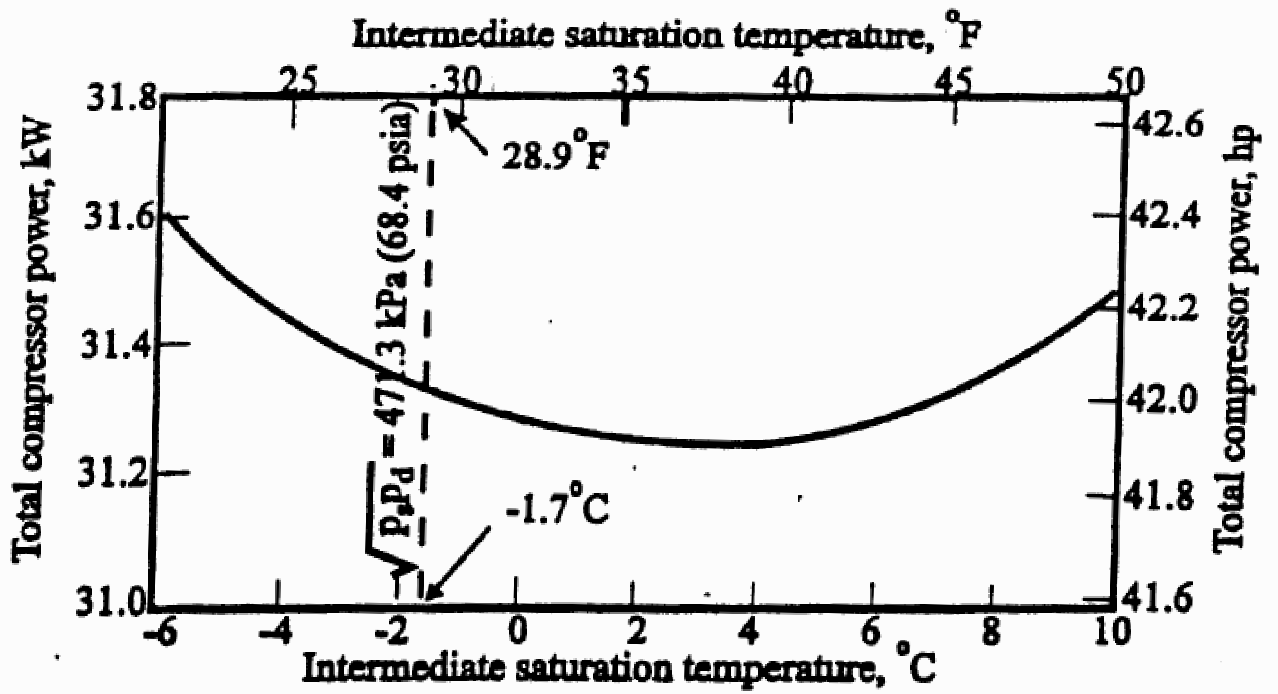


FIGURE 3.18
Total power required in a two-stage R-22 system with a refrigerating capacity of 100 kW (28.4 tons of refrigeration) as a function of the saturated intermediate temperature. The evaporating temperature is −30°C (−22°F), and the condensing temperature is 35°C (95°F).

Dynamic Response

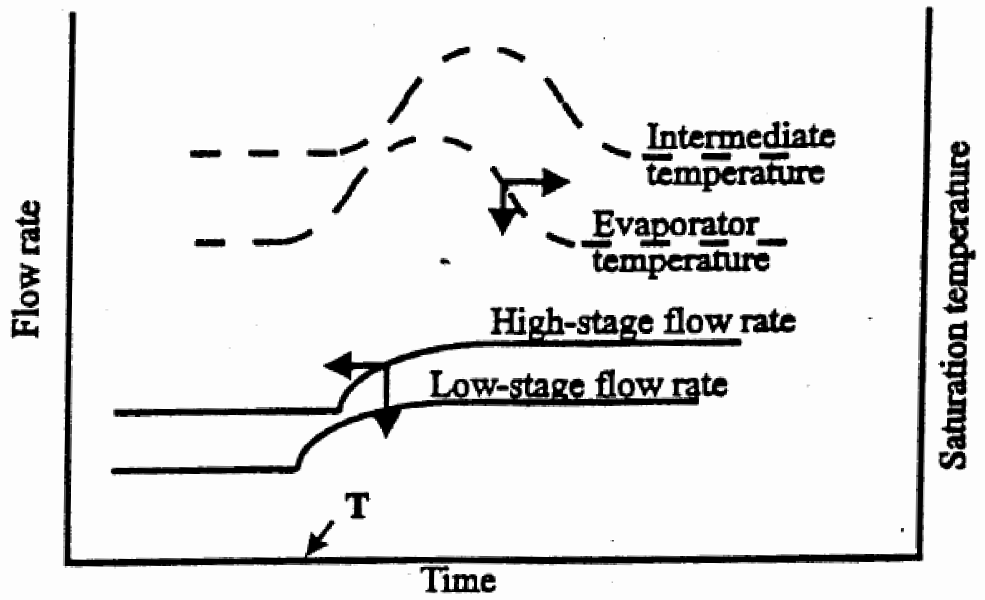
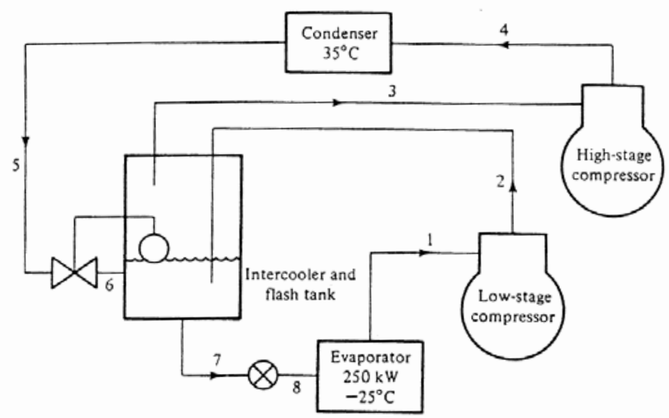
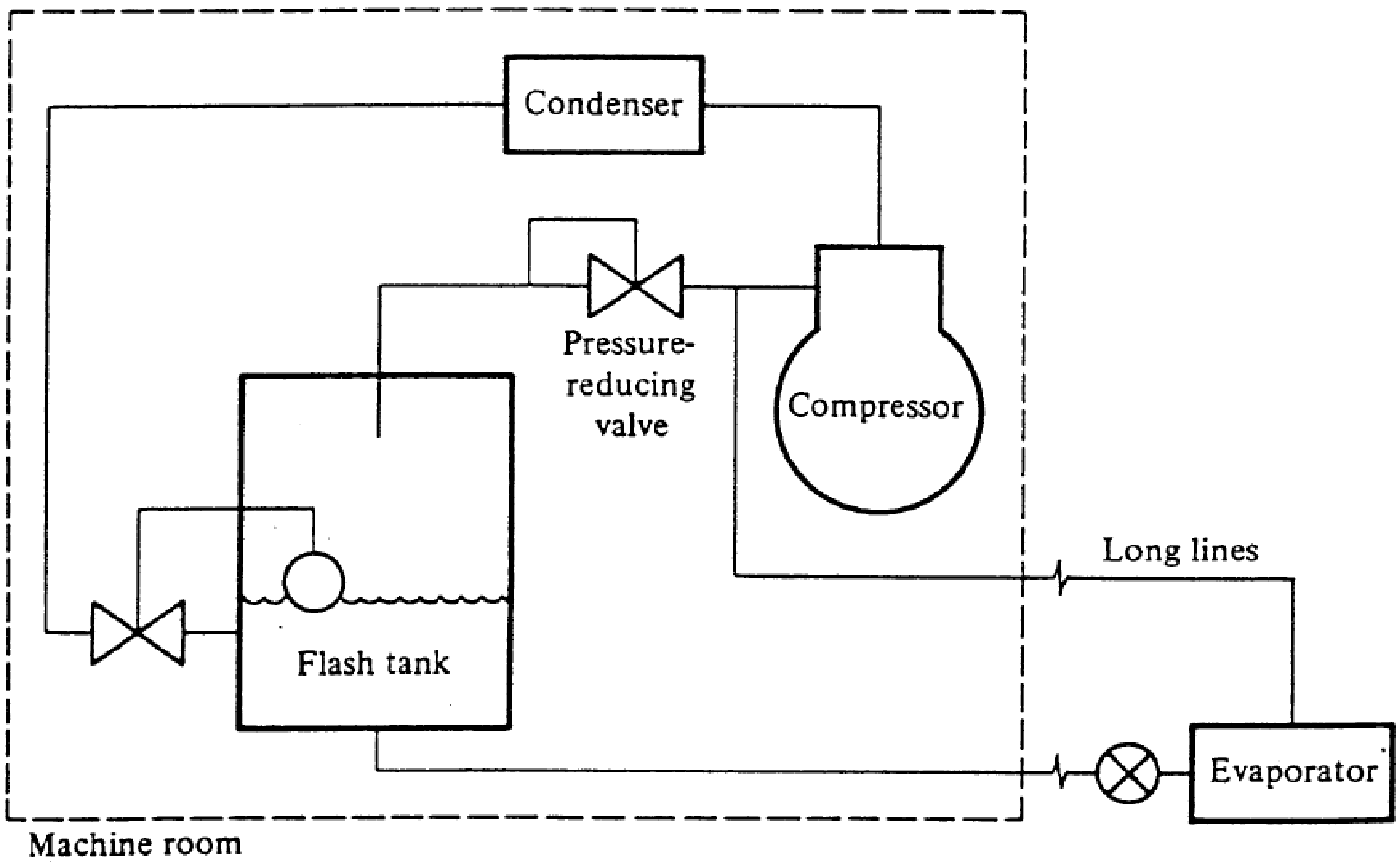


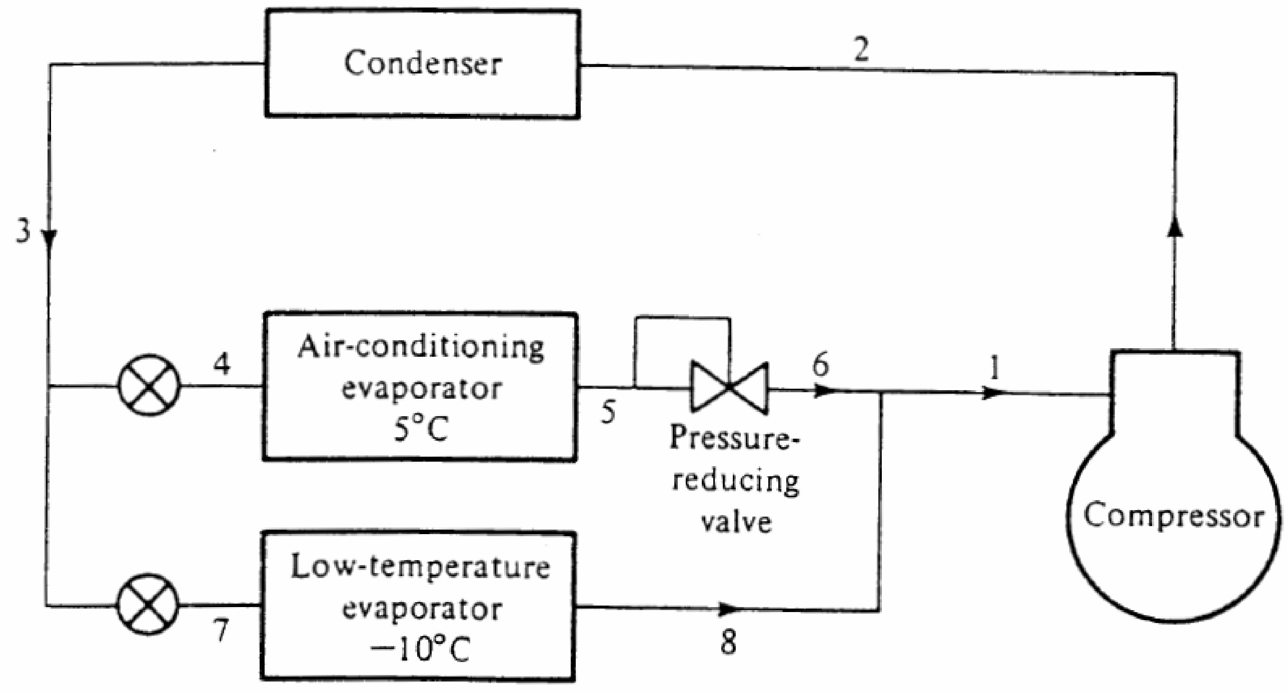
FIGURE 3.23
Response of flow rates and saturation temperatures of a two-stage system to an increase in refrigeration load.



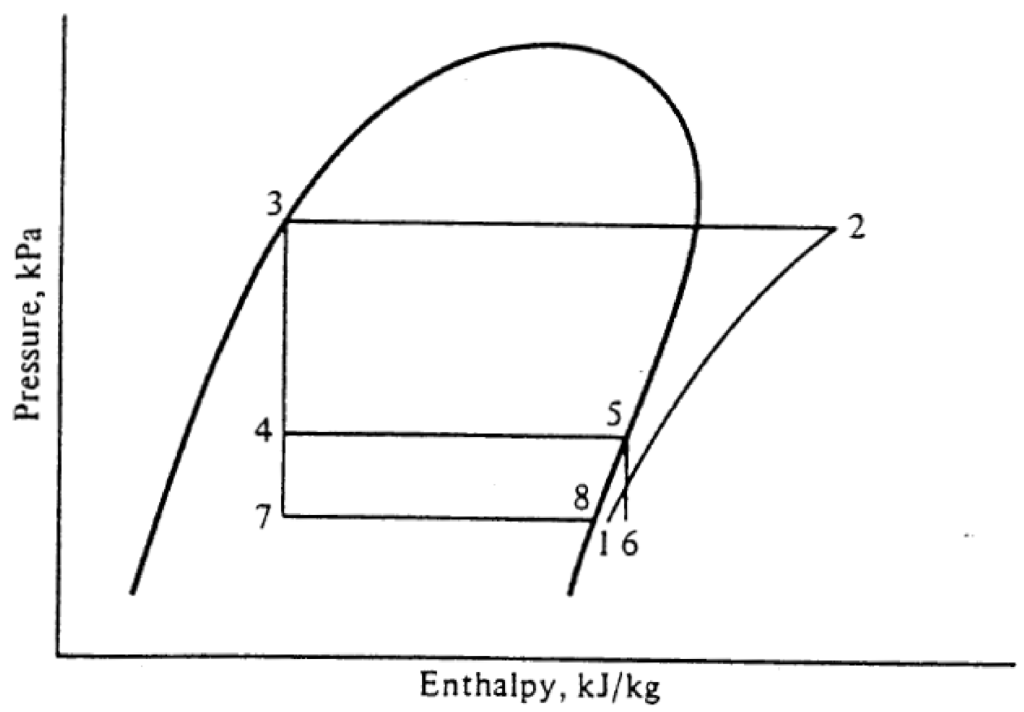
System with one compressor and one evaporator using a flash tank

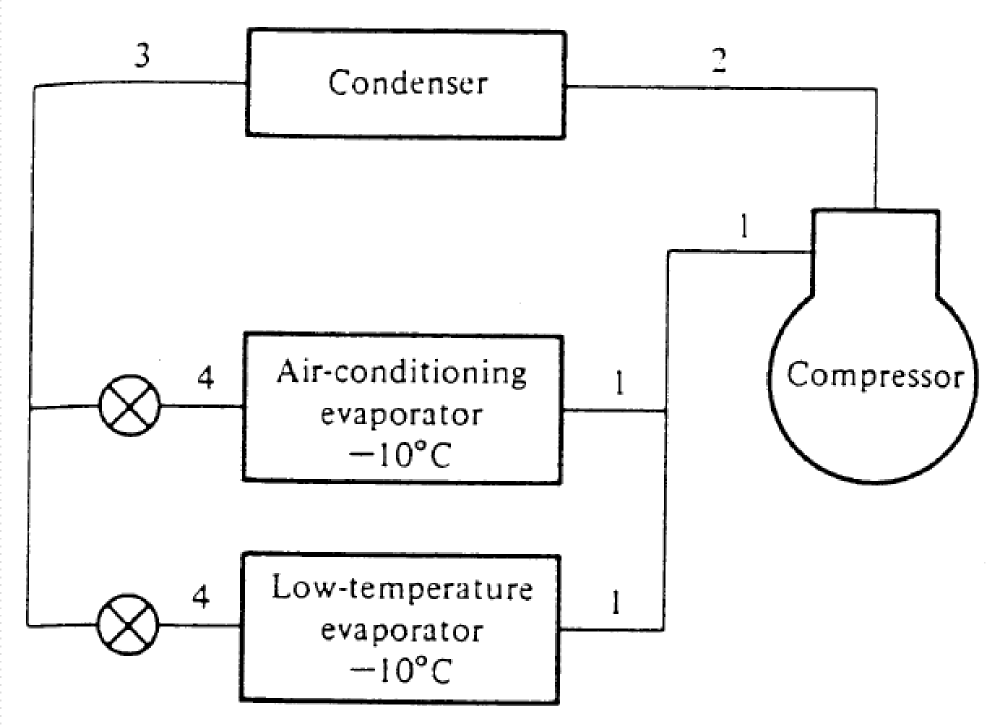


One compressor and two evaporators

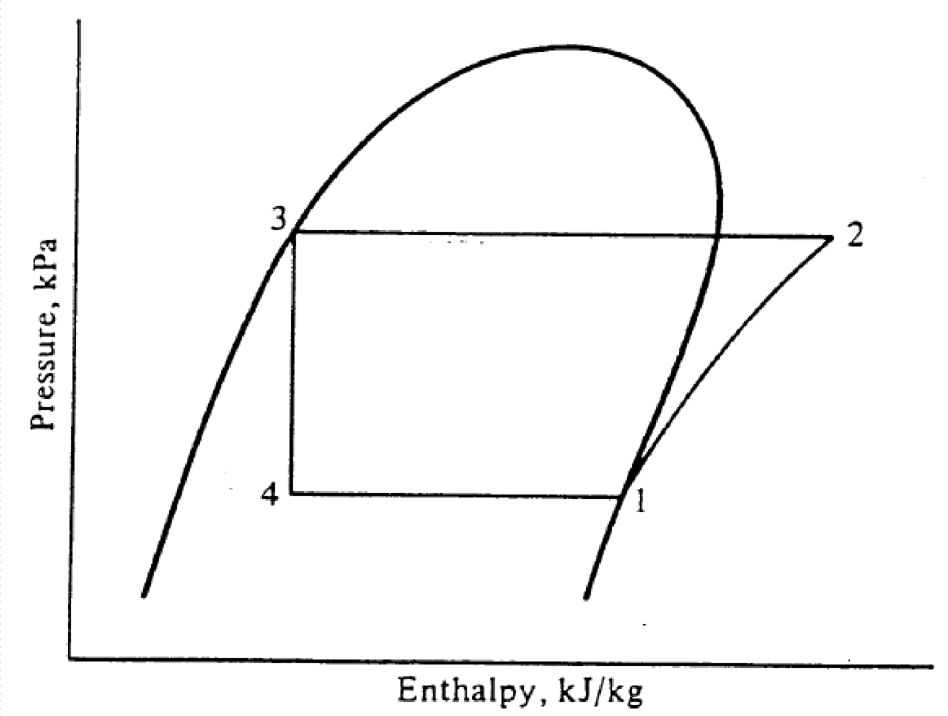


One compressor and two evaporators

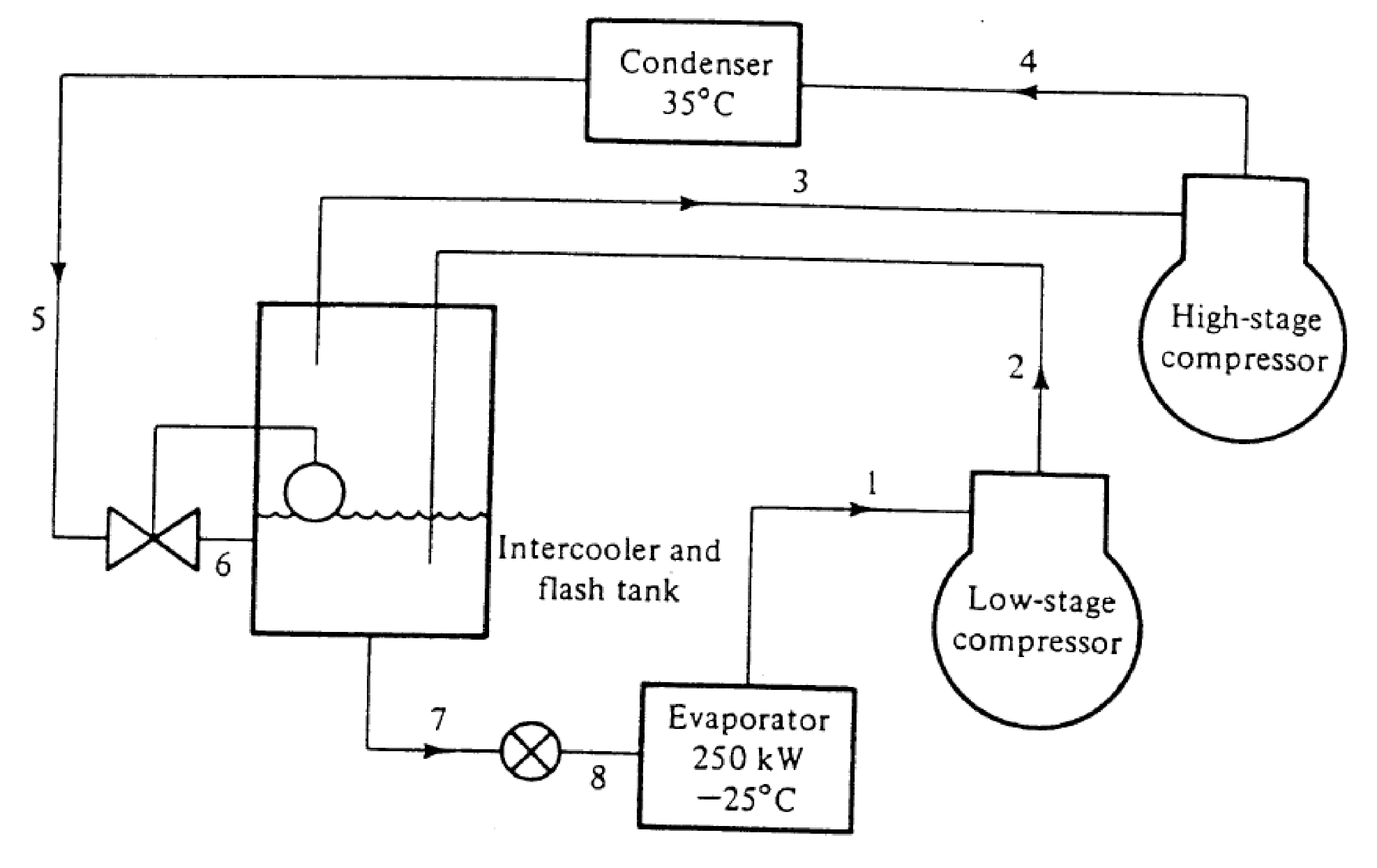



3
Condenser
2
1
Compressor
4
Air-conditioning evaporator
−10°C
1
4
Low-temperature evaporator
−10°C
1

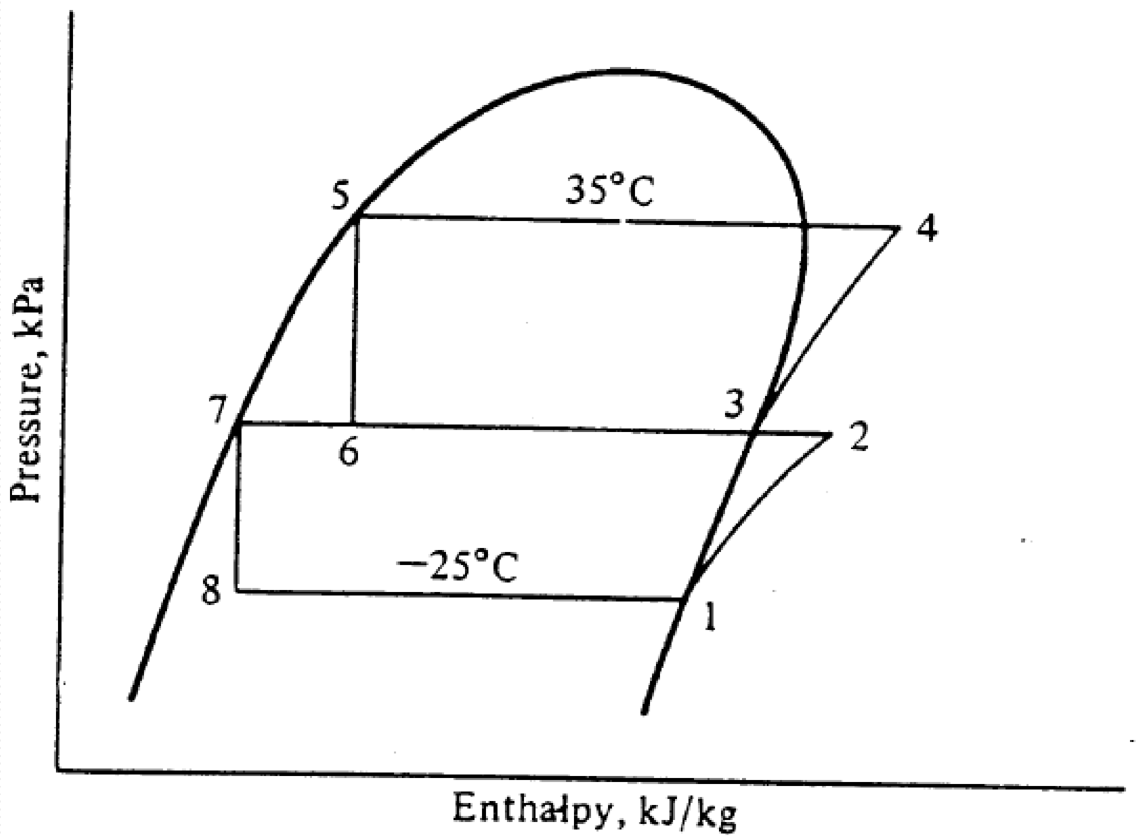
One compressor and two evaporators



Two compressor and one evaporators

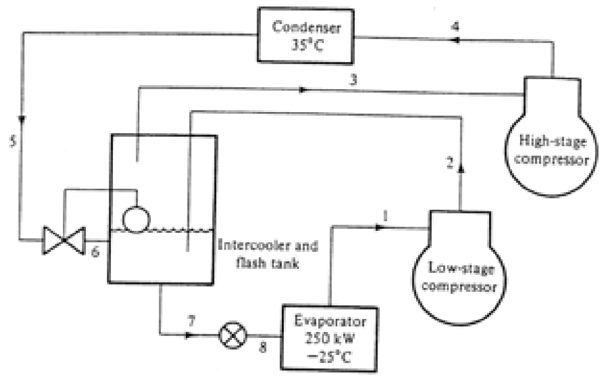


Two compressor and one evaporators

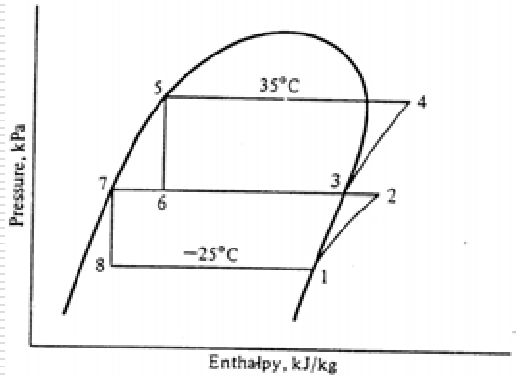


Example

Example 16-3 Calculate the power required by the two compressors in an ammonia system which serves a 250-kW evaporator at -25°C. The system uses two-stage compression with intercooling and removal of flash gas. The condensing temperature is 35°C.



p_s = saturation pressure at -25°C = 152 kPa
p_d = saturation pressure at 35°C = 1352 kPa
$p_i = \sqrt{152(1352)} = 453$ kPa



$h_1 = h_g$ at -25°C = 1430 kJ/kg
$h_2 = h$ at 453 kPa after isentropic compression = 1573
$h_3 = h_g$ at 453 kPa = 1463
$h_4 = h$ at 1352 kPa after isentropic compression = 1620
$h_5 = h_f$ at 35°C = 366 $\quad h_6 = h_5 = 366$
$h_7 = h_f$ at 453 kPa = 202 $\quad h_8 = h_7 = 202$

Example

Heat balance about the evaporator:

$$w_1 = \frac{250 \text{ kW}}{1430 - 202} = 0.204 \text{ kg/s}$$

$$w_1 = w_2 = w_7 = w_8 = 0.204 \text{ kg/s}$$

Heat and mass balance about the intercooler:

$$w_2 h_2 + w_6 h_6 = w_7 h_7 + w_3 h_3$$

$$w_6 = w_3 \quad \text{and} \quad w_7 = w_2$$

$$0.204(1573) + w_3(366) = 0.204(202) + w_3(1463)$$

$$w_3 = 0.255 \text{ kg/s}$$

Low-stage power: (0.204 kg/s) (1573 – 1430 kJ/kg) = 29.2 kW
High-stage power: (0.255 kg/s) (1620 – 1463 kJ/kg) = 40.0 kW
Total power: 29.2 + 40.0 = 69.2 kW

Example

This power requirement can be compared with that of a single-compressor system developing 250 kW of refrigeration at -25°C with a condensing temperature of 35°C. The pressure-enthalpy diagram is shown in Fig. 16-11. The enthalpies are:

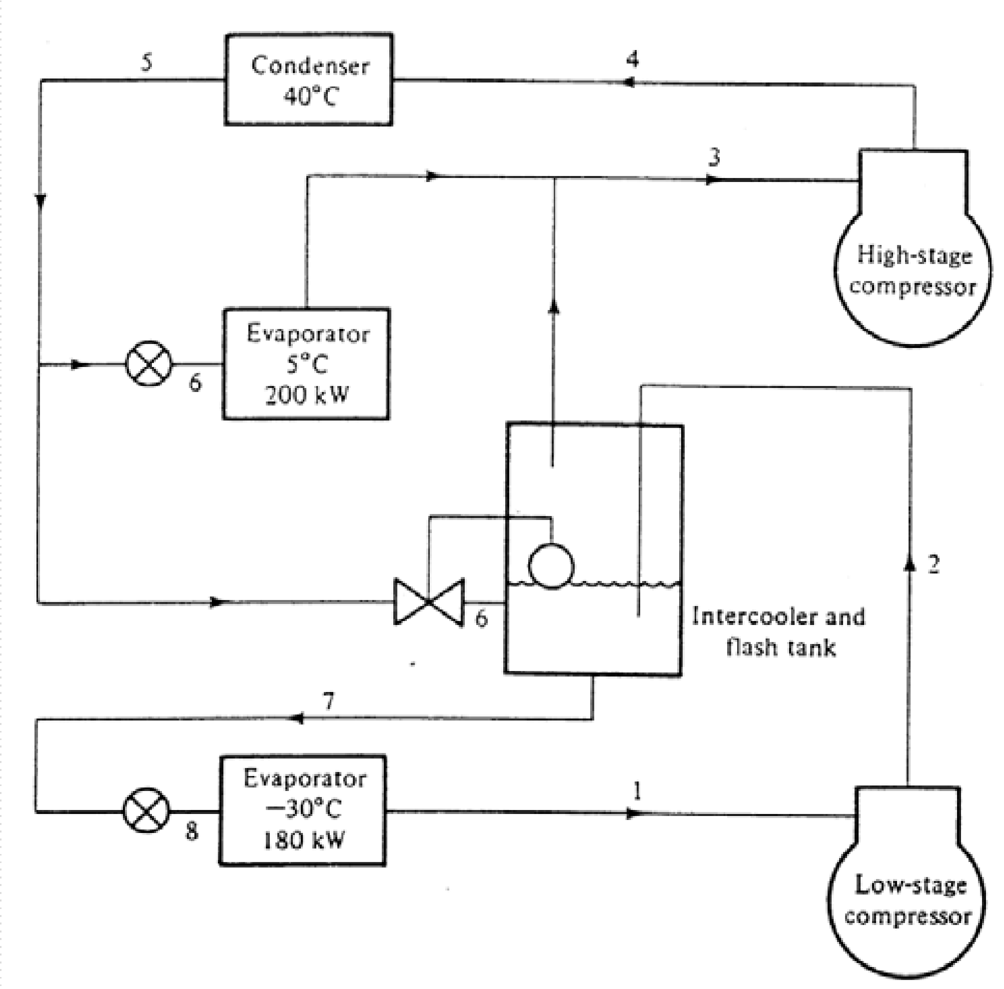
$h_1 = 1430$ kJ/kg $\quad h_2 = 1765 \quad h_3 = h_4 = 366$

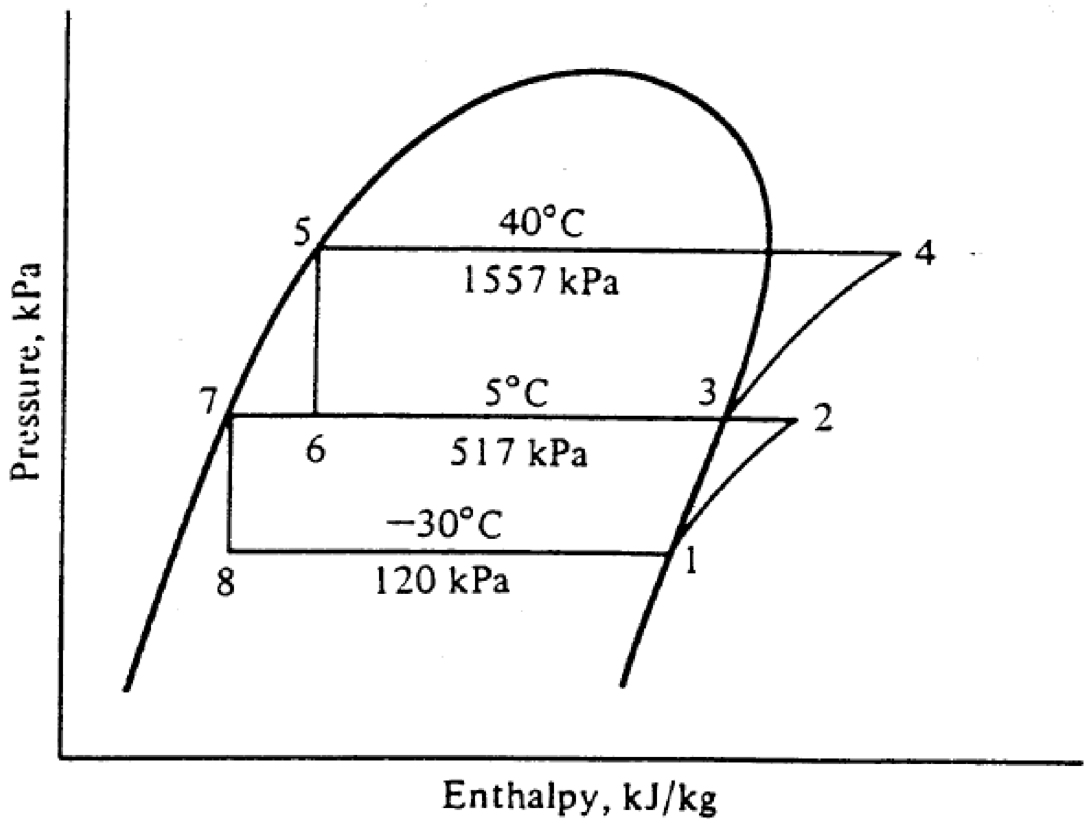
$$w_1 = \frac{250 \text{ kW}}{1430 - 366} = 0.235 \text{ kg/s}$$

$$\text{Power} = 0.235(1765 - 1430) = 78.7 \text{ kW}$$

The two-stage compressor system requires 69.2 kW, or 12 percent less power than the single-compressor system.

Two compressor and two evaporators




Pressure, kPa
5
40°C
1557 kPa
4
7
5°C
3
2
6
517 kPa
−30°C
1
8
120 kPa
Enthalpy, kJ/kg

Example

Example 16-4 In an ammonia system one evaporator is to provide 180 kW of refrigeration at -30°C and another evaporator is to provide 200 kW at 5°C. The system uses two-stage compression with intercooling and is arranged as in Fig. 16-12*a*. The condensing temperature is 40°C. Calculate the power required by the compressors.

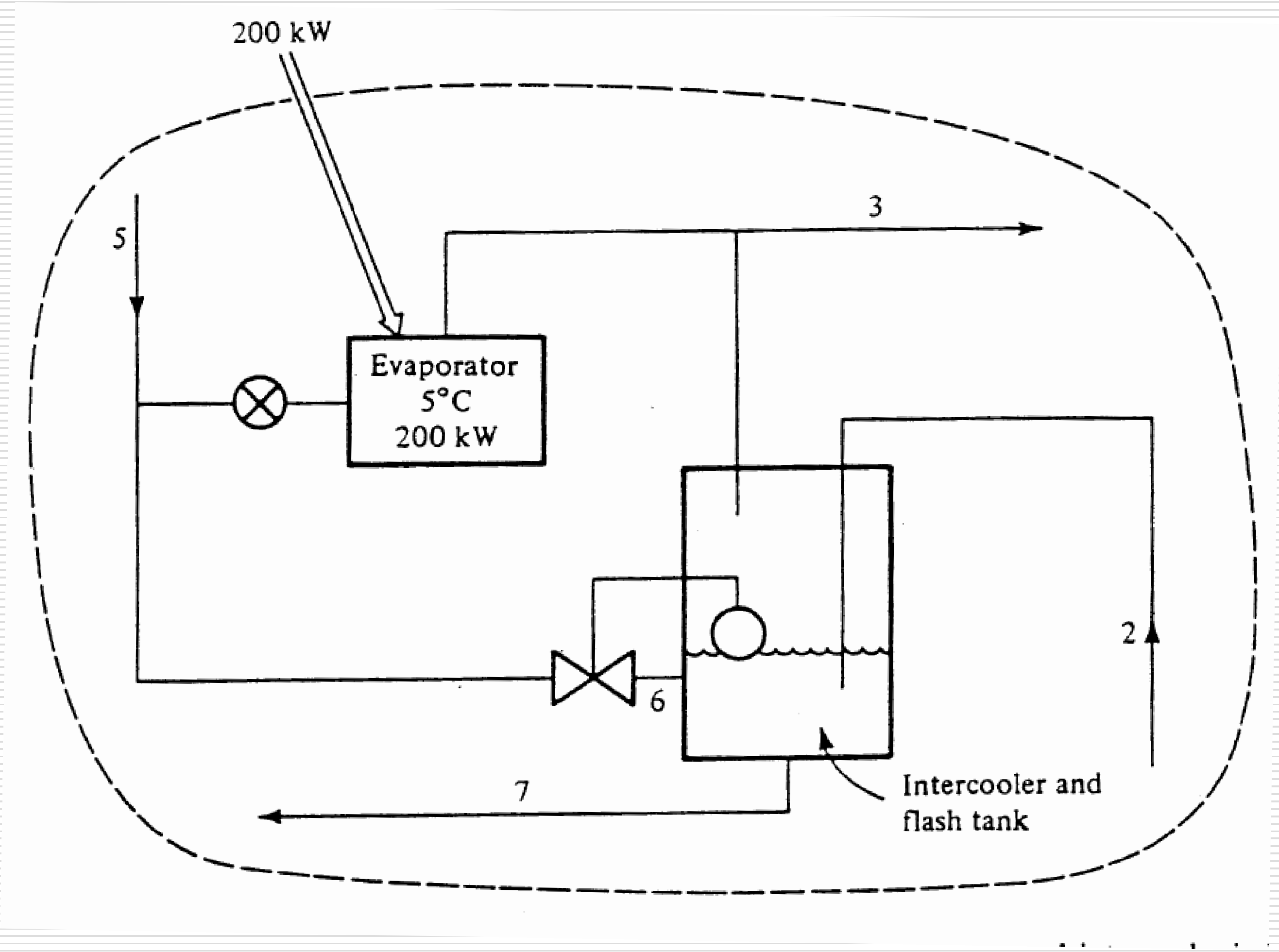
$h_1 = h_g$ at -30°C = 1423 kJ/kg
$h_2 = h$ at 517 kPa after isentropic compression = 1630
$h_3 = h_g$ at 5°C = 1467
$h_4 = h$ at 1557 kPa after isentropic compression = 1625
$h_5 = h_f$ at 40°C = 390.6 $\quad h_6 = h_5 = 390.6$
$h_7 = h_f$ at 5°C = 223 $\quad h_8 = h_7 = 223$

The mass rates of flow are

$$w_1 = \frac{180 \text{ kW}}{1423 - 223} = 0.150 \text{ kg/s}$$

$$w_7 = w_8 = w_2 = w_1 = 0.150 \text{ kg/s}$$

Example



200 kW
3
5
Evaporator
5°C
200 kW
2
6
7
Intercooler and
flash tank

Example

Heat balance:

$$w_5 h_5 + 200\ \text{kW} + w_2 h_2 = w_3 h_3 + w_7 h_7$$

Mass balance:

$$w_2 = w_7 = 0.150\ \text{kg/s}$$

Therefore

$$w_5 = w_3$$

Combining gives

$$390.6 w_3 + 200 + 0.150(1630) = 1467 w_3 + 0.150(223)$$

Solving leads to

$$w_3 = 0.382\ \text{kg/s}$$

The power required by the compressors can now be calculated:

Low-stage power:	0.150(1630 – 1423) =	31.1 kW
High-stage power:	0.382(1625 – 1467) =	60.4
	Total	91.5 kW

Example

If one compressor served each evaporator in single-stage compression, the power requirements of the two compressors would have been as follows:

Flow through low-temperature evaporator:

$$\frac{180 \text{ kW}}{1423 - 390.6} = 0.174 \text{ kg/s}$$

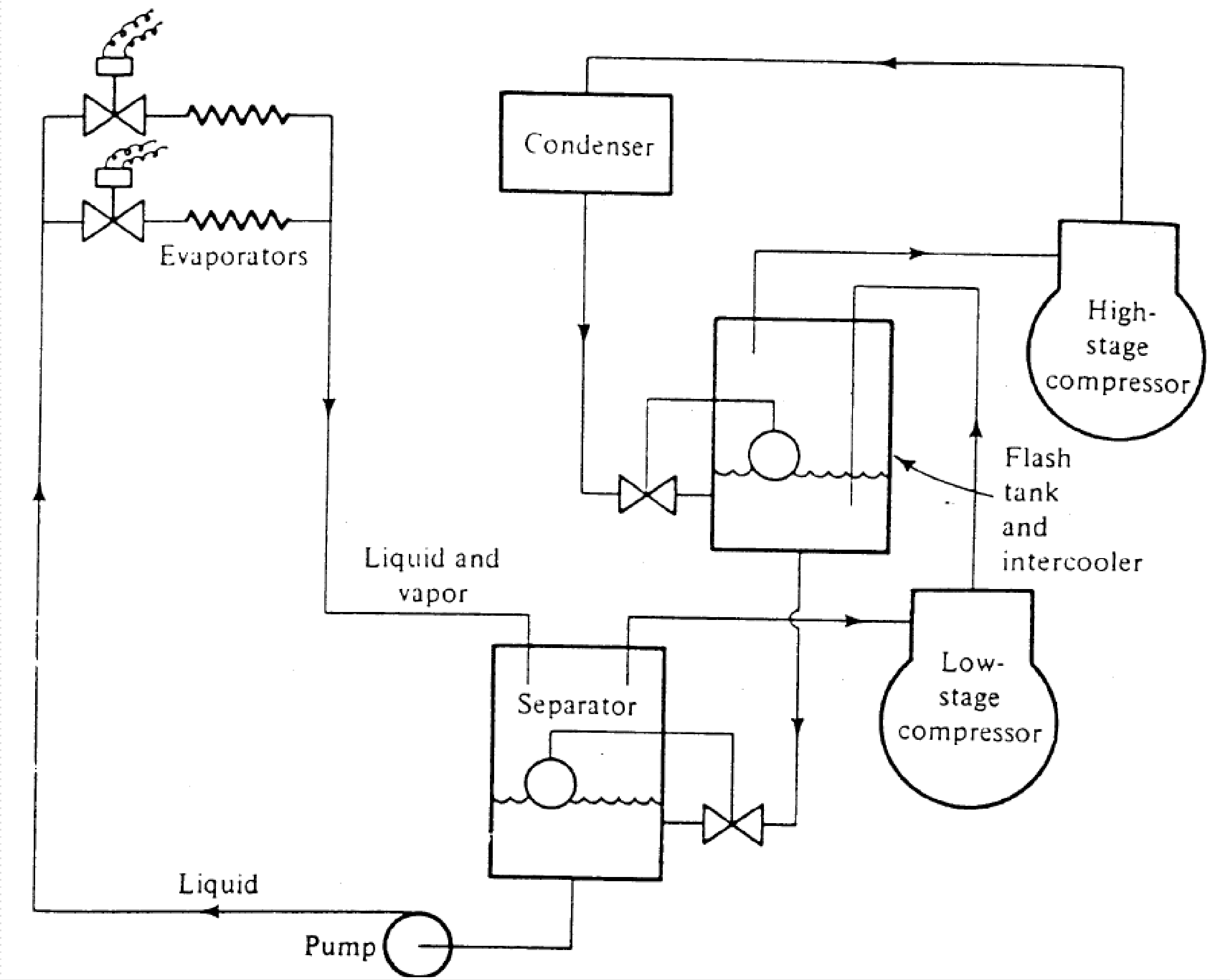
Flow through high-temperature evaporator:

$$\frac{200 \text{ kW}}{1467 - 390.6} = 0.186 \text{ kg/s}$$

Power for low-temperature system: 0.174(1815 − 1423) = 68.2 kW
Power for high-temperature system: 0.186(1625 − 1467) = 29.4

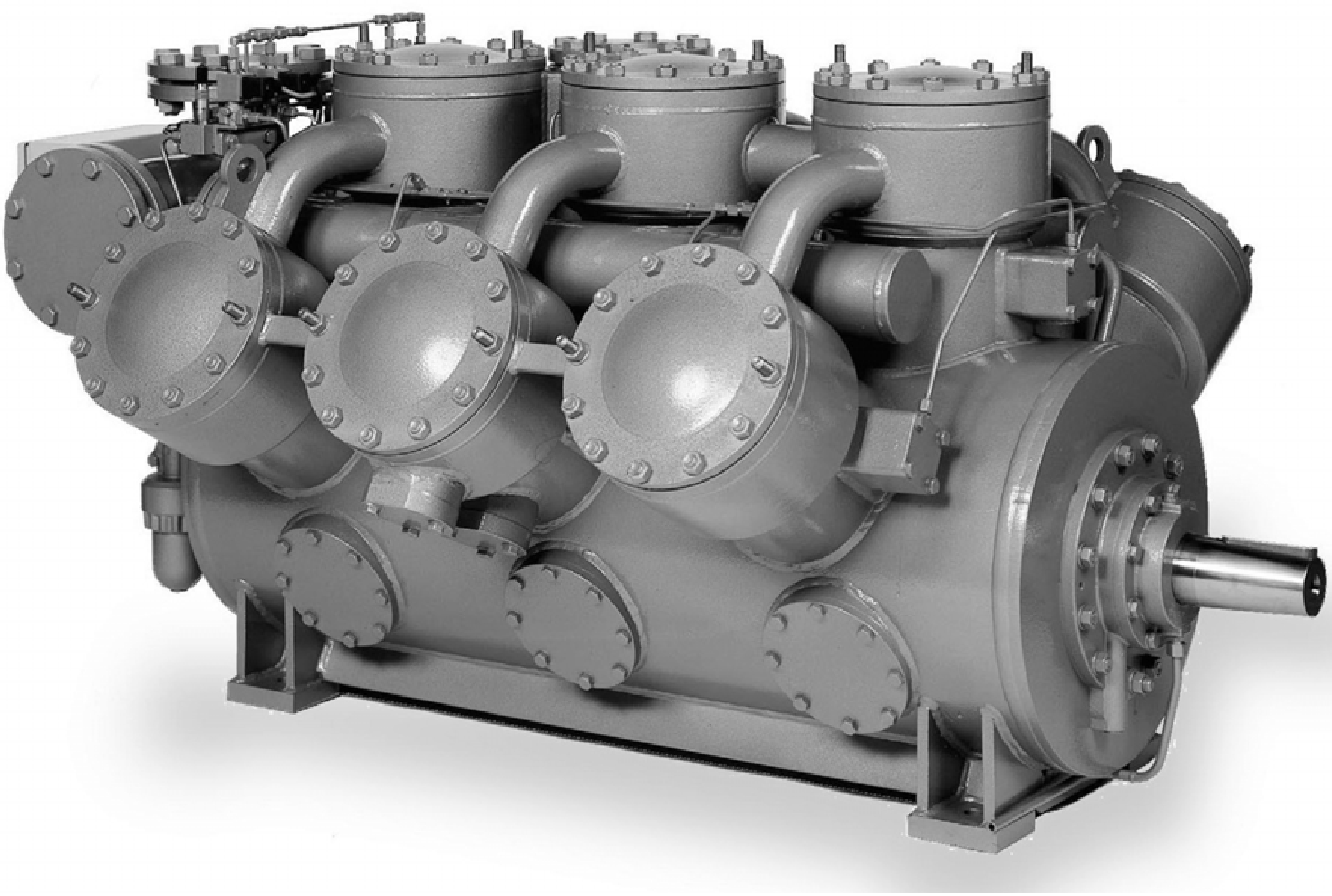
Total 97.6 kW

Liquid Recirculation System



Reciprocating Compressors

李魁鵬




Grasso RC12E Series
THE REVOLUTION
Reciprocating Compressors:
15 types, 6 single-stage and 9 two-stage versions with 2, 3, 4, 6, 9 and 12 cylinders
Swept volumes from 398 to 2388 m³/h (234 to 1406 CFM) at 1500 min⁻¹ (rpm)
Cylinder bore: 160 mm (6.3 in)
Piston stroke: 110 mm (4.3 in)
Speed: max. 1500 min⁻¹ (rpm)
Max. discharge pressure: 24 bar(a) (348 psia)
Type RC4212E, two-stage
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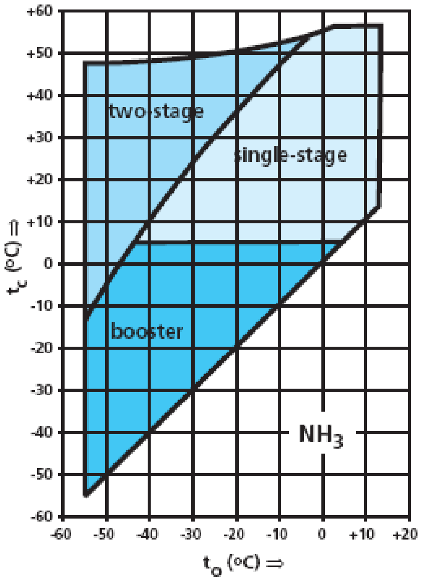
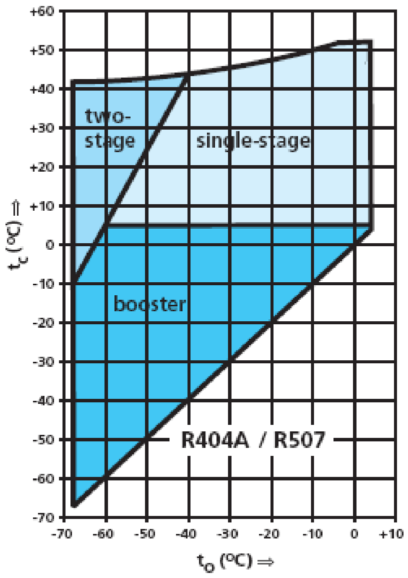
Fields of Application

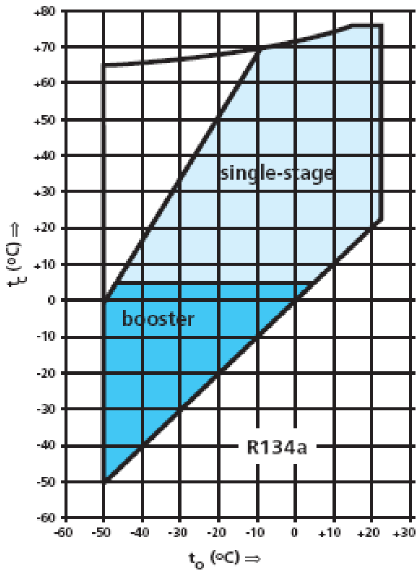
NH_3



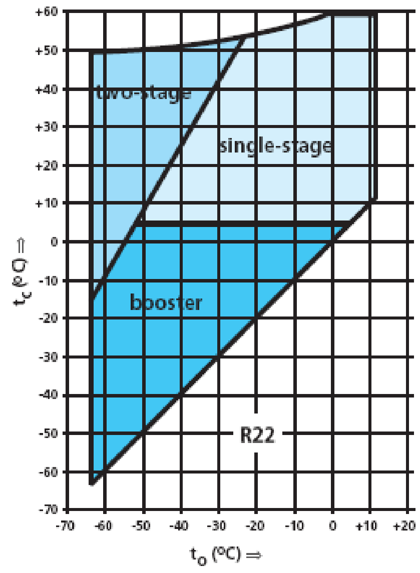
R404A/R507

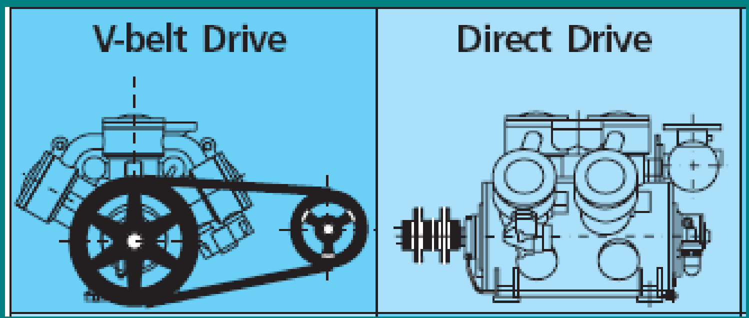


R134a

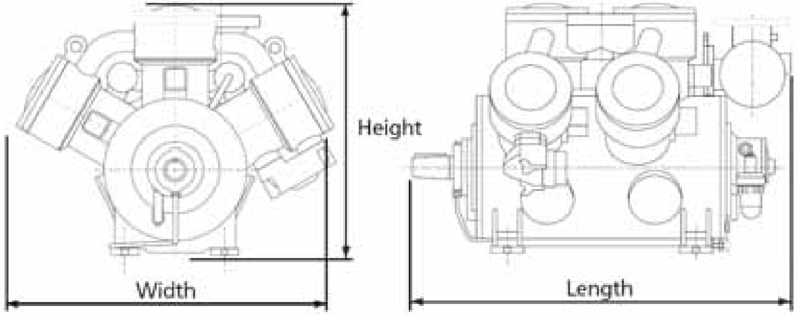


R22





Dimensions



Type		Dimensions (mm)			Mass	Swept Volume
	Grasso RC12E	Length	Width	Height	(kg)	at 1500/min-1*
Single-stage	212E	987	1077	881	553	398
	312E	987	1171	931	660	597
	412E	1317	1077	884	810	796
	612E	1377	1171	931	1050	1194
	912E	1816	1171	931	1390	1791
	1212E	2206	1171	931	1630	2388
Two-stage	2112E	987	1171	931	660	398
	3112E	1317	1077	884	885	597
	4212E	1377	1171	931	1100	796
	5112E	1377	1171	931	1100	995
	6312E	1816	1171	931	1440	1194
	7212E	1816	1171	931	1440	1393
	8412E	2206	1171	931	1700	1592
	9312E	2206	1171	931	1700	1791
	10212E	2206	1171	931	1700	1990

* *For NH_3 only*

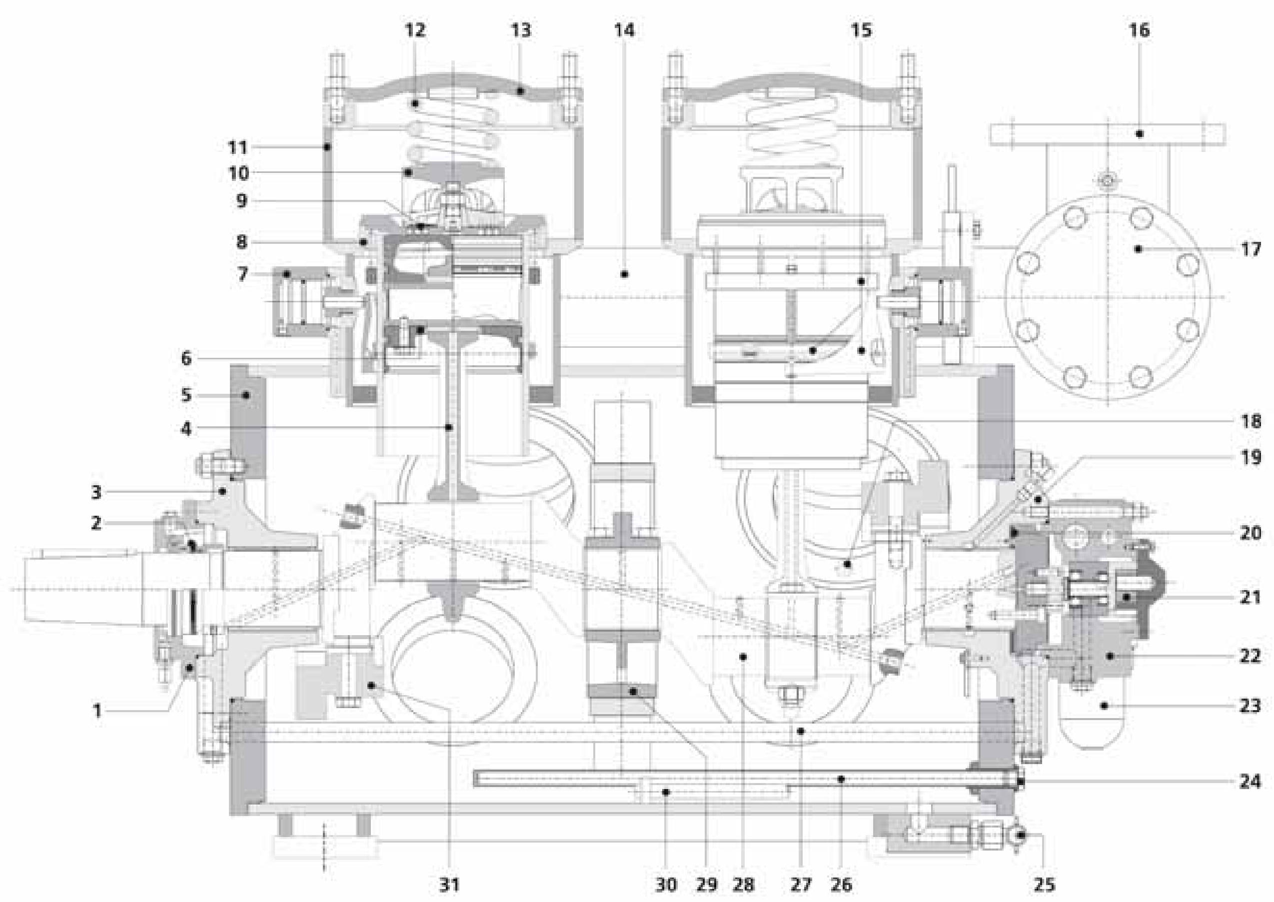
Refrigeration Capacities [kW]

	Type	NH_3 at 1450 min-1		R22 at 1200 min-1	
Single-stage	Grasso RC12E	-15 °C	-5 °C	-15 °C	-5 °C
	212E	179.7	288.1	141.1	213.8
	312E	269.5	432.1	211.6	320.7
	412E	359.4	576.2	282.2	427.6
	612E	539.1	864.3	423.3	641.4
	912E	808.6	1296.5	634.9	962.1
	1212E	1078.2	1728.6	846.6	1282.8
Two-stage	Grasso RC12E	-40 °C	-30°C	-40 °C	-30 °C
	2112E	71.4	114.5	64.8	96.2
	3112E	98.8	157.6	87.4	129.0
	4212E	142.8	229.0	129.6	192.4
	5112E	142.6	-	122.6	-
	6312E	214.2	343.5	194.4	288.6
	7212E	221.7	354.0	194.5	-
	8412E	285.6	458.0	259.2	384.8
	9312E	296.4	472.8	262.2	387.0
	10212E	285.2	-	245.2	-

Condensing temperature = +30 °C, Liquid subcooling = 0 K, Suction superheat = 0 K, Intermediate superheat = 6 K, Temperature difference interstage cooler = 10 K, Economiser B

Table 1.2-1 Technical Data of Grasso 12 series

COMPRESSOR TYPE			Single-stage						Two-stage								
			212	312	412	612	912	1212	2112	3112	4212	5112	6312	7212	8412	9312	10212
Number of cylinders	z_L		2	3	4	6	9	12	2	3	4	5	6	7	8	9	10
Number of cylinders	z_H								1	1	2	1	3	2	4	3	2
Cylinder arrangement			1xV	1xW	2xV	2xW	3xW	4xW	1xW	2xV	2xW	2xW	3xW	3xW	4xW	4xW	4xW
Cylinder bore	D	mm	160														
Piston stroke	S	mm	110														
LP Swept volume at full-load and: $n=1000/min^{-1}$	V_s	m^3/h	265	398	531	796	1194	1592	265	398	531	664	796	929	1062	1194	1327
Ratio LP/HP swept volume (Z_L/Z_H) at full load	φ								2	3	2	5	2	3.5	2	3	5
Standard direction of rotation			counter-clockwise when facing shaft end														
Standard compres-sor speeds (with V-belt drive) at motor speed: 1450 min^{-1} (50 Hz)	n	min^{-1}	535- 600- 675- 720- 765- 860- 965														
Standard compres-sor speeds (with V-belt drive) at motor speed: 1750 min^{-1} (60 Hz)	n	min^{-1}	520- 550- 580- 650- 725- 820-8 70- 920														
Standard steps of capacity control (expressedin % of full-load swept volume):	S[a]	%	100-50	100-67- 33	100-75- 50	100-83-67-50 - 33	100 -8967-44- 33	100-83-67-50- 25									
	B[b]	%	100-50	100-67- 33	100-75- 50	100-83-67-50	100-67- 44	100-75-58-42									
	S[a]	%	100-50	100-67- 33	100-75- 50	100-83-67-50- 33	100-89-78-67-56-44- 33	100-92-83-75-67-58-50-42-33-25									
	B[b]	%	100-50	100-67-33	100-75- 50	100-83-67-50	100-67- 44	100-75-58-42									
	T[c]	%							100	100-67	100-75- 50	100-80-60-40	100-67- 33	100-86-72-57-43-29	100-75-50-25	100-78-67-56-45-33	100-80-70-60-50-30



Legend			
1	Shaft seal housing	17	Suction gas filter housing
2	Rotary shaft seal	18	Plug with oil return orifice
3	Bearing cover (drive end)	19	Bearing cover (oil pump end)
4	Connecting rod	20	Thrust bearing
5	Crankcase	21	Oil pump
6	Piston	22	Oil pump housing
7	Valve lifting control cylinder	23	Oil discharge filter
8	Cylinder liner	24	Plugged off connection for crankcase heater
9	Suction and discharge valve assembly	25	Oil charge and drain valve
10	Buffer spring cup	26	Sleeve for heating element
11	Cylinder jacket	27	Oil supply line
12	Buffer spring	28	Crankshaft
13	Cylinder head cover	29	Intermediate bearing
14	Suction manifold	30	Oil intake line
15	Valve lifting mecanism	31	Counterweight

16	Suction connection

Effect of the Evaporating Temperature on Volumetric Efficiency

The volumetric efficiency is a key term in explaining trends in the refrigerating capacity and power requirement of reciprocating compressors. The volumetric efficiency of a compressor, η_v in percent, is defined by the equation:

$$\eta_v = \frac{\text{volume rate entering compressor, m}^3/\text{s (ft}^3/\text{min)}}{\text{displacement rate, m}^3/\text{s (ft}^3/\text{min)}}(100) \qquad (4.1)$$

Effect of the Evaporating Temperature on Volumetric Efficiency

Example 4.1. What is the volumetric efficiency of an eight-cylinder Vilter 458XL ammonia compressor operating at 1200 rpm when the saturated suction temperature is −1°C (30.2°F) and the condensing temperature is 30°C (86°F)? The bore and stroke of the compressor are 114.3 by 114.3 mm (4-1/2 by 4-1/2 in). The catalog lists the refrigerating capacity at this condition as 603.1 kW (171.5 tons).

Effect of the Evaporating Temperature on Volumetric Efficiency

Solution. The volume swept by one piston during a stroke is:

$$\text{swept volume} = (\pi\ 0.1143^2/4\ \text{m}^2)(0.1143\ \text{m}) = 0.001173\ \text{m}^3\ (0.0414\ \text{ft}^3)$$

The displacement rate is the displacement volume of one cylinder multiplied by the number of cylinders and the number of strokes per second:

$$\text{displacement rate} = (0.001173\ \text{m}^3)(20\ \text{rev/s})(8\ \text{cylinders})$$
$$= 0.1877\ \text{m}^3/\text{s}\ (397.7\ \text{cfm})$$

The mass flow rate can be computed by dividing the refrigerating capacity by the refrigerating effect. The enthalpy of ammonia leaving the condenser and entering the evaporator is 342.0 kJ/kg (138.7 Btu/lb) and the enthalpy leaving the evaporator is 1460.8 kJ/kg (619.6 Btu/lb). The mass flow rate $\dot{m}$ is:

$$\dot{m} = \frac{603.1\ \text{kW}}{1460.8 - 342\ \text{kJ/kg}} = 0.539\ \text{kg/s}\ (71.3\ \text{lb/min})$$

Effect of the Evaporating Temperature on Volumetric Efficiency

The specific volume of the refrigerant entering the compressor is 0.3007 m^3/kg (4.82 ft^3/lb), so the actual volume flow rate is:

$$\begin{aligned}\textbf{volume flow rate} &= (0.539\ \text{kg/s})(0.3007\ \text{m}^3/\text{kg}) \\ &= 0.1621\ \text{m}^3/\text{s}\ (343.4\ \text{cfm})\end{aligned}$$

Equation 4.1 can now be applied to compute η_v:

$$\eta_v = \text{volumetric efficiency} = \frac{0.1621\ \text{m}^3/s}{0.1877\ \text{m}^3/s}(100) = 86.4\%$$

Effect of the Evaporating Temperature on Volumetric Efficiency

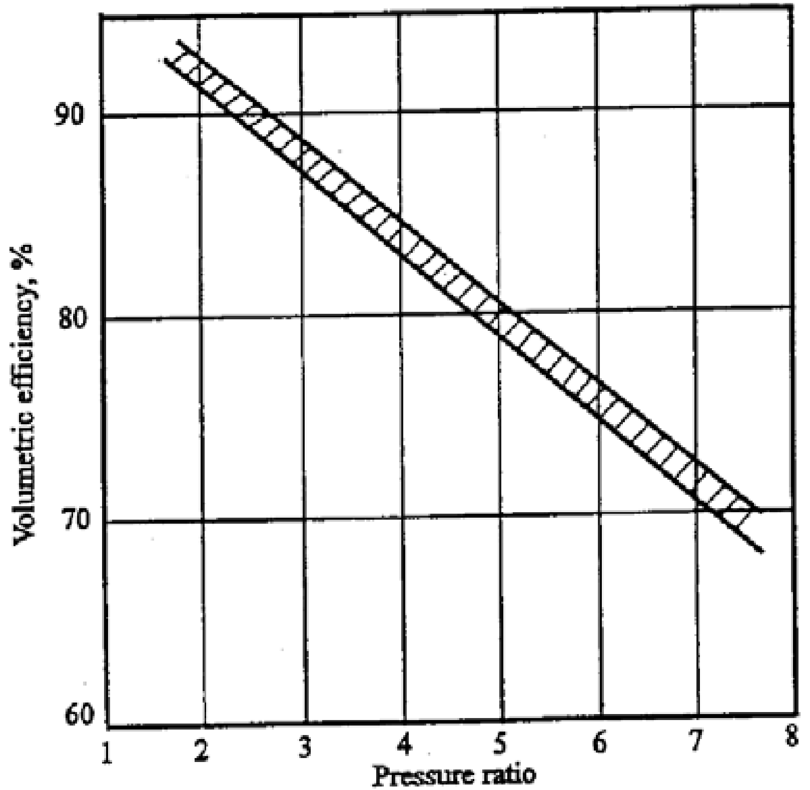


FIGURE 4.2
Band of volumetric efficiencies of an 8-cylinder Sabroe 108L ammonia compressor operating at 1170 rpm.

Effect of the Evaporating Temperature on Volumetric Efficiency

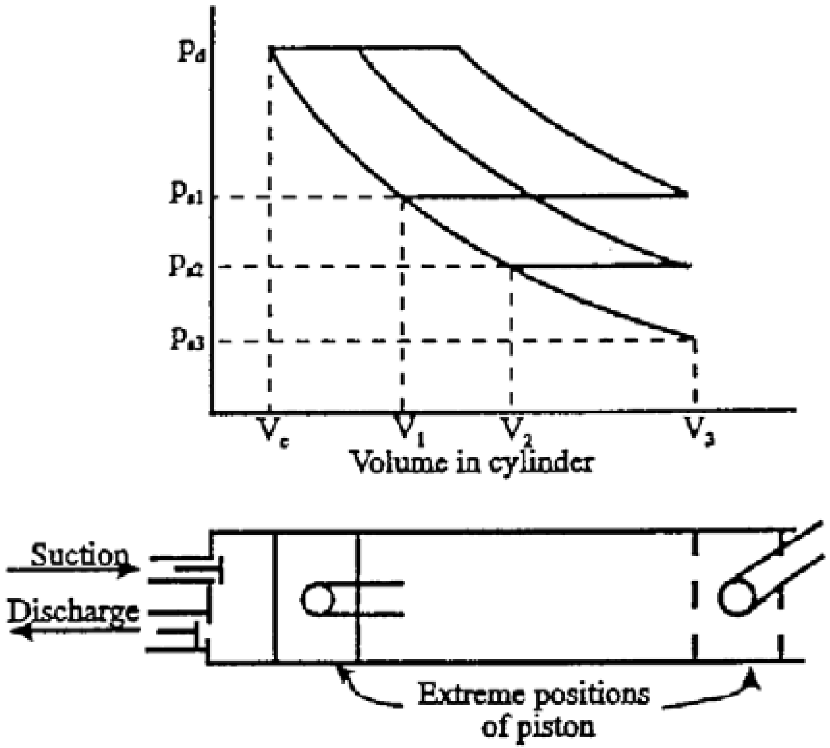


FIGURE 4.3
Pressure-volume diagram corresponding to several piston positions.

$$\eta_{vc} = \left(\frac{V_3 - V_1}{V_3 - V_c}\right) 100$$

clearance volumetric efficiency, η_{vc}

Effect of the Evaporating Temperature on Volumetric Efficiency

$$\eta_{vc} = \left(\frac{V_3 - V_1}{V_3 - V_c}\right) 100 \qquad (4.2)$$

The magnitude of V_c is a function of the design and construction of the compressor, and most manufacturers try to keep this volume to a minimum. V_c is often expressed as a percentage of the swept volume in a term called the percent clearance, m

$$\text{Percent clearance} = m = \left(\frac{V_c}{V_3 - V_c}\right) 100 \qquad (4.3)$$

Effect of the Evaporating Temperature on Volumetric Efficiency

$$\eta_{vc} = \left(\frac{V_3 - V_c + V_c - V_1}{V_3 - V_c}\right)100 = 100 + \left(\frac{V_c - V_1}{V_3 - V_c}\right)100$$

$$\eta_{vc} = 100 - 100\left(\frac{V_c}{V_3 - V_c}\right)\left(\frac{V_1}{V_c} - 1\right) = 100 - m\left(\frac{V_1}{V_c} - 1\right) \qquad (4.4)$$

Effect of the Evaporating Temperature on Volumetric Efficiency

Finally, the volume V_1 can be related to V_c by assuming that the expansion of the clearance gas to the suction pressure is an isentropic expansion—the reverse of an isentropic compression. The relationship of pressures and specific volumes in an isentropic process between point a and point b can be approximated by the relationship:

$$p_a v_a^n = p_b v_b^n \quad \text{or} \quad \left(\frac{v_b}{v_a}\right) = \left(\frac{p_a}{p_b}\right)^{1/n} \tag{4.5}$$

where the exponent n is unique for each substance. For ammonia the value of n is about 1.28 and for R-22 its value is approximately 1.11.

