DIGITAL NOTES

ADVANCED THERMAL ENGINEERING
(R17A0312)

B.Tech – III Year – I Sem

DEPARTMENT OF MECHANICAL ENGINEERING

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

(An Autonomous Institution – UGC, Govt.of India)

Recognizes under 2(f) and 12(B) of UGC ACT 1956

(Affiliated to JNTUH, Hyderabad, Approved by AICTE –Accredited by NBA & NAAC-"A" Grade-ISO 9001:2015 Certified)

MALLA REDDY COLLEGE OF ENGINEERING & TECHNOLOGY

III Year B. Tech, ME-I Sem

L	T/P/D	C
4	1	4

(R17A0312) ADVANCED THERMAL ENGINEERING

Objectives:

- Applications and the principles of thermodynamics to components and systems.
- The purpose of this course is to enable the student to gain an understanding of how thermodynamic principles govern the behavior of various systems
- Student have knowledge of methods of analysis and design of complicated thermodynamic systems

UNIT-I

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

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

UNIT-II

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

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

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. Treatise on thermal engineering/ V.P.Vasandhani and D.S.Kumar/Metropolitan

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. Gas Turbines / V. Ganesan / TMH.

OUTCOMES: At the end of the course, student will be able to

- Recognize main and supplementary elements of Steam engines and Gas turbine engines and define operational principles.
- Describe the most important combustion concepts and problems in concern with Steam turbine engines and Gas turbine engines.
- be able to analyze energy distribution in an external combustion engine like steam turbine and gas turbine engines
- Develop problem solving skills through the application of thermodynamics.
- Solve problems associated with turbo machines like steam and gas turbines.

CONTENTS

Course Coverage:

S.NO	NAME OF THE UNIT	Text Book	Chapters Referred
1	Steam Boilers and Condensors	T.E. by Rajput R.K	C11 – C14 C20
2	Basic Concepts of Rankine Cycle Steam Nozzle	T.E. by Rajput R.K	C15 C18
3	Steam Turbines	T.E. by Rajput R.K	C19
4	Gas Turbines	T.E. by Rajput R.K	C21
5	Jet Propulsion	T.E. by Vasandani & D.S.Kumar	C18 C19

UNIT-I

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.

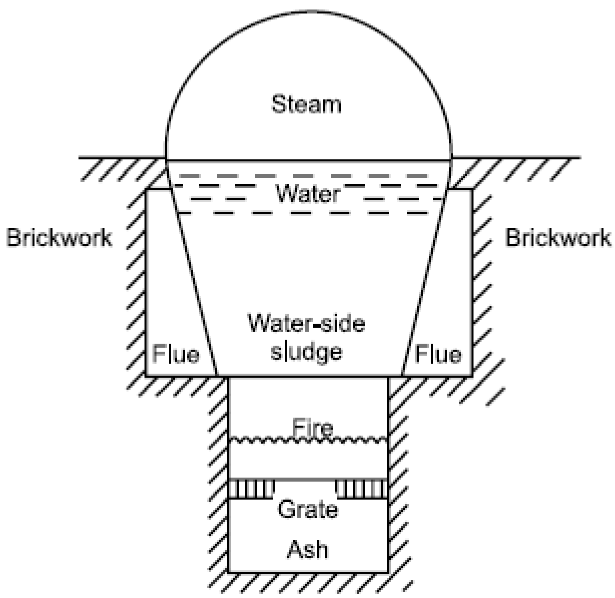


Fig. 1.1 *Wagon boiler of Watt,* (1788)

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.

Boiler technology got revolutionized during Second World War, when the need arose for the boilers to supply steam to field installations. Field requirements were critical as the boiler installation and commissioning should take place in minimum time. Therefore the 'Package boilers' which were complete with all auxiliaries as one unit came up and gradually transformed into modern boiler having lot of accessories and mountings. Thus in a boiler other than heat supplying unit, shell and tubes, a number of other devices are used for its control, safe and efficient operation. Devices which are mounted on boiler for its control and safe operation are called "mountings" while devices which are mounted on boiler for improving its performance are called "accessories". Thus boiler mountings are necessary while boiler accessories are optional.

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 boiler, 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 and Water tube Boilers

Fire tube boilers are those boilers in which hot gases (combustion products) flow inside the tubes and water surrounds them. Water extracts heat for its phase transformation from the hot gases flowing inside the tubes, thus heat is indirectly transferred from hot gas to water through a metal interface.

Such boilers came up in eighteenth century and were extensively used for steam generation in variety of applications. With the passage of time and coming up of another types of boilers the fire tube boilers have lost their charm to some extent due to limitations in terms of steam pressure. Fire tube boilers are used for applications having small steam requirement. Different types of fire tube boilers have been discussed ahead.

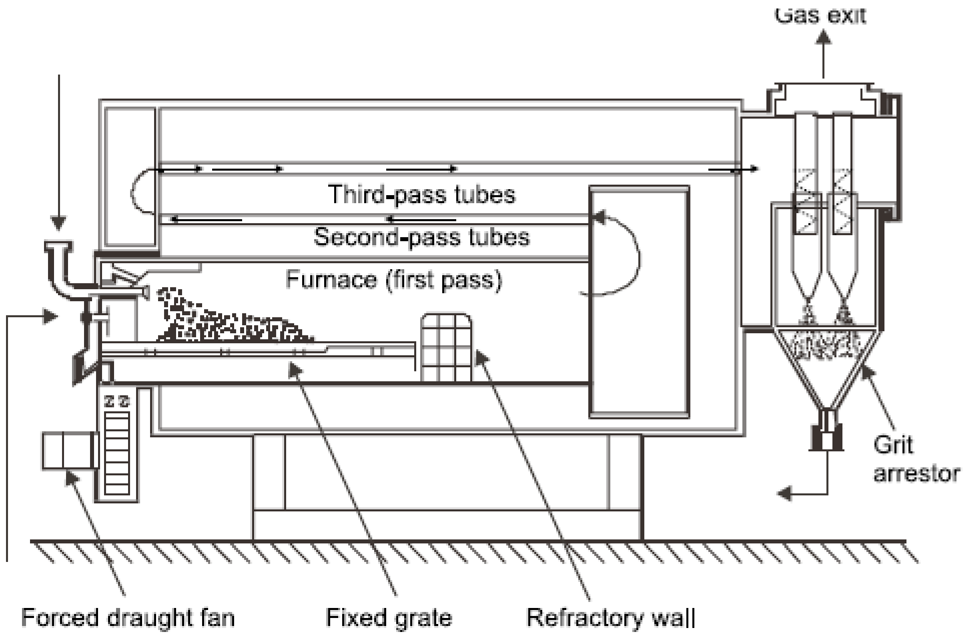


Fig. 1.2 *Fire tube boiler*

Water tube boilers are those boilers in which water flows inside the tubes and hot gases surround them. This type of boilers came up as a solution to the problem of explosion faced in fire tube boilers when the pressure and steam generation capacity were increased. In such boilers the shell behaved as heated pressure vessel subjected to internal pressure which set up tensile stresses (hoop stress) in walls.

Mathematically, this stress can be given as,

Hoop stress = $P*D/(2t)$

Where P is internal working pressure, D is diameter of shell and t is thickness of shell wall.

Above expression shows that if 'P' (pressure) increases then either 'D' (diameter) should be decreased or't' (thickness) be increased to keep stress within acceptable limits. While increasing thickness the mass of boiler and cost of manufacturing both increase therefore the reduction of 'D' (diameter) is an attractive option. This became the basis for water tube boilers in which small diameter of tube facilitated quite high pressure steam generation.

Such boilers came up in late eighteenth and nineteenth century. George Babcock and Stephen Wilcox gave straight-tube boiler of water tube boiler type in 1867 which was subsequently modified and developed as present ‘Babcock and Wilcox boiler’.

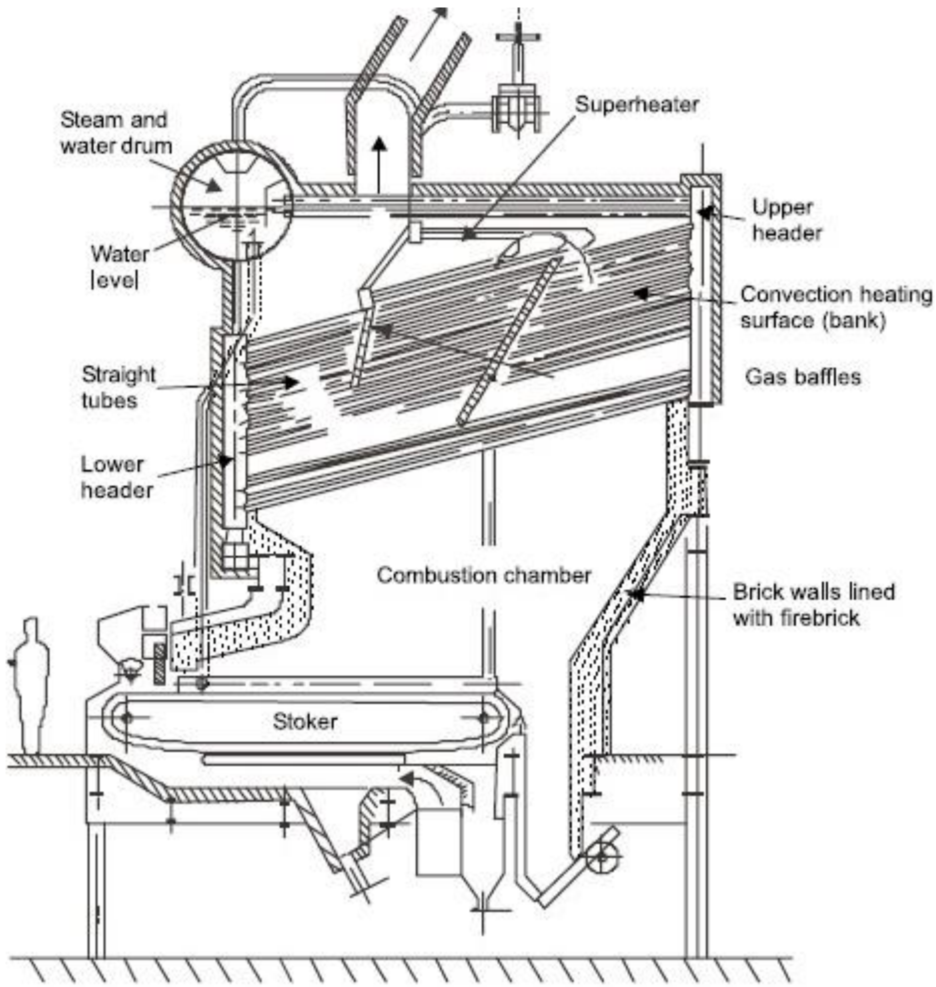


Fig. 1.3 *Water tube boiler*

Water tube boilers may be further classified based on type of tubes employed. These can be *Straight water tube boilers* and *Bent water tube* boilers. Straight water tube boilers are those in which tubes carrying water are straight from one end to the other end. At the two ends headers are provided.

In general water comes down from drum into down header and after passing through tubes get heated and evaporated to steam which is carried back to drum through up-comer header or riser. Circulation of water is caused by the density difference as density of feed water is more than density of hot water/wet/dry steam due to lower temperature of feed water.

Bent water tube boilers are those in which bent tubes are employed for carrying water. Bent water tubes are advantageous over straight water tubes in many respects. Bent tubes offer better access into boiler and ease of inspection and maintenance. Also tube arrangement can be modified so as to maximize heating surface and exposure of tubes to hot gases.

Circulation is better in case of bent tube boilers as compared to straight tube, since the orientation of tubes in case of former is generally at inclination from vertical while for later it is horizontal. Stirling boiler is one such boiler. In water tube boilers the heat distribution generally occurs amongst economizer tubes, evaporator tubes, super heater tubes. Hottest gases are designed to come in contact with super heater tubes. The evaporator tubes are in between super heater and economizer tubes.

Simple Vertical Boiler

Simple vertical boiler shown in Fig. 1.4 has a vertical boiler shell of cylindrical shape. It has fire box of cylindrical type inside the shell. Vertical passage of tubular type called uptake is provided over fire box for exhaust of flue gases. Cross tubes are provided for improving water circulation and increasing heating surface. At the bottom of fire box a fire grate is provided for burning fuel. Total heating surface area is about 7–10 times grate area. Man hole and hand holes are provided in the shell for access to inside of shell. Hot gases raising from fire grate go upwardly and heat the water contained in shell and tubes.

Steam generated in shell can be tapped through a steam stop valve placed on the crown of shell. Such boilers have steam generation capacity up to 1000 kg per hour and maximum steam pressure up to 10 bar. Size of the boiler ranges from 0.6 m diameter to 2 m diameter and height from 1.2 m to 4 m high. Boiler efficiency is nearly 50%.

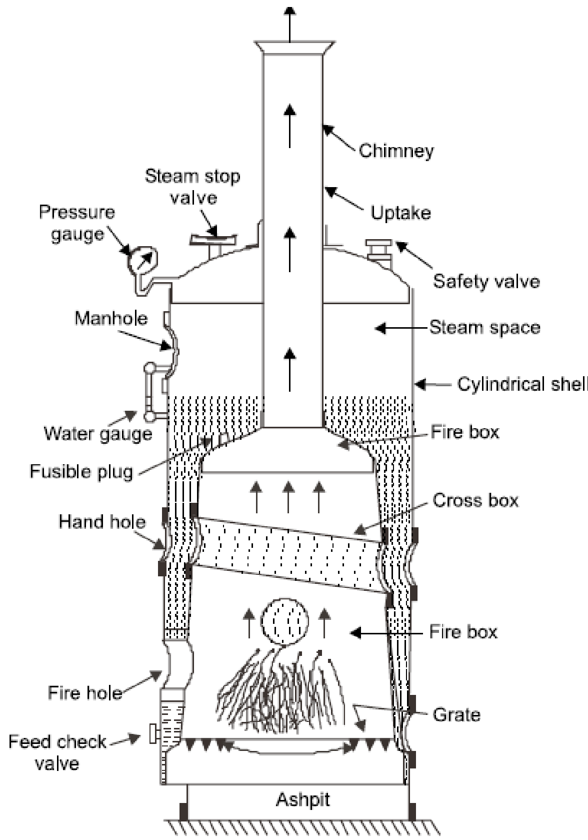


Fig. 1.4 *Simple vertical boiler*

Cochran Boiler

This is a fire tube boiler of vertical type and came up as a modification over the simple vertical boiler in order to maximize heating surface. Total heating surface area is 10–25 times the grate area. It has cylindrical shell with hemispherical crown. Hemispherical geometry offers maximum volume space for given mass of material and is also very good for strength and maximization of radiant heat absorption. Figure. 1.5 shows the schematic of Cochran boiler with various mountings upon it. Fire box is also of hemispherical form. Flue gases flow from fire box to refractory material lined combustion chamber through a flue pipe. Incomplete combustion if any can get completed in combustion chamber and hot gases subsequently enter into tubes. After coming out of fire tubes hot gases enter into smoke box having chimney upon it. As the fire box is separately located so any type of fuel such as wood, paddy husk, oil fuel etc. can be easily burnt. These boilers are capable of generating steam up to pressure of 20 bar and steam generating capacity from 20 kg/hr to 3000 kg/hr. Boilers have dimensions ranging from 1m diameter and 2 m height to 3 m diameter and 6 m height. Efficiency of such boilers ranges between 70 and 75%.

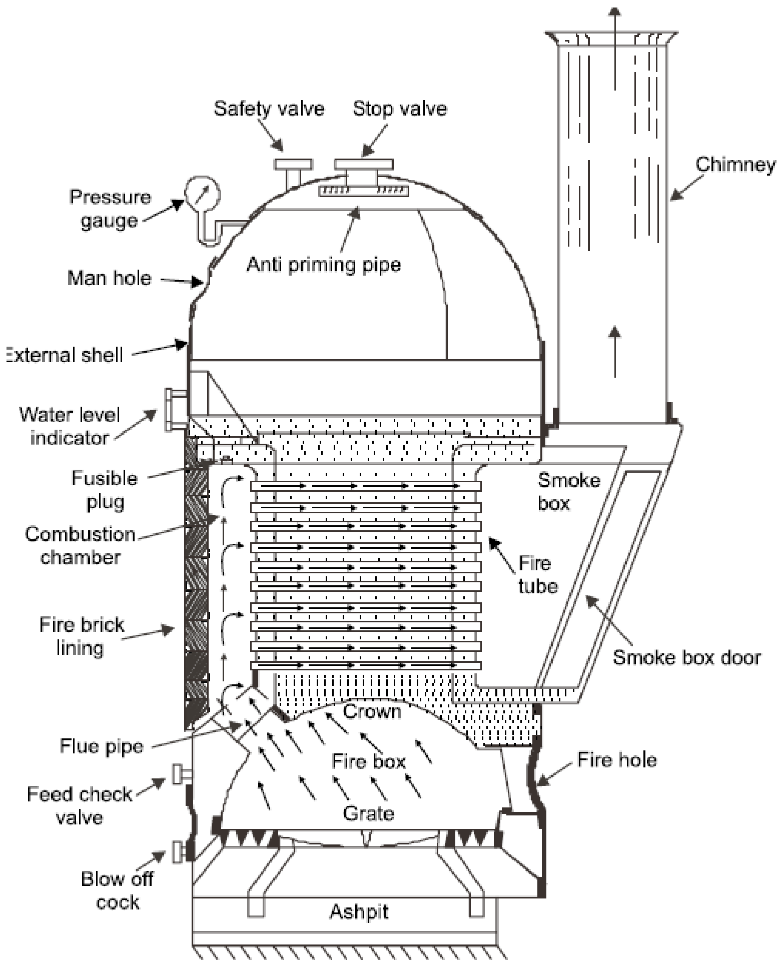


Fig. 1.5 *Cochran boiler*

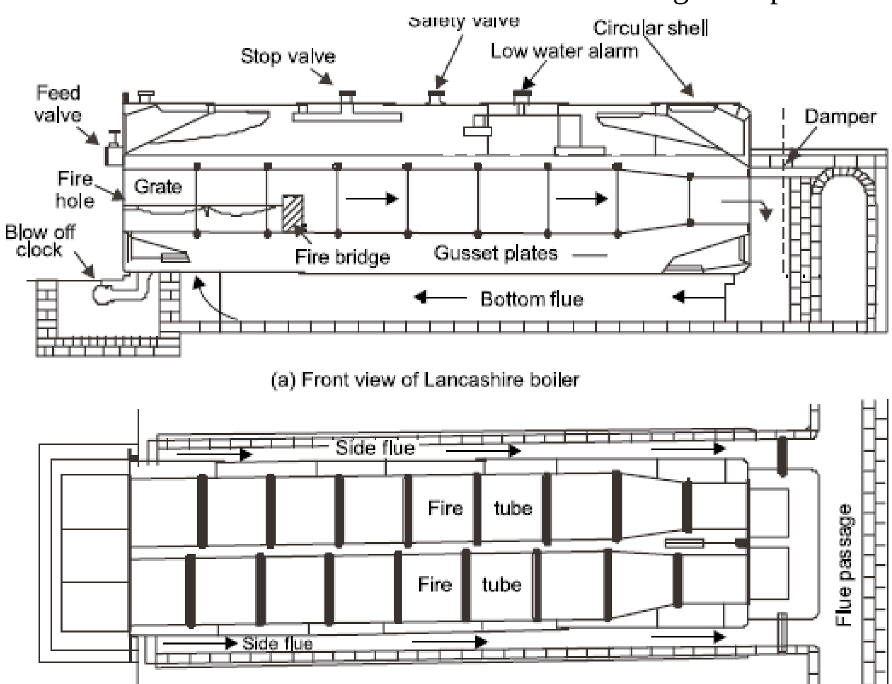
Lancashire Boiler

It is a horizontal fire tube boiler. General arrangement in the boiler is shown in Fig. 1.6.

Boiler is mounted on brickwork *setting* with front end of shell sloping about 1: 250 for emptying the shell. It has a circular shell connected *to end plates* supported by gusset plates. Two *fire tubes* run throughout the length of the boiler. Fire tubes are of diameter less than half the diameter of shell and diameter of fire tubes is reduced as shown to have access to lower side of boiler.

Fire Bridge is provided to prevent fuel from falling over the end of furnace. Fire bridge also helps in producing a better mixture of air and gases for perfect combustion by partly enveloping the combustion space. Hot gases start from grate area, enter into fire tubes and come out at back of boiler from where these gases flow towards the front of boiler through *bottom flue*. Upon reaching the front these hot gases flow through the *side flues* and enter the *main outlet*. Outlet passage may also be used commonly by more than one boilers. About 85% of actual heat transferred is transferred through surface of fire tubes while 15% is transferred through bottom and side flues.

Plan, elevation and side views of Lancashire boiler shown in figure explain the furnace,



(a) Front view of Lancashire boiler

(b) Top view of Lancashire boiler

different firetubes, bottom flues, side flues etc. Dampers are provided at the end of side flues for regulating the pressure difference (draught) for exit of burnt gases. Other mountings and accessories are shown in the elevation of Lancashire boiler.

Cornish Boiler

This is a horizontal fire tube boiler having single flue gas tube. General arrangement is very similar to Lancashire boiler. Water surrounds the flue gas tube in the shell. Hot flue gases after passing through the tube are divided into two portions at the end of boiler and pass through side flue passages to reach upto the front of boiler and then enter into bottom flue gas passage for escaping out through chimney after traversing the entire length of bottom passage. Hot gases thus traverse complete length of passage from end to end of boiler thrice i.e. through main flue gas tube, side flues and bottom flues. Heat transfer is more from side flues than bottom flue due to sedimentation in bottom. These boilers are generally capable of producing steam up to the rate of 1350 kg/hr and maximum steam pressure up to 12 bar. Shell is generally of length 4 to 7 m and diameter 1.2 to 1.8 m.

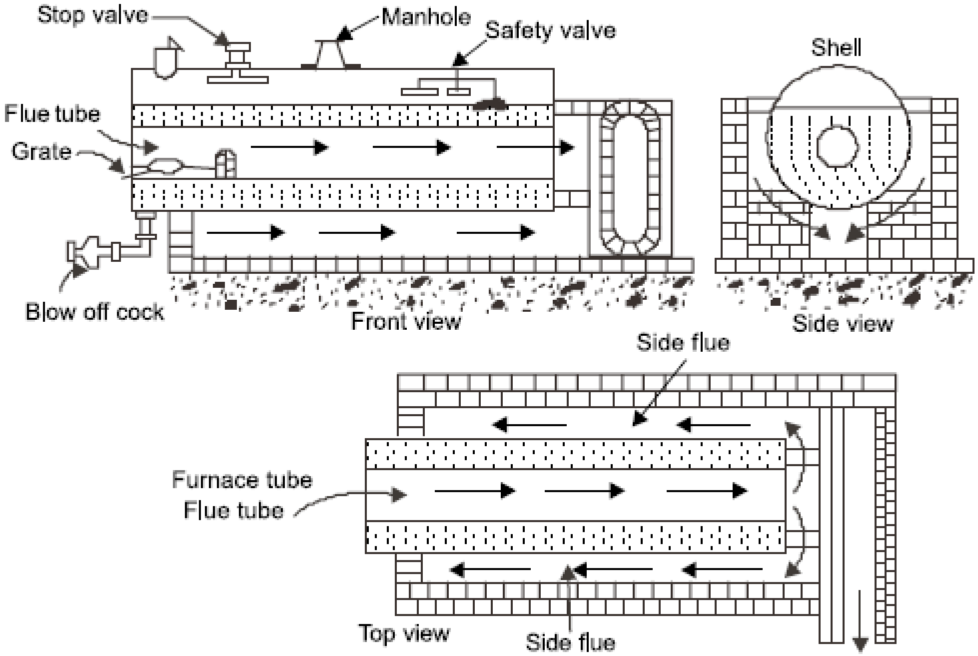


Fig. 1.7 *Cornish boiler*

Locomotive Boiler

These boilers were invented for getting steam to run a steam engine used in locomotives. These are fire tube type of boilers. It has basically three parts i.e. smoke box, shell and fire box. Figure 1.8 shows a general arrangement in locomotive boiler.

Inside *fire box* the fuel (coal) is burnt over the *grate*. For feeding fuel the *fire* hole is used. Hot gases produced in fire box are diverted by *fire brick arch* and enter into the *fire tubes* surrounded with water. Steam produced gets collected in a steam drum fitted on top of the shell. Arrangement for super heating is there in these boilers.

As shown the wet steam goes through inlet headers of super heater and after passing through tubes, it returns to the outlet header of super heater and is taken out for steam engine. A very large door is provided at the end of smoke box so as to facilitate cleaning and maintenance of complete boiler.

As it is a moving boiler, therefore, its chimney is completely eliminated. For expelling the burnt gases (draught) the exhaust steam coming out from steam engine is being used. Thus it is an artificial draught used in these boilers for expelling burnt gases.

Babcock and Wilcox Boiler

It is a water tube boiler suitable for meeting demand of increased pressure and large evaporation capacity or large sized boiler units. Figure 1.9 shows the Babcock and Wilcox boiler. It has three main parts:

(i) Steam and water drum
(ii) Water tubes
(iii) Furnace.

Steam and water drum is a long drum fabricated using small shells riveted together. End *cover plates* can be opened as and when required. Mountings are mounted on drum as shown. Drum is followed by water tubes which are arranged below drum and connected to one another and drum through headers. Header in which water flows from drum to tubes is called *down take header* while headers in which flow is from tubes to drum is called *uptake header*.

Soot deposition takes place in mud box which is connected to down take header. “Blow off cock” for blowing out the sediments settled in *mud box* is shown in figure. Super heater tubes are also shown in the arrangement, which are U-shape tubes placed horizontally between drum and water tubes. Superheating of steam is realized in super heater tubes.

Below the super heater and water tubes is the *furnace*, at the front of which *fuel feed hopper* is attached. *Mechanical stoker* is arranged below the hopper for feeding fuel. Bridge wall and baffles made of fire resistant bricks are constructed so as to facilitate hot gases moving upward from the *grate* area, then downwards and again upwards before escaping to the chimney. A *smoke box* is put at the back of furnace through which smoke goes out via *chimney*, put at top of smoke box. A *damper* is used for regulating pressure difference (draught) causing expulsion of hot gases.

The complete boiler unit with all mountings and accessories is suspended by steel slings from girders resting on steel columns. It is done so as to permit free expansion and contraction of boiler parts with temperature.

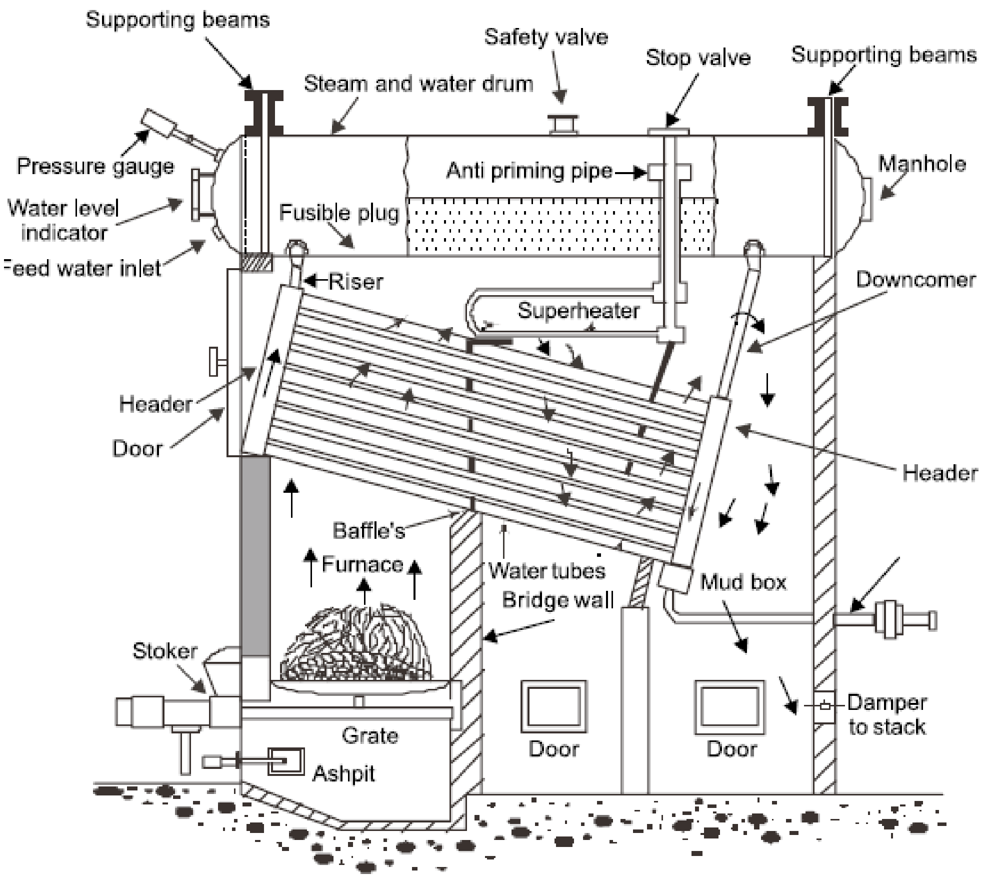


Fig. 1.9 *Babcock and Wilcox boiler*

High Pressure Boiler

High pressure boilers generally operate in supercritical range. Need of such boilers is felt because high pressure and temperature of steam generated in boiler improves plant efficiency. These boilers have forced circulation of water/steam in the boiler. This forced circulation is maintained by employing suitable pump. The steam drum is of very small size and in some cases it may be even absent too. This is because of using forced circulation. In case of natural circulation drum size has to be large. Schematic of high pressure boiler is shown in figure 1.10. In fact the high pressure boilers have been possible because of availability of high temperature resistant materials. Here direct heating of water tubes is done by the excessively hot gases present in fire box. The fire box has large volume as otherwise exposed water tubes shall melt. Heat is picked by number of parallel tubes containing water. These parallel tubes appear as if it is a wall due to close spacing of tubes. Water circulation circuit is shown in line diagram.

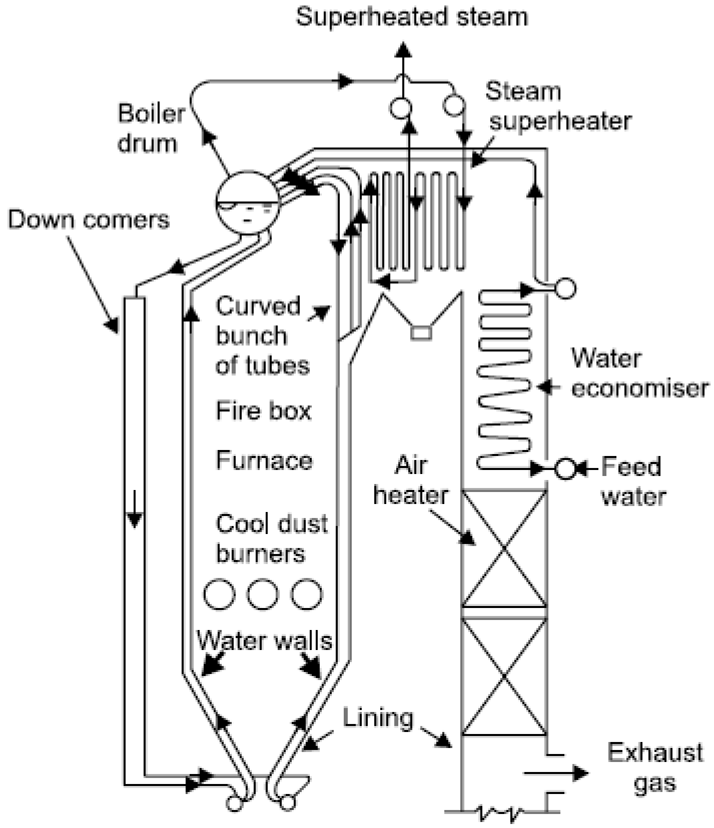


Fig. 1.10 *High pressure boiler with natural circulation*

High pressure boilers may have natural circulation in case the steam pressure desired lies between 100 and 170 bar and size is not constraint. High pressure boilers have capability of generating larger quantity of steam per unit of furnace volume.

High pressure boilers are disadvantageous from safety point of view and therefore, stringent reliability requirements of mountings is there.

Benson Boiler

It is a water tube boiler capable of generating steam at supercritical pressure. Figure 1.11 Shows the schematic of Benson boiler. Mark benson, 1992 conceived the idea of generating steam at supercritical pressure in which water flashes into vapour without any latent heat requirement. Above critical point the water transforms into steam in the absence of boiling and without any change in volume i.e. same density. Contrary to the bubble formation on tube surface impairing heat transfer in the normal pressure boilers, the supercritical steam generation does not have bubble formation and pulsations etc. due to it. Steam generation also occurs very quickly in these boilers. As the pressure and temperatures have to be more than critical point, so material of construction should be strong enough to withstand thermal stresses. Feed pump has to be of large capacity as pressure inside is quite high, which also lowers the plant efficiency due to large negative work requirement. Benson boilers generally have steam generation pressure more than critical pressure and steaming rate of about 130–135 tons/hr. Thermal efficiency of these boilers is of the order of 90%.

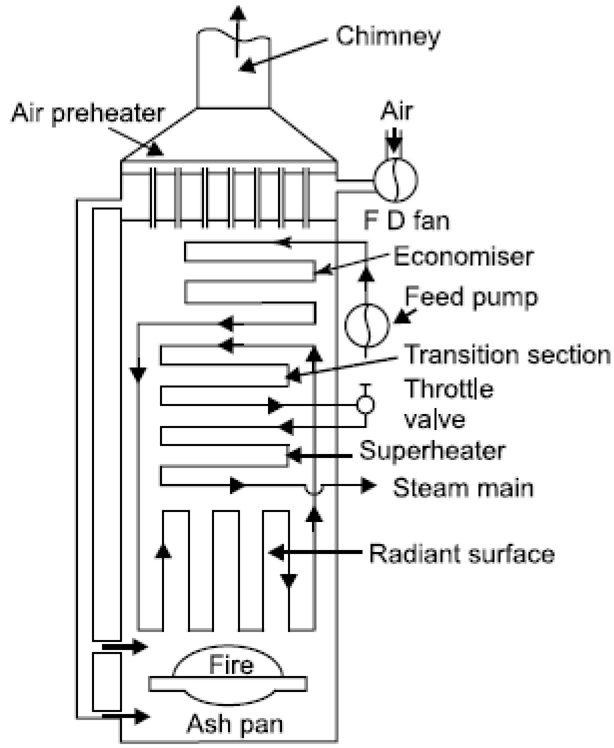


Fig. 1.11 *Benson boiler*

La Mont Boiler

This is a water tube boiler having forced circulation. Schematic showing the arrangement inside boiler is given in Fig. 1.12. Boiler has vertical shell having three distinct zones having water tubes in them, namely evaporator section, superheater section and economizer section.

Feed water is fed from feed pump to pass through economizer tubes. Hot water from economizer goes into drum from where hot feed water is picked up by a circulating pump. Centrifugal pump may be steam driven or of electric driven type. Pump increases pressure and water circulates through evaporation section so as to get converted into steam and enters back to drum. Steam available in drum enters into superheater tubes and after getting superheated steam leaves through steam main.

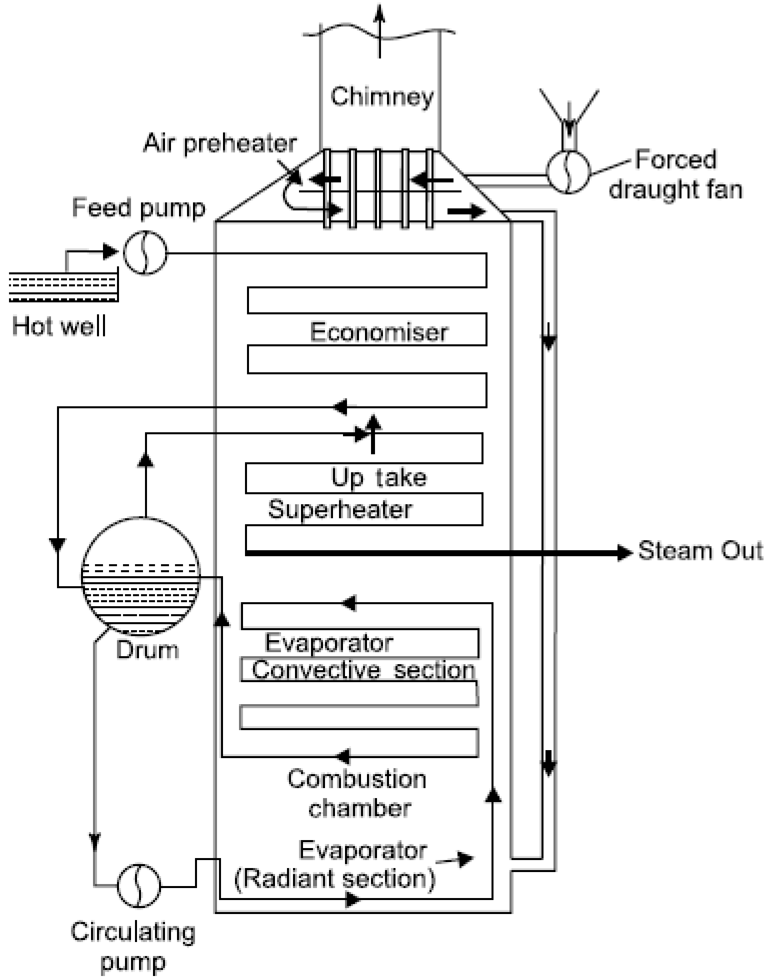


Fig. 1.12 *La Mont boiler*

Boiler Mountings and Accessories

Boiler mountings and accessories have been defined earlier and shown on the different boilers. Different mountings are

(i) Water level indicator
(ii) Safety valves
(iii) High steam and low water safety valves
(iv) Fusible plug
(v) Pressure gauge
(vi) Stop valve
(vii) Feed check valve
(viii) Blow off cock
(ix) Manhole and mud box

Various boiler accessories are:

(i) Superheater
(ii) Economiser
(iii) Air preheater
(iv) Feed pump

Water level indicator: It is used for knowing the level of water in boiler as water level inside boiler should not go below a certain limit. General arrangement is shown in Fig. 1.13 with the different parts in it. It has two tubes one is front glass tube while other is metal tube. Water level is seen through glass tube which is made strong enough to withstand high steam pressure and temperature. Two control cocks are provided for regulating steam and water passage from boiler to glass tube.

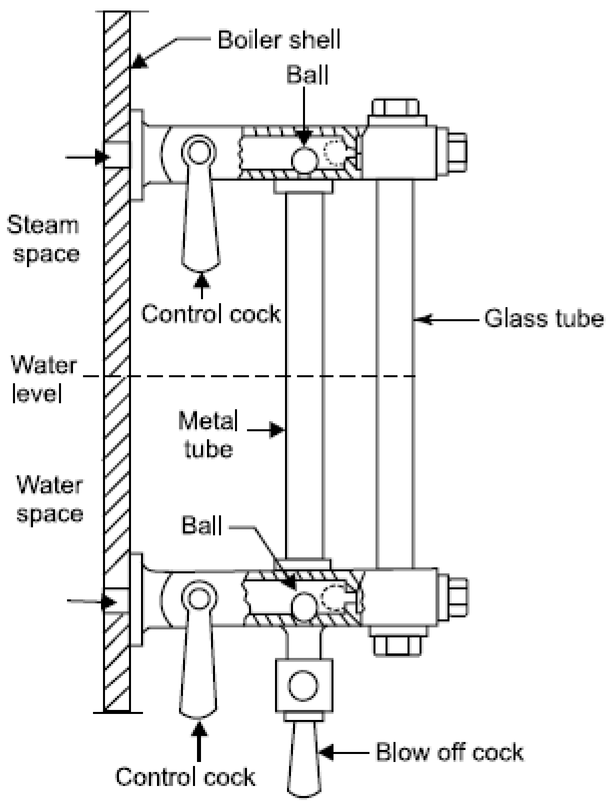


Fig. 1.13 *Water level indicator*

For blow off purpose a blowing cock is also provided as shown. In case of breakage of glass tube the possibility of accident is prevented by providing two balls. As glass tube breaks the rush of water and steam carries the two balls with it and closes the openings for glass tube, thus water and steam flowing out can be prevented. Numbers of other types of water level indicators are also available.

Safety valve: Its function is to prevent the steam pressure from exceeding a limiting maximum pressure value. Safety valve should operate automatically by releasing excess steam and bring pressure down within safe limits. These are of different types such as 'dead weight safety valve', 'lever safety valve' 'spring loaded safety valve' etc. Figure 1.14 gives the general description of 'dead weight safety valve'.

It has a large vertical pipe on the top of which a valve seat is fixed. Valve rests upon this valve seat. A weight carrier is hung on the top of valve upon which cast iron rings enclosed in cast iron cover are placed in weight carrier as dead weight.

When the pressure of steam exceeds the total weight of valve, it is lifted and falls back as steam pressure gets reduced.

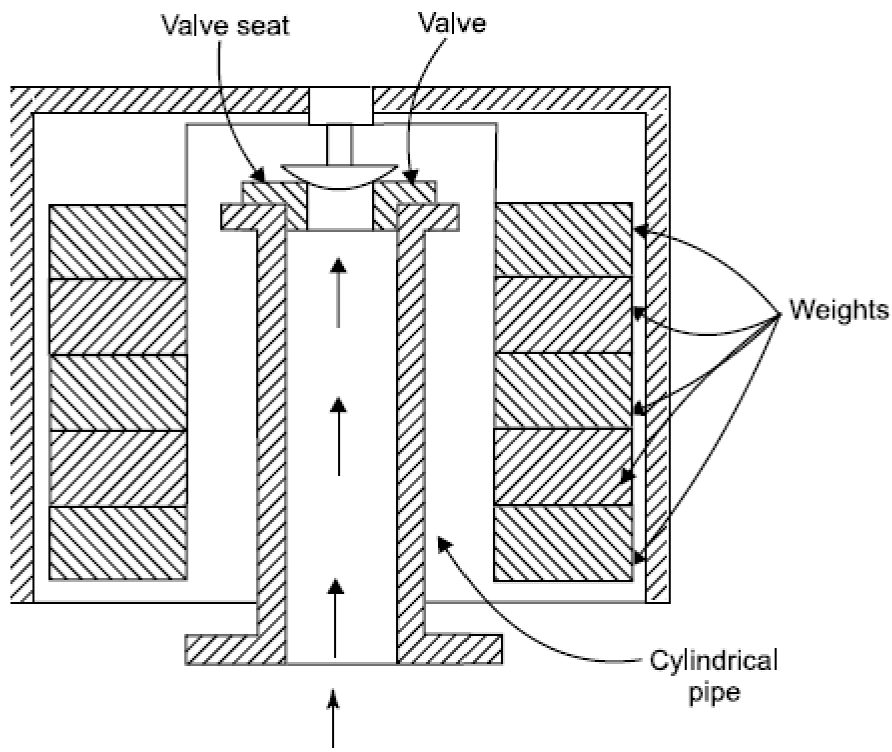


Fig. 1.14 *Dead weight safety valve*

It has a large vertical pipe on the top of which a valve seat is fixed. Valve rests upon this valve seat. A weight carrier is hung on the top of valve upon which cast iron rings enclosed in cast iron cover are placed in weight carrier as dead weight.

When the pressure of steam exceeds the total weight of valve, it is lifted and falls back as steam pressure gets reduced.

Fusible plug: It is a safety device used for preventing the level of water from going down below a critical point and thus avoid overheating. Fusible plug is mounted at crown plate of combustion chamber.

Fusible plug has gun metal body and a copper plug put with fusible metal at interface of copper plug and gun metal body. As water level goes down the heat available from furnace could not be completely utilized for steam formation and so the overheating may cause melting of fusible metal.

Fusible metal is a low melting point metal. Thus upon melting of lining the copper plug falls down and water falls from this opening onto furnace and thus quenches fire.

Pressure gauge: It is mounted at front top. Generally Bourdon type pressure gauge are being used for pressure measurement. Pressure is continuously monitored so as to avoid occurrence of over shooting of boiler pressure. Although safety devices to protect boiler against pressure rising beyond a limit are provided but pressure gauges are also used for monitoring pressure.

Stop valve: It regulates the flow of steam from the boiler as shown in Fig 1.15. This is generally mounted on highest part of boiler shell and performs function of regulating the flow of steam from boiler. Stop valve generally has main body of cast steel, valve, valve seat and nut etc. are of brass. Stop valve can be easily operated by rotating the hand wheel which causes lifting or lowering of spindle, thus causing opening or closing of valve.

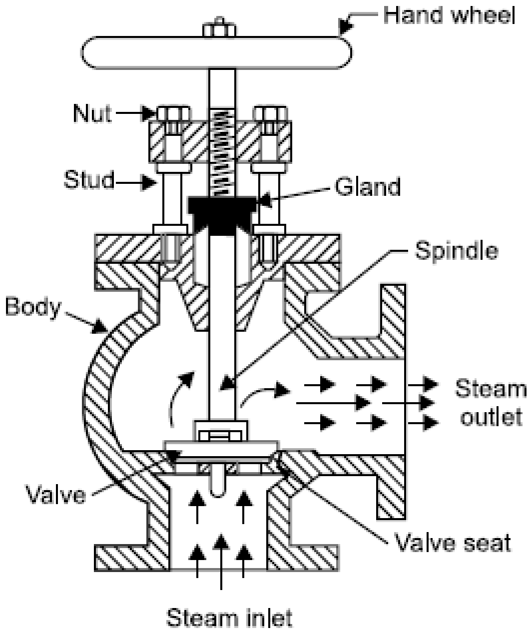


Fig. 1.15 *Stop valve*

Feed check valve: It is a non return valve at the end of delivery pipe from feed water pump and is placed on boiler shell slightly below normal water level. Figure 1.16 shows the arrangement in a feed check valve. It has a check valve whose opening and closing are regulated by the position of spindle. By hand wheel rotation the position of spindle can be altered suitably. Feed check valve permits unidirectional flow of water from feed pump to be boiler shell. Under normal running the pressure of feed water coming from pump is more than pressure inside the boiler and so the feed water continues to enter the shell. While during the non working of feed pump the pressure in boiler shell is more and so the check valve gets closed.

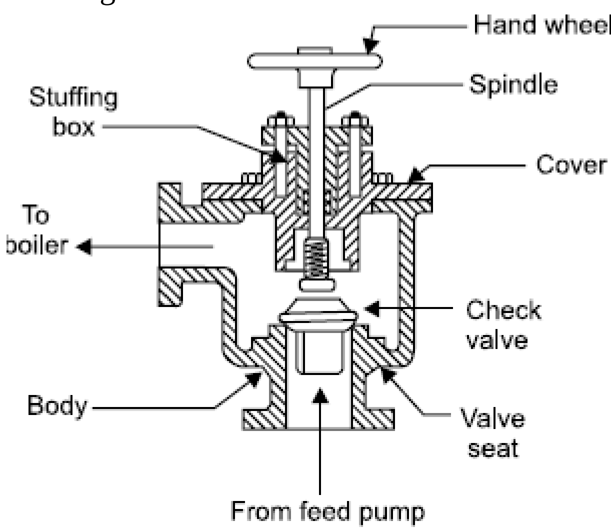


Fig. 1.16 *Feed check valve*

Blow off cock*:* It is used for periodical cleaning by discharging the water and sediments from bottom of boiler. Figure 1.17 shows the blow off cock. Blow off cock is fitted to the bottom of boiler shell. Blow off cock has a plug of conical type put into the mating casing. Plug position is altered for opening and closing the flow. Plug has rectangular opening which when comes in line with inlet and outlet passage then blow off cock is open and when opening is not in line then cock is closed. Plug is rotated by spindle.

