2.4 Variations on P-h Chart in accordance with changes in operating conditions

In order to track down the operating conditions of air conditioners and chillers, conjuring the P-h Chart enables accurate understanding of a variety of their symptoms.

2.4.1 Factors influencing on equipment

1. Insufficient refrigerating effect

Less refrigerating effect causes the reduction in the amount of heat absorbed per 1-kg mass refrigerant flowing in the evaporator and the degradation of the refrigerating capacity. Furthermore, it decreases the coefficient of performance, thus resulting in operation of decreased efficiency.

2. Excessive specific volume of suction gas

Excessive specific volume of suction gas causes the reductions in the specific weight of the suction gas, the weight of refrigerant circulated discharging from the compressor, and the refrigerating capacity. The reduced weight of refrigerant circulated results in less running current.

3. Excessive compression ratio

Excessive compression ratio causes a significant difference in pressure ratio between the suction gas pressure and the discharge gas pressure, which increases the volumetric expansion of discharge gas remaining in the cylinder top clearance, thus resulting in the reductions in the suction gas amount and the refrigerating capacity.

The thermal equivalent of compressor work increases and the coefficient of performance decreases. If there are no changes in the specific volume of the suction gas, the running current increases in proportion to the increase in the thermal equivalent of compressor work.

4. Too high discharge gas temperature

Refrigerating oil gets mixed with refrigerant gas and circulates. If the discharge gas temperature is too high, the chiller oil temperature becomes high to develop oil deterioration (carbonization), thus causing clogged dryer or faulty startup of the compressor.

5. Superheated degree other than 5℃

Too high superheated degree abnormally increases the temperature of the motor coil in the (semi-) hermetic system compressor, resulting in actuated compressor protective thermostat (C.T.P.) and shortened motor life span, and furthermore increased discharge gas temperature.

When the superheated degree reaches 0˚C, that is, the system turns into wet compression, uneven temperature is caused in the motor coil, resulting in burnt motor. In addition, if the liquid refrigerant melts in the lubricating oil, diluted oil or oil-forming symptom occurs, thus resulting in a drop in hydraulic pressure. Furthermore, in the extreme case, the liquid compression (liquid hammering) occurs to cause a broken valve.

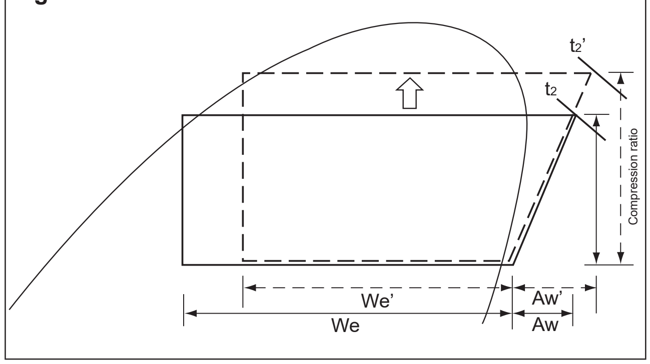
6. Insufficient sub-cooled degree

If there is a large pressure loss in the liquid pipe between the condenser and the expansion valve or the cooling load increases, flushing gas is generated and moist vapor enters the expansion valve, thus resulting in increased dryness factor at the expansion valve outlet to degrade the refrigerating effect.

2.4.2 Changes on P-h Chart and problems while in malfunctions

Take the basic cycle under the operation conditions by model described in Chapter 3 as standard operation. If the operating conditions vary with the conditions of indoor and outdoor air, external contamination, or gas leaks, the operation differs from the standard operation, thus causing problems described in Section 4-1 Factors influencing on equipment.
The following section shows the changes on the P-h Chart due to the changes in the basic conditions.
Actually, the system operates in a cycle with different conditions conspired.

Fig.2-23



1. Abnormal rise of high pressure

Possible causes (Example)

Water-cooled type:	Insufficient cooling water Dirty condenser Poor heat exchange in cooling tower
Air-cooled type:	Dirty heat exchanger Short-circuit of hot air
Common:	Over change of refrigerant Air mixed in refrigerant system

Symptoms

The low pressure slightly rises as the high pressure rises. In the case of units using a capillary tube, the low pressure sometimes rises noticeably. In this case, the superheated degree decreases and the discharge gas temperature becomes significantly high.
The sub-cooled degree rises only if the refrigerant is overcharged, while in other cases it shows little change or, if anything, a downward tendency.

