COURSE MATERIAL

IV Year B. Tech I- Semester

MECHANICAL ENGINEERING

HEATING VENTILATION AND AIR CONDITIONING

R17A0331

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

DEPARTMENT OF MECHANICAL ENGINEERING

(Autonomous Institution-UGC, Govt. of India)
Secunderabad-500100, Telangana State, India.
www.mrcet.ac.in

CONTENTS

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(Autonomous Institution – UGC, Govt. of India)

VISION

- To establish a pedestal for the integral innovation, team spirit, originality and competence in the students, expose them to face the global challenges and become technology leaders of Indian vision of modern society.

MISSION

- To become a model institution in the fields of Engineering, Technology and Management.
- To impart holistic education to the students to render them as industry ready engineers.
- To ensure synchronization of MRCET ideologies with challenging demands of International Pioneering Organizations.

QUALITY POLICY

- To implement best practices in Teaching and Learning process for both UG and PG courses meticulously.
- To provide state of art infrastructure and expertise to impart quality education.
- To groom the students to become intellectually creative and professionally competitive.
- To channelize the activities and tune them in heights of commitment and sincerity, the requisites to claim the never - ending ladder of **SUCCESS** year after year.

For more information: www.mrcet.ac.in

VISION

To become an innovative knowledge center in mechanical engineering through state-of-the-art teaching-learning and research practices, promoting creative thinking professionals.

MISSION

The Department of Mechanical Engineering is dedicated for transforming the students into highly competent Mechanical engineers to meet the needs of the industry, in a changing and challenging technical environment, by strongly focusing in the fundamentals of engineering sciences for achieving excellent results in their professional pursuits.

Quality Policy

- ✓ To pursuit global Standards of excellence in all our endeavors namely teaching, research and continuing education and to remain accountable in our core and support functions, through processes of self-evaluation and continuous improvement.
- ✓ To create a midst of excellence for imparting state of art education, industry-oriented training research in the field of technical education.

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

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Department of Mechanical Engineering

PROGRAM OUTCOMES

Engineering Graduates will be able to:

1. **Engineering knowledge:** Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems.
2. **Problem analysis**: Identify, formulate, review research literature, and analyze complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.
3. **Design/development of solutions**: Design solutions for complex engineering problems and design system components or processes that meet the specified needs with appropriate consideration for the public health and safety, and the cultural, societal, and environmental considerations.
4. **Conduct investigations of complex problems**: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.
5. **Modern tool usage**: Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modeling to complex engineering activities with an understanding of the limitations.
6. **The engineer and society**: Apply reasoning informed by the contextual knowledge to assess societal, health, safety, legal and cultural issues and the consequent responsibilities relevant to the professional engineering practice.
7. **Environment and sustainability**: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.
8. **Ethics**: Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.
9. **Individual and teamwork**: Function effectively as an individual, and as a member or leader in diverse teams, and in multidisciplinary settings.
10. **Communication**: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation, make effective presentations, and give and receive clear instructions.
11. **Project management and finance**: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one’s own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.

12. **Life-long learning**: Recognize the need for and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

PROGRAM SPECIFIC OUTCOMES (PSOs)

PSO1 Ability to analyze, design and develop Mechanical systems to solve the Engineering problems by integrating thermal, design and manufacturing Domains.

PSO2 Ability to succeed in competitive examinations or to pursue higher studies or research.

PSO3 Ability to apply the learned Mechanical Engineering knowledge for the Development of society and self.

Program Educational Objectives (PEOs)

The Program Educational Objectives of the program offered by the department are broadly listed below:

PEO1: PREPARATION

To provide sound foundation in mathematical, scientific and engineering fundamentals necessary to analyze, formulate and solve engineering problems.

PEO2: CORE COMPETANCE

To provide thorough knowledge in Mechanical Engineering subjects including theoretical knowledge and practical training for preparing physical models pertaining to Thermodynamics, Hydraulics, Heat and Mass Transfer, Dynamics of Machinery, Jet Propulsion, Automobile Engineering, Element Analysis, Production Technology, Mechatronics etc.

PEO3: INVENTION, INNOVATION AND CREATIVITY

To make the students to design, experiment, analyze, interpret in the core field with the help of other inter disciplinary concepts wherever applicable.

PEO4: CAREER DEVELOPMENT

To inculcate the habit of lifelong learning for career development through successful completion of advanced degrees, professional development courses, industrial training etc.

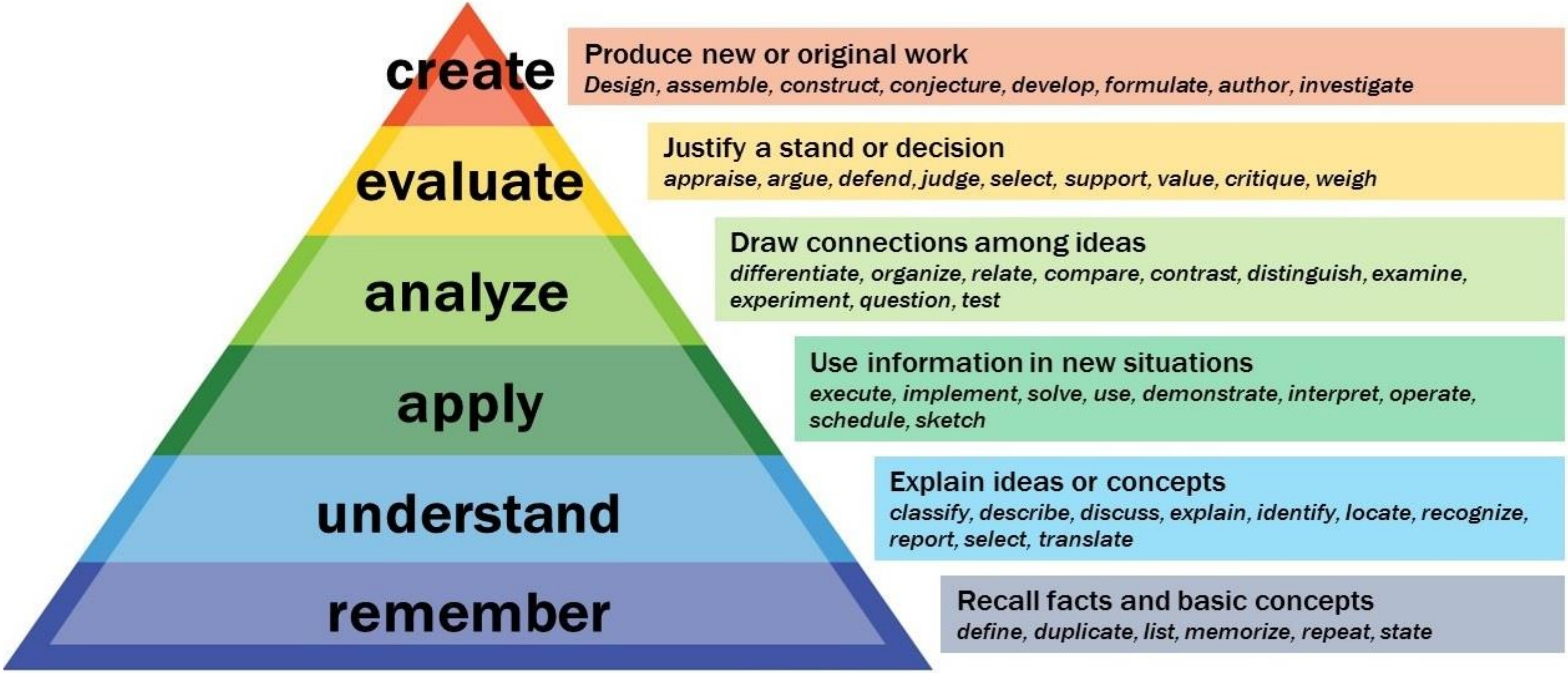
PEO5: PROFESSIONALISM

To impart technical knowledge, ethical values for professional development of the student to solve complex problems and to work in multi-disciplinary ambience, whose solutions lead to significant societal benefits.

Blooms Taxonomy

Bloom's Taxonomy is a classification of the different objectives and skills that educators set for their students (learning objectives). The terminology has been updated to include the following six levels of learning. These 6 levels can be used to structure the learning objectives, lessons, and assessments of a course.

1. **Remembering**: Retrieving, recognizing, and recalling relevant knowledge from long- term memory.
2. **Understanding**: Constructing meaning from oral, written, and graphic messages through interpreting, exemplifying, classifying, summarizing, inferring, comparing, and explaining.
3. **Applying**: Carrying out or using a procedure for executing or implementing.
4. **Analyzing**: Breaking material into constituent parts, determining how the parts relate to one another and to an overall structure or purpose through differentiating, organizing, and attributing.
5. **Evaluating**: Making judgments based on criteria and standard through checking and critiquing.
6. **Creating**: Putting elements together to form a coherent or functional whole; reorganizing elements into a new pattern or structure through generating, planning, or producing.



create
Produce new or original work
Design, assemble, construct, conjecture, develop, formulate, author, investigate
evaluate
Justify a stand or decision
appraise, argue, defend, judge, select, support, value, critique, weigh
analyze
Draw connections among ideas
differentiate, organize, relate, compare, contrast, distinguish, examine, experiment, question, test
apply
Use information in new situations
execute, implement, solve, use, demonstrate, interpret, operate, schedule, sketch
understand
Explain ideas or concepts
classify, describe, discuss, explain, identify, locate, recognize, report, select, translate
remember
Recall facts and basic concepts
define, duplicate, list, memorize, repeat, state

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

IV Year B. Tech, ME-I Sem

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(R17A0331) HEATING VENTILATION AND AIR CONDITIONING
(CORE ELECTIVE -III)

Course Objectives:

- Thecourseaimstoemphasizetheimportanceofheatingandventilationsystems.
- This program includes heating, ventilation and airconditioning.
- Graduateswillpossesstheskillsnecessarytoobtainanentry-levelHVACTechnician position.
- GraduateswillhaveanunderstandingofsafeHVACpracticesandhowimportantthey are in the HVACenvironment.
- Graduates will understand the importance of professional behavior and life-long learning,andwillmeetthechallengesofcontinuedtechnologicalgrowthwithinthefield.

UNIT I

INTRODUCTION TO HVAC: Fundamentals-Modes of Heat Transfer-Sensible Heat and Latent Heat-**Basic Components of Air-Conditioning and Refrigeration machines**-Basic Refrigeration SystemorVaporCompressionCycle-Pressure –EnthalpyChart-Function&TypesofCompressor-Function & Types of Condenser-Function & Types of Expansion Valves, Function & Types of Evaporator-Accessories used in the System-Refrigerant andBrines

UNIT II

CLASSIFICATION OF AIR-CONDITIONING SYSTEM: Window A/C-Working of Window A/C with Line Diagrams-**Split A/C**-Types - Working of Split A/C with Line Diagrams-**Ductable Split A/C**-Working of Ductable Split A/C with Line Diagrams-Variable Refrigerant Volume (VRV)/ Variable Refrigerant Flow (VRF)-**Ductable Package A/C**-Working of Ductable Package A/C with Line Diagrams

UNIT III

STUDY OF PSYCHROMETRIC CHARTS: Dry Bulb Temperature-Wet Bulb Temperature-Dew Point Temperature-Relative Humidity-Humidity Ratio-Processes, Heating, Cooling, Cooling and Dehumidification, Heating and Humidification

UNIT IV

LOADCALCULATION: SurveyofBuilding-CoolingLoadSteps-FindingTemperaturedifference(ΔT)-Wall, Glass, Roof, partition-Finding ‘U’ Factor-Wall, Glass, Roof, Partition-Finding Ventilation requirement for IAQ-Load Calculations (Manually using E-20 form)- ESHF, ADP & Air Flow Rate (CFM)Calculation

UNIT V

STATIC PRESSURE CALCULATION: Selection of Motor HP-Selection Fan/Blower RPM-**Hydronic System-**ClassificationofWaterPiping-Pipesizingforchillwatersystem-FittingsusedintheHVAC Piping System-Valves used in the HVAC Piping System-Function of Valves-Openings for CHW Pipes passing through Wall-Sectional drawing @ CHW Pipe supports-Pump Head Calculation-Selection of Pump

REFERENCES:

1. HVAC Fundamentals Volume-l / James E. Brumbou / Audel / 4Edition
2. Fundamentals of HVAC Systems / Robert Mcdowall / Academic Press /2007
3. Home Heating & Air Conditioning systems / James Kittle /MGH
4. HVAC Fundamentals / Samuel C. Sugarman / Fairmont Press /2005.
5. R&AC Hand Book byISHRAE
6. Ventilation Systems: Design and Performance/ Hazim B. Awbi. / Routledge /2007.
7. Portable Ventilation Systems Hand Book / Neil McManus / CRC Press /2000.
8. Design of Industrial Ventilation Systems / John L Alden / Industrial Press / 5Edition.
9. Industrial Ventilation Applications / ISHRAE Hand Book /2009.
10. HVAC Hand book /ISHRAE.

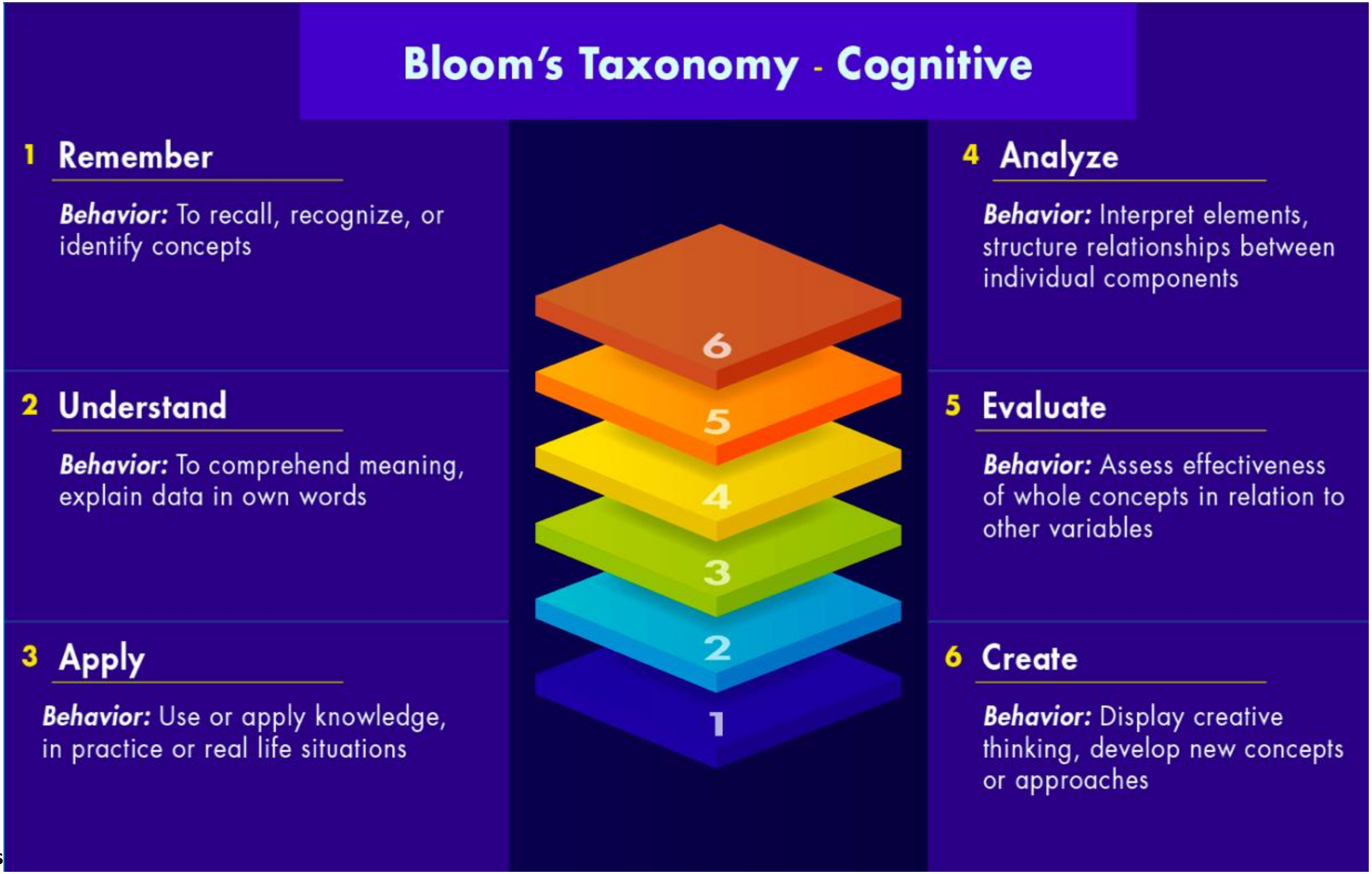
Course Outcomes:

- Students will assist in the installations of Heating, Air Conditioning and Refrigeration equipment.
- Perform preventive maintenance on heating and air conditioningsystems.
- Students will identify sitehazards.
- The student shall understand the principles and working HVACsystems.
- To be able to study and analyze psychrometric chart in refrigeration systems. Develop problem solving skills through the application ofthermodynamics.

HEATING VENTILATION AND AIR CONDITIONING (R17A0331)

COURSE OBJECTIVES

UNIT - 1	CO1: To Explain in detail Basic Components of Air-Conditioning & Refrigeration machines, Basic Refrigeration cycle, Accessories & Refrigerants
UNIT - 2	CO2: To Know Detail classification of Air-Conditioning System
UNIT - 3	CO3: To Study Psychrometric Chart and various terminology
UNIT - 4	CO4:To Explain the Load calculations of Survey of Building, Ventilation requirement for IAQ, ESHF, ADP & Air Flow Rate(CFM)Calculation
UNIT - 5	CO5: Explain Hydronic System, Water Piping,Fittings used in the HVAC Piping System Function, CHW Pipe supports &Pump Head Calculation


Bloom's Taxonomy - Cognitive
1 Remember
Behavior: To recall, recognize, or identify concepts
2 Understand
Behavior: To comprehend meaning, explain data in own words
3 Apply
Behavior: Use or apply knowledge, in practice or real life situations
4 Analyze
Behavior: Interpret elements, structure relationships between individual components
5 Evaluate
Behavior: Assess effectiveness of whole concepts in relation to other variables
6 Create
Behavior: Display creative thinking, develop new concepts or approaches
6
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<u>COURSE OUTLINE</u>

UNIT – 1

NO OF LECTURE HOURS: 15

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	INTRODUCTION TO HVAC: Fundamentals-Modes of Heat Transfer-Sensible Heat and Latent Heat	Conduction, Convection& radiation, latent heat & Sensible Heat	Understand the basics of modes of Heat Transfer (B2). To remember the Sensible & latent Heat (B1)
2.	Basic Components of Air-Conditioning and Refrigeration machines.	Compressors, Condensers, Evaporators & Expansion Values	Understand the different components of the Air-Conditioning and Refrigeration machines(B2)
3.	Basic refrigeration System or Vapor Compression Cycle Pressure –Enthalpy Chart-Function	VCR CYCLE	Understand the working of cycles (B2) To analyze the cycle in enthalpy chart (B4)
4.	Types of Compressor-Function	WORKING OF COMPRESSORS	To understand Function of compressor (B2) To Apply the compressors in hvac (B3)
5.	Types of Condenser-Function	WORKING OF CONDENSORS	To understand Function of condensors (B2) To Apply the compressors in hvac (B3)

6.	Types of Expansion Valves, Functions	WORKING OF EXPANSION VALVES	To understand Function of Expansion Values (B2) To Apply the compressors in hvac (B3)
7.	Types of Evaporator functions	WORKING OF EVAPORATOR	To understand Function of Evaporators (B2) To Apply the compressors in hvac (B3)
8.	Accessories used in the System-Refrigerant and Brines	REFRIGERANT TYPES	To able to understand which type of refrigerant is used in HVAC (B4)

UNIT – 2

NO OF LECTURE HOURS: 10

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	CLASSIFICATION OF AIR-CONDITIONING SYSTEM	Classifications	To understand Classification of Air Conditioning System(B2)
2.	Window A/C-Working of Window A/C with Line Diagrams	Working of A/C Systems	To understand of Window A/C Systems (B2) To analyze the Each Component (B4)
3.	Split A/C-Types - Working of Split A/C with Line Diagrams-	Working of Split A/C Systems	To understand of Split A/C Systems (B2) To analyze the Each Component (B4)
4.	Ductable Split A/C- Working of Ductable Split A/C with Line Diagrams	Working of Ductable A/C	To understand of Ductable A/C Systems (B2) To analyze the Each Component (B4)
5.	Variable Refrigerant Volume (VRV)/ Variable Refrigerant Flow (VRF)-	Working of (VRV)/VRF	To understand of VRV /VRF (B2) To analyze the Each Component (B4)

UNIT – 3

NO OF LECTURE HOURS: 8

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	STUDY OF PSYCHROMETRIC CHARTS	To learn what is PSYCHROMETRIC CHARTS	To understand & apply the PSYCHROMETRIC CHARTS (B2),(B3).
2.	Dry Bulb Temperature-Wet Bulb Temperature-	DBT & WBT Definations	
3.	Dew Point Temperature-Relative Humidity-Humidity Ratio-Processes,	DPT	
4.	Heating, Cooling, Cooling and Dehumidification, Heating and Humidification	TO MARK IN PSYCHROMETRIC CHARTS CURVES	To Able to Analyze in HVAC systems.(B4)

UNIT – 4

NO OF LECTURE HOURS:12

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	LOAD CALCULATION: Survey of Building-Cooling Load Steps	Survey of Building	To Understand the load Calculations of Building (B2). To apply the load calculations in hvac systems (B3). To Analyze the Requirements of IAQ (B4).
2.	Finding Temperature difference(ΔT)- Wall, Glass, Roof, Partition	Various element of building	
3.	Finding 'U' Factor-Wall, Glass, Roof, Partition-Finding Ventilation	To calculate u factor	
4.	Requirement for IAQ-Load Calculations (Manually using E-20 form)- ESHF, ADP	IAR	
5.	Load Calculations (Manually using E-20 form)- ESHF, ADP	ESHF	
6.	Air Flow Rate (CFM)Calculation	CFM CALCULATION	

UNIT – 5

NO OF LECTURE HOURS: 12

LECTUR E	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Selection of Motor HP-Selection Fan/Blower RPM	VARIOUS FAN/BLOWER INHVAC	To able to select fan/Blower Based on RPM(B3)
2.	Hydronic System		To understand the Hydrsonic System(B2).
3.	Classification of Water Piping-Pipe sizing for chill water system-	WATER PIPING SYSTEM	To Understand the Classification of Piping(B2) To able to select the pipe in chill water System(B3).
4.	Fittings used in the HVAC Piping System	FITTINGS	To Understand the fittings in HVAC (B2)
5.	Valves used in the HVAC Piping System-Function of Valves	VALUES & FUNCTIONS	To Understand the VALUES in HVAC (B2)
6.	Openings for CHW Pipes passing through Wall-Sectional drawing @ CHW Pipe supports-Pump Head Calculation-Selection of Pump	CHW PIPES	To Understand the CHW (B2) Analyze the pump calculations(B4)

Mapping of COs and POs:

Course Outcomes	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3
CO1	X	X	X	X	X	X	X	X	X	-	X	X	X	X	X
CO2	X	X	X	X	X	X	X	X	X	-	X	X	X	X	X
CO3	X	X	X	X	X	X	X	X	X	-	X	X	X	X	X
CO4	X	X	X	X	X	X	X	X	X	-	X	X	X	X	X
CO5	X	X	X	X	X	X	X	X	X	-	X	X	X	X	X

Course Outcomes	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3
CO1	2	3	3	3	3	**2**	2	1	2	-	1	2	2	2	2
CO2	3	2	2	2	2	2	1	1	1	-	2	2	3	2	2
CO3	3	2	3	2	2	2	2	1	2	-	2	2	2	2	2
CO4	2	2	2	2	2	2	1	2	1	-	2	2	3	2	2
CO5	3	2	2	2	2	2	1	2	2	-	2	3	2	3	2

Mode of Evaluation:

- 75% of marks for External Evaluation.
- 20% of marks for Internal Evaluation.
- 5% of marks for Continuous Evaluation assignments.

UNIT 1

INTRODUCTION TO HVAC

COURSE OBJECTIVE: To Explain in detail Basic Components of Air-Conditioning & Refrigeration machines, Basic Refrigeration cycle, Accessories & Refrigerants

COURSE OUTCOME: The student shall understand the principles and working HVACsystems.

1.1 INTRODUCTION TO HVAC

Fundamentals-Modes of Heat Transfer Sensible Heat and Latent Heat

Heat transfer is defined as energy-in-transit due to temperature difference. Heat transfer takes place whenever there is a temperature gradient within a system or whenever two systems at different temperatures are brought into thermal contact. Heat, which is energy-in-transit, cannot be measured or observed directly, but the effects produced by it can be observed and measured. Since heat transfer involves transfer and/or conversion of energy, all heat transfer processes must obey the first and second laws of thermodynamics. However unlike thermodynamics, heat transfer deals with systems not in thermal equilibrium and using the heat transfer laws it is possible to find the rate at which energy is transferred due to heat transfer. From the engineer's point of view, estimating the rate of heat transfer is a key requirement.

Heating ventilation & Air Conditioning involves heat transfer; hence a good understanding of the fundamentals of heat transfer is a must for a student of Heating ventilation & Air Conditioning. This section deals with a brief review of heat transfer relevant to Heating ventilation & Air Conditioning.

Generally heat transfer takes place in three different modes: conduction, convection and radiation. In most of the engineering problems heat transfer takes place by more than one mode simultaneously, i.e., these heat transfer problems are of multi-mode type.

Conduction heat transfer:

Conduction heat transfer takes place whenever a temperature gradient exists in a stationary medium. Conduction is one of the basic modes of heat transfer. On a microscopic level, conduction heat transfer is due to the elastic impact of molecules in fluids, due to molecular vibration and rotation about their lattice positions and due to free electron migration in solids.

The fundamental law that governs conduction heat transfer is called Fourier's law of heat conduction, it is an empirical statement based on experimental observations and is given by:

$$Q_x = -k.A.\frac{dT}{dx}$$

In the above equation, Qx is the rate of heat transfer by conduction in x-direction, (dT/dx) is the temperature gradient in x-direction, A is the cross-sectional area normal to the x-direction and k is proportionality constant and is a property of the conduction medium, called thermal conductivity. The '-' sign in the above equation is a consequence of 2^{nd} law of thermodynamics, which states that in spontaneous process heat must always flow from a high temperature to a low temperature (i.e., dT/dx must be negative).

The thermal conductivity is an important property of the medium as it is equal to the conduction heat transfer per unit cross-sectional area per unit temperature gradient. Thermal conductivity of materials varies significantly. Generally it is very high for pure metals and low for non-metals. Thermal conductivity of solids is generally greater than that of fluids. Table 7.1 shows typical thermal conductivity values at 300 K. Thermal conductivity of solids and liquids vary mainly with temperature, while thermal conductivity of gases depend on both temperature and pressure. For isotropic materials the value of thermal conductivity is same in all directions, while for anisotropic materials such as wood and graphite the value of thermal conductivity is different in different directions. In Heating ventilation & Air Conditioning conductivity materials are used in the construction of heat exchangers, while low thermal conductivity materials are required for insulating refrigerant pipelines,refrigerated cabinets, building walls etc.

Material	**Thermal conductivity (W/m K)**
Copper (pure)	399
Gold (pure)	317
Aluminum (pure)	237
Iron (pure)	80.2
Carbon steel (1 %)	43
Stainless Steel (18/8)	15.1
Glass	0.81
Plastics	0.2 – 0.3
Wood (shredded/cemented)	0.087
Cork	0.039
Water (liquid)	0.6
Ethylene glycol (liquid)	0.26
Hydrogen (gas)	0.18
Benzene (liquid)	0.159
Air	0.026

General heat conduction equation:

Fourier’s law of heat conduction shows that to estimate the heat transfer through a given medium of known thermal conductivity and cross-sectional area, one needs the spatial variation of temperature. In addition the temperature at any point in the medium may vary with time also. The spatial and temporal variations are obtained by solving the heat conduction equation. The heat conduction equation is obtained by applying first law of thermodynamics and Fourier’s law to an elemental control volume of the conducting medium. In rectangular coordinates, the general heat conduction equation for a conducting media with constant thermo-physical properties is given by:

$$\frac{1}{\alpha}\frac{\partial T}{\partial \tau} = \left[\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right] + \frac{q_g}{k}$$

In the above equation, $\alpha = \frac{k}{\rho c_p}$ is a property of the media and is called as thermal diffusivity,

qg is the rate of heat generation per unit volume inside the control volume and τ is the time.

Convection Heat Transfer:

Convection heat transfer takes place between a surface and a moving fluid, when they are at different temperatures. In a strict sense, convection is not a basic mode of heat transfer as the heat transfer from the surface to the fluid consists of two mechanisms operating simultaneously. The first one is energy transfer due to molecular motion (conduction) through a fluid layer adjacent to the surface, which remains stationary with respect to the solid surface due to no-slip condition. Superimposed upon this conductive mode is energy transfer by the macroscopic motion of fluid particles by virtue of an external force, which could be generated by a pump or fan (forced convection) or generated due to buoyancy, caused by density gradients.

When fluid flows over a surface, its velocity and temperature adjacent to the surface are same as that of the surface due to the no-slip condition. The velocity and temperature far away from the surface may remain unaffected. The region in which the velocity and temperature vary from that of the surface to that of the free stream are called as hydrodynamic and thermal boundary layers, respectively. fluid with free stream velocity U∞ flows over a flat plate. In the vicinity of the surface , the velocity tends to vary from zero (when the surface is stationary) to its free stream value U∞. This happens in a narrow region whose thickness is of the order of ReL-0.5 (ReL = U∞L/ν) where there is a sharp velocity gradient. This narrow region is called hydrodynamic boundary layer. In the hydrodynamic boundary layer region the inertial terms are of same order magnitude as the viscous terms. Similarly to the velocity gradient, there is a sharp temperature gradient in this vicinity of the surface if the temperature of the surface of the plate is different from that of the flow stream. This region is called thermal boundary layer, δt whose thickness is of the order of (ReLPr) , where Pr is the Prandtl number, given by:

$$Pr = \frac{c_{p,f}\mu_f}{k_f} = \frac{\nu_f}{\alpha_f}$$

Radiation heat transfer:

Radiation is another fundamental mode of heat transfer. Unlike conduction and convection, radiation heat transfer does not require a medium for transmission as energy transfer occurs due to the propagation of electromagnetic waves. A body due to its temperature emits electromagnetic radiation, and it is emitted at all temperatures. It is propagated with the speed of light (3×10^8 m/s) in a straight line in vacuum. Its speed decreases in a medium but it travels in a straight line in homogenous medium. The speed of light, c is equal to the product of wavelength λ and frequency ν, that is,

$$C = \lambda \nu$$

Thermal radiation lies in the range of 0.1 to 100 μm, while visible light lies in therange of 0.35 to 0.75 μm. Propagation of thermal radiation takes place in the form ofdiscrete quanta, each quantum having energy of

$$\nu = h \; E$$

Where, h is Plank's constant, h = 6.625 x 10^{-34} Js.

The radiation energy is converted into heat when it strikes a body.

The radiation energy emitted by a surface is obtained by integrating Planck's equation

over all the wavelengths. For a real surface the radiation energy given by StefanBoltzmann's law is:

$$Q_r = \varepsilon.\sigma.A.Ts$$

where Qr = Rate of thermal energy emission, W

ε = Emissivity of the surface

σ = Stefan-Boltzmann's constant, 5.669 X 10^{-8}

A=Surface area, m

Ts = Surface Temperature,

The emissivity is a property of the radiating surface and is defined as the emissive power (energy radiated by the body per unit area per unit time over all the wavelengths) of the surface to that of an ideal radiating surface. The ideal radiator is called as a "black body", whose emissivity is 1. A black body is a hypothetical body that absorbs all the incident (all wave lengths) radiation. The term 'black' has nothing to do with black colour. A white coloured body can also absorb infrared radiation as much as a black coloured surface. A hollow enclosure with a small hole is an approximation to black body. Any radiation that enters through the hole is absorbed by multiple reflections within the cavity. The hole being small very small quantity of it escapes through the hole.

The radiation heat exchange between any two surfaces 1 and 2 at different temperatures T1 and T2 is given by:

$$Q_{1\text{-}2} = \sigma.A.F_\varepsilon F_A (T_1^4 - T_2^4)$$

where	$Q_{1\text{-}2}$	=	Radiation heat transfer between 1 and 2, W
	F_ε	=	Surface optical property factor
	F_A	=	Geometric shape factor
	T_1, T_2	=	Surface temperatures of 1 and 2, K

Type of fluid and flow	Convective heat transfer coefficient h_c (W/m^2 K)
Air, free convection	6 – 30
Water, free convection	20 – 100
Air or superheated steam, forced convection	30 – 300
Oil, forced convection	60 – 1800
Water, forced convection	300 – 18000
Synthetic refrigerants, boiling	500 - 3000
Water, boiling	3000 – 60000
Synthetic refrigerants, condensing	1500 - 5000
Steam, condensing	6000 – 120000

Traditionally, from the manner in which the convection heat transfer rate is defined, evaluating the convective heat transfer coefficient has become the main objective of the problem. The convective heat transfer coefficient can vary widely depending upon the type of fluid and flow field and temperature difference. Table shows typical order-of-magnitude values of convective heat transfer coefficients for different conditions.

Sensible heat

Sensible heat is when energy is transferred as heat to an object, changing the temperature but not its state. If you can measure the temperature of the heat, it is sensible. A body (solid, liquid or gas) of mass m and specific heat c is heated to change its temperature from T1 to T2 without changing its state. Indeed, the volume or the pressure of the body is unchanged. The energy received by the body responsible for its risen temperature is given by the relation:

Q=m*c*(T2-T1) in joules

Q=m*c*(T2-T1)1055,06 in BTU

Latent heat

In contrast to sensible heat, latent heat is the energy released or absorbed that changes the state of a body during a constant temperature process. This process leaves temperature unaffected - it won't get higher or lower. The most common forms of latent heat are fusion and vaporization.

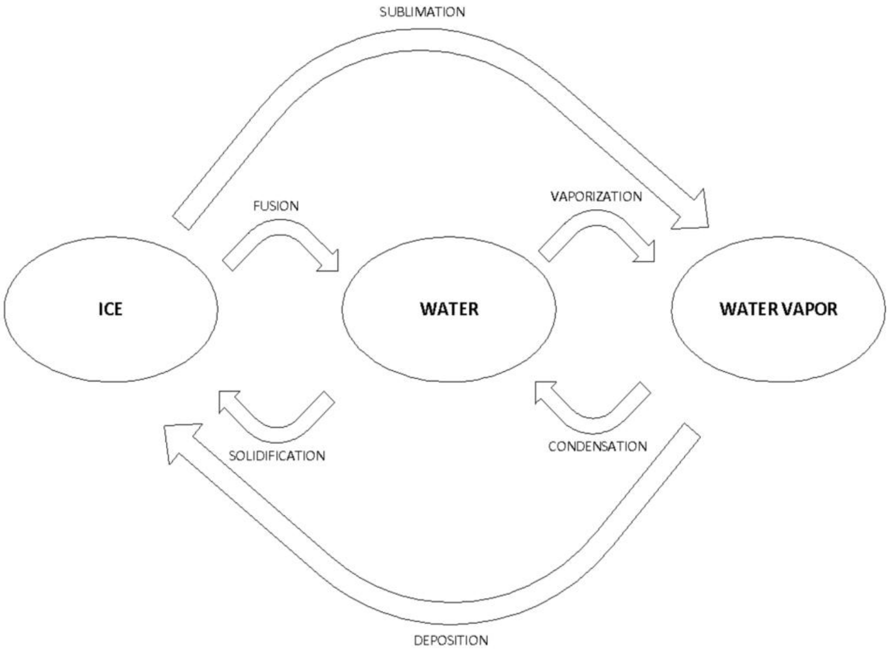
Fusion

Fusion is the passage of a body from solid state to liquid state. During the process of changing phasis, the temperature stays the same. Energy is supplied to a solid in order to melt it and energy is released from a liquid when it freezes. The best example is an ice cube melting at 32 °F (0°C).

Vaporization

Vaporization is the passage of a body from the liquid state to the vapor state. If conditions allow, the formation of vapor bubbles within a liquid, (known as boiling). Heat must be supplied to a liquid to effect vaporization. If there is not enough heat, it may come from the system itself as a reduction in temperature. The atoms or molecules of a liquid are held together by cohesive forces, and these forces must be overcome in separating the atoms or molecules to form the vapor. The heat of vaporization is a direct measure of these cohesive forces. The best example is a pot of water boiling at 212 °F (100°C).

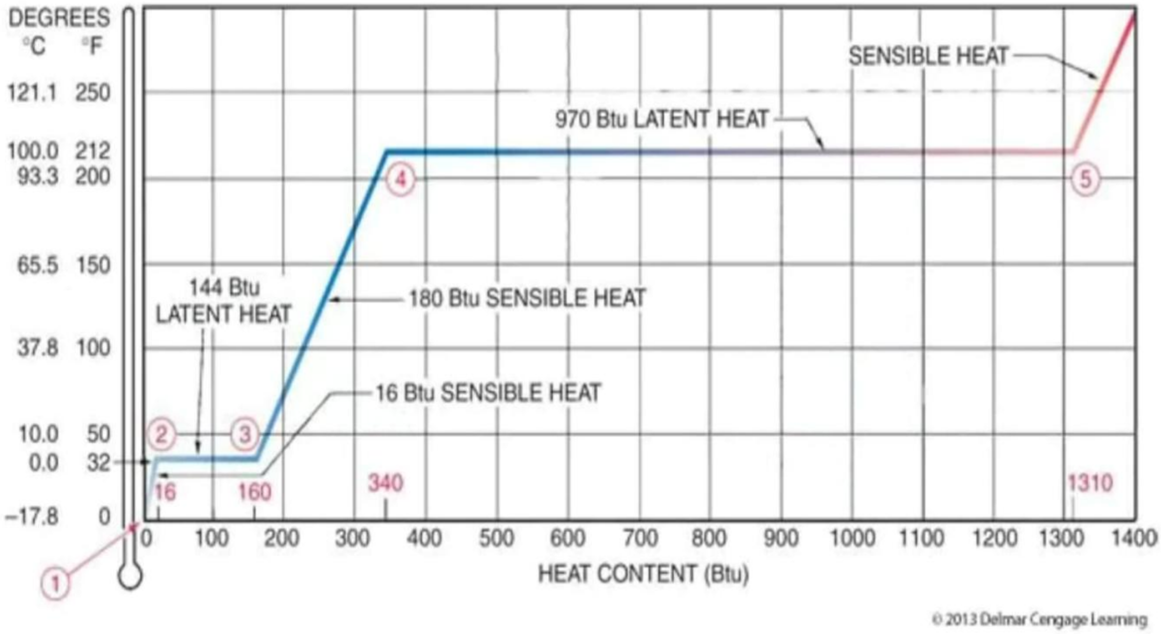
Water is an excellent example because it can go through fusion and vaporization. Take a look at the diagram below.



In application

In an HVAC system, we can look at the total power corresponding to the sum of the sensible power required to lower the temperature of the air and the latent power necessary to dehumidify this air.

For a better understanding of the concept of sensible and latent heat, let’s look at this water diagram phase:



Looking at the diagram, we can see that at stage 1, the water is solid.

From stage 1 to stage 2, the ice is heated and the temperature increases. The energy spent is 16 Btu. (Sensible heat)

From stage 2 to stage 3, the ice is changing phase to become liquid. The energy spent now is 144 Btu. (Latent Heat)

From stage 3 to stage 4, the liquid is heated once again and the temperature increases. The energy spent is 180 Btu (Sensible heat).

From stage 4 to stage 5, the liquid is at a saturation point. It has absorbed all the sensible heat it can absorb. The more heat added will now change the liquid to vapor. The energy spent is 970 Btu (Latent heat).

Once all the liquid has turn to vapor, any more heat added will increase the temperature of the steam as sensible heat.

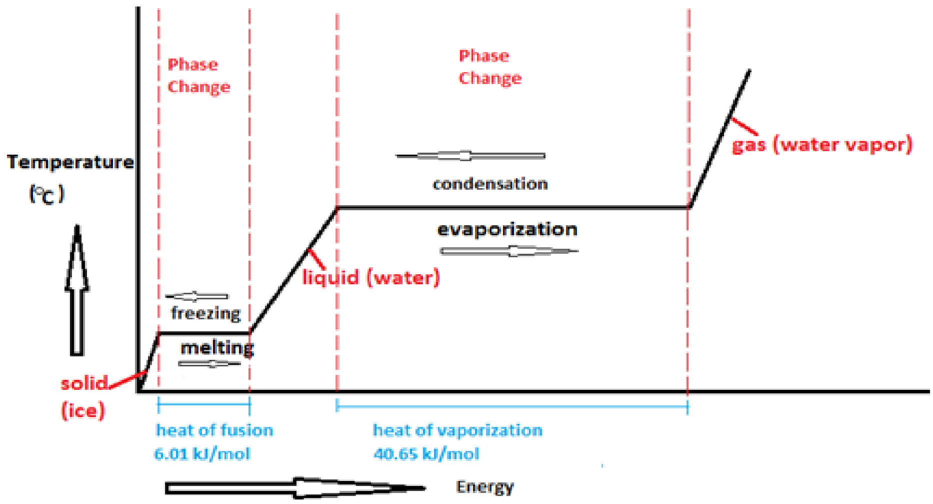
Finally, we observe that latent heat uses much more energy than sensible heat to change the phase of a liquid, solid or steam. All this is part of the basic principle of heat pump and refrigeration cycle. A heat pump is a machine or device that moves heat from one location at lower temperature to another location at a higher temperature, using mechanical work or a high temperature heat source. Refrigeration use the same principle but in reverse. Heat is now moving from a cold place to a warm place.

Latent Heat:

Latent heat is defined as the heat or energy that is absorbed or released during a phase change of a substance. It could either be from a gas to a liquid or liquid to solid and vice versa. Latent heat is related to a heat property called enthalpy.

However, an important point that we should consider regarding latent heat is that the temperature of the substance remains constant. As far as the mechanism is concerned, latent heat is the work that is needed to overcome the attractive forces that hold molecules and atoms together in a substance.

Let's take an example. Suppose a solid substance is changing to a liquid, it needs to absorb energy to push the molecules into a wider, more fluid volume. Similarly, when a substance changes from a gas phase to a liquid, their density levels also need to go from lower to a higher level wherein the substance then needs to release or lose energy so that the molecules come closer together. In essence, this energy that is required by a substance to either freeze, melt or boil is said to be latent heat.



1.2 BASIC COMPONENTS OF AIR-CONDITIONING AND REFRIGERATION MACHINES

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.

Standard Vapour Compression Refrigeration System (VCRS)

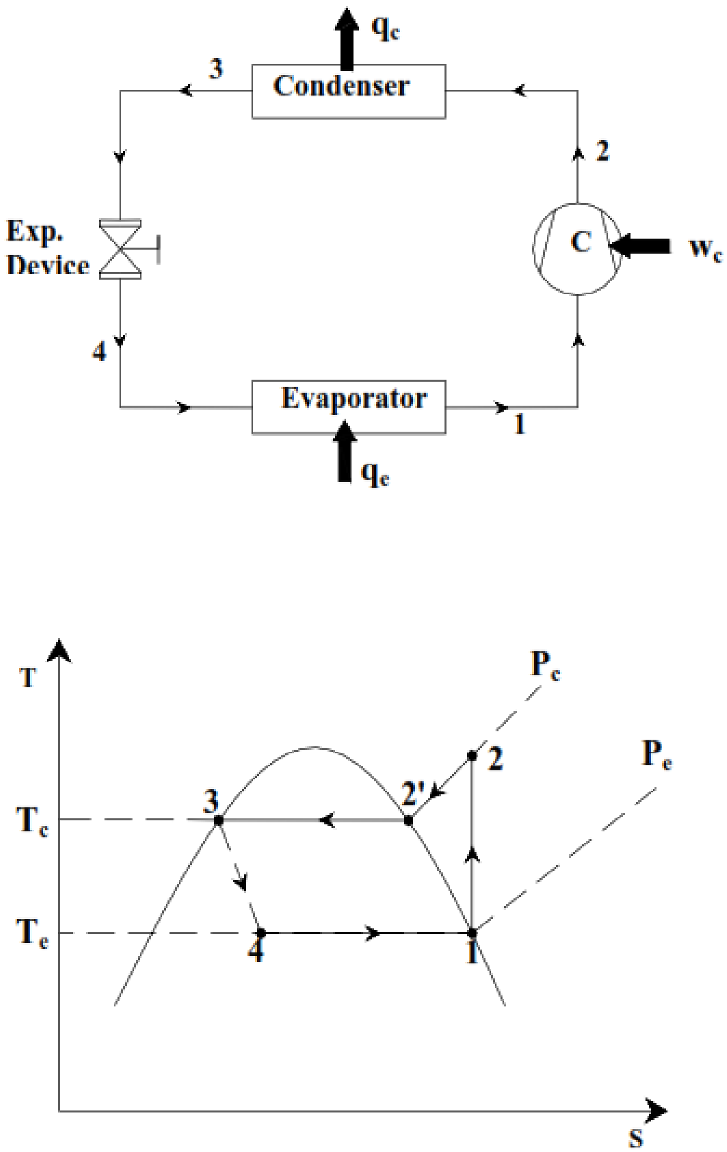


Fig.10.5. Standard Vapour compression refrigeration system

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.

Figure 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.

A simple analysis of standard vapour compression refrigeration system can be carried out

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, Qe is given by:

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

where mr is the refrigerant mass flow rate in kg/s, h1 and h4 are the specific enthalpies (kJ/kg)at the exit and inlet to the evaporator, respectively. (h_1-h_4) 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 Pe is the saturation pressure corresponding to evaporator temperature Te, i.e.,

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

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

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

Where h2 and h1 are the specific enthalpies (kJ/kg) at the exit and inlet to the compressor, respectively. (h2-h1) 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, Qc is given by:

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

where h3 and h2 are the specific enthalpies (kJ/kg) at the exit and inlet to the condenser, respectively.

The condenser pressure Pc is the saturation pressure corresponding to evaporator temperature T , i.e.,

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

Expansion device:

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

$$h_3 = h_4$$

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

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

where x4 is the quality of refrigerant at point 4, hf,e, hg,e, hfg enthalpy, are the saturated liquid saturated vapour enthalpy and latent heat of vaporization at evaporator pressure, respectively.

The COP of the system is given by:

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

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

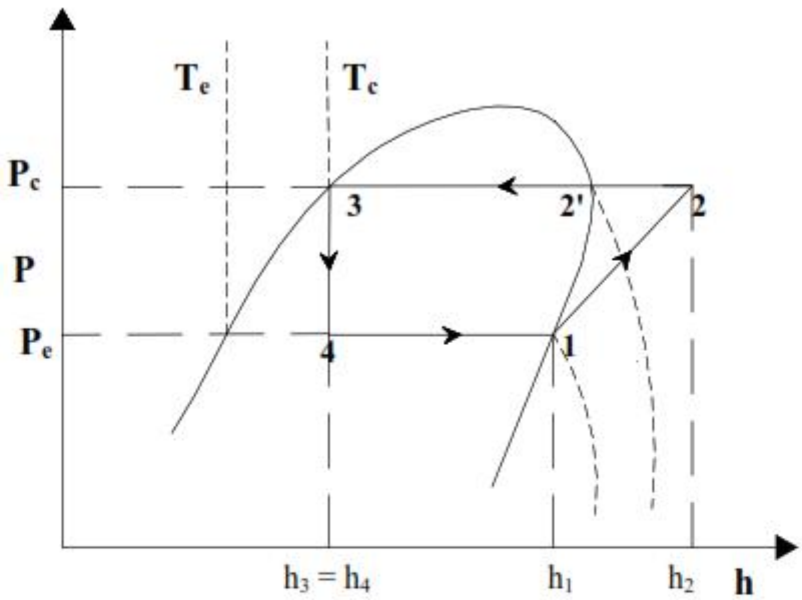


Fig.10.9. Standard vapour compression refrigeration cycle on a P-h chart

Since the various performance parameters are expressed in terms of enthalpies, it is very addition to the P-h and T-s charts one can also use thermodynamic property tables 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 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.

COMPONENTS OF REFRIGERATION SYSTEM:

A typical refrigeration system consists of several basic components such as compressors, condensers, expansion devices, evaporators, in addition to several accessories such as controls, filters, driers, oil separators etc. For efficient operation of the refrigeration system, it is essential that there be a proper matching between various components. Before analyzing the balanced performance of the complete system, it is essential to study the design and performance characteristics of individual components. Except in special applications, the refrigeration system components are standard components manufactured by industries specializing in individual components. Generally for large systems, depending upon the design specifications, components are selected from the manufacturers' catalogs and are assembled at site. Even though most of the components are standard off-the-shelf items, sometimes components such as evaporator may be made to order. Small capacity refrigeration systems such as refrigerators, room and package air conditioners, water coolers are available as complete systems. In this case the manufacturer himself designs or selects the system components, assembles them at the factory, tests them for performance and then sells the complete system as a unit.

COMPRESSORS

A compressor is the most important and often the costliest component (typically 30 to 40 percent of total cost) of any vapour compression refrigeration system (VCRS). The function of a compressor in a VCRS is to continuously draw the refrigerant vapour from the evaporator, so that a low pressure and low temperature can be maintained in the evaporator at which the refrigerant can boil extracting heat from the refrigerated space. The compressor then has to raise the pressure of the refrigerant to a level at which it can condense by rejecting heat to the cooling medium in the condenser.

CLASSIFICATION OF COMPRESSORS

Compressors used in refrigeration systems can be classified in several ways:

a) Based on the working principle:

i. Positive displacement type

ii. Roto-dynamic type

In positive displacement type compressors, compression is achieved by trapping a refrigerant vapour into an enclosed space and then reducing its volume. Since a fixed amount of refrigerant is trapped each time, its pressure rises as its volume is reduced. When the pressure rises to a level that is slightly higher than the condensing pressure, then it is expelled from the enclosed space and a fresh charge of low-pressure refrigerant is drawn in and the cycle continues. Since the flow of refrigerant to the compressor is not steady, the positive displacement type compressor is a pulsating flow device. However, since the operating speeds are normally very high the flow appears to be almost steady on macroscopic time scale. Since the flow is pulsating on a microscopic time scale, positive displacement type compressors are prone to high wear, vibration and noise level.

Depending upon the construction, positive displacement type compressors used in refrigeration and air conditioning can be classified into:

i. Reciprocating type

ii. Rotary type with sliding vanes (rolling piston type or multiple vane type)

iii. Rotary screw type (single screw or twin-screw type)

iv. Orbital compressors, and

v. Acoustic compressors.

In roto-dynamic compressors, the pressure rise of refrigerant is achieved by imparting kinetic energy to a steadily flowing stream of refrigerant by a rotating mechanical element and then converting into pressure as the refrigerant flows through a diverging passage. Unlike positive displacement type, the roto-dynamic type compressors are steady flow devices, hence are subjected to less wear and vibration.

Depending upon the construction, roto-dynamic type compressors can be classified into:

i. Radial flow type, or

ii. Axial flow type.

Centrifugal compressors (also known as turbo-compressors) are radial flow type, roto-dynamic compressors. These compressors are widely used in large capacity refrigeration and air conditioning systems. Axial flow compressors are normally used in gas liquefaction applications.

b) **Based on arrangement of compressor motor or external drive:**

i. Open type

ii. Hermetic (or sealed) type

iii. Semi-hermetic (or semi-sealed) type

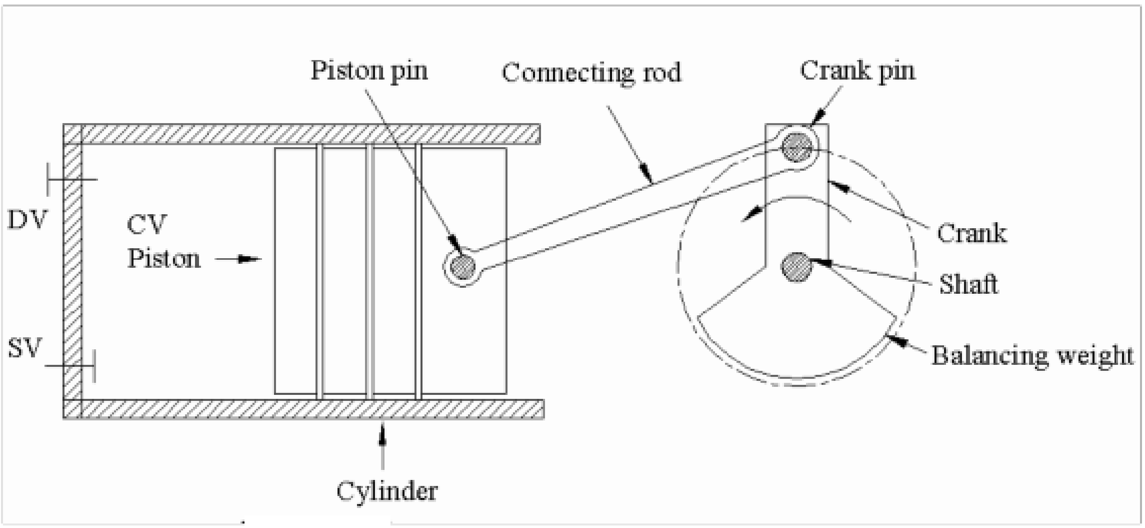
In open type compressors the rotating shaft of the compressor extends through a seal in the crankcase for an external drive. The external drive may be an electrical motor or an engine (e.g. diesel engine). The compressor may be belt driven or gear driven. Open type compressors are normally used in medium to large capacity refrigeration system for all refrigerants and for ammonia (due to its incompatibility with hermetic motor materials). Open type compressors are characterized by high efficiency, flexibility, better compressor cooling and service ability. However, since the shaft has to extend through the seal, refrigerant leakage from the system cannot be eliminated completely. Hence refrigeration systems using open type compressors require a refrigerant reservoir to take care of the refrigerant leakage for some time, and then regular maintenance for charging the system with refrigerant, changing of seals, gaskets etc.

In hermetic compressors, the motor and the compressor are enclosed in the same housing to prevent refrigerant leakage. The housing has welded connections for refrigerant inlet and outlet and for power input socket. As a result of this, there is virtually no possibility of refrigerant leakage from the compressor. All motors reject a part of the power supplied to it due to eddy currents and friction, that is, inefficiencies. Similarly the compressor also gets heated-up due to friction and also due to temperature rise of the vapor during compression. In

Open type, both the compressor and the motor normally reject heat to the Surrounding air for efficient operation. In hermetic compressors heat cannot be rejected to the surrounding air since both are enclosed in a shell. Hence, the cold suction gas is made to flow over the motor and the compressor before entering the compressor. This keeps the motor cool. The motor winding is in direct contact with the refrigerant hence only those refrigerants, which have high dielectric strength, can be used in hermetic compressors. The cooling rate depends upon the flow rate of the refrigerant, its temperature and the thermal properties of the refrigerant. If flow rate is not sufficient and/or if the temperature is not low enough the insulation on the winding of the motor can burn out and short-circuiting may occur. Hence, hermetically sealed compressors give satisfactory and safe performance over a very narrow range of design temperature and should not be used for off-design conditions.

Reciprocating compressors

Reciprocating compressor is the workhorse of the refrigeration and air conditioning industry. It is the most widely used compressor with cooling capacities ranging from a few Watts to hundreds of kilowatts. Modern day reciprocating compressors are high speed (≈ 3000 to 3600 rpm), single acting, single or multi-cylinder (upto 16 cylinders) type. Reciprocating compressors consist of a piston moving back and forth in a cylinder, with suction and discharge valves to achieve suction and compression of the refrigerant vapor.



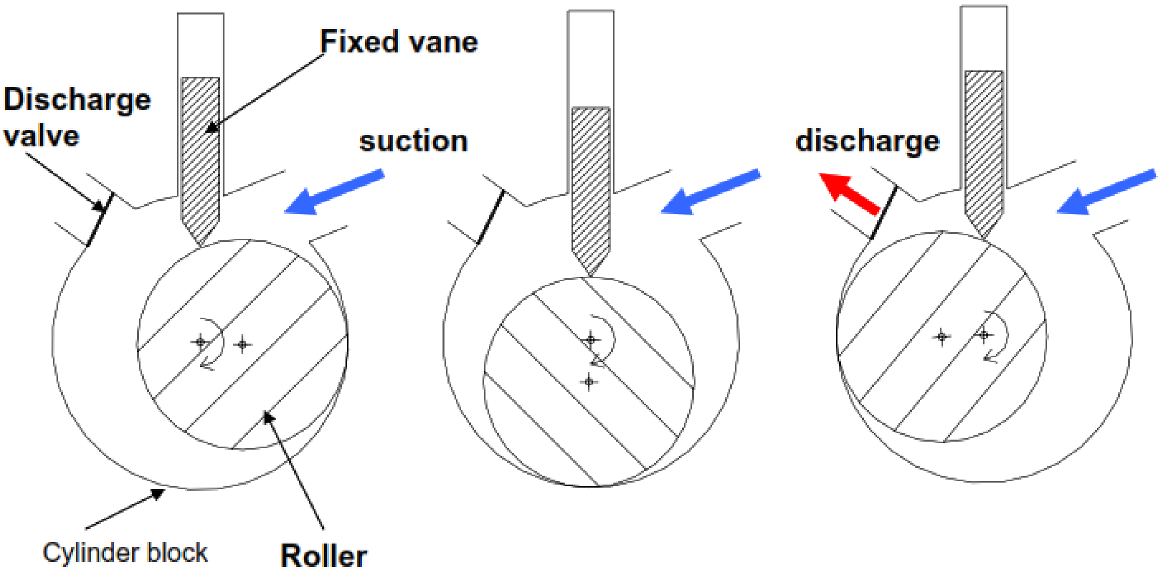
Schematic of a reciprocating compressor

Its construction and working are somewhat similar to a two-stroke engine, as suction and compression of the refrigerant vapor are completed in one revolution of the crank. The suction side of the compressor is connected to the exit of the evaporator, while the discharge side of the compressor is connected to the condenser inlet. The suction (inlet) and the discharge (outlet) valves open and close due to pressure differences between the cylinder and inlet or outlet manifolds respectively. The pressure in the inlet manifold is equal to or slightly less than the evaporator pressure. Similarly the pressure in the outlet manifold is equal to or slightly greater than the condenser pressure. The purpose of the manifolds is to provide stable inlet and outlet pressures for the smooth operation of the valves and also provide a space for mounting the valves.

The valves used are of reed or plate type, which are either floating or clamped. Usually, backstops are provided to limit the valve displacement and springs may be provided for smooth return after opening or closing. The piston speed is decided by valve type. Too high a speed will give excessive vapour velocities that will decrease the volumetric efficiency and the throttling loss will decrease the compression efficiency.

Rolling piston (fixed vane) type compressors:

Rolling piston or fixed vane type compressors are used in small refrigeration systems (upto 2 kW capacity) such as domestic refrigerators or air conditioners. These compressors belong to the class of positive displacement type as compression is achieved by reducing the volume of the refrigerant. In this type of compressors, the rotating shaft of the roller has its axis of rotation that matches with the center line of the cylinder, however, it is eccentric with respect to the roller. This eccentricity of the shaft with respect to the roller creates suction and compression of the refrigerant as shown in Fig.20.1. A single vane or blade is positioned in the non-rotating cylindrical block. The rotating motion of the roller causes a reciprocating motion of the single vane.



Working principle of a rolling piston type compressor

This type of compressor does not require a suction valve but requires a discharge valve. The sealing between the high and low pressure sides has to be provided:

-Along the line of contact between roller and cylinder block

- Along the line of contact between vane and roller, and

- between the roller and end-pates

The leakage is controlled through hydrodynamic sealing and matching between the mating components. The effectiveness of the sealing depends on the clearance, compressor speed, surface finish and oil viscosity. Close tolerances and good surface finishing is required to minimize internal leakage. Unlike in reciprocating compressors, the small clearance volume filled with high-pressure refrigerant does not expand, but simply mixes with the suction refrigerant in the suction space. As a result, the volumetric efficiency does not reduce drastically with increasing pressure ratio, indicating small re-expansion losses. The compressor runs smoothly and is relatively quiet as the refrigerant flow is continuous.

The mass flow rate of refrigerant through the compressor is given by:

$$\dot{m} = \eta_V \left(\frac{\dot{V}_{SW}}{v_e} \right) = \left(\frac{\eta_V}{v_e} \right) \left(\frac{\pi}{4} \right) \left(\frac{N}{60} \right) (A^2 - B^2) L$$

where A = Inner diameter of the cylinder

B = Diameter of the roller

L = Length of the cylinder block

N = Rotation speed, RPM

η_V = Volumetric efficiency

v_e = specific volume of refrigerant at suction

Multiple vane type compressors:

In multiple vane type compressor, the axis of rotation coincides with the center of the roller (O), however, it is eccentric with respect to the center of the cylinder (O'). The rotor consists of a number of slots with sliding vanes. During the running of the compressor, the sliding vanes, which are normally made of non-metallic materials, are held against the cylinder due to centrifugal forces. The number of compression strokes produced in one revolution of the rotor is equal to the number of sliding vanes, thus a 4-vane compressor produces 4 compression strokes in one rotation.

In these compressors, sealing is required between the vanes and cylinder, between the vanes and the slots on the rotor and between the rotor and the end plate. However, since pressure difference across each slot is only a fraction of the total pressure difference, the sealing is not as critical as in fixed vane type compressor.

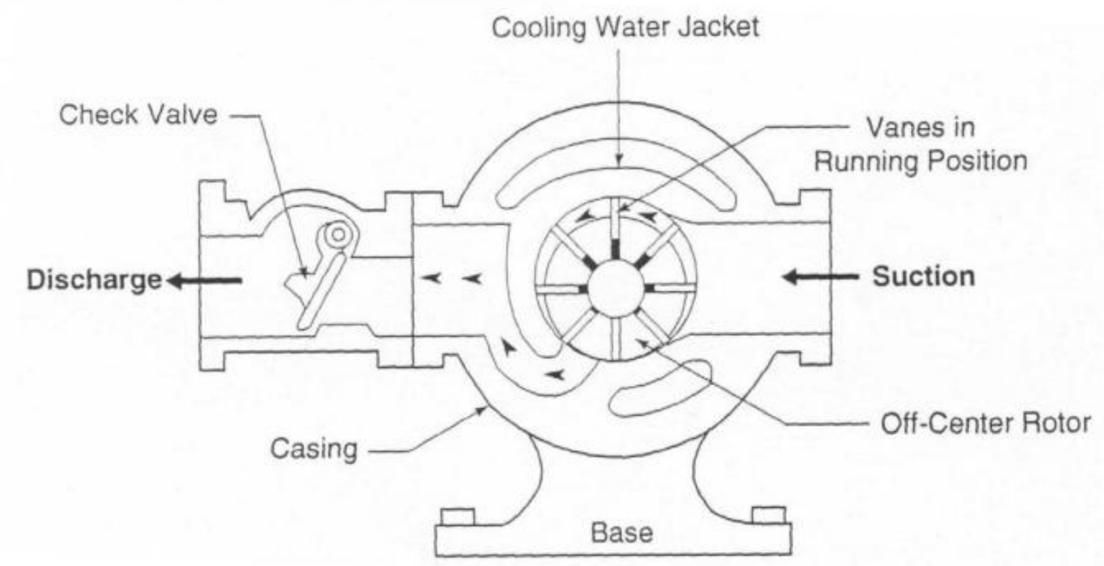
This type of compressor does not require suction or discharge valves, however, as shown in Fig., check valves are used on discharge side to prevent reverse rotation during off-time due to pressure difference. Since there are no discharge valves, the compressed refrigerant is opened to the discharge port when it has been compressed through a fixed volume ratio, depending upon the geometry. This implies that these compressors have a fixed built-in volume ratio. The built-in volume ratio is defined as "the ratio of a cell as it is closed off from the suction port to its volume before it opens to the discharge port". Since the volume ratio is fixed, the pressure ratio, rp is given by:

$$r_p = \left(\frac{P_d}{P_s}\right) = V_b{}^k$$

where Pd and Ps are the discharge and suction pressures, Vb is the built-in volume ratio and k is the index of compression. Since no centrifugal force is present when the compressor is off, the multiple vanes will not be pressed against the cylinder walls during the off-period.

As a result, high pressure refrigerant from the discharge side can flow back into the side and

pressure equalization between high and low pressure sides take place. This is beneficial from the compressor motor point-of-view as it reduces the required starting torque. However, this introduces cycling loss due to the entry of high pressure and hot refrigerant liquid into the evaporator. Hence, normally a non-return check valve is used on the discharge side which prevents the entry of refrigerant liquid from high pressure side into evaporator through the compressor during off-time, at the same time there will be pressure equalization across the vanes of the compressor. As a result,



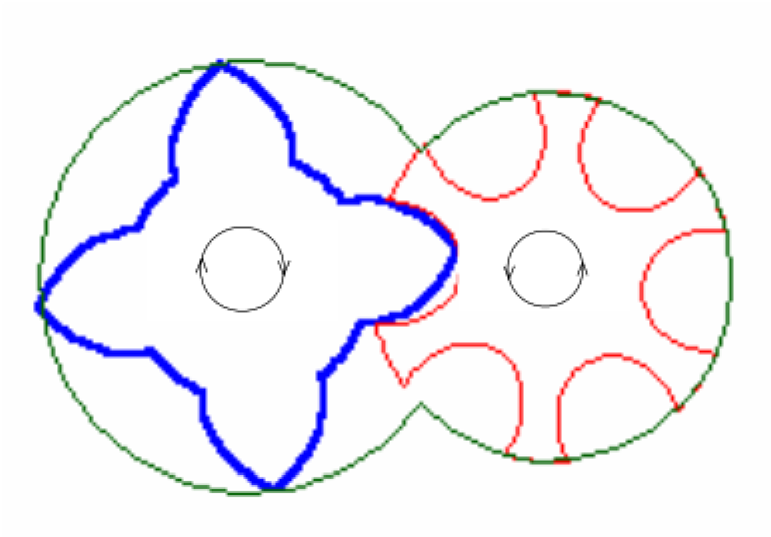
Sectional view of a multiple vane, rotary compressor

ROTARY, SCREW COMPRESSORS:

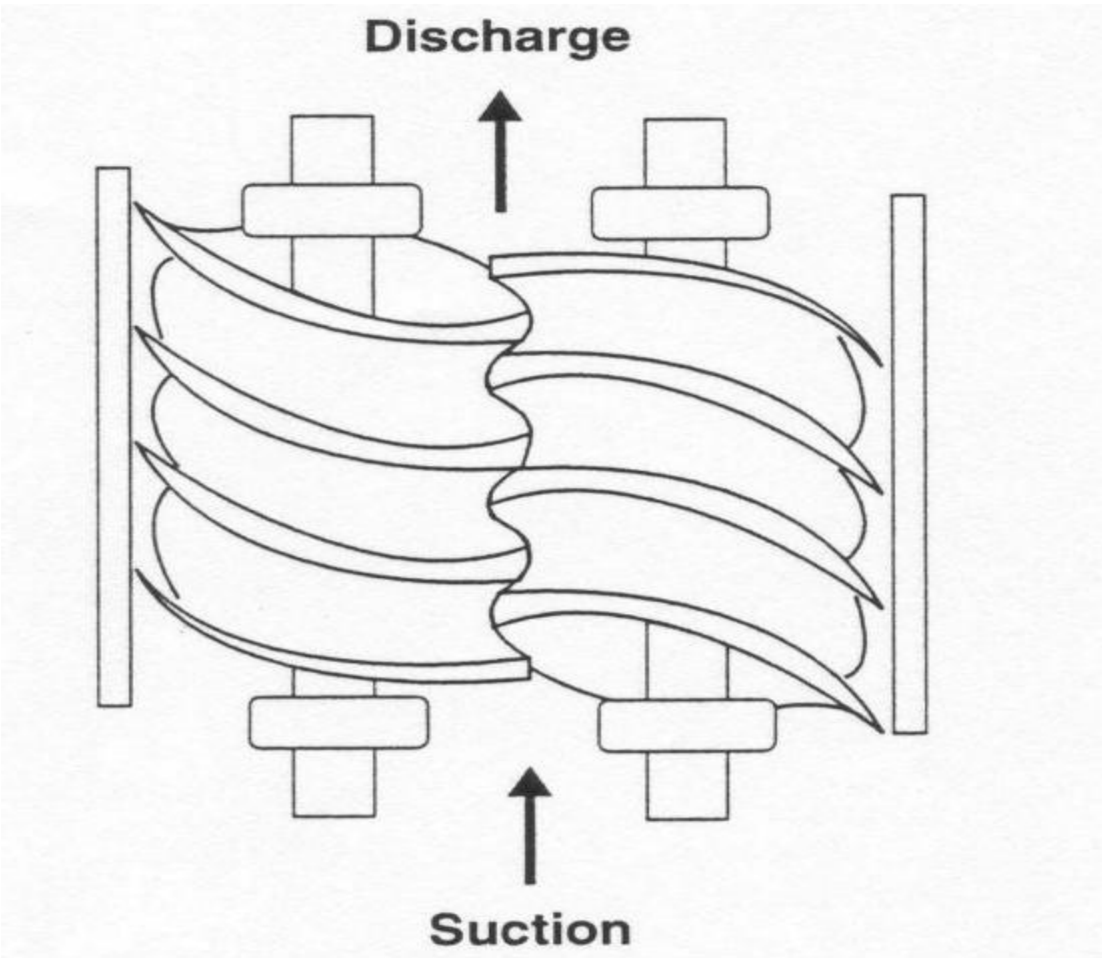
The rotary screw compressors can be either twin-screw type or single-screw

Twin-screw compressor:

The twin-screw type compressor consists of two mating helically grooved rotors, one male and the other female. Generally the male rotor drives the female rotor. The male rotor has lobes, while the female rotor has flutes or gullies. when the male rotor rotates at 3600RPM, the female rotor rotates at 2400 RPM. The flow is mainly in the axial direction. Suction an compression take place as the rotors unmesh and mesh. When one lobe-gully combination begins to unmesh the opposite lobe-gully combination begins to mesh.With 4 male lobes rotating at 3600 RPM, 4 interlobe volumes are per revolution, thus giving 4 X 3600 = 14400 discharges per minute.



Twin-screw compressor with 4 male lobes and 6 female gullies



Direction of refrigerant flow in a twin-screw compressor

Discharge takes place at a point decided by the designed built-in volume ratio, which depends entirely on the location of the delivery port and geometry of the compressor.

