COURSE MATERIAL

II Year B. Tech II- Semester

MECHANICAL ENGINEERING

AY: 2022-23

Applied Thermodynamics

R20A0309

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MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

DEPARTMENT OF MECHANICAL ENGINEERING

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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.

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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.

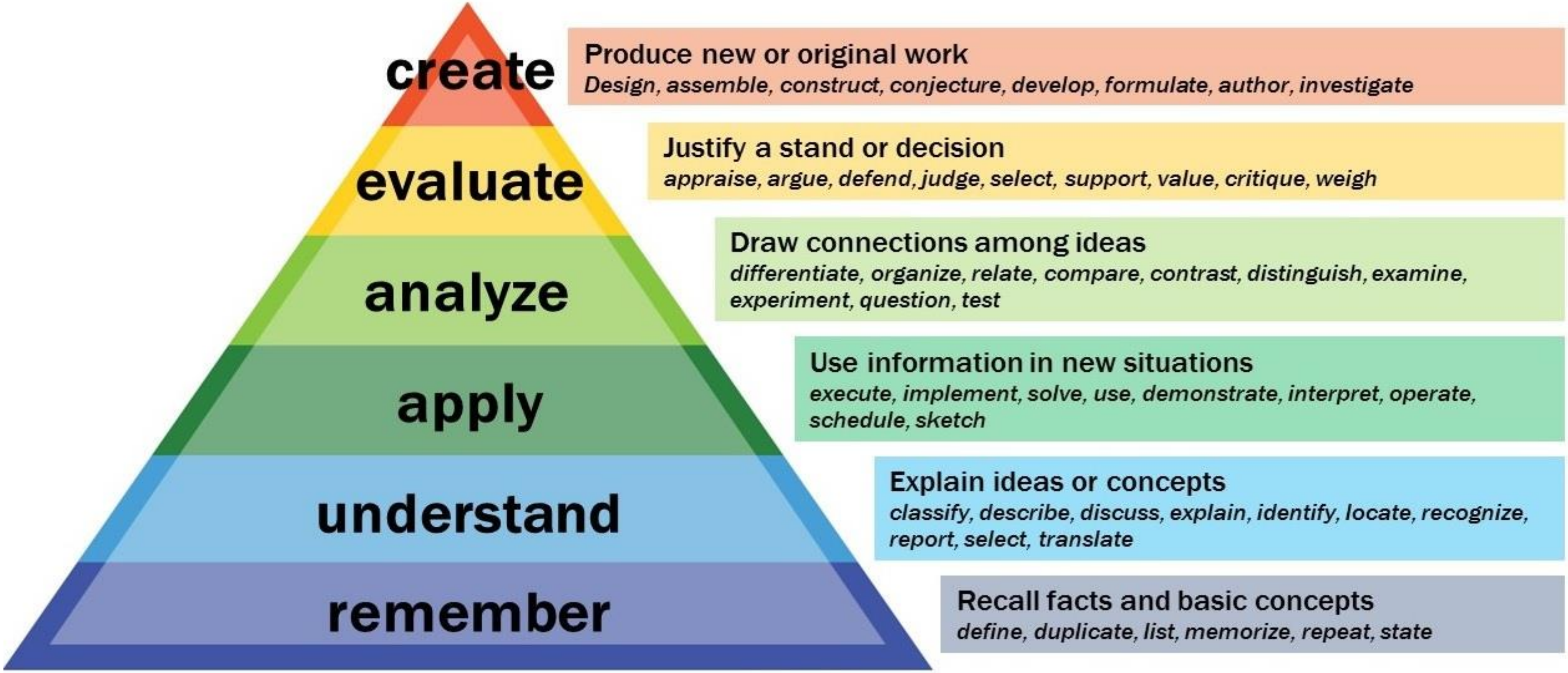
PEO5: PROFESSIONALISM

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 AND TECHNOLOGY

II Year B.Tech. ME- II Sem **L/T/P/C**
3/-/-/3

(R20A0309) APPLIED THERMODYNAMICS

COURSE OBJECTIVES:

1. To have Knowledge in steam power plants and their components, performance and analysis of Steam Turbines, Gas Turbines.
2. To understand Steam nozzles, Steam Condensers and their performances in Industries.
3. The purpose of this course is to enable the student to gain an understanding of how thermodynamic principles govern the behavior of various systems.
4. Evaluate the performance of critical components and accessories steam and gas power plants.
5. To understand the concept of jet propulsion, Rockets and their propellants.

UNIT-I

Basic Concepts: Rankine cycle – Schematic layout, Thermodynamic Analysis, Concept of Mean Temperature of heat addition, Methods to improve cycle performance Regeneration & reheating

Boilers: Classification - Working principles with sketches including H.P. Boilers - Mountings and Accessories - Working principle.

UNIT-II

Steam Nozzles: Function of nozzle - Applications and Types- Flow through nozzles- Thermodynamic analysis.

Steam Condensers: Requirements of steam condensing plant - Classification of condensers - Working principle of different types.

UNIT-III

Steam Turbines: Classification - Impulse turbine; Mechanical details - Velocity diagram - Effect of friction - Power developed, axial thrust, Blade or diagram efficiency - Condition for Maximum efficiency.

Reaction Turbine: Mechanical details - Principle of operation, Thermodynamic analysis of a stage, Degree of reaction - Velocity diagram - Parson's reaction turbine - Condition for Maximum efficiency.

UNIT-IV

Gas Turbines: Simple gas turbine plant - Ideal cycle, essential components - Parameters of Performance - Actual cycle - Regeneration, Inter cooling and Reheating - Closed and Semi - Closed cycles - Merits and Demerits.

UNIT-V

Jet Propulsion: Principle of Operation - Classification of jet propulsive engines – Working Principles with schematic diagrams and representation on T-S diagram- Thrust, Thrust Power and Propulsion Efficiency - Turbo jet engines - Needs and Demands met by Turbo jet -
Schematic Diagram, Thermodynamic Cycle, Performance Evaluation Thrust Augmentation - Methods.
Rockets: Application - Working Principle - Classification - Propellant Type - Thrust, Propulsive Efficiency - Specific Impulse - Solid and Liquid propellant Rocket Engines

TEXT BOOKS:

1. Thermal Engineering / Rajput / Lakshmi Publications.
2. Gas Turbines / V. Ganesan / TMH.
3. Thermal Engineering /P.L. Ballaney / Khanna Publishers, NewDelhi.

REFERENCE BOOKS:

1. Gas Turbines and Propulsive Systems / P. Khajuria & S.P. Dubey / Dhanapatrai Pub.
2. Thermal Engineering / R.S. Khurmi & J.K. Gupta / S. Chand Pub.
3. Thermodynamics and Heat Engines / R. Yadav / Central Book Depot

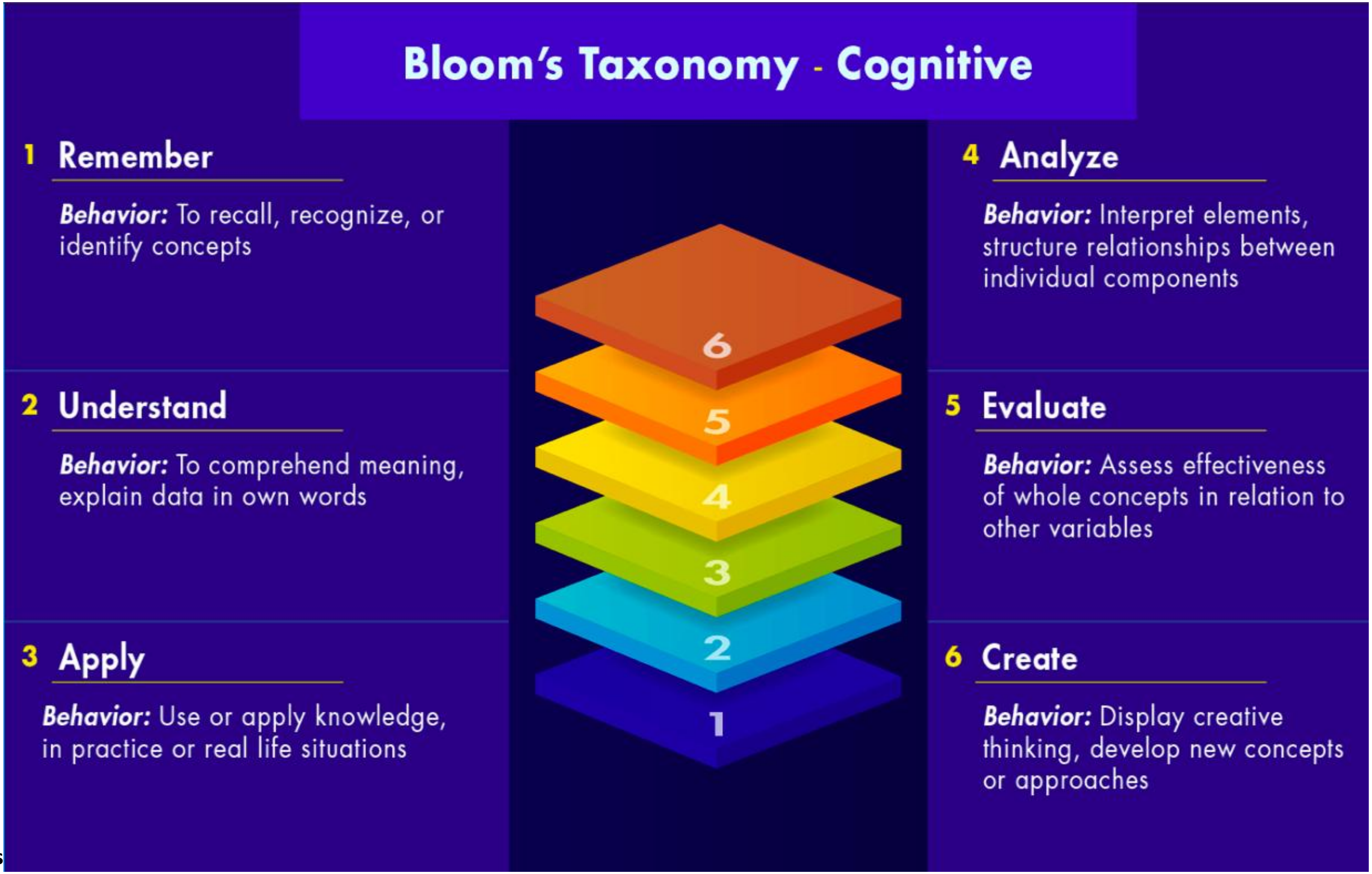
COURSE OUTCOMES:

1. Describe knowledge of Rankine cycle and heat equation in different processes, and improving efficiency techniques.
2. Demonstrate knowledge of ability to identify & apply fundamentals to solve problems involving nozzles and turbines, jet propulsion systems and rockets.
3. Design nozzles, turbines and condensers with desired needs within realistic constraints related thermal fields like different types of power plants etc.
4. Explore their knowledge & ability to design the constructional features of various types of boilers in various fields of energy transfer equipments and to understand the velocity triangles in Steam Turbines & Reaction Turbines
5. Knowledge of impact of engineering solutions on the society and also on contemporary issues related to different types of steam cycles and propulsion systems.

APPLIED THERMODYNAMICS (R18A0308)

COURSE OBJECTIVES

UNIT - 1	CO1: To Explain in detail Basic Components of Rankine cycle. Methods to improve cycle performance. Types of Boilers with their working principles and applications.
UNIT - 2	CO2: To Know the function of nozzles. Thermodynamic analysis of nozzles. Steam condensers and their requirement.
UNIT - 3	CO3: To Study the different types of Turbines. Conditions for maximum efficiency. Parsons reaction turbine.
UNIT - 4	CO4: To study the different types gas turbines. Applications of gas turbines. Efficiency improvement methods.
UNIT - 5	CO5: To know the working principles of different Jet engines, thermodynamic cycles. Rocket engines and their working principles.


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
5
4
3
2
1

S

COURSE OUTLINE

UNIT – 1

NO OF LECTURE HOURS: 15

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Introduction to Rankine cycle: Schematic lay out	p-V and T-s diagrams	Understand the basics of Rankine Cycle. (B1)
2.	Basic Components of Rankine Cycle and thermal power plant.	Boiler, Turbine, Condenser, Pump.	Understand the different components of the thermal power plant (B2)
3.	Methods to improve the Rankine cycle efficiency	Cycle efficiency	Understand the working of Regenerative and reheat cycles (B2)
4.	Types of Boilers and their Function	Bob-cock Wilcox, La-mont, Loeffler boilers etc.	To understand Function of boilers (B3)
5.	High Pressure Boilers and Low pressure boilers	Yarrow Boiler, Benson Boiler	To understand the working principles of High pressure and Low pressure Boilers (B2)
6.	Boiler mountings and accessories.	Pressure Gauge, Fusible plug, water level indicator etc.	To understand Function of mountings and accessories (B3)

UNIT – 2

NO OF LECTURE HOURS: 10

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Functions of Nozzles	nozzles	To understand the function of Nozzles.(B2)
2.	Types of nozzles	Convergent, divergent, Convergent-divergent nozzles	Classification of nozzles (B4)
3.	Thermodynamic analysis	Expansion of steam	To understand change in enthalpy (B2)
4.	Steam condensers requirement	Condensers	To understand the requirement of condensers (B2)
5.	Working principle of different types of condensers	Working principles	To understand the working of condensers (B2) To analyze the Each Component (B4)

UNIT – 3

NO OF LECTURE HOURS: 12

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Classification of steam turbines	Steam Turbines	Types of condensers (B2)
2.	Principle of operation of turbines	Working	Working principles of turbines.(B4)
3.	To draw velocity diagrams	Velocity, diagram	To understand velocity diagrams(B2)
4.	Maximum efficiency calculations	Maximum efficiency	To analyse maximum efficiency (B4)
5.	Working of reaction turbine	Reaction turbine, Parsons turbine.	To be familiar with reaction turbines(B2)

UNIT – 4

NO OF LECTURE HOURS:10

LECTURE	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Simple gas turbine plant	Gas turbine	To Understand the working of Gas turbines (B2).
2.	Ideal cycle, essential components	Components, cycle	To understand the Ideal cycle and its components (B4)
3.	Regeneration cycle	Regeneration	To Analyze the Regeneration cycle (B4).
4.	Inter cooling and Reheating	Inter cooling, Reheating	To understand the Inter cooling and reheat cycles (B4)
5.	Closed and Semi - closed cycle gas turbines	Closed cycle, semi closed cycle	To analyse the semi closed and semi closed gas turbines

UNIT – 5

NO OF LECTURE HOURS: 10

LECTUR E	LECTURE TOPIC	KEY ELEMENTS	LEARNING OBJECTIVES (2 to 3 objectives)
1.	Introduction to Jet propulsion. Classification of Jet engines.	Jet engine, Jet propulsion	To understand jet engine and Jet propulsion. (B2)
2.	Working Principles with schematic diagrams and representation on T-S diagram	T-S diagram	To analyse the T-S diagram (B2).
3.	Thrust, Thrust Power and Propulsion Efficiency	Thrust, Propulsion efficiency	To Understand the thrust power and propulsion (B2).
4.	Needs and Demands met by Turbo jet	Turbo Jet	To Understand the needs of turbo jet (B3)
5.	Thermodynamic Cycle, Performance Evaluation	Performance	To analyse thermodynamic analysis (B4)
6.	Rockets and their working principles	Rockets	To Understand the working of Rocket engines (B2)

Mapping of COs and POs:

Course Outcomes	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3
C308.1	X	X	X	X	X	X	X	X	X	-	-	X	X	X	X
C308.2	X	X	X	X	-	X	X	X	-	-	-	X	X	X	X
C308.3	X	X	X	X	-	X	X	X	-	-	-	X	X	X	X
C308.4	X	X	X	-	-	X	X	X	-	-	-	X	X	X	X
C308.5	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**
C308.1	3	3	2	1	1	3	3	3	1	-	-	3	3	3	2
C308.2	2	3	2	1	-	2	2	2	-	-	-	2	3	2	3
C308.3	2	3	2	1	-	2	2	2	-	-	-	2	3	2	3
C308.4	2	2	3	-	-	2	2	3	-	-	-	3	2	2	2
C308.5	3	2	3	-	-	2	2	3	2	-	-	3	2	2	2

Mode of Evaluation: X

- 70% of marks for External Evaluation.
- 24% of marks for Internal Evaluation.
- 6% of marks for Continuous Evaluation assignments.

UNIT 1

BASIC CONCEPTS & BOILERS

Course Objective:

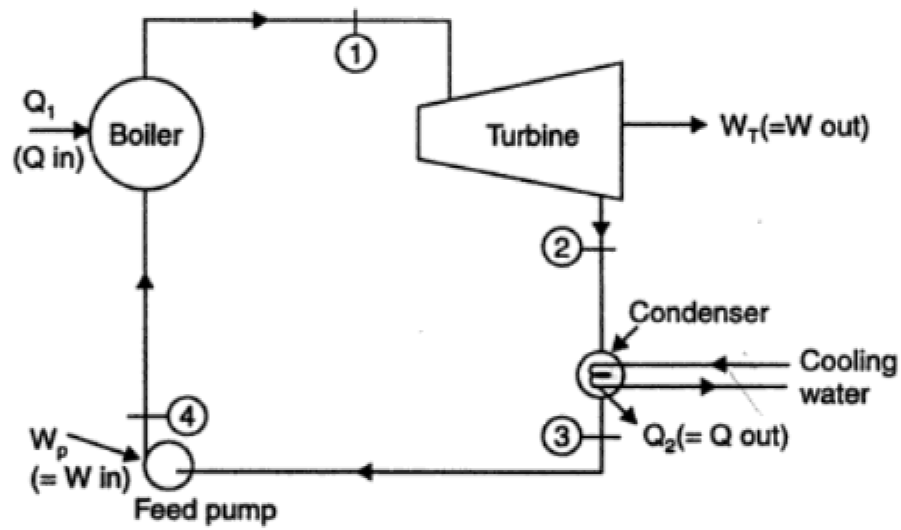
The purpose of this course is to enable the student to gain an understanding of how thermodynamic principles govern the behavior of various systems

Course Outcome:

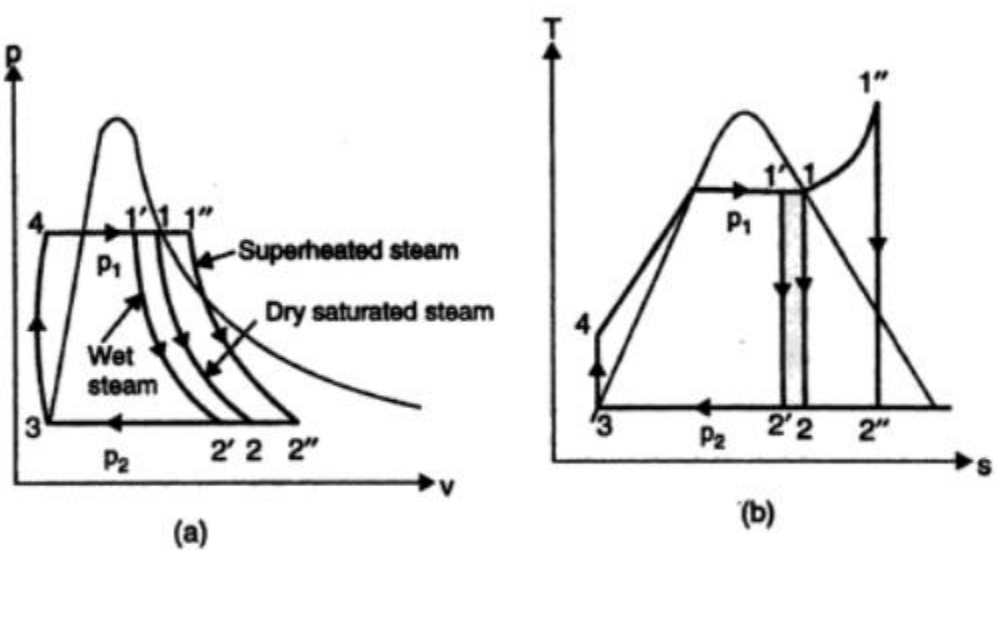
To be able to describe the most important combustion concepts and problems in concern to power plants.

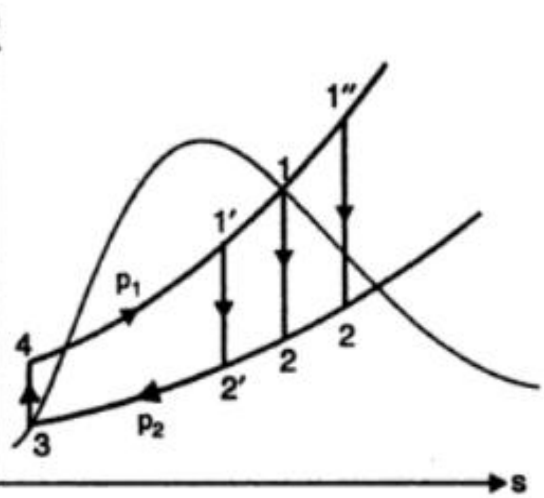
Rankine Cycle:

Rankine cycle is the theoretical cycle on which steam turbine works. In an ideal Rankine cycle, the system executing the cycle undergoes a series of four processes: two isentropic (reversible adiabatic) processes alternated with two isobaric processes:



Rankine cycle.





(c)

(a) p-v **diagram ; *(b) T-s* diagram ; *(c) h-s* diagram for Rankine cycle.**

- **Isentropic expansion** (expansion in a steam turbine) – Steam from the boiler expands adiabatically from state 1 to state 2 in a steam turbine to produce work and then is discharged to the condenser (partially condensed). The steam does work on the surroundings (blades of the turbine) and loses an amount of enthalpy equal to the work that leaves the system. The work done by turbine is given by
  $\mathbf{W_T = H_1 - H_2.}$
  Again the entropy remains unchanged.
- **Isobaric heat rejection (in a heat exchanger)** – In this phase the cycle completes by a constant-pressure process in which heat is rejected from the partially condensed steam. There is heat transfer from the vapor to cooling water flowing in a cooling circuit. The vapor condenses and the temperature of the cooling water increases. The net heat rejected is given by
  $\mathbf{Q_{re} = H_2 - H_3}$

- **Isentropic compression** (compression in centrifugal pumps) –
  The liquid condensate is compressed adiabatically from state 3 to state 4 by centrifugal pumps (usually by condensate pumps and then by feed water pumps). The liquid condensate is pumped from the condenser into the higher pressure boiler. In this process, the surroundings do work on the fluid, increasing its enthalpy (h = u+pv) and compressing it (increasing its pressure). On the other hand the entropy remains unchanged.

  The work required for the compressor is given by $\mathbf{W_{Pumps} = H_4 - H_3.}$

- **Isobaric heat addition** (in a heat exchanger – boiler) – In this phase (between state 2 and state 3) there is a constant-pressure heat transfer to the liquid condensate from an external source, since the chamber is open to flow in and out. The feed water (secondary circuit) is heated from to the boiling point (2 → 3a) of that fluid and then evaporated in the boiler (3a → 3). The net heat added is given by

  $\mathbf{Q_{add} = H_1 – H_4}$

Thermal Efficiency of Rankine Cycle

Considering 1 kg of fluid :

Applying *steady flow energy equation* (S.F.E.E.) to boiler, turbine, condenser and pump :

(*i*) **For boiler** (as control volume), we get

$$h_{f_4} + Q_1 = h_1$$

$$\therefore \quad Q_1 = h_1 - h_{f_4}$$

(*ii*) **For turbine** (as control volume), we get

$$h_1 = W_T + h_2, \text{ where } W_T = \text{turbine work}$$

$$\therefore \quad W_T = h_1 - h_2$$

(*iii*) **For condenser**, we get

$$h_2 = Q_2 + h_{f_3}$$

$$\therefore \quad Q_2 = h_2 - h_{f_3}$$

(*iv*) **For the feed pump**, we get

$$h_{f_3} + W_P = h_{f_4}, \quad \text{where, } W_P = \text{Pump work}$$

$$\therefore \quad W_P = h_{f_4} - h_{f_3}$$

Now, efficiency of Rankine cycle is given by

$$\eta_{\text{Rankine}} = \frac{W_{\text{net}}}{Q_1} = \frac{W_T - W_P}{Q_1}$$

$$= \frac{(h_1 - h_2) - (h_{f_4} - h_{f_3})}{(h_1 - h_{f_4})}$$

The feed pump handles liquid water which is incompressible which means with the increase in pressure its density or specific volume undergoes a little change. Using general property relation for reversible adiabatic compression, we get

$$Tds = dh - vdp$$

$\because$ $ds = 0$

$\therefore$ $dh = vdp$

or $\Delta h = v\,\Delta p$ ...... (since change in specific volume is negligible)

or $h_{f_4} - h_{f_3} = v_3\,(p_1 - p_2)$

When p is in bar and v is in m^3/kg, we have

$$h_{f_4} - h_{f_3} = v_3\,(p_1 - p_2) \times 10^5 \text{ J/kg}$$

The feed pump term $(h_{f_4} - h_{f_3})$ being a small quantity in comparison with turbine work, W_T, is usually neglected, *especially when the boiler pressures are low.*

Then, $\eta_{\text{Rankine}} = \dfrac{h_1 - h_2}{h_1 - h_{f_4}}$

Comparison Between Rankine Cycle and Carnot Cycle

The following points are worth noting :

(*i*) Between the same temperature limits Rankine cycle provides a higher specific work output than a Carnot cycle, consequently Rankine cycle *requires a smaller steam flow rate resulting in smaller size plant for a given power output.* However, Rankine cycle calls for higher *rates of heat transfer in boiler and condenser.*

(*ii*) Since in Rankine cycle only part of the heat is supplied isothermally at constant higher temperature T_1, therefore, its *efficiency is lower* than that of Carnot cycle. The efficiency of the Rankine cycle will approach that of the Carnot cycle more nearly if the *superheat temperature rise is reduced.*

(*iii*) The advantage of using pump to feed liquid to the boiler instead to compressing a wet vapour is obvious that the *work for compression is very large compared to the pump.*

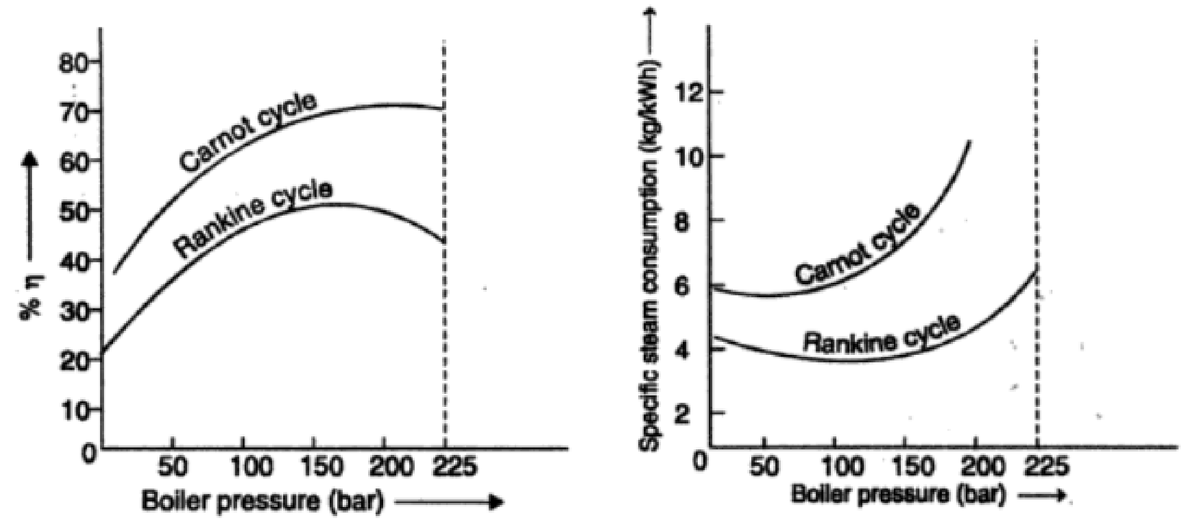
Fig. 15.4 shows the plots between efficiency and specific steam consumption against boiler pressure for Carnot and ideal Rankine cycles.

Effect of Operating Conditions on Rankine Cycle Efficiency

The Rankine cycle efficiency can be *improved* by :

(*i*) ***Increasing the average temperature at which heat is supplied.***

(*ii*) ***Decreasing/reducing the temperature at which heat is rejected.***



(a) (b)

This can be achieved by making suitable changes in the conditions of steam generation or condensation, as discussed below :

1. ***Increasing boiler pressure.*** It has been observed that by increasing the boiler pressure (other factors remaining the same) the cycle tends to rise and reaches a maximum value at a boiler pressure of about 166 bar [Fig. 15.5 (*a*)].

2. ***Superheating.*** All other factors remaining the same, if the steam is superheated before allowing it to expand the Rankine cycle efficiency may be increased [Fig. 15.5 (*b*)]. The use of superheated steam also ensures longer turbine blade life because of the absence of erosion from high velocity water particles that are suspended in wet vapour.

3. ***Reducing condenser pressure.*** The thermal efficiency of the cycle can be amply improved by reducing the condenser pressure [Fig. 15.5 (*c*)] (hence by reducing the temperature at which heat is rejected), especially in high vacuums. But the increase in efficiency is obtained at the *increased cost of condensation apparatus.*

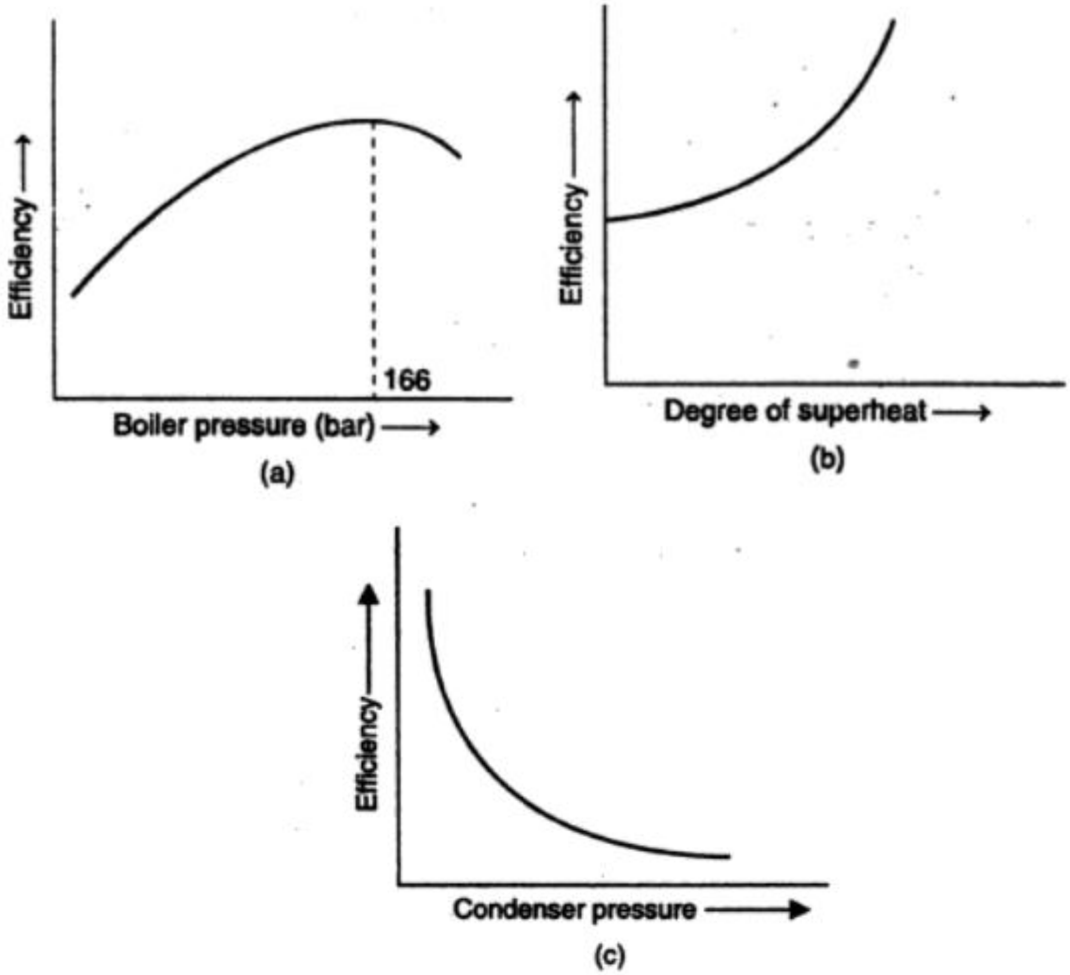


Fig. 15.5. Effect of operating conditions on the thermal efficiency of the Rankine cycle.

The thermal efficiency of the Rankine cycle is also *improved* by the following methods :

(i) By regenerative feed heating.

(ii) By reheating of steam.

(iii) By water extraction.

(iv) By using binary-vapour.

☞**Example 15.1.** *The following data refer to a simple steam power plant :*

S. No.	Location	Pressure	Quality/temp.	Velocity
1.	Inlet to turbine	6 MPa (= 60 bar)	380°C	—
2.	Exit from turbine inlet to condenser	10 kPa (= 0.1 bar)	0.9	200 m/s
3.	Exit from condenser and inlet to pump	9 kPa (= 0.09 bar)	Saturated liquid	—
4.	Exit from pump and inlet to boiler	7 MPa (= 70 bar)	—	—
5.	Exit from boiler	6.5 MPa (= 65 bar)	400°C	—
	Rate of steam flow = 10000 kg/h.			

Calculate :

(i) Power output of the turbine.

(ii) Heat transfer per hour in the boiler and condenser separately.

(iii) Mass of cooling water circulated per hour in the condenser. Choose the inlet temperature of cooling water 20°C and 30°C at exit from the condenser.

(iv) Diameter of the pipe connecting turbine with condenser.

Solution. Refer Fig. 15.6.

(i) Power output of the turbine, P :

At 60 bar, 380°C : From steam tables,

$$h_1 = 3043.0 \text{ (at 350°C)} + \frac{3177.2 - 3043.0}{(400 - 350)} \times 30 \text{ ... By interpolation}$$

$$= \mathbf{3123.5 \ kJ/kg}$$

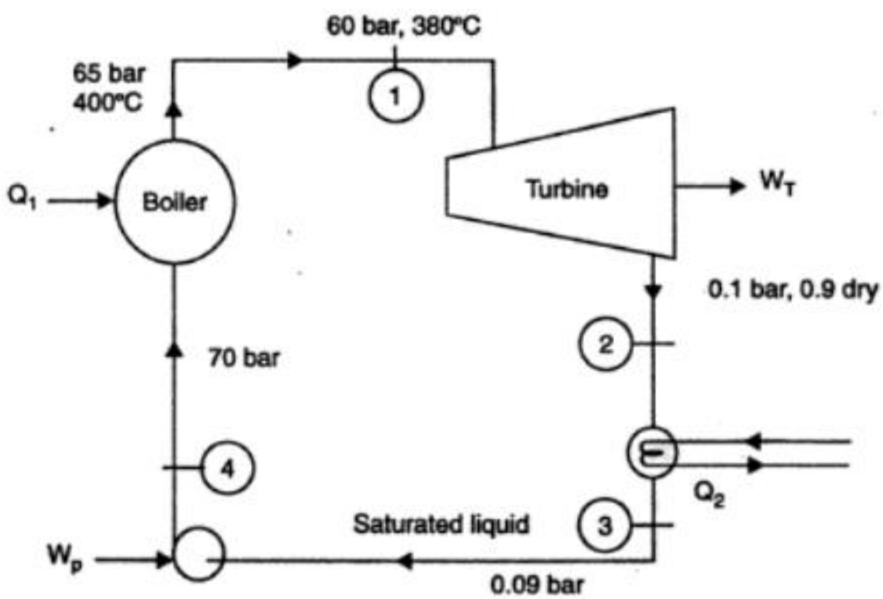


Fig. 15.6

Fig. 15.6

At 0.1 bar :

$$h_{f_2} = 191.8 \text{ kJ/kg}, \; h_{fg_2} = 2392.8 \text{ kJ/kg (from steam tables)}$$

$$x_2 = 0.9 \text{ (given)}$$

$$\therefore \quad h_2 = h_{f_2} + x_2 \, h_{fg_2} = 191.8 + 0.9 \times 2392.8 = 2345.3 \text{ kJ/kg}$$

Power output of the turbine = $m_s (h_1 - h_2)$ kW,

[where m_s = Rate of steam flow in kg/s and h_1, h_2 = Enthalpy of steam in kJ/kg]

$$= \frac{10000}{3600} (3123.5 - 2345.3) = 2162 \text{ kW}$$

Hence power output of the turbine = **2162 kW. (Ans.)**

(*ii*) **Heat transfer per hour in the boiler and condenser :**

At 70 bar : $h_{f_4} = 1267.4$ kJ/kg

$$\textit{At 65 bar, 400°C} : h_a = \frac{3177.2 \, (60 \text{ bar}) + 3158.1 \, (70 \text{ bar})}{2} = 3167.6 \text{ kJ/kg}$$

......(By interpolation)

$\therefore$ *Heat transfer per hour in the boiler,*

$$Q_1 = 10000 \, (h_a - h_{f_4}) \text{ kJ/h}$$

$$= 10000 \, (3167.6 - 1267.4) = \mathbf{1.9 \times 10^7 \text{ kJ/h.}} \quad \textbf{(Ans.)}$$

At 0.09 bar : $h_{f_3} = 183.3$ kJ/kg

Heat transfer per hour in the condenser,

$$Q_1 = 10000 \, (h_2 - h_{f_3})$$

$$= 10000 \, (2345.3 - 183.3) = \mathbf{2.16 \times 10^7 \text{ kJ/h.}} \quad \textbf{(Ans.)}$$

(*iii*) **Mass of cooling water circulated per hour in the condenser, $\mathbf{m_w}$:**

Heat lost by steam = Heat gained by the cooling water

$$Q_2 = m_w \times c_{pw} \, (t_2 - t_1)$$

$$2.16 \times 10^7 = m_w \times 4.18 \, (30 - 20)$$

$$\therefore \quad \mathbf{m_w} = \frac{2.16 \times 10^7}{4.18 \, (30 - 20)} = \mathbf{1.116 \times 10^7 \text{ kg/h.}} \quad \textbf{(Ans.)}$$

(iv) Diameter of the pipe connecting turbine with condenser, d :

$$\frac{\pi}{4} d^2 \times C = m_s x_2 v_{g_2} \quad ...(i)$$

Here, d = Diameter of the pipe (m),

C = Velocity of steam = 200 m/s (given),

m_s = Mass of steam in kg/s,

x_2 = Dryness fraction at '2', and

v_{g_2} = Specific volume at pressure 0.1 bar (= 14.67 m³/kg).

Substituting the various values in eqn. (i), we get

$$\frac{\pi}{4} d^2 \times 200 = \frac{10000}{3600} \times 0.9 \times 14.67$$

$$d = \left(\frac{10000 \times 0.9 \times 14.67 \times 4}{3600 \times \pi \times 200}\right)^{1/2} = 0.483 \text{ m or } \mathbf{483 \text{ mm.}} \quad \textbf{(Ans.)}$$

Example 15.2. *In a steam power cycle, the steam supply is at 15 bar and dry and saturated. The condenser pressure is 0.4 bar. Calculate the Carnot and Rankine efficiencies of the cycle. Neglect pump work.*

Solution. Steam supply pressure, p_1 = 15 bar, x_1 = 1

Condenser pressure, p_2 = 0.4 bar

Carnot and Rankine efficiencies :

From steam tables :

At 15 bar : t_s = 198.3°C, h_g = 2789.9 kJ/kg, s_g = 6.4406 kJ/kg K

At 0.4 bar : t_s = 75.9°C, h_f = 317.7 kJ/kg, h_{fg} = 2319.2 kJ/kg,

s_f = 1.0261 kJ/kg K, s_{fg} = 6.6448 kJ/kg K

$$T_1 = 198.3 + 273 = 471.3 \text{ K}$$

$$T_2 = 75.9 + 273 = 348.9 \text{ K}$$

$$\therefore \quad \eta_{carnot} = \frac{T_1 - T_2}{T_1} = \frac{471.3 - 348.9}{471.3}$$

$$= \mathbf{0.259 \text{ or } 25.9\%.} \quad \textbf{(Ans.)}$$

$$\eta_{Rankine} = \frac{\text{Adiabatic or isentropic heat drop}}{\text{Heat supplied}} = \frac{h_1 - h_2}{h_1 - h_{f_2}}$$

where $h_2 = h_{f_2} + x_2 \, h_{fg_2} = 317.7 + x_2 \times 2319.2$ ...(*i*)

Value of x_2 :

As the steam expands isentropically,

$$\therefore \quad s_1 = s_2$$

$$6.4406 = s_{f_2} + x_2 \, s_{fg_2} = 1.0261 + x_2 \times 6.6448$$

$$\therefore \quad x_2 = \frac{6.4406 - 1.0261}{6.6448} = 0.815$$

$$\therefore \quad h_2 = 317.7 + 0.815 \times 2319.2 = 2207.8 \text{ kJ/kg} \qquad \text{[From eqn. } (i)\text{]}$$

Hence, $\eta_{\text{Rankine}} = \dfrac{2789.9 - 2207.8}{2789.9 - 317.7} = \mathbf{0.2354 \text{ or } 23.54\%.}$ **(Ans.)**

Example 15.3. *In a steam turbine steam at 20 bar, 360°C is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work and the cycle efficiency.*

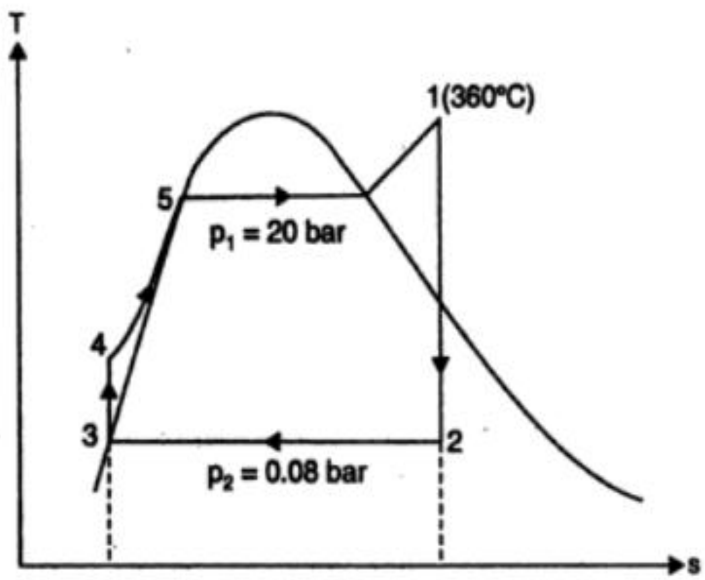


Fig. 15.7

Solution. Boiler pressure, $p_1 = 20$ bar (360°C)

Condenser pressure, $p_2 = 0.08$ bar

From steam tables :

At 20 bar (p_1), 360°C : $h_1 = 3159.3$ kJ/kg

$s_1 = 6.9917$ kJ/kg-K

At 0.08 bar (p_2) : $h_3 = h_{f(p_2)} = 173.88$ kJ/kg,

$s_3 = s_{f(p_2)} = 0.5926$ kJ/kg-K

$h_{fg(p_2)} = 2403.1$ kJ/kg, $\quad s_{g(p_2)} = 8.2287$ kJ/kg-K

$v_{f(p_2)} = 0.001008$ m³/kg $\quad \therefore \quad s_{fg(p_2)} = 7.6361$ kJ/kg-K

Now $\quad s_1 = s_2$

$$6.9917 = s_{f(p_2)} + x_2\, s_{fg(p_2)} = 0.5926 + x_2 \times 7.6361$$

$$\therefore \quad x_2 = \frac{0.69917 - 0.5926}{7.6361} = 0.838$$

$$\therefore \quad h_2 = h_{f(p_2)} + x_2\, h_{fg(p_2)}$$

$$= 173.88 + 0.838 \times 2403.1 = 2187.68 \text{ kJ/kg.}$$

Net work, W_{net} :

$$W_{net} = W_{turbine} - W_{pump}$$

$$W_{pump} = h_{f_4} - h_{f(p_2)} \; (= h_{f_3}) = v_{f(p_2)} \; (p_1 - p_2)$$

$$= 0.00108 \text{ (m}^3\text{/kg)} \times (20 - 0.08) \times 100 \text{ kN/m}^2$$

$$= 2.008 \text{ kJ/kg}$$

[and $\quad h_{f_4} = 2.008 + h_{f(p_2)} = 2.008 + 173.88 = 175.89$ kJ/kg]

$$W_{turbine} = h_1 - h_2 = 3159.3 - 2187.68 = 971.62 \text{ kJ/kg}$$

$\therefore \quad W_{net} = 971.62 - 2.008 =$ **969.61 kJ/kg. (Ans.)**

Cycle efficiency, η_{cycle} :

$$Q_1 = h_1 - h_{f_4} = 3159.3 - 175.89 = 2983.41 \text{ kJ/kg}$$

$\therefore \quad \eta_{cycle} = \dfrac{W_{net}}{Q_1} = \dfrac{969.61}{2983.41} =$ **0.325 or 32.5%. (Ans.)**

Example 15.4. *A Rankine cycle operates between pressures of 80 bar and 0.1 bar. The maximum cycle temperature is 600°C. If the steam turbine and condensate pump efficiencies are 0.9 and 0.8 respectively, calculate the specific work and thermal efficiency. Relevant steam table extract is given below.*

p(bar)	t(°C)	Specific volume (m³/kg)		Specific enthalpy (kJ/kg)			Specific entropy (kJ/kg K)		
		v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g
0.1	45.84	0.0010103	14.68	191.9	2392.3	2584.2	0.6488	7.5006	8.1494
80	295.1	0.001385	0.0235	1317	1440.5	2757.5	3.2073	2.5351	5.7424

80 bar, 600°C	v	0.486 m³/kg
Superheat	h	3642 kJ/kg
table	s	7.0206 kJ/kgK

(GATE, 1998)

Solution. Refer Fig. 15.8

At 80 bar, 600°C :

h_1 = 3642 kJ / kg ;

s_1 = 7.0206 kJ / kg K.

Since $s_1 = s_2$,

$\therefore$ $7.0206 = s_{f_2} + x_2\, s_{fg_2}$

$= 0.6488 + x_2 \times 7.5006$

$$x_2 = \frac{7.0206 - 0.6488}{7.5006} = 0.85$$

Now, $h_2 = h_{f_2} + x_2\, h_{fg_2}$

$= 191.9 + 0.85 \times 2392.3$

$= 2225.36$ kJ/kg

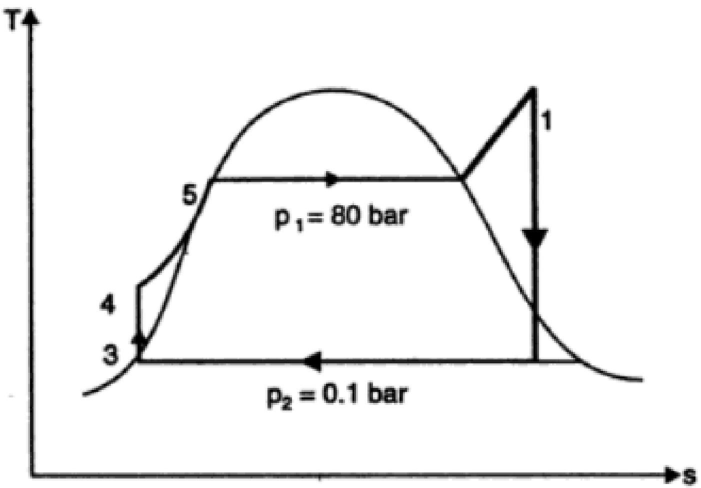


Fig. 15.8

Actual turbine work

$= \eta_{turbine} \times (h_1 - h_2)$

$= 0.9\,(3642 - 2225.36) = 1275$ kJ/kg

Pump work $= v_{f(p_2)}\,(p_1 - p_2)$

$$= 0.0010103\,(80 - 0.1) \times \frac{10^5}{10^3}\ \text{kN/m}^2 = 8.072\ \text{kJ/kg}$$

Actual pump work $= \dfrac{8.072}{\eta_{pump}} = \dfrac{8.072}{0.8} = 10.09$ kJ/kg

Specific work $(W_{net}) = 1275 - 10.09 =$ **1264.91 kJ / kg. (Ans.)**

Thermal efficiency $= \dfrac{W_{net}}{Q_1}$

ıere, $Q_1 = h_1 - h_{f_4}$

But $h_{f_4} = h_{f_3} + \text{pump work} = 191.9 + 10.09 = 202$ kJ/kg

$\therefore$ Thermal efficiency, $\eta_{th} = \dfrac{1264.91}{3642 - 202} =$ **0.368 or 36.8 %. (Ans.)**

Example 15.5. *A simple Rankine cycle works between pressures 28 bar and 0.06 bar, the initial condition of steam being dry saturated. Calculate the cycle efficiency, work ratio and specific steam consumption.*

Solution.

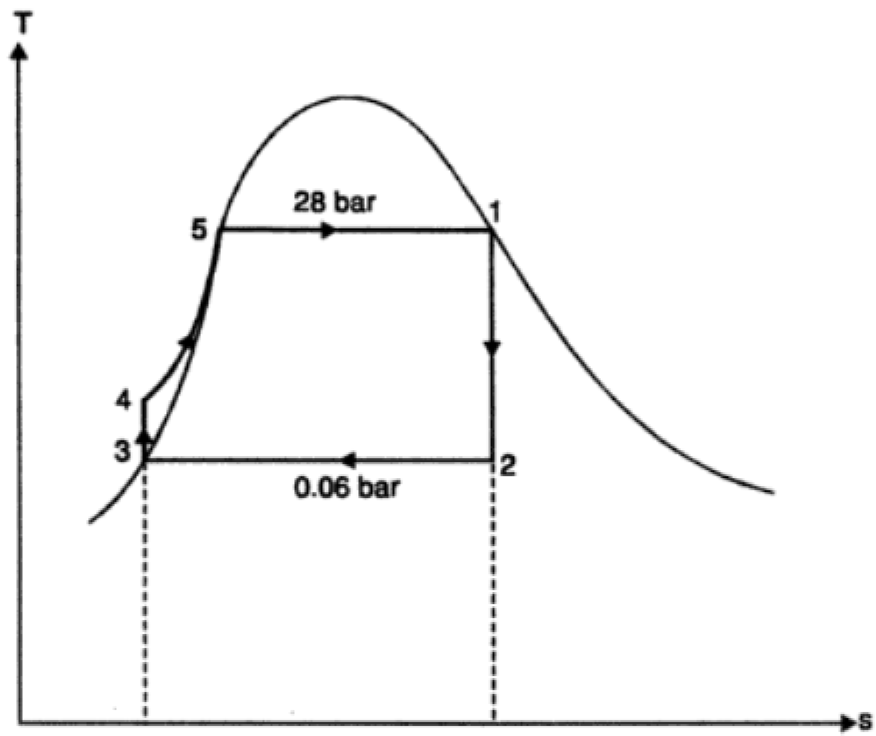


Fig. 15.9

From steam tables,

At 28 bar : $h_1 = 2802$ kJ/kg, $s_1 = 6.2104$ kJ/kg K

At 0.06 bar : $h_{f_2} = h_{f_3} = 151.5$ kJ/kg, $h_{fg_2} = 2415.9$ kJ/kg,

$s_{f_2} = 0.521$ kJ/kg K, $s_{fg_2} = 7.809$ kJ/kg K

$v_f = 0.001\ \text{m}^3/\text{kg}$

Considering turbine *process 1-2*, we have :

$$s_1 = s_2$$

$$6.2104 = s_{f_2} + x_2\, s_{fg_2} = 0.521 + x_2 \times 7.809$$

$$\therefore \quad x_2 = \frac{6.2104 - 0.521}{7.809} = 0.728$$

$$\therefore \quad h_2 = h_{f_2} + x_2\, h_{fg_2}$$

$$= 151.5 + 0.728 \times 2415.9 = 1910.27 \text{ kJ/kg}$$

$\therefore$ Turbine work, $W_{turbine} = h_1 - h_2 = 2802 - 1910.27 = 891.73$ kJ/kg

Pump work, $W_{pump} = h_{f_4} - h_{f_3} = v_f(p_1 - p_2)$

$$= \frac{0.001\,(28 - 0.06) \times 10^5}{1000} = 2.79 \text{ kJ/kg}$$

$[\because \quad h_{f_4} = h_{f_3} + 2.79 = 151.5 + 2.79 = 154.29 \text{ kJ/kg}]$

$\therefore$ Net work, $W_{net} = W_{turbine} - W_{pump}$

$$= 891.73 - 2.79 = 888.94 \text{ kJ/kg}$$

Cycle efficiency $= \dfrac{W_{net}}{Q_1} = \dfrac{888.94}{h_1 - h_{f_4}}$

$$= \frac{888.94}{2802 - 154.29} = \mathbf{0.3357 \text{ or } 33.57\%.} \quad \textbf{(Ans.)}$$

Work ratio $= \dfrac{W_{net}}{W_{turbine}} = \dfrac{888.94}{891.73} = \mathbf{0.997.}$ **(Ans.)**

Specific steam consumption $= \dfrac{3600}{W_{net}} = \dfrac{3600}{888.94} = \mathbf{4.049\ kg/kWh.}$ **(Ans.)**

W_{net} 000.04

☞**Example 15.6.** *In a Rankine cycle, the steam at inlet to turbine is saturated at a pressure of 35 bar and the exhaust pressure is 0.2 bar. Determine :*

(*i*) *The pump work,* (*ii*) *The turbine work,*

(*iii*) *The Rankine efficiency,* (*iv*) *The condenser heat flow,*

(*v*) *The dryness at the end of expansion.*

Assume flow rate of 9.5 kg/s.

Solution. Pressure and condition of steam, at inlet to the turbine,

$p_1 = 35$ bar, $x = 1$

Exhaust pressure, $p_2 = 0.2$ bar

Flow rate, $\dot{m} = 9.5$ kg/s

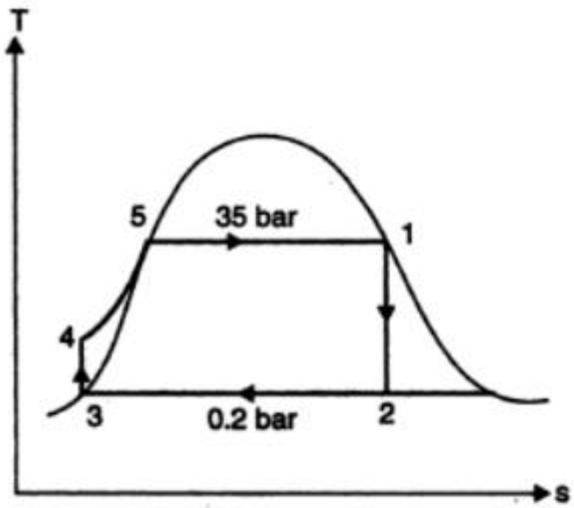


Fig. 15.10

From steam tables :

At 35 bar : $h_1 = h_{g_1} = 2802$ kJ/kg, $s_{g_1} = 6.1228$ kJ/kg K

At 0.26 bar : $h_f = 251.5$ kJ/kg, $h_{fg} = 2358.4$ kJ/kg,

$v_f = 0.001017$ m^3/kg, $s_f = 0.8321$ kJ/kg K, $s_{fg} = 7.0773$ kJ/kg K.

(*i*) **The pump work :**

Pump work $= (p_4 - p_3)\, v_f = (35 - 0.2) \times 10^5 \times 0.001017$ J or 3.54 kJ/kg

$$\begin{bmatrix} \text{Also } h_{f_4} - h_{f_3} = \text{Pump work} = 3.54 \\ \therefore \quad h_{f_4} = 251.5 + 3.54 = 255.04 \text{ kJ / kg} \end{bmatrix}$$

Now power required to drive the pump

$= 9.5 \times 3.54$ kJ/s or **33.63 kW. (Ans.)**

(*ii*) **The turbine work :**

$$s_1 = s_2 = s_{f_2} + x_2 \times s_{fg_2}$$

$$6.1228 = 0.8321 + x_2 \times 7.0773$$

$$\therefore \quad x_2 = \frac{6.1228 - 0.8321}{7.0773} = 0.747$$

$$\therefore \quad h_2 = h_{f_2} + x_2\, h_{fg_2} = 251.5 + 0.747 \times 2358.4 = 2013 \text{ kJ/kg}$$

$\therefore$ *Turbine work* $= \dot{m}\,(h_1 - h_2) = 9.5\,(2802 - 2013) =$ **7495.5 kW. (Ans.)**

It may be noted that pump work (33.63 kW) is very small as compared to the turbine work (7495.5 kW).

(iii) The Rankine efficiency :

$$\eta_{rankine} = \frac{h_1 - h_2}{h_1 - h_{f_2}} = \frac{2802 - 2013}{2802 - 251.5} = \frac{789}{2550.5} = 0.3093 \text{ or } 30.93\%. \quad \textbf{(Ans.)}$$

(iv) The condenser heat flow :

The condenser heat flow $= \dot{m}\ (h_2 - h_{f_2}) = 9.5\ (2013 - 251.5) =$ **16734.25 kW. (Ans.)**

(v) The dryness at the end of expansion, x_2 :

The dryness at the end of expansion,

$$x_2 = 0.747 \text{ or } 74.7\%. \quad \textbf{(Ans.)}$$

Example 15.7. *The adiabatic enthalpy drop across the primemover of the Rankine cycle is 840 kJ/kg. The enthalpy of steam supplied is 2940 kJ/kg. If the back pressure is 0.1 bar, find the specific steam consumption and thermal efficiency.*

Solution. Adiabatic enthalpy drop, $h_1 - h_2 = 840$ kJ/kg

Enthalpy of steam supplied, $h_1 = 2940$ kJ/kg

Back pressure, $p_2 = 0.1$ bar

From steam tables, corresponding to 0.1 bar : $h_f = 191.8$ kJ/kg

Now,
$$\eta_{rankine} = \frac{h_1 - h_2}{h_1 - h_{f_2}} = \frac{840}{2940 - 191.8} = 0.3056 = 30.56\%. \quad \textbf{(Ans.)}$$

Useful work done per kg of steam = 840 kJ/kg

$$\therefore \quad \textit{Specific steam consumption} = \frac{1}{840} \text{ kg/s} = \frac{1}{840} \times 3600 = \textbf{4.286 kg/kWh.} \quad \textbf{(Ans.)}$$

Example 15.8. *A 35 kW (I.P.) system engines consumes 284 kg/h at 15 bar and 250°C. If condenser pressure is 0.14 bar, determine :*

(i) *Final condition of steam ;* (ii) *Rankine efficiency ;*
(iii) *Relative efficiency.*

Solution. Power developed by the engine = 35 kW (I.P.)
Steam consumption = 284 kg/h
Condenser pressure = 0.14 bar
Steam inlet pressure = 15 bar, 250°C.
From steam tables :

At 15 bar, 250°C : h = 2923.3 kJ/kg, s = 6.709 kJ/kg K

At 0.14 bar : h_f = 220 kJ/kg, h_{fg} = 2376.6 kJ/kg,
s_f = 0.737 kJ/kg K, s_{fg} = 7.296 kJ/kg K

(i) **Final condition of steam :**

Since steam expands isentropically.

$$\therefore \quad s_1 = s_2 = s_{f_2} + x_2 \, s_{fg_2}$$

$$6.709 = 0.737 + x_2 \times 7.296$$

$$\therefore \quad x_2 = \frac{6.709 - 0.737}{7.296} = \mathbf{0.818 \simeq 0.82.} \quad \textbf{(Ans.)}$$

$$\therefore \quad h_2 = h_{f_2} + x_2 \, h_{fg_2} = 220 + 0.82 \times 2376.6 = 2168.8 \text{ kJ/kg.}$$

(ii) **Rankine efficiency :**

$$\eta_{\text{rankine}} = \frac{h_1 - h_2}{h_1 - h_{f_2}} = \frac{2923.3 - 2168.8}{2923.3 - 220} = \mathbf{0.279 \text{ or } 27.9\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Relative efficiency :**

$$\eta_{\text{thermal}} = \frac{\text{I.P.}}{\dot{m}\left(h_1 - h_{f_2}\right)} = \frac{35}{\frac{284}{3600}(2923.3 - 220)} = 0.1641 \text{ or } 16.41\%$$

$$\eta_{\text{relative}} = \frac{\eta_{\text{thermal}}}{\eta_{\text{rankine}}} = \frac{0.1641}{0.279}$$

$$= \mathbf{0.588 \text{ or } 58.8\%.} \quad \textbf{(Ans.)}$$

Example 15.9. *Calculate the fuel oil consumption required in a industrial steam plant to generate 5000 kW at the turbine shaft. The calorific value of the fuel is 40000 kJ/kg and the Rankine cycle efficiency is 50%. Assume appropriate values for isentropic turbine efficiency, boiler heat transfer efficiency and combustion efficiency.* **(AMIE Summer, 2000)**

Solution. Power to be generated at the turbine shaft, $P = 5000$ kW

The calorific value of the fuel, $C = 40000$ kJ/kg

Rankine cycle efficiency, $\eta_{\text{rankine}} = 50\%$

Fuel oil combustion, m_f :

Assume : $\eta_{\text{turbine}} = 90\%$; $\eta_{\text{heat transfer}} = 85\%$; $\eta_{\text{combustion}} = 98\%$

$$\eta_{\text{rankine}} = \frac{\text{Shaft power} / \eta_{\text{turbine}}}{m_f \times C \times \eta_{\text{heat transfer}} \times \eta_{\text{combustion}}}$$

or

$$0.5 = \frac{(5000/0.9)}{m_f \times 40000 \times 0.85 \times 0.98}$$

$$\therefore \quad m_f = \frac{(5000/0.9)}{0.5 \times 40000 \times 0.85 \times 0.98} = \mathbf{0.3335 \text{ kg/s or } 1200.6 \text{ kg/h.}} \quad \textbf{(Ans.)}$$

REGENERATIVE CYCLE

In the Rankine cycle it is observed that the condensate which is fairly at low temperature has an irreversible mixing with hot boiler water and this results in decrease of cycle efficiency. Methods are, therefore, adopted to heat the feed water from the hot well of condenser irreversibly by interchange of heat within the system and thus improving the cycle efficiency. This heating method is called regenerative feed heat and the cycle is called *regenerative cycle*.