Problems

When the high pressure reaches the set point of the pressure switch, the machine stops running, or it does not stop running but the following problems may be caused.

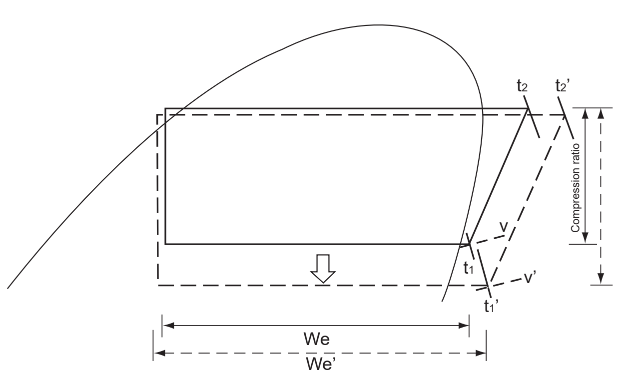
1. The discharge gas temperature becomes too high, thus exerting an adverse influence upon equipment.
2. The refrigeration effect decreases while the compression ratio increases, thus causing a reduction in the refrigerating capacity.
3. The coefficient of performance drops to degrade the operating efficiency.
4. The thermal equivalent of compression work increases to cause increases of running current, that is, power consumption.

2. Abnormal drop of low pressure while in superheated compression

Possible causes (Example)

Insufficient refrigerant (gas leak)
Clogged dryer
Clogged filter
Clogged expansion valve or capillary tube
Faulty operation of expansion valve

Fig.2-24



Symptoms

In this case, the weight of refrigerant circulated has decreased. Therefore, with the reduction of the amount of evaporated heat, the amount of condensed heat reduces, thus slightly decreasing the condensing temperature (pressure).

The suction gas increases in the temperature and specific volume.

The discharge gas temperature becomes significantly high as well.

The subcooling degree decreases in the case of insufficient refrigerant, while it increases in the case of other causes due to clogging.

Problems

The suction gas temperature has become too high. Safety devices such as the compressor protective thermostat may be actuated or the machine may stop running due to the actuation of the low pressure switch.

Or the machine does not stop running but the following problems may be caused.

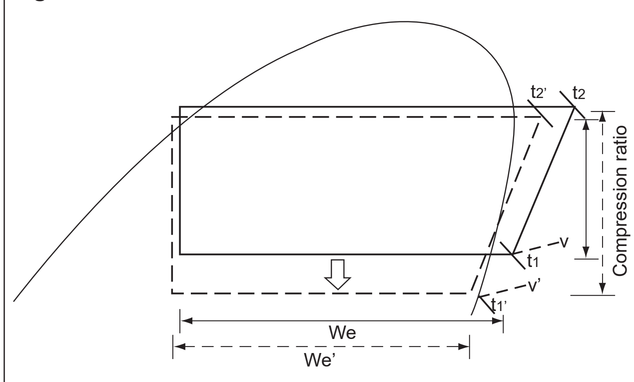
1. The discharge gas temperature becomes too high, thus exerting an adverse influence upon equipment.
2. Even though the refrigerating effect increases, the compression ratio as well as the specific volume of the suction gas becomes larger, thus resulting in decreased weight of refrigerant circulated and in a substantial reduction in refrigerating capacity.
3. Regardless of little change in the thermal equivalent of compression work, since the specific volume of the suction gas is large, the running current decreases.

3. Abnormal drop of low pressure while in wet compression

Possible causes (Example)

Insufficient air quantity
Insufficient cooling water
Dirty evaporator
Inadequate cooling load

Fig.2-25



Symptoms

In this case, the heat to the evaporator has decreased. Therefore, with the decrease of the evaporating temperature (pressure), the condensing temperature (pressure) shows a slight decrease.

The suction gas shows a hunting phenomenon between the moist vapor and the superheated vapor on units using an expansion valve while it turns into moist vapor on units using a capillary tube. In any of these cases, the specific volume becomes larger.

The discharge vapor temperature decreases.

Problems

Since the compressor sucks in moist vapor, the hydraulic pressure is not built up. Therefore, the pressure switch for hydraulic pressure protection or the low-pressure pressure switch may be actuated.

If the moist vapor is sucked in, the following problems may be caused.