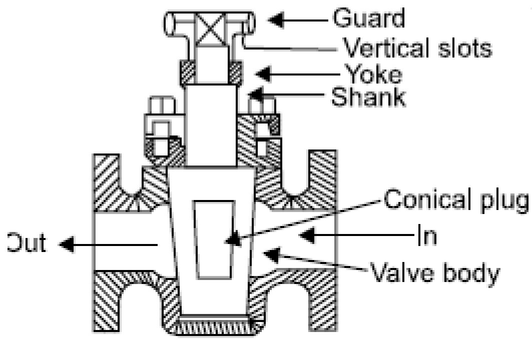
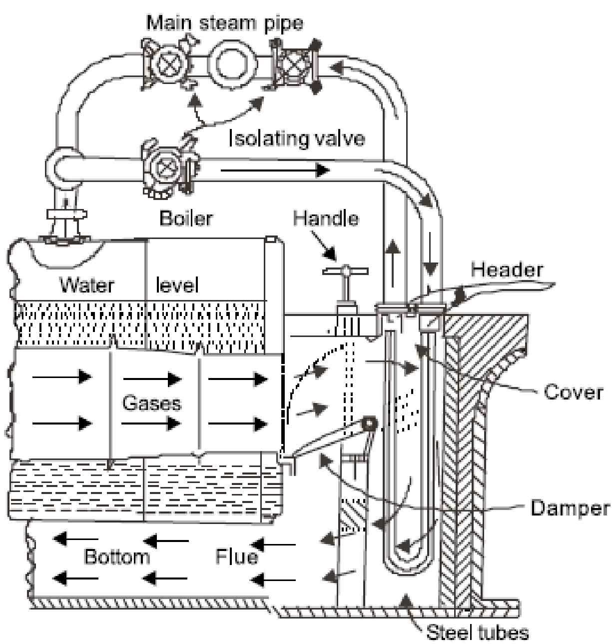


Fig. 1.17 *Blow off cock*

Blow off cock also helps in regulating the salt concentration as frequent draining helps in throwing out the salt deposited over period of time. Opening blow off cock removes deposited sediments in boiler.

Superheater: Its purpose is to super heat steam and is a type of heat exchanger in which steam flows inside tubes and hot gases surround it. Figure 1.18 shows the smooth tube hairpin type super heater (Sudgen's superheater) and convective and radiant superheater.

In hair pin superheater the steam generated is passed through isolating valve to U-shaped steel tubes. Superheated steam leaves superheater through tube connected to steam stop valve. Hot gases from fire tube are diverted over superheater tubes by damper as shown. These hot gases upon passing over steel tubes leave boiler through bottom flue. The convective and radiant superheater as shown has two set of tubes picking up heat through convection and radiation.



(a) Smooth tube hairpin type superheater

Fig 1.18

Economizer: It is also a heat recovery device in which feed water is heated from heat available with exhaust gases. Thus hot feed water available from economizer lowers the fuel requirement in combustion. It is also a type of heat exchanger having exhaust gas and feed water as two fluids. General arrangement in economizer is shown in Fig. 1.19. Economizer also helps in removal of dissolved gases by preheating of water and thus minimizes tendency of corrosion and pitting. Hotter feed water also reduces thermal strain in boiler parts. Economizer is located in the boiler structure so as to expose the economizer surface to hot gases. Its location varies with the boiler designs. Typical economizer called Green’s economizer as shown in Fig. 1.19 has vertical pipes of cast iron fitted with two headers at bottom and top respectively. Feed water passes through bottom header, economizer tubes and top header to boiler. Thus economizer is simply a heat exchanger where heat is transferred from hot flue gases to water inside the tubes through metal interface. Top header is also provided with a safety valve so as to avoid explosion due to excessive pressure of water developing inside economizer tubes. Bottom header is also provided with a blow off valve so as to throw out the sediments deposited in feed water. Economizer is also provided with scrapers fitted to clean pipes from the deposition of soot carried by the flue gases. Continuous scrapping is always desired so as to maximize heat transfer rate. Economizer also has a by pass provided so that flue gases can be diverted when economizer is out of full or part operation due to failure or cleaning purpose or feed water temperature control.

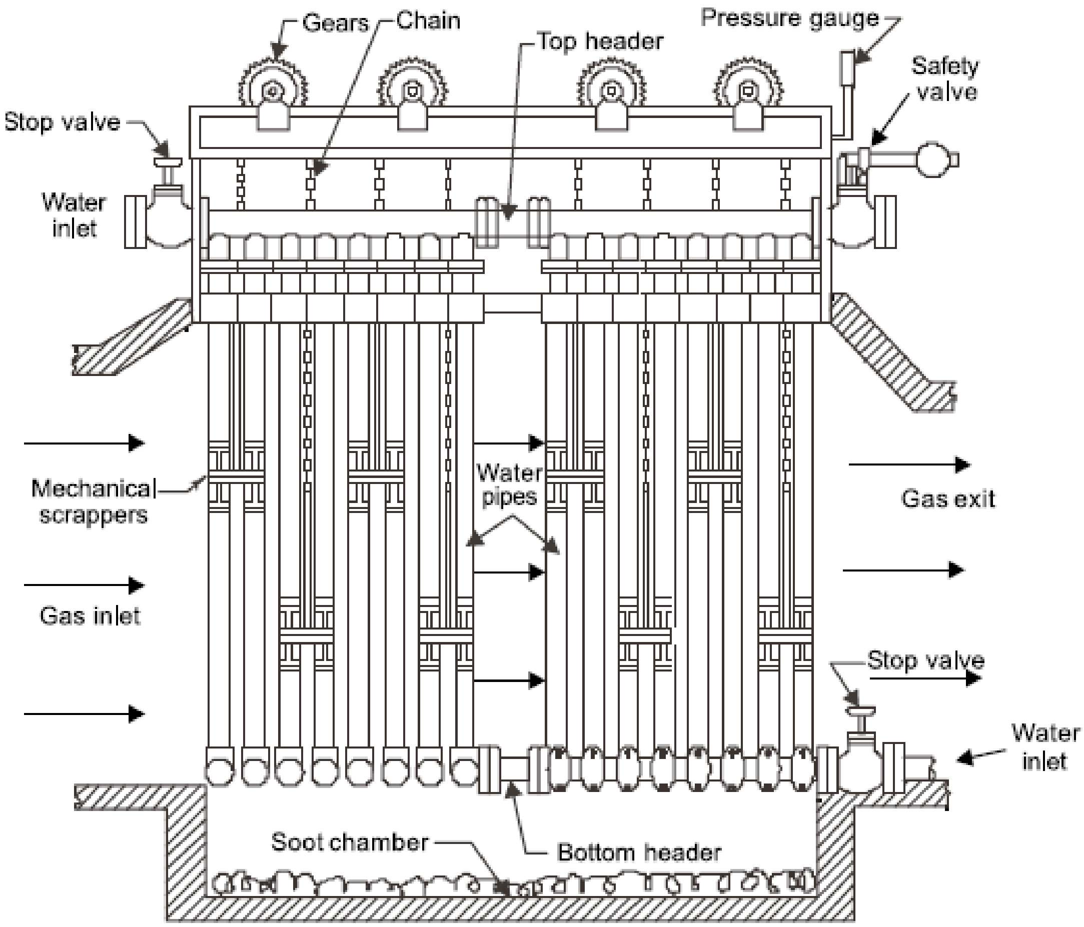


Fig. 1.19 *Economizer*

Air preheater: It is used for recovering the heat going along with exhaust gases by the air before being sent to furnace. Heat is recovered by passing exhaust gases through an air to air heat exchanger as shown in Fig. 1.20. Air preheaters are generally placed after economizer and before chimney. Air when preheated before supply to furnace/combustion chamber helps in achieving ‘faster rate of combustion’, ‘possibility of burning inferior quality coal/fuel’ and ‘increased rate of evaporation from boiler’ etc.

Air preheaters are of tubular type, plate type and regenerative type. This classification of air preheaters bases upon the kind of arrangement used for heat exchange between two fluids. Generally, tubular type air preheater are generally used in small boilers. Tubular air preheater has hot flue gases passing inside tubes and air blown over these tubes.

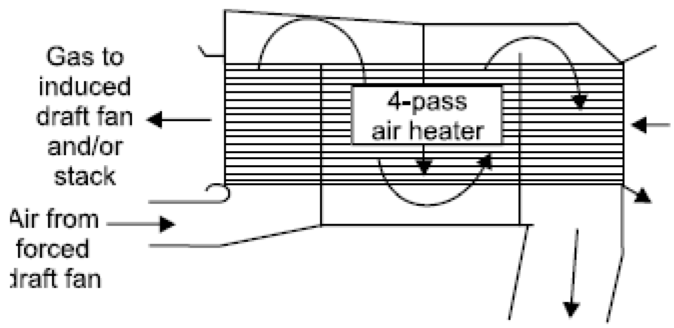


Fig. 1.20 *Tubular air preheater*

In case of plate type air preheater there are number of plates having air and flue gases flowing through alternative spacing. In regenerative type air preheater there is a wire mesh rotor which is alternatively heated and cooled by the hot flue gases and air to be used for combustion.

Feed pump: Feed pump is used for sending water into boiler at the pressure at which steam generation takes place. It is generally of three types i.e. centrifugal pump, reciprocating pump and injectors.

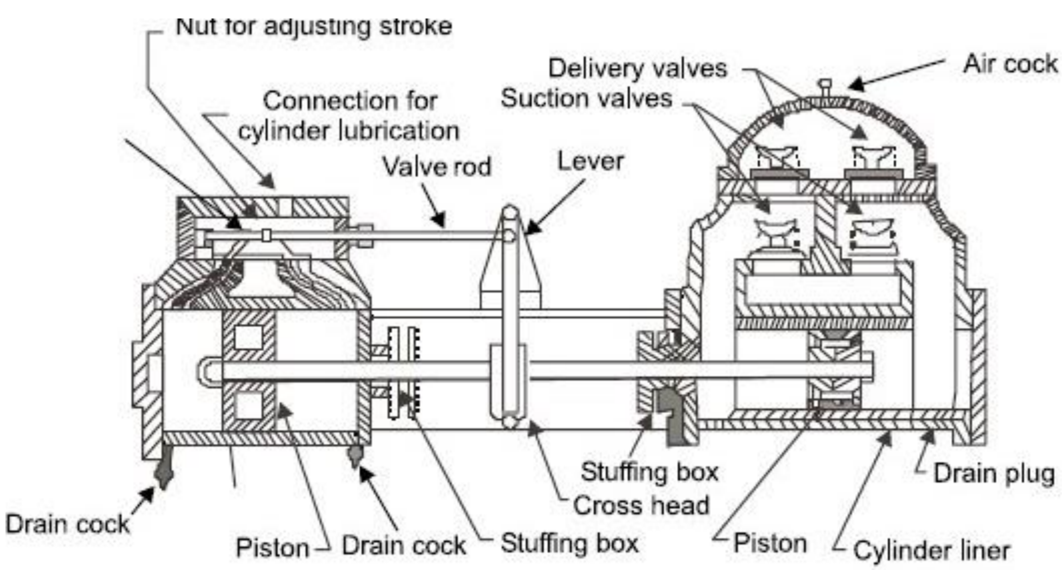


Fig. 1.21 *Reciprocating type pump, Duplex feed pump*

A reciprocating type feed pump is shown in Fig. 1.21. In boilers the pumps raise feed water pressure to the value more than the highest operating pressure of boiler. Pumps also have capability to deliver feed water in excess to the maximum evaporation rate of boiler. This excess capacity of feed pump is generally 15–20% of maximum continuous rating and is required to meet one or more of following situations.

(i) Sometimes excessive steam demand may occur.
(ii) Since boilers are to be blown out frequently to remove depositions and salts, therefore excess capacity is required.
(iii) Malfunctioning of boiler may cause carrying away of water with steam, thereby causing water shortage in boiler.
(iv) Over a period of time pump capacity decreases and so excess pump capacity is desired.

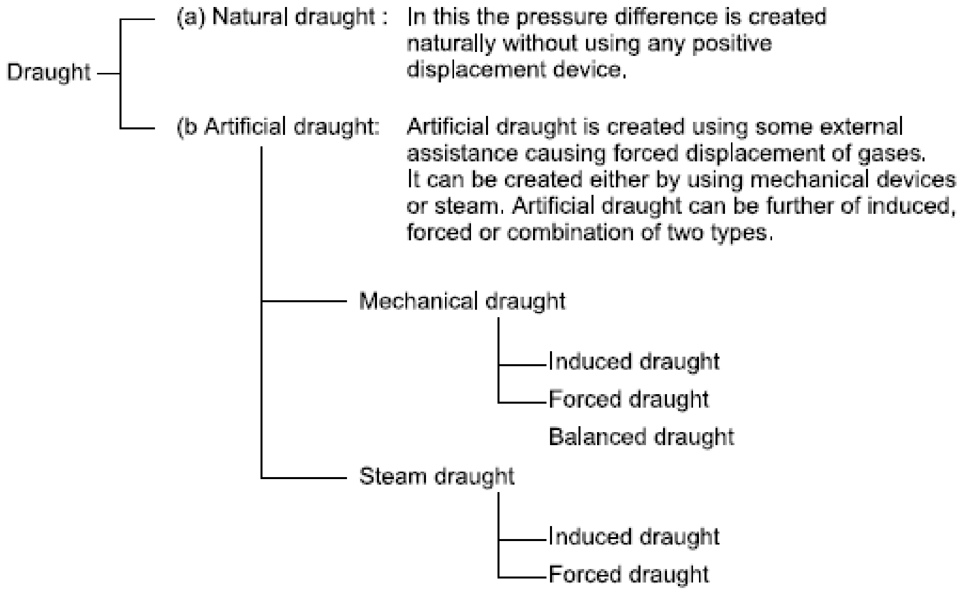
Boiler Draught

Draught refers to the pressure difference created for the flow of gases inside the boiler. Boiler unit has a requirement of the expulsion of combustion products and supply of fresh air inside furnace for continuous combustion. The obnoxious gases formed during combustion should be discharged at such an height as will render the gases unobjectionable. A chimney or stack is generally used for carrying these combustion products from inside of boiler to outside, i.e. draught is created by use of chimney.

Draught may be created naturally or artificially by using some external device. Draught can be classified as below:

1. In this the pressure difference is created naturally without using any positive displacement device.
2. Artificial draught is created using some external assistance causing forced displacement of gases. It can be created either by using mechanical devices or steam. Artificial draught can be of induced type, forced type or combination of two types.

Thus the draught in boiler may be said to be required for, ‘providing and maintaining the supply of sufficient air for combustion’, ‘expulsion of combustion products from furnace region’ and ‘discharge of burnt gases to atmosphere’. The amount of draught required shall depend upon, ‘type of boiler’, ‘rate of fuel burning’, rate at which combustion products are produced’ and ‘the air requirement rate’. As the pressure difference is very small so draught is measured in ‘mm’ of water. Mathematically, pressure due to 1 mm of water column is equivalent to 1 kgf/m^2



Natural Draught

It is produced employing chimney. The natural draught is produced by a chimney due to the fact that the hot gases inside the chimney are lighter than the outside cold air i.e. density difference of hot gases inside chimney and cold atmospheric air. Thus in a boiler unit the combustion products (hot) rise from fuel bed through chimney, and are replaced by fresh air (cold) entering the grate. It means that amount of draught produced by a chimney depends upon flue-gas temperature. Intensity of draught produced by chimney also depends upon height of chimney. Draught produced by a taller chimney is large as the difference in weight between the column of air inside and that of air outside increases with height. Generally draught is less than 12 kgf/m^2 in chimneys.

In stricter terms the word ‘chimney’ is used for brick or concrete structure and ‘stack’ is used for metallic one. Chimneys are generally made of steel, brick or reinforced concrete. Steel chimneys or stacks are most desirable for smaller boiler units due to small initial cost, ease of construction and erection. On account of small space requirement as compared to other stacks, self sustaining steel stacks are used in some large power plants. Steel stacks have problem of rust and corrosion, so painting requirements are quite stringent. Brick chimneys are required where permanent chimney with longer life is required. Such chimneys have inherent disadvantages of leakages etc. across the construction, therefore careful construction is required. Leakage of air across chimney wall effects intensity of draught.

Brick chimneys are constructed of round, octagonal, or square section. Generally brick chimney has two walls with air space between them and inner wall having fire brick lining. Concrete chimneys are used due to the absence of joints, light weight and space economy as compared with brick chimneys.

Also the reinforced concrete chimney is less expensive compared to brick chimney along with minimum chances of leakage across walls.

Calculations: As it is obvious from earlier discussion that the vertical duct called chimney creates natural draught so estimation of height of chimney is very important. Figure 1.22 shows the schematic of chimney in a boiler unit. During no working of boiler the pressure inside boiler is atmospheric pressure. Pressure at outlet of chimney will be less than atmospheric pressure due to altitude difference.

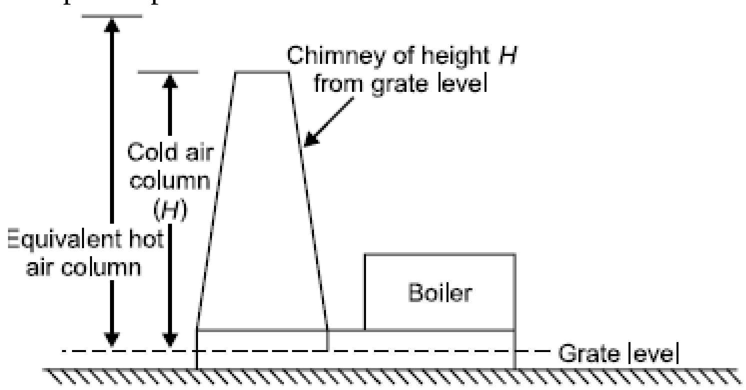


Fig. 1.22 *Schematic of chimney*

During boiler operation the chimney shall be filled with hot gases and the pressure at bottom of chimney (p_b) shall be summation of 'pressure at chimney outlet', (p_0) and 'pressure due to hot gas column of height *H*'.

Pressure at bottom of chimney = Pressure at outlet + Pressure due to hot gas column

$$P_b = P_0 + \rho_g \; g \; H$$

However, the pressure at grate level remains unchanged. Pressure difference between the grate level pressure and bottom of chimney causes flow of gases. This pressure difference is also called static draught.

Let us consider the combustion of fuel in furnace. Combustion products are released as a result of this combustion process. Fuel may be considered to be comprising of hydrocarbons.

Fuel + Air → Combustion products + Heating value

In the hydrocarbon fuel major constituents are carbon, hydrogen, nitrogen etc. As a result of complete combustion carbon gets transformed into carbon dioxide and hydrogen yields steam. The combustion products have major fraction of carbon dioxide and steam, but the volume of steam is negligible compared to volume of combustion product. The volume of combustion products can be taken equal to volume of air supplied, measured at same temperature and pressure.

Let us assume various properties as,

T_a = Atmospheric temperature, K
T_g = Average temperature of hot gases inside chimney, K

T_0 = Absolute zero temperature, 273.15_K

ρ_a = Density of air at absolute temperature, 1.293 kg/m3

ρ_g = Density of hot gases inside chimney, kg/m3

Δp = Pressure difference, draught in Pa

$\Delta p'$ = Pressure difference, draught in kgf/m^2

h_g = Equivalent height of hot gas column to produce draught, _p in 'metres'.

h_w = Equivalent height of water column in 'mm' to produce draught, _*p*

m = Mass of air supplied per kg of fuel

C = Hot gas velocity in chimney, m/s

M_g = Discharge rate through chimney, kg/s

Pressure of hot gases in chimney at grate level = Density of hot gases x height of chimney x gravitational acceleration

$$= \left\{\left(\frac{m+1}{m}\right)\frac{\rho_a \cdot T_0}{T_g}\right\} H \cdot g$$

Pressure due to cool air (outside) column of height *H* at grate level

= Density of air at *Ta* x Height x *g*

$$= \frac{\rho_a \cdot T_0 \cdot H \cdot g}{T_a}$$

Natural Draught produced = Difference of pressures due to cool air column and hot gas column of height '*H*'.

$$\boxed{\Delta p = \rho_a \cdot T_0 \cdot H \cdot g \left\{ \frac{1}{T_a} - \left(\frac{m+1}{m} \right) \cdot \frac{1}{T_g} \right\}, \text{ Pa}}$$

Substituting values for T_0, ρ_a and rounding off values we get,

$$\boxed{\Delta p' = 353 \cdot H \left\{ \frac{1}{T_a} - \left(\frac{m+1}{m} \right) \cdot \frac{1}{T_g} \right\}, \text{ kgf/m}^2}$$

Height of hot gas column equivalent to draught produced,

$$h_g = \frac{\Delta p', \text{ kgf/m}^2}{\text{Density of hot gases, kg/m}^3 \text{ at } T_g}$$

$$= \frac{\rho_a \cdot T_0 \cdot H \cdot \left\{ \frac{1}{T_a} - \left(\frac{m+1}{m} \right) \cdot \frac{1}{T_g} \right\}}{\left\{ \frac{\rho_a \cdot T_0}{T_g} \cdot \left(\frac{m+1}{m} \right) \right\}}$$

$$h_g = H \cdot \left\{ \left(\frac{m}{m+1} \right) \cdot \frac{T_g}{T_a} - 1 \right\}, \text{ metres}$$

Actually in boilers this draught requirement is different from that theoretically estimated due to the draught losses. The magnitude of these losses varies from boiler to boiler due to different arrangements within them. Some of generic losses shall be because of:

- Frictional losses due to resistance offered by passage surface roughness, different equipments as grate, superheater, air preheater, economizer etc. through which gas passes.
- Pressure losses in bends, baffles, supports etc.
- Kinetic energy required with gases for moving at certain velocity throughout.

Therefore while designing the chimney due considerations should be made for overcoming above losses. These losses constitute about 20% of the static draught produced.

***Hot gas velocity in chimney**:* Assuming chimney to be frictionless the hot gas velocity in chimney could be given using the equivalent hot gas column height;

$$C = \sqrt{2g \cdot h_g}\ ,\ \text{m/s}$$

For the chimney having friction losses, the hot gas velocity shall be lesser. If the equivalent pressure loss due to friction in hot gas column is given by h_f then the velocity of hot gases

$$C = \sqrt{2g \cdot (h_g - h_f)}\ ,\ \text{m/s}$$

$$= \sqrt{2g \cdot h_g\left(1 - \frac{h_f}{h_g}\right)}$$

Substituting values for 'g'

$$C = 4.43\ \sqrt{h_g\left(1 - \frac{h_f}{h_g}\right)}$$

or $$\boxed{C = K \cdot \sqrt{h_g}}$$

Where K is a constant and its value is available for different types of chimneys as given below. It depends upon the friction loss fraction.

K = 0.825 for brick chimney, and

K = 1.1 for steel chimney

Diameter of chimney: Diameter of chimney could be estimated from the mass flow rate of hot gases through chimney and its velocity.

$$\text{Chimney cross-sectional area, } A = \frac{\text{Discharge rate}}{\text{Velocity of hot gases} \times \text{Density of hot gas}}$$

$$\text{Diameter of chimney} = \sqrt{\frac{4}{\pi} \times \frac{M_g}{C \cdot \rho_g}}$$

Discharge through chimney: Mass flow rate of hot gases through chimney could be obtained as,

$$M_g = \text{Cross-sectional area} \times \text{Velocity of hot gas} \times \text{Density}$$

$$M_g = A \times C \times \rho_g$$

$$\boxed{M_g = A \times \rho_g \times K \cdot \sqrt{h_g}}$$

Discharge through chimney can be mathematically maximized for certain conditions. A look at expression of discharge given earlier shows that for a particular chimney.

$$M_g = \text{Constant} \cdot \left[\frac{H}{T_g^2} \cdot \left\{ \left(\frac{m}{m+1} \right) \cdot \frac{T_g}{T_a} - 1 \right\} \right]^{1/2}$$

Differentiating discharge with respect to hot gas temperature and equating it to zero for optimum condition,

$$\frac{d\,M_g}{d\,T_g} = 0$$

we get, $\boxed{\dfrac{T_g}{T_a} = \dfrac{2(m+1)}{m}}$

or $\quad T_g = \dfrac{2(m+1)}{m} \cdot T_a$

Upon substituting T_g value in $\dfrac{d^2 M_g}{d\,T_g^2} = 0$ we see that it is condition for maxima of M_g.

Thus discharge through chimney is maximum for the hot gas temperature given by

$$\boxed{T_g = 2\left(\frac{m+1}{m} \right) \cdot T_a}$$

For the condition of maximum discharge, draught can be obtained as,

$$\Delta p_{\text{for max. discharge}} = \rho_a \cdot T_0 \cdot H \cdot g \left\{\frac{1}{2\,T_a}\right\}$$

$$\boxed{\Delta p_{\text{for max. discharge}} = \frac{\rho_a \cdot T_0 \cdot H \cdot g}{2\,T_a}}$$

In terms of water column,

$$\boxed{h_{w,\text{ for max. discharge}} = \frac{\rho_a \cdot T_0 \cdot H}{2\,T_a}} \quad \text{mm of water column.}$$

Efficiency of chimney: It has been explained in natural draught that the pressure difference is created due to density difference caused by temperature difference. Hence it is obvious that the flue gases should leave at quite high temperature for creating required density difference. Thus the flue gases leave with sufficient heat energy, which could be used in boiler if some other mechanism is employed for exhaust of flue gases such as artificial draught. Therefore, efficient chimney should have such a design so that flue gases leave at lowest possible temperature. Efficiency of chimney is quantification of the cost of natural draught in terms of energy, i.e. the large amount of usable energy going along waste hot gases. Normally this efficiency of chimney is less than 1 percent.

Chimney efficiency is defined as the ratio of "energy with unit mass of gas in natural draught" and "the extra heat carried by same mass of gas due to high temperature in natural draught as compared to that in artificial draught".

$$\text{Chimney efficiency} = \frac{\text{Energy with unit mass of gas in natural draught}}{\text{Extra heat carried away in natural draught compared to artifical draught by unit mass of gas}}$$

$$\eta_{\text{chimney}} = \frac{9.81 \times H\left\{\left(\frac{m}{m+1}\right)\cdot\frac{T_g}{T_a} - 1\right\}}{C_{p,g}\left(T_g - T_{g,a}\right)}$$

Artificial Draught

Artificial draught refers to the externally created draught employing some equipment for it. Its requirement is felt, when the natural draught becomes insufficient for exhaust of flue gases. In general it is seen that for draught requirements being more than 40 mm of water, the natural draught does not work and becomes highly uneconomical. In the modern large power plants this draught produced by chimney is insufficient and requires some artificial method. Also the size of boiler units in use today forbid the use zof natural draught as the flue gas handling capacity is limited. In case of natural draught the fuel rate upto (20 kg/hr per m2 of grate area could be handled while with artificial draught it goes up to 300 kg/hr per m2 of grate area. Apart from these limitations the economy of using artificial draught over natural draught beyond a limit also make it attractive. For same steam generation the fuel consumption gets reduced by up to 15% with use of artificial draught in a boiler.

Artificial draught may be produced either by mechanical means such as fans, blowers etc. or by using steam jet for producing draught. Thus artificial draught can be classified as,

(i) Mechanical draught

(ii) Steam jet draught.

Artificial draught systems do not require tall chimney/stack, but small stack is always required for discharge of flue gases to certain height in atmosphere for minimizing pollution.

Mechanical Draught

Mechanical draught produced using fans, blowers etc. could be of forced type, induced type or the combination of the two. Line diagram showing the arrangements is shown in Fig. 11.37.

(i) *Forced draught:* It is the arrangement in which high pressure air is delivered to the furnace so as to force flue gases out through stack. Air under pressure may be fed to stokers or grate for which a fan/blower is put at the bottom of furnace. As due to pressurized air the pressure inside furnace becomes more than atmospheric pressure so it should be properly sealed, otherwise gas may leak through the cracks in setting into the boiler unit. Also the flames from furnace may flare out upon opening the fire door, so it should be equipped with dampers to shut off air supply when furnace doors are opened. It is obvious from here that the fan in case of forced draught shall handle fresh atmospheric air.

(ii) *Induced draught:* Induced draught is the one in which the suction created on furnace side draws flue gases and throws them out through small chimney/stack. Fan is located at base of chimney in induced draught so as to reduce pressure at fuel bed below atmospheric pressure. The fan in induced draught shall handle hot flue gases. Power required to drive the fan/blower in case of induced draught is less than that in case of forced draught fan.

Mathematically it can be given as below.

For volume of fluid handled being V (m^3/s) at pressure of p, the power required shall be,

$$= \frac{p \cdot V}{\eta_{\text{mech}}}$$

In case of *induced draught* fan, as hot flue gases are to be handled, so, total mass handled by fan for mf kg of fuel burnt per unit time = $mf + m * mf$

$$= mf\,(1 + m),\ \text{kg/s}$$

Therefore, volume handled by fan $= \dfrac{\text{Mass}}{\text{Density}}$

$$= \frac{m_f(1+m)\cdot m\cdot T_g}{\rho_a\cdot(1+m)\cdot T_0}$$

$$V_{\text{induced}} = \frac{m\cdot m_f\cdot T_g}{\rho_a\cdot T_0}$$

Power required $= \dfrac{p\cdot V_{\text{induced}}}{\eta_{\text{mech}}}$

$$\boxed{\text{Induced draught fan power requirement} = \frac{p\cdot m\cdot m_f\cdot T_g}{\rho_a\cdot T_0\cdot\eta_{\text{mech}}}}$$

In case of *forced draught* the fan handles air at atmospheric temperature so,
total mass of air handled $= m_f\cdot m$, kg/s

Volume of air at absolute zero temperature $= \dfrac{m_f\cdot m}{\rho_a}$

Volume of air at atmospheric temperature $= \dfrac{m_f\cdot m\cdot T_a}{\rho_a\cdot T_0}$ m^3/s

Power required in forced draught fan $= \dfrac{p\cdot m_f\cdot m\cdot T_a}{\rho_a\cdot T_0\cdot\eta_{\text{mech}}}$

$$\boxed{\text{Forced draught fan power requirement} = \frac{p\cdot m_f\cdot m\cdot T_a}{\rho_a\cdot T_0\cdot\eta_{\text{mech}}}}$$

Comparing the two power requirements,

$$\frac{\text{Power required in induced draught}}{\text{Power required in forced draught}} = \frac{T_g}{T_a} = \text{More than 1}$$

As $T_g > T_a$ so the power requirement in induced draught is more than that of forced draught.

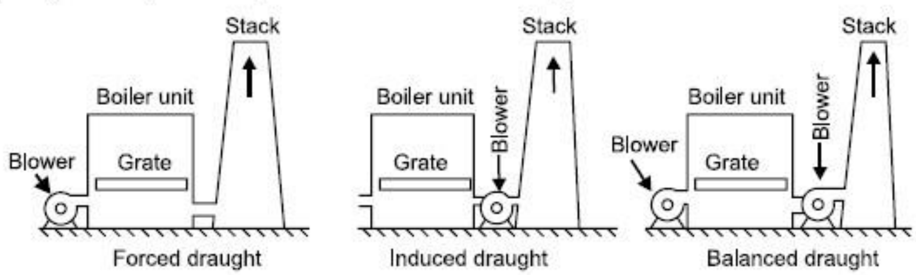


Fig. 11.37 *Mechanical draught*

Equivalent Evaporation

From earlier discussions it is seen that there exists a large variety of the boilers in terms of their arrangement, efficiency, steam generation rate, steam condition, type of fuel used, firing method and draught etc. For comparing one boiler with other any of the above parameters cannot be considered as they are interdependent. Therefore, for comparing the capacity of boilers working at different pressures, temperatures, and different final steam conditions etc. a parameter called "equivalent evaporation" can be used. Equivalent evaporation actually indicates the amount of heat added in the boiler for steam generation. Equivalent evaporation refers to the quantity of dry saturated steam generated per unit time from feed water at 100^0C to steam at 100^0C at the saturation pressure corresponding to 100^0C.

Sometimes it is also called equivalent evaporation from and at 100^0C. Thus, mathematically it could be given as,

$$\text{Equivalent evaporation} = \frac{\begin{array}{c}\text{Mass of steam generated per hour} \times \\ \left(\text{Heat supplied to generate steam in boiler}\right)\end{array}}{\begin{array}{c}\text{Heat supplied for steam generation at} \\ 100^\circ\text{C from water at } 100^\circ\text{C } \left(\text{i.e. Latent heat}\right)\end{array}}$$

Boiler Efficiency

Boiler efficiency quantifies how effectively the heat is being used in boiler. Thus it could be given by the ratio of heat actually used for steam generation and total heat available due to combustion of fuel in boiler.

$$\text{Boiler efficiency} = \frac{\text{Heat used in steam generation}}{\text{Total heat available due to fuel burning}}$$

$$= \frac{m\left(h - h_w\right)}{m_f \times \text{C.V.}}$$

Here m_f is the mass of fuel burnt per hour, C.V. is calorific value of fuel used (kcal/kg), m is mass of steam generated per hour and enthalpies h and hw are that of final steam and feed water, kcal/kg. Generally high heating value of fuel is used as calorific value of fuel.

STEAM CONDENSERS

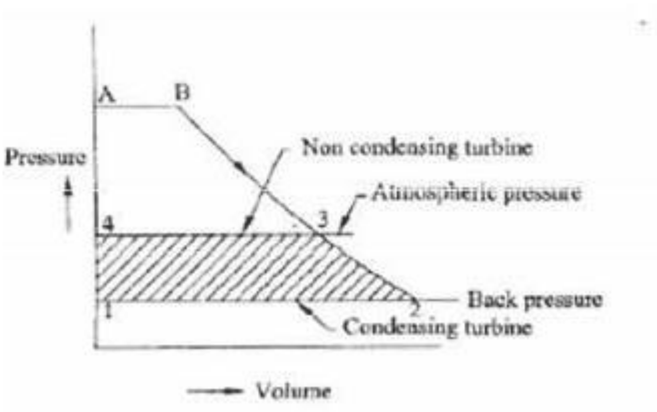
Condenser is one of the important elements of a steam power plant. It is placed at the exhaust end of a steam engine or steam turbine. A steam condenser is a closed vessel in which steam is exhausted from steam engine or steam turbine. The steam is cooled by-means of cooling water. The condensed steam formed together with air and other non-condensable gases is removed by pumps. The condensed steam is called condensate.

The exhaust steam leaving the prime mover (steam engine or steam turbine) still contains considerably large amount of heat. So, the exhaust steam is passed into a condenser in which heat exchange between steam and cooling water takes place.

The work done and efficiency of a turbine plant are increased if back or exhaust pressure of turbine is reduced. This is because the average temperature at which heat is rejected in the cycle is reduced.

There exists a relation between temperature and pressure. If the back or exhaust pressure of steam is low, the temperature is also low. Thus, by lowering the back pressure, the temperature at which heat is rejected is reduced and in turn work done and efficiency are increased. In order reduce the back pressure for increasing work done and efficiency, the steam should be exhausted in a closed vessel where it is condensed. The condensation of steam in a closed vessel enables expansion of steam to a lower back pressure (Temperature).

In a non condensing plant (plants which doesn't employ a condenser), the back pressure should be higher or equal to atmospheric pressure otherwise steam can't exhaust to atmosphere.



P-V Diagram of a Steam Turbine

Area A-5-3-4 shows the work done by a steam turbine which is not fitted with a condenser (Non condensing turbine). Area A B 2 1 shows work done by a condensing steam turbine. The shaded area 1-2-3-4 shows the increase in work done by fitting a condenser to a non condensing steam turbine.

The condensation of steam in a closed vessel produces partial vacuum as volume reduces greatly. I kg of dry steam at 1.013 bar and with a volume of 1.67 m^3, if condensed in a closed vessel, it will be condensed into water at a temperature of 100°C, liquid condensed will occupy only 0.001 m3 and pressure falls to about 0.2 bar. It means that the exhaust pressure can be lowered from 1.013 bar to 0.2 bar.

So, by fitting a condenser to a steam turbine or steam engine, the range of expansion of steam can be increased as the exhaust steam can be discharged at a pressure below atmospheric and hence work output from turbine can be increased.

Steam engines can't take advantage of very low vacuum as they are intermittent flow machines and they have to force the expanded steam out of cylinder through restricted exhaust ports and passages.

Steam turbines being continuous flow machines (steam flow takes place continuously from inlet to outlet without any obstruction) can take advantage of low vacuum because they have large exhaust outlets through which steam can be discharged after expansion.

There is a limit for reduction in back pressure beyond which it is not economical. This limit is based on the increased cost involved in the creation and maintenance of higher vacuum. In steam turbines, steam can be expanded upto 0.035 bar or even less depending upon the temperature of the cooling water and capacity of the plant.

The condensed steam (condensate) still contains considerable amount of heat and it can be reused as feed water to the boiler.

OBJECTS OF CONDENSER

A steam condenser has 2 objects:

1. The primary object is to maintain ve4y low back pressure on the exhaust side of steam turbine. This enables steam to expand to a greater extent by which maximum possible energy from steam can be converted into mechanical work.

2. The secondary object of steam condenser is to supply pure, hot feed water to the boiler. Thus, by fitting a condenser, the thermal efficiency of steam power plant can be greatly increased and also the capacity without increasing the size

ADVANTAGES OF CONDENSER IN STEAM POWER PLANT

The following advantages can be obtained by employing a condenser in a steam power plant:

1. It increases expansion ratio of available steam (increased available enthalpy drops) and thus increases efficiency of the plant.

2. It reduces back pressure of steam, thus more work can be obtained.

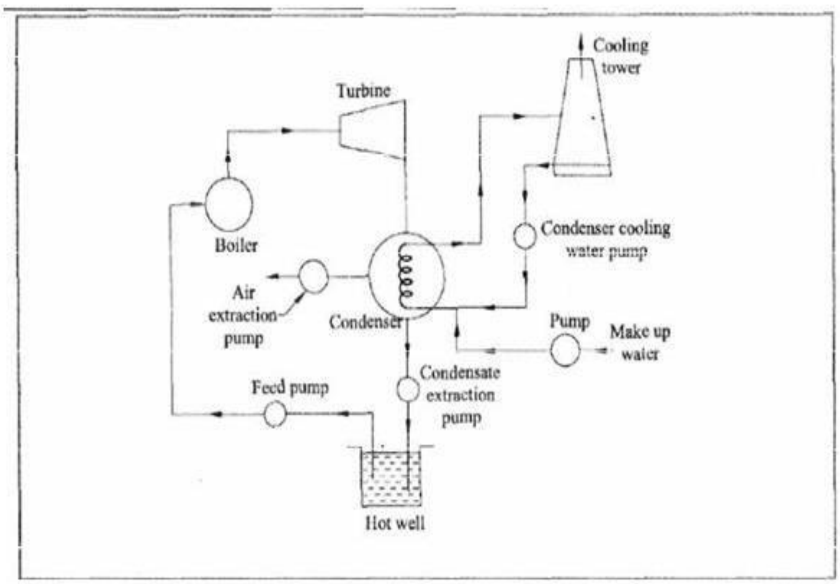
3. The condensate can be recovered and can be reused as feed water to the boiler. This reduces cost of power generation. Recovery of condensate is important in case of marine plants.

4. Recovery of condensate reduces make up water that must be added.

5. Supplying hot feed water reduces thermal stresses; steam consumption also can be reduced.

6. As condensate recovered is pure, the cost of water treatment is saved.

ELEMENTS OF A STEAM CONDENSING PLANT



A Steam Condensing Plant

The following are the important elements in a steam condensing plant:

1. Condenser.

2. Condensate pump

3. Hot well.

4. Boiler feed pump.

5. Air extraction pump.

6. Cooling tower.

Steam condenser is a closed vessel in which steam is condensed. Steam gives up heat energy to cooling water during the process of condensation.

A pump that removes condensed steam from the condenser and supplies to hot well is called -condensate extraction pump. Dry air pump is a pump for removing non condensable gases from the condenser. If a single pump is used for removing air and condensate, it is known as wet air pump.

Hot well is a reservoir for collecting condensate from a condenser. From the hot well, the water is fed to boiler. Boiler feed pump pumps condensate from hot well to boiler.

A cooling tower is an arrangement for re-cooling the cooling water of condenser. It is essential where there's scarcity of water.

Cooling Tower: The cooling water is placed at a certain height. The hot water falls down in radial sprays from a height and atmospheric air enters from base of tower. Partial evaporation of water takes place which reduces the temperature of circulating water. This cooled water is collected in the pond at the base of the tower a pumped back into the condenser.

CLASSIFICATION OF CONDENSERS

Depending upon the way of condensing the exhaust steam, steam condensers are of 2 types:

1. Jet condensers or mixing type condensers.

2. Surface condensers or non mixing type condensers.

In jet condensers, cooling water comes in direct contact with the exhaust steam. As the name implies, cooling water is sprayed into the exhaust steam in the form of a jet so that rapid condensation takes place. The temperature of cooling water and condensate is same when leaving the condenser.

With these condensers, the condensate can't be used as feed water to the boiler as it is not free from salts and impurities. Because of the loss of condensate and high power requirement for jet condenser pumps, these condensers are rarely used in modern power plants. These are employed where water of good quality is easily available in sufficient quantity.

In surface condensers, there is no direct contact between the steam to be condensed and the cooling water. Cooling water passes through number of tubes while exhaust steam passes over the outer surface of the tubes. Here, the temperature of condensate may be higher than cooling water at exit. Both cooling water and condensate are separately with drawn. The condensate is pure and can be used as feed water to the boiler.

This type of condenser is essential in ships and other marine applications which can carry limited quantity of fresh water for the boiler. Also widely used in land installations where impure water can be used for cooling or better quality of water for feed is to be used economically.

A jet condenser is much simplier and less costly than a surface condenser.

JET CONDENSERS

There are 3 classes of jet condensers:

1. Low level jet condensers (parallel flow and counter flow).

2. High level or Barometric jet condensers.

3. Ejector condenser.

In low level jet condensers, the condensing chamber is at low elevation and overall height of the unit is low enough so that condenser can be placed directly beneath the steam turbine.

Combined or separate pumps are required to extract cooling water, condensate and air from the condenser.

In high level jet condensers, the condensing chamber is placed at sufficiently high level to enable water to drain away by gravity. No water pump is required to remove condensate and cooling water but an air pump is required to remove air and other gases from condenser.

In ejector condensers, the steam and water mix in a series of combining cones and the kinetic energy of water is utilised in removing condensate and air from the condensers. No separate pump is required to remove condensate and air.

LOW LEVEL JET CONDENSERS

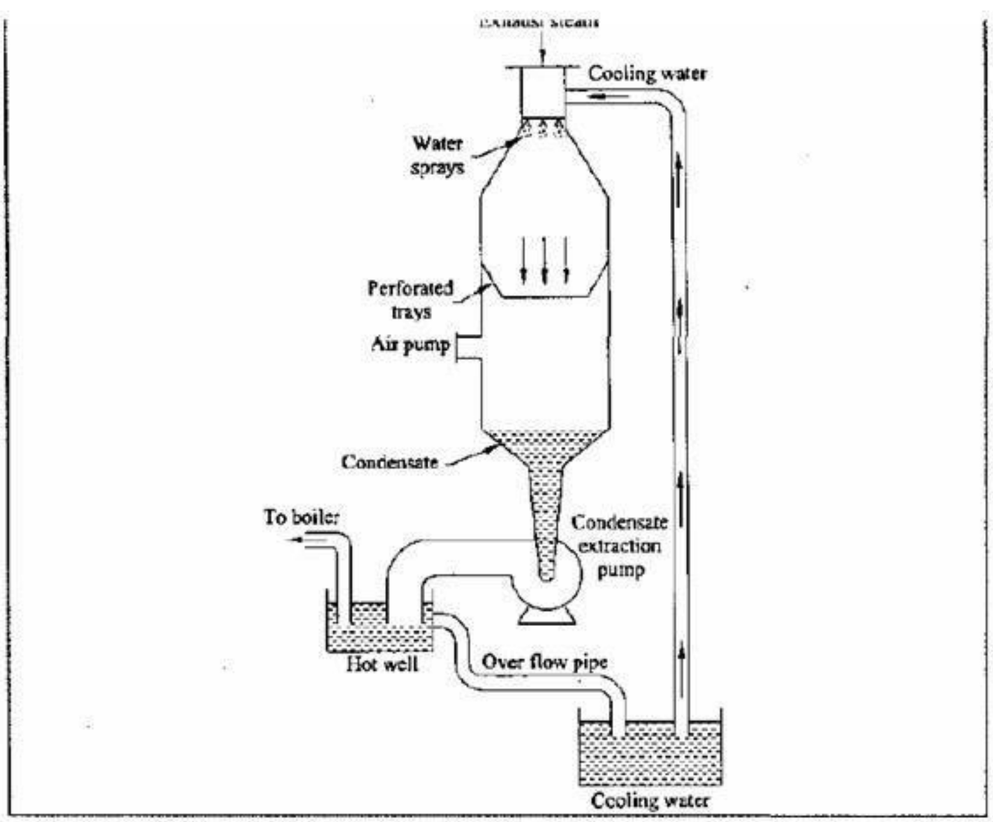
According to direction of flow of water and steam, low level jet condensers are sub classified as :

1. Parallel flow jet condensers.

2. Counter flow jet condensers.

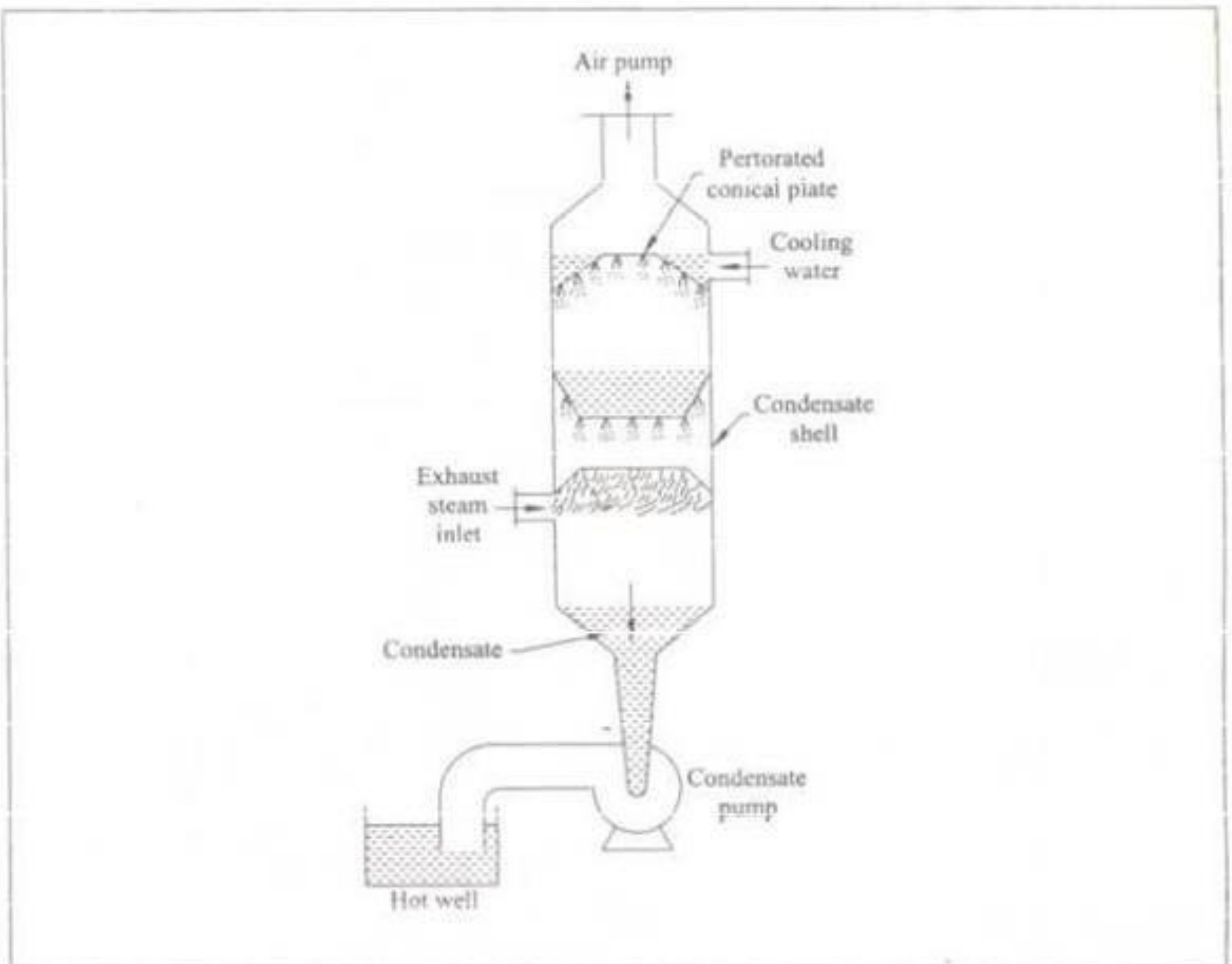
In parallel flow jet condensers, both steam and water enter at top and fall together to bottom where the mixture is removed by an extraction pump. This arrangement is best suited for turbine work where exhaust steam comes from underside of turbine.