The principle of regeneration can be practically utilised by extracting steam from the turbine at several locations and supplying it to the regenerative heaters. The resulting cycle is known as *regenerative or bleeding cycle*. The heating arrangement comprises of : (*i*) For medium capacity turbines—not more than 3 heaters ; (*ii*) For high pressure high capacity turbines—not more than 5 to 7 heaters ; and (*iii*) For turbines of super critical parameters 8 to 9 heaters. The most advantageous condensate heating temperature is selected depending on the turbine throttle conditions and this determines the number of heaters to be used. The final condensate heating temperature is kept 50 to 60°C below the boiler saturated steam temperature so as to prevent evaporation of water in the feed mains following a drop in the boiler drum pressure. The conditions of steam bled for each heater are so selected that the temperature of saturated steam will be 4 to 10°C higher than the final condensate temperature.

Fig. 15.15 (*a*) shows a diagrammatic layout of a condensing steam power plant in which a surface condenser is used to condense all the steam that is not extracted for feed water heating. The turbine is double extracting and the boiler is equipped with a superheater. The cycle diagram (*T-s*) would appear as shown in Fig. 15.15 (*b*). This arrangement constitutes a *regenerative cycle*.

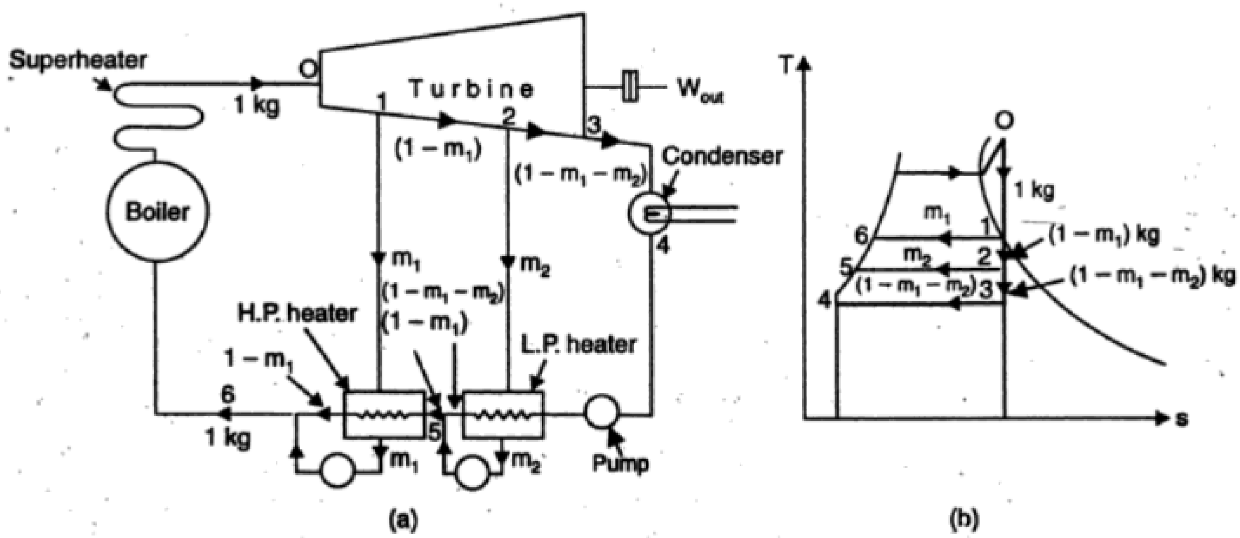


Fig. 15.15. Regenerative cycle.

Energy/Heat balance equation for H.P. heater :

$$m_1 (h_1 - h_{f_6}) = (1 - m_1)(h_{f_6} - h_{f_5})$$

or $$m_1[(h_1 - h_{f_6}) + (h_{f_6} - h_{f_5})] = (h_{f_6} - h_{f_5})$$

or $$m_1 = \frac{h_{f_6} - h_{f_5}}{h_1 - h_{f_5}} \quad ...(15.8)$$

Energy/Heat balance equation for L.P. heater :

$$m_2 (h_2 - h_{f_5}) = (1 - m_1 - m_2)(h_{f_5} - h_{f_3})$$

or $$m_2[(h_2 - h_{f_5}) + (h_{f_5} - h_{f_3})] = (1 - m_1)(h_{f_5} - h_{f_3})$$

or $$m_2 = \frac{(1 - m_1)(h_{f_5} - h_{f_3})}{(h_2 - h_{f_3})} \quad ...(15.9)$$

All enthalpies may be determined ; therefore m_1 and m_2 may be found. The maximum temperature to which the water can be heated is dictated by that of bled steam. The condensate from the bled steam is added to feed water.

Neglecting pump work :

The heat supplied externally in the cycle

$$= (h_0 - h_{f_6})$$

Isentropic work done $= m_1 (h_0 - h_1) + m_2 (h_0 - h_2) + (1 - m_1 - m_2)(h_0 - h_3)$

The thermal efficiency of regenerative cycle is

$$\eta_{\text{thermal}} = \frac{\text{Work done}}{\text{Heat supplied}}$$

$$= \frac{m_1 (h_0 - h_1) + m_2 (h_0 - h_2) + (1 - m_1 - m_2)(h_0 - h_3)}{(h_0 - h_{f_6})} \quad ...(15.10)$$

[The work done by the turbine may also be calculated by summing up the products of the steam flow and the corresponding heat drop in the turbine stages.

i.e., Work done $= (h_0 - h_1) + (1 - m_1)(h_1 - h_2) + (1 - m_1 - m_2)(h_2 - h_3)$]

Advantages of Regenerative cycle over Simple Rankine cycle :

1. The heating process in the boiler tends to become reversible.

2. The thermal stresses set up in the boiler are minimised. This is due to the fact that temperature ranges in the boiler are reduced.

3. The thermal efficiency is improved because the average temperature of heat addition to the cycle is increased.

4. Heat rate is reduced.

5. The blade height is less due to the reduced amount of steam passed through the low pressure stages.

6. Due to many extractions there is an improvement in the turbine drainage and it reduces erosion due to moisture.

7. A small size condenser is required.

Disadvantages :

1. The plant becomes more complicated.

2. Because of addition of heaters greater maintenance is required.

3. For given power a large capacity boiler is required.

4. The heaters are costly and the gain in thermal efficiency is not much in comparison to the heavier costs.

Note. In the absence of precise information (regarding actual temperature of the feed water entering and leaving the heaters and of the condensate temperatures) the following assumption should always be made while doing calculations :

1. Each heater is ideal and bled steam just condenses.

2. The feed water is heated to saturation temperature at the pressure of bled steam.

3. Unless otherwise stated the work done by the pumps in the system is considered negligible.

4. There is equal temperature rise in all the heaters (usually 10°C to 15°C).

Example 15.12. *A steam turbine is fed with steam having an enthalpy of 3100 kJ/kg. It moves out of the turbine with an enthalpy of 2100 kJ/kg. Feed heating is done at a pressure of 3.2 bar with steam enthalpy of 2500 kJ/kg. The condensate from a condenser with an enthalpy of 125 kJ/kg enters into the feed heater. The quantity of bled steam is 11200 kg/h. Find the power developed by the turbine. Assume that the water leaving the feed heater is saturated liquid at 3.2 bar and the heater is direct mixing type. Neglect pump work.*

Solution. Arrangement of the components is shown in Fig. 15.16.

At 3.2 bar, $h_{f_2} = 570.9$ kJ/kg.

Consider m kg out of 1 kg is taken to the feed heater (Fig. 15.16).

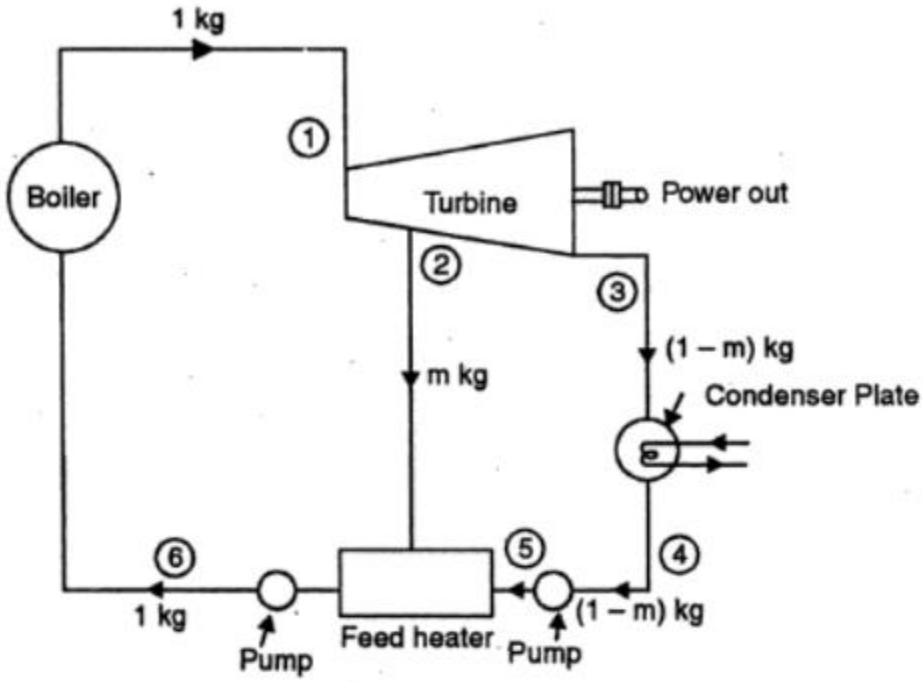


Fig. 15.16

Energy balance for the feed heater is written as :

$$mh_2 + (1 - m)\ h_{f_5} = 1 \times h_{f_2}$$

$$m \times 2100 + (1 - m) \times 125 = 1 \times 570.9$$

$$2100\ m + 125 - 125\ m = 570.9$$

$$1975\ m = 570.9 - 125$$

$\therefore$ $m = 0.226$ kg per kg of steam supplied to the turbine

$\therefore$ Steam supplied to the turbine per hour

$$= \frac{11200}{0.226} = 49557.5 \text{ kg/h}$$

Net work developed per kg of steam

$$= (h_1 - h_2) + (1 - m)\ (h_2 - h_3)$$

$$= (3100 - 2500) + (1 - 0.226)\ (2500 - 2100)$$

$$= 600 + 309.6 = 909.6 \text{ kJ/kg}$$

$\therefore$ **Power developed by the turbine**

$$= 909.6 \times \frac{49557.5}{3600} \text{ kJ/s}$$

$= 12521.5$ **kW. (Ans.)** $\quad (\because\ 1 \text{ kJ/s} = 1 \text{ kW})$

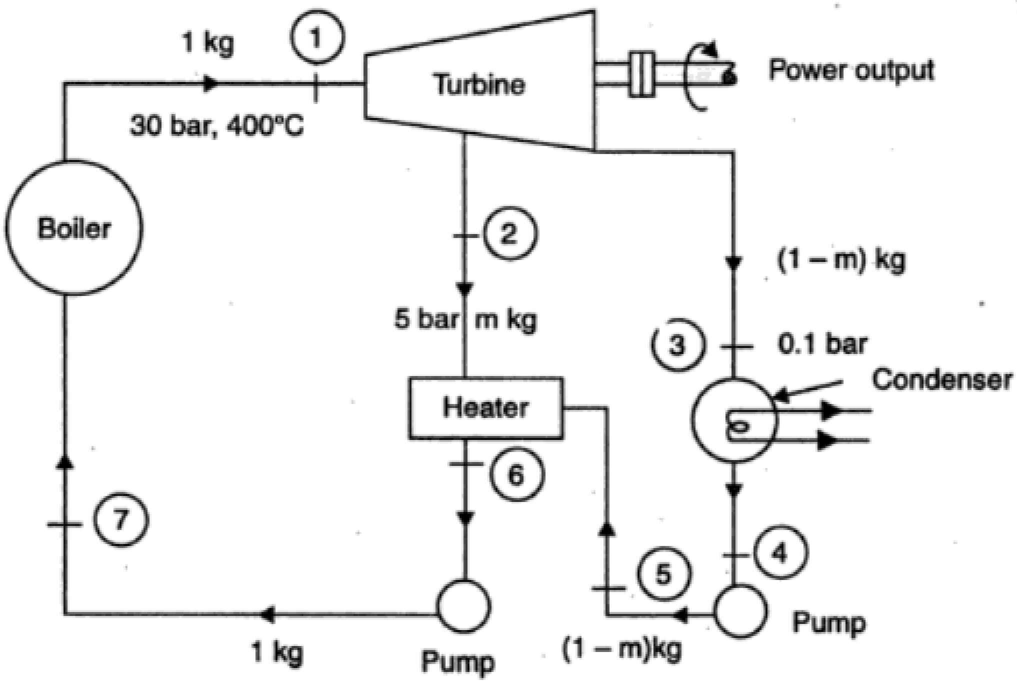
☞**Example 15.13.** *In a single-heater regenerative cycle the steam enters the turbine at 30 bar, 400°C and the exhaust pressure is 0.10 bar. The feed water heater is a direct contact type which operates at 5 bar. Find :*

(*i*) *The efficiency and the steam rate of the cycle.*

(*ii*) *The increase in mean temperature of heat addition, efficiency and steam rate as compared to the Rankine cycle (without regeneration).*

Pump work may be neglected.

Solution. Fig. 15.17 shows the flow, *T-s* and *h-s* diagrams.



(a)

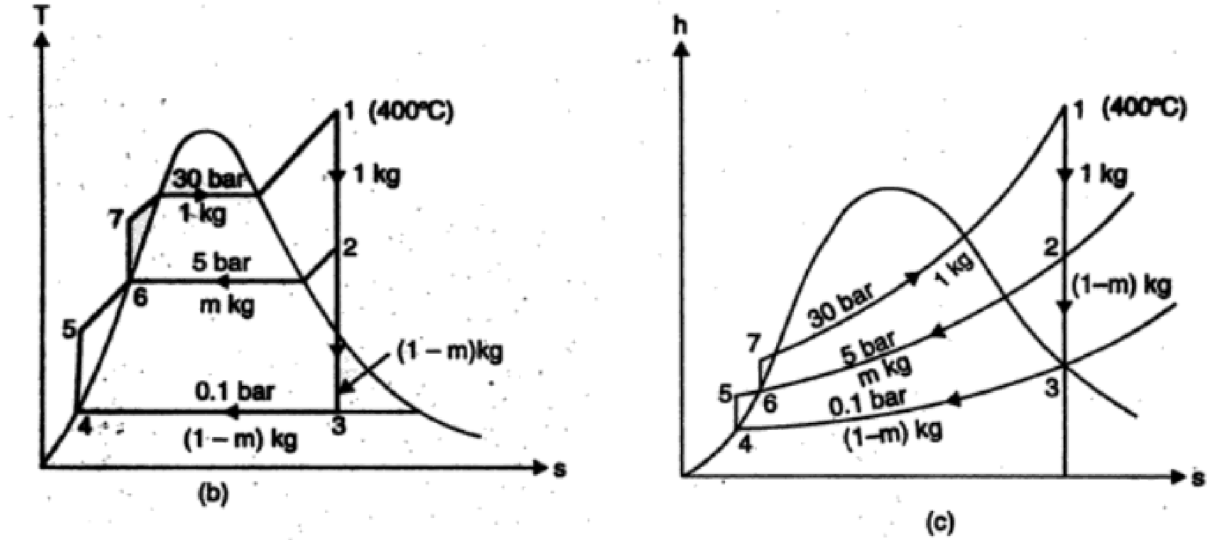


Fig. 15.17

From steam tables :

At 30 bar, 400°C : $h_1 = 3230.9$ kJ/kg, $s_1 = 6.921$ kJ/kg K $= s_2 = s_3$,

At 5 bar : $s_f = 1.8604$, $s_g = 6.8192$ kJ/kg K, $h_f = 640.1$ kJ/kg

Since $s_2 > s_g$, the state 2 must lie in the superheated region. From the table for superheated steam $t_2 = 172$°C, $h_2 = 2796$ kJ/kg.

At 0.1 bar : $s_f = 0.649$, $s_{fg} = 7.501$, $h_f = 191.8$, $h_{fg} = 2392.8$

Now, $s_2 = s_3$

i.e.,
$$6.921 = s_{f_3} + x_3\, s_{fg_3} = 0.649 + x_3 \times 7.501$$

$$\therefore \quad x_3 = \frac{6.921 - 0.649}{7.501} = 0.836$$

$$\therefore \quad h_3 = h_{f_3} + x_3\, h_{fg_3} = 191.8 + 0.836 \times 2392.8 = 2192.2 \text{ kJ/kg}$$

Since pump work is neglected

$$h_{f_4} = 191.8 \text{ kJ/kg} = h_{f_5}$$

$$h_{f_6} = 640.1 \text{ kJ/kg (at 5 bar)} = h_{f_7}$$

Energy balance for heater gives

$$m\,(h_2 - h_{f_6}) = (1 - m)\,(h_{f_6} - h_{f_5})$$

$$m\,(2796 - 640.1) = (1 - m)\,(640.1 - 191.8) = 448.3\,(1 - m)$$

$$2155.9\,m = 448.3 - 448.3\,m$$

$\therefore$ $$m = 0.172 \text{ kg}$$

$\therefore$ **Turbine work,**

$$W_T = (h_1 - h_2) + (1 - m)\,(h_2 - h_3)$$

$$= (3230.9 - 2796) + (1 - 0.172)\,(2796 - 2192.2)$$

$$= 434.9 + 499.9 = 934.8 \text{ kJ/kg}$$

Heat supplied, $$Q_1 = h_1 - h_{f_6} = 3230.9 - 640.1 = 2590.8 \text{ kJ/kg.}$$

(*i*) Efficiency of cycle, η_{cycle} :

$$\eta_{cycle} = \frac{W_T}{Q_1} = \frac{934.8}{2590.8} = \mathbf{0.3608 \text{ or } 36.08\%.} \quad \textbf{(Ans.)}$$

$$\textbf{Steam rate} = \frac{3600}{934.8} = \mathbf{3.85 \text{ kg/kWh.}} \quad \textbf{(Ans.)}$$

(*ii*) $$T_{m_1} = \frac{h_1 - h_{f_7}}{s_1 - s_7} = \frac{2590.8}{6.921 - 1.8604} = 511.9 \text{ K} = 238.9°\text{C.}$$

T_{m_1} (without regeneration)

$$= \frac{h_1 - h_{f_4}}{s_1 - s_4} = \frac{3230.9 - 191.8}{6.921 - 0.649} = \frac{3039.1}{6.272} = 484.5 \text{ K} = 211.5°\text{C.}$$

Increase in T_{m_1} due to regeneration

$$= 238.9 - 211.5 = \mathbf{27.4°C.} \quad \textbf{(Ans.)}$$

W_T (without regeneration)

$$= h_1 - h_3 = 3230.9 - 2192.2 = 1038.7 \text{ kJ/kg}$$

Steam rate without regeneration

$$= \frac{3600}{1038.7} = 3.46 \text{ kg/kWh}$$

$\therefore$ **Increase in steam rate due to regeneration**

$$= 3.85 - 3.46 = \mathbf{0.39 \text{ kg/kWh.}} \quad \textbf{(Ans.)}$$

$$\eta_{cycle} \text{ (without regeneration)} = \frac{h_1 - h_3}{h_1 - h_{f_4}} = \frac{1038.7}{3230.9 - 191.8} = \mathbf{0.3418 \text{ or } 34.18\%.} \quad \textbf{(Ans.)}$$

Increase in cycle efficiency due to regeneration

$$= 36.08 - 34.18 = \mathbf{1.9\%.} \quad \textbf{(Ans.)}$$

$= 36.08 - 34.18 = 1.9\%$. (Ans.)

Example 15.14. *Steam is supplied to a turbine at a pressure of 30 bar and a temperature of 400°C and is expanded adiabatically to a pressure of 0.04 bar. At a stage of turbine where the pressure is 3 bar a connection is made to a surface heater in which the feed water is heated by bled steam to a temperature of 130°C. The condensed steam from the feed heater is cooled in a drain cooler to 27°C. The feed water passes through the drain cooler before entering the feed heater. The cooled drain water combines with the condensate in the well of the condenser.*

Assuming no heat losses in the steam, calculate the following :

(i) Mass of steam used for feed heating per kg of steam entering the turbine ;

(ii) Thermal efficiency of the cycle.

Solution. Refer Fig. 15.18.

From steam tables :

At 3 bar : $t_s = 133.5°C$, $h_f = 561.4$ kJ/kg.

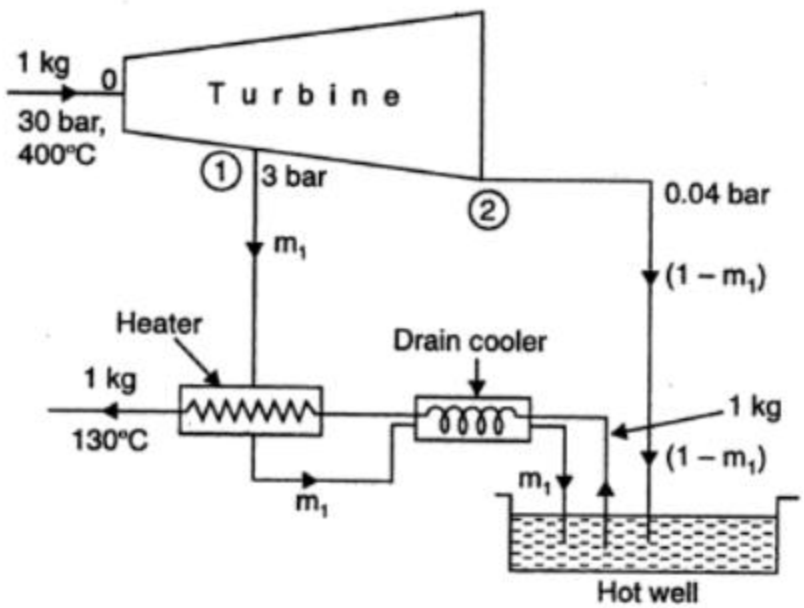
At 0.04 bar : $t_s = 29°C$, $h_f = 121.5$ kJ/kg.

From Mollier chart :

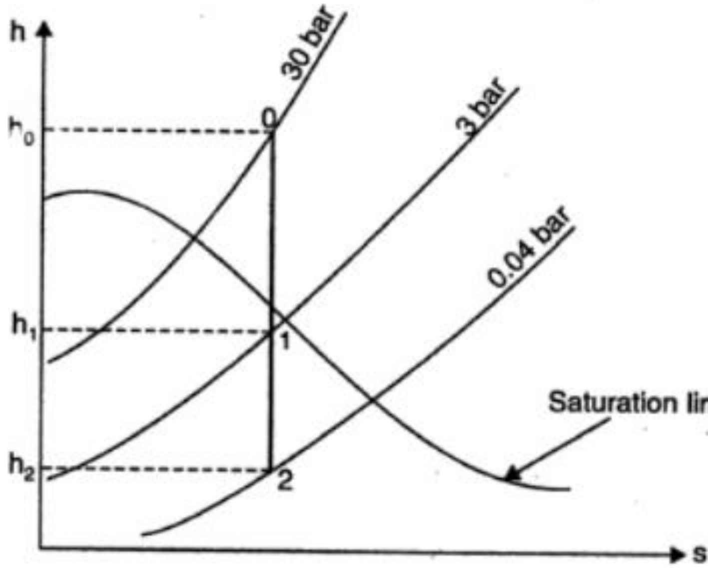
$h_0 = 3231$ **kJ/kg** (at 30 bar, 400°C)

$h_1 = 2700$ **kJ/kg** (at 3 bar)

$h_2 = 2085$ **kJ/kg** (at 0.04 bar).



(a)



(b)

Fig. 15.18

Fig. 15.18

(*i*) **Mass of steam used, m_1 :**

Heat lost by the steam = Heat gained by water.

Taking the feed-heater and drain-cooler combined, we have :

$$m_1 (h_1 - h_{f_2}) = 1 \times 4.186 (130 - 27)$$

$$m_1 (2700 - 121.5) = 4.186 (130 - 27)$$

$$\therefore \qquad \mathbf{m_1} = \frac{4.186 (130 - 27)}{(2700 - 121.5)} = \mathbf{0.1672 \ kg.} \quad \textbf{(Ans.)}$$

(*ii*) **Thermal efficiency of the cycle :**

Work done per kg of steam

$$= 1(h_0 - h_1) + (1 - m_1)(h_1 - h_2)$$

$$= 1(3231 - 2700) + (1 - 0.1672)(2700 - 2085)$$

$$= 1043.17 \text{ kJ/kg}$$

Heat supplied per kg of steam $= h_0 - 1 \times 4.186 \times 130$

$$= 3231 - 544.18 = 2686.82 \text{ kJ/kg.}$$

$$\eta_{\text{Thermal}} = \frac{\text{Work done}}{\text{Heat supplied}} = \frac{1043.17}{2686.82} = \mathbf{0.3882 \ or \ 38.82\%.} \quad \textbf{(Ans.)}$$

15.5. REHEAT CYCLE

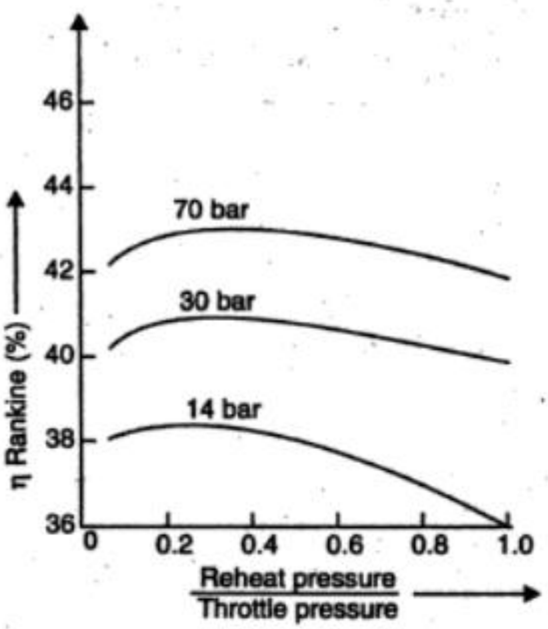
For attaining greater thermal efficiencies when the initial pressure of steam was raised beyond 42 bar it was found that resulting condition of steam after, expansion was increasingly wetter and exceeded in the safe limit of 12 per cent condensation. It, therefore, became necessary to *reheat* the steam after part of expansion was over so that the resulting condition after complete expansion fell within the region of permissible wetness.

The reheating or resuperheating of steam is now universally used when high pressure and temperature steam conditions such as 100 to 250 bar and 500°C to 600°C are employed for throttle. For plants of *still higher pressures and temperatures, a double reheating may be used.*

In actual practice reheat *improves* the cycle efficiency by about 5% for a 85/15 bar cycle. A *second reheat* will give a *much less gain* while the initial cost involved would be so high as to prohibit use of two stage reheat except in case of very high initial throttle conditions. The cost of reheat equipment consisting of boiler, piping and controls may be 5% to 10% more than that of the conventional boilers and this additional expenditure is justified only if gain in thermal efficiency is sufficient to promise a return of this investment. *Usually a plant with a base load capacity of 50000 kW and initial steam pressure of 42 bar would economically justify the extra cost of reheating.*

The improvement in thermal efficiency due to reheat is greatly dependent upon the *reheat pressure* with respect to the original pressure of steam.

Fig. 15.23 shows the reheat pressure selection on cycle efficiency.



Condenser pressure : 12.7 mm Hg
Temperature of throttle and heat : 427°C
Fig. 15.23. Effect of reheat pressure selection on cycle efficiency.

Fig. 15.24 shows a schematic diagram of a theoretical single-stage reheat cycle. The corresponding representation of ideal reheating process on T-s and h-s chart is shown in Fig. 15.22 (*a* and *b*).

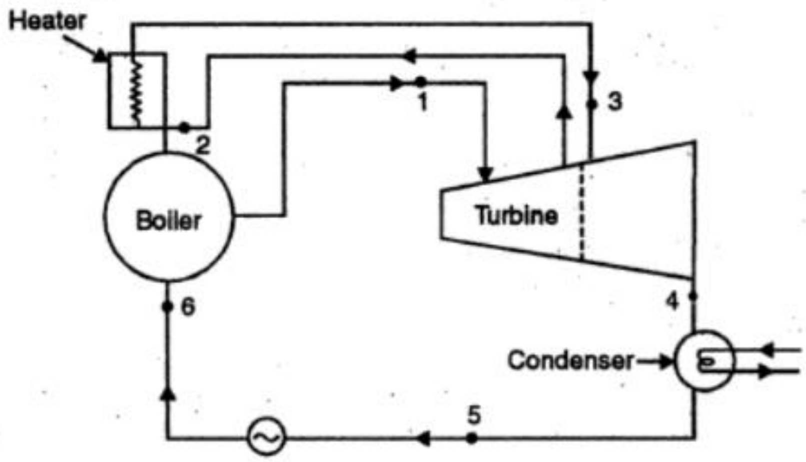


Fig. 15.24. Reheat cycle.

Refer Fig. 15.24, (*a*). 5-1 shows the formation of steam in the boiler. The steam as at state point 1 (*i.e.*, pressure p_1 and temperature T_1) enters the turbine and expands isentropically to a certain pressure p_2 and temperature T_2. From this state point 2 the whole of steam is drawn out of the turbine and is reheated in a reheater to a temperature T_3. (Although there is an *optimum pressure* at which the steam should be removed for reheating, if the highest return is to be obtained, yet, for simplicity, the whole steam is removed from the high pressure exhaust, where the pressure is about *one-fifth* of boiler pressure, and after undergoing a 10% pressure drop, in circulating through the heater, it is returned to intermediate pressure or low pressure turbine). This reheated steam is then readmitted to the turbine where it is expanded to condenser pressure isentropically.

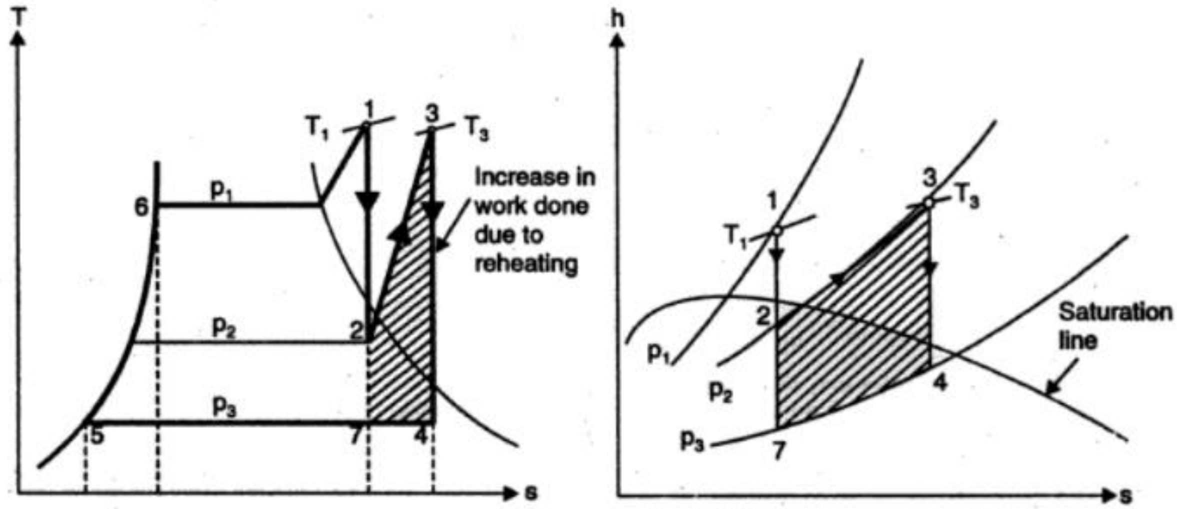


Fig. 15.25. Ideal reheating process on *T-s* and *h-s* chart.

Note. Superheating of steam. The primary object of superheating steam and supplying it to the primemovers is *to avoid too much wetness at the end of expansion*. Use of inadequate degree of superheat in *steam engines would cause greater condensation* in the engine cylinder ; while in case of *turbines* the moisture content of steam would result in *undue blade erosion*. The *maximum wetness in the final condition of steam that may be tolerated without any appreciable harm to the turbine blades is about 12 per cent*. Broadly each *1 per cent* of moisture in steam reduces the efficiency of that part of the turbine in which wet steam passes by 1 per cent to 1.5 per cent and in engines about 2 per cent.

Advantages of superheated steam :

(*i*) Superheating reduces the initial condensation losses in steam engines.

(*ii*) Use of superheated steam results in *improving the plant efficiency* by effecting a *saving in cost of fuel*. This saving may be of the order of 6% to 7% due to first 38°C of superheat and 4% to 5% for next 38°C and so on. This saving results due to the fact that the heat content and consequently the capacity to do work in superheated steam is increased and the quantity of steam required for a given output of power is reduced. Although additional heat has to be added in the boiler there is reduction in the work to be done by the feed pump, the condenser pump and other accessories due to reduction in quantity of steam used. It is estimated that the *quantity of steam may be reduced by 10% to 15% for first 38°C of superheat and somewhat less for the next 38°C of superheat in the case of condensing turbines.*

(*iii*) When a superheater is used in a boiler it helps in *reducing the stack temperatures* by extracting heat from the flue gases before these are passed out of chimney.

Thermal efficiency with '*Reheating*' (neglecting pump work) :

Heat supplied $= (h_1 - h_{f_4}) + (h_3 - h_2)$

Heat rejected $= h_4 - h_{f_4}$

Work done by the turbine = Heat supplied – heat rejected

$$= (h_1 - h_{f_4}) + (h_3 - h_2) - (h_4 - h_{f_4})$$

$$= (h_1 - h_2) + (h_3 - h_4)$$

Thus, theoretical thermal efficiency of reheat cycle is

$$\eta_{\text{thermal}} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_4}) + (h_3 - h_2)} \qquad \text{...(15.11)}$$

If pump work, $W_p = \dfrac{v_f\,(p_1 - p_b)}{1000}$ kJ/kg is considered, the thermal efficiency is given by :

$$\eta_{\text{thermal}} = \frac{[(h_1 - h_4) + (h_3 - h_4)] - W_p}{[(h_1 - h_{f_4}) + (h_3 - h_2)] - W_p} \qquad \text{...(15.12)}$$

W_p is usually small and *neglected.*

Thermal efficiency *without reheating* is

$$\eta_{\text{thermal}} = \frac{h_1 - h_7}{h_1 - h_{f_4}} \quad (\because \quad h_{f_4} = h_{f_7}) \qquad \text{...(15.13)}$$

Advantages of 'Reheating' :

1. There is an increased output of the turbine.
2. Erosion and corrosion problems in the steam turbine are eliminated/avoided.
3. There is an improvement in the thermal efficiency of the turbines.
4. Final dryness fraction of steam is improved.
5. There is an increase in the nozzle and blade efficiencies.

Disadvantages :

1. Reheating requires more maintenance.

2. The increase in thermal efficiency is not appreciable in comparison to the expenditure incurred in reheating.

Example 15.18. *Steam at a pressure of 15 bar and 250°C is expanded through a turbine at first to a pressure of 4 bar. It is then reheated at constant pressure to the initial temperature of 250°C and is finally expanded to 0.1 bar. Using Mollier chart, estimate the work done per kg of steam flowing through the turbine and amount of heat supplied during the process of reheat. Compare the work output when the expansion is direct from 15 bar to 0.1 bar without any reheat. Assume all expansion processes to be isentropic.*

Solution. Refer Fig. 15.26.

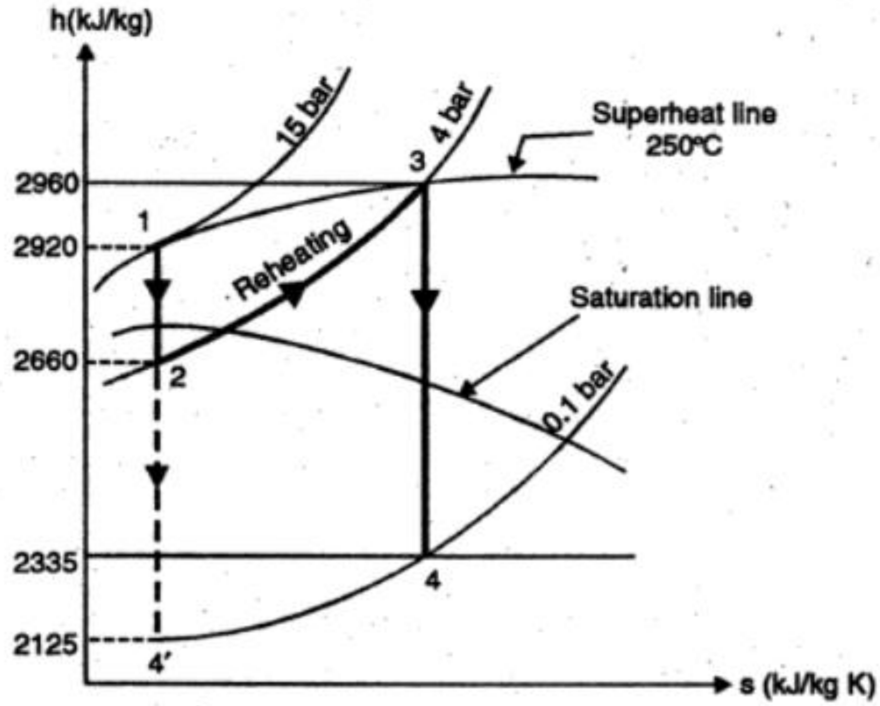


Fig. 15.26

Fig. 15.26

Pressure, $p_1 = 15$ bar ;

$p_2 = 4$ bar ;

$p_4 = 0.1$ bar.

Work done per kg of steam,

$$W = \text{Total heat drop}$$

$$= [(h_1 - h_2) + (h_3 - h_4)] \text{ kJ/kg} \qquad ...(i)$$

Amount of heat supplied during process of reheat,

$$h_{reheat} = (h_3 - h_2) \text{ kJ/kg} \quad ...(ii)$$

From Mollier diagram or *h-s* chart,

$$h_1 = 2920 \text{ kJ/kg}, \; h_4 = 2660 \text{ kJ/kg}$$
$$h_3 = 2960 \text{ kJ/kg}, \; h_4 = 2335 \text{ kJ/kg}$$

Now, by putting the values in eqns. (*i*) and (*ii*), we get

$$W = (2920 - 2660) + (2960 - 2335)$$
$$= \mathbf{885 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

Hence *work done per kg of steam* = **885 kJ/kg. (Ans.)**

Amount of heat supplied during reheat,

$$h_{reheat} = (2960 - 2660) = \mathbf{300 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

If the expansion would have been continuous without reheating *i.e.*, 1 to 4′, the work output is given by

$$W_1 = h_1 - h_4'$$

From Mollier diagram,

$$h_{4'} = 2125 \text{ kJ/kg}$$

$$\therefore \quad W_1 = 2920 - 2125 = \mathbf{795 \text{ kJ/kg.}} \quad \textbf{(Ans.)}$$

Example 15.19. *A steam power plant operates on a theoretical reheat cycle. Steam at boiler at 150 bar, 550°C expands through the high pressure turbine. It is reheated at a constant pressure of 40 bar to 550°C and expands through the low pressure turbine to a condenser at 0.1 bar. Draw T-s and h-s diagrams. Find :*

(*i*) *Quality of steam at turbine exhaust ;* (*ii*) *Cycle efficiency ;*

(*iii*) *Steam rate in kg/kWh.* **(AMIE Summer, 1999)**

Solution. Refer Fig. 15.27 and 15.28

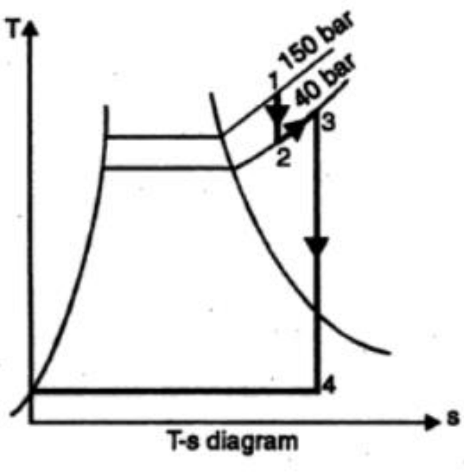


Fig. 15.27

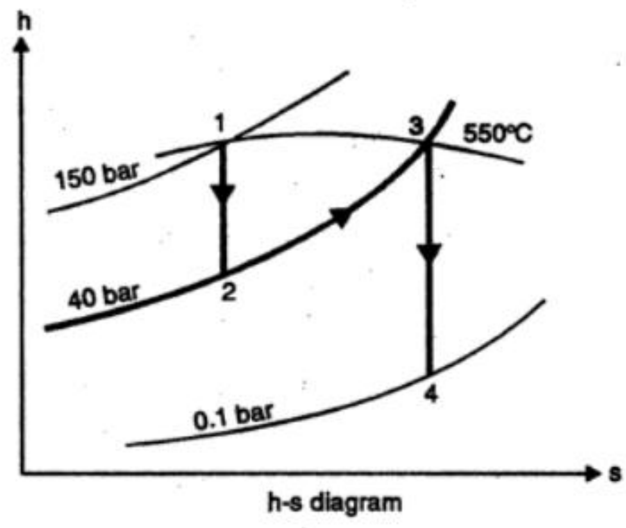


Fig. 15.28

From Mollier diagram (*h-s* diagram) :

$$h_1 = 3450 \text{ kJ/kg} ;\ h_2 = 3050 \text{ kJ/kg} ;\ h_3 = 3560 \text{ kJ/kg} ;\ h_4 = 2300 \text{ kJ/kg}$$

$$h_{f_4} \text{ (from steam tables, at 0.1 bar)} = 191.8 \text{ kJ/kg}$$

(*i*) **Quality of steam at turbine exhaust, x_4 :**

$$x_4 = 0.88 \text{ (From Mollier diagram)}$$

(*ii*) **Cycle efficiency, η_{cycle} :**

$$\eta_{cycle} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_4}) + (h_3 - h_2)}$$

$$= \frac{(3450 - 3050) + (3560 - 2300)}{(3450 - 191.8) + (3560 - 3050)} = \frac{1660}{3768.2} = \mathbf{0.4405 \text{ or } 44.05\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Steam rate in kg/kWh :**

$$\text{Steam rate} = \frac{3600}{(h_1 - h_2) + (h_3 - h_4)} = \frac{3600}{(3450 - 3050) + (3560 - 2300)}$$

$$= \frac{3600}{1660} = \mathbf{2.17 \text{ kg/kWh.}} \quad \textbf{(Ans.)}$$

Boilers

Steam is extensively used for various applications such as power production, industrial processes, work interaction, heating etc. With the increasing use of steam in different engineering systems the steam generation technology has also undergone various developments starting from 100 B.C. when Hero of Alexandria invented a combined reaction turbine and boiler. Boiler, also called steam generator is the engineering device which generates steam at constant pressure. It is a closed vessel, generally made of steel in which vaporization of water takes place. Heat required for vaporization may be provided by the combustion of fuel in furnace, electricity, nuclear reactor, hot exhaust gases, solar radiations etc. Earlier boilers were closed vessels made from sheets of wrought iron which were lapped, riveted and formed into shapes of simple sphere type or complex sections such as the one shown in Fig. 1.1. It is the 'Wagon boiler' of Watt developed in 1788.

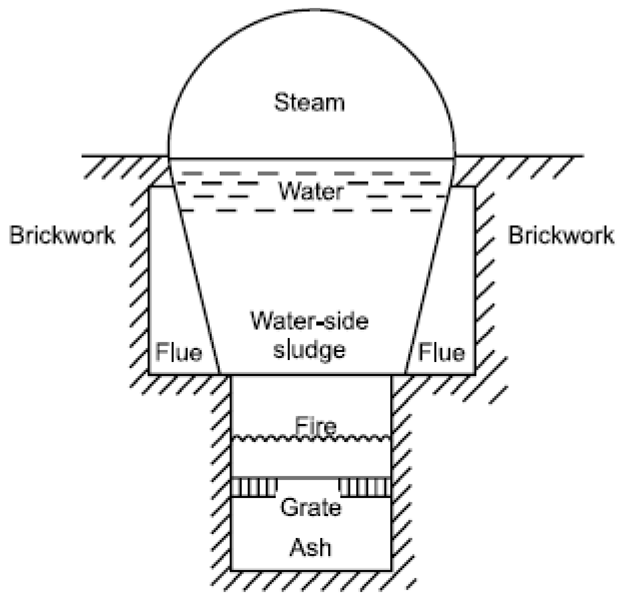


Fig. 1.1 *Wagon boiler of Watt*, (1788)

According to A.S.M.E. (American Society of Mechanical Engineers, U.S.A.) code a boiler is defined as a combination of apparatus for producing, furnishing or recovering heat together with the apparatus for transferring the heat so made available to water which could be heated and vaporized to steam form.

Types of Boilers

Boilers are of many types. Depending upon their features they can be classified as given under:

(a) Based upon the orientation/axis of the shell: According to the axis of shell boiler can be classified as vertical boiler and horizontal boiler.

(i) Vertical boiler has its shell vertical.

(ii) Horizontal boiler has its shell horizontal.

(iii) Inclined boiler has its shell inclined.

(b) Based upon utility of boiler: Boilers can be classified as

(i) Stationery boiler, such boilers are stationery and are extensively used in power plants, industrial processes, heating etc.

(ii) Portable boiler, such boilers are portable and are of small size. These can be of the following types,

Locomotive boilers, which are exclusively used in locomotives. Marine boiler, which are used for marine applications.

(c) Based on type of firing employed: According to the nature of heat addition process boilers can be classified as,

(i) Externally fired boilers, in which heat addition is done externally i.e. furnace is outside the boiler unit. Such as Lancashire boiler, Locomotive boiler etc.

(ii) Internally fired boilers, in which heat addition is done internally i.e. furnace is within the boiler unit. Such as Cochran boiler, Bobcock Wilcox boiler etc.

(d) Based upon the tube content: Based on the fluid inside the tubes, boilers can be,

(i) Fire tube boilers, such boilers have the hot gases inside the tube and water is outside surrounding them. Examples for these boilers are, Cornish boiler, Cochran boiler, Lancashire boiler, Locomotive boiler etc.

(ii) Water tube boilers, such boilers have water flowing inside the tubes and hot gases surround them. Examples for such boilers are Babcock-Wilcox boiler, Stirling boiler, La-Mont boiler, Benson boiler etc.

(e) Based on type of fuel used: According to the type of fuel used the boilers can be,

(i) Solid fuel fired boilers, such as coal fired boilers etc.

(ii) Liquid fuel fired boilers, such as oil fired boilers etc.

(iii) Gas fired boilers, such as natural gas fired boilers etc.

(f) Based on circulation: According to the flow of water and steam within the boiler circuit the boilers may be of following types,

(i) Natural circulation boilers, in which the circulation of water/steam is caused by the density difference which is due to the temperature variation.

(ii) Forced circulation boilers, in which the circulation of water/steam is caused by a pump i.e. externally, assisted circulation.

(g) Based on extent of firing: According to the extent of firing the boilers may be,

(i) Fired boilers, in which heat is provided by fuel firing.

(iii) Unfired boilers, in which heat is provided by some other source except fuel firing such as hot flue gases etc.

(iv) Supplementary fired boilers, in which a portion of heat is provided by fuel firing and remaining by some other source.

FIRE-TUBE BOILERS

Since the late eighteenth century, fire-tube boilers have been used in various forms to produce steam for industrial purposes and also for hauling railway locomotives and river launches. They are no longer used in utility power plants and steam locomotives have also mostly disappeared. However, they are still often used in industrial plants to produce saturated steam at the upper limits of about 18 bar pressure and 6.2 kg/s steaming capacity.

For small steam requirements, fire-tube boilers are suitable. They have certain inherent advantages like (1) low first cost, (2) reliability in operation, (3) need of only unskilled labour, (4) less draught required, and (5) quick response to load changes.

A fire-tube boiler is so named because the products of combustion pass through its tubes or flues, which are surrounded by water. They may be either:

(a) Externally fired (e.g. locomotive type boilers, Lancashire boilers, horizontal return tubular (HRT) boiler etc.), or

(b) Internally fired (e.g. Scotch-marine boilers, package boilers etc.).

Figure 6.1 shows a typical externally fired fire-tube boiler in which the furnace is outside the boiler shell. Coal is entered manually by shovels on to the grate by opening the fire-door. The products of combustion flow through the tubes which are immersed in the shell containing water. A fusible plug made up of a low melting point alloy (lead-based) is installed on the roof of the crown in the furnace. If the water-level in the shell falls below a certain level, the fusible plug melts due to overheating and water pours down through the hole formed and puts out the fire. There is a spring-loaded safety-valve provided to keep the boiler pressure within the safety limit. The spring is set in such a way that the upward thrust of steam against the lid is balanced by the downward thrust of the spring. If the operating steam pressure exceeds this value, the upward thrust of steam will then be greater than the downward spring thrust and the difference will force open the lid upward, as a result of which steam will be released with a hissing sound, the steam pressure inside the shell will go down till the lid is forced down to be back on its seat.

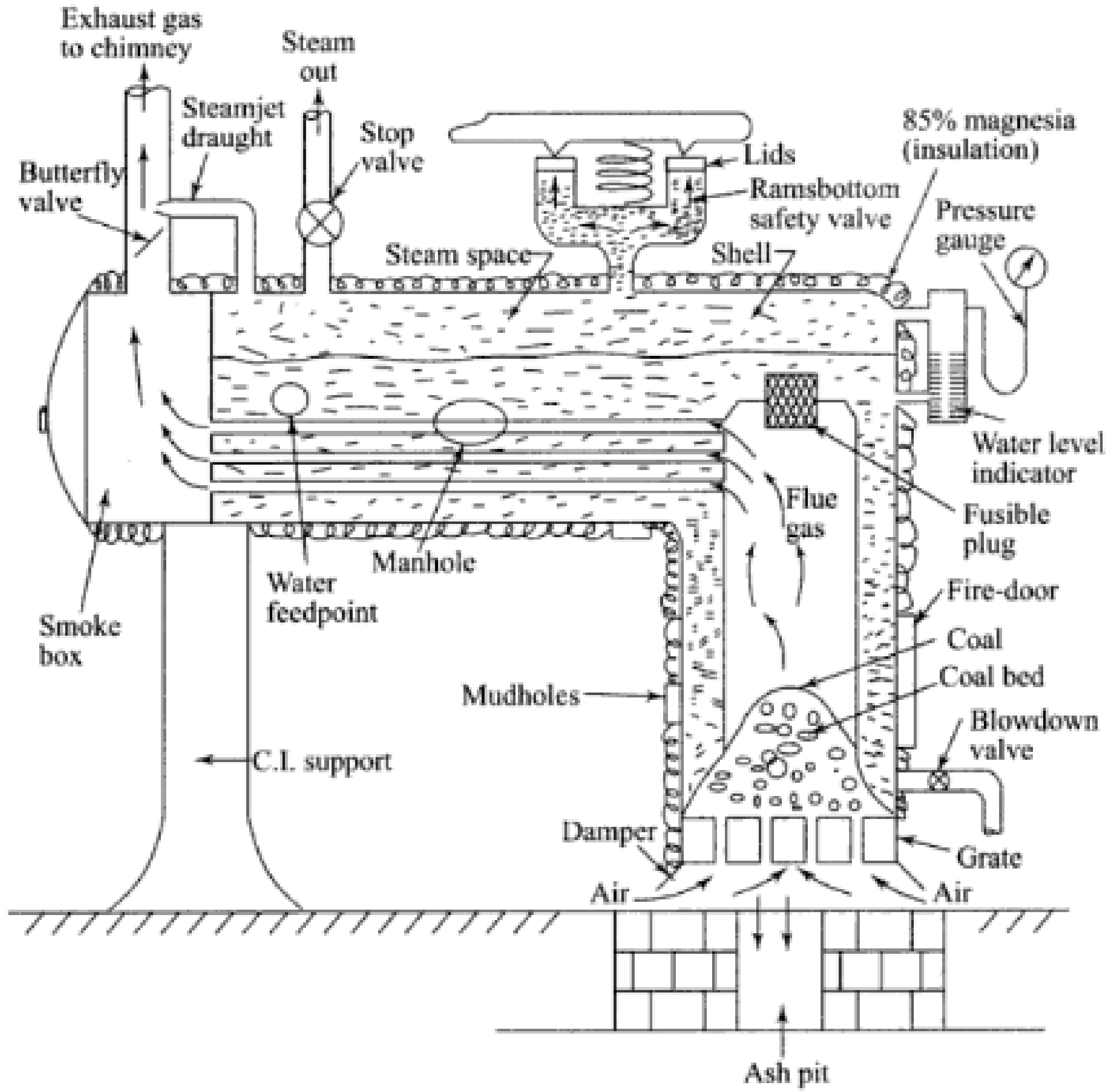


Fig. 6.1 *A typical fire-tube boiler*

As the hot flue gases flow through the tubes, heat is transferred from gas to water all along the length. The gas is cooled and the water is heated till there is nucleate boiling around the tubes and steam is formed. Steam is taken out at the required rate by opening the main stop valve. Auxiliary steam may also be taken to operate a steam jet water injector to feed water into the shell. The shell is insulated all around by asbestos and 85% magnesia to reduce heat loss to the surroundings. Air flow from below the grate is regulated by operating dampers according to the requirement of combustion. An elliptic manhole is provided for a man to go in to do cleaning or repair as the need arises.

WATER-TUBE BOILERS

Water-tube boilers were developed to permit increases in boiler pressure and capacity with reasonable metal stresses. As mentioned above, with higher steam pressures and capacities, fire-tube boilers would need large-diameter shells, and with such large diameters, the shells would have to operate under such extreme pressure and temperature stresses that the thicknesses would become very large. They are also subjected to large scale deposit and susceptible to boiler explosions, and become very costly.

The water-tube boiler, where water flows through the tubes and flue gases flow outside them, puts the pressure in the tubes and the relatively small-diameter drums, which are capable of withstanding extreme pressures of the modern steam generator.

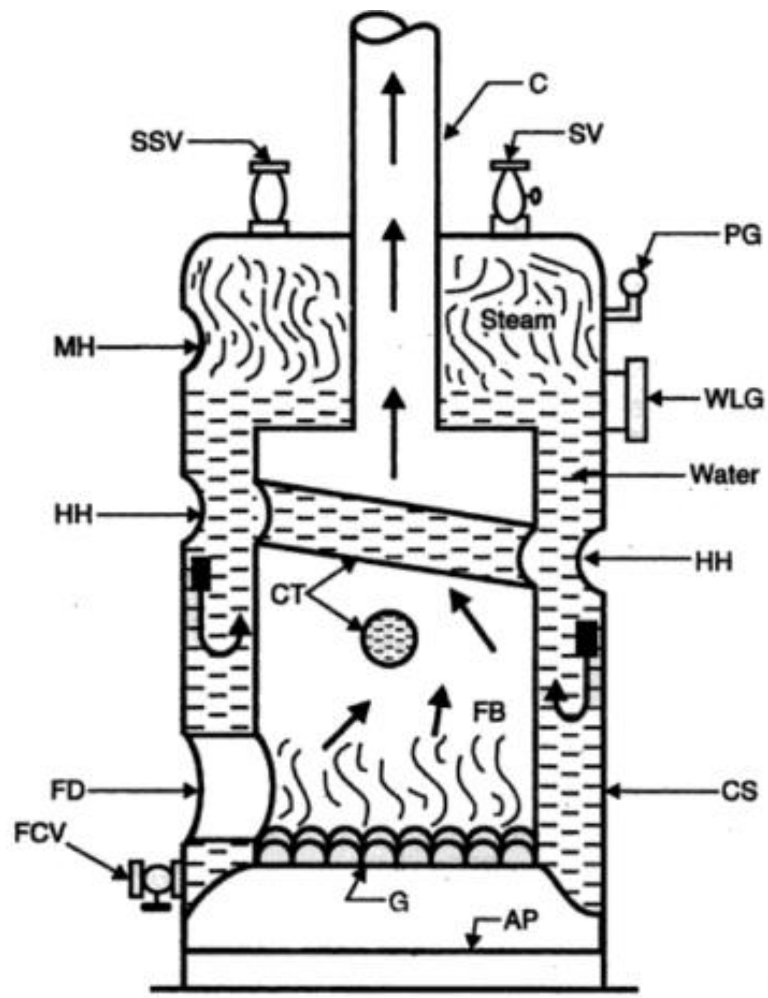
FIRE TUBE BOILERS

The various fire tube boilers are described as follows :

Simple Vertical Boiler

Refer Fig. 11.1. It consists of a cylindrical shell, the greater portion of which is full of water (which surrounds the fire box also) and remaining is the steam space. At the bottom of the fire box is grate on which fuel is burnt and the ash from it falls in the ash pit.

The fire box is provided with two cross tubes. This increases the heating surface and the circulation of water. The cross tubes are fitted inclined. This ensures efficient circulation of water. At the ends of each cross tube are provided hand holes to give access for cleaning these tubes. The combustion gases after heating the water and thus converting it into steam escape to the atmosphere through the chimney. Man hole, is provided to clean the interior of the boiler and exterior of the combustion chamber and chimney. The various mountings shown in Fig. 11.1 are (*i*) Pressure gauge, (*ii*) Water level gauge or indicator, (*iii*) Safety valve, (*iv*) Steam stop valve, (*v*) Feed check valve, and (*vi*) Man hole.



CS = Cylindrical shell	C = Chimney
MH = Man hole	HH = Hand hole
CT = Cross tubes	FD = Fire door
G = Grate	FB = Fire box
PG = Pressure gauge	AP = Ash pit
SV = Safety valve	SSV = Steam stop valve
WLG = Water level gauge	FCV = Feed check valve

Fig. 11.1. Simple vertical boiler.

The rate of production in such a boiler normally does not exceed 2500 kg/hr and pressure is normally limited to 7.5 to 10 bar.

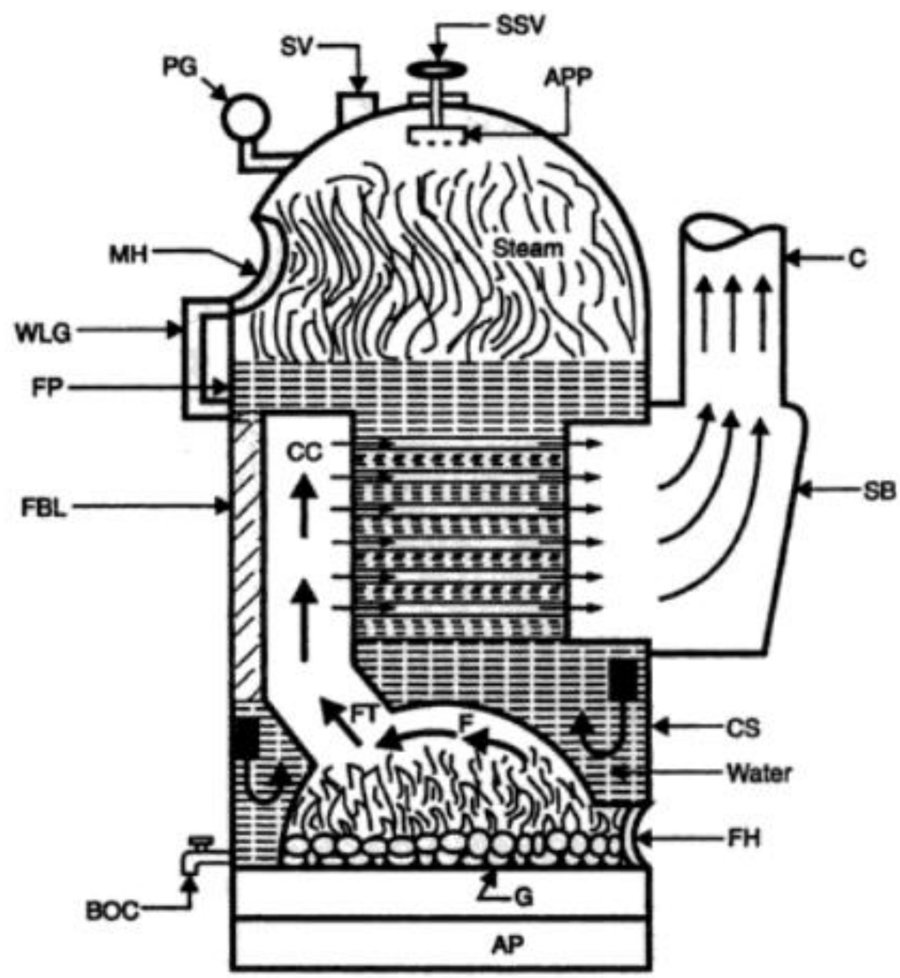
A simple vertical boiler is self-contained and can be transported easily.