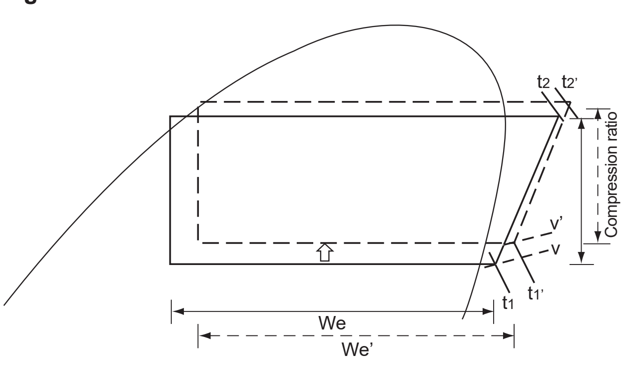
1. The hydraulic pressure is not built up, developing wear of bearings or metal parts in the compressor.
2. The refrigerating effect is small, while the compression ratio of the suction gas as well as the specific volume of the suction gas is large. Therefore, the refrigerating capacity shows a substantial reduction.
3. Regardless of a slight increase in the thermal equivalent of compressor work, since the specific volume of the suction gas is large, the running current decreases.

4. Abnormal rise of low pressure while in superheated compression

Possible causes (Example)

Increased cooling load
Wrong selection of unit (too small)

Fig.2-26



Symptoms

The high pressure slightly rises as the low pressure rises.
The suction gas increases in the temperature while decreases in the specific volume.
The discharge gas temperature becomes high.
The sub-cooled degree becomes low.

Problems

The suction gas temperature has become too high. Safety devices such as the compressor protective thermostat may be actuated to stop the machine running.

Or the machine does not stop running but the following problems may be caused.

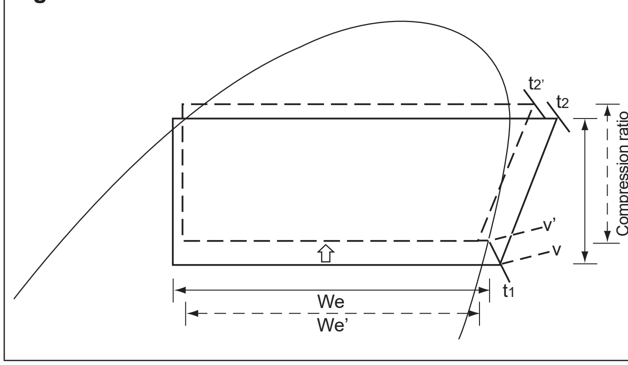
1. The discharge gas temperature becomes too high, thus exerting an adverse influence upon equipment.
2. Since the specific volume of the suction gas decreases, the weight of refrigerant circulated and the refrigerating capacity increases. Furthermore, the running current increases as well, thus resulting in increased power consumption.

5. Abnormal rise of low pressure while in wet compression

Possible causes (Example)

Faulty function of expansion valve
(Faulty installation of feeler bulb)
Overcharged refrigerant
(In the case of units using a capillary tube)

Fig.2-27



Symptoms

Units using an expansion valve show a downward tendency of sub-cooled degree. By contrast, units using a capillary tube show an upward tendency of sub-cooled degree. In either of these two cases, the high pressure rises.

The suction gas decreases in the specific volume while the temperature remains roughly the same as that in the standard operation.

The discharge gas increases in the high pressure, while it shows a slight decrease in the temperature.

Problems

Since the compressor sucks in moist vapor, the oil pressure is not built up. Therefore, the pressure switch for oil pressure protection may be actuated.

If the high-pressure moist vapor is sucked in, the following problems may be caused.

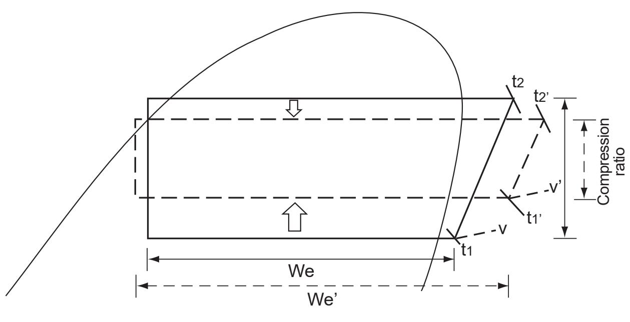
1. The oil pressure is not built up, developing wear of bearings or metal parts in the compressor.
2. Since the specific volume of the suction gas decreases, the refrigerating capacity increases, thus increasing the running current and the power consumption.