Since the built-in volume ratio is fixed by the geometry, a particular compressor is designed for a particular built-in pressure ratio. However, different built-in ratios can be obtained by changing the position of the discharge port. The built-in pressure ratio, rp given by:

$$r_p = \left(\frac{P_d}{P_s}\right) = V_b^{\,k}$$

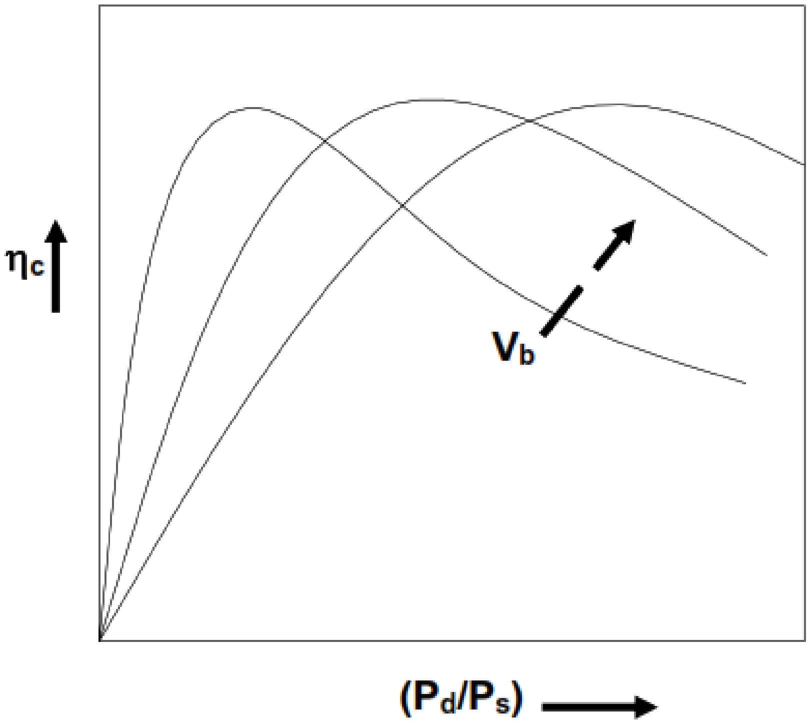
Where Pd and Ps are the discharge and suction pressures, Vb is the built-in volume ratio and k is the index of compression.

If the built-in pressure at the end of compression is less than the condensing pressure, high pressure refrigerant from discharge manifold flows back into the interlobe space when the discharge port is uncovered. This is called as under compression. On the other hand, if the built-in pressure at the end of compression is higher than the condensing pressure, then the compressed refrigerant rushes out in an unrestrained expansion as soon as the port is uncovered (over-compression).

Both under-compression and over-compression are undesirable as they lead to loss in efficiency. Lubrication and sealing between the rotors is obtained by injecting lubricating oil between the rotors. The oil also helps in cooling the compressor, as a result very high pressure ratios (upto 20:1) are possible without overheating the compressor.

The capacity of the screw compressor is normally controlled with the help of a slide valve. As the slide valve is opened, some amount of suction refrigerant escapes to the suction side without being compressed. This yields a smooth capacity control from 100 percent down to 10 percent of full load. It is observed that the power input is approximately proportional to refrigeration capacity upto about 30 percent, however, the efficiency decreases rapidly, there after. Figure shows the compression efficiency of a twin-screw compressor as a function of pressure ratio and built-in volume ratio. It can be seen that for a given built-in volume ratio, the efficiency reaches a peak at a particular optimum pressure ratio. The value of this optimum pressure ratio increases with built-in volume ratio as shown in the figure. If the design condition corresponds to the optimum pressure ratio, then the compression efficiency drops as the system operates at off-design conditions. However, when operated at the optimum pressure ratio, the efficiency is much higher than other types of compressors.

As the rotor normally rotates at high speeds, screw compressors can handle fairly large amounts of refrigerant flow rates compared to other positive displacement type compressors. Screw compressors are available in the capacity range of 70 to 4600 kW. They generally compete with high capacity reciprocating compressors and low capacity centrifugal compressors. They are available for a wide variety of refrigerants and applications. Compared to reciprocating compressors, screw compressors are balanced and hence do not suffer from vibration problems.



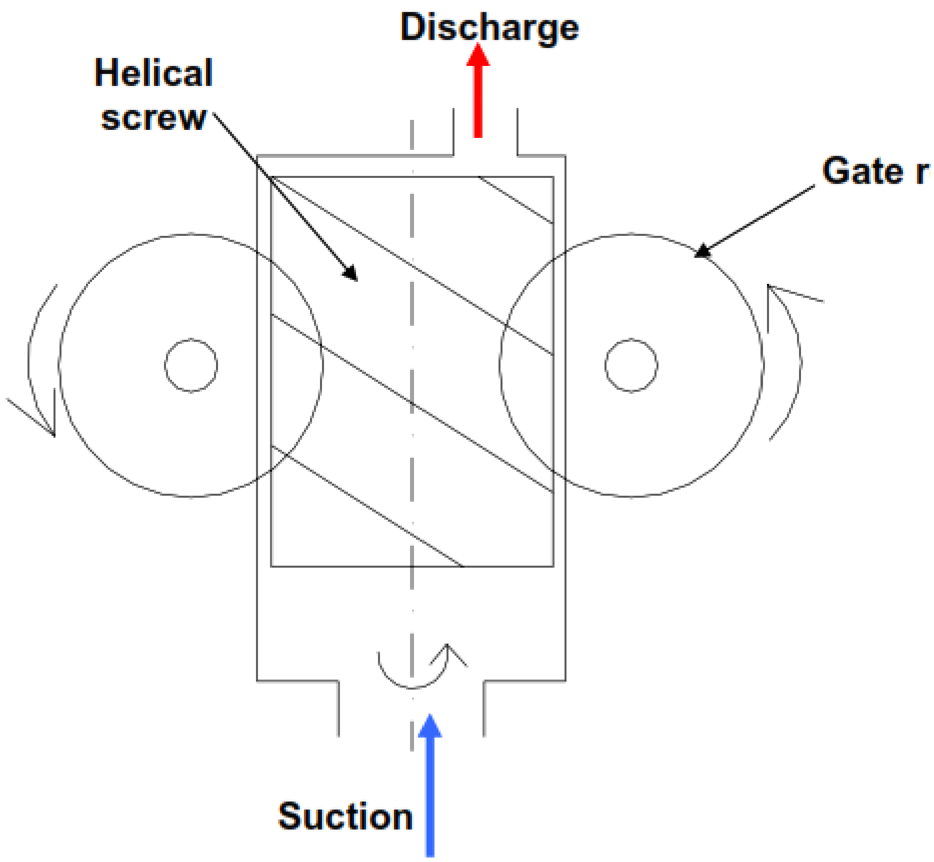
Variation of compression efficiency of a twin-screw compressor with pressure ratio and built-in volume ratio

Twin-screw compressors are rugged and are shown to be more reliable than reciprocating compressors; they are shown to run for 30000 – 40000 hours between major overhauls. They are compact compared to reciprocating compressors in the high capacity range.

Single-screw compressors:

As the name implies, single screw compressors consist of a single helical screw and two planet wheels or gate rotors. The helical screw is housed in a cylindrical casing with suction port at one end and discharge port at the other end as shown in Fig. Suction and compression are obtained as the screw and gate rotors unmesh and mesh. The high and low pressure regions in the cylinder casing are separated by the gate rotors.

The single screw is normally driven by an electric motor. The gate rotors are normally made of plastic materials. Very small power is required to rotate the gate rotors as the frictional losses between the metallic screw and the plastic gate rotors is very small. It is also possible to design the compressors with a single gate rotor. Similar to twin-screw, lubrication, sealing and compressor cooling is achieved by injecting lubricating oil into the compressor. An oil separator, oil cooler and pump are required to circulate the lubricating oil. It is also possible to achieve this by injecting liquid refrigerant, in which case there is no need for an oil separator. f plastic materials. Very small power is required to rotate the gate rotors as the frictional losses between the metallic screw and the plastic gate rotors is very small. It is also possible to design the compressors with a single gate rotor. Similar to twin-screw, lubrication, sealing and compressor cooling is achieved by injecting lubricating oil into the compressor. An oil separator, oil cooler and pump are required to circulate the lubricating oil. It is also possible to achieve this by injecting liquid refrigerant, in which case there is no need for an oil separator.

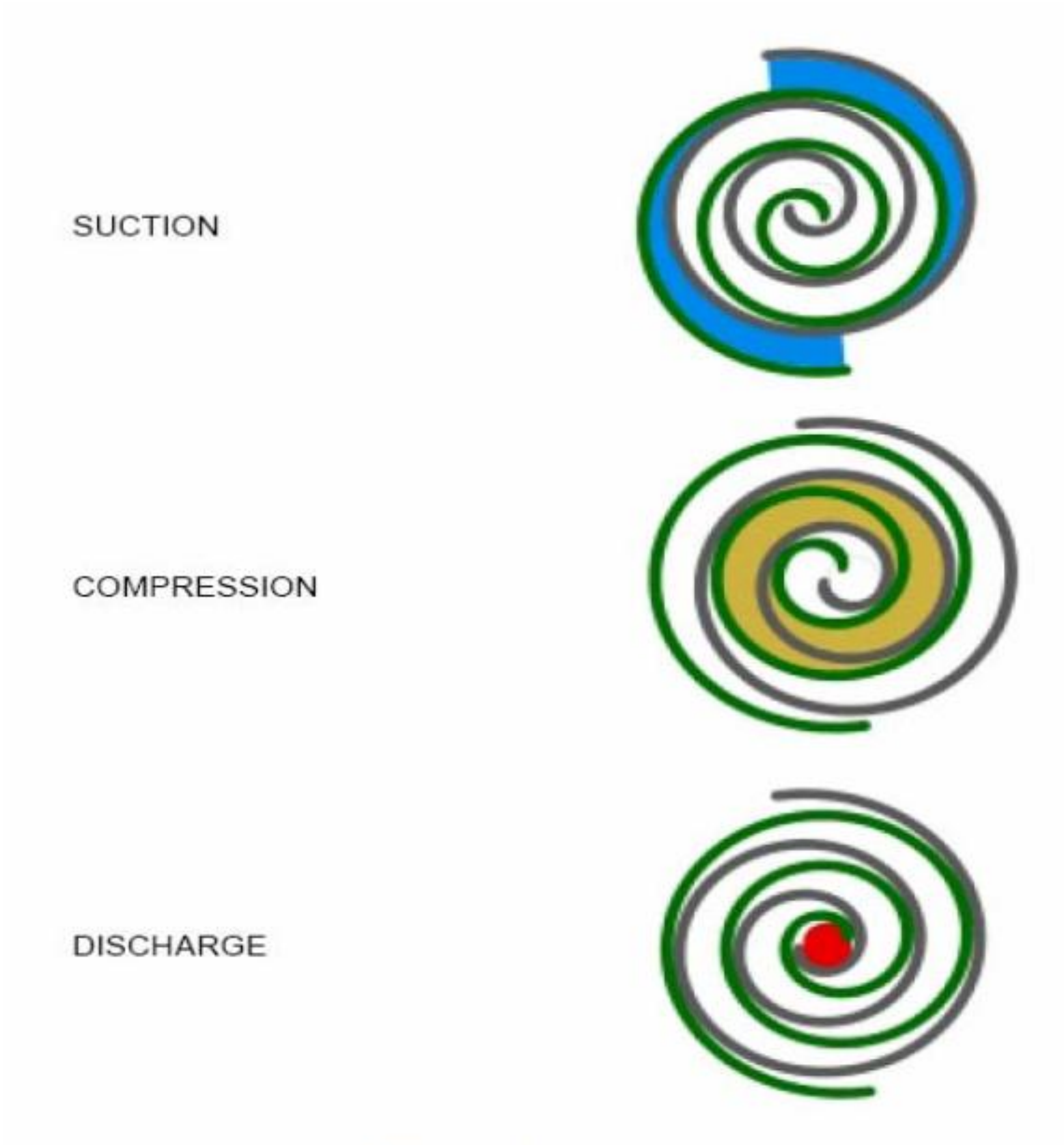


Working principle of a single-screw compressor

Scroll compressors:

Scroll compressors are orbital motion, positive displacement type compressors, in which suction and compression is obtained by using two mating, spiral shaped, scroll members, one fixed and the other orbiting. Figure shows the working principle of scroll compressors. Figures show the constructional details of scroll compressors. As shown in Fig, the compression process involves three orbits of the orbiting scroll. In the first orbit, the scrolls ingest and trap two pockets of suction gas. During the second orbit, the two pockets of gas are compressed to an intermediate pressure. In the final orbit, the two pockets reach discharge pressure and are simultaneously opened to the discharge port. This simultaneous process of suction, intermediate compression, and discharge leads to the smooth continuous compression process of the scroll compressor. One part that is not shown in this diagram but is essential to the operation of the scroll is the antirotation coupling. This device maintains a fixed angular relation of 180 degrees between the fixed and orbiting scrolls. This

fixed angular relation, coupled with the movement of the orbiting scroll, is the basis for the formation of gas compression pockets.



Working principle of a scroll compressor

Currently, the scroll compressors are used in small capacity (3 to 50 kW) refrigeration, air conditioning and heat pump applications. They are normally of hermetic type. Scroll compressors offer several advantages such as:

1. Large suction and discharge ports reduce pressure losses during suction and discharge

2. Physical separation of suction and compression reduce heat transfer to suction gas, leading to high volumetric efficiency

3. Volumetric efficiency is also high due to very low re-expansion losses and continuous flow over a wide range of operating conditions

4. Flatter capacity versus outdoor temperature curves

5. High compression efficiency, low noise and vibration compared to reciprocating compressors

6. Compact with minimum number of moving parts

As shown in, each scroll member is open at one end and bound by a base plate at the other end. They are fitted to form pockets of refrigerant between their respective base plates and various lines of contacts between the scroll walls. Compressor capacity is normally controlled by variable speed inverter drives.

Centrifugal compressors;

Centrifugal compressors; also known as turbo-compressors belong to the roto-dynamic type of compressors. In these compressors the required pressure rise takes place due to the continuous conversion of angular momentum imparted to the refrigerant vapour by a high-speed impeller into static pressure. Unlike reciprocating compressors, centrifugal compressors are steady-flow devices hence they are subjected to less vibration and noise.

Figure shows the working principle of a centrifugal compressor. As shown in the figure, low-pressure refrigerant enters the compressor through the eye of the impeller

(1). The impeller

(2) consists of a number of blades, which form flow passages (3) for refrigerant. From the eye, the refrigerant enters the flow passages formed by the impeller blades, which rotate at very high speed. As the refrigerant flows through the blade passages towards the tip of the impeller, it gains momentum and its static pressure also increases. From the tip of the impeller, the refrigerant flows into a stationary diffuser (4). In the diffuser, the refrigerant is decelerated and as a result the dynamic pressure drop is converted into static pressure rise, thus increasing the static pressure further. The vapour from the diffuser enters the volute casing (5) where further conversion of velocity into static pressure takes place due to the

divergent shape of the volute. Finally, the pressurized refrigerant leaves the compressor from the volute casing (6).

The gain in momentum is due to the transfer of momentum from the high speed impeller blades to the refrigerant confined between the blade passages. The increase in static pressure is due to the self-compression caused by the centrifugal action. This is analogous to the gravitational effect, which causes the fluid at a higher level to press the fluid below it due to gravity (or its weight). The static pressure produced in the impeller is equal to the static head, which would be produced by an equivalent gravitational column. If we assume the impeller blades to be radial and the inlet diameter of the impeller to be small, then the static head, h developed in the impeller passage for a single stage is given by:

$$h = \frac{V^2}{g}$$

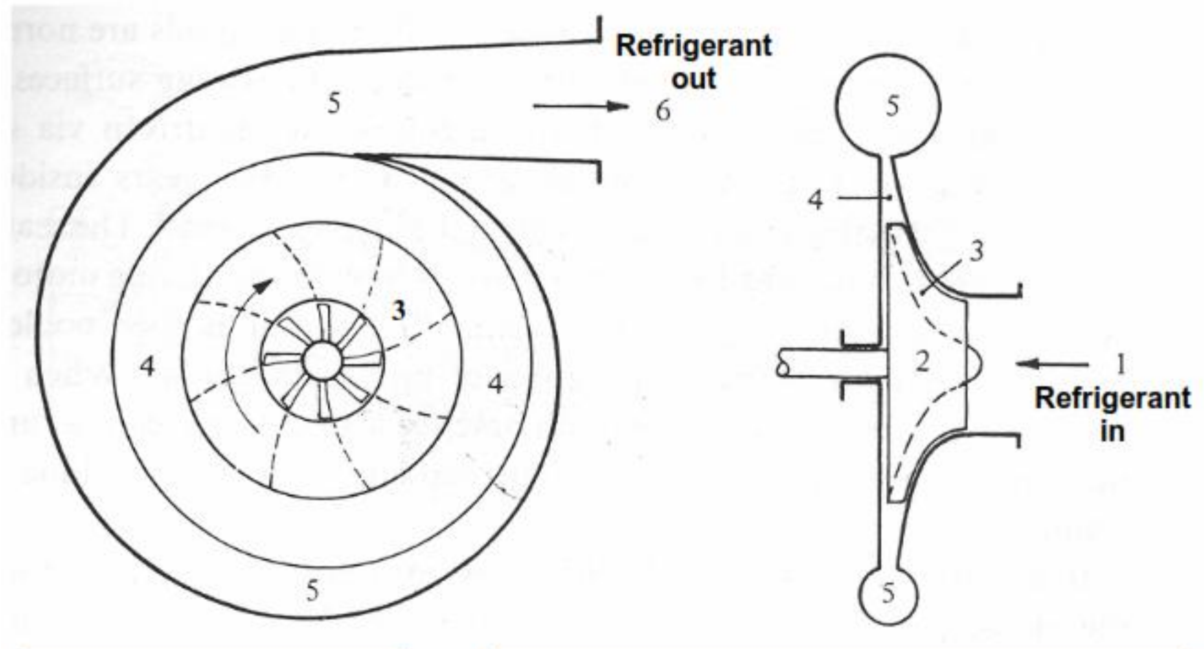
where h = static head developed, m

V = peripheral velocity of the impeller wheel or tip speed, m/s

g = acceleration due to gravity, m/s^2

Hence increase in total pressure, ΔP as the refrigerant flows through the passage is given by:

$$\Delta P = \rho g h = \rho V^2$$



Centrifugal Compressor
1: Refrigerant inlet (eye); 2: Impeller; 3: Refrigerant passages
4: Vaneless diffuser; 5: Volute casing; 6: Refrigerant discharge

Thus it can be seen that for a given refrigerant with a fixed density, the pressure rise depends only on the peripheral velocity or tip speed of the blade. The tip speed of the blade is proportional to the rotational speed (RPM) of the impeller and the impeller diameter. The maximum permissible tip speed is limited by the strength of the structural materials of the blade (usually made of high speed chrome-nickel steel) and the sonic velocity of the refrigerant. Under these limitations, the maximum achievable pressure rise (hence maximum achievable temperature lift) of single stage centrifugal compressor is limited for a given refrigerant. Hence, multistage centrifugal compressors are used for large temperature lift applications. In multistage centrifugal compressors, the discharge of the lower stage compressor is fed to the inlet of the next stage compressor and so on. In multistage centrifugal compressors, the impeller diameter of all stages remains same, but the width of the impeller becomes progressively narrower in the direction of flow as refrigerant density increases progressively.

The blades of the compressor or either forward curved or backward curved or radial. Backward curved blades were used in the older compressors, whereas the modern centrifugal compressors use mostly radial blades.

The stationary diffuser can be vaned or vaneless. As the name implies, in vaned diffuser vanes are used in the diffuser to form flow passages. The vanes can be fixed or adjustable. Vaned diffusers are compact compared to the vaneless diffusers and are commonly used for high discharge pressure applications. However, the presence of vanes in the diffusers can give rise to shocks, as the refrigerant velocities at the tip of the impeller blade could reach sonic velocities in large, high-speed centrifugal compressors. In vaneless diffusers the velocity of refrigerant in the diffuser decreases and static pressure increases as the radius increases. As a result, for a required pressure rise, the required size of the vaneless diffuser could be large compared to vaned diffuser. However, the problem of shock due to supersonic velocities at the tip does not arise with vaneless diffusers as the velocity can be diffused smoothly.

Generally adjustable guide vanes or pre-rotation vanes are added at the inlet (eye) of the impeller for capacity control.

Commercially centrifugal compressors are available for a wide variety of refrigeration and air conditioning applications with a wide variety of refrigerants.

These machines are available for the following ranges:

Evaporator temperatures	:	$-100^{o}C$ to $+10^{o}C$
Evaporator pressures	:	14 kPa to 700 kPa
Discharge pressure	:	upto 2000 kPa
Rotational speeds	:	1800 to 90,000 RPM
Refrigeration capacity	:	300 kW to 30000 kW

As mentioned before, on the lower side the capacity is limited by the impeller width and tipspeeds and on the higher side the capacity is limited by the physical size (currently the maximum impeller diameter is around 2 m).

Since the performance of centrifugal compressor is more sensitive to evaporator and condensing temperatures compared to a reciprocating compressor, it is essential to reduce the pressure drops when a centrifugal compressor is used in commercial systems. Commercial refrigeration systems using centrifugal compressors normally incorporate flash intercoolers to improve the system performance. Since the compressor is normally multi-staged, use of flash intercooler is relatively easy in case of centrifugal compressors.

Centrifugal compressors are normally lubricated using an oil pump (force feed) which can be driven either directly by the compressor rotor or by an external motor. The lubrication system consists of the oil pump, oil reservoir and an oil cooler. The components requiring lubrication are the main bearings, a thrust bearing (for the balancing disc) and the shaft seals. Compared to reciprocating compressors, the lubrication for centrifugal compressors is simplified as very little lubricating oil comes in direct contact with the refrigerant. Normally labyrinth type oil seals are used on the rotor shaft to minimize the leakage of lubricating oil to the refrigerant side. Sometimes oil heaters may be required to avoid excessive dilution of lubricating oil during the plant shutdown.

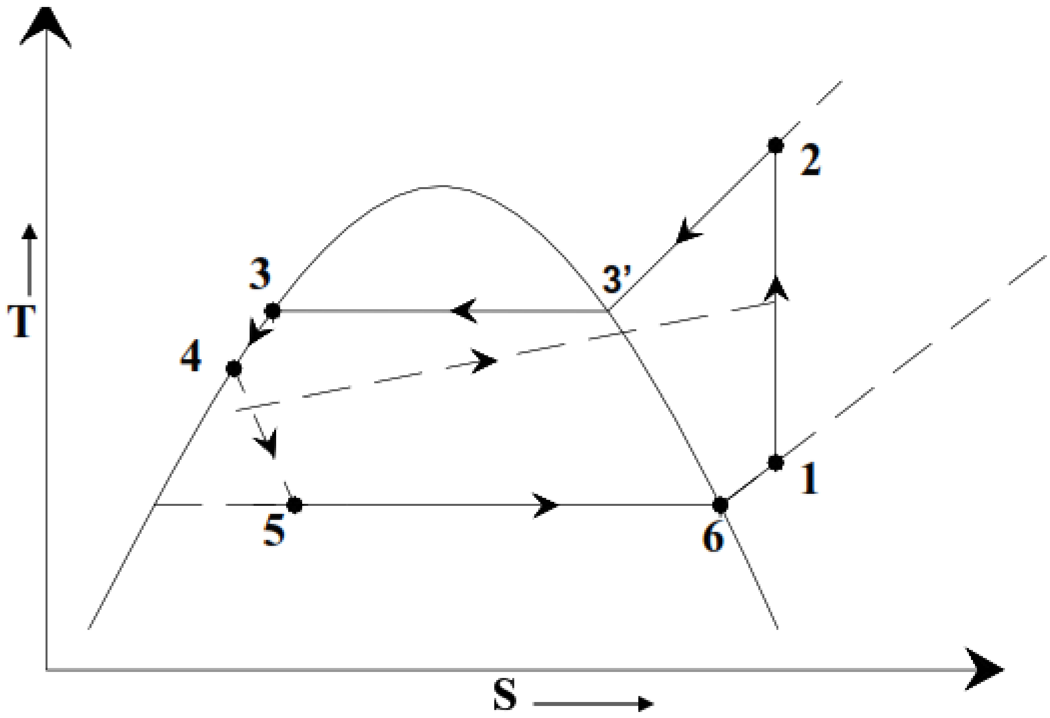
Commercially both hermetic as well as open type centrifugal compressors are available. Open type compressors are driven by electric motors, internal combustion engines (using a wide variety of fuels) or even steam turbines.

CONDENSORS:

Introduction to condensers

Condensers and evaporators are basically heat exchangers in which the refrigerant undergoes a phase change. Next to compressors, proper design and selection of condensers and evaporators is very important for satisfactory performance of any refrigeration system. Since both condensers and evaporators are essentially heat exchangers, they have many things in common as far as the design of these components is concerned. However, differences exist as far as the heat transfer phenomena is concerned. In condensers the refrigerant vapour condenses by rejecting heat to an external fluid, which acts as a heat sink. Normally, the external fluid does not undergo any phase change, except in some special cases such as in cascade condensers, where the external fluid (another refrigerant) evaporates. In evaporators, the liquid refrigerant evaporates by extracting heat from an external fluid (low temperature heat source). The external fluid may not undergo phase change, for example if the system is used for sensibly cooling water, air or some other fluid. There are many refrigeration and air conditioning applications, where the external fluid also undergoes phase change. For example, in a typical summer air conditioning system, the moist air is dehumidified by condensing water vapour and then, removing the condensed liquid water. In many low temperature refrigeration applications freezing or frosting of evaporators takes place. These aspects have to be considered while designing condensers and evaporators.

Condenser is an important component of any refrigeration system. In a typical refrigerant condenser, the refrigerant enters the condenser in a superheated state. It is first de-superheated and then condensed by rejecting heat to an external medium. The refrigerant may leave the condenser as a saturated or a sub-cooled liquid, depending upon the temperature of the external medium and design of the condenser. Figure shows the variation of refrigeration cycle on T-s diagram. In the figure, the heat rejection process is represented by 2-3’-3-4. The temperature profile of the external fluid, which is assumed to undergo only sensible heat transfer, is shown by dashed line. It can be seen that process 2-3’ is a de-superheating process, during which the refrigerant is cooled sensibly from a temperature T2 to the saturation temperature corresponding condensing pressure, T3’. Process 3’-3 is the condensation process, during which the temperature of the refrigerant remains constant as it undergoes a phase change process. In actual refrigeration systems with a finite pressure drop in the condenser or in a system using a zeotropic refrigerant mixture, the temperature of the refrigerant changes during the condensation process also. However, at present for simplicity, it is assumed that the refrigerant used is a pure refrigerant (or an azeotropic mixture) and the condenser pressure remains constant during the condensation process. Process 3-4 is a sensible, sub cooling process, during which the refrigerant temperature drops from T3 to T4 .



Refrigeration cycle on T-s diagram

Classification of condensers:

Based on the external fluid, condensers can be classified as:

a) Air cooled condensers

b) Water cooled condensers, and

c) Evaporative condensers

Air-cooled condensers:

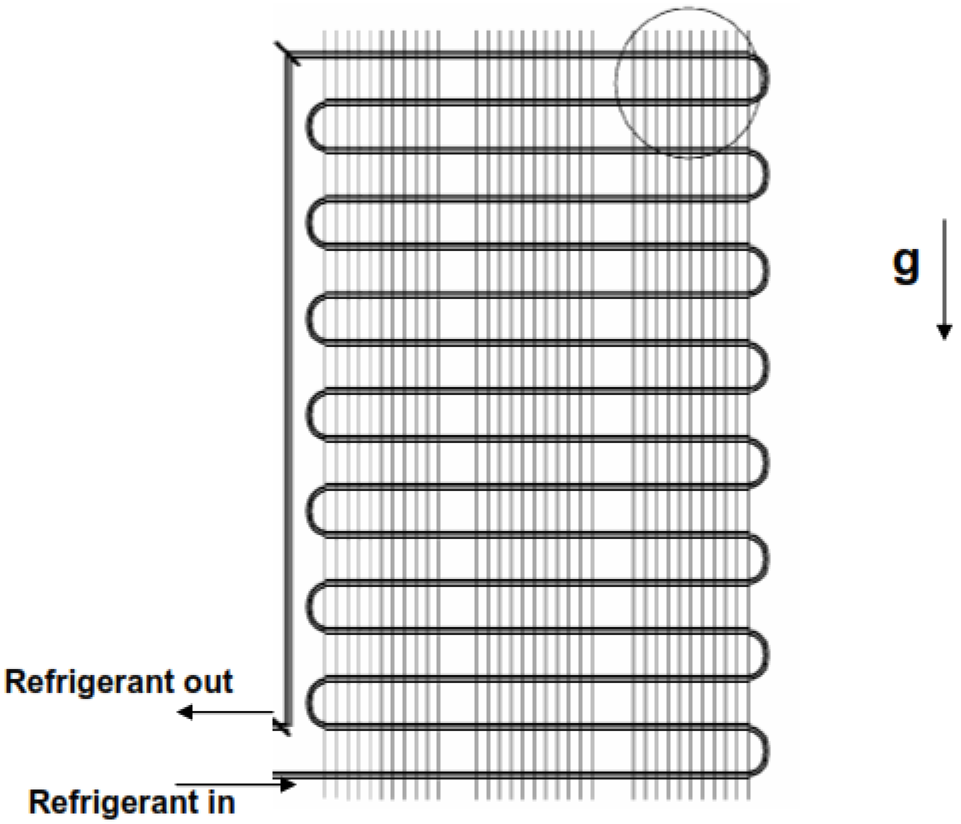
As the name implies, in air-cooled condensers air is the external fluid, i.e., the refrigerant rejects heat to air flowing over the condenser. Air-cooled condensers can be further classified into natural convection type or forced convection type.

Natural convection type:

In natural convection type, heat transfer from the condenser is by buoyancy induced natural convection and radiation. Since the flow rate of air is small and the radiation heat transfer is also not very high, the combined heat transfer coefficient in these condensers is small. As a result a relatively large condensing surface is required to reject a given amount of heat. Hence these condensers are used for small capacity refrigeration systems like household refrigerators and freezers. The natural convection type condensers are either plate surface type or finned tube type. In plate surface type condensers used in small refrigerators and freezers, the refrigerant carrying tubes are attached to the outer walls of the refrigerator. The whole body of the refrigerator (except the door) acts like a fin. Insulation is provided between the outer cover that acts like fin and the inner plastic cover of the refrigerator. It is for this reason that outer body of the refrigerator is always warm. Since the surface is warm, the problem of moisture condensation on the walls of the refrigerator does not arise in these systems. These condensers are sometimes called as flat back condensers.

The finned type condensers are mounted either below the refrigerator at an angle or on the backside of the refrigerator. In case, it is mounted below, then the warm air rises up and to assist it an air envelope is formed by providing a jacket on backside of the refrigerator. The fin spacing is kept large to minimize the effect of fouling by dust and to allow air to flow freely with little resistance.

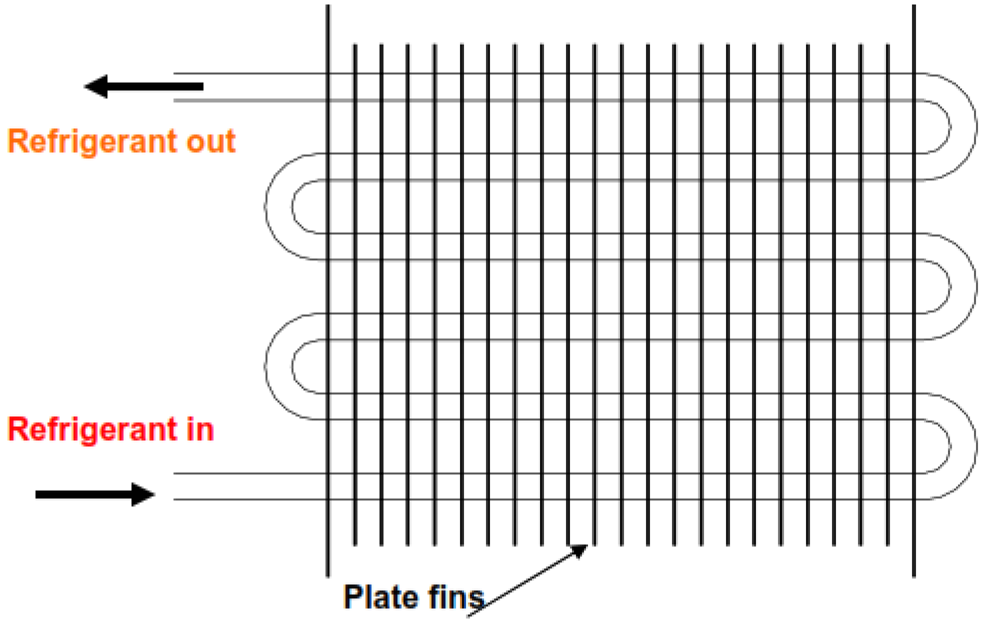
In the older designs, the condenser tube (in serpentine form) was attached to a plate and the plate was mounted on the backside of the refrigerator. The plate acted like a fin and warm air rose up along it. In another common design, thin wires are welded to the serpentine tube coil. The wires act like fins for increased heat transfer area. Figure shows the schematic of a wire-and-tube type condenser commonly used in domestic refrigerators. Regardless of the type, refrigerators employing natural convection condenser should be located in such a way that air can flow freely over the condenser surface.



Schematic of a wire-and-tube type condenser used in small refrigeration systems

Forced convection type:

In forced convection type condensers, the circulation of air over the condenser surface is maintained by using a fan or a blower. These condensers normally use fins on air-side for good heat transfer. The fins can be either plate type or annular type. Figure 22.3 shows the schematic of a plate-fin type condenser. Forced convection type condensers are commonly used in window air conditioners, water coolers and packaged air conditioning plants. These are either chassis mounted or remote mounted. In chassis mounted type, the compressor, induction motor, condenser with condenser fan, accumulator, HP/LP cut- out switch and pressure gauges are mounted on a single chassis. It is called condensing unit of rated capacity. The components are matched to condense the required mass flow rate of refrigerant to meet the rated cooling capacity. The remote mounted type, is either vertical or roof mounted horizontal type. Typically the air velocity varies between 2 m/s to 3.5 m/s for economic design with airflow rates of 12 to 20 cmm per ton of refrigeration (TR). The air specific heat is 1.005 kJ/kg-K and density is 1.2 kg/m3.. Therefore for 1 TR the temperature rise Δta = 3.5167/(1.2x1.005 x 16/60) = 10.9oC for average air flow rate of 16 cm. Hence, the air temperature rises by 10 to 15oC as compared to 3 to 6oC for water in water cooled condensers.



Forced convection, plate fin-and-tube type condenser

The area of the condenser seen from outside in the airflow direction is called face area. The velocity at the face is called face velocity. This is given by the volume flow rate divided by the face area. The face velocity is usually around 2m/s to 3.5 m/s to limit the pressure drop due to frictional resistance. The coils ofthe tube in the flow direction are called rows. A condenser may have two to eight rows of the tubes carrying the refrigerant. The moist air flows over the fins while the refrigerant flows inside the tubes. The fins are usually of aluminum and tubes are made of copper. Holes of diameter slightly less than the tube diameter are punched in the plates and plates are slid over the tube bank. Then the copper tubes are pressurized which expands the tubes and makes a good thermal contact between the tube and fins. This process is also known as bulleting. For ammonia condensers mild steel tubes with mild steel fins are used. In this case the fins are either welded or galvanizing is done to make a good thermal contact between fin and tube. In case of ammonia, annular crimpled spiral fins are also used over individual tubes instead of flat-plate fins. In finned tube heat exchangers the fin spacing may vary from 3 to 7 fins per cm. The secondary surface area is 10 to 30 times the bare pipe area hence; the finned coils are very compact and have smaller weight.

Water Cooled Condensers:

In water cooled condensers water is the external fluid. Depending upon the construction, water cooled condensers can be further classified into:

1. Double pipe or tube-in-tube type

2. Shell-and-coil type

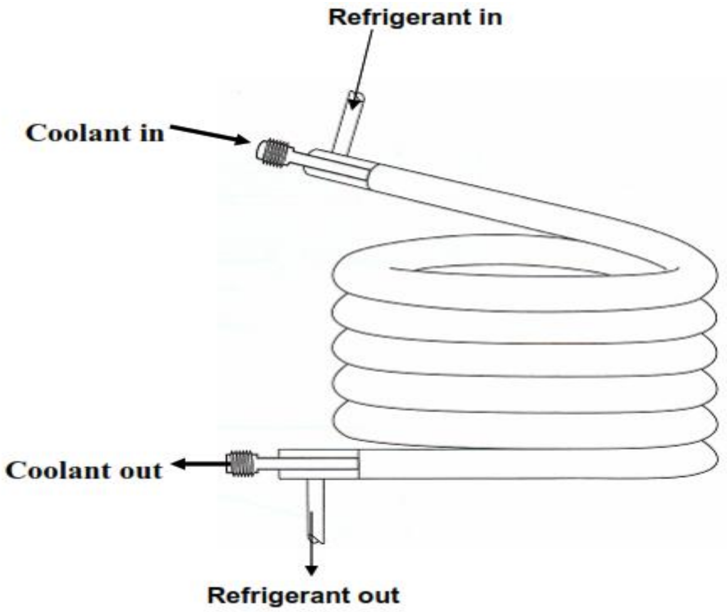
3. Shell-and-tube type

Double Pipe or tube-in-tube type:

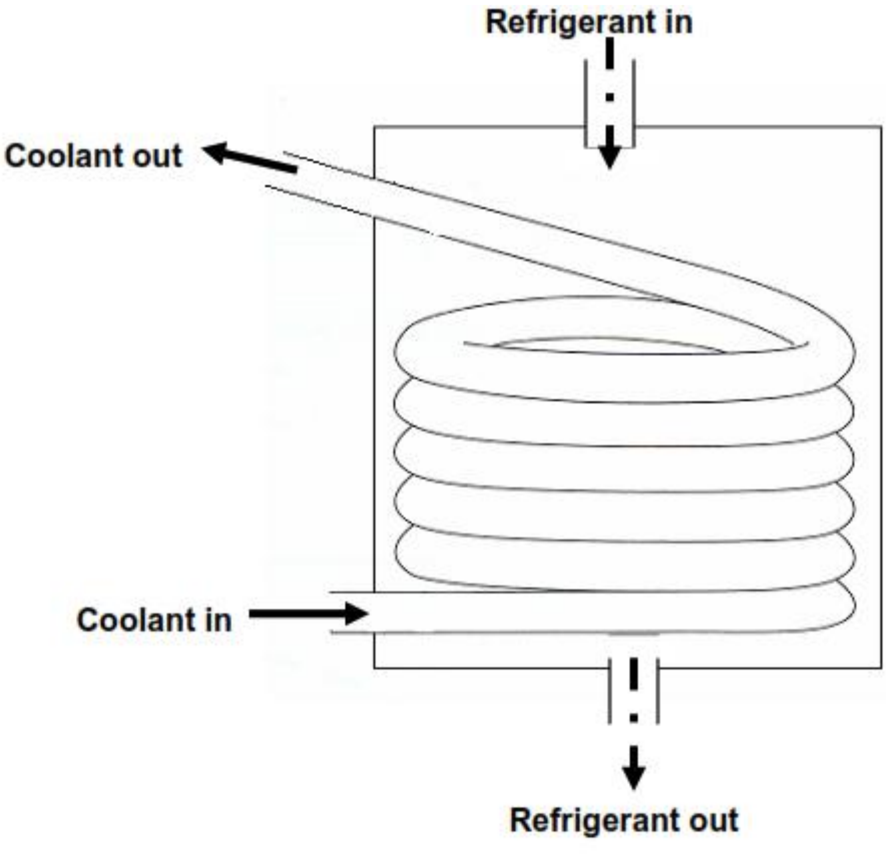
Double pipe condensers are normally used up to 10 TR capacity. Figure shows the schematic of a double pipe type condenser. As shown in the figure, in these condensers the cold water flows through the inner tube, while the refrigerant flows through the annulus in counter flow. Headers are used at both the ends to make the length of the condenser small and reduce pressure drop. The refrigerant in the annulus rejects a part of its heat to the surroundings by free convection and radiation. The heat transfer coefficient is usually low because of poor liquid refrigerant drainage if the tubes are long.

Shell-and-coil type:

These condensers are used in systems up to 50 TR capacity. The water flows through multiple coils, which may have fins to increase the heat transfer coefficient. The refrigerant flows through the shell. In smaller capacity condensers, refrigerant flows through coils while water flows through the shell. Figure shows a shell-and-coil type condenser. When water flows through the coils, cleaning is done by circulating suitable chemicals through the coils.



Double pipe (tube-in-tube) type condenser

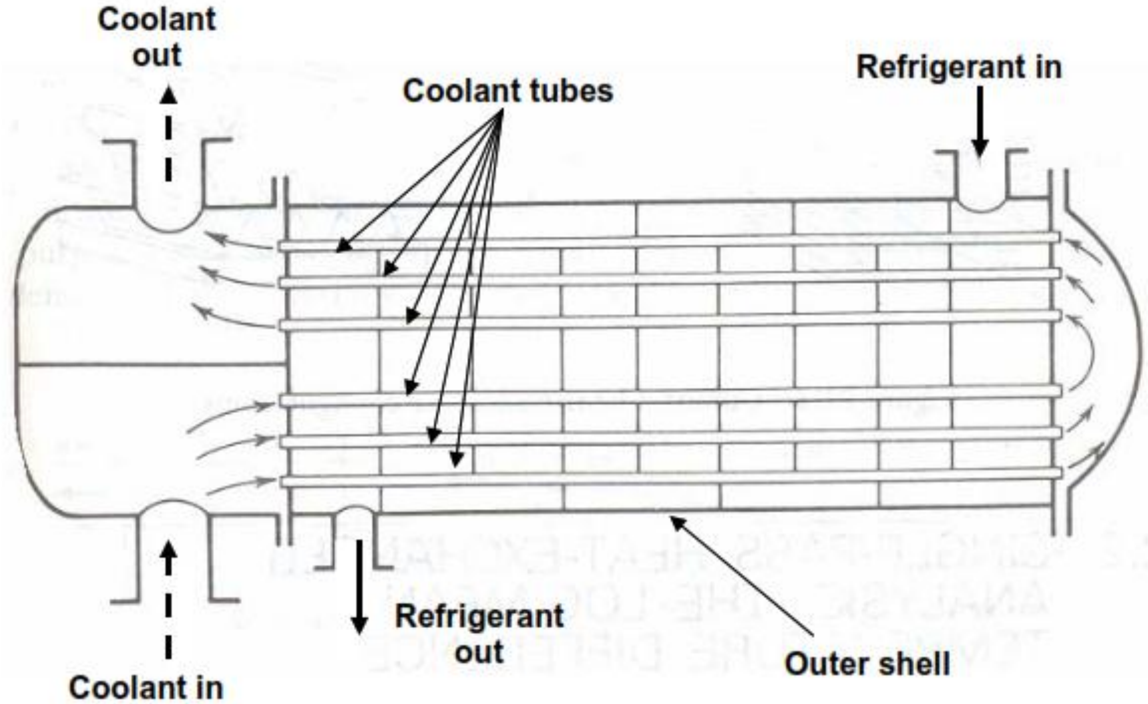


Shell-and-coil type condenser

Shell-and-tube type:

This is the most common type of condenser used in systems from 2 TR upto thousands of TR capacity. In these condensers the refrigerant flows through the shell while water flows through the tubes in single to four passes. The condensed refrigerant collects at the bottom of the shell. The coldest water contacts the liquid refrigerant so that some subcooling can also be obtained. The liquid refrigerant is drained from the bottom to the receiver. There might be a vent connecting the receiver to the condenser for smooth drainage of liquid refrigerant. The shell also acts as a receiver. Further the refrigerant also rejects heat to the surroundings from the shell. The most common type is horizontal shell type. A schematic diagram of horizontal shell-and-tube type condenser is shown in Fig.

Vertical shell-and-tube type condensers are usually used with ammonia in large capacity systems so that cleaning of the tubes is possible from top while the plant is running.

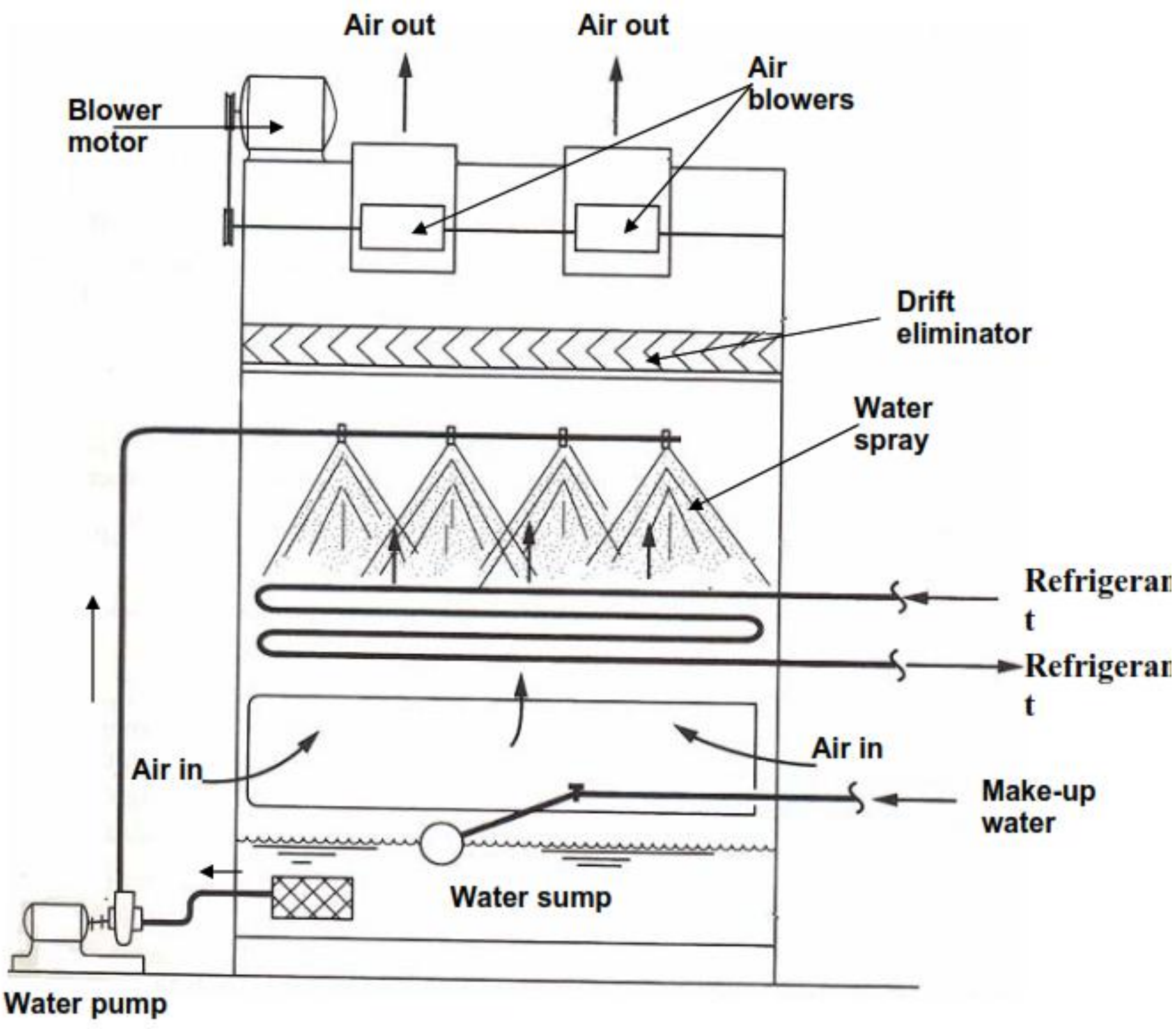


A two-pass, shell-and-tube type condenser

Evaporative condensers:

In evaporative condensers, both air and water are used to extract heat from the condensing refrigerant. Figure shows the schematic of an evaporative condenser. Evaporative condensers combine the features of a cooling tower and water-cooled condenser in a single unit. In these condensers, the water is sprayed from top part on a bank of tubes carrying the refrigerant and air is induced upwards. There is a thin water film around the condenser tubes from which evaporative cooling takes place. The heat transfer coefficient for evaporative cooling is very large. Hence, the refrigeration system can be operated at low condensing temperatures (about 11 to 13 K above the wet bulb temperature of air). The water spray counter current to the airflow acts as cooling tower. The role of air is primarily to increase the rate of evaporation of water.

The required air flow rates are in the range of 350 to 500 m3/h per TR of refrigeration capacity.



Schematic of an evaporative condenser

Evaporative condensers are used in medium to large capacity systems. These are normally cheaper compared to water cooled condensers, which require a separate cooling tower. Evaporative condensers are used in places where water is scarce. Since water is used in a closed loop, only a small part of the water evaporates. Make-up water is supplied to take care of the evaporative loss. The water consumption is typically very low, about 5 percent of an equivalent water cooled condenser with a cooling tower. However, since condenser has to be kept outside, this type of condenser requires a longer length of refrigerant tubing, which calls for larger refrigerant inventory and higher pressure drops. Since the condenser is kept outside, to prevent the water from freezing, when outside temperatures are very low, a heater is placed in the water tank. When outside temperatures are very low it is possible to switch-off the water pump and run only the blowers, so that the condenser acts as an air cooled condenser.

Another simple form of condenser used normally in older type cold storages is called as atmospheric condenser. The principle of the atmospheric condenser is similar to evaporative condenser, with a difference that the air flow over the condenser takes place by natural means as no fans or blowers are used. A spray system sprays water over condenser tubes. Heat transfer outside the tubes takes by both sensible cooling and evaporation, as a result the external heat transfer coefficient is relatively large. The condenser pipes are normally large, and they can be either horizontal or vertical. Though these condensers are effective and economical they are being replaced with other types of condensers due to the problems such as algae formation on condenser tubes, uncertainity due to external air circulation etc.

Evaporators

An evaporator, like condenser is also a heat exchanger. In an evaporator, the refrigerant boils or evaporates and in doing so absorbs heat from the substance being refrigerated. The name evaporator refers to the evaporation process occurring in the heat exchanger.

Classification

There are several ways of classifying the evaporators depending upon the heat transfer process or refrigerant flow or condition of heat transfer surface.

Natural and Forced Convection Type

The evaporator may be classified as natural convection type or forced convection type. In forced convection type, a fan or a pump is used to circulate the fluid being refrigerated and make it flow over the heat transfer surface, which is cooled by evaporation of refrigerant. In natural convection type, the fluid being cooled flows due to natural convection currents arising out of density difference caused by temperature difference. The refrigerant boils inside tubes and evaporator is located at the top. The temperature of fluid, which is cooled by it,decreases and its density increases. It moves downwards due to its higher density and the warm fluid rises up to replace it.

Refrigerant Flow Inside or Outside Tubes

The heat transfer phenomenon during boiling inside and outside tubes is very different; hence, evaporators are classified as those with flow inside and outside tubes.

In natural convection type evaporators and some other evaporators, the refrigerant is confined and boils inside the tubes while the fluid being refrigerated flows over the tubes.

The direct expansion coil where the air is directly cooled in contact with the tubes cooled by refrigerant boiling inside is an example of forced convection type of evaporator where refrigerant is confined inside the tubes.

In many forced convection type evaporators, the refrigerant is kept in a shell and the fluid being chilled is carried in tubes, which are immersed in refrigerant. Shell and tube type brine and water chillers are mainly of this kind.

Flooded and Dry Type

The third classification is flooded type and dry type. Evaporator is said to be flooded type if liquid refrigerant covers the entire heat transfer surface. This type of evaporator uses a float type of expansion valve. An evaporator is called dry type when a portion of the evaporator is used for superheating the refrigerant vapour after its evaporation.

Natural Convection type evaporator coils

These are mainly used in domestic refrigerators and cold storages. When used in cold storages, long lengths of bare or finned pipes are mounted near the ceiling or along the high sidewalls of the cold storages. The refrigerant from expansion valve is fed to these tubes. The liquid refrigerant evaporates inside the tubes and cools the air whose density increases. The high-density air flows downwards through the product in the cold storage. The air becomes warm by the time it reaches the floor as heat is transferred from the product to air. Some free area like a passage is provided for warm air to rise up. The same passage is used for loading and unloading the product into the cold storage.

The advantages of such natural convection coils are that the coil takes no floor space and it also requires low maintenance cost. It can operate for long periods without defrosting the ice formed on it and it does not require special skill to fabricate it. Defrosting can be done easily (e.g. by scraping) even when the plant is running. These are usually welded at site. However, the disadvantage is that natural convection heat transfer coefficient is very small hence very long lengths are required which may cause excessive refrigerant side pressure drops unless parallel paths are used. The large length requires a larger quantity of refrigerant than the forced convection coils. The large quantity of refrigerant increases the time required for defrosting, since before the defrosting can start all the liquid refrigerant has to be pumped out of the evaporator tubes. The pressure balancing also takes long time if the system trips or is to be restarted after load shedding. Natural convection coils are very useful when low air velocities and minimum dehumidification of the product is required. Household refrigerators, display cases, walk-in-coolers, reach-in refrigerators and obviously large cold storages are few of its applications. Sufficient space should be provided between the evaporator and ceiling to permit the air circulation over the top of the coil.

Baffles are provided to separate the warm air and cold air plumes. Single ceiling mounted is used for rooms of width less than 2.5 m. For rooms with larger widths more evaporator coils are used. The refrigerant tubes are made of steel or copper. Steel tubes are used for ammonia and in large capacity systems.

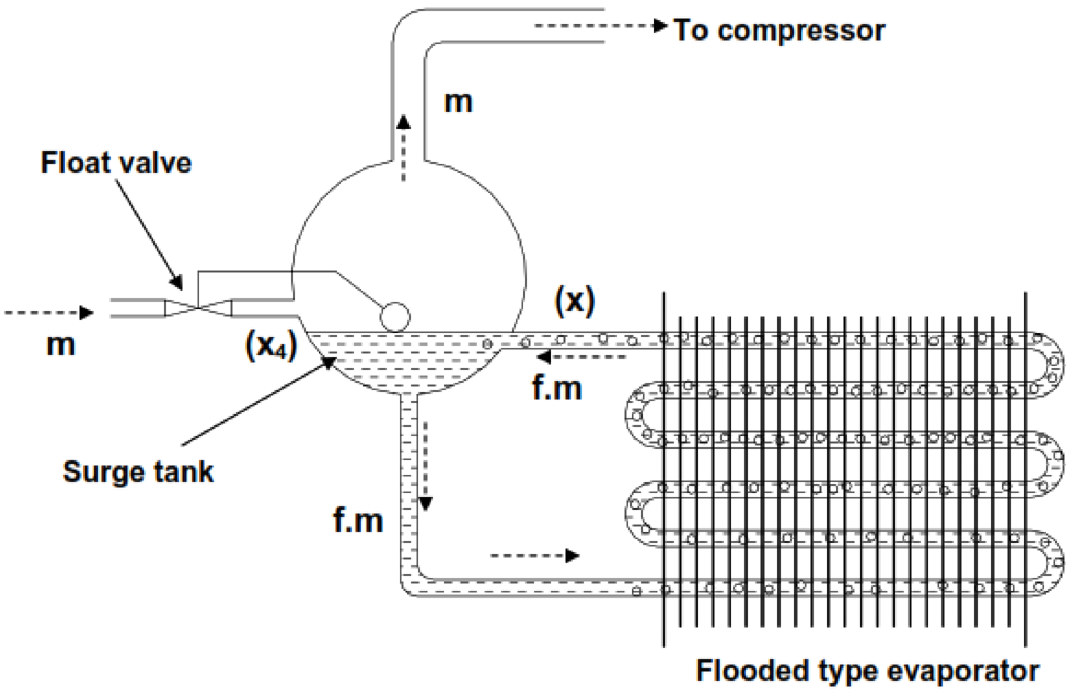
Flooded Evaporator

This is typically used in large ammonia systems. The refrigerant enters a surge drum through a float type expansion valve. The compressor directly draws the flash vapour formed during expansion. This vapour does not take part in refrigeration hence its removal makes the evaporator more compact and pressured drop due to this is also avoided. The liquid refrigerant enters the evaporator from the bottom of the surge drum. This boils inside the tubes as heat is absorbed. The mixture of liquid and vapour bubbles rises up along the evaporator tubes. The vapour is separated as it enters the surge drum. The remaining unevaporated liquid circulates again in the tubes along with the constant supply of liquid refrigerant from the expansion valve. The mass flow rate in the evaporator tubes is m.f where m is the mass flow rate through the expansion valve and to the compressor. The term f is called recirculation factor. Let x4 be the quality of mixture after the expansion valve and x be the quality of mixture after boiling in the tubes as shown in Figure. In steady state mass flow rate from expansion valve is same as the mass flow rate to the compressor hence mass conservation gives

$$x_4 . \dot{m} + x.f.\dot{m} = \dot{m}$$

$$\therefore f = \frac{(1 - x_4)}{x}$$

For x4 = x = 0.25, for example, the circulation factor is 3, that is mass flow rate through the evaporator is three times that through the compressor. Since, liquid refrigerant is in contact with whole of evaporator surface, the refrigerant side heat transfer coefficient will be very high. Sometimes a liquid refrigerant pump may also be used to further increase the heat transfer coefficient. The lubricating oil tends to accumulate in the flooded evaporator hence an effective oil separator must be used immediately after the compressor.



Schematic of a flooded evaporator

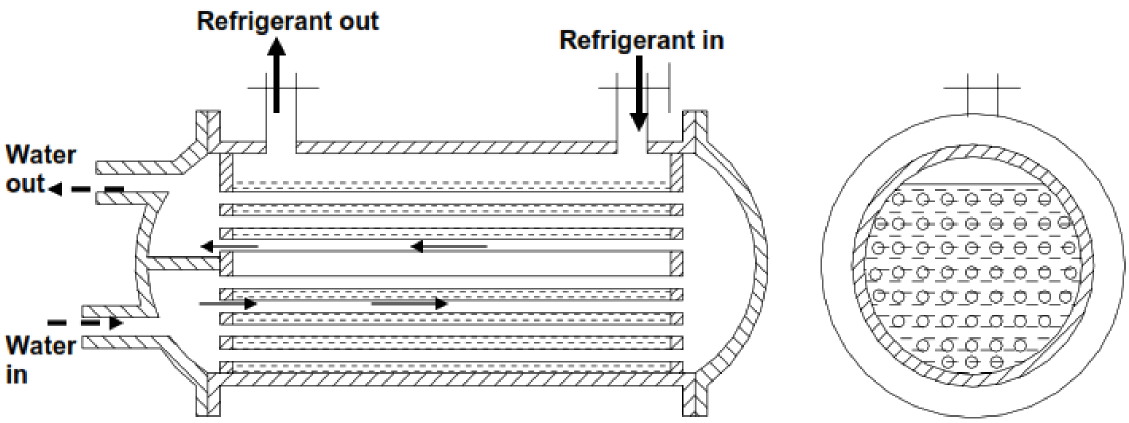
Shell-and-Tube Liquid Chillers

The shell-and-tube type evaporators are very efficient and require minimum floor space and headspace. These are easy to maintain, hence they are very widely used in medium to large capacity refrigeration systems. The shell-and-tube evaporators can be either dry type or flooded type. As the name implies, a shell-and-tube evaporator consists of a shell and a large number of straight tubes arranged parallel to each other. In dry expansion type, the refrigerant flows through the tubes while in flooded type the refrigerant is in the shell. A pump circulates the chilled water or brine. The shell diameters range from 150 mm to 1.5 m. The number of tubes may be less than 50 to several thousands and length may be between 1.5 m to 6 m. Steel tubes are used with ammonia while copper tubes are used with freons. Ammonia has a very high heat transfer coefficient while freons have rather poor heat transfer coefficient hence fins are used on the refrigerant side. Dry expansion type uses fins inside the tubes while flooded type uses fins outside the tube. Dry-expansion type requires less charge of refrigerant and have positive lubricating oil return. These are used for small and medium capacity refrigeration plants with capacity ranging from 2 TR to 350 TR. The flooded type evaporators are available in larger capacities ranging from 10 TR to thousands of TR.

Flooded Type Shell-and-Tube Evaporator

Figure shows a flooded type of shell and tube type liquid chiller where the liquid (usually brine or water) to be chilled flows through the tubes in double pass just like that in shell and tube condenser. The refrigerant is fed through a float valve, which maintains a constant level of liquid refrigerant in the shell. The shell is not filled entirely with tubes as shown in the end view of Fig. This is done to maintain liquid refrigerant level below the top of the shell so that liquid droplets settle down due to gravity and are not carried by the vapour leaving the shell. If the shell is completely filled with tubes, then a surge drum is provided after the evaporator to collect the liquid refrigerant.

Shell-and-tube evaporators can be either single pass type or multipass type. In multipass type, the chilled liquid changes direction in the heads. Shell and-tube evaporators are available in vertical design also. Compared to horizontal type, vertical shell-and-tube type evaporators require less floor area. The chilled water enters from the top and flows downwards due to gravity and is then taken to a pump, which circulates it to the refrigeration load. At the inlet to tubes at the top a special arrangement introduces swirling action to increase the heat transfer coefficient.



Schematic of a flooded type shell-and-tube evaporator

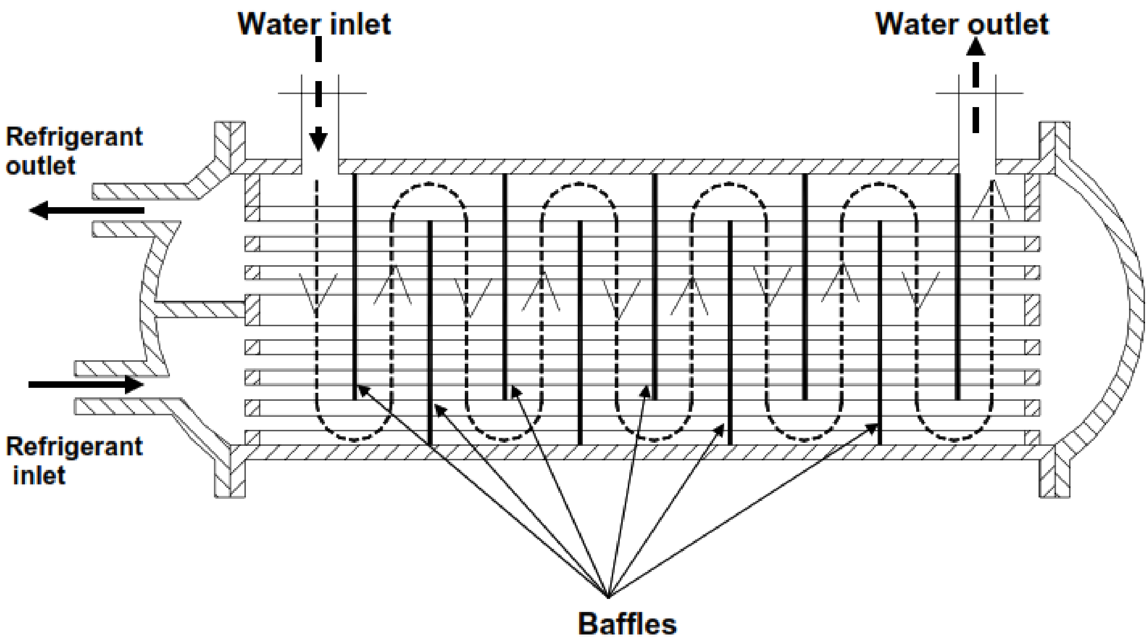
Direct expansion type, Shell-and-Tube Evaporator

Figure 23.3 shows a liquid chiller with refrigerant flowing through the tubes and water flowing through the shell. A thermostatic expansion valve feeds the refrigerant into the tubes through the cover on the left. It may flow in several passes through the dividers in the covers of the shell on either side. The liquid to be chilled flows through the shell around the baffles. The presence of baffles turns the flow around creating some turbulence thereby increasing the heat transfer coefficient. Baffles also prevent the short-circuiting of the fluid flowing in the shell. This evaporator is of dry type since some of the tubes superheat the vapour. To maintain the chilled liquid velocity so as to obtain good heat transfer coefficient, the length and the spacing of segmental baffles is varied. Widely spaced baffles are used when the flow rate is high or the liquid viscosity is high. The number of passes on the refrigerant side are decided by the partitions on the heads on the two sides of the heat exchanger. Some times more than one circuit is also provided. Changing the heads can change the number of passes. It depends upon the chiller load and the refrigerant velocity to be maintained in the heat exchanger.

Shell-and-Coil type evaporator

These are of smaller capacity than the shell and tube chillers. These are made of one or more spiral shaped bare tube coils enclosed in a welded steel shell. It is usually dry-expansion type with the refrigerant flowing in the tube and chilled liquid in the shell. In some cases the chiller operates in flooded mode also with refrigerant in the shell and chilled water flowing thorough the spiral tube. The water in the shell gives a large amount of thermal storage capacity called hold-up capacity. This type is good for small but highly infrequent peak loads. It is used for cooling drinking water in stainless steel tanks to maintain sanitary conditions. It is also used in bakeries and photographic laboratories.

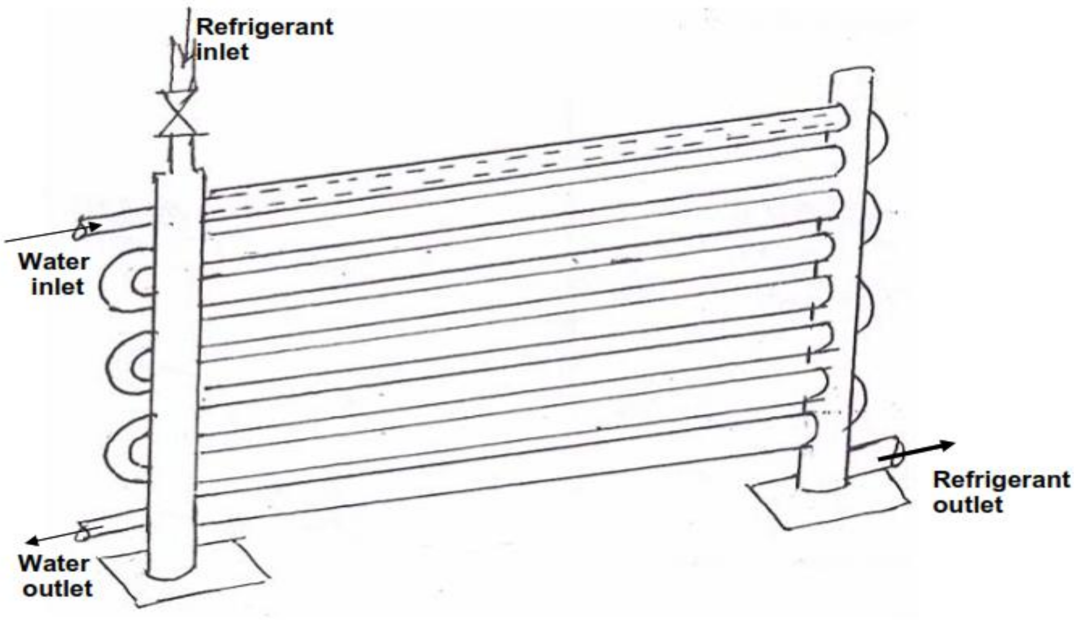
When the refrigerant is in the shell that is in flooded mode it is called instantaneous liquid chiller. This type does not have thermal storage capacity, the liquid must be instantaneously chilled whenever required. In the event of freeze up the water freezes in the tube, which causes bursting of the tubes since water expands upon freezing. When water is in the shell there is enough space for expansion of water if the freezing occurs. The flooded types are not recommended for any application where the temperature of chilled liquid may be below 3^{o}C.



Schematic of a direct expansion type, Shell-and-Tube evaporator

Double pipe type evaporator

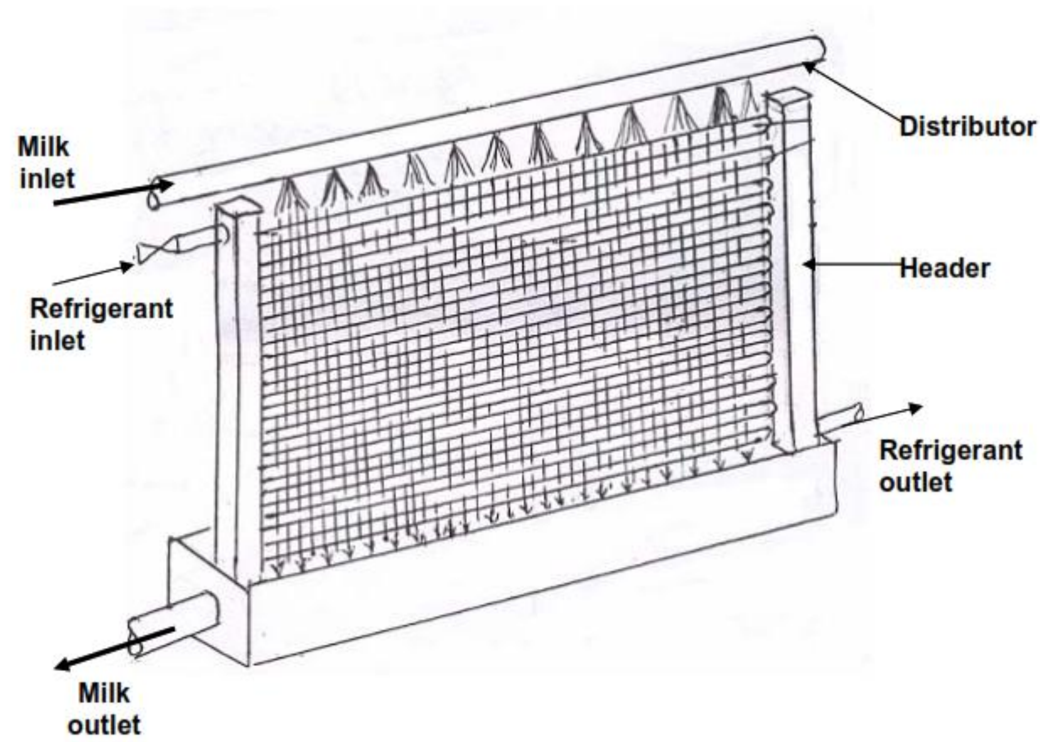
This consists of two concentric tubes, the refrigerant flows through the annular passage while the liquid being chilled flows through the inner tube in counter flow. One design is shown in Fig. 23.4 in which the outer horizontal tubes are welded to vertical header tubes on either side. The inner tubes pass through the headers and are connected together by 180o bends. The refrigerant side is welded hence there is minimum possibility of leakage of refrigerant. These may be used in flooded as well as dry mode. This requires more space than other designs. Shorter tubes and counter flow gives good heat transfer coefficient. It has to be insulated from outside since the refrigerant flows in the outer annulus which may be exposed to surroundings if insulation is not provided.



Schematic of a double pipe type evaporator

Baudelot type evaporators

This type of evaporator consists of a large number of horizontal pipes stacked one on top of other and connected together to by headers to make single or multiple circuits. The refrigerant is circulated inside the tubes either in flooded or dry mode. The liquid to be chilled flows in a thin layer over the outer surface of the tubes. The liquid flows down by gravity from distributor pipe located on top of the horizontal tubes as shown in Figure The liquid to be chilled is open to atmosphere, that is, it is at atmospheric pressure and its aeration may take place during cooling. This is widely used for cooling milk, wine and for chilling water for carbonation in bottling plants. The liquid can be chilled very close to its freezing temperature since freezing outside the tubes will not damage the tubes. Another advantage is that the refrigerant circuit can be split into several parts, which permit a part of the cooling done by cold water and then chilling by the refrigerant.