11.7.2. Cochran Boiler

It is *one of the best* types of vertical multi-tubular boiler, and has a number of horizontal fire tubes.

Dimensions, working pressure, capacity, heating surface and efficiency are given below :

Shell diameter	...... 2.75 m
Height	...... 5.79 m
Working pressure	...... 6.5 bar (max. pressure = 15 bar)
Steam capacity	...... 3500 kg/hr (max. capacity = 4000 kg/hr)
Heating surface	...... 120 m^2
Efficiency	...... 70 to 75% (depending on the fuel used)



CS = Cylindrical shell	*FT* = Flue tube
CC = Combustion chamber	*SB* = Smoke box
FBL = Fir brick lining	*C* = Chimney
F = Furnace (dome shaped)	*FH* = Fire hole
BOC = Blow off cock	*G* = Grate
SSV = Steam stop valve	*AP* = Ash pit
APP = Antipriming pipe	*SV* = Safety valve
PG = Pressure gauge	*MH* = Man hole
	WLG = Water level gauge

Fig. 11.2. Cochran boiler.

Cochran boiler consists of a cylindrical shell with a dome shaped top where the space is provided for steam. The furnace is one piece construction and is seamless. Its crown has a hemi-spherical shape and thus provides maximum volume of space. The fuel is burnt on the grate and ash is collected and disposed of from ash pit. The gases of combustion produced by burning of fuel enter the combustion chamber through the flue tube and strike against fire brick lining which directs them to pass through number of horizontal tubes, being surrounded by water. After which the gases escape to the atmosphere through smoke box and chimney. A number of hand-holes are provided around the outer shell for cleaning purposes.

The various boiler mountings shown in the Fig. 6.2 are : (*i*) Water level gauge, (*ii*) Safety valve, (*iii*) Steam stop valve, (*iv*) Blow off cock, (*v*) Manhole and, (*vi*) Pressure gauge.

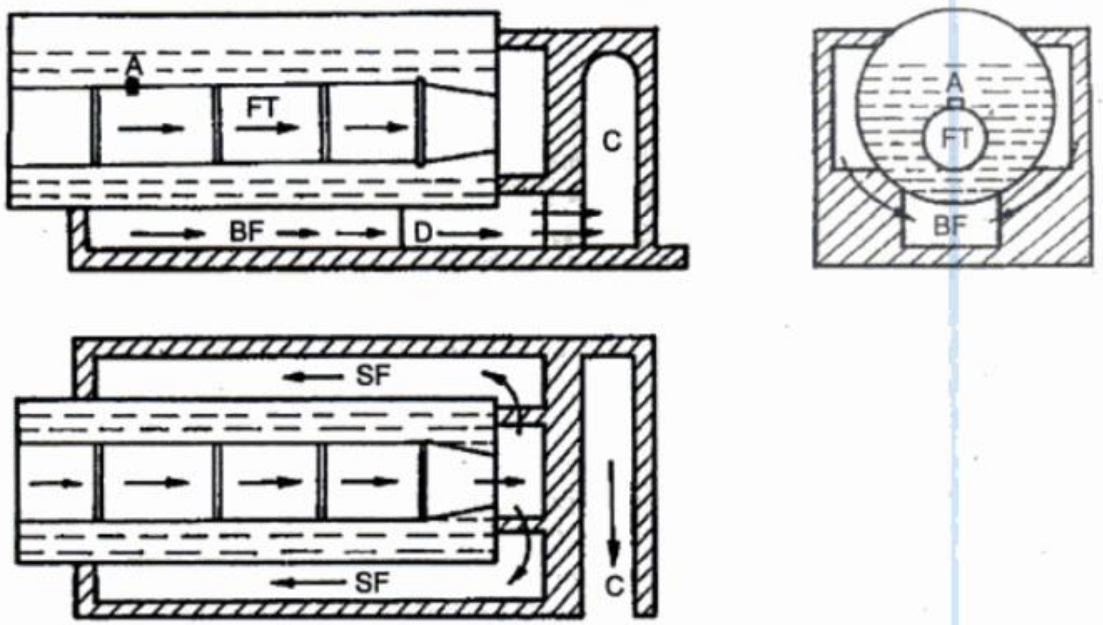
The path of combustion of gases and circulation of water are shown by arrows in Fig. 6.2.

6.7.3. Cornish Boiler

This form of boiler was first adopted by Trevithick, the Cornish engineer, at the time of introduction of high-pressure steam to the early Cornish engine, and is still used.

The specifications of Cornish boiler are given below :

No. of flue tubes	...... One
Diameter of the shell	...... 1.25 to 1.75 m
Length of the shell	...... 4 to 7 m
Pressure of the steam	...... 10.5 bar
Steam capacity	...... 6500 kg/h.



A = Fusible plug
BF = Bottom flue
D = Damper
FT = Flue tube
SF = Side flue
C = Passage to chimney

Fig. 6.3. Cornish boiler.

Refer Fig. 6.3. It consists of a cylindrical shell with flat ends through which passes a smaller flue tube containing the furnace. The products of combustion pass from the fire grate forward over the brick work bridge to the end of the furnace tube ; they then return by the two side flues to the front end of the boiler, and again pass to the back end of a flue along the bottom of the boiler to the chimney.

The various boiler mountings which are used on this boiler are : (*i*) Steam stop valve, (*ii*) Pressure gauge, (*iii*) Water gauge, (*iv*) Fusible plug, (*v*) Blow off cock, (*vi*) High steam low water safety valve, (*vii*) Feed check valve and (*viii*) Manhole.

The advantage possessed by this type of boiler is that the sediment contained in the water falls to the bottom, where the plates are not brought into contact with the hottest portion of the furnace gases. The reason for carrying the product of combustion first through the side flues, and lastly through the bottom flue, is because the gases, having parted with much of their heat by the time they reach the bottom flue, are less liable to unduly heat the plates in the bottom of the boiler, where the sediment may have collected.

6.7.4. Lancashire Boiler

This boiler is *reliable*, has *simplicity of design*, *ease of operation* and *less operating and maintenance costs*. It is commonly used in *sugar-mills* and *textile industries* where alongwith the power steam and steam for the process work is also needed. In addition this boiler is used where larger reserve of water and steam are needed.

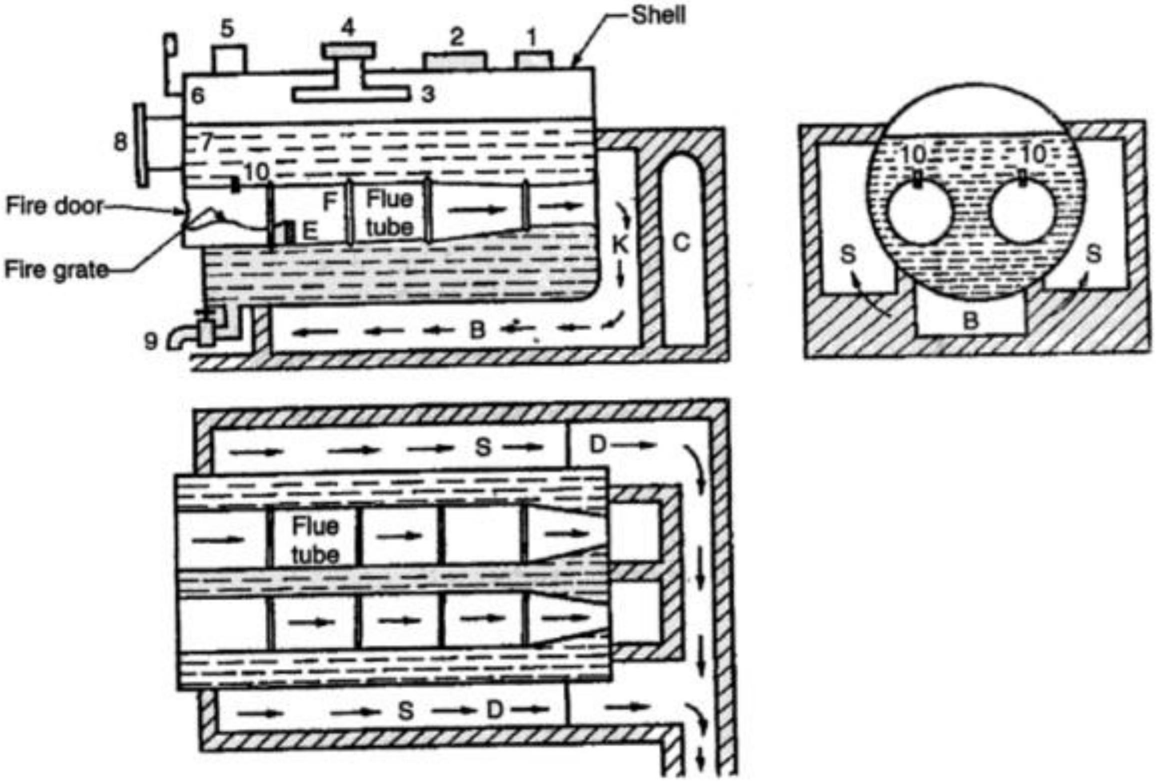
The specifications of Lancashire boiler are given below :

Diameter of the shell	...... 2 to 3 m
Length of the shell	...... 7 to 9 m
Maximum working pressure	...... 16 bar
Steam capacity	...... 9000 kg/h
Efficiency	...... 50 to 70%

Refer Fig. 6.4. The Lancashire boiler consists of a cylindrical shell inside which two large tubes are placed. The shell is constructed with several *rings* of cylindrical from and it is placed horizontally over a brickwork which forms several channels for the flow of hot gases. These two tubes are also constructed with several rings of cylindrical form. They pass from one and of the shell to the other and are covered with water. The furnace is placed at the front end of each tube and they are known as furnace tubes. The coal is introduced through the fire hole into the grate. There is low brickwork fire bridge at the back of the gate to prevent the entry of the burning coal and ashes into the interior of the furnace tubes.

The combustion products from the grate pass upto the back end of the furnace tubes and then in downward direction. Thereafter they move through the bottom channel or bottom flue upto the front end of the boiler where they are divided and pass upto the side flues. Now they move along the two side flues and come to the chimney flue from where they lead to the chimney. To control the flow of hot gases to the chimney, dampers (in the form of sliding doors) are provided. As a result the flow of air to the grate can be controlled. The various mountings used on the boiler are shown in Fig. 6.4.

Note. In Cornish and Lancashire boilers, conical shaped cross tubes known as galloway tubes (not shown) may be fitted inside the furnace tubes to increase their heating surfaces and circulation of water. But these tubes have now become absolete for their considerable cost of fitting. Moreover, they cool the furnace gases and retard combustion.



B = Bottom flue
C = Chimney
D = Dampers
E = Fire-bridge
F = Flue tube
K = Main flue
S = Side flue

1. High steam low water safety valve
2. Main hole
3. Antipriming pipe
4. Steam stop valve
5. Safety valve
6. Pressure gauge
7. Feed check valve
8. Water gauge
9. Blow down cock
10. Fusible plug

Fig. 6.4. Lancashire boiler.

6.7.5. Locomotive Boiler

It is mainly employed in locomotives though it may also be used as a stationary boiler. It is compact and its capacity for steam production is quite high for its size as it can raise large quantity of steam rapidly.

Dimensions and the specifications of the locomotive boilers (made at Chitranjan works in India) are given below :

Barrel diameter	...... 2.095 m
Length of the barrel	...... 5.206 m
Size of the tubes (superheater)	...... 14 cm
No. of superheater tubes	 38
Size of ordinary tubes	...... 5.72 cm
No. of ordinary tubes	 116
Steam capacity	...... 9000 kg/h
Working pressure	...... 14 bar
Grate Area	...... 4.27 m^2
Coal burnt/hr	...... 1600 kg
Heating surface	...... 271 m^2
Efficiency	...... 70%

Refer Fig. 6.5. The locomotive boiler consists of a cylindrical barrel with a rectangular fire box at one end and a smoke box at the other end. The coal is introduced through the fire hole into the grate which is placed at the bottom of the fire box. The hot gases which are generated due to burning of the coal are deflected by an arch of fire bricks, so that walls of the fire box may be heated properly. The fire box is entirely surrounded by water except for the fire hole and the ash pit which is situated below the fire box which is fitted with dampers at its front and back ends. The dampers control the flow of air to the grate. The hot gases pass from the fire box to the smoke box through a series of fire tubes and then they are discharged into the atmosphere through the chimney. The fire tubes are placed inside the barrel. Some of these tube are of larger diameter and the others of smaller diameter. The superheater tubes are placed inside the fire tubes of larger diameter. The heat of the hot gases is transmitted into the water through the heating surface of the fire tubes. The steam generated is collected over the water surface.

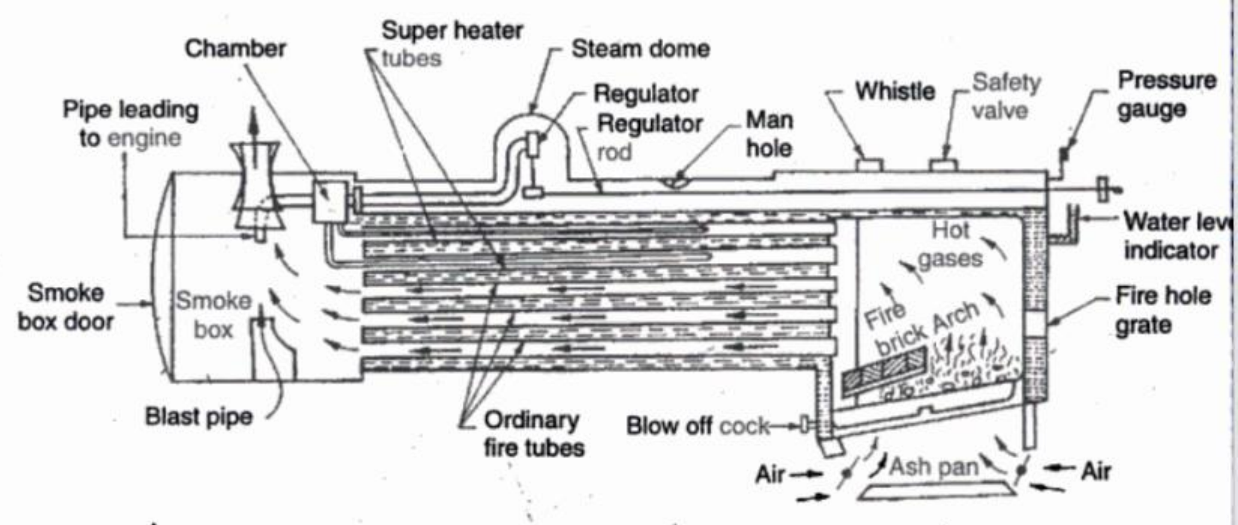


Fig. 6.5. Locomotive boiler.

A dome shaped chamber known as *steam dome* is fitted on the upper part of the barrel, from where the steam flows through a steam pipe into the chamber. The flow of steam is regulated by means of a regulator. From the chamber it passes through the superheater tubes and returns to the superheated steam chamber (not shown) from which it is led to the cylinders through the pipes, one to each cylinder.

In this boiler natural draught cannot be obtained because it requires a very high chimney which cannot be provided on a locomotive boiler since it has to run on rails. Thus some artificial arrangement has to be used to produce a correct draught. As such the draught here is *produced by exhaust steam* from the cylinder which is discharged through the blast pipe to the chimney. When the locomotive is standing and no exhaust steam is available from the engine fresh steam from the boiler is used for the purpose.

The various boiler mountings include :

Safety valves, pressure gauge, water level indicator, fusible plug, man hole, blow off cock and feed check valve.

A locomotive boiler entails the following *merits* and *demerits* :

Merits :

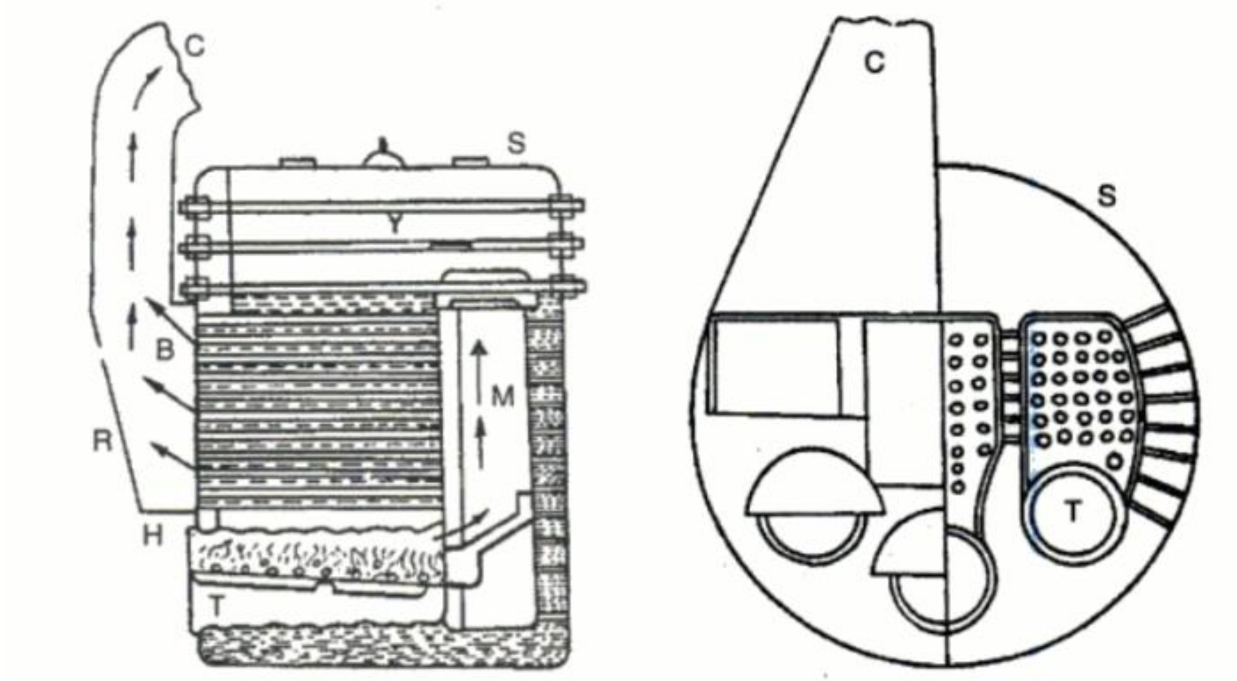
1. High steam capacity.
2. Low cost of construction.
3. Portability.
4. Low installation cost.
5. Compact.

Demerits :

1. There are chances to corrosion and scale formation in the water legs due to the accumulation of sediments and the mud particles.
2. It is difficult to clean some water spaces.
3. Large flat surfaces need bracing.
4. It cannot carry high overloads without being damaged by overheating.
5. There are practical constructional limits for pressure and capacity which do not meet requirements.

6.7.6. Scotch Boiler

The scotch type marine boiler is probably the *most popular* boiler for steaming capacities upto about 1000 kg/h and pressure of about 17 bar. It is of compact size and occupies small floor space.



T = Furnace tube
H = Fire hole
M = Combustion chamber
B = Smoke box
R = Smoke box door
C = Chimney
S = Shell
Y = Bar stays

Fig. 6.6. Scotch boiler.

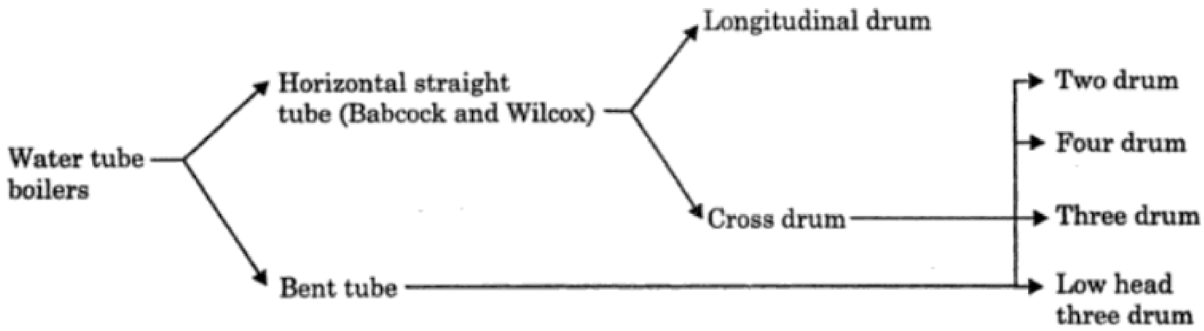
Fig. 6.6 shows a single ended scotch type marine boiler. It consists of a cylindrical shell in which are incorporated one to four cylindrical, corrugated steel furnaces. The furnaces are internally fired and surrounded by water. A combustion chamber is located at the back end of the furnace and is also surrounded by water. Usually each furnace has its own combustion chamber. A nest of fire tubes run from the front tube plate to the back tube plate.

The hot gases produced due to burning of fuel move to the combustion chambers (by means of the draught). Then they travel to the smoke box through the fire tubes and finally leave the boiler *via* uptake and the chimney.

In a double ended scotch boiler furnaces are provided at each end. They look like single ended boilers placed back to back. A double ended boiler for same evaporation capacity, is cheaper and occupies less space as compared to single ended boiler.

WATER TUBE BOILERS

The types of water tube boilers are given below :



6.8.1. Babcock and Wilcox Water-tube Boiler

The water-tube boilers are used exclusively, when pressure above 10 bar and capacity in excess of 7000 kg of steam per hour is required. Babcock and Wilcox water-tube boiler is an example of horizontal straight tube boiler and may be designed for stationary or marine purposes.

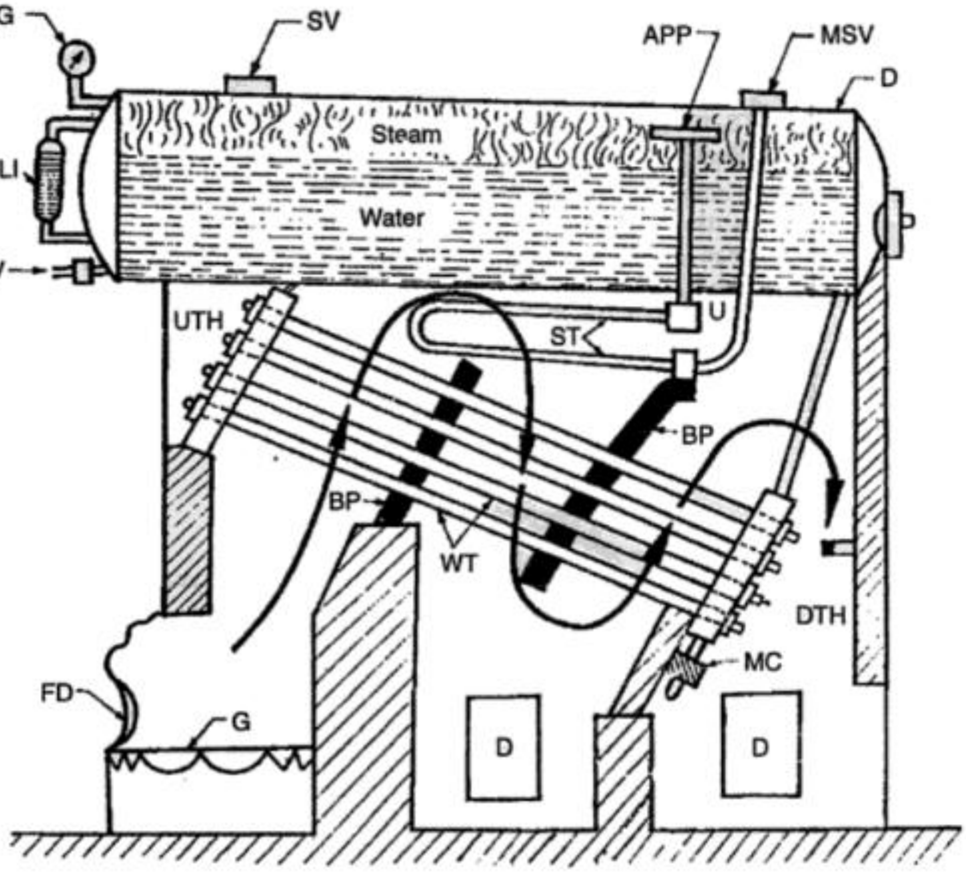
The particulars (dimensions, capacity etc.) relating to this boiler are given below :

Diameter of the drum	1.22 to 1.83 m
Length	6.096 to 9.144 m
Size of the water tubes	7.62 to 10.16 cm
Size of superheater tubes	3.84 to 5.71 cm
Working pressure	40 bar (max.)
Steaming capacity	40000 kg/h (max.)
Efficiency	60 to 80%

Fig. 6.7 shows a Babcock and Wilcox boiler with longitudinal drum. It consists of a drum connected to a series of front end and rear end header by short riser tubes. To these headers are connected a series of inclined water-tubes of solid drawn mild steel.

The angle of inclination of the water tubes to the horizontal is about 15° or more. A hand hole is provided in the header in front of each tube for cleaning and inspection of tubes. A feed valve is provided to fill the drum and inclined tubes with water the level of which is indicated by the water level indicator. Through the fire door the fuel is supplied to grate where it is burnt. The hot gases are forced to move upwards between the tubes by baffle plates provided. The water from the drum flows through the inclined tubes *via* downtake header and goes back into the shell in the form of water and steam *via* uptake header. The steam gets collected in the steam space of the drum. The steam

then enters through the antipriming pipe and flows in the superheater tubes where it is further heated and is finally taken out through the main stop valve and supplied to the engine when needed.



D = Drum	*PG* = Pressure gauge
DTH = Down take header	*ST* = Superheater tubes
WT = Water tubes	*SV* = Safety valve
BP = Baffle plates	*MSV* = Main stop valve
D = Doors	*APP* = Antipriming pipe
G = Grate	*L* = Lower junction box
FD = Fire door	*U* = Upper junction box
MC = Mud collector	*FV* = Feed valve
WLI = Water level indicator	

Fig. 6.7. Babcock and wilcox boiler.

At the lowest point of the boiler is provided a mud collector to remove the mud particles through a blow-down-cock.

The entire boiler except the furnace are hung by means of metallic slings or straps or wrought iron girders supported on pillars. This arrangement enables the drum and the tubes to expand or contract freely. The brick work around the boiler encloses the furnace and the hot gases.

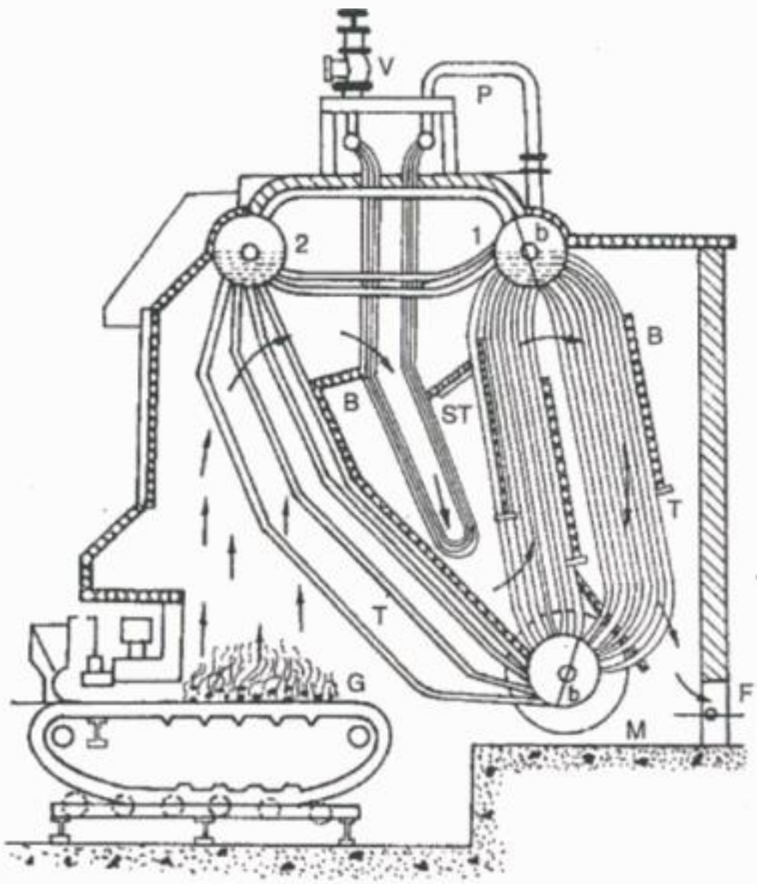
The various mountings used on the boiler are shown in Fig. 6.7.

A Babcock Wilcox water tube boiler with cross draw differs from longitudinal drum boiler in a way that how drum is placed with reference to the axis of the water tubes of the boiler. The longitudinal drum restricts the number of tubes that can be connected to one drum circumferentially and limits the capacity of the boiler. In the cross drum there is no limitation of the number of connecting tubes.

The pressure of steam in case of cross drum boiler may be as high as 100 bar and steaming capacity upto 27,000 kg/h.

6.8.2. Stirling Boiler

Stirling water tube boiler is an example of *bent tube* boiler. The main elements of a bent type water tube boiler are essentially drum or drums and headers connected by bent tubes. For large central power stations these boilers are very popular. They have steaming capacities as high as 50,000 kg/h and pressure as high as 60 bar.



V = Stop valve
P = Steam pipe
b = Water baffle
B = Baffle wall
ST = Superheater tubes
T = Water tubes
G = Grate
M = Mud drum

Fig. 6.8. Stirling boiler.

Fig. 6.8 shows a small-sized stirling water tube boiler. It consists of two upper drums known as steam drums and a lower drum known as mud or water drum. The steam drums are connected to mud drum by banks of bent tubes. The steam and water space of the steam drums are interconnected with each other, so that balance of water and steam may be obtained. For carrying out cleaning operation a manhole at one end of each drum is provided. The feed water from the economiser (not shown) is delivered to the steam drum-1 which is fitted with a baffle. The baffle deflects the water to move downwards into the drum. The water flows from the drum 1 to the mud drum through the rearmost water tubes at the backside. So the mud particles and other impurities will move to the mud drum, where these particles may be deposited. As this drum is not subjected to high temperature, so the impurities may not cause harm to the drum. The blow off cock blows off the impurities. The baffle provided at the mud drum deflects the pure water to move upwards to the drum 1 through the remaining half of the water tubes at the back. The water also flows from it to the drum 2 through the water tubes which are just over the furnace. So they attain a higher temperature than the remaining portion of the boiler and a major portion of evaporation takes place in these tubes. The steam is taken from the drum 1 through a steam pipe and then it passes through the superheater tubes where the steam is superheated. Finally the steam moves to the stop valve from where it can be supplied for further use.

The combustion products ensuing from the grate move in the upward and downward directions due to the brickwall baffles and are finally discharged through the chimney into the atmosphere. Fire brick arch gets incandescent hot and helps in combustion and preventing the chilling of the furnace when fire door is opened and cold air rushes in.

The steam drums and mud drum are supported on steel beams independent of the brickwork.

It is lighter and more flexible than the straight tube boilers. But it is comparatively more difficult to clean and inspect the bent tubes.

6.9. HIGH PRESSURE BOILERS

6.9.1. Introduction

In applications where steam is needed at pressure, 30 bar, and individual boilers are required to raise less than about 30000 kg of steam per hour, *shell boilers are considerably cheaper than the water tube boilers.* Above these limits, shell boilers (generally factory built) are difficult to transport if not impossible. There are no such limits to water tube boilers. These can be site erected from easily transportable parts, and moreover the pressure parts are of smaller diameter and therefore can be thinner. The geometry can be varied to suit a wide range of situations and furnace is not limited to cylindrical form. Therefore, *water tube boilers are generally preferred for high pressure and high output whereas shell boilers for low pressure and low output.*

The modern high pressure boilers employed for power generation are for steam capacities 30 to 650 tonnes/h and above with a pressure upto 160 bar and maximum steam temperature of about 540°C.

6.9.2. Unique Features of the High Pressure Boilers

Following are the unique features of high pressure boilers :

1. Method of water circulation
2. Type of tubing
3. Improved method of heating.

1. Method of water circulation. The circulation of water through the boiler may be *natural circulation* due to density difference or *forced circulation.* In all modern high pressure boiler plants, the water circulation is maintained with the help of pump which forces the water through the boiler plant. The use of natural circulation is limited to sub-critical boilers due to its limitations.

2. Type of tubing. In most of the high pressure boilers, the water circulated through the tubes and their external surfaces are exposed to the flue gases. In water tube boilers, if the flow takes place through one continuous tube, the large pressure drop takes place due to friction. This is *considerably reduced by arranging the flow to pass through parallel system of tubing.* In most of the cases, several sets of the tubings are used. This type of arrangement helps to *reduce the pressure loss,* and better control over the quality of the steam.

3. Improved method of heating. The following improved methods of heating may be used to increase the heat transfer :

(*i*) The saving of heat by *evaporation of water* above critical pressure of the steam.

(*ii*) The heating of water can be made by *mixing the superheated steam*. The mixing phenomenon *gives highest heat transfer co-efficient*.

(*iii*) The overall heat transfer co-efficient can be increased by *increasing the water velocity inside the tube and increasing the gas velocity above sonic velocity*.

6.9.3. Advantages of High Pressure Boilers

The following are the *advantages* of high pressure boilers.

1. In high pressure boilers pumps are used to maintain forced circulation of water through the tubes of the boiler. This *ensures positive circulation of water and increases evaporative capacity of the boiler and less number of steam drums will be required.*
2. The *heat of combustion is utilised more efficiently* by the use of small diameter tube in large number and in multiple circuits.
3. *Pressurised combustion* is used which *increases rate of firing of fuel* thus increasing the rate of heat release.
4. Due to compactness less floor space is required.
5. The tendency of scale formation is eliminated due to high velocity of water through the tubes.
6. All the parts are uniformly heated, therefore the danger of overheating is reduced and thermal stress problem is simplified.
7. The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas and air leakages.
8. The components can be arranged horizontally as high head required for natural circulation is eliminated using forced circulation. There is a greater flexibility in the components arrangement.
9. The steam can be raised quickly to meet the variable load requirements without the use of complicated control devices.
10. The *efficiency of plant is increased upto 40 to 42 per cent* by using high pressure and high temperature steam.
11. A very rapid start from cold is possible if an external supply of power is available. Hence the boiler can be used for carrying peak loads or stand by purposes with hydraulic station.
12. Use of high pressure and high temperature steam is economical.

6.9.4. LaMont Boiler

This boiler works on a forced circulation and the circulation is maintained by a centrifugal pump, driven by a steam turbine using steam from the boiler. For emergency an electrically-driven pump is also fitted.

Fig. 6.9 shows a LaMont steam boiler. The feed water passes through the economiser to the drum from which it is drawn to the circulation pump. The pump delivers the feed water to the tube evaporating section which in turn sends a mixture of steam and water to the drum. The steam in the drum is then drawn through the superheater. The superheated steam so obtained is then supplied to the prime mover.

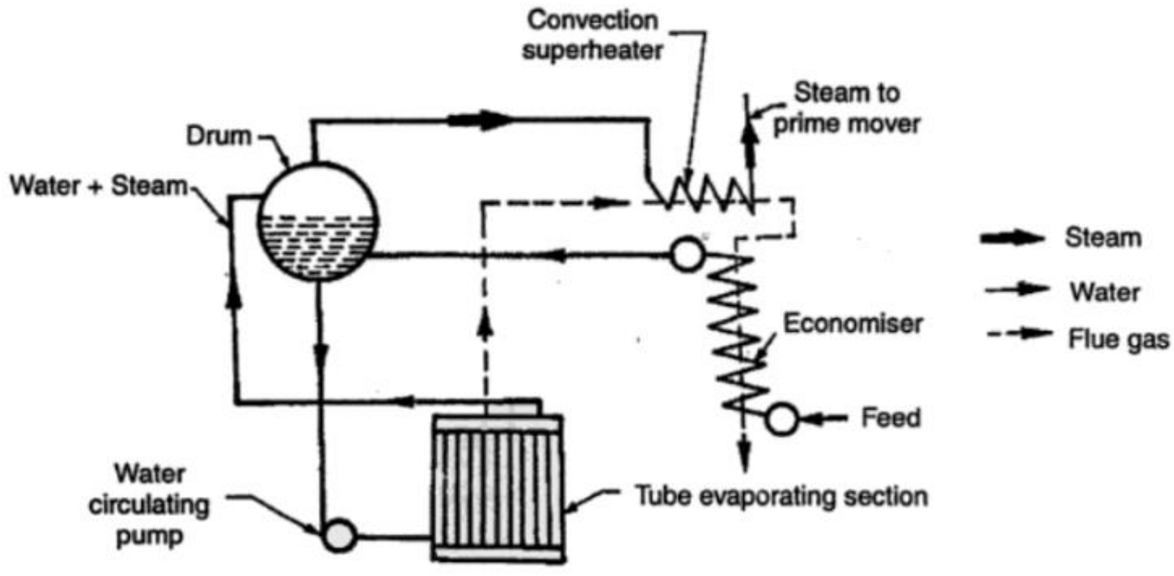


Fig. 6.9. LaMont boiler.

These boilers have been built to generate of 45 to 50 tonnes of superheated steam at a pressure of 130 bar and at a temperature of 500°C.

6.9.5. Loeffler Boiler

In a LaMont boiler the major difficulty experienced is the deposition of salt and sediment on the inner surfaces of the water tubes. The deposition reduces the heat transfer and ultimately the generating capacity. This further increases the danger of overheating the tubes due to salt deposition as it has high thermal resistance. This difficulty was solved in Loeffler boiler by *preventing the flow* of water into the boiler tubes.

This boiler also makes use of forced circulation. Its novel principle is *the evaporating of the feed water by means of superheated steam from the superheater, the hot gases from the furnace being primarily used for superheating purposes.*

Fig. 6.10 shows a diagrammatic view of a Loeffler boiler. The high pressure feed pump draws water through the economiser (or feed water heater) and deliver it into the evaporating drum. The steam circulating pump draws saturated steam from the evaporating drum and passes it through radiant and convective superheaters where steam is heated to required temperature. From the superheater about one-third of the superheated steam passes to the prime mover (turbine) the remaining two-thirds passing through the water in the evaporating drum in order to evaporate feed water.

This boiler can carry higher salt concentrations than any other type and is more compact than indirectly heated boilers having natural circulation. These qualities fit it for land or sea transport power generation.

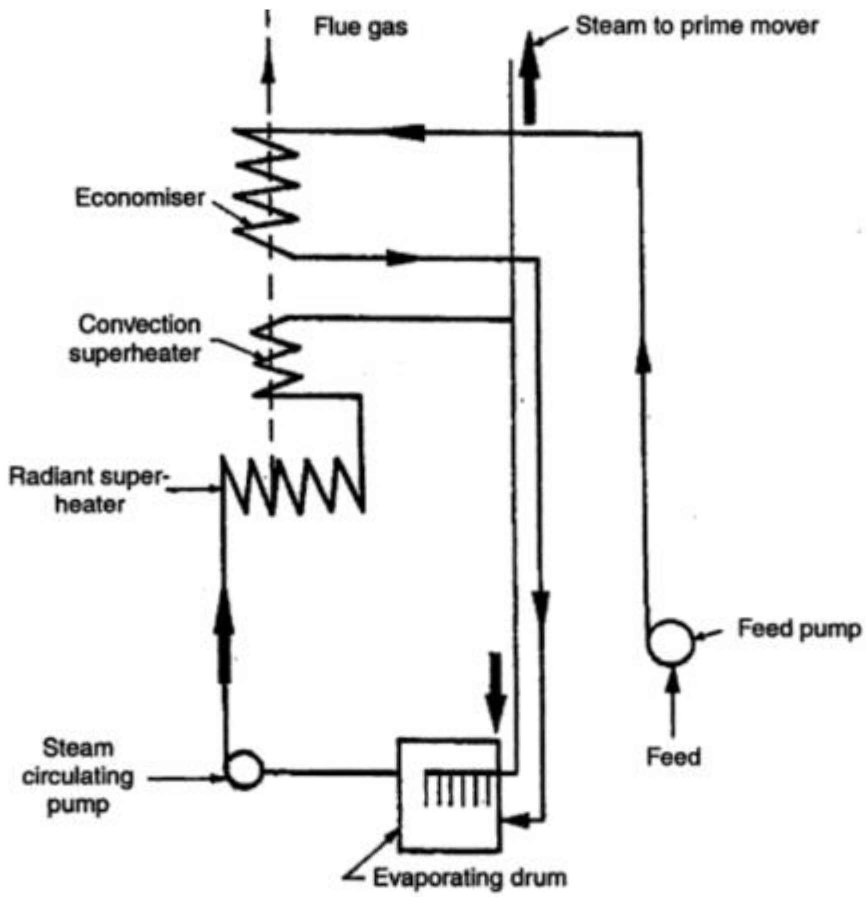


Fig. 6.10. Loeffer boiler.

6.9.6. Benson Boiler

In the LaMont boiler, the main difficult experienced *is the formation and attachment of bubbles on the inner surfaces of the heating tubes. The attached bubbles to the tube surfaces reduce the heat flow and steam generation as it offers high thermal resistance than water film. Benson in 1922 argued that if the boiler pressure was raised to critical pressure (225 atm.), the steam and water have the same density and therefore, the danger of bubble formation can be easily eliminated.* The first high pressure Benson boiler was put into operation in 1927 in West Germany.

This boiler too makes use of forced circulation and uses oil as fuel. It chief novel *principle is that it eliminates the latent heat of water by first compressing the feed to a pressure of 235 bar, it is then above the critical pressure and its latent heat is zero.*

Fig. 6.11 shows a schematic diagram of a Benson boiler. This boiler does not use any drum. The feed water after circulation through the economic tubes flows through the radiant parallel tube section to evaporate partly. The steam water mixture produced then moves to the transit section where this mixture is converted into steam. The steam is now passed through the convection superheater and finally supplied to the prime mover.

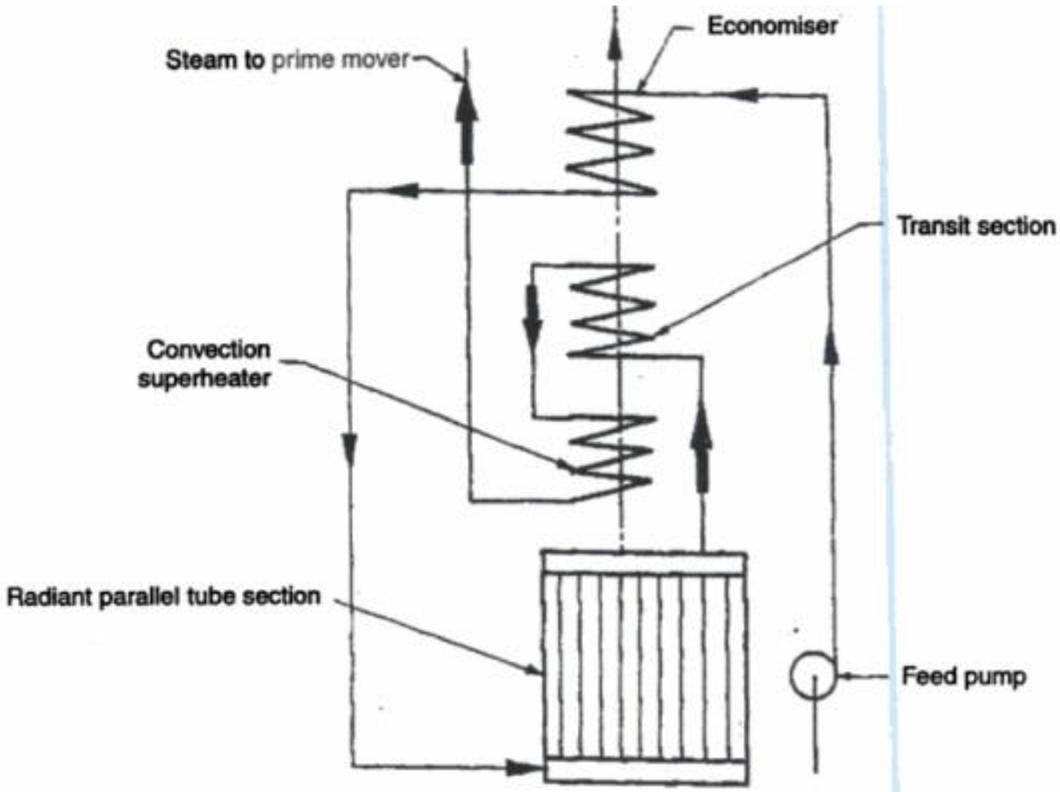


Fig. 6.11. Benson boiler.

Boilers having as high as 650°C temperature of steam had been put into service. The maximum working pressure obtained so far from commercial Benson boiler is 500 atm. The Benson boilers of 150 tonnes/hr generating capacity are in use.

Advantages of a Benson Boiler :

The Benson boiler possesses the following advantages :

1. It can be erected in a comparatively smaller floor area.
2. The total weight of a Benson boiler is 20% less than other boilers, since there are no drums. This also reduces the cost of the boiler.
3. It can be started very quickly because of welded joints.
4. Natural convection boilers require expansion joints but these are not required for Benson boiler as the pipes are welded.
5. The furnace walls of the boiler can be more efficiently protected by using smaller diameter and closed pitched tubes.
6. The transfer of parts of the boiler is easy as no drums are required and majority of the parts are carried to the site without pre-assembly.
7. It can be operated most economically by varying the temperature and pressure at partial loads and overloads. The desired temperature can also be maintained constant at any pressure.

8. The blow-down losses of the boiler are hardly 4% of natural circulation boiler of the same capacity.
9. Explosion hazards are not severe as it consists of only tubes of small diameter and has very little storage capacity.
10. The superheater in a Benson boiler is an integral part of forced circulation system, therefore no special starting arrangement for superheater is required.

6.9.7. Velox Boiler

It is a well known fact that when the gas velocity exceeds the sound-velocity, the heat is transferred from the gas at a much higher rate than rates achieved with sub-sonic flow. The advantage of this theory is taken to effect the large heat transfer from a smaller surface area in this boiler.

This boiler makes use of **pressurised combustion.**

The gas turbine drives the axial flow compressor which raises the incoming air from atmosphere pressure to furnace pressure. The combustion gases after heating the water and steam flow through the gas turbine to the atmosphere. The feed water after passing through the economiser is pumped by a water circulating pump to the tube evaporating section. Steam separated in steam separating section flows to the superheater, from there it moves to the prime mover.

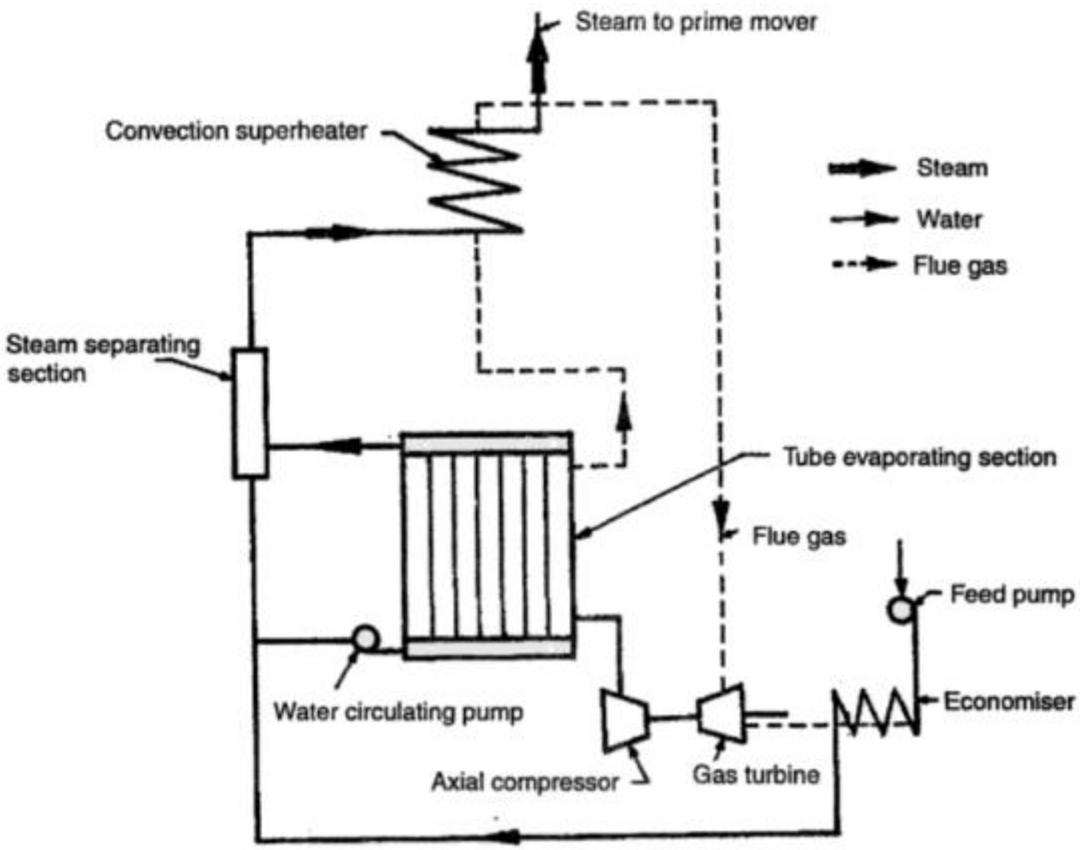


Fig. 6.12. Velox boiler.

The size of the Velox boiler is limited to 100 tonnes/h because 600 B.H.P. is required to run the air compressor at this output. The power developed by the gas turbine is not sufficient to run the compressor and therefore some power from external source must be supplied.

Advantages

1. The boiler is very compact and has greater flexibility.
2. Very high combustion rates are possible.
3. It can be quickly started.
4. Low excess air is required as the pressurised air is used and the problem of draught is simplified.

6.9.8. Super-Critical Boilers

A large number of steam generating plants are designed between working ranges of 125 atm. and 510°C to 300 atm. and 660°C ; these are basically characterised as *sub-critical and super-critical.*

Usually *a sub-critical boiler* consists of three distinct section as *preheater* (economiser), *evaporator* and *superheater.*

A *super-critical boiler* requires only *preheater* and *superheater.*

The constructional layout of both the above types of boilers is, however, practically identical.

These days it has become a rule to use *super-critical boilers above 300 MW capacity units.*

The super-critical boilers claim the following *advantages* over critical type :

1. Large heat transfer rates.
2. Owing to less heat capacity of the generator the pressure level is more stable and therefore gives better response.
3. Because of absence of two phase mixture the problems of erosion and corrosion are minimised.
4. More adaptable to load fluctuations (because of great ease of operation, simplicity and flexibility).
5. The turbo-generators connected to super-critical boilers can generate peak loads by changing the pressure of operation.
6. Higher thermal efficiency.

Presently, 246 atm. and 538°C are used for unit size above 500 MW capacity plants.

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6. Higher thermal efficiency.

Presently, 246 atm. and 538°C are used for unit size above 500 MW capacity plants.

6.9.9. Supercharged Boiler

In a supercharged boiler, the combustion is carried out under pressure in the combustion chamber by supplying the compressed air. The exhaust gases from the combustion chamber are used to run the gas turbine as they are exhausted to high pressure. The gas turbine runs the air compressor to supply the compressed air to the combustion chamber.

Advantages :

1. Owing to very high overall heat transfer co-efficient the heat transfer surface required is hardly 20 to 25% of the heat transfer surface of a conventional boiler.
2. The part of the gas turbine output can be used to drive other auxiliaries.
3. Small heat storage capacity of the boiler plant gives better response to control.
4. Rapid start of the boiler is possible.
5. Comparatively less number of operators are required.

B. BOILER MOUNTINGS AND ACCESSORIES

6.10. BOILER MOUNTINGS AND ACCESSORIES

6.10.1. Introduction to boiler

Boiler mountings. These are different fittings and devices which are *necessary for the operation and safety of a boiler*. Usually these devices are mounted over boiler shell.

In accordance with the Indian boiler regulation the following mountings should be fitted to the boilers.

- Two safety valves
- Two water level indicators
- A pressure gauge
- A steam stop valve
- A feed check valve
- A blow off cock
- An attachment for inspector's test gauge
- A man hole
- Mud holes or sight holes.

Boilers of Lancashire and Cornish type should be fitted with a *high pressure and low water safety valve.*

All land boilers should have a *fusible plug* in each furnace.

Boiler Accessories. These are *auxiliary plants required for steam boilers for their proper operation and for the increase of their efficiency.* Commonly used boiler accessories are :

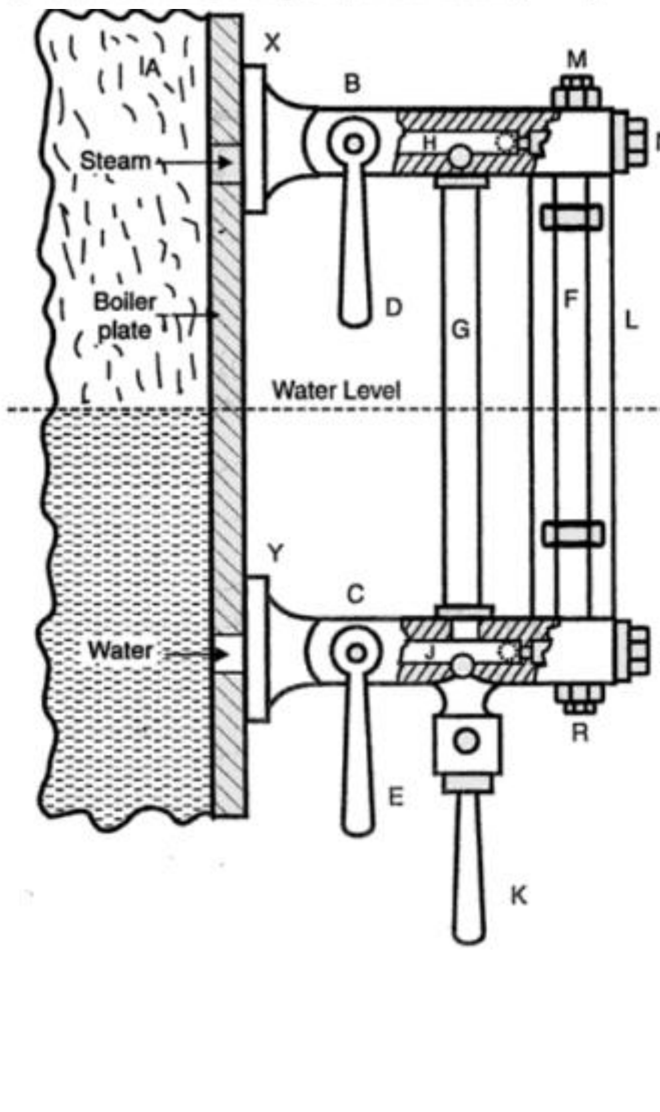
- Feed pumps
- Injector
- Economiser
- Air preheater
- Superheater
- Steam separator
- Steam trap.

6.10.2. Boiler Mountings

The various boiler mountings are discussed as follows :

6.10.2.1. Water level indicator

- *The function of a water level indicator is to indicate the level of water in the boiler constantly.* It is also called *water gauge.* Normally two water level indicators are fitted at the front end of every boiler. Where the boiler drum is situated at considerable height from the floor, the water gauge is often inclined to make the water level visible from any position. When the water being heated in the boiler transforms into steam the level of water in the boiler shell goes on decreasing. For the proper working of the boiler, the water must be kept at safe-level. If the water level falls below the safe level and the boiler goes on producing steam without the addition of feed water, great *damage like crack and leak can occur to the parts of the boiler which get uncovered from water.* This can result in the *stoppage of steam generation and boiler operation.*
- Fig. 6.13 shows a Hopkinson's water gauge. It is a common form of glass tube water-level gauge. *A* is the front end plate of the boiler. *F* is a very hard glass tube indicating water level and is connected to the boiler plate through stuffing boxes in hollow gun metal castings (*B, C*) having flanges *X, Y* for bolting the plate.



A = End plate of boiler	H and J = Two balls
B and C = Hollow gun metal castings	K = Drain cock
D and E = Cocks	L = Guard glass
F = Gauge glass	M, N, P, R = Screwed caps
G = Hollow metal column	X, Y = Flanges

Fig. 6.13. Water level indicator.

For controlling the passage of steam and water cocks D and E are provided. When these cocks are opened the water stands in the glass tube at the same level as in the boiler. K is the drain cock to blow out water at intervals so as not to allow any sediments to accumulate. Upper and lower stuffing boxes are connected by a hollow metal column G. Balls J and H rest in the position shown in the normal working of the gauge. When the glass tube breaks due to rush of water in the bottom passage the balls move to dotted positions and shut off the water and steam. Then the cocks D and E can be safely closed and broken glass tube replaced. M, N, P and R are screwed caps for internal cleaning of the passage after dismantling. L is the guard glass ; it is tough and does not give splinters on breaking. Thus when the gauge glass breaks, and this guard glass which normally will hold flying pieces, also gives way, the pieces will not fly one and *hurt* the attendant.

6.10.2.2. Pressure gauge

- ***The function of a pressure gauge is to measure the pressure exerted inside the vessel.*** The gauge is usually mounted on the front top of the shell or the drum. It is usually constructed to indicate upto double the maximum working pressure. Its dial is graduated to read pressures in kgf/cm^2 (or bar) gauge (*i.e.*, above atmospheric). There are two types of pressure gauges : (*i*) Bourdon tube pressure gauge and (*ii*) Diaphragm type pressure gauge. A pointer, which rotates over a circular graduated scale, indicates the pressure.

A pressure gauge is known as *compound pressure gauge* if it is designed in such a fashion so as to measure pressures above and below the atmosphere on the same dial.

Fig. 6.14 shows a Bourdon pressure gauge (single tube) a common type of pressure gauge used. The essential feature of this gauge is the elliptical spring tube which is made of a special

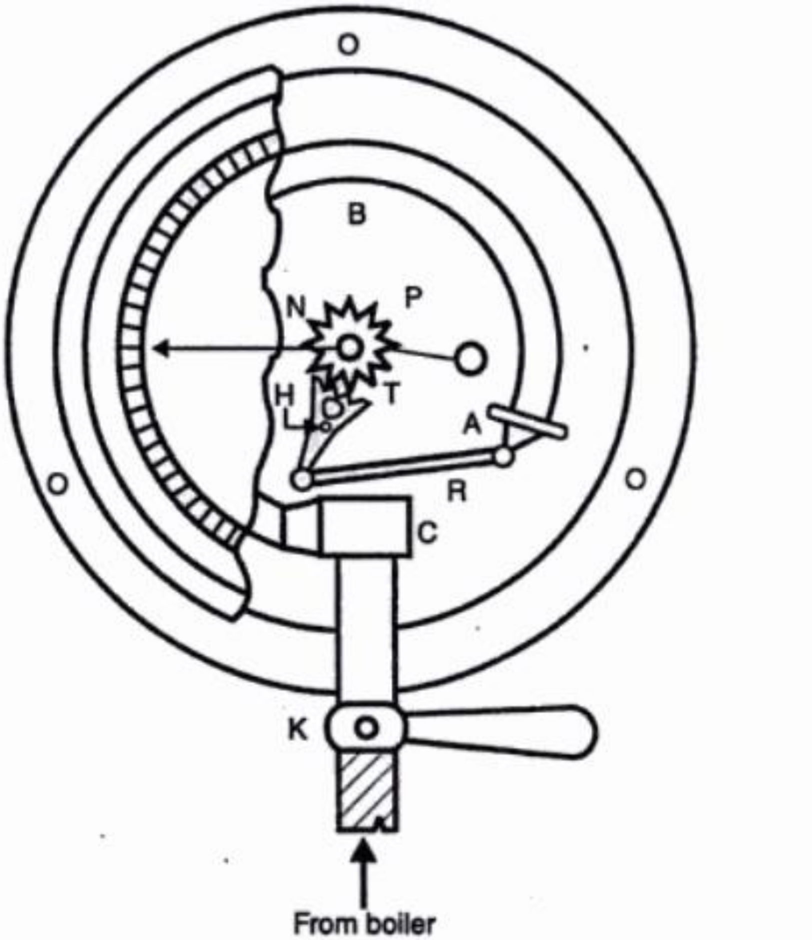
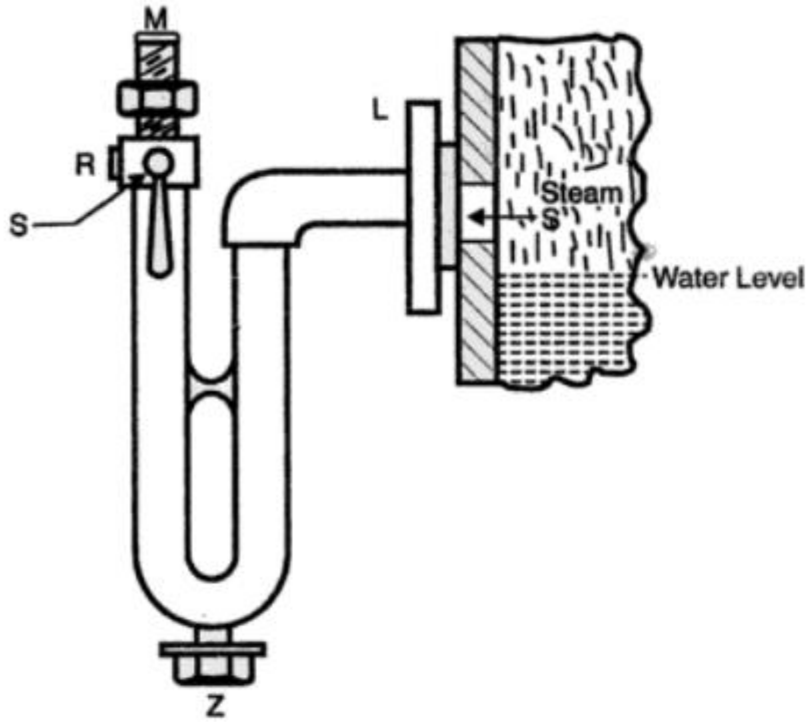


Fig. 6.14. Bourdon pressure gauge.

quality of bronze and is solid drawn. One end *A* is closed by a plug and the other is connected with a block *C*, the block is connected with a syphon tube (which is full of condensed water). The steam pressure forces the water from the syphon tube into elliptical tube and this causes the tube to become circular is cross-section. As the tube is fixed at *C*, the other end *A* moves outwards. This outward movement is magnified by the rod *R* and transmitted to toothed sector *T*. This toothed sector is hinged at the point *H* and meshes with the pinion *P* fixed to the spindle of the pointer *N*. Thus the pointer moves and registers the pressure on a graduated dial.

