

Department of Mechanical Engineering

Course File

AUTOMATION IN MANUFACTURING SYSTEM
(Course Code: ME621PE)

III B.Tech II Semester

2023-24

Mr.K.VEERANAJENYULU
Assistant Professor



Ananthagiri, Kodad, Telangana 508 206, India.

REFRIGERATION & AIR CONDITIONING

Check List

S.No	Name of the Format	Page No.
1	Syllabus	1
2	Timetable	3
3	Program Educational Objectives	4
4	Program Objectives	4
5	Course Objectives	5
6	Course Outcomes	5
7	Guidelines to study the course	6
8	Course Schedule	7
9	Course Plan	10
10	Unit Plan	14
11	Lesson Plan	19
12	Assignment Sheets	41
13	Tutorial Sheets	46
14	Evaluation Strategy	51
15	Assessment in relation to COB's and CO's	53
16	Mappings of CO's and PO's	53
17	Rubric for course	55
18	Mid-I and Mid-II question papers	56
19	Mid-I mark	60
20	Mid-II mark	61
21	Sample answer scripts and Assignments	62
22	Course materials like Notes, PPT's, etc.	63

Int. Marks:25 Ext. Marks:75 Total Marks:100

B.Tech ME III Year II-Semester

**L T P C
3 1 0 4**

(ME621PE) REFRIGERATION & AIR CONDITIONING

UNIT – I:

Introduction to Refrigeration: Necessity and Applications, Carnot Refrigerator, Unit of Refrigeration, COP, Different Refrigeration Methods, Air Refrigeration: Bell-Coleman Cycle, Ideal and Actual Cycles, Open and Dense Air Systems - Refrigeration Needs of Air Crafts, Application of Air Refrigeration, Types of systems – Problems.

UNIT – II:

Vapour Compression Refrigeration (VCR) System: Basic cycle- Working principle and essential components of the plant – COP – Representation of cycle on T-S and p-h charts – effect of sub cooling and super heating – cycle analysis – Actual cycle - numerical problems.

Refrigerants: Desirable Properties – Classification of Refrigerants Used – Nomenclature-Secondary Refrigerants- Lubricants – Ozone Depletion – Global Warming- Newer Refrigerants.

UNIT – III:

Vapour Absorption Refrigeration (VAR) System: Description and working of NH₃ – water system and Li Br – water (Two shell & four shell) system - Calculation of max COP. Principle and operation of Three Fluid absorption refrigeration system.

Steam Jet Refrigeration System: Working Principle and Basic Components, Principle and operation of i) Thermo-Electric Refrigerator, ii) Vortex Tube or Hilsch tube.

UNIT – IV:

Introduction to Air Conditioning: Psychrometric Properties & Processes – Characterization of Sensible and Latent Heat Loads — Need For Ventilation, Consideration of Infiltrated Air – Heat Load Concepts.

Air Conditioning Systems: Air Cooler (Evaporative Cooling), Window, Split, summer, winter, Year Round, Central Air Conditioning Systems.

UNIT V:

Human Comfort: Requirements of Temperature, Humidity and Concept of Effective Temperature, Comfort Chart. Heat Pump – Heat Sources – Different Heat Pump Circuits. Air Conditioning Equipment: Humidifiers – Dehumidifiers – Air Filters, Fans and Blowers.

Text Book:

1. Refrigeration and Air Conditioning-CP Arora-TMH.
2. A Course in Refrigeration and Air conditioning-SC Arora & Domkundwar-Dhanpatrai
3. Refrigeration and Air Conditioning- Manohar Prasad -New Age.

References:

1. Principles of Refrigeration – Dossat- Pearson Education.
2. Refrigeration and Air Conditioning - P.L. Ballaney
3. Basic Refrigeration and Air Conditioning –P.N. Ananthanarayanan-TMH.

Timetable

III B.Tech. II Semester – R&AC

Day/Hour	9.40-10.20	10.20-11.10	11.20-12.10	12.10-01.00	01.40-02.25	02.25-03.10	03.15-04.00
Monday						R&AC	
Tuesday					R&AC		
Wednesday						R&AC	
Thursday							
Friday			R&AC				
Saturday					R&AC		

Vision of the Institute

To be a premier Institute in the country and region for the study of Engineering, Technology and Management by maintaining high academic standards which promotes the analytical thinking and independent judgment among the prime stakeholders, enabling them to function responsibly in the globalized society.

Mission of the Institute

To be a world-class Institute, achieving excellence in teaching, research and consultancy in cutting-edge Technologies and be in the service of society in promoting continued education in Engineering, Technology and Management.

Quality Policy

To ensure high standards in imparting professional education by providing world-class infrastructure, top-quality-faculty and decent work culture to sculpt the students into Socially Responsible Professionals through creative team-work, innovation and research

Vision of the Department

To equip the Mechanical Engineering students with the best analytical skills in the state of the latest technologies and the best communication skills to meet the Mechanical Engineering manpower requirement both nationally and internationally to responds to the demands of the market which are dynamic in nature.

Mission of the Department

To equip the Mechanical Engineering students with the best analytical skills in the state of the latest technologies and the best communication skills to meet the Mechanical Engineering manpower requirement both nationally and internationally to responds to the demands of the market which are dynamic in nature.

Program Educational Objectives (B.Tech. – ME)
Graduates will be able to

- PEO 1: To transcend in a professional career by acquiring knowledge in basic sciences, mathematics and mechanical engineering.**
- PEO 2: To exhibit problem solving skills on par with global requirements in industry and R&D.**
- PEO 3: To adopt the latest technologies, evolve as entrepreneurs, solving mechanical engineering problems, dealing with environmental society and ethical issues.**

Program Outcomes (B.Tech. – ME)

At the end of the Program, a graduate will have the ability to

PO 1: An ability to apply the knowledge of mathematics, science and engineering fundamentals.

PO 2: An ability to conduct Investigations using design of experiments, analysis and interpretation of data to arrive at valid conclusions.

PO 3: An ability to design mechanical engineering components and processes within economic, environmental, ethical and manufacturing constraints.

PO 4: An ability to function effectively in multidisciplinary teams.

PO 5: An ability to identify, formulates, analyze and solve Mechanical Engineering problems.

PO 6: An ability to understand professional, ethical and social responsibility.

PO 7: An ability to communicate effectively through written reports or oral presentations.

PO 8: The broad education necessary to understand the impact of engineering solutions in a global, economic, environmental, and societal context.

PO 9: An ability to recognize the need and to engage in independent and life-long learning.

PO 10: A knowledge of contemporary issues.

PO 11: An ability to use the appropriate techniques and modern engineering tools necessary for engineering practice.

PO 12: An ability to demonstrate knowledge and understanding of engineering.

COURSE OBJECTIVES

On completion of this Subject/Course the student shall be able to:

S.No	Objectives
1	To impart knowledge on working principle of refrigeration cycle.
2	To apply the principles of Thermodynamics to analyze different types of refrigeration systems.
3	To impart knowledge on different types of refrigeration systems.
4	To impart knowledge on Fundamentals of psychrometry.
5	To impart knowledge on Applications of air-conditioning and heat pumps.

COURSE OUTCOMES

The expected outcomes of the Course/Subject are:

S.No	Outcomes
1.	Ability to understand various refrigeration systems.
2.	Ability to understand the operation of various devices of VCR system.
3.	Ability to demonstrate the working of refrigeration equipment.
4.	Ability to understand various psychrometric processes.
5.	Ability to explain the air-conditioning equipment and heat pump circuits.



Signature of faculty

Note: Please refer to Bloom's Taxonomy, to know the illustrative verbs that can be used to state the outcomes.

GUIDELINES TO STUDY THE COURSE / SUBJECT

Course Design and Delivery System (CDD):

- The Course syllabus is written into number of learning objectives and outcomes.
- Every student will be given an assessment plan, criteria for assessment, scheme of evaluation and grading method.
- The Learning Process will be carried out through assessments of Knowledge, Skills and Attitude by various methods and the students will be given guidance to refer to the text books, reference books, journals, etc.

The faculty be able to –

- Understand the principles of Learning
- Understand the psychology of students
- Develop instructional objectives for a given topic
- Prepare course, unit and lesson plans
- Understand different methods of teaching and learning
- Use appropriate teaching and learning aids
- Plan and deliver lectures effectively
- Provide feedback to students using various methods of Assessments and tools of Evaluation
- Act as a guide, advisor, counselor, facilitator, motivator and not just as a teacher alone



Signature of HOD

Date:



Signature of faculty

Date:

COURSE SCHEDULE

S. No.	Description	Duration (Date)		Total No. of Periods
		From	To	
1.	UNIT – I:Introduction to Refrigeration: Necessity and Applications, Carnot Refrigerator, Unit of Refrigeration, COP, Different Refrigeration Methods, Air Refrigeration: Bell-Coleman Cycle, Ideal and Actual Cycles, Open and Dense Air Systems - Refrigeration Needs of Air Crafts, Application of Air Refrigeration, Types of systems – Problems.	16.06.2023	30.06.2023	15
2.	UNIT – II:Vapour Compression Refrigeration (VCR) System: Basic cycle- Working principle and essential components of the plant – COP – Representation of cycle on T-S and p-h charts –effect of sub cooling and super heating – cycle analysis – Actual cycle - numerical problems.Refrigerants: Desirable Properties – Classification of Refrigerants Used – Nomenclature-Secondary Refrigerants- Lubricants – Ozone Depletion – Global Warming- Newer Refrigerants.	04.07.20223	19.07.2023	15
3.	UNIT – III:Vapour Absorption Refrigeration (VAR) System: Description and working of NH ₃ – water system and Li Br – water (Two shell & four shell) system - Calculation of max COP. Principle and operation of Three Fluid absorption refrigeration system.Steam Jet Refrigeration System: Working Principle and Basic Components, Principle and operation of i) Thermo-Electric Refrigerator, ii) Vortex Tube or Hilsch tube.	24.07.2023	14.08.2023	20
4.	UNIT – IV:Introduction to Air Conditioning: Psychrometric Properties & Processes – Characterization of Sensible and Latent Heat Loads — Need For Ventilation, Consideration of Infiltrated Air – Heat Load Concepts.Air Conditioning Systems: Air Cooler (Evaporative Cooling), Window, Split, summer, winter, Year Round, Central Air Conditioning Systems.	16.08.2023	06.09.2023	10
5.	UNIT V:Human Comfort: Requirements of Temperature, Humidity and Concept of Effective Temperature, Comfort Chart. Heat Pump – Heat Sources – Different Heat Pump Circuits. Air Conditioning Equipment: Humidifiers – Dehumidifiers – Air Filters, Fans and Blowers.	08.09.2023	06.10.2023	15

The Schedule for the whole Course / Subject is:

Total No. of Instructional periods available for the course: 75 Hours

SCHEDULE OF INSTRUCTIONS - COURSE PLAN

Unit No.	Lesson No.	Date	No. of Periods	Topics / Sub-Topics	Objectives & Outcomes Nos.	References (Textbook, Journal)
I	1	22.01.2024	1	Introduction to Refrigeration	1 1	Power Plant Engineering – R.K.Rajput
	2	23.01.2024	1	Necessity and Applications	1 1	Power Plant Engineering – R.K.Rajput
	3	24.01.2024	1	Carnot Refrigerator	1 1	Power Plant Engineering – R.K.Rajput
	4	27.01.2024	1	Unit of Refrigeration	1 1	Power Plant Engineering – R.K.Rajput
	5	29.01.2024	1	COP	1 1	Power Plant Engineering – R.K.Rajput
	6	30.01.2024	1	Different Refrigeration Methods	1 1	Power Plant Engineering – R.K.Rajput
	7	31.01.2024	1	Air Refrigeration	1 1	Power Plant Engineering – R.K.Rajput
	8	02.02.2024	1	Bell-Coleman Cycle	1 1	Power Plant Engineering – R.K.Rajput
	9	03.02.2024	1	Ideal and Actual Cycles	1 1	Power Plant Engineering – R.K.Rajput
	10	05.02.2024	1	Open and Dense Air Systems	1 1	Power Plant Engineering – R.K.Rajput
	11	06.02.2024	1	Refrigeration Needs of Air Crafts	1	Power Plant Engineering – R.K.Rajput
	12	07.02.2024	1	Application of Air Refrigeration	1 1	Power Plant Engineering – R.K.Rajput
	13	09.02.2024	1	Types of systems	1 1	Power Plant Engineering – R.K.Rajput
	14	12.02.2024	1	Problems	1	Power Plant Engineering – R.K.Rajput
	15	13.02.2024	1	Problems	1	Power Plant Engineering – R.K.Rajput
	16	14.02.2024	1	Vapour Compression Refrigeration (VCR)	2 2	Power Plant Engineering

II				System		– R.K.Rajput
	17	16.02.2024	1	Basic cycle- Working principle and essential components of the plant	2 2	Power Plant Engineering – R.K.Rajput
	18	17.02.2024	1	COP	2 2	Power Plant Engineering – R.K.Rajput
	19	19.02.2024	1	Representation of cycle on T-S and p-h charts	2 2	Power Plant Engineering – R.K.Rajput
	20	20.2.2024	1	effect of sub cooling and super heating	2 2	Power Plant Engineering – R.K.Rajput
	21	21.02.2024	1	cycle analysis	2 2	Power Plant Engineering – R.K.Rajput
	22	23.02.2024	1	Actual cycle	2 2	Power Plant Engineering – R.K.Rajput
	23	24.02.2024	1	numerical problems	2 2	Power Plant Engineering – R.K.Rajput
	24	26.02.2024	1	Refrigerants: Desirable Properties	2 2	Power Plant Engineering – R.K.Rajput
	25	27.02.2024	1	Classification of Refrigerants Used	2 2	Power Plant Engineering – R.K.Rajput
	26	28.02.2024	1	Nomenclature	2 2	Power Plant Engineering – R.K.Rajput
	27	29.02.024	1	Secondary Refrigerants- Lubricants	2 2	Power Plant Engineering – R.K.Rajput
	28	01.03.2024	1	Ozone Depletion	2 2	Power Plant Engineering – R.K.Rajput
	29	02.03.2024	1	Global Warming	2 2	Power Plant Engineering – R.K.Rajput
	30	04.03.2024	1	Newer Refrigerants.	2 2	Power Plant Engineering – R.K.Rajput
	31	5.03.2024	1	Vapour Absorption Refrigeration (VAR) System	3 3	Power Plant Engineering – R.K.Rajput
	32	06.03.2024	1	Description and working of NH ₃	3 3	Power Plant Engineering – R.K.Rajput
	33	11.03.2024	1	water system and Li Br – water(Two	3 3	Power Plant Engineering

III				shell & four shell) system		– R.K.Rajput
	34	12.03.2024	1	Calculation of max COP	3 3	Power Plant Engineering – R.K.Rajput
	35	13.03.2024	1	Calculation of max COP	3 3	Power Plant Engineering – R.K.Rajput
	36	15.03.2024	1	Principle and operation	3 3	Power Plant Engineering – R.K.Rajput
	37	16.03.2024	1	Three Fluid absorption	3 3	Power Plant Engineering – R.K.Rajput
	38	22.03.2024	1	refrigeration system	3 3	Power Plant Engineering – R.K.Rajput
	39	23.03.2024	1	Steam Jet Refrigeration System	3 3	Power Plant Engineering – R.K.Rajput
	40	26.03.2024	1	Working Principle	3 3	Power Plant Engineering – R.K.Rajput
	41	27.03.2024	1	Basic Components	3 3	Power Plant Engineering – R.K.Rajput
	42	30.3.2024	1	Principle and operation	3 3	Power Plant Engineering – R.K.Rajput
	43	1.04.2024	1	Thermo-Electric Refrigerator	3 3	Power Plant Engineering – R.K.Rajput
	44	02.04.2024	1	Vortex Tube or Hilsch tube.	3 3	Power Plant Engineering – R.K.Rajput
	45	03.04.2024	1	Vortex Tube or Hilsch tube.	3 3	Power Plant Engineering – R.K.Rajput
		46	06.04.2024	1	Introduction to Air Conditioning	4 4
	47	08.04.2024	1	Psychrometric Properties	4 4	Power Plant Engineering – R.K.Rajput
	48	10.04.2024	1	Processes – Characterization	4 4	Power Plant Engineering – R.K.Rajput
	50	16.04.2024	1	Sensible and Latent Heat Loads	4 4	Power Plant Engineering – R.K.Rajput
	51	19.04.2024	1	Need For Ventilation	4 4	Power Plant Engineering – R.K.Rajput

IV	52	20.04.2024	1	Consideration of Infiltrated Air	4 4	Power Plant Engineering – R.K.Rajput
	53	22.04.2024	1	Heat Load Concepts.	4 4	Power Plant Engineering – R.K.Rajput
	54	23.04.2024	1	Air Conditioning Systems	4 4	Power Plant Engineering – R.K.Rajput
	55	24.04.2024	1	Air Cooler	4 4	Power Plant Engineering – R.K.Rajput
	56	26.04.2024	1	Evaporative Cooling	4 4	Power Plant Engineering – R.K.Rajput
	57	27.04.2024	1	Window AC	4 4	Power Plant Engineering – R.K.Rajput
	58	29.04.2024	1	Split AC	4 4	Power Plant Engineering – R.K.Rajput
	59	30.04.2024	2	summer, winter, Year Round,	4 4	Power Plant Engineering – R.K.Rajput
	60	30.04.2024		Central Air Conditioning Systems	4 4	Power Plant Engineering – R.K.Rajput
V	61	01.05.2024	1	Human Comfort	5 5	Power Plant Engineering – R.K.Rajput
	62	03.05.2024	1	Requirements of Temperature	5 5	Power Plant Engineering – R.K.Rajput
	63	04.05.2024	1	Humidity and Concept	5 5	Power Plant Engineering – R.K.Rajput
	64	06.05.2024	1	Effective Temperature	5 5	Power Plant Engineering – R.K.Rajput
	65	07.05.2024	1	Comfort Chart	5 5	Power Plant Engineering – R.K.Rajput
	66	08.05.2024	1	Comfort Chart	5 5	Power Plant Engineering – R.K.Rajput
	67	10.05.2024	1	Heat Pump	5 5	Power Plant Engineering – R.K.Rajput
	68	03.06.2024	1	Heat Sources	5 5	Power Plant Engineering – R.K.Rajput
	69	04.06.2024	1	Different Heat Pump Circuits	5 5	Power Plant Engineering – R.K.Rajput

	70	05.06.2024	1	Air Conditioning	5 5	Power Plant Engineering – R.K.Rajput
	71	07.06.2024	1	Equipment: Humidifiers	5 5	Power Plant Engineering – R.K.Rajput
	72	10.06.2024	1	Dehumidifiers	5 5	Power Plant Engineering – R.K.Rajput
	73	11.06.2024	1	Air Filters	5 5	Power Plant Engineering – R.K.Rajput
	74	12.06.2024	2	Air Filters	5 5	Power Plant Engineering – R.K.Rajput
	75	12.06.2024		Fans and Blowers.	5 5	Power Plant Engineering – R.K.Rajput



Signature of HOD

Date:



Signature of faculty

Date:

Note:

1. Ensure that all topics specified in the course are mentioned.
2. Additional topics covered, if any, may also be specified in bold.
3. Mention the corresponding course objective and outcome numbers against each topic.

LESSON PLAN (U-I)

Lesson No: 01-15

Duration of Lesson: 50 min

Lesson Title: Introduction to Refrigeration: Necessity and Applications, Carnot Refrigerator, Unit of Refrigeration, COP, Different Refrigeration Methods, Air Refrigeration: Bell-Coleman Cycle, Ideal and Actual Cycles, Open and Dense Air Systems - Refrigeration Needs of Air Crafts, Application of Air Refrigeration, Types of systems – Problems.

Instructional / Lesson Objectives:

At the end of unit – I, Students will know

- The types of refrigeration systems
- The necessity and applications of refrigeration
- Unit of refrigeration and C.O.P
- The Bell Coleman cycle and Brayton Cycle, Open and Dense air systems
- The Refrigeration needs of Air crafts

Teaching AIDS : PPTs, Digital Board

Time Management of Class :

5 min for taking attendance 5 for revision of previous class 35 min for the lecture delivery 5 min for doubts session
--

Assignment / Questions:

(Note: Mention for each question the relevant Objectives and Outcomes Nos.1,2,3,4 & 1,3..)

Refer assignment – I & tutorial-I sheets



Signature of faculty

LESSON PLAN (U-II)

Lesson No: 16-30

Duration of Lesson: 50min

Lesson Title : Vapour Compression Refrigeration (VCR) System: Basic cycle- Working principle and essential components of the plant – COP – Representation of cycle on T-S and p-h charts –effect of sub cooling and super heating – cycle analysis – Actual cycle - numerical problems. Refrigerants: Desirable Properties – Classification of Refrigerants Used – Nomenclature-Secondary Refrigerants- Lubricants – Ozone Depletion – Global Warming- Newer Refrigerants.

Instructional / Lesson Objectives:

At the end of Unit – II Students will know

- The working principle of simple Vapour compression refrigeration system
- The essential components of the refrigeration plant
- Representation of refrigeration cycle on T-S and p-h charts
- The effects of sub cooling and super heating on COP
- The influence of various parameters on refrigeration system performance
- Types of refrigerants
- Understand the required properties of refrigerants
- The causes of Ozone Depletion and global Warming.

Teaching AIDS : PPTs, Digital Board

Time Management of Class :

5 min for taking attendance 5 for revision of previous class 35 min for the lecture delivery 5 min for doubts session
--

Assignment / Questions:

(Note: Mention for each question the relevant Objectives and Outcomes Nos.1,2,3,4 & 1,3..)

Refer assignment – I & tutorial-I sheets



Signature of faculty

LESSON PLAN (U-III)

Lesson No: 31-50

Duration of Lesson: 50 Mins

Lesson Title: Vapour Absorption Refrigeration (VAR) System: Description and working of NH₃ – water system and Li Br – water (Two shell & four shell) system - Calculation of max COP. Principle and operation of Three Fluid absorption refrigeration system. Steam Jet Refrigeration System: Working Principle and Basic Components, Principle and operation of i) Thermo-Electric Refrigerator, ii) Vortex Tube or Hilsch tube.

Instructional / Lesson Objectives:

- At the end of Unit – III Students will know
- The Vapor Absorption System
 - Max COP
 - The working principles of NH₃ – water system
 - The working principles of Li Br –water (Two shell & Four shell) System.
 - The principle of operation of Three Fluid absorption system, salient features.
 - Electrolux refrigerator
 - The Steam Jet Refrigeration System – Working Principle and Basic Components
 - The working Principle and operation of Thermoelectric refrigerator
 - The working Principle and operation of Vortex tube or Hilsch tube

Teaching AIDS : PPTs, Digital Board

Time Management of Class :

5 min for taking attendance 5 for revision of previous class 35 min for the lecture delivery 5 min for doubts session
--

Assignment / Questions:

(Note: Mention for each question the relevant Objectives and Outcomes Nos.1,2,3,4 & 1,3..)

Refer assignment-II & tutorial-II sheets.



Signature of faculty

LESSON PLAN (U-IV)

Lesson No: 51-60

Duration of Lesson: 50 Mins

Lesson Title: Introduction to Air Conditioning: Psychrometric Properties & Processes – Characterization of Sensible and Latent Heat Loads — Need For Ventilation, Consideration of Infiltrated Air – Heat Load Concepts. Air Conditioning Systems: Air Cooler (Evaporative Cooling), Window, Split, summer, winter, Year Round, Central Air Conditioning Systems.

Instructional / Lesson Objectives:

- At the end of Unit – IV Students will know
- Acquainted with Psychrometric Properties & Processes
 - Able to identify Sensible and latent heat loads
 - the need for Ventilation, Consideration of Infiltration
 - The Load concepts of RSHF, GSHF- Problems, Concept of ESHF and ADP.
 - the Air conditioning Load Calculations

Teaching AIDS : PPTs, Digital Board

Time Management of Class :

5 min for taking attendance 5 for revision of previous class 35 min for the lecture delivery 5 min for doubts session
--

Assignment / Questions:

(Note: Mention for each question the relevant Objectives and Outcomes Nos.1,2,3,4 & 1,3..)

Refer assignment-II & tutorial-II sheets.



Signature of faculty

LESSON PLAN (U-V)

Lesson No: 61-75

Duration of Lesson: 50 Mins

Lesson Title: Requirements of Temperature, Humidity and Concept of Effective Temperature, Comfort Chart. Heat Pump – Heat Sources – Different Heat Pump Circuits. Air Conditioning Equipment: Humidifiers – Dehumidifiers – Air Filters, Fans and Blowers.

Instructional / Lesson Objectives:

- At the end of Unit – V Students will know
- Understand the Air Conditioning systems
 - The AC equipment classification
 - Understand the cooling, heating humidification and dehumidification,
 - Know the working principle of filters, grills and registers, fans and blowers.
 - Know various heat sources – different heat pump circuits.
 - Understand the requirements of human comfort
 - the concept of effective temperature- Comfort chart –Comfort Air conditioning
 - the requirements of Industrial air conditioning

Teaching AIDS : PPTs, Digital Board

Time Management of Class :

5 min for taking attendance 5 for revision of previous class 35 min for the lecture delivery 5 min for doubts session
--

Assignment / Questions:

(Note: Mention for each question the relevant Objectives and Outcomes Nos.1,2,3,4 & 1,3..)

Refer assignment-II & tutorial-II sheets.



Signature of faculty

ASSIGNMENT – 1

This Assignment corresponds to Unit No. 1

Question No.	Question	Objective No.	Outcome No.
1	Define refrigeration and Tonne of refrigeration. Explain the working principle of the Bell-Coleman refrigeration cycle with schematic-V and T-S diagrams.	1	1
2	What is the necessity of aircraft refrigeration and list out the methods of air refrigeration? Briefly explain the working of a simple air cooling system with schematic and T-S diagrams.	1	1
3	Describe the Open and Dense air refrigeration systems.	1	1



Signature of HOD

Date:



Signature of faculty

Date:

ASSIGNMENT – 2

This Assignment corresponds to Unit No. 2

Question No.	Question	Objective No.	Outcome No.
1	With the help of a schematic diagram, explain the working of simple VCRS. Also draw T-S and P-H diagrams, State its advantages and disadvantages over air refrigeration.	2	2
2	What is a refrigerant? Classify the refrigerants and list out the desirable properties of an ideal refrigerant.	2	2
3	Describe the causes of Ozone Depletion and global Warming.	2	2



Signature of HOD

Date:



Signature of faculty

Date:

ASSIGNMENT – 3

This Assignment corresponds to Unit No. 3

Question No.	Question	Objective No.	Outcome No.
1	With the help of a neat sketch, explain the working principle of a practical VAR system and state its advantages over a VCR system.	3	3
2	Explain the working principle of the three fluid refrigeration (Electrolux) systems with a schematic diagram. Also explain the Li-Br absorption refrigeration system.	3	3
3	Explain the working Principle and operation of Vortex tube or Hilsch tube.	3	3



Signature of HOD

Date:



Signature of faculty

Date:

ASSIGNMENT – 4

This Assignment corresponds to Unit No. 4

Question No.	Question	Objective No.	Outcome No.
1	What is psychrometry? Explain all Psychrometric properties in detail.	4	4
2	What is the importance of the Psychrometric chart and briefly explain the Psychrometric chart with neat sketch.	4	4
3	Describe the concept of effective temperature- Comfort chart – Comfort Air conditioning.	4	4



Signature of HOD

Date:



Signature of faculty

Date:

ASSIGNMENT – 5

This Assignment corresponds to Unit No. 5

Question No.	Question	Objective No.	Outcome No.
1	Briefly explain the factors that affect human comfort. Explain the comfort chart with a neat diagram.	5	5
2	What is the function of a heat pump and classify the heat pumps. With the help of a neat sketch, explain any two heat pump circuits.	5	5
3	Write about various heat sources and different heat pump circuits.	5	5



Signature of HOD

Date:



Signature of faculty

Date:

TUTORIAL – 1

This tutorial corresponds to Unit No. 1 (Objective Nos.: 1, Outcome Nos.: 1)

1. There are two cooling turbines in

- (a) Reduced ambient system of Refrigeration
- (b) Boot-strap system of Refrigeration
- (c) Regenerative system of Refrigeration
- (d) Boot strap evaporative system of Refrigeration

2. There are two stages of compression of air in

- (a) Reduced ambient system of Refrigeration
- (b) Boot-strap system of Refrigeration
- (c) Regenerative system of Refrigeration
- (d) Simple evaporative system of Refrigeration

3. The amount of heat absorbed by the system at low temperature is

- (a) COP
- (b) refrigerating effect
- (c) work done on the system
- (d) refrigeration efficiency



Signature of HOD

Date:



Signature of faculty

Date:

TUTORIAL – 2

This tutorial corresponds to Unit No. 2 (Objective Nos.: 2, Outcome Nos.: 2)

1. Heat is rejected by the refrigerant, during vapour compression refrigeration cycle in

- (a) Compressor (b) Condenser
(c) Evaporator (d) Throttle valve

2. In a simple saturated vapour compression cycle, the refrigerant is in superheated condition

- (a) Before entering compressor (b) before entering condenser
(c) Before entering throttle valve (d) before entering evaporator

3. Sub-cooling is a process of cooling the refrigerant in vapour compression refrigeration system

- (a) After compression (b) before compression
(c) Before Throttling (d) none of the above



Signature of HOD

Date:



Signature of faculty

Date:

TUTORIAL SHEET – 3

This tutorial corresponds to Unit No. 3 (Objective Nos.: 3, Outcome Nos.: 3)

1. The function of compressor in vapour compression refrigeration system is performed in vapour absorption system by

- (A) Generator (b) absorber
(c) Generator, absorber and liquid pump (d) absorber and liquid pump

2. Electrolux refrigerators has the following working substances

- (a)Hydrogen (b) Ammonia and Hydrogen
(c) Ammonia and water (d) Ammonia, hydrogen and water

3. Which of the following system can be called as mechanical system of refrigeration

- (a)Vapour absorption system (b) Vapour compression system
(c) Steam jet refrigeration system (d) None of the above



Signature of HOD

Date:



Signature of faculty

Date:

TUTORIAL – 4

This tutorial corresponds to Unit No. 4 (Objective Nos.: 4, Outcome Nos.: 4)

1. In a Psychrometric chart, the vertical lines parallel to the ordinate indicate

- a) Dry bulb Temperature
- b) Wet bulb Temperature
- c) Specific humidity
- d) Enthalpy of saturation

2. When moisture is added to air at constant dry bulb temperature, the process is known as

- a) Sensible cooling
- b) humidification
- c) Dehumidification
- d) Chilling

3. Which of the following can be measured by a sling psychrometer?

- a) Wet bulb temperature
- b) Dry bulb as well as wet bulb temperatures
- c) Specific humidity
- d) Absolute humidity



Signature of HOD

Date:



Signature of faculty

Date:

TUTORIAL – 5

This tutorial corresponds to Unit No. 5 (Objective Nos.: 5, Outcome Nos.: 5)

1. Which of the following statements are TRUE?

- a) As the amount of clothing increases, the surrounding DBT should be increased to maintain thermal comfort
- b) As the amount of clothing increases, the surrounding DBT should be decreased to maintain thermal comfort
- c) As the activity level increases, DBT of air should be increased to maintain thermal comfort
- d) As the activity level increases, DBT of air should be decreased to maintain thermal comfort

2. In ‘comfort chart’ the effective temperature is represented by _____ lines.

3. Which of the following statements are TRUE?

- a) Effective temperature combines the affects of dry bulb temperature and air velocity into a single index
- b) Effective temperature combines the affects of dry bulb temperature and wet bulb temperature into a single index
- c) Mean radiant temperature combines the affects of dry bulb temperature and surrounding surface temperature into a single index
- d) Operative temperature combines the affects of dry bulb temperature and mean radiant temperature into a single index



Signature of HOD

Date:



Signature of faculty

Date:

EVALUATION STRATEGY

Target (s)

- a. Percentage of Pass : 95%

Assessment Method (s) (Maximum Marks for evaluation are defined in the Academic Regulations)

- a. Daily Attendance
- b. Assignments
- c. Online Quiz (or) Seminars
- d. Continuous Internal Assessment
- e. Semester / End Examination

List out any new topic(s) or any innovation you would like to introduce in teaching the subjects in this semester

Case Study of any one existing application



Signature of HOD

Date:



Signature of faculty

Date:

COURSE COMPLETION STATUS

Actual Date of Completion & Remarks if any

Units	Remarks	Objective No. Achieved	Outcome No. Achieved
Unit 1	completed on 30.06.2023	1	1
Unit 2	completed on 19.07.2023	2	2
Unit 3	completed on 14.08.2023	3	3
Unit 4	completed on 06.09.2023	4	4
Unit 5	completed on 24.09.2023	5	5



Signature of HOD

Date:



Signature of faculty

Date:

Mappings

1. Course Objectives-Course Outcomes Relationship Matrix

(Indicate the relationships by mark "X")

Course-Objectives \ Course-Outcomes	1	2	3	4	5
1	H				
2		H			
3			H		
4				H	
5					H

2. Course Outcomes-Program Outcomes (POs) & PSOs Relationship Matrix

(Indicate the relationships by mark "X")

P-Outcomes \ C-Outcomes	a	b	c	d	e	f	g	h	i	j	k	l	PSO 1	PSO 2
1	H	H			M								H	
2	H	M	H			M		M			M		H	H
3	M		H		H				M					M
4	M	H							M		H		M	
5	H		M		H									M

III B.TECH VI SEMESTER I MID EXAMINATIONS - MARCH 2024

Branch : B.Tech. (ME)

**Subject : REFRIGERATION & AIR
CONDITION**

Max. Marks : 20M

Date : 21.03.2024 FN

Time : 90 Minutes

PART - A

ANSWER ALL THE QUESTIONS.

5 X 1M = 5M

Q.No	Question	CO	BTL
1.	What is the difference between a refrigerator and a heat pump?	CO1	2
2.	Differentiate the open and closed air refrigeration cycles.	CO1	2
3.	What are the advantages of VCR system over air refrigeration system?	CO2	2
4.	Draw P-H and T-S diagrams for Simple vapour refrigeration cycle.	CO2	2
5.	What is the function of the Absorber in VAR system?	CO3	2

PART - B

ANSWER ALL THE QUESTIONS.

3 X 5M = 15M

Q.No	Question	CO	BTL
6.	A boot strap cooling system of 10 TR capacity is used in an aeroplane. The ambient air temperature and pressure are 20C and 0.85 bar respectively. The pressure of air increases from 0.85 bar to 1 bar due to ramming action of air. The pressure of air discharged from the main compressor is 3 bar. The discharge pressure of air from the auxiliary compressors is 4 bar. The isentropic efficiency of each of the compressor is 80%, while that of turbine is 85%. 50% of enthalpy of air discharged from the main compressor is removed in the first heat exchanger and 30% of the enthalpy of air discharged from the auxiliary compressor is removed in the second heat exchanger using rammed air. Assuming ramming action to be isentropic, the required cabin pressure of 0.9 bar and temperature of the air leaving the cabin not more than 20C. Find: i) The power required to operate the system ii) The C.O.P of the system.	CO1	4
OR			
7.	A refrigerator working on the Bell-Coleman cycle operates between pressure limits of 1.05 bar and 8.5 bar. Air is drawn from the cold chamber at 10C, compressed and then it is cooled to 30C before entering the expansion cylinder. The expansion and compression follows the law $p v^{1.3} = \text{constant}$. Determine the theoretical COP of the system.	CO1	3
8.	Sketch and explain the T-S and P-H diagram for the vapour compression refrigeration system when: i) Cycle with saturated vapour after compression; ii) Cycle with superheated vapour after compression	CO2	3
OR			
9.	Draw the line diagram of a actual VCR cycle and explain its working.	CO2	3
10.	Draw a neat sketch of a practical VAR cycle and explain its working.	CO3	3



III B.TECH VI SEMESTER II MID EXAMINATIONS - JUNE 2024

Branch : B.Tech. (ME)

Max. Marks : 20M

Date : 20-Jun-2024 Session : Morning

Time : 90 Min

Subject : REFRIGERATION & AIR CONDITIONING,ME621PE

PART - A

ANSWER ALL THE QUESTIONS

5 X 1M = 5M

Q.No	Question	CO	BTL
1.	What is the function of an expansion device in a refrigeration system ?	CO3	1
2.	Define the following : a) Dry bulb temperature b) Wet bulb temperature	CO4	2
3.	Define the following : a) Specific Humidity b) Absolute Humidity	CO4	2
4.	Define by-pass factor of the cooling or heating coil.	CO5	2
5.	List out the equipment used in air conditioning plant.	CO5	1

PART - B

ANSWER ALL THE QUESTIONS

3 X 5M = 15M

Q.No	Question	CO	BTL
6.	With the help of a neat sketch, explain the steam jet refrigeration system.	CO3	3
OR			
7.	List out the classification of evaporators and explain any one with schematic diagram.	CO3	2
8.	Explain the following with the help of a neat sketch : a) Summer air conditioning system b) Winter air conditioning system	CO4	3
OR			
9.	A sling psychrometer reads 40C DBT and 28C WBT, the air is cooled at 20C DBT without changing its humidity. Calculate the following : a) Specific humidity b) Relative humidity c) DPT d) Enthalpy e) Specific volume	CO4	4
10.	Construct a line diagram and explain the working of air to water heat pump circuit.	CO5	3
OR			
11.	Explain comfort air conditioning system with the help of comfort chart.	CO5	3

ANURAG Engineering College

(An Autonomous Institution)

Ananthagiri (V & M), Suryapet (Dt.), Telangana - 508206.

Mechanical Engineering

III B.Tech II Semester Mid Marks List

Faculty: K.Veeranjaneyulu			Subject: R&AC						
S.No	H.T.No.	Name of the Student	Mid - I	Assignment - I	Mid - I Total	Mid - II	Assignment - II	Mid - II Total	AVG
1	21C11A0301	KARTHIK KOMPELLI	18	5	23	20	5	25	24
2	21C11A0302	KESHAVA SAI HEMANTH ABRAPURI	12	5	17	16	5	21	19
3	21C11A0305	SANDEEP REDDY KUNDURU	13	5	18	10	5	15	17
4	21C15A0312	SRINIVASA RAO ANNAM	17	5	22	15	5	20	21
5	22C15A0301	JAGAN S	16	5	21	14	5	19	20
6	22C15A0302	MAHENDRA CHEDDE	17	5	22	18	5	23	23
7	22C15A0303	RAGOTHAM UPPULA	19	5	24	18	5	23	24
8	22C15A0304	SHIVA ETIKALA	14	5	19	17	5	22	21
9	22C15A0305	SYED RIZWAN	9	5	14	18	5	23	19
Signature of the Faculty									

COURSE MATERIAL

III Year B. Tech II-

Semester

MECHANICAL

ENGINEERING

REFRIGERATION & AIR

CONDITIONING

Objective:

To impart knowledge on working principle of refrigeration cycle.

Outcome:

Ability to understand various refrigeration systems.

Introduction to Refrigeration: Necessity and Applications, Carnot Refrigerator, Unit of Refrigeration, COP, Different Refrigeration Methods, Air Refrigeration: Bell-Coleman Cycle, Ideal and Actual Cycles, Open and Dense Air Systems - Refrigeration Needs of Air Crafts, Application of Air Refrigeration, Types of systems – Problems.

UNIT – I

INTRODUCTION

In an air refrigeration cycle, the air is used as a refrigerant. In olden days, air was widely used in commercial applications because of its availability at free of cost. Since air does not change its phase i.e. remains gaseous throughout the cycle, therefore the heat carrying capacity per kg of air is very small as compared to vapour absorbing systems. The air-cycle refrigeration systems, as originally designed and installed, are now practically obsolete because of their low coefficient of performance and high power requirements. However, this system continues to be favored for air refrigeration because of the low weight and volume of the equipment. The basic elements of an air cycle refrigeration system are the compressor, the cooler or heat exchanger, the expander and the refrigerator.

Coefficient of Performance of a Refrigerator

The coefficient of performance (briefly written as C.O .P.) is the ratio of heat extracted in the refrigerator to the work done on the refrigerant. It is also known as theoretical coefficient of performance. Mathematically,

$$\text{Theoretically } C.O.P = \frac{Q}{W}$$

Where

Q= Amount of heat extracted in the refrigerator (or the amount of refrigeration produced, or the capacity of a refrigerator), and .

W = Amount of work done.

Notes: 1. For per unit mass, $C.O.P. = \frac{q}{w}$

2. The coefficient of performance is the reciprocal of the efficiency (i.e. $1/\eta$) of a heat engine. It is thus obvious, that the value of C.O.P is always greater than unity.

3. The ratio of the actual C.O.P to the theoretical. C.O.P. is known as relative coefficient of performance. Mathematically,

$$\text{Relative } C.O.P = \frac{\text{Actual } C.O.P}{\text{Theoretical } C.O.P}$$

Example 1: Find the C.O.P. of a refrigeration system if the work input is 80 kJ/kg and refrigeration effect produced is 160 kJ/kg of refrigerant flowing.

Solution: Given: $w = 80$ kJ/kg ; $q = 160$ kJ/kg

We know that C.O.P. of a refrigeration system
$$= \frac{q}{w} = \frac{160}{80} = 2$$

Difference Between a Heat Engine, Refrigerator and Heat Pump:

In a heat engine, as shown in Fig.1 (a), the heat supplied to the engine is converted into useful work. If Q_2 is the heat supplied to the engine and Q_1 is the heat rejected from the engine, then the net work done by the engine is given by

$$W_E = Q_2 - Q_1$$

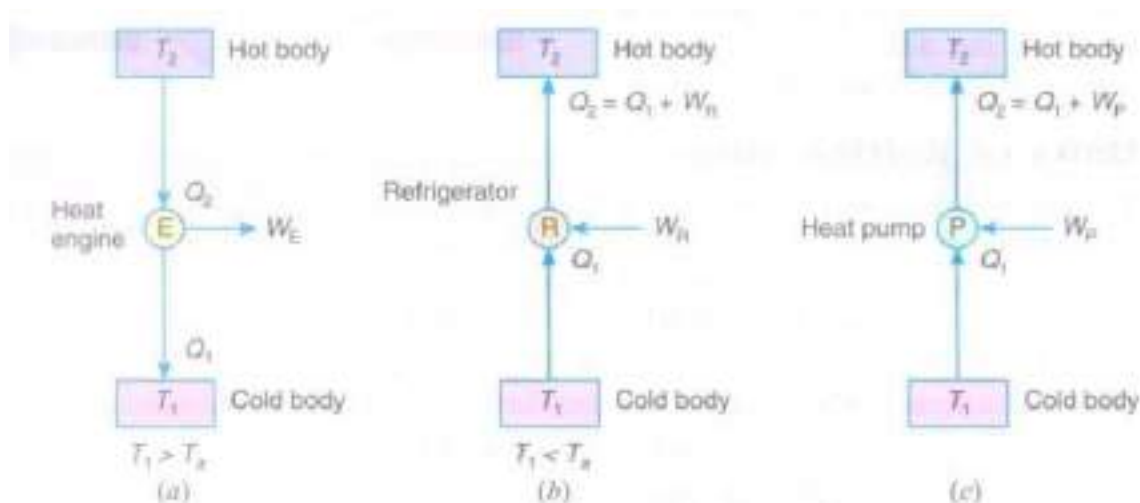


Fig: 1 Difference between a heat engine, refrigerator and heat pump

The performance of a heat engine is expressed by its efficiency. We know that the efficiency or coefficient of performance of an engine,

$$\eta_E \text{ or (C.O.P)}_E = \frac{\text{Work done}}{\text{Heat Supplied}} = \frac{W_E}{Q_2} = \frac{Q_2 - Q_1}{Q_2}$$

A refrigerator as shown in Fig. 1(b), is a reversed heat engine which either cool or maintain the temperature of a body (T_1) lower than the atmospheric temperature (T_a). This is done by extracting the heat (Q_1) from a cold body and delivering it to a hot body (Q_2). In doing so, work W_R is required to be done on the system. According to First Law of Thermodynamics,

$$W_R = Q_2 - Q_1$$

The performance of a refrigerator is expressed by the ratio of amount of heat taken from the cold body (Q_1) to the amount of work required to be done on the system (W_R). This ratio is called coefficient of performance. Mathematically, coefficient of performance of a refrigerator,

$$(C.O.P)_R = \frac{Q_1}{W_R} = \frac{Q_1 Q_2}{-Q_1}$$

Any refrigerating system is a heat pump as shown in Fig 1 (c), which extracts heat (Q_1) from a cold body and delivers it to a hot body. Thus there is no difference between the cycle of operations of a heat pump and a refrigerator. The main difference between the two is in their operating temperatures. A refrigerator works between the cold body temperature (T_1) and the atmospheric temperature (T_a) whereas the heat pump operates between the hot body temperature (T_2) and the atmospheric temperature (T_a). A refrigerator used for cooling in summer can be used as a heat pump for heating in winter.

In the similar way, as discussed for refrigerator, we have

$$W_p = Q_2 - Q_1$$

The performance of a heat pump is expressed by the ratio of the amount of heat delivered to the hot body (Q_2) to the amount of work required to be done on the system (W_P) This

ratio is called coefficient of performance or energy performance ratio (E.P.R.) of a heat pump.

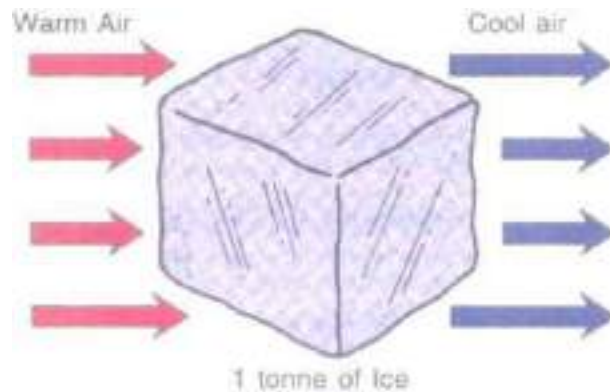
Mathematically, coefficient of performance or energy performance ratio of a heat pump,

$$\begin{aligned}(C.O.P)_p = E.P.R. &= \frac{Q_2}{W_p} = \frac{Q_1 Q_2}{-Q_1} \\ &= \frac{Q_1 Q_2}{-Q_1} + 1 = (C.O.P)_R + 1\end{aligned}$$

From above we see that the C.O.P. may be less than one or greater than one depending on the type of refrigeration system used. But the C.O.P. of a heat pump is always greater than one.

Mechanical refrigeration:

Before discussing the air refrigeration cycles, we should first know about the unit of refrigeration, coefficient of performance of a refrigerator and the difference between the heat engine, a refrigerator and a heat pump.



One tonne (1000 kg) of ice requires 335 KJ/Kg to melt. When this is accomplished in 24 hours, it is known as a heat transfer rate of 1 tonne of refrigeration (1TR)

Units of Refrigeration:

The practical unit of refrigeration is expressed in terms of tonne of refrigeration (briefly written as TR). A tonne of refrigeration is defined as the amount of refrigeration effect produced by the uniform melting of one tonne (1000 kg) of ice from and at 0°C in 24 hours.

Since the latent heat of ice is 335 kJ/kg, therefore one tonne of refrigeration,

$$1 \text{ TR} = 1000 \times 335 \text{ KJ in 24 hours}$$

$$= \frac{1000 \times 335}{24 \times 60} = 232.6 \text{ KJ/min}$$

In actual practice, one tonne of refrigeration is taken as equivalent to 210 kJ/min or 3.5 kW (i.e. 3.5 kJ/s).

Air Refrigerator Working on Reversed Carnot Cycle

In refrigerating systems, the Carnot cycle considered is the reversed Carnot cycle. We know that a heat engine working on Carnot cycle has the highest possible efficiency. Similarly, a refrigerating system working on the reversed Carnot cycle, will have the maximum possible coefficient of performance. We also know that it is not possible to make an engine working on the Carnot cycle. Similarly, it is also not possible to make a refrigerating machine working on the reversed Carnot cycle. However, it is used as the ultimate standard of comparison.

A reversed Carnot cycle, using air as working medium (or refrigerant) is shown on p - v and T - s diagrams in Fig. 1(a) and (b) respectively. At point 1, let p_1 , v_1 , T_1 be the pressure, volume and temperature of air respectively.

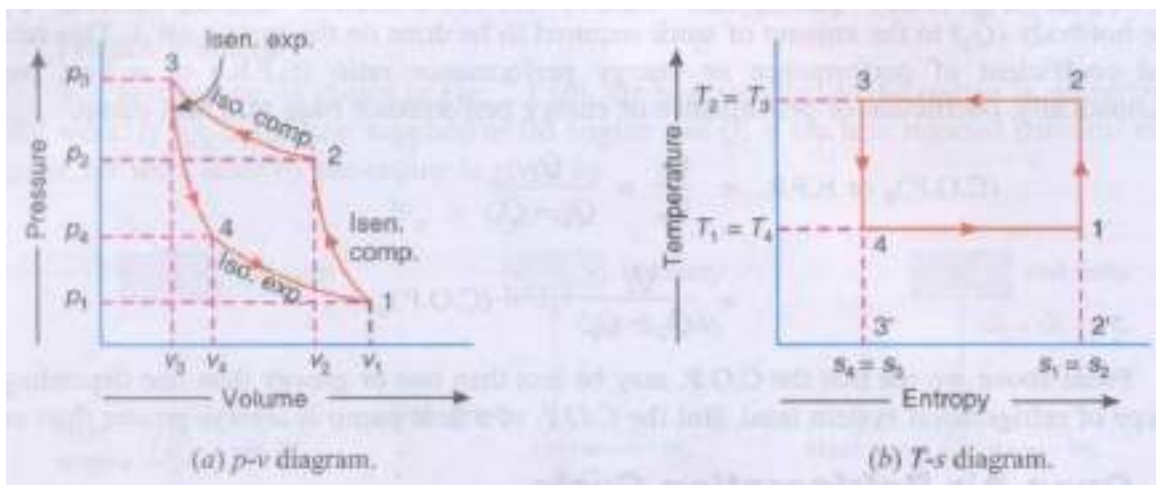


Fig: 1 Reserved Carnot Cycle

The four processes of the cycle are as follows:

1. Isentropic compression process: The air is compressed isentropic ally as shown by the curve 1-2 on p - v and T - s diagrams. During this process, the pressure of air increases from P_1 to P_2 specific volume decreases from V_1 to V_2 and temperature increases from T_1 to T_2 . We know that during isentropic compression, no heat is absorbed or rejected by the air.