6. Abnormal rise of low pressure and drop of high pressure

Possible causes (Example)

Faulty compression
Faulty four-way valve
Faulty check valve

Fig.2-28



Symptoms

The suction gas substantially increases in the temperature but decreases in the specific volume.
The discharge gas substantially increases in the temperature as well.
Since the compression ratio is small, the running current decreases.
If the four-way valve is faulty, both of suction gas temperatures and discharge gas temperature do not rise.

Problems

The suction gas temperature has become too high. Safety devices such as the compressor protective thermostat may be actuated to stop the machine running.
Or the machine does not stop running but the following problems may be caused.
1. Since the compressor piston is in a freewheeling condition, the refrigerating capacity decreases substantially.
2. Even with pump-down operation, the low pressure does not fall to 0MPa G or less.
3. Even if the cooling water supply is interrupted, it may take time for the high pressure to rise and high-pressure cut may not be performed.

2.5 Refrigeration capacity calculation

By drawing refrigeration cycle on the P-h Chart, refrigerating effect, that is, the amount of heat per 1-kg mass of refrigerant during the evaporating process, and the state of suction gas to the compressor can be found.
However, the above data is not enough to reveal the refrigerating (cooling) capacity. In order to find the refrigerating capacity, it is required to calculate the capacity using numerical values on the P-h Chart and a variety of the compressor parameters.
The following section describes reciprocating compressors.

2.5.1 Compressor Parameters

1. Piston displacement V [m³/h]

The piston displacement is the total volume swept through the piston suction and compression strokes per unit of time.
The piston displacement of a reciprocating compressor is calculated by the following expression according to the compressor specifications.

Table 2-3

Examples of compressor specifications

Type	2T55HF	3T55RF
Nos. of cylinders	2	3
Cylinder diameter	55mm	55mm
Cylinder stroke	20.2mm	25.4mm
Revolutions speed (50/60)	2900/3450rpm	2900/3450rpm

$$V = \left[\frac{\pi}{4}\right] \cdot D^2 \cdot L \cdot Z \cdot n \times 60 \ [m^3/h]$$

Where

Va= Piston displacement [m³/h]

$\left[\frac{\pi}{4}\right]$ = Constant

D= Cylinder diameter (m)

L= Piston stroke (m)

Z= Number of cylinders

n = Revolutions per minute (rpm)

Cylinder volume[cm] $\left[\frac{\pi}{4}\right] \cdot D^2 \cdot L \times 10^6$

Exercise

Find the piston displacement for the 2T55HF compressor operating at 60 Hz.

D = 0.055 m L = 0.0202 m Z = 2 n = 3450

$$V = \left[\frac{\pi}{4}\right] \times \overset{D}{0.055} \times \overset{D}{0.055} \times \overset{L}{0.0202} \times \overset{Z}{2} \times \overset{n}{3450} \times 60$$

$\fallingdotseq$ 19.9 m³/h

How to find the piston displacement from the Japanese legal refrigeration ton (i.e., nominal refrigerating capacity)

The legal refrigeration ton in technical information represents refrigerating capacity during operation in a standard refrigeration cycle, which is referred to as nominal refrigerating capacity. (Refer to information in Section 3-1.)
In the Refrigeration Safety Regulation in the High Pressure Gas Control Law (Japan), the nominal refrigerating capacity is represented by refrigeration ton. Furthermore, assuming that the refrigerating capacity R of compressor having a piston replacement of V[m³/h] is computed by using the formula R = V/C, the value of the constant C is defined by the type of refrigerant.
Thus, when the legal refrigeration ton is known, the piston displacement can be found:

V = R·C

V = Piston displacement
R = Legal refrigeration ton
C = Constant

1 refrigeration ton = 19,300 kJ/h = 3.86 kW (3,320 kcal/h)

Table 2-4

Nominal refrigerating capacity:
Calculation reference coefficient (C)

Refrigerant	Volume of a single cylinder	
	5000 cm^3 or less	More than 5000 cm^3
R12	13.9	13.1
R22	8.5	7.9
R500	12.0	11.3
R502	8.4	7.9

Example

Find the piston displacement, assuming that the 2T55HF compressor operating at 60 Hz has a legal refrigeration ton of 2.34.