Effect of the Evaporating Temperature on Volumetric Efficiency

The term V_1/V_c in Eq. 4.4 can be expressed in terms of the pressure ratio, $(p_c/p_1)^{1/n}$. The expression for the clearance volumetric efficiency then becomes:

$$\eta_{vc} = 100 - m[(\text{pressure ratio})^{1/n} - 1] \qquad (4.6)$$

Example 4.2. The percent clearance of the ammonia compressor whose actual volumetric efficiency is shown in Fig. 4.2 is 3%. At a pressure ratio of 5.0, what is the clearance volumetric efficiency?

Solution. Applying Eq. 4.6,

$$\eta_{vc} = 100 - 3(5^{1/1.28} - 1) = 92.5\%$$

Influence of the Evaporating Temperature on Refrigerating Capacity

The overall equation that expresses the refrigeration rate is:

$$q_r = \dot{V}_d\,(\eta_v/100)\,(1/v_s)\,\Delta h_{ev} \qquad (4.7)$$

where:

q_r	=	refrigeration rate, kW [(tons of refrigeration)200]
$\dot{V}_d$	=	displacement rate, m^3/s (ft^3/min)
η_v	=	actual volumetric efficiency, percent
v_s	=	specific volume of gas entering the compressor, m^3/kg (ft^3/lb)
Δh_{ev}	=	refrigerating effect, kJ/kg (Btu/lb)

Influence of the Evaporating Temperature on Refrigerating Capacity

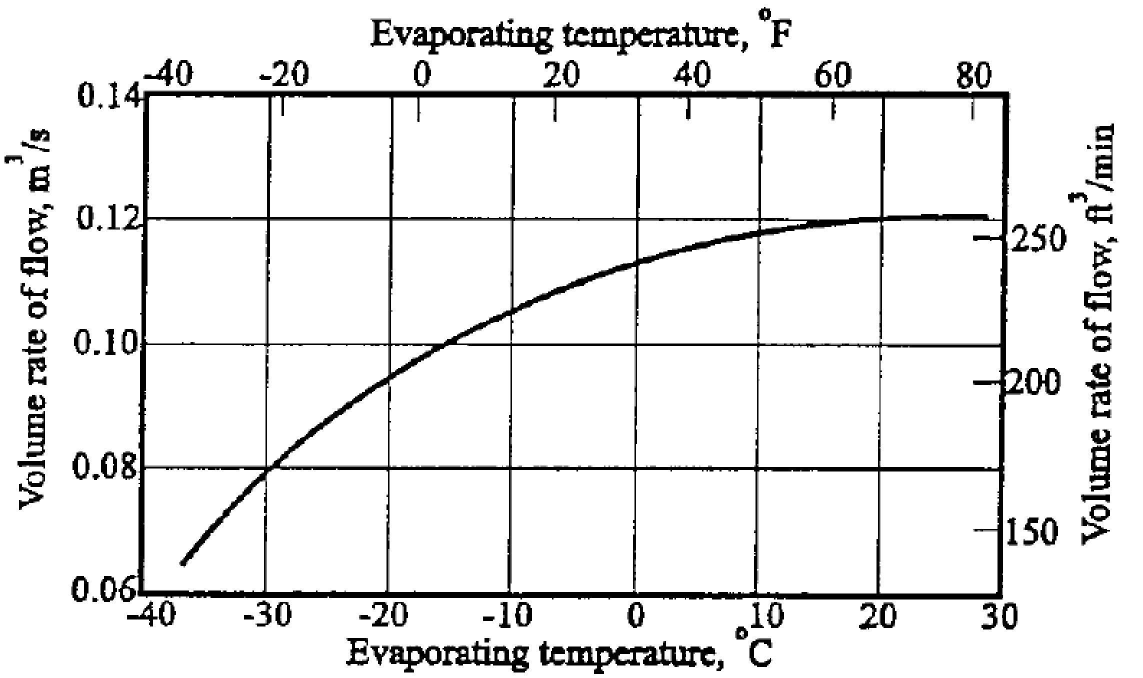


FIGURE 4.4
Effect of evaporating temperature on volume rate of flow measured at the compressor suction of an 8-cylinder compressor with a displacement rate of 0.123 m^3/s (260 cfm) operating with a condensing temperature of 30°C (86°F).

Influence of the Evaporating Temperature on Refrigerating Capacity

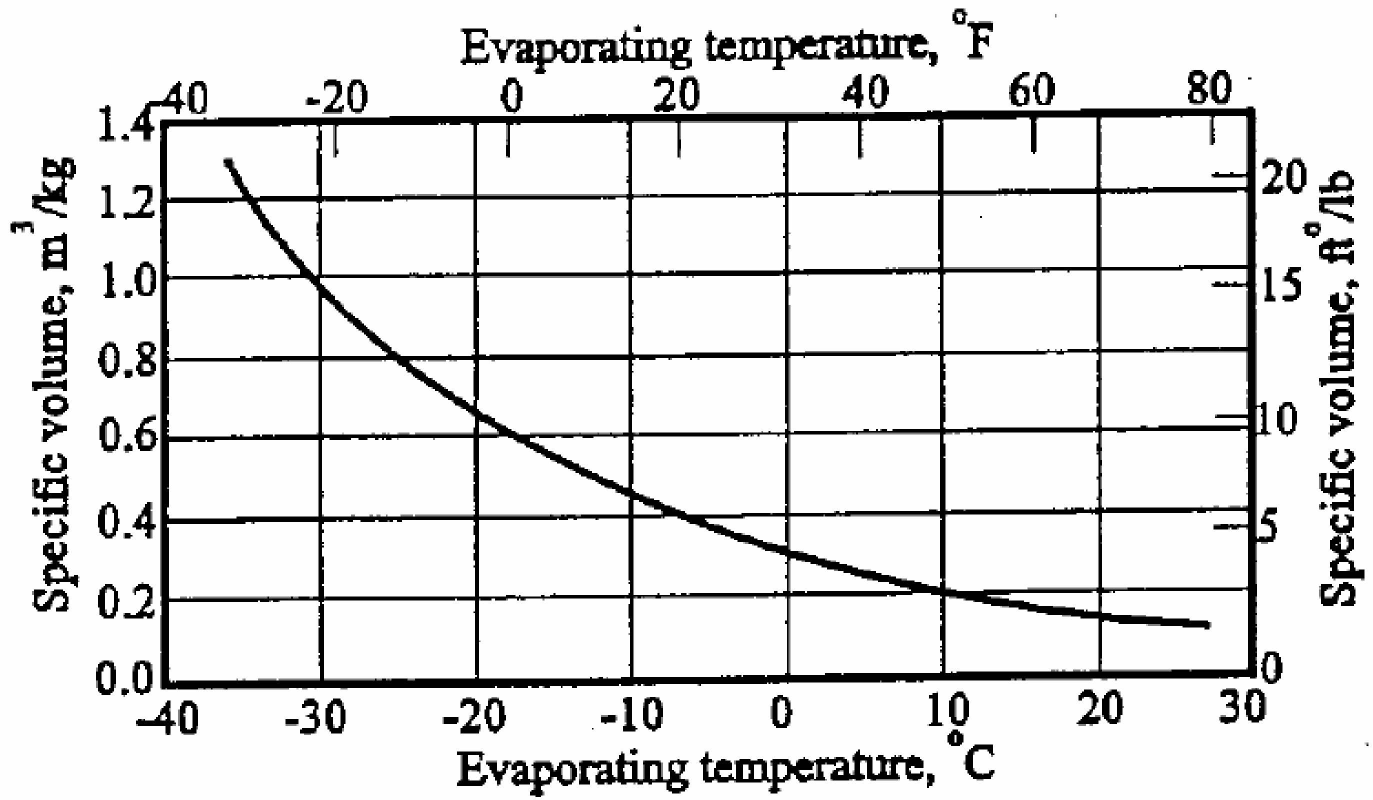


FIGURE 4.5
Variation in the specific volume of ammonia suction vapor with evaporating temperature.

Influence of the Evaporating Temperature on Refrigerating Capacity

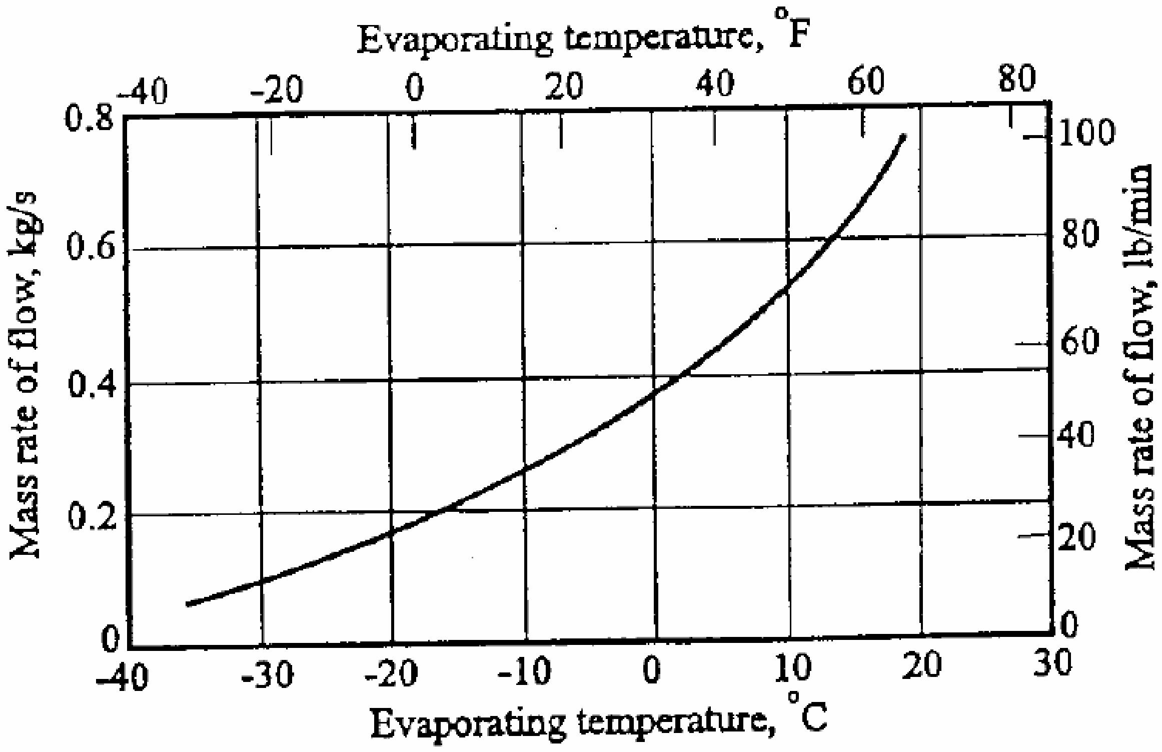


FIGURE 4.6
Influence of the evaporating temperature on the mass rate of flow of ammonia for the reciprocating compressor introduced in Figure 4.4.

Influence of the Evaporating Temperature on Refrigerating Capacity

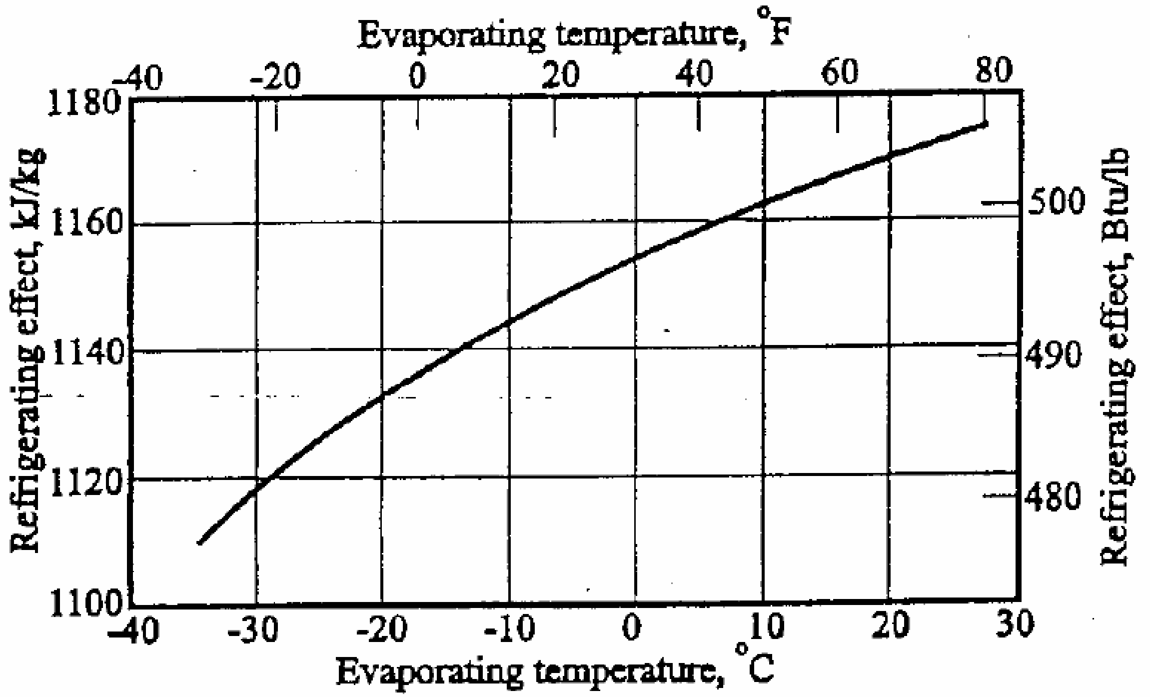


FIGURE 4.7
Refrigerating effect for ammonia as a function of the evaporating temperature, with a condensing temperature of 30°C (10°F) assuming 5.6°C (10°F) of liquid subcooling and 5.6°C (10°F) suction superheat.

Influence of the Evaporating Temperature on Refrigerating Capacity

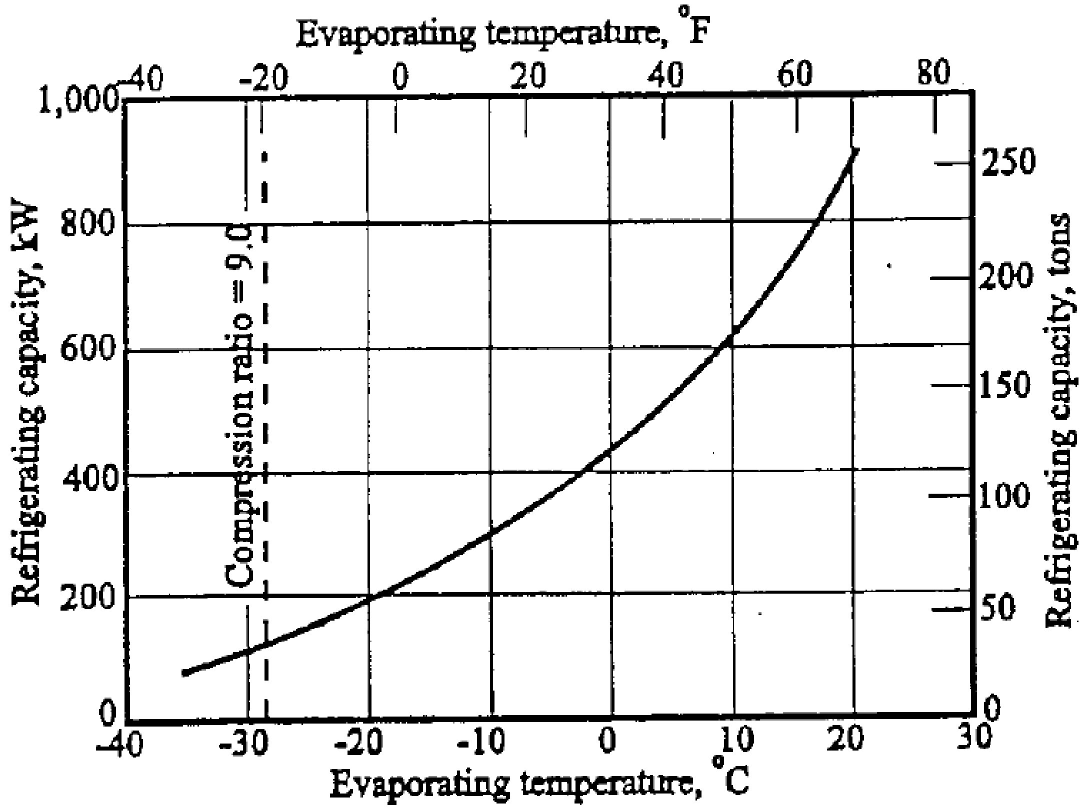


FIGURE 4.8
Effect of the evaporating temperature on refrigerating capacity of the ammonia compressor in Figure 4.4. The condensing temperature is constant at 30°C (86°F).

Influence of the Condensing Temperature on Refrigerating Capacity

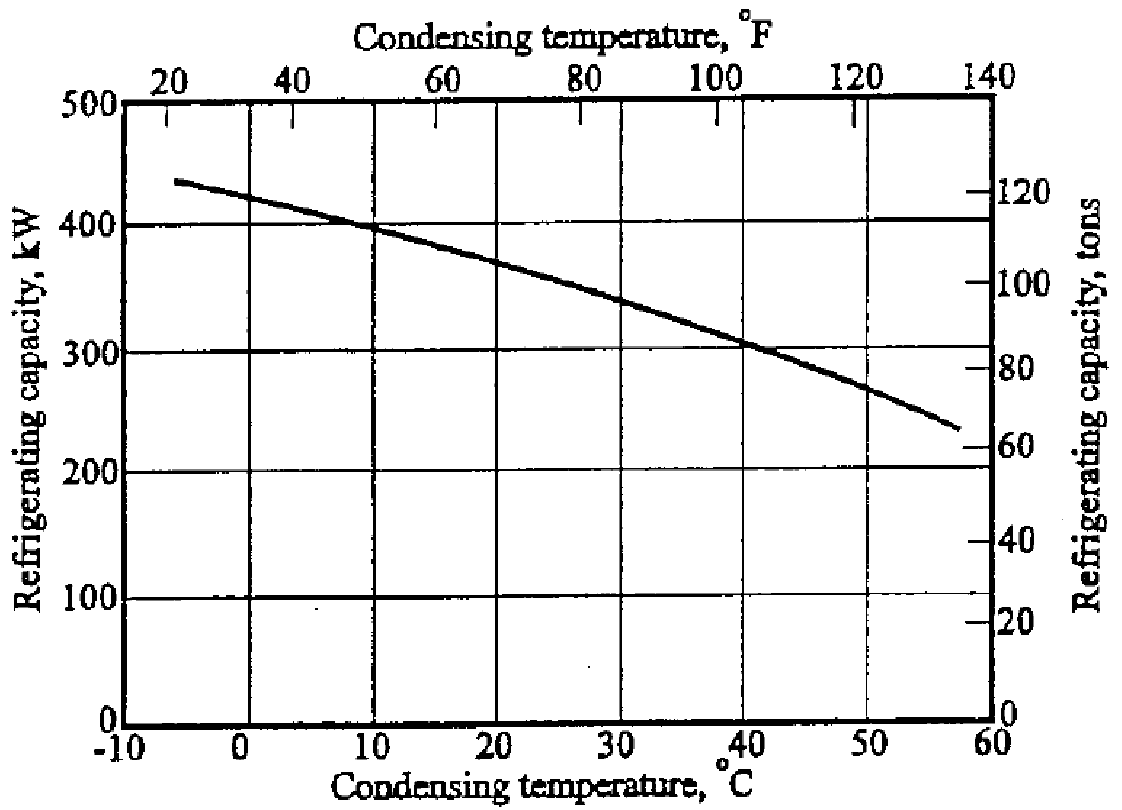


FIGURE 4.9
Effect of condensing temperature on the refrigerating capacity of an 8-cylinder ammonia compressor with a displacement rate of 0.123 m^3/s (260 cfm) operating with an evaporating temperature of −10°C (14°F).

Influence of the Condensing Temperature on Refrigerating Capacity

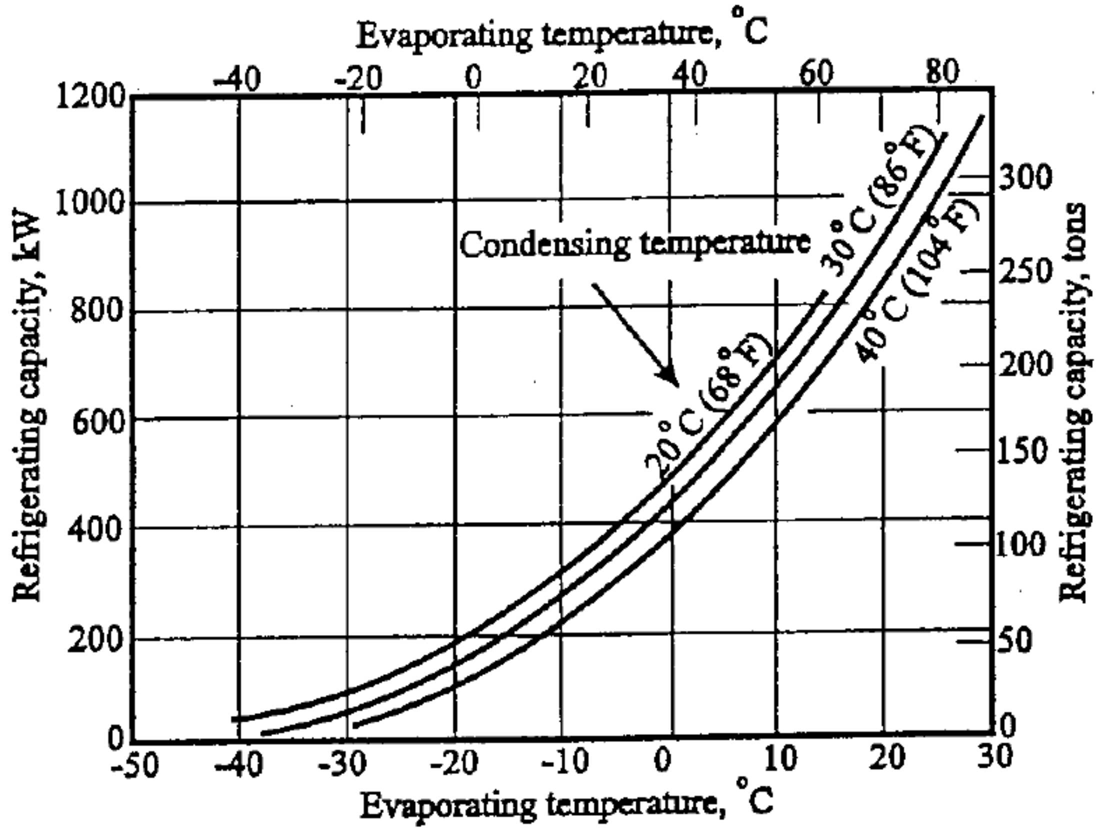


FIGURE 4.10
Effect of both the evaporating and condensing temperatures on refrigeration capacity for the 8-cylinder ammonia compressor.

Power Required by a Reciprocating Compressor

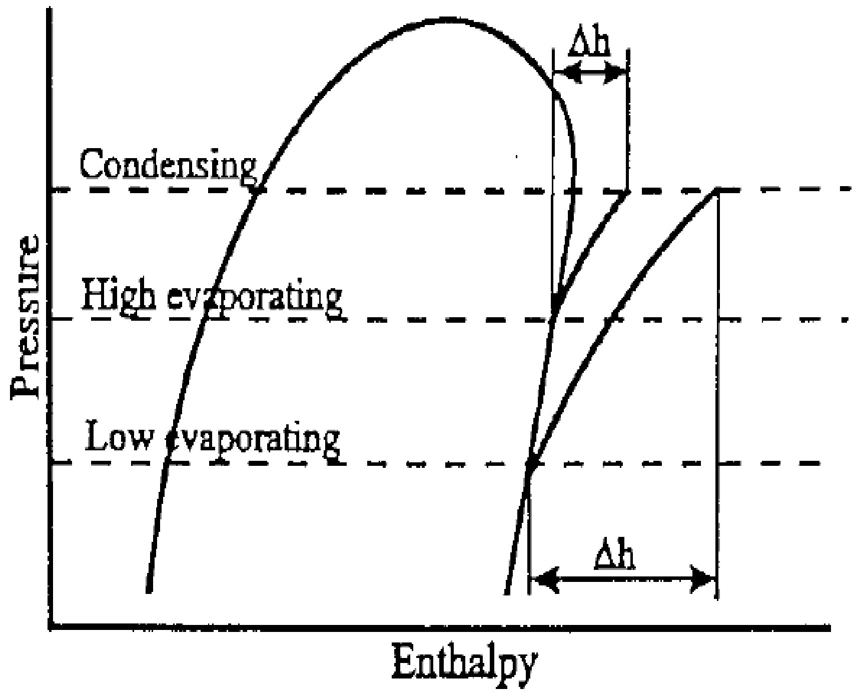


FIGURE 4.11
Effect of evaporating temperature on the ideal work of compression.

$$P' = \dot{m}\,\Delta h_{ideal}$$

Power Required by a Reciprocating Compressor

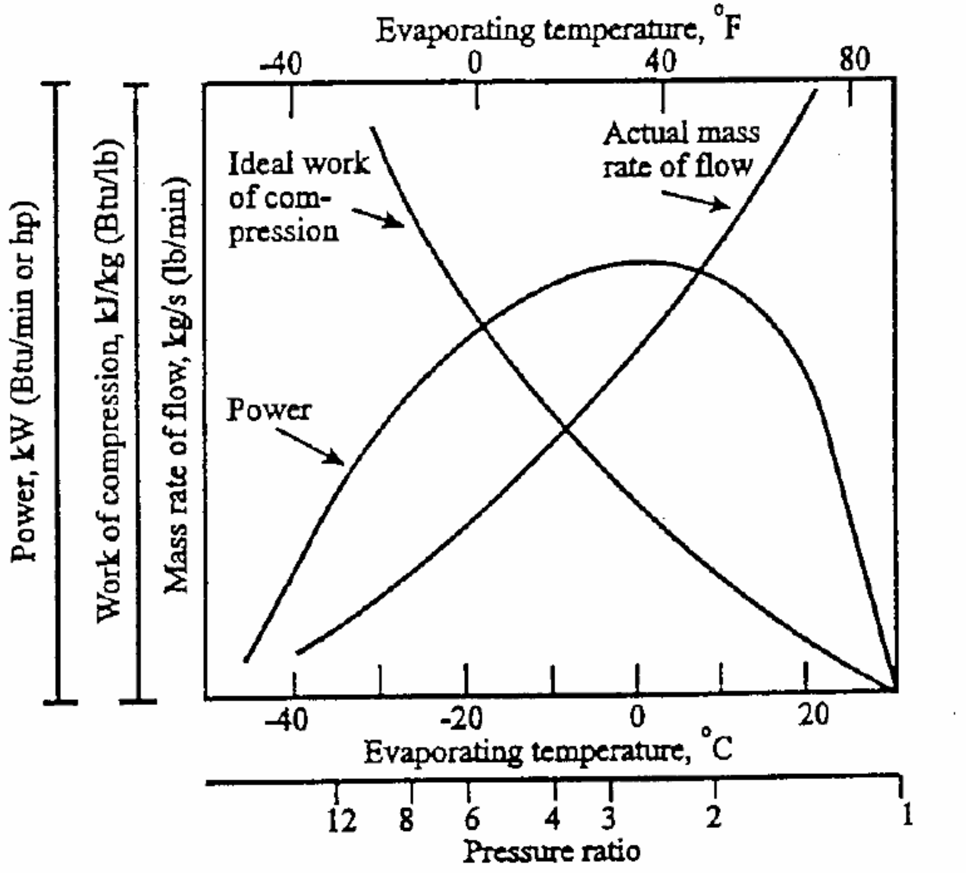


FIGURE 4.12
Effect of evaporating temperature on the mass rate of flow, the ideal work of compression and the compressor power requirement. The condensing temperature is 30°C (86°F).

Power Required by a Reciprocating Compressor

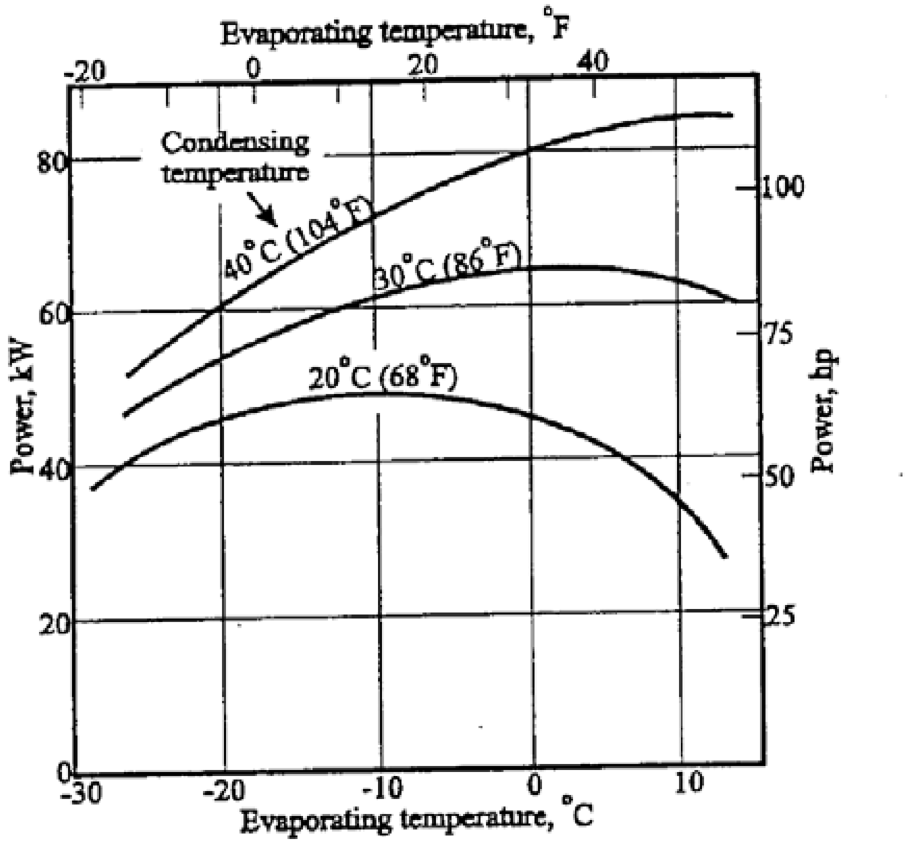


FIGURE 4.13
Actual power requirement of an 8-cylinder Sabroe 108L ammonia compressor operating at 1170 rpm.

Adiabatic Compression Efficiency

calculated from the equation:

$$P = \dot{m}\,\Delta h_{comp} \tag{4.11}$$

where P is the power required by the actual compressor.

The ratio of the ideal to the actual work of compression is defined as the adiabatic compression efficiency, η_c:

$$\eta_c = \frac{\Delta h_{ideal}}{\Delta h_{comp}} \tag{4.12}$$

Adiabatic Compression Efficiency

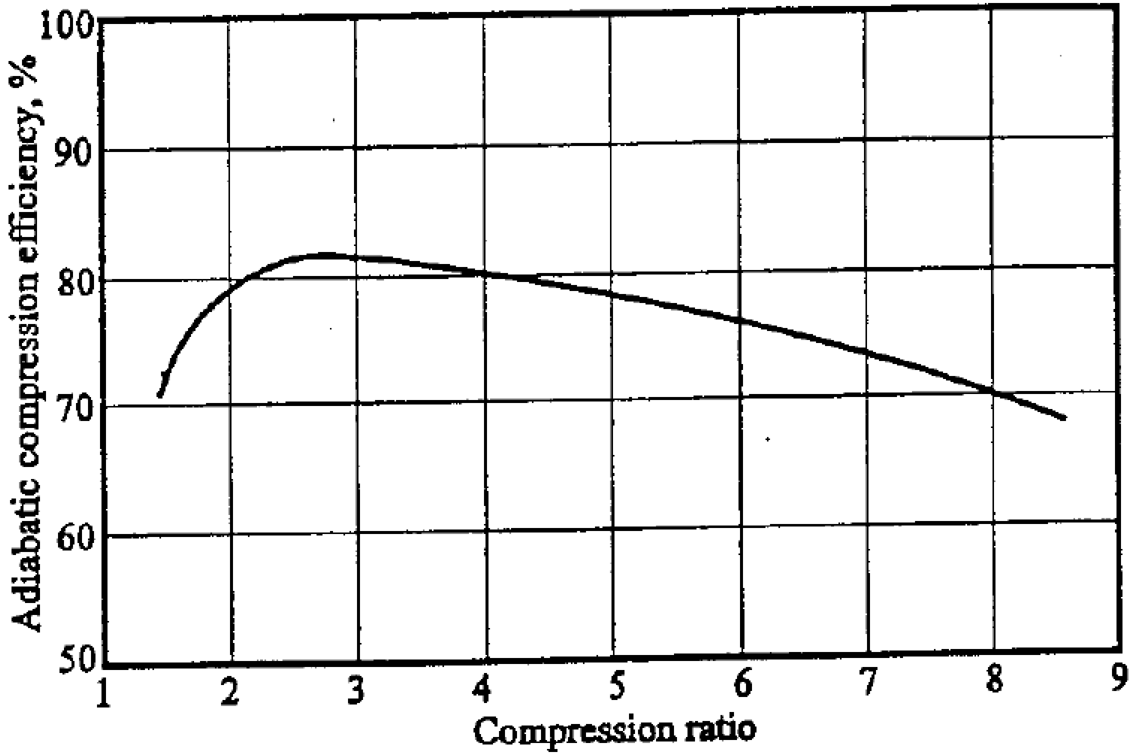


FIGURE 4.14
Adiabatic compression efficiency as a function of the compression ratio.

Effect of the Evaporating and Condensing Temperature on System Efficiency

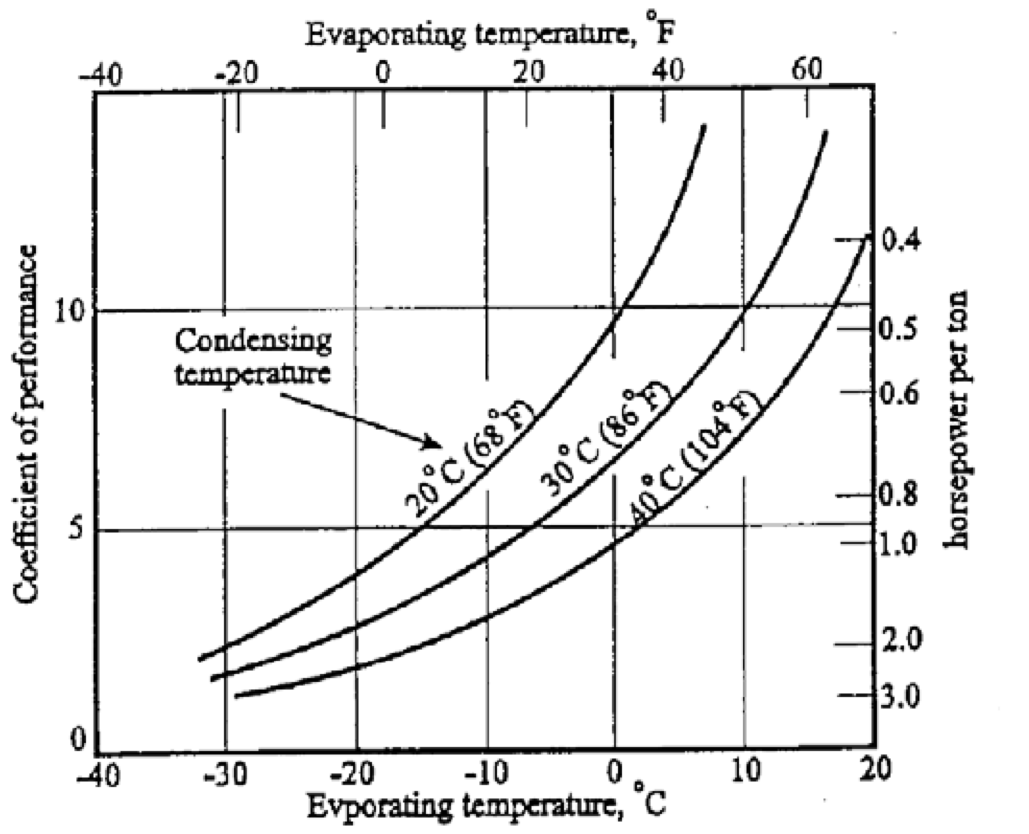


FIGURE 4.15
Coefficient of performance and horsepower per ton as a function of the evaporating and condensing temperatures.

Effect of the Evaporating and Condensing Temperature on System Efficiency

TABLE 4.1
Percent reduction in power per °C (°F).

	Small lift of temperature	Large lift of temperature
Increase of evaporating temperature	5^+% per °C 2.8^+% per °F	3^+% per °C 1.7^+% per °F
Decrease of condensing temperature	5^-% per °C 2.8^-% per °F	3^-% per °C 1.7^-% per °F

Effect of the Evaporating and Condensing Temperature on System Efficiency

TABLE 4.2

Reduction in compressor capacity caused by a drop in saturation temperature in the suction line.

Refrigerant	Evaporating temperature, °C (°F)			
	−20°C (−4°F)		−40°C (−40°F)	
	0.5°C drop	1.0°C drop	0.5°C drop	1.0°C drop
Ammonia	2.1%	4.2%	2.6%	5.2%
R-22	2.0%	3.9%	2.3%	4.6%

Effect of the Evaporating and Condensing Temperature on System Efficiency

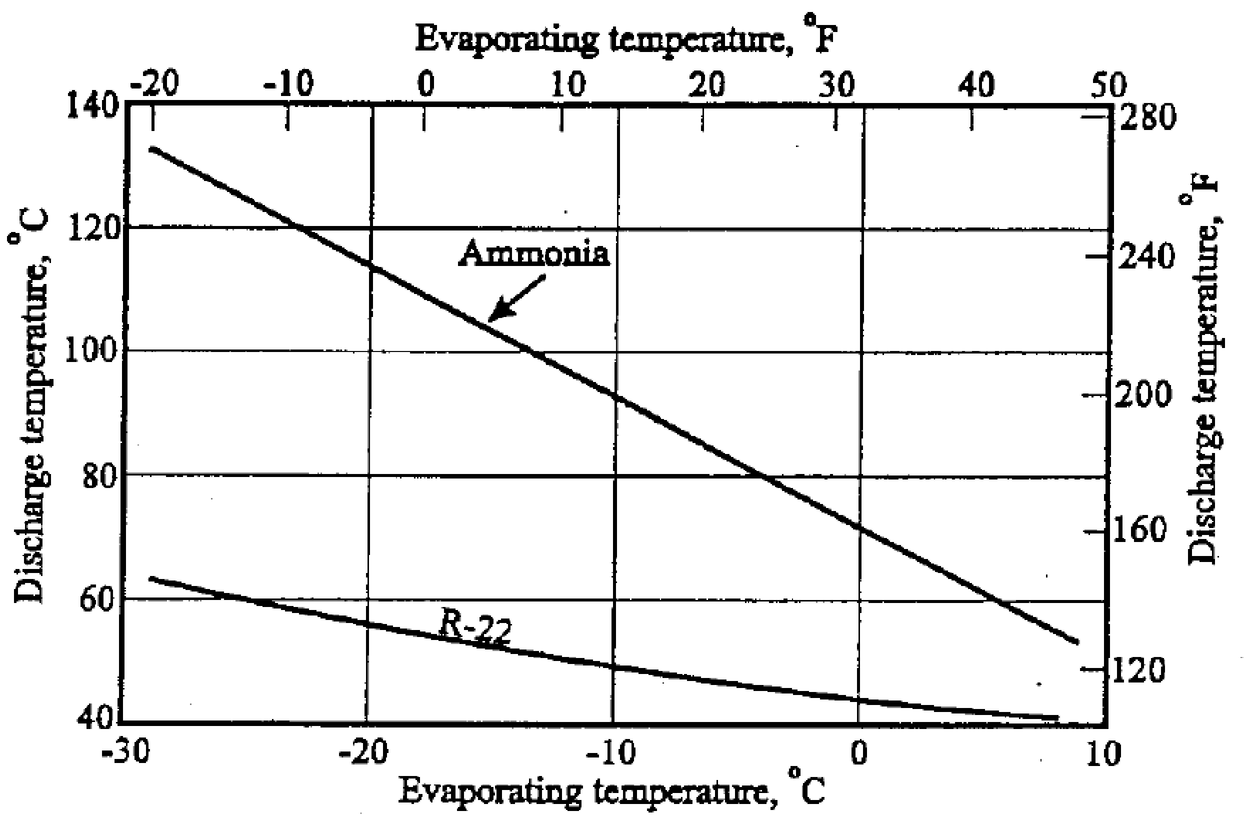


FIGURE 4.19
Discharge temperatures for ideal, adiabatic compressions from saturated vapor to a pressure corresponding to a condensing temperature of 30°C (86°F).

Effect of the Evaporating and Condensing Temperature on System Efficiency

TABLE 4.3

Effect of rotative speed on the volumetric and compression efficiencies for a compressor operating at 35°C (95°F) condensing and −10°C (14°F) evaporating temperatures.

	800 rpm	1200 rpm	1600 rpm
Volumetric efficiency	80%	80%	80%
Compression efficiency	80%	76%	72%

typically must be higher than about 100 kPa (15 psi). The cutout could be set to shut down the compressor after a 90-second duration of low pressure. This time delay permits the compressor a time interval to build up the oil pressure on startup.

Capacity Regulation

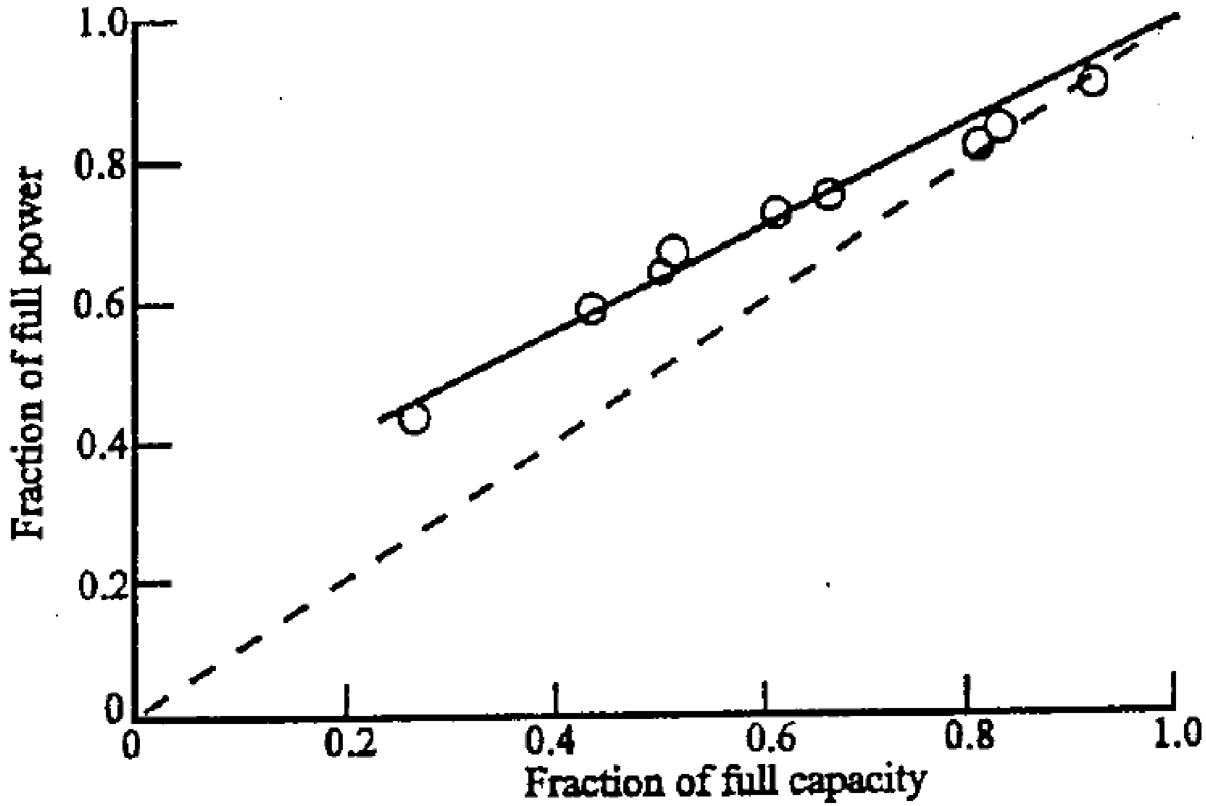
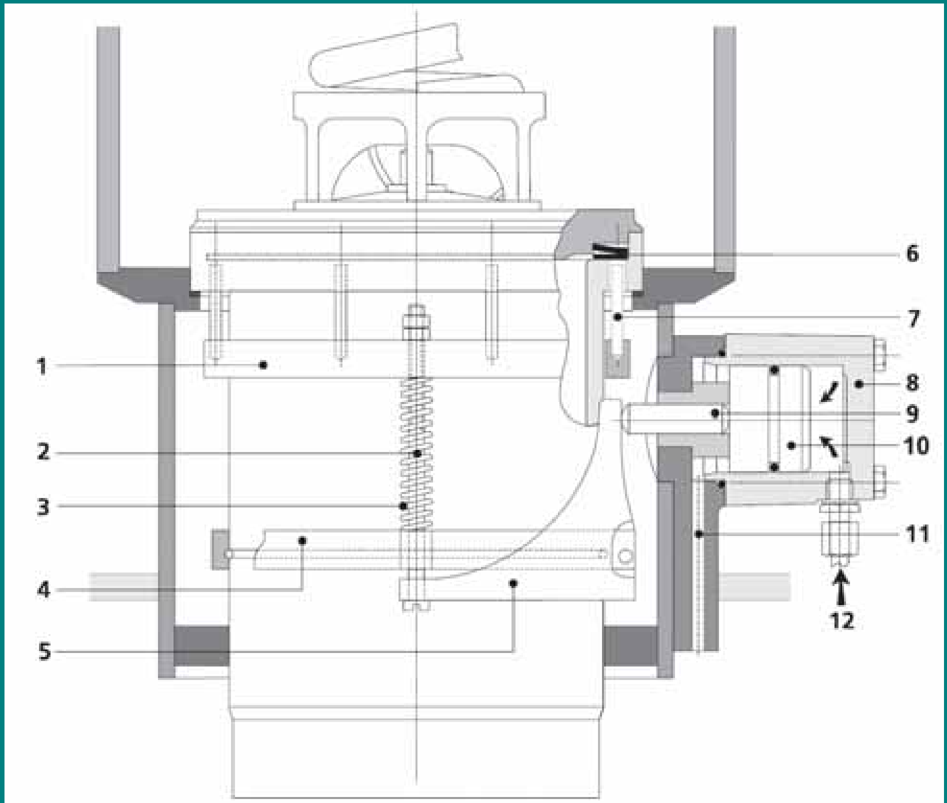


FIGURE 4.21
Power-capacity relationship of a 70-kW (20-ton) water chiller during cylinder unloading.



Legend	
1	Pressure ring
2	Tie bolt
3	Spring
4	Supporting ring
5	Semi-circular lever
6	Suction valve ring
7	Push pin
8	Piston housing
9	Stem
10	Piston
11	Bore for pressure equalizing
12	Control oil pressure

Screw compressors:

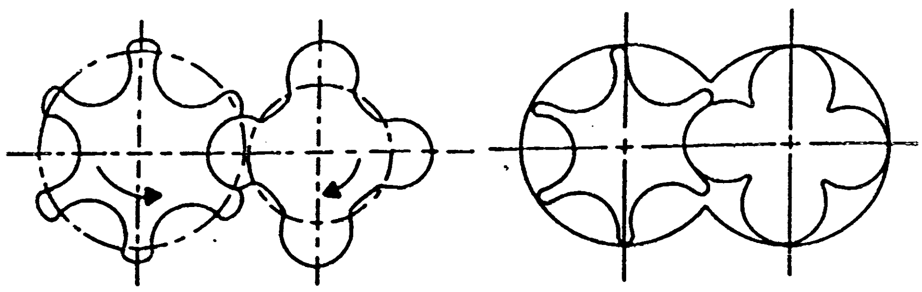
= [illegible] ~ [illegible]/s (100 ~ 33[illegible])

Power = 25 ~ [illegible] kW (33.5 hp ~ 1676 hp) 往復式 ~ [illegible]

RPM = 3550 rpm (2[illegible] rpm with [illegible] Hz) $\frac{P_{dis}}{P_{suc}}$ ≒ [illegible]

No. of lobe/gullies = [illegible] 3/5, 5/7, 5/6 ([illegible]鐘)

$\frac{P_{dis}}{P_{suc}}$ ≒ ~20



(a) (b)

FIGURE 5.1
Screw compressor rotors with *(a)* symmetric profile, and *(b)* asymmetric profile.

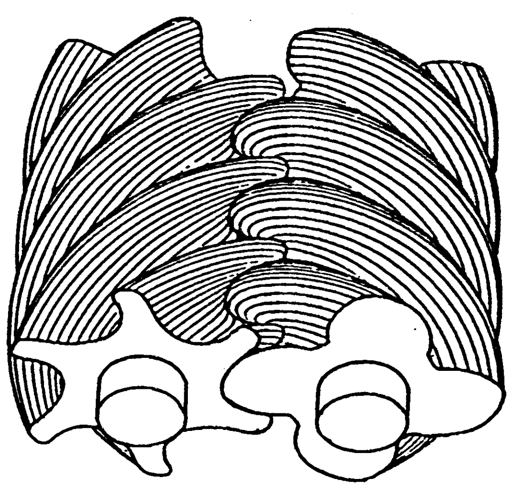


FIGURE 5.2
Screw compressor rotors.

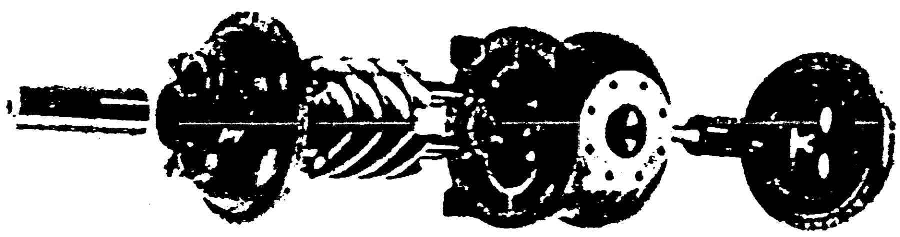


FIGURE 5.3
Exploded view of main elements of a screw compressor. *(Courtesy Sullair Refrigeration)*

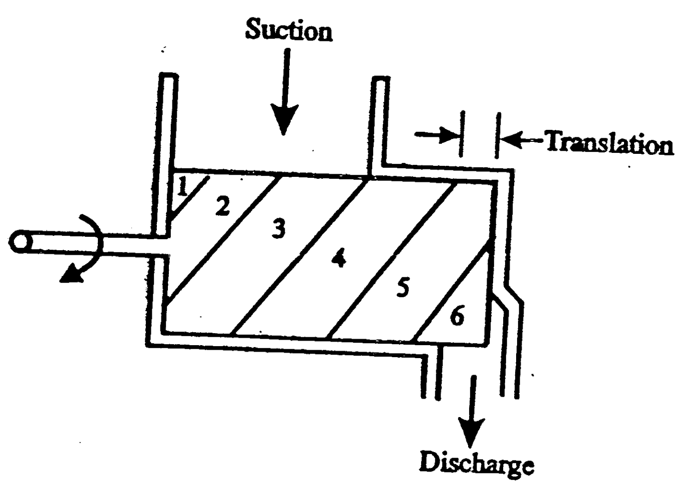


FIGURE 5.4
Visualization of the intake, compression and discharge processes of a screw compressor.

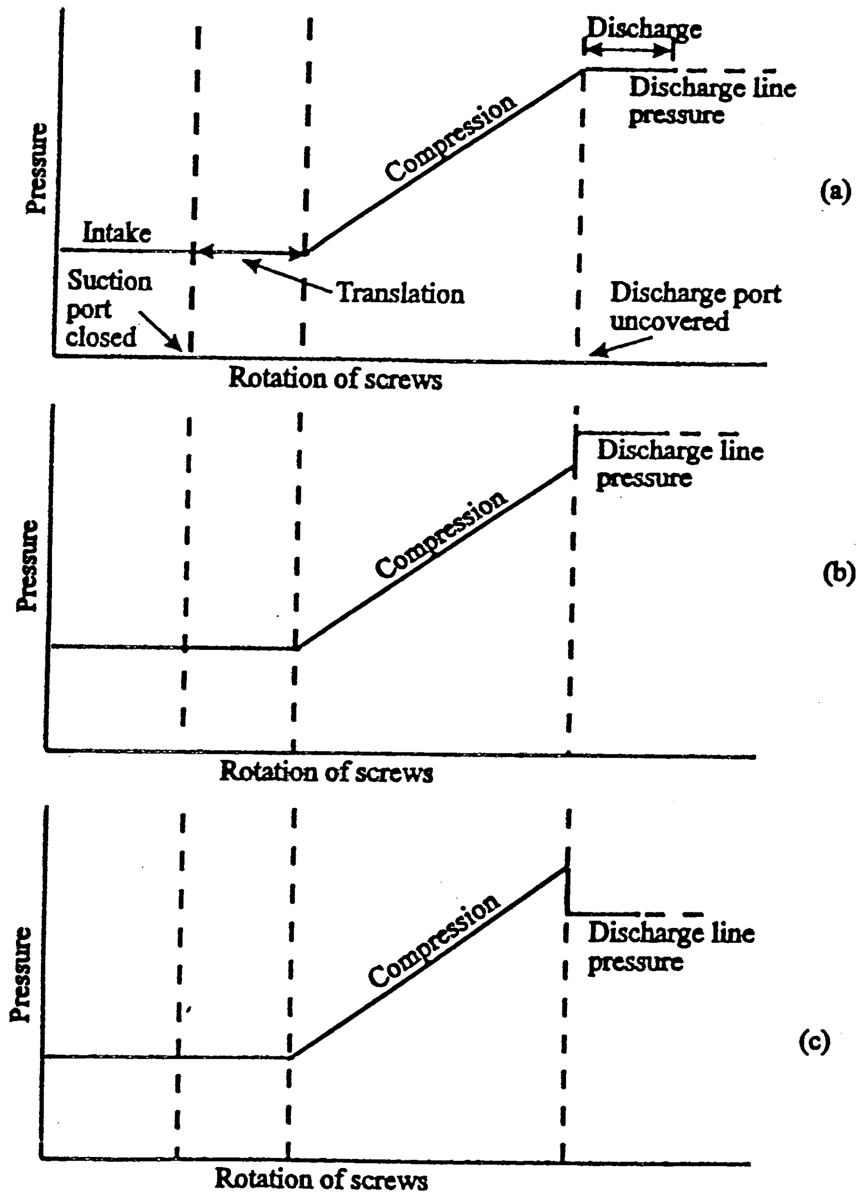


FIGURE 5.5
Pressures during intake, translation, compression, and discharge when (a) the discharge-line pressure equals, (b) when the discharge-line pressure is higher, and (c) the discharge-line pressure is lower than the built-in discharge pressure.

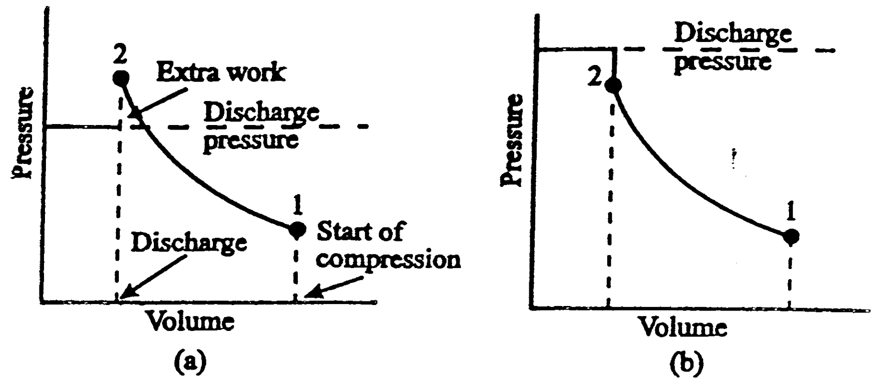


FIGURE 5.6
(a) Over-compression and (b) under-compression shown on a pressure-volume diagram where the area under the curves indicate work applied to the refrigerant.

$$v_i = \frac{\text{volume in cavity when suction port closes}}{\text{volume in cavity when discharge port uncovers}}$$

$$\text{Pressure ratio} = \left(\frac{\text{suction volume}}{\text{discharge volume}}\right)^k = v_i^k$$

where k = ratio of specific heats, c_p/c_v, which is approximately 1.29 for ammonia and 1.18 for R-22.

TABLE 5.1
Pressure ratios corresponding to built-in volume ratios for ideal compression.

Built-in volume ratio	Ammonia	R-22
2.6	3.4	3.1
3.6	5.3	4.5
4.2	6.4	8.9
5.0	8.0	6.7

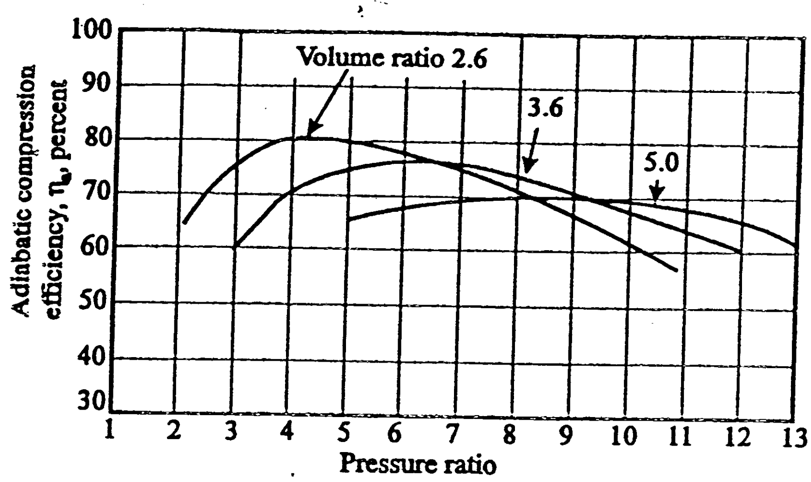


FIGURE 5.7
Adiabatic compression efficiency of ammonia screw compressors.

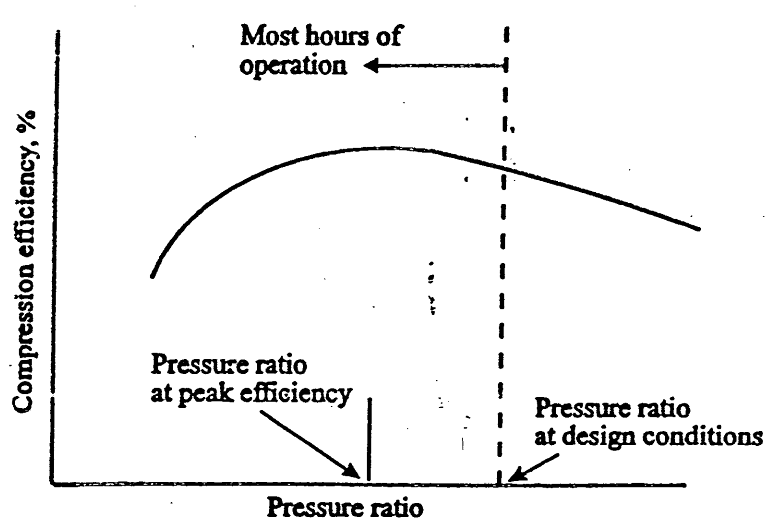


FIGURE 5.8
Selecting a compressor with its peak efficiency occurring lower than the design pressure ratio.

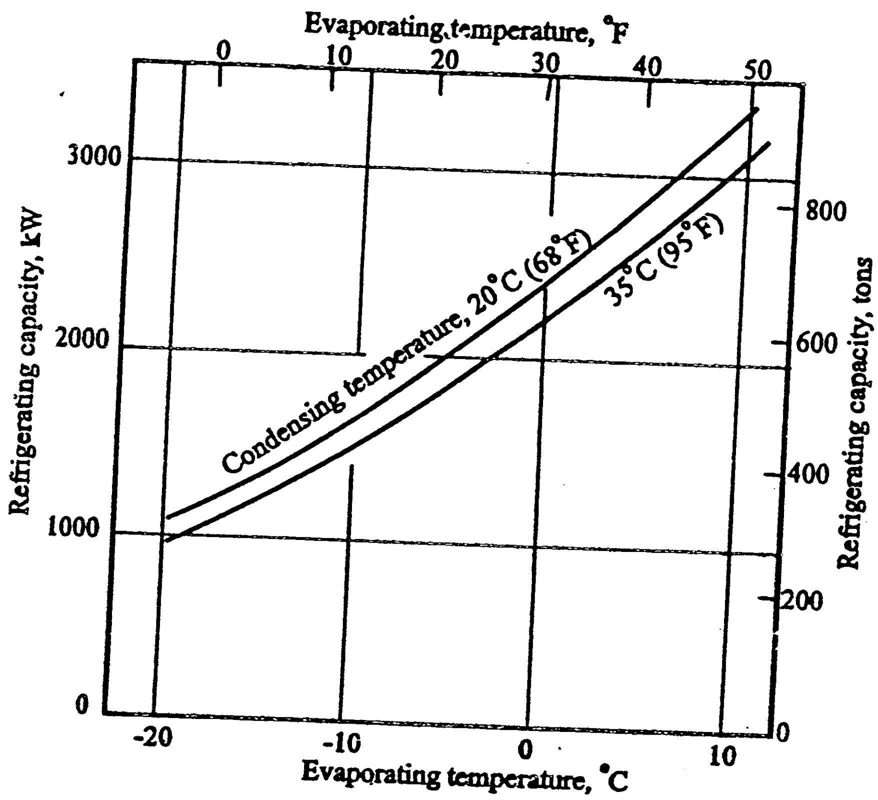


FIGURE 5.9
Effect of evaporating and condensing temperatures on the refrigerating capacity of an ammonia screw compressor. (*Model RWB-II 222, Frick Company*)

TABLE 5.2
Comparison of refrigerating capacity and power of screw and reciprocating compressor with changes in evaporating and condensing temperatures. Values shown are percentages referred to the base evaporating temperature of 5°C (41°F).

Evaporating temperature, °C (°F)	Capacity, kW (tons)				Power, kW (hp)			
	Condensing temperature				Condensing temperature			
	20°C (68°F)		35°C (95°F)		20°C (68°F)		35°C (95°F)	
	Recip.	Screw	Recip.	Screw	Recip.	Screw	Recip.	Screw
5 (41)	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
0 (32)	0.838	0.834	0.714	0.835	1.02	1.05	0.972	1.01
−10 (14)	0.572	0.564	0.481	0.565	0.986	1.09	0.872	0.980
−20 (−4)	0.38	0.375	0.321	0.371	0.848	1.09	0.710	0.877

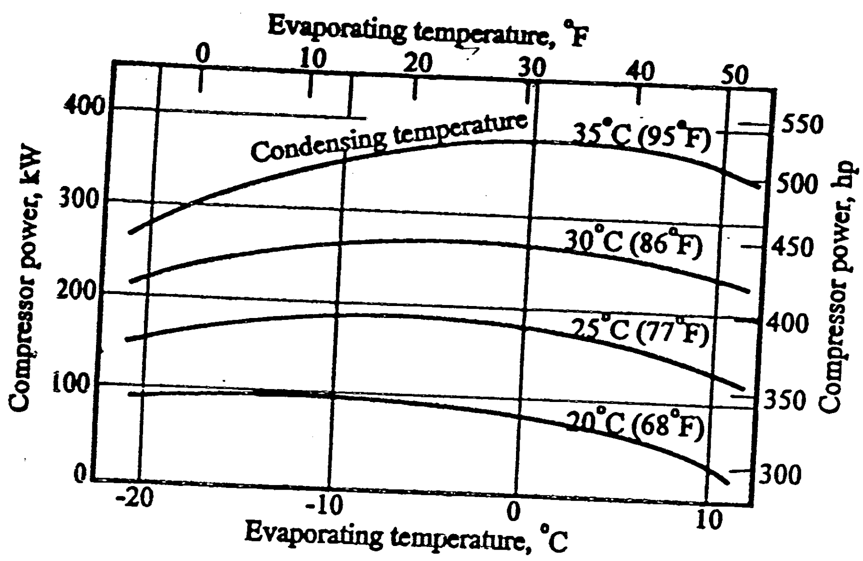


FIGURE 5.10
Effect of evaporating and condensing temperatures on the power requirement of an ammonia screw compressor. (*Model RWB-II 222, Frick Company*)

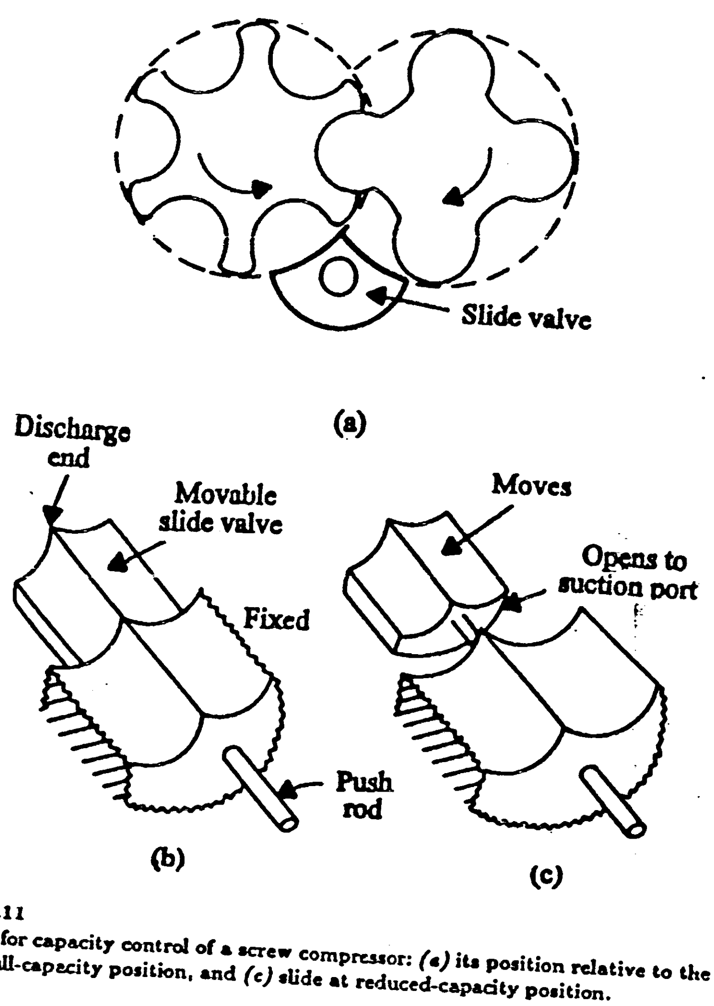


FIGURE 5.11
A slide valve for capacity control of a screw compressor: (*a*) its position relative to the rotors, (*b*) slide at full-capacity position, and (*c*) slide at reduced-capacity position.

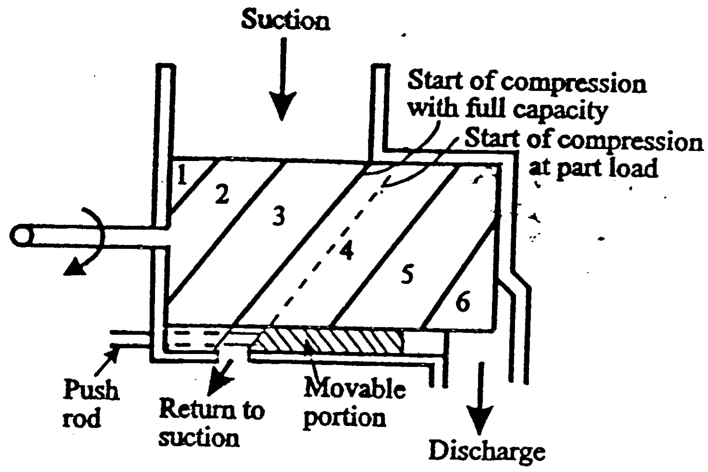


FIGURE 5.12
Side view of the function of the slide valve at *(a)* full capacity, and *(b)* partial capacity.

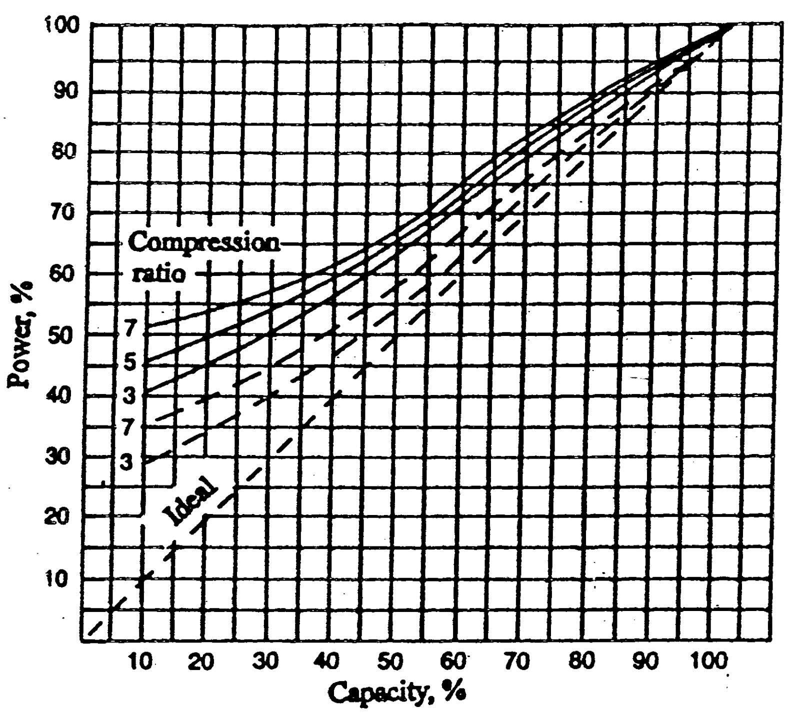


FIGURE 5.13
Part-load power requirements of a screw compressor. The solid lines apply to constant condensing and evaporating temperatures, while the dashed lines reflect a drop in condensing temperature and increase in evaporating temperature at part load.