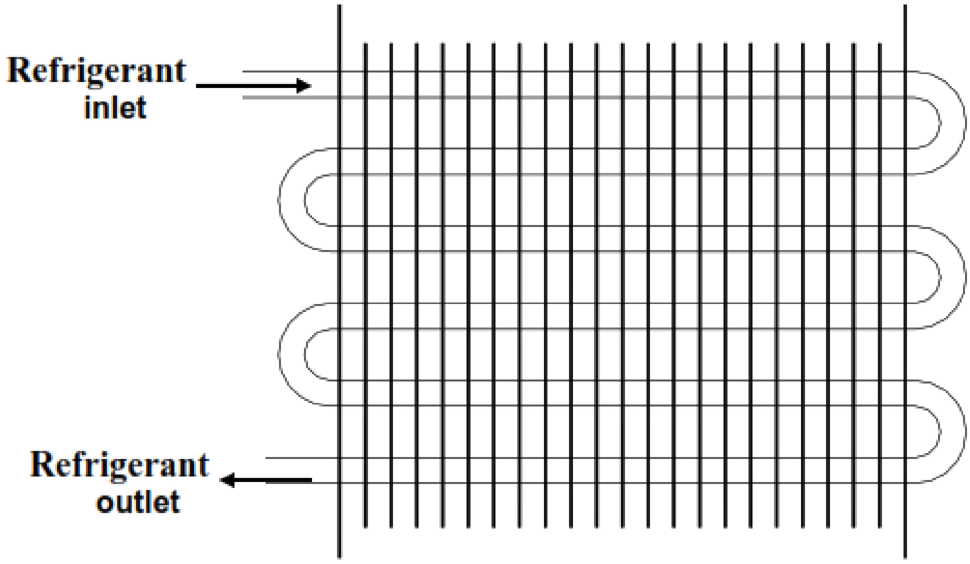


Schematic of a Baudelot type evaporator for chilling of milk

Direct expansion fin-and-tube type

These evaporators are used for cooling and dehumidifying the air directly by the refrigerant flowing in the tubes. Similar to fin-and-tube type condensers, these evaporator consists of coils placed in a number of rows with fins mounted on it to increase the heat transfer area. Various fin arrangements are used. Tubes with individual spiral straight fins or crimpled fins welded to it are used in some applications like ammonia. Plate fins accommodating a number of rows are used in air conditioning applications with ammonia as well as synthetic refrigerants such as fluorocarbon based refrigerants.

The liquid refrigerant enters from top through a thermostatic expansion valve as shown in Fig.. This arrangement makes the oil return to compressor better rather than feeding refrigerant from the bottom of the coil. When evaporator is close to the compressor, a direct expansion coil is used since the refrigerant lines are short, refrigerant leakage will be less and pressure drop is small. If the air-cooling is required away from the compressor, it is preferable to chill water and pump it to air-cooling coil to reduce the possibility of refrigerant leakage and excessive refrigerant pressure drop, which reduces the COP.



Schematic of a direct expansion fin-and-tube type

The fin spacing is kept large for larger tubes and small for smaller tubes. 50 to 500 fins per meter length of the tube are used in heat exchangers. In evaporators, the atmospheric water vapour condenses on the fins and tubes when the metal temperature is lower than dew point temperature. On the other hand frost may form on the tubes if the surface temperature is less than 0 C.Hence for low temperature coils a wide spacing with about 80 to 200 fins per m is used to avoid restriction of flow passage due to frost formation. In air-conditioning o applications a typical fin spacing of 1.8 mm is used. Addition of fins beyond a

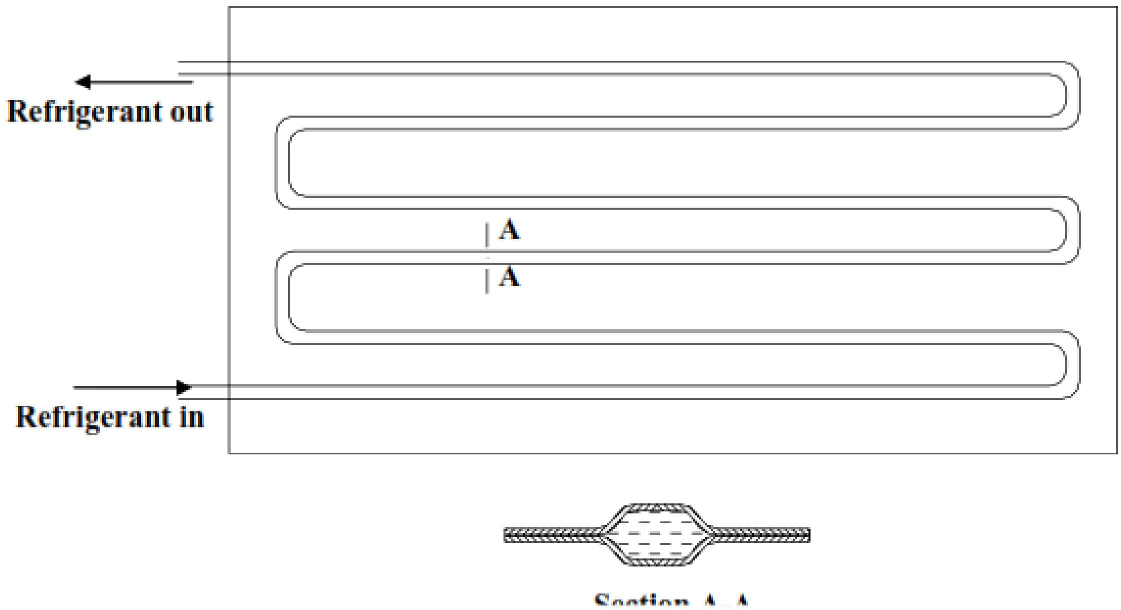
certain value will not increase the capacity of evaporator by restricting the airflow. The frost layer has a poor thermal conductivity hence it decreases the overall heat transfer coefficient apart from restricting the flow. Therefore, for applications in freezers below 0oC, frequent defrosting of the evaporator is required.

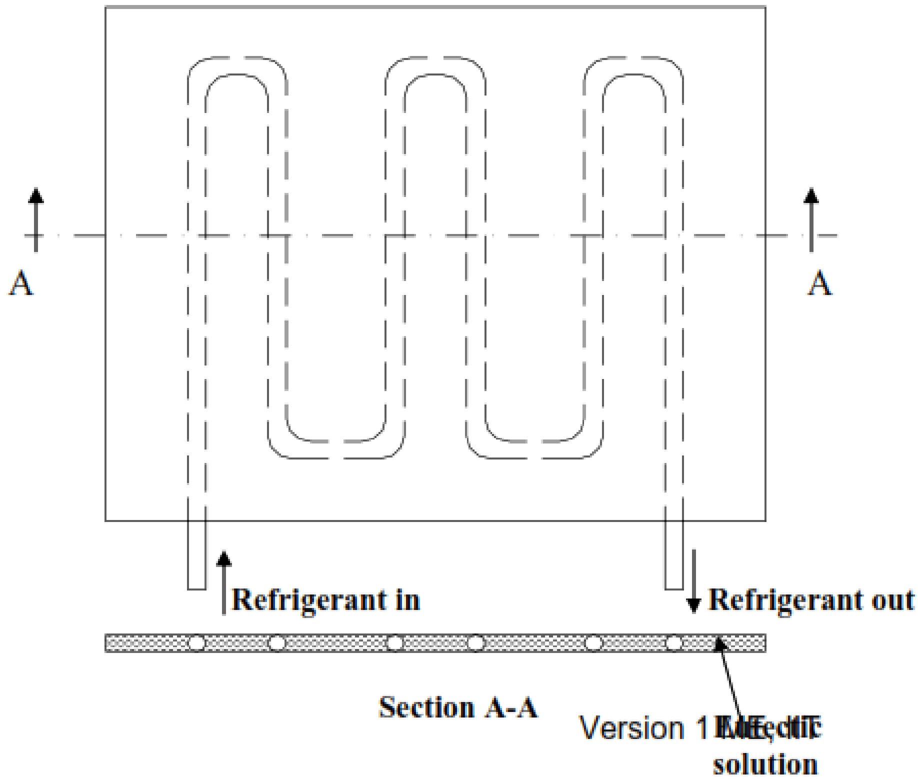
Plate Surface Evaporators

These are also called bonded plate or roll-bond type evaporators. Two flat sheets of metal (usually aluminum) are embossed in such a manner that when these are welded together, the embossed portion of the two plates makes a passage for refrigerant to flow. This type is used in household refrigerators.

In another type of plate surface evaporator, a serpentine tube is placed between two metal plates such that plates press on to the tube. The edges of the plates are welded together. The space between the plates is either filled with a eutectic solution or evacuated. The vacuum between the plates and atmospheric pressure outside, presses the plates on to the refrigerant carrying tubes making a very good contact between them. If eutectic solution is filled into the void space, this also makes a good thermal contact between refrigerant carrying tubes and

the plates. Further, it provides an additional thermal storage capacity during offcycle and load shedding to maintain a uniform temperature. These evaporators are commonly used in refrigerated trucks. Figure shows an embedded tube, plate surface evaporator.





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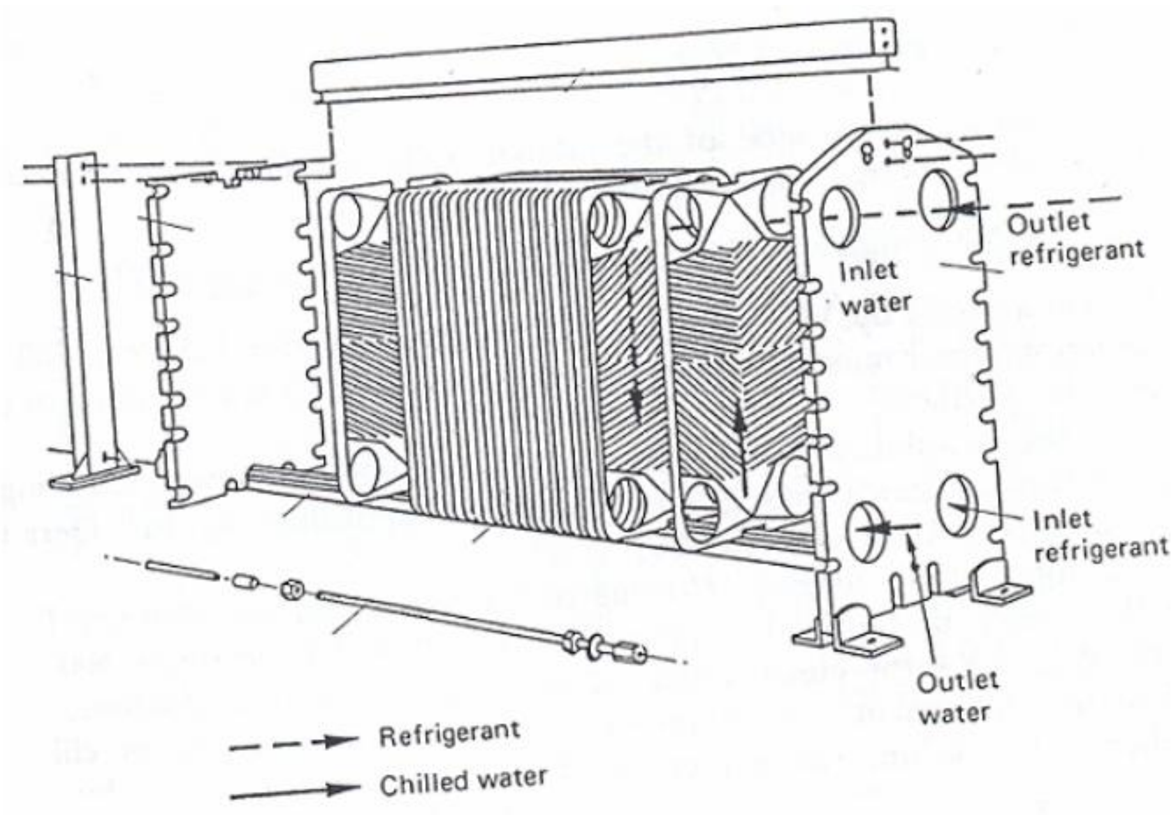
chematic of an embedded tube, plate surface evaporator

Plate type evaporators:

Plate type evaporators are used when a close temperature approach (0.5 K or less) between the boiling refrigerant and the fluid being chilled is required. These evaporators are widely used in dairy plants for chilling milk, in breweries for chilling beer. These evaporators consist of a series of plates (normally made of stainless steel) between which alternately the milk or beer to be cooled and refrigerant flow in counterflow direction. The overall heat transfer coefficient of these plate type evaporators is very high (as high as 4500 W/m K in case of ammonia/water and 3000 W/m2 K in case of R 22/water). In addition they also require very less refrigerant inventory for the same capacity (about 10 percent or even less than that of shell-and-tube type evaporators). Another important advantage when used in dairy plants and breweries is that, it is very easy to clean the evaporator and assemble it back as and when required. The capacity can be increased or decreased very easily by adding or removing plates. Hence these evaporators are finding widespread use in a variety of applications. Figure shows the schematic of a plate type evaporator.



Schematic of a plate type evaporator

Expansion Devices:

An expansion device is another basic component of a refrigeration system. The basic functions of an expansion device used in refrigeration systems are to:

1. Reduce pressure from condenser pressure to evaporator pressure, and

2. Regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator.

Under ideal conditions, the mass flow rate of refrigerant in the system should be proportional to the cooling load. Sometimes, the product to be cooled is such that a constant evaporator temperature has to be maintained. In other cases, it is desirable that liquid refrigerant should not enter the compressor. In such a case, the mass flow rate has to be controlled in such a manner that only superheated vapour leaves the evaporator. Again, an ideal refrigeration system should have the facility to control it in such a way that the energy requirement is minimum and the required criterion of temperature and cooling load are satisfied. Some additional controls to control the capacity of compressor and the space temperature may be required in addition, so as to minimize the energy consumption.

The expansion devices used in refrigeration systems can be divided into fixed opening type or variable opening type. As the name implies, in fixed opening type the flow area remains fixed, while in variable opening type the flow area changes with changing mass flow rates. There are basically seven types of refrigerant expansion devices. These are:

1.Hand (manual) expansion valves

2. Capillary Tubes

3. Orifice

4. Constant pressure or Automatic Expansion Valve (AEV)

5. Thermostatic Expansion Valve (TEV)

6. Float type Expansion Valve

   a) High Side Float Valve
   b) Low Side Float Valve

7. Electronic Expansion Valve

Of the above seven types, Capillary tube and orifice belong to the fixed opening type, while the rest belong to the variable opening type. Of the above seven types, the hand operated expansion valve is not used when an automatic control is required. The orifice type expansion is used only in some special applications. Hence these two are not discussed here.

Capillary Tube

A capillary tube is a long, narrow tube of constant diameter. The word "capillary" is a misnomer since surface tension is not important in refrigeration application of capillary tubes. Typical tube diameters of refrigerant capillary tubes range from 0.5 mm to 3 mm and the length ranges from 1.0 m to 6 m. The pressure reduction in a capillary tube occurs due to the following two factors:

1. The refrigerant has to overcome the frictional resistance offered by tube walls. This leads to some pressure drop, and
2. The liquid refrigerant flashes (evaporates) into mixture of liquid and vapour as its pressure reduces. The density of vapour is less than that of the liquid. Hence, the average density of refrigerant decreases as it flows in the tube. The mass flow rate and tube diameter (hence area) being constant, the velocity of refrigerant increases since $m = \rho VA$. The increase in velocity or acceleration of the refrigerant also requires pressure drop.

Several combinations of length and bore are available for the same mass flow rate and pressure drop. However, once a capillary tube of some diameter and length has been installed in a refrigeration system, the mass flow rate through it will vary in such a manner that the total pressure drop through it matches with the pressure difference between condenser and the evaporator. Its mass flow rate is totally dependent upon the pressure difference across it; it cannot adjust itself to variation of load effectively.

Advantages and disadvantages of capillary tubes

Some of the advantages of a capillary tube are:

1. It is inexpensive.

2. It does not have any moving parts hence it does not require maintenance

3. Capillary tube provides an open connection between condenser and the evaporator hence during off-cycle, pressure equalization occurs between condenser and evaporator. This reduces the starting torque requirement of the motor since the motor starts with same pressure on the two sides of the compressor. Hence, a motor with low starting torque (squirrel cage Induction motor) can be used.

4. Ideal for hermetic compressor based systems, which are critically charged and factory assembled.

Some of the disadvantages of the capillary tube are:

1. It cannot adjust itself to changing flow conditions in response to daily and seasonal variation in ambient temperature and load. Hence, COP is usually low under off design conditions.

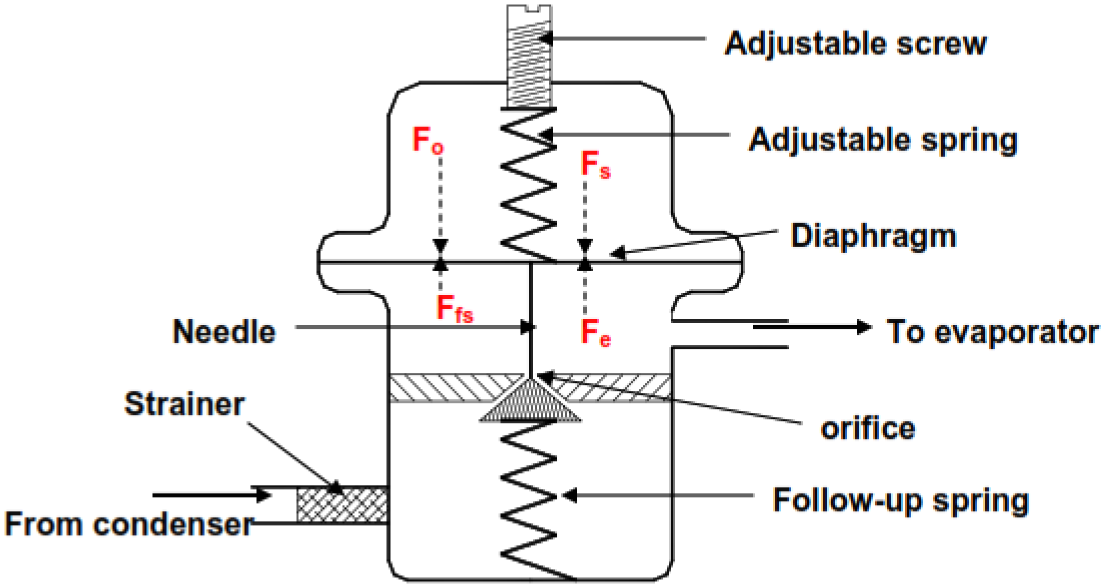
2. It is susceptible to clogging because of narrow bore of the tube, hence, utmost care is required at the time of assembly. A filter-drier should be used ahead of the capillary to prevent entry of moisture or any solid particles

3. During off-cycle liquid refrigerant flows to evaporator because of pressure difference between condenser and evaporator. The evaporator may get flooded and the liquid refrigerant may flow to compressor and damage it when it starts. Therefore critical charge is used in capillary tube based systems. Further, it is used only with hermetically sealed compressors where refrigerant does not leak so that critical charge can be used .Normally an accumulator is provided after the evaporator to prevent slugging of compressor.

Automatic Expansion Valve (AEV)

An Automatic Expansion Valve (AEV) also known as a constant pressure expansion valve acts in such a manner so as to maintain a constant pressure and thereby a constant temperature in the evaporator. The schematic diagram of the valve is shown in Fig. As shown in the figure, the valve consists of an adjustment spring that can be adjusted to maintain the required temperature in the evaporator. This exerts force Fs on the top of the diaphragm. The atmospheric pressure, Po also acts on top of the diaphragm and exerts a force of

F = Po .Ad , Ad being the area of the diaphragm. The evaporator pressure Pe acts below the diaphragm. The force due to evaporator pressure is F e = Pe. Ad. The net downward force Fs + Fo - Fe is fed to the needle by the diaphragm. This net force along with the force due to follow-up spring F_{fs} controls the location of the needle with respect to the orifice and thereby controls the orifice opening.



Schematic of an Automatic Expansion Valve

If $F_e + F_{fs} > F_s + F_o$ the needle will be pushed against the orifice and the valve will be fully closed.

On the other hand if $F_e + F_{fs} < F_s + F_o$, the needle will be away from the orifice and the valve will be open. Hence the relative magnitude of these forces controls the mass flow rate through the expansion valve.

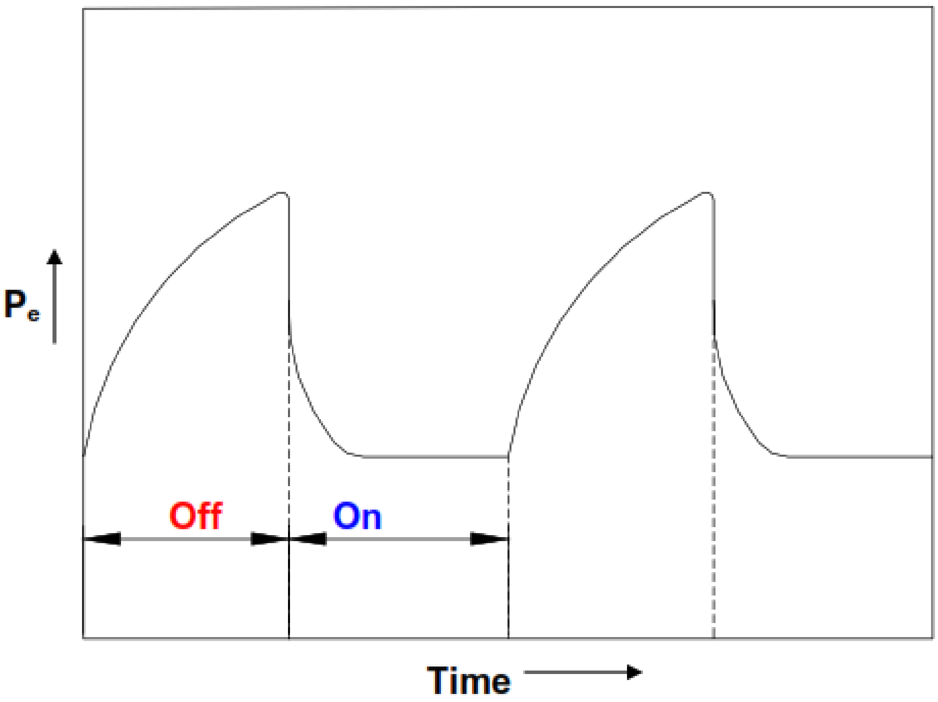
The adjustment spring is usually set such that during off-cycle the valve is closed, that is, the needle is pushed against the orifice. Hence,

$$F_{eo} + F_{fso} > F_{so} + F_o$$

Where, subscript $_o$ refers to forces during off cycle. During the off-cycle, the refrigerant remaining in the evaporator will vaporize but will not be taken out by the compressor, as a result the evaporator pressure rises during the off-cycle as shown in Fig.24.10.

When the compressor is started after the off-cycle period, the evaporator pressure P_e starts decreasing at a very fast rate since valve is closed; refrigerant is not fed to evaporator while the compressor removes the refrigerant from the evaporator. This is shown in Fig.24.10. As P_e decreases the force F_e decreases from F_{eo} to $(F_{eo} - \Delta F_e)$. At one stage, the sum $F_e + F_{fs}$ becomes less than $F_s + F_o$, as a result the needle stand moves downwards (away from the needle stand) and the valve opens. Under this condition,

$$(F_{eo} - \Delta F_e) + F_{fso} < F_{so} + F_o$$



Variation of evaporator pressure during on- and off-cycles of an AEV based refrigeration system

When the refrigerant starts to enter the evaporator, the evaporator pressure does not decrease at the same fast rate as at starting time. Thus, the movement of the needle stand will slow down as the refrigerant starts entering. As the needle moves downwards, the adjustment spring elongates, therefore the force F_s decreases from its off-cycle value of F_{s0}, the decrease being proportional to the movement of the needle.

As the needle moves downwards, the follow-up spring is compressed; as a result, F_{fs} increases from its off-cycle value. Hence, the final equation may be written as,

$$(F_{eo} - \Delta F_e) + (F_{fso} + \Delta F_{fs}) = (F_{so} - \Delta F_s) + F_o \quad \text{or}$$

$$F_e + F_{fs} = F_s + F_o \quad = \text{constant}$$

The constant is sum of force due to spring force and the atmospheric pressure, hence it depends upon position of adjustment spring. This will be the equilibrium position. Then onwards, the valve acts in such a manner that the evaporator pressure remains constant as long as the refrigeration load is constant. At this point, the mass flow rate through the valve is the same as that through the compressor.

Applications of automatic expansion valve

The automatic expansion valves are used wherever constant temperature is required, for example, milk chilling units and water coolers where freezing is disastrous. In air-conditioning systems it is used when humidity control is by DX coil temperature. Automatic expansion valves are simple in design and are economical. These are also used in home freezers and small commercial refrigeration systems where hermetic compressors are used. Normally the usage is limited to systems of less than 10 TR capacities with critical charge. Critical charge has to be used since the system using AEV is prone to flooding. Hence, no receivers are used in these systems. In some valves a diaphragm is used in place of bellows.

Thermostatic Expansion Valve (TEV)

Thermostatic expansion valve is the most versatile expansion valve and is most commonly used in refrigeration systems. A thermostatic expansion valve maintains a constant degree of superheat at the exit of evaporator; hence it is most effective for dry evaporators in preventing the slugging of the compressors since it does not allow the liquid refrigerant to enter the compressor. The schematic diagram of the valve is given in Figure. This consists of a feeler bulb that is attached to the evaporator exit tube so that it senses the temperature at the exit of evaporator. The feeler bulb is connected to the top of the bellows by a capillary tube. The feeler bulb and the narrow tube contain some fluid that is called power fluid. The power fluid may be the same as the refrigerant in the refrigeration system, or it may be different. In case it is different from the refrigerant, then the TEV is called TEV with cross charge. The pressure of the power fluid Pp is the saturation pressure corresponding to the temperature at the evaporator exit. If the evaporator temperature is Te and the corresponding saturation evaporator pressure is Pe, then the purpose of TEV is to maintain a temperature Te +ΔTs at the evaporator exit, where ΔTs is the degree of superheat required from the TEV. The power fluid senses this temperature Te +ΔTs by the feeler bulb and its pressure Pp is the saturation pressure at this temperature. The force Fp exerted on top of bellows of area A due to this pressure is given by:

$$\mathbf{F_p = A_b\, P_p}$$

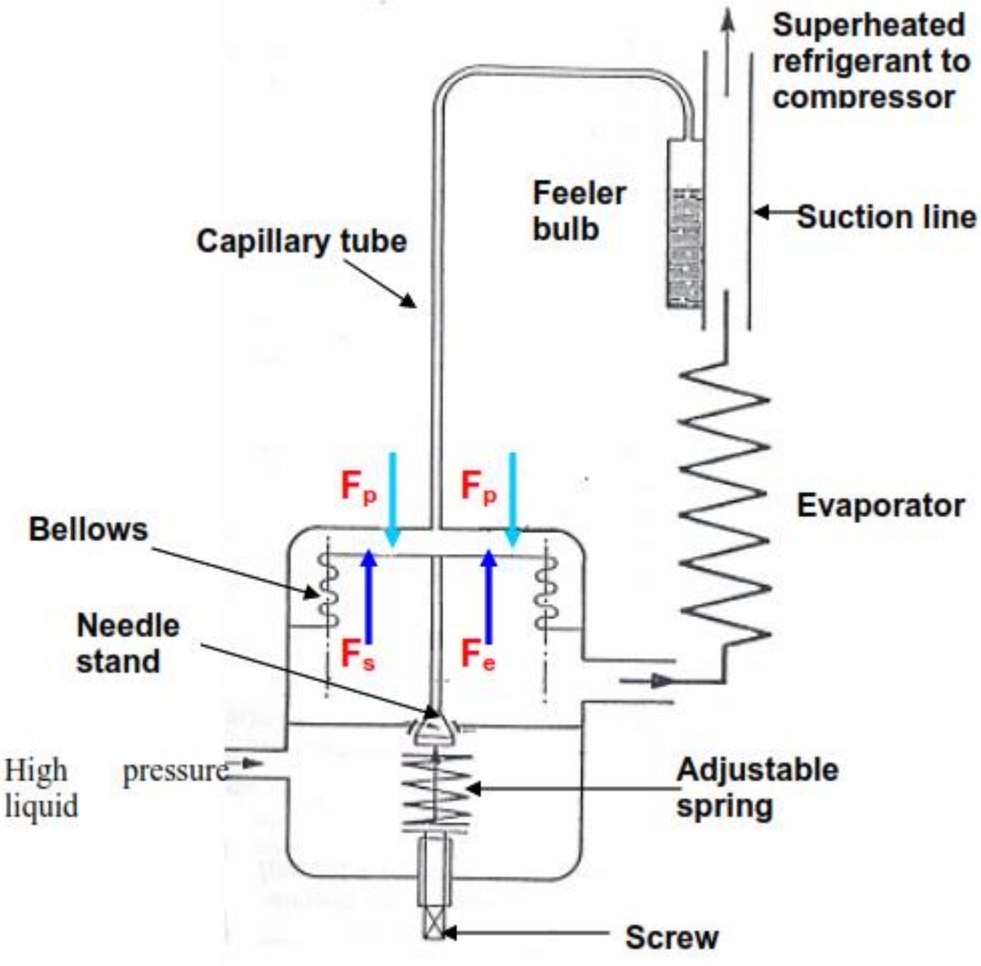
The evaporator pressure is exerted below the bellows. In case the evaporator is large and has a significant pressure drop, the pressure from evaporator exit is fed directly to the bottom of the bellows by a narrow tube. This is called pressure equalizing connection. Such a TEV is called TEV with external equalizer, otherwise it is known as TEV with internal equalizer. The force Fe exerted due to this pressure Pe on the bottom of the bellows is given by

$$F_e = A_b\, P_e$$

The difference of the two forces Fp and Fe is exerted on top of the needle stand. There is an adjustment spring below the needle stand that exerts an upward spring force Fs on the needle stand. In steady state there will be a force balance on the needle stand, that is,

$$F_s = F_p - F_e$$

During off-cycle, the evaporator temperature is same as room temperature throughout, that is, degree of superheat ΔT_s is zero. If the power fluid is the same as the refrigerant, then $P_p = P_e$ and $F_p = F_e$. Therefore any arbitrarily small spring force F_s acting upwards will push the needle stand against the orifice and keep the TEV closed. If it is *TEV with cross charge* or if there is a little degree of superheat during off-cycle then for TEV to remain closed during off-cycle, F_s should be slightly greater than $(F_p - F_e)$.



Schematic of a Thermostatic Expansion Valve (TEV)

As the compressor is started, the evaporator pressure decreases at a very fast rate hence the force Fe decreases at a very fast rate. This happens since TEV is closed and no refrigerant is fed to evaporator while compressors draws out refrigerant at a very fast rate and tries to evacuate the evaporator. The force Fp does not change during this period since the evaporator temperature does not change. Hence, the difference Fp -Fe, increases as the compressor runs for some time after starting. At one point this difference becomes greater than the spring force Fs and pushes the needle stand downwards opening the orifice. The valve is said to open up. Since a finite downward force is required to open the valve, a minimum degree of superheat is required for a finite mass flow rate. As the refrigerant enters the evaporator it arrests the fast rate of decrease of evaporator pressure. The movement of needle stand also slows down. The spring, however gets compressed as the needle stand moves downward to open

the orifice. If F_{s0} is the spring force in the rest position, that is, off-cycle, then during open valve position

$$\mathbf{F_s = F_{s0} + \Delta F_s}$$

Eventually, the needle stand reaches a position such that,

$$F_s = F_p - F_e = A_b\,(P_p - P_e)$$

That is, F_p is greater than F_e or P_p is greater than P_e. The pressure P_p and P_e are saturation pressures at temperature ($T_e + \Delta T_s$) and T_e respectively. Hence, for a given setting force F_s of the spring, TEV maintains the difference between F_p and F_e or the degree of superheat ΔT_s constant.

$$\begin{aligned}\Delta T_s &\propto (F_p - F_e)\\ &\propto F_s\end{aligned}$$

This is irrespective of the level of P_e, that is, evaporator pressure or temperature, although degree of superheat may be slightly different at different evaporator temperatures for same spring force, F_s. It will be an ideal case if the degree of superheat is same at all evaporator temperatures for a given spring force.

Advantages, disadvantages and applications of TEV

The advantages of TEV compared to other types of expansion devices are:

1. It provides excellent control of refrigeration capacity as the supply of refrigerant to the evaporator matches the demand

2. It ensures that the evaporator operates efficiently by preventing starving under high load conditions

3. It protects the compressor from slugging by ensuring a minimum degree of superheat under all conditions of load, if properly selected.

However, compared to capillary tubes and AEVs, a TEV is more expensive and proper precautions should be taken at the installation. For example, the feeler bulb must always be in good thermal contact with the refrigerant tube. The feeler bulb should preferably be insulated to reduce the influence of the ambient air. The bulb should be mounted such that the liquid is always in contact with the refrigerant tubing for proper control.

The use of TEV depends upon degree of superheat. Hence, in applications where a close approach between the fluid to be cooled and evaporator temperature is desired, TEV cannot be used since very small extent of superheating is available for operation. A counter flow arrangement can be used to achieve the desired superheat in such a case. Alternately, a subcooling HEX may be used and the feeler bulb mounted on the vapour exit line of the HEX. The valves with bellows have longer stroke of the needle, which gives extra sensitivity compared to diaphragm type of valve. But valves with bellows are more expensive.

Thermostatic Expansion Valves are normally selected from manufacturers' catalogs. The selection is based on the refrigeration capacity, type of the working fluid, operating temperature range etc. In practice, the design is different to suit different requirements such as single evaporators, multi-evaporators etc.

Float type expansion valves:

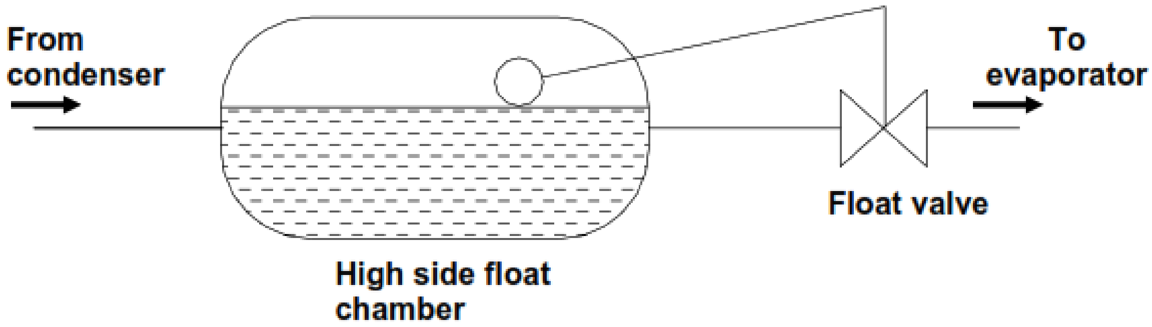
Float type expansion valves are normally used with flooded evaporators in large capacity refrigeration systems. A float type valve opens or closes depending upon the liquid level as sensed by a buoyant member, called as float. The float could take the form of a hollow metal or plastic ball, a hollow cylinder or a pan. Thus the float valve always maintains a constant liquid level in a chamber called as float chamber. Depending upon the location of the float chamber, a float type expansion valve can be either a low-side float valve or a high-side float valve.

Low-side float valves:

A low-side float valve maintains a constant liquid level in a flooded evaporator or a float chamber attached to the evaporator. When the load on the system increases, more amount of refrigerant evaporates from the evaporator. As a result, the refrigerant liquid level in the evaporator or the low-side float chamber drops momentarily. The float then moves in such a way that the valve opening is increased and more amount of refrigerant flows into the evaporator to take care of the increased load and the liquid level is restored. The reverse process occurs when the load falls, i.e., the float reduces the opening of the valve and less amount of refrigerant flows into the evaporator to match the reduced load. As mentioned, these valves are normally used in large capacity systems and normally a by-pass line with a hand-operated expansion is installed to ensure system operation in the event of float failure.

High-side float valves:

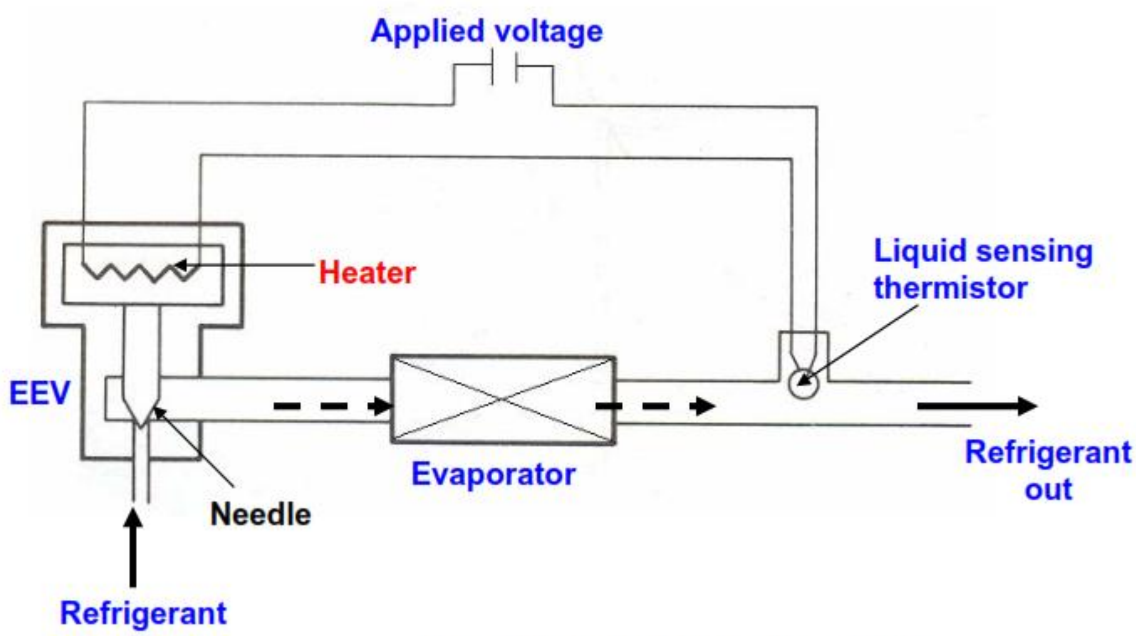
Figure shows the schematic of a high-side float valve. As shown in the figure, a high-side float valve maintains the liquid level constant in a float chamber that is connected to the condenser on the high pressure side. When the load increases, more amount of refrigerant evaporates and condenses. As a result, the liquid level in the float chamber rises momentarily. The float then opens the valve more to allow a higher amount of refrigerant flow to cater to the increased load, as a result the liquid level drops back to the original level. The reverse happens when the load drops. Since a high-side float valve allows only a fixed amount of refrigerant on the high pressure side, the bulk of the refrigerant is stored in the low-pressure side (evaporator). Hence there is a possibility of flooding of evaporator followed by compressor slugging. However, unlike lowside float valves, a high-side float valve can be used with both flooded as well as direct expansion type evaporators.



Schematic of a high-side float valve

Electronic Type Expansion Valve

The schematic diagram of an electric expansion valve is shown in Fig. As shown in the figure, an electronic expansion valve consists of an orifice and a needle in front it. The needle moves up and down in response to magnitude of current in the heating element. A small resistance allows more current to flow through the heater of the expansion valve, as a result the valve opens wider. A small negative coefficient thermistor is used if superheat control is desired. The thermistor is placed in series with the heater of the expansion valve. The heater current depends upon the thermistor resistance that depends upon the refrigerant condition. Exposure of thermistor to superheated vapour permits thermistor to selfheat thereby lowering its resistance and increasing the heater current. This opens the valve wider and increases the mass flow rate of refrigerant. This process continues until the vapour becomes saturated and some liquid refrigerant droplets appear. The liquid refrigerant will cool the thermistor and increase its resistance. Hence in presence of liquid droplets the thermistor offers a large resistance, which allows a small current to flow through the heater making the valve opening narrower. The control of this valve is independent of refrigerant and refrigerant pressure; hence it works in reverse flow direction also. It is convenient to use it in year-round-air-conditioning systems, which serve as heat pumps in winter with reverse flow. In another version of it the heater is replaced by stepper motor, which opens and closes the valve with a great precision giving a proportional control in response to temperature sensed by an element.



Schematic of an electronic expansion valve

Refrigerants

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 0C, 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:

i. Thermodynamic and thermo-physical properties

ii.Environmental and safety properties, and

iii. Economics.

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 mass flow rate per unit cooling capacity will be small

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 a value 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 nonexplosive. 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) Dilelectric 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 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 – 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

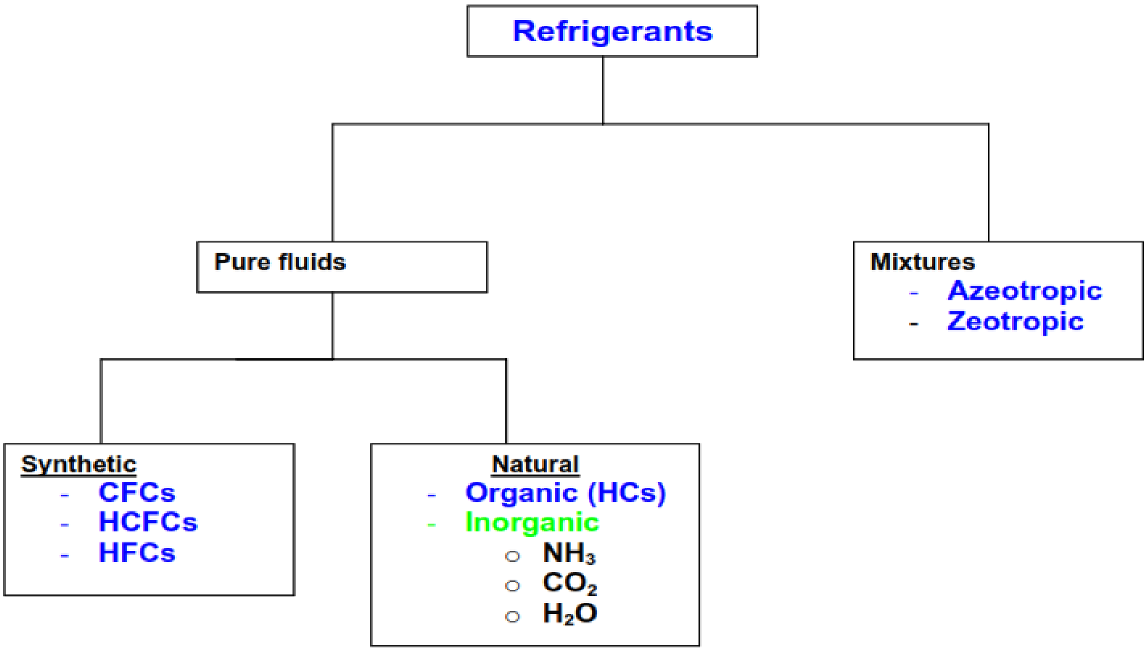


Fig.26.1: *Classification of fluids used as refrigerants*

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 (C_3H_8)	:	R 290
n-butane (C_4H_{10})	:	R 600
iso-butane (C_4H_{10})	:	R 600a

Unsaturated Hydrocarbons: R1150 (C_2H_4)
R1270 (C_3H_6)

Refrigerant	Application	Substitute suggested Retrofit(R)/New (N)
R 11(CFC) NBP = 23.7°C h_{fg} at NBP=182.5 kJ/kg T_{cr} =197.98°C Cp/Cv = 1.13 ODP = 1.0 GWP = 3500	Large air conditioning systems Industrial heat pumps As foam blowing agent	R 123 (R,N) R 141b (N) R 245fa (N) n-pentane (R,N)
R 12 (CFC) NBP = -29.8°C h_{fg} at NBP=165.8 kJ/kg T_{cr} =112.04°C Cp/Cv = 1.126 ODP = 1.0 GWP = 7300	Domestic refrigerators Small air conditioners Water coolers Small cold storages	R 22 (R,N) R 134a (R,N) R 227ea (N) 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 Cp/Cv = 1.166 ODP = 0.05 GWP = 1500	Air conditioning systems Cold storages	R 410A, R 410B (N) 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 Cp/Cv = 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 (NH_3) NBP = -33.35°C h_{fg} at NBP=1368.9 kJ/kg T_{cr} =133.0°C Cp/Cv = 1.31 ODP = 0.0 GWP = 0.0	Cold storages Ice plants Food processing Frozen food cabinets	**No replacement required** * Toxic and flammable * Incompatible with copper * Highly efficient * Inexpensive and available
R 744 (CO_2) NBP = -78.4°C h_{fg} at 40°C=321.3 kJ/kg T_{cr} =31.1°C Cp/Cv = 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

Refrigerants, their applications and substitutes

Refrigerant	Application	Substitute suggested Retrofit(R)/New (N)
R718 (H_2O) NBP = 100.°C h_{fg} at NBP=2257.9 kJ/kg T_{cr} =374.15°C Cp/Cv = 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 Cp/Cv = 1.086 ODP = 0.0 GWP = 3.0	Replacement for R 12 Domestic refrigerators Water coolers	**No replacement required** * Flammable * Eco-friendly

INDUSTRIAL APPLICATIONS

Vapour compression refrigeration cycle finds it place in following applications:

- Domestic refrigerator
- Domestic Air conditioner
- Chiller for cold water around 5 to 6 degree centigrade
- Central air conditioning system
- Cold storage to cool vegetables and fruits
- Ice plant to make ice
- Industry application like process industry
- And many applications where cooling is desired.
- Refrigerated trucks and railroad cars.
- large-scale warehouses for chilled or frozen storage of foods and meats
- Oil refineries, petrochemical and chemical processing plants,
- Natural gas processing plants are among the many types of industrial plants that often utilize large vapor-compression refrigeration systems.

TUTORIAL QUESTIONS

1. List important components of a vapour compression refrigeration system
2. Classify refrigerant compressors based on their working principle and based on the arrangement of compressor motor/external drive?
3. Draw the schematic of a reciprocating compressor and explain its working principle
4. Compare air-cooled condensers with water-cooled condensers
5. Explain the basic functions of expansion devices in refrigeration systems
6. Describe advantages, disadvantages and applications of different types of expansion valves,
7. Classify refrigerant evaporators and discuss the salient features of different types of evaporators
8. List important thermodynamic and environmental properties influencing refrigerant selection?
9. What are the applications of refrigeration?
10. Write the desirable properties of the Refrigerents and Classification of Refrigerants?.
11. Explain with the neat sketch the working of vapour compression refrigeration system?
12. Explain different modes of heat transfer? explain latent heat and Sensible Heat?

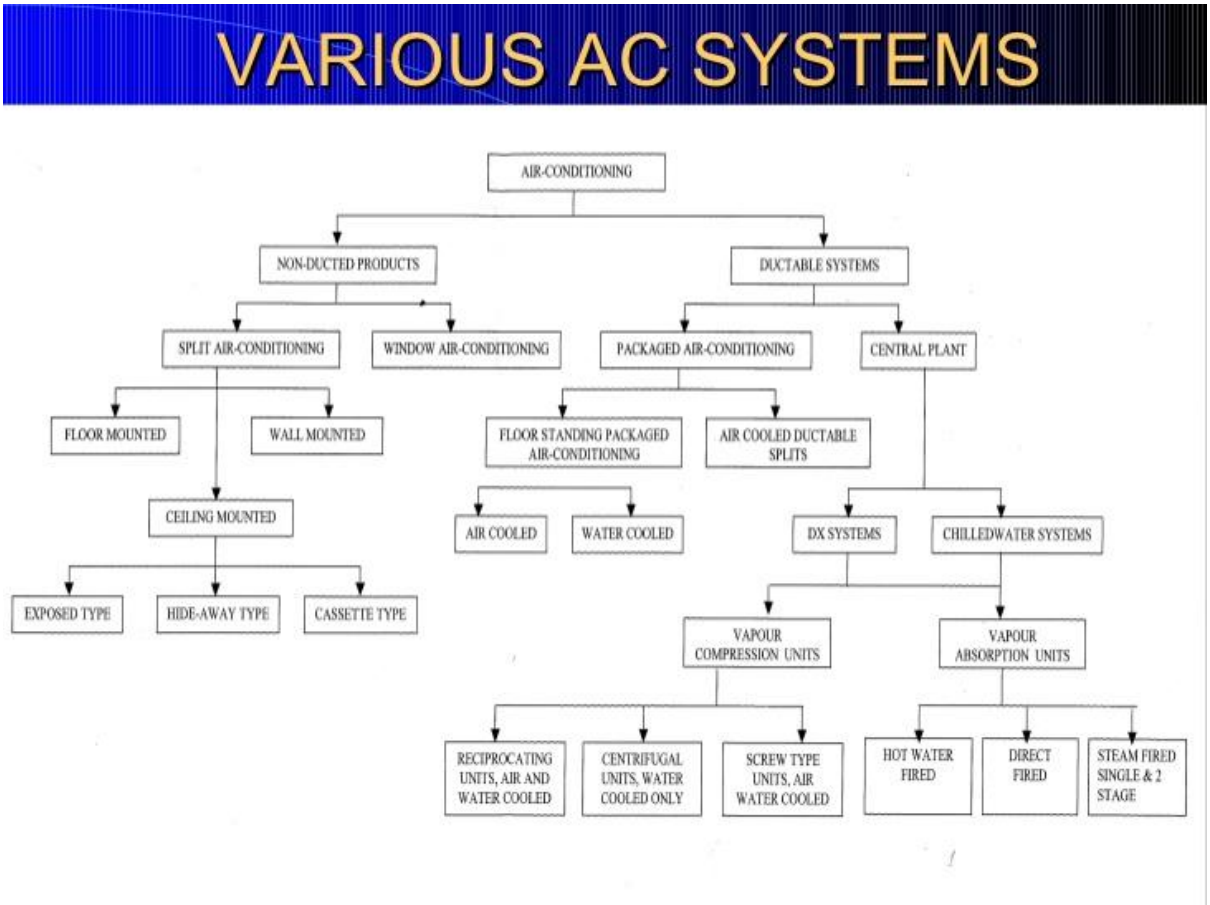
UNIT II

CLASSIFICATION OF AIR-CONDITIONING SYSTEM

COURSE OBJECTIVE: To Know Detail classification of Air-Conditioning System

COURSE OUTCOME: Students will assist in the installations of Heating, Air Conditioning and Refrigeration equipment.

1.**CLASSIFICATION OF AIR-CONDITIONING SYSTEM:**



1.WINDOW AIR CONDITIONING SYSTEM

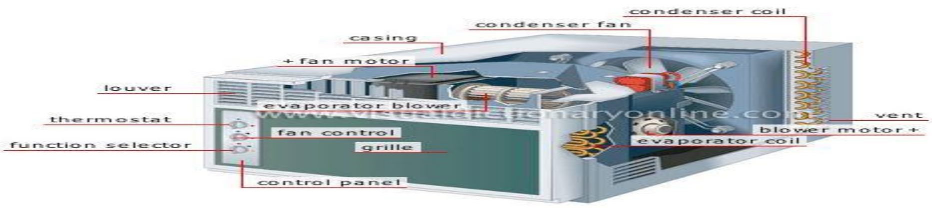
Windows air conditioners are one of the most widely used types of air conditioners because they are the simplest form of the air conditioning systems. Window air conditioner comprises of the rigid base on which all the parts of the window air conditioner are assembled. The base is assembled inside the casing which is fitted into the wall or the window of the room in which the air conditioner is fitted.

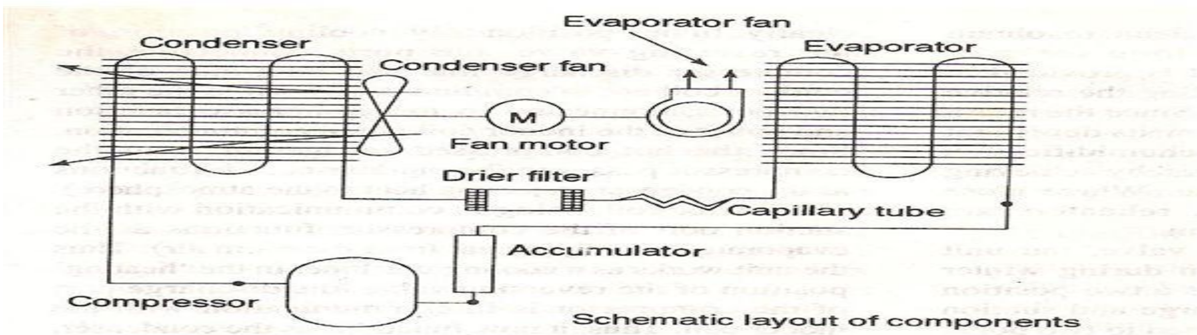
The whole assembly of the window air conditioner can be divided into two compartments: the room side, which is also the cooling side and the outdoor side from where the heat absorbed by the room air is liberated to the atmosphere. The room side and outdoor side are separated from each other by an insulated partition enclosed inside the window air conditioner assembly (refer fig 1 below).

In the front of the window air conditioner on the room side there is beautifully decorated front panel on which the supply and return air grills are fitted (the whole front panel itself is commonly called as front grill). The louvers fitted in the supply air grills are adjustable so as to supply the air in desired direction. There is also one opening in the grill that allows access to the control panel or operating panel in front of the window air conditioner.

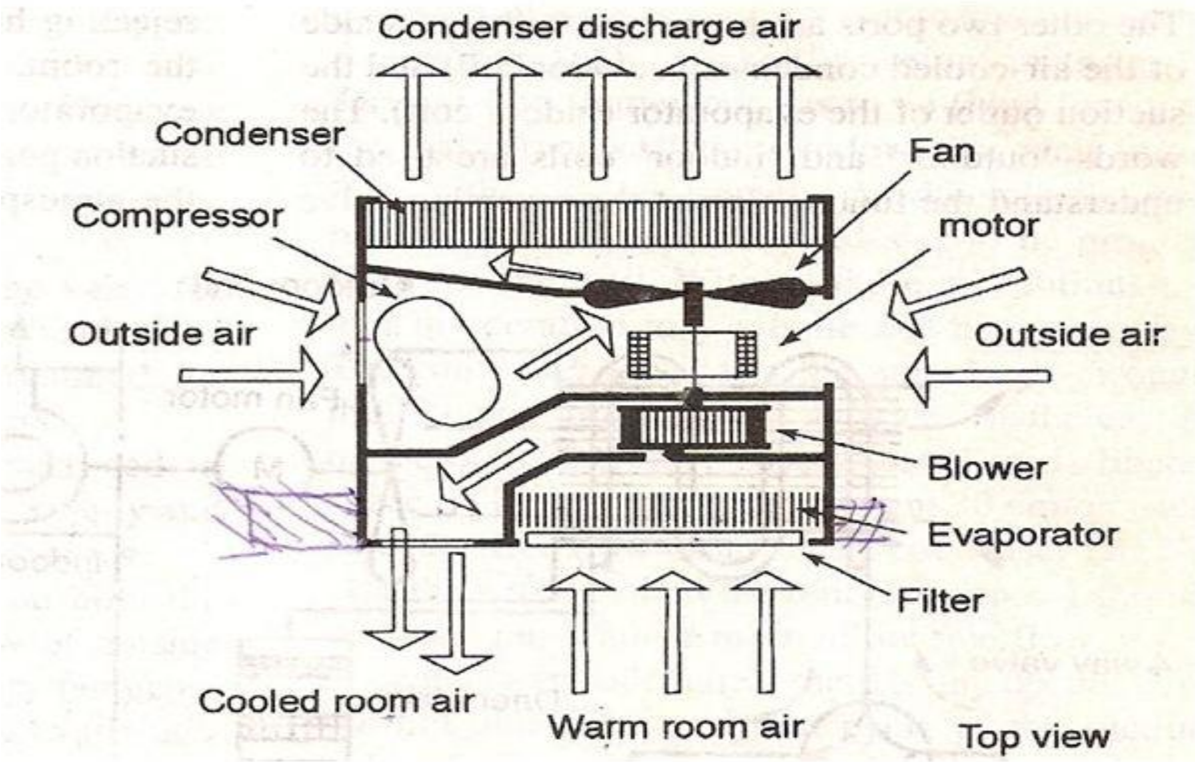
The various parts of the window air conditioner can be divided into following categories: the refrigeration system, air circulation system, ventilation system, control system, and the electrical protection system. All these have been discussed in details below along with the front panel and other parts.

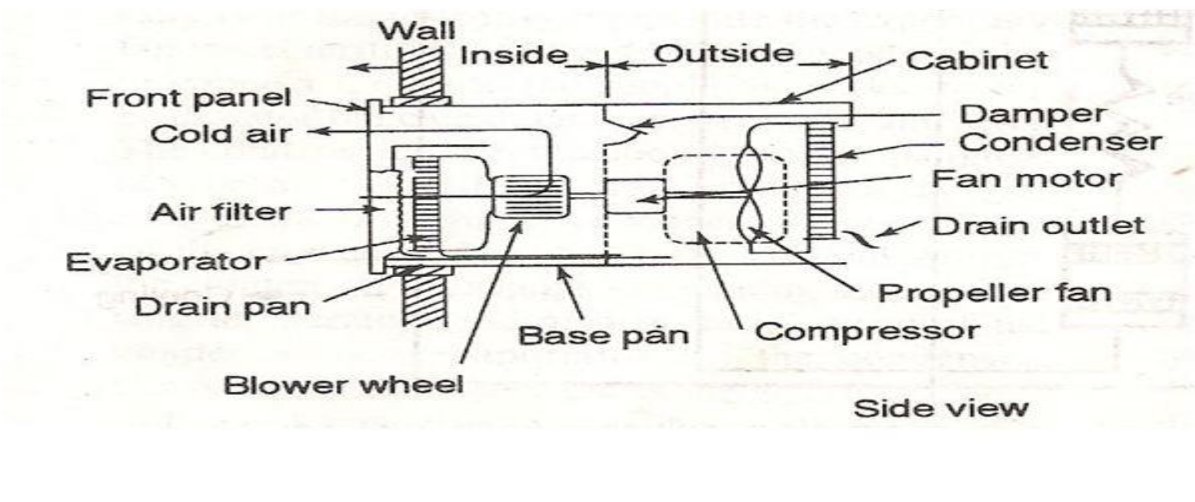
Window Air Conditioner





Schematic layout of components


Condenser discharge air
Condenser
Fan
Compressor
motor
Outside air
Outside air
Blower
Evaporator
Filter
Cooled room air
Warm room air
Top view


Wall
Inside
Outside
Cabinet
Front panel
Damper
Cold air
Condenser
Fan motor
Air filter
Drain outlet
Evaporator
Propeller fan
Drain pan
Base pan
Compressor
Blower wheel
Side view

The Refrigeration System of the Window Air Conditioner

The refrigeration system of the window air conditioner comprises of all the important parts of the refrigeration cycle. These include the compressor, condenser, expansion valve and the evaporator. All these components have been shown in fig 3 above. The refrigerant used in most of the window air conditioners is R22.

The compressor used in the window air conditioners is hermetically sealed type, which is portable one. This compressor has long life and it carries long warranty periods. In case of the maintenance problems it can be replaced easily from the company. The condenser is made up of copper tubing and it is cooled by the atmospheric air. The condenser is covered with the fins to enable faster heat transfer rate from it.

The capillary tubing made up of various rounds of the copper coil is used as the expansion valve in the window air conditioners. Just before the capillary there is drier filter that filters the refrigerant and also removes the moisture particles, if present in the refrigerant.

Like condenser, the evaporator is also made up of copper tubing of number of turns and is covered with the fins. The evaporator is also called as the cooling coil since the rooms air passes over it and gets cooled. Just in front of the evaporator there is air filter fitted in the front panel or front grill. As the room air is absorbed, it is first passed over the filter so that it gets filtered. The filtered air is then blown over the cooling coil and the chilled air is passed into the room.

The refrigerant after leaving the cooling coil enters the accumulator where it is accumulated and then it is again sucked by the compressor for recirculation over the whole cycle.

Air Circulation System of the Window Air Conditioner

The air circulation system of the window air conditioner comprises of the following parts (please refer fig 4 & 5).

1) Blower: This is the small blower that is fitted behind the evaporator or cooling coil inside the assembly of the window air conditioner system. The blower sucks the air from the room which first passes over the air filter and gets filtered. The air then passes over the cooling coil and gets chilled. The blower then blows this filtered and chilled air, which passes through the supply air compartment inside the window air conditioner assembly. This air is then delivered into the room from the supply air grill of the front panel.

2) **Propeller fan or the condenser fan**: The condenser fan is the forced draft type of propeller fan that sucks the atmospheric air and blows it over the condenser. The hot refrigerant inside the condenser gives up the heat to the atmospheric air and its temperature reduces.

3) Fan motor: The motor inside the window air conditioner assembly is located between the condenser and the evaporator coil. It has double shaft on one side of which the blower is fitted and on the other side the condenser fan is fitted. This makes the whole assembly of the blower, the condenser fan and the motor highly compact.

Control System of the Window Air Conditioners

In front of the window air conditioner there is control panel or the operating panel that carries various control buttons. This control panel can be easily accessed from the front panel of the window air conditioner. The three important parameters that are to be controlled inside the window air conditioner are the room air temperature, the flow rate of the air and the direction of the air. All these controls are discussed below and also shown in figure 1.

1) **Thermostat for controlling the room air temperature**: For controlling the temperature inside the room there is thermostat. The thermostat sensor is connected directly to the cooling coil to sense its temperature. The thermostat is also connected to the switch in control panel and it has the knob for setting the temperature. The person inside the room can easily set the temperature required by rotating this knob. In the modern window air conditioners, there is printed circuit board (PCB) to which the thermostat is connected. This PCB has remote sensor so the setting of the thermostat can be easily changed by the remote control.

2**) Air flow rate inside the room**: The motor connected to the blower is of multispeed type, so one can change the speed of the motor. As the speed of the motor changes the amount of air sucked by it and blown by it also changes and so the amount of air delivered in the room also changes. The speed of the motor can be changed by the knob provided in the control panel or by the remote control if the air conditioner has PCB fitted into it.

3) Direction of the air flow inside the room: In front panel of the window air conditioner there are horizontal lovers. Additionally, in front of the air conditioner body and attached to it are the vertical louvers. The chilled air from blown by the blower passes into the room through these louvers. The horizontal louvers in the front panel enable changing the vertical motion direction of the air inside the room. The position of these lovers can be changed manually.

The vertical louvers enable changing the horizontal motion of the air inside the room. These louvers are connected to the small motor. The vertical louvers can be kept moving in the vertical direction so that the air flows throughout the room uniformly or they can be kept in the fixed direction so that the air flows in particular desired direction only. The operation of the motor of the vertical lovers can be controlled by the small button on the control panel of the window air conditioner (refer the figure below especially figure 5 & 6). In case of the automatic window air conditioner with PCB, the motion of the vertical lovers can be controlled by the remote. The horizontal louvers in the front panel and the vertical louvers enable fine control of the distribution of air inside the room.

Front Panel of the Window Air Conditioner

The very front covering of the window air conditioner assembly that is visible to the person is the front panel (many times called as the front grill). For novice people the front panel itself is the whole air conditioner. These days lots of importance is being given to the aesthetics of the front panel so the window air conditioner also serves as the decorative item inside the room. The front panel has two important compartments: return air and supply air compartments. These are described below (please refer the figure above, especially figure 5 & 6).

1) **Return air compartment:** The return air compartment of the front panel comprises of the return air grill and the air filter. When the blower sucks the air, it is first over the return air grill and then over the air filter. Since the return air from the room comes inside the air conditioner via this part of the front panel, it is called as return air compartment of the grill.

2) Supply air compartment: The supply air compartment of the front panel comprises of the horizontal louvers as described above. The horizontal louvers help changing the vertical direction of air inside the room and their position can be changed manually as per the requirement.

There is another opening in the front panel that provides access to the control panel of the window air conditioner. The front panel of the window air conditioner can be removed easily for carrying out the maintenance works. If you want to remove the filter from the front panel, one can easily slide it out from the side without removing the whole panel.

Drainage System of the Window Air Conditioner

When the room air is chilled by the cooling coil the dew from the air is accumulated on the coil. This dew drops in the bottom base of the window air conditioner and it has to be removed by some system else the water will leak inside the room. For collecting the dew, the window air conditioned is installed with slightly tilted angle toward outside due to which all the dew water gets collected towards the back. There is small opening at the end for the drainage of this water. This opening can be left open or it can be connected to the small drain pan and the piping so that the water is drained out easily.

Electrical Protection System

The hermetically sealed compressor has motor fitted inside it. The compressor is the most important part of the air conditioning system so the motor connected to it should be protected against getting overheated and burning. Due to running of the air conditioner for long time, sometimes the winding gets overheated. To prevent the burning of the coil there is thermostat that senses the temperature of the coil. When the coil temperature reaches certain level, it trips the compressor and stops it until it gets cooled and restarts only after certain lower limit of the temperature is attained.

Air Filter

The air filter is very important part of the window air conditioner. It performs one of the most important functions of the window air conditioner, which is cleaning of the air. The air cleaner is fitted in the front of the air conditioner in the front panel. The room air first passes over the air filter and then over the cooling coil. Thus the filtered and chilled air is passed to the room. For proper working of the window air conditioner system cleaning the air filter once every two weeks is very important. If this is not done dust will get accumulated in the filter and the air will not be absorbed and supplied by the AC. Due to dirt the temperature of the evaporator may become too low resulting in the formation of ice and ultimately complete blockage of the cooling coil.

Working of Window AC

Now that we have seen the various **parts of the window air conditioner**, let us see its working. For understanding the working of the window AC please refer the figures given below. The working of window air conditioner can be explained by separately considering the two cycles of air: room air cycle and the hot air cycle. The compartments of the room and hot air are separated by an insulated partition inside the body of the air conditioner. The setting of thermostat and its working has also been explained in the discussions below.

Working of Window AC

Room Air Cycle

The air moving inside the room and in the front part of the air conditioner where the cooling coil is located is considered to be the room air. When the window AC is started the blower starts immediately and after a few seconds the compressor also starts. The evaporator coil or the cooling gets cooled as soon as the compressor is started.

The blower behind the cooling coil starts sucking the room air, which is at high temperature and also carries the dirt and dust particles. On its path towards the blower, the room air first passes through the filter where the dirt and dust particles from it get removed.

The air then passes over the cooling coil where two processes occur. Firstly, since the temperature of the cooling coil is much lesser than the room air, the refrigerant inside the cooling coil absorbs the heat from the air. Due to this the temperature of the room air becomes very low, that is the air becomes chilled.

Secondly, due to reduction in the temperature of the air, some dew is formed on the surface of the cooling coil. This is because the temperature of the cooling coil is lower than the dew point temperature of the air. Thus the moisture from the air is removed so the relative humidity of the air reduces. Thus when the room air passes over the cooling coil its temperature and relative humidity reduces.

This air at low temperature and low humidity is sucked by the blower and it blows it at high pressure. The chilled air then passes through small duct inside the air conditioner and it is then thrown outside the air conditioner through the opening in the front panel or the grill. This chilled air then enters the room and chills the room maintaining low temperature and low humidity inside the room.

The cool air inside the room absorbs the heat and also the moisture and so its temperature and moisture content becomes high. This air is again sucked by the blower and the cycle repeats. Some outside air also gets mixed with this room air. Since this air is sent back to the blower, it is also called as the return room air. In this way the cycle of this return air or the room air keeps on repeating.

Hot Air Cycle

The hot air cycle includes the atmospheric air that is used for cooling the condenser. The condenser of the window air conditioner is exposed to the external atmosphere. The propeller fan located behind the condenser sucks the atmospheric at high temperature and it blows the air over the condenser.

The refrigerant inside the condenser is at very high temperature and it has to be cooled to produce the desired cooling effect. When the atmospheric air passes over the condenser, it absorbs the heat from the refrigerant and its temperature increases. The atmospheric air is already at high temperature and after absorbing the condenser heat, its temperature becomes even higher. The person standing behind the condenser of the window AC can clearly feel the heat of this hot air. Since the temperature of this air is very high, this is called as hot air cycle.

The refrigerant after getting cooled enters the expansion valve and then the evaporator. On the other hand, the hot mixes with the atmosphere and then the fresh atmospheric air is absorbed by the propeller fan and blown over the condenser. This cycle of the hot air continues.

Setting the Room Temperature with Thermostat

The temperature inside the room can be set by using the thermostat knob or the remote control. If your window AC has knob, you would see some numbers or the round scale round the knob that will enable setting the temperature desired in the room. If your AC has come with the remote control, then you will see the room temperature on the digital indicator placed in the control panel of the window AC. You would probably also see the temperature on the small screen of the remote control. With the buttons provided on the remote control you can easily set the temperature inside the room.

When the desired temperature is attained inside the room, the thermostat stops the compressor of the AC. After some time when the temperature of the air becomes higher again, the thermostat restarts the compressor to produce the cooling effect. One should set the thermostat at the required temperature and not keep it at very low temperature to avoid high electricity bills.

Setting the Speed of the Air

The Speed of the air can be set by the fan motor button provided on the control panel. If your AC has the remote control you can see the fan speed button on it. The motor of the blower is of multispeed that type that enable changing the speed or the flow of air inside the room.

Important Part of the Window AC: Air Filter

The filter is a very important part of the AC since it cleans the air before it enters the room. For proper functioning of the filter it is very important to clean it every two weeks. If this is not done the filter will get choked and it won't be able to clean the air. Soon the dirt will also enter the evaporator coil and choke it. If this happens the AC will stop functioning and cleaning the evaporator becomes a very tedious process. Cleaning the filter hardly takes five minutes, do it regularly and enjoy the comforts of window AC on long-term basis.

2. SPLIT A/C

A split air conditioner consists of two main parts – a compressor located outside and an inside air outlet unit. Unlike a system that requires a series of ductwork networked throughout the ceiling, split air conditioners rely on a set of pipes to connect the outdoor to the inside air unit which is why there are referred to as a ductless mini-split air conditioner installation. Refrigerant is dispersed through the copper pipes that cycle through the system to generate either heated or cold air.

There are two main parts of the split air conditioner. These are:

1. **Outdoor unit**: This unit houses important components of the air conditioner like the compressor, condenser coil and also the expansion coil or capillary tubing. This unit is installed outside the room or office space which is to be cooled. The compressor is the maximum noise making part of the air conditioner, and since in the split air conditioner, it is located outside the room, the major source of noise is eliminated. In the outdoor unit there is a fan that blows air over the condenser thus cooling the compressed Freon gas in it. This gas passes through the expansion coil and gets

converted into low pressure, low temperature partial gas and partial liquid Freon fluid.

2. **Indoor unit**: It is the indoor unit that produces the cooling effect inside the room or the office. This is a beautiful looking tall unit usually white in color, though these days a number of stylish models of the indoor unit are being launched. The indoor unit houses the evaporator coil or the cooling coil, a long blower and the filter. After passing from the expansion coil, the chilled Freon fluid enters the cooling coil. The blower sucks the hot, humid and filtered air from the room and it blows it over the cooling coil. As the air passes over cooling coil its temperature reduces drastically and also loses the excess moisture. The cool and dry air enters the room and maintains comfortable conditions of around 25-27 degree Celsius as per the requirements.

The temperature inside the space can be maintained by thermostat setting. The setting should be such that comfortable conditions are maintained inside the room, and there is also chance for the compressor to trip at regular intervals. If the compressor keeps running continuously without break, its life will reduce.

These days multi-split air conditioners are also being used commonly. In units for one outdoor unit there are two indoor units which can be placed in two different rooms or at two different locations inside a large room.

Since there is long distance between the indoor and the outdoor unit, there is always loss of some cooling effect; hence for the same tonnage, split air conditioners produce somewhat less cooling effect than **window air conditioners**. However, with modern insulation material this gap has been reducing between the two. In any case, there are number of instances where there is just no alternative to the split air conditioners.

