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REFRIGERATION AND AIR CONDITIONING



C P ARORA

and material



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Preface

The need for a modern textbook in the field of refrigeration and air conditioning has been felt for a long time. This book presents a basic as well as applied thermodynamic treatment of the subject in a very comprehensive manner based on years of teaching and learning effort at the Indian Institutes of Technology, Mumbai and Delhi, and interaction with the industry.

The book is intended to serve as a text for undergraduate and to some extent postgraduate students of engineering. It should also serve as a useful reference for practising engineers. A few texts follow the extremely rigorous approach, whereas others are restricted to merely the elementary and empirical form. In this text a conscious effort has been made to maintain a reasonable level of rigour, but at the same time to employ simple techniques for solving fairly complex problems. Throughout the book, emphasis has been laid on physical understanding while at the same time relying on simple analytical treatment. A sound physical basis has also been laid for obtaining fairly precise estimates of refrigeration and air-conditioning equipment.

The presentation of the subject follows the classical line of separately treating the topics in refrigeration and air conditioning, the two being linked via the medium of the refrigerant evaporator. Accordingly, Chapters 1 to 13 are devoted to refrigeration and Chapters 14 to 22 to air conditioning. Chapters 23 and 24 deal with motors and controls and applications of refrigeration and air-conditioning process in food preservation.

The text and illustrative examples are in SI units throughout the book. Charts and tables, such as pressure-enthalpy diagrams for refrigerant 11 and carbon dioxide, enthalpy-composition diagrams for ammonia-water and lithium bromide-water systems, tables for solar radiation heat gain through glass, equivalent temperature differentials for walls and roofs, etc., have been adapted in SI units and are provided along with others, such as pressure-enthalpy diagram for refrigerant 12, psychrometric chart, etc.

Any claim to originality that may be advanced for the material presented here in refrigeration is with respect to (i) Ewing's construction to find the suction state for maximum COP, (ii) a comparison of refrigerants based on normal boiling points thus introducing the concept of thermodynamic similarity, (iii) a study of azeotropes,

(iv) class of service of compressors, (v) illustrative examples on both air-cooled and water-cooled condensers, (vi) the sizing of the capillary tube according to Fanno-line flow, (vii) the influence of a refrigerant on the augmentation of boiling heat transfer, (viii) heat-transfer analysis of both dry and flooded evaporators, (ix) the simulation of the vapour compression system, and (x) the analysis and calculations for mixtures in the vapour-absorption system using enthalpy-composition diagrams.

The approach to the subject of air conditioning is both fundamental and practice-oriented. A basic calculation procedure is given for the preparation of psychrometric charts. Lucid explanations, expressions and diagrams are given to develop the understanding of sensible, latent and total heat processes and loads. A separate chapter is devoted to solar radiation, leading not only to the study of solar-heat gains and cutting-solar load, but also to provide to the reader the basic knowledge to enable him to design systems for solar-energy utilization. The chapter on air-conditioning equipment design makes use of the concept of enthalpy potential involving simultaneous heat and mass transfer. Examples on air transmission include the static regain method of duct designing which leads to a balanced air-distribution system.

Chapter 23 adequately fills the need to provide essential information on the electrical aspects of the control of refrigeration and air-conditioning equipment. It also gives methods for the control of room conditions at partial loads. Finally, Chapter 24 takes up typical applications of refrigeration and air-conditioning to food preservation. These include chilling, freezing, freeze-drying and heat-drying.

The twentieth century saw large scale development in commercial refrigeration and air conditioning, particularly after du Pont introduced a family of chloro-fluoro-carbons, the so-called CFCs with the trade name of Freons. Now, as the new century begins, another revolution is taking place in the industry for replacing these very CFCs with alternatives on account of the ozone-depletion-potential of these refrigerants. The author, therefore, considers that it is his duty, and he owes it to the readers to present this updated version with exhaustive revision of the contents of the book.

Many research and postgraduate students are interested in evaluating thermodynamic properties of new refrigerants and refrigerant mixtures. The basic procedure to evaluate the thermodynamic properties of *pure refrigerants* is, therefore, given in Chapter 1, and the same for *ideal and non-ideal mixtures* and particularly *Propane/Isobutane mixtures* in Chapter 4. Chapter 4 on refrigerants contains an exhaustive treatment of the topics *substitutes for CFC Refrigerants*, particularly CFC 12, and *Non-isothermal Refrigeration* using non-azeotropic mixtures of refrigerants. In addition, empirical relations for thermophysical properties of refrigerants, and *supercritical vapour compression cycle* for CO₂ as refrigerant with a potential to substitute for CFCs are also given in this chapter.

Chapter 9 on Evaporators includes many illustrative examples for *simulation and design of flooded and direct-expansion chillers* which include pressure drop calculations and use of *Slipcevic correlations* for tubes with roughened surfaces.

Since water-lithium bromide system has recently gained some popularity with the use of waste heat for refrigeration, the representation of vapour absorption cycle on *Inp versus 1/T* diagram and practical *single-effect* and *double-effect water-lithium bromide vapour absorption cycles* have been described in Chapter 12 on Vapour Absorption System.

In Chapter 20 on Design of A/C Apparatus the treatment of the topic has been greatly extended to include determination of *air-side heat transfer coefficient* and *cooling tower selection*. Examples include those on induced-draft counterflow and crossflow atmospheric cooling towers.

Prominent features added in the second edition were

- (i) *Standard rating cycle for domestic refrigerators and second law efficiency* in Chapter 3
- (ii) *Calorimetric method of determining refrigerating capacity of hermetic compressors* in Chapter 6, *R22 centrifugal compressors* in Chapter 6 also due to the present trend of their use as substitutes for R11 chillers
- (iii) *Linde–Hampson process for liquefaction of gases* in Chapter 11; also, *reversed stirling cycle* in this chapter due to the application of this cycle in a big way in Philips Liquefier
- (iv) *Clean spaces* in Chapter 16 and processing and transmission of air in clean rooms in Chapter 21
- (v) *Flat-plate solar collector* in Chapter 17 as an extension of the topic of solar radiation
- (vi) *Water vapour transmission* and use of vapour barriers in Chapter 18
- (vii) *Building design features* and measures for conservation of energy in Chapter 19
- (viii) *Static regain method of duct design* in Chapter 21
- (ix) Example on *conversion of split-phase motor into capacitor-start motor* to increase starting torque which may help using compressor of one refrigerant with another refrigerant in Chapter 23
- (x) *Freeze-drying of Yoghurt* in Chapter 24

Further, a major contribution to this edition is in the form of a detailed Appendix which is now presented in three parts as follows:

- A. *Correlations* on thermodynamic properties of refrigerants R12, R134a, R152a, R22, R290 and R600a
- B. *Tables* on thermodynamic properties of the above and other refrigerants, R290/R600a mixtures, etc.
- C. *Charts*

When the second edition was published in 2000, the refrigeration and air-conditioning industry was embarking on to an era of new refrigerants. Due to the problem of the depletion of the ozone layer, CFC refrigerants R11, R12, R113, R114, and R502 were to be phased out on 31.12.2000, and alternative HFC and HCFC refrigerants were to be used from 1.1.2001.

The second edition did provide a study of the alternative refrigerants which were planned. But since 2000, certain new refrigerants have taken their place as substitutes. They have come to be accepted by the industry, and plants working on them have been designed and installed. For example, HFC 134a now occupies place of pride as a substitute for CFC R12. However, HCFC R22 continues to be used and loved by the industry, although an HFC blend R410A is also favoured by some. At the same time, there is a newfound enthusiasm for ammonia. Further, HCFC R123 has now replaced CFC R11. Both the HCFCs, R22 and R123, are permitted for use till 2030.

Hence, it had become absolutely necessary to revise the book.

In this revision, topics on R11 and R12 have been retained to an extent for the sake of comparison. But there is greater emphasis on R123 and R134a. Emphasis on R22 and ammonia remains as such. Detailed comparisons have, however, been made between HCFC R22 and HFC alternatives R410A and R407C. Similarly, comparisons have been made between HCFC R123 and the HFC alternative R245fa. Accordingly, a number of comparison tables, and solved problems have been introduced in Chapters 3, 4, and 6 in the edition.

For the same reasons, tables of properties of HCFC R123, and HFCs R134a, R404A, R407C, R410A, and R507A have been added in Appendix B. In addition, vapour-region pressure-enthalpy diagrams of R123 and R134a have been included in Appendix C.

There are other inclusions in this edition. ‘Scroll compressors’ are the new positive displacement machine. They were developed a decade ago, but have become very popular only in recent years. They are being employed with R134a, and with R22 in low-to-medium capacity machines in the range of 1 to 12 TR. Hence, a section on the working of scroll compressors has been devoted in Chapter 6 on compressors.

Also, taking note of the need of students to learn more about the practical aspects of a system, a detailed section on ‘Installation, Service, and Maintenance’ has been included in Chapter 10 on Complete Vapour Compression System.

An interesting feature of air conditioning is the ‘comfort zone’. As it forms the basis of design, an ASHRAE ‘Comfort Chart’ has now been included in Chapter 16 on Design Conditions.

Lastly, to ignite the imagination of the student on the wide variety of Industrial air-conditioning applications, three typical HVAC applications, ‘Tunnels Ventilation’, ‘Station Air Conditioning’, and ‘Mine Ventilation and Air Conditioning’ have been described in Chapter 20 on Applications.

I bow with gratitude before the Divine Father, Mother, Friend, and Beloved, the source of all knowledge, Who made me an instrument to write this book.

At this juncture, I remember my father’s words: “My investment is in my children”. Truly speaking, the benefits of this book flow from the investment made by my father.

I want to express my heartfelt gratitude to the Divine for the Love, Kindness and Affection bestowed on me through my children and their spouses: Sangeeta–Vivek, Smita–Rajat, Shubhra–Hemant, and Amitabh–Shailaja and grandchildren Himali, Ishika, Vaibhav, Aakriti, Shreya, Atyant, and two new and loving grandchildren, Anisha and Rishi, born since the publication of the last edition.

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May this wonderful subject of Refrigeration and Air Conditioning, and this book inspire teachers, students, and practicing engineers to explore new vistas in the field. Please feel free to send in your feedback at the book’s website.

C P Arora

List of Principal Symbols

Capital letters

A	Area
A_F	Face area
C	Velocity, thermal conductance, concentration (in mass transfer), clearance factor, heat capacity/specific heat
D	Diameter, diffusion coefficient, mass of vapour distilled from generator
E	Emissive power
F	Force, genometric factor, rich solution circulation
G	Mass velocity
H	Enthalpy, head
I	Solar radiation intensity
I_D	Intensity of direct solar radiation
I_d	Identity of diffuse solar radiation
K	Dynamic loss coefficient
L	Fin width, length, air mass
M	Molecular weight, stability criterion in finite difference approximation
N	Number of tubes
P	Perimeter, power requirement
Q	Heat transfer
Q_L	Latent heat transfer
Q_s	Sensible heat transfer
\dot{Q}_v	Volume flow rate of air
R	Gas constant, thermal resistance
S	Entropy
T	Absolute temperature
U	Internal energy, overall heat transfer coefficient
V	Volume
V_p	Piston displacement
W	Work, moisture content of material
X	Bypass factor

Small letters

a	Velocity of sound, absorptivity
c	Specific heat
c_p	Specific heat at constant pressure
c_v	specific heat at constant volume

d	Solar declination angle
f	Heat transfer coefficient, friction factor, specific rich solution circulation
g	Acceleration due to gravity
h	Specific enthalpy, heat transfer coefficient, hour angle
h_M	Mass transfer coefficient
k	Thermal conductivity
k_d	Diffusion coefficient
k_ω	Diffusion coefficient based on specific humidity
l	Fin height, tube length
m	Mass, polytropic index of expansion
n	Polytropic index of compression, number of moles, recirculation number
p	Pressure
Δ_p	Pressure loss
p_s	Static pressure
p_T	Total pressure
p_v	Velocity pressure
q	Heat flux, heat transfer per unit mass
r	Radius, compression ratio, reflectivity
s	Specific entropy
t	Celsius temperature
t_e	Sol-air temperature
Δt_E	Effective temperature difference
u	Specific internal energy, tangential velocity
v	Specific volume
w	Specific work, moisture removal
x	Distance, dryness fraction, liquid phase mole fraction
y	Vapour phase mole fraction
z	Height above datum

Greek letters

α	Thermal diffusivity, wall solar azimuth angle
β	Coefficient of thermal expansion, solar altitude
γ	Adiabatic index, solar azimuth angle
δ	Joule Thomson coefficient
ξ	Coefficient of performance
ε	Emissivity, heat exchanger effectiveness
λ	Decrement factor
η	Efficiency
η_p	Polytropic efficiency
ϕ	Flow coefficient, relative humidity, time lag
σ	Stefan-Boltzman constant, surface tension
μ	Dynamic viscosity, head coefficient, degree of saturation
ν	Kinematic viscosity
ρ	Density
ψ	Zenith angle
ξ	Concentration by weight

τ	Time, transmissivity
θ	Angle of incident, excess temperature
ω	Specific humidity, angular velocity
\mathcal{H}_u	Lockhart-Martinelli parameter for two phase turbulent flow

Dimensionless numbers

Bi	Biot number
Bo	Boiling number
Co	Condensation number
Fo	Fourier number
Gr	Grashof number
K_f	Load factor in boiling
Le	Lewis number
Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number
Sc	Schmidt number
Sh	Sherwood number
St	Stanton number
θ	Trotout number

Subscripts

A	Absorber
C	Convective
I	Infiltration
R	Radiative
S	Apparatus dew point, wetted surface
TP	two phase
a	Ambient, poor solution, dry air
b	Black body
c	Cold, clearance, condensate, critical
d	Dynamic loss, diffusion, vapour from generator, dew point, discharge
e	Entrainment
f	Friction, saturated liquid, fin, fouling, fluid
fg	Vaporization
g	Glass, saturated vapour, air-side
h	Generator, hot
i	Inside, initial
is	Isentropic
k	Heat rejection
m	Log mean
max	Maximum
min	Minimum
n	Nozzle, normal to surface
o	Outside, heat absorption or refrigeration, molar, stagnation
r	Radial, refrigerant-side, rich solution, reduced property

rel	Relative
s	Suction, at normal boiling point, saturation, saturated solid
sd	Shading
sg	Sublimation
t	Total, based on extended surface side area
u	Tangential
v	Vapour, volumetric
w	Wall, water
x	x-direction
∞	Free stream

Superscripts

*	Per ton refrigeration, thermodynamic wet bulb
'	Pure substance
L	Saturated liquid mixture, wet bulb
V	Saturated vapour mixture

Visual Preview

Introduction

The student is first introduced to the theories and concepts regarding the working of an air conditioner and refrigerator.

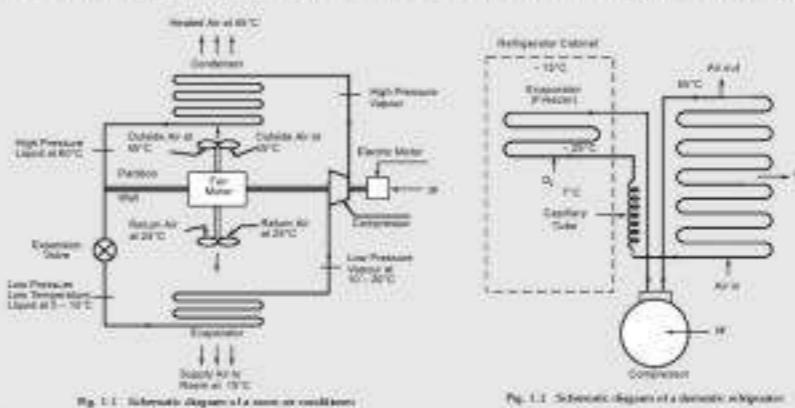
1



Introduction

1.1 A BRIEF HISTORY OF REFRIGERATION

The methods of production of cold by mechanical processes are quite recent. Long back in 1748, William Cullen of Glasgow University produced refrigeration by creating partial vacuum over ethyl ether. But, he could not implement his experience



Vapour Compression System 5

(e) Carnot COP

$$T_{max} = \frac{273}{40 - (0)} = 6.8$$

COP of the cycle

$$\varepsilon = \frac{h_1 - h_4}{h_2 - h_1} = \frac{112.8}{213.96 - 187.5} = 4.3$$

Example 3.2 R 134a System

Chlorine in the Freon 12 (CCl_2F_2) molecule depletes the ozone layer in the earth's upper atmosphere. R 12 has now been replaced by the ozone-friendly R 134a ($C_2H_4F_2$).

For the conditions of Example 3.1, do calculations for R 134a, and compare results.

Solution From the table of properties of R 134a in the Appendix, we have

$$\begin{aligned} p_0 &= 0.2958 \text{ MPa} & p_1 &= 1.0166 \text{ MPa} & h_4 &= 256.41 \text{ kJ/kg} \\ h_1 &= 398.6 \text{ kJ/kg} & v_1 &= 0.06931 \text{ m}^3/\text{kg} \\ s_1 &= 1.7541 \text{ kJ/kgK} & s'_1 &= 1.7111 \text{ kJ/kg K} & h'_2 &= 419.43 \text{ kJ/kg} \\ C_p \text{ at } p_1 &= 1.145 \text{ kJ/kg K} & T'_2 &= 313 \text{ K} \end{aligned}$$

(a) For isentropic compression,

$$\begin{aligned} s_2 &= s_1 = 1.7541 + C_p \ln \frac{T_2}{313} \\ \Rightarrow T_2 &= 317.6 \text{ K (44.4°C)} \\ h_2 &= h'_1 + C_p (T_2 - T'_2) \\ &= 419.43 + 1.145(4.4) = 424.5 \text{ kJ/kg} \\ w &= h_2 - h_1 = 424.5 - 398.6 = 25.9 \text{ kJ/kg} \\ q_o &= h_1 - h_4 = 398.6 - 256.41 = 142.19 \text{ kJ/kg} \end{aligned}$$

Solved Examples

Solved Examples are provided in sufficient number in each chapter and at appropriate locations to aid in understanding of the text material.

Practice Problems

Over 150 Practice Problems are given to provide hands-on practice to students in problem-solving.

Revision Exercises

- 3.1 A 15 TR Freon 22 vapour compression system operates between a condenser temperature of 40°C and an evaporator temperature of 5°C.
 - (a) Determine the compressor discharge temperature:
 - (i) Using the $p-h$ diagram for Freon 22.
 - (ii) Using saturation properties of Freon 22 and assuming the specific heat of its vapour as 0.8 kJ/kg. K.
 - (iii) Using superheat tables for Freon 22.
 - (b) Calculate the theoretical piston displacement and power consumption of the compressor per ton of refrigeration.
- 3.2 A simple saturation ammonia compression system has a high pressure of 1.35 MN/m² and a low pressure of 0.19 MN/m². Find per 400,000 kJ/h of refrigerating capacity, the power consumption of the compressor and COP of the cycle.
- 3.3 (a) A Freon 22 refrigerating machine operates between a condenser temperature of 40°C and an evaporator temperature of 5°C. Calculate the increase (per cent) in the theoretical piston displacement and the power consumption of the cycle:

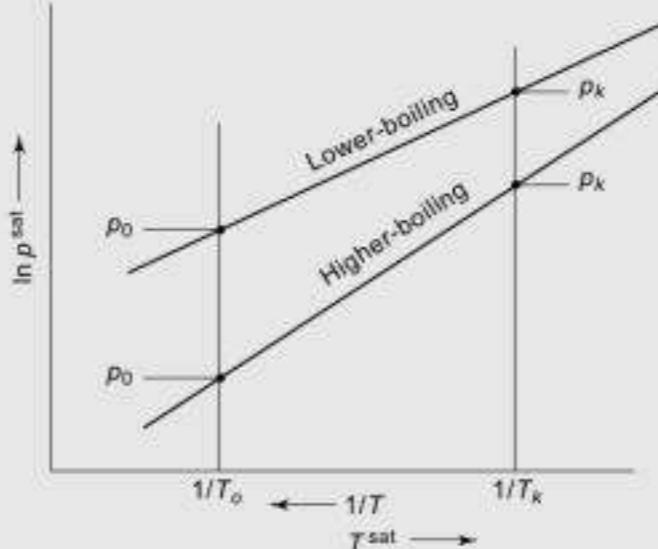


Fig. 4.1(b) Comparison of pressures of lower-boiling and higher-boiling refrigerants at given evaporator and condenser temperatures

Concepts

Normal boiling point of refrigerants is emphasized as an important performance criterion.

Comfort Airconditioning

Chapters 14 to 22 are primarily for comfort air conditioning topics like ASHRAE comfort chart, solar radiation heat gain, pyrometric calculations for cooling and heating, design of A/c apparatus and fan-duct system interaction.

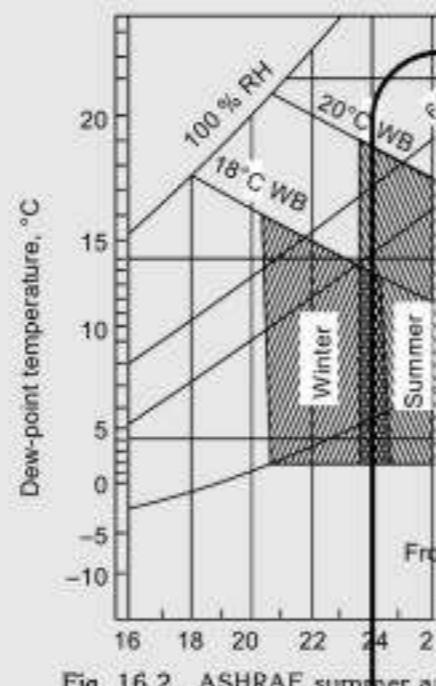


Fig. 16.2 ASHRAE summer air chart

$$\dot{Q}_B' = \dot{Q}_B \sqrt{\frac{\Delta p'}{\Delta p_B}}$$

$$\dot{Q}_C' = \dot{Q}_C \sqrt{\frac{\Delta p'}{\Delta p_C}}$$

$$\dot{Q}_D' = \dot{Q}_D \sqrt{\frac{\Delta p'}{\Delta p_D}}$$

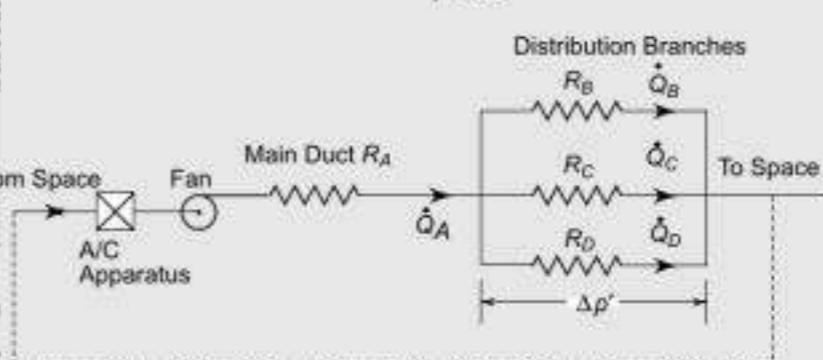


Fig. 22.11 Simple fan-system network

Tables

Refrigerants in use before 2000, and alternative refrigerants have been compiled in a list and are compared, and comparison of CFC 11 alternatives is summarized for centrifugal compressors.

Table 4.9: Common CFCs and possible alternatives with normal boiling points

Designation	Category	Chemical Formula	N.B.P., °C	Flammability
R 113	CFC	$\text{C}_2\text{Cl}_2\text{F}_3$	47.68	Non-flammable
R 141b	HCFC	$\text{CH}_3\text{CCl}_2\text{F}$	32.1	Slightly flammable
R 152	HFC	$\text{CH}_2\text{FC}\text{CH}_2\text{F}$	30.7	Flammable
R 123	HCFC	$\text{C}_2\text{HCl}_2\text{CF}_3$	27.82	Non-flammable
R 11	CFC	CCl_3F	23.7	Non-flammable
R 245fa	HFC		14.9	Flammable
R 600a (Isobutane)	HC	$(\text{CH}_3)_2\text{CH}$	-11.67	Flammable
R 134	HFC	CHF_2CHF_2	-19.8	Non-flammable
R 152a	HFC	CH_3CHF_2	-24.02	Slightly flammable
R 134a	HFC	$\text{CF}_3\text{CH}_2\text{F}$	-26.07	Non-flammable
R 12	CFC	CCl_2F_2	-29.8	Non-flammable
R 717 (Ammonia)		NH_3	-33.3	Flammable
R 22	HCFC	CHCl_2F_2	-40.8	Non-flammable
R 290 (Propane)	HC	C_3H_8	-42.1	Flammable
R 407C	HFC		-43.63/-36.63	-
R 502	CFC		-45.4	Non-flammable
R 404 A	HFC		-46.22/-45.47	-
R 507 A	HFC		-46.74	-
R 145a	HFC	CH_3CF_3	-47.35	Slightly flammable
R 125	HFC	CHF_2CF_3	-48.55	Non-flammable
R 410 A	HFC		-51.44/-51.36	-
R 32	HFC	CH_2F_2	-52.024	Slightly flammable

Calculation of Enthalpy of Mixture in Vapour Phase

Method developed by Agarwal and Arora² will now be described. Fig. 4.15 shows the vapour-liquid domes of pure components 1 and 2, and mixture of certain composition on a $p-h$ diagram.

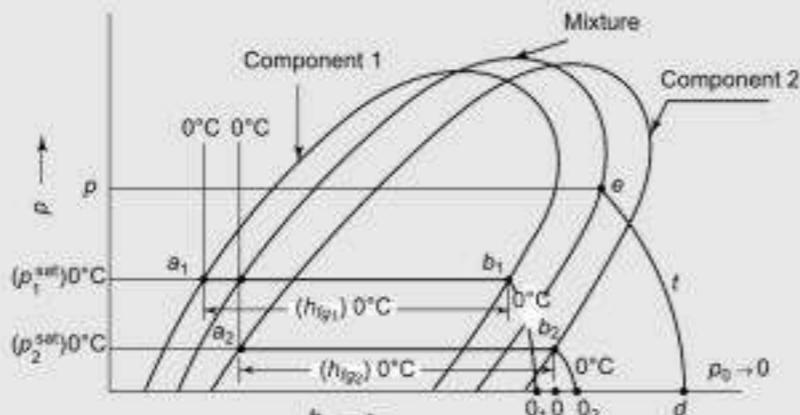


Fig. 4.15 Proposed method for vapour mixture enthalpy calculation

The figure illustrates how the enthalpies of saturated liquid and saturated vapour may be calculated. The proposed method assumes values for reference

Figures

Apart from numerous self-explanatory figures, an innovative new pH diagram has been developed to estimate the properties of mixed refrigerants, as the need of the day is to find new refrigerant blends as alternatives. This is for PG and research students.

Simulation Problems

For advanced students, procedures to develop computer methods for design, simulation and optimization of refrigeration systems are given.

Example 9.4 Estimation of D-X Chiller Capacity (Simulation)

The following specifications are given for an R 22 D-X Chiller.

Condensing temperature, t_c	43°C
Saturated suction temperature	2°C
Number of passes, n	8
Tubes in each pass	12, 16, 20, 24, 30, 32, 32, 34
Evaporator superheat	5°C
Inlet water temperature, t_{w_1}	11.1°C
Outlet water temperature, t_{w_2}	7.2°C
Refrigerant pressure drop in evaporator 0.14 bar (assumed)	
Shell diameter, D_s	0.406 m
Tube length between tube sheets, l	2.213 m
Tube ID, D_t	0.0158 m
Tube OD, D_o	0.0191 m
Tube pitch (triangular), P_T	0.0222 m
Number of baffles	21
Baffle pitch, P_B	0.0762 m
Baffle cut	0.094 m

Appendices

For the benefit of students pursuing postgraduate studies and research, correlations of properties of refrigerants are given in Appendix A.

where ρ_L is in kg/dm³, and the constants are as follows:

$$\begin{aligned} D_1 &= 0.2477199 & D_2 &= -0.1480948 & D_3 &= 0.008001550 \\ D_4 &= -0.01962269 & D_5 &= 0.0023223 & D_6 &= -0.0001057677 \end{aligned}$$

Zero-Pressure Constant Volume Specific Heat

$$C_v^0 = C_{v1} + C_{v2} T + C_{v3} T^2 + C_{v4} T^3 \quad (\text{A.1.8})$$

where the units of specific heat are in kJ/kg.K, and the constants are

$$\begin{aligned} C_{v1} &= 0.0479836 & C_{v5} &= -2.94985 \times 10^{-6} \\ C_{v2} &= 0.00238154 & C_{v6} &= 1.37374 \times 10^{-9} \end{aligned}$$

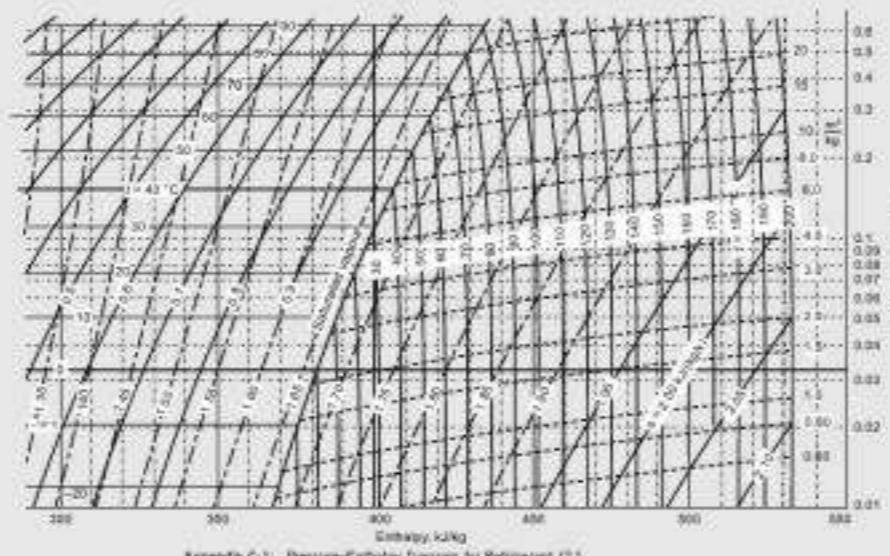
A.2 CORRELATIONS FOR THERMODYNAMIC PROPERTIES OF R 134a

The correlations given by Wilson and Basu* have been used:

Vapour Pressure Correlation

$$\ln P_r = P_1 + \frac{P_2}{T_r} + P_3 T_r + P_4 T_r^2 + \frac{P_5 (P_6 - T_r)}{T_r} \ln (P_s - T_r) \quad (\text{A.2.1})$$

* Wilson D.P. and Basu R.S., 'Thermodynamic properties of a new stratospherically safe working fluid-Refrigerant 134a', *ASHRAE Trans.*, Vol. 94, pp. 2095-2118, 1988.



Charts

Many new tables and charts have been introduced in Appendices B and C to expand the scope of study and problem-solving.



Introduction



1.1 A BRIEF HISTORY OF REFRIGERATION

The methods of production of cold by mechanical processes are quite recent. Long back in 1748, William Coolen of Glasgow University produced refrigeration by creating partial vacuum over *ethyl ether*. But, he could not implement his experience in practice. The first development took place in 1834 when Perkins proposed a hand-operated compressor machine working on ether. Then in 1851 came Gorrie's air refrigeration machine, and in 1856 Linde developed a machine working on ammonia.

The pace of development was slow in the beginning when steam engines were the only prime movers known to run the compressors. With the advent of electric motors and consequent higher speeds of the compressors, the scope of applications of refrigeration widened. The pace of development was considerably quickened in the 1920 decade when du Pont put in the market a family of new working substances, the fluoro-chloro derivatives of methane, ethane, etc.—popularly known as chloro fluorocarbons or CFCs—under the name of *Freons*. Recent developments involve finding alternatives or substitutes for Freons, since it has been found that chlorine atoms in Freons are responsible for the depletion of ozone layer in the upper atmosphere. Another noteworthy development was that of the ammonia-water vapour absorption machine by Carre. These developments account for the major commercial and industrial applications in the field of refrigeration.

A phenomenon called *Peltier* effect was discovered in 1834 which is still not commercialized. Advances in *cryogenics*, a field of very low temperature refrigeration, were registered with the liquefaction of oxygen by Pictet in 1877. Dewar made the famous Dewar flask in 1898 to store liquids at cryogenic temperatures. Then followed the liquefaction of other permanent gases including helium in 1908 by Onnes which led to the discovery of the phenomenon of *superconductivity*. Finally in 1926, Giaque and Debye independently proposed adiabatic demagnetization of a paramagnetic salt to reach temperatures near absolute zero.

Two of the most common refrigeration applications, viz., a window-type room air conditioner and a domestic refrigerator, have been described in the following pages.

1.1.1 Room Air Conditioner

Figure 1.1 shows the schematic diagram of a typical window-type room air conditioner, which works according to the principle described below:

Consider that a room is maintained at constant temperature of 25°C . In the air conditioner, the air from the room is drawn by a fan and is made to pass over a *cooling coil*, the surface of which is maintained, say, at a temperature of 10°C . After passing over the coil, the air is cooled (for example, to 15°C) before being supplied to the room. After picking up the room heat, the air is again returned to the cooling coil at 25°C .

Now, in the cooling coil, a liquid working substance called a *refrigerant*, such as CHClF_2 (monochloro-difluoro methane), also called *Freon* 22 by trade name, or simply *Refrigerant* 22 (R 22), enters at a temperature of, say, 5°C and evaporates, thus absorbing its latent heat of vaporization from the room air. This equipment in which the refrigerant evaporates is called an *evaporator*.

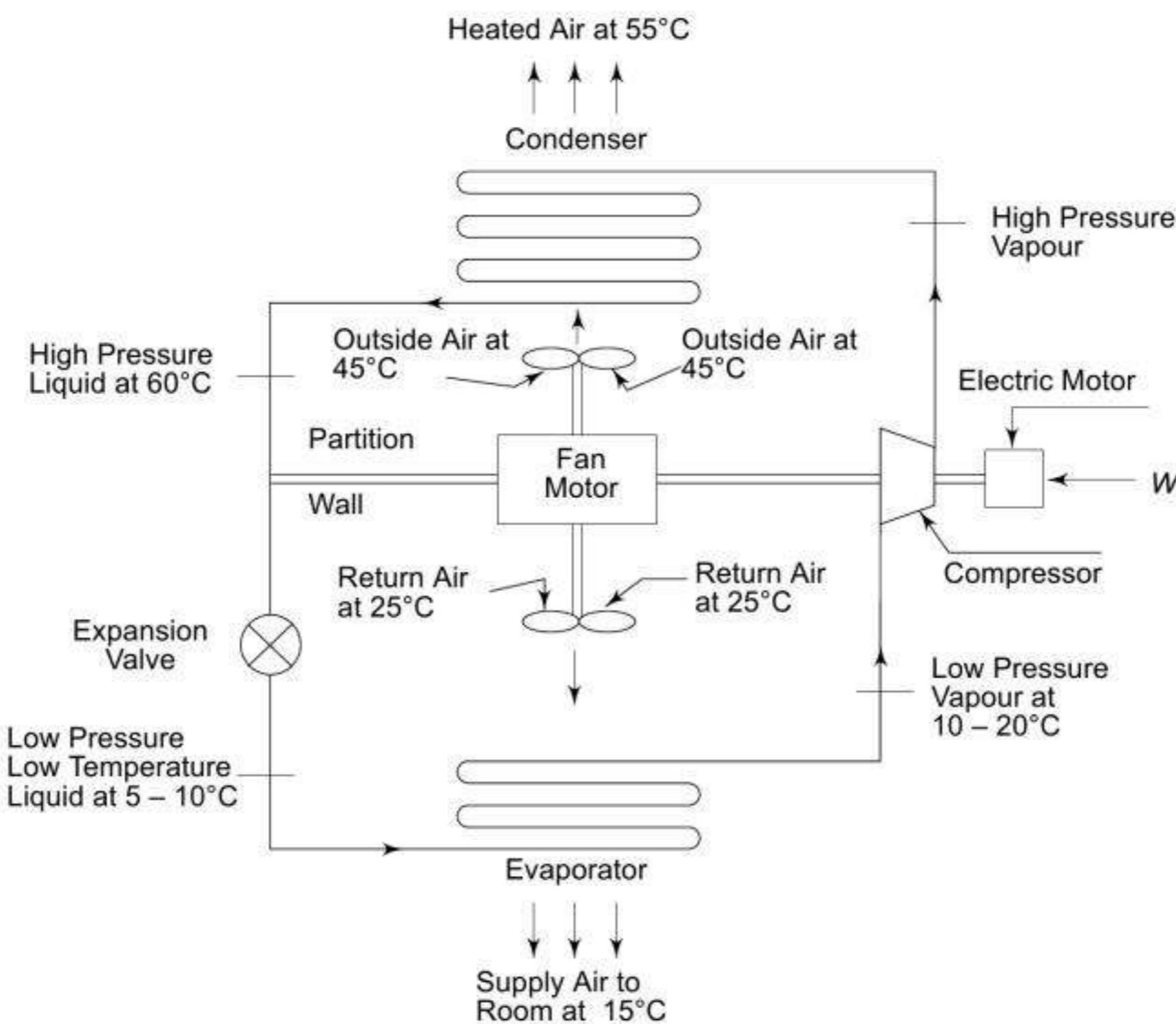


Fig. 1.1 Schematic diagram of a room air conditioner

After evaporation, the refrigerant becomes vapour. To enable it to condense back and to release the heat—which it has absorbed from the room while passing through the evaporator—its pressure is raised by a *compressor*. Following this, the high pressure vapour enters the *condenser*. In the condenser, the outside atmospheric air, say, at a temperature of 45°C in summer, is circulated by a fan. After picking up the

latent heat of condensation from the condensing refrigerant, the air is let out into the environment, say, at a temperature of 55°C . The condensation of refrigerant may occur, for example, at a temperature of 60°C .

After condensation, the high pressure liquid refrigerant is reduced to the low pressure of the evaporator by passing it through a pressure reducing device called the *expansion device*, and thus the cycle of operation is completed. A partition wall separates the high temperature side of the condenser from the low temperature side of the evaporator.

The principle of working of large air conditioning plants is also the same, except that the condenser is *water cooled* instead of being *air cooled*.

1.1.2 Domestic Refrigerator

The working principle of a domestic refrigerator is exactly the same as that of an air conditioner. A schematic diagram of the refrigerator is shown in Fig. 1.2. Like the air conditioner, it also consists of the following four basic components:

- (i) Evaporator; (ii) Compressor; (iii) Condenser; (iv) Expansion device.

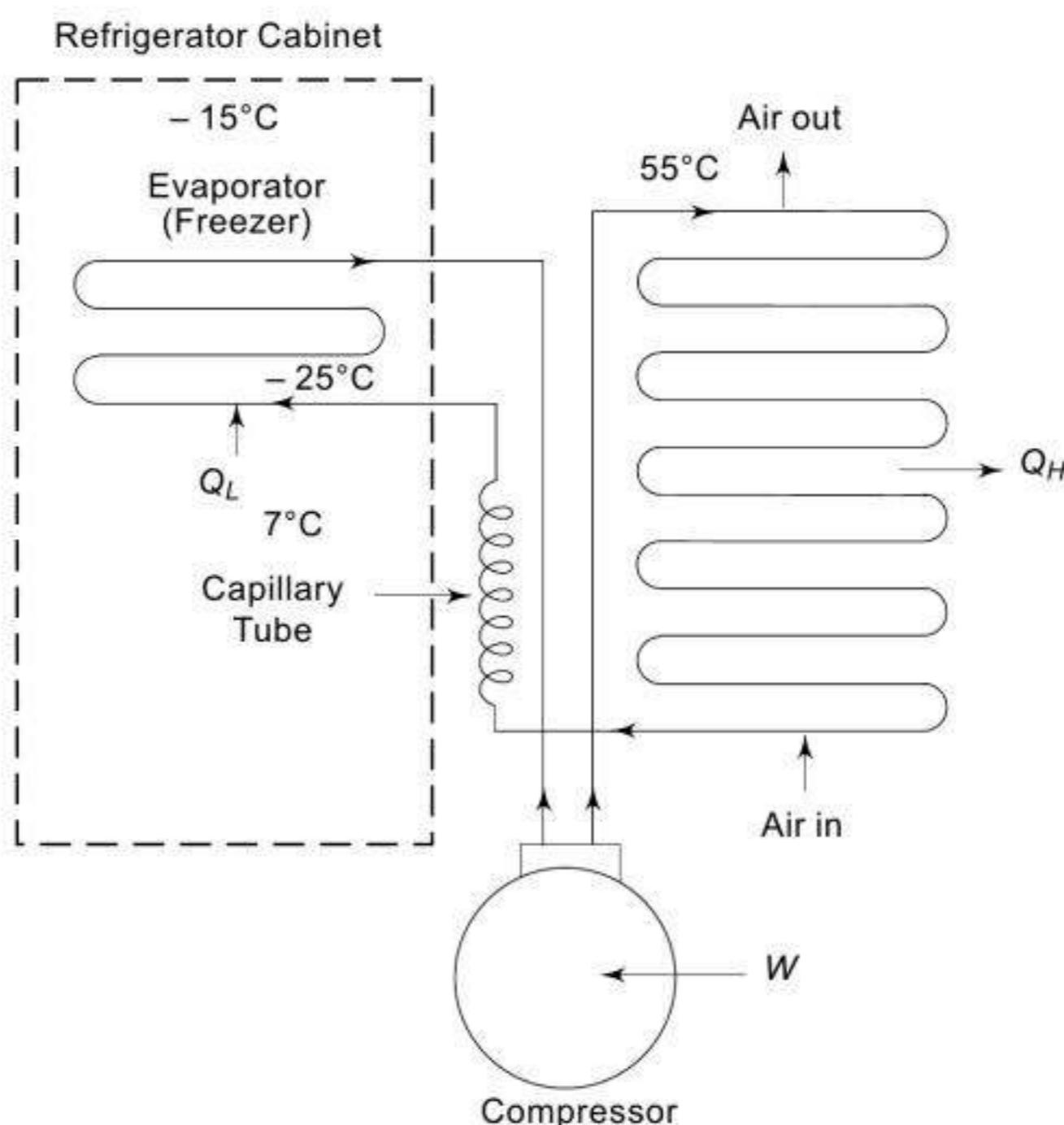


Fig. 1.2 Schematic diagram of a domestic refrigerator

But there are some design features which are typical of a refrigerator. For example, the evaporator is located in the *freezer* compartment of the refrigerator. The freezer forms the coldest part of the cabinet with a temperature of about -15°C , while the refrigerant evaporates inside the evaporator tubes at -25°C . Just below the

freezer, there is a *chiller tray*. Further below are compartments with progressively higher temperatures. The bottom-most compartment which is meant for vegetables is the least cold one. The cold air being heavier flows down from the freezer to the bottom of the refrigerator. The warm air being lighter rises from the vegetable compartment to the freezer, gets cooled and flows down again. Thus, a *natural convection current* is set up which maintains a temperature gradient between the top and the bottom of the refrigerator. The temperature maintained in the freezer is about -15°C , whereas the mean inside temperature of the cabinet is 7°C .

The design of the condenser is also a little different. It is usually a *wire and tube* or *plate and tube* type mounted at the back of the refrigerator. There is no fan. The refrigerant vapour is condensed with the help of surrounding air which rises above by natural convection as it gets heated after receiving the latent heat of condensation from the refrigerant. The standard condensing temperature is 55°C .

Note In both the room air conditioner as well as the refrigerator a long narrow bore tube, called the capillary tube, is employed as the expansion device.

In the modern no-frost refrigerators, the evaporator is located outside the freezer compartment. The cold air is made to flow by forced convection by a fan.

Working Substances in Refrigerating Machines The working substance being used in air conditioners is R22. In refrigerators R12 has been used before the year 2000. But R12 is a CFC (chloro-fluoro carbon). Because of the ozone-layer depletion problem, alternatives such as the following are being used in place of R12.

- (i) Refrigerant 290 or R290, viz., Propane (C_3H_8).
- (ii) Refrigerant 134a or R134a, viz., Tetra-fluoroethane ($\text{C}_2\text{H}_2\text{F}_4$)
- (iii) Refrigerant 600a or R 600a, viz., Isobutane (C_4H_{10}).

1.2 SYSTEME INTERNATIONAL D'UNITES (SI UNITS)

SI or the International System of Units is the purest form and an extension and refinement of the traditional metric system. In SI, the main departure from the traditional metric system is in the use of *Newton* as the unit of force.

There are six basic SI units as given in Table 1.1. The units of other thermodynamic quantities may be derived from these basic units.

Table 1.1 Basic SI units

Quantity	Unit	Symbol
Length	metre	m
Mass	kilogram	kg
Time	second	s
Temperature	kelvin	K
Electric current	ampere	A
Luminous intensity	candela	cd

The unit of temperature is kelvin which measures the absolute temperature given by

$$T = t + 273.15$$

where t is the Celsius temperature in °C.

1.2.1 Unit of Force

Force F is proportional to mass m and acceleration a , so that

$$F = C(m)(a) \quad (1.1)$$

where C is a proportionality constant. The SI unit of force, viz., Newton denoted by the symbol N is derived from unit values taking the proportionality constant as unity. Thus, one newton is

$$1\text{N} = (1\text{ kg}) \left(1 \frac{\text{m}}{\text{s}^2} \right) = 1 \frac{\text{kg} \cdot \text{m}}{\text{s}^2}$$

The MKS unit of force, kgf, defined by Eq. (1.1) is

$$1\text{ kgf} = \frac{1}{9.80665} (1\text{ kg}) \left(9.80665 \frac{\text{m}}{\text{s}^2} \right) = 1\text{ kgf}$$

which represents a unit weight or the gravitational force on one kilogram mass. In the above definition, the value of the constant C is taken as equal to the standard gravitational acceleration so that one kilogram mass has one kilogram weight.

It can be seen that

$$1\text{ kgf} = (1\text{ kg}) \left(9.80665 \frac{\text{m}}{\text{s}^2} \right) = 9.80665\text{ N}$$

Also, we know that

$$1\text{ lbf} = 0.453592\text{ kgf}$$

1.2.2 Unit of Pressure

The SI unit of pressure p can also be derived from its definition as force per unit area. Thus

$$[p] = \frac{[F]}{[A]} = \text{N/m}^2$$

The unit is also called pascal and is denoted by the symbol Pa.

Another common SI unit of pressure is bar which is equivalent to a pressure of 10^5 N/m^2 or 0.1 MN/m^2 or 100 kN/m^2 . Its conversion to MKS and FPS units is as follows

$$\begin{aligned} 1\text{ bar} &= \frac{10^5 / 9.80665\text{ kgf}}{10^4\text{ cm}^2} = 1.0197\text{ kgf/cm}^2 \text{ or ata} \\ &= \frac{1.02(2.54)^2}{0.453592} = 14.5\text{ lbf/in}^2 \end{aligned}$$

It can be seen that one *standard atmosphere* is given by

$$\begin{aligned} 1 \text{ atm} &= 1.033 \text{ kgf/cm}^2 = 14.696 \text{ lbf/in}^2 \\ &= \frac{1.033}{1.0197} = 1.01325 \text{ bar} \\ &= 760 \text{ mm Hg or 760 torr} \end{aligned}$$

Accordingly,

$$1 \text{ torr} = 1 \text{ mm Hg} = \frac{1}{760} \text{ atm} = 133 \text{ N/m}^2$$

The conversion of one *technical atmosphere*, i.e. ata is obtained as:

$$\begin{aligned} 1 \text{ ata} &= 1 \text{ kgf/cm}^2 = (9.80665) (10^4) = 980665 \text{ N/m}^2 \\ &= 0.980665 \text{ bar} \\ &= (0.980665) (14.5) = 14.22 \text{ lbf/in}^2 \\ &= \frac{980665}{133} = 736 \text{ torr or mm Hg} \end{aligned}$$

The conversion of other units of pressure are

$$\begin{aligned} 1 \text{ cm H}_2\text{O} &= \left(\frac{10^4 \times 1}{1000} \right) \text{ kg} \left(9.80665 \frac{\text{m}}{\text{s}^2} \right) = 98.1 \text{ N/m}^2 \\ 1 \text{ in Hg} &= (25.4 \text{ mm}) \left(133 \frac{\text{N/m}^2}{\text{mm Hg}} \right) = 3390 \text{ N/m}^2 \end{aligned}$$

1.2.3 Unit of Energy (Work and Heat)

The unit of work or energy is obtained from the product of force and distance moved. The SI unit of work is *Newton metre* denoted by Nm or Joule denoted by J. Thus

$$1 \text{ Nm} = 1 \text{ J} = (1 \text{ kg m/s}^2) (1 \text{ m}) = 1 \text{ kg m}^2/\text{s}^2$$

Since both heat and work are energy, the SI unit of heat is the same as the unit of work, viz., joule. The conversion of the MKS unit of heat, viz., kcal, is obtained from its mechanical equivalent of heat which is 427 kcal/kgfm. Thus:

$$\begin{aligned} 1 \text{ kcal} &= 427 \text{ kgf m} = (427) (9.80665 \text{ N})\text{m} \\ &= 4186.8 \text{ Nm or J} = 4.1868 \text{ kJ} \end{aligned}$$

Also

$$\begin{aligned} 1 \text{ kcal} &= (1 \text{ kg of water}) (1^\circ\text{C}) \\ &= \frac{1}{0.453} \times \frac{9}{5} \text{ lb } ^\circ\text{F} = 3.968 \text{ Btu} \end{aligned}$$

Hence

$$\begin{aligned} 1 \text{ kcal} &= 4.1868 \text{ kJ} = 3.968 \text{ Btu} \\ 1 \text{ kJ} &= 0.948 \text{ Btu} = 0.239 \text{ kcal} \\ 1 \text{ Btu} &= 0.252 \text{ kcal} = 1.055 \text{ kJ} \end{aligned}$$

1.2.4 Unit of Power

The SI unit of power is *watt*, denoted by the symbol W. It is defined as the rate of doing 1 Nm of work per second. Thus

$$1 \text{ W} = 1 \text{ J/S} = 1 \text{ Nm/s}$$

It may also be noted that watt also represents the electrical unit of work defined by

$$1 \text{ W} = 1(\text{volt}) \times 1(\text{ampere}) = 1 \text{ J/s}$$

The conversion of the horsepower unit can also be obtained

$$1 \text{ hp (imperial)} = 550 \frac{\text{ft.1bf}}{\text{s}} = \frac{(550 \times 0.3048 \text{ m})(0.453592 \times 9.80665 \text{ N})}{\text{s}}$$

$$= 746 \text{ Nm/s or J/s or W}$$

$$1 \text{ hp(metric)} = 75 \frac{\text{kgf.m}}{\text{s}} = (75 \times 9.80665 \text{ N}) \frac{\text{m}}{\text{s}}$$

$$= 736 \text{ Nm/s or J/s or W}$$

Further, the units of energy can be derived from those of power. Thus

$$1 \text{ J} = 1 \text{ Ws}$$

$$1 \text{ kWh} = 3,600,000 \text{ J} = 3,600 \text{ kJ} = 860 \text{ kcal} = 3,410 \text{ Btu}$$

$$1 \text{ hp/hr} = 746 \times 3,600 \text{ J} = 2,680 \text{ kJ} = 641 \text{ kcal} = 2,540 \text{ Btu}$$

(imperial)

$$1 \text{ hp/hr} = 736 \times 3,600 \text{ J} = 2,650 \text{ kJ} = 632 \text{ kcal} = 2,510 \text{ Btu}$$

(metric)

1.2.5 Unit of Enthalpy

The interconversion of units of enthalpy are as follows

$$1 \text{ kJ/kg} = 0.239 \text{ kcal/kg} = 0.42 \text{ Btu/lb}$$

$$1 \text{ kcal/kg} = 4.19 \text{ kJ/kg} = 1.8 \text{ Btu/lb}$$

$$1 \text{ Btu/lb} = 0.556 \text{ kcal/kg} = 2.33 \text{ kJ/kg}$$

Note The definition of enthalpy (H) (and specific enthalpy (h)) is obtained by the application of the First Law of Thermodynamics to a thermodynamic process.

1.2.6 Units of Entropy and Specific Heat

These are expressed as

$$1 \text{ kJ/kg.K} = 0.239 \text{ kcal/kg}^{\circ}\text{C} \text{ or } \text{Btu/lb}^{\circ}\text{F}$$

$$1 \text{ kcal/kg}^{\circ}\text{C} = 1 \text{ Btu/lb}^{\circ}\text{F} = 4.1868 \text{ kJ/kg.K}$$

Note The definition of entropy (S) (and specific entropy (s)) is obtained by the application of the Second Law of Thermodynamics to a thermodynamic process.

1.2.7 Unit of Refrigerating Capacity

The standard unit of refrigeration in vogue is *ton refrigeration* or simply ton denoted by the symbol TR. It is equivalent to the production of cold at the rate at which heat is to be removed from one US tonne of water at 32°F to freeze it to ice at 32°F in one day or 24 hours. Thus

$$1 \text{ TR} = \frac{1 \times 2,000 \text{ lb} \times 144 \text{ Btu/lb}}{24 \text{ hr}}$$

$$= 12,000 \text{ Btu/hr} = 200 \text{ Btu/min}$$

where the latent heat of fusion of ice has been taken as 144 Btu/lb. The term one ton refrigeration is a carry over from the time ice was used for cooling. In general 1 TR always means 12,000 Btu of heat removal per hour, irrespective of the working substance used and the operating conditions, viz., temperatures of refrigeration and heat rejection. This unit of refrigeration is currently in use in the USA, the UK and India. In many countries, the standard MKS unit of kcal/hr is used.

It can be seen that

$$\begin{aligned} 1 \text{ TR} &= 12,000 \text{ Btu/hr} \\ &= \frac{12,000}{3.968} = 3,024.2 \text{ kcal/hr} \\ &= 50.4 \text{ kcal/min} \approx 50 \text{ kcal/min} \end{aligned}$$

Also, since 1 Btu = 1.055 kJ, the conversion of ton into equivalent SI unit is:

$$\begin{aligned} 1 \text{ TR} &= 12,000 \times 1.055 = 12,660 \text{ kJ/hour} \\ &= 211 \text{ kJ/min} = 3.5167 \text{ kW} \end{aligned}$$

Example 1.1 The performance test of an air conditioning unit rated as 140.7 kW (40 TR) seems to be indicating poor cooling. The test on heat rejection to atmosphere in its condenser shows the following:

Cooling water flow rate: 4 L/s
 Water temperatures: In 30°C: Out 40°C
 Power input to motor: 48 kW (95% efficiency)

Calculate the actual refrigerating capacity of the unit.

Solution Heat rejected in condenser

$$\begin{aligned} \dot{Q}_{\text{condenser}} &= \dot{m}_w C_w \Delta t_w \\ &= 4 (4.1868) (40 - 30) = 167.5 \text{ kW} \end{aligned}$$

Work input

$$\dot{W} = 48 (0.95) = 45.6 \text{ kW}$$

Refrigeration capacity (by energy balance)

$$\begin{aligned} \dot{Q}_{\text{refrigeration}} &= \dot{Q}_{\text{condenser}} - \dot{W} \\ &= 167.5 - 45.6 = 121.9 \text{ kW (34.7 TR)} \end{aligned}$$

The unit is operating below its rated capacity of 40 TR.



1.3 THERMODYNAMIC SYSTEMS, STATE, PROPERTIES, PROCESSES, HEAT AND WORK

Thermodynamic systems are of two types. They are either *closed* or *open* as illustrated in Fig. 1.3. A closed system is one across whose *boundary* only heat Q and work W flow. In an open system the working fluid also crosses the *control surface* drawn around the system. Everything outside the system is *surroundings*. The system plus surroundings combine to make the *universe*.

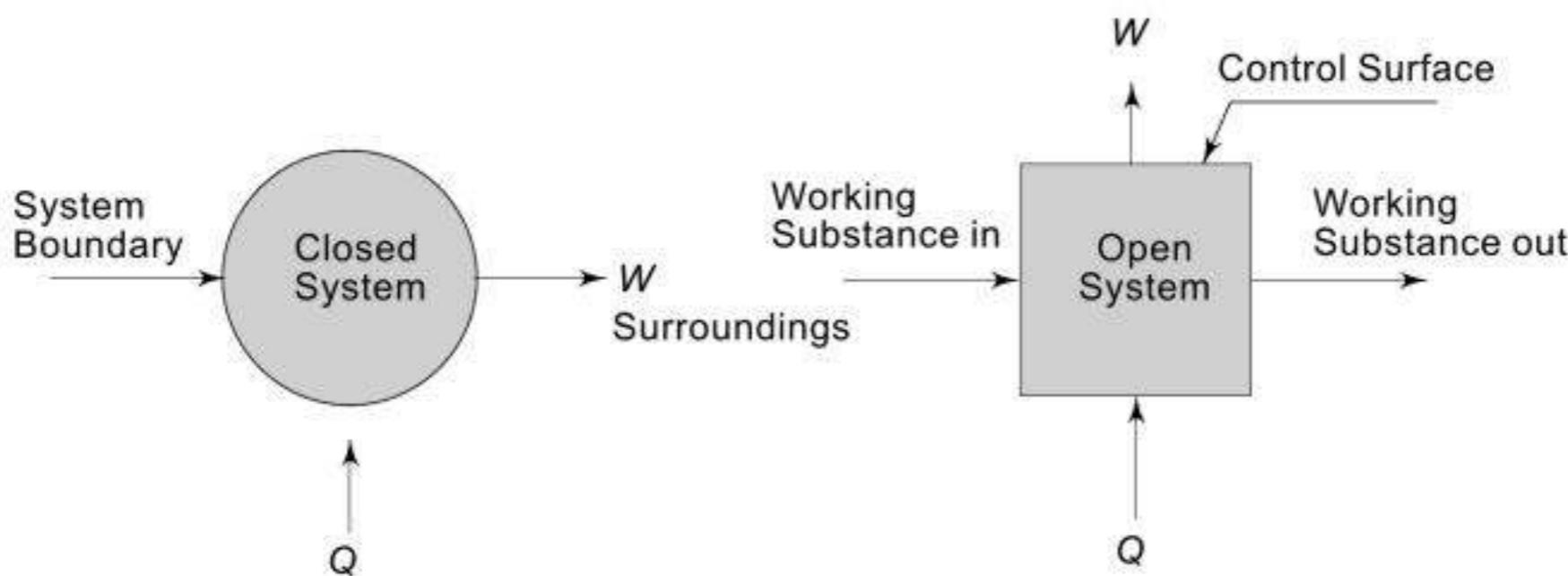


Fig. 1.3 Closed and open systems

The state of a thermodynamic system is characterised by its *properties*. The change of state of the working substance represents a *thermodynamic process*. Thermodynamic processes occurring in a closed system are called *non-flow processes*. Likewise, thermodynamic processes occurring in an open system are called *flow processes*.

Further, the processes that can be reversed such that the system and environment, both, can be restored to the initial state are called *reversible processes*. The processes which, when reversed, will not restore *both* the system and environment to the initial state are called *irreversible processes*.

The properties are either *intensive or extensive*. Intensive properties do not depend on the size of the system. These are, e.g., pressure p and temperature T . The extensive properties depend on the size of the system, e.g., volume V , internal energy U , enthalpy H , entropy S , etc. Their numerical values per unit mass of the working substance are called the *specific properties* denoted by lower case symbols, viz., v , u , h , s , etc. The specific properties are intensive properties.

A *thermodynamic process* is accompanied with *heat* and *work* interactions between the system and the surroundings. The heat added to the system is considered as positive, and that rejected by the system as negative. The sign convention for work is the opposite. The work done by the system is positive and the work done *on* the system is negative.

The heat and work interactions per unit mass of the *working substance* in the system are denoted as q and w .

Note The work done in a reversible process in a simple compressible system is given by

$$W = \int p \, dV$$

Note that in an irreversible process, the work is not given by $\int p \, dV$.



1.4 FIRST LAW OF THERMODYNAMICS

The first law of thermodynamics is mathematically stated as follows:

$$\oint \delta Q = \oint \delta W \quad (1.2)$$

Accordingly, during a *thermodynamic cycle*, viz., a cyclic process the system undergoes, the cyclic integral of heat added is equal to the cyclic integral of work done. Equation (1.2) can also be written for a cycle as

$$\oint(\delta Q - \delta W) = 0$$

Equation (1.3) below is a corollary of the first law. It shows that there exists a property U , named *internal energy* of the system/substance, such that a change in its value is equal to the difference in heat entering and work leaving the system. Accordingly, for a process in a closed system, the first law can be written as:

$$\delta Q = \delta U + \delta W \quad (1.3)$$

For the change of state of a system from initial state 1 to final state 2, this becomes

$$Q = U_2 - U_1 + W$$

Another property named *enthalpy* H can also be defined now as a combination of properties U , p and V ,

$$H = U + pV, h = u + pv$$

For a reversible process, since $\delta W = pdV$, the first law can also be written as

$$\delta Q = dU + pdV, \delta q = du + pdv \quad (1.4a)$$

$$\delta Q = dH - Vdp, \delta q = dh - v dp \quad (1.4b)$$

The first law can be applied to a process in an open system. Figure 1.4 represents an open system undergoing a *steady-state steady-flow (SSSF) process*. For the process, the first law takes the form of a *steady-flow energy equation* as in Eq. (1.5)

$$\begin{aligned} \dot{Q} &= \dot{m}[(u_2 - u_1) + (p_2 v_2 - p_1 v_1) + \frac{1}{2}(C_2^2 - C_1^2) \\ &\quad + g(z_2 - z_1)] + \dot{W} \\ &= \dot{m}[(h_2 - h_1) + \frac{1}{2}(C_2^2 - C_1^2) + g(z_2 - z_1)] + \dot{W} \quad (1.5) \end{aligned}$$

Here, in addition to change in internal energy, changes in kinetic and potential energies are also considered since these are significant. In addition, work, equal to $(p_2 v_2 - p_1 v_1)$, to make the fluid enter and leave the system called the *flow work* is also considered.

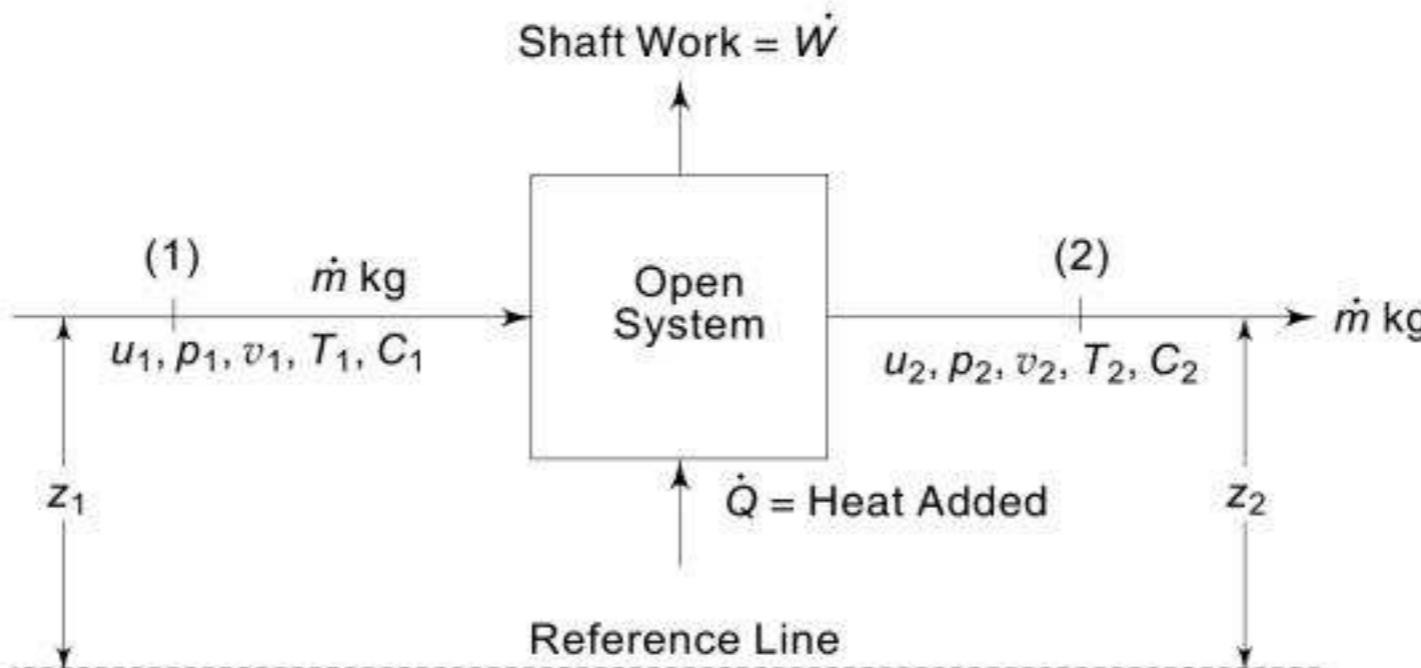


Fig. 1.4 Representation of a steady-state steady-flow process

Writing Eq. (1.5) on the basis of a unit mass entering and leaving the system, we have Eq. (1.6)

$$q + h_1 + \frac{C_1^2}{2} + gz_1 = h_2 + \frac{C_2^2}{2} + gz_2 + w \quad (1.6)$$

1.5 SECOND LAW OF THERMODYNAMICS

The second law of thermodynamics can be mathematically state for a thermodynamic cycle in the form of *Clausius Inequality* as given in Eq. (1.7)

$$\oint \frac{\delta Q}{T} \leq 0 \quad (1.7)$$

The equality holds for a reversible cycle, and the inequality for an irreversible cycle.

Just as the application of first law to a thermodynamic process led to the establishment of a new property, named internal energy (U), the application of the second law to a process leads to the establishment of another new property named *entropy* (S), defined as follows in Eq. (1.8)

$$dS \equiv \left(\frac{\delta Q}{T} \right)_{rev} \quad (1.8)$$

Thus, for a reversible process, between two given states, from initial state 1 to final state 2 in a closed system, or inlet state 1 to exist state 2 in an open system, the change in entropy is given by

$$S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T} \right)_{rev}, s_2 - s_1 = \int_1^2 \left(\frac{\delta q}{T} \right)_{rev}$$

It is found by applying Clausius inequality that for an irreversible process

$$S_2 - S_1 > \int_1^2 \left(\frac{\delta Q}{T} \right), s_2 - s_1 > \int_1^2 \left(\frac{\delta q}{T} \right)$$

Note For a reversible process in a compressible system work done $W = \int_1^2 pdV$. Hence, the area under the curve on P-V diagram gives work done in the process. Similarly, for a reversible process, heat transfer $Q = \int_1^2 TdS$. Hence, the area under the curve on T-S diagram gives heat transfer during the process.

1.6 NON-FLOW PROCESSES

Processes in a closed system are referred to as *non-flow processes*. Since the velocities are small, and hence dissipation due to friction is negligible, most non-flow processes are considered as reversible.

In a *reversible constant volume process*, $W = \int pdV = 0$.

Hence, from first law, $Q = U_2 - U_1$.

In a *reversible constant pressure process*, $W = \int pdV = p(V_2 - V_1)$, and from first law, $Q = (U_2 - U_1) + p(V_2 - V_1) = H_2 - H_1$.

Also, from second law and property relation, $Q = \int TdS$
 $= T(S_2 - S_1) = H_2 - H_1$.

In an *isothermal process*, $Q = T(S_2 - S_1) = (U_2 - U_1) + W$.

In an *adiabatic process*, $Q = 0$ and $W = -(U_2 - U_1)$. In a *reversible adiabatic process*, in addition, we have from second law, $Q = \int TdS = 0$. Hence, $S_2 = S_1$. A reversible adiabatic process is, therefore, an *isentropic process*.

A general process can be represented by the *polytropic relation*

$$p_1 V_1^n = p_2 V_2^n = pV^n = \text{Constant}$$

in which n is the *polytropic index*. For the polytropic process, we have Eq. (1.9) for work

$$\begin{aligned} W &= \int pdV = \frac{n}{1-n} (p_2 V_2 - p_1 V_1) \\ &= \frac{n}{1-n} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right] \end{aligned} \quad (1.9)$$

1.7 STEADY-FLOW PROCESSES

The steady-flow energy equation is applicable to flow processes, viz. processes in an open system. In most flow processes, the irreversibility due to viscous friction cannot be neglected on account of significant velocities encountered. It is, therefore, necessary to calculate the unknown quantity by first assuming the process as reversible, and then multiplying or dividing the result by a *process efficiency*.

Boiling and Condensation

$$q = -h_2 - h_1 + \frac{1}{2}(C_2^2 - C_1^2) \equiv h_2 - h_1$$

as the velocities are small, and the change in kinetic energy can be neglected. If there is a significant pressure drop as in *direct-expansion evaporators* in refrigeration, the process is not completely reversible.

Isothermal Process Neglecting kinetic and potential energies

$$q - w = h_2 - h_1$$

Adiabatic Process In nozzles and diffusers, there is no heat transfer or work done. In these devices, there is interconversion between kinetic energy and enthalpy. Thus we have

$$\frac{1}{2}(C_2^2 - C_1^2) = h_1 - h_2$$

In turbines and compressor, assuming no heat transfer and neglecting changes in kinetic and potential energies, we obtain for work

$$w = h_1 - h_2$$

To account for irreversibility in processes, the efficiencies are defined as follows:

Nozzle and Turbine Efficiency

$$\eta = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

Diffuser and Compressor Efficiency

$$\eta = \frac{h_1 - h_{2s}}{h_1 - h_2}$$

In these equations, h_2 represents the enthalpy at the end in the actual process and h_{2s} represents the enthalpy if the process is executed reversibly, viz., isentropically between the pressure limits p_1 and p_2 .

Throttling Process It is an irreversible adiabatic process. It is employed to reduce the pressure of a fluid by introducing a restriction in the flow passage as illustrated in Fig. 1.5.

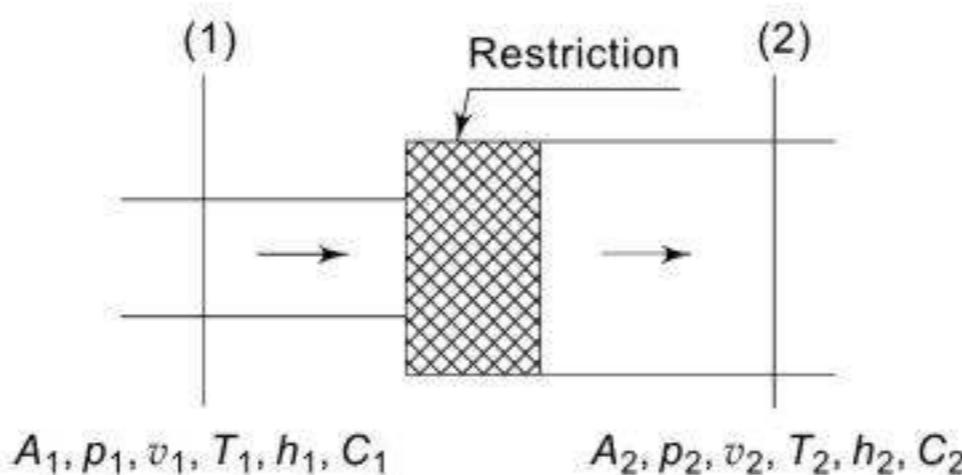


Fig. 1.5 Representation of a throttling process

As the process is adiabatic and no external work is done, we have

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2}$$

Since $p_2 < p_1$, $V_2 > V_1$. Hence if $A_1 = A_2$, $C_2 > C_1$. Making $A_2 > A_1$ such that $C_2 = C_1$, or since kinetic energy change is negligible, we find that in a throttling process

$$h_1 = h_2$$

Accordingly, throttling is also referred to as an *isenthalpic-expansion process*.



1.8 THERMODYNAMIC STATE OF A PURE SUBSTANCE

The working substance used in refrigerating machines is called a *refrigerant*. A refrigerant is usually a pure substance, though research is on to use mixtures also.

A knowledge of two independent properties is required to determine the thermodynamic state of a pure substance. In the case of equilibrium between two phases of a pure substance, only one independent property is required to define the state.

The equilibrium between phases, and the state of a pure substance are best described by the *phase diagrams* such as the $p - v$ and $T - s$ diagrams shown in Figs 1.7(a) and 1.7(b). The continuous lines on these diagrams represent the locii of equilibrium/saturation states in which phase change occurs, e.g., s and f_1 between solid and liquid, and f_2 and g between liquid and vapour. The temperature for change from solid state s to liquid state f_1 or vice versa, is the *melting/freezing/fusion temperature*. The temperature for change from liquid state f_2 to vapour state g , and vice versa, is the *boiling/condensation temperature*. Similarly the temperature for change between solid state s and vapour state g is the *sublimation/ablimation temperature*.

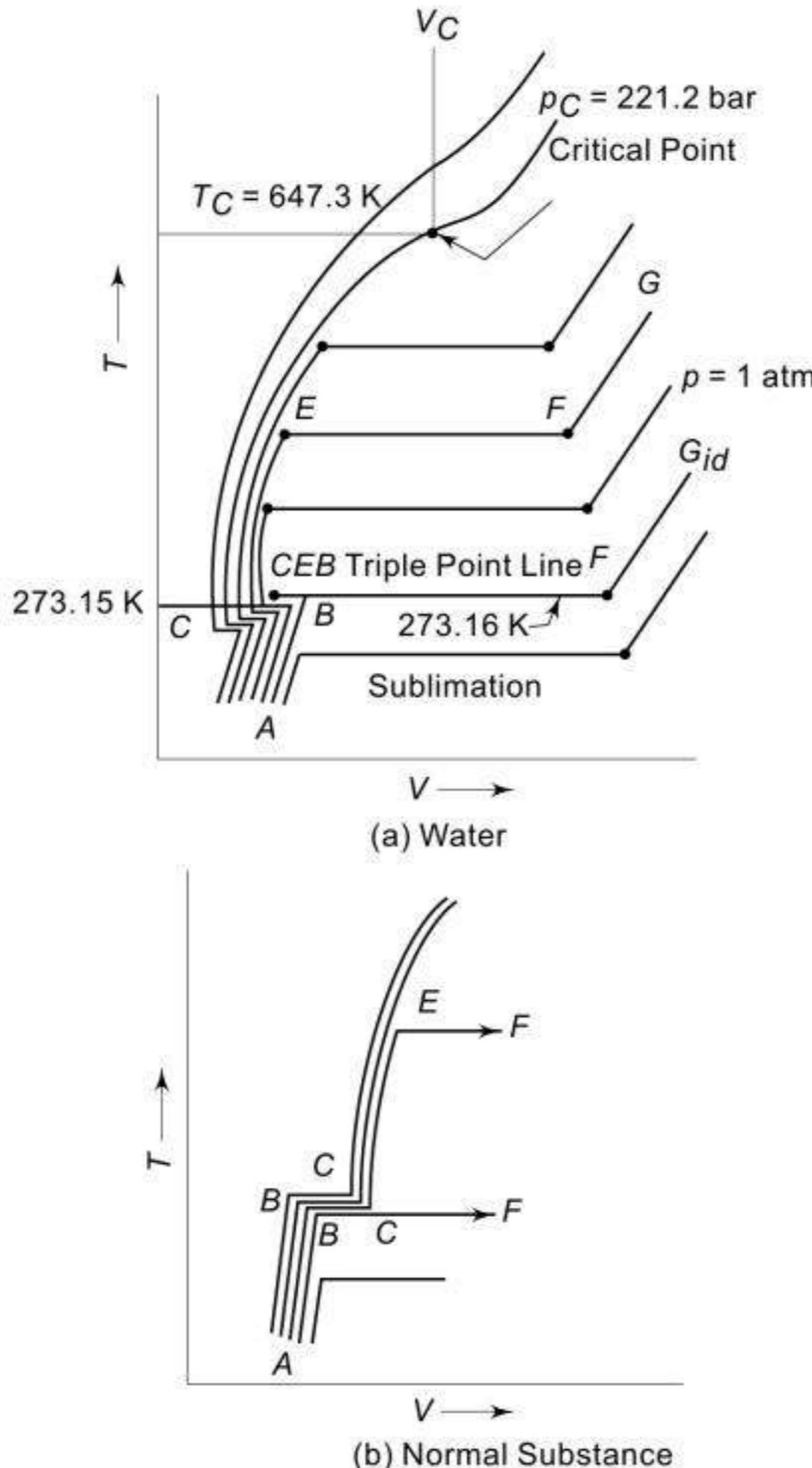


Fig. 1.6 Temperature-specific volume phase diagrams for water and normal substances

Consider a unit mass of solid at S below its melting point at 1 atm pressure. Let heat be supplied to it, and let us follow the events that occur at constant pressure as shown in Fig. 1.6(a) for water and Fig. 1.6(b) for a normal substance, Figs. 1.7(a) and 1.7(b) are for a normal substance. We observe the following from Fig. 1.7.