The movement of the free end of the elliptical tube *is proportional to the difference between external and internal pressure on the tube.* Since the outside pressure on the tube is atmospheric, the movement of the free end is a measure of the boiler pressure above atmospheric *i.e., gauge pressure.*



L = Flange
M = Connection pressure gauge
Z = Plug for cleaning syphon
R = Plug for testing pressure gauge
S = Three way cock

Fig. 6.15. U-tube syphon.

Fig. 6.15 shows a U-tube syphon which connects the gauge to the boiler. The U-tube syphon is connected to the steam space of the boiler and contains condensed steam which enters the gauge tube. The condensed water transmits pressure to the gauge, and at the sametime prevents steam from entering the pressure gauge. In case steam passes into the gauge tube it will expand the tube and reading obtained will be false. Furthermore metal may be affected. Plug R is used for connecting the inspector's standard gauge and testing accuracy of boiler pressure gauge while in service. Plug Z is employed for cleaning the syphon. Three way cock S is used for either connecting the boiler pressure gauge to steam space or inspector's pressure gauge to the steam space.

Note. *The double-tube Bourdon gauge is more rigid than the single tube and more suitable for locomotive and portable boilers.*

6.10.2.3. Safety valves

The function of a safety valve is to release the excess steam when the pressure of steam inside the boiler exceeds the rated pressure. As soon as the pressure of steam inside the boiler exceeds the rated pressure the safety valve automatically opens and excess steam rushes out into the atmosphere till the pressure drops down to the normal value. A safety valve is generally mounted on the top of the shell.

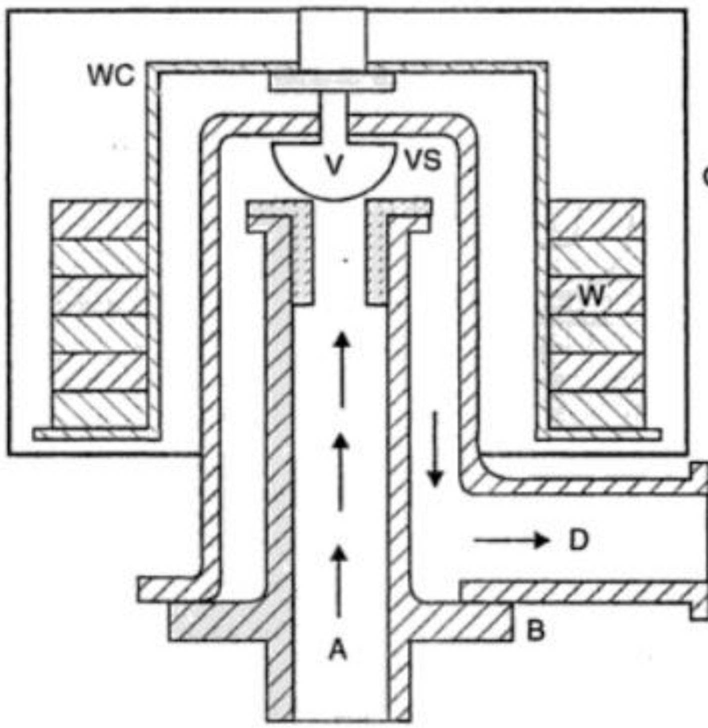
As per boiler regulations every boiler must be fitted at least with two safety valves.

The various types of safety valves are enumerated and discussed as follows :

1. Dead weight safety valve.
2. Lever safety valve.
3. Spring loaded safety valve.
4. High steam and low water safety valve.

1. Dead weight safety valve

Fig. 6.16 shows a dead weight safety valve. *A* is the vertical cast iron pipe through which steam pressure acts. *B* is the bottom flange directly connected to seating block on the boiler shell communicating to the steam space. *V* is the gun metal valve and *VS* is the gun metal valve seat. *D* is another cast iron pipe for discharge of excess steam from the boiler. *W* are the weights in the form of cylindrical disc of cast iron. *WC* is the weight carrier carrying the weights *W*. The cover plate *C* covers these weights. The steam pressure acts in the upward direction and is balanced by the force of the dead weights *W*. The total dead-weights consist of the sum of the weights *W*, weight of the valve *V*, weight of the weight carrier and weight of the cover plate *C*.



A = Cast-iron pipe
B = Bottom flange
V = Gun metal valve
W = Cast-iron weights
WC = Weight carrier
D = Discharge pipe
VS = Gun metal valve seat
C = Cover plate

Fig. 6.16. Dead weight safety valve.

When the steam pressure is greater than the working pressure it lifts the valve with its weights. So the steam escapes from the boiler and the steam pressure thereby decreases.

Merits of dead weight safety valve

1. Simplicity of design.
2. Gives quite a satisfactory performance during operation.
3. It cannot be easily tempered from the pressure adjustment view-point.

Demerits

1. Unsuitable for use on any boiler *where extensive vibration and movement are experienced* (*e.g.* locomotive and marine work).

2. It is not suitable for high pressure boilers because a large amount of weight is required to balance the steam pressure.

Uses. It is mainly used for *low pressures, low capacity, stationary boilers* of the Cornish and Lancashire types.

2. *Lever safety valve*

Refer Fig. 6.17. It consists of a lever and weight W. The valve (made of gun metal) rests on the valve seat (gun metal) which is screwed into the valve body ; the valve seat can be replaced if required. The valve body is fitted on the boiler shell. One end of the lever is hinged while at the other is suspended a weight W. The strut presses against the valve on seat against the steam pressure below the valve. The slotted lever guide allows vertical movement to the lever.

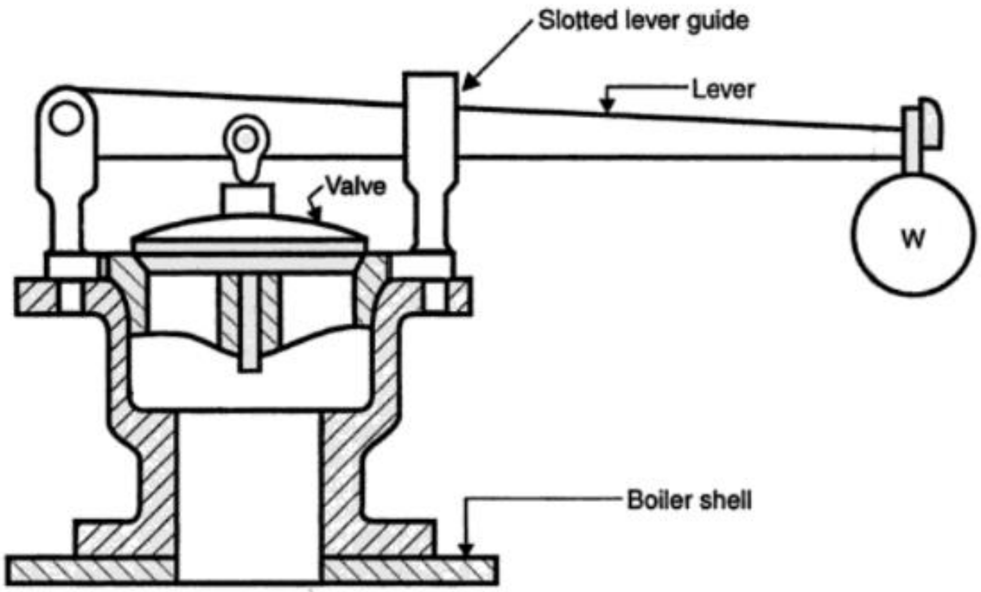


Fig. 6.17. Lever safety valve.

When the steam pressure becomes greater than the normal working pressure, the valve is lifted with the lever and the weight. Consequently, the steam escapes through the passages between the valve and seat and the steam pressure decreases.

The disadvantages of this valve is that it admits of being tempered with, and the effect of a small addition to the weight is magnified considerably in its action on the valve.

Fig. 6.18 shows the loading arrangement on the lever.

Let p = Steam pressure (gauge)

d = Diameter of the valve

W = Weight suspended on the lever

W_l = Weight of the lever acting at the centre of gravity G

W_v = Weight of the valve

A = Area of the valve.

Taking moments about the fulcrum F, we get

$$W \times AF + W_l \times GF + W_v \times VF = p \times a \times VF, \text{ where } a = \frac{\pi}{4} d^2.$$

From the above equation we can find the weight W or length of lever for a given pressure of steam (p).

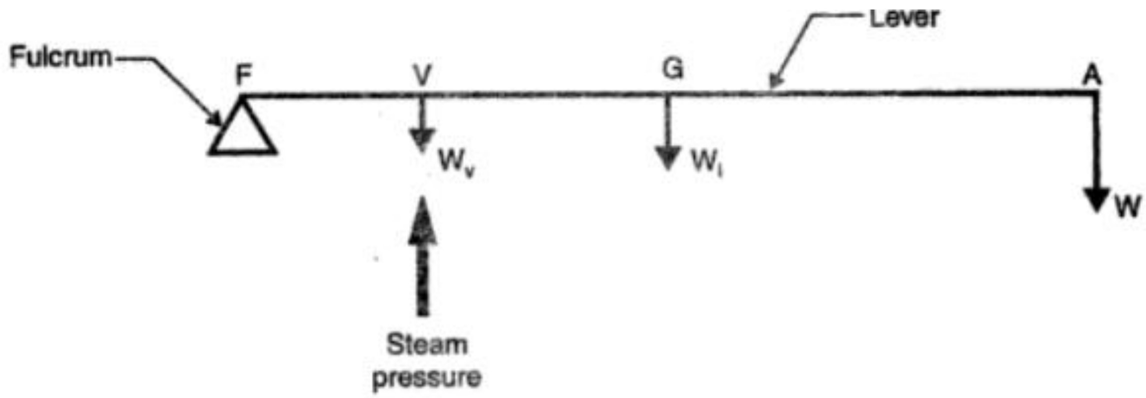


Fig. 6.18. Loading arrangement on the lever.

3. *Spring loaded safety valve*

For locomotives and marine engines both the lever and dead-weight types are unsuitable for abvious reasons, and the valve must be spring loaded, as such valve is *unaffected by vibration or deviations from the vertical*.

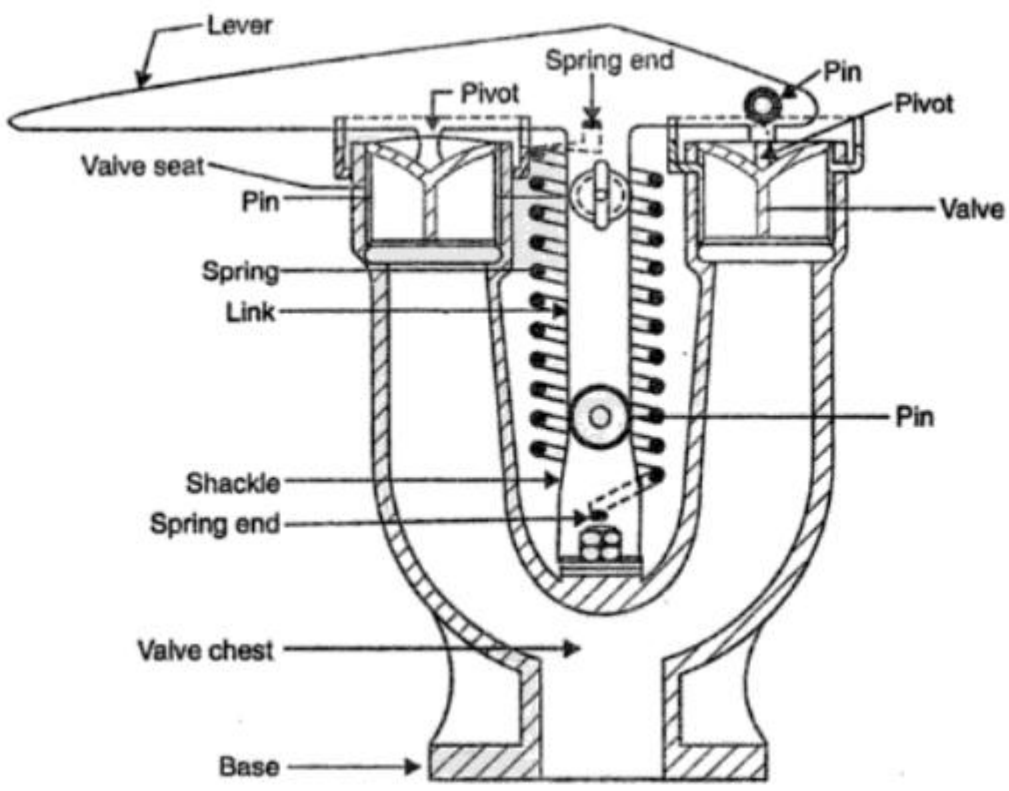


Fig. 6.19. Ramsbottom spring loaded safety valve.

Fig. 6.19. Ramsbottom spring loaded safety valve.

Fig. 6.19 illustrates what is known as Ramsbottom spring loaded safety valve. It consists of two separate valves and seatings having one lever, bearing on the two valves, and loaded by a spring, the spring being placed between the valves. The tension on the spring can be adjusted by the nuts. By pulling or raising the lever the operator/driver can relieve the pressure from either valve separately, and ascertain it is not sticking on the seating.

One *disadvantage* of the spring-loaded safety valve is that *the load on the valve increases as the valve lifts, so that pressure required just to lift the valve is less than that required to open it fully.* From this reason in some cases *it is arranged that the area acted on by the steam is greater when the valve is open than the valve is closed.*

6.10.2.5. Fusible plug

The function of a fusible plug is to protect the boiler against damage due to overheating for low water level. It is fitted on the fire box crown plate or over the combustion chamber at its appropriate place.

A common from of fusible plug is illustrated in Fig. 6.21. It consists of a hollow gun metal body screwed into the fire box crown. The body has a hexagonal flange to tighten it into the shell. A gun metal plug having a hexagonal flange is screwed into the gun metal body. There is another hollow gun metal plug separated from the metal plug by an annulus of fusible metal. The fusible metal is protected from fire by flange on the hollow gun metal plug.

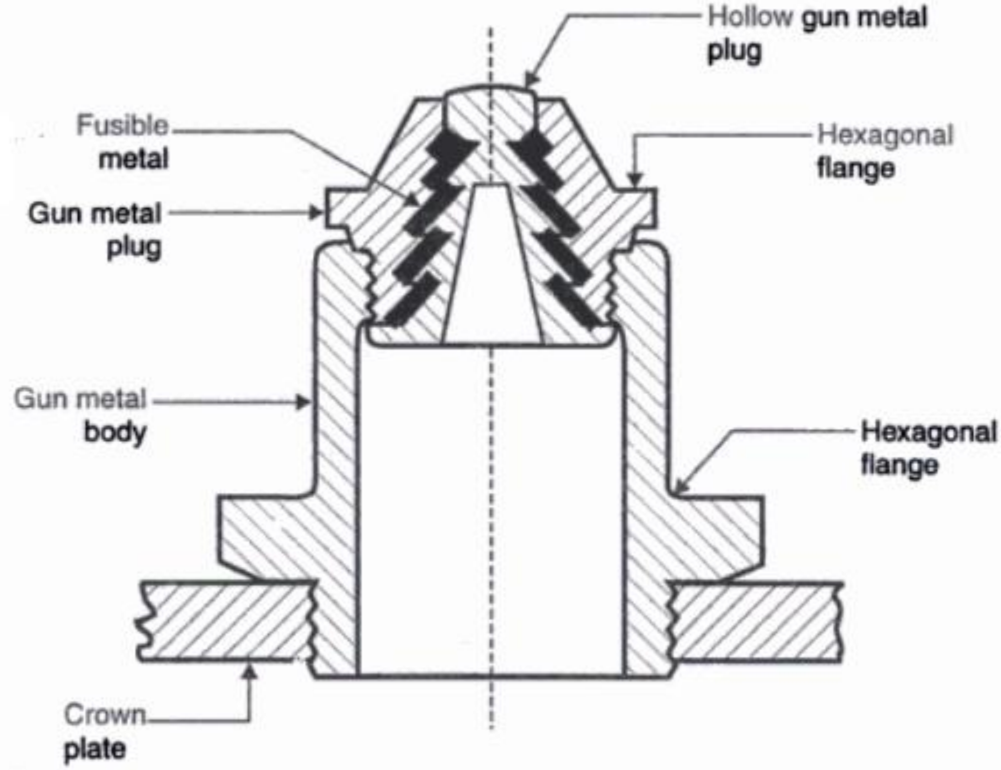


Fig. 6.21. Fusible plug.

Boiler accessories

Boiler accessories are those components which are installed either inside or outside the boiler to increase the efficiency of the plant and to help in the proper working of the plant.

Boiler Accessories: These are the devices which are used as integral parts of a boiler and help in running efficiently.

Various boiler accessories are:

1. Feed pump
2. Super heater
3. Economiser
4. Air Preheater
5. Injector

1. Feed pump

It is used to deliver water to the boiler. A feed pump may be of centrifugal type or reciprocating type. But a double acting reciprocating pump is commonly used as a feed pump.

2. Super Heater

Function

It superheats the steam generated by the boiler and increases the temperature steam above saturation temperature at constant pressure.

Location

Superheaters are placed in the path of flue gases to recover some of their heat. In bigger installations, the superheaters are placed in an independently fired furnace. Such superheaters are called separately fired or portable superheaters.

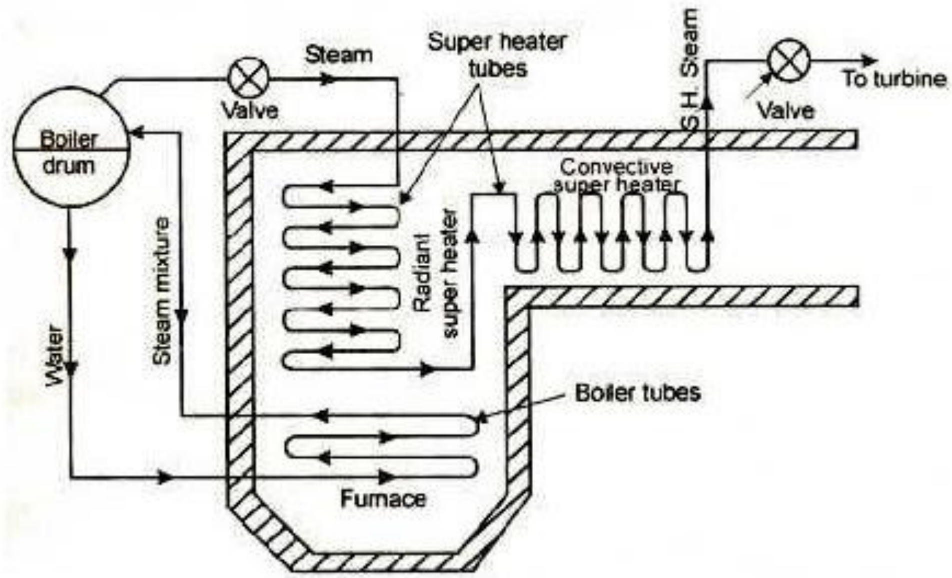


Fig Super heater (radiant and convective)

Construction

There are many types of super heaters. A combination type of radiant and convective super heater is shown in figure. Both these super heaters are arranged in series in the path of flue gases. Radiant super heater receives heat from the burning fuel by radiation process. Convective super heater is placed adjacent to the furnace wall in the path of flue gases. It receives heat by convection.

Working

Steam stop valve is opened. The steam (wet or dry) from the evaporator drum is passed through the superheated tubes. First the steam is passed through the radiant superheater and then to the convective super heater. The steam is heated when it passes through these super heaters and converted into superheated steam. This superheated steam is supplied to the turbine through a valve.

Applications

This type of super heaters is used in modern high pressure boilers.

Advantages of superheated steam (super heaters)

1. Work output is increased for the same quantity of steam.
2. Loss due to condensation of steam in the steam engine and is the steam mains is minimized.
3. Capacity of the plant is increased.
4. Thermal efficiency is increased since the temperature of superheated steam is high.

3. Economiser

Function:

An economizer pre –heats (raise the temperature) the feed water by the exhaust flue gases. This pre –heated water is supplied to the boiler from the economizer.

Location:

An economizer is placed in the path of the flue gases in between the boiler and the air pre – heater or chimney.

Construction:

An economizer used in modern high pressure boilers is shown by a line sketch. It consists of a series of vertical tubes. These tubes are hydraulically pressed into the top and bottom headers. The bottom header is connected to feed pump. Top header is connected to the water space of the boiler. It is provided with a safety valve which opens when water pressure exceeds a certain limit. To keep the surface of the tubes clean from soot and ash deposits, scrapers are provided in the tubes. These scrapers are slowly moved up and down to clean the surfaces of the tubes. The action of adjacent pairs of scraper is in opposite direction. i.e., when one scraper moves up, the other moves down.

Economizers may be parallel or counter-flow types. When the gas flow and water flow are in the same direction, it is called parallel flow economizer. In counter-flow, the gas flow and water flow are in opposite direction.

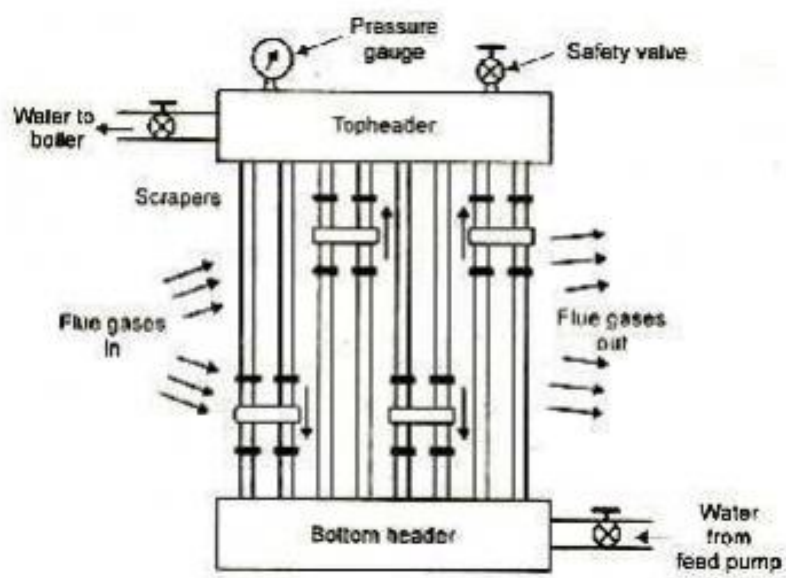


Fig. Economizer

Working

The feed water is pumped to the bottom header and this water is carried to the top header through a number of vertical tubes. Hot flue gases are allowed to pass over the external surface of the tubes. The feed water which flows upward in the tubes is thus heated by the flue gases. This pre-heated water is supplied to the boiler.

Advantages

1. Feed water to the boiler is supplied at high temperature. Hence heat required in the boiler is less. Thus fuel consumption is less.
2. Thermal efficiency of the plant is increased.
3. Life of boiler is increased.
4. Loss of heat in flue gases is reduced.
5. Steaming capacity is increased.

4. Air pre-heater

Function

Air pre-heater pre-heats (increases the temperature) the air supply to the furnace with the help of hot the gases.

Location

It is installed between the economizer and the chimney.

Construction

A tubular type air pre-heater is shown in figure. It consists of a large number of tubes. Flue gases pass through the tube. Air flows over the tubes. Baffles are provided to pass the air number of times over the tubes. A soot hopper is provided at the bottom to collect the soot.

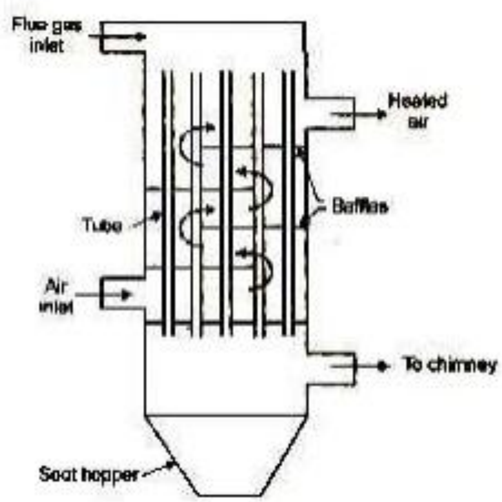


Figure: Air pre-heater

Working

Hot flue gases pass through the tubes of air pre-heater after leaving the boiler or economizer. Atmospheric air is allowed to pass over these tubes. Air and flue gases flow in opposite directions. Baffles are provided in the air pre-heater and the air passes number of times over the tubes. Heat is absorbed by the air from the flue gases. This pre-heater air is supplied to the furnace to air combustion.

Advantages

1. Boiler efficiency is increased.
2. Evaporative rate is increased.
3. Combustion is accelerated with less soot, smoke and ash.
4. Low grade and inferior quality fuels can be used.

Injector

The function of an injector is to feed water into the boiler. **It is commonly employed for vertical and locomotive boilers and does not find its application in large capacity high pressure boilers. It is also used where the *space is not available for the installation of a feed pump*.**

In an injector the water is delivered to the boiler by steam pressure ; the *kinetic energy of steam is used to increase the pressure and velocity of the feed water*.

Fig. 6.26 shows an injector. It consists of a spindle *P*, a steam cone *S*, a combining cone *K*, a delivery cone *D*, and a handle *H*, with a pointer *T*. The spindle's upper end is provided with a handle while the lower end serves the purpose of a valve. The pointer on the handle indicates the 'shut' and 'open' position of the valve. The lower part of the spindle has a screw which works in a nut which is integral part of the steam cone. The key *E* checks the rotation of steam cone. With the rotation of the handle steam cone moves up or down and consequently the valve controls the steam flow through the steam cone. The steam enters through the steam pipe *A*, while the feed water enters through the water pipe *B*. The flow of water is also regulated due to sliding motion of the steam cone by its lower end. The water mixes with the steam at the combining cone where it is condensed. The mixture then passes through the delivery cone and there its kinetic energy is converted into pressure energy. The final pressure must be greater than the steam pressure of boiler otherwise water will not enter into the boiler. The excess water finds its way through the overflow pipe.

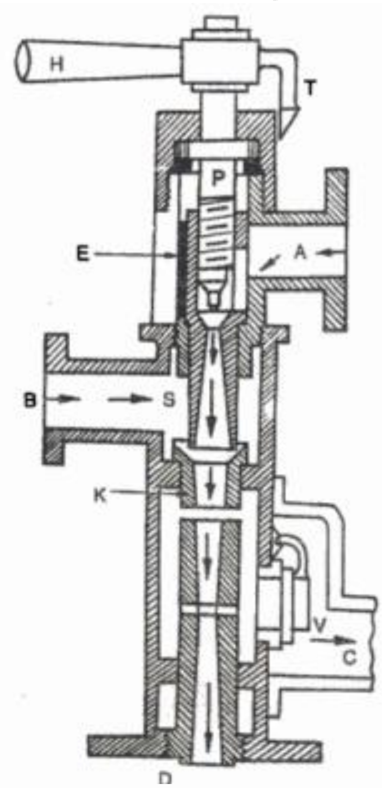
Advantages of an injector

1. Low initial cost.
2. Simplicity.
3. Compactness.
4. Absence of dynamic parts.
5. Thermal efficiency very high (about 99%).
6. Ease of operation.

Disadvantages

1. Pumping efficiency is low.
2. It cannot force very hot water.
3. Irregularity of operation under extreme variation in steam pressure.

Note. An injector is more efficient than a feed pump because all the heat in the operating steam is returned to boiler and in addition to performing the work of a pump, the injector acts as a feed water heater. But when a large quantity of feed water is involved (*e.g.*, marine and large installations) feed pumps are employed because they have greater reliability and require lesser amount of attention.

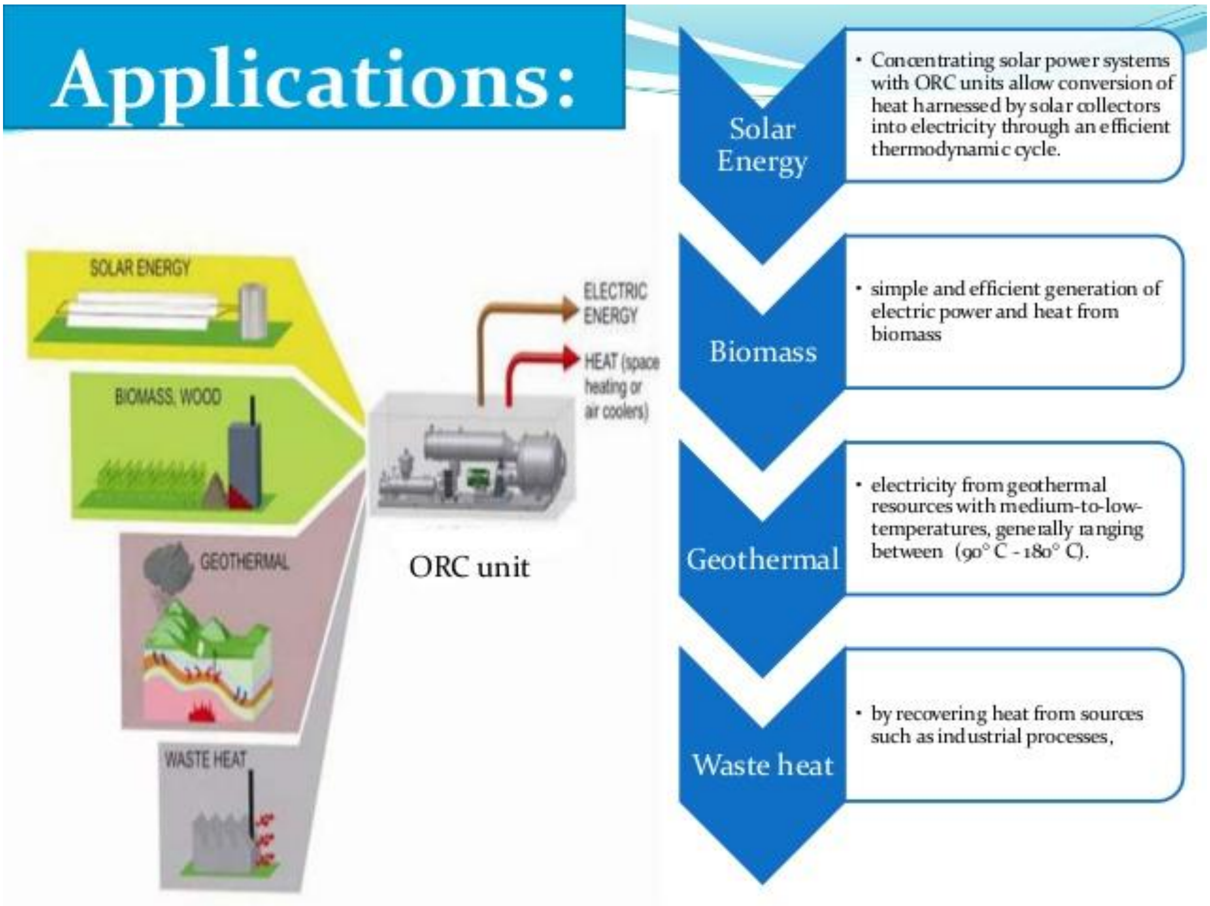


S = Steam cone	E = Key
K = Combining cone	H = Handle
D = Delivery cone	T = Pointer
P = Spindle	V = Valve
A = Steam pipe	C = Overflow pipe
B = Water pipe	

Fig. 6.26. Injector.

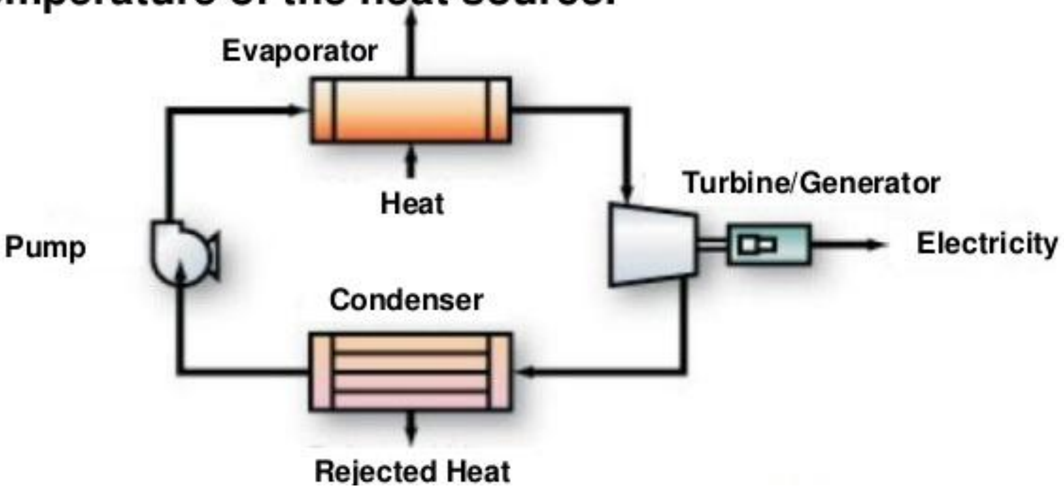
INDUSTRIAL APPLICATIONS

Industrial Applications of Rankine Cycle



Organic Rankine Cycle (ORC)

An Organic Rankine Cycle is best suited to making electricity from low-grade heat sources, using an organic fluid to match the fluid's properties with the temperature of the heat source.



Mostly Industrial Boilers Used in

1. Food Processing Industry
2. Milk & Dairies Industry
3. Rice Mills Industry
4. Textile Industry
5. Pharmaceuticals Industry
6. Rubber Industry
7. Thermocol Industry
8. Plywood Industry
9. Metal Forging Industry
10. Health Care Industry
11. Chemical Industry
12. Automobiles Industry
13. Construction Industry
14. Paper Industry
15. Refineries & Distilleries
16. Sugar Industry

High pressure boilers are used in industrial, commercial, or manufacturing applications for central heating systems,autoclaves, hot water supply and other important factory or plant processes.

TUTORIAL QUESTIONS

Theory Questions:

1. Mention the different operations of Rankine cycle. Draw the schematic for an ideal Rankine cycle. Draw p-v, T-s and h-s diagrams for this cycle.
2. Explain regeneration cycle with the help of neat sketches of layout, p-v and T-s plots.
3. What are the different thermodynamic variables affecting efficiency and output of Rankine cycle. Explain their influence on Rankine cycle.
4. Draw diagram of ‘reheat cycle’ and state the advantages and disadvantages of reheating
5. Sketch the process diagram of a ‘regenerative cycle’. State the advantages of regenerative cycle over simple Rankine cycle.
6. How boilers are classified on different accounts with examples for each category.
7. Write any six comparisons between fire tube and water tube boilers.
8. Explain the working of Babcock and Wilcox boiler with the help of a neat sketch.
9. Sketch and describe a Cochran boiler. What are its special features?
10. Explain Lancashire boiler with neat sketch.
11. What are the functions of boiler mountings and accessories? Explain any one accessory
12. Explain vilox Boiler with Neat Sketch?
13. Explain Lamont Boiler with Neat sketch

NUMERICAL PROBLEMS

1. Consider a regenerative vapour power cycle with a feed water heater. The steam enters the first stage turbine at 8 MPa, 500^{o}C and expands to 0.7 MPa, where some of the steam is extracted and diverted to feed water heater operating at 0.7 MPa. The remaining steam expands through the second stage turbine to a condenser pressure of 0.008 MPa. The saturated liquid exits the feed water heater at 0.7 MPa. The isentropic efficiency of each turbine is 85%, while each pump operates isentropically. If the net power output of the cycle is 105 MW, determine

   1. Thermal efficiency of the cycle
   2. The mass flow rate of steam entering the first turbine stage.
2. In a Rankine cycle, the steam at inlet to turbine is saturated at pressure of 30 bar and exhaust pressure is 0.25 bar. Determine (i) The pump work (ii) Turbine work (iii)

ASSIGNMENT QUESTIONS

Rankine efficiency (iv) Condenser heat flow (v) dryness at the end of expansion. Assume flow rate of 10 kg/s.

3. A steam power plant equipped with combined reheat and regenerative arrangements is supplied with steam to H.P turbine at 80 bar and 470ºC. For feed heating a part of steam is extracted at 7 bar and the remainder of steam is reheated to 350ºC in a reheater and then expanded in L.P turbine down to 0.035 bar. Determine (i) amount of steam bled off for feed heating (ii) amount of steam in L.P turbine (iii) heat supplied in boiler and reheater (iv) Output of turbine (v) cycle efficiency.

ASSIGNMENT QUESTIONS

1. A power generating plant uses steam as working fluid and operates at boiler pressure of 50 bar, dry saturated and a condenser pressure of 0.1bar. Calculate for these limits:The cycle efficiency; ii) the work ratio and specific steam consumption for Rankine cycle. Take pumping work also into account
2. a) Explain a regenerative cycle with a diagram
   b) Draw diagram of 'reheat cycle' and state the advantages and disadvantages of reheating
3. a) Explain ***any two*** of the following with neat sketches
   i) Super heater ii) Air Preheater iii) Economizer
   b) List the advantages of high pressure boilers.
4. Explain the working of Babcock and Wilcox boiler with the help of a neat sketch.
5. In a reheat cycle steam enters the H.P turbine at 100 bar and 500ºC. The expansion is continued to a pressure of 8.5 bar with isentropic efficiency of 80%. There is a pressure drop of 1.5 bar in the reheater and then steam enters the L.P turbine at 7 bar and 500ºC in which expansion is continued to a back pressure of 0.04 bar with isentropic efficiency of 85%.Determine i) thermal efficiency ii) specific steam consumption.

UNIT 2

STEAM NOZZLES & STEAM CONDENSERS

Course Objective:

Applications and the principles of thermodynamics to components and systems.

Course Outcome:

To be able to analyze energy distribution in turbines ,nozzles and condensers

Steam Nozzles and Types

Nozzle is a duct by flowing through which the velocity of a fluid increases at the expense of pressure drop. if the fluid is steam, then the nozzle is called as Steam nozzle.
The flow of steam through nozzles may be taken as adiabatic expansion. The steam possesses very high velocity at the end of the expansion, and the enthalpy decreases as expansion occurs. Friction exists between the steam and the sides of the nozzle; heat is produced as the result of the resistance to the flow. The phenomenon of super saturation occurs in the steam flow through nozzles. This is because of the time lag in the condensation of the steam during the expansion.

The area of such duct having minimum cross-section is known as throat.

A fluid is called compressible if its density changes with the change in pressure brought about by the flow.

If the density changes very little or does not changes, the fluid is said to be incompressible. Generallythe gases and vapors are compressible, whereas liquids are incompressible.

Types of Nozzles:

There are three types of nozzles

1. Convergent nozzle
2. Divergent nozzle
3. Convergent-divergent nozzle.

Convergent Nozzle:

A typical convergent nozzle is shown in the Fig.1. In a convergent nozzle, the cross sectional area decreases continuously from its entrance to exit. It is used in a case where the back pressure is equal to or greater than the critical pressure ratio.

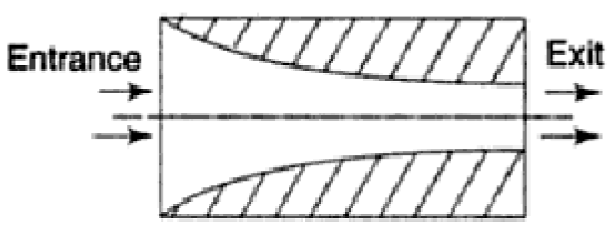


Fig 1. Convergent nozzle

Divergent nozzle:

The cross sectional area of divergent nozzle increases continuously from its entrance to exit. It is used in a case where the back pressure is less than the critical pressure ratio.

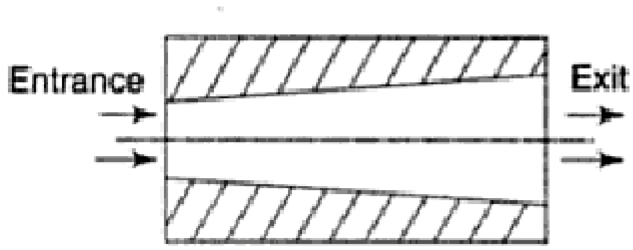


Fig 2. Divergent nozzle

Convergent – Divergent nozzle:

In this condition, the cross sectional area first decreases from its entrance to the throat and then again increases from throat to the exit. This case is used in the case where the back pressure is less than the critical pressure. Also, in present day application, it is widely used in many types of steam turbines.

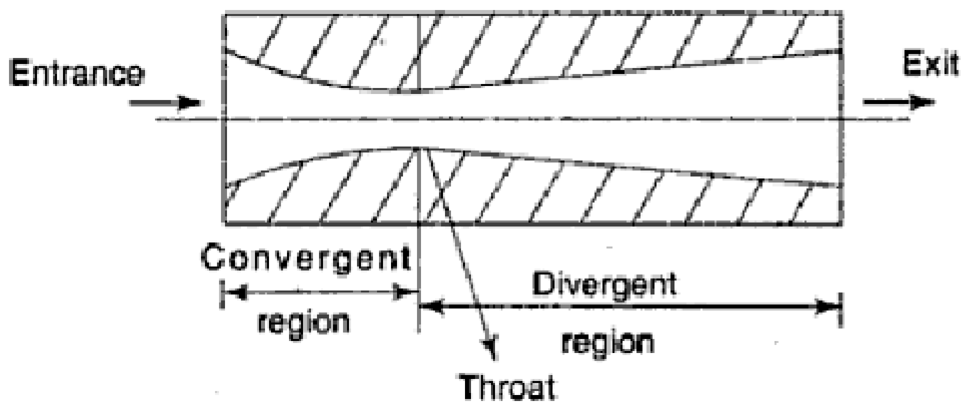


Fig 3. Convergent-Divergent nozzle

Flow of steam Through Nozzle

Supersaturated flow or metastable flow of in Nozzles: As steam expands in the nozzle, the pressure and temperature in it drop, and it is likely that the steam start condensing when it strikes the saturation line. But this is not always the situation. Due to the high velocities, the time up to which the steam resides in the nozzle is small, and there may not be sufficient time for the needed heat transfer and the formation of liquid droplets due to condensation. As a result, the condensation of steam is delayed for a while. This phenomenon is known as super saturation, and the steam that remains in the wet region without holding any liquid is known as supersaturated steam. The locus of points where condensation occurs regardless of the initial temperature and pressure at the entrance of the nozzle is called the Wilson line. The Wilson line generally lies between 4 and 5 percent moisture curves in the saturation region on the h-s diagram in case of steam, and is often taken as 4 percent moisture line. The phenomenon of super saturation is shown on the h-s chart below:

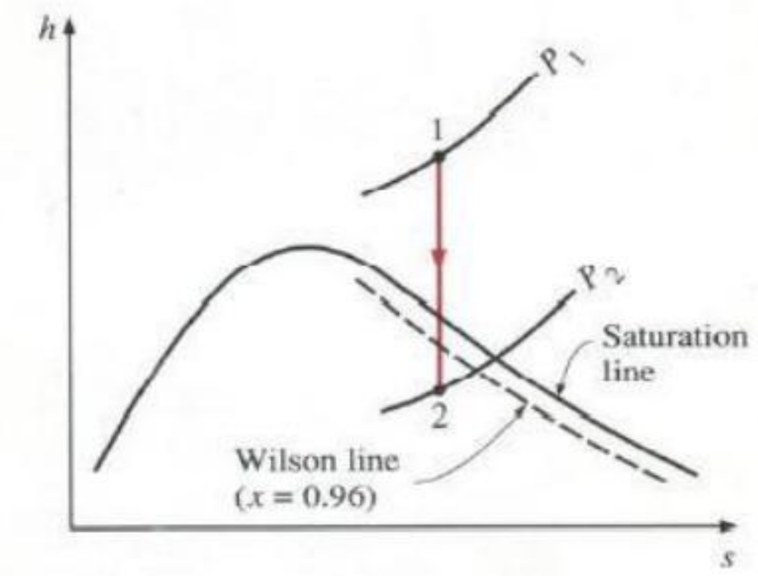


Fig 4. The h-s diagram for the expansion of steam in the nozzle

Effects of Supersaturation:

The following are the effects of supersaturation in a nozzle.

1. The temperature at which the steam becomes supersaturated will be less than the saturation temperature corresponding to that pressure. Therefore, supersaturated steam will have the density more than that of equilibrium condition which results in the increase in the mass of steam discharged.
2. Supersaturation causes the specific volume and entropy of the steam to increase.
3. Supersaturation reduces the heat drop. Thus the exit velocity of the steam is reduced.
4. Supersaturation increases the dryness fraction of the steam.

Effect of Friction on Nozzles:

1. Entropy is increased.
2. The energy available decreases.
3. Velocity of flow at the throat get decreased.
4. Volume of flowing steam is decreased.
5. Throat area required to discharge a given mass of steam is increased.

Continuity and steady flow energy equations through a certain section of the nozzle:

Where m denotes the mass flow rate, v is the specific volume of the steam, A is the area of cross-section and C is the velocity of the steam.

For steady flow of the steam through a certain apparatus, principle of conservation of energy states:

$h1 + C12/2 + gz1 + q = h2 + C22/2 + gz2 + w$

For nozzles, changes in potential energies are negligible, w = 0 and q = 0.

h1 + C12 /2 = h2 + C22 /2
which is the expression for the steady state flow energy equation.

i.e., $$0.25 = \frac{\frac{\pi}{4}(D_2)^2 \times 529}{0.22 \times 10^4}$$

$$\therefore\ (D_2)^2 = \frac{0.25 \times 0.22 \times 10^4 \times 4}{\pi \times 529} = 1.324$$

$\therefore$ Diameter, $D_2 = \sqrt{1.324} = 1.15$ cm i.e., **11·5 mm**

Diameter of the section of the nozzle at a point where the pressure is 9·5 bar = 11·5 mm.

(ii) For exit :

From $H - \Phi$ chart, Enthalpy drop from inlet to exit, $H_e = H_1 - H_3 = 600$ kJ/kg and dryness fraction, $x_3 = 0.89$.

Velocity at exit, $V_3 = 44.72\sqrt{H_e} = 44.72\sqrt{600} = 1{,}095$ m/sec.

From steam tables, at 0·7 bar, $v_{s3} = 2.365$ m^3/kg.

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0.89 \times 2.365$ m^3/kg.

For mass continuity, $m = \dfrac{A_3 V_3}{v_3}$

$$\therefore\ A_3 = \frac{\pi}{4}(D_3)^2 \times \frac{1}{10^4} = \frac{m \times v_3}{V_3} = \frac{0.25 \times (0.89 \times 2.365)}{1{,}095}$$

$$\therefore\ (D_3)^2 = \frac{0.25 \times 0.89 \times 2.365 \times 10^4 \times 4}{\pi \times 1{,}095} = 6.12$$

$\therefore$ Exit diameter, $D_3 = \sqrt{6.12} = 2.47$ cm, i.e., **24·7 mm.**

INTRODUCTION

***A steam nozzle** may be defined as a passage of varying cross-section, through which heat energy of steam is converted to kinetic energy. Its major function is to produce steam jet with high velocity to drive steam turbines.* A turbine nozzle performs *two functions* :

(*i*) It transforms a portion of energy of steam (obtained from steam generating unit) into kinetic energy.

(*ii*) In the impulse turbine it directs the steam jet of high velocity against blades, which are free to move in order to convert kinetic energy into shaft work. In reaction turbines the nozzles which are free to move, discharge high velocity steam. The reactive force of the steam against the nozzle produces motion and work is obtained.

The cross-section of a nozzle at first tapers to a smaller section (*to allow for changes which occur due to changes in velocity, specific volume and dryness fraction as the steam expands*) ; the smallest section being known as ***throat***, and then it diverges to a large diameter. The nozzle which converges to throat and diverges afterwards is known as ***convergent-divergent*** nozzle (Fig. 18.1). In convergent nozzle there is no divergence after the throat as shown in Fig. 18.2.

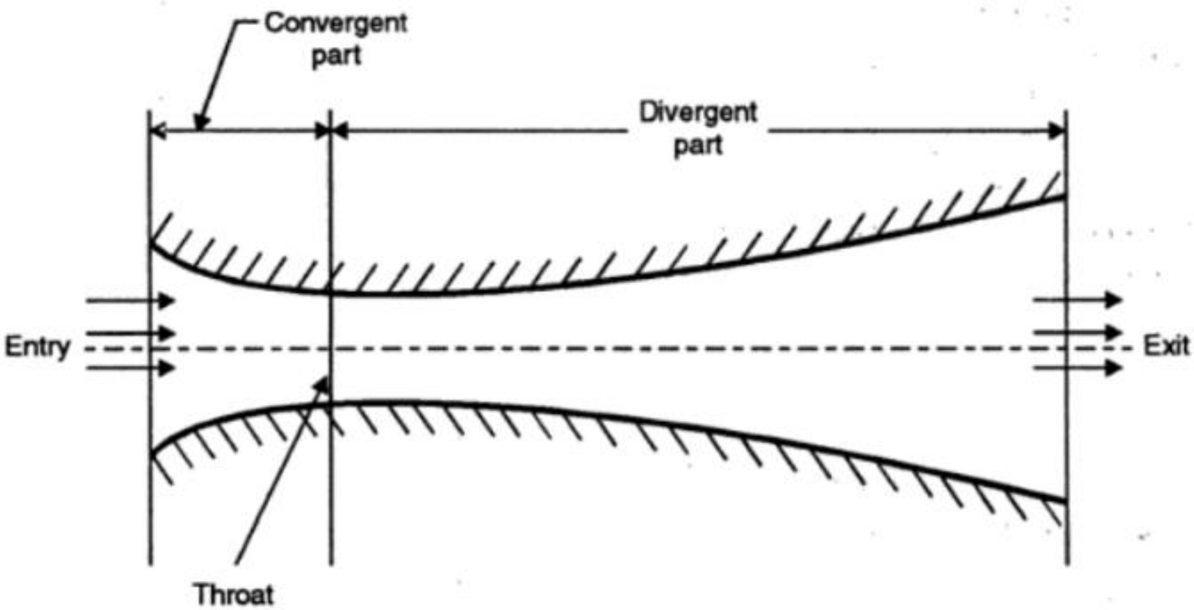


Fig. 18.1. Convergent-divergent nozzle.

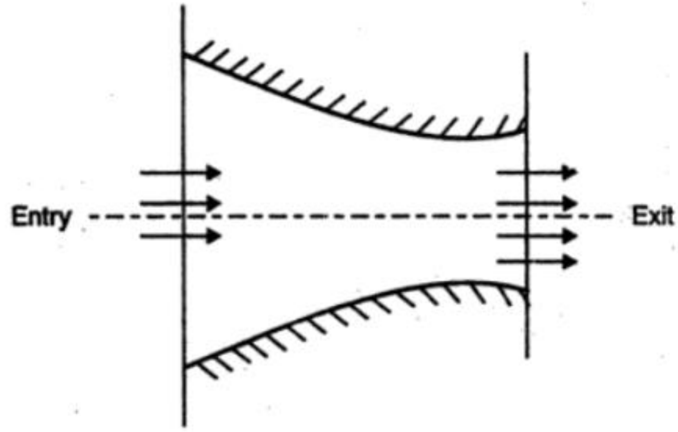


Fig. 18.2. Convergent nozzle.

In a *"convergent-divergent nozzle", because of the higher expansion ratio, addition of divergent portion produces steam at higher velocities as compared to a convergent nozzle.*

STEAM FLOW THROUGH NOZZLES

The steam flow through the nozzle may be assumed as *adiabatic flow* since during the expansion of steam in nozzle neither any heat is supplied nor rejected, work, however, is per formed by increasing the kinetic energy of the steam. As the steam passes through the nozzle it loses its pressure as well as the heat. *The work done is equal to the adiabatic heat drop which in turn is equal to Rankine area.*

Velocity of Steam

Steam enters the nozzle with high pressure and low initial velocity (it is so small as compared to the final velocity that it is generally *neglected*) and leaves it with high velocity and low pressure. This is due to the reason that heat energy of steam is *converted* into kinetic energy as it (steam) passes through the nozzle. The final or outlet velocity of steam can be found as follows :

Let C = Velocity of steam at the section considered (m/sec),

h_1 = Enthalpy of steam entering the nozzle,

h_2 = Enthalpy of steam at section considered, and

h_d = Heat drop during expansion of steam is the nozzle = $(h_1 - h_2)$.

Considering 1 kg of steam and flow to be frictionless adiabatic, we have :

Gain in kinetic energy = Adiabatic heat drop

$$\frac{C^2}{2} = h_d$$

$$\therefore \quad C = \sqrt{2 \times 1000\, h_d}\text{ , where } h_d \text{ is in kJ}$$

$$= 44.72\ \sqrt{h_d} \qquad \text{...(18.1)}$$

In practice, there is loss due to friction in the nozzle and its value varies from 10 to 15 per cent of total heat drop. Due to this, total heat drop is minimized. Let heat drop after deducting friction loss be kh_d.

The velocity, $C = 44.72\ \sqrt{kh_d}$ ...(18.2)

18.2.2. Discharge through the Nozzle and Conditions for its Maximum Value :

Let p_1 = Initial pressure of steam,

v_1 = Initial volume of 1 kg of steam at pressure p_1 (m^3),

p_2 = Steam pressure at the throat,

v_2 = Volume of 1 kg of steam at pressure p_2 (m^3),

A = Cross-sectional area of nozzle at throat (m^2), and

C = Velocity of steam (m/s).

The steam flowing through the nozzle follows approximately the equation given below :

$$pv^n = \text{Constant}$$

where, n = 1.135 for **saturated steam**

and = 1.3 for **superheated steam.**

[For **wet steam,** the value of n can be calculated by Dr. *Zenner's equation,*

$n = 1.035 + 0.1x$, where x is the initial dryness fraction of steam]

Work done per kg of steam during the cycle (Rankine area)

$$= \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

and, Gain in kinetic energy = Adiabatic heat drop

= Work done during Rakine cycle

or
$$\frac{C^2}{2} = \frac{n}{n-1}(p_1v_1 - p_2v_2)$$

$$= \frac{n}{n-1}\, p_1v_1\left(1 - \frac{p_2v_2}{p_1v_1}\right) \qquad ...(18.3)$$

Also
$$p_1v_1^{\,n} = p_2v_2^{\,n}$$

or
$$\frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n} \qquad ...(18.4)$$

or
$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{1/n} \qquad ...(18.5)$$

Putting the value of v_2/v_1 from eqn. (18.4) in eqn. (18.3), we get

$$\frac{C^2}{2} = \frac{n}{n-1}\, p_1v_1\left[1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right] = \frac{n}{n-1}\, p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}\right]$$

$$= \frac{n}{n-1}\, p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C^2 = 2\left(\frac{n}{n-1}\right) p_1v_1\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]$$

$$C = \sqrt{2\left(\frac{n}{n-1}\right) p_1 v_1 \left\{1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right\}} \qquad ...(18.6)$$

If m is the mass of steam discharged in kg/sec.,

Then $$m = \frac{AC}{v_2} \qquad ...(18.7)$$

Substituting the value of v_2 from eqn. (18.5) in eqn. (18.7),

$$m = \frac{AC}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}}$$

or $$m = \frac{A}{v_1\left(\frac{p_1}{p_2}\right)^{1/n}} \sqrt{2\left(\frac{n}{n-1}\right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \frac{A}{v_1} \sqrt{\left[2\left(\frac{n}{n-1}\right) p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}} \left(\frac{p_2}{p_1}\right)^{2/n}\right\}\right]}$$

$$= \frac{A}{v_1} \sqrt{\left[2\left(\frac{n}{n-1}\right) p_1 v_1 \left\{\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}\right]} \qquad ...(18.8)$$

A convergent-divergent nozzle is required to discharge 350 kg of steam per hour. The nozzle is supplied with steam at 8.5 bar and 90% dry and discharges against a back pressure of 0.4 bar. Neglecting the effect of friction, find the throat and exit diameters.

Let suffixes, 1,2 and 3 represent conditions at entry, throat and exit of the nozzle respectively as shown in fig. 8-6.

As the steam is initially wet, critical or throat pressure,

$p_2 = 0{\cdot}582 \times p_1 = 0{\cdot}582 \times 8{\cdot}5 = 4{\cdot}95$ bar.

As shown in fig. 8-6, vertical line 1-2-3 is drawn. The values read off from the $H - \Phi$ chart (Mollier chart) are :

Enthalpy drop from entry to throat, $H_t = H_1 - H_2 = 102$ kJ/kg,

Enthalpy drop from entry to exit, $H_e = H_1 - H_3 = 456$ kJ/kg,

Dryness fraction of steam at throat, $x_2 = 0{\cdot}87$ and

Dryness fraction of steam at exit, $x_3 = 0{\cdot}777$

Velocity at throat, $V_2 = 44.72\sqrt{H_t} = 44{\cdot}72\sqrt{102} = 452$ m/sec. [eqn. (8.5)]

Specific volume of dry saturated steam at 4.95 bar (by arithmetical interpolation from steam tables), $v_{s2} = 0.3785$ m^3/kg.

$\therefore$ Actual volume of wet steam at throat,

$v_2 = x_2 \times v_{s2} = 0{\cdot}87 \times 0{\cdot}3785 = 0{\cdot}33$ m^3/kg.

For mass continuity, $m = \dfrac{A_2 V_2}{v_2}$ kg/sec. [eqn. (8.2)]

i.e., $\dfrac{350}{3{,}600} = \dfrac{A_2 \times 452}{0{\cdot}33 \times 10^4}$

$\therefore A_2 = \dfrac{350 \times 0{\cdot}33 \times 10^4}{3{,}600 \times 452} = 0{\cdot}71 \text{ cm}^2$

$\therefore$ Throat diameter, $D_2 = \sqrt{\dfrac{0{\cdot}71 \times 4}{\pi}} = 0{\cdot}951$ cm i.e., **9·51 mm**

Similarly velocity at exit,

$V_3 = 44{\cdot}72\sqrt{456} = 955$ m/sec.

Specific volume of dry saturated steam at 0.4 bar (from steam tables),

$v_{s3} = 3{\cdot}993$ m^3/kg.

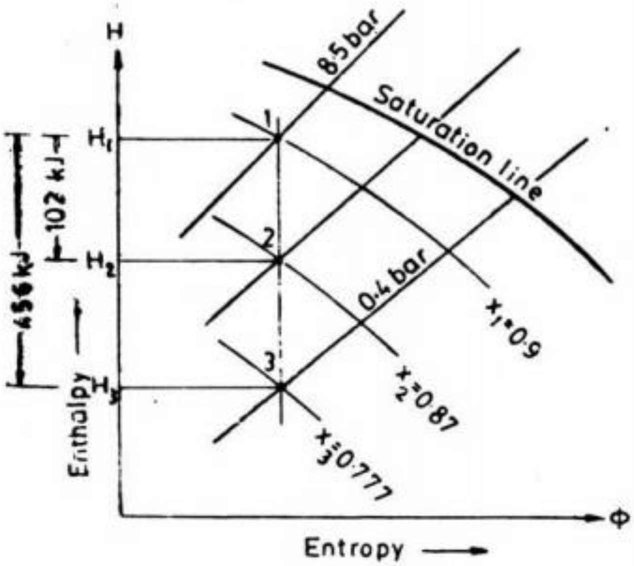


Fig. 8-6. H - φ diagram.

∴ Actual volume of wet steam at exit, $v_3 = x_3 \times v_{s3}$

$= 0.777 \times 3.993 = 3.11\ m^3/kg.$

Again for mass continuity,

$$m = \frac{A_3 V_3}{v_3} \qquad [\text{eqn. } (8.2)]$$

i.e., $$\frac{350}{3,600} = \frac{A_3 \times 955}{3.11 \times 10^4}$$

$$\therefore\ A_3 = \frac{350 \times 3.11 \times 10^4}{955 \times 3,600} = 3.16\ cm^2$$

∴ Exit diameter, $D_3 = \sqrt{\frac{3.16 \times 4}{\pi}} = 2.01$ cm i.e., **20.1mm**

Problem – 3 : *An impulse turbine which is to develop 175 kW with probable steam consumption of 11 kg per kW-hour is supplied with dry saturated steam at 10 bar. Find the number of nozzles each of about 6 mm diameter at the throat that will be required for the purpose and estimate the exact diameters at the throat and exit of the nozzles. The condenser pressure is 0.15 bar. Neglect the effect of friction in nozzles. Assume index of expansion as 1.135.*

Let suffixes, 1, 2 and 3 represent conditions at entry, throat and exit of the nozzle.