2. Isothermal compression process: The air is now compressed isothermally (i.e. at constant temperature, $T_2 = T_3$) as shown by the curve 2-3 on p-v and T-s diagrams. During this process, the pressure of air increases from P_2 to P_3 and specific volume decreases from V_2 to V_3 . We know that the heat rejected by the air during isothermal compression per kg of air.

$$q_{2-3} = \text{Area } 2-3-3'-2'$$

$$T_3(s_2 - s_3) = T_2(s_2 - s_3)$$

3. Isentropic expansion process: The air is now expanded isentropic ally as shown by the curve 3-4 on p-v and T-s diagrams. The pressure of air decreases from P_3 to P_4 specific volume increases from V_3 to V_4 and the temperature decreases from T_3 to T_4 , We know that during isentropic expansion, no heat is absorbed or rejected by the air.

4. Isothermal expansion process: The air is now expanded isothermally (i.e. at constant temperature, $T_4 = T_1$) as shown by the curve 4-1 on p-v and T-s diagrams. The pressure of air decreases from P_4 to P_1 and specific volume increases from V_4 to V_1 . We know that the heat absorbed by the air (or heat extracted from the cold body) during isothermal expansion per kg of air,

$$q_{4-1} = \text{Area } 4-1-2'-3'$$

$$= T_3(s_2 - s_3) = T_2(s_2 - s_3)$$

We know that work done during the cycle per kg of air

$$= \text{Heat rejected} - \text{Heat absorbed} = q_{2-3} - q_{4-1}$$

$$= T_2(s_2 - s_3) - T_1(s_2 - s_3) = (T_2 - T_1)(s_2 - s_3)$$

Coefficient of performance of the refrigeration system working on reversed Carnot cycle

$$(C.O.P)_R = \frac{\text{Heat absorbed}}{\text{Work done}} = q_{4-1}$$

Though the reversed Carnot cycle is the most efficient between the fixed temperature limits, yet no refrigerator has been made using this cycle. This is due to the reason that the isentropic processes of the cycle require high speed while the isothermal processes require an extremely low speed. This variation in speed of air is not practicable.

Temperature Limitations for Reversed Carnot Cycle

We have seen in the previous article that the C.O.P. of the refrigeration system working on reversed Carnot cycle is given by

$$(C.O.P)_R = \frac{T_1 T_2}{-T_1}$$

Where T_1 = Lower temperature, and

T_2 = Higher temperature.

The C.O.P. of the reversed Carnot cycle may be improved by

1. Decreasing the higher temperature (i.e. temperature of hot body, T_2), or
2. Increasing the lower temperature (i.e. temperature of cold body, T_1).

This applies to all refrigerating machines, both theoretical and practical. It may be noted that temperatures T_1 and T_2 cannot be varied at will, due to certain functional limitations. It should be kept in mind that the higher temperature (T_2) is the temperature of cooling water or air available for rejection of heat and the lower temperature (T_1) is the temperature to be maintained in the refrigerator. The heat transfer will take place in the right direction only when the higher temperature is more than the temperature of cooling water or air to which heat is to be rejected, while the lower temperature must be less than the temperature of substance to be cooled.

Thus, if the temperature of cooling water or air (i.e., T_2) available for heat rejection is low, the C.O.P. of the Carnot refrigerator will be high. Since T_2 in winter is less than T_2 in summer, therefore, C.O.P. in winter will be higher than C.O.P. in summer. In other words, the Carnot refrigerators work more efficiently in winter than in summer. Similarly, if the lower temperature fixed by the refrigeration application is high, the C.O.P of the Carnot refrigeration will be high. Thus a Carnot refrigerator used for making ice at 0°C (273 K) will have less C.O.P than a Carnot refrigerator used for air-conditioned plant in summer at 20°C when the atmosphere temperature is 40°C . In other words, we can say that the Carnot C.O.P of a domestic refrigerator is less than the Carnot C.O.P of a domestic air-conditioner.

Air Refrigerator Working on a Bell-Coleman Cycle (or Reversed Brayton or Joule Cycle)

A Bell-Coleman air refrigeration machine was developed by Bell-Coleman and Light Foot by reversing the Joule's air cycle. It was one of the earliest types of refrigerators used in ships carrying frozen meat. Fig.1 shows a schematic diagram of such a machine which consists of a compressor, a cooler, an expander and a refrigerator.

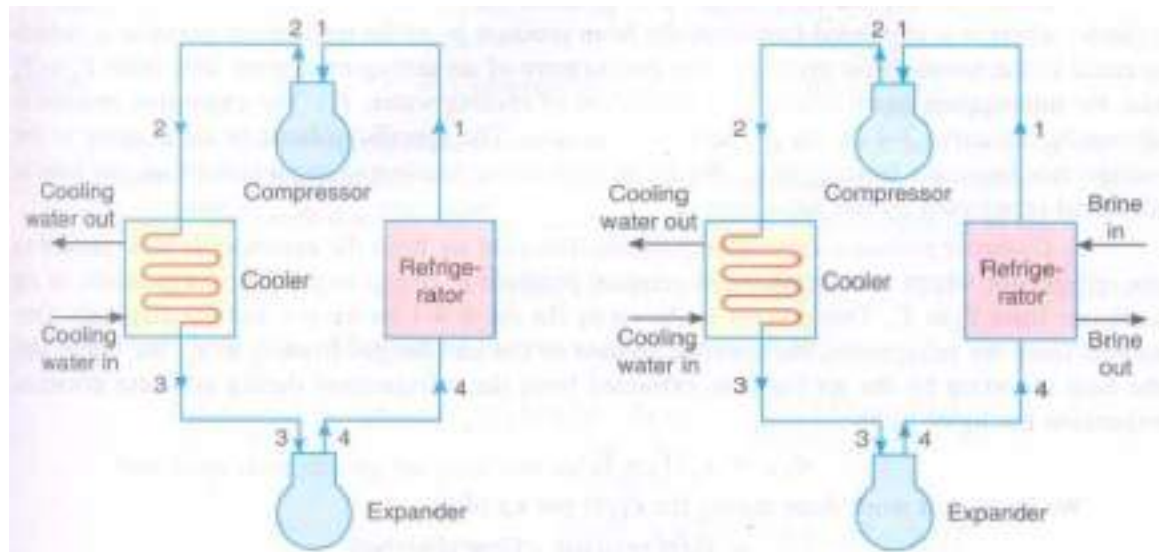


Fig.1 Open cycle air Bell-Coleman Refrigerator

Fig: 2 Closed cycle or Dense air Bell-Coleman Refrigerator

The Bell-Coleman cycle (also known as reversed Brayton or Joule cycle) is a modification of reversed Carnot cycle. The cycle is shown on p-V and T-s diagrams in Fig.3 (a) and (b). At point 1, let P_1 , V_1 and T_1 be the pressure, volume and temperature of air respectively. The four processes of the cycle are as follows:

1. Isentropic compression process: The cold air from the refrigerator is drawn into the compressor cylinder where it is compressed isentropically in the compressor as shown by the curve 1-2 on p-V and T-s diagrams. During the compression stroke, both the pressure and temperature increases and the specific volume of air at delivery from compressor reduces from V_1 to V_2 . We know that during isentropic compression process, no heat is absorbed or rejected by the air.

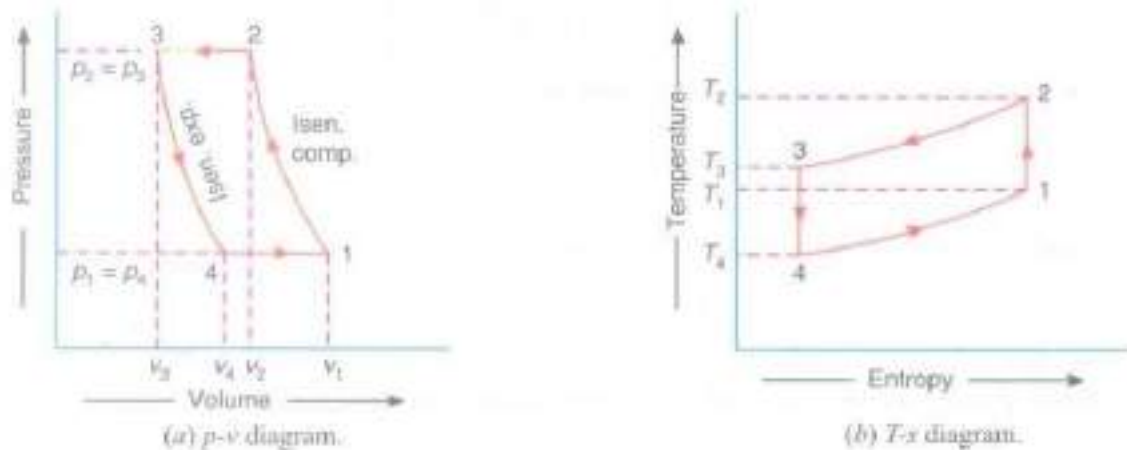


Fig: 3 Bell-Coleman Cycle.

2. Constant pressure cooling process: The warm air from the compressor is now passed into the cooler where it is cooled at constant pressure P_3 (equal to p_2), reducing the temperature from T_2 to T_3 (the temperature of cooling water) as shown by the curve 2-3 on p - v and T - s diagrams. The specific volume also reduces from V_2 to V_3 . We know that heat rejected by the air during constant pressure per kg of air,

$$Q_{2-3} = c_p (T_2 - T_3)$$

3. Isentropic expansion process: The air from the cooler is now drawn into the expander cylinder where it is expanded isentropically from pressure P_3 to the refrigerator pressure P_4 which is equal to the atmospheric pressure. The temperature of air during expansion falls from T_3 to T_4 (i.e. the temperature much below the temperature of cooling water, T_3). The expansion process is shown by the curve 3-4 on the p - v and T - s diagrams. The specific volume of air at entry to the refrigerator increases from V_3 to V_4 . We know that during isentropic expansion of air, no heat is absorbed or rejected by the air.

4. Constant pressure expansion process: The cold air from the expander is now passed to the refrigerator where it is expanded at constant pressure P_4 (equal to P_1). The temperature of air increases from T_4 to T_1 . This process is shown by the curve 4-1 on the p - v and T - s diagrams. Due to heat from the refrigerator, the specific volume of the air

changes from V_4 to V_1 . We know that the heat absorbed by the air (or heat extracted from the refrigerator) during constant pressure expansion per kg of air is

$$q_{4-1} = c_p (T_1 - T_4)$$

We know that work done during the cycle per kg of air

$$= \text{Heat rejected} - \text{Heat absorbed}$$

$$= c_p (T_2 - T_3) - c_p (T_1 - T_4)$$

\therefore Coefficient of performance,

$$C.O.P = \frac{\text{Heat absorbed}}{\text{work done}} = \frac{c_p (T_1 - T_4)}{c_p (T_2 - T_3) - c_p (T_1 - T_4)}$$

Open Air Refrigeration Cycle

In an open air refrigeration cycle, the air is directly led to the space to be cooled (i.e. a refrigerator), allowed to circulate through the cooler and then returned to the compressor to start another cycle. Since the air is supplied to the refrigerator at atmospheric pressure, therefore, volume of air handled by the compressor and expander is large. Thus the size of compressor and expander should be large. Another disadvantage of the open cycle system is that the moisture is regularly carried away by the air circulated through the cooled space. This leads to the formation of frost at the end of expansion process and clog the line. Thus in an open cycle system, a drier should be used.

Closed or Dense Air Refrigeration Cycle

In a closed or dense air refrigeration cycle, the air is passed through the pipes and component parts of the system at all times. The air, in this system, is used for absorbing heat from the other fluid (say brine) and this cooled brine is circulated into the space to be cooled. The air in the closed system does not come in contact directly with the space to be cooled.

The closed air refrigeration cycle has the following thermodynamic advantages:

1. Since it can work at a suction pressure higher than that of atmospheric pressure, therefore the volume of air handled by the compressor and expander are smaller as compared to an open air refrigeration cycle system.
2. The operating pressure ratio can be reduced, which results in higher coefficient of performance

Introduction

The advent of high-speed passenger aircraft, jet aircraft and missiles has introduced the need for compact, and simple refrigeration systems, capable of high capacity, with minimum reduction of pay load. When the power requirements, needed to transport the additional weight of the refrigerating system are taken into account, the air cycle systems usually prove to be the most efficient. The cooling demands per unit volume of space are heavy. An ordinary passenger aircraft requires a cooling system capable of 8 TR capacity and a super constellation requires a cooling system of more than 8 TR capacity. A jet fighter traveling at 950 km/h needs a cooling system capable of 10 to 20 TR capacity. To dissipate the heat load from 10 kW of electronic equipment in a missile or other high speed flight system, approximately 3 TR of cooling capacity are required. The miniaturization of electronic equipment concentrates a heavy cooling load in a small area. It creates difficulty in transferring heat to air at high altitudes. Moreover, low pressure of air further complicates the refrigeration design requirements.

Methods of Air Refrigeration Systems

The various methods of air refrigeration systems used for air crafts these days are as follows:

1. Simple air cooling system
2. Simple air evaporative cooling system
3. Boot strap air cooling system
4. Boot strap air evaporative cooling system
5. Reduced ambient air cooling system, and
6. Regenerative air cooling system.

Now we shall discuss all the above mentioned cooling systems, one by one, in the following pages.

Simple Air Cooling System

A simple air cooling system for aircrafts is shown in Fig.1. The main components of this system are the main compressor driven by a gas turbine, a heat exchanger, a cooling turbine and a cooling air fan. The air required for refrigeration system is bled off from the main compressor. This high pressure and high temperature air is cooled initially in the heat exchanger where ram air is used for cooling. It is further cooled in the cooling turbine by the process of expansion. The work of this turbine is used to drive the cooling fan which draws cooling air through the heat exchanger. This system is good for ground surface cooling and for low flight speeds.

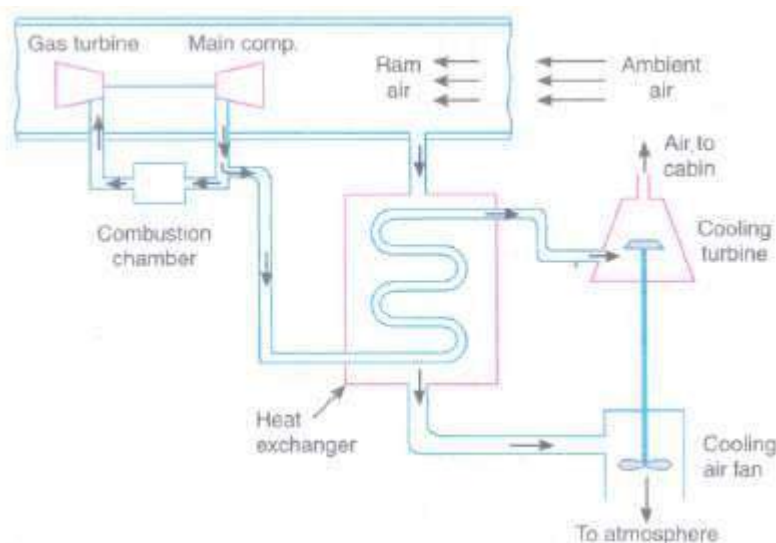


Fig:1 Simple air cooling system

The T-s diagram for a simple air cooling system is shown in Fig. 2. The various processes are discussed below:

1. Ramming process: Let the pressure and temperature of ambient air is P_1 and T_1 respectively. The ambient air is rammed isentropically from pressure P_1 and temperature T_1 to the pressure P_2 and temperature T_2 . This ideal ramming action is shown by the vertical line 1-2 in Fig.2. In actual practice, because of internal friction due to irreversibilities, the temperature of the rammed air is more than T_2 . Thus the actual ramming process is shown by the curve 1-2 which is adiabatic but not isentropic due to friction. The pressure and temperature of the rammed air is now P_2 and T_2 , respectively.

During the ideal or actual ramming process, the total energy or enthalpy remains constant i.e. $h_2 = h_2'$ and $T_2 = T_2'$.

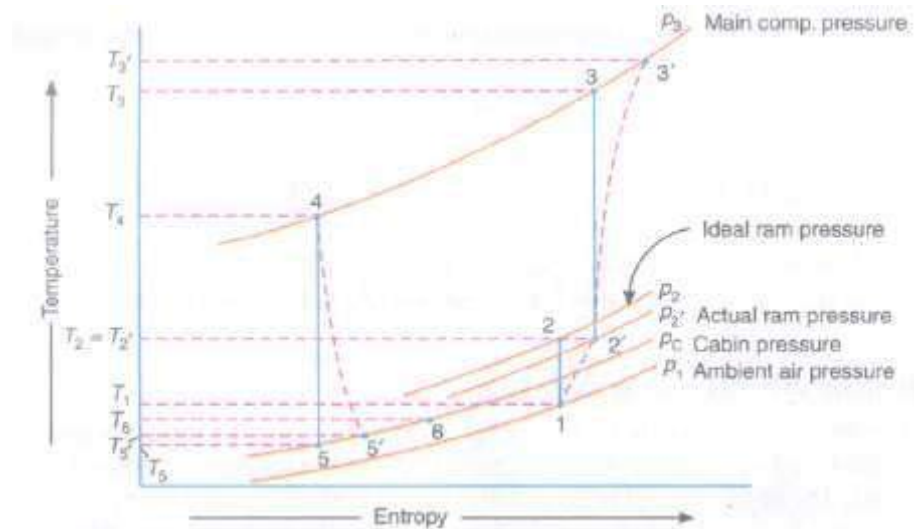


Fig: 2 T-s diagram for simple air cycle cooling system

Simple Air Evaporative Cooling system

A simple air evaporative cooling system is shown in Fig.1. It is similar to the simple cooling system except that the addition of an evaporator between the heat exchanger and cooling turbine. The evaporator provides an additional cooling effect through evaporation of a refrigerant such as water. At high altitudes, the evaporative cooling may be obtained by using alcohol or ammonia. The water, alcohol and ammonia have different refrigerating effects at different altitudes. At 20000 metres height, water boils at 40°C, alcohol at 9°C and ammonia at - 70°C.



Air evaporative cooling system

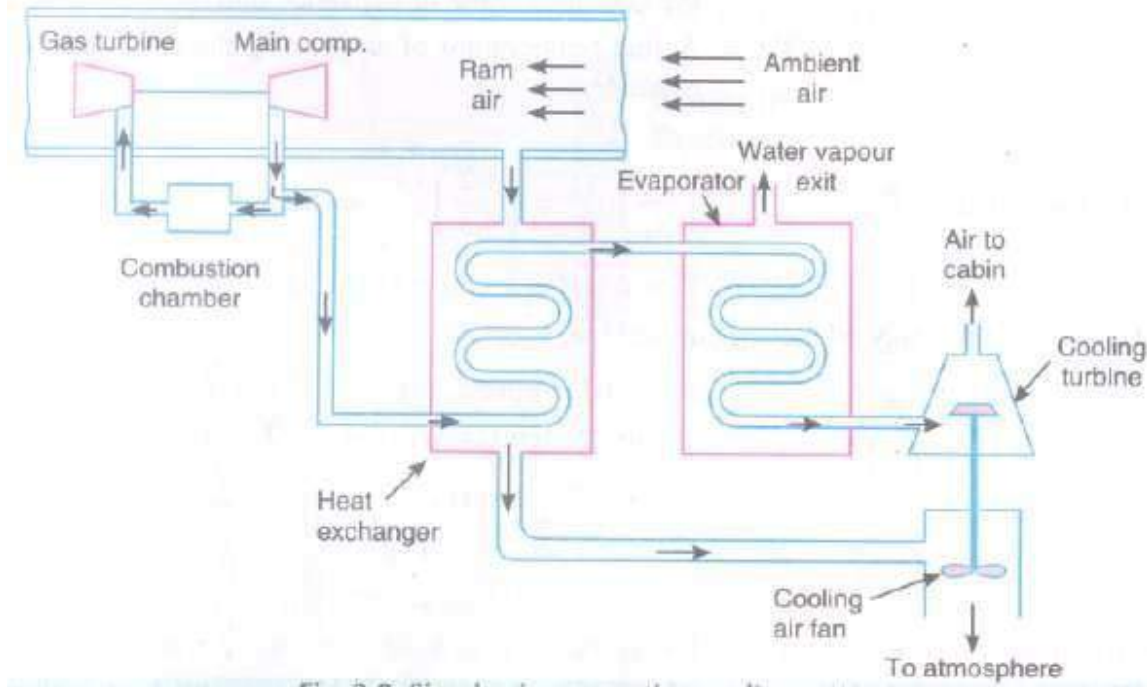


Fig: 3 Simple air evaporative cooling system

The T - s diagram for a simple air cycle evaporative cooling system is shown in Fig.3. The various processes are same as discussed in the previous article, except that the cooling process in the evaporator is shown by 4 - 4' in Fig.4.

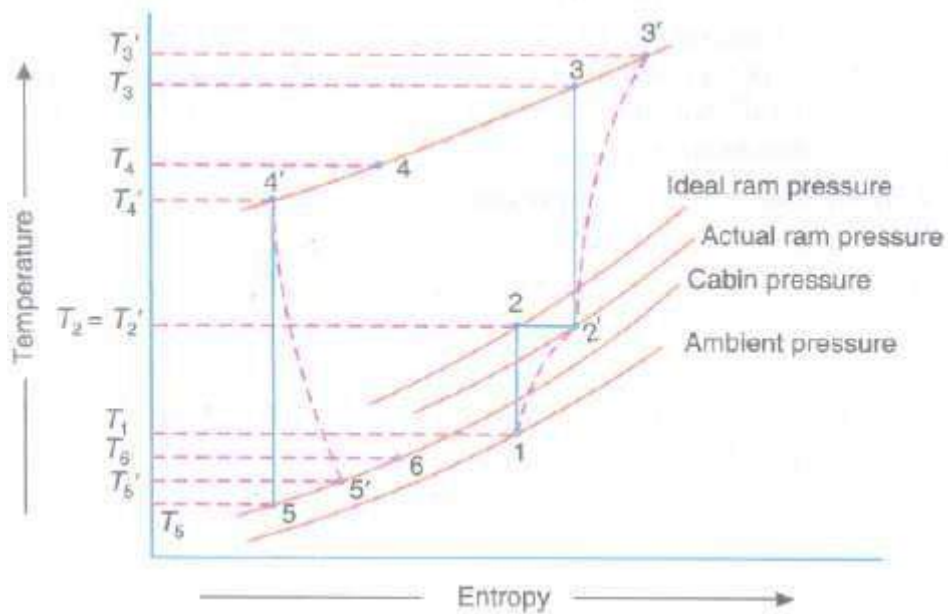


Fig: 4 T-s diagram for simple evaporative cooling system

$$m_a = \frac{210 Q}{c_p (T_1 - T_2)} \text{ Kg/min}$$

Power required for the refrigerating system,

$$P = \frac{m_a c_p (T_3 - T_2)}{60} \text{ KW}$$

And C.O.P of the refrigerating system

$$= \frac{210 Q}{m_a c_p (T_3 - T_2)} = \frac{210 Q}{P \times 60}$$

The initial mass of evaporant (m_e) required to carried for the given flight time is given by

$$m_e = \frac{Q_e \cdot t}{h_{fg}}$$

Where

Q_e = Heat to be removed in evaporation in KJ/ min

t = Flight time in minutes, and

h_{fg} = Latent heat of vaporization of evaporant in KJ/Kg

Boot-strap Air Cooling System

A boot-strap air cooling system is shown in Fig.1. This cooling system has two heat exchangers instead of one and a cooling turbine drives a secondary compressor instead of cooling fan. The air bled from the main compressor is first cooled by the ram air in the first heat exchanger. This cooled air, after compression in the secondary compressor, is led to the second heat exchanger where it is again cooled by the ram air before passing to the cooling turbine. This type of cooling system is mostly used in transport type aircraft.

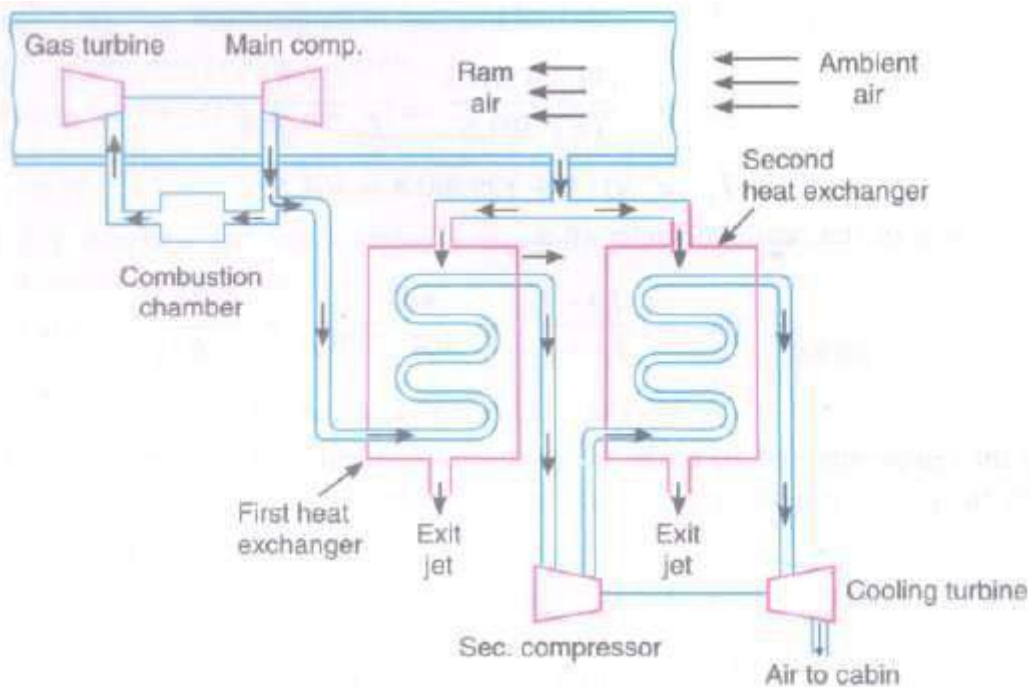


Fig:5 Boot-strap air cooling system

The T-s diagram for a boot-strap air cycle cooling system is shown in Fig.5. The various processes are as follows:

1. The process 1- 2 represents the isentropic ramming of ambient air from pressure P_1 and temperature T_1 to pressure P_2 and temperature T_2 . The process 1- 2' represents the actual ramming process because of internal friction due to irreversibilities.
2. The process 2'- 3 represents the isentropic compression of air in the main compressor and the process 2'- 3' represents the actual compression of air because of internal friction due to irreversibilities.

3. The process 3'-4 represents the cooling by ram air in the first heat exchanger. The pressure drop in the heat exchanger is neglected.
4. The process 4 - 5 represents the isentropic compression of cooled air, from first heat exchanger, in the secondary compressor. The process 4 - 5' represents the actual compression process because of internal friction due to irreversibilities.
5. The process 5'- 6 represents the cooling by ram air in the second heat exchanger. The pressure drop in the heat exchanger is neglected.
6. The process 6 - 7 represents the isentropic expansion of cooled air in the cooling turbine up to the cabin pressure. The process 6 - 7' represents actual expansion of the cooled air in the cooling turbine.
7. The process 7'- 8 represents the heating of air up to the cabin temperature T_8 .

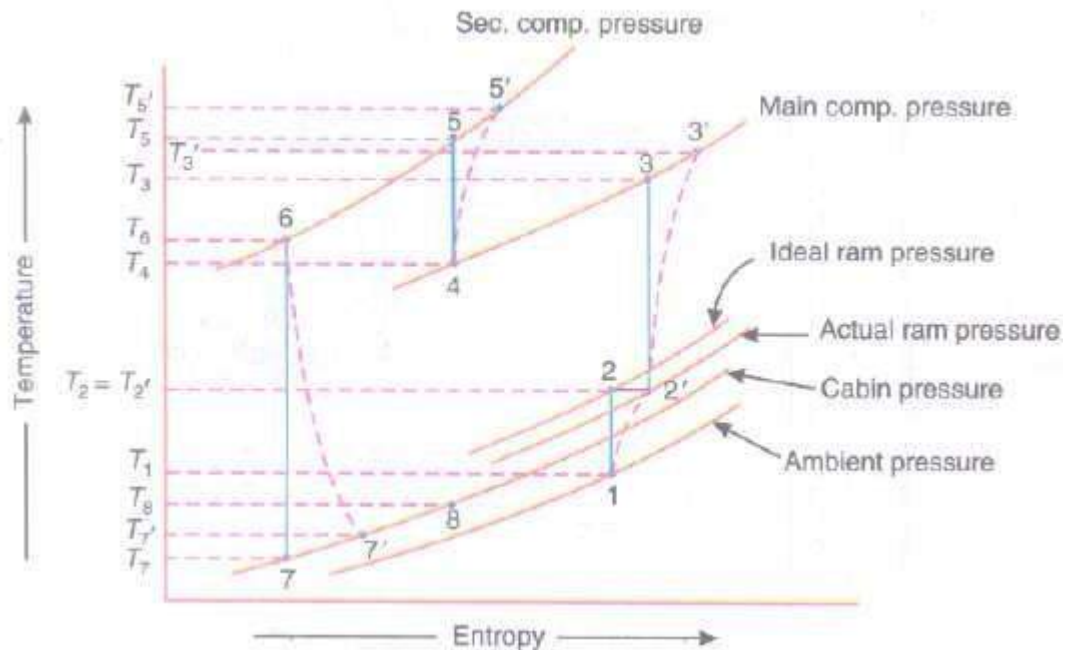


Fig: 6 T-s diagram for boot strap air cooling system

If Q tonnes of refrigeration is the cooling load in the cabin, then the quantity of air required for the refrigeration purpose will be

$$m_a = \frac{210 Q}{c (T - T')} \text{ Kg/min}$$

Unit -II**Objective:**

To apply the principles of Thermodynamics to analyse different types of refrigeration systems.

Outcome:

Ability to understand the operation of various devices of VCR system.

Vapour Compression Refrigeration (VCR) System: Basic cycle- Working principle and essential components of the plant – COP – Representation of cycle on T-S and p-h charts – effect of sub cooling and super heating – cycle analysis – Actual cycle - numerical problems.

Refrigerants: Desirable Properties – Classification of Refrigerants Used – Nomenclature Secondary Refrigerants- Lubricants – Ozone Depletion – Global Warming- Newer Refrigerants.

Vapour Compression Refrigeration Systems:

As mentioned, vapour compression refrigeration systems are the most commonly used among all refrigeration systems. As the name implies, these systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems. Vapour compression refrigeration systems are available to suit almost all applications with the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle. Before the actual cycle is discussed and analysed, it is essential to find the upper limit of performance of vapour compression cycles. This limit is set by a completely reversible cycle.

Comparison between gas cycles and vapor cycles:

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. As mentioned in the previous chapter, in a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration

Notes-

take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

Standard Vapour Compression Refrigeration System (VCRS):

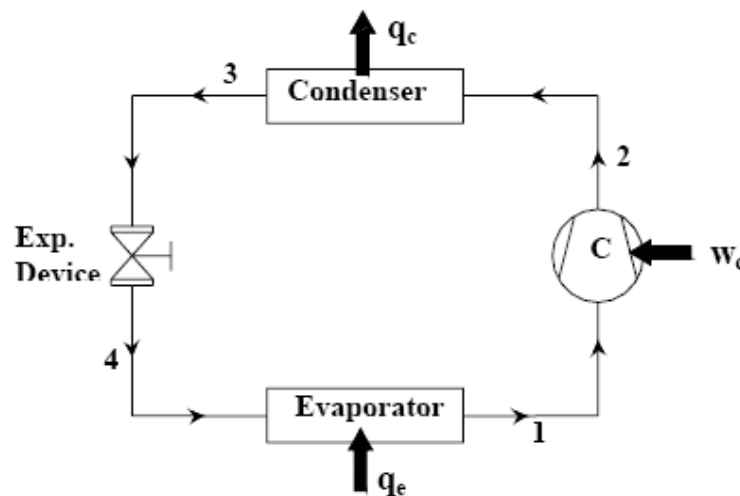
Figure 5 shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a $T-s$ diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor

Process 2-3: Isobaric heat rejection in condenser

Process 3-4: Isenthalpic expansion of saturated liquid in expansion device

Process 4-1: Isobaric heat extraction in the evaporator



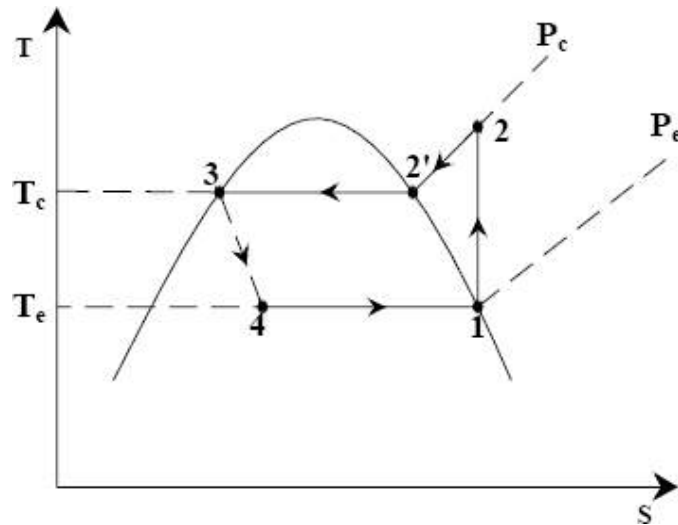


Fig.5. Standard Vapour compression refrigeration system

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3- 4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the cycle diagrams on T s charts.

Use of Pressure-enthalpy (P-h) charts:

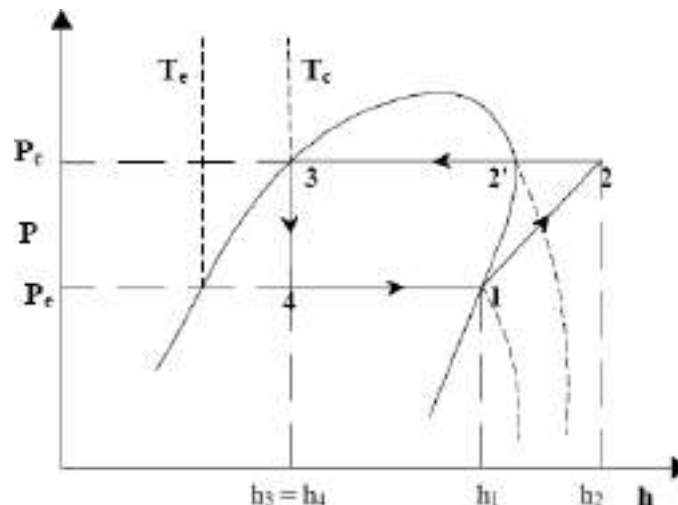


Fig. 1. Standard vapour compression refrigeration cycle on a P-h chart Since the various performance parameters are expressed in terms of

enthalpies, it is very convenient to use a pressure – enthalpy chart for property evaluation and performance analysis. The use of these charts was first suggested by Richard Mollier. Figure 1 shows the standard vapour compression refrigeration cycle on a P-h chart. As discussed before, in a typical P-h chart, enthalpy is on the x-axis and pressure is on y-axis. The isotherms are almost vertical in the subcooled region, horizontal in the two-phase region (for pure refrigerants) and slightly curved in the superheated region at high pressures, and again become almost vertical at low pressures. A typical P-h chart also shows constant specific volume lines (isochors) and constant entropy lines (isentropes) in the superheated region. Using P-h charts one can easily find various performance parameters from known values of evaporator and condenser pressures.

In addition to the P-h and T-s charts one can also use thermodynamic property tables from solving problems related to various refrigeration cycles.

Performance of SSS cycle:

The performance of a standard VCRS cycle can be obtained by varying evaporator and condensing temperatures over the required range. Figure 1 shows the effects of evaporator and condensing temperatures on specific and volumetric refrigeration effects of a standard VCRS cycle. As shown in the figure, for a given condenser temperature as evaporator temperature increases the specific refrigeration effect increases marginally. It can be seen that for a given evaporator temperature, the refrigeration effect decreases as condenser temperature increases. These trends can be explained easily with the help of the P-h diagram. It can also be observed that the volumetric refrigeration effect increases rapidly with evaporator temperature due to the increase in specific refrigeration effect and decrease in specific volume of refrigerant vapour at the inlet to the compressor. Volumetric refrigeration effect increases marginally as condenser temperature decreases.

The performance of a standard VCRS cycle can be obtained by varying evaporator and condensing temperatures over the required range. Figure 2 shows the effects of evaporator and condensing temperatures on specific and volumetric refrigeration effects of a standard VCRS cycle. As shown in the figure, for a given condenser temperature as evaporator temperature increases the specific

Notes-

refrigeration effect increases marginally. It can be seen that for a given evaporator temperature, the refrigeration effect decreases as condenser temperature increases. These trends can be explained easily with the help of the P-h diagram. It can also be observed that the volumetric refrigeration effect increases rapidly with evaporator temperature due to the increase in specific refrigeration effect and decrease in specific volume of refrigerant vapour at the inlet to the compressor. Volumetric refrigeration effect increases marginally as condenser temperature decreases.

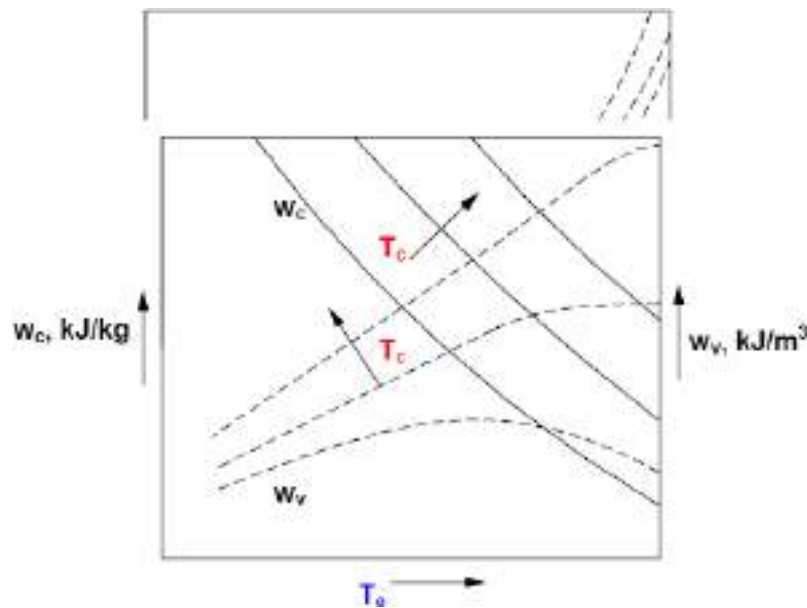


Fig.2: Effect of evaporator and condenser temperatures on specific and volumic works of compression of a standard VCRS cycle

Figure 2 shows that the specific work of compression decreases rapidly as the evaporator temperature increases and condenser temperature decreases. Once again these effects can be explained using a $T-s$ or $P-h$ diagram. For a given condenser temperature, the volumic work of compression increases initially, reaches a peak, then starts decreasing. This is due to the fact that as evaporator temperature increases the specific work of compression decreases and the specific volume at the inlet to the compressor also decreases. As a result, an optimum evaporator temperature exists at which the volumic work of compression reaches a maximum. Physically, the volumic work of compression is analogous to mean effective pressure of the compressor, as multiplying this with the volumetric flow rate gives the power input to the compressor. For a given power input, a high volumic work of compression implies smaller volumetric flow rates and hence a smaller compressor.

Figure 3 shows the effect of evaporator and condenser temperatures on COP of the SSS cycle. As expected, for a given condenser temperature the COP increases rapidly with evaporator temperature, particularly at low condensing temperatures. For a given evaporator temperature, the COP decreases as condenser temperature increases. However, the effect of condenser temperature becomes marginal at low evaporator temperatures.

The above results show that at very low evaporator temperatures, the COP becomes very low and also the size of the compressor becomes large (due to small volumic refrigeration effect). It can also be shown that the compressor discharge temperatures also increase as the evaporator temperature decreases. Hence, single stage vapour compression refrigeration systems are not viable for very low evaporator temperatures. One has to use multistage or cascade systems for these applications. These systems will be discussed in the next lecture. One can also observe the similarities in performance trends between SSS cycle and Carnot cycle, which is to be expected as the VCRS cycle is obtained by modifying the SSS cycle.

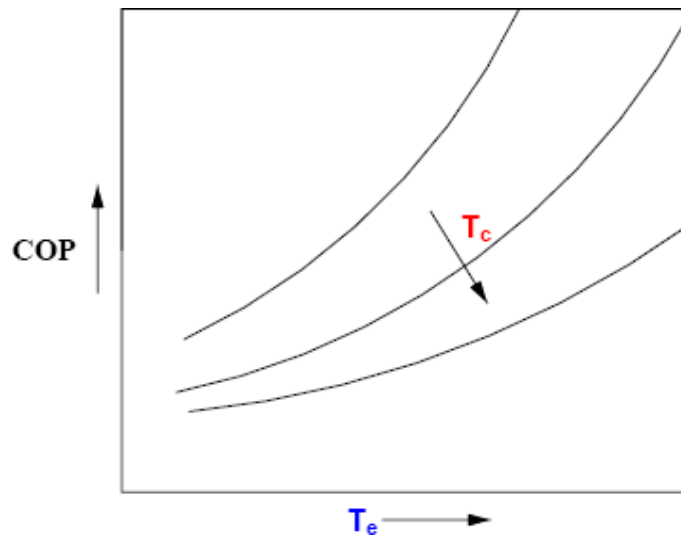


Fig.3: Effect of evaporator and condenser temperatures on COP of a standard VCRS cycle

Comparison between gas cycles and vapor cycles:

Thermodynamic cycles can be categorized into gas cycles and vapour cycles. As mentioned in the previous chapter, in a typical gas cycle, the working fluid (a gas) does not undergo phase change, consequently the operating cycle will be away from the vapour dome. In gas cycles, heat rejection and refrigeration take place as the gas undergoes sensible cooling and heating. In a vapour cycle the working fluid undergoes phase change and refrigeration effect is due to the vaporization of refrigerant liquid. If the refrigerant is a pure substance then its temperature remains constant during the phase change processes. However, if a zeotropic mixture is used as a refrigerant, then there will be a temperature glide during vaporization and condensation. Since the refrigeration effect is produced during phase change, large amount of heat (latent heat) can be transferred per kilogram of refrigerant at a near constant temperature. Hence, the required mass flow rates for a given refrigeration capacity will be much smaller compared to a gas cycle. Vapour cycles can be subdivided into vapour compression systems, vapour absorption systems, vapour jet systems etc. Among these the vapour compression refrigeration systems are predominant.

Vapour Compression Refrigeration Systems:

As mentioned, vapour compression refrigeration systems are the most commonly

used among all refrigeration systems. As the name implies, these systems belong to the general class of vapour cycles, wherein the working fluid (refrigerant) undergoes phase change at least during one process. In a vapour compression refrigeration system, refrigeration is obtained as the refrigerant evaporates at low temperatures. The input to the system is in the form of mechanical energy required to run the compressor. Hence these systems are also called as mechanical refrigeration systems. Vapour compression refrigeration systems are available to suit almost all applications with the refrigeration capacities ranging from few Watts to few megawatts. A wide variety of refrigerants can be used in these systems to suit different applications, capacities etc. The actual vapour compression cycle is based on Evans-Perkins cycle, which is also called as reverse Rankine cycle. Before the actual cycle is discussed and analysed, it is essential to find the upper limit of performance of vapour compression cycles. This limit is set by a completely reversible cycle.

The Carnot refrigeration cycle:

Carnot refrigeration cycle is a completely reversible cycle, hence is used as a model of perfection for a refrigeration cycle operating between a constant temperature heat source and sink. It is used as reference against which the real cycles are compared. Figures 1(a) and (b) show the schematic of a Carnot vapour compression refrigeration system and the operating cycle on T-s diagram.

As shown in Fig.1(a), the basic Carnot refrigeration system for pure vapour consists of four components: compressor, condenser, turbine and evaporator. Refrigeration effect ($q_{4-1} = q_e$) is obtained at the evaporator as the refrigerant undergoes the process of vaporization (process 4-1) and extracts the latent heat from the low temperature heat source. The low temperature, low pressure vapour is then compressed isentropically in the compressor to the heat sink temperature T_c . The refrigerant pressure increases from P_e to P_c during the compression process (process 1-2) and the exit vapour is saturated. Next the high pressure, high temperature saturated refrigerant undergoes the process of condensation in the condenser (process 2-3) as it rejects the heat of condensation ($q_{2-3} = q_c$) to an external heat sink at T_c . The high pressure saturated liquid then flows through the turbine and undergoes isentropic

expansion (process 3-4). During this process, the pressure and temperature fall from P_c, T_c to P_e, T_e . Since a saturated liquid is expanded in the turbine, some amount of liquid flashes into vapour and the exit condition lies in the two-phase region. This low temperature and low pressure liquid-vapour mixture then enters the evaporator completing the cycle. Thus as shown in Fig.1(b), the cycle involves two isothermal heat transfer processes (processes 4-1 and 2-3) and two isentropic work transfer processes (processes 1-2 and 3-4). Heat is extracted isothermally at evaporator temperature T_e during process 4-1, heat is rejected isothermally at condenser temperature T_c during process 2-3. Work is supplied to the compressor during the isentropic compression (1-2) of refrigerant vapour from evaporator pressure P_e to condenser pressure P_c , and work is produced by the system as refrigerant liquid expands isentropically in the turbine from condenser pressure P_c to evaporator pressure P_e . All the processes are both internally as well as externally reversible, i.e., net entropy generation for the system and environment is zero.

Applying first and second laws of thermodynamics to the Carnot refrigeration cycle,

$$\oint \delta q = \oint \delta w$$

$$\oint \delta q = q_{4-1} - q_{2-3} = q_e - q_c$$

Refrigeration and Air $\oint \delta W = W_{3-4} - W_{1-2} = W_T - W_C = -W_{net}$ Unit-

$$\Rightarrow (q_c - q_e) = w_{net} \quad (1)$$

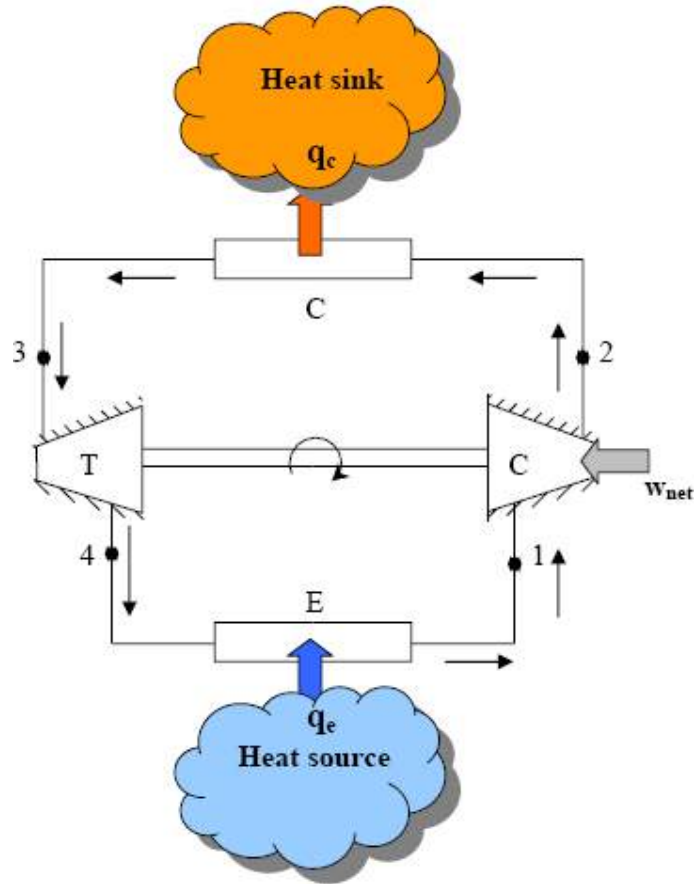


Fig.1(a): Schematic of a Carnot refrigeration system

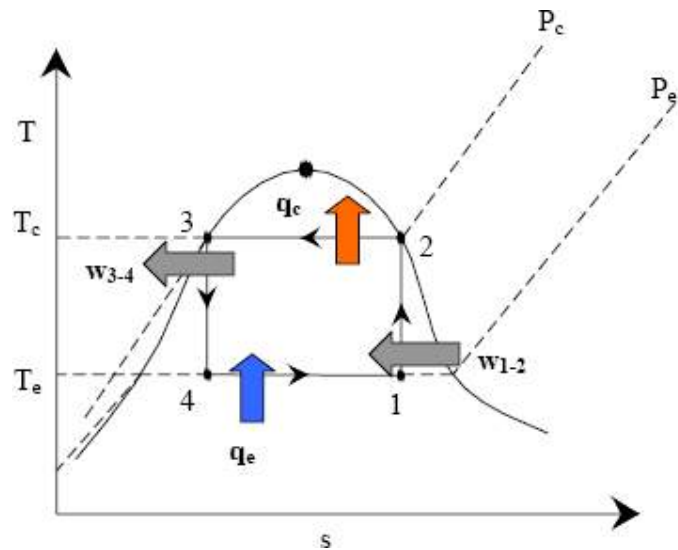


Fig.1(b): Carnot refrigeration cycle on T-s diagram

now for the reversible, isothermal heat transfer processes 2-3 and 4-1, we can write:

$$q_e = q_{4-1} = \int_4^1 T.ds = T_e (s_1 - s_4) \quad (2)$$

$$q_c = -q_{2-3} = -\int_2^3 T.ds = T_c (s_2 - s_3) \quad (3)$$

where T_e and T_c are the evaporator and condenser temperatures, respectively, and,

$$s_1 = s_2 \text{ and } s_3 = s_4 \quad (4)$$

the Coefficient of Performance (COP) is given by:

$$\text{COP}_{\text{Carnot}} = \frac{\text{refrigeration effect}}{\text{net work input}} = \frac{q_e}{w_{\text{net}}} = \frac{T_e (s_1 - s_4)}{T_c (s_2 - s_3) - T_e (s_1 - s_4)} = \left(\frac{T_e}{T_c - T_e} \right) \quad (5)$$

thus the COP of Carnot refrigeration cycle is a function of evaporator and condenser temperatures only and is independent of the nature of the working substance. This is the reason why exactly the same expression was obtained for air cycle refrigeration systems operating on Carnot cycle (Lesson 9). The Carnot COP sets an upper limit for refrigeration systems operating between two constant temperature thermal reservoirs (heat source and sink). From Carnot's theorems, for the same heat source and sink temperatures, no irreversible cycle can have COP higher than that of Carnot COP.