C = 8.5 (R22)
V = 2.34×8.5=19.9 m^3/h

2. Volumetric efficiency ηv

Volumetric efficiency is the ratio of the gas volume actually sucked into the cylinder to the cylinder volume.
When the gas sucked into the cylinder is compressed and discharged, the gas remains in the cylinder top clearance. Due to this residual gas, the volume of fresh suction gas decreases. In addition, since there are some gas leaks from the piston ring, the piston displacement becomes 70% to 80% of the cylinder volume.
The volumetric efficiency is found using the compression ration. The larger the compression ratio is, the smaller the volumetric efficiency becomes. By contrast, the smaller the compression ratio is, the larger the volumetric efficiency becomes.

Fig.2-29

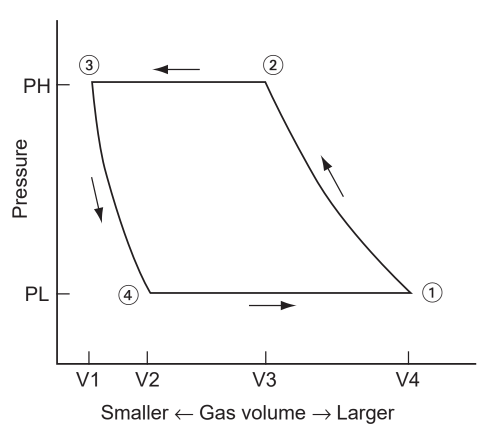


Fig.2-30

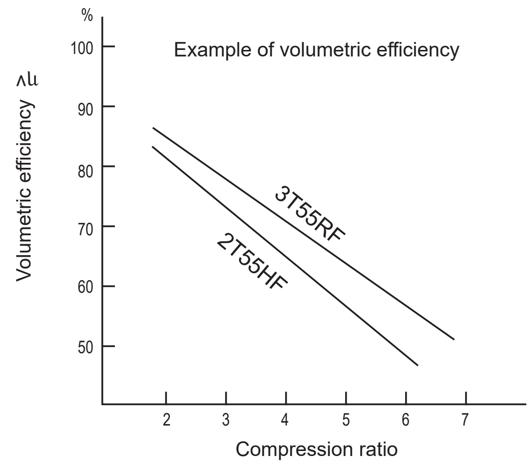
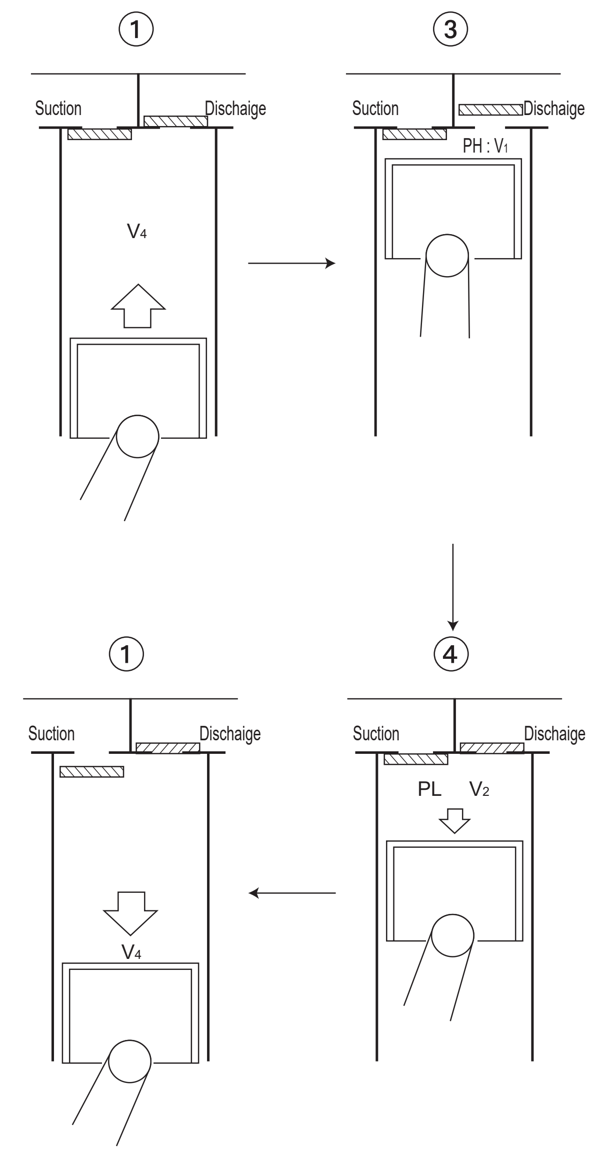


Fig.2-31



3. Actual volume of suction vapor compressed qv [m³/h]

The actual piston displacement [qv] can be found by multiplying the suction vapor volume [V] obtained according to the compressor specifications by the volumetric efficiency [ηv].