From eqn. (8.9), $\frac{p_2}{p_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$

Putting $n = 1.135$, $\frac{p_2}{p_1} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}} = (0.936)^{8.4} = 0.578$

Critical or throat pressure, $p_2 = 0.578 \times p_1 = 0.578 \times 10 = 5.78$ bar.

Enthalpy drop from entry to throat, $H_t = H_1 - H_2 = 122$ kJ/kg and

Dryness fraction of steam at throat, $x_2 = 0{\cdot}957$ (from $H - \Phi$ chart).

Velocity at throat, $V_2 = 44{\cdot}72\sqrt{H_t} = 44{\cdot}72\sqrt{122} = 494$ m/sec.,

From steam tables at 5·78 bar, $v_{s2} = 0{\cdot}327$ m^3/kg by arithmetical interpolation.

Specific volume at throat, $v_2 = x_2 \times v_{s2} = 0{\cdot}957 \times 0{\cdot}327$ m^3/kg.

For mass continuity, $m = \dfrac{A_2 V_2}{v_2} = \dfrac{\dfrac{\pi}{4}\left(\dfrac{6}{10}\right)^2 \times 494}{10^4 \times 0{\cdot}957 \times 0{\cdot}327} = 0{\cdot}0446$ kg/sec.

Steam consumption per sec. $= \dfrac{11 \times 175}{3{,}600} = 0{\cdot}5347$ kg/sec.

$\therefore$ Number of nozzles required $= \dfrac{0{\cdot}5347}{0{\cdot}0446} = 11{\cdot}99$ say 12

$\therefore$ Exact diameter at throat, $D_2 = 6\sqrt{\dfrac{11{\cdot}99}{12}} =$ **5·997 mm.**

For exit :

Enthalpy drop from entry to exit, $H_e = H_1 - H_3 = 655$ kJ/kg and

Dryness fraction of steam at exit, $x_3 = 0{\cdot}85$ (from $H - \Phi$ chart).

$\therefore$ Velocity at exit, $V_3 = 44{\cdot}72\sqrt{H_e} = 44{\cdot}72\sqrt{655} = 1{,}145$ m/sec.

From steam tables at 0.15 bar, $v_{s_3} = 10{\cdot}022$ m^3/kg.

$\therefore$ Taking the number of nozzles as 12,

Mass of steam per nozzle $= \dfrac{0{\cdot}5347}{12} = 0{\cdot}0446$ kg/sec.

Again for mass continuity, $m = \frac{A_3 V_3}{v_3}$ or $A_3 = \frac{m v_3}{V_3} = \frac{m \times V_{s_3} \times x_3}{V_3}$

$$\therefore A_3 = \frac{0.0446 \times (10.022 \times 0.85) \times 10^4}{1,145} = 3.646 \text{ cm}^2.$$

$\therefore$ Exact diameter at exit, $D_3 = \sqrt{\frac{3.646 \times 4}{\pi}} = 2.155$ cm i.e., **21.55 mm.**

Problem – 4 : *Steam expands from 17 bar and 80°C superheat to 0.7 bar in a*

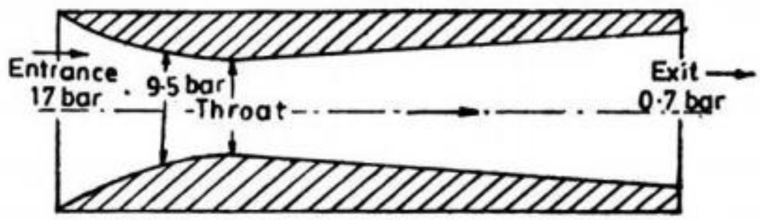


Fig. 8-7

convergent-divergent nozzle. Assuming that the expansion is frictionless adiabatic, and the steam discharged is 0.25 kg/sec., calculate the diameters of the sections of nozzle (i) at a point where the pressure is 9.5 bar, and (ii) at exit.

Take K_p *of superheated steam as 2.3 kJ/kg K.*

Referring to fig. 8-7,

Let suffixes, 1,2 and 3 represent conditions at entrance, section of the nozzle where pressure is 9·5 bar and exit respectively.

As the steam is initially superheated,

critical or throat pressure $= 0{\cdot}546 \times p_1 = 0{\cdot}546 \times 17 = 9{\cdot}28$ bar.

It means that the nozzle is still converging where the pressure is 9·5 bar.

(1) For section of the nozzle where the pressure is 9·5 bar :

Enthalpy drop from entry to section of nozzle, where the pressure is 9·5 bar,

$H_1 - H_2 = 140$ kJ/kg (from $H - \Phi$ chart);

Temperature of steam, $t_2 = 213°C$ (from $H - \Phi$ chart).

At 9·5 bar, saturation temperature, $t_s = 177{\cdot}69°C$ (from steam tables).

$\therefore$ Steam at section where pressure is 9·5 bar is superheated, i.e., steam is still superheated after expansion.

At 9.5 bar, $v_{s2} = 0.2042$ m^3/kg (from steam tables).

Specific volume at 9·5 bar and 213°C,

$$v_2 = V_{s_2} \times \frac{T_{sup2}}{T_{sat2}} = 0{\cdot}2042 \times \frac{(213 + 273)}{(177{\cdot}69 + 273)} = 0{\cdot}22 \text{ m}^3\text{/kg}$$

Velocity at section, where pressure is 9·5 bar, $V_2 = 44{\cdot}72\sqrt{140} = 529$ m/sec.

For mass continuity, $m = \dfrac{A_2 V_2}{v_2}$

i.e. $0{\cdot}25 = \dfrac{\frac{\pi}{4}(D_2)^2 \times 529}{}$

i.e., $0{\cdot}25 = \dfrac{\frac{\pi}{4}(D_2)^2 \times 529}{0{\cdot}22 \times 10^4}$

$\therefore\ (D_2)^2 = \dfrac{0{\cdot}25 \times 0{\cdot}22 \times 10^4 \times 4}{\pi \times 529} = 1{\cdot}324$

$\therefore$ Diameter, $D_2 = \sqrt{1{\cdot}324} = 1{\cdot}15$ cm i.e., **11·5 mm**

Diameter of the section of the nozzle at a point where the pressure is 9·5 bar = 11·5 mm.

(ii) For exit :

From $H - \Phi$ chart, Enthalpy drop from inlet to exit, $H_e = H_1 - H_3 = 600$ kJ/kg and dryness fraction, $x_3 = 0.89$.

Velocity at exit, $V_3 = 44{\cdot}72\sqrt{H_e} = 44{\cdot}72\sqrt{600} = 1{,}095$ m/sec.

From steam tables, at 0·7 bar, $v_{s3} = 2{\cdot}365$ m^3/kg.

Specific volume at exit, $v_3 = x_3 \times v_{s3} = 0{\cdot}89 \times 2{\cdot}365$ m^3/kg.

For mass continuity, $m = \dfrac{A_3 V_3}{v_3}$

$\therefore\ A_3 = \dfrac{\pi}{4}(D_3)^2 \times \dfrac{1}{10^4} = \dfrac{m \times v_3}{V_3} = \dfrac{0{\cdot}25 \times (0{\cdot}89 \times 2{\cdot}365)}{1{,}095}$

$\therefore\ (D_3)^2 = \dfrac{0{\cdot}25 \times 0{\cdot}89 \times 2{\cdot}365 \times 10^4 \times 4}{\pi \times 1{,}095} = 6{\cdot}12$

$\therefore$ Exit diameter, $D_3 = \sqrt{6{\cdot}12} = 2{\cdot}47$ cm, i.e., **24·7 mm.**

Steam Condensers

INTRODUCTION

A steam condenser *is a device or an appliance in which steam condenses and heat released by steam is absorbed by water*. It serves the following purposes :

1. It *maintains a very low back pressure* on the exhaust side of the piston of the steam engine or turbine. Consequently, the steam expands to a greater extent which results in an *increase in available heat energy* for converting into mechanical work. The shaded area in Fig. 20.1. (*i.e.*, area 44′5′5) shows the increase in work obtained by fitting a condenser to a non-condensing engine. *The therm.al efficiency of a condensing unit therefore is higher than that of non-condensing unit for the same available steam.*

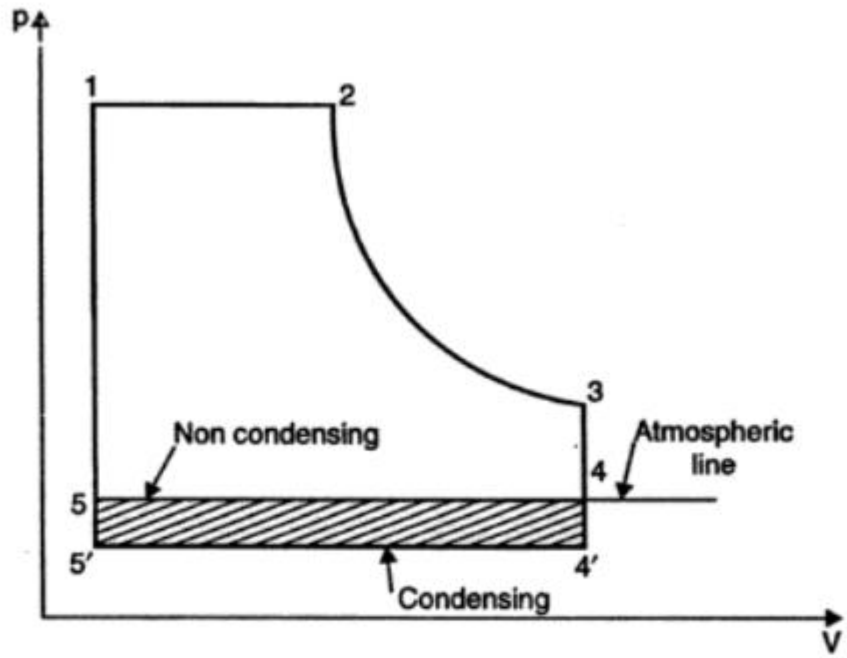


Fig. 20.1

2. It supplies to the boiler pure and hot feed water as the condensed steam which is discharged from the condenser and collected in a hot well, can be used as feed water for the boiler.

20.2. VACUUM

Vacuum is *sub-atmospheric pressure*. It is measured as the *pressure depression below atmospheric*. The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low pressure steam or vapour. The back pressure in steam engine or steam turbine can lowered from 1.013 to 0.2 bar abs. or even less. Since the steam engines are intermittent flow machines and as such cannot take the advantage of a very low vacuum, therefore, for most steam engines the exhaust pressure is about 0.2 to 0.28 bar abs. On the other hand, in steam turbines, which are continuous flow machines, the back pressure may be about 0.025 bar abs.

ORGANS OF A STEAM CONDENSING PLANT

A steam condensing plant mainly consists of the following *organs/elements* :

1. Condenser (To condense the steam).
2. Supply of cooling (or injection) water.
3. Wet air pump (To remove the condensed steam, the air and uncondensed water vapour and gases from the condenser ; separate pumps may be used to deal with air and condensate).
4. Hot well (where the condensate can be discharged and from which the boiler feed water is taken).
5. Arrangement for recooling the cooling water in case surface condenser is employed.

CLASSIFICATION OF CONDENSERS

Mainly, condensers are of two types : (1) Jet condensers, (2) Surface condenser.

In ***jet condensers***, the *exhaust steam and water come in direct contact with each other and temperature of the condensate is the same as that of cooling water leaving the condenser*. The cooling water is usually sprayed into the exhaust steam to cause, rapid condensation.

In ***surface condensers***, *the exhaust steam and water do not come into direct contact*. The steam passes over the outer surface of tubes through which a supply of cooling water is maintained. There may be single-pass or double-pass. In single-pass condensers, the water flows in one direction only through all the tubes, while in two-pass condenser the water flows in one direction through the tubes and returns through the remainder.

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A jet condenser is simpler and cheaper than a surface condenser. It should be installed when the cooling water is cheaply and easily made suitable for boiler feed or when a cheap source of boiler and feed water is available. A surface condenser is most commonly used because the condensate obtained is not thrown as a waste but returned to the boiler.

20.4.1. Jet Condensers

These condensers may be classified as :

(*a*) Parallel flow type

(*b*) Counter flow type

(*c*) Ejector type.

Parallel flow and counter flow condensers are further sub-divided into two types : (*i*) Low level type (*ii*) High level type.

In *parallel-flow type* of condenser, both the exhaust steam and cooling water find their entry at the top of the condenser and then flow downwards and condensate and water are finally collected at the bottom.

In *counter-flow type*, the steam and cooling water enter the condenser from opposite directions. Generally, the exhaust steam travels in upward direction and meet the cooling water which flows downwards.

Low level jet condenser (Parallel-flow)

In the Fig. 20.2 is shown a line sketch of a low level parallel flow condenser. The exhaust steam is entering the condenser from the top and cold water is being sprayed on its way. The baffle plate provided in it ensures the proper mixing of the steam and cooling water. An extraction pump at the bottom discharges the condensate to the hot well from where it may be fed to the boiler if the cooling water being used is free from impurities. A separate dry pump may be incorporated to maintain proper vacuum.

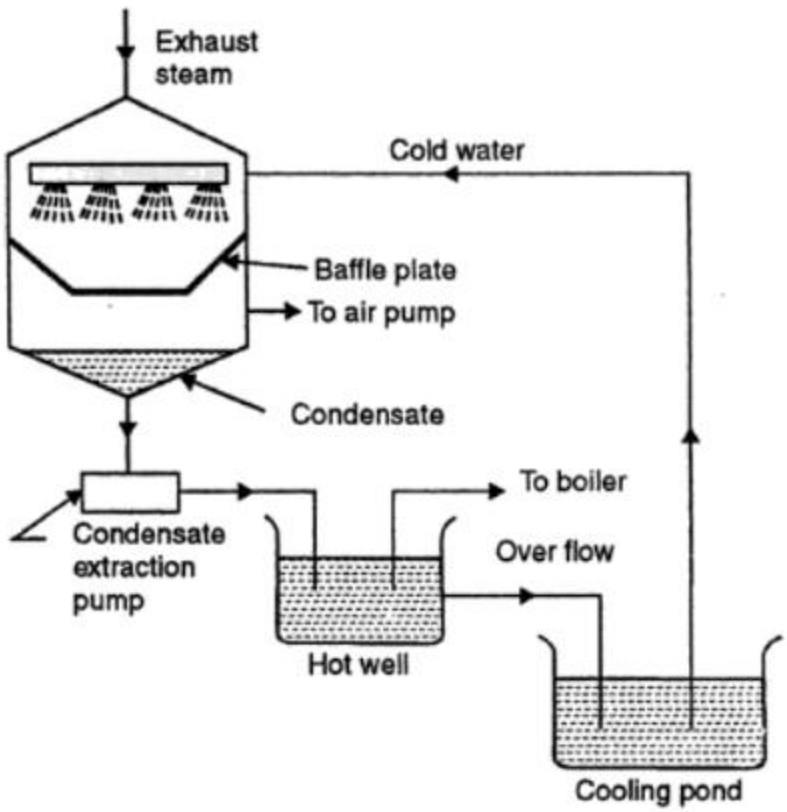


Fig. 20.2. Low level jet condenser (Parallel flow).

Low level jet condenser (Counter-flow)

Refer Fig. 20.3 (on next page). *L, M* and *N* are the perforated trays which break up water into jets. The steam moving upwards comes in contact with water and gets condensed. The condensate and water mixture is sent to the hot well by means of an extraction pump and the air is removed by an air suction pump provided at the top of the condenser.

High level jet condenser (Counter-flow type).

In Fig. 20.4 is shown a high level counter-flow jet condenser. It is also called ***barometric condenser***. In this case the shell is placed at a height about 10.363 metres above hot well and thus the necessity of providing an extraction pump can be obviated. However provision of own injection pump has to be made if water under pressure is not available.

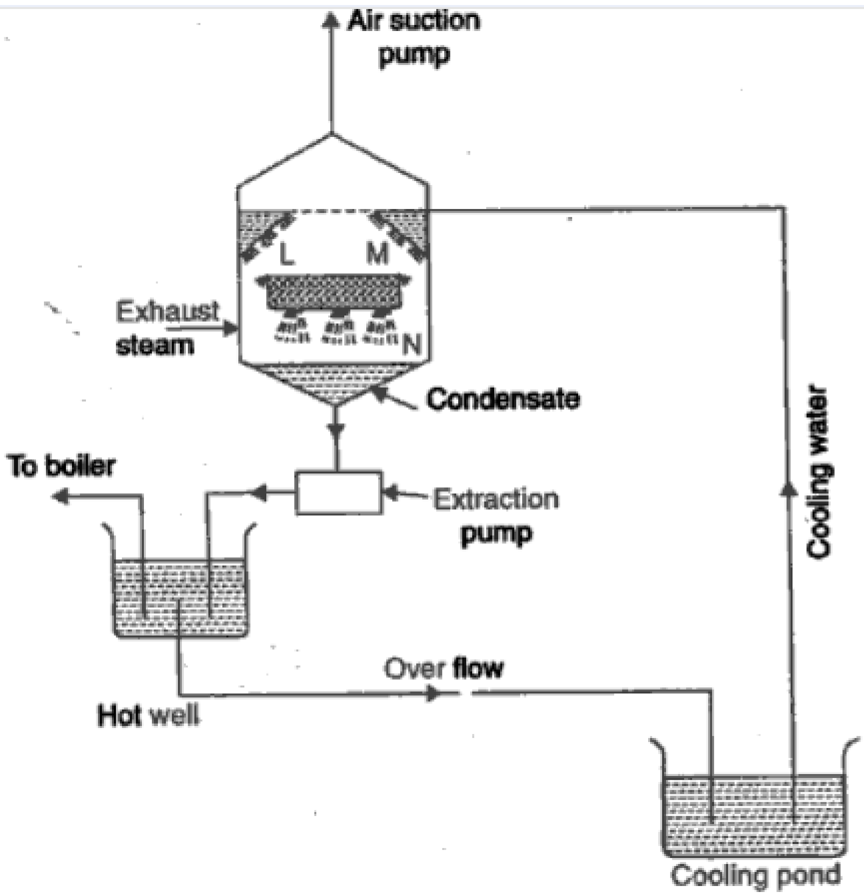


Fig. 20.3. Low level jet condenser (Counter-flow).

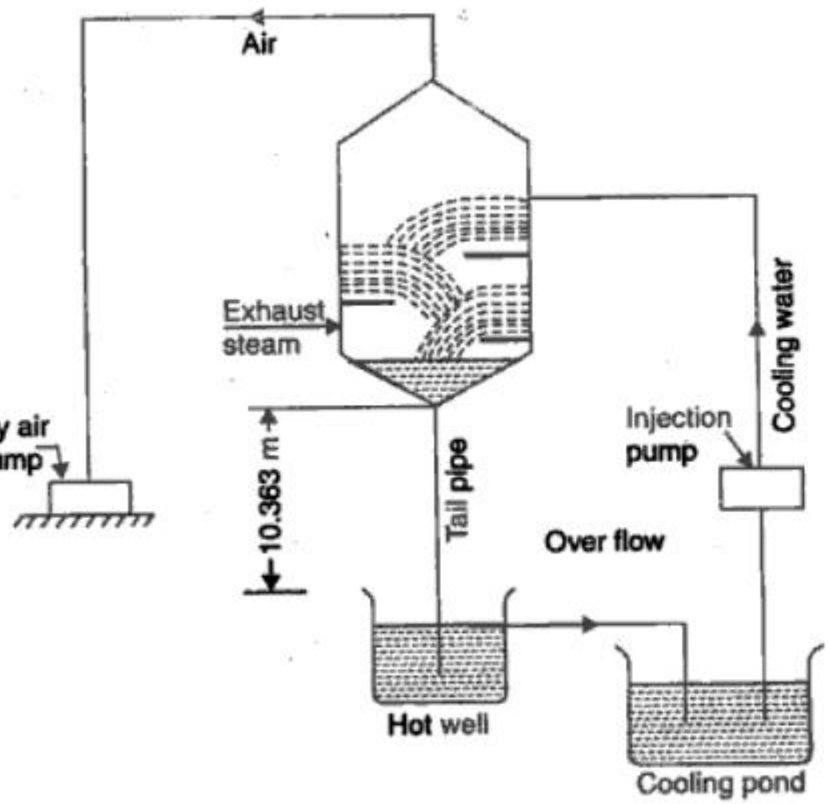


Fig. 20.4. High level jet condenser (Counter-flow type).

Ejector condenser

Fig. 20.5 shows the schematic sketch of an ejector condenser. Here the exhaust steam and cooling water mix in hollow truncated cones. The cold water having a head of about 6 metres flow down through the number of cones and as it moves its velocity increases and drop in pressure results. Due to this decreased pressure exhaust steam along with associated air is drawn through

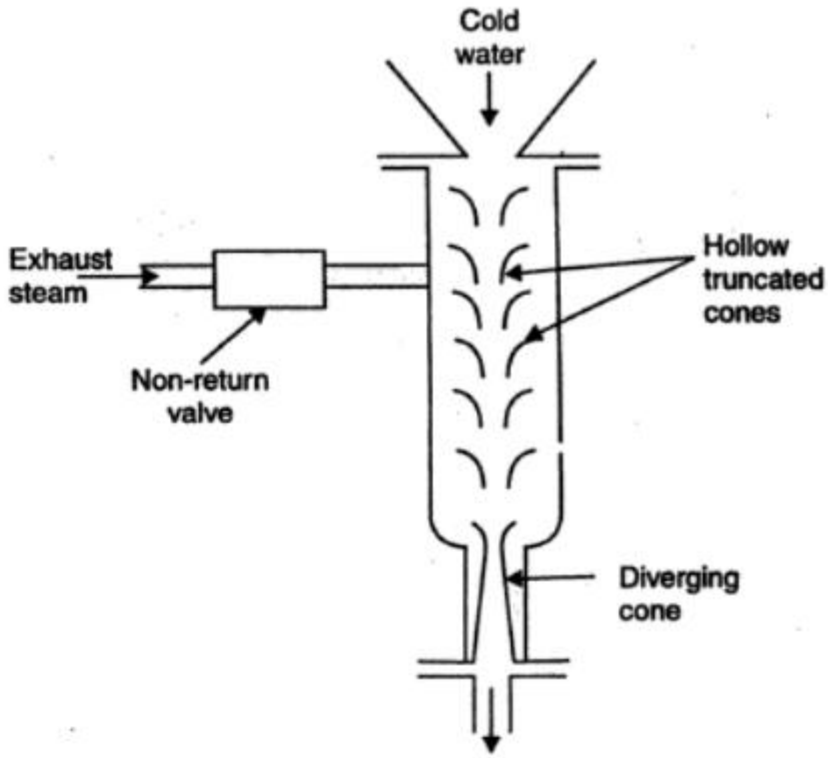


Fig. 20.5. Ejector condenser.

the truncated cones and finally lead to diverging cone. In the diverging cone, a portion of kinetic energy gets converted into pressure energy which is more than the atmospheric so that condensate consisting of condensed steam, cooling water and air is discharged into the hot well. The exhaust steam inlet is provided with a non-return valve which does not allow the water from hot well to rush back to the engine in case a failure of cooling water supply to condenser.

20.4.2. Surface Condensers

Most condensers are generally classified on the direction of flow of condensate, the arrangement of the tubing and the position of the condensate extraction pump. The following is the main classification of surface condensers :

(*i*) Down-flow type
(*ii*) Central-flow type
(*iii*) Inverted-flow type
(*iv*) Regenerative type
(*v*) Evaporative type.

(*i*) Down-flow type

In Fig. 20.6. is shown a down flow type of surface condenser. It consists of a shell which is generally of cylindrical shape ; though other types are also used. It has cover plates at the ends and furnished with number of parallel brass tubes. A baffle plate partitions the water box into two

20.4.4. Comparison between Jet and Surface Condensers

	Jet Condenser	Surface Condenser
1.	Cooling water and steam are mixed up.	Cooling water and steam are not mixed up.
2.	Low manufacturing cost.	High manufacturing cost.
3.	Lower up keep.	Higher upkeep.
4.	Requires small floor space.	Requires large floor space.
5.	The condensate cannot be used as feed water in the boilers unless the cooling water is free from impurities.	Condensate can be reused as feed water as it does not mix with the cooling water.
6.	More power is required for air pump.	Less power is needed for air pump.
7.	Less power is required for water pumping.	More power is required for water pumping.
8.	It requires less quantity of cooling water.	It requires large quantity of cooling water.
9.	The condensing plant is simple.	The condensing plant is complicated.
10.	Less suitable for high capacity plants due to low vacuum efficiency.	More suitable for high capacity plants as vacuum efficiency is high.

20.5. SOURCES OF AIR IN CONDENSERS

The *main sources of air* found in condensers are given below :

1. There is a leakage of air from atmosphere at the joints of the parts which are internally under a pressure less than that of atmosphere. The quantity of air that leaks in can be reduced to a great extent if design and making of the vacuum joints are undertaken carefully.
2. Air is also accompanied with steam from the boiler into which it enters dissolved in feed water. The quantity of air depends upon the treatment which the feed water receives before it enters the boiler. However, the amount of air which enters through this source is relatively small.
3. In jet condensers, a little quantity of air accompanies the injection water (in which it is dissolved).

Note. (*i*) In jet condensers, the quantity of air dissolved in injection water is about 0.5 kg/10000 kg of water.

(*ii*) In surface condensers of reciprocating steam engines, the air leakage is about 15 kg/10000 kg of steam whereas in surface condensers of well designed and properly maintained steam turbine plants the air leakage is about 5 kg/10000 kg of steam.

In order to check whether there is air leakage in the condenser, the following procedure is adopted :

1. Keep the plant running until the temperature and pressure conditions are steady in the condenser.
2. The steam condenser be isolated by shutting off steam supply and simultaneously closing the condensate and air extraction pumps.

In case there is a leakage the readings of vacuum gauge and thermometer will record a fall.

The following methods are used to check the source of air leakage.

1. Put the steam condenser under air pressure and note its effect on soap water at the points where infiltration is likely to occur.
2. Put the peppramerit oil on the suspected joint (when the condenser is operating) and make a check on the peppramerit odour in the discharge of air ejector.
3. Large leakages in steam condenser under vacuum can be detected by moving/passing candle flame over possible openings.

20.6. EFFECTS OF AIR LEAKAGE IN A CONDENSER

The following are the *effects of air leakage in a condenser* :

1. **Lowered thermal efficiency.** The leaked air in the condenser results in increased back-pressure on the primemover which means there is loss of heat drop and consequently thermal efficiency of steam power plant is lowered.

2. **Increased requirement of cooling water**. The leaked air in the condenser lowers the partial pressure of steam which means a lowered saturation temperature of steam. As the saturation temperature of steam lowers, its latent heat increases. So it will require increased amount of cooling water for increased latent heat.

3. **Reduced heat transfer**. Air has poor thermal conductivity. Hence leaked air reduces the rate of heat transfer from the vapour, and consequently it requires surface of the tubes of a surface condenser to be increased for a given condenser capacity.

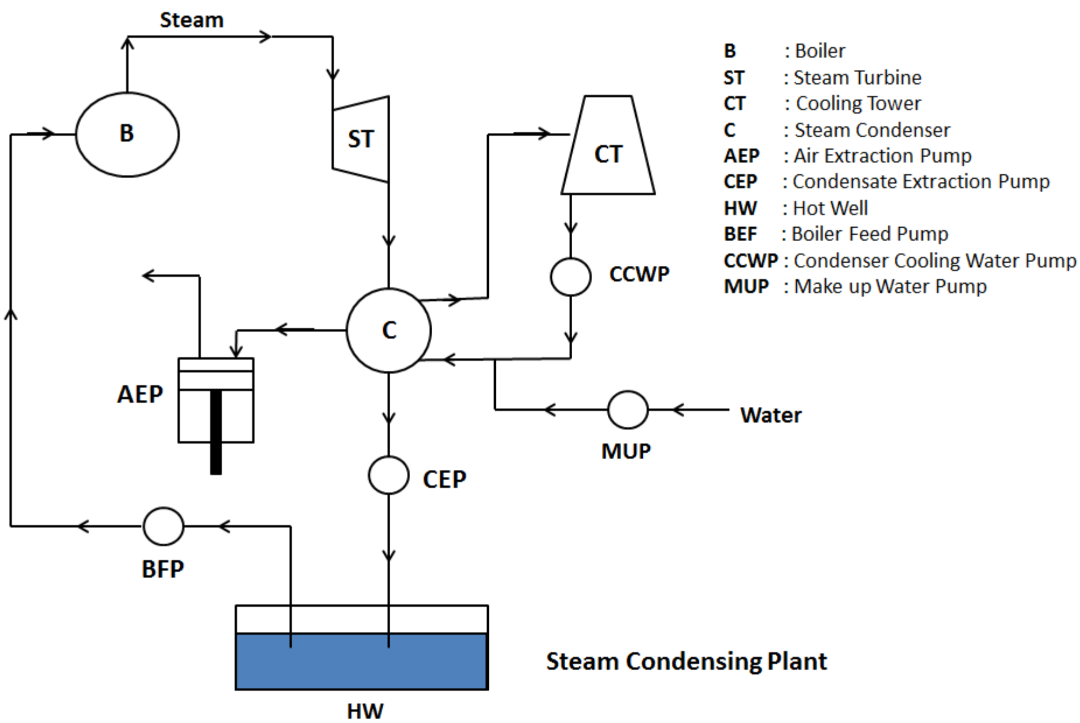
4. **Corrosion.** The presence of air in the condenser increases the corrosive action.

20.7. METHODS FOR OBTAINING MAXIMUM VACUUM IN CONDENSERS

Following are some of the methods used to obtain *maximum possible vacuum in condensers used in modern steam power plants.*

1. **Air pump.** Air pumps are provided to maintain a desired vacuum in the condenser by extracting the air and other non-condensable gases. They are usually classified as : (*a*) ***Wet air pumps*** which remove a *mixture of condensate and non-condensable gases.* (*b*) ***Dry air pump*** which removes the *air only*.

2. **Steam air ejector.** When a wet air pump (also called extraction pump) is employed then use is made of steam air ejectors to remove air from the mixture. The operation of the ejector consists in utilising the viscous drag of a high velocity steam jet for the ejection of air and other non-condensable gases from a chamber ; it is chiefly used for exhausting the air from steam condensers. In the case of ejectors used for steam plants where a high vacuum pressure is main-



Steam Condensing Plant

Requirements of Steam Condensing Plant

1. Condenser:

It is a closed vessel used to condense the steam. The low pressure steam gives off its heat to the coolant (here water from cooling tower) and gets converted into water during the process of condensation.

2. Condensate Extraction Pump: It is a pump which is installed in between the condenser and hot well. It transfers the condensate from the condenser to the hot well.

3. Hot Well: It is a sump that lies in between the condenser and boiler. It receives the condensate from the condenser by condensate pump. The feed water is transferred from the hot well to the boiler.

4. Boiler Feed Pump: It is a pump installed in between the hot well and boiler. It pumps the feed water from the hot well to the boiler. And this is done by increasing the pressure of condensate above boiler pressure.

5. Air Extraction Pump: It is a pump used to extracts or removes the air from the steam condenser.

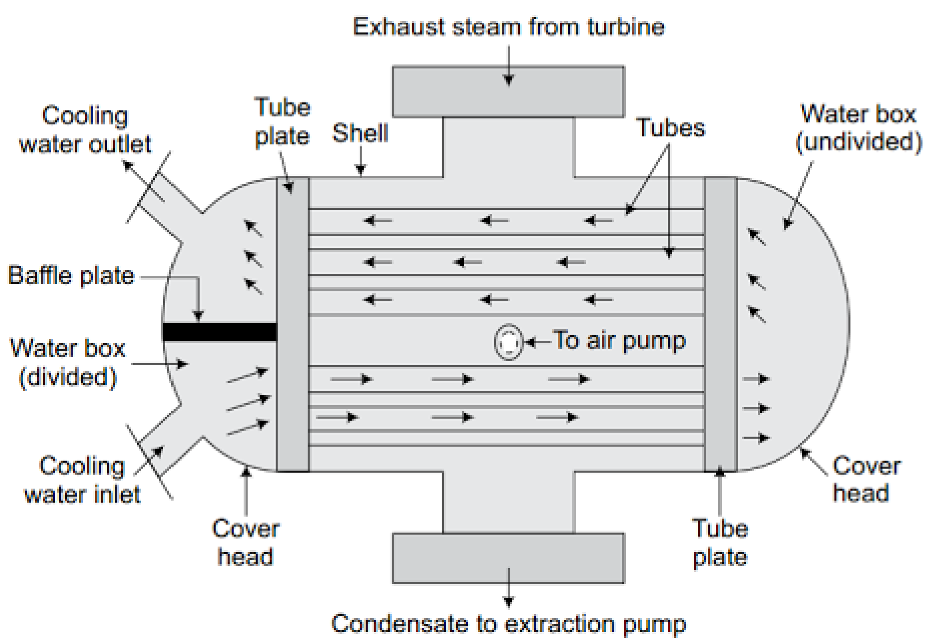
6. Cooling Tower: It is a tower which contains the cold water and this water is made to circulate within the condenser for cooling of steam.

7. Cooling Water Pump: It is a pump lies in between the cooling tower and condenser. It circulates the cooling water through the condenser.

Working

The steam condenser receives the exhaust steam from one end and comes in contact with the cooling water circulated within it form the cooling tower. As the low pressure steam comes in contact with the cooling water, it condenses and converts into water. It is connected to the air extraction pump and condensate extraction pump. After the condensation of steam, the condensate is pumped to the hot well with the help of condensate extraction pump. The air extraction pump extracts the air from the condenser and creates the vacuum inside it. The vacuum created helps in the circulation of cooling water and flow of condensate downward.

Surface Condensers



Surface Condenser

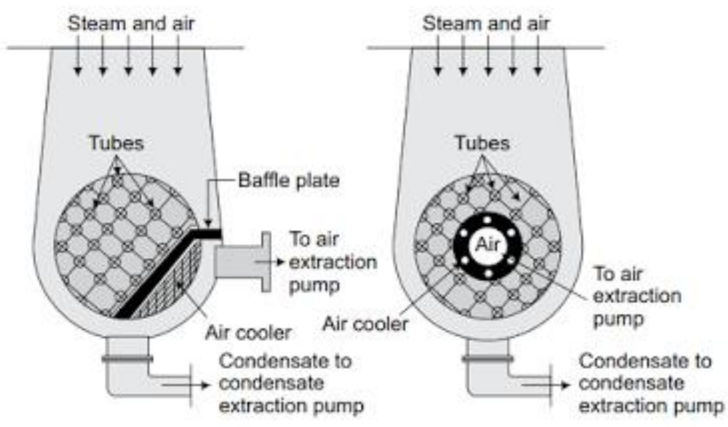
Surface condenser is a type of steam condenser in which the steam and cooling water do not mix with each other. And because of this, the whole condensate can be used as boiler feed water. It is also called as non-mixing types condenser.

The figure above shows the longitudinal section of a two pass surface condenser. It consists of a horizontal cylindrical vessel made of cast iron and packed with tubes. The cooling water flows through these tubes. The ends of the condensers are cut off by the perforated type plates. The tubes are fixed into these perforated type plates. It is fixed in such a manner that any leakage of water into the center of condensing space is prevented. The water tubes are passed horizontally through the main condensing space. The exhaust steam from the turbine or engine enters at the top and forced to move downward due to the suction of the air extraction pump. In this steam condenser, the cooling water enters into boiler through lower half of the tubes in one direction and returns in opposite direction through the upper half as shown in the figure above.

This type of condenser is used in ships as it can carry only a limited quantity of water for the boiler. It is also widely used for the land installation where there is a scarcity of good quality of water.

Types of Surface Condensers
The surface condenser on the basis of direction of flow of condensate, the arrangement of the tubing system and the position of the extraction pump are classified as

(i) Down Flow



Down Flow **Central Flow**

In Down flow surface condenser, the steam enters at the top of the condenser and flows downwards over the tubes due to the gravity and air extraction pumps. The condensate gets collected at the bottom and then pumped with the help of condensate extraction pump. The pipe of dry air extraction pump is provided near the bottom and it is covered by baffle plates so as to prevent the entry of the condensate into it.

The steam in down flow condenser flows perpendicular to the direction of flow of cooling water, so it is also called as cross-surface condenser.

(ii) Central Flow
In central flow condenser, the steam enters at the top of the condenser and flows in downward direction. In this the suction pipe of the air extraction pump is provided in the center of the tube nest as shown in the figure. Due to this placement of the suction pipe in the center of the tube nest, the exhaust

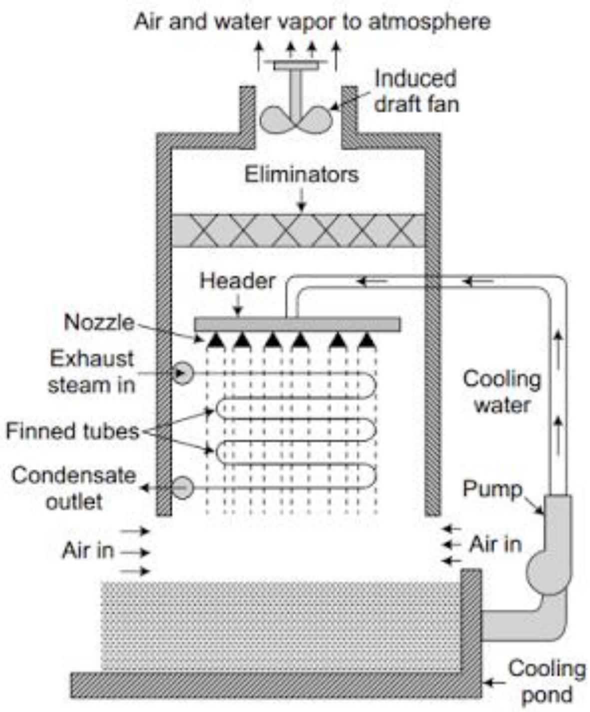
steam flows radially inward over the tubes towards the suction pipe. The condensate is collected at the bottom of the condenser and pumped to the hot well.

We can say that it is the improved form of the down flow surface condenser.

(iii) Regenerative

In regenerative surface condensers, the condensate is heated by the use of regenerative method. In that the condensate is passed through the exhaust steam coming out from the turbine or engine. This raises its temperature and it is used as the feed water for the boiler.

(iv) Evaporative



Evaporative Condenser

In evaporative surface condensers, the steam enters at the top of the condenser in a series of pipes over which a film of cold water is falling. At the same time, current of air is made to circulate over the film of water. As the air circulates over the water film, it evaporates some of the cooling water. As a result of this rapid evaporation, the steam circulating inside the series of pipes gets condensed. Remaining cooling water that left is collected at an increased temperature and reused. It is brought to the original temperature by adding required quantity of cold water.

Advantages of Steam Condenser

- It increases the efficiency of the plant.
- It reduces the back pressure of the steam and as a result of this, more work can be done.
- It reduces the temperature of the exhaust steam and this allows to obtain more work.
- It allows the reuse of condensate for the feed water and hence reduces the cost of power generation.

- The temperature of the condensate is more than the feed water. This reduces the supply of heat per kg of steam.

Comparison of Jet and Surface Condenser in Tabular Form

S.no	Jet Condenser	Surface Condenser
1.	Exhaust steam and cooling water mixed with each other.	Exhaust steam and cooling water are not mixed with each other.
2.	It is less suitable for high capacity plants.	It is more suitable for high capacity plants.
3.	The condensing plant using this type of steam condenser is simple and economical.	The condensing plant using surface condenser is costly and complicated.
4.	Condensate is wasted and cannot be reused.	The condensate is reused.
5.	Less quantity of circulating water is required.	Large quantity of circulating water is required.
6.	It has low maintenance cost.	It has high maintenance cost.
7.	In jet condenser, more power is required for the air pump.	In surface condenser, less power is required for the air pump.
8.	High power is required for water pumping.	Less power is required for water pumping.

Example 20.13. *To check the leakage of air in a condenser, the following procedure is adopted. After ruming the plant to reach the steady conditions the steam supply to the condenser and the air and condensate pump are shut down, thus completely isolating the condenser. The temperature and vacuum readings are noted at shut down and also after a period of 10-minutes. They are 39°C and 685 mm Hg and 28°C and 480 mm Hg respectively. The barometer reads 750 mm Hg. The effective volume of the condenser is 1.5 m³. Determine (i) quantity of air leakage into the condenser during the period of observation ; (ii) the quantity of water vapour condensed during the period.*

Solution. ***At shut down :***

From steam tables, corresponding to $t_s = 39°C$:

$$p_s = 0.07 \text{ bar} = \frac{0.07}{0.001333} = 52.5 \text{ mm Hg}$$

and $$v_g = 20.53 \text{ m}^3/\text{kg}$$

The combined pressure of steam and air in the condenser,

$$p = p_a + p_s = 750 - 685 = 65 \text{ mm Hg}$$

$$\therefore \quad p_a = p - p_s = 65 - 52.5 = 12.5 \text{ mm Hg} = 12.5 \times .001333 = 0.0167 \text{ bar}$$

Now, mass of air in 1.5 m³

$$m_a = \frac{p_a V_a}{RT_a} = \frac{0.0167 \times 10^5 \times 1.5}{287 \times (273 + 39)} = 0.028 \text{ kg}$$

and mass of steam in 1.5 m³

$$m_s = \frac{1.5}{20.53} = 0.073 \text{ kg}$$

After 10 minutes, observed duration,

From steam tables, corresponding to $t_s = 28°C$:

$$p_s = 0.0378 \text{ bar} = \frac{0.0378}{0.001333} = 28.36 \text{ mm Hg}, \; v_g = 36.69 \text{ m}^3/\text{kg}$$

Total pressure in the condenser,

$$p = 750 - 480 = 270 \text{ mm Hg}$$

∴ Air pressure, $$p_a = p - p_s = 270 - 28.36 = 241.64 \text{ mm Hg}$$
$$= 241.64 \times 0.001333 = 0.322 \text{ bar}$$

Mass of air, $$m_a = \frac{p_a V_a}{RT_a} = \frac{0.322 \times 10^5 \times 1.5}{287 \times (273 + 28)} = 0.559 \text{ kg.}$$

Mass of steam, $$m_s = \frac{1.5}{36.69} = 0.0408 \text{ kg}$$

∴ **Air leakage** *in 10 minutes period*

$$= (0.559 - 0.028) = \mathbf{0.531 \text{ kg.}} \quad \textbf{(Ans.)}$$

steam condensed *in 10 minutes period*

$$= (0.073 - 0.0408) = \mathbf{0.0322 \text{ kg.}} \quad \textbf{(Ans.)}$$

Example 20.10. *In a surface condenser a section of the tubes near to the air pump suction is screened off so that the air is cooled to a temperature below that of the condensate, separate extraction pumps being provided to deal with air and condensate respectively. 5448 kg of steam are condensed per hour and the air leakage is 4.54 kg/h. The temperature of the exhaust steam is 31°C, the temperature of the condensate is 27°C, and the temperature at the air pump suction is 21.1°C. Assuming a constant vacuum throughout the condenser, find :*

(*i*) *The mass of steam condensed per hour in the air cooler :*

(*ii*) *The volume of air in* m^3/h *to be dealt with by the air pump ;*

(*iii*) *The percentage reduction in necessary air pump capacity following the cooling of the air.*

Solution. From steam tables, pressure of steam at 31°C,

$$p_s = 0.045 \text{ bar abs.}$$

and partial pressure of steam at air pump suction at 21.1°C,

$$p_s = 0.025 \text{ bar abs.}$$

Partial pressure of air, $p_a = 0.045 - 0.025 = 0.02$ bar abs.

Also, at air pump suction ; $T = 21.1 + 273 = 294.1$ K

(*i*) Using characteristic equation of gas for 4.54 kg of air,

$$p_a V = mRT$$

$$V = \frac{mRT}{10^4 p_a} = \frac{4.54 \times 287 \times 294.1}{0.02 \times 10^5} = 191.6 \text{ m}^3\text{/h.}$$

As per Dalton's law this is also the volume of the steam mixed with air at the air pump suction.

Specific volume of steam at partial pressure of 0.025 bar = 54.25 m³/kg

$\therefore$ *Mass of steam at air pump suction/hour* $= \dfrac{191.6}{54.25} =$ **3.53 kg. (Ans.)**

(*ii*) **Volume of air/hour to be dealt = 191.6 m³/h. (Ans.)**

(*iii*) Temperature at the pump *without air cooling*,

$$T = 27 + 273 = 300 \text{ K.}$$

Partial pressure of steam at 27°C = 0.0357 bar (From steam tables).

Then, partial pressure of air = 0.045 − 0.0357 bar = 0.0093 bar

Again, using the characteristic gas equation to 4.54 kg of air at partial pressure of 0.0093

$$V = \frac{mRT}{p_a} = \frac{4.54 \times 287 \times 300}{0.0093 \times 10^5} = 420.3 \text{ m}^3\text{/h.}$$

Percentage reduction in air pump capacity due to air cooling

$$= \frac{420.3 - 191.6}{420.3} = \mathbf{0.544 \text{ or } 5.44\%.} \quad \textbf{(Ans.)}$$

INDUSTRIAL APPLICATIONS

Industrial Applications.

Nozzles are used in

1. Steam turbines, gas turbines, water turbines and in jet engines, Jet propulsion.
2. Nozzles are used for flow measurement e.g. in venturimeter.
3. Nozzles are used to remove air from a condenser.
4. Injectors for pumping feed water to boilers.

Condensers have proven to be reliable and are chosen for applications throughout the world, a few of which are below:

1. Power
2. Chemical processing
3. Refinery
4. HVAC
5. Low oxygen condensate
6. Marine

TUTORIAL QUESTIONS

Theory Questions:

1. Mention various types of nozzles and distinguish their features.
2. Define nozzle velocity coefficient and how it is related to nozzle efficiency and discharge coefficient as applied to nozzles.
3. Derive an expression for maximum mass flow per unit area of flow through a convergent- divergent nozzle when steam expands isentropic ally from rest.
4. Discuss the process of super-saturation in steam nozzles with the help of enthalpy entropy diagram.
5. Define degree of super-saturation and degree of under-cooling. Explain in detail the physical significance of abrupt change at Wilson`s line.
6. Differentiate the terms "over expanding" and "under expanding" as applied to a fluid flow through a nozzle.
7. What are the components of a steam condensing plant? What are the functions of each component working in steam condensing plant?
8. Classify steam condensers. What are the differences between the jet Condensers and surface condensers? List out the advantages of condenser in a steam power plant.
9. Draw the schematic diagram of parallel flow jet condenser.
10. Draw the schematic diagram of Evaporative condenser and Explain Briefly?

Numerical Problems:

1. In a convergent-divergent nozzle, the steam enters at 15 bar and 3000C and leaves at a pressure of 2 bar. The inlet velocity to the nozzle is 150 m/s. Find the required throat and exit areas for a mass flow rate of 1 kg/s. Assume nozzle efficiency to be 90 percent and Cps = 2.4 kJ/kg.K
2. A convergent nozzle is used to expand ethane gas at 780 kPa and 3500K isentropically into a chamber at 370 kPa. Find the nozzle exit area for a mass flow rate of 1400 kg/s. Assume the initial velocity is zero, C p=1.9 kJ/kgK, R=277 J/kg K and A. INDEX =1.17
3. Dry saturated steam expands through a nozzle from a pressure of 13.7 bar down to 9.6 bar. Assuming the flow to be frictionless and adiabatic, estimate velocity of steam jet.

ASSIGNMENT QUESTIONS

4. For a nozzle, show the area on p-v diagram which represents the conversion of heat energy to kinetic energy. Prove that this area equals the heat drop during expansion. Assume isentropic flow in a nozzle. Further show the expansion for steam on T-s and h-s charts and for air on T-s chart.

ASSIGNMENT QUESTIONS

1. What are the differences between the jet Condensers and surface condensers?
   Draw the schematic diagram of Evaporative condenser and Explain Briefly?
2. Starting from the fundamentals, show that the maximum discharge through the nozzle,the ratio of throat pressure to inlet pressure is given by $(2/n+1)^{n/n-1}$, where n is the index for isentropic expansion through the nozzle
3. In a convergent-divergent nozzle, the steam enters at 15 bar and 300^{o}C and leaves at a pressure of 2 bar. The inlet velocity to the nozzle is 150 m/s. Find the required throat and exit areas for a mass flow rate of 1 kg/s. Assume nozzle efficiency to be 90 percent and C_{ps} = 2.4 kJ/kg.K

UNIT 3

STEAM TURBINES & REACTION TURBINE

Course Objective:

Student have knowledge of methods of analysis and design of complicated thermodynamic systems

Course Outcome:

To understand the velocity triangles and boilers concepts in a lucid manner.

Steam Turbine

14.1 INTRODUCTION

In steam engines, the pressure energy of steam is utilised. Steam engines can be operated without any drop in pressure in the cylinder, but the operation of steam turbine purely depends on the dynamic action of steam.

The first steam turbine was made by Hero of Alexandria in about 120 BC. This turbine worked on pure reaction principle and had no provision for driving anything. It consisted of a hollow ball mounted on between the two pivots. Steam which was produced in a cauldron placed beneath the turbine was supplied through one pivot. Two converging nozzles were provided in the ball. The steam was expanded through these nozzles to the atmosphere which caused reactive force on the ball and thus initiated rotation.

It has undergone several changes in its design since. The steam turbine has been used as a prime mover in all thermal power stations. Now, single steam turbine of 1000 MW capacity is built in many countries.

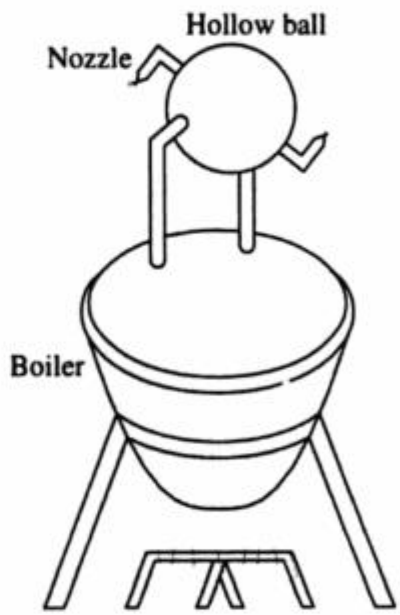


Fig. 14.1 *Working of the hero's turbine*

CLASSIFICATION OF TURBINES

Steam turbines are mainly divided into two groups:

(a) Impulse turbine and (b) impulse reaction turbine (in practice known as reaction turbine).

Impulse Turbine

In impulse turbine, steam coming out at a very high velocity through the fixed nozzle strikes the blades fixed on the periphery of a rotor. The blades change the direction of the steam flow without changing its pressure. The force due to change of momentum causes the rotation of the turbine shaft.

The examples of impulse turbine are De-Laval, Curties and Rateau.

Impulse-Reaction Turbine

In an impulse reaction turbine, steam expands both in fixed and moving blades continuously as the steam passes over them. The pressure drop occurs continuously over both moving and fixed blades.

The example of such turbine is Parsons's turbine.

DIFFERENCE BETWEEN IMPULSE AND REACTION TURBINE

(a) In an impulse turbine the steam completely expands in the nozzle and its pressure remains constant during its flow through the blade passages.

In reaction turbines steam expands partially in the nozzle and further expansion takes place in the rotor blades.

(b) The relative velocity of steam passing over the blade of impulse turbine remains constant (assuming there is no friction).

The relative velocity of steam passing over the blade of reaction turbine increases as the steam passing over the blade expands.

(c) The impulse turbine blades are symmetrical but the reaction turbine blades are asymmetrical.

(d) The pressure on both ends of the moving blade of an impulse turbine is the same but in a reaction turbine it is different.

(e) The number of stages required for reaction turbine are more compared with impulse turbine for the same power developed as the pressure drop in each stage is small.

(f) The blade efficiency curve for the reaction turbine is more flat compared to that of the impulse turbine.

(g) The steam velocity in a reaction turbine is not very high and therefore the speed of the turbine is low.

SIMPLE IMPULSE TURBINE

The simple impulse turbine is called De-Laval turbine after the name of its inventor. It consists of one set of nozzles and a set of moving blades as shown in Fig. 14.2. The expansion of steam from its initial pressure to final pressure takes place in one set of nozzles. Due to high drop in pressure in the nozzles the velocity of the steam increases in the nozzle.

The steam leaves the nozzle with a very high velocity and strikes the blades mounted on a wheel with this high velocity. The pressure of the steam when it moves over the blades remains constant but the velocity decreases.

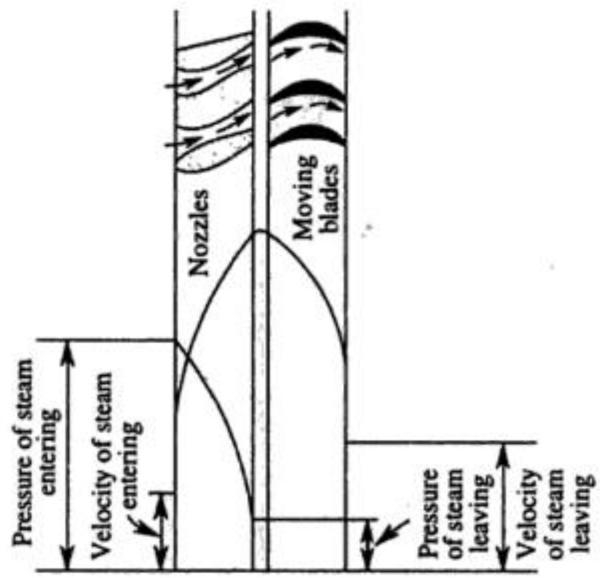


Fig. 14.2 *Diagrammatic sketch of a simple impulse turbine*

But this turbine is not commonly used due to the following disadvantages.

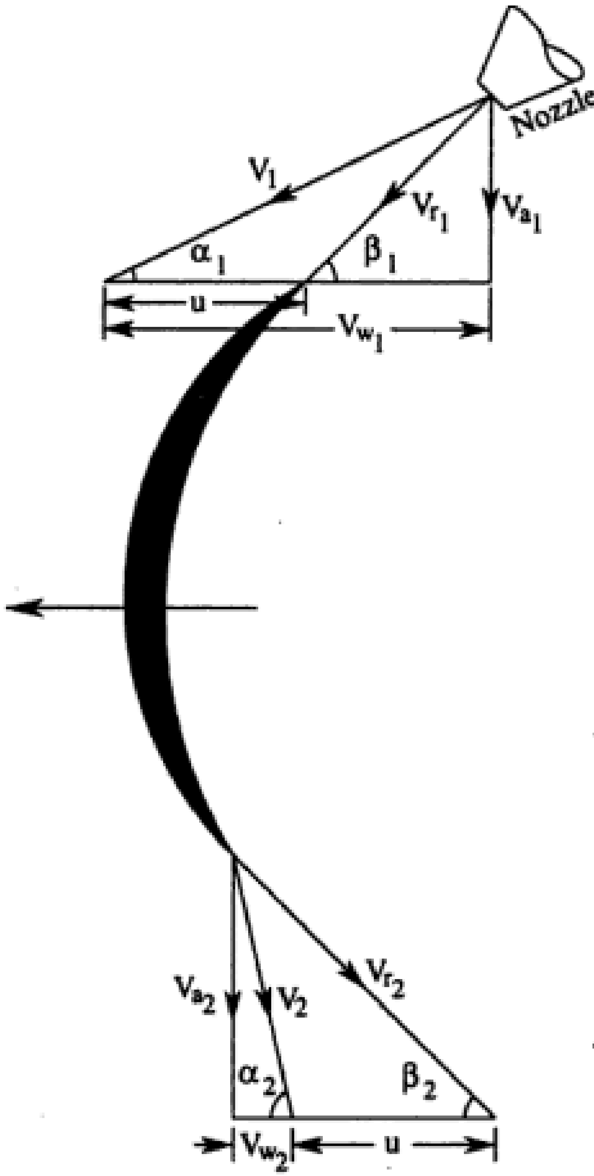
(a) Since all the kinetic energy of the high velocity steam has to be absorbed in only one ring of moving blades, so the velocity of the turbine is too high i.e. up to 30,000 r.p.m. for practical purposes.

(b) The veloctiy of steam at exit is sufficiently high which means that there is a considerable loss of kinetic energy.

VELOCITY DIAGRAM FOR MOVING BLADES FOR AN IMPULSE TURBINE

Figure 14.3 shows the velocity diagram of a single stage turbine.

V_1 = Absolute velocity of steam



at inlet in m/s

α_1 = Nozzle inlet angle

u = Blade velocity in m/s

V_{r_1} = Relative velocity of steam at inlet in m/s

V_{w_1} = Tangential velocity of steam at inlet m/s.

V_{a_1} = Axial velocity of steam at inlet in m/s

β_1 = Blade inlet angle

β_2 = Blade outlet angle

V_{r_2} = Relative velocity of steam at outlet in m/s.

V_{w_2} = Tangential velocity of steam at outlet in m/s.

V_{a_2} = Axial velocity of steam at outlet in m/s.

K = Blade velocity coefficient $= \frac{V_{r_2}}{V_{r_1}}$

V_2 = Absolute velocity of steam at outlet in m/s.

α_2 = Angle made by absolute velocity V_2 with the tangent of the wheel at outlet.

Fig. 14.3 *Velocity diagram for moving blades for an impulse turbine*

COMBINED VELOCITY DIAGRAM

The procedure for drawing the combined velocity diagram is given below:

1. Draw a horizontal line AB equal to blade velocity u to some suitable scale.
2. Draw a line AC at an angle α_1 with AB. Cut $AC = V_1$.

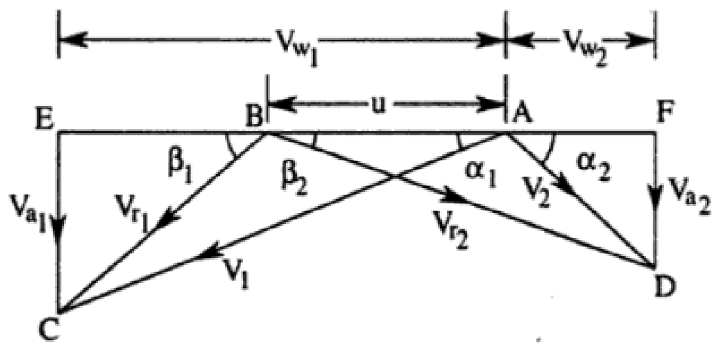


Fig. 14.4 *Combined velocity diagram*

3. Join B and C. The line BC represents the relative velocity at inlet. The blade inlet angle β_1 is measured and its value is noted down.
4. From point C draw a perpendicular CE on AB produced. CE represents axial velocity at inlet and AE represents tangential velocity at inlet.
5. From point B draw a line BD at an angle β_2 (the blade outlet angle). Cut $BD = V_{r_2} = KV_{r_1}$. Join A and D. AD represents the absolute velocity at outlet. The angle α_2 is measured and noted down.
6. From point D draw a perpendicular DF on BA.Then AF represents the tangential velocity of steam at outlet and DF represents the axial velocity outlet. This completes the velocity triangle.
   (a) Force in the tangential direction = Rate of change of momentum in the tangential direction.

   $$= \text{mass per second} \times \text{change of velocity} = m\left(V_{w_1} \pm V_{w_2}\right) \text{ N} \qquad (14.1)$$

   (b) Force in the axial direction or Axial thrust

   = Rate of change of momentum in axial direction

   $$= m\left(V_{a_1} - V_{a_2}\right) \text{ N} \qquad (14.2)$$

   (c) Work done by steam on blades $= m\left(V_{w_1} \pm V_{w_2}\right)u$ Nm/s (14.3)

   (d) Power developed by the turbine $= \dfrac{m(V_{w_1} \pm V_{w_2})u}{1000}$ kW (14.4)

(e) Blade efficiency $= \dfrac{\text{Work done on the blade/s}}{\text{Energy supplied to the blade/s}}$

$$\eta_b = \frac{m(V_{w_1} \pm V_{w_2})u}{\frac{1}{2} \times m \times V_1^{\,2}} = \frac{2u(V_{w_1} \pm V_{w_2})}{V_1^2} \qquad (14.5)$$

(f) Energy lost due to blade friction $= \dfrac{1}{2} m\left(V_{r_1}^2 - V_{r_2}^2\right)$ Nm/s (14.6)

(g) Stage efficiency $= \dfrac{\text{Work done on the blade/s}}{\text{Total energy supplied per stage}}$

$$= \frac{m(V_{w_1} \pm V_{w_2})u}{m(H_1 - H_2)} = \frac{(V_{w_1} \pm V_{w_2})u}{H_d} \qquad (14.7)$$

where $H_d = H_1 - H_2 =$ heat drop in the nozzle ring.

MAXIMUM WORK AND MAXIMUM DIAGRAM EFFICIENCY

From the combined velocity diagram, Fig. 14.4

$V_{w_1} = V_1 \cos\alpha_1 = V_{r_1} \cos\beta_1 + u$, $V_{w_2} = V_2 \cos\alpha_2 = V_{r_2} \cos\beta_2 - u$

$$\therefore V_{w_1} + V_{w_2} = Vr_1 \cos\beta_1 + V_{r_2} \cos\beta_2 = V_{r_1} \cos\beta_1 \left[1 + \frac{V_{r_2}}{V_{r_1}} . \frac{\cos\beta_2}{\cos\beta_1}\right] = Vr_1 \cos\beta_1 [1 + KC]$$

where $K = \frac{V_{r_2}}{V_{r_1}}$ and $c = \frac{\cos\beta_2}{\cos\beta_1}$

$$V_{w_1} + V_{w_2} = (V_1\cos\alpha_1 - u)(1 + KC)$$

Rate of doing work per kg of steam per second $= (V_1\cos\alpha_1 - u)(1 + KC)u$

Diagram efficiency $\eta_b = \frac{(V_1\cos\alpha_1 - u)(1 + KC)2u}{V_1^2}$

Let $\rho = \frac{u}{V_1}$ = Blade speed ratio

$$\eta_b = 2(\rho\cos\alpha_1 - \rho^2)(1 + KC) \qquad (14.8)$$

It is obvious from Eq. (14.8) that the value of diagram efficiency depends upon the following factors:

(i) Nozzle angle α_1 (ii) Blade speed ratio ρ (iii) Blade angles β_1 and β_2 (iv) Blade velocity co-efficient K.