In counter flow jet condensers, water and steam flow in opposite directions. Steam enters at bottom and flows upwards while water enters at top and falls downwards. Air extraction pump is at the top.



Parallel Flow Jet Condenser

The exhaust steam enters at top into the condenser. Cold water enters from top and is sprayed. The baffles or trays ensure proper mixing of steam and water. A condensate extraction pump discharges condensate to the hotwell. Surplus condensate from the hotwell gravitates to cooling pond. A separate dry pump may be incorporated to maintain proper vacuum.

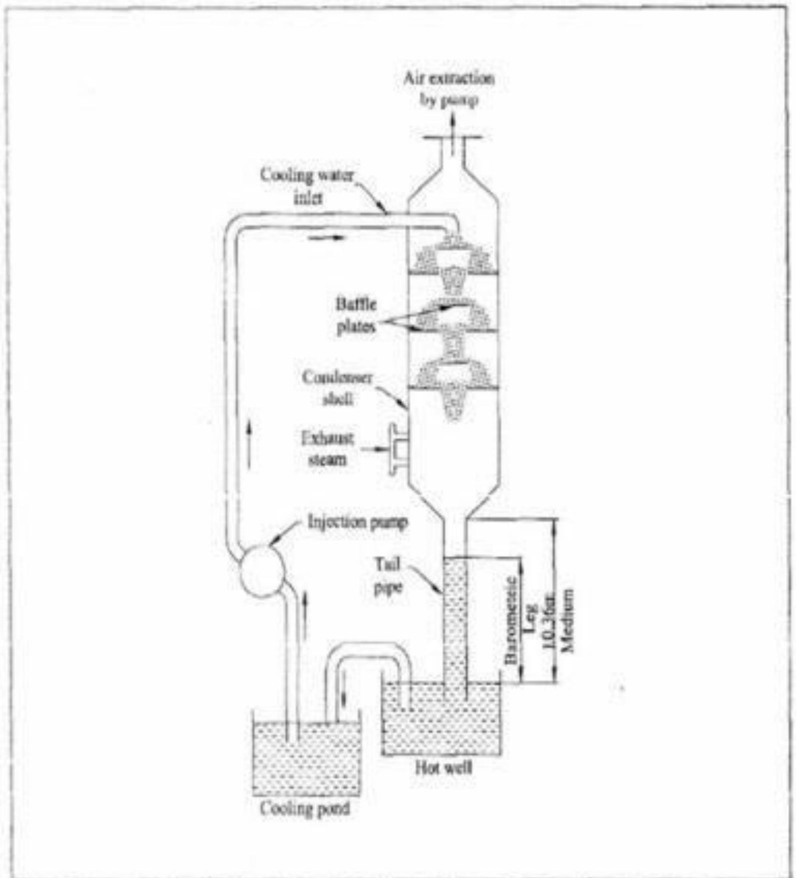


Counter Flow Jet Condenser

COUNTER FLOW JET CONDENSER

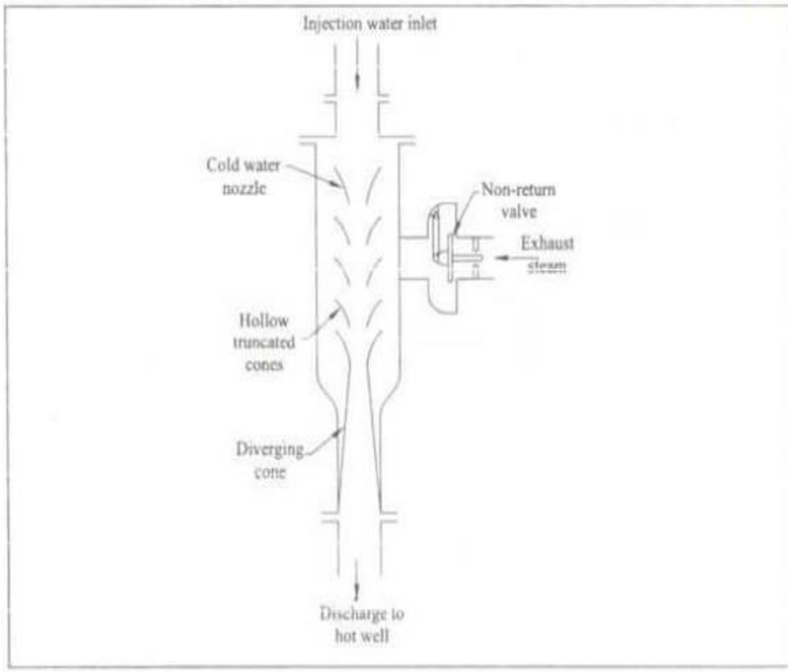
In these condensers, exhaust steam enters at the bottom, flows upwards and meets the down coming cooling water. Air pump is placed at the top of the condenser shell.

Vacuum is created by the air pump. This draws the supply of cooling water which falls in large number of jets through perforated conical plates. The falling water is caught in the trays from which it escapes into a second series of jets and meets the exhaust steam entering at the bottom. Rapid condensation occurs, condensate and cooling water descend through a vertical pipe to the condensate pump from which it is delivered to hot well.



HIGH LEVEL OR BAROMETRIC JET CONDENSER

It is so named because it is placed at a height greater than that of water barometer. If a long pipe over 10 metres was closed at one end filled with water, inverted without slipping any water and the open end is submerged in an open tank of water, the atmospheric pressure would hold water up in the pipe to a height of 10.368 m at sea level. This fact is made use of in a barometric condenser by making the discharge pipe more than 10 metres in height and thus making it impossible for any vacuum in the condenser to cause the water to rise high enough and flood the engine.



High Level Jet Condenser

Exhaust steam enters at bottom, flows upwards, and meets the down coming cooling water. Vacuum is created by air pump placed at the top of the condenser shell. The condensate and cooling water flow by gravity into hot well and thus, there is no need of condensate extraction pump. The surplus water from hot well flows to cooling pond through an overflow pipe. The shell is placed at a height greater than 10.36 metres - barometric height of water column.

EJECTOR CONDENSER

The principle of operation of this condenser is that the momentum of flowing water ejects throws out the condensate and air without the aid of a pump.

In these condensers, steam and water mix up while passing through a series of metal cones. The exhaust steam enters the condenser through a non return valve arrangement. Water enters at top and while flowing through the truncated cones its velocity increases and pressure drops. Due to the decreased pressure, the exhaust steam is drawn through the cones and finally lead to diverging cone.

In diverging cone, part of kinetic energy is transformed into pressure energy so that the condensate is discharged into hot well. So, the condenser acts as an air pump and as well as a condenser.

The non returns valve prevents sudden back rush of water into engine exhaust pipe in case of sudden failure of water supply to condenser.
An ejector condenser requires more cooling water than any other type of jet condensers.

SURFACE CONDENSERS

These may be subdivided into 2 types:

1. Condensers in which exhaust steam passes over a series of tubes through which cooling water flows.

2. Evaporative surface condenser in which steam passes through a series of tubes and cooling water flows in the form of a thin film outside the tubes. Surface condensers may also be classified as - 2 flow or multi flow condensers.

Depending upon the direction of flow of condensate, surface condensers are classified as:

1. Down flow surface condenser.

2. Central flow surface condenser.

3. Inverted flow surface condenser.

4. Regenerative surface condenser.

5. Evaporative surface condenser.

2 FLOW SURFACE CONDENSERS

It consists of a cast iron shell cylindrical in shape and closed at each end to form a water box. Numbers of water tubes are fixed to the tube plates. Exhaust steam enters at top and condensed by coming in contact with cold surface of tubes through which cooling water circulates.

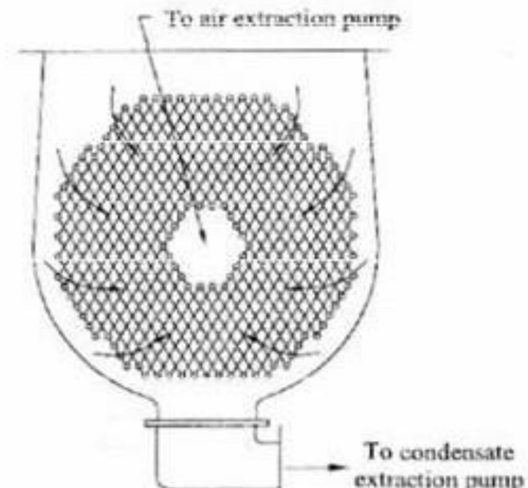
The cooling water enters at one end of tubes in one half of the condenser (one pass-one flow) and then enters into the tubes of second half of the condenser (second pass). Next, the water goes to outlet.

The resulting water from condensation of steam and air associated with uncondensed water vapour are extracted from the bottom of the condenser.

This type of condenser requires 2 pumps: Wet air pump to remove air and condensate, water circulating pump to circulate cooling water under pressure.

This is called 2 flow or 2 pass condenser because the cooling water circulates the whole length of condenser twice. By introducing more partitions in the water boxes, the same condenser may be converted into 3 flow or even 4 flow condenser. The rate of heat transmission increases with increase in number of flows but power required to circulate water also increases. As steam flows in a direction right angle to the direction of flow of water, it is also called as -Cross flow surface condenser.

DOWN FLOW SURFACE CONDENSER

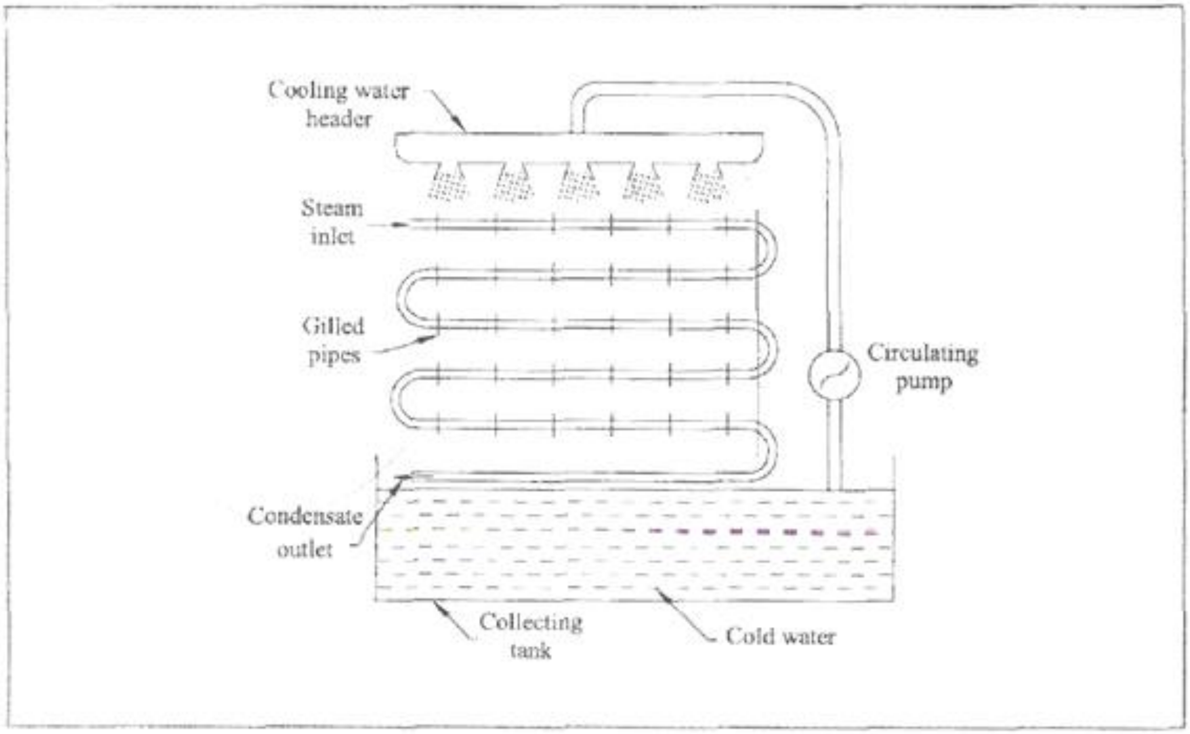


Down flow surface condenser

gravity as well as suction of extraction pump located at the bottom. The condensate is collected at the bottom and then pumped by the extraction pump.

A baffle is provided to cover the suction pipe of the dry air pump to prevent entry of condensed steam into it. As steam flows perpendicular to the direction of flow of water (inside the tubes), this is also called as - Cross flow surface condenser.

CENTRAL FLOW SURFACE CONDENSER



In this, the exhaust steam enters at the top of the condenser shell and flows downwards. The suction pipe of the air extraction pump is placed in the centre of the 'nest' of the tubes. This causes the steam to flow radially inwards towards the centre. The tubes carry circulating cooling water. The condensate is collected at the bottom and then pumped by the extraction pump.

This condenser is more efficient than down flow type as steam has access to the whole periphery of the tubes.

INVERTED FLOW SURFACE CONDENSER

This condenser has air extraction pump at the top. The exhaust steam enters at the bottom, rises up and then again flows down to the bottom of the condenser by flowing near outer surface of the condenser. Condensate extraction pump is located at the bottom.

REGENERATIVE SURFACE CONDENSER

In this condenser, the condensate is heated by a regenerative method. The condensate after leaving the tubes is passed through the exhaust steam from the steam engine or steam turbine. Thus, it raises the temperature of the condensate for use as feed water to the boiler.

EVAPORATIVE CONDENSER

It consists of sheets of gilled piping which is bent backwards and forward and placed in a vertical plane

The steam to be condensed enters at the top of a series of gilled pipes outside of which a film of cold water falls from a water header. At the same time, a current of air circulates over the water film causing rapid evaporation of some of the cooling water. As a result, the steam gets condensed. The water which is not evaporated falls into a collecting tank from which it is reused again. Its original temperature is restored by adding requisite quantity of cold water.

This condenser can run on minimum quantity of water and even without cooling water in cold weather and on light loads.

REQUIREMENTS OF A GOOD SURFACE CONDENSER

For a surface condenser to work effectively, the following requirements should be met:

1. There should be no leakage of air in the condenser. The pressure in the condenser also depends upon the amount of air. Owing to high vacuum pressure in the condenser, it is impossible to prevent air from leaking through the joints thereby increasing the pressure in the condenser and thus limiting the amount of work done by unit mass of steam in steam engine or steam turbine. Air leakage also results in lowering the partial pressure of steam and temperature.

This means that latent heat increases and so more cooling water is required which results in low overall efficiency.

2. The steam should enter the condenser with least possible resistance.

3. The drop in pressure of steam should be minimum (To extract more work) and steam should be well distributed in the vessel for effective condensation.

4. The circulating cooling water should flow through the tubes with least resistance and with a velocity consistent with high efficiency.

5. The condensate should be removed as quickly as possible at maximum practicable to obtain higher thermal efficiency.

6. There should be no under cooling of condensate. This can be achieved by regulating quantity of cooling water such that the temperature of exit water is equal to saturation temperature of steam.

7. Air should be removed from the condenser with minimum possible expenditure of energy.

COMPARISON OF JET AND SURFACE CONDENSERS

The following are the advantages and disadvantages of jet and surface condensers.

Advantages of Jet Condensers:

1. Intimate mixing -There is a thorough mixing of exhaust steam and cooling water. So, heat transfer is much better.

2. Smaller quantity of cooling water — Due to direct mixing of steam with cooling water, small quantity of cooling water is enough for condensation of given quantity of exhaust steam.

3. It requires less building space.

4. The equipment is simple and cost is less.

5. Maintenance cost is low.

6. Use of impure water is possible.

7. With barometric condenser, there is no danger of flooding.

8. Cooling water pump is not required for barometric condenser.

Disadvantages:

1. The condensate is wasted. Due to direct mixing of steam with cooling water (which may be not pure); condensate can't be used as feed water to the boiler and is wasted.

2. Failure of condensate pump would flood the engine.

3. As cooling water leaves the dissolved air, lower vacuum is achieved.

4. More power is required for air extraction pump.

5. It is less suitable for high capacity plants due to low vacuum efficiency.

6. The barometric condenser requires a long pipe which increases the cost of condenser and possibility of more air leakages. Also, it is difficult to accommodate.

Advantages of Surface Condensers:

1. Steam and cooling water are not mixed and hence condensate can be used again and again as boiler feed water.

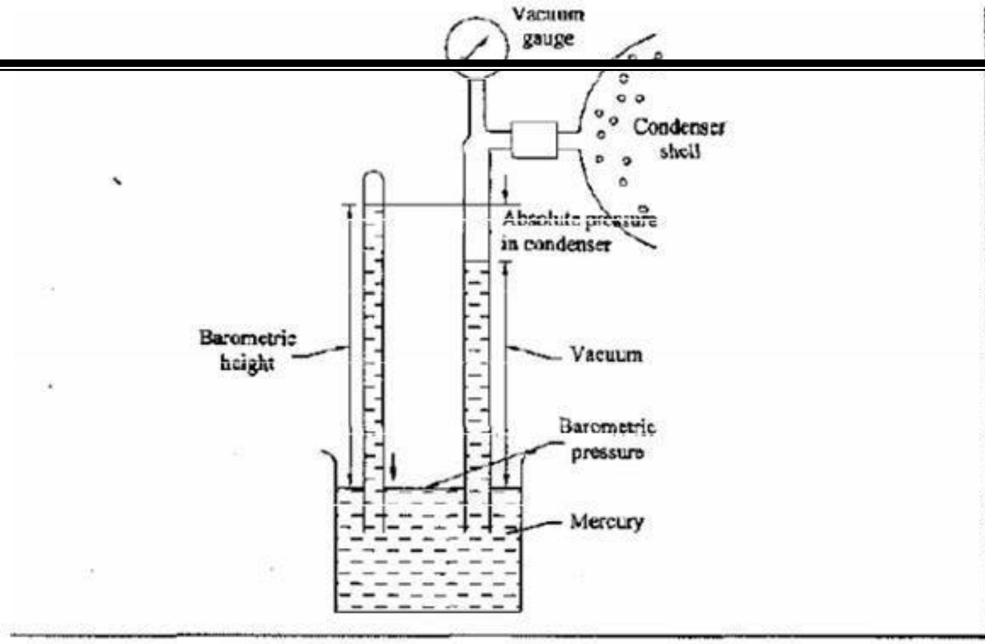
2. It requires less quantity of makeup water which saves the cost of feed water treatment.

3. High vacuum can be obtained which results in greater plant efficiency.

4. It is suitable for large capacity plants.

5. Any type of cooling water can be used.

6. Power required by air and cooling water pumps is much less. Disadvantages:



1. High initial cost.

2. Maintenance cost is more.

3. As water doesn't mix with steam, more cooling water is needed to affect perfect condensation.

4. It requires large floor space.

5. The plant is complicated.

MEASUREMENT OF VACUUM IN A CONDENSER

In case of condensers, vacuum means pressure below atmospheric. It is usually expressed in mm of mercury. The pressure in the condenser should be minimum possible to obtain maximum work from the steam engine or turbine. The vacuum that can be obtained

depends upon - Temperature of water, tightness of joints and valves, amount of air infiltration. This vacuum is not uniform throughout the condenser. It is least at the air pump suction, high in condenser body and still higher at engine exhaust valve.

Measurement of Vacuum in a Condenser

The vacuum depends upon the Barometric pressure(Atmospheric pressure) and absolute pressure in the condenser.

Barometric pressure is a variable quantity and it varies from place to place. For the purpose of comparison, it is convenient to refer vacuum gauge reading to a standard barometric pressure (Atmospheric pressure) of 760 mm of Mercury. Mathematically, vacuum gauge reading is 'corrected' to a standard barometer.

Standard or corrected vacuum = (760 mm of Hg - Absolute pressure in condenser - in mm of Hg) = [760 - (Barometer reading
- Vacuum gauge reading)] mm of Hg) 760 mm of Hg =1.013 bar

1 bar = $105 N/m^2$

1 mm of Hg = $133\ N/m^2$

= 0.00133 bar

The loss of vacuum in a condenser is mainly due to - air infiltration, reduced circulation of cooling water, accumulation of scale inside and outside the tubes and plugging of ejector jet.

METHODS TO OBTAIN MAXIMUM VACUUM

Following are some of the methods employed to obtain maximum possible vacuum in condensers:

1. Air Pump: Air pumps are provided to maintain desired vacuum in the condenser by extracting air and other non condensable gases. There are 2 types of pumps: Dry air pump which removes air only and wet air pump - which removes a mixture of condensate and non condensable gases.

2. Steam Air Ejector: When a wet air pump is used, then steam air ejectors are employed to remove air from the mixture. The operation consists in utilising the viscous drag of a high velocity steam jet for the ejection of air and other non condensable gases from the chamber.

3. De-aerated Water: If dissolved air is removed from feed water, then it is called - Deaerated water. The de-aerated water helps in maintaining better vacuum and controls corrosion of boiler shell and piping of theplant.

4. Air Tight Joints: Various joints in the plant should be made air tight to avoid aiinfiltration. DALT0NS

LAW 0F PARTIAL PRESSURES

This law is very helpful for analytical treatment of problems dealing with a mixture of gases or of gas and vapour.

This law states that - "In a mixture of perfect gases that don't react chemically with one another, total pressure exerted by the mixture is the sum of partial pressures which each gas would exert if separately occupy the whole volume at the temperature of the mixture".

Let there is a mixture of air and steam in condenser. Let t =

Temperature of the mixture

Pa = Partial pressure of air at temperature t

Ps = Partial pressure of steam (water vapour) at temperature P = Total pressure in the condenser

Then, according to Dalton's law, $P = P_a + P_s$

So, the total pressure is the sum of partial pressures of steam and non condensable gases. The non condensable gases lie in dissolved form in the water and get separated on heating the water. The non condensable gases chiefly consist of CO_2 and air. Amount of CO_2 is extremely small compared to air and can be neglected.

The Dalton's law also states that - "Each constituent of the mixture in the container occupies the whole volume of the container and exerts its own partial pressure in the container".

Let V = Volume of container - m3

ma = Mass of air in the condenser - Kg ms= Mass of steam in the condenser -Kg

Va= Specific volume of air at temperature t - m3/kg

Vs = Specific volume of steam at temperature t - m3/kg According to Dalton's law, $V = m_a.v_a = m_s.v_s$

$$\therefore \frac{m_a}{m_s} = \frac{v_s}{v_a}$$

$$\text{Mass of air}/m^3 \text{ of container} = \frac{m_a}{V} = \frac{1}{v_a}$$

$$\text{Mass of steam}/m^3 \text{ of container} = \frac{m_s}{V} = \frac{1}{v_s}.$$

Total mass of mixture in the container =

m=ma+ ms

$$= m_s\left(1 + \frac{m_a}{m_s}\right)$$

$$= m_s\left(1+\frac{v_s}{v_a}\right) \qquad \text{(or)}$$

$$m = m_a\left(1+\frac{v_a}{v_s}\right)$$

VACUUM EFFICIENCY

In steam condensers, we have a mixture of air and steam. If no air is present in the condenser, then total absolute pressure would be equal to partial pressure of steam and maximum vacuum would be obtained in the condenser.

The ratio of actual vacuum obtained at the steam inlet to the condenser to the maximum vacuum (Ideal vacuum) which could be obtained in a perfect condensing plant (No air is present) is called - Vacuum efficiency.

$$\therefore \qquad \text{Vacuum efficiency} = \frac{\text{Actual Vacuum}}{\text{Ideal Vacuum}}$$

Actual vacuum = Barometric pressure - Absolute pressure in condenser. (Actual pressure).

Ideal vacuum = Barometric pressure - Absolute pressure of steam corresponding to temperature of condensation. (Ideal pressure). So, vacuum efficiency is a measure of the degree of perfection to maintain desired vacuum in the condenser. Always, there will be some amount of air present in the condenser due to leakage and dissolved air present in the steam. So, vacuum efficiency depends upon the quantity of air removed from the condenser by the air pump. Generally, the vacuum efficiency is about 98%.

The performance of a condenser is given by the term - Condenser efficiency. It is the ratio of actual temperature rise of cooling water to maximum possible rise.

Condenser efficiency is defined as the ratio of difference between outlet and inlet temperatures of cooling water to difference between saturation temperature corresponding to absolute pressure in the condenser and inlet temperature of cooling water

$$\therefore \quad \text{Condenser efficiency} = \frac{\text{Rise in temperature of cooling water}}{\begin{array}{c}\text{saturation temperature Corresponding}\\ \text{to absolute pressure in the condenser}\end{array} - \begin{array}{c}\text{Inlet temperature of}\\ \text{cooling water}\end{array}}$$

The condenser efficiency generally varies from 75% to 85%. QUANTITY OF

CIRCULATING COOLING WATER REQUIRED

The function of circulating cooling water in a condenser is to absorb heat from steam and thereby to condense it. In surface condenser, the temperature of condensate and exit water is not the same while in jet condenser, it is the same. To design a condenser, it is essential to calculate the quantity of cooling water necessary for a certain capacity of steam to be condensed.

Determination the amount of cooling water is a problem of simple heat exchange. Heat lost by steam = Heat gained by cooling water

Heat lost by steam = ms . (H_T - H_c) Where m_s = Mass of condensate kgs/hr

HT = Total heat of steam entering the condenser kJ/kg HC= Total heat in the condensate kJ/kg

Heat gained by cooling water = m_w . c_w . (t_0 - t_i)

Where mw= Mass of cooling water required - kgs/hr c_w = Specific heat of cooling water kJ/kg

t_i = Inlet temperature of cooling water. t_0 = Outlet temperature of cooling water.

Although the steam is supplied in superheated state to steam engine or steam turbine, usually, it is wet when enters the condenser.

Heat lost by wet steam = Latent heat + Sensible heat due to cooling of condensate due to air which leaks into condenser

where

x = Dryness fraction of steam entering the condenser.

L = Latent heat of steam entering the condenser.

t_s = Saturation temperature of exhaust steam corresponding to condenser vacuum.

t_c = Temperature of condensate leaving the condenser.

Heat gained by cooling water = $m_w \cdot c_w \, (t_o - t_i)$

$$\therefore \quad m_s\left[x \cdot L + c_w\,(t_s - t_c)\right] = m_w \cdot c_w \cdot (t_o - t_i)$$

$$\therefore \quad m_w = \frac{m_s\,[x \cdot L + c_w\,(t_s - t_c)]}{c_w \cdot (t_o - t_i)} \; kgs/hr$$

In case of Jet condenser, $t_c = t_o$.

SOURCES OF AIR IN A CONDENSER

The performance of a condenser is adversely affected by the presence of air in the condenser. Following are the main sources through which air may enter into condenser:

1. The dissolved air in the feed water enters boiler which in turn enters condenser with the exhaust steam. The amount of air coming in depends upon the treatment of feed water.

2. Air leaks from atmosphere through various joints which are internally under less pressure than atmosphere. The amount of air depends upon accurate workmanship, care in design and making of joints.

3. Leakage through condenser accessories like atmospheric relief valve etc.

4. In case of jet condenser, some air comes in with injection water - cooling water in which it is dissolved.

EFFECTS OF AIR IN A CONDENSER

The following are the important effects of presence of air in a condenser:

1. With increased amount of air, condenser pressure or back pressure increases. This reduces the useful work done.

2. Presence of air lowers partial pressure of steam and so lowers saturation temperature of steam. With the lowering saturation temperature, latent heat of steam increases and so more cooling water is required.

3. Air is a poor conductor of heat and so reduces rate of heat transmission. So, surface area of the tubes has to be increased for a given condenser capacity.

The presence of air reduces the rate of condensation of steam since abstraction of heat by cooling water is partly from steam and partly from air.

5. The air extraction pump is required to remove air only but some quantity of steam escapes with air. This reduces the amount of condensate. Also, the condensate is under cooled with the result that more heat has to be supplied to the feed water in the boiler.

6. Larger the amount of air present in the condenser, capacity of air pump increases and greater is the corrosive action by the air. Corrosive action is roughly proportional to the concentration of oxygen present in the condenser.

It is most important to check all air leakages and to remove any air that may be in the condenser. In practice, it is impossible to remove all the air. So, it is continuously removed by air pump which sucks air from condenser, compresses it to a little above atmospheric pressure so that it is forced out.

AIR PUMPS

It is essential to extract the air present in the condenser as it adversely affects the performance of a condenser. Air extraction pumps remove air and other non condensable gases from the condenser. The primary function of an air pump is to maintain vacuum in the condenser as nearly as possible corresponding to exhaust steam temperature. This is done by removing air and other non condensable gases from the condenser.

There are 2 types of air pumps:

1. Dry air pump which removes air only.

2. Wet air pump which removes air and non condensable gases along with condensate. Air pumps may be classified as:

1. Reciprocating piston or bucket pumps.

2. Rotary pumps.

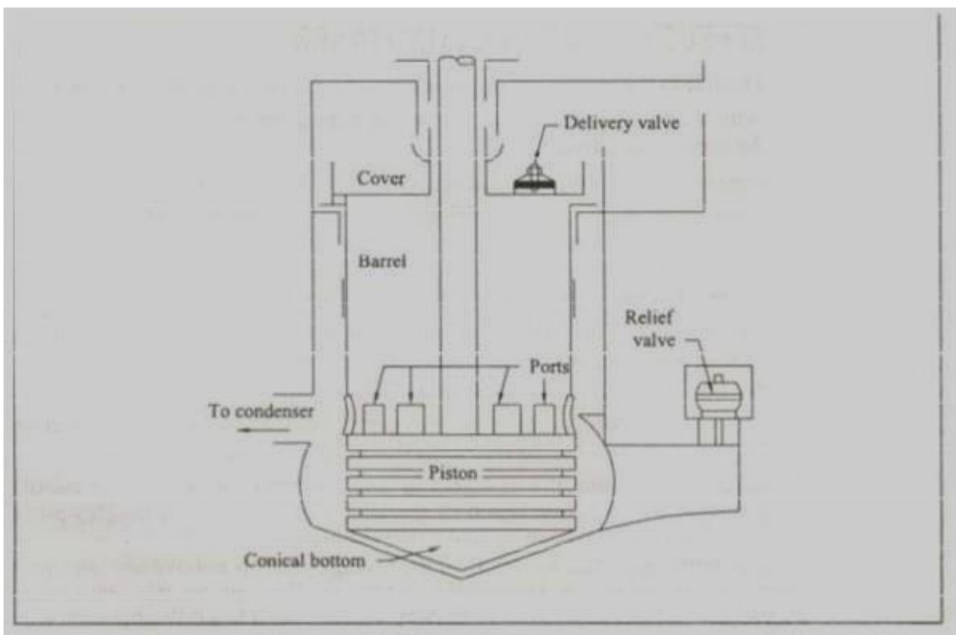
3. Steam jet air pumps.

4. Wet jet pumps.

Here, we discuss about one type of reciprocating piston or bucket pump only

EDWARD'S AIR PUMP

This is a commonly used reciprocating piston wet air pump. The fig shows an Edward's air pump.



The delivery valve is placed in the cover which is on the top of the pump barrel lever. The reciprocating piston is flat on its upper surface and conical at bottom. The pump lever has a ring of ports around its lower end for whole circumference. This communicates with the condenser.

When the piston is at the top of the barrel, the condensate and air from the condenser is collected in the conical portion of lower part of barrel through the ports. On the downward stroke of the piston, vacuum is produced above it since the delivery valves are closed and sealed by water. The piston uncovers the ports. When it moves downwards, the mixture rushes into space above the piston. This mixture is compressed when the piston goes to top and raises the pressure slightly above atmospheric pressure. The delivery valves are now open which allow mixture to pass on the top of the cover. Condensate flows to hot well which is at atmospheric pressure. A relief valve is placed to release the pressure. This pump is most suitable for condensing vapour in land and marine steam plants.

UNIT-II

STEAM POWER PLANT

It is a combination of several components or devices whose objective is to convert heat energy of fuel into mechanical work and then to obtain electric power.

To have heat energy, we need certain source of energy. This source of heat energy is called Fuel. Energy remains locked in the fuel. By burning the fuel, the chemical energy of fuel gets converted into thermal or heat energy. This burning of fuel is known as -combustion. The place where combustion of fuel takes place is called Boiler.

We can't convert heat energy directly into work. There must be a medium which takes up heat energy released by combustion of fuel and acts on another device to produce mechanical work. In steam power plant, steam that comes from water is used as working medium as water is cheaply available in large quantity, safe and good conductor of heat.

Water absorbs heat from combustion of fuel, changes its phase and turns into steam. In that process, it absorbs high amount of heat - latent heat of evaporation.

With continuous absorption of heat, steam changes its state - wet steam to dry saturated steam and then to superheated steam and the pressure of steam increases.

After attaining required pressure, the high-pressure steam is allowed to flow through a passage called nozzle over f ring of moving blades attached to a shaft. The unit which houses the shaft with blades (number of blades mounted circumferentially over a shaft) is called -Turbine.

The high-pressure steam while flowing through the nozzle increases its kinetic energy and then expands over the blades of turbine and in doing so, imparts rotary motion to the blades. In this way, heat energy of fuel is converting into mechanical work.

Now, the low pressure, expanded steam from turbine goes to atmosphere. This exhaust steam from turbine still contains enough heat and can be used for different purposes - for process work, expansion in low-pressure turbines or for preheating feed water to the boiler.

To increase the work done from the turbine, a unit called condenser is placed after the turbine. With the inclusion of condenser, steam in turbine can be expanded to a greater extent and we can get more work from the turbine.

In condenser, the exhaust steam from condenser is cooled and then recirculated as feed water to the boiler. These are the important components in a steam power plant. To increase the efficiency of the plant and to have satisfactory running of the plant, other components are also included in the plant.

The fig. 2.1 shows the phases of energy transformation that take place in a steam power plant.

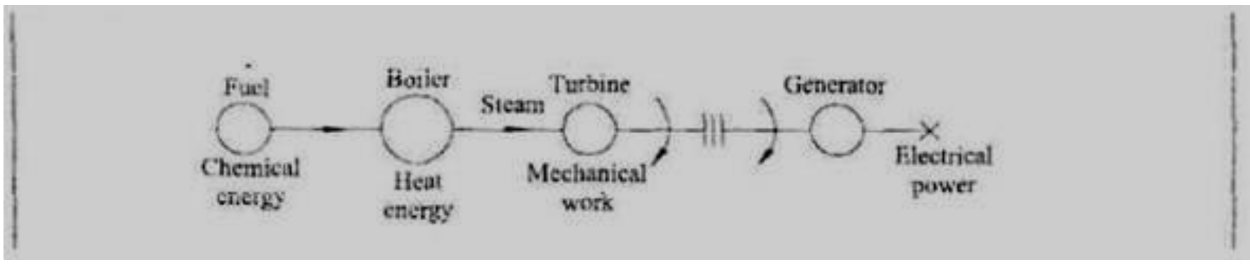


FIGURE 2.1 : PHASES OF ENERGY TRANSFORMATION IN A STEAM POWER PLANT

LAYOUT OF A STEAM POWER PLANT

The purpose of a steam power plant is to generate electric power only or to generate steam for industrial purpose besides generation of electric power. Steam is extensively used in industries like Textile, Sugar, Paper, Refinery, and Chemical etc.

BOILER

It is a closed vessel inside which combustion of fuel takes place. Tubes are arranged in the walls of the boiler through which water passes. The water by absorbing heat of combustion turns into steam.

TURBINE

Steam from super heater passes through nozzles and enters into turbine. The high-pressure steam now expands over the blades of the turbine rotor (shaft upon which the circumferential blades are mounted). The pressure of steam drops down along with its enthalpy (total heat content). This drop in heat energy (enthalpy) is converted into mechanical energy. As a result, the shaft of the turbine rotates.

CONDENSER

This is located after the turbine so that steam after expansion in the turbine exhausts into the condenser. The exhaust steam from the turbine enters the condenser and major portion of it gets condensed. The condensed steam is called condensate and is recirculated as feed water to the boiler. The steam that remains in the condenser is used to preheat the feed water to the boiler, with the addition of a condenser, we can extract more work from the turbine.

FEED PUMP

It is a pump which takes up purified, preheated water and then forces into the boiler with pressure. As boiler works at higher pressure, feed pumps are necessary to raise the pressure of water for its entry to boiler.

VAPOUR POWER CYCLES

A common method of producing mechanical work is by transfer of heat from a heat reservoir at a high temperature to a working fluid which undergoes through a thermodynamic cycle.

A heat engine cycle or thermodynamic cycle is a combination of thermodynamic processes through which the working fluid passes in a certain sequence. The cycle begins with one set of conditions (pressure, volume, temperature etc.), undergoes different changes in different processes and while doing so, converts part of the heat energy into mechanical work and rejects the remaining heat to a low temperature reservoir called - sink and finally comes back to original state - attains initialconditions.

Any machine designed to carry out a thermodynamic cycle and converts heat energy into mechanical work is called a - heat engine. The thermodynamic cycle upon which it operates is known as - heat engine cycle or power cycle.

The power cycles may use vapour or gas as the working fluid. Cycle which use vapour as the working substance are known as vapour power cycles. The most commonly used vapour is -steam. In thermodynamic cycles, the transfer of heat from high temperature reservoir (also called -source) to working fluid and from working fluid to sink will be irreversible. But, the processes of working fluid itself may be reversible. The process is internally reversible. The cycles composed of reversible processes are called - ideal cycles.

In vapour power cycles, change of phase of working fluid takes place while in gas power cycles, the working fluid remains in one phase throughout the cycle. In vapour power cycles, water is universally used as a working fluid as it is easy to change its phase, ease of handling and its chemical stability.

The vapour power cycle consists of a series of steady flow processes, each process carried out in a separate component designed for that purpose. Each component in the cycle / plant constitutes an open system and as the working fluid passes through each component, it passes through a cycle of mechanical and thermodynamic states.

In the analysis of the cycles, all the processes are assumed to be reversible.
RANKINE CYCLE

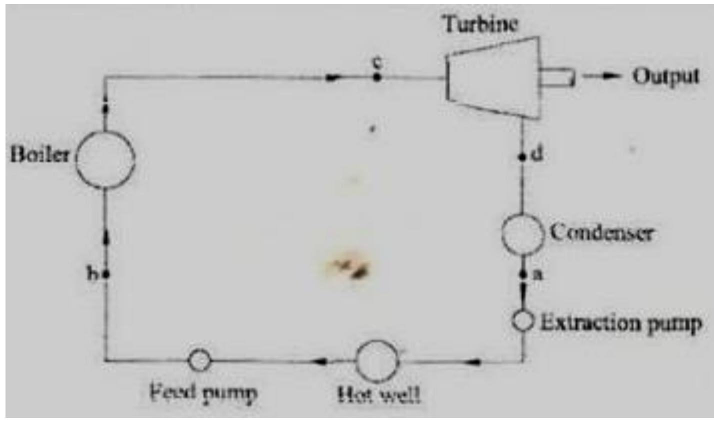
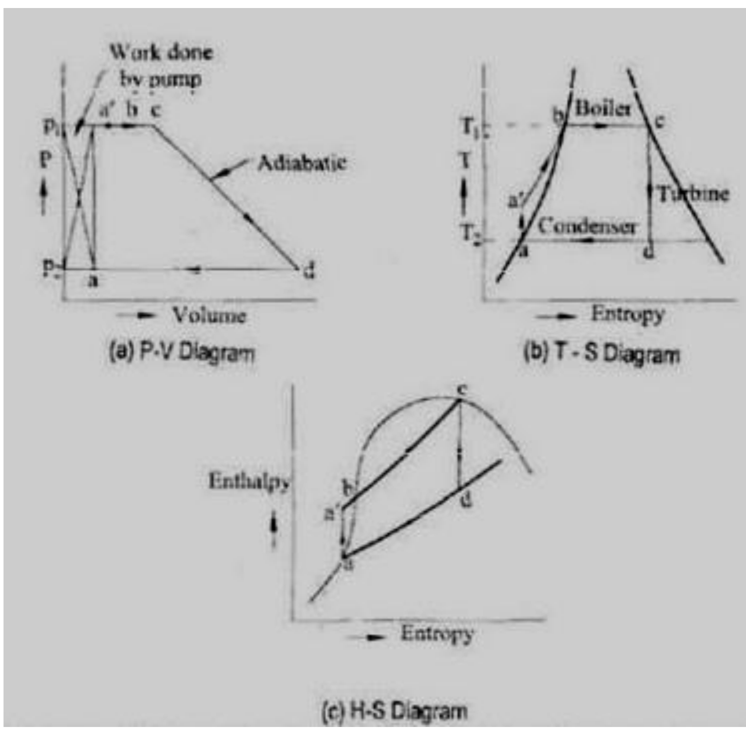


FIGURE 2.3 COMPONENTS IN A RANKINE CYCLE



The fig. 2.4 shows P-V, T-S and H-S diagrams of a Rankine cycle which uses steam as its working fluid.

A Rankine cycle is a basic cycle for a simple steam power plant. It is a theoretical cycle upon which steam engine or steam turbine works. It is a modified form of Carnot cycle and an ideal cycle for comparing the performance of steam power plants."

Processes of Rankine Cycle:

Process a-a’: The point 'a' represents water at initial pressure p1 and corresponding saturation temperature T2. The water is pumped into the boiler by feed pump raising its pressure to boiler pressure p1. The pumping process is isentropic compression. During this process, the temperature rises slightly.

Process a' -b : As water enters the boiler, water is first heated upto evaporation temperature at constant pressure px. The temperature increases to T1. Sensible heat is supplied during this process.

Process b - c: Water evaporates completely at temperature T1 and constant pressure p1.During this process, the heat supplied is latent heat of vaporization. The final condition (Point C) may be wet, dry or super heated depending upon the quantity of heat supplied.

Process c - d: Steam expands isentropically in the turbine from P1, T1 to p2, T2 and does the work.

Process d-a: The exhaust steam from the turbine at constant pressure p2 and temperature T2 is condensed in a condenser where latent heat of steam is removed. The process is isothermal compression.

At point a; the working fluid restores its original conditions - returns to its original state. Thus the cycle gets completed. In a steam power plant, supply of heat and rejection of heat are more easily performed at constant pressure than at constant temperature. In the operation of the cycle, the work done in pumping feed water to the boiler is very small at low pressures and is usually neglected. The fig. 1.5 shows the P-V and T-S diagrams for the Rankine cycle, neglecting pumping work.

ASSUMPTIONS IN THE WORKING OF RANKINE CYCLE

The following assumptions are made in the working of Rankine cycle:

1. The same working fluid is repeatedly circulated in a closed circuit;

2. Heat is added in boiler only and rejected in condenser only. Except boiler and condenser, there is no heat transfer between working fluid and surroundings.

3. There is no pressure drop in the piping system.

4. Expansion in the prime mover occurs without friction or heat transfer i.e., expansion is isentropic in which case entropy of working fluid entering and leaving the prime mover is same.

5. The working fluid is not under cooled in the condenser i.e., the temperature of water

leaving the condenser is same as saturation temperature corresponding to the exhaust pressure.

EFFICIENCY OF RANKINE CYCLE

Refer the fig. 1.4 , Let

H1 = Enthalpy of 1kg of steam at pressure p1 at entrance to the prime movers at c, c' or c''.
H2 = Enthalpy of 1 kg of steam at pressure p2 as its leaves the prime mover at d, d' or d''.

h2 - Enthalpy of 1kg of water at pressure p2 as it enters the feed pump at a work done by the prime mover

$$\eta_{Rankine} = \frac{\text{Work done}/kg \text{ of steam}}{\text{Heat supplied}/kg \text{ of steam}} = \frac{(H_1 - H_2)}{(H_1 - h_2)}$$

(H1 - H2) is known as isentropic enthalpy drop or Rankine heat drop.

VARIABLES AFFECTING EFFICIENCY OF RANKINE CYCLE
The important thermodynamic variables in a Rankine cycle are :

1. Steam pressure at inlet to turbine.

2. Steam temperature or degree of superheat at inlet to turbine.

3. Steam pressure at exhaust or condenser pressure. (Also called - Back pressure).

At higher pressure, heat rejection is less and so thermal efficiency increases. With increase in maximum pressure average temperature of heat addition increases and so, thermal efficiency increases. But, increase in maximum pressure increases the wetness of the vapour after expansion which decreases the adiabatic efficiency of the prime mover and causes erosion of blades.

By using superheated steam at the entrance to the turbine, the cycle efficiency increases. Also, superheating reduces specific steam consumption as work done per unit mass of steam is greater. For same condenser pressure, with superheated steam, dryness fraction at exhaust increases, or for same value of dryness fraction, work done increases. To avoid erosion of blades by water droplets, minimum dryness fraction at turbine exhaust should be 0.88.

The thermal efficiency of Rankine cycle can be greatly improved by reducing condenser pressure. By reducing condenser pressure, a large amount of heat drop is available as work done. But, by reducing back pressure, wetness of steam increases. So, it can be concluded that the cycle efficiency increases with increase in maximum pressure (upto a certain limit in case of un-superheated cycle), with increase in initial temperature and with decrease in back pressure.

METHODS OF INCREASING THERMAL EFFICIENCY

The thermal efficiency of Rankine cycle may be increased by :

1. Increasing the average temperature at which heat is added.

2. Decreasing the average temperature at which heat is rejected.

Based on these principles, methods of increasing thermal efficiency are :

(a) Increasing inlet pressure of steam to turbine.

(b) Using superheated steam to turbine.