$qv = V \times \eta v$

Example

Find the actual volume of suction vapor compressed while the 2T55HF compressor is operating under the following conditions.

High pressure : 1.9MPa G → 2.0MPa abs

Low pressure : 0.5MPa G → 0.6MPa abs

Compression ratio $\frac{2.0}{0.6}$ = 3.33 Referring to the volumetric efficiency chart, $\eta v = 0.7$

From Example 5-1(1), $V = 19.9\ m^3/h$

Thus, $qv = 19.9 \times 0.7 = 13.93\ m^3/h$

4. Weight of refrigerant circulated qm [kg/h]

The weight of refrigerant circulated is that of refrigerant circulated per hour by the compressor, which is equal to the weight of the suction vapor of the compressor. If the refrigerating capacity (· kJ/h(and the refrigerating effect [We kJ/kg] are known, the weight of refrigerant circulated can be is found using the following formula:

$$qm\ (kg/h) = \frac{\phi \cdot [kJ/h]}{We[kJ/kg]}$$

The refrigerating capacity, however, cannot be determined according to the operating state. By multiplying the actual volume of suction vapor qv [m³/h] by the density of the suction gas 1/v [kg/m³], the weight of refrigerant circulated qm [kg/h] can be found.

$qm\ [kg/h] = [m^3/h] \times [kg/m^3]$

Example

Assuming that the specific volume of the suction gas is 0.04 m³/kg with reference to Example 5-1(3), find the weight of refrigerant circulated.

$qv = 13.93\ m^3/h$

$v = 0.04\ m^3/kg$

$$qm = 13.93 \times \frac{1}{0.04} \fallingdotseq 348.3\ kg/h$$

2.5.2 Refrigeration capacity calculation

As mentioned above, the weight of refrigerant circulated has been found according to the compressor parameters. In order to find the refrigeration capacity according to the actual operating state, multiply the weight of refrigerant circulated qm [kg/h] by the refrigeration effect We [kJ/kg].

$\phi[kJ/h] = qm\ [kg/h] \times We\ [kJ/kg]$

Example

Find the refrigerating capacity while the 2HC55HF (60Hz) compressor is operating under the conditions in Section 3-3.

From Example 5-1(4), the weight of refrigerant circulated (qm) is 348.3 kg/h, and

from operating data in Section 3-3, the refrigerating effect (We) is 153 kJ/kg,

$\phi = 348.3 \times 153 \fallingdotseq 53.290\ kJ/h$

Summary

In order to calculate the refrigerating capacity according to the P-h Chart:

1. Draw a refrigeration cycle on the P-h (Mollier) Chart according to the operating state.
2. Find the refrigerating effect, specific volume of suction gas, and compression ratio.
3. Calculate the piston displacement of the compressor.
4. Find the volumetric efficiency according to compression ratio and calculate the actual volume of suction vapor of the compressor.
5. Calculate the weight of refrigerant circulated according to the specific volume of the suction gas and the actual weight of suction vapor of the compressor.
6. By multiplying the refrigerating effect by the weight of refrigerant circulated, the refrigerating capacity can be found.

Exercise 5

For air conditioners using a 3T55RF compressor (with legal refrigerating capacity of 3.70/4.41 tons) operating under the following conditions, find the refrigeration capacity according to the P-h Chart.