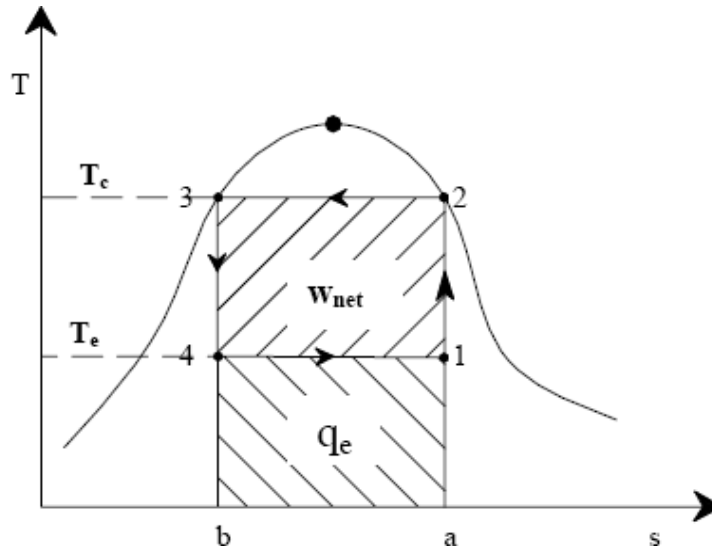


Fig.2. Carnot refrigeration cycle represented in T-s plane

It can be seen from the above expression that the COP of a Carnot refrigeration system increases as the evaporator temperature increases and condenser temperature decreases. This can be explained very easily with the help of the T-s diagram (Fig.2). As shown in the figure, COP is the ratio of area a-1-4-b to the area 1-2-3-4. For a fixed condenser temperature T_c , as the evaporator temperature T_e increases, area a-1-4-b (q_e) increases and area 1-2-3-4 (W_{net}) decreases as a result, COP increases rapidly. Similarly for a fixed evaporator temperature T_e , as the condensing temperature T_c increases, the net work input (area 1-2-3-4) increases, even though cooling output remains constant, as a result the COP falls. Figure 3 shows the variation of Carnot COP with evaporator temperature for different condenser temperatures. It can be seen that the COP increases sharply with evaporator temperatures, particularly at high condensing temperatures. COP reduces as the condenser temperature increases, but the effect becomes marginal at low evaporator temperatures. It will be shown later that actual vapour compression refrigeration systems also behave in a manner similar to that of Carnot refrigeration systems as far as the performance trends are concerned.

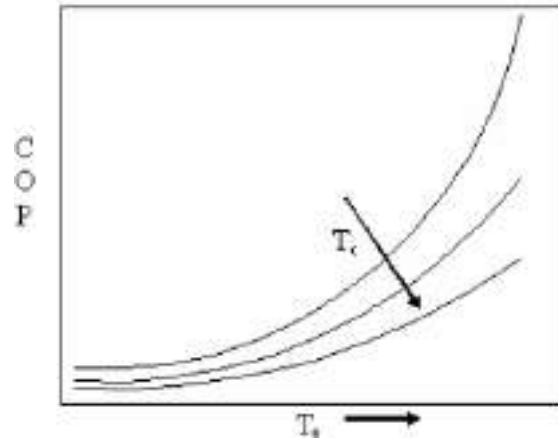


Fig.10.3. Effects of evaporator and condenser temperatures on Carnot COP Practical difficulties with Carnot refrigeration system:

It is difficult to build and operate a Carnot refrigeration system due to the following practical difficulties:

- i. During process 1-2, a mixture consisting of liquid and vapour have to be compressed isentropically in the compressor. Such a compression is known as wet compression due to the presence of liquid. In practice, wet compression is very difficult especially with reciprocating compressors. This problem is particularly severe in case of high speed reciprocating compressors, which get damaged due to the presence of liquid droplets in the vapour. Even though some types of compressors can tolerate the presence of liquid in vapour, since reciprocating compressors are most widely used in refrigeration, traditionally dry compression (compression of vapour only) is preferred to wet compression.
- ii. The second practical difficulty with Carnot cycle is that using a turbine and extracting work from the system during the isentropic expansion of liquid refrigerant is not economically feasible, particularly in case of small capacity systems. This is due to the fact that the specific work output (per kilogram of refrigerant) from the turbine is given by:

$$w_{3-4} = \int_{P_e}^{P_c} v \cdot dP$$

(6)

since the specific volume of liquid is much smaller compared to the specific volume of a vapour/gas, the work output from the turbine in case of the liquid will be small. In addition, if one considers the

inefficiencies of the turbine, then

the net output will be further reduced. As a result using a turbine for extracting the work from the high pressure liquid is not economically justified in most of the cases. One way of achieving dry compression in Carnot refrigeration cycle is to have two compressors – one isentropic

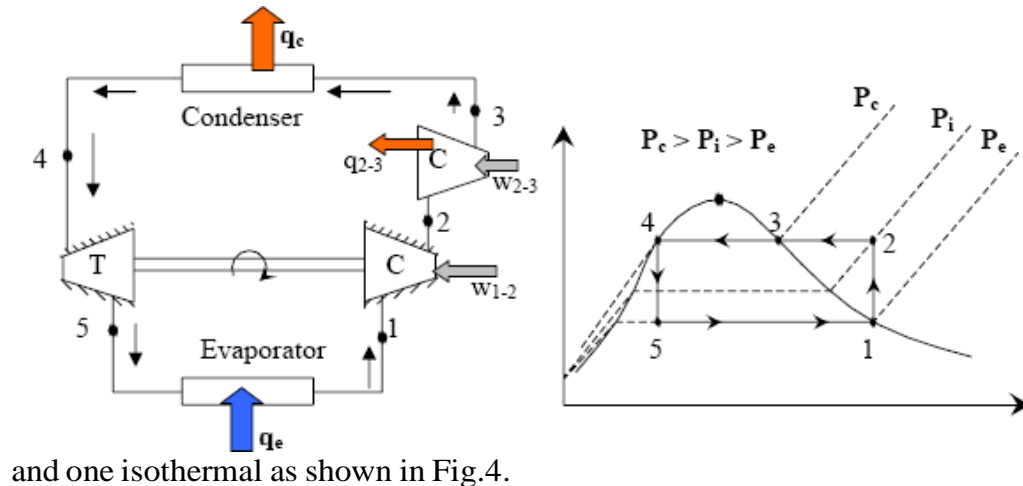


Fig .4. Carnot refrigeration system with dry compression

As shown in Fig.4, the Carnot refrigeration system with dry compression consists of one isentropic compression process (1-2) from evaporator pressure P_e to an intermediate pressure P_i and temperature T_c , followed by an isothermal compression process (2-3) from the intermediate pressure P_i to the condenser pressure P_c . Though with this modification the problem of wet compression can be avoided, still this modified system is not practical due to the difficulty in achieving true isothermal compression using high speed compressors. In addition, use of two compressors in place of one is not economically justified.

From the above discussion, it is clear that from practical considerations, the Carnot refrigeration system need to be modified. Dry compression with a single compressor is possible if the isothermal heat rejection process is replaced by isobaric heat rejection process. Similarly, the isentropic expansion process can be replaced by an isenthalpic throttling process. A refrigeration system, which incorporates these two changes is known as

Evans-Perkins or reverse Rankine cycle. This is the theoretical cycle on which the actual vapour compression refrigeration systems are based.

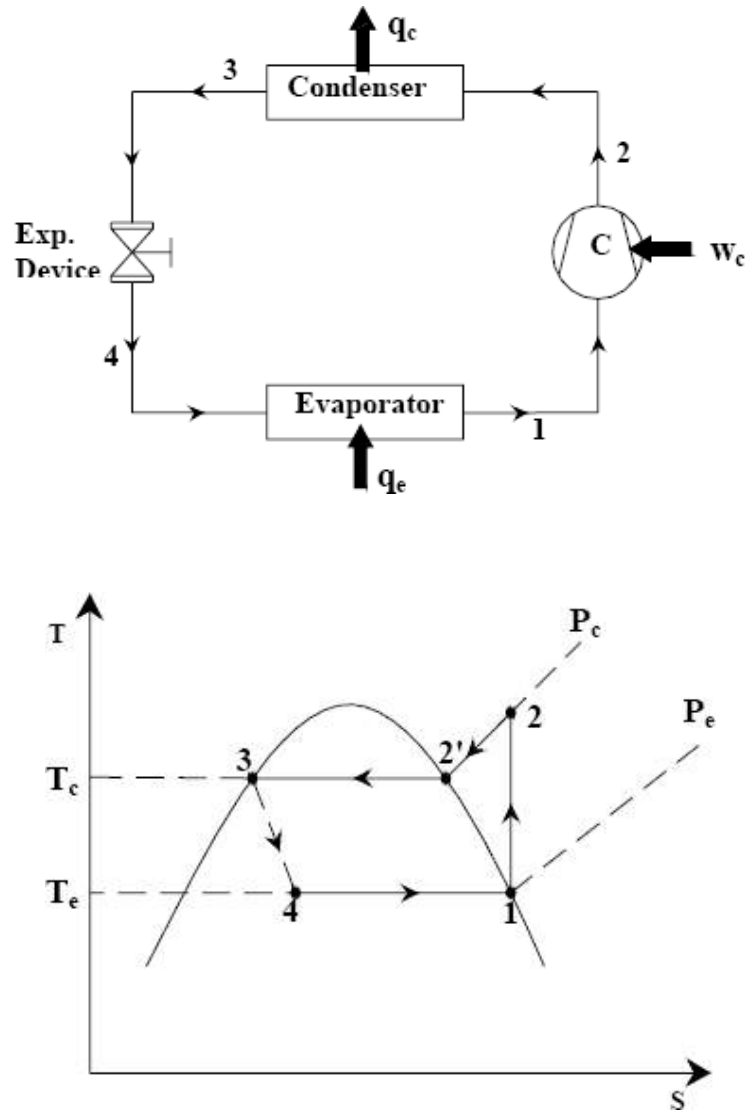


Fig.5. Standard Vapour compression refrigeration system

Standard Vapour Compression Refrigeration System (VCRS):

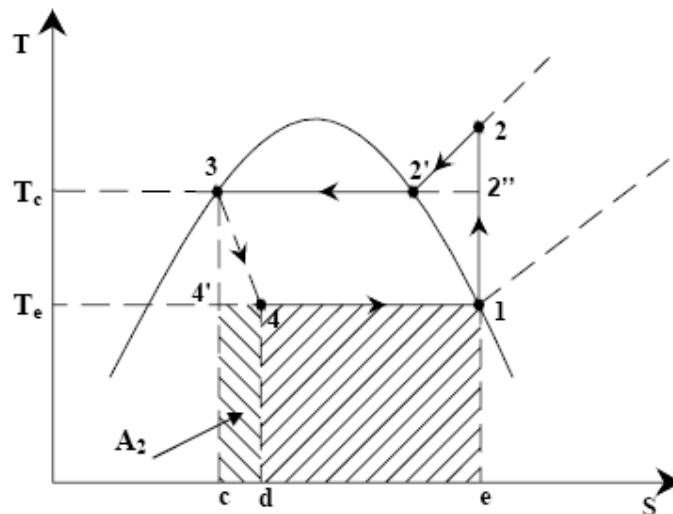
Figure 5 shows the schematic of a standard, saturated, single stage (SSS) vapour compression refrigeration system and the operating cycle on a T s diagram. As shown in the figure the standard single stage, saturated vapour compression refrigeration system consists of the following four processes:

Process 1-2: Isentropic compression of saturated vapour in compressor

Process 2-3: Isobaric heat rejection in condenser

Process 3-4: Isenthalpic expansion of saturated liquid in expansion device
 Process 4-1: Isobaric heat extraction in the evaporator

By comparing with Carnot cycle, it can be seen that the standard vapour compression refrigeration cycle introduces two irreversibilities: 1) Irreversibility due to non isothermal heat rejection (process 2-3) and 2) Irreversibility due to isenthalpic throttling (process 3- 4). As a result, one would expect the theoretical COP of standard cycle to be smaller than that of a Carnot system for the same heat source and sink temperatures. Due to these irreversibilities, the cooling effect reduces and work input increases, thus reducing the system COP. This can be explained easily with the help of the



cycle diagrams on T s charts. Figure 6(a) shows comparison between Carnot and standard VCRS in terms of refrigeration effect.

Fig. 6(a). Comparison between Carnot and standard VCRS

The heat extraction (evaporation) process is reversible for both the Carnot cycle and VCRS cycle. Hence the refrigeration effect is given by:

For Carnot refrigeration cycle (1-2''-3-4'):

$$q_{e,Carnot} = q_{4'-1} = \int_{4'}^1 T ds = T_e (s_1 - s_{4'}) = \text{area } e-1-4'-c-e \quad (7)$$

For VCRS cycle (1-2-3-4):

$$q_{e, \text{VCRS}} = q_{4-1} = \int_1^4 T_e ds = T_e (s_1 - s_4) = \text{area } e-1-4-d-e \quad (8)$$

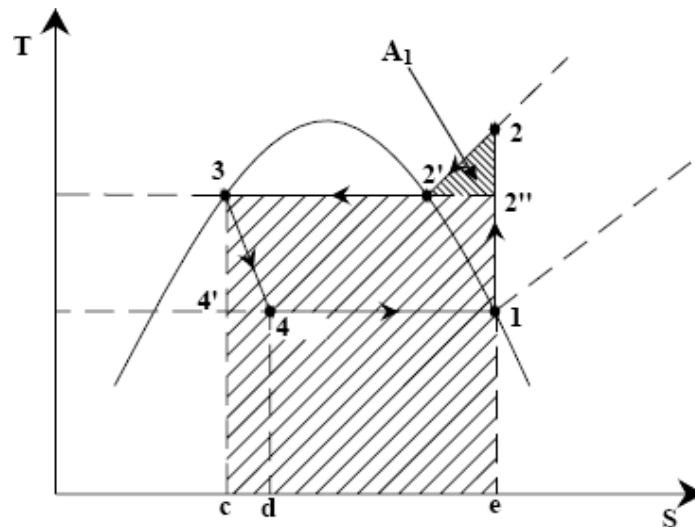
thus there is a reduction in refrigeration effect when the isentropic expansion process of Carnot cycle is replaced by isenthalpic throttling process of VCRS cycle, this

reduction is equal to the area $d-4-4'-c-d$ (area A_2) and is known as throttling loss. The throttling loss is equal to the enthalpy difference between state points 3 and 4', i.e.,

$$q_{e, \text{Carnot}} - q_{\text{VCRS}} = \text{area } d-4-4'-c-d = (h_3 - h_{4'}) = (h_4 - h_{4'}) = \text{area } A_2 \quad (9)$$

It is easy to show that the loss in refrigeration effect increases as the evaporator temperature decreases and/or condenser temperature increases. A practical consequence of this is a requirement of higher refrigerant mass flow rate.

The heat rejection in case of VCRS cycle also increases when compared to



Carnot cycle.

Fig. 6(b). Comparative evaluation of heat rejection rate of VCRS and Carnot cycle

As shown in Fig. 6(b), the heat rejection in case of Carnot cycle (1-2''-3-4') is given by:

$$q_{c, \text{Carnot}} = -q_{2''-3} = - \int_{2''}^3 T_c ds = T_c (s_2'' - s_3) = \text{area } e-2''-3-c-e \quad (10)$$

In case of VCRS cycle, the heat rejection rate is given by:

$$q_{c, \text{VCRS}} = -q_{2-3} = -\int_2^3 T \cdot ds = \text{area } e-2-3-c-e \quad (11)$$

Hence the increase in heat rejection rate of VCRS compared to Carnot cycle is equal to the area $2''-2-2'$ (area A_1). This region is known as *superheat horn*, and is due to the replacement of isothermal heat rejection process of Carnot cycle by isobaric heat

Since the heat rejection increases and refrigeration effect reduces when the Carnot cycle is modified to standard VCRS cycle, the net work input to the VCRS increases compared to Carnot cycle. The net work input in case of Carnot and VCRS cycles are given by: rejection in case of VCRS.

$$W_{\text{net, Carnot}} = (q_c - q_e)_{\text{Carnot}} = \text{area } 1-2'-3-4'-1$$

$$W_{\text{net, VCRS}} = (q_c - q_e)_{\text{VCRS}} = \text{area } 1-2-3-4'-c-d-4-1 \quad (12)$$

$$(13)$$

As shown in Fig.6(c), the increase in net work input in VCRS cycle is

$$W_{\text{net, VCRS}} - W_{\text{net, Carnot}} = \text{area } 2''-2-2' + \text{area } c-4'-4-d-c = \text{area } A_1 + \text{area } A_2$$

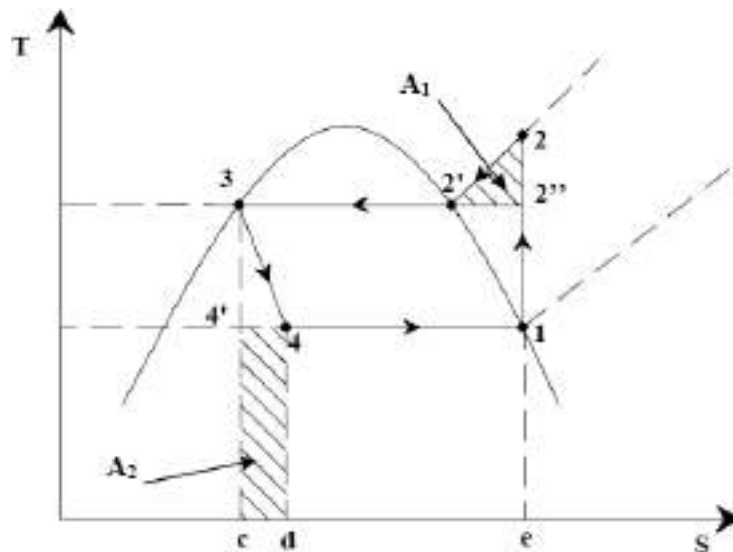


Fig. 6(c). Figure illustrating the increase in net work input in VCRS cycle

To summarize the refrigeration effect and net work input of VCRS cycle are given by:

$$q_{e, \text{VCRS}} = q_{e, \text{Carnot}} - \text{area } A_2 \quad (15)$$

$$W_{\text{net, VCRS}} = W_{\text{net, Carnot}} + \text{area } A_1 + \text{area } A_2$$

The COP of VCRS cycle is given by:

$$\text{COP}_{\text{VCRS}} = \frac{q_{e,\text{VCRS}}}{W_{\text{net,VCRS}}} = \frac{q_{e,\text{Carnot}} - \text{area } A_2}{W_{\text{net,Carnot}} + \text{area } A_1 + \text{area } A_2} \quad (17)$$

If we define the cycle efficiency, η_R as the ratio of COP of VCRS cycle to the COP of Carnot cycle, then:

$$\eta_R = \frac{\text{COP}_{\text{VCRS}}}{\text{COP}_{\text{Carnot}}} = \left[\frac{1 - \left(\frac{\text{area } A_2}{q_{e,\text{Carnot}}} \right)}{1 + \left(\frac{\text{area } A_1 + \text{area } A_2}{W_{\text{net,Carnot}}} \right)} \right] \quad (18)$$

The cycle efficiency (also called as second law efficiency) is a good indication of the deviation of the standard VCRS cycle from Carnot cycle. Unlike Carnot COP, the cycle efficiency depends very much on the shape of T s diagram, which in turn depends on the nature of the working fluid. we assume that the potential and kinetic energy changes during isentropic compression. If process 1-2 are negligible, then the work input w_{1-2} is given by:

$$w_{1-2,\text{VCRS}} = (h_2 - h_1) = (h_2 - h_f) - (h_1 - h_f)$$

(19)

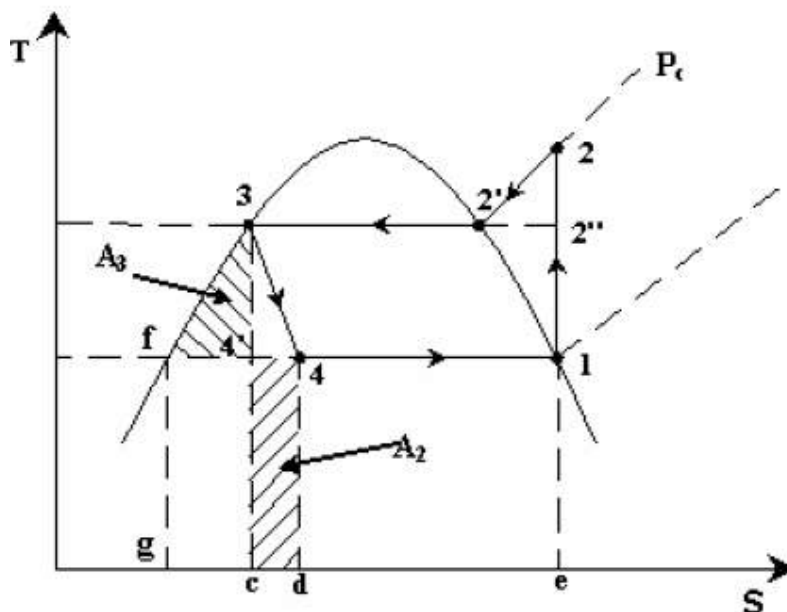


Fig.7. Figure showing saturated liquid line 3-f coinciding with the constant pressure line

Now as shown in Fig.7, if we further assume that the saturated liquid line 3-f coincides with the constant pressure line P_c in the subcooled region (which is a reasonably good assumption), then from the 2nd Tds relation;

$$Tds = dh - v dP = dh; \text{ where } p \text{ is constant}$$

$$\therefore (h_2 - h_f) = \int_2^f Tds = \text{area } e-2-3-f-g-e \quad (20)$$

$$\text{and, } (h_1 - h_f) = \int_1^f Tds = \text{area } e-1-f-g-e \quad (21)$$

Substituting these expressions in the expression for net work input, we obtain the compressor work input to be equal to area 1-2-3-f-1. Now comparing this with the earlier expression for work input (area 1-2-3-4'-c-d-4-1), we conclude that area A_2 is equal to area A_3 .

As mentioned before, the losses due to superheat (area A_1) and throttling (area $A_2 \approx A_3$) depend very much on the shape of the vapor dome (saturation liquid and vapour curves) on T s diagram. The shape of the saturation curves depends on the nature of refrigerant. Figure 8 shows T s diagrams for three different types of refrigerants

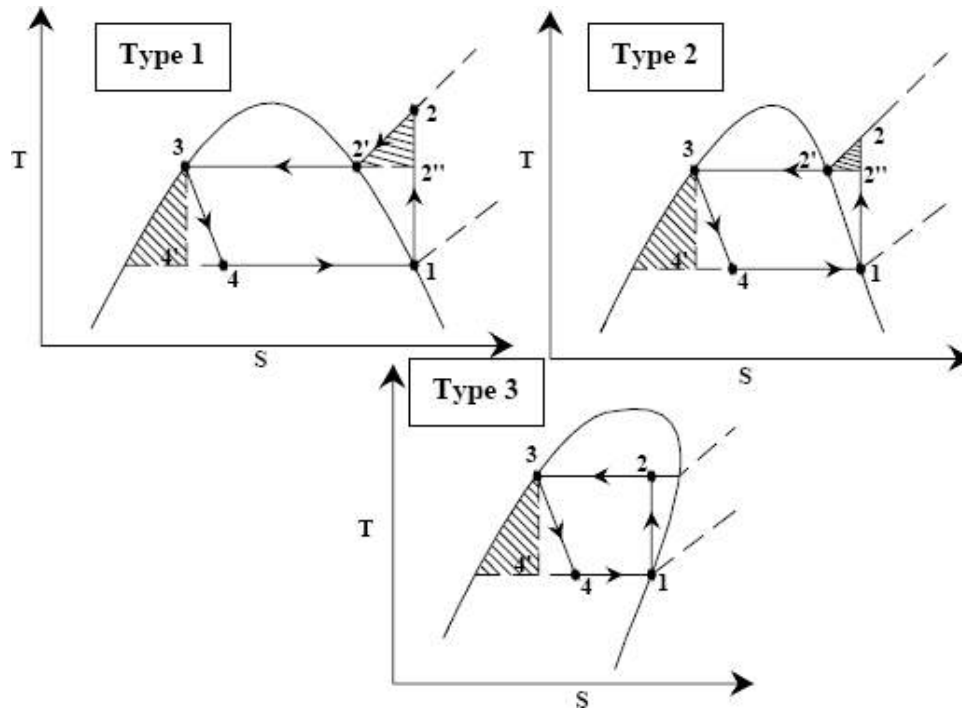


Fig. 8. T-s diagrams for three different types of refrigerants

Refrigerants such as ammonia, carbon di-oxide and water belong to Type 1. These refrigerants have symmetrical saturation curves (vapour dome), as a result both the superheat and throttling losses (areas A_1 and A_3) are significant. That means deviation of VCRS cycle from Carnot cycle could be significant when these refrigerants are used as working fluids. Refrigerants such as CFC11, CFC12, HFC134a belong to Type 2, these refrigerants have small superheat losses (area A_1) but large throttling losses (area A_3). High molecular weight refrigerants such as CFC113, CFC114, CFC115, iso-butane belonging to Type 3, do not have any superheat losses, i.e., when the compression inlet condition is saturated (point 1), then the exit condition will be in the 2-phase region, as a result it is not necessary to superheat the refrigerant. However, these refrigerants experience significant throttling losses. Since the compressor exit condition of Type refrigerants may fall in the two-phase region, there is a danger of wet compression leading to compressor damage. Hence for these refrigerants, the compressor inlet condition is chosen such that the exit condition does not fall in the two-phase region. This implies that the refrigerant at the inlet to the compressor should be superheated, the extent of which depends on the refrigerant.

Superheat and throttling losses:

It can be observed from the discus from the throttling loss. The superheat loss increases only the work input to the compressor, it does not effect the refrigeration effect. In heat pumps superheat is not a loss, but a part of the useful heating effect. However, the process of throttling is inherently irreversible, and it increases the workinput and also reduces the refrigeration effect.

Analysis of standard vapour compression refrigeration system:

pression refrigeration system can be carried out A simple analysis of standard vapour com by assuming a) Steady flow; b) negligible kinetic and potential energy changes across each component, and c) no heat transfer in connecting pipe lines. The steady flow energy equation is applied to each of the four components.

Evaporator: Heat transfer rate at evaporator or *refrigeration capacity*, Q_e is given by:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) \quad (22)$$

where is the refrigerant mass flow rate in kg/s, h_1 and h_4 are the specific enthalpies at t_r m (kJ/kg) he exit and inlet to the evaporator, respectively. $(h_1 - h_4)$ is known as specific refrigeration effect or simply *refrigeration effect*, which is equal to the heat

transferred at the evaporator per kilogram of refrigerant. The evaporator pressure P_e is the saturation pressure corresponding to evaporator temperature T_e , i.e.,

$$P_e = P_{sat}(T_e) \quad (23)$$

Compressor: Power input to the compressor, W_c is given by:

$$\dot{W}_c = \dot{m}_r (h_2 - h_1) \quad (24)$$

Where h_2 and h_1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, w respectively. $(h_2 - h_1)$ is known as specific work of compression or simply work of compression, which is equal to the work input to the compressor per kilogram of refrigerant.

Condenser: Heat transfer rate at condenser, Q_c is given by:

$$\dot{Q}_c = \dot{m}_r (h_2 - h_3) \quad (25)$$

where h_3 and h_2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively. The condense pressure P_c is the saturation pressure corresponding to evaporator temperature T_c , i.e.,

$$P_c = P_{sat}(T_c) \quad (26)$$

Expansion device: For the isenthalpic expansion across the expansion device could be considerable, however, if we take the control process, the kinetic energy change volume, well downstream of the expansion device, then the kinetic energy gets dissipated due to viscous effects, and

$$h_3 = h_4 \quad (27)$$

The exit condition of the expansion device lies the definition of quality (or dryness fraction), we can write: in the two-phase region, hence applying

$$h_4 = (1 - x_4)h_{f,e} + x_4 h_{g,e} = h_f + x_4 h_{fg} \quad (28)$$

where x_4 is the quality of refrigerant at point 4, $h_{f,e}$, $h_{g,e}$, h_{fg} enthalpy, saturated vapour enthalpy and latent heat of vaporization at evaporator are the saturated liquid pressure, respectively.

The COP of the system is given by:

$$\text{COP} = \left(\frac{\dot{Q}_e}{\dot{W}_c} \right) = \left(\frac{\dot{m}_r (h_1 - h_4)}{\dot{m}_r (h_2 - h_1)} \right) = \frac{(h_1 - h_4)}{(h_2 - h_1)} \quad (29)$$

At any point in the cycle, the mass flow rate of refrigerant \dot{m}_r can be written in terms of volumetric flow rate and specific volume at that point, i.e.,

$$\dot{m}_r = \frac{\dot{V}}{v}$$

applying this equation to the inlet condition of the compressor,

(30)

$$\dot{m}_r = \frac{\dot{V}_1}{v_1} \quad (31)$$

Where V_1 is the volumetric flow rate at compressor inlet. We can also write, the refrigeration capacity in terms of volumetric flow rate V_1 inlet and v_1 is the specific volume at compressor inlet. At a given compressor speed, V_1 is an indication of the size of the compressor as:

$$\dot{Q}_e = \dot{m}_r (h_1 - h_4) = \dot{V}_1 \left(\frac{h_1 - h_4}{v_1} \right) \quad (32)$$

$$\frac{h_1 - h_4}{v_1}$$

Where $\frac{h_1 - h_4}{v_1}$ is called as volumetric refrigeration effect (kJ/m³ of refrigerant).

Generally, the type of refrigerant, required refrigeration capacity, evaporator temperature and condenser temperature are known. Then from the evaporator and condenser temperature one can find the evaporator and condenser pressures and enthalpies at the exit of evaporator and condenser (saturated vapour enthalpy at evaporator pressure and saturated liquid enthalpy at condenser pressure). Since the exit condition of the compressor is in the superheated region, two independent properties are required to fix the state of refrigerant at this point. One of these independent properties could be the condenser pressure, which is already known. Since the compression process is isentropic, the entropy at the exit to the compressor is same as the entropy at the inlet, s_1 which is the saturated vapour entropy at evaporator pressure (known). Thus from the known pressure and entropy the exit state of the compressor could be fixed, i.e.,

$$h_2 = h(P_c, s_2) = h(P_c, s_1) \quad (33)$$

$$s_1 = s_2$$

The quality of refrigerant at the inlet to the evaporator (x_4) could be obtained from known values of h_3 , $h_{f,e}$ and $h_{g,e}$.

Once all the state points are known, then from the required refrigeration capacity and various enthalpies one can obtain the required refrigerant mass flow rate, volumetric flow rate at compressor inlet, COP, cycle efficiency etc.

Actual VCRS systems:

The cycles considered so far are internally reversible and no change of refrigerant state takes place in the connecting pipelines. However, in actual VCRS several irreversibilities exist. These are due to:

1. Pressure drops in evaporator, condenser and LSHX
2. Pressure drop across suction and discharge valves of the compressor
3. Heat transfer in compressor
4. Pressure drop and heat transfer in connecting pipe lines

Figure 1 shows the actual VCRS cycle on P-h and T-s diagrams indicating various irreversibilities. From performance point of view, the pressure drop in the evaporator, in the suction line and across the suction valve has a significant effect on system performance. This is due to the reason that as suction side pressure drop increases the specific volume at suction, compression ratio (hence volumetric efficiency) and discharge temperature increase. All these effects lead to reduction in system capacity, increase in power input and also affect the life of the compressor due to higher discharge temperature. Hence this pressure drop should be as small as possible for good performance. The pressure drop depends on the refrigerant velocity, length of refrigerant tubing and layout (bends, joints etc.). Pressure drop can be reduced by reducing refrigerant velocity (e.g. by increasing the inner diameter of the refrigerant tubes), however, this affects the heat transfer coefficient in evaporator. More importantly a certain minimum velocity is required to carry the lubricating oil back to the compressor for proper operation of the compressor.

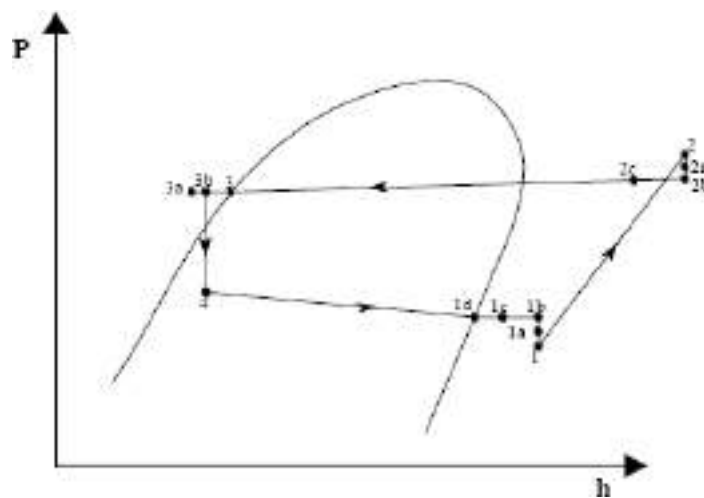
Heat transfer in the suction line is detrimental as it reduces the density of refrigerant vapour and increases the discharge temperature of the compressor. Hence, the suction lines are normally insulated to minimize heat transfer. In actual systems the compression process involves frictional effects and heat transfer. As a result, it cannot be reversible, adiabatic (even though it can be

isentropic).

In many cases cooling of the compressor is provided deliberately to maintain the maximum compressor temperature within safe limits. This is particularly true in case of refrigerants such as ammonia. Pressure drops across the valves of the compressor increase the work of compression and reduce the volumetric efficiency of the compressor. Hence they should be as small as possible.

Compared to the vapour lines, the system is less sensitive to pressure drop in the condenser and liquid lines. However, this also should be kept as low as possible. Heat transfer in the condenser connecting pipes is not detrimental in case of refrigeration systems. However, heat transfer in the subcooled liquid lines may affect the performance.

In addition to the above, actual systems are also different from the theoretical cycles due to the presence of foreign matter such as lubricating oil, water, air, particulate matter inside the system. The presence of lubricating oil cannot be avoided, however, the system design must ensure that the lubricating oil is carried over properly to the compressor. This depends on the miscibility of refrigerant-lubricating oil. Presence of other foreign materials such as air (non-condensing gas), moisture, particulate matter is detrimental to system performance. Hence systems are designed and operated such that the concentration of these materials is as low as possible.



(a)

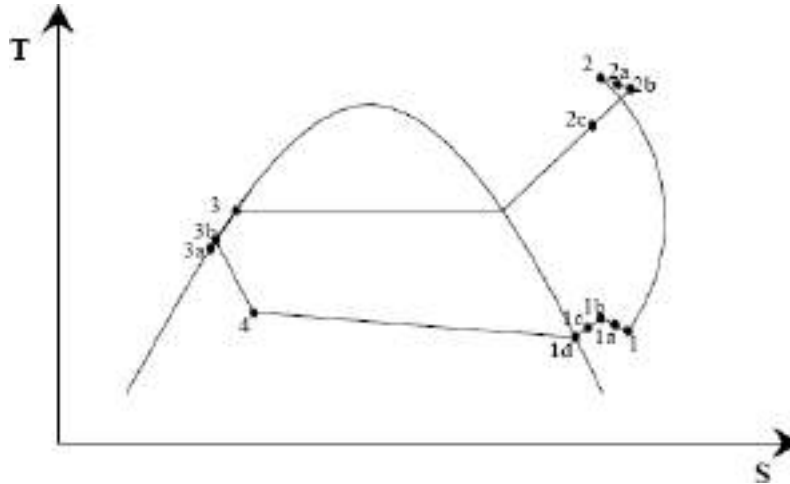


Fig. 1: Actual VCRC cycle on P-h and T-s diagrams

Process	State
Pressure drop in evaporator	4-1d
Superheat of vapour in evaporator	1d-1c
Useless superheat in suction line	1c-1b
Suction line pressure drop	1b-1a
Pressure drop across suction valve	1a-1
Non-isentropic compression	1-2
Pressure drop across discharge valve	2-2a
Pressure drop in the delivery line	2a-2b
Desuperheating of vapour in delivery pipe	2b-2c
Pressure drop in the condenser	2b-3
Subcooling of liquid refrigerant	3-3a
Heat gain in liquid line	3a-3b

The COP of actual refrigeration systems is sometimes written in terms of the COP of Carnot refrigeration system operating between the condensing and evaporator temperatures (COP_{Carnot}), cycle efficiency (η_{cyc}), isentropic efficiency of the

compressor (η_{is}) and efficiency of the electric motor (η_{motor}), as given by the equation shown below:

$$COP_{act} = \eta_{cyc} \eta_{is} \eta_{motor} COP_{Carnot} \quad (1)$$

An approximate expression for cycle efficiency (η_{cyc}) in the evaporator temperature range of $-50^{\circ}C$ to $+40^{\circ}C$ and condensing temperature range of

+10°C ton+60°C for refrigerants such as ammonia, R12 and R22 is suggested by Linge in 1966. This expression for a refrigeration cycle operating without ($\Delta T_{\text{sub}} = 0$) and with subcooling

($\Delta T_{\text{sub}} = T_c - T_{r,\text{exit}} > 0$ K) are given in Eqns. (2) and (3), respectively:

$$\eta_{\text{cyc}} = \left(1 - \frac{T_c - T_e}{265} \right) \text{ without subcooling} \quad (2)$$

$$\eta_{\text{cyc}} = \left(1 - \frac{T_c - T_e}{265} \right) \left(1 + \frac{\Delta T_{\text{sub}}}{250} \right) \text{ with subcooling} \quad (3)$$

In the above equations T_c and T_e are condensing and evaporator temperatures, respectively.

The isentropic efficiency of the compressor (η_{is}) depends on several factors such as the compression ratio, design of the compressor, nature of the working fluid etc. However, in practice its value generally lies between 0.5 to 0.8. The motor efficiency (η_{motor}) depends on the size and motor load. Generally the motor efficiency is maximum at full load. At full load its value lies around 0.7 for small motors and about 0.95 for large motors.

Complete vapour compression refrigeration systems:

In addition to the basic components, an actual vapour compression refrigeration consists of several accessories for safe and satisfactory functioning of the system. These include: compressor controls and safety devices such as overload protectors, high and low pressure cutouts, oil separators etc., temperature and flow controls, filters, driers, valves, sight glass etc. Modern refrigeration systems have automatic controls, which do not require continuous manual supervision.

Introduction:

The thermodynamic efficiency of a refrigeration system depends mainly on its operating temperatures. However, important practical issues such as the

system design, size, initial and operating costs, safety, reliability, and serviceability etc. depend very much on the type of refrigerant selected for a given application. Due to several environmental issues such as ozone layer depletion and global warming and their relation to the various refrigerants used, the selection of suitable refrigerant has become one of the most important issues in recent times. Replacement of an existing refrigerant by a completely new refrigerant, for whatever reason, is an expensive proposition as it may call for several changes in the design and manufacturing of refrigeration systems. Hence it is very important to understand the issues related to the selection and use of refrigerants. In principle, any fluid can be used as a refrigerant. Air used in an air cycle refrigeration system can also be considered as a refrigerant. However, in this lecture the attention is mainly focused on those fluids that can be used as refrigerants in vapour compression refrigeration systems only.

Primary and secondary refrigerants:

Fluids suitable for refrigeration purposes can be classified into primary and secondary refrigerants. Primary refrigerants are those fluids, which are used directly as working fluids, for example in vapour compression and vapour absorption refrigeration systems. When used in compression or absorption systems, these fluids provide refrigeration by undergoing a phase change process in the evaporator. As the name implies, secondary refrigerants are those liquids, which are used for transporting thermal energy from one location to other. Secondary refrigerants are also known under the name brines or antifreezes. Of course, if the operating temperatures are above 0°C , then pure water can also be used as secondary refrigerant, for example in large air conditioning systems. Antifreezes or brines are used when refrigeration is required at sub-zero temperatures. Unlike primary refrigerants, the secondary refrigerants do not undergo phase change as they transport energy from one location to other. An important property of a secondary refrigerant is its freezing point. Generally, the freezing point of a brine will be lower than the freezing point of its constituents. The temperature at which freezing of a brine takes place its depends on its concentration. The concentration at which

a lowest temperature can be reached without solidification is called as eutectic point. The commonly used secondary refrigerants are the solutions of water and ethylene glycol, propylene glycol or calcium chloride. These solutions are known under the general name of brines.

In this lecture attention is focused on primary refrigerants used mainly in vapour compression refrigeration systems. As discussed earlier, in an absorption refrigeration system, a refrigerant and absorbent combination is used as the working fluid.

Refrigerant selection criteria:

Selection of refrigerant for a particular application is based on the following requirements:

Thermodynamic and thermo-physical properties:

The requirements are:

- a) Suction pressure: At a given evaporator temperature, the saturation pressure should be above atmospheric for prevention of air or moisture ingress into the system and ease of leak detection. Higher suction pressure is better as it leads to smaller compressor displacement
 - b) Discharge pressure: At a given condenser temperature, the discharge pressure should be as small as possible to allow light-weight construction of compressor, condenser etc.
 - c) Pressure ratio: Should be as small as possible for high volumetric efficiency and low power consumption
 - d) Latent heat of vaporization: Should be as large as possible so that the required massflow rate per unit cooling capacity will be small
- The above requirements are somewhat contradictory, as the operating pressures, temperatures and latent heat of vaporization are related by Clausius- Clapeyron Equation:

In the above $\ln(P_{sat}) = -\frac{h_{fg}}{RT} + \frac{s_{fg}}{R}$ saturation pressure (in atm.) at a temperature T , h_{fg} and s_{fg} are enthalpy and entropy of vaporization and R is the gas constant. Since the change in entropy of

vaporization is relatively small, from the above equation it can be shown that:

In the above equation, P_c and P_e are the condenser and evaporator pressures, T_c and T_e are condenser and evaporator temperatures. From the above equation, it can be seen that for given condenser and evaporator temperatures as the latent heat of vaporization increases, the pressure ratio also increases. Hence a trade-off is required between the latent heat of vaporization and pressure ratio.

In addition to the above properties; the following properties are also important:

- e) Isentropic index of compression: Should be as small as possible so that the temperature rise during compression will be small
- f) Liquid specific heat: Should be small so that degree of subcooling will be large leading to smaller amount of flash gas at evaporator inlet
- g) Vapour specific heat: Should be large so that the degree of superheating will be small
- h) Thermal conductivity: Thermal conductivity in both liquid as well as vapour phase should be high for higher heat transfer coefficients
- i) Viscosity: Viscosity should be small in both liquid and vapour phases for smaller frictional pressure drops

The thermodynamic properties are interrelated and mainly depend on normal boiling point, critical temperature, molecular weight and structure. The normal boiling point indicates the useful temperature levels as it is directly related to the operating pressures. A high critical temperature yields higher COP due to smaller compressor superheat and smaller flash gas losses. On the other hand since the vapour pressure will be low when critical temperature is high, the volumetric capacity will be lower for refrigerants with high critical temperatures. This once again shows a need for trade-off between high COP and high volumetric capacity. It is observed that for most of the refrigerants the ratio of normal boiling point to critical temperature is in

the

range of 0.6 to 0.7. Thus the normal boiling point is a good indicator of the critical temperature of the refrigerant.

The important properties such as latent heat of vaporization and specific heat depend on the molecular weight and structure of the molecule. Trouton's rule shows that the latent heat of vaporization will be high for refrigerants having lower molecular weight. The specific heat of refrigerant is related to the structure of the molecule. If specific heat of refrigerant vapour is low then the shape of the vapour dome will be such that the compression process starting with a saturated point terminates in the superheated zone (**i.e., compression process will be dry**). However, a small value of vapour specific heat indicates higher degree of superheat. Since vapour and liquid specific heats are also related, a large value of vapour specific heat results in a higher value of liquid specific heat, leading to higher flash gas losses. Studies show that in general the optimum value of molar vapour specific heat lies in the range of **40 to 100 kJ/kmol.K**.

The freezing point of the refrigerant should be lower than the lowest operating temperature of the cycle to prevent blockage of refrigerant pipelines.

Environmental and safety properties:

Next to thermodynamic and thermophysical properties, the environmental and safety properties are very important. In fact, at present the environment friendliness of the refrigerant is a major factor in deciding the usefulness of a particular refrigerant. The important environmental and safety properties are:

- a) Ozone Depletion Potential (ODP): According to the Montreal protocol, the ODP of refrigerants should be zero, i.e., they should be non-ozone depleting substances. Refrigerants having non-zero ODP have either already been phased-out (e.g. R 11, R 12) or will be phased-out in near-future (e.g. R22). Since ODP depends mainly on the presence of chlorine or bromine in the molecules, refrigerants having either chlorine (i.e., CFCs and HCFCs) or bromine cannot be used under the new regulations
- b) Global Warming Potential (GWP): Refrigerants should have as low a GWP value

as possible to minimize the problem of global warming. Refrigerants with zero ODP but a high value of GWP (e.g. R134a) are likely to be regulated in future.

c) Total Equivalent Warming Index (TEWI): The factor TEWI considers both direct (due to release into atmosphere) and indirect (through energy consumption) contributions of refrigerants to global warming. Naturally, refrigerants with as a low avalue of TEWI are preferable from global warming point of view.

d) Toxicity: Ideally, refrigerants used in a refrigeration system should be nontoxic. However, all fluids other than air can be called as toxic as they will cause suffocation when their concentration is large enough. Thus toxicity is a relative term, which becomes meaningful only when the degree of concentration and time of exposure required to produce harmful effects are specified. Some fluids are toxic even in small concentrations. Some fluids are mildly toxic, i.e., they are dangerous only when the concentration is large and duration of exposure is long. Some refrigerants such as CFCs and HCFCs are non-toxic when mixed with air in normal condition. However, when they come in contact with an open flame or an electrical heating element, they decompose forming highly toxic elements (e.g. phosgene- COCl_2). In general the degree of hazard depends on:

- Amount of refrigerant used vs total space
- Type of occupancy
- Presence of open flames
- Odor of refrigerant, and
- Maintenance condition

Thus from toxicity point-of-view, the usefulness of a particular refrigerant depends on the specific application.

e) Flammability: The refrigerants should preferably be non-flammable and non-explosive. For flammable refrigerants special precautions should be taken to avoid accidents. Based on the above criteria, ASHRAE has divided refrigerants into six safety groups (A1 to A3 and B1 to B3). Refrigerants belonging to Group A1 (e.g. R11, R12, R22, R134a, R744, R718) are least hazardous, while refrigerants belonging to Group B3 (e.g. R1140) are most hazardous. Other important properties are:

f) Chemical stability: The refrigerants should be chemically stable as long as they are inside the refrigeration system.

- g) Compatibility with common materials of construction (both metals and nonmetals)
- h) Miscibility with lubricating oils: Oil separators have to be used if the refrigerant is not miscible with lubricating oil (e.g. ammonia). Refrigerants that are completely miscible with oils are easier to handle (e.g. R12). However, for refrigerants with limited solubility (e.g. R 22) special precautions should be taken while designing the system to ensure oil return to the compressor
- i) Dielectric strength: This is an important property for systems using hermetic compressors. For these systems the refrigerants should have as high a dielectric strength as possible
- j) Ease of leak detection: In the event of leakage of refrigerant from the system, it should be easy to detect the leaks.

Economic properties:

The refrigerant used should preferably be inexpensive and easily available.

Designation of refrigerants:

Figure 1 shows the classification of fluids used as refrigerants in vapour compression refrigeration systems. Since a large number of refrigerants have been developed over the years for a wide variety of applications, a numbering system has been adopted to designate various refrigerants. From the number one can get some useful information about the type of refrigerant, its chemical composition, molecular weight etc. All the refrigerants are designated by R followed by a unique number.

i) Fully saturated, halogenated compounds: These refrigerants are derivatives of alkanes (C_nH_{2n+2}) such as methane (CH_4), ethane (C_2H_6). These refrigerants are designated by R XYZ, where:

X+1 indicates the number of Carbon (C) atoms

Y-1 indicates number of Hydrogen (H) atoms,

and Z indicates number of Fluorine (F) atoms

The balance indicates the number of Chlorine atoms. Only 2 digits indicates that the value of X is zero.