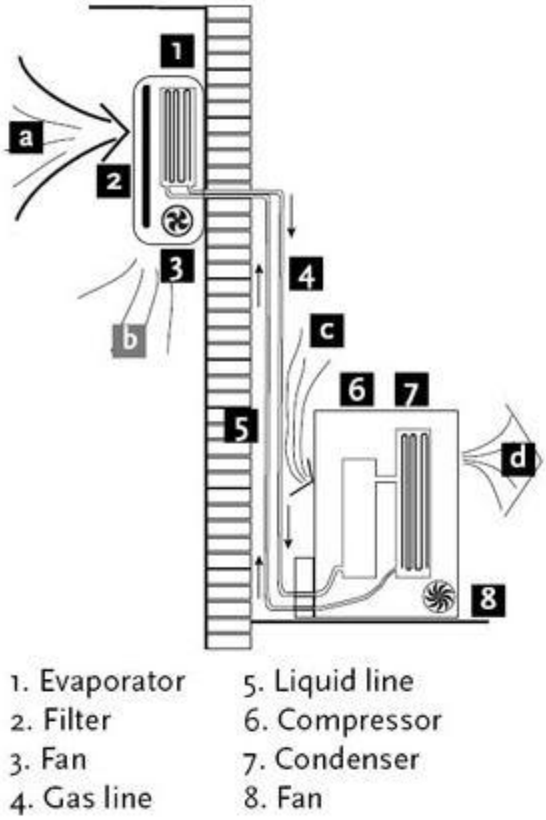
ntroduction

The split air conditioner is one of the most widely used type of the air conditioners. Earlier window air conditioner was used most widely, but the split air conditioner is now catching up with it. The major reasons behind the popularity if split air conditioner are their silent operation and elegant looks. Another advantage of the split air conditioner is that you don't have to make the hole in the wall of the air conditioner and destroy the beauty of the room.

These days the indoor units of the split air conditioner are available in wide range of color and designs.

There are two main parts of the split air conditioner: the indoor unit and the outdoor unit (see fig below). The indoor unit of the split AC is installed inside the room that is to be air conditioned or cooled while the outdoor unit is installed outside the room in open space where the unit can be installed and maintained easily. Apart from these two major parts there is copper tubing connecting the indoor and the outdoor units. Let us see the various parts of the indoor and the outdoor units of the split ACs.

Parts of Split Conditioner



Outdoor Unit

As mentioned previously the outdoor unit is installed outside the room to be air conditioned in the open space. In outdoor unit lots of heat is generated inside the compressor and the condenser, hence there should be sufficient flow of the air around it. The outdoor unit is usually installed at the height above the height of the indoor unit inside the room though in many cases the outdoor is also installed at level below the indoor unit.

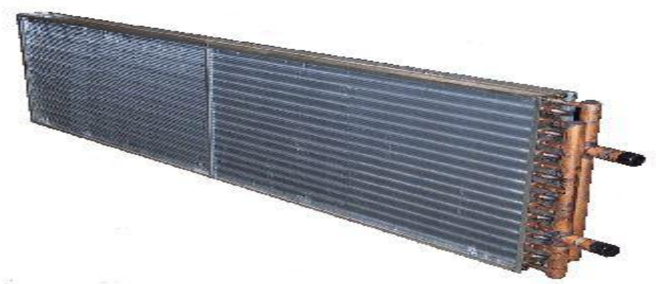
The outdoor unit contains the important parts of the split AC like compressor, condenser, expansion valve etc. Let us see these parts in more details:

1. Compressor:

The compressor is most important part of the any air conditioner. It compresses the refrigerant and increases its pressure before sending it to the condenser. The size of the compressor varies depending on the desired air conditioning load. In most of the domestic split air conditioners hermetically sealed type of compressor is used. In such compressors the motor used for driving the shaft is located inside the sealed unit and it is not visible externally. External power has to be supplied to the compressor, which is utilized for compressing the refrigerant and during this process lots of heat is generated in the compressor, which has to be removed by some means.

2. Condenser:

The condenser used in the outdoor unit of split air conditioners is the coiled copper tubing with one or more rows depending on the size of the air conditioning unit and the compressor. Greater the tonnage of the air conditioner and the compressor more are the coil turns and rows. The high temperature and high pressure refrigerant from the compressor comes in the condenser where it has to give up the heat. The tubing is made up of copper since it rate of conduction of heat is high. The condenser is also covered with the aluminum fins so that the heat from the refrigerant can be removed at more faster rate.



3. Condenser Cooling Fan:

The heat generated within the compressor has to be thrown out else the compressor will get too hot in the long run and its motor coils will burn leading to complete breakdown of the compressor and the whole air conditioner. Further, the refrigerant within the condenser coil has to be cooled so that after expansion its temperature become low enough to produce the cooling effect. The condenser cooling fan is an ordinary fan with three or four blades and is driven by a motor. The cooling fan is located in front of the compressor and the condenser coil. As the blades of the fan rotate it absorbs the surrounding air from the open space and blows it over the compressor and the condenser with the aluminum fins thus cooling them. The hot air is thrown back to the open space and the circulation of air continues unhindered.

4. Expansion Valve:

The expansion valve is usually a copper capillary tubing with several rounds of coils. In the split air conditioners of bigger capacities thermostatic expansion valve is used which is operated electronically automatically. The high pressure and medium temperature refrigerant leaves the condenser and enters the expansion valve, where its temperature and pressure drops suddenly.

Refrigerant Piping or Tubing

The refrigerant piping is made up of copper tubing and it connects the indoor and the outdoor unit (see images above). The refrigerant at low temperature and low pressure leaves the expansion valve and enters the copper tubing, which is connected to the evaporator or the cooling coil at the other end.

The distance between the indoor and the outdoor unit can be short or long depending on the distance at which the open space is available in the home or office building. The longer the distance longer is the refrigerant piping between the two. When the refrigerant flows from the indoor unit to the outdoor unit in the tubing there is some loss of the cooling effect on the way, hence the distance between the indoor and the outdoor unit should be kept as minimum as possible. For the distance up to 15 meters there is not much appreciable loss of the cooling effect, however beyond that the losses become higher.

The refrigerant inside the tubing is at very low temperature and length of piping between and outdoor unit and indoor unit is quite long. Further, the tubing is exposed to the open atmosphere which is at very high temperature. Due to this, if the tubing is left uncovered all the cooling effect will be lost to the open atmosphere and by the time the refrigerant enters the cooling coil its temperature will already be too high and the purpose of producing the cooling effect will not be served. To avoid this, the refrigerant tubing connecting the indoor and the outdoor unit is covered with the insulation. This prevents the loss of the cooling effect to the atmosphere and low temperature refrigerant will produce the desired cooling effect inside the room.

After producing the cooling effect inside the room in the indoor unit, the refrigerant has to comeback to the outdoor unit for getting compressed and re-circulating. There is another refrigerant tubing that connects the indoor and the outdoor unit so that the refrigerant can travel from cooling coil back to the compressor. This tubing is also covered with insulation so that the refrigerant enters the compressor at minimum possible temperature to increase the refrigeration efficiency of the air conditioner. Thus there are two tubing connecting the indoor and the outdoor unit and both are covered with the insulation tape.

The refrigerant tubing are made up of copper since it is highly ductile and malleable element. The tubing can be easily manufactured from this material and they are flexible enough so they can turned into angles and coiled easily. The copper tubing used for condenser and evaporator facilitate high rate of heat conduction.

Wall Mounted Indoor Unit

It is the indoor unit that produces the cooling effect inside the room. The indoor unit of the split air conditioner is a box type housing in which all the important parts of the air conditioner are enclosed. The most common type of the indoor unit is the wall mounted type though other types like ceiling mounted and floor mounted are also used. We shall discuss all these types in separate articles, here we shall discuss the wall mounted type of the indoor unit.

These days the companies give utmost importance to the looks and aesthetics of the indoor unit. In the last couple few years the purpose of the indoor unit has changed from being a mere cooling effect producing devise to a beautiful looking cooling devise adding to the overall aesthetics of the room. This is one of the major reasons that the popularity of the split units has increased tremendously in the last few years. Let us see the various parts enclosed inside the indoor unit of the split air conditioner:

1. Evaporator Coil or the Cooling Coil:

The cooling coil is a copper coil made of number turns of the copper tubing with one or more rows depending on the capacity of the air conditioning system. The cooling coil is covered with the aluminum fins so that the maximum amount of heat can be transferred from the coil to the air inside the room.

The refrigerant from the tubing at very low temperature and very low pressure enters the cooling coil. The blower absorbs the hot room air or the atmospheric air and in doing so the air passes over the cooling coil which leads to the cooling of the air. This air is then blown to the room where the cooling effect has to be produced. The air, after producing the cooling effect is again sucked by the blower and the process of cooling the room continues.

After absorbing the heat from the room air, the temperature of the refrigerant inside the cooling coil becomes high and it flows back through the return copper tubing to the compressor inside the outdoor unit. The refrigerant tubing supplying the refrigerant from the outdoor unit to the indoor unit and that supplying the refrigerant from indoor unit to the outdoor unit are both covered with the insulation tape.

2. Air Filter:

The air filter is very important part of the indoor unit. It removes all the dirt particles from the room air and helps supplying clean air to the room. The air filter in the wall mounted type of the indoor unit is placed just before the cooling coil. When the blower sucks the hot room air, it is first passed through the air filter and then though the cooling coil. Thus the clean air at low temperature is supplied into the room by the blower.

3. Cooling Fan or Blower:

Inside the indoor unit there is also a long blower that sucks the room air or the atmospheric air. It is an induced type of blower and while is sucks the room air it is passed over the cooling coil and the filter due to which the temperature of the air reduces and all the dirt from it is removed. The blower sucks the hot and unclean air from the room and supplies cool and clean air back. The shaft of the blower rotates inside the bushes and it is connected to a small multiple speed motor, thus the speed of the blower can be changed. When the fan speed is changed with the remote it is the speed of the blower that changes.

4. Drain Pipe:

Due to the low temperature refrigerant inside the cooling coil, its temperature is very low, usually much below the dew point temperature of the room air. When the room air is passed over the cooling due the suction force of the blower, the temperature of the air becomes very low and reaches levels below its dew point temperature. Due to this the water vapor present in the air gets condensed and dew or water drops are formed on the surface of the cooling coil. These water drops fall off the cooling coil and are collected in a small space inside the indoor unit. To remove the water from this space the drain pipe is connected from this space extending to the some external place outside the room where water can be disposed off. Thus the drain pipe helps removing dew water collected inside the indoor unit.

To remove the water efficiently the indoor unit has to be a tilted by a very small angle of about 2 to 3 degrees so that the water can be collected in the space easily and drained out. If this angle is in opposite direction, all the water will get drained inside the room. Also, the if the tilt angle is too high, the indoor unit will shabby inside the room.

5. Louvers or Fins:

The cool air supplied by the blower is passed into the room through louvers. The louvers help changing the angle or direction in which the air needs to be supplied into the room as per the requirements. With louvers one easily change the direction in which the maximum amount of the cooled air has to be passed.

There are two types of louvers: horizontal and vertical. The horizontal louvers are connected to a small motor and there position can set by the remote control. Once can set a fixed position for the horizontal louvers so that chilled air is passed in a particular direction only or one can keep it in rotation mode so that the fresh air is supplied throughout the room. The vertical louvers are operated manually and one can easily change their position as per the requirements. The horizontal louvers control flow of air in upper and downward directions of the room, while vertical louvers control movement of air in left and right directions.

Floor Air Conditioner

Floor air conditioner is another type of air conditioner unit that is commonly used in places such as restaurants, halls, motels, data centers. The reason for its name is obvious in that it is basically standing on the floor. Other types of indoor units are the wall mounted, ducted, ceiling exposed, portable and window.

These fan coil units are all located indoor. The outdoor unit or condensers are located outside the building where the heat is rejected through the vapor compression cycle.

These two units are all that are required for the air conditioning system to function. They are connected to each other through the cooper tubes which are the gas and liquid lines.

The cooling capacity of the floor air conditioner can range from 24,000 Btu/hr to 200,000 Btu/hr depending on the model. There is also the **inverter** and non-inverter type with the inverter type commanding a better performance but more costly. Here are some of the brands that you may encounter for this type of unit.

As with many air conditioners equipment, you have a choice of the type of refrigerant that is being used.

R22 is the older and most cost effective but this refrigerant is not ozone-friendly and its production has been stopped. Due to recycling program, this refrigerant is still available but will be totally phase out by 2030.

The more ozone-friendly **refrigerant** include R407C and R410A. Many manufacturers are producing these air conditioners and selling them in the market.

However, it was discovered that these refrigerants can cause global warming. A newer refrigerant that has less global warming effect is the **R32**.

Ionizer

The ionizer feature is offered by some brands. These ionizer feature helps to get rid of pollen, dust and other particles to create a cleaner air in your room.

Auto-Restart Feature

This feature is widely available in most units but it is good to confirm its availability on the floor air conditioner unit that you have chosen. It basically remembers most of the settings that you set before the power failure occurred. Once the power is restored, it will be operating in its previous settings without you having to set it all over again.

Sound Pressure

Check the specifications of the **sound level** emitted by the indoor unit and outdoor unit. The indoor unit's sound level will be heard by the people in the room and should be as low as possible for the same capacity and fan speed. For instance, the maximum sound pressure could be 52dBA and minimum of 47dBA for a 24,000 Btu/hr unit.

Column air conditioner

Floor mounted cabinet air conditioner

FLOOR MOUNTED AIR CONDITIONER

Floor mounted split air conditioners can really be subdivided in to two types. Firstly **column air conditioners** which are large, high-capacity units (up to around 45000 BTU) used where a large room is to be cooled and where there may be building reasons why several smaller outlets cannot be used. Typical applications include lobbies, reception and waiting areas. The high output of these air conditioners means that they produce a strong flow of cool air which does not allow occupants to be in close proximity to the air conditioner.

Secondly, there are smaller cabinet style air conditioners which are far smaller, more like the dimensions of a storage heater than a tall upright freezer dimensions of a column air conditioner. Typically their rated capacities are up to about 15000BTU, and they are ideal for providing high-efficiency climate control to new extensions and conservatories. These both types of air conditioners are typically installed by refrigeration engineers as they do not usually have quick connect pipes, therefore the pipe work and cabling between the inside and outside elements is installed bespoke to the building and then the system is charged with refrigerant.

CEILING CASSETTE AIR CONDITIONER

Commonplace in offices with suspended ceilings, the ceiling cassette air conditioner, sometimes known as a cartridge air conditioner, is usually designed to be fitted within a one or two ceiling tile spaces. The bulk of the unit is unseen as it is above the ceiling line and the only visible part is the decorative lower facing with its central inlet grille and 4 edge outlet louvers. The main advantage of these units is aesthetics, but also that a centrally mounted unit can deliver an increased cooling (or heating) capacity across a wide area because of the air being distributed in 4 directions. Typically, a single ceiling cassette air conditioner can do the same job as 3 or 4 wall mounted units.

There is another type of ceiling air conditioner, which is an **under ceiling air conditioner** These are used where there is no suspended ceiling to install a cassette and where there is sufficient ceiling height to suspend an under ceiling unit. As these are designed to be entirely within a room, they are made to be reasonably aesthetic, however, the under ceiling air conditioners do inevitably look like overly cumbersome items to hang from a ceiling. Generally they are designed to lift air vertically into the unit and discharge treated air horizontally along the ceiling avoiding direct discharge directly onto occupants, and some allow air discharge from four sides.

Ceiling cassette air conditioner

Advantages

The main advantages of mini splits are their small size and flexibility for zoning or heating and cooling individual rooms. Many models can have as many as four indoor air handling units (for four zones or rooms) connected to one outdoor unit. The number depends on how much heating or cooling is required for the building or each zone (which in turn is affected by how well the building is insulated). Each of the zones will have its own thermostat, so you only need to condition that space when it is occupied, saving energy and money.

Ductless mini split systems are also often easier to install than other types of space conditioning systems. For example, the hook-up between the outdoor and indoor units generally requires only a three-inch (~8 centimeter [cm]) hole through a wall for the conduit. Also, most manufacturers of this type of system can provide a variety of lengths of connecting conduits. So, if necessary, you can locate the outdoor unit as far away as 50 feet (~15 meters [m]) from the indoor evaporator. This makes it possible to cool rooms on the front side of a building house with the compressor in a more advantageous or inconspicuous place on the outside of the building.

Since mini splits have no ducts, they avoid the energy losses associated with ductwork of central forced air systems. Duct losses can account for more than 30% of energy consumption for space conditioning, especially if the ducts are in an unconditioned space such as an attic.

Compared with other add-on systems, mini splits offer more flexibility in interior design options. The indoor air handlers can be suspended from a ceiling, mounted flush into a drop ceiling, or hung on a wall. Floor-standing models are also available. Most indoor units have profiles of about seven inches (~18 cm) deep and usually come with sleek, high-tech-looking jackets. Many also offer a remote control to make it easier to turn the system on and off when it's positioned high on a wall or suspended from a ceiling. Split-systems can also help to keep your home safer, because there is only a small hole in the wall. Through-the-wall and window mounted room air-conditioners can provide an easy entrance for intruders.

Disadvantages

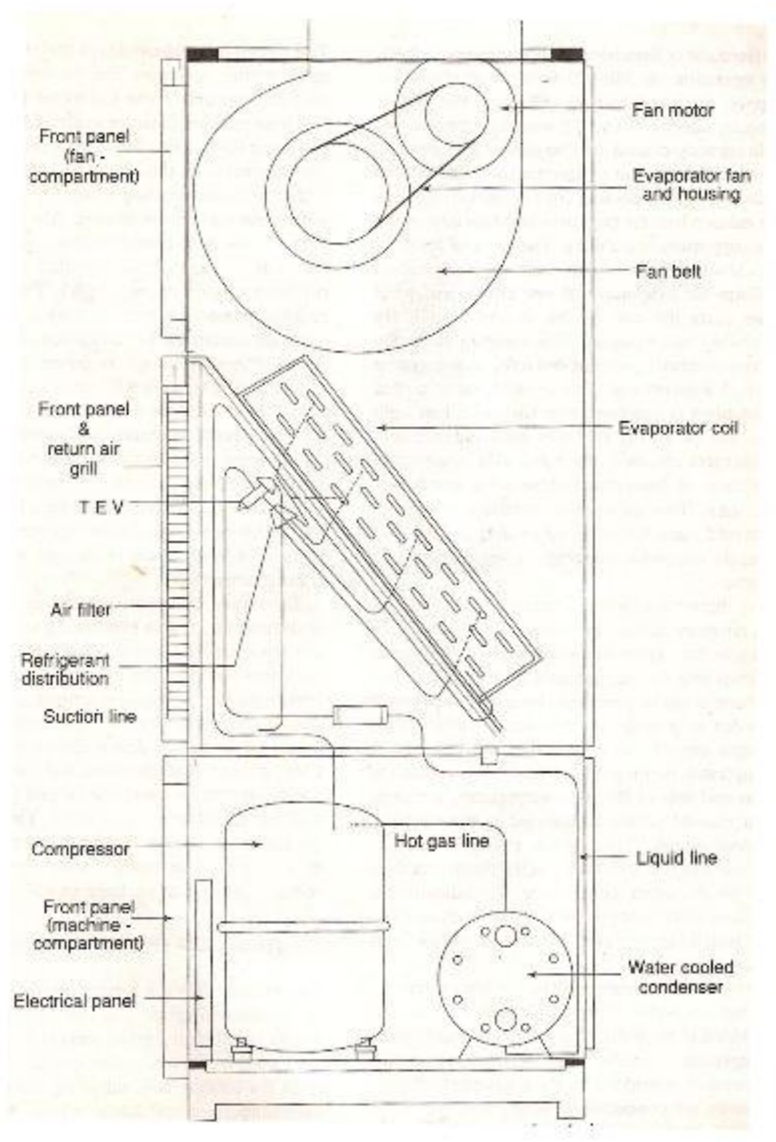
The primary disadvantage of mini splits is their cost. Such systems cost about $1,500 to $2,000 per ton (12,000 Btu per hour) of cooling capacity. This is about 30% more than central systems (not including ductwork) and may cost twice as much as window units of similar capacity.

The installer must also correctly size each indoor unit and judge the best location for its installation. Oversized or incorrectly located air-handlers often result in short-cycling, which wastes energy and does not provide proper temperature or humidity control. Too large a system is also more expensive to buy and operate.

Some people may not like the appearance of the indoor part of the system. While less obtrusive than a window room air conditioner, they seldom have the built-in look of a central system. There must also be a place to drain condensate water near the outdoor unit.

Qualified installers and service people for mini splits may not be easy to find. In addition, most conventional heating and cooling contractors have large investments in tools and training for sheet metal duct systems. They need to use (and charge for) these to earn a return on their investment, so they may not recommend ductless systems except where a ducted system would be difficult for them to install.

Packaged Air Conditioners



The window and split air conditioners are usually used for the small air conditioning capacities up to 5 tons. The central air conditioning systems are used for where the cooling loads extend beyond 20 tons. The packaged air conditioners are used for the cooling capacities in between these two extremes. The packaged air conditioners are available in the fixed rated capacities of 3, 5, 7, 10 and 15 tons. These units are used commonly in places like restaurants, telephone exchanges, homes, small halls, etc.

As the name implies, in the packaged air conditioners all the important components of the air conditioners are enclosed in a single casing like window AC. Thus the compressor, cooling coil, air handling unit and the air filter are all housed in a single casing and assembled at the factory location.

Depending on the type of the cooling system used in these systems, the packaged air conditioners are divided into two types: ones with water cooled condenser and the ones with air cooled condensers. Both these systems have been described below:

Packaged Air Conditioners with Water Cooled Condenser

In these packaged air conditions the condenser is cooled by the water. The condenser is of shell and tube type, with refrigerant flowing along the tube side and the cooling water flowing along the shell side. The water has to be supplied continuously in these systems to maintain functioning of the air conditioning system.

The shell and tube type of condenser is compact in shape and it is enclosed in a single casing along with the compressor, expansion valve, and the air handling unit including the cooling coil or the evaporator. This whole packaged air conditioning unit externally looks like a box with the control panel located externally.

In the packaged units with the water cooled condenser, the compressor is located at the bottom along with the condenser (refer the figure below). Above these components the evaporator or the cooling coil is located. The air handling unit comprising of the centrifugal blower and the air filter is located above the cooling coil. The centrifugal blower has the capacity to handle large volume of air required for cooling a number of rooms. From the top of the package air conditioners the duct comes out that extends to the various rooms that are to be cooled.

All the components of this package AC are assembled at the factory site. The gas charging is also done at the factory thus one does not have to perform the complicated operations of the laying the piping, evacuation, gas charging, and leak testing at the site. The unit can be transported very easily to the site and is installed easily on the plane surface. Since all the components are assembled at the factory, the high quality of the packaged unit is ensured.

Package AC with Water Cooled Condenser

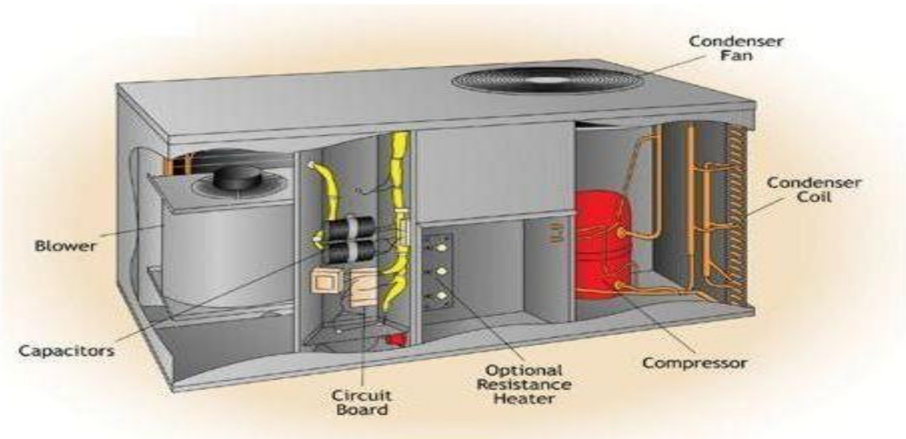
Packaged Air Conditioners with Air Cooled Condensers

In this packaged air conditioners the condenser of the refrigeration system is cooled by the atmospheric air. There is an outdoor unit that comprises of the important components like the compressor, condenser and in some cases the expansion valve (refer the figure below).

The outdoor unit can be kept on the terrace or any other open place where the free flow of the atmospheric air is available. The fan located inside this unit sucks the outside air and blows it over the condenser coil cooling it in the process. The condenser coil is made up of several turns of the copper tubing and it is finned externally. The packaged ACs with the air cooled condensers are used more commonly than the ones with water cooled condensers since air is freely available it is difficult maintain continuous flow of the water.

The cooling unit comprising of the expansion valve, evaporator, the air handling blower and the filter are located on the floor or hanged to the ceiling. The ducts coming from the cooling unit are connected to the various rooms that are to be cooled.

Package Air Conditioner Air Cooled Condenser



Central Air-Conditioning Plants

The central air conditioning plants or the systems are used when large buildings, hotels, theaters, airports, shopping malls etc are to be air conditioned completely. The window and split air conditioners are used for single rooms or small office spaces. If the whole building is to be cooled it is not economically viable to put window or split air conditioner in each and every room. Further, these small units cannot satisfactorily cool the large halls, auditoriums, receptions areas etc.

In the central air conditioning systems there is a plant room where large compressor, condenser, thermostatic expansion valve and the evaporator are kept in the large plant room. They perform all the functions as usual similar to a typical refrigeration system. However, all these parts are larger in size and have higher capacities. The compressor is of open reciprocating type with multiple cylinders and is cooled by the water just like the automobile engine. The compressor and the condenser are of shell and tube type. While in the small air conditioning system capillary is used as the expansion valve, in the central air conditioning systems thermostatic expansion valve is used.

The chilled is passed via the ducts to all the rooms, halls and other spaces that are to be air conditioned. Thus in all the rooms there is only the duct passing the chilled air and there are no individual cooling coils, and other parts of the refrigeration system in the rooms. What is we get in each room is the completely silent and highly effective air conditions system in the room. Further, the amount of chilled air that is needed in the room can be controlled by the openings depending on the total heat load inside the room.

The central air conditioning systems are highly sophisticated applications of the air conditioning systems and many a times they tend to be complicated. It is due to this reason that there are very few companies in the world that specialize in these systems. In the modern era of computerization a number of additional electronic utilities have been added to the central conditioning systems.

There are two types of central air conditioning plants or systems:

1. **Direct expansion or DX central air conditioning plant**: In this system the huge compressor, and the condenser are housed in the plant room, while the expansion valve and the evaporator or the cooling coil and the air handling unit are housed in separate room. The cooling coil is fixed in the air handling unit, which also has large blower housed in it. The blower sucks the hot return air from the room via ducts and blows it over the cooling coil. The cooled air is then supplied through various ducts and into the spaces which are to be cooled. This type of system is useful for small buildings.
2. **Chilled water central air conditioning plant:** This type of system is more useful for large buildings comprising of a number of floors. It has the plant room where all the important units like the compressor, condenser, throttling valve and the evaporator are housed. The evaporator is a shell and tube. On the tube side the Freon fluid passes at extremely low temperature, while on the shell side the brine solution is passed. After passing through the evaporator, the brine solution gets chilled and is pumped to the various air handling units installed at different floors of the building. The air handling units comprise the cooling coil through which the chilled brine flows, and the blower. The blower sucks hot return air from the room via ducts and blows it over the cooling coil. The cool air is then supplied to the space to be cooled through the ducts. The brine solution which has absorbed the room heat comes back to the evaporator, gets chilled and is again pumped back to the air handling unit.

-Variable Refrigerant Volume (VRV)/ Variable Refrigerant Flow (VRF)

Variable refrigerant flow (VRF), also known as variable refrigerant volume (VRV).

VRFs are typically installed with an air conditioner inverter which adds a DC inverter to the compressor in order to support variable motor speed and thus variable refrigerant flow rather than simply perform on/off operation. By operating at varying speeds, VRF units work only at the needed rate allowing for substantial energy savings at load conditions. Heat recovery VRF technology allows individual indoor units to heat or cool as required, while the compressor load benefits from the internal heat recovery. Energy savings of up to 55%

are predicted over comparable unitary equipment.[1] [3] This also results in greater control of the building's interior temperature by the building's occupants.

VRFs come in two system formats, two pipe and three pipe systems. In a heat pump two pipe system all of the zones must either be all in cooling or all in heating. Heat Recovery (HR) systems have the ability to simultaneously heat certain zones while cooling others; this is usually done through a three pipe design, with the exception of Mitsubishi and Carrier, whose systems are able to do this with a two pipe system using a branch circuit (BC) controller to the individual indoor evaporator zones. In this case the heat extracted from zones requiring cooling is put to use in the zones requiring heating. This is made possible because the heating unit is functioning as a condenser, providing sub-cooled liquid back into the line that is being used for cooling. While the heat recovery system has a greater initial cost, it allows for better zoned thermal control of a building and overall greater efficiencies.[4] In heat recovery VRF systems, some of the indoor units may be in cooling mode while others are in heating mode, reducing energy consumption. If the coefficient of performance in cooling mode of a system is 3, and the coefficient of performance in heating mode is 4, then heat recovery performance can reach more than 7. While it is unlikely that this balance of cooling and heating demand will happen often throughout the year, energy efficiency can be greatly improved when the scenario occurs.[5]

VRF systems may be air or water cooled. If air cooled, VRF condensing units are exposed to outside air and may be outdoors, and condensing units are the size of large refrigerators, since they need to contain a large condenser (heat exchanger) to transfer heat to the surrounding air, because air doesn't have a high heat capacity. If water cooled, the condensing units are placed indoors and are much smaller and cooled, using water, possibly by a cooling tower.

INDUSTRIAL APPLICATIONS

Air-conditioning is an important part of human society. Day by day its the environment that we live-in is in verge of pullution overtake. It’s important to us that we breath good conditioned air. Air conditioners have following applications.

1. Air conditioning can be defined as conditioning the air for a natural and comfortable atmosphere within the living area particularly in our home or office.
2. Filtering air for dust particles, mould, insects, and much other micro organism living in air.
3. Employed in large super computer halls to small desktops rooms for keeping them their cool and for their prolonged working.
4. Constant temperature is to be maintained in tool room as you know metal are not so trusted with changing temperature for their dimensions.
5. Air conditioning helps the shop owners for a good sale. Or may be that is why we hang out in malls more often in college days.
6. Air conditioner keeps the boss cool in office. Just imagine what it would like a sweaty red boss.
7. Did i said that some of the Air conditioner breed can maintain the humity level.
8. Air conditioner can keep your food fresh for a little longer. Some times even for 2 year in cold rooms.
9. Air conditioner is used in Operation theatre so that patients could get well soon other than getting his wounds septic due to the micro organisms present in air, which as stated, in a condition without AC.
10. Air conditioning keeps your office toilet smell free its 80% fresh air for every intake air.
11. Did i mentioned that AC's are used in car and other such vehicle for the confort of the rider. Aeroplane too.
12. Testing rooms are generally air conditioned.
13. Parking lot needs Air conditioning. That doesn't mean to cool the air but just to circulate the air. A fan specificly exhaust fan will do the work.
14. Air conditioning of stair well.
15. Air conditioning in airports and bus stand for the comfort of passengers.

TUTORIAL QUESTIONS

1. Explain detail classification of Air Conditioning System?
2. Explain with neat sketch working of window air-conditioning?
3. Difference between the window and split air-conditioning?
4. Explain with neat sketch working of packaged Air Conditioning System ?
5. Explian the working of Split A/c System with neat diagram?
6. Write the Merits and Demerits of Split a/c?
7. Describe briefly about Variable Refrigerant Volume (VRV)/ Variable Refrigerant Flow (VRF)?
8. Write Down the applications of ductable ac

UNIT III

STUDY OF PSYCHROMETRIC CHARTS

COURSE OBJECTIVE: To Study Psychrometric Chart and various terminology.

COURSE OUTCOME: To be able to study and analyze psychrometric chart in refrigeration systems. Develop problem solving skills through the application of thermodynamics

CONCEPT OF PSYCHROMETRY AND PSYCHROMETRICS

Air comprises of fixed gases principally, nitrogen and oxygen with an admixture of water vapour in varying amounts. In atmospheric air water is always present and its relative weight averages less than 1% of the weight of atmospheric air in temperate climates and less than 3% by weight under the most extreme natural climatic conditions, it is nevertheless one of most important factors in human comfort and has significant effects on many materials. Its effect on human activities is in fact altogether disproportionate to its relative weights. The art of measuring the moisture content of air is termed **psychrometry** . The science which investigates the thermal properties of moist air, considers the measurement and control of the moisture content of air, and studies the effect of atmospheric moisture on material and human comfort may properly be termed **psychrometrics'**'.

DEFINITIONS

Some of the more important definitions are given below :

1. **Dry air.** The international joint committee on Psychrometric Data has adopted the following exact composition of air expressed in mole fractions (Volumetric) Oxygen 0.2095, Nitrogen 0.7809, Argon 0.0093, Carbon dioxide 0.0003. Traces of rare gases are neglected. Molecular weight of air for all air conditioning calculations will be taken as 28.97. Hence the gas constant,

$$R_{air} = \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K}$$

Dry air is never found in practice. Air always contains some moisture. Hence the common designationair usually means moist air. The termdry air' is used to indicate the water free contents of air having any degree of moisture.

2. **Saturated air.** Moist air is said to be saturated when its condition is such that it can co-exist in natural equilibrium with an associated condensed moisture phase presenting a flat surface to it. For a given temperature, a given quantity of air can be saturated with a fixed quantity of moisture. At higher temperatures, it requires a larger quantity of moisture to saturate it. At saturation, vapour pressure of moisture in air corresponds to the saturation pressure given in steam tables corresponding to the given temperature of air.

3. **Dry-bulb temperature (DBT).** It is the temperature of air as registered by an ordinary thermometer (t_{db}).

4. **Wet-bulb temperature (WBT).** It is the temperature registered by a thermometer when the bulb is covered by a wetted wick and is exposed to a current of rapidly moving air (t_{wb}).

5. **Adiabatic saturation temperature.** It is the temperature at which the water or ice can saturate air by evaporating adiabatically into it. It is numerically equivalent to the measured wet bulb temperature (as corrected, if necessary for radiation and conduction) (t_{db}– t_{wb}).

6. **Wet bulb depression.** It is the difference between dry-bulb and wet bulb temperatures.

7. **Dew point temperature (DPT).** It is the temperature to which air must be cooled at constant pressure in order to cause condensation of any of its water vapour. It is equal to

steam table saturation temperature corresponding to the actual partial pressure of water vapour in the air (t_{dp}).

8. **Dew point depression.** It is the difference between the dry bulb and dew point temperatures (t_{db} – t_{dp}).

9. **Specific humidity (Humidity ratio).** It is the ratio of the mass of water vapour per unit mass of dry air in the mixture of vapour and air, it is generally expressed as grams of water per kg of dry air. For a given barometric pressure it is a function of dew point temperature alone.

10. **Relative humidity (RH), ().** It is the ratio of the partial pressure of water vapour in the mixture to the saturated partial pressure at the dry bulb temperature, expressed as percentage.

11. **Sensible heat.** It is the heat that changes the temperature of a substance when added to or abstracted from it.

12. **Latent heat.** It is the heat that does not affect the temperature but changes the state of substance when added to or abstracted from it.

13. **Enthalpy.** It is the combination energy which represents the sum of internal and flow energy in a steady flow process. It is determined from an arbitrary datum point for the air mixture and is expressed as kJ per kg of dry air (h).

Note. When air is saturated DBT, WBT, DPT are equal.

PSYCHROMETRIC RELATIONS

Pressure

Dalton's law of partial pressure is employed to determine the pressure of a mixture of gases.

This law states that the total pressure of a mixture of gases is equal to the sum of partial pressures which the component gases would exert if each existed alone in the mixture volume at the mixture temperature. Precise measurements made during the last few years indicate that this law as well as Boyle‘s and Charle‘s laws are only approximately correct. Modern tables of atmospheric air properties are based on the correct versions. For calculating partial pressure of water vapour in the air many equations have been proposed, probably Dr.Carrier‘s equation is most widely used.

$$p_v = (p_{vs})_{wb} - \frac{[pt - (p_{vs})_{wb}](t_{db} - t_{wb})}{1527.4 - 1.3\, t_{wb}}$$

where

p_v = Partial pressure of water vapour,

p_{vs}= Partial pressure of water vapour when air is

fully saturated, p_t = Total pressure of moist air,

t_{db}= Dry bulb temperature (ºC), and

t_{wb}= Wet bulb temperature (ºC).

Specific humidity W:

$$\text{Specific humidity} = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}}$$

$$W = \frac{m_v}{m_a}$$

$$\text{Also,} \quad m_a = \frac{p_a V}{R_a T}$$

$$m_v = \frac{p_v \times V}{R_v \times T}$$

Where,

p_a= Partial pressure of dry air,

p_v= Partial pressure of water vapour,

V= Volume of mixture,

R_a= Characteristic gas constant for dry air, and

R_v= Characteristic gas constant for water vapour.

$$W = \frac{p_v \times V}{R_v \times T} \times \frac{R_a\,T}{p_a\,V} = \frac{R_a}{R_v} \times \frac{p_v}{p_a}$$

$$R_a = \frac{R_0}{M_a} \qquad \left(= \frac{8.3143}{28.97} = 0.287 \text{ kJ/kg K in SI units}\right)$$

$$R_v = \frac{R_0}{M_v} \qquad \left(= \frac{8.3143}{18} = 0.462 \text{ kJ/kg K in SI units}\right)$$

Wher
e

R_0 = Universal gas constant,
M_a=Molecular weight of air, and
M_v = Molecular weight of water vapour.

$$W = \frac{0.287}{0.462} \cdot \frac{p_v}{p_a} = 0.622 \frac{p_v}{p_t - p_v}$$

$$W = 0.622 \frac{p_v}{p_t - p_v}$$

The masses of air and water vapour in terms of specific volumes are given by expression as

$$m_a = \frac{V}{v_a} \qquad \text{and} \qquad m_v = \frac{V}{v_v}$$

Where

$_a$= Specific volume of dry air, and
= Specific volume of water vapour.

$$W = \frac{v_a}{v_v}$$

Degree of saturation (μ):

$$\text{Degree of saturation} = \frac{\text{Mass of water vapour associated with unit mass of dry air}}{\text{Mass of water vapour associated with saturated unit mass of dry saturated air}}$$

$$\mu = \frac{W}{W_s}$$

W_s = Specific humidity of air when air is fully saturated

$$\mu = \frac{0.622\left(\dfrac{p_v}{p_t - p_v}\right)}{0.622\left(\dfrac{p_{vs}}{p_t - p_{vs}}\right)} = \frac{p_v(p_t - p_{vs})}{p_{vs}(p_t - p_v)}$$

$$= \frac{p_v}{p_s}\left[\frac{\left(1 - \dfrac{p_{vs}}{p_t}\right)}{\left(1 - \dfrac{p_v}{p_t}\right)}\right]$$

Where

p_{VS} = Partial pressure of water vapour when air is fully saturated (p_{VS} can be calculated from steam tables corresponding to the dry bulb temperature of the air).

Relative humidity (RH) :

$$\text{Relative humidity, } \phi = \frac{\text{Mass of water vapour in a given volume}}{\text{Mass of water vapour in the same volume if saturated at the same temp.}}$$

$$= \frac{m}{m_{vs}} = \frac{\dfrac{p_v T}{R_v T}}{\dfrac{p_{vs} T}{R_v T}} = \frac{p_v}{p_{vs}}$$

$$\phi = \frac{p_a W}{0.622} \times \frac{1}{p_{vs}} = 1.6\ W\ \frac{p_a}{p_{vs}}$$

Note 1. Relative humidity as compared to specific humidity plays a vital role in comfort air-conditioning and industrial air-conditioning. Relative humidity signifies the absorption capacity of air. If initial relative humidity of air is less it will absorb more moisture.

Note **2.** W, μ and cannot be conveniently measured as they require measurement of p_V and p_{VS}. The value of p_V can be obtained from the measurement of the wet bulb temperature and the value of p_{VS} can be calculated from steam tables corresponding to given air temperature.

Enthalpy of moist air

It is the sum of enthalpy of dry air and enthalpy of water vapour associated with dry air. It is expressed in kJ/kg of dry air.

$$h = h_{air} + W \cdot h_{vapour}$$
$$= c_p t_{db} + W \cdot h_{vapour}$$

where h = Enthalpy of mixture/kg of dry air,

h_{air} = Enthalpy of 1 kg of dry air,

h_{vapour} = Enthalpy of 1 kg of vapour obtained from steam tables,

W = Specific humidity in kg/kg of dry air, and

c_p = Specific heat of dry air normally assumed as 1.005 kJ/kg K.

Also $h_{vapour} = h_g + c_{ps}(t_{db} - t_{dp})$

where h_g = Enthalpy of saturated steam at dew point temperature,

and c_{ps} = 1.88 kJ/kg K.

$$\therefore \quad h = c_p t_{db} + W[h_g + c_{ps}(t_{db} - t_{dp})]$$
$$= (c_p + c_{ps} W)\, t_{db} + W(h_g - c_{ps} t_{dp})$$
$$= c_{pm} t_{db} + W(h_g - c_{ps} t_{dp})$$

Where $C_{pm} = (C_p + C_{ps} W)$ is the specific heat of humid air or humid specific heat.

The value of C_{pm} is taken as 1.021 kJ/kg dry air per K. It is the heat capacity of (1 + *W*) kg of moisture per kg of dry air.

$h_{vapour} = h_g$ at dry bulb temperature. So,

$$h = c_p t_{db} + W h_g.$$

However, a better approximation is given by the following relationship:

$$h_{vapour} = 2500 + 1.88 t_{db} \text{ kJ/kg of water vapour}$$

Where t_{db} is dry bulb temperature in ºC, and the datum state is liquid water at 0ºC.

$$h = 1.005\, t_{db} + W(2500 + 1.88\, t_{db}) \text{ kJ/kg dry air.}$$

PSYCHROMETRIC CHARTS

The psychrometric charts are prepared to represent graphically all the necessary moist air properties used for air conditioning calculations. The values are based on actual measurements verified for thermodynamic consistency.

For psychrometric charts the most convenient co-ordinates are dry bulb temperature of air vapour mixture as the abcissa and moisture content (kg/kg of dry air) or water vapour pressure as the ordinate. Depending upon whether the humidity contents are abcissa or ordinate with temperature co-ordinate, the charts are generally classified as Mollier chart and

Carrier chart. Carrier chart having t_{db} as the abcissa and W as the ordinate finds a wide application. The chart is constructed as under :

1. The dry bulb temperature (ºC) of unit mass of dry air for different humidity contents or humidity ratios are indicated by vertical lines drawn parallel to the ordinate.

2. The mass of water vapour in kg (or grams) per kg of dry air is drawn parallel to the abcissa for different values of dry bulb temperature. It is the major vertical scale of the chart.

3. Pressure of water vapour in mm of mercury is shown in the scale at left and is the absolute pressure of steam.

4. Dew point temperatures are temperatures corresponding to the boiling points of water at low pressures of water vapour and are shown in the scale on the upper curved line. The dew points for different low pressures are read on diagonal co-ordinates.

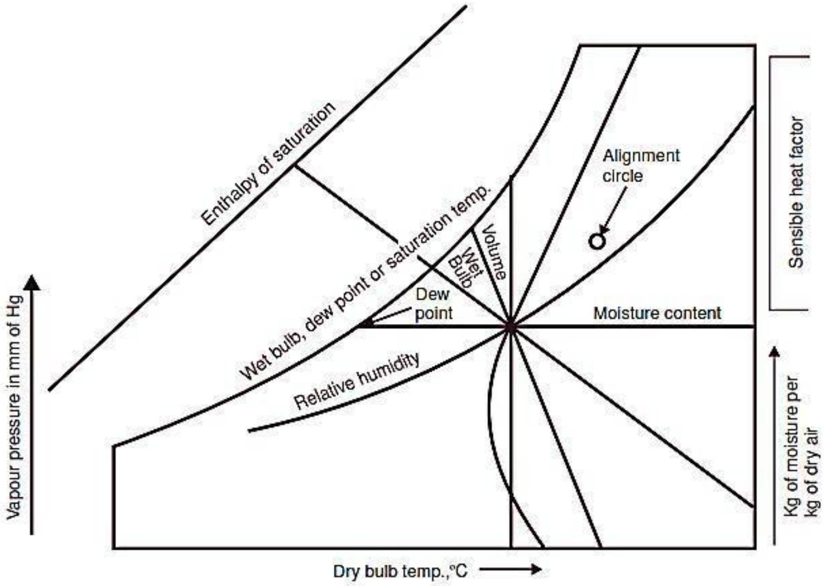


Fig.a. Skeleton psychrometric chart.

5. Constant relative humidity lines in per cent are indicated by marking off vertical distances between the saturation line or the upper curved line and the base of the chart. The relative humidity curve depicts quantity (kg) of moisture actually present in the air as a percentage of the total amount possible at various dry bulb temperatures and masses of vapour.

6. Enthalpy or total heat at saturation temperature in kJ/kg of dry air is shown by a diagonal system of co-ordinates. The scale on the diagonal line is separate from the body of the chart and is indicated above the saturation line.

7. Wet bulb temperatures are shown on the diagonal co-ordinates coinciding with heat coordinates.

The scale of wet bulb temperatures is shown on the saturation curve. The diagonals run downwards to the right at an angle of 30º to the horizontal.

8. The volume of air vapour mixture per kg of dry air (specific volume) is also indicated by a set of diagonal co-ordinates but at an angle of 60º with the horizontal.

The other properties of air vapour mixtures can be determined by using formulae (already discussed). In relation to the psychrometric chart, these terms can quickly indicate many things about the condition of air, for example:

1. If dry bulb and wet bulb temperatures are known, the relative humidity can be read from the chart.

2. If the dry bulb and relative humidity are known, the wet bulb temperature can be determined.

3. If wet bulb temperature and relative humidity are known, the dry bulb temperature can be found.

4. If wet bulb and dry bulb temperatures are known, the dew point can be found.

5. If wet bulb and relative humidity are known, dew point can be read from the chart.

6. If dry-bulb and relative humidity are known, dew point can be found.

7. The quantity (kg) of moisture in air can be determined from any of the following combinations:

(i) Dry bulb temperature and relative humidity;
(ii) Dry bulb temperature and dew point;
(iii) Wet bulb temperature and relative humidity;
(iv) Wet bulb temperature and dew point temperature;
(v) Dry bulb temperature and wet bulb temperature; and
(vi) Dew point temperature alone.

Figs. a and b show the skeleton psychrometric chart and lines on carrier chart respectively.

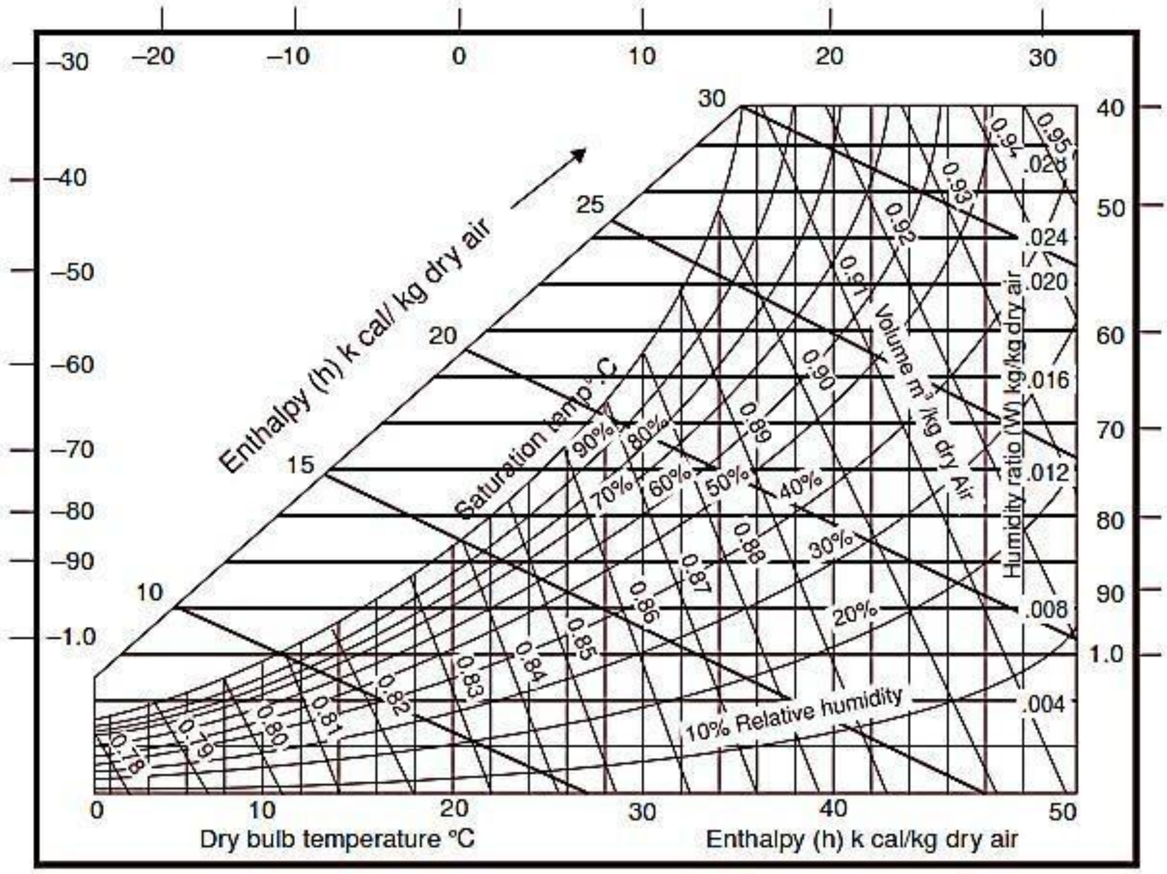


Fig b.. Carrier chart.

PSYCHROMETRIC PROCESSES

In order to condition air to the conditions of human comfort or of the optimum control of an industrial process required, certain processes are to be carried out on the outside air available. The processes affecting the psychrometric properties of air are called **psychrometric processes.** These processes involve mixing of air streams, heating, cooling, humidifying, dehumidifying, adiabatic saturation and mostly the combinations of these. The important psychrometric processes are enumerated and explained in the following text:

1. Mixing of air streams

2. Sensible heating

3. Sensible cooling

4. Cooling and dehumidification

5. Cooling and humidification

6. Heating and dehumidification

7. Heating and humidification.

Mixing of Air Streams

Refer Figs. C and D. Mixing of several air streams is the process which is very frequently used in air conditioning. This mixing normally takes place without the addition or rejection of either heat or moisture, i.e., adiabatically and at constant total moisture content. Thus we can write the following equations :

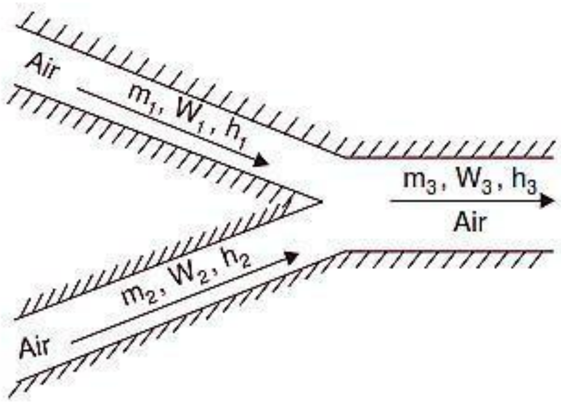
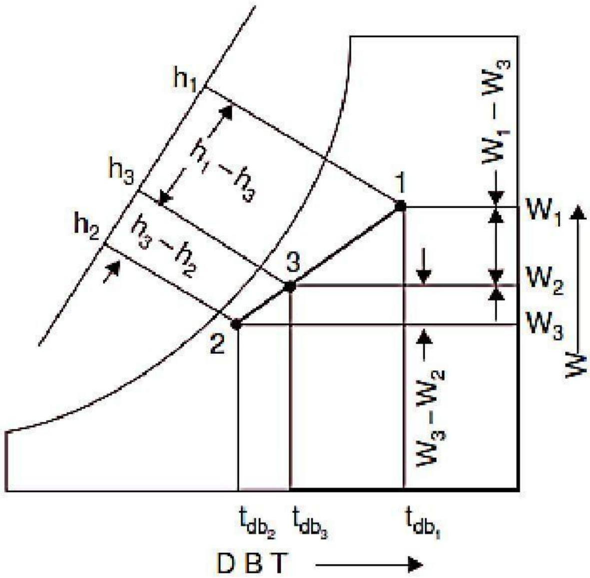


Fig. C. Mixing of air streams.

$$m_1 + m_2 = m_3$$
$$m_1W_1 + m_2W_2 = m_3W_3$$
$$m_1h_1 + m_2h_2 = m_3h_3$$



$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

Rearranging of last two equations gives the following:

$$m_1(W_1 - W_3) = m_2(W_3 - W_2)$$
$$m_1(h_1 - h_3) = m_2(h_3 - h_2)$$
$$\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3} = \frac{h_3 - h_2}{h_1 - h_3}$$

Where, m = Mass of dry air at particular state points W= Specific humidity at particular state points h = Enthalpy at particular state points

On the psychrometric chart, the specific humidity and enthalpy scales are linear, ignoring enthalpy deviations. Therefore, the final state 3 lies on a straight line connecting the initial states of the two streams before mixing, and the final state 3 divides this line into two parts that are in the same ratio as were the two masses of air before mixing. If the air quantities are known in volume instead of mass units, it is generally sufficiently accurate to units of m^3 or m^3/min. in the mixing equations. The inaccuracy introduced is due to the difference in specific volume at two initial states. This difference in densities is small for most of the comfort air conditioning problems.

Sensible Heating

When air passes over a dry surface which is at a temperature greater than its (air) dry bulb temperature, it undergoes sensible heating. Thus the heating can be achieved by passing the air over heating coil like electric resistance heating coils or steam coils. During such a process, the specific humidity remains constant but the dry bulb temperature rises and approaches that of the surface. The extent to which it approaches the mean effective surface temperature of the coil is conveniently expressed in terms of the equivalent **by-pass factor.**

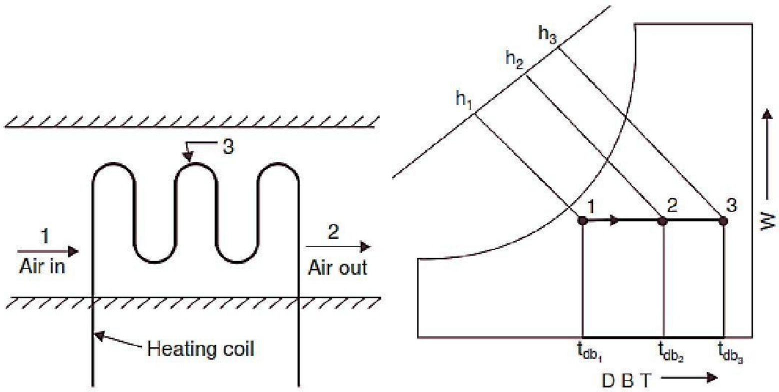
The by-pass factor (BF) for the process is defined as the ratio of the difference between the mean surface temperature of the coil and leaving air temperature to the difference between the mean surface temperature and the entering air temperature.

Thus on Fig. E, air at temperature t_{db1}, passes over a heating coil with an average surface temperature t_{db3} and leaves at temperature t_{db2}

The by-pass factor is expressed as follows :

Fig.E. Sensible heating

$$BF = \frac{t_{db_3} - t_{db_2}}{t_{db_3} - t_{db_1}}$$



The value of the by-pass factor is a function of coil design and velocity. The heat added to the air can be obtained directly from the entering and leaving enthalpies (h_2-h_1) or it can be obtained from the humid specific heat multiplied by the temperature difference (t_{db2}-t_{db1}). In a complete air conditioning system the preheating and reheating of air are among the familiar examples of sensible heating.

Note. By-pass factor' can be considered to represent the fraction of air which does not come into contact with coil surface.

Sensible Cooling

Refer Fig. G. Air undergoes *sensible cooling* whenever it passes over a surface that is at a temperature less than the *dry bulb* temperature of the *air but greater than the dew point temperature.* Thus sensible cooling can be achieved by passing the air over cooling coil like *evaporating coil of the refrigeration cycle or secondary brine coil.* During the process, *the specific humidity remains constant* and *dry bulb temperature decreases*, approaching the

mean effective surface temperature. On a psychrometric chart the process will appear as a horizontal line 1–2 (Fig. H), where point 3 represents the effective surface temperature. For this process:

$$\text{By-pass factor BF} = \frac{t_{db_2} - t_{db_3}}{t_{db_1} - t_{db_3}}$$

The heat removed from air can be obtained from the enthalpy difference (h_1-h_2) or from humid specific heat multiplied by the temperature difference (t_{db1}-t_{db2}).

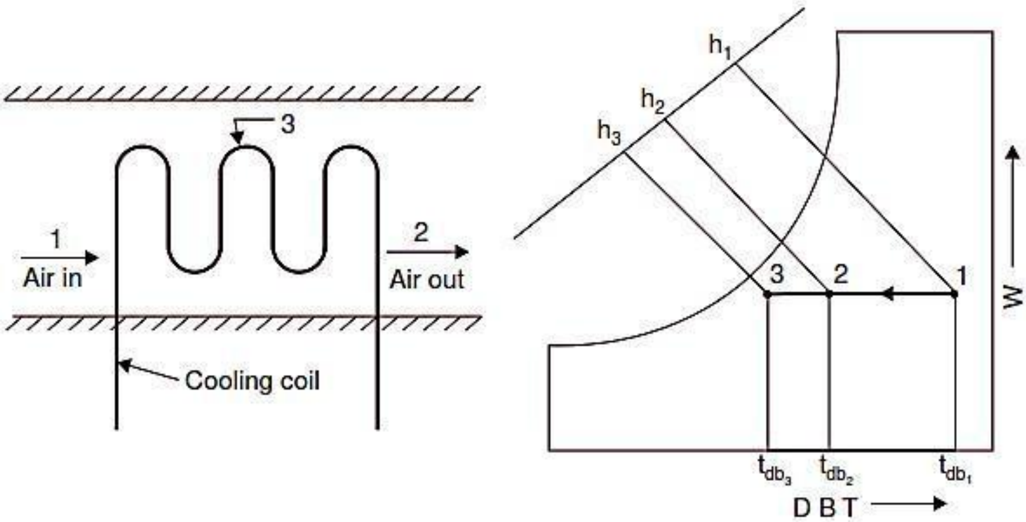


Fig.G. Sensible cooling.

Cooling and Dehumidification

Refer Fig. I. Whenever air is made to pass over a surface or through a spray of water that is at a temperature less than the dew point temperature of the air, condensation of some of the water vapour in air will occur simultaneously with the sensible cooling process. Any air that comes into sufficient contact with the cooling surface will be reduced in temperature to the mean surface temperature along a path such as 1-2-3 in Fig. I, with condensation and therefore dehumidification occurring between points 2 and 3. The air that does not contact the surface will be finally cooled by mixing with the portion that did, and the final state point will somewhere on the straight line connecting points 1 and 3. The actual path of air during the path will not be straight line shown but will be something similarly to the curved dashed line 1–4. It will result from a continuous mixing of air which is connecting a particular part of the coil and air which is by passing it. It is convenient, however to analyse the problem with the straight line shown, and to assume that the final air state results from the mixing of air that has completely by passed the coil with air that has been cooled to the mean effective surface temperature. If there is enough contact between air and surface for all the air to come to the mean surface temperature, the process is one of zero by pass. In any practical system, complete saturation is not obtained and final state will be a point such as 4 in Fig. I with an effective surface temperature, e.g t_{db3} in Fig. I is called **apparatus dew point**‘ (ADP). The final state point of air passing through a cooling and dehumidifying apparatus is in effect a mixture condition that results from mixing the fraction of the air, which is equal to the equivalent by-pass factor (BF) and is at initial state point and the remaining fraction which is equal to one minus by pass factor (1–BF) and is saturated at the apparatus dew point (ADP).

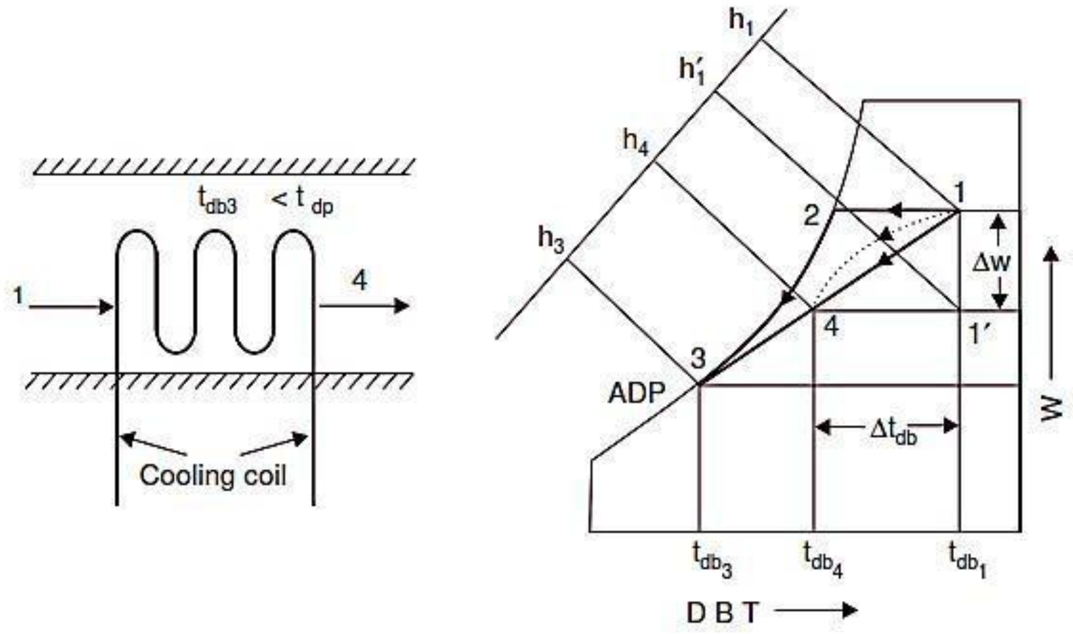


Fig. I Cooling and dehumidification

Total heat removed from the air is given by

$$Q_t = h_1 - h_4 = (h_1 - h_1') + (h_1' - h_4)$$
$$= Q_L + Q_S$$

where, W_L = Latent heat removed $(h_1 - h_1')$, and

Q_S = Sensible heat removed $(h_1' - h_4)$

The ratio $\frac{Q_S}{Q_L}$ *is called sensible heat factor* (SHF) Or *sensible heat ratio* (SHR)

$$\text{SHF} = \frac{Q_S}{Q_L + Q_S}$$

The ratio fixes the slope of the line 1—4 on the psychrometric chart. Sensible heat factor slope lines are given on the psychrometric chart. If the initial condition and SHF are known for the given process, then the process line can be drawn through the given initial condition at a slope given by SHF on the psychrometric chart.

The capacity of the cooling coil in *tonnes* of refrigeration is given by,

$$\text{Capacity in TR} = \frac{m_a(h_1 - h_4) \times 60}{14000},$$

where ma = mass of air, kg/min and h = enthalpy in kJ/kg of air.

Cooling and Humidification

If unsaturated air is passed through a spray of continuously recirculated water, the specific humidity will increase while the dry bulb temperature decreases. This is the process of **adiabatic saturation or evaporative cooling.** This process is one of constant adiabatic-saturation temperature and for all practical purposes, one of constant wet bulb temperature. The process is illustrated as path 1-2 on Fig. J, with wet bulb temperature of air being that of point 3, which is also equilibrium temperature of the recirculated water if there is sufficient contact between air and spray, the air will leave at a condition very close to that of point 3. The concept of equivalent by pass can be applied to this process but another term is more used to describe the performance of a humidifying apparatus. It is the **'saturating'** or **'humidifying efficiency'** which is defined as the ratio of dry-bulb temperature decrease to the entering wet bulb depression usually expressed as percentage. Thus, from Fig. J, the saturating efficiency is :

$$\% \eta_{sat} = \left(\frac{t_{db_1} - t_{db_2}}{t_{db_1} - t_{db_3}} \right) \times 100$$

As a fraction, it is equal to one minus the by pass factor for the process. This adiabatic process, for all practical purposes, is line of constant enthalpy. The moisture added can be obtained from the increase in specific humidity.

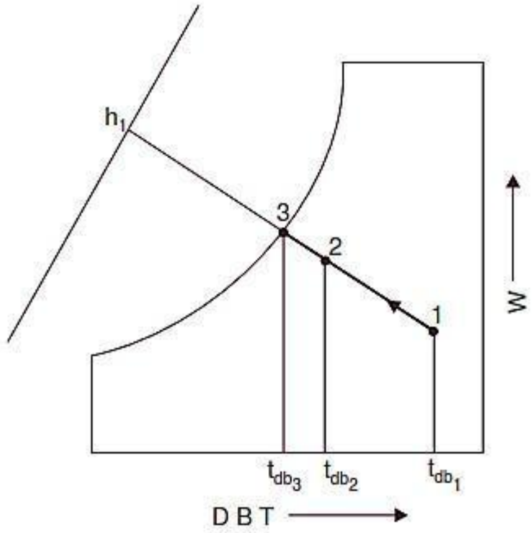


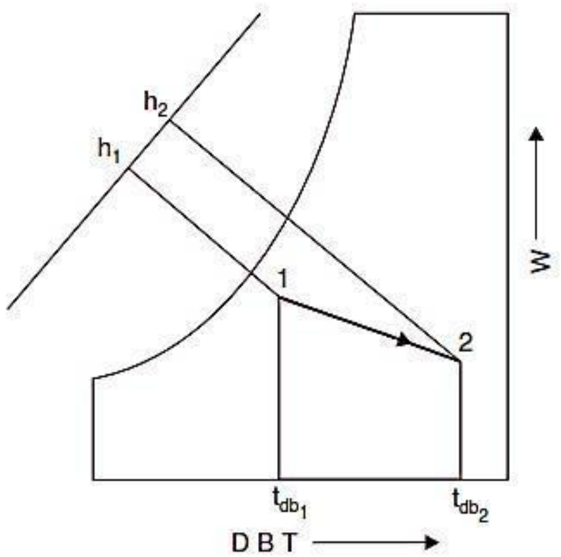
Fig J Cooling and humidification.

Heating and Dehumidification

If air is passed over a solid absorbent surface or through a liquid absorbent spray simultaneous heating and dehumidification is accompanied. In either case the dehumidification results from adsorbent or absorbent having a lower water vapour pressure than air. Moisture is condensed out of the air, and consequently the latent heat of condensation is liberated, causing sensible heating of air. If these were the only energies involved, the process would be the inverse of the adiabatic saturation process. There is, however, an additional energy absorbed or liberated by the active material, termed the heat of adsorption or absorption. For the solid adsorbents used commercially, such as silica gel or activated alumina, and for the more common liquid absorbents, such as solutions of organic salts or inorganic compounds like ethylene, glycol, heat is involved and results in additional sensible heating. Thus the path lies above a constant wet bulb line on the psychrometric chart such as path 1-2 in Fig. K

Heating and Humidification

If air is passed through a humidifier which has heated water sprays instead of simply recirculated spray, the air is humidified and may be heated, cooled or unchanged in temperature. In such a process the air increases in specific humidity and the enthalpy, and the dry bulb temperature will increase or decrease according to the initial temperature of the air and that of the spray. If sufficient water is supplied relative to the mass flow of air, the air will approach saturation at water temperature. Examples of such processes are shown on Fig. L



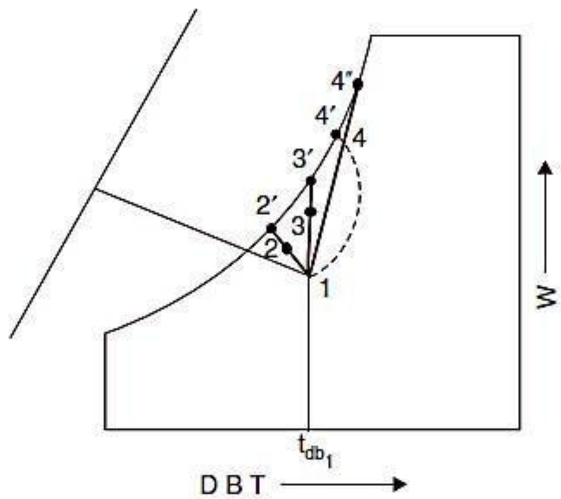


Fig. K. Heating and dehumidification. Fig. L. Heating and humidification.

Process 1-2 : It denotes the cases in which the temperature of the heated spray water is less than the air DBT

Process 1-3 : It denotes the cases in which the temperature is equal to the air DBT. Process 1-4 : It denotes the cases in which a spray temperature is greater than air DBT.

As in the case of adiabatic saturation, the degree to which the process approaches saturation can be expressed in terms of the by-pass factor or a saturating efficiency.

If the water rate relative to the air quantity is smaller, the water temperature will drop significantly during the process. The resultant process will be a curved line such as the dashed 1-4 where 4 represents the leaving water temperature.

Note. It is possible to accomplish heating and humidification by evaporation from an open pan of heated water, or by direct injection of heated water or steam. The latter is more common. The process line for it is of little value because the process is essentially an instantaneous mixing of steam and the air. The final state point of the air can be found, however by making a humidity and enthalpy balance for the process. The solution of such a problem usually involves cut-and-try procedure.

INDUSTRIAL APPLICATIONS

The psychrometric chart plays a vital role in the design, analyses and optimization of various food engineering systems and processing equipments.

The most common food processes where there is heat and moisture transfer between the food and the surrounding air. These processes are food drying, chilling, storage of grains and frozen storage of foods to draw the attention of industry in the overall understanding of the psychrometrics in food production and provide a positive insight to the literature of psychrometrics in those processes.

TUTORIAL QUESTIONS

1. Explain the various psychrometric process in detail with the psychrometric chart.
2. The values obtained from a sling psychrometer are Tdb=30 degree celsius and Twb=20 degree celsius. The barometric readings are 740 mm of Hg. Calculate (i) dew point temp and relative humidity (ii) degree of saturation (iii) specific humidity (iv) specific volume (v) specific enthalpy.
3. Define the following psychrometric terms a. Dry Bulb temperature b.Wet bulb temperature. c. Dew point temperature d. humidity ratio
4. Explain briefly about sensible heating and Sensible cooling?
5. Explain the Heating and Humidification process with neat diagram?
6. Explain the process Cooling and Dehumidification?

UNIT IV

LOAD CALCULATION

COURSE OBJECTIVE: To Explain the Load calculations of Survey of Building, Ventilation requirement for IAQ, ESHF, ADP & Air Flow Rate(CFM)Calculation

COURSE OUTCOME: Students will assist in the installations of Heating, Air Conditioning and Refrigeration equipment.