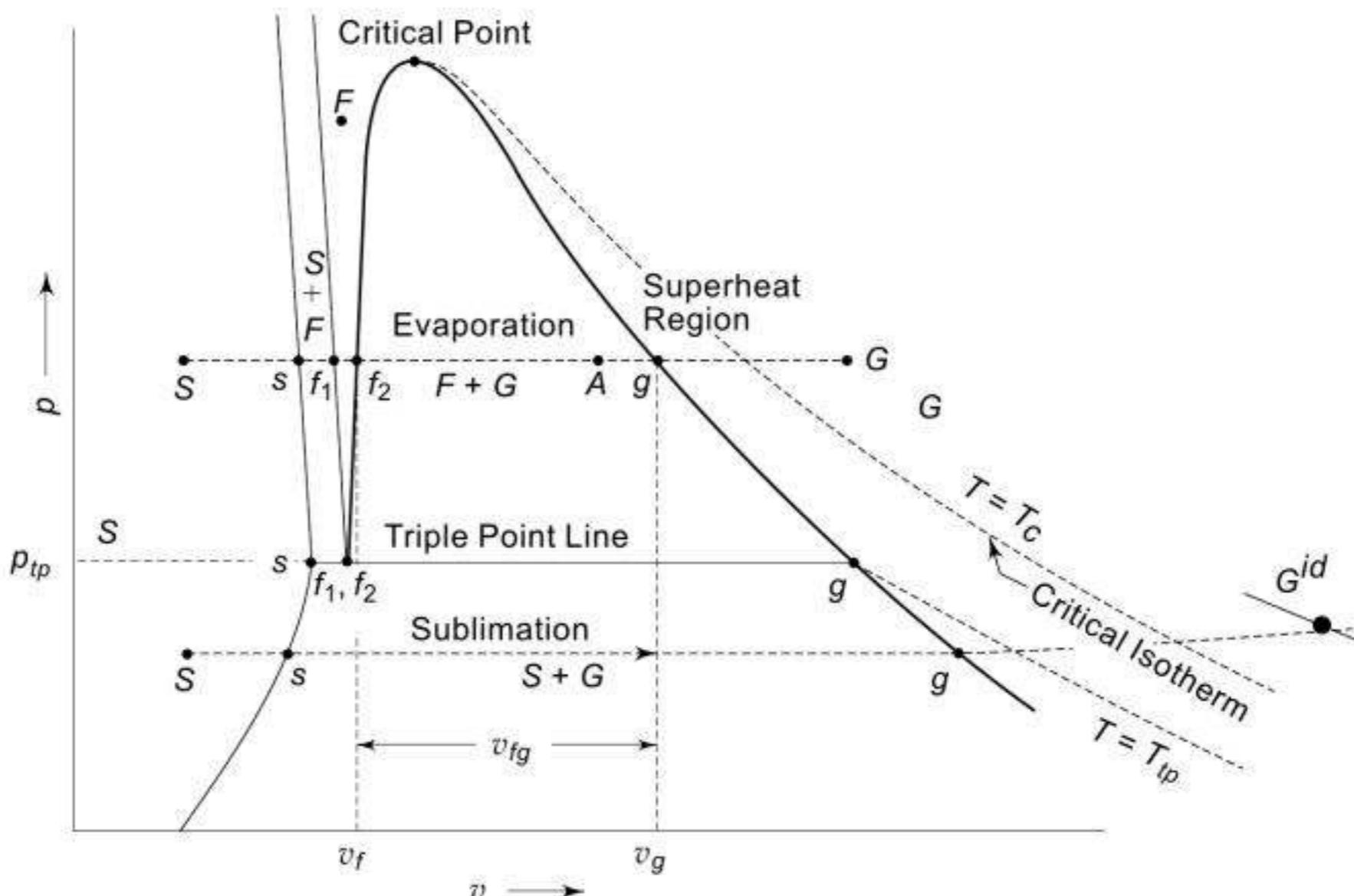


Fig. 1.7(a) p-v phase Diagram for a Normal Substance

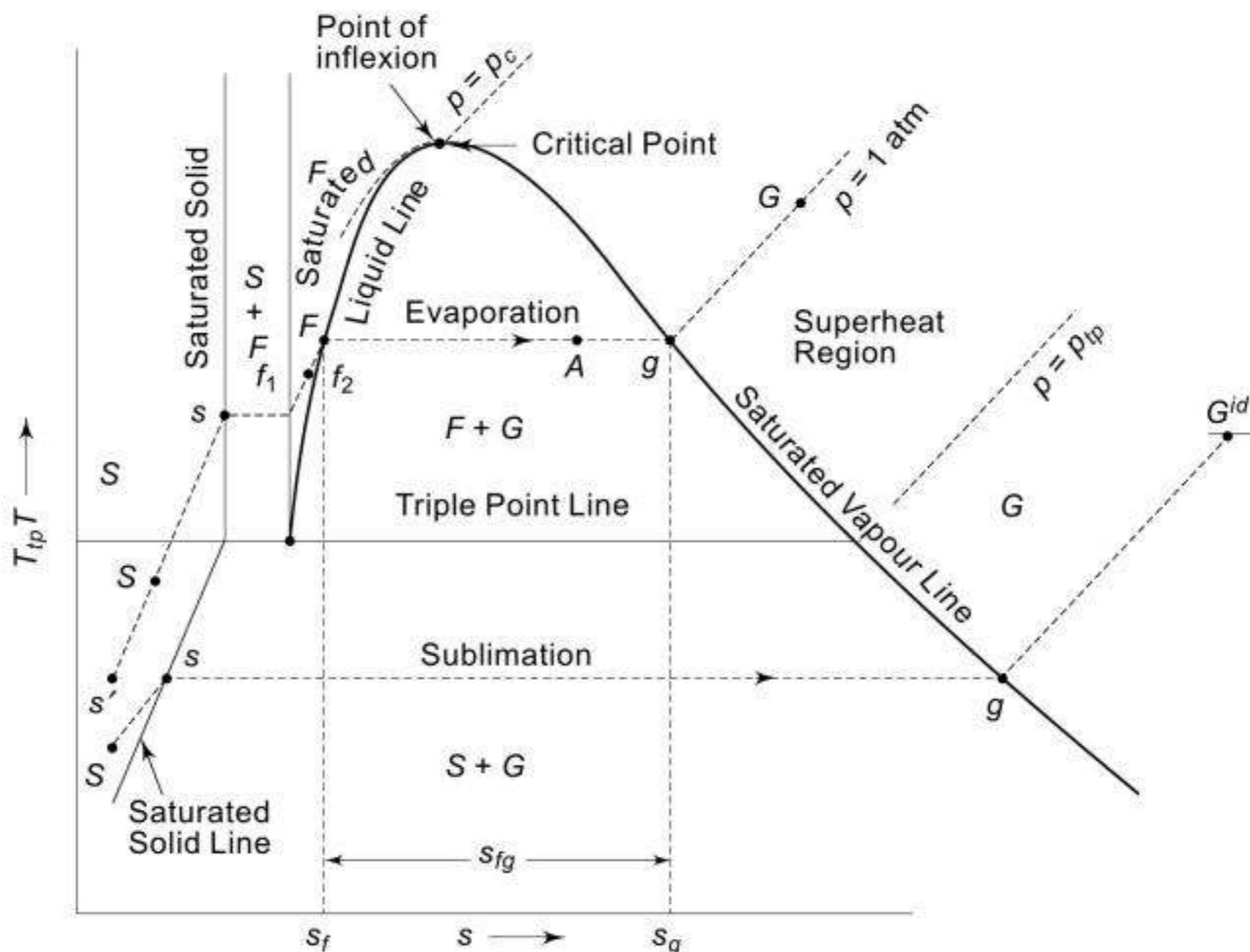


Fig. 1.7(b) T-s phase diagram for a normal substance

- Temperature rises until point s is reached.
- Further heating results in melting of solid at constant (melting point) temperature until point f_1 is reached.
- Change of phase from solid to liquid is complete at f_1 . Heating of liquid now results in rise in temperature until point f_2 is reached.

- (iv) Further heating at f_2 results in vaporization of liquid at constant (boiling point) temperature until point g is reached.
- (v) Change of phase from liquid to vapour is complete at g . Heating of vapour/gas at g results in rise in temperature again until, say, point G above the boiling point temperature is reached.

States, f_1 , f_2 and g are saturation states. f_1 represents saturated liquid state in equilibrium with saturated solid state s , and f_2 represents saturated liquid state in equilibrium with saturated vapour stage g . Now onwards, the subscripts 1 and 2 from f_1 and f_2 will be dropped, and the context will tell which one we are referring to.

Note that there is a large change in volume equal to $v_{fg} = v_g - v_f$ during vaporization from f_2 to g . Similarly, we have latent heat of vaporization $h_{fg} = h_g - h_f$, and entropy of vaporization $s_{fg} = s_g - s_f$.

State S below the melting point temperature $T_s = T_{f_1}$ is a *subcooled solid state*. State G above the boiling point temperature $T_{f_2} = T_g$ is a *superheated vapour state*. Any liquid state F between f_1 and f_2 below the boiling point temperature is a *subcooled liquid state*. Any state A in the liquid plus vapour ($F + G$) region represents a mixture of vapour at g and liquid at f_2 . The position of A is governed by the quality or dryness fraction x of vapour and $(1 - x)$ of liquid.

Consider now the reverse process of cooling of superheated vapour at G . The processes followed will be *desuperheating* from G to g , *condensation* from g to f_2 , *subcooling* from f_2 to f_1 , *freezing* from f_1 to s , and subcooling from s to S .

Consider now that the heating of solid is carried out at lower pressures. As the pressure is lowered, there is a marked decrease in boiling point temperature, and an increase in volume and enthalpy accompanying vaporization. This continues until *triple point pressure* p_{tp} and *temperature* T_{tp} are reached. On the *triple point line*, all the saturation states s , f_1 , f_2 and g lie. Thus, all the three phases exist in equilibrium at the triple point. The triple point for water is at 273.16 K (0.01°C) temperature, and 0.006112 bar pressure.

Below the triple point pressure, saturated solid at s on heating directly changes into saturated vapour at g . This is called *sublimation*. Similarly, saturated vapour at g on cooling directly changes to saturated solid at s . This is called *ablimation* or *freeze-condensation*.

Consider now the heating at pressures above atmospheric. In general, at higher pressures, the boiling point increases, and there is a marked decrease in v_{fg} , h_{fg} and s_{fg} . At a certain high pressure, v_{fg} , h_{fg} and s_{fg} become zero. This is referred to as the *critical point*. The properties at this point are denoted as *critical pressure* p_c , *critical temperature* T_c and *critical volume* v_c . The same for water are:

$$p_c = 221.2 \text{ bar}, T_c = 647.3 \text{ K}, v_c = 0.00317 \text{ m}^3/\text{kg}$$

At pressures above critical, there is no definite transition between liquid and vapour phases.

Note that the gaseous state G^{id} at low pressures (tending to zero) and high temperatures represents the *ideal* or *perfect gas* state following the equation $pv = RT$.

1.9 HEAT EXCHANGE PROCESSES

Heat is normally absorbed or rejected by a working substance at a constant pressure. It has been seen that when a working substance exchanges heat, then either

- (i) the temperature of the substance changes and the substance remains in a single phase, or
- (ii) the temperature of the substance remains constant but a phase change occurs.

The heat transferred without a phase change results in a temperature rise and is called *sensible heat*, and that transferred resulting in a change of phase at constant temperature is called *latent heat*.

Sensible heat is measured by the expression

$$Q_s = mC_p \Delta T$$

where m is the mass, C_p is the specific heat at constant pressure, and ΔT is the temperature change of the working substance. In general, chilled water or salt solutions of either NaCl or CaCl₂ called *brines* are used as carriers of refrigeration for the absorption of heat. Chilled water is used for air conditioning in central air-conditioning plants. Brines have freezing points lower than 0°C and are, therefore, used as coolants in applications below 0°C refrigeration temperature, such as in cold storages, ice plants, skating rinks, etc.

Latent heat exchange processes correspond to those of melting, evaporation and sublimation, and vice versa, viz., fusion, condensation and desublimation and the heat transferred is measured by

$$Q_L = m\Delta h$$

where Δh is the latent heat for the corresponding process, e.g., h_{fg} for latent heat of vaporization, h_{sg} for sublimation and h_{sf} for fusion. The subscripts s , f and g denote, solid, liquid and gas respectively. Evaporation is the most commonly used method in refrigeration for the absorption of heat. Sublimation is used in a process called *freeze drying*. In this, the product is first frozen and cooled until it reaches below its triple point temperature. It is then placed in a chamber which is evacuated to a pressure sufficiently below the triple point pressure. Heat is then supplied to the product and in a *freeze-condenser* the sublimated vapours are condensed at a temperature of about -40°C. The food thus dried generally retains its original flavour and value and can be reconstituted into its original condition by the addition of water. Freeze drying is used in the manufacture of certain high-value food products and medicines.

Wide use is made of the melting of ice in homes, stores, transports, etc., for the preservation of products at refrigerated temperatures. The main drawbacks of ice are the temperature limitation of 0°C and the spoilage caused by the melted water. These drawbacks can be avoided by the use of solid carbon-dioxide, also called *dry ice*, which sublimates at atmospheric pressure. Dry ice is rather widely used in refrigerated air transport. In recent times, liquid nitrogen has also become quite popular in transport refrigeration.

Example 1.2 100 kg of ice at -5°C is placed in a bunker to cool some vegetables. 24 hours later, the ice has melted into water at 10°C. What is the average rate of cooling in kJ/h and TR provided by the ice? Given:

Specific heat of ice, $C_{ps} = 1.94 \text{ kJ/kg.K}$

Specific heat of water, $C_{pf} = 4.1868 \text{ kJ/kg.K}$

Latent heat of fusion of ice at 0°C, $h_{sf} = 335 \text{ kJ/kg}$.

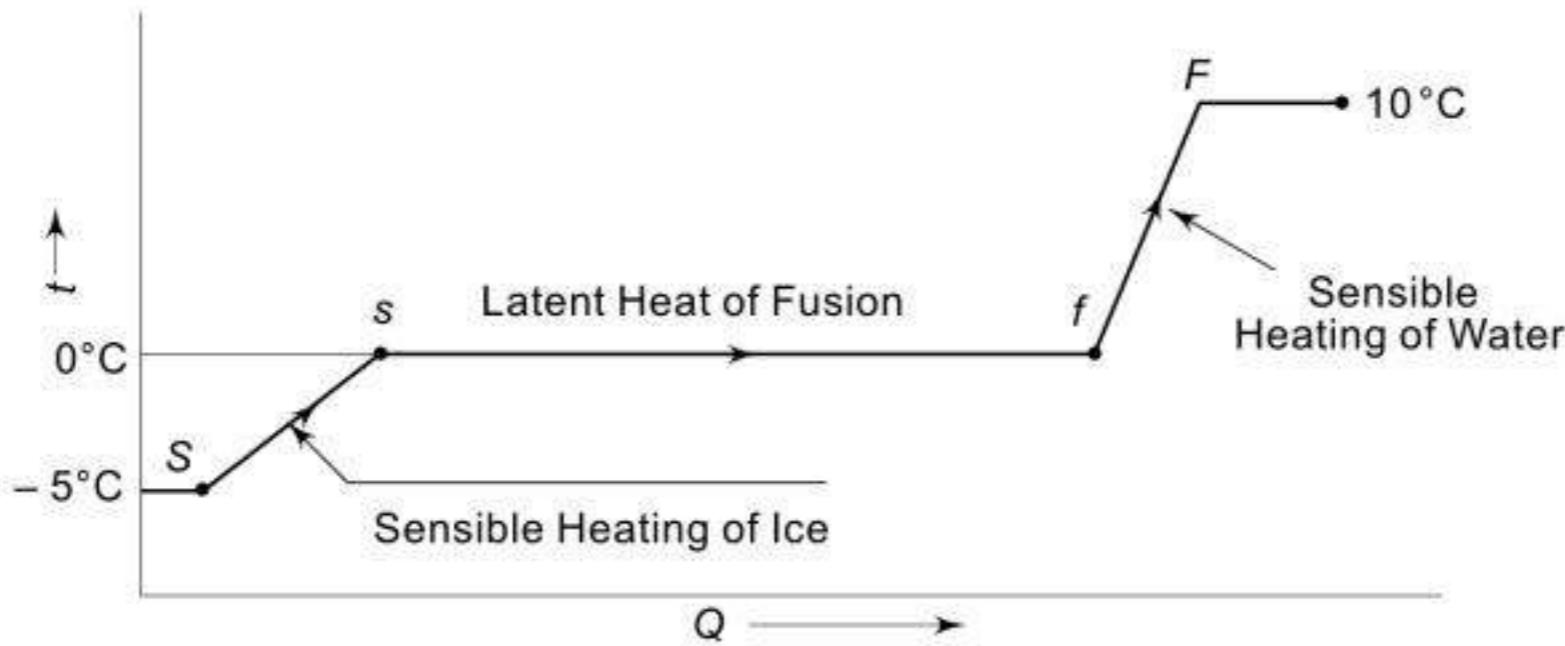


Fig. 1.8 Figure for Example 1.2

Solution The change of temperature with heat supplied is shown in Fig. 1.8.

$$\begin{aligned} Q &= m [C_{ps}(t_s - t_s) + h_{sf} + C_{pf}(t_F - t_f)] \\ &= 100 [1.94(0 + 5) + 355 + 4.1868(10 - 0)] = 38,660 \text{ kJ} \end{aligned}$$

$$\begin{aligned} \text{Capacity} &= \frac{38,660}{24} = 1611 \text{ kJ/h} \\ &= \frac{1611}{12,660} = 0.127 \text{ TR} \end{aligned}$$

1.10 PRODUCTION OF LOW TEMPERATURES

The various principles and processes involved in the production of low temperatures are as follows:

- (i) Throttling expansion of a liquid with flashing.
- (ii) Reversible adiabatic expansion of a gas.
- (iii) Irreversible adiabatic expansion (throttling) of a real gas.
- (iv) Thermoelectric cooling.
- (v) Adiabatic demagnetization.

1.10.1 Expansion of a Liquid with Flashing

Consider the throttling of a saturated liquid initially at 1 at pressure p_1 as shown in Fig. 1.9. The state after expansion is at 2. The process is accompanied by an increase in entropy along with a drop in pressure due to which the volume increases and a part of the liquid is vaporized, thus cooling the remaining liquid. The expansion, with the vaporization of a part of the liquid causing the lowering of its temperature, is called *flashing*. This method is most commonly used for obtaining low temperatures. It will be seen that for the process

$$h_1 = h_2 = h_{f_2} + x_2 h_{fg2} \quad v_2 = v_{f_2} + x_2 v_{fg2}$$

It may be noted that the temperature will similarly drop from T_1 to T_2 if the liquid had isentropically expanded from 1 to 2'. But the isentropic expansion of a liquid is not employed in refrigeration for reasons explained in Sec. 3.1.

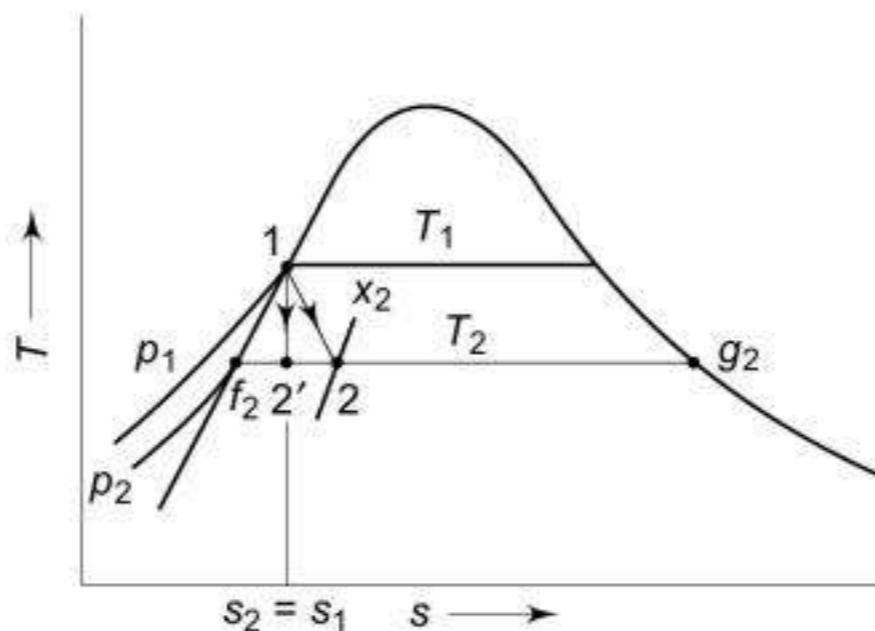


Fig. 1.9 Reversible and irreversible adiabatic expansion of a liquid

1.10.2 Reversible Adiabatic Expansion of a Perfect/Ideal Gas

This method is used with permanent gases such as air. The gas is initially compressed and cooled and then expanded reversibly in an adiabatic reciprocating or turbo-type expander, thus doing external work. The reversible adiabatic expansion of an ideal gas from p_1 to p_2 follows the path $pV^\gamma = C$. The work done is given by the shaded area shown in Fig. 1.10. The expression for work is obtained by adding the flow work components to the non-flow work. Thus, we have for specific work.

$$w = \oint p dv = \int_a^1 p dv + \int_1^2 p dv + \int_2^b p dv + \int_b^a p dv = - \oint \frac{1}{v} v dp$$

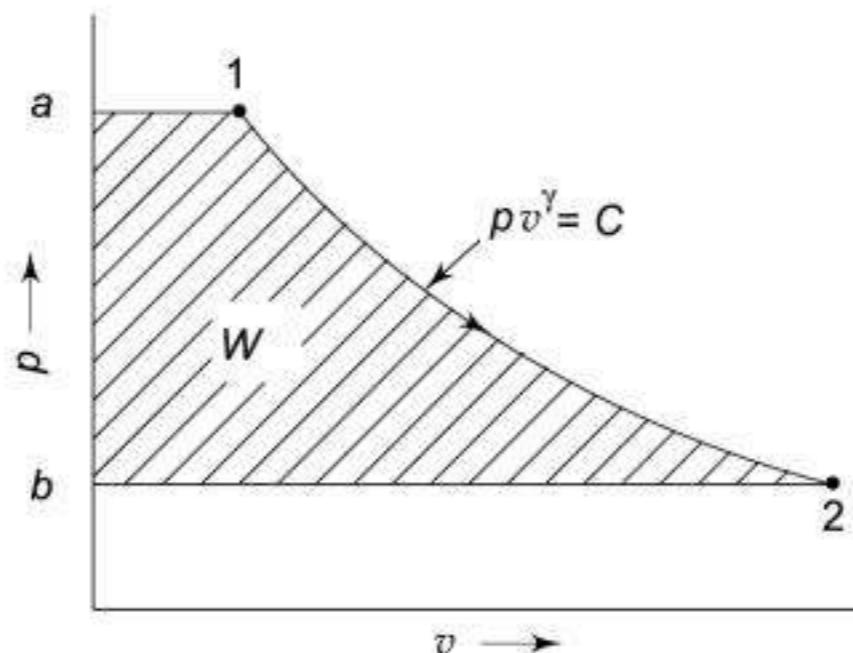


Fig. 1.10 Work done in a reversible adiabatic-expansion process

which gives on integration for a perfect gas

$$w = - \int v_1 \left(\frac{p_1}{p} \right)^{1/\gamma} dp = \frac{\gamma}{\gamma - 1} (p_1 v_1 - p_2 v_2)$$

Also since $q = 0$

we obtain from the steady-state steady-flow energy equation

$$h_1 - h_2 = w = - \int v dp = \frac{\gamma}{\gamma - 1} (p_1 v_1 - p_2 v_2)$$

which for an ideal gas, on putting $\frac{\gamma R}{\gamma - 1} = C_p$, becomes

$$w = h_1 - h_2 = C_p (T_1 - T_2)$$

The temperature after expansion is given by

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}}$$

1.10.3 Irreversible Adiabatic Expansion (Throttling) of a Real Gas

Herein again, the initially compressed and cooled gas is expanded, but irreversibly, through a throttle device with restriction to flow. It has been shown that for this process

$$h_1 = h_2$$

and for a perfect gas $T_1 = T_2$, since enthalpy is a function of temperature only.

Real gases, however, show a departure from ideal gas behaviour and usually produce a substantial decrease in temperature under certain conditions, viz., initial low temperature and high pressure of the gas. The decrease in temperature dT corresponding to a drop in pressure dp , defined by

$$\left(\frac{\partial T}{\partial p} \right)_h = \delta \quad \text{or} \quad \mu_J$$

is known as the *Joule-Thomson coefficient*. For an ideal gas this coefficient is zero. For a real gas, we have:

For cooling	δ is positive	Initial state of gas at high pressure and low temperature
For no change in temperature	$\delta = 0$	Ideal gas behaviour
For heating	δ is negative	Initial state of gas at high temperature.

1.10.4 Thermoelectric Cooling

In 1834, Jean Peltier, a French watchmaker and an amateur scientist, discovered that cooling is produced at one junction of two dissimilar metals, if a current is passed through them. Simultaneously, heat is produced at the other junction (Fig. 1.11). The Peltier heats absorbed at the cold end and rejected at the hot end are given by

$$Q_c = \Pi_c I$$

$$Q_h = \Pi_h I$$

where Π_c and Π_h are the *Peltier coefficients* which are functions of temperatures T_c and T_h of the cold and hot ends respectively. It may be seen that if T_h is maintained at ambient temperature, T_c will be lower than the ambient temperature. It is also to be noted that which of the junctions or ends will become cold or hot depends on the direction of flow of the current.

The phenomenon is called the *Peltier effect*. The actual cooling produced at the cold end would, however, be reduced due to irreversible processes of Joule and Fourier heat transfers. The energy balance of the system is

$$\dot{Q}_k - \dot{Q}_o = EI$$

where I is the current, E is the emf applied and \dot{Q}_o and \dot{Q}_k are the actual amounts of heat flows at the cold and hot ends respectively.

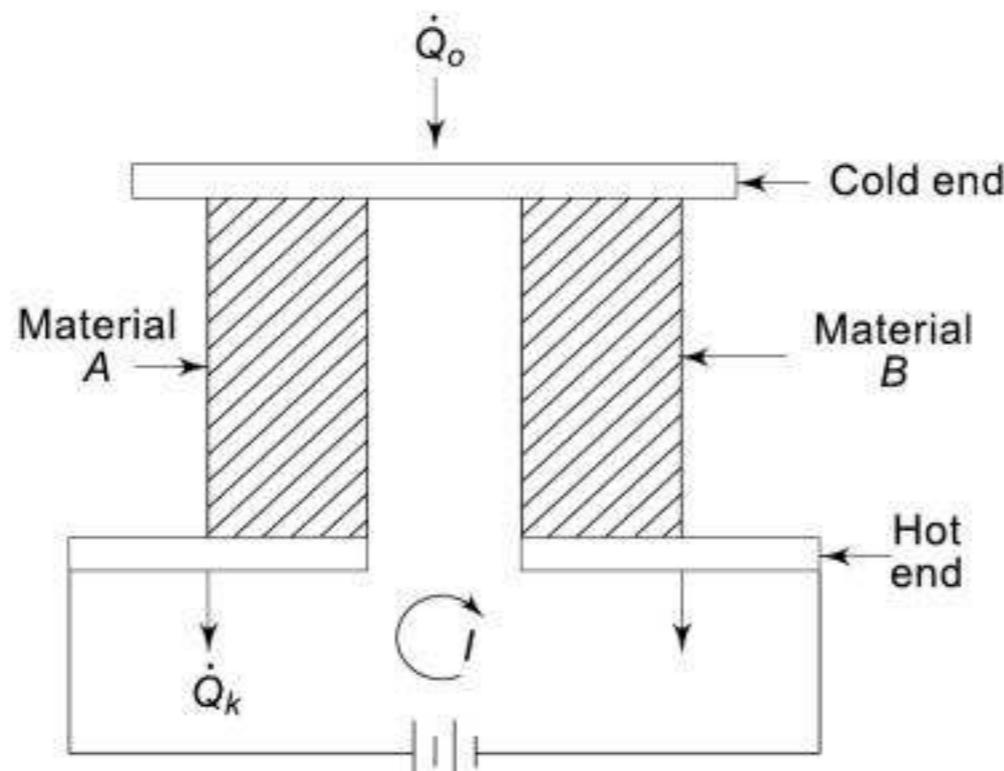


Fig. 1.11 Peltier effect

1.10.5 Adiabatic Demagnetization

In 1926, Giaque and Deby independently proposed that cooling could be produced by the adiabatic demagnetization of a paramagnetic salt. Figure 1.12 provides a schematic representation of such a process.

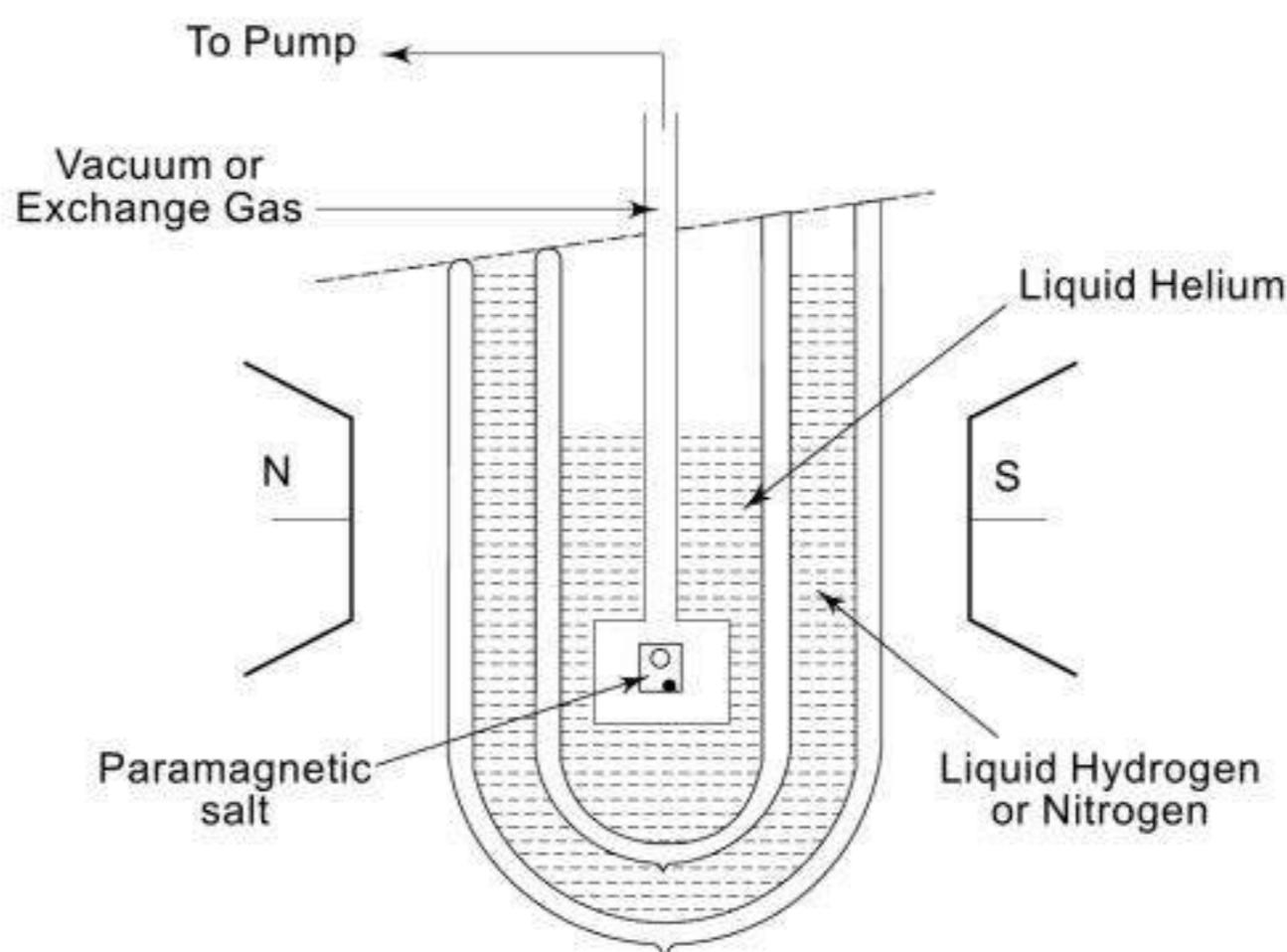


Fig. 1.12 Arrangement for adiabatic demagnetization of a paramagnetic salt

The paramagnetic salt is suspended in a tube containing low pressure gaseous helium as exchange gas to provide thermal communication with the surrounding bath of liquid helium. The liquid helium bath and the salt are first cooled to about 1 K by pumping helium to the lowest practical pressure. A magnetic field is applied, causing magnetization of the salt. The heat produced is removed by the helium bath such that the temperature of the bath again approaches 1 K. Next, the exchange gas is removed by pumping and the magnetic field is turned off. The temperature of the salt then decreases as a result of *adiabatic demagnetization*.

1.11 SATURATION PRESSURE VERSUS SATURATION TEMPERATURE RELATIONSHIP

The whole matter of phase equilibrium is best summarised in phase diagram in Fig. 1.13 between saturation pressure p^{sat} and saturation temperature T^{sat} of a substance. Any point on a line in this diagram represents two phases in equilibrium state such as s and f_1 on *fusion line*, f_2 and g on *vaporization line* and s and g on *sublimation line*. The triple point is one unique point where all saturation states s, f_1, f_2 and g conjoin. The vaporization line begins at the triple point, and ends at the critical point. It is called the *vapour pressure curve*.

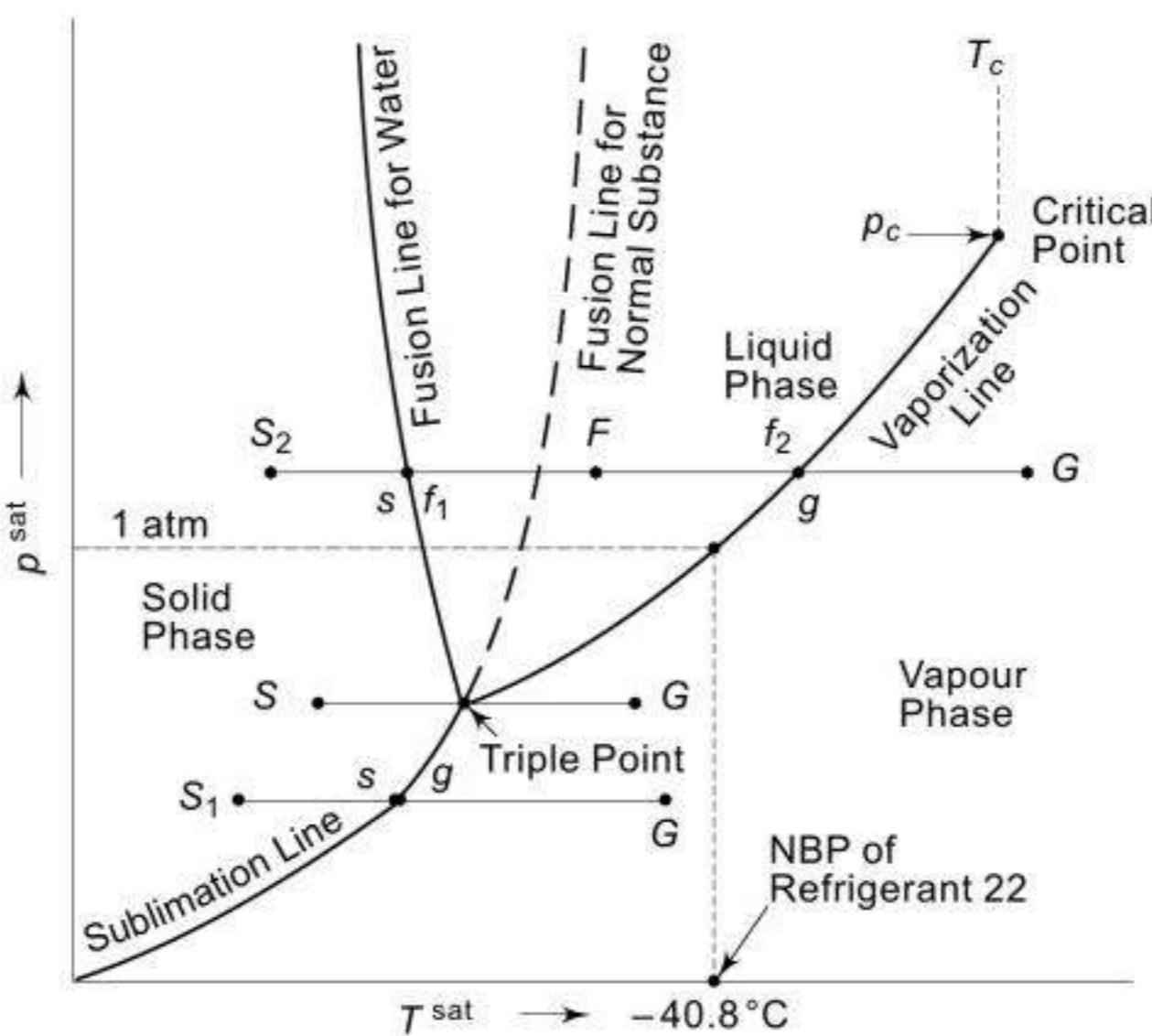


Fig. 1.13 Saturation pressure versus saturation temperature phase diagram of a pure substance

The saturation temperature at 1 atm pressure is the *normal boiling point* of the substance. The figure shows the N.B.P. of one of the most commonly used refrigerants, viz., Refrigerant 22 (CHClF_2) as -40.8°C . Equation recommended by Reynolds¹⁴ represents the saturation pressure versus saturation temperature data of Refrigerant 22 very accurately.

$$\ln p^{\text{sat}} = 71.55415 - \frac{4,818.96}{T^{\text{sat}}} - 7.861 \ln T^{\text{sat}} + 9.0807 \times 10^{-3} T^{\text{sat}} \\ + 0.445747 \frac{(381.17 - T^{\text{sat}})}{T^{\text{sat}}} \ln (381.17 - T^{\text{sat}}) \quad (1.10)$$

Herein, the pressure is in Pa.

Example 1.3 Find saturation pressure of Refrigerant 22 at 40°C (313 K).

Solution Substituting values in Eq. (1.10), we obtain

$$\begin{aligned}\ln p^{\text{sat}} &= 71.55415 - 15.396 - 45.1711 + 2.8423 + 0.4099 = 14.236 \\ \Rightarrow p^{\text{sat}} &= 1,527,660 \text{ Pa}\end{aligned}$$

Note $\ln p^{\text{sat}}$ versus $1/T^{\text{sat}}$ relationship can be simplified; and can be expressed in the form of Antoine equation as follows in Eq. (1.11)

$$\ln p^{\text{sat}} = A - \frac{B}{T^{\text{sat}}} \quad (1.11)$$

which is a straight line, and is quite accurate.

Example 1.4 Using the vapour pressure data of Freon 22 and propane at 40°C and -30°C given below, find the values of the constants of the Antoine equation for the two refrigerants, and verify their validity for pressure at 5°C .

t^{sat} $^\circ\text{C}$	T^{sat} K	p^{sat}	
		Freon 22 Pa	Propane Pa
40	313.15	1,533,500	1,366,400
5	278.15	584,000	547,750
-30	243.15	163,500	166,400

Solution Substituting values and solving, we get the Antoine equations:

For Freon 22

$$\ln p^{\text{sat}} = 22.01864 - \frac{2.43492 \times 10^3}{T^{\text{sat}}}$$

Putting $T^{\text{sat}} = 278.15$ K, we get

$$p^{\text{sat}} = 5.765 \times 10^5 \text{ Pa}$$

The error in comparison to actual value of 5.84×10^5 Pa is 1.28% only.

For Propane

$$\ln p^{\text{sat}} = 21.50251 - \frac{2.309424 \times 10^3}{T^{\text{sat}}}$$

At $T^{\text{sat}} = 278.15$ K, $p^{\text{sat}} = 5.402 \times 10^5$ Pa

The error in comparison to actual value of 547,750 Pa is 1.38% only.

Note If a number of data points are used, and least squares method is employed for finding A and B, the accuracy would still be better.

1.12 THE GASEOUS PHASE: EQUATION OF STATE

Pressure p , specific volume v and temperature T are the three measurable properties of a substance. The equation expressing their relationship in the gaseous phase is

called the *equation of state*. This functional relationship between p - v - T of a gas can be either theoretical, or generalized or an empirical equation fitted from experimental data.

The simplest theoretical equation of state is the *ideal* or *perfect gas* equation representing behaviour of a gas at low pressures (*tending to zero*) and high temperatures such as at point G^{id} in Figs 1.6 and 1.7. This equation is

$$v^{id} = \frac{RT}{p}$$

where $R = \bar{R}/M$ is the *gas constant* for the particular gas, \bar{R} is the *universal gas constant* having the value 8.3143 kJ/kmol K, and M is the molecular mass of the substance.

In addition, the internal energy and enthalpy of an ideal gas are functions of temperature only. These are given by,

$$\begin{aligned} du &= C_{v_o} dT, u_2 - u_1 = C_{v_o} (T_2 - T_1) \\ dh &= C_{p_o} dT, h_2 - h_1 = C_{p_o} (T_2 - T_1) \end{aligned}$$

where C_{v_o} and C_{p_o} are *zero-pressure constant volume* and *zero-pressure constant pressure* specific heats. It can be shown that

$$C_{p_o} - C_{v_o} = R$$

and the ratio of specific heats C_{p_o}/C_{v_o} is denoted as γ . Their values for air as a perfect gas are:

$$\begin{aligned} C_{p_o} &= 1.005 \text{ kJ/kg. K}, C_{v_o} = 0.718 \text{ kJ/kg.K}, R = 0.287 \text{ kJ/kg. K} \\ \gamma &= 1.4, M = 28.966 \end{aligned}$$

For a *real gas*, the actual volume can be expressed by the general relation

$$v = v^{\text{real}} = z \frac{RT}{p}$$

where $z = v^{\text{real}}/v^{id} = pv/RT$ is called the *compressibility* of the gas.

Many different equations of state have been proposed to represent the real volume of gases. In 1949, Redlich-Kwong¹² (R-K) proposed the equation as follows in Eq. (1.12)

$$p = \frac{\bar{R} T}{\bar{v} - b} - \frac{a}{\sqrt{T} (\bar{v}^2 + \bar{v} b)} \quad (1.12)$$

in which v is the *molar volume* (volume of M kg of gas). Its constant a and b found from conditions of critical isotherm at the critical point are expressed in terms of critical constants T_c and p_c as follows:

$$a = 0.42748 \frac{R^2 T_c^{5/2}}{p_c}, b = 0.08664 \frac{R T_c}{p_c}$$

Often constants a and b are replaced by constants A and B written as:

$$A^2 = \frac{a}{R^2 T^{2.5}}, B = \frac{b}{RT}, \text{ and } h = \frac{b}{v} = \frac{b}{z RT/p} = \frac{Bp}{z}$$

so that the R-K equation can be written in terms of z in place of v as follows in Eq. (1.13)

$$\begin{aligned}
 z &= \frac{1}{1-h} - \frac{a}{bRT^{1.5}} \left(\frac{h}{1+h} \right) \\
 &= \frac{1}{1-h} - \frac{A^2}{B} \left(\frac{h}{1+h} \right) = \frac{p v}{RT}
 \end{aligned} \tag{1.13}$$

Note that h here is a function equal to b/v , and not the enthalpy.

A modification of the $R-K$ equation is Peng-Robinson¹¹ ($P-R$) equation as given in Eq. (1.14)

$$p = \frac{RT}{v-b} - \frac{a}{v^2 + 2bv - v^2} \tag{1.14}$$

where

$$a = \frac{0.45724 R^2 T_c^2}{p_c} \left[1 + f(\omega) \left\{ 1 - \left(\frac{T}{T_c} \right)^{1/2} \right\} \right]^2$$

$$b = \frac{0.0778 RT_c}{p_c}$$

$$f(\omega) = 0.37464 + 1.54226 \omega - 0.26992 \omega^2$$

$$\omega = \text{Accentric factor} = -1.0 - \log(p_r^{\text{sat}}) T_{r=0.7}$$

$$T_r = \text{Reduced temperature} = \frac{T}{T_c}$$

$$p_r = \text{Reduced pressure} = \frac{p}{p_c}$$

A very popular empirical equation is the *Martin-Hou⁹ equation*. A modified Martin-Hou equation, as recommended by Reynolds¹⁴, which applies with great accuracy to one of the most commonly used refrigerants, viz., Refrigerant 22 (CHClF_2), is given in Eq. (1.15) in simplified form:

$$p = \frac{RT}{v-b} + \sum_{i=2}^5 \frac{A_i + B_i T + C_i e^{-4.2 T/T_c}}{(v-b)^i} \tag{1.15}$$

where R is the gas constant for the refrigerant equal to 96.15 J/kg.K, p is in Pa, v is in m^3/kg , and the constants are:

$$b = 1.248556 \times 10^{-4}$$

$$A_2 = -1.169819 \times 10^2 \quad B_2 = 1.164312 \times 10^{-1} \quad C_2 = -1.1841 \times 10^3$$

$$A_3 = -2.929526 \times 10^{-2} \quad B_3 = 2.303194 \times 10^{-4} \quad C_3 = 2.48896$$

$$A_4 = 2.419193 \times 10^{-4} \quad B_4 = -6.796677 \times 10^{-7} \quad C_4 = 0.0$$

$$A_5 = -2.434584 \times 10^{-7} \quad B_5 = 6.302018 \times 10^{-10} \quad C_5 = -1.2062 \times 10^{-6}$$

Example 1.5 Solution Procedure for Martin-Hou Equation

Using Eq. (1.15), find the specific volume of saturated vapour of Refrigerant 22 at 40°C (313 K) temperature. Critical temperature of the refrigerant is 369 K.

Solution The saturation pressure of Refrigerant 22 at 40°C is 1,533,500 Pa. At the pressure and 313 K temperature, the ideal gas volume is

$$v^{id} = \frac{RT}{p} = \frac{96.15(313)}{1,533,500} = 0.01962 \text{ m}^3/\text{kg}$$

Using a value close to this as *first approximation*, we find, by substituting values in Eq. (1.15) by successive approximations, that at $v = 0.0151 \text{ m}^3/\text{kg}$, the right hand side is

$$\begin{aligned}\text{RHS} &= 2,009,60 - 508,900 + 33,760 + 580 - 100 \\ &= 1,534,940 \text{ Pa} \equiv \text{LHS}\end{aligned}$$

The error is only 0.01%. Hence, the required volume is 0.0151 m³/kg.

1.13 CLAPEYRON EQUATION

The *Clapeyron equation* relates the enthalpy of vaporization of a substance with the slope $dp^{\text{sat}}/dT^{\text{sat}}$ of its vapour pressure curve. It is written as given in Eq. (1.16)

$$\left(\frac{dp}{dT}\right)^{\text{sat}} = \frac{h_{fg}}{T(v_g - v_f)} \quad (1.16)$$

The example below illustrates the application of this equation to find the latent heat of vaporization of a refrigerant.

Example 1.6 Calculation of Latent Heat of Vaporization of R22.

From the following handbook data for R22,

t °C	p bar	v_f L/kg	v_g m ³ /kg
-4	4.358	0.77	0.0536
0	4.976	0.778	0.0471
+4	5.657	0.787	0.0416

Calculate the latent heat of vaporization of R22 of 0°C.

Solution

At 0°C,

$$\left(\frac{\Delta p}{\Delta T}\right)^{\text{sat}} = \frac{129.9 \text{ kPa}}{8^\circ \text{C}} = 16.237 \text{ kPa}/^\circ \text{C}$$

$$\begin{aligned}(h_{fg})0^\circ \text{C} &= \left(\frac{\Delta p}{\Delta T}\right)^{\text{sat}} T(v_g - v_f) \\ &= 16.237 (273.16) (0.0471 - 0.00077) \\ &= 205.48 \text{ kJ/kg}\end{aligned}$$

Note The value of $dp^{\text{sat}}/dT^{\text{sat}}$ at 0°C can also be found by differentiation from Eq. (1.10).

1.14 PROPERTY RELATIONS

$\delta Q = dU + \delta W$ is the first law equation which is applicable to *all process* in a closed system. For the particular case of reversible processes in a closed system, since $\delta W = pdV$ and $\delta Q = TdS$, this equation takes the form

$$TdS = dU + pdV \quad (1.17a)$$

$$TdS = dH - Vdp \quad (1.17b)$$

By dividing both sides by mass m , these can also be written on the basis of a unit mass of the substance as given below in Eqs. (1.18a) and (1.18b)

$$Tds = du + pdv \quad (1.18a)$$

$$Tds = dh - vdp \quad (1.18b)$$

These are the two well-known T - ds equations. Although derived for a reversible process in a closed system, these are actually in terms of properties of a system/substance. Hence, these are applicable to all processes whether reversible or irreversible, and whether in a closed system or in an open system. These equations are referred to as *property relations*, and are used to evaluate the changes in entropy in terms of changes in other properties employing the relationships as illustrated in Eqs. (1.19a) and (1.19b)

$$ds = \frac{1}{T} du + \frac{p}{T} dv \quad (1.19a)$$

$$ds = \frac{1}{T} dh - \frac{v}{T} dp \quad (1.19b)$$

1.15 THERMODYNAMIC PROPERTIES OF REFRIGERANTS²

For establishing the thermodynamic properties of refrigerants, and for that matter, for any pure substance, the following minimum experimental data/correlations are required:

- (i) p^{sat} versus T^{sat} , or T^{sat} versus p^{sat} .
- (ii) p - v - T data or equation of state for gaseous phase.
- (iii) Liquid density ρ or specific volume v_f .
- (iv) Liquid specific heat C_f .
- (v) Zero-pressure (ideal gas) constant pressure specific heat C_{p_0} or constant volume specific heat C_{v_0} ($= C_{p_0} - R$) of the gaseous phase.

Then the latent heat is calculated from the Clapeyron equation. The only other properties that need to be calculated are internal energy, enthalpy and entropy. For the purpose, the relations given in Eqs (1.20), (1.21), (1.22), and (1.23) for changes in u , h and s at constant temperature in the gaseous phase are used:

$$(u_2 - u_1)_T = \int_1^2 \left[T \left(\frac{\partial p}{\partial T} \right)_v - p \right] dv \quad (1.20)$$

$$(h_2 - h_1)_T = \int_1^2 \left[v - T \left(\frac{\partial v}{\partial T} \right)_p \right] dp \quad (1.21)$$

$$(s_2 - s_1)_T = - \int_1^2 \left(\frac{\partial v}{\partial T} \right)_v dp \quad (1.22)$$

$$(s_2 - s_1)_T = \int_1^2 \left(\frac{\partial p}{\partial T} \right)_v dv \quad (1.23)$$

Note that Eq. (1.20) for internal energy change requires a p -explicit equation of state, whereas Eq. (1.21) for enthalpy change requires a v -explicit equation of state. If only a p -explicit equation of state is available, then enthalpy change can be found from internal energy change using the relationship:

$$(h_2 - h_1)_T = (u_2 - u_1)_T + (p_2 v_2 - p_1 v_1)_T \quad (1.24)$$

1.15.1 Enthalpy Calculations

Figure 1.14 illustrates the method of calculation of enthalpy with the help of pressure-enthalpy ($p - h$) diagram. Since there is no absolute value of enthalpy, and only differences in enthalpies of state points are required in calculations, a *reference state* has to be chosen for the purpose to which an arbitrary value of enthalpy is assigned. In Fig. 1.14, the reference state chosen is that of saturated liquid at point 1. In the case of water, point 1, usually, is the saturated liquid state at its triple point (0.01°C) to which a value of $h_1 = h_{f_1} = 0 \text{ kJ/kg}$ is assigned. In the case of refrigerants, the reference state chosen is that of saturated liquid at 0°C . And since the refrigerants work at temperatures below 0°C also, in order to avoid negative values of enthalpies in calculations, the value of enthalpy assigned at the reference temperature is, usually, $h_1 = h_{f_1} = 200 \text{ kJ/kg}$. Note that the pressure at 1 is $(p^{\text{sat}})_{0^\circ\text{C}}$.

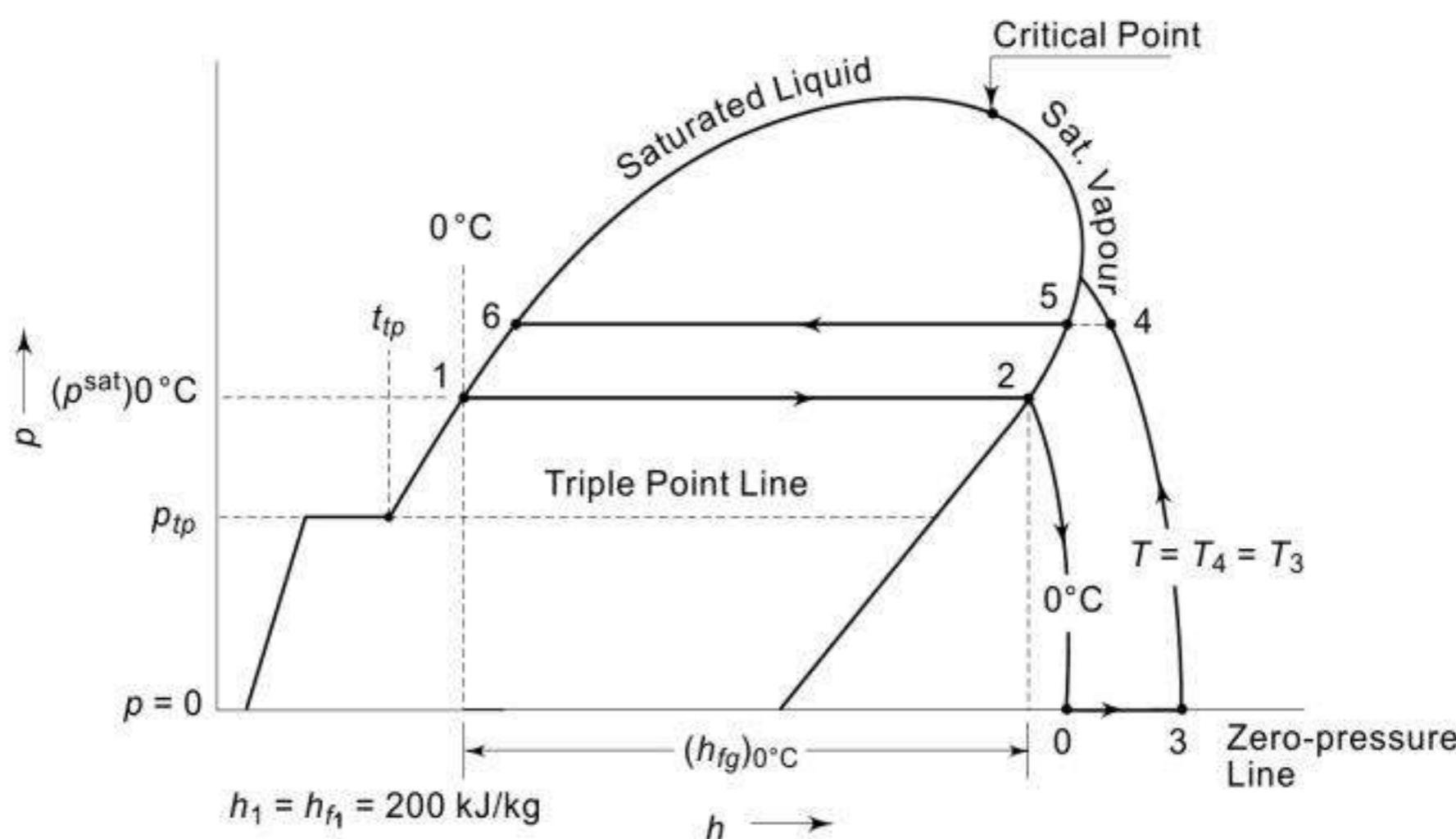


Fig. 1.14 Figure demonstrating method of calculation of enthalpy

Now the enthalpy of saturated vapour at 2 at the reference temperature and pressure is given by

$$h_2 = h_1 + (h_{fg})_{0^\circ\text{C}} = 200 + (h_{fg})_{0^\circ\text{C}}$$

Then, for the calculation of the enthalpy of vapour at any state 4 at temperature T_4 and pressure p_4 , the path followed is from 2 to 0, 0 to 3 and 3 to 4. State 0 is at the same temperature as 2 but at zero pressure. Similarly, state 3 is at the same temperature as 4 but at zero pressure. Both states 0 and 3 are, therefore, ideal gas states at temperatures 0°C and T respectively. It is seen that

$$h_4 = h_2 + (h_0 - h_2) + (h_3 - h_0) + (h_4 - h_3)$$

where

$$h_3 - h_0 = \int_{T_3}^{T_4} C_{p_0} dT$$

Now, $(h_4 - h_3)$ represents the change in enthalpy at constant temperature $T_4 = T_3$ as a result of change in pressure from 0 to $p_4 = p$. Similarly, $(h_2 - h_0)$ represents the change in enthalpy at 0°C as a result of change in pressure from 0 to p_2 . These represent differences in enthalpies of real gas and ideal gas at the same temperature. The difference $h^R = (h - h^{id})_T$ is termed *residual enthalpy*, and can be found either from Eq. (1.21) or Eq. (1.24) by integrating between pressure limits of $p = 0$ to p , or between volume limits if $v = \infty$ (at $p = 0$) to v .

The enthalpy of saturated vapour at any point 5 can be similarly calculated. Then, the enthalpy of saturated liquid at 6 can be found from

$$h_6 = h_5 - (h_{fg})_T$$

or from

$$h_6 = h_1 + \int_{T_1}^{T_6} c_f dT$$

Example 1.7 Calculation of Enthalpy of Vapour Using R-K Equation of State

Using Redlich-Kwong equation of state, calculate the enthalpy of superheated vapour of propane at 2 MPa and 350 K. Take $h_f = 0$ at 200 K ($p^{sat} = 19.97 \text{ kPa}$) as the reference state. Critical data for propane are:

$$T_c = 369.8 \text{ K}, p_c = 4.236 \text{ MPa}, v_c = 0.005066 \text{ m}^3/\text{kg}$$

Assume zero-pressure constant pressure specific heat of propane as constant as

$$C_{p_0} = 1.6794 \text{ kJ/kg.K}$$

Solution Refer to Fig. 1.14. Reference state 1 is at 200 K, and not 0°C . From Clapeyron equation, we find:

$$(h_{fg})_{200 \text{ K}} = 456.24 \text{ kJ/kg}$$

Hence,

$$h_2 = h_1 + (h_{fg})_{200 \text{ K}} = 456.24 \text{ kJ/kg}$$

Gas constant for propane

$$R = \frac{\bar{R}}{M} = \frac{8.3143}{44} = 0.18855 \text{ kJ/kg.K}$$

Constants of Redlich Kwong equation

$$a = 0.42748 R^2 \frac{T_c^{5/2}}{p_c}$$

$$= 0.42748 \frac{(188.55)^2 (369.8)^{5/2}}{4.236 \times 10^6} = 9416$$

$$b = 0.08664 \frac{R T_c}{p_c}$$

$$= 0.08664 \frac{(188.55) (369.8)}{4.236 \times 10^6} = 1.426 \times 10^{-3}$$

Substituting values in R-K equation, we get

$$v_2 = 1.865 \text{ m}^3/\text{kg}$$

$$v_4 = 0.02534 \text{ m}^3/\text{kg}$$

$$z_2 = \frac{p_2 v_2}{RT_2} = 0.9876$$

$$z_4 = \frac{p_4 v_4}{RT_4} = 0.7677$$

Redlich-Kwong¹² derived the following expression for residual enthalpy

$$\frac{h^R}{RT} = \frac{h - h^{id}}{RT} = -\frac{3}{2} \frac{A^2}{B} \ln \left[1 + \frac{Bp}{z} \right] + (z - 1) \quad (1.25)$$

The constants at points 2 and 4 are:

$$A_2^2 = \frac{a}{R^2 T_2^{2.5}} = 4.692 \times 10^{-7}$$

$$A_4^2 = \frac{a}{R^2 T_4^{2.5}} = 1.1582 \times 10^{-7}$$

$$B_2 = \frac{b}{RT_2} = 3.782 \times 10^{-8}$$

$$B_4 = \frac{b}{RT_4} = 2.16 \times 10^{-8}$$

Residual enthalpies at points 2 and 4 are:

$$\left(\frac{h - h^{id}}{RT} \right)_2 = -\frac{3}{2} \cdot \frac{4.692 \times 10^{-7}}{3.782 \times 10^{-8}} \ln \left[1 + \frac{3.782 \times 10^{-8} \times 19.97 \times 10^3}{0.9876} \right]$$

$$+ (0.9876 - 1) = -0.02663$$

$$\Rightarrow (h - h^{id})_2 = -(0.18855) (200) (0.02663) = -1.0 \text{ kJ/kg}$$

$$\left(\frac{h - h^{id}}{RT} \right)_4 = -\frac{3}{2} \cdot \frac{4.692 \times 10^{-7}}{3.782 \times 10^{-8}} \ln \left[1 + \frac{3.782 \times 10^{-8} \times 2 \times 10^6}{0.7677} \right]$$

$$+ (0.7677 - 1) = -0.6723$$

$$\Rightarrow (h - h^{id})_4 = -(0.18855)(350)(0.6723) = -44.35 \text{ kJ/kg}$$

Change in enthalpy at zero pressure

$$h_3 - h_0 = \int_{200}^{350} C_{p_0} dT = 1.6794 (350 - 200) = 251.91 \text{ kJ/kg}$$

Enthalpy of superheated propane vapour at 4

$$\begin{aligned} h_4 &= h_2 + (h_0 - h_2) + (h_3 - h_0) + (h_4 - h_3) \\ &= h_2 - (h - h^{id})_2 + (h_3 - h_0) + (h - h^{id})_4 \\ &= 456.24 + 1.0 + 251.91 - 44.35 = 664.8 \text{ kJ/kg} \end{aligned}$$

Note The value from table is 641.6 kJ/kg. The error is due to the following reasons:

- (i) Redlich-Kwong equation has been used in place of an equation fitting actual p-v-T data.
- (ii) A constant and approximate value of C_{p_0} has been used.

1.15.2 Enthalpy from Residual Internal Energy

If only a p -explicit equation is available, the enthalpy change at constant temperature between points 2 and 0, and 4 and 3 can be calculated by using Eq. (1.24). For the purpose, we have for *residual internal energy* from Eq. (1.20)

$$\begin{aligned} u^R &= (u - u^{id})_T = \int_{v=\infty}^v \left(T \left(\frac{\partial p}{\partial T} \right)_v - p \right) dv_T \\ &= \int_v^{\infty} \left[p - T \left(\frac{\partial p}{\partial T} \right)_v \right] dv_T \end{aligned}$$

Using this expression, one can find u_0 and then h_0 from Eq. (1.24) at $t_0 = 0^\circ\text{C}$ ($T_0 = 273.15 \text{ K}$) and $p_0 = 0$. Then, one can find the enthalpy at 4 using the relationship

$$\begin{aligned} h_4 &= h_0 + (h_3 - h_0) + (h_4 - h_3) \\ &= h_0 + \int_{T_0}^{T_3} C_{p_0} dT + \left| (u - u^{id})_T \right|_{v=\infty}^{v=v_4} + p_4 v_4 - p_3 v_3 \end{aligned} \quad (1.25a)$$

Putting $C_{p_0} = C_{v_0} + R$, and $p_3 v_3 = RT_3$ (ideal gas state at T_3), we have

$$h_4 = h_0 - RT_0 + p_4 v_4 + \int_{T_0}^{T_3} C_{v_0} dT + \left| (u - u^{id})_T \right|_{v=\infty}^{v=v_4} \quad (1.25b)$$

Note Often, state 0 with enthalpy h_0 is employed as the new reference state for further calculation of enthalpies, and entropies in the entire vapour region.

Example 1.8 Residual Internal Energy from Modified Martin-Hou Equation

Derive an expression for $(u - u^{id})_T$ from Eq. (1.15).

Solution We have Eqs. (1.26), (1.27), (1.28) by differentiating partially with respect to T

$$\left(\frac{\partial p}{\partial T}\right)_v = \frac{R}{v - b} + \sum_{i=2}^5 \frac{1}{(v - b)^i} \left[B_i - C_i \frac{k}{T_c} e^{-kT/T_c} \right] \quad (1.26)$$

$$\begin{aligned} \left[p - \left(\frac{\partial p}{\partial T}\right)_v T \right] &= \cancel{\frac{RT}{v-b}} + \sum_{i=2}^5 \frac{A_i + B_i T + C_i e^{-kT/T_c}}{(v-b)^i} \\ &- \cancel{\frac{RT}{v-b}} + \sum_{i=2}^5 \frac{B_i T - \frac{C_i k T}{T_c} e^{-kT/T_c}}{(v-b)^i} \end{aligned} \quad (1.27)$$

$$\begin{aligned} \int_v^{v=\infty} \left[p - \left(\frac{\partial p}{\partial T}\right)_v T \right] dv &= \int_v^{v=\infty} \sum_{i=2}^5 \frac{A_i + C_i e^{-kT/T_c} \left(1 + \frac{kT}{T_c}\right)}{(v-b)^i} \\ &= \sum_{i=2}^5 - \left[\frac{A_i + C_i e^{-kT/T_c} \left(1 + \frac{kT}{T_c}\right)}{(-i+1)(v-b)^{i-1}} \right] \end{aligned} \quad (1.28)$$

Example 1.9 Reference State Enthalpy of Freon 22 Vapour

Find the value of ideal gas enthalpy h_0 at $T_0 = 273.15\text{ K}$ and $p_0 = 0$ of Freon 22 vapour using the modified Martin-Hou equation. T_c for Freon 22 is 369 K.

Solution Refer to Fig. 1.14.

First we find $p_2 = (p^{\text{sat}})_{273.15\text{ K}} = 4.976 \times 10^5 \text{ Pa}$ from Eq. (1.10)

Next we find $v_2 = 0.0471 \text{ m}^3/\text{kg}$ at p_2 and T_2 from Eq. (1.15)

Then,

$$\begin{aligned} h_2 &= h_{f_1} + (h_{fg})_{273.15\text{ K}} = 200 + 205.48 = 405.48 \text{ kJ/kg} \\ u_2 &= h_2 - p_2 v_2 \\ &= 405.48 - 497.6 (0.0471) = 382.04 \text{ kJ/kg} \end{aligned}$$

Now, substituting values of A_i , C_i , k , T_c , and b in Eq. (1.28), we have for $T_2 = 273.15\text{ K}$ and $v_2 = 0.0471 \text{ m}^3/\text{kg}$,

$$\begin{aligned} u_2 - u_2^{id} &= u_2 - u_0 = -7020 \text{ J/kg} = -7.02 \text{ kJ/kg} \\ \Rightarrow u_0 &= u_2 + 7.02 = 389.06 \text{ kJ/kg} \\ \Rightarrow h_0 &= u_0 + R T_0 = 389.06 + 0.09615 (273.15) \\ &= 415.32 \text{ kJ/kg} \end{aligned}$$

Example 1.10 Enthalpy of Freon 22 Vapour

Taking the value of reference state enthalpy h_0 as found in Example 1.9, calculate the enthalpy of Freon 22 superheated vapour at 1650 kPa and 95°C (368.15 K).

Solution From Eq. (1.15), $v = 0.0187 \text{ m}^3/\text{kg}$ at 1650 kPa and 368.15 K. Substituting values of constants, and v and T , we get:

$$\left(u - u^{id} \right)_{368.15 \text{ K}} \Bigg|_{v = \infty}^{v = 0.0187} = -27,440 \text{ J/kg} = -27.44 \text{ kJ/kg}$$

The expression for C_{v_a} for Freon 22 from Appendix is as follows in Eq. (1.29)

$$C_{v_o} = C_1 + C_2 T + C_3 T^2 - \frac{C_4}{T} \text{ J/kg.K} \quad (1.29)$$

where $C_1 = 117.767818$, $C_2 = 1.6997296$

$$C_3 = -8.83043292 \times 10^{-4}, C_4 = 3.32541759 \times 10^5$$

$$\int_{273.15}^{368.15} C_{v_o} dT = 70,750 \text{ J/kg} = 70.75 \text{ kJ/kg}$$

$$\begin{aligned}
 h &= h_0 - RT_0 + pv + \int_{T_o}^T C_{v_o} dT + \left| \left(u - u^{id} \right)_T \right|_{v=\infty}^v \\
 &= 415.32 - 0.09615(273.15) + 1650(0.0187) + 70.75 - 27.44 \\
 &= 463.22 \text{ kJ/kg}
 \end{aligned} \tag{1.30}$$

1.15.3 Entropy Calculations

The method of calculation of entropy is similarly illustrated in Fig. 1.15. The value of entropy assigned to the reference state of saturated liquid at 0°C is usually $s_1 = s_{f_1} = 1.0 \text{ kJ/kg}\cdot\text{K}$. Then, the entropy at 2 is

$$s_2 = s_1 + (s_{fg})_{0^\circ\text{C}} = 1.0 + \frac{(h_{fg})_{0^\circ\text{C}}}{273.15} \text{ kJ/kg.K}$$

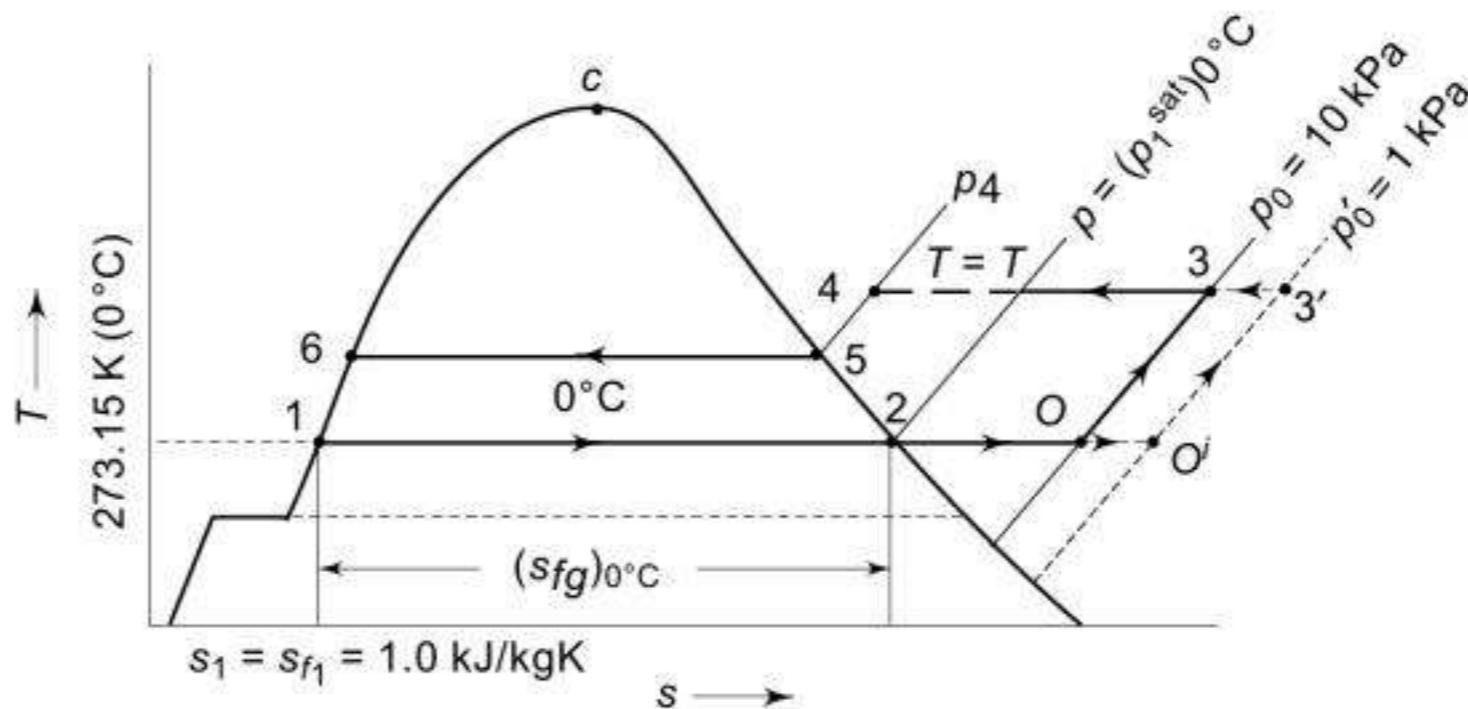


Fig. 1.15 Figure demonstrating method of calculation of entropy

And the entropy of vapour at any point 4 is given by

$$s_4 = s_2 + (s_0 - s_2) + (s_3 - s_0) + (s_4 - s_0)$$

$$= s_2 - \left| (s - s^{id})_{0^\circ C} \right|_{p_o}^{p_2} + \int_{T_o}^{T_3} C_{p_o} \frac{dT}{T} + \left| (s - s^{id})_{T_4} \right|_{p_o}^{p_4}$$

We note that in the calculation of residual entropy $s^R = (s - s^{id})$, the lower limit of integration is not $p_o = 0$ but p_o equal to some very low pressure, say 10 kPa, at which the gas behaves as an ideal gas. Correspondingly, the volume at 0 is $v_0^{id} = RT_0/p_0$, and the volume at 3 is $v_3^{id} = RT_3/p_0$. This is done because of a mathematical anomaly since $s \rightarrow \infty$ as $p \rightarrow 0$. The path followed for entropy calculation is shown in Fig. 1.15. In case a lower reference pressure is chosen such as $p_0' = 1, 0.1, 0.01$, etc., kPa, this will not affect the values at 4 since the new path followed will now be $2 - 0', 0' - 3'$, and $3' - 4$.

Example 1.11 Residual Entropy from Modified Martin-Hou Equation

Derive an expression for residual entropy from Eq. (1.15).

$$\begin{aligned} \text{Solution } s^R &= (s - s^{id})_T = \int_{v^{id}}^v \left(\frac{\partial P}{\partial T} \right)_v dv \\ &= \int_{v^{id}}^v \frac{R}{v-b} dv + \int_{v^{id}}^v \sum_{i=2}^5 \frac{B_i - C_i}{(v-b)^i} \frac{k}{T_c} e^{-kT/T_c} \\ &= R \ln \frac{p_o v}{RT} + \sum_{i=2}^5 \frac{B_i - C_i}{(-i+1)} \frac{k}{T_c} \frac{e^{-kT/T_c}}{(v-b)^{i-1}} \quad (1.31) \end{aligned}$$

Note In the integration of the first term $(v-b)$ has been approximated to v . The term containing $1/(v^{id}-b)^{i-1}$, obtained after the integration of the second term, is approximated to zero as $v^{id} \gg v$.

Example 1.12 Reference State Entropy of Freon 22 Vapour

Find the reference state entropy s_0 of Freon 22 vapour at 273.15 K and 10 kPa using modified Martin-Hou equation.

Solution Ideal gas volume at 273.15 K

$$v_0^{id} = \frac{R T_0}{p_0} = \frac{0.09615 (273.15)}{10} = 2.62625 \text{ m}^3/\text{kg}$$

Specific volume of saturated Freon 22 vapour at 273.15 K

$$v_2 = 0.0471 \text{ m}^3/\text{kg}, \text{ from Eq. (1.15)}$$

Substituting values in Eq. (1.31), we get

$$(s - s^{id})_{0^\circ\text{C}} = s_2 - s_0 = -0.4016 \text{ kJ/kg}$$

$$\text{Now, } s_2 = s_1 + \frac{(h_{fg})_{0^\circ\text{C}}}{273.15} = 1 + \frac{205}{273.15} = 1.7518 \text{ kJ/kg.K}$$

$$\Rightarrow s_0 = s_2 + 0.4016 = 2.1534 \text{ kJ/kg.K}$$

Note It will be interesting to see how the value of s_0 will change if $p'_0 = 1 \text{ kPa}$ is taken. It will be seen that $s'_0 > s_0$. But, it will not effect the value of s_4 since $s'_3 > s_3$. See Fig. 1.15.

Example 1.13 Entropy of Freon 22 Vapour

Find the entropy of superheated Freon 22 vapour at 1650 kPa and 95°C (368.15 K).

Solution Ideal gas volume at 368.15 K

$$v_4^{id} = \frac{RT}{p_0} = \frac{0.09615(368.15)}{10} = 3.5398 \text{ m}^3/\text{kg}$$

Specific volume of superheated vapour at 1650 kPa and 368.15 K

$$v_4 = 0.0187 \text{ m}^3/\text{kg}, \text{ from Eq. (1.15)}$$

Using Eq. (1.29)

$$\begin{aligned} s_3 - s_0 &= \int_{T_0}^T C_{p_o} \frac{dT}{T} = \int_{T_0}^T C_{v_o} \frac{dT}{T} + R \ln \frac{T}{T_0} \\ &= R \ln \frac{T}{T_0} + \left[C_1 \ln \frac{T}{T_0} + C_2(T - T_0) \right. \\ &\quad \left. + \frac{C_3}{2} (T^2 - T_0^2) - 2C_4 \left(\frac{1}{T} - \frac{1}{T_0} \right) \right] 10^{-3} \\ &= 0.827 \text{ kJ/kg.K} \end{aligned}$$

$$s_4 - s_3 = \left| (s - s^{id}) \right|_{368.15 \text{ K}} \Bigg|_{\substack{v=0.0187 \\ v^{id}=3.5398}} = -1.1488 \text{ kJ/kg.K}$$

Hence,

$$\begin{aligned} s_4 &= s_0 + (s_3 - s_0) + (s_4 - s_3) \\ &= 2.1534 + 0.827 - 1.1488 = 1.8316 \text{ kJ/kg.K} \end{aligned}$$

1.16 MODES OF HEAT TRANSFER

The difference in temperature provides the necessary potential for heat transfer. There are three *modes of heat transfer*. They are *conduction*, *convection* and *radiation*.

Essentially heat is transferred within a *stationary* medium by conduction, viz., from particle to particle, whether it be solid, liquid or gas. In convection, there must be a bulk flow of the fluid. Heat is carried away from the wall surface by the flowing fluid. Convection, however, takes place in two ways, viz., *forced convection* and *natural or free convection*. In forced convection, the flow of the fluid is produced by an external source such as a pump or a fan. Examples are the shell and tube condenser of a refrigeration plant in which the flow of water is maintained by a pump, and the air-cooled condenser of an air conditioner in which the flow of air is maintained by a fan. In natural or free convection, the flow of the fluid is produced

by the difference in density due to temperature difference. The higher temperature fluid, being lighter, rises up and the lower temperature fluid, being heavier, settles down. Thus a natural convection current is set up in the fluid. One example is the air-cooled condenser of a domestic refrigerator. In radiation, heat is transferred in the form of electromagnetic waves. For radiative heat transfer, therefore, the presence of a medium is not necessary.

1.17 LAWS OF HEAT TRANSFER

The *heat flux* q is the heat transfer rate \dot{Q} per unit area A normal to the direction of flow of heat. The various laws relate the heat transfer rate or heat flux to temperature difference ΔT . The unit of ΔT is degrees kelvin (K) or degrees celsius ($^{\circ}\text{C}$). Both have the same numerical value.

1.17.1 Fourier Law of Heat Conduction

The heat flux by conduction is proportional to the *temperature gradient* within a body. Thus at any point P in a body, if the temperature gradient is $\frac{\partial T}{\partial x}$ as shown in

Fig. 1.16, then the heat flux is given by

$$\begin{aligned} q &= \frac{\dot{Q}}{A} \propto \frac{\partial T}{\partial x} \\ \text{or} \quad \dot{Q} &= -kA \frac{\partial T}{\partial x} \end{aligned} \quad (1.32)$$

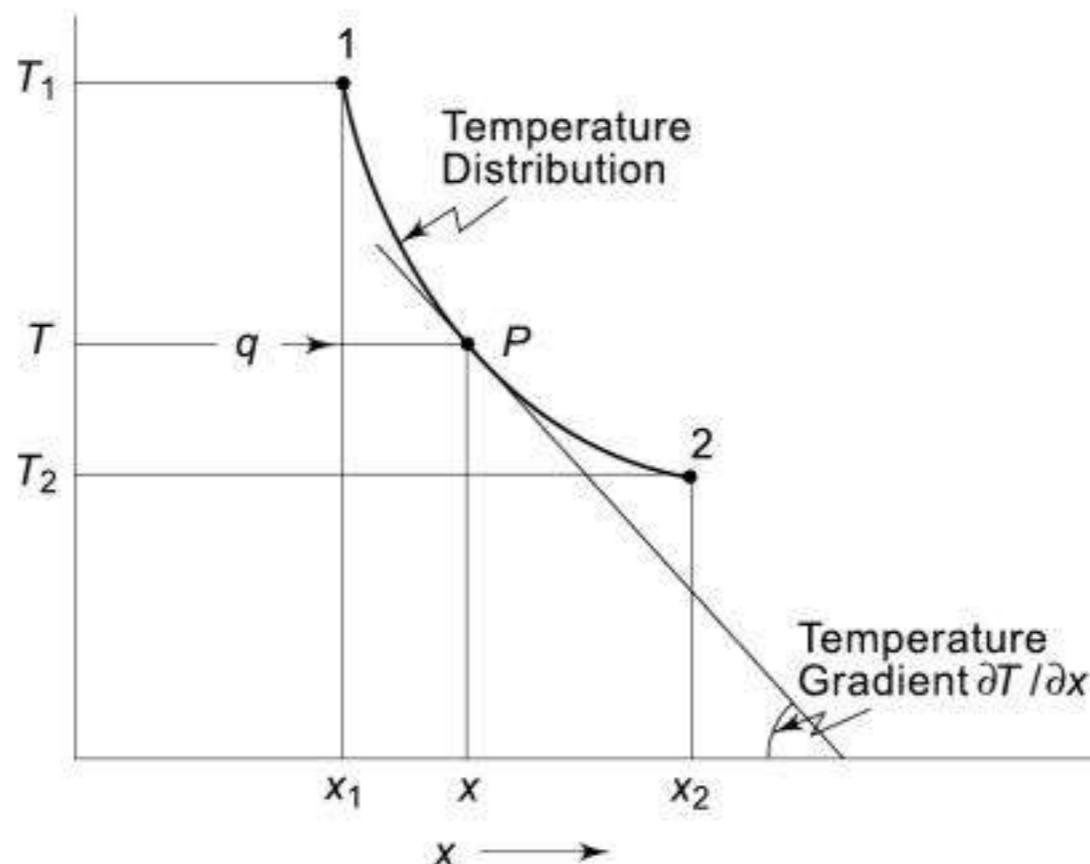


Fig. 1.16 Temperature distribution in a conduction medium

where k is the constant of proportionality, called *thermal conductivity* which is a property of the material of the body. The minus sign is inserted to make \dot{Q} positive since $\frac{\partial T}{\partial x}$ is negative. Equation (1.32) is called the *Fourier law of heat conduction*.

The SI unit of thermal conductivity can be derived as below

$$\begin{aligned}[k] &= \frac{[\dot{Q}] [\Delta x]}{[A] [\Delta T]} \\ &= \frac{(\text{J/s}) \text{ m}}{\text{m}^2 \text{ K}} = \text{Wm}^{-1} \text{ K}^{-1}\end{aligned}$$

Physically, thermal conductivity represents the amount of heat that will flow per unit time, per unit area normal to the direction of flow of heat, through a unit thickness of the material and when the temperature difference across the material is unity. Experimental measurements of q and $\Delta T/\Delta x$ can be made to determine the thermal conductivity of materials. It is seen that the order of decreasing thermal conductivity is as follows:

Metals; non-metals; liquids; insulating materials; gases.

Metals

Common metals used in heat exchangers are copper, aluminium and iron. Their thermal conductivities in $\text{Wm}^{-1} \text{ K}^{-1}$ units are as follows:

Copper	387
Aluminium	203
Iron	73

It is seen that copper has the highest thermal conductivity. It is, therefore, used in condensers, evaporators, etc., in Freon refrigeration systems. However, ammonia attacks copper. Hence, iron is used in ammonia heat exchangers. As aluminium is cheap, and widely available, attempts are being made to use it in Freon systems.