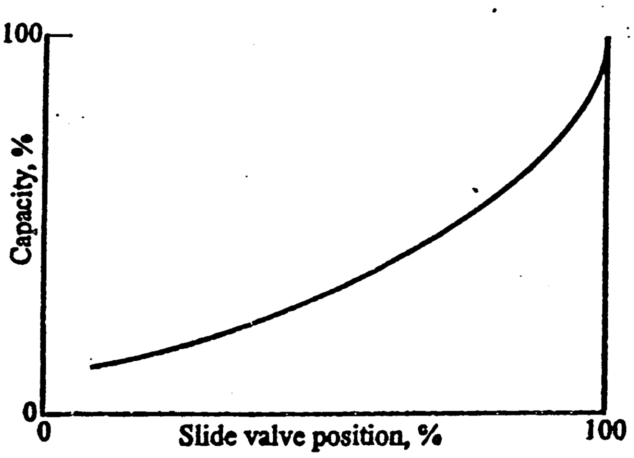


FIGURE 5.14
Variation in the compressor capacity as a function of the slide valve position

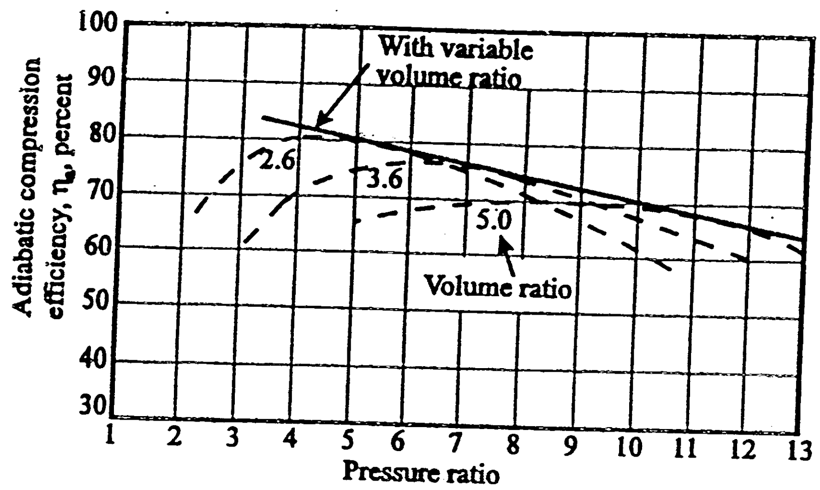


FIGURE 5.16
Maintaining peak compression efficiency with a variable-volume ratio device during changes in the pressure ratio.

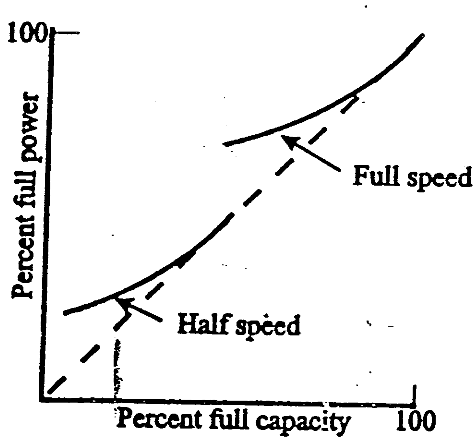


FIGURE 5.15
The power-capacity curve of a screw compressor driven by a two-speed motor.

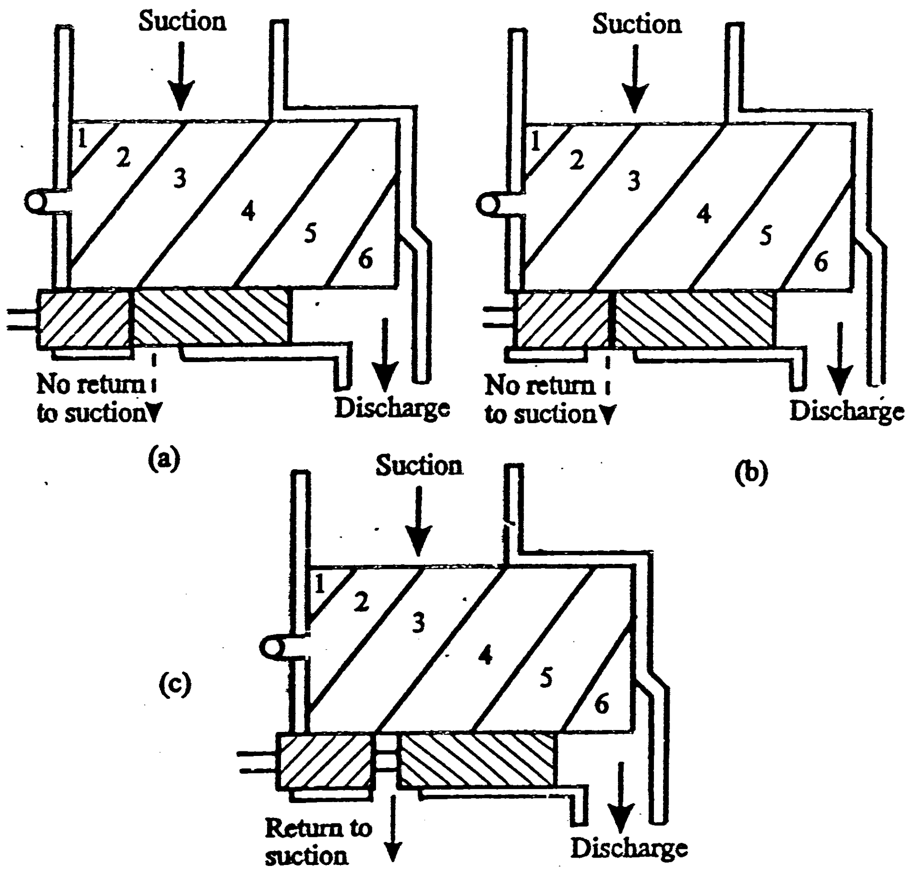
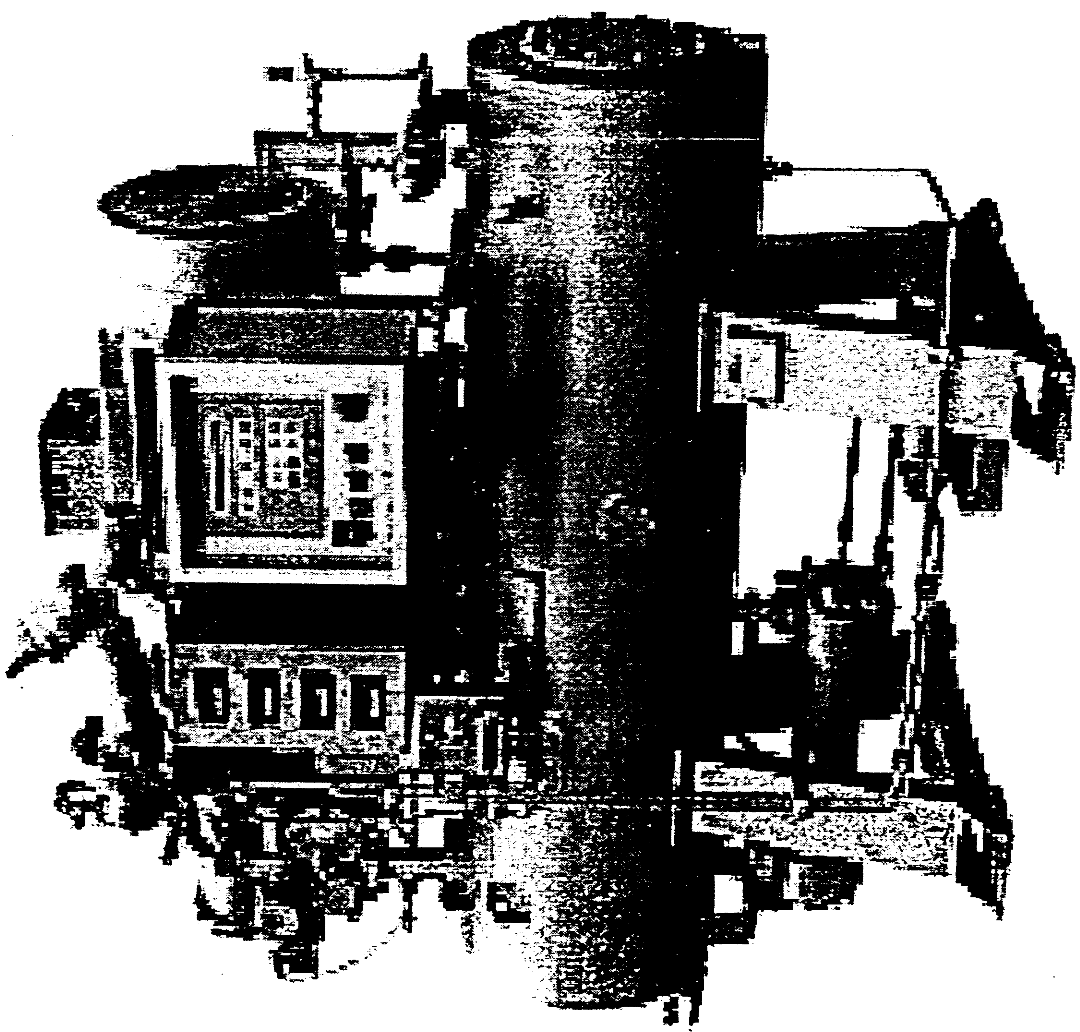


FIGURE 5.17
A variable v_i device at the following operating conditions: (a) full load and low v_i, (b) full load and high v_i, and (c) part load and high v_i.



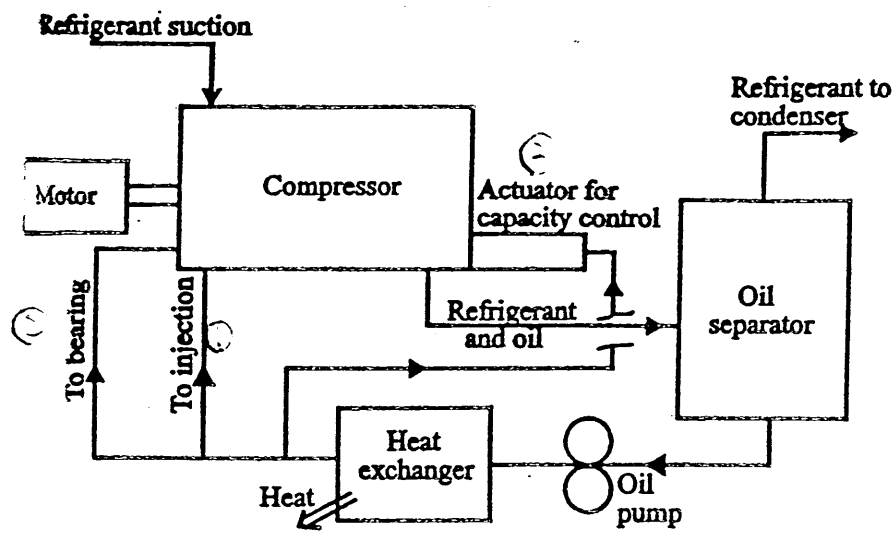


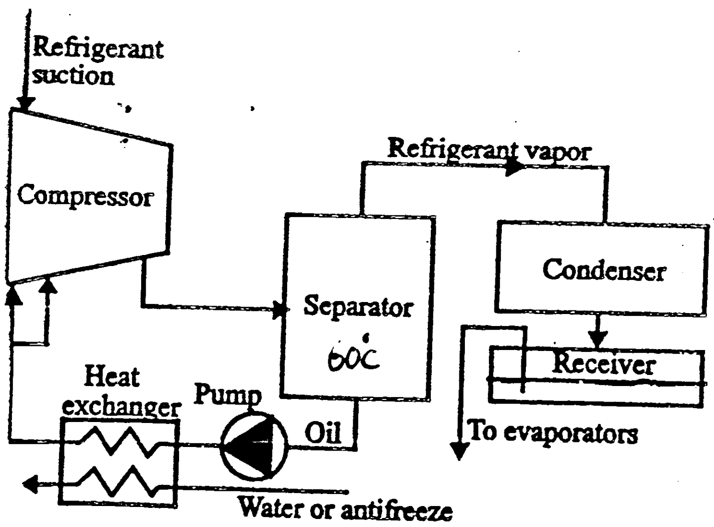
FIGURE 5.18
Flow and distribution of oil serving a screw compressor.

The screw compressor is provided with oil to serve three purposes:

(1) sealing of internal clearances

(2) lubrication of bearings

(3) actuation of the slide valve.



<I.>

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

<II>

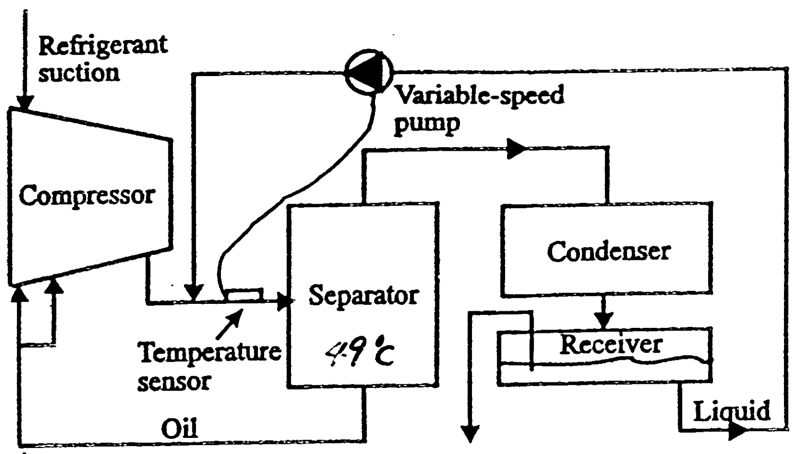


FIGURE 5.20
Cooling oil by injection of liquid into the compressor discharge line.

<III>

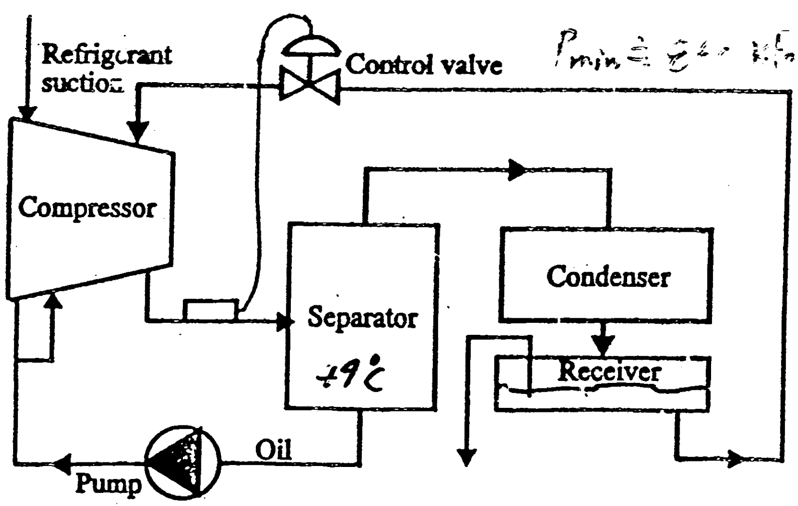


FIGURE 5.21
Cooling oil by direct injection of liquid refrigerant at an early stage of the compression process.

Advantage : the lowest

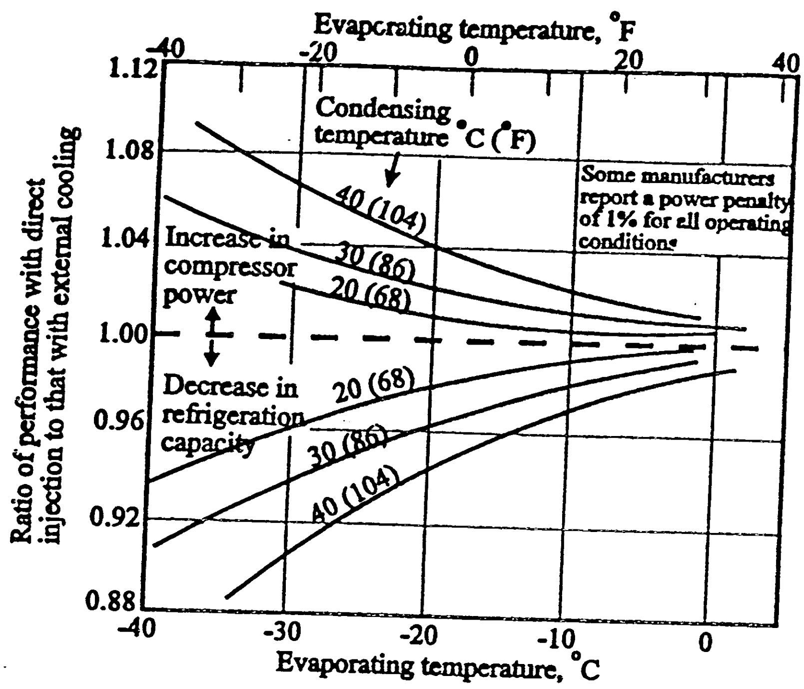
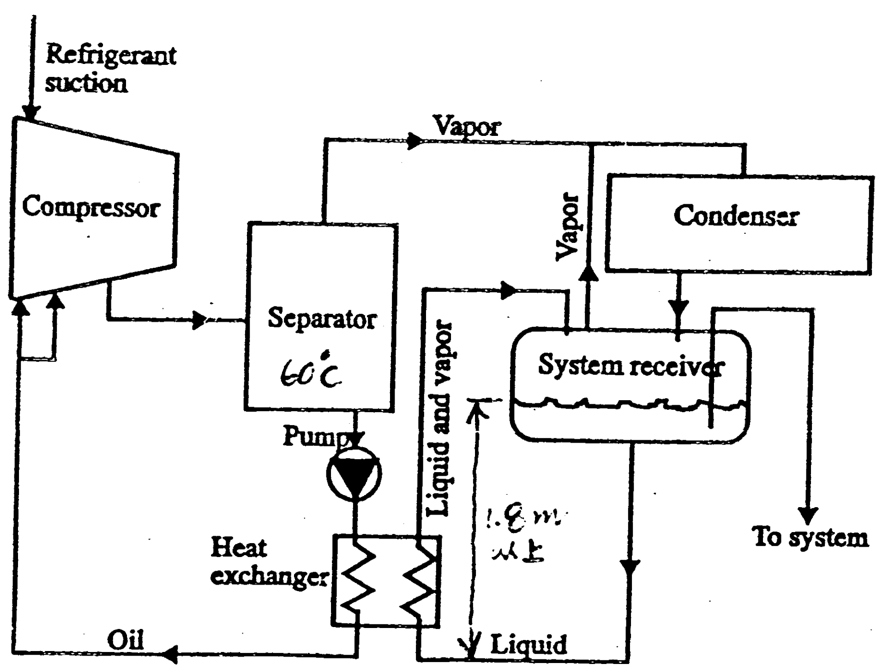


FIGURE 5.22
Penalties in refrigeration capacity and power requirement for ammonia screw compressors provided with direct-injection oil cooling.



< IV. >

FIGURE 5.23
A thermosiphon oil cooling installation where the level of the system receiver is above the level of the heat exchanger.

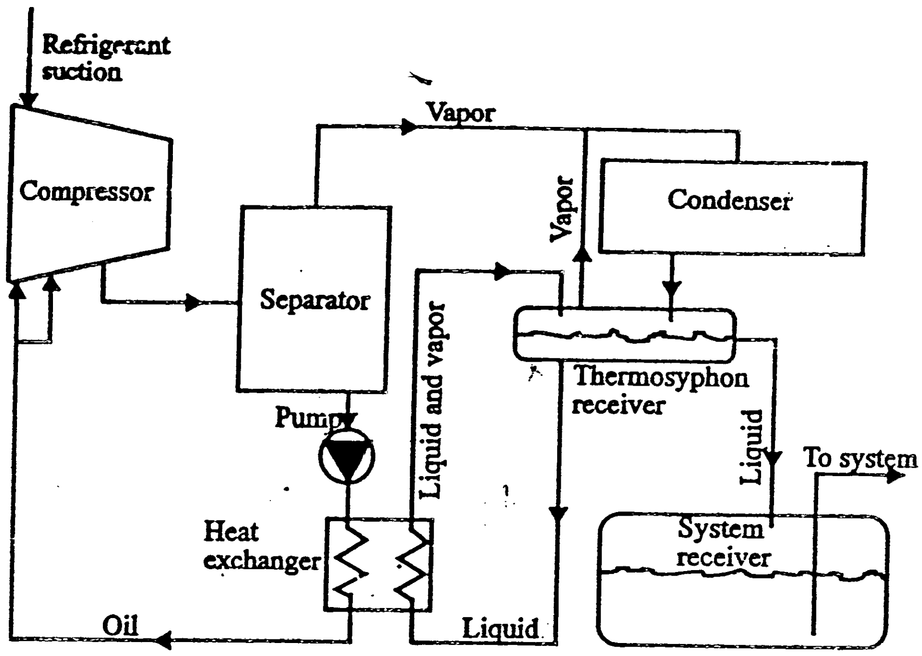


FIGURE 5.24
A thermosiphon oil cooling installation where the level of the system receiver is at or below the level of the heat exchanger, requiring an additional receiver.

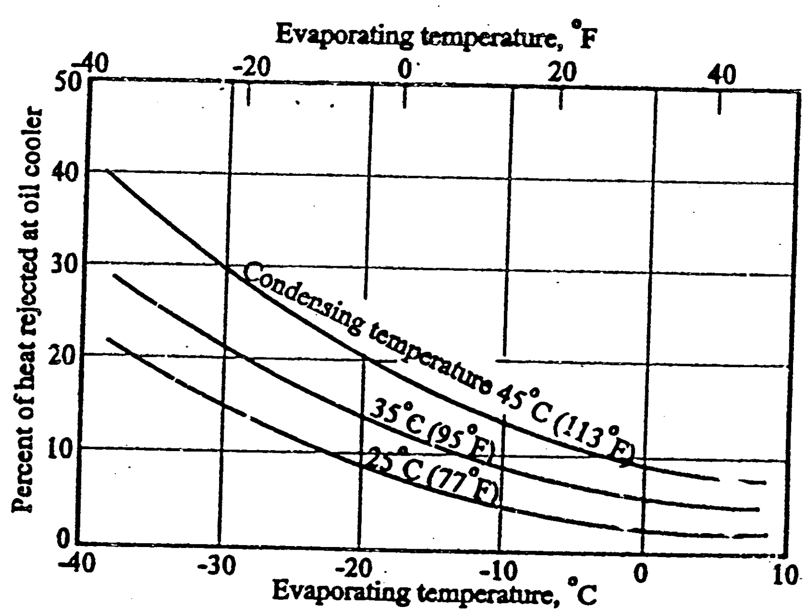


FIGURE 5.25
Percentage of heat input (total of refrigeration load and compressor power) that is absorbed by the injected oil in a screw compressor.

2-27

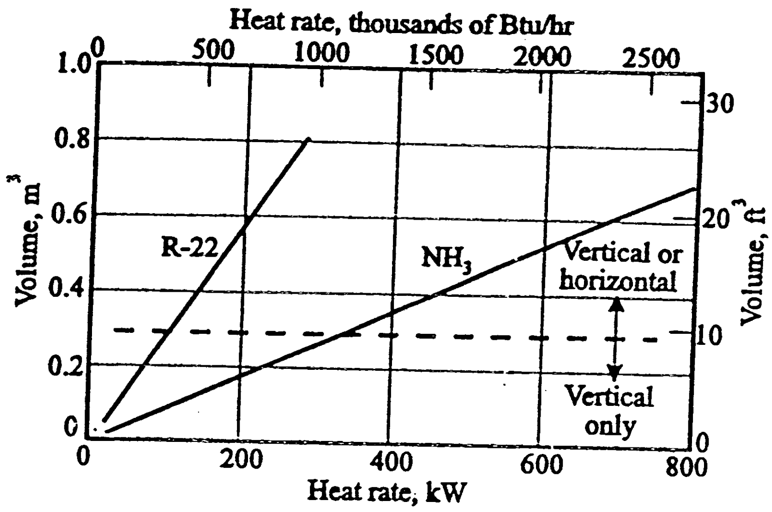


FIGURE 5.26
Volume of the thermosiphon receiver as a function of the heat rate of the oil cooling heat exchanger.

TAPLE 5.3
Flow-rate carrying capacities of various line sizes in the vent pipe between the receiver and the condenser.

Pipe size, inch	Ammonia		R- 22	
	kg/s	lb/min	kg/s	lb/min
1-1/2	0.0529	7	0.121	16
2	0.907	12	0.219	29
2-1/2	0.166	22	0.378	50
3	0.295	39	0.680	90
4	0.529	70	1.17	155
5	0.907	120	2.12	280
6	1.66	220	3.63	480
8	5.29	700	7.56	1000

(1) Determine the heat rejection rate at the oil cooler,

$q_{oc} = (q_{tot})$(percent from Fig. 5-25)/100

(2) Compute $\dot{m}_{ev}$

$$\dot{m}_{ev} = \frac{q_{oc}}{h_g - h_f}$$

(3) $\dot{m}_{oc} = 2\,\dot{m}_{ev}$ for R-22, and $\dot{m}_{oc} = 3\,\dot{m}_{ev}$ for NH_3

(4) Volume of receiver $= \dfrac{2 \times (5\,\text{min})(\dot{m}_{ev})}{\rho_l}$

(5) Liquid line from receiver to the HX.

For NH_3, in $= 0.472 (\dot{m}_{oc}, lb/min)^{0.37}$ 22.6 Pa/m

$= 2.88 (\dot{m}_{oc}, kg/s)^{0.37}$

For R22, in $= 0.350 (\dot{m}_{oc}, lb/min)^{0.37}$ 113 Pa/m

$= 2.13 (\dot{m}_{oc}, kg/s)^{0.37}$

(6) Liquid/vapor line from heat exchanger to thermosiphon receiver

For NH_3

$D_{,in} = 0.572 (\dot{m}_{oc}, lb/min)^{0.37}$ 9.04 Pa/m

$= 3.49 (\dot{m}_{oc}, kg/s)^{0.37}$

For R22

$D_{,in} = 0.423 (\dot{m}_{oc}, lb/min)^{0.37}$

$= 2.58 (\dot{m}_{oc}, kg/s)^{0.37}$ 45.2 Pa/m

(7) Difference in elevation from the receiver to the H.X. $\gtrsim 1.8$ m

Vapor line from receiver to condenser

Example 5.1. Design the thermosiphon oil-cooling system serving an ammonia screw compressor operating with an evaporating temperature of −20°C (−4°F) and a condensing temperature of 35°C (95°F). The full- load refrigerating capacity and power requirement at these conditions are 1025 kW (291.4 tons of refrigeration) and 342 kW (458.5 hp), respectively.

Solution. The combined refrigeration and power input is:

$$1025 \text{ kW} + 342 \text{ kW} = 1367 \text{ kW}$$

or

$$(291.4 \text{ tons})(12{,}000) + (458.5 \text{ hp})(2545) = 4{,}664{,}000 \text{ Btu/hr}$$

Figure 5.25 indicates that at the prevailing evaporating and condensing temperatures, 14% of the total energy input is absorbed by the oil:

$$q_{oc} = (1367 \text{ kW})(0.14) = 191.4 \text{ kW } (653{,}000 \text{ Btu/hr})$$

As a preliminary step, compute the evaporation rate of ammonia,

$$\dot{m}_{ev} = \frac{q_{oc}}{h_g - h_f} = \frac{191.4 \text{ kW}}{1124 \text{ kJ/kg}} = 0.1703 \text{ kg/s}$$

$$\text{or} \quad \dot{m}_{ev} = \frac{653{,}000 \text{ Btu/hr}}{(483.2 \text{ Btu/lb})(60 \text{ min/hr})} = 22.52 \text{ lb/min}$$

Designing for a circulation ratio of 3, which is typical for ammonia,

$$\dot{m}_{oc} = 3\dot{m}_{ev} = 0.511 \text{ kg/s } (67.6 \text{ lb/min})$$

Thermosiphon receiver. If one-half the receiver should be able to contain a five-minute evaporation rate, the volume of the receiver, V_{rec} is

$$V_{rec} = \frac{2(5 \text{ min})(60 \text{ sec/min})(0.1703 \text{ kg/s})}{(\rho_{liquid} = 587.6 \text{ kg/m}^3)} = 0.174 \text{ m}^3 \ (6.31 \text{ ft}^3)$$

a size which corresponds to Fig. 5.26.

Choosing a length L of 1.83 m (6 ft) in the equation for volume $\pi(D^2/4)L$,

$$D = \sqrt{\frac{(0.174 \text{ m}^3)(4)}{\pi(1.83)}} = 0.348 \text{ m } (13.7 \text{ in})$$

Choose the next largest diameter which is 16 in (0.4064 m).

Line size from receiver to heat exchanger. Applying Eq. 5.4 to the flow rate of $\dot{m}_{oc}$,

$$D = 0.472(67.6 \text{ lb/min})^{0.37} = 2.88(0.511 \text{ kg/s})^{0.37} = 2.24\text{in}$$

Choose a 2-1/2 in size.

Liquid/vapor line size between the heat exchanger and receiver. Using Eq. 5.6,

$$D = 0.572(67.6 \text{ lb/min})^{0.37} = 3.49(0.511 \text{ kg/s})^{0.37} = 2.72 \text{ in}$$

Choose a 3-inch line.

Vapor line from receiver to condenser. Entering Table 5.3 with the evaporation rate of 0.1703 kg/s (22.52 lb/min), we find that a 2-1/2-inch size would almost be adequate, but choose a 3-inch line to minimze the pressure drop.

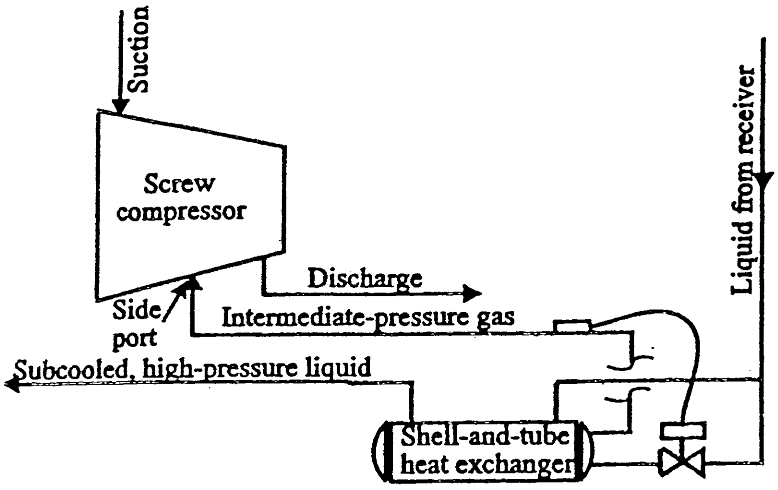


FIGURE 5.27
Using the side port of a screw compressor to provide a two-stage benefit.

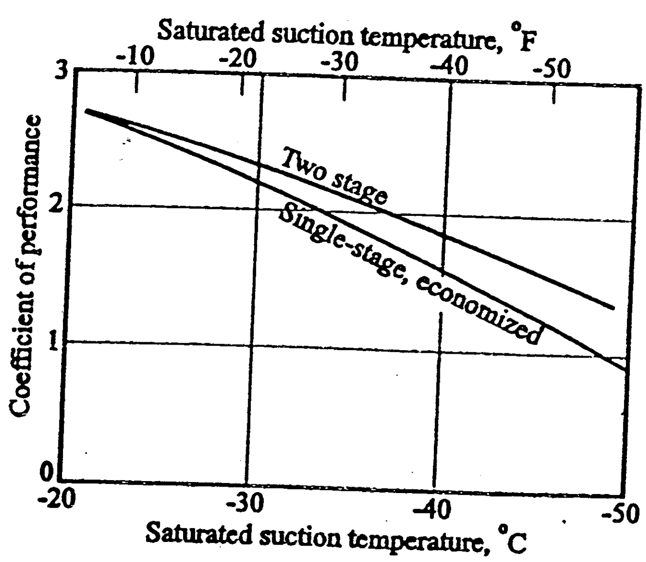


FIGURE 5.28
Comparison of the coefficients of performance of a two-stage ammonia system with an economized single-stage compressor equipped with a flash-type subcooler.

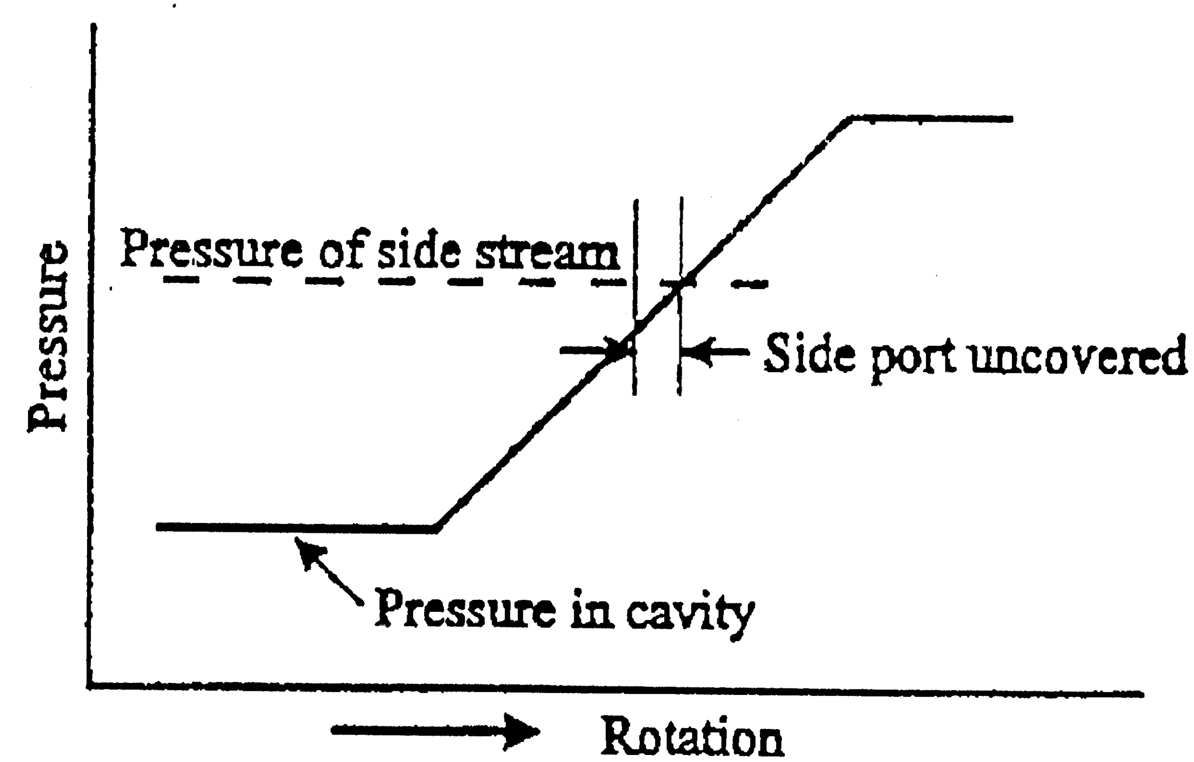


FIGURE 5.29
Unrestrained expansion of side-port gas during the admission into the compressor.

the optimum intermediate pressure no longer prevails

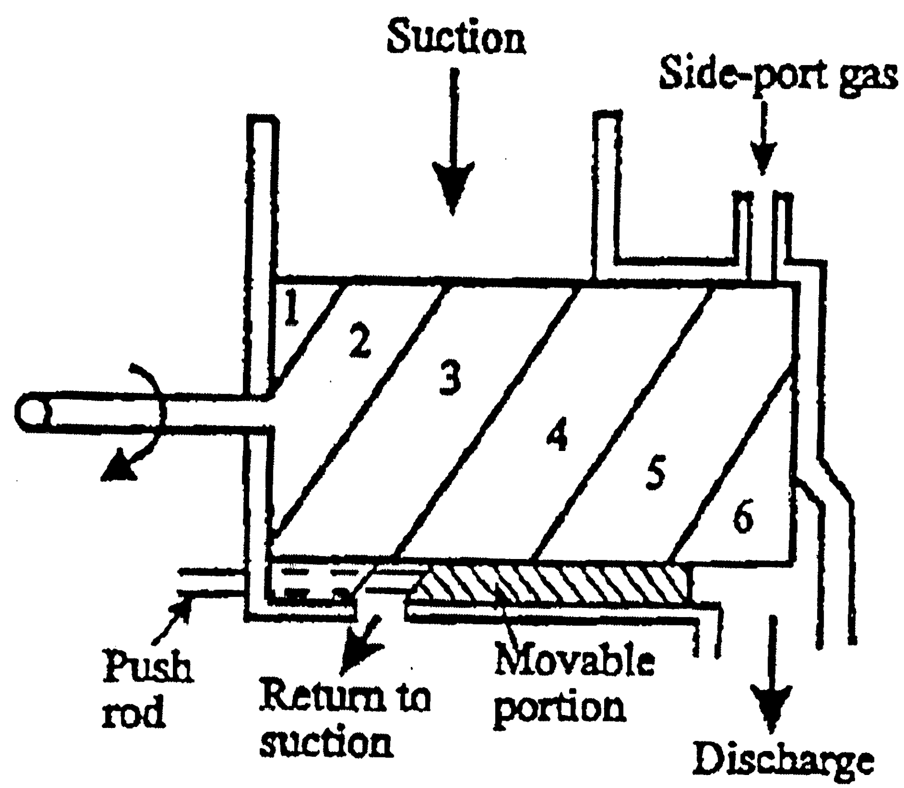


FIGURE 5.31
Opening of the slide valve drops the side-port pressure.

2-31-2

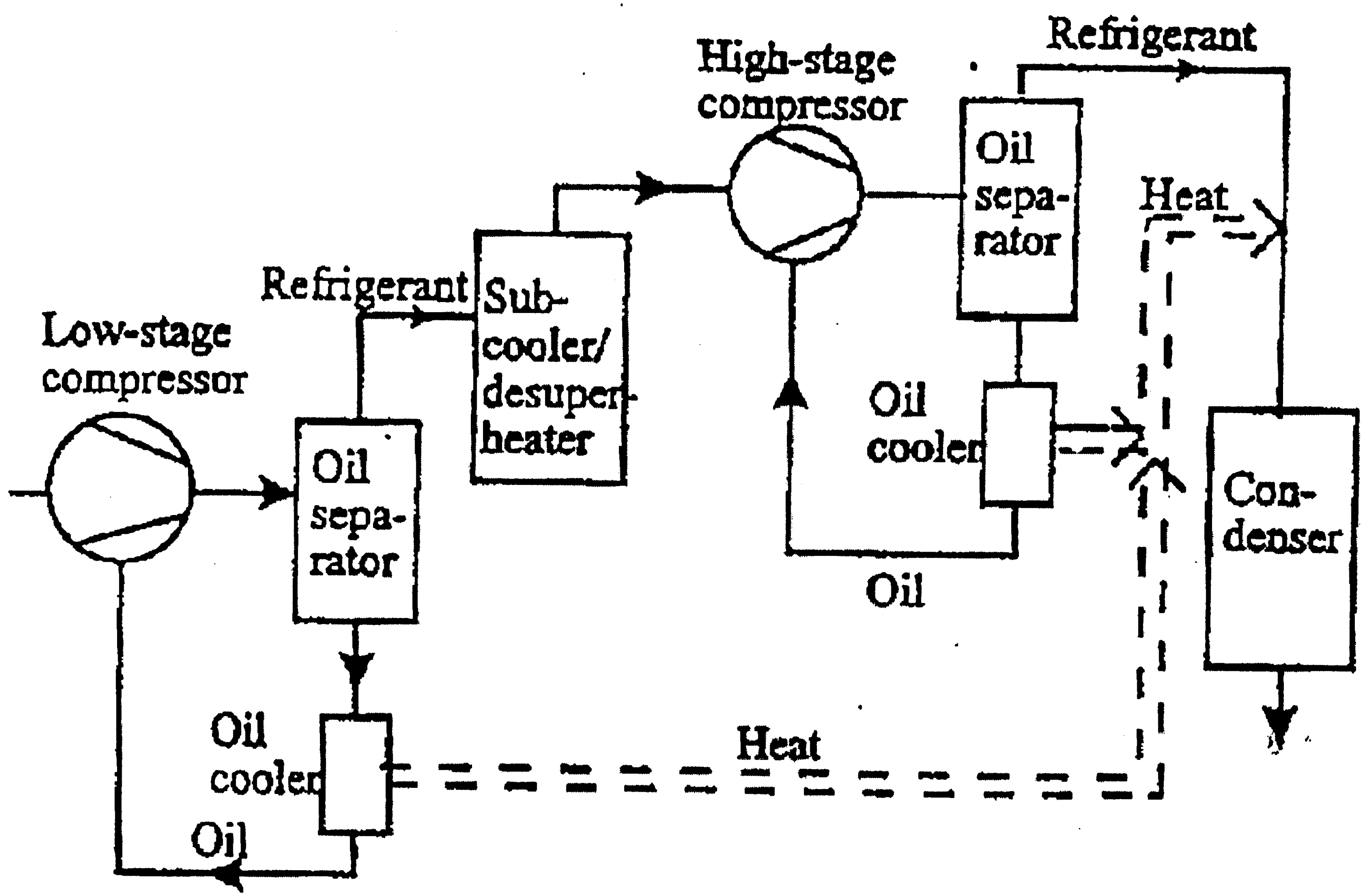


FIGURE 5.32
Transfer of heat from the low-stage oil cooler directly to the condenser when using a thermosyphon or external liquid oil cooler.

2-31-3

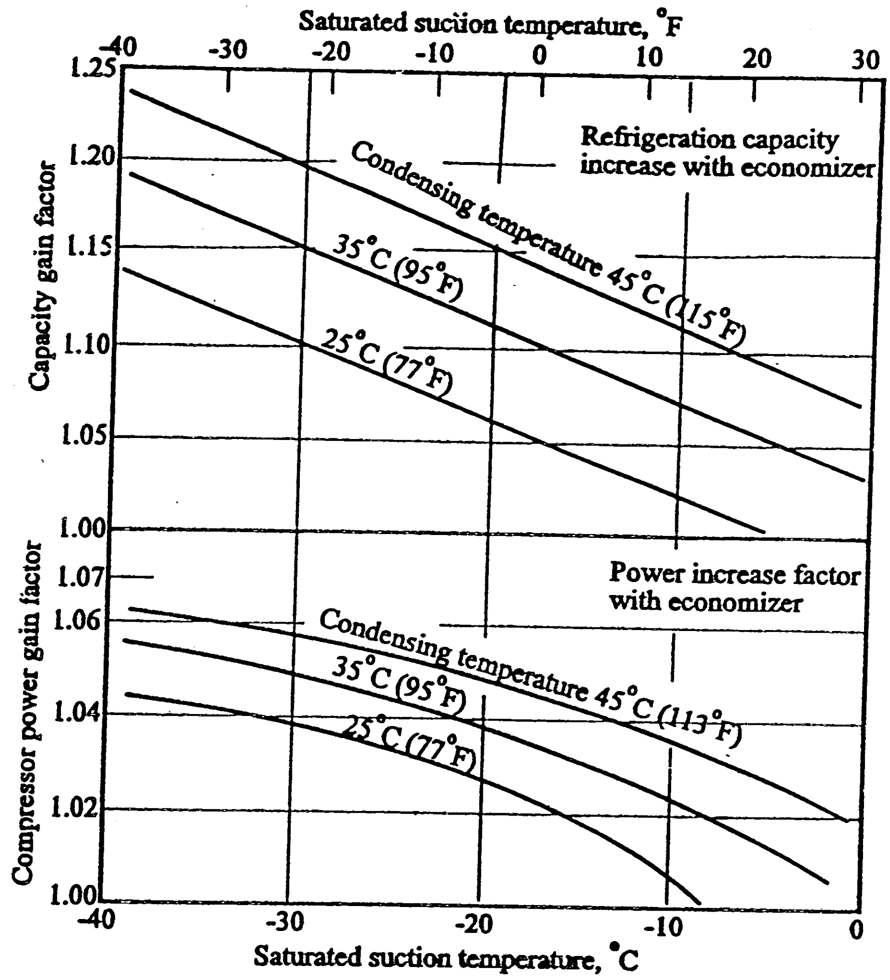
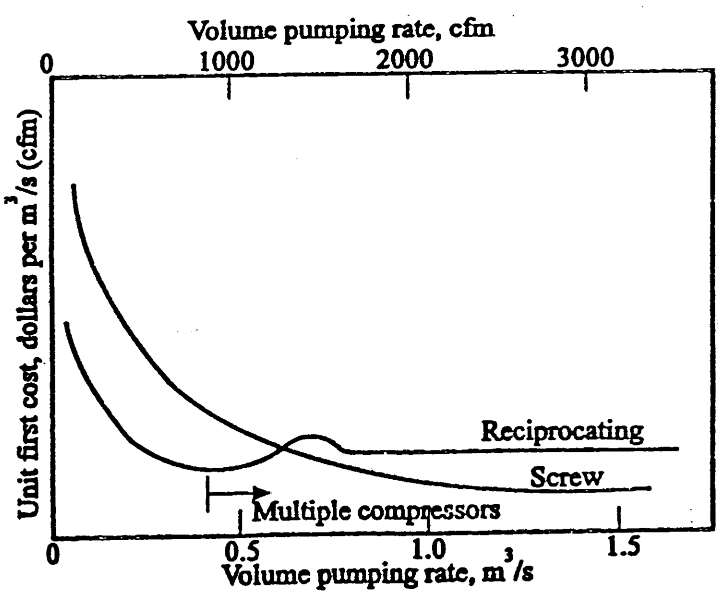


FIGURE 5.30
Multiplying factors for the refrigerating capacity and the power requirement in an ammonia system when operating with an economized cycle.



The [illegible] comp. as the Low-Stage comp.

FIGURE 5.34
Comparative first costs of reciprocating and screw compressors.

Refrigeration Engineering

-Refrigerant

李魁鵬

■ 何謂冷媒

冷凍空調系統中，用以傳遞熱能，產生冷凍效果之工作流體。依工作方式分類可分為一次(Primary)冷媒與二次(Secondary)冷媒。依物質屬性分類可分為自然(Natural)冷媒與合成(Synthetic)冷媒。

理想的冷媒條件

1. 無毒
2. 不爆炸
3. 對金屬及非金屬無腐蝕作用
4. 不燃燒
5. 洩漏時易於察覺
6. 化學性安定
7. 對潤滑油無破壞性
8. 具有較的蒸發潛熱
9. 對環境無害

■ 理想的冷媒物理特性

1. 蒸發壓力要高

蒸發溫度會隨應用溫度而變化，例如冰水機之蒸發溫度約爲0~5℃，冷凍庫主機之蒸發溫度約爲-20 ~ -30℃，家用空調機之蒸發溫度約爲5~10℃。蒸發溫度愈低，蒸發壓力亦愈低，若冷媒之蒸發壓力低於大氣壓力時，則空氣易侵入系統，系統處理上較爲困難，因此希望冷媒在低溫蒸發時，其蒸發壓力可高於大氣壓力。

■ 理想的冷媒物理特性

2. 蒸發潛熱要大

冷媒之蒸發潛熱大，表示使用較少的冷媒便可以吸收大量的熱量。

3. 臨界溫度要高

臨界溫度高，表示冷媒凝結溫度高，則可以用常溫的空氣或水來冷卻冷媒而達到凝結液化的作用。

4. 冷凝壓力要低

冷凝壓力低，表示用較低壓力即可將冷媒液化，壓縮機之壓縮比小，可節省壓縮機之馬力。

■ 理想的冷媒物理特性

5. 凝固溫度要低
冷媒之凝固點要低，否則冷媒在蒸發器內凍結而無法循環。
6. 氣態冷媒之比容積要小
氣態冷媒之比容積愈小愈好，則壓縮機之容積可縮小使成本降低，且吸氣管及排氣管可以用較小的冷媒配管。
7. 液態冷媒之密度要高
液態冷媒之密度愈高，則液管可用較小的配管。
8. 可溶於冷凍油，則系統不必裝油分離器

■ 理想的冷媒化學特性

1. 化學性質穩定

蒸發溫度會隨應用溫度而變化，例如冰水機之蒸發溫度約爲0~5℃，冷在冷凍循環系統中，冷媒只有物理變化，而無化學變化，不起分解作用。

2. 無腐蝕性

對鋼及金屬無腐蝕性，氨對銅具有腐蝕性，因此氨冷凍系統不得使用銅管配管；絕緣性要好，否則會破壞壓縮機馬達之絕緣，因此氨不得使用於密閉式壓縮機，以免與銅線圈直接接觸。

■ 理想的冷媒化學特性

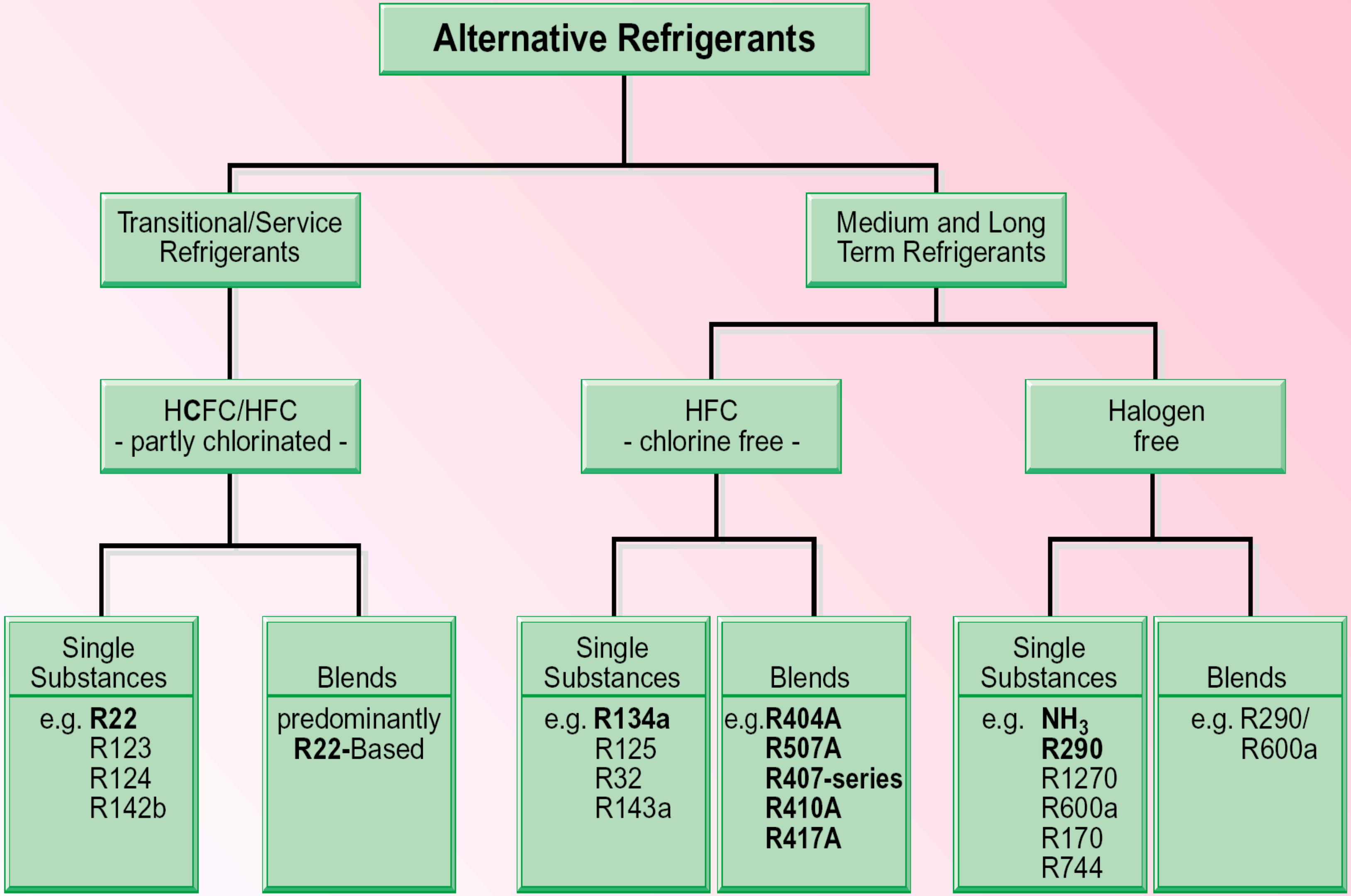
3. 無環境污染性
   對自然環境無害，不破壞臭氧層，溫室效應低。

4. 無毒性

5. 不具爆炸性與燃燒性

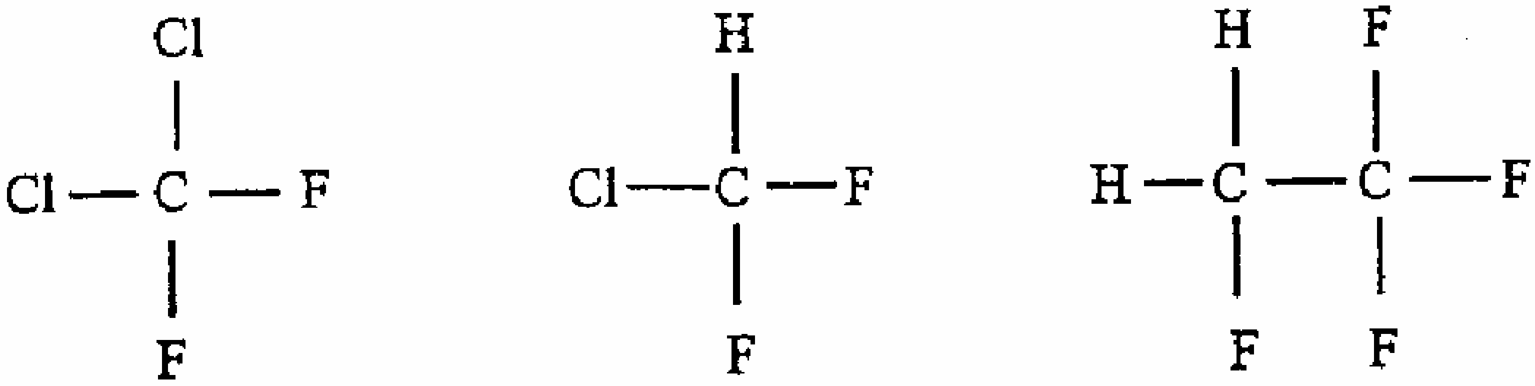
冷媒種類

Halocarbons (鹵碳化合物冷媒)
Azeotropes (共沸冷媒)
Zeotropes(非沸冷媒)
Organic compounds (有機化合物冷媒)
Inorganic compounds (無機化合物冷媒)



General survey of the alternative refrigerants **Fig. 1**

■ 冷媒的命名



CFC (R-12) (a)

HCFC (R-22) (b)

HFC (R-134a) (c)

- The first digit on the right is the number of fluorine atoms
- The second digit from the right is one more than the number of hydrogen atoms
- The third digit from the right is one less than the number of carbon atoms

■ 冷媒種類

TABLE 12.2
Numerical designation of some refrigerants.

Family	Numerical designation	Chemical name	Chemical formula
Halocarbons	12	Dichlorodifluoromethane	CCl_2F_2
	13	Chlorotrifluoromethane	$CClF_3$
	22	Chlorodifluoromethane	$CHClF_2$
	23	Trifluoromethane	CHF_3
	32	Difluoromethane	CH_2F_2
	125	Pentafluoroethane	CHF_2CF_3
	134a	Tetrafluoroethane	CH_2FCF_3
Azeotrope	R-507		R-125/R-143a
Hydrocarbons	170	Ethane 乙烷	C_2H_6
	290	Propane 丙烷	C_3H_8
	600	Butane 丁烷	C_4H_{10}
Inorganic compounds	717	Ammonia	NH_3
	744	Carbon dioxide	CO_2

■ 非共沸冷媒

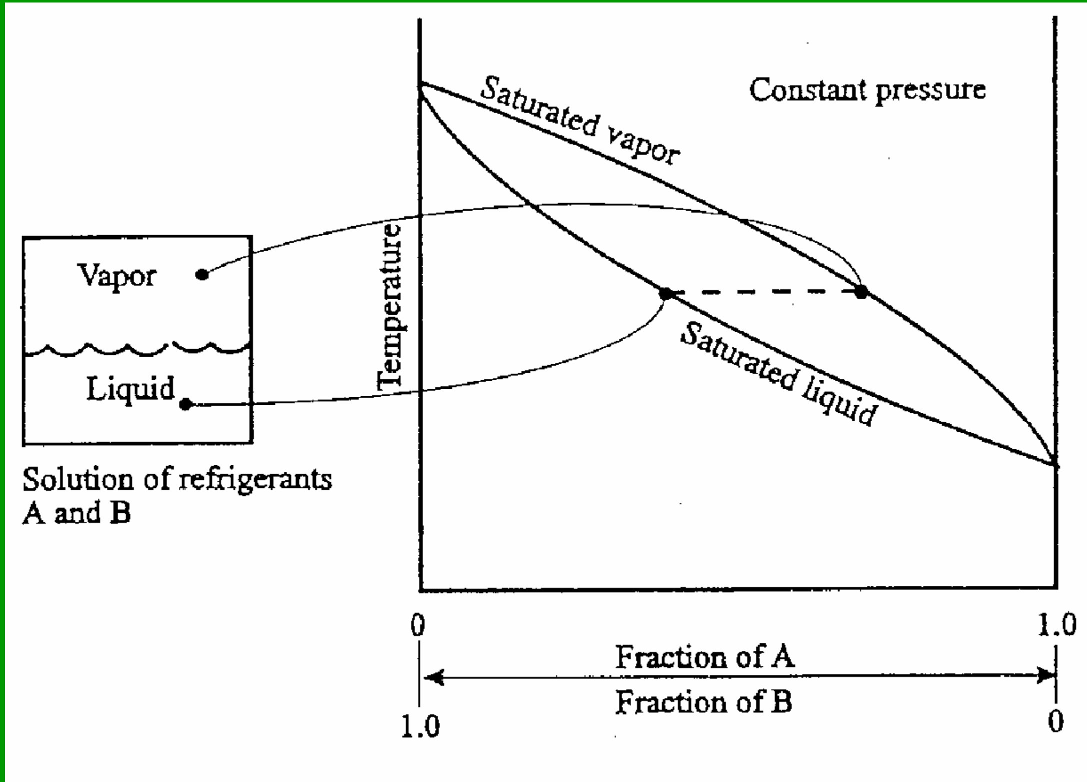


FIGURE 12.2
An ideal zeotropic mixture of substances A and B.

■ 非共沸冷媒

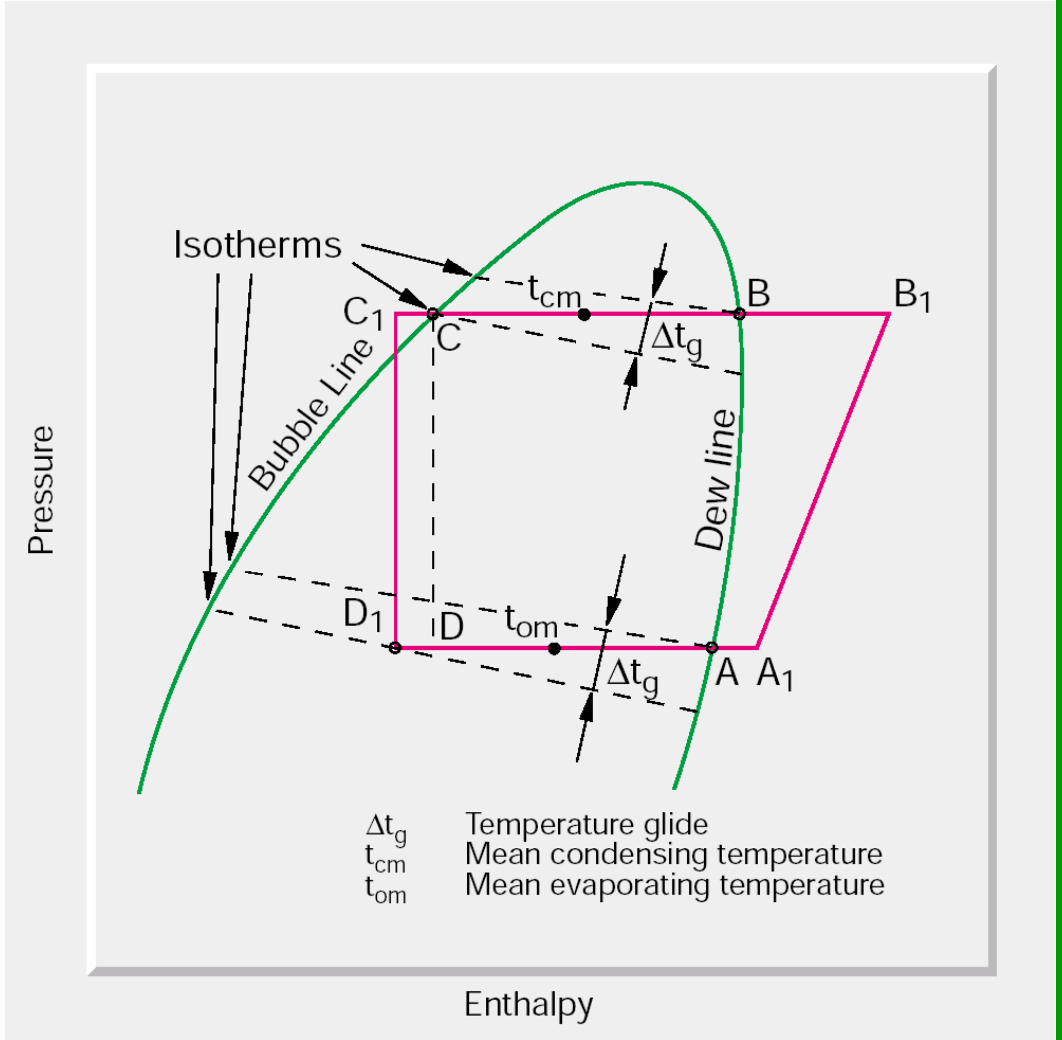


Fig. 13 Evaporating and condensing behavior of zeotropic blends

■ 共沸冷媒

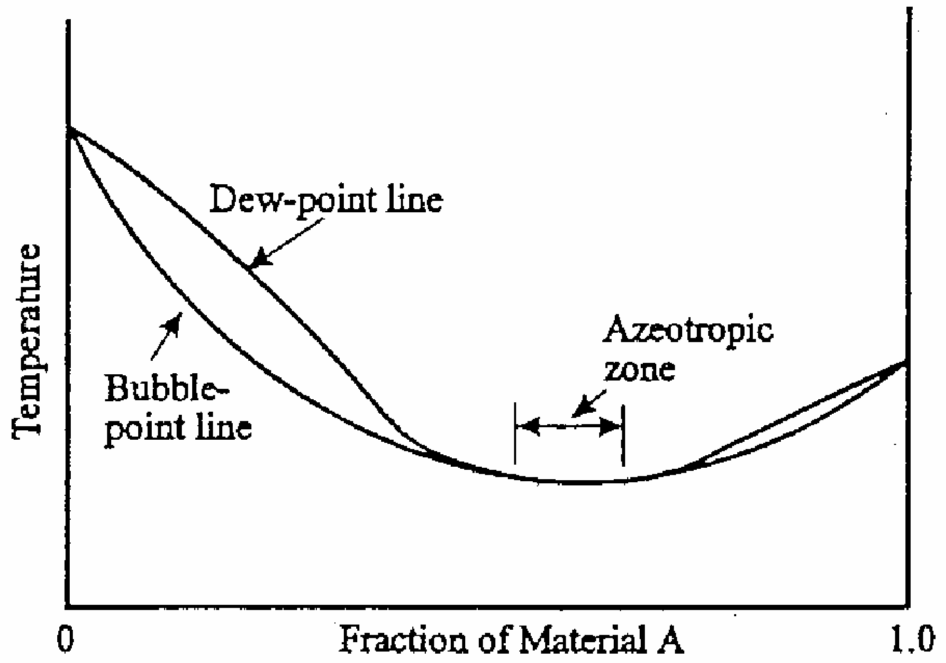


FIGURE 12.3
An azeotropic mixture of Materials A and B in the range of 50-to-60% of A.

二次冷媒

考慮因素：

- Lowest solidification temperature
- Flammability
- Compatibility with food
- Corrosion tendency and inhibition possibilities

- Viscosity
- Specific heat
- Specific gravity or density
- Thermal conductivity

■ 二次冷媒種類

Ethyl alcohol (甲醇)

Methl alcohol (乙醇)

Calcium chloride(氯化鈣)

Sodium chloride (氯化鈉)

Ethylene glycol (乙烯乙二醇)

Propylene glycol (丙烯乙二醇)

Halocarbones (鹵碳化合物)

Polymers (聚合物)

■ 二次冷媒種類

TABLE 20.1
Lowest freezing temperatures of some aqueous solutions and the concentration at which these minimum temperatures occur[2]. The practical operating temperature will be somewhat higher than these temperatures.

Solute	Minimum freezing temperature, °C (°F)	Concentration of solute, percent by mass
Acetone	−94.6 (−138)	100
Calcium chloride ($CaCl_2$)	−55 (−67)	30
d-limonene	−97 (−142)	100
Ethyl alcohol	−112 (−170)	100
Ethylene glycol	−48.3 (−55)	60
Methyl alcohol	−97.8 (−144)	100
Polydimethylsiloxane[3]	−111 (−168)	100
Propylene glycol	−51.1 (−60)	60
Sodium chloride (NaCl)	−20 (−4)	23

■ 二次冷媒相圖

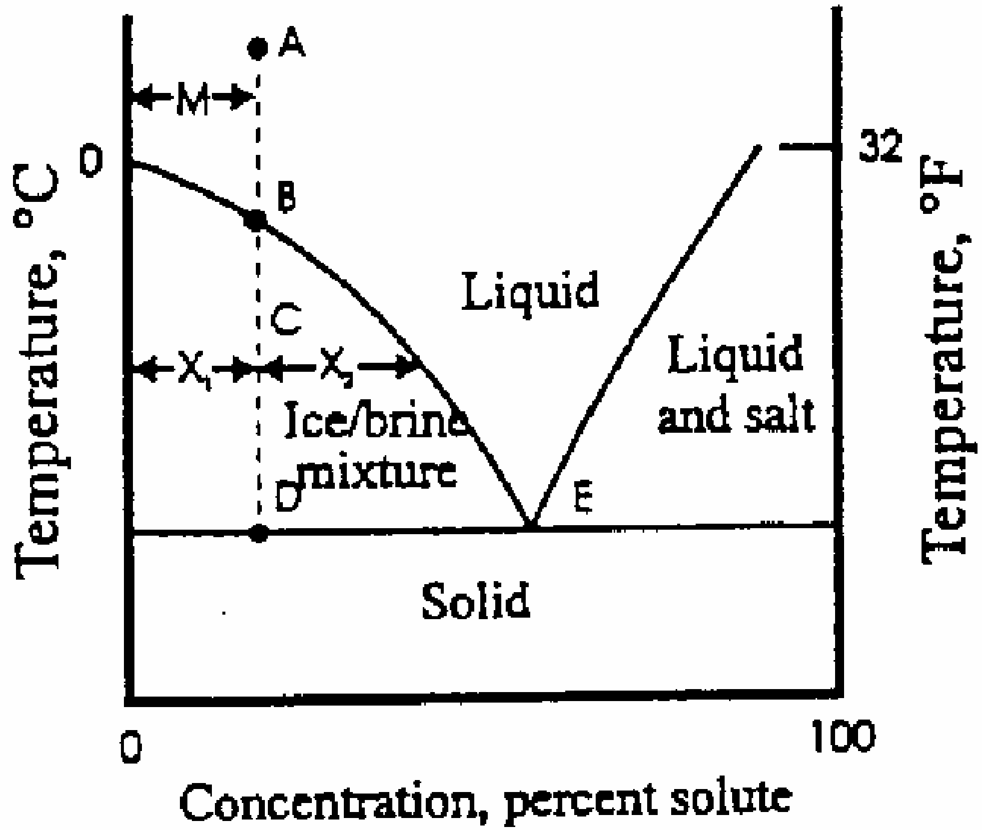


FIGURE 20.1
Phase diagram of an aqueous secondary coolant.

■ 二次冷媒相圖

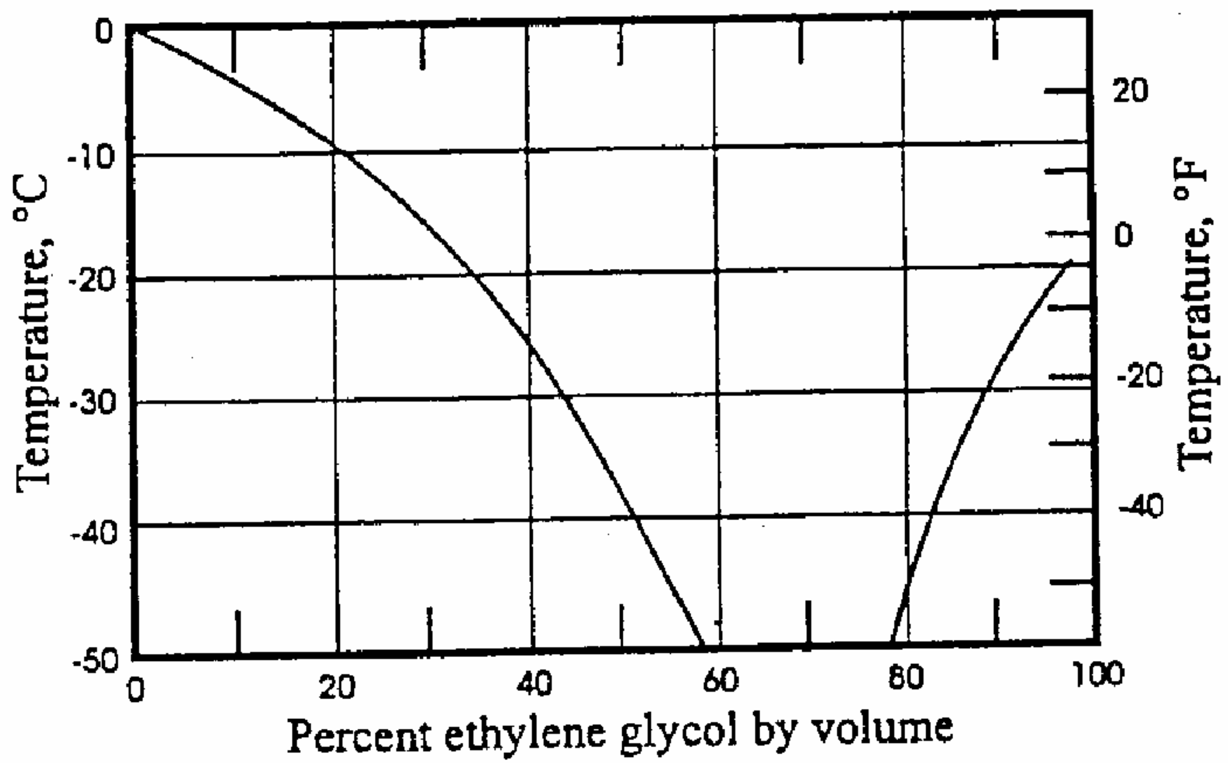


FIGURE 20.13
Freezing temperature of inhibited ethylene glycol solution[5].