1. SURVEY OF BUILDING

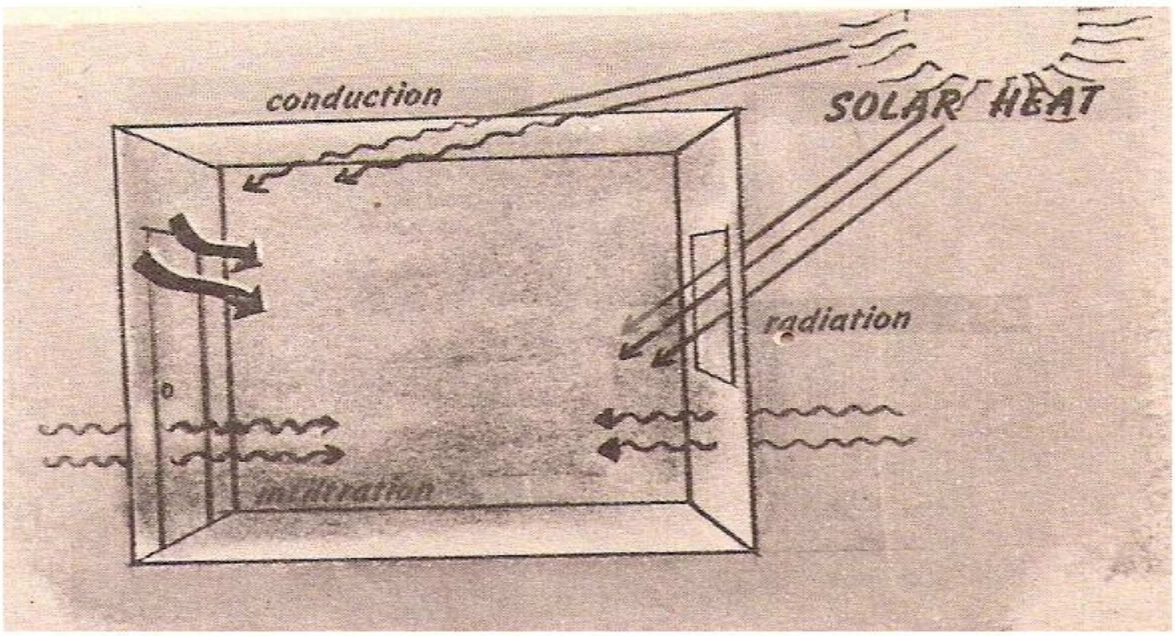
Heat Load Calculations Form

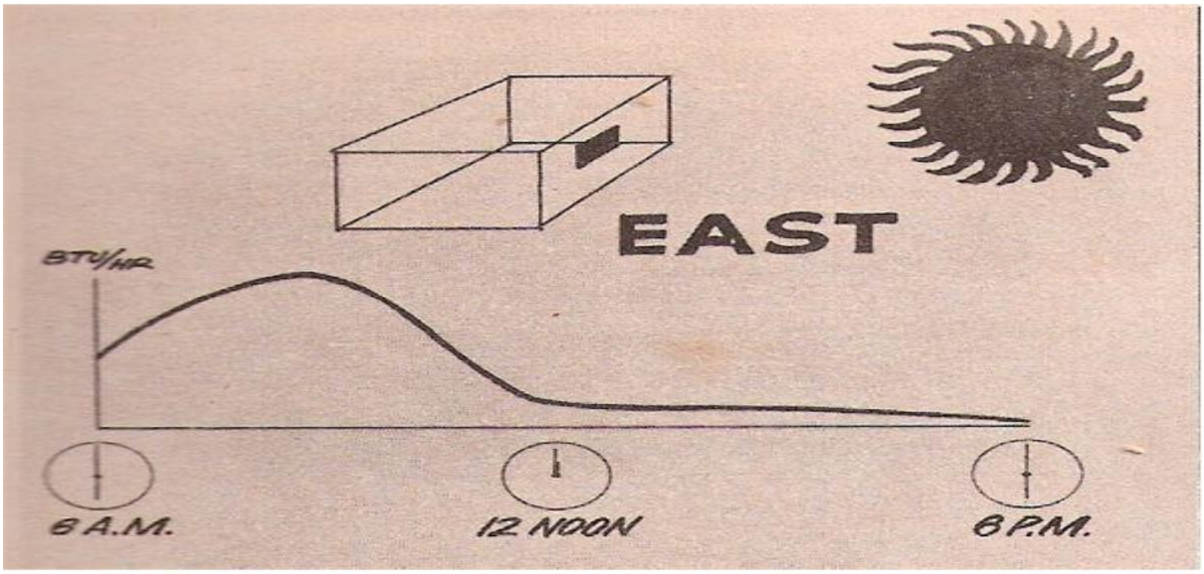
Sources of Heat inside the Residential Room

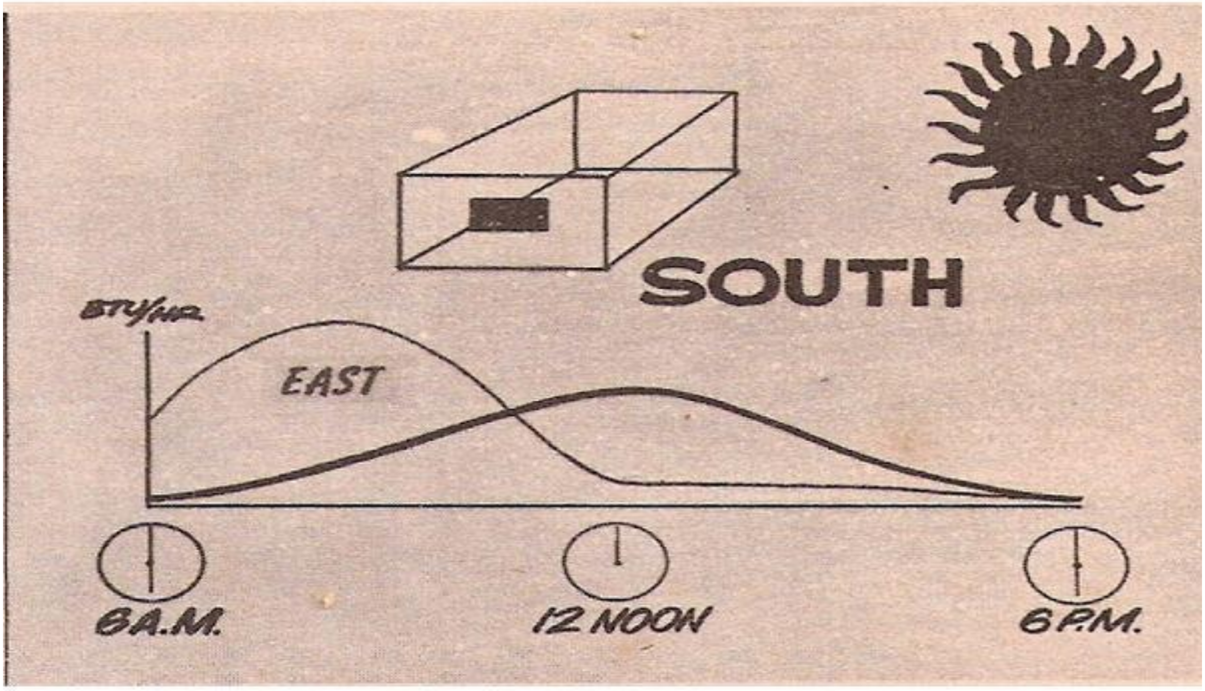
The application of air conditioning is required because heat is generated inside the room from various sources. The main purpose of the air conditioning systems is to remove this heat and create comfort conditions. The temperature and humidity required for the human comfort are 25 degree C and 50% respectively. Due to various heat sources the temperature inside the room becomes very high so all the heat generated inside the room has to be removed.

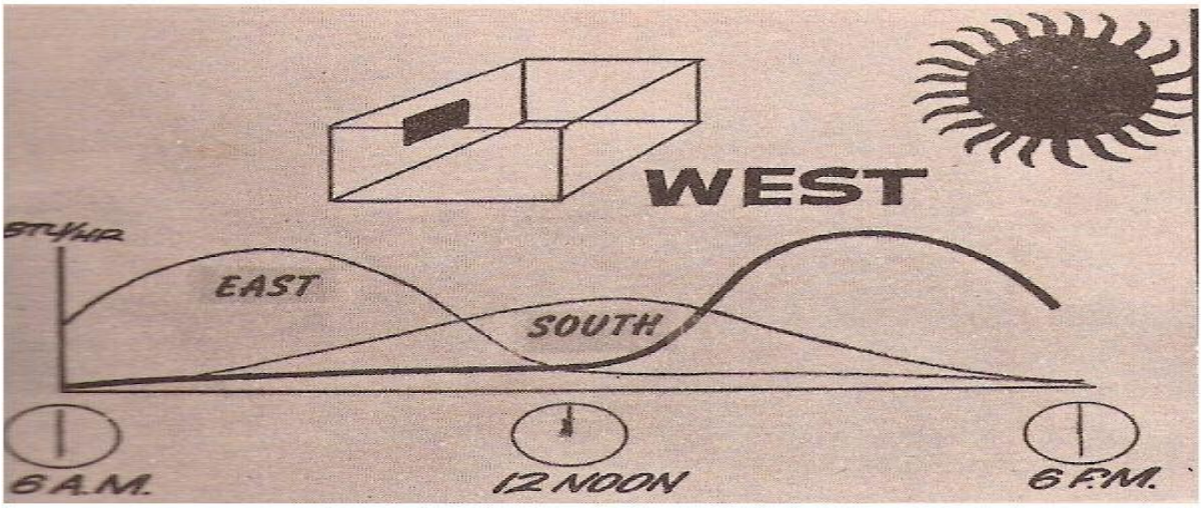
Before actually starting the heat load calculations let us try to see the various sources of heat inside the room mentioned in the items list of the heat load calculations form (please refer the attached form above).

Solar Heat Gain by the Windows

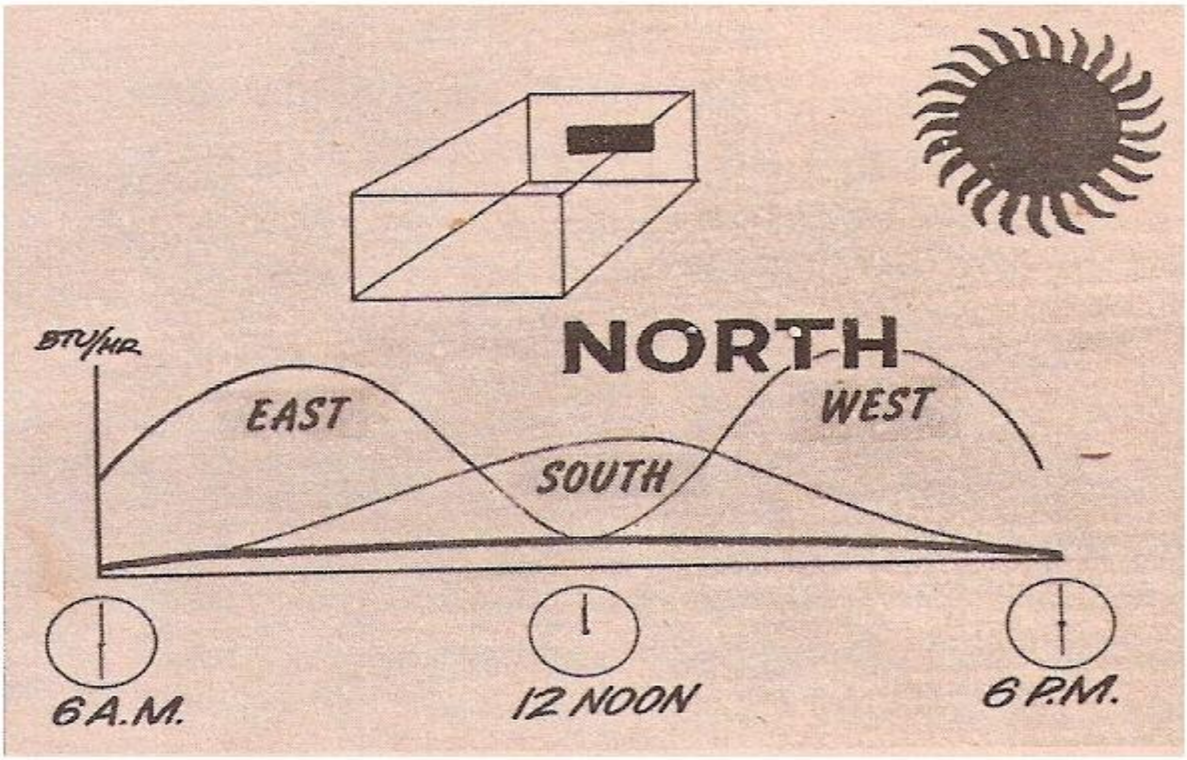



EAST
BTU/HR
6 A.M.
12 NOON
6 P.M.


SOUTH
BTU/HR
EAST
6A.M.
12 NOON
6 P.M.



WEST
BTU/HR
EAST
SOUTH
6 A.M.
12 NOON
6 P.M.



NORTH
BTU/HR
EAST
WEST
SOUTH
6 A.M.
12 NOON
6 P.M.

1. Solar Heat Gain from the Windows

The solar heat is one of the prominent sources of heat inside the room. It enters the rooms via windows, walls and the roof. The solar heat entering the room via windows produces the heating effect immediately. The amount of heat entering the room via windows depends on following factors:

a) The size of the window: Larger the size of the window more is the heat gained from it.

b) Orientation of the window: This is the direction of the window in the room. As per the various positions of the sun throughout the day, it has been found that in the morning the maximum amount of heat is absorbed by the windows in the eastern direction (see the images below). This means the windows in east direction absorb maximum heat in the morning when sun rises. In the afternoon the sun reaches overhead position so the windows in south absorb maximum heat in the afternoon. But this intensity of heat is lesser than that absorbed from east and west. In the late afternoon sun reaches western side and its temperature becomes maximum around 4pm. Thus the windows in west absorb maximum heat in the late afternoon. The amount of heat absorbed by the windows in east and west directions is maximum, it's lesser for the windows in south direction and least for the windows in north direction since sun does not move to north.

c) The glass used for the windows: The double glass used for the windows helps reducing the solar heat gained from the windows. Similarly, if the glass is covered with black or other color shades the amount of solar heat absorbed by it reduces.

d) The awnings used for the windows: The awnings built outside the windows help reduce the amount of direct heat absorbed by the window.

e) The curtains used for the windows: The curtains also play important role in absorption of heat by the windows. The curtains can be thick or thin, inside the room or outside the room or on both sides, the color of the curtains can be dark or light. All the curtains help reduce the amount of heat absorbed by the windows. The dark shades, thick curtains and curtains on both the sides are more effective.

2. Solar Heat Gained by the Walls

Just like the windows, the walls also gain solar heat by conduction and radiation heat transfer methods. The heat gained by the wall is not released inside the room immediately, rather the heat gained by the wall is stored inside it and it is released in the room in late night. This heat creates uncomfortable conditions and it has to be removed from the room. The amount of heat absorbed by the wall depends on following factors:

a) **Size of the wall**

b) Orientation of the wall

c) Thickness of the wall

d) Material of construction of the wall

e) Insulation on the wall, if any and it is of one inch or two inch

3. **Heat Gained by the Partitions**:

There can be partitions inside the room with air conditioned room or non-air conditioned room. The amount of heat absorbed by the partitions with non-air conditioned rooms is higher.

4. **Solar Heat Gained by the Roof of the Room**

Just like the windows and walls the solar heat is also absorbed by the roof of the room. Like the walls, the solar heat absorbed by the roof also reaches the room slowly. The roof exposed to the sun absorbs the heat continuously throughout the day, so it absorbs the maximum heat of all the factors. The amount of heat absorbed by the roof depends on:

a) The size of the roof

b) Thickness of the roof

c) Material of construction for roof

d) Insulation material stuck to the roof and its thickness

5. **Heat Gained from the Ceiling**:

The ceiling of the room is not exposed to the sun directly. Above the ceiling of the room there could be air conditioned room or non-air conditioned room. The heat is absorbed by the ceiling above which there is non-air conditioned room.

6. **Heat Absorbed by the Floor**:

If the floor of the room is exposed to some source of heat is absorbs the heat and it should be taken into account.

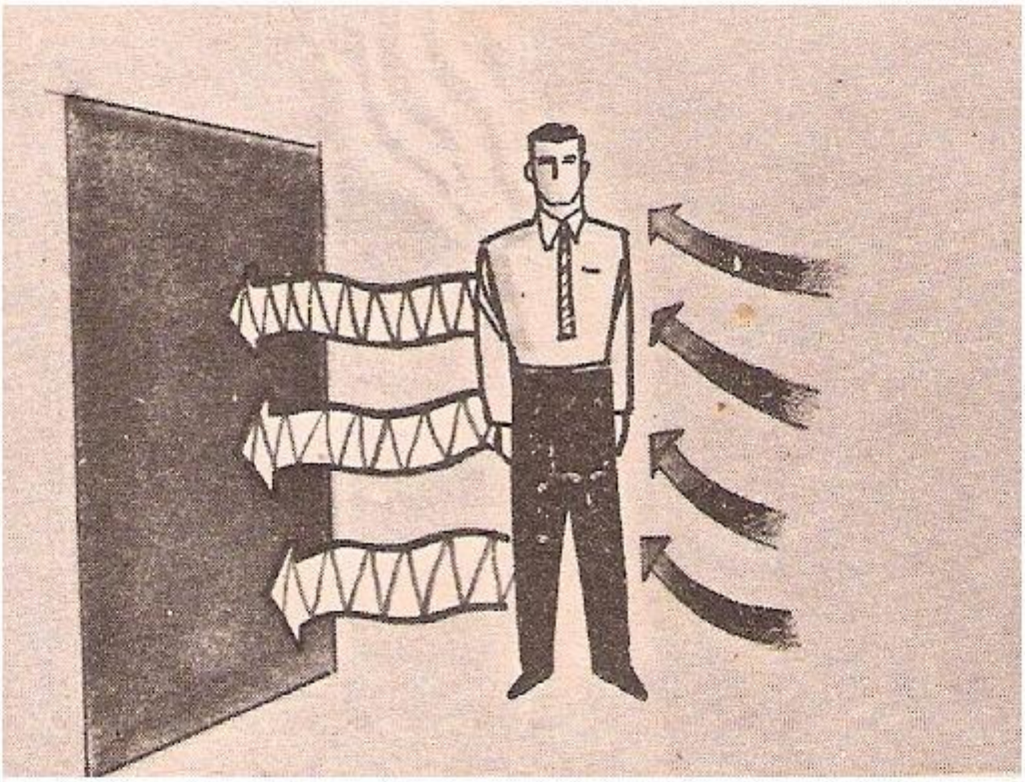
7. **Heat Gained from Outside air or Infiltrated Air**:

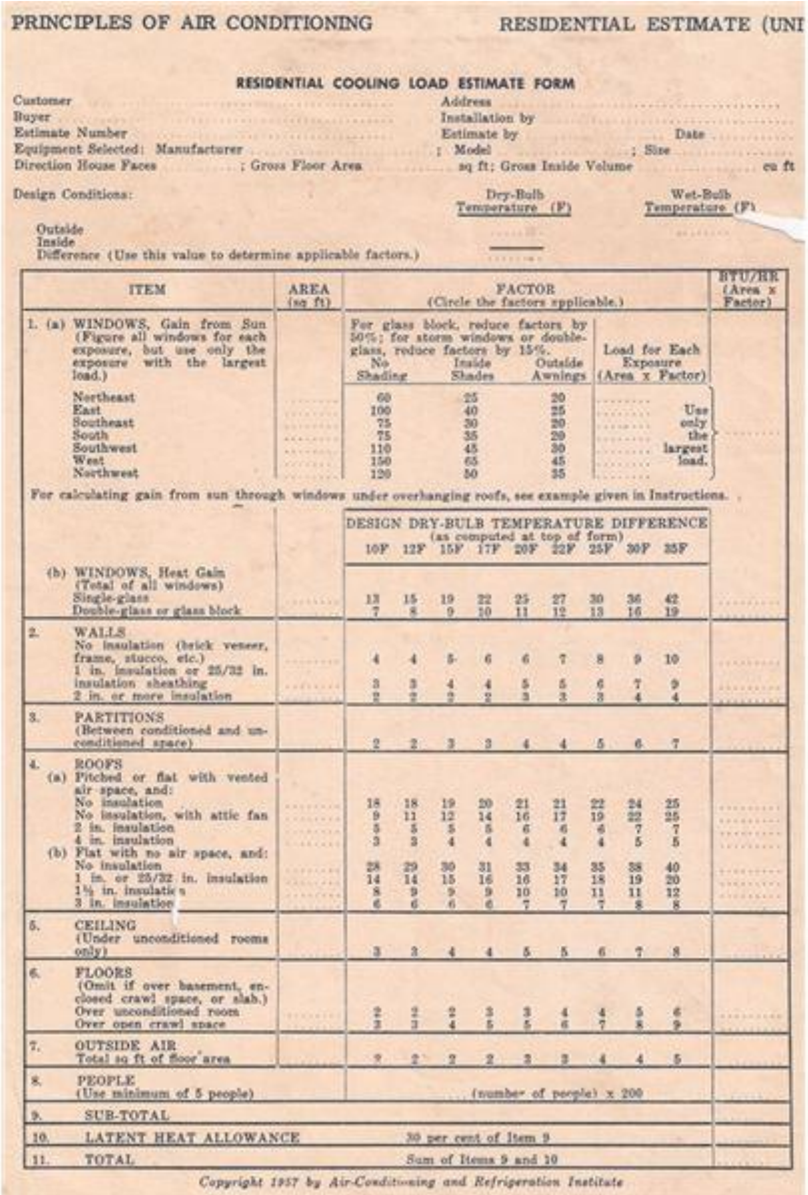
The outside air is at higher temperature than the room air. When it comes inside the room it raises the room temperature. The outside air can come inside the room due to opening of the door or it can leak inside through various openings in doors and windows, this air is also called as infiltrated air.

8. **Heat Gained from the People**

The people inside the room release lots of latent heat and the sensible heat. More the number of people inside the room more is the heat released inside it.

Once all the sources of heat described in the items list of the heat load calculations have been understood, we are now ready to perform the heat load calculations for the residential room. The first and the foremost step in heat calculations is carryout the survey of the room or building. In the next article we shall see various factors of the building to be surveyed.





PRINCIPLES OF AIR CONDITIONING RESIDENTIAL ESTIMATE (UNI

RESIDENTIAL COOLING LOAD ESTIMATE FORM

Customer .... Address ....
Buyer .... Installation by ....
Estimate Number .... Estimate by .... Date ....
Equipment Selected: Manufacturer .... ; Model .... ; Size ....
Direction House Faces .... ; Gross Floor Area .... sq ft; Gross Inside Volume .... cu ft

Design Conditions: Dry-Bulb Temperature (F) Wet-Bulb Temperature (F)

Outside
Inside
Difference (Use this value to determine applicable factors.)

ITEM	AREA (sq ft)	FACTOR (Circle the factors applicable.) For glass block, reduce factors by 50%; for storm windows or double-glass, reduce factors by 15%. No Shading	Inside Shades	Outside Awnings	Load for Each Exposure (Area x Factor)	BTU/HR (Area x Factor)
1. (a) WINDOWS, Gain from Sun (Figure all windows for each exposure, but use only the exposure with the largest load.)						
Northeast	....	60	25	20	....	
East	....	100	40	25	.... Use	
Southeast	....	75	30	20	.... only	
South	....	75	35	20	.... the	....
Southwest	....	110	45	30	.... largest	
West	....	150	65	45	.... load.	
Northwest	....	120	50	35	....	

For calculating gain from sun through windows under overhanging roofs, see example given in Instructions.

ITEM	AREA	DESIGN DRY-BULB TEMPERATURE DIFFERENCE (as computed at top of form) 10F	12F	15F	17F	20F	22F	25F	30F	35F	BTU/HR
(b) WINDOWS, Heat Gain (Total of all windows)											
Single-glass	....	13	15	19	22	25	27	30	36	42	....
Double-glass or glass block		7	8	9	10	11	12	13	16	19	
2. WALLS											
No insulation (brick veneer, frame, stucco, etc.)	....	4	4	5	6	6	7	8	9	10	....
1 in. insulation or 25/32 in. insulation sheathing	....	3	3	4	4	5	5	6	7	9	....
2 in. or more insulation	....	2	2	2	2	3	3	3	4	4	....
3. PARTITIONS (Between conditioned and unconditioned space)		2	2	3	3	4	4	5	6	7	
4. ROOFS											
(a) Pitched or flat with vented air space, and:											
No insulation	....	18	18	19	20	21	21	22	24	25	....
No insulation, with attic fan	....	9	11	12	14	16	17	19	22	25	....
2 in. insulation	....	5	5	5	5	6	6	6	7	7	....
4 in. insulation	....	3	3	4	4	4	4	4	5	5	....
(b) Flat with no air space, and:											
No insulation	....	28	29	30	31	33	34	35	38	40	....
1 in. or 25/32 in. insulation	....	14	14	15	16	16	17	18	19	20	....
1½ in. insulation	....	8	9	9	9	10	10	11	11	12	....
3 in. insulation		6	6	6	6	7	7	7	8	8	
5. CEILING (Under unconditioned rooms only)		3	3	4	4	5	5	6	7	8	
6. FLOORS (Omit if over basement, enclosed crawl space, or slab.)											
Over unconditioned room	....	2	2	2	3	3	4	4	5	6	....
Over open crawl space		3	3	4	5	5	6	7	8	9	
7. OUTSIDE AIR Total sq ft of floor area		2	2	2	2	3	3	4	4	5	
8. PEOPLE (Use minimum of 5 people)		(number of people) x 200									
9. SUB-TOTAL											
10. LATENT HEAT ALLOWANCE		30 per cent of Item 9									
11. TOTAL		Sum of Items 9 and 10									

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Survey of the Building or Room

Once you have understood the various sources of heat in the residential building, we have to now carry out the survey of the building for heat load calculations. The survey of the building involves measuring various dimensions of the building, orientation of the building, and applying various factors to calculate the heat load calculations. With the help of survey one can also decide the best air conditioning system suitable for the building and also the methods to install it.

For the survey of the building the accompanying heat load calculation form has to be used. To carry out the heat load calculations you have to determine all the parameters mentioned in the form, fill them in the form and carry out various calculations, which will eventually help you find the total heat load of a single room. Let us see the various parameters of the building to be surveyed as mentioned in the heat load calculations form. The discussions that follow will explain you all the parts of the form, the survey to be carried out and how to fill the form. For each of the rooms separate heat load calculations form has to be filled. For your convenience the heat load calculations has been attached below, please refer it for all further reading and calculations.

Heat Load Calculations Form

Topmost Portion of the Heat Load Calculations Form

In topmost part of the heat load calculations form you have to fill the basic details like name of the customer, their address, the person who buys the air conditioning system, the person who will install the air conditioning system, estimate number, heat load estimated by, and date. After carrying our heat load calculations you can also fill the details of proposed equipment manufacturer, model, and size of the machine. After the survey of the building you will also be able to fill the details of the direction of the house or room, total floor area for which the heat load calculations are done, and the inside volume of the space that has been surveyed.

After these basic details, you have to fill the design conditions for the room. These are very important and will have a major impact on the total tonnage of the air conditioning system required for the room. In the design conditions space you must note down the outside and desired inside dry bulb and wet bulb temperatures. The difference between the two, as mentioned in the form, has also to be noted.

Orientation of the Room and Various Dimensions

Now you can come to the lower part of the heat load calculations form and start the survey of the room. For this you should have with you two instruments: the magnetic compass and the measuring tape. The magnetic compass will help you determine the orientation of the room including the directions of the all the walls and the windows. The measuring tape will help you determine all the dimensions of the all the walls, partitions, windows, floor, roof, ceiling etc. If you are ready with these instruments let us carry out the following measurements:

1. **Solar Heat Gained by the Windows**

Please refer the heat load calculations form above.

Firstly, find the direction of all the windows using magnetic compass. In the accompanying heat load calculations form, you will notice six directions of the windows: northeast, east, southeast etc. Separate north direction has not been mentioned since it is assumed that heat absorbed by the windows in north direction is very small. For all the windows in all the directions measure the dimension of windows and fill the details in appropriate column. For instance, if there is a window of 24 sq ft in north east direction, fill the details for North East in the "Item" column the area of 24 in the "Area" column. Similarly, fill the areas for all the windows located in all the directions.

Next, select the proper factor for each of the windows from "Factor" column and round the appropriate factor. For example, for the northeast direction window if there is external awning round off the factor 20 associated with outside awning. Reduce this factor by appropriate margin for glass block windows, storm windows or windows with double glass and find corrected factor. In a similar manner, round off the associated factors for all the windows in all the directions and apply correction factor wherever necessary.

Next, multiply the area of each window with the corrected factor associated with it. These details are filled in the "Area x Factor" column. The maximum value of this column has to be filled in the last column of BTU/HR that indicates the solar heat gained by the windows.

Next, you have to find out the solar heat gained by the windows due to designed dry bulb temperature difference between outside and inside. For this firstly, find the total area of all the windows and fill them in the area column. Find out if the windows are of single glass or double glass and fill the details in the appropriate areas column. Now, round off the proper factor for the designed dry bulb temperature difference. For example, if the desired dry bulb temperature difference between the atmosphere and the room is 17F the associated factor would be 22 for single glass window and 10 for double glass window. Multiply the total area of the window by this rounded factor and fill the result in the last column that indicates the total BTU/HR gained by the windows.

Solar Heat Gained by the Walls

To find the heat gained by the wall find the total area of all the walls of the room (excluding partitions). Find out if walls are non-insulated or insulated and the thickness of insulation if any. Accordingly, fill the details in the areas column for the appropriate type of wall. Now round off the associated factor with it for the designed dry bulb temperature difference. Multiply the total area of the wall with this factor and fill the result in the last column of BTU/HR that indicates the total heat gained by the walls per hour.

Heat Gained by the Partitions

In a similar manner, find the total area of the partitions, round off the factor associated with it, multiply the two and fill the result in the last column to find the total BTU absorbed by the partition per hour. Only the partitions that are connected to the non-air conditioned rooms are to be considered.

Solar Heat Gained by the Roof

To find the total heat gained by the roof, find out the total area of the roof and check if it is insulated or non-insulated. Find the thickness of insulation if it is insulated. Now round off the associated factor. Multiply the total area with this factor and fill the result in the last column to find the total BTU of heat gained by roof per hour**.**

Heat Gained by the Ceiling and Floor

By the same procedure as explained for the roof, find the heat gained by the ceiling and the floor.

Heat Gained by the Room Air from the Outside Air or Infiltrated Air

The total outside air or infiltrated air that enters the room has been linked with the total floor area of the room. Find the total floor area of the room and multiply it with the associated factor that gives the total BTU gained by room from the outside air.

Heat Gained by Room Air from the People

To take into account the total heat gained by the room air form the people inside the room, find the average number of people that will stay in the room most of the time. Multiply it by 200 as mentioned in the form and put the result in the last column for total heat gained from the people per hour.

Subtotal of Heat Gained by the Room Air

The subtotal gives the total heat gained by the room air from the windows, walls, partitions, roofs, ceiling, floor, outside air and people inside the room.

Total Heat Load Inside the Room and Total Tonnage of AC Required

To the subtotal of heat gained by the room add additional 30% of the subtotal to account for the latent heat inside the room. This will give the total heat load inside the room. The air conditioner of suitable tonnage that can remove all the total heat gained by the room should be selected. The suitable manufacturer and best possible model of the air conditioner can also be recommended to the customer. This ends the total process of heat load calculations using the ready-made form.

Example Heat Load Calculations

Now that we have seen the various heat loads inside the room and also surveyed the room, let us see one example heat load calculations for the residential building using the heat load calculations form shown below. To start with, fill the details given at the top of the form. These are given below:

Customer: Mr. Allan Smith

Address: New York

Buyer: Mrs. Smith

Installation by: Mr. Garry and Mr. Ronny

Estimate number: 0022

Estimate by: Ms. Sheena Roy

Equipment Selected: Manufacturer, Model and Size (to be filled at the end of heat load estimate): 2.5 TR Split type.

Direction House Faces: North

Gross Floor area (of the house): 1500 sq ft.

Gross inside volume (of the room for which heat load calculations are being done): 300 sq ft

Sample Heat Load Calculations for Residential Building

Design Conditions:

Dry Bulb Temp (DBT) F

Outside 100

Inside 78

Difference 22

Direct Solar Heat Gain by the Windows

There are three windows in the room each of the size 6 x 4 = 24 sq ft. There is one window each in east, south and west direction. The glass of the all windows is single, there is no shading and no outside awnings.

Fill the details of areas in the heat load calculations form as shown in the attached form. Round off the proper associated factor with window in each direction for from no shading option. If the windows has shades or outside awning, one has to round of the factors from those columns. For this particular example the rounded factors have been shown in the form. For window in east direction it is 100, while for window in south and west directions it is 75 and 150 respectively.

Now multiply the area of each of the window by factor associated with it as shown in the form. For window in east direction it is 24 x 100 = 2400, for window in south it is 24 x 75 = 1800, for window in west it is 24 x 150 = 3600. The highest of all these, 3600 has to be selected and filled in the last column. Thus the total solar heat gained by the window is 3600 BTU/HR.

Solar Heat Gained by the Windows due to Designed Conditions (Internal and External Temperature Difference):

The total area of three windows is 24 + 24 + 24 = 72 sq ft and they are all of single glass. Fill this in the area column for single glass window as shown in the form. Since the difference between external and internal dry bulb temperature is 22F, the factor associated with it would be 27, so it has to be rounded of. The product of 72 and 27 is 1944. Thus the solar heat gained by the windows due to design temperature is 1944 BTU/HR.

Heat Gained by the Walls

Let the size of the room is 20 ft x 15 ft = 300 sq ft, which is the total floor area of the room. Let us suppose the height of each wall is 12 ft and none of them are insulated. Two walls of this room if length 20 ft and 15 ft are exposed directly to the sun, while remaining two are partitions.

The total area of walls exposed directly to the sun is 20 x 12 + 15 x 12 = 420 sq ft. Since the designed temperature difference is 22F and there is no insulation, the factor associated with it is 7. The product of 420 and 7 is 2940, which is the total BTU/HR gained by the walls exposed directly to the sun.

Heat Gained by the Partitions

There are two partitions in the room of size 20 x 12 = 240 sq ft and 15 x 12 = 180 sq ft. The first one is with air conditioned room and the other with non-air conditioned room. For heat load calculations we have to consider only the second one. The factor associated with designed temperature difference of 22F is 4. Hence the total heat gained by partition is 180 x 4 = 540 BTU/HR.

Heat Gained by the Roof

The size of roof is same as the size of the floor, which is 20 x 15 = 300 sq ft. The roof is exposed directly to the sun, it is flat with no vented air and it is non-insulated. For 22F of design temperature difference, the factor associated with it is 34. Thus the total heat gained by the roof is 300 x 34 = 10200 BTU/HR.

Heat Gained by the Ceiling

Since the roof is directly exposed to the sun, there is no ceiling for the room, hence there is no heat gained by the ceiling.

Heat Gained by the Floor

The size of the floor is 20 x 15 = 300 sq ft. Let us consider that the room is located over other non-conditioned room, so it gains some heat from it. For the designed temperature difference of 22F, the associated factor is 4. Thus the heat gained by the floor is 300 x 4 = 1200 BTU/HR.

Heat Gained by the Room Air from the Outside Air

The total amount of outside air or the infiltrated air inside the room is proportional to the floor area of the room, thus the total floor area, 300 sq ft, of the room has to be considered. The factor associated with it for designed temperature difference of 22F is 3. Thus the heat gained by the room air from the outside air is 300 x 3 = 900 BTU/HR.

Heat Gained by the Room Air from the People or Occupants

Let us suppose the average number of people inside the room would be six. Thus the heat gained by the room air from people is 6 x 200 = 1200 BTU/HR.

Subtotal of all the Heat Gained by the Room Air

The subtotal heat gained by the room air is total of all the heat gains as mentioned. The total heat gained is 3600 + 1944 + 2940 + 540 + 10200 + 1200 + 900 + 1200 = 22524 BTU/HR.

Latent Heat Allowance

The latent heat allowance includes heat absorbed from the moisture and other small sources. The latent heat allowance is 30% of subtotal, which is 0.3 x 22525 = 6757.5 BTU/HR.

Total Heat Load inside the Room

Thus the total heat load inside the room is 22524 + 6757.5 = 29281.5 BTU/HR.

Recommended Total Tonnage of AC and Type of AC

One ton of AC = 12000 BTU/HR. Thus total tonnage required in the room is 29281.5/12000 = 2.44 tons, which can be taken as 2.5 TR. The total recommended tonnage for the room is 2.5. For this tonnage split air conditioner is the best option. One can install wall mounted split air conditioners of 1.5 ton and 1.0 ton at two different locations inside the room.

COOLING LOAD CALCULATIONS:

As mentioned before, load calculations involve a systematic and stepwise procedure that takes into account all the relevant building energy flows. The cooling load experienced by a building varies in magnitude from zero (no cooling required) to a maximum value. The design cooling load is a load near the maximum magnitude, but is not normally the maximum. Design cooling load takes into account all the loads experienced by a building under a specific set of assumed conditions.

The assumptions behind design cooling load are as follows:

1. Design outside conditions are selected from a long-term statistical database. The conditions will not necessarily represent any actual year, but are representative of the location of the building. Design data for outside conditions for various locations of the world have been collected and are available in tabular form in various handbooks.

2. The load on the building due to solar radiation is estimated for clear sky conditions.

3. The building *occupancy* is assumed to be *at full design capacity*.

4. All building equipment and appliances are considered to be operating at a reasonably representative capacity.

The total building cooling load consists of heat transferred through the building envelope (walls, roof, floor, windows, doors etc.) and heat generated by occupants, equipment, and lights. The load due to heat transfer through the envelope is called as **external load**, while all other loads are called as **internal loads**. The percentage of external versus internal load varies with building type, site climate, and building design. The total cooling load on any building consists of both **sensible** as well as **latent** load components. The sensible load affects dry bulb temperature, while the latent load affects the moisture content of the conditioned space.

Buildings may be classified as **externally loaded and internally loaded**. In externally loaded buildings the cooling load on the building is mainly due to heat transfer between the surroundings and the internal conditioned space. Since the surrounding conditions are highly variable in any given day, the cooling load of an externally loaded building varies widely. In internally loaded buildings the cooling load is mainly due to internal heat generating sources such as occupants or appliances or processes. In general the heat generation due to internal heat sources may remain fairly constant, and since the heat transfer from the variable surroundings is much less compared to the internal heat sources, the cooling load of an internally loaded building remains fairly constant. Obviously from energy efficiency and economics points of view, the system design strategy for an externally loaded building should be different from an internally loaded building. Hence, prior knowledge of whether the building is externally loaded or internally loaded is essential for effective system design.

As mentioned before, the total cooling load on a building consists of external as well as internal loads. The external loads consist of heat transfer by conduction through the building walls, roof, floor, doors etc, heat transfer by radiation through fenestration such as windows and skylights. All these are sensible heat transfers. In addition to these the external load also consists of heat transfer due to infiltration, which consists of both sensible as well

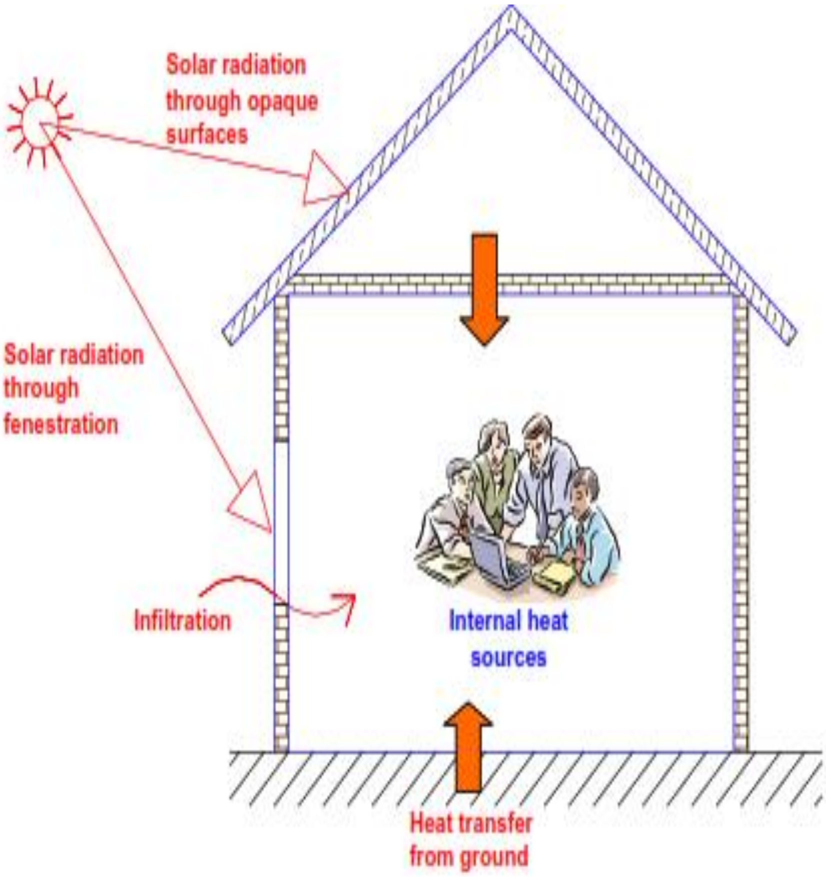


Fig.*: Various cooling load components*

as latent components. The heat transfer due to ventilation is not a load on the building but a load on the system. The various internal loads consist of sensible and latent heat transfer due to occupants, products, processes and appliances, sensible heat transfer due to lighting and other equipment. Figure shows various components that constitute the cooling load on a building.

Estimation of cooling load involves estimation of each of the above components from the given data. In the present chapter, the cooling load calculations are carried out based on the CLTD/CLF method suggested by ASHRAE. For more advanced methods such as TFM, the reader should refer to ASHRAE and other handbooks.

Estimation of external loads:

a) Heat transfer through opaque surfaces: This is a sensible heat transfer process. The heat transfer rate through opaque surfaces such as walls, roof, floor, doors etc. is given by:

$$Q_{opaque} = U.A.CLTD$$

where U is the overall heat transfer coefficient and A is the heat transfer area of the surface on the side of the conditioned space. CLTD is the cooling load temperature difference.

For sunlit surfaces, CLTD has to be obtained from the CLTD tables as discussed in the previous chapter. Adjustment to the values obtained from the table is needed if actual conditions are different from those based on which the CLTD tables are prepared.

For surfaces which are not sunlit or **which have negligible thermal mass** (such as doors), the CLTD value is simply equal to the temperature difference across the wall or roof. For example, for external doors the CLTD value is simply equal to the difference between the design outdoor and indoor dry bulb temperatures, $T_{out}-T_{in}$.

For interior air conditioned rooms surrounded by non-air conditioned spaces, the CLTD of the interior walls is equal to the temperature difference between the surrounding non-air conditioned space and the conditioned space. Obviously, if an air conditioned room is surrounded by other air conditioned rooms, with all of them at the same temperature, the CLTD values of the walls of the interior room will be zero.

Estimation of CLTD values of floor and roof with false ceiling could be tricky. For floors standing on ground, one has to use the temperature of the ground for estimating CLTD. However, the ground temperature depends on the location and varies with time. ASHRAE suggests suitable temperature difference values for estimating heat transfer through ground. If the floor stands on a basement or on the roof of another room, then the

CLTD values for the floor are the temperature difference across the floor (i.e., difference between the temperature of the basement or room below and the conditioned space). This discussion also holds good for roofs which have non-air conditioned rooms above them. For sunlit roofs with false ceiling, the U value may be obtained by assuming the false ceiling to be an air space. However, the CLTD values obtained from the tables may not exactly fit the specific roof. Then one has to use his judgement and select suitable CLTD values.

b) Heat transfer through fenestration: Heat transfer through transparent surface such as a window, includes heat transfer by conduction due to temperature difference across the window and heat transfer due to solar radiation through the window. The heat transfer through the window by convection is calculated using Eq.(35.3), with CLTD being equal to the temperature difference across the window and A equal to the total area of the window. The heat transfer due to solar radiation through the window is given by:

$$Q_{trans} = A_{unshaded} . SHGF_{max} . SC . CLF \qquad (4.4)$$

where $A_{unshaded}$ is the area exposed to solar radiation, $SHGF_{max}$ and SC are the maximum Solar Heat Gain Factor and Shading Coefficient, respectively, and **CLF** is the Cooling Load Factor. As discussed in a previous chapter, the unshaded area has to be obtained from the dimensions of the external shade and solar geometry. $SHGF_{max}$ and SC are obtained from ASHRAE tables based on the orientation of the window, location, month of the year and the type of glass and internal shading device.

The **Cooling Load Factor (CLF)** accounts for the fact that all the radiant energy that enters the conditioned space at a particular time does not become a part of the cooling load[1] instantly. As solar radiation enters the conditioned space, only a negligible portion of it is absorbed by the air particles in the conditioned space instantaneously leading to a minute change in its temperature. Most of the radiation is first absorbed by the internal surfaces, which include ceiling, floor, internal walls, furniture etc. Due to the large but finite thermal capacity of the roof, floor, walls etc., their temperature increases slowly due to absorption of solar radiation. As the surface temperature increases, heat transfer takes place between these surfaces and the air in the conditioned space. Depending upon the thermal capacity of the wall and the outside temperature, some of the absorbed energy due to solar radiation may be conducted to the outer surface and may be lost to the outdoors. Only that fraction of the solar radiation that is transferred to the air in the conditioned space becomes a load on the building, the heat transferred to the outside is not a part of the cooling load. Thus it can be seen that the radiation heat transfer introduces a time lag and also a decrement factor depending upon the dynamic characteristics of the surfaces. Due to the time lag, the effect of radiation will be felt even when the source of radiation, in this case the sun is removed. The CLF values for various surfaces have been calculated as functions of solar time and orientation and are available in the form of tables in ASHRAE Handbooks. Table 35.2 gives

typical CLF values for glass with interior shading.

Solar	**Direction the sunlit window is facing**								
	N	**NE**	**E**	**SE**	**S**	**SW**	**W**	**NW**	**Horiz.**
6	**0.73**	**0.56**	**0.47**	**0.30**	**0.09**	**0.07**	**0.06**	**0.07**	**0.12**
7	**0.66**	**0.76**	**0.72**	**0.57**	**0.16**	**0.11**	**0.09**	**0.11**	**0.27**
8	**0.65**	**0.74**	**0.80**	**0.74**	**0.23**	**0.14**	**0.11**	**0.14**	**0.44**
9	**0.73**	**0.58**	**0.76**	**0.81**	**0.38**	**0.16**	**0.13**	**0.17**	**0.59**
10	**0.80**	**0.37**	**0.62**	**0.79**	**0.58**	**0.19**	**0.15**	**0.19**	**0.72**
11	**0.86**	**0.29**	**0.41**	**0.68**	**0.75**	**0.22**	**0.16**	**0.20**	**0.81**
12	**0.89**	**0.27**	**0.27**	**0.49**	**0.83**	**0.38**	**0.17**	**0.21**	**0.85**
13	**0.89**	**0.26**	**0.26**	**0.33**	**0.80**	**0.59**	**0.31**	**0.22**	**0.85**
14	**0.86**	**0.24**	**0.24**	**0.28**	**0.68**	**0.75**	**0.53**	**0.30**	**0.81**
15	**0.82**	**0.22**	**0.22**	**0.25**	**0.50**	**0.83**	**0.72**	**0.52**	**0.71**
16	**0.75**	**0.20**	**0.20**	**0.22**	**0.35**	**0.81**	**0.82**	**0.73**	**0.58**
17	**0.78**	**0.16**	**0.16**	**0.18**	**0.27**	**0.69**	**0.81**	**0.82**	**0.42**
18	**0.91**	**0.12**	**0.12**	**0.13**	**0.19**	**0.45**	**0.61**	**0.69**	**0.25**

Table 4.2*: Cooling Load Factor (CLF) for glass with interior shading and located in north latitudes (ASHRAE)*

c) Heat transfer due to infiltration: Heat transfer due to infiltration consists of both sensible as well as latent components. The sensible heat transfer rate due to infiltration is given by:

[1] At any point of time, cooling load may be equated to the heat transfer rate to the air in the conditioned space. If heat is transferred to the walls or other solid objects, then it does not become a part of the cooling load at that instant.

$$Q_{s,inf} = \dot{m}_o c_{p,m}(T_o - T_i) = \dot{V}_o \rho_o c_{p,m}(T_o - T_i) \qquad (35.5)$$

where $\dot{V}_o$ is the infiltration rate (in m^3/s), ρ_o and $c_{p,m}$ are the density and specific heat of the moist, infiltrated air, respectively. T_o and T_i are the outdoor and indoor dry bulb temperatures.

The latent heat transfer rate due to infiltration is given by:

$$Q_{l,inf} = \dot{m}_o h_{fg}(W_o - W_i) = \dot{V}_o \rho_o h_{fg}(W_o - W_i)$$

where h_{fg} is the latent heat of vaporization of water, W_o and W_i are the outdoor and indoor humidity ratio, respectively.

As discussed in an earlier chapter, the infiltration rate depends upon several factors such as the tightness of the building that includes the walls, windows, doors etc and the prevailing wind speed and direction. As mentioned before, the infiltration rate is obtained by using either the air change method or the crack method.

The infiltration rate by air change method is given by:

$$\dot{V}_o = (ACH).V/3600 \quad m^3/s$$

where **ACH is the number of air changes per hour** and **V is the gross volume of the conditioned space** in m^3. Normally the ACH value varies from 0.5 ACH for tight and well-sealed buildings to about 2.0 for loose and poorly sealed buildings. For modern buildings the ACH value may be as low as 0.2 ACH. Thus depending upon the age and condition of the building an appropriate ACH value has to be chose,

using which the infiltration rate can be calculated.

The infiltration rate by the crack method is given by

where **A is the effective leakage area of the cracks, C is a flow coefficient** which depends on the type of the crack and the nature of the flow in the crack, Δ**P is the difference between outside and inside pressure (P_o-P_i)** and **n is an exponent** whose value depends on the nature of the flow in the crack. The value of n varies between 0.4 to 1.0, i.e., $0.4 \le n \le 1.0$. The pressure difference Δ**P** arises due to pressure difference due to the wind (ΔP_{wind}), pressure difference due to the stack effect (ΔP_{stack}) and pressure difference due to building pressurization (ΔP_{bld}), i.e.,

$$\Delta P = \Delta P_{wind} + \Delta P_{stack} + \Delta P_{bld}$$

Semi-empirical expressions have been obtained for evaluating pressure difference due to wind and stack effects as functions of prevailing wind velocity and direction, inside and outside temperatures, building dimensions and geometry etc.

Representative values of infiltration rate for different types of windows, doors walls etc. have been measured and are available in tabular form in air conditioning design handbooks.

d) Miscellaneous external loads: In addition to the above loads, if the cooling coil has a positive by-pass factor (BPF > 0), then some amount of ventilation air directly enters the conditioned space, in which case it becomes a part of the building cooling load. The sensible and latent heat transfer rates due to the by-passed ventilation air

can be calculated using equations (35.5) and (35.6) by replacing $\dot{V}_o$ with $\dot{V}_{vent}.BPF$, where cooling coil.

$\dot{V}_{vent}$ is the ventilation rate and BPF is the by-pass factor of the

In addition to this, sensible and latent heat transfer to the building also occurs due to heat transfer and air leakage in the supply ducts. A safety factor is usually provided to account for this depending upon the specific details of the supply air ducts.

If the supply duct consists of supply air fan with motor, then power input to the fan becomes a part of the external sensible load on the building. If the duct consists of the electric motor, which drives the fan, then the efficiency of the fan motor also must be taken into account while calculating the cooling load. Most of the times, the power input to the fan is not known *a priori* as the amount of supply air required is not known at this stage. To take this factor into account, initially it is assumed that the supply fan adds about 5% of the room sensible cooling load and cooling loads are then estimated. Then this value is corrected in the end when the actual fan selection is done.

4.2. Estimation of internal loads:

The internal loads consist of load due to occupants, due to lighting, due to equipment and appliances and due to products stored or processes being performed in the conditioned space.

a) Load due to occupants: The internal cooling load due to occupants consists of both sensible and latent heat components. The rate at which the sensible and latent heat transfer take place depends mainly on the population and activity level of the occupants. Since a portion of the heat transferred by the occupants is in the form of radiation, a Cooling Load Factor (CLF) should be used similar to that used for radiation heat transfer through fenestration. Thus the sensible heat transfer to the conditioned space due to the occupants is given by the equation:

$$\mathbf{Q_{s,occupants} = (No.\ of\ people).(Sensible\ heat\ gain / person).CLF}$$

Table 35.3 shows typical values of total heat gain from the occupants and also the sensible heat gain fraction as a function of activity in an air conditioned space. However, it should be noted that the fraction of the total heat gain that is sensible depends on the conditions of the indoor environment. If the conditioned space temperature is higher, then the fraction of total heat gain that is sensible decreases and the latent heat gain increases, and vice versa.

Activity	Total heat gain, W	Sensible heat gain fraction
Sleeping	**70**	**0.75**
Seated, quiet	**100**	**0.60**
Standing	**150**	**0.50**
Walking @ 3.5 kmph	**305**	**0.35**
Office work	**150**	**0.55**
Teaching	**175**	**0.50**
Industrial work	**300 to 600**	**0.35**

Table 35.3*: Total heat gain, sensible heat gain fraction from occupants*

The value of Cooling Load Factor (CLF) for occupants depends on the hours after the entry of the occupants into the conditioned space, the total hours spent in the conditioned space and type of the building. Values of CLF have been obtained for different types of buildings and have been tabulated in ASHRAE handbooks.

Since the latent heat gain from the occupants is instantaneous the CLF for latent heat gain is 1.0, thus the latent heat gain due to occupants is given by:

$$Q_{l,occupants} = (\text{No. of people}).(\text{Latent heat gain / person})$$

b) Load due to lighting: Lighting adds sensible heat to the conditioned space. Since the heat transferred from the lighting system consists of both radiation and convection, a Cooling Load Factor is used to account for the time lag. Thus the cooling load due to lighting system is given by:

$$Q_{s,lighting} = (\text{Installed wattage})(\text{Usage Factor})(\text{Ballast factor})CLF$$

The usage factor accounts for any lamps that are installed but are not switched on at the time at which load calculations are performed. The ballast factor takes into account the load imposed by ballasts used in fluorescent lights. A typical ballast factor value of 1.25 is taken for fluorescent lights, while it is equal to 1.0 for incandescent lamps. The values of CLF as a function of the number of hours after the lights are turned on, type of lighting fixtures and the hours of operation of the lights are available in the form of tables in ASHRAEhandbooks.

c) Internal loads due to equipment and appliances: The equipment and appliances used in the conditioned space may add both sensible as well as latent loads to the conditioned space. Again, the sensible load may be in the form of radiation and/or convection. Thus the internal sensible load due to equipment and appliances is given by:

$$Q_{s,appliances} = (\text{Installed wattage}).(\text{Usage Factor}).CLF$$

The installed wattage and usage factor depend on the type of the appliance or equipment. The CLF values are available in the form of tables in ASHARE handbooks.

The latent load due to appliances is given by:

$$Q_{l,appliance} = (\text{Installed wattage}).(\text{Latent heat fraction})$$

Table shows typical load of various types of appliances.

Appliance	Sensible load, W	Latent load, W	Total load, W
Coffee brewer, 0.5 gallons	**265**	**65**	**330**
Coffee warmer, 0.5 gallons	**71**	**27**	**98**
Toaster, 360 slices/h	**1500**	**382**	**1882**
Food warmer/m^2 plate area	**1150**	**1150**	**2300**

***Table**: Typical appliance load (C.P. Arora)*

For other equipment such as computers, printers etc, the load is in the form of sensible heat transfer and is estimated based on the rated power consumption. The CLF value for these equipment may be taken as 1.0 as the radiative heat transfer from these equipment is generally negligible due to smaller operating temperatures. When the equipment are run by electric motors which are also kept inside the conditioned space, then the efficiency of the electric motor must be taken into account. Though the estimation of cooling load due to appliance and equipment appears to be simple as given by the equations, a large amount of uncertainty is introduced on account of the usage factor and the difference between rated (nameplate) power consumption at full loads and actual power consumption at part loads. Estimation using nameplate power input may lead to overestimation of the loads, if the equipment operates at part load conditions most of the time.

If the conditioned space is used for storing products (**e.g. cold storage**) or for carrying out certain processes, then the sensible and latent heat released by these specific products and or the processes must be added to the internal cooling loads. The sensible and latent heat release rate of a wide variety of live and dead products commonly stored in cold storages are available in air conditioning and refrigeration handbooks. Using these tables, one can estimate the required cooling capacity of cold storages.

Thus using the above equations one can estimate the sensible **($Q_{s,r}$)**, latent **($Q_{l,r}$)** and total cooling load **($Q_{t,r}$)** on the buildings. Since the load due to sunlitsurfaces varies as a function of solar time, it is preferable to calculate the cooling loads at different solar times and choose the maximum load for estimating the system capacity. From the sensible and total cooling loads one can calculate the Room Sensible Heat Factor (RSHF) for the building. As discussed in an earlier chapter, from the RSHF value and the required indoor conditions one can draw the RSHF line on the psychrometric chart and fix the condition of the supply air.

35.5. Estimation of the cooling capacity of the system:

In order to find the required cooling capacity of the system, one has to take into account the sensible and latent loads due to ventilation, leakage losses in the return air ducts and heat added due to return air fan (if any).

Load on the system due to ventilated air:

Figure 35.2 shows a schematic of an air conditioning system with the cooling coil, supply and return ducts, ventilation and fans. The cooling coil has a by-pass factor X. Then the cooling load on the coil due to sensible heat transfer of the ventilated air is given by:

$$Q_{s,vent} = \dot{m}_{vent}(1-X).c_{p,m}(T_o - T_i) = \dot{V}_{vent}\,\rho_o(1-X).c_{p,m}(T_o - T_i) \qquad (35.12)$$

where $\dot{m}_{vent}$ and $\dot{V}_{vent}$ are the mass and volumetric flow rates of the ventilated air and X is the by-pass factor of the coil.

The latent heat load on the coil due to ventilation is given by:

$$Q_{l,vent} = \dot{m}_{vent}(1-X).h_{fg}(W_o - W_i) = \dot{V}_{vent}\,\rho_o(1-X).h_{fg}(W_o - W_i) \qquad (35.13)$$

where W_o and W_i are the humidity ratios of the ambient and conditioned air, respectively and h_{fg} is the latent heat of vapourization of water.

Load on the coil due to leakage in return air duct and due to return air fan:

If there is leakage of air and heat from or to the return air duct, additional capacity has to be provided by the cooling coil to take care of this. The sensible heat transfer to the return duct due to heat transfer from the surroundings to the return duct depends on the surface area of the duct that is exposed to outside air ($A_{exposed}$), amount of insulation (U_{ins}) and temperature difference between outdoor air and return air, i.e.,

$$Q_{s,duct} = U_{ins}.A_{exposed}(T_o - T_i)$$

The amount of sensible and latent heat transfer rates due to air leakage from or to the system depends on the effectiveness of the sealing provided and the condition of the outdoor air and return air. Since the load due to return air duct including the return air fan ($Q_{return\ duct}$) are not known a priori an initial value (e.g. as a fraction of total building cooling load) is assumed and calculations are performed. This value is modified at the end by taking into account the actual leakage losses

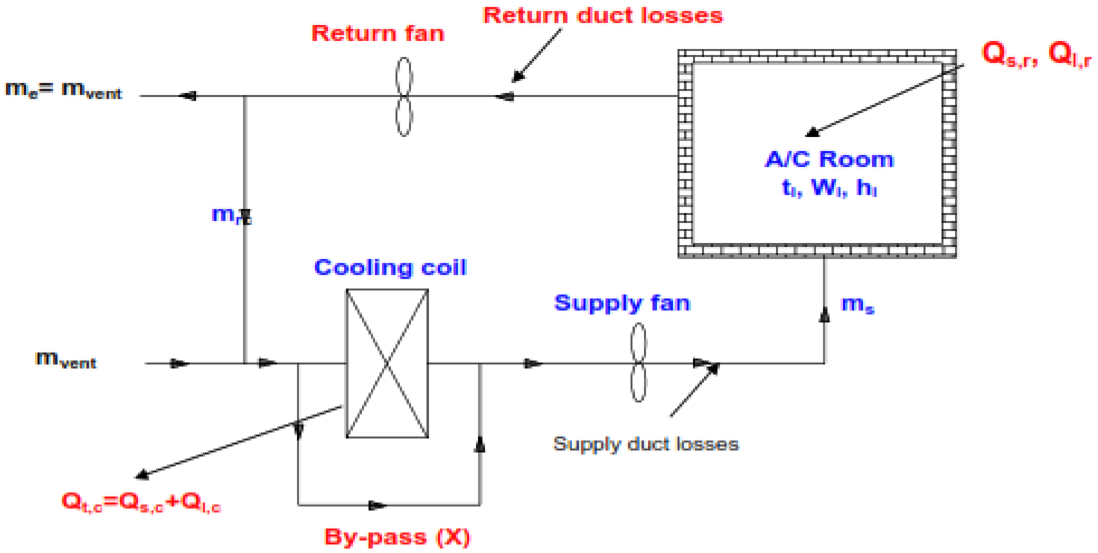


Fig. *A typical summer air conditioning system with a cooling coil of non-zero by-pass factor*

and return fan power consumption.

Now the total sensible load on the coil (**$Q_{s,c}$**) is obtained by summing up the total sensible load on the building (**$Q_{s,r}$**), sensible load due to ventilation (**$Q_{s,vent}$**) and sensible load due to return air duct and fan (**$Q_{s,retrun\ duct}$**), that is:

$$Q_{s,c} = Q_{s,r} + Q_{s,vent} + Q_{s,return\ duct}$$

Similarly the total latent load on the coil (**$Q_{l,c}$**) is obtained by summing up the total latent load on the building (**$Q_{l.r}$**), latent load due to ventilation (**$Q_{l,vent}$**) and latent load due to return air duct and fan (**$Q_{l,retrun\ duct}$**), that is:

$$Q_{l,c} = Q_{l,r} + Q_{l,vent} + Q_{l,return\ duct}$$

Finally the required cooling capacity of the system which is equal to the total load on the coil is obtained from the equation:

$$\text{Required cooling capacity}, Q_{t,c} = Q_{s,c} + Q_{l,c}$$

One can also calculate the sensible heat factor for the coil (CSHF) and draw the process line on the psychrometric chart and find the required coil Apparatus Dew Point Temperature (coil ADP) from the above data as discussed in an earlier chapter.

As mentioned, the method discussed above is based on CLTD/CLF as suggested by ASHRAE. It can be seen that with the aid of suitable input data and building specifications one can manually estimate the cooling load on the building and the required cooling capacity of the system. A suitable **safety factor** is normally used in the end to account for uncertainties in occupants, equipment, external infiltration, external conditions etc. This relatively simple method offers reasonably accurate results for most of the buildings. However, it should be noted that the data available in ASHRAE handbooks (e.g. CLTD tables, SHGF tables) have been obtained for a specific set of conditions. Hence, any variation from these conditions introduces some amount of error. Though this is generally taken care by the safety factor (i.e., by selecting a slightly oversized cooling system), for more accurate results one has to resort actual building simulation taking into account on all relevant factors that affect the cooling load. However, this could be highly complex mathematically and hence time consuming and expensive. The additional cost and effort may be justified for large buildings with large amount of cooling loads, but may not be justified for small buildings. Thus depending upon the specific case one has to select suitable load calculation method.

Heating load calculations:

As mentioned before, conventionally steady state conditions are assumed for estimating the building heating loads and the internal heat sources are neglected. Then the procedure for heating load calculations becomes fairly simple. One has to estimate only the sensible and latent heat losses from the building walls, roof, ground, windows, doors, due to infiltration and ventilation. Equations similar to those used for cooling load calculations are used with the difference that the **CLTD values are simply replaced by the design temperature difference between the conditioned space and outdoors**. Since a steady state is assumed, the required heating capacity of the system is equal to the total heat loss from the building. As already mentioned, by this method, the calculated heating system capacity will always be more than the actual required cooling capacity. However, the difference may not be very high as long as the internal heat generation is not very large (i.e., when the building is not internally loaded). However, when the internal heat generation rate is large and/or when the building has large thermal capacity with a possibility of storing solar energy during day time, then using more rigorous unsteady approach by taking the internal heat sources into account yields significantly small heating small capacities and hence low initial costs. Hence, once again depending on the specific case one has to select a suitable and economically justifiable method for estimating heating loads.

VENTILATION REQUIRMENTS FOR IAQ

Indoor air quality, which is a function of outdoor and indoor air pollutants, thermal comfort, and sensory loads (odors, “freshness”), can affect the health of children and adults and may affect student learning and teacher productivity.

Pollutants are generated from many sources. Outdoor pollutants include ozone, which has been associated with absenteeism among students. Pollutants and allergens in indoor air—mold, dust, pet dander, bacterial and fungal products, volatile organic compounds, and particulate matter—are associated with asthma and other respiratory symptoms and with a set of building-related symptoms (eye, nose, and throat irritations; headaches; fatigue; difficulty breathing; itching; and dry, irritated skin). In some cases, outdoor pollutants react with indoor chemicals to create new irritants.

Thermal comfort is influenced by temperature, relative humidity, and perceived air quality (sensory loads) and has been linked to student achievement as measured by task performance. Relative humidity is also a factor in the survival rates of viruses, bacteria, and fungi and their effects on human health.

Heating, ventilation, and air-conditioning (HVAC) systems are intended to provide effective outside air delivery to rapidly dilute or filter out air contaminants and (2) thermal comfort for building occupants by heating or cooling outside air coming into occupied spaces. Ventilation can be supplied through mechanical systems, which draw air into and push air out of a building, or “naturally,” through the opening and closing of doors and windows and by uncontrolled leakage points through a building’s envelope. A variety of mechanical systems is available, including hybrid systems that use both natural and mechanical ventilation.

HVAC systems must be properly designed and sized to handle the sensible and latent heat loads of outside and recirculated air. If not properly designed, operated, and maintained, HVAC systems can themselves generate pollutants and excess moisture, thereby affecting the health of occupants. The principal standards and guidelines for HVAC system design and operation in the United States are (1) American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Standard 62.1-2004, “Ventilation for Acceptable Indoor Air Quality”; (2) American National Standards Institute (ANSI)/ASHRAE Standard 55-2004, “Thermal Environmental Conditions for Human Occupancy”; (3) the Department of Energy’s EnergySmart Schools guidelines; and (4) individual state codes, some of which are based on or refer to the International Building Code or other codes. Because industry standards for ventilation and energy efficiency have been developed separately, they have, in some cases, had the net effect of increasing relative indoor humidity.

As shown in Figure 4.1, the complex interactions between indoor and outdoor pollutants, moisture/humidity, HVAC systems, operations and maintenance practices can affect occupants' health, comfort, and productivity. These topics are discussed in greater detail in the rest of the chapter.

1. POLLUTANT SOURCES

Pollutants are generated by many sources both internal and external to a school. External sources include combustion products; biological material; and particulate matter and ozone entering through air intakes and the building envelope. People themselves can carry pollen and allergen sources, such as dust mites and pet dander, into a school on their shoes, skin, and clothes. Internal sources include but are not limited to combustion products; building materials and equipment; educational materials; cleaning products; biological agents; and human activity. In some cases, outdoor pollutants react with indoor chemicals to produce new irritants.

2. Outdoor Sources of Pollutants

Outdoor air pollutants can affect the health of children and adults in two ways. First, students, teachers, administrators, and support staff are exposed to outdoor pollutants before they enter a building, which can lead to increased respiratory symptoms . Second, outdoor

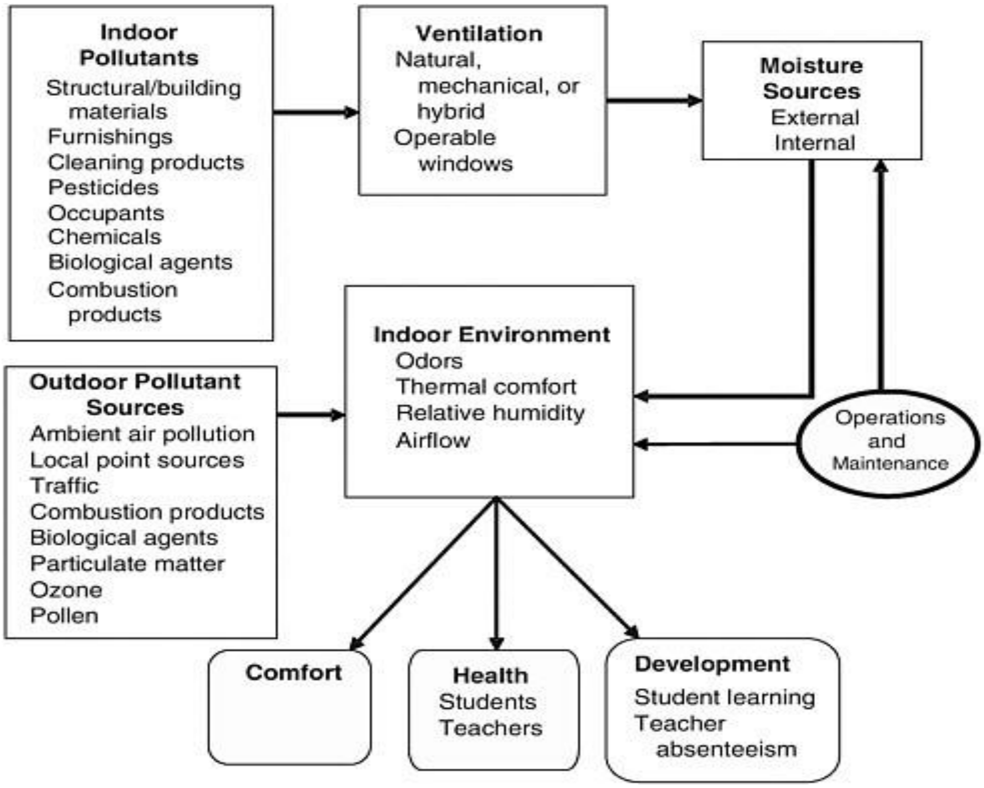


FIGURE 4.1 Relationships between pollutants, moisture, and ventilation and human comfort, health, and development.

sources of pollution can contribute to indoor air pollutant concentrations when outdoor air is drawn into a school building through air intakes located at the rooftop, at ground level, or from below-grade “wells.” Outside air also enters the building through doors, windows, ventilation shafts, and leaks in the building envelope.

They concluded that there was strongly suggestive evidence that absence from school increased with exposure to ozone at higher concentrations. However, the findings were mixed on the associations of school absence with exposure to outdoor nitrogen oxides, carbon dioxide, and particles <10 µm.

Site location can be an important determinant of outdoor pollutants. Schools next to high-traffic areas or with school buses idling their engines next to school doorways, windows, and air intakes may have higher levels of outdoor air pollutants being drawn indoors Other significant sources of outdoor pollutants are plant-derived materials, or biomass, which can generate bioaerosols, including molds, fungi, and pollen. An IOM study (2002, p. 8) found as follows:

Although there is sufficient evidence to conclude that pollen exposure is associated with exacerbation of existing asthma in sensitized individuals, and pollen allergens have been documented in both dust and indoor air, there is inadequate or insufficient information to determine whether indoor air exposure to pollen is associated with exacerbation of asthma.

The IOM study also noted that “there is relatively little information on the impact of ventilation and air cleaning measures on indoor pollen levels, although it is clear that shutting windows and other measures that limit the entry rate of unfiltered air can be effective

Indoor Sources of Pollutants

Indoor pollutants include chemicals, allergens, volatile organic compounds (VOCs), particulate matter, and biological particles or organisms. Chemicals in indoor environments include combustion products such as nitrogen oxides (NO_x), sulfur oxides (SO_x), and carbon monoxide (CO). Combustion products can be generated by gas-fired pilot lights in kitchens and laboratories. Other sources of indoor chemical pollutants include building materials (e.g., structural materials such as particleboard, adhesives, insulation); furnishings (carpets, paints, furniture); products used in a building (cleaning materials, pesticides, markers, art supplies); and equipment (copiers and printers).

Indoor allergen sources—house dust mites, pet dander, cockroaches, rodents, and seasonal pollens—can be brought into a building by occupants, can be generated by furry animals kept in classrooms, or can be attracted to food sources in, for example, school kitchens and cafeterias. Daisey et al. (2003) found that a variety of bioaerosols (primarily molds and fungi, dust mites, and animal antigens) could be found in school environments.

Volatile organic compounds (VOCs) and semivolatile organic compounds (SVOCs) are chemical compounds used extensively in building materials such as adhesives for wood products and structural materials, paints, and carpet adhesives. They also are found in art supplies, paints and lacquers, paint strippers, cleaning supplies, pesticides, office equipment such as copiers and printers, correction fluids and carbonless copy paper, graphics and craft materials, markers, and photographic solutions. In fact, there are no places in schools where VOCs and SVOCs are not found.