High pressure:
1.9MPa
Low pressure:
0.5MPa
Suction gas temperature: 8˚C
Liquid temperature at expansion valve inlet: 45˚C
Power supply: 3
phase, 200 VAC, 60 Hz

However, assume that the volumetric efficiency is 0.75

Answers

Q1. Low pressure: 0.6MPa abs
High pressure: 1.8MPa abs

Q2. Moist vapor: X = 0.65

Q3. Superheated vapor: h = 450 kJ/kg

Q4. Sub-cooled liquid: h = 200 kJ/kg

Q5. V = 0.07 m^3/kg
h = 427 kJ/kg

Q6. 0.29

Exercise 1

Table 2-5

	P Absolute pressure MPa abs	t Temperature C	h Specific enthalpy kJ/kg	s Specific entropy kJ/(kg•K)	v Specific volume m^3/kg	x Dryness factor
Point A	0.8	80	460	1.88	0.04	Superheated vapor
Point	1.0	0	200			Subcooled liquid
Point C	0.2	-25	350			0.8
Point D	0.4	60	450	1.92	0.078	Superheated vapor
Point	0.25	0	410	1.83	0.1	Superheated vapor

Exercise 2

Table 2-6

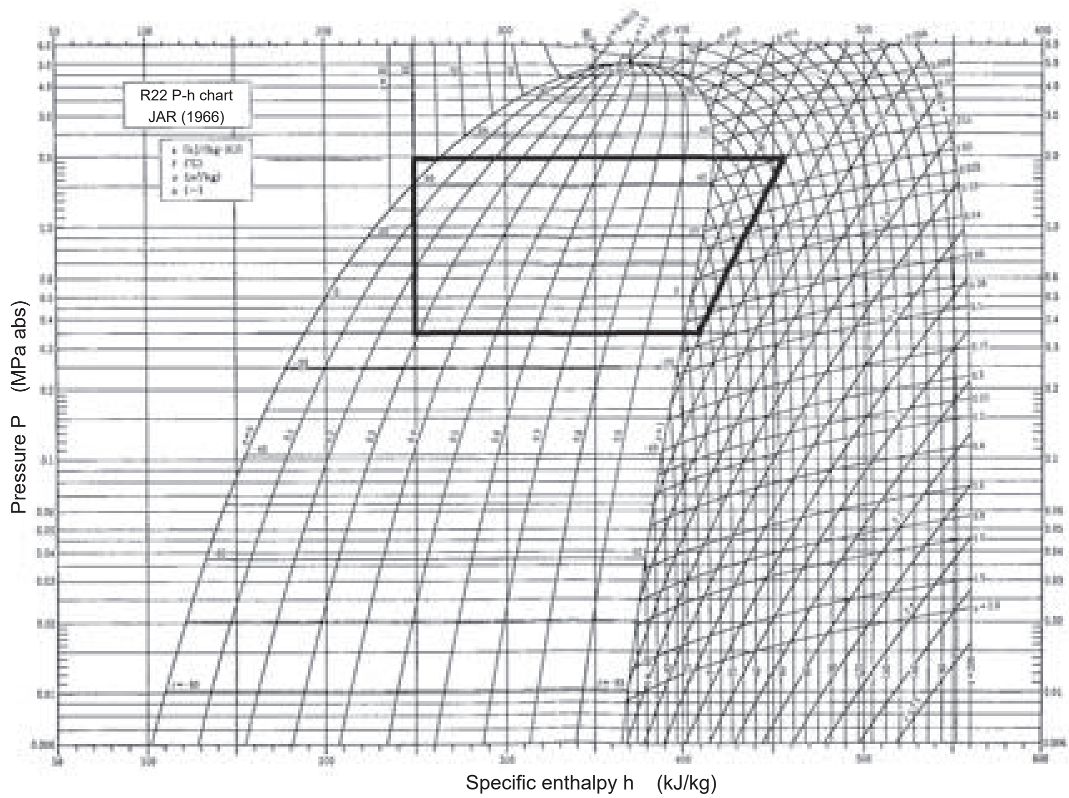
	P Absolute pressure MPa abs	t Temperature C	h Specific enthalpy kJ/kg	v Specific volume m^3/kg	x Dryness factor	s Specific entropy kJ/(kg•K)
Point 1	0.6	11	412	0.041		1.76
Point 2	1.4	55	432	0.019		1.76
Point 3	1.4	31	238			
Point 4	0.6	6	238		0.16	