■ 二次冷媒相圖

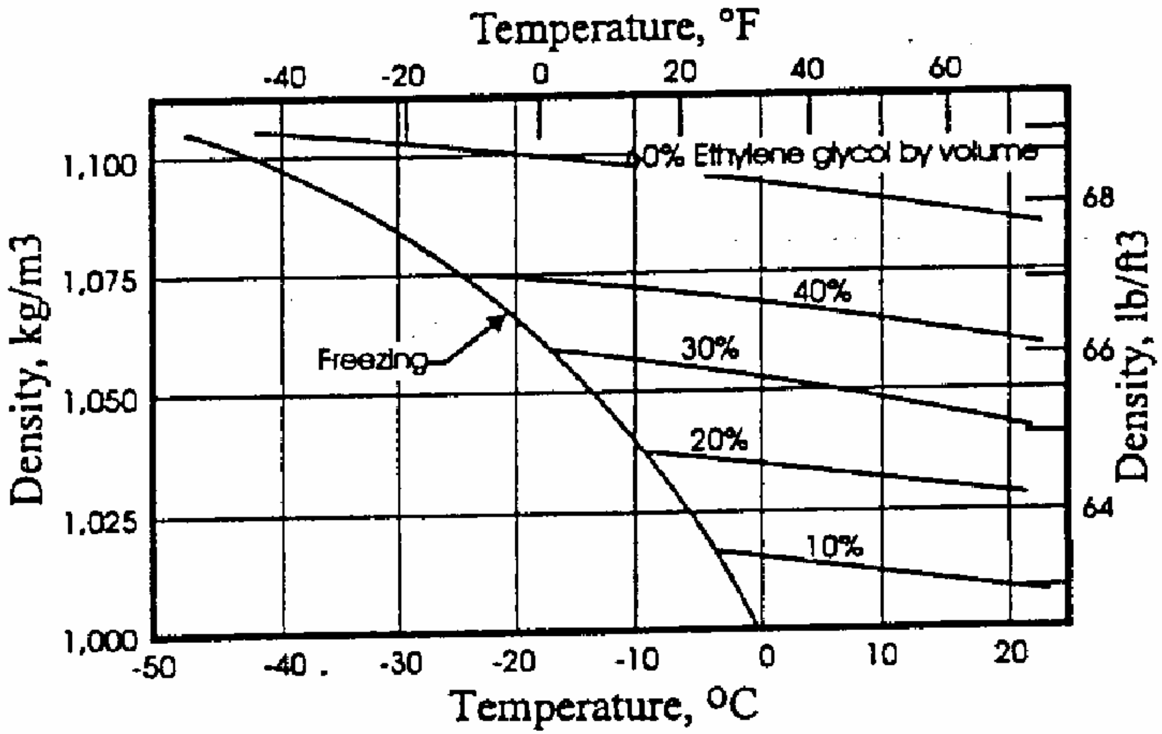
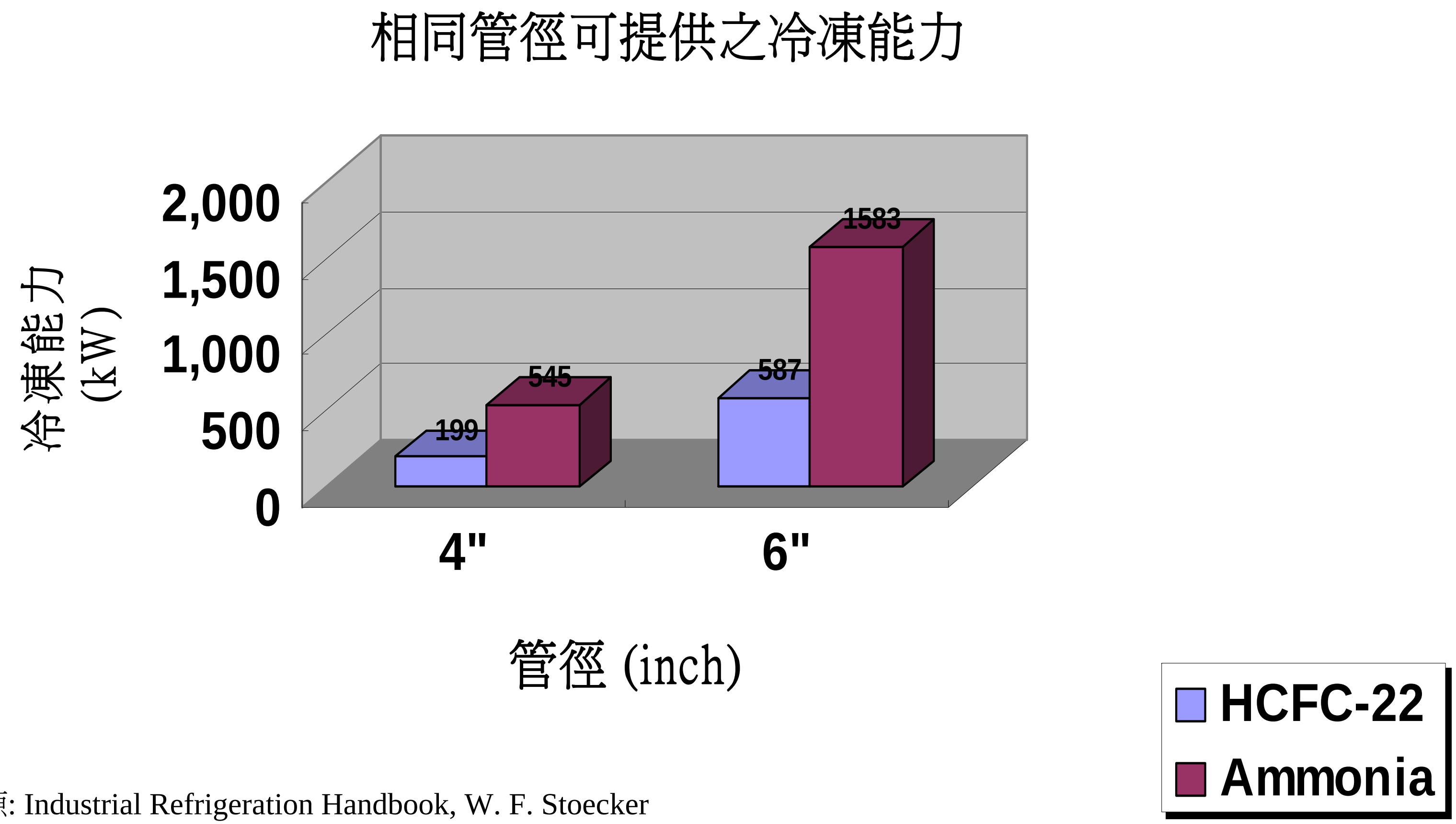


FIGURE 20.14
Density of inhibited ethylene glycol solution[5].

R-22與氨冷媒之比較

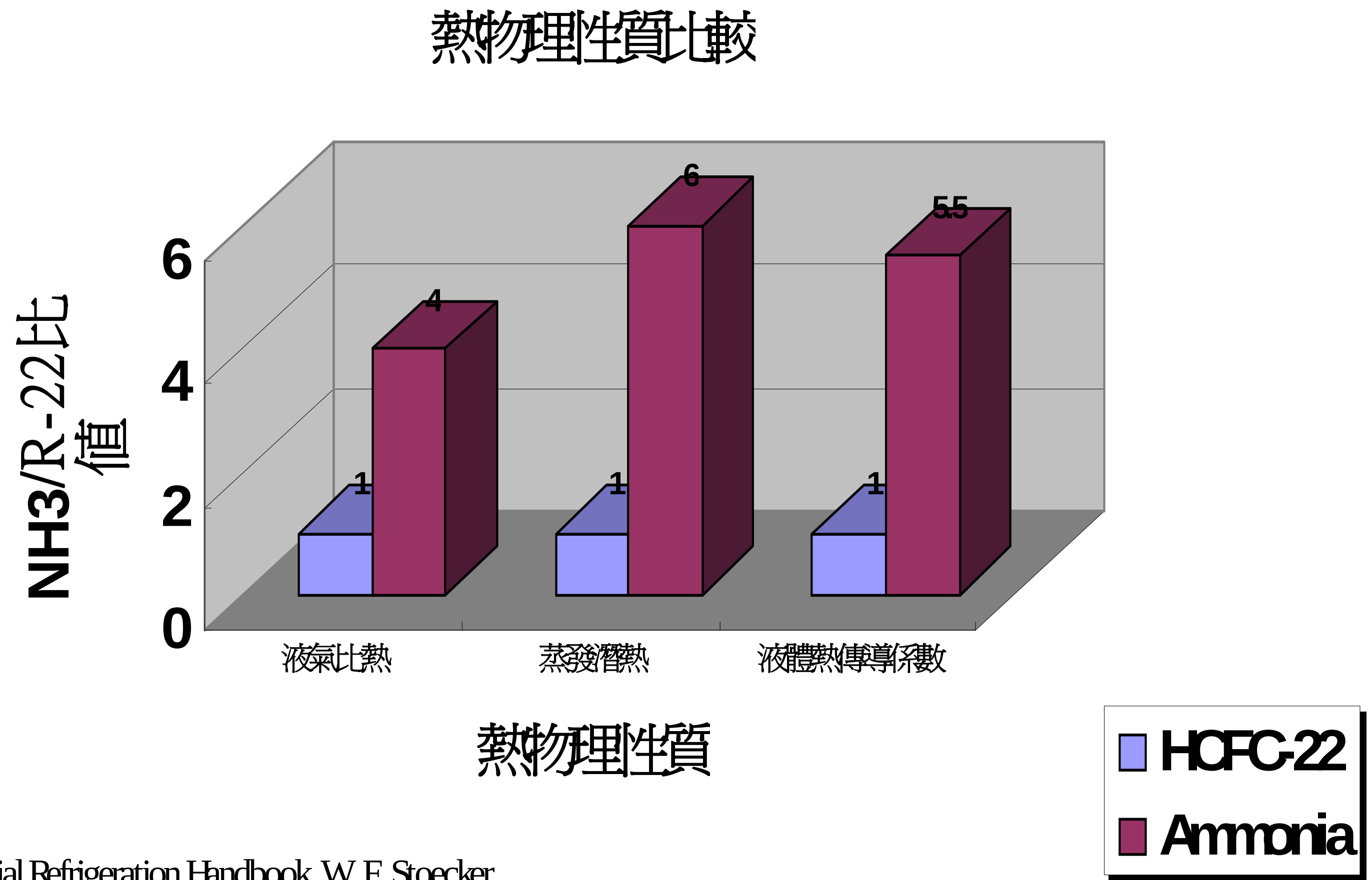
在相同管徑的條件下所能提供之冷凍能力比較



資料來源: Industrial Refrigeration Handbook, W. F. Stoecker

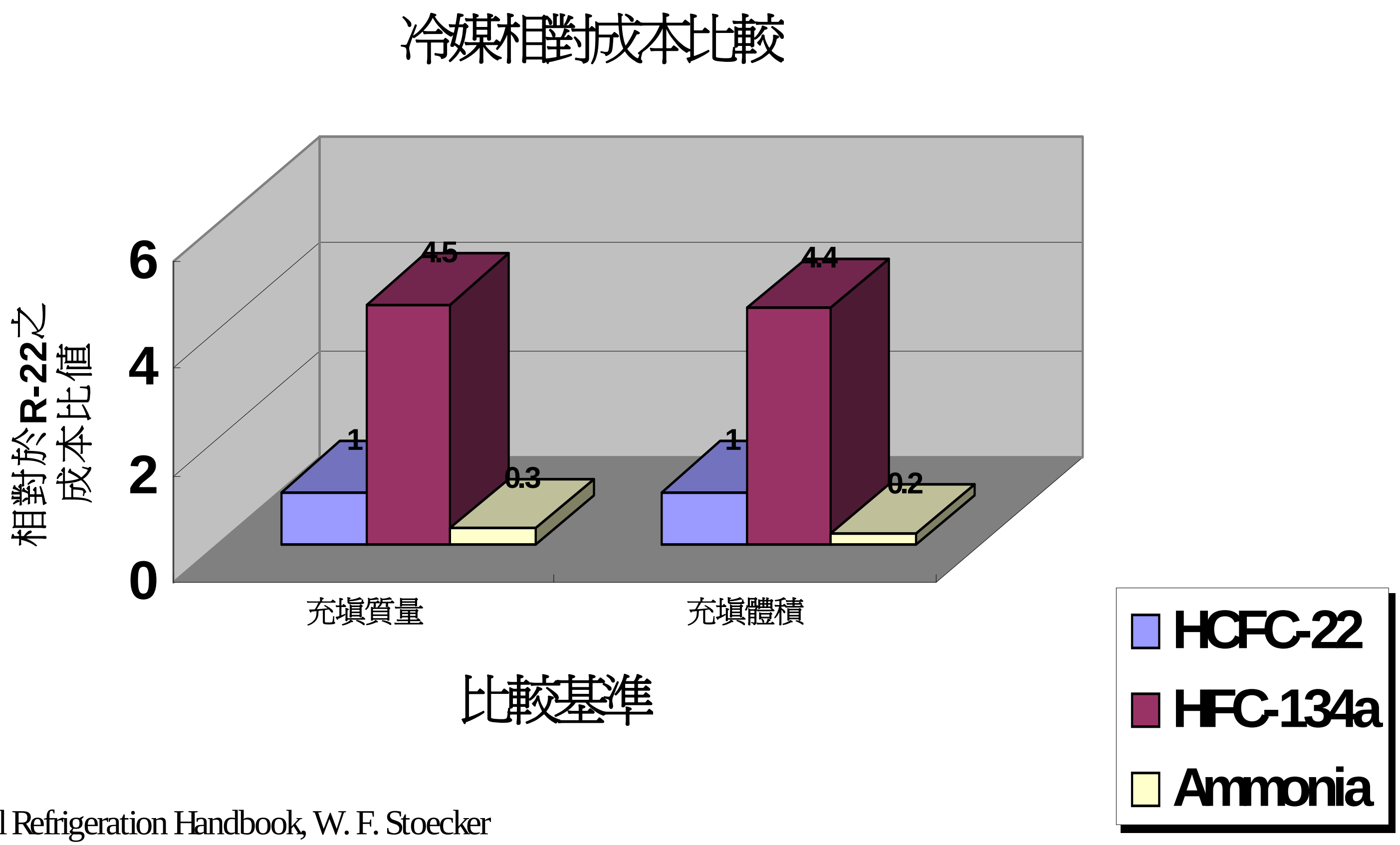
操作條件：**30.5**公尺長**(100 ft)**，**–17.8℃ (0℉)**的飽和溫度吸入管。**(0.56℃ (1℉)**飽和溫度之降低)

氨冷媒之熱物理性質高於R-22之倍數



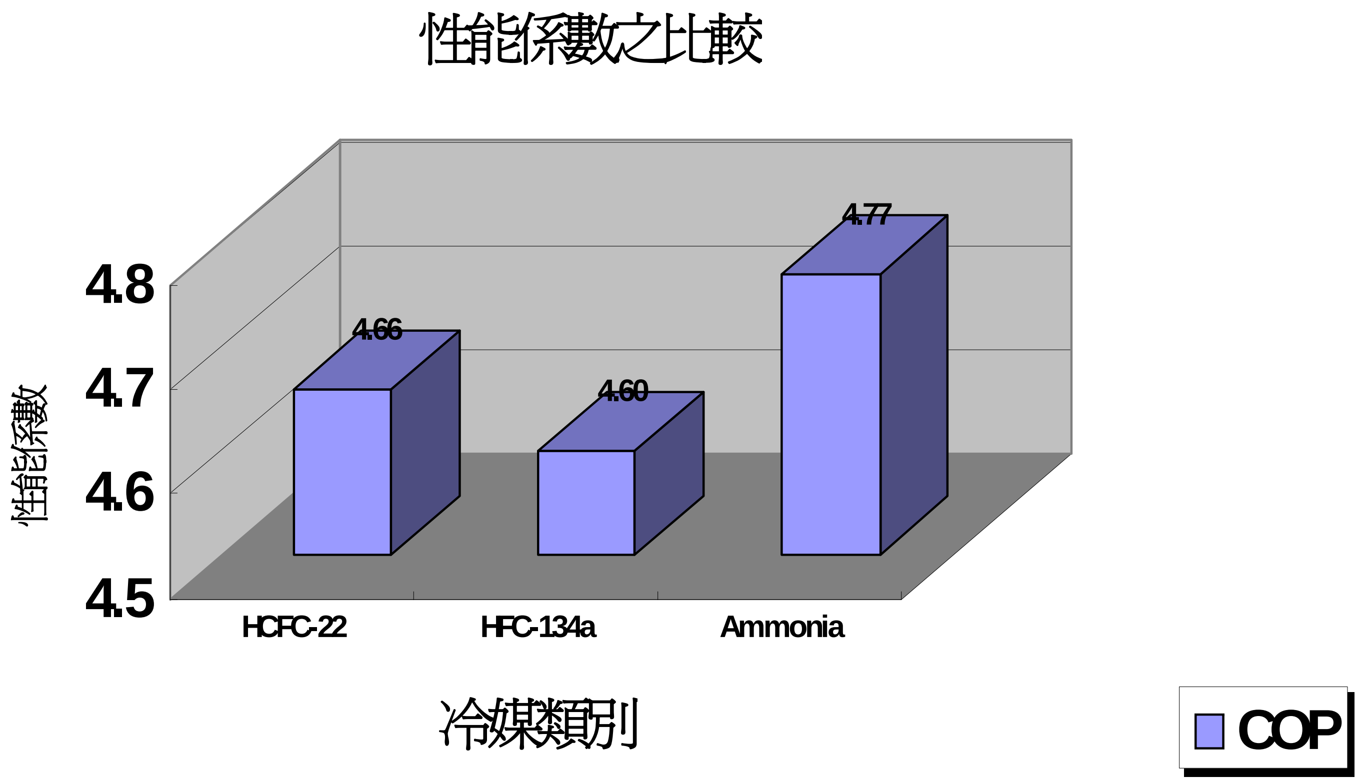
資料來源 Industrial Refrigeration Handbook, W. F. Stoecker

冷媒相對成本之比較



資料來源 Industrial Refrigeration Handbook, W. F. Stoecker

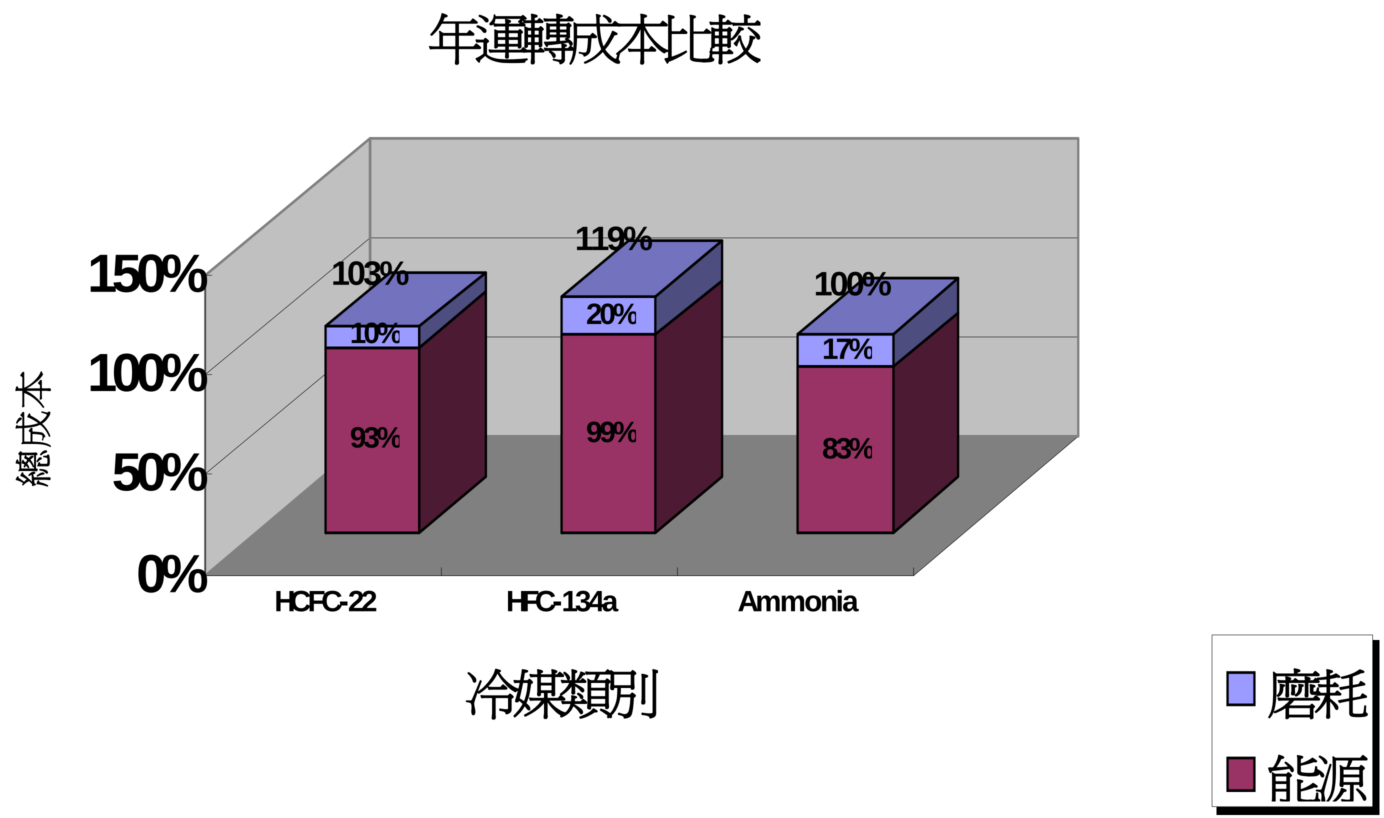
壓縮冷凍循環性能係數比較



資料來源: Industrial Refrigeration Handbook, W. F. Stoecker

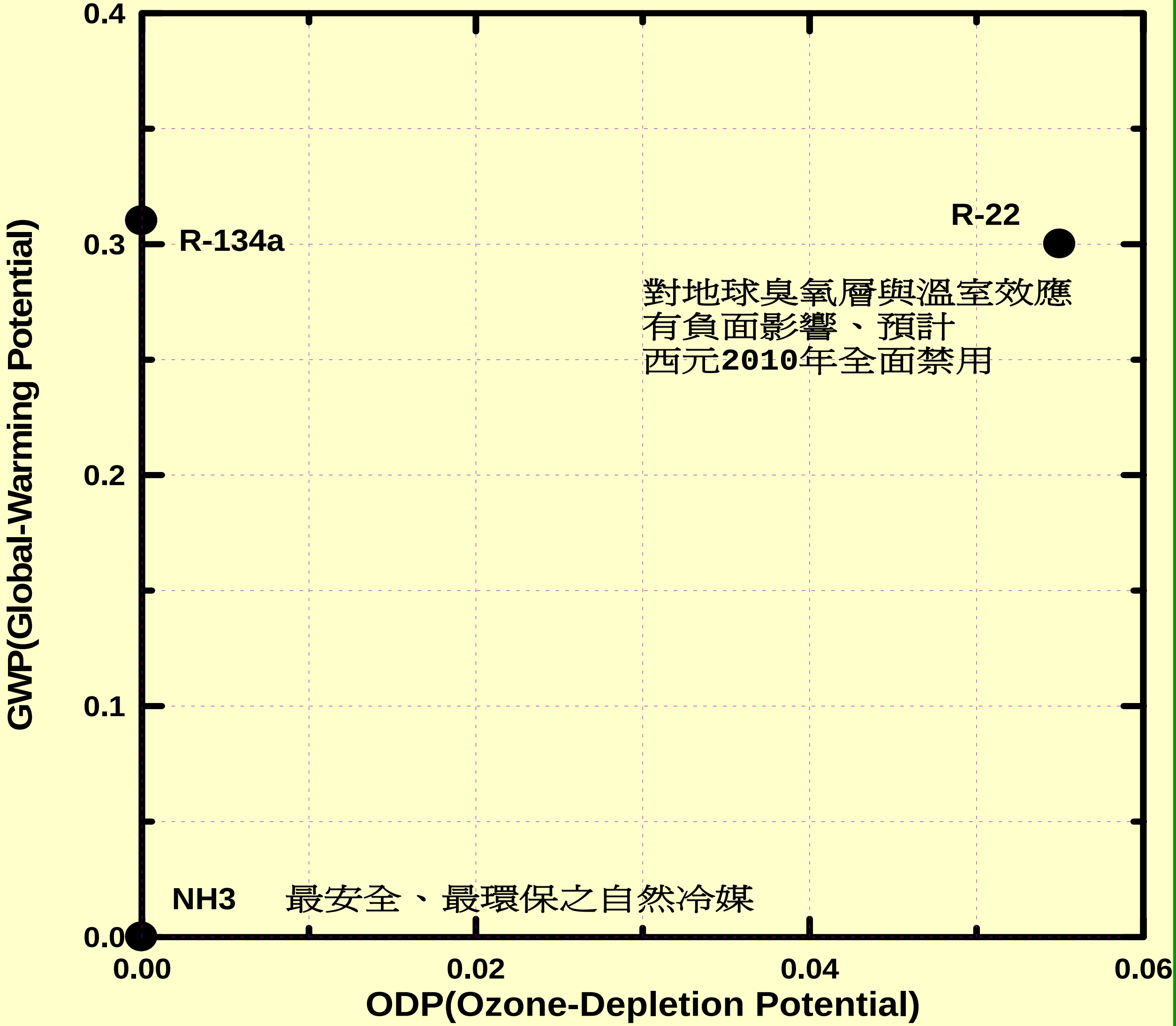
蒸發溫度**-15℃ (5℉)**、冷凝溫度**30℃ (86℉)**

年運轉成本比較



資料來源: Grasso

R-22與氨冷媒之臭氧層破壞潛能(ODP)與全球溫暖化潛能指標比較(GWP)



資料來源: Industrial Refrigeration Handbook, W. F. Stoecker

總結

- 1. 就各種熱流與冷凍性能比較結果，**NH3**中央冷凍系統明顯優於**R-22**個別冷凍系統。
- 2. 就長遠的投資角度而言，**NH3**中央冷凍系統的運轉成本明顯低於**R-22**個別冷凍系統。
- 3. 中央冷凍系統提供優於個別冷凍系統之經濟、穩健且安全的系統備載與彈性調配之能力。
- 4. 就長遠的前途而言，**NH3**中央冷凍系統的運轉將生生不息，**R-22**個別冷凍系統終將因對地球臭氧層與溫室效應之負面影響，而窮途末路，預計西元**2010**年全面禁用**R-22**冷媒，屆時可能面對更高的冷媒成本以及被迫修改系統。

Refrigerant Piping

李魁鵬

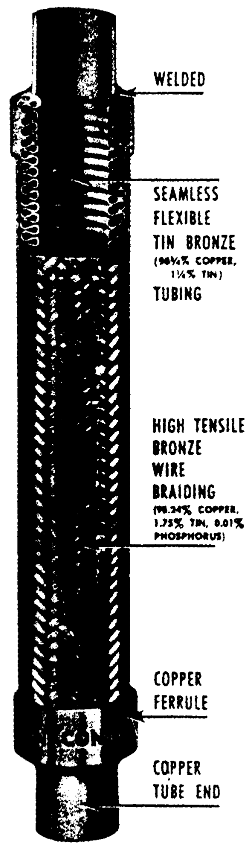


Fig. 19-1 Vibration eliminator. (Courtesy of Anaconda Metal Hose Division, The American Brass Company.)

General Design Considerations

1. Ensure an adequate supply of refrigerant to all evaporators.
2. Ensure positive and continuous return of oil to the compressor crankcase.
3. Avoid excessive refrigerant pressure losses that unnecessarily reduce the capacity and efficiency of the system.
4. Prevent liquid refrigerant from entering the compressor during either the running or off cycles, or during compressor start-up.
5. Avoid the trapping of oil in the evaporator or suction line which may subsequently return to the compressor in the form of a large "slug" with possible damage to the compressor.

TABLE 19-1A MINIMUM TONNAGE FOR OIL ENTRAINMENT UP SUCTION RISERS (TYPE L COPPER TUBING)

		Pipe OD											
		$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{5}{8}$	$2\frac{1}{8}$	$2\frac{5}{8}$	$3\frac{1}{8}$	$3\frac{5}{8}$	$4\frac{1}{8}$
	Sat. suction	Area, sq. in.											
Refrigerant	temp, F	0.146	0.233	0.348	0.484	0.825	1.256	1.78	3.094	4.77	6.812	9.213	11.97
	−40	0.061	0.110	0.182	0.27	0.54	0.91	1.4	2.79	4.78	7.49	10.9	15.1
	−20	.077	.138	.228	.34	.67	1.13	1.75	3.49	5.99	9.36	13.7	19.0
R-12*	0	.093	.167	.278	.42	.82	1.38	2.14	4.26	7.32	11.4	16.6	23.2
	20	.112	.201	.332	.50	.97	1.65	2.55	5.1	8.73	13.6	19.9	27.6
	40	.132	.238	.390	.59	1.15	1.94	3.0	6.0	10.3	16.1	23.4	32.6
	−40	0.09	0.16	0.27	0.41	0.79	1.34	2.1	4.1	7.1	11.1	16.1	22.4
	−20	.11	.20	.33	.50	.96	1.60	2.5	5.0	8.7	13.5	19.6	27.4
R-22*	0	.13	.24	.39	.59	1.2	1.96	3.0	6.1	10.4	16.2	23.6	32.8
	20	.16	.28	.46	.70	1.4	2.30	3.5	7.1	12.1	18.9	27.6	38.1
	40	.18	.33	.54	.81	1.6	2.70	4.1	8.2	14.1	22.0	32.1	44.6
	−40	0.068	0.12	0.20	0.31	0.60	1.0	1.6	3.1	5.4	8.4	12.2	16.9
	−20	.086	.16	.26	.39	.75	1.3	2.0	3.9	6.8	10.5	15.3	21.4
R-500*	0	.110	.19	.31	.47	.92	1.6	2.4	4.8	8.2	12.8	18.7	26.0
	20	.130	.23	.37	.56	1.1	1.9	2.9	5.7	9.9	15.3	22.4	31.2
	40	.150	.27	.44	.67	1.3	2.2	3.4	6.8	11.6	18.2	26.6	36.8
	−60	0.053	0.10	0.16	0.24	0.46	0.78	1.2	2.4	4.1	6.4	9.4	13.0
	−40	.070	.12	.20	.30	.59	1.0	1.5	3.1	5.3	8.3	12.0	16.8
R-502†	−20	.084	.15	.25	.38	.74	1.3	1.9	3.8	6.6	10.3	15.0	20.9
	0	.104	.19	.31	.47	.91	1.5	2.4	4.7	8.1	12.7	18.4	25.7
	20	.120	.22	.37	.56	1.1	1.8	2.9	5.7	9.8	15.2	22.2	30.8
	40	.146	.26	.43	.65	1.3	2.2	3.3	6.7	11.4	17.8	26.0	36.1

Minimum tonnage values are based on the indicated saturation temperatures (SST) with 15 F deg of superheat and 90 F liquid temperature.

* For R-12, R-22, and R-500, reduce or increase table values 1% for 10 F deg less or more superheat.

† For R-502, reduce or increase table values 2% for 10 F deg less or more superheat.

For liquid temperatures other than 90 F, multiply the table values by the corresponding factor listed in the following table:

Liquid temperature, F		50	60	70	80	90	100	110	120	130	140
Correction	R-12, R-22, R-500	1.20	1.15	1.10	1.05	1.00	0.95	0.90	0.85	0.80	0.75
Factors	R-502	1.26	1.20	1.13	1.07	1.00	0.94	0.88	0.82	0.76	0.70

TABLE 19-1B MINIMUM TONNAGE FOR OIL ENTRAINMENT UP HOT GAS RISERS (TYPE L COPPER TUBING)

Refrigerant	Sat. discharge temp, F	Pipe OD											
		1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
		Area, sq. in.											
		0.146	0.233	0.348	0.484	0.825	1.256	1.78	3.094	4.77	6.812	9.213	11.97
R-12*	80	.17	.31	.50	.77	1.51	2.54	3.93	7.84	13.5	21.0	30.7	42.6
	90	.17	.31	.51	.77	1.51	2.54	3.92	7.84	13.5	21.0	30.7	42.6
	100	.17	.31	.51	.77	1.51	2.54	3.92	7.84	13.5	21.0	30.7	42.6
	110	.17	.31	.51	.77	1.50	2.53	3.90	7.81	13.4	20.9	30.5	42.2
	120	.17	.30	.50	.75	1.47	2.49	3.84	7.66	13.2	20.6	30.0	41.6
	130	.17	.30	.49	.72	1.45	2.44	3.77	7.54	12.9	20.3	29.4	40.8
	140	.16	.28	.47	.71	1.38	2.33	3.61	7.20	12.4	19.4	28.2	39.9
R-22*	80	.23	.42	.69	1.04	2.0	3.4	5.3	10.6	18.2	28.3	41.5	57.5
	90	.23	.42	.69	1.04	2.0	3.4	5.3	10.6	18.2	28.2	41.3	57.3
	100	.23	.42	.69	1.03	2.0	3.4	5.3	10.5	18.0	28.1	41.0	56.7
	110	.23	.41	.67	1.02	2.0	3.4	5.2	10.4	17.9	27.9	40.8	56.5
	120	.22	.40	.66	1.00	2.0	3.3	5.1	10.2	17.5	27.4	39.9	55.4
	130	.22	.39	.64	.98	1.9	3.2	5.0	10.0	17.2	26.8	39.0	54.0
	140	.21	.38	.63	.96	1.9	3.2	4.9	9.7	16.7	26.1	38.0	52.6
R-500*	80	.20	.36	.59	.89	1.73	2.92	4.51	9.0	15.5	24.2	35.4	49.0
	90	.20	.35	.58	.88	1.73	2.86	4.49	8.9	15.4	24.0	35.0	48.5
	100	.20	.35	.58	.88	1.73	2.86	4.47	8.8	15.3	23.8	34.9	48.2
	110	.20	.35	.57	.87	1.70	2.86	4.45	8.7	15.2	23.7	34.7	48.0
	120	.19	.34	.56	.86	1.66	2.82	4.44	8.7	15.0	23.3	34.1	47.3
	130	.19	.34	.56	.85	1.64	2.78	4.29	8.6	14.7	23.0	33.6	46.5
	140	.18	.33	.54	.83	1.61	2.71	4.20	8.4	14.4	22.5	32.8	45.5
R-502†	80	.18	.32	.53	.80	1.55	2.7	4.1	8.2	14.1	21.9	32.5	44.3
	90	.17	.31	.51	.77	1.49	2.52	3.92	7.8	13.4	20.9	30.5	42.3
	100	.165	.30	.50	.74	1.44	2.45	3.8	7.55	13.0	20.2	29.5	40.9
	110	.160	.29	.48	.72	1.41	2.38	3.71	7.35	12.7	19.7	28.7	39.8
	120	.154	.28	.46	.69	1.33	2.26	3.52	7.0	12.4	18.7	27.3	37.9
	130	.145	.26	.43	.65	1.27	2.14	3.34	6.62	11.4	17.8	25.9	35.9
	140	.135	.24	.40	.61	1.18	1.98	3.08	6.15	10.6	16.4	24.0	33.3

* Minimum tonnages are based on a saturated suction temperature of +20 F with 15°F of superheat at the indicated saturated condensing temperatures with 15°F subcooling and actual discharge temperature based on 70% compressor efficiency. For suction temperatures other than 20°F, multiply the table values by the following factors:

Sat. Suct. Temperature	−40	−20	0	+20	+40
Correction Factor	0.85	0.90	0.95	1.0	1.06

† Minimum tonnages are based on a saturated temperature of −20°F. All other conditions are the same as above. For suction temperatures other than −20°F, multiply the table values by the following factors:

Sat. Suct. Temperature	−60	−40	−20	0	+20	+40
Correction Factor	0.87	0.94	1.0	1.08	1.15	1.21

Example 19-1 Determine the minimum size suction riser that will ensure oil return at minimum loading for a 75-ton R-502 system that is equipped with a reciprocating compressor having capacity steps of 25%, 50%, 75%, and 100% if the design saturated suction temperature at minimum loading is −20°F and the liquid refrigerant approaching the refrigerant flow control is 70°F.

Solution Since the minimum system capacity will occur when the compressor is operating at the lowest step of capacity, which is 25% of full load, the minimum capacity is (75 tons × 0.25) 18.75 tons. For R-502 at a saturated suction temperature of −20°F, Table 19-1A indicates a minimum capacity of 15 tons for a $3\frac{5}{8}$ in. OD copper pipe. The correction factor listed for 70°F liquid is 1.13, so that the corrected minimum capacity for the $3\frac{5}{8}$ in. OD pipe is (15 tons × 1.13) 16.95 tons. Since minimum pipe capacity is less than the system minimum load, this size suction riser will ensure oil return during periods of minimum loading.

TABLE 19-2 REFRIGERANT LINE CAPACITIES FOR REFRIGERANT-12 (SINGLE- OR HIGH-STAGE APPLICATIONS) (TONS OF REFRIGERATION RESULTING IN A LINE FRICTION DROP PER 100 FT EQUIVALENT PIPE LENGTH CORRESPONDING TO 2°F (ΔT) CHANGE IN SATURATION TEMPERATURE)

Line size type L copper OD	Suction lines*					Discharge lines* $\Delta P = 3.66$			Liquid lines line size type L copper OD	Condenser to receiver velocity = 100 fpm	Receiver* to system $\Delta T = 1$ F $\Delta P = 1.8$ psi
	Suction temp F										
	−40 $\Delta P = 0.49$	−20 $\Delta P = 0.72$	0 $\Delta P = 1.01$	20 $\Delta P = 1.38$	40 $\Delta P = 1.82$	Sat. −40	Suct. 0	Temp F 40			
1/2				0.21	0.31	0.46	0.54	0.67	1/2	1.16	2.03
5/8		0.17	0.26	0.40	0.58	0.85	0.98	1.23	5/8	2.65	3.81
7/8	0.25	0.42	0.68	1.04	1.50	2.23	2.58	3.22	7/8	6.94	10.10
1 1/8	0.51	0.87	1.39	2.10	3.10	4.60	5.30	6.65	1 1/8	11.85	20.5
1 3/8	0.87	1.52	2.40	3.70	5.36	7.8	9.0	11.3	1 3/8	18.10	35.1
1 5/8	1.41	2.44	3.86	5.82	8.50	12.4	14.4	18.0	1 5/8	25.5	57.5
2 1/8	2.94	5.03	8.00	12.1	17.6	25.8	30.0	37.4	2 1/8	44.4	117.8
2 5/8	5.20	8.94	14.2	21.3	31.4	45.5	52.5	66.0	2 5/8	68.4	207.8
3 1/8	8.35	14.3	22.7	34.0	49.5	73.0	85.0	106.0	3 1/8	97.5	344.0
3 5/8	12.4	21.2	33.8	50.6	73.5	107.0	124.0	155.0	3 5/8	132.0	508.0
4 1/8	17.4	29.9	47.7	71.0	103.0	152.0	176.0	220.0	4 1/8	173.0	704.0
5 1/8	31.7	54.0	85.3	128.0	187.0	270.0	314.0	392.0			
6 1/8	50.8	86.0	137.0	206.0	299.0	428.0	494.0	620.0			

Steel IPS	SCH									Steel IPS	SCH		
½	40			0.30	0.45	0.64	0.92	1.07	1.34	½	80	3.43	3.23
¾	40	0.24	0.41	0.64	0.96	1.39	1.96	2.26	2.83	¾	80	6.25	7.27
1	40	0.46	0.78	1.22	1.82	2.68	3.75	4.35	5.42	1	80	10.4	14.3
1¼	40	0.97	1.60	2.52	3.78	5.41	7.8	9.0	11.3	1¼	80	18.6	30.1
1½	40	1.50	2.41	3.76	5.62	8.12	11.4	13.2	16.5	1½	80	25.5	47.3
2	40	2.81	4.69	7.40	10.9	15.7	21.6	25.1	31.4	2	40	48.0	111.9
2½	40	4.44	7.42	11.6	17.3	24.7	34.7	40.2	50.2	2½	40	68.3	173.0
3	40	8.04	13.2	20.6	30.6	43.8	61.0	70.8	88.4	3	40	104.0	311.8
4	40	16.03	27.0	42.8	62.9	90.2	125.0	146.0	182.0	4	40	179.0	634.0
5	40	30.0	49.1	78.7	114.0	165.0	228.0	264.0	330.0				
6	40	48.2	78.6	124.0	182.0	268.0	365.0	421.0	528.0				
8	48	98.4	161.0	254.0	376.0	541.0	745.0	865.0	1080.0				
10	40	180.0	297.0	458.0	678.0	972.0	1350.0	1570.0	1960.0				
12	ID	286.0	475.0	729.0	1080.0	1520.0	2130.0	2460.0	3090.0				

NOTES:

* (1) Basis of table: 100°F condensing temperature, 2 $F\,\Delta T$ per 100 ft equivalent length (except liquid lines).

(2) For other ΔT's and equivalent lengths (L_e),

$$\text{Line capacity (tons)} = \text{Table tons} \times \left(\frac{\text{Actual } \Delta T \text{ loss desired}}{\text{Table } \Delta T \text{ loss}}\right)^{0.55}$$

(3) For other tons and equivalent lengths in a given pipe size,

$$\Delta T = \text{Table } \Delta T \times \left(\frac{L_e}{100}\right) \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$

(4) Values based on 100°F condensing temperature. For capacities at other condensing temperatures, multiply table value by line capacity multiplier below:

Line	Condensing temperature, F					
	80	90	100	105	110	120
Suction Lines	1.11	1.06	1.00	0.97	0.94	0.88
Discharge Lines	0.88	0.94	1.00	1.04	1.07	1.16

(5) Tabulated data taken from Chapter 9 of the *ASRE Data Book,* Design Volume, 1957–58 Edition. Initially developed from ARI preliminary data.

(6) Pressure drop equivalent of saturation temperature loss

$$\text{Actual } \Delta P = \text{Table } \Delta P \times \left(\frac{\text{Actual } \Delta T}{\text{Table } \Delta T}\right)$$

From *ASRE Data Book,* Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers.

TABLE 19-3 REFRIGERANT LINE CAPACITIES FOR REFRIGERANT-22 (SINGLE- OR HIGH-STAGE APPLICATIONS) (TONS OF REFRIGERATION RESULTING IN A LINE FRICTION DROP PER 100 FT EQUIVALENT PIPE LENGTH CORRESPONDING TO 2°F (ΔT) CHANGE IN SATURATION TEMPERATURE)

Line size type L copper OD		Suction lines*					Discharge lines* $\Delta P = 6.1$			Liquid lines			
		Suction temp, F								Line size type L copper OD		Condenser to reciever velocity = 100 fpm	Receiver to system $\Delta T = 1$ F
		−40 $\Delta P = 0.79$	−20 $\Delta P = 1.15$	0 $\Delta P = 1.6$	20 $\Delta P = 2.22$	40 $\Delta P = 2.93$	Sat. −40	Suct. 0	Temp 40				
½					0.40	0.59	1.0	1.1	1.2	½		2.24	3.5
⅝			0.32	0.49	0.75	1.10	2.1	2.3	2.5	⅝		3.57	6.4
⅞		0.35	0.87	1.31	2.00	1.89	4.9	5.4	5.2	⅞		7.41	17.0
1⅛		1.08	1.74	2.65	4.03	5.82	9.8	10.7	11.8	1⅛		12.7	34.4
1⅜		1.88	3.01	4.61	7.03	9.98	17.0	18.6	20.5	1⅜		19.2	60.0
1⅝		2.90	4.78	7.23	10.26	15.95	26.4	29.0	31.9	1⅝		27.2	95.0
2⅛		6.21	9.97	15.2	23.2	33.2	55.0	60.3	66.3	2⅛		47.3	200.0
2⅝		10.8	17.5	26.5	40.3	58.1	96.0	105.0	115.6	2⅝		73.2	354.0
3⅛		17.3	27.1	43.3	64.5	23.1	155.0	170.0	187.3	3⅛		104.1	572.0
3⅝		25.9	41.9	63.9	96.8	139.5	233.0	255.0	281.0	3⅝		141.1	860.0
4⅛		36.9	59.2	89.7	136.0	196.0	327.0	358.0	394.0	4⅛		183.0	1200.0
5⅛		66.1	106.7	162.0	245.0	355.0	588.0	644.0	709.0				
Steel IPS	SCH									Steel IPS	SCH		
½	40		0.38	0.56	0.84	1.20	2.0	2.2	2.4	½	80	4.66	5.5
¾	40	0.49	0.78	1.18	1.77	2.52	4.1	4.5	5.0	¾	80	6.17	12.2
1	40	0.94	1.49	2.24	3.24	4.72	7.8	8.5	9.4	1	80	13.2	24.4
1¼	40	1.95	3.01	4.67	6.83	9.73	16.1	17.6	19.4	1¼	80	22.9	51.5
1½	40	2.89	4.63	7.04	10.4	14.7	24.1	26.5	29.1	1½	80	37.1	78.0
2	40	5.60	8.90	13.0	19.9	28.2	46.6	51.0	56.2	2	40	51.5	185.0
2½	40	8.90	14.2	21.5	31.9	45.9	74.7	82.0	90.0	2½	40	73.3	297.0
3	40	15.9	25.2	38.0	56.5	80.1	132.0	144.0	159.0	3	40	113.0	510.0
3½	40	23.1	36.1	55.1	81.0	116.0	189.0	207.0	228.0	3½	40	151.5	704.0

4	40	32.1	50.8	76.7	112.7	159.5	260.0	285.0	314.0
5	40	57.8	91.0	138.6	204.0	292.0	477.0	520.0	575.0
6	40	94.1	148.6	224.0	329.0	472.0	775.0	850.0	937.0
8	40	199.0	316.0	474.0	704.0	996.0	1650.0	1810.0	1992.0
10	40	294.2	550.0	840.0	1226.0	1760.0	2880.0	3150.0	3470.0
12	ID	555.0	877.0	1340.0	1935.0	2795.0	4640.0	5080.0	5590.0

4	40	195.0	1060.0

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating and Air Conditioning Engineers.

NOTES:

* (1) Basis of table 105°F Condensing Temperature, 2 F ΔT per 100 ft equivalent length (except liquid lines).

(2) For other ΔT's and equivalent lengths (L_e),

$$\text{Line capacity (tons)} = \text{Table tons} \times \left(\frac{\text{Actual } \Delta T \text{ desired}}{\text{Table } \Delta T \text{ loss}}\right)^{0.55}$$

(3) For other tons and equivalent lengths in a given pipe size,

$$\Delta T = \text{Table } \Delta T \times \left(\frac{L_e}{100}\right) \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$

(4) Pressure drop equivalent of saturation temperature loss

$$\text{Actual } \Delta P = \text{Table } \Delta P \times \left(\frac{\text{Actual } \Delta T}{\text{Table } \Delta T}\right)$$

(5) Tabulated data taken from Chapter 9 of the *ASRE DATA Book*, Design Volume, 1957–58 Edition. Initially developed from ARI preliminary data.

(6) For other condensing temperatures, multiply table tons by the following factors:

Condensing temp F	Suction lines	Hot gas lines
80	1.13	0.77
90	1.08	0.86
100	1.03	0.95
110	0.97	1.04
120	0.91	1.13

TABLE 19-4 REFRIGERANT LINE CAPACITIES FOR REFRIGERANT-502 (SINGLE- OR HIGH-STAGE APPLICATIONS) (TONS OF REFRIGERATION RESULTING IN A LINE FRICTION DROP (ΔP in PSI) PER 100 FT EQUIVALENT PIPE LENGTH AS SHOWN, WITH CORRESPONDING (ΔT) CHANGE IN SATURATION TEMPERATURE)

Line size type L copper, OD	Suction lines ΔT = 2 F						Discharge lines ΔT = 1.0 F ΔP = 3.15			Liquid lines[a]		
	Suction temp, F						Saturated suction temp			Line size type L copper OD	Velocity = 100 fpm	ΔT = 1 F ΔP = 3.15
	−60 ΔP = 0.31	−40 ΔP = 0.94	−20 ΔP = 1.33	0 ΔP = 1.83	20 ΔP = 2.43	40 ΔP = 3.14	−40	0	40			
1/2	0.10	0.11	0.15	0.22	0.34	0.49	0.61	0.62	0.78	1/2	1.61	2.40
5/8	0.11	0.15	0.26	0.42	0.63	0.91	1.14	1.27	1.45	5/8	2.58	4.52
7/8	0.23	0.41	0.68	1.09	1.64	2.39	2.98	3.34	3.80	7/8	5.35	12.01
1 1/8	0.46	0.82	1.38	2.20	3.33	4.83	6.02	6.74	7.66	1 1/8	9.13	24.43
1 3/8	0.80	1.44	2.42	3.84	5.80	8.41	10.49	11.74	13.34	1 3/8	13.90	42.71
1 5/8	1.27	2.28	3.83	6.07	9.16	13.29	16.51	18.49	21.01	1 5/8	19.68	67.69
2 1/8	2.65	4.76	7.97	12.63	18.98	27.45	34.03	38.14	43.36	2 1/8	34.23	140.87
2 5/8	4.71	8.44	14.12	22.29	33.50	48.38	59.93	67.18	76.35	2 5/8	52.79	249.43
3 1/8	7.56	13.54	22.58	35.56	53.38	77.02	95.34	107.2	121.5	3 1/8	75.35	398.62
3 5/8	11.30	20.15	33.58	52.83	79.25	114.56	141.4	158.6	180.1	3 5/8	101.9	593.10
4 1/8	15.98	28.47	47.39	74.49	111.78	160.90	199.0	223.1	253.5	4 1/8	132.5	837.24
5 1/8	28.71	51.07	84.85	133.32	199.37	286.92	354.3	397.2	451.2	—	—	—
6 1/8	46.35	82.31	136.77	214.07	319.89	459.97	567.6	636.5	723.1	—	—	—

From *ASHRAE Data Book*, Fundamentals Volume, 1972 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

NOTES:

(1) For other ΔT's and equivalent lengths, L_e,

$$\text{Line capacity (tons)} = \text{Table tons} \times \left(\frac{100}{L_e}\right) \times \left(\frac{\text{Actual } \Delta T \text{ loss desired}}{\text{Table } \Delta T \text{ loss}}\right)^{0.55}$$

(2) For other tons and equivalent lengths in a given pipe size

$$\Delta T = \text{Table } \Delta T \times \left(\frac{L_e}{100}\right) \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$

(3) Values are based on 105°F condensing temperature. For other condensing temperatures, multiply table tons by the following factors:

Condensing temp F	Suction lines	Hot gas lines
80	1.20	.83
90	1.12	.91
100	1.04	.97
110	.96	1.02
120	.88	1.08
130	.80	1.16

TABLE 19-5 REFRIGERANT LINE CAPACITIES FOR REFRIGERANT-717 (AMMONIA)(SINGLE- OR HIGH-STAGE APPLICATIONS) (TONS OF REFRIGERATION RESULTING IN A LINE FRICTION DROP PER 100 FT EQUIVALENT PIPE LENGTH CORRESPONDING TO 1°F (ΔT) CHANGE IN SATURATION TEMPERATURE)

		Suction lines*						Liquid lines			
		Suction temperature F									
Line size IPS	Line size SCH	−40 $\Delta P = 0.32$	−20 $\Delta P = 0.52$	0 $\Delta P = 0.78$	20 $\Delta P = 1.08$	40 $\Delta P = 1.48$	Discharge lines $\Delta P = 3.3$	Line size IPS	Line size SCH	Condenser to receiver velocity = 100 fpm	Receiver to system $\Delta P = 3.3$
3/8	80										
1/2	80						3.63	1/2	80	13.5	29.7
3/4	80				2.58	3.75	7.98	3/4	80	24.9	66.7
1	80		2.11	3.46	5.14	7.50	15.9	1	80	41.5	130.0
1 1/4	40	3.24	5.57	8.90	13.4	19.4	41.2	1 1/4	40	86.2	281.0
1 1/2	40	4.83	8.75	13.70	20.2	29.4	57.5	1 1/2	40	117.2	439.0
2	40	9.34	16.4	26.2	39.4	57.3	118.9	2	40	193.5	1004.0
2 1/2	40	15.0	26.0	42.2	62.5	91.2	187.2	2 1/2	40	276.0	1599.0
3	40	26.9	46.0	73.9	111.0	162.0	338.2	3	40	425.0	2341.0
4	40	56.1	94.5	151.0	226.0	327.0	676.0	4	40	736.0	5750.0
5	40	102.0	172.0	272.0	408.0	592.0	1228.0	5	40		
6	40	160.0	280.0	445.0	662.0	958.0	1986.0	6	40		
8	40	338.0	570.0	908.0	1355.0	1960.0	4120.0	8	40		
10	40	605.0	1030.0	1640.0	2430.0	3555.0		10	40		
12	ID	975.0	1660.0	2640.0	3940.0	5680.0		12	ID		

NOTES:

(1) Basis of table: 100°F condensing temperature, 1 F ΔT per 100 ft equivalent length. Discharge and liquid lines based on 0°F suction.

(2) For other ΔT's and equivalent lengths,

$$\text{Line capacity (tons)} = \text{Table tons} \times \left(\frac{100 \times \text{Actual } \Delta T \text{ loss desired, F}}{\text{Actual equiv. length, ft}}\right)^{0.55}$$

(3) For other tons and equivalent lengths,

$$\Delta T \text{ for a given pipe size} = \left(\frac{\text{Actual equiv. length, ft}}{100}\right) \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$

(4) Values based on 100°F condensing temperature. For capacities at other condensing temperatures, multiply table value by line capacity multiplier:

	Condensing temperature F			
Line	70	80	90	100
Suction lines	1.0	1.0	1.0	1.0
Discharge lines	0.70	0.80	0.90	1.0

(5) Pressure drop equivalent to saturation temperature loss

$$\text{Actual } \Delta P = \text{Table } \Delta P \times \left(\frac{\text{Actual } \Delta T}{\text{Table } \Delta T}\right)$$

(6) Tabulated data taken from Chapter 9 of the *ASRE Data Book*, Design volume, 1957–58 Edition. Reprinted by permission of ASHRAE. Initially developed from ARI preliminary data.

TABLE 19-6 R-134a REFRIGERANT LINE SIZING SELECTIONS FOR TYPE L COPPER TUBING (DIMENSIONS IN OD; DATA BASED ON 120° CONDENSING)

Table 1. R-134a Refrigerant Line Sizing Selections for Type L Copper Tubing (Dimensions in OD; data based on 120° condensing)

Evaporator Capacity (tons)	Suction Line Sizes to Limit Pressure Drop to 2° Equivalent									Discharge Line Sizes for 1° Equivalent			Liquid Line Sizes for 1° Equivalent		
	−40° Evap.			0° Evap.			40° Evap.								
	Equivalent Piping Length (ft)									Equivalent Piping Length (ft)					
	25	50	100	25	50	100	25	50	100	25	50	100	25	50	100
1/4	3/4	7/8	1-1/8	1/2	5/8	3/4	3/8	1/2	1/2	3/8	3/8	3/8	3/8	3/8	3/8
1/2	1-1/8	1-1/8	1-3/8	5/8	3/4	7/8	1/2	5/8	5/8	3/8	1/2	1/2	3/8	3/8	3/8
3/4	1-1/8	1-3/8	1-5/8	3/4	7/8	1-1/8	5/8	5/8	3/4	1/2	1/2	5/8	3/8	3/8	3/8
1	1-3/8	1-3/8	1-5/8	7/8	1-1/8	1-1/8	5/8	3/4	3/4	1/2	5/8	5/8	3/8	3/8	3/8
1-1/2	1-3/8	1-5/8	2-1/8	1-1/8	1-1/8	1-3/8	3/4	7/8	7/8	5/8	5/8	3/4	3/8	3/8	1/2
2	1-5/8	2-1/8	2-1/8	1-1/8	1-3/8	1-5/8	3/4	7/8	1-1/8	5/8	3/4	7/8	3/8	3/8	1/2
3	2-1/8	2-1/8	2-5/8	1-3/8	1-3/8	1-5/8	7/8	1-1/8	1-1/8	3/4	7/8	7/8	1/2	1/2	5/8
5	2-1/8	2-5/8	3-1/8	1-5/8	1-5/8	2-1/8	1-1/8	1-3/8	1-3/8	7/8	1-1/8	1-1/8	1/2	5/8	3/4
7-1/2	2-5/8	3-1/8	3-5/8	1-5/8	2-1/8	2-5/8	1-3/8	1-3/8	1-5/8	1-1/8	1-1/8	1-3/8	5/8	3/4	3/4
10	3-1/8	3-1/8	3-5/8	2-1/8	2-1/8	2-5/8	1-3/8	1-5/8	2-1/8	1-1/8	1-3/8	1-3/8	5/8	3/4	7/8
15	3-5/8	3-5/8	5-1/8	2-1/8	2-5/8	3-1/8	1-5/8	2-1/8	2-1/8	1-3/8	1-3/8	1-5/8	3/4	7/8	1-1/8
20	3-5/8	4-1/8	5-1/8	2-5/8	3-1/8	3-1/8	2-1/8	2-1/8	2-5/8	1-3/8	1-5/8	2-1/8	7/8	1-1/8	1-1/8
25	4-1/8	5-1/8	5-1/8	2-5/8	3-1/8	3-5/8	2-1/8	2-1/8	2-5/8	1-5/8	2-1/8	2-1/8	7/8	1-1/8	1-1/8
30	4-1/8	5-1/8	6-1/8	3-1/8	3-1/8	3-5/8	2-1/8	2-5/8	2-5/8	1-5/8	2-1/8	2-1/8	1-1/8	1-1/8	1-3/8
40	5-1/8	6-1/8	6-1/8	3-1/8	3-5/8	4-1/8	2-5/8	2-5/8	3-1/8	2-1/8	2-5/8	2-5/8	1-1/8	1-3/8	1-3/8

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CHART 19-1 R-134a VAPOR PRESSURE DROP IN COPPER TUBING. REPRINTED BY PERMISSION FROM *ASHRAE* JOURNAL, APRIL 1990.

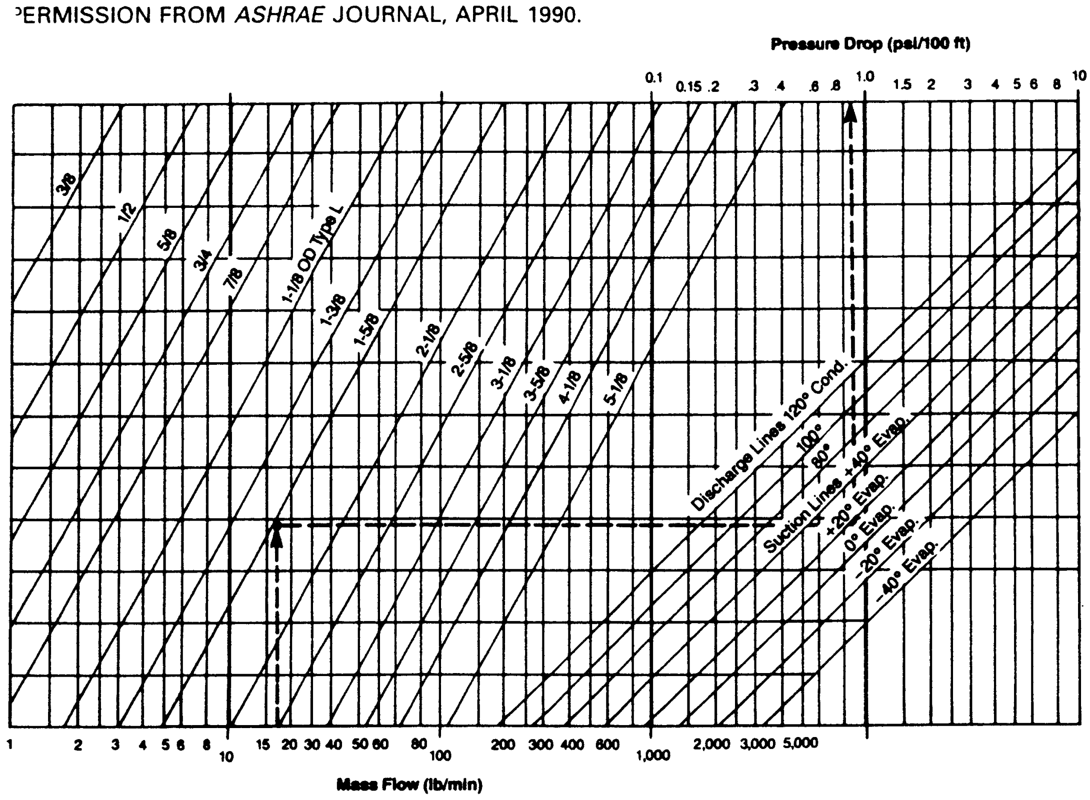

CHART 19-2 R-134a VAPOR VELOCITIES IN COPPER TUBING. REPRINTED BY PERMISSION FROM *ASHRAE* JOURNAL, APRIL 1990.

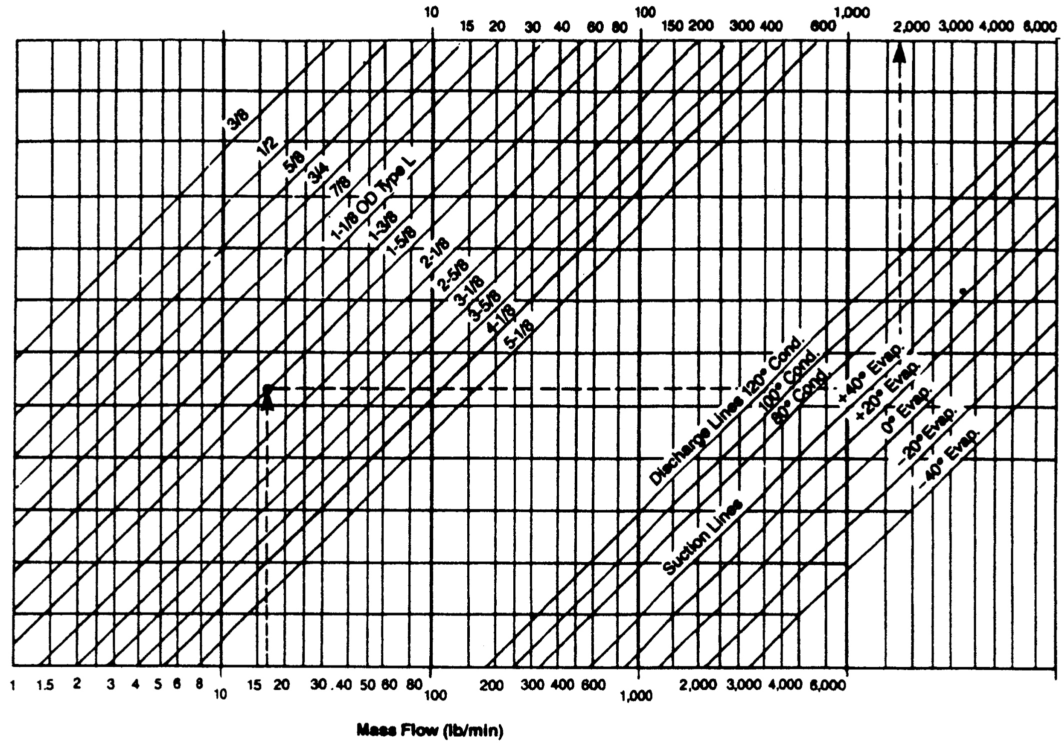

CHART 19-3 VELOCITY AND PRESSURE DROP FOR R-134a BASIS 90°F LIQUID IN COPPER TUBING. REPRINTED BY PERMISSION FROM *ASHRAE* JOURNAL, APRIL 1990.

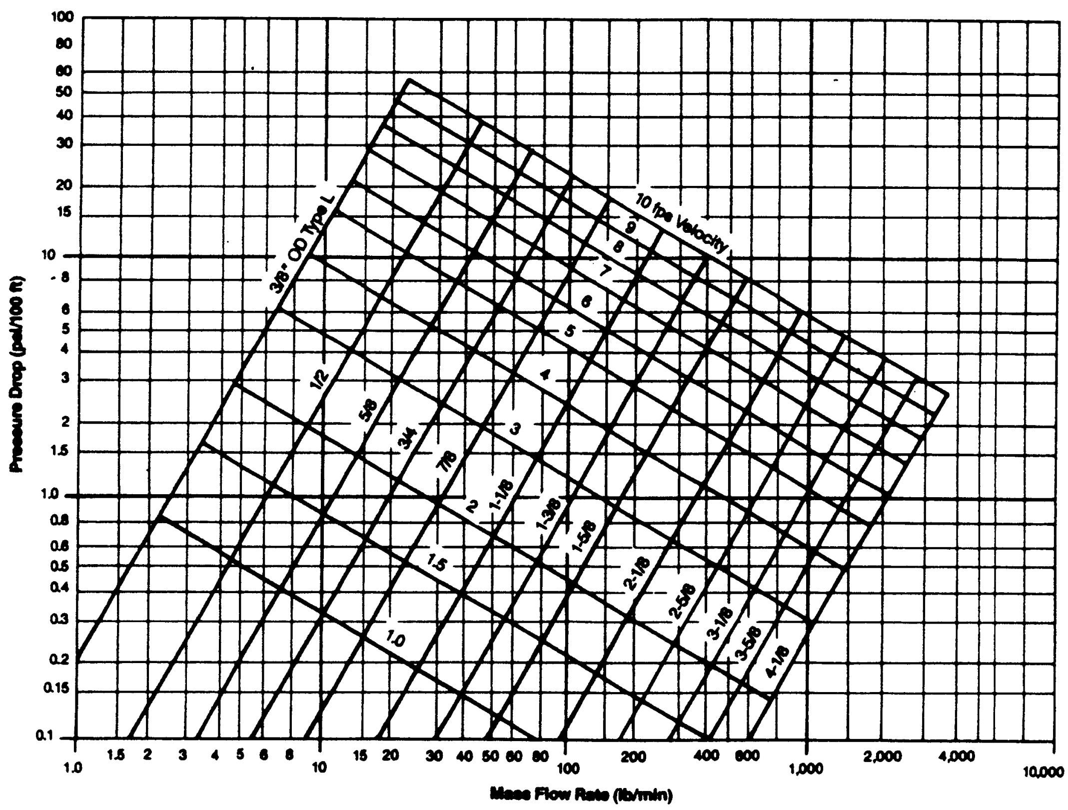

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CHART 19-4 MASS FLOW PER TON OF REFRIGERATION FOR R-134a. REPRINTED BY PERMISSION FROM *ASHRAE* JOURNAL, APRIL 1990.