Outdoor sources of VOCs and SVOCs include fuels and combustion, biological organisms, and pesticides. Research has shown that concentrations of VOCs are consistently higher indoors than outdoors (Adgate et al., 2004; Wallace, 1991), and studies in homes suggest that indoor concentrations vary depending on the specific VOC (Weisel et al., 2005; Meng et al., 2005). One study also showed that building renovation contributes significantly to total VOC concentrations (Crump et al., 2005).

Particulate matter (PM) includes solid particles ranging in size from ultrafine (<0.1 μm) to relatively large (>10 μm). These particles come from outdoors (including dusts and particles from traffic, stationary sources, and microorganisms), and indoors (humans, building materials, fibers, bioaerosols, mold, pet dander) (Afshari et al., 2005). Larger PM remains suspended in air for relatively short periods of time, instead settling on floors, surfaces, and furnishings. Smaller PM has longer suspension times—i.e., it remains airborne longer. Particulate matter has been implicated in a number of health effects, primarily respiratory and cardiac (Nel, 2005). Particulate matter can absorb VOCs, which may affect occupants' health and comfort (Nilsson et al., 2004).

Larger PM tends to be related to housekeeping practices, ineffective filtration by HVAC systems, and local activity. Finer PM tends to be more independent of these factors, and a fraction of finer PM will even diffuse through structures and so be not removable by HVAC filtration.

One important group of PM is the airborne allergens, including molds and fungi, dander and other body fragments, dust mites, and cockroach antigens. Because these bioaerosols can induce an immune response, they are capable of causing illness at very low exposure levels and also of causing more severe respiratory disease than PM from nonbiological sources. The strength of the association of each of these bioaerosols with illness was summarized in *Clearing the Air: Asthma and Indoor Air Exposures* (IOM, 2000), and many of them were found to be more strongly related to asthmatic symptoms than were moisture and mold.

Improperly maintained HVAC systems can themselves be a source of pollutants. Several findings in *Damp Indoor Spaces and Health* (IOM, 2004) pertain specifically to the design and operation of HVAC systems as a critical factor in the control of moisture and mold growth in buildings:

- Although relatively little attention has been directed to dampness and mold growth in HVAC systems, there is evidence of associated health effects .
- Liquid water is often present at several locations in or near commercial-building HVAC systems, facilitating the growth of microorganisms that may contribute to symptoms or illnesses
- Microbial contamination of HVAC systems has been reported in many case studies and investigated in a few multibuilding efforts
- Sites of reported contamination include outside air louvers, mixing boxes (where outside air mixes with recirculated air), filters, cooling coils, cooling coil drain pans, humidifiers, and duct surfaces.
- Bioaerosols from contaminated sites in an HVAC system may be transported to occupants and deposited on previously clean surfaces, making microbial contamination of HVAC systems a potential risk factor for adverse health effects.

3. `Indoor Air Chemistry

Ozone (O_3) is a primary pulmonary irritant that also plays an important role in indoor chemistry. Although ozone concentrations are generally higher outdoors than indoors, indoor ozone concentrations can be appreciable, infiltrating a building through windows, doors, and the envelope (Weschler et al., 1992). Ozone concentrations might be expected to be higher in naturally ventilated buildings. Indoor ozone sources include printers, copiers, and electrostatic air cleaners if they are not adequately maintained or are improperly exhausted. Sources of indoor terpenes and other unsaturated hydrocarbons are numerous and include cleaning products and air fresheners (Nazaroff and Weschler, 2004).

Reactions among reactive gases (such as ozone) and commonly occurring, nonirritating organic compounds (certain terpenes such as limonene and pinene) can generate products that are highly irritating and can impact human health and comfort (Karlberg et al., 1992; Weschler and Shields, 1997). The process of these ongoing reactions has been termed “indoor air chemistry” (Weschler et al., 1992). Ozone/terpene reaction products have been shown to cause greater airway irritation than either original product (Wolkoff et al., 2006; Weschler, 2004).

concentrations of products generated by reactions among indoor pollutants increased as ventilation decreased. This increase in reaction products is independent of the diurnal variation in ozone levels or of outside ozone levels. These results suggest that maintaining adequate ventilation rates may reduce the potential for reactions among airborne pollutants that generate even more reactive and irritating products.

4. VENTILATION

Ventilation rate is based on the outdoor air requirements of a ventilation system. Ventilation effectiveness is based on the ability of the system to distribute conditioned air within occupied spaces to dilute and remove air contaminants. The principal standard for ventilation rates is American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) Standard 62.1-2004, "Ventilation for Acceptable Indoor Air Quality." However, Daisey et al. (2003), in a comprehensive review of the literature related to indoor air quality, ventilation, and health symptoms in schools, found that reported ventilation and carbon dioxide (CO_2) levels indicated that a significant proportion of classrooms did not meet (then) ASHRAE Standard 62-1999 for minimum ventilation rate.[1]

A number of studies in schools have reviewed the effect of ventilation rates on health, productivity, and airborne pollutant control. Typically, these studies also look at a second variable, such as temperature or humidity, both of which are components of thermal comfort, to identify any confounding or synergistic effects.

5. THERMAL COMFORT

Human perception of the thermal environment depends on four parameters: air temperature, radiant temperature, relative humidity, and air speed (Kwok, 2000). Perception is modified by personal metabolic rates and the insulation value of clothing. Thermal comfort standards are essentially based on a set of air and radiant temperatures and relative humidity levels that will satisfy at least 80 percent of the occupants at specified metabolic rates and clothing values.

There is a robust literature on the effects of temperature and humidity on occupant comfort and productivity, primarily from studies in office buildings (Fanger, 2000; Sepännen and Fisk, 2005; Wyon, 2004; Wang et al., 2005). These studies show that productivity declines if temperatures go too high (Federspiel et al., 2004). However, there is a paucity of studies investigating the relationship between room temperatures in schools and occupant comfort or productivity (Mendell and Heath, 2004).

ASHRAE has codified the air temperature, relative humidity, radiant temperature, and air movement conditions under which occupants should feel "thermally neutral." Guidance is found in ASHRAE Standard 55-2004, "Thermal Environmental Conditions for Human Occupancy," which provides a range of temperatures and relative humidity for winter and summer conditions. When applying current standards, several points are relevant to the school environment**:**

6. PERCEPTION OF AIR QUALITY (SENSORY LOADS)

An expanded definition of comfort includes the perception of air "quality." Occupants may perceive indoor air as heavy, stale, smelly, unpleasant, refreshing, or crisp. As the air is sensed, many attributes are integrated—its temperature, moisture content, odor, and chemical properties. Materials and educational supplies emit odorous compounds as do dirty filters and ducts, cleaning agents, kitchens, bathrooms, gymnasiums, art rooms, moldy surfaces, computers, and copying machines. Chemical reactions that occur indoors also give rise to particles and a host of odorous and irritating compounds.

Fanger (2000) discusses perceived air quality and ventilation requirements in the context of indoor sensory pollution loads from occupants and materials. Exhaled breath, skin, sweat, dirty clothing, perfume, deodorants, and other body odors make the occupants themselves a source of the sensory pollution load degrading perceived indoor air quality. Using nonsmoking adults at 1 Met (metabolic rate) as a reference, kindergarten children at 2.7 Mets contribute 20 percent more to the sensory pollution load. Teenagers 14-16 years old at 1-2 Met activity levels contribute 30 percent more to the sensory pollution load that ventilation air has to handle to achieve the equivalent acceptance.

loads from common indoor objects like carpets, building materials, and personal computers decreased text typing performance as the percent of subjects dissatisfied with air quality increased. They reported a 0.8 percent decrease in text typing for a 10 percent decrease in perceived air quality. Wargocki et al. (2000) showed that increasing ventilation from 3 to 10 to 30 L/s per person improved simulated office work (typing rate and computation rate). These and other studies show that sensory pollution loads indoors are perceived by occupants and that dissatisfaction with perceived indoor air quality may have subtle effects on performance.

Moisture and relative humidity also play a role in the perception of air quality. Moisture in the air can lead to oxidation and chemical reactions by hydrolysis and decomposition, including enzymatic digestion by molds. These processes yield compounds that contribute to sensory pollution loads indoors. Fang et al. (1999a,b) found that perception depended on the enthalpy (heat content) of the air. Air that was cool and dry was perceived as "fresh" and "more pleasant" than air that was warm and moist. Figure 4.3, from Fang et al.

(1999a,b) shows that in the absence of odorous sources people prefer air that is cooler and drier than the air commonly found indoors.

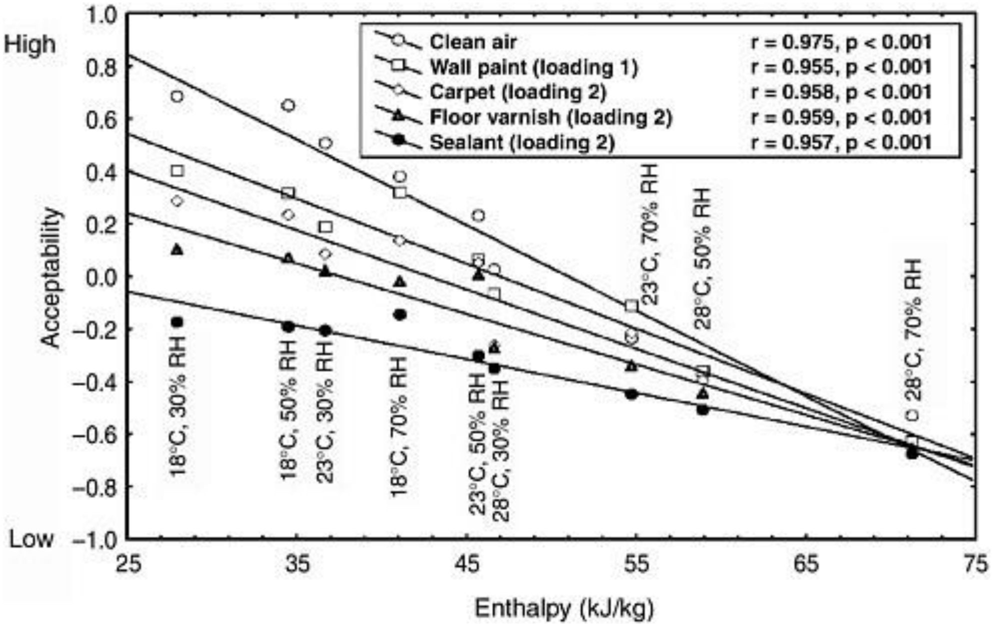


FIGURE 4.3 Acceptability of air quality as a function of enthalpy (heat content) and odorous sources. SOURCE: Adapted from Fang et al. (1999a,b).

The introduction of odor sources was perceived to degrade air quality whether introduced individually or in various combinations. Interestingly, when the enthalpy was high, objectionable odors could not be recognized as easily as when the enthalpy was low. In other words, people prefer cooler, drier air but are then more likely to detect odors, which diminish the perceived air quality.

TABLE 4.1 HVAC Systems and Risk Factors for Building-Related Symptoms

HVAC System Type	Risk
Natural ventilation with operable windows	No particle removal via filtration; poor indoor temperature and control; noise from outdoors; inability to control the pressure difference across the building envelope and exclude pollutant infiltration or penetration of moisture into structure; low ventilation rates in some weather; possible low ventilation rates in some portions of the occupied space.
Systems with ducts and fans but no cooling or	HVAC components may be dirty when installed or become dirty and release pollutants and odors; poor control of

humidification (simple mechanical ventilation)	indoor temperature due to absence of cooling; low humidity in winter in cold climates; high humidity during humid weather; noise generated by forced air flow and fans; draft caused by forced air flows.
Systems with ducts, fans, and cooling coils (air conditioning systems)	Additional risk factors from cooling coils: very high relative humidity or condensed moisture (e.g., in cooling coils and drain pans) and potential microbial growth; biocides used to treat wet surfaces such as drain pans and sometimes applied to nearby insulation.
Systems with ducts, fans, cooling coils, and humidifiers of various types	Additional risk from humidifiers: microbial growth in humidifiers; transport of water droplets downstream of humidifiers, causing wetting of surfaces; leakage and overflow of humidifier water; condensation from humid air; biocides in humidifiers; chemical water treatments in steam generators.
Systems with recirculation of return air (recirculation may occur in all mechanical HVAC systems)	Additional risks[a] from recirculation: indoor-generated pollutants are spread throughout the section of building served by the air-handling system; typically higher indoor air velocities increase risk of draft and HVAC noise; supply ducts and filters of HVAC system may become contaminated by recirculated indoor-generated pollutants.
Sealed or openable windows (windows may be sealed or openable with all types of mechanical HVAC systems)	Additional risk with sealed windows: no control of the environment if HVAC systems fails; psychological effect of isolation from outdoors. Additional risk with operable windows: more exposure to outdoor noise and pollutants.
Decentralized systems (cooling and heating coils located throughout building, rather than just in mechanical rooms)	Additional risk of decentralization: potentially poorer maintenance because components are more numerous or less accessible; potentially more equipment failures due to larger number of components.

[a]However, recirculation facilitates removal of indoor-generated pollutants using air cleaners, e.g., particle filters and may also decrease concentrations of pollutants near

pollutant sources.

SOURCE: Sepännen and Fisk (2002).

7. VENTILATION SYSTEM STANDARDS

ASHRAE Standard committees periodically update the various standards documents. The ASHRAE Standard 62 series addresses ventilation in buildings and the 90.1 series addresses energy efficiency. After initially lowering ventilation requirements in response to the early 1970s energy crisis, ASHRAE Standard 62-2001, "Ventilation for Acceptable Indoor Air Quality," now requires substantially higher ventilation rates for schools and other buildings. ASHRAE Standard 90.1-2001, "Energy Standard for Buildings Except Low-Rise Residential Buildings," and other energy-saving measures such as more efficient motors, office equipment, and lighting, along with better thermal insulation for building envelopes, have systematically reduced the sensible heat loads of buildings. This has implications for HVAC design and operations.

In a report commissioned for AirXchange Corporation, TIAX (2003) demonstrated the consequence of systematic changes in buildings as a result of shedding heat loads and increasing ventilation. The net effect of lowering sensible heat loads while increasing ventilation rates without specifically dealing with latent heat loads has been to increase indoor relative humidity. Heat gains from within buildings have decreased. Henderson (2003) and Shirey (2003) report that in certain common conditions the cycling time of HVAC systems is shortened, leaving condensed moisture on coils that can reevaporate, adding moisture to the building supply air. Maintaining optimally comfortable humidity (between 40 and 50 percent) is more difficult. Higher humidity increases condensation on cooled indoor surfaces and thermal bridges. Humidity that remains above 65 percent for appreciable time increases the opportunity for mold growth.

Although there are studies looking at the energy efficiency and health effects of HVAC system operation, few if any studies directly compare the energy efficiency and health trade-offs, if any, of HVAC system operation (Engvall et al., 2005).

8. Ventilation

Ventilation systems are designed to manage the sensible and latent heat loads of buildings. Outside air is needed for ventilation to provide thermal comfort as well as for diluting and removing indoor pollutants, odors, and moisture. Depending on the design and operational parameters of an HVAC system, the air supplied to the spaces can be entirely outdoor air (no recirculated air) or outdoor air mixed with indoor air drawn from the indoor spaces (return air). Using no recirculated air in a space requires more energy because large amounts of air must be conditioned for temperature and humidity levels when a mechanical system is used. In most cases using a percentage of return air mixed with the outside air is desirable for energy conservation. Increasing the amount of outside air in this air mixture to as high a level as is practical could potentially result in higher levels of human health, comfort, and productivity.

Ensuring that the air supplied is as clean as possible requires controlling the sources of pollutants and moisture within the ventilation system itself, cleaning the incoming outside air as much as possible prior to mixing it with the return air (most commonly this is going to be particulate filtration only, but where the outside air is very contaminated, gas-phase air filtration may also be used), effectively and continuously maintaining the hygiene of the HVAC system, and controlling indoor pollutant sources to minimize the spread of airborne pollutants. Additionally, a ventilation system should be capable of effectively distributing the ventilation air into occupied spaces and exhausting the return air from those spaces. Balancing the ventilation system for effective supply and exhaust rates is critical.

Many schools use unit-ventilator systems, a type of decentralized system, because their first costs (design and installation) are generally less than those of central systems: Unit ventilators eliminate the requirement for ducted supplies and returns (plenum or ducted). They distribute all the air from a single location, usually on the external wall of a room, thereby reducing ventilation effectiveness. They also typically do not meet the requirements for low ambient noise, necessary for acoustical quality associated with student learning. Teachers often use the top of a unit ventilator as a storage shelf, so if this type of ventilation system is used in a green school, helping teachers understand the importance of not blocking air vents on the system is critical.

Central HVAC systems, which may supply small blocks of classrooms or entire sections of a school, require supply ducts and an air return system (plenum or ducted) to move the air to occupied spaces. Central system can have multiple supply and return vents in a single classroom, potentially increasing ventilation effectiveness. Additionally, student comfort might improve since there is a lower probability of air blowing on students sitting on one side of a room.

The design of and materials used in the supply air ducts may have an influence on the long-term health and well-being of the students and on system maintainability. The noise of air moving in the ducts and its potential impact on student learning and teacher health Where the air has a high moisture content, the use of fiberglass-lined ductwork to attenuate noise transmission can support the growth of microbial contamination, if the system is not properly maintained.

The type of ventilation system used may depend on the climate. Throughout much of the United States, ventilation systems need to control humidity as well as temperature and ventilation rates throughout the year. This is particularly true where schools are used year-round. The ventilation system is the primary mechanism for indoor humidity control, particularly in hot and humid climates. Excess humidity in the ventilation system, ductwork, and the building spaces increases the probability of indoor microbial contamination. Active humidity control systems, such as desiccant systems, may be effective for controlling humidity through ventilation systems in hot and humid climates (Fischer and Bayer, 2003). Displacement ventilation is another form of active humidity control in cold climates (Melikov et al., 2005).

School buildings are intended to be used for many years, so it is critical that the ventilation system be designed to allow effective operations and maintenance practices. Sepännen et al. (2004b), in a literature review on the association of ventilation rate and human responses, reported that better hygiene, commissioning, operation and maintenance of air handling systems may be particularly important for reducing the negative effects of HVAC systems. Ventilation may also have harmful effects on indoor air quality and climate if not properly designed, installed, maintained, and operated. To be well-designed, HVAC systems should be easily accessible to facility maintenance staff for maintenance and repair activities.

9. Filtration

Indoor and outdoor particulates and certain VOCs can be effectively removed by filtration. Most filters are designed to collect particles larger than 10 m but are relatively inefficient at removing submicron-sized particles. The location of the filters is critical and should ensure that both outside air and recirculated air are effectively filtered for particulate and VOCs removal before the airstream reaches occupants. In addition, filters should be located such that they can be consistently maintained. Efficient and effective filtration that removes particulate contamination to a level that protects building occupants and not just the equipment is essential (IOM, 2000, pp. 360-382). Particulate filtration having a Minimum Efficiency Reporting Value (MERV) of 11 or higher should be on all HVAC equipment supplying air to the occupied spaces of a building. Filters should fit snugly to prevent the bypass of air around the filter(s). The filters should be changed frequently and regularly to prevent them from becoming a source of indoor air pollution (Clausen, 2004; Hanssen, 2004). Additionally, filters should be kept dry, since wet filters may become microbially contaminated and thereby spread contamination throughout the area served by the ventilation system.

Use of gaseous-phase filtration to remove gaseous pollutants from the supply air stream may be desirable in areas with significant amounts of outdoor air pollution. Gaseous-phase filters or filter media should be changed frequently and regularly.

10. Cleaning

Although to date no systematic research has examined the relationship of cleaning effectiveness to student and teacher health, student learning, or teacher productivity (Berry, 2005), a few studies have related methods for source reduction or control in schools to exposures to pollutants. Smedje and Nörback (2001) observed that classrooms with more frequent cleaning had lower concentrations of cat and dog antigen in settled dust. However, the study could not be repeated. Few studies have looked systematically at changes in exposure, health, or productivity in relation to changes in school building materials, cleaning products, or cleaning practices.

The effects of air pollutants in schools can be reduced through proper design and maintenance practices for HVAC filters, drip pans, cooling coils and other elements. Simple measures such as closing windows during pollen season or prohibiting furry pets in a school may also be effective. In other cases, more subtle design considerations may be needed, for example, limiting food preparation, vending, and eating to certain areas with structural and surface finishes that allow for cleaning and easy pest control.

HVAC HEATING LOSS CALCULATIONS & PRINCIPLES

Introduction

There are two different but related calculated values of interest to the heating system designer. The first is to estimate the maximum rate of heat loss to properly size the heating equipment (furnace). The second calculated value that must be determined is the annual heating bill. This is determined by calculating the annual energy requirement based from the design heat loss rate.

In this course, we will learn to determine the rate at which heat is lost through building elements using a process called heat loss calculation. You will learn how to extrapolate your calculation of a maximum hourly rate into an annual energy usage rate. You will also learn some useful tips on saving heating energy.

The section-3 of the course includes one sample example.

Factors Affecting Comfort in winter

1. TEMPERATURE difference between the inside and outside of the building is the primary cause of heat loss in the winter months. The greater this difference, the higher the rate of heat loss. Since most buildings are controlled to a constant inside temperature by the occupants, higher heat loss occurs when it is colder outside. This also means that the annual heating bill can be reduced by lowering the setting on the thermostat …. (but only if the occupants agree to it!)

2. WIND is the second greatest source of heat loss during the winter. High winds can occur on the cold nights and when they do, heat loss can be higher because of air scrubbing the outside of the space covering. Winds can also force their way through cracks in the structure, causing infiltration and drafts. In fact, up to one-third of the annual heating energy goes to heat this moving infiltration air many times each winter day.

3. HUMIDITY levels can also affect the comfort within a structure. Very low humidity levels (less than 20% relative humidity) cause scratchy throats and dry noses in most people.

Very high humidity levels (over 60%) are also uncomfortable, since the body's ability to perspire is restricted.

4. RADIATION sources can also affect comfort in a structure. The sun shining through a window will make a room very comfortable in winter; that same sun could make it unbearable in summer. Walls and windows also release and absorb radiation. A Trombe wall heated by the sun will keep a room feeling warm with an air temperature less than 60°F. A large expanse of cold glass windows can also make a room feel chilly.

Remember that these same four factors are also important in determining cooling requirements, but control of humidity and solar gain are much more important during that season.

HEATING LOSS ESTIMATION

The heat loss is divided into two groups:

1) The conductive heat losses through the building walls, floor, ceiling, glass, or other surfaces, and

2) The convective infiltration losses through cracks and openings, or heat required to warm outdoor air used for ventilation.

Normally, the heating load is estimated for winter design temperature usually occurring at night; therefore, in determining the heating load, credit for heat generation from internal heat sources such as lights, machinery, appliances, and people is usually ignored. Also in determining the heating load, credit for solar heat gain is usually NOT included and is generally ignored. Credit for solar heat gain is a plus factor in winter heating.

HEAT LOSS FROM BUILDING ENVELOPE (Wall, Roof, Glass)

Heat loss occurs from a building structure primarily due to conduction. Because heat moves in all directions, when calculating the heat loss of a building, we much consider all surfaces (external walls, roof, ceiling, floor, and glass) that divide the inside, heated space from the outside. We refer to that dividing line as the ***Building Envelope.*** The heat loss is determined by equation:

$$Q = A * U * (Ti – To)$$

Where

- Q = Total hourly rate of heat loss through walls, roof, glass, etc in Btu/hr
- U = Overall heat-transfer coefficient of walls, roof, ceiling, floor, or glass in Btu/hr ft^2 °F
- A = Net area of walls, roof, ceiling, floor, or glass in ft^2
- Ti = Inside design temperature in °F
- To = Outside design temperature in °F

Let's examine each one of these terms, starting at the bottom with the outside design temperature.

Outside Design Temperature (To)

Look up for location

Since the inside of the building is controlled to a fixed temperature by the thermostat, the maximum rate of heat loss will occur during the record cold temperature. When designing the heating system for a structure, the first step is to obtain data on the local micro climate of the region. This information is available from a variety of sources, but HVAC designers normally use the ASHRAE Fundamentals Handbook for ready reference. As a basis for design, the most unfavorable but economical combination of temperature and wind speed is chosen. The winter month heating load conditions are based on annual percentiles of 99.6 and 99%, which suggests that the outdoor temperature is equal to or lower than design data 0.4% and 1% of the time respectively. For example, the Pittsburgh, PA, 99% design temperature is 4°F. Only one percent of the hours in a typical heating season (about 35 hour's total) fall at or below that temperature. Since most of these hours are during the night-time when most people are sleeping, and because these extremes are buffered by the large storage mass of the building, these cooler periods usually go unnoticed.

Inside Design Temperature (Ti)

Always use 65°F

The inside design temperature is traditionally taken as 65°F, because in most buildings there is enough heat internally generated from people, lighting, and appliances. Today people are keeping thermostats set lower, so load predictions based on this method are usually conservative, and will result in furnace size recommendations that are slightly larger than actually needed.

Note that the temperature difference between the inside and outside of the building is the primary cause of heat loss in the winter months. The greater this difference, the higher the rate of heat loss. Since most buildings are controlled to a constant inside temperature by the occupants, higher heat loss occurs when it is colder outside.

Net Area (A)

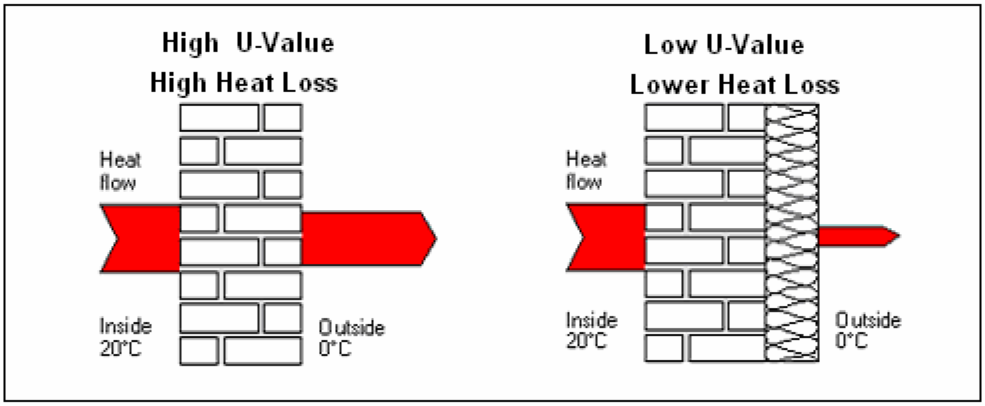
Measured on the drawing/building

The net area of each building section is determined from either the drawings (in new construction) or from field measurements (in retrofit situations). In addition to the areas of the four walls, floor, and ceiling, we must also consider heat loss from doors and windows. We will also need to determine the volume of the building to estimate the rate of infiltration into the building measured in air changes per hour.

Overall Coefficient of Heat Transfer (U)

Look up for materials used

The letter *"U"* represents the overall coefficient of heat transfer. The U-value measures how well a building component, e.g. a wall, roof or a window, keeps heat inside a building. For those living in a warm climate the U-value is also relevant as it is an indicator of how easy it is to keep the inside of the building cold.



The higher the U-value the more heat flows through so a good U-value is a low one as you want to keep heat inside the building or outside depending on the climate you live in. A house built with low U-value building components will use less energy and thus the building owner saves money on the energy bill. Using less energy is good for the environment.

"U" factor is the inverse of "R" factor, ("U" = 1 / "R"); *the larger the R-value or the lower the "U" factor, the lower the heat loss.* Calculating the U-value is often complicated by the fact that the *total* resistance to the flow of heat through a wall made of several layers is the sum of the resistances of the individual layers. This aspect is discussed in detail in subsequent sections.

Heat Loss (Q)

Total hourly rate of heat loss through walls, roof, glass is given by equation Q = U * A * ΔT.

For example: 10 sq-ft. of single glass [U value of 1.13] with an inside temperature of 70°F and an outside temperature of 0°Fwould have 791 BTUH heat loss:

A (10) x U (1.13) x ΔT (70) = 791 Btu/hr

Since the building structure is made of different materials, for example a wall that contains windows and door, just calculate the heat loss through each of the components separately, then add their heat losses together to get the total amount.

Q (wall) =Q (framed area) +Q (windows) +Q (door)

In North America, heat loss is typically expressed in terms of total British Thermal Units per Hour or Btu/hr.

HEAT LOSS FROM FLOORS ON SLAB

Heat loss from floors on slab can be estimated by equation:

$Q = F * P * (T_i - T_o)$

Where:

1) F is the Heat Loss Coefficient for the particular construction in Btu/hr- ft-°F

2) P is the perimeter of slab in ft

3) T_i is the inside temperature in °F

4) T_o is the outside temperature in °F

Heat loss from slab-on- grade foundations is a function of the slab perimeter rather than the floor area. Perimeter is the part of the foundation or slab nearest to the surface of the ground outside. The losses are from the edges of the slab and insulation on these edges will significantly reduce the heat losses.

For basement walls, the paths of the heat flow below the grade line are approximately concentric circular patterns centered at the intersection of the grade line and the basement wall. The thermal resistance of the soil and the wall depends on the path length through the soil and the construction of the basement wall. A simplified calculation of the heat loss through the basement walls and floor is given by equation:

$Q = A * U_{base} * (T_{base} – To)$

Where

- *A* = *A*rea of basement wall or floor below grade in ft^2
- U_{base} = Overall heat-transfer coefficient of wall or floor and soil path, in Btu/hr ft^2 °F
- T_{base} is the basement temperature to be maintained in °F

- T_o is the outside temperature in °F

The values of U_{base} are roughly given as follows:

	0 to 2 ft below grade	Lower than 2 ft
Un insulated wall	0.35	0.15
Insulated wall	0.14	0.09
Basement floor	0.03	0.03

Source: *ASHRAE Handbook 1989*, *Fundamentals*

Calculating heat loss through a basement or slab on grade is more difficult for two main reasons: First because the soil can hold a large quantity of heat, second because the temperature in the ground is not the same as outside temperature (in fact it varies little by season). Because of these reasons, buildings loose more heat through their perimeter and the standard practice is to insulate basement walls and 2-4 feet under the slab near those walls. The ASHRAE method is to calculate heat loss for this situation is to look up a perimeter heat loss factor (called "F") in a table based on the "R" value of perimeter insulation used.

Note that the portion of heat transmission from basement is usually neglected unless the weather in winter is severe and the values are significant in comparison with other forms of heat transmission.

HEAT LOSS DUE TO INFILTRATION & VENTILATION

The second type of heat loss in buildings is infiltration. To calculate this, you need to know the volume of the space (i.e. sq ft of floor times ceiling height) and how much air typically leaks out , which is often stated as how many times per hour the entire air in the building space is lost to outside and referred to as air changes per hour or ACH. Infiltration can be considered to be 0.15 to 0.5 air changes per hour (ach) at winter design conditions. The more the windows on the external walls, the greater will be the infiltration.

The infiltration/ventilation air quantity estimation is usually done by one of the three methods 1) air change method, 2) infiltration through the cracks and 3) based on occupancy i.e. number of people in the space.

Ventilation rate based on Air change method:

V = ACH * A * H / 60

Where

- V = Ventilation air in CFM
- ACH = Air changes per hour usually 0.15 to 0.5 ACH depending on the construction of the building
- A = Area of the space in ft^2
- H = Height of the room in ft

Note A * H is the volume of the space.

Ventilation rate based on Crack method:

Volume of air = I * A

Where

- V = Ventilation air in CFM
- I = Infiltration rate usually 0.15 cfm/ft^2
- A = Area of cracks/openings in ft^2

Ventilation rate based on Occupancy method:

V = N * 20

Where

- V = Ventilation air in CFM
- N = Number of people in space usually 1 person per 100 sq-ft for office application
- 20 = Recommended ventilation rate is 20 CFM/person [based on ASHRAE 62 standard for IAQ]

In heat loss estimation, we choose the method that gives the most amount of load.

As soon as the volume flow rate of infiltrated air, CFM, is determined, the sensible heat loss from infiltration can be calculated as

$Q = V * \rho_{air} * C_p * (Ti – To) * 60$

Where:

- $Q_{sensible}$ is sensible heat load in (Btu/hr)
- V = volumetric air flow rate in (cfm)
- ρ_{air} is the density of the air in (lbm/ft^3)
- C_p = specific heat capacity of air at constant pressure in (Btu/lbm -F)
- Ti = indoor air temperature in (°F)
- To = outdoor air temperature in (°F)

ANNUAL HEATING VALUE

The annual heating value is the function of the "degree days" of heating.

Heating **degree day** is defined as a measure of the coldness of the weather experienced. The degree-day concept has traditionally been used to determine the coldness of a climate. When the weather is slightly cool, a little bit of heat might be needed for a few hours in the evening or early morning to stay comfortable. On a very cold day, a lot of heat will be needed all day and all night. A day's average temperature gives some idea of how much heat will be needed on that day. Climatologists use a measurement known as heating degree-days (HDDs) to estimate heating needs more precisely. They assume that people will use at least some heat on any day that has an average outdoor temperature of less than 65ºF. They then calculate the heating needs for each day by subtracting the day's average temperature from 65. The result is the number of heating degrees for that day or HDDs. The higher the number, the more fuel will be used in heating your home or building.

Example for any given day:

High Temp = 50° F

Low Temp = 20° F

Average Temperature = $\frac{50° + 20° F}{2}$ = 35° F

Degree Day = 65°F - 35° F = 30° F

Therefore, the day was a 30 Degree Day.

From the above data, we can make an educated guess about the annual heat loss. To determine the annual heat loss, divide the energy loss rate by the design temperature difference and then multiply it by 24 hours per day and the number of annual degree days (from the weather files of the location).

For example, a house with a design heating load of 30,000 Btu/hr in Pittsburgh (average temperature of 4°F) will use:
[30,000 Btu/hr * 24 hr/day / (65 - 4) (°F)] x 6000 DD/yr = 71million Btu/yr

The concept of degree days is used primarily to evaluate energy demand for heating and cooling services. In the United States, for example, Pittsburgh, Columbus, Ohio, and Denver, Colorado, have comparable annual degree days (about 6000 DD/year). It can be expected that the same structure in all three locations would have about the same heating bill. Move the building to Great Falls, MT (7800 DD/year), it would have a higher heating bill; but in Albuquerque, NM, (4400 DD/year), it would have a relatively lower heating cost.

Although the degree day reading is useful, keep in mind that other factors such as sun load or excessive infiltration due to high wind also affect the heating requirements of a building and are not taken into account by the degree day calculation.
We will learn more about the Degree days and the Heat loss estimation in a sample example presented in section-3 of the course but before that let’s briefly discuss the concepts of heat transmission.

SECTION # 2 HEAT TRANSMISSION THROUGH BUILDINGS

The Physics of Heat Transmission

Although it is not necessary to understand the physics of heat movement, it is useful to understand it in general terms. Heat transfer is the tendency of heat or energy to move from a warmer space to a cooler space until both spaces are the same temperature. Obviously the greater the difference in temperatures, the greater will be the heat flow. There are three types of heat transfer:

1. Via Conduction - This occurs when two objects are in direct contact, for example the air against a window or the soil against a foundation. In buildings, this is typically the most significant method of heat transfer. Conduction moves in all directions at the same time. The total heat transferred by conduction varies *directly* with time, area, and temperature difference, and *inversely* with the thickness of the material through which it passes.

2. Via Convection - This occurs within a fluid medium (e.g. air or water) and is the result of the warmer part of the fluid rising while the colder part sinks. Convection results in the entire fluid rapidly reaching the same temperature. The old saying that "heat rises" is really a misstatement that should say "warm air rises". Heat has no sense of direction, but warm air being lighter rises due to being displaced by colder air which has a greater pull of gravity. The heated air leaking out through door and window openings is an example of convection.

3. Via Radiation - This occurs between a warm object and a colder object when they are separated only by a medium which is transparent to infrared radiation. This is easiest to understand by just standing in the sun: while the sun is very far away, it is also very big and very hot while space and the atmosphere block very little of that incoming radiation. With smaller and much cooler objects, radiation is a much less significant source of heat transfer, although its affects can still easily be noticed. In a home, windows are transparent to some heat radiation (more about this in solar power), but the rest of the building is relatively opaque.

The primary heat loss is via conduction and convection. Let's discuss these further.

Heat Loss by Conduction

With buildings, we refer to heat flow in a number of different ways: “k” values, “C” values, "R" values and “U” values.

What it all means?

Basically all these letter symbols denote heat transfer factors and describe the same phenomenon; however, some are described as determined by material dimensions and boundaries.

---

k = Thermal Conductivity

The letter *"k"* represents thermal conductivity, which is the rate of heat transfer through one inch of a homogeneous material. A material is considered homogeneous when the value of its thermal conductivity does not depend on its dimension. It is the same number regardless of the thickness. Thermal Resistance, or “R” is the reciprocal of thermal conductivity i.e. R = 1/k. Thermal conductivity is expressed in (Btu-in/hr ft^2 °F). Materials with lower k-values are better insulators.

Example:

Calculate the heat loss through a 3” thick insulation board that has an area of $2ft^2$ and has a k-value of 0.25. Assume the average temperature difference across the material is 70°F.

Solution:

Q = 0.25 *(k)* * 2 (ft^2) * 70°F (ΔT) / 3 (in. of thickness)

Q = 35 / 3 = 11.66 Btu/hr

It should be apparent from the example that in order to reduce heat transfer, the thermal conductivity must be as low as possible and the material be as thick as possible. Most good insulating materials have a thermal conductivity (k) factor of approximately 0.25 or less, and rigid foam insulations have been developed with thermal conductivity (k) factors as low as 0.12 to 0.15.

Note: In some technical literature, k-values are based on thickness per foot instead of per inch.

C = Thermal Conductance

The letter "C" represents thermal conductance, which, like thermal conductivity, is a measure of the rate of heat transfer through a material but it differs from conductivity (k -value) in one significant way*. Thermal conductance is a specific factor for a given thickness of material whereas thermal conductivity is a heat transfer factor per inch of thickness.* The lower the C value, the better the insulator or lower the heat loss.

Typically, building components such as walls or ceilings consist of a "series" or layers of different materials as you follow the heat flow path out. The overall C value is not additive because if you were to take two insulating materials with a C-value of .5 each and were to add them together, you get the result of a total C-value of 1.0. This would mean that the heat flow rate has increased with the addition of more insulating material. Obviously then you cannot add C-values to find the "series" value.

Therefore, we now have to bring in the perhaps more familiar "R"-value which is a measure of a material's Resistance to heat flow and is the inverse or reciprocal of the material's C-value (R=1/C).

So if a material has a C-value of .5, it has an R-value of 2 (1/.5). If you have to add two materials in series or layers, say each with a C-value of .5, you take the inverse of both to get an R-value for each of 2. These can be added together to get a total R-value of 4.

---

h = Film or Surface Conductance.

Heat transfer through any material is affected by the resistance to heat flow offered by its surface and air in contact with it. The degree of resistance depends on the type of surface, its relative roughness or smoothness, its vertical or horizontal position, its reflective properties, and the rate of airflow over it. It is similar to thermal conductance and is expressed in Btu/ (hr ^{0}F ft^2).

R = Thermal Resistance

The thermal resistance (R) is a measure of the ability to retard heat flow in a given thickness of material. By definition, the resistance of a material to the flow of heat is the reciprocal of its heat transfer coefficient. In other words, the *R-value* is the reciprocal of either the k-value or the C-value.

When a building structure is composed of various layers of construction elements, the overall total resistance is the sum of all individual resistances for whole wall, internal air spaces, insulation materials and air films adjacent to solid materials. Individual R-values for common building materials can be checked from the ASHARE fundamentals handbook.

U = Overall Coefficient of Heat Transmission

The U-value is the rate of heat flow passing through a square foot of the material in an hour for every degree Fahrenheit difference in temperature across the material (Btu/ft^{2}hr°F).

For thermal heat loss calculations, we normally use U-values (U for Unrestrained heat flow) which is a material's C-value but also includes the insulating effect of the air films on either side of the material. So it is, therefore, a smaller number (less heat flow).

As with C-values discussed above, you can not add U-values for series calculations. To obtain a U-value for such an assembly, you add the individual R-values of the layers and the air films on either side of the assembly. Then you take the reciprocal of the total R-value to get the total U-value of the assembly ($U = 1/R_{Total}$).

Here are a few of the most common covering materials and their associated “U” factors:

Material	**“U” Value (Btu / hr-ft^2-°F)**
Glass, single	1.13
Glass, double glazing	.70
Single film plastic	1.20
Double film plastic	.70

Material	"U" Value (Btu / hr-ft^2-°F)
Corrugated FRP panels	1.20
Corrugated polycarbonate	1.20
Plastic structured sheet; 16 mm thick 8 mm thick 6 mm thick	 .58 .65 .72
Concrete block, 8 inch	.57

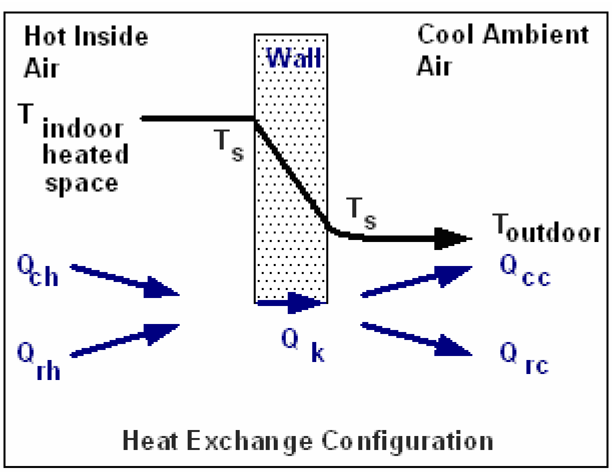
Note that the windows are commonly described by their U-values while descriptions of building walls, floors, or ceilings, often use R-values which is than converted to U-values by inverse relationship.

Combined Modes of Heat Transfer

1) Heat transfer by convection Q_{ch} and radiation Q_{rh} from the hot air and surrounding surfaces to the wall surface,

2) Heat transfer by conduction through the wall Q_k

3) Heat transfer by convection Q_{cc} and radiation Q_{rc} from the wall surface to the cold air and surrounding surfaces.



When one side of the wall is warmer than the other side, heat will conduct from the warm side into the material and gradually move through it to the colder side. A temperature gradient is established across the thickness of the wall. The temperature gradient is linear between the two surfaces for a homogenous wall and the slope of temperature gradient is proportional to the resistances of individual layers for a composite structure.

If both sides are at constant temperatures--say the inside heated surface at 77°F (25°C) and the outside surface at 40°F (4.4°C)--conductivity will carry heat inside the building at an easily predicted rate.

Under steady state conditions, the total rate of heat transfer (Q) between the two fluids is:

$$Q = Q_{ch} + Q_{rh} = Q_k = Q_{cc} + Q_{rc}$$

In real-life situations, however, the inside and outside temperatures are not constant. In fact the driving force for conductive heat flow can further increase as night falls to still lower outside air temperatures.

Calculation Methods

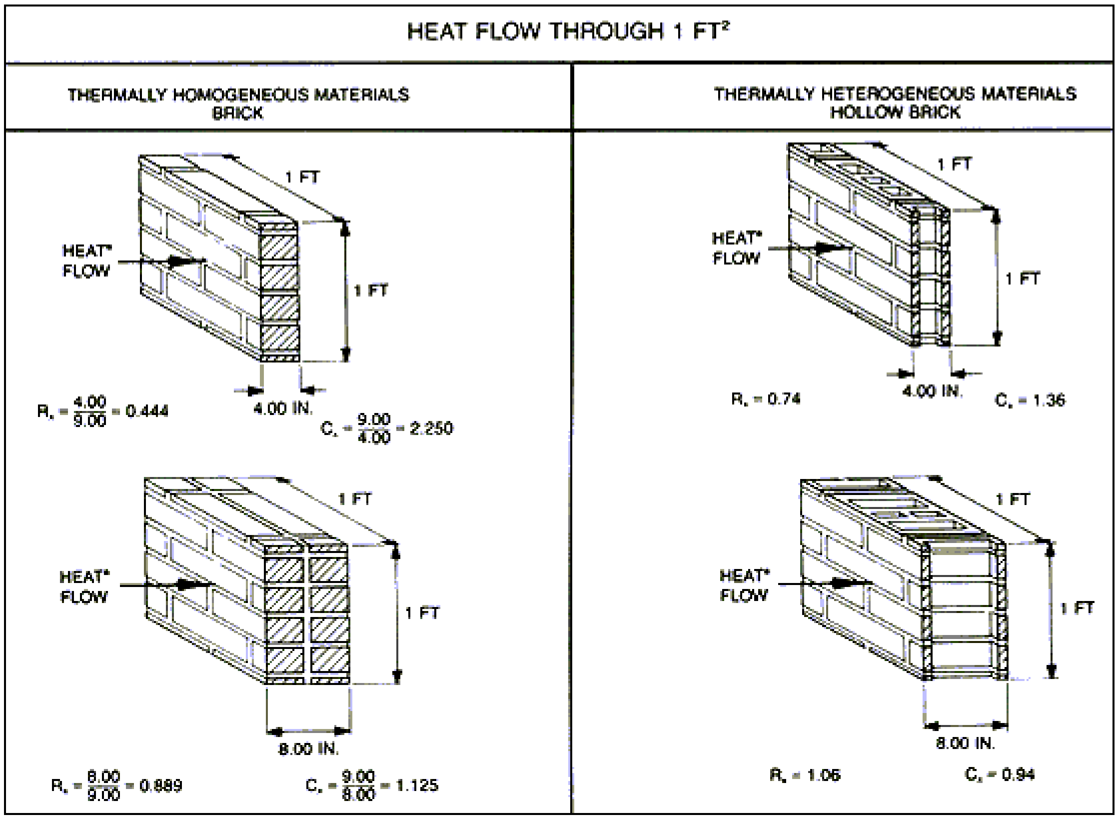
Conductance and resistances of homogeneous material of any thickness can be obtained from the following formula:
C_x=k/ x, and R_x=x /k

Where:

- x=thickness of material in inches
- k = thermal conductivity

Materials in which heat flow is identical in all directions are considered thermally homogeneous.

This calculation for a homogeneous material is shown in figure below. The calculation only considers the brick component of the wall assembly. Whenever an opaque wall is to be analyzed, the wall assembly should include both the outside and inside air surfaces. The inclusion of these air surfaces makes all opaque wall assemblies layered construction.



Thermal Transmittance through Materials

In computing the heat transmission coefficients of layered construction, the paths of heat flow should first be determined. If these are in series, the resistances are additive, but if the paths of heat flow are in parallel, then the thermal transmittances are averaged. The word "series" implies that in cross-section, each layer of building material is one continuous material. However, that is not always the case. For instance, in a longitudinal wall section, one layer could be composed of more than one material, such as wood studs and insulation, hence having parallel paths of heat flow within that layer. In this case, a weighted average of the thermal transmittances should be taken.

Series heat flow

To calculate the "R_{Total}" value of anything that is composed of multiple different materials, just add up the "R" values of each of the components. For example for composite wall (layered construction), the overall thermal resistance is:

$R_{Total} = R_1 + R_2 + ...$

Or

$R_{Total} = 1/h_o + x_1/k_1 + ...+1/C + x_2/k_2 + 1/h_i$

Where:

- h_o, h_i are the outdoor and indoor air film conductance in Btu/hr.ft^2.F
- k_1, k_2 are the thermal conductivity of materials in Btu/hr.ft^2.F
- x_1,x_2 are the wall thickness (in)
- C is the air space conductance in Btu/hr.ft^2.F

And the overall coefficient of heat transmission is:

$U = 1/R_{Total}$

Or

$$U = \frac{1}{R_i + R_1 + R_2 + ...R_o}$$

Where:

- R_i = the resistance of a "boundary layer" of air on the inside surface.
- R_1, R_2 ...= the resistance of each component of the walls for the actual thickness of the component used. If the resistance per inch thickness is used, the value should be multiplied by the thickness of that component.
- R_o = the resistance of the "air boundary layer" on the outside surface of the wall.

The formula for calculating the U factor is complicated by the fact that the total resistance to heat flow through a substance of several layers is the sum of the resistance of the various layers. The resistance to heat flow is the reciprocal of the conductivity. Therefore, in order to calculate the overall heat transfer factor, it is necessary to first find the overall resistance to heat flow, and then find the reciprocal of the overall resistance to calculate the U factor.

Note that in computing U-values, the component heat transmissions are not additive, but the overall U-value is actually less (i.e., better) than any of its component layers. The U-value is calculated by determining the resistance of each component and then taking the reciprocal of the total resistance. Thermal resistances (R-values) must first be added and the total resistance (R-Total) divided into 1 to yield the correct U-factor.

Correct:

$$U = \frac{1}{R_1 + R_2 + R_3 + \ldots + R_n} = \frac{1}{R_{Total}}$$

Incorrect:

$$U = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \ldots + \frac{1}{R_n} = U_1 + U_2 + U_3 + \ldots + U_n$$

The total R-value should be calculated to two decimal places, and the total U-factor to three decimal places.

Example #1

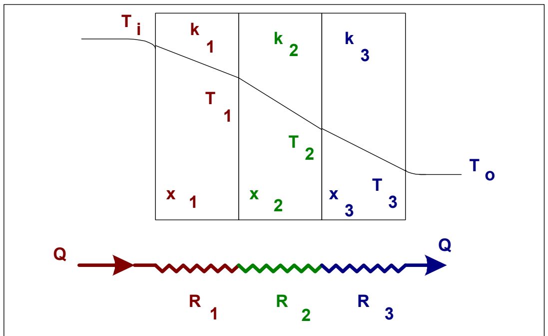
Determine the U-value for a layered wall construction assembly composed of three materials:

1) Plywood, 3/4-inch thick (R_1 = 3/4 X 1.25 = 0.94)

2) Expanded polystyrene, 2-inches thick (R_2 = 2" X 4.00 = 8.00)

3) Hardboard, 1/4-inch thick (R_3 = 0.18)

Assume resistance of inside still air is Ri = 0.68 and resistance of outside air at 15 mph wind velocity is Ro = 0.17



Thermal Resistance of Composite Wall

The U-values is:

$$U = \frac{1}{R_i + R_1 + R_2 + R_3 + R_{o_o}}$$

$$= \frac{1}{0.68 + 0.18 + 8.00 + 0.94 + 0.17}$$

$$= \frac{1}{9.97} = 0.10 \frac{\text{BTU}}{\text{hr. - sq. ft. - °F}}$$

To calculate heat loss for say for 100 square feet of wall with a 70° F temperature difference, the Q will be:

$$Q = (.10)\,(100)\,(70) = 700 \ \text{BTU/ HR}$$

In the calculations above the ΔT is taken as 70°F, which is temperature difference between indoor and outside air. If the sun shines on a wall or roof of a building and heats the surface much hotter than the air (as typical in the summer), the heat flow through the wall or roof would be greatly influenced by the hot surface temperature; hence, use a surface temperature rather than air to obtain a more realistic heat flow rate. Similarly, when calculating the heat flow through a floor slab resting on the ground, there will not be an air boundary-layer resistance underneath (Ro = 0) and the temperature (t_o) will be the ground temperature (not the outside air temperature).

Example # 2

Calculate the heat loss through 100 ft^2 wall with an inside temperature of 65°F and an outside temperature of 35°F. Assume the exterior wall is composed of 2" of material having a ‘k’ factor of 0.80, and 2" of insulation having a conductance of 0.16.

Solution:

U value is found as follows:

R total = 1/C + x1/k1 or

R total = 1/0.16 + 2/0.80

R total = 8.75

U = 1/R or 1/8.75 = 0.114 Btu/hr ft^2 °F

Once the U factor is known, the heat gain by transmission through a given wall can be calculated by the basic heat transfer equation:

$Q = U \times A \times \Delta T$

Q = 0.114 x 100 x 30

Q = 342 Btu/hr

Conductance and resistance coefficients of various wall elements are listed in Table below: These coefficients were taken from the 1981 ASHRAE *Handbook of Fundamentals,* Chapter 23.

HEAT TRANSMISSION COEFFICIENTS OF COMMON BUILDING MATERIALS

Material Description	***Density Lb/ft^3***	***Conduction (k) Btu-in/hr ft^2 °F***	***(C) Btu/hr ft^2 ^{0}F***	***Resistance (R) Per inch thickness x/k***	***For thickness listed 1/C***
Masonary Units					
Face Brick	130	9.00		0.11	
Common Brick	120	5.00		0.20	
Hollow Brick					
4” (62.9% solid)	81		1.36		0.74
6” (67.3% solid)	86		1.07		0.93
8” (61.2% solid)	78		0.94		1.06

Material Description	***Density Lb/ft^3***	***Conduction (k) Btu-in/hr ft^2 °F***	***(C) Btu/hr ft^2 ^{0}F***	***Resistance (R) Per inch thickness x/k***	***For thickness listed 1/C***
10” 60.9% solid)	78		0.83		1.20
Hollow Brick vermiculite fill					
4” (62.9% solid)	83		0.91		1.10
6” (67.3% solid)	88		0.66		1.52
8” (61.2% solid)	80		0.52		1.92
10” 60.9% solid)	80		0.42		2.38
Lightweight concrete block-100 Lb density concrete					
4”	78		0.71		1.40
6”	66		0.65		1.53
8”	60		0.57		1.75
10”	58		0.51		1.97
12”	55		0.47		2.14
Lightweight concrete block vermiculite fill - 100 Lb density concrete					
4”	79		0.43		2.33
6”	68		0.27		3.72
8”	62		0.21		4.85
10”	61		0.17		5.92
12”	58		0.15		6.80
Building Board					
3/8” -Drywall Gypsum	50		3.10		0.32
1/2” -Drywall Gypsum	50		2.25		0.45
Plywood	34	0.80		1.25	
½” Fiberboard sheathing	18		0.76		1.32
Siding					
7/16” hard board	40			1.49	0.67
½” by 8” Wood bevel	32			1.23	0.81
Aluminum or steel over sheathing				1.61	0.61
Insulating Material					
Boards					
• Expanded Polystrene	1.80	0.25		4.00	
• Expanded Polyurethane	1.50	0.16		6.25	
• Poly isocyanurate	2.0	0.14		7.14	
Loose Fill					
• Vermiculite	4 - 6	0.44		2.27	
• Perlite	5 - 8	0.37		2.70	
Woods					
Hard woods	45.0	1.1		0.91	

Material Description	*Density Lb/ft^3*	*Conduction* ***(k) Btu-in/hr ft^2 °F***	***(C) Btu/hr ft^2 ^{0}F***	*Resistance (R) Per inch thickness x/k*	*For thickness listed 1/C*
Soft woods	32.0	0.80		1.25	
Metals					
Steel	-	312		0.003	
Aluminum	-	1416		0.0007	
Copper	-	2640		0.0004	
Air Space					
¾” to 4”- winter			1.03		0.97
¾” to 4” - summer			1.16		0.86
Air Surfaces					
Inside – Still air			1.47		0.68
Outside – 15 mph wind-winter			5.88		0.17
Outside – 7.5 mph wind - summer			4.00		0.25

Heat Loss by Convection

The other mechanism of heat loss is convection, or heat loss by air movement. In homes, this is principally heat loss by exfiltration and infiltration. Exfiltration is the loss of heated air through building cracks and other openings. Infiltration is the introduction of outside cold air into the building. This air movement also causes discomfort (drafts) to occupants in addition to the heat loss itself.

The driving force for this exchange of air is the difference between indoor and outdoor air pressures. Air pressure differences are principally caused by wind pressures and the "stack" effect of warm inside air that tends to rise. Mechanically induced air pressure differences can also occur due to such things as exhaust fans and furnace venting.

To calculate the heat loss by convection, we go back to the general heat loss calculation and modify it to:

Heat Loss = Heat Capacity of Air * Air Volume Exchanged/hour * Temp. Difference

The volume exchanged can be determined by measuring or judging how many air changes that a building goes through in an hour. You can assume a rate between .25 and .50 air changes per hour (ACH), usually with a lower rate for basements with little outside air exposure, and higher rates for living areas or exposed basements.

The heat capacity of air is product of ρ_{air} * Cp and is equivalent to 0.018 Btu per (°F) (cu.ft.)

Where

- o ρ_{air} is the density of the air in (lbm/ft³)
- o C_p = specific heat capacity of air at constant pressure in (Btu/lbm -F)

Example

If you have a 1500 square foot house on a crawl space with 8-foot ceilings, the calculation of the volume exchanged can be:

1500 sq. ft. x 8 ft x .25 ACH

= 3000 ft^3/hr

Heat Loss

The heat capacity of air is a physical constant and is .018 Btu per (°F) (cu. ft.). Considering an outside temperature of -20°F and indoor temperature of 70°F, the heat loss due to infiltration will be:

.018 Btu/ (°F) (ft^3) x 3,000 ft^3/hr x 90°

= 4860 Btu/hr

Another method of determining heat loss by convection is the crack method. For this method you obtain the air leakage rates in cubic feet per minute for the doors and windows from their manufacturers and multiply by the lineal feet of sash crack or square feet of door area. (A more exact analysis would multiply the door infiltration rates by 1 or 2 due to open/close cycles and add .07 CFM per linear feet of foundation sill crack). This gives an air change rate per minute. This has to be converted to an hourly rate by multiplying by 60. Then you substitute this figure for the air change rate in the infiltration heat loss equation above.

A heating system is to be designed for the top 3 floors of an office building in Montreal area with following specifications:

Specification:

Location:		Montreal
Type of building:		office
Number of floors:		3
Floor area:		64ft x 80ft =5120 ft
Floor to floor height:		12 ft
Window area:		25% of wall area
Wall construction:	Face brick-	4 in
	Styrofoam insulation -	2 inches
	Concrete block -	8 inches
	Air space -	1.5 in
	Plaster board-	0.5 in
Roof construction:	Tar and gravel (built-up) -	0.375 in
	Rigid insulation -	4 in
	Concrete -	8 in
	Air space -	16 in
	Acoustic tile -	0.5 in
Windows:		double glazed (U value = 0.70)

Ventilation:	mechanical
Recommended ventilation:	minimum ½ air change per hour
	0.05 to 0.25 cfm/ft^2
	20 cfm/person, 8 persons/1000 ft^2 (max)

Calculation Methodology:

To calculate a design heating load, we should prepare the following design about building design and weather data at design conditions.

1. Outdoor design weather conditions: temperature, wind speed.
2. Decide on the Indoor air temperature
3. Divide the building into thermal zones (exterior and interior)
4. Determine heat transfer coefficients (U-values) for outside walls, glass and roof by finding the inverse of the sum of individual R-values for each layer of material
5. Determining the net area of outside walls, glass and roof.
6. Computing heat transmission losses for each kind of wall, glasses and roof.
7. Computing infiltration around outside doors, windows porous building materials and other openings.
8. The sum of the transmission losses or heat transmitted through walls, ceiling and glass plus the energy associated with cold air entering by infiltration or the ventilation air required to replace mechanical exhaust, represents the total heating load.

Design Conditions:

Location: Montreal

Outdoor air: - 20°F

Indoor air: 70°F

Wind velocity: 15 mph

Reference: 1993 ASHRAE Handbook- Fundamentals, pp.24. 17

Zone Division:

Identifying the thermal zones is the first step in the design of any HVAC system. Thermal Zoning is a method of designing and controlling the HVAC system so that occupied areas can be maintained at a different temperature than unoccupied areas using independent setback thermostats.

A zone is defined as a space or group of spaces in a building having similar heating and cooling requirements throughout its occupied area so that comfort conditions may be controlled by a single thermostat. In practice the corner rooms and the perimetric spaces of the building have variations in load as compared to the interior core areas. The buildings may be zoned into individual floors, rooms, or spaces with distinct loads, such as perimeter and interior zones. Smaller buildings are usually divided into two major zones.

a. Exterior Zone: The area inward from the outside wall (usually 12 to 18 feet if rooms do not line the outside wall). The exterior zone is directly affected by outdoor conditions during summer and winter.

b. Interior Zone: The area contained by the external zone. The interior zone is only slightly affected by outdoor conditions. Thus, the interior zone usually has uniform cooling. Heating is generally provided from the exterior zone.

In our example, the whole building envelope is divided into six zones:

1. Corner zone on the 1st and 2nd floors (zone 1)

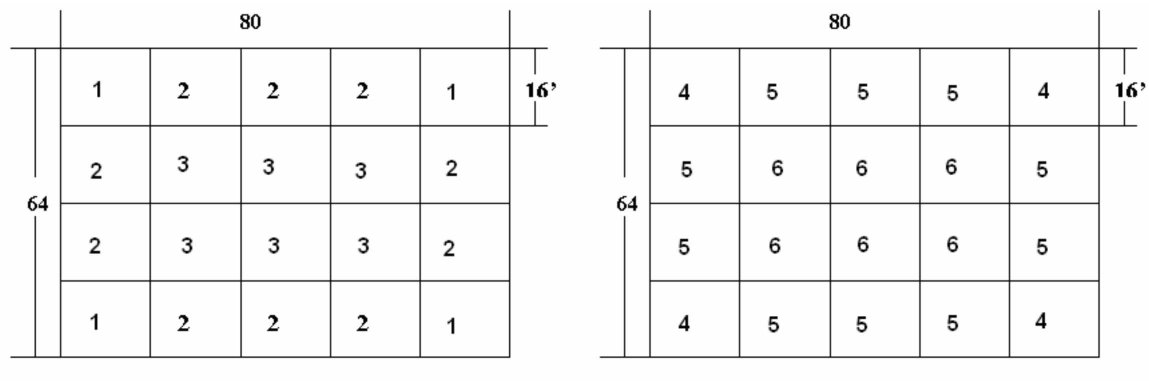
2. A corner zone on the 3rd floor (zone 4)

3. A central zone on the 1st and 2nd floors (zone 3)

4. A central zone on the 3rd floor (zone 6)

5. An interior zone on the 1st and 2nd floors (zone 2)

6. An interior zone on the 3rd floor (zone 5)



1st and 2nd Floor

3rd Floor

HEAT LOSS CALCULATION

Heat losses from the different zones will be calculated in steps and the overall heat loss is obtained from the sum of the heat loss through the individual zones.

There are two types of heat losses from the building envelope that will be considered.

1. $Q_{Conductive}$

2. $Q_{Infiltration}$

The total heat loss is the summation of conductive and infiltration loss.

$Q_{Total} = Q_{Conductive} + Q_{Infiltration}$

CONDUCTIVE HEAT LOSS ($Q_{Conductive}$)

Step – 1:

Calculate the U-Value of Wall Material by finding the inverse of the sum of individual R-values for each layer of material.

Table – 1 Total resistance of Wall Construction

Layer	x (inch)	(k) Btu-in/hr ft^2 °F	(C) Btu/hr ft^2 ^0F	(h) Btu/hr ft^2 ^0F	R =x/k =1/c = 1/h ft^2 ^0F hr / Btu
Outside air	Film	-	-	5.88	0.17
Face brick	4”	9	-	-	0.44
Styrofoam	2”	-	0.151	-	6.62
Concrete	8”	-	0.57	-	1.75
Air space	1.5”	-	1.03	-	0.97
Plaster board	0.5”	-	2.25	-	0.44
Inside air	Film	-	-	1.47	0.68
Total					11.07

Source: 1997 ASHARE Fundamentals Handbook, Tables 22-1, 22-2, 22-4

Heat transfer coefficient for the wall U = 1/R = 1/11.07 = 0.09 Btu/hr ft^2 ^0F

Step -2: Calculate the U-Value of Roof Construction

Table – 2 Total resistance of the roof components:

Layer	x (inch)	(k) Btu-in/hr ft^2 °F	(C) Btu/hr ft^2 ^0F	(h) Btu/hr ft^2 ^0F	R =x/k =1/c = 1/h ft^2 ^0F hr / Btu
Outside air	Film	-	-	5.88	0.17
Tar-gravel	0.375”	-	2.99	-	0.33
Insulation	2”	0.14	-	-	14.29
Concrete	8”	-	0.57	-	1.75
Air space	4”	-	1.03	-	0.970

Layer	**x (inch)**	**(k) Btu-in/hr ft^2 °F**	**(C) Btu/hr ft^2 ^{0}F**	**(h) Btu/hr ft^2 ^{0}F**	**R =x/k =1/c = 1/h ft^2 ^{0}F hr / Btu**
Acoustic tile	0.5”	-	0.14	-	7.14
Inside air	Film	-	-	1.63	0.61
Total					25.26

Source Reference: 1997 ASHARE Fundamentals Handbook, Tables 22.1, 22.2, 22.4

Heat transfer coefficient for the roof U = 1/R =1/25.26 = 0.04 Btu/hr ft^2 ^{0}F

Step - 3: Calculate the Heat Loss

Heating load of the surfaces:

$Q_{Conduction}$ = Q wall + Q roof + Q window

The calculation is made for center zone on the 3rd floor – zone 4:

Conductive Loss thru Wall (Q wall)

Q wall = U *A * ΔT

U = 0.09 Btu/hr ft^2 ^{0}F

ΔT = 90°F ---- [Ti = 70°F and To = -20°F]

Net Area of Wall

Area of surface = 16 * 12 = 192 ft^2

Area of glazing = 25% * 192 = 48 ft^2

Area of walls = Area of surface – Area of glazing = 192 – 48 = 144 ft^2

Number of surface walls = 8 no --- [refer to zoning diagram for 3rd floor]

Total Area of walls in zone - 4 = 1152 ft^2

Q wall = 0.09 * 1152 * 90

Q wall = <u>9331 Btu/hr</u>

<u>Conductive Loss thru Roof (Q roof)</u>

Q roof =U * A * ΔT

U = 0.04 Btu/hr ft^2 ^{0}F

ΔT = 90°F

<u>Net Area of Roof</u>

Area of surface = 16 * 16 = 256 ft^2

Number of zone-4 roofs = 4 no

Total Area of roof = 256 * 4 = 1024 ft^2

Q roof = 0.04 * 1024 * 90

Q roof = <u>3686 Btu/hr</u>

<u>Conductive Loss thru Glazing (Q window)</u>

Q window =U * A * ΔT

U = 0.70 Btu/hr ft^2 ^{0}F

ΔT = 90°F

A = 48 ft^2 [25% of the wall area]

Total Glazing Area = 8 * 48 = 384 ft^2 [… there are 8 surface walls in zone-4]

Q window = 0.70 * 384 * 90

Q window = <u>24192 Btu/hr</u>

So: $Q_{Conductive}$ = Q wall + Q roof + Q window

Or: Q1 = 9331 + 3686 + 24192

Q1 = 37209 Btu/hr

Total Conduction heat losses:

The conduction heat loss above is done for Zone-4, third floor. If you repeat this for all other zones of the building, you could obtain the total heat loss through the envelope at design temperatures.

Refer to the zoning diagram –

Zone 1 has 8 walls on 1st floor and 8 walls on the second floor – thus total number of surfaces = 16 number.

The other zones are calculated likewise in table below:

<u>Table 3</u> Conduction heat losses:

Zone	Surface	Area ft^2	Number of surfaces; n	U Btu/hr $ft^2\ {}^0F$	ΔT (° F)	Q=U *A * ΔT Btu/hr	Total, Q Btu/hr
1.	Wall Window Roof	144 48 -	16 16 -	0.09 0.70 -	90 90 -	18662 48384 -	67046
2.	Wall Window Roof	144 48 -	20 20	0.09 0.70 -	90 90 -	23328 60480 -	83808
3.	Wall Window Roof	- - -	- - -	- - -	- - -	- - -	-
4.	Wall Window Roof	144 48 256	8 8 4	0.09 0.70 0.04	90 90 90	9331 24192 3686	37209
5.	Wall Window	144 48	10 10	0.09 0.70	90 90	11664 30240	51120

Zone	Surface	Area ft^2	Number of surfaces; n	U Btu/hr ft^2 ^0F	ΔT (° F)	Q=U *A * ΔT Btu/hr	Total, Q Btu/hr
	Roof	256	10	0.04	90	9216	
6.	Wall Window Roof	- - 256	- - 6	- - 0.04	- - 90	- - 5530	5530

Total = 244713 Btu/hr

HEAT LOSS BY VENTILATION ($Q_{Infiltration}$)

Calculation of volume of air:

For finding ventilation rate in "cfm", we choose the method that gives the most amount of load. For this reason we calculate cfm based on three methods (air change, crack and the people).

<u>Ventilation rate based on Air change method:</u>

According to division, the area of each zone is 16 * 16 = 256 ft^2

Volume of each zone = 256 * 12 = 3072 ft^3

Recommended air change/hr = 0.5

Volume of air = ½ ACH * 3072 / 60

Volume of air = 25.6 cfm/zone

<u>Ventilation rate based on Crack method:</u>

According to division, the area of each zone is 16 * 16 = 256 ft^2

Expected infiltration = 0.15 cfm/ft^2

Volume of air = 0.15 * 256

Volume of air = 38.4 cfm/zone

Ventilation rate based on Occupancy method:

According to division, the area of each zone is 16 * 16 = 256 ft^2

Recommended ventilation rate = 20 cfm/person [based on ASHRAE 62 recommendation for IAQ]

Number of people = 8 people/1000 sq-ft

Volume of air = cfm/person * number of people in one zone (256 ft^2 area)

Volume of air = 20 * 8 * 256/1000

Volume of air = 40.96 cfm/zone

Or

Total ventilation for the building (60 zones) = 40.96 * 60 = 2457cfm

Here, cfm according to people is more than the other ones and therefore, we will consider this as method of ventilation for calculating heat loss.

Heat Loss by Ventilation

$Q_{Ventilation} = V * \rho_{air} * C_p * (Ti - To) * 60$

Where:

- $Q_{sensible}$ is sensible heat load in (Btu/hr)
- V = volumetric air flow rate in (cfm)
- ρ_{air} is the density of the air in (lbm/ft^3)
- C_p = specific heat capacity of air at constant pressure in (Btu/lbm -F)
- Ti = indoor air temperature in (°F)
- To = outdoor air temperature in (°F)

Heat loss for ventilation from one zone:

$Q_{ventilation}$ = 0.075 * 40.96 * 0.24 * 90 * 60 = 3981Btu/hr

RESULTS

Table - 4: Total Heat Loss:

Zone Designation	**No. Of Zones**	**Ventilation heat loss per zone Btu/hr**	**$Q_{ventilation}$ Btu/hr**	**$Q_{conductance}$ Btu/hr**	**Total heat loss Q_{Total} Btu/hr**
1.	8	3981	31848	67046	98894
2.	20	3981	79620	83808	163428
3.	12	3981	47772	-	47772
4.	4	3981	15924	37209	53133
5.	10	3981	39810	51120	90930
6.	6	3981	23886	5530	29416
Total	60		238860	244713	**483573**

Q_{Total} with 10% safety factor = 483573 * 1.1 = 531930 Btu/hr

CONCLUSION

In this example, the total heating load for the building is 531930 Btu/hr with 10% safety factor. This value shall be used for sizing the heating furnace.

Total ventilation required for the total building is 2457cfm that with 10% safety factor is equal 2702cfm.

ANNUAL HEAT LOSS

After determining the total heat loss rate, we are going to take our calculation one step further to determine the annual heating loss and its related cost. If you want to figure the total seasonal heat loss, you would perform a degree day calculation.

The table below gives heating and cooling needs for 13 locations in each of the ten provinces and three territories of Canada. Heating needs in Vancouver are about half those in Winnipeg, although differences between other cities in southern Canada are less dramatic. Cooling needs, on the other hand, differ much more widely across the country.