Ex: R 22

$X = 0 \Rightarrow$ No. of Carbon atoms = $0+1 = 1 \Rightarrow$ derivative of methane

$(CH_4)Y = 2 \Rightarrow$ No. of Hydrogen atoms = $2-1 = 1$

$Z = 2 \Rightarrow$ No. of Fluorine atoms = 2

The balance = $4 - \text{no. of (H+F) atoms} = 4-1-2 = 1 \Rightarrow$ No. of Chlorine atoms = 1

\therefore The chemical formula of R 22 = $CHClF_2$

Similarly it can be shown that the chemical formula of:

R12 = CCl_2F_2

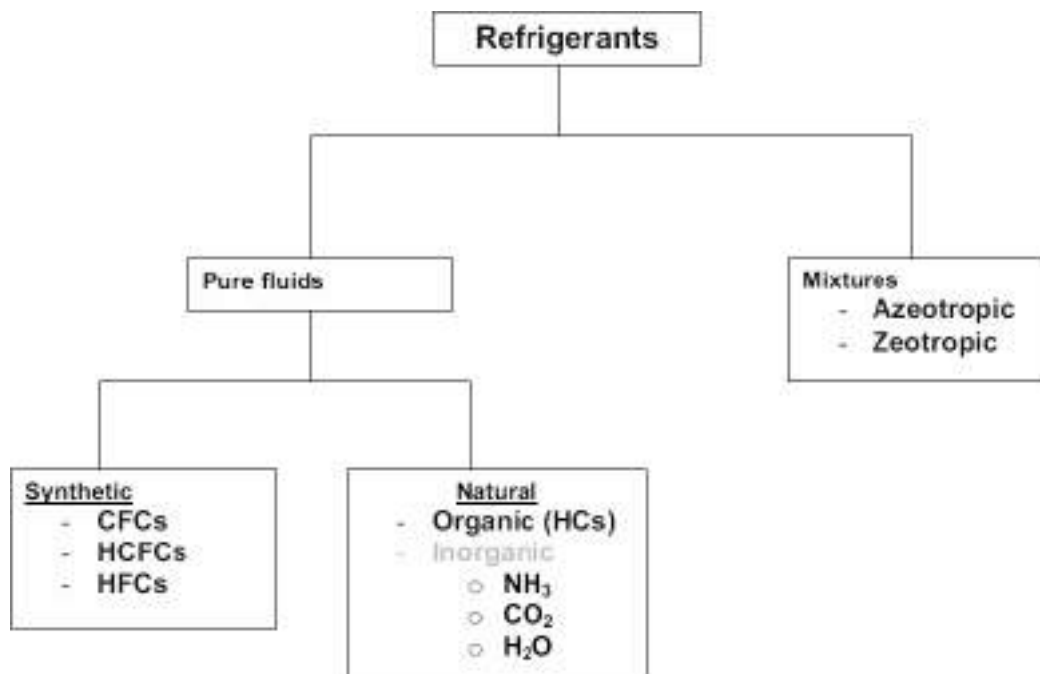
R134a = $C_2H_2F_4$ (derivative of ethane)

(letter a stands for isomer, e.g. molecules having same chemical composition but different atomic arrangement, e.g. R134 and R134a)

ii) Inorganic refrigerants: These are designated by number 7 followed by the molecular weight of the refrigerant (rounded-off).

Ex.: Ammonia: Molecular weight is 17, \therefore the designation is R

717 Carbon dioxide: Molecular weight is 44, \therefore the designation is



R 744 Water: Molecular weight is 18, \therefore the designation is R 718

Fig.1: Classification of fluids used as refrigerants

iii) Mixtures: Azeotropic mixtures are designated by 500 series, where as zeotropic refrigerants (e.g. non-azeotropic mixtures) are designated by 400 series.

Azeotropic mixtures:

R 500: Mixture of R 12 (73.8 %) and R 152a (26.2%)

R 502: Mixture of R 22 (48.8 %) and R 115 (51.2%)

R503: Mixture of R 23 (40.1 %) and R 13 (59.9%)

R507A: Mixture of R 125 (50%) and R 143a (50%)

Zeotropic mixtures:

R404A : Mixture of R 125 (44%), R 143a (52%) and R 134a

(4%) R407A : Mixture of R 32 (20%), R 125 (40%) and R 134a (40%)

R407B : Mixture of R 32 (10%), R 125 (70%) and R 134a (20%)

R410A : Mixture of R 32 (50%) and R 125 (50%)

iv) Hydrocarbons:

Propane (C₃H₈) : R

290 n-butane (C₄H₁₀) :

R 600

iso-butane (C₄H₁₀) : R 600a

Unsaturated Hydrocarbons: R1150 (C₂H₄)

R1270 (C₃H₆)

Comparison between different refrigerants:

Synthetic refrigerants that were commonly used for refrigeration, cold storage and air conditioning applications are: R 11 (CFC 11), R 12 (CFC 12), R 22 (HCFC 22), R 502 (CFC 12+HCFC 22) etc. However, these refrigerants have to be phased out due to their Ozone Depletion Potential (ODP). The synthetic replacements for the older refrigerants are: R-134a (HFC-134a) and blends of HFCs. Generally, synthetic refrigerants are non-toxic and non-flammable. However, compared to the natural refrigerants the synthetic refrigerants offer lower performance and they also have higher Global Warming Potential (GWP). As a result, the synthetic refrigerants face an uncertain future. The most commonly used natural refrigerant is ammonia. This is also one of the oldest known refrigerants. Ammonia has good thermodynamic, thermophysical and environmental properties. However, it is toxic and is not

compatible with some of the common materials of construction such as copper, which somewhat restricts its application. Other natural refrigerants that are being suggested are hydrocarbons (HCs) and carbon dioxide (R-744). Though these refrigerants have

some specific problems owing to their eco-friendliness, they are being studied widely and are likely to play a prominent role in future.

Prior to the environmental issues of ozone layer depletion and global warming, the most widely used refrigerants were: R 11, R 12, R 22, R 502 and ammonia. Of these, R 11 was primarily used with centrifugal compressors in air conditioning applications. R 12 was used primarily in small capacity refrigeration and cold storage applications, while the other refrigerants were used in large systems such as large air conditioning plants or cold storages. Among the refrigerants used, except ammonia, all the other refrigerants are synthetic refrigerants and are non-toxic and non-flammable. Though ammonia is toxic, it has been very widely used due to its excellent thermodynamic and thermophysical properties. The scenario changed completely after the discovery of ozone layer depletion in 1974. The depletion of stratospheric ozone layer was attributed to chlorine and bromine containing chemicals such as Halons, CFCs, HCFCs etc. Since ozone layer depletion could lead to catastrophe on a global level, it has been agreed by the global community to phase out the ozone depleting substances (ODS). As a result except ammonia, all the other refrigerants used in cold storages had to be phased-out and a search for suitable replacements began in earnest. At the same time, it was also observed that in addition to ozone layer depletion, most of the conventional synthetic refrigerants also cause significant global warming. In view of the environmental problems caused by the synthetic refrigerants, opinions differed on replacements for conventional refrigerants. The alternate refrigerants can be classified into two broad groups:

- i) Non-ODS, synthetic refrigerants based on Hydro-Fluoro-Carbons (HFCs) and their blends
- ii) Natural refrigerants including ammonia, carbon dioxide, hydrocarbons and

their blends

It should be noted that the use of natural refrigerants such as carbon dioxide, hydrocarbons is not a new phenomena, but is a revival of the once-used- and discarded technologies in a much better form. Since the natural refrigerants are essentially making a comeback, one advantage of using them is that they are familiar in terms of their strengths and weaknesses. Another important advantage is that they are completely environment friendly, unlike the HFC based refrigerants, which do have considerable global warming potential. The alternate synthetic refrigerants are normally non-toxic and non-flammable. It is also possible to use blends of various HFCs to obtain new refrigerant mixtures with required properties to suit specific applications. However, most of these blends are non-azeotropic in nature, as a result there could be significant temperature glides during evaporation and condensation, and it is also important take precautions to prevent leakage, as this will change the composition of the mixture. Table 1 shows a list of refrigerants being replaced and their replacements.

Refrigerant	Application	Substitute suggested dRetrofit(R)/New (N)
R 11(CFC) NBP = 23.7°C h_{ig} at NBP=182.5 kJ/kg T_{cr} =197.98°C C_p/C_v = 1.13 ODP = 1.0 GWP = 3500	Large air conditioning systems	R 123 (R,N)
	Industrial heat pumps	R 141b (N)
	As foam blowing agent	R 245fa (N)
		n-pentane (R,N)
R 12 (CFC) NBP = -29.8°C h_{ig} at NBP=165.8 kJ/kg T_{cr} =112.04°C C_p/C_v = 1.126 ODP = 1.0 GWP = 7300	Domestic refrigerators	R 22 (R,N)
	Small air conditioners	R 134a (R,N)
	Water coolers	R 227ea (N)
	Small cold storages	R 401A,R 401B (R,N)
		R 411A,R 411B (R,N)
		R 717 (N)

R 22 (HCFC) NBP = -40.8°C h_{fg} at NBP = 233.2 kJ/kg T_{cr} = 96.02°C C_p/C_v = 1.166 ODP = 0.05 GWP = 1500	Air conditioning systems	R 410A
	Cold storages	R 417A (R,N)
		R 407C (R,N)
		R 507, R 507A (R,N)
		R 404A (R,N)
		R 717 (N)
R 134a (HFC) NBP = -26.15°C h_{fg} at NBP = 222.5 kJ/kg T_{cr} = 101.06°C C_p/C_v = 1.102 ODP = 0.0 GWP = 1200	Used as replacement for R 12 in domestic refrigerators, water coolers, automobile A/Cs etc	No replacement required * Immiscible in mineral oils * Highly hygroscopic
R 717 (NH₃) NBP = -33.35°C	Cold storages Ice plants	No replacement required * Toxic and flammable

h_{fg} at NBP=1368.9 kJ/kg T_{cr} =133.0°C C_p/C_v = 1.31 ODP = 0.0 GWP = 0.0	Food processing Frozen food cabinets	* Incompatible with copper * Highly efficient * Inexpensive and available
R 744 (CO₂) NBP = -78.4°C h_{fg} at 40°C=321.3 kJ/kg T_{cr} =31.1°C C_p/C_v = 1.3 ODP = 0.0 GWP = 1.0	Cold storages Air conditioning systems Simultaneous cooling and heating (Transcritical cycle)	No replacement required * Very low critical temperature * Eco-friendly * Inexpensive and available

Table 1: Refrigerants, their applications and substitutes

Refrigerant	Application	Substitute suggested Retrofit(R)/New (N)
R718 (H₂O) NBP = 100.0 C h_{fg} at NBP=2257.9 kJ/kg T_{cr} =374.15°C C_p/C_v = 1.33 ODP = 0.0 GWP = 1.0	Absorption systems Steam jet systems	No replacement required * High NBP * High freezing point * Large specific volume * Eco-friendly * Inexpensive and available
R600a (iso-butane) NBP = -11.73°C h_{fg} at NBP=367.7 kJ/kg T_{cr} =135.0°C C_p/C_v = 1.086 ODP = 0.0 GWP = 3.0	Replacement for R 12 Domestic refrigerators Water coolers	No replacement required * Flammable * Eco-friendly

Table 1: Refrigerants, their applications and substitutes (contd.)

Unit -III

Objective:

To impart knowledge on different types of refrigeration systems.

Outcome:

Ability to demonstrate the working of refrigeration equipment.

Vapour Absorption Refrigeration (VAR) System: Description and working of NH₃ – water system and Li Br – water (Two shell & four shell) system - Calculation of max COP.

Principle and operation of Three Fluid absorption refrigeration system.

Steam Jet Refrigeration System: Working Principle and Basic Components, Principle and operation of i) Thermo-Electric Refrigerator, ii) Vortex Tube or Hilsch tube.

Introduction

Vapour Absorption Refrigeration Systems (VARs) belong to the class of vapour cycles similar to vapour compression refrigeration systems. However, unlike vapour compression refrigeration systems, the required input to absorption systems is in the form of heat. Hence these systems are also called as heat operated or thermal energy driven systems. Since conventional absorption systems use liquids for absorption of refrigerant, these are also sometimes called as wet absorption systems. Similar to vapour compression refrigeration systems, vapour absorption refrigeration systems have also been commercialized and are widely used in various refrigeration and air conditioning applications. Since these systems run on low-grade thermal energy, they are preferred when low-grade energy such as waste heat or solar energy is available. Since conventional absorption systems use natural refrigerants such as water or ammonia they are environment friendly.

In this lesson, the basic working principle of absorption systems, the maximum COP of ideal absorption refrigeration systems, basics of properties of mixtures and simple absorption refrigeration systems will be discussed.

Basic principle

When a solute such as lithium bromide salt is dissolved in a solvent such as water, the boiling point of the solvent (water) is elevated. On the other hand, if the temperature of the solution (solvent + solute) is held constant, then the effect of dissolving the solute is to reduce the vapour pressure of the solvent below that of the saturation pressure of pure solvent at that temperature. If the solute itself has some vapour pressure (i.e., volatile solute) then the total pressure exerted over the solution is the sum total of the partial pressures of solute and solvent. If the solute is nonvolatile (e.g. lithium bromide salt) or if the boiling point difference between the solution and solvent is large ($\geq 300^{\circ}\text{C}$), then the total pressure exerted over the solution will be almost equal to the vapour pressure of the solvent only. In the simplest absorption refrigeration system, refrigeration is obtained by connecting two vessels, with one vessel containing pure solvent and the other containing a solution. Since the pressure is almost equal in both the vessels at equilibrium, the temperature of the solution will be higher than that of the pure solvent. This means that if the solution is at ambient temperature, then the pure solvent will be at a temperature lower than the ambient.

Hence refrigeration effect is produced at the vessel containing pure solvent due to this temperature difference. The solvent evaporates due to heat transfer from the surroundings, flows to the vessel containing solution and is absorbed by the solution. This process is continued as long as the composition and temperature of the solution are maintained and liquid solvent is available in the container.

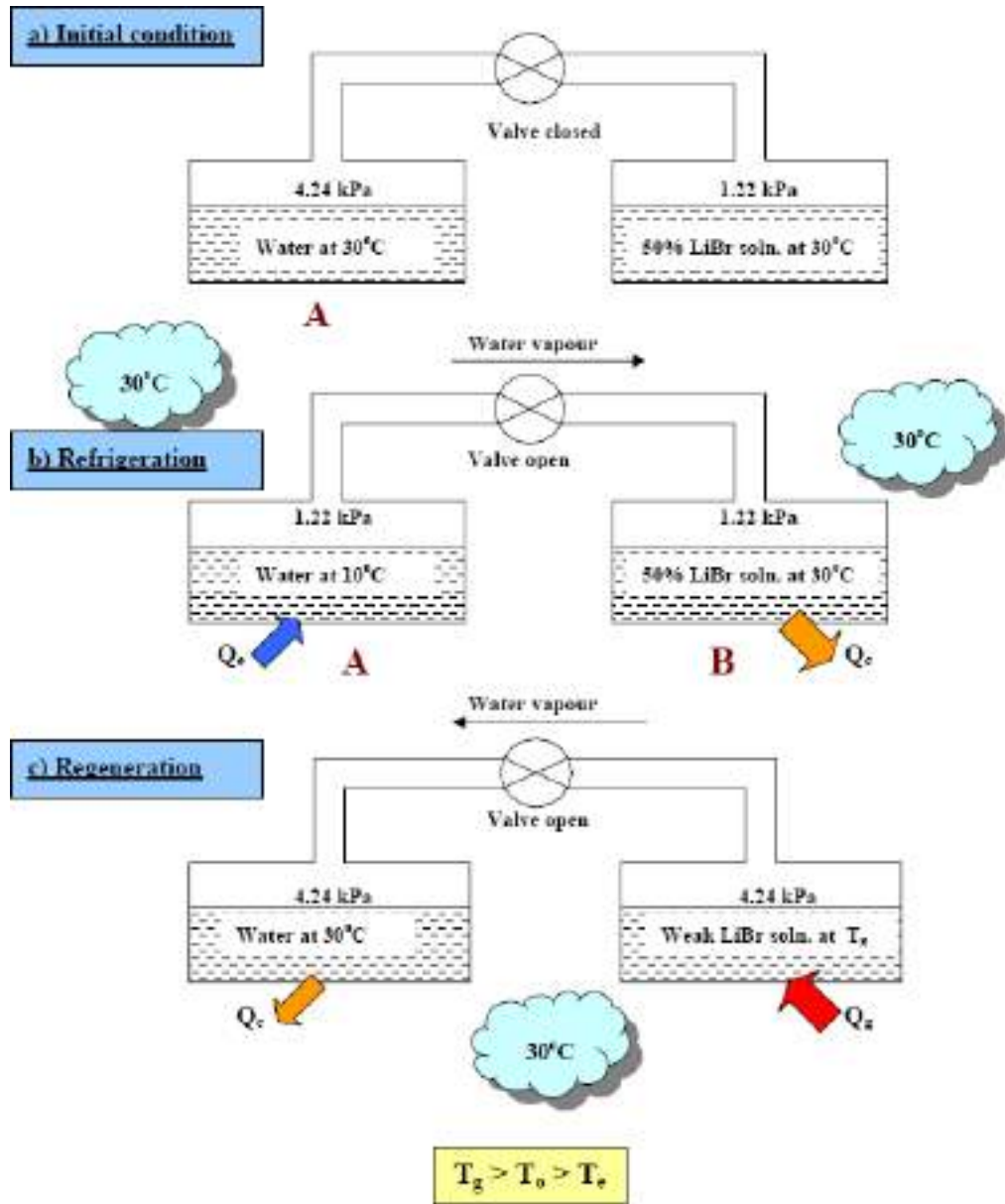


Fig1: Basic principle of vapour absorption systems

For example, Fig.1 shows an arrangement, which consists of two vessels A and B connected to each other through a connecting pipe and a valve. Vessel A is filled with pure water, while vessel B is filled with a solution containing on mass basis 50 percent of water and 50 percent lithium bromide (LiBr salt). Initially the valve connecting these two vessels is closed, and both vessels are at thermal equilibrium with the surroundings, which is at 30oC. At 30oC, the saturation pressure of water is 4.24 kPa, and the equilibrium vapour pressure of water-lithium bromide solution (50 : 50 by mass) at 30oC is 1.22 kPa.

Thus at initial equilibrium condition, the pressure in vessel A is 4.24 kPa, while it is

1.22 kPa in vessel B. Now the valve between vessels A and B is opened. Initially due to pressure difference water vapour will flow from vessel A to vessel B, and this vapour will be absorbed by the solution in vessel B. Since absorption in this case is exothermic, heat will be released in vessel B. Now suppose by some means the concentration and temperature of vessel B are maintained constant at 50 % and 30oC, respectively. Then at equilibrium, the pressure in the entire system (vessels A and B) will be 1.22 kPa (equilibrium pressure of 50 % LiBr solution at 30oC). The temperature of water in vessel A will be the saturation temperature corresponding to

1.22 kPa, which is equal to about 10oC, as shown in the figure. Since the water temperature in A is lower than the surroundings, a refrigeration effect (Q_e) can be produced by transferring heat from the surroundings to water at 10oC. Due to this heat transfer, water vaporizes in A, flows to B and is absorbed by the solution in B. The exothermic heat of absorption (Q_a) is rejected to the surroundings.

Now for the above process to continue, there should always be pure water in vessel A, and vessel B must be maintained always at 50 percent concentration and 30oC. This is not possible in a closed system such as the one shown in Fig.14.1. In a closed system with finite sized reservoirs, gradually the amount of water in A decreases and the solution in B becomes diluted with water. As a result, the system pressure and temperature of water in A increase with time. Hence the refrigeration effect at A reduces gradually due to the reduced

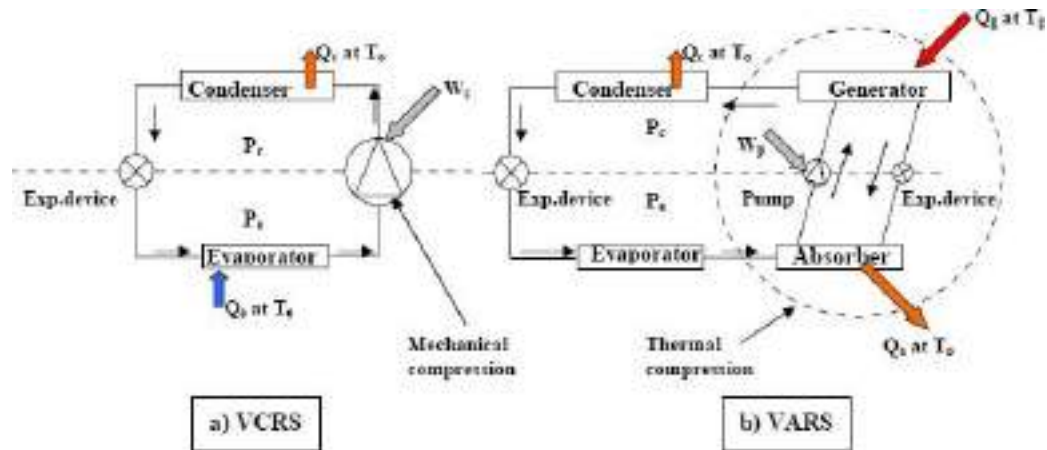
temperature difference between the surroundings and water. Thus refrigeration produced by systems using only two vessels is intermittent in nature. In these systems, after a period, the refrigeration process has to be stopped and both the vessels A and B have to be brought back to their original condition. This requires removal of water absorbed in B and adding it back to vessel A in liquid form, i.e., a process of regeneration as shown in Fig.1(c).

Assume that before regeneration is carried out, the valve between A and B is closed and both A and B are brought in thermal equilibrium with the surroundings (30°C), then during the regeneration process, heat at high temperature T_g is supplied to the dilute LiBr solution in B, as a result water vapour is generated in B. The vapour generated in B is condensed into pure water in A by rejecting heat of condensation to the surroundings. This process has to be continued till all the water absorbed during the refrigeration process (14.1(b)) is transferred back to A. Then to bring the system back to its original condition, the valve has to be closed and solution in vessel B has to be cooled to 30°C. If we assume a steady-flow process of regeneration and neglect temperature difference for heat transfer, then the temperature of water in A will be 30°C and pressure inside the system will be 4.24 kPa. Then the temperature in vessel B, T_g depends on the concentration of solution in B. The amount of heat transferred during refrigeration and regeneration depends on the properties of solution and the operating conditions. It can be seen that the output from this system is the refrigeration obtained Q_e and the input is heat supplied to vessel B during vapour regeneration process, Q_g .

The system described may be called as an Intermittent Absorption Refrigeration System. The solvent is the refrigerant and the solute is called as absorbent. These simple systems can be used to provide refrigeration using renewable energy such as solar energy in remote and rural areas. As already explained, these systems provided refrigeration intermittently, if solar energy is used for regenerating the refrigerant, then regeneration process can be carried out during the day and refrigeration can be produced during the night.

Though the intermittent absorption refrigeration systems discussed above are simple in design and inexpensive, they are not useful in applications that require

continuous refrigeration. Continuous refrigeration can be obtained by having a modified system with two pairs of vessels A and B and additional expansion valves and a solution pump.



Figs.2: (a) Vapour compression refrigeration system (VCRS)
 (b) Vapour Absorption Refrigeration System (VARS)

Figure 2(a) and (b) show a continuous output vapour compression refrigeration system and a continuous output vapour absorption refrigeration system. As shown in the figure in a continuous absorption system, low temperature and low pressure refrigerant with low quality enters the evaporator and vaporizes by producing useful refrigeration Q_e . From the evaporator, the low temperature, low pressure refrigerant vapour enters the absorber where it comes in contact with a solution that is weak in refrigerant. The weak solution absorbs the refrigerant and becomes strong in refrigerant. The heat of absorption is rejected to the external heat sink at T_o . The solution that is now rich in refrigerant is pumped to high pressure using a solution pump and fed to the generator. In the generator heat at high temperature T_g is supplied, as a result refrigerant vapour is generated at high pressure. This high pressure vapour is then condensed in the condenser by rejecting heat of condensation to the external heat sink at T_o . The condensed refrigerant liquid is then throttled in the expansion device and is then fed to the evaporator to complete the refrigerant cycle. On the solution side, the hot, high-pressure solution that is weak in refrigerant is throttled to the absorber pressure in the solution

expansion valve and fed to the absorber where it comes in contact with the refrigerant vapour from evaporator. Thus continuous refrigeration is produced at evaporator, while heat at high temperature is continuously supplied to the generator. Heat rejection to the external heat sink takes place at absorber and condenser. A small amount of mechanical energy is required to run the solution pump. If we neglect pressure drops, then the absorption system operates between the condenser and evaporator pressures. Pressure in absorber is same as the pressure in evaporator and pressure in generator is same as the pressure in condenser.

It can be seen from Fig.2, that as far as the condenser, expansion valve and evaporators are concerned both compression and absorption systems are identical. However, the difference lies in the way the refrigerant is compressed to condenser pressure. In vapour compression refrigeration systems the vapour is compressed mechanically using the compressor, whereas in absorption system the vapour is first converted into a liquid and then the liquid is pumped to condenser pressure using the solution pump. Since for the same pressure difference, work input required to pump a liquid (solution) is much less than the work required for compressing a vapour due to

$$(w = - \int v \, dP)$$

very small specific volume of liquid the mechanical energy required to operate vapour absorption refrigeration system is much less than that required to operate a compression system. However, the absorption system requires a relatively large amount of low-grade thermal energy at generator temperature to generate refrigerant vapour from the solution in generator. Thus while the energy input is in the form of mechanical energy in vapour compression refrigeration systems, it is mainly in the form of thermal energy in case of absorption systems. The solution pump work is often negligible compared to the generator heat input. Thus the COPs for compression and absorption systems are given by:

$$\text{COP}_{\text{VCRS}} = \frac{Q_e}{W_c}$$

$$\text{COP}_{\text{VARS}} = \frac{Q_e}{Q_g + W_p} \approx \frac{Q_e}{Q_g}$$

Thus absorption systems are advantageous where a large quantity of low-grade thermal energy is available freely at required temperature. However, it will be seen that for the refrigeration and heat rejection temperatures, the COP of vapour compression refrigeration system will be much higher than the COP of an absorption system as a high grade mechanical energy is used in the former, while a low-grade thermal energy is used in the latter. However, comparing these systems based on COPs is not fully justified, as mechanical energy is more expensive than thermal energy. Hence, sometimes the second law (or exergetic) efficiency is used to compare different refrigeration systems. It is seen that the second law (or exergetic) efficiency of absorption system is of the same order as that of a compression system.

Maximum COP of ideal absorption refrigeration system

In case of a single stage compression refrigeration system operating between constant evaporator and condenser temperatures, the maximum possible COP is given by Carnot COP:

$$\text{COP}_{\text{Carnot}} = \frac{T_e}{T_c - T_e}$$

If we assume that heat rejection at the absorber and condenser takes place at same external heat sink temperature T_o , then a vapour absorption refrigeration system operates between three temperature levels, T_g , T_o and T_e . The maximum possible COP of a refrigeration system operating between three temperature levels can be obtained by applying first and second laws of thermodynamics to the system. Figure 1 shows the various energy transfers and the corresponding temperatures in an absorption refrigeration system.

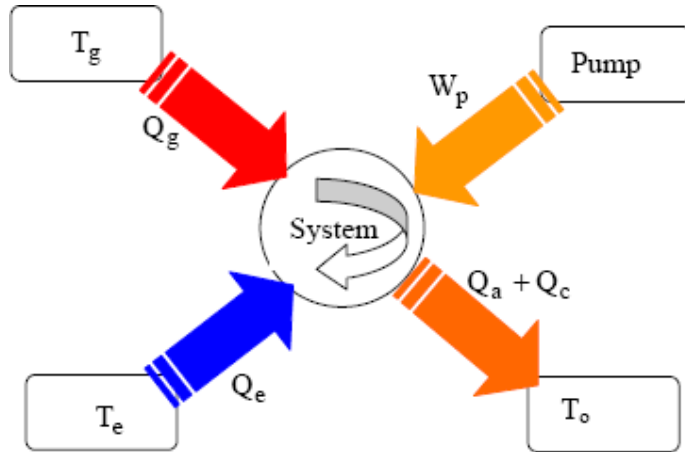


Fig.1: Various energy transfers in a vapour absorption refrigeration system

From first law of thermodynamics,

$$Q_e + Q_g - Q_{c+a} + W_p = 0$$

where Q_e is the heat transferred to the absorption system at evaporator temperature T_e , Q_g is the heat transferred to the generator of the absorption system at temperature T_g , Q_{a+c} is the heat transferred from the absorber and condenser of the absorption system at temperature T_o and W_p is the work input to the solution pump.

From second law of thermodynamics,

$$\Delta S_{\text{total}} = \Delta S_{\text{sys}} + \Delta S_{\text{surr}} \geq 0$$

where ΔS_{total} is the total entropy change which is equal to the sum of entropy change of the system ΔS_{sys} and entropy change of the surroundings ΔS_{surr} . Since the refrigeration system operates in a closed cycle, the entropy change of the working fluid of the system undergoing the cycle is zero, i.e., $\Delta S_{\text{sys}} = 0$. The entropy change of the surroundings is given by:

$$\Delta S_{\text{surr}} = -\frac{Q_e}{T_e} - \frac{Q_g}{T_g} + \frac{Q_{a+c}}{T_o} \geq 0$$

Substituting the expression for first law of thermodynamics in the above equation

$$Q_g \left(\frac{T_g - T_o}{T_g} \right) \geq Q_e \left(\frac{T_o - T_e}{T_e} \right) - W_p$$

Neglecting solution pump work, W_p ; the COP of VARS is given by:

$$\text{COP}_{\text{VARS}} = \frac{Q_e}{Q_g} \leq \left(\frac{T_e}{T_o - T_e} \right) \left(\frac{T_g - T_o}{T_g} \right)$$

An ideal vapour absorption refrigeration system is totally reversible (i.e., both internally and externally reversible). For a completely reversible system the total entropy change (system+surroundings) is zero according to second law, hence for an

ideal VARS $\Delta S_{\text{total,rev}} = 0 \Rightarrow \Delta S_{\text{surr,rev}} = 0$. Hence:

$$\Delta S_{\text{surr,rev}} = -\frac{Q_e}{T_e} - \frac{Q_g}{T_g} + \frac{Q_{a+c}}{T_o} = 0$$

Hence combining first and second laws and neglecting pump work, the maximum possible COP of an ideal VARS system is given by:

$$\text{COP}_{\text{ideal VARS}} = \frac{Q_e}{Q_g} = \left(\frac{T_e}{T_o - T_e} \right) \left(\frac{T_g - T_o}{T_g} \right)$$

Thus the ideal COP is only a function of operating temperatures similar to Carnot system. It can be seen from the above expression that the ideal COP of

VARS system is equal to the product of efficiency of a Carnot heat engine operating between T_g and T_o and COP of a Carnot refrigeration system operating between T_o and T_e , i.e.,

$$\text{COP}_{\text{ideal VARS}} = \frac{Q_e}{Q_g} = \left(\frac{T_e}{T_o - T_e} \right) \left(\frac{T_g - T_o}{T_g} \right) = \text{COP}_{\text{Carnot}} \cdot \eta_{\text{Carnot}}$$

Thus an ideal vapour absorption refrigeration system can be considered to be a combined system consisting of a Carnot heat engine and a Carnot refrigerator as shown in Fig.2. Thus the COP of an ideal VARS increases as generator temperature (T_g) and evaporator temperature (T_e) increase and heat rejection temperature (T_o) decreases. However, the COP of actual VARS will be much less than that of an ideal VARS due to various internal and external irreversibilities present in actual systems.

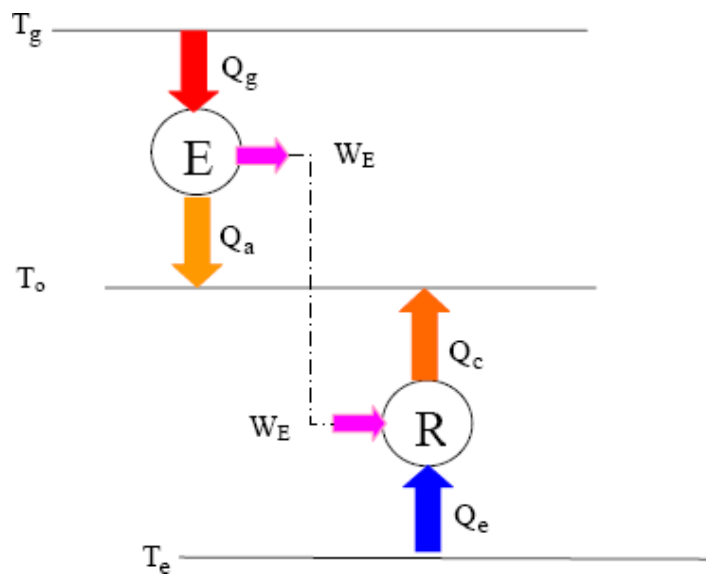


Fig.2: Vapour absorption refrigeration system as a combination of a heat engine and a refrigerator

Properties of refrigerant-absorbent mixtures

The solution used in absorption refrigeration systems may be considered as a homogeneous binary mixture of refrigerant and absorbent. Depending upon the boiling point difference between refrigerant and absorbent and the operating temperatures, one may encounter a pure refrigerant vapour or a mixture of refrigerant and absorbent vapour in generator of the absorption system. Unlike pure substances, the thermodynamic state of a binary mixture (in liquid or vapour phase) cannot be fixed by pressure and temperature alone. According to Gibbs' phase rule, one more parameter in addition to temperature and pressure is required to completely fix the thermodynamic state. Generally, the composition of the mixture is taken as the third independent parameter. The composition of a mixture can be expressed either in mass fraction or in mole fraction. The mass fraction of components 1 and 2 in a binary mixture are given by:

$$\xi_1 = \frac{m_1}{m_1 + m_2}; \quad \xi_2 = \frac{m_2}{m_1 + m_2}$$

where m_1 and m_2 are the mass of components 1 and 2, respectively. The mole fraction of components 1 and 2 in a binary mixture are given

$$x_1 = \frac{n_1}{n_1 + n_2}; \quad x_2 = \frac{n_2}{n_1 + n_2}$$

where n_1 and n_2 are the number of moles of components 1 and 2, respectively. An important property of a mixture is its miscibility. A mixture is said to be completely miscible if a homogeneous mixture can be formed through any arbitrary range of concentration values. Miscibility of mixtures is influenced by the temperature at which they are mixed. Some mixtures are miscible under certain conditions and immiscible at other conditions. The refrigerant-absorbent mixtures used in absorption refrigeration systems must be completely miscible under all conditions both in liquid and vapour phases.

Refrigerant-absorbent combinations for VARS

The desirable properties of refrigerant-absorbent mixtures for VARS are:

- i. The refrigerant should exhibit high solubility with solution in the absorber. This is to say that it should exhibit negative deviation from Raoult's law at absorber.
- ii. There should be large difference in the boiling points of refrigerant and absorbent (greater than 200°C), so that only refrigerant is boiled-off in the generator. This ensures that only pure refrigerant circulates through refrigerant circuit (condenser expansion valve-evaporator) leading to isothermal heat transfer in evaporator and condenser.
- iii. It should exhibit small heat of mixing so that a high COP can be achieved. However, this requirement contradicts the first requirement. Hence, in practice a trade-off is required between solubility and heat of mixing.
- iv. The refrigerant-absorbent mixture should have high thermal conductivity and low viscosity for high performance.
- v. It should not undergo crystallization or solidification inside the system.
- vi. The mixture should be safe, chemically stable, non-corrosive, inexpensive and should be available easily.

The most commonly used refrigerant-absorbent pairs in commercial systems are:

1. Water-Lithium Bromide ($\text{H}_2\text{O-LiBr}$) system for above 0°C applications such as air conditioning. Here water is the refrigerant and lithium bromide is the absorbent.
2. Ammonia-Water ($\text{NH}_3\text{-H}_2\text{O}$) system for refrigeration applications with ammonia as refrigerant and water as absorbent of late efforts are being made to develop other refrigerant-absorbent systems using both natural and synthetic refrigerants to overcome some of the limitations of ($\text{H}_2\text{O-LiBr}$) and ($\text{NH}_3\text{-H}_2\text{O}$) systems.

Currently, large water-lithium bromide ($\text{H}_2\text{O-LiBr}$) systems are extensively used in air conditioning applications, whereas large ammonia-water ($\text{NH}_3\text{-H}_2\text{O}$) systems are used in refrigeration applications, while small ammonia-water systems with a third inert gas are used in a pumpless form in small domestic refrigerators (triple fluid vapour absorption systems).

Introduction

In vapour absorption refrigeration systems based on ammonia-water pair, ammonia is the refrigerant and water is the absorbent. These systems are more versatile than systems based on water-lithium bromide as they can be used for both sub-zero (refrigeration) as well as above 0°C (air conditioning) applications. However, these systems are more complex in design and operation due to the smaller boiling point temperature difference between the refrigerant and absorbent (about 133°C). Due to the smaller boiling point temperature difference the vapour generated in the generator consists of both ammonia as well as water. If water is allowed to circulate with ammonia in the refrigerant circuit, then:

- i. Heat transfer in condenser and evaporator becomes non-isothermal
- ii. Evaporator temperature increases
- iii. Evaporation will not be complete
- iv. Water may get accumulated in the evaporator leading to malfunctioning of the plant
- v. Circulation ratio increases

Since all the above effects are detrimental to the performance of the system, it is necessary to minimize the concentration of water vapour in ammonia at the inlet to the condenser. This requires additional components, namely a rectification column and a dephlegmator between generator and absorber, which increases the design complexity and cost and also reduces the system COP compared to water-lithium bromide system.

Properties of ammonia-water solutions

Composition

Similar to water-lithium bromide solutions, the composition of ammonia-water solution is also expressed either in mass fraction (ξ) or mole fraction (x). However, for ammonia-water solutions, the mass and mole fractions are

defined in terms of ammonia.

For example the mass fraction ξ is defined as the ratio of mass of ammonia to the total mass of solution, i.e.,

$$\xi = \frac{m_A}{m_A + m_W}$$

where m_A and m_W are the mass of ammonia and water in solution, respectively.

Similarly, the mole fraction of ammonia-water solution is defined as:

$$x = \frac{n_A}{n_A + n_W}$$

where n_A and n_W are the number of moles of ammonia and water in solution, respectively. The number of moles of ammonia and water can easily be obtained from their respective masses in solution and molecular weights, thus;

$$n_A = \frac{m_A}{M_A}; \text{ and } n_W = \frac{m_W}{M_W}$$

where M_A (= 17.0 kg/kmol) and M_W (= 18.0 kg/kmol) are the molecular weights of ammonia and water respectively.

Vapour pressure of ammonia-water solutions

Liquid ammonia and water are completely miscible in all proportions, hence can form solutions of all concentrations from 0 to 1, at normal temperatures. The effect of ammonia in water is to lower the vapour pressure of water, similarly the effect of water in ammonia is to lower ammonia's vapour pressure. Thus the total pressure over ammonia water solutions is made up of partial pressure of ammonia and partial pressure of water vapour, and is always in between the saturation pressures of pure ammonia and water.

If Raoult's law is applied to ammonia-water mixtures, then the total pressure at any temperature, P_{total} is given by:

$$P_{\text{total}} = xP_A + (1 - x)P_W$$

where x is the liquid phase mole fraction of ammonia, P_A and P_W are the saturation pressures of pure ammonia and pure water at that temperature.

However, similar to water-lithium bromide solutions, ammonia-water solutions also deviate from ideal solution behaviour predicted by Raoult's law in a negative manner,

i.e., at a given temperature of the solution the actual vapour pressure will be less than that predicted by Raoult's law (activity coefficient is much smaller than 1.0). For example, at a mass fraction of 0.4 and temperature of 40°C , Raoult's law predicts a vapour pressure of 6.47 bar, whereas the measured vapour pressure is 3.029 bar. The vapour pressure data of ammonia-water solutions is also available in the form of

Dühring and other P - T - ξ plots.

Composition of ammonia-water vapour

Since the vapour above ammonia-water liquid consists of both ammonia and water vapour, it is essential to distinguish between the composition in liquid phase and composition in vapour phase. The superscripts L and V will be used to distinguish

between liquid and vapour phase compositions. Thus ξ_L stands for liquid phase mass fraction and ξ_V stands for vapour phase mass fraction. Though the vapour phase composition, can be obtained by assuming ideal solution behaviour, it is observed that

the actual vapour composition deviates from that predicted by ideal mixture

equations. Based on experimental measurements, charts have been developed for obtaining composition of ammonia-water mixture in vapour phase in equilibrium with a solution of ammonia and water at different temperatures. Figure 1 shows the construction of such a chart using which one can obtain the composition of mixture in

vapour phase from known values of liquid phase mass fraction (ξ_L) and saturated temperature of pure ammonia or pressure.

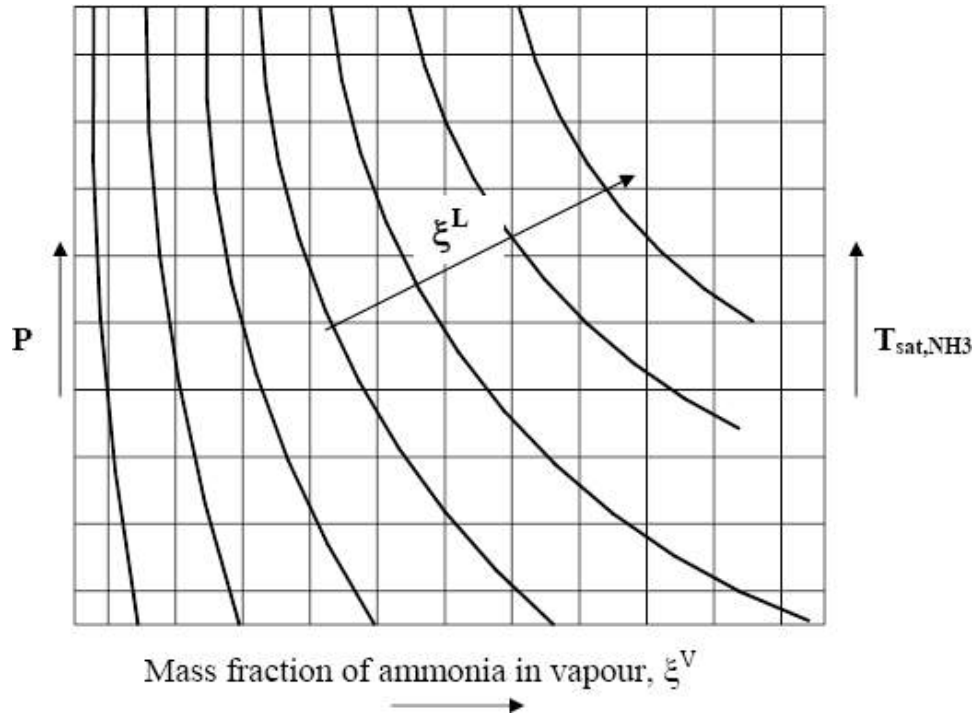


Fig.1. Vapour-liquid equilibrium chart for ammonia-water solution

Bubble point and dew point for ammonia-water mixtures

Figure 2 shows a cylinder containing mixture of ammonia and water. The pressure on the mixture is maintained constant with the help of a free-floating piston with fixed weights. Initially (State 1) the cylinder consists of subcooled solution of ammonia water mixture. Now heat is supplied to the system and the temperature of the solution

is increased steadily, the mass fraction of the solution remains constant at ξ^L initially. At a certain temperature the first vapour bubble appears. The temperature at which the first bubble appears is called as bubble point (=T bubble) of the solution at that concentration and pressure. Further heating results in increase in temperature and formation of more vapour as shown in the figure (State 2). If heating is continued further, then the temperature increases continuously, as more liquid is converted into vapour, and finally at a particular temperature the last liquid droplet vaporizes. The temperature at which the last liquid droplet evaporates is called as dew point temperature (T_{dew}). When heating is continued further the mixture enters into superheated vapour state (State 3). It should be noted that unlike pure fluids, the temperature of the ammonia-water mixture increases continuously as the liquid

undergoes vaporization. This is to say that the phase change process is characterized by a temperature glide, which is the difference between the dew point and bubble point temperatures. If this process is repeated with different initial concentrations starting from 0 (pure water) to 1 (pure ammonia) and at the same pressure, different values of bubble and dew points will be obtained. Of course when the concentration is 0 (pure water) or 1 (pure ammonia) the bubble and dew points coincide. Now if we plot the temperatures (bubble point and dew point) against concentration and join all the bubble points by a curve and all the dew points by another curve, then we would get the equilibrium Temperature vs concentration curve for ammonia-water mixtures at that pressure as shown in Fig.1. The loci of all the bubble points is called as bubble point line and the loci of all the dew points is known as the dew point line. The bubble point line is the saturated liquid line and the dew point line is the saturated vapour line for the mixture at that pressure. The region between the bubble and dew point lines is the two phase region where both liquid and vapour coexist in equilibrium. Different bubble point and dew point lines will be obtained if the experiment is carried out with different pressures. For example, Figure 4 shows the bubble and dew point lines for two different pressures, P1 and P2. The same results can also be obtained if one starts the experiment initially with superheated vapour and then start cooling it. In this case, the dew point is the temperature at which the first liquid droplet forms from the vapour and the bubble point is the temperature at which the last vapour bubble condenses.

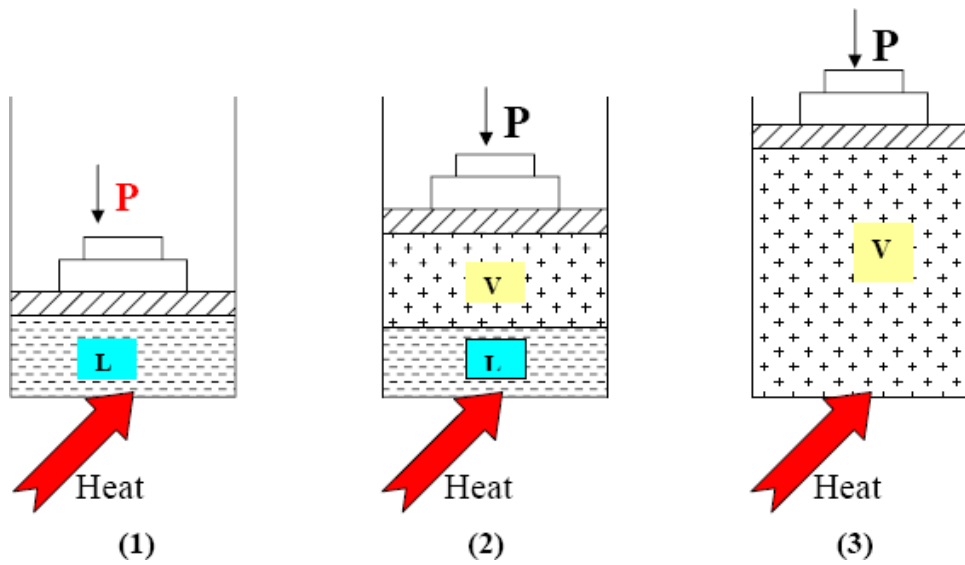


Fig.2: A simple experiment illustrating the principle of bubble and dew points

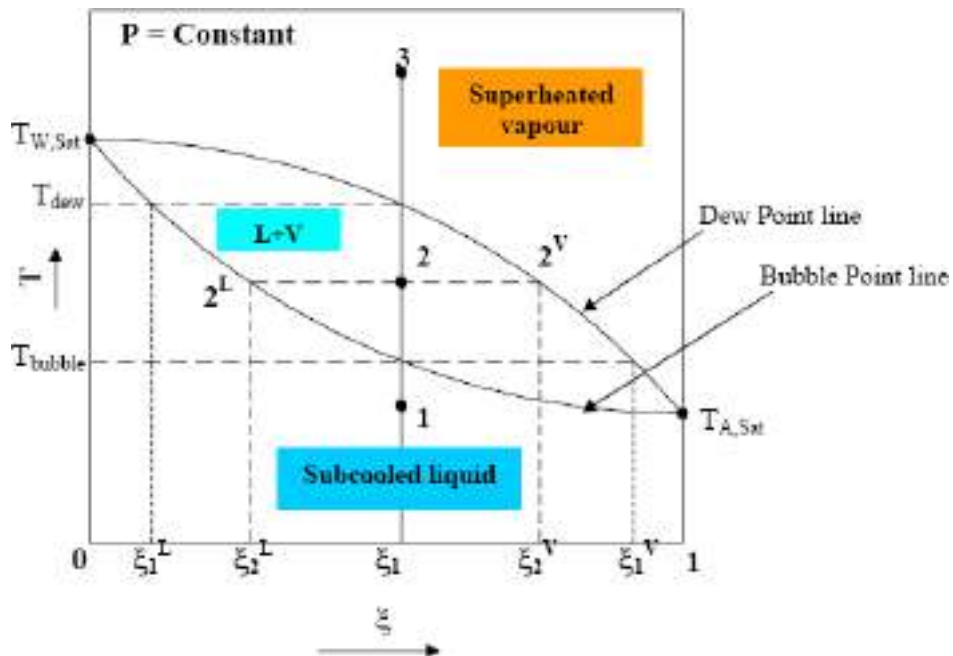


Fig.3: Equilibrium temperature-concentration curve for $\text{NH}_3\text{-H}_2\text{O}$ at a constant pressure

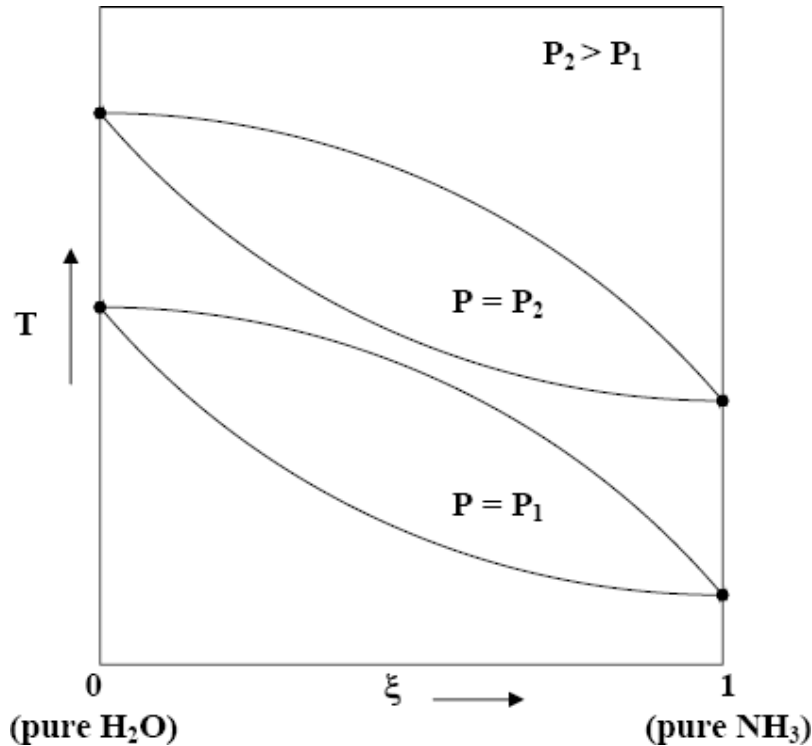


Fig.4: Bubble point and dew point curves at two different pressures

Now since the process is carried out in a closed system, the mass of both ammonia and water will be conserved. The concentration of subcooled liquid will be same as the concentration of superheated vapour. However, in the two-phase region in which the saturated liquid exists in equilibrium with saturated vapour, the concentration of liquid and vapour will be different. For example, at point 2 in Fig.3, the temperature of saturated liquid and vapour will be same as they are in equilibrium, hence, the

concentration of liquid will be ξ_2 (**intersection of constant temperature line with bubble point line**) and that of vapour will be $\xi_2 V$ (**intersection of constant temperature line with dew point line**) as shown in the figure.