If α_1, K and C are assumed to be constant, the diagram efficiency depends upon the value of ρ. In order to determine the optimum value of ρ for maximum diagram efficiency the first differential of the equation.

$\eta_b = 2(\rho\cos\alpha_1 - \rho^2)(1 + KC)$ be equated to zero

$$\therefore\ 2(1 + KC)(\cos\alpha_1 - 2\rho) = 0$$

or $\rho = \dfrac{\cos\alpha_1}{2}$ since$(1+KC)$ is not equal to zero

$$\therefore\ \eta_{b(\max)} = 2(1+KC)\left[\frac{\cos\alpha_1}{2}.\cos\alpha_1 - \frac{\cos^2\alpha_1}{4}\right] = (1+KC)\frac{\cos^2\alpha_1}{2} \qquad (14.9)$$

Assuming that the blades are symmetrical and friction is absent. $\beta_1 = \beta_2, \therefore C = 1$, and $K = 1$

$$\eta_{b(\max)} = \cos^2\alpha_1 \qquad (14.10)$$

Rate of doing work per kg of steam per second

$$= (V_1\cos\alpha_1 - u)(1+KC)u$$

$$\rho = \frac{u}{V_1} = \frac{\cos\alpha_1}{2} \quad \therefore V_1 = \frac{2u}{\cos\alpha_1}$$

Maximum rate of doing work per kg of steam per second

$$= \left(\frac{2u}{\cos\alpha_1}.\cos\alpha_1 - u\right)(1+1.1)u = 2u^2 \qquad (14.11)$$

METHODS OF REDUCING ROTOR SPEED

In case of simple impulse turbine, the steam is expanded from the boiler pressure to the condenser pressure in one stage only. Hence the speed of the rotor becomes very high for practical purposes. In order to make the rotor speed practicable compounding of steam turbine is done. Compounding is the method in which multiple system of rotors are keyed to a common shaft in series and the steam pressure or jet velocity is absorbed in stages as it flows over the rotor blades. The rotor speed can be reduced by the following methods of compounding steam turbine.

(a) Velocity compounding
(b) Pressure compounding
(c) Pressure-velocity compounding.

Velocity Compounding

Figure 14.5 (a) shows a section of velocity compounded turbine. It consists of a set of nozzles and a few rows of moving blades which are fixed to the shaft and rows of fixed blades which are attached to the casing. In the figure two rows of moving blades are separated by a row of fixed blades. The steam is expanded from the boiler pressure to the condenser pressure in the nozzles only. Due to the decrease in pressure the steam acquires a very high velocity. This high velocity steam first enters the first row of moving blades, where some portion of the velocity is absorbed. Then it enters the ring of fixed blades where the direction of steam is changed to suit the second ring of moving blades. There is no

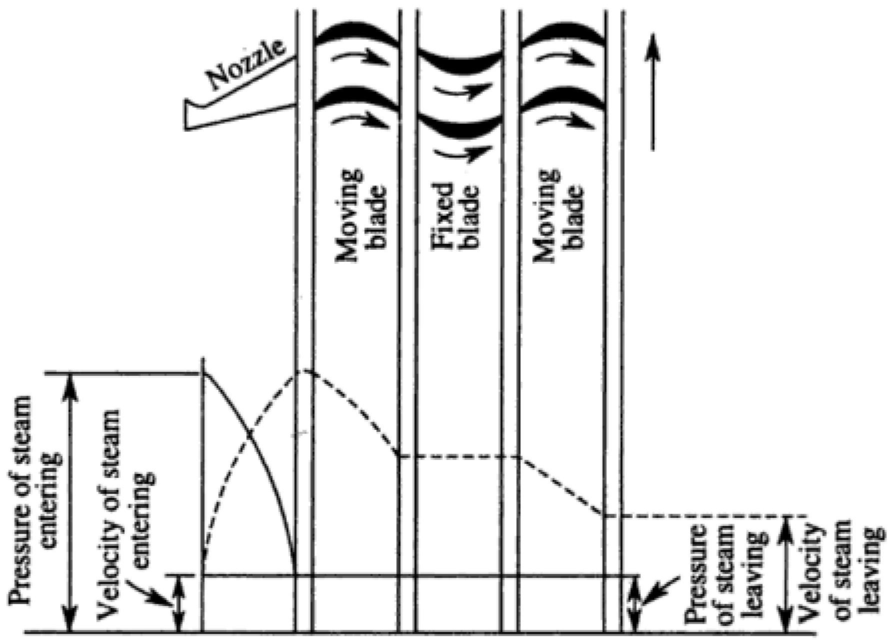


Fig. 14.5(a) *Velocity compounding*

change in the velocity as the steam passes over the fixed blades.

The steam then passes on to the second row of moving blades where velocity is further reduced. Thus a fall in velocity occurs everytime when steam passes over the row of moving blades. Steam thus leaves the turbine with a low velocity. The variation of pressure and velocity of steam as it passes over the moving and fixed blades is shown in the figure. It is clear that the pressure drop takes place only in the nozzle and there is no further drop of pressure as the steam passes over the blades. This method of velocity compounding is used in Curtis turbine.

Advantages of velocity compounded turbines

(a) The arrangement has less number of stages and hence less initial cost.
(b) The arrangement requires little space.
(c) The system is reliable and easy to operate.
(d) The fall of pressure in the nozzle is considerable, so the turbine itself need not work in high pressure surroundings and the turbine housing need not be very strong.

Disadvantages of velocity compounded turbines

(a) Due to very high steam velocity in the nozzles, friction losses are large.
(b) The efficiency is also low because the ratio of blade velocity to steam velocity is not optimum for all the wheels.
(c) Power developed in the later rows is only a fraction of power developed in the first row. Still all the stages require same space, material and cost of fabrication.

Velocity Diagram for Velocity Compounded Turbine

Figures 14.5(d) and (e) shows the velocity diagram for a two stage velocity compounded turbine.

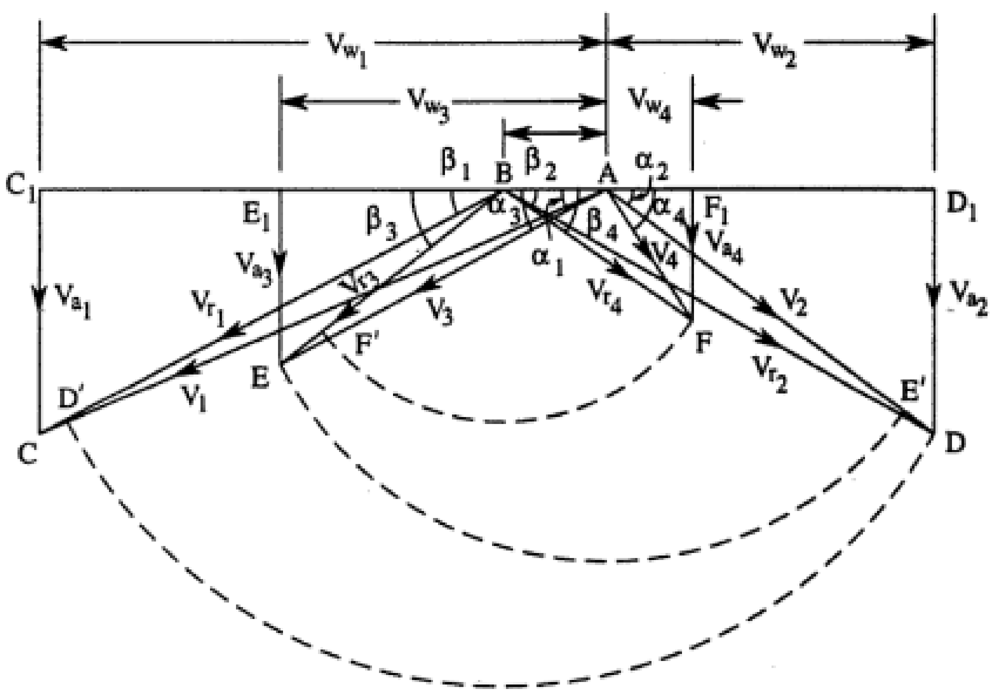


Fig. 14.5 (e) *Velocity diagram for velocity compounded two stage turbine*

Given,

u	=	Blade velocity in m/s
V_1	=	Absolute velocity of steam at inlet to blade in m/s.
V_{r_1}	=	Velocity of steam relative to the 1st row of moving blades at entrance in m/s.
α_1	=	Nozzle angle at entrance.
β_1	=	Entrance angle to the 1st row of moving blades
K	=	Friction factor
V_{r_2}	=	Velocity of steam relative to the 1st row of moving blade at exit in m/s.
	=	KVr_1
β_2	=	Exit angle from 1st row of moving blades.
V_2	=	Absolute velocity of steam at exit from 1st row of moving blade in m/s.
α_2	=	Entrance angle to the fixed blade.
V_3	=	Absolute velocity of steam at exit from the fixed blades in m/s = $K\,V_2$
	=	Absolute velocity of steam entering the second row of moving blades.
α_3	=	Exit angle from the fixed blades.
V_{r_3}	=	Velocity of steam relative to the second row of moving blades at entrance in m/s.
β_3	=	Entrance angle to the 2nd row of moving blades.
Vr_4	=	Velocity of steam relative to the second row of moving blades at exit in m/s.
β_4	=	Exit angle from 2nd row of moving blades.
V_4	=	Absolute velocity of steam at exit from 2nd row of moving blades in m/s.
α_4	=	Entrance angle to the next row of fixed blades.
$\left(V_{w_1} \pm V_{w_2}\right)$	=	C_1D_1 = Tangential velocity or velocity of whirl for the 1st row of moving blades in m/s.
$\left(V_{w_3} \pm V_{w_4}\right)$	=	E_1F_1 = Tangential velocity or velocity of whirl for the 2nd row of moving blades in m/s.
V_{a_1}	=	Axial velocity of steam at entrance to the 1st row of moving blades in m/s.

m/s.

V_{a_3} = Axial velocity of steam at entrance to the 2nd row of moving blade in m/s.

V_{a_4} = Axial velocity of steam at exit to the 2nd row of moving blades in m/s.

Let AB represent the blade velocity u to any scale. At A draw $AC = V_1$ at an angle α_1 to AB. Join BC. Then, $BC = Vr_1$ and $\angle CBC_1 = \beta_1$. Mark BD' such that $BD' = KBC = Vr_2$. Draw BD such that $\angle DBA = \beta_2$. Join AD; then $AD = V_2$ and $\angle DAD_1 = \alpha_2$. From C draw perpendicular CC_1 on AB produced. Then $AC_1 = Vw_1$ and $CC_1 = Va_1$. From D draw perpendicular DD_1 on BA produced. Then $AD_1 = Vw_2$ and $DD_1 = Va_2$. At B draw $AE = V_3 = KV_2$ at an angle α_3. Join BE. Then $BE = Vr_3$.

From E draw perpendicular EE_1 on AB produced. Then $EE_1 = Va_3$ and $\angle EBE_1 = \beta_3$. Mark BF' such that $BF' = KBE = Vr_4$. Draw BF such that $\angle FBA = \beta_4$. Join AF, then $AF = V_4$. From F draw perpendicular on BA produced. Then $FF_1 = Va_4$, $AE_1 = Vw_3$ and $AF_1 = Vw_4$.

Work done from the 1st row of moving blades $= m(Vw_1 \pm Vw_2)u$ Nm/s

$$= m.C_1D_1.u \text{ Nm/s} \tag{14.12}$$

Work done from the 2nd row of moving blades $= m(Vw_3 \pm Vw_4).u$ Nm/s

$$= m.E_1F_1.u \text{ Nm/s} \tag{14.13}$$

$\therefore$ (a) Work done from the two stages $= m.C_1D_1.u + m.E_1F_1.u$

$$= m(C_1D_1 + E_1F_1).u \text{ Nm/s or J/s} \tag{14.14}$$

(b) Power developed by a two stage impulse turbine $= m.u.(C_1D_1 + E_1F_1)$ W

$$= m.u.\frac{(C_1D_1 + E_1F_1)}{1000} \text{ kW} \tag{14.15}$$

(c) Diagram or blade efficiency

$$\eta_b = \frac{u(C_1D_1 + E_1F_1)}{\frac{V_1^2}{2}} = \frac{2u(C_1D_1 + E_1F_1)}{V_1^2} \times 100\% \tag{14.16}$$

$$\text{(d) Stage efficiency} = \frac{u(C_1D_1 + E_1F_1)}{1000\,H} \tag{14.17}$$

where $H = H_1 - H_2$ = enthalpy drop in nozzle in kJ/kg.

(e) Total axial thrust $= m[(Va_1 - Va_2) + (Va_3 - Va_4)]$N (14.18)

In order to have maximum efficiency of the turbine, the out going absolute velocity, V_4 of the steam should be at right angles to the blade having no tangential component i.e. $\alpha_4 = 90°$.

The velocity diagrams at the inlet and outlets for the different blades are drawn on the same side as shown in Fig. 14.5 (f) for frictionless flow and symmetrical blading under maximum efficiency condition.

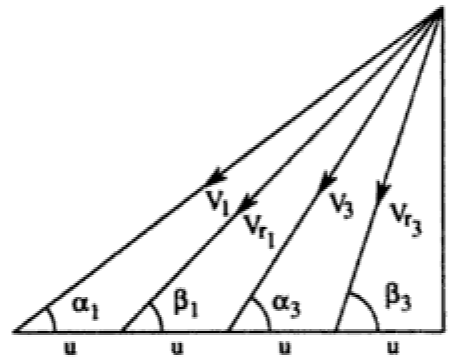


Fig. 14.5(f)

$V_1\cos\alpha_1 = 4u$ or $u = \dfrac{V_1\cos\alpha_1}{4} = \dfrac{V_1\cos\alpha_1}{2 \times \text{No. of stages}}$

From Fig. 14.5 (f) we find that, $V_2\cos\alpha_2 = 2u$

$Vr_4\cos\beta_4 = Vr_3\cos\beta_3 = u$

$\therefore$ Maximum work in the 1st stage $= m(V_1\cos\alpha_1 + V_2\cos\alpha_2)u = m(4u + 2u)u = m.6u^2$

Maximum work in the 2nd stage $= m(Vr_3\cos\beta_3 + Vr_4\cos\beta_4)u = m(u + u)u = m.2u^2$

$\therefore$ Total work $= 6mu^2 + 2mu^2 = 8mu^2$

Maximum blade efficiency, $\eta_{b(\max)} = \dfrac{8mu^2}{\frac{1}{2}mV_1^2}$

$$\eta_{b(\max)} = \frac{8mu^2 \times 2}{m \times \frac{16u^2}{\cos^2\alpha_1}} = \cos^2\alpha_1 \qquad (14.19)$$

Thus the velocity compounded turbine has the same maximum efficiency as the single stage turbine.

1. Steam with absolute velocity 360 m/s enters the stage of an impulse turbine provided with a single row wheel. The nozzles are inclined at 20° to the plane of the wheel. The blade rotor with diameter 95.5 cm rotates with a speed of 3000 r.p.m. Find (a) suitable inlet and outlet angle for the moving blade so that there is no axial thrust on the blade. It may be assumed that friction in blade passages is 19% of the kinetic energy corresponding to relative velocity at inlet to blades, (b) Power developed in blading for a steam flow of 1 kg/s, and (c) Kinetic energy of steam finally leaving the stage.

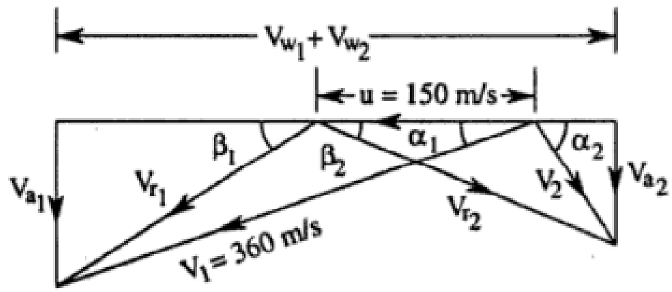


Fig. 14.10

Solution: Given, $V_1 = 360$ m/s, $\alpha_1 = 20°$, $d = 95.5$ cm, $N = 3000$ r.p.m, $V_{a_1} = V_{a_2}$

Mean blade speed, $u = \dfrac{\pi d N}{60} = \dfrac{\pi \times 0.955 \times 3000}{60} = 150$ m/s

From the given relation

$$\frac{V_{r_2}^2}{2} = (1 - 0.19)\frac{V_{r_1}^2}{2} \quad \therefore V_{r_2} = 0.9 V_{r_1}$$

Scale of velocity diagram, 1 cm = 50 m/s

From the velocity triangle

(a) Blade inlet angle, $\beta_1 = 34°$

Blade outlet angle, $\beta_2 = 38°$

$V_{w_1} + V_{w_2} = 348$ m/s

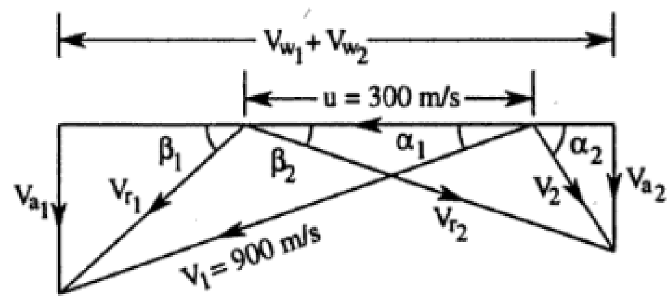
(b) Power output for a steam flow of 1 kg/s

$$= \frac{(V_{w_1} + V_{w_2})u}{1000} \text{ kW.} = \frac{348 \times 150}{1000} = 52.2 \text{ kW}$$

(c) Absolute velocity of steam at exit, $V_2 = 126$ m/s

Kinetic energy of steam finally leaving the stage $= \dfrac{V_2^2}{2} = \dfrac{(126)^2}{2} = 7938$ Nm/kg.

2. The blade speed of a single ring of an impulse turbine is 300 m/s and the nozzle angle is 20°. The isentropic heat drop is 473 kJ/kg and the nozzle efficiency is 0.85. Given that the blade velocity co-efficient is 0.7 and the blades are symmetrical, draw the vector diagrams and calculate for a mass flow of 1 kg/s:



(a) axial thrust on the blading.
(b) steam consumption per B.P. hour if the mechanical efficiency is 90 per cent.
(c) blade efficiency, stage efficiency and maximum blade efficiency.
(d) heat equivalent of the friction of blading.

Solution: $V_1 = 44.7\sqrt{(H_1 - H_2)\eta_n} = 44.7\sqrt{473 \times 0.85} = 900$ m/s

Given, $\alpha_1 = 20°$, $u = 300$ m/s, $\beta_1 = \beta_2$, $V_{r_2} = 0.7V_{r_1}$

From this data the velocity diagram is drawn and the following results are obtained.

$\beta_1 = \beta_2 = 29.5°$; $V_{w_1} + V_{w_2} = 927.5$ m/s; $V_{a_1} - V_{a_2} = 92.5$ m/s; $V_{r_1} = 630$ m/s; $V_{r_2} = 441$ m/s

(a) Axial thrust per kg $= V_{a_1} - V_{a_2} = 92.5$ N

(b) Power $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000}$ kW $= \dfrac{1 \times 927.5 \times 300}{1000} = 278.25$ kW

Brake power $= 278.25 \times 0.9 = 250.425$ kW

$\therefore$ Steam consumption per B.P. hour $= \dfrac{3600}{250.425} = 14.33$ kg

(c) Blade efficiency $= \dfrac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 300 \times 927.5}{(900)^2} = 0.688 = 68.8\%$

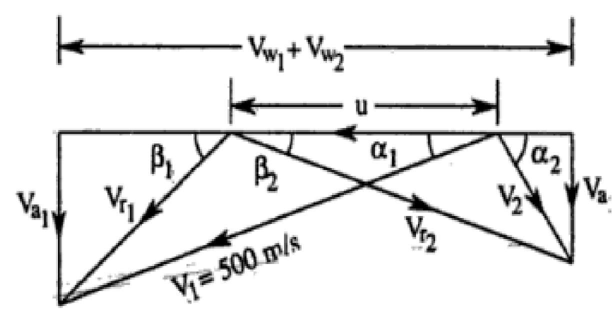
Maximum blade efficiency, $\eta_{b(\max)} = \cos^2\alpha_1 = \cos^2 20 = 0.88 = 88\%$

Stage efficiency $= \dfrac{\text{Work done on blade}}{\text{Total energy supplied to blade}} = \dfrac{278.25}{473} = 0.588 = 58.8\%$

(d) Energy lost due to blade friction

$$= \frac{Vr_1^2 - Vr_2^2}{2} = \frac{(630)^2 - (441)^2}{2} = 101209.5 \text{ J} = 101.2095 \text{ kJ.}$$

3. The data pertaining to impulse turbine is as follows:
Steam velocity = 500 m/s; Blade speed = 200 m/s; Exit angle of moving blade = 25°; Nozzle angle = 20°. Neglecting the effect of friction when passing through the blade passages, calculate (a) inlet angle of moving blade, (b) exit velocity and direction, (c) work done per kg of steam, (d) axial thrust and power for a steam flow rate of 5 kg/s, and (e) diagram efficiency.



Solution: Given, $V_1 = 500$ m/s; $u = 200$ m/s; $\beta_2 = 25°$; $\alpha_1 = 20°$ $V_{r_1} = V_{r_2}$

The following results are obtained from the velocity diagram

(a) Inlet angle of moving blade, $\beta_1 = 33°$

(b) Exit velocity, $V_2 = 162.5$ m/s

Direction of exit velocity, $\alpha_2 = 56°$

(c) $V_{w_1} + V_{w_2} = 555$ m/s

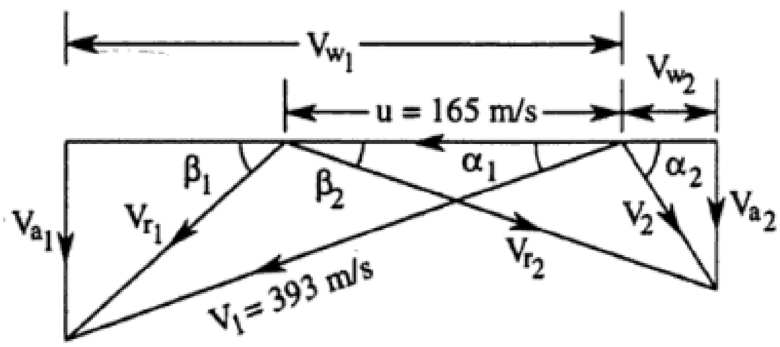
Work done per kg of steam $= m(V_{w_1} + V_{w_2})u = 1 \times 555 \times 200 = 111000$ Nm/s

(d) Axial thrust $= m(V_{a_1} - V_{a_2}) = 5 \times 45 = 225$ N

$$\text{Power} = \frac{5 \times 111000}{1000} = 555 \text{ kW.}$$

(e) $$\text{Diagram efficiency} = \frac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 200 \times 555}{(500)^2} = 0.888 = 88.8\%$$

5. The mean diameter of the blades of an impulse turbine with a single row wheel is 105 cm and the speed is 3000 r.p.m. The nozzle angle is 18°, the ratio of blade speed to steam speed is 0.42 and the ratio of the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle. The steam flow is 8 kg per sec. Draw the velocity diagram for the blades and derive the (a) resultant t! n the blades, (b) tangential thrust on the blades, (c) axial thrust on the blades, (d) power developed in blades, and (e) blading efficiency



Solution: Given, $d = 105\text{ cm} = 1.05\text{ m}; N = 3000; \alpha_1 = 18°$

$$\frac{u}{V_1} = 0.42; \quad \frac{V_{r_2}}{V_r} = 0.84; \quad \beta_2 = \beta_1 - 3°; \quad m = 8\text{ kg/s}$$

Blade speed, $u = \dfrac{\pi d N}{60} = \dfrac{\pi \times 1.05 \times 3000}{60} = 165\text{ m/s}$

Absolute velocity of steam at inlet, $V_1 = \dfrac{165}{0.42} = 393\text{ m/s}$

The velocity diagram is shown in Fig. 14.14.

The following results are obtained from the velocity diagram:

$\beta_1 = 30.2°; V_{w_1} + V_{w_2} = 389\text{ m/s}; V_{a_1} = 122\text{ m/s}; V_{a_2} = 93\text{ m/s}$

(a) Resultant thrust $= \sqrt{(3112)^2 + (232)^2} = 3120.56\text{ N}$

(b) Tangential thrust $= m(V_{w_1} + V_{w_2}) = 8 \times 389 = 3112\text{ N}$

(c) Axial thrust $= m(V_{a_1} - V_{a_2})\text{ N} = 8(122 - 93) = 232\text{ N}$

(d) Power developed $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000}\text{ kW} = \dfrac{8 \times 389 \times 165}{1000} = 513.48\text{ kW}$

(e) Blading efficiency $= \dfrac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 165 \times 389}{(393)^2} = 0.83 = 83\%.$

REACTION TURBINE

A turbine in which steam pressure decreases gradually while expanding through the moving blades as well as through the fixed blades is known as *reaction turbine*. It consists of a large number of stages, each stage consisting of set of fixed and moving blades. The heat drop takes place throughout in both fixed and moving blades. No nozzles are provided in a reaction turbine. The fixed blades act both as nozzles in which the velocity of steam is increased and direct the steam to enter the ring of moving blades. As the pressure drop takes place both in the fixed and moving blades all blades are nozzle shaped. The steam expands while flowing over the moving blades and thus gives reaction to the moving blades. Hence the turbine is known as reaction turbine. The fixed blades are attached with the casing whereas moving blades are fixed with rotor. It is also called Pearson's Reaction turbine.

The work done per kg of steam in the stage

(Per pair) $= u(Vw_1 + Vw_2)$ Nm or joules.

Work done by the steam per second per pair $= m.u(Vw_1 + Vw_2)$ N.m/s or J/s

$$\text{Power developed per pair} = \frac{m.u(Vw_1 + Vw_2)}{1000} \text{ kW} \quad (14.20)$$

where, m = mass of steam flowing over blades in kg/s.

$$\text{Efficiency} = \frac{\text{Work done per pair per kg of steam}}{\text{Enthalpy drop per pair}} = \frac{u(Vw_1 + Vw_2)}{1000H} \quad (14.21)$$

where, H = enthalpy drop per pair in kJ/kg.

Degree of Reaction (R)

The degree of reaction is defined as the ratio of isentropic heat drop in the moving blades to isentropic heat drop in the entire stage of reaction turbine. The degree of Reaction R is given by,

$$R = \frac{\text{Enthalpy drop in the moving blade}}{\text{Enthalpy drop in the stage}} = \frac{dH_2}{dH_1 + dH_2} \quad (14.22)$$

where dH_1 = enthalpy drop in the fixed blade per kg of steam $= \dfrac{V_1^2 - V_2^2}{2}$ kJ/kg.

dH_2 = enthalpy drop in the moving blade per kg of steam $= \dfrac{V_{r_2}^2 - V_{r_1}^2}{2}$ kJ/kg $= H_2 - H_3$.

Also, $dH_1 + dH_2$ = enthalpy drop in the stage per kg of steam

$= H_1 - H_3$ = work done by the steam in the stage.

$$dH_1 + dH_2 = u(Vw_1 + Vw_2) \quad \therefore R = \frac{Vr_2^2 - Vr_1^2}{2u(Vw_1 + Vw_2)} \quad (14.23)$$

In Parsons Reaction turbine, the degree of reaction is 50%, then $\alpha_1 = \beta_2, \alpha_2 = \beta_1$. Which means that the moving blade and fixed blade have the same shape. When degree of reaction $R = 0$, we have the simple impulse turbine. When degree of reaction $R = 1$, we have the pure reaction turbine.

For symmetrical triangles, $V_{r_2} = V_1, V_{r_1} = V_2$.

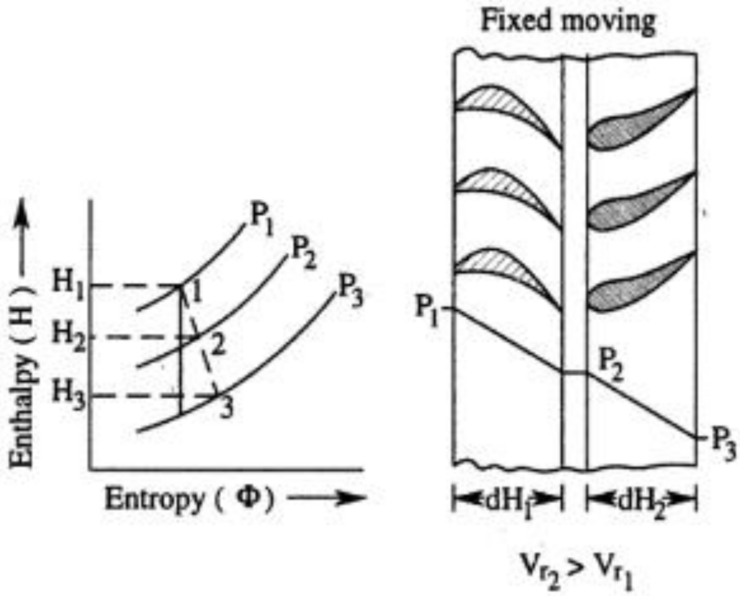


Fig. 14.6 *Reaction turbine stage isentropic expansion*

Efficiency of Reaction Turbine

The condition for maximum efficiency is calculated considering the following assumptions.

(a) The degree of reaction is 50%.

(b) The fixed and moving blades are symmetrical.

The kinetic energy supplied to the fixed blade per kg of steam $= \frac{V_1^2}{2}$

The kinetic energy supplied to the moving blade per kg of steam $= \frac{V_{r_2}^2 - V_{r_1}^2}{2}$

$$\therefore \text{ Total energy supplied} = \frac{V_1^2}{2} + \frac{V_{r_2}^2 - V_{r_1}^2}{2} \qquad (14.24)$$

As $V_{r_2} = V_1$

Total energy supplied $= V_1^2 - \frac{V_{r_1}^2}{2}$

From velocity triangle, $V_{r_1}^2 = V_1^2 + u^2 - 2.u.V_1\cos\alpha_1$

$$\therefore \text{ Total energy supplied} = V_1^2 - \frac{V_1^2 + u^2 - 2.u.V_1\cos\alpha_1}{2} \qquad (14.25)$$

The work done per kg of steam is given by, work done $= u(Vw_1 + Vw_2)$

$= u(V_1\cos\alpha_1 + Vr_2\cos\beta_2 - u) = u(2V_1\cos\alpha_1 - u)$ As $\alpha_1 = \beta_2$ and $V_1 = Vr_2$

$$\text{Diagram efficiency} = \frac{\text{Work done}}{\text{Total energy supplied}}$$

$$= \frac{u(2V_1\cos\alpha_1 - u)}{V_1^2 - \frac{V_1^2 + u^2 - 2u.V_1\cos\alpha_1}{2}} = \frac{2u(2V_1\cos\alpha_1 - u)}{V_1^2 - u^2 + 2.u.V_1\cos\alpha_1}$$

$$= \frac{2u\,V_1\left(2\cos\alpha_1 - \frac{u}{V_1}\right)}{V_1^2\left(1 - \frac{u^2}{V_1^2} + 2.\frac{u}{V_1}\cos\alpha_1\right)} = \frac{2.\frac{u}{V_1}\left(2\cos\alpha_1 - \frac{u}{V_1}\right)}{\left(1 - \frac{u^2}{V_1^2} + 2.\frac{u}{V_1}.\cos\alpha_1\right)}$$

$$= \frac{2\rho(2\cos\alpha_1 - \rho)}{(1 - \rho_2 + 2\rho\cos\alpha_1)} \tag{14.26}$$

where $\rho = \frac{u}{V_1}$, the blade speed ratio.

The efficiency is maximum when $1 - \rho^2 + 2\rho\cos\alpha_1$ is minimum or when its differential with respect to ρ is zero.

$\therefore$ For maximum efficiency $\frac{d}{d\rho}(1 - \rho^2 + 2\rho\cos\alpha_1) = 0$

$$-2\rho + 2\cos\alpha_1 = 0 \text{ or, } \rho = \cos\alpha_1 \tag{14.27}$$

Putting the value of ρ in Eq. 14.26.

In a Parson's turbine of 50% degree of reaction running at 25 r.p.s. the available enthalpy drop for an expansion is 62.8 kJ/kg. If the mean diameter of the rotor is 1 m, find the number of rows of moving blades required. The blade outlet angle is 20° and speed ratio is 0.7. Assume stage efficiency as 80%.

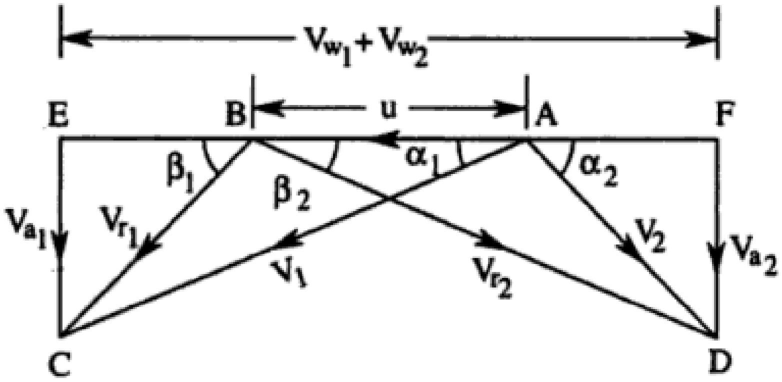


Fig. 14.31

Solution: Given, $\beta_2 = \alpha_1 = 20°$,

Blade speed, $u = \dfrac{\pi D_m N}{60} = \dfrac{\pi \times 1 \times 25 \times 60}{60} = 78.5$ m/s

Speed ratio $= \dfrac{u}{V_1} = 0.7 \quad \therefore \quad V_1 = \dfrac{78.5}{0.7} = 112$ m/s

$$AF = V_{r_2}\cos\beta_2 - u = V_1\cos\beta_2 - u = 112\cos 20° - 78.5 = 27$$

$$V_{w_1} + V_{w_2} = V_1\cos\alpha_1 + AF = 112 \times \cos 20° + 27 = 132.5 \text{ m/s}$$

Work done per stage $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000} = \dfrac{1 \times 132.5 \times 78.5}{1000} = 10.4675$ kJ/kg.

$$\text{Stage efficiency} = \frac{\text{Work done per stage}}{\text{Enthalpy drop per stage}}$$

$\therefore$ Enthalpy drop per stage $= \dfrac{10.4675}{0.8} = 13.084$ kJ/kg.

$\therefore$ Number of rows required $= \dfrac{\text{Total enthalpy drop}}{\text{Enthalpy drop per stage}} = \dfrac{62.8}{13.084} = 5$ stages.

At a stage of a reaction turbine, the mean rotor diameter is 140 cm. The speed ratio is 0.7. Find the inlet angle of the blade if the outlet angle of the blade is 20°. The speed of the turbine is 3000 r.p.m. Find the diagram efficiency.

If the rotor is designed to run at the best theoretical speed and the exit angle remains the same, find the percentage increase in diagram efficiency and rotor speed.

Solution: $u = \frac{\pi DN}{60} = \frac{\pi \times 1.4 \times 3000}{60} = 220$ m/s

Speed ratio, $\rho = 0.7 = \frac{u}{V_1} \quad \therefore \quad V_1 = \frac{220}{0.7} = 314$ m/s

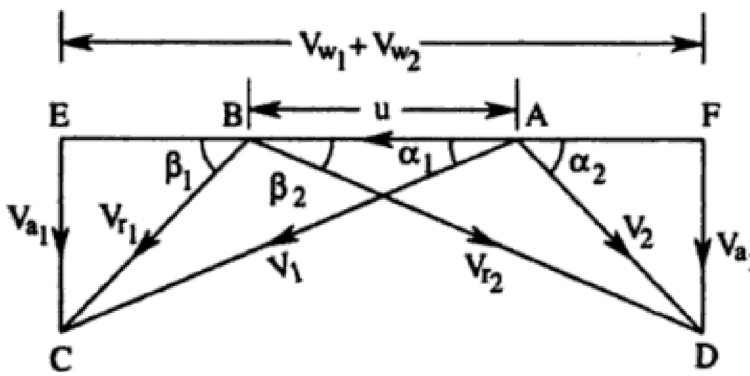


Fig. 14.32

$\alpha_1 = \beta_2 = 20°$

From ΔABC

$$V_{r_1}^2 = V_1^2 + u^2 - 2V_1 u\cos\alpha_1 = (314)^2 + (220)^2 - 2\times 314\times 220\times \cos 20° = 17168.068$$

$\therefore V_{r_1} = 131.02$ m/s.

Work done per kg of steam $= m(V_{w_1} + V_{w_2})u$

$= m(2V_1\cos\alpha_1 - u)u = 1(2\times 314\ \cos 20° - 220)220$ Nm $= 81521.1$ Nm.

Energy supplied $= \frac{V_1^2 + V_{r_2}^2 - V_{r_1}^2}{2} = \frac{2V_1^2 - V_{r_1}^2}{2}$ $[\therefore V_1 = Vr_2]$

$$= \frac{2\times(314)^2 - (131.02)^2}{2} = 90012.88 \text{ Nm.}$$

$\therefore$ Diagram efficiency $= \frac{\text{Work done}}{\text{Energy supplied}} = \frac{81521.1}{90012.88} = 0.9056 = 90.56\%$

For maximum efficiency, $u = V_1 \cos\alpha_1 = 314 \ \cos 20^\circ = 295$ m/s

For this blade speed of 295 m/s, $V_{r_1}^2 = (314)^2 + (295)^2 - 2\times 314 \times 295 \ \cos 20^\circ$

$\therefore Vr_1 = 110$ m/s.

$\therefore$ Diagram efficiency $= \frac{\text{Work done}}{\text{Energy supplied}}$

$$= \frac{2u(2V_1\cos\alpha_1 - u)}{\left(V_1^2 + V_{r_2}^2 - V_{r_1}^2\right)} = \frac{2\times 295(2\times 314\cos 20^\circ - 295)}{(314)^2 + (314)^2 - (110)^2} = 0.938 = 93.8\%$$

$\therefore$ Percentage increase in diagram efficiency $= \frac{0.938 - 0.905}{0.905} = 0.0365 = 3.65\%$

$\therefore$ The best rotor speed

$$N = \frac{60u}{\pi D} = \frac{60\times 295}{\pi \times 1.4} = 4025 \text{ r.p.m.}$$

In a power plant the steam is supplied at a pressure of 30 bar and temperature 300°C to the high pressure side of the turbine where it is expanded to 5 bar. The steam is then removed and reheated to 300°C at constant pressure. It is then expanded to the low pressure side of the turbine to 0.05 bar. Find the efficiency of the cycle with and without reheating.

Solution: Given, $P_1 = 30$ bar,

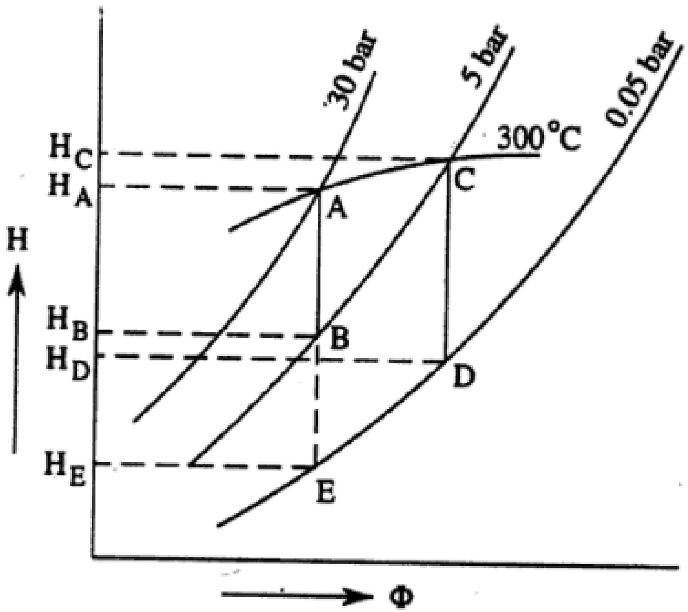


Fig. 14.33

$T_1 = 300°C$, $P_2 = 5$ bar, $P_3 = 0.05$ bar

Temperature of reheated steam = 300°C

From Mollier diagram

$H_A = 2990$ kJ/kg, $H_b = 2625$ kJ/kg, $H_C = 3075$ kJ/kg,

$H_D = 2595$ kJ/kg, $H_E = 2280$ kJ/kg.

From steam table $h_D = h_E = 340.6$ kJ/kg.

With reheating

$$\eta = \frac{(H_A - H_B) + (H_C - H_D)}{H_A + H_C - H_B - h_D} = \frac{(2990 - 2625) + (3075 - 2595)}{2990 + 3075 - 2625 - 340.6} = 0.273 = 27.3\%.$$

1.1. The velocity of steam leaving the nozzle of an impulse turbine is 900 m/s and nozzle angle is 20°. The blade velocity is 300 m/s and and the blade velocity coefficient is 0.7. Calculate for a mass flow of 1 kg/s and symmetrical blading.

(a) The blade inlet angle.

(b) The driving force on the wheel.

(c) The axial thrust.

(d) The diagram power. **[JNVU, 1990]**

Solution : Given :

$$C_1 = 900\ m/s,\ \ \alpha_1 = 20^o, U = 300\ m/s \quad k = 0.7,\ \ \dot{m} = 1\ kg/s$$

From inlet velocity triangle

$$C_{a1} = C_1 \sin \alpha_1 = 900 \sin 20^o = 307.8\ m/s$$

$$C_{w1} = C_1 \cos \alpha_1 = 900 \cos 20^o = 845.72\ m/s$$

$$\therefore C_{w1} - U = 845.72 - 300 = 545.72$$

Now

$$\tan \beta_1 = \frac{C_{a1}}{C_{w1} - U} = \frac{307.8}{545.72} = 0.5640255$$

$$\therefore \beta_1 = \tan^{-1}(0.5640255) = 29^o.25'$$

$$V_1 = \frac{C_{a1}}{\sin \beta_1} = \frac{307.8}{\sin 29^o.25'} = \frac{307.8}{0.4912702} = 626.54\ m/s$$

$$V_2 = K\ V_1 = 0.7 \times 626.54 = 438.58\ m/s$$

Given $\beta_1 = \beta_2 = 29^o25'$

$$C_{w2} = V_2 \cos \beta - U = 438.58 \cos 29^o25' - 300$$

$$= 382 - 300 = 82\ m/s$$

$$\therefore \Delta C_w = C_{w1} - (-C_{w2}) = 845.72 + 82$$

$$= 927.72\ m/s$$

(b) The driving force on the wheel

$$F = \dot{m}\,\Delta\, C_w = 1 \times 927.72 = 927.72\ N/kg/s$$

$$C_{a2} = V_2\ \sin\ \beta_2 = 438.58\ \sin\ 29^o25' = 215.46$$

$$\therefore \Delta\ C_a = C_{a1} - C_{a2} = 307.8 - 215.46 = 92.34\ m/s$$

(c) The axial thrust = $\dot{m}\ \Delta\ C_a$

$$= 1 \times 92.34 = 92.34\ N/kg/s$$

(d) The diagram power

$$= \frac{\dot{m}\ \Delta\ C_w \times U}{1000}$$

$$= \frac{1 \times 927.72 \times 300}{1000} = 278.316\ kW$$

1.2 In a simple impulse turbine the nozzles are inclined at 20º to the direction of motion of the moving blades. The steam leaves the nozzle at 375 m/s. The blade speed is 165 m/s. Calculate suitable inlet and outlet angles for the blades in order that the axial thrust is zero. The relative velocity of steam as it flows over the blades is reduced by 15% by friction. Also, determine the power developed for a flow rate of 10 kg/s. **[JNVU, 1991]**

Solution : Given

$$C_1 = 375\ m/s,\ \ \alpha_1 = 20^o\ \ C_{a1} = C_{a2}\ \text{(No axial thrust)}\ \ U = 165\ m/s$$

From inlet velocity triangle

$$C_{w1} = C_1 \cos \alpha_1 = 375 \cos 20^o = 352.38 \; m/s$$

$$C_{a1} = C_1 \sin \alpha_1 = 375 \sin 20^o = 128.25 \; m/s$$

$$= C_{a2}$$

$$\tan \beta_1 = \frac{C_{a1}}{C_{w1} - U} = \frac{128.25}{187.38} = 0.6844$$

$$\therefore \beta_1 = 34.38^o$$

$$V_1 = \frac{C_{a1}}{\sin \beta_1} = \frac{128.25}{\sin 34.38^o} = 227.12 \; m/s$$

$$V_2 = 0.85 \; V_1 = 0.85 \times 227.12 = 193.05 \; m/s$$

$$\sin \beta_2 = \frac{C_{a2}}{V_2} = \frac{128.75}{190.05} = 0.6669$$

$$\therefore \beta_2 = 41.82^o$$

$$C_{w2} = U - V_2 \cos \beta_2$$

$$= 165 - 193.05 \; \cos 41.82^o$$

$$= 165 - 143.86 = 21.14 \; m/s$$

$$\Delta C_w = C_{w1} - C_{w2} = 352.38 - 21.14 = 331.24 \; m/s$$

$$P = \frac{\dot{m} \; \Delta C_w \; U}{1000} = \frac{10 \times 331.24 \times 165}{1000} = 546.546 \; kW$$

1.3 The mean diameter of the blades of an impulse turbine with a single row wheel is 1 m and the speed is 3000 rpm. The nozzle angle is 18°, the ratio of linear velocity of blade to absolute velocity of steam entering moving blade is 0.42, and the relative velocity at outlet from the blades to that at inlet is 0.84. The outlet angle of the blade is to be made 3° less than the inlet angle of the blade. The steam rate of flow is 8 kg/s. Determine :

(a) Tangential thrust on the blades.

(b) Axial thrust on the blades.

(c) Resultant thrust on the blades.

(d) Power developed. **[JNVU, 1998]**

Solution : Given :

$$D = 1m, \quad N = 3000\ rpm \quad \alpha_1 = 18^o, \quad \rho = \frac{U}{C_1} = 0.42$$

$$\frac{V_2}{V_1} = 0.84, \quad \beta_2 = \beta_1 - 3^o \quad \dot{m} = 8\ kg/s$$

$$U = \frac{\pi\ DN}{60} = \frac{\pi \times 1 \times 3000}{60} = 157.1 \quad m/s$$

$$\frac{U}{C_1} = 0.42 \qquad \therefore C_1 = \frac{157.1}{0.42} = 374 \quad m/s$$

$$C_{w1} = C_1\ \cos\ \alpha_1 = 374\ \cos\ 18^o = 355.7\ \ m/s$$

$$C_{a1} = C_1\ \sin\ \alpha_1 = 374\ \sin\ 18^o = 115.57\ m/s$$

$$\tan\ \beta_1 = \frac{C_{a1}}{C_{w1} - U} = \frac{115.57}{355.7 - 157.1} = \frac{115.57}{198.6} = 0.58$$

$$\therefore \beta_1 = \tan^{-1}(0.58) = 30.196^o$$

$$\therefore \beta_2 = \beta_1 - 3^o = 30.196 - 3 = 27.196^o$$

$$V_1\ \cos\ \beta_1 = C_{w1} - U = 198.6$$

$$\therefore V_1 = \frac{198.6}{\cos\ 30.196^o} = 229.78\ m/s$$

$$V_2 = 0.84 \times 229.78 = 193\ \ m/s$$

$$C_{a2} = V_2\ \sin\ \beta_2 = 193\ \sin\ 27.196^o = 88.2\ m/s$$

$$V_2 \cos \beta_2 = U + C_{w2}$$

$$193 \cos 27.196^{\circ} = 157.1 + C_{w2}$$

$$\therefore C_{w2} = 171.676 - 157.1 = 14.576 \ m/s$$

$$\Delta \ C_w = C_{w1} + C_{w2} = 355.7 + 14.576 = 370.276$$

$$\Delta \ C_a = C_{a1} - C_{a2} = 115.57 - 88.2 = 27.37$$

(a) $$F_t = \dot{m} \ \Delta \ C_w$$

$$= 8 \times 370.276 = 2962.2 \ N$$

(b) $$F_a = \dot{m} \ \Delta \ C_a$$

$$= 8 \times 27.37 = 218.96 \ N$$

(c) $$F_r = \sqrt{F_t^2 + F_a^2}$$

$$= \sqrt{8774676.235 + 47943.48}$$

$$= \sqrt{8822619.717} = 2970.29 \ N$$

(d) $$P = \frac{\dot{m} \ \Delta \ C_w \ U}{1000} = \frac{2962.2 \times 157.1}{1000}$$

$$= 465.36 \ kW$$

1.4 A simple impulse steam turbine has one ring of moving blades running at 150 m/s. The absolute velocity of steam at the exit from the stage is 85 m/s at an angle 80° from the tangential direction. Blade velocity coefficient is 0.82 and the rate of steam flowing through the stage is 2.5 kg/s. If the blades are equiangular, determine (a) blade angles, (b) nozzle angle, (c) absolute velocity of steam isuing from the nozzle, (d) axial thrust, (e) Power developed. (f) If the velocity of steam reaching nozzle as 90 m/s and nozzle efficiency 0.85, find the enthalpy drop in the nozzle. **[JNVU, 2000]**

Solution : Given :

$$U = 150\ m/s,\quad C_2 = 85\ m/s,\quad \alpha_2 = 80^o \quad K = 0.82,\quad \dot{m} = 2.5\ kg/s \qquad C_o = 90\ m/s$$

From the outlet velocity triangle

$$C_{a2} = C_2 \sin\alpha_2 = 85 \sin 80^o = 83.7\ m/s$$

$$C_{w2} = C_2 \cos\alpha_2 = 85 \cos 80^o = 14.76\ m/s$$

$$\tan\beta_2 = \frac{C_{a2}}{U + C_{w2}} = \frac{83.7}{150 + 14.76} = \frac{83.7}{164.76} = 0.5080$$

$$\therefore\ \beta_2 = \tan^{-1}(0.5080) = 26.94^o$$

(a) As per question, $\beta_1 = \beta_2 = 26.93^o$

$$V_2 \cos\beta_2 = U + C_{w2} = 164.76$$

$$\therefore V_2 = \frac{164.76}{\cos 26.93^o} = 184.8\ m/s$$

$$\therefore V_1 = \frac{V_2}{K} = \frac{184.8}{0.82} = 225.3675\ m/s$$

From inlet velocity triangle

$$C_{a1} = V_1 \sin\beta_1 = 225.3675 \sin 26.93^o$$

$$= 225.3675 \times 0.4529 = 102.1\ m/s$$

$$C_{w1} = V_1 \cos\beta_1 + U$$

$$= 225.3675 \cos 26.93^o + 150$$

$$= 200.927 + 150 = 350.927\ m/s$$

$$\tan\alpha_1 = \frac{C_{a1}}{C_{w1}} = \frac{102.1}{350.927} = 0.2909$$

(b) $\therefore \alpha_1 = \tan^{-1}(0.2909) = 16.22^o$

$$C_{w1} = C_1 \cos \alpha_1$$

(c) $\therefore C_1 = \dfrac{C_{w1}}{\cos \alpha_1} = \dfrac{350.927}{\cos 16.22^o} = 365.48\ m/s$

(d) $F_a = \dot{m}\,(C_{a1} - C_{a2})$

$$= 2.5\,(102.1 - 83.7) = 2.5 \times 18.4 = 46\ N$$

(e) $P = \dfrac{\dot{m}\ \Delta\ C_w\ U}{1000}$

$$= \frac{2.5 \times (350.927 + 14.76) \times 150}{1000} = 137.13\ kW$$

(f) Applying SFEE for the nozzle

$$h_0 + \frac{C_0^2}{2000} = h_1 + \frac{C_1^2}{2000}$$

$$\therefore \frac{C_1^2 - C_0^2}{2000} = (h_0 - h_1)$$

$$\frac{(365.48)^2 - (90)^2}{2000} = \Delta\ h$$

$$\therefore \Delta\ h = 62.73\ kJ/kg$$

Considering the nozzle efficiency

$$\therefore \ \Delta h = 62.73 \ kJ/kg$$

Considering the nozzle efficiency

$$\Delta h = \frac{62.73}{0.85} = 73.8 \ kJ/kg \qquad \text{Ans.}$$

1.5 Following data refer to one stage of an impulse turbine. Velocity of steam at inlet to the nozzle = 30 m/s Isentropic heat drop = 84 kJ Reheat of the steam due to friction is equal to 10% of the isentropic heat drop.

$$\frac{\text{Blade speed}}{\text{Tangential component of the steam speed}} = 0.5$$

Nozzle angle = 20°

Velocity coefficient of the blades = 0.95

Find the blade angles so that the steam may enter them without shock and leave them in an axial direction. **[JNVU, 1999]**

Solution : Given :

$$C_0 = 30 \ m/s, \quad \alpha_1 = 20^o, \quad \Delta h = 84 \ kJ \quad \alpha_2 = 90^o \quad K = 0.95, \quad \frac{U}{C_{w1}} = 0.5$$

10% of the isentropic heat drop $= 0.9 \times 84 = 75.6$

For the nozzle

$$\frac{C_1^2}{2000} = 75.6 + \frac{C_0^2}{2000}$$

$$C_1 = \sqrt{75.6 \times 2000 + (30)^2}$$

$$= \sqrt{151200 + 900} = \sqrt{152100} = 390\ m/s$$

$$\frac{U}{C_{w1}} = 0.5$$

$$\therefore\ U = C_1 \cos \alpha_1 \times 0.5 = 390 \cos 20^o \times 0.5 = 183.24\ m/s$$

$$C_{w1} = C_1 \cos \alpha_1 = 390 \cos 20^o = 366.48\ m/s$$

$$C_{a1} = C_1 \sin \alpha_1 = 390 \sin 20^o = 133.38\ m/s$$

$$C_{w1} - U = 366.48 - 183.24 = 183.24\ m/s$$

$$\tan \beta_1 = \frac{C_{a1}}{C_{w1} - U} = \frac{133.38}{183.24} = 0.72789$$

$$\therefore\ \beta_1 = 36.05^o$$

$$V_1 = \frac{C_{a1}}{\sin \beta_1} = \frac{133.38}{\sin 36.05^o} = 226.643\ m/s$$

$$V_2 = k\ V_1 = 0.95 \times 226.643 = 215.31\ m/s$$

From outlet velocity diagram $\left(C_{w2} = 0 \text{ Given}\right)$

$$V_2 \cos \beta_2 = U$$

$$\cos \beta_2 = \frac{U}{V_2} = \frac{183.24}{215.31} = 0.85$$

$$\therefore\ \beta_2 = 31.67^o$$

1.6 A single row, impulse turbine has blades whose inlet angle is 40° and exit angle 37°. The mean blade speed is 230 m/s and the nozzles are inclined at an angle of 27° to the plane of rotation of the blades. There is a 10% loss of relative velocity due to friction in the blades. The turbine uses 550 kg/hr of steam. Determine

(a) The nozzle velocity of the steam.

(b) The absolute velocity of the steam at exit.

(c) The power output of the turbine.

(d) The end thrust on the turbine. **[JNVU, 2005]**

Solution : Given :

$\alpha_1 = 27^\circ, \quad \beta_1 = 40^\circ, \quad \beta_2 = 37^\circ, \quad U = 230\ m/s$

From inlet velocity triangle

Using sine rule

$$\frac{U}{\sin 13^\circ} = \frac{C_1}{\sin 140^\circ} = \frac{V_1}{\sin 27^\circ}$$

$$\therefore \quad C_1 = \frac{U \sin 140^\circ}{\sin 13^\circ} = \frac{230 \times 0.64278}{0.22495} = 657.21\ m/s$$

$$V_1 = \frac{U \sin 27^\circ}{\sin 13^\circ} = \frac{230 \times 0.4539}{0.22495} = 464.18\ m/s$$

$$\therefore\ V_2 = 0.9\ V_1 = 417.762\ \ m/s$$

$$C_{a1} = V_1 \sin\ \beta_1 = 464.18\ \sin 40^\circ$$

$$C_{w1} = C_1 \cos 27^\circ = 657.21\ \cos 20^\circ = 585.578\ m/s$$

From outlet velocity triangle

$$\begin{aligned} C_{w2} &= V_2 \cos \beta_2 - U \\ &= 417.762 \cos 37^\circ - 230 \\ &= 333.639 - 230 \\ &= 103.639\ \ m/s \end{aligned}$$

$$C_{a2} = V_2 \sin \beta_2 = 417.762 \sin 37^\circ = 251.415\ m/s$$

From the triangle

$$C_2^2 = (C_{a2})^2 + (C_{w2})^2$$

$$= (251.415)^2 + (103.639)^2$$

$$= 73950.7696$$

$$\therefore \quad C_2 = 271.93 \quad m/s$$

$$\Delta C_w = C_{w1} + C_{w2} = 585.578 + 103.639 = 689.217$$

$$P = \frac{\dot{m} \ \Delta C_w \ U}{1000} = \frac{550}{60} \times \frac{689.217 \times 230}{1000}$$

$$= 24.218 \ kW$$

Axial thrust

$$F_a = \dot{m} \left(C_{a1} - C_{a2}\right)$$

$$= \frac{550}{3600} \left(298.369 - 251.415\right)$$

$$= 0.153 \times 46.99 = 7.1895 \ N$$

INDUSTRIAL APPLICATIONS

Industrial Applications

Steam Turbine

Steam turbines are a part of various industries, from medium to large scale, and include dozens of institutional applications.

- Chemical Industry: Providing heat and electricity to drive different processes in the chemical and pharmaceutical industries, steam turbines are integrated in the process of producing power.
- Waste Plants: Steam turbines help generate the power needed to harness energy from wastes.
- Oil & Gas: Used as a pump drive or a compressor, steam turbines support dozens of operations in the oiland gas industry.
- Sugar Mills: Offering high levels of efficiency and sustainable operations, steam turbines are used to produce green carbon-dioxide energy from bagasse.

Some of the most popular applications of a steam turbine in different industries include the following:

1. **Combined Heat and Power**

Steam turbines are an essential component of most CHP systems. They support combined heat and power systems that are used to power industrial processes, under conditions where waste fuels are available for the boiler to safely utilize. When used for CHPs, the steam emitted by the steam turbine can be used directly. Steam turbine powered CHPs are typically found in paper mills, where there is an abundance of waste fuels ranging from black liquor to hog fuel, each equally successfully in powering the boiler. They can also be found in chemical plants that make excessive use of steam turbines; followed by their use of metals.

2. **Driving Mechanical Equipment**

Steam turbines are a far more efficient alternative to electrical power. Especially when it comes to driving different equipment like air compressors, boiler feed water pumps, refrigerator chillers, etc.

3. **District Heating & Cooling Systems**

Different institutions throughout different cities rely on district cooling and heating systems. These systems usually have a steam turbine placed between the boiler and the distribution system or placed as a replacement for a pressure reduction station.It is to be noted that, more often, boilers are capable of producing moderate-pressure steam while the distribution requires low pressure steam. Bridging this gap between the two, a steam turbine generates energy using the high pressure steam and emits low pressure steam into the distribution system.

4. **Combined Cycle Power Plants**

Steam turbines allow power plants to generate power using a gas turbine and utilize gas and heat produced in the process to generate steam that, in turn, produces additional power. Combined cycle power plants supported by steam turbines are capable of producing or accomplishing electric generation efficiencies extending beyond the 50-percent mark and are used in large industrial applications.

Reaction Turbine

- Reaction turbine is used in wind power mills to generate electricity
- It is most widely used turbine in hydro-power plants, to generate electricity.
- It is the only turbine to get maximum power output from a low available water head and high velocity other than cross-flow turbine which not that efficient.

TUTORIAL QUESTIONS

Theory Questions:

1. What is turbine and classify them.
2. Derive the expression for maximum efficiency of reaction turbine. What is the condition for maximum blade efficiency of a 50% reaction turbine and its value?
3. Derive an expression for optimum stage efficiency of a reaction turbine.
4. Write a note on degree of reaction. Derive an expression for degree of reaction and show that inlet and outlet velocity triangles are symmetrical for a 50% degree of reaction turbine.
5. Sketch how efficiency varies with blade-steam velocity ratio.
6. Deduce an expression for work done per stage of a reaction blading?
7. Explain with neat sketch of impulse turbine with Pressure and velocity curves

Numerical Problems:

1. In a single-stage impulse turbine, the steam jet leaves the nozzles at 20º to the plane of the wheel at a speed of 670 m/s and it enters the moving blades at an angle of 35º to the drum axis. The moving blades are symmetrical in shape. Determine the blade velocity and diagram efficiency.
2. Write short notes on De-Laval Turbine and about its features. Steam leaves the nozzle of a single-stage impulse turbine at 840 m/s. The nozzle angle is 18º and the blade angles are 29º at the inlet and outlet. The friction coefficient is 0.9. Calculate
   (i) blade velocity (ii) steam mass flow rate in kg/h to develop 300 kW power.