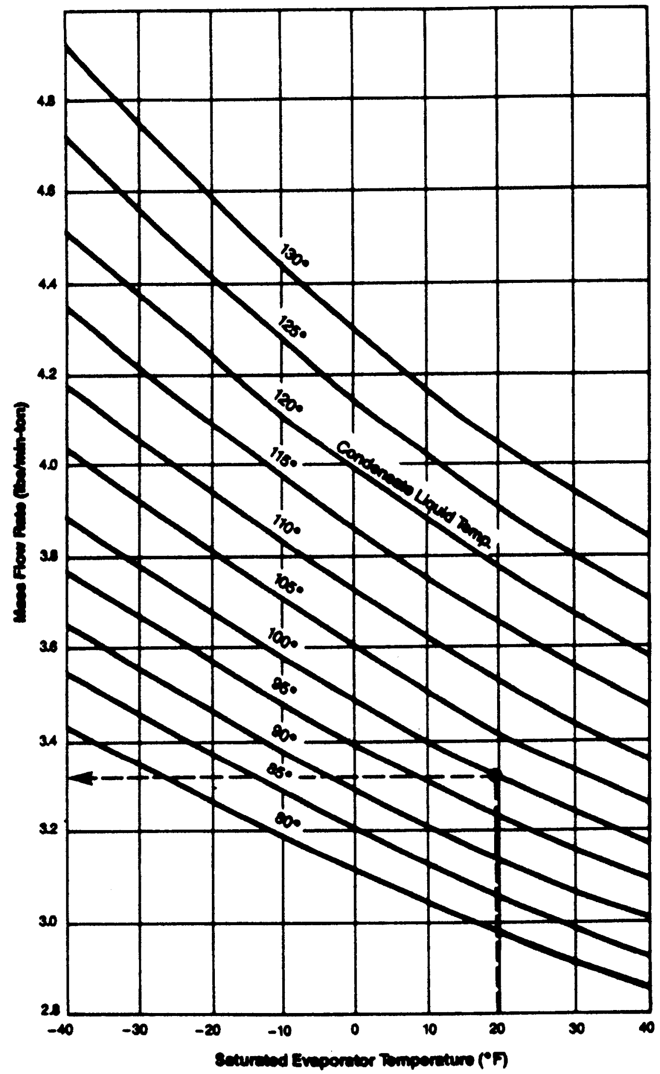

TABLE 19-7 REFRIGERANT LINE CAPACITIES FOR INTERMEDIATE- OR LOW-STAGE DUTY (TONS) FOR R-12, R-22, AND AMMONIA

Refrigerant and ΔT Equivalent of Friction Drop*	Line Size Type L Copper OD	Suction Lines* Suction Temp F −90	−80	−70	−60	−50	−40	−30	Discharge Lines*	Line Size Type L Copper OD	Liquid Lines: Condenser to Receiver V = 100 fpm	Liquid Lines: Receiver to System*
	1/2									1/2		2.1
Refrigerant 12	5/8									5/8		3.9
	7/8				0.2	0.3	0.4	0.5	0.9	7/8		11.0
	1 1/8	0.17	0.24	0.3	0.4	0.6	0.8	1.0	1.8	1 1/8		21.5
	1 3/8	0.30	0.42	0.6	0.8	1.1	1.4	1.7	3.2	1 3/8		37.0
	1 5/8	0.47	0.67	0.9	1.2	1.7	2.2	2.7	5.0	1 5/8		60.0
	2 1/8	1.00	1.40	1.9	2.5	3.5	4.6	5.7	10.5	2 1/8		125.0
2 F ΔT Per 100 ft	2 5/8	1.7	2.4	3.3	4.5	6.1	8.0	10.0	18.5	2 5/8		220.0
Equiv. Length	3 1/8	2.8	3.9	5.4	7.3	10.0	13.0	16.2	30.0	3 1/8	See Table 19-2	350.0
	3 5/8	4.1	5.9	8.2	10.8	15.0	19.5	24.3	44.0	3 5/8		
	4 1/8	6.0	8.5	11.7	15.6	21.5	28.0	35.0	65.0	4 1/8		
	5 1/8	10.6	15.1	20.8	27.8	38.5	50.0	62.5	113.0	5 1/8		
	6 1/8	18.1	25.8	35.4	47.2	65.4	85.0	106.0	180.0	6 1/8		
	1/2									1/2		3.6
Refrigerant 22	5/8								0.6	5/8		7.0
	7/8	0.16	0.23	0.31	0.44	0.57	0.75	0.94	1.5	7/8		18.0
	1 1/8	0.34	0.48	0.65	0.91	1.19	1.55	1.93	3.0	1 1/8		36.0
	1 3/8	0.59	0.81	1.12	1.59	2.07	2.7	3.4	5.2	1 3/8		63.0
	1 5/8	0.93	1.34	1.8	2.5	3.3	4.3	5.4	8.5	1 5/8		100.0
	2 1/8	1.9	2.8	3.7	5.2	6.8	8.9	11.1	17.5	2 1/8		210.0
2 F ΔT Per 100 ft	2 5/8	3.5	5.0	6.6	9.4	12.3	16.0	20.0	31.0	2 5/8		375.0
Equiv. Length	3 1/8	5.5	8.0	10.6	15.0	19.6	25.5	32.0	50.0	3 1/8	See Table 19-3	
	3 5/8	8.4	12.0	16.0	22.6	29.5	38.5	48.0	75.0	3 5/8		
	4 1/8	12.0	17.2	22.9	32.3	42.3	55.0	68.8	105.0	4 1/8		
	5 1/8	21.2	30.6	41.0	57.5	75.0	98.0	122.0	190.0	5 1/8		
	6 1/8	34.8	50.0	66.5	94.0	123.0	160.0	200.0	305.0	6 1/8		
	Steel IPS SCH				−60	−50	−40	−30		Steel IPS SCH		
										3/8 80		17.0
	1/2 40				0.26	0.38	0.50	0.62	1.0	1/2 80		34.0
	3/4 40				0.55	0.76	1.05	1.30	2.1	3/4 80		75.0
	1 40				1.05	1.53	2.00	2.50	4.1	1 80		150.0
Refrigerant 717	1 1/4 40				2.15	3.15	4.10	5.10	8.5	1 1/4 80		305.0
(Ammonia)	1 1/2 40				3.4	5.0	6.5	8.1	12.5	1 1/2 80		490.0
	2 40				6.3	9.2	12.0	15.0	25.0	2 40	See Table 19-4	
	2 1/2 40				10.3	15.0	19.5	24.3	40.0	2 1/2 40		
1 F ΔT Per 100 ft	3 40				18.4	26.8	35.0	43.7	71.0	3 40		
Equiv. Length	3 1/2 40				27.3	39.8	52.0	65.0	105.0	3 1/2 40		
	4 40				37.8	55.2	72.0	90.0	145.0	4 40		
	5 40				68.3	100.0	130.0	162.0	260.0	5 40		
	6 40				110.0	161.0	210.0	262.0	425.0	6 40		
	8 40				258.0	376.0	490.0	610.0		8 40		

NOTES:

* *(1) Values in this table are tons of refrigeration resulting in a line friction drop per 100 ft of equivalent pipe length corresponding to the (ΔT) change in saturation temp indicated in the left hand column under the refrigerant designation.*

(2) Values based on 0 F saturated discharge temp. For capacities at other saturated discharge temp, multiply table value by proper line capacity multiplier:

Sat. Discharge Temp, F	Line Capacity Multipliers: Refrigerant 12 Suction	Refrigerant 12 Discharge	Refrigerant 22 Suction	Refrigerant 22 Discharge	Ammonia Discharge
−30	1.12	0.55	1.09	0.58	
−20	1.07	0.70	1.06	0.71	
−10	1.03	0.85	1.03	0.85	0.77
0	1.00	1.00	1.00	1.00	1.00
10	0.96	1.25	0.97	1.20	1.23
20	0.93	1.50	0.94	1.45	1.45
30	0.90	1.80	0.90	1.80	1.67

(3) For other ΔT's and Equivalent Lengths,

$$\text{Line Capacity (Tons)} = \text{Table Tons} \times \left(\frac{100}{\text{Actual Equiv. Length, ft}} \times \frac{\text{Actual } \Delta T \text{ Loss Desired, F}}{\text{Table } \Delta T \text{ Loss, F}} \right)^{0.55}$$

(4) For other Tons and Equivalent Lengths in a given pipe size,

$$\Delta T(F) = \text{Table } \Delta T \times \frac{\text{Actual Equiv. Length, ft}}{100} \times \left(\frac{\text{Actual Tons}}{\text{Table Tons}} \right)^{1.8}$$

(5) Values obtained from Carrier Corp. data.

* From *ASRE Data Book,* Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

Example 19-2 A 5-ton system employing R-134a is operating with an evaporating temperature of 20°F and a condensing temperature of 100°F. If the equivalent length of the suction pipe is 125 feet, determine

(a) the size of the suction pipe using the smallest size of type L copper tube wherein the pressure loss in the pipe will not result in more than a 2°F drop in saturation temperature,

(b) the velocity of the suction vapor in feet per minute.

Solution From Table 19-6, for a 5-ton system and a 100-foot length of pipe, the suggested pipe size is $2\frac{1}{8}$ in. OD at a 0°F evaporating temperature and $1\frac{3}{8}$ in. OD at a 40°F evaporating temperature. Since a 20°F evaporating temperature will fall midway between these two temperatures, try using a pipe size of $1\frac{5}{8}$ in. OD, which is a pipe size midway between the two pipe sizes.

From Chart 19-4, using a 20°F evaporating temperature and a 100°F condensing temperature, note that the refrigerant mass flow rate will be 3.3 lb/min/ton, so that a 5-ton system will require 16.5 lb/min (5 × 3.3). Entering Chart 19-1 with the mass flow rate of 16.5 lb/min, move vertically up to the selected pipe size of $1\frac{5}{8}$ in. OD, then horizontally across to the evaporating temperature of 20°F, then vertically up again to the top of the chart, and read the pressure loss as 0.9 psi per 100 feet of pipe. The total pressure loss in the suction pipe will be 1.13 psi (0.9×1.25).

On examining the pressure–temperature relationship of R-134a in Table 16-5 it will be found that in the 20°F temperature range the pressure change per degree of temperature change is approximately 0.7 psi, so that for a 2°F drop in saturation temperature the pressure loss would be approximately 1.4 psi. Since the pressure loss in the selected pipe size is only 1.13 psi, the size selected meets the stated requirements.

Entering the bottom of Chart 19-2 with the mass flow rate of 16.5 lb/min, move vertically up to the selected pipe size of $1\frac{5}{8}$ in. OD, then horizontally across to the evaporator temperature of 20°F, then vertically up again to the top of the chart, and read the vapor velocity as 1850 fpm.

Example 19-3 A 40-ton, R-12 system has an evaporator temperature of 20°F and a condensing temperature of 110°F. If a suction pipe 30 ft long containing three standard elbows is required, determine the following:

(a) the size of type L copper tubing required,

(b) the overall pressure drop in the suction line in pounds per square inch.

Suction line saturation temperature loss in °F,

$$\text{Table } \Delta T \times \left(\frac{L_e}{100}\right) \times \left(\frac{\text{Actual tons}}{\text{Table tons}}\right)^{1.8}$$

$$= 2°\text{F} \times \left(\frac{54}{100}\right) \times \left(\frac{40}{31.96}\right)^{1.8}$$

$$= 2°\text{F} \times 0.54 \times 1.5$$

$$= 1.62°\text{F}$$

Pressure drop equivalent of saturation temperature loss,

$$\text{Table } \Delta P \times \left(\frac{\text{Actual } \Delta T}{\text{Table } \Delta T}\right)$$

$$= 1.38 \text{ psi} \times \left(\frac{1.62°\text{F}}{2°\text{F}}\right)$$

$$= 1.1 \text{ psi}$$

Solution

(a) From Table 19-2, $3\frac{1}{8}$ in. OD copper tubing has a capacity of 34 tons based on a condensing temperature of 100°F and a suction line pressure loss equivalent to 2°F per 100 ft of pipe. Since the pressure loss is proportional to the length of pipe and since the length of pipe is relatively short in this instance, this pipe size may be sufficient, and a trial calculation should be made. From Table 15-1, $3\frac{1}{8}$ in. OD (3 in. nominal) standard elbows have an equivalent length of 8 ft.

Actual equivalent length of suction piping:

Straight pipe length	= 30
3 ells at 8 ft	= 24 ft
Total equivalent length	= 54 ft

(b) Correction factor from Table 19-2 to correct tonnage for 110°F condensing temperature is 0.94.

$$\text{Corrected tonnage} = 34 \times 0.94 = 31.96 \text{ tons}$$

Example 19-4 An R-12 system with a condensing temperature of 100°F has a capacity of 35 tons. The equivalent length of the liquid line including fittings and accessories is 60 ft. If the line contains a 20-ft riser, determine the following:

(a) the size of the liquid line required,
(b) the overall pressure drop in the line,
(c) the amount of subcooling (°F) required to prevent flashing of the liquid.

Solution

(a) From Table 19-2, $1\frac{3}{8}$ in. OD copper tubing has a capacity of 35.1 tons based on a 1.8 psi pressure drop per 100 equivalent feet of pipe.

(b)

For 60 ft equivalent length, the friction loss in the pipe	= 1.8 psi × 0.6
	= 1.00 psi
From Table 16-3, the density of 100°F liquid	= 78.8 lb/ft^3
Pressure loss per foot of lift	= 78.8/144 = 0.55
Static pressure loss	= (0.55 psi/ft) (20 ft)
	= 11.0 psi
Overall pressure loss in liquid line	= 1 psi + 11 psi
	= 12.0 psi

(c) Assuming the condensing temperature to be 100°F, the pressure at the condenser is 131.6 psia. The pressure at the refrigerant control is 119.6 psia, which corresponds to a saturation temperature of approximately 93°F. The amount of subcooling required is approximately 7°F (100° − 93°).

General Design of Suction Piping

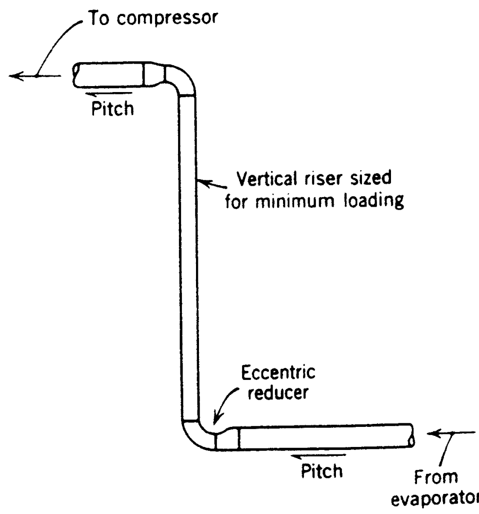

Fig. 19-2 Illustrating method of reducing the size of a vertical suction riser. (Courtesy of York Corporation.)

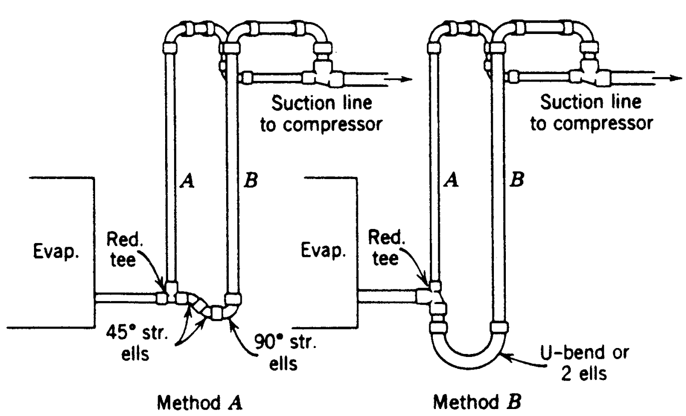

Fig. 19-3 Double suction riser construction. (Courtesy of Carrier Corporation.)

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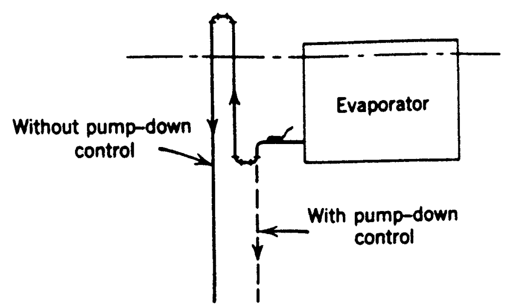


Fig. 19-4 Evaporator located above compressor.

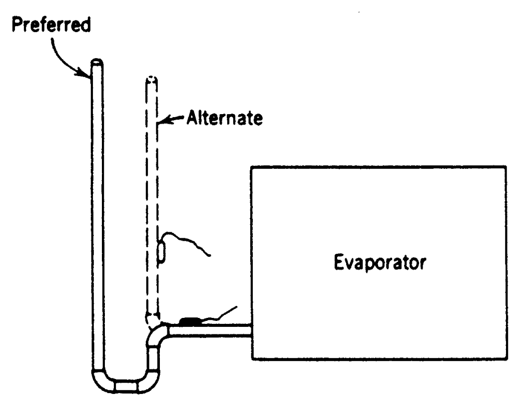


Fig. 19-5 Evaporator below compressor.

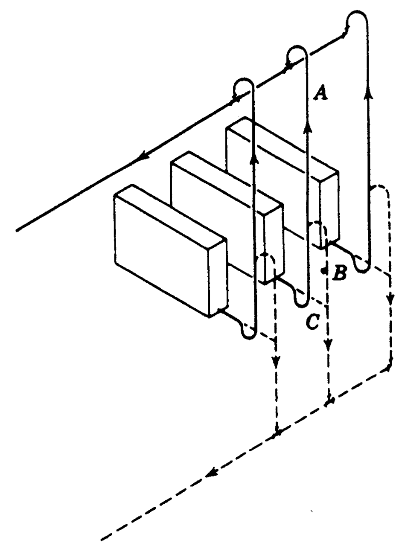


Fig. 19-6 Multiple evaporators, individual suction lines.

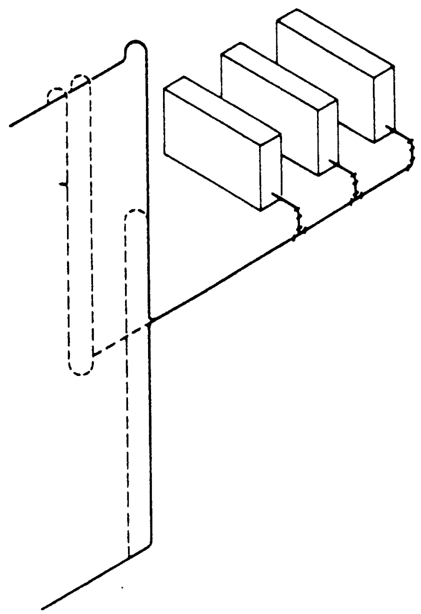


Fig. 19-7 Multiple evaporators, common suction line.

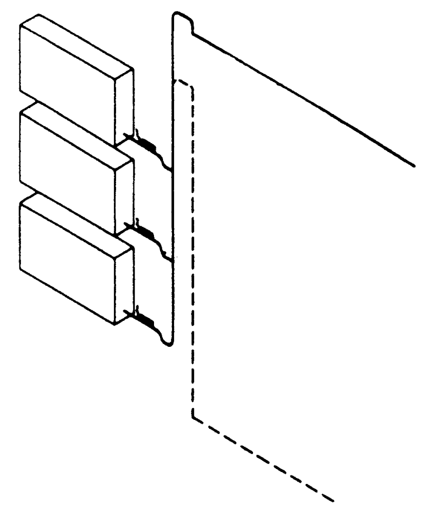


Fig. 19-8 Evaporators at different levels connected to a common suction riser.

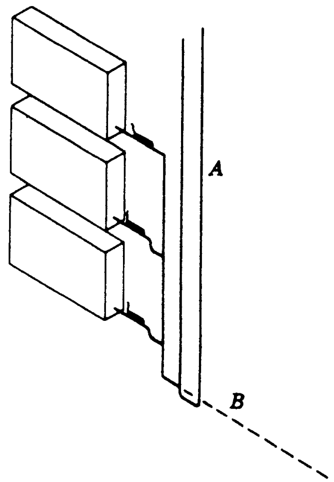


Fig. 19-9 Evaporators at different levels connected to a double suction riser.

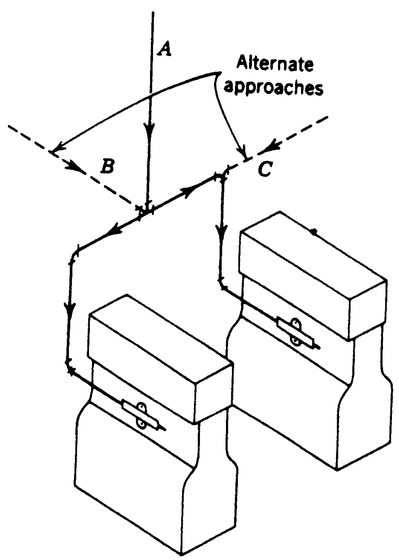


Fig. 19-10 Suction piping for compressors connected in parallel.

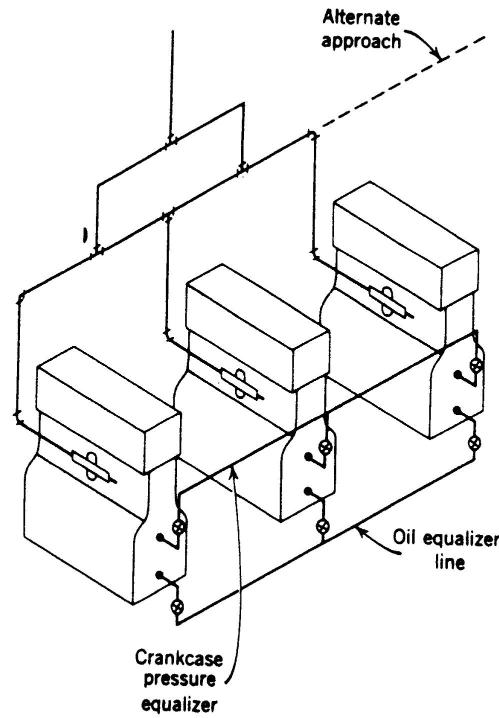


Fig. 19-11 Suction piping for compressors connected in parallel.

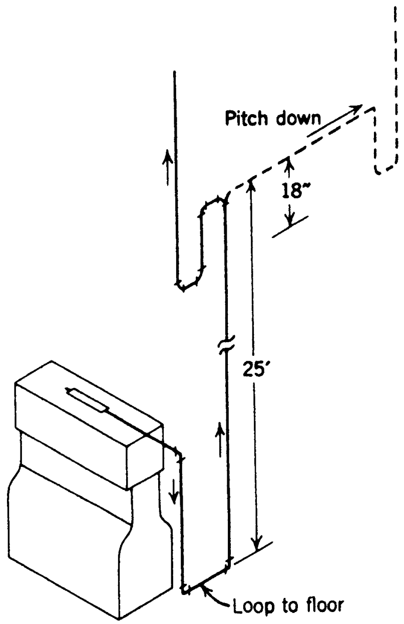


Fig. 19-13 Piping of discharge riser.

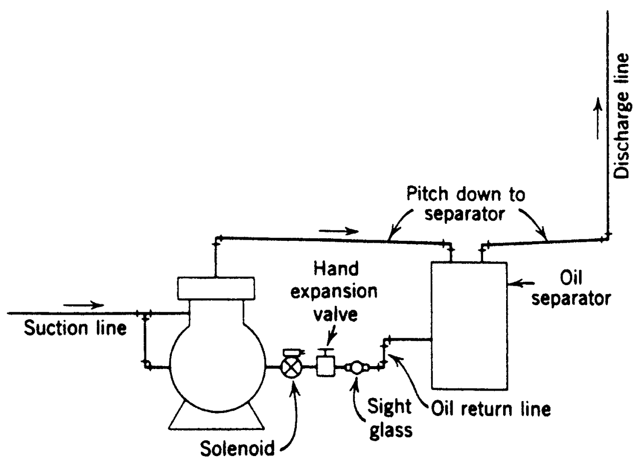


Fig. 19-12 Arrangement for preventing liquid return to compressor crankcase.

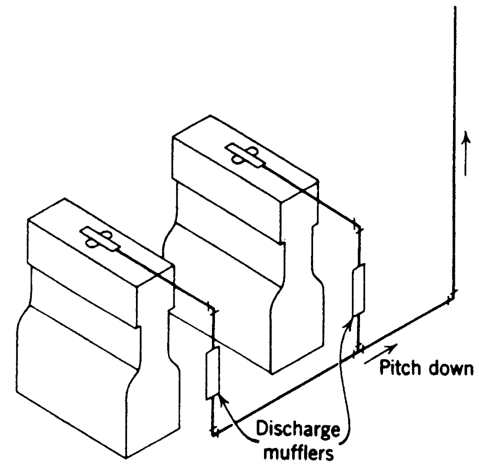


Fig. 19-14 Discharge piping of multiple compressors connected in parallel.

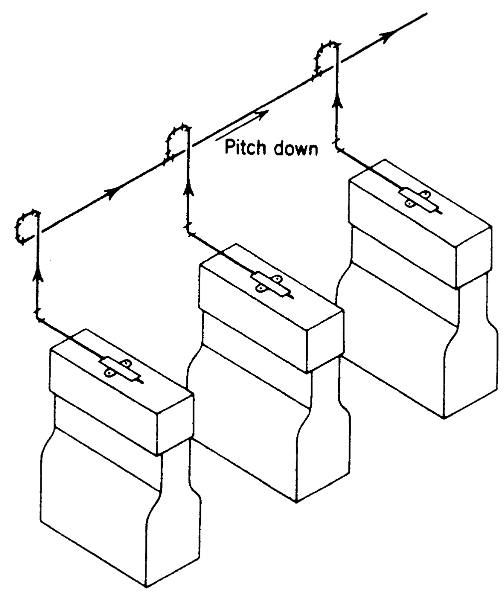


Fig. 19-15 Discharge piping for multiple compressors connected in parallel.

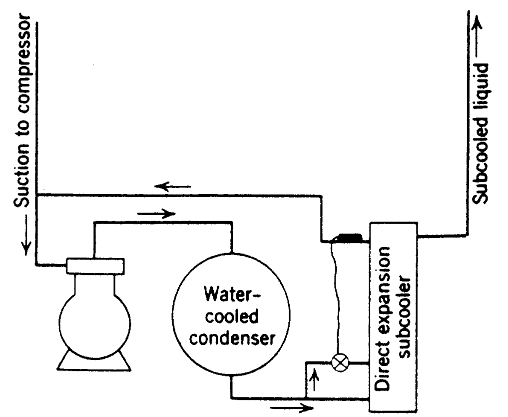


Fig. 19-16 Subcooling liquid refrigerant with direct expansion subcooler.

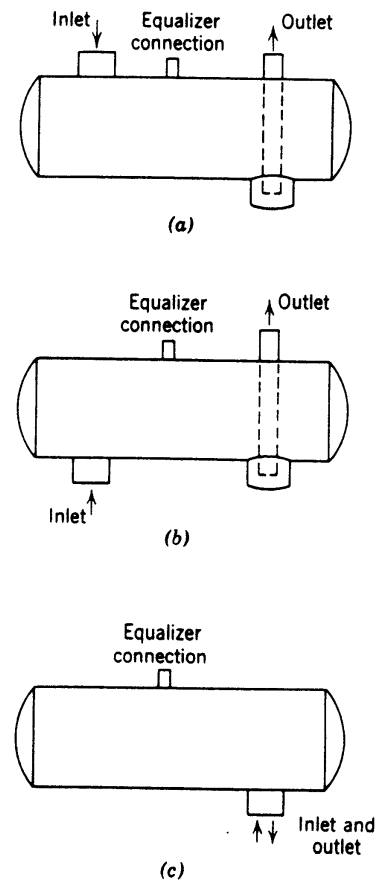


Fig. 19-17 (*a*) Top inlet through-flow receiver. (*b*) Bottom inlet through-flow receiver. (*c*) Surge-type receiver.

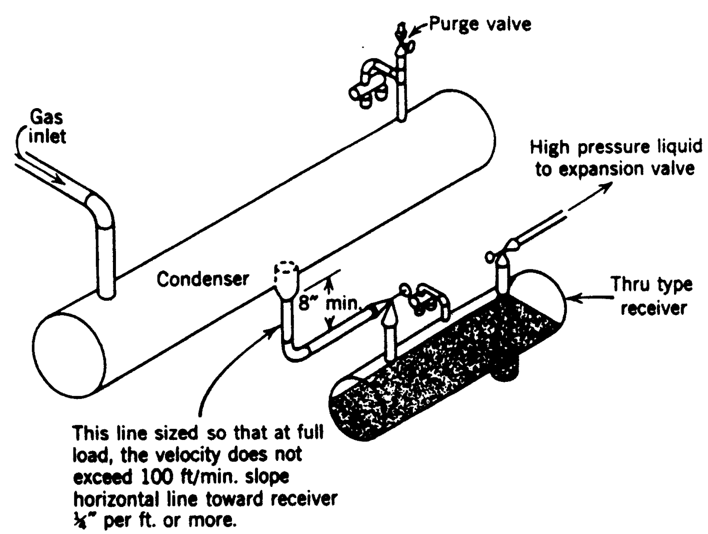
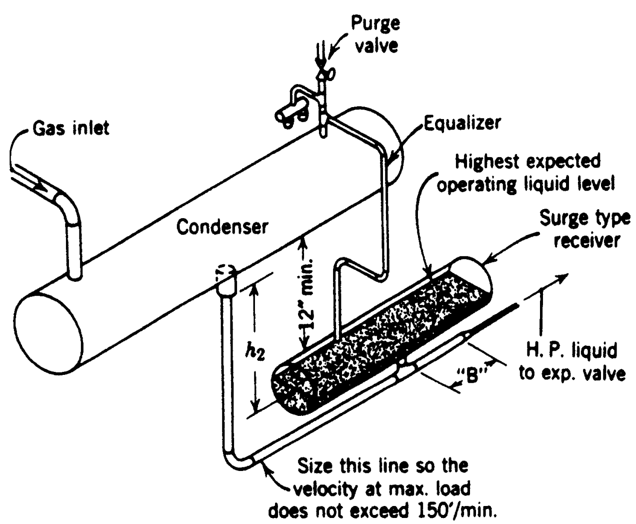


Fig. 19-18 Top inlet through-type receiver hookup. (Courtesy of York Corporation.)



Maximum Velocity of Drain line, lbs/min	Type Valve between Condenser and Receiver	h_2 Required inches
150	None	14
150	Angle	16
150	Globe	28
100	None, Angle, or Globe	14

Size drain line to receiver for maximum velocity of 150 ft/min. If a valve is located in this line, the trapping height limitation may require a larger size line to minimize the pressure drop.

Fig. 19-19 Surge-type receiver hookup. (Courtesy of York Corporation.)

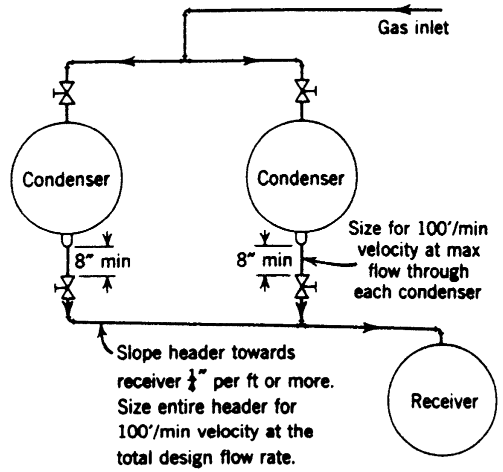
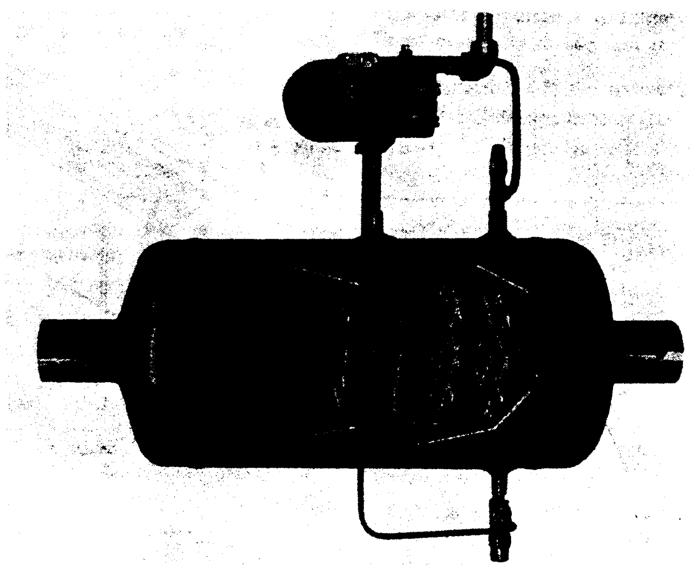


Fig. 19-20 Parallel shell-and-tube condensers with top inlet receiver. (Courtesy of York Corporation.)



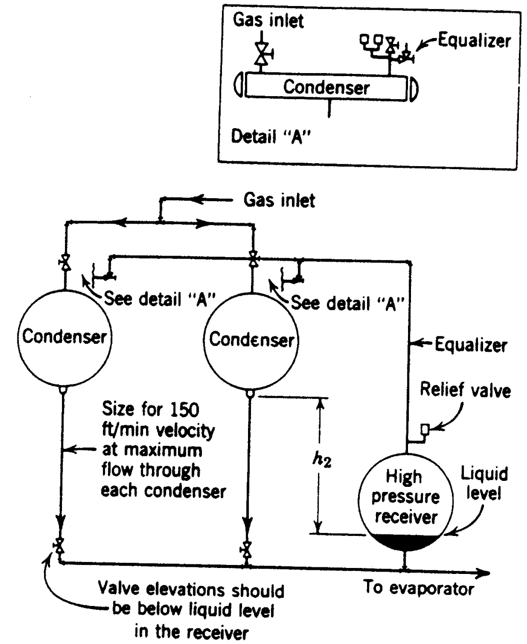


Fig. 19-21 Parallel shell-and-tube condensers with bottom inlet receiver. (Courtesy of York Corporation.)

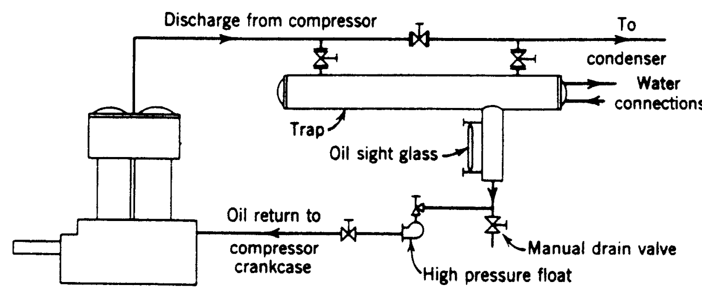


Fig. 19-23 Application of chiller-type oil separator. (Courtesy of York Corporation.)

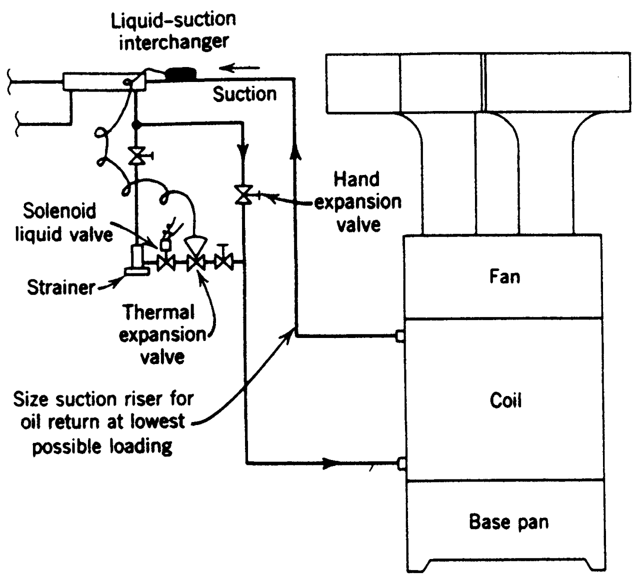


Fig. 19-24 Forced circulation air cooler with direct-expansion feed of flooded-type coil. (Courtesy of Carrier Corporation.)

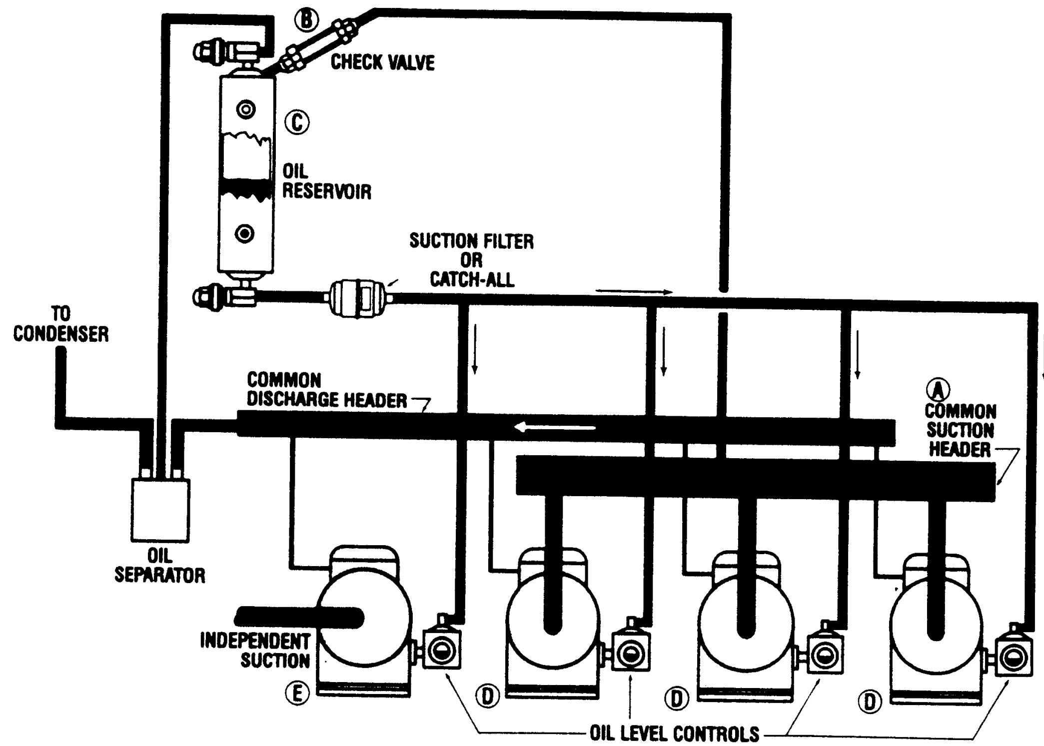


Fig. 19-25 A multicompressor system employing oil level controls. (Courtesy of Sporlan Valve Company.)

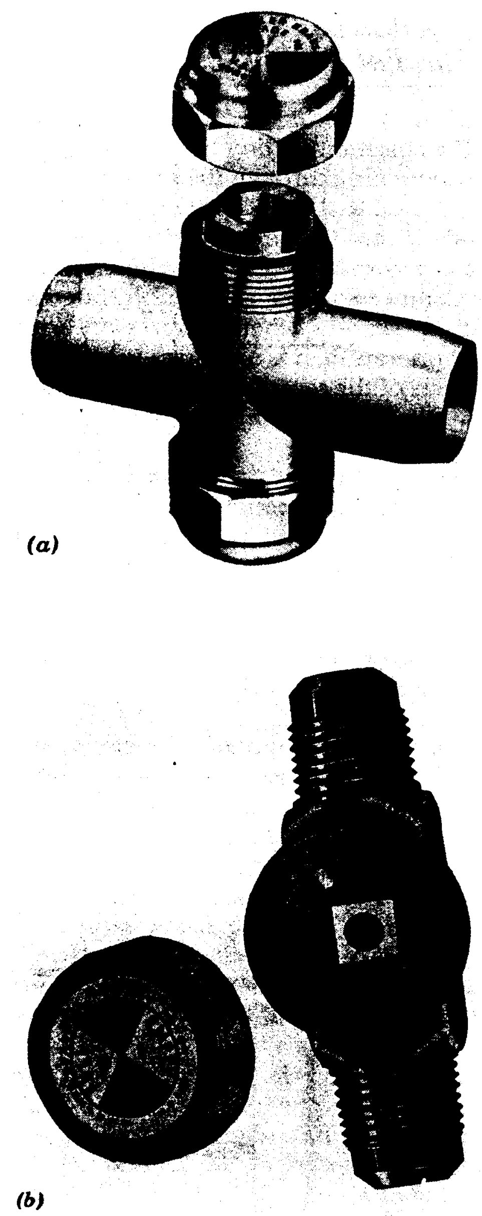


Fig. 19-26 Typical liquid indicators or sight glasses. Notice moisture indicator incorporated in single-port sight glass. The color of the moisture indicator denotes the relative moisture content of the system. (*a*) Double-port sight glass. (*b*) Single-port sight glass. (Courtesy of Mueller Brass Company.)

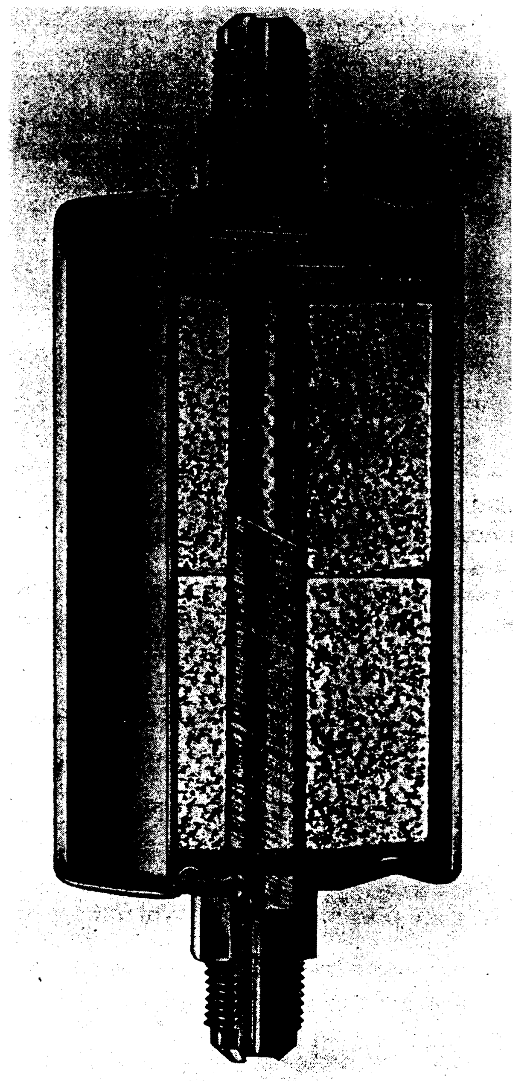


Fig. 19-27 Straight-through, nonrefillable drier. (Courtesy of Mueller Brass Company.)

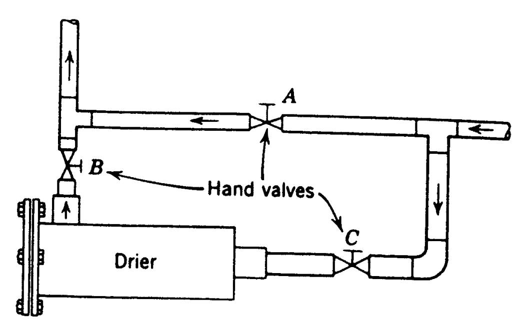


Fig. 19-28 Side outlet drier installed in bypass line.

34.

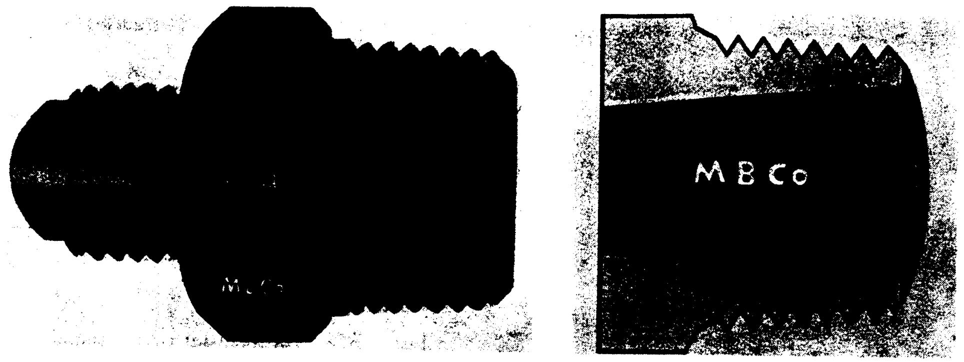
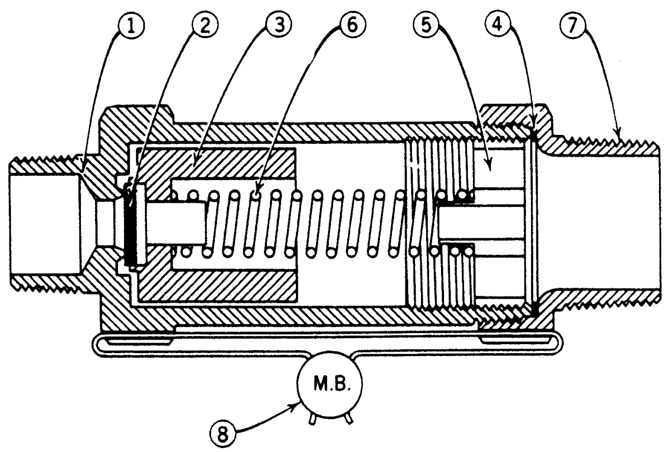


Fig. 19-30 Fusible plugs. (Courtesy of Mueller Brass Company.)



1. Valve body
2. Seat disc
3. Disc holder
4. Gasket
5. Spring retainer
6. Spring
7. Outlet connection
8. Lead seal and locking wire (prevents alteration of factory setting)

Fig. 19-29 Typical relief valve. (Courtesy of Mueller Brass Company.)

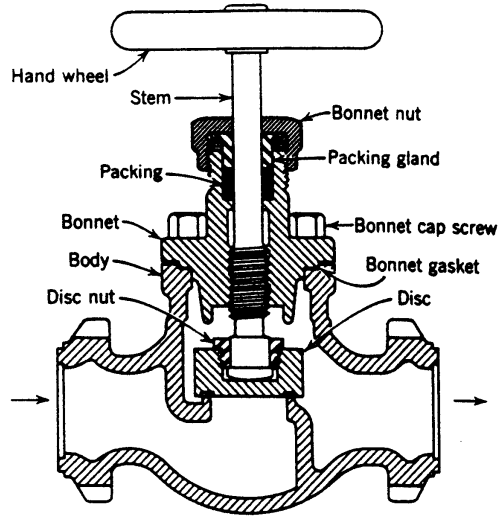


Fig. 19-33 Packed-type manual valve. (Courtesy of Vilter Manufacturing Company.)

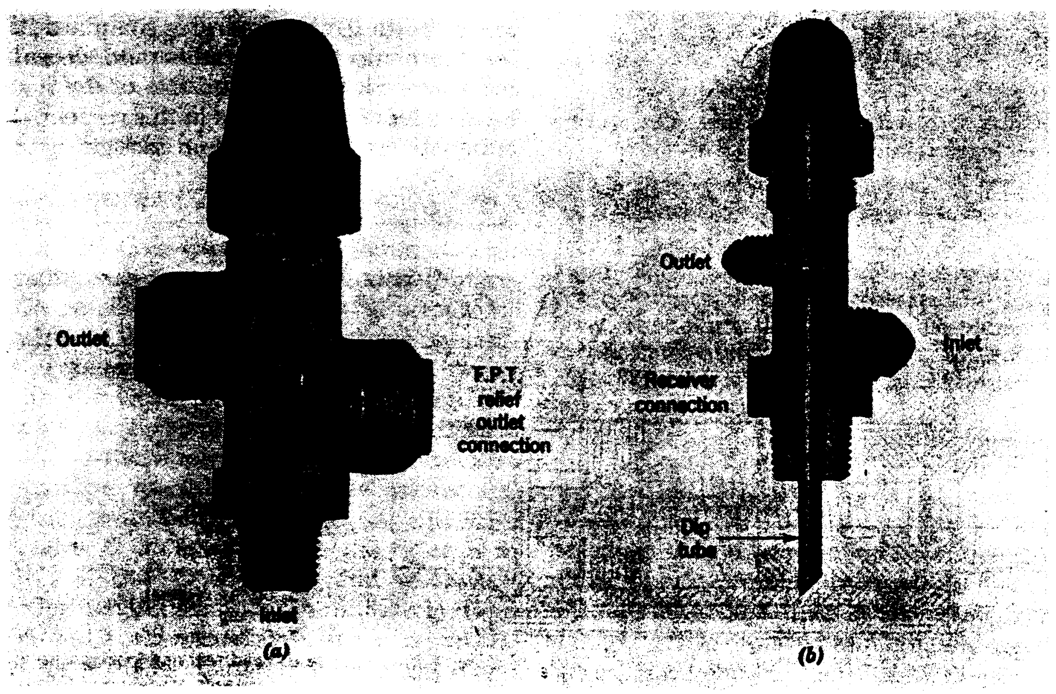


Fig. 19-31 Receiver tank valves. (*a*) Angle type with pressure relief outlet (nonback-seating type). (*b*) Angle type with dip tube (nonback-seating type). (Courtesy of Mueller Brass Company.)

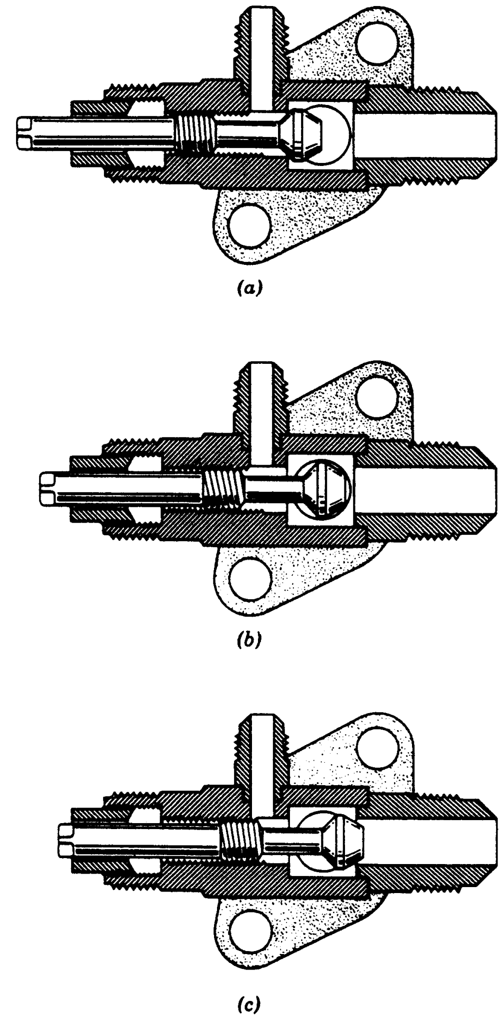


Fig. 19-32 Compressor service valve. (*a*) Back-seated. (*b*) Intermediate position. (*c*) Front-seated.

Refrigeration Engineering

-Condenser

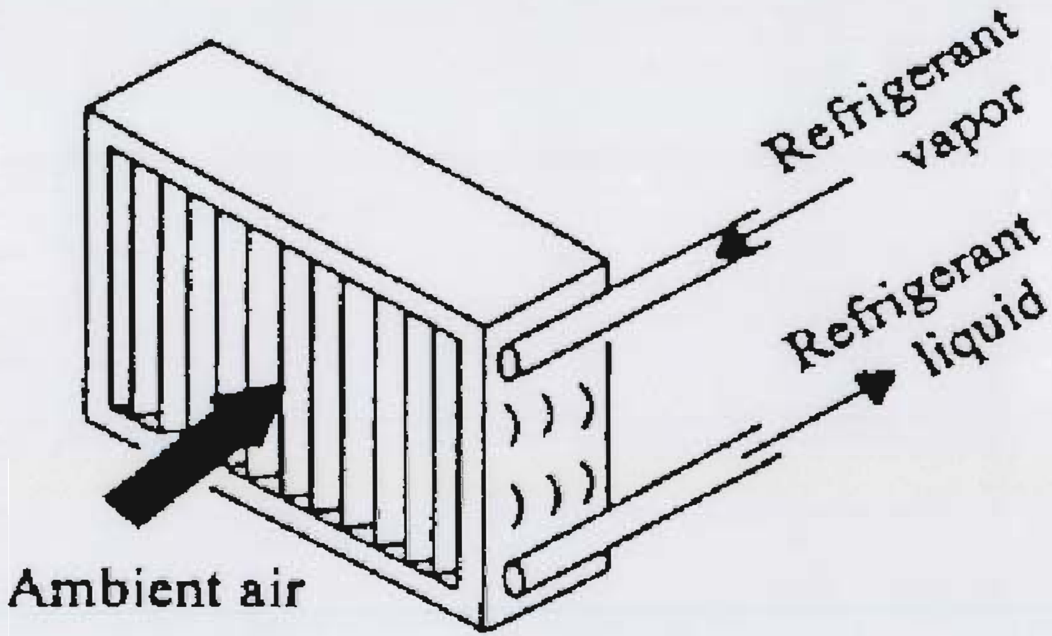
李魁鵬

■ Types of condenser:

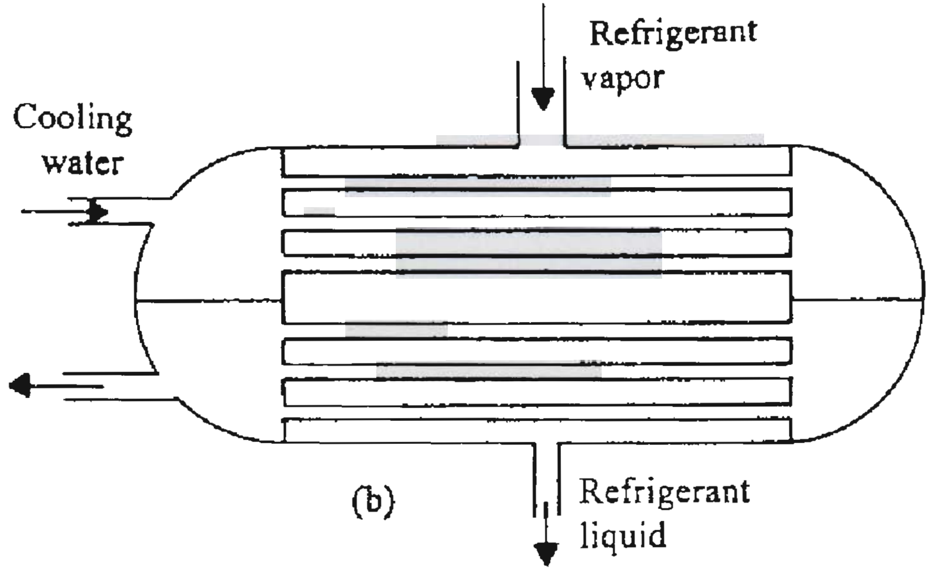
The three main types of condensers used in general refrigeration systems are:

air-cooled, water-cooled, evaporative

- air-cooled

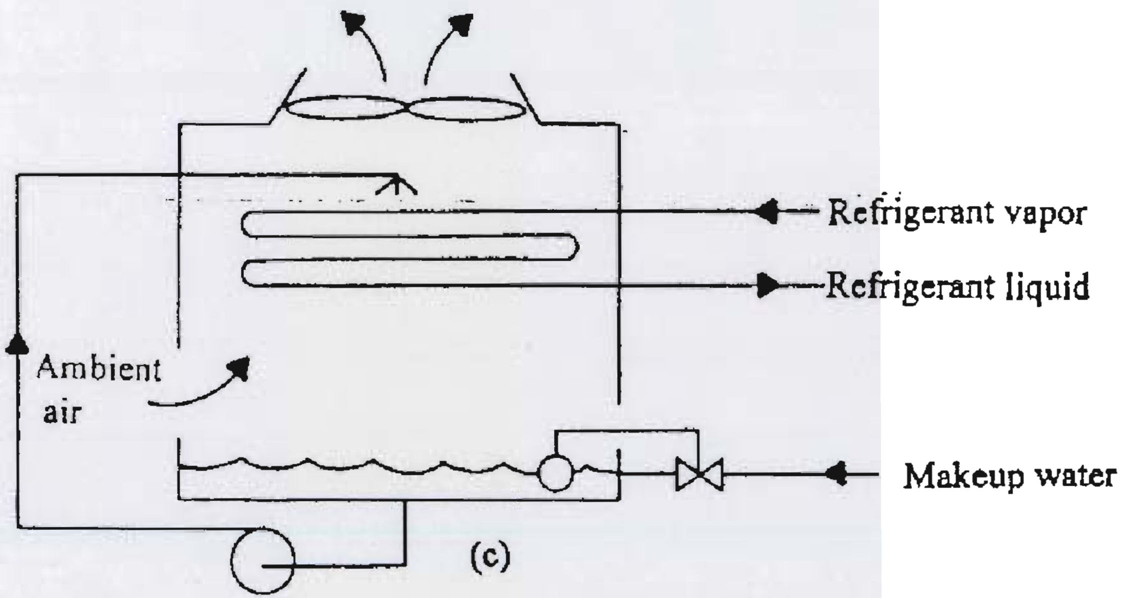


(a)



(b)

b) water-cooled



(c)

c) evaporative

■ The condensing process: a cold vertical surface

$$h_c = 0.943\left(\frac{g\rho^2 h_{fg} k^3}{\mu \Delta t L}\right)^{1/4} \quad (7.1)$$

where h_c = mean condensing coefficient, $W/m^2 \cdot °C$ ($Btu/hr \cdot ft^2 \cdot °F$)

g = acceleration due to gravity = 9.81 m/s^2 (4.17×10^8 ft/hr^2)

ρ = density of condensate, kg/m^3 (lb/ft^3)

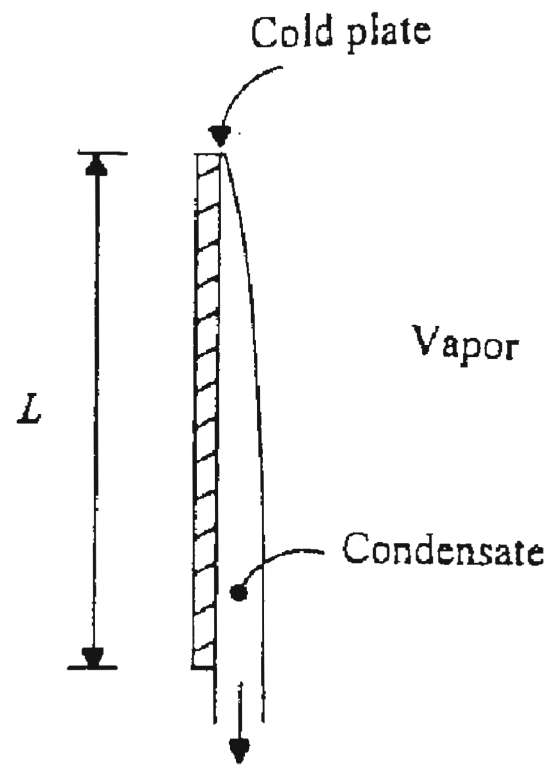
h_{fg} = latent heat of vaporization of the refrigerant, kJ/kg (Btu/lb)

k = conductivity of condensate, $W/m \cdot °C$ ($Btu/hr \cdot ft \cdot °F$)

μ = viscosity of condensate, $Pa \cdot s$ ($lb/ft \cdot hr$)

Δt = temperature difference, vapor to the plate, °C (°F)

L = vertical length of plate, m (ft)



■ The condensing process: a horizontal shell-and tube condenser

Average N tubes of diameter D in a vertical row:

$$h_c = 0.64 \left(\frac{g\rho^2 h_{fg} k^3}{\mu \Delta t N D} \right)^{1/4} \qquad (7.2)$$

TABLE 7.1
Condensing coefficients on the outside of tubes for several refrigerants. The condensing temperature is 30°C (86°F) and there are six 25-mm (1-in) tubes in a vertical row.

Refrigerant	Condensing coefficient	
	W/m²·°C	Btu/hr·ft²·°F
R-22	1142	201
R-134a	1046	184
Ammonia	5096	897

■ Condensation inside tubes

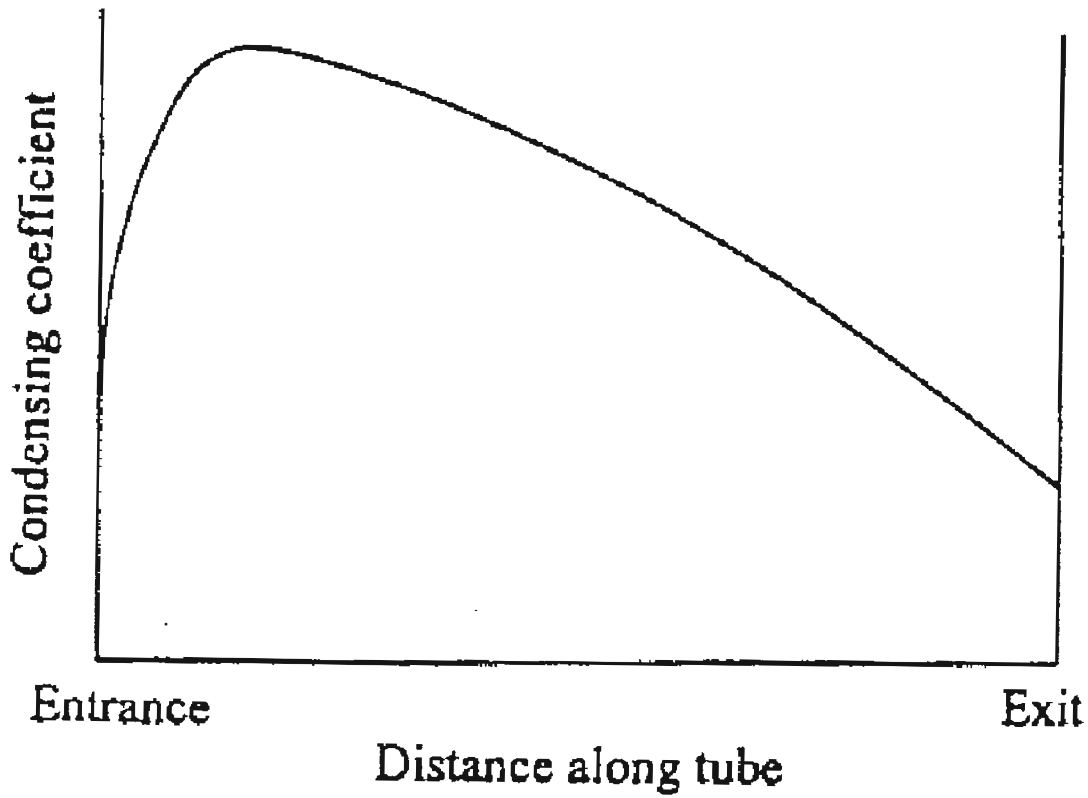


FIGURE 7.3
Variation in the condensing heat-transfer coefficient inside a tube.

Heat rejection ratio

$$\text{HRR} = \frac{\text{rate of heat rejection at condenser}}{\text{rate of heat absorbed at evaporator}}$$

$$\text{HRR} = \frac{\text{refrigerating capacity} + \text{compressor power}}{\text{refrigerating capacity}} \tag{7.3}$$

1) Carnot cycle:

$$\text{HRR} = \frac{T_{cond}}{T_{refrig}} \tag{7.4}$$

2) Improved expression:

$$\text{HRR} = \left(\frac{T_{cond}}{T_{refrig}}\right)^{1.7} \tag{7.5}$$

- Heat rejection ratio of open type compressors:

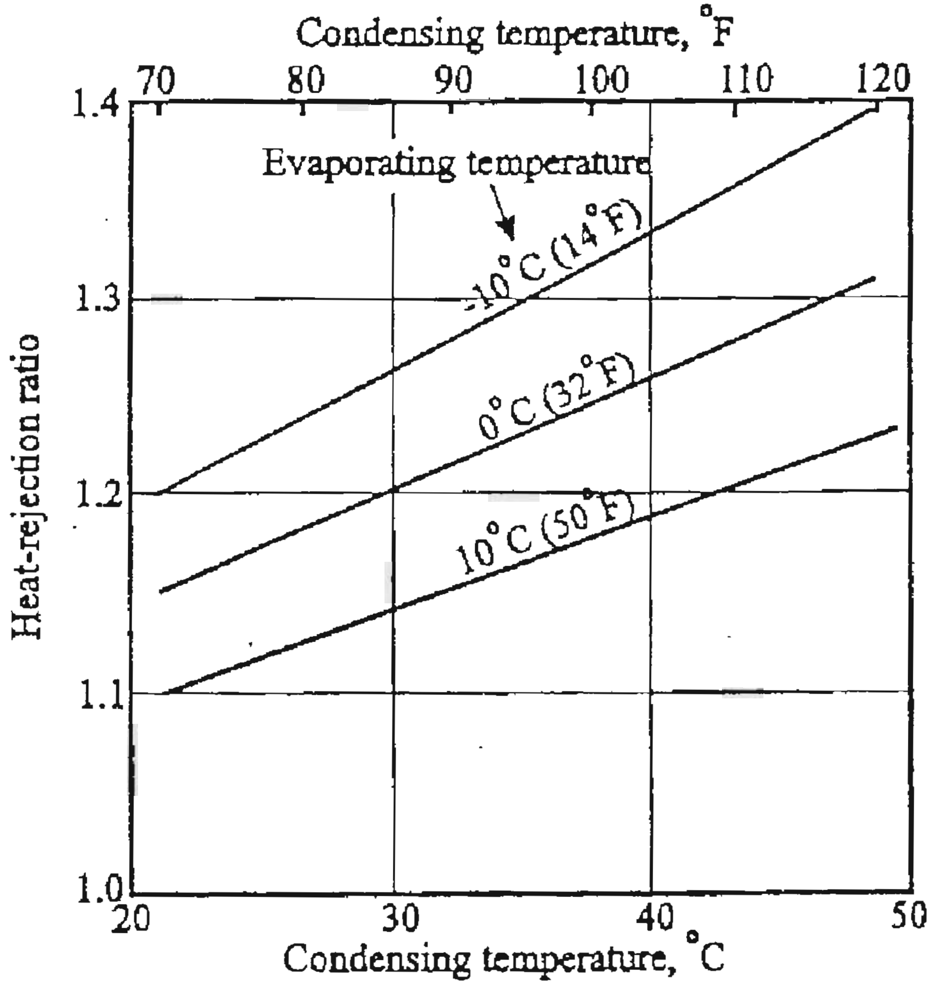


FIGURE 7.4
Typical values of the ratio of the heat rejected at the condenser to the refrigerating capacity, HRR, for ammonia and halocarbon refrigerants.

- Performance of air and water-cooled condensers

$$q = UA \left[\frac{t_o - t_i}{\ln\left(\frac{t_c - t_i}{t_c - t_o}\right)} \right] \quad (7.6)$$

where q = rate of heat transfer, kW (Btu/hr)

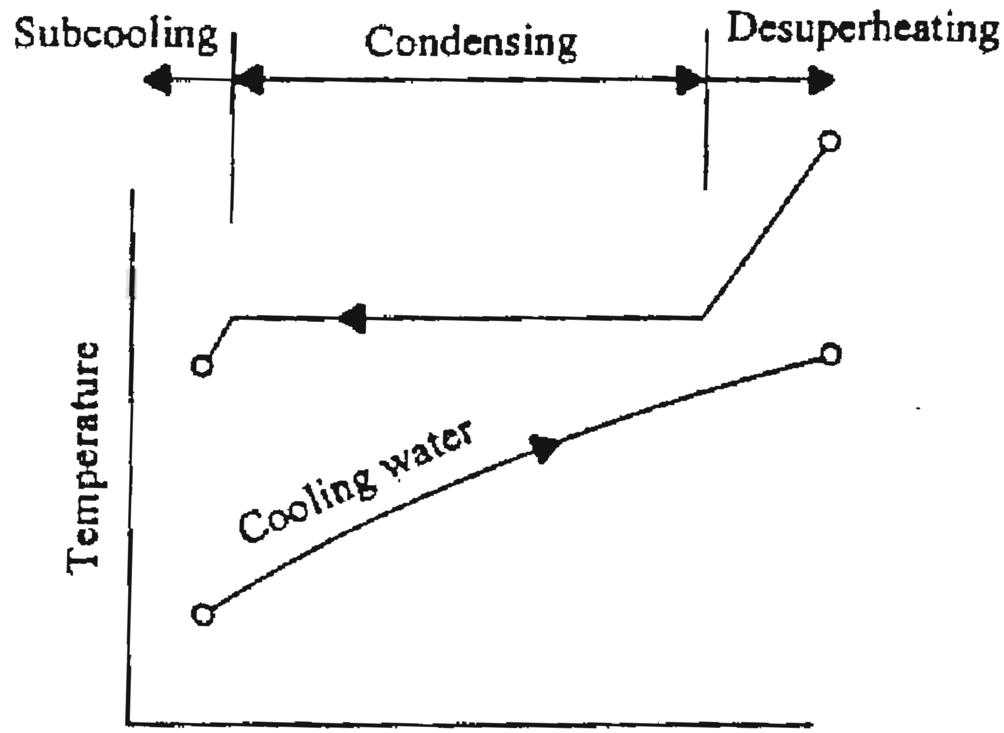
UA = product of overall heat-transfer coefficient and area to which it applies, kW/°C (Btu/hr per °F)

t_c = temperature of condensing refrigerant, °C (°F)

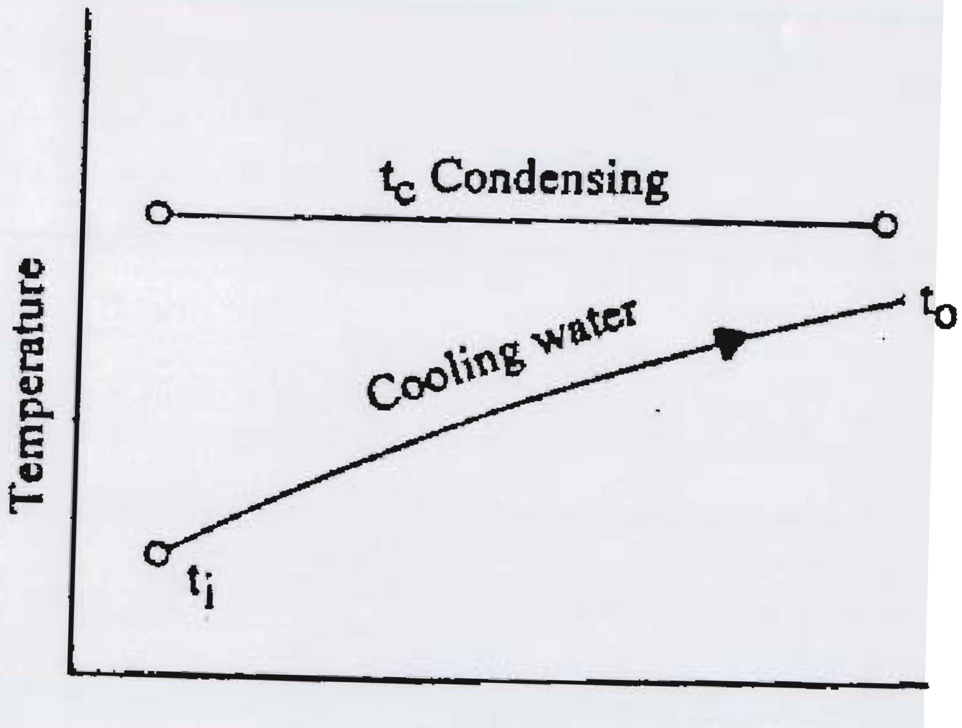
t_i = temperature of entering cooling water, °C (°F)

t_o = temperature of leaving cooling water, °C (°F)

a) Actual temperature profile



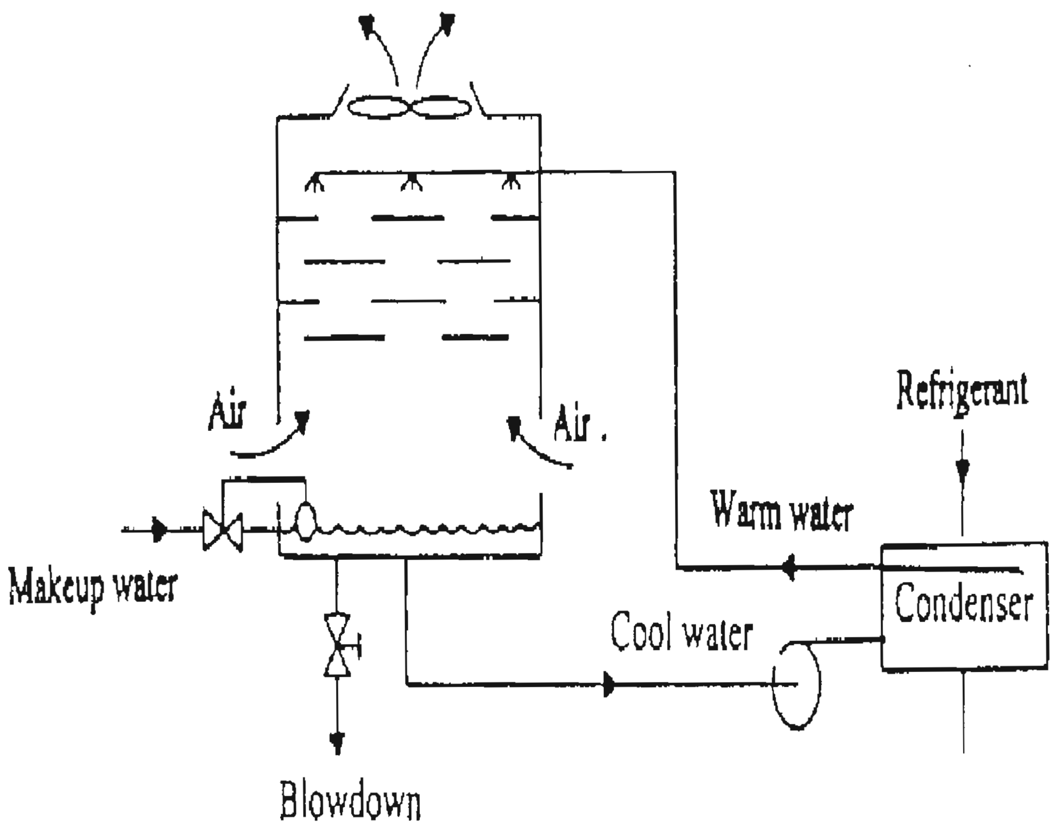
(a)

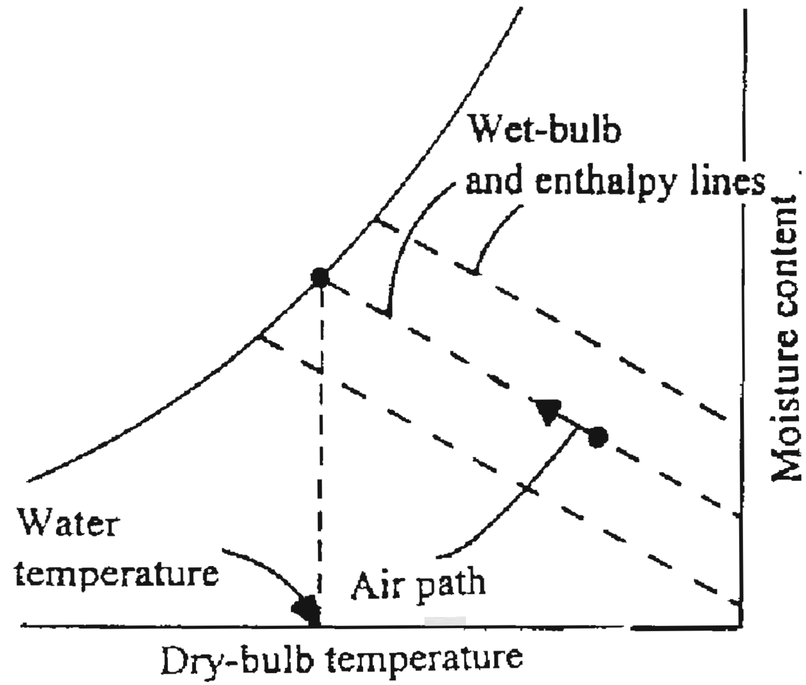


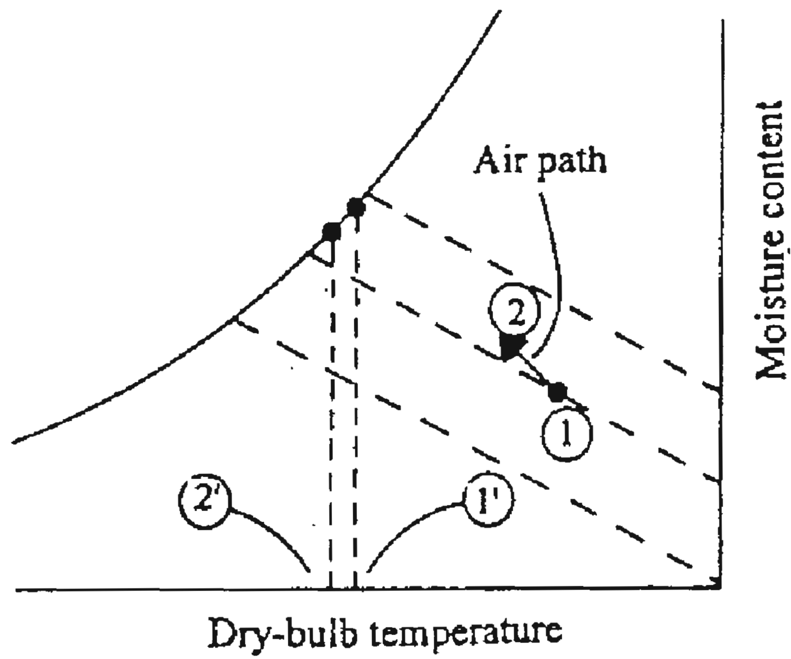
(b)

b) Idealized temperature profile

A cooling tower







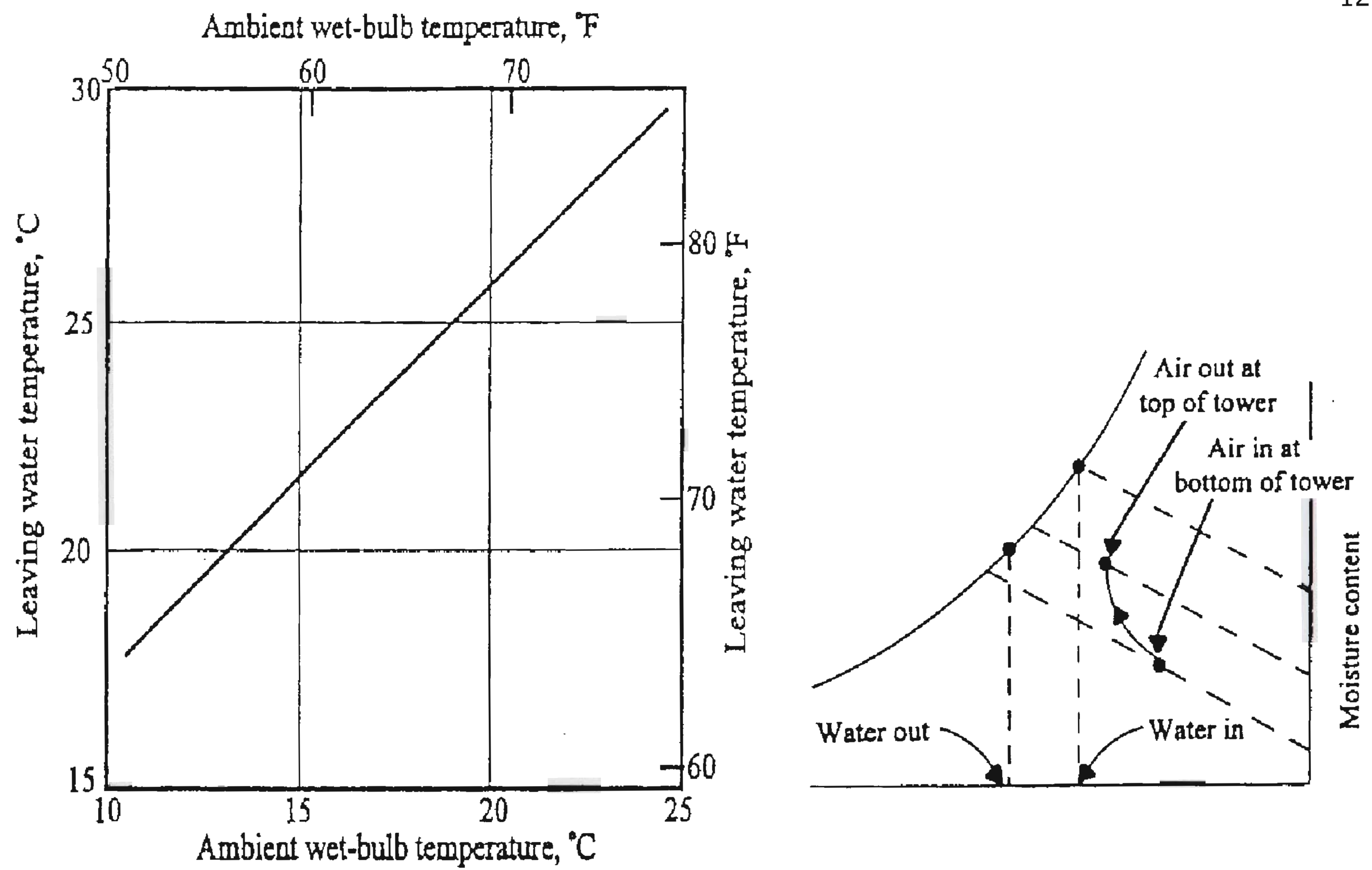


FIGURE 7.10
Leaving water temperature from a cooling tower as the ambient wet-bulb temperature changes.
The heat load and water-flow rate are constant.