(c) Reducing the condenser pressure.

(d) By reheating the steam.

(e) By regenerative feed heating.

REHEATING OF STEAM

It is mentioned that efficiency of ordinary Rankine cycle can be improved by increasing the pressure of steam at inlet to the turbine. With increased pressure, the steam will expand to a greater extent and becomes quite wet at the end of expansion. The wet steam contains suspended water particles. These water particles are heavier than steam particles cause erosion of the blades and increase internal losses. Ultimately, it will lead to reduction in blade efficiency of the turbine.

In order to increase the life of the turbine blades, it is necessary to keep steam dry during expansion. This is done by taking out steam from the turbine at a section where it becomes just dry saturated and is reheated at constant pressure by flue gases until it is again superheated to about the same temperature as on entry to the turbine. It is then taken to next stage of turbine where further expansion takes place. This process is known as - Reheating. Generally, the expansion is carried out in several' stages and steam is reheated between stages at suitable points. The final dryness fraction should not be less than 0.88 in a steam turbine.

To certain extent, the process of reheating causes increase in work done. But, this increase in work done is at the cost of additional heat supplied in reheating the steam and so, there will be no appreciable change in efficiency.

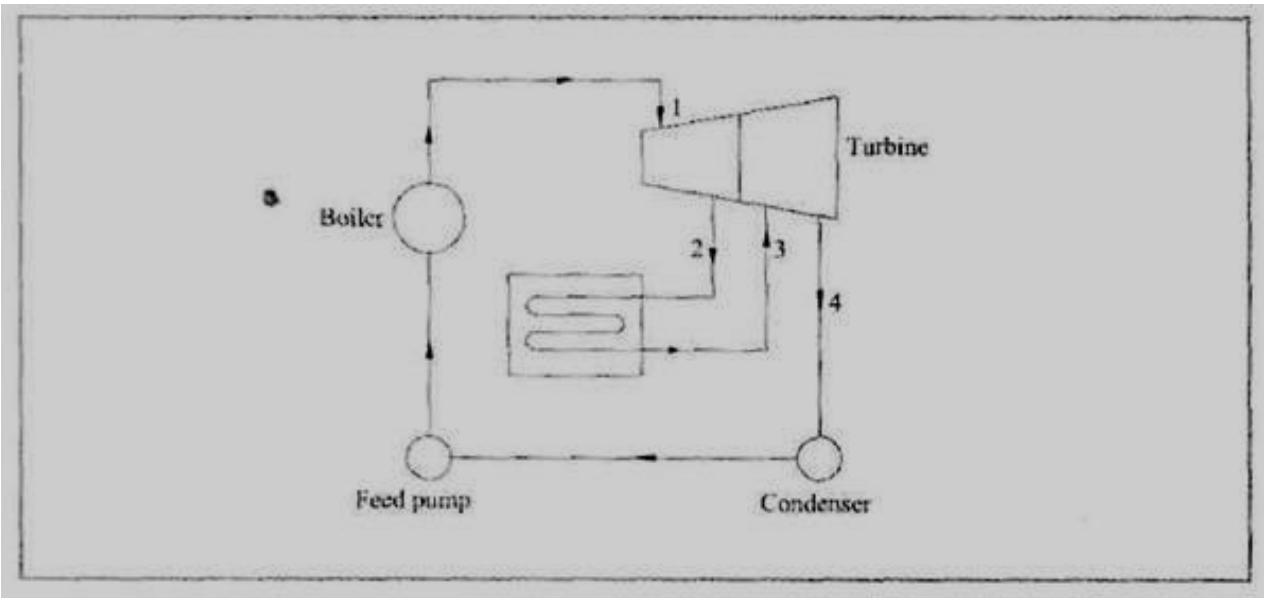


figure 2.7: reheating of steam

The main purpose of reheating is to avoid wet condition in the turbine thereby avoiding erosion of blades and frictional losses both of which reduce nozzle and blade efficiency. Reheating is generally employed when pressures are high (Above 100 bar), at one point. For still higher pressures, reheating may be carried out twice. Reheating has become essential for supercritical boilers.

The improvement in thermal efficiency due to reheating of steam is dependent to a large extent upon the reheat pressure with respect to original pressure of steam.

The reheater may be incorporated in the walls of the boiler or it may be a separately fired super heater or heated by high-pressure superheated steam.

Reheating should be done at proper pressure for economy. If steam is reheated early in its expansion, additional quantity of heat supplied will be less and so, gain in thermal efficiency will be less. If it is carried out late, though large amount of additional heat is supplied, much of it will be discarded in the condenser.

Advantages of Reheating Steam:

Reheating of steam in a turbine has the following advantages:

1. It increases output of the turbine.

2. Erosion and corrosion problems are avoided.

3. The thermal efficiency increases.
4. Nozzle and blade efficiencies increase.

Disadvantages:

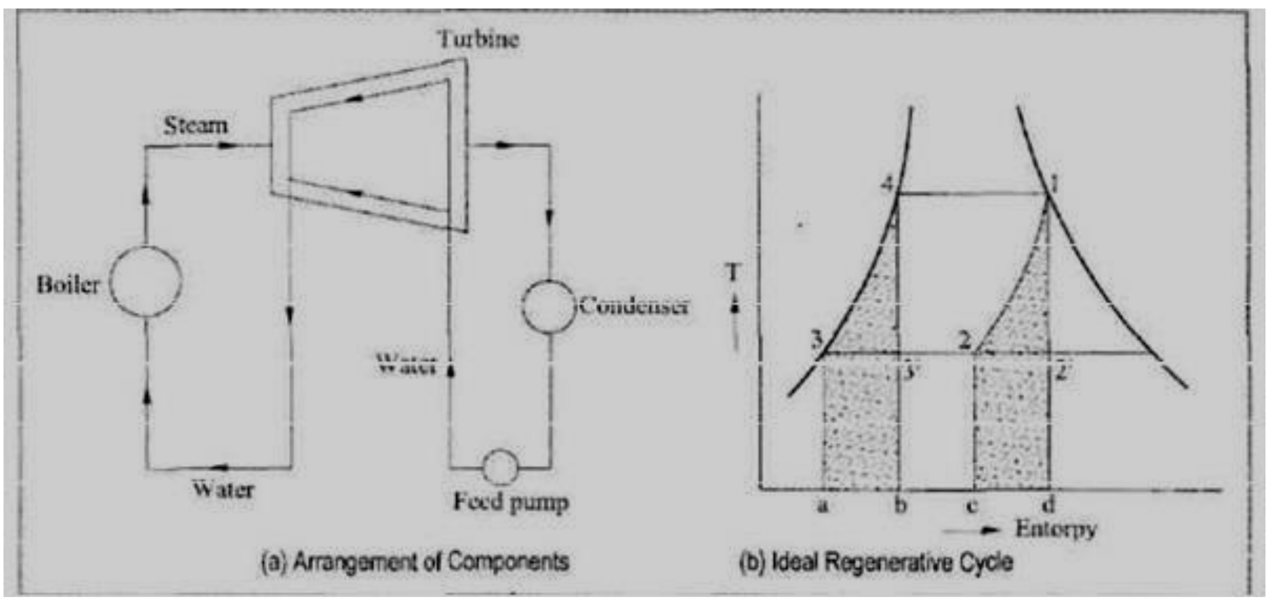
1. Maintenance is more.

2. Relative to cost of reheating, increase in thermal efficiency is not appreciable. Efficiency of reheat

cycle= work done/heat supplied

REGENERATIVE CYCLE

In this, dry saturated steam from boiler enters the turbine at a higher temperature Tl and then expands to temperature T2- Now, the condensate from condenser is pumped back and circulated around turbine casing in a direction opposite to that of expanding steam in the turbine.



The fig. 2.9 shows the ideal regenerative cycle.

Thus, the steam is heated before entering into the boiler, such a system of heating is known as -regenerative heating as steam is used to heat the steam itself. At all points, the temperature difference is infinitesimal between water and steam and so the process is reversible. Due to loss of heat the expansion in the turbine is not isentropic but follows the path 1-2. The heat gained by feed water during 3-4 (Area 34ba) is equal to heat gives by steam during 1-2 (Area ldc2). The heat supplied from external source is equal to 4\db and heat rejected to external sink is equal to 2ca3 (= 2' db3'). Areas of heat supplied and heat rejected have the same value as that of Carnot cycle. So, the efficiency of this cycle is equal to that of Carnot cycle for same temperature limits.

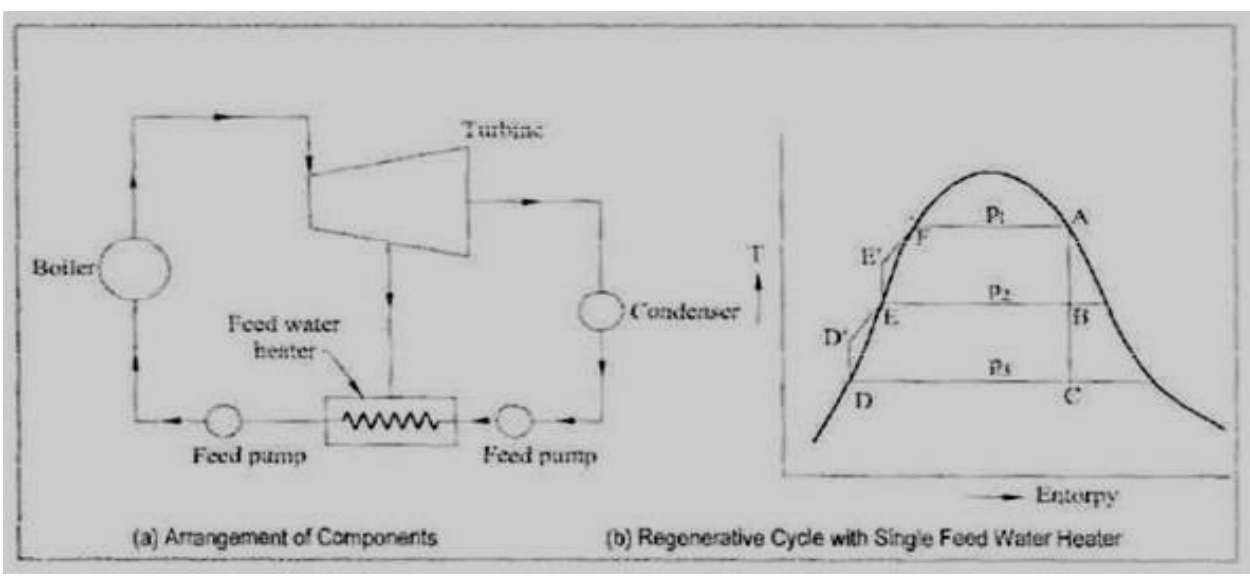
Compared to Rankine cycle, the advantage in regenerative cycle is rejection of less amount of heat in the condenser.

The ideal regenerative cycle can't be obtained because:

1. It is not practicable to design a turbine which acts as a heat exchanger as well as expansion device. So, necessary heat transfer is not possible.

2. The dryness fraction of expanding steam will be very low. So, in actual practice, advantage of principle of regeneration is taken by bleeding a part of steam at certain stages of expansion so that the dryness fraction of remaining part is not greatly reduced. The resulting cycle is known as - regenerative or bleeding cycle. The process of draining steam from turbine at certain locations during its expansion and using this steam for heating feed water (in feed water heaters) supplied to the boiler is called Bleeding and the process of heating is called - regenerative feed heating. The corresponding steam is said to be bled.

The result of this process is to supply hot feed water to the boiler. This increases the efficiency of the plant but there is a loss of small amount of work done by the turbine. This cycle is not an ideal cycle as mixing in feed water heater is irreversible. By employing more number of heaters, mixing becomes reversible and efficiency can be increased but correspondingly, cost also increases.



Regenerative Cycle with Single Feed Water Heater

Consider 1kg of steam (at pressure P1) enters the turbine at point A. During its expansion, at some suitable location B (at pressure p2), m kg of steam is bled off from the turbine and taken to a feed water heater. The remaining steam (1 - m) kg is expanded further in the turbine to condenser pressure and leaves the turbine at C. The exhaust steam from turbine goes to a condenser and after condensation to D; (1 - m) kg of water (condensate) is compressed in the feed pump to the bleeding pressure p2.
It is then mixed with m kg of bled steam in the feed water

heater and 1 kg of mixture leaves at E. This water is compressed by second feed pump to boiler pressure p1.

Let H1 = enthalpy of steam entering the turbine.

H2 = enthalpy of bled steam.

H3 = enthalpy of steam leaving the turbine.

h2 = sensible heat of head water leaving the heater. h3 = sensible heat of steam leaving the

condenser. m = amount of steam bled / kg of steam supplied. We know that,

Heat lost by bled steam = Heat gained by feed water :: m(H2-h2) = (1 -m) (h 2 -h 3)

$$m = \frac{(h_2 - h_3)}{(H_2 - h_3)}$$

For 1 kg of steam at entrance to turbine

Work done in turbine during A - B = (H1 - H2)

Mass of steam between B - C = (1 - m) kg

Work done in turbine during B - C = (1 - m) (H2 - H3) Total work done = (H1- H2) + (1 - m)

(H2 - H3)

Total heat supplied per kg of feed water = (H1 - h2) Efficiency of regenerative cycle=

$$\eta_{regen} = \frac{\text{work done}}{\text{heat supplied}}$$

$$= \frac{(H_1 - H_2) + (1 - m)(H_2 - H_3)}{(H_1 - h_2)}$$

If there is no regenerative feed heating, then efficiency of the cycle will be same as that of Rankine cycle. The efficiency of regenerative cycle is greater than that of Rankine cycle and the cycle efficiency is a maximum when the temperature of bled steam is approximately equal to mean of boiler and condenser temperatures.

Advantages:

The main advantages of bleeding are:

1. It increases efficiency as heat of bled steam is not lost in the condenser but utilized in feed water heating which increases the average temperature at which heat is added.

2. Due to bleeding, volume flow rate is reduced and due to this, dimensions of turbine blades can be reduced. Also, the size of condenser can be reduced.

3. Due to higher temperature of feed water, thermal stresses in the boiler are minimized.

Disadvantages:

1. For given output higher capacity boiler is required.

2. With more heaters, maintenance is more and cost is also more.

STEAM NOZZLES

In steam turbines, the overall transformation of heat energy of steam into mechanical work takes place in two stages. The available energy of steam is first converted into kinetic energy and then this kinetic energy is transformed into mechanical work. The first step is accomplished with devices called steam nozzles.

A steam nozzle is a duct or passage of smoothly varying cross sectional area which converts heat energy of steam into kinetic energy. The shape of nozzle is designed such that it will perform this conversion of energy with minimum loss.

When steam flows through a nozzle, expansion of steam takes place. During this expansion, the pressure of steam decreases and also the heat content (Enthalpy). With the expenditure of enthalpy, the velocity and specific volume increase. Also, with the expansion of steam, there will be condensation of steam with varying dryness fraction.

The mass of steam passing through any section of nozzle remains constant. So, the variation of pressure and the cross section of nozzle depend upon the velocity, specific volume and dryness fraction of steam. The velocity increases continuously from entrance to exit of the nozzle.

The cross section of the nozzles may be circular, rectangular, elliptical or square. The smallest section in the nozzle is known as throat. The nozzles are used in steam and gas turbines, jet engines, for propulsion of rocket motors, flow measurements, in injectors for pumping water, in ejectors for removing air from condensers etc. The major function ofnozzles is to produce a jet of steam or gas with high velocity to drive steam or gas turbines. So, the nozzles are located just before the steam or gas turbines. When the nozzles velocity gas is produced and there will be no question of condensation and hence dryness fraction.

When the nozzles are used with steam turbines, they perform the following functions.

1. They convert part of heat energy of steam (obtained from boiler) into kinetic energy.

2. In case of impulse turbines (details of steam turbines are given in the chapter - steam turbines), the nozzles direct the jet of high velocity steam against the blades of rotor which then convert the kinetic energy of steam into mechanical (shaft) work.

In case of reaction turbines, the nozzles discharge high velocity steam on to the rotor blades. The reactive force of steam against the nozzle produces motion ofrotor and work is obtained. When a fluid is decelerated in a duct or passage (velocity decreases) causing a rise in pressure during the travel along the stream, then the duct or passage is known as - Diffuser. Diffusers are extensively used in centrifugal, axial flow compressors, ramjets and combustion chambers etc.

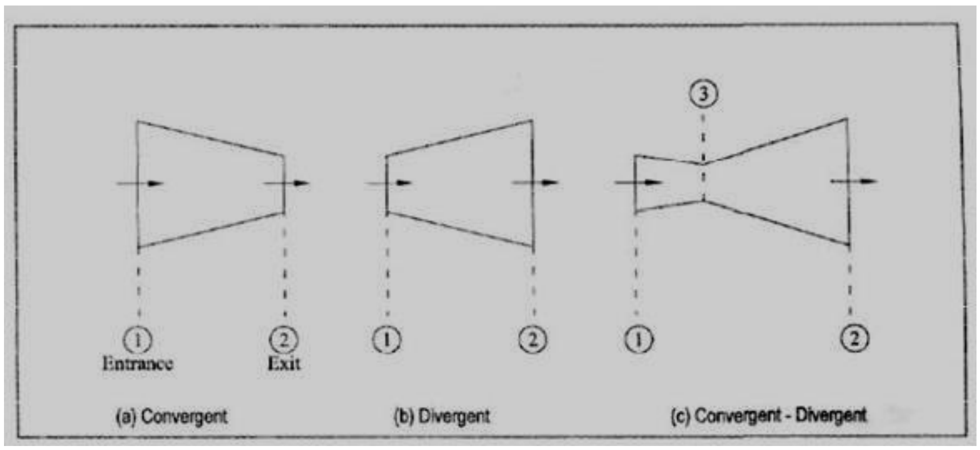
TYPES OF STEAM NOZZLES

There are three important types of steam nozzles:

1. Convergent nozzle.

2. Divergent nozzle.

3. Convergent - divergent nozzle.



Types of Steam Nozzles

If the cross section of the nozzle decreases continuously from entrance to exit; then it is called convergent nozzle.

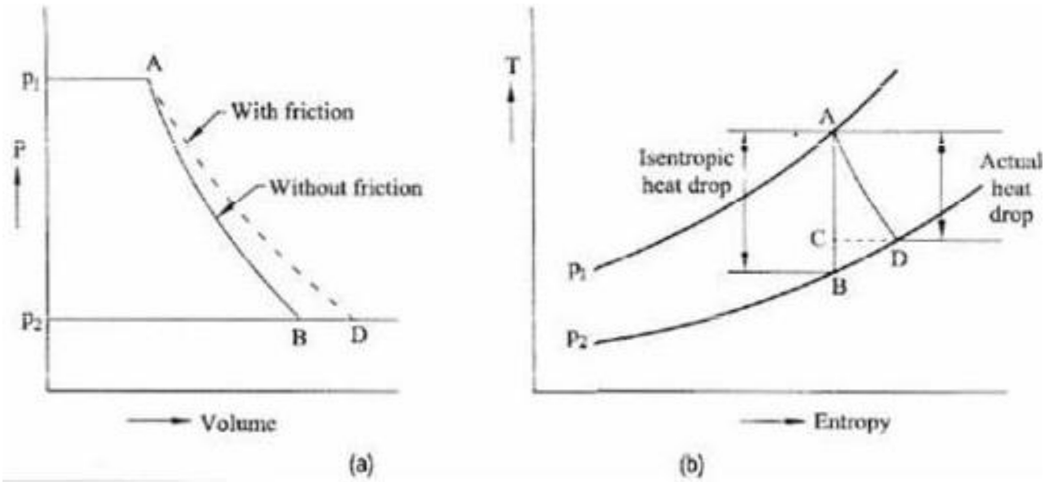
If the cross section of a nozzle increases continuously from entrance to exit then it is called Divergent nozzle.

If the cross section of a nozzle decreases first up to certain length and then increases up to exit; then it is called - Convergent - Divergent nozzle. This is used mostly in various types of steam turbines.

The three types of the nozzle are shown in above figure.

The flow of steam through nozzles may be regarded as adiabatic expansion because in nozzles, the velocity of steam is so high that there will be no time available for heat exchange with surroundings and so heat is neither supplied nor rejected. However, work is performed by increasing the kinetic energy of steam. Also, in a nozzle, the change of potential energy is negligible and no work is done on or by the fluid.

The expansion of steam in a nozzle is not a free expansion and the steam is not throttled because it has a very high velocity at the end of expansion and the pressure as well as enthalpy decrease as expansion takes place. The pressure at which steam leaves the nozzle is known as Back pressure.



In actual practice, always some friction is produced between steam and the walls of the nozzle; this friction causes resistance for the flow of steam, which is converted into heat. This heat tends to dry the steam. So, for the design of a nozzle, the effect of friction has to be considered. There is a phenomenon known as - super saturation that occurs in the flow of steam through nozzles. This is due to time lag in the condensation of steam during expansion. This super saturated flow affects mass and condition of the steam discharged. So, the flow of steam through a nozzle may be regarded as either:

1. Reversible adiabatic or isentropic flow.

2. Adiabatic flow modified by friction.

3. Super saturated flow.

Expansion of Steam

The point in the nozzle where area is minimum is called throat and the pressure at the throat is called - critical pressure. At this section; the mass flow per unit area is maximum.

The velocity of fluid at the throat of a nozzle operating at its designed pressure ratio (when the flow rate is maximum) is equal to velocity of sound, and it is called - Sonic velocity. The flow up to throat is sub sonic and the flow after throat is supersonic (greater than velocity of sound).As we know, the velocity increases continuously in a nozzle from inlet to exit. After throat, the fluid velocity becomes greater than sonic velocity and to accelerate flow; the area must increase or the nozzle must diverge resulting in diverging portion of nozzle. The ratio of fluid velocity to local sound velocity is known as - Machnumber.

Convergent - Divergent Nozzle

A convergent nozzle is used if exit pressure is equal to or more than the critical pressure and convergent - divergent nozzle is used if exit pressure is less than the critical pressure.

As already mentioned, the velocity of steam at inlet to a nozzle is very small compared to exit velocity. Low velocity implies large inlet area and most nozzles are shaped in such as way that the inlet area is large and converges rapidly tothroat area.

Note: A ventruimeter which is used for flow measurement of fluids is also convergent divergent in shape. But, in it, there is no continuous rise or fall of pressure. So, it is neither a complete nozzle nor a diffuser. In its convergent portion, the pressure is decreasing, velocity is rising and this portion acts as a sub sonic nozzle. In the divergent portion, pressure is rising, velocity is falling and this portion acts as subsonic diffuser. The pressure at throat may not necessarily imply sonic velocity.

The ratio of critical pressure to initial pressure is called - critical pressure ratio (p2/p1)- At the throat, the pressure is critical (velocity of fluid equals to sound velocity), area is minimum and mass flow per unit area is maximum.

With liquids, convergent - divergent shape is never used because the sonic velocity in liquids is very high (About 1500 m/sec compare to about 330 m/sec in air) which is out of the limit of practical velocities used.

STEADY FLOW ENERGY EQUATION

Consider steady flow of 1kg of steam through a nozzle.

Let

P1 and p2= Pressures at inlet and exit - bar.

V1 and V2 = Velocities at inlet and exit - m/sec

Vs1 and Vs2 = Specific volumes at inlet and exit - m3/kg u1 and u2 = Internal energy at inlet and exit - KJ/kg

Z1 and Z2 = Elevation at inlet and exit - m

h1 and h2 = Enthalpy at inlet and exit - KJ/kg

q = Heat supplied if any - KJ/kg

w = Work done if any - KJ/kg

For a steady flow process (without any accumulation of the fluid between inlet and exit), by the principle of conservation of energy;

Energy at entrance or inlet = Energy at exit.

Work done in forcing 1kg of steam into nozzle+ initial internal energy + initial kinetic energy

+ initial potential energy + heat supplied if any from the surroundings= work done in sending out 1 kg of steam from nozzle+ final internal energy + final kinetic energy + final potential energy + work done if any to the surroundings.

$P_1V_1+u_1+V_1^2/2+gz_1+q=P_2V_2+u_2+V_2^2/2+gz_2+w$ $P_1 V_1 + u_1 = h_1$ = Enthalpy of stem at inlet $P_2 V_2 + u_2 = h_2$ = Enthalpy of stem at exit

Generally, changes in potential energy are negligible.

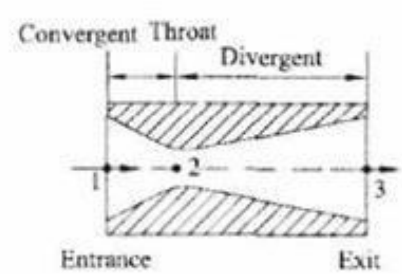
$$z_1 = z_2$$

If no heat is supplied from surroundings; then $q = 0$. If no work is done to the surroundings, then $w = 0$.

$$\therefore \quad h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

This is the steady flow energy equation of a nozzle. In this equation; the effect of friction is not considered.

Flow of Steam Through a Convergent - Divergent Nozzle



In the converging portion 1-2 (From inlet to throat), there is a drop in steam pressure with a rise in its velocity. Also, there is a drop in the enthalpy of the steam. This drop of enthalpy is not utilized in doing external work but converted into kinetic energy. In the divergent portion2-3.

(From throat to exit), there is further drop of steam pressure with a further rise in its velocity.

Again, there is a drop in the enthalpy which is converted into kinetic energy. Now, at the outlet, steam leaves the nozzles with high velocity and low pressure.

EFFECT OF FRICTION IN A NOZZLE; NOZZLE EFFICIENCY

When steam flows through a nozzle, for a given pressure drop, the final velocity of steam gets reduced because of the following losses :

1. The friction between steam and walls of nozzle.

2. Internal friction of steam itself.

3. Shock losses.

Most of the friction in a convergent divergent nozzle occurs in the divergent portion - between throat and exit. Due to the effect of friction, the actual flow through a nozzle is not isentropic but still approximately adiabatic. The effects of friction are :

1. The enthalpy drop is reduced and hence the final velocity.

2. The kinetic energy gets converted into heat due to friction and is absorbed by the steam. Due to this, the final dryness fraction ofsteam increases.

3. Steam becomes more dry due to increased dryness fraction and hence specific volume of steam increases and mass flow rate decreases.

The effect of friction is shown on the h-s diagram or Mother chart in fig.3.8.

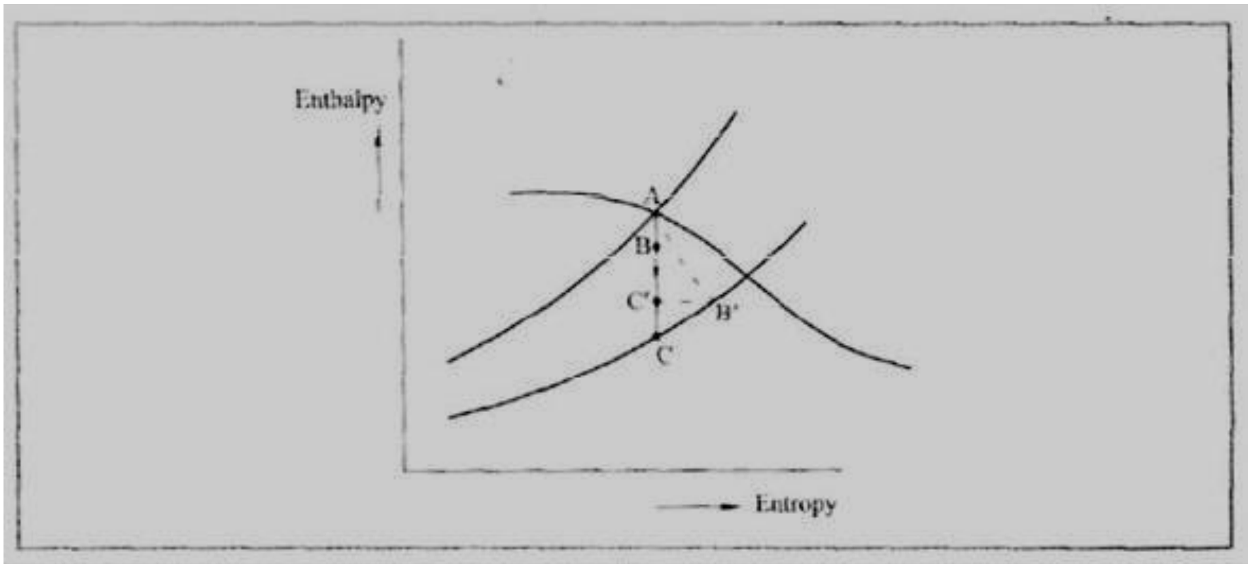


FIGURE 2.8: EFFECT Of FRICTION IN A NOZZLE

Point A represents the initial condition of steam which enters the nozzle in a dry saturated state. If the effect of friction is neglected, the expansion of steam from entrance to throat is represented by A-B and that from throat to exit by B-C. The whole expansion from A to C is isentropic. The heat drop (hA - hc) is known as - Isentropic heat drop or Rankine heat drop. In actual practice, the expansion process is modified by friction. Let point B' represent the final condition of steam. Now, AB' represents the actual expansion - Adiabatic expansion. Dryness fraction at B' is more than at C. So, the effect of friction is to improve the quality of the steam.

The heat drop (hA - hB ') is the actual enthalpy drop during the expansion of steam when effect of friction is considered and is known as - useful heat drop. The useful heat drop is less than the isentropic heat drop.

If the steam enters the nozzle in a super heated condition, then during expansion, the friction tends to super heat the steam.

The ratio of actual or useful heat drop to isentropic heat drop is known as - Coefficient of nozzle or nozzle efficiency. K.

nozzle = K = Nozzle efficiency

$$= \frac{\text{Actual/useful heat drop}}{\text{Isentropic heat drop}}$$

$$= \frac{AB'}{AC} = \frac{h_A - h_B{}'}{h_A - h_C} = \frac{\Delta h_{act}}{\Delta h_{isen}}$$

The efficiency of a nozzle generally varies from 0.85 to 0.95.

VELOCITY COEFFICIENT

In the problems of nozzles, sometimes, the term velocity coefficient is used for accounting the effects of friction.

Velocity coefficient is defined as the ratio of actual exit velocity to exit velocity when the flow is isentropic for the same pressure drop.

The velocity coefficient depends upon the dimensions of the nozzle, roughness of the nozzle walls, velocity of flow, friction etc.

VELOCITY OF STEAM

Steam enters the nozzle with high pressure and low velocity and leaves the nozzle with high velocity and low pressure. The initial velocity compared to exit velocity is so small and is generally neglected. Let

$$= \frac{n}{n-1} p_1 \cdot v_1 \left(1 - \frac{p_2 v_2}{p_1 v_1}\right) \qquad \ldots\ldots\ldots (1)$$

$$\therefore \qquad \frac{v_2}{v_1} = \left(\frac{p_1}{p_2}\right)^{1/n} \qquad \ldots\ldots\ldots (2)$$

$$V_2{}^2 = \frac{n}{n-1} p_1 v_1 \left[1 - \frac{p_2}{p_1}\left(\frac{p_1}{p_2}\right)^{1/n}\right]$$

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

$$m = \frac{\text{Volume of steam flowing}/sec}{\text{Specific volume of steam at } p_2}$$

$$= \frac{A \cdot V_2}{v_2}$$

$$\therefore \qquad m = \frac{A}{v_2} \cdot \sqrt{2\left(\frac{n}{n-1}\right) p_1 v_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

$$m = \frac{A}{v_1 \left(\frac{p_1}{p_2}\right)^{1/n}} \sqrt{2\left(\frac{n}{n-1}\right) p_1 V_1 \left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

V1 = Velocity of steam at entrance of nozzle - m/sec. V2 = Velocity of steam at any section - m/sec

h1 = Enthalpy of entering steam - J/kg

h2 = Enthalpy of steam at the section considered - J/kg

For unit mass flow of steam, we have the steady flow energy equation:

$$= \frac{n}{n-1}(p_1 v_1 - p_2 v_2)$$

The gain in kinetic energy between any two sections is equal to loss of enthalpy. Enthalpy drop hd = (hl - h2)

$$\therefore \quad \frac{V_2^{\,2}}{2} = \frac{n}{n-1}(p_1 v_1 - p_2 v_2)$$

Neglecting the velocity of entering steam or velocity of approach;

V_{22}= 2 hd =2000 h_dJ/kg

In actual practice, always certain amount of friction exits between steam and the surfaces of the nozzle. This reduces the enthalpy drop by 10-15 percent and hence the exit velocity of steam is also reduced correspondingly.

K = Nozzle efficiency or coefficient of nozzle.

MASS OF STEAM DISCHARGED THROUGH A NOZZLE

The steam flowing through a nozzle approximately follows the equation pV^n = constant.
Where n = 1.135 for saturated steam= 1.300 for superheated steam.
Let p1 = Initial pressure of steam - N/m2
v1 = Initial volume of 1 kg of steam - m3
p2 = Pressure of steam at throat - N/m2
v2 - Volume of steam at pressure p2 - m3/kg
A = Cross sectional area of nozzle - m2
V2 = Velocity of leaving steam - m/sec.
Work done during Rankine cycle (Rankine area)

= Drop in enthalpy

$$\therefore \quad \left(\frac{p_2}{p_1}\right)^{1-n} = \left[\frac{n+1}{2}\right]^{-n}$$

$$\therefore \quad \frac{p_2}{p_1} = \left[\frac{2}{n+1}\right]^{\frac{n}{n-1}} \quad \ldots\ldots\ldots (6)$$

$$= \frac{A}{v_1}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}} \sqrt{2\left(\frac{n}{n-1}\right)p_1 V_1\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\right]}$$

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

$$= A\sqrt{\frac{2n}{n-1}\cdot\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{2/n}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]} \quad \ldots\ldots\ldots (4)$$

$$\frac{m}{A} = \sqrt{\frac{2n}{n-1}\frac{p_1}{v_1}\left[\left(\frac{p_2}{p_1}\right)^{2/n}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]} \quad \ldots\ldots\ldots (5)$$

$$\left[\left(\frac{p_2}{p_1}\right)^{2/n} - \left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right]$$

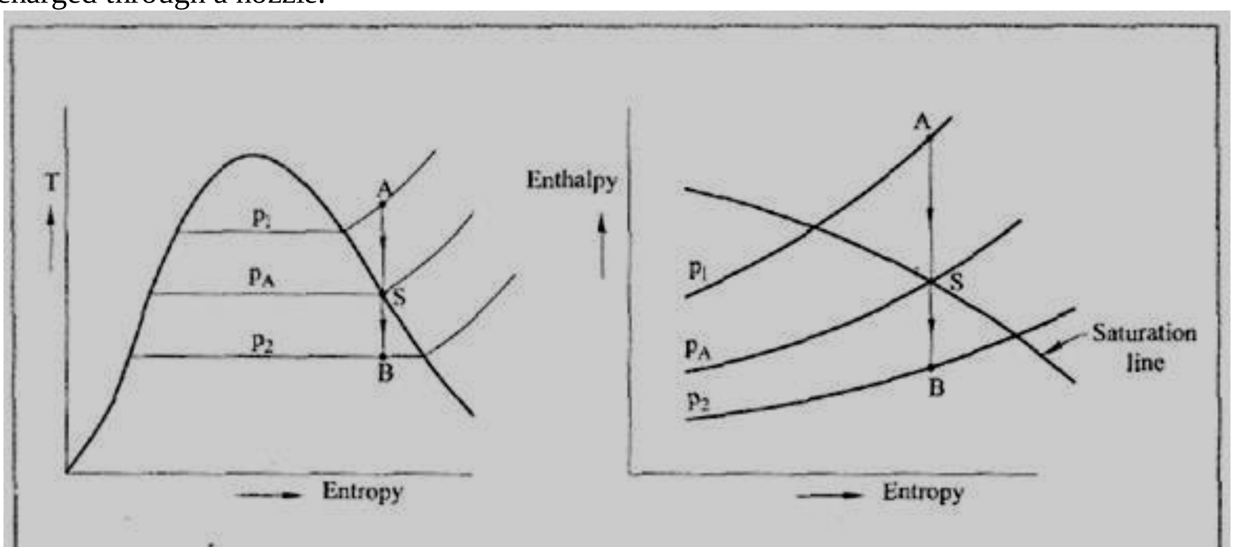
$$\therefore \quad \frac{2}{n}\left[\frac{p_2}{p_1}\right]^{\frac{2}{n}-1} - \frac{n+1}{n}\left[\frac{p_2}{p_1}\right]^{\frac{n+1}{n}-1} = 0$$

$$\therefore \quad \left(\frac{p_2}{p_1}\right)^{\frac{2-n}{n}} = \frac{n+1}{2}\left(\frac{p_2}{p_1}\right)^{1/n} \quad \text{(or)}$$

CRITICAL PRESSURE RATIO :From equation (4); the rate of mass flow of steam per unit are is given by :
The ratio (p2/p1) is known as - Critical pressure ratio and its value depends upon the value of index n. The pressure at throat is known as - Critical pressure and the ratio of pressure at minimum cross section i.e., throat (p2) to initial pressure - pressure at entrance (p1) is known as —critical pressure ratio. The area of throat of all steam nozzles should be designed on this ratio.

CONDITION FOR MAXIMUM DISCHARGE AND MAXIMUM DISCHARGE
Normally, a nozzle is designed for maximum discharge by designing a certain throat pressure (p2) which produces this condition. For only one value of pressure ratio (p2/p1) the discharge will be maximum. That ratio is Critical pressure ratio - Ratio of throat pressure to inlet pressure (p2/p1) For maximum discharge

This condition should be met to obtain maximum discharge from a nozzle. This equation gives pressure ratio for a maximum discharge per unit area through the nozzle. When this condition is met and the discharge is maximum, then the flow through the nozzle is called choking flow. Nozzles are always designed for choked flow. We know that mass of steam discharged through a nozzle.

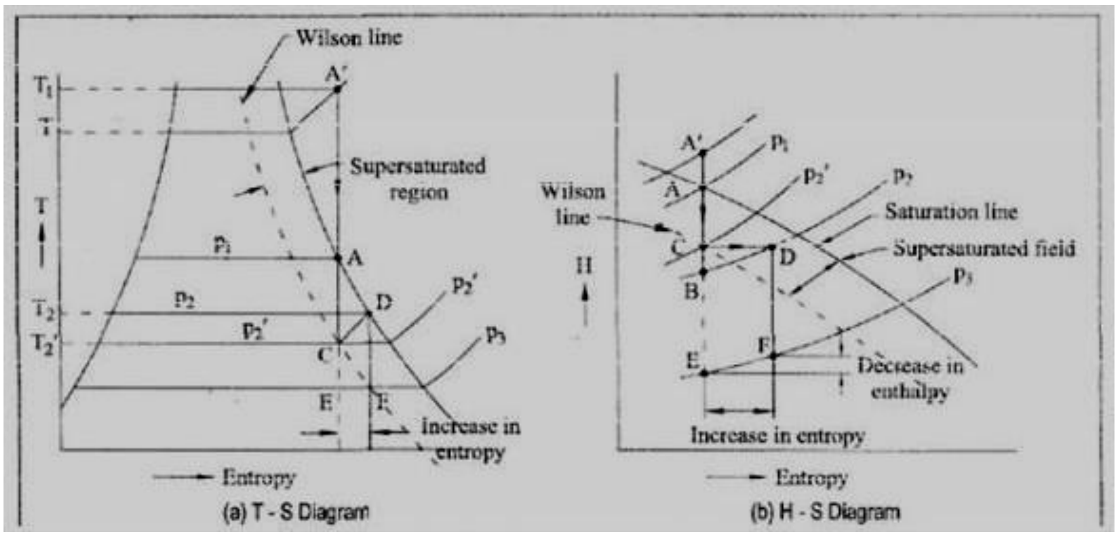


(a) T - S Diagram (b) H - S Diagram

So, in a nozzle, the maximum mass flow depends only on initial conditions of steam (p1, v1) and the throat area and is independent of the exit conditions of steam. The mass flow being constant at all sections of the nozzle, maximum discharge per unit area occurs at throat which has minimum area. In a convergent - divergent nozzle, the discharge remains constant after throat. The divergent portion doesn't affect the discharge of steam, passing through the nozzle but it only accelerates the steam leaving the nozzle. When the initial pressure p1 is constant, the discharge through the nozzle increases as the pressure p2 at throat decreases during the expansion process. Once the throat pressure p2 reaches the value given by equation (6); the discharge reaches a maximum value and it remains constant after throat.

SUPER SATURATED OR META STABLE FLOW

When a super heated vapour expands adiabatically or isentropically, the vapour begins to condense when saturated vapour line is reached. As expansion continues below this line into wet region, condensation proceeds gradually and steam becomes more and more wet. There is always a stable mixture of steam and condensate (liquid) at any point during expansion. This type of expansion is in thermal equilibrium and is shown in Fig. 3.13 on T-S and H-S diagrams.



The point S in expansion lies on saturation line and represents the point at which condensation within the vapour just begins. The condensation of steam occurs when steam passes through certain distance in the nozzle and after certain short interval of time.

When steam flows through the nozzle, the discharge of steam through the nozzle will be slightly less than the theoretical discharge due to the effect of friction. But, during the flow of wet steam through the nozzle, the measured discharge is slightly greater than the theoretical discharge even though we consider the effect of friction.

Normally, condensation starts around tiny dust particles which are always present in commercial steam plants in sufficient quantity. When steam is free of foreign particles, condensation process is delayed and the temperature of the steam continues to fall. This is known as - super saturation. When certain degree of super saturation is reached, the presence of dust particles has no effect on condensation and equilibrium between vapour and liquid phases is attained completely and also instantaneously.

.

In normal condensation, the random kinetic energy of the molecules fall to a level which is insufficient to overcome the attractive forces of the molecules and some of the slower moving molecules join together to form tiny droplets of water. A certain time interval is essential for the collection of these molecules to form droplets.

In actual practice, the velocity of steam in sonic or even super sonic and the convergent portion of the nozzle is so short the molecules of steam find no sufficient time to collect and form droplets and steam doesn't condense at the saturation temperature corresponding to the pressure but continues to expand with fall in temperature but without condensation. As a result, equilibrium between liquid and vapour phases is delayed. The expansion takes place very rapidly and condensation can't keep pace with expansion and lags behind. Due to this, the steam remains in an unnatural dry or super heated state. The steam in such conditions is said to be 'super - saturated' or ' meta - stable'. It is also called -Super cooled steam and its temperature at any pressure is less than the

saturation temperature; corresponding to that pressure. The flow of super saturated steam through the nozzle is called -super saturated or meta stable or non - equilibrium flow. Super saturation means that steam doesn't condense at the saturation temperature corresponding to the pressure as it occurs in case of equilibrium pressure as it occurs in case of equilibrium flow.

In the state of 'super saturation', the steam is under cooled to a temperature less than that corresponding to its pressure; hence, the density of steam increases and hence the measured discharge increases than the calculated theoretical discharge. Experiments showed that in the absence of dust; dry saturated steam when suddenly expanded, doesn't condense until its density is about 8 times that of saturated vapour of the same pressure.

The reasons for super saturated flow are :

1. The flow of steam is so rapid that it doesn't allow time for transfer of heat. It may take about 0.001 second only for steam to travel from inlet to exit of nozzle.
2. There may not be any dust particles which generally form nucleus for condensation. At certain instant, the supersaturated steam condenses suddenly to its natural state.
   Thus, flow of steam through a nozzle may be regarded as either ideal adiabatic or adiabatic flow modified by friction and super saturation.

Super Saturated Flow

Point A' represents the position of initial super heated steam at entrance pressure p1. The line A' - A represents isentropic expansion of steam in thermal equilibrium upto saturation line. Line AC represents isentropic expansion of steam in super saturated region. Upto the point at which condensation occurs, the state of steam is not of stable equilibrium not unstable equilibrium either, since a small disturbance will cause condensation to commence. So, steam in this condition is said to be in meta stable state. Point C represents the meta stable state. It is obtained by drawing a vertical line from points to Wilson line. At C; the steam condenses suddenly. Line CD represents condensation of steam at constant enthalpy. Point D is obtained by drawing a horizontal line through C to meet throat pressure p2 of the nozzle. Line DF represents isentropic expansion of steam in the divergent portion in thermal equilibrium.

During the partial condensation of steam DF, sufficient amount of heat is released which raises the temperature back to saturation temperature.

EFFECTS OF SUPER SATURATION

The following are the important effects that occur during super saturated flow of steam in a nozzle.

1. As the condensation doesn't take place during super saturated expansion, the temperature at which super saturation occurs will be less than the super saturation temperature corresponding to the pressure. So, the density of super saturated steam will be more than that for equilibrium conditions. (Generally 8 times that of ordinary saturated vapour at the corresponding pressure). Which gives increase in the mass of steam discharged.

2. Due to super saturation, the entropy and specific volume increase.

3. Super saturation increases slightly the dryness fraction.

4. For some pressure limits, super saturation reduces enthalpy drop slightly. As velocity is proportional to square root of enthalpy drop; exit velocity is also reduced slightly.

When meta stable conditions exist in the nozzle; Mollier chart (H-S chart) should not be used and the expansion must be considered to follow the law $pv^{1.3} = C$ i.e., with index of expansion for super heated steam. The problems on super saturated flow can't be solved by Mollier chart unless Wilson line is drawn on it.

WILSON LINE

Generally, there is a limit upto which super saturated flow is possible. This limit of super saturation is represented by a curve known as - Wilson line, on the Mollier diagram. Above this curve, steam is super saturated and super heated. Beyond Wilson line, there is no super saturation. At Wilson line condensation occurs suddenly and irreversibly at constant enthalpy and then remains in stable condition. The result is to reduce heat drop slightly during expansion causing corresponding reduction in exit velocity and final dryness fraction increases slightly.

The limiting condition of under cooling at which condensation begins and restores the conditions of thermal equilibrium is called Wilson line. Generally, Wilson line closely follows 0.96 dryness fraction line. In nozzles, this limit may be within the nozzle or after the vapour leaves the nozzle.

DEGREE OF UNDER COOLING

It is the difference between super saturated steam temperature and saturation temperature at that pressure. The temperature T2 ' is less than the normal temperature of steam at pressure p2. The state C is known as - Under cooled as the temperature of steam is lesser than the saturation temperature at pressure p2'. The amount of under cooling (Difference in temperatures) is known as - Degree of under cooling. Degree of under cooling = T2 - T2 '

There is a limit to the degree of under cooling possible and the limit to which the super saturated flow is possible is given by - Wilson line. The region between the Wilson line and the dry saturated line is called - Super saturatedzone.

When Wilson line is reached, condensation begins at constant enthalpy and pressure remains unaltered.

DEGREE OF SUPER SATURATION

The ratio of pressures corresponding to temperature of super saturated steam and saturation temperature is known as - Degree of super saturation

$$\text{Degree of super saturation} = \frac{\text{Pressure corresponding } T_2}{\text{Pressure corresponding to } T_2'} = \frac{p_2}{p_2'}$$

Numerical Problems – Steam Nozzles

1. Steam at a pressure of 10 bar and 210°C is supplied to a convergent divergent nozzle with a throat area of 15 cm^2. The exit is below critical pressure. Find the coefficient of discharge, if flow is 7200 kg of steam per hour.

Solution: From steam table at 10 bar and 210°C, $v = 0.2074\ m^3/kg$.