Heating and Cooling Degree-Days for Selected Canadian Cities (Average Annual Totals, 1971–2000)

	HEATING DEGREE-DAYS	COOLING DEGREE-DAYS
St. John's, Newfoundland & Labrador	4,881	32
Charlottetown, Prince Edward Island	4,715	100
Halifax, Nova Scotia	4,367	104
Saint John, New Brunswick	4,754	37
Montreal, Quebec	4,575	235
Toronto, Ontario	4,066	252
Winnipeg, Manitoba	5,777	186
Regina, Saskatchewan	5,661	146
Edmonton, Alberta	5,708	28
Vancouver, British Columbia	2,926	44
Yellowknife, Northwest Territories	8,256	41
Whitehorse, Yukon	6,811	8
Resolute, Nunavut	12,526	0

Source: *Environment Canada*

From our heat loss calculations for Montreal, we know that the expected rate of energy loss per hour is 483573 Btu/hr. For Montreal the heating degree days are 4575 and the average winter temperature is 5°F.

The annual heat loss can be calculated by following equation:

[483573 Btu/hr * 24 hr/day / (65 – 5)] * 4575 DD/yr = 88 million Btu/yr.

Note that the value is rounded to the nearest million Btu. Since the numbers we are using in our calculations are very squishy (the infiltration rate can change by over 200%), the answer we get is really nothing more than an educated guess.

SELECTING FUEL & HEATING SYSTEM

Selecting the fuel and heating system best suited for your needs depends on many factors. These include: the cost and availability of the fuel or energy source; the type of appliance used to convert that fuel to heat and how the heat is distributed in your space; the cost to purchase, install, and maintain the heating appliance; the heating appliance's and heat delivery system's efficiency; and the environmental impacts associated with the heating fuel.

One somewhat simple way to evaluate heating options is to compare the cost of the fuel. To do that, you have to know the energy content of the fuel and the efficiency by which it is converted to useful heat.

Natural Gas: *The heating capacity of gas heating appliances is measured in British thermal units per hour (Btu/h). (One Btu is equal to the amount of energy it takes to raise the temperature of one pound of water by 1 degree Fahrenheit.) Most gas heating appliances have heating capacities of between 40,000 and 150,000 Btu/h.*

Consumption of natural gas is measured in cubic feet (ft^3). This is the amount that the gas meter registers and the amount that the gas utility records when a reading is taken. One cubic foot of natural gas contains about 1,007 Btu of energy.

Propane: *Propane, or liquefied petroleum gas (LPG), can be used in many of the same types of equipment as natural gas. It is stored as a liquid in a tank, so it can be used anywhere, even in areas where natural gas hookups are not available. Consumption of propane is usually measured in gallons; propane has an energy content of about 92,700 Btu per gallon.*

Fuel Oil: *Several grades of fuel oil are produced by the petroleum industry, but only # 2 fuel oil is commonly used for space heating. The heating (bonnet) capacity of oil heating appliances is the steady-state heat output of the furnace, measured in Btu/h. Typical oil-fired central heating appliances have heating capacities of between 56,000 and 150,000 Btu/h. Oil use is generally billed by the gallon. One gallon of #2 fuel oil contains about 140,000 Btu of potential heat energy.*

Electricity: *The watt (W) is the basic unit of measurement of electric power. The heating capacity of electric systems is usually expressed in kilowatts (kW); 1 kW equals 1,000 W. A kilowatt-hour (kWh) is the amount of electrical energy supplied by 1 kW of power over a 1-*

hour period. Electric systems come in a wide range of capacities, generally from 10 kW to 50 kW.

When converted to heat in an electric resistance heating element, one kWh produces 3,413 Btu of heat.

ANNUAL HEATING COST

To convert Btu/yr values into dollars per year for the annual heating cost, we have to guess at how much energy costs. Again these values vary widely, depending on season, geographic location and type of fuel.

Comparing Fuel Costs

Comparing fuel costs is generally based on knowing two parameters viz. the efficiency of the appliance and the unit price of the fuel. Follow the steps below:

1) Convert the Btu content of the fuel per unit to millions of Btu by dividing the fuel's Btu content by 1,000,000. For example: 3,413 Btu/kWh (electricity) divided by 1,000,000 = 0.003413 millions Btu per unit.

2) Use the following equation to estimate energy cost:

   Energy cost ($ per million Btu) = Cost per unit of fuel ÷ [Fuel energy content (in millions Btu per unit) × Heating system efficiency (in decimal)]

The table below provides examples of heat cost tabulation for different fuels and heating equipment.

Heating Equipment	**Fuel**	**Fuel Cost (Note #1)**	**Fuel energy content (in millions Btu per unit)**	**Heating System Efficiency (Note #2)**	**Heat Cost in $ per million Btu (Note #3)**
Resistance	Electric	$0.086 per	0.003412	0.99	= $25.46

Heating Equipment	Fuel	Fuel Cost (Note #1)	Fuel energy content (in millions Btu per unit)	Heating System Efficiency (Note #2)	Heat Cost in $ per million Btu (Note #3)
Baseboard		kWh			
Heat Pump	Electric	$0.086 per kWh	0.003412	2	= $12.60
Medium Efficiency Furnace	Natural Gas	$9.96 per thousand cubic feet	1.03	0.90	= $10.74
Medium Efficiency Furnace	Fuel Oil	$1.25 per gallon	0.14	0.85	= $10.5
Medium Efficiency Furnace	Propane	$1.09 per gallon	0.0913	0.85	= $14.05

Note #1: The fuel costs used are the national annual average residential fuel prices in 2001 according to the Energy Information Administration (EIA), U.S. Department of Energy. Prices will vary by location and season.

Note #2: The system efficiencies used are assumed examples only.

Note #3: Energy cost ($ per million Btu) = Cost per unit of fuel ÷ [Fuel energy content (in millions Btu per unit) × Heating system efficiency (in decimal)]

The average Btu content of fuel values make comparisons of fuel types possible. For example:

- The heat content of one gallon of fuel oil roughly equals that of 41 kWh of electricity, 137 cubic feet of natural gas, 1.5 gallons of propane, 17.5 pounds of air-dried wood, 17 pounds of pellets, a gallon of kerosene, or 10 pounds of coal.

- One million Btu is the heat equivalent of approximately 7 gallons of No. 2 heating oil or kerosene, 293 kWh of electricity, 976 cubic feet of natural gas, 11 gallons of propane, 125 pounds of air-dried wood, 121 pounds of pellets, or 71 pounds of coal.

Since this is an introductory course, we will assume one value for all situations. This assumption is too general to use for making large economic decisions, but it is certainly easier than trying to keep up with these constantly changing values. For the purposes of this course, all energy will cost exactly $10 per million Btu. At today's energy prices, this average value is high for gas heat (by about a factor of 2), about right for fuel oil, and low for electric resistance heat (by about a factor of 2). Even these prices vary substantially across the nation.

So for our example building above using 88 million Btu/yr, we would calculate the heating cost to be 88 * $10 = $880 per year. But in reality the heating cost might range from under $440 for gas heat to over $1400 for electric resistance heat.

WINDOWS

Windows provide light, ventilation and in many cases passive solar heating, but are otherwise a source of great heat loss.

In the heat loss calculation, all windows are created equal, no matter which direction they face. Disallowing for wind factors, similar types of glazing's lose heat at the same rate. On the other hand, when calculating heat gain, windows facing east and west GAIN more heat that those facing north and south. This results in larger quantities of air being distributed to rooms with east and west facing windows. This air is necessary for cooling but not for heating. In the more northern climates, heat loss occurs equally from all windows regardless of which direction they face. This will restore the emphasis on a balanced distribution system rather than one weighted toward solar radiation.

A decently insulated wall easily achieves R21, while most windows do no better than R3: meaning that a window loses seven times more heat per square foot than a wall does! So clearly we have to pay attention to how we use windows.

Windows Ratings

Windows are rated according to a standard set by NFRC (the National Fenestration Rating Council) and consist of four values that tell about the performance of the window:

1. Heat Transmission Coefficient (U-value) - tells how much heat the window will loose. In the past manufacturers measured the "U" value at the center of the glass, because it is often higher than for the whole unit. While this practice has been abandoned, buyers should verify that the "U" value given is for the entire window unit. Typically values are "U" for a double pane, Low-E, argon gas fill window is around .33 (i.e., R3), while a triple pane super window achieves a "U" of about .15 (i.e. R6.6). By comparison, an old style double pane window has a "U" of about .5 (i.e. R2). Because the frame of the window often lets more heat out than the glass, larger window units have better overall "U" values. Likewise, using true divided lights (consisting of multiple glass panes instead of one), reduces the "R" value significantly.

2. Visible Transmission (VT) - represents the percentage of the available visible light that is allowed to pass through the window. Even clear glass isn't perfectly transparent, and multiple glazing and Low-E coatings reduce this value.

3. Solar heat gain coefficient (SHGC) - signifies the percentage of the available solar gain that is allowed to pass through the window. As with visible transmission, this value is lower when multiple glazing and Low-E coatings are present. A double pane window with a Low-E coating that stops solar gain and allow only 30-40% of the solar gain through, while a Low-E designed to allow solar gain lets in only slightly less than plain clear glass, about 75%. By comparison, a super-window which has a U of .15 has a SHGC of only 50%. Note that these numbers are for glass only, and must be reduced to account to the space taken up by the frame and any pane dividers that shrink the overall glass area.

4. Air Leakage (AL) - air leakage, the amount of air leakage through the window. Note that this does not include the air leakage around the window unit where it is attached to the wall, which should be sealed tightly. The air leakage amount for a window that opens is higher than one that doesn't (due to greater difficulty in sealing) and that for double hung or sliding windows is greater than that for awning or casement.

The key recommendations for energy savings include:

- Single-pane windows are impractical in heating-dominated climates. In these regions, multiple-pane, low-E, and gas-filled window configurations are advisable.

- Avoid aluminum frame windows or specify aluminum-frame windows with thermal breaks. Even in milder climates, these windows tend to have low inside surface temperatures during the heating season, giving rise to condensation problems. Wood, vinyl, and fiberglass are the best frame materials for insulating value.

- Buy windows with energy efficient label. The window energy label lists the U-factor, solar heat gain coefficient, visible light transmittance, and air leakage rating.

INSULATING MATERIALS

Insulation is the material added to a building structure when the building materials themselves don't provide the desirable amount of resistance to heat transfer. The amount of insulation that can be added is limited to the available space between the framing materials, and is typically the most significant factor in determining how well a wall insulates. Since the framing material itself is at best a mediocre insulator, framing act as a *thermal bridge* leaking heat, and reducing the overall "R" value of the wall. Advanced framing is a method of making six inch thick walls (instead of the traditional four inch), without increasing the amount of wood used. Alternatively, foam board sheets can be attached to the exterior of a standard 2x4 wall also yielding a better insulating wall. The use of light-gauge steel framing to replace wood creates a problem because it conducts heat so well, and so must always be used with an exterior layer of foam board insulation to stop the thermal bridging of the steel. In all cases light-gauge steel framed walls have a lower overall "R" value to a wood framed wall.

There are many kinds of insulating materials, each of which has its own set of advantages and disadvantages, and none of which are the perfect solution. A material "R" value will differ based on how it is manufactured and how it is installed and possibly other conditions as well. The numbers used here are typical, and should be used for relative comparison purposes only.

1. **Fiberglass**: In its familiar form, glass fibers are spun together and formed into batts with glue and then typically also attached to a vapor barrier backing. This glue is a skin irritant, making this form of insulation unpleasant to handle. Fiberglass in this form is a good insulator (between R3 and R3.5 for every inch installed), but does loose some insulation ability as the outside temperature gets very cold due to having large air spaces which will transfer heat by convection. Fiberglass is also available without the glues in a blown in systems that creates a higher density, and hence somewhat higher "R" value, and presumably less susceptibility to convection heat transfer. There is some concern that fiberglass fibers break down with age and create tiny sharp filaments causing a disease called silicosis.

2. **Mineral Wool:** A fibrous product made out of various mineral by-products and all having similar properties to fiberglass. Mineral wool is no longer popular in modern buildings.

3. **Polystyrene**: Polystyrene is a plastic (known mostly by the brand name "Styrofoam") made from petroleum that is "blown" with some kind of gas (i.e. filled with lots of bubbles) and formed into boards. It's "R" value is somewhat better than Fiberglass, about R4 per inch. Polystyrene is the insulation of choice for around foundations and under concrete slabs. It is also sometimes used on roofs and in other places where its board form is more convenient.

4. **Polyurethane/PolyIsocyanurate:** Like Polystyrene, these are plastics blown with a gas, but promising a higher "R" value: up to R7.5 per inch for PolyIsocyanurate. Polyurethane can be "foamed in place" in an existing wall cavity. “Icycene” brand polyurethane is marketing itself as more environmentally friendly foam, and has been used on a number of "Healthy House" demonstration projects sponsored by the American Lung Association.

5. **Air Crete:** This unique product is cement with a lot of air in it, installed as foam. Its "R" value is similar to fiberglass, but it has all the environmental properties of cement (and therefore does not have problems with silicosis). Its raw materials are abundant, but it takes a lot of energy to make it. It’s recyclable, but there is currently not a strong market for cement products.

VENTILATION & INFILTRATION

Design the HVAC system with the outdoor air rates required by ASHRAE Standard-62 to maintain indoor air quality. “Build Tight & Ventilate Right”.

Build Tight / Ventilate Right

Air leaks out of a building due to two main driving forces: 1) Wind and 2) a temperature difference, each of which creates a pressure difference between inside and outside forcing air through the cracks in the building. What this means is that on cold windy days we loose a lot of heat, and on calm days where the inside temperature is near the outside temperate, even if we open the windows wide, we get very little air movement! Clearly, we'd like to even these extremes out as much as possible to provide for a steady supply of fresh air.

Of course there is a compromise, and most people don't want to live in a hermetically sealed building. There is a point where we've tightened our house up enough so that we're not paying a big energy penalty, but not so much that mechanical ventilation is the only source of fresh air. Since air leakage varies with weather, even a very tight house that has only 1/10 ACH under normal conditions, might see 1 ACH on a cold, windy day. As with many areas of Green Building, there are no fixed answers and each person must find their own compromise. In most buildings it is difficult to get it super tight anyhow, so this question won't come up unless you're taking extra air tightening measures.

Select windows with air leakage ratings of 0.2 cubic feet per minute per square foot of window area (cfm/ft^2) or less. Check the seals between window components for air tightness. To minimize infiltration around installed windows, caulk and weather-strip cracks and joints.

Air Sealing Techniques

In general, the majority of air leaks will be found around doors & windows, followed by any place where two parts of the building meet, such as the walls against the floor or ceiling. Air, like water will find a way through any place that can possibly be gotten through.

The tightness of a house is measured in the number of times per hour all the air in the house is lost, and is often abbreviated by ACH (air changes per hour). To determine how tight a house is, a test called *the blower door test* is done. To do this, all openings in the house are closed except an exterior door and that opening is filled with a device the size of a door that contains a powerful fan, a flow meter and a pressure gage that measures the difference between inside and outside. The fan removes air from the house until there is a pressure difference of 50 Pascals (a metric pressure measurement), at which point the fan speed is adjusted so as to maintain that constant pressure difference. The amount of airflow is now equal to the air leakage of the home (at a pressure of 50 Pascals).

Remember that the amount of air leakage on any given day is determined by the wind velocity and the temperature outside, so the result of the blower door test is not how much air your house will actually leak (which varies greatly), but a relative measure. By using a formula, the air leakage under mild weather conditions can be estimated.

The blower door can also be used to find air leaks by walking around with a device that gives off smoke and looking for places where moving air moves the smoke. This is typically done once the structural work is completed, but before any finish work is done to allow for easy fixes for any leaks found.

The most important and least cost technique is to make sure you have adequate caulking and weather stripping around all windows and doors. During construction, your contractor, or an air tightening specialist should walk around the house sealing all the potential leaks, typically with either caulk or expanding spray foam.

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USEFUL TERMS

1) Ambient Air - The air surrounding a building; outside air

2) Air Change - The term air change is a rate at which outside air replaces indoor air in a space. It can be expressed in one of two ways: the number of changes of outside air per unit of time air changes per hour (ACH); or the rate at which a volume of outside air enters per unit of time - cubic feet per minute (CFM).

3) Building Envelope - The term building envelope indicates the surfaces that separate the inside from the outdoors. This includes the parts of the building: all external building materials, windows, walls, floor and the roof. Essentially the building envelope is a barrier between the conditioned indoor environment and the outdoors.

4) Building Location Data- Building location data refers to specific outdoor design conditions used in calculating heating and cooling loads.

5) British thermal unit (BTU): Theoretically, it is approximate heat required to raise 1 lb. of water 1 deg Fahrenheit, from 59^0F to 60^0F. Its unit of heat and all cooling and heating load calculations are performed in Btu per hour in US.

6) Cooling load: The rate at which heat is removed from a space to maintain the constant temperature and humidity at the design values

7) Cooling Load Temperature Difference (CLTD) – A value used in cooling load calculations for the effective temperature difference (delta T) across a wall or ceiling, which accounts for the effect of radiant heat as well as the temperature difference. CLTD value calculates the instantaneous external cooling load across a wall or roof. CLTD value is used to convert the space sensible heat gain to space sensible cooling load.

8) Cooling Coil Load – The rate at which heat is removed at the cooling coil that serves one or more conditioned spaces and is equal to the sum of all the instantaneous space cooling loads.

9) Cubic feet per minute (CFM) - The amount of air, in cubic feet, that flows through a given space in one minute. 1 CFM equals approximately 2 liters per second (l/s). A typical system produces 400 CFM per ton of air conditioning.

10) Comfort Zone- The range of temperatures, humidity’s and air velocities at which the greatest percentages of people feel comfortable.

11) Design Conditions- Cooling loads vary with inside and outside conditions. A set of conditions specific to the local climate is necessary to calculate the expected cooling load for a building. Inside conditions of 75°F and 50% relative humidity are usually recommended as a guideline. Outside conditions are selected for the 2.5% climate occurrence.

12) Exfiltration- Uncontrolled air leakage out of a building through window and door openings

13) Exhaust - The airflow leaving the treated space from toilets, kitchens, laboratories or any hazardous area where negative pressure is desired.

14) Enthalpy - Heat content or total heat, including both sensible and latent heat.

15) Fenestration – is an architectural term that refers to the arrangement, proportion and design of window, skylight and door systems within a building. Fenestration consists of glazing, framing and in some cases shading devices and screens.

16) Heating load: The heating load is a rate at which heat is added to the space to maintain the indoor conditions.

17) Infiltration- Leakage of air inward into a space through walls, crack openings around doors and windows or through the building materials used in the structure.

18) Latent Cooling Load- The net amount of moisture added to the inside air by plants, people, cooking, infiltration, and any other moisture source. The amount of moisture in the air can be calculated from a combination of dry-bulb and wet-bulb temperature measurements. The latent loads will affect absolute (and relative) humidity.

19) Latent Heat Gain – is the energy added to the space when moisture is added to the space by means of vapor emitted by the occupants, generated by a process or through air infiltration from outside or adjacent areas.

20) Radiant Heat Gain – is the rate at which heat absorbed by the surfaces enclosing the space and the objects within the space is transferred by convection when the surface or objects temperature becomes warmer than the space temperature.

21) Sensible Cooling Load- The heat gain of the building due to conduction, solar radiation, infiltration, appliances, people, and pets. Burning a light bulb, for example, adds only sensible load to the house. This sensible load raises the dry-bulb temperature.

22) Space Heat gain: The rate at which heat enters to and/or is generated within a space during a time interval.

23) Space Heat loss: The rate at which energy is lost from the space during a time interval.

24) Sensible Heat Gain or Loss – is the heat directly added to or taken away the conditioned space by conduction, convection and/or radiation. The sensible loads will affect dry bulb air temperature.

25) Space Cooling Load – the rate at which energy must be removed from a space to maintain a constant space air temperature. Note that "space *heat gain* $\neq$ *space-cooling load."*

26) Space Heat Extraction Rate: The rate at which energy is removed from the space by the cooling and dehumidification equipment. Space heat extraction rate is usually the same as the space-cooling load if the space temperature remains constant.

27) Shading- The effectiveness of a fenestration product plus shade assembly in stopping heat gain from solar radiation is expressed as the Solar Heat Gain Coefficient (SHGC). SHGC values range from 0 to almost 1. The more effective at stopping heat gain, the lower the SHGC value.

28) Solar Heat Gain Coefficient (SHGC) - Solar heat gain coefficient (SHGC) is the ratio of the solar heat gain entering the space through the fenestration area to the incident solar radiation. Solar heat gain includes directly transmitted solar heat and absorbed solar radiation, which is then reradiated, conducted, or convected into the space. Solar Heat Gain Coefficient (SHGC) replaces the Shading Coefficient (SC) used in earlier versions of the standards as a measure of the solar heat gain due to windows and shading devices.

29) Temperature, Dry Bulb – is the temperature of a gas or mixture of gases indicated by an accurate thermometer after correction for radiation.

30) Temperature, Wet Bulb – is the temperature at which liquid or solid water, by evaporating into air, can bring the air to saturation adiabatically at the same temperature.

31) Temperature, Dewpoint – is the temperature at which the condensation of water vapor is a space begins for a given state of humidity and pressure as the temperature of the air is reduced.

32) Thermal conductivity – is the time rate of heat flow through a unit area and unit thickness of a homogenous material under steady conditions when a unit temperature gradient is maintained in the direction perpendicular to the area.

33) Thermal Transmittance or Coefficient of Heat Transfer (U-factor) – is the time rate of heat flow per unit area under steady conditions from the fluid on the warm side of a barrier to the fluid on the cold side, per unit temperature difference between the two fluids.

34) Thermal Conduction – is the process of heat transfer through a material medium in which kinetic energy is transmitted by particles of the material from particle to particle without gross displacement of the particles.

35) Thermal Convection – is the transfer of heat by movement of fluid. Forced convection is the transfer of heat from forced circulation of fluid as by a fan, jet or pump. Natural convection is the transfer of heat by circulation of gas or liquid due to differences in density resulting from temperature changes.

36) Thermally Light Buildings- A building whose heating and cooling requirements are proportional to the weather is considered a thermally light building. That is, when the outdoor temperature drops below the desired room temperature, heating is required and when the outdoor temperature goes above the desired room temperature, cooling is needed. In a thermally light building, the thermal performance of the envelope becomes a dominant factor in energy use and can usually be seen as seasonal fluctuations in utility consumption.

37) Thermally Heavy Buildings- When factors other than weather determine the heating and cooling requirements, the building can be considered thermally heavy. The difference between thermally light and thermally heavy buildings is the amount of heat generated by people, lighting, and equipment within the building. Thermally heavy buildings typically have high internal heat gains and, to a certain extent, are considered to be self-heating and more cooling dominated. This need to reject heat makes them less dependent on the thermal performance of the building envelope.

38) Thermal Weight- A simple "rule of thumb" for determining the thermal weight of a building is to look at heating and cooling needs at an outdoor temperature of 60°Ft. If the building requires heat at this temperature, it can be considered thermally light and if cooling is needed, it is thermally heavy.

39) Ton - A unit of measure for cooling capacity; One ton = 12,000 BTUs per hour

40) U-Factor- The U-factor is the “overall coefficient of thermal transmittance of a construction assembly, in Btu/ (hr ft^2 ºF), including air film resistances at both surfaces."

41) Zone- Occupied space or spaces within a building which has its heating or cooling controlled by a single thermostat or zone is a is a space or group of spaces within a building with heating and/or cooling requirements sufficiently similar so that comfort conditions can be maintained throughout by a single controlling device.

42) Zoning - A system in which living areas or groups of rooms are divided into separate spaces and each space's heating/air conditioning is controlled independently.

TUTORIAL QUESTIONS

1. Describe the various factors affecting survey of building?
2. Explain ventilation requirements of IAQ?
3. Write about the steps in cooling load calculations?
4. Explain about the u factor of wall,roof?
5. Explain about the ventilation systems standards?

UNIT V

STATIC PRESSURE CALCULATION

COURSE OBJECTIVE: Explain Hydronic System, Water Piping,Fittings used in the HVAC Piping System Function, CHW Pipe supports &Pump Head Calculation

COURSE OUTCOME: Perform preventive maintenance on heating and air conditioning systems.

SELECTION OF MOTOR FOR HP

Mechanical engineers tend to focus on individual equipment selections and forget to analyze how the motor and drive system will operate as part of the whole HVAC system. The common HVAC design approach is to break a system down into disjointed components, optimize the selection of each discrete piece of equipment to suit a particular design condition, and then assemble them all together and declare them a system. The advantage of this method is that it simplifies the design; the disadvantage is that it ignores the important interactions among different components.

On the other hand, a system approach allows the engineer to analyze both the demand and supply sides of the system and how they work together. The engineer evaluates how end-use requirements can be accomplished most effectively and efficiently. Focusing on systems means expanding possibilities, from looking for one piece of equipment that can meet worst-case needs to evaluating whether components can be configured to maintain high performance over the entire range of operating conditions.

An example of where this system approach would be beneficial is in the selection of the fans and pumps for the HVAC system. The engineer is responsible for analyzing the motor selection to ensure meeting not only the peak load but also its most efficient operating range. A typical example is selecting a motor that is most efficient at one-half to full load to operate at less than one-half or into the service factor.

The engineer is required to analyze the calculated loads as well as the energy model to determine how the system will perform throughout the year. This system analysis will inform the design and will help the engineer to select a motor and drive size that will maximize the energy efficiency of the system. Using a component approach will not identify how the motor will be loaded throughout the year and therefore would result in an inefficient motor and drive selection.

A VAV system is one of the most popular air distribution systems for commercial buildings. In this system, a motor-driven fan in an AHU supplies air to several VAV boxes and has an airflow requirement that the system ductwork was designed to handle. The fans are specified according to the requirements of this airflow condition. However, actual operating conditions can vary according to the season, the time of day, and the occupancy pattern for the building.

To handle the need for variable flow rates, the system is equipped with VFDs and dampers. Dampers are one of the least-efficient methods of controlling flow in an HVAC system. However, a damper used in combination with a VFD drive is the most efficient means of operating a VAV system. This is because the motor power use can be better managed by using the VFD to modulate the power consumption based on system load profile. In addition to increasing energy costs, an inefficient motor and drive system often increases maintenance costs. When systems do not operate efficiently, the stress on the system caused by energy losses must be dissipated by piping, structures, dampers, and valves.

Reviewing a submittal – What to look for

A significant amount of effort is focused on design, but that same amount of input is seldom spent on the review of submittals during construction. Typically, graduate engineers are burdened with the task of approving HVAC equipment that they did not select, and they often do not understand what to look for when reviewing the submittal. This construction administration phase should really be regarded the most important part of the equipment selection process. It is in this phase that engineers have the final say on whether the designed equipment will meet the system demand.

It has now become commonplace for manufacturers' representatives or sales people to reselect the designed equipment in a value engineering exercise orchestrated by owners (or simply to beat the competition). This pressure can result in AHU fans and pumps being selected to run at speeds of up 100 Hz in order to meet the design loads. At that point, any number of project variables make it difficult to sign off on the design with confidence that it will operate with the performance and efficiency intended.

Take, for example, a motor with a service factor of 1 and an FLA of 25, which is being operated at 25 amps and 60 Hz. Increasing the frequency to 65 hertz would over-amp the motor. Changing cycles (Hz) and not changing voltage proportionally will eventually burn up the motor. A VFD changes frequency (Hz) in direct proportion to volts, thus producing variable flows while protecting the motor; this is the principle behind VFD drives. Increasing the frequency output to the motor does not by itself cause the motor to draw more current. Current draw is a function of torque and slip. If the motor torque is enough to keep the slip from increasing, then the current draw will not increase. But increasing the frequency above the motor design point means you begin to lose torque. Therefore if the motor was selected with too low a service factor, it may overload the motor and cause it to fail.

In the past it was common practice for mechanical engineers to build in a 20% "fudge factor" in motor design selection. So if the torque required for a motor design is X ft/lb. at a speed of 1,725 RPM, that would equate to Y Horse Power (HP). Based on the closest available motor size from the manufacturer of the rated HP plus 5% (Y+5%) and the next increment of motor size of HP plus 20% (Y+20%); traditionally, the designer would choose the larger motor (Y+20%) which would equate to a maximum torque 20% more than the design load (X+20%). That practice has all but ceased in the spirit of energy efficiency. So if Y HP is required for the motors and the closest standard size is Y+5% HP, then that would be the motor that will be selected. That means if the motor is operated at more than 5% of the available torque, the motor can no longer perform as designed. The lower torque means more slip, higher slip means more current draw, more current draw means more heat (or over load), and more heat means shorter life.

Additionally, centrifugal loads such as pumps and fans are governed by the fan and pump affinity laws (Figure 1).

Fan Affinity Laws

Volume Capacity

The volume capacity of a centrifugal fan can be expressed as:

$q_1 / q_2 = (n_1 / n_2)(d_1 / d_2)^3$ ………………………….. (1)

Where:

q = volume flow capacity (m^3/s, gpm, cfm)

n = wheel velocity - revolution per minute - (rpm)

d = wheel diameter

Head or Pressure

The head or pressure of a centrifugal fan can be expressed like:

$dp_1 / dp_2 = (n_1 / n_2)^2 (d_1 / d_2)^2$ ……………………… (2)

Where:

dp = head or pressure (m, ft, Pa, psi, ..)

Power

The power consumption of a centrifugal fan can be expressed as:

$P_1 / P_2 = (n_1 / n_2)^3 (d_1 / d_2)^5$ ………………………..(3)

Where:

P = power (W, bhp)

Changing the Wheel Velocity

If the wheel diameter is constant - change in fan wheel velocity can simplify the affinity laws to:

Volume Capacity

$q_1 / q_2 = (n_1 / n_2)$ …………………………………..(1a)

Head or Pressure

$dp_1 / dp_2 = (n_1 / n_2)^2$ ………………………… (2a)

Power

$P_1 / P_2 = (n_1 / n_2)^3$ ……………………………(3a)

From the simplified equation for a centrifugal load, the following is true:

• Change in flow is proportional to the change in shaft speed

• Change in head (pressure) is proportional to the square of the change in shaft speed

• Change in power consumed is proportional to the cube of the change in shaft speed

For a VFD application, the alternate current (AC) motor is controlled by varying the speed; this will result in a change in the torque and HP for the motor. From Figure 1, the motor is rated at 100 HP at a frequency of 60 Hz. At 60 Hz, both the HP and torque are 100%. At speeds below 60 Hz the affinity law will govern the performance of the motor. However, as the motor speed increases above base motor speed, the horsepower will remain constant but the torque decreases. The torque decreases because the motor impedance increases with increasing frequency. Since the VFD cannot supply a voltage above the supplied voltage, the current decreases as the frequency increases, in turn decreasing the available torque.

The rate of decrease in torque is not proportional to the increase in frequency; theoretically, it is reduced by the ratio of the base speed to the higher speed (i.e., 60 Hz/ 105 Hz * 100 = 57%). However, in a design application the actual decrease in torque is usually further reduced by the increase in bearing friction, windage loss, and fan loading. The National Electrical Manufacturers Association

(NEMA) MG1, 12.44, states that alternating-current motors shall operate successfully under running conditions at rated load and at rated voltage with a variation in the frequency up to 10% above the rated frequency for two minutes. Performance within this frequency variation will not necessarily be in accordance with the standards established for operation at rated frequency. When running a motor higher than the rated frequency, the torque must be derated. Rotor balance, critical speed, and bearing life are huge concerns for HVAC motors when operating at frequencies in excess of the rated frequency.

From Table 2 (taken from the NEMA MG1 guidelines), we see that the over-speed of the motor will depend on the maximum operating speed. Take for example a fan for an AHU that is designed to operate at 1,800 RPM; from the table, the maximum over-speed that can be applied to the motor for two minutes is operating the VFD at 75 Hz (1.25 x 60 Hz). Likewise, the maximum over-speed for the motor of a fan for an AHU operating at 3,600 RPM is operating the VFD at 72 Hz (1.20 x 60).

A motor's speed capability is most often limited by the mechanical stress limits of the rotating structure. For continuous operation, motors operating above 90 Hz with constant voltage above 60 Hz may not have the required torque to sustain a constant horsepower load. Maximum safe operating speeds should be based on Table 2. The control's maximum speed should be set such that the motor is not unintentionally operated beyond the recommended speeds. If a continuous speed greater than the operating speed listed in NEMA MG 1 is required, the motor manufacturer should be consulted.

HYDRONIC SYSTEM

Hydronics is the use of water as the heat-transfer medium in heating and cooling systems. A hydronic piping system is used to circulate chilled or hot water with the connections between the piping and the terminal units made in a series loop. The terminal units are the heat exchangers that transfer the thermal energy between the water and the spaces to be cooled or heated.

Hydronic systems may be used for both a chilled and a heated water loop with chillers and cooling towers used separately or together as a means to provide water cooling, while boilers heat the water

Types of hydronic piping systems are:

Hydronic systems may be divided into several general piping arrangement categories:

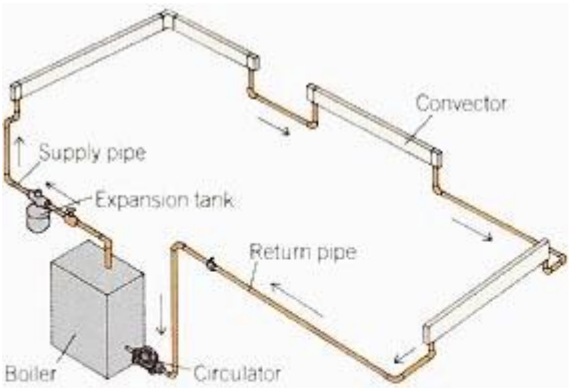
- Single or one-pipe.
- Two pipe steam (direct return or reverse return)
- Three pipe.
- Four pipe.
- Series loop.

1. **The Series Loop** – This system is aptly named because all of the units are in series, and one loop is formed. In this system the entire water supply flows through each terminal unit and then

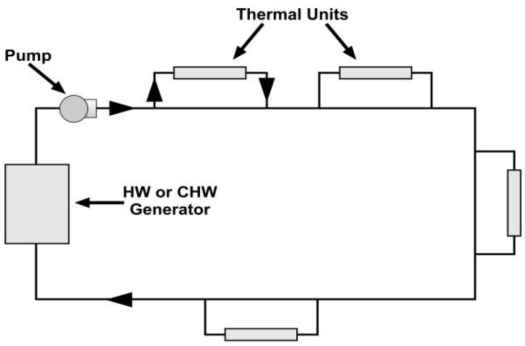
returns to the generator and pump. Although it is a simple arrangement, this setup has its disadvantages:

- To maintain or repair any terminal unit, it requires a shutdown of the entire system.
- The number of units is limited because in heating systems the water temperature continually decreases as it gives up heat in each unit in series. That can cause a low temperature in the far units in the system which may not provide adequate heat for comfort.

The series loop arrangement is basic, inexpensive and mostly used for residences**.**

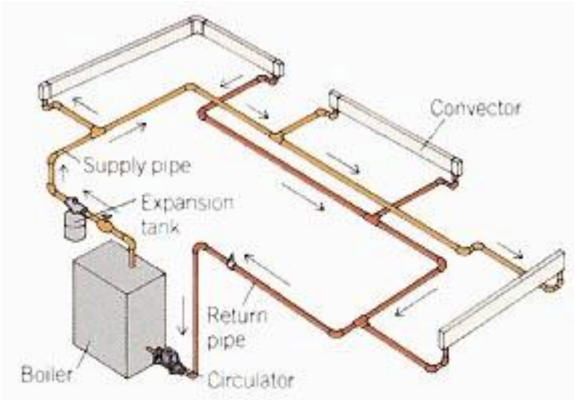


2. **One-Pipe Main –** With this system, each terminal unit is connected by a supply and a return branch pipe to the main. By locating valves in the branch lines, each unit can be separately controlled and serviced. In this system, like in the series loop, if there are too many units the heated water going to the far units may be not sufficient for room comfort.

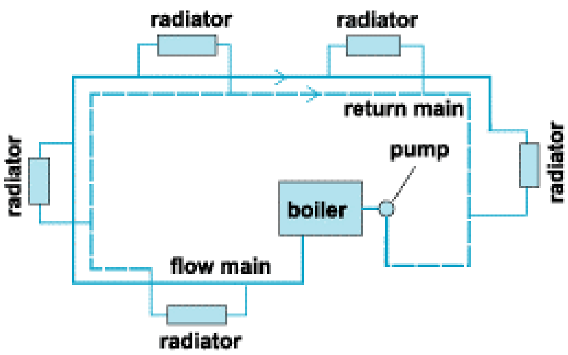


3. **Two-Pipe Direct Return –** This is generally used for larger systems and consists of two mains. One main is used for supply and one main is used for return. This system is more expensive than the one-pipe main and series loop, but it allows each terminal unit to be separately controlled and serviced because the supply water temperature to each unit is the

same. The two-pipe system is called direct return because the return main is routed to bring the water back to the source by the shortest path.



4. **Two-Pipe Reverse Return** – Here we have a supply and a return that are equal in length and size. The first terminal supplied is the last terminal returned and vise-versa, making it is easy to balance the flow rates.



Combination arrangements can also be made to create a three-pipe or four-pipe system. In the three-pipe arrangement, simultaneous heating or cooling can be made available. There are two-supply mains, one circulating chilled water, the other hot water. Three-way control valves in the branch to each terminal unit will determine whether the unit receives hot or chilled water and the return main receives the water from each unit. However, the three-pipe system can waste energy because the return main mixes chilled and hot water. In this mixing process, the chilled water is warmed and the hot water is cooled, which results in extra heating and cooling in the boiler and/or chiller. The four-pipe arrangement is expensive, but it separates two-pipe systems – one for chilled water and one for hot water. Therefore, no mixing occurs making it an ideal arrangement to avoid wasted energy.

Pipe Materials

Typically the piping used in an HVAC system is either schedule 40 black steel welded or cut-grooved pipe, or lighter gauge rolled-groove steel pipe for sizes 2-1/2-in. diameter and above. Type L copper or threaded schedule 40 black steel pipe is normally used for 2-in. diameter and smaller. In some closed-loop water source heat pump applications, schedule 40 PVC piping has been used where local codes and inspectors permit. If PVC is used on the exterior of a building it should be protected from the elements with insulation so that the piping does not deteriorate from extended exposure.

Use of PVC Piping

Schedule 40 PVC is sometimes used on the interior of the building where codes allow or, in some climates, on the exterior as a means to reduce cost.

Figure 10

Materials Used for Water Piping

Joints

Steel pipe is offered with weld, mechanical groove or threaded connections. Copper pipe is offered with solder or mechanical rolled-groove connections. PVC pipe is offered with solvent or mechanical grooved connections.

Weld-joint pipe has beveled ends so that when two pieces are butted together they form a groove for welding.

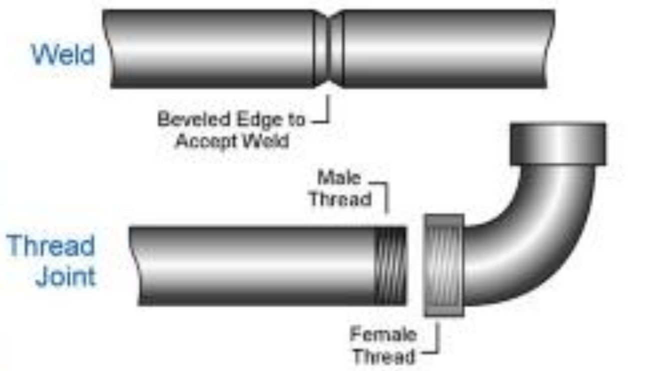


Figure 11

Weld and Threaded Joint

The threaded joint design has male tapered threads at the end of the pipe that screw into the female thread of a fitting.

Solder (sweat) joints and solvent joints are formed by one end that slips inside the other; solder or glue is used to seal the joint.

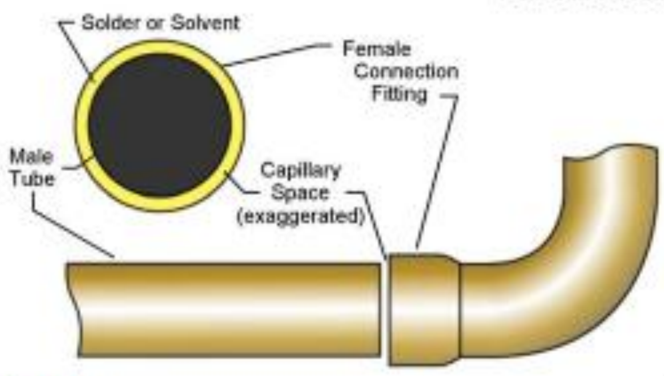


Figure 12

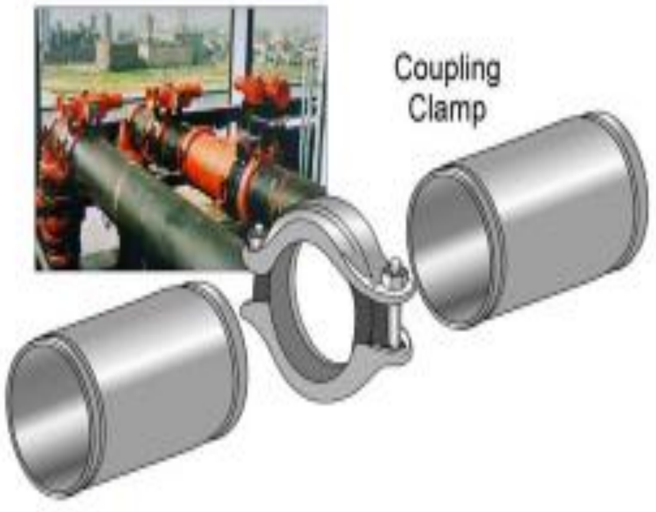


Figure 13

Mechanical (Groove) Joint
Actual photo courtesy of Victaulic Company

Mechanical groove joints have a groove that is cut or rolled into the end of the pipe and fitting. The joint is then completed with a mechanical coupling that locks into the grooves. Each coupling has a rubber gasket that seals the joint.

Grooved Joints

Large equipment like centrifugal chillers are often available with a choice of either weld stub-outs or groove stub-outs factory-installed for the chilled water and the condenser water connections.

Fittings

Numerous fittings are available such as 90 and 45-degree elbows, tees, concentric reducers, eccentric reducers, flanges, etc. Fittings that allow for the least pressure drop, best routing and proper drainage should be used. The friction loss that best represents the type of fittings for a specific project (standard radius elbow versus long radius elbow for instance) can be most easily found in the Fitting Equivalent Length Pressure Drop Charts in the Appendix when calculating total system pressure drop.

Equivalent lengths for unusual fittings not covered in the Tables will have to be determined by consulting with the manufacturer.

Figure 14

Valves

Many types of valves are available in the HVAC industry. Each type of valve has certain characteristics that make it better for certain applications such as shutoff, balancing, control (also referred to as "throttling"), or one-way flow. Some valves are suitable for multiple applications. A brief description of the different types of valves and their applications are listed below.

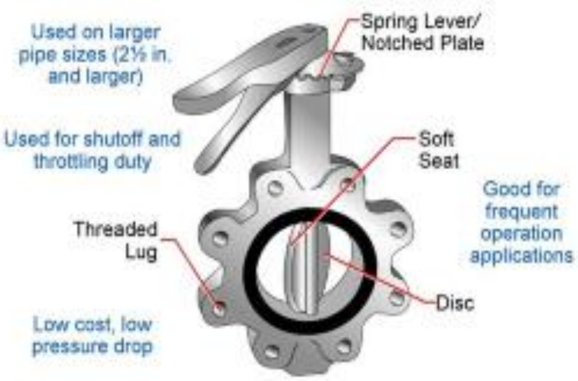


Figure 15

Butterfly Valves, Lug Pattern

Butterfly valves are generally found on larger sized systems and are used for shutoff duty, throttling duty and where there is frequent operation. They have good flow control (linear relationship between percent open and percent of full flow through the valve), low cost, high capacity and low pressure drop. They typically have bigger valves and are used on pipe sizes 2 ½-in. and larger. Lug-pattern will either through-bolt between two flanges, or be secured at the end of a pipe section, while a wafer-pattern is a more economical style that just sits between the bolted flanges without its own lugs.

Gate valves, also known as "stop valves," are designed for shutoff duty. When the valve is in the wide-open position, the gate is completely out of the fluid stream, thus providing straight through flow and a very low pressure drop. Gate valves should not be used for throttling. They are not designed for this type of service and consequently it is difficult to control fluid flow with any degree of accuracy. Vibration and chattering of the disc occurs when the valve is used for throttling, resulting in damage to the seating surface. The flow rate arrows in the figure indicate that a gate valve can be installed without regard to direction of flow within the pipe; they can seat in either direction. The globe valves shown next need to seat against the flow, which is why there is only one flow direction arrow on the figure.

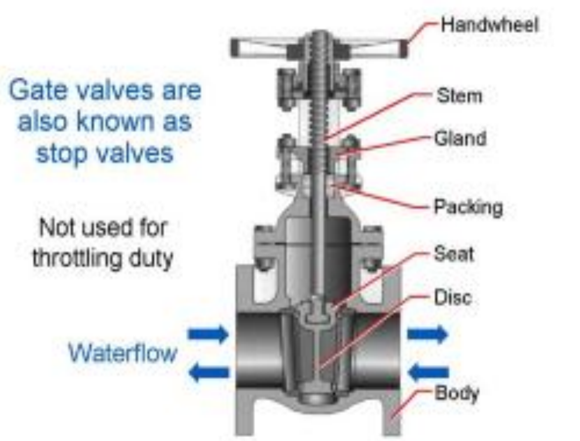


Figure 16

Gate Valve

Figure 20

Plug Valve

Plug valves, also called plug cocks, are primarily used for balancing flow rates in systems not subject to frequent flow changes. They come with cylindrical or tapered plugs that are usually lubricated to reduce galling, turning torque, and face leakage. Plug valves have approximately the same loss as a gate valve when in the fully open position. When partially closed for balancing, this line loss increases substantially. For large flow rate applications, a globe or butterfly will be used instead of a plug valve. Their sizes are limited to smaller applications because of cost.

Ball valves are used for full open/closed service, with limited requirement for precise control. They are best suited for quick-open linear control. Their advantage is low cost, high capacity, low leakage, and tight sealing.

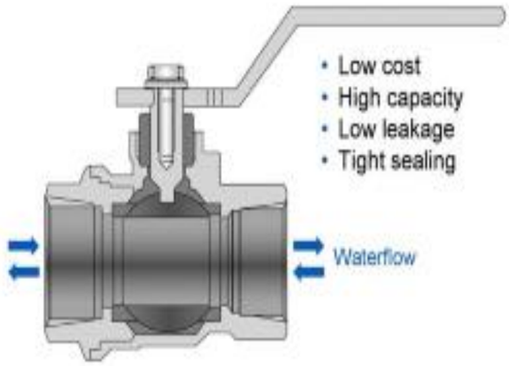


Figure 21

Ball Valve

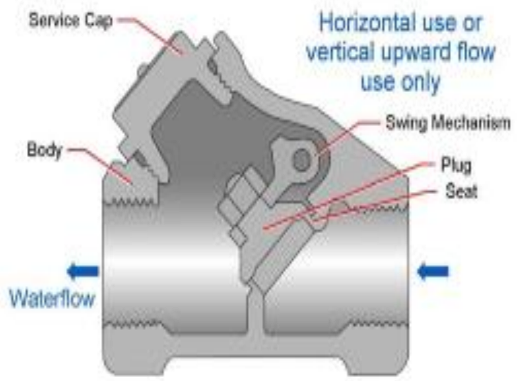


Figure 22

Swing Check Valve

Check valves prevent the flow of water in the reverse direction. There are two basic designs of check valves, the swing check and the lift check. The swing check valve may be used in a horizontal line or in a vertical line if flow is upward. The flow through the swing check is in a straight line and without restriction at the seat. Swing checks are generally used in combination with gate valves.

Globe, angle, and "Y" valves are of the same basic design and are designed primarily for throttling (balancing) duty. The angle or Y-pattern valve is recommended for full flow service since it has a substantially lower pressure drop at this condition than the globe valve. Another advantage of the angle valve is that it can be located to replace an elbow, thus eliminating one fitting.

Figure 17

Globe Valve

Figure 18

Angle Globe Valve

Globe, angle and Y valves can be opened or closed substantially faster than a gate valve because of the shorter lift of the disc. When valves are to be operated frequently, the globe design provides the more convenient operation. The seating surfaces of the globe, angle or Y valves are subject to less wear and the plug and seat are easy to replace compared to the gate valve discussed previously.

Figure 19

Y-Globe Valve

The lift check operates in a manner similar to that of a globe valve and, like the globe valve, its flow is restricted. The disc is seated by backflow or by gravity when there is no flow, and is free to rise and fall, depending on the pressure under it. The lift check should only be installed in horizontal piping and usually is used in combination with globe, angle and Y valves.

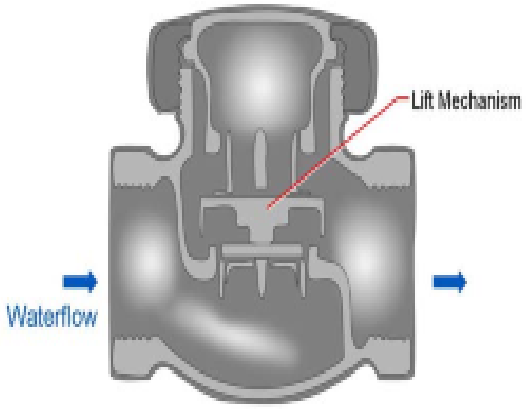


Figure 23

Lift Check Valve

Control Valves: 3-Way and 2-Way

Control valves can be 2-position (open or closed), 2-way modulating (modulates to vary flow through the coil and system), or 3-way modulating (modulates flow through the coil by bypassing water back to the return thereby maintaining a nearly constant flow through the system). Three-way valves are used for hot and cold water flow control on chillers, boilers, air coils, and most all HVAC hydronic units where temperature control is necessary.

Three-way mixing valves have two inlets and one outlet. Three-way diverting valves have one inlet and two outlets. Mixing valves are typically used to vary the flow through a load (such as a chilled or hot water coil). Diverting valves are used to direct the flow one way or another and are useful in applications like 2-pipe changeover or in bypass applications.

Three-way valves are used in many applications such as flow rate variation, temperature variation, and primary-secondary pumping systems in both 2-pipe and 4-pipe systems.

Two-way modulating valves are used for variable flow through heating and cooling coils. They throttle the flow for part-load control instead of bypassing the flow around the coil.

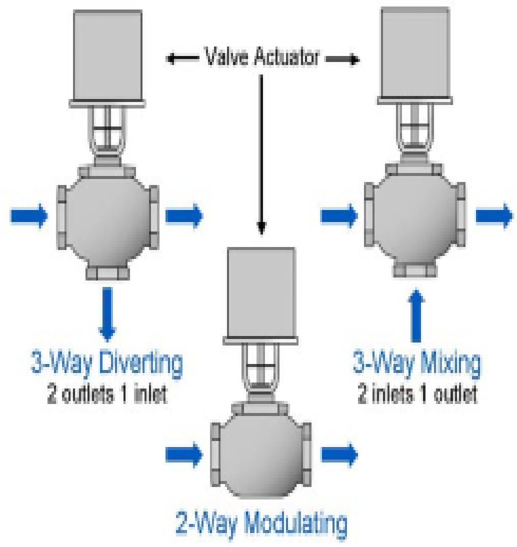


Figure 24

Control Valve Types

Pipe Sizing and Pump Selection Example

Steps 1-7 use a condenser water piping loop as the example system. Step 8 is for sizing the chilled-water piping loop. Once the piping system with all pipe routing, piping accessories, and equipment has been drawn and the flow rate for each piece of equipment has been determined, it becomes necessary to size the piping. Sizing of the piping will allow the total resistance (head) in the system to be determined so the pumps can be selected.

Step 1: Determine Water Velocity in Piping

Pipe size is limited by velocity based on noise and pipe erosion considerations. Both sound and erosion increase as the velocity increases. The table below gives recommended velocity limits, which are based on experience and are designed to give good balance between pipe size and system life.

Water Velocity

For our first example, sizing the condenser water piping on a single chiller system we will stay between 5 and 10 fps.

Recommended Water Velocities

Service	Velocity Range (fps)
Pump discharge	8 to 12
Pump suction	4 to 7
Drain line	4 to 7
Header	4 to 15
Mains and Riser	3 to 10
Branches and Runouts	5 to 10
City water	3 to 07

Figure 60

Recommended Water Velocities

The header pipe is close to the pump and carries fluid to the mains and risers. Mains (horizontal) and risers (vertical) distribute the fluid to the various areas of the building where branches and runouts feed the water flow to the air terminals, fan coils, baseboard, etc.

Step 2: Determining Piping Friction Losses

Friction Loss rate for pipe can be found by using Charts 1, 2 and 3, which are found in the Appendix. Charts 1 and 2 are normally used for larger size steel pipes, $\geq$2 ½-in., and Chart 3 is usually for smaller pipes, $\leq$2-in., where copper tubing is commonly used.

Chart 1 applies to new, smooth, clean, standard weight, steel pipe and can be used to determine the friction loss rate in a closed-loop piping system, such as a chilled water or hot water recirculating system.

Chart 2 applies to standard weight steel pipe that has been subject to scaling. This chart can be used to determine the friction loss rate in an open re-circulating piping system such as a condenser water system with cooling tower.

Chart 3 is used to determine the friction loss in copper tubing, which can be expected to stay clean throughout its normal life. Chilled or hot water systems that use copper piping would be sized with this chart.

Each chart gives the friction loss or head in feet of water per 100 ft of straight pipe.

Table 4 is the physical properties of steel pipe. This is helpful for inside areas, pipe and water weights.

Friction loss in valves and fittings can be determined by using equivalent length Tables 5, 6, 7 and 8.

The equivalent length tables were derived using the published manufacturer's Cv values. This was done to simplify and streamline the process for determining the friction loss in valves and fittings.

All friction losses are in equivalent length (feet) of straight pipe.

Cv

Cv is a flow coefficient used in the valve and controls industry. It is defined as the gallons of water (at 60° F) that will pass through any valve in one minute at a one-pound pressure drop.

Step 3: Gather Job Specific Component Pressure Drops and Design Data

Given:

100- ton cooling load

Entering chilled water temperature (54° F)

Leaving chilled water temperature (44° F)

Entering condenser water temperature (85° F)

Leaving condenser water temperature (95° F)

AHU-1, 45- ton load

AHU-2, 55- ton load

Equipment selections should be done to determine the pressure drop through each piece of equipment. This can be done by using the manufacturer's computerized selection program or published data.

Chiller	30 Series	Cooler PD	12.4 ft wg
		Condenser PD	11.0 ft wg
AHU-1	39 Series	Clg. Coil PD	9.9 ft wg
AHU-2	39 Series	Clg. Coil PD	13.4 ft wg
Tower	From Vendor	Unbalanced head	6.5 ft wg
	Required Nozzle Pressure		12.5 ft wg
Air Separator	From Vendor	From Vender PD	1.3 ft wg

Figure 61

Equipment Selection

Step 4: Review the Highest Pressure Drop Circuit and Calculate Water Flows

Cooling tower details usually do not show the exact exit point of the water from the distribution nozzles or the exact water height in the basin. We need to determine the unbalanced head. It doesn't have to be exact; in fact, if you just use the water inlet height minus the outlet height, you will be close enough. For our example, we have a nozzle pressure drop of 12.5 feet and a height of 6.5 feet for a total tower pressure drop of 19 ft wg.

You should contact the cooling tower manufacturer for the pressure drop of the cooling tower selected. Most cooling tower electronic selection programs will show this information. If this information is not available, an approximation of 15 ft wg will be close enough in most cases.

The condenser flow rate can normally be obtained from the actual chiller selection. For our example, we will use 3 gpm/ton, which is the ARI standard condenser flow rate.

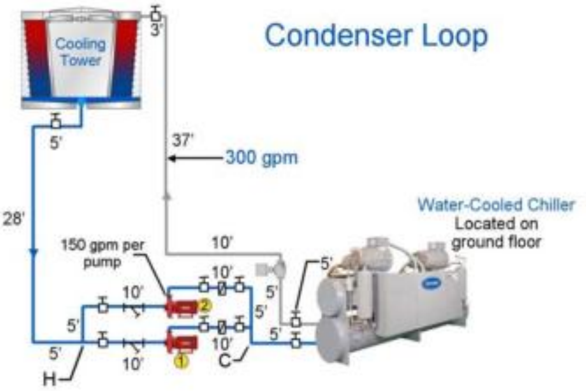


Figure 62

Sizing the Condenser Water Piping

The chilled water flow rate can be obtained from the actual chiller selection that was based on tons and water Δt or it can be calculated based on the following formula:

$$gpm = \frac{tons * 12{,}000\ Btuh / ton}{\Delta t * 60 \min / hr * 8.33 lb / gal * sp\ gr * sp\ ht}$$, or more simply,

gpm = tons * 24 ÷ Δt

100 * 24 ÷ 10 = 240 gpm

So we will use 300 gpm for the condenser flow rate and 240 gpm for the chilled water flow in the cooler (evaporator).

Calculation of gpm

Note: The above assumes "fresh water" in the cooler and condenser circuit. The water will be chemically treated as discussed earlier, but is still considered "fresh" water. If, for instance, 20 percent propylene glycol (PG) was used in the evaporator loop, the flow rate calculations would need to reflect the properties of the 20 percent PG mixture.

Step 5: Size the Pipe; Find the Friction Rate/100 ft

First, let's size the open-loop system (condenser water system) using Chart 2 to determine the piping friction losses.

Referring to Chart 2 in the Appendix, we find that 300 gpm intersects a 4-in. line size at 8.3 ft per second velocity and 10.0 ft of friction loss per 100 ft of pipe. Since we also have two pumps that handle 50 percent of the flow, we must determine the line size at the pumps. Referring to Chart 2, we find that 150 gpm intersects a 3-in. line size at 6.9 ft per second velocity and 9.3 ft of friction loss per 100 ft of pipe.

The 300 gpm sections of pipe will be 4-in. size, and the 150 gpm sections of pipe will be 3-in. size.

These sizes are within the recommended water velocity and pressure drop per 100 ft recommendations.

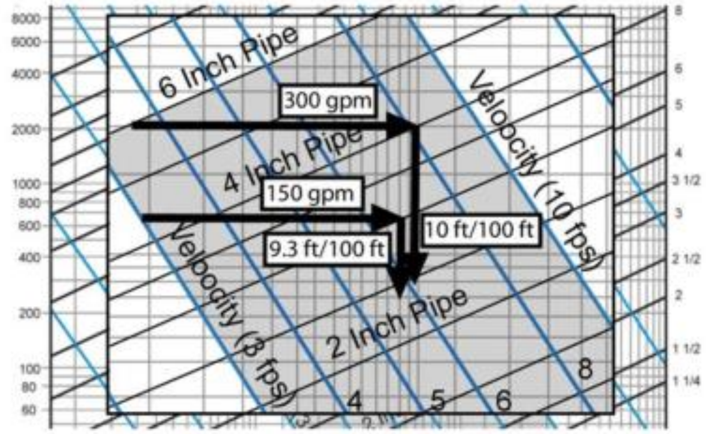


Figure 63

Sizing the Open-Loop Condenser Water Piping Using Chart 2

Step 6: Find the Longest Circuit Pressure Drop

Next let's add up the lengths of straight pipe in each size from the example. We will start at the pump suction and go in the direction of water flow around the loop.

(Note: When two pieces of equipment are piped in parallel, only the circuit with the highest pressure drop should be used in the head calculation.)

3-in. straight pipe (at pump #2) = 5 + 10 + 10 + 5 = 30 ft

4-in. straight pipe = 5 + 5 + 5 + 10 + 37 + 3 + 5 + 28 + 5 = 103 ft

This circuit is represented by C-H

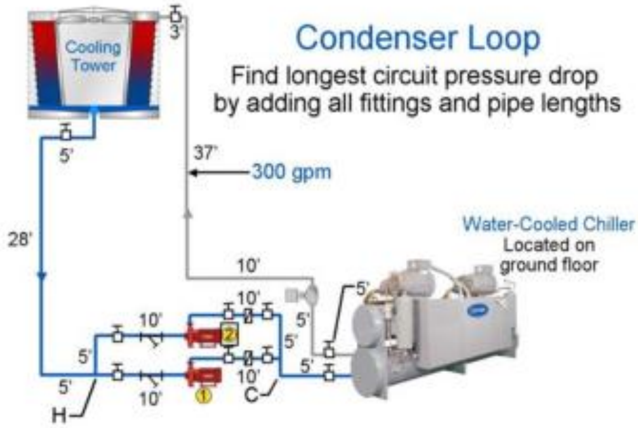


Figure 64

Example Condenser Water Loop

Next let's list the valves and the elbows and any other accessories in the loop in terms of their straight pipe equivalent lengths.

3-in. equivalent lengths (ft)

Pipe (from above)	=	30
Std. ells (qty of 2 at 7.67 ft each) (Table 6)	=	15
Butterfly Valves (qty of 3 @ 11.51 ft each) (Table 5)	=	35
Lift Check Valve (qty of 1 @ 14.06 ft) (Table 5)	=	14
Strainer (qty of 1 at 42 ft) (Table 8)	=	42
Total	=	136 ft

4-in. equivalent lengths (ft)

Pipe (from above)	=	103
Std. ells (qty of 7 at 10.07 ft each) (Table 6)	=	70
Tees (qty of 2 at 6.71 ft each) (Table 5)	=	14
Butterfly valves (qty of 4 at 15.1 ft each) (Table 5)	=	30
Control valve, butterfly (qty of 1 at 16 ft) (Table 8)	=	16
Total	=	233 ft

Step 7: Sum All the Pressure Drops for Pump Selection

Total Friction Loss = Equivalent ft * loss/100 ft

For 3-in. pipe, 136 equiv. ft * 9.3 ft /100

For 4-in. pipe, 233 equiv. ft * 10.0 ft /100

Total = 12.65 + 23.3 = 35.95 ft wg (round off to 36 ft wg pressure drop)

Enclosed is the actual pump curve for the condenser water pump based on 150 gpm each at 66 ft wg of head.

Head on Condenser Water Pump (ft)

Friction head	=	36.0
Unbalanced head	=	6.5
Pressure drop through condenser	=	11.0
Pressure drop through nozzles	=	12.5
Total head across pump	=	66.0

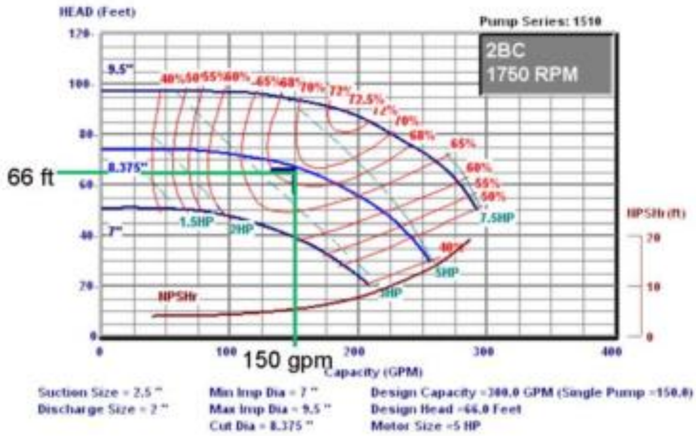


Figure 66

Example Condenser Water Pump Selection – Parallel Pumps, Single Pump Performance
Screen capture courtesy of Bell & Gossett

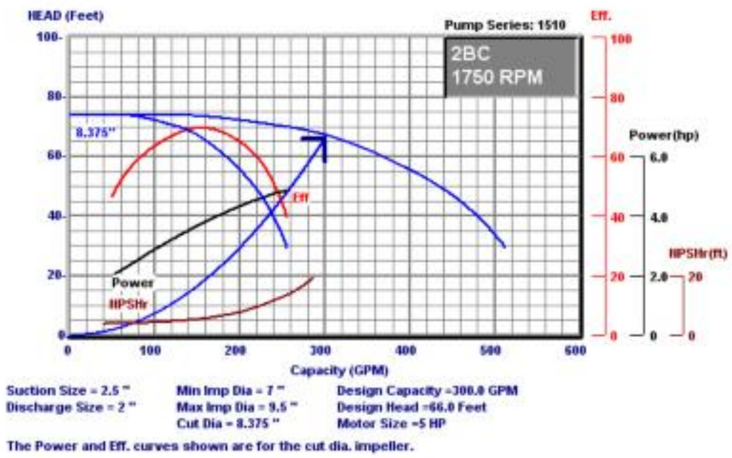


Figure 65

Example Condenser Water Pump Selection – Parallel Pumps, Performance of Both Pumps
Screen capture courtesy of Bell & Gossett

Step 8: Size the Chilled Water Loop

The closed-loop system (chilled water system) should be sized using Chart 1 to determine the piping friction losses.

Referring to Chart 1, we find that 240 gpm intersects a 4-in. line size at 6.3 fps velocity and 3.4 ft of friction loss per 100 ft of pipe. Since we also have two pumps that handle 50 percent of the flow, we must determine the line size at the pumps. Referring to Chart 1, we find that 120 gpm intersects a 3-in. line size at 5.5 fps velocity and 3.5-ft of friction loss per 100 ft of pipe.

Pipe Sizes

Note: 4-in. piping was used in lieu of 3.5-in. because 3.5- in. pipe is not readily available. Three-in. piping was used in lieu of 2.5-in.because it was border line on velocity.

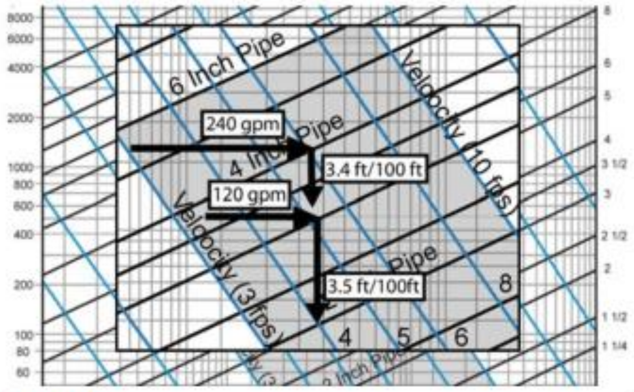


Figure 67

Sizing the Chilled Water Piping Using Chart 1

The two air-handling units are piped in parallel so we must use the one that has the highest pressure drop when the piping and AHU pressure drops are summed. Since AHU-2 has the longest length of piping and the coil pressure drop is higher we will use the piping to AHU-2 in our calculation. Referring to Chart 1, (55 tons $*$ 24 $\div$ 10 = 132 gpm) we find that 132 gpm intersects the 3-in. line size at 6.0 fps velocity and 4.0-ft of friction per 100 ft of pipe (not shown in the text). Notice the water flow in this circuit is clockwise with the pump pushing water through the chillers. The longest path of water flow starts at point C, goes through the chiller, up to AHU-2, back down through the air separator, to point N and then through pump number 3.

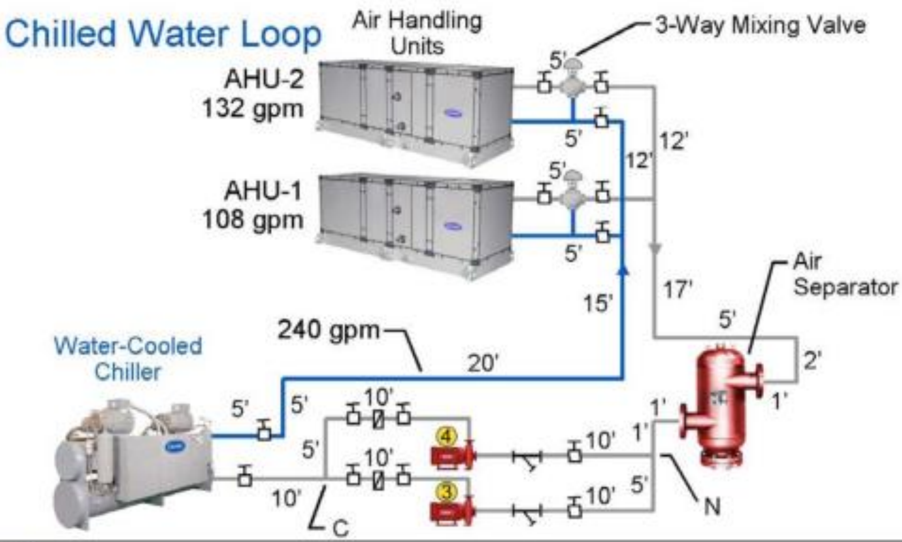


Figure 68

Find longest pressure drop circuit and calculate water flow.

Straight pipe from the example:

3-in. pipe @ pump #3	=	10 + 10 + 5	=	25 ft
3-in. pipe @ AHU #2	=	12 + 5 + 5 + 12	=	34 ft
4-in. pipe (C-N)	=	10 + 5 + 5 + 20 + 15 + 17 + 5 + 2 + 1 + 1 + 1	=	82 ft

3-in. equivalent lengths @ pump (ft)

pipe (from above)	=	25
Std. Ells (qty. of 1 @ 7.67 ft) (Table 6)	=	8
Butterfly Valves (qty. of 3 @ 11.51 ft each) (Table 5)	=	35
Lift Check Valve (qty. of 1 @ 14.06 ft) (Table 5)	=	14
Strainer (qty. of 1 @ 42 ft) (Table 8)	=	42
Total	=	124 ft

3-in. equivalent lengths @ AHU #2 (ft)

pipe (from above)	=	34
Std. Ells (qty. of 2 @ 7.67 ft each) (Table 6)	=	15
Tees (qty. of 1 @ 5.11 ft) (Table 6)	=	5
Butterfly Valves (qty. of 3 @ 11.51 ft each) (Table 5)	=	35
Control Valve, butterfly (qty. of 1 @ 11 ft) (Table 8)	=	11
Total	=	100 ft

4-in. equivalent lengths (ft)

pipe (from above)	=	82
Ells (qty. of 7 @ 10.07 ft each) (Table 6)	=	70
Tees (qty. of 4 @ 6.71 ft each) (Table 6)	=	27
Butterfly Valves (qty. of 2 @ 15.1 ft each) (Table 5)	=	30
Total	=	209 ft

Total Friction Loss = Equivalent ft * loss/100 ft of pipe

For (3-in. pipe @ pump, 124 equiv ft * 3.5 /100) + (3-in. pipe @ AHU #2, 100 equiv. ft * 4 /100) + (4-in. pipe, 209 equiv. ft * 3.4 /100)

Total = 4.22 + 4.0 + 7.32 = 15.54 ft wg

Head

Head (hd) is an energy unit that is usually expressed in feet of the liquid being pumped. In a closed system, friction is the only loss or head that the pump has to overcome. The height of water on the suction side of the pump is always exactly equal to the height of the discharge side piping. In open systems this is not true, there is always a difference in head between the suction side and discharge side of the pump. In a cooling tower for instance, the height between the water level in the basin and the exit from the distribution nozzles at the top of the tower represents the unbalanced head that must be overcome by the pump. If the distribution system consists of spray nozzles that require a specific pressure to force water through the nozzles, this pressure must be added to the static head also. The total head of the pump will consist of the following: pipe friction loss, valves including control valves, accessories, equipment such as coolers, condensers coils, air separators, etc., and any unbalanced head.