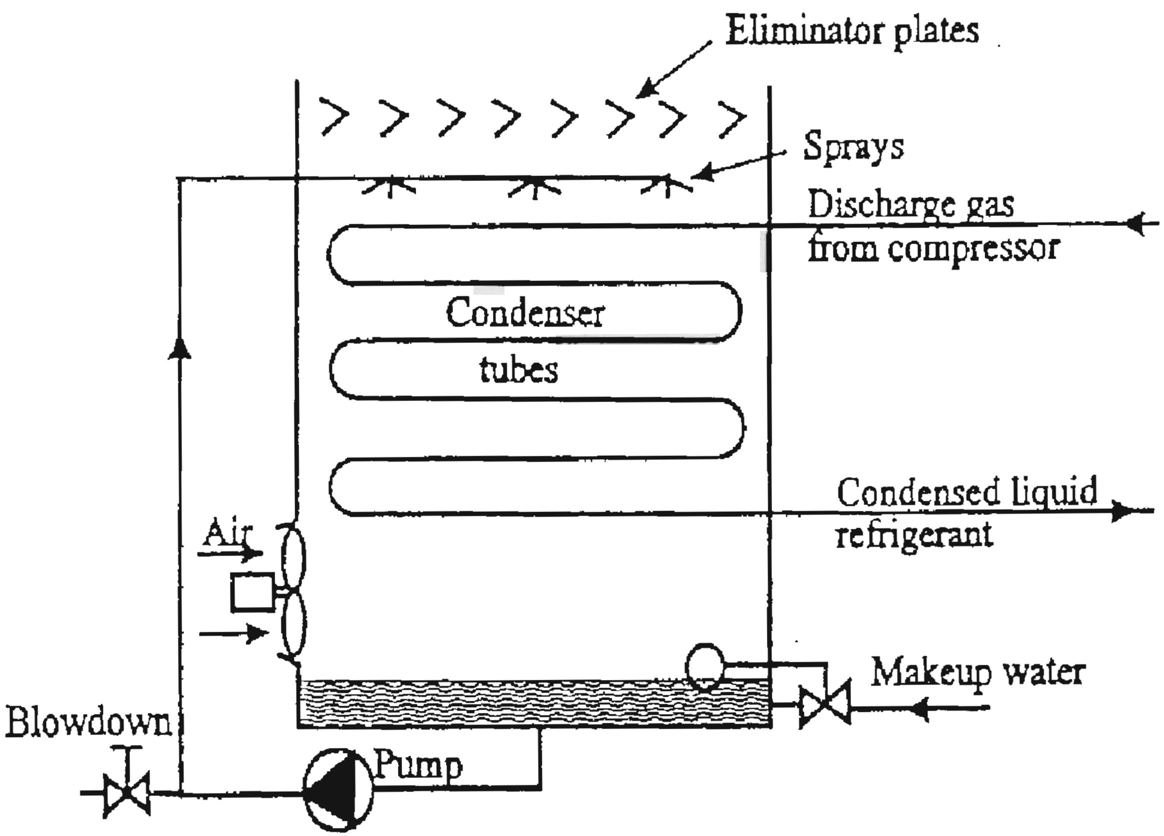
■ Characteristics of condensers

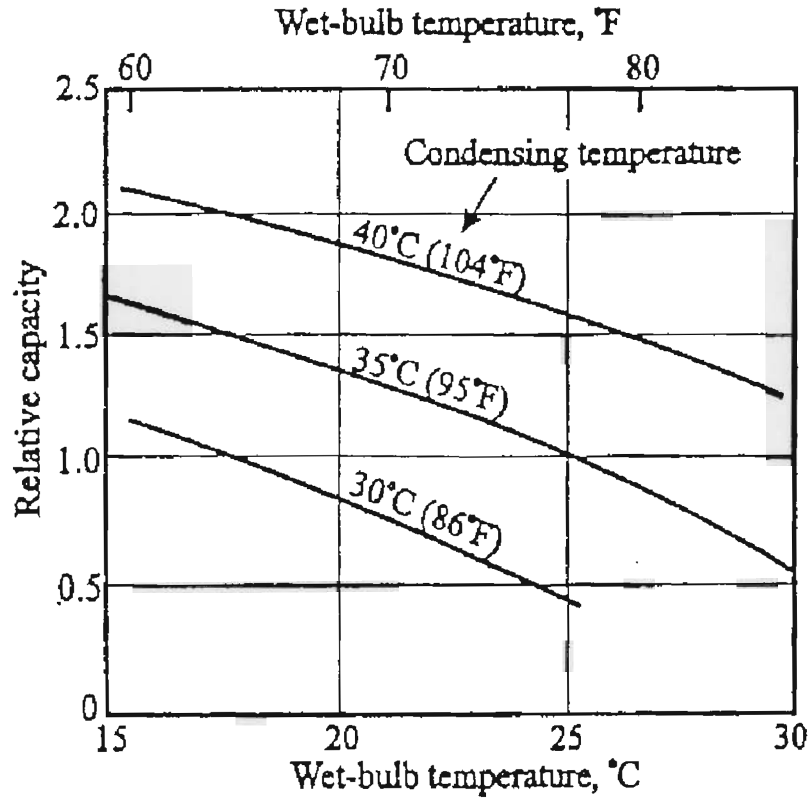
Air-cooled condenser. Usually lowest first cost of the three, and least maintenace cost as well, because no water circulates or evaporates.

Water-cooled condenser with cooling tower. Lower condensing temperature than with an air-cooled condenser, because the wet-bulb rather than the dry-bulb temperature of the air is the sink toward which the condensing temperature drives. When the distance between the compressor and the point of heat rejection is long, water can be piped to the cooling tower, rather than sending refrigerant, as must be done with the evaporative or air-cooled condenser.

Evaporative condenser. Compact and provides lower condensing temperatures than the air-cooled condenser and also lower than the water-cooled condenser/cooling tower combination. Figure 7.11 shows an evaporative condenser with a bit more detail than was presented in Fig. 7.1c.

■ Evaporative condensers





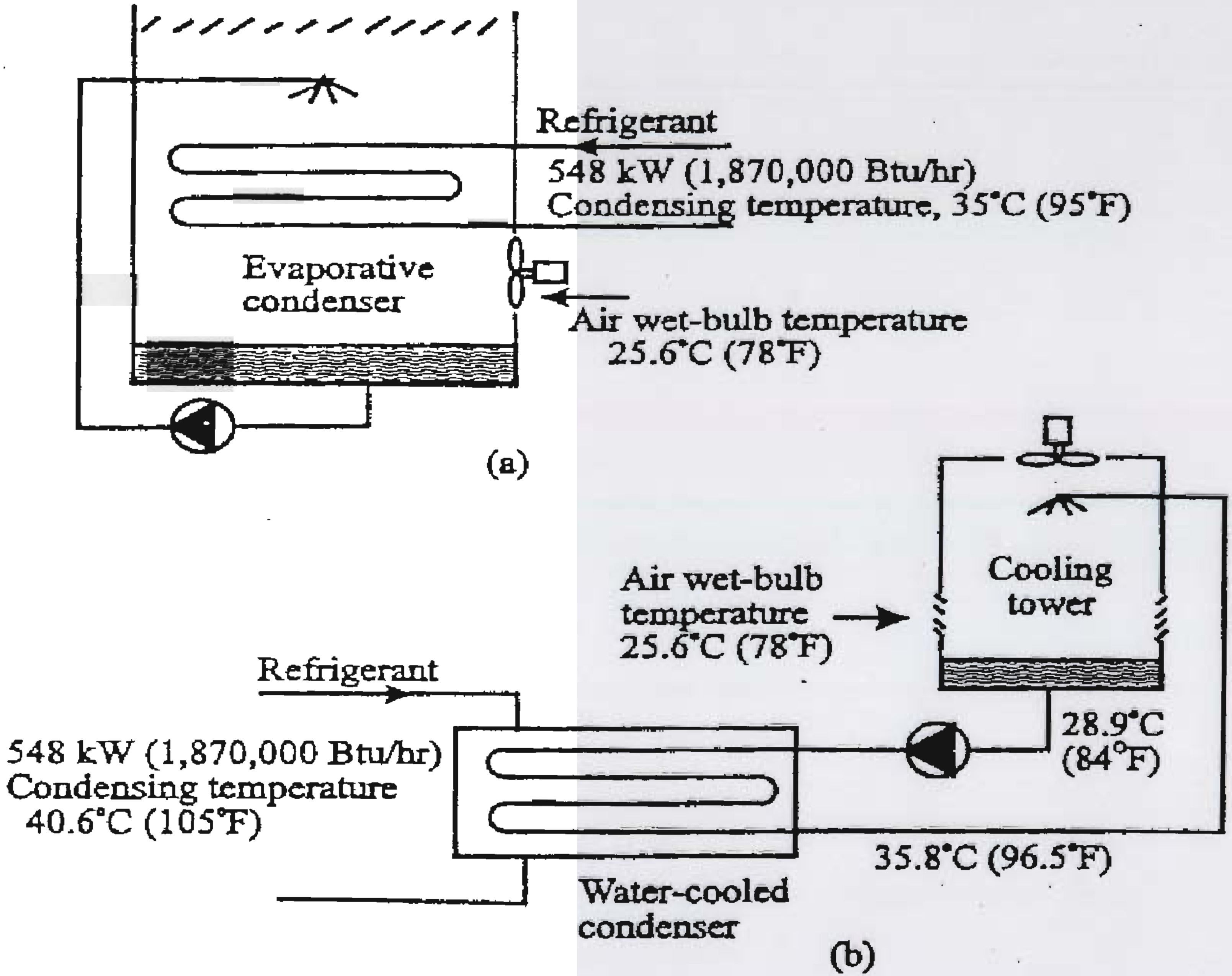


FIGURE 7.12
Achieving a lower condensing temperature with an evaporative condenser in comparison to the combination of a water-cooled condenser and cooling tower.

Nominal sizes and rates for evaporative condensers

Heat-transfer area:

0.25 m^2 per kW of heat rejection (0.8 ft^2 per 1000 Btu/hr)

Spray water circulating rate:

0.018 L/s per kW of heat rejection (5 gph per 1000 Btu/hr)

Air volume flow rate:

0.03 m^3/s per kW of heat rejection (18 cfm per 1000 Btu/hr)

Air pressure drop through the condenser:

250-375 Pa (1 to 1-1/2 inches of water)

Rate of water evaporated:[10]

1.5 L/hr per kW of heat rejection (0.12 gph per 1000 Btu/hr)

Total rate of water consumption:[10]

with good quality makeup water the bleed rate may be as low as 50% of the evaporation rate, so the total rate evaporated and blown down may be about 2.2 L/hr per kW of heat rejection (0.18 gph per 1000 Btu/hr).

■ Capacity control

儘量讓冷凝器全載運轉以降低冷凝溫度，除非出現以下狀況時，必須進行容量控制：

- the condensing pressure is too low to adequately feed level-control valves and expansion valves
- the pressure of defrost gas is too low to achieve a satisfactory defros,
- if the plant uses screw compressors with their oil cooled by direct injection of refrigerant, the pressure of the liquid must be high enough to force an adequate flow rate of liquid into the compressor
- savings in compressor power by further lowering of the condensing temperature are less than savings that would be possible in pump and fan motors of the compressors

Capacity control- varying the air flow rate

$$\text{Condenser capacity} = (\text{constant})(\text{air flow rate})^{0.48}$$

- Variable-frequency drive of fan motor
- Two speed fan motors
- Pony motors
- Fan dampers
- Fan cycling on a single-fan unit
- Shutting down one fan in a multiple-fan condenser

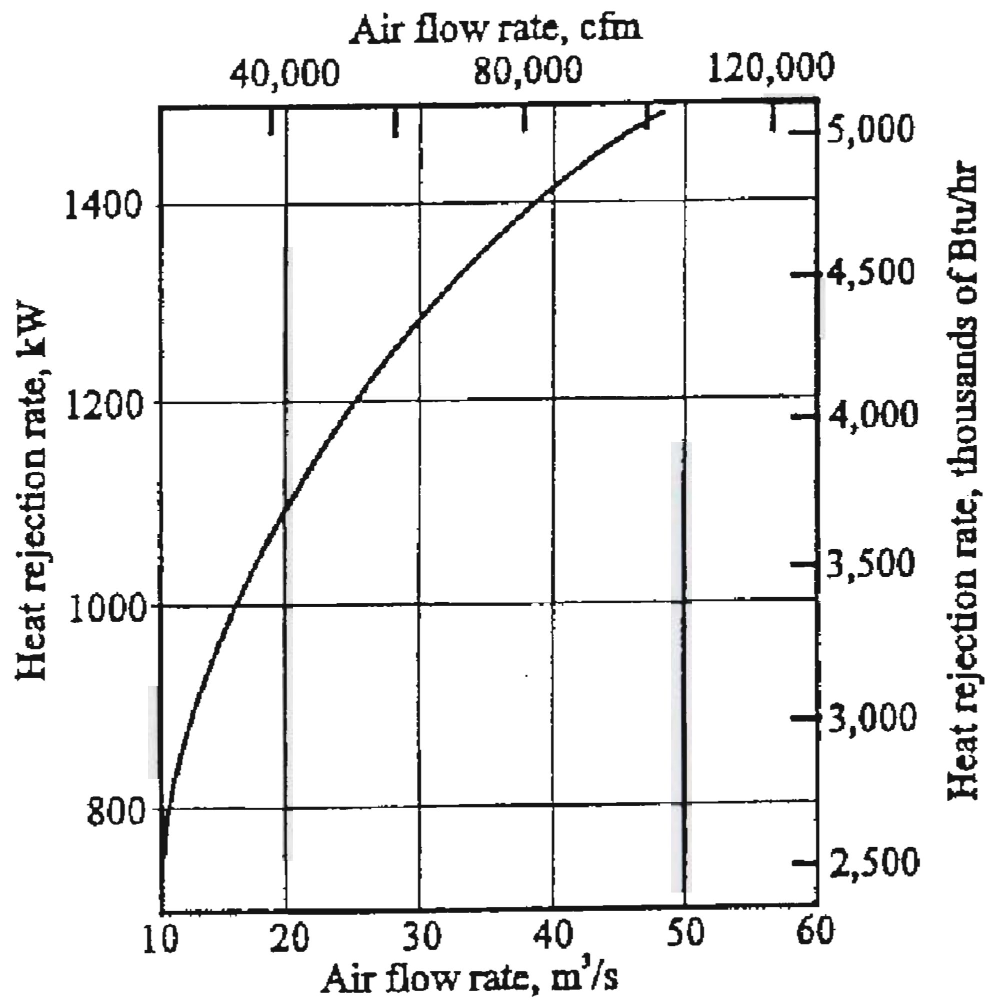


FIGURE 7.16
Effect of air-flow rate on the heat-rejection capacity of an evaporative condenser with given condensing and wet-bulb temperatures.

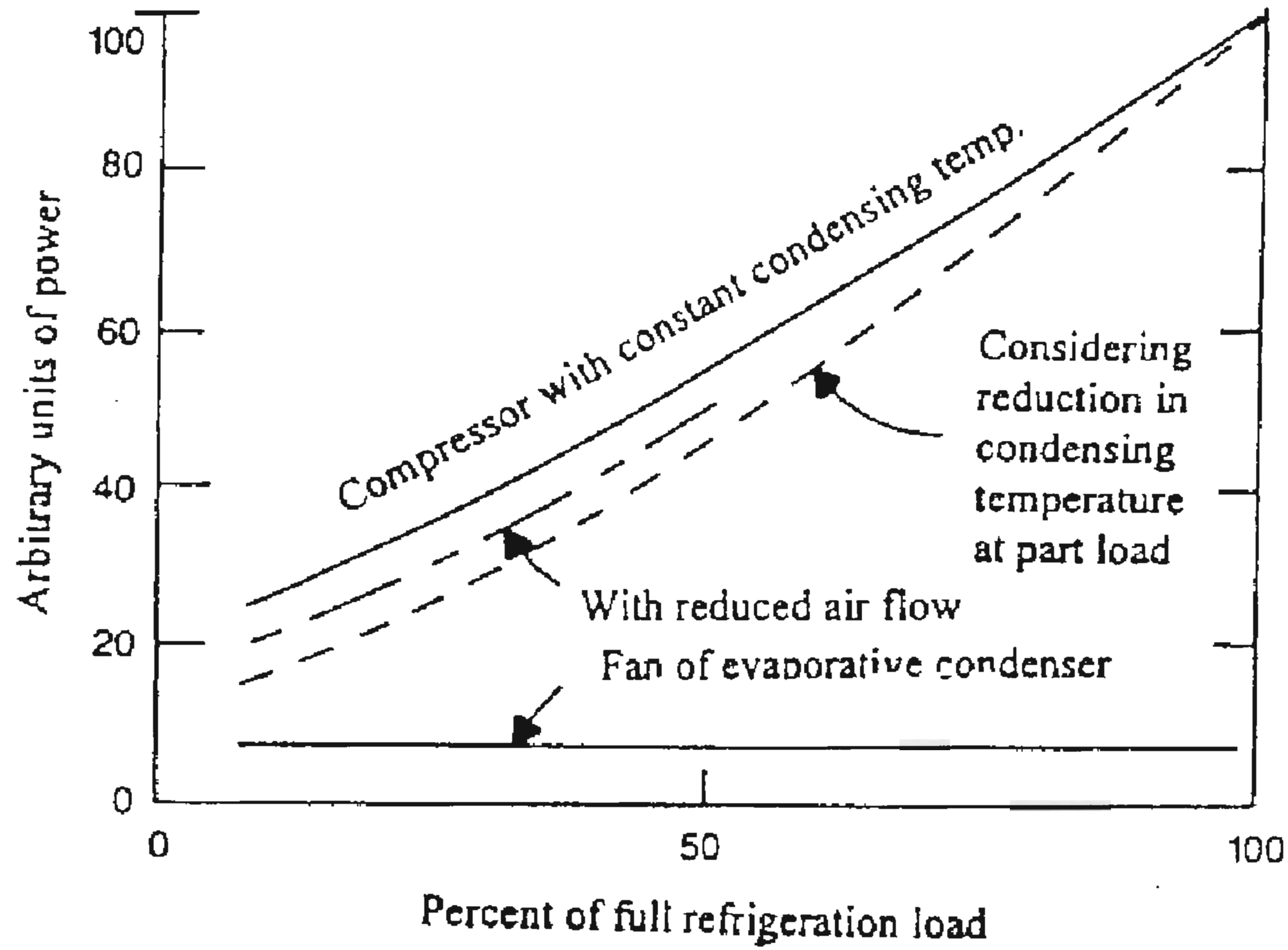


FIGURE 7.17

Relative power requirements of the compressor and the fan of an evaporative condenser. The evaporating temperature is in the range of 5°C (41°F) and the wet-bulb temperature is constant.

全載時風扇馬力約為壓縮機之5%～8%

- Positioning the condenser

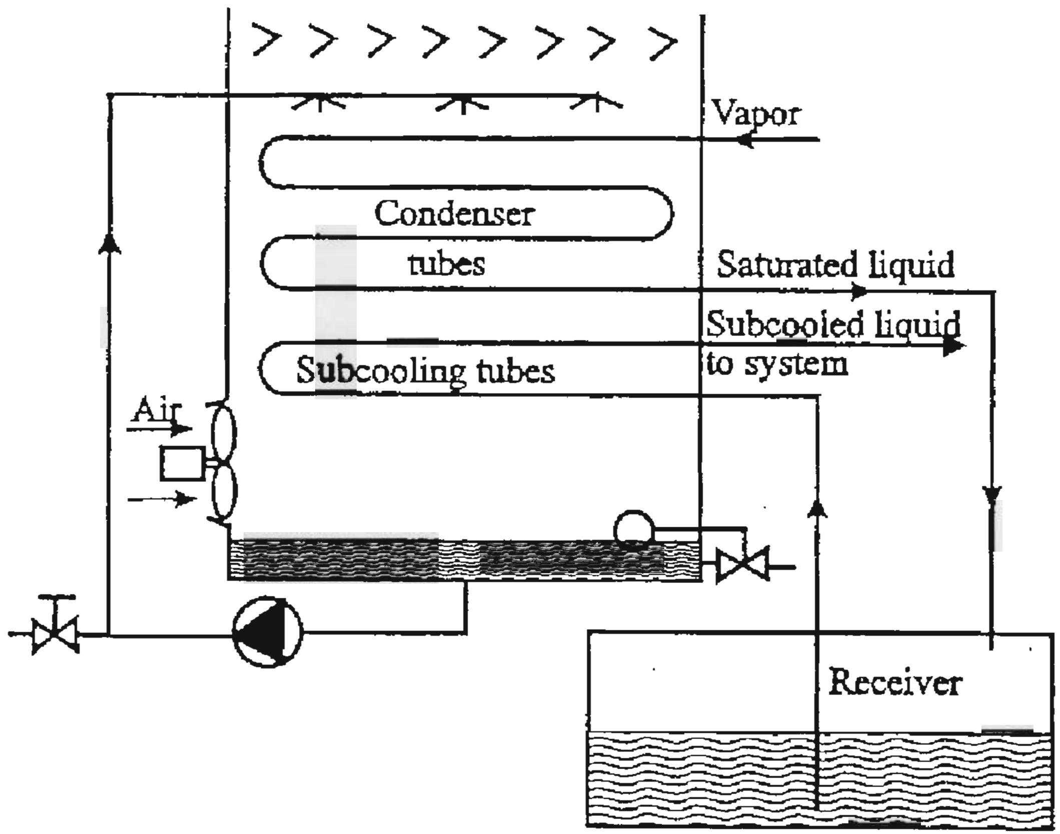
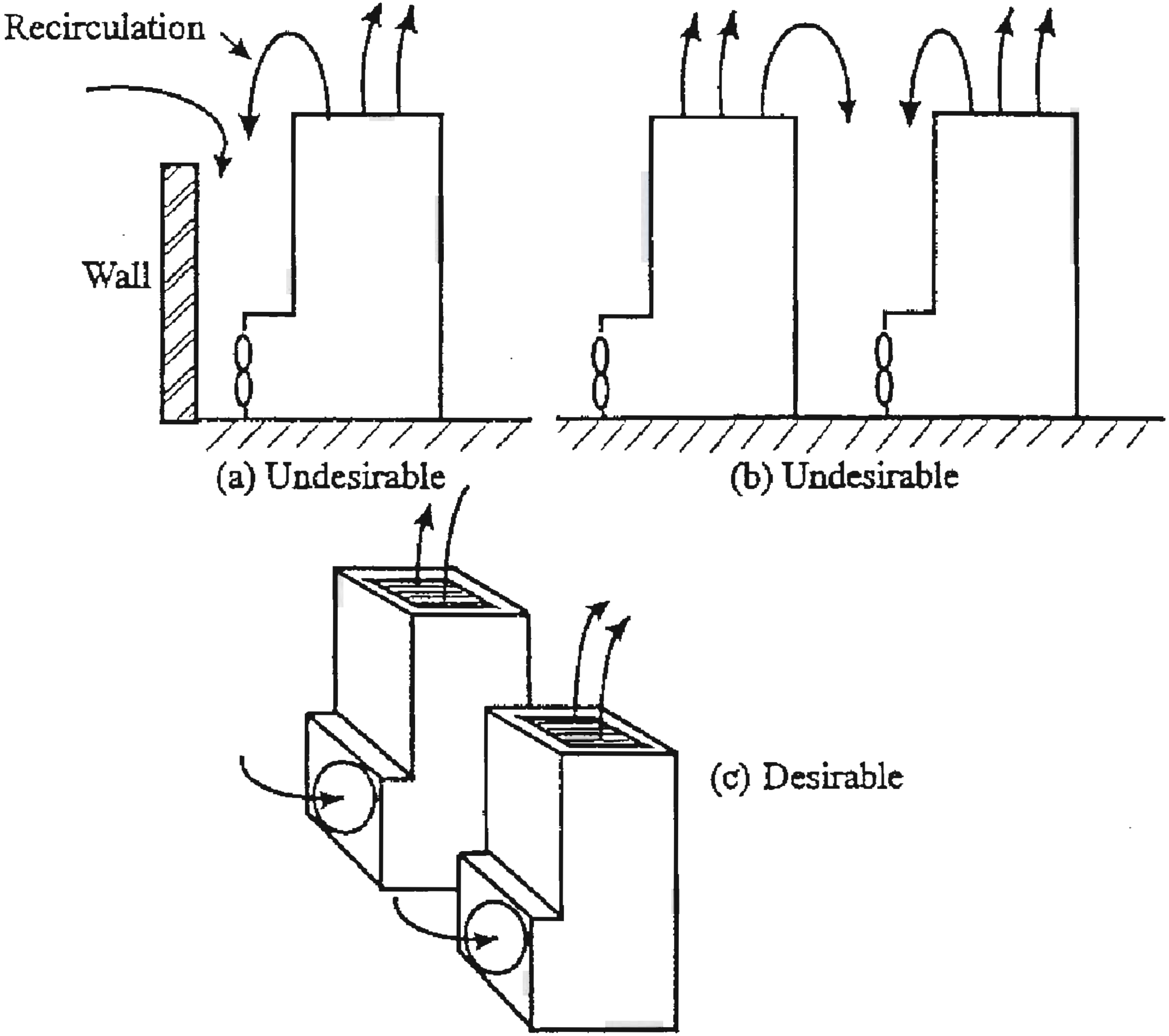


FIGURE 7.18
Liquid subcooler as auxiliary coil in the condenser case.


Recirculation
Wall
(a) Undesirable
(b) Undesirable
(c) Desirable

■ Purging the condenser of air

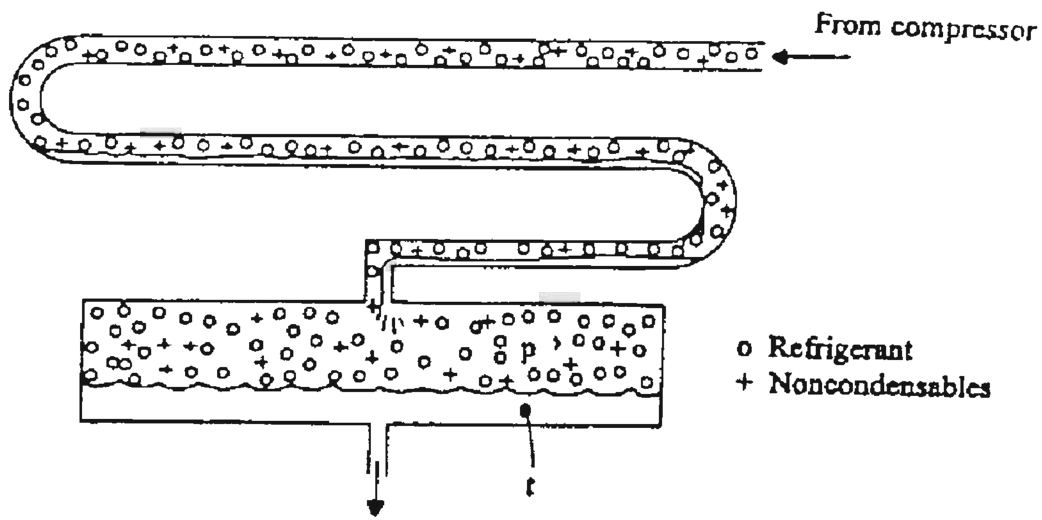


FIGURE 7.22
Noncondensables in a condenser.

TABLE 7.3
Evaporating temperatures corresponding to standard atmospheric pressure of 101 kPa (14.7 psia).

Refrigerant	Evaporating temperature below which air could be drawn into the system through leaks
Ammonia	−33.5°C (−28.3°F)
R-22	−40.8°C (−41.5°F)
R-404a	−46°C (−50.8°F)
R-507	−46.7°C (−52°F)

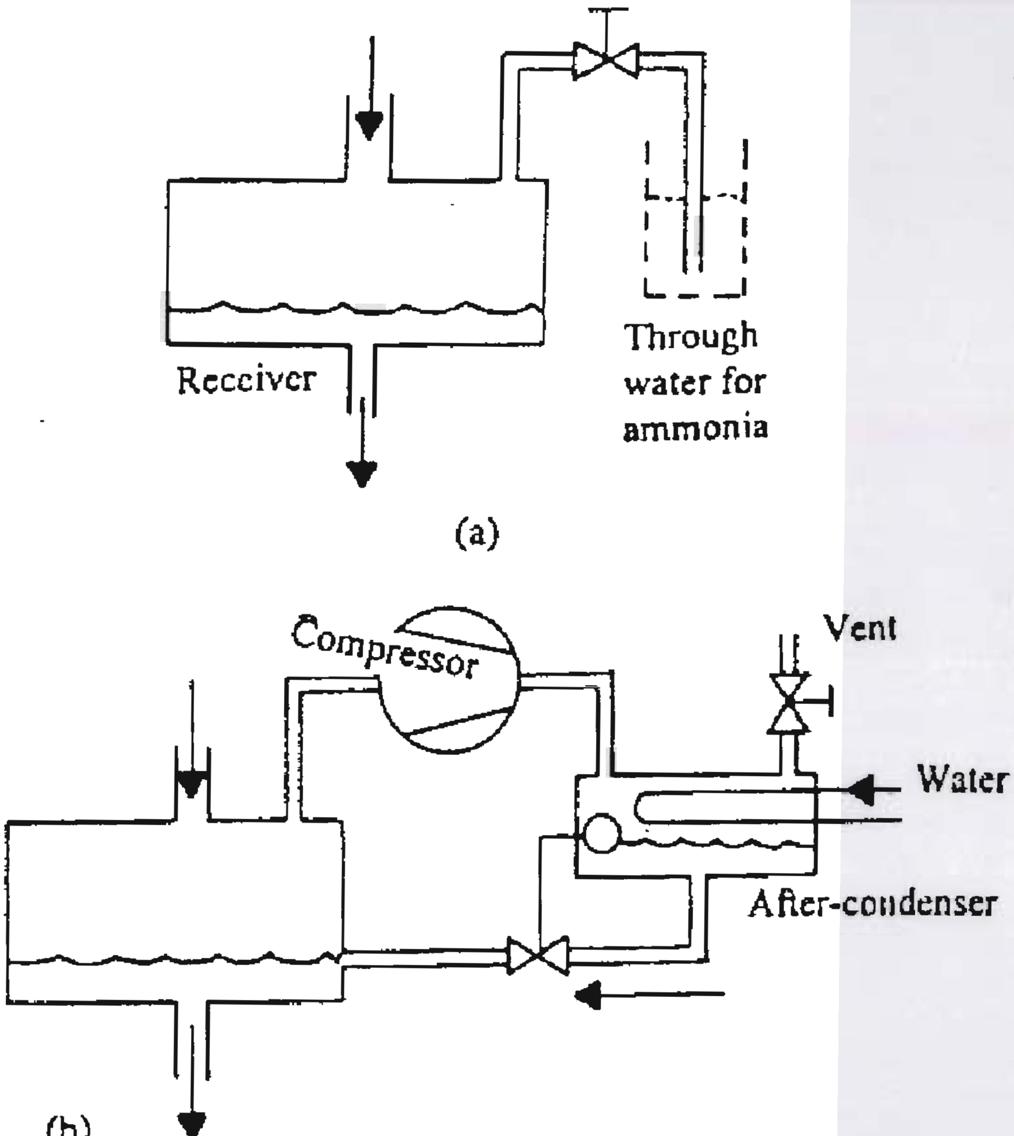


FIGURE 7.23
Two methods of purging noncondensables, (a) direct venting, and (b) compression of the refrigerant followed by condensation.

Refrigeration Engineering

-Evaporator

李魁鵬

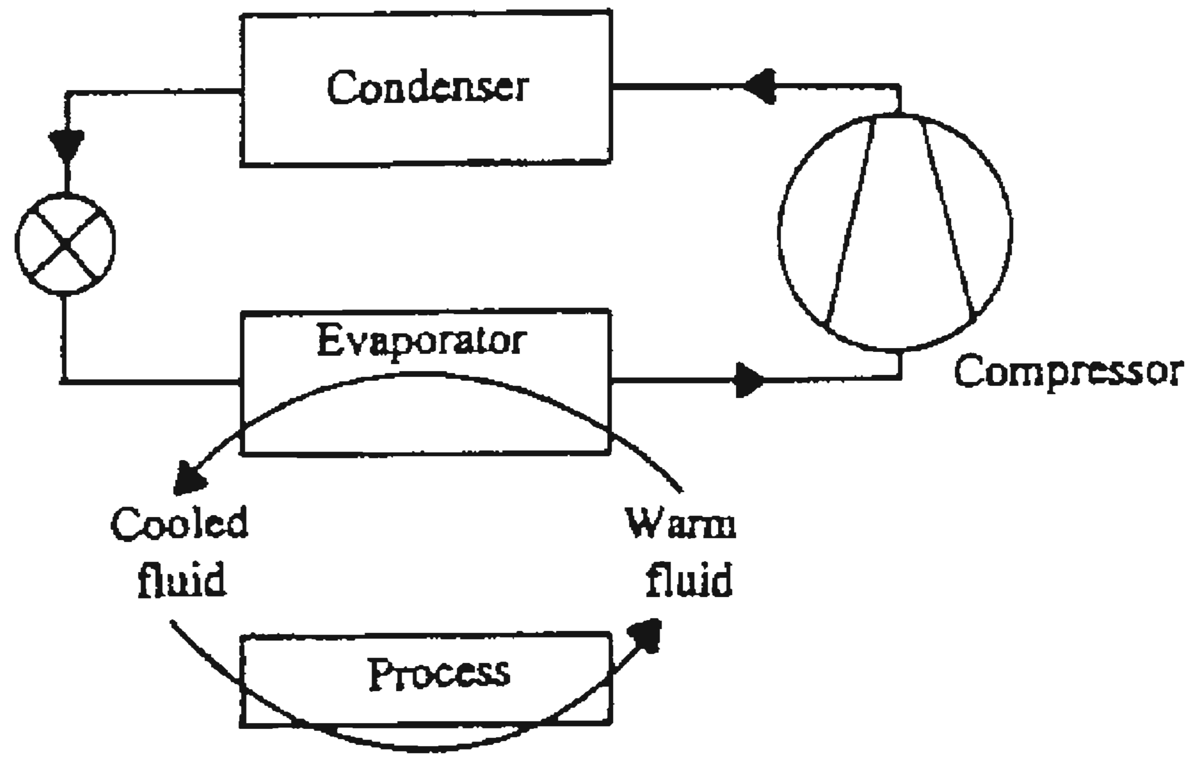


FIGURE 6.1
Cooling of a fluid stream at the evaporator.

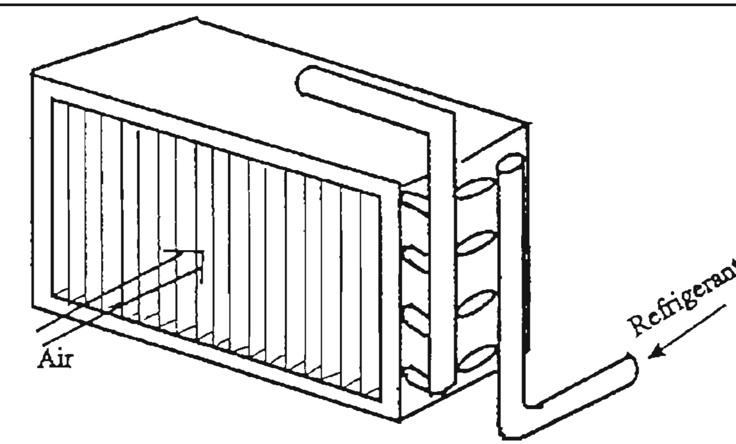
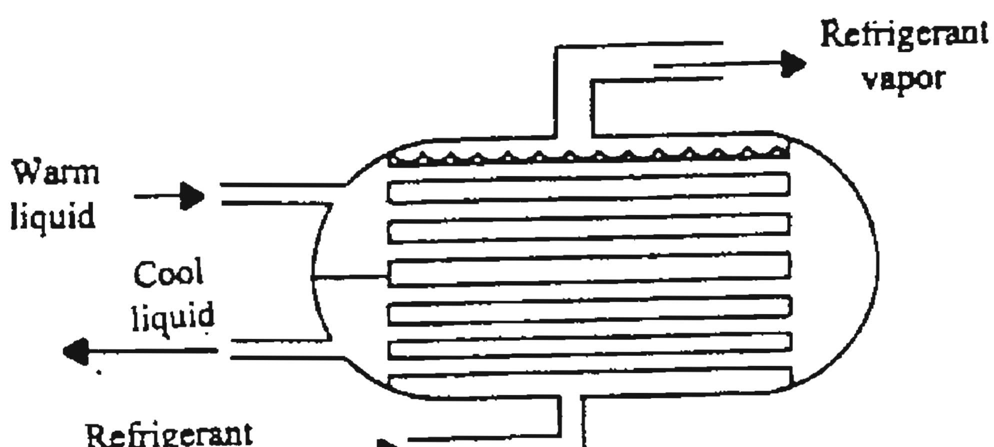
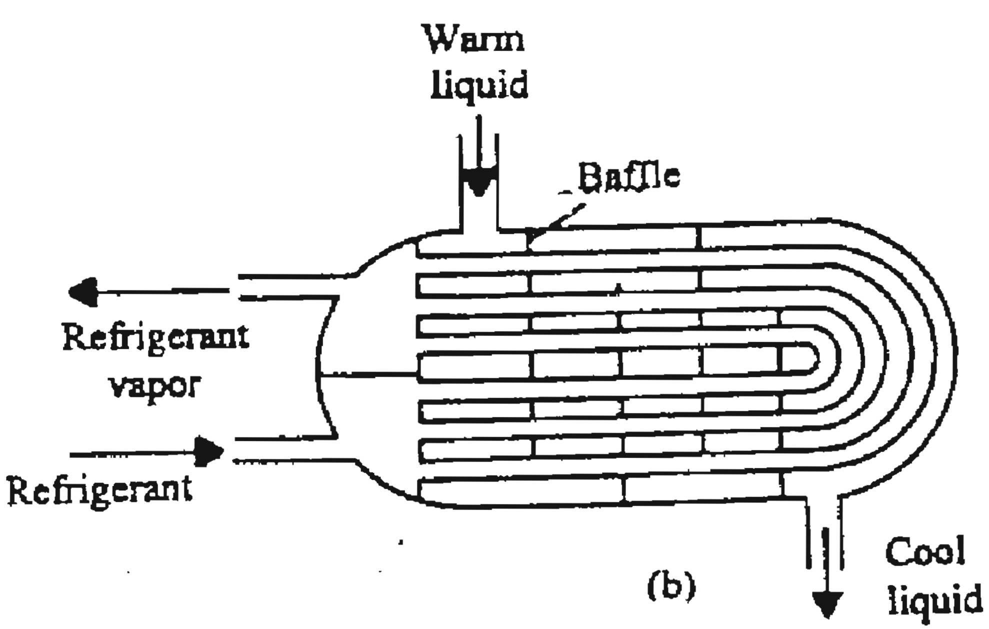


FIGURE 6.2
An air cooling coil.



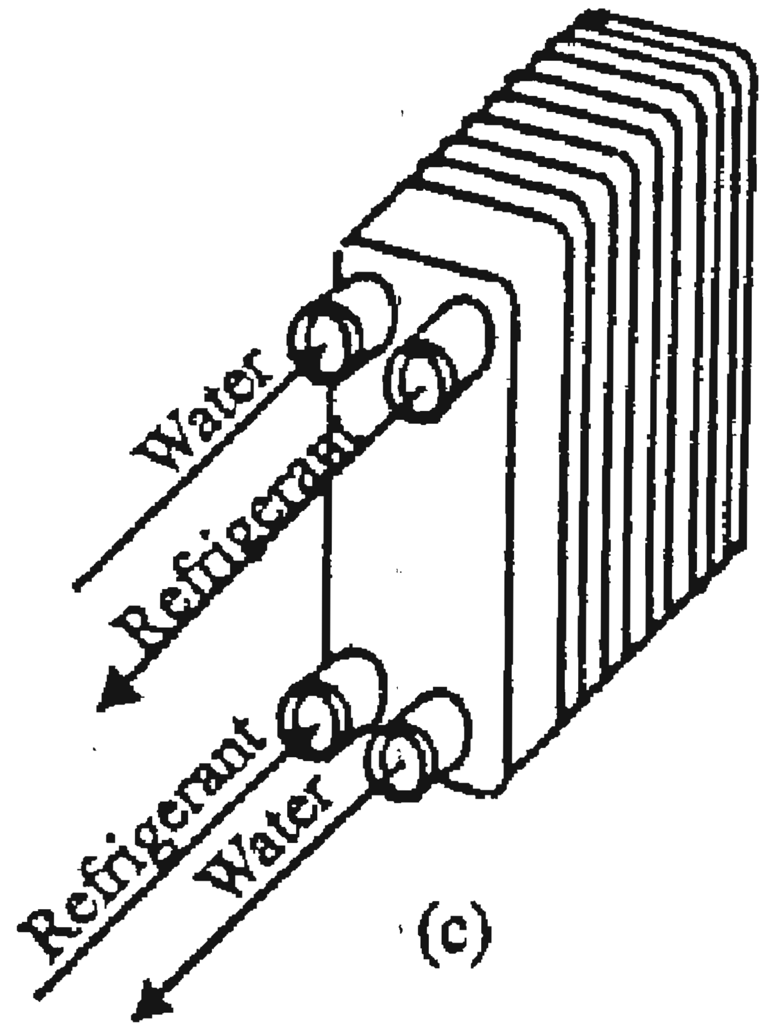
(a)

Shell-and-tube liquid chilling evap.



(b)

Shell-and-tube liquid chilling evap.



(c)

Plate-type liquid chilling evap.

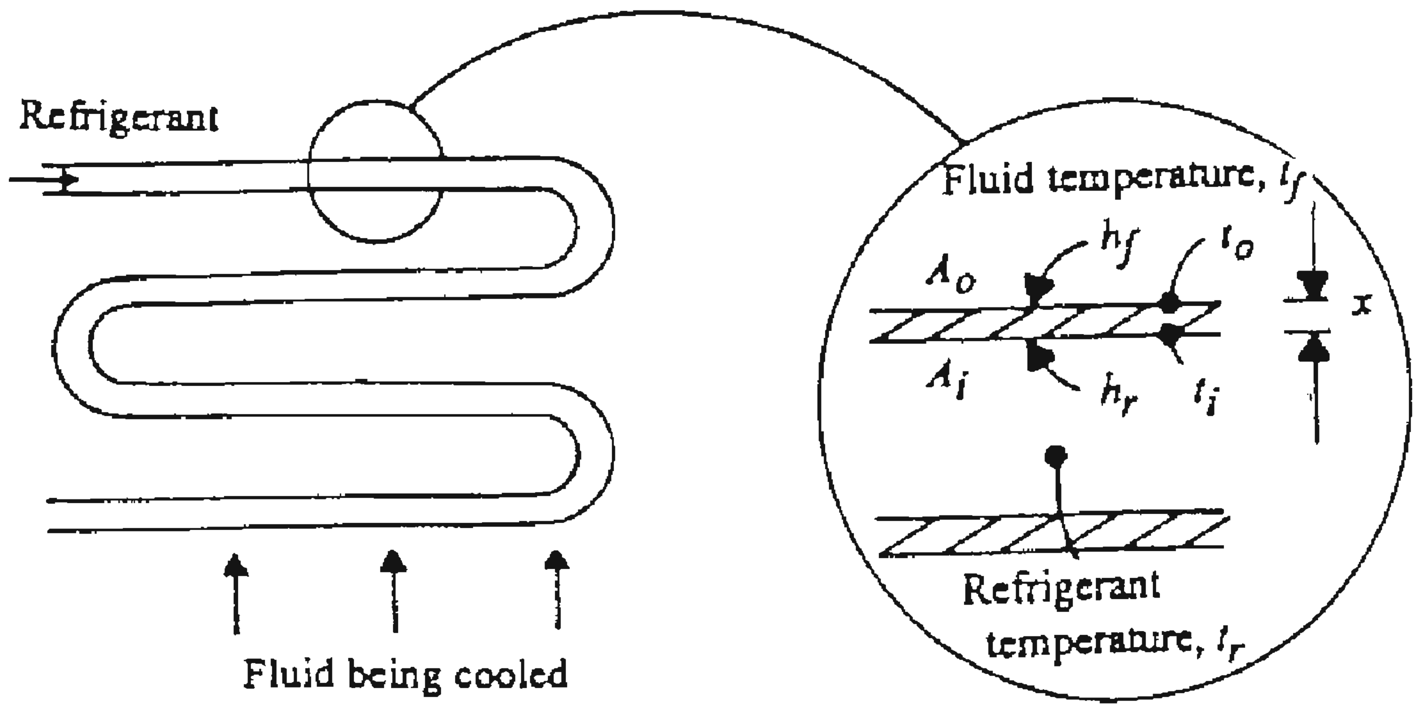


FIGURE 6.4
Heat-transfer coefficients in an evaporator.

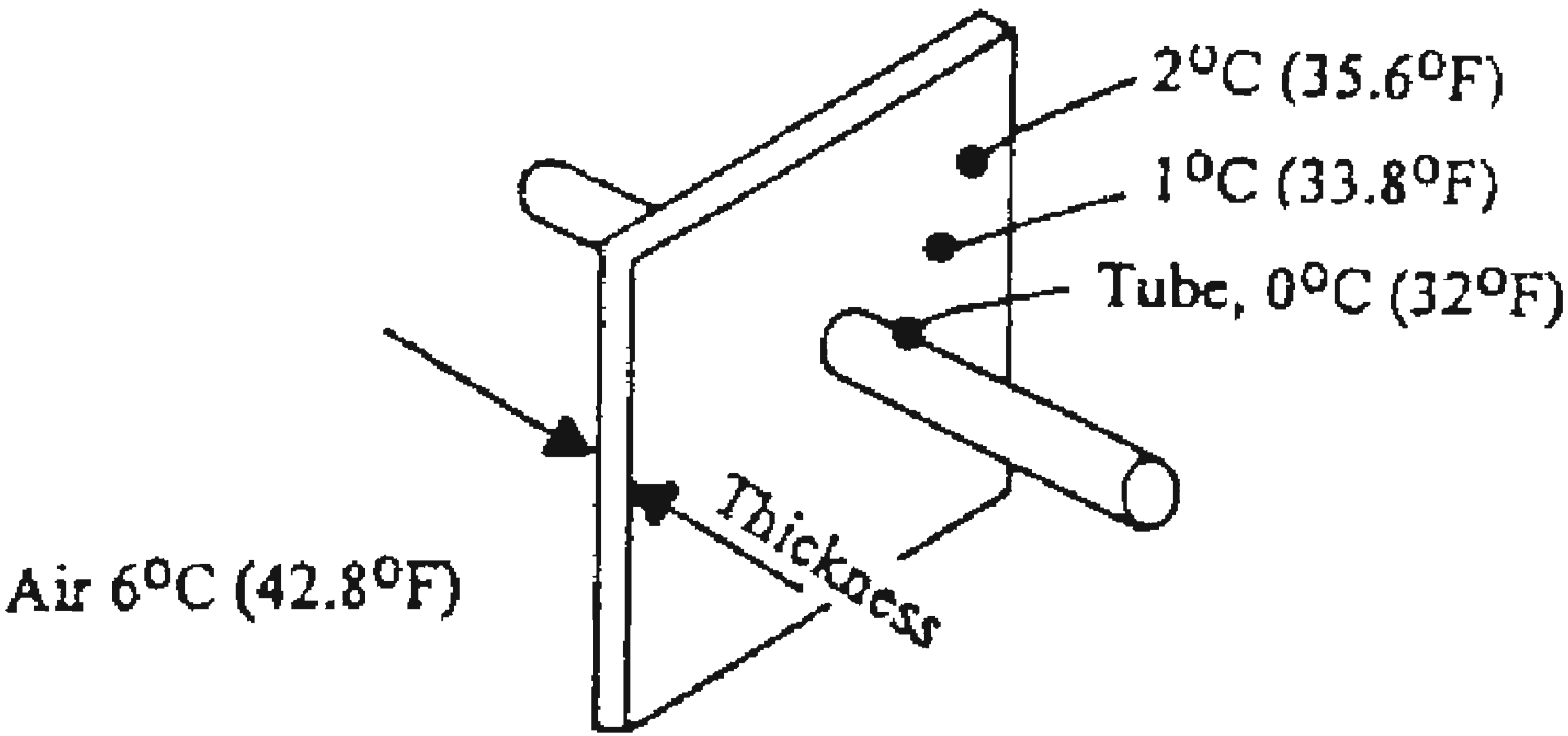
$$q = UA(t_f - t_r)$$

where U = overall heat-transfer coefficient, $W/m^2 \cdot °C$ ($Btu/hr \cdot ft^2 \cdot °F$).

$$\frac{1}{UA} = \frac{1}{h_f A_o} + \frac{x}{k A_{mean}} + \frac{1}{h_r A_i} \qquad \frac{1}{UA} = \frac{1}{U_o A_o} = \frac{1}{U_i A_i}$$

4

- η : Fin effectiveness (0.3~0.7)



$$\frac{1}{U_i} = \frac{A_i}{h_f A_o \eta} + \frac{x A_i}{k A_{mean\ tube}} + \frac{1}{h_r}$$

TABLE 6.2
Influence of some choices of fin dimensions and materials on the fin effectiveness η and the overall heat-transfer capacity of the coil.

Increase of variable	Effect on fin effectiveness η	Effect on overall heat transfer of coil	Effect on cost
Distance between tubes	Decrease	Increase	Increases
Air-side heat-transfer coefficient	Decrease	Increase	Increase of fan power
Thermal conductivity	Increase $k_{aluminum} = 4k_{steel}$	Increase	See Sec. 6.26
Fin thickness	Increase	Increase	Increase because of more metal

■ LMTD : Log-Mean-Temperature Difference

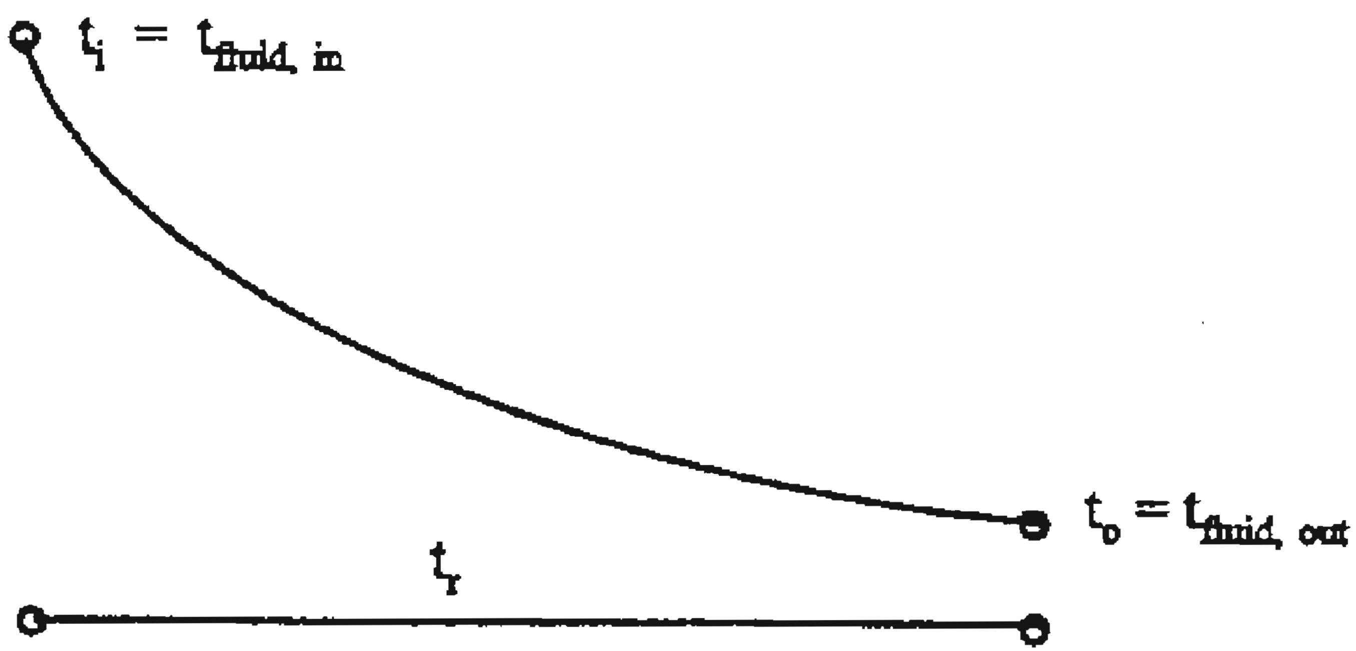


FIGURE 6.7
Distributions of the temperatures of the refrigerant and the fluid being chilled in an evaporator.

$$q,\ \mathrm{kW\ (Btu/hr)} = UA\left[\frac{(t_i - t_r) - (t_o - t_r)}{\ln[(t_i - t_r)/(t_o - t_r)]}\right]$$

$$q = UA\left[\frac{t_i - t_o}{\ln[(t_i - t_r)/(t_o - t_r)]}\right]$$

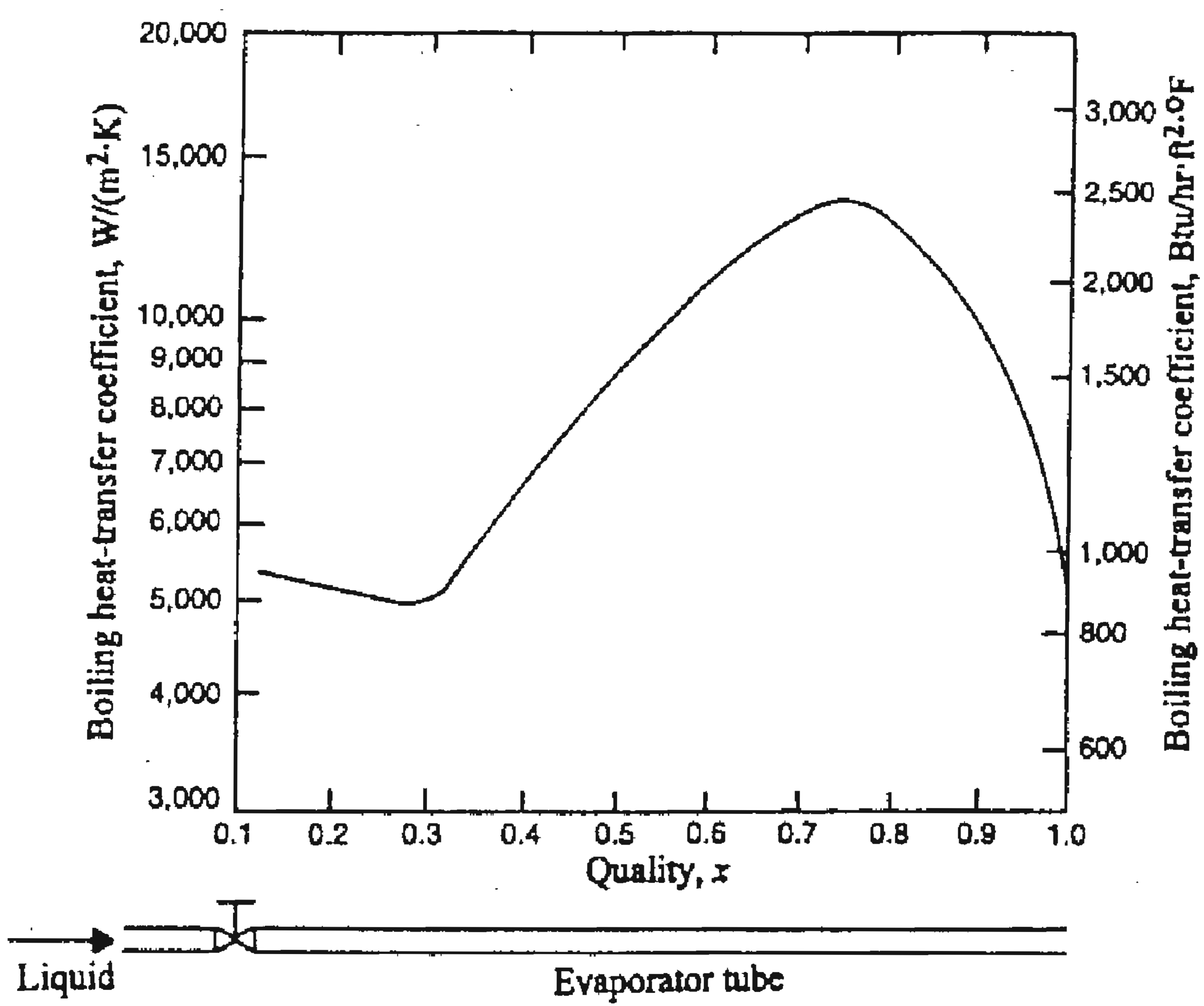
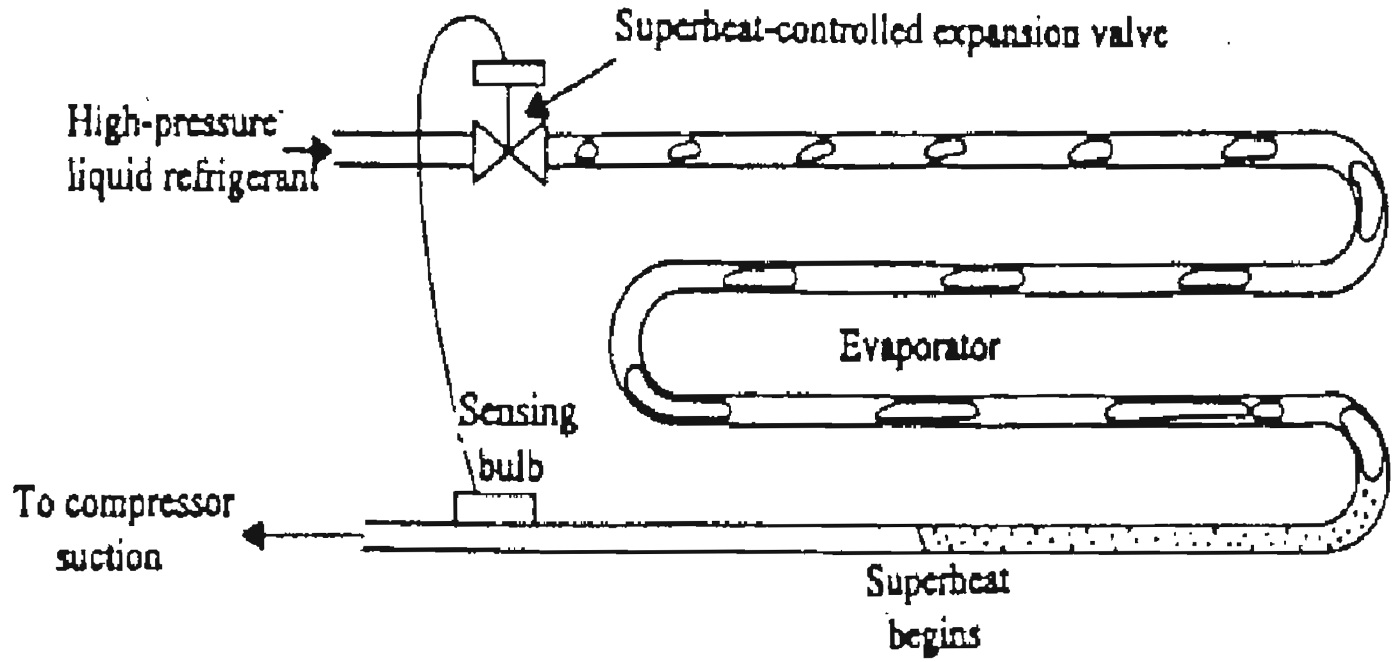
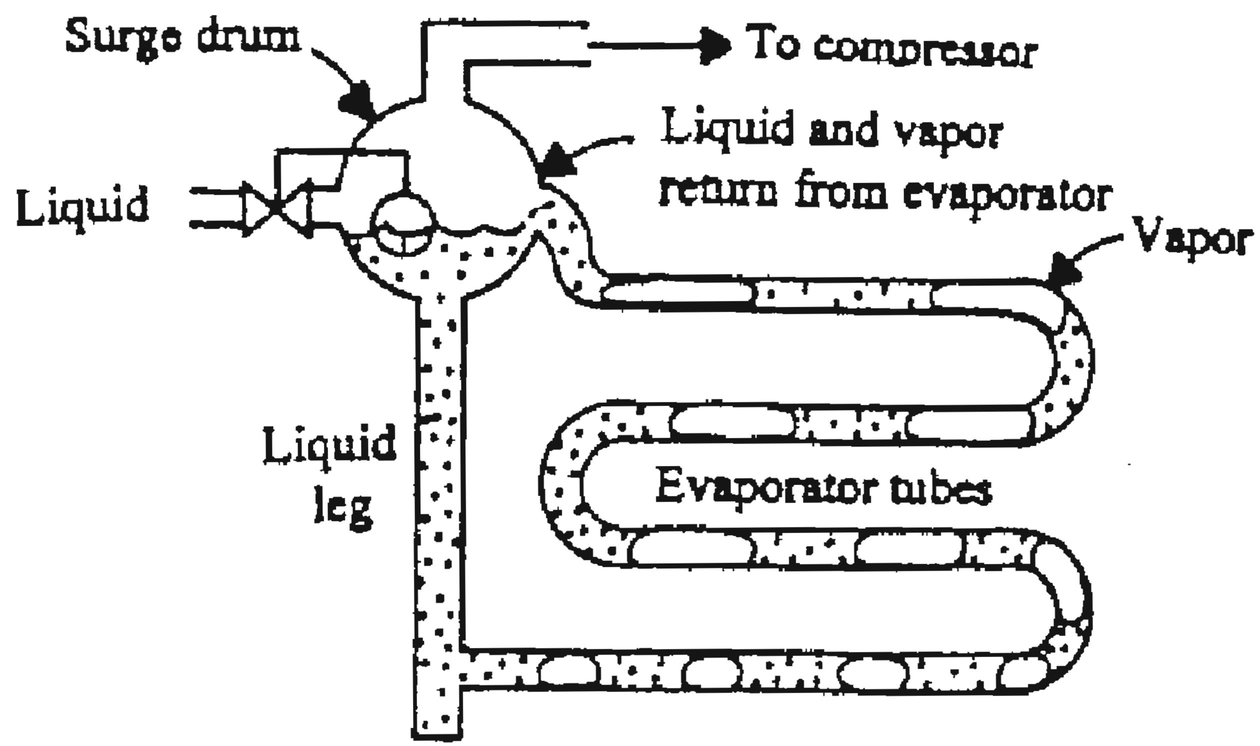


FIGURE 6.8
Heat-transfer coefficient as refrigerant boils in an evaporator tube[1].

8



A direct-expansion evaporator



A flooded evaporator

Advantages of flooded evaporators in comparison to direct expansion:

- 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.

- Disadvantages of flooded evaporators in comparison to direct expansion:

- 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.