As steam is initially superheated $n = 1.3$

Now

$$m = A\sqrt{1000\,n \times \frac{P_1}{v_1} \times \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}} = 15\times10^{-4}\sqrt{1000\times1.3\times\frac{10\times10^2}{0.2074}\times\left(\frac{2}{1.3+1}\right)^{\frac{1.3+1}{1.3-1}}} = 2.1977.$$

$$\text{Co-efficient of discharge} = \frac{\text{Actual discharge}}{\text{Theoretical discharge}} = \frac{7200}{2.1977\times3600} = 0.91$$

2. Dry saturated steam at a pressure of 8 bar enters a convergent divergent nozzle and leaves it at a pressure of 1.5 bar. If the flow is isentropic and if the corresponding expansion index is 1.133, find the ratio of cross-sectional area at exit and throat for maximum discharge.

Solution: Throat pressure $P_2 = P_1\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} = 8\left(\frac{2}{1.133+1}\right)^{\frac{1.133}{1.133-1}} = 4.6227$ bar.

At 8 bar, from steam tables $\phi_s = 6.6628$ kJ/kg K, $H = 2769.1$ kJ/kg

At 4.6227 bar, $V_s = 0.40366\ \text{m}^3/\text{kg}$, $h = 627.542$ kJ/kg
$L = 2117.63$ kJ/kg, $\phi_w = 1.8308$ kJ/kg K, $\phi_s = 6.8475$ kJ/kg K.

At 1.5 bar, $V_s = 1.1593\ \text{m}^3/\text{kg}$, $h = 467.11$ kJ/kg $L = 2226.5$ kJ/kg, $\phi_w = 1.4336$, $\phi_s = 7.2233$ kJ/kg K.

For throat, entropy before expansion = entropy after expansion

$$\phi_1 = \phi_2$$

$$\phi_{s_1} = \phi_{w_2} + x_2(\phi_{s_2} - \phi_{w_2}),\ 6.6628 = 1.8308 + x_2(6.8475 - 1.8308)$$

$$\therefore x_2 = 0.963$$

$$H_2 = h_2 + x_2 L_2 = 627.542 + 0.963 \times 2117.63 = 2666.8197 \text{ kJ/kg.}$$

Enthalpy drop form entry to throat $= H_1 - H_2$

Velocity at throat $V_2 = 44.72\sqrt{H_1 - H_2} = 44.72\sqrt{2769.1 - 2666.8197} = 452.27$ m/s.

$$\therefore m = \frac{A_2 V_2}{v_2}, \quad \therefore \quad A_2 = \frac{m \times v_2}{V_2} = \frac{m \times x_2 \times v_{s_2}}{V_2}$$

$$A_2 = \frac{m \times 0.963 \times 0.40366}{452.27} = 0.0008594 \text{ m}$$

For exit, $\phi_1 = \phi_3, \phi_{s_1} = \phi_{w_3} + x_3(\phi_{s_3} - \phi_{w_3})$

$$6.6628 = 1.4336 + x_3(7.2233 - 1.4336),\ \therefore x_3 = 0.903$$

$$.H_3 = h_3 + x_3 L_3 = 467.11 + 0.903 \times 2226.5 = 2477.64 \text{ kJ/kg.}$$

$\therefore$ Enthalpy drop from entrance to exit $= H_1 - H_3$

Velocity at exit, $V_3 = 44.72\sqrt{H_1 - H_3} = 44.72\sqrt{2769.1 - 2477.64} = 763.468$ m/s.

Now, $m = \frac{A_3 V_3}{v_3} = \frac{A_3 V_3}{x_3 \times v_{s_3}}$

$$\therefore A_3 = \frac{m \times x_3 \times v_{s_3}}{V_3} = \frac{m \times 0.903 \times 1.1593}{763.468} = 0.001371 \text{ m}$$

$$\therefore \frac{\text{Area at exit}}{\text{Area at throat}} = \frac{A_3}{A_2} = \frac{0.001371 \text{ m}}{0.0008594 \text{ m}} = 1.595$$

A convergent divergent adiabatic steam nozzle is supplied with steam at 10 bar and 250°c.the discharge pressure is 1.2 bar.assuming that the nozzle efficiency is 100% and initial velocity of steam is 50 m/s. find the discharge velocity.

Given Data:-

Initial pressure(p_1)=10bar Initial

Temperature(T_1)=250°c

Exit pressure(p_2)=1.2 bar

Nozzle efficiency(η_{nozzle})=100%

Initial velocity of steam (v_1)=50m/s

Solution:-

From steam table, For 10 bar, 250°c, h_1=2943 KJ/kg s_1=6.926 KJ/kgk

From steam table, For 1.2 bar,

h_{f2} =439.3 KJ/kg ; h_{fg2}=2244.1 KJ/kg;

s_{f2}=1.3 61 KJ/kg K ; s_{fg2}=5.937 KJ/kgK.

Since $s_1=s_2$,

S1=sf 2+x2sfg2

$6.926=1.361+x_2(5.937)$

X_2=0 .9373

We know that,

$$h2=hf2+x2hfg2$$

$$= 439.3+(0.9373)2244.1$$

$$h_2 = 2542KJ/Kg$$

Exit velocity $(V2) = Rt[(2000(2943)\ \ 2542) + 50^2]$

$$= 896.91m/s.$$

Dry saturated steam at 6.5 bar with negligible velocity expands isentropically in a convergent divergent nozzle to 1.4 bar and dryness fraction 0.956. De termine the final velocity of steam from th e nozzle if 13% heat is loss in friction. Find th e % reduction in the final velocity.

Given data:

Exit pressure (P2) = 1.4 bar

Dryness fract ion (X2) = 0.956

Heat loss = 13%

To Find:

The percent reduction in final velocity

Solution:

From steam table for initial pressure P1 = 6.5bar, take values h_1 =

$$h_1 = 2758.8 KJ/Kg$$

Similarly, at 1.4 bar,

$$h_{fg2} = 2231.9 \ KJ/Kg$$

$$h_{f2} = 458.4 KJ/Kg$$

$$h2 = hf2 + X2 \ hfg2$$

$$= 458.4 + (0.956)\ 2231.6$$

$$h_2 = 2592.1 \ KJ/Kg$$

$h_2 = 2592.1$ KJ/Kg

Final velocit y (V2) = Rt (2000(h1-h2))

V2 = 577.39 m/s

Heat drop is 13%= 0.13

Nozzle efficiency (η) = 1- 0.13 = 0.87

Velocity of s team by considering the nozzle efficiency,

y (V2) = Rt (2000(h1-h2)) x η

V2 = 538.55 m/s

= % reductio n in final velocity = 6.72%

'.A convergent divergent nozzle receives steam at 7bar and 200°c and it expands isentropically into a space of 3bar neglecting the inlet velocity calculat e the exit area required for a mass flow of 0.1Kg/sec . when the flow is in equilibrium through all and super saturated with $PV^{1.3}$=C.

Given Data:

Initiall pressure (P_1) = 7bar = $7\times 10^5 N/m^2$

Initiall temperature (T_1) = 200°C

Press ure (P_2) = 3bar = $3\times 10^5 N/m^2$

Mass flow rate (m) = 0.1Kg/sec

$PV^{1.3}$ =C

To Find:

Exit area

Solution:

From st eam table for P1 = 7bar and T1 = 200°C V_1 =

0.2999

h_1 = 2844.2

S1 = 6.886

Similarly for P2 = 3bar

Vf2 = 0.001074 Vg2 = 0.60553 hf2 =

561.5 hfg2 = 2163.2

Sf2 = 1.672 Sfg2 = 5.319

We know that, S1 = S2 = St

$S_1 = S_{f2} + X_2 S_{fg2}$

$6.886 = 1.672 + X_2 (5.319)$ X_2 =

0.98

$$h2 = hf2 + X2\ hfg2$$

$$h_2 = 561.5 + 0.98\ (2163.2)$$

(i) Flow is in equilibriu m through all:

$$V_2 = 569.56$$

$$v_2 = X_2 \times v_{g2}$$

$$= 0.98 \times 0.60553 = 0.5934$$

$$V_2 = \sqrt{2000\,(h_1 - h_2)}$$

$$V_2 = \sqrt{2000\,(2844.2 - 2681.99)}\ V_2 =$$

569.56

$$v_2 = X_2 \times v_{g2}$$

$= 0.98 \times 0.60553 = 0.5934$

$$m = \frac{[(A]_2 \times V_2)}{v_2}$$

$$A_2 = \frac{[m \times V_2)}{v_2} = \frac{0.5934 \times 0.1}{569.56}$$

$$A_2 = 1.041 \times 10^{-4} m^2$$

(ii) For saturated flow:

$$v_2 = \sqrt{\frac{2n}{n-1}\ (P_1 v_1)(1 - (\frac{P_2}{P_1})^{\frac{n-1}{n}}}$$

$$v_2 = \sqrt{\frac{2(1.3)}{1.3-1}\ (7 \times 10^5 \times 0.2999)\ (1 - \frac{3 \times 10^5}{7 \times 10^5})^{\frac{1.2-1}{1.2}}}$$

v2 = 568.69 m/s

specific volume of steam at exit. For super saturated flow, $P_1 v_1^n = P_2$

$$\left(\frac{v_2}{v_1}\right)^n = \frac{P1}{P2}$$

$$\left(\frac{v_2}{v_1}\right)^n = \frac{P1}{P2}$$

$$v_2 = \left(\frac{7}{3}\right)^{\frac{1}{1.3}} \times 0.2999$$

$$v_2 = 0.5754$$

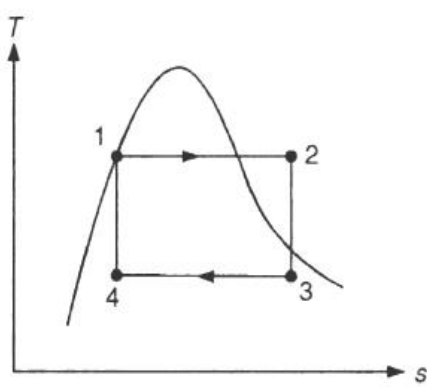
$$A_2 = \frac{[m \times V_2)}{v_2}$$

$$= \frac{0.1 \times 0.5754}{568.69}$$

$$A_2 = 1.011 \times 10^{-4} m^2$$

EXAMPLE 9.1

A steam power plant operates between a boiler pressure of 4 MPa and 300°C and a condenser pressure of 50 kPa. Determine the thermal efficiency of the cycle, the work ratio, and the specific steam flow rate. assuming the cycle to be a Carnot cycle



Solution

(a) The T–s diagram of a Carnot cycle is shown in the adjacent figure.

Process 1–2 is reversible and isothermal heating of water in the boiler.

Process 2–3 is isentropic expansion of steam at state 2 in the turbine.

Process 3–4 is reversible and isothermal condensation of steam in the condenser.

Process 4–1 is isentropic compression of steam to initial state.

At state 1: $P_1 = 4$ MPa, $T_1 = 300°C$

At state 2: P_2 = 50 kPa, the steam is in a saturated state.

From the saturated water-pressure table (Table 4 of the Appendix), at 50 kPa, we get $T_2 = T_{min} = T_{sat} = 81.33°C$

Therefore, the thermal efficiency for the given Carnot cycle is

$$\eta_{th,carnot} = 1 - \frac{T_{min}}{T_{max}} = 1 - \frac{81.33 + 273.15}{300 + 273.15} = 0.3815$$

$$= \boxed{38.15 \text{ per cent}}$$

$$\text{The work ratio} = \frac{\text{net work output}}{\text{gross work output}} = \frac{w_{net,out}}{w_{gross,out}}$$

Heat supplied = $h_2 - h_1 = h_{fg\,@\,4MPa}$ = 1714.1 kJ/kg (From Table 4 of the Appendix)

$$\eta_{th,carnot} = \frac{w_{net,out} - w_{net,in}}{\text{gross heat supplied}} = 0.3815$$

Therefore,

$$w_{net,\,out} - w_{net,\,in} = 0.3815 \times 1714.1 = 653.9 \text{ kJ/kg}$$

That is, the net work output = 653.9 kJ/kg.

To find the expansion work for the process 2–3, h_3 is required.

From Table 4, h_2 = 2801.4 kJ/kg and $s_2 = s_3$ = 6.0701 kJ/(kg K)

But
$$s_3 = 6.0701 = s_{f3} + x_3 s_{fg3} = 1.0910 + x_3(7.5939 - 1.0910)$$
or
$$x_3 = 0.766$$

Now,
$$h_3 = h_{f3} + x_3 h_{fg3} = 340.49 + 0.766(2645.9 - 340.49) = 2106.4 \text{ kJ/kg}$$

Therefore,
$$w_{32} = h_2 - h_3 = 2801.4 - 2106.4 = 695 \text{ kJ/kg}$$

That is, the gross work output, $w_{gross,out}$ = 695 kJ/kg

Therefore,

$$\text{Work ratio} = \frac{w_{net,out}}{w_{gross,out}} = \frac{653.9}{695} = \boxed{0.94}$$

The specific steam flow rate (ssfr) is the steam flow required to develop unit power output. That is,

$$\text{ssfr} = \frac{\dot{m}_{\text{steam}}}{\dot{m}_s w_{\text{out}}} = \frac{1}{w_{\text{net, out}}}$$

$$= \frac{1}{653.9} = \boxed{0.00153 \text{ kg/kW}}$$

2. A steam power plant uses steam as working fluid and operates at a boiler pressure of 5 MPa, dry saturated and a condenser pressure of 5 kPa. Determine the cycle efficiency for (a) Carnot cycle (b) Rankine cycle. Also show the T-s representation for both the cycles.

Solution:

From steam tables:

At 5 MPa $\quad h_{f,\,5\text{MPa}} = 1154.23$ kJ/kg, $s_{f,\,5\text{ MPa}} = 2.92$ kJ/kg · K

$h_{g,\,5\text{MPa}} = 2794.3$ kJ/kg, $s_{g,\,5\text{ MPa}} = 5.97$ kJ/kg · K

At 5 kPa

$h_{f,\,5\text{kPa}} = 137.82$ kJ/kg, $s_{f,\,5\text{kPa}} = 0.4764$ kJ/kg · K

$h_{g,\,5\text{kPa}} = 2561.5$ kJ/kg, $s_{g,\,5\text{kPa}} = 8.3951$ kJ/kg · K

$v_{f,\,5\text{kPa}} = 0.001005$ m³/kg

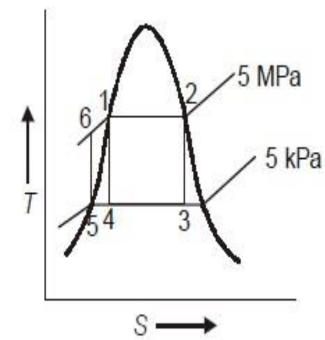


Fig. 8.30

Carnot cycle : 1–2–3–4–1

Rankine cycle : 1–2–3–5–6–1

As process 2-3 is isentropic, so $s_2 = s_3$

and

$$s_3 = s_{f,\,5\text{kPa}} + x_3 \cdot s_{fg,\,5\text{kPa}} = s_2 = s_{g,\,5\text{MPa}}$$

$$x_3 = 0.694$$

Hence enthalpy at 3,

$$h_3 = h_{f,\,5\text{kPa}} + x_3 \cdot h_{fg,\,5\text{kPa}}$$

$$h_3 = 1819.85 \text{ kJ/kg}$$

Enthalpy at 2. $\quad h_2 = h_{g,\,5\text{MPa}} = 2794.3$ kJ/kg

Process 1-4 is isentropic, so $s_1 = s_4$

$$s_1 = 2.92 = 0.4764 + x_4 \cdot (8.3951 - 0.4764)$$
$$x_4 = 0.308$$

Enthalpy at 4,

$$h_4 = 137.82 + (0.308 \times (2561.5 - 137.82))$$
$$h_4 = 884.3 \text{ kJ/kg}$$

Enthapy at 1,

$$h_1 = h_{f \text{ at 5 MPa}}$$
$$h_1 = 1154.23 \text{ kJ/kg}$$

Carnot cycle (1-2-3-4-1) efficiency:

$$\eta_{\text{carnot}} = \frac{\text{Net work}}{\text{Heat added}}$$
$$= \frac{(h_2 - h_3) - (h_1 - h_4)}{(h_2 - h_1)}$$
$$= \frac{\{(2794.3 - 1819.85) - (1154.23 - 884.3)\}}{(2794.3 - 1154.23)}$$
$$\eta_{\text{carnot}} = 0.4295$$

In Rankine cycle, 1-2-3-5-6-1

$$\text{Pump work, } h_6 - h_5 = v_{f,5}(p_6 - p_5)$$

$$= 0.001005\ (5000 - 5)$$

$$h_6 - h_5 = 5.02$$

$$h_5 = h_{f \text{ at } 5\text{kPa}} = 137.82 \text{ kJ/kg}$$

$$\text{Hence } h_6 = 137.82 + 5.02 = 142.84 \text{ kJ/kg}$$

$$h_6 = 142.84 \text{ kJ/kg}$$

$$\text{Net work in Rankine cycle} = (h_2 - h_3) - (h_6 - h_5)$$

$$= 974.45 - 5.02$$

$$= 969.43 \text{ kJ/kg}$$

$$\text{Heat added} = h_2 - h_6$$

$$= 2794.3 - 142.84$$

$$= 2651.46 \text{ kJ/kg}$$

$$\text{Rankine cycle efficiency} = \frac{969.43}{2651.46}$$

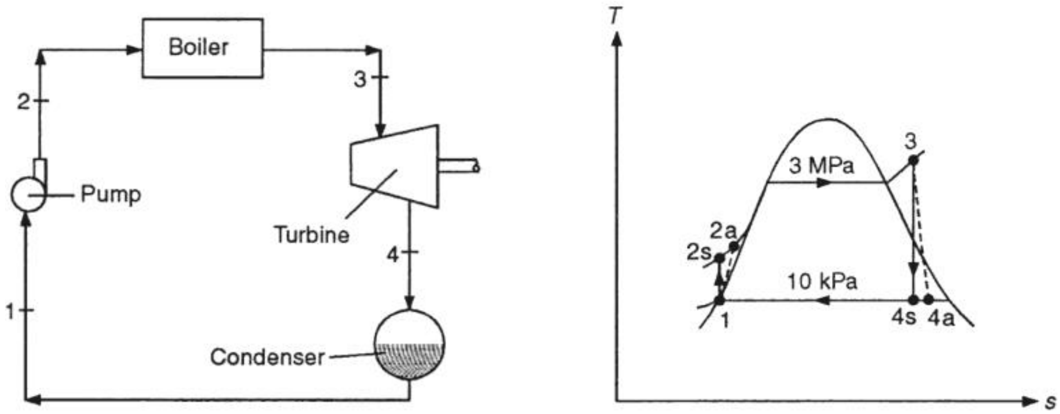
$$\eta_{\text{Rankine}} = 0.3656$$

or

$$\eta_{\text{Rankine}} = \mathbf{36.56\%} \quad \textbf{Ans.}$$

EXAMPLE 2

A steam power plant operates on the cycle shown below with 3 MPa and 400°C at the turbine inlet and 10 kPa at the turbine exhaust. The adiabatic efficiency of the turbine is 85 per cent and that of the pump is 80 per cent. Determine (a) the thermal efficiency of the cycle, and (b) the mass flow rate of the steam if the power output is 20 MW.



Solution

All the components are treated as steady-flow devices. The changes, if any, in the kinetic and potential energies are assumed to be negligible. Losses other than those in the turbine and pump are neglected.

(a) $$w_{\text{pump,in}} = \frac{v_1(P_2 - P_1)}{\eta_P} = \frac{0.001010\,(3000 - 10)}{0.80} = 3.77 \text{ kJ/kg}$$

Turbine work output is

$$w_{\text{turb,out}} = \eta_T w_{\text{turb,in}} = \eta_T (h_3 - h_{4s})$$

$$= 0.85(3230.90 - 2192.21) = 882.89 \text{ kJ/kg}$$

Boiler heat input is

$$q_{in} = h_3 - h_2 = 3230.9 - 195.59 = 3035.31 \text{ kJ/kg}$$

Thus,

$$w_{net,out} = w_{turb,out} - w_{pump,in} = 882.89 - 3.77 = 879.12 \text{ kJ/kg}$$

$$\eta_{th} = \frac{w_{net,out}}{q_{in}} = \frac{879.12}{3035.31} = 0.2896 = \boxed{28.96 \text{ per cent}}$$

If there are no losses in the turbine and the pump, the thermal efficiency would be 28.99 per cent.

(b) The power generated by the power plant is

$$\dot{W}_{net,out} = \dot{m} w_{net,out} = 20{,}000 \text{ kW}$$

Therefore, the mass flow rate, $\dot{m} = \frac{20{,}000}{879.12} = \boxed{22.75 \text{ kg/s}}$

3. A steam turbine plant operates on Rankine cycle with steam entering turbine at 40 bar, 350°C and leaving at 0.05 bar. Steam leaving turbine condenses to saturated liquid inside condenser. Feed pump pumps saturated liquid into boiler. Determine the net work per kg of steam and the cycle efficiency assuming all processes to be ideal. Also show cycle on T-s diagram. Also determine pump work per kg of steam considering linear variation of specific volume.

Solution:

From steam table

$$h_2 = h_{\text{at 40 bar, 350°C}} = 3092.5 \text{ kJ/kg}$$
$$s_2 = s_{\text{at 40 bar, 350°C}} = 6.5821 \text{ kJ/kg·K}$$
$$h_4 = h_{f \text{ at 0.05 bar}} = 137.82 \text{ kJ/kg}$$
$$s_4 = s_{f \text{ at 0.05 bar}} = 0.4764 \text{ kJ/kg}$$
$$v_4 = v_{f \text{ at 0.05 bar}} = 0.001005 \text{ m}^3\text{/kg}$$

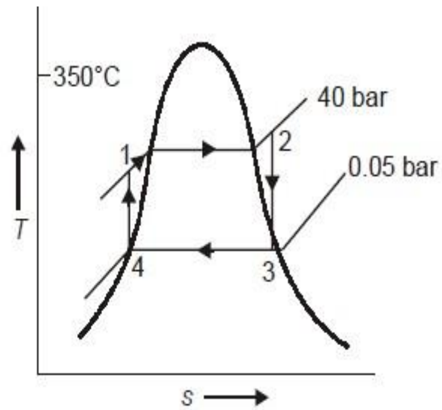


Fig. 8.31

Let dryness fraction at state 3 be x_3,

For ideal process, 2-3, $s_2 = s_3$

$$s_2 = s_3 = 6.5821 = s_{f \text{ at } 0.05 \text{ bar}} + x_3 \cdot s_{fg \text{ at } 0.05 \text{ bar}}$$

$$6.5821 = 0.4764 + x_3 \cdot 7.9187$$

$$x_3 = 0.7711$$

$$h_3 = h_{f \text{ at } 0.05 \text{ bar}} + x_3 \cdot h_{fg \text{ at } 0.05 \text{ bar}}$$

$$= 137.82 + (0.7711 \times 2423.7)$$

$$h_3 = 2006.74 \text{ kJ/kg}$$

For pumping process

$$h_1 - h_4 = v_4 \cdot \Delta p = v_4 \times (p_1 - p_4)$$

$$h_1 = h_4 + v_4 \times (p_1 - p_4)$$

$$= 137.82 + (0.001005 \times (40 - 0.05) \times 10^2)$$

$$h_1 = 141.84 \text{ kJ/kg}$$

Pump work per kg of steam = $(h_1 - h_4)$ = 4.02 kJ/kg

Net work per kg of steam = (Expansion work – Pump work) per kg of steam

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

$$= 1081.74 \text{ kJ/kg}$$

$$\text{Cycle efficiency} = \frac{\text{Net work}}{\text{Heat added}}$$

$$= \frac{1081.74}{(h_2 - h_1)}$$

$$= \frac{1081.74}{(3092.5 - 141.84)}$$

$$= 0.3667 \text{ or } 36.67\%$$

$$\boxed{\begin{aligned} \text{Net work per kg of steam} &= \mathbf{1081.74\ kJ/kg} \\ \text{Cycle efficiency} &= \mathbf{36.67\%} \\ \text{Pump work per kg of steam} &= \mathbf{4.02\ kJ/kg} \end{aligned}}$$ **Ans.**

A steam power plant running on Rankine cycle has steam entering HP turbine at 20 MPa, 500°C and leaving LP turbine at 90% dryness. Considering condenser pressure of 0.005 MPa and reheating occurring upto the temperature of 500°C determine,

(a) the pressure at wich steam leaves HP turbine

(b) the thermal efficiency

Solution:

Let us assume that the condensate leaves condenser as saturated liquid and the expansion in turbine and pumping processes are isentropic.

From steam tables,

$$h_2 = h_{\text{at 20 MPa, 500°C}} = 3238.2 \text{ kJ/kg}$$

$$s_2 = 6.1401 \text{ kJ/kg} \cdot \text{K}$$

$$h_5 = h_{\text{at 0.005 MPa, 0.90 dry}}$$

$$h_5 = h_{f \text{ at 0.005 MPa,}} + 0.9 \times h_{fg \text{ at 0.005 MPa}}$$

$$= 137.82 + (0.9 \times 2423.7)$$

$$h_5 = 2319.15 \text{ kJ/kg}$$

$$s_5 = s_{f \text{ at 0.005 MPa,}} + 0.9 \times s_{fg \text{ at 0.005 MPa}}$$

$$= 0.4764 + (0.9 \times 7.9187)$$

$$s_5 = 7.6032 \text{ kJ/kg} \cdot \text{K}$$

$$h_6 = h_{f \text{ at 0.005 MPa}} = 137.82 \text{ kJ/kg}$$

It is given that temperature at state 4 is 500°C and due to isentropic process $s_4 = s_5 = 7.6032$ kJ/kg ·K. The state 4 can be conveniently located on Mollier chart by the intersection of 500°C constant temperature line and entropy value of 7.6032 kJ/kg · K and the pressure and enthalpy obtained. But these shall be approximate.

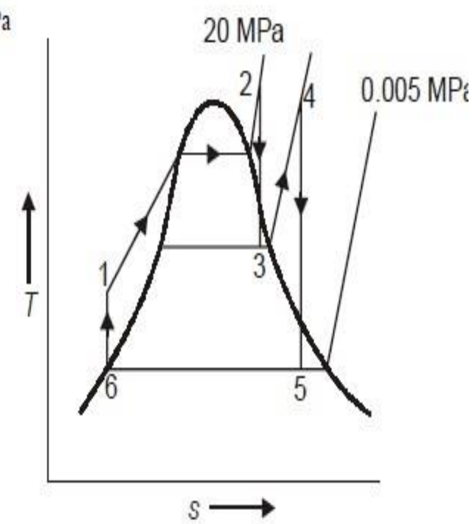


Fig. 8.32

The state 4 can also be located by interpolation using steam table. The entropy value of 7.6032 kJ kg · K lies between the superheated steam states given under, p = 1.20 MPa, $s_{\text{at 1.20 MPa and 500°C}}$ = 7.6759 kJ/kg · K

p = 1.40 MPa, $s_{\text{at 1.40 MPa and 500°C}}$ = 7.6027 kJ/kg · K

By interpolation state 4 lies at pressure

$$= 1.20 + \frac{(1.40-1.20)}{(7.6027-7.6759)}\ (7.6032 - 7.6759)$$

$$= 1.399 \text{ MPa} \approx 1.40 \text{ MPa}$$

Thus, steam leaves HP turbine at 1.4 MPa

Enthalpy at state 4, h_4 = 3474.1 kJ/kg

For process 2-3, $s_2 = s_3$ = 6.1401 kJ/kg · K. The state 3 thus lies in wet region as $s_3 < s_{g\text{ at 1.40 MPa}}$. Let dryness fraction at state 3 be x_3.

$$s_3 = s_{f\text{ at 1.4 MPa}} + x_3 \cdot s_{fg\text{ at 1.4 MPa}}$$

$$6.1401 = 2.2842 + x_3 \cdot 4.1850$$

$$x_3 = 0.9214$$

$$h_3 = h_{f\text{ at 1.4 MPa}} + x_3 \cdot h_{fg\text{ at 1.4 MPa}}$$

$$= 830.3 + (0.9214 \times 1959.7) = 2635.97 \text{ kJ/kg}$$

$$\text{Enthalpy at 1, } h_1 = h_6 + v_6(p_1 - p_6)$$

$$= h_{f\text{ at 0.005 MPa}} + v_{f\text{ at 0.005 MPa}}\ (20 - 0.005) \times 10^3$$

$$= 137.82 + (0.001005 \times 19.995 \times 10^3)$$

$$h_1 = 157.91 \text{ kJ/kg}$$

$$\text{Net work per kg of steam} = (h_2 - h_3) + (h_4 - h_5) - (h_1 - h_6)$$

$$= 1737.09 \text{ kJ/kg}$$

$$\text{Heat added per kg of steam} = (h_2 - h_1) = 3080.29 \text{ kJ/kg}$$

$$\text{Thermal efficiency} = \frac{\text{Net work}}{\text{Heat added}} = \frac{1737.09}{3080.29} = 0.5639 \text{ or } 56.39\%$$

Pressure of steam leaving HP turbine = **1.40 MPa**
Thermal efficiency = **56.39%** **Ans.**

The following readings were taken during the test on a boiler for a period of 1 h: Steam generated 5000 kg; coal burnt 700 kg, CV of coal 31402 kJ/kg and quality of seam 0.92. If the boiler pressure is 1.2 MPa and feed water temperature is 45°C. Find the boiler equivalent evaporation and thermal efficiency.

Solution: Given $m_s = 5000$ kg/h; $m_f = 700$ kg/h; CV = 31402 kJ/kg; $x = 0.92$; $p = 1.2$ MN/m^2 = 12 bar; $T_w = 45$°C.

From steam table (temperature basis) at 45°C, we have

$$h_{fw} = h_f = 188.35 \text{ kJ/kg}$$

From steam table (saturation pressure basis) at 12 bar, we have

$$h_f = 798.43 \text{ kJ/kg}$$
$$h_{fg} = 1984.3 \text{ kJ/kg}$$

Enthalpy of wet steam is given by

$$\begin{aligned} h_s &= h_f + x\,h_{fg} \\ &= 798.43 + 0.92 \times 1984.3 \\ &= 2623.99 \text{ kJ / kg} \end{aligned}$$

Equivalent mass of evaporation is given by

$$m_e = \frac{m_s}{m_f} = \frac{5000}{700} = 7.143$$

Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$= \frac{7.143 \times (2623.99 - 188.35)}{2257}$$

$$= 7.7 \text{ kg/kg of coal}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}}$$

$$= \frac{7.143 \times (2623.99 - 188.35)}{31402}$$

$$= 0.554 \text{ or } 55.4\%$$

A boiler is to produce 6000 kg/h of steam at 25 bar and 350°C. The feed water temperature is 40°C. If the calorific value of the fuel oil is 42000 and the expected thermal efficiency is 75%, determine the fuel consumption rate and the equivalent evaporation "from and at 100°C".

Solution: Given m_s = 6000 kg/h; p = 25 bar; T_{sup} = 350°C; T_w = 40°C; CV = 42000 kJ/kg; $\eta_{\text{boiler}} = 75\%$.
From steam table (temperature basis) at 40°C, we have

$$h_{fw} = h_f = 167.45 \text{ kJ/kg}$$

From steam table (saturation pressure basis) at 25 bar, we have

$$T_s = 223.94°\text{C}$$

$$h_f = 961.96 \text{ kJ/kg}$$

$$h_{fg} = 1839.0 \text{ kJ/kg}$$

$$h_g = 2800.9 \text{ kJ/kg}$$

Assuming specific heat at constant pressure of the steam, C_{ps} as 2.3 kJ/kg, we have
Enthalpy of superheated steam is given by

$$h_s = h_g + C_{ps}\left(T_{sup} - T_s\right)$$

$$= 2800.9 + 2.3 \times (350 - 223.94)$$

$$= 3090.838 \text{ kJ / kg}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_s\left(h_s - h_{fw}\right)}{m_f \times \text{CV}}$$

$$0.75 = \frac{6000 \times (3090.838 - 167.45)}{m_f \times 42000}$$

$$m_f = 556.8\ \text{kg / h}$$

Equivalent mass of evaporation is given by

$$m_e = \frac{m_s}{m_f} = \frac{6000}{556.8} = 10.775$$

Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$= \frac{10.775 \times \left(3090.838 - 167.45\right)}{2257}$$

$$= 13.96\ \text{kg / kg of oil}$$

The equivalent evaporation of a boiler from and at 100°C is found to be 12 kg of steam per kg of fuel burnt. The calorific value of the fuel is 35000 kJ/kg. Determine the efficiency of the boiler. If the boiler produces 15000 kg/h of steam at 20 bar from feed water at 40°C and the fuel consumption is 1800 kg/h, determine the condition of steam produced.

Solution: Given E = 12 kg/kg of fuel; CV = 35000 kJ/kg; m_s = 15000 kg/h; p = 20 bar; T_w = 40°C; m_f = 1800 kg/h.
Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$12 = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$m_e\left(h_s - h_{fw}\right) = 12 \times 2257$$

$$= 27084\ \text{kJ / kg of coal}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}}$$

The equivalent evaporation of a boiler from and at 100°C is found to be 12 kg of steam per kg of fuel burnt. The calorific value of the fuel is 35000 kJ/kg. Determine the efficiency of the boiler. If the boiler produces 15000 kg/h of steam at 20 bar from feed water at 40°C and the fuel consumption is 1800 kg/h, determine the condition of steam produced.

Solution: Given E = 12 kg/kg of fuel; CV = 35000 kJ/kg; m_s = 15000 kg/h; p = 20 bar; T_w = 40°C; m_f = 1800 kg/h.

Equivalent evaporation is given by

$$E = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$12 = \frac{m_e\left(h_s - h_{fw}\right)}{2257}$$

$$m_e\left(h_s - h_{fw}\right) = 12 \times 2257$$

$$= 27084 \text{ kJ / kg of coal}$$

Boiler efficiency is given by

$$\eta_{\text{boiler}} = \frac{m_e\left(h_s - h_{fw}\right)}{\text{CV}}$$

UNIT-III

STEAM TURBINES

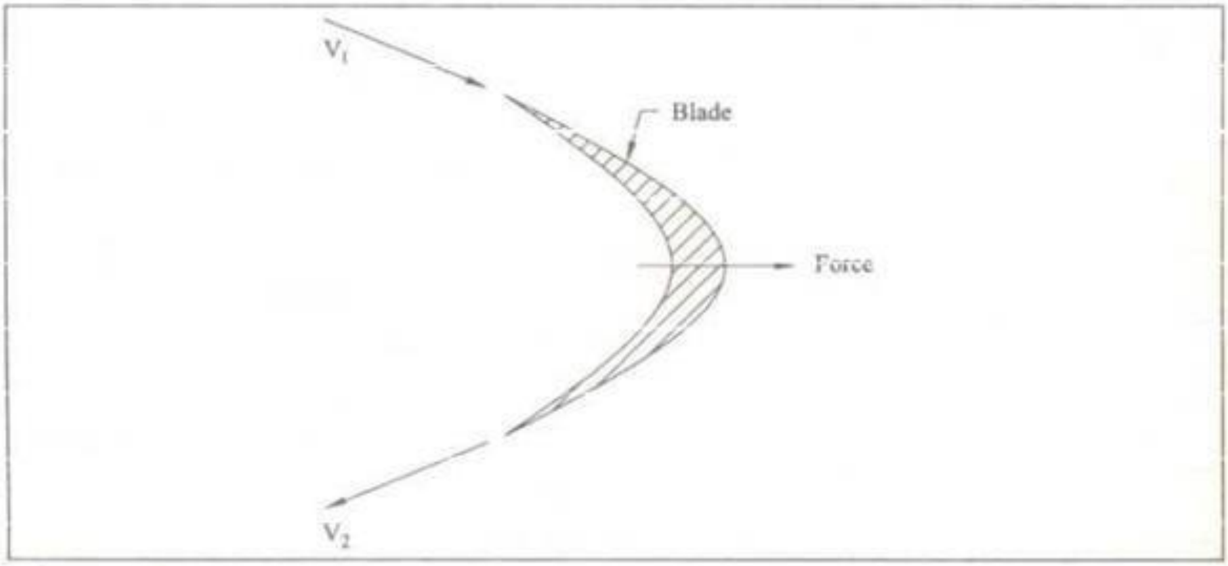
A steam turbine is a key unit in a steam power plant from which we get power. A steam turbine is a turbo-machine and a prime mover in which potential energy of steam is transformed into kinetic energy and this kinetic energy is then transformed into mechanical energy of rotation of shaft of turbine. In reciprocating steam engines, the pressure energy of steam is utilised and dynamic action of Steam is negligible. In steam engines, steam acts on piston as a load or weight and so, the action of steam is - static. Steam engines may be operated without any expansion or drop of pressure in the cylinder. The expansive property of steam is not utilised to fullest extent even in the best types of multi expansion steam engines.

Steam turbines can't be operated as that of steam engines. The turbine depends wholly upon the dynamic action of steam. The turbine utilises the kinetic or velocity energy of steam instead of pressure energy only. The expansive property of steam is almost utilised in the turbine either in admission nozzles or in the turbine blading.

The steam is caused to fall in pressure in a nozzle during admission to the turbine, due to this fall in pressure; certain amount of heat energy is converted into kinetic energy. A steam turbine consists of a number of curved blades fixed uniformly on the rim of a wheel which is fastened to a shaft and we obtain power from this shaft.

The high velocity steam from nozzles impinges on the blades of turbine, suffers a change in the direction of motion and thus gives rise to change in momentum and so a force. This constitutes the driving force of the turbine. The blades obtain no motive force from the static pressure of steam or from any impact of steam jet because blades are designed and curved in such a ways that steam enters the blades without any shock and will glide ON and OFF the blades.

According to Newton's second law of motion, the force is proportional to rate of change of momentum (Mass x velocity). If the rate of change of momentum is caused by allowing a high velocity steam jet to pass over a curved blade, then, steam will impart a force to the blade. If the blade is free, then it will move (rotate) in the direction of force.



PRINCIPLE OF OPERATION

1. A nozzle in which heat energy of high pressure steam is converted into kinetic energy so that steam issues from the nozzle with very high velocity.

2. Blades which change the direction of steam issuing from the nozzle so that a force acts on blades due to change of momentum and rotates them.

So, the basic principle of operation a steam turbine is generation of high velocity steam jet by expansion of high pressure steam in a nozzle and motive power in the turbine is obtained by change in momentum of the high velocity steam jet by allowing it to impinge on curved blades. Steam turbines are steady flow machines, have large exhaust outlets (for discharging used steam) and the speed of flow is very high. So, they can handle large volume of steam and produce higher power and the processes are assumed to be adiabatic. Steam turbines are capable of expanding steam to the lowest exhaust pressure obtainable in the condenser. The turbine is a constant high speed machine and really must be operated condensing in order to take full advantage of greater range of steam expansion.

Steam turbines are mainly used for electric power generation and for large marine propulsion. These are also used for direct drives of fans, compressors, pumps etc.

When properly designed and constructed, a steam turbine is the most durable prime-mover.

TYPES OF STEAM TURBINES

Steam turbines may be classified in many ways. Considering the action of steam which is most important factor, steam turbines are mainly classified as :

1. Impulse turbines.

2. Impulse reaction turbines (In practice known as - reaction turbines).

If the flow of steam through the nozzles and moving blades of a turbine takes place in such a way that steam is expanded and entire pressure drop takes place in nozzles only and pressure at the outside of blades is equal to inside of blades, then such a turbine is known as - impulse turbine.

In these turbines, the pressure drop takes place in nozzles only and not in moving blades. This is obtained by making the blade passage of constant cross sectional area.

In impulse reaction turbines, the pressure drop takes place in nozzles as well as moving blades. The drop of pressure of steam while flowing through the moving blades results in the generation of kinetic energy within the moving blades giving rise to reaction and adds to the driving force which is then transmitted through the rotor to the turbine shaft. This turbine works on the principles of both impulse and reaction. This is achieved by making the blade passage of varying cross sectional area.

IMPULSE TURBINE

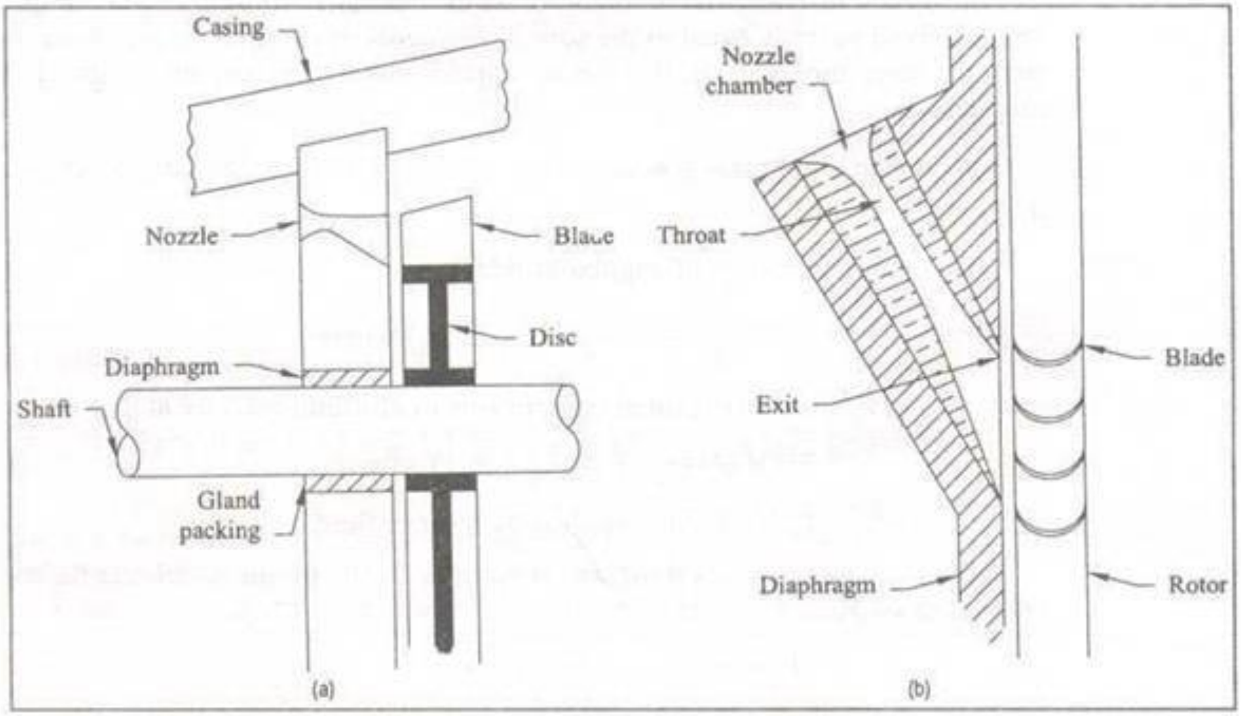
The turbines in which complete process of expansion of steam takes place in stationary nozzles and the kinetic energy is converted into mechanical work on the turbine blades are known as -Impulse turbines. In impulse turbines, the entire pressure drop takes place in nozzles only. The pressure drops from steam chest pressure to condenser or exhaust pressure. The pressure in the blade passages remains approximately constant and is equal to condenser pressure.

An impulse turbine for its operation, depends wholly on the impulsive force of high velocity steam jets, which are obtained by expansion of steam in nozzles. The action of steam jet impinging on the blades is said to be impulse and the rotation of rotor is due to impulsive forces of steam jets.

Generally, converging - diverging nozzles are used. Due to relatively large expansion ratio, steam leaves the nozzles at a very high velocity (Even supersonic). The steam at high velocity impinges over blades, both pressure and enthalpy remain constant, work transfer takes place, velocity reduces gradually and steam comes out with appreciable velocity . The nozzle angle is inclined at a fixed angle to tangent of rotor wheel.

Mostly, impulse turbines are axial flow turbines and they have zero degree of reaction (discussed later). The entire pressure drop takes place in nozzles resulting in enthalpy drop. The energy transfer is derived from a change of absolute velocity.

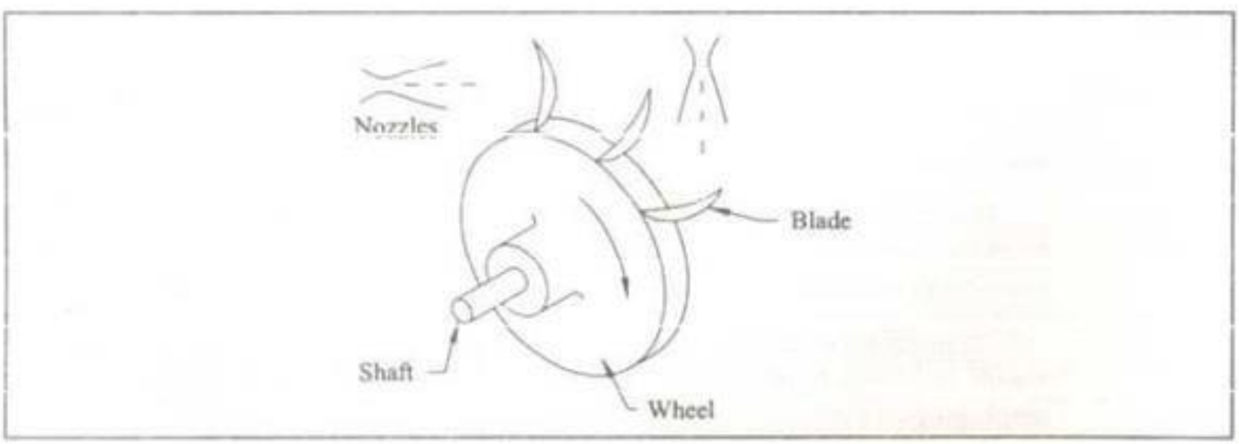
Impulse turbines are generally employed where relatively small amounts of power are required and where rotor diameter is fairly small.



ARRANGEMENT OF A SIMPLE IMPULSE TURBINE.

DE-LAVEL TURBINE

A De-lavel turbine named after Swedish Engineer De-lavel is the simplest impulse turbine and is commonly used.



RUNNER AND BUCKET OF DE-LAVEL TURBINE

The essential parts of an impulse turbine are - nozzles, blades and casing.

In nozzles, the expansive property of steam is utilised to produce jets of steam with very high velocity. The nozzle guides the steam to flow in the designed direction. It also

regulates the flow of steam. It is kept very close to turbine blades to minimise wind age losses.
The runner or rotor consists of a circular disc mounted on a shaft. On the periphery of the runner, a number of buckets or curved blades are fixed uniformly.