Thermal conductivities of other important substances (average values) are:

Liquids

Water	0.556
Ammonia	0.54
Freon 22	0.093

Gases

Air	0.024
Water vapour	0.0206

Low temperature insulating materials

Expanded polystyrene (Thermocole)	0.037
Polyurethane foam(PUF)	0.0173

The low values of thermal conductivities of these insulating materials are essentially due to the low conductivity of gas/air trapped inside, as these are manufactured by blowing agents like air/R134a into the melted polystyrene and polyurethane respectively.

1.17.2 Newton's Law of Cooling for Convection

When a fluid flows over a wall which is at a different temperature than the fluid, heat will flow from the wall to the fluid or from the fluid to the wall depending on the direction of the temperature gradient. Although this mode of heat transfer is named convection, the physical mechanism of heat transfer at the wall is a conduction process.

Consider a fluid flowing along a wall maintained at temperature T_w as shown in Fig. 1.17. The free stream temperature of the fluid is T_∞ . Then a temperature field varying from T_w to T_∞ will establish itself in the fluid near the wall. Let δ_T be the distance from the wall of the point in the fluid at which the temperature of the fluid just becomes equal to the free stream temperature T_∞ . The distance δ_T is termed the *thermal boundary layer*. The distance δ from the wall in which the fluid velocity becomes equal to the free stream velocity is called the *hydrodynamic boundary layer*.

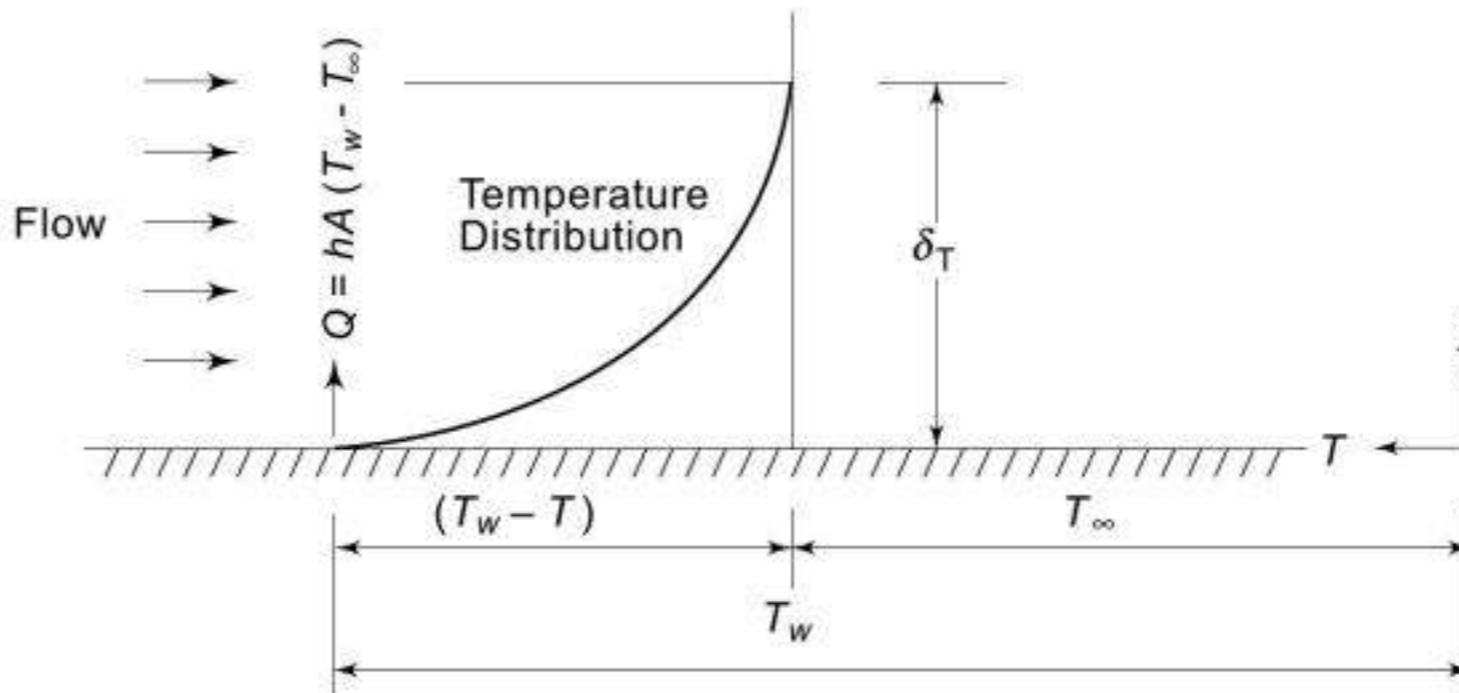


Fig. 1.17 Temperature distribution in thermal boundary layer in convection

The concept of the *heat transfer coefficient* or *film coefficient* or *surface conductance*, denoted by the symbols h or f , was introduced by Newton. He recommended the following equation to evaluate the heat transfer rate by convection.

$$\dot{Q} = hA (T_w - T_\infty) = hA \Delta T \quad (1.33)$$

where A is the wall surface area and $\Delta T = (T_w - T_\infty)$.

It must be noted that this h or f is not a property of the fluid. However, it depends on the *thermophysical* or *heat transport* properties of the fluid such as thermal conductivity k , dynamic viscosity μ , density ρ and specific heat C_p . In addition, it also depends on the hydrodynamic or flow parameters such as velocity of flow and characteristic dimensions.

The SI unit of the heat transfer coefficient can now be derived as follows:

$$\begin{aligned} [h] &= \frac{[\dot{Q}]}{[A] [\Delta T]} \\ &= \frac{\text{J/s}}{\text{m}^2 \text{K}} = \text{W m}^{-2} \text{K}^{-1} \end{aligned}$$

Typical values of h are given in Table 1.2.

Table 1.2 Typical values of convection heat transfer coefficients

Mode and Medium	$h, \text{W m}^{-2} \text{K}^{-1}$
Free convection, air	5 – 25
Forced convection, air	10 – 100
Forced convection, water	5,000 – 10,000
Boiling refrigerant	500 – 2,000
Condensing refrigerant	1,500 – 2,500

1.17.3 Thermal Radiation

The thermal radiation emitted by a body per unit area of its surface is called *emissive power* which is proportional to the fourth power of its absolute temperature.

For an *ideal radiator* or *black body*, the emissive power E_b is given by the *Stefan-Boltzman law* that follows in Eq. (1.34)

$$E_b = \sigma T^4 \quad (1.34)$$

where σ is the constant of proportionality called the *Stefan-Boltzman constant* having a numerical value of $5.669 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}$.

The emissive power of an actual radiator is expressed by the relation given in Eq. (1.35)

$$E = \epsilon E_b = \epsilon \sigma T^4 \quad (1.35)$$

where

$\epsilon = \frac{E}{E_b}$ is the emissivity of the actual surface.

Thermal radiations emitted by two bodies of areas A_1 and A_2 , at temperatures T_1 and T_2 will then be given by

$$\dot{Q}_1 = \epsilon_1 A_1 \sigma T_1^4 \text{ and } \dot{Q}_2 = \epsilon_2 A_2 \sigma T_2^4$$

However, the heat exchange by thermal radiation between two bodies will also depend on the extent to which the two bodies “see” each other geometrically. The expression for such a heat exchange may be expressed as given in Eq. (1.36)

$$\dot{Q} = \sigma A_1 F_{12} (T_1^4 - T_2^4) = \sigma A_2 F_{21} (T_1^4 - T_2^4) \quad (1.36)$$

where $A_1 F_{12} = A_2 F_{21}$ is the *reciprocity relation*, and F_{12} and F_{21} are *geometric factors* which depend on the emissivities ϵ_1 and ϵ_2 , and the geometry and orientation of the bodies.

In the case of a small body of area A_1 surrounded by a large body of area A_2 , completely seeing each other, we have

$$F_{12} = \epsilon_1 \quad (1.36)$$

so that

$$\dot{Q} = \epsilon_1 A_1 \sigma (T_1^4 - T_2^4) \quad (1.37)$$

Often, the heat exchange by radiation \dot{Q}_R between two surfaces is expressed in terms of a *radiation coefficient* h_R defined by

$$\dot{Q}_R = h_R A_1 (T_1 - T_2) = A_1 F_{12} \sigma (T_1^4 - T_2^4)$$

so that

$$h_R = \frac{F_{12} \sigma (T_1^4 - T_2^4)}{(T_1 - T_2)}$$

1.18 ELECTRICAL ANALOGY¹

It is found convenient to handle complicated heat transfer problems involving composite materials and multi-modes by unifying the concept of heat transfer with that

of the flow of electric current. Comparing the flow of electrical energy and the flow of heat, it is found that the following similarities hold.

Quantity	Electrical Energy	Heat
Driving potential	Voltage, V	Temperature difference, ΔT
Flow	Current, I	Heat transfer rate, \dot{Q}

The governing law for the transfer of electrical energy is Ohm's law

$$I = \frac{V}{R_e}$$

where R_e is the electrical resistance. One can similarly express the heat transfer rate by Eq. (1.38)

$$\dot{Q} = \frac{\Delta T}{R} = C \Delta T \quad (1.38)$$

where R is the *thermal resistance* and $C = 1/R$ is the *thermal conductance*. The units of the two quantities are $\text{W}^{-1} \text{K}$ and WK^{-1} respectively.

Comparing Eq. (1.38) with Eqs (1.32), (1.33) and (1.36) we obtain expressions for conductive, convective and radiative resistances respectively as follows:

$$R_{\text{COND}} = \frac{\Delta x}{k A}, \text{ for plane wall of thickness } \Delta x \quad (1.39)$$

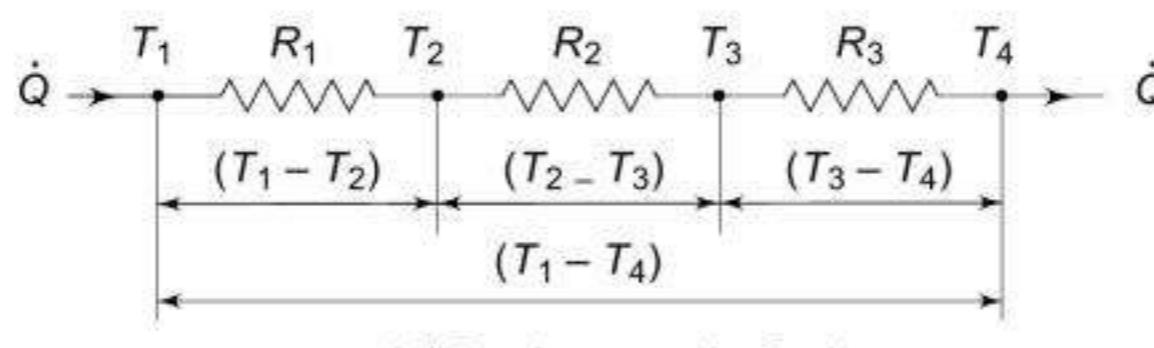
$$R_{\text{CONV}} = \frac{1}{h A} \quad (1.40)$$

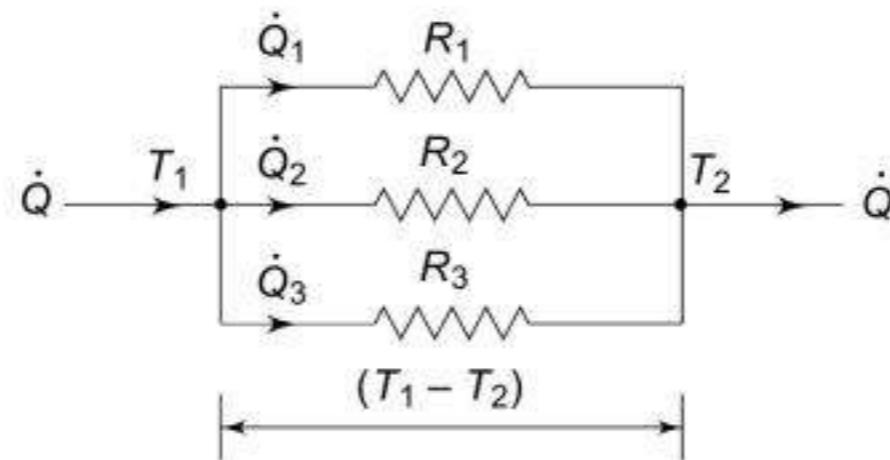
$$R_{\text{RAD}} = \frac{\Delta T}{\sigma A_1 F_{12} (T_1^4 - T_2^4)} \quad (1.41)$$

1.18.1 Resistances in Series

When resistances are in series as shown in Fig. 1.18 (a), it implies that the heat transfer rate through all the resistances is the same. We can then write: as given in Eq. (1.42)

$$\dot{Q} = \frac{T_1 - T_2}{R_1} = \frac{T_2 - T_3}{R_2} = \frac{T_3 - T_4}{R_3} = \frac{\Delta T}{R} \quad (1.42)$$





(b) Resistances in Parallel

Fig. 1.18 Thermal resistances in series and in parallel

where $\Delta T = (T_1 - T_4)$ is the overall temperature difference, and R is the overall thermal resistance. From Eq. (1.42), the individual temperature drops are:

$$T_1 - T_2 = \dot{Q} R_1$$

$$T_2 - T_3 = \dot{Q} R_2$$

$$T_3 - T_4 = \dot{Q} R_3$$

Adding,

$$\overline{T_1 - T_4 = \dot{Q} (R_1 + R_2 + R_3)}$$

$$\Rightarrow \dot{Q} = \frac{T_1 - T_4}{R_1 + R_2 + R_3} = \frac{\Delta T}{R}$$

so that the overall thermal resistance R is given by Eq. (1.43)

$$R = R_1 + R_2 + R_3 + \dots \quad (1.43)$$

1.18.2 Resistances in Parallel

When resistances are in parallel, as shown in Fig. 1.18(b), the net heat transfer rate is equal to the sum of the heat transfer rates through all sections. At the same time, the temperature drop ΔT across each resistance is the same. Hence,

$$\dot{Q}_1 = \frac{\Delta T}{R_1}$$

$$\dot{Q}_2 = \frac{\Delta T}{R_2}$$

$$\dot{Q}_3 = \frac{\Delta T}{R_3}$$

Adding

$$\overline{\dot{Q} = \dot{Q}_1 + \dot{Q}_2 + \dot{Q}_3}$$

$$= \Delta T \left[\frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} \right] = \frac{\Delta T}{R}$$

whence we get for the overall thermal resistance Eq. (1.44)

$$\frac{1}{R} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \dots + \dots \quad (1.44)$$



1.19 STEADY-STATE CONDUCTION

In general, the temperature distribution throughout a body may vary with location and time. Under *steady-state* conditions, however, the temperature does not change with time. We shall now examine some steady-state heat conduction problems in one dimension.

1.19.1 Heat Flow Through a Slab or Plane Wall

Let there be a slab of thickness Δx , the two faces of which are maintained at temperature T_1 and T_2 as shown in Fig. 1.19. Consider a section at a distance x from one end. Let the temperature gradient at this section be dT/dx . Then heat entering the wall, per unit time, at this section is

$$\dot{Q}_x = -kA \frac{dT}{dx} \quad (1.45)$$

Similarly, heat leaving the wall at $(x + dx)$ is

$$\dot{Q}_{x+dx} = \dot{Q}_x + \frac{d\dot{Q}_x}{dx} dx \quad (1.46)$$

Under steady-state conditions

$$\dot{Q}_x = \dot{Q}_{x+dx}$$

so that from Eqs (1.45) and (1.46) we have Eq. (1.47)

$$\frac{d^2T}{dx^2} = 0 \quad (1.47)$$

Using the boundary conditions

(i) $T = T_1$ at $x = 0$, (ii) $T = T_2$ at $x = \Delta x$

and solving the differential Eq. (1.47), we obtain for the temperature distribution in the wall

$$\frac{T - T_1}{T_2 - T_1} = \frac{x}{\Delta x} \quad (1.48)$$

which is linear with respect to x . The temperature gradient is

$$\frac{dT}{dx} = \frac{T_2 - T_1}{\Delta x} \quad (1.49)$$

which is constant. The heat transfer rate is given by the Fourier law in Eq. (1.50)

$$\dot{Q} = -kA \frac{dT}{dx} = \frac{kA (T_1 - T_2)}{\Delta x} = \frac{T_1 - T_2}{\Delta x / kA} \quad (1.50)$$

It is seen that the thermal resistance of the wall is

$$R = \frac{\Delta x}{kA}$$

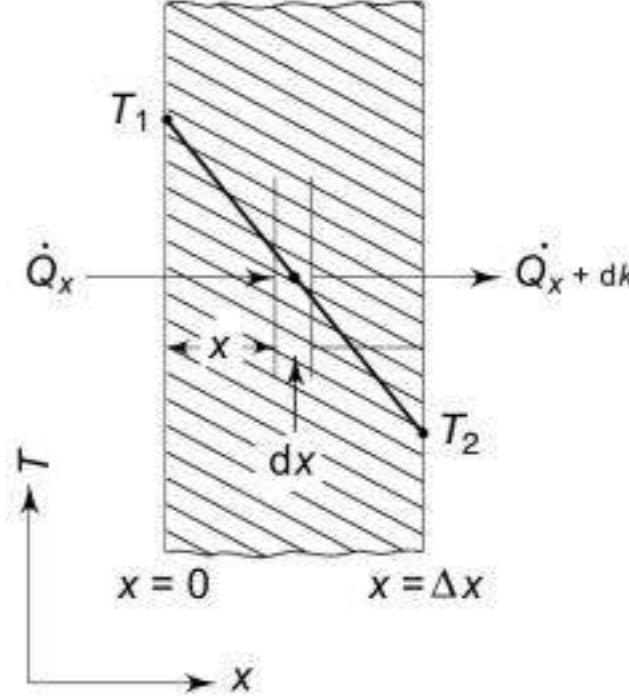


Fig. 1.19 Heat conduction through a plane wall

1.19.2 Heat Flow Through a Cylinder

Consider a cylindrical shell of thickness dr at a distance r from the axis of a cylindrical tube or pipe of length L and outer and inner radii of r_0 and r_1 respectively, as shown in Fig. 1.20. The area of the inside surface of the shell will be $2\pi rL$. Then the heat flows in the radial direction at r and $r + dr$ are:

$$\dot{Q}_r = -k(2\pi rL) \frac{dT}{dr}, \quad \dot{Q}_{r+dr} = \dot{Q}_r + \frac{d\dot{Q}_r}{dr} \cdot dr$$

Under steady-state conditions, equating the two we get

$$\begin{aligned} \frac{d\dot{Q}_r}{dr} &= 0 \\ \Rightarrow \quad \frac{d}{dr} \left(r \frac{dT}{dr} \right) &= 0 \end{aligned} \quad (1.51)$$

Using the boundary conditions

- (i) $T = T_i$ at $r = r_1$
- (ii) $T = T_o$ at $r = r_0$

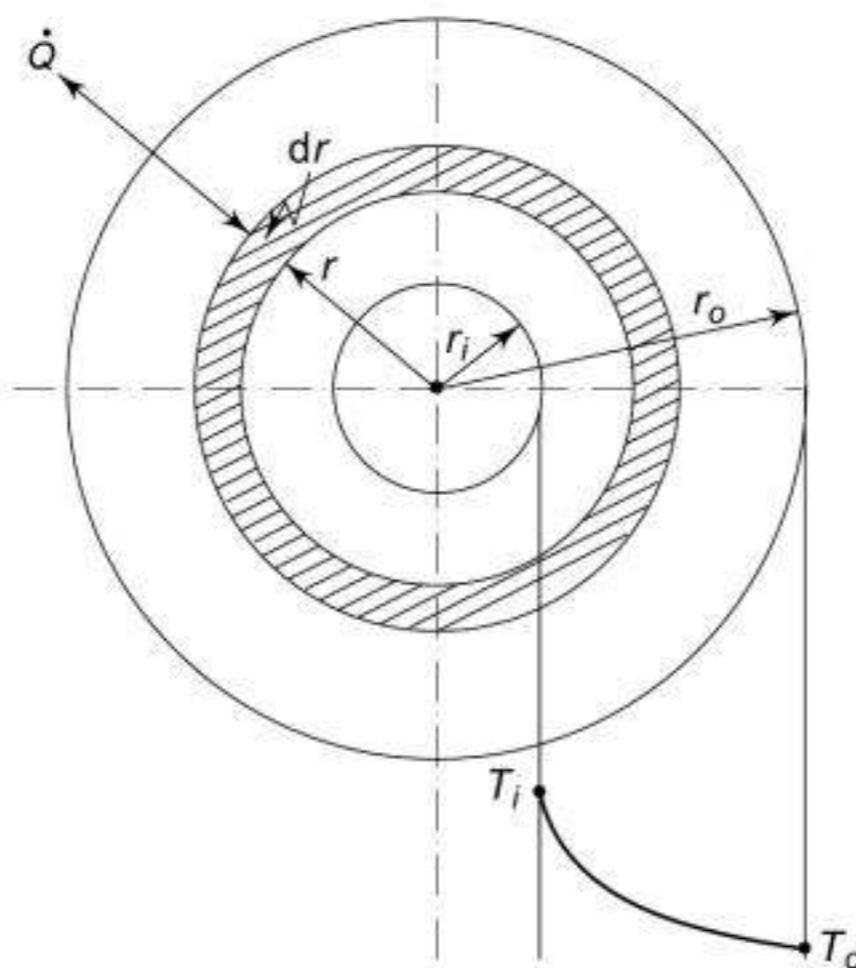


Fig. 1.20 Heat conduction through a hollow cylinder

and solving Eq. (1.51) we get for the temperature distribution

$$\frac{T - T_o}{T_i - T_o} = \frac{\ln\left(\frac{r}{r_o}\right)}{\ln\left(\frac{r_i}{r_o}\right)} \quad (1.51a)$$

and for heat flow we have Eq. (1.52)

$$\dot{Q} = -kA \frac{dT}{dr} = -k(2\pi rL) \frac{dT}{dr}$$

$$= \frac{T_i - T_o}{\left(\frac{1}{2\pi kL} \right) \ln \left(\frac{r_o}{r_i} \right)} \quad (1.52)$$

The thermal resistance of the cylindrical shell is then as follows in Eq. (1.53)

$$R = \frac{\ln \left(\frac{r_o}{r_i} \right)}{2\pi kL} \quad (1.53)$$

1.19.3 Heat Flow Through a Composite Wall with Convection Boundaries

Consider a wall comprising of more than one material and convection at the two surfaces as shown in Fig. 1.21. It is seen that all the thermal resistances are in series. An equivalent electrical analogue of the wall is also shown in the figure. The overall thermal resistance is given by Eq. (1.54)

$$R = \frac{1}{h_o A} + \frac{\Delta x_A}{k_A A} + \frac{\Delta x_B}{k_B A} + \frac{\Delta x_C}{k_C A} + \frac{1}{h_i A} \quad (1.54)$$

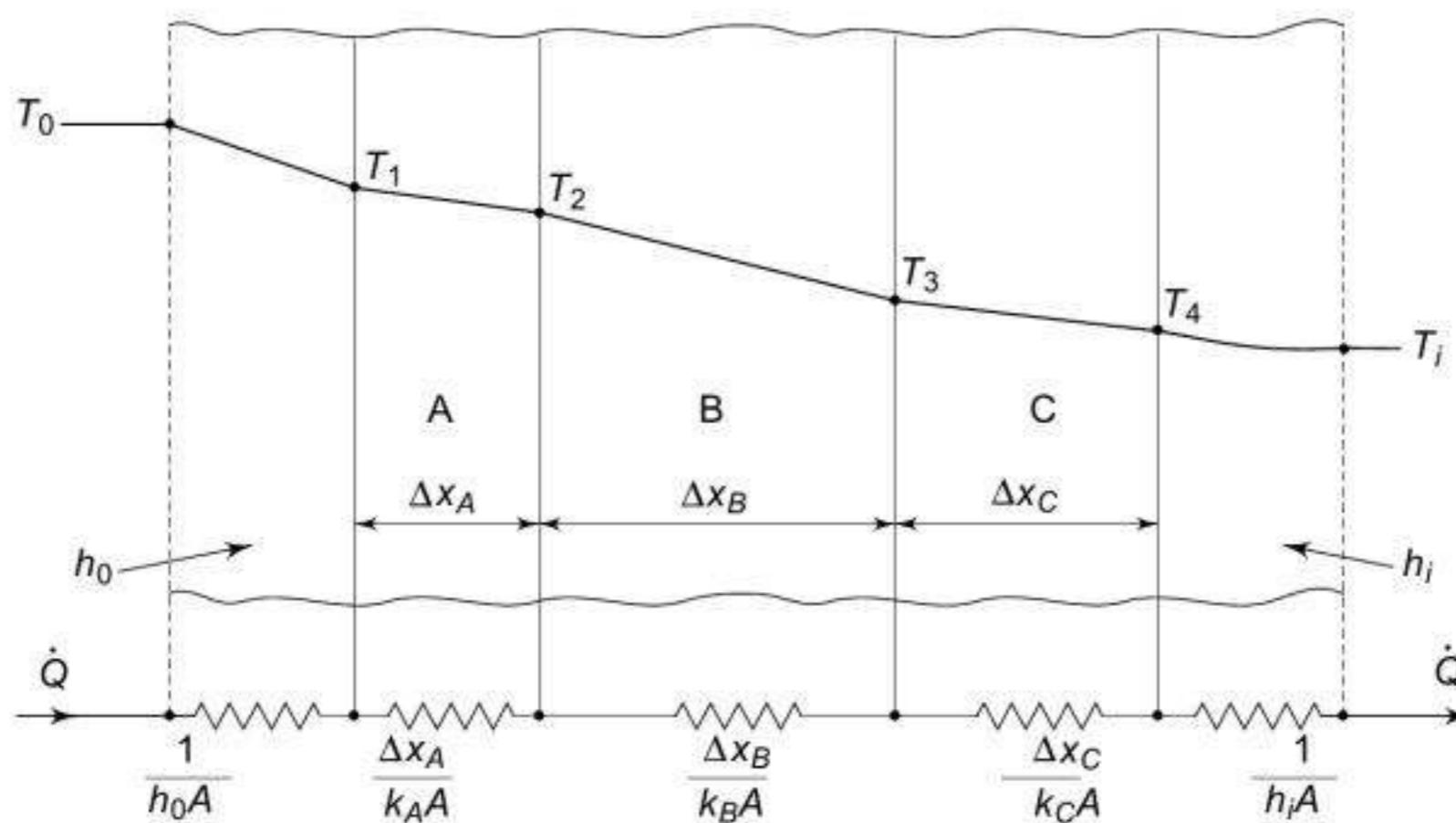


Fig. 1.21 Composite plane wall with resistances in series

where subscripts *A*, *B* and *C* refer to the three materials of the wall and h_i and h_o are the convective heat transfer coefficients between the inside and outside wall surfaces and surrounding air.

Similarly, if some of the resistances are in parallel while some are in series as shown in Fig. 1.22, then the overall thermal resistance is given by

$$R = \frac{1}{h_o A} + R_A + \left[\frac{1}{\frac{1}{R_B} + \frac{1}{R_C} + \frac{1}{R_D}} \right] + R_E + \frac{1}{h_i A} \quad (1.55)$$

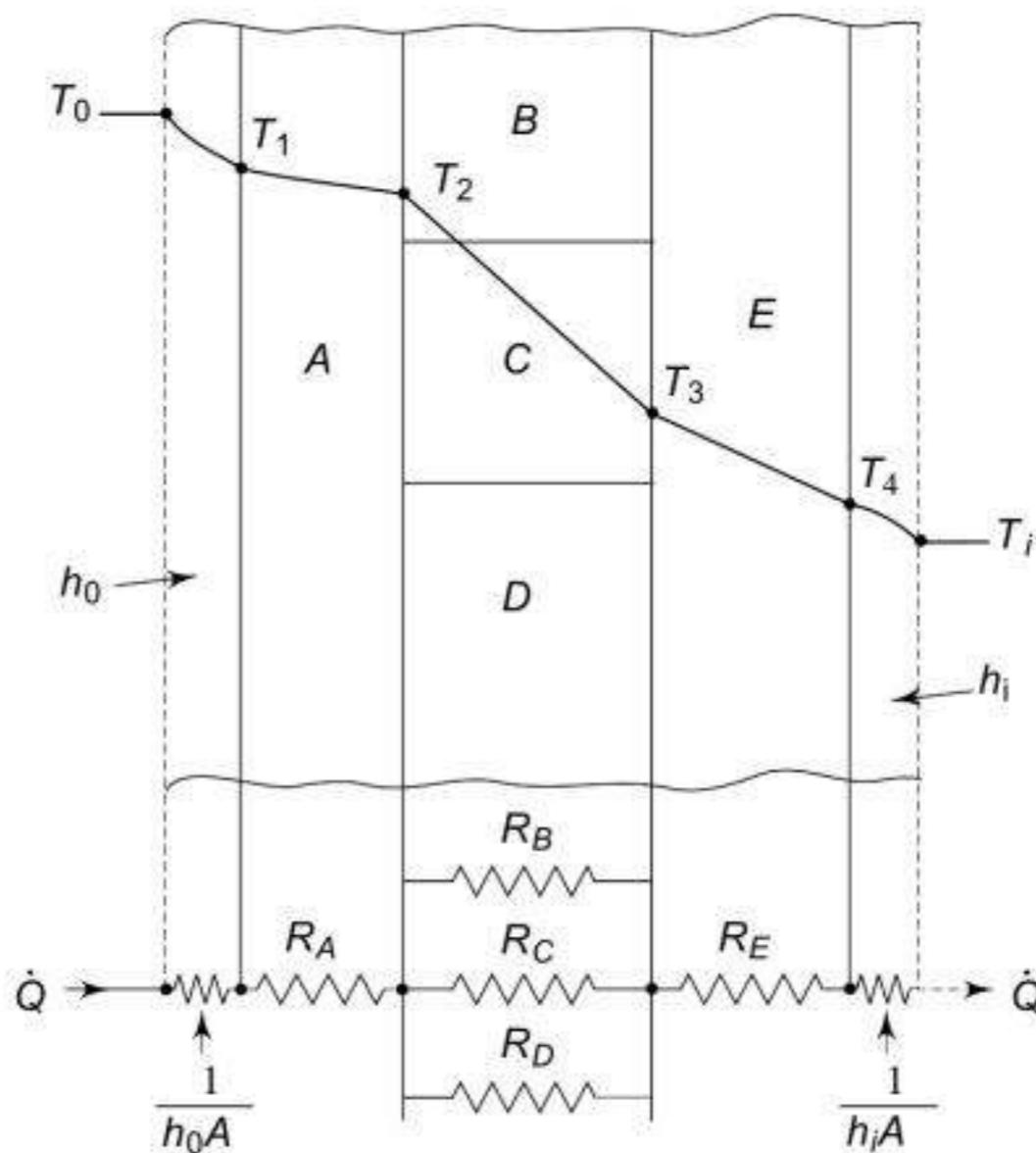


Fig. 1.22 Composite plane wall with resistances in series as well as in parallel

In both cases, if T_o and T_i are the outside and inside air temperatures respectively, then the heat flow rate is given by

$$\dot{Q} = \frac{T_o - T_i}{R}$$

Example 1.14 An exterior wall of a house consists of 10.2 cm brick and 3.8 cm gypsum plaster. What thickness of loosely-packed rockwool insulation should be added to reduce the heat transfer through the wall by 80 per cent?

The thermal conductivities of brick, gypsum plaster and rockwool are 0.7, 0.48 and $0.065 \text{ W.m}^{-1} \text{ K}^{-1}$ respectively.

Solution Since ΔT is the same in both cases, we have $(\dot{Q}/R)' = 0.2 (\dot{Q}/R) = \Delta T$

$$\Rightarrow \frac{\dot{Q}' \text{ with insulation}}{\dot{Q} \text{ without insulation}} = 0.2 = \frac{R \text{ without insulation}}{R' \text{ with insulation}}$$

The resistances of brick and plaster are (per unit area of wall)

$$R_{\text{brick}} = \frac{0.102}{0.7} = 0.145 \text{ m}^2 \cdot \text{K.W}^{-1}$$

$$R_{\text{plaster}} = \frac{0.038}{0.48} = 0.079 \text{ m}^2 \cdot \text{K.W}^{-1}$$

Then the resistance without insulation is

$$\begin{aligned} R &= R_{\text{brick}} + R_{\text{plaster}} \\ &= 0.145 + 0.079 = 0.224 \text{ m}^2 \cdot \text{K.W}^{-1} \end{aligned}$$

and the resistance with insulation is

$$R' = \frac{R}{0.2} = \frac{0.224}{0.2} = 1.122 \text{ m}^2 \cdot \text{K.W}^{-1}$$

Hence, the resistance of rockwool is

$$\begin{aligned} R_{\text{rockwool}} &= R' - R \\ &= 1.122 - 0.224 = 0.898 \text{ m}^2 \cdot \text{K.W}^{-1} \end{aligned}$$

The required thickness of rockwool insulation is, therefore,

$$\begin{aligned} \Delta x_{\text{rockwool}} &= (k A R)_{\text{rockwool}} \\ &= (0.065) 1 (0.898) = 0.0585 \text{ m} (\approx 6 \text{ cm}) \end{aligned}$$

1.19.4 Heat Flow Through a Composite Cylinder

Consider a composite tube of three materials with fluids flowing inside as well as outside the tube as shown in Fig. 1.23. Summing up all the resistances which are in series, we obtain Eq. (1.56)

$$R = \frac{1}{h_i A_i} + \frac{\ln \frac{r_2}{r_1}}{2 \pi k_A L} + \frac{\ln \frac{r_3}{r_2}}{2 \pi k_B L} + \frac{\ln \frac{r_4}{r_3}}{2 \pi k_C L} + \frac{1}{h_o A_o} \quad (1.56)$$

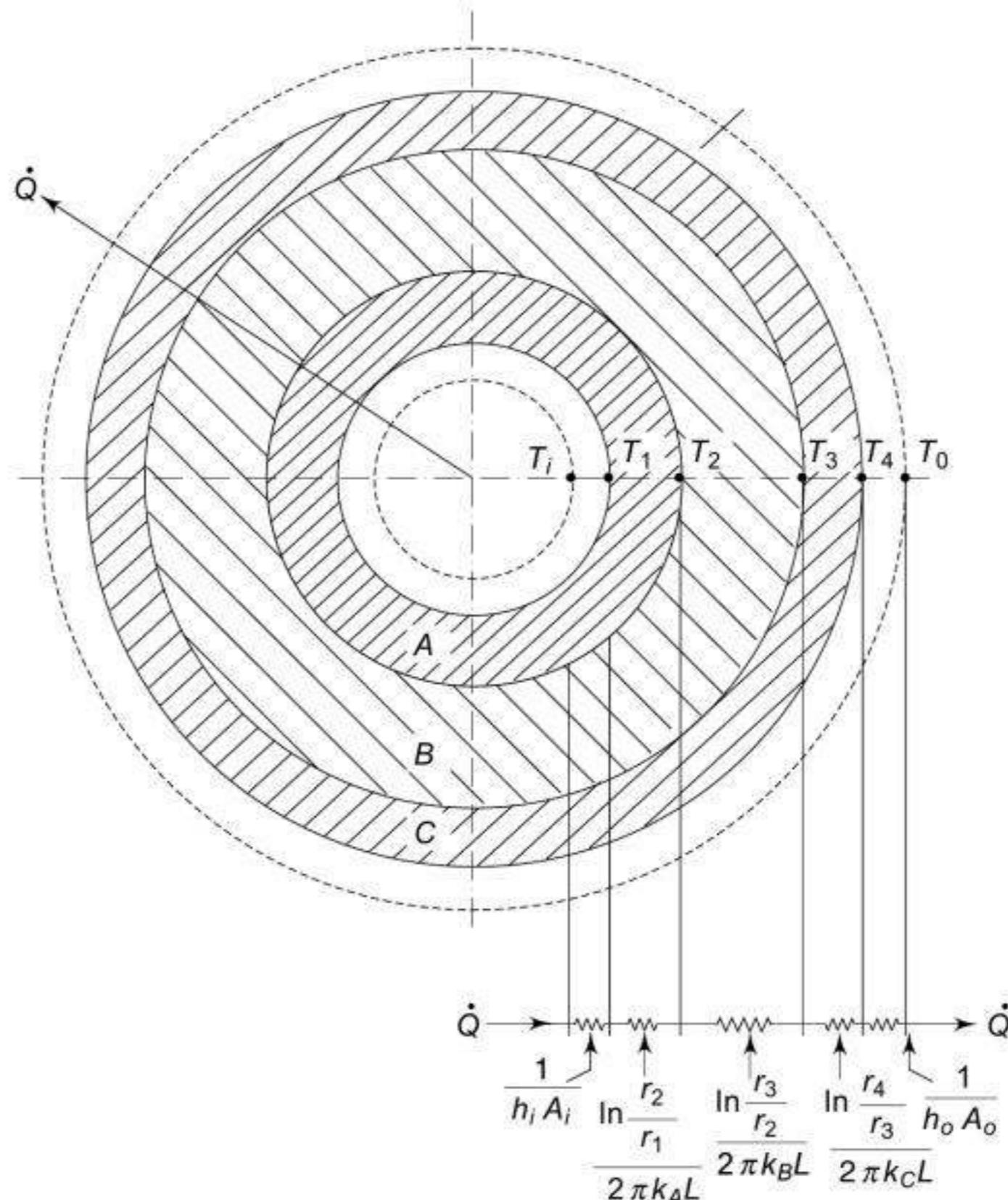


Fig. 1.23 Heat flow through a composite cylinder with convection boundaries

For the case of a single tube as in heat exchangers, this becomes

$$R = \frac{1}{h_i A_i} + \frac{\ln \frac{r_o}{r_i}}{2\pi k L} + \frac{1}{h_o A_o} \quad (1.57)$$

where A_o and A_i are the outside and inside tube surface areas respectively. Then the heat flow rate can be determined using the expression in Eq. (1.57).

Example 1.15 A 3 cm OD pipe is to be covered with two layers of insulation, each having a thickness of 2.5 cm. The average thermal conductivity of one insulation is five times that of the other. Determine the percentage decrease in heat transfer if the better insulating material is next to the pipe than if it is the outer layer.

Assume that the outside and inside surface temperatures of the composite insulation are fixed.

Solution Let the thermal conductivity of the poorer insulation be $k_1 = 5k$, and that of the better insulation be $k_2 = k$.

Case I: Better insulating material next to pipe

$$\begin{aligned} \dot{Q}_I &= \frac{\Delta T}{\frac{1}{2\pi k_1 L} \ln \frac{r_2}{r_1} + \frac{1}{2\pi k_2 L} \ln \frac{r_3}{r_2}} \\ &= \frac{2\pi L \Delta T}{\frac{1}{k} \ln \frac{4}{2.5} + \frac{1}{5k} \ln \frac{65}{4}} = \frac{2\pi k L \Delta T}{1.0774} \end{aligned}$$

Case II: Poorer insulating material next to pipe

$$\dot{Q}_{II} = \frac{2\pi L \Delta T}{\frac{1}{5k} \ln \frac{4}{2.5} + \frac{1}{k} \ln \frac{65}{4}} = \frac{2\pi k L \Delta T}{0.683}$$

Percentage reduction in heat flow

$$\frac{\dot{Q}_{II} - \dot{Q}_I}{\dot{Q}_{II}} \times 100 = \frac{\frac{1}{0.683} - \frac{1}{1.0774}}{\frac{1}{0.683}} \times 100 = 36.6\%$$

1.19.5 Overall Heat Transfer Coefficient

In order to calculate the rate of heat flow through a combination of resistances, the concept of *overall heat transfer coefficient* is introduced. Denoting it by the symbol U we express it by the relation

$$\dot{Q} = UA \Delta T = \frac{\Delta T}{R}$$

so that

$$\frac{1}{UA} = R$$

For a plane composite wall with all resistances in series as in Fig. 1.21, we find by comparison with Eq. (1.54) that

$$\begin{aligned}\frac{1}{UA} &= \frac{1}{h_i A} + \frac{\Delta x_A}{k_A A} + \frac{\Delta x_B}{k_B A} + \dots + \frac{1}{h_o A} \\ \Rightarrow \quad \frac{1}{U} &= \frac{1}{h_i} + \frac{\Delta x_A}{k_A} + \frac{\Delta x_B}{k_B} + \dots + \frac{1}{h_o} \quad (1.58)\end{aligned}$$

For a composite cylinder, the overall heat transfer coefficient can be based on either the outside or the inside tube surface area. Thus, we have

$$\dot{Q} = U_o A_o \Delta T = U_i A_i \Delta T = \frac{\Delta T}{R}$$

Then, by comparison with Eq. (1.56), we obtain

$$\frac{1}{U_o A_o} = \frac{1}{U_i A_i} = \frac{1}{h_i A_i} + \frac{\ln \frac{r_2}{r_1}}{2\pi k_A L} + \frac{\ln \frac{r_3}{r_2}}{2\pi k_B L} + \dots + \frac{1}{h_o A_o} \quad (1.59)$$

Equation (1.59) may be used to determine U_o or U_i . We see that $U_o A_o = U_i A_i$.

Example 1.16 (a) Find the overall heat transfer coefficient of a flat built-up roof having the construction shown in Fig. 1.24.

(b) Find the value of U if rigid roof deck insulation of resistance $R = 0.76 \text{ K.W}^{-1}$ is added to this construction.

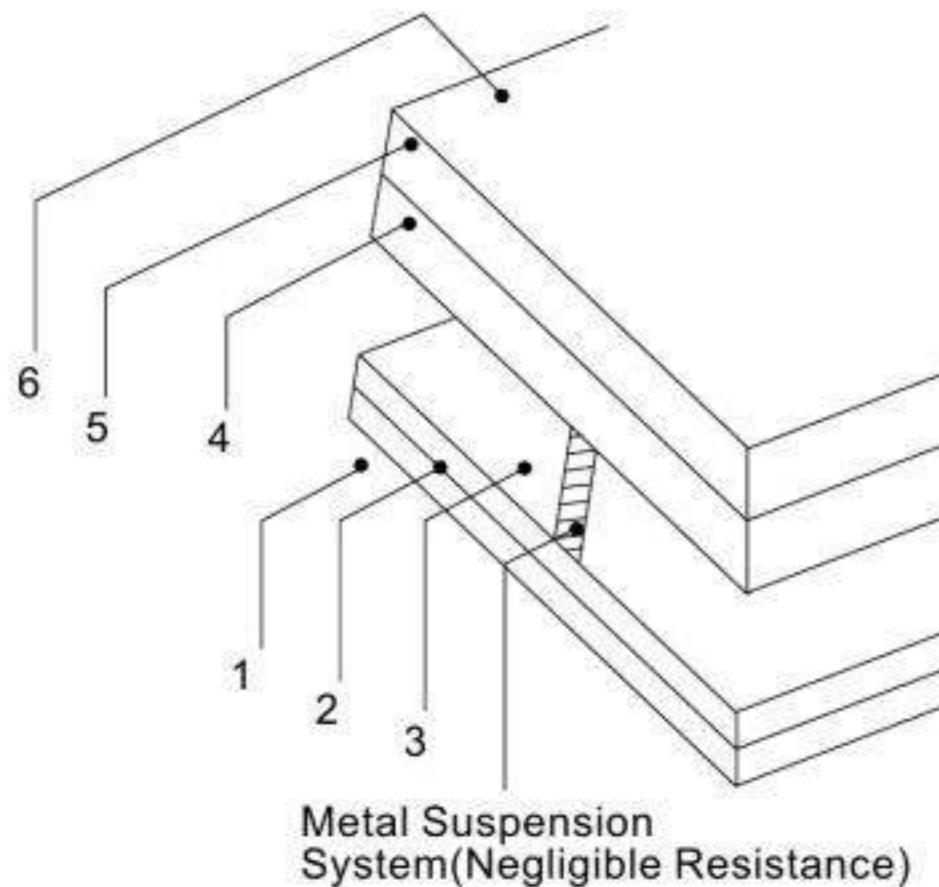


Fig. 1.24 Built-up roof construction for example 1.16

Solution

- (a) The values of resistances per unit area as found from ASHRAE Handbook³ are as follows:

Resistance No.	Construction material	Unit resistance, $\Delta x/k$ or $1/h \text{ m}^2 \cdot \text{K.W}^{-1}$
1.	Inside surface (still air)	0.107
2.	Metal lath and 0.75 inch plaster	0.083
3.	Air space (greater than 9 cm width)	0.164
4.	Concrete slab, 5 cm	0.391
5.	Built-up roofing, 9.5 cm	0.058
6.	Outside surface (20 kmph wind velocity)	0.03
Total thermal resistance, R_a		0.833

Overall heat transfer coefficient for construction 'a'

$$U_a = \frac{1}{R_a} = \frac{1}{0.833} = 1.2 \text{ W.m}^{-2} \cdot \text{K}^{-1}$$

(b) Total thermal resistance for construction 'b'

$$R_b = R_a + 0.76 = 0.833 + 0.76 = 1.593 \text{ m}^2 \cdot \text{K.W}^{-1}$$

Overall heat transfer coefficient for construction 'b'

$$U_b = \frac{1}{R_b} = \frac{1}{1.593} = 0.68 \text{ W.m}^{-2} \cdot \text{K}^{-1}$$



1.20 HEAT TRANSFER FROM EXTENDED SURFACE

A *heat exchanger* is an apparatus which affects the transfer of heat from one fluid to another. The overall heat transfer coefficient of a heat exchanger surface is determined principally by the greatest single resistance. As an illustration, neglecting the thermal resistance of the metal wall of a heat exchanger, the overall heat transfer coefficient between the two fluids is given by

$$\frac{1}{U A} = \frac{1}{h_i A_i} + \frac{1}{h_o A_o}$$

Taking $A_i = A_o = A$, and values of h_i and h_o as 1000 and $10 \text{ W.m}^{-2} \cdot \text{K}^{-1}$ respectively, we see that

$$\frac{1}{U} = \frac{1}{1000} + \frac{1}{10} = 0.101$$

$$\Rightarrow U = 9.9 \text{ W.m}^{-2} \cdot \text{K}^{-1}$$

so that the value of U is less than the value of the lower of the two heat transfer coefficients, viz., $h_o = 10$, in this case. The lower coefficient is, therefore, the *controlling coefficient*. The influence of the higher coefficient is only marginal. The size of the heat exchanger will thus be uneconomically large. There are two methods by which the heat exchange can be improved.

One is to augment the lower heat transfer coefficient.

The other is to employ an extended surface on the side of the lower coefficient.

Both methods decrease the thermal resistance, one by increasing h , and the other by increasing A .

Consider an extended surface in the form of a thin rod protruding from a surface into a surrounding fluid as shown in Fig. 1.25. The *root* or *base* of the rod is at temperature T_1 while the fluid temperature is T_∞ . The length, cross-sectional area and perimeter of the rod are l , A and P respectively. The temperature distribution along the rod is also shown in Fig. 1.25.

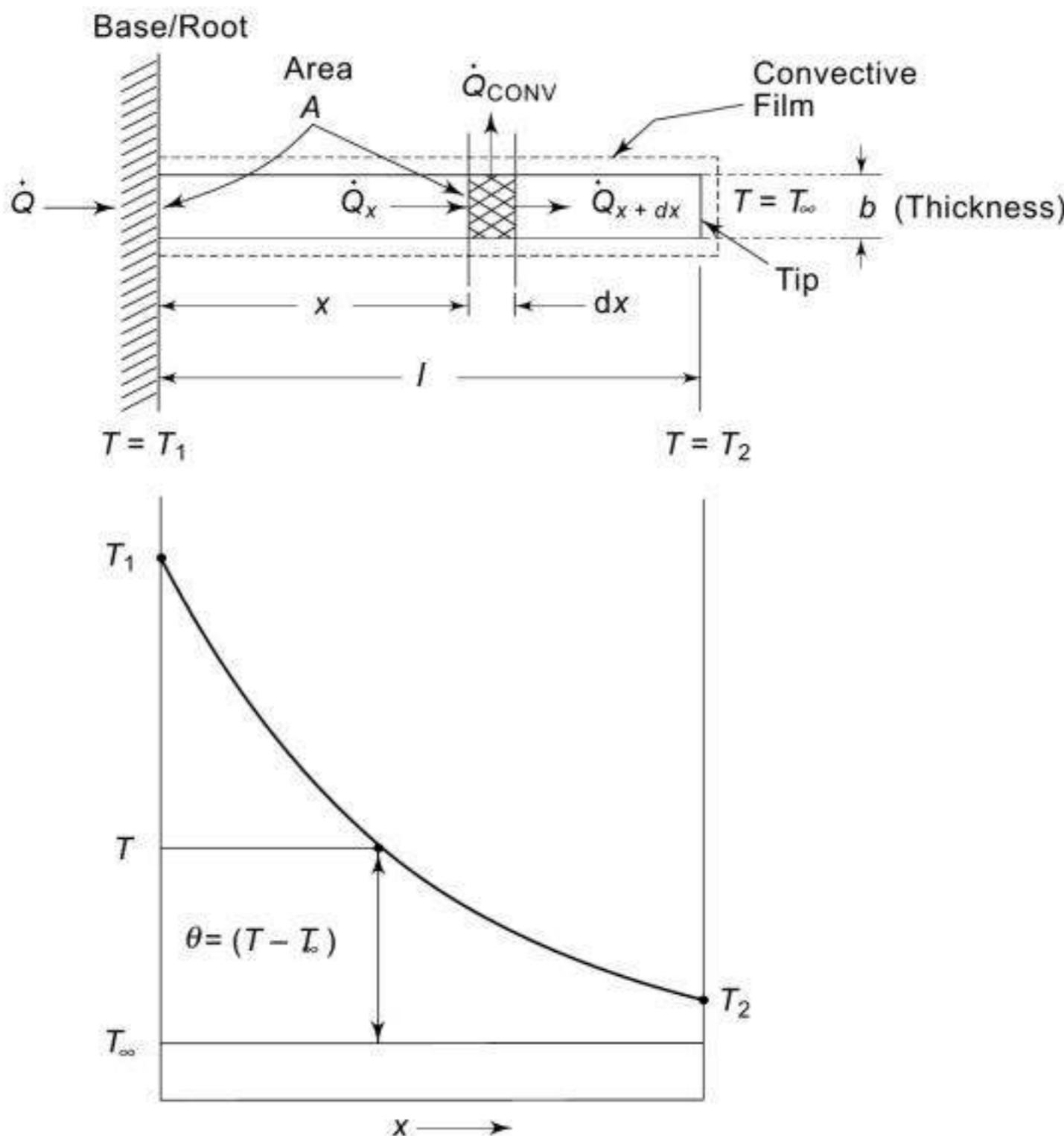


Fig. 1.25 Heat flow and temperature distribution along the length of a rod protruding from a surface

Consider an element dx at a distance x from the base of rod. The heat flows by conduction within the rod, and by convection from its surface to the surrounding fluid. The energy balance over the element dx under steady-state gives

$$\dot{Q}_x - \dot{Q}_{x+dx} = \dot{Q}_{\text{conv}} \\ \left(-kA \frac{dT}{dx} \right)_x - \left(-kA \frac{dT}{dx} \right)_{x+dx} = h(Pdx)(T - T_\infty) \quad (1.60)$$

where T is the temperature of the rod at x , k is the thermal conductivity of the rod, and h is the heat transfer coefficient from the surface to the fluid. Equation (1.60) can be simplified to

$$\frac{d^2\theta}{dx^2} = m^2\theta = 0 \quad (1.61)$$

by putting $\theta = \text{excess temperature} = (T - T_\infty)$,

and

$$m = \sqrt{\frac{hP}{kA}}$$

Using the boundary conditions

- (i) $\theta = T_1 - T_\infty = \theta_1$ at $x = 0$, and
- (ii) $\frac{d\theta}{dx} = 0$ at $x = l$ (no heat conduction at tip)

we obtain the solution of Eq. (1.61) for temperature distribution as

$$\frac{\theta}{\theta_1} = \frac{\cosh m(l - x)}{\cos ml} \quad (1.62)$$

Then the heat flow from the base of the rod (at $x = 0$) is given by Eq. (1.63)

$$\begin{aligned} \dot{Q}_1 &= -kA \left| \frac{d\theta}{dx} \right|_{x=0} \\ &= mkA\theta_1 \tanh ml \\ &= \sqrt{hPkA} \theta_1 \tanh ml \end{aligned} \quad (1.63)$$

The term $\sqrt{hPkA} \theta_1 = mkA\theta_1$ is constant. The value of $\tanh ml$ increases with ml and tends to unity. It becomes 0.964 at $ml = 2$ and 0.9951 at $ml = 3$ as seen from Table 1.3. It is, therefore, apparent that increasing the length of an extended surface beyond $ml = 2$ or 3 does not improve the heat flow.

For a *rectangular fin* of height l , width L and thickness b , in place of a rod, as shown in Fig. 1.25 the cross-sectional area and perimeter are

$$A = bL, P = 2L + 2b = 2(L + b) \approx 2L$$

so that the parameter m is

$$m = \sqrt{\frac{hP}{kA}} = \sqrt{\frac{2h(L + b)}{kbL}} \cong \sqrt{\frac{2h}{kb}}$$

The entire fin is not at temperature T_1 . It drops from T_1 at the root to T_2 at the tip. Hence, the whole fin surface is not equally effective.

The fin efficiency η_f is now defined as the ratio of the actual heat transferred by a fin to that which would be transferred if the entire fin surface were assumed to be at the base temperature. Thus

$$\eta_f = \frac{\dot{Q}_1 \text{ with fin}}{\dot{Q} \text{ with fin surface at base temperature}}$$

For a rectangular fin, therefore,

$$\eta_f = \frac{mkA\theta_1 \tanh ml}{h(Pl)\theta_1} = \frac{\tanh ml}{ml}$$

Table 1.3 gives the values of $\tanh ml$ and η_f as a functions of ml . It is seen that $\tanh ml$, and hence \dot{Q}_1 increase rapidly at first as l increases. The increase, then, slows down finally reaching an asymptotic value at $ml \approx 3$. The fin efficiency is quite high upto $ml \approx 0.5$, but decreases rapidly as ml increases.

Table 1.3 Numerical values of $\tanh ml$ and η_f

ml	0	0.5	1.0	1.5	2	3	4	5	6
$\tanh ml$	1	0.4621	0.7616	0.9052	0.964	0.995	0.9993	0.9999	1
η_f		0.924	0.7616	0.603	0.482	0.332	0.2498	0.20	0.167

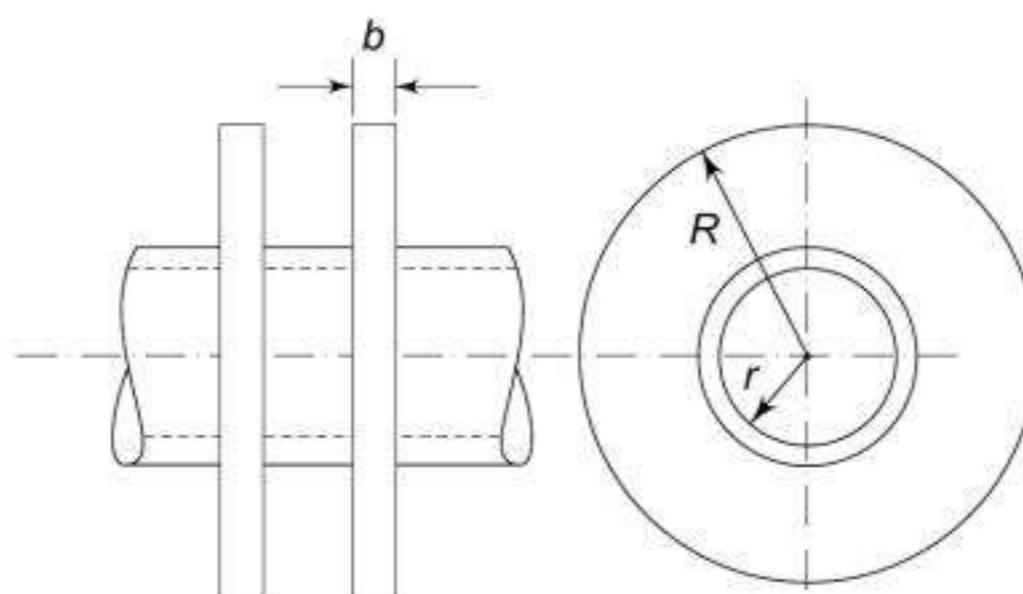
1.20.1 Efficiency of Circular Fins and Finned Tube Arrays

A largely empirical method, developed by Schmidt¹⁵ for finding the efficiency of a circular fin the configuration of which is shown in Fig. 1.26, is summarized as follow in Eqs. (1.64) and (1.65).

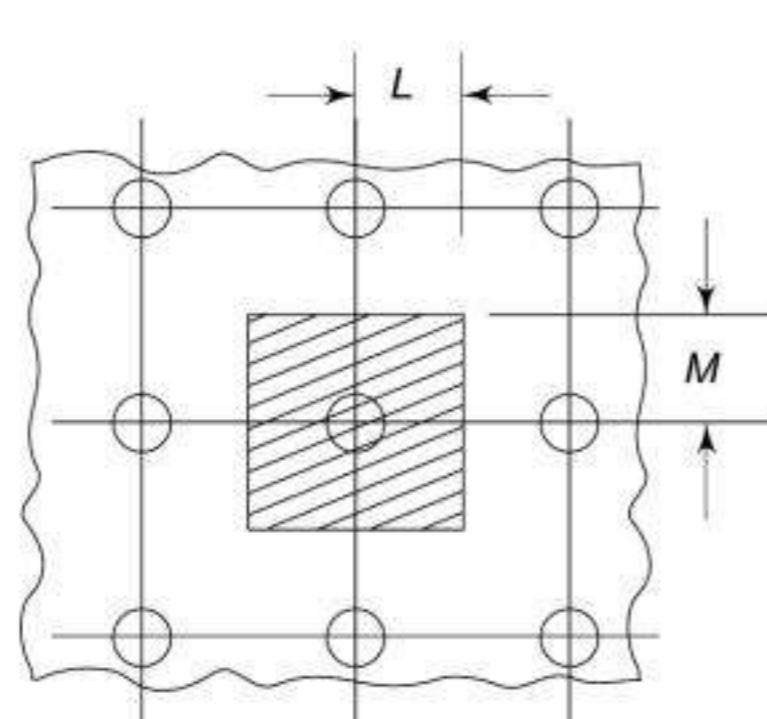
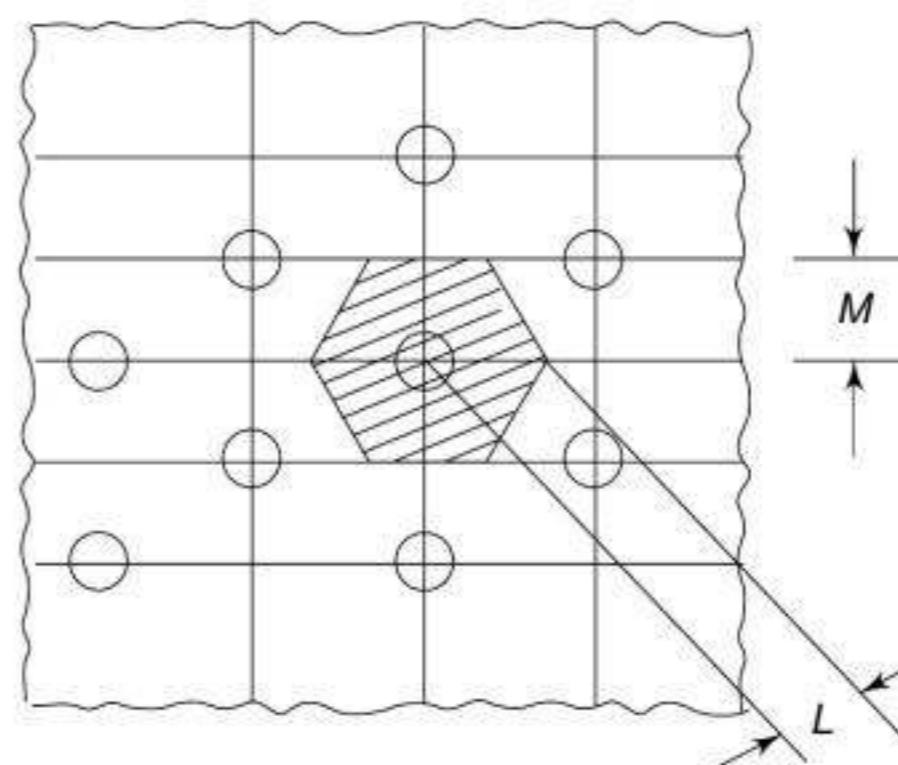
$$\eta_f = \frac{\tanh (mr\phi)}{(mr\phi)} \quad (1.64)$$

where

$$\phi = \left(\frac{R}{r} - 1 \right) \left[1 + 0.35 \ln \left(\frac{R}{r} \right) \right] \quad (1.65)$$

**Fig. 1.26** Circular fins

However, in air conditioning, continuous plate fins are used in finned tube cooling coils. The two array configurations, viz., the rectangular tube array and the angular tube array (hexangular fin) are shown in Figs 1.27 and 1.28 respectively.

**Fig. 1.27** Rectangular tube array fin**Fig. 1.28** Triangular tube array hexangular fin

Schmidt, again, provides an empirical method for calculating the efficiency of these fins. The method is based on selecting a circular fin with an equivalent radius R_e that has the same fin efficiency as the rectangular fin as follows in Eqs. (1.66) and (1.67).

For the Rectangular Tube Array Fin

$$\frac{R_e}{r} = 1.28 \psi(\beta - 0.2)^{1/2} \quad (1.66)$$

For the Triangular Tube Array Hexangular Fin

$$\frac{R_e}{r} = 1.27 \psi(\beta - 0.3)^{1/2} \quad (1.67)$$

In both the above expressions, from Figs. (1.27) and (1.28)

$$\psi = \frac{M}{r} \text{ and } \beta = \frac{L}{M}$$

in which L is always selected to be greater than or equal to M .

1.21 UNSTEADY-STATE CONDUCTION

In unsteady-state conduction, the temperature of the body changes with time. This change in temperature is represented by $\partial T / \partial \tau$ where τ stands for time. Thus, in the case of a plane wall element dx (Fig. 1.19), we have

$$\dot{Q}_x - \dot{Q}_{x+dx} = C_p (A dx) \frac{\partial T}{\partial \tau} \quad (1.68)$$

where ρ is the density, and C_p is the specific heat of the wall.

The right hand side in Eq. (1.68) represents the increase in stored energy of the element. Substituting for \dot{Q}_x and \dot{Q}_{x+dx} from Eqs (1.45) and (1.46) we obtain the unsteady-state heat conduction equation in one dimension as

$$\frac{\partial^2 T}{\partial x^2} = \frac{1}{\alpha} \frac{\partial T}{\partial \tau} \quad (1.69)$$

where $\alpha = k/\rho C_p$ is called the *thermal diffusivity* of the wall material. It will be seen that the units of thermal diffusivity are $m^2 s^{-1}$.

Various analytical methods have been used to solve Eq. (1.69). These include, among others, *the methods of separation of variables* and *the method of transformation of coordinates*. In all cases, the solutions are of the following dimensionless form

$$\frac{\theta}{\theta_i} = f \left(\frac{x}{L}, \text{Bi}, \text{Fo} \right)$$

where

θ = excess temperature at any time τ at $x = t - t_{\text{reference}}$

θ_i = initial excess temperature at x

x/L = dimensionless distance

$$\text{Bi} = \text{Biot number} = \frac{hL}{k}$$

$$\text{Fo} = \text{Fourier number} = \frac{\alpha \tau}{L^2}$$

Here h represents the heat transfer coefficient at the surface and L is the distance between the centre line of the material and the surface. Many charts, such as those of Gurnie-Lurie and Heisler⁵, present the solution of Eq. (1.69) in graphical form. In addition to analytical methods, there are also numerical methods employed to solve this equations. One such method using *finite difference approach* has been employed in Chapter 18.



1.22 FORCED CONVECTION CORRELATIONS

Correlations for the heat transfer coefficient h in forced convection are expressed in the following form

$$\text{Nu} = f(\text{Re}, \text{Pr})$$

where Nu, Re and Pr are dimensionless numbers expressed as follows:

$$\text{Nu} = \text{Nusselt number} = \frac{hL}{k}$$

$$\text{Re} = \text{Reynolds number} = \frac{Lu\rho}{\mu} \quad \text{or} \quad \frac{Du\rho}{\mu}$$

$$\text{Pr} = \text{Prandtl number} = \frac{C_p \mu}{k}$$

Here L is any characteristic length in the flow geometry. In the case of pipes, it is equal to the diameter D . Also u is the mean flow velocity and k , ρ , C_p and μ are the thermal conductivity, density, specific heat and dynamic viscosity of the fluid respectively.

The Reynolds number represents the ratio of inertia forces to viscous forces. It is a flow criterion. Generally, for a pipe, the flow is streamlined or *laminar* when $\text{Re} < 2100$, and *turbulent* when $\text{Re} > 2300$. For intermediate values of Re, the flow is in the transition region.

For turbulent flow in smooth tubes, the following correlation is commonly used. *Dittus-Boelter equation*

$$\text{Nu} = 0.023 (\text{Re})^{0.8} (\text{Pr})^n \quad (1.70)$$

where

$$\begin{aligned} n &= 0.4 \text{ for heating} \\ &= 0.3 \text{ for cooling} \end{aligned}$$

For turbulent flow of fluids perpendicular to tubes such as water in the shell the following correlation (see Kern⁷) is used

$$\text{Nu} = 0.36 (\text{Re})^{0.55} \text{Pr}^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (1.71)$$

For determining Nu and Re, the outside diameter of the tube is taken as the characteristic length. The velocity is measured in the free stream. The properties of the fluid are taken at the bulk mean temperature, whereas μ_w is at the wall temperature.

1.23 FREE CONVECTION CORRELATIONS

In free or natural convection, the flow velocity is developed as a result of the temperature difference causing a buoyancy force to act on the fluid. The flow criterion in the case of free convection is, therefore, the *Grashof number* which is defined by

$$\text{Gr} = \frac{g \beta \rho^2 \Delta T L^3}{\mu^2}$$

in which β is the *coefficient of thermal expansion* of the fluid defined by

$$\beta = \frac{1}{v} \left(\frac{\partial v}{\partial T} \right)_p$$

For an ideal gas

$$v = \frac{RT}{p}, \left(\frac{\partial v}{\partial T} \right)_p = \frac{R}{p}, \text{ so that } \beta = \frac{1}{T}$$

Also, $\Delta T = (T_w - T_\infty)$ is the temperature difference between the wall and the surrounding fluid.

Free convection correlations are found to have the following form

$$\text{Nu} = f(\text{Gr}, \text{Pr})$$

In many cases, the relationship simplifies to

$$\text{Nu} = C (\text{Gr} \cdot \text{Pr})^m = C R_a^m$$

The product $(\text{Gr} \cdot \text{Pr})$ is named Rayleigh number (Ra). It is a criterion of transition from laminar to turbulent flow in free convection.

For laminar flow, $10^4 < \text{Gr} \cdot \text{Pr} < 10^9$

For turbulent flow, $\text{Gr} \cdot \text{Pr} > 10^9$

For the case of air, Eq. (1.72) is simplified as written below in Eqs (1.73a), (1.73b), (1.74a) and (1.74b).

Vertical plate or cylinder to air

$$h = 1.42 \left(\frac{\Delta T}{L} \right)^{1/4} \text{ laminar} \quad (1.73a)$$

$$h = 0.95 (\Delta T)^{1/3} \text{ turbulent} \quad (1.73b)$$

Horizontal cylinder to air

$$h = 1.32 \left(\frac{\Delta T}{D} \right)^{1/4} \text{ laminar} \quad (1.74a)$$

$$h = 1.24 \left(\frac{\Delta T}{D} \right)^{1/3} \text{ turbulent} \quad (1.74b)$$

1.24 DESIGN OF HEAT EXCHANGERS

The thermal design of a heat exchanger is primarily concerned with the determination of the heat transfer area required to transfer a specified amount of heat between two fluids with specified flow rates and inlet temperatures.

In a heat exchanger, usually, the temperatures of fluids on both sides of the heat transfer surface vary as a result of heat exchange except in the case of the fluid undergoing change of phase. This variation has to be taken into account in any design procedure.

There are three basic types of heat exchangers. They are:

- (i) Parallel flow.
- (ii) Counter flow.
- (iii) Cross flow.

In parallel flow, both the fluids flow past the heat transfer surface in the same direction as shown in Fig. 1.29. In counter flow, the two fluids flow in opposite directions as shown in Fig. 1.30. In cross flow, the two fluids flow at right angles to each other.

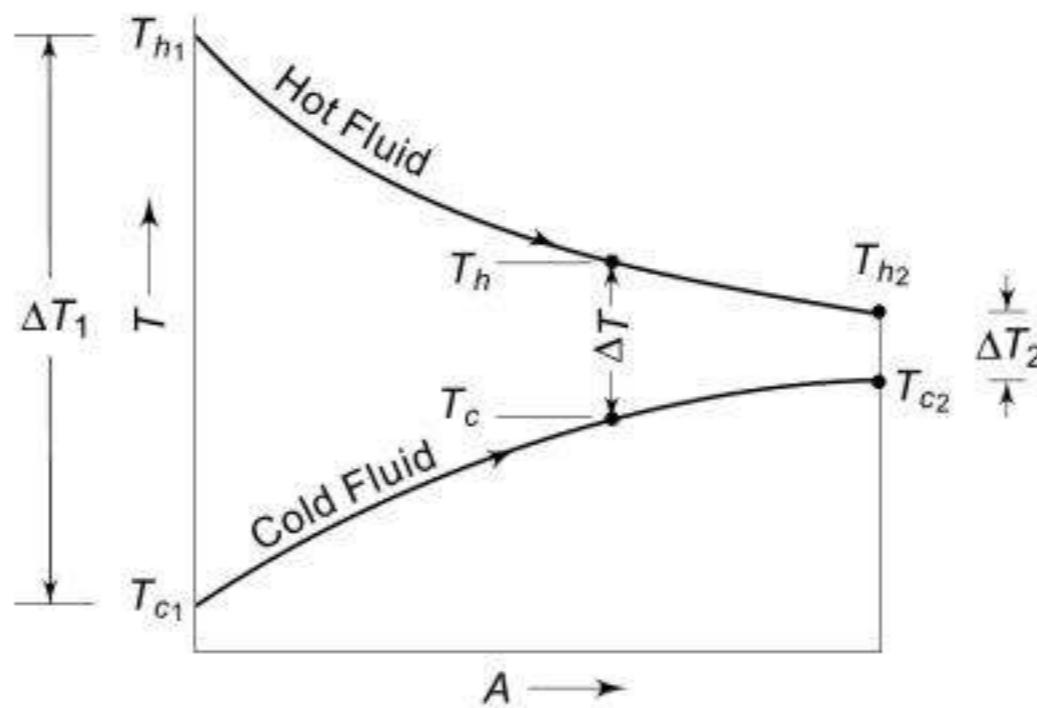


Fig. 1.29 Temperature variation in parallel flow heat exchanger

1.24.1 Log Mean Temperature Difference (LMTD) Method

It is proposed to calculate the required heat transfer surface area by the equation

$$\dot{Q} = UA \Delta T_m$$

where ΔT_m is the *log mean temperature difference* (LMTD) given by

$$\Delta T_m = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \quad (1.75)$$

where ΔT_1 and ΔT_2 are the temperature differentials between the two fluids at the two ends of the heat exchanger.

There are two great advantages of counter flow heat exchangers.

- (i) A counterflow heat exchanger has a higher value of LMTD, for the same *end temperatures* as obtained from Eq. (1.75) and seen from Fig. 1.30. Hence, it requires smaller heat transfer surface area A than parallel flow heat exchanger for a specified heat transfer rate.
- (ii) It can also be seen from Fig. 1.30 that in the case of counterflow heat exchanger, the hot fluid can be cooled to the inlet temperature of the cold fluid (from t_{h1} to t_{c1}), and similarly, cold fluid can be heated to the inlet temperature of the hot fluid (from t_{c1} to t_{h1}) at least theoretically in the limit, if large heat transfer surface area is provided. This is not possible in parallel flow heat exchanger.

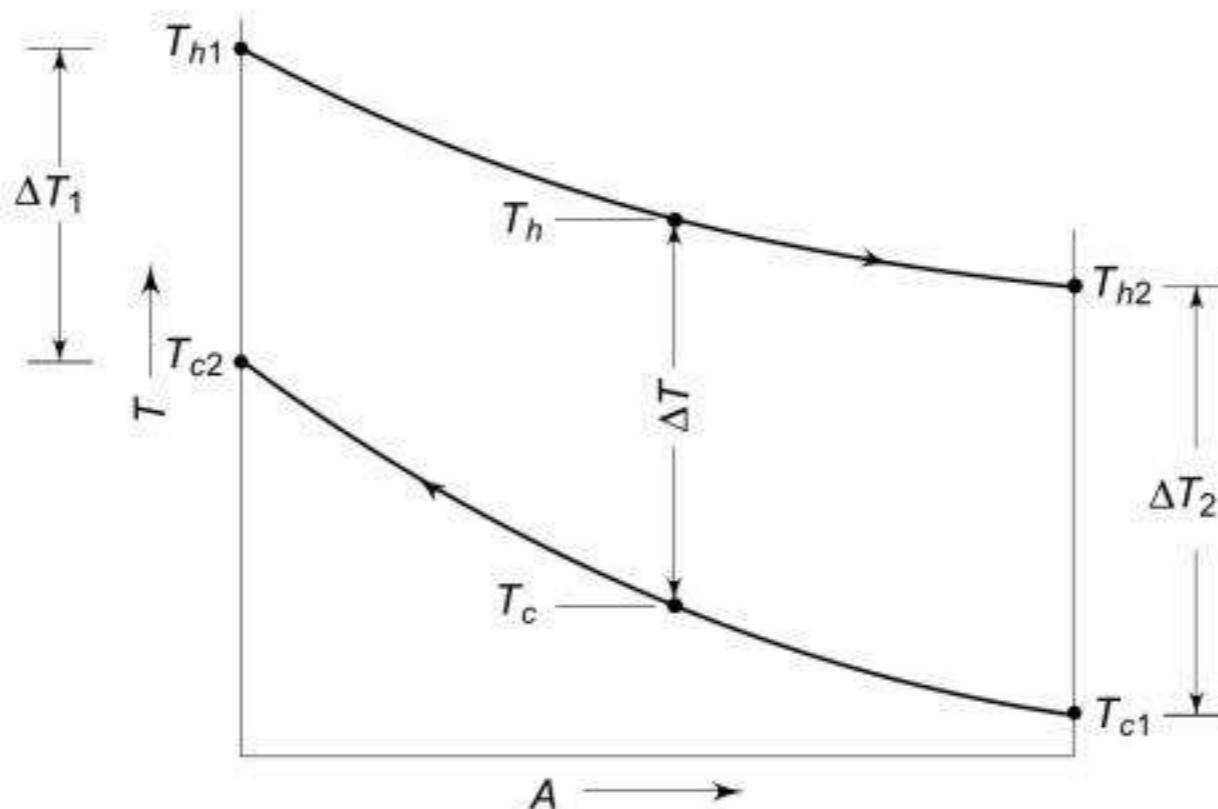


Fig. 1.30 Temperature variation in counter flow heat exchanger

Note In the case of phase change of a fluid, the temperature on one side is constant as in condensers and evaporators. In these cases, both types have the same LMTD, and there is no difference between counterflow and parallel flow.

1.25 MASS TRANSFER

In *mass transfer*, we deal with the movement of species in a multicomponent system. The driving potentials for the transfer of mass can be obtained in various ways. But here we shall confine our discussion to *molecular diffusion* and *convective mass transfer*.

1.25.1 Molecular Diffusion: Fick's Law

Mass transfer by molecular diffusion of a species through another stationary medium is analogous to heat transfer by conduction. *Fick's Law* relates the *diffusion rate* or *mass flow* \dot{m}_A of a species *A* to its driving potential which is the *concentration gradient* $\partial C_A / \partial x$ as follows in Eq. (1.76)

$$\frac{\dot{m}_A}{A} = -D \frac{\partial C_A}{\partial x} \quad (1.76)$$

where *A* is the normal area and *D* is called the *diffusion coefficient* or *mass transfer diffusivity*. The unit of diffusion coefficient is found to be $\text{m}^2 \text{s}^{-1}$. It has the dimensions of kinematic viscosity $\nu = \mu/\rho$ or thermal diffusivity $\alpha = k/\rho C$.

1.25.2 Convective Mass Transfer

In molecular diffusion, bulk velocities are insignificant. Most physical applications of mass transfer, however, involve the bulk motion of fluids. This gives rise to convective mass transfer which is similar to convective heat transfer. Accordingly, we define the *mass transfer coefficient* h_M by the relation

$$\frac{\dot{m}_A}{A} = h_M (C_w - C_\infty) = \frac{k_\omega}{\rho} (\omega_w - \omega_\infty) \quad (1.77)$$

where C_w and C_∞ denote the concentrations of the species A at the wall and free stream. We see that h_M has the units of m s^{-1} . The expression ρh_M is replaced by *diffusion coefficient* k_ω and C by specific humidity ω in the case of diffusion of water vapour into air. Its use is illustrated in Chapters 14 and 20. It is seen that $k_\omega = \rho h_M$ if $\Delta C = \Delta \omega$, which is a satisfactory assumption (see Chap. 14).

1.26 ANALOGY BETWEEN MOMENTUM, HEAT AND MASS TRANSFER

D has the same significance in mass transfer as ν in momentum transfer and α in heat transfer. Dimensionless numbers can be formed from the ratio of any two of these properties.

Prandtl Number (as defined earlier for convective heat transfer)

$$\text{Pr} = \frac{\nu}{\alpha} = \frac{C_p \mu}{k}$$

Lewis Number

$$\text{Le} = \frac{\alpha}{D} = \frac{k}{\rho C_p D}$$

Schmidt number

$$\text{Sc} = \frac{\nu}{D} = \frac{\mu}{\rho D} = \frac{\text{Dynamic viscosity}}{\text{Mass diffusivity}}$$

where the product ρD is termed as *mass diffusivity*. It has the units of $\text{kg m}^{-1} \text{s}^{-1}$.

1.26.1 Analogy between Momentum and Heat Transfer

Then if $\nu = \alpha$ or $\text{Pr} = 1$, the velocity and temperature distributions in the flow will be the same. In that case, the heat transfer coefficient h can be determined from the knowledge of the friction factor f by the similarity relation called *Reynolds analogy*, viz.,

$$\text{St} = \frac{f}{2} \quad (1.78)$$

where St denotes the *Stanton number* defined by

$$\text{St} = \frac{\text{Nu}}{\text{Re} \cdot \text{Pr}} = \frac{h}{\rho u C_p} = \frac{h}{G C_p}$$

Here $G = \rho u$ is the *mass velocity*.

To account for some variation in Pr , Eq. (1.78) is modified to

$$j_H = \text{St} \cdot \text{Pr}^{2/3} = \frac{f}{2} \left(= \frac{h}{G C_p} \text{Pr}^{2/3} \right) \quad (1.79)$$

which is called the *Colburn analogy*, and j_H is the *Colburn j-factor* for heat transfer.

1.26.2 Analogy between Momentum and Mass Transfer

Similarly, the velocity and concentration profiles will have the same shape if $\nu = D$, or $\text{Sc} = 1$. Thus, the Schmidt number plays the same role in mass transfer as does the

Prandtl number in heat transfer. An equivalent of the Nusselt number in mass transfer is the *Sherwood number* defined by

$$\text{Sh} = \frac{h_M L}{D} = \frac{k_\omega L}{\rho D}$$

where L is any characteristic length. The convective mass transfer correlations are, therefore, expressed in the form

$$\text{Sh} = f(\text{Re}, \text{Sc})$$

similar to forced convection heat transfer correlations which are usually expressed in the form $\text{Nu} = f(\text{Re} \cdot \text{Pr})$.

The Stanton number can then be replaced by the number

$$\frac{\text{Sh}}{\text{Re} \cdot \text{Sc}} = \frac{h_M}{u} = \frac{\rho h_M}{G} \quad (1.80)$$

in mass transfer correlations, as given in Eq. (1.80). And similar to Eq. (1.78), we can write for mass transfer factor j_M as

$$j_M = \left(\frac{\text{Sh}}{\text{Re} \cdot \text{Sc}} \right) \text{Sc}^{2/3} = \left(\frac{\rho h_M}{G} \right) \text{Sc}^{2/3} = \frac{f}{2} \quad (1.81)$$

The numerical value of the Schmidt number for the diffusion of water vapour into air at 25°C and 1 atm pressure is 0.6.