ASSIGNMENT QUESTION

1. Discuss the relative advantages and disadvantages of gas turbines and steam turbines.
2. Define the following as related to steam turbines.
   i) Blade Speed ratio ii) blade velocity coefficient iii) diagram efficiency iv) stage efficiency
3. a) Prove that for a 50% reaction turbines $\alpha=\varphi$ and $\theta=\beta$
   b) Explain the difference between an impulse turbine and a reaction turbine
4. Derive the expression for maximum blade efficiency of a single stage impulse turbine.
5. Write the expression for blade efficiency for a single stage reaction turbine for getting the maximum blade efficiency.

UNIT 4

GAS TURBINES

Course Objective:

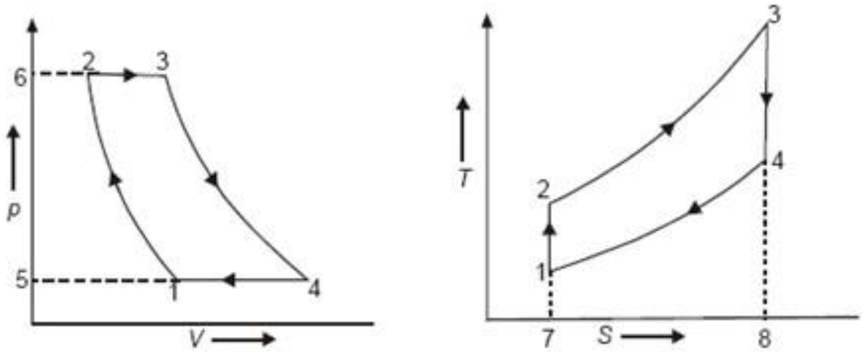
Student have knowledge of methods of analysis and design of complicated thermodynamic systems

Course Outcome:

To be able to recognize main and supplementary elements of turbines and define operational principles.

Brayton Cycle

Brayton cycle, popularly used for gas turbine power plants comprises of adiabatic compression process, constant pressure heat addition, adiabatic expansion process and constant pressure heat release process. A schematic diagram for air-standard Brayton cycle is shown in Fig. 4.1. Simple gas turbine power plant working on Brayton cycle is also shown here.



Brayton cycle on P–V and T–S diagram

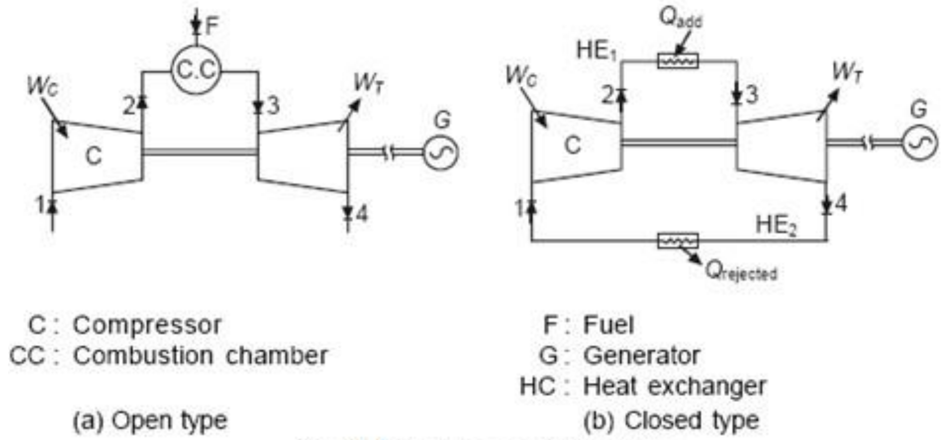


Fig. 4.1 *Simple gas turbine plant*

Thermodynamic cycle shows following processes:

1-2: Adiabatic compression, involving (–ve) work, W_C in compressor.

2-3 : Constant pressure heat addition, involving heat Q_{add} in combustion chamber or heat exchanger.

3-4: Adiabatic expansion, involving (+ve) work, W_T in turbine.

4-1: Constant pressure heat rejection, involving heat, $Q_{rejected}$ in atmosphere or heat exchanger.

In the gas turbine plant layout shown process 1–2 (adiabatic compression) is seen to occur in compressor, heat addition process 2–3 occurs in combustion chamber having open type arrangement and in heat exchanger in closed type arrangement. Process 3–4 of adiabatic expansion occurs in turbine.

In open type arrangement exhaust from turbine is discharged to atmosphere while in closed type, heat rejection occurs in heat exchanger. In gas turbine plant of open type, air entering compressor gets compressed and subsequently brought up to elevated temperature in combustion chamber where fuel is added to high pressure air and combustion occurs. High pressure and high temperature combustion products are sent for expansion in turbine where its' expansion yields positive work. Expanded combustion products are subsequently discharged to atmosphere. Negative work required for compression is drawn from the positive work available from turbine and residual positive work is available as shaft work for driving generator.

In gas turbine plant of closed type the working fluid is recycled and performs different processes without getting contaminated. Working fluid is compressed in compressor and subsequently heated up in heat exchanger through indirect heating. High pressure and high temperature working fluid is sent for getting positive work from turbine and the expanded working fluid leaving turbine is passed through heat exchanger where heat is picked up from working fluid. Thus, the arrangement shows that even costly working fluids can also be used in closed type as it remains uncontaminated and is being recycled.

Air standard analysis of Brayton cycle gives work for compression and expansion as;

$$W_C = m_1 \cdot (h_2 - h_1)$$

$$W_T = m_3 \cdot (h_3 - h_4)$$

for air standard analysis, $m_1 = m_3$, where as in actual cycle

$$m_3 = m_1 + m_f, \quad \text{in open type gas turbine}$$

$$m_3 = m_1, \quad \text{in closed type gas turbine}$$

For the fuel having calorific value CV the heat added in air standard cycle;

$Q_{add} = m_1(h_3 - h_2)$, whereas $Q_{add} = m_f \times CV$ for actual

cycle. Net work = $W_T - W_C$

$W_{net} = \{m_3\ (h_3 - h_4) - m_1(h_2 - h_1)\}$

$$\text{Air standard cycle efficiency} = \frac{W_{net}}{Q_{add}}$$

$$= \frac{m_1\{(h_3 - h_4) - (h_2 - h_1)\}}{m_1(h_3 - h_2)}$$

$$\text{Air standard Brayton cycle efficiency: } \boxed{\eta_{Brayton} = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}}$$

Thus, it is obvious from the expression of efficiency that it depends only on pressure ratio (r) and nature of gas (γ). For pressure ratio of unity, efficiency shall be zero. For a particular gas the cycle efficiency increases with increasing pressure ratio. Here the variation of efficiency with pressure ratio is shown for air (γ= 1.4) and monatomic gas as

argon (γ= 1.66).

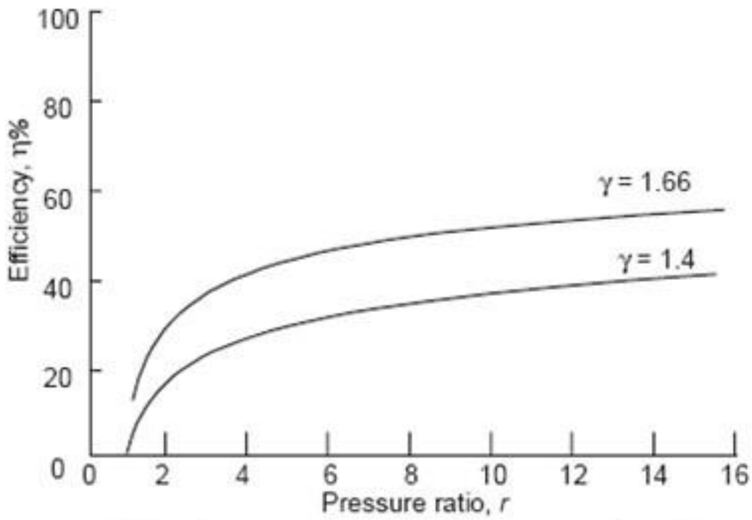


Fig. 4.2 *Efficiency vs. pressure ratio in simple cycle*

Regenerative gas turbine cycle

Regenerative air standard gas turbine cycle shown ahead in Fig. 4.3 has a regenerator (counter flow heat exchanger) through which the hot turbine exhaust gas and comparatively cooler air coming from compressor flow in opposite directions. Under ideal conditions, no frictional pressure drop occurs in either fluid stream while turbine exhaust gas gets cooled from 4 to 4′ while compressed air is heated from 2 to 2′. Assuming regenerator effectiveness as 100% the temperature rise from 2–2′ and drop from 4 to 4′ is shown on *T-S* diagram.

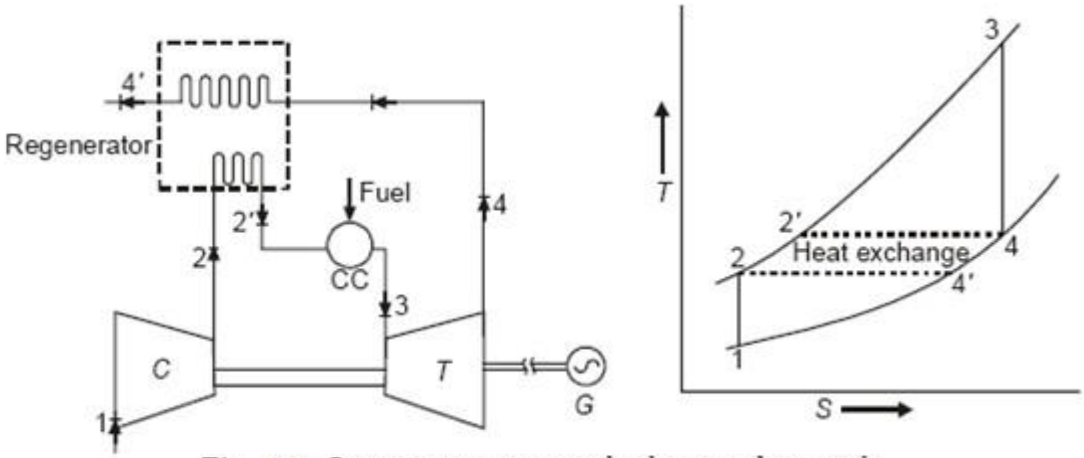


Fig. 4.3 *Regenerative air standard gas turbine cycle.*

Regenerator effectiveness, $\varepsilon = \dfrac{h_{2'} - h_2}{h_4 - h_2}$,

Thus, thermodynamically the amount of heat now added shall be

$$Q_{add,\, regen} = m\,(h_3 - h_2')$$

Where as without regenerator the heat added; $Q_{add} = m\,(h_3 -$

$h_2)$ Here it is obvious that, $Q_{add,\ regen} < Q_{add}$

This shows an obvious improvement in cycle thermal efficiency as every thing else remains same. Net work produced per unit mass flow is not altered by the use of regenerator.

$$\text{Air standard cycle thermal efficiency, } \eta_{regen} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_{2'})}$$

$$\eta_{regen} = \frac{c_p \cdot (T_3 - T_4) - c_p\,(T_2 - T_1)}{c_p\,(T_3 - T_{2'})}$$

eheat gas turbine cycle

Reheat gas turbine cycle arrangement is shown in Fig. 4.4. In order to maximize the
work available from the simple gas turbine cycle one of the option is to increase enthalpy of fluid entering gas turbine and extend its expansion upto the lowest possible enthalpy value.

C: Compressor HPT: High pressure turbine

CC: Combustion chamber LPT: Low pressure turbine

G: Generator RCC: Reheat combustion

chamber f: Fuel

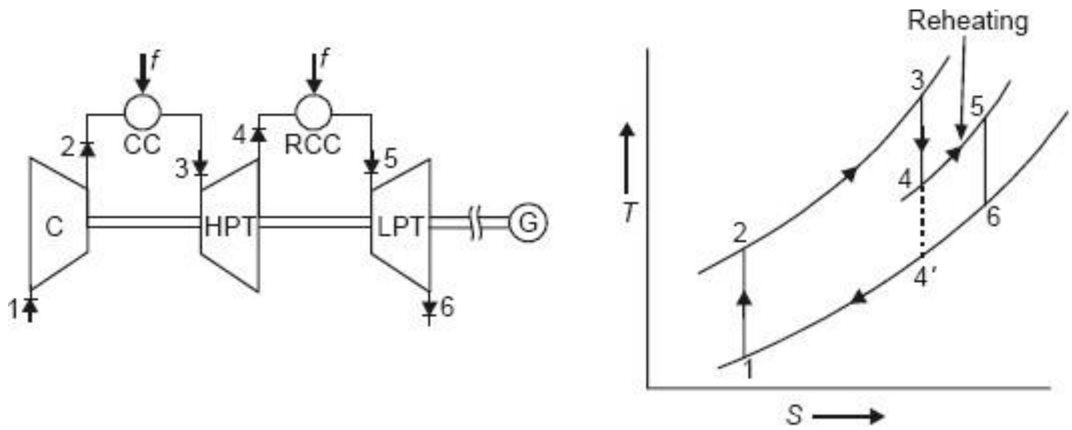


Fig. 4.4 Reheat gas turbine cycle

This can also be said in terms of pressure and temperature values i.e. inject fluid at high pressure and temperature into gas turbine and expand upto lowest possible pressure value. Upper limit at inlet to turbine is limited by metallurgical limits while lower pressure is limited to near atmospheric pressure in case of open cycle. Here in the arrangement shown ambient air enters compressor and compressed air at high pressure leaves at 2. Compressed air is injected into combustion chamber for increasing its temperature up to desired turbine inlet temperature at state 3. High pressure and high temperature fluid enters high pressure turbine (HPT) for first phase of expansion and expanded gases leaving at 4 are sent to reheat combustion chamber (reheater) for being further heated. Thus, reheating is a kind of energizing the working fluid.

Assuming perfect reheating (in which temperature after reheat is same as temperature attained in first combustion chamber), the fluid leaves at state 5 and enters low pressure turbine (LPT) for remaining expansion upto desired pressure value. Generally,

temperature after reheating at state 5 is less than temperature at state 3. In the absence of reheating the expansion process within similar pressure limits goes upto state 4′. Thus, reheating offers an obvious advantage of work output increase since constant pressure

lines on *T-S* diagram diverge slightly with increasing entropy, the total work of the two stage turbine is greater than that of single expansion from state 3 to state 4′. i.e.,

$$(T3 - T4) + (T5 - T6) > (T3 - T4').$$

Here it may be noted that the heat addition also increases because of additional heat supplied for reheating. Therefore, despite the increase in net work due to reheating the cycle thermal efficiency would not necessarily increase.

A plot showing variation of efficiency with pressure ratio 'r' is shown in Fig. 4.5 along with simple cycle efficiency variation. It indicates that reheating offers increase in specific work output at the

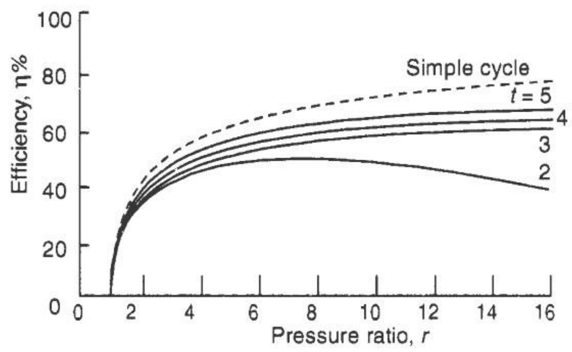


***Fig.** 4.5 Reheat cycle efficiency vs. cycle pressure ratio*

cost of cycle efficiency. This reduction in efficiency may be attributed to the addition of a less efficient cycle 4564′ to a simple cycle. 4564′ is a less efficient cycle since it operates over a smaller temperature range. Variation of specific work output with pressure ratio is shown in Fig. 4.6. It shows how specific work output shows increase with increasing pressure ratio upto optimum pressure ratio. It may also be noted that in reheat cycle, the temperature of exhaust gases at exit of gas turbine gets increased as compared to simple cycle within similar limits. Therefore, reheat cycle offers potential for use of regenerator for harnessing the hotter exhaust from gas turbine.

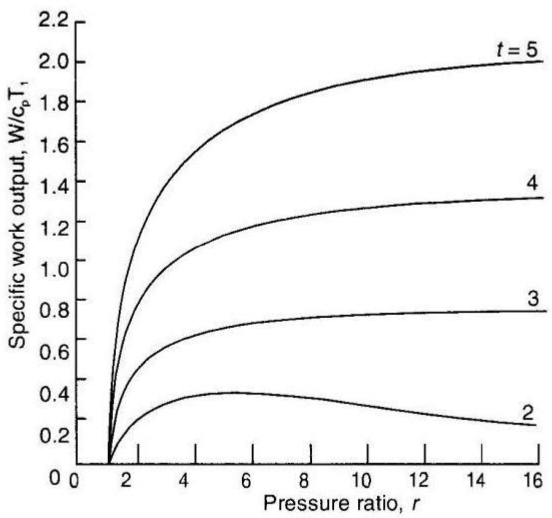
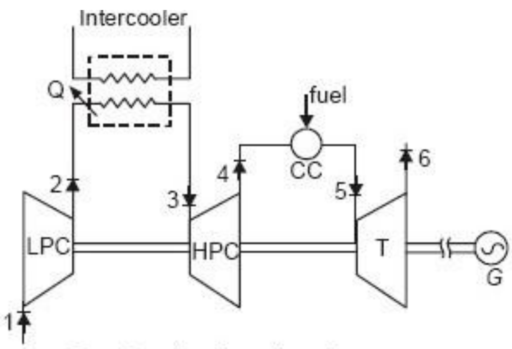


***Fig.** 4.6 Reheat cycle specific work output vs. cycle pressure ratio*

GAS TURBINE CYCLE WITH INTERCOOLING

Net work output from gas turbine cycle can also be increased by reducing negative work i.e. compressor work. Multistaging of compression process with intercooling in between is one of the approach for reducing compression work. It is based on the fact that for a fixed compression ratio higher is the inlet temperature higher shall be compression work requirement and vice-a-versa. Schematic for intercooled gas turbine cycle is given in Fig. 4.7.

Thermodynamic processes involved in multistage intercooled compression are shown in Figs.
4.8, 4.9. First stage compression occurs in low pressure compressor (LPC) and compressed air leaving LPC at '2' is sent to intercooler where temperature of compressed air is lowered down to state 3 at constant pressure. In case of perfect intercooling the temperatures at 3 and 1 are same. Intercooler is a kind of heat exchanger where heat is picked up from high temperature compressed air. The amount of compression work saved due to intercooling is obvious from *p-V* diagram and shown by area 2342′. Area 2342′ gives the amount of work saved due to intercooling between compression.



CC : Combustion chamber
LPC : Low pressure compressor
HPC : High pressure compressor
T : Turbine
G : Generator

Fig. 4.7 *Gas turbine cycle with intercooling*

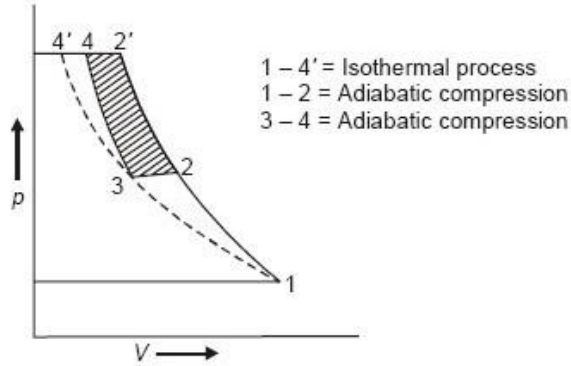


Fig. 4.8 *Intercooled compression*

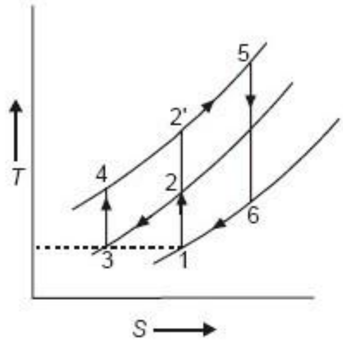


Fig. 4.9 *T-S diagram for gas turbine*

for gas turbine cycle with intercooling shows that in the absence of intercooling within same pressure limits the state at the end of compression would be 2′ while with perfect intercooling
this state is at 4 i.e., $T2' > T4$. The reduced temperature at compressor

exit leads to additional heat requirement in combustion chamber i.e. more amount of fuel is to be burnt for attaining certain turbine inlet temperature as compared to simple cycle without intercooling.

Thus, intercooled cycle thermal efficiency may not increase with intercooling because of simultaneous increase in heat addition requirement. The lower temperature at compressor exit enhances the potential for regeneration so when intercooling is used in conjunction with regeneration an appreciable increase in thermal efficiency can result.

Net work output in gas turbine cycle with intercooling;

$$W_{\text{net, intercool}} = m\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}$$

$$W_{\text{net, intercool}} = mc_p\{(T_5 - T_6) - (T_4 - T_3) - (T_2 - T_1)\}$$

Cycle thermal efficiency;

$$\eta_{\text{intercool}} = \frac{\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}}{\{h_5 - h_4\}}$$

GAS TURBINE CYCLE WITH REGENERATION, REHEAT AND INTERCOOLING

Regenerative gas turbine employing reheating during expansion and intercooling during compression is considered here as shown in Fig. 4.13 . This combination offers considerable increase in net work output and thermal efficiency.

C_1 : Low pressure compressor stage
C_2 : High pressure compressor stage
CC : Combustion chamber
RCC : Reheat combustion chamber
T_1 : High pressure turbine stage
T_2 : Low pressure turbine stage
G : Generator

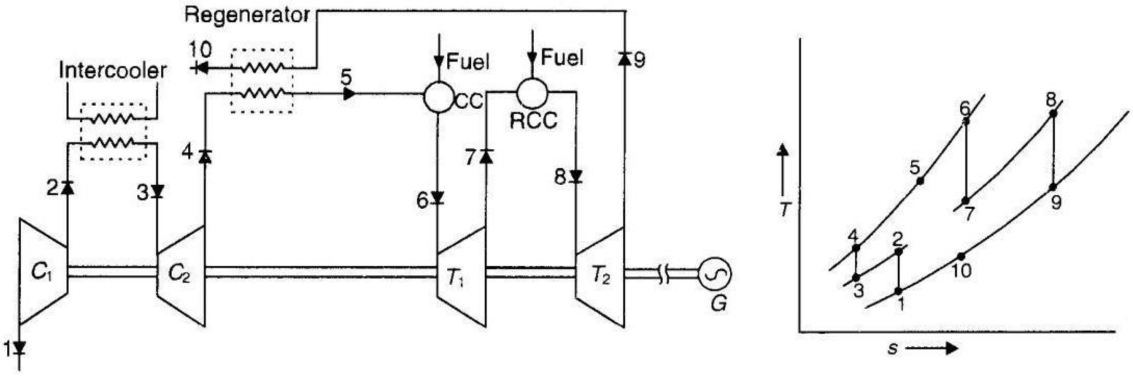


Fig. 4.13 *Schematic for gas turbine cycle with regeneration, reheat and intercooling and T-S diagram*

Based upon air standard cycle considerations thermodynamic analysis gives the Net work output from cycle,

$$W_{net} = m\ \{(h_6 - h_7) + (h_8 - h_9) - (h_4 - h_3) - (h_2 - h_1)\}$$

Heat added

$$Q_{add} = m\{(h_6 - h_5) + (h_8 - h_7)\}$$

Cycle thermal efficiency,

$$\eta_{cycle} = \frac{\{(h_6 - h_7) + (h_8 - h_9) - (h_4 - h_3) - (h_2 - h_1)\}}{\{(h_6 - h_5) + (h_8 - h_7)\}}$$

GAS TURBINE IRREVERSIBILITIES AND LOSSES

Till now the discussions have been confined to air standard Brayton cycle. But the realistic gas turbine cycle has deviations from air standard cycle due to,

(*i*) frictional effects within compressor and turbine which causes increase in specific entropy of working fluid across these components.

(*ii*) friction which shall cause drop in pressure of working fluid across the constant pressure processes.

Apart from above irreversibilities of the gas turbine power plant the irreversibilities of combustion chamber are quite significant.

Salient state points of realistic gas turbine Brayton cycle with above irreversibilities and losses are shown below:

Isentropic efficiency of turbine and compressor can be mathematically given as

$$\eta_{\text{isen, }t} = \left\{\frac{h_3 - h_4}{h_3 - h_{4s}}\right\}$$

$$\text{i.e. } \eta_{\text{isen, }t} = \frac{\text{Actual expansion work}}{\text{Ideal expansion work}}$$

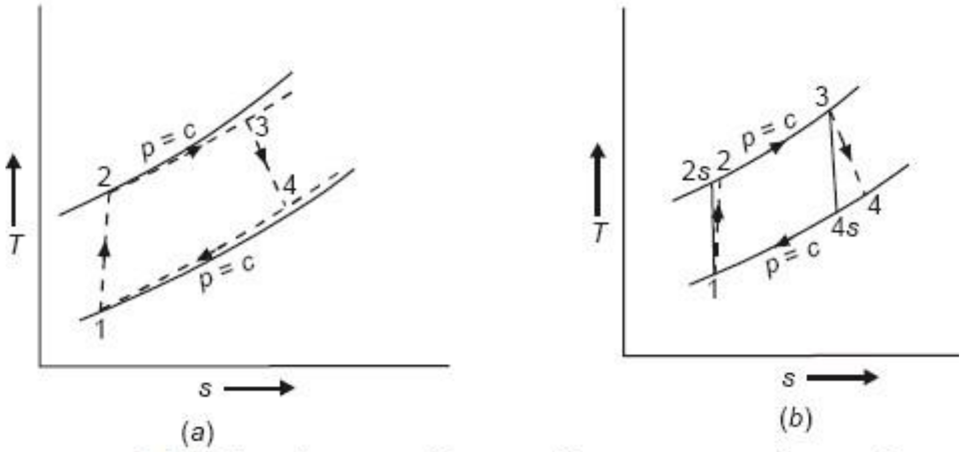


Fig. 4.14 *Effect of irreversibilities and losses in gas turbine cycle.*

Isentropic efficiency of compressor

$$\eta_{\text{isen, }c} = \left\{\frac{h_{2s} - h_1}{h_2 - h_1}\right\}$$

$$\eta_{\text{isen, }c} = \frac{\text{Ideal compressor work}}{\text{Actual compressor work}}$$

Other factors causing the real cycle to be different from ideal cycle are as given below:

(i) Fluid velocities in turbomachines are very high and there exists substantial change in kinetic energy between inlet and outlet of each component. In the analysis carried out earlier the changes in kinetic energy have been neglected whereas for exact analysis it cannot be.

(ii) In case of regenerator the compressed air cannot be heated to the temperature of gas leaving turbine as the terminal temperature difference shall always exist.

(iii) Compression process shall involve work more than theoretically estimated value in order to overcome bearing and windage friction losses.

Different factors described above can be accounted for by stagnation properties, compressor and turbine isentropic efficiency and polytropic efficiency.

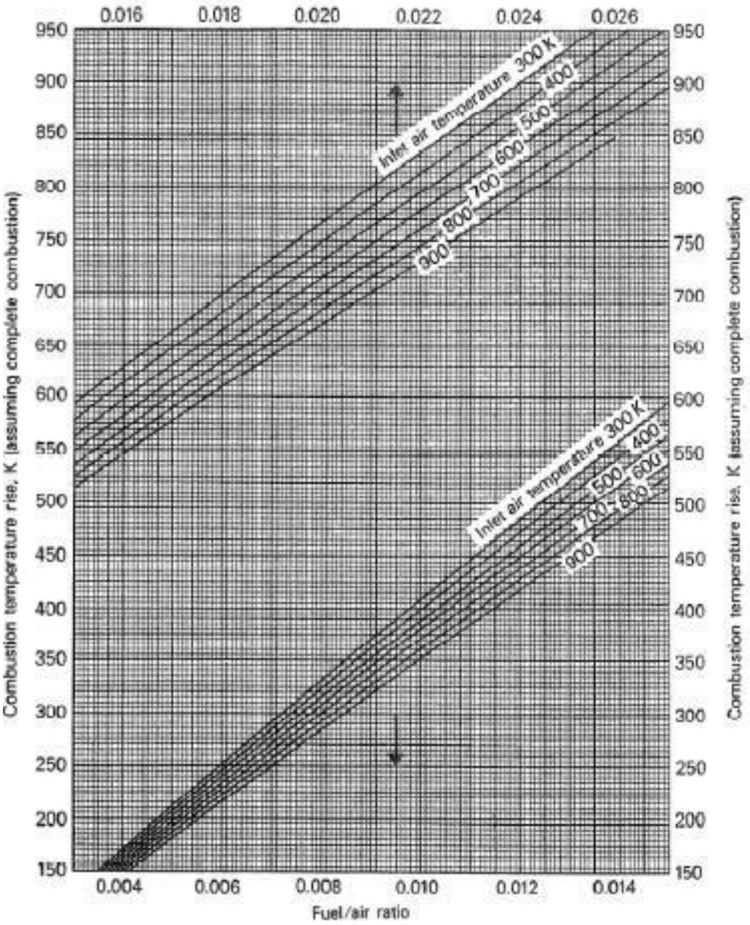
$T_{02} = 164.7 + 288 = 452.7$K, and $T_{04} = 1100\text{-}264.8 = 835.2$ K

Hence,

$T_{05} = 0.80*382.5+452.7 = 758.7$ K

For a combustion chamber inlet air temperature of 759 K and a combustion temperature rise of (1100-759) = 341 K, the theoretical fuel/air ratio required is 0.0094 (from the chart of slide 12), and thus

$$f = \frac{theoretical\_f}{\eta_b} = \frac{0.0094}{0.98} = 0.0096$$

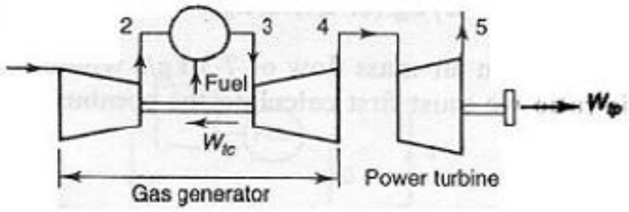


Theoretical Fuel-Air Ratio

Combustion temperature rise vs theoretical fuel-air ratio

Determine the specific work output, specific fuel consumption and cycle efficiency for a simple cycle gas turbine with a free power turbine (see figure) given the following specification:

Compressor pressure ratio	12.0
Turbine inlet temperature	1350 K
Isentropic efficiency of compressor, η_c	0.86
Isentropic efficiency of each turbine, η_t	0.89
Mechanical efficiency of each shaft, η_m	0.99
Combustion efficiency	0.99
Combustion chamber pressure loss	6 % compressor delivery pressure
Exhaust pressure loss	0.03 bar
Ambient conditions, p_a, T_a	1 bar, 288 K



Free turbine unit

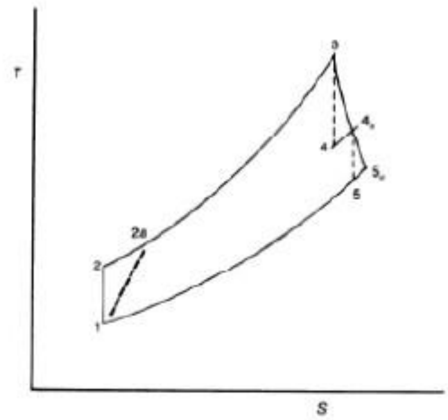
Solution:

Proceeding as in the previous example,

$$T_{02} - T_{01} = \frac{288}{0.86}\left[12^{1/3.5} - 1\right] = 346.3\,\text{K}$$

$$W_{tc} = \frac{1.005 * 346.3}{0.99} = 351.5\,\text{kJ/kg}$$

$$p_{03} = 12.0(1 - 0.06) = 11.28\,\text{bar}$$



The intermediate pressure between the two turbines, p_{04}, is unknown, but can be determined from the fact that the compressor turbine produces just sufficient work to drive the compressor. The temperature equivalent of the compressor turbine work is, therefore,

$$T_{03} - T_{04} = \frac{W_{tc}}{c_{pg}} = \frac{351.5}{1.148} = 306.2\,\text{K}$$

The corresponding pressure ratio can be found using the relation

$$T_{03} - T_{04} = \eta_t T_{03}\left[1-\left(\frac{1}{p_{03}/p_{04}}\right)^{\gamma-1/\gamma}\right]$$

$$306.2 = 0.89*1350\left[1-\left(\frac{1}{p_{03}/p_{04}}\right)^{0.25}\right]$$

$$\frac{p_{03}}{p_{04}} = 3.243$$

$$T_{04} = 1350 - 306.2 = 1043.8\,K$$

The pressure at entry to the power turbine, p_{04}, is then found to be

$$p_{04} = \frac{p_{03}}{p_{03}/p_{04}} = 11.28/3.243 = 3.478 \text{ bar}$$

and the power turbine pressure ratio is

$$p_{04}/p_{05} = 3.478/(1+0.03) = 3.377$$

The temperature drop in the power turbine can now be obtained

$$T_{04} - T_{05} = 0.89*1043.8\left[1-\left(\frac{1}{3.377}\right)^{0.25}\right] = 243.7\text{ K}$$

and the specific work output, i.e. power turbine work per unit air mass flow, is

$$W_{tp} = c_{pg}(T_{04} - T_{05})\eta_m$$

$$W_{tp} = 1.148(243.7)0.99 = 277.0 \text{ kJ/kg (or kW/kg)}$$

The compressor delivery temperature is 288+346.3 = 634.3 K and the combustion temperature rise is 1350 – 634.3 = 715.7 K

The theoretical fuel/air ratio required is 0.0202 (from the chart in slide 12), giving an actual fuel/air ratio of 0.0202/0.99 = 0.0204

The SFC and cycle efficiency, η, are then given by

$$SFC = \frac{f}{W_{tp}} = \frac{3600*0.0204}{277.9} = 0.265 \text{ kg/kWh}$$

$$\eta = \frac{3600}{0.265*43100} = 0.315$$

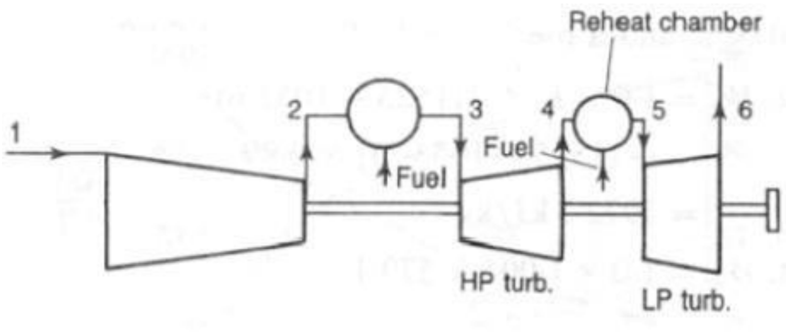
- The cycle calculations are carried out to determine the overall performance. It However, they also provide information required by other groups such as the aerodynamic and control design groups.
- *E.g.*, the temperature at entry to the power turbine, T_{04}, may be required as a control parameter to prevent operation above the metallurgical limiting temperature of the compressor turbine.
- The exhaust gas temperature (EGT), T_{05}, would be important if the gas turbine were to be considered for combined cycle or cogeneration plant.
- The temperature, $T_{05} = 1043.8 - 243.7 = 800.1\text{K}$ or 527° C, is suitable for use with a waste heat boiler.
- For a combined cycle plant, a higher T I T might be desirable because there would be a consequential increase in EGT, permitting the use of a higher steam temperature and a more efficient steam cycle.
- If the cycle pressure ratio were increased to increase the efficiency of the gas cycle, however, the EGT would be decreased resulting in a lower steam cycle efficiency.

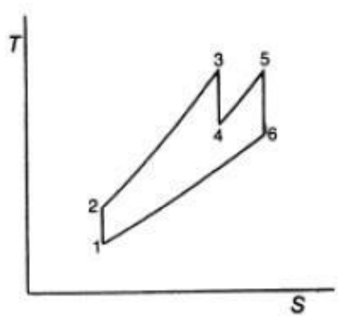
Example 3:

Consider the design of a high pressure ratio, single-shaft cycle with reheat at some point in the expansion when used either as a separate unit,or as part of a combined cycle. The power required is 240 MW at 288 K and 1.01 bar

Compressor pressure ratio	30
Polytropic efficiency	
(compressor and turbines)	0.89
Turbine inlet temperature	
(both turbines)	1525 K
$\Delta p/p_{02}$ (1 st combustor)	0.02
$\Delta p/p_{04}$ (2 nd combustor)	0.04
Exhaust pressure	1.02 bar

Solution:





A heat Exchanger is not used because it would result in an exhaust temperature that would be too low for use with a high efficiency steam cycle.

The Assumptions are as follows

- Let us assume that the mass flow rate is constant throughout, ignoring the effect of substantial cooling bleeds that would be required with high turbine inlet temperatures specified.
- The reheat pressure is not specified. So, as a starting point we use a value giving equal pressure ratio in each turbine.
- This division of the expansion leads to equal work in each turbine and a maximum net work output for the ideal reheat cycle).

From polytropic relations:

$$\text{for compression,} \quad \frac{n-1}{n} = \frac{1}{\eta_{\infty c}}\left(\frac{\gamma-1}{\gamma}\right) = \frac{1}{0.89}\left(\frac{0.4}{1.4}\right) = 0.3210$$

$$\text{for expansion,} \quad \frac{n-1}{n} = \eta_{\infty t}\left(\frac{\gamma-1}{\gamma}\right) = 0.89\left(\frac{0.333}{1.333}\right) = 0.2223$$

Assuming that $p_{01} = p_a$ and $T_{01} = T_a$, we have $T_{02}/T_{01} = (30)^{0.3210}$

$$T_{02} = 858.1 \text{ K}$$

$$T_{02} - T_{01} = 570.1 \text{ K}$$

$$p_{02} = 30*1.01 = 30.3 \text{ bar}$$

$$p_{03} = 30.3(1.00\text{-}0.02) = 29.69 \text{ bar}$$

$$p_{06} = 1.02 \text{ bar, so} \quad p_{03}/p_{06} = 29.11$$

Theoretically, the optimum pressure ratio for each turbine would be

$$\sqrt{(29.11)} = 5.395$$

A pressure loss of 4 % in the reheat combustor has to be considered, so a value of 5.3 for p_{03}/p_{04} could be assumed. Then,

$$\frac{T_{03}}{T_{04}} = (5.3)^{0.2223}$$

$$T_{04} = 1052.6K$$

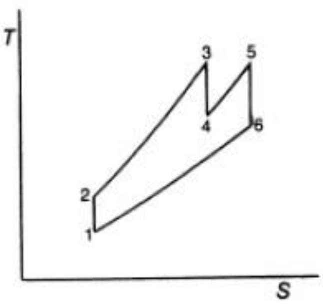
$$p_{04} = 29.69/5.3 = 5.602bar$$

$$p_{05} = 5.602(1.00 - 0.04) = 5.378bar$$

$$p_{05}/p_{06} = 5.378/1.02 = 5.272$$

$$\frac{T_{05}}{T_{06}} = (5.272)^{0.2223}$$

$$T_{06} = 1053.8K$$



Assuming unit flow of 1.0 kg/s and a mechanical efficiency of 0.99,

Turbine output, W_t = 1.0*1.148{(1525-1052.6)+(1525-1053.8)}*0.99

= 1072.3 kJ/kg

Compressor input, W_c = 1.0*1.005*570.1

= 573.0 kJ/kg

Net work output, W_N = 1072.3-573.0 =499.3 kJ/kg

Flow required for 240 MW is given by

m = 240000/499.3

= 480.6 kg/s

- For the first combustor, temperature rise = 1525 - 858 = 667 K, inlet temperature = 858 K and fuel/air ratio = 0.0197 (*from the chart of Slide 12*)
- For the second combustor, temperature rise = 1525-1052.6 = 472.4 K, inlet temperature = 1052.6 K and fuel/air ratio = 0.0142(*from the chart of Slide 12*)

- Actual total fuel/air ratio $f = \dfrac{0.0197 + 0.0142}{0.99} = 0.0342$
- And, thermal efficiency $\eta = \dfrac{499.3}{0.0342 * 43100} = 33.9\%$

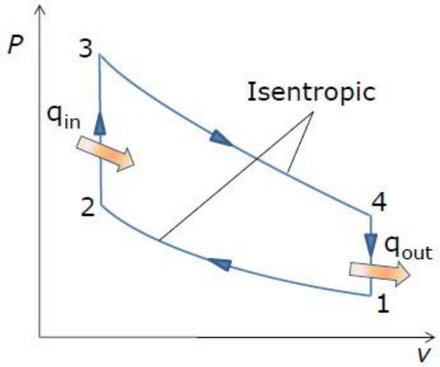
- This is a reasonable efficiency for simple cycle operation, and the specific output is excellent.
- However, the turbine exit temperature (T_{06} = 1053.8 K or 780.8° C) is too high for efficient use in a combined cycle plant. A reheat steam cycle using conventional steam temperatures of about 550° -575° C would require a turbine exit temperature of about 600° C.

- The turbine exit temperature could be reduced by increasing the reheat pressure, and if the calculations are repeated for a range of reheat pressures, then the results obtained are as shown in the Slide 26.
- It can be seen that a reheat pressure of 13 bar gives an exhaust gas temperature (EGT) of 605° C; the specific output is about 10 percent lower than the optimum value, but the thermal efficiency is substantially improved to 37.7 per cent. Further increases in reheat pressure would give slightly higher efficiencies, but the EGT would be reduced below 600° C resulting in a less efficient steam cycle.

Problems for practice

In an air standard Otto cycle, the compression ratio is 7 and the compression begins at 35oC and 0.1 MPa. The maximum temperature of the cycle is 1100oC. Find (a) the temperature and the pressure at various points in the cycle, (b) the heat supplied per kg of air, (c) work done per kg of air, (d) the cycle efficiency and (e) the MEP of the cycle.

Solution: Problem 1



T_1=35°C=308 K
P_1=0.1 Mpa
T_3=1100°C=1373 K
$r=v_1/v_2=7$

- Since process, 1-2 is isentropic,

$$\frac{P_2}{P_1}=\left(\frac{v_1}{v_2}\right)^{\gamma}=7^{1.4}=15.24$$

- Hence, P_2=1524 kPa

$$\frac{T_2}{T_1}=\left(\frac{v_1}{v_2}\right)^{\gamma-1}=7^{1.4-1}=2.178$$

- Hence, T_2=670.8 K

- For process, 2-3,

$$\frac{P_2 v_2}{T_2} = \frac{P_3 v_3}{T_3}, \ \therefore P_3 = \frac{T_3}{T_2} P_2 = \frac{1373}{607.8} \times 1524 = 3119.34$$

- P_3=3119.34 kPa.
- Process 3-4 is again isentropic,

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = 7^{1.4-1} = 2.178$$

$$\therefore T_4 = \frac{1373}{2.178} = 630.39\,\mathrm{K}$$

- Hence, T_2=630.39 K

- Heat input,

$Q_{in}=c_v(T_3-T_2)$
$=0.718(1373–670.8)$
$=504.18$ kJ/kg

- Heat rejected,

$Q_{out}=c_v(T_4-T_1)$
$=0.718(630.34–308)$
$=231.44$ kJ/kg

- The net work output, $W_{net}=Q_{in}-Q_{out}$

- The net work output,

$W_{net}=Q_{in}-Q_{out}$
$=272.74$ kJ/kg

- Thermal efficiency, $\eta_{th,otto}=W_{net}/Q_{in}$
$=0.54$
$=54\ \%$

- Otto cycle thermal efficiency,

$\eta_{th,otto} = 1-1/r^{\gamma-1} = 1-1/7^{0.4}$
$= 0.54$ or $54\ \%$

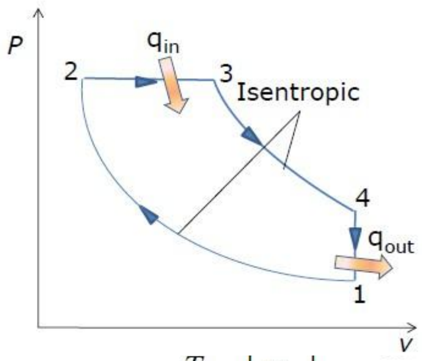
- $v_1=RT_1/P_1$
$=0.287\text{x}308/100=0.844\ m^3/kg$

- $MEP = W_{net}/(v_1-v_2) = 272.74/v_1\,(1-1/r)$
$=272.74/0.844(1-1/7)$
$=360$ kPa

Problem 2

In a Diesel cycle, the compression ratio is 15. Compression begins at 0.1 Mpa, 40oC. The heat added is 1.675 MJ/kg. Find (a) the maximum temperature in the cycle, (b) work done per kg of air (c) the cycle efficiency (d) the temperature at the end of the isentropic expansion (e) the cut-
off ratio and (f) the MEP of the cycle.

Solution: Problem 2



T_1=40°C=313 K
P_1=0.1 Mpa
Q_{in}=1675 MJ/kg
$r=v_1/v_2=15$

$$T_1 \quad (v_2)$$

$$T_2 = 313 \times 2.954 = 924.66\,\text{K}$$

$$Q_{in} = 1675 = 1.005(T_3 - 924.66)$$
$$\therefore T_3 = 2591.33\ \text{K} = T_{\max}$$

- Hence, the maximum temperature is 2591.33 K

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = 15^{1.4} = 44.31$$
$$\therefore P_2 = 4431\,\text{kPa}$$
$$\frac{P_2 v_2}{T_2} = \frac{P_3 v_3}{T_3} \rightarrow v_3 = \frac{T_3}{T_2} v_2 = \frac{2591.33}{924.66} \times 0.06 = 0.168\,\text{m}^3/\text{kg}$$
$$r_c = \frac{v_3}{v_2} = \frac{0.168}{0.06} = 2.8$$

- The cut-off ratio is 2.8.

$$T_4 = T_3\left(\frac{v_3}{v_4}\right)^{\gamma-1} = 2591.33 \times \left(\frac{0.168}{0.898}\right)^{0.4}$$
$$= 1325.37\,\text{K}$$
$$\text{Q}_{\text{out}} = c_v(T_4 - T_1) = 0.718(1325.4 - 313) = 726.88\,\text{kJ/kg}$$
$$\text{Net work done, } W_{net} = Q_{in} - Q_{out} = 1675 - 726.88$$
$$= 948.12 \text{ kJ/kg}$$

- Therefore, thermal efficiency,

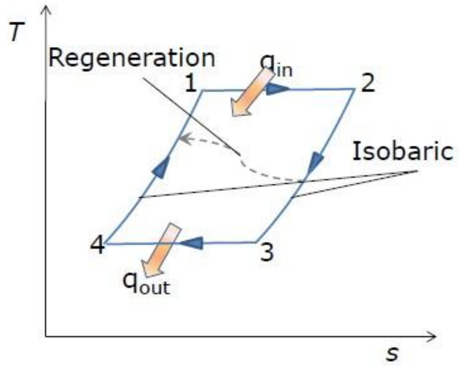
$$\eta_{th} = W_{net}/Q_{in}$$
$$= 948.12/1675 = 0.566 \text{ or } 56.6\%$$

- The cycle efficiency can also be calculated using

Problem 3

An air-standard Ericsson cycle has an ideal regenerator. Heat is supplied at 1000°Cand heat is rejected at 20°C. If the heat added is 600 kJ/kg,find the compressor work, the turbine work, and the cycle efficiency.

Solution: Problem 3



$T_1=T_2=1000°C=1273.15$ K
$T_3=T_4=20°C=293.15$ K

Since the regenerator is given as ideal, *-Q2-3= Q1-4*

Also in an Ericsson cycle, the heat is input during the isothermal expansion process, which is the turbine part of the cycle. Hence the turbine work is 600 kJ/kg.

- Thermal efficiency of an Ericsson cycle is equal to the Carnot efficiency.

$$\eta_{th}=\eta_{th,\ Carnot}=1-T_L/T_H$$
$$=1-293.15/1273.15$$
$$=0.7697$$

- Therefore the net work output is equal to:

$$w_{net}=\eta_{th}Q_H$$
$$=0.7697\times600=461.82 \text{ kJ/kg}$$

- The compressor work is equal to the difference between the turbine work and the net work output:

$$w_c=w_t-w_{net}$$
$$=600-461.82=138.2 \text{ kJ/kg}$$

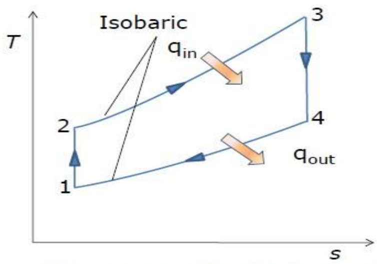
- In the Ericsson cycle the heat is rejected isothermally during the compression process. Therefore this compressor work is also equal to the heat rejected during the cycle.

Problem 4

In a Braytoncycle based power plant, the air at the inlet is at 27oC, 0.1 MPa.
The pressure ratio is 6.25and the maximum temperature is 800oC.

Find (a) the compressor work per kg of air (b) the turbine work per kg or air (c) the heat supplied per kg of air, and (d) the cycle efficiency.

Solution: Problem 4



T_1 = 27°C = 300 K
P_1 = 100 kPa
r_p = 6.25
T_3 = 800°C = 1073 K

- Since process, 1-2 is isentropic,

$$\frac{T_2}{T_1} = r_p^{(\gamma-1)/\gamma} = 6.25^{(1.4-1)/1.4} = 1.689$$
$$T_2 = 506.69\ \text{K}$$
$$W_{comp} = c_p(T_2 - T_1) = 1.005(506.69 - 300)$$
$$= 207.72\ \text{kJ/kg}$$

- The compressor work per unit kg of air is 207.72 kJ/kg

- Process 3-4 is also isentropic,

$$\frac{T_3}{T_4} = r_p^{(\gamma-1)/\gamma} = 6.25^{(1.4-1)/1.4} = 1.689$$
$$T_4 = 635.29\ \text{K}$$
$$W_{turb} = c_p(T_3 - T_4) = 1.005(1073 - 635.29)$$
$$= 439.89\ \text{kJ/kg}$$

- The turbine work per unit kg of air is 439.89 kJ/kg
- Heat input, Q_{in},

$$Q_{in} = c_p(T_3 - T_2) = 1.005(1073 - 506.69)$$
$$= 569.14\ \text{kJ/kg}$$

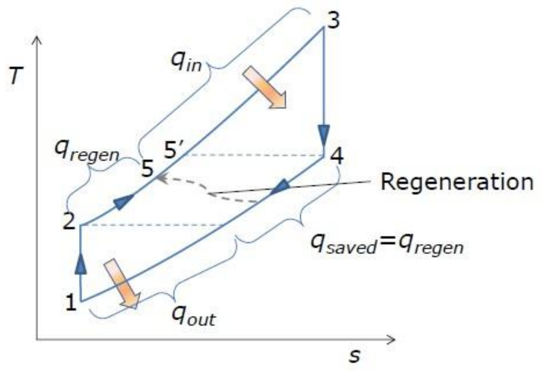
- Heat input per kg of air is 569.14 kJ/kg
- Cycle efficiency,

η_{th}=(W_{turb}-W_{comp})/Q_{in}
=(439.89-207.72)/569.14
=0.408 or 40.8%

Problem 5

- Solve Problem 3 if a regenerator of 75% effectiveness is added to the plant.

Solution: Problem 5



$$\varepsilon = \frac{T_5 - T_2}{T_4 - T_2} = 0.75$$

$$or, \frac{T_5 - 506.69}{635.29 - 506.69} = 0.75$$

$$T_5 = 603.14\,K$$

- T_4, W_{comp}, W_{turb} remain unchanged
- The new heat input, $Q_{in}=c_p(T_3-T_5)$
  $=472.2$ *kJ/kg*
- *Therefore* $\eta_{th}=(W_{turb}-W_{comp})/Q_{in}$
  $=(439.89-207.72)/472.2$
  $=0.492$ *or* $49.2\ \%$

INDUSTRIAL APPLICATIONS

Industrial Applications

The following are the applications of a gas turbine:

1. They are used to propel air-crafts and ships,
2. Gas turbine plants are used as standby plants for the hydroelectric power plants.
3. Gas turbine power plants may be used as peak loads plant and standby plants for smaller power units.
4. The shaft can be connected to other machinery to do various types of work such as: turning a helicopter rotor, running a compressor (which "crushes" a gas to a condensed form for use in industrial applications) or generating electric power.
5. The gas turbine is useful to our modern world because it is relatively compact in size and makes a lot of power. Gas turbines are used in backup power systems in Manhattan for example, when the grid goes down due to natural disaster, gas turbines power up and can produce power for emergency uses.
6. Gas turbines are used on oil platforms to make power. The oil platform is like a small city, isolated out on the water, so it requires a lot of power and does not have a lot of space. Gas turbines are also used in oil refineries to make power for the cracking process.

TUTORIAL QUESTIONS

UNIT – IV

Theory Questions:

1. Explain about the open cycle and closed cycle turbines with neat sketches and also draw P-V & T-S diagrams.
2. State the merits of gas turbines over IC engines.
3. Draw the gas turbine power plant with inter cooling
4. List out the advantages of open cycle gas turbine over closed cycle gas turbine.
5. List different applications of gas turbine power cycles in power sector industries.
6. Discuss the relative advantages and disadvantages of gas turbines and steam turbines.
7. What are the different methods to improve the efficiency of gas turbines?
8. What are the different types of combustion chambers in gas turbine engines? Explain them in detail with relevant sketches.
9. Draw the schematic diagram of closed cycle gas turbine and explain its working.
10. Explain the operating principle of Brayton cycle with a schematic diagram p-v and T-s diagrams.
11. Why Re-heater is necessary in gas turbine? What are its effects
12. What are the requirements of a good combustion chamber for a gas turbine?
13. Explain with neat sketch the gas turbine cycles with intercooling and reheating and what will be the condition of maximum output

Numerical Problems:

1. A simple gas turbine cycle works with a pressure ratio of 8. The compressor and turbine inlet temperatures are 300 K and 800 K respectively. If the volume flow rate of air is 250 m3/s, compute the power output and thermal efficiency
2. A constant pressure open cycle gas turbine plant works between temperature range of 15ºC and 700ºC and pressure ratio of 6. Find the mass of air circulating in the installation, if it develops 1100 kW. Also find the heat supplied by the heating chamber.

ASSIGNMENT QUESTIONS

3. In a gas turbine plant, air is drawn at 1 bar, 150 C and the pressure ratio is 6. The expansion takes place in two turbines. The efficiency of compressor is 0.82, high pressure turbine is

   0.85 and low pressure turbine is 0.84. The maximum cycle temperature is 6250 C. Calculate

   i) Pressure and temperature of gases entering the low pressure turbine.

   ii) Net power developed

   iii) Work ratio

   iv) Thermal efficiency. Work output of high pressure turbine is equal to compressor work
4. In an air standard regenerative gas turbine cycle the pressure ratio is 5. Air enters the compressor at 1 bar, 300 K and leaves at 490 K. The maximum temperature in the cycle is 1000 K. Calculate the cycle efficiency, given that the efficiency of regenerator and the adiabatic efficiency of the turbine are each 80%. Assume for air, the ratio of specific heats is 1.4. Also show the cycle on T-S diagram.
5. A gas turbine unit receives air at 1 bar and 300 K and compresses it adiabatically to 6.2 bar. The compressor efficiency is 88%. The fuel has a heating value of 44186 KJ/kg and

   the fuel air ratio is 0.017 KJ/kg of air. The turbine efficiency is 90 %. Calculate the work of turbine and compressor per kg of air compressed and thermal efficiency. Take Cp=1.005 KJ/kg K, γ=1.4 for the compression process, Cp=1.147 KJ/kg K, γ=1.33 for the expansion process.

ASSIGNMENT QUESTIONS

1. a) Describe with neat sketches the working of a simple constant pressure open cycle gas turbine.

   b) Discuss the relative advantages and disadvantages of gas turbines and steam turbines.
2. Describe with neat diagram a closed cycle gas turbine and explain advantages, disadvantages and applications.
3. Explain with neat sketch the gas turbine cycles with intercooling and reheating and what will be the condition of maximum output.
4. Explain about the open cycle and closed cycle turbines with neat sketches and also draw P-V & T-S diagrams.

UNIT 5

JET PROPULSION & ROCKETS

Course Objective:

Applications and the principles of thermodynamics to components and systems.

Course Outcome:

Develop problem solving skills through the application of thermodynamics.

5
JET PROPULSION ENGINES

5.1 Introduction

Jet propulsion, similar to all means of propulsion, is based on Newton's Second and Third laws of motion.

The jet propulsion engine is used for the propulsion of aircraft, missile and submarine (for vehicles operating entirely in a fluid) by the reaction of jet of gases which are discharged rearward (behind) with a high velocity. As applied to vehicles operating entirely in a fluid, a momentum is imparted to a mass of fluid in such a manner that the reaction of the imparted momentum furnishes a propulsive force. The magnitude of this propulsive force is termed as thrust.

For efficient production of large power, fuel is burnt in an atmosphere of compressed air (combustion chamber), the products of combustion expanding first in a gas turbine which drives the air compressor and then in a nozzle from which the thrust is derived. Paraffin is usually adopted as the fuel because of its ease of atomisation and its low freezing point.

Jet propulsion was utilized in the flying Bomb, the initial compression of the air being due to a divergent inlet duct in which a small increase in pressure energy was obtained at the expense of kinetic energy of the air. Because of this very limited compression, the thermal efficiency of the unit was low, although huge power was obtained. In the normal type of jet propulsion unit a considerable improvement in efficiency is obtained by fitting a turbo–compressor which will give a compression ratio of at least 4 : 1.

5.2 Classification

Jet propulsion engines are classified basically as to their method of operation as shown in fig. 5–1. The two main catagories of jet propulsion systems are the *atmospheric*

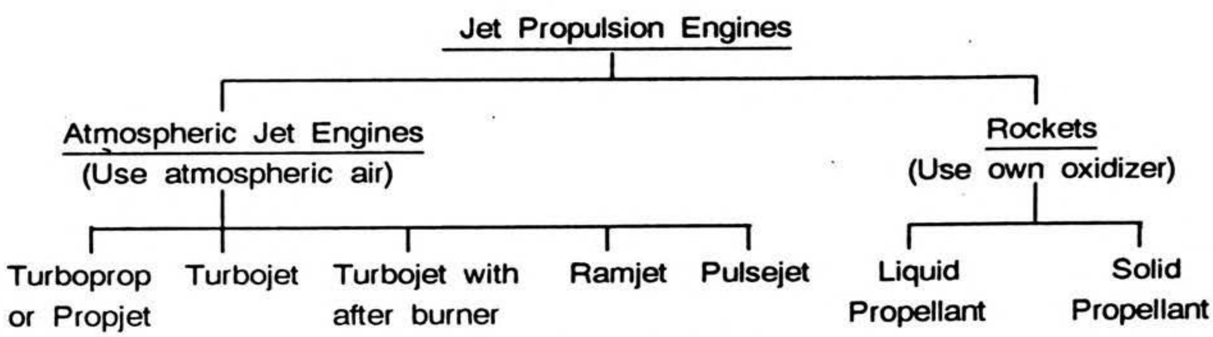


Fig. 5 – 1. Jet propulsion engines.

jet engine and *rocket*. Atmospheric jet engines require oxygen from the atmosperic air for combustion of fuel, *i.e.* they are dependent on atmospheric air for combustion.

The rocket engine carries its own oxidizer for combustion of fuel and is, therefore, independent of the atmospheric air. Rocket engines are discussed in art. 5·6.

The turboprop, turbojet and turbojet with after burner are modified simple open cycle gas turbine engines. In turboprop thrust is not completely due to jet. Approximately 80 to 90 percent of the thrust in turboprop is produced by acceleration of the air outside the engine by the propeller (as in conventional aeroengines) and about 10 to 20 percent of the thrust is produced by the jet of the exhaust gases. In turbojet engine, the thrust is completely due to jet of exhaust gases. The turbojet with after burner is a turbojet engine with a reheater added to the engine so that the extended tail pipe acts as a combustion chamber.

The ramjet and pulsejet are aero–thermo–dynamic–ducts, i.e. a straight duct type of jet engine without compressor and turbine. The ramjet has the simplest construction of any propulsion engine, consisting essentially of an inlet diffuser, a combustion chamber and an exit nozzle of tail pipe. Since the ramjet has no compressor, it is dependent entirely upon ram compression.