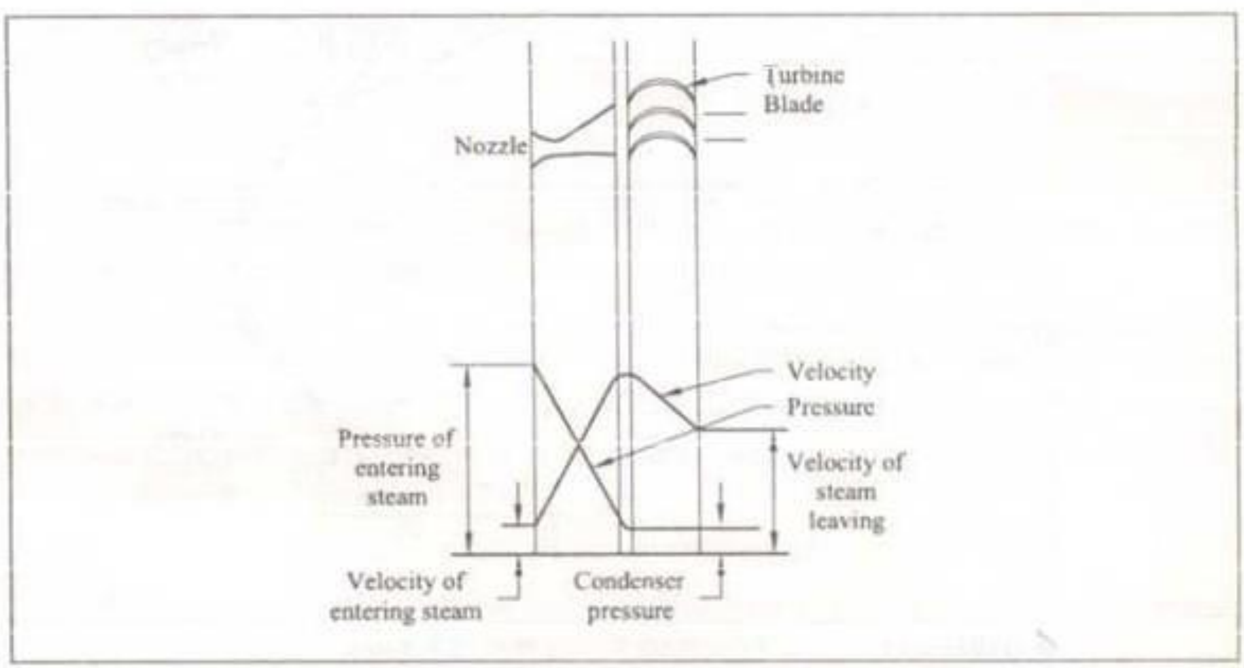
The surface of the blades is made very smooth to minimise losses due to friction. Mostly, the blades are bolted to the disc. Sometimes, the blades and disc are cast as a single unit. The function of blades is to change the direction of steam jet and hence momentum of the jet and so to produce a force which will rotate the blades.

The casing is air tight metallic case which houses the rotor and blades. It controls the flow of steam from blades to condenser and to safeguard the runner against any accident.

A De-lavel turbine consists of a single impulse wheel on which steam jets impinge from several nozzles arranged around the circumference. The blades are made symmetrical with angles of about 30° at inlet and exit. It has spherical bearings. It uses helical gears to reduce high rotational speed to a practical value.

PRESSURE AND VELOCITY VARIATION IN IMPULSE TURBINE

The fig. shows the variation of pressure and velocity of steam in a simple impulse turbine while it flows through nozzles and blades.



VARIATION OF PRESSURE AND VELOCITY IN A SIMPLE IMPULSE TURBINE

The entire pressure drop takes place in nozzles and the pressure remains constant while passing through the blades. As enthalpy drop takes place in nozzles the heat energy is converted into kinetic energy and so velocity of steam increases in the nozzle and is reduced gradually while flowing through the blades.

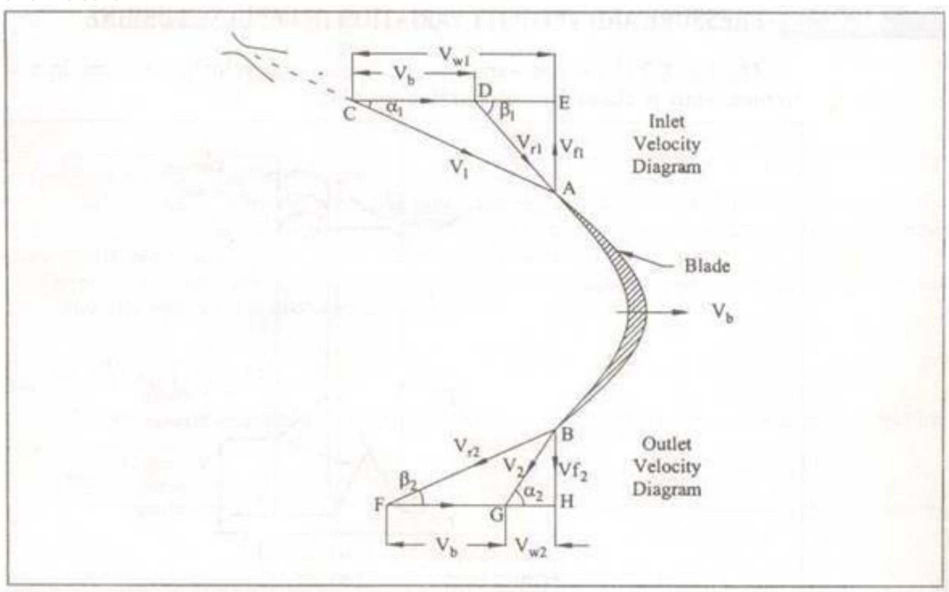
For economy and for maximum work; the speed of the blade should be one half of the velocity of steam. The small rotor employed in simple impulse turbine gives very high rotational speed about 30,000 RPM as most of the kinetic energy is absorbed by one set of moving blades only. Such high speeds can be used to drive the machines or generators with large reduction gearing arrangement.

From the velocity graph; it is clear that the velocity of steam leaving the blades consists of a large portion of velocity of steam leaving the nozzle. This results in loss of energy and this loss of energy due to higher exit velocity is called - carry over loss or leaving energy loss.

VELOCITY DIAGRAM FOR AN IMPULSE TURBINE

We should be able to estimate the propelling force that would be applied to a turbine rotor under any given set of conditions. With this, we can estimate the work done and hence the power. Since force is due to change of momentum mainly caused by change in the direction of flow of steam, it is essential to draw velocity diagram that shows how velocity of steam varies during its passage through the blades.

Velocity is a vector quantity as it has both magnitude and direction. So, we can represent velocity by a straight line and the length of the straight line indicates its magnitude and its direction is indicated by the direction of the line with reference to some fixed direction.



VELOCITY DIAGRAMS FOR AN IMPULSE TURBINE

The steam jet after leaving the nozzle, impinges on one end the blade, glides over the inside surface of the blade and finally leaves from the other edge.
Let Vb = Linear velocity of moving blade.

V1 = Absolute velocity of steam at inlet to moving blade i.e., exit velocity of nozzle. Vw1 = Tangential component of entering steam. Vw1 Also known as velocity of whirl at entrance.

Vr1 = Relative velocity of steam with respect to tip of blade at inlet. It is the vectorial difference between Vb and V1

Vf1=Velocity of flow = Axial velocity at entrance to moving blades. It is the vertical component of V1

α_1= Angle of nozzle = Angle which the entering steam makes with the moving blade at entrance - with the tangent of the wheel at entrance.

β_1 = Angle which the relative velocity makes with the tangent of the wheel - direction of motion of blade. It is also known as blade angle at inlet.

The above notations stand for inlet triangle.

V2, Vw2, Vf2, Vr2, α2, β2 are the corresponding values at the exit of the moving blades. They stand for outlet triangle.

The steam jet with absolute velocity V1 impinges on the blade at an angle of a1 to the tangent of the blade. The absolute velocity V2 can be considered as having two components. The tangential component called whirl component Vw1 = V1cosα1 is parallel to direction of rotation of blades and axial or flow component Vf1 = V sin α1 is perpendicular to the direction of rotation of blades.

The tangential component of the steam jet does work on the blade because it is in the same direction as the motion of the blade. The axial component doesn't work on the blades because it is perpendicular to the direction of motion of blade. It is responsible for the flow of steam through the turbine. Change of velocity in this component causes an axial thrust on the rotor. As the blade moves with a tangential velocity in peripheral direction, the entering steam jet will have relative velocity to the blades. If there is no friction loss at the blade, relative velocity at inlet is equal to relative velocity at outlet i.e., Vr1= Vr2.

As the steam glides over the blades without shock, the surface of the blade at inlet must be parallel to relative velocity Vr1. So, the moving blade at inlet must be inclined to the tangent of the blade at an angle β1. In other words, to avoid shock at entrance, vector Vr1 must be tangential to the blade tip at entry i.e, β1 must be equal to angle of blade at entrance. The blade is designed on this principle.

From the above analysis, following points are to be noted.

1. No expansion of steam takes place in the moving blades. The blades only deflect steam. This causes change in momentum and consequently force.

2. If the steam has to enter and leave the blades without shock, angle β1, should be angle of blade at inlet and angle β2 should be angle of blade at outlet. This is an essential condition.

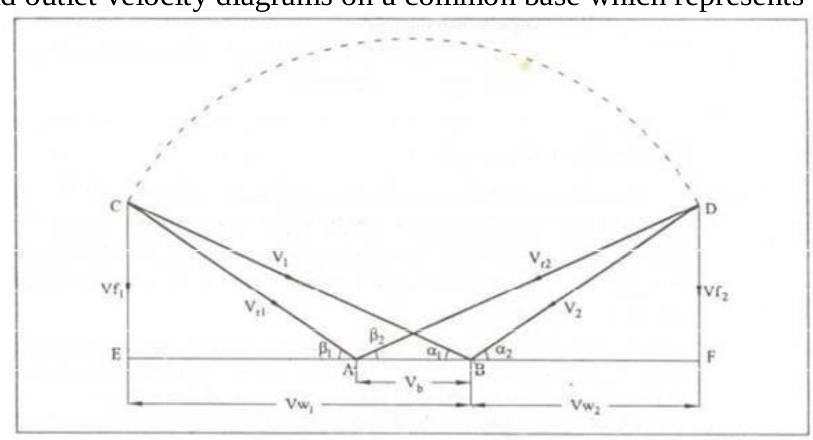
3. Since there is no pressure drop in the moving blades, the pressure on the two sides of the blades is equal.

4. α_1 is the outlet angle of nozzle. If steam has to enter the next nozzle ring without shock, its inlet angle must be equal to α2.

5. In a simple impulse turbine, the loss at exit is the whirl component at outlet - $V_2 \cos \alpha_2$. For minimum loss, this quantity should be minimum, i.e., α2 should be equal to 90°. In that case the turbine discharges axially and it is called axialturbine.

COMBINED VELOCITY DIAGRAM

To solve problems on turbines conveniently, it is common practice to combine both the inlet and outlet velocity diagrams on a common base which represents the blade velocity



Construction of combined velocity diagram :

1. First, draw a horizontal line and cut off AB equal to velocity of blade to some suitable

COMBINED VELOCITY DIAGRAM FOR AN IMPULSE TURBINE

scale.

2. From 5, draw a line BC at an angle a1, with AB. Cut off BC equal to V_1 to scale.

3. Join AC. It represents Vrl.

4. From A; draw a line AD at an angle β_2 with AB. With A as centre and radius equal to AC, draw an arc that meets the line through A at D such that AC = AD. Or $V_{r1} = V_{r2}$.

5. Join BD. It represents absolute velocity at exit to scale.

6. From C and D draw perpendiculars to meet the line AB produced at E and F.

7. Now; to scale,

EB = velocity of whirl at entrance.
BF = velocity of whirl at exit.
CE = velocity of flow at inlet.
DF = velocity of flow at outlet.

When friction is neglected, there will be no fall in steam pressure as it flows over the blades and $V_{r1} = V_{r2}$.

Also, when friction is absent,
$\beta_1 = \beta_2$ and $V_{f1} = V_{f2}$

FORCES ON BLADE AND WORK DONE BY BLADE

The work done may be found out from the change of momentum of steam jet during its flow
over the blades. As mentioned earlier, velocity of whirl is responsible for work on the blade.

1. Force on Rotor:

$$= m \times (V_{\omega 1} - V_{\omega 2}) \text{ Where}$$

According to Newton's second law of motion,

Tangential force on rotor = mass x tangential acceleration

$$F_t = \dot{m} (V_{\omega 1} \pm V_{\omega 2}) \text{ Newton.}$$

m = Mass rate of steam flow - kgs/sec.

Actually, Vw2 is negative as the steam is discharged in opposite direction to blade motion.

So, Vw1 ,and Vw2 are added together. Generally,

$$= F_t \, . \, V_b N - m/sec$$

$$= m \, . \, (V_{\omega 1} \pm V_{\omega 2}) \, . \, V_b N - m/sec$$

$$= m \, . \, (V_{\omega 1} +\_ V_{\omega 2}) \, . \, V_b \text{ watts.}$$

Positive sign is to be used when Vw2 and Vb are in opposite direction as shown above and negative sign is to be used when Vw2 and Vb are in same direction.

2. Work done as Blade:

Work done = force x distance

= Tangential force x distance moved in unit time in the direction of force.

$$\eta_{bl} = \frac{\text{Work done on blade}}{\text{Energy supplied blade}}$$

$$= \frac{m \cdot (V_{\omega 1} \pm V_{\omega 2} \cdot V_b}{\frac{1}{2} m V_1^{\,2}}$$

$$= \frac{2 V_b \cdot (V_{\omega 1} \pm V_{\omega 2})}{V_1^{\,2}}$$

3. Power Developed by the Turbine:

Power = Rate of doing work

(1 Watt= 1 N-m/sec)

This power is known as Rim power or diagram power to distinguish it from shaft power.

4. Axial Thrust on Rotor:

Axial force Fa= Mass x Axial acceleration

= Mass x change in velocity of flow

This axial force must be balanced or must be taken by a thrust bearing. EFFICIENCIES

The following efficiencies are common to both impulse and reaction turbines :

1. Blading or diagram efficiency.

2. Gross or stage efficiency.

3. Nozzle efficiency

1. **Blading or Diagram Efficiency**:

It is defined as the ratio of work done on blades to energy supplied to blades.

Let V1 = Absolute velocity of steam at inlet —m/sec

m = Mass of steam supplied — kg/sec.

Energy of steam supplied to blade = 1/2*m*V_{12}

Work done on blade = m . (Vw1 ± Vw2) . Vb J/sec Diagram or blading efficiency

This is called diagram efficiency because the quantities involved in it are obtained from velocity diagram

2. **Gross or Stage Efficiency**:

A stage consists of a set of nozzles and a row of moving blades and so, stage efficiency includes the performance of nozzles also.

Stage efficiency is defined as the ratio of work done on blades per kg of steam to total energy supplied per stage per kg of steam. If h1 and h2 represent before and after expansion of steam through the nozzles, then the enthalpy drop (h1- h2) is the enthalpy drop through a stage, i.e., the heat energy (h1 - h2) is the energy supplied per stage per kg of steam.

$$\text{Stage efficiency} = \frac{\text{Work done on blade}/kg \text{ of steam}}{\text{Total energy supplied}/\text{stage}/kg \text{ of steam}}$$

$$\eta_{\text{stage}} = \frac{(V_{\omega 1} \pm V_{\omega 2} \cdot V_b}{h_1 - h_2}$$

3. Nozzle Efficiency:

It is defined as the ratio of energy supplied to blades per kg of steam to total energy supplied per stage per kg of steam.Energy supplied to blades per kg of steam = ½*m*V_{12} Total energy supplied per stage per kg of steam = (h1-h2)

$$\text{Nozzle efficiency} = \frac{\text{Energy availbale at entrance}/kg}{\text{Enthalpy drop through a stage}/kg \text{ of steam}}$$

$$\eta_{\text{nozzle}} = \frac{\frac{1}{2} V_1^{\ 2}}{(h_1 - h_2)}$$

$$= \frac{V_1^{\ 2}}{2\,(h_1 - h_2)}$$

Stage efficiency = blade efficiency x nozzle efficiency.

Energy converted to heat by blade friction= Loss of kinetic energy during flow over the blades

$$= \frac{1}{2} m \cdot \left(V_{r1}{}^2 - V_{r2}{}^2\right) J$$

EFFECT OF BLADE FRICTION

In an impulse turbine, the relative velocity remains same as steam passes over the blades if friction is neglected. In actual practice, the flow of steam the blades is resisted by friction. The effect of this friction is to reduce the relative velocity of steam while passing over the blades. Generally, there is a loss of 10-15% in relative velocity. Owing to friction in blades. Vr2 is less than Vr1 and we may write $V_{r2} = K \cdot V_{r1}$ $\quad K = \frac{V_{r2}}{V_{r1}}$

The ratio of Vr2 to Vr1 is called blade velocity coefficient or coefficient of velocity friction factor K. The effect of blade friction is to reduce relative velocity at outlet and consequently Vw2 This in turn will cause reduction in work done and blade efficiency. Depending upon the shape of the blades, value of K varies from 0.75 to 0.85

CONDITION FOR MAXIMUM EFFICIENCY OF AN IMPULSE TURBINE

Efficiency of an impulse turbine is

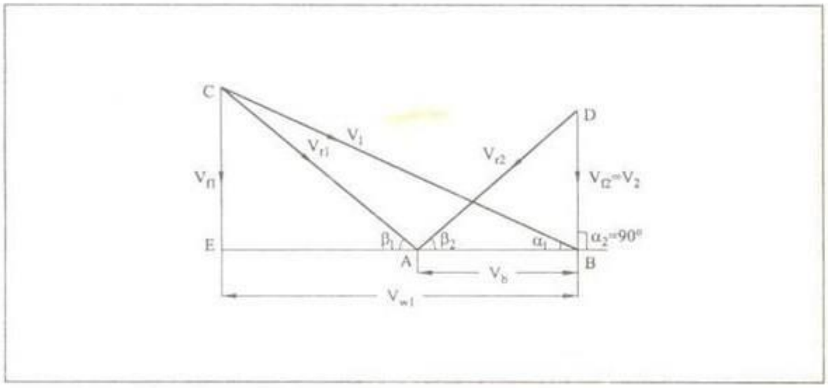
$$\eta_{bl} = \frac{2\,(V_{\omega 1} \pm V_{\omega 2}) \cdot V_b}{V_1{}^2}$$

The blading efficiency will be maximum when V1 is minimum. From combined velocity diagram, we can observe that, value of V1, will be minimum when α2=90°. So, for maximum efficiency, the steam should leave the turbine blades at right angles to their motion.

Also, for maximum efficiency, $V_{w2}=0$

$$\eta_{bl(max)} = \frac{2\, V_{\omega 1} \cdot V_b}{V_1{}^2}$$

COMBINED VELOCITY DIAGRAM FOR MAXIMUM EFFICIENCY Neglecting blade friction, $V_{r1} = V_{r2}$



When we consider friction; $V_b = \frac{1}{2} V_{\omega 1}$

$$[\because \quad \Delta EAC = \Delta ADB]$$

$$\eta_{bl(max)} = 2 \,.\, V_{\omega 1}\, 0.5 \frac{V_{\omega 1}}{V_1^{\,2}}$$

$$= \frac{V_{\omega 1}^{\,2}}{V_1^{\,2}}$$

$$\left[\because \quad \cos \alpha_1 = \frac{V_{\omega 1}}{V_1}\right]$$

$$\eta_{bl(max)} = (1 + KC) \cdot \frac{\cos^2 \alpha_1}{2}, \text{ where}$$

$$K = \text{Blade velocity coefficient} = \frac{V_{r2}}{V_{r1}}$$

$$C = \frac{\cos \beta_2}{\cos \beta_1} = A \text{ constant}$$

Here, we introduce another term — blade speed ratio or blade velocity ratio ρ which is defined as the ratio of speed of blade to absolute velocity at inlet.

$$\rho = \frac{V_b}{V_1}$$

This is a very important factor in the design of turbines and efficiency of a turbine depends largely on the value of ρ

Optimum Value of Blade Speed Ratio:

$$\rho_{opt} = \frac{\cos \alpha_1}{2}$$

$$\eta_{bl} = 2(1 + KC)(\rho \cos \alpha_1 - \rho^2)$$

For maximum blade efficiency.

$$\frac{d\eta_{bl}}{d\rho} = 0 = 2(1 + KC)(\cos \alpha_1 - 2\rho)$$

For maximum blade efficiency,

$2(1 + KCT)(\cos \alpha 1 - 2\rho) = 0$

$\cos \alpha 1 - 2\rho = 0$

$$\rho_{opt} = \frac{\cos \alpha_1}{2}$$

Maximum work done : Wmax = 2 . V_{b2}

For maximum work developed per kg of steam or for maximum efficiency, the blade velocity should be approximately half of absolute velocity of steam jet coming out from nozzle.

COMPOUNDING OF IMPULSE TURBINES

We already know that, in impulse turbines, the entire pressure drop takes place in nozzles only. If the entire pressure drop from boiler pressure to condenser pressure (say 125 bar to 1 bar) is carried out "in one stage (one set of nozzles) only, then, the velocity of the steam will be extremely high. It will make the turbine rotor to run at very high speeds (upto 30.000 RPM). In practice, such a high speed of a turbine is of no use and will have number of disadvantages. The leaving loss also becomes high. It is usually necessary to reduce the speed by gearing which will be of undue proportions.

So, it is essential to make improvement in the impulse turbine to make it more efficient, practical - to reduce the high speed of the rotor to practical limits. This is achieved by making use of more than one set of nozzles, blades and rotors in series keyed to a common shaft so that either pressure of steam or its velocity is absorbed in stages and in doing so, the speed gets reduced. This also reduces leaving loss. This process of absorbing pressure or velocity of steam in stages to reduce the speed of the turbine rotor is called - compounding.

There are three important methods of compounding:

1. Pressure compounding.

2. Velocity compounding.

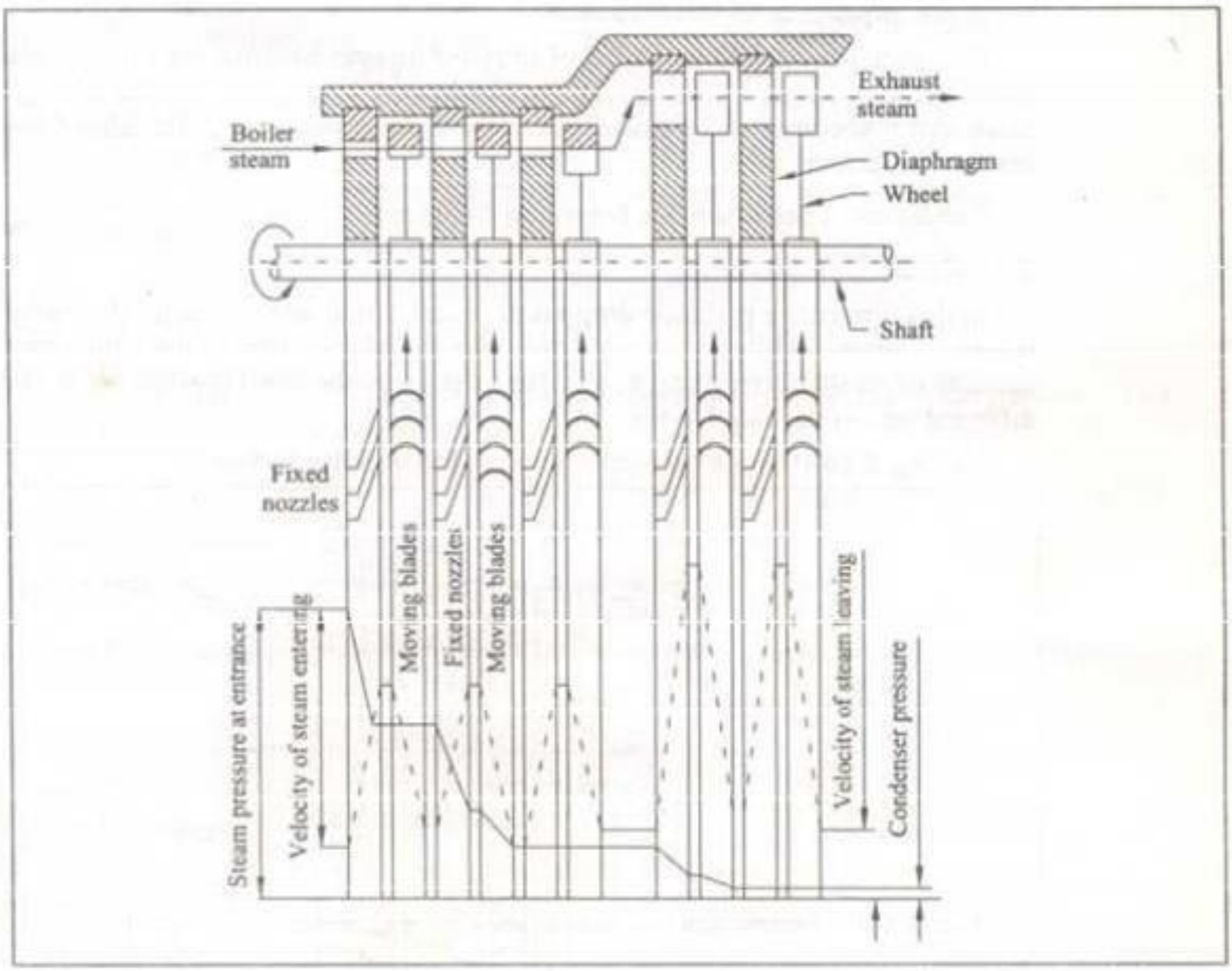
3. Pressure - velocity compounding.

1. Pressure Compounding:

In this, the whole expansion of steam is carried out in a number of steps by employing a number of simple impulse turbines in series on same shaft as shown in fig

We can arrange a number of simple impulse turbines in series on same shaft allowing exhaust steam from one turbine to enter the nozzles of next turbine. Then, each of the simple impulse turbine is termed as - stage of the turbine, each stage containing a set of nozzles and blades.

This is equivalent to splitting the whole pressure drop into a series of smaller pressure drops and so it is called - pressure compounding. The total pressure drop of steam doesn't take place in the first set of nozzles but divided equally among all nozzle sets and the pressure remains constant while flowing over the moving blades



The nozzles are usually fitted into partitions termed as diaphragms which separate one wheel chamber from the next.

The steam from boiler pressure is passed through the first set of nozzles (A number of nozzles are arranged around the circumference of the wheel. All nozzles for one wheel constitute one set of nozzles); where only a small pressure drop occurs with an increase in velocity of steam. While flowing over the first set of moving blades, pressure remains constant but velocity decreases. This constitutes one stage. A stage consists of a set of fixed nozzles and a set of moving blades. A stage itself is a simple impulse turbine.

The steam from first set of moving blades enters the second stage - into second set of nozzles where its pressure is further reduced. Consequently, the velocity increase again. Now, the steam enters the second set of moving blades in which pressure remains constant but velocity decreases.

This process is repeated in the remaining stages also until condenser pressure is reached.

As pressure drop per stage is reduced, the velocity of steam is reduced which in turn reduces the blade or rotor velocity. The speed of the turbine can be reduced further by increasing number of stages. The leaving velocity of the last stage of the turbine is much less compared to simple impulse or De-lavel turbine.

This is the most efficient type of impulse turbine because the ratio of blade velocity, to steam velocity remains constant. But to obtain very low speed, number of stages required are more and it becomes more expensive. Now-a-days, pressure compounded impulse turbines are not being used. Rateau and Zoelly turbines belong to this group.

2. Velocity Compounding:

In this, the entire pressure drop takes place in one set of nozzles thereafter, the pressure remains constant while the steam flows over the blades. Due to the entire pressure drop, the velocity of steam becomes high, and this velocity is absorbed in steps while steam flows over different sets of moving blades.

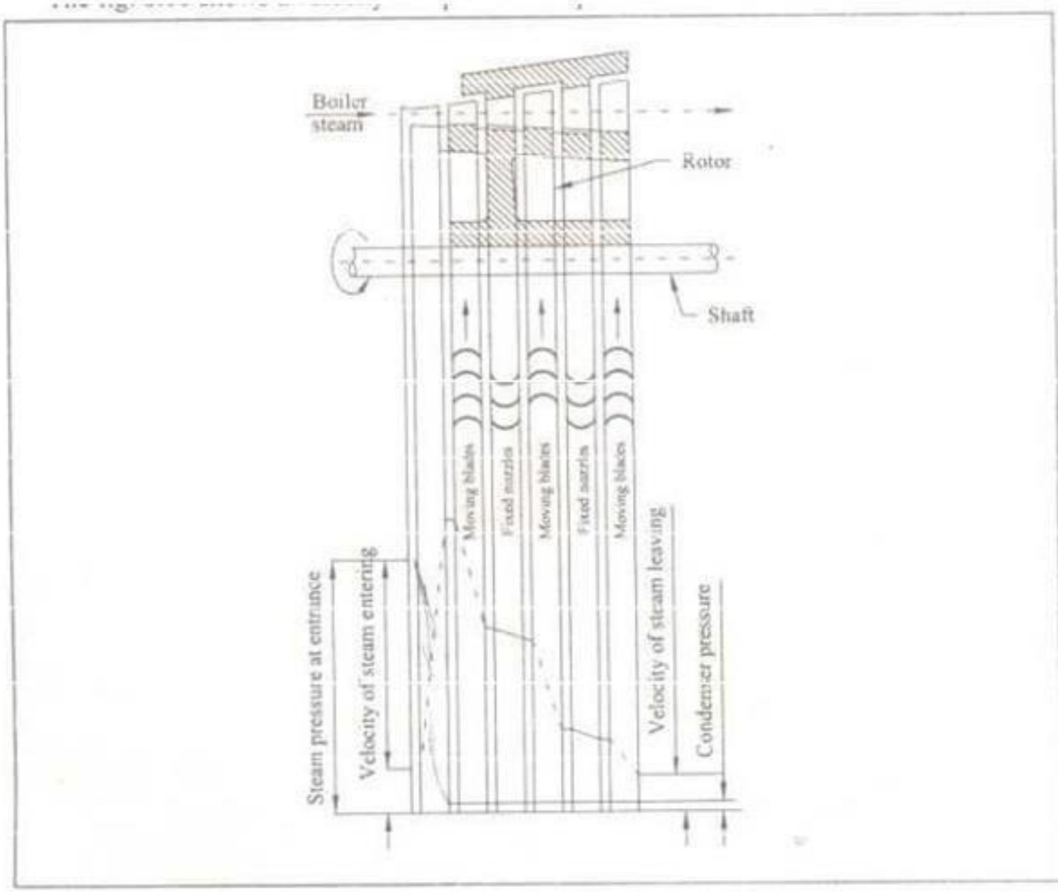
Here, the turbine consists of a set of nozzles and a wheel fitted with two or more rows of moving blades. There are fixed or guide blades arranged between moving blades and set in reverse manner.

The expansion of steam takes place in the set of nozzles from boiler pressure to condenser pressure. The resulting high velocity of steam is utilized by as many sets of rotor blades as necessary.

A portion of initial high velocity of steam is absorbed by the first set of moving blades. The steam from first set of moving blades comes out with a fairly high velocity. It then enters the fixed or stationary or guide blades which change the direction of steam and direct the steam into second set of moving blades; without affecting the velocity appreciably. There is slight drop in velocity in guide blades due to friction. While passing through the second set of moving blades, steam suffers a change of momentum and gives up another portion of its velocity -kinetic energy to the rotor.

The process is repeated and the steam finally enters the condenser from the last set of moving blades.

The entire pressure drop takes place in the nozzles only and no pressure drop occurs in fixed (guide) blades or moving blades. This method of velocity compounding is known as - Curtis principle and Curtis turbine is an example of velocity compounded impulse turbine.



3. Pressure - Velocity Compounding:

In this, both the principles of pressure compounding and velocity compounding are used. Total pressure drop of steam is divided into stages and velocity in each stage is also compounded.

This type allows bigger pressure drop in each stage and hence less number of stages are required. So, for a given pressure drop, this is more compact than a pressure compounded turbine.

In this turbine, each stage has a set of nozzles, two or more rows of moving blades and one or more rows of guide blades both placed alternately. Each stage is separated from adjacent stage by a diaphragm containing a nozzle.

In this turbine, the whole pressure drop takes place in different sets of nozzles, i.e., whole pressure drop doesn't take place on set of nozzles but divided into small drops. So, it is pressure compounded.

While flowing over different sets of moving blades in different stages, the velocity is reduced. So, it is velocity compounded.

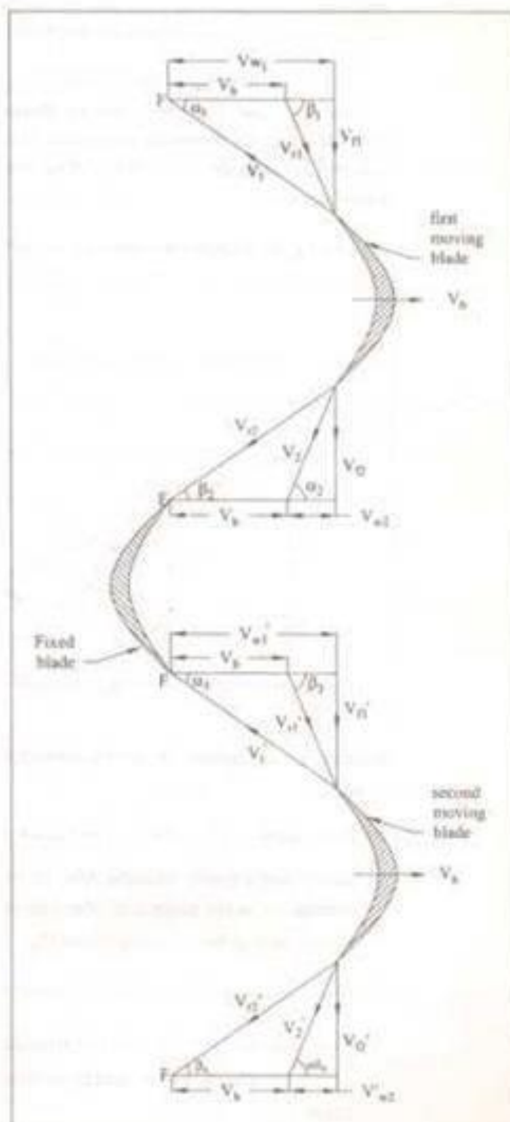
The diameter of this turbine is increased at each stage to allow increasing volume of steam at lower pressures. This type of compounding is used in Curtis turbine.

VELOCITY DIAGRAM FOR VELOCITY COMPOUNDED IMPULSE TURBINE

In a single stage turbine, steam after leaving the nozzle impinges on one end of the blades, glides over the inner surface of the blades and leaves the blades at the other end. A velocity compounded impulse turbine consists of one set of nozzles, two or more sets of moving blades and guide blades. If we consider two rows or two sets of moving blades only, then, steam after expansion in the nozzles, enters the first set of moving blades and after leaving the first set of moving blades, enters first set or first row of fixed or guide blades

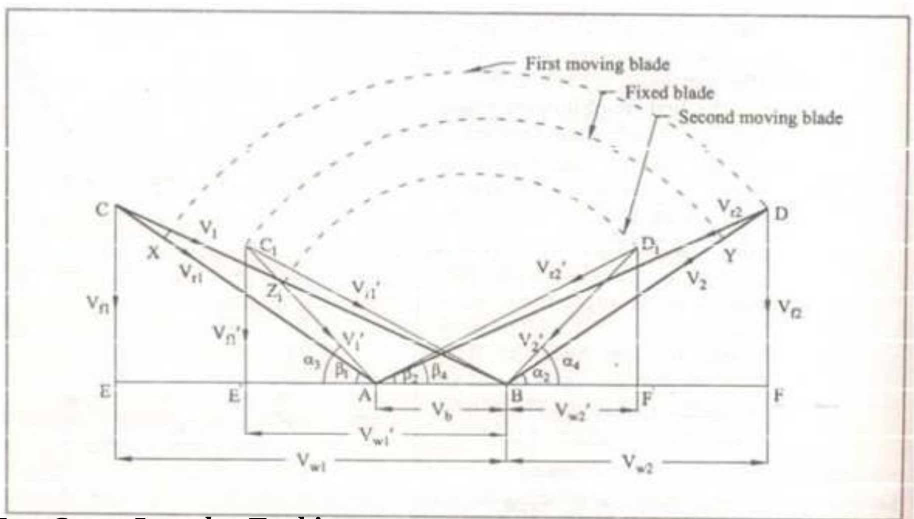
There is no enthalpy drop in the guide blades. Their function is to change the direction only.

But, there may be slight reduction in velocity due to friction. The guide blades are also called as fixed blades as they do not rotate but arc attached to the casing. The moving blade rows are attached to the rotor.



The blade velocity is constant for both the stages as there are mounted on same shaft. The absolute velocity at exit from the first moving blade is the entry velocity to the fixed blade. Similarly, the exit velocity from the fixed blade is the entry velocity to the second moving blade.

We can draw combined velocity diagrams for the first and second row of moving blades individually with a similar procedure as given for a single stage impulse turbine. We can combine these individual velocity diagrams to obtain final combined velocity diagram for the whole turbine



Combined Velocity Diagram for Two Stage Impulse Turbine

Calculations:

I. Total work done per kg of steam passing through both stages:

Wt = work done in first moving blade set + work done in second moving blade set.

$$= 2\,V_b\,.\,(V_1 \cos\alpha_1 - V_b) + 2\,V_b\,(V_1\,.\cos\alpha_1 - 3\,V_b).$$

$$= 4\,V_b\,(V_1\,.\cos\alpha_1 - 2\,V_b).$$

Blading or diagram efficiency for a two stage turbine:

$$\eta_{bl} = \frac{\text{Work done}}{\text{Energy supplied}}$$

$$= \frac{\omega_t}{\frac{1}{2}\,.\,m\,.\,V_1^{\,2}} = \frac{\omega_t}{\frac{1}{2}V_1^{\,2}} \qquad [m = 1\ kg]$$

$$= \frac{4\,V_b\cdot(V_1\cos\alpha_1 - 2\,V_b)\,.\,2}{V_1^{\,2}}$$

$$= \frac{8V_b}{V_1^{\,2}}(V_1\cos\alpha_1 - 2\,V_b)$$

$$= 8\,.\,\frac{V_b}{V_1}\left(\cos\alpha_1 - 2\,.\,\frac{V_b}{V_1}\right)$$

$$= 8\,\rho\,(\cos\alpha_1 - 2\,\rho) \text{ where}$$

$$\rho = \frac{V_b}{V_1} = \text{blade speed ratio.}$$

For maximum efficiency, $\rho_{opt} = \dfrac{\cos\alpha_1}{4}$

Maximum efficiency $\eta_{bl(max)} = \cos^2\alpha_1.$

If there are 'n' number of rows, then optimum blade speed ratio $\rho_{opt} = \dfrac{\cos\alpha_1}{2n}$

Maximum efficiency implies minimum rejection of energy which is obtained when discharge is axial i.e., $\alpha 4 = 90°$.

For any given blade speed, a two row wheel (two sets of moving blades) can utilize four times the enthalpy drop of a simple impulse turbine

$$\omega_t = 4\, V_b \,.\, (V_1 \cos \alpha_1 - 2\, V_b)$$

3. Maximum work done:

$$\text{Optimum value of } \rho = \frac{\cos \alpha_1}{4}$$

We know that work done

$$\frac{V_b}{V_1} = \frac{\cos \alpha_1}{4}$$

$$V_1 = \frac{4 V_b}{\cos \alpha_1}$$

$$\therefore \qquad \omega_{t(max)} = 4 V_b \left(\frac{4 V_b}{\cos \alpha_1} \,.\, \cos \alpha_1 - 2 V_b \right)$$

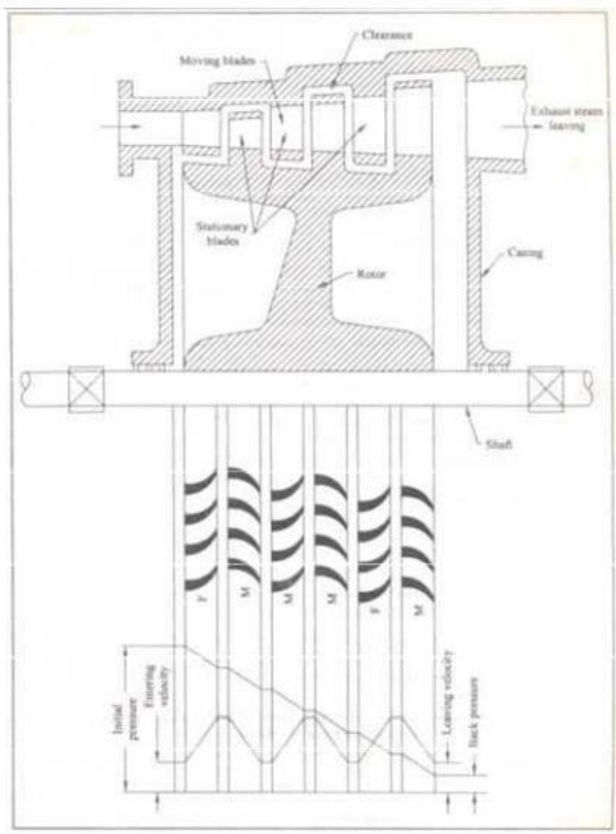
$$= 8 V_b^{\,2}$$

From the velocity diagram;

Total work done $= m\,(EF + E'\,F')\,.\,V_b$.

REACTION TURBINE

In a reaction turbine, steam enters the rotor under pressure and flows over the blades. While gliding, steam propels the blades and makes them to move. The rotor is rotated by reactive forces of steam jets. The motion of blades is similar to recoil of a gun. Pure reaction turbine is not possible in actual practice and all turbines employ both impulse and reaction principles. The driving force is partly impulsive and partly reactive.



In this turbine there are number of rows of moving blades fixed to shaft with equal number of fixed blades attached to the casing. The fixed blades are fixed in reverse manner compared to moving blades and corresponds to nozzles referred to in impulse turbine.

The impulse turbines are partial admission turbines as nozzles do not occupy the complete circumference leading into blade annulus. In impulse - reaction turbines, instead of a set of nozzles, steam is admitted for the whole circumference and so there is full, complete admission. A set of fixed blades are positioned at the entrance in place of nozzles.

In passing through the first row of fixed blades, steam suffers a small drop in pressure and hence its velocity increases. It then enters the first row of moving blades and as in impulse turbine, suffers a change in direction and so results in momentum. This results in an impulse on blades.

But, here the passage of moving blades is so designed that there is a small pressure drop in moving blades giving rise to increase in velocity - kinetic energy. This kinetic energy gives rise to a reaction in the direction opposite to that of added velocity. Thus, the driving force is vector sum of impulse and reaction forces. Normally this type of turbine is known as - Reaction turbine. It is also called as - Parson's reaction turbine.

In this turbine, the pressure drop takes place in both fixed and moving blades and blade passages in both are of convergent nozzles shape. The steam velocities in this turbine are comparatively low and the maximum value is about equal to blade velocity. In this turbine, as pressure falls, specific volume increase and so, height of blades is increased progressively. This type of turbine is very successful in practice and popular in power plants.

PARSON'S REACTION TURBINE

A Parson's reaction turbine is the simplest type of reaction turbine and is commonly used. The main components of it are:

1. Casing.
2. Guide mechanism.
3. Runner.
4. Draft tube.

The casing is an air tight metallic case in which steam from boiler under high pressure is distributed around the fixed blades which are positioned at the entrance. The casing is so designed that steam enters the fixed blades with uniform velocity.

The guide mechanism consists of fixed or guide blades. They allow the steam to enter the rotor without shock and they allow required quantity of steam to enter the turbine. The guide blades may be opened or closed by a regulating shaft which allows steam to flow according to the need.

The runner consists of moving blades. These blades are designed properly to allow steam to enter and leave the blades without shock. The steam after passing through the rotor flows to condenser through a draft tube. It minimises losses due to eddies.

In impulse turbines, steam pressure on both sides of moving blades is same and axial thrust is negligible. But, in a reaction turbine, this thrust is considerable due to fall of pressure within the blades and difference between blade sizes in various steps. (To accommodate increased volume, height of blades in increased progressively). Thrust bearings are used to balance this thrust.

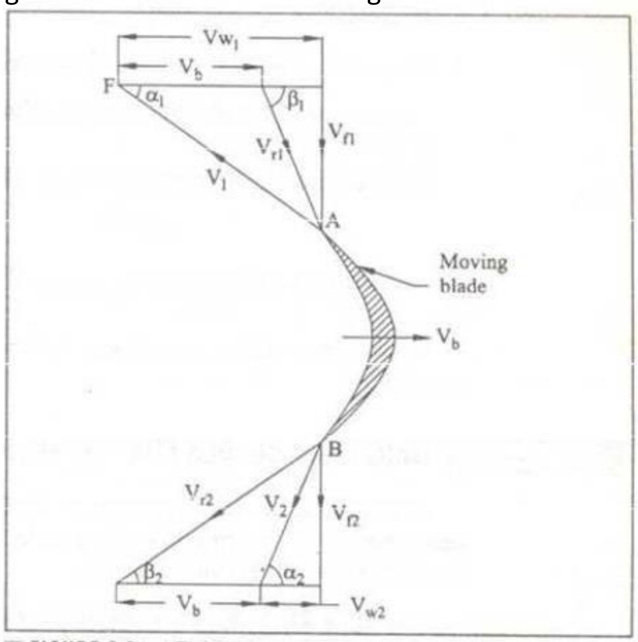
In a reaction turbine, the pressure is reduced in both fixed and moving blades. The velocity increases in fixed blades and reduces while passing through moving blades.

VELOCITY DIAGRAM OF A REACTION TURBINE

The fig shows the inlet and outlet velocity diagrams for moving blade of a reaction turbine.

The steam jet enters the blades along CA with absolute velocity V1 at an angle a1 in the direction of motion of blade. DA represents relative velocity of steam at entrance Vr1 with respect to blade. CE represents velocity of whirl at entrance Vw1 which causes the work done. EA represents axial component of absolute velocity V1 known as velocity of flow at entrance Vf1. It causes steam to flow through the blades and also exerts an axial thrust on the rotor. It does not work on the blade. Mean velocity of blade is represented by $V_{b.}$

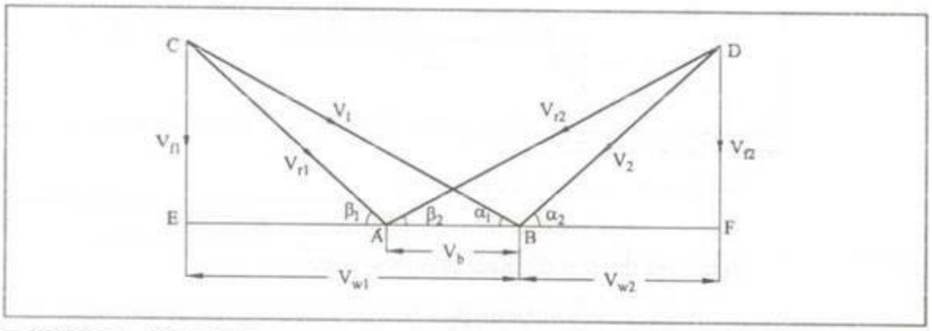
The steam jet glides over and leaves the blades at the other end. V2Vr2, Vf2, Vw2, α_1, β_2 represents corresponding values at outlet of the moving blade.



$$\eta_{bl} = \frac{\text{Rate of doing work}}{\text{Energy input}}$$

$$= \frac{2\rho\,(2\cos\alpha_1 - \rho)}{1 - \rho^2 + 2\rho\cos\alpha_1}$$

In Parson's reaction turbine, both the fixed and moving blades are made identical. SO, $\alpha_1 = \beta_2$ and $\beta_1 = \alpha_2$. So, the velocity diagram for Parson's reaction turbine will be symmetrical about vertical centre line and Vf1 = Vf2; V1 = Vr2; V2 =Vr1.