Obviously the vapour

formed initially will be richer in the low boiling point substance (ammonia) and the

liquid remaining will be rich in high boiling point substance (water). For example, as shown in Fig.16.3, the concentration of the first vapour bubble will be $\xi_1 V$ and the concentration of the last liquid droplet will be ξ_1

L. Since the total mass as well as mass of individual components is always conserved, we can write mass balance for total mass (m_{total}) and ammonia (m_A) mass at state 2 as:

$$m_{total} = m_2^L + m_2^V$$

$$m_A = \xi_2^L m_2^L + \xi_2^V m_2^V = \xi_1 m_{total}$$

where m_2^L and m_2^V are the mass of liquid and vapour at state 2, respectively. From the above equations it can be easily shown that:

$$\frac{m_2^L}{m_2^V} = \left(\frac{\xi_2^V - \xi_1}{\xi_1 - \xi_2^L} \right), \text{ or}$$

$$m_2^L (\xi_1 - \xi_2^L) = m_2^V (\xi_2^V - \xi_1)$$

The above equation is called as the mixing rule or lever rule for the binary mixtures such as ammonia and water. It implies that the fraction of liquid and vapour in the two-phase mixture is inversely proportional to the distance between the mixture condition 2 and the saturated liquid and vapour states 2L and 2V, respectively.

Enthalpy of ammonia-water mixtures Liquid phase:

The enthalpy of ammonia-water solution in liquid phase, h^L is calculated in a manner similar to that of water-lithium bromide solutions, i.e., by the equation:

$$h^L = \xi^L h_A^L + (1 - \xi^L) h_W^L + \Delta h_{mix}$$

where ξ^L is the liquid phase mass fraction of ammonia, h_A^L and h_W^L are liquid phase enthalpies of pure ammonia and water respectively. Δh_{mix} is the heat of mixing, which is negative (exothermic) similar to water-lithium bromide mixtures.

Using the above equation

one can calculate the specific enthalpy of ammonia water solutions at any

concentration and temperature provided the heat of mixing is known from measurements. Thus enthalpy charts for solution are plotted as a field of

isotherms against mass fraction by taking suitable reference values for enthalpy of ammonia and water. Since pressure does not have a significant effect on liquid enthalpy (except at critical point), normally pressure lines are not shown on typical solution enthalpy charts. Also **enthalpy of subcooled liquid is generally assumed to be equal to the** saturated enthalpy at that temperature without loss of much accuracy Vapour phase:

Evaluation of enthalpy of a mixture of vapours of ammonia and water is more complicated compared to liquid phase enthalpy. This is due to the dependence of vapour enthalpy on both temperature and pressure. However, to simplify the problem, it is generally assumed that ammonia and water vapour mix without any heat of mixing. Then the enthalpy of the vapour mixture, h^V is given by:

$$h^V = \xi^V h_A^V + (1 - \xi^V) h_W^V$$

where ξ^V is the vapour phase mass fraction of ammonia and h_A^V and h_W^V are the specific enthalpies of ammonia vapour and water vapour respectively at the temperature of the mixture. However, since vapour enthalpies depend on temperature as well as pressure, one has to evaluate the vapour enthalpy at suitable pressure, which is not equal to the total pressure. An approximate, but practically useful method is to evaluate the vapour

enthalpies of ammonia and water at pressures, $\xi^V P$ and $(1 - \xi^V) P$ given by:

$$P_A = \xi^V P_{\text{total}}$$

$$P_W = (1 - \xi^V) P_{\text{total}}$$

where ξ^V is the vapour phase mole fraction of ammonia and P_{total} is the total pressure. It should be noted that P_A and P_W are equal to the partial pressures of ammonia and water only if they behave as ideal gases. However since ammonia and water vapour may not approach the ideal gas behaviour at all temperatures and pressures, in general P_A and P_W are not equal to the partial pressures. Using this method enthalpies of ammonia-water mixtures in vapour phase have been obtained as functions of temperature and mass fraction.

The complete enthalpy-composition diagram for ammonia-water mixtures:

Normally, charts of enthalpy-temperature-mass fraction are available which give both liquid phase as well as vapour enthalpy of mixtures. Figure 5 shows one such chart. Figure 6 shows the enthalpy-composition diagram at a constant pressure P. In the figure point **a** represents the condition of saturated liquid mixture at a temperature T

with a liquid phase mass fraction of ξ^L . The liquid phase enthalpy corresponding to this condition is given by h^L . The composition and enthalpy of vapour mixture in equilibrium with the liquid mixture at temperature T and pressure P are obtained by drawing a vertical line from **a** upto the auxiliary line and then drawing a horizontal line to the right from the intersection of the vertical line with the auxiliary line. The intersection of this horizontal line with the dew point line **a'** gives the vapour phase

mass fraction ξ^V and the vapour phase enthalpy h^V as shown in the figure. The isotherm T in the two-phase region is obtained by joining points **a** and **a'** as shown in the figure. Point **b** in the figure lies in the two-phase region. The specific enthalpy of this point h_b is given by:

$$h_b = (1 - \psi_b)h^L + \psi_b h^V$$

where ψ_b is the quality or dryness fraction of the two-phase mixture at b. Since points **a**, **a'** and **b** are co-linear, the dryness fraction ψ_b is given by:

$$\psi_b = \frac{\xi_b - \xi^L}{\xi^V - \xi^L}$$

In actual enthalpy-composition diagrams the isotherms are not shown in two-phase region as a different set of them exist for each pressure.

It is important to note that it is not possible to fix the state of the mixture (subcooled, saturated, two-phase or superheated) just from temperature and mass fraction alone, though one can calculate enthalpy of the mixture from temperature and mass fraction. This is due to the reason that at a given mass fraction and temperature, depending upon the pressure the point can be subcooled or saturated or superheated.

For example, a liquid mixture with a mass fraction of 0.4 and temperature of 80°C has an enthalpy of 210 kJ/kg, and it will be in subcooled condition if the pressure is 4.29 bar and saturated if the pressure is 8.75 bar.

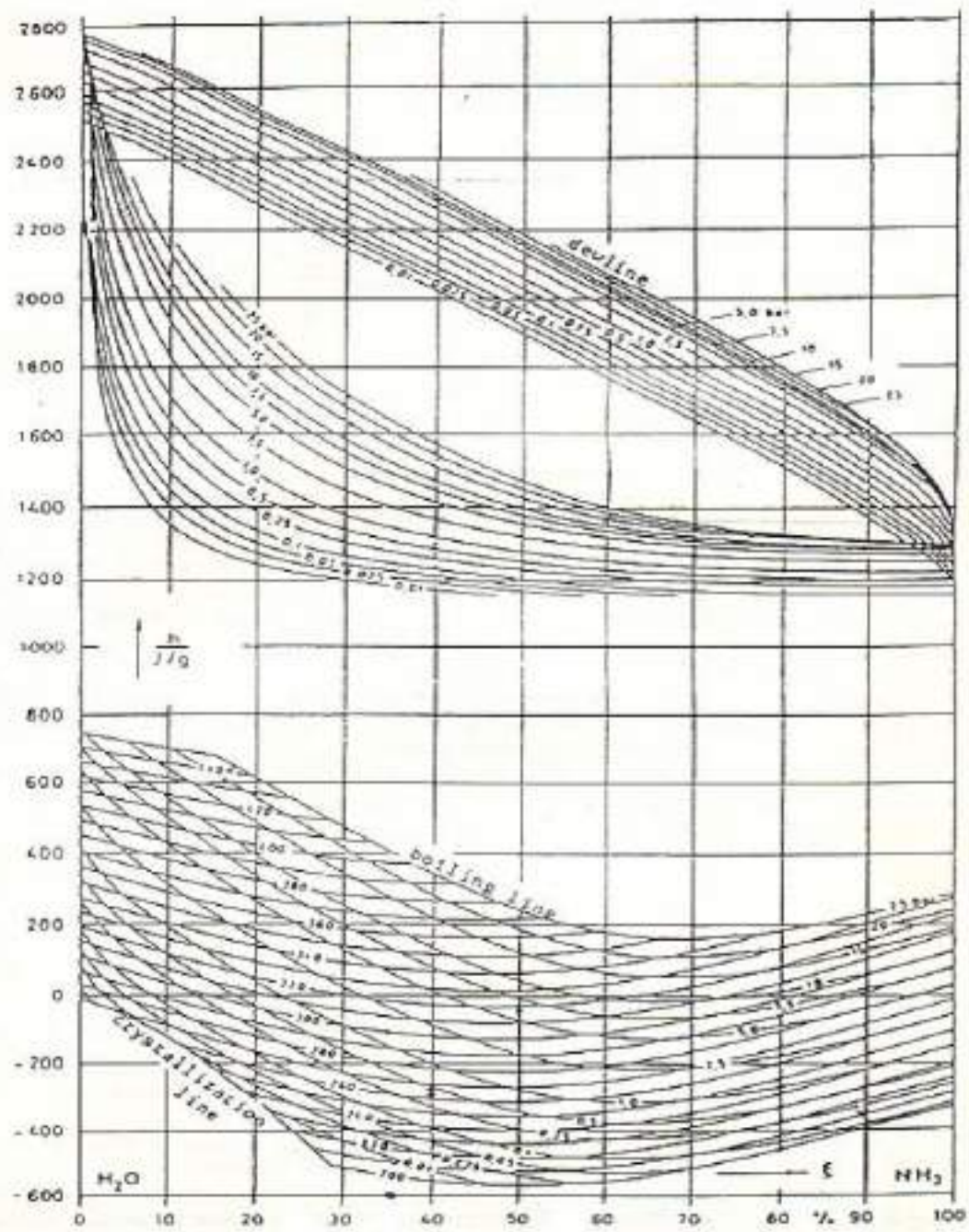


Fig.5: h-T-ξ chart for ammonia-water solution

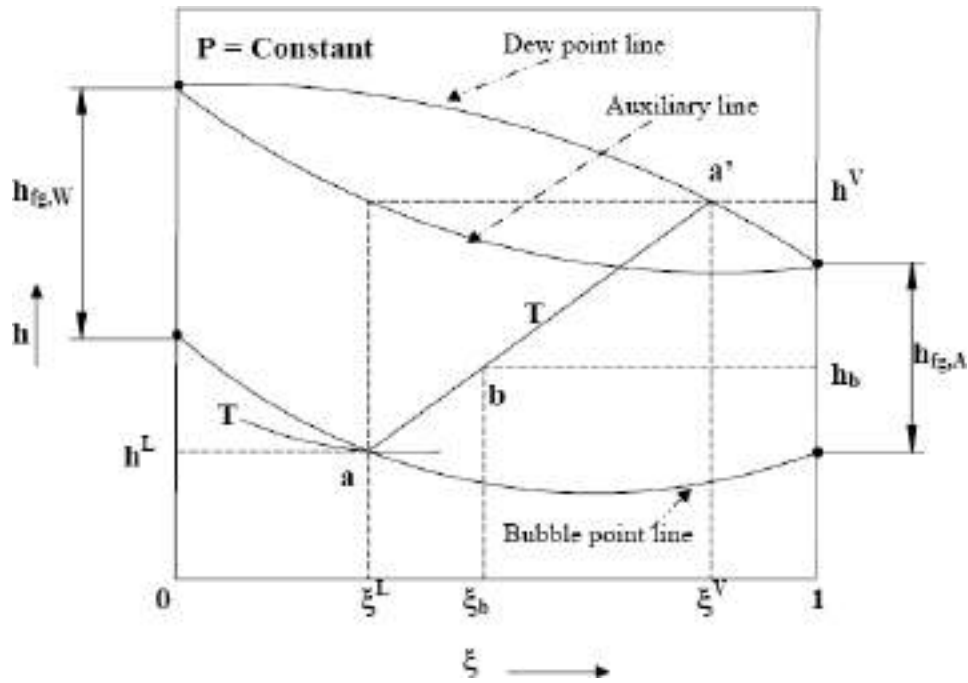


Fig.6: Enthalpy-composition diagram of NH₃-H₂O at a constant pressure P

Determination of temperature of mixture in two-phase region:

A trial-and-error method has to be used to determine the temperature of a point in two-phase region if its enthalpy, liquid phase mass fraction and pressure are known. The trial-and-error method can be graphical or numerical. Figure 6 shows a graphical method for finding the temperature of point x in the two-phase region which is at a

known pressure P_x , liquid phase mass fraction ξ_x and enthalpy h_x . To start with, point a' is obtained as shown in the figure by drawing a vertical line from point x upto the auxiliary line and then drawing a horizontal line from the intersection point a'' upto the dew point line, the intersection of which gives a'. Then a straight line a'-x-a is drawn as shown. Next point b' is obtained by drawing a vertical line upto the auxiliary line and then drawing a horizontal line from b'' upto the dew point line to get b'. Then line b'-x-b is drawn passing through x. This procedure is repeated until convergence is obtained.

Numerically the temperature can be obtained from the equation, which needs to be satisfied for each end of the isotherm passing through x, i.e.,

$$\frac{h^V - h_x}{\xi^V - \xi_x} = \frac{h_x - h^L}{\xi_x - \xi^L}$$

To start with guess values of h^L and ξ^L are assumed by taking some point on the bubble point line. Then saturated vapour properties h^V and ξ^V are obtained from the enthalpy composition charts using the guess values of h^L and ξ^L . Then using the above equation new values of h^L and ξ^L are obtained. Then these new values are used to obtain next set of h^V and ξ^V . This procedure is repeated till the values converge.

Once the converged values of h^L and ξ^L are obtained then the temperature is read from the enthalpy composition chart.

Vapour Absorption Refrigeration systems based on Ammonia water

Pair Introduction:

Vapour absorption refrigeration system based on ammonia-water is one of the oldest refrigeration systems. As mentioned earlier, in this system ammonia is used as refrigerant and water is used as absorbent. Since the boiling point temperature difference between ammonia and water is not very high, both ammonia and water are generated from the solution in the generator. Since presence of large amount of water in refrigerant circuit is detrimental to system performance, rectification of the generated vapour is carried out using a rectification column and a dephlegmator. Since ammonia is used as the refrigerant, these systems can be used for both refrigeration and air conditioning applications. They are available in very small (as pumpless systems) to large refrigeration capacities in applications ranging from domestic refrigerators to large cold storages. Since ammonia is not compatible with materials such as copper or brass, normally the entire system is fabricated out of steel. Another important difference between this system and water-lithium bromide systems is in the operating pressures. While water-lithium bromide systems operate under very low (high vacuum) pressures, the ammonia-water system is operated at pressures much higher than atmospheric. As a result, problem of air leakage into the system is eliminated. Also this system does not suffer from the problem of crystallization encountered in water-lithium bromide systems. However, unlike water, ammonia is both toxic and flammable. Hence, these systems need safety precautions.

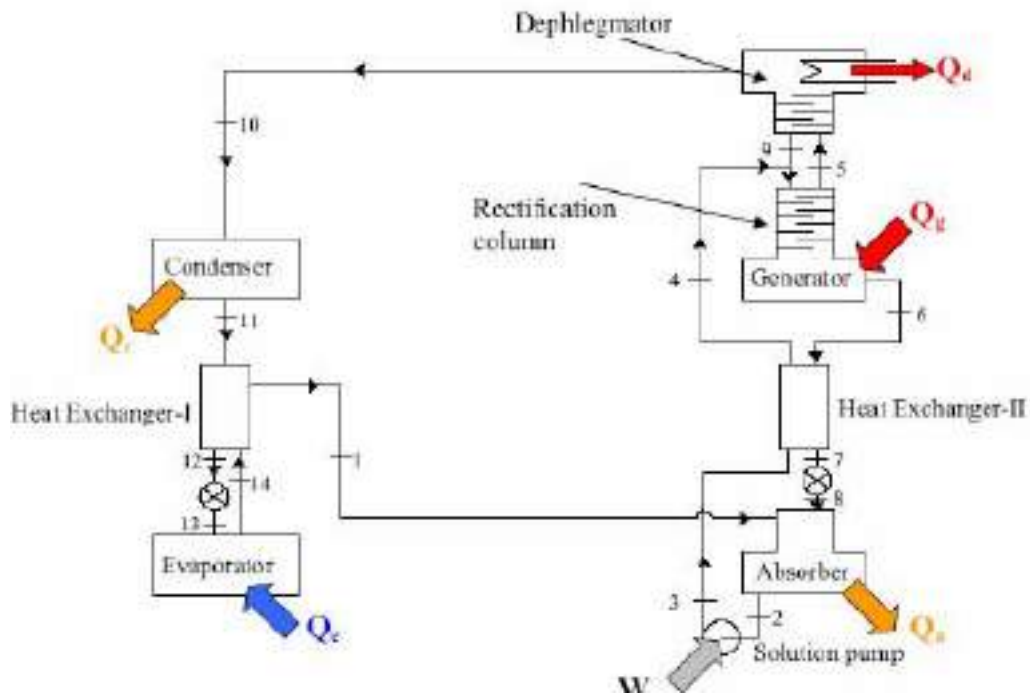


Fig. 7. Schematic of NH₃ – H₂O/H₂O based vapour absorption refrigeration system

Working principle

Figure 7 shows the schematic of an ammonia-water absorption refrigeration system. Compared to water-lithium bromide systems, this system uses three additional components: a rectification column, a dephlegmator and a subcooling heat exchanger (Heat Exchanger-I). As mentioned before, the function of rectification column and dephlegmator is to reduce the concentration of water vapour at the exit of the generator. Without these the vapour leaving the generator may consist of five to ten percent of water. However, with rectification column and dephlegmator the concentration of water is reduced to less than one percent. The rectification column could be in the form of a packed bed or a spray column or a perforated plate column in which the vapour and solution exchange heat and mass. It is designed to provide a large residence time for the fluids so that high heat and mass transfer rates could be obtained. The subcooling heat exchanger, which is normally of counterflow type is used to increase the refrigeration effect and to ensure liquid entry into the refrigerant expansion valve.

As shown in the figure, low temperature and low pressure vapour (almost pure ammonia) at state 14 leaves the evaporator, exchanges heat with the condensed

liquid in Heat Exchanger-I and enters the absorber at state 1. This refrigerant is absorbed by the weak solution (weak in ammonia) coming from the solution expansion valve, state 8. The heat of absorption, Q_a is rejected to an external heat sink. Next the strong solution that is now rich in ammonia leaves the absorber at state 2 and is pumped by the solution pump to generator pressure, state 3. This high pressure solution is then pre-heated in the solution heat exchanger (Heat Exchanger-II) to state 4. The preheated solution at state 4 enters the generator and exchanges heat and mass with the hot vapour flowing out of the generator in the rectification column. In the generator, heat is supplied to the solution (Q_g). As a result vapour of ammonia and water are generated in the generator. As mentioned, this hot vapour with five to ten percent of water exchanges heat and mass with the rich solution descending from the top. During this process, the temperature of the vapour and its water content are reduced. This vapour at state 5 then enters the dephlegmator, where most of the water vapour in the mixture is removed by cooling and condensation. Since this process is exothermic, heat (Q_d) is rejected to an external heat sink in the dephlegmator. The resulting vapour at state 10, which is almost pure ammonia (mass fraction greater than 99 percent) then enters the condenser and is condensed by rejecting heat of condensation, Q_c to an external heat sink. The condensed liquid at state 11 is subcooled to state 12 in the subcooling heat exchanger by rejecting heat to the low temperature, low pressure vapour coming from the evaporator. The subcooled, high pressure liquid is then throttled in the refrigerant expansion valve to state 13. The low temperature, low pressure and low quality refrigerant then enters the evaporator, extracts heat from the refrigerated space (Q_e) and leaves the evaporator at state 14. From here it enters the subcooling heat exchanger to complete the refrigerant cycle. Now, the condensed water in the dephlegmator at state 9 flows down into the rectifying column along with rich solution and exchanges heat and mass with the vapour moving upwards. The hot solution that is now weak in refrigerant at state 6 flows into the solution heat exchanger where it is cooled to state 7 by preheating the rich solution. The weak, but high pressure solution at state 7 is then throttled in the solution expansion valve to state 8, from where it enters the absorber to complete its cycle. As far as various energy flows out of the system are concerned, heat is supplied to the

system at generator and evaporator, heat rejection takes place at absorber, condenser and dephlegmator and a small amount of work is supplied to the solution pump.

Introduction

Vapour absorption refrigeration systems using water-lithium bromide pair are extensively used in large capacity air conditioning systems. In these systems water is used as refrigerant and a solution of lithium bromide in water is used as absorbent. Since water is used as refrigerant, using these systems it is not possible to provide refrigeration at sub-zero temperatures. Hence it is used only in applications requiring refrigeration at temperatures above 0°C. Hence these systems are used for air conditioning applications. The analysis of this system is relatively easy as the vapour generated in the generator is almost pure refrigerant (water), unlike ammonia-water systems where both ammonia and water vapour are generated in the generator. Properties of water-lithium bromide solutions

Composition:

The composition of water-lithium bromide solutions can be expressed either in mass fraction (ξ) or mole fraction (x). For water-lithium bromide solutions, the mass fraction ξ is defined as the ratio of mass of anhydrous lithium bromide to the total mass of solution, i.e.,

$$\xi = \frac{m_L}{m_L + m_W}$$

where m_L and m_W are the mass of anhydrous lithium bromide and water in solution, respectively.

The composition can also be expressed in terms of mole fraction of lithium bromide as:

$$x = \frac{n_L}{n_L + n_W}$$

where n_L and n_W are the number of moles of anhydrous lithium bromide and water in solution, respectively. The number moles of lithium bromide and water can easily be obtained from their respective masses in solution and molecular weights, thus;

$$n_L = \frac{m_L}{M_L}; \text{ and } n_W = \frac{m_W}{M_W}$$

where M_L (= 86.8 kg/kmol) and M_W (= 18.0 kg/kmol) are the molecular weights of anhydrous lithium bromide and water respectively.

Vapour pressure of water-lithium bromide solutions

Applying Raoult's law, the vapour pressure of water-lithium bromide solution with the vapour pressure exerted by lithium bromide being negligibly small is given by:

$$P = (1 - x)P_W$$

where P_W is the saturation pressure of pure water at the same temperature as that of the solution and x is the mole fraction of lithium bromide in solution. It is observed that Raoult's law is only approximately correct for very dilute solutions of water lithium bromide (i.e., as $x \rightarrow 0$). Strong aqueous solutions of water-lithium bromide are found to deviate strongly from Raoult's law in a negative manner.

For example, at 50 percent mass fraction of lithium bromide and 25°C, Raoult's law predicts a vapour pressure of 26.2 mbar, whereas actual measurements show that it is only 8.5 mbar.

The ratio of actual vapour pressure to that predicted by Raoult's law is known as **activity coefficient**. For the above example, the activity coefficient is 0.324.

The vapour pressure data of water-lithium bromide solutions can be very conveniently represented in a Dühring plot. In a Dühring plot, the temperature of the solution is plotted as abscissa on a linear scale, the saturation temperature of pure water is plotted as ordinate on the right hand side (linear scale) and the pressure on a logarithmic scale is plotted as ordinate on the left hand side. The plot shows the pressure-temperature values for various constant concentration lines (isosters), which are linear on Dühring plot. Figures 15.1 shows the Dühring plot. The Dühring plot can be used for finding the vapour pressure data and also for plotting the operating cycle. Figure 15.2 shows the water-lithium bromide based absorption refrigeration system on Dühring plot.

Other types of charts showing vapour pressure data for water-lithium bromide systems are also available in literature. Figure 15.3 shows another chart wherein the mass fraction of lithium bromide is plotted on abscissa, while saturation temperature of pure water and vapour pressure are plotted as ordinates. Also shown are lines of constant solution temperature on the chart. Pressure-temperature composition data are also available in the form of empirical equations.

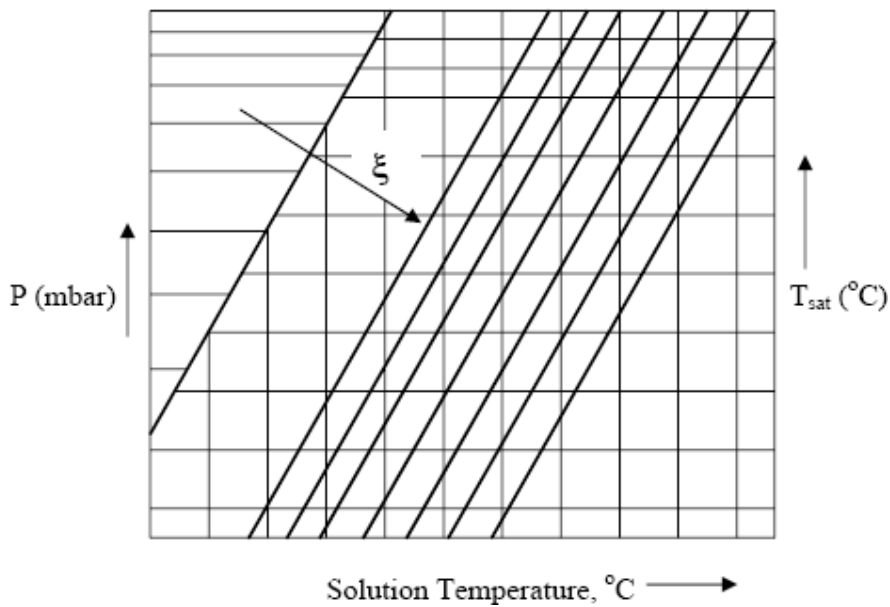


Fig.15.1.: A typical Dühring plot

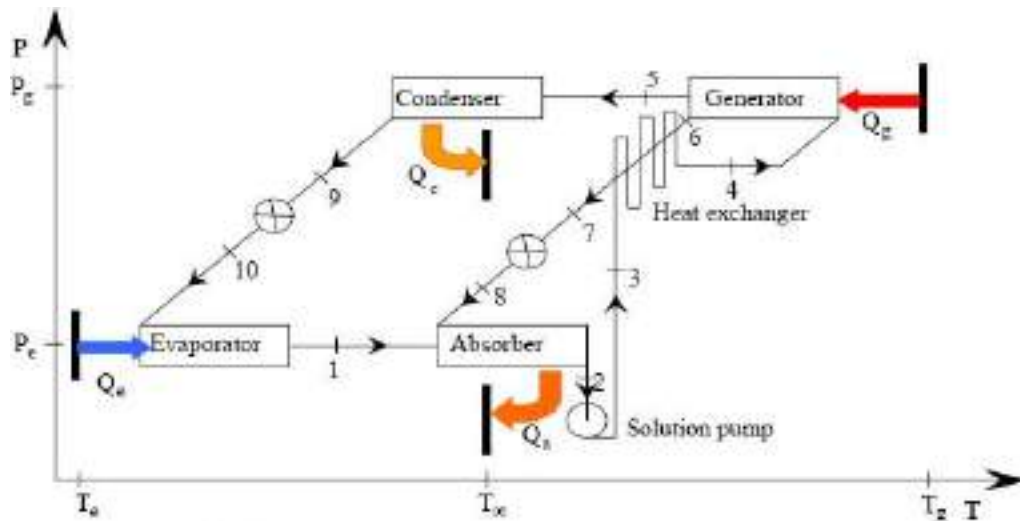


Fig.15.2: H₂O-LiBr system with a solution heat exchanger on Diluting plot

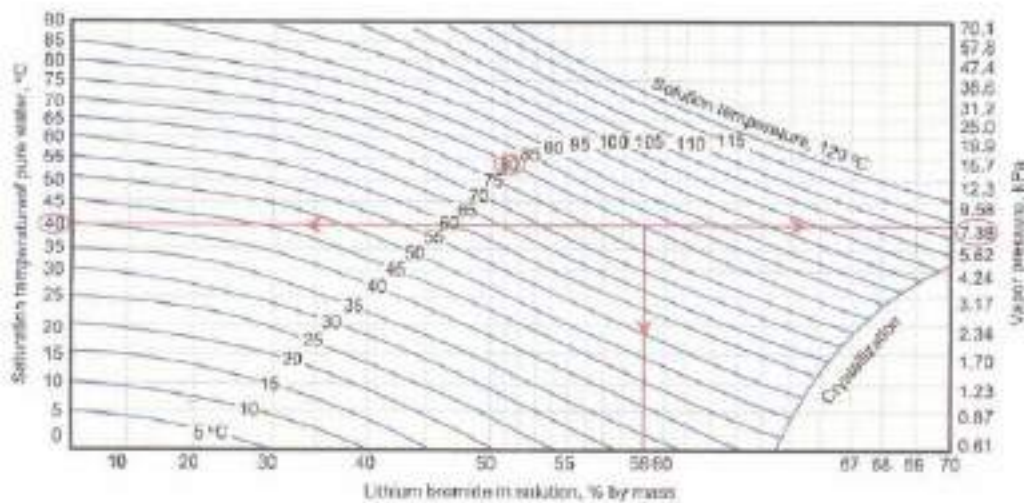


Fig.15.3: Pressure-Temperature-Concentration diagram for H₂O-LiBr solution

15.2.3. Enthalpy of water-lithium bromide solutions

Since strong water-lithium bromide solution deviates from ideal solution behaviour, it is observed that when water and anhydrous lithium bromide at same temperature are mixed adiabatically, the temperature of the solution increases considerably. This indicates that the mixing is an exothermic process with a negative heat of mixing. Hence the specific enthalpy of the solution is given by:

$$h = \xi h_L + (1 - \xi)h_W + \Delta h_{\text{mix}}$$

where h_L and h_W are the specific enthalpies of pure lithium bromide and water, respectively at the same temperature. Figure 15.4 shows a chart giving the specific enthalpy-temperature-mass fraction data for water-lithium bromide solutions. The chart is drawn by taking reference enthalpy of 0 kJ/kg for liquid water at 0°C and solid anhydrous lithium bromide salt at 25°C.

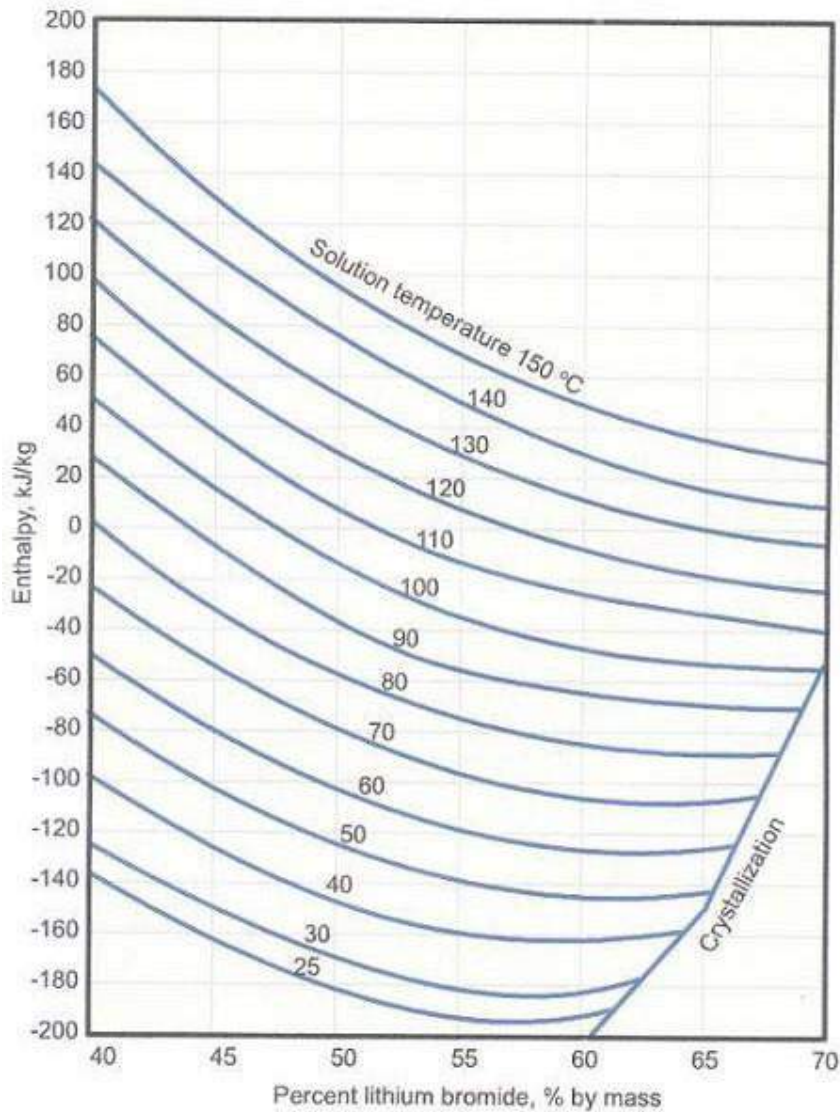


Fig. 4: Enthalpy –Temperature - Concentration diagram for H₂O-LiBr solution

Enthalpy values for pure water (liquid and superheated vapour)

The enthalpy of pure water vapour and liquid at different temperatures and

pressures can be obtained from pure water property data. For all practical purposes, liquid water enthalpy, $h_{W,liquid}$, liquid at any temperature T can be obtained from the equation:

$$h_{W,liquid} = 4.19(T - T_{ref}) \text{ kJ/kg}$$

where T_{ref} is the reference temperature, 0°C .

The water vapour generated in the generator of water-lithium bromide system is in super heated condition as the generator temperature is much higher than the saturation water temperature at that pressure. The enthalpy of superheated water vapour, $h_{W,sup}$ at low pressures and temperature T can be obtained approximately by the equation:

$$h_{W,sup} = 2501 + 1.88(T - T_{ref})$$

Crystallization

The pressure-temperature-mass fraction and enthalpy-temperature-mass fraction charts (Figs. 15.3 and 15.4) show lines marked as crystallization in the lower right section. The region to the right and below these crystallization lines indicates solidification of LiBr salt. In the crystallization region a two-phase mixture (slush) of water-lithium bromide solution and crystals of pure LiBr exist in equilibrium. The water-lithium bromide system should operate away from the crystallization region as the formation of solid crystals can block the pipes and valves. Crystallization can occur when the hot solution rich in LiBr salt is cooled in the solution heat exchanger to low temperatures. To avoid this the condenser pressure reduction below a certain value due to say, low cooling water temperature in the condenser should be avoided. Hence in commercial systems, the condenser pressure is artificially maintained high even though the temperature of the available heat sink is low. This actually reduces the performance of the system, but is necessary for proper operation of the system.

It should be noted from the property charts that the entire water-lithium

bromide system operates under vacuum.

Steady flow analysis of Water-Lithium Bromide Systems

Figure 5 shows the schematic of the system indicating various state points.

A steady flow analysis of the system is carried out with the following assumptions:

- i. Steady state and steady flow
- ii. Changes in potential and kinetic energies across each component are negligible
- iii. No pressure drops due to friction
- iv. Only pure refrigerant boils in the generator.

generator. The nomenclature followed is:

\dot{m} = mass flow rate of refrigerant, kg

\dot{m} = mass flow rate of strong solution (rich in LiBr), kg/s

\dot{m} = mass flow rate of weak solution (weak in LiBr), kg/s

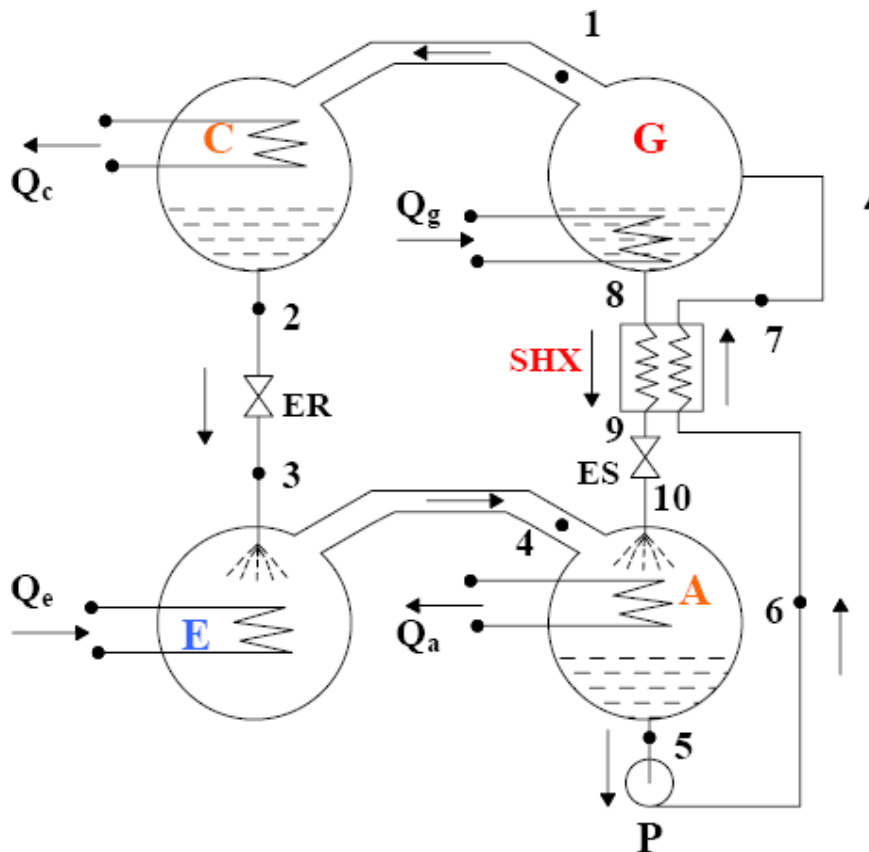


Fig.5: Schematic of a H₂O-LiBr system

A: Absorber;

C: Condenser;

E: Evaporator; valve;

ES: Solution Expansion valve

ERESG

G: Generator; P: Solution Pump

SHX: Solution HX; ER: Refrigerant Expansion

The circulation ratio (λ) is defined as the ratio of strong solution flow rate to refrigerant flow rate. It is given by:

$$\lambda = \frac{\dot{m}_{ss}}{\dot{m}}$$

this implies that the strong solution flow rate is given by:

$$\dot{m}_{ss} = \lambda \dot{m}$$

The analysis is carried out by applying mass and energy balance across each component.

Condenser:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}$$

$$Q_c = \dot{m}(h_1 - h_2)$$

$$P_c = P_{sat}(T_c)$$

where T_c is the condenser temperature

Expansion valve (refrigerant):

$$\dot{m}_2 = \dot{m}_3 = \dot{m}$$

$$h_2 = h_3$$

Evaporator:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}$$

$$Q_e = \dot{m}(h_4 - h_3)$$

$$P_e = P_{sat}(T_e)$$

where T_e is the evaporator temperature

Absorber:

From total mass balance:

$$\dot{m} + \dot{m}_{ss} = \dot{m}_{ws}$$

$$\dot{m}_{ss} = \lambda \dot{m} \Rightarrow \dot{m}_{ws} = (1 + \lambda) \dot{m}$$

From mass balance for pure water:

$$\dot{m} + (1 - \xi_{ss}) \dot{m}_{ss} = (1 - \xi_{ws}) \dot{m}_{ws}$$

$$\Rightarrow \lambda = \frac{\xi_{ws}}{\xi_{ss} - \xi_{ws}}$$

$$Q_a = \dot{m} h_4 + \lambda \dot{m} h_{10} - (1 + \lambda) \dot{m} h_5$$

$$\text{or, } Q_a = \dot{m} [(h_4 - h_5) + \lambda (h_{10} - h_5)]$$

The first term in the above equation represents the enthalpy change of water as changes its state from vapour at state 4 to liquid at state 5. The second term represents the sensible heat transferred as solution at state 10 is cooled to solution at state 5. $\dot{m}(h_4 - h_5)$

$\dot{m} (h_{10} - h_5)$

Solution

pump:

$$\dot{m}_5 = \dot{m}_6 = \dot{m}_{ws}$$

$$W_p = \dot{m}_{ws} (h_6 - h_5) = (1 + \lambda) \dot{m} (h_6 - h_5)$$

however, if we assume the solution to be incompressible, then:

$$W_p = (1 + \lambda) \dot{m} v_{sol} (P_6 - P_5) = (1 + \lambda) \dot{m} v_{sol} (P_c - P_e)$$

where v_{sol} is the specific volume of the solution which can be taken to be approximately equal to 0.00055 m³/kg. Even though the solution pump work is small it is still required in the selection of suitable pump.

Solution heat exchanger:

$$\dot{m}_6 = \dot{m}_7 = \dot{m}_{ws}$$

$$\dot{m}_8 = \dot{m}_9 = \dot{m}_{ss}$$

heat transfer rate in the solution heat exchanger, QHX is given by:

$$Q_{HX} = (1 + \lambda) \dot{m} (h_7 - h_6) = \lambda \dot{m} (h_8 - h_9)$$

Generator:

$$\dot{m}_7 = \dot{m}_8 + \dot{m}_1$$

Heat input to the generator is given by:

$$Q_g = \dot{m} h_1 + \lambda \dot{m} h_8 - (1 + \lambda) \dot{m} h_7$$

$$\text{or, } Q_g = \dot{m} [(h_1 - h_7) + \lambda (h_8 - h_7)]$$

in the above equation the 1st term on the RHS represents energy required to generate water vapour at state 1 from solution at state 7 and the 2 $\dot{m}(h_1 - h_7)$ represents the sensible heat required to heat the solution from state 7 to state 8. $\lambda \dot{m} (h_8 - h_7)$

Solution expansion valve:

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{ws}$$

$$h_9 = h_{10}$$

The COP of the system is given by:

$$COP = \frac{Q_e}{Q_g + W_p} \approx \frac{Q_e}{Q_g}$$

The second law (exergetic) efficiency of the system η_{II} is given by:

$$\eta_{II} = \frac{COP}{COP_{max}} = \left(\frac{Q_e}{Q_g} \right) \left(\frac{T_g}{T_g - T_c} \right) \left(\frac{T_c - T_e}{T_e} \right)$$

In order to find the steady-state performance of the system from the above set of equations, one needs to know the operating temperatures, weak and strong solution concentrations, effectiveness of solution heat exchanger and the refrigeration capacity. It is generally assumed that the solution at the exit of absorber and generator is at equilibrium so that the equilibrium P-T- ξ and h-T- ξ charts can be used for evaluating solution property data. The effectiveness of solution heat exchanger, ϵ_{HX} is given by:

$$\epsilon_{HX} = \frac{(T_7 - T_6)}{(T_8 - T_6)}$$

From the above equation the temperature of the weak solution entering the generator (T7) can be obtained since T6 is almost equal to T5 and T8 is equal to the generator temperature Tg. The temperature of superheated water vapour at state 1 may be assumed to be equal to the strong solution temperature T8.

DOMESTICELECTRO LUX REFRIGERATOR

- The domestic absorption type refrigerator was developed from an invention by Carl Munters and Baltzer Von Platen. This system is often called "Munters Platen System".
- This type of refrigerator is also called "Three-fluids absorption system". The three fluids used in this system are ammonia, hydrogen and water.
- The "ammonia" is used as a refrigerant because it possesses most of the desirable properties. Though it is toxic, and not otherwise preferred in domestic appliances, it is very safe in this system due to absence of any moving parts in the system and, therefore, there is the least chance of any leakage.
- The "hydrogen" being the lightest gas, is used to increase the rate of evaporation (the lighter the gas, faster is the evaporation) of the liquid ammonia passing through the evaporator. The hydrogen is also non-

corrosive and insoluble in water. This is used in the low-pressure side of the system.

- The "water" is used as a solvent because it has the ability to absorb ammonia readily.

Principle and Working of Electrolux Refrigerators:

Fig. 1 shows a schematic diagram of an 'Electrolux refrigerator'. It is a domestic refrigerator and is the best known absorption type of refrigerator. Here pump is dispensed with. The small energy supply is by means of a heater which may be electric or gas.

Principle: The principle involved makes use of the properties of gas-vapour mixtures. If a liquid is exposed to an inert atmosphere, it will evaporate until the atmosphere is saturated with the vapour of the liquid. This evaporation requires heat which is taken from the surroundings in which the evaporation takes place. A cooling effect is thus produced. The partial pressures of the refrigerant vapour (in this case ammonia) must be low in the evaporator, and higher in condenser. The total pressure throughout the circuit must be constant so that the only movement of the working fluid is by convection currents. The partial pressure of ammonia is kept low in requisite parts of the circuit by concentrating hydrogen in those parts.

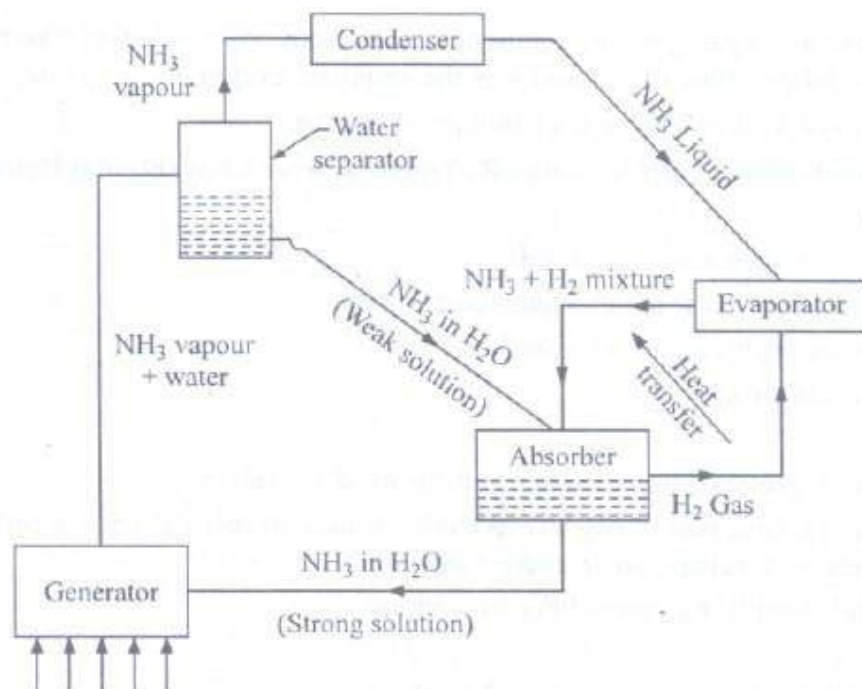


Fig.1. Electrolux refrigerator

Working: The ammonia liquid leaving the condenser enters the evaporator and evaporates into the hydrogen at the low temperature corresponding to its low partial pressure. The mixture of ammonia and hydrogen passes to the absorber into which is also admitted water from the separator. The water absorbs the ammonia and the hydrogen returns to the evaporator. In the absorber the ammonia therefore passes, from the ammonia circuit into water circuit as ammonia in water solution. This strong solution passes, to the generator where it is heated and the vapour given off rises to the separator. The water with the vapour is separated out and a weak solution of ammonia is passed back to the absorber, thus completing the water circuit. The ammonia vapour rises from the separator to the condenser where it is condensed and then returned to the evaporator.

The actual plant includes refinements and practical modifications (which are not included here).

The following points are worth noting:

- The complete cycle is carried out entirely by gravity flow of the refrigerant.
- The hydrogen gas circulates only from the absorber to the evaporator and back.
- With this type of machine efficiency is not important since the energy input is small.
- It has not been used for industrial applications as the C.O.P. of the system is very low.

Role of Hydrogen: By the presence of hydrogen it is possible to maintain uniform total pressure throughout the system and at the same time permit the refrigerant to evaporate at low temperature in the evaporator corresponding to its partial pressure. Thus the condenser and evaporator pressures of the refrigerant are maintained as below:

- (i) In the condenser only ammonia is present, and the total pressure is the condensing pressure.
- (ii) In the evaporator hydrogen and ammonia are present, their relative masses are adjusted such that the partial pressure of ammonia is the required evaporator pressure. These are achieved without the use of pumps or valves.

Advantages and Disadvantages of Electrolux Refrigerator over Conventional Refrigerators:

Advantages:

1. No pump or compressor is required.
2. No mechanical troubles, maintenance cost is low.
3. No lubrication problem; no wear and tear.
4. Completely leak proof.
5. Noiseless.
6. No chance of pressure unbalancing and no need of valves.
7. System may be designed to use any available source of thermal energy- process steam, exhaust from engines or turbines, solar energy etc.
8. Easy control, simply by controlling heat input.

Disadvantages:

1. More complicated in construction and working.
2. C.O.P. very low.
3. The major disadvantage of this type of refrigerator is that if it is spoiled once, it cannot be repaired and has to be replaced fully.

Unit -IV

Objective:

To impart knowledge on Fundamentals of psychrometry.

Outcome:

Ability to understand various psychometric processes.

Introduction to Air Conditioning: Psychrometric Properties & Processes – Characterization of Sensible and Latent Heat Loads — Need For Ventilation, Consideration of Infiltrated Air – Heat Load Concepts. Air Conditioning Systems: Air Cooler (Evaporative Cooling), Window, Split, summer, winter, Year Round, Central Air Conditioning Systems.

Introduction:

Atmospheric air makes up the environment in almost every type of air conditioning system. Hence a thorough understanding of the properties of atmospheric air and the ability to analyze various processes involving air is fundamental to air conditioning design.

Psychrometry: is the study of the properties of mixtures of air and water vapour.

Atmospheric air is a mixture of many gases plus water vapour and a number of pollutants (Fig.1). The amount of water vapour and pollutants vary from place to place. The concentration of water vapour and pollutants decrease with altitude, and above an altitude of about 10 km, atmospheric air consists of only dry air. The pollutants have to be filtered out before processing the air. Hence, what we process is essentially a mixture of various gases that constitute air and water vapour. This mixture is known as *moist air*.

The moist air can be thought of as a mixture of dry air and moisture. For all practical purposes, the composition of dry air can be considered as constant. In 1949, a standard composition of dry air was fixed by the International Joint Committee on Psychrometric data. It is given in Table 1.