Exercise 3

(1) 174 kJ/kg
(2) 20 kJ/kg
(3) 194 kJ/kg
(4) 8.7
(5) 2.33
(6) 24.4 kg/m^3

Exercise 4

Fig.2-32



■ Data

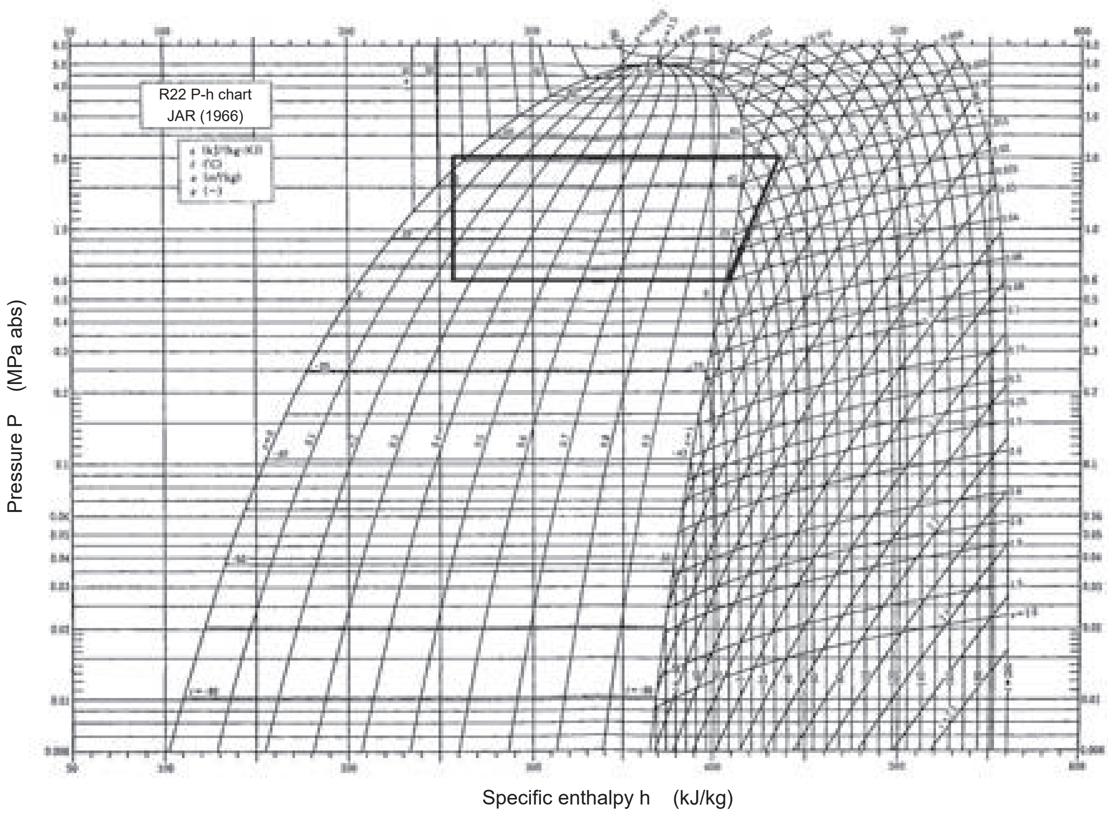
Evaporating pressure	0.35MPa abs
Suction gas	
Temperature	0˚C
Specific enthalpy	410 kJ/kg
Specific volume	0.07 m³/kg
Specific enthalpy at expansion valve inlet	250 kJ/kg
Sub-cooled degree	10˚C
Refrigerating effect	160 kJ/kg
Thermal equivalent of compressor work	45 kJ/kg
Condensing load	205 kJ/kg
Coefficient of performance	3.56
Compression ratio	5.43
Condensing pressure	1.9MPa abs
Discharge gas	
Temperature	88˚C
Specific enthalpy	455 kJ/kg

Al Mustaqbal University
College of Engineering & Technology
Air Conditioning and Refrigeration Engineering
Techniques Department

Subject: Maintenance AC System
Name of lecturer: Mustafa M.G
Stage : 3rd
Lecture No:
Date: / /

Exercise 5

Fig.2-33



According to the P-h Chart,

h_1 = 408 kJ/kg

h_2 = 436 kJ/kg

$h_3 = h_4$ = 256 kJ/kg

Refrigerating effect We = 152 kJ/kg

Specific volume of suction gas v = 0.04 m^3/kg

Piston displacement

V = 4.41 tons × 8.5=37.485 m^3/h

Actual volume of suction vapor

Since the volumetric efficiency is 0.75,

qv = 37.485 × 0.75 = 28.11 m^3/h

Weight of refrigerant circulated

qm = 28.11 × 1/0.04 = 702.8 kg/h

Refrigeration capacity

ϕ = 702.8 kg/h ×152 kJ/kg ≒ 106.826 kJ/h