//

TABLE 6.3

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

	Increase in evaporating temperature, °C per m (°F per ft)	
Evaporating temperature	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)

適合應用在小系統

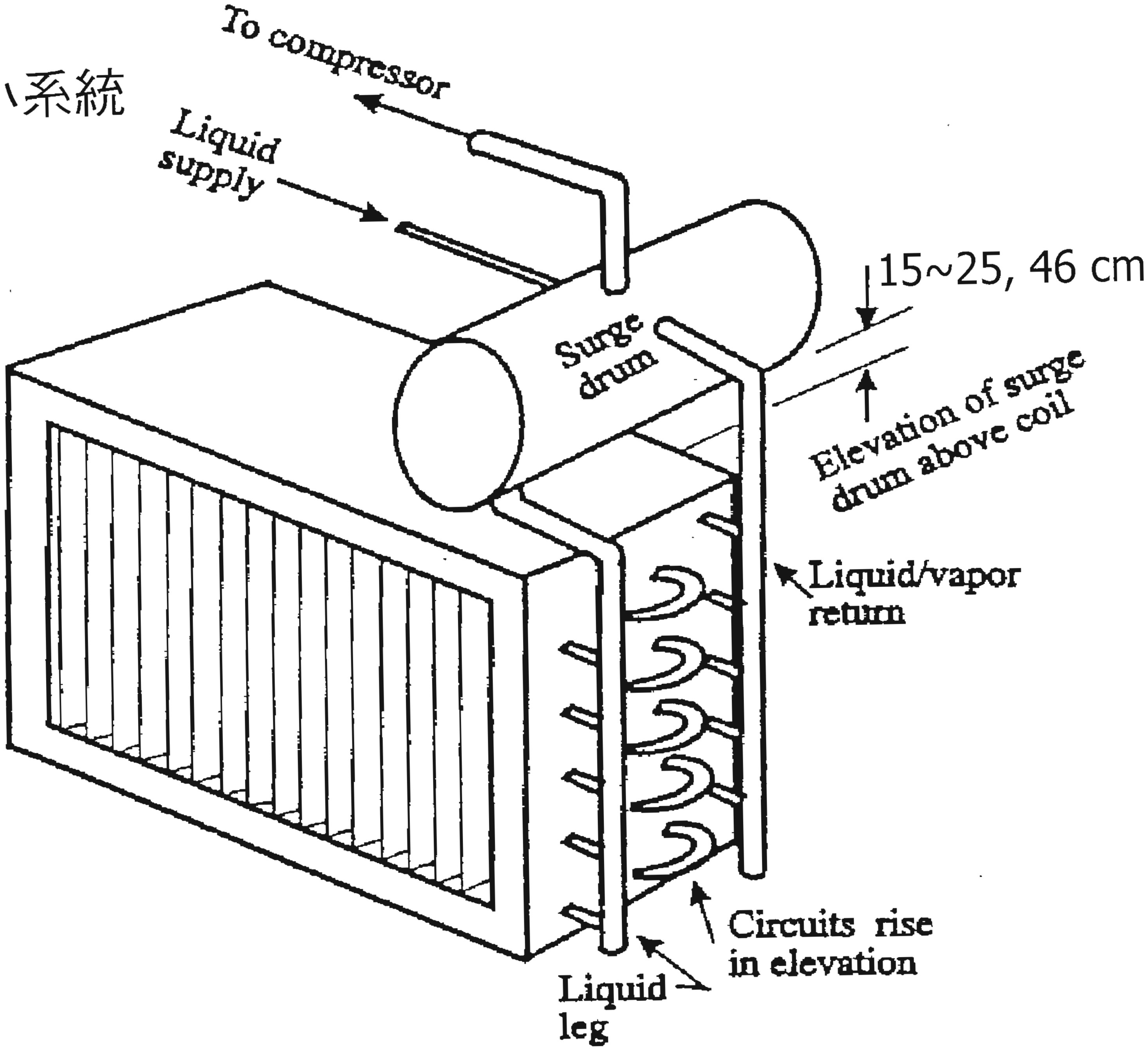


FIGURE 6.11
A flooded air-cooling coil for low-temperature application.

適合應用在大系統

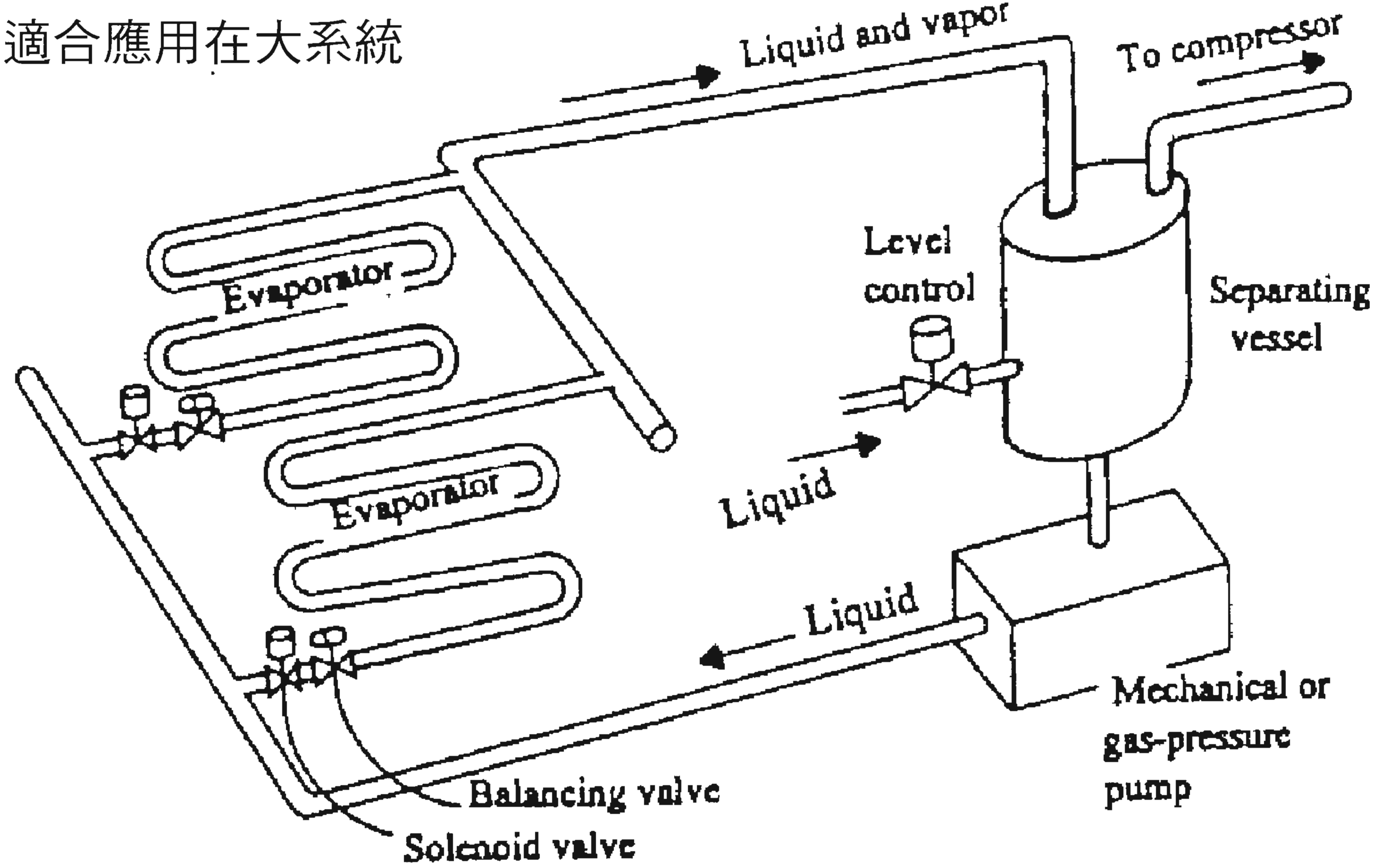


FIGURE 6.12
Forced liquid recirculation.

- Optimum evaporating temperature

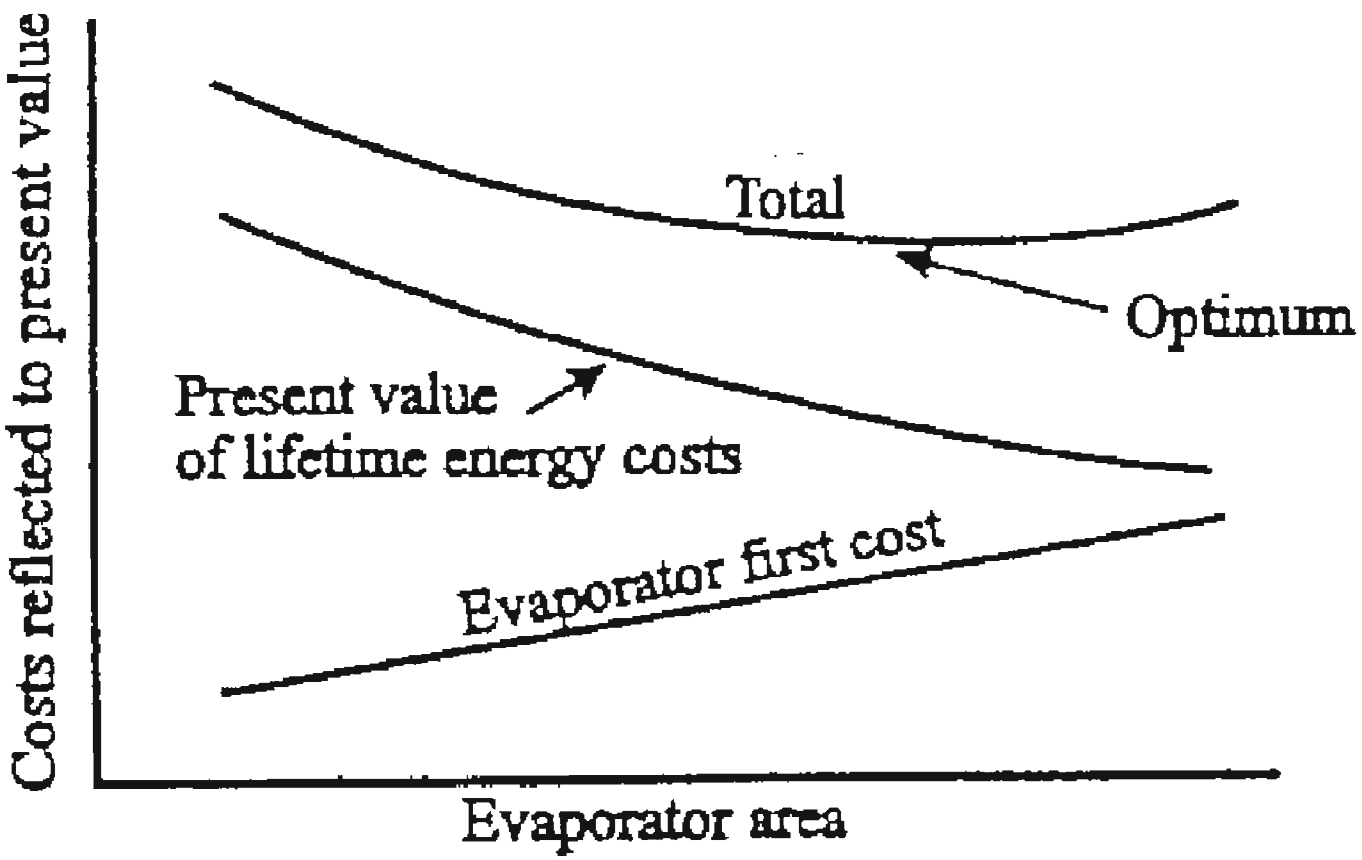
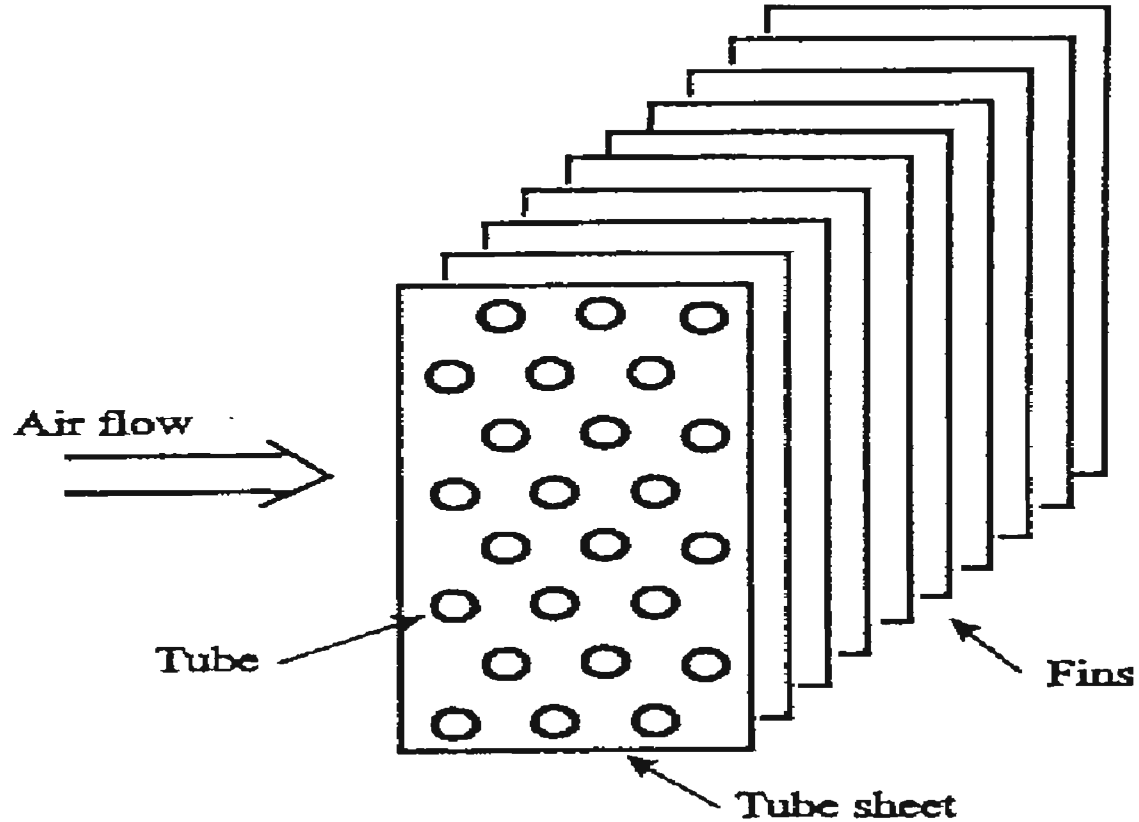


FIGURE 6.13
Optimum evaporator area for minimum total of the first cost of the evaporator and the present worth of the lifetime compressor energy cost.

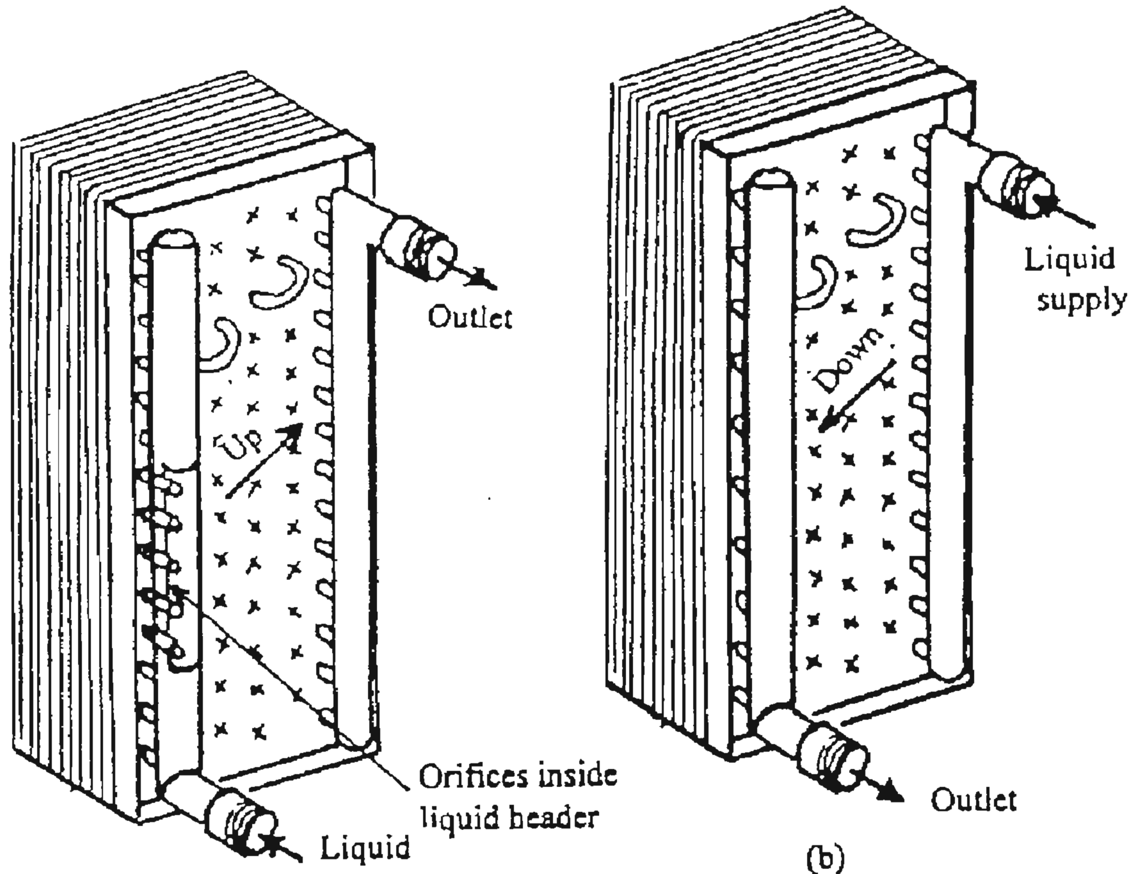
■ Tube sheet and fins of a coil



k value :
Stainless≒1/4 Al.

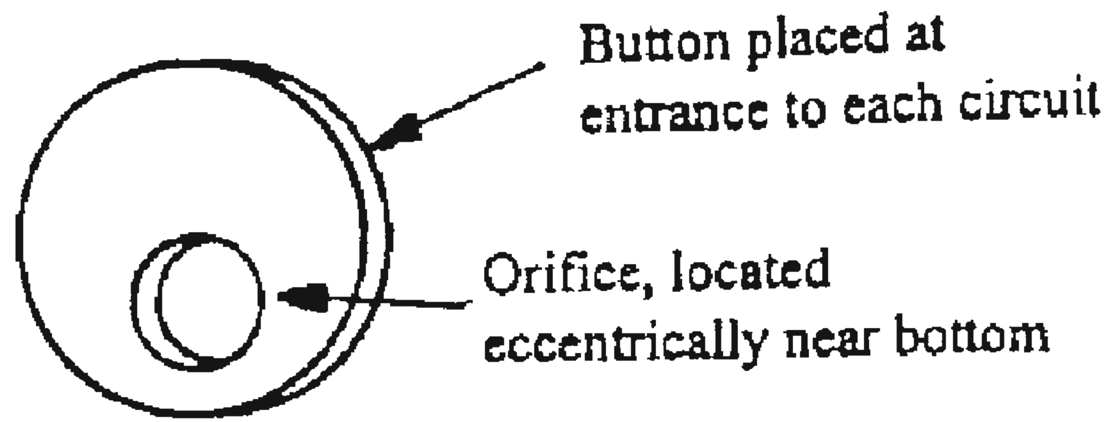
- copper tube/aluminum fin for halocarbon air-cooling coils
- aluminum tube/aluminum fin for halocarbon or ammonia air-cooling coils
- carbon steel tube/carbon steel fin for air-cooling coils using ammonia, halocarbons, antifreezes or water in the tubes
- stainless steel tube/stainless steel fin when special cleaning provisions are required on the air side

The application of the coil, particularly whether it will become frosted, determines to a large extent the spacing of fins. In air conditioning coils with thin aluminum fins, the spacing may be 470 per m (12 fins per inch, FPI), while industrial coils are usually built with 118 or 158 fins per m (3 or 4 FPI). Coils serving spaces where the air temperature is below freezing usually have a fin spacing of 118 per m (3 per inch).


Outlet
Up
Orifices inside
liquid header
Liquid
supply
Liquid
supply
Down
Outlet

(a)

(b)


Button placed at
entrance to each circuit
Orifice, located
eccentrically near bottom

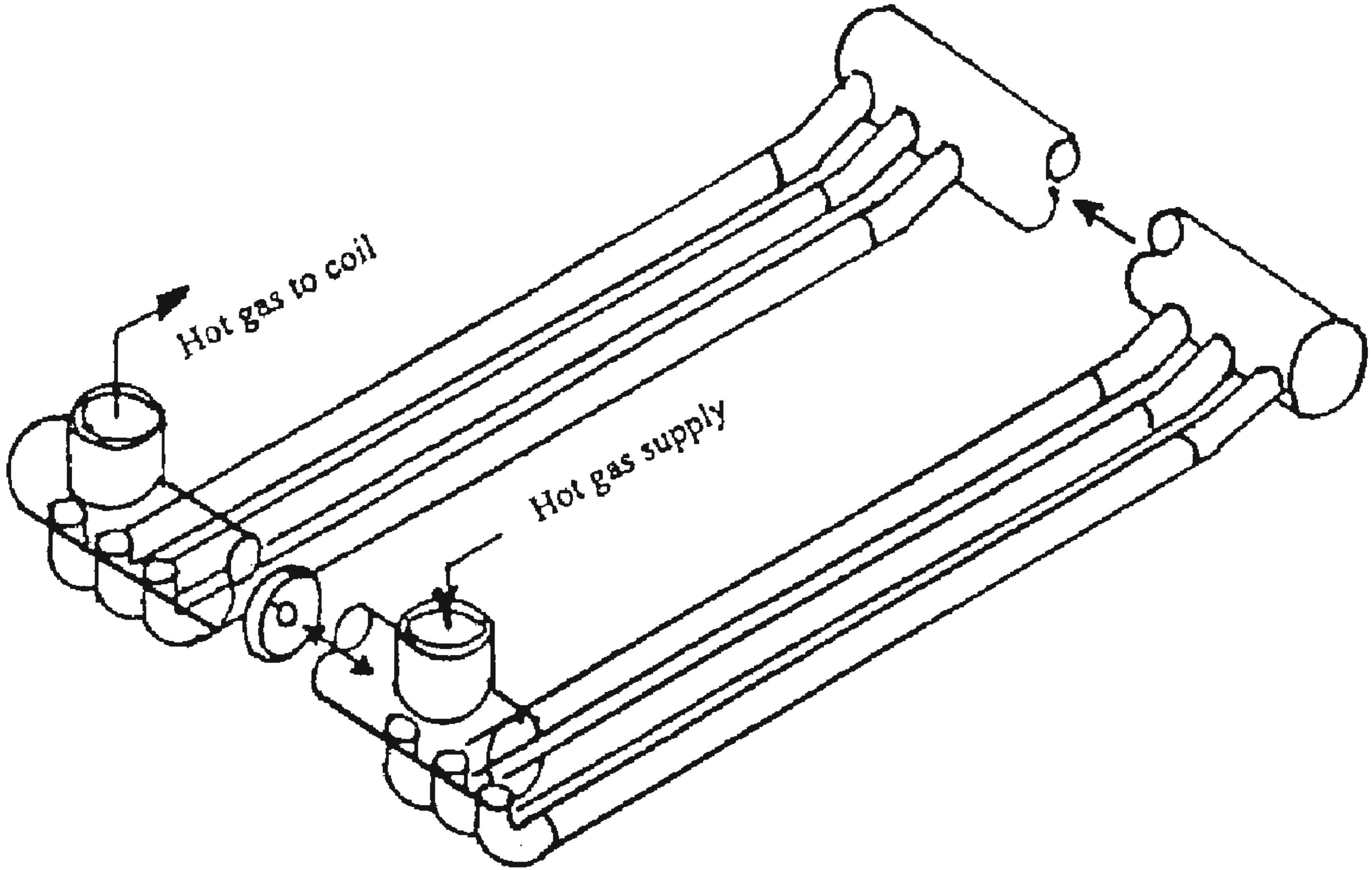
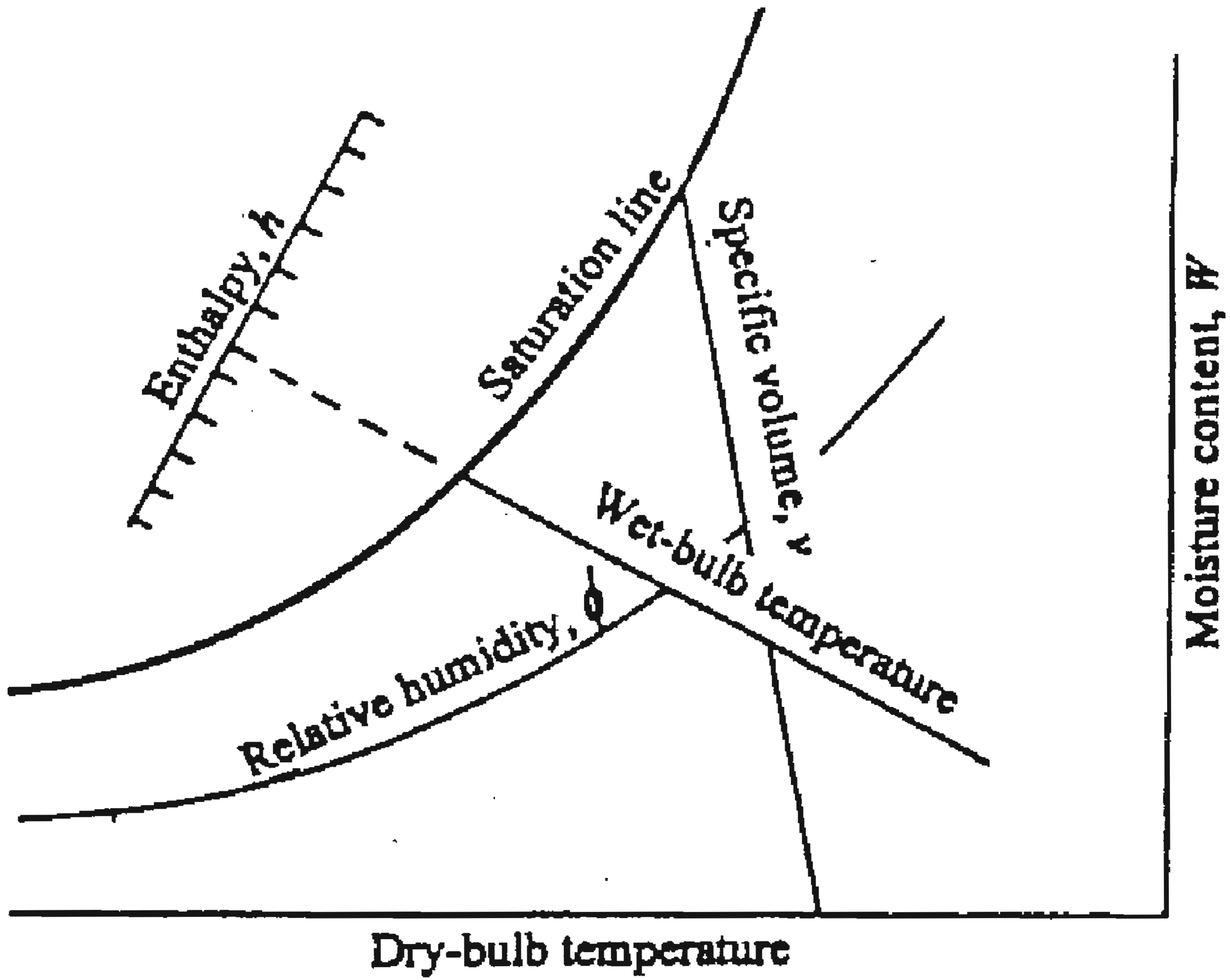


FIGURE 6.17
Pipe coils embedded in the drain pan in order to facilitate drainage of the melted frost in a low-temperature coil.

- Psychrometric chart



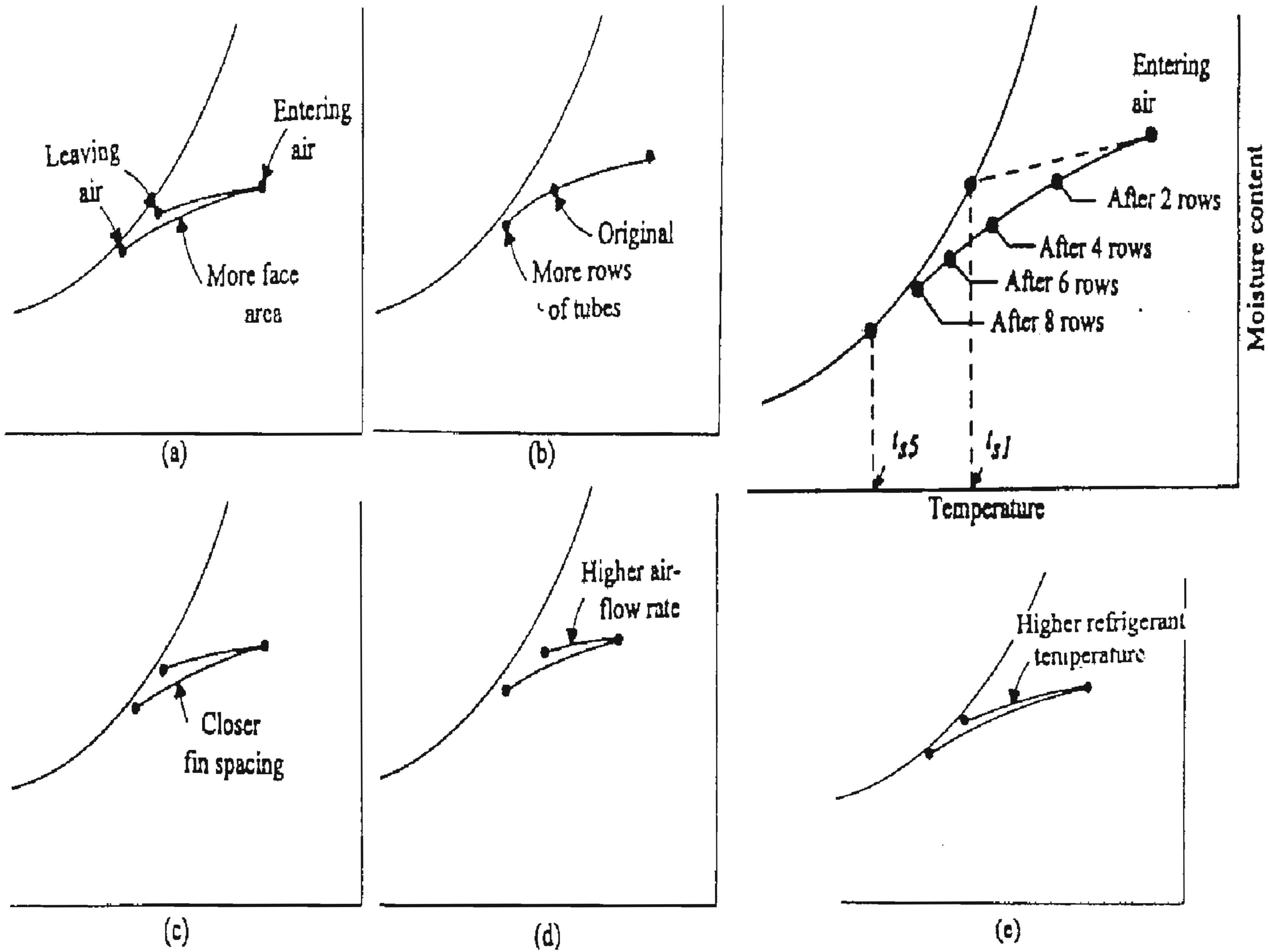

Leaving air
Entering air
More face area
(a)
Original
More rows of tubes
(b)
Entering air
After 2 rows
After 4 rows
After 6 rows
After 8 rows
t_{s5}
t_{s1}
Temperature
Moisture content
Closer fin spacing
(c)
Higher air-flow rate
(d)
Higher refrigerant temperature
(e)

TABLE 6.5
Influence of design or operating parameters on outlet air conditions from an evaporator coil.

Parameter, increase of:	Effect on outlet air conditions		Refrigerating capacity	Typical range
	temperature	moisture content		
Face area	lower	lower	higher	depends on refrigerating capacity
Number of rows of tubes deep	lower	lower	higher	four to eight
Number of fins per unit length	lower	lower	higher	115 to 300 fins per m (3 to 8 fins per in)
Air flow rate	higher	higher	higher	Face velocity 2 to 4 m/s (400 to 800 fpm)
Refrigerant temperature	higher	higher	lower	3 to 8°C (5 to 15°F) below entering air

Typical information

- the space temperature
- the saturated suction temperature
- the refrigeration capacity
- which refrigerant is to be used
- the type of feed, for example, whether liquid recirculation, flooded coils, or direct expansion. If liquid recirculation is chosen whether top or bottom feed should be specified as well as the recirculation rate.

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.

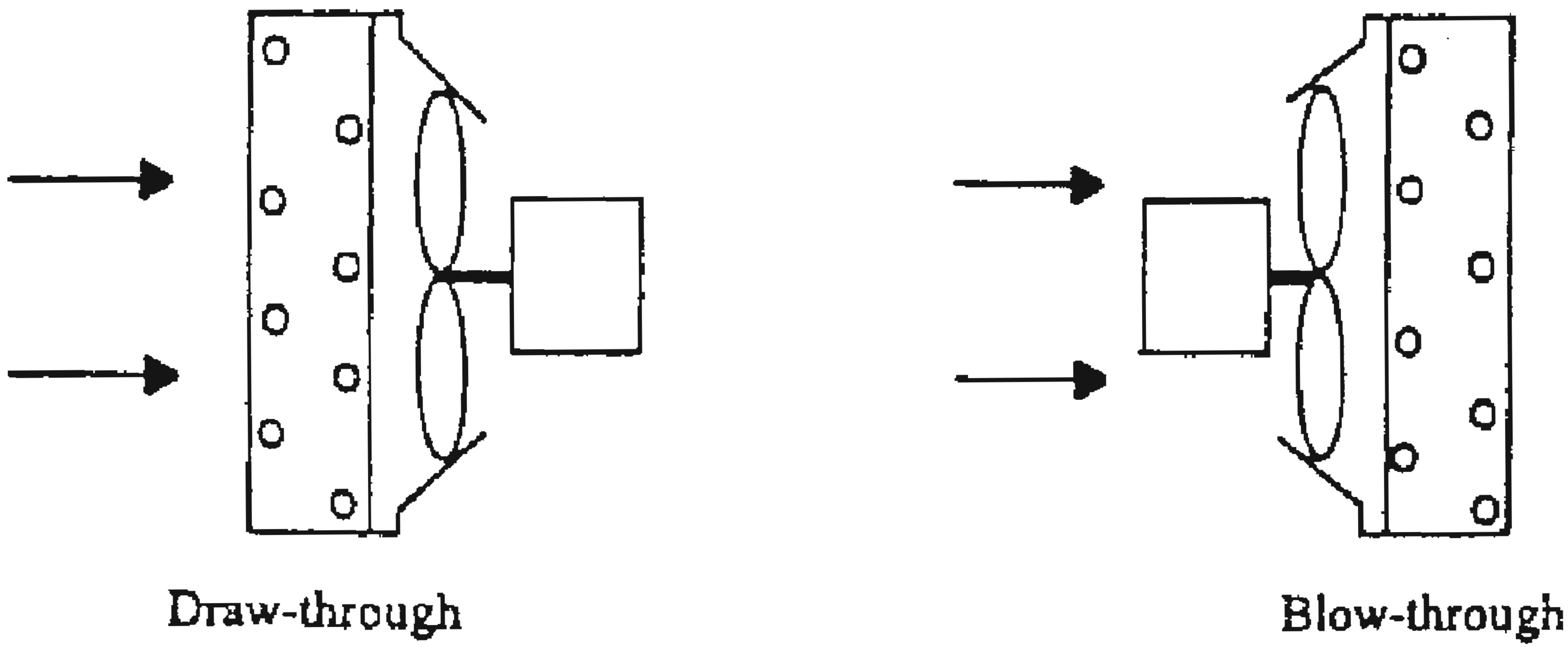


FIGURE 6.36
Draw-through and blow-through arrangements of the fan and coil.

25

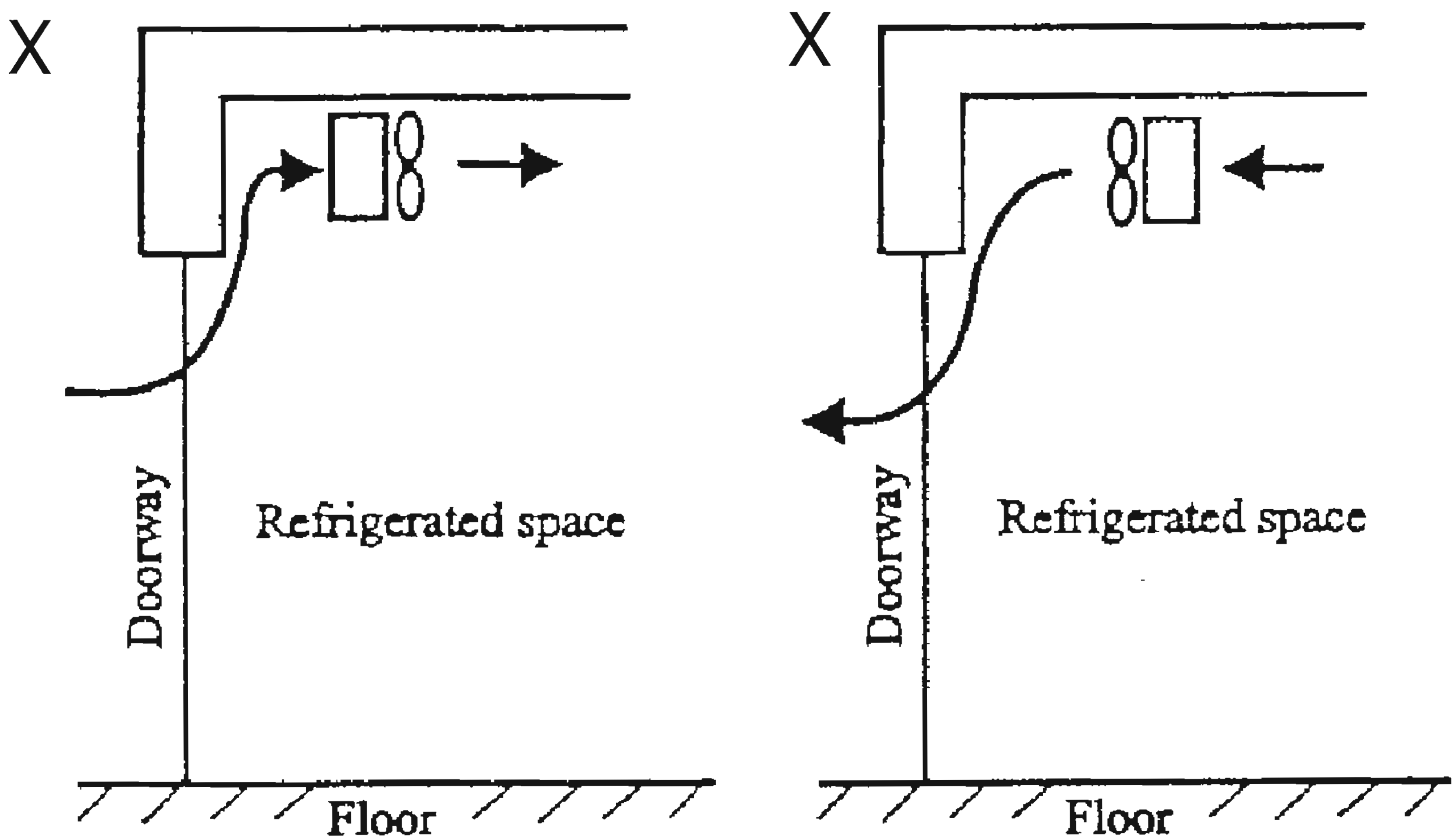
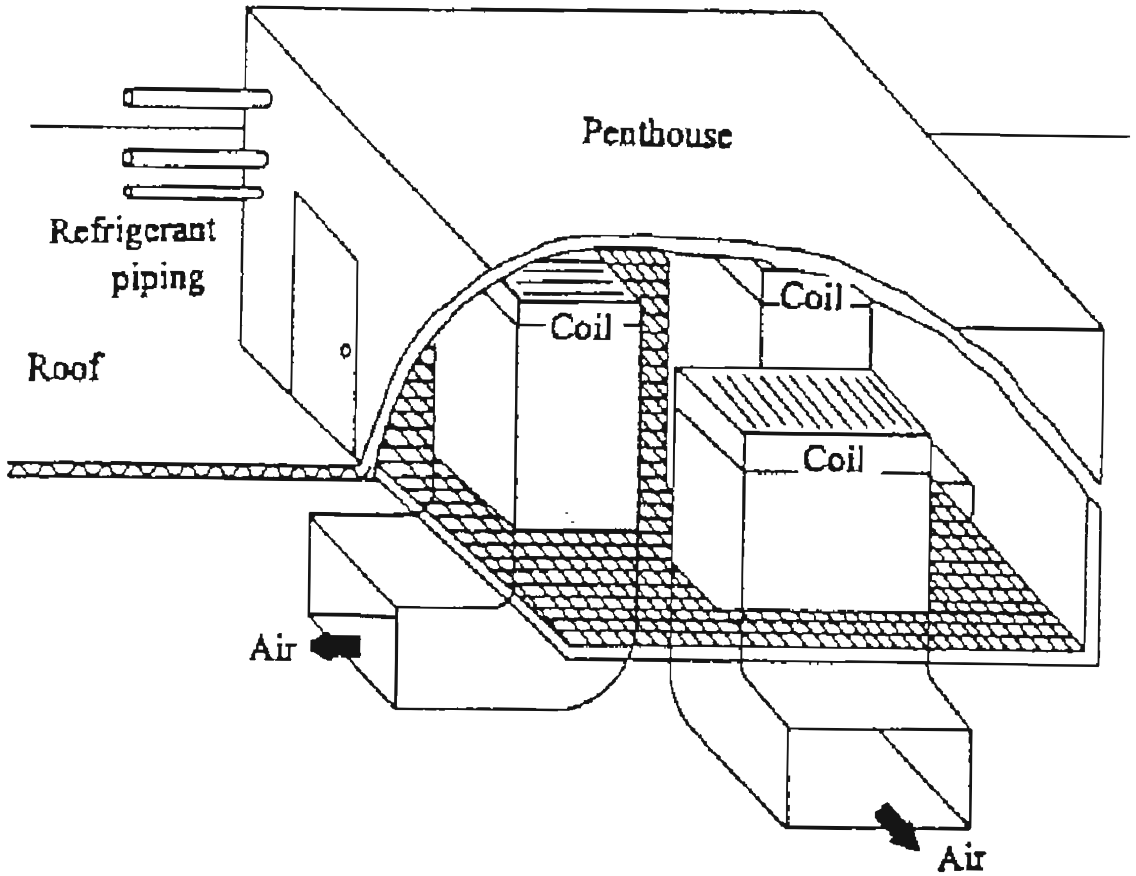
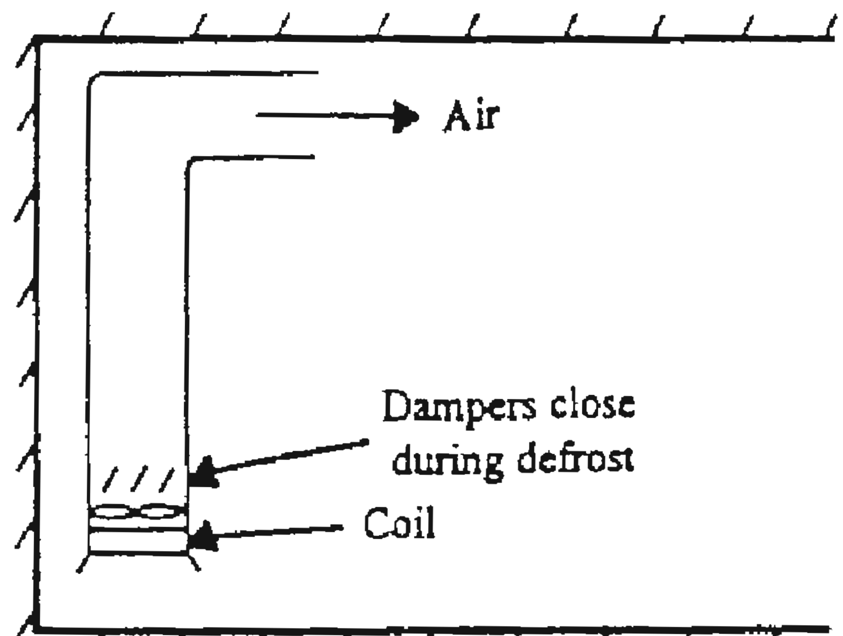


FIGURE 6.39
Do not mount coils above a doorway such that they draw warm air in through or discharge cold air out through the doorway. Instead, arrange for the air to flow past the doorway.


Penthouse
Refrigerant piping
Roof
Coil
Coil
Coil
Air
Air


Air
Dampers close during defrost
Coil

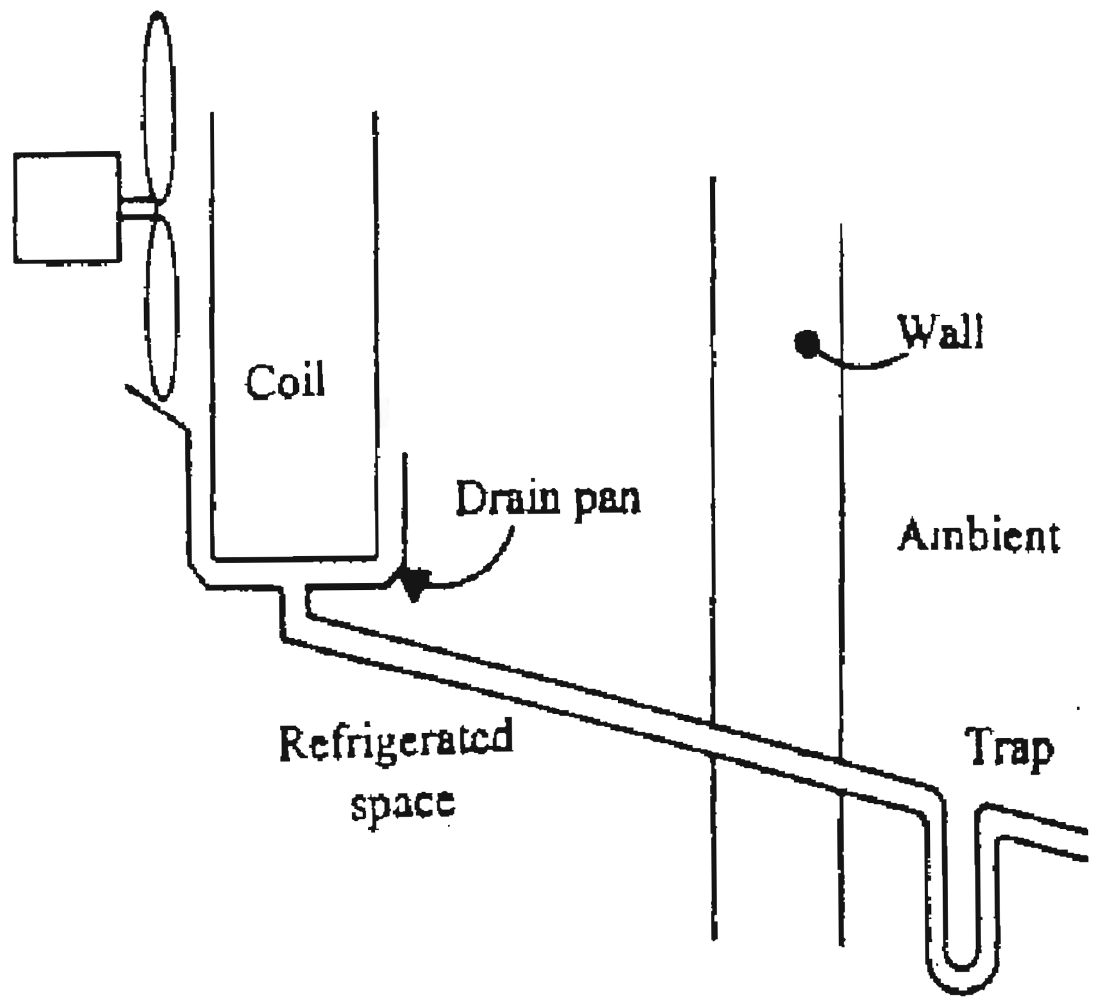


FIGURE 6.45
A trap in the drain line that carries defrost water to outside the refrigerated space.

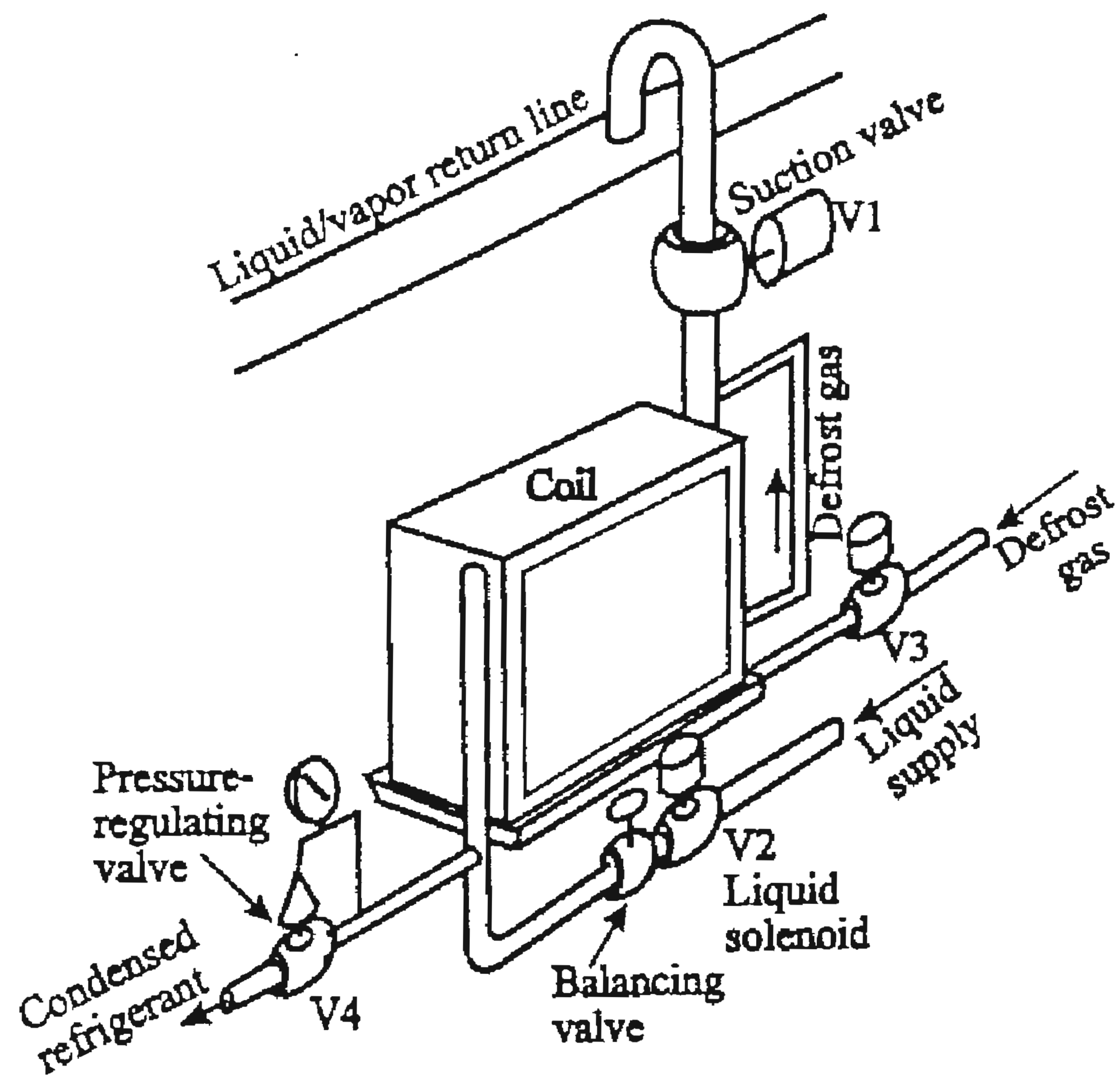


FIGURE 6.46
Elements of a bottom-feed liquid circulation coil equipped with hot-gas defrost.

TABLE 6.9
Status of valves in Fig. 6.46 during refrigeration and during defrost operation.

Operation	V1	V2	V3	V4
Refrigeration	Open	Open	Closed	Closed, because valve is attempting to raise pressure
Defrost	Closed	Closed	Open	Opens when coil pressure rise rises above pressure setting

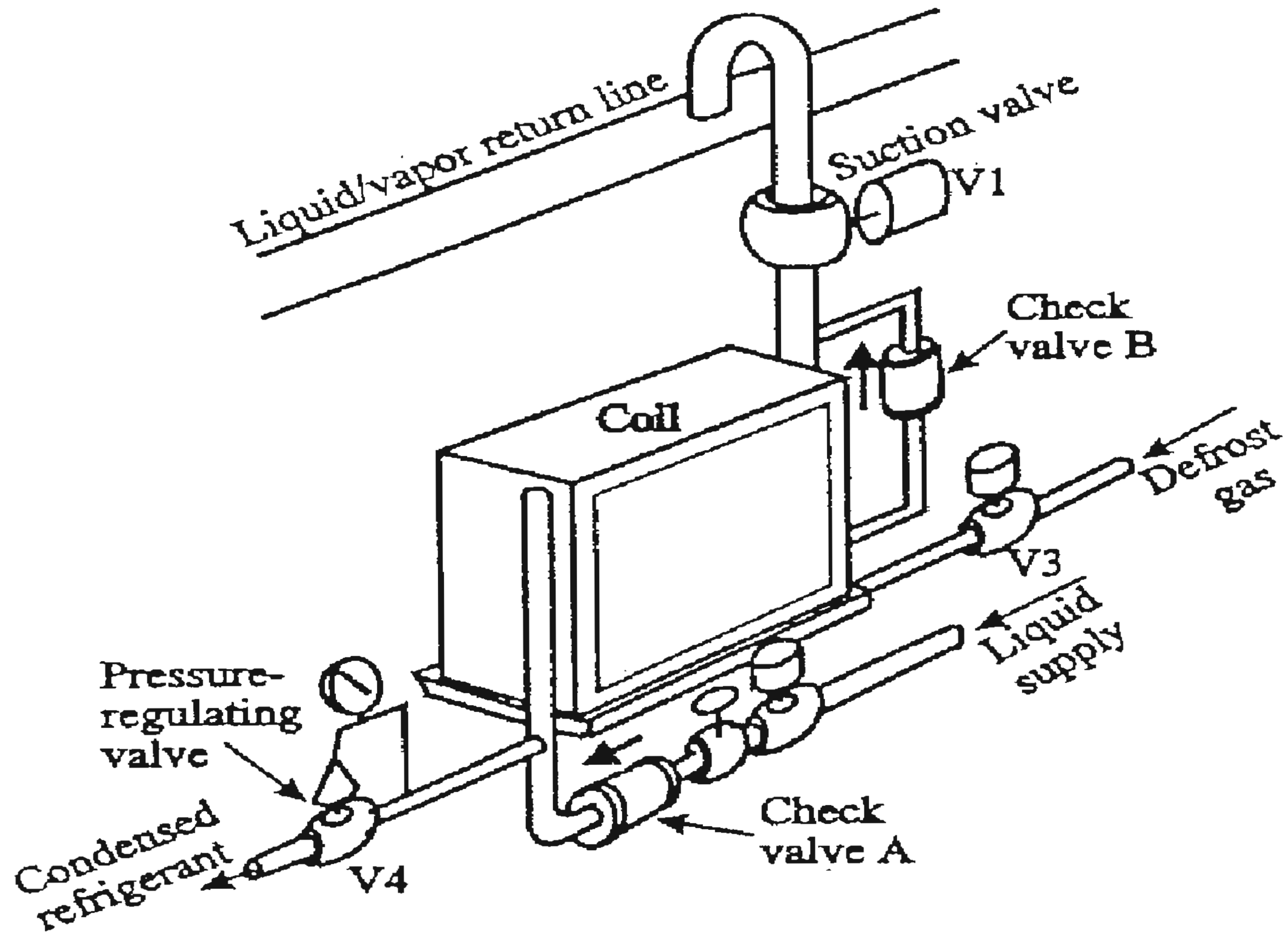


FIGURE 6.47
Addition of two check valves.

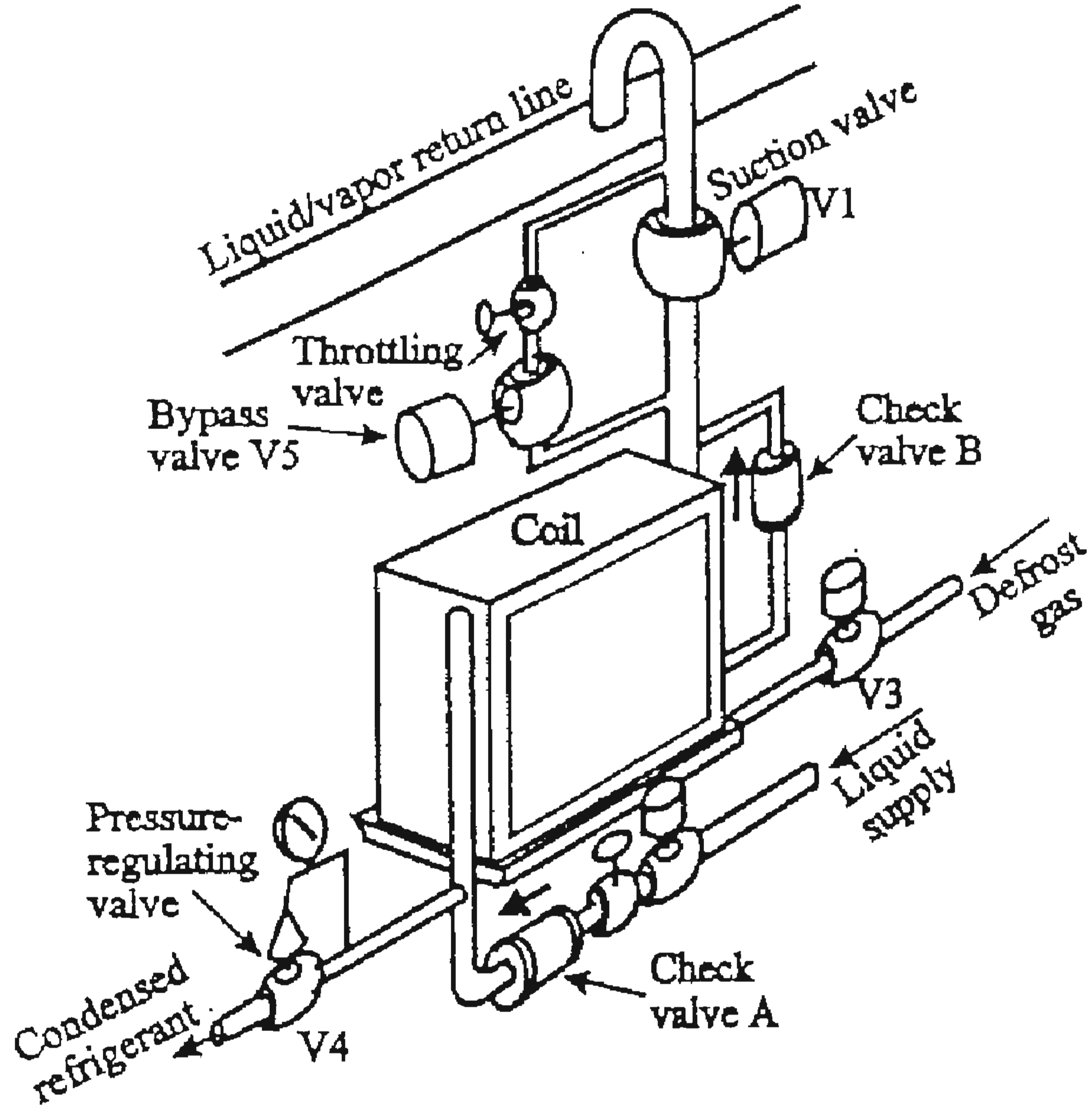


FIGURE 6.48
Bypass around the suction valve to slowly relieve the pressure in the coil at the termination of defrost.

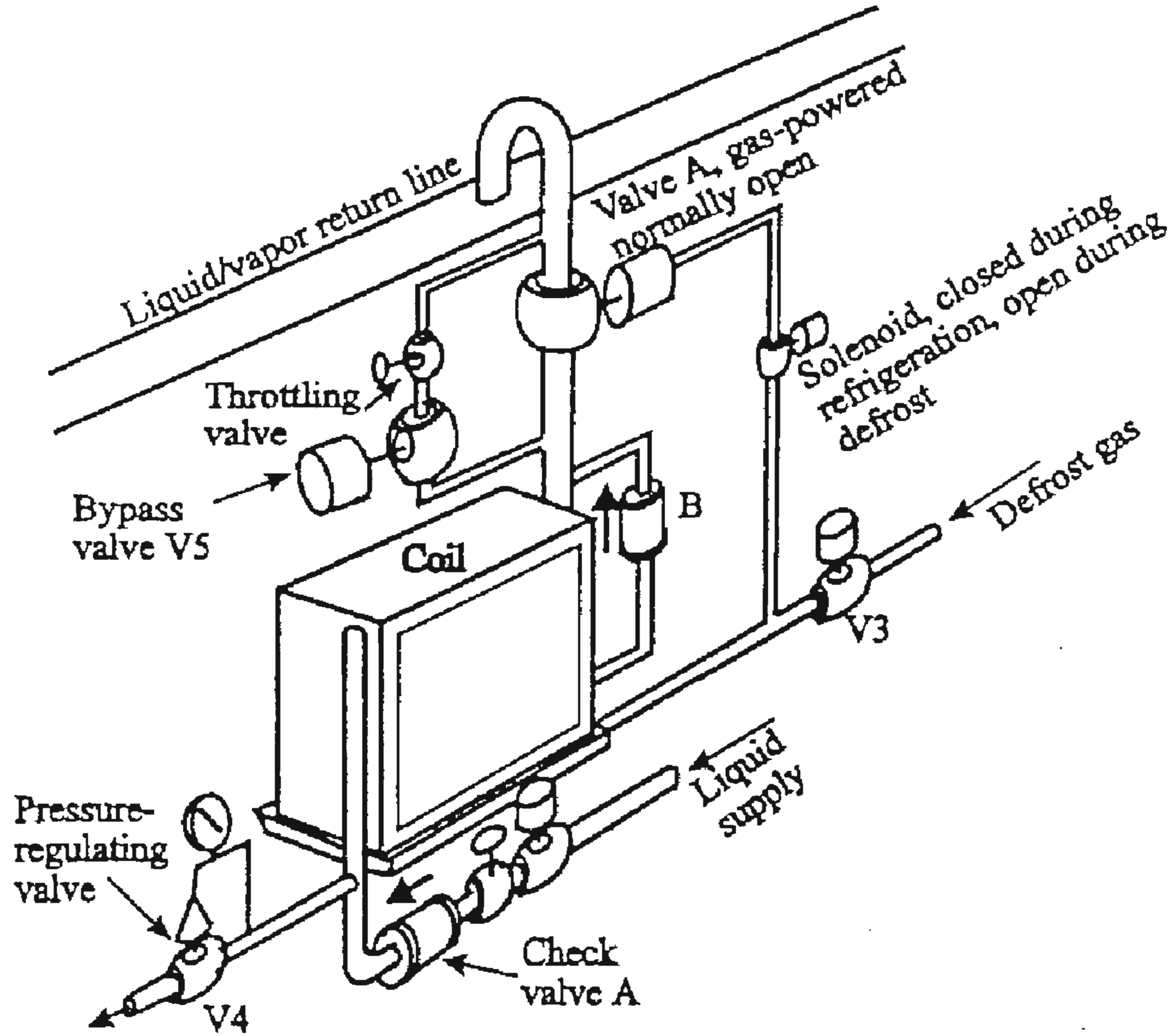


FIGURE 6.49
Defrost control group with gas-powered suction valve, bottom-feed liquid recirculation coil.

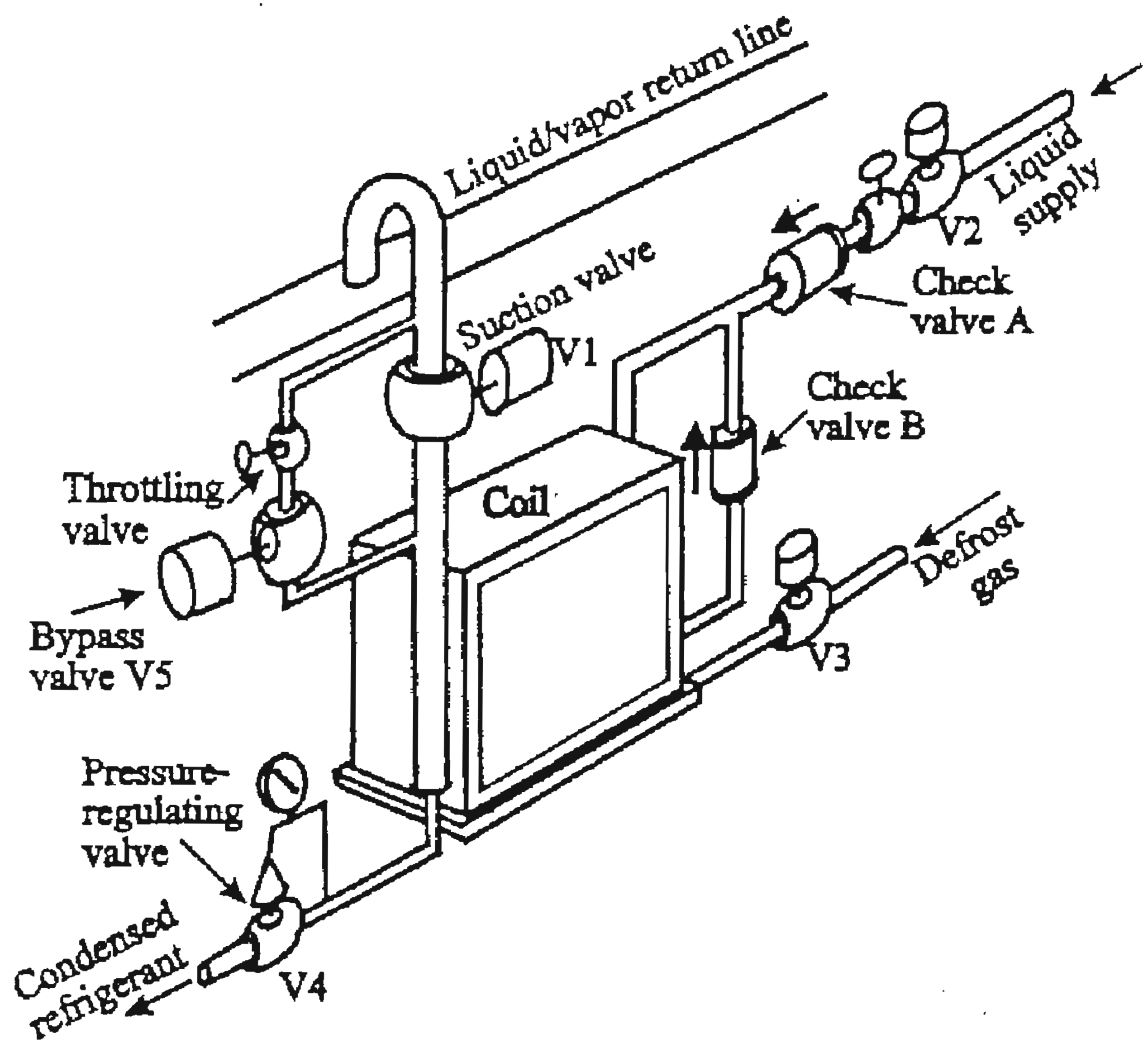


FIGURE 6.50
Defrost control group for a top-feed liquid-recirculation coil.

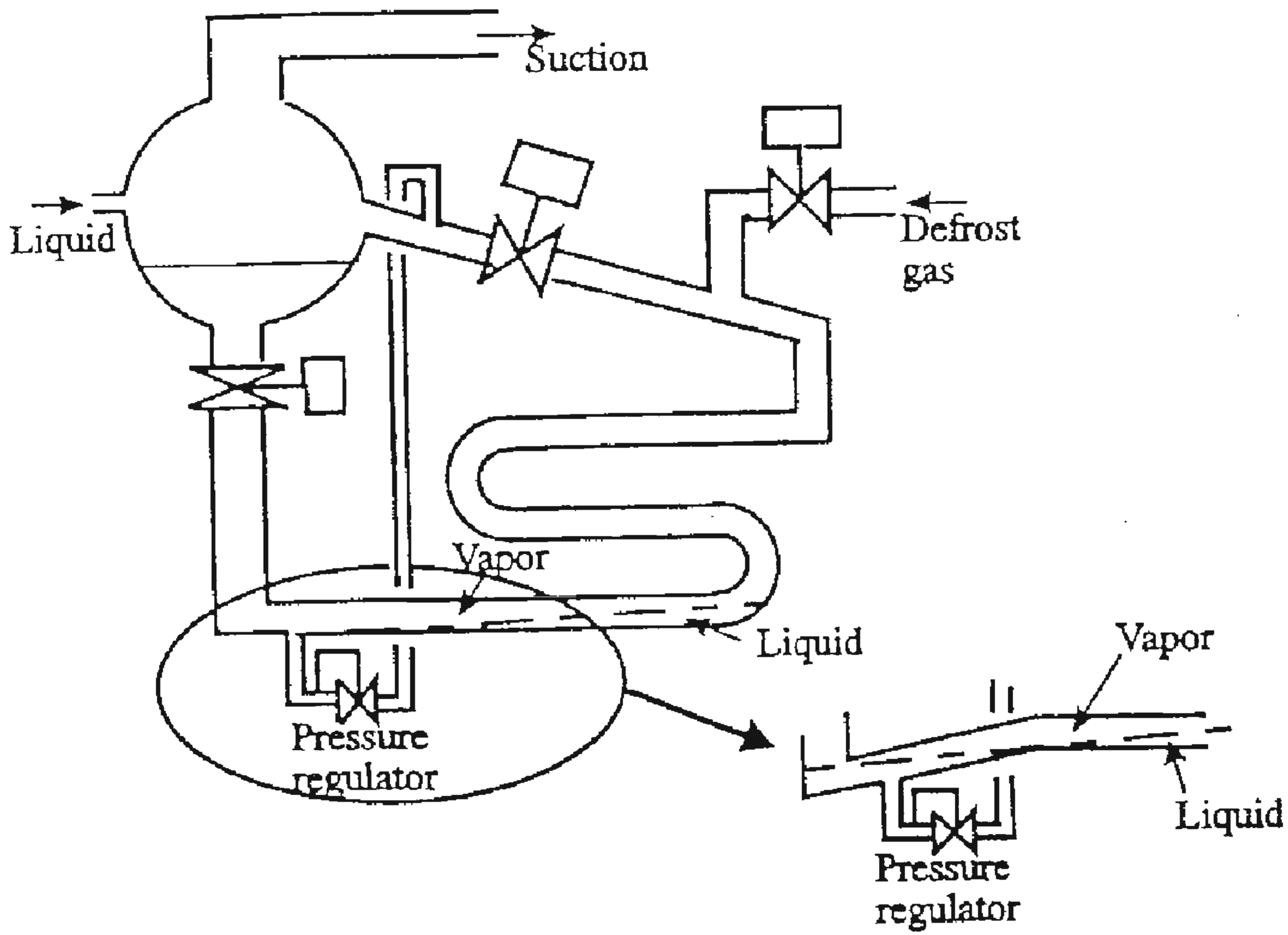


FIGURE 6.51
Valves and piping for defrosting a flooded coil, with the line from the liquid leg to the coil sloped upward to achieve a liquid seal during defrost.