Constituent	Molecular weight	Mol fraction
Oxygen	32.000	0.2095
Nitrogen	28.016	0.7809

Argon	39.944	0.0093
Carbon dioxide	44.010	0.0003

Table 1: Composition of standard air

Based on the above composition the *molecular weight of dry air is found to be 28.966 and the gas constant R is 287.035 J/kg.K.*

As mentioned before the air to be processed in air conditioning systems is a mixture of dry air and water vapour. While the composition of dry air is constant, the amount of water vapour present in the air may vary from zero to a maximum depending upon the temperature and pressure of the mixture (dry air + water vapour).

At a given temperature and pressure the dry air can only hold a certain maximum amount of moisture. When the moisture content is maximum, then the air is known as *saturated air*, which is established by a *neutral equilibrium between the moist air and the liquid or solid phases of water.*

For calculation purposes, the molecular weight of water vapour is taken as 18.015 and its gas constant is 461.52 J/kg.K.

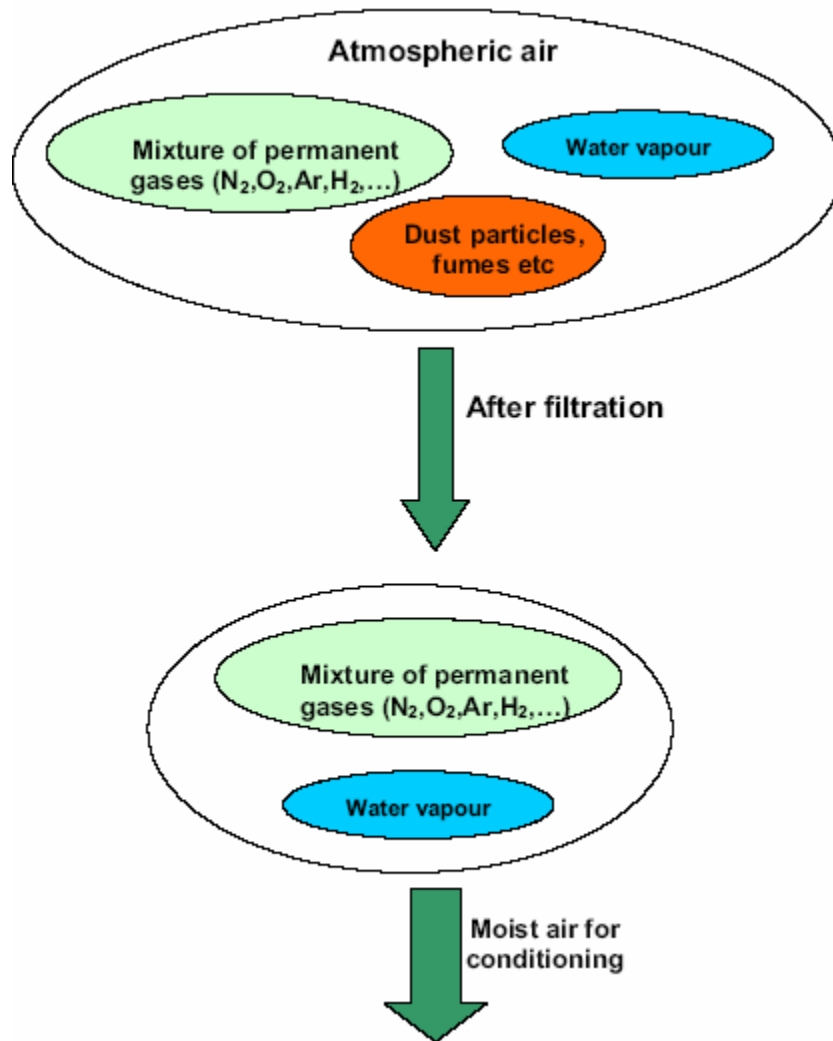


Fig.1: Atmospheric air

Methods for estimating properties of moist air:

In order to perform air conditioning calculations, it is essential first to estimate various properties of air. It is difficult to estimate the exact property values of moist air as it is a mixture of several permanent gases and water vapour. However, moist air upto 3 atm. pressure is found to obey perfect gas law with accuracy sufficient for engineering calculations. For higher accuracy Goff and Gratch tables can be used for estimating moist air properties. These tables are obtained using mixture models based on fundamental principles of statistical mechanics that take into account the real gas behaviour of dry air and water vapour. However, these tables are valid for a barometric pressure of 1 atm. only. Even though the calculation procedure is quite complex, using the mixture models it is possible to estimate moist air properties at other pressures also. However, since in most cases the pressures involved are low, one can apply the perfect gas model to

estimate psychrometric properties.

Basic gas laws for moist air:

According to the *Gibbs-Dalton law* for a mixture of perfect gases, the total pressure exerted by the mixture is equal to the sum of partial pressures of the constituent gases. According to this law, for a homogeneous perfect gas mixture occupying a volume V and at temperature T , each constituent gas behaves as though the other gases are not present (i.e., there is no interaction between the gases). Each gas obeys perfect gas equation. Hence, the partial pressures exerted by each gas, $p_1, p_2, p_3 \dots$ and the total pressure p_t are given by:

$$p_1 = \frac{n_1 R_u T}{V}; p_2 = \frac{n_2 R_u T}{V}; p_3 = \frac{n_3 R_u T}{V} \dots\dots$$
$$p_t = p_1 + p_2 + p_3 + \dots\dots \quad \square \quad 1$$

where n_1, n_2, n_3, \dots are the number of moles of gases 1, 2, 3, ...

Applying this equation to moist air.

$$p = p_t = p_a + p_v \quad \square \quad (2)$$

where $p = p_t$ = total barometric pressure

p_a = partial pressure of dry air

p_v = partial pressure of water vapour

Important psychrometric properties:

Dry bulb temperature (DBT) is the temperature of the moist air as measured by a standard thermometer or other temperature measuring instruments. Saturated vapour pressure (p_{sat}) is the saturated partial pressure of water vapour at the dry bulb temperature. This is readily available in thermodynamic tables and charts. ASHRAE suggests the following regression equation for saturated vapour pressure of water, which is valid for 0 to 100°C.

$$\ln(p_{sat}) = \frac{c_1}{T} + c_2 + c_3 T + c_4 T^2 + c_5 T^3 + c_6 \ln(T) \quad \square \quad 3$$

where p_{sat} = saturated vapor pressure of water in kiloPascals

T = temperature in K

The regression coefficients c1 to c6 are given by:

$$\begin{aligned} c1 &= -5.80022006E+03, & c2 &= -5.516256E+00, & c3 &= -4.8640239E-02 \\ c4 &= 4.1764768E-05, & c5 &= -1.4452093E-08, & c6 &= 6.5459673E+00 \end{aligned}$$

Relative humidity (ϕ) is defined as the ratio of the mole fraction of water vapour in moist air to mole fraction of water vapour in saturated air at the same temperature and pressure. Using perfect gas equation we can show that:

$$\phi = \frac{\text{partial pressure of water vapour}}{\text{saturation pressure of pure water vapour at same temperature}} = \frac{p_v}{p_{\text{sat}}}$$

□ 4

Relative humidity is normally expressed as a percentage. When ϕ is 100 percent, the air is saturated.

Humidity ratio (W): The humidity ratio (or specific humidity) W is the mass of water associated with each kilogram of dry air¹. Assuming both water vapour and dry air to be perfect gases², the humidity ratio is given by:

$$W = \frac{\text{kg of water vapour}}{\text{kg of dry air}} = \frac{p_v V / R_v T}{p_a V / R_a T} = \frac{p_v / R_v}{(p_t - p_v) / R_a} \quad \square 5$$

Substituting the values of gas constants of water vapour and air R_v and R_a in the above equation; the humidity ratio is given by:

$$W = 0.622 \frac{p_v}{p_t - p_v} \quad \square 6$$

For a given barometric pressure p_t , given the DBT, we can find the saturated vapour pressure p_{sat} from the thermodynamic property tables on steam. Then using the above equation, we can find the humidity ratio at saturated conditions, W_{sat} .

It is to be noted that, W is a function of both total barometric pressure and vapor pressure of water.

Dew-point temperature: If unsaturated moist air is cooled at constant pressure, then the temperature at which the moisture in the air begins to condense is known as *dew-point temperature (DPT)* of air. An approximate equation for dew-point temperature is given

by:

$$DPT = \frac{4030(DBT + 235)}{4030 - (DBT + 235)\ln\phi} - 235 \quad \square 7$$

where ϕ is the relative humidity (in fraction). DBT & DPT are in °C. Of course, since from its definition, the dew point temperature is the saturation temperature corresponding to the vapour pressure of water vapour, it can be obtained from steam tables or using Eqn.(3).

Degree of saturation μ : The degree of saturation is the ratio of the humidity ratio W to the humidity ratio of a saturated mixture W_s at the same temperature and pressure, i.e.,

$$\mu = \left. \frac{W}{W_s} \right|_{t,P} \quad \square 8$$

Enthalpy: The enthalpy of moist air is the sum of the enthalpy of the dry air and the enthalpy of the water vapour. Enthalpy values are always based on some reference value. For moist air, the enthalpy of dry air is given a zero value at 0°C, and for water vapour the enthalpy of saturated water is taken as zero at 0°C.

The enthalpy of moist air is given by:

$$h = h_a + Wh_g = c_p t + W(h_{fg} + c_{pw} t) \quad \square 9$$

where c_p = specific heat of dry air at constant pressure, kJ/kg.K

c_{pw} = specific heat of water vapor, kJ/kg.K

t = Dry-bulb temperature of air-vapor mixture, °C

W = Humidity ratio, kg of water vapor/kg of dry air

h_a = enthalpy of dry air at temperature t , kJ/kg

h_g = enthalpy of water vapor at temperature t , kJ/kg

h_{fg} = latent heat of vaporization at 0°C, kJ/kg

The unit of h is kJ/kg of dry air. Substituting the approximate values of c_p and h_g , we obtain:

$$h = 1.005 t + W(2501 + 1.88t) \quad \square 10$$

Humid specific heat: From the equation for enthalpy of moist air, the humid specific heat of moist air can be written as:

$$c_{pm} = c_p + W.c_{pw} \quad \square \quad 11$$

where c_{pm} = humid specific heat, kJ/kg.K

c_p = specific heat of dry air, kJ/kg.K

c_{pw} = specific heat of water vapor, kJ/kg

W = humidity ratio, kg of water vapor/kg of dry air

Since the second term in the above equation ($w.c_{pw}$) is very small compared to the first term, for all practical purposes, the humid specific heat of moist air, c_{pm} can be taken as 1.0216 kJ/kg dry air.K

Specific volume: The specific volume is defined as the number of cubic meters of moist air per kilogram of dry air. From perfect gas equation since the volumes occupied by the individual substances are the same, the specific volume is also equal to the number of cubic meters of dry air per kilogram of dry air, i.e.,

$$v = \frac{R_a T}{p_a} = \frac{R_a T}{p_t - p_v} \quad m^3 / kg \text{ dry air} \quad \square \quad 12$$

Psychrometric chart

A *Psychrometric chart* graphically represents the thermodynamic properties of moist air. Standard psychrometric charts are bounded by the dry-bulb temperature line (abscissa) and the vapour pressure or humidity ratio (ordinate). The Left Hand Side of the psychrometric chart is bounded by the saturation line. Figure 2 shows the schematic of a psychrometric chart. Psychrometric charts are readily available for standard barometric pressure of 101.325 kPa at sea level and for normal temperatures (0-50°C). ASHRAE has also developed psychrometric charts for other temperatures and barometric pressures (for low temperatures: -40 to 10°C, high temperatures 10 to 120°C and very high temperatures 100 to 120°C)

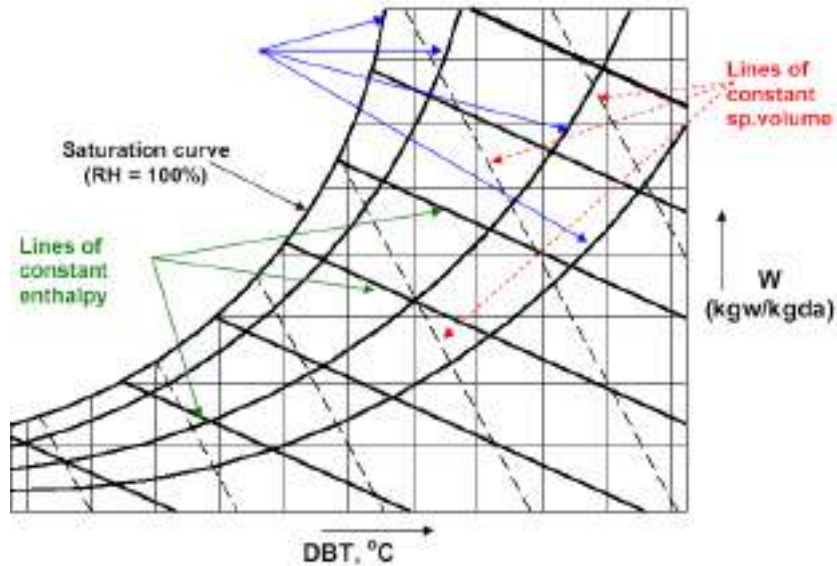


Fig.2: Schematic of a psychrometric chart for a given barometric pressure

Measurement of psychrometric properties:

Based on Gibbs' phase rule, the thermodynamic state of moist air is uniquely fixed if the barometric pressure and two other independent properties are known. This means that at a given barometric pressure, the state of moist air can be determined by measuring any two independent properties. One of them could be the dry-bulb temperature (DBT), as the measurement of this temperature is fairly simple and accurate. The accurate measurement of other independent parameters such as humidity ratio is very difficult in practice. Since measurement of temperatures is easier, it would be convenient if the other independent parameter is also a temperature. Of course, this could be the dew-point temperature (DPT), but it is observed that accurate measurement of dew-point temperature is difficult. In this context, a new independent temperature parameter called the *wet-bulb temperature* (WBT) is defined. Compared to DPT, it is easier to measure the wet-bulb temperature of moist air. Thus knowing the dry-bulb and wet-bulb temperatures from measurements, it is possible to find the other properties of moist air.

To understand the concept of wet-bulb temperature, it is essential to understand the process of combined heat and mass transfer.

Combined heat and mass transfer; the straight line law

The straight line law states that “when air is transferring heat and mass (water) to or from a wetted surface, the condition of air shown on a psychrometric chart drives towards the saturation line at the temperature of the wetted surface”. For example, as

shown in Fig.26.3, when warm air passes over a wetted surface its temperature drops from 1 to 2. Also, since the vapor pressure of air at 1 is greater than the saturated vapor pressure at t_w , there will be moisture transfer from air to water, i.e., the warm air in contact with cold wetted surface cools and dehumidifies. According to the straight line law, the final condition of air (i.e., 2) lies on a straight line joining 1 with t_w on the saturation line. This is due to the value of unity of the Lewis number, that was discussed in an earlier chapter on analogy between heat and mass transfer.

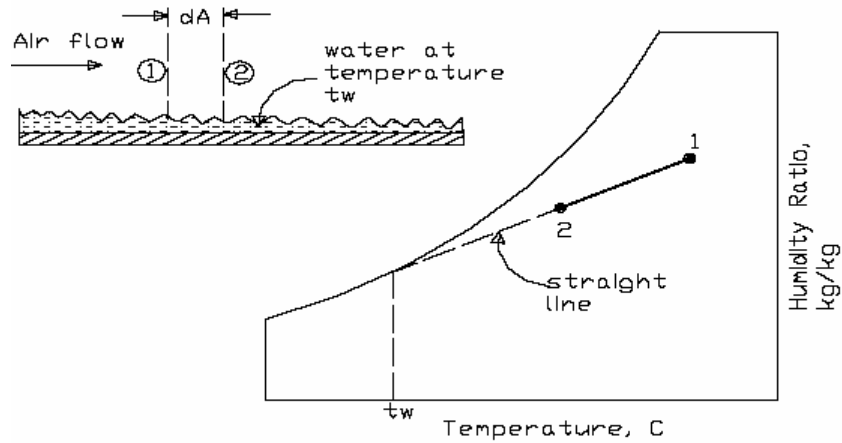


Fig.3: Principle of straight-line law for air-water mixtures

Adiabatic saturation and thermodynamic wet bulb temperature:

Adiabatic saturation temperature is defined as that temperature at which water, by evaporating into air, can bring the air to saturation at the same temperature adiabatically. An adiabatic saturator is a device using which one can measure theoretically the adiabatic saturation temperature of air.

As shown in Fig.4, an adiabatic saturator is a device in which air flows through an infinitely long duct containing water. As the air comes in contact with

water in the duct, there will be heat and mass transfer between water and air. If the duct is infinitely long, then at the exit, there would exist perfect equilibrium between air and water at steady state. Air at the exit would be fully saturated and its temperature is equal to that of water temperature. The device is adiabatic as the walls of the chamber are thermally insulated. In order to continue the process, makeup water has to be provided to compensate for the amount of water evaporated into the air. The temperature of the make-up water is controlled so that it is the same as that in the duct.

After the adiabatic saturator has achieved a steady-state condition, the temperature

indicated by the thermometer immersed in the water is the *thermodynamic wet-bulb temperature*. The thermodynamic wet bulb temperature will be less than the entering air DBT but greater than the dew point temperature.

Certain combinations of air conditions will result in a given sump temperature, and this can be defined by writing the energy balance equation for the adiabatic saturator. Based on a unit mass flow rate of dry air, this is given by:

$$h_1 = h_2 - (W_2 - W_1)h_f \quad \square \quad 13$$

where h_f is the enthalpy of saturated liquid at the sump or thermodynamic wet-bulb temperature, h_1 and h_2 are the enthalpies of air at the inlet and exit of the adiabatic saturator, and W_1 and W_2 are the humidity ratio of air at the inlet and exit of the adiabatic saturator, respectively.

It is to be observed that the thermodynamic wet-bulb temperature is a thermodynamic property, and is independent of the path taken by air. Assuming the humid specific heat to be constant, from the enthalpy balance, the thermodynamic wet-bulb temperature can be written as:

$$t_2 = t_1 - \frac{h_{fg,2}}{c_{pm}} (w_2 - w_1) \quad \square \quad 14$$

where $h_{fg,2}$ is the latent heat of vaporization at the saturated condition 2. Thus measuring the dry bulb (t_1) and wet bulb temperature (t_2) one can find the inlet humidity ratio (W_1) from the above expression as the outlet saturated humidity ratio (W_2) and latent heat of vaporizations are functions of t_2 alone (at fixed barometric pressure).

On the psychrometric chart as shown in Fig.4, point 1 lies below the line of constant enthalpy that passes through the saturation point 2. $t_2 = f(t_1, W_1)$ is not a unique function, in the sense that there can be several combinations of t_1 and W_1 which can result in the same sump temperature in the adiabatic saturator. A line passing through all these points is a constant wet bulb temperature line. Thus all inlet conditions that result in the same sump temperature, for example point 1' have the same wet bulb temperature. The line is a straight line according to the straight line law. The straight-line joining 1 and 2 represents the path of the air as it passes through the adiabatic saturator.

Normally lines of constant wet bulb temperature are shown on the psychrometric chart. The difference between actual enthalpy and the enthalpy obtained by following constant wet-bulb temperature is equal to $(w_2-w_1)hf$.

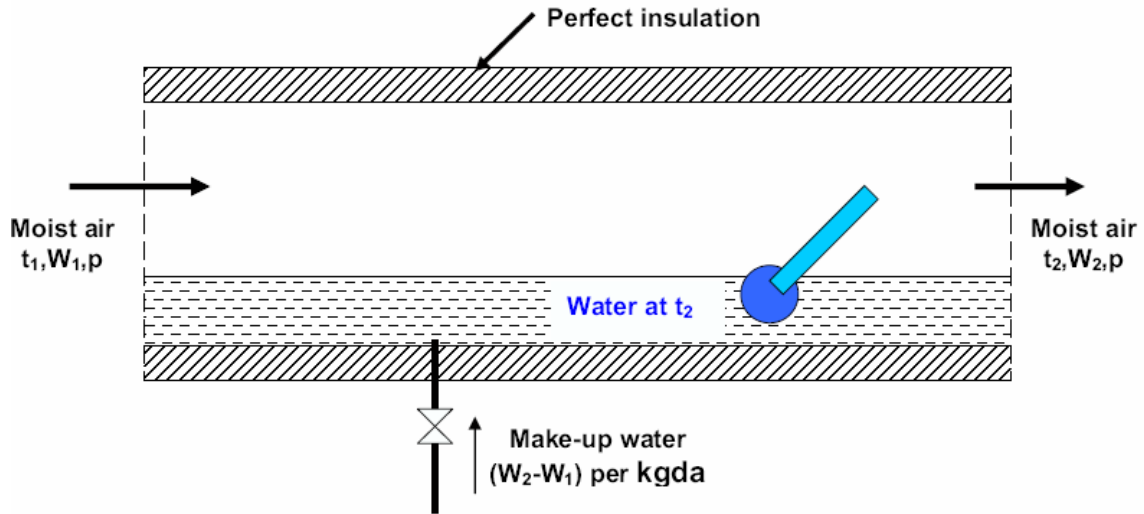


Fig.4: The process of adiabatic saturation of air

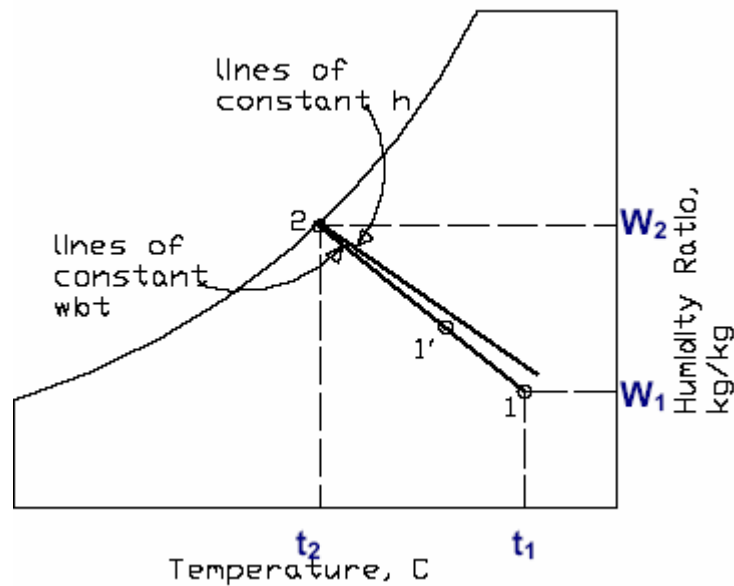


Fig.5: Adiabatic saturation process 1-2 on psychrometric chart

Wet-Bulb Thermometer:

In practice, it is not convenient to measure the wet-bulb temperature using an adiabatic saturator. In stead, a thermometer with a wetted wick is used to measure the wet bulb

temperature as shown in Fig.6. It can be observed that since the area of the wet bulb is finite, the state of air at the exit of the wet bulb will not be saturated, in stead it will be point 2 on the straight line joining 1 and i, provided the temperature of water on the wet bulb is i. It has been shown by Carrier, that this is a valid assumption for air-water mixtures. Hence for air-water mixtures, one can assume that the temperature measured by the wet-bulb thermometer is equal to the thermodynamic wet-bulb temperature⁴. For other gas-vapor mixtures, there can be appreciable difference between the thermodynamic and actual wet-bulb temperatures.

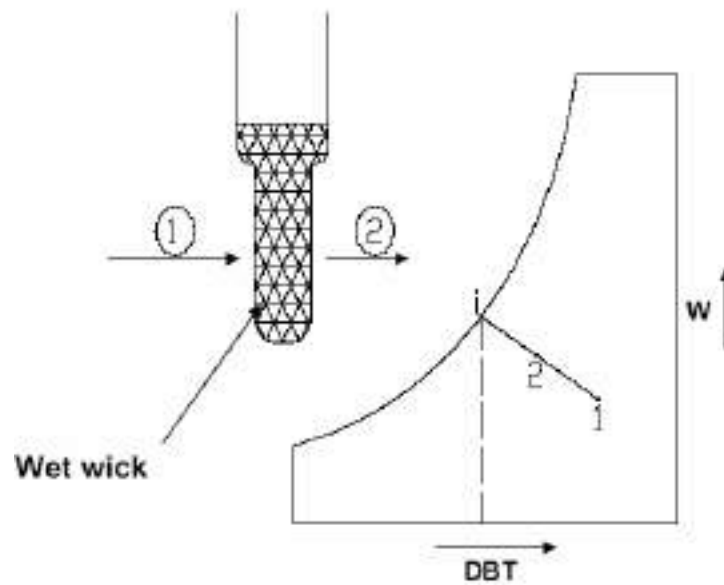


Fig.6: Schematic of a wet-bulb thermometer and the process on psychrometric chart

Calculation of psychrometric properties from p, DBT and WBT:

As mentioned before, to fix the thermodynamic state of moist air, we need to know three independent properties. The properties that are relatively easier to measure, are: the barometric pressure, dry-bulb temperature and wet-bulb temperature. For a given barometric pressure, knowing the dry bulb and wet bulb temperatures, all other properties can be easily calculated from the psychrometric equations. The following are the empirical relations for the vapor pressure of water in moist air:

i) Modified Apjohn equation:

$$p_v = p'_v - \frac{1.8p(t - t')}{2700} \quad \square \quad 15$$

ii) **Modified Ferrel equation:**

$$p_v = p'_v - 0.00066p(t - t') \left[1 + \frac{1.8t}{1571} \right] \quad \square 16$$

iii) **Carrier equation:**

$$p_v = p'_v - \frac{1.8(p - p'_v)(t - t')}{2800 - 1.3(1.8t + 32)} \quad \square 17$$

where t = dry bulb temperature, °C

t' = wet bulb temperature, °C

p = barometric pressure

p_v = vapor pressure

p'_v = saturation vapor pressure at wet-bulb temperature

The units of all the pressures in the above equations should be consistent.

Once the vapor pressure is calculated, then all other properties such as relative humidity, humidity ratio, enthalpy, humid volume etc. can be calculated from the psychrometric equations presented earlier.

Psychrometer:

Any instrument capable of measuring the psychrometric state of air is called a psychrometer. As mentioned before, in order to measure the psychrometric state of air, it is required to measure three independent parameters. Generally two of these are the barometric pressure and air dry-bulb temperature as they can be measured easily and with good accuracy.

Two types of psychrometers are commonly used. Each comprises of two thermometers with the bulb of one covered by a moist wick. The two sensing bulbs are separated and shaded from each other so that the radiation heat transfer between them becomes negligible. Radiation shields may have to be used over the bulbs if the surrounding temperatures are considerably different from the air temperature.

The *sling psychrometer* is widely used for measurements involving room air or other applications where the air velocity inside the room is small. The sling psychrometer consists of two thermometers mounted side by side and fitted in a frame with a handle

for whirling the device through air. The required air circulation (≈ 3 to 5 m/s) over the sensing bulbs is obtained by whirling the psychrometer (≈ 300 RPM). Readings are taken when both the thermometers show steady-state readings.

In the *aspirated psychrometer*, the thermometers remain stationary, and a small fan, blower or syringe moves the air across the thermometer bulbs.

The function of the wick on the wet-bulb thermometer is to provide a thin film of water on the sensing bulb. To prevent errors, there should be a continuous film of water on the wick. The wicks made of cotton or cloth should be replaced frequently, and only distilled water should be used for wetting it. The wick should extend beyond the bulb by 1 or 2 cms to minimize the heat conduction effects along the stem.

Other types of psychrometric instruments:

1. Dunmore Electric Hygrometer
2. DPT meter
3. Hygrometer (Using horse's or human hair)

Introduction:

In the design and analysis of air conditioning plants, the fundamental requirement is to identify the various processes being performed on air. Once identified, the processes can be analyzed by applying the laws of conservation of mass and energy. All these processes can be plotted easily on a psychrometric chart. This is very useful for quick visualization and also for identifying the changes taking place in important properties such as temperature, humidity ratio, enthalpy etc. The important processes that air undergoes in a typical air conditioning plant are discussed below.

Important psychrometric processes:

a) Sensible cooling:

During this process, the moisture content of air remains constant but its temperature decreases as it flows over a cooling coil. For moisture content to remain constant, the surface of the cooling coil should be dry and its surface temperature should be greater than the dew point temperature of air. If the cooling coil is 100% effective, then the exit temperature of air will be equal to the coil temperature.

However, in practice, the exit air temperature will be higher than the cooling coil temperature. Figure 1 shows the sensible cooling process O-A on a psychrometric chart. The heat transfer rate during this process is given by:

$$Q_c = m_a(h_o - h_A) = m_a c_{pm}(T_o - T_A) \quad \square \quad 1$$

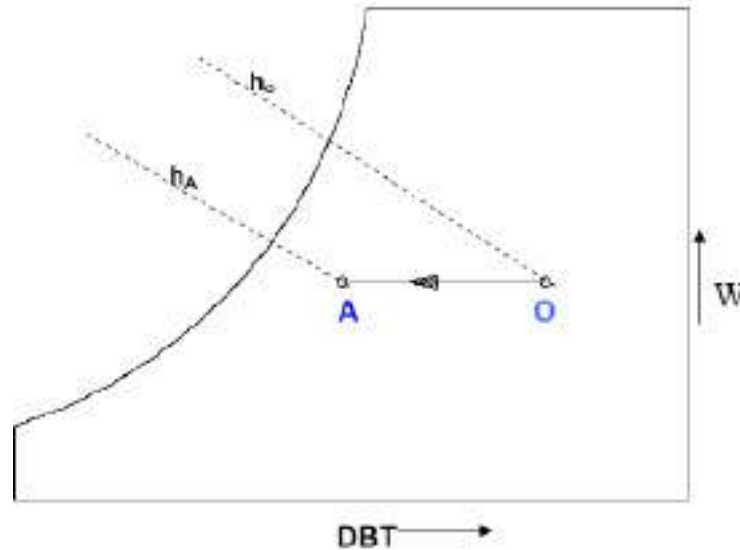


Fig.1: Sensible cooling process O-A on psychrometric chart

b) Sensible heating Process (O-B):

During this process, the moisture content of air remains constant and its temperature increases as it flows over a heating coil. The heat transfer rate during this process is given by:

$$Q_h = m_a (h_B - h_O) = m_a c_{pm} (T_B - T_O) \quad \square \quad 2$$

where c is the humid specific heat ($\square 1.0216$ kJ/kg dry air) and m_a is the mass flow rate of dry air (kg/s). Figure 2 shows the sensible heating process on a psychrometric chart.

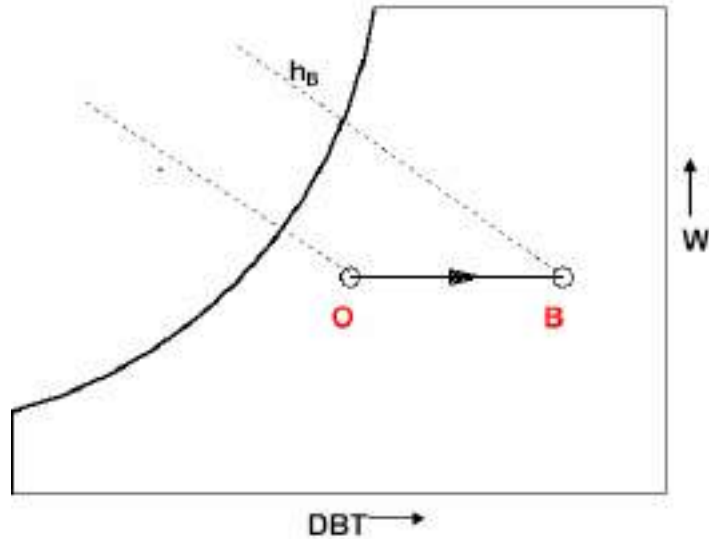


Fig.2: Sensible heating process on psychrometric chart

c) Cooling and dehumidification (Process O-C):

When moist air is cooled below its dew-point by bringing it in contact with a cold surface as shown in Fig.3, some of the water vapor in the air condenses and leaves the air stream as liquid, as a result both the temperature and humidity ratio of air decreases as shown. This is the process air undergoes in a typical air conditioning system. Although the actual process path will vary depending upon the type of cold surface, the surface temperature, and flow conditions, for simplicity the process line is assumed to be a straight line. The heat and mass transfer rates can be expressed in terms of the initial and final conditions by applying the conservation of mass and conservation of energy equations as given below:

By applying mass balance for the water:

$$m_a \cdot w_O = m_a \cdot w_C + m_w$$

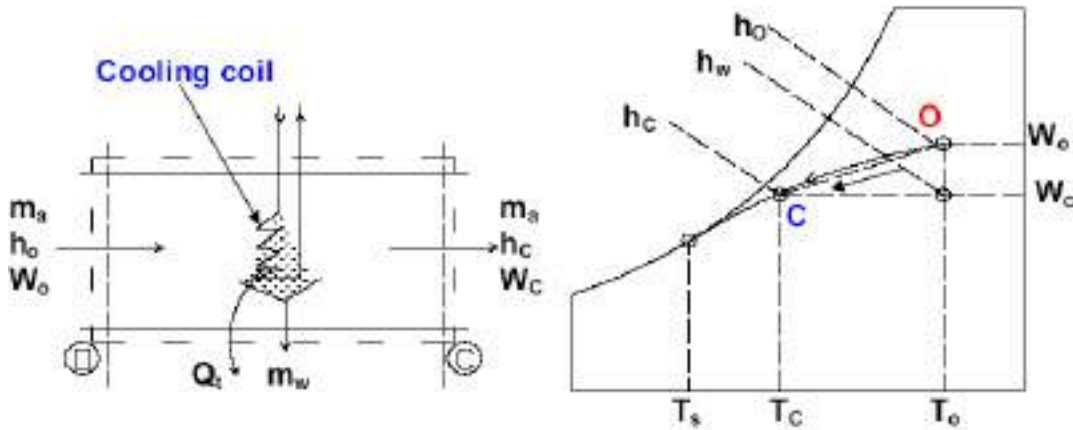


Fig..3: Cooling and dehumidification process (O-C)

By applying energy balance:

$$m_a \cdot h_o = Q_t + m_w \cdot h_w + m_a \cdot h_c \quad \square 4$$

from the above two equations, the load on the cooling coil, Q is given by:

$$Q_t = m_a (h_o - h_c) - m_a (w_o - w_c) h_w \quad \square 5$$

the 2nd term on the RHS of the above equation is normally small compared to the other terms, so it can be neglected. Hence,

$$Q_t = m_a (h_o - h_c) \quad \square 6$$

It can be observed that the cooling and de-humidification process involves both latent and sensible heat transfer processes, hence, the total, latent and sensible heat transfer rates (Q_t , Q_l and Q_s) can be written as:

$$\begin{aligned} Q_t &= Q_l + Q_s \\ \text{where } Q_l &= m_a (h_o - h_w) = m_a \cdot h_{fg} (w_o - w_c) \\ Q_s &= m_a (h_w - h_c) = m_a \cdot c_{pm} (T_o - T_c) \end{aligned} \quad \square 7$$

By separating the total heat transfer rate from the cooling coil into sensible and latent heat transfer rates, a useful parameter called Sensible Heat Factor (SHF) is defined. SHF is defined as the ratio of sensible to total heat transfer rate, i.e.,

$$\text{SHF} = Q_s / Q_t = Q_s / (Q_s + Q_l) \quad \square 8$$

From the above equation, one can deduce that a SHF of 1.0 corresponds to no latent heat transfer and a SHF of 0 corresponds to no sensible heat transfer. A SHF of 0.75 to 0.80 is quite common in air conditioning systems in a normal dry-climate. A lower value of SHF, say 0.6, implies a high latent heat load such as that occurs in a humid climate.

From Fig.3, it can be seen that the slope of the process line O-C is given by:

$$\tan c = \frac{\Delta w}{\Delta T} \quad \square 9$$

From the definition of SHF,

$$\frac{1 - \text{SHF}}{\text{SHF}} = \frac{Q_l}{Q_s} = \frac{m_a h_{fg} \Delta w}{m_a c_{pm} \Delta T} = \frac{2501 \Delta w}{1.0216 \Delta T} = 2451 \frac{\Delta w}{\Delta T} \quad \square 10$$

From the above equations, we can write the slope as:

$$\tan c = \frac{1}{2451} \left(\frac{1 - \text{SHF}}{\text{SHF}} \right) \quad \square 11$$

Thus we can see that the slope of the cooling and de-humidification line is purely a function of the sensible heat factor, SHF. Hence, we can draw the cooling and dehumidification line on psychrometric chart if the initial state and the SHF are known. In some standard psychrometric charts, a protractor with different values of SHF is provided. The process line is drawn through the initial state point and in parallel to the given SHF line from the protractor as shown in Fig.4.

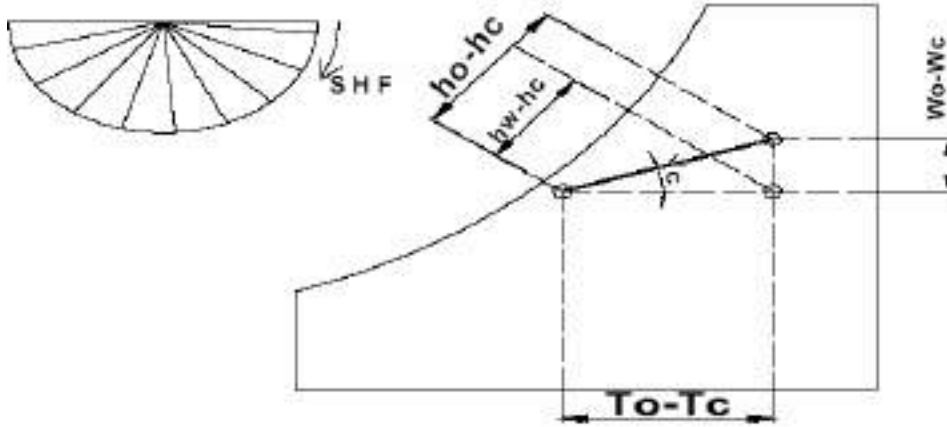


Fig.4: A psychrometric chart with protractor for SHF lines

In Fig.3, the temperature T_s is the effective surface temperature of the cooling coil, and is known as apparatus dew-point (ADP) temperature. In an ideal situation, when all the air comes in perfect contact with the cooling coil surface, then the exit temperature of air will be same as ADP of the coil. However, in actual case the exit temperature of air will always be greater than the apparatus dew-point temperature due to boundary layer development as air flows over the cooling coil surface and also due to temperature variation along the fins etc. Hence, we can define a by-pass factor (BPF) as:

$$\text{BPF} = \frac{T_c - T_s}{T_o - T_s} \quad \square \quad 12$$

It can be easily seen that, higher the by-pass factor larger will be the difference between air outlet temperature and the cooling coil temperature. When BPF is 1.0, all the air by-passes the coil and there will not be any cooling or de-humidification. In practice, the by-pass factor can be increased by increasing the number of rows in a cooling coil or by decreasing the air velocity or by reducing the fin pitch.

Alternatively, a contact factor(CF) can be defined which is given by:

$$\text{CF} = 1 - \text{BPF} \quad \square \quad 13$$

d) Heating and Humidification (Process O-D):

During winter it is essential to heat and humidify the room air for comfort. As shown in Fig.5., this is normally done by first sensibly heating the air and then adding water vapour to the air stream through steam nozzles as shown in the figure.

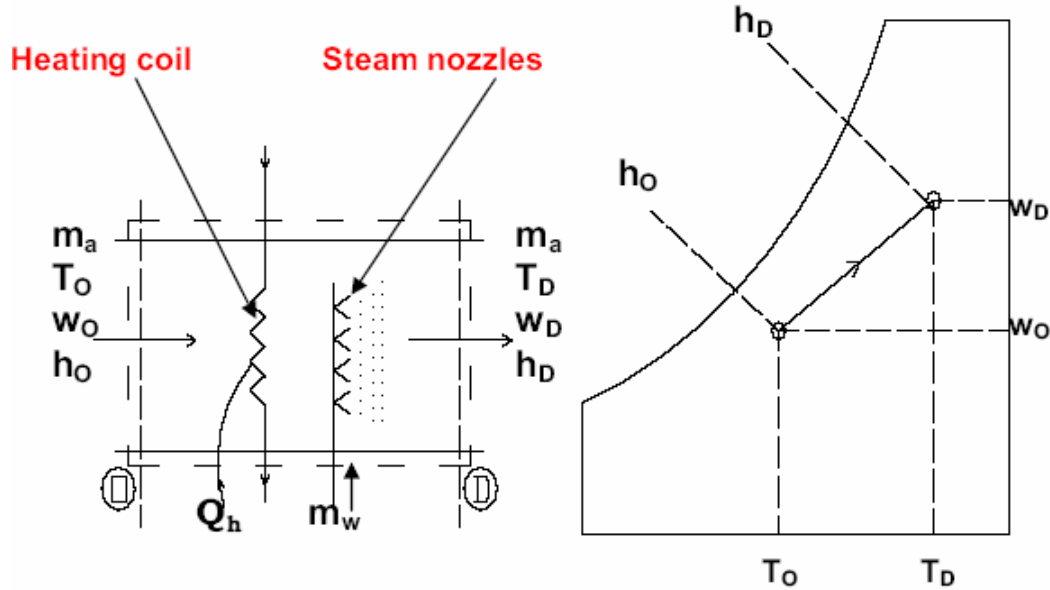


Fig.5: Heating and humidification process

Mass balance of water vapor for the control volume yields the rate at which steam has to be added, i.e., m_w :

$$m_w = m_a (w_D - w_o) \quad \square$$

14 where m is the mass flow rate of dry air. a

From energy balance:

$$Q_h = m_a (h_D - h_o) - m_w h_w \quad \square \quad 15$$

where Q_h is the heat supplied through the heating coil and h_w is the enthalpy of steam. Since this process also involves simultaneous heat and mass transfer, we can define a sensible heat factor for the process in a way similar to that of a cooling and dehumidification process.

e) Cooling & humidification (Process O-E):

As the name implies, during this process, the air temperature drops and its humidity increases. This process is shown in Fig.6. As shown in the figure, this can be

achieved by spraying cool water in the air stream. The temperature of water should be lower than the dry-bulb temperature of air but higher than its dew-point temperature to avoid condensation ($T_{DPT} < T_w < T_o$).

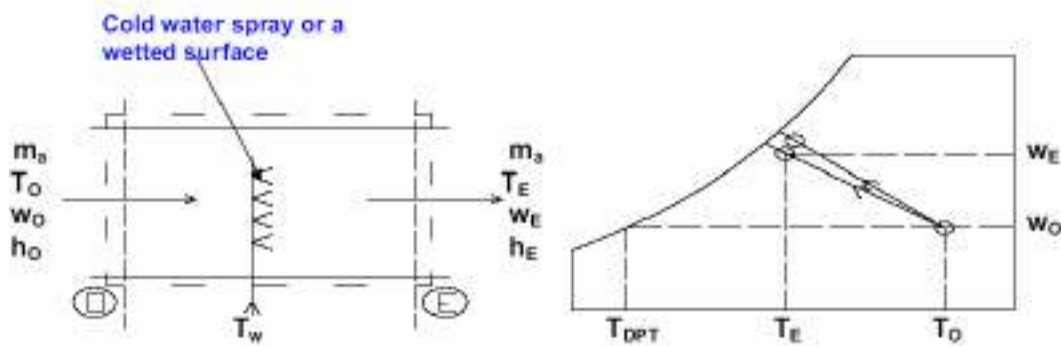


Fig.6: Cooling and humidification process

It can be seen that during this process there is sensible heat transfer from air to water and latent heat transfer from water to air. Hence, the total heat transfer depends upon the water temperature. If the temperature of the water sprayed is equal to the wetbulb temperature of air, then the net transfer rate will be zero as the sensible heat transfer from air to water will be equal to latent heat transfer from water to air. If the water temperature is greater than WBT, then there will be a net heat transfer from water to air. If the water temperature is less than WBT, then the net heat transfer will be from air to water. Under a special case when the spray water is entirely recirculated and is neither heated nor cooled, the system is perfectly insulated and the make-up water is supplied at WBT, then at steady-state, the air undergoes an adiabatic saturation process, during which its WBT remains constant. This is the process of adiabatic saturation discussed in Chapter The process of cooling and humidification is encountered in a wide variety of devices such as evaporative coolers, cooling towers etc.

f) Heating and de-humidification (Process O-F):

This process can be achieved by using a hygroscopic material, which absorbs or adsorbs the water vapor from the moisture. If this process is thermally isolated, then the enthalpy of air remains constant, as a result the temperature of air increases as its moisture content decreases as shown in Fig.7. This hygroscopic material can be a solid or a liquid. In general, the absorption of water by the hygroscopic material is an

exothermic reaction, as a result heat is released during this process, which is transferred to air and the enthalpy of air increases.

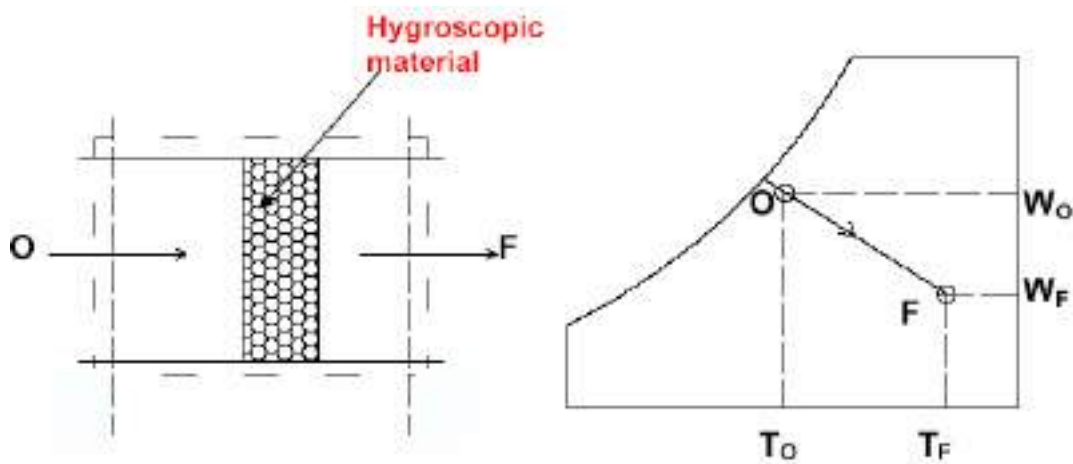


Fig.7. Chemical de-humidification process

g) Mixing of air streams:

Mixing of air streams at different states is commonly encountered in many processes, including in air conditioning. Depending upon the state of the individual streams, the mixing process can take place with or without condensation of moisture.

i) Without condensation: Figure 8 shows an adiabatic mixing of two moist air streams during which no condensation of moisture takes place. As shown in the figure, when two air streams at state points 1 and 2 mix, the resulting mixture condition 3 can be obtained from mass and energy balance.

From the mass balance of dry air and water vapor:

$$m_{a,1}w_1 + m_{a,2}w_2 = m_{a,3}w_3 = (m_{a,1} + m_{a,2})w_3 \quad \square 16$$

From energy balance:

$$m_{a,1}h_1 + m_{a,2}h_2 = m_{a,3}h_3 = (m_{a,1} + m_{a,2})h_3 \quad \square 17$$

From the above equations, it can be observed that the final enthalpy and humidity ratio of mixture are weighted averages of inlet enthalpies and humidity ratios. A generally valid approximation is that the final temperature of the mixture is the weighted average

of the inlet temperatures. With this approximation, the point on the psychrometric chart representing the mixture lies on a straight line connecting the two inlet states. Hence, the ratio of distances on the line, i.e., (1-3)/(2-3) is equal to the ratio of flow rates $m_{a,2} / m_{a,1}$. The resulting error (due to the assumption that the humid specific heats being constant) is usually less than 1 percent.

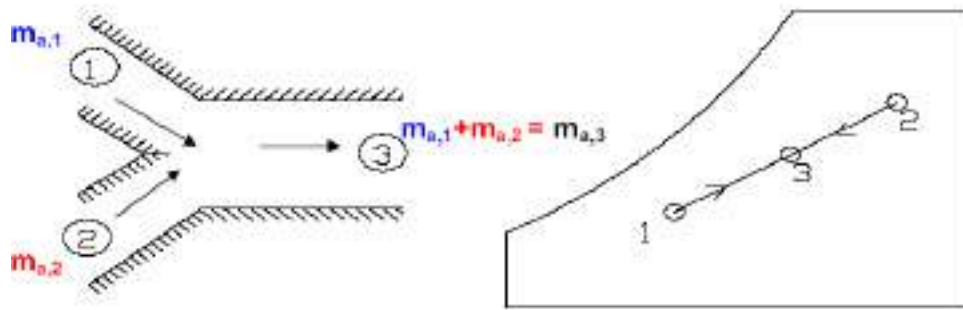


Fig.8. *Mixing of two air streams without condensation*

ii) Mixing with condensation:

As shown in Fig.9, when very cold and dry air mixes with warm air at high relative humidity, the resulting mixture condition may lie in the two-phase region, as a result there will be condensation of water vapor and some amount of water will leave the system as liquid water. Due to this, the humidity ratio of the resulting mixture (point 3) will be less than that at point

4. Corresponding to this will be an increase in temperature of air due to the release of latent heat of condensation. This process rarely occurs in an air conditioning system, but this is the phenomenon which results in the formation of fog or frost (if the mixture temperature is below 0oC). This happens in winter when the cold air near the earth mixes with the humid and warm air, which develops towards the evening or after rains.

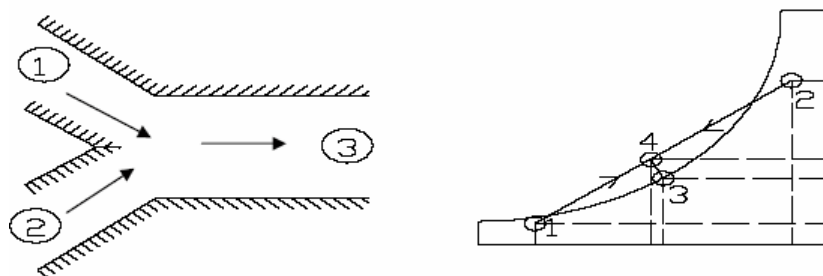


Fig.9. *Mixing of two air streams with condensation*

3. Air Washers:

An air washer is a device for conditioning air. As shown in Fig.10, in an air washer air comes in direct contact with a spray of water and there will be an exchange of heat and mass (water vapour) between air and water. The outlet condition of air depends upon the temperature of water sprayed in the air washer. Hence, by controlling the water temperature externally, it is possible to control the outlet conditions of air, which then can be used for air conditioning purposes.