1.26.3 Analogy between Heat and Mass Transfer

Combining Eqs. (1.78) and (1.80) to eliminate $f/2$, we obtain a similarity relation between heat transfer and mass transfer coefficients as follows in Eq. (1.82)

$$\begin{aligned} \frac{h}{h_M} &= \rho C_p \left(\frac{\text{Sc}}{\text{Pr}} \right)^{2/3} \\ &= \rho C_p \left(\frac{\alpha}{D} \right)^{2/3} \\ &= \rho C_p \text{Le}^{2/3} \end{aligned} \quad (1.82)$$

where $\text{Le} = \alpha/D$ is the *Lewis number*.

If $\text{Le} = 1$, the temperature and concentration profiles are the same and then $j_M = j_H$. Incidentally for air and water vapour mixtures at atmospheric pressure, the value of the Lewis number is approximately equal to unity. This simplifies the design of air-conditioning equipment. Writing h as f_g for heat transfer coefficient of air, and h_M as k_ω/ρ for diffusion of water vapour in air, we have for $\text{Le} = 1$

$$k_\omega = \frac{f_g}{C_p}$$

as used in Chapters 14 and 16.

Empirical relations for Sh and Nu for air over water droplets can be expressed as follows:

$$\text{Sh} = \frac{h_M L}{D} = \frac{k_\omega L}{\rho D} = 2 + 0.6 \text{Re}^{0.5} \text{Sc}^{0.33} \quad (1.83a)$$

$$\text{Nu} = \frac{hL}{k} = 2 + 0.6 \text{ Re}^{0.5} \text{ Pr}^{0.33} \quad (1.83\text{b})$$

The average value of diffusivity D of water vapour in air is $2.495 \times 10^{-5} \text{ m}^2 \text{ s}^{-1}$.

Example 1.17 Mass Transfer Coefficient of Water Vapour in Air.

Air at 35°C and 1 atm flows at a mean velocity of 30 m/s over a flat plate 0.5 m long. Calculate the mass transfer coefficient of water vapour from the plate into air. Assume the concentration of vapour in air as very small. The diffusion coefficient of water vapour into air is $0.256 \times 10^{-4} \text{ m}^2 \text{ s}^{-1}$. The Colburn j -factor for heat transfer coefficient is given by $j_H = 0.0296 \text{ Re}^{-0.2}$.

Solution Properties of air at 35°C , 1 atm

$$\rho = 1.146 \text{ kg. m}^{-3}$$

$$C_p = 1.006 \text{ kJ. kg}^{-1} \text{ K}^{-1}$$

$$\mu = 2 \times 10^{-5} \text{ kg. m}^{-1} \text{ s}^{-1}$$

$$\text{Pr} = 0.706$$

$$\text{Sc} = \frac{\mu}{\rho D} = \frac{2 \times 10^{-5}}{(1.146)(0.256 \times 10^{-4})} = 0.682$$

$$\text{Re} = \frac{Lu\rho}{\mu} = \frac{(0.5)(30)(1.146)}{2 \times 10^{-5}} = 859,600$$

$$j_M = j_H = 0.0296 \text{ Re}^{-0.2} \quad (1.84)$$

$$= 0.0296 (859,600)^{-0.2} = 1.925 \times 10^{-3}$$

$$h_M = j_M u / (\text{Sc})^{2/3}$$

$$= \frac{(0.001925)(30)}{(0.682)^{2/3}} = 0.075 \text{ m. s}^{-1}$$



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Revision Exercises

- 1.1** (a) A creamery must cool 11,350 kg of milk received each day from 30°C to 3°C in 3 hours. What must be the capacity of the refrigerating machine? Take the specific heat of milk as 3.92 kJ/kg.K.
 (b) If the compressor in the above case operates 8 hours per day with surplus refrigeration accumulated in brine storage tanks, what must the capacity be?
- 1.2** Cold salt brine at – 2°C is used in a packing plant to chill food slabs from 40°C to 3.4°C in 18 hours. Determine the weight of brine required to cool 100 food slabs of 225 kg each if the final temperature of brine is 1.7°C, Given:
 Specific gravity of brine = 1.05
 Specific heat of brine = 3.77 kJ/kg.K
 Specific heat of food slabs = 3.15 kJ/kg.K
- 1.3** A cold-storage room has inside dimensions of 12 m × 7.5 m × 3 m. The overall coefficient of heat transfer through the walls and ceiling is $0.55 \text{ W.m}^{-2}\text{K}^{-1}$ and for the floor is $2.2 \text{ W.m}^{-2}\text{K}^{-1}$. The inside temperature is to be maintained at –5°C. If the outside air is at 45°C and the air adjacent to the floor is at 30°C, determine the capacity of the refrigerating unit for 70 per cent running time.
- 1.4** 10 kg of air at 65°C and 3.5 bar expands reversibly and polytropically to 1.5 bar. The index of expansion is 1.25. Find the final temperature, work done, heat transferred and change of entropy. Assuming the process to be adiabatic, find the final temperature and work done.
- 1.5** Air at 3.5 bar and 30°C flows at the rate of 0.5 kg/s through an insulated turbine. If the air delivers 11.5 hp to the turbine blades and the change in kinetic energy is negligible, at what temperature does the air leave the turbine?

- 1.6** A fluid expands from 3 bar and 150°C to 1 bar in a nozzle. The initial velocity is 90 m/s. The isentropic efficiency is likely to be 0.95. Estimate the final velocity assuming the fluid to be steam and air.
- 1.7** A real gas expands isenthalpically from 135 bar and -75°C to 7 bar. If the Joule-Thomson coefficient in the range is 0.2°C/bar, what is the final temperature of the gas?
- 1.8** R 22 saturated liquid at 35°C is throttled to a pressure corresponding to a temperature of 5°C. Determine the extent of flashing of the liquid into vapour.
- 1.9** The exterior wall of a house is made of 10 cm of common brick ($k = 0.7 \text{ W.m}^{-1}\text{K}^{-1}$) followed by a 3.79 cm layer of gypsum plaster ($k = 0.48 \text{ W.m}^{-1}\text{K}^{-1}$). What thickness of thermocole insulation ($k = 0.037 \text{ W.m}^{-1}\text{K}^{-1}$) should be added to reduce the heat loss or gain through the wall to 25%?
- 1.10** Twelve thin brass fins, 0.75 mm thick, are placed axially on a 5 cm diameter \times 5 cm long cylinder which stands vertically and is surrounded by air at 38°C. The fins extend 2.5 cm from the cylinder surface which is at a temperature of 150°C. The thermal conductivity of brass is $78 \text{ W.m}^{-1}\text{K}^{-1}$ and the heat transfer coefficient for air is $23 \text{ W.m}^{-2}\text{K}^{-1}$. What is the rate of heat transfer to air from the cylinder?
- 1.11** Find the overall heat transfer coefficient between water and oil if the water flows through a copper pipe 1.8 cm ID and 2.1 cm OD while the oil flows through the annulus between the copper pipe and a steel pipe. The water and oil side film heat transfer coefficients are 4600 and $1250 \text{ W.m}^{-2}\text{K}^{-1}$ respectively. The fouling factors on the water and oil sides may be taken as 0.0004 and $0.001 \text{ m}^2\text{K.W}^{-1}$ respectively. The thermal conductivity of the tube wall is $330 \text{ W.m}^{-1}\text{K}^{-1}$.
- 1.12** In a R134a condenser of a domestic refrigerator, having wire-and-tube construction, $A_o/A_i = 12$. Heat transfer coefficient of refrigerant condensing inside tubes is $h_i = 1950 \text{ W.m}^{-2}\text{K}^{-1}$. Heat transfer coefficient by natural convection of air outside tubes is $h_o = 19.5 \text{ W.m}^{-2}\text{K}^{-1}$. Air enters at 35°C and leaves at 43°C. Heat to be rejected in the condenser is 180 W. Determine the outside tube surface area.
- 1.13** Plot $\ln p^{\text{sat}}$ versus $1/T^{\text{sat}}$ diagrams for CO₂, R 22 (CHClF₂), NH₃, C₃H₈, R 12 (CCl₂F₂), R 134a (C₂H₂F₄), R 152 a (C₂H₄F₂), Iso - C₄H₁₀ and R₁₁ (C Cl₂F). Find values of constants of Antonie equation.
- 1.14** Calculate the latent heat of vaporization of ammonia at 40°C using the following data from the saturation table of ammonia.

t °C	p bar	v_f L/kg	v_g m ³ /kg
30	11.67	1.68	0.11
35	13.5	1.7	0.095
40	15.54	1.73	0.083
45	17.82	1.75	0.073
50	20.33	1.78	0.063

1.15 Determine the specific volume of saturated vapour by Redlich-Kwong equation for the following, and compare results with those given in the tables of properties.

- (a) Propane at -25°C where $p^{\text{sat}} = 2.018$ bar.
- (b) Propane at 50°C where $p^{\text{sat}} = 17.119$ bar.
- (c) Isobutane at -25°C where $p^{\text{sat}} = 0.5865$ bar.
- (d) Isobutane at 50°C where $p^{\text{sat}} = 6.92$ bar.



Refrigerating Machine and Reversed Carnot Cycle

2.1 REFRIGERATING MACHINES

There are essentially two categories of thermal plants. These are:

- (i) Thermal power plants or *work producing plants*.
- (ii) Refrigeration/heat pump plants or *work consuming plants*.

The work producing plants or *heat engines* lead to the conversion of heat to work. The work consuming plants, viz., *refrigerators/heat pumps*, are not those which are in any way related to the conversion of work into heat. No ingenuity at all is required for the conversion of work into heat. In fact, all work (mechanical/electrical energy) that is consumed in machinery is ultimately dissipated as heat to the environment. The objective of work consuming plants, actually, is to lead to the flow of heat from a low temperature body to a high temperature body. The work is consumed to achieve this.

Examples of common work consuming plants, viz., refrigerators are the following:

Cold storages. Central air conditioning plants. Domestic refrigerators. Room air conditioners. Ice Plants. Food freezing and freeze-drying plants. Air liquefaction plants. etc.

Heat pumps are heating plants. But they operate in the same way as refrigerators.

Refrigeration equipment, in general, is relatively smaller in size as compared to work producing plants. The capacity of a power plant is in MW, whereas the capacity of a refrigeration system is in kW or even less. A very large super cold storage or a central air conditioning plant for a multistoreyed building may consume power in the range of 2000 to 5000 kW. A window-type room air conditioner may consume only 2.5 kW of power, and a domestic refrigerator just 100 to 250 W only.

2.2 A REFRIGERATING MACHINE—THE SECOND LAW INTERPRETATION

A *refrigerating machine* is a device which will either cool or maintain a body at a temperature below that of the surroundings. Hence, heat must be made to flow from

a body at low temperature to the surroundings at high temperature.

However, this is not possible on its own. We see in nature that heat *spontaneously* flows from a high temperature body to a low temperature body.

The reverse process to complete the thermodynamic cycle, in which heat Q will flow back from the low temperature body to the high temperature body, is not possible. Thus, we see that a thermodynamic cycle involving heat transfer alone as shown in Fig. 2.1 cannot be devised. The logical conclusion is that there must be a process in which some work is done.

The second law of thermodynamics, like the first law, is based on the observations of actually existing processes and devices in nature. Figure 2.2 shows the schematic diagram of an actual refrigeration system which works on the well-known *vapour compression cycle*. Most refrigeration devices/plants, including air conditioners and refrigerators such as the ones illustrated in Chap. 1, work on this cycle only.

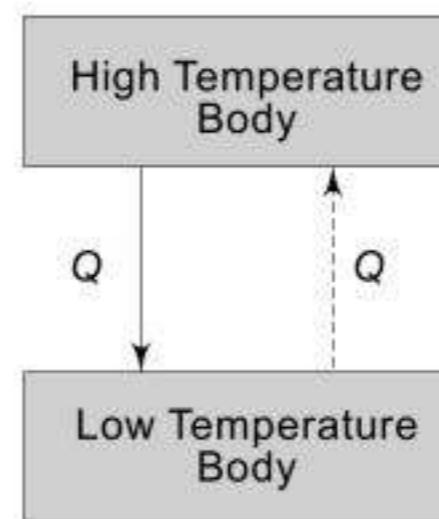


Fig. 2.1 A thermodynamic cycle involving heat transfer alone: Not possible

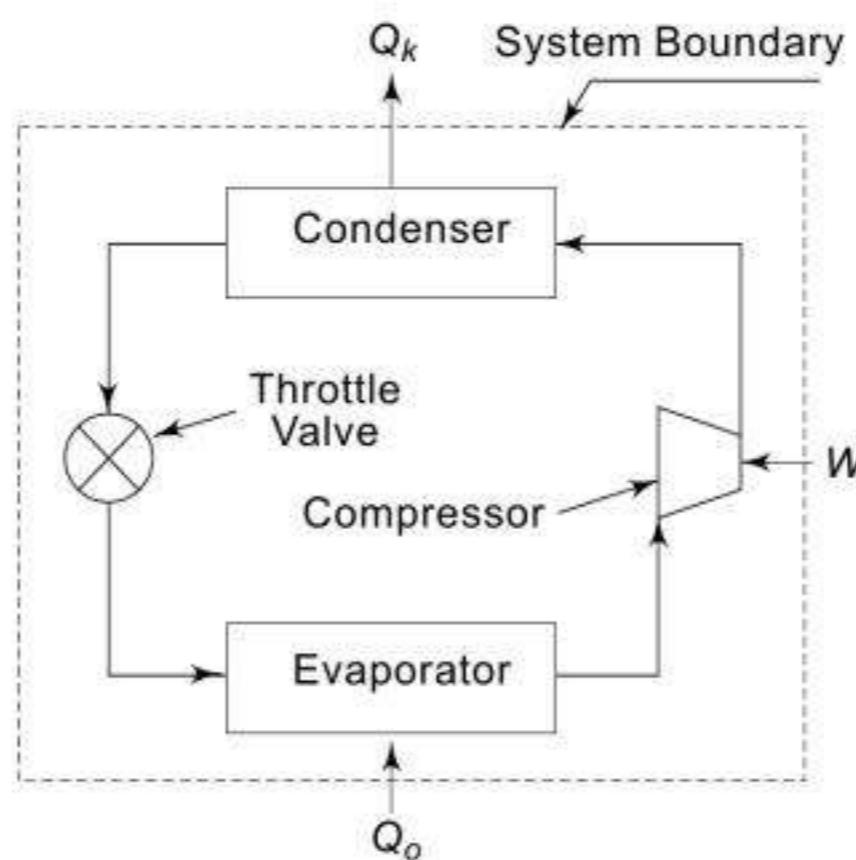


Fig. 2.2 Illustration of an actual refrigerator/heat pump: The simple vapour compression system

The heat and work interactions of the processes of the cycle are as follows:

- Heat Q_0 is absorbed in the evaporator by the evaporation of a liquid refrigerant at a low pressure p_0 , and corresponding low saturation temperature T_0 .
- The evaporated refrigerant vapour is compressed to a high pressure p_k in the compressor consuming work W . The pressure after compression is such that the corresponding saturation temperature T_k is higher than the temperature of the surroundings.
- Heat Q_k is then rejected in the condenser to the surroundings at high temperature T_k .

The application of the first law, $\oint \delta Q = \oint \delta W$, to the cycle gives:

$$-Q_k + Q_o = -W$$

$$Q_k - Q_o = W$$

This, also, represents an *energy balance* of the system in Fig. 2.2 obtained by drawing a boundary around it.

There are two statements of the second law of thermodynamics, the *Kelvin-Planck statement*, and the *Clausius statement*. The Kelvin-Planck statement pertains to heat engines such as E represented in Fig. 2.3.

The Clausius statement pertains to refrigerators/heat pumps R. The above observation from illustration of actually existing refrigerators/heat pumps leads to the Clausius statement which is as follows:

"It is impossible to construct a device which will operate in a cycle and produce no effect other than the transfer of heat from a low temperature body to a high temperature body".

The statement implies that a refrigerator R of the type shown in Fig. 2.4 which will absorb heat Q_o from a low temperature body and transfer it to a high temperature body is impossible.

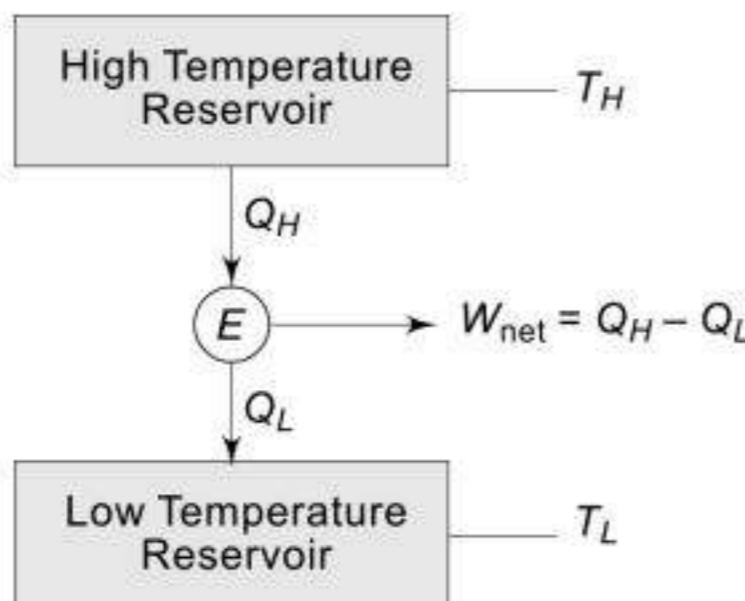


Fig. 2.3 Schematic representation of a heat engine

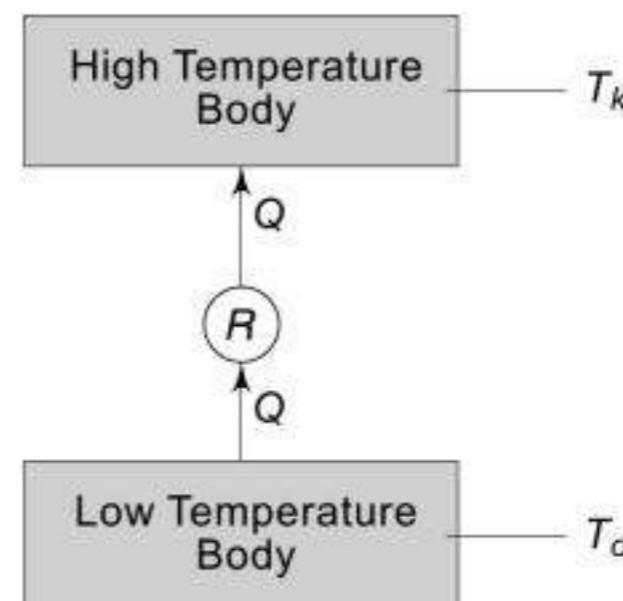


Fig. 2.4 Refrigerator without any Work Input: Impossible

The only alternative is that there must be some work input W_{in} . Accordingly, we obtain a schematic representation of a refrigerating machine/heat pump as shown in Fig. 2.5, and from the first law $W_{in} = Q_k - Q_o$. Accordingly, heat transferred Q_k is more than heat absorbed Q_o by the amount of work input W_{in} .

Now, the stress on the words 'operating in a cycle' is significant. For, in a single process, it may be possible to obtain removal of heat from a low temperature body without doing external work, e.g., by the evaporation of a refrigerant, for instance liquid Freon 22 after throttling expansion to, say, 1 bar correspond-

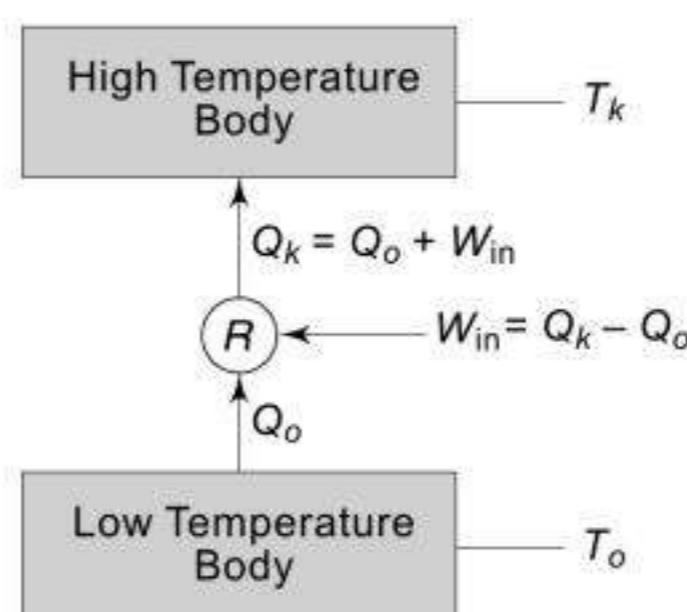


Fig. 2.5 Schematic representation of a refrigerator/heat pump

ing to a saturation temperature of -40°C as shown in Fig. 2.6. However, this could not happen continuously. This process could continue only if one had an infinite supply of high pressure Freon 22 liquid in the cylinder. But that is not possible in nature. To obtain refrigeration continuously, the refrigerant vapour after evaporation at low pressure will have to be brought back to the initial state of high pressure liquid again. That will mean forming a complete thermodynamic cycle.

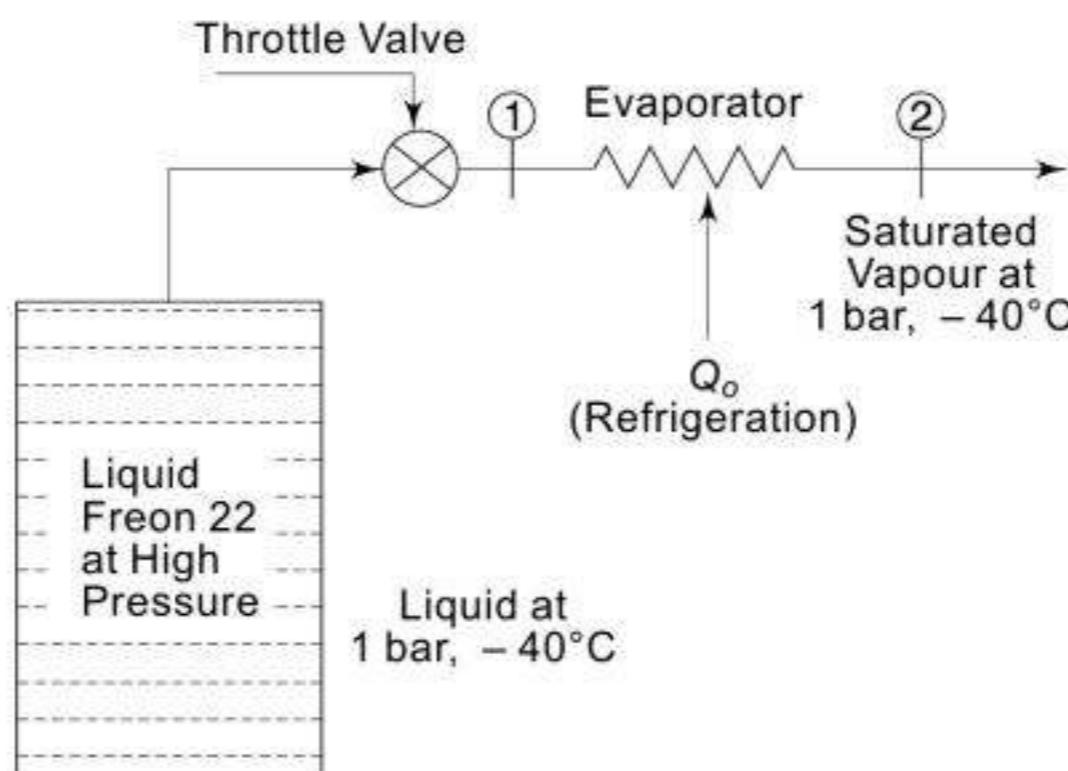


Fig. 2.6 A process: Producing refrigeration while $W = 0$

The Clausius statement eliminates the possibility of obtaining refrigeration without doing work. The statement necessitates a further clarification regarding heat-operated refrigerating machines such as the vapour absorption type or ejector type, using heat directly to produce refrigeration. Such systems may be considered as a combination of a heat engine and a refrigerating machine. The heat engine part of the system utilizes heat from a body at a higher temperature than the surroundings and delivers the required mechanical work, *within the system*, which is directly used by the refrigerating machine part. Thus the usual process of the conversion of thermal energy, first into work (or electrical energy) and then its utilization in a refrigerating machine, is replaced by a combined process.



2.3 HEAT ENGINE, HEAT PUMP AND REFRIGERATING MACHINE

It may be concluded from the preceding discussion that a reversible heat engine may be converted into a refrigerating machine by running it in the reversed direction. Schematically, therefore, a refrigerating machine is a reversed heat engine which can be seen by comparing Figs. 2.5 and 2.3.

As for the *heat pump*, there is no difference in the cycle of operation between a refrigerating machine and a heat pump. The same machine can be utilized either

- (i) to absorb heat from a cold body (a cooled space) at temperature T_o and reject it to the surroundings at temperature $T_k \geq T_a$, or
- (ii) to absorb heat from the surroundings at temperature $T_o \leq T_a$ and reject it to a hot body (a heated space) at temperature T_k ,

where T_a is the temperature of the surroundings.

Figure 2.7 illustrates the manner of application of heat engine E , heat pump H and refrigerating machine R . It implies that the same machine can be used either for cooling or for heating. When used for cooling, it is called a refrigerating machine and when used for heating it is called a heat pump.

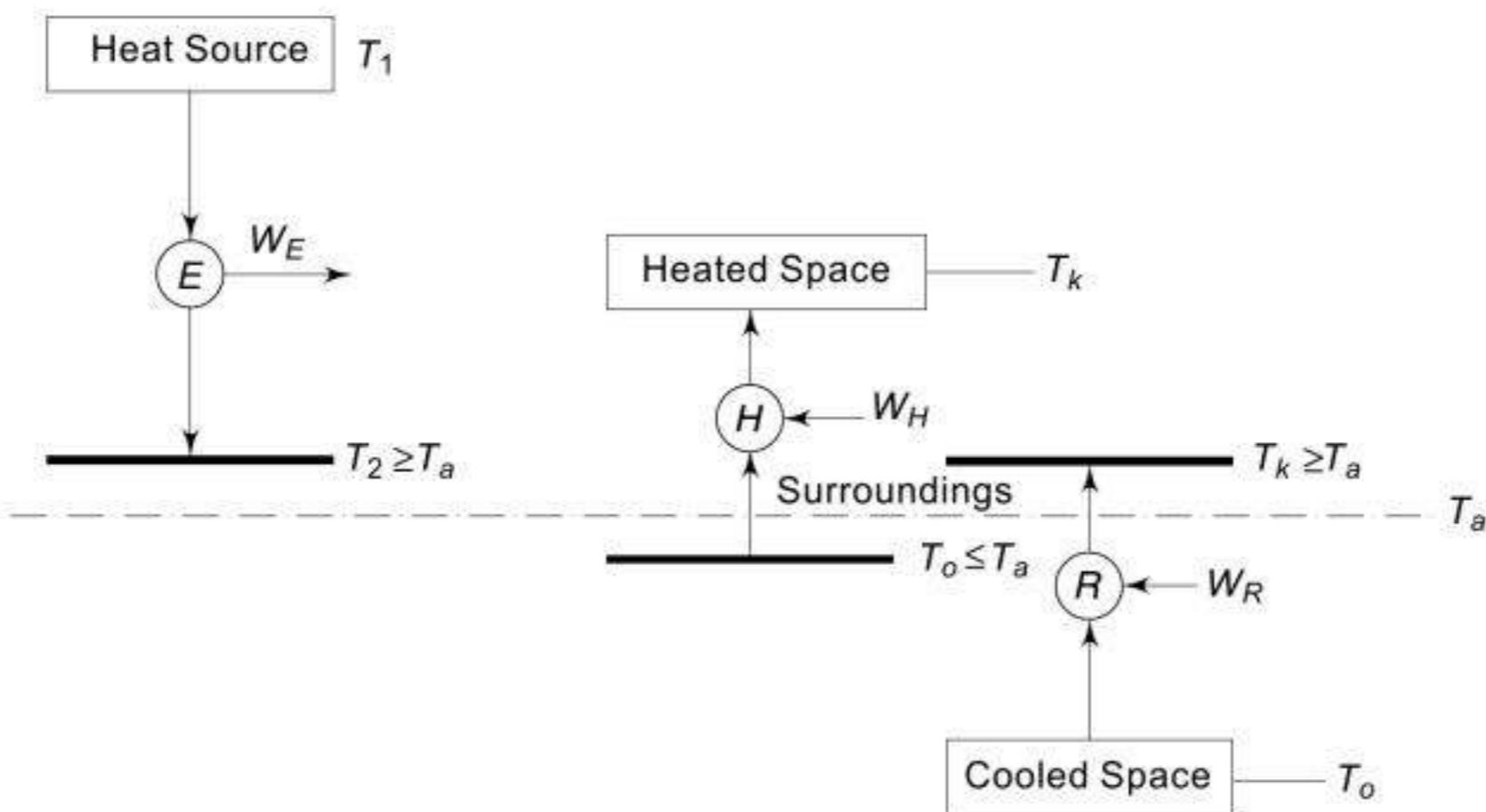


Fig. 2.7 Comparison of heat engine, heat pump and refrigerating machine

The main difference between the two is in their operating temperatures. A refrigerating machine operates between the ambient temperature $T_a \approx T_k$ and a low temperature T_o . A heat pump operates between the ambient temperature $T_a \approx T_o$ and a high temperature T_k . The heat engine operates between the heat source temperature T_1 and ambient temperature $T_a \approx T_2$.

Another essential difference is in their useful function. In a refrigerating machine, the heat exchanger that absorbs heat is connected to the conditioned space. In a heat pump, instead, the heat exchanger that rejects heat is connected to the conditioned space. The other heat exchanger in each case is connected to the surroundings. Thus if a refrigerating machine, that is used for cooling in summer, is to be used as a heat pump for heating in winter, it will be necessary, either

- to rotate the machine by 180° to interchange the positions of the two heat exchangers between the space and the surroundings, or
- to exchange the functions of the two heat exchangers by the operation of valves, e.g., a four-way valve in a window-type air conditioner.

Such an operation of a refrigerating machine is termed as *reversed cycle heating*.

2.3.1 Energy Ratios or Coefficients of Performance

The performance of a heat engine is described by its thermal efficiency. The performance of a refrigerating machine or a heat pump is expressed by the ratio of useful heat to work, called the *energy ratio* or *coefficient of performance (COP)*. Thus we have for a refrigerating machine:

Cooling energy ratio, or COP for cooling

$$\varepsilon_c = \frac{Q_o}{W} = \frac{Q_o}{Q_k - Q_o} = \frac{1}{\frac{Q_k}{Q_o} - 1} \quad (2.1)$$

And we have for a heat pump:

Heating energy ratio, or COP for heating

$$\varepsilon_h = \frac{Q_k}{W} = \frac{Q_k}{Q_k - Q_o} = \frac{1}{1 - \frac{Q_o}{Q_k}} \quad (2.2)$$

An idea about the approximate magnitude of the numerical values of these coefficients can be had from the following approximate calculations. The thermal efficiency of a heat engine is of the order of 30 per cent (say) so that

$$\eta_{th} = \frac{Q_k - Q_o}{Q_k} = 0.3$$

Then, if the engine is reversed in operation to work as a refrigerator or a heat pump with operating conditions unchanged (although in actual practice the operating temperatures will be different), we should have

$$\varepsilon_c = \frac{Q_o}{Q_k - Q_o} = \frac{1 - \eta_{th}}{\eta_{th}} = 2.33$$

and

$$\varepsilon_h = \frac{Q_k}{Q_k - Q_o} = \frac{1}{\eta_{th}} = 3.33$$

For vapour compression systems, ε_c is of the order of 3 for water-cooled and 2 for air-cooled air-conditioning applications and 1 for domestic refrigerators.

For air cycle refrigeration systems, $\varepsilon_c \approx 1$ and for vapour absorption systems, it is well below unity. Steam ejector machines have still lower values.

However, the latter two are heat-operated refrigerating machines, and the definition of their coefficients of performance is altogether different. Hence, no comparison need be made at this stage.

2.3.2 Power Consumption of a Refrigerating Machine

Power consumption \dot{W} of a refrigerating machine is determined in terms of kW. However, the power consumption of the motors is sometimes rated in horsepower (HP). We have

$$HP = \frac{\dot{W}}{0.746}$$

where \dot{W} is in kW.

A quantity which is frequently used for comparison is horsepower per ton refrigeration. It is determined as follows:

$$HP/kW \text{ refrigeration} = \frac{\dot{W}}{0.746 \dot{Q}_o}$$

where \dot{Q}_o is the refrigerating capacity in kW. Then, since 1 TR = 3.5167 kW,

$$\begin{aligned} \text{HP/TR} &= \frac{\dot{W}}{0.746 \dot{Q}_o} \quad (3.5167) \\ &= \frac{4.71 \dot{W}}{\dot{Q}_o} \\ &= \frac{4.71}{\mathcal{E}_c} \end{aligned} \quad (2.3)$$

Thus HP/TR is inversely proportional to COP for cooling. In the above derivation, imperial horsepower has been used. If the horsepower is metric, then

$$\begin{aligned} \text{HP} &= \frac{\dot{W}}{0.736} \\ \text{HP/TR} &= \frac{4.78}{\mathcal{E}_c} \end{aligned}$$

2.3.3 Heat Pump vs. Electric Resistance Heater

It may be seen from the simple transposition of Eqs. (2.1) and (2.2) that

$$\begin{aligned} \frac{Q_k}{W} &= 1 + \frac{Q_o}{W} \\ \mathcal{E}_h &= 1 + \mathcal{E}_c \end{aligned} \quad (2.4)$$

The above relationship expresses a very interesting feature of a heat pump. According to Eq. (2.4), COP for heating is always greater than unity. It is so since Q_k is always greater than Q_o by the amount W .

Thus for the purpose of heating, it is far more economical to use a heat pump rather than an electric-resistance heater. For example, if W is the energy consumption of an electric resistance heater, the heat released to the space will be W only. But if this electrical energy W is utilized in a heat pump, the heat pumped to the space will be

$$Q_k = \mathcal{E}_h W = (1 + \mathcal{E}_c) W$$

Therefore, whatever is the value of \mathcal{E}_c (even zero), Q_k will always be greater than or equal to W . The value of \mathcal{E}_h for air-conditioning applications is of the order of 3. Then the heat pumped will be 3 W in a heat pump unit while the power consumption is only W . The heat pump, therefore, is a definite advancement over the simple electric-resistance heater. Only the cost of the heat pump (which is a refrigerating machine also) is prohibitive. But when an air-conditioning plant is already installed for cooling in summer, it would always be prudent to use it for heating as well in winter, operating as a heat pump.

Suppose a room loses \dot{Q}_k kW of heat during winter. Then, it requires an equal amount of heat addition to maintain it at a desired temperature. Figure 2.8 illustrates the two methods for heating the room:

- (i) Using electric resistance heater. The power consumption is

$$\dot{W} = \dot{Q}_k$$

(ii) Using heat pump. The power consumption is

$$\dot{W} = \frac{\dot{Q}_k}{\mathcal{E}_h} \cong \frac{1}{3} \dot{Q}_k$$

if $\mathcal{E}_h = 3$. Thus, power consumption of the heat pump is very much lower.

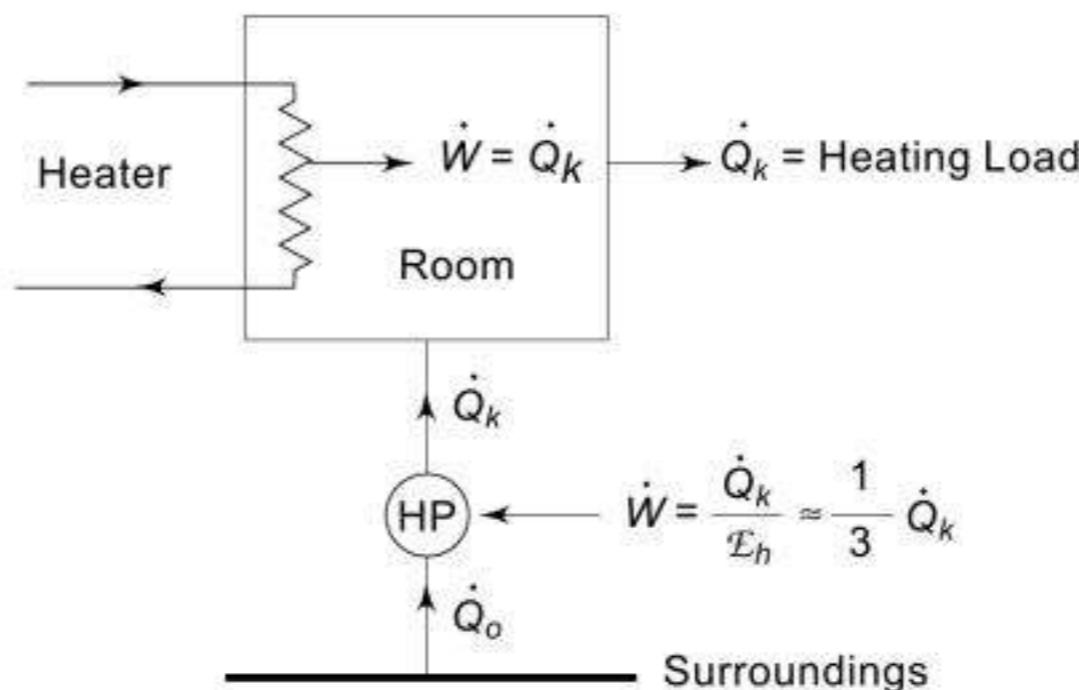


Fig. 2.8 Heating of a room by electric heater and heat pump



2.4 BEST REFRIGERATION CYCLE: THE CARNOT PRINCIPLE

It is possible to show that the cooling energy ratio of a refrigeration cycle working between two temperatures will be maximum when the cycle is a reversible one. For the purpose, consider a reversible refrigerating machine R and another irreversible refrigerating machine I , both working between two heat reservoirs at temperatures T_k and T_o , and absorbing the same quantity of heat Q_o from the cold reservoir at T_o as shown in Fig.

2.9 (a).

Now, to prove the contrary, let us assume that the COP of the irreversible machine is higher than that of the reversible machine, viz., $\mathcal{E}_I > \mathcal{E}_R$. Hence

$$\frac{\dot{Q}_o}{W_I} > \frac{\dot{Q}_o}{W_R}$$

$$W_R > W_I$$

And since

$$Q_{k,R} = Q_o + W_R$$

$$Q_{k,I} = Q_o + W_I$$

we have
and

$$Q_{k,R} > Q_{k,I}$$

$$Q_{k,R} - Q_{k,I} = W_R - W_I = W_{net}$$

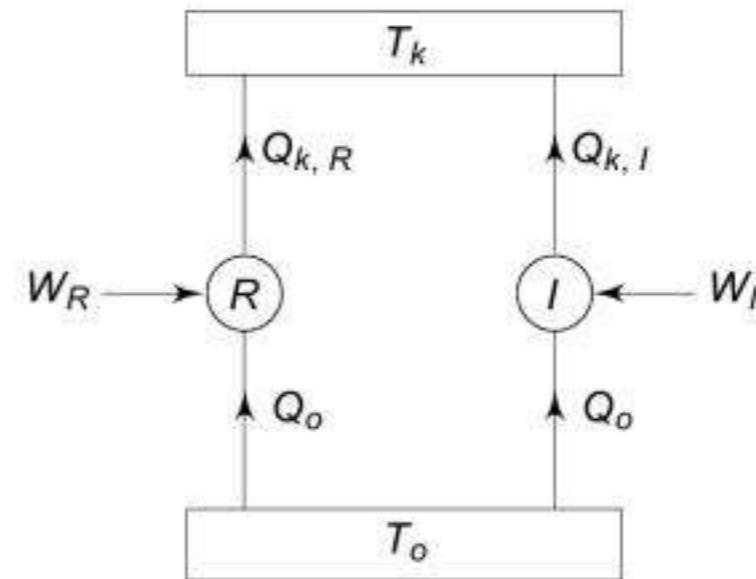


Fig. 2.9 (a) Reversible and irreversible refrigerating machines

If now, the reversible refrigerating machine is made to work as a heat engine and the irreversible refrigerating machine continues to work as a refrigerating machine, as shown in Fig. 2.9(b), the resultant combined system will work as a perpetual motion machine of the second kind taking heat equal to $Q_{k,R} - Q_{k,I}$ from the hot reservoir and converting it completely into work, thus violating the Kelvin-Planck statement of the Second Law applicable to heat engines as shown in Fig. 2.10.

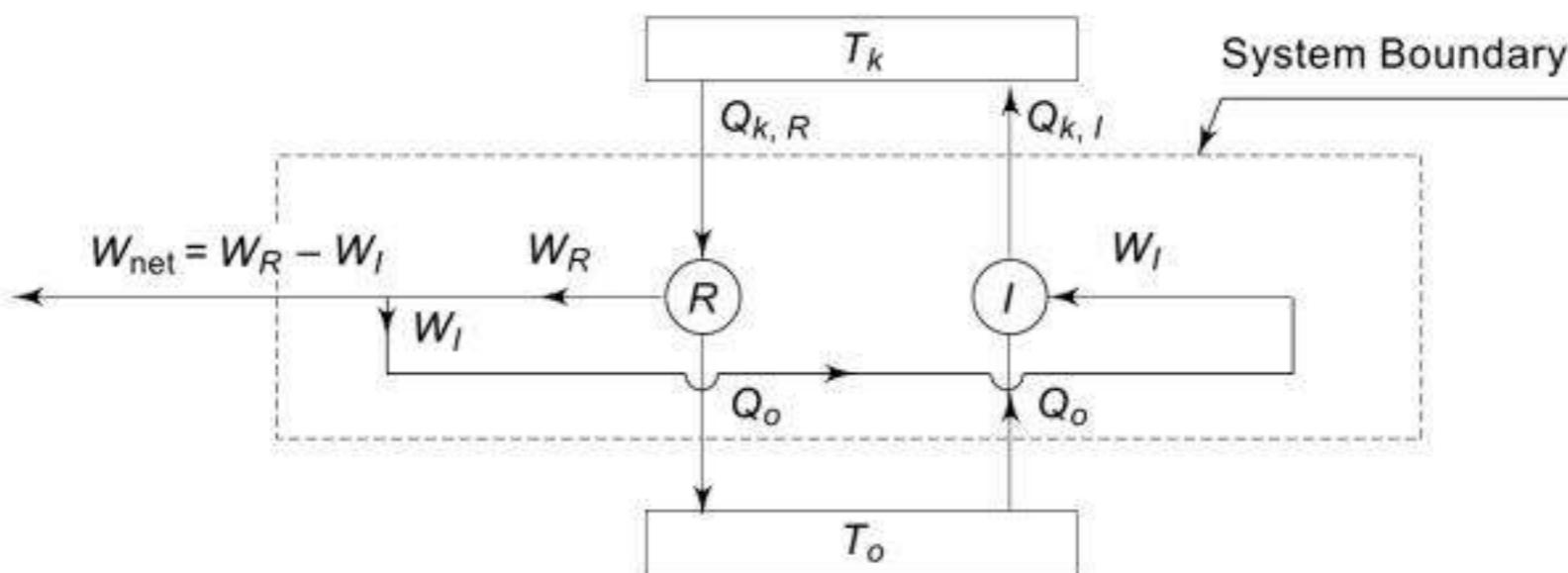


Fig. 2.9(b) Reversible refrigerating machine working as a heat engine in combination with an irreversible refrigerating machine

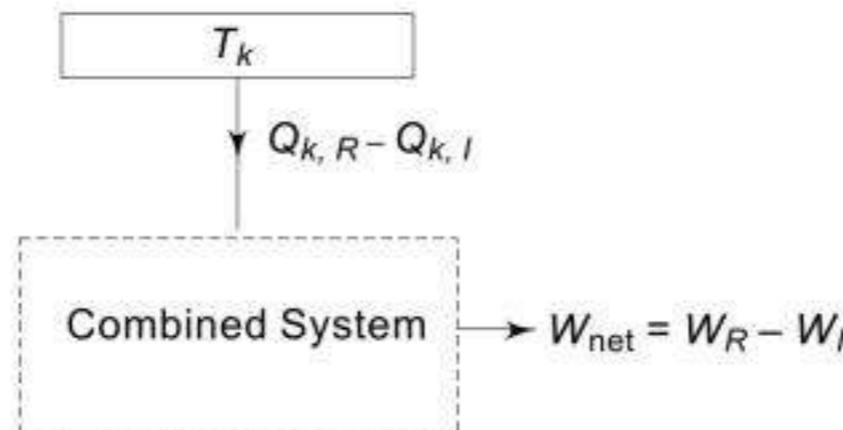


Fig. 2.10 Combined system resulting in a perpetual motion machine of the second kind thus violating the Second Law

It is, therefore, concluded that a refrigeration cycle operating reversibly between two heat reservoirs has the highest coefficient of performance. Likewise, it can also be shown that all reversible refrigeration cycles have the same COP. These are two corollaries of Second Law comprising the *Carnot Principle*.

2.4.1 Reversed Carnot Cycle

We now know that a reversible refrigeration cycle has the maximum COP. We know further that a reversible heat engine can be reversed in operation to work as a refrigerating machine.

Sadi Carnot, in 1824, proposed a reversible heat-engine cycle as a measure of maximum possible conversion of heat into work. A reversed Carnot cycle can, therefore, be employed as a reversible refrigeration cycle, which would be a measure of maximum possi-

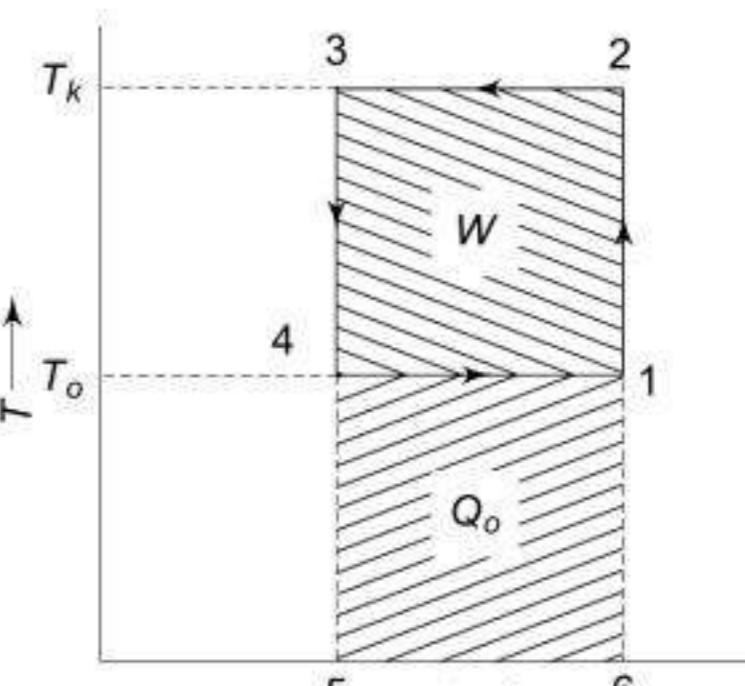


Fig. 2.11 Reversed Carnot Cycle

ble COP of a refrigerating machine operating between two temperatures T_k of heat rejection and T_o of refrigeration.

A reversed Carnot cycle, for a unit mass of the working substance, is shown in Fig. 2.11 on a $T-s$ diagram. The cycle consists of two isothermals and two isentropics as follows:

- Process 1—2 Isentropic compression, $s_1 = s_2$
- Process 2—3 Isothermal heat rejection to the hot reservoir at $T_k = \text{const.}$
- Process 3—4 Isentropic expansion, $s_3 = s_4$
- Process 4—1 Isothermal heat absorption from the cold reservoir at $T_o = \text{const.}$

The areas on the $T-s$ diagram, representing $\int T ds$ give the heat transfers, and work done in the cycle as follows:

$$\text{Heat absorbed from cold body, } Q_o = T_o \Delta s = \text{area 1-4-5-6}$$

$$\text{Heat rejected to hot body, } Q_k = T_k \Delta s = \text{area 2-3-5-6}$$

$$\text{Work done, } W = Q_k - Q_o = (T_k - T_o) \Delta s = \text{area 1-2-3-4}$$

Hence we obtain Carnot values of COP for cooling and heating as

$$\mathcal{E}_{c, \text{Carnot}} = \frac{\dot{Q}_o}{W} = \frac{T_o}{T_k - T_o} = \frac{I}{\frac{T_k}{T_o} - I}$$

$$\mathcal{E}_{h, \text{Carnot}} = \frac{\dot{Q}_k}{W} = \frac{T_k}{T_k - T_o} = \frac{I}{I - T_o/T_k}$$

Effect of Operating Temperatures We, thus, see that the Carnot COP depends on the operating temperatures T_k and T_o only. It does not depend on the working substance (refrigerant) used.

For cooling, T_o is the refrigeration temperature and T_k is the temperature of heat rejection to the surroundings. The lowest possible refrigeration temperature is $T_o = 0$ (absolute zero) at which $\mathcal{E}_c = 0$. The highest possible refrigeration temperature is $T_o = T_k$, i.e., when the refrigeration temperature is equal to the temperature of the surroundings (ambient) at which $\mathcal{E}_c = \infty$. Thus, Carnot COP for cooling varies between 0 and ∞ .

For heating, T_o is the temperature of heat absorption from the surroundings and T_k is the heating temperature. Theoretically, the COP for heating varies between 1 and ∞ .

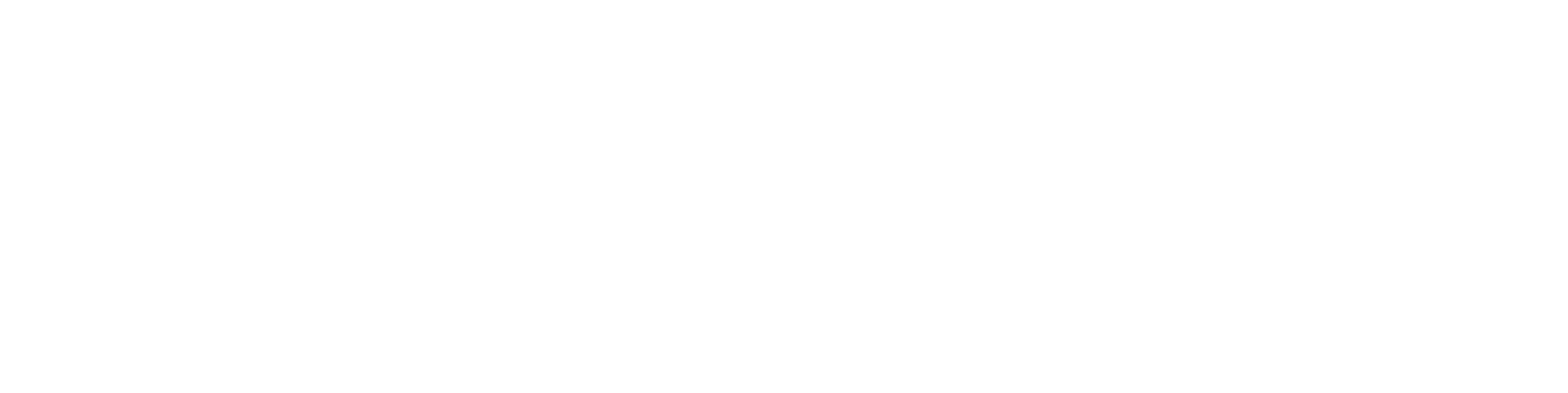
It may, therefore, be noted that to obtain maximum possible COP in any application,

- (i) the cold body temperature T_o should be as high as possible, and
- (ii) the hot body temperature T_k should be as low as possible.

The lower the refrigeration temperature required, and higher the temperature of heat rejection to the surroundings, the larger is the power consumption of the refrigerating machine. Also, the lower is the refrigeration temperature required, the lower is the refrigerating capacity obtained.

Consider now, for example, a domestic refrigerator which produces refrigeration at -25°C (248 K). Let heat be rejected to the ambient at 60°C (333 K).

The maximum possible COP of this refrigerator would be



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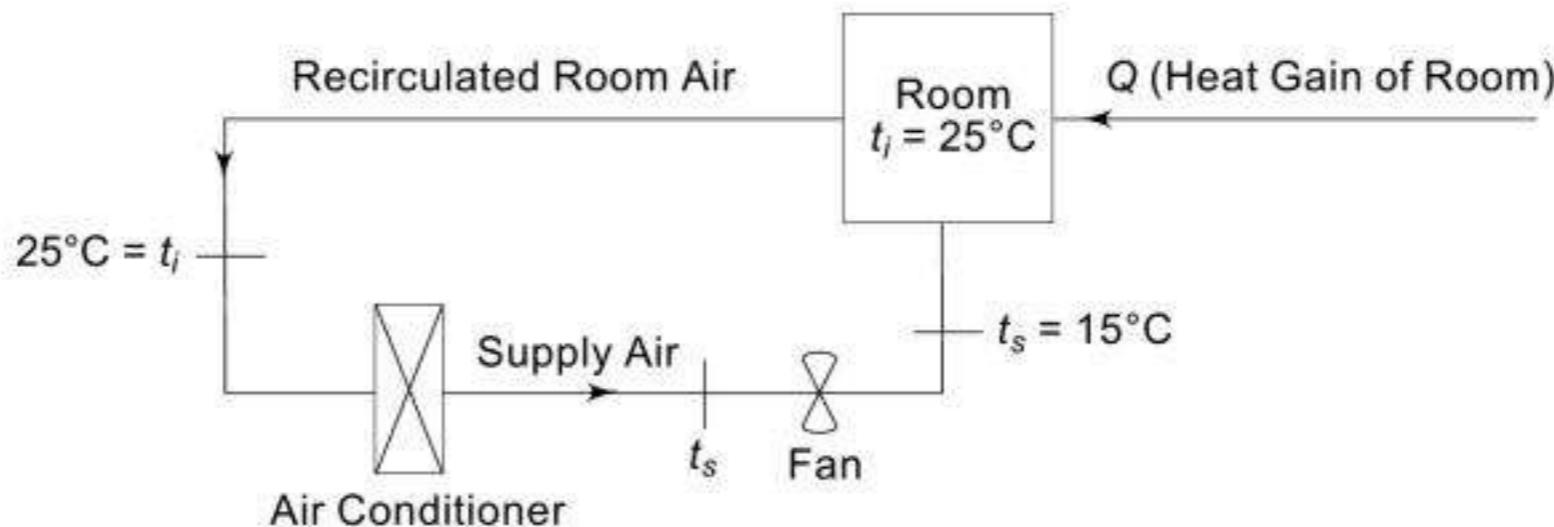


Fig. 2.12(a) Simple summer air-conditioning system

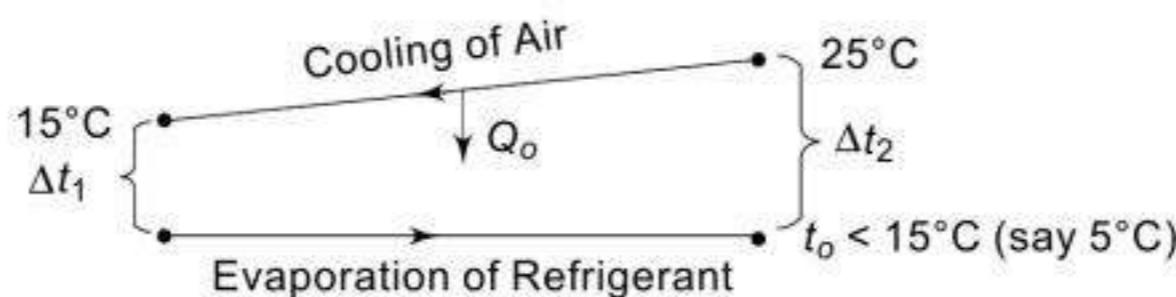


Fig. 2.12(b) Temperatures in a cooling coil

The selection of temperature T_k depends on the surrounding medium used for heat rejection. There are three possible media in the surroundings to which heat Q_k may be rejected, viz., air, water and ground. The units that use air as a cooling medium are called the *air-cooled units* and those using water are called the *water-cooled units*.

Consider an air-cooled unit. Let the surrounding air temperature (in summer) be 45°C . Also, let the rise in temperature of air, after absorbing heat Q_k , be 10°C , as shown in Fig. 2.13(a). Hence the temperature of heat rejection t_k has to be greater than 55°C , say 65°C , so that the temperature differentials Δt_1 and Δt_2 , across the heat exchanger (HE) are 20°C and 10°C respectively, and the arithmetic mean temperature difference is $(20 + 10)/2 = 15^\circ\text{C}$.

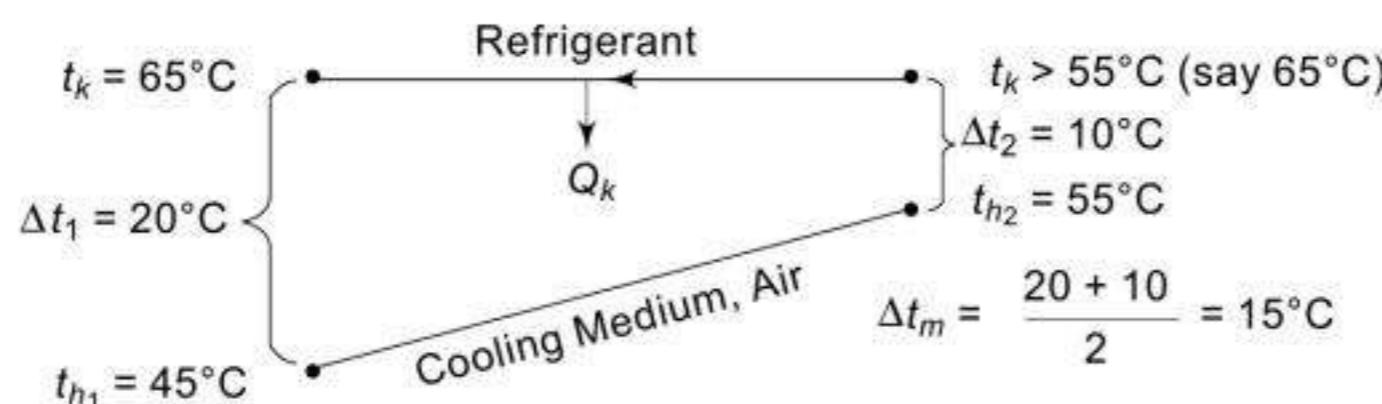


Fig. 2.13(a) Temperatures during heat rejection in an air-cooled refrigerating machine

Air is widely used as a cooling medium in small refrigerating machines such as refrigerators, water coolers, window-type air conditioners and small-package units.

Water as a cooling medium is preferable to air as it affords a lower value of T_k because of the following reasons:

- It is available at a temperature lower than that of air. Its temperature approaches the wet bulb temperature of the surrounding air. This is the limiting temperature to which heated water can be cooled in a cooling tower or a spray pond.



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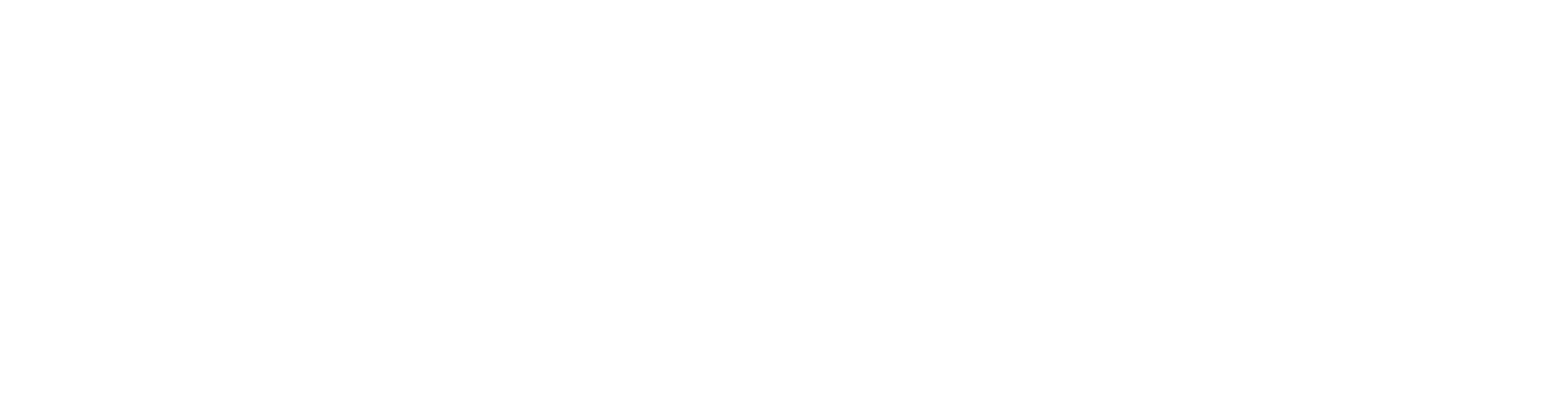
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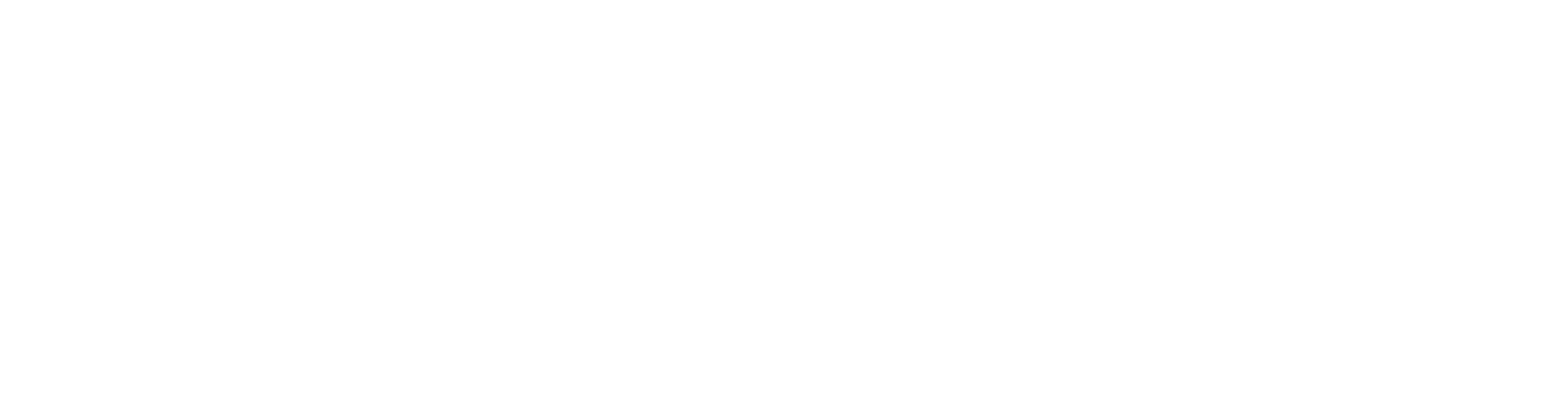
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The four processes of the cycle are analysed as non-flow processes as follows in Eqs. (2.6), (2.7), (2.8) and (2.9).

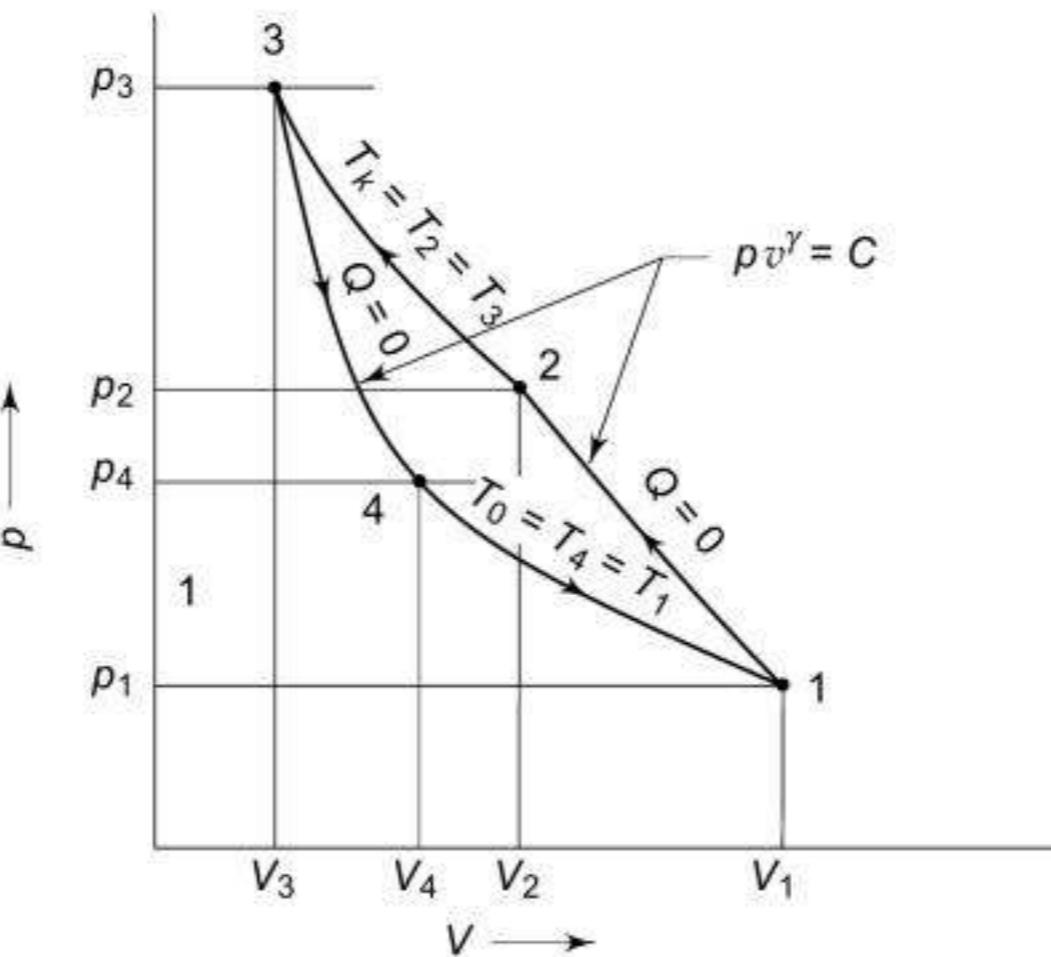


Fig. 2.15(b) Reversed Carnot cycle with gas as a refrigerant on p - v diagram

1–2 Isentropic compression: $Q = 0$

Pressure increases from p_1 to p_2

Specific volume reduces from v_1 to v_2

Temperature increases from $T_o = T_1$ to $T_k = T_2$

$$\text{Work done, } |w_{1-2}| = \frac{p_2 v_2 - p_1 v_1}{\gamma - 1} = \frac{R(T_k - T_o)}{\gamma - 1} \quad (2.6)$$

2–3 Isothermal compression and heat rejection: $T_2 = T_3 = T_k$

Pressure increases from p_2 to p_3

Specific volume reduces from v_2 to v_3

$$\text{Work done, } |w_{2-3}| = p_2 v_2 \ln \frac{v_2}{v_3} = RT_k \ln \frac{v_2}{v_3} \quad (2.7)$$

Heat rejected, $q_k = q_{2-3} = w_{2-3}$ (for a perfect gas in an isothermal process)

3–4 Isentropic expansion: $Q = 0$

Pressure falls from p_3 to p_4

Specific volume increases from v_3 to v_4

Temperature decreases from $T_k = T_3$ to $T_o = T_4$

$$\text{Work done, } w_{3-4} = \frac{p_3 v_3 - p_4 v_4}{\gamma - 1} = \frac{R(T_k - T_o)}{\gamma - 1} \quad (2.8)$$

4–1 Isothermal expansion and heat absorption: $T_4 = T_1 = T_o$

Pressure falls from p_4 to p_1

Specific volume increases from v_4 to v_1

$$\text{Work done, } w_{4-1} = p_4 v_4 \ln \frac{v_1}{v_4} = RT_o \ln \frac{v_1}{v_4} \quad (2.9)$$

Refrigerating effect, $q_o = q_{4-1} = w_{4-1}$ (for a perfect gas)

Net work of the cycle.

$$w = |w_{2-3}| - w_{4-1} = RT_k \ln \frac{v_2}{v_3} - RT_o \ln \frac{v_1}{v_4} \quad (2.10)$$

$$\text{Refrigerating effect, } q_o = q_{4-1} = RT_o \ln \frac{v_1}{v_4} \quad (2.11)$$

Now for the isentropic processes 1-2 and 3-4

$$\frac{T_k}{T_o} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_3}{p_4} \right)^{\frac{\gamma-1}{\gamma}}$$

and

$$\frac{T_k}{T_o} = \left(\frac{v_1}{v_2} \right)^{\gamma-1} = \left(\frac{v_4}{v_3} \right)^{\gamma-1} = r^{\gamma-1}$$

where r is the compression ratio for the isentropic processes.

$$\text{Hence } \frac{p_2}{p_1} = \frac{p_3}{p_4}, \text{ and } r = \frac{v_1}{v_2} = \frac{v_4}{v_3}$$

$$\text{or } \frac{v_2}{v_3} = \frac{v_1}{v_4} = \frac{p_3}{p_2} = \frac{p_4}{p_1}$$

$$\text{since } p_2 v_2 = p_3 v_3 \text{ and } p_4 v_4 = p_1 v_1$$

We have, for the COP for cooling

$$\begin{aligned} E_c &= \frac{q_o}{w} = \frac{RT_o \ln \frac{v_1}{v_4}}{RT_k \ln \frac{v_2}{v_3} - RT_o \ln \frac{v_1}{v_4}} \\ &= \frac{T_o}{T_k - T_o} \\ &= \frac{1}{\frac{T_k}{T_o} - 1} = \frac{1}{r^{\gamma-1} - 1} \end{aligned} \quad (2.12)$$

Thus COP is a function of compression ratio only.

2.7 LIMITATIONS OF REVERSED CARNOT CYCLE

It is found that serious practical difficulties are encountered in the application of Carnot cycle.

In the reversed Carnot cycle with vapour as refrigerant, the isothermal processes of condensation and evaporation are internally reversible processes, and they are easily achievable in practice although there may be some problem in having only partial evaporation. However, isentropic compression and expansion processes have some limitations which are discussed in Chap. 3. In brief, it is difficult to design an expander to handle a mixture of largely liquid and partly vapour for the process 3-4. Also, because of the internal irreversibilities in the compressor and the expander, the actual COP of the Carnot cycle is very low, though the ideal cycle COP is the maximum. A cycle which is closest to the reversed Carnot vapour cycle is the vapour compression cycle described in Chap. 3.

There are two drawbacks of reversed Carnot cycle with gas as a refrigerant:

- (i) Firstly, it is not possible to devise, in practice, isothermal processes of heat absorption and rejection, 4-1 and 2-3 in Fig. 2.15 with gas as the working substance. These are impractical as these will be infinitely slow.
- (ii) Secondly, the cycle on $p-v$ diagram is very narrow since the volume is changing both during the reversible isothermal and reversible adiabatic processes. Drawn correctly to scale, the Carnot $p-v$ diagram is much thinner than the diagram illustrated in Fig. 2.15. As a result, the stroke volume of the cylinder is very large. The cycle, therefore, suffers from poor actual COP as a result of irreversibilities of the compressor and expander.

A gas refrigeration cycle, which is closest to reversed Carnot cycle with gas as a refrigerant, is described in Chap. 11.

2.8 ACTUAL REFRIGERATION SYSTEMS

Although the Carnot cycle is theoretically the most efficient cycle between given temperatures T_k and T_o , it has limitations for practical use. It is, therefore, found useful only as a criterion of perfection of cycle. In an actual cycle, the COPs, \mathcal{E}_c and \mathcal{E}_h , will be less than their Carnot values. For the purpose of comparison between the actual and Carnot values, we define the *second law efficiency* or *exergetic efficiency* for cooling and heating, $(\eta_{II})_c$ and $(\eta_{II})_h$ as below:

$$(\eta_{II})_c = \frac{\mathcal{E}_c}{\mathcal{E}_{c, \text{Carnot}}}$$

$$(\eta_{II})_h = \frac{\mathcal{E}_h}{\mathcal{E}_{h, \text{Carnot}}}$$

Note that \mathcal{E}_c and \mathcal{E}_h are the first law COPs.

The conventional refrigeration systems work on the *vapour compression cycle* which is closest to the Carnot vapour cycle and has a high COP. *Gas cycle refrigeration* is used in aircraft refrigeration. Among the less conventional ones are the heat-operated refrigerating machines working on the *vapour absorption cycle* and *steam ejector cycle*.

There are also the low temperature refrigeration or cryogenic cycles, e.g., Linde cycle, Claude cycle, etc., used for the liquefaction of gases. Also, we have Philips liquefier which employs a cycle approaching the reversible Stirling cycle.



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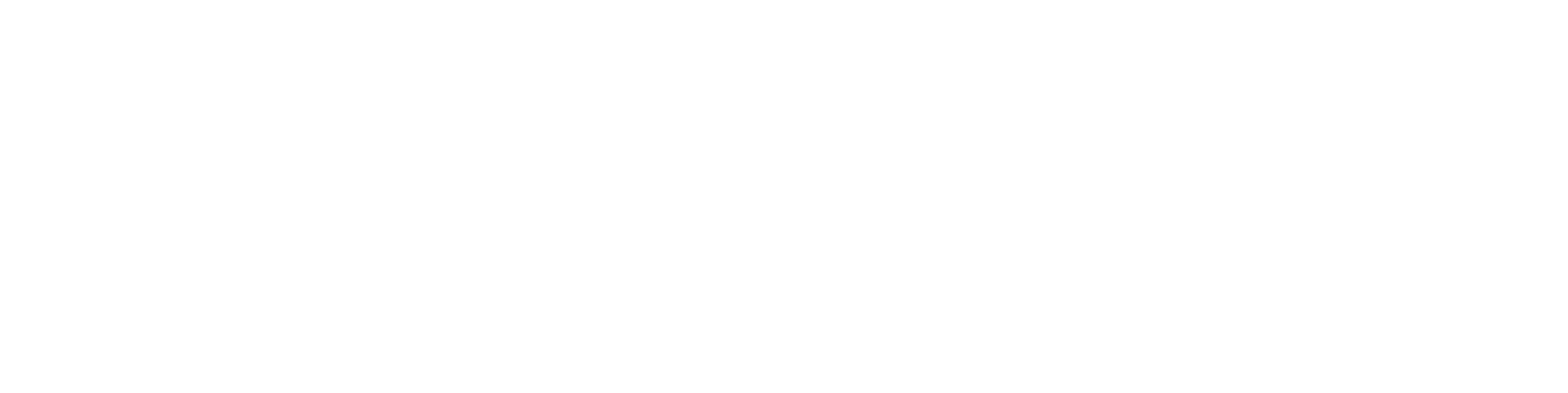
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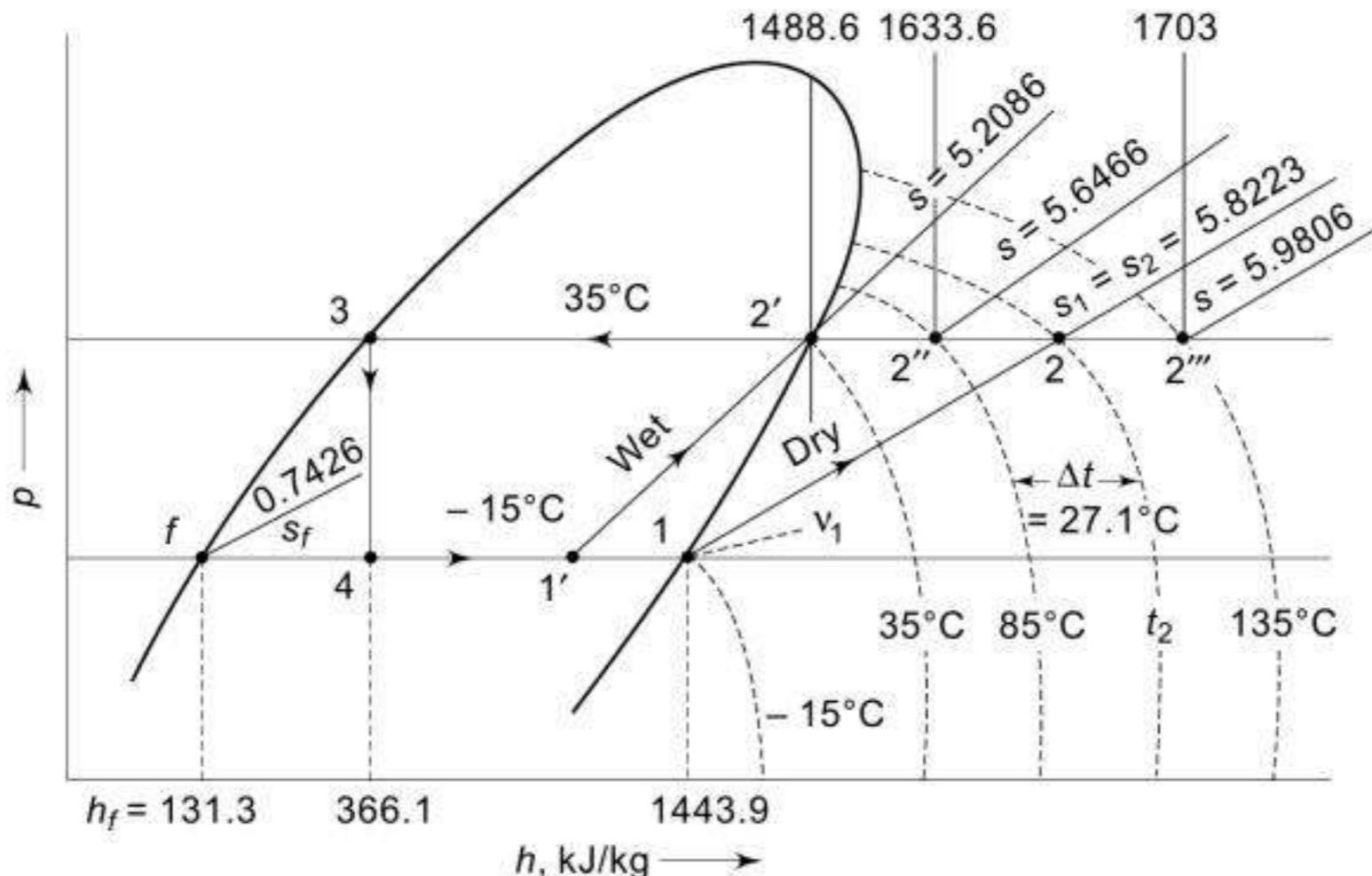


Fig. 3.8 Figure for Examples 3.3 and 3.4

Refrigerant circulation rate per ton refrigeration

$$m^* = \frac{211}{1077.8} = 0.1956 \text{ kg/(min)} (\text{TR})$$

(i) Theoretical piston displacement per ton

$$\dot{V}^* = m^* v_1 \times 60 = 0.1956 \times 0.509 \times 60 = 5.975 \text{ m}^3/\text{h}$$

(ii) Theoretical horsepower per ton

$$\text{HP}^* = \frac{m^* w}{90 \times 0.736} = \frac{0.1956 \times 259.4}{60 \times 0.736} = 1.15 \text{ hp/TR}$$

$$(iii) \text{ COP} = \frac{q_0}{w} = \frac{1078.5}{259.1} = 4.16$$

(b) *Wet compression* The cycle is shown as 1'-2'-3-4 in Fig. 3.8. The dryness at suction at 1' can be calculated by interpolation for entropy.

$$x'_1 = \frac{5.2086 - 0.7426}{5.8223 - 0.7426} = 0.88$$

$$h'_1 = 131.3 + 0.88 (1443.9 - 131.3) = 1286.1 \text{ kJ/kg}$$

$$v'_1 = v_f + x (v_g - v_f) = 0.00152 + 0.88 (0.509 - 0.00152) \\ = 0.448 \text{ m}^3/\text{kg}$$

$$q_0 = h'_1 - h_4 = 1286.1 - 366.1 = 920 \text{ kJ/kg}$$

$$w = h'_2 - h'_1 = 1488.6 - 1286.1 = 202.5 \text{ kJ/kg}$$

Refrigerant circulation rate per ton

$$m^* = \frac{211}{920} = 0.229 \text{ kg/(min)} (\text{TR})$$

(i) Theoretical piston displacement per ton

$$V^* = 0.229 \times 0.448 \times 60 = 6.159 \text{ m}^3/\text{h}$$

(ii) Theoretical horsepower per ton

$$\text{HP}^* = \frac{0.229 \times 202.5}{60 \times 0.736} = 1.05 \text{ hp/TR}$$

$$\text{(iii) COP} = \frac{920}{202.5} = 4.55$$

Note It is seen that higher COP and lower power consumption is obtained with wet compression in the case of ammonia. However, compressor piston displacement is increased by $(6.159 - 5.975)/5.975$, viz., 3% though.