Blading or Diagram Efficiency of Parson's Turbine:

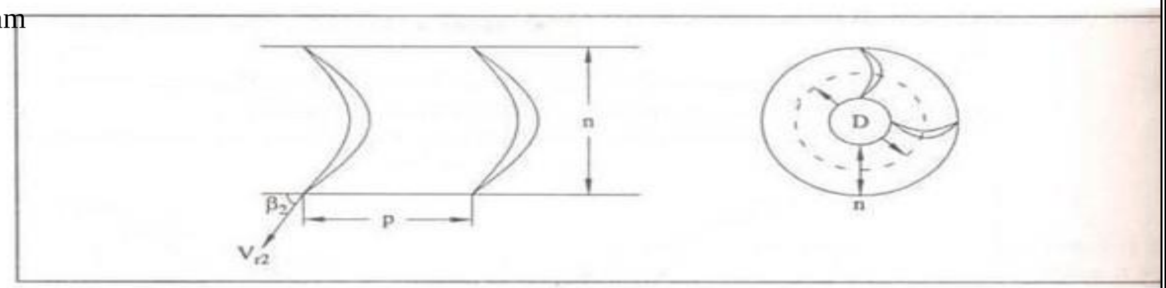
$$\rho = \rho_{opt} = \cos \alpha_1.$$

$$\text{Maximum efficiency} = \eta_{bl(max)} = \frac{2\cos^2 \alpha_1}{1 + \cos^2 \alpha_1}$$

In reaction turbines, both fixed and moving blades act as nozzles as pressure drop takes place in both

HEIGHT OF BLADES FOR A REACTION TURBINE:

In designing blades, the height of blade plays an important role. In a reaction turbine, the steam enters the moving blades over the whole circumference. So, the area through which steam flows is always full of steam



$$A_2 = \frac{L}{p} h \,.\, (p \sin \beta_2 - t)$$

where

L = Effective length of area over which steam flows.

p = pitch of blades.

h = height of blades.

t = thickness of blades.

n = number of blades per row $= \frac{\pi D}{p}$. Where

D = mean diameter of rotor drum

Neglecting blade thickness

Similarly for moving blades inlet; $\pi D.h\,.\,V_{r2}\,.\sin\beta_2 = \pi D.h.\,V_{f2} = m.v_2$

$$\pi D.h.V_1 \sin\alpha_1 = \pi D.h.\,V_{f1} = m.v_1$$

For a Parson's reaction turbine, $\alpha 1 = \beta 2$, $V1 = Vr2$ and v - specific volume of steam is assumed to be constant, the blade height for fixed as well as moving blades is same in a stage.

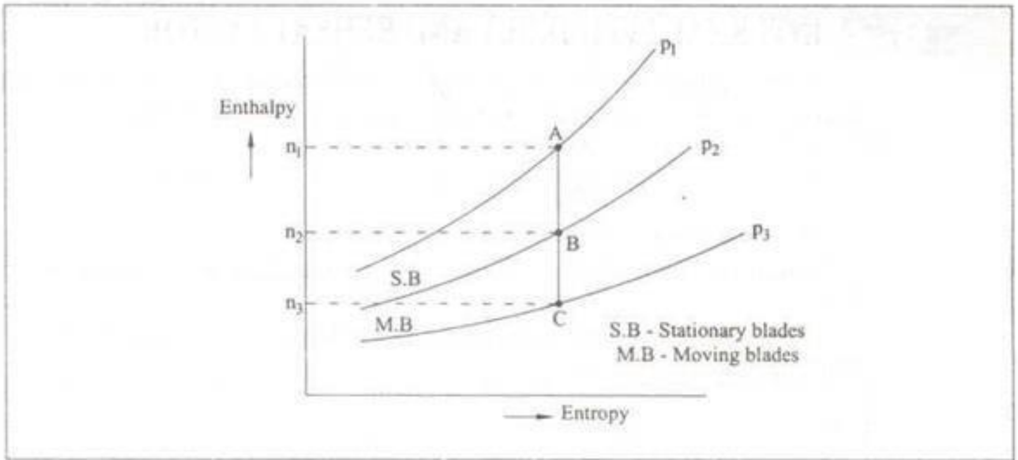
DEGREE OF REACTION

The energy transfer is by change of dynamic pressure and by change of dynamic pressure in rotor passage. The ratio of energy transfer by means of change of static pressure in the rotor to total energy static pressure in the rotor to total energy transfer in the rotor is called the degree of reaction.

In case of turbines, a stage consists of a set of nozzles (or fixed blades) and rotor having moving blades. The fig shows the H-S diagram for isentropic flow of steam through stationary (fixed) and moving blade.

$$\therefore \quad \text{Degree of reaction} = \frac{\text{Enthalpy drop in moving blades}}{\text{Enthalpy drop in stage}} = \frac{h_2 - h_3}{h_1 - h_3}$$

The degree of reaction is defined as the ratio of enthalpy drop in moving blades to total enthalpy drop in a stage.



FLOW OF STEAM IN A TURBINE

In an impulse turbine, the total enthalpy drop takes place in nozzles only and no enthalpy drop takes place in the rotor. (Both enthalpy and static pressure remain constant in rotor passage). So for impulse turbines, degree of reaction is zero.

In case of reaction turbines, enthalpy drop is divided in fixed blades and moving blades. (Pressure changes in both fixed and moving blades). So, for a reaction turbine,

$$D.R = \frac{\text{Enthalpy drop in moving blades}}{\text{Enthalpy drop in moving blades + Enthalpy drop in fixed blades}}$$

$$= \frac{h_2 - h_3}{(h_1 - h_2) + (h_2 - h_3)} = \frac{(h_2 - h_3)}{(h_1 - h_3)}$$

$$(h_1 - h_2) = \frac{1}{2}\left(V_1^{\,2} - V_2^{\,2}\right) J/kg \text{ of steam.}$$

$$(h_2 - h_3) = \frac{1}{2}\left(V_{r2}^{\,2} - V_{r1}^{\,2}\right) J/kg \text{ of steam.}$$

For a Parson's reaction turbine;

$$V_1 = V_{r2} \text{ and } V_2 = V_{r1}.$$

Degree of reaction = 0.5 = 50%.

So, a Parson's reaction turbine is also known as 50% reaction turbine. In it, half of the total enthalpy drop takes place in fixed blades and the remaining half of the enthalpy drop takes place in moving blades.

COMPARISON OF IMPULSE AND REACTION TURBINES

The following are the important differences between impulse and reaction turbines.

1 Steam flows through nozzles and impinges on moving blades. Steam (lows through fixed blades and then flows over moving blades

2 pressure drop takes place in nozzles and pressure remains constant in moving blades. Pressure drop takes place both in fixed and moving blades.

3 Blade passage is of constant cross sectional area as there is no expansion. Due to expansion, the blade passage is of variable cross sectional area. 4 Blade shape is profile type and easy to manufacture. Blade shape is aerofoil type and difficult to manufacture.

5 Nozzles are located in diaphragms and rotor is disc or wheel type.

Fixed blades attached to casing serve as nozzles and rotor construction is drum type.

6 Admission of steam is partial over the circumference Admission of steam is full over the whole circumference.

7 Because of large pressure drop, number of stages is less. Because of small pressure drop, for same pressure drop, larger numbers of stages are required. Reaction turbines are multi stage turbines only.

8 Because of large pressure drop, velocity of steam and velocity of blade are higher.

Because of small pressure drop, the velocity of steam and velocity of blade are lower.

9 The diagram efficiency decreases rapidly with change in designed blade speed ratio.

Greater working range is possible.

10 Suitable for small power requirements. Suitable for medium and higher power requirements

11 Occupies less space per unit power. Occupies more space per unit power.

12 Degree of reaction is zero Degree of reaction is greater thanzero

The choice of stages in a steam turbine depends upon relative importance of capital cost and turbine efficiency. Higher turbine efficiency results in lower steam consumption and smaller size of boiler and condenser which in turn reduces the capital cost of total steam power plant.

Numerical Problems – Steam Turbines

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.

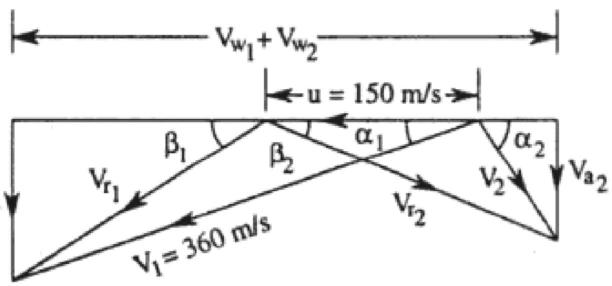


Fig. 14.10

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

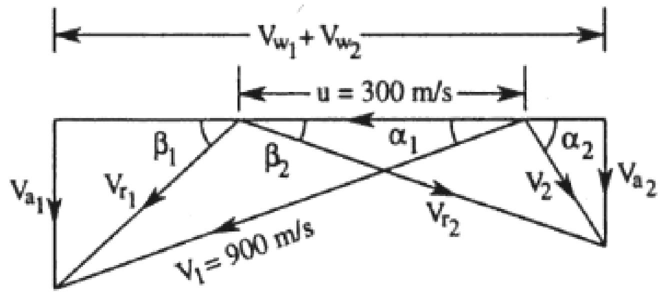
(b) Power output for a steam flow of 1 kg/s

$$= \frac{(V_{w_1} + V_{w_2})u}{1000} \text{ kW} = \frac{348 \times 150}{1000} = 52.2 \text{ kW}$$

(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 $= \dfrac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 300 \times 927.5}{(900)^2} = 0.688 = 68.8\%$

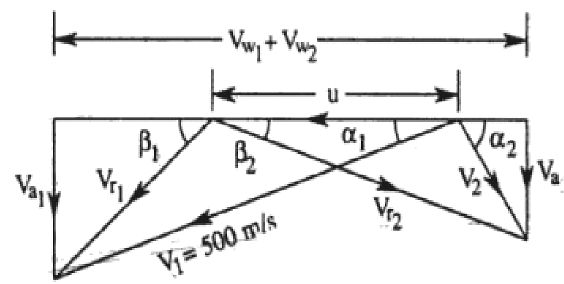
Maximum blade efficiency, $\eta_{b(\max)} = \cos^2\alpha_1 = \cos^2 20 = 0.88 = 88\%$

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

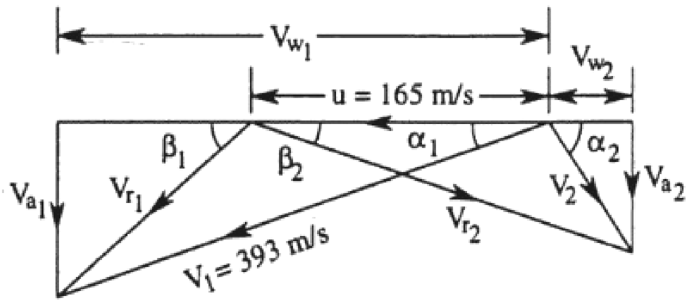
Work done per kg of steam $= m(V_{w_1} + V_{w_2})u = 1 \times 555 \times 200 = 111000$ Nm/s

(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[illegible]n the blades, (b) tangential thrust on the blades, (c) axial thrust on the blades, (d) power [illegible]eveloped in blades, and (e) blading efficiency



Solution: Given, $d = 105$ cm $= 1.05$ 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, $u = \dfrac{\pi d N}{60} = \dfrac{\pi \times 1.05 \times 3000}{60} = 165$ m/s

Absolute velocity of steam at inlet, $V_1 = \dfrac{165}{0.42} = 393$ m/s

The velocity diagram is shown in Fig. 14.14.
The following results are obtained from the velocity diagram:
$\beta_1 = 30.2°$; $V_{w_1} + V_{w_2} = 389$ m/s; $V_{a_1} = 122$ m/s; $V_{a_2} = 93$ m/s

(a) Resultant thrust $= \sqrt{(3112)^2 + (232)^2} = 3120.56$ N

(b) Tangential thrust $= m(V_{w_1} + V_{w_2}) = 8 \times 389 = 3112$ N

(c) Axial thrust $= m(V_{a_1} - V_{a_2})$ N $= 8(122 - 93) = 232$ N

(d) Power developed $= \dfrac{m(V_{w_1} + V_{w_2})u}{1000}$ kW $= \dfrac{8 \times 389 \times 165}{1000} = 513.48$ kW

(e) Blading efficiency $= \dfrac{2u(V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 165 \times 389}{(393)^2} = 0.83 = 83\%$.

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%.

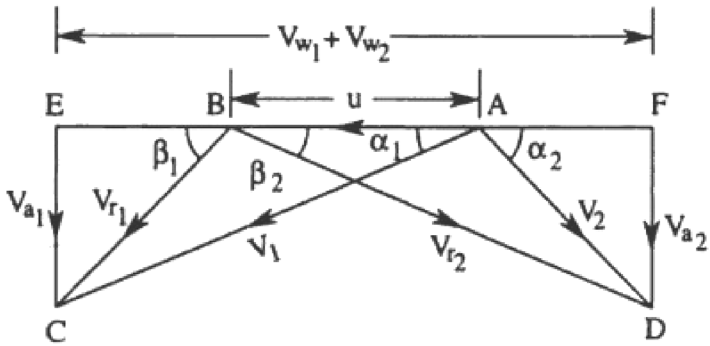


Fig. 14.31

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.

Stage efficiency $= \dfrac{\text{Work done per stage}}{\text{Enthalpy drop per stage}}$

$\therefore$ 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

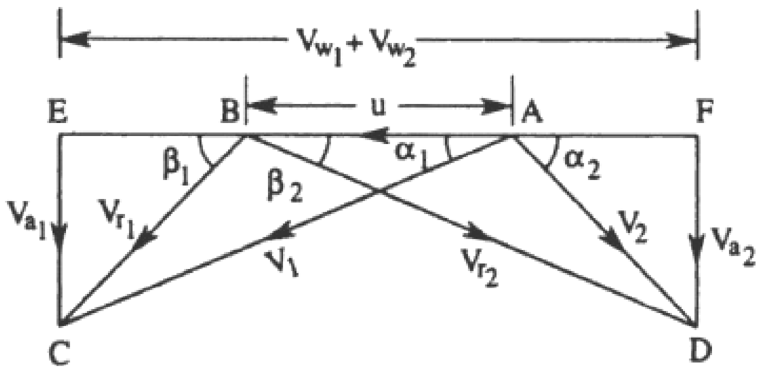


Fig. 14.32

$\alpha_1 = \beta_2 = 20°$

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.

$$\text{Energy supplied} = \frac{V_1^2 + V_{r_2}^2 - V_{r_1}^2}{2} = \frac{2V_1^2 - V_{r_1}^2}{2} \quad [\therefore V_1 = Vr_2]$$

$$= \frac{2 \times (314)^2 - (131.02)^2}{2} = 90012.88 \text{ Nm.}$$

$$\therefore \text{Diagram efficiency} = \frac{\text{Work done}}{\text{Energy supplied}} = \frac{81521.1}{90012.88} = 0.9056 = 90.56\%$$

For maximum efficiency, $u = V_1 \cos\alpha_1 = 314 \cos 20° = 295$ m/s

For this blade speed of 295 m/s, $V_{r_1}^2 = (314)^2 + (295)^2 - 2 \times 314 \times 295 \cos 20°$

$\therefore Vr_1 = 110$ m/s.

$$\therefore \text{Diagram efficiency} = \frac{\text{Work done}}{\text{Energy supplied}}$$

$$= \frac{2u(2V_1 \cos\alpha_1 - u)}{\left(V_1^2 + V_{r_2}^2 - V_{r_1}^2\right)} = \frac{2 \times 295(2 \times 314 \cos 20° - 295)}{(314)^2 + (314)^2 - (110)^2} = 0.938 = 93.8\%$$

$$\therefore \text{Percentage increase in diagram efficiency} = \frac{0.938 - 0.905}{0.905} = 0.0365 = 3.65\%$$

$\therefore$ 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,

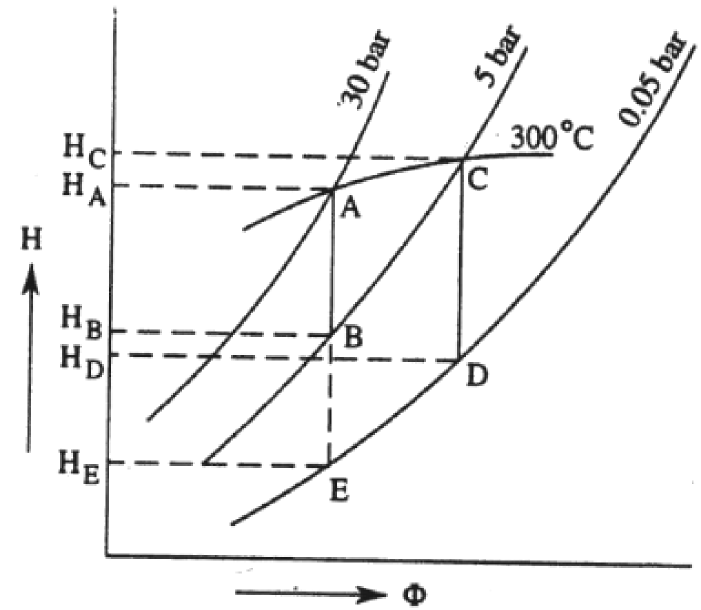


Fig. 14.33

$T_1 = 300°C$, $P_2 = 5$ 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.

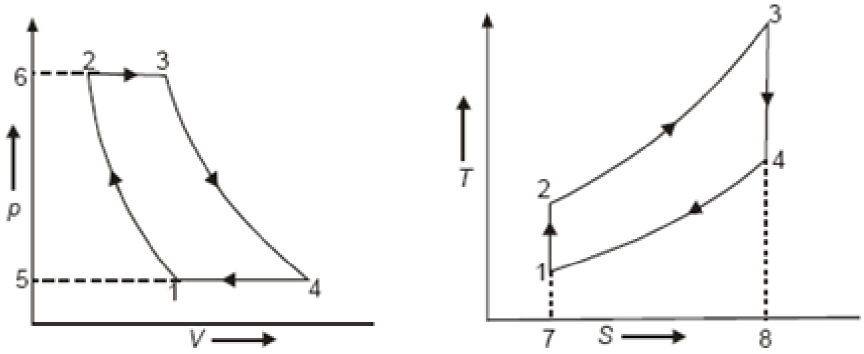
With reheating

$$\eta = \frac{(H_A - H_B) + (H_C - H_D)}{H_A + H_C - H_B - h_D} = \frac{(2990 - 2625) + (3075 - 2595)}{2990 + 3075 - 2625 - 340.6} = 0.273 = 27.3\%.$$

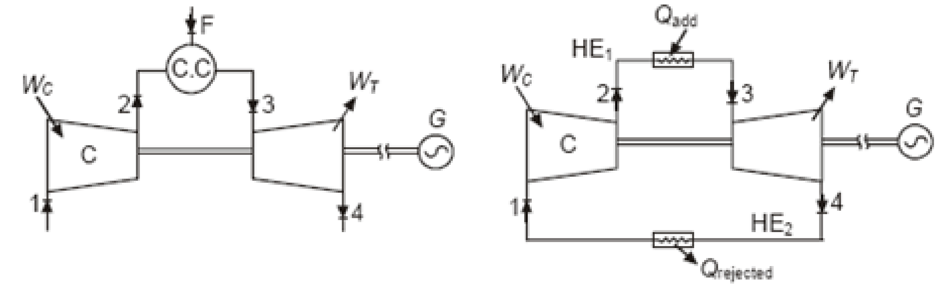
UNIT-IV

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



C : Compressor
CC : Combustion chamber
F : Fuel
G : Generator
HC : Heat exchanger

(a) Open type (b) Closed type

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 = $W_T - W_C$

$W_{net} = \{m_3 (h_3 - h_4) - m_1(h_2 - h_1)\}$

$$\text{Air standard cycle efficiency} = \frac{W_{\text{net}}}{Q_{\text{add}}}$$

$$= \frac{m_1\{(h_3 - h_4) - (h_2 - h_1)\}}{m_1(h_3 - h_2)}$$

Air standard Brayton cycle efficiency: $\boxed{\eta_{\text{Brayton}} = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}}$

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).

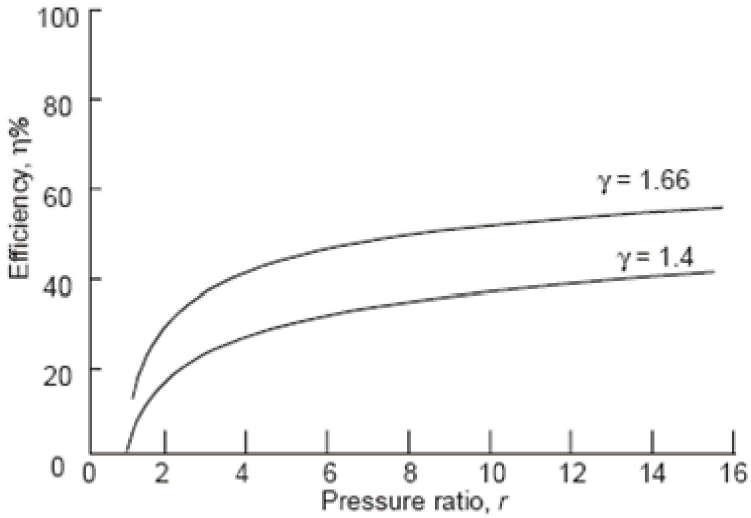


Fig. 4.2 *Efficiency vs. pressure ratio in simple cycle*

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.

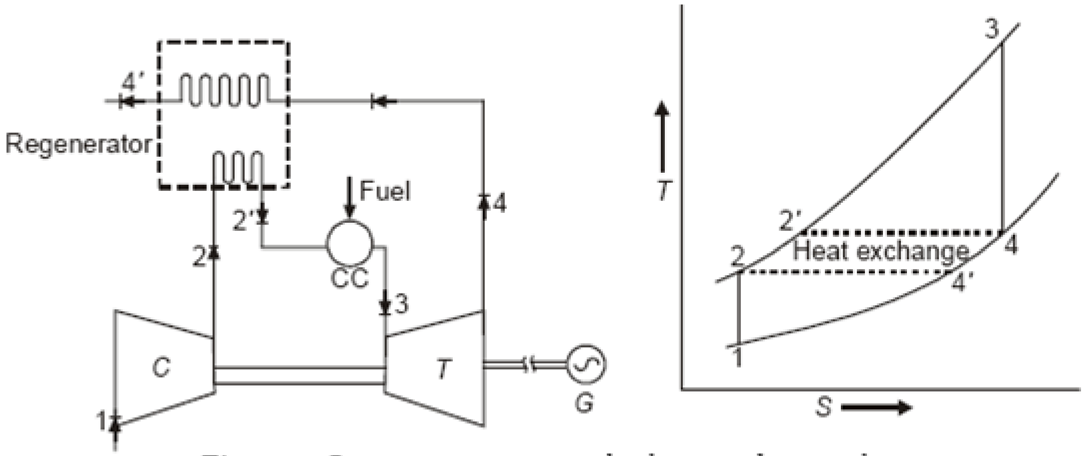


Fig. 4.3 *Regenerative air standard gas turbine cycle.*

$$\text{Regenerator effectiveness, } \varepsilon = \frac{h_{2'} - h_2}{h_4 - h_2},$$

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.

$$\text{Air standard cycle thermal efficiency, } \eta_{regen} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_{2'})}$$

$$\eta_{regen} = \frac{c_p \cdot (T_3 - T_4) - c_p\,(T_2 - T_1)}{c_p\,(T_3 - T_{2'})}$$

Reheat 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

CC: Combustion chamber

G: Generator

f: Fuel

HPT: High pressure turbine

LPT: Low pressure turbine

RCC: Reheat combustion chamber

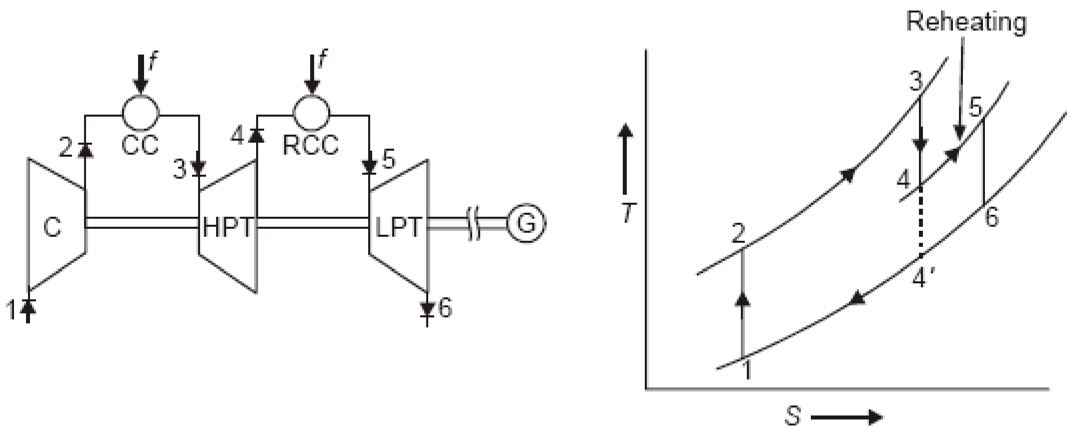


Fig. 4.4 Reheat gas turbine cycle

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

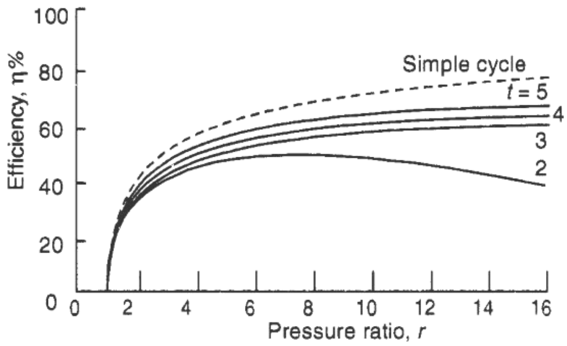


Fig.* 4.5 *Reheat cycle efficiency vs. cycle pressure ratio

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.

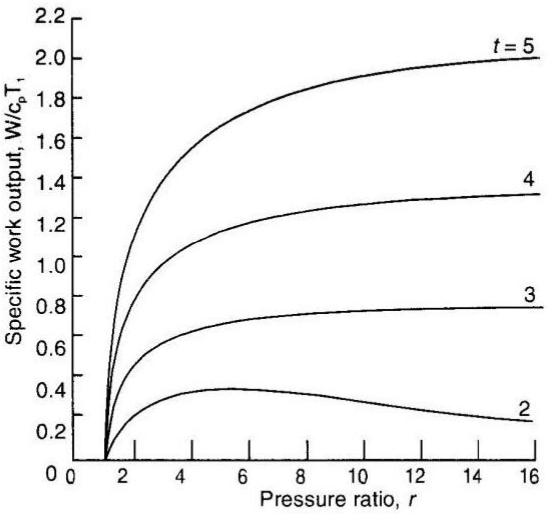
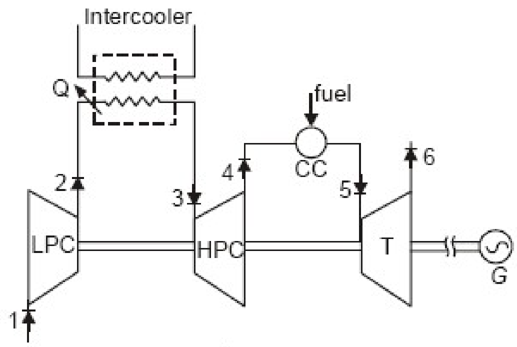


Fig.* 4.6 *Reheat cycle specific work output vs. cycle pressure ratio

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*

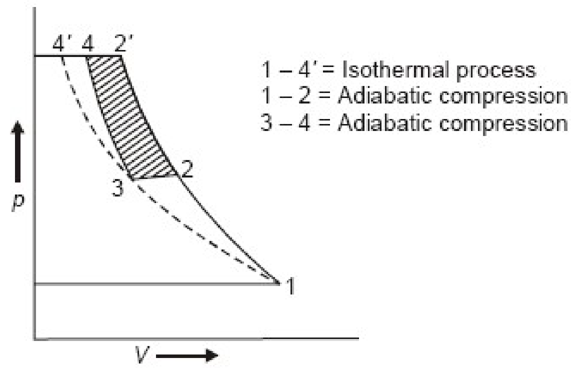


***Fig.* 4.8** *Intercooled compression*

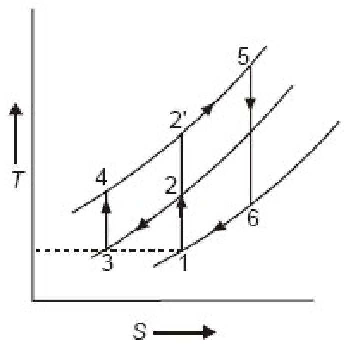


***Fig.* 4.9** *T-S diagram for gas turbine*

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;

$$\eta_{\text{intercool}} = \frac{\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}}{\{h_5 - h_4\}}$$

Here the reduction in thermal efficiency due to reheat is overcome by adding regeneration. Higher gas turbine exhaust gas temperature is thus fully utilized in regenerator (heat exchanger). Additional heat supplied for reheating in order to increase work output is supplemented to some extent by regenerator. It is seen that in this kind of combustion thermal efficiency is increased.

Net work output,

$$W_{\text{net}} = m\{(h_6 - h_7) + (h_4 - h_5) - (h_2 - h_1)\}$$

Heat added,

$$Q_{\text{add}} = m\{(h_4 - h_3) + (h_6 - h_5)\}$$

Thermal efficiency;

$$\eta_{\text{reheat and regenerator}} = \frac{\{(h_6 - h_7) + (h_4 - h_5) - (h_2 - h_1)\}}{\{(h_4 - h_3) + (h_6 - h_5)\}}$$

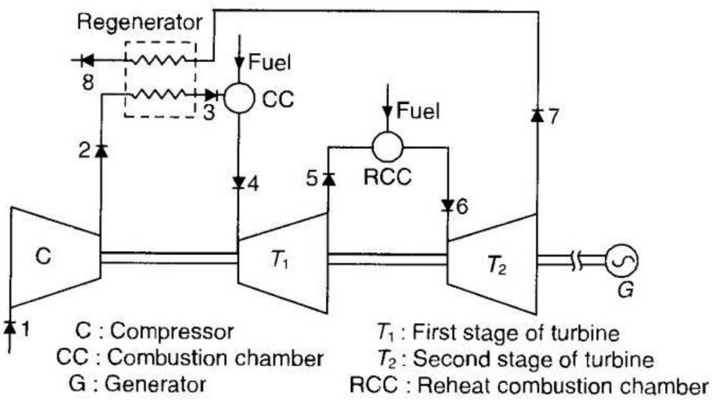


Fig. **4.10** *Schematic for gas turbine cycle with reheat and regeneration*

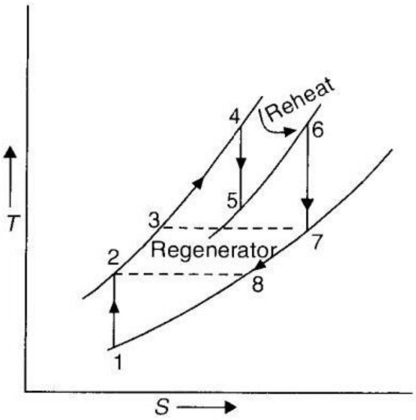


Fig. **4.11** *Thermodynamic representation of gas turbine cycle with reheat and regenerator*

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;

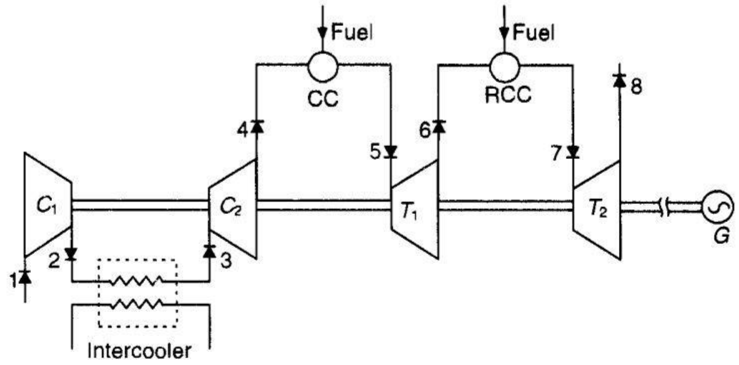
$$W_{\text{net, intercool}} = m\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}$$
$$W_{\text{net, intercool}} = mc_p\{(T_5 - T_6) - (T_4 - T_3) - (T_2 - T_1)\}$$

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 REHEAT AND INTERCOOLING

Schematic arrangement for gas turbine cycle with reheating during expansion and intercooling during compression is given in Fig. 4.12 along with *T-S* diagram. Here reheating and intercooling offer appreciable increase in specific work output but the thermal efficiency deteriorates as reheating and intercooling both have additional heat requirement. Although due to higher temperature the gas turbine exhaust offers potential for regeneration.



C_1 : Low pressure compressor stage
C_2 : High pressure compressor stage
CC : Combustion chamber
G : Generator
RCC : Reheat combustion chamber
T_1 : High pressure turbine stage
T_2 : Low pressure turbine stage

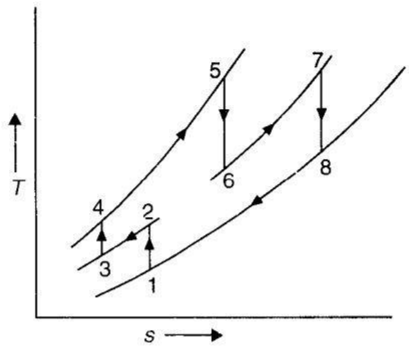


Fig. 4.12 *Schematic for gas turbine cycle with reheat and intercooling and T-S diagram representation*

Air standard cycle analysis for this arrangement gives,

$$W_{net} = m\,\{(h_5 - h_6)\} + (h_7 - h_8) - (h_4 - h_3) - (h_2 - h_1)\}$$

Heat added

$$Q_{add} = m\{(h_5 - h_4) + (h_7 - h_6)\}$$

Cycle thermal efficiency,

$$\eta_{cycle} = \frac{\{(h_5 - h_6) + (h_7 - h_8) - (h_4 - h_3) - (h_2 - h_1)\}}{\{(h_5 - h_4) + (h_7 - h_6)\}}$$

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

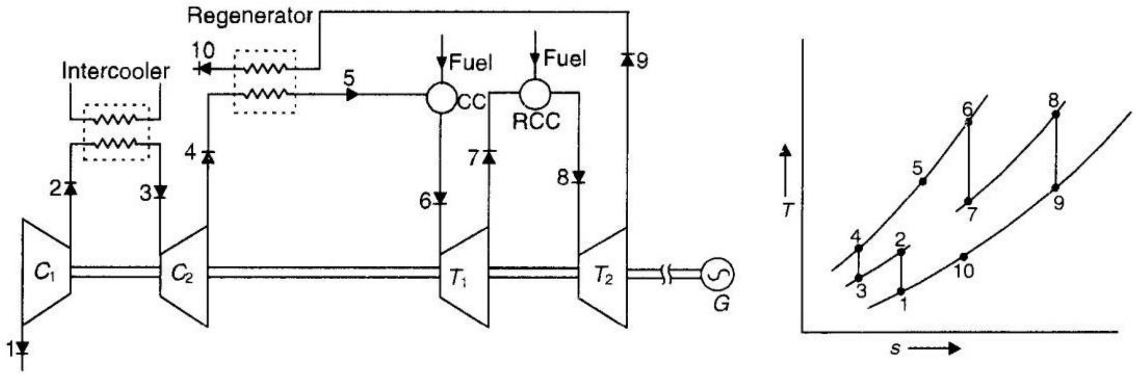


Fig. 4.13 *Schematic for gas turbine cycle with regeneration, reheat and intercooling and T-S diagram*

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

$$\eta_{isen,\ t} = \left\{\frac{h_3 - h_4}{h_3 - h_{4s}}\right\}$$

$$\text{i.e. } \eta_{isen,\ t} = \frac{\text{Actual expansion work}}{\text{Ideal expansion work}}$$

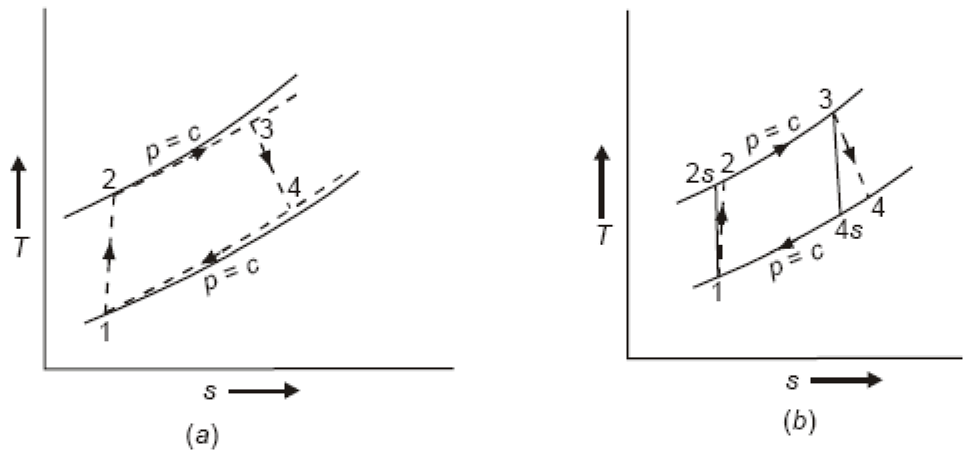


Fig. **4.14** *Effect of irreversibilities and losses in gas turbine cycle.*

Isentropic efficiency of compressor

$$\eta_{\text{isen},\, c} = \left\{ \frac{h_{2s} - h_1}{h_2 - h_1} \right\}$$

$$\therefore \eta_{\text{isen},\, c} = \frac{\text{Ideal compressor work}}{\text{Actual compressor work}}$$

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.

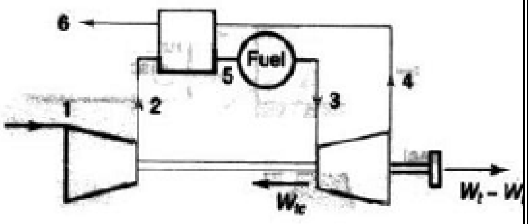
Different factors described above can be accounted for by stagnation properties, compressor and turbine isentropic efficiency and polytropic efficiency.

Worked Out Examples:

Example 1:

Determine the **specific work output**, **specific fuel consumption** and **cycle efficiency** for a heat-exchange cycle, having the following specifications:

compressor pressure ratio	4.0
Turbine inlet temperature	1100 K
Isentropic efficiency of compressor, η_c	0.85
Isentropic efficiency of turbine , η_t	0.87
Mechanical transmission efficiency, η_m	0.99
Combustion efficiency , η_b	0.98
Heat-exchanger effectiveness	0.80
Pressure losses---	
combustion chamber, Δp_b	2% comp. delivery pressure
heat-exchanger air-side, Δp_{ha}	3% comp. delivery pressure
heat –exchanger gas-side, Δp_{hg}	0.04 bar
Ambient conditions, p_a, T_a	1 bar, 288 K

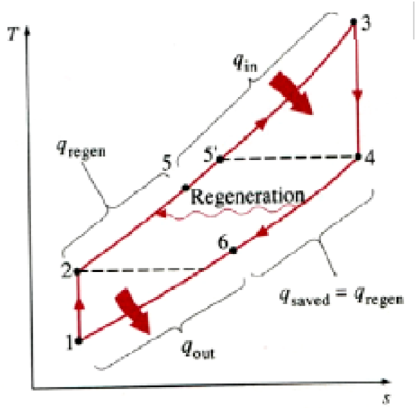


Solution:

Since $T_{01} = T_a$ and $p_{01} = p_a$, and $\gamma = 1.4$, the temperature equivalent of the compressor work is given by

$$T_{02} - T_a = \frac{T_a}{\eta_c}\left[\left(\frac{p_{02}}{p_a}\right)^{(\gamma-1)/\gamma} - 1\right]$$

$$= \frac{288}{0.85}\left[4^{1/3.5} - 1\right] = 164.7K$$



T-s diagram of a Brayton cycle with regeneration.

Turbine work per unit mass flow, required to drive the compressor, is

$$W_{tc} = \frac{c_{pa}(T_{02} - T_a)}{\eta_m} = \frac{1.005 * 164.7}{0.99} = 167.2kJ/kg$$

$$p_{03} = p_{02}\left(1 - \frac{\Delta p_b}{p_{02}} - \frac{\Delta p_{ha}}{p_{02}}\right) = 4.0(1 - 0.02 - 0.03) = 3.8bar$$

$$p_{04} = p_a + \Delta p_{hg} = 1.04bar, \text{ and hence } p_{03}/p_{04} = 3.654$$

Since $\gamma = 1.333$ for the expanding gases, the temperatures equivalent of the total turbine work is

$$T_{03} - T_{04} = \eta_t T_{03}\left[1 - \left(\frac{1}{p_{03}/p_{04}}\right)^{(\gamma-1)/\gamma}\right]$$

$$= 0.87 * 1100\left[1 - \left(\frac{1}{3.654}\right)^{1/4}\right] = 264.8K$$

Total turbine work per unit mass flow is

$W_t = c_{pg}\,(T_{03} - T_{04}) = 1.148 * 264.8 = 304.0$ kJ/kg

- Remembering that the mass flow is to be assumed the same throughout the unit, the specific work output is simply

  $W_t - W_{tc} = 304 - 167.2 = 136.8$ kJ/kg (or kW s/kg)

- It follows that for a 1000 kW plant an air mass flow of 7.3 kg/s would be required. To find the fuel/air ratio we must first calculate the combustion temperature rise ($T_{03} - T_{05}$).

- Heat exchanger effectiveness is given by:

$$\text{Heat-exchanger effectiveness} = 0.80 = \frac{T_{05} - T_{02}}{T_{04} - T_{02}}$$

The specific fuel consumption is therefore

$$SFC = \frac{f}{W_t - W_{tc}} = \frac{3600 * 0.0096}{136.8} = 0.253kg/kWh$$

Finally, the cycle efficiency is

$$\eta = \frac{3600}{SFC * Q_{net,p}} = \frac{3600}{0.253 * 43100} = 0.331$$

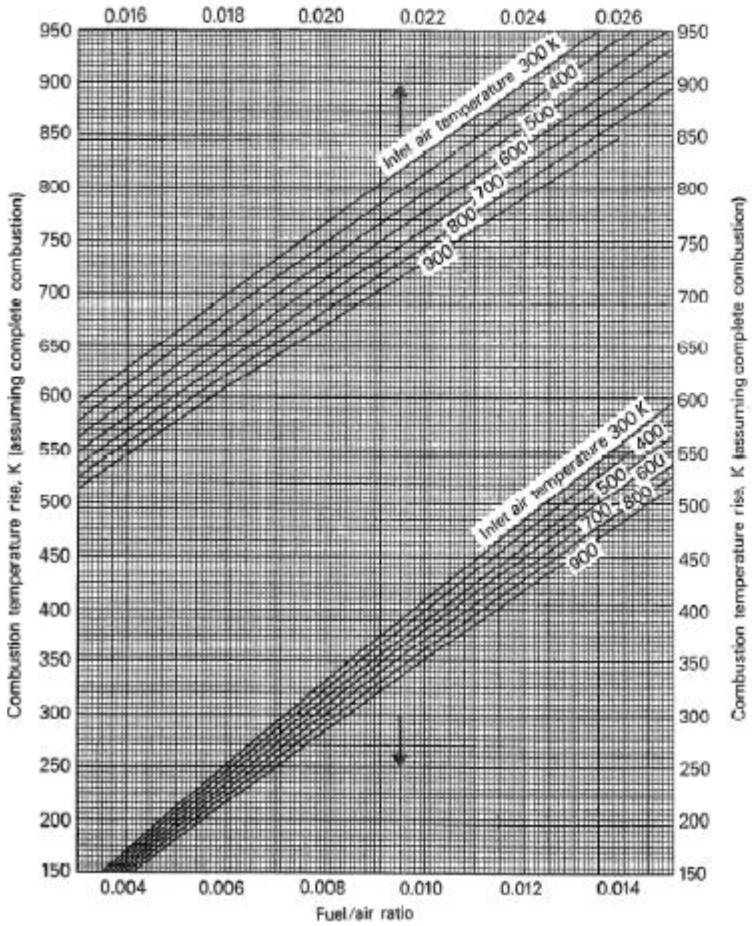
$T_{02} = 164.7 + 288 = 452.7$K, and $T_{04} = 1100 - 264.8 = 835.2$ K

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

$$f = \frac{theoretical\_f}{\eta_b} = \frac{0.0094}{0.98} = 0.0096$$



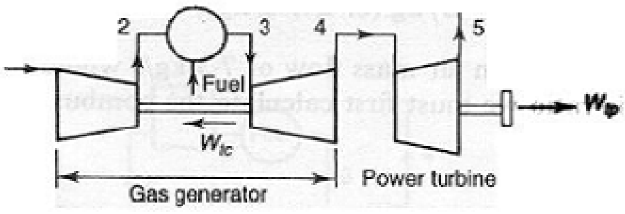
Theoretical Fuel-Air Ratio

Combustion temperature rise vs theoretical fuel-air ratio

Example 2:

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

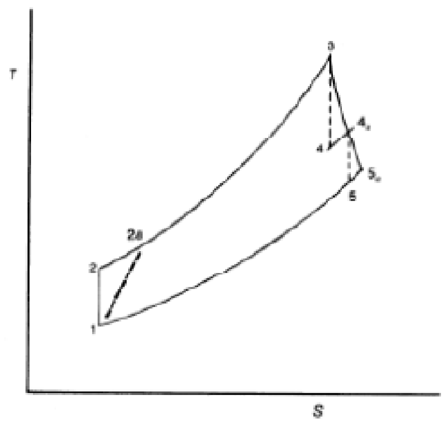
Solution:

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

$$SFC = \frac{f}{W_{tp}} = \frac{3600*0.0204}{277.9} = 0.265 \text{ kg/kWh}$$

$$\eta = \frac{3600}{0.265*43100} = 0.315$$

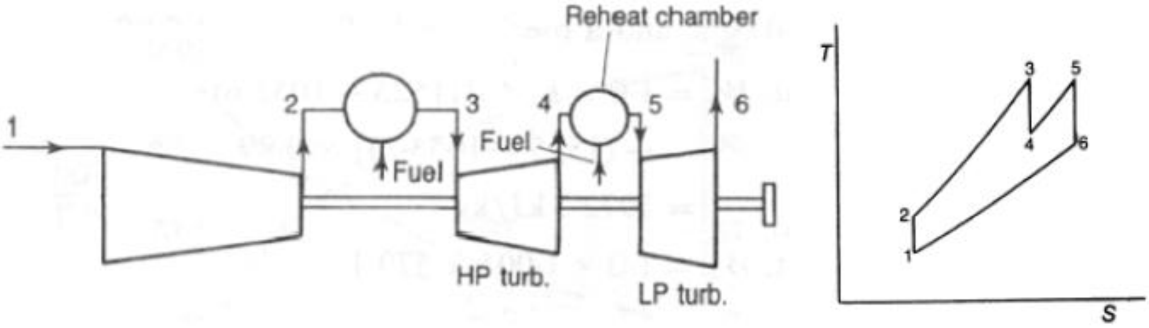
- The cycle calculations are carried out to determine the overall performance. It However, they also provide information required by other groups such as the aerodynamic and control design groups.
- *E.g.*, the temperature at entry to the power turbine, T_{04}, may be required as a control parameter 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 = 800.1K 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,

$$\frac{T_{03}}{T_{04}} = (5.3)^{0.2223}$$

$$T_{04} = 1052.6K$$

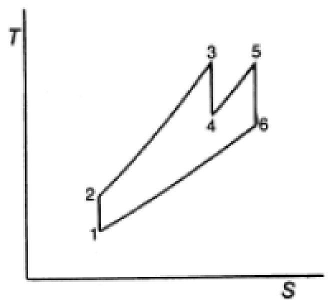
$$p_{04} = 29.69/5.3 = 5.602bar$$

$$p_{05} = 5.602(1.00-0.04) = 5.378bar$$

$$p_{05}/p_{06} = 5.378/1.02 = 5.272$$

$$\frac{T_{05}}{T_{06}} = (5.272)^{0.2223}$$

$$T_{06} = 1053.8K$$



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

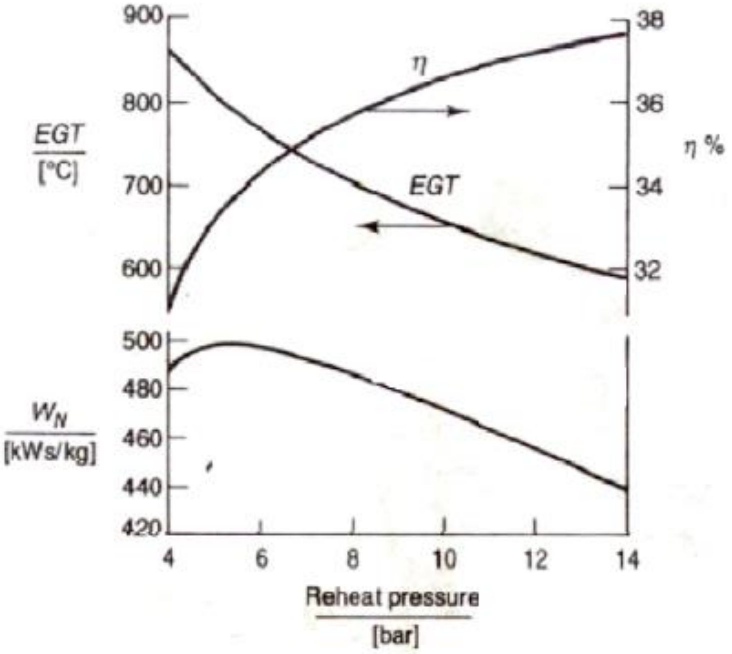
Flow required for 240 MW is given by

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.