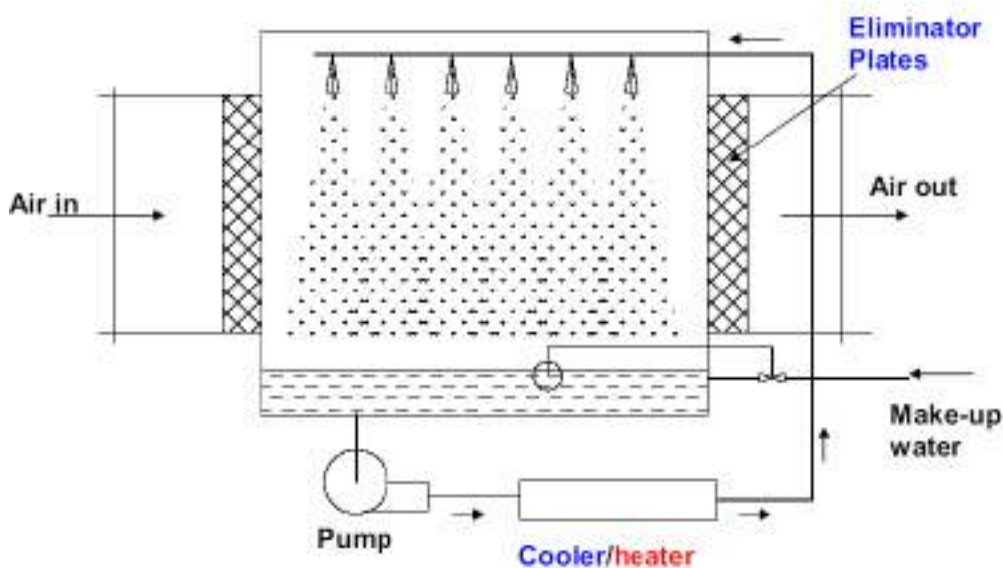


Fig.10: Air washer

In the air washer, the mean temperature of water droplets in contact with air decides the direction of heat and mass transfer. As a consequence of the 2nd law, the heat transfer between air and water droplets will be in the direction of decreasing temperature gradient. Similarly, the mass transfer will be in the direction of decreasing vapor pressure gradient. For example,

a) Cooling and dehumidification: $t_w < t_{DPT}$. Since the exit enthalpy of air is less than its inlet value, from energy balance it can be shown that there is a transfer of total energy from air to water. Hence to continue the process, water has to be externally cooled. Here both latent and sensible heat transfers are from air to water. This is shown by Process O-A in Fig.11.

b) Adiabatic saturation: $t_w = t_{WBT}$. Here the sensible heat transfer from air to water is exactly equal to latent heat transfer from water to air. Hence, no external cooling or

heating of water is required. That is this is a case of pure water recirculation. This is shown by Process O-B in Fig.11. This the process that takes place in a perfectly insulated evaporative cooler.

c) Cooling and humidification: $t_{DPT} < t_w < t_{WBT}$. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from air to water, hence, water has to be cooled externally. This is shown by Process O-C in Fig.11.

d) Cooling and humidification: $t_{WBT} < t_w < t_{DBT}$. Here the sensible heat transfer is from air to water and latent heat transfer is from water to air, but the total heat transfer is from water to air, hence, water has to be heated externally. This is shown by Process O-D in Fig.11. This is the process that takes place in a cooling tower. The air stream extracts heat from the hot water coming from the condenser, and the cooled water is sent back to the condenser.

e) Heating and humidification: $t_w > t_{DBT}$. Here both sensible and latent heat transfers are from water to air, hence, water has to be heated externally. This is shown by Process O-E in Fig.11.

Thus, it can be seen that an air washer works as a year-round air conditioning system. Though air washer is a and extremely useful simple device, it is not commonly used for comfort air conditioning applications due to concerns about health resulting from bacterial or fungal growth on the wetted surfaces. However, it can be used in industrial applications.

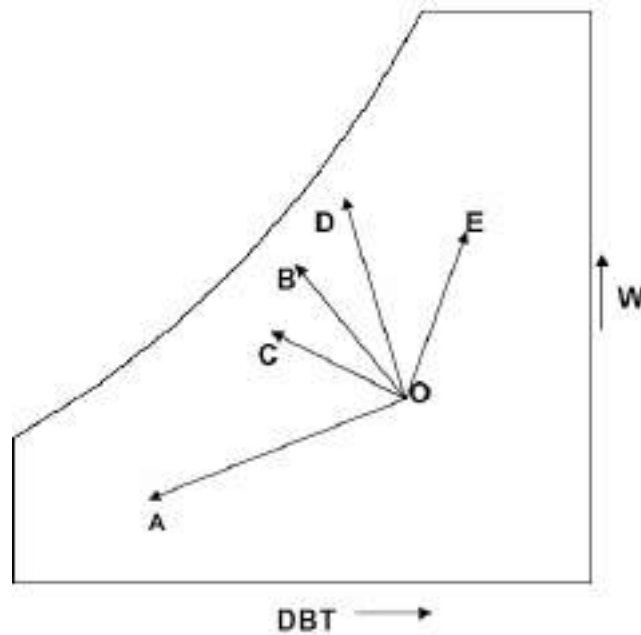


Fig.11: Various psychrometric processes that can take place in an air washer

4. Enthalpy potential:

As shown in case of an air washer, whenever water (or a wetted surface) and air contact each other, there is possibility of heat and moisture transfer between them. The directions of heat and moisture transfer depend upon the temperature and vapor pressure differences between air and water. As a result, the direction of the total heat transfer rate, which is a sum of sensible heat transfer and latent heat transfers also depends upon water and air conditions. The concept of enthalpy potential is very useful in quantifying the total heat transfer in these processes and its direction.

The sensible (Q_s) and latent (Q_L) heat transfer rates are given by:

$$Q_S = h_C A_S (t_i - t_a)$$

$$Q_L = \dot{m}_w \cdot h_{fg} = h_D \cdot A_S (w_i - w_a) \cdot h_{fg} \quad \square 18$$

the total heat transfer Q_T is given by:

$$Q_T = Q_S + Q_L = h_C A_S (t_i - t_a) + h_D \cdot A_S (w_i - w_a) \cdot h_{fg} \quad \square 19$$

where t_i = dry-bulb temperature of air, °C

t_a = temperature of water/wetted surface,

°C w_a = humidity ratio of air, kg/kg

w_i = humidity ratio of saturated air at t_i , kg/kg

h_c = convective heat transfer coefficient, W/m²

°C h_D = convective mass transfer coefficient,

kg/m² hfg = latent heat of vaporization, J/kg

Since the transport mechanism that controls the convective heat transfer between air and water also controls the moisture transfer between air and water, there exists a relation between heat and mass transfer coefficients, h_c and h_D as discussed in an earlier chapter. It has been shown that for air-water vapor mixtures,

$$h_D \approx \frac{h_c}{c_{pm}} \text{ or } \frac{h_c}{h_D \cdot c_{pm}} = \text{Lewis number} \approx 1.0 \quad \square \quad 20$$

where c_{pm} is the humid specific heat \square 1.0216 kJ/kg.K.

Hence the total heat transfer is given by:

$$Q_T = Q_S + Q_L = \frac{h_c A_s}{c_{pm}} [(t_i - t_a) + (w_i - w_a) \cdot h_{fg}] \quad \square$$

21 by manipulating the term in the parenthesis of RHS, it can be shown that:

$$Q_T = Q_S + Q_L = \frac{h_c A_s}{c_{pm}} [(h_i - h_a)]$$

thus the total heat transfer and its direction depends upon the enthalpy difference (or potential) between water and air ($h_i - h_a$). i a

if $h_i < h_a$; then the total heat transfer is from water to air and water gets cooled

if $h_i > h_a$; then the total heat transfer is from air to water and water gets heated

if $h_i = h_a$; then the net heat transfer is zero, i.e., the sensible heat transfer rate is equal to but in the opposite direction of latent heat transfer. Temperature of water remains at its wet bulb temperature value

The concept of enthalpy potential is very useful in psychrometric calculations and is frequently used in the design and analysis of evaporative coolers, cooling towers, air washers etc.

Unit -V

Objective:

To impart knowledge on Applications of air-conditioning and heat pumps.

Outcome:

Ability to explain the air-conditioning equipment and heat pump circuits.

Human Comfort: Requirements of Temperature, Humidity and Concept of Effective Temperature, Comfort Chart. Heat Pump – Heat Sources – Different Heat Pump Circuits.

Air Conditioning Equipment: Humidifiers – Dehumidifiers – Air Filters, Fans and Blowers.

THERMODYNAMICS OF HUMAN BODY

A human body feels comfortable thermodynamically when the heat produced by the metabolism of human body is equal to the sum of the heat dissipated to the surroundings and the heat stored in human body by raising the temperature of body tissues. These phenomena can be represented by the following equation.

$$Q_M = \pm Q_s + Q_E \pm Q_R \pm Q_c \quad \dots(1)$$

where Q_M = Metabolic heat produced within the body (kJ/h.)

$\pm Q_s$ = Stored energy in temperature rise of body tissues; +ve when the tissue temperature rises and -ve when it falls.

Q_E = Evaporative heat loss due to the water evaporating at skin surface and the lungs expressed in kJ/h.

$\pm Q_R$ = Heat loss and gain by radiation; +ve when heat is lost to the surrounding, and -ve, when it is gained from the surroundings kJ/h.

Q_c = Heat loss or gain by conduction and convection; +ve when heat is lost, and -ve, when heat is gained from the surroundings in kJ/h.

- The rate at which the body produces heat is termed as the metabolic rate, the value of which depends upon a number of factors such as individual's

health, his physical activity and his environments. The metabolic heat produced depends on the rate of food energy consumption in body.

— A fasting man, the weak or sick, will have less metabolic heat production.

If Q_p , Q_R and Q_c are high and +ve, Q_s will become -ve when Q_M is low and hence sick, old weak or fasting man feels more cold when exposed to low temperature, high draft air.

— A man gets fever, when internal body activities increase Q_M to such an extent that Q_s becomes +ve for given Q_E , Q_K and Q_c . The stored energy Q_s has the maximum and minimum limits which when exceeded brings death.

— Body temperature when exceeds 40.5°C and falls below 36.7°C is dangerous. The usual body temperature when $Q_s = 0$ is 37°C for a normal man. There is some kind of thermostatic control in human body, which tries to maintain temperature of human body at the normal level of 37°C .

• Whenever the environment temperature changes, the human body reacts by trying many physiological adjustments to keep the deep temperature of the body constant. There are two types of adjustments generally adopted by the human control system.

(i) Vasomotor control. This control regulates the blood supply to the skin. It acts by causing vasodilation of the peripheral blood vessels, if environment temperature is increased. The increased circulation of the blood increases the convective transport of heat from the interior of the body to the surface. In other words, the thermal conductance of the skin is increased, thus minimising body heating. The vasomotor mechanism alone is sufficient to maintain the heat balance at a low level of heat load, and is therefore known as "First Line of Defence".

If environment temperature is decreased, the control causes vasoconstriction of the peripheral blood vessels. This decreases blood circulation and reduces body cooling.

(ii) Sudomotor Control. This control regulates sweat production. Whenever the heat loss by convection and radiation becomes negative due to high temperature of atmosphere compared to body temperature, the only mode of heat transfer to dissipate the heat is by evaporation.

The sudomotor control acts by initiating sweat gland activity. The sweating capacity differs according to persons degree of acclimatization to heat and work.

FACTORS AFFECTING COMFORT

Air-conditioning of buildings mainly concerns the comfort of people and not the maintenance of exact conditions as required for products. The conditions conducive to comfort depend upon the following factors:

1. Temperature
2. Humidity
3. Air motion
4. Air purity.

Temperature. Proper control of temperature of the air medium surrounding the body removes a physiological stress of accommodation, thereby making for greater comfort and improved physical well-being and health.

Humidity. It is through evaporation from skin that a large proportion of body heat is lost. Since evaporation is promoted by a low relative humidity of the air and is related by a high humidity control has an important effect on the comfort. Extremes of humidity not only results undesirable physiological relations but also effect (usually adversely) the properties of many substances in the treated space, clothing and furniture in particular.

Air motion. Air movement over the body increases rate of heat and moisture dissipation above the still-air rate, thereby modifying the feeling of warmth or coldness. Depending on the amounts of motion there are pleasing and displeasing effects.

Air Purity. People feel uncomfortable when breathing contaminated air even if it is within acceptable temperature and humidity ranges. Therefore proper filtration, cleaning and purification of air is necessary to keep it away from dust, dirt and other properties.

EFFECTIVE TEMPERATURE

In addition to major four factors of temperature, humidity, air movement, purity of air there are several other factors such as clothing, economical status of persons, climate (a person living in the cold climates takes about 7 to 9 days to become acclimatized to the hot environments) to which a person is accustomed to, etc., which influence human comfort and deserve due consideration. Unfortunately it is not practically feasible to study the combined effects of all the factors on human comfort. Therefore, it is usual practice to consider the aforesaid four factors for most applications which are included in a term called the 'effective temperature*.

- Effective Temperature is defined as a sensory index that combines into a single factor the effects of temperature, humidity and air movement on human comfort in a noise free pure air environment. The effective temperature corresponds to the dry-bulb temperature of the saturated air at which a given percentage of people feel comfortable. For example, at 21°C (and RH = 100% and air movement 8m/min.) most people feel comfortable.

Although a particular effective temperature may have a humidity ranging all the way from 0 to 100% and air motion from still air to high velocities, it does not follow that all these combinations are equally comfortable. Each combination produces the same sensation of warmth or coolness but other affects may contribute a feeling of discomfort.

— Too low a humidity produces a parched condition of skin, mouth, and nose.

— Too high humidity causes an accumulation of moisture in clothing and promotes the production of body odours.

— High air motion will usually cause annoyance because of its directional effects such as blowing of hair.

— When the ranges of humidity and air motion are limited to avoid these physiological or psychological regions of discomfort the effective temperature index may be said to measure comfort fairly closely. This is on the assumption that air purity and noise level is maintained at an acceptable standard.

• Refer Fig.1. This chart shows as to how the human comfort remains invariant with temperature of confinement when the air motion and humidity are changed. For example, if a room is maintained at 25°C, a man feels the same comfort provided the moisture and air movement are increased. Similarly the effects of other parameters can also be explained. The amount of moisture is varied by controlling the wet bulb temperature. The above effects can be better understood with the help of thermal analysis for human body.

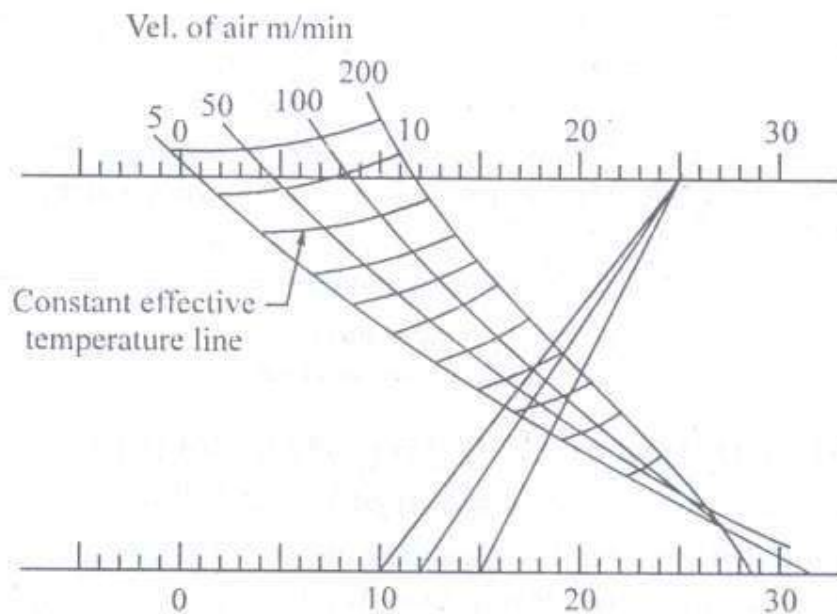


Fig 1. Effective temperature chart.

Comfort Air conditioning

The empirically limited effective temperature chart is known as 'Comfort chart'. This chart is prepared on the two scales (Fig.1) having constant relative humidity lines. The effective temperature is represented by broken lines. The per cent of people feeling comfortable is shown for summer as well

as winter season. Effective temperature for summer air-conditioning is 21°C when 100% people feel comfortable. On the other hand for winter air-conditioning the temperature has to be kept at about 21.7°C for 100% comfort. However, the statistical study on the comfort air-conditioning shows that in the slightly cool condition the comfort of human beings is not influenced by moisture from RH = 30% to 90%. But as the temperature increases, the comfort is influenced by moisture content. At $t_{db} = 25.3^{\circ}\text{C}$, 70% people feel comfortable from RH = 25% to 70%. But any moisture increase beyond this value requires reduction in dry bulb temperature in order to maintain the level of comfort. At around $t_{db} = 28^{\circ}\text{C}$ slightly warm condition and $t_{db} = 30.5^{\circ}\text{C}$, 40% people feel comfortable from RH = 25% to 60%. Thereafter the increase in moisture requires decrease in DBT.

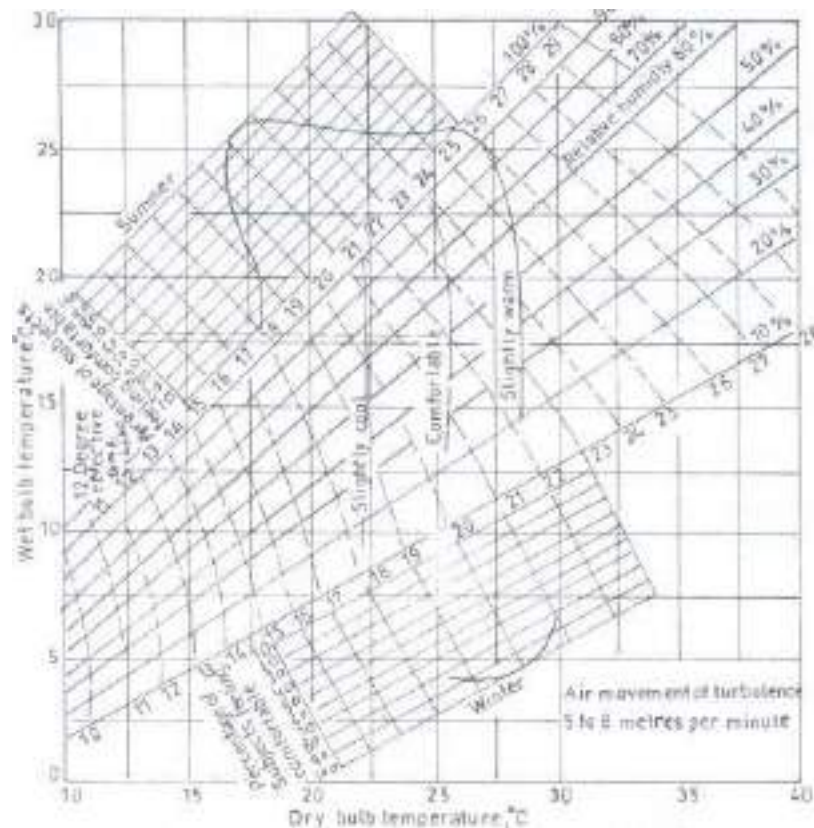


Fig. 1. Comfort chart.

FACTOR GOVERNING OPTIMUM EFFECTIVE TEMPERATURE

The optimum effective temperature is affected/governed by the following important factors:

1. Climatic and seasonal difference. It is generally agreed that people living in the colder climates are comfortable in effective temperatures lower than for the persons living in warmer regions. The optimum indoor effective temperature appears to bear some relation to the average outdoor effective temperature. This changes slowly with the season of the year, and day-to-day variations seem to have relatively little effect. The total shift in the optimum from summer to winter is shown in the comfort chart.
2. Duration of occupancy. By experience it has been observed that the shorter the occupancy, the more the effective temperature line tends to shift from the continuous- occupancy optimum in the direction of the outdoor effective temperature. This shift, while not great, is large enough to be taken into account.
3. Clothing. In winter, much of the effect due to climate and variable occupancy is compensated, because people can dress comfortably to suit both outdoor and indoor conditions by merely adding or removing garments. In considering any application, therefore, allowance should be made for the amount of clothing worn. For example, a formal dining room with women present in evening dress will require a high temperature than the average restaurant.
4. Age and sex. As compared to men, women require 0.5°C higher effective temperature. Furthermore, both men and women over forty on the average require an effective temperature about 0.5°C higher than do those below forty.
5. Activity. The desirable room effective temperature is influenced by the rate of bodily activity. The men performing heavy manual labour in factories need a lower temperature than occupants seated at rest in auditoriums.
6. Radiant heat. The effect of body radiant heat from person to person particularly in a crowded place like theatre or a auditorium is large enough to require a slight lowering in effective temperature that could otherwise be maintained.

7. Latitude. With every 5° reduction in latitude 0.5°C increase in effective temperature is desired.

8. Shock effect. The shock effect is due to sudden entrance from or exit to outdoors, and is likewise mostly a summer problem. This effect is due to two factors: (i)

relatively inflexibility in the amount of clothing so that evaporation of perspiration makes for a chilling effect and (ii) outdoor exertion gives beneficial warmth in winter and objectionable overheating of the body in summer. Careful tests have proved, however, that the shock effect is not physiologically harmful.

INTRODUCTION

The primary requirement of cooling or heating equipment is that these must be able to remove or add heat at the rate at which it is produced, or removed and maintain the given comfort conditions in the room.

- The proportion of sensible heat to latent heat decides the slope of 'sensible heat ratio' or enthalpy humidity difference ratio' and hence also the condition at which the conditioned air must enter the room because it lies on the line drawn from the room condition at the slope of above ratios.
- The cost of the equipment will be quite high if it is designed for maximum heating or cooling loads. The owner may be prepared to tolerate some uncomfot for a short time to save capital cost investment. If the equipment is designed for average loads, there may be long periods of uncomfot and the equipment may soon become unpopular. Hence the capacity equipment must be estimated at a value, which wisely accounts for the physical and economical comfort of the owner. It is only the exact and detailed estimation of heating and cooling loads which helps in this decision.

The Estimation of load involves the following variables:

1. Magnitude and direction of wind velocity.
2. Outside humidity and temperature.
3. Nature of construction, materials used.

4. Orientation of openings. windows and doors.
5. Periods of occupancy and the number of persons in the room, activities of the persons etc.
 - One has to be satisfied with the estimation under assumed conditions keeping the variables fixed and allowances must be made for, when the actual conditions are different from those assumed. Thus to make precise load calculation is unfortunately not simple and some of the sources of the load are I difficult to predict and evaluate. For a practising engineer it is not possible to make detailed calculations I and therefore he must consult latest editions of 'Heating, ventilation and air conditioning guide' and make quick easy estimates.

COOLING-LOAD ESTIMATE

For .air-conditioning the cooling-load can be classified as follows:

1. Room load-which falls on the room directly.
2. Total load-which falls on the air-conditioning apparatus.

Room Load

Room sensible heat (RSH):

- (i) Solar and transmission heat gain through walls, roof, etc.
- (ii) Solar and transmission heat gain through glass.
- (iii) Transmission gain through partition walls, ceiling, floor etc.
- (iv) Infiltration.
- (v) Internal heat gain from people, power, lights, appliances etc.
- (vi) Additional heat gain not accounted above, safety factor etc.
- (vii) Supply duct heat gain, supply duct leakage loss and fan power.

Room Latent Heat (RLH):

- (i) Infiltration.
- (ii) Internal heat from people. steam, appliances etc.
- (iii) Vapour transmission.
- (iv) Additional heat gain not accounted above, safety factor etc.
- (v) Supply duct leakage loss.

It may be noted that if we add by-passed air load to room sensible heat, we shall get effective room sensible heat (ERSH). Similarly adding by-passed air load to room latent heat we shall get effective room latent heat' (ERLH).

Grand total load (on air-conditioning apparatus)(a)Sensible heat:

- (i) Effective room sensible heat.
- (ii) Sensible heat of the outside air that is not by-passed.
- (iii) Return duct heat gain, return duct leakage gain, dehumidifier pump power and dehumidifier and piping losses.

Total sensible heat (TSH) is obtained by adding items (i) to (iii).

(b) Latent heat

- (i) Effective room latent heat.
- (ii) Latent heat of outside air which is not by-passed.
- (iii) Return duct leakage gain.

Total latent heat (TLH) is obtained by adding items (i) to (iii).

Grand total heat (GTH) = Total sensible heat (TSH) + Total latent heat (TLH) i.e., $GTH = TSH + TLH$

HEATING-LOAD ESTIMATE

Heating-load estimate is prepared on the basis of 'maximum probable heat/ass' of the room or space to be heated. Following points should be considered while making heat-load calculations:

1. Transmission heat loss. The transmission heat loss from walls, roof, etc., is calculated on the basis of just the outside and inside temperature difference.
2. Solar radiation. Normally there is no solar radiation present and hence no solar heat gain at the time of the peak load which normally occurs in the early morning hours.
3. Internal heat gains. The heating requirement is reduced due to internal heat gains from occupants, lights, motors and machinery etc.

SOLAR RADIATION:

- The solar radiation intensity normal to the sun's rays incident upon

a plane surface situated in the limits of the earth's atmosphere, varies with the time of the year as the distance of the earth from the sun changes. Its value when the earth is at its mean distance from the sun is called the solar constant. The normal value of the solar constant is assumed as $5045 \text{ kJ/m}^2\text{-h}$. Radiation received at the surface of the earth is much less because, much of it, while passing through earth's atmosphere, is scattered and absorbed by dust and vapour particles and the gases in the atmosphere.

- The solar heat reaches part of the earth's surface in the form of two radiations:
 1. Beam or direct radiation: The part of the sun's radiation which travels through the atmosphere and reaches the earth's surface directly is called Beam or direct radiation. It is maximum when the surface is normal to the sun rays. The intensity of the radiation can be increased or decreased by changing the orientation of the surface.
 2. Diffuser sky radiation: A large part of the sun's radiation is scattered, reflected back into space and absorbed by the earth's atmosphere. A part of this radiation is re-radiated and reaches the earth's surface uniformly from all directions. It is called diffuse or sky radiation. It does not normally change with orientation of the surface.
- The total solar radiation reaching a surface is equal to the sum of the direct and diffuse radiations.

SOLAR HEAT GAIN THROUGH GLASS

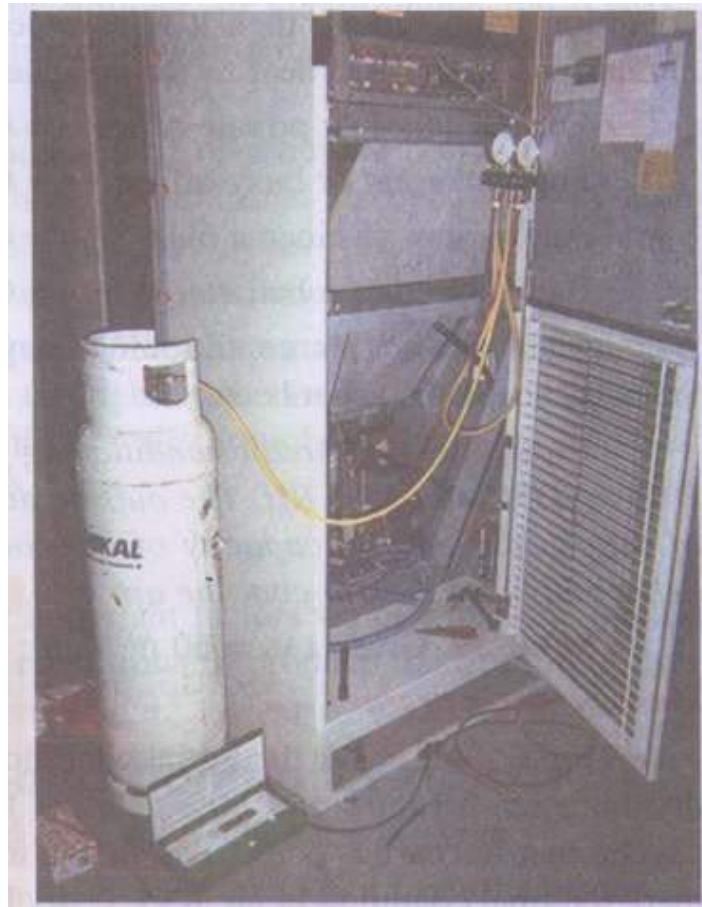
- Glass which is major material of most buildings, provides the most direct route for entry of solar radiation. For these reasons, the proper estimation of heat gain through glass is necessary.
- Heat transmitted through a glass surface depends on the wavelength of radiation and physical and chemical characteristics of the glass. Part of the radiation is absorbed, part is reflected and the rest is transmitted. Glass is opaque to the radiant energy emitted from sources below 200°C . Thus glass has high transmittivity for short wavelength and low transmittivity for long wave length radiation.
- Direct solar heat gain can be reduced by using different types of glass,

glass construction and shades as given below:

- (i) Double pan glass reduces the solar heat by 10% to 20%.
- (ii) Special heat absorbing glass reduces the solar heat by 25%.
- (iii) Stained glass can reduce it upto 65% depending upon its colour.
- (iv) Shading devices installed on the outside of windows reduce sun load upto 15%.
- (v) Ventilation blinds and curtain shades reduce it by 30 to 35%.

Industrial Air Conditioning System

It is an important system of air conditioning these days in which the inside dry bulb temperature and relative humidity of the air is kept constant for proper working of the machines and for the proper research and manufacturing processes. Some of the sophisticated electronic and other machines need a particular dry bulb temperature and relative humidity. Sometimes, these machines also require a particular method of psychrometric processes. This type of air conditioning system is used in textile mills, paper mills, machine-parts manufacturing plants, tool rooms, photo-processing plants etc.



Industrial air-conditioning system.

Air Conditioning System

We have already discussed in Art. 18.2, the four important factors which affect the human comfort. The system which effectively controls these conditions to produce the desired effects upon the occupants of the space is known as an air conditioning system.

Equipments Used in an Air Conditioning System

Following are the main equipments or parts used in an air conditioning system:

1. Circulation fan. The main function of this fan is to move air to and from the room.
2. Air conditioning unit. It is a unit, which consists of cooling and dehumidifying processes for summer air conditioning or heating and humidification processes for winter air conditioning.
3. Supply duct. It directs the conditioned air from the circulating fan to the space to be air conditioned at proper point.
4. Supply outlets. These are grills, which distribute the conditioned air evenly in the room.
5. Return outlets. These are the openings in a room surface which allow the room air to enter the return duct.
6. Filters. The main function of the filters is to remove dust, dirt and other harmful bacteria from the air.

Classification of Air Conditioning Systems: The air conditioning systems may be broadly classified as follows:

1. According to the purpose
 - (a) Comfort air conditioning system, and
 - (b) Industrial air conditioning system.
2. According to season of the year
 - (a) Winter air conditioning system,
 - (b) Summer air conditioning system, and
 - (c) Year-round air conditioning system.
3. According to the arrangement of equipment
 - (a) Unitary air conditioning system, and
 - (b) Central air conditioning system.

AIR Filters: There are different types of air filters which are used for removing dust from the air. Adoption of a particular air-filter depends on the nature of the dust, type of dust and required cleanliness in the air-conditioned space. The filters are broadly classified in five groups as given below

1. Dry filters
2. Electric filters
3. Viscous filters
4. Centrifugal dust collectors
5. Wet filters

The purpose of all dust removal equipments is to remove or reduce the concentration of dust to a very small fraction of its original in the conditioned air. This provides health and reduces interior cleaning cost. The most effective system works at the lowest cost of cleaning equipment and with smallest drop in pressure through the cleaning-equipment.

1. Dry-Filters: Dry-filters are subdivided into two forms as cleanable filters and throw-away filters. Dry-filters are usually made of cloth, coarse paper, wool or cellulose felt. The dust in the air will be trapped or screened when it is passed through this filtering medium. The velocity of air allowed through these filters ranges between 2 m to 15 m/min. The filtering materials are often arranged in bag forms to provide necessary surface without excessive surface requirements. The cleaning of these filters can be done by providing shaking or rapping action of the filters during operation. Dry filters are capable of collecting 99% or more dusts as small as 0.5 micron. A bag form dry filter is shown, in Fig.1.

Throw-away filters are made of glass wool, plastic fibres, steel wool, animal hairs or vegetable fibres. These filters are available in the market in different sizes and forms. The pads of above mentioned materials are set into permanent steel frames. If a resin powder is added with dry wool, then electrostatic charge develops which is sufficient to attract and retain small diameter aerosols. Addition of resin powder increases the filtering efficiency.

The dry-type air filters remove tiny dust particles of 0.3 to 10 μ m diameter very satisfactorily. Since it has a limited dust holding capacity, it cannot be used where the dust concentration in the air is higher than 2.5 grams per 1000 m³. The dry type filters are suitable for removing dust in all ordinary applications in metropolitan and industrial communities but are not suitable for filtering the air drawn through a hood over a sand blasting or a grinding operation in an industrial plant. The dry-type filters are not capable to remove the smoke from air.

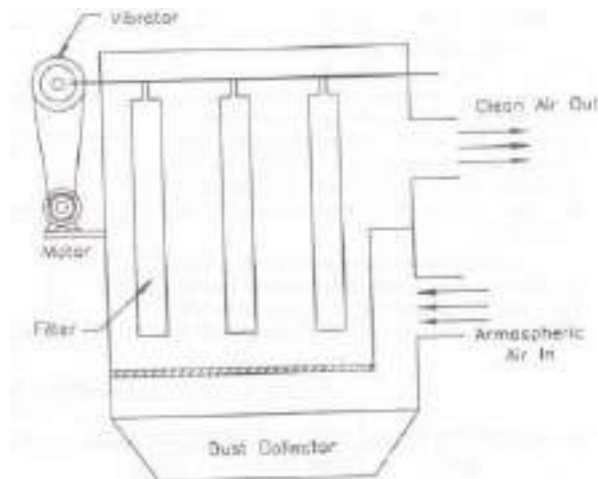


Fig: 1 Bag type filter.

2. Viscous Filters: This types of filters are better than any other type. These filters are made in form of pads and bats using glass wool, steel wool, plastic fibres or copper mesh. These pads are impregnated with viscosine which is oily substance. The cardboard frame can be used so that the filter can be thrown away when it is full of dust. Some viscous filters can be washed in gasoline and can be used again. The viscosine used must have the following properties.

1. It must have constant viscosity over a wide temperature range in order to be sticky enough when cold and it should not flow when hot.
2. It must contain some germicidal action to prevent the growth of bacteria.

3. It should not evaporate more than 1% of its weight during the life time of the filter.

Self-cleaning Viscous Filters: The self-cleaning viscous filters take the form of a continuous roll of material coated with the oil and is driven by motor across the air stream as shown in Fig.2. The filtering screen is continuously running over rolls at the top and bottom while the bottom roll is in a sump filled with viscous liquid which acts to wash the screen and to recharge with the fresh viscous liquid.

The filter curtain is rotated by a simple ratchet at an intermittent speed resulting in a travel of 10 cm. per hour. The principal advantage of self-cleaning - filter over others is the reduction of the maintenance cost. It has a cleaning efficiency of 97% with - 0.7 cms of water head loss at 150 m/min air velocity through the filter. Viscous filters are available from 80 m³/min to 600 m³/min capacity. Any number of units may be grouped in form of bank up to 250 m³/min flow of air.

2. **Wet-Filters:** Water spray type air washer is also known as air filter. In this type of air-filter, the dust particles are wetted by water spray and then owing to the additional weight of the water, the particles fall to the bottom. The effectiveness of washer in removing the dust depends on the "wettability" of the dust by water. It is almost impossible to wet a greasy particle like pollens.

These types of filters are extensively used in industrial area for the absorption of soluble gases, which are dangerous and unhealthy for the occupants. Sometimes, chemicals are added in water to absorb some specific gases.

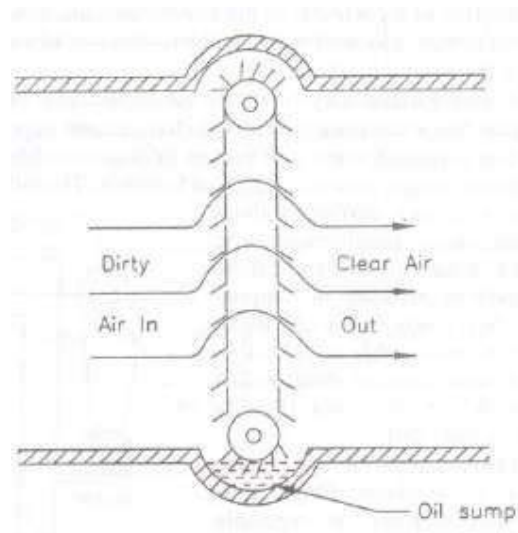


Fig: 2 Self viscous filters

The effectiveness of this washer can be increase by increase the number of the sprays along the direction of air-flow as shown in Fig.3. It is never recommended to use an air-washer of any sort without a filter. This can be successfully used in combination with dry or viscous type filters for removing the dissolving gases like SO_2 .

4. Electronic Filters: The application of electronic filters for removing the dust is very recent. Dr. Cottell in 1906 developed. The commercial form but only in 1937 commercial equipment was applied for practical purposes. The principle for electric filter is illustrated in Fig. 4.

Air is passed between a pair of oppositely charged conductors and it become ionised as the voltage applied between the conductors is sufficiently large (8000 to 15000 V). As the air is passed through this ionised chamber, both negative and positive ions are formed, the latter being larger in quantity (20% negatively charged and 80% positively charged). The air carrying the ions and coming out of the ionising chamber is further passed through the collecting unit. This unit consists of a set of vertical metal plates, spaced 15 to 20 mm apart. Alternate plates are positively charged and earthed and attract the negatively and positively charged dust particles respectively. The voltage applied to the plates is approximately half the potential of the ionizing wires. As the alternate plates are grounded, high intensity electrostatic field exists between the plates. When

the charged dust particles are passed between the plates, the electrostatic field exerts a force on charged particles and drives them towards the grounded plates. To remove tile dust accumulation, the collector plates are cleaned periodically by washing them with hot water spray.

Odour Removal: Control of odours in occupied spaces is a health as well as a comfort necessity. The principle sources of air contamination are body odours resulting from breathing, insensible perspiration and products of organic decomposition. Contamination of the air occurs either directly from the bodies of the occupants and from their clothes. The seriousness of the odour problem varies with the personal cleanliness, density of occupancy and degree of exertion of the occupants. Odours emitted from manufacturing plants are usually complex because they are made up from the mixture of many materials.

Sources of odour are so many and so diverse that it is difficult to suggest specific remedies for the emissions they create. Many of the products causing odours are self-oxidizing and disappear in a short period when the time interval and mixing volume are sufficient.

The different types of materials which are responsible for odour carried into air are given in table 1. If the odour is from outside air (industrial area) then the odour absorber is placed in the path of outside air supply before entering into the room, but if the odour is from the conditioned space itself then the absorber is placed in the return recirculated air.

Table 1 Odour Elements

odour Causing Element	Source	Concentration in Kg per million cu. m. of air
Butyric acid	Body-odour	1.5
Valerie acid	Body-odour	0.01
Pyridine	Burning tobacco	0.035
Iodoform	--	0.0255
Creso	--	0.85
Aerolein	Frying fat	1.00
Essential oils	Perfumes	1.00

Only small portion of the total air recirculated needs to pass through the adsorber as the activated carbon is practically 100% efficient and the resultant mixture of odour free air with the balance of the recirculated air is of sufficiently low concentration to give satisfactory results. The arrangement of the system is shown in Fig.1.

In a restaurant and similar installations, the minimum outside air is 0.35 m³/min per person for keeping the effect of odour below tolerable concentration. This increased outdoor air quantity will always require a large cooling in summer and a large heating in winter with resultant increased operating costs. This can be minimised by introducing inside odour absorber system.

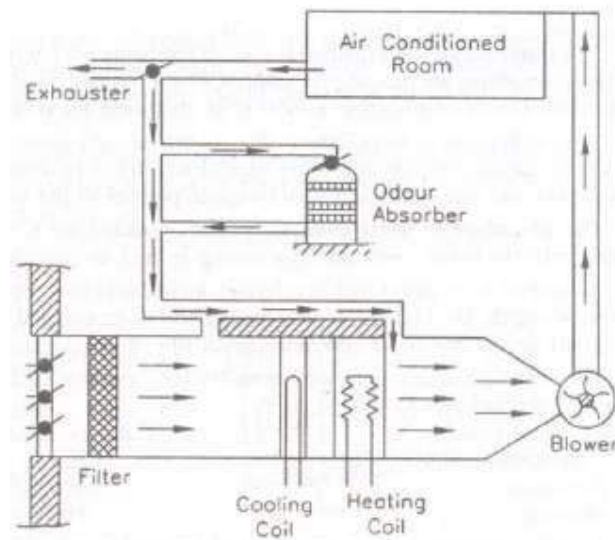


Fig. 1 Odour absorber system for inside source

Most odours are either eliminated or reduced in concentration by the following methods.

1. Modification of the Process
2. Dilution with outdoor air
3. Masking and neutralization
4. Adsorption
5. Combustion or incineration
6. Condensation

7. Scrubbing

1. Process Modification: Many times, slight changes in process minimise odour generation and are more effective and more economical than control devices. Modification in manufacture process could involve lowering the process temperatures or such parameters. This is commonly done in foundry ovens, mineral wool plants,

varnish cookers and paint baking ovens. The temperature in fertilizer plant using brewery waste is adjusted to assure that there is no vaporization.

2. Dilution Method: The odours can be eliminated at or near the source whenever possible. If the odour is not dangerous, it can be diluted until the concentration is below the objectionable level. This is very common approach in many industries. For example, in producing polyurethane foam, the tolylene-diisocyanate is soon diluted to a nondetectable concentration.

3. Sensory Modification Method: There are two sensory modification methods. Masking does not alter the composition of the origin, an odour where neutralization does modify the original odour. In sensory modification, perfume or deodorant may be added directly to the process. Perfumes, colognes and deodorants are odour masking elements which release a pleasant odour to overcome an unpleasant smell.

Use of ozone for odour control is also a masking method. The concentration of ozone required to destroy odours is normally toxic. The toxic qualities of ozone cause drowsiness and headache and reduce their ability to perceive odours. Ozone treatment has been employed in processing of fish and other foods, paint and varnish, plastics and petroleum, paper and fertilizer plants as well as in breweries. Odour neutralization is the method used to eliminate or diminish the intensity of the original odour. Selection of neutralization agents depends mainly on the experience of the person in that field. A series of precatalogued agents is available for treating a variety of specific odours.

4. Adsorption: Charcoal (active carbon) is used mostly in locations where

noxious gases are plentiful. Charcoal is an absorbent of odours in the same way that silica-gel is an adsorbent for moisture except that odour absorption does not generate heat. Charcoal is able to absorb condensable vapour and gases which come in contact with its surface and hold them until released by reactivation. It has a strong affinity for organic gases and hydrocarbons, the small charcoal containers are capable of handling 1 cu. m of air with a pressure drop of 0.5 cm. of water.

HUMIDIFIERS

The humidification of air is one of the important phase of air-conditioning system. The success of many air-conditioning systems depends upon the proper functioning of humidifiers. The humidification is achieved by using one of the following four methods:

(A) Injecting the steam (B) Atomizing the water (C) Evaporating the water (D) By air washing.

Common Considerations for Humidification

The selection of water quality for humidification is very important as the water introduced into the air to raise the RH affects humidification equipment operation. As water evaporates, a white crystalline solid is usually formed. The solid consists primarily of calcium and sodium salts contained in the water. The effect of solids varies with the type of equipment selected and its material. Commercial water softening replaces the hardness causing minerals (normally calcium) with sodium, which creates another type of mineral problem. Another water related problem is algae. It can be found in stagnant water in reservoirs of some types of humidification equipment. Adding an algicide to the water normally prevents the growth.

All humidification systems except steam humidifier consist of an arrangement for exposing water surface either in the form of small droplets or as wet surfaces. In order to change water to vapour nearly 2400 kJ of heat is required for every kg of water. This heat is taken from the air which is to be humidified and cooling of air takes place. In adiabatic humidifying process,

the total heat of air before and after humidification remains constant therefore the WBT also remains constant.

Humidification by Injecting the Steam

In this system, humidification of air is carried out by injecting steam into the air just above atmospheric pressure. The steam condenses to a very fine mist as it is dispersed, and evaporates almost instantly to the gaseous state, raising the R.H. The process does not raise the temperature to appreciable amount (0.5 to 1°C). Steam humidifiers are widely used whenever excess capacity is readily available from a boiler providing steam for heating or processing. In some applications, steam used in humidifiers would otherwise be wasted since its pressure is often too low for process

applications. The steam humidifier when the steam flows from the available source through a strainer and pressure reducer into the outer steam jacket of the distribution manifold. The steam heats the manifold and prevents condensation from forming inside. The strainer prevents dirt from reaching the working parts and the pressure reducer regulates the steam at the required pressure for the humidifier. When the control valve opens (as humidistat operates), steam flows through the separator to remove condensate and then enters the distribution manifold. Condensate from the separator goes out of the trap to an open drain to an open drain or into the condensate return system.

DEHUMIDIFIERS

Dehumidification of air (removal of moisture) is necessary for the storage, manufacture and packing of variety of products in addition to comfort air-conditioning.

Many storage areas are maintained at a specific RH to protect products from corrosion. Rust and oxidation type corrosive reactions will not occur if the RH is below 40%. Material stored in such atmosphere need not be coated with surface protecting materials.

In the food industry, storage room humidity must be kept low to prevent formation of mold, deterioration of the product and destruction of packing. Hygroscopic materials such as sugar, salt and synthetics are stored in humidity controlled areas to prevent them from taking moisture from the surrounding atmosphere.

There are many uses of dry air in manufacturing, processing and production operations. Dehumidification prevents hard sugar candy from becoming sticky. In the pharmaceutical industry, many products can be mixed, packaged only in extremely dry atmosphere. For example, in a tableting operation, powder will not compress if the moisture content is not controlled.

Handling powdery hygroscopic products such as flour, cocoa, sugar and salt is practically impossible if moisture content of the air is not regulated. In electronic industry, moisture causes a multitude of problems. These range from pitting of electrical contacts because of excessive arcing when humidity is high to bursting of casings on electrical heating elements when moisture is sealed in casing during manufacture.

Many packing operations require dry air. If only a small amount of water vapour is present during the packaging of potato chips, caramel corn, instant coffee and other extremely hygroscopic materials, they will become rancid and stale after a very short period. Using dehumidified air during packaging keeps moisture out of the container and prevents such problems. There are three common methods to accomplish dehumidification.

1. By reducing the temperature of the air below its DPT This is accomplished by passing the air over cooling coil whose surface temperature is maintained below DPT of the air.
2. By absorption of moisture from the air. This is accomplished by passing the air through adsorption bed. The moisture in the air does not enter into chemical combination with the medium through which it is

passed. It is simply removed from the air.

3. By adsorption of moisture from the air. This is accomplished by passing the air through a chemical. The moisture in the air enters into chemical combination with dry agent

INDUSTRIAL APPLICATIONS OF HEAT PUMP

The applications of heat pump for warming and cooling of homes and offices have attracted considerable attention during recent years. The industrial field offers equal and more diverse opportunities for the utilization of the heat pump. There are many applications in different industries in which large quantities of cold and hot liquids are required. The necessary cooling and heating can be achieved with the help of heat pump simultaneously.

The different industrial applications of heat pump are listed below and few of them are discussed in detail.

1. Purification of water.
2. Concentration of juices, milk and sugar syrups.
3. The purification of salty water from sea source.
4. The concentration of dyes and chemicals.
5. Use for preparing powdered milk and table salt.
6. For the recovery of valuable solvents from different manufacturing process.
7. For year-round air-conditioning.

1. Thermo-Compressor: The compression type distillation plant was brought to a high degree of perfection during World War II. The army and navy had need for the production of drinking water from sea water in many areas and aboard ships.

The arrangement of the system is shown in colored Plate I. In this arrangement, the external heat source is removed by combining the condenser and evaporator into a single piece of equipment.

The low pressure vapour from the evaporator shell flows to the thermo-compressor and is discharged at a higher pressure to the condenser coil which is mounted in a submerged position, within the evaporator shell. The saturation temperature thus maintained on the condenser coil is sufficiently above the saturation temperature in the evaporator shell to support evaporation. Thus, the condenser acts as the heat source for the evaporator. There is adequate heat available for this purpose because the compressor adds the work of compression and delivers to the condenser. The thermal performance can be further improved by the inclusion of preheaters as shown in colored plate-I. Here, the cold feed on its way to the evaporator abstracts the heat of the distillate.

This arrangement offers thermal and practical advantages which justify its use. The EPR of the system lies between 10 to 15.

If the entire operating temperature level is raised, then the equipment can run at positive pressures both in evaporator and in condenser. This eliminates the need for vacuum pump to remove non-condensable gases. It further reduces the size of all parts and passages because of high fluid density.

If a source of power is available at a reasonable cost, the thermo-compressor offers substantial lower operating cost and less investment which have made its application entirely practical in many industries. The heat-pump, particularly can be used in hydroelectric areas where the electric power is cheaply available.

2. Juice Concentration Plant: The systematic arrangement of heat pump used for juice concentration is shown in colored Plate II. The high pressure vapour refrigerant from compressor is passed into the refrigerant condenser through desuperheater. The juice is pumped into the refrigerant condenser through the pump P_3 . The heat of condensation of vapour refrigerant is used to evaporate the water from weak juice which is pumped through P_3 . A recirculating pump P_1 is necessary in the circuit to make the juice concentrated. Thus the water vapour formed in the condenser C_1 is condensed in the condenser C_2 , which also works as an evaporator for the refrigerant cycle. The heat from the water vapour is given out to the low pressure refrigerant mixture coming out through expansion valve and separator. This heat is utilised to evaporate the low-pressure

refrigerant liquid as shown in Fig. and evaporated vapour is passed to the compressor, and the cycle is repeated. The water condensed in condenser C_2 is removed by pump P_4 .