3.4 EWING'S CONSTRUCTION

The theoretical COP of the vapour compression cycle with any *refrigerant* working between given evaporation and condenser temperatures is found to depend on the state of the suction vapour which may be wet, dry saturated or superheated. For example, in the limiting case, when the vapour is compressed immediately after throttling from 4 itself to 5 as shown in Fig. 3.9, the refrigerating effect as well as the COP are both zero. As the suction state is moved to the right, the refrigerating effect and COP go on increasing until point 1 is reached. After this point, the compression partly takes place in the superheat region and the heat is not rejected at constant temperature in the condenser. A trend of increasing COP *may continue* until the suction state 1 m for maximum COP is reached.

For some refrigerants and certain operating conditions, the maximum COP occurs with the suction state 1 m in the two phase region and for some others in the superheat region.

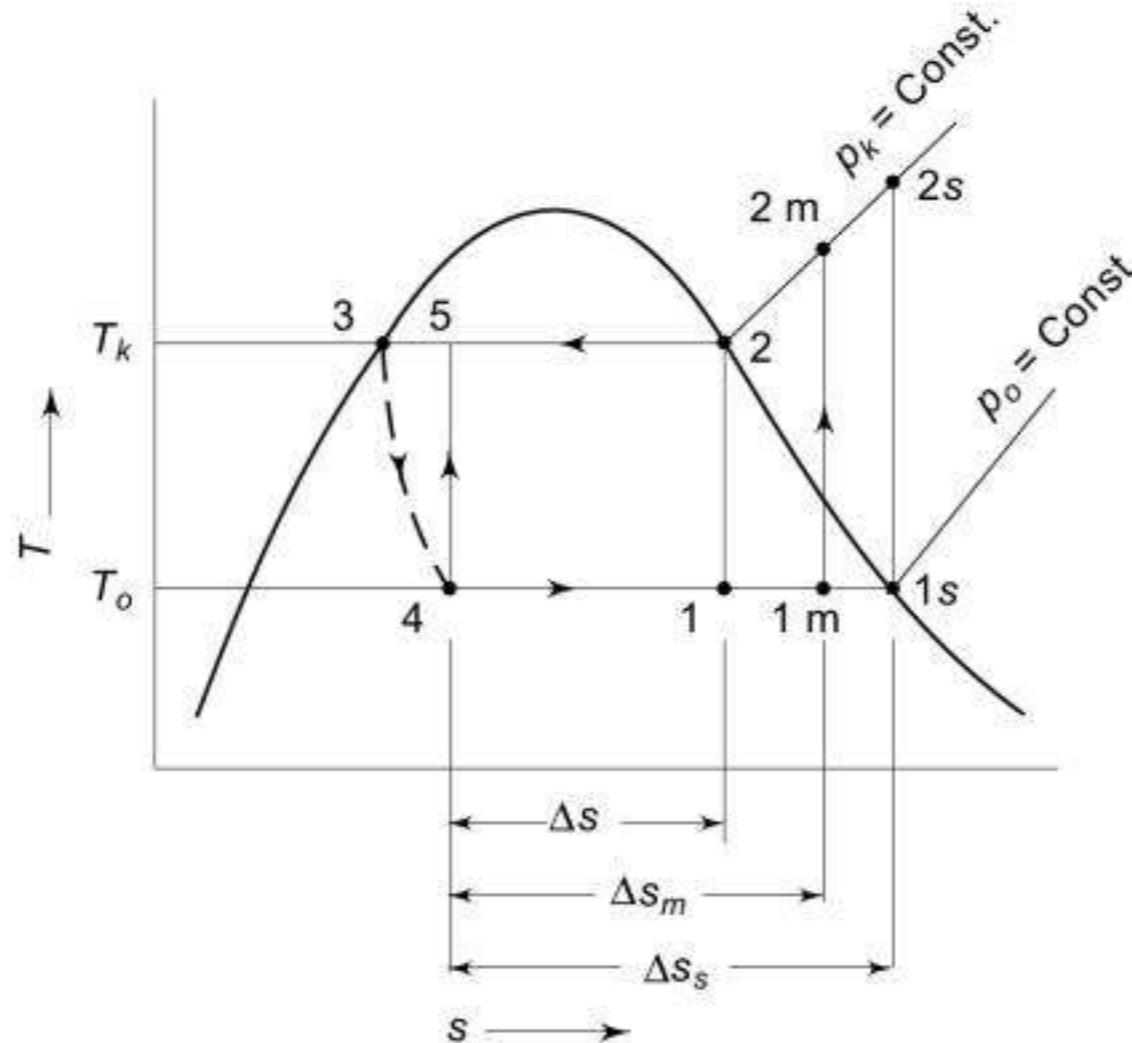


Fig. 3.9 Effect of variable suction state

Ewing¹ has shown that when this maximum occurs with state 1 m in the two-phase region, the COP is equal to that of a reversed Carnot cycle operating between the

evaporator temperature T_0 and the temperature of superheated discharge vapour T_{2m} . To show this, consider the expression for COP

$$\varepsilon = \frac{h_1 - h_4}{h_2 - h_1} = \frac{(h_1 - h_4)/\Delta s}{(h_2 - h_4)/\Delta s - (h_1 - h_4)/\Delta s} \quad (3.15)$$

Now, an enthalpy-entropy diagram may be employed for Ewing's construction as shown in Fig. 3.10. It may be seen that the term $(h_1 - h_4)/\Delta s$ is the gradient of the evaporator pressure $p_0 = \text{const.}$ line and is equal to T_0 , as long as point 1 is in the two-phase region. The expression for COP, with the suction state in the two-phase region, is then as given in Eq. (3.16)

$$\varepsilon = \frac{T_0}{(h_2 - h_4)/\Delta s - T_0} \quad (3.16)$$

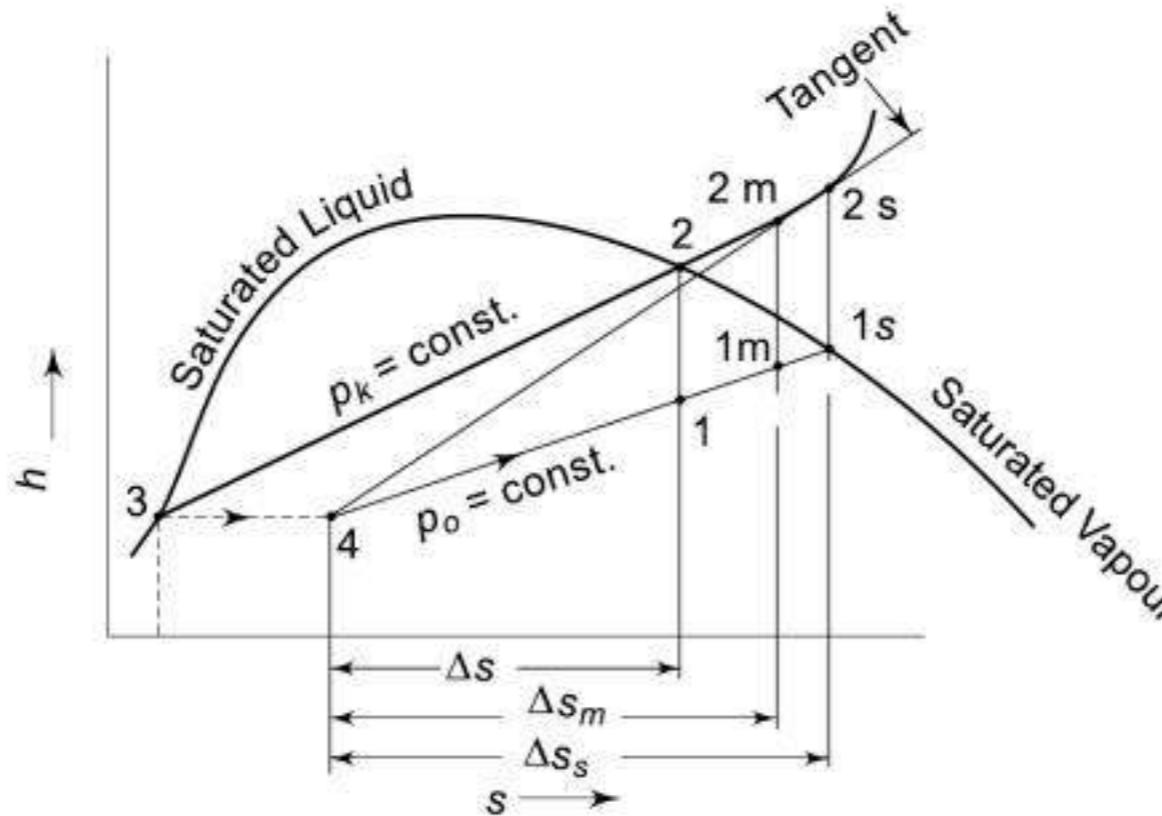


Fig. 3.10 Ewing's construction

The numerical value of the term $(h_2 - h_4)/\Delta s$, however, changes as point 2 shifts along with point 1. It can be seen that the COP is maximum when this gradient is minimum, i.e., when a line drawn from 4 to $p_k = \text{const.}$ line makes a tangent at 2m. The corresponding suction state for maximum COP is obtained on the isentropic line at 1m. Using the thermodynamic relation

$$\left(\frac{\Delta h}{\Delta s}\right)_p = T$$

we find that

$$\left[\frac{h_{2m} - h_4}{\Delta s_m} \right]_{p_k = \text{const.}} = T_{2m} \quad (3.17)$$

so that the slope of the tangent $\Delta h/\Delta s$ from point 4 to the $p_k = \text{const.}$ line is equal to the discharge temperature T_{2m} . The expression for maximum COP is then as follows in Eq. (3.18)

$$\varepsilon_{\max} = \frac{T_0}{\left[\frac{h_2 - h_4}{\Delta s} \right]_{\min} - T_0} = \frac{T_0}{T_{2m} - T_0} \quad (3.18)$$



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Hence, we have

$$\begin{aligned}
 s_1 &= 2.3073 = 1.2458 + x_2 (1.074) = s_2 \\
 \Rightarrow x_2 &= 0.988 \text{ kJ/kg} \\
 h_2 &= h_3 + x_2 h_{fg} = 272.37 + (0.988) (325.59) \\
 &= 598.12 \text{ kJ/kg} \\
 E_s &= \frac{h_1 - h_4}{h_2 - h_1} = \frac{536.19 - 272.37}{598.12 - 536.19} = \frac{263.8}{61.9} \\
 &= 4.26 \\
 \frac{T_0}{T_{2s} - T_0} &= \frac{258.15}{30 - (-15)} = \frac{258.15}{45} \\
 &= 5.74 < E_s
 \end{aligned}$$

Isobutane, therefore, belongs to Case II like Freon 12 and R 134a. The suction state for maximum COP lies in the superheat region.



3.5 STANDARD RATING CYCLE AND EFFECT OF OPERATING CONDITIONS

For the purpose of comparison, a case is made for the use of standard operating conditions. According to the current practice, the testing of single-stage compressors for air conditioning applications is carried out at $t_0 = 5^\circ\text{C}$, $t_k = 40^\circ\text{C}$ and a suction temperature of 20°C .

The following analysis shows the effect of change in operating conditions on the performance of the vapour compression cycle.

3.5.1 Effect of Evaporator Pressure

Consider a simple saturation cycle 1-2-3-4 with R 134a as the refrigerant as shown in Fig. 3.12 for operating conditions of $t_k = 40^\circ\text{C}$ and $t_0 = 0^\circ\text{C}$. It has been seen in Example 3.2 that for this cycle:

$$\text{Volume of suction vapour, } \dot{V}^* = 0.103 \text{ m}^3/(\text{min}) (\text{TR})$$

$$\text{Unit power consumption, } \dot{W}^* = \frac{9.61}{15} = 0.64 \text{ kW/TR}$$

Now consider a change in the evaporator pressure corresponding to a decrease in the evaporator temperature to -10°C . The changed cycle is shown as 1'-2'-3-4' in Fig. 3.12. Similar calculations for the changed conditions give:

$$\text{Volume of suction vapour, } \dot{V}^* = 0.154 \text{ m}^3/(\text{min}) (\text{TR})$$

$$\text{Unit power consumption, } \dot{W}^* = 0.87 \text{ kW/TR}$$

It is, therefore, seen that a drop in evaporator pressure corresponding to a drop of 10°C in saturated suction temperature increases the volume of suction vapour and hence decreases the capacity of a reciprocating compressor by

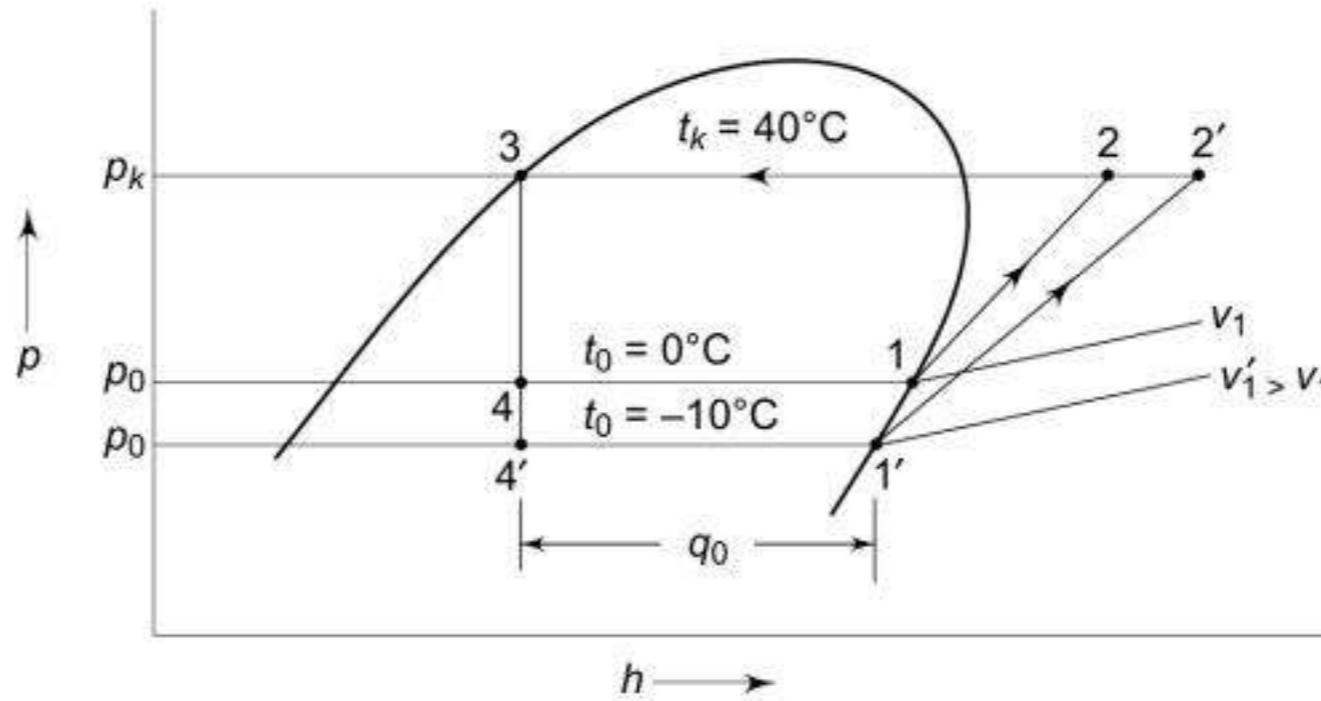


Fig. 3.12 Effect of evaporator pressure

$$\left(1 - \frac{V^*}{V^{*'}}\right) \times 100 = 33.3\%$$

and increases the power consumption per unit refrigeration by

$$\left(\frac{W^{*'}}{W^*} - 1\right) \times 100 = 36.4\%$$

It is observed that a decrease in evaporator temperature results in:

- (i) Decrease in refrigerating effect from $(h_1 - h_4)$ to $(h'_1 - h'_4)$
- (ii) Increase in the specific volume of suction vapour from v_1 to v'_1
- (iii) Decrease in volumetric efficiency, due to increase in the pressure ratio, from η_v to η'_v .
- (iv) Increase in compressor work from $(h_2 - h_1)$ to $(h'_2 - h'_1)$ due to increase in the pressure ratio as well as change from *steeper* isentropic 1-2 to *flatter* isentropic 1' - 2' as discussed in Sec. 3.5.3.

Since

$$\dot{Q}_0 = \dot{m}q_0 = \frac{\eta_v \dot{V}_p}{v_1} q_0$$

and

$$\dot{W}^* = m^* w \quad (3.23)$$

expressions for the dependence of capacity and unit power consumption may now be written as follows:

$$\dot{Q}_0 \propto q_0 = (h_1 - h_4)$$

$$\begin{aligned} &\propto \frac{1}{v_1} \\ &\propto \eta_v \end{aligned}$$

and

$$W^* \propto m^* \propto \frac{1}{q_0} = \frac{1}{h_1 - h_4}$$

$$\propto w = h_2 - h_1$$

Note The term *volumetric efficiency* η_v is only relevant in the case of positive displacement compressors.

Hence, denoting the capacity and unit power consumption of the changed cycle ($t_0 = -10^\circ\text{C}$) by \dot{Q}'_0 and $\dot{W}^{*\prime}$, we have expression as follows in Eqs. (3.24) and (3.25)

$$\frac{\dot{Q}'_0}{\dot{Q}_0} = \frac{h'_1 - h'_4}{h_1 - h_4} \cdot \frac{v_1}{v'_1} \cdot \frac{\eta'_v}{\eta_v} \quad (3.24)$$

$$\frac{\dot{W}^{*\prime}}{\dot{W}^*} = \frac{\mathcal{E}_c}{\mathcal{E}'_c} = \frac{h_1 - h_4}{h'_1 - h'_4} \cdot \frac{h'_2 - h'_1}{h'_2 - h_1} \quad (3.25)$$

Example 3.6 Variation in Capacity of Condensing Unit with Refrigeration Temperature

A Freon 22 condensing unit is specified to give 40 TR capacity for **air conditioning** under standard operating conditions of 40°C condensing and 5°C evaporating temperatures. What would be its capacity in TR for **food freezing** for which the evaporator temperature is -35°C ?

Solution Refer to Fig. 3.13.

(i) At $t_0 = 5^\circ\text{C}$

$$\dot{Q}_0 = 40 \text{ TR} = 40 (3.5167) = 140.67 \text{ kW}$$

$$q_0 = h_1 - h_4 = 407.15 - 249.7 = 157.45 \text{ kJ/kg}$$

$$\dot{m} = \frac{\dot{Q}_0}{q_0} = \frac{140.67}{157.45} = 0.8934 \text{ kg/s}$$

$$\dot{V}_p = \dot{m} v_1 = 0.8934 (0.04035) = 0.036 \text{ m}^3/\text{s}$$

$$w = h_2 - h_1 = 26.85 \text{ kJ/kg}$$

$$\dot{W} = \dot{m} w = 24 \text{ kW}$$

$$\dot{q}_k = h_2 - h_3 = 434 - 249.7 = 184.3 \text{ kJ/kg}$$

$$\dot{Q}_k = \dot{m} q_k = 0.8934 (184.3) = 164.7 \text{ kW}$$

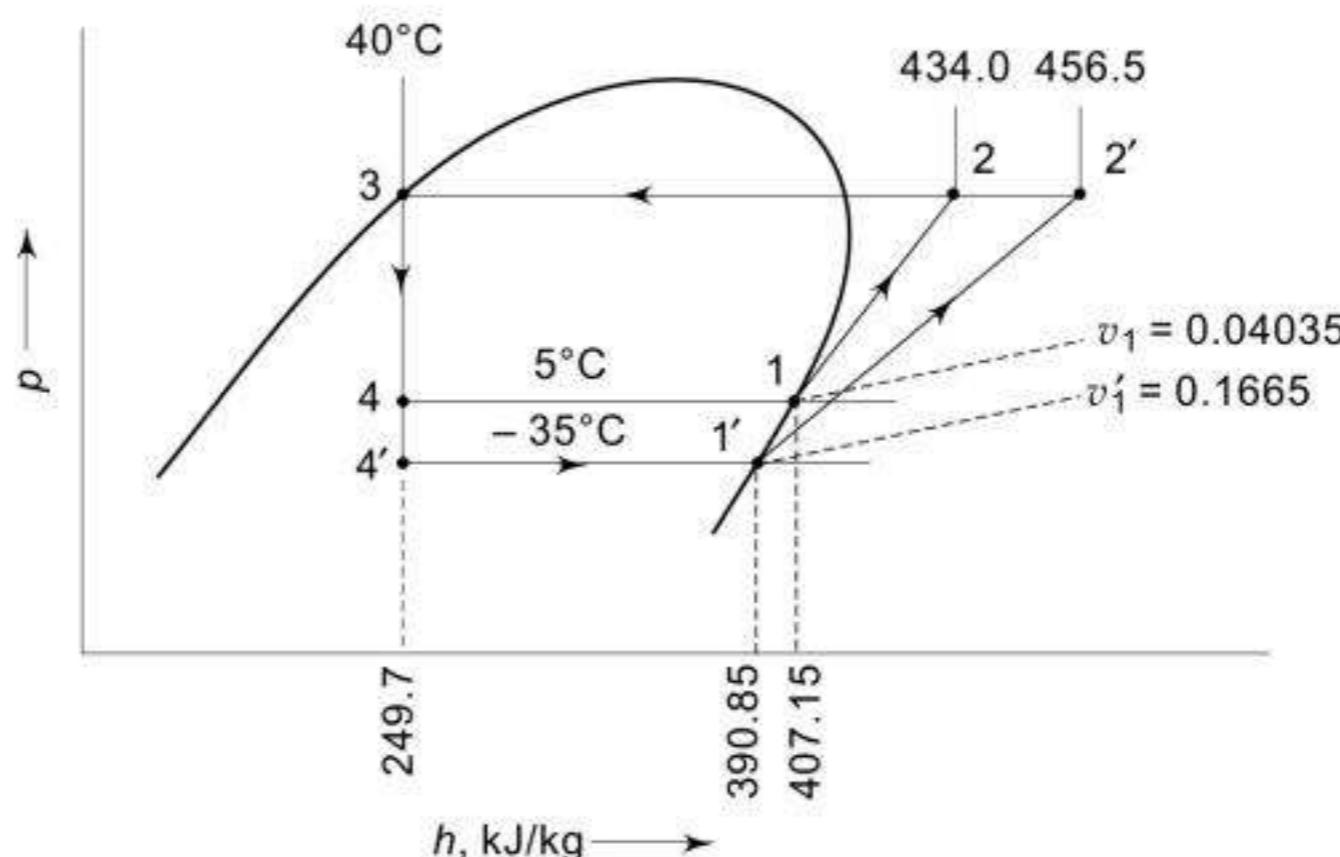


Fig. 3.13 p-h diagram for Example 3.6

(ii) At $t_0 = -35^\circ\text{C}$, since \dot{V}_p remains the same

$$\dot{m}' = \frac{\dot{V}_p}{v'_1} = \frac{0.036}{0.1665} = 0.2164 \text{ kg/s}$$

$$q'_0 = h'_1 - h'_4 = 390.85 - 249.7 = 141.15 \text{ kJ/kg}$$

$$\dot{Q}'_0 = \dot{m}' q'_0 = 0.2164 (141.15) = 30.15 \text{ kW (6.6 TR)}$$

Thus, the compressor capacity would be reduced to 6.6 TR only from 40 TR. The compressor motor would become very much undersize as its power requirement would be

$$\begin{aligned}\dot{W} &= \dot{m}' w' = 0.2164 (456.5 - 390.85) \\ &= 142.1 \text{ kW}\end{aligned}$$

as against the existing motor of 24 kW only.

On the other hand, the condenser would be very much oversize as the heat rejected would be reduced to only

$$\begin{aligned}\dot{Q}'_k &= \dot{m}' q'_k = 0.2164 (456.5 - 249.7) \\ &= 44.8 \text{ kW}\end{aligned}$$

3.5.2 Effect of Condenser Pressure

An increase in condenser pressure, similarly results in a decrease in the refrigerating capacity and an increase in power consumption, as seen from the changed cycle 1-2'-3'-4' for $t'_k = 45^\circ\text{C}$ in Fig. 3.14. The decrease in refrigerating capacity is due to a decrease in the refrigerating effect and volumetric efficiency. The increase in power consumption is due to increased mass flow (due to decreased refrigerating effect) and an increase in specific work (due to increased pressure ratio), although the isentropic line remains unchanged. Accordingly, one can write for the ratios

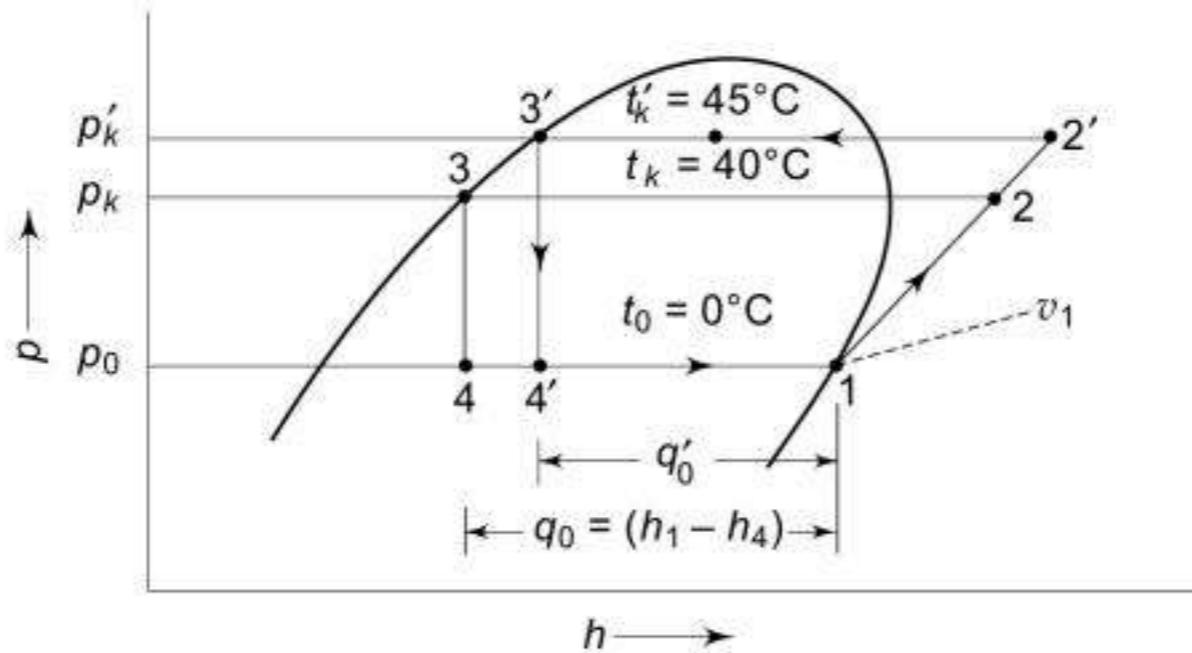
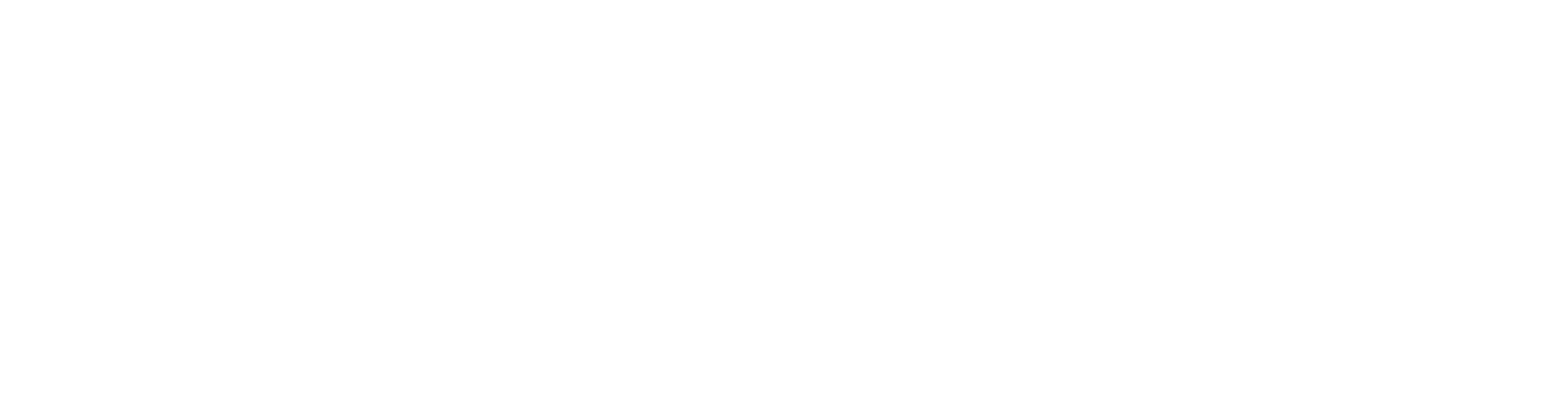


Fig. 3.14 Effect of condenser pressure

$$\frac{Q'_0}{Q_0} = \frac{h_1 - h'_4}{h_1 - h_4} \cdot \frac{\eta'_v}{\eta_v} \quad (3.26)$$

$$\frac{W^{*'}}{W^*} = \frac{\mathcal{E}'_c}{\mathcal{E}_c} = \frac{h_1 - h_4}{h_1 - h'_4} \cdot \frac{h'_2 - h_1}{h_2 - h_1} \quad (3.27)$$



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- (a) A Freon 12 simple saturation cycle operates at temperatures of 35°C and -15°C for the condenser and evaporator respectively. Determine the COP and HP/TR of the system.
- (b) If a liquid-vapour heat exchanger is installed in the system, with the temperature of the vapour leaving the heat exchanger at 15°C, what will be the change in the COP and HP/TR?

Solution (a) Simple saturation cycle Refer to Fig. 3.18. We have from tables of properties of Freon 12

t °C	h_f kJ/kg	h_g kJ/kg	s_g kJ/(kg.K)	Superheated			
				20 K		40 K	
				h kJ/kg	s kJ/(kg.K)	h kJ/kg	s kJ/(kg.K)
35	69.5	201.5	0.6839	216.4	0.731	231.0	0.7741
-15		181.0	0.7052	193.2	0.751	205.7	0.7942

Hence

$$h_3 = h_4 = 69.5 \text{ kJ/kg}$$

$$h_1 = 181 \text{ kJ/kg}$$

$$s_1 = 0.7052 \text{ kJ/(kg.K)} = s_2$$

By interpolation for the degree of superheat at discharge

$$\Delta t = \frac{0.7052 - 0.6839}{0.731 - 0.6839} (20) = 9.04^\circ\text{C}$$

Hence

$$h_2 = 201.5 + \frac{10.2}{20} (216.4 - 201.5) = 208.2 \text{ kJ/kg}$$

$$q_0 = 181 - 69.5 = 111.5 \text{ kJ/kg}$$

$$w = 208.2 - 181 = 27.2 \text{ kJ/kg}$$

$$\text{COP} = \frac{111.5}{27.2} = 4.09$$

$$\text{HP/TR} = \frac{4.761}{4.09} = 1.16$$

- (b) Liquid-vapour heat exchanger cycle

Degree of superheat at suction

$$t'_1 - t_1 = 15 - (-15) = 30^\circ\text{C}$$

By interpolation for superheated vapour

$$h'_1 = 193.2 + \frac{10}{20} (205.7 - 193.2) = 199.45 \text{ kJ/kg}$$

$$s'_1 = 0.751 + \frac{10}{20} (0.7942 - 0.751) = 0.7726 \text{ kJ/(kg.K)} = s'_2$$

$$\text{Now, } h'_1 - h_1 = 199.45 - 181 = 18.45 = h_3 - h'_3$$

$$\text{Hence, } h'_3 = 69.5 - 18.45 = 51.05 \text{ kJ/kg}$$

By interpolation for the degree of superheat

$$\Delta t = 20 + \frac{0.7726 - 0.731}{0.7741 - 0.731} (20) = 20 + 19.3 = 39.3^\circ\text{C}$$

Hence $h'_2 = 216.4 + \frac{19.2}{20} (231 - 216.4) = 230.5 \text{ kJ/kg}$
 $q'_0 = h_1 - h'_3 = 181 - 51.05 = 129.95 \text{ kJ/kg}$
 $w = 230.4 - 199.45 = 30.95$

$$\text{COP} = \frac{129.95}{30.95} = 4.199$$

Increase in $\text{COP} = \frac{4.199 - 4.09}{4.09} (100) = 2.56\%$

$$\text{HP/TR} = \frac{4.761}{4.199} = 1.134$$

Decrease in HP/TR = $\frac{1.16 - 1.134}{1.2} (100) = 2.5\%$

Note Theoretically, the increase in COP is not very large for Freon 12. And for Freon 22 there is, in fact, a decrease in COP. However, superheat improves the performance by ensuring complete vaporization of liquid.

In refrigerators and air conditioners, the capillary tube is joined to the suction line, thus forming a regenerative heat exchanger.

Example 3.8 Using Liquid-Vapour Regenerative Heat Exchanger in R 134a Systems

- (a) An R 134a simple saturation cycle refrigerator operates at 40°C condenser and -16°C evaporator temperatures. Determine COP and HP/TR.
- (b) If a liquid-vapour regenerative heat exchanger is installed in the system, with the suction vapour at 15°C, what will be the effect on COP and HP/TR?

Solution Refer Fig. 3.18

At $t_k = 40^\circ\text{C}$	$p_k = 1.0166 \text{ MPa}$
$h_3 = 256.41 \text{ kJ/kg}$	$h_g = 419.43 \text{ kJ/kg}$
$C_p = 1.145 \text{ kJ/kg K}$	$s_g = 1.711 \text{ kJ/kg}$
At $t_0 = -16^\circ\text{C}$	$p_0 = 0.15728 \text{ MPa}$
$h_1 = 389.02 \text{ kJ/kg}$	$s_1 = 1.7379 \text{ kJ/kg K}$
$C_p = 0.831 \text{ kJ/kg K}$	

- (a) *Simple saturation cycle* Calculations for isentropic compression give for discharge temperature

$$s_2 = s_g + C_p \ln \frac{T_2}{T_g} = s_1$$

$$1.7111 + 1.145 \ln \frac{320.5}{313} = 1.7379$$

Discharge temperature and enthalpy

$$T_2 = 320.5 \text{ K (47.4°C)}$$

$$\begin{aligned} h_2 &= h_g + C_p(t_2 - t_g) \\ &= 419.43 + 1.145(47.4 - 40) = 427.92 \text{ kJ/kg} \end{aligned}$$

Refrigerating effect and specific work

$$\begin{aligned} q_0 &= h_1 - h_3 = 389.02 - 256.41 = 132.61 \text{ kJ/kg} \\ w &= h_2 - h_1 = 427.92 - 389.02 = 38.9 \text{ kJ/kg} \end{aligned}$$

COP and HP/TR

$$\begin{aligned} E_c &= \frac{q_0}{w} = \frac{132.61}{38.9} = 3.41 \\ \text{HP/TR} &= \frac{4.761}{E_c} = \frac{4.761}{3.41} = 1.396 \end{aligned}$$

- (b) *Liquid-vapour regenerative heat exchanger cycle* Enthalpy and entropy of suction vapour

$$\begin{aligned} h'_1 &= h_1 + C_p(t'_1 - t_1) \\ &= 389.02 + 0.831(31) = 414.78 \text{ kJ/kg} \end{aligned}$$

$$\begin{aligned} s'_1 &= s_1 + C_p \ln \frac{T'_1}{T_1} \\ &= 1.7379 + 0.831 \ln \frac{273 + 15}{273 - 16} = 1.8325 \text{ kJ/kg K} \end{aligned}$$

Isentropic compression gives

$$s'_2 = s'_1 = 1.8325 = s_g + C_p \ln \frac{T'_2}{T_g} = 1.7111 + 1.145 \ln \frac{T'_2}{313}$$

Discharge temperature and enthalpy

$$\begin{aligned} \Rightarrow T'_2 &= 348 \text{ K (75°C)} \\ h'_2 &= h_g + C_p(t'_2 - t_g) \\ &= 419.43 + 1.145(75 - 40) = 459.5 \text{ kJ/kg} \end{aligned}$$

Energy balance of regenerative heat exchanger gives

$$q = h'_1 - h_1 = 414.78 - 389.02 = 25.76 \text{ kJ/kg} = h_3 - h'_3$$

Refrigerating effect and specific work

$$q'_0 = q_0 + q = 132.61 + 25.76 = 158.37 \text{ kJ/kg}$$

$$w' = h'_2 - h'_1 = 459.5 - 414.78 = 44.72 \text{ kJ/kg}$$

COP and HP/TR

$$\begin{aligned} E_c &= \frac{q'_0}{w'} = \frac{158.37}{44.72} = 3.54 \\ \text{HP/TR} &= \frac{4.761}{E_c} = \frac{4.761}{3.54} = 1.345 \end{aligned}$$

There is an increase of 3.8% in COP, and decrease in HP/TR by the same percentage nearly.



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- (iii) Pressure drop in the suction line, 1b-1a.
- (iv) Pressure drop due to *wire drawing* at the compressor-suction valve, 1a-1.
- (v) Polytropic compression 1-2 with friction and heat transfer to the surroundings instead of isentropic compression.
- (vi) Pressure drop at the compressor-discharge valve, 2-2a.
- (vii) Pressure drop in the delivery line, 2a-2b.
- (viii) Heat loss and desuperheating of the vapour in the delivery line, 2b-2c.
- (ix) Pressure drop in the condenser, 2b-3.
- (x) Subcooling of the liquid in the condenser or subcooler, 3-3a.
- (xi) Heat gain in the liquid line, 3a-3b. The lines 3-3a and 3a-3b are along the saturated liquid line on the *T-s* diagram as the constant pressure lines in liquid region run close to it.
- (xii) Pressure drop in the evaporator, 4-1d.

It may be noted that the pressure drop in the evaporator is large. This is due to the cumulative effect of two factors. Firstly, the pressure drops in the evaporator due to friction. This is called the *frictional pressure drop*. Secondly, as evaporation proceeds, the volume increases, and hence the velocity must also increase. The increase in kinetic energy comes from a decrease in enthalpy and, therefore, from a further pressure drop. This pressure drop is called the *momentum pressure drop*.

In the condenser, the pressure drop is not significant, since the frictional pressure drop is positive and the momentum pressure drop is negative.

It is, therefore, to be noted that pressure drop in the condenser is not very critical. But, it is very much so in the design of evaporators as it would increase power consumption greatly or alternatively reduce refrigerating capacity (see Sec. 9.5). The condensers, in any case, are usually oversized to keep the condenser pressure low and also to let condensers function as receivers. However, in the case of evaporators, both undersizing and oversizing are harmful.

It may also be noted that due to the pressure drop in the evaporator from p_{01} to p_{02} , the temperature in the evaporator does not remain constant. It correspondingly changes from t_{01} to t_{02} .

Further, due to various pressure drops, the capacity of the plant is decreased and the unit power consumption (per unit of refrigeration) is increased. Correspondingly, the COP of the actual cycle is reduced.

The compressor cylinder is hotter than the surroundings. Thus it loses heat to the refrigerant. The cooling of the compressor reduces the work of compression. Freon compressors are, therefore, provided with parabolic fins built into the body of the compressor cylinders on the outside during casting. The compressors are thus cooled by air by natural convection. Ammonia and even Freon 22 compressors are *water-jacketed* for the same purpose. Friction, however, increases the work of compression. The actual work of compression can be determined by knowing the initial and the final states and finding out n , the polytropic index of the compression curve. The heat rejected in the polytropic compression process can be obtained by applying the SSSF energy equation

$$\begin{aligned} q &= (h_2 - h_1) + w \\ \Rightarrow -w &= (h_2 - h_1) - q \end{aligned} \quad (3.35)$$

Equation (3.35) represents the energy balance of the compressor. It means:

Work of compression = Increase in enthalpy of gas + Heat lost in cooling

Example 3.9 The following data were obtained from a test on a twin cylinder, single acting 15 cm × 20 cm, 320 rpm compressor ammonia refrigeration plant.

Temperatures of refrigerant:

After expansion valve, entering brine cooler	-25°C
Leaving brine cooler	-18°C
Entering compressor	-8°C
Leaving compressor	140°C
Entering condenser	130°C
Leaving condenser	30°C
Entering expansion valve	32°C

Pressures of refrigerant:	
Compressor discharge and condenser	13.5 bar
Compressor suction	1.324 bar
Brine circulation rate	102 kg/min
Temperature drop of brine in cooler	7°C
Specific heat of brine	3.14 kJ/(kg) (K)
Input to motor	18.8 kW
Motor efficiency at this load	92%
Compressor jacket cooling water	5 kg/min
Temperature rise of jacket water	8.9°C

Show the thermodynamic states at various points on p-h and T-s diagrams and calculate:

- Refrigerating capacity in TR assuming 2 per cent loss of useful refrigeration by heat gain from room in brine cooler.
- Ammonia circulated.
- Compressor IHP and mechanical efficiency.
- Compressor volumetric efficiency.
- COP of the cycle.

Solution Referring to Fig. 3.19, the various state points are given below:

State 3b: $t = 32^\circ\text{C}$	$p = 13.5 \text{ bar}$	$h = 351.5 \text{ kJ/kg}$ (at 32°C)
State 4: $t = -25^\circ\text{C}$	$p = 1.516 \text{ bar}$	$h = 351.5 \text{ kJ/kg}$ (saturation)
State 1d: $t = -28^\circ\text{C}$ (saturation)		$p = 1.324 \text{ bar}$
State 1c: $t = -18^\circ\text{C}$	$p = 1.324 \text{ bar}$	$h = 1451.3 \text{ kJ/kg}$
State 1b: $t = -8^\circ\text{C}$	$p = 1.324 \text{ bar}$	$h = 1473.6 \text{ kJ/kg}$
State 1: $t = -8^\circ\text{C}$		$h = 1473.6 \text{ kJ/kg}$ ($= h_{1b}$)
States 2, 2a and 2b:		
	$t = 140^\circ\text{C}$	$h = 1777 \text{ kJ/kg}$
State 2c: $t = 130^\circ\text{C}$	$p = 13.5 \text{ bar}$	$h = 1751 \text{ kJ/kg}$
State 3: $t = 35^\circ\text{C}$ (saturation at 13.5 bar)		$p = 13.5 \text{ bar}$
State 3a: $t = 30^\circ\text{C}$	$p = 13.5 \text{ bar}$	$h = 341.8 \text{ kJ/kg}$

- (a) Refrigerating capacity (from brine)

$$\begin{aligned}\dot{Q}_0 &= 102 \times 3.14 \times 7 \times 1.02 = 2287 \text{ kJ/min} \\ &= \frac{2287}{211} = 10.84 \text{ TR}\end{aligned}$$

(b) Refrigerating effect

$$q_0 = h_{1c} - h_4 = 1451.3 - 351.5 = 1099.8 \text{ kJ/kg}$$

Ammonia circulated

$$\dot{m} = \frac{2287}{1099.8} = 2.08 \text{ kg/min}$$

- (c) Note: From 2 to 2b, it is an isenthalpic process. It has been assumed here that enthalpy is a function of temperature only and hence $t_{2b} = t_2$.
Enthalpy increase during compression

$$h_2 - h_1 = 1777 - 1473.6 = 303.4 \text{ kJ/kg}$$

Total enthalpy increase of ammonia

$$H_2 - H_1 = \dot{m} (h_2 - h_1) = 2.08 (303.4) = 631.1 \text{ kJ/min}$$

$$\text{Heat to jacket water, } \dot{Q}_j = 5 \times 4.1868 \times 8.9 = 186.3 \text{ kJ/min}$$

$$\begin{aligned} \text{Compressor work, } |\dot{W}| &= (H_2 - H_1) + |\dot{Q}_j| \\ &= 631.1 + 186.3 = 817.4 \text{ kJ/min} \\ &= 13.62 \text{ kW} \end{aligned}$$

$$\text{Compressor IHP} = \frac{13.62 \times 10^3}{746} = 18.26$$

$$\begin{aligned} \text{Compressor input} &= \text{Power consumption of motor} \times \text{motor efficiency} \\ &= 18.8 \times 0.92 = 17.3 \text{ kW} \end{aligned}$$

$$\text{Compressor BHP} = \frac{17.3 \times 10^3}{746} = 23.19$$

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{IHP}}{\text{BHP}} = 0.787$$

(d) Gas enters compressor at 1b

$$v_{1b} = v_g \cdot \frac{273 + (-8)}{273 + (-28)}$$

But $v_g = 0.88 \text{ m}^3/\text{kg}$ (saturated vapour at -28°C)

Hence

$$v_{1b} = 0.88 \left(\frac{265}{245} \right) = 0.952 \text{ m}^3/\text{kg} \quad (\text{It can also be obtained directly from the } p\text{-}h \text{ diagram})$$

Actual volume flow rate of refrigerant

$$\dot{V} = \dot{m}v_{1b} = 2.08 \times 0.952 = 1.99 \text{ m}^3/\text{min}$$

Compressor piston displacement

$$\dot{V}_p = \frac{\pi}{4} D^2 L N (2) = \frac{\pi}{4} (0.15)^2 (0.2) (320) (2) = 2.62 \text{ m}^3/\text{min}$$

Volumetric efficiency of the compressor

$$\eta_v = \frac{\dot{V}}{\dot{V}_p} = \frac{1.99}{2.62} = 0.756 \text{ (or 75.6%)}$$

$$(e) \text{ COP} = \frac{\dot{Q}_0}{\dot{W}} = \frac{2287}{60 \times 13.62} = 2.797$$



3.7 STANDARD RATING CYCLE FOR DOMESTIC REFRIGERATORS³

Figure 3.21 shows a standard *ten-state-points cycle*. This cycle approximates the design and operating conditions of a domestic refrigerator. The ten states points correspond to the following conditions:

- (i) Vapour in cylinder before compression begins at $p_1 = p_{10} - \Delta p_s$, $t_1 = t_{10}$ where Δp_s is pressure drop at suction valve.
- (ii) Vapour in cylinder after compression ends, 2s after isentropic, and 2n after actual polytropic compression at $p_2 = p_k + \Delta p_d$, where Δp_d is pressure drop at delivery valve.
- (iii) Vapour at compressor shell outlet/condenser inlet at $p_3 = p_k$, $t_3 = t_2$.
- (iv) Saturated vapour state in condenser at $t_4 = t_k = 55^\circ\text{C}$ condensing temperature.
- (v) Saturated liquid state in condenser at $t_5 = t_k = 55^\circ\text{C}$.
- (vi) Subcooled liquid leaving condenser at $t_6 = 43^\circ\text{C}$.
- (vii) Subcooled liquid leaving regenerator/entering capillary at $t_7 = 32^\circ\text{C}$.
- (viii) Liquid-vapour mixture exiting capillary/entering evaporator at $t_8 = t_0 = -25^\circ\text{C}$ evaporator temperature.
- (ix) Saturated vapour leaving evaporator at $p_9 = p_8 - \Delta p_E$ where Δp_E is pressure drop in evaporator.
- (x) Superheated vapour leaving regenerator/entering compressor at $t_{10} = 32^\circ\text{C}$.

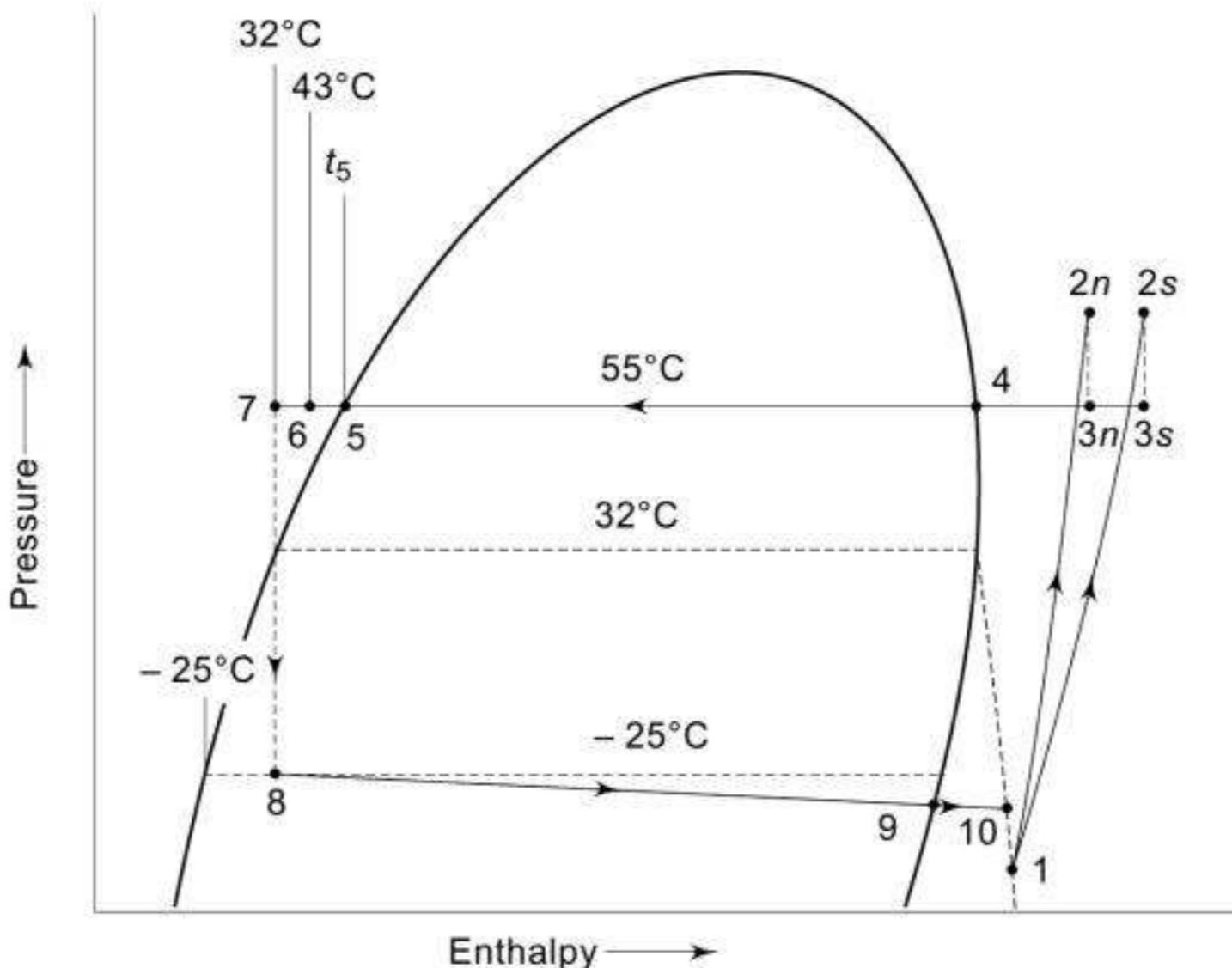


Fig. 3.21 Standard ten state point cycle

Note that this cycle corresponds to an ambient temperature of 43°C. Subcooling of liquid from $t_6 = 43^\circ\text{C}$ to $t_7 = 32^\circ\text{C}$ takes place in the regenerator. Superheating of vapour to $t_{10} = 32^\circ\text{C}$ takes place in the suction line, regenerator and the compressor. For Freon 12, the following pressure drops are assumed:

$$\Delta p_E = 0.1 \text{ bar}, \Delta p_s = 0.1 \text{ bar}, \Delta p_d = 0.25 \text{ bar}$$

Example 3.10 Thermodynamic Calculations for 165 L Freon 12 Domestic Refrigerator

An 89 W refrigerating capacity 165 L Freon 12 domestic refrigerator operates on the standard cycle (Fig. 3.21). Determine:

- (i) Isentropic discharge temperature.
- (ii) Actual discharge temperature if experimental value of polytropic index n is found to be 1.032.
- (iii) Motor watts (isentropic).
- (iv) Heat rejected in the condenser.
- (v) Volumetric efficiency of the compressor if its cylinder volume is 4.33 cc, and rpm of its motor is 2800.

Solution The pressures are:

$$p_k = p_3 = p_4 = p_5 = p_6 = p_7 = (p^{\text{sat}})_{55^\circ\text{C}} = 13.61 \text{ bar}$$

$$p_0 = p_8 = (p^{\text{sat}})_{-25^\circ\text{C}} = 1.24 \text{ bar}$$

$$p_9 = p_{10} = p_8 - \Delta p_E = 1.24 - 0.1 = 1.14 \text{ bar}$$

$$t_9 = (t^{\text{sat}})_{1.14 \text{ bar}} = -27^\circ\text{C}$$

$$p_1 = p_{10} - \Delta p_s = 1.14 - 0.1 = 1.04 \text{ bar}$$

$$t_1 = t_{10} = 32^\circ\text{C}$$

$$p_2 = p_3 + \Delta p_d = 13.61 + 0.25 = 13.86 \text{ bar}$$

From the table of properties of Freon 12, we have:

$$h_5 = (h_f)_{55^\circ\text{C}} = 90.3 \text{ kJ/kg}$$

$$h_6 = (h_f)_{43^\circ\text{C}} = 77.65 \text{ kJ/kg}$$

$$h_7 = (h_f)_{32^\circ\text{C}} = 66.56 \text{ kJ/kg} = h_8$$

$$h_9 = (h_g)_{-27^\circ\text{C}} = 175.6 \text{ kJ/kg}$$

$$h_{10} = 211 \text{ kJ/kg} = h_1$$

$$s_1 = 0.85 \text{ kJ/kg. K}$$

$$v_{10} = 0.18 \text{ m}^3/\text{kg}$$

(i) Isentropic discharge temperature,

$$s_{2s} = s_1 = 0.85 \text{ kJ/kg. K} \text{ at } p_2 = 13.86 \text{ bar} \text{ gives}$$

$$t_{2s} = 138^\circ\text{C}$$

$$\Rightarrow h_{2s} = 267 \text{ kJ/kg} = h_{3s}$$

(ii) Actual discharge temperature,

$$T_{2n} = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma-1}{\gamma}} = 305.15 \left(\frac{13.86}{1.04} \right)^{\frac{0.032}{1.032}} = 330.7 \text{ K}$$

$$t_{2n} = 57.5^\circ\text{C}$$

$$\Rightarrow h_{2n} = 234 \text{ kJ/kg} = h_{3n}$$

(iii) Refrigerating effect, mass flow rate and motor watts

$$q_0 = h_9 - h_8 = 175.6 - 66.56 = 109.1 \text{ kJ/kg}$$

$$\dot{m} = \frac{\dot{Q}_0}{q_0} = \frac{89 \times 10^{-3}}{109.1} = 0.817 \times 10^{-3} \text{ kg/s}$$

$$\dot{W} = \dot{m} w = \dot{m} (h_{2s} - h_1) = 0.817 \times 10^{-3} (267 - 211)$$

$$= 0.046 \text{ kW (46 W)}$$

Note The actual power of motor in the hermetic unit of 165 L refrigerator is 110 W. This takes care of compressor and motor efficiencies.

(iv) Condenser heat rejected

$$\dot{Q}_k = \dot{m} q_k = \dot{m} (h_{3n} - h_6) = 0.817 \times 10^{-3} (234 - 77.65)$$

$$= 0.128 \text{ kW (128 W)}$$

(v) Suction vapour volume

$$V_s = \frac{60\dot{m}}{N} v_{10} = \frac{0.817 \times 10^{-3} (0.18)}{2800/60}$$

$$= 3.1513 \times 10^{-6} \text{ m}^3/\text{rev} = 3.1513 \text{ cc/rev}$$

Volumetric efficiency of compressor

$$\eta_v = \frac{V_s}{V_p} = \frac{3.1513}{4.33} = 0.728 (72.8\%)$$

Note Results for alternatives to CFC 12 in refrigerators are discussed in Chap. 4.

3.7.1 Pull-Down Characteristic and Ice-Making Time of Refrigerators³

According to the Indian Standards Institution (ISI) specification, for the testing of refrigerators, the environment temperature is maintained at 43°C. The *no-load test* is performed by adjusting the thermostat position corresponding to an average cabinet temperature of 7°C. The purpose of this test is to find the *pull-down period*, the no-load power consumption, and the percentage running time as per ISI code.

The pull-down period is the time required to reach the specified temperatures inside the cabinet after switching on the unit. The temperature at the geometric centre of the evaporator shall not exceed –5°C at pull-down.

Another test performed on the refrigerators is the *ice-making time* test. For this test, after stable operating conditions are obtained following pull-down period, a quantity of 0.5 kg of water at $30 \pm 1^\circ\text{C}$ in two standard ice trays is kept in the freezer, and the time for each ‘ON’ and ‘OFF’ is noted. After specified time of 3 hours, the ice trays are examined for the formation of ice, and the *running time* of the unit during the 3 hours of operation is obtained, say, from the data acquisition system monitoring ‘ON-OFF’ cycle. This running time of the unit is the ice-making time. If ice is not formed during 3 hours of operation, the system operation is to be continued.

For the case of Freon 12 refrigerator of Example 3.8, the pull down period is 90 minutes, and the ice making time is 125 minutes.

Purpose of these tests is to compare the values for different refrigerators, and for different refrigerants. Increase in pull-down period, running time, and ice making time implies inadequate capacity.

3.8 HEAT PUMP

In mild winter climate, a heat pump output of 2 to 3 times the compressor power input may be realized. If a condensing temperature of 46°C is assumed, while the room is maintained at 24°C , for the heat-pump cycle the following values of COP for heating versus evaporator temperature may be obtained:

$$4.5^{\circ}\text{C} \text{ evaporator, } \text{COP}_H = 6.5$$

$$-18^{\circ}\text{C} \text{ evaporator, } \text{COP}_H = 4$$

Note that evaporator temperature depends on outdoor air temperature.

These values are theoretical. These do not include losses due to pressure drops, *coil frosting* and *defrosting*, etc. Inspite of the lowering of COP with the outdoor air temperature and hence the evaporator temperature specially at the time of maximum heating load, these *air-to-air heat pumps* are finding increasing applications in mild climates.

However, despite overall actual COP ranging from 1.5 to 3, the operating cost of air-to-air heat pumps exceeds the cost with conventional *fossil-fuel-equipment*. But, *water-to-air* and *water-to-water* heat pumps which use relatively higher temperatures can compete with *fossil-fuel-equipment*. Warm water is thus a better heat source than outside air. In *solar-assisted heat pumps*, solar panels can provide warm water in the $20\text{--}35^{\circ}\text{C}$ temperature range even in extreme cold climates.

Example 3.11 A simple Refrigerant 134a (Tetrafluoro-ethane) heat pump for space heating operates between temperature limits of 15 and 50°C . The heat required to be pumped is 100 MJ/h .

Determine:

- the dryness fraction of refrigerant entering the evaporator,
- the discharge temperature assuming the specific heat of vapour as $0.996 \text{ kJ/(kg)} (\text{K})$,
- the theoretical piston displacement of the compressor,
- the theoretical horsepower of the compressor, and
- the COP.

The specific volume of Refrigerant 134a saturated vapour at 15°C is $0.04185 \text{ m}^3/\text{kg}$. The other relevant properties of R134a are given below.

Saturation temperature	Pressure	Specific enthalpy		Specific entropy	
		h_f	h_g	s_f	s_g
$^{\circ}\text{C}$	MN/m^2	kJ/kg		$\text{kJ/(kg)} (\text{K})$	
15	0.4887	220.48	417.1	1.0725	1.72
50	1.318	271.62	423.4	1.24	1.7072



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Table 4.2 (Contd)

No. of F atoms	No. of H atoms					
	6-H	5-H	4-H	3-H	2-H	1-H
3-F				(b) CH ₂ F-CHF ₂ -35	CHClF-CHF ₂ 17	CHF ₃ -CCl ₂ F 38
				(a) CH ₃ -CF ₃ -47.24	CHClF-CClF ₂ 7	CClF ₃ -CCl ₂ F 47.68
R 143					CH ₂ Cl-CF ₃ 8	CCl ₃ -CF ₃ 45.9
					CHCl ₂ -CF ₃ 27.82	
					R 123	R 113
				(a) CH ₂ F-CF ₃ -26.1	CHClF-CF ₃ -12	CCl ₂ F-CF ₃ -12
4-F					(b) CHF ₂ -CHF ₂ -20	CClF ₂ -CClF ₂ 3.6
					R 134	R 114
						CHF ₂ -CF ₃
5-F						-48.1
						R 125
						R 115
6-F					C ₂ F ₆	
						-78.3
						R 116



4.3 COMPARATIVE STUDY OF METHANE DERIVATIVES IN USE BEFORE THE YEAR 2000

A study of fluoro-chloro derivatives of methane, along with their normal boiling points and designations as given in Table 4.1, makes an interesting reading. A comparative study of refrigerants in use before 2000 AD may be begun with the well-known refrigerant CFC 12 or R 12, viz., CCl_2F_2 , with its normal boiling point (N.B.P.) of -29.8°C . The refrigerant has commonly been used in small-capacity unitary equipment with reciprocating compressors, such as domestic refrigerators, water coolers, car air conditioners, etc., for refrigeration from 0 to -25°C . It is evident that R 12 will maintain positive pressure in the evaporator at -25°C . Further, because of low value of γ , R 12 has low discharge temperature. It was, therefore, found very suitable for hermetically sealed units of refrigerators on which the manufacturers gave 5 to 10 years warranty. Chances of burn-out of motor windings in hermetically sealed units are few if discharge temperature and hence winding temperature is low.

Just above R 12, along the column with zero H atoms, we have CFC 11 or R 11, viz., CCl_3F , with an N.B.P. of 23.7°C . This is a *higher boiling* refrigerant. Its boiling point is 53.5°C higher than that of R 12, and hence it is a low pressure substance. It has vacuum in the evaporator even in air conditioning applications at $t_0 = 5^\circ\text{C}$. It, therefore, has a large specific volume of the suction vapour at any evaporator temperature and was, therefore, found suitable for use in large capacity centrifugal compressors (300 TR and above) for water chillers (water at about 7°C) for cooling air in central air-conditioning plants. R 11 could not be considered suitable for use with reciprocating compressors.

Above R 11, we have R 10, i.e., CCl_4 or carbon tetrachloride. It has a very high boiling point and was not found suitable even for centrifugal compressors.

Below R 12, in the same column, we have CFC 13 or R 13 having N.B.P. of -81.5°C , and CFC 14 or R 14 with N.B.P. of -127.8°C . Both are *lower boiling*, and hence high pressure substances. Their critical temperatures are also below the normal ambient temperatures. These, therefore, cannot be condensed at normal ambient temperatures. Because of these reasons, these substances were not used in common refrigeration and air conditioning applications.

To the left of R 12, in the row with one H atom, we have another well-known refrigerant HCFC 22 or R 22, viz., CHClF_2 . It has an N.B.P. of -40.8°C which is about 10°C lower than that of R 12. It is, therefore, a comparatively higher pressure refrigerant. Consequently, it has a smaller specific volume of the suction vapour and gives about 40 per cent more capacity in a positive displacement compressor as compared to R 12. Previously, R 22 was employed for air conditioning in large capacity plants and package units above 5 TR only. But now, because of the capacity bonus, it is used even in one ton window-type air conditioners. However, its value of γ is higher than that of R 12. Therefore, it has higher compressor discharge and winding temperatures. It is, therefore, necessary to affect greater cooling of motor windings in R 22 hermetically sealed units with the help of incoming suction vapours, and also by adequate heat transfer to surrounding air, even by forced con-

vection, from its body. In any case, R 22 units have more frequent burn-out problems than R 12. Warranties are given only for 1–2 years unlike 5–10 years for R 12.

Because of its lower boiling point, R 22 also finds application in food freezing, freeze drying, etc., maintaining positive pressure in evaporators with temperatures as low as -40°C .

Above R 22, in the 1-H column, we have R 21. It is, however, a little chemically unstable. Further above is R 20 which is a very high boiling point substance unsuitable as a refrigerant. Below R 22, we have R 23 with an N.B.P. of -82.2°C . It is similar to R 13.

Other substances in Table 4.1, are either chemically unstable or highly inflammable due to the presence of H atoms in the molecule. Methane is the most flammable of all. R 40 or methyl chloride (CH_3Cl) with an N.B.P. of -23.74°C , used to be a very popular refrigerant in early 20th century. It is, however, poisonous and is odourless at the same time. Its leakage, therefore, gives no warning and may lead to death. Its use is, therefore, banned.

Note Here, we notice that substances with normal boiling points in the range -50 to $+50^{\circ}\text{C}$ only find application as refrigerants in commercial refrigeration and air conditioning.

4.4 COMPARATIVE STUDY OF ETHANE DERIVATIVES IN USE BEFORE THE YEAR 2000

A similar study can be made of ethane derivatives given in Table 4.2. These are mostly higher molecular weight and hence higher boiling substances. A few lower down in the table have their N.B.P. in the range -50 to 50°C . One of them is R 113 or $\text{C}_2\text{Cl}_3\text{F}_3$. It has two isomers. One has an N.B.P. of 45.9°C . The N.B.P. of the other is 47.6°C . It was commonly used as a refrigerant for centrifugal machines for air conditioning. It has higher boiling point than R 11. Hence it has lower pressure and larger volumes of the suction vapour as compared to R 11. It has the advantage that centrifugal units of smaller capacity, i.e., of the order of 150 TR, can be designed as against 300–500 TR of R11. However, with R 113, there is vacuum, in both, evaporator and condenser.

Below R 113, we have R 114 with N.B.P. of 3.6°C . It is neither suitable for reciprocating nor for centrifugal compressors. It was generally employed in rotary or low temperature centrifugal compressors.

Another known refrigerant is R 115. Having N.B.P. of -38°C . It is similar to R 22 in its thermodynamic characteristics. It is used as a component of the azeotropic mixture R 502. The other component in the mixture is R 22. The azeotrope has a lower pressure ratio as compared to R 22. It was therefore, used in high condensing and low evaporating temperature applications, such as heat pumps, air-cooled frozen food cabinets, high ambient temperature applications, etc., so that the pressure ratios and, consequently, the discharge temperatures are low. This is a great advantage in hermetically sealed compressor units as the temperature of the compressor body and, therefore, the temperature of the enclosed motor windings depends on the discharge temperature of the gas. A lower winding temperature ensures a lower *burn-out* rate and hence a longer life of the sealed unit.

Then we have R 116 with N.B.P. of -78.3°C which is too low. Other derivatives of ethane have not been used as refrigerants *in the past*.

Note Because of the problem of ozone layer depletion R11, R12, R113, R114, R115 and R502, all CFCs have been phased out.



4.5 REFRIGERANTS IN USE AFTER THE YEAR 2000

After the finding that CFCs, and to a lesser extent HCFCs deplete the ozone layer, over 100 countries adopted Montreal Protocol (MP) of 1987 to phase out CFCs in the year 2000, and HCFCs by the year 2030.

HFCs and HCs do not deplete the ozone layer. They can be used even after 2030. Their production and use is not regulated by the MP. These refrigerants include R32, R125, R134a, R143a, R236 fa, and R245 fa, and their azeotropic or near azeotropic blends R404A, R407C, and R410A.

HFCs have some global warming potential though some governments may consider regulating their use too.

Out of the refrigerants being used prior to 2000 AD, all CFCs R11, R12, R113, R114, and the azeotropic mixture R502 have already been phased out.

Replacements for all important R12, R11, R22, and R502 to date to be used till 2030 are as follows:

4.5.1 Replacement for R12

The replacement for R12 in unitary equipment has been the easiest. Its place, for all practical purposes, has been taken over completely by R134a.

Compare the NBP of -26.1°C of R134a, which is very close to -29.8°C , the NBP of R12.

As R134a is an HFC, its use will continue well beyond 2030 provided its global warming potential is not considered a disqualification, and provided another more suitable refrigerant is not found.

4.5.2 Replacement for R11

The replacement of R11 in large capacity centrifugal machines for air conditioning has posed problems. So far, R123 with its NBP of 27.82°C , closest to 23.71°C , the NBP of R11, has been chosen as the most efficient option for R11 in centrifugal chillers.

Another advantage of R123 is its atmospheric life-time which is the shortest among the refrigerants.

Moreover, its benefit in reducing global warming is significant. It has very very low global warming potential.

Nevertheless, R123 is an HCFC. Its use is slated for phase-out by 2030 despite its unique qualities. As an exception, R123 may continue to be used well beyond 2030.

R11 has also been replaced by HFC R134a. But since NBP of R134a (-26.1°C) is very very low compared to NBP of R11 ($+23.71^{\circ}\text{C}$), it is a much higher pressure refrigerant and it has very much smaller specific volumes of suction vapour. Hence,

tonnage of a single unit of R134a centrifugal compressor is much greater than 300–350 TR or so of a single R11 centrifugal compressor.

Further, most R134a centrifugal chillers have lower COPs, and they are few in number. R245fa, an HFC, however, is being seriously considered for use in place of R11. The NBP of R245fa is 14.9°C. It is not far from 23.71°C, the NBP of R11.

R245fa offers potential to approach R123 efficiencies. Its commercialization, however, is uncertain. Its cost is high at present. And it has global warming potential and flammability concerns also.

4.5.3 Replacement for R22

As R22 is an HCFC, its use is permitted till 2030. Also, since it is currently the most favoured refrigerant in package units and chillers, its use continues. 70% of commercial refrigeration systems still use R22.

Only in food refrigeration, its use is not being favoured. Before 2000 AD, the trend was to replace ammonia with R22. Now, this trend has been reversed. Ammonia has come back in a big way as it is environment friendly. Ammonia has even higher COP.

Ammonia enjoys wide use today in industrial applications especially with reciprocating compressors in small capacity, and screw compressors in large capacity. Food industry is preparing for R22 phase-out, and its replacement by ammonia.

However, in car air conditioners, R134a has replaced R22. R134a has lower pressures.

But HCFC R22 still dominates in air conditioning systems using positive displacement reciprocating, scroll, and screw compressors in medium capacity package units, and large capacity screw chillers. Some industries are using 50/50 percent R32/R125 blend, viz., R410A also as a replacement for R22.

4.5.4 Replacement for R502

Presently, R404A has found application in place of R502. The NBP of R404A is –46.22°C as against –45.6°C, the NBP of R502. A 50/50 per cent R125/R143a blend is also an attractive alternative to R502.

Note Industries worldwide have turned to HCFCs R22 and R123. While HCFCs have a lower ozone depletion potential than CFCs, they still damage the ozone layer. Nevertheless, use of these two HCFCs may continue well beyond 2030 because of their very favourable properties.

4.6 SELECTION OF A REFRIGERANT

Refrigerants have to be physiologically non-toxic and non-flammable. Thermodynamically, there is no working substance which could be called an ideal refrigerant. Different substances seem to satisfy different requirements and those also sometimes only partially. A refrigerant which is ideally suited in a particular application may be a complete failure in the other. In general, a refrigerant may be



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Table 4.3 (Contd)

Refrigerant	Chemical Formula	Designation	M (Molecular Weight)	t_s (N.B.P.) °C	t_c (Critical Temperature) °C	p_c (Critical Pressure) bar	v_c (Critical Volume) L/kg	t_f (Freezing Point) °C
Pentafluoro Ethane	CF ₃ CHF ₂	R 125	120.02	-48.1	66.25	36.31	1.748	
Tetrafluoro Ethane	CF ₃ CH ₂ F	R 134a	102.03	-26.07	101.06	40.56	1.942	-96.6
Monochloro-Tetrafluoro Ethane	CF ₃ CHClF	R 124	136.475	-12.05	122.5	36.34	1.786	
Dichloro-Trifluoro Ethane	CF ₃ CHCl ₂	R 123	152.93	27.82	183.79	36.74	1.818	
- Do -	CHClFCClF ₂	R 123a	152.93	29.9	189.62	38.89		
Tetrafluoro Ethane	CHF ₂ CHF ₂	R134	102.03	-19.8	118.9			
Monofluoro-Dichloro Ethane	CH ₃ CCl ₂ F	R 141b	116.9	32.1	208.0	43.39		
Trifluoro-Trichloro Ethane	CF ₂ Cl-CFCl ₂	R 113	187.89	47.68	214.1	34.15	1.735	-36.6
Tetrafluoro-Dichloro Ethane	CF ₂ Cl-CF ₂ Cl	R 114	170.91	3.6	145.8	32.75	1.715	-94
Pentafluoro-Monochloro Ethane	CF ₂ ClCF ₃	R 115	154.48	-38.0	80.0	32.36	1.68	-106
Difluoro-Monochloro Ethane	CH ₃ -CF ₂ Cl	R 142b	100.495	-9.25	137.1	42.46	2.3	-131.0
Trifluoro Ethane	CH ₃ -CF ₃	R 143a	84.04	-47.24	73.1	38.11	2.305	-111.3
Difluoro Ethane	CH ₃ -CHF ₂	R 152a	66.05	-24.02	113.3	45.2	2.717	-117.0
Ethyl Chloride	C ₂ H ₅ Cl	R 160	64.52	12.0	187.2	52.47	3.03	-138.7
n-perfluoro Butane	C ₄ F ₁₀	—	238.04	-2.0	113.2	23.3	1.588	—
Difluoro Ethane	CH ₂ F-CH ₂ F	R152	66.05	30.7				

(Contd)

Table 4.3 (Contd)

<i>Refrigerant</i>	<i>Chemical Formula</i>	<i>Designation</i>	<i>M (Molecular Weight)</i>	<i>t_s (N.B.P.) °C</i>	<i>t_c (Critical Temperature) °C</i>	<i>p_c (Critical Pressure) bar</i>	<i>v_c (Critical Volume) L/kg</i>	<i>t_f (Freezing Point) °C</i>
<i>Hydrocarbons</i>								
Ethane	C ₂ H ₄	R 170	30.06	-88.6	32.1	49.3	4.7	-183.2
Propane	C ₃ H ₈	R 290	44.1	-42.1	96.8	42.56	4.545	-187.1
<i>n</i> -butane	C ₄ H ₁₀	R 600	58.1	-0.56	153.0	35.3	4.29	-135
Isobutane	(CH ₃) ₃ CH	R 600a	58.13	-11.67	135.0	36.45	4.526	-159.6
<i>Unsaturated Hydrocarbons</i>								
Propylene	C ₂ H ₄	—	42.08	-47.7	94.4	46.0	4.2	-185
Dichloroethylene	C ₂ H ₂ Cl ₂	—	96.9	90.9	243	54.9	—	-56.6
<i>Aliphatic Amines</i>								
Methylamine	CH ₃ NH ₃	—	31.06	-6.7	156.9	74.5	—	-92.5
Ethylamine	C ₂ H ₅ NH ₃	—	45.08	7.0	164.6	54.7	—	-93

where $b = h_{fg}/R$ represents the slope of $\ln p^{\text{sat}}$ versus $1/T^{\text{sat}}$ line. In the small range of condenser and evaporator pressures, the latent heat of vaporization may be assumed as constant, so that the value of b may also be taken as constant. Thus $\ln p^{\text{sat}} - 1/T^{\text{sat}}$ relationship of refrigerants (pure substances) is nearly a straight line. The equation of the straight line is obtained by integrating Eq. (4.3)

$$\ln p^{\text{sat}} = a - \frac{b}{T^{\text{sat}}} \quad (4.5)$$

Exact equations, however, have some higher order terms as in Eq. (1.10).

Further, from the *Badylkes theory of thermodynamic similarity*¹² it is found that for substances belonging to the same family, the dimensionless *Trotton number* θ is constant. The number is defined by the expression

$$\theta = \frac{M(h_{fg})_s}{T_s} \quad (4.6)$$

where

M = molecular weight

T_s = normal boiling point

$(h_{fg})_s$ = latent heat of vaporization at normal boiling point.

Thus, from Eqs. (4.4) and (4.5), we have

$$b = -\frac{h_{fg}}{R} = -\frac{M h_{fg}}{R} = -\frac{\theta T_s}{R} = f(T_s)$$

Accordingly, the slope of the $\ln p^{\text{sat}} - 1/T^{\text{sat}}$ line is a function of the normal boiling point. Also from Eq. (4.5), at atmospheric pressure

$$a = \frac{b}{T_s} = f'(T_s)$$

Thus, both the constants in Eq. (4.5) are functions of the normal boiling point. The higher the boiling point, the steeper is the slope of $\ln p^{\text{sat}} - 1/T^{\text{sat}}$ line.

Also, the higher the boiling point, the further is the line from the origin. The $\ln p^{\text{sat}} - 1/T^{\text{sat}}$ data for a number of refrigerants is plotted in Fig. 4.1 (a).

Figure 4.1(a) is plotted for $\ln p^{\text{sat}}$ along the ordinate against $1/T^{\text{sat}}$ along the abscissa for a number of refrigerants. Since the slope of the line is negative, the plot is usually made with $-(1/T^{\text{sat}})$ along the abscissa as shown qualitatively in Fig. 4.1(b).

Figure 4.1 (b) compares the pressures of lower-boiling and higher-boiling refrigerants at given evaporator and condenser temperatures. The following points emerge from this figure:

- (i) At given T_0 and T_k , p_0 and p_k are lower for higher-boiling refrigerant. Hence, higher boiling substances are in general low pressure refrigerants. On the contrary, p_0 and p_k are higher for lower-boiling refrigerant. These are, therefore, high pressure refrigerants.
- (ii) Because of the steeper characteristic, the high-boiling refrigerants have higher-pressure ratios p_k/p_0 . And because of the flatter characteristic, the lower-boiling refrigerants have lower pressure ratios.

- (iii) In addition, since slope $b = -h_{fg}/R$, the high-boiling having steeper slope have high latent heat of vaporization. And the lower boiling having flatter slope have low latent heat of vaporization.