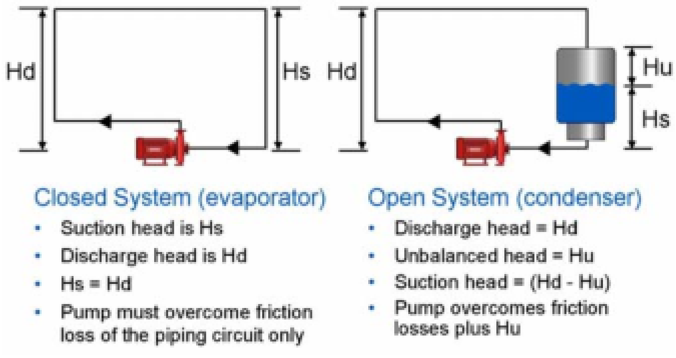


Figure 48

Pumping Head Examples

If you have a high-rise building, the static head can impose significant pressure on the system components. Example – A 50-story building with 12-ft between floors would be 600 ft. high. Since 2.31 ft. equals 1 psi, the pressure on the components at the lowest level would be 600 ft./2.31 ft/psi = 260 psi. The system components at the lowest elevation must be designed for this pressure. In this example, the chiller "waterboxes" would have to be constructed to accommodate 260 psi. If the cooling tower was on the roof, the condenser waterboxes would have to be 300 psi rated as well. If the cooling tower was ground-mounted, the condenser waterbox could be standard construction.

Discharge Head

Discharge head is the head at the pump discharge, made up of the static head at the pump outlet, any positive static pressure in the discharge side of the system, discharge pipe friction loss, and any equipment pressure drop. A pump discharge pressure gauge would indicate total discharge head.

Suction Head

Suction head is the head indicated on a pressure gauge at the pump suction. In a closed-loop system it would be the remaining discharge pressure after subtracting all the piping friction, and the valve and equipment losses. In an open-loop system, suction head includes static head (or lift), entrance loss and friction head in the suction piping, and any positive pressure existing on the suction side. With a closed-loop system in operation, a pressure gauge at the pump suction would indicate a positive suction head. On an open-loop system the gauge would read negative if the pump was above the fluid source being pumped.

Liquid Horsepower

Liquid horsepower is obtained by the formula, gpm $*$ hd $*$ sp gr $\div$ 3960 (for standard water sp gr =1.0), where 3960 converts the equation units into horsepower (33,000 ft $*$ lb per minute, divided by 8.33 lb per gallon).

Brake Horsepower

Brake horsepower (bhp) is the power required to drive the pump and equals the liquid horsepower divided by the overall efficiency of the pump.

Net Positive Suction Head

Net positive suction head (NPSH) is equal to the pressure drop in ft wg of the liquid from the suction flange to the point inside the impeller where pressure starts to rise. NPSH available at the pump suction for the actual application must always be greater than the NPSH required for the pump used. Failure to do so would allow fluid vaporization within the pump (cavitation). Cavitation can cause impeller failure, shaft failure and/or seal failures. To check the available NPSH at the pump suction flange in a given system, use the following formula:

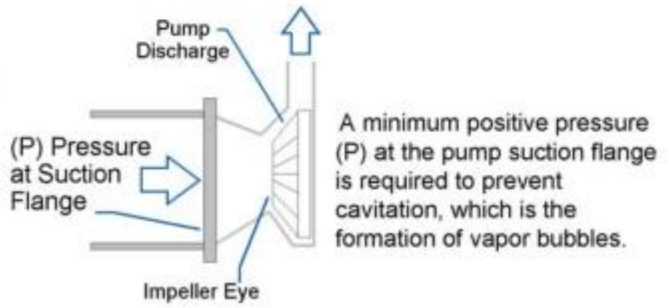


Figure 49

Net Positive Suction Head at Pump Inlet

$$NPSH = 2.31\left(P_a - P_{vp}\right) + \left(H_s - H_f\right)$$

NPSH

Net positive suction head does not tend to be an issue in closed-loop piping systems. In open-loops, if the pump is elevated above the cooling tower basin, it is prudent to check the NPSH requirements of the pump.

If using a fluid other than water use the following formula:

$$NPSH = \frac{2.31\left(P_a - P_{vp}\right)}{sp\ gr} + \left(H_s - H_f\right)$$

Where:

$NPSH$	=	net positive suction head
2.31	=	conversion factor to change 1 psi to pressure head in ft of water
P_a	=	atmospheric pressure (absolute pressure, psia)
P_{vp}	=	vapor pressure corresponding to water temperature at the pump suction. For water returning from a cooling tower at 85° F it is 0.59648, for 86° F water it is 0.61585.
H_s	=	elevation head, static head (ft) above or below the pump suction. (If above, positive static head; if below, negative static head, sometimes termed suction lift.)
H_f	=	friction head (ft), loss in suction line. Must include entrance loss and pressure drop through valves and fittings.
$sp\ gr$	=	specific gravity of the fluid being pumped

Figure 50 shows a cross-section of a typical pump. This figure is of a centrifugal inline pump, which gets its name from the straight inlet and discharge water flow. Other centrifugal pump designs will be discussed later in the TDP Module. The figure shows six components of a typical pump: the motor, coupler, bearings, pump shaft, mechanical seal, and the impeller.

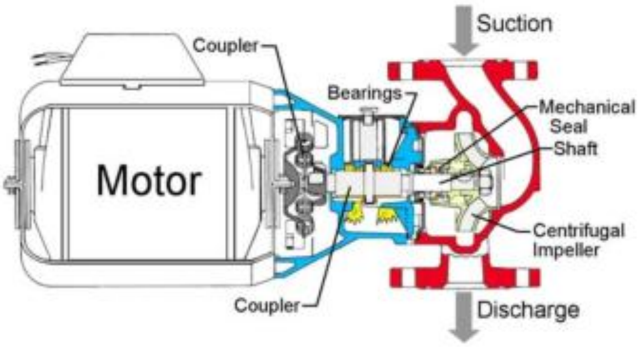


Figure 50

Typical Pump Cross-Section

The motor is typically an open-drip proof type provided by the pump manufacturer and selected specifically for the head and flows required.

A coupler mechanism is provided to attach the pump shaft to the motor assembly.

Specifically designed bearings are utilized to provide constant circulation of oil over all bearing surfaces.

The pump shaft serves the purpose of transmitting the motor torque to the impeller.

A mechanical seal is required to prevent water from entering the motor and bearing compartment.

The impeller moves the water through the pump assembly. It is selected specifically for the flow and head required for the application.

Pump Curve

The optimal pump curve for an air-conditioning application is shown here. The very steep curve is not desirable for HVAC duty because it can lead to surging at low flow rates. The very flat curve can be an issue because large flow rate changes occur with small changes in the head.

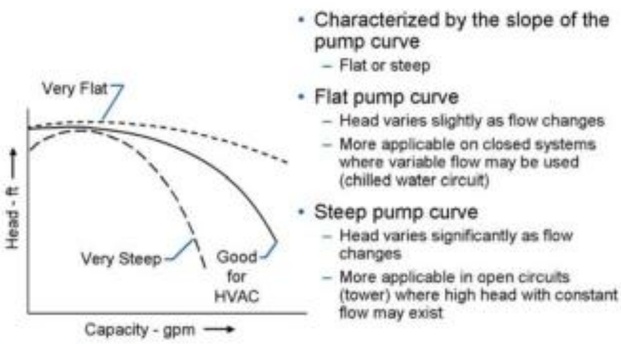


Figure 51

Pump Curve Examples

Important items to understand about pumps are:

1. Varying the speed – proportionally raises or lowers the head and capacity. The whole head curve shifts up or down.
2. Varying the impeller diameter – proportionally raises or lowers the head and capacity. The whole head curve shifts up or down. Increasing the impeller size raises the head and capacity.
3. Varying the impeller diameter – proportionally varies the capacity.

4. Varying the pitch and number of vanes within the impeller changes the shape of the head capacity curve.
5. Varying the impeller and vane designs produce variations in head-capacity relationships. Narrow impellers with larger impeller-to-eye diameter ratios develop a larger head. Wide impellers with low diameter ratios are used for low heads and large flows.

Changes in speed and impeller diameters are reflected in pump performance as follows:

$$\frac{rpm_1}{rpm_2} or \frac{impeller\ dia._1}{impeller\ dia._2} = \frac{gpm_1}{gpm_2} = \left(\frac{head_1}{head_2}\right)^2 = \left(\frac{bhp_1}{bhp_2}\right)^3$$

A given pump operates along its own head-capacity curve. It is a centrifugal device just like a fan. At full capacity flow, the operating point falls at the crossing of the pump head-capacity curve and the system head curve (Point 1). If the pressure drop increases the system curve and the operating point moves up the head-capacity curve (Point 2, reduced water flow). If a greater flow is desired, the pressure drop must be reduced and the operating point would move down the curve (Point 3) or the pump could be speeded up, or the impeller size increased which would move the head capacity curve upward (Point 4). These performance characteristics are just like a fan in a duct system.

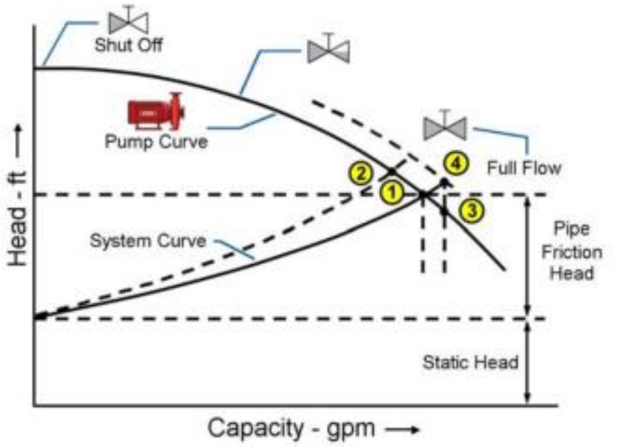


Figure 52

Pump and System Curve Intersection

If the system head is overestimated and the pump is selected with a high head-capacity curve, unfortunate results may occur. The pump will operate on its head-capacity curve to produce an increased flow at decreased head and increased horsepower demand (Point 1). The system head should always be calculated without undue safety factors or as close as practical to the true values to eliminate possible waste of horsepower or possible overloading of pump motor with an unvalved system. If not sized properly, the balancing valve on the pump discharge may have to be throttled or the impeller size decreased to achieve the desired flow. This is especially true when evaluating system head on a system designed with parallel or series pumps.

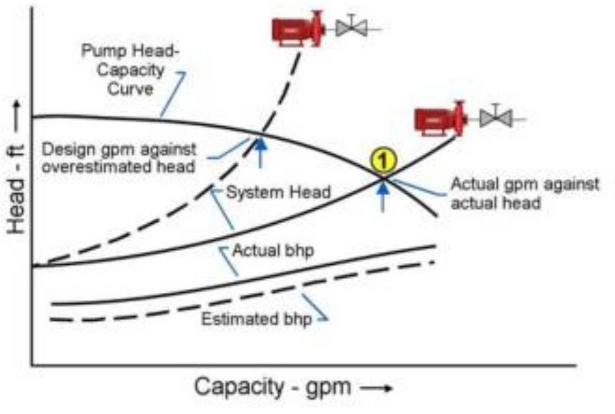


Figure 53

Overestimating Pump Head

Variable Speed Pumping

Variable speed pumping is very common. A pump with a VFD operates much like a fan with a VFD. A differential pressure sensor located near the end-of-run in the piping system sends a signal to the VFD to slow down the pump rpm if pressure in the piping is rising, or increase the rpm if the pressure is falling from set point. The pump moves its rpm along the system curve resulting in variations in flow rate.

Energy savings are excellent at reduced flow rates as the bhp follows the cube ratio of the rpm.

$$bhp_2 = bhp_1 * \left(\frac{rpm_2}{rpm_1}\right)^3$$

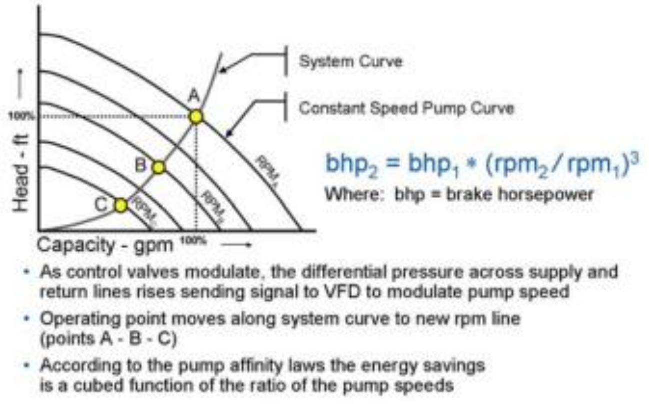


Figure 56

Variable Speed Pumping Characteristics

Selection

Pumps should be selected based on design, size, service, and performance. In terms of performance, a pump should be selected to provide the required flow rate at the design head while trying to achieve the lowest possible horsepower. Pump catalogs, with pump performance curves, allow the proper pump to be selected. Most pump manufacturers also have software programs that can select the optimal pump for your application. The following is an example of a pump selection from a manufacturer's program.

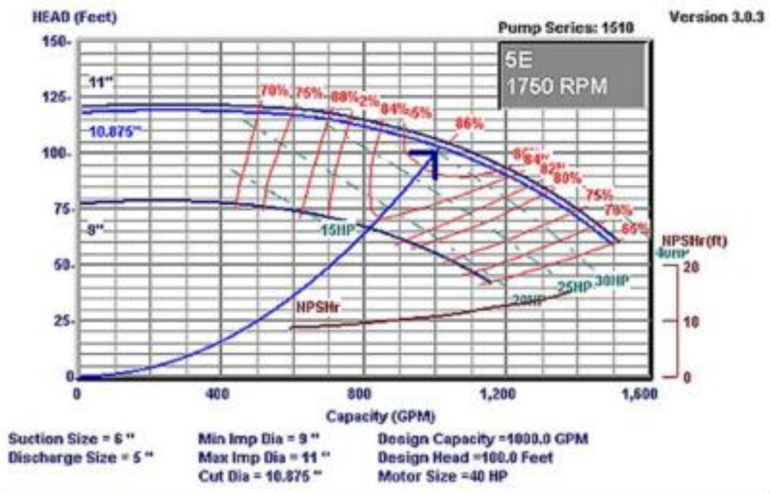


Figure 57

Typical Centrifugal Pump Selection Screen Capture
Courtesy of Bell & Gossett

Fans and blowers provide air for ventilation and industrial process requirements. Fans generate a pressure to move air (or gases) against a resistance caused by ducts, dampers, or other components in a fan system. The fan rotor receives energy from a rotating shaft and transmits it to the air.

Difference between Fans, Blowers and Compressors

Fans, blowers and compressors are differentiated by the method used to move the air, and by the system pressure they must operate against. As per American Society of Mechanical Engineers (ASME) the specific ratio - the ratio of the discharge pressure over the suction pressure – is used for defining the fans, blowers and compressors (see Table 5.1).

TABLE 5.1 DIFFERENCES BETWEEN FANS, BLOWER AND COMPRESSOR

Equipment	Specific Ratio	Pressure rise (mmWg)
Fans	Up to 1.11	1136
Blowers	1.11 to 1.20	1136 – 2066
Compressors	more than 1.20	–

5.2 Fan Types

Fan and blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Fan efficiencies differ from design to design and also by types. Typical ranges of fan efficiencies are given in Table 5.2.

Fans fall into two general categories: centrifugal flow and axial flow.

In centrifugal flow, airflow changes direction twice - once when entering and second when leaving (forward curved, backward curved or inclined, radial) (see Figure 5.1).

In axial flow, air enters and leaves the fan with no change in direction (propeller, tubeaxial, vaneaxial) (see Figure 5.2).

TABLE 5.2 FAN EFFICIENCIES

Type of fan	Peak Efficiency Range
Centrifugal Fans	
Airfoil, backward curved/inclined	79–83
Modified radial	72–79
Radial	69–75
Pressure blower	58–68
Forward curved	60–65
Axial fan	
Vane axial	78–85
Tube axial	67–72
Propeller	45–50

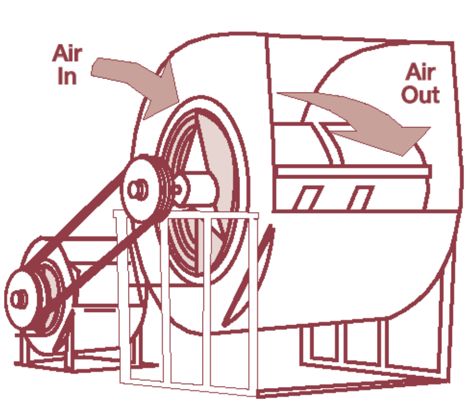


Figure 5.1 Centrifugal Fan

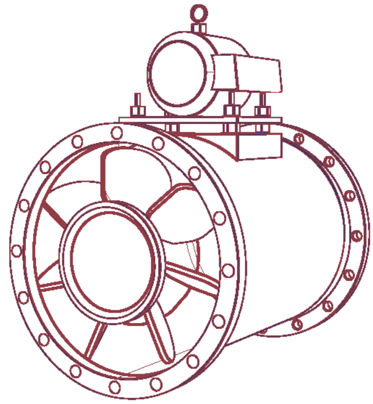


Figure 5.2 Axial Fan

Centrifugal Fan: Types

The major types of centrifugal fan are: *radial, forward curved and backward curved* (see Figure 5.3).

Radial fans are industrial workhorses because of their high static pressures (upto 1400 mm WC) and ability to handle heavily contaminated airstreams. Because of their simple design, radial fans are well suited for high temperatures and medium blade tip speeds.

Forward-curved fans are used in clean environments and operate at lower temperatures. They are well suited for low tip speed and high-airflow work - they are best suited for moving large volumes of air against relatively low pressures.

Backward-inclined fans are more efficient than forward-curved fans. Backward-inclined fans reach their peak power consumption and then power demand drops off well within their useable airflow range. Backward-inclined fans are known as "non-overloading" because changes in static pressure do not overload the motor.

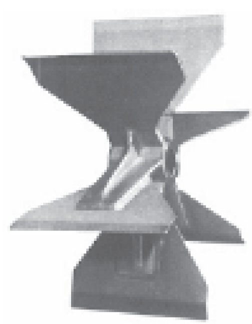
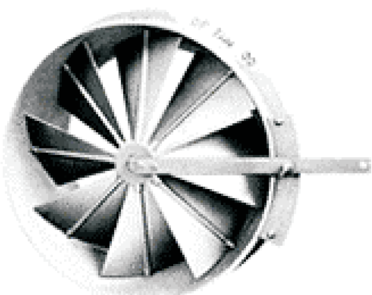
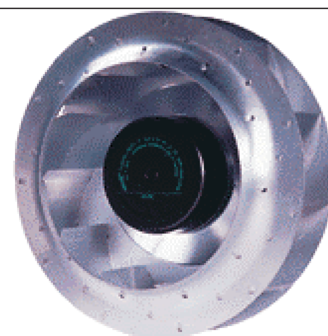
Paddle Blade (Radial blade)	Forward Curved (Multi-Vane)	Backward Curved
		

Figure 5.3 Types of Centrifugal Fans

Axial Flow Fan: Types

The major types of axial flow fans are: *tube axial, vane axial and propeller (see Figure 5.4.)*

Tubeaxial fans have a wheel inside a cylindrical housing, with close clearance between blade and housing to improve airflow efficiency. The wheel turn faster than propeller fans, enabling operation under high-pressures 250 – 400 mm WC. The efficiency is up to 65%.

Vaneaxial fans are similar to tubeaxials, but with addition of guide vanes that improve efficiency by directing and straightening the flow. As a result, they have a higher static pressure with less dependence on the duct static pressure. Such fans are used generally for pressures upto 500 mmWC. Vaneaxials are typically the most energy-efficient fans available and should be used whenever possible.

Propeller fans usually run at low speeds and moderate temperatures. They experience a large change in airflow with small changes in static pressure. They handle large volumes of air at low pressure or free delivery. Propeller fans are often used indoors as exhaust fans. Outdoor applications include air-cooled condensers and cooling towers. Efficiency is low – approximately 50% or less.

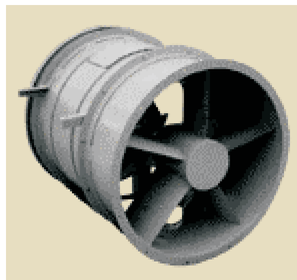
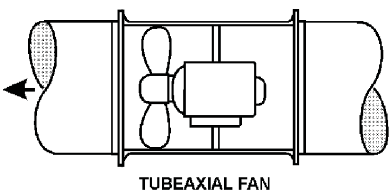
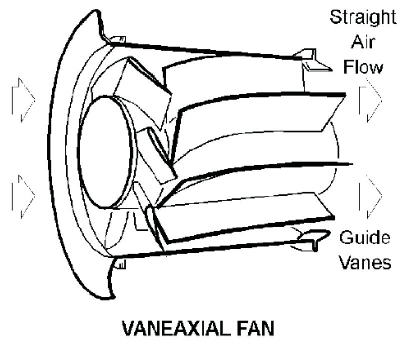
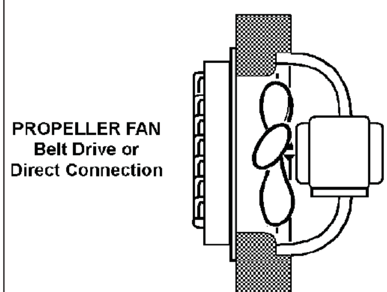
Tube Axial	Vane Axial	Propeller
		
		

Figure 5.4 Types of Axial Fans

The different types of fans, their characteristics and typical applications are given in Table 5.3.

Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 1.20 kg/cm^2. They are also used to produce negative pressures for industrial vacuum systems. Major types are: centrifugal blower and positive-displacement blower.

Centrifugal blowers look more like centrifugal pumps than fans. The impeller is typically gear-driven and rotates as fast as 15,000 rpm. In multi-stage blowers, air is accelerated as it passes through each impeller. In single-stage blower, air does not take many turns, and hence it is more efficient.

Centrifugal blowers typically operate against pressures of 0.35 to 0.70 kg/cm^2, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure

TABLE 5.3 TYPES OF FANS, CHARACTERISTICS, AND TYPICAL APPLICATIONS

Centrifugal Fans			Axial-flow Fans		
Type	**Characteristics**	**Typical Applications**	**Type**	**Characteristics**	**Typical Applications**
Radial	High pressure, medium flow, efficiency close to tube-axial fans, power increases continuously	Various industrial applications, suitable for dust laden, moist air/gases	Propeller	Low pressure, high flow, low efficiency, peak efficiency close to point of free air delivery (zero static pressure)	Air-circulation, ventilation, exhaust
Forward-curved blades	Medium pressure, high flow, dip in pressure curve, efficiency higher than radial fans, power rises continuously	Low pressure HVAC, packaged units, suitable for clean and dust laden air / gases	Tube-axial	Medium pressure, high flow, higher efficiency than propeller type, dip in pressure-flow curve before peak pressure point.	HVAC, drying ovens, exhaust systems
Backward curved blades	High pressure, high flow, high efficiency, power reduces as flow increases beyond point of highest efficiency	HVAC, various industrial applications forced draft fans, etc.	Vane-axial	High pressure, medium flow, dip in pressure-flow curve, use of guide vanes improves efficiencyexhausts	High pressure applications including HVAC systems,
Airfoil type	Same as backward curved type, highest efficiency	Same as backward curved, but for clean air applications			

increases, which can be a disadvantage in material conveying systems that depend on a steady air volume. Because of this, they are most often used in applications that are not prone to clogging.

Positive-displacement blowers have rotors, which "trap" air and push it through housing. Positive-displacement blowers provide a constant volume of air even if the system pressure varies. They are especially suitable for applications prone to clogging, since they can produce enough pressure - typically up to 1.25 kg/cm^2 - to blow clogged materials free. They turn much slower than centrifugal blowers (e.g. 3,600 rpm), and are often belt driven to facilitate speed changes.

5.3 Fan Performance Evaluation and Efficient System Operation

System Characteristics

The term "system resistance" is used when referring to the static pressure. The system resistance is the sum of static pressure losses in the system. The system resistance is a function of the configuration of ducts, pickups, elbows and the pressure drops across equipment-for example back-filter or cyclone. *The system resistance varies with the square of the volume of air flowing through the system*. For a given volume of air, the fan in a system with narrow ducts and multiple short radius elbows is going to have to work harder to overcome a greater system resistance than it would in a system with larger ducts and a minimum number of long radius turns. Long narrow ducts with many bends and twists will require more energy to pull the air through them. Consequently, for a given fan speed, the fan will be able to pull less air through this system than through a short system with no elbows. Thus, the system resistance increases substantially as the volume of air flowing through the system increases; square of air flow.

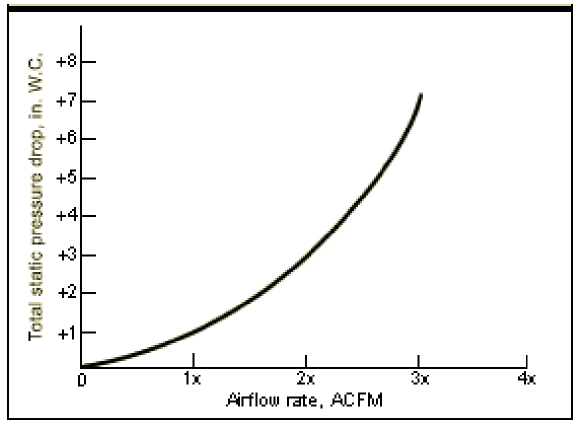


Figure 5.5 System Characteristics

Conversely, resistance decreases as flow decreases. To determine what volume the fan will produce, it is therefore necessary to know the system resistance characteristics.

In existing systems, the system resistance can be measured. In systems that have been designed, but not built, the system resistance must be calculated. Typically a system resistance curve (see Figure 5.5) is generated with for various flow rates on the x-axis and the associated resistance on the y-axis.

Fan Characteristics

Fan characteristics can be represented in form of fan curve(s). The fan curve is a performance curve for the particular fan under a specific set of conditions. The fan curve is a graphical representation of a number of inter-related parameters. Typically a curve will be developed for a given set of conditions usually including: fan volume, system static pressure, fan speed, and brake horsepower required to drive the fan under the stated conditions. Some fan curves will also include an efficiency curve so that a system designer will know where on that curve the fan will be operating under the chosen conditions (see Figure 5.6). In the many curves shown in the Figure, the curve static pressure (SP) vs. flow is especially important.

The intersection of the system curve and the static pressure curve defines the operating point. When the system resistance changes, the operating point also changes. Once the operating point is fixed, the power required could be found by following a vertical line that passes through the operating point to an intersection with the power (BHP) curve. A horizontal line drawn through the intersection with the power curve will lead to the required power on the right vertical axis. In the depicted curves, the fan efficiency curve is also presented.

System Characteristics and Fan Curves

In any fan system, the resistance to air flow (pressure) increases when the flow of air is increased. As mentioned before, it varies as the square of the flow. The pressure required by a system over a range of flows can be determined and a "system performance curve" can be developed (shown as SC) (see Figure 5.7).

This system curve can then be plotted on the fan curve to show the fan's actual operating point at "A" where the two curves (N_1 and SC_1) intersect. This operating point is at air flow Q_1 delivered against pressure P_1.

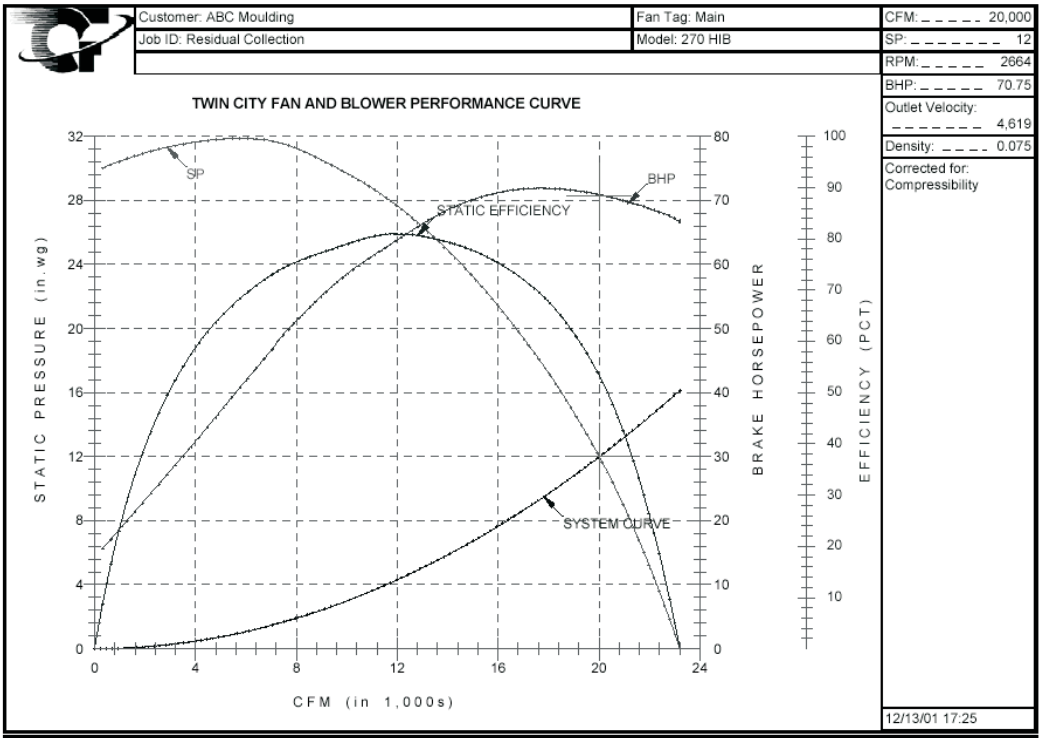


Figure 5.6 Fan Characteristics Curve by Manufacturer

A fan operates along a performance given by the manufacturer for a particular fan speed. (The fan performance chart shows performance curves for a series of fan speeds.) At fan speed N_1, the fan will operate along the N_1 performance curve as shown in Figure 5.7. The fan's actu-

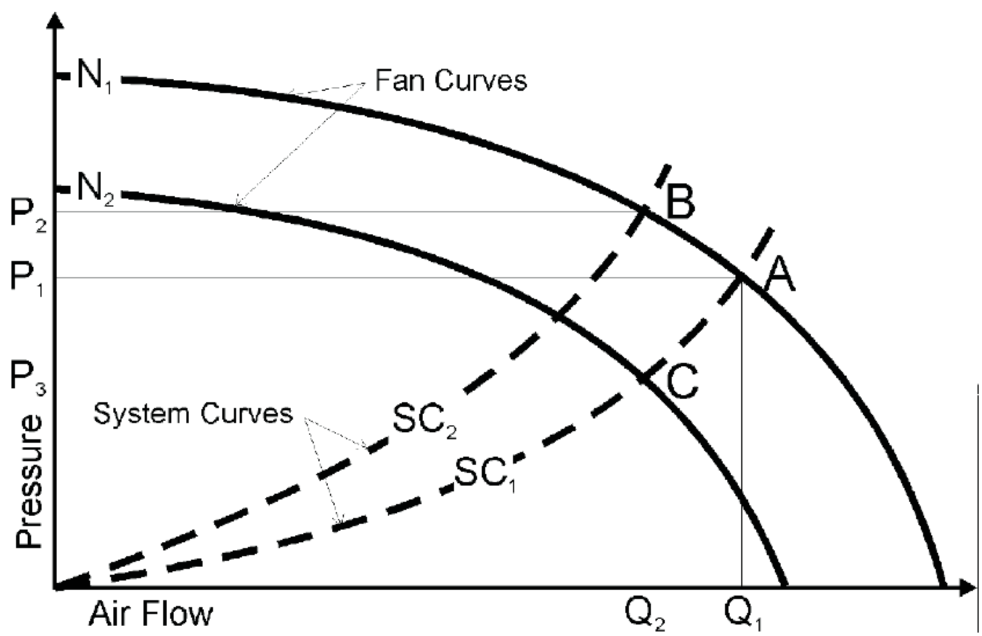


Figure 5.7 System Curve

al operating point on this curve will depend on the system resistance; fan's operating point at "A" is flow (Q_1) against pressure (P_1).

Two methods can be used to reduce air flow from Q_1 to Q_2:

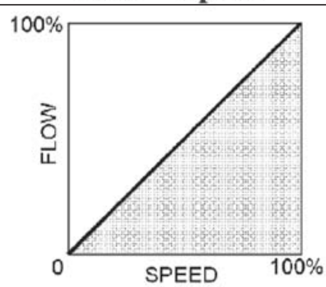
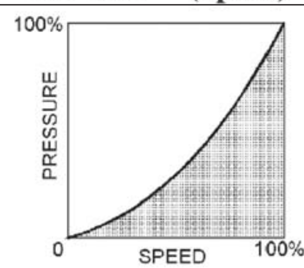
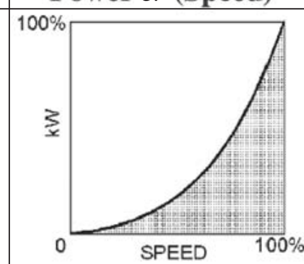
First method is to restrict the air flow by partially closing a damper in the system. This action causes a new system performance curve (SC_2) where the required pressure is greater for any given air flow. The fan will now operate at "B" to provide the reduced air flow Q_2 against higher pressure P_2.

Second method to reduce air flow is by reducing the speed from N_1 to N_2, keeping the damper fully open. The fan would operate at "C" to provide the same Q_2 air flow, but at a lower pressure P_3.

Thus, reducing the fan speed is a much more efficient method to decrease airflow since less power is required and less energy is consumed.

Fan Laws

The fans operate under a predictable set of laws concerning speed, power and pressure. A change in speed (RPM) of any fan will predictably change the pressure rise and power necessary to operate it at the new RPM.

Flow α Speed	**Pressure α (Speed)²**	**Power α (Speed)³**
		
$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$	$\frac{SP_1}{SP_2} = \left(\frac{N_1}{N_2}\right)^2$	$\frac{kW_1}{kW_2} = \left(\frac{N_1}{N_2}\right)^3$
Varying the RPM by 10% decreases or increases air delivery by 10%.	***Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%***	***Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%***

Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)

Fan Design and Selection Criteria

Precise determination of air-flow and required outlet pressure are most important in proper selection of fan type and size. The air-flow required depends on the process requirements; normally determined from heat transfer rates, or combustion air or flue gas quantity to be handled. System pressure requirement is usually more difficult to compute or predict. Detailed analysis should be carried out to determine pressure drop across the length, bends, contractions and expansions in the ducting system, pressure drop across filters, drop in branch lines, etc. These pressure drops should be added to any fixed pressure required by the process (in the case of ventilation fans there is no fixed pressure requirement). Frequently, a very conservative approach is adopted allocating large safety margins, resulting in over-sized fans which operate at flow rates much below their design values and, consequently, at very poor efficiency.

Once the system flow and pressure requirements are determined, the fan and impeller type are then selected. For best results, values should be obtained from the manufacturer for specific fans and impellers.

The choice of fan type for a given application depends on the magnitudes of required flow and static pressure. For a given fan type, the selection of the appropriate impeller depends additionally on rotational speed. Speed of operation varies with the application. High speed small units are generally more economical because of their higher hydraulic efficiency and relatively low cost. However, at low pressure ratios, large, low-speed units are preferable.

Fan Performance and Efficiency

Typical static pressures and power requirements for different types of fans are given in the Figure 5.8.

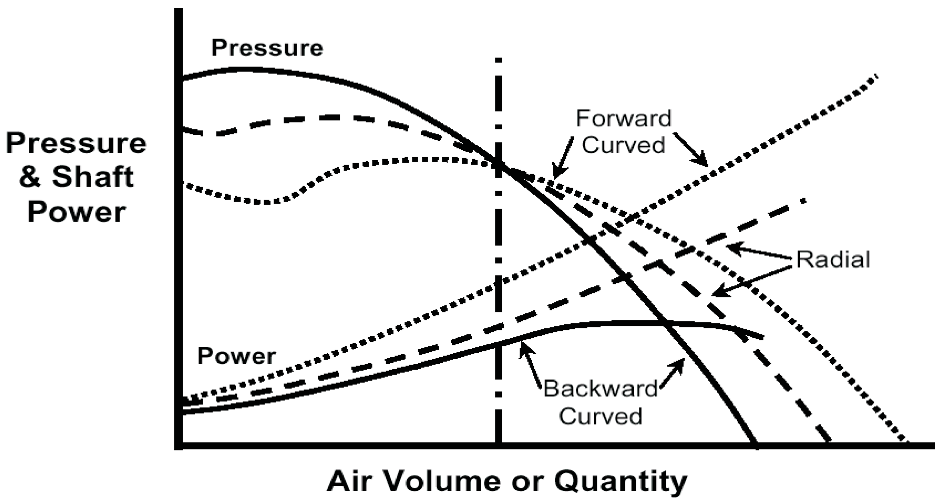


Figure 5.8 Fan Static Pressure and Power Requirements for Different Fans

Fan performance characteristics and efficiency differ based on fan and impeller type (See Figure 5.9).

In the case of centrifugal fans, the hub-to-tip ratios (ratio of inner-to-outer impeller diameter) the tip angles (angle at which forward or backward curved blades are curved at the blade tip - at the base the blades are always oriented in the direction of flow), and the blade width determine the pressure developed by the fan.

Forward curved fans have large hub-to-tip ratios compared to backward curved fans and produce lower pressure.

Radial fans can be made with different heel-to-tip ratios to produce different pressures.

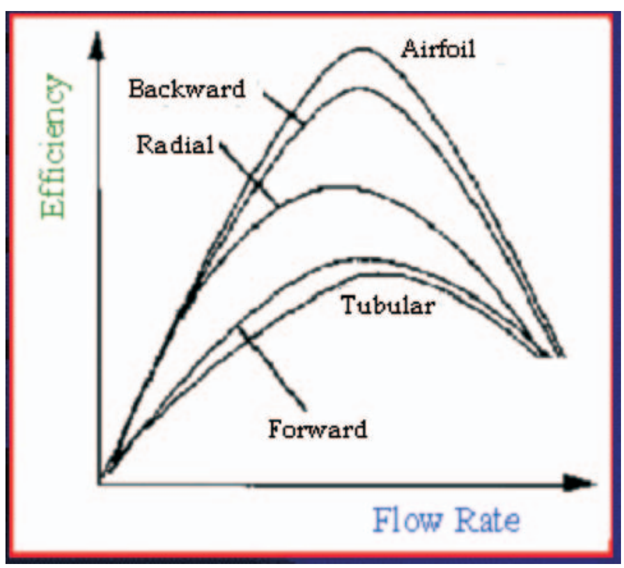


Figure 5.9 Fan Performance Characteristics Based on Fans/ Impellers

At both design and off-design points, backward-curved fans provide the most stable operation. Also, the power required by most backward –curved fans will decrease at flow higher than design values. A similar effect can be obtained by using inlet guide vanes instead of replacing the impeller with different tip angles. Radial fans are simple in construction and are preferable for high-pressure applications.

Forward curved fans, however, are less efficient than backward curved fans and power rises continuously with flow. Thus, they are generally more expensive to operate despite their lower first cost.

Among centrifugal fan designs, aerofoil designs provide the highest efficiency (upto 10% higher than backward curved blades), but their use is limited to clean, dust-free air.

Axial-flow fans produce lower pressure than centrifugal fans, and exhibit a dip in pressure before reaching the peak pressure point. Axial-flow fans equipped with adjustable / variable pitch blades are also available to meet varying flow requirements.

Propeller-type fans are capable of high-flow rates at low pressures. Tube-axial fans have medium pressure, high flow capability and are not equipped with guide vanes.

Vane-axial fans are equipped with inlet or outlet guide vanes, and are characterized by high pressure, medium flow-rate capabilities.

Performance is also dependant on the fan enclosure and duct design. Spiral housing designs with inducers, diffusers are more efficient as compared to square housings. Density of inlet air is another important consideration, since it affects both volume flow-rate and capacity of the fan to develop pressure. Inlet and outlet conditions (whirl and turbulence created by grills, dampers, etc.) can significantly alter fan performance curves from that provided by the manufacturer (which are developed under controlled conditions). Bends and elbows in the inlet or outlet ducting can change the velocity of air, thereby changing fan characteristics (the pressure drop in these elements is attributed to the system resistance). All these factors, termed as System Effect Factors, should, therefore, be carefully evaluated during fan selection since they would modify the fan performance curve.

Centrifugal fans are suitable for low to moderate flow at high pressures, while axial-flow fans are suitable for low to high flows at low pressures. Centrifugal fans are generally more expensive than axial fans. Fan prices vary widely based on the impeller type and the mounting (direct-or-belt-coupled, wall-or-duct-mounted). Among centrifugal fans, aerofoil and backward-curved blade designs tend to be somewhat more expensive than forward-curved blade designs and will typically provide more favourable economics on a lifecycle basis. Reliable cost comparisons are difficult since costs vary with a number of application-specific factors. A careful technical and economic evaluation of available options is important in identifying the fan that will minimize lifecycle costs in any specific application.

Safety margin

The choice of safety margin also affects the efficient operation of the fan. In all cases where the fan requirement is linked to the process/other equipment, the safety margin is to be decided, based on the discussions with the process equipment supplier. In general, the safety margin can be 5% over the maximum requirement on flow rate.

In the case of boilers, the induced draft (ID) fan can be designed with a safety margin of 20% on volume and 30% on head. The forced draft (FD) fans and primary air (PA) fans do not require any safety margins. However, safety margins of 10 % on volume and 20% on pressure are maintained for FD and PA fans.

Some pointers on fan specification

The right specification of the parameters of the fan at the initial stage, is pre-requisite for choosing the appropriate and energy efficient fan.

The user should specify following information to fan manufacturer to enable right selection:

Design operating point of the fan – volume and pressure

Normal operating point – volume and pressure

Maximum continuous rating

Low load operation - This is particularly essential for units, which in the initial few years may operate at lower capacities, with plans for upgradation at a later stage. The initial low load and the later higher load operational requirements need to be specified clearly, so that, the manufacturer can supply a fan which can meet both the requirements, with different sizes of impeller.

Ambient temperature – The ambient temperatures, both the minimum and maximum, are to be specified to the supplier. This affects the choice of the material of construction of the impeller.

The maximum temperature of the gas at the fan during upset conditions should be specified to the supplier. This will enable choice of the right material of the required creep strength.

Density of gas at different temperatures at fan outlet

Composition of the gas – This is very important for choosing the material of construction of the fan.

Dust concentration and nature of dust – The dust concentration and the nature of dust (e.g. bagasse – soft dust, coal – hard dust) should be clearly specified.

The proposed control mechanisms that are going to be used for controlling the fan.

The operating frequency varies from plant-to-plant, depending on the source of power supply. Since this has a direct effect on the speed of the fan, the frequency prevailing or being maintained in the plant also needs to be specified to the supplier.

Altitude of the plant

The choice of speed of the fan can be best left to fan manufacturer. This will enable him to design the fan of the highest possible efficiency. However, if the plant has some preferred speeds on account of any operational need, the same can be communicated to the fan supplier.

Installation of Fan

The installation of fan and mechanical maintenance of the fan also plays a critical role in the efficiency of the fan. The following clearances (typical values) should be maintained for the efficient operation of the impeller.

Impeller Inlet Seal Clearances

- Axial overlap –5 to 10 mm for 1 metre plus dia impeller
- Radial clearance –1 to 2 mm for 1 metre plus dia impeller
- Back plate clearance –20 to 30 mm for 1 metre plus dia impeller
  Labyrinth seal clearance –0.5 to 1.5 mm

The inlet damper positioning is also to be checked regularly so that the "full open" and "full close" conditions are satisfied. The fan user should get all the details of the mechanical clearances from the supplier at the time of installation. As these should be strictly adhered to, for efficient operation of the fan, and a checklist should be prepared on these clearances. A check on these clearances should be done after every maintenance, so that efficient operation of the fan is ensured on a continuous basis.

System Resistance Change

The system resistance has a major role in determining the performance and efficiency of a fan. The system resistance also changes depending on the process. For example, the formation of the coatings / erosion of the lining in the ducts, changes the system resistance marginally. In some cases, the change of equipment (e.g. Replacement of Multi-cyclones with ESP / Installation of low pressure drop cyclones in cement industry) duct modifications, drastically shift the operating point, resulting in lower efficiency. In such cases, to maintain the efficiency as before, the fan has to be changed.

Hence, the system resistance has to be periodically checked, more so when modifications are introduced and action taken accordingly, for efficient operation of the fan.

5.5 Flow Control Strategies

Typically, once a fan system is designed and installed, the fan operates at a constant speed. There may be occasions when a speed change is desirable, i.e., when adding a new run of duct that requires an increase in air flow (volume) through the fan. There are also instances when the fan is oversized and flow reductions are required.

Various ways to achieve change in flow are: pulley change, damper control, inlet guide vane control, variable speed drive and series and parallel operation of fans.

Pulley Change

When a fan volume change is required on a permanent basis, and the existing fan can handle the change in capacity, the volume change can be achieved with a speed change. The simplest way to change the speed is with a pulley change. For this, the fan must be driven by a motor through a v-belt system. The fan speed can be increased or decreased with a change in the drive pulley or the driven pulley or in some cases, both pulleys. As shown in the Figure 5.10, a higher sized fan operating with damper control was downsized by reducing the motor (drive) pulley size from 8" to 6". The power reduction was 15 kW.

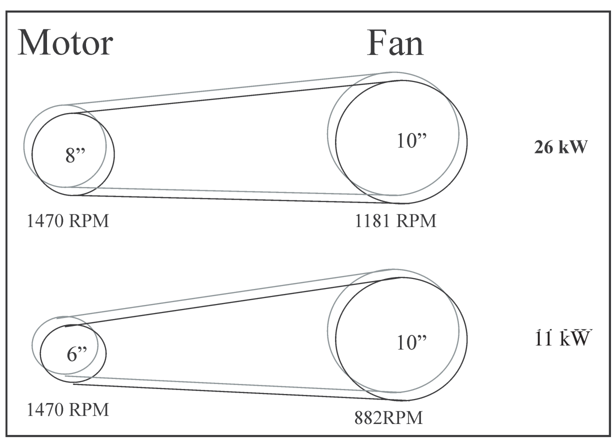


Figure 5.10 Pulley Change

Damper Controls

Figure 5.11 Damper change

Some fans are designed with damper controls (see Figure 5.11). Dampers can be located at inlet or outlet. Dampers provide a means of changing air volume by adding or removing system resistance. This resistance forces the fan to move up or down along its characteristic curve, generating more or less air without changing fan speed. However, dampers provide a limited amount of adjustment, and they are not particularly energy efficient.

Inlet Guide Vanes

Figure 5.12 Inlet Guide Vanes

Inlet guide vanes are another mechanism that can be used to meet variable air demand (see Figure 5.12). Guide vanes are curved sections that lay against the inlet of the fan when they are open. When they are closed, they extend out into the air stream. As they are closed, guide vanes pre-swirl the air entering the fan housing. This changes the angle at which the air is presented to the fan blades, which, in turn, changes the characteristics of the fan curve. Guide vanes are energy efficient for modest flow reductions – from 100 percent flow to about 80 percent. Below 80 percent flow, energy efficiency drops sharply.

Axial-flow fans can be equipped with variable pitch blades, which can be hydraulically or pneumatically controlled to change blade pitch, while the fan is at stationary. Variable-pitch blades modify the fan characteristics substantially and thereby provide dramatically higher energy efficiency than the other options discussed thus far.

Variable Speed Drives

Although, variable speed drives are expensive, they provide almost infinite variability in speed control. Variable speed operation involves reducing the speed of the fan to meet reduced flow requirements. Fan performance can be predicted at different speeds using the fan laws. Since power input to the fan changes as the cube of the flow, this will usually be the most efficient form of capacity control. However, variable speed control may not be economical for systems, which have infrequent flow variations. When considering variable speed drive, the efficiency of the control system (fluid coupling, eddy-current, VFD, etc.) should be accounted for, in the analysis of power consumption.

Series and Parallel Operation

Parallel operation of fans is another useful form of capacity control. Fans in parallel can be additionally equipped with dampers, variable inlet vanes, variable-pitch blades, or speed controls to provide a high degree of flexibility and reliability.

Combining fans in series or parallel can achieve the desired airflow without greatly increasing the system package size or fan diameter. Parallel operation is defined as having two or more fans blowing together side by side.

The performance of two fans in parallel will result in doubling the volume flow, but only at free delivery. As Figure 5.13 shows, when a system curve is overlaid on the parallel performance curves, the higher the system resistance, the less increase in flow results with parallel fan operation. Thus, this type of application should only be used when the fans can operate in a low resistance almost in a free delivery condition.

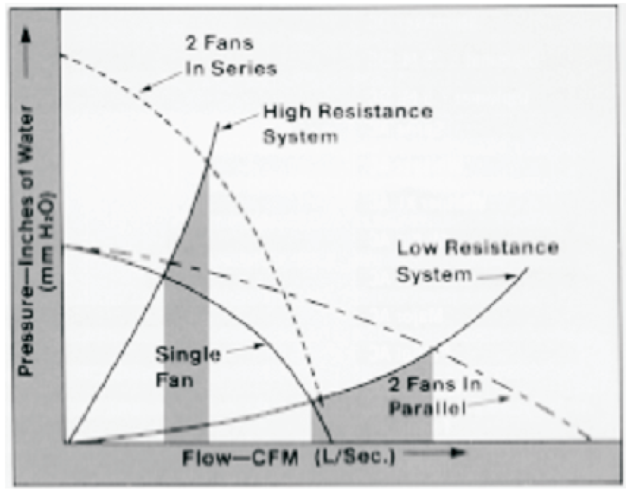


Figure 5.13 Series and Parallel Operation

Series operation can be defined as using multiple fans in a push-pull arrangement. By staging two fans in series, the static pressure capability at a given airflow can be increased, but again, not to double at every flow point, as the above Figure displays. In series operation, the best results are achieved in systems with high resistances.

In both series and parallel operation, particularly with multiple fans certain areas of the combined performance curve will be unstable and should be avoided. This instability is unpredictable and is a function of the fan and motor construction and the operating point.

Factors to be considered in the selection of flow control methods

Comparison of various volume control methods with respect to power consumption (%) required power is shown in Figure 5.14.

All methods of capacity control mentioned above have turn-down ratios (ratio of maximum–to–minimum flow rate) determined by the amount of leakage (slip) through the control elements. For example, even with dampers fully closed, the flow may not be zero due to leakage through the damper. In the case of variable-speed drives the turn-down ratio is limited by the control system. In many cases, the minimum possible flow will be determined by the characteristics of the fan itself. Stable operation of a fan requires that it operate in a region where the system curve has a positive slope and the fan curve has a negative slope.

The range of operation and the time duration at each operating point also serves as a guide to selection of the most suitable capacity control system. Outlet damper control due to its simplicity, ease of operation, and low investment cost, is the most prevalent form of capacity control. However, it is the most inefficient of all methods and is best suited for situations where only small, infrequent changes are required (for example, minor process variations due to seasonal changes. The economic advantage of one method over the other is determined by the time duration over which the fan operates at different operating points. The frequency of flow change is another important determinant. For systems requiring frequent flow control, damper adjustment may not be convenient. Indeed, in many plants, dampers are not easily accessible and are left at some intermediate position to avoid frequent control.

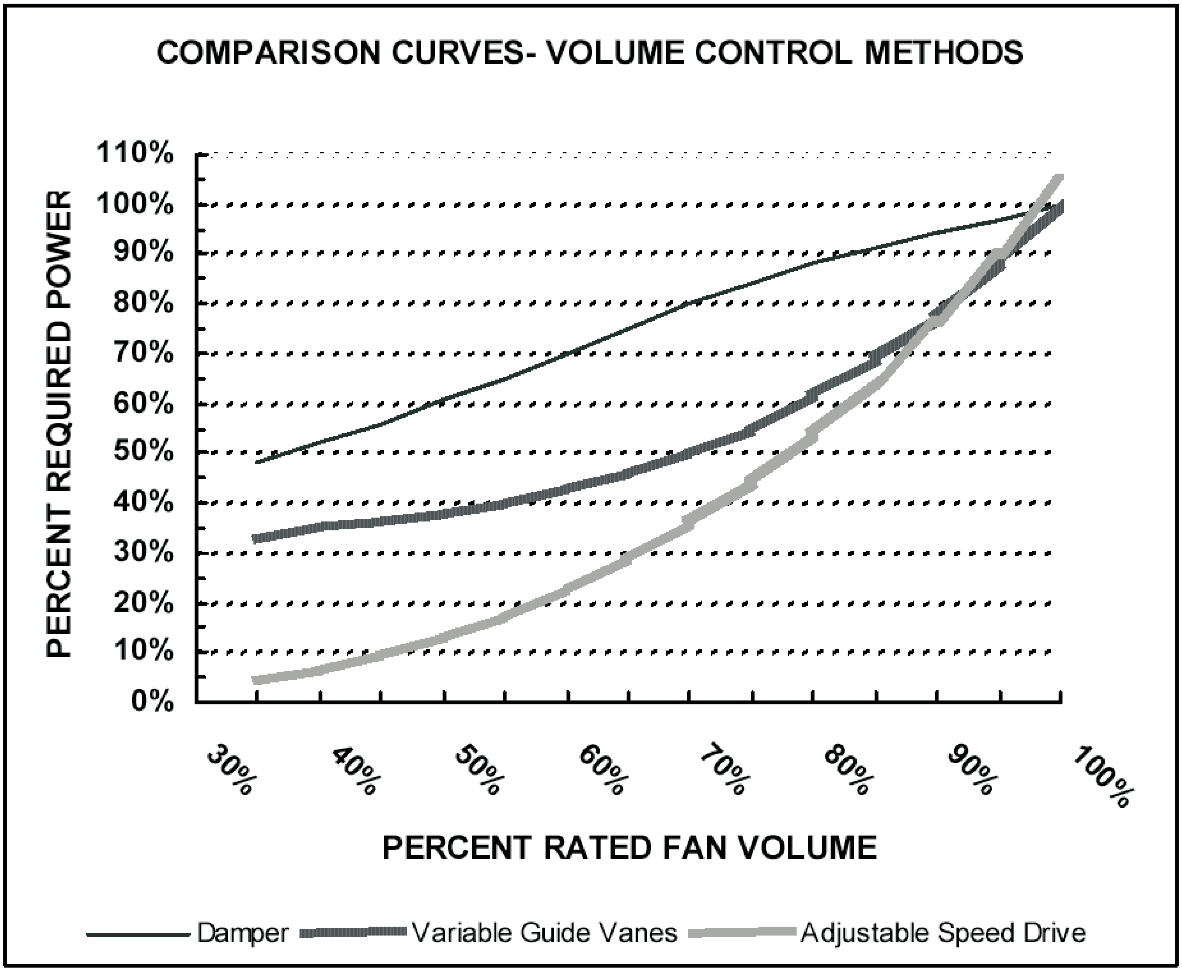


Figure 5.14 Comparison: Various Volume Control Methods

5.6 Fan Performance Assessment

The fans are tested for field performance by measurement of flow, head, temperature on the fan side and electrical motor kW input on the motor side.

Air flow measurement

Static pressure
Static pressure is the potential energy put into the system by the fan. It is given up to friction in the ducts and at the duct inlet as it is converted to velocity pressure. At the inlet to the duct, the static pressure produces an area of low pressure (see Figure 5.15).

Velocity pressure
Velocity pressure is the pressure along the line of the flow that results from the air flowing through the duct. The velocity pressure is used to calculate air velocity.

Total pressure
Total pressure is the sum of the static and velocity pressure. Velocity pressure and static pressure can change as the air flows though different size ducts, accelerating and decelerating the

velocity. The total pressure stays constant, changing only with friction losses. The illustration that follows shows how the total pressure changes in a system.

The fan flow is measured using pitot tube manometer combination, or a flow sensor (differential pressure instrument) or an accurate anemometer. Care needs to be taken regarding number of traverse points, straight length section (to avoid turbulent flow regimes of measurement) up stream and downstream of measurement location. The measurements can be on the suction or discharge side of the fan and preferably both where feasible.

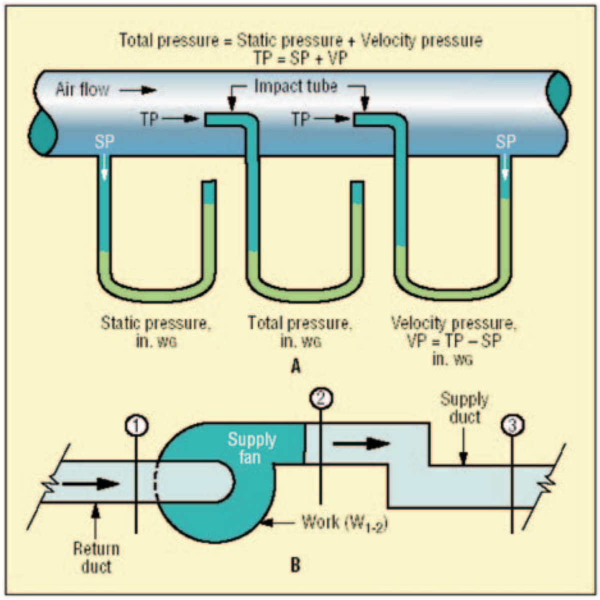


Figure 5.15 Static, Total and Velocity Pressure

Measurement by Pitot tube

The Figure 5.16 shows how velocity pressure is measured using a pitot tube and a manometer. Total pressure is measured using the inner tube of pitot tube and static pressure is measured using the outer tube of pitot tube. When the inner and outer tube ends are connected to a manometer, we get the velocity pressure. For measuring low velocities, it is preferable to use an inclined tube manometer instead of U tube manometer.

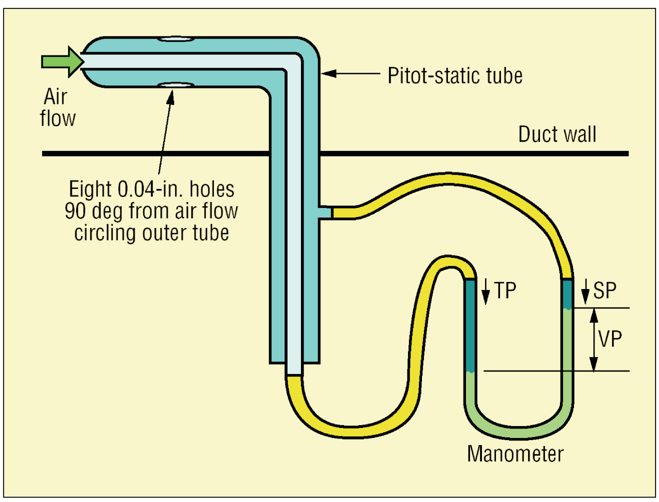


Figure 5.16 Velocity Measurement Using Pitot Tube

Measurements and Calculations

Velocity pressure/velocity calculation

When measuring velocity pressure the duct diameter (or the circumference from which to calculate the diameter) should be measured as well. This will allow us to calculate the velocity and the volume of air in the duct. In most cases, velocity must be measured at several places in the same system.

The velocity pressure varies across the duct. Friction slows the air near the duct walls, so the velocity is greater in the center of the duct. The velocity is affected by changes in the ducting configuration such as bends and curves. The best place to take measurements is in a section of duct that is straight for at least 3–5 diameters after any elbows, branch entries or duct size changes

To determine the average velocity, it is necessary to take a number of velocity pressure readings across the cross-section of the duct. The velocity should be calculated for each velocity pressure reading, and the average of the velocities should be used. Do not average the velocity pressure; average the velocities. For round ducts over 6 inches diameter, the following locations will give areas of equal concentric area (see Figure 5.17).

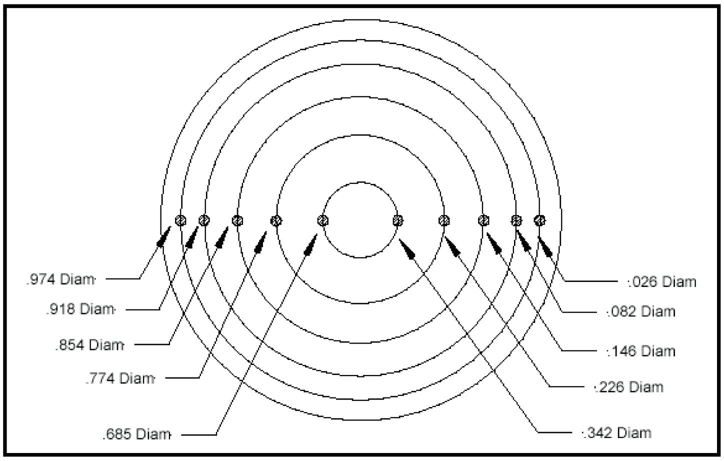


Figure 5.17 Traverse Points for Circular Duct

For best results, one set of readings should be taken in one direction and another set at a 90 ° angle to the first. For square ducts, the readings can be taken in 16 equally spaced areas. If it is impossible to traverse the duct, an approximate average velocity can be calculated by measuring the velocity pressure in the center of the duct and calculating the velocity. This value is reduced to an approximate average by multiplying by 0 .9.

Air density calculation

The first calculation is to determine the density of the air. To calculate the velocity and volume from the velocity pressure measurements it is necessary to know the density of the air. The density is dependent on altitude and temperature.

$$\text{Gas Density}(\gamma) = \frac{273 \quad x \ 1.293}{273 + t^{o}C}$$

t°C – temperature of gas/air at site condition

Velocity calculation

Once the air density and velocity pressure have been established, the velocity can be determined from the equation:

$$\text{Velocity v, m/s} = \frac{C_p \ x \ \sqrt{2 \times 9.81 \times \Delta p \times \gamma}}{\gamma}$$

C_p = Pitot tube constant, 0.85 (or) as given by the manufacturer

Δp = Average differential pressure measured by pitot tube by taking measurement at number of points over the entire cross section of the duct.

γ = Density of air or gas at test condition,

Volume calculation

The volume in a duct can be calculated for the velocity using the equation:

$$\text{Volumetric flow (Q), m}^3\text{/sec} = \text{Velocity,V(m / sec) x Area (m}^2\text{)}$$

Fan efficiency

Fan manufacturers generally use two ways to mention fan efficiency: mechanical efficiency (sometimes called the total efficiency) and static efficiency. Both measure how well the fan converts horsepower into flow and pressure.

The equation for determining mechanical efficiency is:

$$\text{Fan Mechanical Efficiency } \eta_{mechanical}\ \% = \frac{\text{Volume in m}^3\text{/ Sec} \times \Delta p\text{ (total pressure) in mmwc}}{102 \text{ x Power input to the fan shaft in (kW)}} \text{ x } 100$$

The static efficiency equation is the same except that the outlet velocity pressure is not added to the fan static pressure

$$\text{Fan Static Efficiency } \eta_{static}\ \% = \frac{\text{Volume in m}^3\text{/ Sec} \times \Delta p\text{ (static pressure) in mmwc}}{102 \text{ x Power input to the fan shaft in (kW)}} \text{ x } 100$$

Drive motor kW can be measured by a load analyzer. This kW multiplied by motor efficiency gives the shaft power to the fan.

Minimizing demand on the fan.

1. Minimising excess air level in combustion systems to reduce FD fan and ID fan load.
2. Minimising air in-leaks in hot flue gas path to reduce ID fan load, especially in case of kilns, boiler plants, furnaces, etc. Cold air in-leaks increase ID fan load tremendously, due to density increase of flue gases and in-fact choke up the capacity of fan, resulting as a bottleneck for boiler / furnace itself.
3. In-leaks / out-leaks in air conditioning systems also have a major impact on energy efficiency and fan power consumption and need to be minimized.

The findings of performance assessment trials will automatically indicate potential areas for improvement, which could be one or a more of the following:

1. Change of impeller by a high efficiency impeller along with cone.
2. Change of fan assembly as a whole, by a higher efficiency fan
3. Impeller de-rating (by a smaller dia impeller)
4. Change of metallic / Glass reinforced Plastic (GRP) impeller by the more energy efficient hollow FRP impeller with aerofoil design, in case of axial flow fans, where significant savings have been reported
5. Fan speed reduction by pulley dia modifications for derating
6. Option of two speed motors or variable speed drives for variable duty conditions
7. Option of energy efficient flat belts, or, cogged raw edged V belts, in place of conventional V belt systems, for reducing transmission losses.
8. Adopting inlet guide vanes in place of discharge damper control
9. Minimizing system resistance and pressure drops by improvements in duct system

Case Study – 1

VSD Applications

Cement plants use a large number of high capacity fans. By using liners on the impellers, which can be replaced when they are eroded by the abrasive particles in the dust-laden air, the plants have been able to switch from radial blades to forward-curved and backward-curved centrifugal fans. This has vastly improved system efficiency without requiring frequent impeller changes.

For example, a careful study of the clinker cooler fans at a cement plant showed that the flow was much higher than required and also the old straight blade impeller resulted in low system efficiency. It was decided to replace the impeller with a backward-curved blade and use liners to prevent erosion of the blade. This simple measure resulted in a 53 % reduction in power consumption, which amounted to annual savings of Rs. 2.1 million.

Another cement plant found that a large primary air fan which was belt driven through an arrangement of bearings was operating at system efficiency of 23 %. The fan was replaced with a direct coupled fan with a more efficient impeller. Power consumption reduced from 57 kW to 22 kW. Since cement plants use a large number of fans, it is generally possible to integrate the system such that air can be supplied from a common duct in many cases.

For example, a study indicated that one of the fans was operated with the damper open to only 5 %. By re-ducting to allow air to be supplied from another duct where flow was being throttled, it was possible to totally eliminate the use of a 55 kW fan.

The use of variable-speed drives for capacity control can result in significant power savings. A 25 ton-per-hour capacity boiler was equipped with both an induced-draft and forced-draft fan. Outlet dampers were used to control the airflow. After a study of the airflow pattern, it was decided to install a variable speed drive to control air flow. The average power consumption was reduced by nearly 41 kW resulting in annual savings of Rs. 0.33 million. The investment of Rs. 0.65 million for the variable-speed drive was paid back in under 2 years.

The type of variable-speed drive employed also significantly impacts power consumption. Thermal power stations install a hydraulic coupling to control the capacity of the induced-draft fan. It was decided to install a VFD on ID fans in a 200 MW thermal power plant. A comparison of the power consumption of the two fan systems indicated that for similar operating conditions of flow and plant power generation, the unit equipped with the VFD control unit consumed, on average, 4 million units / annum less than the unit equipped with the hydraulic coupling.

Case Study – 2

FRP Fans in Cooling Towers / Humidification Plants

The fans used for cooling tower applications are usually axial flow fans. Such fans are also commonly used in humidification plants. The conventional fans are made from aluminium / steel. These fans are being replaced in recent times by high efficiency FRP (fibre reinforced plastics) fans. The savings potential is shown below:

ILLUSTRATIVE DATA ON ENERGY SAVINGS WITH HIGH EFFICIENCY FRP BLADE AXIAL FLOW FANS

(Source : PCRA Literature)

Fan Data	Type of Fan	Air flow cfm (1 cfm = 1.7 m3/hr	Static Pressure mmWC	Input Power kW	% of Power Saving
Cooling Tower Fans					
24 ft.	Al	664480	3.65	68	Ref.
	FRP	740100	4.61	44	35.29
24 ft.	Al	817650	5.50	60	Ref.
	FRP	919400	6.00	46	23.33
16 ft.	Al	389200	--	45	Ref.
	FRP	391900	--	25	44.44
Humidification Fans					
5.25 ft.	Al	31382	--	7.9	
	FRP	31557	--	5.1	35.80
3.94 ft.	Al	38102	--	13.0	
	FRP	45935	--	13.0	17.05

INDUSTRIAL APPLICATIONS

Hydronics is the use of a liquid heat-transfer medium. In large-scale commercial and industrial buildings, a hydronic system may include a chilled and heated water loop to provide heating and air conditioning. Other hot water applications include high-rise buildings, hotels and multi-unit residences

The Hydronic Heat Solution for construction projects is superior in many ways to all other alternatives.

- No products of combustion or moisture are introduced into the conditioned space.
- Air within the conditioned space is recirculated to eliminate the need to heat vast amounts of outside air.
- Unlimited air ducting strategies are possible.
- Thermostatic temperature control is effective and uniform.
- Air within the conditioned space can be filtered, even to HEPA standards.
- Explosion-proof equipment is available for Volatile environments.
- Heat exchangers can be located all around the structure to provide optimum conditioning uniformity.
- Energy costs per project/per season, are lower than all other methods of temporary space heating.

ENERGY SECTOR

The Hydronic Heat Solution for providing heat (to the oil & gas, mining and diamond drilling industries), is the best method to heat enclosures, tanks with liquids, pipelines and environmental containments.

- Many accessories are available to make use of the “Central Glycol Heaters” for a large variety of heat requirements.
- Explosion-proof equipment is available for Volatile environments.
- No products of combustion (and the resulting moisture) are introduced into conditioned spaces.
- Air within the conditioned space is recirculated to eliminate the need to heat vast amounts of outside air.
- Heat exchangers can be located all around the structure to provide optimum conditioning uniformity.
- Various efficiency air filters can be utilized.

- An incredible cost saving over mobile hot-oil-circulators is achieved.
- Hot glycol distributed in a non-pressure vessel circulation system is a safe reliable alternative to steam pressure vessels which are serious safety hazards and require an operator to have a pressure vessel operator’s certificate.

. AGRICULTURE

The Hydronic Heat Solution for drying grain provides many advantages over the large direct-flame, conventional continuous dryers.

- Drying takes place in the storage bins (on the farm) and utilizes the existing aeration fan-and-duct system which means that handling of the product is, therefore, greatly reduced.
- For Hydronic systems, the products of combustion (and the resulting water vapor) are not forced through the grain as is the case with competitor’s direct burner systems. The result is, NO Contamination of the food product and less moisture to deal with.
- Optimum drying temperature for a Hydronic system is about 100°F, which will not affect the germination ability of the seed in any way. Competitor’s direct-flame systems always run much hotter contact temperatures, which result in loss of germination ability for the seed.
- The comparatively low cost, per bushel, of Hydronic drying, makes it an extremely useful harvest-management tool for the farmer. Threshing can begin when the grain is still high moisture which allows the farmer to preserve top grade product since it prevents bad weather from diminishing the quality of the grain.
- Dry grain stores longer than grain with moisture and therefore reduces spoilage

TUTORIAL QUESTIONS

1. Explain different types of fans & blowers?
2. Explain the Classification of water piping system?
3. Describe about hydronic system?
4. Write about different types of values used in HVAC piping system?
5. Discuss about the fittings used in HVAC piping System?

ASSIGNMENT QUESTIONS

ASSIGNMENT QUESTIONS

UNIT 1

1. Draw the schematic of a reciprocating compressor and explain its working principle
2. Explain the basic functions of expansion devices in refrigeration systems
3. Classify refrigerant evaporators and discuss the salient features of different types of evaporators
4. Write the desirable properties of the Refrigerents and Classification of Refrigerants?.
5. Explain with the neat sketch the working of vapour compression refrigeration system?
6. Explain different modes of heat transfer? explain latent heat and Sensible Heat?

UNIT II

1. Explain with neat sketch working of window air-conditioning?
2. Explain with neat sketch working of packaged Air Conditioning System ?
3. Explian the working of Split A/c System with neat diagram?
4. Describe briefly about Variable Refrigerant Volume (VRV)/ Variable Refrigerant Flow (VRF)?
5. Write Down the applications of ductable A/C

UNIT III

1. Explain the various psychrometric process in detail with the psychrometric chart.
2. Define the following psychrometric terms a. Dry Bulb temperature b.Wet bulb temperature. c. Dew point temperature d. humidity ratio
3. Explain briefly about sensible heating and Sensible cooling?
4. Explain the Heating and Humidification process with neat diagram?
5. Explain the process Cooling and Dehumidification?

UNIT IV

1. Describe the various factors affecting survey of building?
2. Explain ventilation requirements of IAQ?
3. Write about the steps in cooling load calculations?
4. Explain about the u factor of wall,roof?

5. Explain about the ventilation systems standards?

UNIT V

1. Explain different types of fans & blowers?
2. Explain the Classification of water piping system?
3. Describe about hydronic system?
4. Write about different types of values used in HVAC piping system?
5. Discuss about the fittings used in HVAC piping System?