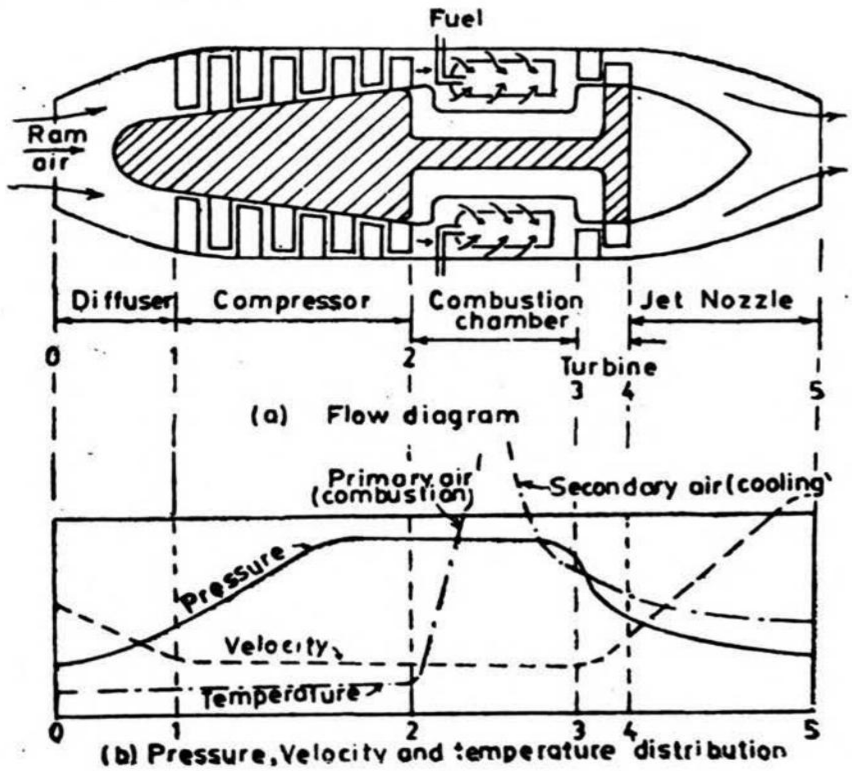


Fig. 5-2 Turbojet engine.

The pulsejet is an intermittent combustion jet engine and it operates on a cycle similar to a reciprocating engine and may be better compared with an ideal Otto cycle rather than the Joule or Bryton cycle. From construction point of view, it is some what similar to a ramjet engine. The difference lies in provision of a mechanical valve arrangement to prevent the hot gases of combustion from going out through the diffuser.

5.3 Turbojet Engine

The turboject engine (fig..5–2) is similar to the simple open cycle constant pressure gas turbine plant (fig. 4–2) except that the exhaust gases are first partially expanded in the turbine to produce just sufficient power to drive the compressor. The exhaust gases leaving the turbine are then expanded to atmospheric pressure in a propelling (discharge) nozzle. The remaining energy of gases after leaving the turbine is used as a high speed jet from which the thrust is obtained for forward movement of the aircraft.

Thus, the essential *components* of a turbojet engine are :

- An entrance air diffuser (diverging duct) in front of the compressor, which causes rise in pressure in the entering air by slowing it down. This is known as *ram*. The pressure at entrance to the compressor is about 1·25 times the ambient pressure.
- A rotary compressor, which raises the pressure of air further to required value and delivers to the combustion chamber. The compressor is the radial or axial type and is driven by the turbine.
- The combustion chamber, in which paraffin (kerosene) is sprayed, as a result of this combustion takes place at constant pressure and the temperature of air is raised.
- The gas turbine into which products of combustion pass on leaving the combustion chamber. The products of combustion are partially expanded in the turbine to provide necessary power to drive the compressor.
- The discharge nozzle in which expansion of gases is completed, thus developing the forward thrust.

A Rolls–Royce Derwent jet engine employs a centrifugal compressor and turbine of the impulse–reaction type. The unit has 550 kg mass. The speed attained is 960 km/hour.

5.3.1 Working Cycle : Air from surrounding atmosphere is drawn in through the diffuser, in which air is compressed partially by ram effect. Then air enters the rotary compressor and major part of the pressure rise is accomplished here. The air is compressed to a pressure of about 4 atmospheres. From the compressor the air passes into the annular combustion chamber. The fuel is forced by the oil pump through the fuel nozzle into the combustion chamber. Here the fuel is burnt at constant pressure. This raises the temperature and volume of the mixture of air and products of combustion. The mass of air supplied is about 60 times the mass of the fuel burnt. This excess air produces sufficient mass for the propulsion jet, and at the same time prevents gas temperature from reaching values which are too high for the metal of the rotor blades.

The hot gases from the combustion chamber then pass through the turbine nozzle ring. The hot gases which partially expand in the turbine are then exhausted through the discharge (propelling nozzle) by which the remaining enthalpy is converted into kinetic energy. Thus, a high velocity propulsion jet is produced.

The oil pump ad compressor are mounted on the same shaft as the turbine rotor. The power developed by the turbine is spent in driving the compressor and the oil pump.

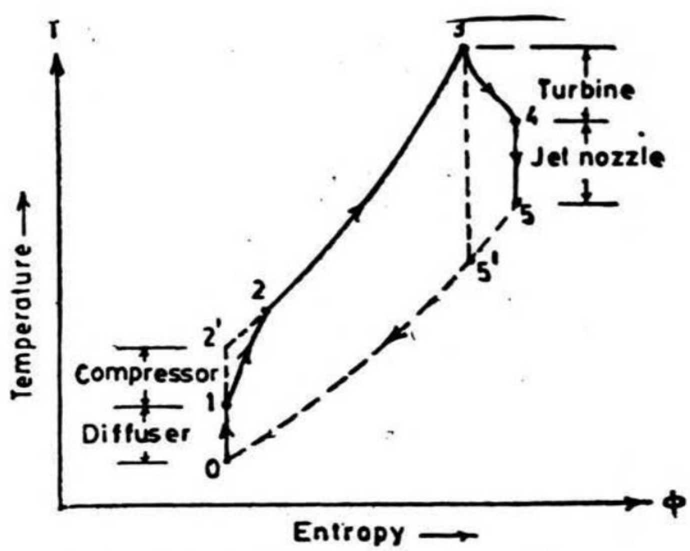


Fig. 5-3. A typical turbojet engine cycle on T – φ diagram.

Some starting devicé such as compressed air motor or electric motor, must be provided in the turbojet plant. Flight speeds upto 800 km per hour are obtained from this type of unit.

The basic thermodynamic cycle for the turbojet engine is the Joule or Brayton cycle as shown in $T - \Phi$ diagram of fig. 5–3. While drawing this cycle, following simplifying assumptions are made :

- There are no pressure losses in combustion chamber.
- Specific heat of working medium is constant.
- Diffuser has ram efficiency of 100 percent *i.e.*, the entering atmospheric air is diffused isentropically from velocity V_o to zero (V_o is the vehicle velocity through the air).
- Hot gases leaving the turbine are expanded isentropically in the exit nozzle *i.e.*, the efficiency of the exit nozzle is 100 percent.

5.3.2 Thrust Power and Propulsive Efficiency : The jet aircraft draws in air and expels it to the rear at a markedly increased velocity. The action of accelerating the mass of fluid in a given direction creates a reaction in the opposite direction in the form of a propulsive force. The magnitude of this propulsive force is defined as thrust. It is dependent upon the rate of change of momentum of the working medium i.e. air, as it passes through the engine.

The basis for comparison of jet engines is the thrust. The thrust, T of a turbojet engine can be expressed as,

$$T = m(V_i - V_o) \quad \ldots (5.1)$$

where, m = mass flow rate of gases, kg/sec.,

V_j = exit jet velocity, m/sec., and,

V_o = vehicle velocity, m/sec.

The above equation is based upon the assumption that the mass of fuel is neglected. Since the atmospheric air is assumed to be at rest, the velocity of the air entering relative to the engine, is the velocity of the vehicle, V_O. The thrust can be increased by increasing the mass flow rate of gas or increasing the velocity of the exhaust jet for given V_o.

Thrust power is the time rate of development of the useful work achieved by the engine and it is obtained by the product of the thrust and the flight velocity of the vehicle. Thus, thrust power TP is given by

$$TP = T\,V_o = m(V_j - V_o)\,V_o \,\frac{\text{N·m}}{\text{sec.}} \quad \ldots (5.2)$$

The kinetic energy imparted to the fluid or the energy required to change the momentum of the mass flow of air, is the difference between the rate of kinetic energy of entering air and the rate of kinetic energy of the exist gases and is called propulsive power. The propulsive power PP is given by

$$PP = \frac{m(V_j^2 - V_0^2)}{2} \text{ N.m/sec.} \quad \ldots (5.3)$$

Propulsive efficiency is defined as the ratio of thrust power (TP) and propulsive power (PP) and is the measure of the effectiveness with which the kinetic energy imparted to the fluid is transformed or converted into useful work. Thus, propulsive efficiency η_p is given by

$$\eta_p = \frac{TP}{PP} = \frac{m(V_j - V_0)\,V_0}{1} \times \frac{2}{m(V_j^2 - V_0^2)}$$

$$\therefore \eta_p = \frac{2(V_j - V_o)V_o}{V_j^2 - V_o^2} = \frac{2V_o}{V_j + V_o} = \frac{2}{1 + \left(\frac{V_j}{V_o}\right)} \qquad \ldots (5.4)$$

From the expression of n_p it may be seen that the propulsion system approaches maximum efficiency as the velocity of the vehicle approaches the velocity of the exhaust gases. But as this occurs, the thrust and the thrust power approach zero. Thus, the ratio of velocities for maximum propulsive efficiency and for maximum power are not the same. Alternatively, the propulsive efficiency can be expressed as

$$\eta_p = \frac{TP}{PP} = \frac{TP}{TP + K.E.\text{ losses}} \qquad \ldots (5.5)$$

Thermal efficiency of a propulsion is an indication of the degree of utilization of energy in fuel (heat supplied) in accelerating the fluid flow and is defined as the increase in the kinetic energy of the fluid (propulsive power) and the heat supplied. Thus,

$$\text{Thermal efficiency, } \eta_T = \frac{\text{Propulsive power}}{\text{Heat supplied}}$$

$$= \frac{\text{Propulsive power}}{\text{Fuel flow rate} \times \text{C.V. of fuel}} \qquad \ldots (5.6)$$

The overall efficiency is the ratio of the thrust power and the heat supplied. Thus, overall efficiency is the product of propulsive efficiency and thermal efficiency. The propulsive and overall efficiencies of the turboject engine are comparable to the mechanical efficiency and brake thermal efficiency respectively, of the reciprocating engine.

Problem – 1 : *A jet propulsion unit, with turbojet engine, having a forward speed of 1,100 km/hr produces 14 kN of thrust and uses 40 kg of air per second. Find: (a) the relative exist jet velocity, (b) the thrust power, (c) the propulsive power, and (d) the propulsive efficiency.*

(a) Forward speed, $V_o = \frac{1{,}100 \times 1{,}000}{3{,}600} = 305{\cdot}55$ m/sec.

Using eqn. (5.1), thrust, $T = m(V_j - V_o)$

i.e., $14{,}000 = 40(V_j - 305{\cdot}55)$

$$\therefore V_j = \frac{14{,}000}{40} + 305{\cdot}55 = 350 + 305{\cdot}55 = 655{\cdot}55 \text{ m/sec.}$$

Thus relative exist jet velocity, V_j = **655·55 m/sec.**

(b) Using eqn. (5.2)

Thrust power, $TP = T \times V_o$

$= 14{,}000 \times 305{\cdot}55 = 42{,}77{,}700$ N.m/sec. or = **4,277.7 kN.m/sec.**

(c) Using eqn. (5.3),

$$\text{Propulsive power, } PP = \frac{m(V_2^2 - V_0^2)}{2}$$

$$= \frac{40\{(655{\cdot}55)^2 - (305{\cdot}55)^2\}}{2}$$

$= 6{,}727 \times 10^3$ N.m/sec = 6,727 KN.m/sec or **6,727 kW**

(d) Using eqn. (5.4),

is not effective and that there are pulsations created in the combustion chamber which affect the air flow in front of the diffuser.

Since the ram jet engine has no turbine, the temperature of the gases of combustion is not limited to a relatively low figure as in the turbojet engine. Air fuel ratios of around 15·1 are used. This produces exhaust temperatures in the range of 2000°C to 2200°C. Extensive research is being conducted on the development of hydrocarbon fuels that will give 30 percent more energy per unit volume than current aviation gasolines. Investigations are carried out to determine the possibility of using solid fuels in the ram jet and in the after burner of the turbojet engine. If powdered aluminium could be utilized as an aircraft fuel, it would deliver over 2·5 times as much heat per unit volume as aviation gasoline, while some other could deliver almost four times as much heat.

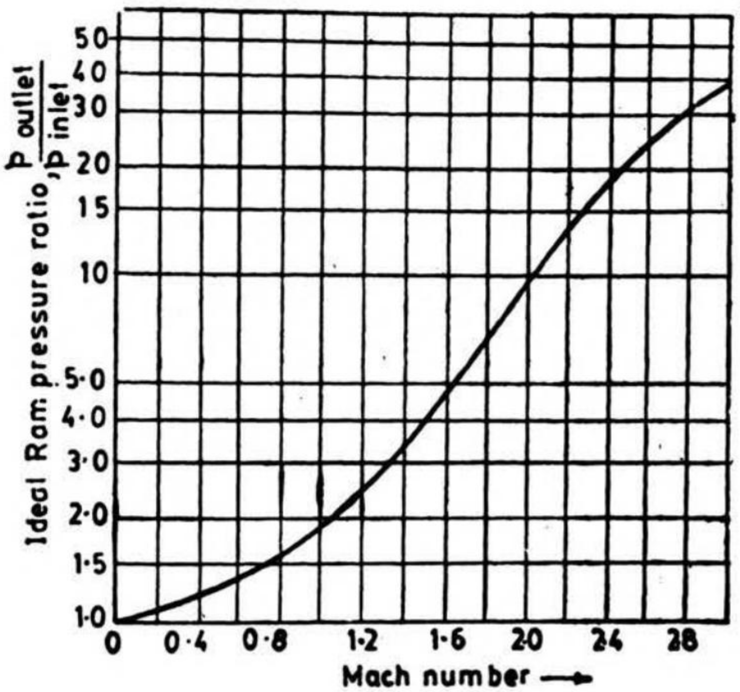


Fig. 5–5. Ram pressure ratio versus Mach number of vehicle for sea level condition.

The temperature, pressure and velocity of the air during its passage through a ram jet engine at supersonic flight are shown in fig. 5–4.

The cycle for an ideal ram jet, which has an isentropic entrance diffuser and exit nozzle, is the Joule cycle as shown by the dotted lines in fig. 5–6. The difference between the actual and ideal jet is due principally to losses actually encountered in the flow system. The sources of these losses are :

- Wall friction and flow separation in the subsonic diffuser and shock in the supersonic diffuser.
- Obstruction of the air stream by the burners which introduces eddy currents and turbulence in the air stream.
- Turbulence and eddy currents introduced in the flow during burning.
- Wall friction in the exit nozzle.

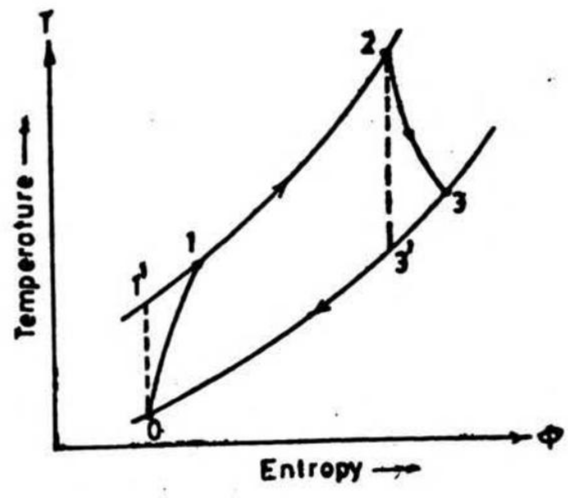


Fig. 5–6. T – ϕ diagram of Ram jet engine.

By far, the most critical component of the ram jet is the diffuser. Due to the peculiarities of steamline flow, a diffuser which is extremely engine weight than any other propulsion engine at supersonic speed with the exception of the rocket engine. The thrust per unit frontal area increases both with the efficiency and the air flow through the engine; therefore much greater thrust per unit area is obtainable at high supersonic speeds. General performance of a ram jet engine in the subsonic range would have a specific fuel consumption between 0.6 to 0·8 kg fuel per N thrust – hr and a specific weight between 0·01 to 0·02 kg per N thrust. The supersonic ram jet engine has a specific fuel consumption between 0·25 to 0·04 and a specific weight between 0·01 to 0·04. Thus, the best performance of the ram jet engine is obtained at flights speeds of 1500 to 3500 km/hr.

5.5 Pulse Jet Engine

The pulse jet engine is somewhat similar to a ram jet engine. The difference is that a mechanical valve arrangement is used to prevent the hot gases of combustion from flowing out through the diffuser in the pulse jet engine.

Paul Schmidt patented principles of the pulse jet engine in 1930. It was developed by Germany during World–War–II, and was used as the power plant for "buzz bomb".

The turbojet and ram jet engines are continuous in operation and are based on the constant pressure heat addition (Bryton) cycle. The pulse jet is an intermittent combusion engine and it operates on a cycle similar to a reciprocating engine and may be better compared with an ideal Otto cycle rather than the Joule or Bryton cycle.

The compression of incoming air is accomplished in a diffuser. The air passes through the spring valves and is mixed with fuel from a fuel spray located behind the valves. A spark plug is used to initiate combustion but once the engine is operating normally, the spark is turned off and residual flame in the combustion chamber is used for ignition. The engine walls also may get hot enough to initiate combustion.

The mechanical valves which were forced open by the entering air, are forced shut when the combustion process raises the pressure within the engine above the pressure in the diffuser. As the combustion products cannot expand forward, they move to the rear at high velocity. The combustion products cannot expand forward, they move to the rear at high velocity. When the combustion products leave, the pressure in the combustion chamber drops and the high pressure air in the diffuser forces the valves open and fresh air enters the engine.

Since the products of combustion leave at a high velocity there is certain scavenging of the engine caused by the decrease in pressure occasioned by the exit gases. There is a stable cycle set up in which alternate waves of high and low pressure travel down the engine. The alternating cycles of combustion, exhaust, induction, combustion, etc. are related to the acoustical velocity at the temperature prevailing in the engine. Since the temperature varies continually, the actual process is complicated, but a workable assumption is that the tube is acting similar to a quarter wave length organ pipe. The series of pressure and rarefaction waves move down it at the speed of sound for an assumed average temperatures.

The frequency of the combustion cycle may be calculated from the following expression:

$$f = \frac{a}{4L} \text{ cycles/sec.} \qquad \ldots (5.7)$$

where, $a = \sqrt{\gamma RT}$ = sound velocity in the medium at temperature, T, and

L = length of engine (from valves to exit).

efficient at a given speed may be quite inadequate at another velocity.

Because of the simplicity of the engine, the ram jet develops greater thrust per unit engine weight than any other propulsion engine at supersonic speed with the exception of the rocket engine. The thrust per unit frontal area increases both with the efficiency and the air flow through the engine; therefore much greater thrust per unit area is obtainable at high supersonic speeds. General performance of a ram jet engine in the subsonic range would have a specific fuel consumption between 0.6 to 0*8 kg fuel per N thrust - hr and a specific weight between 0*01 to 0*02 kg per N thrust. The supersonic ram jet engine has a specific fuel consumption between 0*25 to 0-04 and a specific weight between 0-01 to 0-04. Thus, the best performance of the ram jet engine is obtained at flights speeds of 1500 to 3500 km/hr.

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The frequency of the combustion cycle may be calculated from the following expression:

^ = 4 l cvc*es/sec-

where, a = VfTTT = sound velocity in the medium at temperature, *T*, and

L = length of engine (from valves to exit).

A serious limitation placed upon pulse jet engine is the mechanical valve arrangement. Unfortunately, the valves used have resonant frequencies of their own, and under certain conditions, the valve will be forced into resonant vibration and will be operating when they should be shutting. This limitation of valves also limits the engine because the gas goes out of the diffuser when it should go out of the tail pipe.

Despite the apparent noise and the valve limitation, pulse jet engines have several *advantages* when compared to other thermal jet engines.

. . The pluse jet is very inexpensive when compared to a turbojet.

. . The pulse jet produces static thrust and produces thrust in excess of drag at much lower speed than a ram jet.

. . The potential of the pulse jet is quite considerable and its development and research may well bring about a wide range of application.

5.6 Rocket Motors

The jet propulsion action of the rocket has been recognised for long. Since the early beginning, the use of rockets has been in war time as a weapon and in peace time as a signaling or pyrotechnic displays. Although, the rocket was employed only to an insignificant extent in World War–I, marked advances were made by the research that was undertaken at that time. In World War–II, the rocket became a major offensive weapon employed by all warring powers. Rockets and rocket powered weapons have advanced to a point where they are used effectively in military operations.

Rocket type engine differs from the atmospheric jet engine in that the entire mass of the jet is generated from the propellant carried within the engine i.e. the rocket motor carries both the fuel and the oxidizing agent. As a result, this type of engine is independent of the atmospheric air that other thermal jet engines must rely upon. From this point of view rocket motors are most attractive. There are, however, other operational features that make rocket less useful. Here, the fundamentals of rocket motor theory and its applications are discussed.

Rocket engines are classified as to the type of propellant used in them. Accordingly, there are two major groups:

One type belonging to the group that utilizes liquid type propellants and other group that uses solid type propellants.

The basic theory governing the operation of rocket motor is applied, equally to both the liquid and the solid propellant rocket.

Rocket propulsion, at this time, would not be regarded as a competitor of existing means for propelling airplanes, but as a source of power for reaching objectives unattainable by other methods. The rocket motors are under active development programmes for an increasing number of applications. Some of these *applications* are :

– Artillery barrage rockets,

– Anti–tank rockets,

– All types of guided missiles,

– Aircraft launched rockets,

– Jets assisted take–off for airplanes,

– Engines for long range, high speed guided missiles and pilotless aircrafts, and

– Main and auxiliary propulsion engines on transonic airplanes.

It will be repeated again that the rocket engine differs from the other jet propulsion engines in that the entire mass of the gases in the jet is generated from the propellants carried within the engine. Therefore, it is not dependent on the atmospheric air to furnish the oxygen for combustion. However, since the rocket carries its own oxidiser, the propellant consumption is very high.

The particular advantages of the rocket are :

- Its thrust is practically independent of its environments.
- It requires no atmospheric oxygen for its operation.
- It can function even in a vacuum.
- It appear to be the simplest means for converting the themochemical energy of a propellant combination (fuel plus oxidizer) into kinetic energy associated with a jet flow gases.

Despite its apparent simplicity, the development of a reliable rocket system must be light in weight and the rocket motor must be capable of sustained operation in contact with gases at temperature above 2800° C and at appreciable pressures.The problem of materials in consequently a major one. Furthermore, owing to the enormous energy releases involved, problem of ignition, smooth start up, thrust control, cooling etc. arise.

A major problem of development of rocket is selection of suitable propellant to give maximum energy per premium total weight (propellant plus containing vessels) and convenience factors such as a safety in handling, dependability, corrosive tendencies, cost, availability and storage problems. In general, it can be stated that there is a wide variety of fuels that are satisfactory for rocket purpose, but choice of oxidizers is at present distinctly limited.

5.6.1 Basic Theory : Figure 5–7 shows a schematic diagram of a liquid bi–propellant rocket engine. It consists of an injection system, a combustion chamber, and an exit nozzle. The oxidizer and fuel burnt, in the combustion chamber produces a high pressure. The pressure produced is governed by

– Mass rate of flow of the propellants,

– Chemicals characteristics of the propellants, and

– Cross–section area of the nozzle throat.

The gases are ejected to the atmosphere at supersonic speeds through the nozzle. The enthalpy of high pressure gases is converted into kinetic energy. The reaction to the ejection of the high velocity, produces the thrust on the rocket engine.

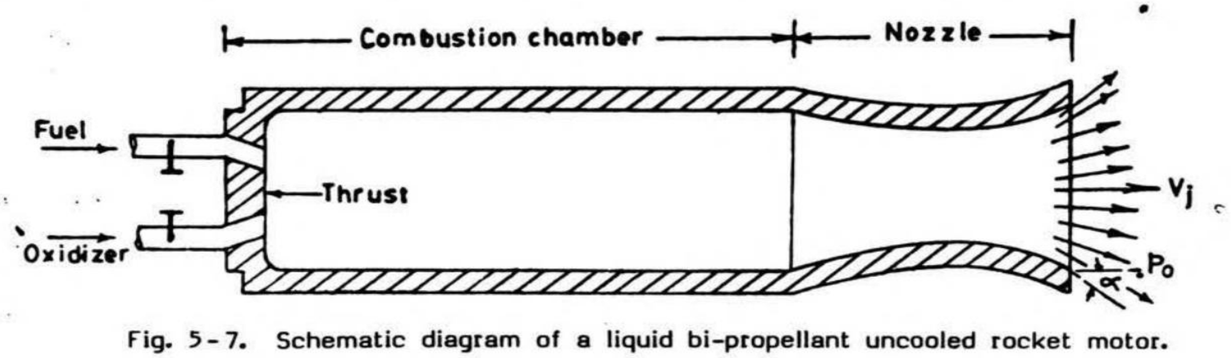

Fig. 5-7. Schematic diagram of a liquid bi-propellant uncooled rocket motor.

The *thrust* developed is a resultant of the pressure forces acting upon the inner and the outer surface of the rocket engine. The resultant internal force acting on the engine is given by

Resultant force = $m_p \ V_j + p_j \ A_j \ N$

where, m_p = Mass rate of propellant consumption, kg/sec,

V_j = Jet velocity relative to nozzle, m/sec,

V_{xj} = Average value of the x–component of the velocity of gases crossing, A_j,

p_j = Exist static pressure, N/m^2, and

A_j = Exit area of nozzle, m^2.

The resultant external forces acting on the rocket engine are $p_o A_o$, where p_o is the atmospheric pressure in N/m^2. The thrust which is a resultant of the total pressure forces becomes

$$T = m_p V_{xj} + A_j (p_j - p_o)\, N \quad \ldots (5.8)$$

Let V_j = the exit velocity of the rocket gases, assumed constant and

let $V_{xj} = \lambda V_j$. Then, eqn. (5.8) becomes

$$T = \lambda m_p V_j + A_j (p_j - p_o)\, N \quad \ldots (5.9)$$

The coefficient λ is the correction factor for the divergence angle a of the exit conical section of the nozzle. λ is given by

$$\lambda = \frac{1 - \cos 2\alpha}{4(1 - \cos \alpha)} = \frac{1}{2}(1 + \cos \alpha) \quad \ldots (5.10)$$

Equation (5.8) shows that thrust of a rocket engine increases as the atmospheric pressure decreases. Therefore, maximum thrust will be obtained when $P_o = 0$, *i.e., rocket engine produces maximum thrust when operating in a vacuum.*

In testing a rocket engine, thrust and propellant consumption for a given time are readily measured. It is convenient then, to express the thrust in terms of the mass rate of flow of propellant and an effective jet velocity, V_{ej}

i.e., Thrust, $T = m_p \times V_{ej}$ $\quad \ldots (5.11)$

The *effective jet exit velocity* is a hypothetical velocity and for convenience in test work it is defined from eqns. (5.9) and (5.11) as under :

$$V_{ej} = \lambda V_j + \frac{A_j}{m_p}(p_j - p_o) \text{ m/sec.} \quad \ldots (5.12)$$

The effective jet exit velocity has become an important parameter in rocket motor performance.

The *thrust power, TP* developed by a rocket motor is defined as the thrust multiplied by the flight velocity, V_o.

$$TP = T V_o = m_p \cdot V_{ej} \cdot V_o \text{ N.m/sec.} \quad \ldots (5.13)$$

The *propulsive efficiency,* η_p is the ratio of the thrust power to propulsive power supplied. The propulsive power is the thrust power plus the kinetic energy lost in the exhaust,

i.e., K.E. Loss $= \frac{1}{2} m_p (V_{ej} - V_o)^2$ N.m/sec.

Therefore, the propulsive efficiency may be expressed as

$$\eta_p = \frac{TP}{TP + \text{K.E. Loss}} = \frac{m_p V_{ej} V_o}{m_p V_{ej} V_o + \frac{1}{2} m_p (V_{ej} - V_o)^2}$$

$$\therefore\ \eta_p = \frac{2\,(V_o/V_{ej})}{1 + (V_o/V_{ej})^2} \qquad \ldots (5.14)$$

Specific Impulse, I_{sp} has become an important parameter in rocket motor performance and is defined as the thrust produced per unit mass rate of propellant consumption.

$$I_{sp} = \frac{T}{m_p} = \frac{m_p \cdot V_{ej}}{m_p} = V_{ej} \qquad \ldots (5.15)$$

Specific impulse, with the units, Newtons of thrust produced per kg of propellant burned per second, gives a direct comparison as to the effectiveness among propellants. It is desirable to use propellants with the greatest possible specific impulse, since, this allows a greater useful load to be carried for a given overall rocket weight.

5.6.2. Types of Rocket Motors : The propellant employed in a rocket motor may be a solid, two liquids (fuel plus oxidizer), or materials containing an adequate supply of available oxygen in their chemical composition (monopropellant). Solid propellants are used for rockets which are to operate for relatively short periods, upto possibly 45 seconds. Their main application is to projectiles, guided missiles, and the assisted take-off aircraft.

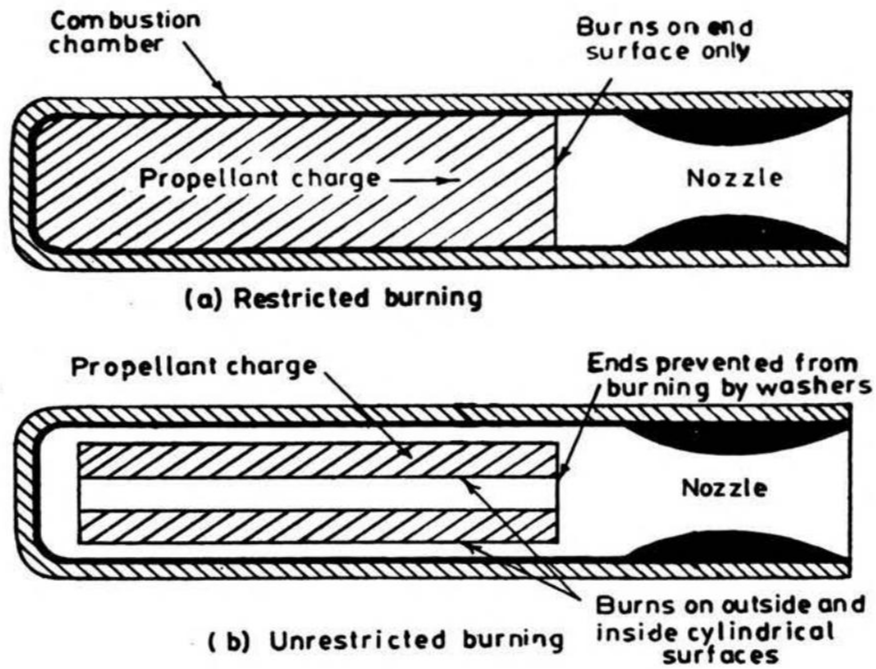

Fig. 5-8 Schematic diagram of a solid propellant rocket.

Solid propellant rockets (fig. 5–8) have been of two basic types :

. . Unrestricted burning types for projectiles and launching rockets; and

. . Restricted burning types for assisted take–off of aircraft and for propelling missiles.

In the unrestricted burning rocket [fig. 5–8(a)] all surfaces of the propellant grain except the ends are ignited; in restricted burning rockets [fig. 5–8(b)] only one surface of the propellant is permitted to burn. Liquid propellant rockets utilizes liquid propellants which are stored in the containers outside the combustion chamber. The basic theory of operation of this type of rocket is same as that for solid propellant rocket. Liquid propellant rockets were developed in order to overcome some of the undesirable features of the

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solid propellant rockets such as short duration of thrust, and no provisions for adequate cooling or control of the burning after combustion starts. Here, the propellant in the liquid

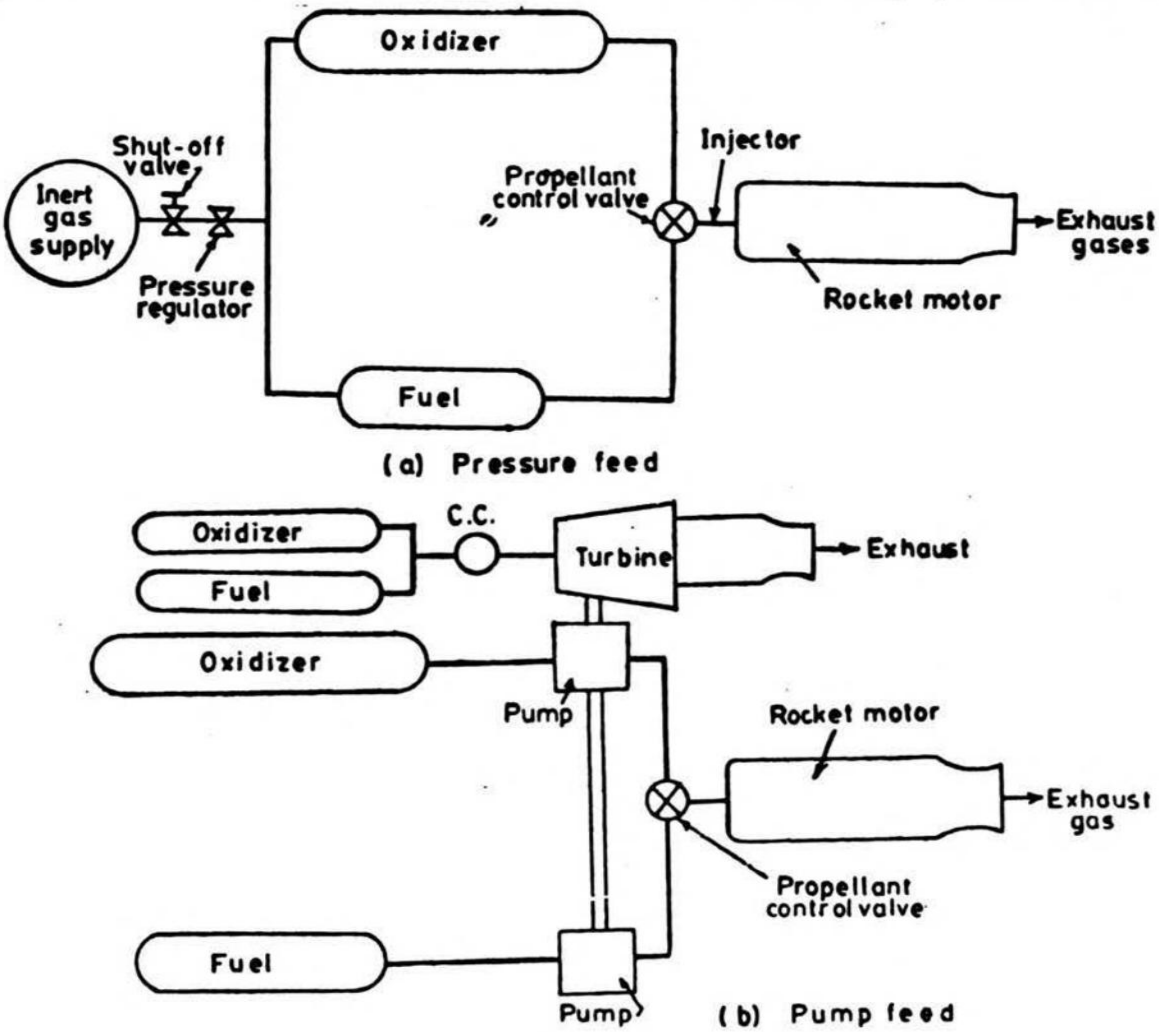

Fig. 5-9. Schematic diagrams of bi-propellant rocket system.

state is injected into a combustion chamber, burned and exhausted at a high velocity through the nozzle. The liquid propellant is also used to cool the rocket motor by circulation of fuels around the walls of the combustion chamber and around the nozzle. Certain liquid fuel, however, such as hydrogen peroxide, burn at such temperatures that no cooling is necessary. Figure 5–9 shows schematic diagrams of pressure feed and pump feed liquid bipropellant rocket systems.

Problem–2 : *The effective exit jet velocity of a rocket is 3000 m/sec, the forward flight velocity is 1500 m/sec and the propellant consumption is 70 kg per sec. Calculate : (a) Thrust, (b) Thrust power, (c) Specific impulse, (d) Specific propellant consumption, and (e) Propulsive efficiency of the rocket.*

(a) Using eqn. (5.11),

Thrust, $T = m_p \times V_{ej} = 70 \times 3{,}000 = 2{,}10{,}000$ N or **210 kN**

(b) Using eqn. (5.13),

Thrust power, $TP = T\ V_o = 2{,}10{,}000 \times 1{,}500 = \mathbf{315 \times 10^6}$ **N.m/s**

(c) Using eqn. (5.14),

Specific impluse, $I_{sp} = \dfrac{T}{m_p} = \dfrac{m_p \cdot V_{ej}}{m_p} = V_{ej} =$ **3,000 N.s/kg**

(d) Specific propellant consumption $= \dfrac{m_p}{T} = \dfrac{m_p}{m_p\, V_{ej}} = \dfrac{1}{V_{ej}} = \dfrac{1}{3{,}000}$

$$= \mathbf{3{\cdot}3 \times 10^{-4}\ kg/N.s}$$

(e) Using eqn. (5.14),

Propulsive efficiency, $\eta_p = \dfrac{2\,(V_o/V_{ej})}{1 + (V_o/V_{ej})^2}$

$$= \frac{2(1500/3000)}{1 + (1500/3000)^2} = 0{\cdot}8 \text{ i.e., } \mathbf{80\%}$$

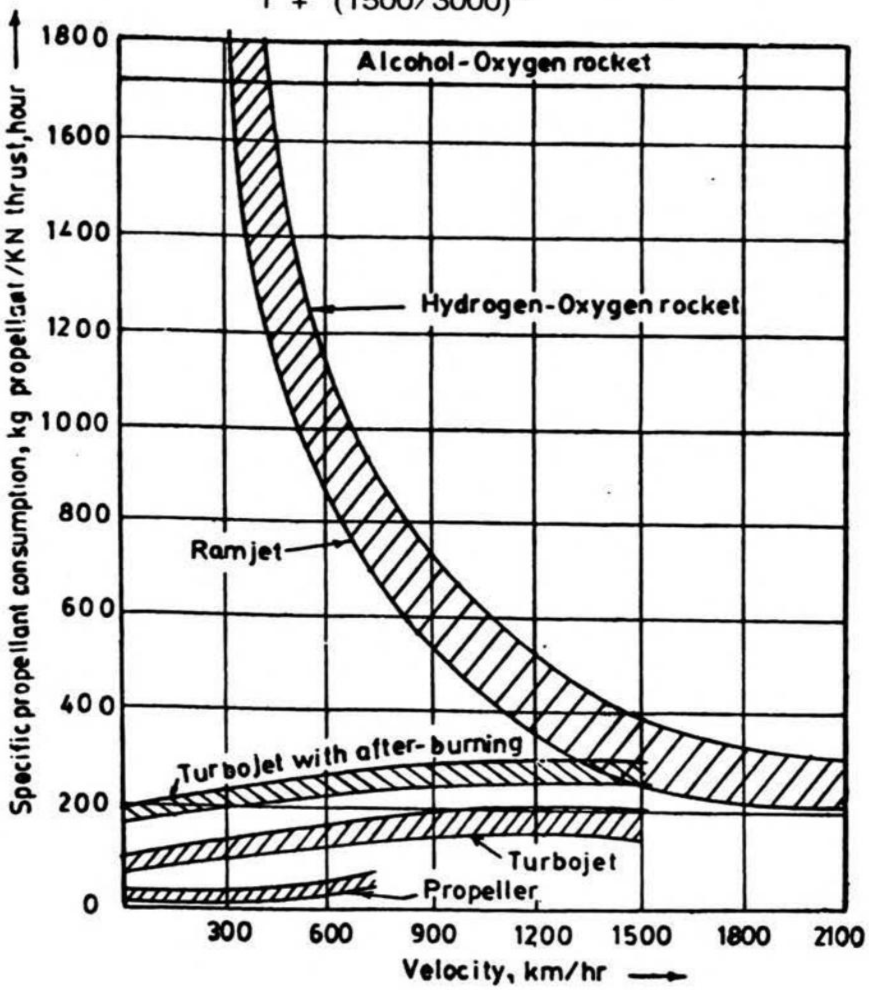

Fig. 5-10. Propellant or fuel consumption versus flight speed for different propulsion systems.

5.7 Comparison of the Various Propulsion Systems

Figure 5–10 shows the specific propellant consumption in kg per kN thrust versus speed for different engines. The curves in this figure indicate that the use of rocket engines to power air planes, as we know them today, is not feasible because of their high fuel consumption. Also, the use of ram jet engines is not economical at lower than 1500 km/hr vehicle speeds.

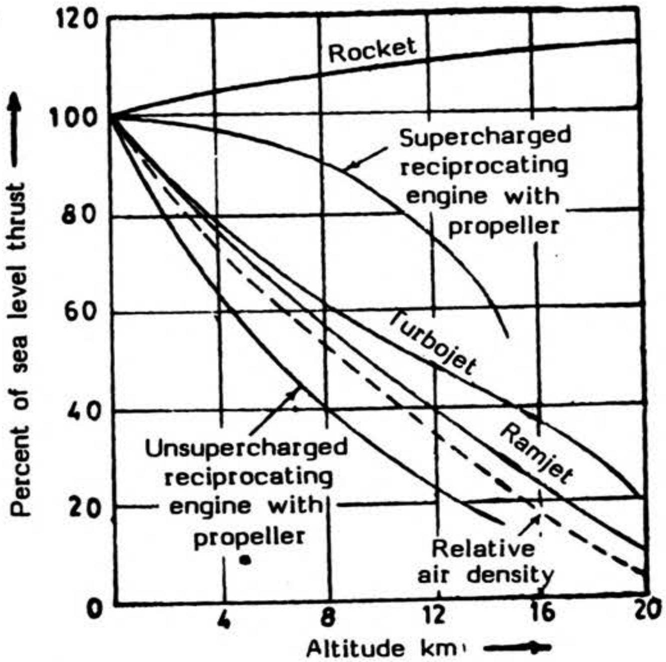

Fig. 5–11 Variation of thrust with altitude for different propulsion systems.

Figure 5–11 shows variation of thrust with altitude for different propulsion systems. It may be noted that the thrust of rocket motor increases with altitude while the thrust of other types of vehicles decreases with altitude.

Figure 5–12 gives relative picture of the probable operating envelope of the various propulsion systems.

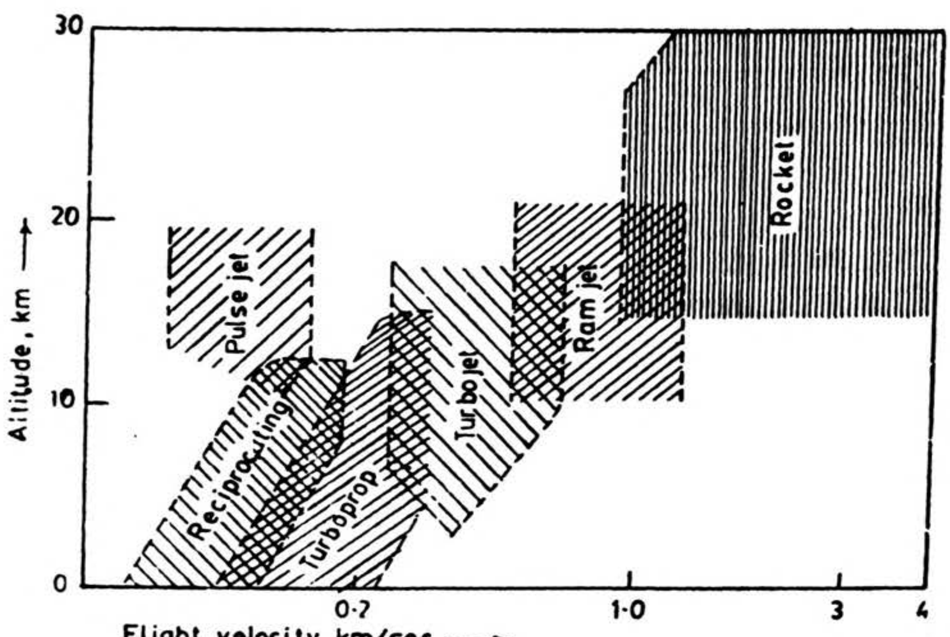

Fig. 5-12 Comparison of probable best performance for various propulsion engines.

INDUSTRIAL APPLICATIONS

INDUSTRIAL APPLICATIONS

- IN AIRCRAFT- Fighter plane, Missiles, Rocket, Airplane.
- Jet propulsion, land and sea transport, racing car.
- The first use of the jet engine was to power military aircraft.
- The General electric company used a "turboprop" jet engine to run an electric generator.
- The jet engine is not only used on aircraft but on boats, where water jets are used to propel the boat forward.
- Normal type of jet engine is used for domestic purpose i.e. Traveling, carrying goods etc.

An aircraft using this type of jet engine could dramatically reduce the time which it takes to travel from one place to another, potentially putting any place on Earth within a 90-minute flight.

Scramjet vehicle has been proposed for a single stage to tether vehicle, where a Mach 12 spinning orbital tether would pick up a payload from a vehicle at around 100 km and carry it to orbit

Rocket applications

1. Satellites in space serve air communication
2. Spacecraft
3. Missiles
4. Jet assisted air planes
5. Pilotless aircraft

TUTORIAL QUESTIONS

Theory Questions:

1. What are the different rocket propulsion systems? Brief the working differences between the propeller-jet, turbojet and turbo-prop.
2. With a neat diagram explain the working of rocket engine
3. Describe briefly about thrust augmentation method used in propulsion.
4. With a neat sketch, explain the working of turbo jet engine.
5. Differentiate between solid propellant and liquid propellant rocket engines.
6. What are the applications of pulse jet engines
7. Give the difference between ramjet and pulse jet engines
8. What are composite and homogeneous solid propellants? How do they work? State their merits and demerits.
9. What is the essential difference between rocket propulsion and turbo-jet propulsion?
10. Write a detailed classification of rockets. Explain liquid propellant rocket with a neat sketch Define and explain the terms:
    i. Thrust
    ii. Thrust power,
    iii. Effective jet exit velocity,
    iv. Propulsive efficiency related to turbojet engines.
11. What are the various applications of rockets?
12. Explain the advantages and disadvantages of bipropellants used in rocket engines over monopropellants.
2. Derive expressions for the thrust and propulsion efficiency of rockets and compare with those of turbojet

Numerical Problems:

1. A jet propulsion system has to create a thrust of 100 tones to move the system at a velocity of 700 km/hr. If the gas flow rate through the system is restricted to a maximum of 30 kg/s. find the exit gas velocity and propulsive efficiency.

2. In a jet propulsion unit, initial pressure and temperature to the compressor are 1.0 bar and 100C. The speed of the unit is 200m/s. The pressure and temperature of the gases before entering the turbine are 7500 C and 3 bar. Isentropic efficiencies of compressor and turbine are 85% and 80%. The static back pressure of the nozzle is 0.5 bar and efficiency of the nozzle is 90%. Determine (a) Power consumed by compressor per kg of air. (b)Air-fuel ratio if calorific value of fuel is 35,000 kJ/kg. Cp of gases=1.12 kJ/kg K, _ =1.32 for gases.
3. A turbo-jet engine flying at a speed of 960 km/h consumes air at the rate of 54.5 kg/s. calculate i). Exit velocity of the jet when the enthalpy change for the nozzle is 200 KJ/kg and velocity coefficient is 0.97. ii).fuel flow rate in kg/s when air fuel ratio is 75:1 iii). Thrust specific fuel consumption iv). Propulsive power v). Propulsive efficiency.
4. A simple turbine jet unit was tested when stationary and the ambient conditions were 1bar and 150C. The pressure ratio for the compressor was 4:1. A fuel consumption of 0.37kg/s was obtained for an air flow of 23kg/s. Calculate the thrust produced if the exhaust gases from the turbine were expanded to atmospheric pressure in a convergent nozzle. Assume the following data:

   Isentropic efficiency of compressor-80% Isentropic efficiency of turbine-85%

   Efficiency of nozzle-93% Transmission efficiency-98%

   Calorific value of fuel-42000kJ/kg Assuming working fluid to be air throughout.

5. In a turbojet, air is compressed in an axial compressor at inlet conditions of 1 bar and 1000C

   3.5 bar. The final temperature is 1.25 times that for isentropic compression. The temperature of gases at inlet to turbine is 4800C. The exhaust gases from turbine are expanded in a velocity of approach is negligible and expansion may be taken to be isentropic in both turbine and nozzle. Value of gas constant R and index r are same for air and flue gases.

   Determine

   i) Power required to drive the compressor per kg of air/sec

   ii) Air-fuel ratio if the calorific value of fuel is 42,000 kJ/kg

   iii) Thrust developed / kg of air / sec.

ASSIGNMENT QUESTIONS

ASSIGNMENT QUESTIONS

1. Why is thrust augmentation necessary? What are the methods for thrust augmentation in a turbojet engine?
2. A turbo-jet engine flying at a speed of 960 km/h consumes air at the rate of 54.5 kg/s. calculate i). Exit velocity of the jet when the enthalpy change for the nozzle is 200 KJ/kg and velocity coefficient is 0.97. ii).fuel flow rate in kg/s when air fuel ratio is 75:1 iii). Thrust specific fuel consumption iv). Propulsive power v). Propulsive efficiency.
3. With a neat diagram explain the working of rocket engine
4. What is turbine and classify them?

PREVIOUS QUESTION PAPERS

R15

Code No: R15A0313

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(Autonomous Institution – UGC, Govt. of India)

III B.Tech I Semester Regular/supplementary Examinations, November 2018

Advanced Thermal Engineering

(ME)

Roll No										

Time: 3 hours **Max. Marks: 75**

Note: This question paper contains two parts A and B
Part A is compulsory which carriers 25 marks and Answer all questions.
Part B Consists of 5 SECTIONS (One SECTION for each UNIT). Answer FIVE Questions, Choosing ONE Question from each SECTION and each Question carries 10 marks.

Note: Steam tables are allowed.

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PART-A (25 Marks)

1). a	Draw the line diagram of a Rankine cycle and mention the various components	[2M]
b	State the advantages of regenerative cycle over simple Rankine cycle.	[3M]
c	State the differences between fire tube and water tube boilers	[2M]
d	What is the function of a safety valve?	[3M]
e	Mention any two differences between jet and the surface condenser	[2M]
f	Write the advantages and disadvantages of steam turbines	[3M]
g	What do you mean by the term gas turbine?	[2M]
h	State the merits of gas turbines over IC engines.	[3M]
i	Draw the gas turbine power plant with inter cooling	[2M]
j	What is thrust augmentation	[3M]

PART-B (50 MARKS)

SECTION-I

2 Explain a regenerative cycle with a diagram and derive the expression for its thermal efficiency. **[10M]**

OR

3 In a Rankinne cycle the steam at inlet to turbine is saturated at a pressure of 35 bar and exhaust pressure is at 0.2 bar. Determine i) the pump work ii) the turbine work iii) Rankine efficiency iv) the condenser heat flow v) the dryness at the end of expansion. **[10M]**

SECTION-II

4 a) Explain ***any two*** of the following with neat sketches (5M) **[10M]**
i) Super heater ii) Air Preheater iii) Economizer
b) List the advantages of high pressure boilers (5M)

OR

5 a) Steam having pressure of 10.5 bar and 0.95 dryness fraction is expanded through a convergent- divergent nozzle and the pressure of steam leaving the nozzle is 0.85 bar. Find the velocity at the throat for maximum discharge **[10M]**

condition. If the index of expansion may be assumed to be 1.135, calculate the mass flow rate of steam through the nozzle. (5M)

b) Explain ***<u>any two</u>*** of the following with sketches

i) pressure gauge ii) water level gauge iii) feed check valve iv) high steam and low water safety valve. (5M)

SECTION-III

6 What are the compounding methods used in reducing the speed of the turbine rotor? Explain any two methods. **[10M]**

OR

7 A single stage steam turbine is supplied with steam at 5 bar, 200 ^{o}C at the rate of 50 kg/min. It expands into a condenser at a pressure of 0.2 bar. The blade speed is 400 m/s. The nozzles are inclined at an angle of 20 degree to the plane of the wheel and outlet blade angle is 30 degrees. Neglecting friction losses calculate i) power developed ii) blade efficiency and iii) stage efficiency. **[10M]**

SECTION-IV

8 a) Describe with neat sketches the working of a simple constant pressure open cycle gas turbine.(5M) **[10M]**

b) Discuss briefly the methods employed for improvement of thermal efficiency of open cycle gas turbine plant. (5M)

OR

9 A gas turbine unit has a pressure ratio of 6:1 and maximum cycle temperature of 610 ^{o}C. The isentropic efficiencies of the compressor and turbine are 0.80 and 0.82 respectively. Calculate the power output in kW of an electrical generator geared to the turbine when the air enters the compressor at 15 ^{o}C at the rate of 16 kg/sec. Take C_p as 1.005 kJ/kgK. γ=1.4 for the compression process and take C_p= 1.11 kJ/kgK and γ =1.333 for the expansion process. **[10M]**

SECTION-V

10 a) Explain the working difference between propeller -jet, turbo-jet and turbo- prop. (5M) **[10M]**

b) State the fundamental differences between jet propulsion and rocket propulsion. (5M)

OR

11(a) With a neat diagram explain the working of rocket engine **[5M]**

(b) Describe briefly about thrust augmentation method used in propulsion. **[5M]**

R15

Code No: R15A0313

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(Autonomous Institution – UGC, Govt. of India)

III B.Tech I Semester Supplementary Examinations, May 2019

Advanced Thermal Engineering

(ME)

Roll No										

Time: 3 hours **Max. Marks: 75**

Note: This question paper contains two parts A and B
Part A is compulsory which carriers 25 marks and Answer all questions.
Part B Consists of 5 SECTIONS (One SECTION for each UNIT). Answer FIVE Questions, Choosing ONE Question from each SECTION and each Question carries 10 marks.

PART-A (25 Marks)

1). a State the methods of increasing the thermal efficiency of a Rankine cycle. **[2M]**

b Draw the PV diagram of a Rankine cycle and mention the different processes. **[3M]**

c Write the differences between forced circulation and free circulation boilers **[2M]**

d What is the function of boiler mountings? Can a boiler work without mountings? **[3M]**

e Define a steam condenser **[2M]**

f What do you understand by diagram efficiency in case of steam turbine **[3M]**

g How are gas turbines classified? **[2M]**

h State the merits of gas turbines over steam turbines. **[3M]**

i Mention the different operating variables that affect the efficiency of a gas turbine plant. **[2M]**

j What are the applications of pulse jet engines? **[3M]**

PART-B (50 MARKS)

<u>SECTION-I</u>

2 a) In a steam power cycle, the steam supply is at 15 bar and dry and saturated. The condenser pressure is 0.4 bar. Calculate the Rankine and Carnot efficiencies of the cycle. Neglect the pump work. **[5M]**

.b) Draw the block diagram of reheat cycle by representing all the components and explain the salient features of the cycle. **[5M]**

OR

3 A reheat Rankine cycle operates between the pressure limits of 26 bar and 0.04 bar. The steam entering the HP turbine and LP turbine has a temperature of 400 ^{o}C. The steam leaves the HP turbine as dry saturated. Compare thermal efficiency and steam rate of Rankine cycle without and with reheating. Neglect the feed pump work. **[10M]**

<u>SECTION-II</u>

4 a) Steam is expanded in a set of nozzles from 10 bar and 200 ^{O}C to 5 bar. Neglecting the initial velocity, find the maximum area of the nozzle required to allow a flow of 3 kg/s under the given conditions. Assume that the expansion of **[5M]**

the steam to be isentropic. Also name the type of nozzle.

b) Clearly explain about any one type of high pressure boiler **[5M]**

OR

5 Dry saturated steam enters a steam nozzle at a pressure of 15 bar and is discharged at a pressure of 2 bar. If the dryness fraction of discharge steam is 0.96, what will be the final velocity of steam? Neglect the initial velocity of the steam. If 10% of the heat drop is lost in friction, find the percentage reduction in the final velocity. **[10M]**

SECTION-III

6 Derive the expression for maximum blade efficiency of a single stage impulse turbine. **[10M]**

OR

7 Define the following as related to steam turbines. **[6M]**

a) i) Blade Speed ratio ii) blade velocity coefficient iii) diagram efficiency iv) stage efficiency

B) explain the difference between an impulse turbine and a reaction turbine **[4M]**

SECTION-IV

8 Describe with neat diagram a closed cycle gas turbine and explain advantages, disadvantages and applications. **[10M]**

OR

9 A gas turbine unit receives air at 1 bar and 300 K and compresses it adiabatically to 6.2 bar. The compressor efficiency is 88%. The fuel has heating value of 44186 kJ/kg and the fuel air ratio is 0.017 kJ/kg of air. The turbine internal efficiency is 90%. Calculate the work of turbine and compressor per kg of air compressed and thermal efficiency for products of combustion, C_p =1.147 kJ/kgK and γ =1.333. **[10M]**

SECTION-V

10 a) Derive the equation for thrust, thrust power of a jet propulsion. **[5M]**

b) The following data pertain to a turbojet flying at a altitude of 9500 m: speed of the turbojet is 800 km/hr, propulsive efficiency=55% overall efficiency of a turbine plant is 17%. Density of air at 9500 m altitude is 0.17kg/m$^{3.}$ Drag on the plane is 6100 N, assuming calorific value of the fuel used as 46000 kJ/kg. Calculate : i) absolute velocity of the jet ii) Volume of air compressed /min. **[5M]**

OR

11 a) With a neat sketch, explain the working of turbo jet engine. **[5M]**

b) Differentiate between solid propellant and liquid propellant rocket engines. **[5M]**

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R15

Code No: R15A0313

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(Autonomous Institution – UGC, Govt. of India)

III B.Tech I Semester supplementary Examinations, May 2018

Advanced Thermal Engineering

(ME)

Roll No										

Time: 3 hours **Max. Marks: 75**

Note: This question paper contains two parts A and B

Part A is compulsory which carriers 25 marks and Answer all questions.

Part B Consists of 5 SECTIONS (One SECTION for each UNIT). Answer FIVE Questions, Choosing ONE Question from each SECTION and each Question carries 10 marks.

Steam tables and Mollier chart may be permitted

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PART – A (25 Marks)

1. (a) How Rankine efficiency can be improved? (2M)
   (b) What are the advantages of Regenerative cycle? (3M)
   (c) What is the function of Fusible plug in steam boilers? (2M)
   (d) What do you understand by the term ‘Critical pressure’ as applied to steam nozzles? (3M)
   (e) Define Degree of reaction? (2M)
   (f) Give the difference between Impulse and Reaction Turbines? (3M)
   (g) What is meant by closed cycle gas turbine? (2M)
   (h) Why Re-heater is necessary in gas turbine? What are its effects? (3M)
   (i) What is Jet Propulsion? (2M)
   (j) Give the difference between ramjet and pulse jet engines? (3M)

PART – B (50 Marks)

SECTION – I

2. a) Show the Rankine cycle on p-v and T-s diagrams and explain the processes involved. Also draw the mechanical system to show different processes of the Rankine cycle. (5M)
   b) In an ideal reheat cycle, the steam enters the turbine at 30 bar and 500^0C.After expansion to 5 bar,the steam is reheated to 500^0C and then expanded to the condenser pressure of 0.1 bar. Determine the cycle thermal efficiency, mass flow rate of steam.
   Take power output as 100MW. (5M)

(OR)

3. a) What are the methods which can lead to increase in thermal efficiency of Rankine cycle? (5M)
   b) A steam power plant has boiler and condenser pressures of 60 bar and 0.1 bar, respectively. Steam coming out of the boiler is dry and saturated. The plant operates on the Rankine cycle. Calculate thermal efficiency. (5M)

SECTION – II

4. a) With the help of neat sketch, explain Cochran Boiler. What are its special features? (5M)
   b) A nozzle is to be designed to expand steam at the rate of 0.10 kg/s from 500kPa, 210^0C to 100kPa. Neglect inlet velocity of steam. For a nozzle efficiency of 0.9, determine the exit area of the nozzle. (5M)

(OR)

5. a) What are the differentiating features between a water tube and fire tube boiler? (5M)
   b) Starting from the fundamentals, show that the maximum discharge through the nozzle,the ratio of throat pressure to inlet pressure is given by $(2/n+1)^{n/n-1}$, where n is the index for isentropic expansion through the nozzle. (5M)

SECTION – III

6. a) What is compounding? Describe various methods of compounding with neat sketches of arrangement, pressure and velocity profiles. (5M)
   b) The following data refers to a single stage impulse turbine: Steam velocity = 800m/s; Blade speed = 300 m/s; Nozzle angle = 20^0; Blade outlet angle = 25^0.Neglecting the effect of friction, calculate the power developed by the turbine for the steam flow rate of 25kg/s. Also calculate the axial thrust on the bearings. (5M)

(OR)

7. a) Prove that for a 50% reaction turbines $\alpha=\varphi$ and $\theta=\beta$. (5M)
   b) In one stage of a reaction turbine, both fixed and moving blades have inlet and outlet blade tip angles of 35^0 and 20^0 respectively. The mean blade speed is 80m/s and the steam consumption is 22500 kg/hr. Determine the power developed and stage efficiency if the isentropic heat drops in both fixed and moving rows is 23.5 kJ/kg in the pair. (5M)

SECTION – IV

8. A gas turbine plant works between the temperature limits of 1152 K and 288 K. Isentropic efficiencies for Compressor and Turbine are 0.85 and 0.8 respectively. Determine the optimum pressure ratio for maximum work output and also find maximum cycle thermal efficiency. (10M)

(OR)

9. Explain with neat sketch the gas turbine cycles with intercooling and reheating and what will be the condition of maximum output. (10M)

SECTION – V

10. Why is thrust augmentation necessary? What are the methods for thrust augmentation in a turbojet engine? (10M)

(OR)

11. What are composite and homogeneous solid propellants? How do they work? State their merits and demerits. (10M)

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