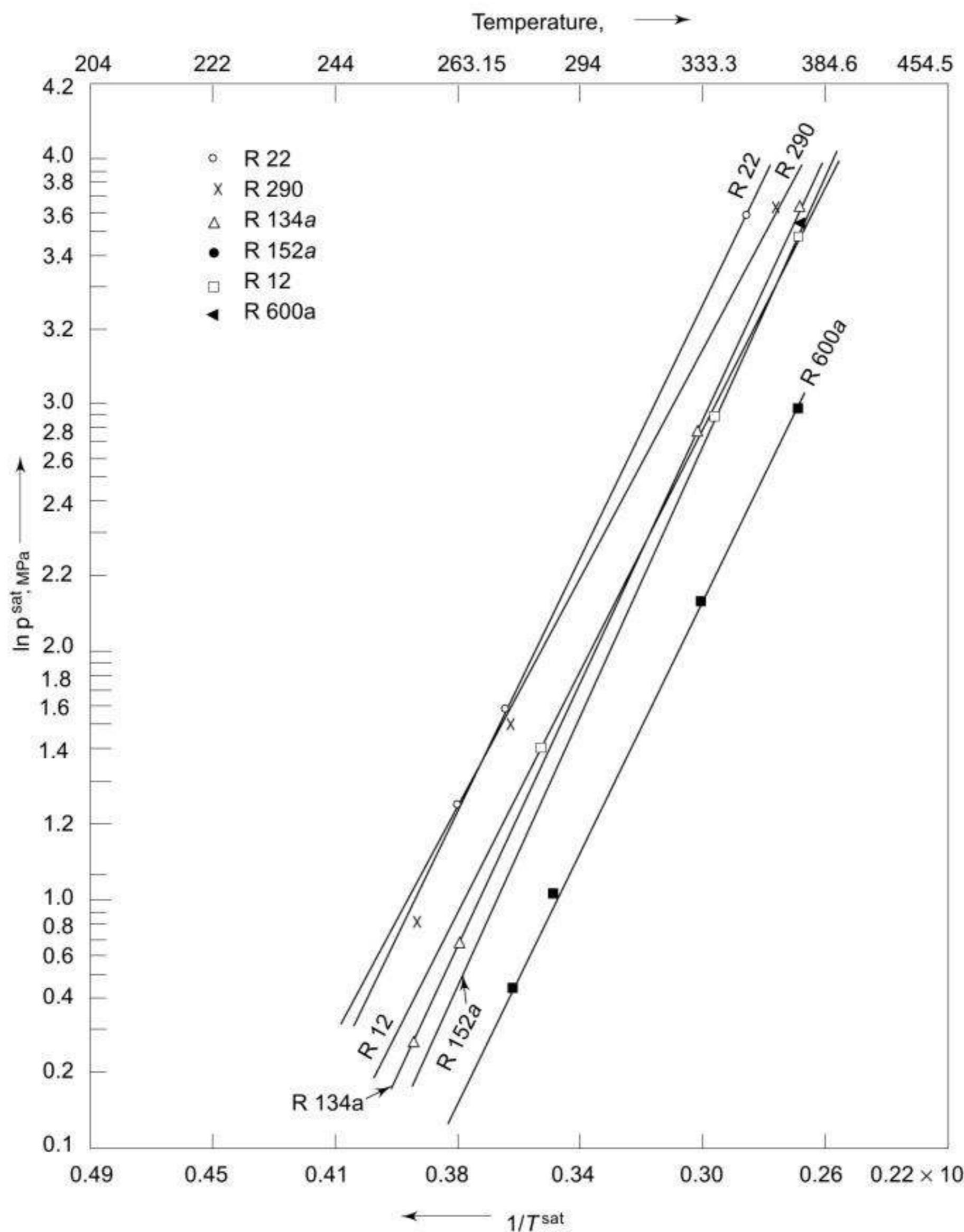


Fig. 4.1 $\ln p^{\text{sat}}$ versus $1/T^{\text{sat}}$ diagram of refrigerants



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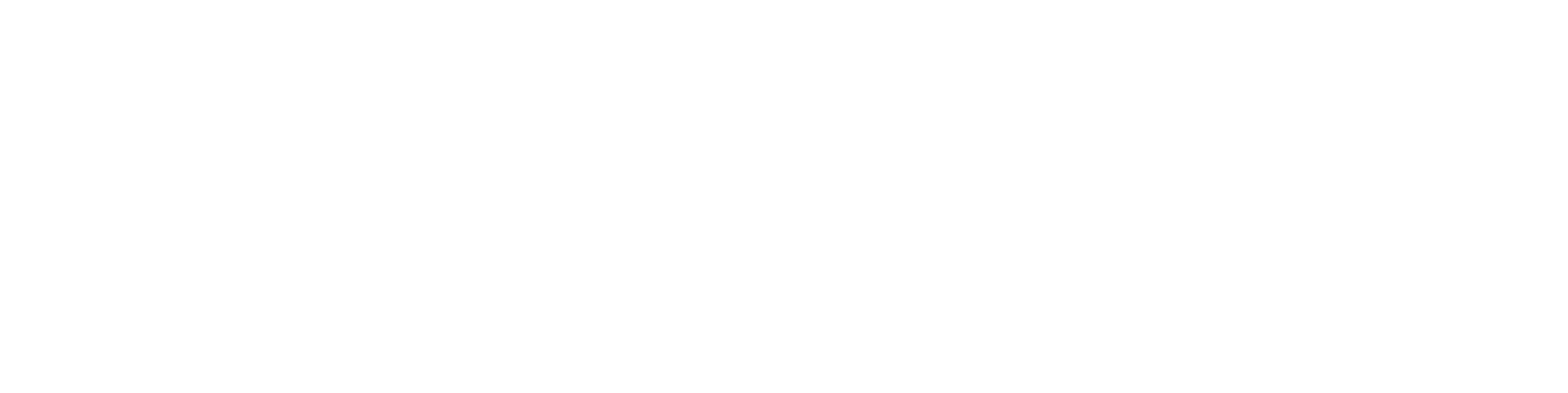
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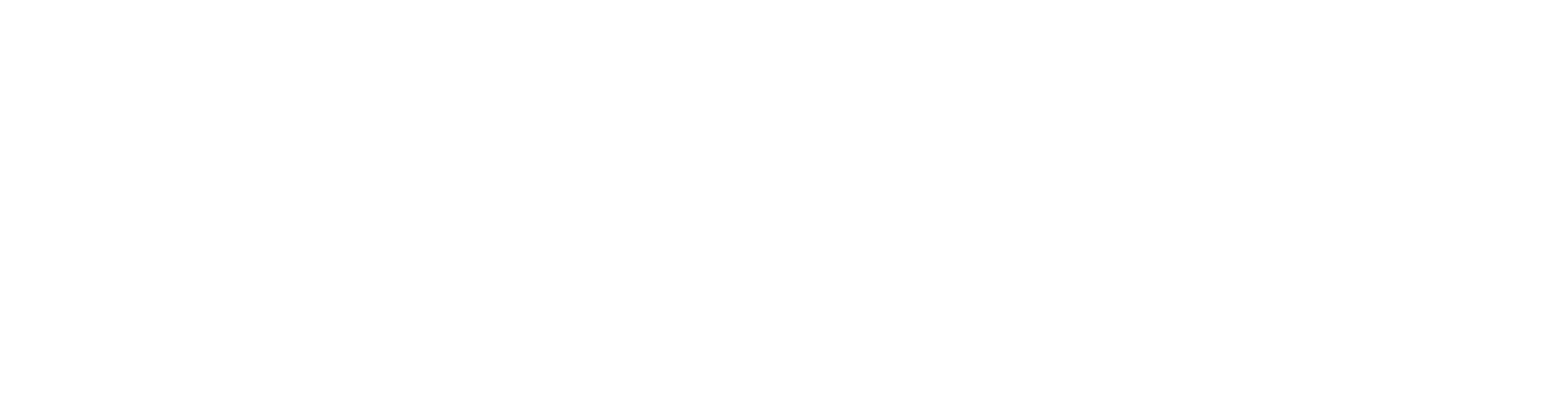
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low index of compression making it suitable for use at extreme pressure ratios, etc., and good motor winding cooling characteristics.

Presently, HFC R 134a (Tetrafluoroethane) is the most preferred substitute for R 12. Its N.B.P. of -26.67°C is quite close to R 12's N.B.P. of -29.8°C . However, it should be noted that R 134a has relatively high GWP. The use of oil in R 134a system requires a very stringent quality control. R134a is not soluble in mineral oil. The polyester-based synthetic oil that is used with it should be *totally dry*. This is difficult considering the fact that the synthetic ester oils are 100 times more *hygroscopic* than mineral oils.

Further, where the pressure ratio does not exceed 10, it is possible to use the transitional substance R 22 in place of R 12. But where the condensing temperature is high or the evaporating temperature is low (as in refrigerators), it is not possible to use R 22 because the motor winding and the compressor would *overheat*. R 22 cannot be used in car air conditioners also because the pressures would be much higher in comparison to R 12. So, with vibrations present, and car A/C not being a completely sealed system, there would be chances of frequent gas leaks. So R 134a is being used in place of R 12. It has lower pressures as well.

Why not hydrocarbons? Hydrocarbons have zero *ODP* and negligible *GWP*. Earlier researches for alternatives had excluded hydrocarbons because of their flammability. However, hydrocarbons are readily available, much cheaper and thermodynamically very suitable. Besides, HC R 290 (Propane) is already being used as refrigerant in petroleum refineries. In any case, the amount of refrigerant charge required in domestic refrigerators is only a few grams. So, propane can be considered for use at least in domestic refrigerators. In fact, it is already in use in some refrigerators in Europe.

Another attractive HC refrigerant from the list in Table 4.9 for use in domestic refrigerators is R 600a (Isobutane). Whereas R 290 is a lower boiling, higher pressure and smaller suction vapour volume substitute, R600a is higher boiling, lower pressure and larger suction vapour volume refrigerant.

But because of their widely different N.B.Ps, neither R 290 nor R 600a can be used as drop-in substitutes in place of R 12. However, by mass, a 50% R 290 + 50% R 600a mixture has exactly the same pressures as R 12. Its volume refrigerating capacity is also the same. Hence, this mixture is favoured as a drop-in substitute.

One more attractive alternative is R 152a (Difluoroethane). Its *GWP* is one order of magnitude less than that of R 134a. Hence, the Environmental Protection Agency of Europe prefers R 152a over R 134a.

4.12.1 Comparative Study of Alternatives to CFC 12 in Domestic Refrigerators

In domestic refrigerators, the inside air is cooled through natural convection by the evaporating refrigerant, and the refrigerant is also condensed through natural convection by the outside air. In frost-free refrigerators, however, a fan is used to circulate air between freezer compartment and the evaporator coil. Both in the evaporator and the condenser, one side fluid is very low thermal conductivity air. This results in very low heat transfer coefficients. Hence, air-side coefficient becomes the *controlling coefficient*. The overall heat transfer coefficient is lower than this controlling coefficient. Accordingly, whichever be the refrigerant used, the overall coefficient would not change. Hence, if an alternative refrigerant is used in a

refrigerator in place of CFC 12, it would not be necessary to change the evaporator and the condenser. Only the compressor, its motor and the capillary tube would have to be changed.

Ashok Babu⁹ has carried out an exhaustive theoretical analysis of five pure refrigerants R 290, R 22, R 134a, R 152a and R 600a, and the binary mixture 50% R 290 + 50% R 600a for finding out the suitability of alternate refrigerants, and for investigating the required modifications.

Analysis was done by Ashok Babu on a 165L, 89W refrigerating capacity, 4.33 cm³ piston displacement refrigerator operating on the standard rating cycle as described in Fig. 3.21. The pressure drops Δp_s and Δp_d at suction and discharge valves were assumed as follows:

For R 290 and R22	$\Delta p_s = 0.2$ bar,	$\Delta p_d = 0.4$ bar
For R 12, R 134a and R 152a	$\Delta p_s = 0.1$ bar,	$\Delta p_d = 0.25$ bar
For R 600a	$\Delta p_s = 0.03$ bar,	$\Delta p_d = 0.05$ bar

The operating parameters are given in Table 4.11. Two sets of calculations were done for performance parameters:

- (i) One using the same R 12 compressor with alternate refrigerants considered as drop-in substitutes.
- (ii) Another with an appropriate sized compressor to obtain the same refrigerating capacity of 89 W with different refrigerants.

The results of calculations are given in Tables 4.12 and 4.13 respectively. From Table 4.12 it is clear that the higher boiling R 600a, R 152a and R 134a give decreased capacity and the lower boiling R 22 and R 290 give increased capacity.

Further comments about the respective alternatives are as follows:

R 600a (Isobutane) It has very large volume of the suction vapour \dot{V}^*/η_v , about 1.8 times that of R12. Hence, it requires a much larger sized compressor. Pressure drop across the capillary is the lowest. Hence, R 600a requires a slightly shorter capillary than 3m of R 12. Further, R 600a has lowest value of γ . Hence, it has the lowest discharge temperature and, therefore, lowest winding temperature inspite of it having the highest pressure ratio. However, it has vacuum in the evaporator. Experiments show that it requires 1.75 times the size of R 12 compressor.

R 152a (Difluoroethane) and R 134a (Tetrafluoroethane) Both these higher boiling substances have similar characteristics except that R 152a has slight vacuum in the evaporator at -25°C, and that the discharge temperature of R 152a is much higher because of its higher value of γ . Otherwise, both refrigerants require larger displacement compressors, viz., 1.35 times for R 134a and 1.43 times for R 152a.

Further, pressure drop in R 152a capillary is the same as in the case of R 12. R 134a has larger pressure drop, 13.85 bar, as against 12.37 bar for R 12. However, the exact capillary size needs to be calculated in each case according to \dot{m} , $p_k - p_o$ and thermophysical properties.

From Table 4.12, it can be seen that the refrigerating capacity is 61 W with R 134a as against 89 W for R 12. To obtain a capacity of 89 W with R 134a as well, the following modifications can be considered:

Table 4.11 Properties data of R 12 and new proposed refrigerants for $t_0 = -25^\circ\text{C}$ and $t_k = 55^\circ\text{C}$.

Refrigerants	NBP °C	p_0 bar	p_k bar	p_2/p_1	$p_k \cdot p_0$ bar	ν_l m^3/kg	h_{fg} kJ/kg	γ
R 290	-42.07	2.02	19.07	10.71	17.05	0.2913	404.99	1.126
R 22	-40.76	2.01	21.74	12.23	19.74	0.1496	223.72	1.166
R 12	-29.79	1.24	13.61	13.47	12.37	0.1803	163.34	1.126
R 134a	-26.2	1.07	14.92	17.64	13.85	0.2592	221.83	1.102
R 152a	-25	0.98	13.32	17.67	12.34	0.4314	328.25	1.134
R 600a	-11.73	0.59	7.82	21.25	7.23	0.8876	379.49	1.086
R290/R600a		1.4	14.22	13.29	12.82	0.4142	390.13	1.104

Table 4.12 Calculated performance parameters of R 12 and new proposed refrigerants using same compressor as in R 12 refrigerator

Refrigerants	$m \times 10^3$ kg/s	n_v	t_{2n} °C	t_{2s} °C	\dot{Q}_o W	\dot{Q}_k W	\dot{W}_{ts} W	$V^* \times 10^3$ $\text{m}^3/\text{s/TR}$	W/TR	COP = \dot{Q}_o/W_{ts} Nm
R 290	0.5896	0.936	125	155	282	808	88.4	4.35	2080	1.76
R 22	1.135	0.925	170	178	307	936	101.9	4.76	1995	1.75
R 12	0.8022	0.772	56.8	89	154	555	47.5	44.47	1786	1.84
R 134a	0.4437	0.608	60.5	125	65	113	518	33.86	32.46	2.77
R 152a	0.2617	0.621	66.8	155	61	106	478	34.92	29.91	1.91
R 600a	0.1941	0.894	53.7	110	48	86	260	25.85	24.68	3.10
R 290/										2.77
R 600a	0.3039	0.612	70.8	116	77	129	609	35.0	28.1	1.75
										1.60
										1.85
										1.60
										2.93



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Note Neither hydrocarbons because of their flammability, nor R 22 because of high pressures and temperatures, would be suitable for use in car air conditioners. R 134a has been chosen for automotive air conditioning.

Example 4.1 Implications of Charging Propane in R 12 Refrigerator

Figure 4.3 shows the p-h diagram of R 12 cycle in an 89 W domestic refrigerator. State 1" leaving evaporator is at -15°C .

State 1 after regenerator and at the end of suction line is at 32°C . Process 3-3' represents subcooling in regenerative heat exchanger between vapour and liquid. State 3 leaving condenser is at 40°C .

What will be the results of charging this refrigerator with propane without changing hermetic compressor? Find to the first approximation.

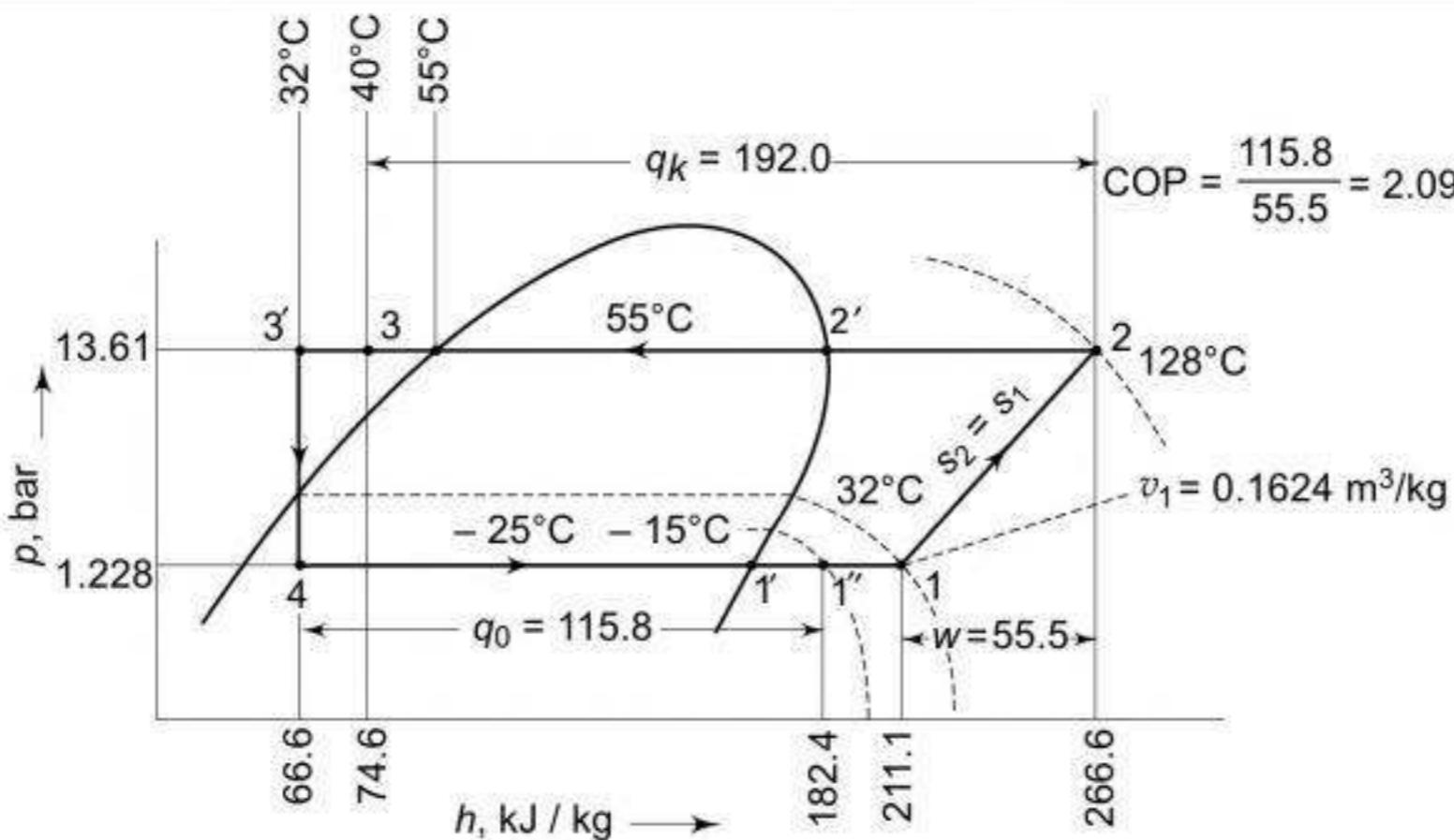


Fig. 4.3 R 12 cycle for refrigerator in Example 4.1

Solution Figure 4.4 shows the standard refrigerator cycles with propane.

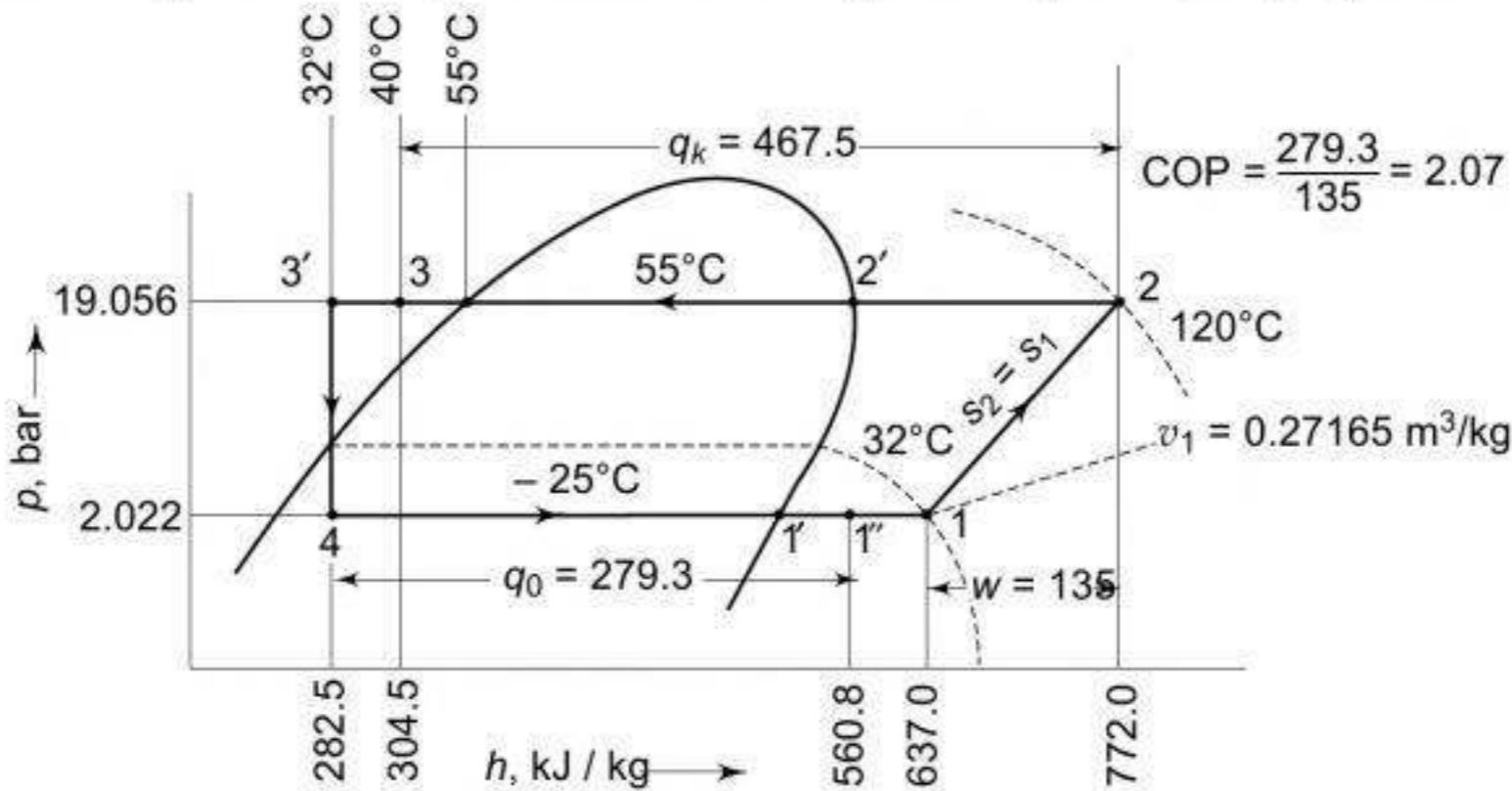


Fig. 4.4 Propane cycle for refrigerator in Example 4.1

Mass flow rate of refrigerant, piston displacement, motor watts and heat rejected in condenser for R 12 refrigerator are as follows:

$$(\dot{m})_{R\ 12} = \frac{\dot{Q}_0}{q_0} = \frac{89 \times 10^{-3}}{115.8} = 7.68 \times 10^{-4} \text{ kg/s}$$

$$\dot{V}_p = (\dot{m}v_1)_{R\ 12} = 7.68 \times 10^{-4} (0.1624) = 1.247 \times 10^{-4} \text{ m}^3/\text{s}$$

$$(\dot{W})_{R\ 12} = (\dot{m}w)_{R\ 12} = 7.68 \times 10^{-4} (55.5 \times 10^3) = 43 \text{ W}$$

$$(\dot{Q}_k)_{R\ 12} = (\dot{m}q_k)_{R\ 12} = 7.68 \times 10^{-4} (192 \times 10^3) = 147 \text{ W}$$

With the same compressor, and hence for the same piston displacement, we have with propane

$$(\dot{m})_{R\ 290} = \frac{\dot{V}_p}{v_1} = \frac{1.247 \times 10^{-4}}{0.27165} = 4.59 \times 10^{-4} \text{ kg/s}$$

$$(\dot{Q}_0)_{R\ 290} = (\dot{m}q_0)_{R\ 290} = 4.59 \times 10^{-4} (279.3 \times 10^3) = 128 \text{ W}$$

$$(\dot{W})_{R\ 290} = (\dot{m}w)_{R\ 290} = 4.59 \times 10^{-4} (135 \times 10^3) = 62 \text{ W}$$

$$(\dot{Q}_k)_{R\ 290} = (\dot{m}q_k)_{R\ 290} = 4.59 \times 10^{-4} (467.5 \times 10^3) = 215 \text{ W}$$

We clearly notice the following advantages in using the hydrocarbon HC 290 in place of CFC 12:

- (i) Refrigerating capacity is increased from 43 W to 62 W, viz., by 44%.
- (ii) COP *nearly* remains the same.
- (iii) Discharge temperature is lowered from 128°C to 120°C.

However, there are implications in charging R 290 as a ‘drop-in’ substitute in a R 12 refrigerator without changing compressor, which are as follows:

- (i) Refrigerant mass flow rate requirement decreases, due to higher latent heat and refrigerating effect of R 290

$$279.3 = (q_0)_{R\ 290} > (q_0)_{R\ 12} = 115.8 \text{ kJ/kg}$$

- (ii) Refrigerant mass flow rate handled by compressor decreases due to large specific volume of R 290 suction vapour

$$0.27165 = (v_1)_{R\ 290} > (v_1)_{R\ 12} = 0.1624 \text{ m}^3/\text{kg}$$

This reduces mass flow handled by compressor as below:

$$4.59 \times 10^{-4} = (\dot{m})_{R\ 290} < (\dot{m})_{R\ 12} = 7.68 \times 10^{-4} \text{ kg/s}$$

- (iii) Conditions (i) and (ii) are complimentary. However, the effect of higher latent heat is greater. As a result, in spite of the larger specific volume and hence smaller mass flow rate handled, the same compressor gives more capacity, and correspondingly increases heat rejected in the condenser for R 290 because of its higher latent heat of vaporization. Thus

$$128 \text{ W} = (\dot{Q}_0)_{R\ 290} > (\dot{Q}_0)_{R\ 12} = 89 \text{ W}$$

$$215 \text{ W} = (\dot{Q}_k)_{R\ 290} > (\dot{Q}_k)_{R\ 12} = 147 \text{ W}$$

- (iv) Motor watts increase for R 290 along with higher refrigerating capacity.

$$62 \text{ W} = (\dot{W})_{R\ 290} > (\dot{W})_{R\ 12} = 43 \text{ W}$$

This will result in considerable overloading and overheating of motor. Consequently, it may burn out inspite of the lower discharge temperature of R 290.

- (v) Due to the enormous amount of heat to be rejected, and use of the same condenser, the condensing temperature and pressure and hence discharge temperature would rise. In case, the mean cooling medium temperature is assumed to be 38°C , the ΔT across R 12 condenser would be $(\Delta T_k)_{\text{R}12} = 55 - 38 = 17^{\circ}\text{C}$, and that across R 290 condenser would be

$$\begin{aligned} (\Delta T_k)_{\text{R}290} &= (\Delta T_k)_{\text{R}12} \frac{(\dot{Q}_k)_{\text{R}290}}{(\dot{Q}_k)_{\text{R}12}} \\ &= 17 \left(\frac{215}{147} \right) = 25^{\circ}\text{C} \end{aligned}$$

Hence, the condensing temperature and pressure with R 290 would be

$$(t_k)_{\text{R}290} = 38 + 25 = 63^{\circ}\text{C}$$

$$(p_k)_{\text{R}290} = 22.582 \text{ bar}$$

And the isentropic discharge temperature would be close to $145\text{--}150^{\circ}\text{C}$ (much above 120°C).

- (vi) Further, if the effect of using the same capillary is also considered, we see that the pressure drops required are as follows:

	R 12	R 290
Condenser pressure	13.61	19.056
Evaporator pressure	1.228	2.022
Pressure drop in capillary	12.382 bar	17.034 bar

Then, assuming same pressure drop as in R 12 capillary, viz., 12.382 bar, and neglecting the difference in thermophysical properties, we see that the pressure of propane at exist from the capillary and entering the evaporator would be

$$(p_0)_{\text{R}290} = 22.582 - 12.382 = 10.2 \text{ bar}$$

This will correspond to an enormously high evaporation temperature of 28°C instead of -25°C . It will also start a vicious cycle of a very high density of suction vapour, extremely large mass flow rate, capacity, motor watts, heat rejected in condenser, condenser temperature and pressure, discharge temperature, etc.

Example 4.2 Make suggestions regarding changes to be made if R 12 in the refrigerator in Example 4.1 is to be replaced with propane.

Solution Mass flow rate of propane required for the same refrigerating capacity of 89 W

$$\dot{m} = \frac{0.089}{279.3} = 3.18 \times 10^{-4} \text{ kg/s}$$

Compressor

Theoretical piston displacement required

$$\dot{V}_p = 3.18 \times 10^{-4} (0.27165) = 8.64 \times 10^{-5} \text{ m}^3/\text{s}$$



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Table 4.16 Comparative Data for CFC 11, HCFC 123, HFC 245fa (Pentafluoropropane) and Ether for $t_k = 40^\circ\text{C}$ and $t_0 = 0^\circ\text{C}$

	CFC 11	HCFC 123	HFC 245fa	HFE 143
Formula	CCl_3F	$\text{CHCl}_2\text{-CF}_3$	$\text{C}_3\text{H}_3\text{F}_5$	$\text{CH}_2\text{FOCHF}_2$
Mol. Mass	137.4	152.9	135.0	100.0
$t_c, ^\circ\text{C}$	197.8	183.68	154.1	186.83
$p_c, \text{ MPa}$	4.37	3.662	3.64	4.141
N.B.P., $^\circ\text{C}$	23.7	27.82	14.9	30.06
$p_0, \text{ MPa}$	0.04	0.03265	0.054	0.029
$p_k, \text{ MPa}$	0.175	0.15447	0.252	0.145
$q_0, \text{ kJ/kg}$	154.9	141.0	151.7	231.91
$V^*, \text{ m}^3/\text{MJ}$	2.604	3.164	2.816	3.312
COP	6.06	6.1	5.861	6.023

4.14 SUBSTITUTES FOR HCFC 22

It is seen that short-term substitutes for HCFC 22 remain HCFC 22 by itself, and HFC 134a both. HFC 134a may well remain the long-term substitute also if no other suitable refrigerant is found.

However, the average of bubble and dew temperatures of HFC R 407C, a mixture of R 32, R 125, and R 134a, is closest to -40.8°C the N.B.P. of HCFC 22. And R 410A is a near azeotropic mixture of HFCs R 32 and R 125. At present, these two blends are serious contenders to replace HCFC 22.

Another HFC blend is R 404A (see Table 4.9). Other blends of HFC 32, HFC 134a etc., may also be found till the deadline of 2030.

Table 4.17 compares the performance parameters of HCFC 22 with the blend R 407C of R 32/R 125/R 134a (23/25/52%), and azeotropic mixture R 410A of R 32/R 125 (50/50%).

Table 4.17 Comparison of Performance Parameters of R 407C and R 410A with HCFC 22 for $t_k = 40^\circ\text{C}$ and $t_0 = 0^\circ\text{C}$

	HCFC 22	R407C		R410A	
$t_c, ^\circ\text{C}$	96.15		86.03		71.36
$p_c, \text{ MPa}$	4.99		4.63		4.903
N.B.P., $^\circ\text{C}$	-40.81	Bubble	Dew	Bubble	Dew
		-43.63	-36.63	-51.44	-51.36
$p_o, \text{ MPa}$	0.498		0.5		0.8
$t_o, ^\circ\text{C}$	0	-3.85	+2.36	-0.03	+0.08
$p_k, \text{ MPa}$	1.5336		1.7		2.4
$t_k, ^\circ\text{C}$	40	43.78	38.84	39.56	39.68
$q_0, \text{ kJ/kg}$	155.4		152.1		155.8
$\dot{m}, \text{ kg/MJ}$	6.435		6.5733		6.55
$v_1, \text{ m}^3/\text{kg}$	0.0471		0.04687		0.03262
$V^*, \text{ m}^3/\text{MJ}$	0.303		0.308		0.2137
$w, \text{ kJ/kg}$	25.0		20.4		28.7
COP	5.8		5.1		6.2

Note that R407C boils through -3.85°C to $+2.36^{\circ}\text{C}$ to give a mean refrigeration temperature of 0°C at 0.5 MPa. It condenses through 43.78°C to 38.84°C to give a mean condensation temperature of 40°C . So it boils and condenses over a range of temperatures of 6.2 and 4.9°C respectively. Hence, evaporators and condensers have to have counter flow designs. Issues relating to separation of refrigerant and oil in the evaporators have also to be examined.

Further, R 407C has much lower COP. But R 407C does have closely similar operating characteristics to R 22.

On the contrary, R 410A is nearly an azeotropic mixture. Its boiling and condensation ranges are negligible. It is like a pure substance. However, it is a much lower-boiling refrigerant than HCFC 22. Hence, it has high pressures.

But it is shown that R 410A has higher COP.

And higher evaporator heat transfer coefficients of R 410A have facilitated system designs with more, not less, dehumidification (refrigeration) capability as compared to R 22.

However, as the critical temperature of R 410A is lower, there is a recognized performance degradation at elevated ambient temperatures. This has effect on dehumidification capability.

On balance, performances of R 410A and R 22 systems are comparable. Hence, it is being used as a substitute to R 22.

For the values presented in Table 4.17, see Example 4.3 for calculations.

Note that R 134a, a medium pressure refrigerant is being used in Europe in conjunction with variable speed screw compressors that are competitive to R 22.

Thus, there is no clear refrigerant choice for future equipment to replace R 22 after 2030.

4.15 SUBSTITUTES FOR CFC R 502

R502 is an azeotropic mixture of HCFCs R 22 and R 115 in 50/50% proportions by mass. Its N.B.P. is -45.6°C .

A new azeotropic mixture R507A formed by blending HFC 134a and HFC 125 in 50-50% proportions by mass, and near azeotropic mixture R 404A formed by HFCs R 125/R 143/R 134a in 44/52/4% proportions make attractive alternatives.

Whereas R507A is an azeotrope having N.B.P. of -46.74°C which is very close to the N.B.P. of R 502, R 404A boils through a range from -46.22°C to -45.47°C at 1 atm pressure of 0.10132 MPa.

Note Small deviations from azeotropic behaviour occur for R 507A. Table B.17 in the Appendix gives averages of dew and bubble pressures for any given temperature.

4.16 ATMOSPHERIC GASES AS SUBSTITUTES FOR CFC REFRIGERANTS

Concern for protecting the environment is on the rise. It is, therefore, suggested that the existing atmospheric gases themselves be used as refrigerants. These are air, water vapour, carbon dioxide, hydrogen, ammonia, etc.

While ammonia is already a very popular refrigerant in industrial applications such as cold storages and ice plants, its application can grow further. This has already put a stop to the replacement of ammonia by HCFC R 22.

It is, however, a challenge to thermal engineers to adapt water and other inert atmospheric gases in varied applications and high performance refrigeration systems.

Air cycle refrigeration (Chapter 11) is already used in aircraft air conditioning. Normalair-Garrett of UK, in cooperation with Hagenuk Faiveley of Germany, have devised the system that uses air cycle technology for car air conditioners. Environmental concerns have breathed life into natural refrigerant technologies including air cycle.

Hydrogen is also used as a working substance in Stirling-cycle in a Philips gas liquifier (Chapter 11). With modifications, it can be adapted for other common applications.

Water vapour refrigeration is used in steam ejector system (Chap. 13) driven by motive steam in very large capacity air conditioning plants.

Carbon dioxide was used as a refrigerant toward the end of nineteenth century. Since its critical temperature is low (31°C only), it cannot be condensed in high ambient conditions. But it can be used in a *supercritical vapour compression cycle*. Improving its COP, however, offers a challenge to thermodynamics engineers.

4.16.1 Supercritical Vapour Compression Cycle with CO_2 as Refrigerant

Figures 4.5 and 4.6 show the supercritical vapour compression cycle with CO_2 (R 744) as refrigerant on T - s and p - h diagrams respectively. The following operating conditions have been chosen for air conditioning cycle:

Evaporation temperature 2°C ($p_0 = 36.8 bar)$

Compressor discharge/Cooler pressure 100 bar

Temperature t_3 after cooler coil 50°C (323 K)

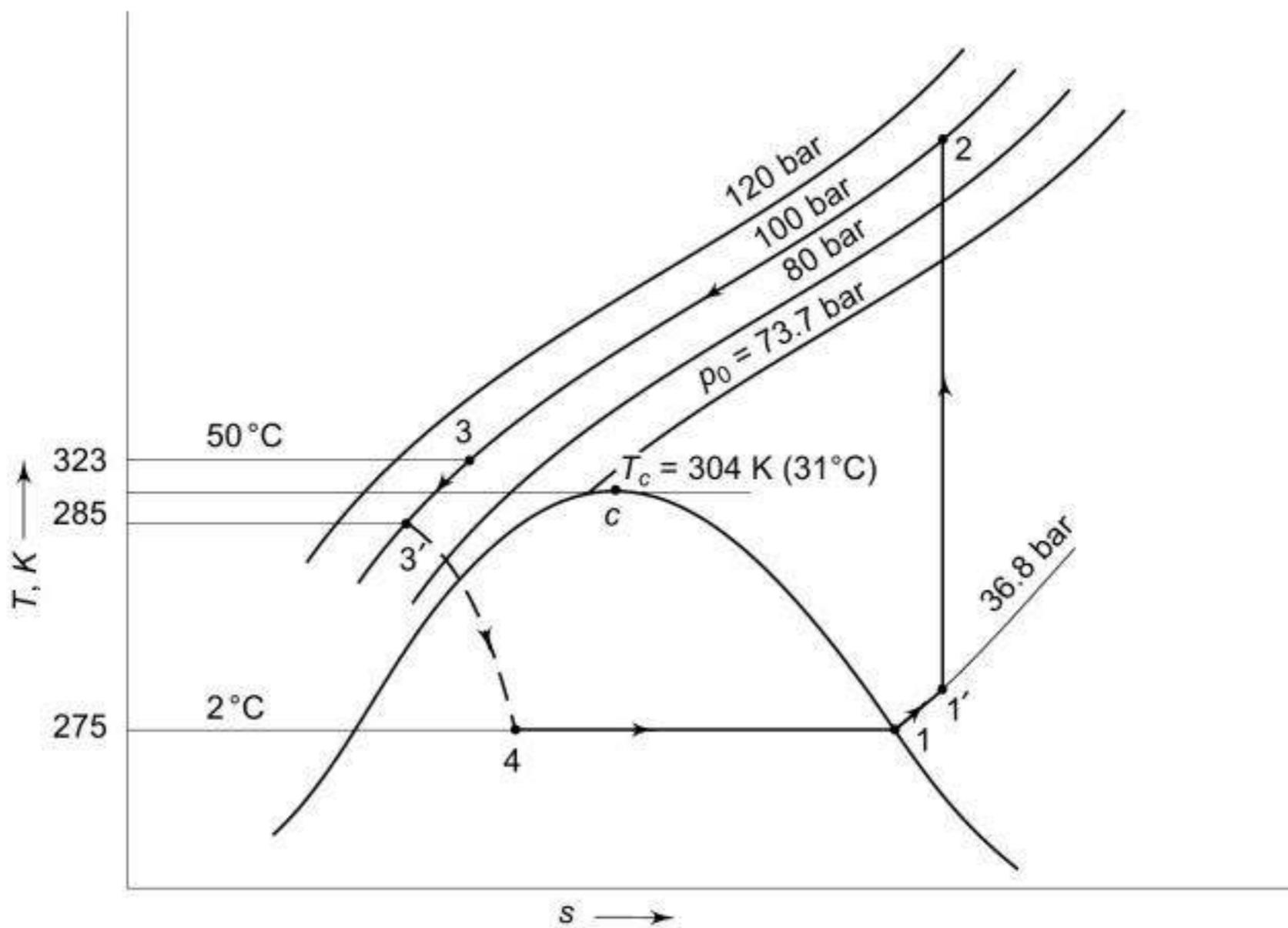


Fig. 4.5 Supercritical vapour compression cycle with CO_2 on T - s diagram

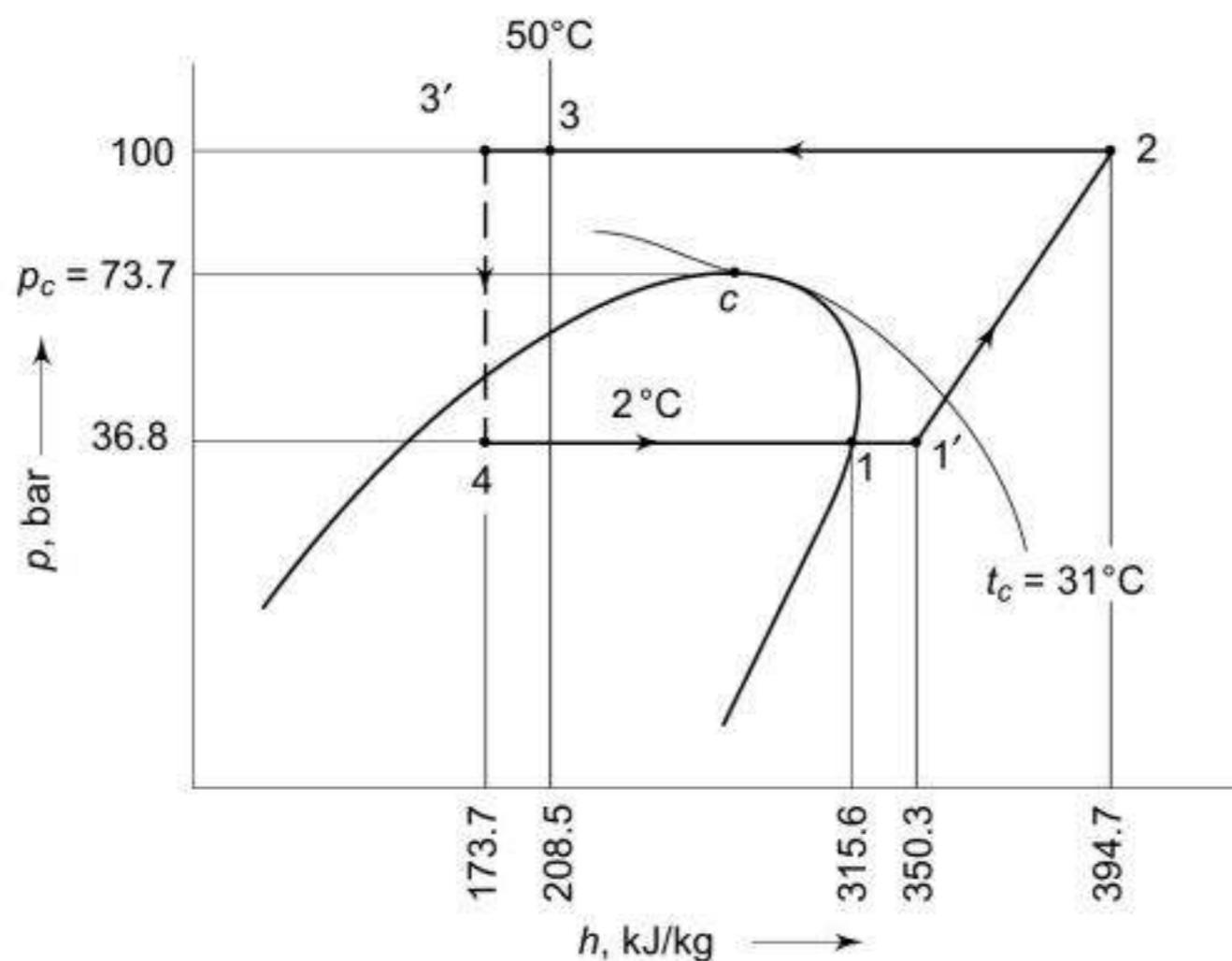


Fig. 4.6 Supercritical vapour compression cycle with CO_2 on p - h diagram

Assuming 20°C superheating of vapour in regenerative heat exchanger, the property values and calculations are as follows:

$$h_1 = 315.6 \text{ kJ/kg}, \quad s_1 = 1.157 \text{ kJ/kg K} \quad h'_1 = 350.3 \text{ kJ/kg}$$

$$h_2 = 394.7 \text{ kJ/kg}, \quad v'_1 = 0.014917 \text{ m}^3/\text{kg}$$

$$h_3 = 208.5 \text{ kJ/kg}, \quad h'_3 = 173.7 \text{ kJ/kg}$$

$$\text{COP}_c = \frac{h_1 - h'_3}{h_2 - h_1} = \frac{142.2}{79.1} = 1.82$$

$$W^* = \frac{35164}{142.2} (79.1) = 1.95 \text{ kW/TR}$$

$$V^* = \frac{35164}{142.2} (0.014917) = 3.7 \times 10^{-4} \text{ m}^3/\text{s (TR)}$$

Results for discharge pressures of 80, 100 and 120 bar, and varying degrees of superheats are given in Table 4.18

Table 4.18 Performance parameters of CO_2 at $t_0 = 2^\circ\text{C}$

<i>Discharge pressure bar</i>	<i>Superheat $^\circ\text{C}$</i>	<i>COP_c</i>	<i>W*</i> kW/TR	<i>V* × 10⁴</i> $\text{m}^3/\text{s (TR)}$
80	10	1.17	3.0	7.0
	20	1.28	2.74	6.8
100	10	1.78	2.01	3.5
	20	1.82	1.95	3.7
	30	1.88	1.87	
120	10	1.51	2.33	3.4
	20	1.55	2.28	3.6

It is observed that high-side pressure of 100 bar is optimum, and superheating of vapour by 30°C gives higher COP. This corresponds to a maximum possible suction temperature of 32°C.

4.17 USING MIXED REFRIGERANTS

The diversification of the refrigeration and air-conditioning industry has led to a continuous search for newer working substances adapted to specific applications. The commonly used fluorocarbons are limited in number and there are gaps in their characteristics owing to their chemistry which need to be filled. The introduction of mixtures in compression refrigerating machines is a step in that direction.

Mcharness and Chapman²⁹, and Downing²⁰ had made experimental studies on a number of refrigerants and their mixtures. Varying the capacity of a refrigerating machine using varying proportions of different refrigerants has now become a well-known possibility.

The most important property of a mixture is the range of temperature at which it boils or condenses. This enables it to achieve non-isothermal refrigeration, and when used in applications involving cooling at a range of temperatures, it results in a higher COP and consequent power saving.^{3,4}

Further, some mixtures form azeotropes which have interesting possibilities.

4.18 BINARY MIXTURES

Two properties are required to define the thermodynamic state of a pure substance. If, however, the state is saturated, only one property needs to be known.

A binary mixture consists of a *higher boiling* and a *lower boiling* component. In the case of a binary system, three independent properties are required to define the state, and two properties when the state is in an equilibrium saturated liquid state or saturated vapour state. One of these properties is invariably the composition.

4.18.1 Measures of Composition in Mixtures

Consider a liquid and/or vapour mixture of two substances. The number of moles of components in the mixture are n_1 and n_2 . Then, the sum

$$n = n_1 + n_2$$

holds for each phase separately. Let x_1 and x_2 denote the mole fractions of the two components in the liquid phase. Then,

$$x_1 = \frac{n_1}{n}, \quad x_2 = \frac{n_2}{n}, \quad x_1 + x_2 = 1$$

Similarly, for the mole fractions y_1 and y_2 in the vapour phase, we have

$$y_1 + y_2 = 1$$

The relations between masses m_1 and m_2 , moles n_1 and n_2 and molecular masses M_1 and M_2 are

$$\begin{aligned} m_1 &= n_1 M_1, \quad m_2 = n_2 M_2 \\ \Rightarrow \quad m &= m_1 + m_2 = n_1 M_1 + n_2 M_2 = nM \end{aligned}$$

where M represents the effective molecular mass of the mixture. Thus,

$$\begin{aligned} M &= \frac{m}{n} = \frac{1}{n} (n_1 M_1 + n_2 M_2) \\ &= x_1 M_1 + x_2 M_2 \text{ for liquid mixture} \\ &= y_1 M_1 + y_2 M_2 \text{ for vapour mixture} \end{aligned}$$

If ξ_1^L and ξ_2^L represent mass fractions in liquid phase, then

$$\xi_1^L = \frac{M_1 x_1}{M_1 x_1 + M_2 x_2} = 1 - \xi_2^L, \quad \xi_2^L = \frac{M_2 x_2}{M_1 x_1 + M_2 x_2} = 1 - \xi_1^L \quad (4.27)$$

Similarly, for mass fractions ξ_1^V and ξ_2^V in vapour phase, we have

$$\xi_1^V = \frac{M_1 y_1}{M_1 y_1 + M_2 y_2} = 1 - \xi_2^V, \quad \xi_2^V = \frac{M_2 y_2}{M_1 y_1 + M_2 y_2} = 1 - \xi_1^V \quad (4.28)$$

The relation between mass fraction and mole fraction is expressed as follows:

$$x_1 \text{ or } y_1 = \frac{\xi_1 M_2}{\xi_1 M_2 + \xi_2 M_1} \quad (4.29)$$

4.18.2 Temperature-Composition Diagram

The nature of this diagram is shown in Fig. 4.7 for any given pressure p . The saturation temperatures of the two components at pressure p are t_1^{sat} and t_2^{sat} , the superscript sat denoting the saturation state of the pure substance. The temperature t_B , at which a liquid mixture begins to boil is called the *bubble point temperature*. A curve can be drawn passing through all such points for various liquid compositions $\xi = \xi^L$, as shown in Fig. 4.7. This is called the *bubble point curve*. It is the locus of saturated liquid states for the mixture.

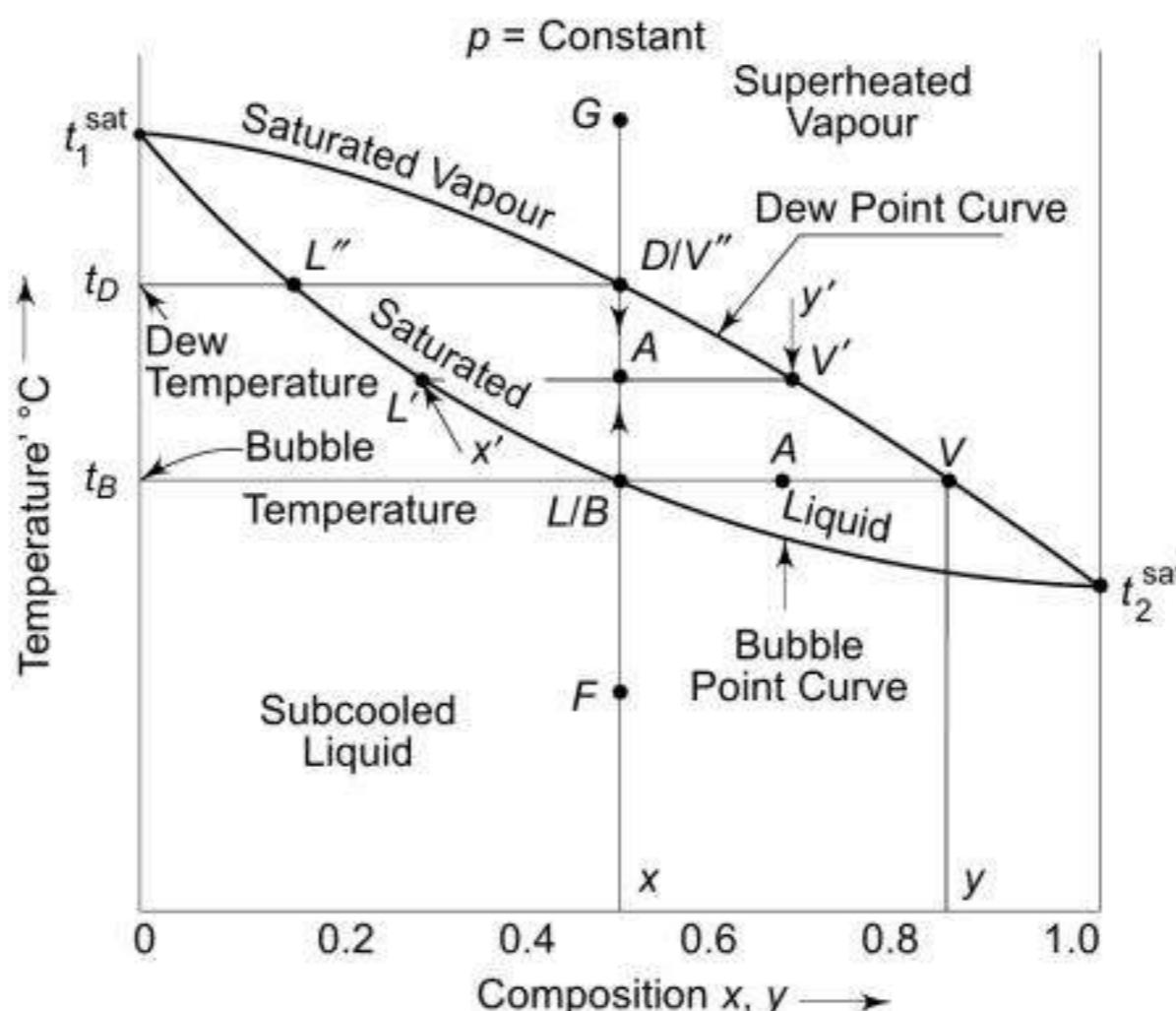


Fig. 4.7 t - x - y diagram of binary mixtures

Similarly, the temperature t_D at which a vapour mixture begins to condense is called the *dew point temperature*. A curve passing through all such points is called the *dew point curve*. It is the locus of saturated vapour states for the mixture. The region below the bubble point curve is the subcooled region, and that above the dew point curve is the superheated region. The region between the two curves is the liquid plus vapour region. The states on these curves are the saturation states. Any state A that falls in the liquid plus vapour region comprises of equilibrium states L of saturated liquid, and V of saturated vapour. The compositions of the two are ξ^L and ξ^V and the overall composition ξ at A is given in terms of the fraction z of vapour and $(1 - z)$ of liquid as given in Eq. (4.30)

$$\xi = z\xi^V + (1 - z)\xi^L \quad (4.30)$$

It may be noted that there will be a different loop like the one in Fig. 4.7 for each pressure.

Thus, when a subcooled liquid of composition x at F in Fig. 4.7 is heated it will first rise in temperature till it reaches the *bubble temperature* t_B at B/L when it begins to boil. The first vapour bubble that is formed is at V having the composition y . As the boiling proceeds, the temperature rises. Subsequent states during boiling are along the vertical such as point A at the same overall composition with liquid at L' and vapour at V' in equilibrium. When boiling is complete, the vapour is at D/V'' at *dew temperature* t_D . The last drop of liquid to vaporize is at L'' .

Further heating will result in superheating of vapour to G . When superheated vapour at G is cooled, the reverse processes of desuperheating to D , condensation to B , and subcooling to F will take place.

It is observed that the *temperature glide/range* during evaporation and condensation is equal to the difference in dew and bubble temperatures ($t_D - t_B$) for the composition.

The temperature-composition diagram represents vapour-liquid equilibrium of mixtures at constant pressure. A counterpart to this diagram is the pressure composition $p-x-y$ or $p - \xi$ diagram at constant temperature as shown in Fig. 4.8.

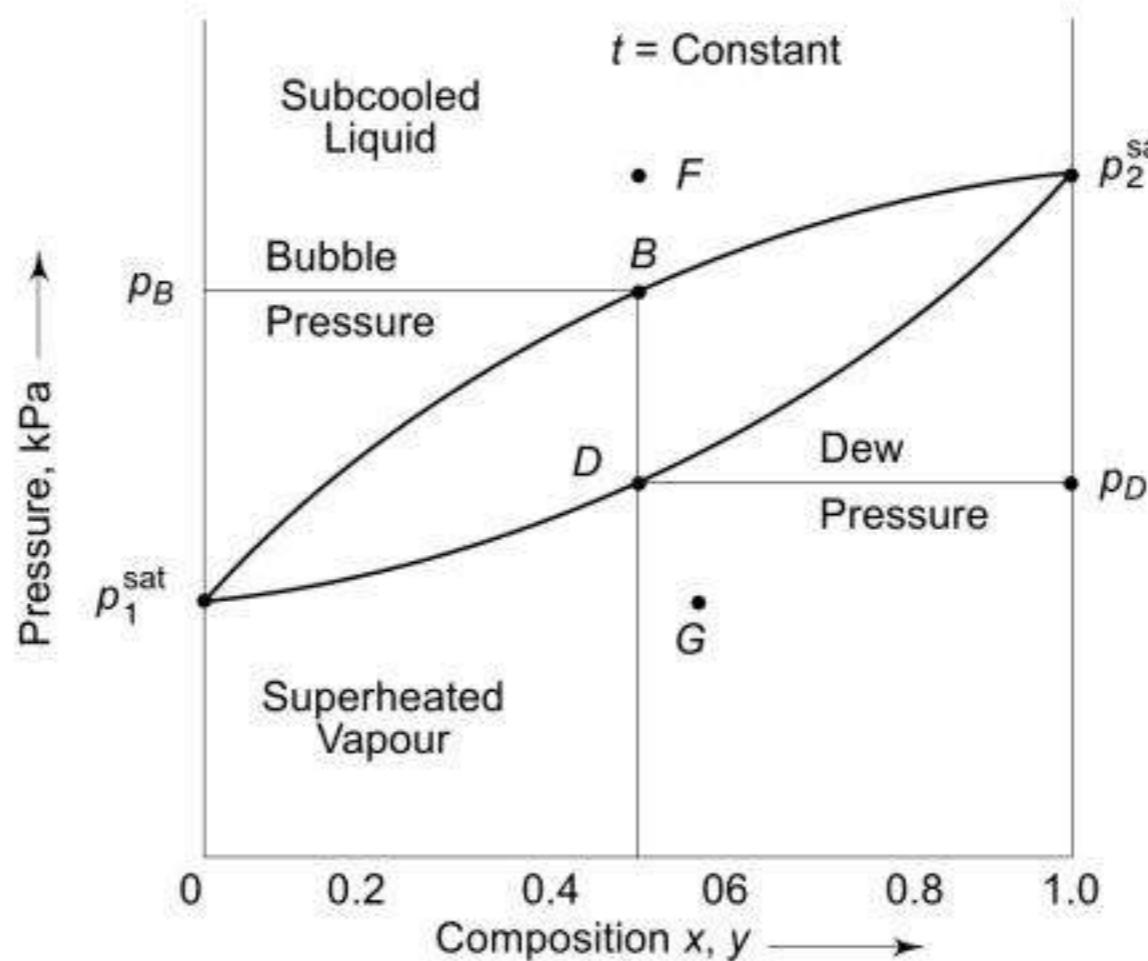


Fig. 4.8 $p-x-y$ Diagram of Binary Mixtures



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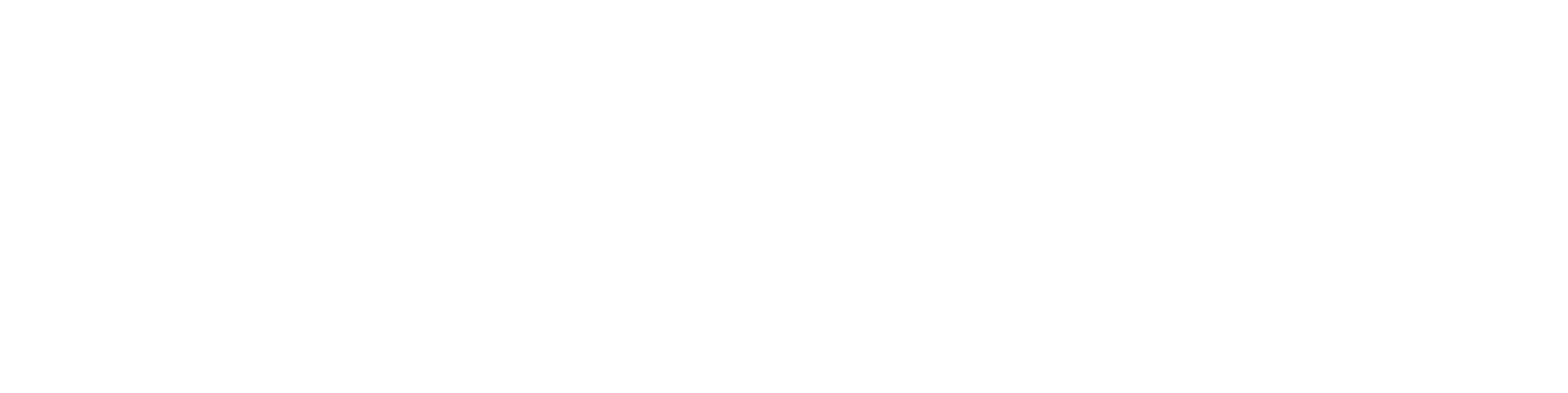
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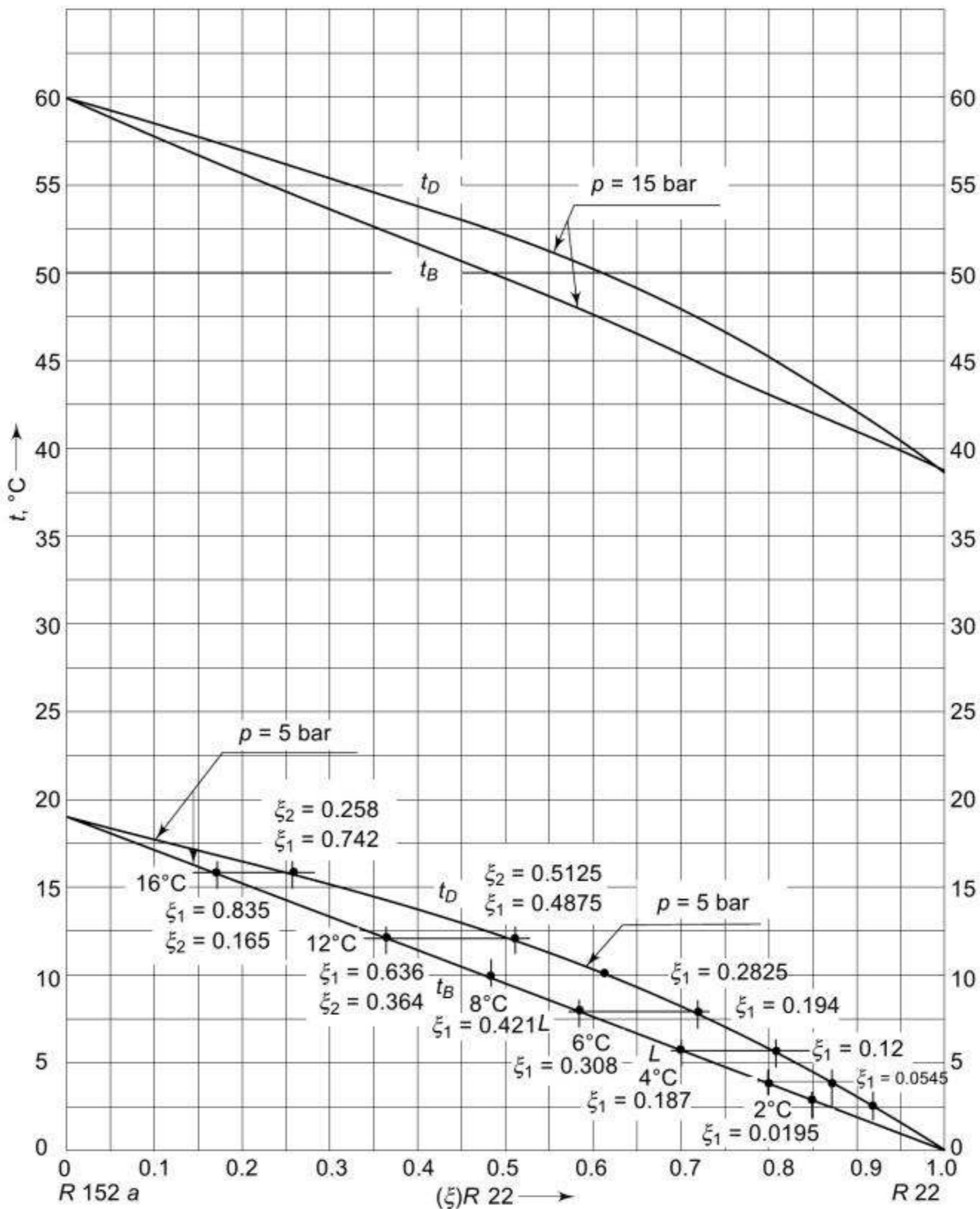


Fig. 4.13 Temperature-composition diagram of R 152/R 22 mixtures at 15 and 5 bar pressures

Table 4.19 Vapour-liquid equilibrium of (1) R 152a and (2) R 22 mixtures at 15 bar pressure

t °C	p_1^{sat} bar	p_2^{sat} bar	x_1	y_1	ξ_1^L	ξ_1^V
39.1		15	0.0	0.0	0.0	0.0
43	9.88	16.42	0.23	0.15	0.18	0.12
47	10.96	18.05	0.43	0.32	0.36	0.26
51	12.125	19.8	0.62	0.5	0.55	0.43
55	13.38	21.67	0.81	0.72	0.76	0.66
57	14.05	22.67	0.89	0.83	0.86	0.79
60	15		1.0	1.0	1.0	1.0

Table 4.20 Vapour-liquid equilibrium of (1) R 152a and (2) R 22 mixtures at 5 bar Pressure

t °C	p_1^{sat} bar	p_2^{sat} bar	x_1	y_1	ξ_1^L	ξ_1^V
0.2		5	0.0	0.0	0.0	0.0
2	2.835	5.308	0.117	0.07	0.0915	0.0545
4	3.04	5.657	0.268	0.163	0.1869	0.12
6	3.257	6.023	0.368	0.24	0.308	0.194
8	3.486	6.406	0.487	0.34	0.421	0.2825
12	3.98	7.226	0.696	0.555	0.636	0.4875
16	4.526	8.123	0.87	0.79	0.835	0.742
19.2	5		1.0	1.0	1.0	1.0

Enthalpy-composition Diagrams From the tables of properties we have the specific heats and saturated liquid and vapour enthalpies of pure refrigerants as follows:

p bar	t_1^{sat} °C	t_2^{sat} °C	h_{f_1} kJ/kg	h_{f_2} kJ/kg	h_{g_1} kJ/kg	h_{g_2} kJ/kg	C_{p_1} kJ/kg.K	C_{p_2} kJ/kg.K
15	60	39.1	311.9	247.9	540.5	415.8	1.61	0.91
5	19.2	0.2	233.0	200.0	520.0	406.0	0.864	0.89

Sample Calculations for Liquid and Vapour Enthalpies At 15 Bar
At $t = 43^\circ\text{C}$, $h^L = \xi_1^L h_1^L + \xi_2^L h_2^L = 0.18 (277.7) + 0.92 (253) = 258 \text{ kJ/kg}$

Note The values of h_1^L and h_2^L have been taken equal to saturation enthalpies h_{f1} and h_{f2} of pure refrigerants.

$$\begin{aligned} h^V &= \xi_1^V [h_{g_1} - C_{p_1} (t_1^{\text{sat}} - t)] + \xi_2^V [h_{g_2} + C_{p_2} (t - t_2^{\text{sat}})] \\ &= 0.12 [540.5 - 1.61 (60 - 43)] + 0.88 [415.8 + 0.91 (43 - 39.1)] \\ &= 431.9 \text{ kJ/kg} \end{aligned}$$

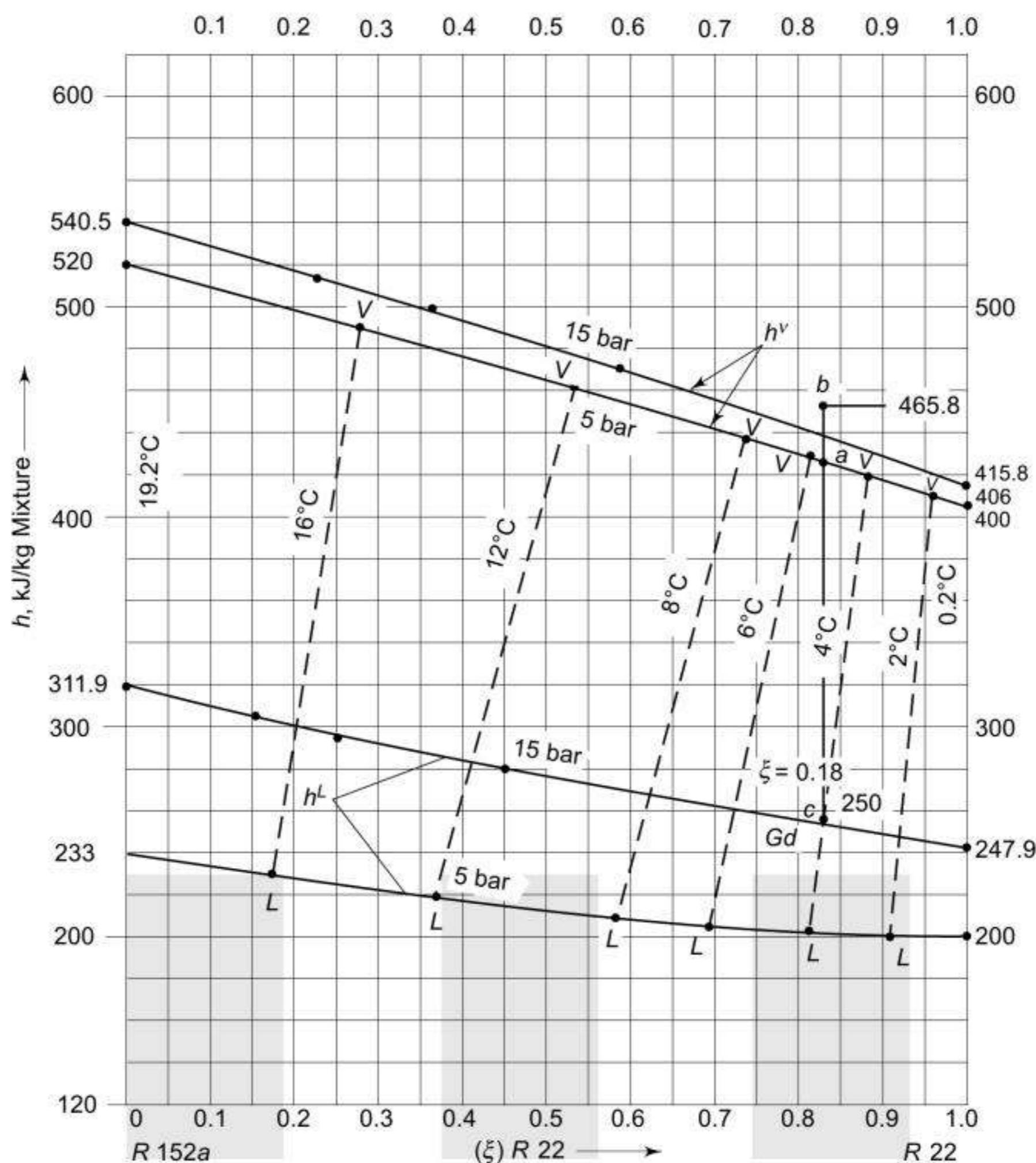
The results are presented in Tables 4.21 and 4.22 for 15 bar and 5 bar pressures, and in Fig. 4.14.

Table 4.21 Saturated liquid and vapour enthalpies of (1) R 152a and (2) R 22 mixtures at 15 bar

t °C	ξ_1^L	ξ_1^V	$h_1^L = h_{f_1}$ kJ/kg	$h_2^L = h_{f_2}$ kJ/kg	h^L kJ/kg	h^V kJ/kg
39.1	0.0	0.0		247.9	247.9	$415.8 = h_{g_2}$
43	0.18	0.12	277.7	253.0	258.0	431.9
47	0.36	0.26	285.5	258.3	268.3	450.4
51	0.55	0.43	293.5	263.6	280.2	472.0
55	0.76	0.66	301.7	269.1	293.1	500.0
57	0.86	0.79	305.7	271.8	300.5	515.5
60	1.0	1.0	311.9		311.9	$540.5 = h_{g_1}$

Table 4.22 Saturated liquid and vapour enthalpies of (1) R 152a and (2) R 22 mixtures at 5 bar

t °C	ξ_1^L	ξ_1^V	$h_1^L = h_{f_1}$ kJ/kg	$h_2^L = h_{f_2}$ kJ/kg	h^L kJ/kg	h^V kJ/kg
0.2	0.0	0.0		200.0	200.0	406.0 = h_{g2}
2	0.0915	0.0545	203.4	202.3	202.4	412.9
4	0.187	0.12	206.7	204.7	205.0	422.2
6	0.308	0.194	210.1	207.1	208.0	430.1
8	0.421	0.2825	213.5	209.4	211.0	440.3
12	0.636	0.4875	220.4	214.2	218.0	464.0
16	0.835	0.742	227.4	219.0	226.0	492.2
19.2	1.0	1.0	233.0			520.0 = h_{g1}

**Fig. 4.14** Enthalpy-composition diagrams of R 152a/R 22 mixtures at 15 and 5 bar pressures

4.20 EVALUATION OF THERMODYNAMIC PROPERTIES OF R 290/R 600a MIXTURES

A more rigorous method of evaluating thermodynamic properties of mixtures has been described in the following pages. The assumptions that have been made in developing the thermodynamic properties of mixtures of R 290 and R 600a are as follows:

- (i) There is no heat of mixing in liquid phase. So it obeys Raoult's law.
- (ii) There is no change in volume due to mixing in the vapour phase. Ideal mixture behaviour has, therefore, been assumed and Dalton's law of partial pressures has been applied. However, real gas behaviour has been taken into account by employing the actual equations of state for the individual gases and the vapour mixture.

Note Entropy of mixing has to be considered for ideal mixtures also, since mixing, in any case, is an irreversible process.

4.20.1 Calculations of Specific Volume, Enthalpy and Entropy of Mixture in Liquid Phase⁸

The specific volume of liquid mixture v^L is calculated from Eq. (4.39)

$$Mv^L = \bar{v}^L = x_1 \bar{v}_1^L + x_2 \bar{v}_2^L \quad (4.39)$$

where x_1 and x_2 are the mole fractions and \bar{v}_1^L and \bar{v}_2^L are the molar volumes of its species.

The specific enthalpy of liquid mixture h^L is similarly calculated using the relations as given in Eq. (4.40)

$$Mh^L = \bar{h}^L = x_1 \bar{h}_1^L + x_2 \bar{h}_2^L \quad (4.40)$$

where \bar{h}_1^L and \bar{h}_2^L are the molar enthalpies of its species. The value of heat of mixing h_m is assumed as zero here.

The specific entropy of liquid mixture s_m^L is calculated from the relation given in Eq. (4.41)

$$Ms^L = \bar{s}^L = x_1 \bar{s}_1^L + x_2 \bar{s}_2^L - \bar{R} (x_1 \ln x_1 + x_2 \ln x_2) \quad (4.41)$$

where \bar{s}_1^L and \bar{s}_2^L are liquid entropies of pure species and the second term is the entropy of mixing.

4.20.2 Calculation of Specific Volume of Mixture in Vapour Phase

For the purpose, we need an equation of state for the components and the mixture. The constants for the mixture can be evaluated from the constants of the components by simple additive rules. So, Peng-Robinson³⁷ Equation (1.14) has been chosen in the following method. The pseudocritical temperature, pressure, and acentric factor for the mixture for use in the equation are given by:

$$\begin{aligned} T_c &= y_1 T_{c_1} + y_2 T_{c_2} \\ p_c &= y_1 p_{c_1} + y_2 p_{c_2} \\ \omega &= y_1 \omega_1 + y_2 \omega_2 \end{aligned}$$

The constants for R 290 and R 600a are given in Table 4.23.

Table 4.23 Critical constants, molecular masses and acentric factors for R 290 and R 600a

Refrigerant	M	p_c MPa	T_c K	ω	$f(\omega)$
R 290	44	4.236	369.8	1.1442	0.60429
R 600a	58	3.685	409.1	1.1424	0.64783

4.20.3 Calculation of Enthalpy of Mixture in Vapour Phase

A method developed by Agarwal and Arora² will now be described.

Figure 4.15 shows the vapour-liquid domes of pure components 1 and 2, and their binary mixture of certain composition on a p - h diagram.

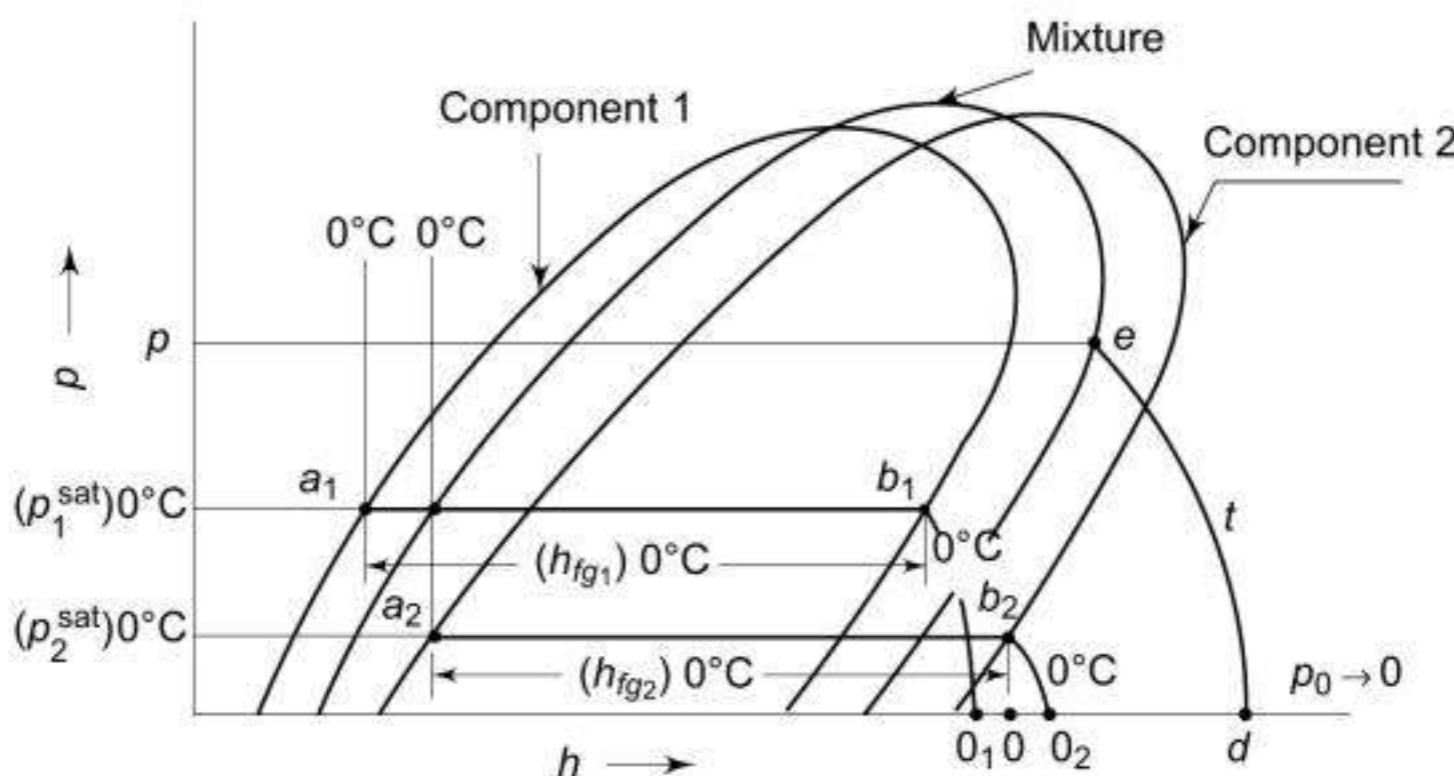


Fig. 4.15 Proposed method for vapour mixture enthalpy calculation

The figure illustrates how the enthalpies of saturated liquid and saturated vapour may be calculated. The proposed method assumes values for reference state enthalpies for saturated liquid states of both the pure components 1 and 2 at a_1 and a_2 at 0°C .

Consider now the saturated vapour state e on the dome for the mixture as shown in Fig. 4.15. Let the pressure and temperature of the mixture at e be p and T respectively.

Now, the state at the same temperature T but at zero pressure is shown by point d in the figure. Then, the enthalpy at e is related to ideal gas state enthalpy at d at temperature T by the residual enthalpy term. Thus

$$h_e = h_d + |h - h^{id}|_{p_0 \rightarrow 0}^p$$

The enthalpy h_d , is related to the ideal gas state enthalpy h_0 at point 0 at $t_0 = 0^\circ\text{C}$ ($T_0 = 273.15$ K) and $p_0 \rightarrow 0$ by the relation

$$h_d - h_0 = \int_{T_0}^T C_{p_0} dT$$

where C_{p_0} is the zero-pressure constant pressure specific heat of the mixture. Now, h_0 is found from

$$h_0 = \xi_1 h_{0_1} + \xi_2 h_{0_2}$$

where h_{0_1} and h_{0_2} are ideal gas state enthalpies of components at point o_1 and o_2 . These, in turn are related to real gas saturated vapour state enthalpies h_{b_1} and h_{b_2} by residual enthalpy terms, while h_{b_1} and h_{b_2} are given by

$$h_{b_1} = h_{a_1} + (h_{fg_1})_{0^\circ\text{C}}, \quad h_{b_2} = h_{a_2} + (h_{fg_2})_{0^\circ\text{C}}$$

Reference state enthalpies h_{a_1} and h_{a_2} have assigned values, say, equal to 200 kJ/kg. Note that the pressures at b_1 and b_2 are $(p_1^{\text{sat}})_{0^\circ\text{C}}$ and $(p_2^{\text{sat}})_{0^\circ\text{C}}$ respectively.