It is necessary to remove the water from the juices at low temperature to maintain its taste and flavour, so the fruit juices and milk are boiled at considerable low temperatures. (Orange juice boils at 10 to 20°C at 10 to 30 mm. of Hg pressure). The required vacuum is maintained in the whole system by introducing a vacuum pump.

3. Desalination of sea water: Freezing processes, for the production of fresh water from sea water, generally fall into one of the two categories. Firstly, there is flash freezing process in which heat is removed from the brine by evaporating water under a high vacuum at the triple point temperature, simultaneously producing ice and water vapour. In the secondary process, a volatile refrigerant is evaporated in direct contact with the brine, thus exchanging the latent heat of evaporation of the refrigerant with the latent heat of fusion of the ice, Both processes produce ice as small crystals of pure water, which need to be separated from the adhering brine and subsequently melted to produce fresh water.

The flash process has the disadvantage that the freezing and melting processes take place under extremely, low pressure, and it is necessary to develop a special type of compressor to transfer the water vapour between these two process stages. Moreover, the physical size of this compressor is quite large, even for modest outputs.

In the secondary refrigerant process, these limitations do not occur. By suitable choice of refrigerant it is possible to carry out two fundamental operations of ice formation and ice melting at a pressure which is substantially at atmospheric. It is possible to use a single compressor for a plant capacity of a few million gallons per day. This system is known as Simen Freeze Desalination Process and has been developed very recently and many installations are in operation and under construction in recent years. The working of the process is described below.

In this process, the brine comes in direct contact with the refrigerant after coming out of the expansion I valve as shown in coloured Plate III. As the temperature of the refrigerant is considerably low (-20°C), the water crystals are formed in the crystallizer and crystals float on the brine due to their lower density I and they are removed in the wash column. The crystals formed are coated with the salt are washed by the pure water supplied from the crystal condenser to the wash chamber as shown in Fig. The washed crystals from the wash-chamber are further removed in the crystal condenser where these ice crystals are melted by giving the heat of compression of the refrigerant coming out from the compressor. This melted ice is pure water which is used for drinking purposes. The high pressure and high temperature refrigerant coming out from the compressor is also liquefied in the crystal-condenser giving its heat to the ice crystals. This high pressure liquefied refrigerant is again passed to the crystallizer through the expansion valve to repeat the cycle. The weak brine coming out from the wash chamber and crystallizer is recirculated again. An installation of 4 mgd capacity working on the above process is installed by U.K. atomic energy for public supply.

An important feature of the secondary refrigerant freeze (SRF) process is the mode of heat transfer. In the freezing and melting operations, heat is transferred by direct contact, without the intervention of metal surfaces. This enables temperature driving forces to be made extremely small (1 to 2°C) and results in minimum power consumption in the compressor. An added advantage of direct contact heat transfer is that scaling, and fouling of surfaces are eliminated which limit the thermodynamic efficiency.

The low temperature operation is of considerable advantage in that corrosion problems are minimized and plant life is correspondingly extended. It permits the use of less expensive plastic materials which are unaffected by brine and refrigerant. Biological fouling is also discouraged by low temperature operation and by the removal of O_2 in the process.

It should be noted that this process may be used for low salinity feed water containing say 5000 to 10,000ppm dissolved solids and considerable reductions in both capital and running costs are possible .with this type of system.

Choice of Refrigerant: The prime requirement for the refrigerant used in the SRF process is that it should be immiscible with water and brine. Also, it must not react chemically with water or brine and must not form hydrates. The vapour pressure, at temperature near the freezing point of brine, should preferably be greater than one atmosphere to avoid the construction of large vacuum vessels but should not be too high. Other wise it will be necessary to construct high cost pressure vessels. The refrigerant should also be non-toxic and non-inflammable. It must be relatively inexpensive and readily available.

n-butane is universally used as refrigerant in this process even it is highly inflammable and its vapour pressure is below atmospheric at the freezing point of brine. However, it is readily available in the quantities and quality demanded for large plants and comparatively inexpensive. Furthermore, it conforms to the more important requirement of limited solubility in water about 150 ppm and does not react chemically with water or form hydrates. In this respect, a comparison must be drawn with iso-butane which has a better vapour pressure, but unfortunately does form a hydrate with water and brine.

One further consideration which must be mentioned is the specific gravity of the liquid refrigerant.

If this is greater than unity, it may lead to difficulties with the crystallization process. If it is near unity, than problems may be experienced with the separation of liquid refrigerant from water and brine. Liquid butane has a specific gravity of about 0.6 and is therefore suitable in both these respects.

4. Room Air-conditioner: This is very convenient unit for cooling the rooms in hot summer. The different capacity units are available as per requirements.

The arrangement of the system is shown in coloured plate IV. The refrigerant cycle components are arranged in a compact unit. The whole unit is divided into two parts A and B as shwon in Fig. Each part is again subdivided into two parts.

The high pressure refrigerant vapour is passed from the compressor into the condenser where it is condensed passing the out-door air over the condenser coil by the fan 'F' The

condensed refrigerant is passed through the capillary into the evaporator where it is evaporated, passing the room air over the evaporator coils by the fan F₂. The room air passing over the evaporator is passed through the air filter 'F' and it is cooled by giving its heat to the low temperature refrigerant. The quantity of air circulated is controlled by the dampers D₁ and D₂ similarly the air passing over the condenser coil is controlled by the openings as shown in the figure. The evaporated refrigerant in the evaporator goes to the compressor and it repeats the cycle.

The air passing over the evaporating coil is dehumidified and water drips into the tray T₁ and the collected water in the tray T₁ is taken to the tray T₂ by the pipe as shown in the figure. The collected water in the tray T₂ evaporates to some extent and helps in cooling the compressor and condenser.

Both the fans F₁ and F₂ draw the air in from the lower portion of the unit and it is discharged to the room and atmosphere after passing over the evaporator and condenser coils.

When the required temperature is reached in the room, the unit automatically stops. This is accomplished by the thermostat and control panel as shown in the figure.

Location of Room Air-conditioner: The condenser operation is more efficient if it is located at cool space. This is more essential in room air-conditioner because the air is used for condensing the refrigerant. If the window unit is exposed to the sun's rays, the heat of radiation and warm air can diminish the efficiency of the exposed condenser and reduces the capacity of the unit. It is advisable to install the window unit for a minimum sun exposure. The order of preferential exposure for optimum performance is north-east or south-west. The shape of the room relative to air circulation, location of windows and the location preferred by the occupants often overrule the above considerations. Air circulation in the room is a primary consideration and good air-pattern should be planned to establish comfortable conditions to the occupants.

5. Heat Pump for Process Drying: There are numerous examples throughout the process industries where the energy required for product drying could be cut drastically by making the drying operation as part of the heating cycle.

In conventional processes, drying is achieved by passing hot air into the dryer and venting the humidified air, sometimes through a heat exchanger, which conserves energy by preheating the inflow air. In conventional dryers, the water evaporated has to leave as vapour usually moved with warm air. For efficient operation, it is generally recognized that the sensible heat lost in the warm air must be minimized and the humidity kept as high as possible. ...

What is not often realized is that the latent heat of the steam is also waste and that the efficiency of drying can be greatly increased by recovering this latent heat. To achieve this, the temperature below DPT should be maintained.

If this latent heat is to be used, it must be returned to the system at a higher temperature and the best way to do this is a heat pump. In a heat-pump dryer, the water leaves as a liquid. Almost invariably, a heat pump will assure higher energy efficiency in drying.

A. Batch Drying System: The basic heat-pump system used for a batch drying is shown in Fig.1. The working of the system is described below.

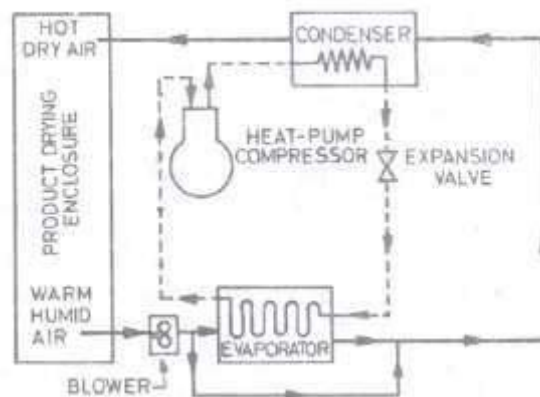


Fig.1 Heat pump assisted drying system for Batch drive

The heat source for dehumidifying heat pump is the humid air of the product drier-environment in which the machine operates. The air is first cooled by evaporating-refrigerant in the evaporator. Depending on the design of the evaporator, a portion of the air is cooled below the DPT. Some of its moisture condenses on the surface of the evaporator and runs to the drain. The cooled and dehumidified air is mixed with air in the chamber. This air cools the condenser and is heated in the process.

The air, having been heated and dehumidified can now absorb more moisture from the material being dried. The heat removed by the evaporator and heat equivalent of the compression work are transferred to the air by the condenser. The temperature during heat transfer depends on the mass flow of the air. To operate a heat pump at a fixed evaporating temperature with high COP, the condensing temperature must be as low as possible. This can be achieved only if the-mass flow of air over the condenser is higher than that over the evaporator.

The rates of evaporation from a solid material is proportional to the temperature difference between the surface and the air stream. The temperature of a wet surface will be identical to the wet-bulb temperature of the air. Evaporation can be increased either by reducing the wet bulb temperature or by increasing DBT.

In case of dry surfaces, evaporation rate is controlled by rate of moisture diffusion to the surface. This rate of diffusion is proportional to the temperature gradient in the solid, which in turn, depends upon the temperature of the air stream. High temperature air has a greater capacity for absorbing water vapour than low temperature air of the RH. The average performance of the high temperature (90°C condenser temperature) heat pump is about 3 Kg/kWh which is about 50% faster than that of available dehumidifier.

B. Thermo-compressor for continuous Drying: This type of heat pump drier is shown in Fig.2. This uses a steam-recompression cycle for continuous drying in a tightly closed chamber. The wet material is dried with superheated steam at atmosphere pressure. Steam enters the drier at about 150°C and leaves having picked up more steam. This additional steam is split off, compressed to high temperature (about 250°C) and then condensed, giving up its latent heat to the main flow.

For materials of high inlet moisture content, such as paper pulp, no additional heat source is required. Here, cold wet material enters the drier and hot, water and dry material leaves the drier. The only energy required is for compressor and circulating fan. The latent heat of the water evaporated is kept within the cycle. This cycle gives overall COP of 3 which corresponds to about 4.5 kg of water per kWh with power density of 0.5 kW/m². If this new drier replaces a process with typical efficiency of 30-40%, primary fuel consumption could drop by a factor of at least 3. The running cost of steam recompression drier will be one-half to one-third that of conventional drier.

For the process to be economical, the product to be dried must have the right range of moisture content and be compact enough to be sealed lightly for the drying chamber.

Leaks from the cycle must be kept minimum as they are more harmful than air leaks in a hot air drier, because both sensible and latent heats are being lost and because the working of the cycle depends on a balance being achieved in enthalpies and mass flows.

Heat sources

(1) Gas Engine Drive Heat pump for Heating: (Water to water). The outdoor swimming pool at royal grammar school at Colehester is heated by an gas engine drive heat pump. This is the first practical application of the gas driven heat pump. This pool (360m²) was opened in 1924 in the memory of pupils serving for world War I and it was electrically heated just few years back with an addition of 135 kW electric heaters. The heat pump system was introduced for heating purposes in 1980 without disturbance to the existing system.

The heat pump is powered by a Ford engine which drives a Denco-Prestcold rotary sliding vane compressor of heat pump system, the fans to move the air through the plant over the evaporator, an alternator to charge the battery and engine cooling pump.

Nearly 7000 gallons of water per hour entered the heat pump from the pool at 24.3°C and (75.7°F) return to pool at 26.5°C (79.7°F). The heat input given to the pool is 2.8 therms per hour whereas to the engine is 1.86 therms per hour. This is actual 151% heat input to the pool when as percentage of energy delivered to the gas in the engine.

The result achieved can be compared with other fuels and heating systems as listed in

table. For example, with conventional gas fired boiler pool heating system, the overall efficiency of the system may hardly exceed 70% when combustion efficiency, plant standing losses and heat exchanger efficiency are considered. The gas driven heat pump has achieved 54% gas saving compared with gas fired boiler.

The gas engine driven heat pump has proved to be a practical, uses less primary energy units than any other conventional heating system and has every indication of being reliable. There is virtually no limit to the possibilities for using a gas driven heat pump although each needs individual arrangement and design to maximise the energy and financial savings.

The arrangement of the system using the heat from the water cooling the engine, engine exhaust and heat pump are shown in Fig. 1.

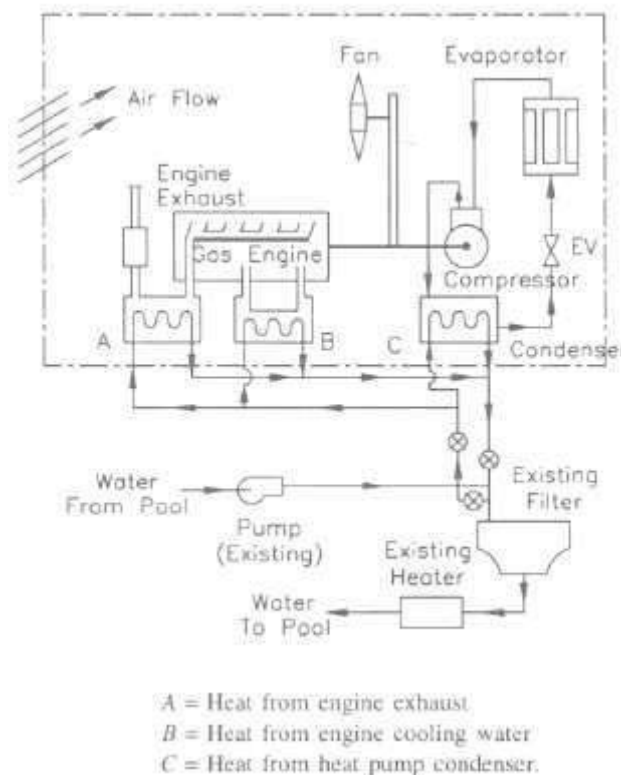


Fig: 1

Some of the largest installations using gas and diesel engines are in Germany. The first installation was used for swimming pool in Dostmund using atmospheric air as source of heat. The total heat supplied to the pool including the heat recovery from engine cooling system was 500 kW. Around 20 indoor and outdoor swimming pools now exist with gas

engine heat pumps.

Also in Dostmund, a building of 64 flats is being heated from a roof top gas engine driven heat pump-83 kW input and 38 kW output from the heat pump condenser and heat of 210 kW was recovered from the engine exhaust. The heat pump used atmospheric air as a source of heat. It runs of its own even at -12°C atmospheric temperature but below this, it is supplemented by a boiler of 186kW capacity. The heat pump saves 60% fuel at 6.8°C air temperature with $\text{COP} = 2.2$ and COP becomes 1.5 at -12°C air temperature. The capital investment is about $\$ 180 \times 10^3$

One of the largest of this type so far is the heat pump used at Panderboon sports centre with an output of 3.8 MW using ground water as a source of heat.

(2) Turbine Driven Heat Pump for Heating and Cooling (water to water). A heat pump normally extracts and upgrades free heat from ambient air or water but needs mechanical energy-usually from an electric motor which drives the compressor. But that energy is produced with a fuel-to-heat energy conversion efficiency of not more than 34%. Therefore, the usual heat pump is not much better than a conventional oil or gas fired furnace. The working of the system is outlined below.

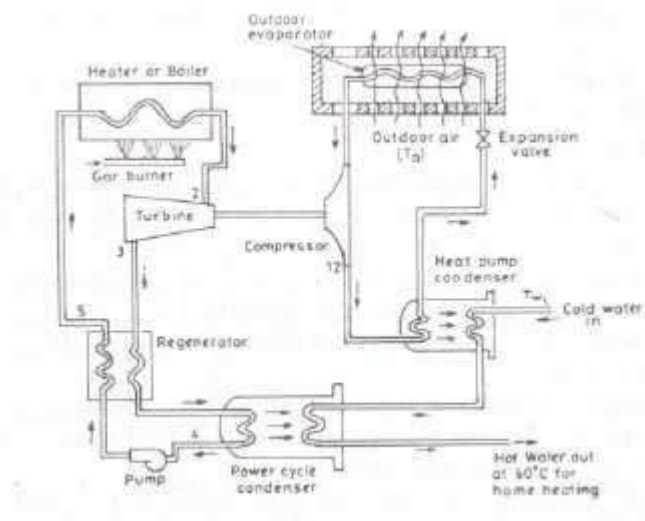


Fig: 2(a) Turbine-driven heat pump

Gas is turned to make a vapour which is used to spin a turbine which in turn runs the heat pump compressor. The fuel energy with external combustion is directly converted into mechanical power. So the effective use of primary energy for heating can be doubled.

Therefore, overall thermal efficiency is twice that of convectional boiler.

Freon-type fluid heated by gas burner expands passing through turbine driving the compressor. Most heat reaches the heat exchanger heating the domestic water circuit as it turns into fluid. Then the fluid is recirculated with the help of the pump back to the heater through a regenerator as shown in Fig: 2(a). The regenerator improves the overall performance of a cycle. This (heater turbine- regenerator condenser and pump) works on conventional Rankine power cycle.

The heat pump follows a traditional vapour compression refrigeration cycle. Low temperature liquid-vapour mixture coming out of expansion valve is warmed by the outdoor evaporator. The warmed vapour is further heated by compressing the vapour passing through a centrifugal compressor. The hot gas coming out condenses and is used as second input to the water circuit. The condensed high pressure liquid is passed through an expansion valve and its pressure and temperature are reduced. This system works on a conventional vapour compression refrigeration system

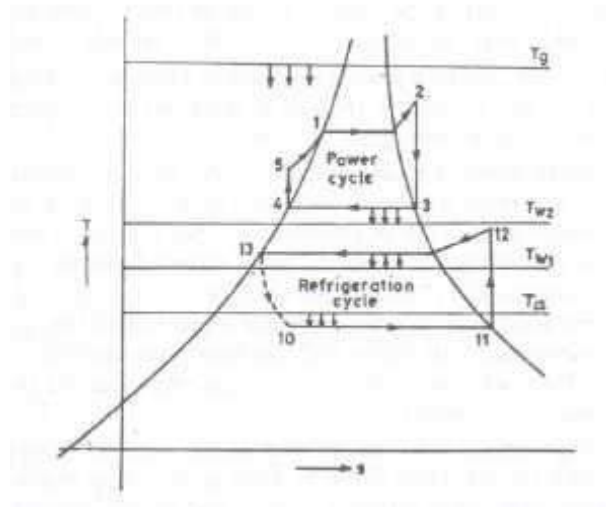


Fig: 2(b) Theoretical heat pump cycle driven by turbine rankine cycle

T_g = Gas Temperature

T_a = Atmospheric air temperature entering into the evaporator

T_{w1} = Water temperature entering into heat-pump condenser

T_{w2} = Water temperature entering into the power cycle condenser shows the direction of heat flow.

The complete cycle of operations (power as well as refrigeration) considering some working fluid is used in both cycles is shown in Fig. 2 (b).

Outstanding Features of the System

(1) Aside from its efficiency and energy conservation, the system is silent, virtually maintenance free and could use any fuel as gas, oil, coal, bio-mass and even solar radiation. No lubrication is needed because the turbine the only critical moving part, literally floats.

(2) The size of the system will be small and compact as a tiny turbine running at 150,000 rpm. will be used. This speed gives the best match between blade speed and nozzle velocity. Gas bearings will be used for the rotor since these are commonly used in dental drills running at 250,000 rpm. without lubrication and are figured to have a 30 year life.

(3) One single working fluid (mixture refrigerant) will be used in the Rankine and Heat-pump cycle for the optimization of the system. Small leakage between the separate fluid circuits with the turbo-compressor are then unimportant and simplifies shaft seals problem.

(4) Possible air-conditioning is another plus point in the favour of the system. For cooling purposes, the flow in the heat-pump cycle could be reversed so the evaporator and condenser swap functions. Shut off the power cycle condenser and you get cooling instead of heat.

(5) Water outlet temperature may be 60°C barely enough for effective space heating. The fan convectors will be most suitable units for a low temperature heating systems. The fan motor may require power of 30 watts only. They will also work well in the cooling mode of the system as forced and reduces the condensation problem with panel radiators.

Heat pump performance, using ambient air, falls off as outdoor temperature drops. To counterbalance this, an auxiliary gas heater for the water circuit is planned as back up in very cold weather conditions.

The initial cost of the system would be two to three times that of conventional boiler but it will be returned within only three years from the fuel savings if the fuel prices remain stable.

ML Strong, a senior project engineer at Glynued Central Research Laboratory in Solihull in England is working on the system for the last 4 years and it is expected by him that the proto type unit could be in the market within short span of time.

(3) Solar-assisted Heat-Pumps: (a) Air temp.-solar assisted heat pump is a practical way to use solar energy and economic way for using fossil fuels for heating purposes. The system uses solar collectors and air temperature heat pump in combination that is hard to beat for performance and energy conservation. It provides everything you need to make money from the sun.

The arrangement of the system is shown in Fig.3. Solar collectors utilize the sun's energy to heat water that is used as sole source of heat when outside temperature are between 8°C to 17°C. When the outdoor temperature falls below 8°C, the heat pump does the heating and when the temperature falls below the thermal balance point, solar heated water supplements heat pump. Solar heated water is also used to reduce domestic hot water bills.

During extreme cold, electric resistance heat provides the heating. Air temp-solar control package activates each system when it can be most proficient in providing warmth. And, of course, in the summer, the Air-temp. heat pump provides cooling and dehumidification. This is known as a parallel heat pump as it is not coupled with solar collector and only helps when collector output is less than required.

The parallel configuration offers only short term advantages. The heat pump provides good auxiliary energy source for an underside conventional solar system. Undersizing and operating efficiency of the heat pump create a cost effective combination utilizing available equipment. The performance requirements imposed on heat pump utilized in the parallel configuration are identical to those imposed by the conventional heat pump operation. As a result, heat pumps are ideal for this application and are readily available from industry which is strongly motivated to develop the standby unit. The economics for this configuration should disappear as the conventional solar configuration becomes cost effective.

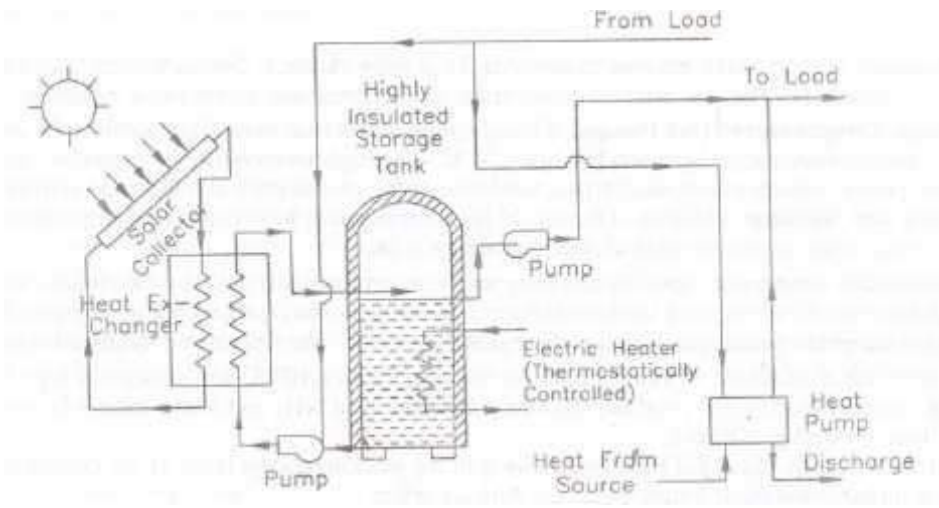


Fig: 3 Parallel operation of heat pump with solar system

(b) Another system is known as series operated solar heat pump. In this arrangement the pump is used to move solar energy from the collector or storage to the load. Thus, the heat pump is in series with the flow of energy as shown in Fig.4.

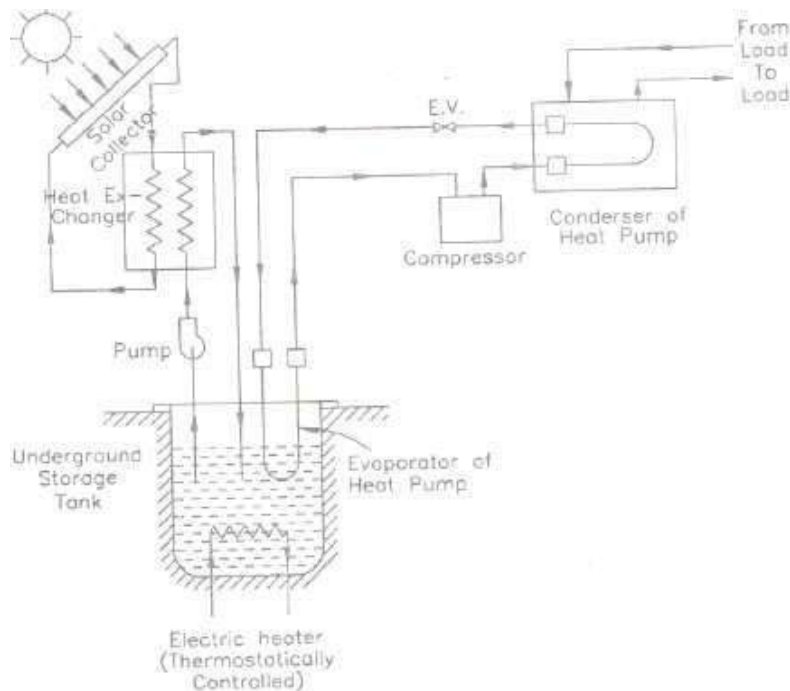


Fig: 4 Series operation of heat pump with solar system

The series solar assisted heat pump configuration has a much greater and far reaching potential. This configuration uses the heat pump as an energy coupling. This relieves the solar collector and storage element of this energy distribution requirement and opens the door for major modification and cost reduction of these elements. Energy collection and storage temperatures are reduced from a high temperature region (60 - 100°C) to a moderate temperature region.

Uninsulated ground tanks become acceptable as storage element. Single glazed structures are also acceptable as collectors. The potential for innovation is unlimited and is yet to be explored.

(4) High Temperature Heat Pumps: Closed compression heat pumps are about to be employed for exploitation temperature range between 100 and 130°C. The high temperature heat pumps can be used in chemical and petrochemical plants, distilleries, food processing plants, posteurization and sterilization plants and blenching and washing industries. The use of high temperature heat pumps for heat recovery already makes economic sense in some cases at present energy costs.

The attainable temperature level by an heat pump is determined primarily by the choice of the working fluid though the type of compression has little influence in compression type heat pump. The possible working media for high temperature compression type heat pump are mainly the fluorinated hydrocarbon refrigerants.

Still there are no suitable working fluids for the range above 100°C for absorption type heat pumps. Because the thermodynamically good and proven combination of NH₃ and water cannot be considered on account of toxicity and pressure.

For all heat pumps, the upper temperature limit of the working media is set by the critical temperature, above which liquefaction is no longer possible. Already when approaching the critical point, conditions for heat pump operation deteriorate so much (owing to the rapidly declining evaporation heat) that an interval of 10 to 15°C (depending on working medium) from the critical temperature ought to be maintained. If this is not maintained, a

series drop in the COP takes place as shown in Fig.5. The threshold set by the structural strength of compressors is still lower for compression type heat pumps (30 bar).

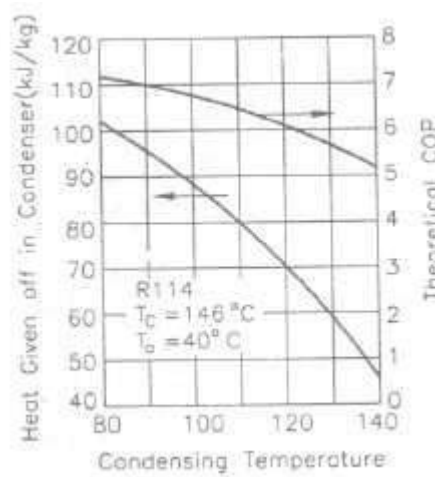


Fig: 5 COP of R₁₁₄ approaching critical temperature

The thermal stability also sets a limit. This is governed by a number of factors as chemical composition of the working fluid, lubricant to be used and moisture. The thermal stability limits for fluorinated hydrocarbons in high temperature range are shown in Fig.6. It is obvious from the figure that there are only two media with the highest critical temperature R₁₁₄ and R₁₁₃ that are mostly restricted by thermal stability, therefore their use ranks below R₁₁₄. The latter reaches the highest value on all three criteria together (critical temperature, stability and pressure). Finally, in the case of lubricated compressors, it is the lubricant that limits the duty range. Hence the oil free designs are the best of the possible compressor types.

Apart from the temperature limits, the volumetric thermal performance (heat gain per unit volume of intake vapour) is also significant for compression heat pumps working media. In this respect, R₁₁₄ is far superior to R₁₁ and R₁₁₃. Its other principal properties—lowest toxicity, solubility in oil and no vacuum when circulation ceases also meet the practical requirements very well. Hence R₁₁₄ is the only most preferred medium for compression heat pumps in the medium temperature range. The application limit should be restricted to 130°C when the critical temperature is approached.

Fluorinol (trifluorothanol) holds promise for higher temperatures ranging up to 160°C. It has been used only in isolated instances and only in expansion turbines. Its volumetric thermal efficiency is only 10% of R₁₁₄.

All three types of compressors may be used though at very different costs. As far as compressors are concerned, the lower performance range is still dominated by the reciprocating compressors, while turbocompressors are preferred for high performance. Between the two, lies the field of the screw compressor.

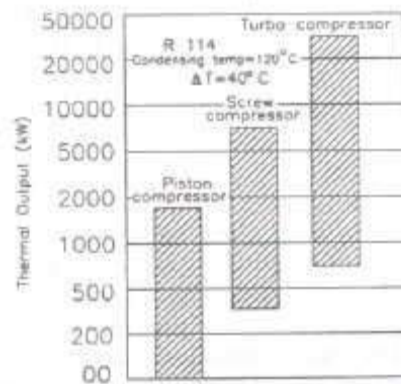


Fig: 6 Performance range of compressors for closed type high temperature compression heat pumps.

INTRODUCTION OF HEAT PUMP

Heating is a major part of the energy consumption of the colder countries of the world. Because of this and shortage of fossil fuels, efforts are being made to find the alternative energy sources. A situation has been reached where the heat pump is suggested as an answer to the reduction of energy consumption by using it in energy recovery systems and by applying it to upgrade lower grade energy in the surrounding environment. Basically a heat pump is a refrigeration unit capable of extracting heat from any source of low grade heat such as atmosphere, sea, ground and upgrading the heat to a useful temperature. Lord Kelvin suggested the use of heat pump for heating purposes in 1852 but due to the lack of efficient components of refrigeration unit, this was not used for actual heating purposes. The heat pump was actually used during the second world war very successfully by the Switzerland engineers as the import of coal from outside Countries was stopped. Nowadays the heat pump is universally used by number of countries not only for heating the factories and homes but for many industrial purposes.

In the commonest applications heat is taken from a body of water or ambient air and elevated to a temperature sufficient to heat dwellings. In this way, energy from practically inexhaustible sources may be exploited almost without cost instead of increasingly scarce primary fuel energy. The heat pump can play a further part in the recovery of rejected heat from industrial processes, by putting it to renewed use instead of discharging it to the environment.

If the heat pump merely raised heat from a lower to a higher temperature level, its action might be compared with that of pumped storage scheme, in which energy is pumped from a lower to a higher altitude. With storage pumps, the energy gain is equal to the energy input but the heat pump is able to recover several times its energy input. This difference is due to the fact that while the working medium of the pumped storage scheme (water) remains in the same state throughout the cycle but in the heat pump system, the medium changes from liquid to vapour and back during the cycle. Thus, in the Process, much more energy is exchanged than is needed to drive the heat pump.

HEAT PUMP CIRCUIT

The heat pump is a primarily refrigerant cycle using heat sources as atmosphere, sea and ground.

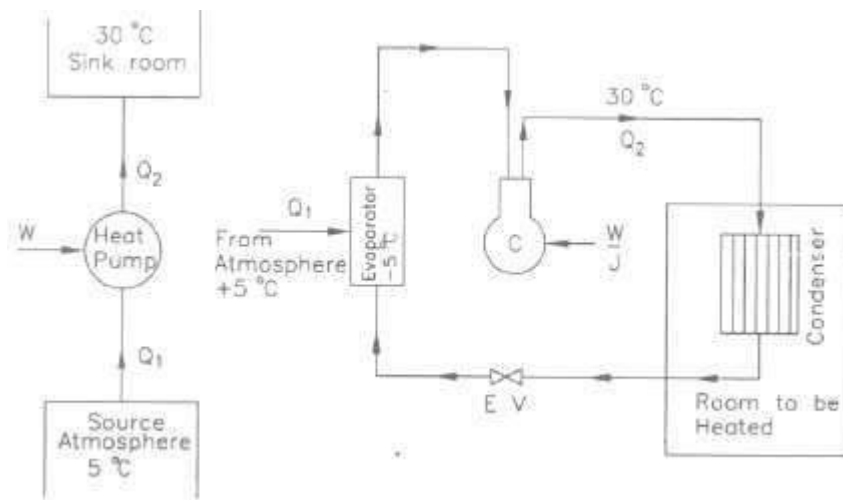


Fig.1

This absorbed heat from the unusable sources is upgraded with the help of compressor and the fluid carrying the heat dissipates at required places through the condenser of the refrigeration system. The heating system using the refrigeration unit is shown in

Fig.1. The refrigerant coming out of expansion valve absorbs heat Q_1 from atmosphere as its temperature (-5°C) is lower than the atmospheric temperature ($+5^\circ\text{C}$). This absorbed heat by refrigerant is upgraded by adding the work through compressor and the total heat $Q_2[Q_2 = Q_1 + (W/J)]$ is dissipated in the room through condenser as shown in the figure. The refrigerant cycle is used for heating purposes only when the atmospheric temperature is below the required temperature for comfort which is known as "Heat Pump" system.

The performance of refrigeration cycle when used for heating purposes is given by a ratio of heat delivered to the work supplied. This ratio is known as "Effective Performance Ratio" (EPR) or "Heat Pump Performance Ratio" and it is given by

DIFFERENT HEAT PUMP CIRCUITS

The basic heat pump circuits which are commonly used are of five types.

(1) Air-to-Air: In this system atmospheric air is used as a source of heat and air is also used for absorbing heat from the condenser. The air-to-air system is further divided into two parts as fixed air circuit heat-pump and fixed refrigerant circuit heat-pump.

Fixed Air-Circuit Heat Pump: In this arrangement, the flow of air is not changed to achieve the change of heating effect from cooling effect or vice versa. The arrangement of refrigeration cycle for heating and cooling purposes is shown in Fig.2 (a). During the heating cycle, the heat is taken from the atmospheric air and it is pumped to the room air and during cooling cycle, the heat taken from the room air and it is pumped into the atmosphere.

During heating cycle, the valves 5, 6, 7 and 8 remain open and the valves 1, 2, 3 and 4 remain close. During cooling cycle, the valves 1, 2, 3 and 4 remain open and valves 5, 6, 7 and 8 remain closed.

Fixed Refrigerant Circuit: In this arrangement, the refrigerator circuit is maintained constant and the air flow direction is changed as per the requirement of heating or cooling in the conditioned space. This air-to-air design eliminates reversing of valves and flow direction of refrigerant therefore allows a simple refrigerant piping circuit. The air circuits are more complicated therefore dampers must be designed air tight and free acting. .

During the heating cycle, the dampers E, F, G, H remain open and A, B, C and D remain

closed. Air from conditioned space passes through dampers E over the condenser and through dampers F to the conditioned space. Atmospheric air passes through dampers G over the evaporator and through damper H back to the atmosphere.

During cooling cycles, dampers A, B, C and D remain open and E, F, G and H remain close. Air from conditioned space passes through dampers A over the evaporator and through damper B back to the conditioned space. Outside air passes through damper

Cover the condenser and through damper D back to the atmosphere. The arrangement of the system is shown in Fig. 2 (b).

(2) Water to Air Design:

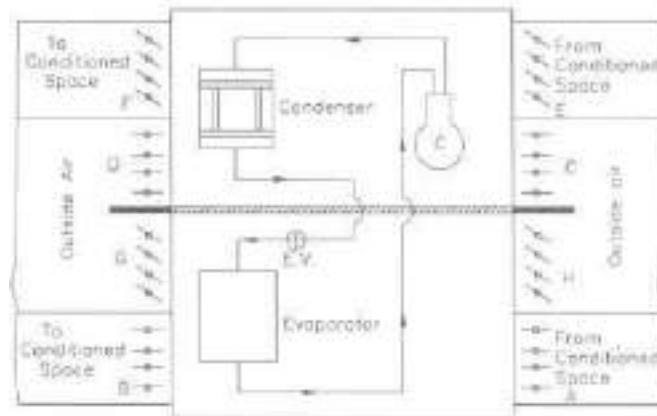


Fig: 2 (b)

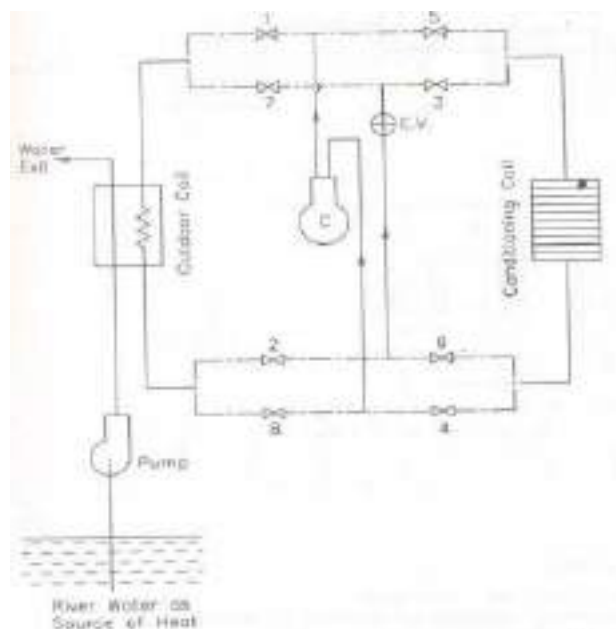


Fig: 2 (a) Water to air design

This type of design is divided into two parts as water to air and air to water design. In water to air design, the atmospheric water is used as a source of heat and air is used to carry out this heat directly into air-conditioned space.

Water to Air Design: In case of water to air design, the valves 1, 2, 3, 4 remain open and valves 5, 6, 7, 8 remain closed during cooling cycle. The valves 5, 6, 7, 8 remain open and valves 1, 2, 3 and 4 remain closed during heating cycle. The arrangement of the system is shown in Fig. 3 (a).

(3) Air to Water Design: In air to water design, outdoor air is used. as the source of heat and water is used as the transferring medium between the air going to the conditioned space and refrigerant coming to water coil. The arrangement of the system is shown in Fig.3 (b). The valve arrangement is just similar to the valve arrangement shown in Fig.3 (a). With this arrangement, it is possible to store the hot water during mild weather conditions which can be used when lower outdoor temperatures are experienced. It is possible to store either hot or cold water depending on the cycle of operation (cooling or heating) during off peak periods and to use when the demand is greatest.

During both heating and cooling cycle, the pump circulates water in a closed loop through the water coil as shown in Fig.3 (b). The valve arrangement for the circulation of refrigerant according to the required heating or cooling cycle is just similar to the valve arrangement shown in Fig. 3 (a).

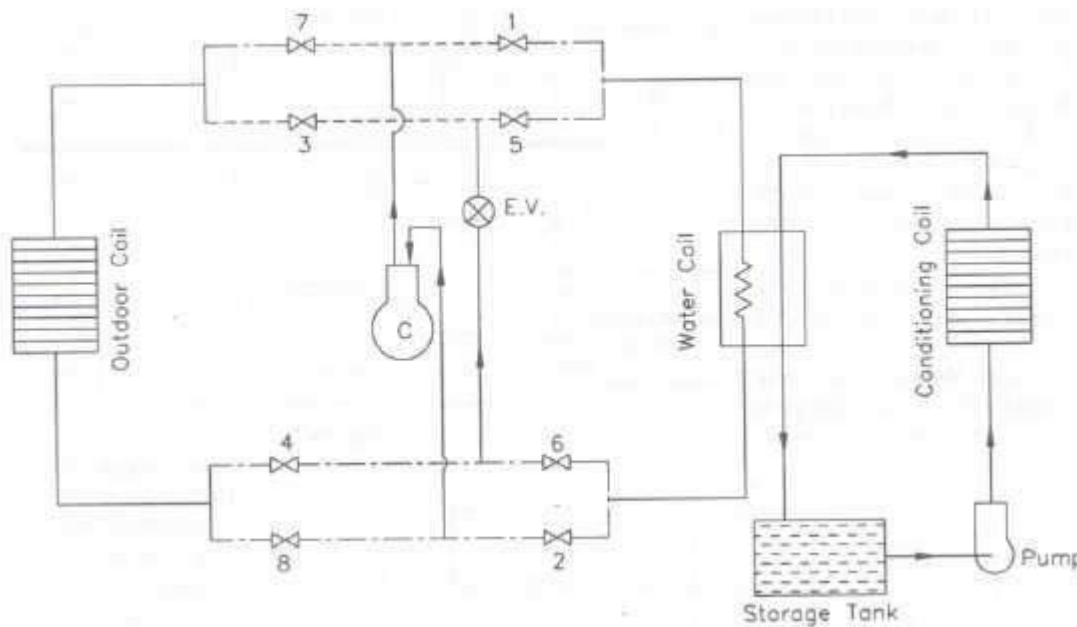


Fig: 3 (b) Air to water design

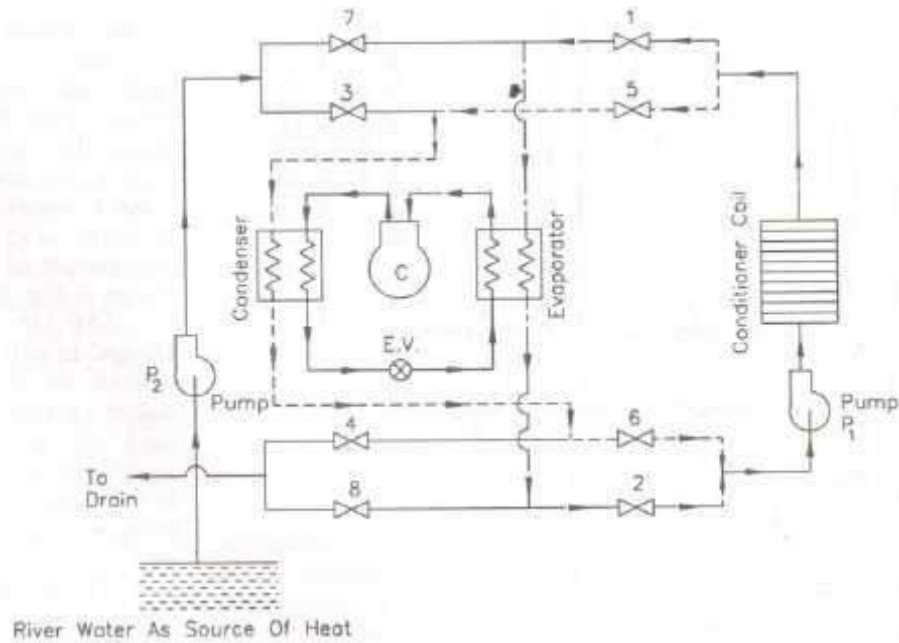


Fig: 4 Water to water design

.....Circulation of water through conditioner coil when heating is required. Circulation of

→ water through conditioner coil when cooling is required.

Water to Water Design: In this arrangement water is used as a source of heat and it is also used as a medium of air-conditioning fluid. The arrangement of the system is shown in Fig. 4. The arrangement of the system is shown in Fig.4. During the heating cycle, valves 5, 6, 7, 8 remain open and valves 1, 2, 3, 4 remain closed. The pump P1 circulates the water through the conditioner coil where it gives up the heat to the air which is passed to the air-conditioned space. The water coming out of the conditioner coil passes through the valve 5 again to the condenser where it absorbs the heat from the refrigerant gas and further it passes through the valve 6 again to the conditioner coil. The pump P2 circulates the water through the valve 7 through the evaporator where heat is absorbed by the refrigerant from the water and the cooled water from evaporator is drained through the valve '8'.

During the cooling cycle, valves 1, 2, 3, 4 remain open and 5, 6, 7, 8 remain close. The path will be just reverse in direction as described above.

(5) Air to Liquid Design. In this type of design, air is used as a source of heat and liquid is used to give this heat to the air which is supplied to the air-conditioned space. The arrangement of the system is shown in Fig. 5 (a).

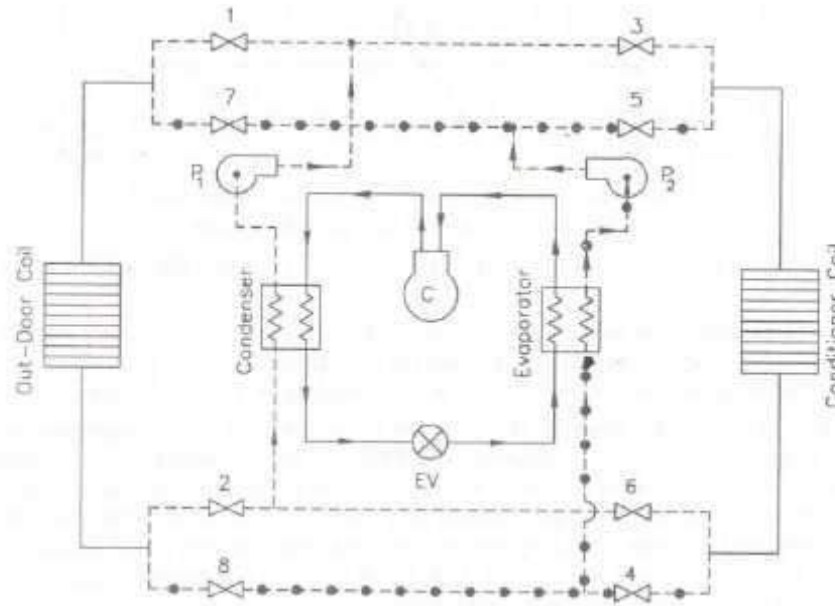


Fig: 5 (a) Air to Liquid design

During the heating cycle, valves 5, 6, 7, 8 remain open and valves 1, 2, 3, 4 remain closed. The pump 'P₁' circulates the hot liquid from the condenser through the valve '5' to the conditioner coil where the heat is given to the air which is supplied to the air- conditioned space. The liquid from the conditioner coil passes through the valve '6' back to the condenser to repeat the cycle. The pump P₂ circulates the cold liquid in the evaporator and through the valve '7' it is passed into the outdoor coil where it absorbs the heat from atmospheric outside air. The liquid coming out from the outdoor coil goes to the evaporator through the valve '8' to repeat the cycle.

During the cooling cycle, valves 1, 2, 3, 4 remain open and the valves 5, 6, 7, 8 remain closed. The air to liquid design can be simultaneously used for heating as well as for cooling purposes. The arrangement of the system is shown in Fig.5 (b).

An auxiliary coil is introduced in the circuit which is shown in Fig.5 (b). When the system is on, the heating cycle, the auxiliary coil provides the cooling when the conditioner coil provides the heating.

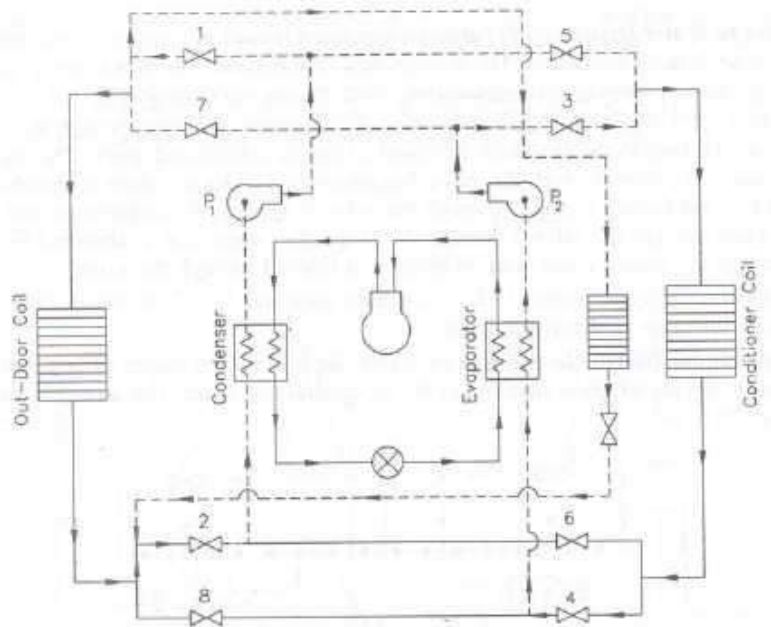


Fig: 5 (b) Air to Liquid Design with auxiliary coil.

When the system is on cooling cycle, the auxiliary coil provides the heating when the conditioner coil provides cooling.

This type of arrangement is useful where cooling and heating are simultaneously required. This is mostly useful in theatres where the cooling of the auditorium is required even on the coldest day because of the heat gain from the occupants where the offices require heating in the cold nights.

Comparison of different Circuits: The air-to-air and water-to-air circuit arrangements are most simple in construction and most economical in running. The EPR for these arrangements are higher than any other arrangement of the system. The temperature gradient necessary between refrigerant and air for effective heat transfer are higher than other circuit arrangements as no intermediate heat transfer surfaces are employed.

In both above mentioned systems, the coils used act as condenser and evaporator so that careful design and construction are necessary. A well constructed leak-proof and reliable design of the valves is necessary as the valves are used for reversing the refrigerant flow.

In water to water and air to liquid designs, there is a fixed refrigerant circuit and refrigerant flows in one direction only. This arrangement reduces the leakage difficulties and it also reduces the difficulties with oil return to the compressor. Simplicity in equipments design is achieved in these two cases because the condenser and evaporator are not required to serve the dual purpose.