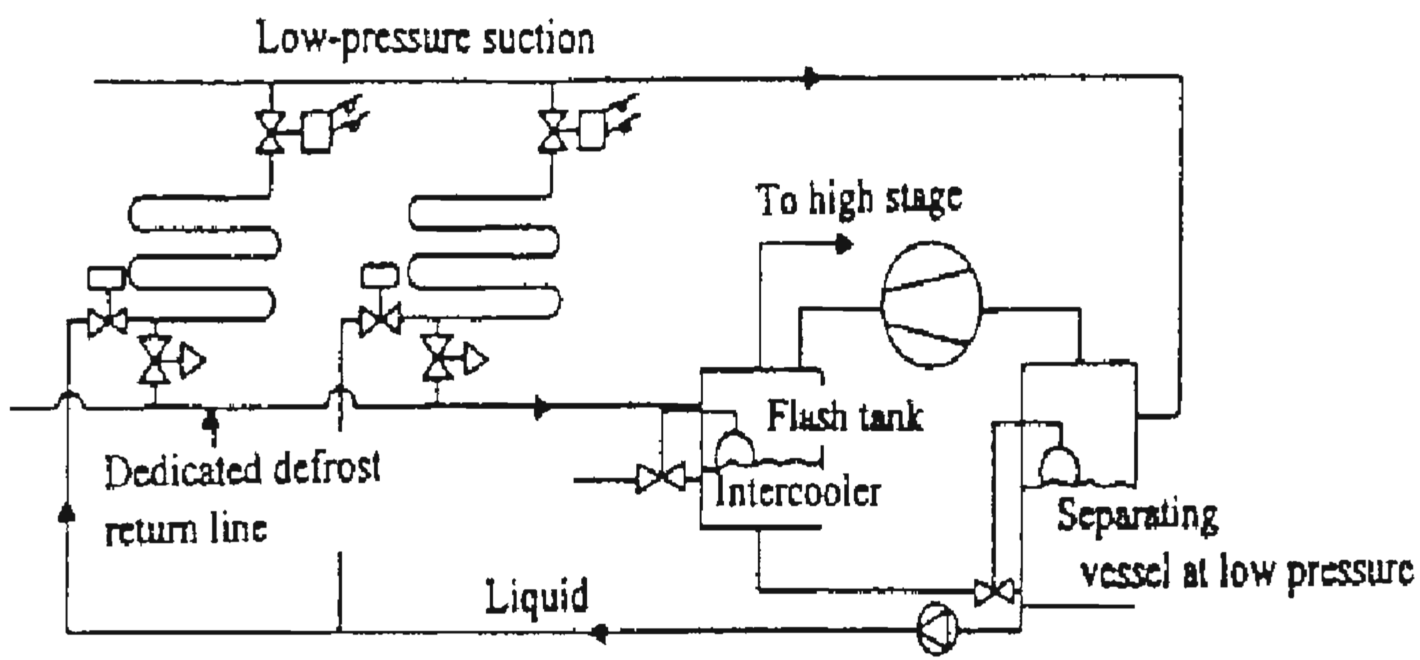


FIGURE 6.53
Returning discharge liquid and vapor from defrosting coils to the intermediate pressure in a two-stage system.

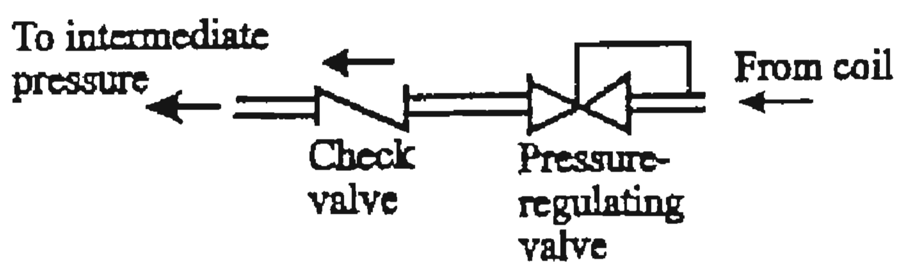


FIGURE 6.54
Placing a check valve in series with the pressure-regulating valve when discharging to intermediate pressure.

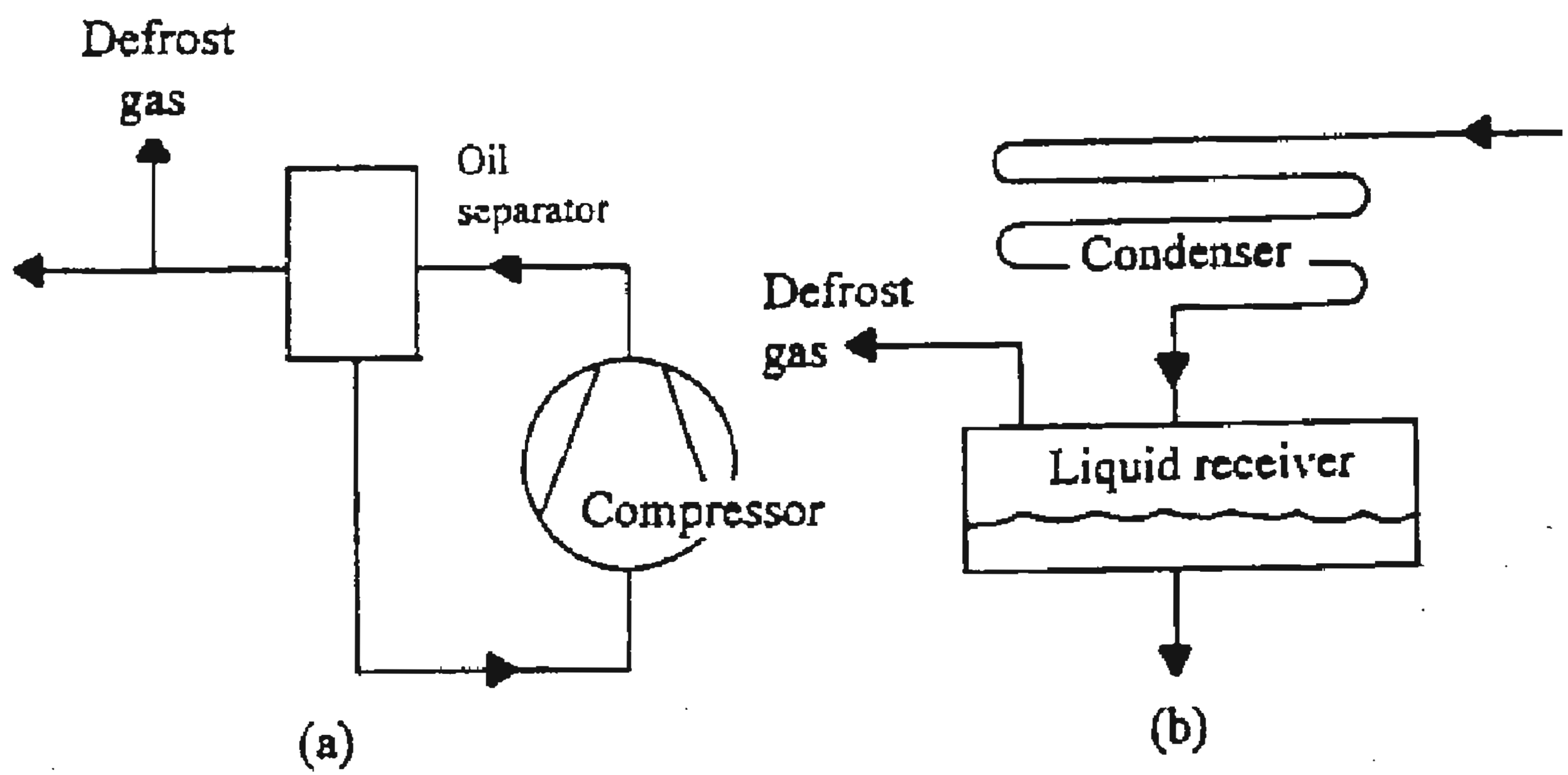


FIGURE 6.55
Source of defrost gas from (a) compressor discharge line, and (b) from the high-pressure liquid receiver.

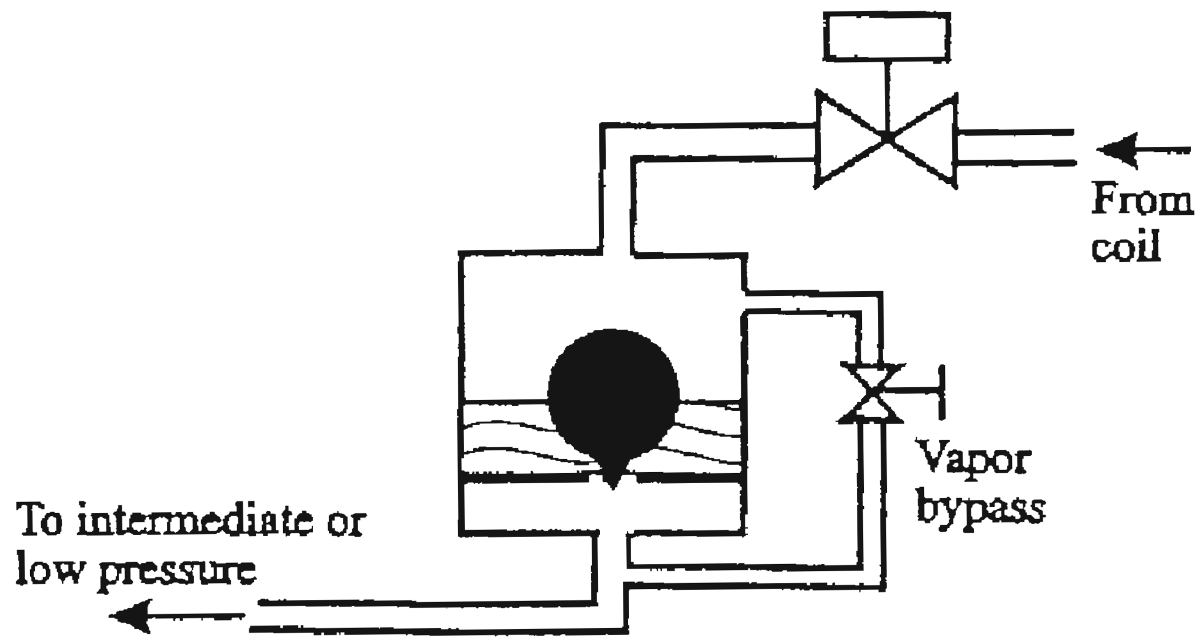


FIGURE 6.56
A liquid drainer replacing the pressure-regulating valve to control the flow of refrigerant condensate from the coil during defrost.

■ Water defrost :

- the rate of defrost water should be between 1 to 1.36 L/s per m^2 of coil face area (1-1/2 to 2 gpm per ft^2).
- a water temperature of about 16°C (61°F) is an acceptable compromise. The higher the water temperature the more rapid will be the defrost, but a high water temperature also results in excessive fogging in the neighborhood of the coil. The rate of vaporization of water into fog is controlled by the water vapor pressure which in turn is a function of the water temperature. Furthermore the water vapor pressure is 3-1/2 times higher at 30°C (86°F) than it is at 10°C (50°F).
- the rate of water to be discharged is that of the melted frost plus the defrost water, so the quantity that the drain pan and drain lines must handle will be considerable more than with a coil defrosted by hot gas.
- the solenoid valve controlling the defrost water should be in a warm environment so that the water line will not freeze. Also, from the position of this valve, the piping should be sloped so that negligible water is retained in the line between the valve and the sprayheads at the coil.
- the pumpout phase to evacuate the coil of refrigerant first specified for hot-gas defrost is equally important for water defrost.

Advantages of aluminum:

- lighter weight
- better heat transfer. While aluminum may possess five times the conductivity of steel, the influence of this factor applies only to heat transfer through the fins and through the tubes. The air-side and refrigerant-side heat-transfer coefficients dominate, so the favorable conductivity of aluminum results in a 10-12% improvement in heat-transfer rate for a coil of given construction.
- less corrosive in acidic and dry SO_2 atmospheres[23] (but see also disadvantages of aluminum)
- achieves more rapid defrost

Disadvantages of aluminum:

- ability to handle stress and physical blows
- higher cost than galvanized steel
- more difficult to repair in the field
- more corrosive when subjected to chlorine in cleaning solutions and when in contact with calcium chloride brine

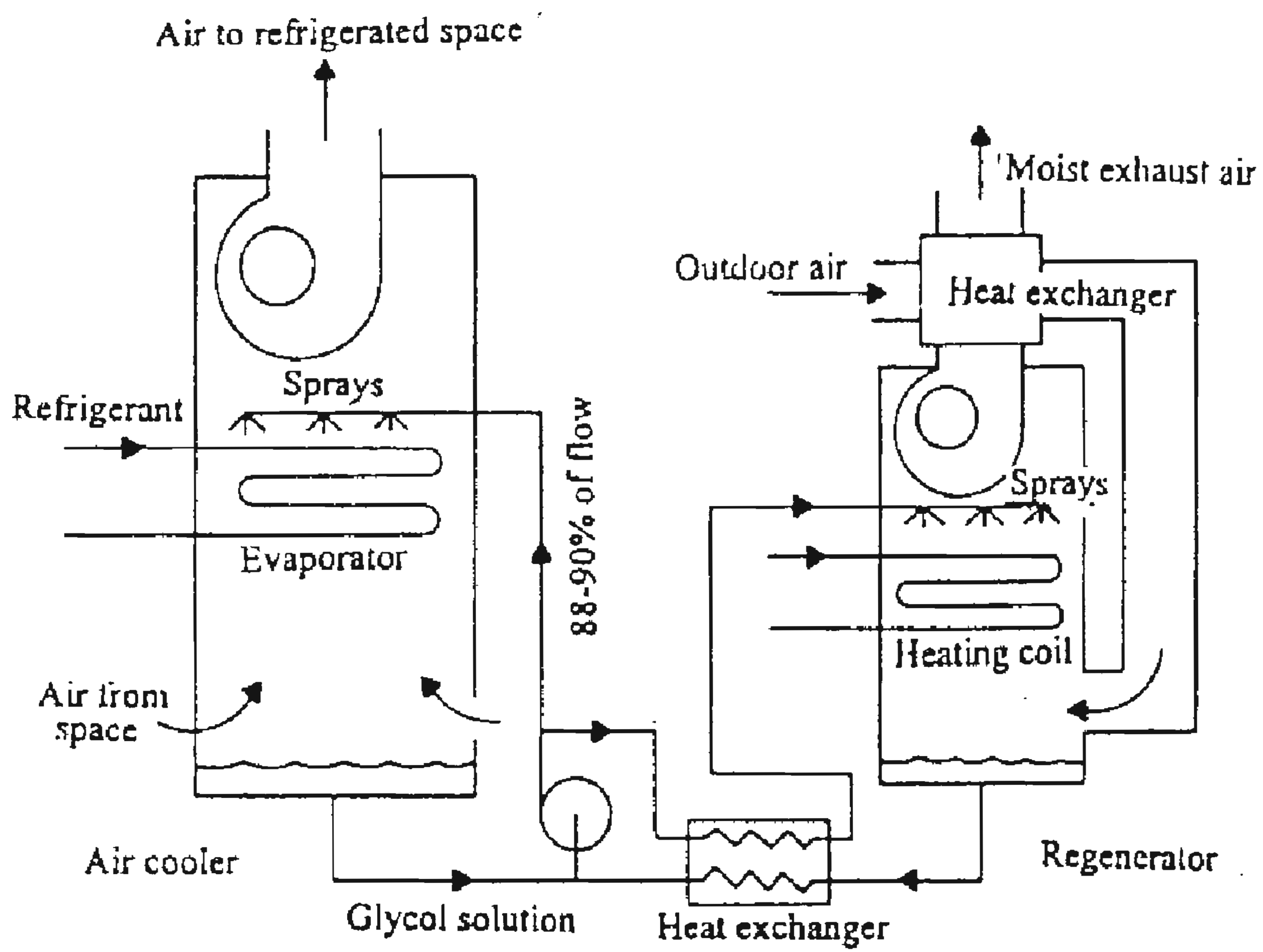


FIGURE 6.57
Schematic diagram of equipment that sprays the evaporator coil with a glycol solution while simultaneously regenerating the solution.

The total resistance to heat transfer, R_{total}, of the chiller is the sum of the individual resistances,

$$\frac{1}{U_o A_o} = R_{total} = \frac{1}{h_{refrig} A_o} + R_{metal} + \frac{1}{h_{liquid} A_i} \tag{6.14}$$

where h = heat-transfer coefficient, $W/m^2 \cdot K$ ($Btu/hr \cdot ft^2 \cdot °F$)

A = heat-transfer area, m^2 (ft^2), with subscript o indicating the outside and i the inside area

Equation 6.14 implies that the refrigerant is in the shell and liquid flows in the tubes. For the tube-side coefficient a standard equation is

$$\frac{hD}{k} = 0.023 \left(\frac{VD\rho}{\mu}\right)^{0.8} \left(\frac{\mu c_p}{k}\right)^{0.4} \tag{6.15}$$

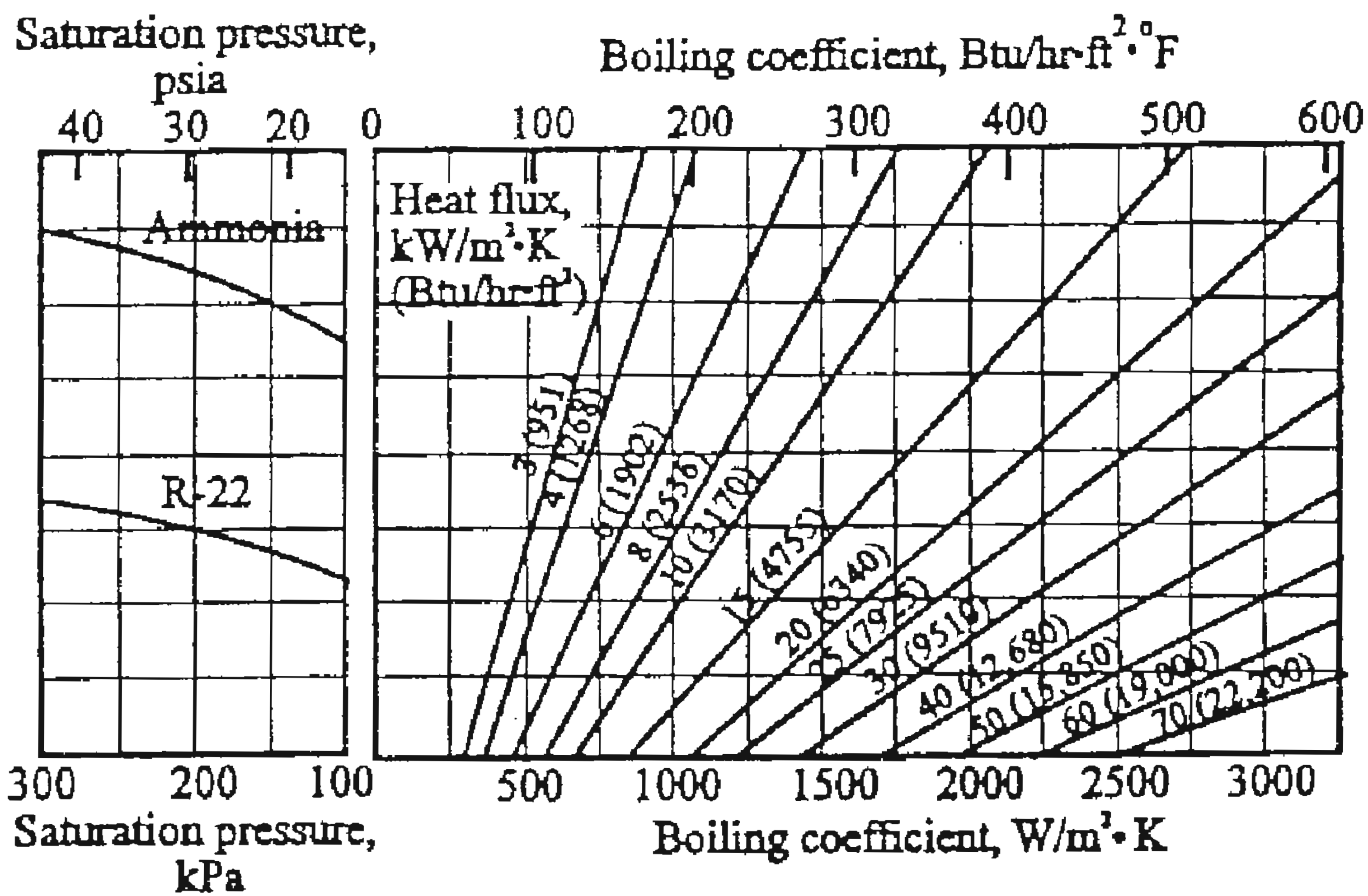


FIGURE 6.61
Heat-transfer coefficient for pool boiling from a horizontal cylinder.

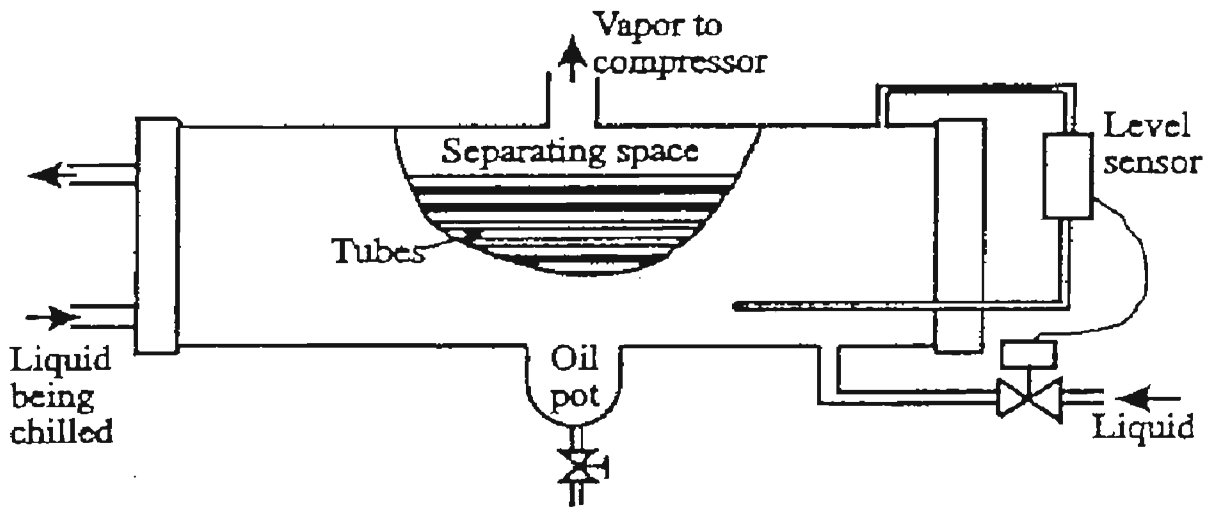


FIGURE 6.59
A shell-and-tube evaporator with refrigerant in the shell and in which separation space is provided above the tubes.

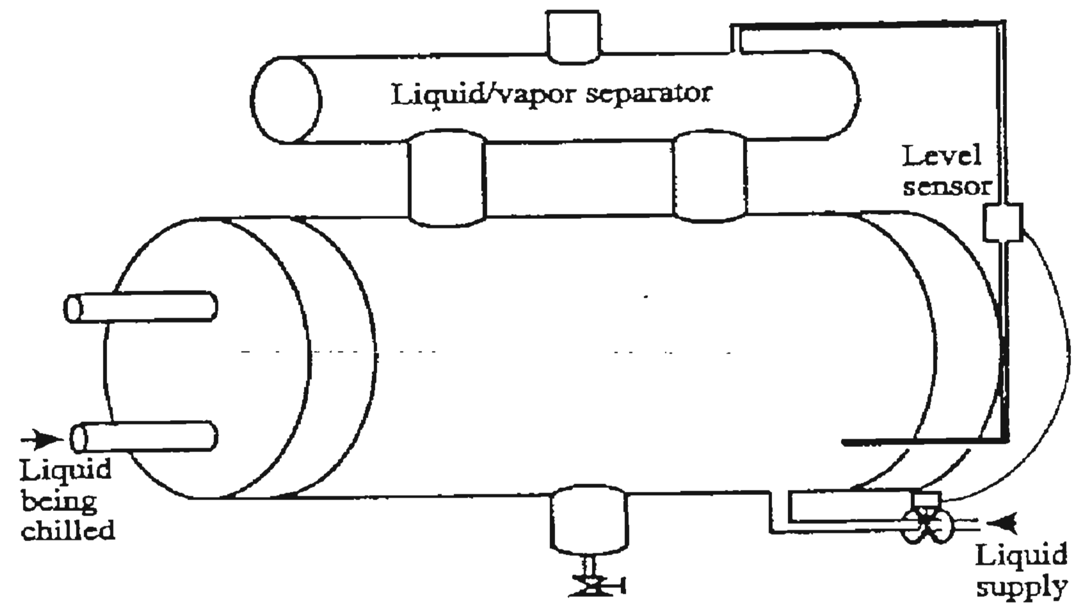
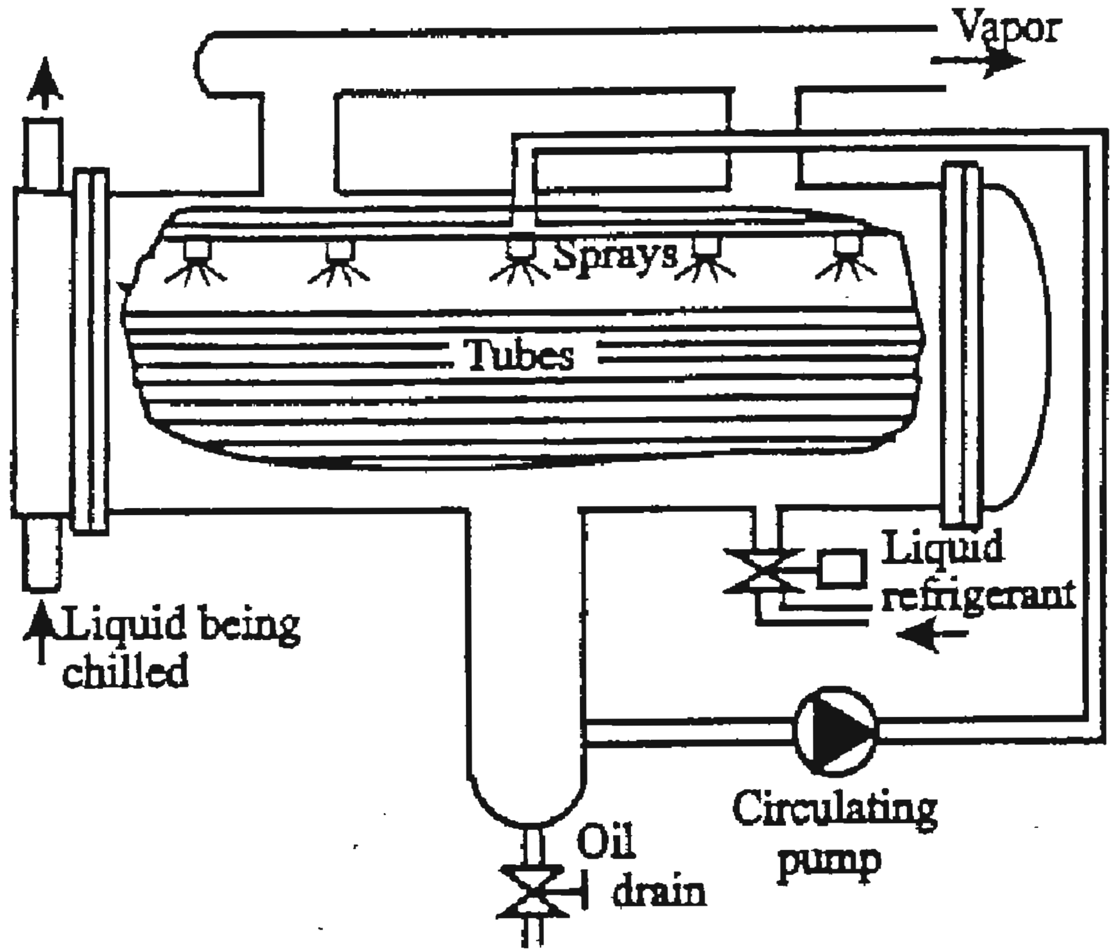
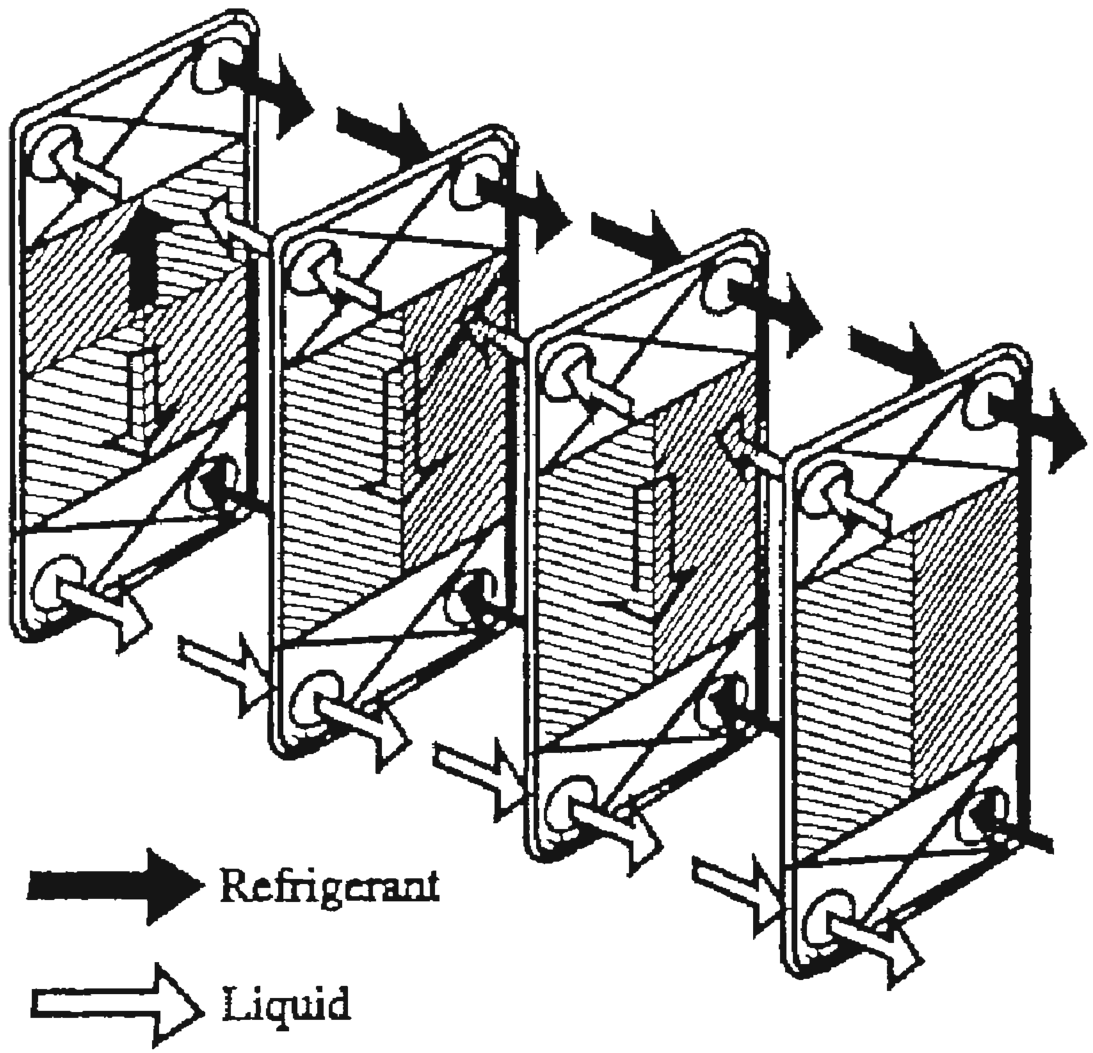


FIGURE 6.60
A shell-and-tube evaporator with refrigerant in the shell and an auxiliary vessel above the main evaporator to facilitate separation of liquid and vapor.


Vapor
Sprays
Tubes
Liquid refrigerant
Liquid being chilled
Circulating pump
Oil drain


Refrigerant
Liquid

Cooling Load

1. heat that leaks into the refrigerated space from the outside by conduction through the insulated walls,
2. heat that enters the space by direct radiation through glass or other transparent materials,
3. heat that is brought into the space by warm outside air entering the space through open doors or through cracks around windows and doors,
4. heat given off by a warm product as its temperature is lowered to the desired level,
5. heat given off by people occupying the refrigerated space,
6. heat given off by any heat-producing equipment located inside the space, such as electric motors, lights, electronic equipment, steam tables, and materials handling equipment.

Equipment Running Time

$$\text{Required Btu/hr equipment capacity} = \frac{\text{Total cooling load, Btu/24 hr}}{\text{Desired running time (hours)}}$$

Factors Determining the Wall Gain Load

$$Q = (A)(U)(D)$$

where Q = the quantity of heat transferred in Btu per hour

A = the outside surface area of the wall (square feet)

U = the overall coefficient of heat transmission in Btu per hour per square foot per degree Fahrenheit

D = the temperature differential across the wall in degrees Fahrenheit

TABLE 10-1 THERMAL CONDUCTIVITY OF MATERIALS USED IN COLD STORAGE WALLS

Material	Description	Thermal conductivity (k)	Thermal conductance (C)
Masonry	Brick, common	5.0	
	Brick, face	9.0	
	Concrete, mortar or plaster	5.0	
	Concrete, sand aggregate	12.0	
	Concrete block		
	Sand aggregate 4 in.		1.40
	Sand aggregate 8 in.		0.90
	Sand aggregate 12 in.		0.78
	Cinder aggregate 4 in.		0.90
	Cinder aggregate 8 in.		0.58
	Cinder aggregate 12 in.		0.53
	Gypsum plaster 1/2 in.		3.12
	Tile, hollow clay 4 in.		0.90
	Tile, hollow clay 6 in.		0.66
	Tile, hollow clay 8 in.		0.54
Woods	Maple, oak, similar hardwoods	1.10	
	Fir, pine, similar softwoods	0.80	
	Plywood 1/2 in.		1.60
	Plywood 3/4 in.		1.07
Roofing	Asphalt roll roofing		6.50
	Built-up roofing 3/8 in.		3.00
Insulating materials	Blanket or batt, mineral or glass fiber	0.27	
	Board or slab		
	Cellular glass	0.40	
	Corkboard	0.30	
	Glass fiber	0.25	
	Polystyrene (extruded)	0.20	
	Polystyrene (molded beads)	0.25	
	Polyurethane (extruded)	0.16	
	Polyurethane (board)	0.18	
	Loose fill		
	Milled paper or wood pulp	0.27	
	Sawdust or shavings	0.45	
	Mineral wool (rock, glass, slag)	0.27	
	Redwood bark	0.26	
	Wood fiber (soft woods)	0.30	
Surface conductance (convection coefficient)	Still air		1.65
	Moving air (7.5 mph)		4.00
	Moving air (15 mph)		6.00
Glass	Single pane		1.13
	Two pane		0.46
	Three pane		0.29
	Four pane		0.21

From *ASHRAE Data Book*, Fundamentals Volume, 1972 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

4.

$$R = \frac{1}{U} = \frac{1}{f_i} + \frac{x}{k_1} + \frac{x}{k_2} + \frac{x}{k_n} + \frac{1}{f_0}$$

Therefore,

$$U = \frac{1}{\frac{1}{f_i} + \frac{x}{k_1} + \frac{x}{k_2} + \cdots \frac{x}{k_n} + \frac{1}{f_0}}$$

where f_i = convection coefficient (surface conductance) of inside wall, floor, or ceiling

f_0 = convection coefficient (surface conductance) of outside wall, floor, or roof

Note When nonhomogeneous materials are used, $1/C$ is substituted for x/k.

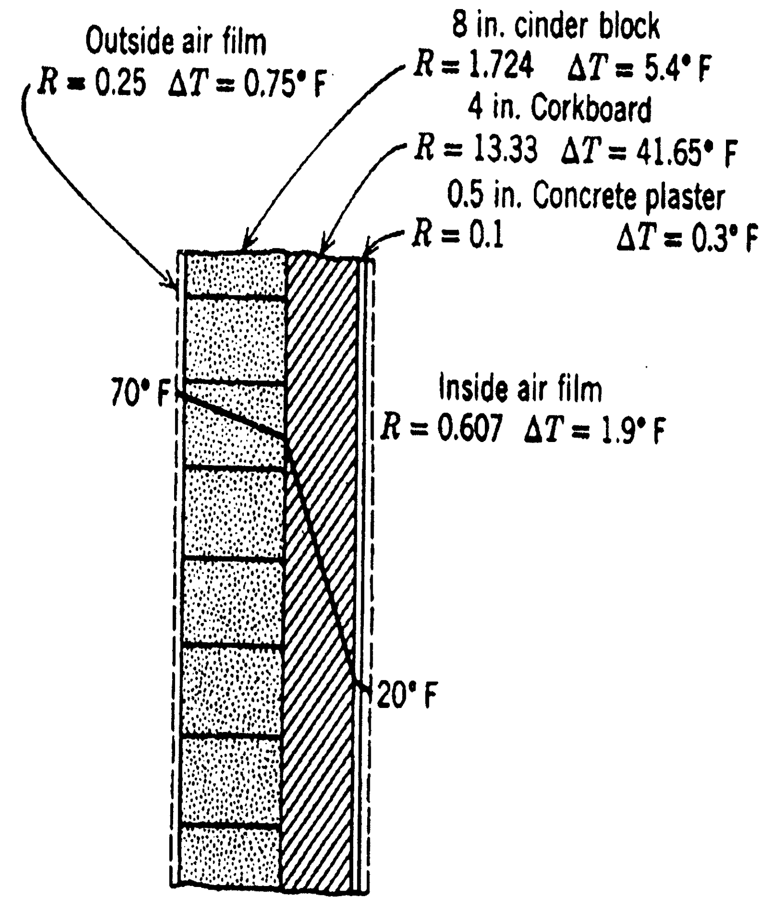


Fig. 10-2 Temperature gradient through a typical cold storage.

Example 10-3 Assuming a wind velocity of 7.5 mph, calculate the value of U for a wall constructed of 8 in. cinder aggregate building blocks, insulated with 4 in. of polyurethane board, and finished on the inside with 0.5 in. of cement plaster.

Solution From Table 10-1,

8 in. cinder aggregate block	$C = 0.58$
Polyurethane board	$k = 0.18$
Cement plaster	$k = 5.00$
Inside convection coefficient	$f_i = 1.65$
Outside convection coefficient	$f_o = 4.00$

Applying Equation 10-5, the overall thermal resistance, $1/U$

$$= \frac{1}{4} + \frac{1}{0.58} + \frac{4}{0.18} + \frac{0.5}{5} + \frac{1}{1.65}$$

$$= 0.25 + 1.724 + 22.222 + 0.1 + 0.606$$

$$= 24.9$$

Therefore, U

$$= 1/24.9$$

$$= 0.04 \text{ Btu/(hr)(ft}^2\text{)(°F)}$$

TABLE 10-2 STORAGE REQUIREMENTS AND PROPERTIES OF PERISHABLE PRODUCTS

Commodity	Storage temperature, °F	Relative humidity, %	Approximate storage life[a]	Water content, %	Highest freezing, °F	Specific heat above freezing,[b] Btu/lb · °F	Specific heat below freezing,[b] Btu/lb · °F	Latent heat,[c] Btu/lb
Vegetables								
Artichokes								
Globe	32	95 to 100	2 weeks	84	29.9	0.87	0.45	120
Jerusalem	31 to 32	90 to 95	4 to 5 months	80	28.0	0.83	0.44	114
Asparagus	32 to 35	95 to 100	2 to 3 weeks	93	30.9	0.94	0.48	133
Beans								
Snap or green	40 to 45	95	7 to 10 days	89	30.7	0.91	0.47	127
Lima	34 to 40	95	3 to 5 days	67	30.0	0.73	0.40	94
Dried	50	70	6 to 8 months	11		0.32	0.23	
Beets								
Roots	32	95 to 100	4 to 6 months	88	30.4	0.90	0.46	126
Bunch	32	98 to 100	10 to 14 days		31.3			
Broccoli	32	95 to 100	10 to 14 days	90	30.9	0.92	0.47	130
Brussels sprouts	32	95 to 100	3 to 5 weeks	85	30.6	0.88	0.46	122
Cabbage, late	32	98 to 100	5 to 6 months	92	30.4	0.94	0.47	132
Carrots								
Topped-immature	32	98 to 100	4 to 6 weeks	88	29.5	0.91	0.46	126
Topped-mature	32	98 to 100	7 to 9 months	88	29.5	0.91	0.46	126
Cauliflower	32	95 to 98	3 to 4 weeks	92	30.6	0.93	0.47	132
Celeriac	32	95 to 100	6 to 8 months	88	30.4	0.91	0.46	126
Celery	32	98 to 100	2 to 3 months	94	31.1	0.95	0.48	135
Collards	32	95 to 100	10 to 14 days	87	30.6	0.90	0.46	125
Corn, Sweet	32	95 to 98	4 to 8 days	74	30.9	0.79	0.42	106
Cucumbers	45 to 50	95	10 to 14 days	96	31.1	0.97	0.49	137
Eggplant	45 to 54	90 to 95	7 to 10 days	93	30.6	0.94	0.48	133
Endive (escarole)	32	95 to 100	2 to 3 weeks	93	31.9	0.94	0.48	133
Frozen vegetables[h]	−10 to 0		6 to 12 months					
Garlic, dry	32	65 to 70	6 to 7 months	61	30.6	0.69	0.40	89
Greens, leafy	32	95 to 100	10 to 14 days	93	31.5	0.94	0.48	133
Horseradish	30 to 32	95 to 100	10 to 12 months	75	28.7	0.78	0.42	104
Jicama	55 to 65	65 to 70	1 to 2 months					
Kale	32	95 to 100	2 to 3 weeks	87	31.1	0.89	0.46	125
Kohlrabi	32	98 to 100	2 to 3 months	90	30.2	0.92	0.47	129
Leeks, green	32	95 to 100	2 to 3 months	85	30.7	0.88	0.46	122
Lettuce, head	32 to 34	95 to 100	2 to 3 weeks	95	31.7	0.96	0.48	136
Mushrooms	32	95	3 to 4 days	91	30.4	0.93	0.47	130
Okra	45 to 55	90 to 95	7 to 10 days	90	28.7	0.92	0.46	129
Onions								
Green	32	95 to 100	3 to 4 weeks	89	30.4	0.91	0.47	127
Dry, and onion sets	32	65 to 75	1 to 8 months	88	30.6	0.90	0.46	126
Parsley	32	95 to 100	1 to 2 months	85	30.0	0.88	0.45	122
Parsnips	32	98 to 100	4 to 6 months	79	30.4	0.84	0.44	112
Peas								
Green	32	95 to 98	1 to 2 weeks	74	30.9	0.79	0.42	106
Dried	50	70	6 to 8 months	12		0.30	0.24	
Peppers								
Dried	32 to 50	60 to 70	6 months	12		0.30	0.24	17
Sweet	45 to 50	90 to 95	2 to 3 weeks	92	30.7	0.94	0.47	132
Potatoes								
Early	38 to 40	90 to 95	4 to 5 months	81	30.9	0.85	0.44	116
Main crop	38 to 40	90 to 95	5 to 8 months	78	30.9	0.82	0.43	111
Sweet	50 to 55	85 to 90	4 to 7 months	69	29.7	0.76	0.41	99
Pumpkins	50 to 55	50 to 75	2 to 3 months	91	30.6	0.92	0.47	130
Radishes								
Spring	32	95 to 100	3 to 4 weeks	95	30.7	0.95	0.48	134
Winter	32	95 to 100	2 to 4 months	95	30.7	0.95	0.48	134
Rhubarb	32	95 to 100	2 to 4 weeks	95	30.3	0.95	0.48	134
Rutabagas	32	98 to 100	4 to 6 months	89	30.0	0.91	0.47	127
Salsify	32	98 to 100	2 to 4 months	79	30.0	0.83	0.44	113
Seed, vegetable	32 to 50	50 to 65	10 to 12 months	7 to 15		0.29	0.23	16
Spinach	32	95 to 98	10 to 14 days	93	31.5	0.94	0.48	133
Squash								
Acorn	45 to 50	70 to 75	5 to 8 weeks		30.6			
Summer	41 to 50	95	5 to 14 days	94	31.1	0.95	0.48	135
Winter	50 to 55	50 to 75	4 to 6 months	85	30.6	0.88	0.45	122
Tamarillos	37 to 40	85 to 95	10 weeks					
Tomatoes								
Mature green	55 to 60	90 to 95	1 to 3 weeks	93	31.0	0.94	0.48	133
Firm, ripe	45 to 50	90 to 95	4 to 7 days	94	31.1	0.95	0.48	134
Turnips								
Roots	32	95	4 to 5 months	92	30.0	0.93	0.47	132
Greens	32	95 to 100	10 to 14 days	90	31.6	0.92	0.47	129
Watercress	32	95 to 100	2 to 3 weeks	93	31.4	0.94	0.48	133
Yams	61	85 to 90	3 to 6 months	74		0.79	0.42	105

TABLE 10-3 REFRIGERATION DESIGN AMBIENT TEMPERATURE GUIDE*

Location	Average ambient temp. °F	Maximum ambient temp. °F	Ground temperature °F
Alabama			
Birmingham	88	99	70
Mobile	88	97	75
Arizona			
Flagstaff	75	90	60
Phoenix	100	113	80
Tucson	84	98	80
Arkansas			
Fort Smith	91	103	70
Little Rock	90	100	70
California			
Bakersfield	96	114	75
Fresno	94	111	80
Los Angeles	83	94	75
Oakland	75	89	65
Sacramento	90	108	80
San Diego	75	80	65
San Francisco	75	83	65
Colorado			
Colorado Springs	83	94	60
Denver	83	98	60
Grand Junction	88	102	60
Pueblo	83	100	55
Connecticut			
Hartford	83	94	65
New Haven	83	95	65
New London	83	93	65
Norwalk	83	96	65
Delaware			
Dover	87	96	65
Milford	87	98	65
Wilmington	87	94	65
District of Columbia	89	98	65
Florida			
Jacksonville	88	96	80
Miami	88	90	80
Orlando	88	97	80
Tallahassee	88	100	80
Tampa	88	95	80
Georgia			
Atlanta	87	95	70
Savannah	89	99	75
Idaho			
Boise	89	105	60
Pocatello	83	100	60
Illinois			
Cairo	89	101	60
Chicago	87	98	60
Peoria	88	100	60
Quincy	90	103	60
Rockford	87	101	60
Springfield	90	102	60
Indiana			
Evansville	90	100	65
Fort Wayne	87	100	60

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TABLE 10-4 ALLOWANCE FOR SOLAR RADIATION

(Degrees Fahrenheit to be added to the normal temperature difference for heat leakage calculations to compensate for sun effect—not to be used for air-conditioning design)

Type of surface	East wall	South wall	West wall	Flat roof
Dark-colored surfaces such as:				
Slate roofing				
Tar roofing	8	5	8	20
Black paints				
Medium-colored surfaces, such as:				
Unpainted wood				
Brick				
Red tile	6	4	6	15
Dark cement				
Red, gray, or green paint				
Light-colored surfaces, such as:				
White stone				
Light-colored cement	4	2	4	9
White paint				

From *ASRE Data Book*, Design Volume, 1957–58 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

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TABLE 10-5 WALL HEAT GAIN

Insulation	(Btu per square foot per 24 hr)																
Cork or equivalent, in.	Temp. difference (ambient temp. minus refrigerator temp.), °F																
	1	40	45	50	55	60	65	70	75	80	85	90	95	100	105	110	115	120
3	2.4	96	108	120	132	144	156	168	180	192	204	216	228	240	252	264	267	288
4	1.8	72	81	90	99	108	117	126	135	144	153	162	171	180	189	198	207	216
5	1.44	58	65	72	79	87	94	101	108	115	122	130	137	144	151	159	166	173
6	1.2	48	54	60	66	72	78	84	90	96	102	108	114	120	126	132	138	144
7	1.03	41	46	52	57	62	67	72	77	82	88	93	98	103	108	113	118	124
8	0.90	36	41	45	50	54	59	63	68	72	77	81	86	90	95	99	104	108
9	0.80	32	36	40	44	48	52	56	60	64	68	72	76	80	84	88	92	96
10	0.72	29	32	36	40	43	47	50	54	58	61	65	68	72	76	79	83	86
11	0.66	26	30	33	36	40	43	46	50	53	56	60	63	66	69	73	76	79
12	0.60	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
13	0.55	22	25	28	30	33	36	39	41	44	47	50	52	55	58	61	63	66
14	0.51	20	23	26	28	31	33	36	38	41	43	46	49	51	54	56	59	61
Single glass	27.0	1080	1220	1350	1490	1620	1760	1890	2030	2160	2290	2440	2560	2700	2840	2970	3100	3240
Double glass	11.0	440	500	550	610	660	715	770	825	880	936	990	1050	1100	1160	1210	1270	1320
Triple glass	7.0	280	320	350	390	420	454	490	525	560	595	630	665	700	740	770	810	840

Note: Where wood studs are used multiply the above values by 1.1.

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Example 10-4 The east wall of a cold storage warehouse in Dallas, Texas, is 100 ft × 20 ft. The outside surface of the wall is dark concrete, and the wall U factor is 0.046 Btu/(hr)(ft^2)(°F). If the inside temperature is maintained at 38°F, compute the wall gain in Btu/24 hr.

Solution From Table 10-3, the outdoor design temperature for Dallas is 92°F, and from Table 10-4, the temperature difference for a medium-colored outside wall facing east is 6°F. Applying Equation 10-3,

$$\text{Wall gain} = (100 \times 20)(0.046)(92° + 6° - 38°)(24)$$

$$= 132{,}480 \text{ Btu/24 hr}$$

$$\text{Wall gain load} = \text{Outside surface area} \times \text{wall gain factor}$$

Example 10-5 The floor of a cold storage warehouse in Green Bay, Wisconsin, measures 100 ft × 150 ft and is laid directly on the ground. Assuming that the inside air temperature is 35°F, if the *U* factor for the floor is 0.045 Btu/(hr)(ft²)(°F), what is the heat gain through the floor in Btu/24 hr?

Solution From Table 10-3, the ground temperature for Green Bay is 55°F. Applying Equation 10-3,

$$Q = (100 \times 150)(0.045)(55^\circ - 35^\circ)(24)$$

$$= 324{,}000 \text{ Btu/24 hr}$$

TABLE 10-6A AVERAGE AIR CHANGES PER 24 HOURS FOR STORAGE ROOMS ABOVE 32°F DUE TO DOOR OPENING AND INFILTRATION (Does not apply to rooms using ventilating ducts or grilles.)

Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr
250	38.0	1,000	17.5	6,000	6.5	30,000	2.7
300	34.5	1,500	14.0	8,000	5.5	40,000	2.3
400	29.5	2,000	12.0	10,000	4.9	50,000	2.0
500	26.0	3,000	9.5	15,000	3.9	75,000	1.6
600	23.0	4,000	8.2	20,000	3.5	100,000	1.4
800	20.0	5,000	7.2	25,000	3.0		

NOTE: For storage room with anterooms, reduce air changes to 50% of values in table.
For heavy duty usage, add 50% to values given in table.

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TABLE 10-6B AVERAGE AIR CHANGES PER 24 HOURS FOR STORAGE ROOMS BELOW 32°F DUE TO DOOR OPENING AND INFILTRATION (Does not apply to rooms using ventilating ducts or grilles.)

Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr	Volume cu ft	Air changes per 24 hr
250	29.0	1,000	13.5	5,000	5.6	25,000	2.3
300	26.2	1,500	11.0	6,000	5.0	30,000	2.1
400	22.5	2,000	9.3	8,000	4.3	40,000	1.8
500	20.0	2,500	8.1	10,000	3.8	50,000	1.6
600	18.0	3,000	7.4	15,000	3.0	75,000	1.3
800	15.3	4,000	6.3	20,000	2.6	100,000	1.1

NOTE: (1) For storage rooms with anterooms, reduce air changes to 50% of values in table.
For heavy duty usage, add 50% to values given in table.
(2) For locker plant rooms, double the above table values.

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Calculating the Air Change Load

$$\text{Air change load} = (\text{Inside volume})(\text{air changes})(0.075)(h_o - h_i)$$

Example 10-6 A storage cooler located in a hotel kitchen has inside dimensions of 16 ft × 25 ft × 10 ft. From Table 10-6A, determine the number of air changes per 24 hours.

Solution The inside volume of the cooler is 4000 ft^2 (16 × 25 × 10) and, being located in a hotel kitchen, usage is heavy. From Table 10-6A, the number of air changes per 24 hr is 12.3 (8.2 + 4.1).

Example 10-7 The inside volume of a storage cooler in a busy supermarket is 6000 ft^3 and usage is heavy. The inside is maintained at 35°F and 90% RH and the outside conditions are 75°F and 50% RH. Calculate the air change load.

Solution From Table 10-6A, the number of air changes is 9.75 (6.5 + 3.25). From the psychrometric chart, the enthalpy of the inside air is 12.4 Btu/lb and the enthalpy of the outside air is 28.3 Btu/lb. Applying Equation 10-7,

$$\text{Air change load} = (6000)(9.75)(0.075)(28.3 - 12.4)$$
$$= 69{,}700 \text{ Btu/24 hr}$$

Calculating the Product Load

- $Q = (m)(c)(\Delta T)$
- $Q = \dfrac{(m)(c)(\text{TD})(24\text{ hr})}{\text{desired cooling time (hr)}}$

where
Q = the quantity of heat in Btu
m = mass of the product (pounds)
c = the specific heat above freezing, Btu/(lb) (°F)
ΔT = the change in the product temperature (°F)

Example 10-8 Seventy-five hundred pounds of fresh lean beef enter a chilling cooler at 102°F and are chilled to 45°F each day. Compute the product load in Btu per 24 hr.

Solution From Table 10-2, the specific heat of beef above freezing is 0.75 Btu/lb °F. Applying Equation 10-8,

$$\text{Product load} = (7500)(0.75)(102^\circ - 45^\circ)$$

$$= 320{,}600 \text{ Btu}/24 \text{ hr}$$

Example 10-9 Determine the product load in Btu per 24 hr assuming that the beef described in Example 10-8 is chilled in 20 hr rather than over the entire 24-hr period.

Solution Applying Equation 10-9,

$$\text{Product load} = \frac{(7500)(0.75)(102^\circ - 45^\circ)(24\text{ hr})}{20\text{ hr}}$$

$$= 384{,}750\text{ Btu}/24\text{ hr}$$

Product Freezing and Storage

When a product is to be frozen and stored at some temperature below its freezing temperature, the product load is calculated in three parts:

1. The heat given off by the product in cooling from the entering temperature to its freezing temperature
2. The heat given off by the product in solidifying or freezing
3. The heat given off by the product in cooling from its freezing temperature to the final storage temperature.

$$Q = (m)(h_{if})$$

where m = the mass of the product in pounds
h_{if} = the product latent heat in Btu per pound

Example 10-10 Five hundred pounds of poultry enter a chiller at 40°F and are frozen and chilled to a final temperature of −5°F for storage in 12 hr. Compute the product load in Btu per 24 hr.

Solution From Table 10-2,

Specific heat above freezing = 0.80 Btu/lb °F

Specific heat below freezing = 0.42 Btu/lb °F

Latent heat = 106 Btu/lb

Freezing temperature = 27°F

To cool poultry from entering temperature to freezing temperature, applying Equation 10-8 $= (500)(0.80)(40° - 27°)$ $= 5200$ Btu

To freeze, applying Equation 10-10 $= (500)(106)$ $= 53{,}000$ Btu

To cool from freezing temperature to final storage temperature, applying Equation 10-8 $= 500 \times 0.42 \times [27° - (-5°)]$ $= 6720$ Btu

Total heat given up by product (summation of 1, 2, and 3) $= 64{,}920$ Btu

Equivalent product load for 24-hr period Btu/24 hr

$$= \frac{64{,}920 \times 24 \text{ hr}}{12 \text{ hr}}$$

$$= 129{,}840 \text{ Btu/24 hr}$$

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Respiration Heat

$$
\begin{aligned}
Q\,(\text{Btu}/24\text{ hr}) = {} & \text{mass of product (lb)} \\
& \times \text{respiration heat (Btu/lb hr)} \\
& \times 24\text{ hr}
\end{aligned}
$$

Containers and Packing Materials

TABLE 10-7 RESPIRATION HEAT FROM FRUITS AND VEGETABLES

Fruits

Commodity	Temperature deg F	Btu per hr per lb
Apples	32	.018
	40	.030
	60	.120
Apricots	32	.023
	40	.036
	60	.170
Bananas		
Holding	54	.069
Ripening	68	.190
Chilling	70–56	.500§
Berries	36	.115
	60	.345
Cherries	32	.032
	60	.250
Cranberries	32	.014
	40	.019
	50	.036
Dates, fresh	32	.014
	40	.019
	50	.036
Grapefruit	32	.0096
	40	.022
	60	.058
Grapes	32	.0075
	40	.014
	60	.050
Lemons	32	.012
	40	.017
	60	.062
Limes	32	.012
	40	.017
	60	.062
Oranges	32	.017
	40	.029
	60	.104
Peaches	32	.023
	40	.036
	60	.170
Pears	32	.016
	60	.230
Plums	32	.032
	60	.250
Quinces	32	.018
	40	.030
	60	.120
Strawberries	32	.068
	40	.120
	60	.360

Vegetables

Commodity	Temperature deg F	Btu per hr per lb
Asparagus	32	.035
	40	.170
Beans, lima	32	.170
	60	.820
Beans, string	32	.099
	40	.140
	60	.470
Beets	32	.055
	40	.085
	60	.150
Brussel sprouts	32	.059
	40	.095
	60	.280
Cabbage	32	.059
	40	.095
	60	.280
Cauliflower	32	.059
	40	.095
	60	.280
Carrots	32	.045
	40	.073
	60	.170
Celery	32	.059
	40	.095
	60	.280
Corn, sweet	32	.035
	40	.170
Cucumber	32	.028
	40	.041
	60	.175
Endive	40	.200
Lettuce	32	.240
	40	.330
	60	.960
Melons (except watermelons)	32	.028
	40	.041
	60	.175
Mushrooms	32	.130
	50	.460
Onions	32	.018
	50	.039
	70	.075
Parsnips	32	.045
	40	.073
	60	.170
Peas	32	.170
	60	.820
Peppers	32	.057
	60	.180
Potatoes	32	.014
	40	.030
	70	.060
Spinach	40	.200
Sweet Potatoes	40	.070
Tomatoes (Green)	60	.130
(Ripe)	40	.027
Turnips	32	.040
	40	.050

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Example 10-11 Three thousand lug boxes of apples are stored at 35°F in a storage cooler. The apples enter the cooler at a temperature of 75°F and at a rate of 200 lug boxes per day for the 15-day harvest period. The average weight of apples per lug box is 59 pounds. The lug boxes have an average weight of 4.5 pounds and a specific heat of 0.6 Btu/(lb)(°F). Calculate the product load in Btu/24 hr.

Solution The product load is calculated for the fifteenth day when the product load is greatest. From Table 10-2, the specific heat of apples is 0.87 Btu/(lb)(°F) and, by interpolation from Table 10-7, the respiration heat is 0.025 Btu/(lb)(hr).

Chilling load $= (m)(c)(t_e - t_s)$

Apples $= (200 \times 59)(0.87)(75° - 35°)$

$= 410{,}600$ Btu/24 hr

Lug boxes $= (200 \times 4.5)(0.6)(75° - 35°)$

$= 21{,}600$ Btu/24 hr

Respiration load $= (m)$(respiration heat)(24 hr)

$= (3000 \times 59)(0.025)(24)$

$= 106{,}200$ Btu/24 hr

Total product load $= 538{,}400$ Btu/24 hr

Calculating the Miscellaneous Load

Lights: wattage X 3.42 Btu/watt hr X 24 hr

Electric motors: factor (Table 10-8) X horsepower X number of hours

People: factor (Table 10-9) X number of people X number of hours

TABLE 10-8 HEAT EQUIVALENT OF ELECTRIC MOTORS

Motor hp	Btu/hp-hr: Connected load in refr. space[1]	Motor losses outside refr. space[2]	Connected load outside refr. space[3]
$\frac{1}{8}$ to $\frac{1}{2}$	4250	2545	1700
$\frac{1}{2}$ to 3	3700	2545	1150
3 to 20	2950	2545	400

TABLE 10-9 HEAT EQUIVALENT OF OCCUPANCY

Cooler temperature, °F	Heat equivalent/person Btu/hr
50	720
40	840
30	950
20	1050
10	1200
0	1300
−10	1400

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-8 HEAT EQUIVALENT OF ELECTRIC MOTORS

Motor hp	Btu/hp-hr: Connected load in refr. space[1]	Btu/hp-hr: Motor losses outside refr. space[2]	Btu/hp-hr: Connected load outside refr. space[3]
$\frac{1}{8}$ to $\frac{1}{2}$	4250	2545	1700
$\frac{1}{2}$ to 3	3700	2545	1150
3 to 20	2950	2545	400

[1] For use when both useful output and motor losses are dissipated within refrigerated space; motors driving fans for forced circulation unit coolers.

[2] For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; pump on a circulating brine or chilled water system, fan motor outside refrigerated space driving fan circulating air within refrigerated space.

[3] For use when motor heat losses are dissipated within refrigerated space and useful work expended outside of refrigerated space; motor in refrigerated space driving pump or fan located outside of space.

From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

TABLE 10-9 HEAT EQUIVALENT OF OCCUPANCY

Cooler temperature, °F	Heat equivalent/person Btu/hr
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From *ASRE Data Book*, Design Volume, 1949 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

Use of Safety Factor

The total cooling load for a 24-hr period is the summation of the heat gains as calculated in the foregoing sections. It is common practice to add 5% to 10% to this value as a safety factor. The percentage used depends on the reliability of the information used in calculating the cooling load. As a general rule 10% is used.

Example 10-12 A storage cooler 18 ft × 10 ft × 10 ft and used for the short-term storage of produce is constructed of panels insulated with 4 in. of polystyrene molded beads (equivalent to 5 in. of corkboard). The outside conditions are 75°F and 50% RH, and the inside is to be maintained at 40°F and 95% RH. One thousand pounds of mixed vegetables are cooled 10°F to the storage temperature each day. A 100-watt light burns in the space 8 hours per day, and one person works in the space approximately 4 hours per day. If the usage is average, calculate the cooling load in Btu/hr.

Solution From Table 10-1, the *k* factor for polystyrene molded bead insulation is 0.25 (Btu)(in.)/(hr)(ft²)(°F) and, from Table 10-6A, the number of air changes per 24 hours is 14. From the psychrometric chart the enthalpy of the outdoor air is 28.3 Btu/lb and the enthalpy of the inside air is 14.9 Btu/lb. From Table 10-2, an average specific heat for mixed vegetables is 0.9 Btu/(lb)(°F) and from Table 10-7, an average respiration heat at 40°F is 0.09 Btu/(lb)(hr). The total outside surface area is 920 ft^2 and, since the inside dimensions are 8 in. less than the outside dimensions, the inside volume is approximately 1500 ft^3 (17.33 × 9.33 × 9.33).

$$\text{Wall gain load} = (A)(U)(t_o - t_i)(24\text{ hr})$$

$$= (920)(0.25/4)(75° - 40°)(24)$$

$$= 48{,}300\text{ Btu}/24\text{ hr}$$

Air change load = (inside volume) (air changes/hr) (0.075) ($h_o - h_i$)

= (1500) (14) (0.075) (28.3 − 14.9)

= 21,100 Btu/24 hr

Product load

Chilling = (m) (c) ($t_e - t_s$)

= (1000) (0.9) (10°F)

= 9000 Btu/24 hr

Respiration = (m) (respiration heat) (24 hr)

= (1000) (0.09) (24)

= 2160 Btu/24 hr

Miscellaneous load

People = (number) (heat equivalent) (hr)

= (1) (800) (4)

= 3200 Btu/24 hr

Lights = (watts) (3.4 Btu/(W) (hr) × (hr)

= (100) (3.4) (8)

= 2720 Btu/24 hr

Summation = 86,480 Btu/24 hr

Safety factor = 8650 (10%)

Total load = 95,130 Btu/24 hr

Required equipment capacity

$$= \frac{\text{Total cooling load}}{\text{Desired running time}}$$

$$= \frac{95{,}130 \text{ Btu/24 hr}}{16 \text{ hr}}$$

= 5950 Btu/hr

Short Method Load Calculations

$$\text{Usage load} = \text{Interior volume} \times \text{usage factor}$$

Example 10-13 Recalculate the cooling load for the cooler described in Example 10-12 using the short form.

Solution From Example 10-12, the outside surface area is 920 ft^2 and the inside volume is 1500 ft^3. From Table 10-5, the wall gain factor is 1.44 Btu/(ft^2)(°F)(24 hr) and from Table 10-10 the usage factor is 0.92 Btu/(°F)(24 hr)

$$\text{Wall gain load} = (\text{Area})(\text{Wall gain factor})$$
$$= (920)(1.44 \times 35°)$$
$$= 46{,}400 \text{ Btu/24 hr}$$

$$\text{Usage load} = (\text{Inside volume})(\text{Usage factor})$$
$$= (1500)(0.92 \times 35°)$$
$$= 48{,}300 \text{ Btu/24 hr}$$

$$\text{Total cooling load} = 94{,}700 \text{ Btu/24 hr}$$

$$\text{Required equipment capacity} = \frac{94{,}700 \text{ Btu/24 hr}}{16 \text{ hr}}$$
$$= 5920 \text{ Btu/hr}$$

TABLE 10-10 USAGE HEAT GAIN

Volume cu ft	Service*	Temperature difference (ambient temp minus storage room temp), F deg: 1	40	50	55	60	65	70	75	80	90	100
20	Average	4.68	187	234	258	281	305	328	351	374	421	468
	Heavy	5.51	220	276	303	331	358	386	413	441	496	551
30	Average	3.30	132	165	182	198	215	231	248	264	297	330
	Heavy	4.56	182	228	251	274	297	319	342	365	410	456
50	Average	2.28	91	114	126	137	148	160	171	182	205	228
	Heavy	3.55	142	177	196	213	231	249	267	284	320	355
75	Average	1.85	74	93	102	111	120	130	139	148	167	185
	Heavy	2.88	115	144	158	173	188	202	216	230	259	288
100	Average	1.61	64	81	84	97	105	113	121	129	145	161
	Heavy	2.52	101	126	139	151	164	176	189	202	227	252
200	Average	1.38	55	69	76	83	90	97	103	110	124	138
	Heavy	2.22	90	111	122	133	144	155	166	178	200	222
300	Average	1.30	52.0	65	71.5	78	84.5	91	97.5	104	117	130
	Heavy	2.08	83.2	104	114	125	135	146	156	166	187	208
400	Average	1.24	49.6	62	68.2	74.4	80.6	86.8	93	99.2	112	124
	Heavy	1.96	78.4	98	108	118	128	137	147	157	176	196
500	Average	1.21	48.4	60.5	66.6	72.6	78.7	84.7	90.7	96.8	109	121
	Heavy	1.87	74.8	93.5	103	112	122	131	140	150	168	187
600	Average	1.17	46.8	58.5	64	70	76	82	88	94	105	117
	Heavy	1.85	74.0	92.5	102	111	120	130	139	148	167	185
800	Average	1.11	44.4	55.5	61.1	66.6	72.2	77.7	83.3	88.8	100	111
	Heavy	1.76	70.4	88.0	96.8	106	115	123	132	141	158	176
1,000	Average	1.10	44.0	55.0	60.5	66	71.5	77	82.5	88	99	110
	Heavy	1.67	66.8	83.5	91.9	100	108	117	125	134	150	167
1,200	Average	.995	39.8	49.8	54.7	59.7	64.7	69.7	74.7	79.6	89.6	99.5
	Heavy	1.58	63.2	79.0	86.9	94.8	103	111	119	126	142	158
1,500	Average	.920	36.8	46.0	50.6	55.2	59.8	64.4	69	73.6	82.8	92
	Heavy	1.50	60.0	75.0	82.5	90.0	97.5	105	113	120	135	150
2,000	Average	.835	33.4	41.8	45.9	50.1	54.3	58.5	62.7	66.8	75.2	83.5
	Long storage	.775	31.0	38.8	42.6	46.5	50.4	54.3	58.1	62	69.8	77.5
3,000	Average	.750	30.0	37.5	41.3	45.0	48.8	52.5	56.2	60.0	67.5	75.0
	Long storage	.576	23.0	28.8	31.7	34.6	37.3	40.3	43.2	46.1	51.8	57.6
5,000	Long storage	.403	16.1	20.2	22.2	24.2	26.2	28.2	30.2	32.2	36.3	40.3
7,500	Long storage	.305	12.2	15.3	16.8	18.3	19.8	21.4	22.9	24.4	27.5	30.5
10,000	Long storage	.240	9.6	12.0	13.2	14.4	15.6	16.8	18.0	19.2	21.6	24.0
20,000	Long storage	.187	7.48	9.35	10.3	11.2	12.2	13.1	14.0	15.0	16.8	18.7
50,000	Long storage	.178	7.12	8.90	9.79	10.7	11.6	12.5	13.4	14.2	16.0	17.8
75,000	Long storage	.176	7.04	8.80	9.68	10.6	11.5	12.3	13.2	14.1	15.8	17.6
100,000	Long storage	.173	6.92	8.65	9.52	10.4	11.2	12.1	13.0	13.8	15.6	17.3

* For average and heavy service, product load is based on product entering at 10 deg above the refrigerator temperature; for long storage the entering temperature is approximately equal to the refrigerator temperature.

Where the product load is unusual, do not use this table.

From *ASHRAE Data Book*, Fundamentals Volume, 1972 Edition, by permission of the American Society of Heating, Refrigerating, and Air-Conditioning Engineers.

Example 10-14 A storage cooler 10 ft × 12 ft × 9 ft equipped with four 2 ft × 5 ft triple-pane glass doors is used for general-purpose storage. The walls are panels insulated with the equivalent of 5 in. of corkboard and the inside volume is approximately 930 ft^3. The cooler is maintained at 35°F, and the service load is heavy. If the ambient temperature is 80°F, determine the cooling load in Btu/hr based on a 16-hr operating time.

Solution The glass area is 40 ft^2 (4 × 2 × 5) and the net wall area is 596 ft^2 (636 − 40). From Table 10-5, the wall gain factor is 65 Btu/(ft^2)(24 hr) and the factor for triple-pane glass is 320 Btu/(ft^2)(24 hr). From Table 10-10, by interpolation, the usage factor is approximately 1.72 Btu/(ft^3)(°F)(24 hr).

$$
\begin{aligned}
\text{Wall gain load} &= (\text{Wall area})(\text{wall gain factor}) \\
&= (596)(65) \\
&= 38{,}750 \text{ Btu/24 hr} \\
&= (\text{Glass area})(\text{glass factor}) \\
&= (40)(320) \\
&= 12{,}800 \text{ Btu/24 hr}
\end{aligned}
$$

$$
\begin{aligned}
\text{Usage load} &= (\text{Inside volume})(\text{usage factor}) \\
&= (930)(1.72 \times 45°\text{F}) \\
&= 71{,}980 \text{ Btu/24 hr}
\end{aligned}
$$

$$\text{Total cooling load} = 123{,}530 \text{ Btu/24 hr}$$

$$
\begin{aligned}
\text{Average hourly load} &= \frac{123{,}530 \text{ Btu/24 hr}}{16 \text{ hr}} \\
&= 7720 \text{ Btu/hr}
\end{aligned}
$$