Now, to find h_e from h_d for the mixture, and h_{o_1} and h_{o_2} from h_{b_1} and h_{b_2} for the components, we have to evaluate residual enthalpy terms:

$$\left| h - h^{id} \right|_0^p \text{ for the mixture}$$

$$\left| h - h^{id} \right|_0^{p_1^{\text{sat}}} \text{ and } \left| h - h^{id} \right|_0^{p_2^{\text{sat}}} \text{ for the components}$$

Since Peng-Robinson equation is a p -explicit equation, we find $|h - h^{id}|$ from $|u - u^{id}|$ using the procedure described in Sec. 1.15.2. For the purpose, we find the differential $(\partial p / \partial T)_v$ from Eq. (1.14), as given below in Eq. (4.42)

$$\left(\frac{\partial p}{\partial T} \right) = \frac{R}{v - b} + \frac{k f(\omega) \left[1 + f(\omega) \left(1 - T_r^{\frac{1}{2}} \right) \right]}{\left[(v + b)^2 - 2b^2 \right] \sqrt{T_r T_c}} \quad (4.42)$$

in which

$$k = \frac{0.45724 R^2 T_c^2}{p_c}$$

Combining the relations as above, we find the expression for vapour phase enthalpy as written below in Eq. 4.43

$$h = h_o + pv - RT_o + \int_{T_o}^T (C_{p_o} - R) dT - \frac{d}{2\sqrt{2b}} \ln \left[\frac{v + b - \sqrt{2b}}{v + b + \sqrt{2b}} \right] \quad (4.43)$$

in which

$$d = a + kf(\omega) \sqrt{T_r} \left[1 + f(\omega) \left(1 - \sqrt{T_r} \right) \right]$$

C_{p_o} for mixture is obtained from Gibbs law

$$\bar{C}_{p_o} = y_1 \bar{C}_{p_{o1}} + y_2 \bar{C}_{p_{o2}}$$

For 50% R 290 and 50% R 600a mixture with $h_{a_1} = h_{a_2} = 200$ kJ/kg as the reference state, it is found that

$$h_o = 568 \text{ kJ/kg}$$

at $t_0 = 0^\circ\text{C}$ and $p_o = 0$ for the mixture.

4.20.4 Calculation of Entropy of Mixture in Vapour Phase

Vapour phase entropy of the mixture is calculated using the relation similarly derived by Smith and Van Ness³⁷ as given below:

$$s = s_o - R (y_1 \ln y_1 + y_2 \ln y_2) + R \ln \left[\frac{RT(v - b)}{p_o} \right]$$

$$+ \int_{T_o}^T (C_{p_o} - R) \frac{dT}{T} + X \left[\frac{1}{2\sqrt{2b}} \log \left(\frac{v + b - \sqrt{2b}}{v + b + \sqrt{2b}} \right) \right] \quad (4.44)$$

$$X = \frac{k f(\omega) \left[1 + f(\omega) \left(1 - \frac{T}{T_c} \right)^{\frac{1}{2}} \right]}{\sqrt{T T_c}}$$

where

The second term on right hand side in Eq. (4.44) represents entropy of mixing, and

$$s_o = \xi_1^V s_{o_1} + \xi_2^V s_{o_2}$$

For 50% R 290 and 50% R 600a mixture with $s_{a_1} = s_{a_2} = 1 \text{ kJ/kg.K}$ as the reference state entropies, it is found that

$$s_o = 2.6593 \text{ kJ/kg.K}$$

at $t_o = 0^\circ\text{C}$ and $p_o = 1.01 \text{ bar}$. Note that ideal gas state is taken, not at $p_o = 0$ but at $p_o = 1.01 \text{ bar}$ for calculations of residual entropy in order to avoid the mathematical anomaly of $s \rightarrow \infty$ as $p \rightarrow 0$ as explained in Chap. 1.

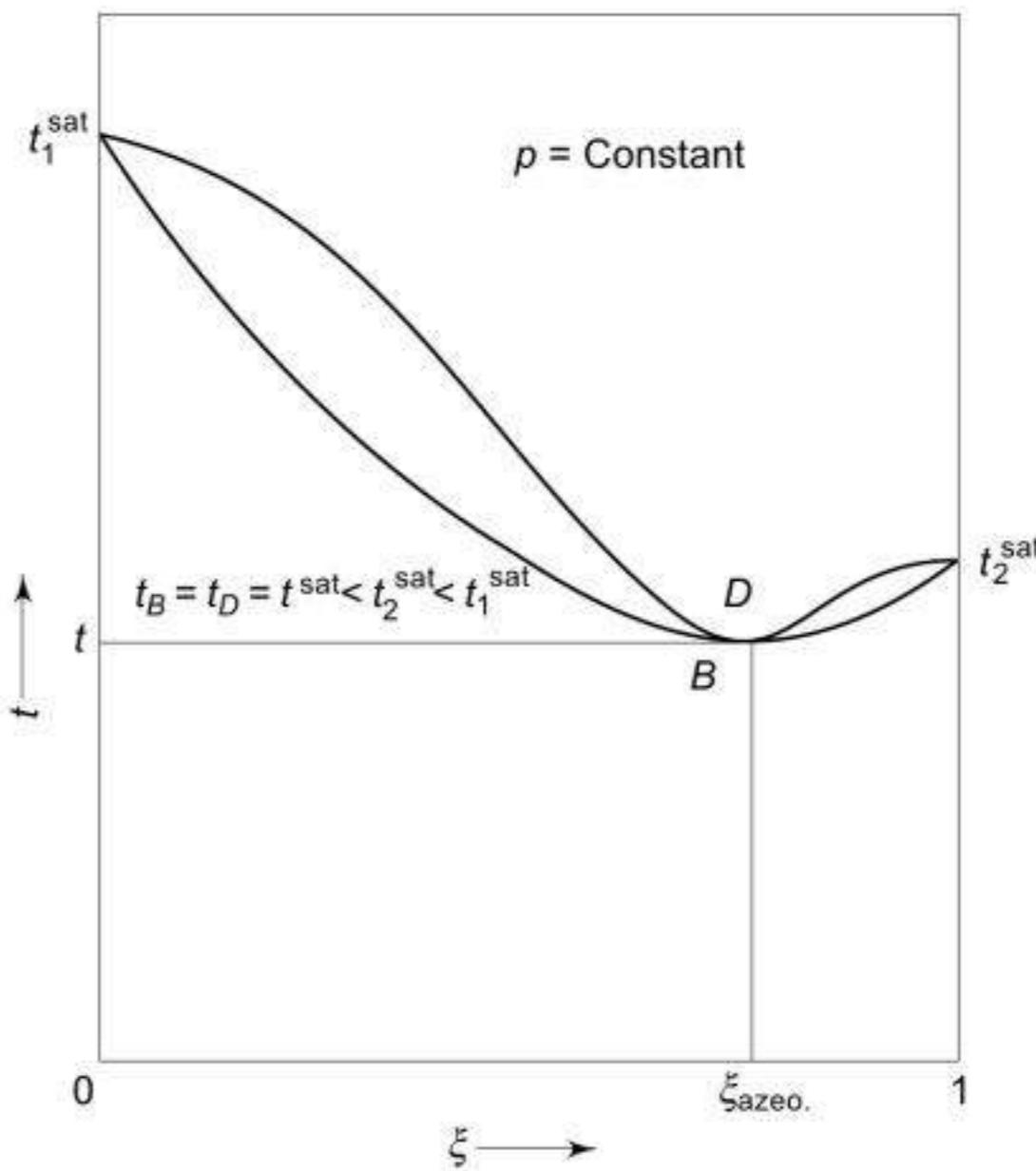
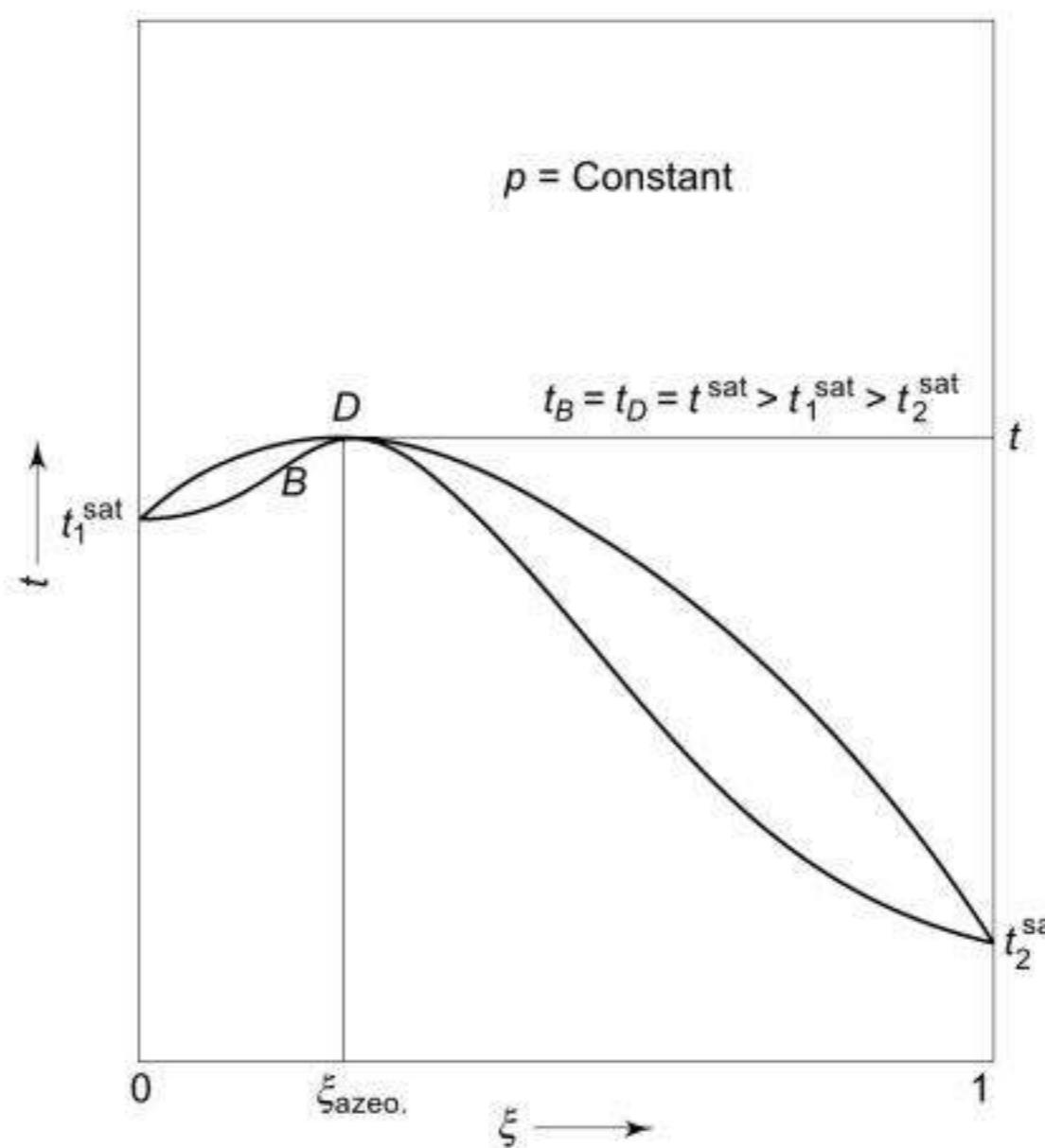
Note Tables in the Appendix give the properties of the mixture as calculated by the procedure described above.

4.21 AZEOTROPIC MIXTURES

As stated earlier, azeotropes are essentially a class of non-ideal mixtures having bubble point temperature equal to dew point temperature. Hence, they boil and condense at constant temperature like pure substances. Azeotropes are generally formed when the difference in the boiling points of the two components is not very large and when the deviations from ideal behaviour are large enough. Hence, there are azeotropes with positive deviation from Raoult's law as well as those with negative deviation from it as described by Figs. 4.16 and 4.17. An example of an azeotrope which has positive deviation from Raoult's law is R 22/R 12 azeotrope. Such an azeotrope has a bubble or dew point which is lower than the boiling point of either of the components as shown in Fig. 4.16 and is, therefore, called a *minimum-boiling azeotrope*. On the other hand, an azeotrope with negative deviation from Raoult's law has a bubble point which is higher than the boiling point of either of the components as shown in Fig. 4.17, and is called a *maximum boiling azeotrope*.

It is to be noted from Figs. 4.16 and 4.17 that there is an *azeotropic composition* ξ_{azeo} for the mixture at the given pressure and temperature. This composition changes with the variation in pressure and temperature. However, near the azeotropic composition the bubble and dew point curves become flat. Thus in the range of usual condensation and evaporation temperatures in refrigerating machines, the azeotropic concentration more or less remains the same. Therefore, a system charged with an azeotrope may be considered as working at all sections without any change in composition.

It is also to be noted that a maximum boiling azeotrope, on account of the negative heat of mixing, will have a higher latent heat of vaporization compared to the molal average latent heat. On the contrary, a minimum boiling azeotrope will have a lower latent heat.

Fig. 4.16 t - ξ diagram of a minimum boiling azeotropeFig. 4.17 t - ξ diagram of a maximum boiling azeotrope



4.22 USE OF MINIMUM AND MAXIMUM BOILING AZEOTROPES

Eiseman²¹ has done a study of a large number of azeotropes. Most fluorocarbon mixtures have positive deviations from Raoult's law. Thus, they form minimum boiling azeotropes.

The main property of a minimum boiling azeotrope is its boiling point which is lower than the N.B.P. of even the lower boiling component. It enables the system with an azeotrope to have a higher evaporator pressure as compared to the lower boiling component. This, in general, results in an increased density of the suction vapour and, therefore, higher capacity.

Further, due to the depression of the boiling point of the azeotrope, slope ' b ' of the vapour pressure line of the azeotrope is flatter than the slope of the line of the lower boiling component. Thus, below a certain temperature, say point A , in Fig. 4.18, the vapour pressure of the azeotrope is higher, and above this temperature, it is lower than the vapour pressure of the lower boiling component. Accordingly, for the azeotrope, the evaporator pressure p_0 at T_0 is higher, and the condenser pressure p_k at T_k is lower than the values for the lower boiling component. Thus the compression ratio and discharge temperature are considerably reduced.

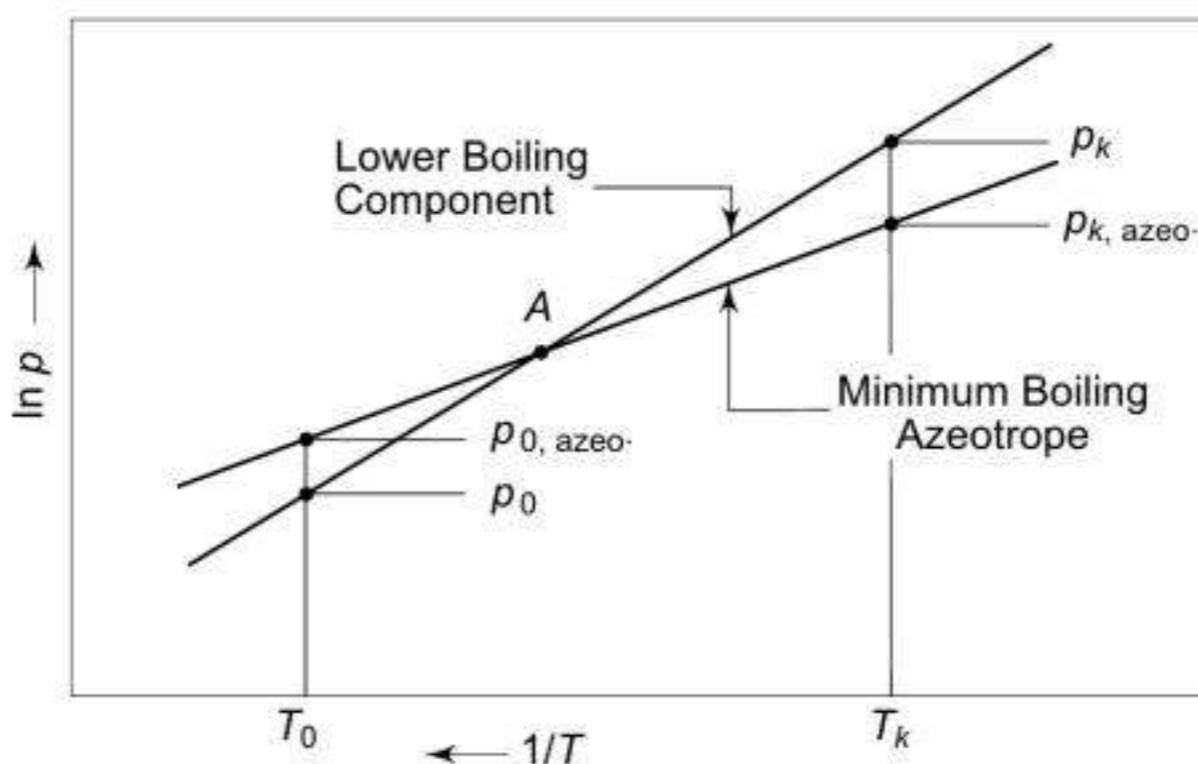


Fig. 4.18 Comparison of $\ln p-1/T$ diagrams of a minimum boiling azeotrope and lower boiling component

It is not intended here to describe all the known azeotropes. However, Table 4.24 gives examples of such common minimum boiling azeotropes.

Refrigerant 500 R 500 was discovered in 1950 as a special refrigerant of the Carrier Corporation under the trade name of Carrene 7. This azeotrope consists of 73.8 per cent R 12 and 26.2 per cent R 152, difluoroethane.

The N.B.P. of R 500 is about 3.5°C lower than that of R 12. The azeotrope has a refrigerating effect per unit of swept volume about 18 per cent more than that of pure R 12. In the past this azeotrope was used to replace R 12 in 60-cycle units when they were operated on a 50-cycle current. This, essentially, kept the capacity of the unit same even with decreased speed.

Table 4.24 Data of minimum boiling azeotropes

Azeotrope	Refrigerants	Weight %	N.B.P.	N.B.P. of Lower Boiling Component, °C.
R 500	R 12/R 152	73.8/26.2	-33.3	-29.8 (R12)
R 501	R 22/R 12	75/25	<-40.8	-40.8 (R22)
R 502	R 22/R 115	48.8/51.2	-45.6	-40.8 (R22)
R 507A	R 125/R 143a	50/50	-46.74	-48.09(R125)
R 404 A	R 125/R143a/R134a	44/52/4	-46.22 / -45.47	-48.09(R125)
R410 A	R32 / R125	50/50	-51.44 / -51.36	-51.65 (R32)

Refrigerant 501 Spauchus³⁹ has published vapour pressure data of R 22 + R 12 mixtures. Using his values, pressure ratios are determined for $t_k = 50^\circ\text{C}$ and $t_0 = 0^\circ\text{C}$ and are given in Table 4.25.

Table 4.25 Pressure ratios at various compositions of R 22 in R 22 + R 12 mixtures

Mole Fraction, x_{22}	1	0.974	0.95	0.9	0.85	—	0
Pressure ratio, p_k/p_0	3.906	3.897	3.897	3.841	3.85	—	3.947

Table 4.25 shows that for the azeotropic mixture the pressure ratio is lower than that for pure R 22. It exhibits a minimum in the neighbourhood of 85 mole per cent of R 22. Lower pressure ratios—coupled with a lower value of the compression index for the mixture as compared to R 22—yield lower discharge temperatures and hence lower winding temperatures in hermetically-sealed units. A higher density of the suction vapour results in increased cooling and further lowering of the temperature of windings.

Refrigerant 502 The behaviour of R 502 is similar to that of the R 22/R 12 azeotrope. It boils at a temperature of about 4.8°C lower than R 22. Thus, the higher capacity and lower compression ratio of the azeotrope, in relation to R 22, result in significantly lower discharge temperatures and lower winding temperatures. The use of R 502 had, therefore, proven very advantageous in supermarket air-cooled and low temperature frozen-food cabinets, and heat pump applications.

Note All the three R 500, R 501 and R 502 contain CFCs. Hence, these have been phased out.

Refrigerant 507A Döring *et al.*¹⁹ have shown that a blend containing, 50–50% by mass, R 143a with an N.B.P. of -47.24°C and R 125 with an N.B.P. of -48.09°C shows azeotropic behaviour, making it an attractive alternative to replace R 502, and even R 22, in certain cases, since both R 125 and R 143a are HFCs (no chlorine atom/s in the molecules). In a test conducted with an open-type compressor, it was found that R 507 has compressor discharge temperature approximately 8°C below that of even R 502. It gives approximately 5-6% higher capacity than R 502.

Refrigerant 404A This *near-azeotrope refrigerant mixture* (NARM) is a *ternary* blend. It is also considered to be a zero-ODP replacement for R 502. Snelson *et al.*³⁸ have conducted experiments on this mixture using an open-type compressor. The lubricating oil used has to be polyol ester oil because of the presence of R 134a in the mixture. The results show that the volumetric refrigerating capacity of R 404A and R 502 are nearly the same. R 404A has about 2-4% higher pressure ratio than R 502. Inspite of this, it has 5-6°C lower compressor discharge temperature which would make it a suitable substitute for R 502 in high compression ratio applications. However, its COP is lower.

Refrigerant 410A R 410A is another near azeotropic mixture of 50/50% R 32 and R125. It is already being used as a substitute for R 22.

Interesting possibilities exist for the use of the maximum boiling azeotropes which are a result of negative deviations from Raoult's law. They have a higher latent heat of vaporization and hence a higher refrigerating effect. Thus, an increased coefficient of performance can be obtained.

Unfortunately, maximum boiling azeotropes are rare in nature. A known example is that of an azeotrope formed by R 22 and dimethyl ether, $(\text{CH}_3)_2\text{O}$, in about 50/50 per cent composition by weight. The azeotrope boils at -20.3°C as compared to -40.8°C for R 22, and -23.7°C for dimethyl ether.

4.23 NON-ISOTHERMAL REFRIGERATION

There are two kinds of duties a refrigerating machine may be required to perform:

- Isothermal refrigeration:* Pumping of heat from a constant low temperature.
- Non-isothermal refrigeration:* Removal of heat at varying temperature.

The former is generally termed as *space refrigeration* and the latter as *process refrigeration*.

In space refrigeration, the cold body is at a constant low temperature, such as in domestic refrigerators, cold storages, etc. The vapour compression cycle used for the purpose suffers from the irreversibility of the heat transfer processes during evaporation and condensation. This is due to the finite temperature difference between the refrigerant, the cold body and the coolant. This finite temperature difference could, however, be made small or even made to approach zero by the use of infinite heat transfer surfaces. This, therefore, does not present a thermodynamic obstacle to remove this source of inefficiency.

In process refrigeration, a body is cooled through a range of temperature. This also has numerous applications, such as in food freezing, beer chilling, most chemical processes and even cooling of air for air conditioning. Here again, the conventional vapour compression system is used. The evaporation temperature t_o of the refrigerant has to be lower than the lowest cold body temperature t_c , and the condensation temperature t_k , similarly, higher than the highest coolant temperature t_h . This is shown in Fig. 4.19. Thus, $t_o < t_{c_1} < t_{c_2}$ and $t_k > t_{h_1} > t_{h_2}$. The system, as for space refrigeration, suffers from the inherent irreversibility of the heat transfer processes during evaporation and condensation, and in the case of process refrigeration, this cannot be made to approach zero even with infinite surfaces. For even if it were

possible to make temperature differentials— $(t_{c_2} - t_o)$ and $(t_k - t_{h_2})$ at one end of the evaporator and condenser respectively—approach zero by the use of infinitely large heat transfer surfaces, there would still be existing finite temperature differentials— $(t_{c_1} - t_o)$ and $(t_k - t_{h_1})$ —at the other ends.

The scope of using non-azeotropic mixtures is described in detail by Tchaikovski and Arora.⁴¹ The most important property of a mixture is its ability to boil through a range of temperature. If a non-azeotropic mixture of two or more refrigerants is used, a definite boiling range can be obtained for the evaporation as well as condensation processes. The actual range will depend upon the proportion of individual components and the temperature and pressure of the mixture. Thus in Fig. 4.19, the evaporation could be made to proceed along path t_o to t'_o and condensation along t_k to t'_k . In this manner, the excessive temperature differences at the other two ends could also be made small. The average refrigeration temperature could thus be increased. And, the average heat rejection temperature could be decreased at the same time. Since the Carnot COP of a refrigerating machine is given by

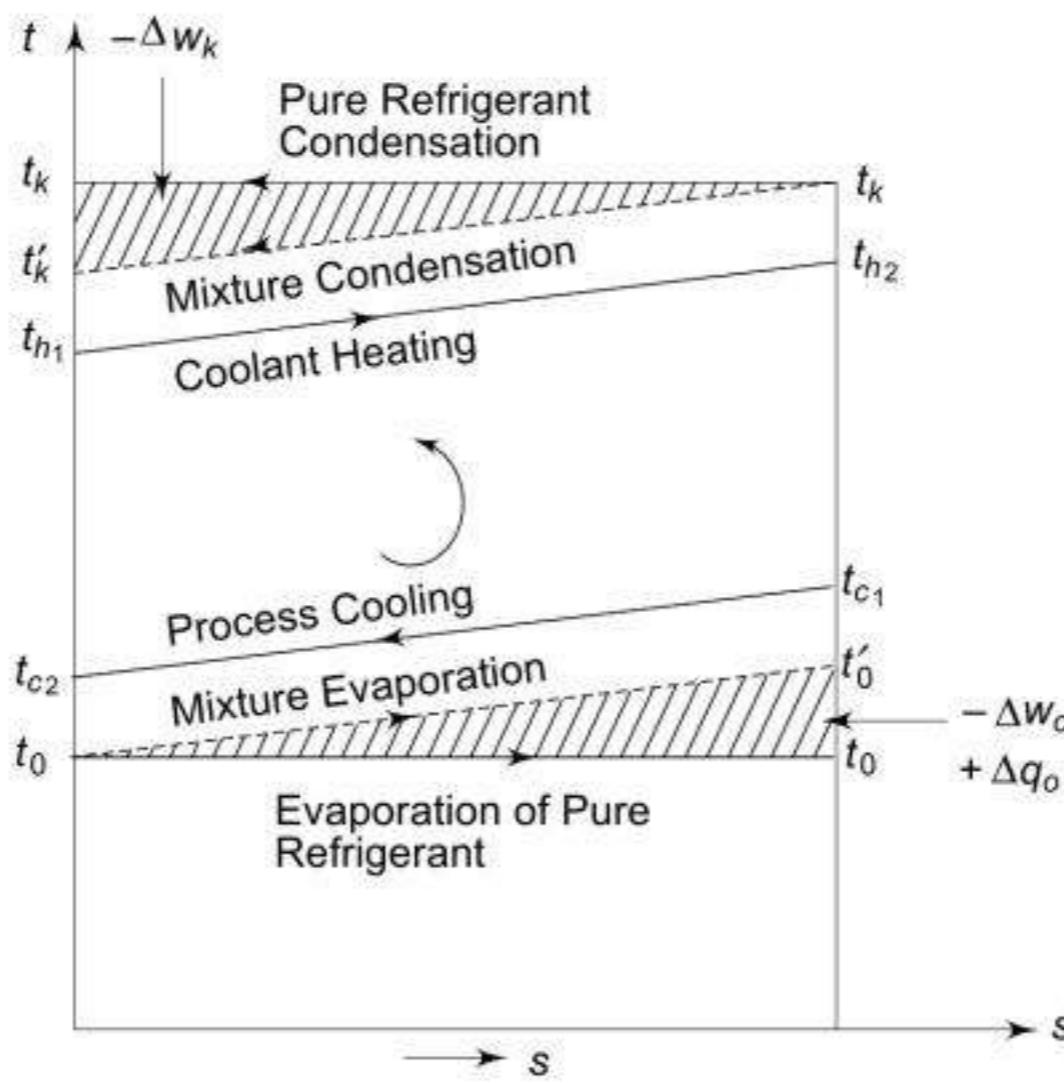


Fig. 4.19 Non-isothermal refrigeration with a mixed refrigerant

$$E_{\text{Carnot}} = \frac{T_0}{T_k - T_0}$$

we find that due to the increase in average temperature T_0 and decrease in average temperature T_k , the COP of the cycle will be increased. The thermodynamic obstacle to the elimination of irreversibility of the heat transfer processes can thus be removed by a judicious selection of a mixture of refrigerants and its composition.

It can be seen from Fig. 4.19 that the use of mixed refrigerant could decrease the work by

area Δw_0 due to non-isothermal evaporation and
area Δw_k due to non-isothermal condensation.

The area $\Delta w_0 = \Delta q_0$ also represents increase in refrigerating effect. It would, therefore, result in increase in COP and hence power saving.

Consider for example a case in which air is cooled from $t_{c_1} = 25^\circ\text{C}$ to $t_{c_2} = 15^\circ\text{C}$, and temperature of cooling air in condenser rises from $t_{h_1} = 45^\circ\text{C}$ to $t_{h_2} = 55^\circ\text{C}$. These are typical values for a window-type air conditioner. Then if pure refrigerant is used, the practical evaporator and condenser temperatures could be $t_0 = 0^\circ\text{C}$ and $t_k = 65^\circ\text{C}$. The corresponding value of Carnot COP would be 4.55.

Now, if a mixture of refrigerants is used for evaporating from $t_0 = 0^\circ\text{C}$ to $t_0' = 10^\circ\text{C}$, and condensing from $t_k = 60^\circ\text{C}$ to $t_k' = 50^\circ\text{C}$, then the Carnot COP would be 5.55. The saving in power would be 18%.

Note It is important to observe that this effect can be utilized only if there is counterflow of the two fluids which is possible only if the mixture is boiling and condensing inside tubes as in window-type air conditioner. Otherwise, if for example, the condensation is outside tubes as in shell and tube condensers, the condensation temperature will rise above the outlet temperature of the coolant ($t_k' \geq 55^\circ\text{C}$ and $t_k > t_k'$). This will cause increase in average temperature of heat rejection, and hence in condenser pressure, work, and power consumption.

Example 4.6 Vapour Compression Cycle of a Hydrocarbon Mixture Refrigerator on p-h Diagram

A hydrocarbon refrigerator operating on equal proportions of propane and isobutane by mass works on simple saturation cycle. The condenser and evaporator pressures are 14 bar and 1.4 bar. Show the cycle on p-h diagram and determine its discharge temperature and theoretical COP.

Solution The values of bubble and dew temperatures at these pressures have already been determined in Example 4.4. Accordingly, the p-h diagram is drawn as shown in Fig. 4.20.

From tables of properties for 50-50% R 290/R 600a mixture in Appendix

$$\begin{aligned} h_1 &= 545.0 \text{ kJ/kg}, h_3 = 341.15 \text{ kJ/kg} = h_4 \\ s_1 &= 2.47 \text{ kJ/kg.K} = s_2 \end{aligned}$$

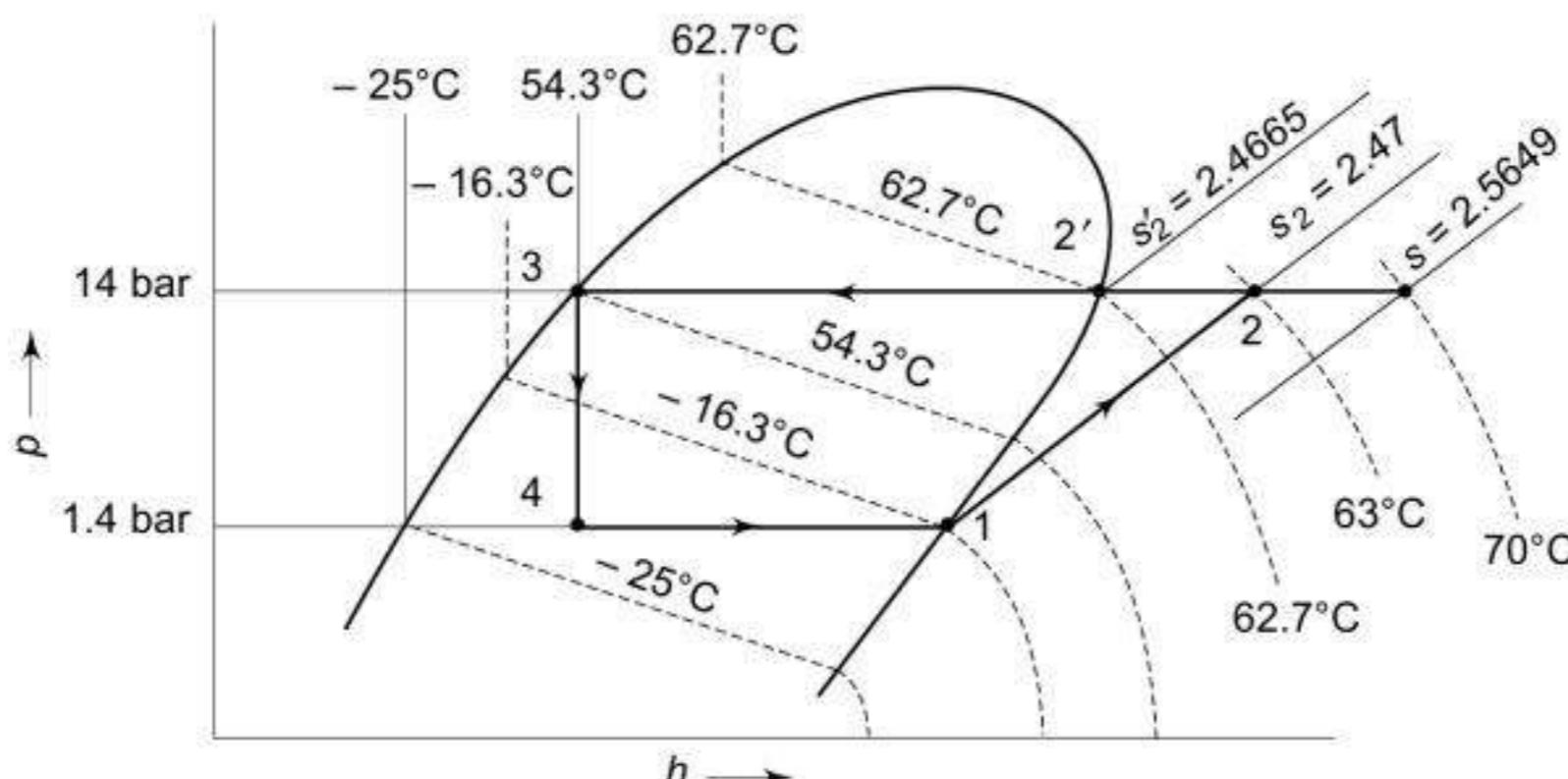


Fig. 4.20 Vapour compression cycle for a refrigerator with 50-50% propane/isobutane mixture

From superheat table for the mixture, for known p_2 and s_2 , we have:
Discharge temperature

$$\begin{aligned} t_2 &= 63^\circ\text{C} \\ \Rightarrow h_2 &= 661.0 \text{ kJ/kg} \\ \text{Hence } \text{COP} &= \frac{q_0}{w} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{545 - 34.15}{661.0 - 545} = 4.7 \end{aligned}$$

Example 4.7 Using Binary Mixture R152a/R22 for Non-Isothermal Refrigeration in Air Conditioners.

It is proposed to substitute R 22 by a mixture of 18% R 152a and 82% R 22 by mass in an air conditioner operating on simple saturation cycle. The condenser and evaporator pressures are 15 bar and 5 bar. Locate all the four state points of the cycle on t - ξ , h - ξ and p - h diagrams.

Also, find the discharge temperature, temperature after throttling, and theoretical COP of the cycle.

Solution The cycle $abcd$ with the mixture is shown on p - h diagram in Fig. 4.21.

The parameters of states as obtained from t - ξ and h - ξ diagrams in Figs. 4.13 and 4.14, and plotted in Figs. 4.22 (a) and (b) are as follows:

State a Saturated vapour

$$p = 5 \text{ bar}, t_a = 5.75^\circ\text{C}, h_a = 428 \text{ kJ/kg}$$

State b Superheated vapour

$$p = 15 \text{ bar}$$

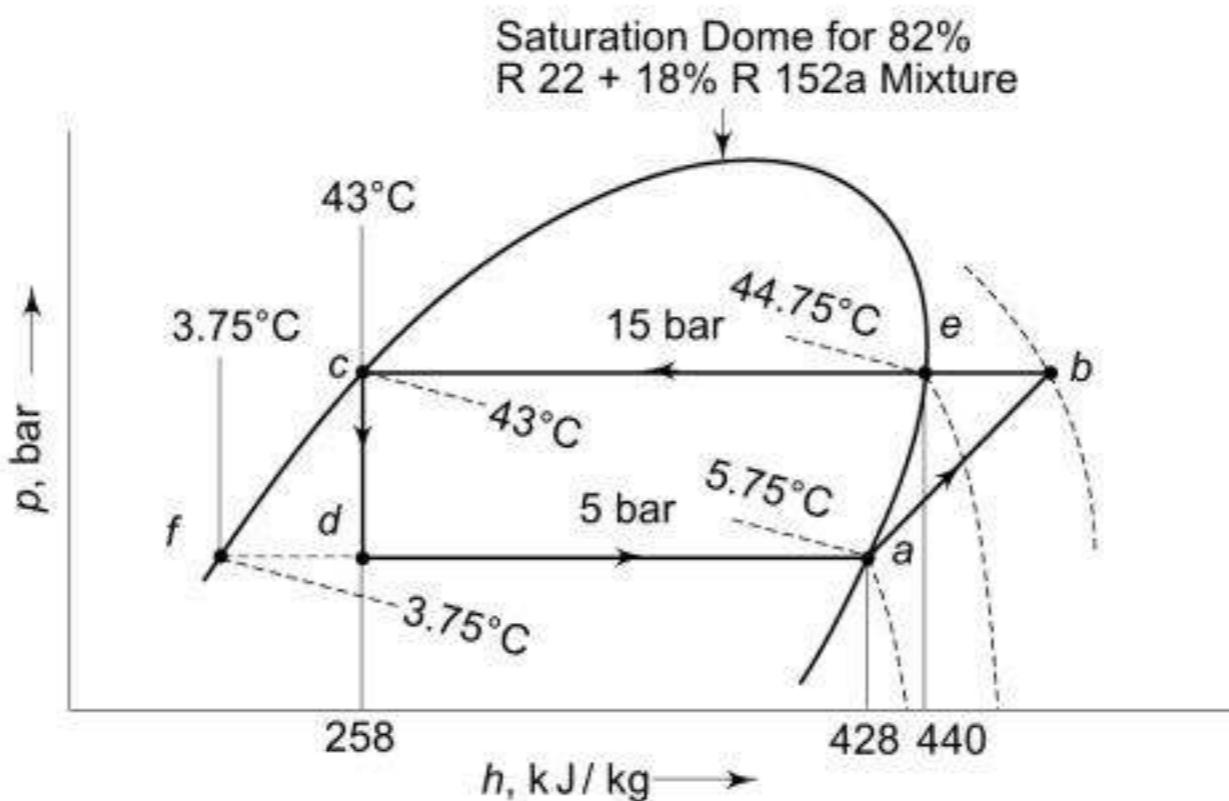
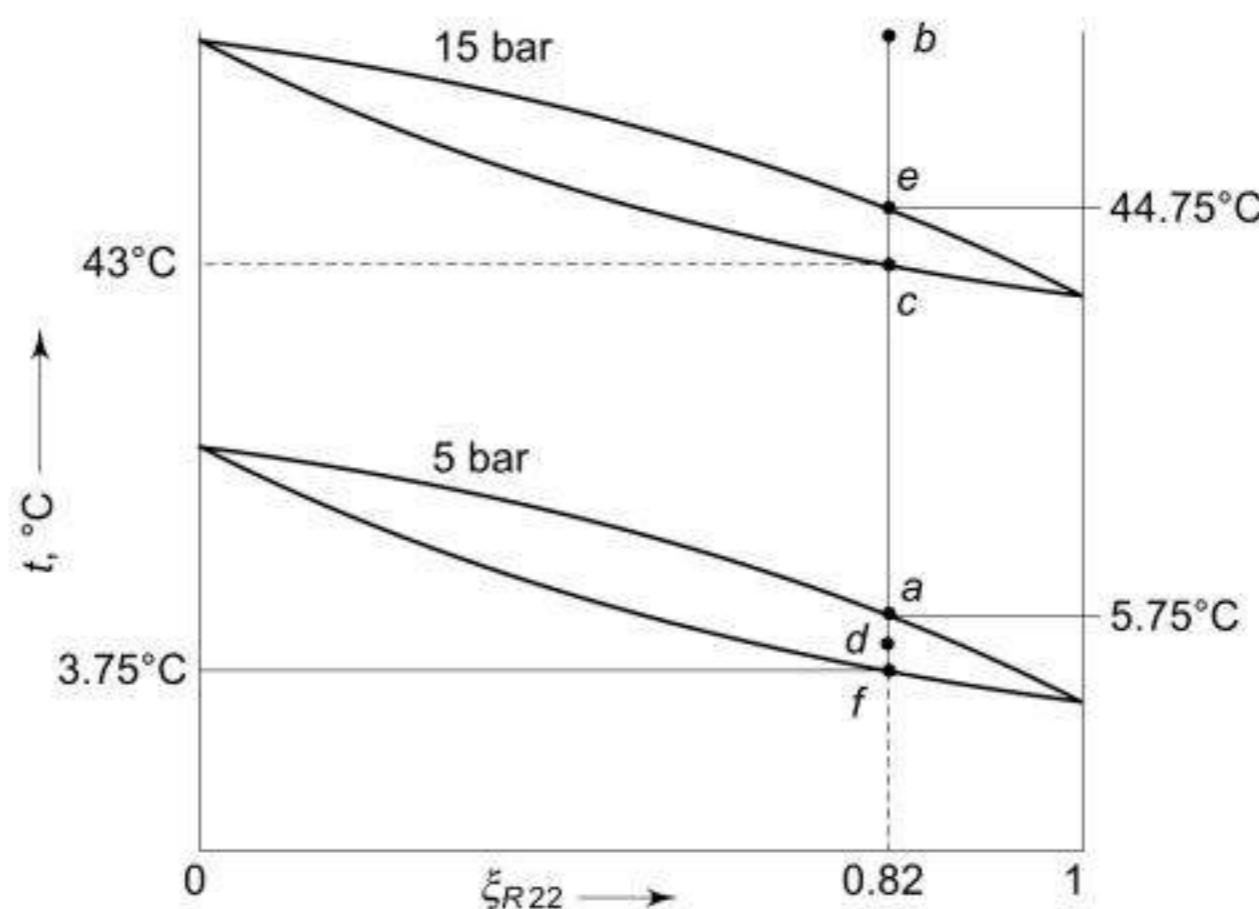


Fig. 4.21 Pressure-enthalpy diagram for R 22/R 152a mixture for Example 4.7

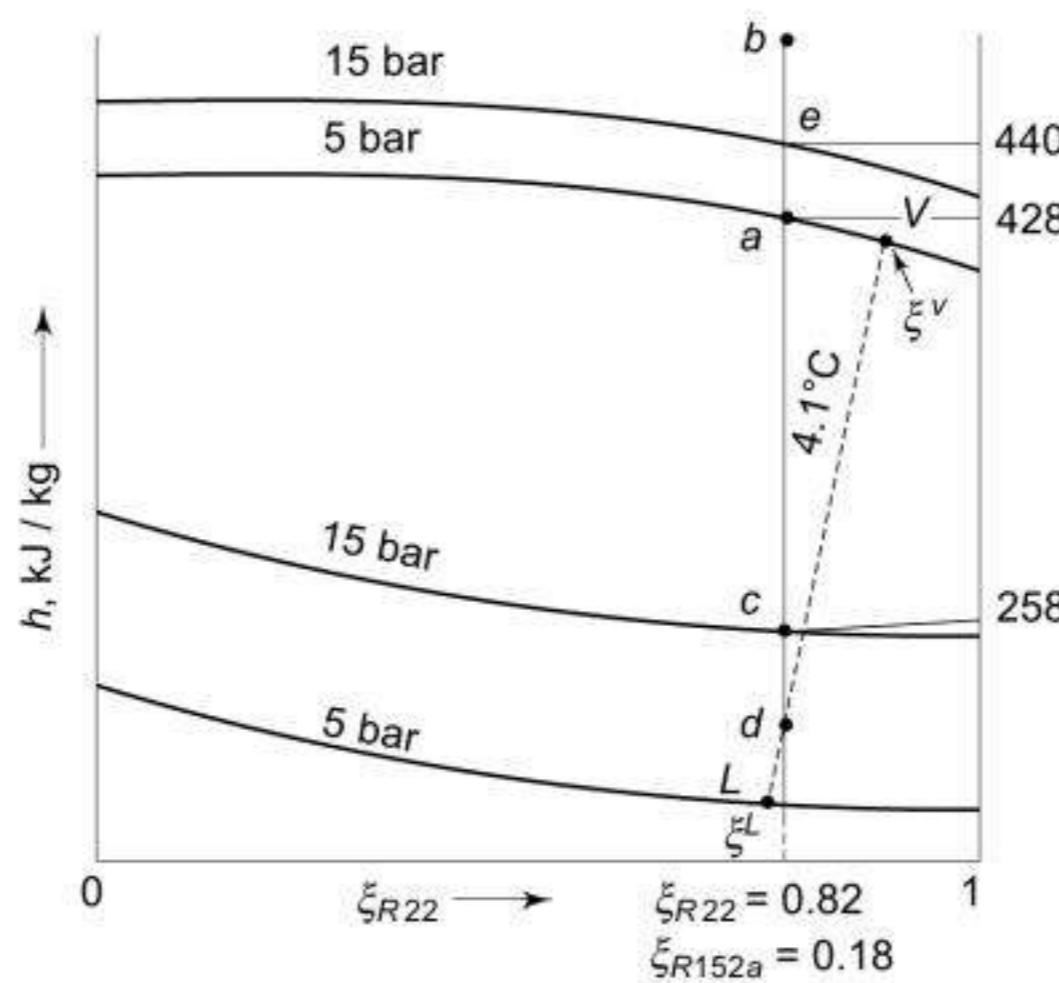
To find its temperature from $s_b = s_a$

State c Saturated liquid

$$p = 15 \text{ bar}, t_c = 43^\circ\text{C}, h_c = 258 \text{ kJ/kg}$$



(a) Temperature-composition diagram



(b) Enthalpy composition diagram

Fig. 4.22 Figure for Example 4.7

State d Liquid-vapour mixture after throttling

$$p = 5 \text{ bar}, h_d = 258 \text{ kJ/kg} = h_c$$

Temperature at *d* lies between 3.75°C at *f* and 5.75°C at *a*.

State e Saturated vapour mixture at 15 bar is at 44.75°C , and its enthalpy is 440 kJ/kg

Entropy at *a*

$$s_a = (\xi s)_{R152a} + (\xi s)_{R22} + \Delta s_m$$

Values of specific entropies of R 152a and R 22 are found from their superheat tables at 5 bar and 5.75°C . And Δs_m is the entropy of mixing. Thus

$$\begin{aligned} s_a &= 0.18(2.1183) + 0.82(1.7518) + \Delta s_m \\ &= 1.8178 + \Delta s_m \text{ kJ/kg.K} \end{aligned}$$

Entropy at e is similarly found at 15 bar and 44.75°C (317.75 K)

$$\begin{aligned}s_e &= 0.18(2.0838) + 0.82(1.6996) + \Delta s_m \\ &= 1.7686 + \Delta s_m \text{ kJ/kg.K}\end{aligned}$$

Discharge temperature at b

The equation to solve is

$$\begin{aligned}s_b &= s_e + \xi_1 C_{p_1} \ln \frac{T_b}{317.75} + \xi_1 C_{p_2} \ln \frac{T_b}{317.75} = s_a \\ 1.7686 + (0.18 \times 1.61 + 0.82 \times 0.91) \ln \frac{T_b}{317.75} &= 1.8178 \\ \Rightarrow T_b &= 324 \text{ K } (t_b = 51^{\circ}\text{C})\end{aligned}$$

Enthalpy at b

$$\begin{aligned}h_b &= h_e + \xi_1 C_{p_1} (t_b - t_e) + \xi_1 C_{p_2} (t_b - t_e) \\ &= 440 + (0.18 \times 1.61 + 0.82 \times 0.91)(51 - 44.75) = 455.8 \text{ kJ/kg}\end{aligned}$$

Temperature at d

Overall composition and enthalpy during throttling process remain same. Hence
Composition balance equation

$$(\xi_1)_d = [\xi_1^L (1 - z) + \xi_1^V z]_d = (\xi_1)_c = 0.18$$

Enthalpy Balance equation

$$h_d = (h^L)_f (1 - z) + (h^V)_a z = 202 (1 - z) + 428z = h_c = 258$$

The two equations can be solved for t_d by iteration. The equilibrium L-V states at 5 bar from Tables 4.19 and 4.21 are plotted on t - ξ and h - ξ diagrams in Figs. 4.22 (a) and 4.22(b). It is seen that state d is at a temperature slightly above 4°C . The method of solution by iteration is as follows:

- (i) Assume t_d .
- (ii) Find ξ_1^L and ξ_1^V .
- (iii) Find z from the composition-balance equation.
- (iv) Verify if enthalpy-balance equation is satisfied.
- (v) If not, assume another value of t_d .

The solution is

$$t_d = 4.1^{\circ}\text{C}$$

Theoretical COP of the cycle

$$\epsilon = \frac{h_a - h_d}{h_b - h_a} = \frac{428 - 258}{455.8 - 428} = \frac{170}{27.8} = 6.1$$

Note Ghosh and Sarkar²² obtained a saving of 2.7 to 4.3% in power consumption with 9.87% R152a in R 22/R 152a mixture.

Example 4.8 Estimating Power Saving in Air Conditioners with R 152a/R 22 Mixture.

Find the theoretical COP of R 22 cycle equivalent to the mixture cycle in Example 4.7. Estimate the power saving, and discuss the other effects.

Solution In order to maintain the condenser and evaporator sizes same, we have to design for same LMTD in the two heat exchangers. This means, we have to maintain mean condensation and evaporation temperatures as same. Hence, for the equivalent R 22 cycle,

$$t_k = \frac{t_e + t_c}{2} = \frac{44.75 + 43}{2} = 43.9^\circ\text{C}$$

$$t_0 = \frac{t_a + t_d}{2} = \frac{5.75 + 4.1}{2} = 4.9^\circ\text{C}$$

The corresponding R 22 cycle is shown in Fig. 4.23.

COP of equivalent R 22 cycle

$$\varepsilon_{R22} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{407.5 - 254.1}{435.5 - 407.5} = 5.5$$

Power saving by using mixture for same refrigerating capacity

$$\frac{W_{R22} - W_{mix}}{W_{R22}} = \frac{1/\varepsilon_{R22} - 1/\varepsilon_{mix}}{1/\varepsilon_{R22}} = \frac{1/5.5 - 1/6.1}{1/5.5} = 0.1 \text{ (10%)}$$

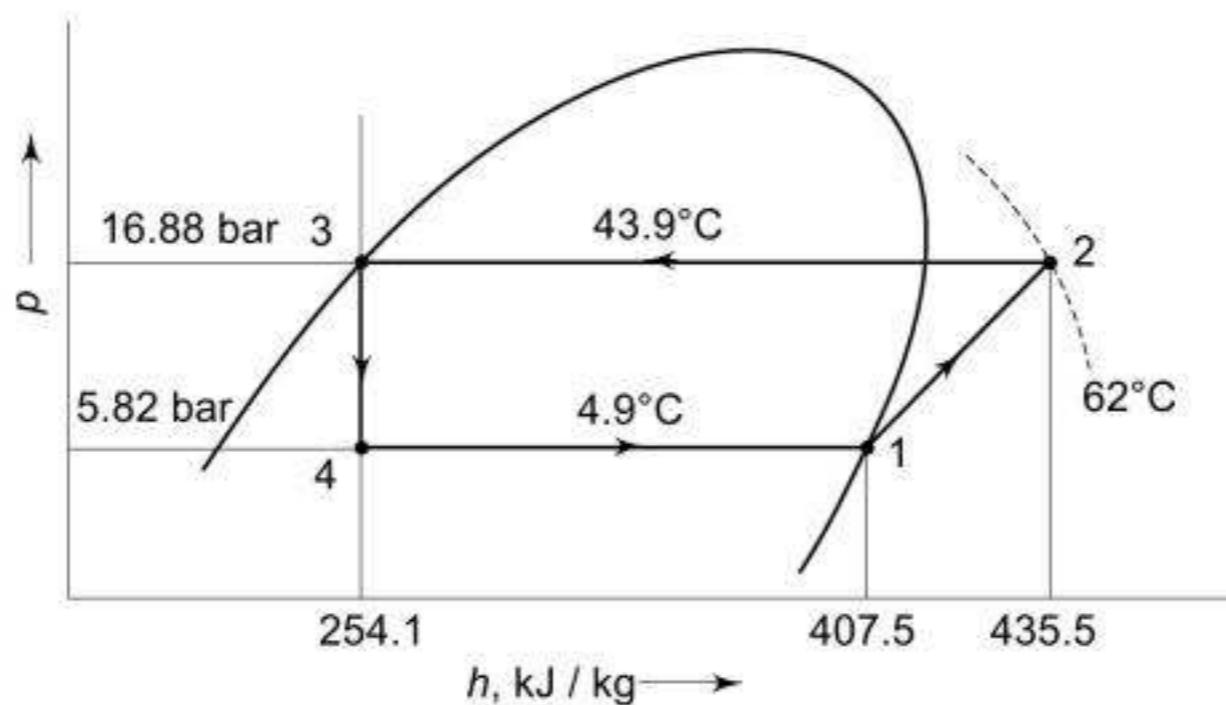


Fig. 4.23 R 22 cycle for Example 4.8

Other Effects:

- (i) Discharge temperature is lowered from 62°C for R 22 to 51°C for mixture.
- (ii) Compressor displacement volume will increase for mixture due to larger volume of R 152a which is a higher boiling substance.
- (iii) R 152a improves the miscibility of R 22 in mineral oil.

4.24 REFRIGERANT PIPING AND DESIGN

A well-designed refrigeration plant is one with a well-sized and well-laid refrigerant piping. Some salient features of this are discussed below. The material used for fluorocarbon refrigerant piping is either seamless copper tubing or iron, whereas for ammonia only iron pipes are used. Their sizes are given in terms of the outside diameter (OD) for copper tubing and normal iron pipe size (IPS) for iron pipes.

4.24.1 Location and Arrangement of Piping

The following factors are important:

- (i) To minimize tubing and refrigerant charge requirement and pressure drops, refrigerant lines should be as short and direct as possible.
- (ii) Piping is to be planned for a minimum number of joints using as few elbows and fittings as possible, although providing for sufficient flexibility to absorb compressor vibrations.
- (iii) It is to be so arranged that normal inspection and servicing of the compressor and equipment is not hindered.
- (iv) It should be so run that it does not interfere with the removal of compressor-cylinder heads, access plates or any other internal part such as the stator in hermetically-sealed units.
- (v) Sufficient clearance is to be provided between the piping and wall or between pipes for insulation.
- (vi) Locations where copper tubing will be exposed to mechanical injury, should be avoided.
- (vii) In the case of iron pipes, *hangers* should be provided close to vertical risers to and from compressors to keep the piping weight off the compressors.
- (viii) Hangers should not be placed more than 2 to 3 m apart and should be within 0.6 m of the change in direction of piping.
- (ix) Valves should be located in inlet and outlet lines to condensers, receivers, evaporators and long lengths of pipes to permit isolation and to facilitate *pumping down* and *pumping out*. Valve stems should be horizontal so that there is less chance of dirt or scale accumulating on the valve seat.

4.24.2 Vibration and Noise in Piping

Vibration and noise can be eliminated by proper design and support of the piping.

The undesirable effects of vibration are:

- (a) Breaking of brazed or soldered joint and consequent loss of charge.
- (b) Transmission of noise through piping and building.

The vibration of piping is caused by the rigid connection of the refrigerant piping to a reciprocating compressor. It is impossible to eliminate it completely. But steps such as the following can be taken to mitigate its effects:

- (i) Run the suction and discharge lines at least 15 pipe diameters in each of two or three directions before securing the first hanger. In this manner, piping can absorb vibrations without being overstressed.
- (ii) A flexible metal hose is often used to absorb vibration transmitted along small rises of pipes. For great effectiveness, it should be installed at right angle to the direction of vibration. Most compressor vibrations have the greatest amplitude in the horizontal plane. Therefore, the flexible hose should be installed in the vertical lines near the compressor. Two isolators, one vertical and one horizontal, may be required to do an efficient job.
- (iii) Vibration and noise are also caused by gas pulsations and turbulence in gas which is increased at high velocities. It is usually more important in discharge lines. Under some conditions, these pulsations may occur at such a frequency as to cause resonance, sufficiently strong to break piping connections. In such a



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required to offset friction and static pressure drop. The static pressure drop associated with an increase in elevation is: 0.1275 bar for every 1.11 m lift for R 22.

When a system is equipped with a receiver to maintain a seal of the liquid refrigerant in the control devices, the liquid line entering the receiver is usually designed generously. A velocity of 0.5 m/s is typical to ensure pressure equalization between the condenser and receiver and to prevent vapour locking. With this velocity a vapour-equalising line should be provided from the top of the condenser to the top of the receiver. Otherwise, a lower velocity can be used.

For liquid line between the receiver and expansion device, it is recommended to design it on the basis of pressure loss of 0.05 bar per 100 m of length.

In general, the pressure drop should not be greater than that corresponding to 0.9°C change in the saturation temperature for ammonia, and 0.9 to 1.8°C for fluorocarbons. A change of 0.9°C in the saturation temperature at 38°C condenser comes to a pressure drop of approximately 0.227 bar for ammonia, and 0.2 bar for R 22.

4.24.6 Suction Lines

Suction lines are the most critical from the viewpoints of design and construction. The considerations involved are the following:

- (i) Correct size for practical pressure drop.
- (ii) Capability to return oil to the compressor by entrainment by the suction vapour under minimum loading conditions, especially in the case of fluorocarbons.
- (iii) If there are suction risers, gas velocities to ensure oil return have to be increased. Thus a minimum tonnage is prescribed depending on the size of the suction line for oil entrainment in suction risers as given in Table 4.26 for R 22.

Table 4.26 Tonnage of suction piping up suction risers

<i>Refrigerant</i>	<i>t₀, °C</i>	<i>Pipe OD, cm</i>			
		1.27	1.59	1.905	2.86
R-22	-40	0.09	0.16	0.27	0.79
	-10	0.13	0.24	0.39	1.2
	5	0.18	0.33	0.54	1.6

- (iv) Double-suction risers may be used for full-load operation and single risers for part-load operation as shown in Fig. 4.24. Thus, excessive pressure drop at full load is avoided, and oil return at part load is ensured. When the load reduces, the oil cannot be entrained in the beginning which collects in the U-bend and forms a seal. Afterwards, the gas flows only through one riser.
- (v) Prevention of drainage of oil from an active evaporator into an idle evaporator. For this purpose, arrangements are made as shown in Figs 4.25 to 4.27. The common-suction line should either be horizontal or pitched down towards the compressor.

It is customary to design suction lines so that the total loss in pressure is equivalent to drops of about 1.8 and 0.9°C in saturation temperatures for fluorocarbons and ammonia respectively. It is also desirable to provide for less pressure drop in low-

temperature installations because of increased penalties on the compressor size and performance. Since the pressure loss is also a function of the length of the line and number of fittings, it is obviously desirable to have the shortest possible runs of suction lines.

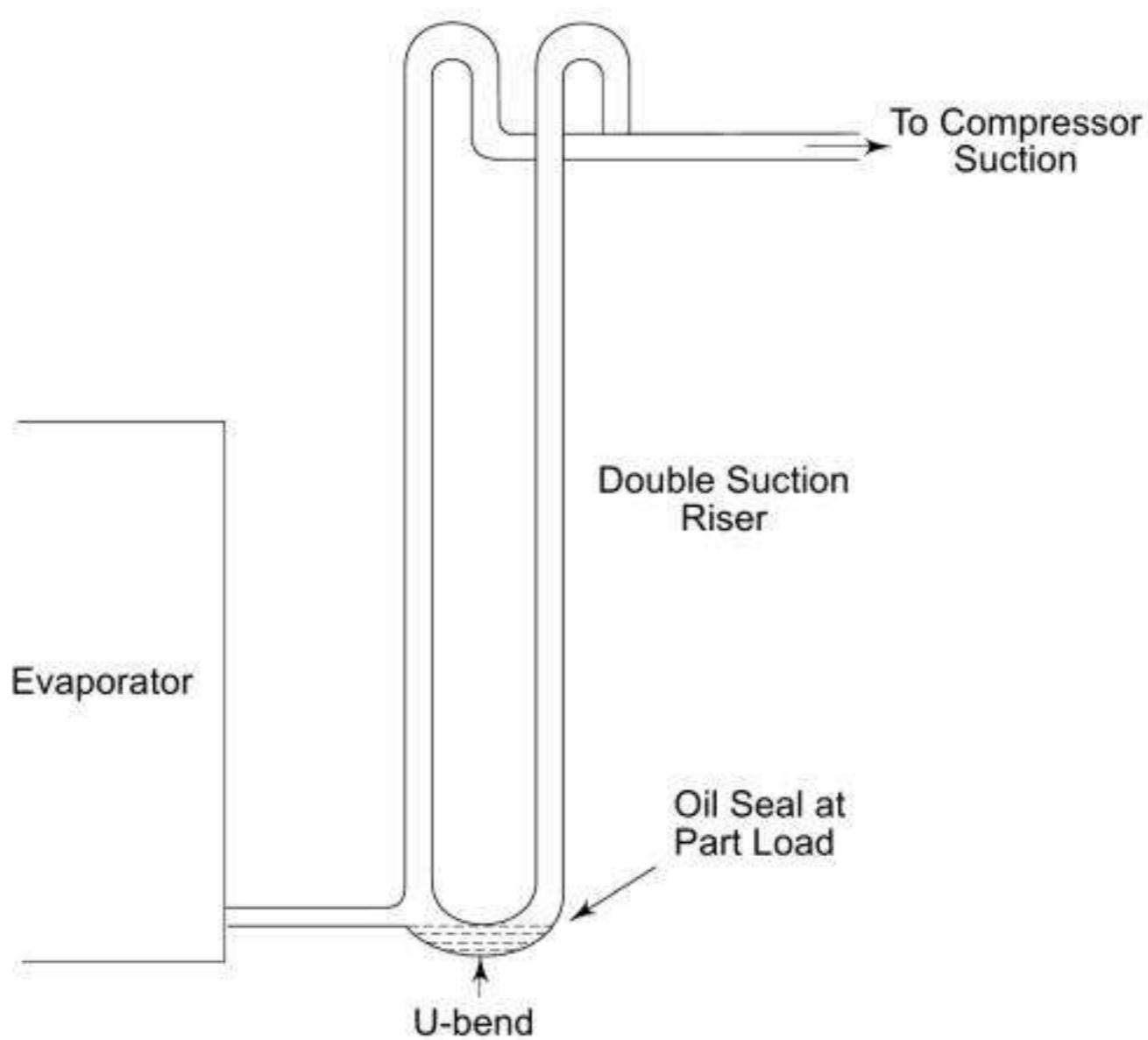


Fig. 4.24 Double-suction riser

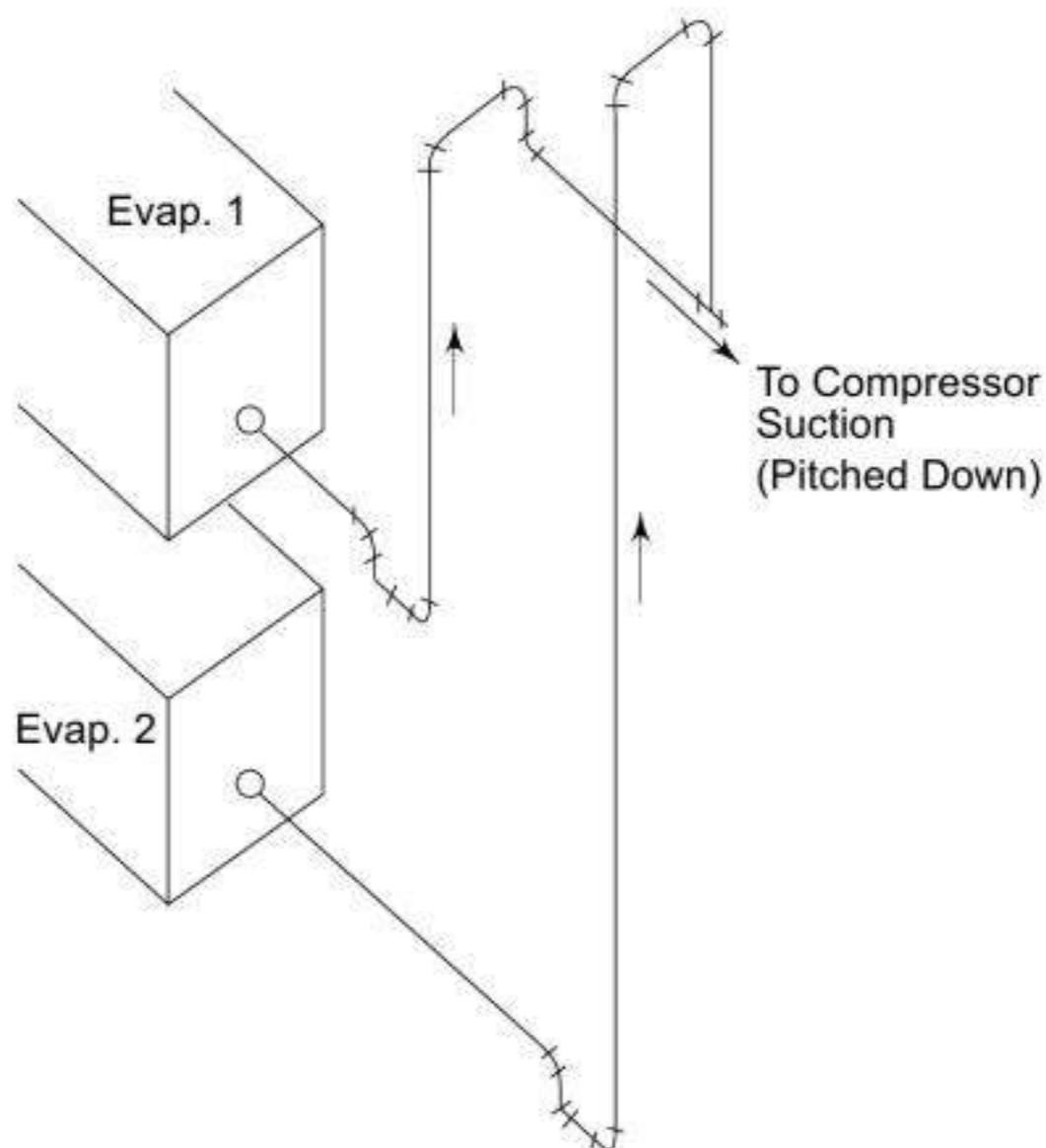


Fig. 4.25 Multiple evaporators at different levels with compressor above

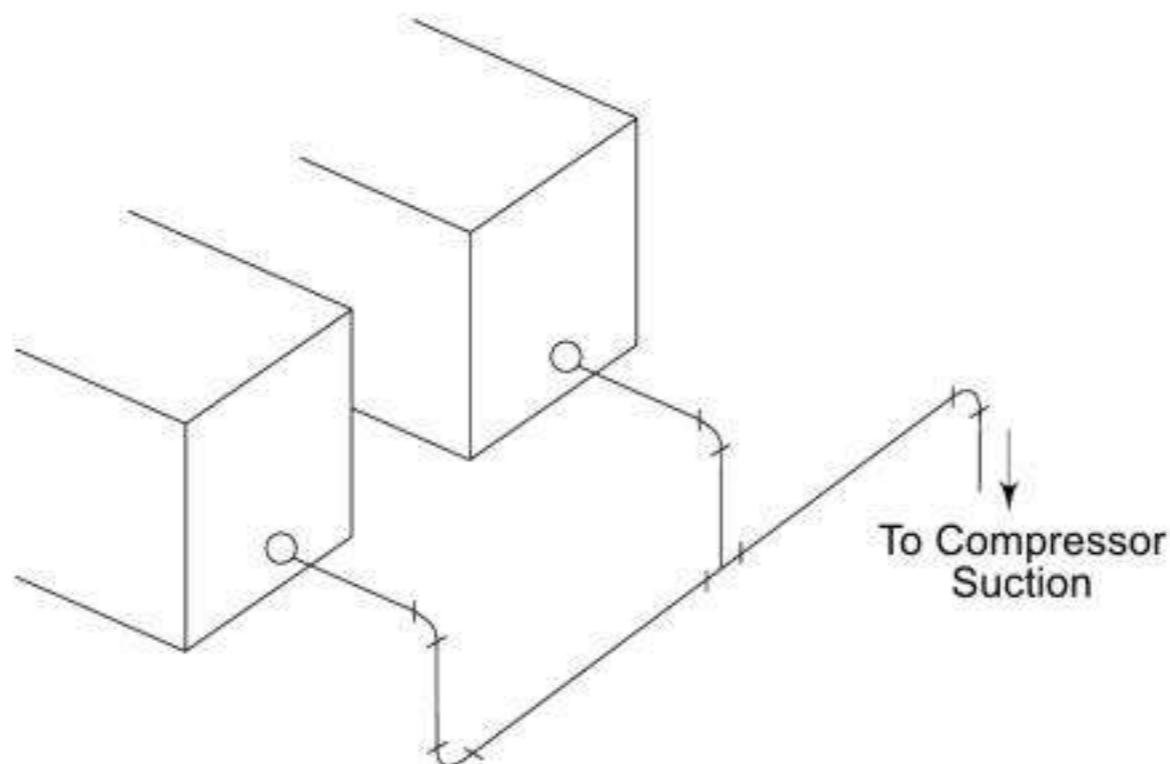


Fig. 4.26 Multiple evaporators at same level with compressor below

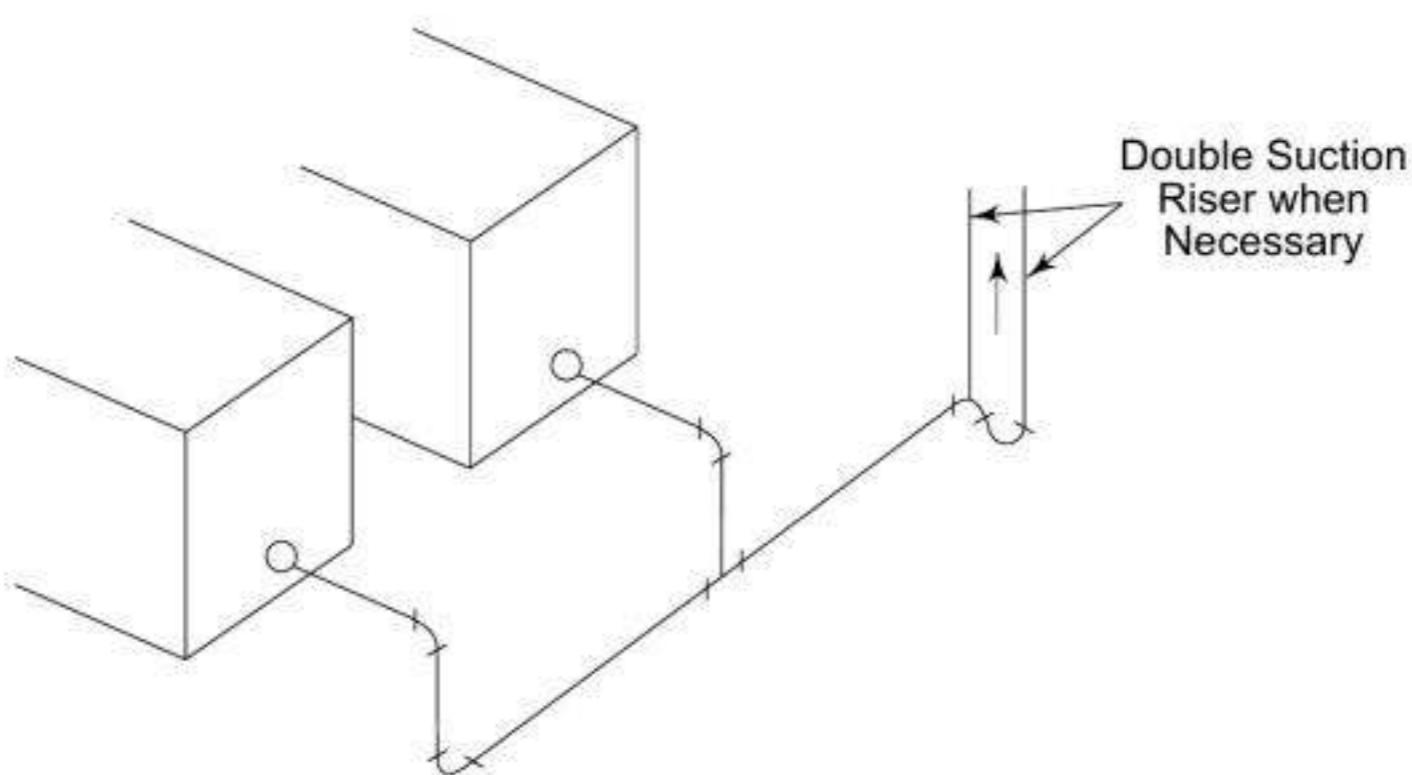


Fig. 4.27 Multiple evaporators at same level with compressor above

4.24.7 Discharge Lines

Even though a low pressure drop is desired, the discharge lines should not be oversized to the extent that the gas velocities are reduced so much that the refrigerant will not be able to carry along the entrained oil. They should be so designed and constructed as to prevent the refrigerant and oil in the line from condensing and draining back to the compressor, especially during shutdown or operation at low ambient where long outdoor discharge lines are required as in evaporative condensers.

Whenever the condenser is located above the compressor, the discharge line should loop to the floor before rising to the compressor as shown in Fig. 4.28. It is not desirable to exceed a pressure drop equivalent to 1 or 2°C in the saturation temperature because of the penalty on volumetric efficiency and hence the capacity of the compressor. Discharge lines should be selected on the basis of 0.45 to 0.67 bar per 100 m pressure drop.

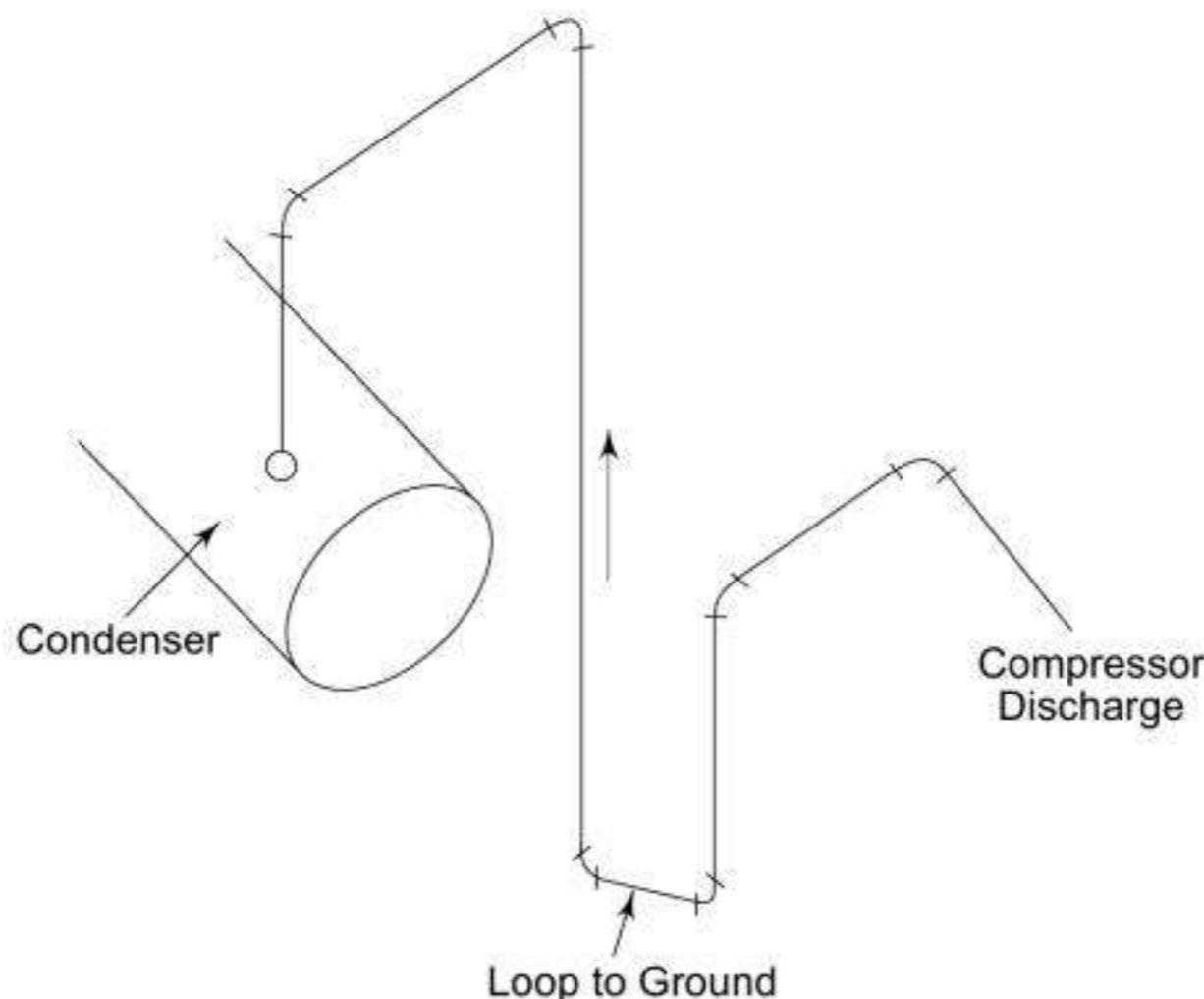


Fig. 4.28 Hot-gas loop

4.24.8 Piping at Multiple Compressors

The piping of compressors operating in parallel must be carefully done to ensure proper operation.

There should be a common suction header. This will ensure equal oil return to each compressor and that all compressors run at the same suction pressure. The suction header should be run above the level of the compressor suction inlet so that all oil can drain into the compressor by gravity. If the header is below the compressor-suction inlets, it can become an oil trap. Further, branch-suction lines should be taken from the side of the header to ensure equal distribution of oil and to prevent the accumulation of liquid refrigerant in an idle compressor in the case of slop over. The horizontal takeoffs to the various compressors should be of the same size as the suction header. No reduction is to be made in the branch suction lines until the vertical drop is reached.

It is always desirable, even with two or more compressors, to use a single condenser (parallel operation).

4.25 LUBRICANTS IN REFRIGERATION SYSTEMS

In refrigeration systems, oil must perform certain functions other than minimizing friction, such as sealing the gas between suction and discharge ports, acting as a coolant to transfer heat from the crank-case to the compressor shell and to dampen the noise generated by moving parts. For hermetic units, it must also have a high dielectric strength. It must have adequate fluidity at low temperatures so that it may easily return to the compressor from the evaporator. For good heat transfer in the evaporator, oil should remain miscible with the refrigerant at evaporator temperatures. Further, it should not contain any suspended matter, wax or moisture which might choke the

expansion device. In hermetic units, oil is charged only once for the life-time of the unit (minimum five years). It should, therefore, be chemically stable in the presence of the refrigerant, metals, winding insulation and extraneous contaminants.

The oils of mineral origin contain paraffins, napthenes, aromatics and non-hydrocarbons. Paraffinic and napthemic oils are saturates. They have excellent chemical stability but poor solubility for refrigerants, such as R 22, and are also poor lubricants. Aromatics have good solubility and lubrication properties. Non-hydrocarbons are the most reactive but are good for lubrication. The resultant properties depend on the proportional composition of the four constituents.

Synthetic oils, such as alkylbenzenes and phosphate esters, have desirable properties for use with R 22. Polyalkylene glycols (PAGs), modified PAGs, and esters are primary lubricants that are being tested for use with R 134a.

Normally, one should choose an oil with the lowest viscosity which will, at the same time, give the necessary sealing properties with the refrigerant used. Thus, the oil which gives the maximum volumetric efficiency is the best. The prescribed oil kinematic viscosities for reciprocating compressors for use with ammonia and R 22 are 150-300 SSU (Saybolt Universal Units) at 38°C.

4.26 SECONDARY REFRIGERANTS

In large refrigeration plants, secondary refrigerants or coolants such as water, brines, glycols and sometimes even halocarbons are used for carrying refrigeration from the plant room to the space where it is usefully applied, instead of directly obtaining it by the evaporating refrigerant at the place of application. This is done in order to reduce the quantity of the refrigerant charge in the system and to reduce pressure losses in lines. The desirable properties of secondary coolants are low freezing point, low viscosity, non-flammability, good stability and low vapour pressure. Chilled water is used as a secondary refrigerant in air-conditioning applications. For low-temperature applications, brines, glycols and hydrocarbons are used.

4.26.1 Brines

Brines are formed by dissolving salt in water. The phase diagram for a brine solution is shown in Fig. 4.29.

The function of salt in water is to depress its freezing point. The temperature at which the freezing point is lowest is called the *eutectic temperature*, and the composition at this temperature is called the *eutectic composition*.

If a brine solution has less than eutectic composition, such as at *A*, its crystallization temperature or freezing point will be lowered to t_B . If this solution at *A* is cooled, ice crystals will begin to form at *B*. As a result, the solution will become richer in salt content. At any point *C*, the mixture will consist of ice at *C₂* and solution at *C₁*. At point *D*, the solution will have the composition corresponding to *D₁*. On further cooling, the solution will freeze as a whole without any separation of ice crystals.

Similarly, if a brine solution of more than eutectic composition such as at *E* is cooled, it will first separate into salt and solution, until it reaches point *D₁* again, whereafter the solution will freeze as a whole.



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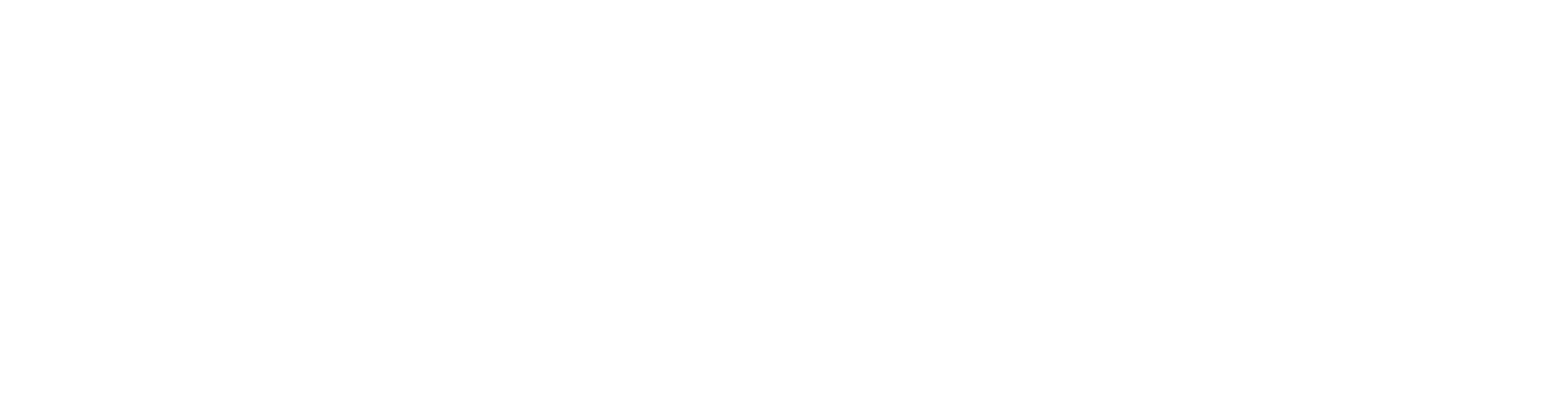
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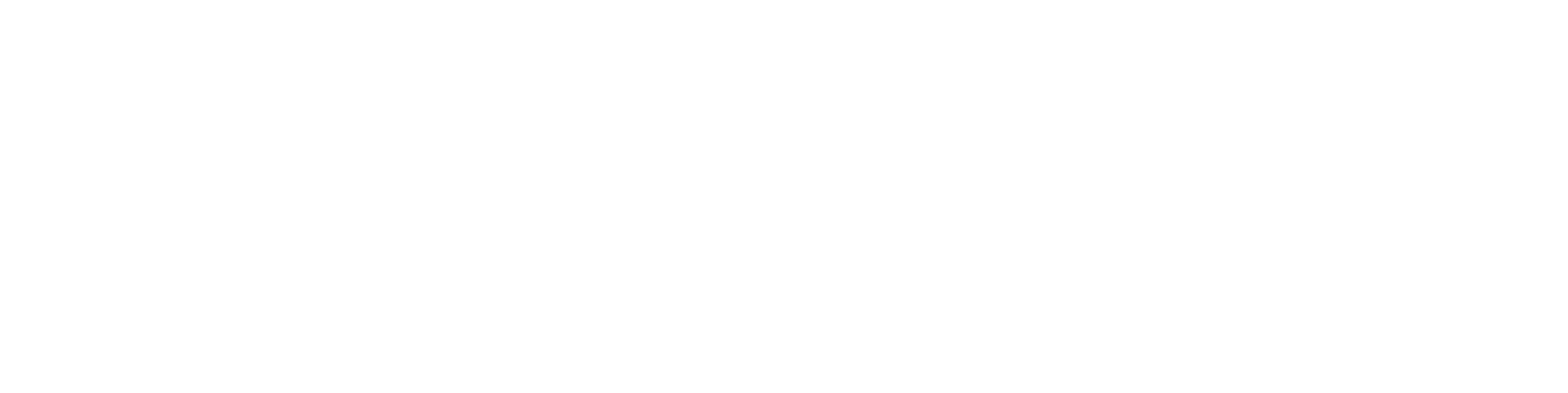
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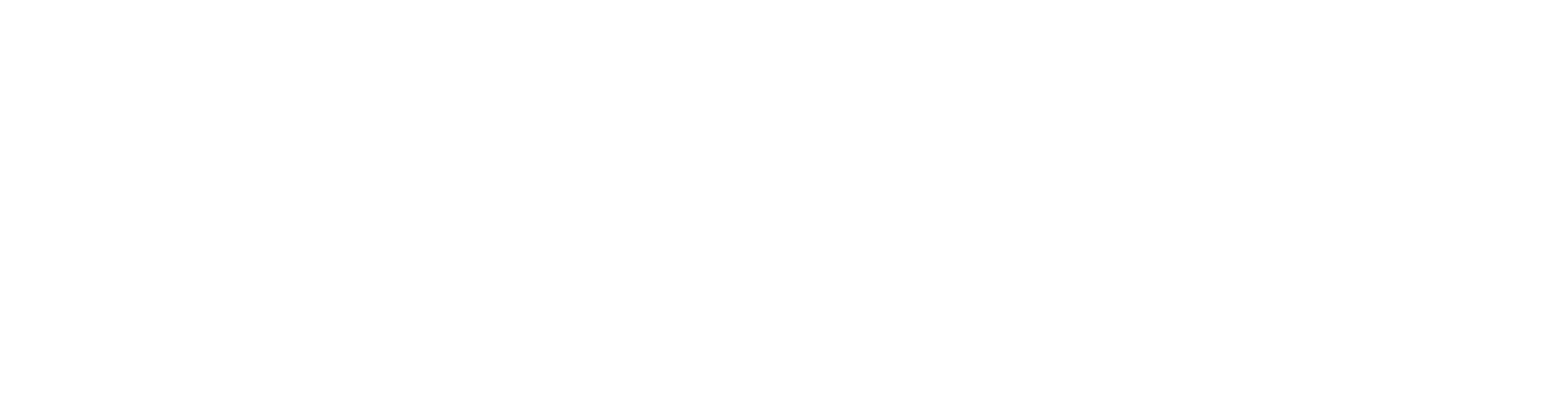
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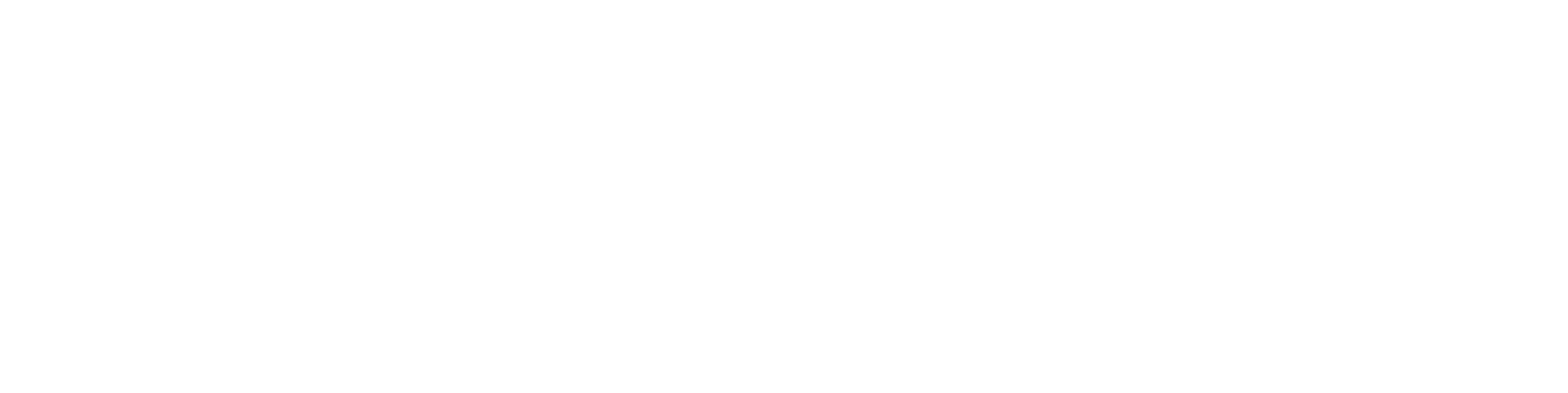
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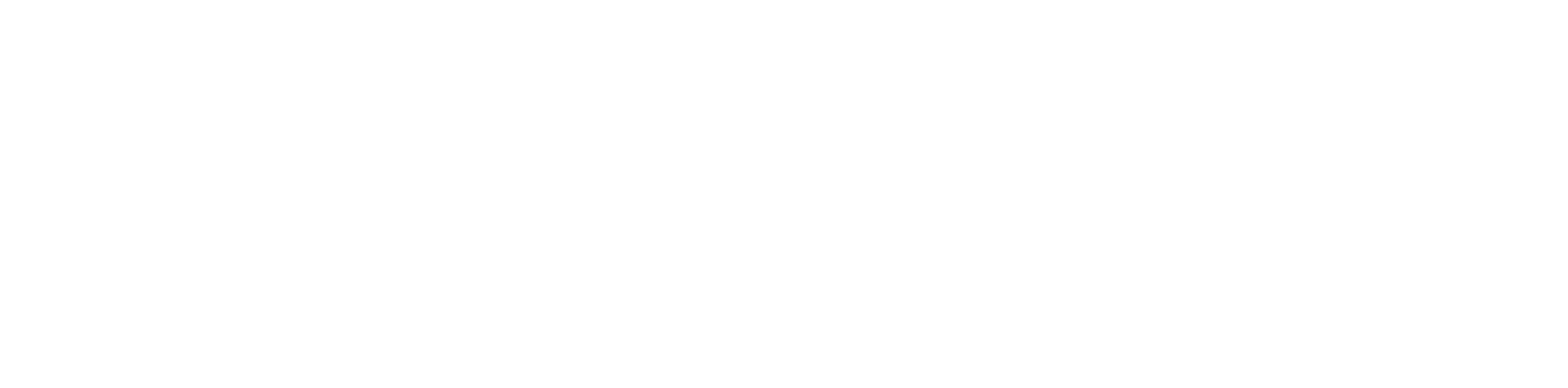
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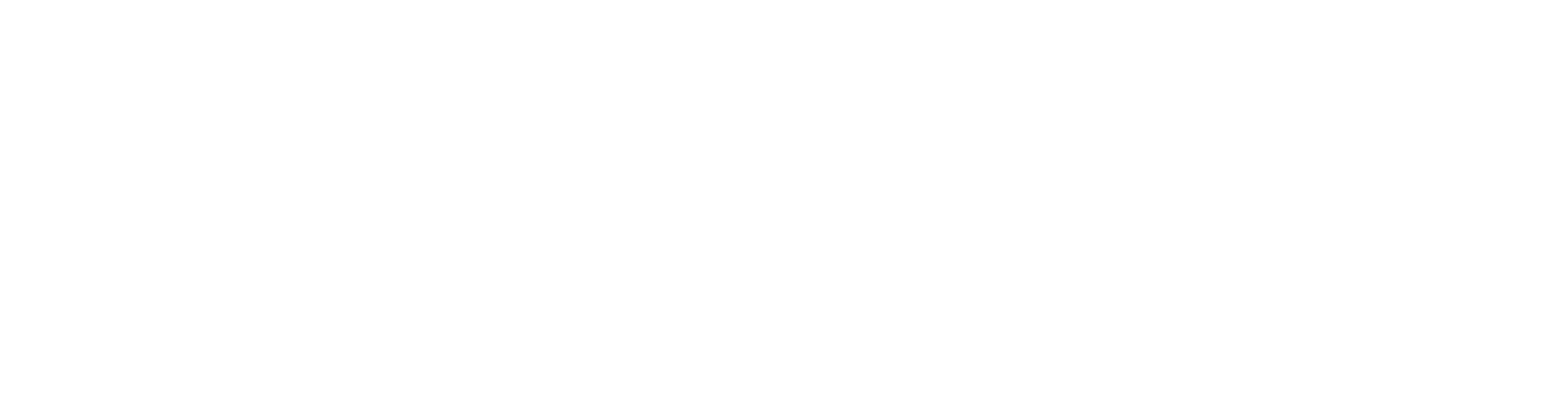
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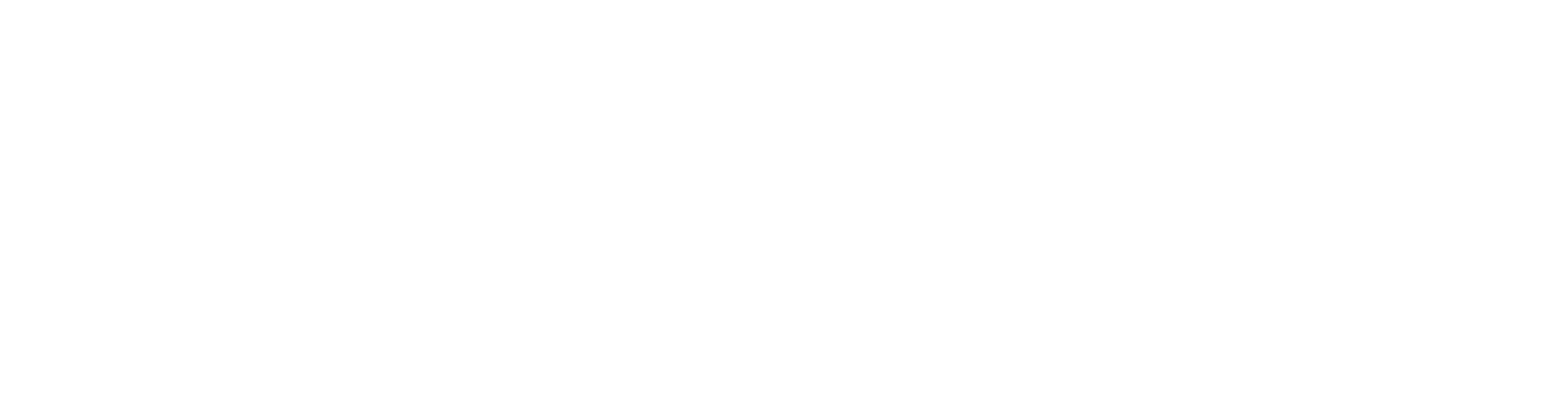
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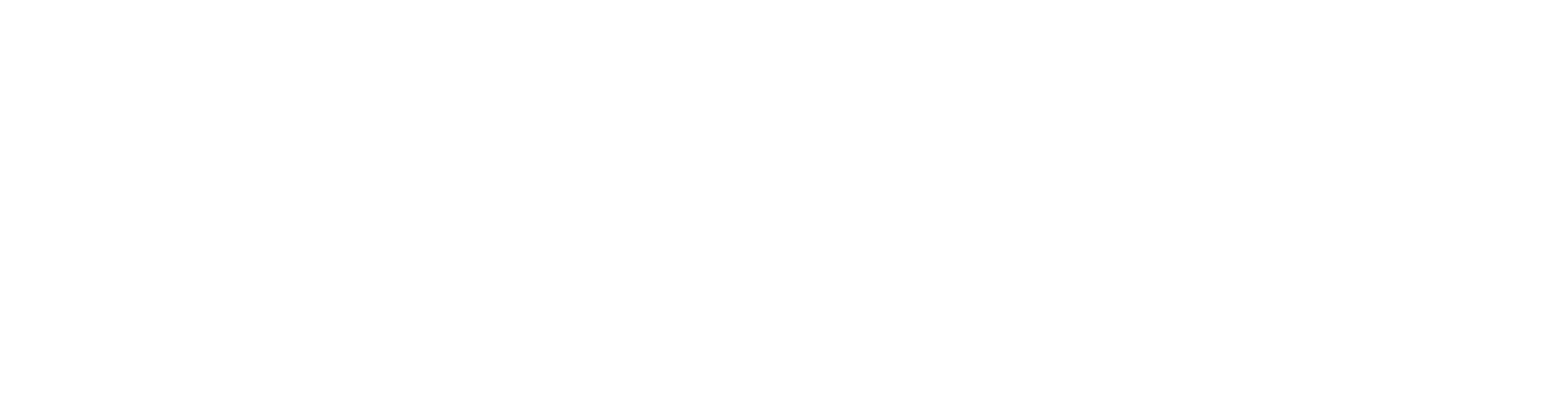
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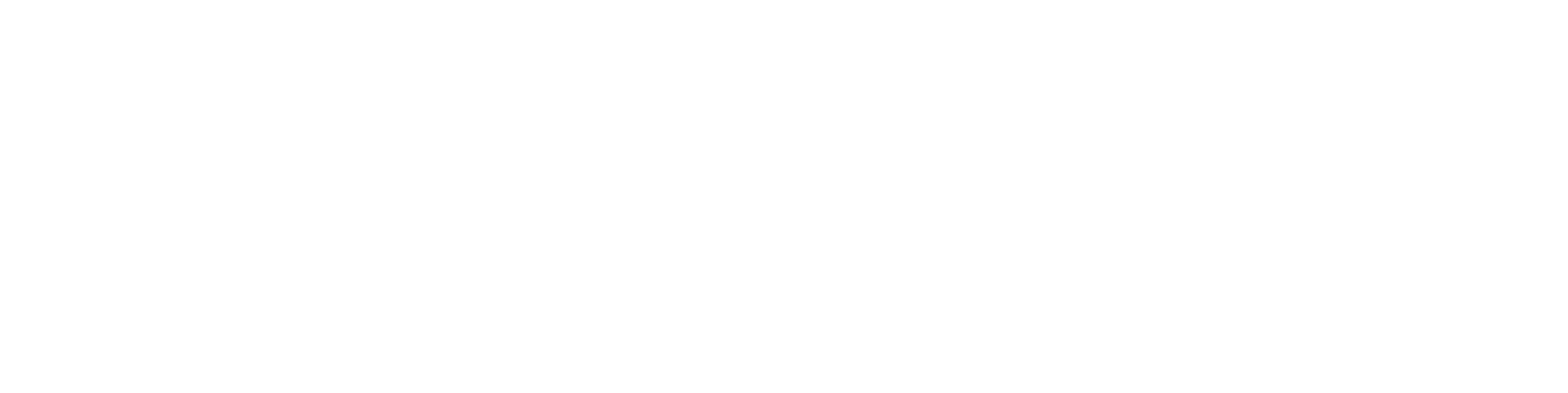
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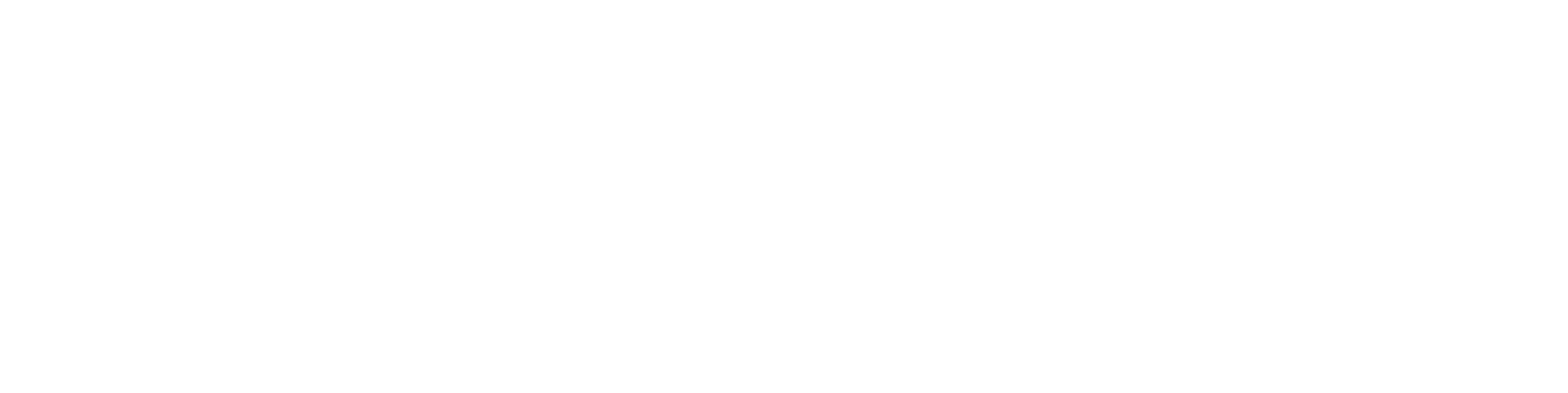
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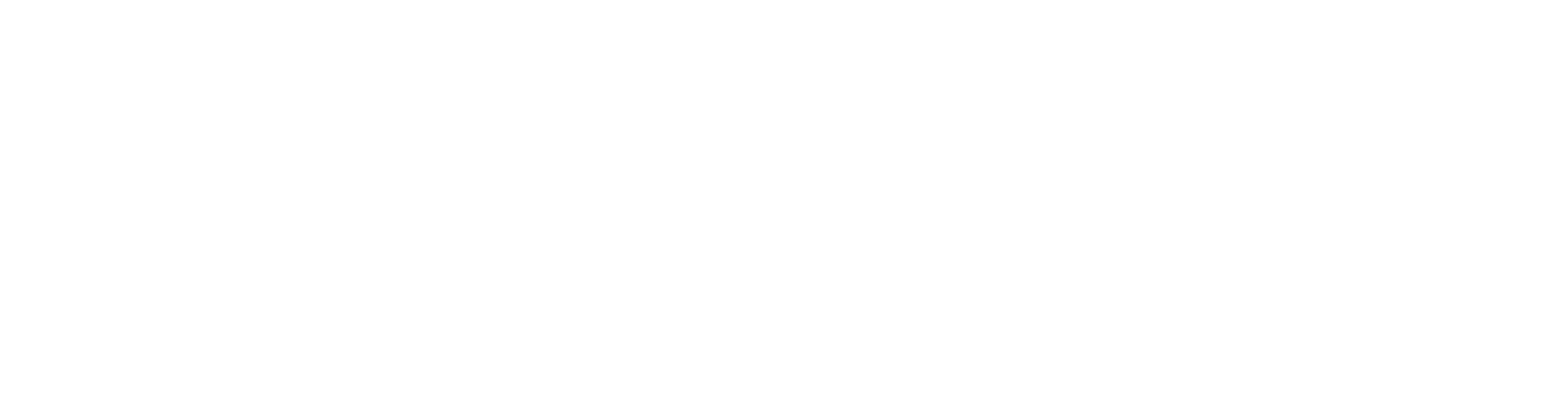
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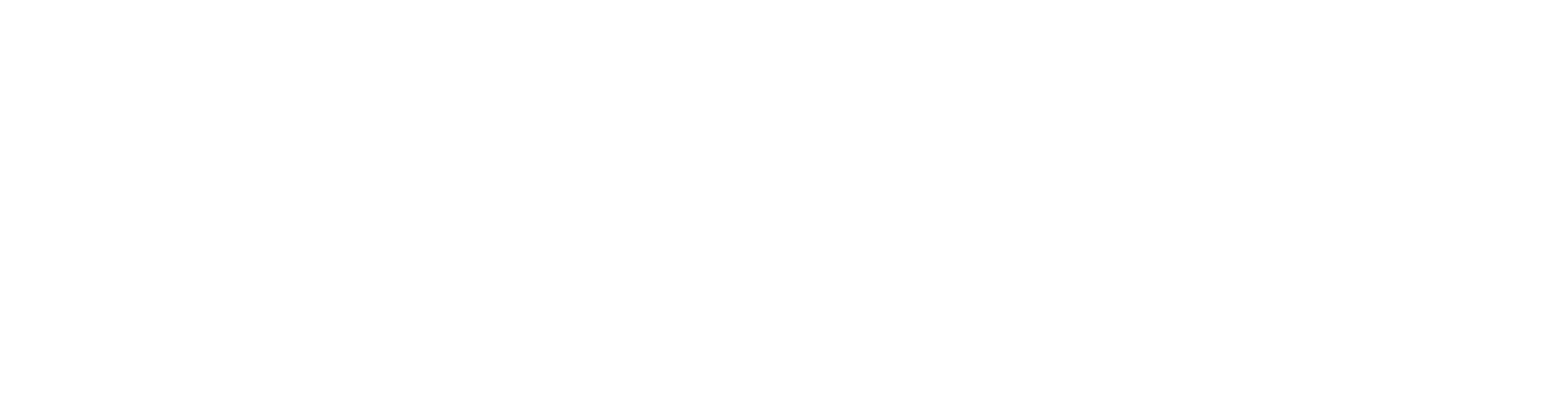
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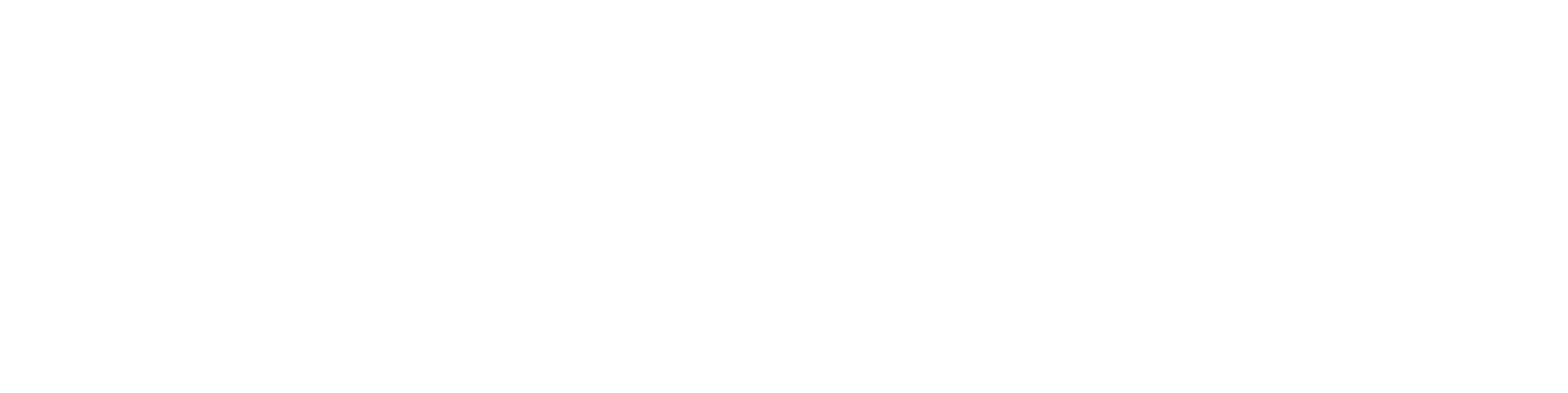
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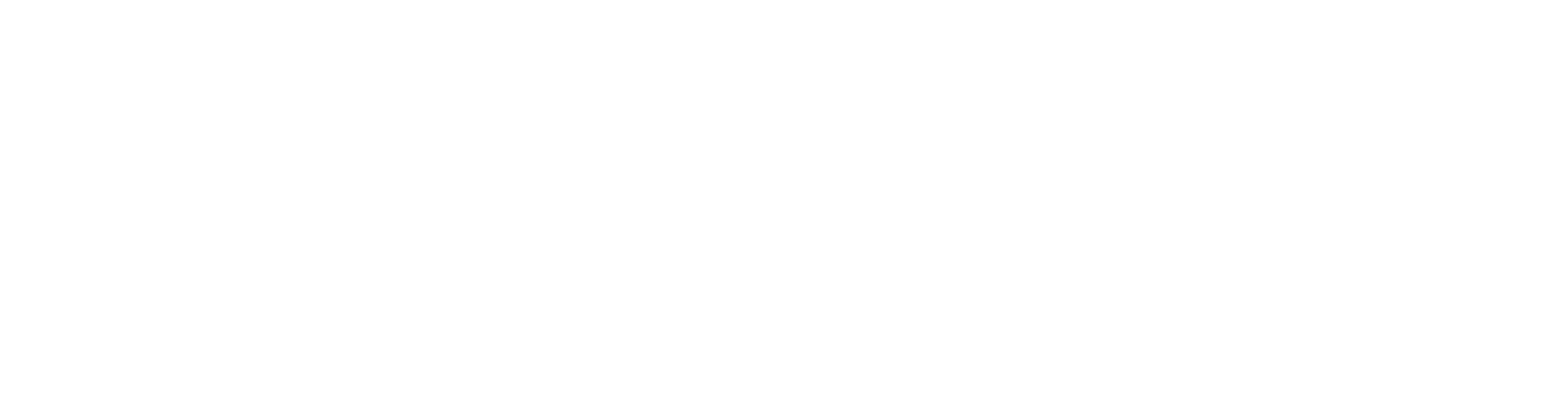
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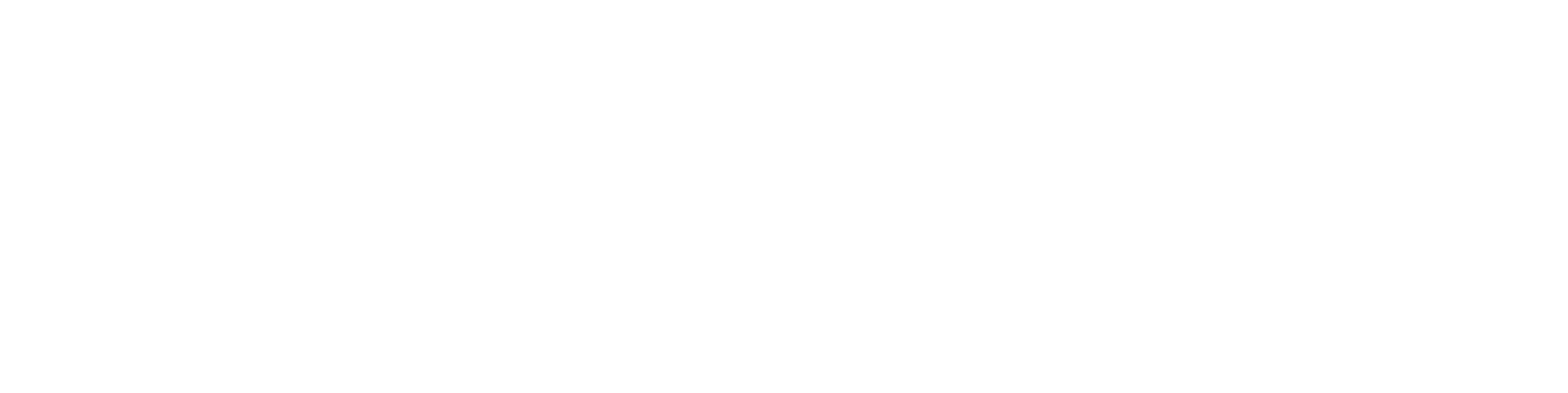
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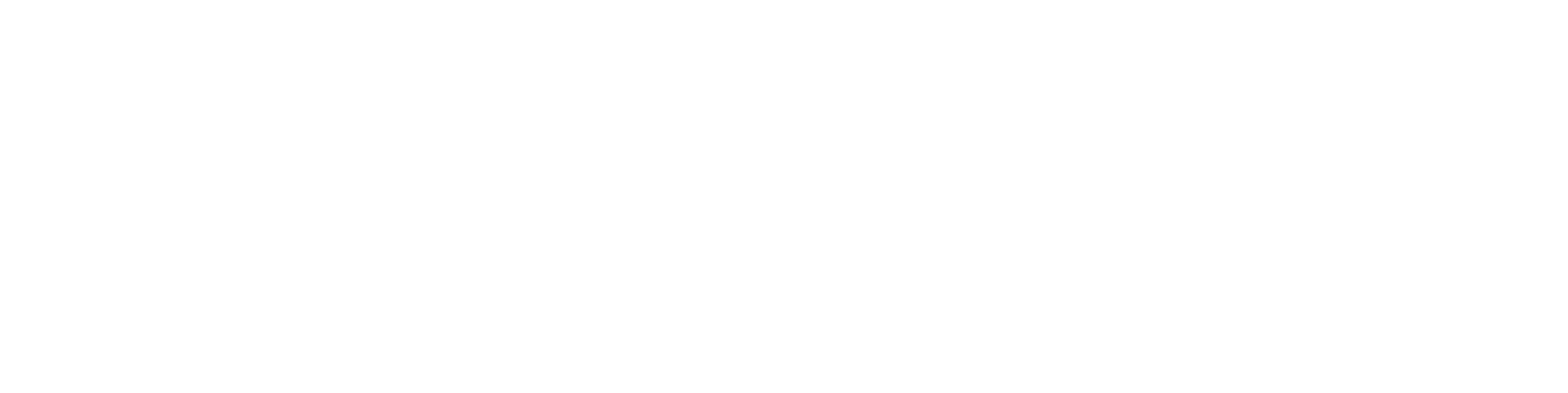
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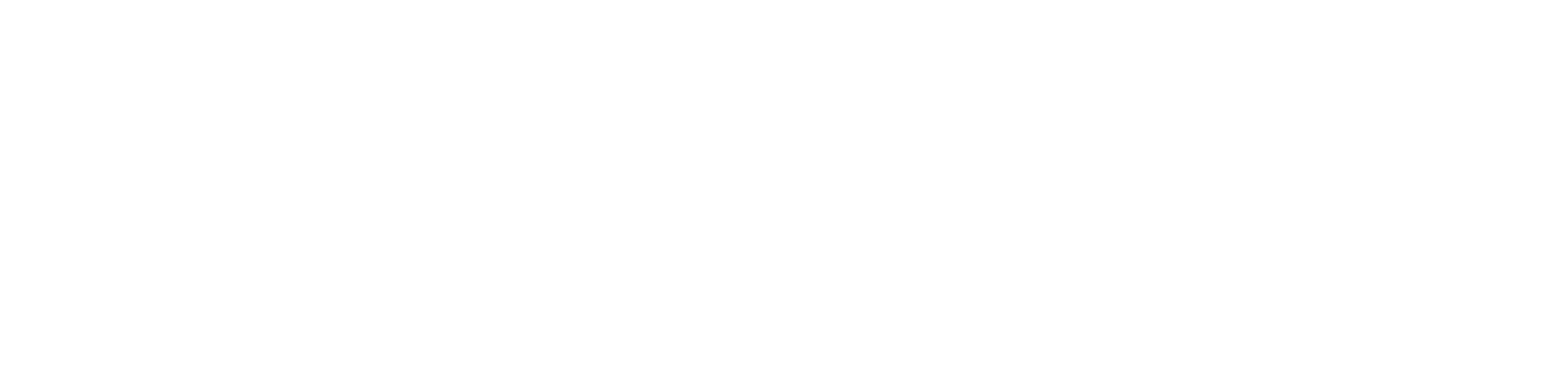
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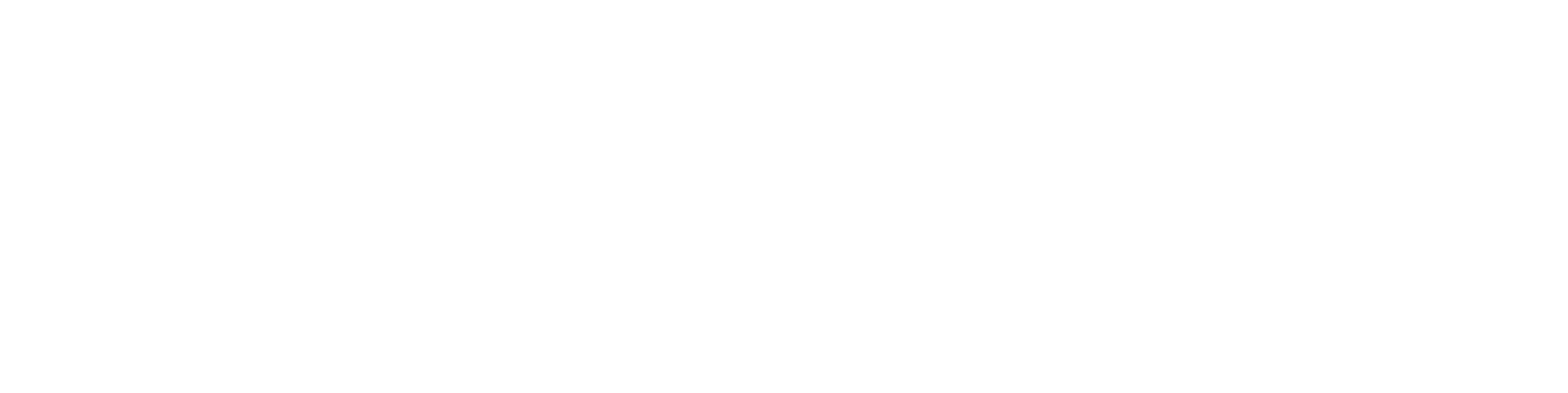
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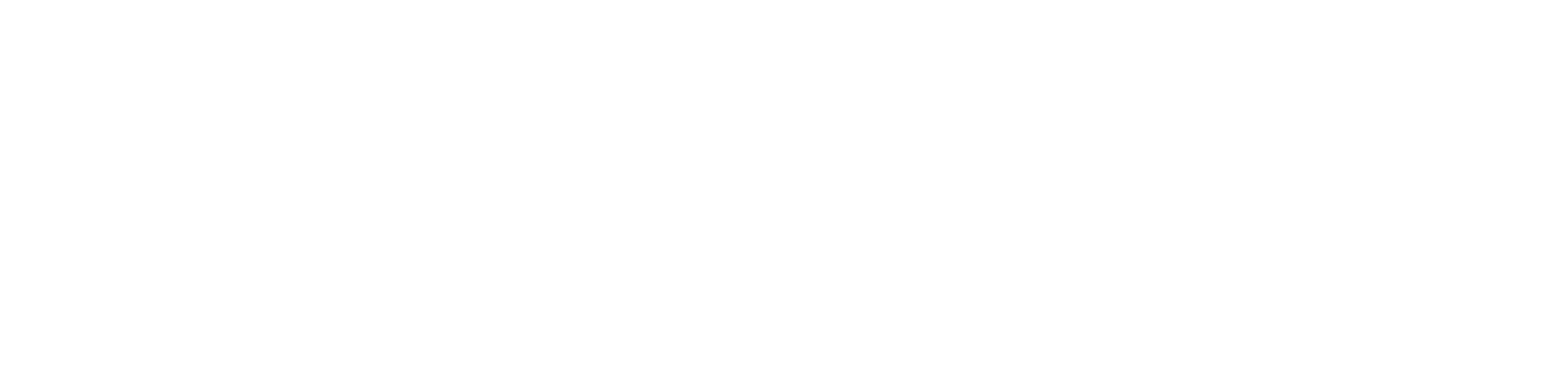
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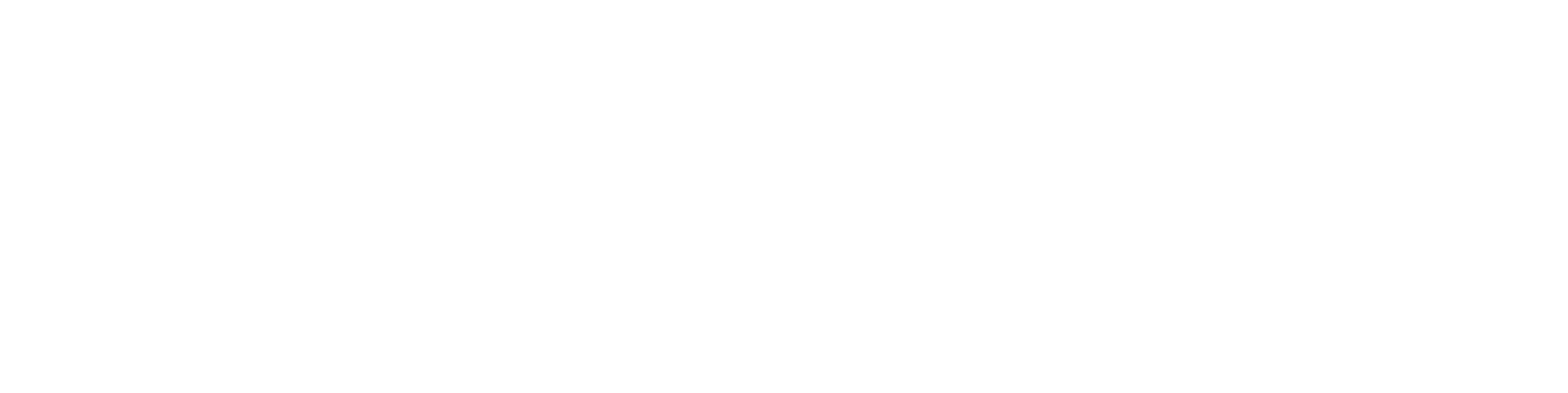
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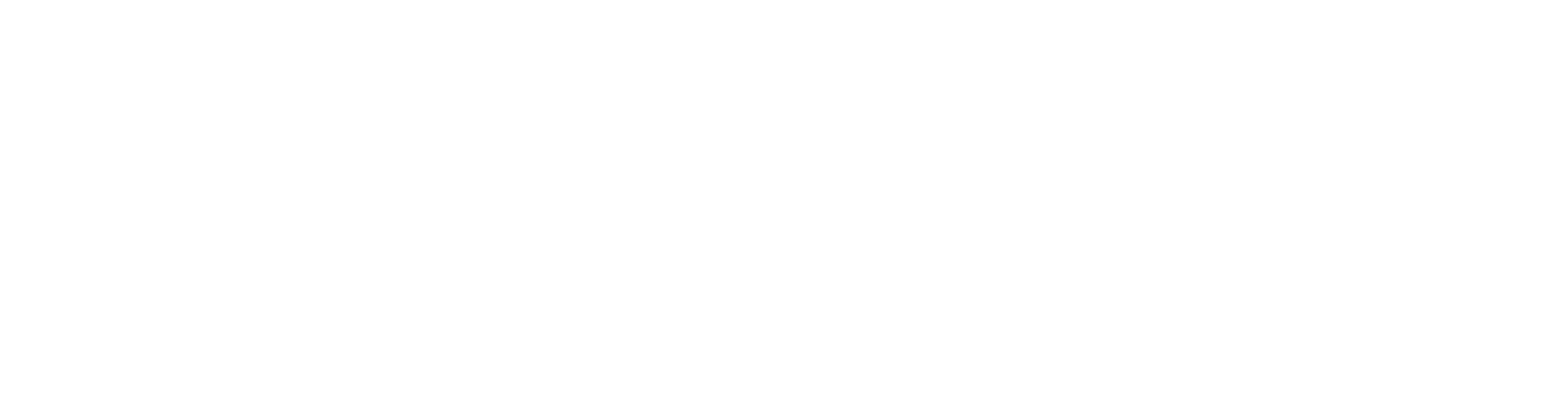
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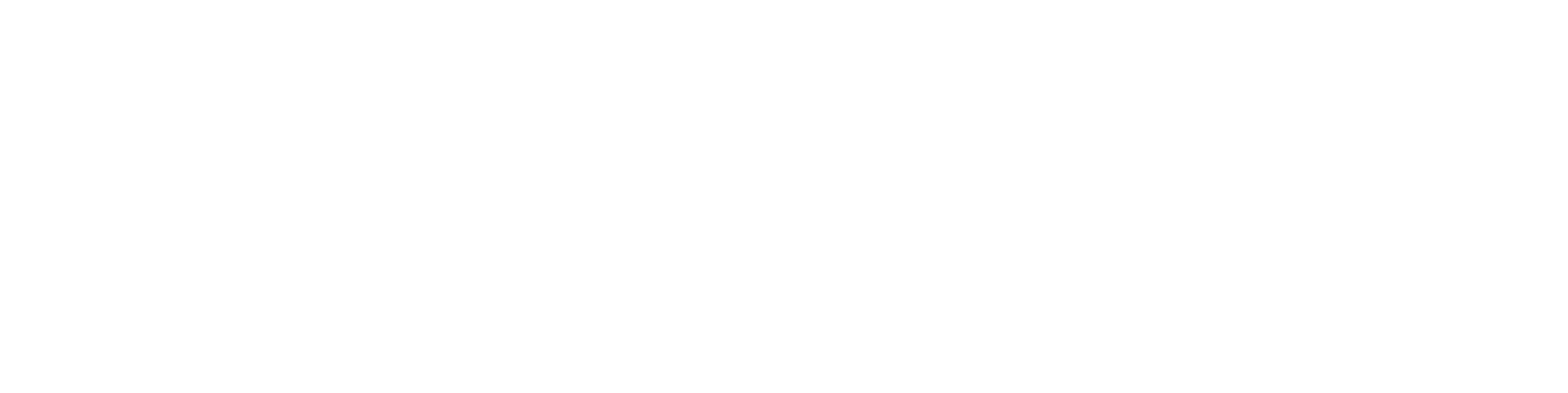
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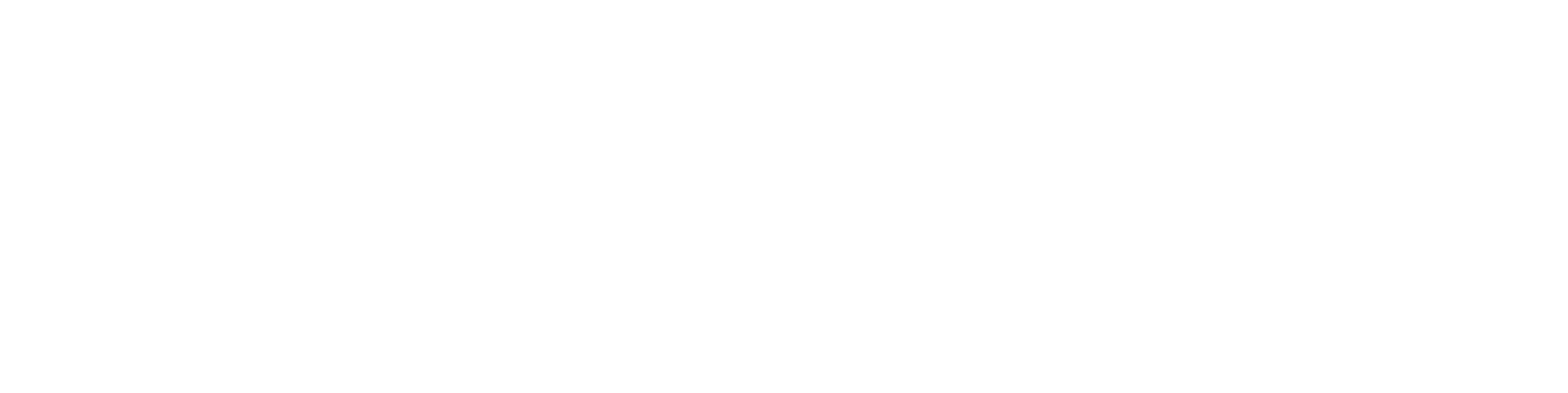
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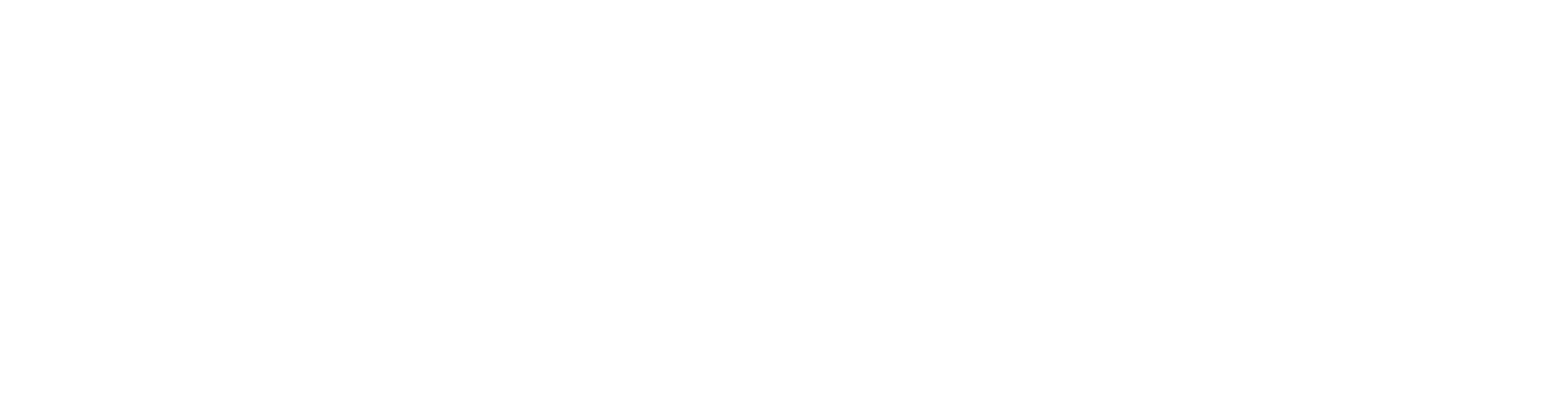
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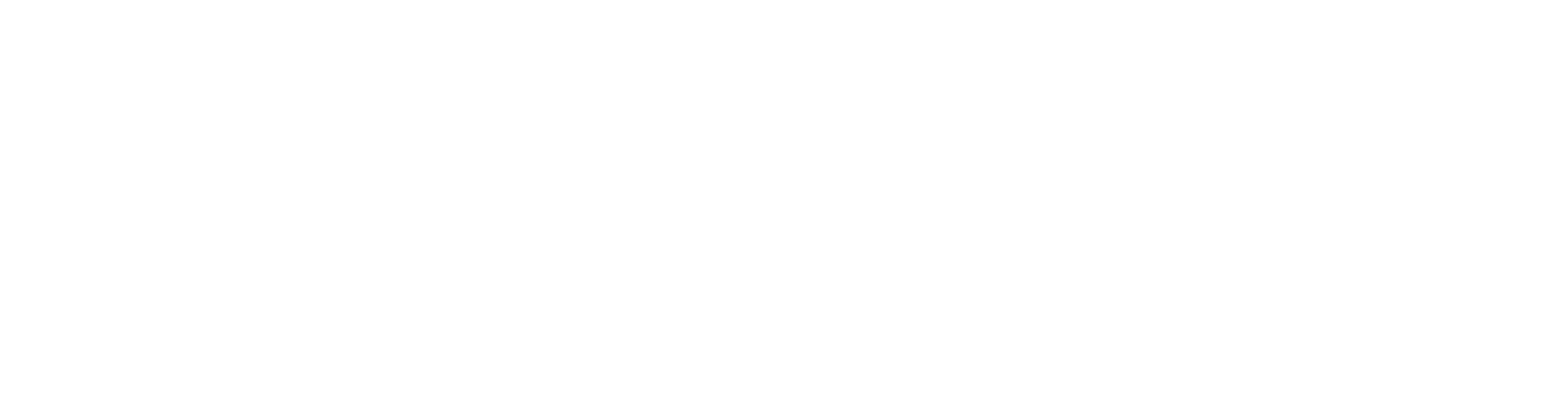
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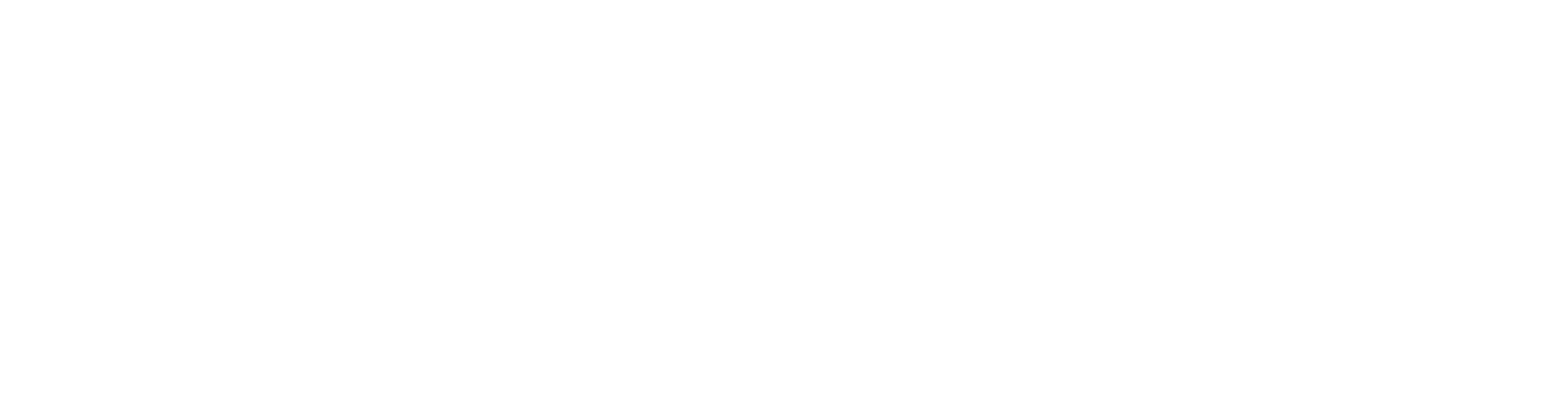
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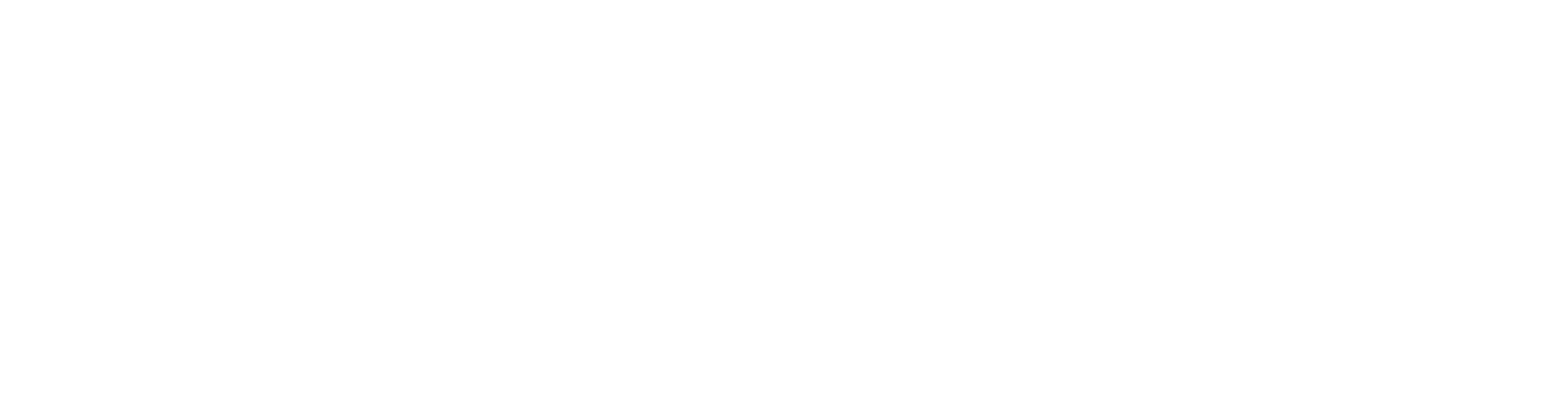
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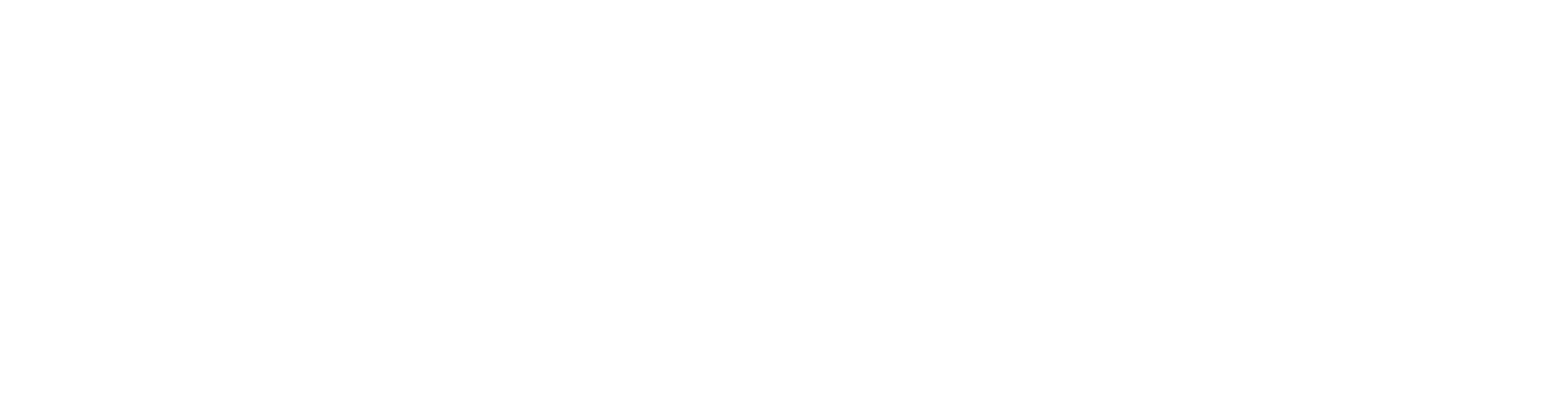
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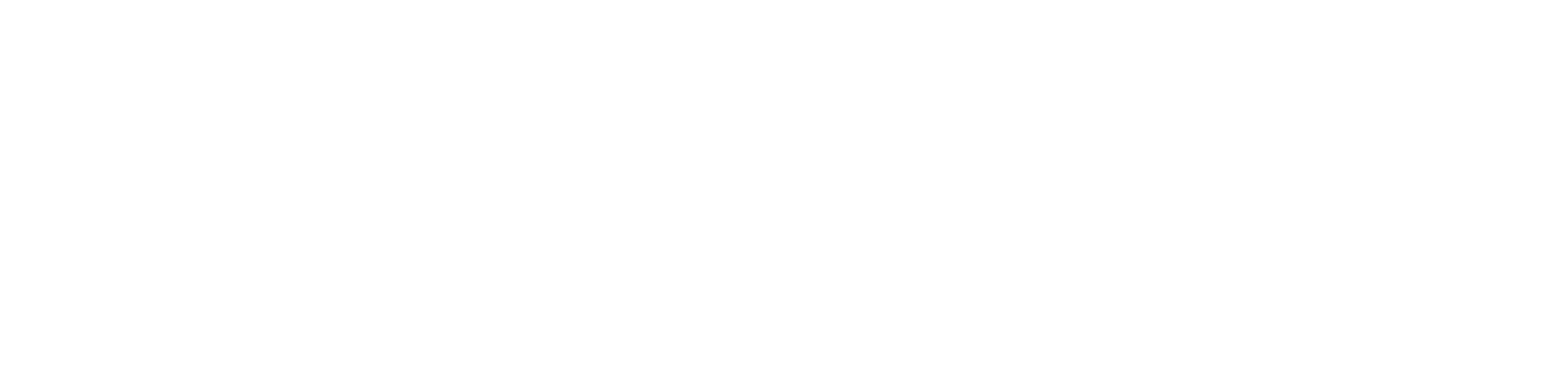
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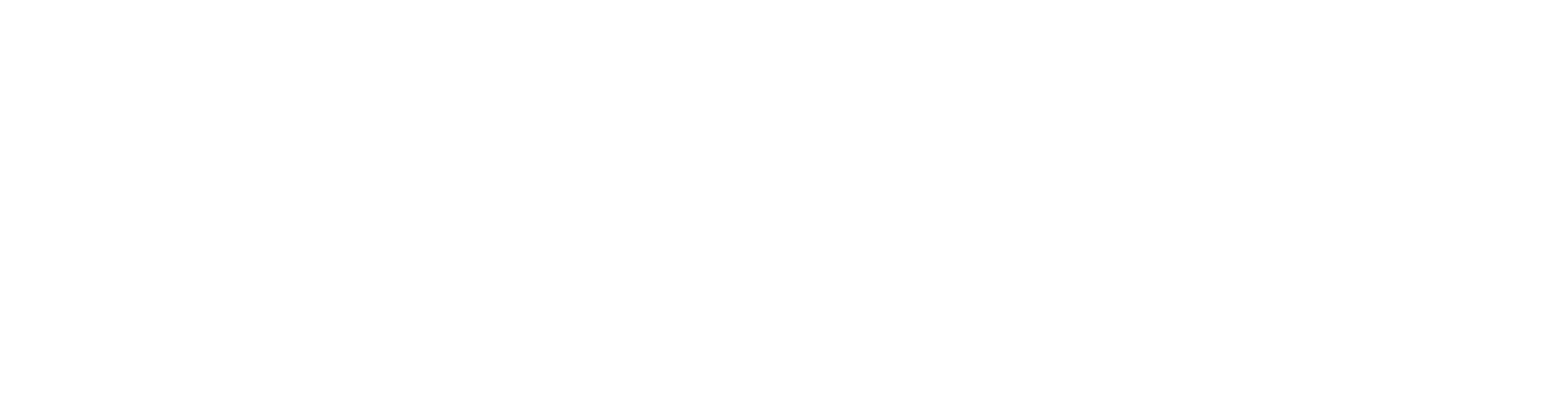
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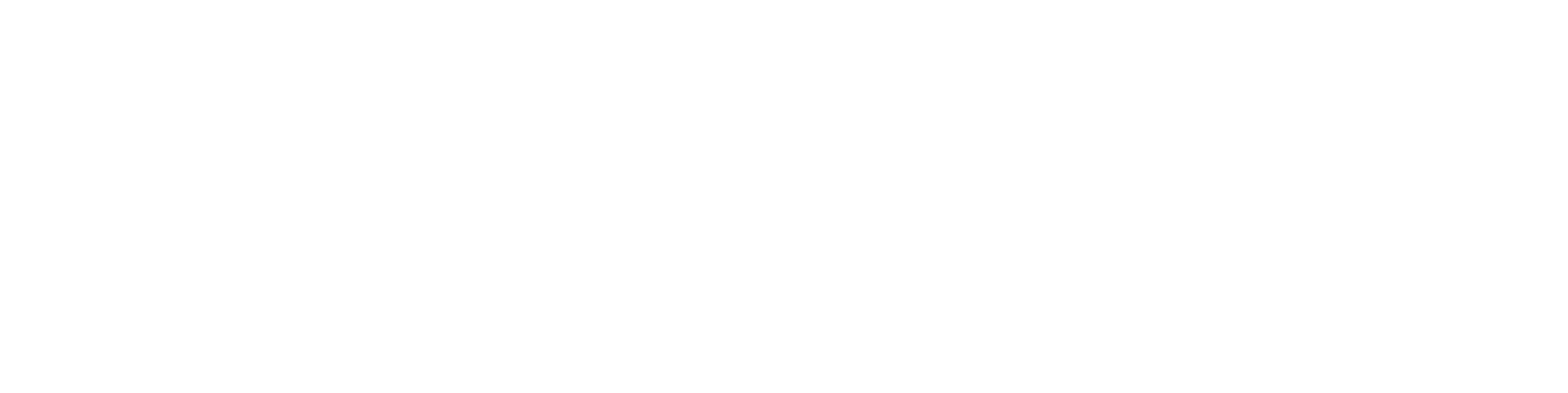
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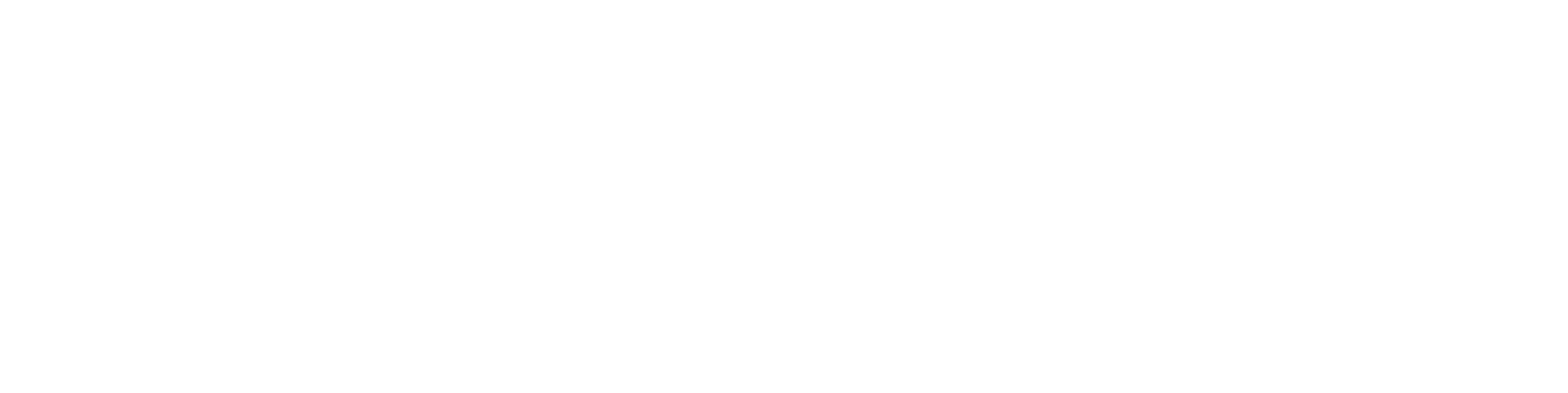
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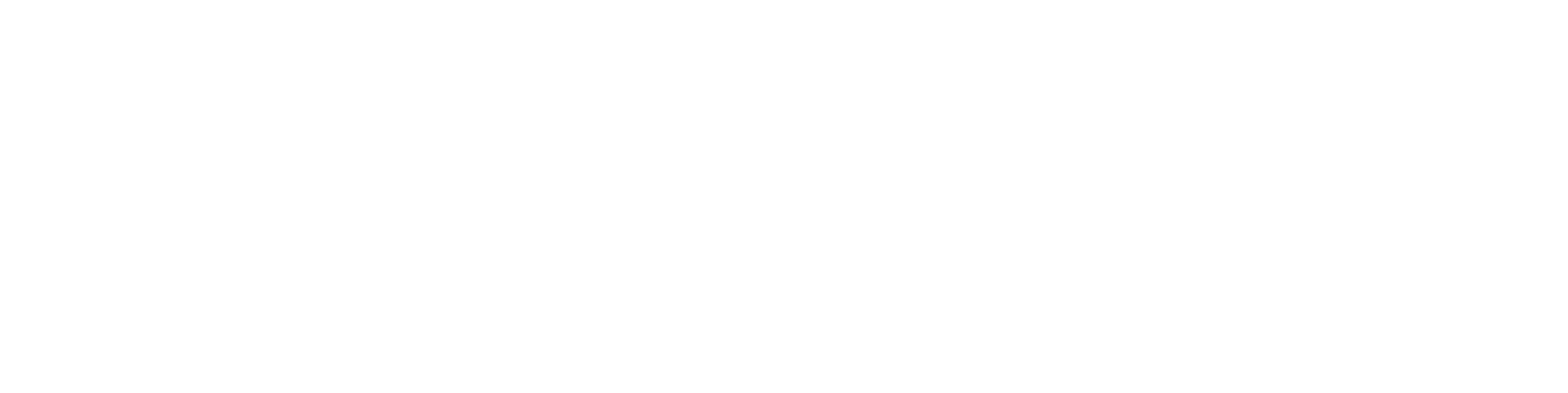
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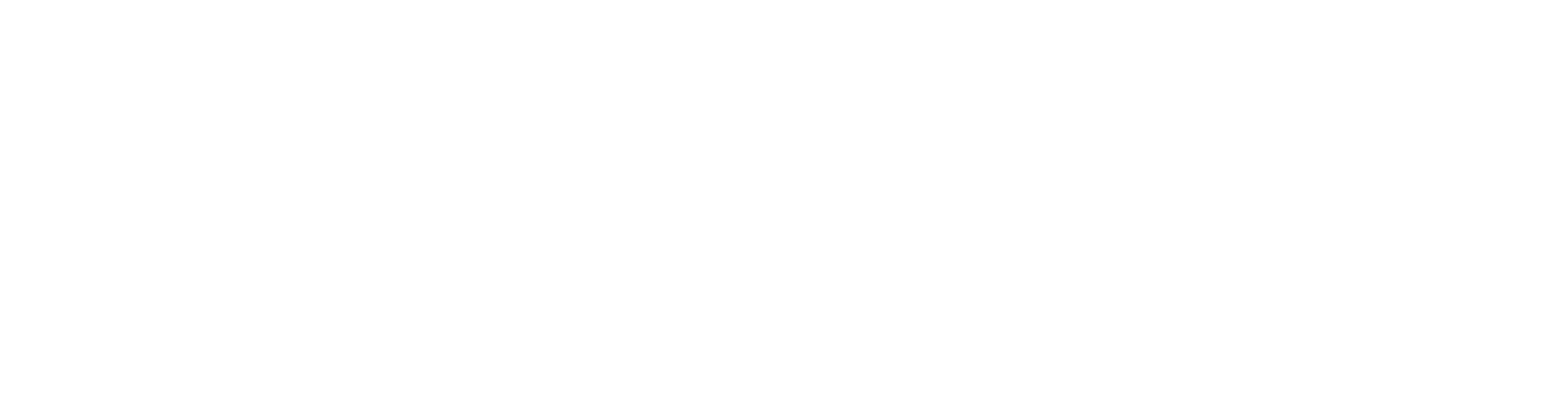
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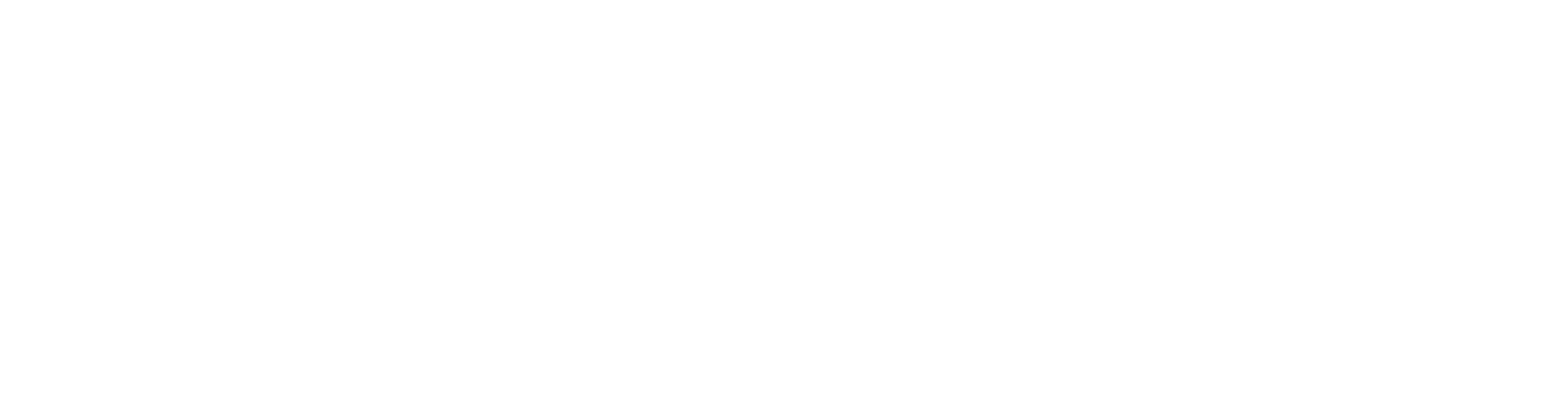
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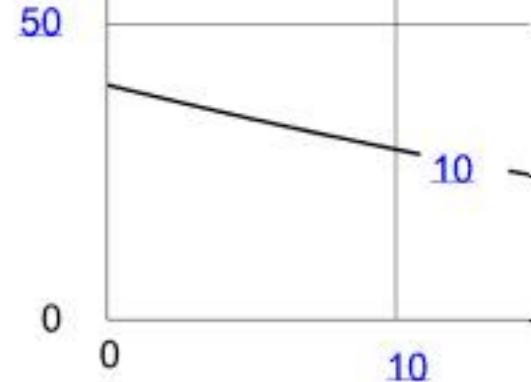
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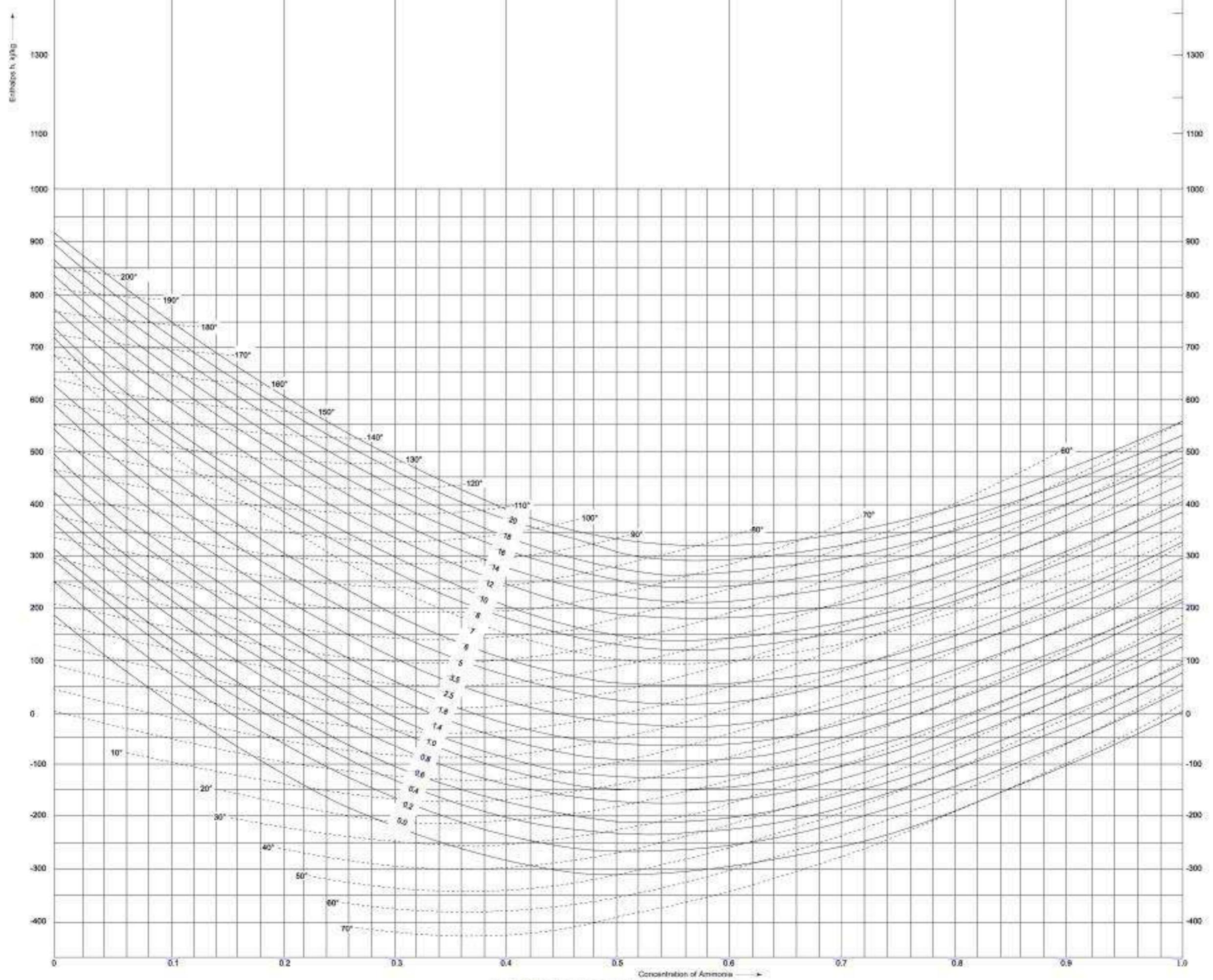
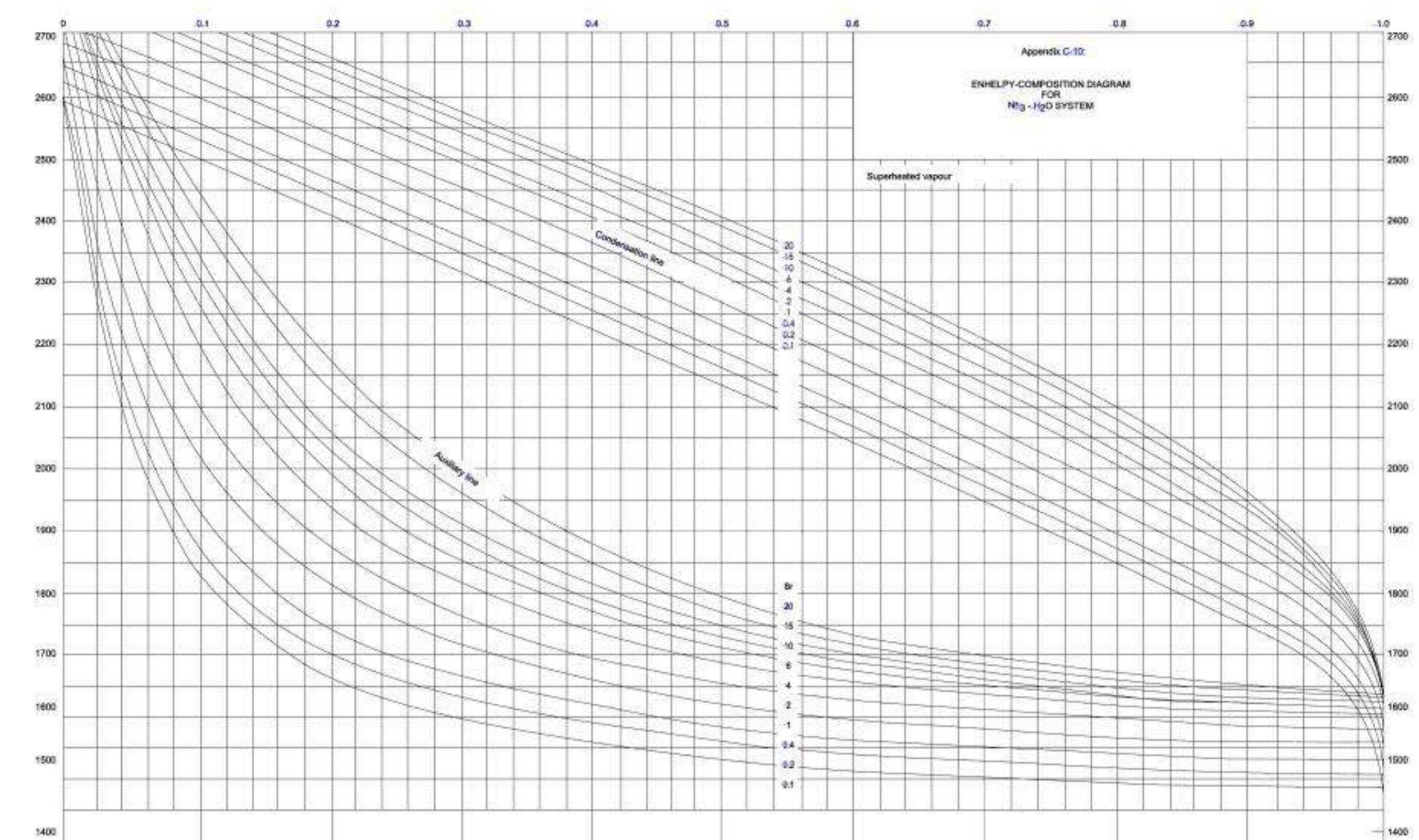


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Appendix C-10: Enthalpy-Composition Diagram for $\text{NH}_3\text{-H}_2\text{O}$ System

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