- With a reheat pressure of 13 bar, the first turbine has a pressure ratio of 2.284 while the second has a pressure ratio of 12.23, differing markedly from the equal pressure ratios we assumed at the outset.
- This example illustrates some of the problems that arise when a gas turbine must be designed for more than one application.

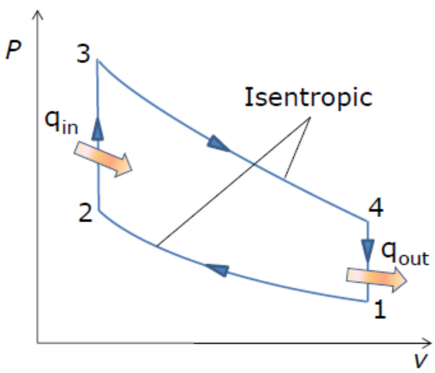


Effect of varying reheat pressure

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,

$$Q_{in} = c_v(T_3 - T_2)$$
$$= 0.718(1373 - 670.8)$$
$$= 504.18 \text{ kJ/kg}$$

- Heat rejected,

$$Q_{out} = c_v(T_4 - T_1)$$
$$= 0.718(630.34 - 308)$$
$$= 231.44 \text{ kJ/kg}$$

- The net work output, $W_{net} = Q_{in} - Q_{out}$

- The net work output,

$$W_{net}=Q_{in}-Q_{out}$$
$$=272.74 \text{ kJ/kg}$$

- Thermal efficiency, $\eta_{th,otto}=W_{net}/Q_{in}$

$$=0.54$$
$$=54\ \%$$

- Otto cycle thermal efficiency,

$$\eta_{th,otto} = 1-1/r^{\gamma-1} = 1-1/7^{0.4}$$
$$= 0.54 \text{ or } 54\ \%$$

- $v_1=RT_1/P_1$

$$=0.287\text{x}308/100=0.844 \text{ m}^3/\text{kg}$$

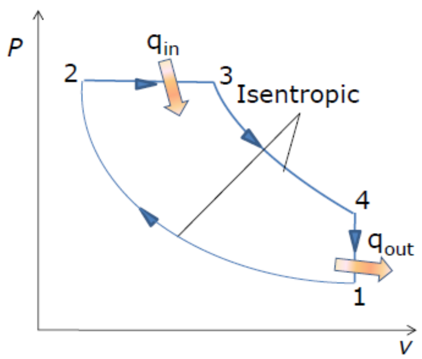
- $MEP = W_{net}/(v_1-v_2) = 272.74/v_1\,(1-1/r)$

$$=272.74/0.844(1-1/7)$$
$$=360 \text{ kPa}$$

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



$T_1=40°C=313\ K$
$P_1=0.1\ Mpa$
$Q_{in}=1675\ MJ/kg$
$r=v_1/v_2=15$

$$v_1 = \frac{RT_1}{P_1} = \frac{0.287 \times 313}{100} = 0.898\,\mathrm{m^3/kg}$$

$$v_2 = v_1 / 15 = 0.898 / 15 = 0.06\,\mathrm{m^3/kg}$$

- It is given that Q_{in} = 1675 MJ/kg

$$Q_{in} = c_p (T_3 - T_2)$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} = 15^{0.4} = 2.954$$

$$T_2 = 313 \times 2.954 = 924.66\,\mathrm{K}$$

$$Q_{in} = 1675 = 1.005(T_3 - 924.66)$$

$$\therefore T_3 = 2591.33\ \mathrm{K} = T_{\max}$$

- Hence, the maximum temperature is 2591.33 K

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{\gamma} = 15^{1.4} = 44.31$$

$$\therefore P_2 = 4431\,\mathrm{kPa}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_3 v_3}{T_3} \rightarrow v_3 = \frac{T_3}{T_2} v_2 = \frac{2591.33}{924.66} \times 0.06 = 0.168\,\mathrm{m^3/kg}$$

$$r_c = \frac{v_3}{v_2} = \frac{0.168}{0.06} = 2.8$$

- The cut-off ratio is 2.8.

$$T_4 = T_3 \left(\frac{v_3}{v_4}\right)^{\gamma-1} = 2591.33 \times \left(\frac{0.168}{0.898}\right)^{0.4}$$

$$= 1325.37\,\mathrm{K}$$

$$\mathrm{Q_{out}} = c_v (T_4 - T_1) = 0.718(1325.4 - 313) = 726.88\,\mathrm{kJ/kg}$$

$$\text{Net work done, } W_{net} = Q_{in} - Q_{out} = 1675 - 726.88$$

$$= 948.12\ \mathrm{kJ/kg}$$

- 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 the Diesel cycle efficiency determined earlier.

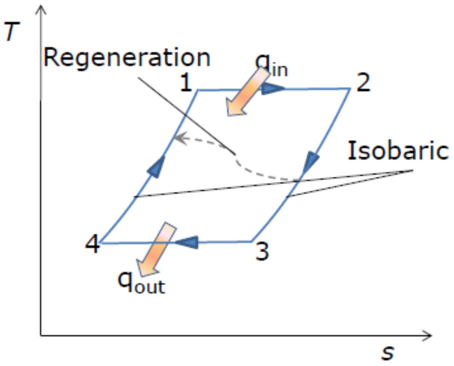
$$MEP = \frac{W_{net}}{v_1 - v_2} = \frac{948.12}{0.898 - 0.06} = 1131.4\,\mathrm{kPa}$$

- The mean effective pressure is 1131. 4 Kpa.

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:

$$w_c=w_t-w_{net}$$
$$=600-461.82 =138.2 \text{ kJ/kg}$$

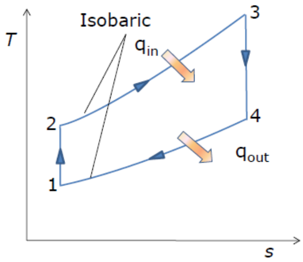
- In the Ericsson cycle the heat is rejected isothermally during the compression process. Therefore this compressor work is also equal to the heat rejected during the cycle.

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},

$$Q_{in} = c_p(T_3 - T_2) = 1.005(1073 - 506.69)$$
$$= 569.14\ \text{kJ/kg}$$

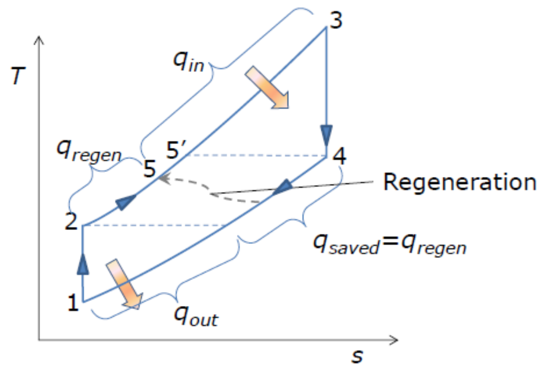
- Heat input per kg of air is 569.14 kJ/kg
- Cycle efficiency,

$$\eta_{th} = (W_{turb} - W_{comp})/Q_{in}$$
$$= (439.89 - 207.72)/569.14$$
$$= 0.408 \text{ or } 40.8\%$$

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 $49.2\ \%$

UNIT-V

Principle of Jet Propulsion

Jet propulsion refers to the imparting of forward motion to the object as a reaction to exit of high velocity gas/liquid stream from the rear end of object. Jet propulsion devices are popularly used in high speed, high altitude aircraft/missile/spacecrafts etc. Simple example of jet propulsion is forward motion of an inflated balloon when air is suddenly released from it.

Jet propulsion is based on the principle of Newton's second law and third law of motion. In a jet propulsion engine the objective is to get the propelling thrust for the engine and for getting it the momentum change occurs in fluid stream such that the reaction to the impulse created by momentum change gives propelling thrust. Thus, the change of momentum of the fluid stream flowing across engine and the reaction to the impulse due to momentum change are responsible for jet propulsion. For realizing momentum change the high temperature and pressure gas stream is expanded through a nozzle so that the gas stream comes out in atmosphere with significantly high velocity, thus giving change of momentum. This momentum change yields impulse force whose reaction produces propelling thrust. Figure. 5.1 gives block diagram for jet propulsion engine.

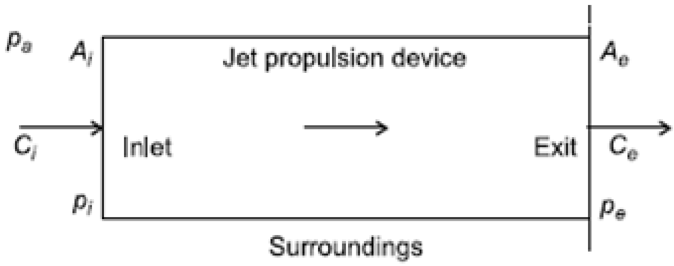


Fig. 5.1 *Principle of jet propulsion engine*

Let us consider jet propulsion device as shown in Fig. 5.1. Let the inlet and exit of device be indicated by subscript '*i*' and '*e*'. Cross-sectional area, velocity of gas, pressure of gas at inlet and exit are given by *Ai* , *Ci* , *pi* and *Ae*, *Ce*, *pe* respectively. Let us analyze for air entering device and flowing through device such that velocity changes from inlet '*Ci*' to exit '*Ce*'.

Atmospheric air enters device at atmospheric pressure *pa*, velocity *Ci*. Consider mass of air entering into engine be at the rate of *ma* kg/s. Mass of gas leaving engine will be at the rate of (*ma* + *mf*) kg/s where *mf* is mass flow rate of fuel required for running. Expansion occurring in engine shall be complete when the pressure at exit *pe* equals to atmospheric pressure, *pa*. In actual case this expansion may not be complete.

For simplicity the mass flowing through device may be assumed constant *i.e.,* fuel added is negligible, then the thrust *T* may be estimated by the rate of change of momentum. This is called as momentum thrust.

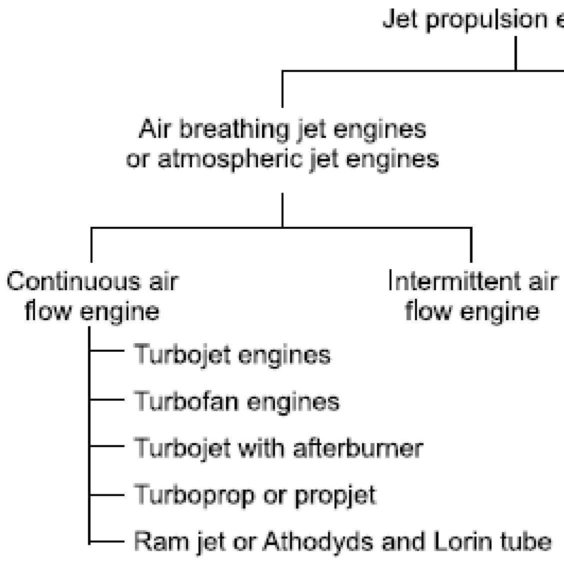
$$T_m = m_a (C_e - C_i)$$

Total thrust, T = Momentum thrust, T_m + Pressure thrust, T_p

$$T = m_a (C_e - C_i) + A_e (p_e - p_i)$$

Classification of Jet propulsion Engines

Jet propulsion engines can be broadly classified based on type of suction. In case of jet engine sucking atmospheric air it is called air-breathing jet engines or atmospheric jet engines. Jet engines may be fed with oxygen carried separately in engine and such engines do not induct atmospheric air. These engines which carry their own oxidizer for combustion of fuel are called non-air breathing jet engines or rocket engines.



Performance of Jet propulsion engines

(i) **Thrust power (TP):** Thrust power indicates the actual power available for propulsion. It refers to the work done per unit time by the engine. This thrust power can be expressed by the product of thrust and velocity with which engine moves (flight velocity).

$$TP = T \times C_a$$
$$= \left[\left\{\left(1 + \frac{m_f}{m_a}\right)\cdot C_e - C_a\right\} + \frac{A_e}{m_a}(p_e - p_a)\right]\cdot C_a$$

(ii) **Propulsive power (PP):** Propulsive power indicates the total energy available for propulsion. It can be estimated by the difference between the rate of kinetic energy entering with air and leaving with jet of exit gases. Mathematically;

$$PP = \frac{1}{2}\left\{\left(1 + \frac{m_f}{m_a}\right)C_e^2 - C_a^2\right\}; \quad \text{W/kg of air}$$

(iii) **Propulsive efficiency (η_{Prop}):** Propulsive efficiency is measure of effectiveness by which propulsive power is transformed into thrust power *i.e.,*

how efficiently propelling duct can propel th engine. Mathematically, it can be given by ratio of thrust power (TP) to propulsive power (PP). Propulsive efficiency is also called Froude efficiency.

$$\eta_{\text{prop}} = \frac{TP}{PP} = \frac{\left[\left(1 + \frac{m_f}{m_a}\right)C_e - C_a\right]C_a}{\frac{1}{2}\left[\left(1 + \frac{m_f}{m_a}\right)C_e^2 - C_a^2\right]}$$

Turbo Jet Engine

Fig. 5.2 shows the schematic of turbojet engine. It has a diffuser section at inlet for realizing some compression of air passing through this section. Due to this air reaching compressor section has pressure more than ambient pressure. This action of partly compressing air by passing it through diffuser section is called “ramming action” or “ram effect”. Subsequently compressor section compresses air which is fed to combustion chamber and fuel is added to it for causing, Combustion products available at high pressure and temperature are then passed through turbine and expanded there. Thus, turbine yields positive work which is used for driving compressor.

Expanding gases leaving turbine are passed through exit nozzle where it is further expanded and results in high velocity jet at exit. This high velocity jet leaving nozzle is responsible for getting desired thrust for propulsion. Pressure variation along the length of turbojet is shown in Fig. 5.2. Velocity and temperature variations are also shown there in. Different salient states shown are plotted on temperature- entropy (*T–s*) diagram as given in Fig. 5.2. Ambient air at state ‘*a*’ enters the diffuser section and ram effect is seen from ‘*a*’ to ‘1’ resulting into small pressure rise. Desired compression occurs between ‘1’ to ‘2’. Subsequent combustion occurs in combustor section resulting into combustion products at state ‘3’. Expansion occurs in turbine from ‘3’ to ‘4’ and subsequently expanded from ‘4’ to ‘5’.

Between states ‘*a*’ and ‘1’ diffusion of air entering occurs isentropically with velocity varying from *Ca* to zero for diffuser section having 100% efficiency. Actually diffuser efficiency may be 0.9–0.95. T–s diagram depicts both theoretical states and actual states. Actual states have been shown as 1′, 2′, 4′ and 5′, while theoretical states are 1, 2, 3, 4 and 5.

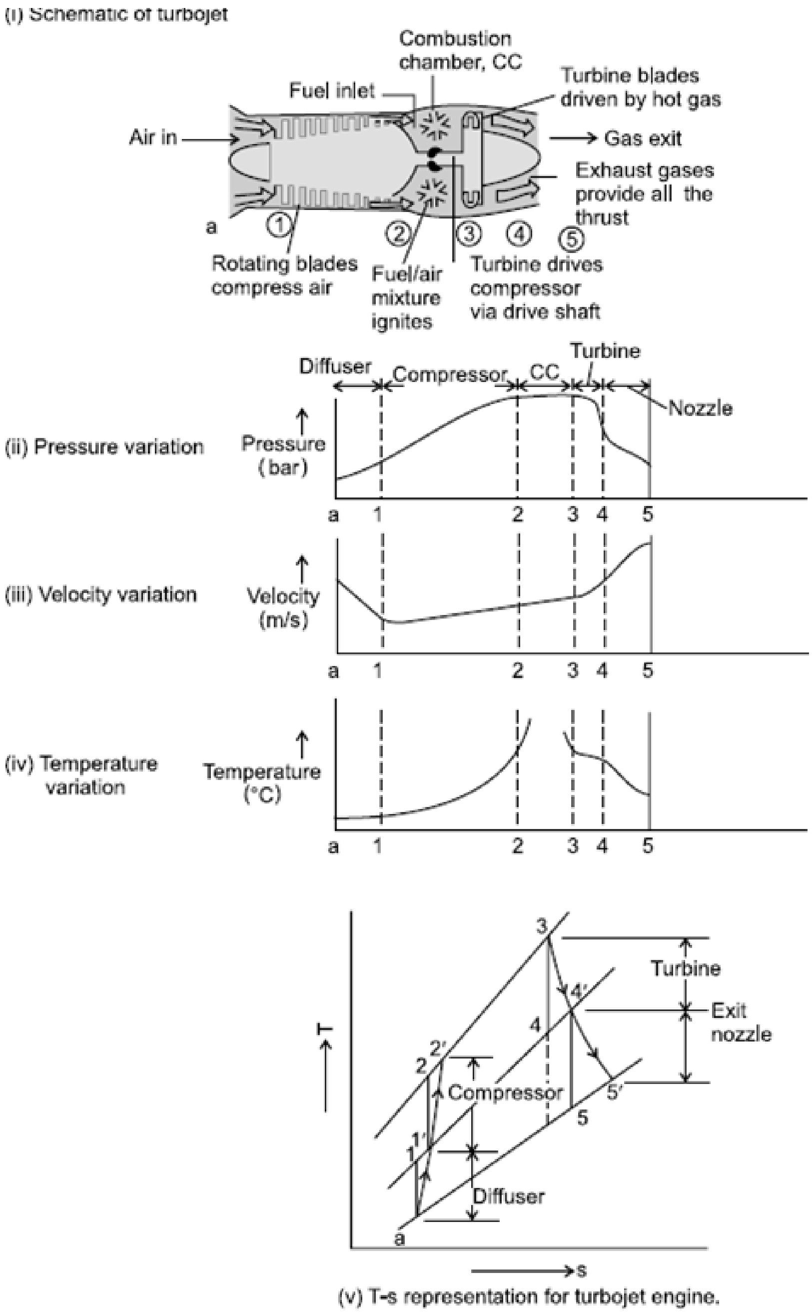
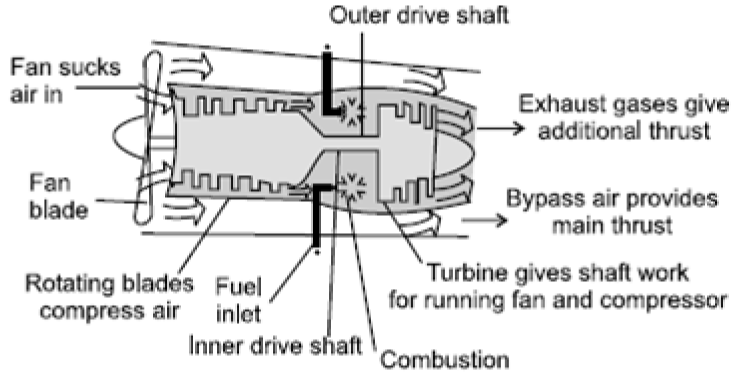


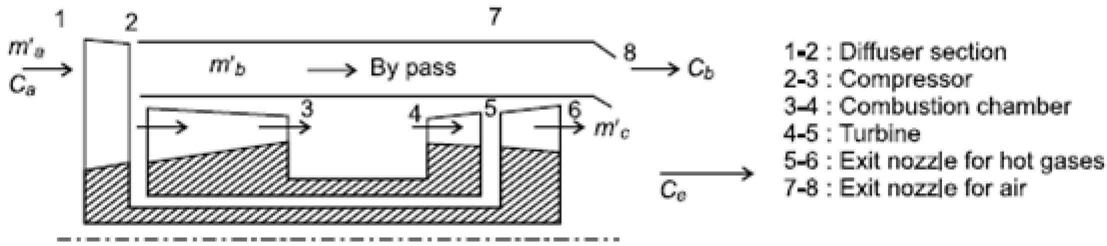
Fig. 5.2 *Turbojet engine*

Turbo Fan Engine

Turbofan engine is the modified turbojet engine in which additional thrust is realized by putting fan at the entry of the engine casing. Fan blades propel by pass air around engine core between inner and outer engine casing. This air does not participate in combustion but provides additional thrust while leaving through exit nozzle. Figure. 5.3 show the schematic of turbofan engine. The propeller fan put at inlet to engine sucks air and it passes through by pass passage as shown up to the exit nozzle end. Thus there are two streams of air flowing, one air stream gets rammed, compressed, burnt, expanded in turbine and finally passes through exit nozzle and other air stream passes through passage between outer and inner casings from inlet to nozzle exit. Total thrust created will be due to two jet streams one due to cold air or fan air and other due to burnt gases leaving turbine.



(a) Turbofan engine arrangement



(b) Line diagram showing different sections (not to scale)

Fig. 5.3 *Turbofan engine*

Turbo fan engine's performance primarily depends on the fan pressure ratio, bypass ratio, overall pressure ratio, turbine inlet temperature, cruising speed and altitude. Amongst these thermodynamic parameters. For optimum performance of turbofan engine the plot of specific fuel consumption for the varying bypass ratio is shown in Fig. 5.4.

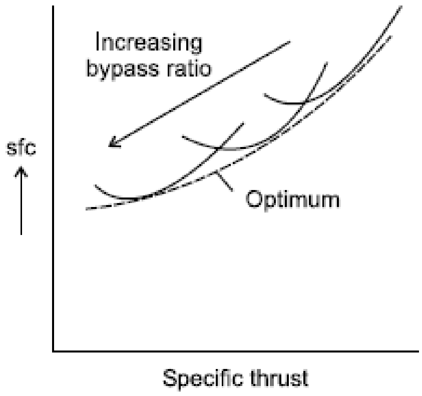


Fig. 5.4

Turboprop Engine

Turboprop (Turbo-propeller) engine, also called turboshaft engine or propjet engine differs slightly from turbofan engine. It uses thrust to turn a propeller. It consists of a compressor for compressing the inlet air, combustion chamber and turbine followed by exit nozzle. A part of turbine output is used to drive the compressor and remaining for driving propeller. Thus some stages of turbine give shaft work for driving compressor and some stages produce shaft work for driving propeller as shown in Fig. 5.5. It can also be understood as if the gases expand through main turbine which drives compressor and also expands through power turbine which drive propeller through suitable reduction gear box. Turboprop engines are used in small passenger planes, cargo planes etc.

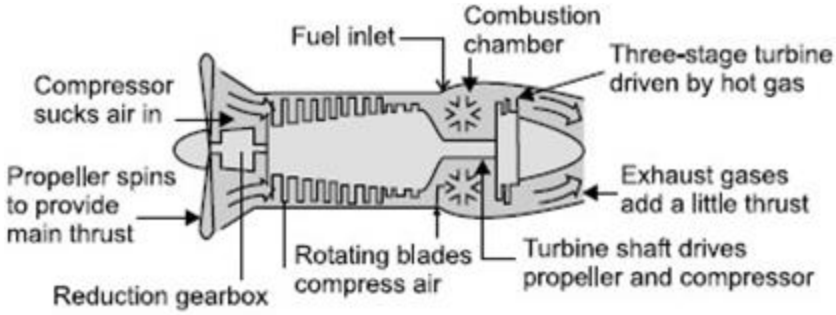


Fig. 5.5 *Turboprop engine*

Ramjet Engine

Ramjet engine is the simplest of jet engines having no moving parts. Ramjet is a typically shaped duct open at both ends with air being compressed merely due to forward motion of engine. Fuel is subsequently added for combustion and thus high pressure, high temperature gases exit from exhaust nozzle. High pressure air is continuously available as engines keeps on moving forward. These ramjets are extensively used for propulsion in number of high speed aircrafts.

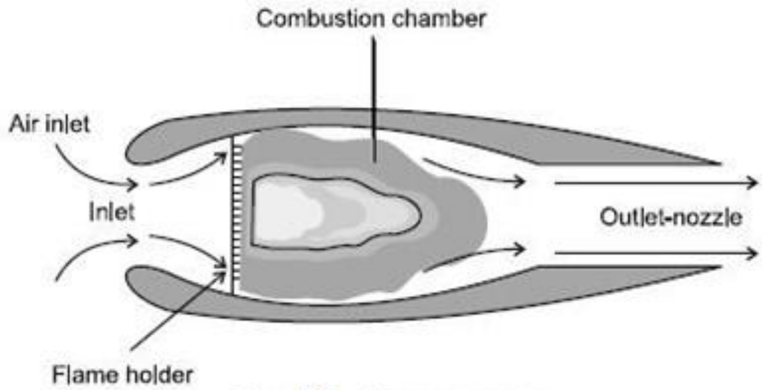


Fig. 5.6 *Ramjet engine*

Ramjet is also called Athodyd Lorin tube based on the name of its inventor Rene Lorin, a French engineer. First ramjet engine was developed in 1913 and it had steady flow through inlet diffuser, combustion chamber and outlet nozzle. These can not operate under stationary condition as sole compression of air occurs due to ram compression. Ram pressure ratio is relatively small initially as forward speed is slow but increases as the speed increases. Ramjet is boosted up to substantially high speed using turbojet or rocket engine for getting desired thrust by ramjet. Ramjet are boosted up to speed of 300 km/hr. The efficiency of ramjet largely depends upon the design of diffuser section.

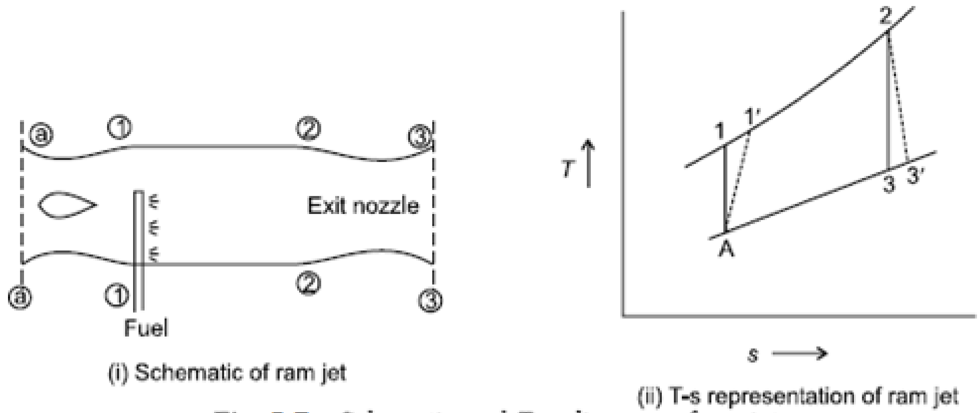
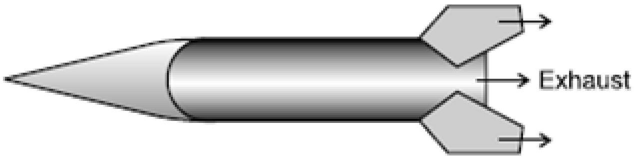


Fig. 5.7 *Schematic and T–s diagram of ramjet*

Principle of rocket propulsion

Rocket engines are non-air breathing engines and carry their own oxidizer for burning of fuel. Rocket propulsion is realized by the thrust produced by combustion products leaving exit nozzle. It has injection system for fuel and oxidizer followed by combustion chamber and exit nozzle as shown in Fig. 5.8.



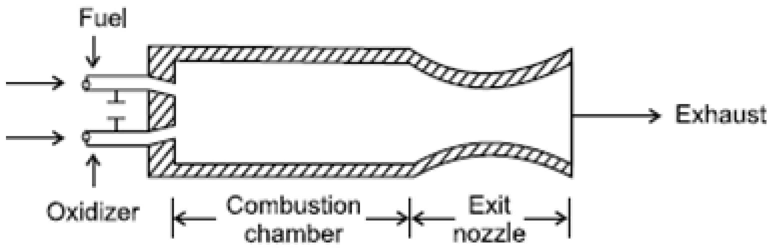


Fig. 5.8 *Layout for rocket engine*

In rocket engines the combustion products get discharged from the exit nozzle with supersonic velocity and thus have very high kinetic energy. Rocket gets desired thrust by the reaction available from the nozzle stream. Thrust is available due to change of momentum and pressure with which jet comes out.

Net thrust available; $T = m'_p C_e + A_e (p_e - p_a)$

Where m'_p is mass flow rate of propellant, jet exit velocity C_e, area of exit nozzle A_e, pressure of exit jet p_e and atmospheric pressure is p_a.

Thrust power in case of rocket engine can be given as;

$$TP = T \cdot C_a = m'_p \cdot C_{ej} \cdot C_a$$

Propulsive efficiency: For rocket engine

$$\eta_{prop} = \frac{2\,(C_a/C_{ej})}{1 + (C_a/C_{ej})^2}$$

Rocket Engine

Rocket engines as described earlier can be classified based on type of propellant carried by them. These could be solid propellant and liquid propellant resulting into,

(i) Solid propellant rocket engine
(ii) Liquid propellant rocket engine.

Since Rocket engines carry propellant fuel and oxidizer along with, therefore, the characteristics of propellant have to be such as to give best performance with minimum oxidizer requirement.

Applications of rocket engines have proved boon for our civilization as satellites in space which are serving air communication and other requirements are rocket engines. Rocket engines are also extensively used in spacecrafts, missiles, jet assisted air planes, pilotless aircraft, etc.

Solid propellant rocket engines

These rocket engines use solid propellants which burns using oxidizer present within it. There are two types of solid propellant rockets depending upon type of burning *i.e.*, restricted burning and unrestricted burning. Figure 5.9 shows the different arrangements in solid propellant rocket engines. Solid propellant has composition such that all essential requirements for combustion are met. Specific requirements of solid propellant are,

(i) Propellant should have sufficient compressive and impact strength at low temperature.
(ii) Propellant should give uniform burning.
(iii) Propellant should give high specific impulse.

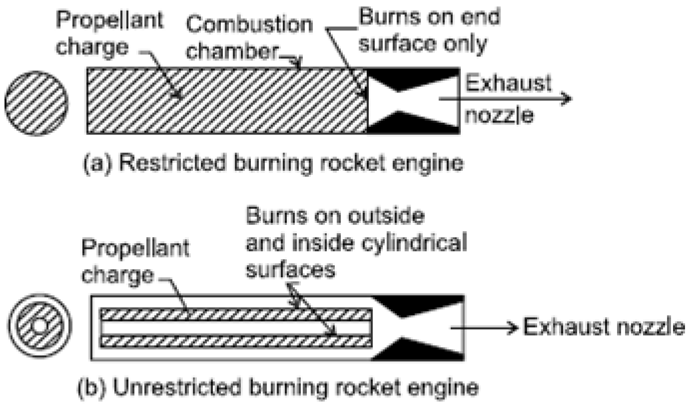


***Fig.* 5.9** *Solid propellant rocket engines*

Arrangement shows that it has seamless sturdy steel tube closed at one end. At the open end exit nozzle is provided and this may be single nozzle type or multi nozzle type exit. Propellant is filled in tube while in liquid state and gets solidified gradually upon cooling and fits tube completely. Solid propellant filled inside tube is burnt gradually so as to give combustion products for producing desired thrust. Restricted burning rocket engines have the propellant burnt only on its exposed surface towards exit nozzle.

Gradually complete propellant is burnt and combustion products move out through exit nozzle. Thus restricted burning nozzles are very similar to cigarette whose burning begins at one end and goes till end. Restricted burning rockets are suited for small thrust applications for longer duration.

Unrestricted burning in rocket engine occurs such that all exposed surfaces of solid propellant are burnt simultaneously inside tube. Since burning of whole solid fuel is initiated so it gets burnt quickly and produces large thrust.

Liquid propellant rocket engines

Figure 5.10 shows the schematic of liquid propellant rocket engines. These contain liquid propellants stored in containers outside combustion chamber. Liquid propellant fuel is being fed using either pressure feed or pump feed to transfer fuel from storage tank to combustion chamber. Pressure feed system has pump driven by small turbine. This feed system is relatively cheap and simple compared to other feed systems.

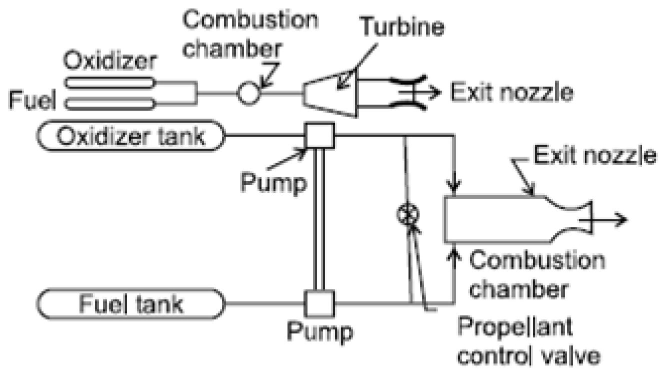


Fig. 5.10 *Liquid propellant rocket engines*

There could be mono-propellant or bi-propellant type rocket engines. Mono-propellant systems are heavier in view of additional quantity of fuel required. Schematic shows two liquid bi-propellant rocket system with rocket motor and propellant system etc. Rocket motor has propellant injector for injecting liquid fuel and oxidizer, ignition systems, combustion chamber and exit nozzle as shown.

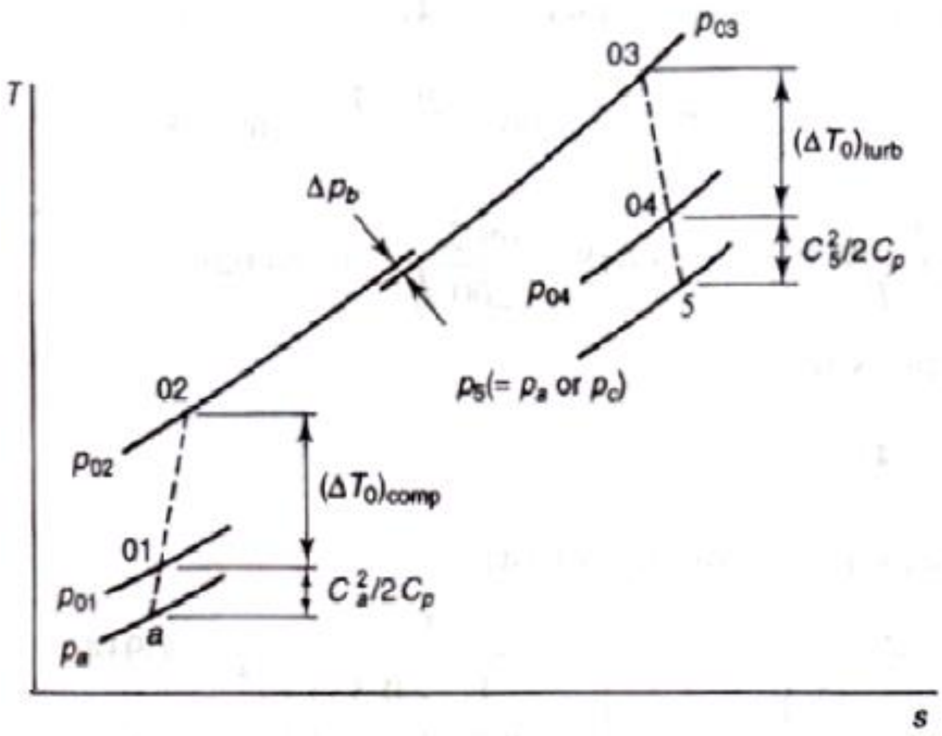
Liquid propellant is heated up by circulating it around the combustion chamber and nozzle walls for cooling it before being injected for combustion. Cooling is done so as to maintain temperature of walls low, so that excessive thermal stresses are not developed. Liquid propellants are selected based on specific impulse, availability, cost, ease of handling and density etc. There are number of oxidizer and liquid propellants available, namely, liquid oxygen + liquid hydrogen, liquid oxygen + ethyl alcohol, liquid oxygen + high quality gasoline, nitric acid + aniline etc.

Example 1(Turbojet engine)

Determination of the specific thrust and SFC for a simple turbojet engine, having the following component performance at the design point at which the cruise speed and altitude are M 0.8 and 10000 m.

Compressor pressure ratio	8.0
Turbine inlet temperature	1200 K
Isentropic efficiency:	
of compressor, η_c	0.87
of turbine, η_t	0.90
of intake, η_i	0.93
of propelling nozzle, η_j	0.95
Mech. transmission efficiency, η_m	0.99
Combustion efficiency, η_b 0.98	
Combustion pressure loss, Δp_b	4% of compressor delivery pressure

Solution:



Turbojet cycle with losses

- From the ISA table, at 10000 m

  $p_a = 0.2650$ bar, $T_a = 223.4$ K and $a = 299.5$ m/s

- The stagnation conditions after the intake may be obtained as follows:

$$\frac{C_a^{\ 2}}{2C_p} = \frac{(0.8*299.5)^2}{2*1.005*1000} = 28.6\text{ K}$$

$$T_{01} = T_a + \frac{C_a^{\ 2}}{2C_p} = 223.3 + 28.6 = 251.9\text{ K}$$

$$\frac{p_{01}}{p_a} = \left[1 + \eta_i \frac{C_a^{\ 2}}{2c_p T_a}\right]^{\frac{\gamma}{(\gamma-1)}} = \left[1 + \frac{0.93*28.6}{223.3}\right]^{3.5} = 1.482$$

$$p_{01} = 0.2650*1.482 = 0.393\text{ bar}$$

At outlet from the compressor,

$$p_{02} = \left(\frac{p_{02}}{p_{01}}\right) p_{01} = 8.0 * 0.393 = 3.144 \text{ bar}$$

$$T_{02} - T_{01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{p_{02}}{p_{01}}\right)^{(\gamma-1)/\gamma} - 1\right] = \frac{251.9}{0.87}\left[8.0^{1/3.5} - 1\right] = 234.9 \text{ K}$$

$$T_{02} = 251.9 + 234.9 = 486.8 \text{ K}$$

$$W_t = W_c/\eta_m \text{ and hence } T_{03} - T_{04} = \frac{C_{pa}(T_{02} - T_{01})}{C_{pg}\eta_m} = \frac{1.005 \text{x} 234.9}{1.148 \text{x} 0.99} = 207.7 \text{ K}$$

$$T_{04} = 1200 - 207.7 = 992.3 \text{ K}$$

$$p_{03} = p_{02}\left(1 - \frac{\Delta p_b}{p_{02}}\right) = 3.144(1 - 0.04) = 3.018 \text{ bar}$$

$$T_{04}' = T_{03} - \frac{1}{\eta_t}(T_{03} - T_{04}) = 1200 - \frac{207.7}{0.90} = 969.2 \text{ K}$$

$$p_{04} = p_{03}\left(\frac{T_{04}'}{T_{03}}\right)^{\frac{\gamma}{(\gamma-1)}} = 3.018\left(\frac{969.2}{1200}\right)^4 = 1.284 \text{ bar}$$

- The nozzle pressure ratio is, therefore

$$\frac{p_{04}}{p_a} = \frac{1.284}{0.265} = 4.845$$

- The critical pressure ratio is

$$\frac{p_{04}}{p_c} = \frac{1}{\left[1 - \frac{1}{\eta_j}\left(\frac{\gamma-1}{\gamma+1}\right)\right]^{\frac{\gamma}{(\gamma-1)}}} = \frac{1}{\left[1 - \frac{1}{0.95}\left(\frac{0.333}{2.333}\right)\right]^4} = 1.914$$

Since $p_{04}/p_a > p_{04}/p_c$, the nozzle is choking, and

$$T_5 = T_c = \left(\frac{2}{\gamma+1}\right) \qquad T_{04} = \frac{2 * 992.3}{2.333} = 850.7 \text{ K}$$

$$p_5 = p_c = p_{04}\left(\frac{1}{p_{04}/p_c}\right) = 1.284/1.914 = 0.671 \text{ bar}$$

$$\rho_5 = \frac{p_c}{RT_c} = \frac{100 * 0.671}{0.287 * 850.7} = 0.275 \text{ kg/m}^3$$

$$C_5 = (\gamma R T_c)^{\frac{1}{2}} = (1.333 * 0.287 * 850.7 * 1000)^{\frac{1}{2}} = 570.5 \text{ m/s}$$

$$\frac{A_5}{m} = \frac{1}{\rho_5 C_5} = \frac{1}{0.275 * 570.5} = 0.006374 \text{ m}^2\text{s/kg}$$

The specific thrust is

$$F_s = (C_5 - C_a) + \frac{A_5}{m}(p_c - p_a)$$
$$= (570.5-239.6) + 0.006374(0.671-0.265)10^5$$
$$= 330.9 + 258.8 = 589.7 \text{ Ns/kg}$$

For $T_{02} = 486.8$ K and $T_{03} - T_{02} = 1200-486.9 = 713.2$ K,

we find that the theoretical fuel/air ratio required is 0.0194.

Thus the actual fuel/air ratio is

$$f = \frac{0.0194}{0.98} = 0.0198$$

The specific fuel consumption is therefore

$$\text{SFC} = \frac{f}{F_s} = \frac{0.0198 * 3600}{589.7} = 0.121 \text{ kg/h N}$$

- For cycle optimisation, calculations would normally be done on the basis of specific thrust and SFC. A common problem, however, is the determination of actual engine performance to meet a particular aircraft thrust requirement. The engine designer needs to know the airflow, fuel flow and nozzle area ; the airflow and nozzle area are also important to the aircraft designer who must determine the installation dimensions.
- If, for example, the cycle conditions in the example were selected to meet a thrust requirement of 6000 N, then

$$m = \frac{F}{F_s} = 10.17 \text{ kg/s}$$

- the fuel flow is given by
  $m_f = f_m = 0.0198 * 10.17 * 3600 = 725.2$ kg/h
  (it should be noted that fuel flow is normally measured and indicated in kg/h rather than kg/s)

- The nozzle area follows from the continuity equation :
  $A5 = .006374 * 10.17 = 0.0648 \text{ m}^2$

Example-2 (Turbofan Engine)

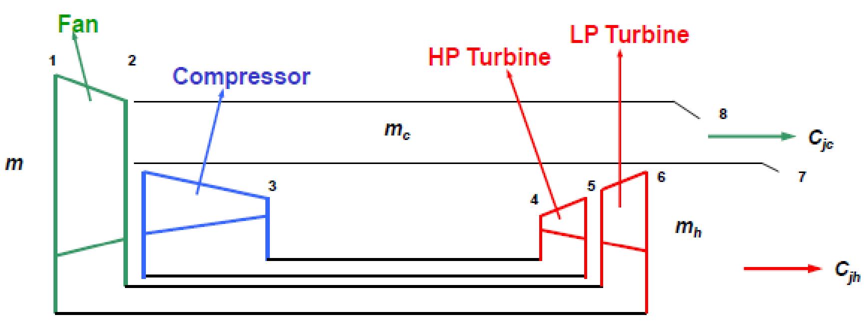
The following data apply to a twin-spool turbofan engine, with the fan driven by the LP turbine and the compressor by the HP turbine. Separate cold and hot nozzles are used.

Overall pressure ratio	25.0
Fan pressure ratio	1.65
Bypass ratio m_c/m_h	5.0
Turbine inlet temperature	1550 K
Fan, compressor and turbine polytropic efficiency	0.90
Isentropic efficiency of each propelling nozzle	0.95
Mechanical efficiency of each spool	0.99
Combustion pressure loss	1.50 bar
Total air mass flow rate	215 kg/s

It is required to find the thrust and SFC under sea-level static conditions where the ambient pressure and temperature are 1.0 bar and 288 K.

Solution:

Twin Spool Turbofan Engine



$$B = \frac{m_c}{m_h} \quad \text{and} \quad m = m_c + m_h \Rightarrow \quad m_c = \frac{mB}{B+1} \qquad m_h = \frac{m}{B+1}$$

The values of *(n-1)/n* for the polytropic compression and expansion are:

$$\text{for compression, } \frac{n-1}{n} = \frac{1}{\eta_{\infty c}}\left(\frac{\gamma-1}{\gamma}\right)_a = \frac{1}{0.9*3.5} = 0.3175$$

$$\text{for expansion, } \frac{n-1}{n} = \eta_{\infty t}\left(\frac{\gamma-1}{\gamma}\right)_g = \frac{0.9}{4} = 0.225$$

Under static conditions, $T_{01} = T_a$ and $p_{01} = p_a$, so that,

$$\frac{T_{02}}{T_{01}} = \left(\frac{p_{02}}{p_{01}}\right)^{(n-1)/n} \Rightarrow T_{02} = 288 * 1.65^{0.3175} = 337.6\,\mathrm{K}$$

$$T_{02} - T_{01} = 337.6 - 288 = 49.6\,\mathrm{K}$$

$$\frac{p_{03}}{p_{02}} = \frac{25.0}{1.65} = 15.15$$

$$T_{03} = T_{02}\left(\frac{p_{03}}{p_{02}}\right)^{(n-1)/n} = 337.6 * 15.15^{0.3175} = 800.1\,\mathrm{K}$$

$$T_{03} - T_{02} = 800.1 - 337.6 = 462.5K$$

The cold nozzle pressure ratio is

$$\frac{p_{02}}{p_a} = FPR = 1.65$$

and the critical pressure ratio for this nozzle is

$$\frac{p_{02}}{p_c} = \frac{1}{\left[1 - \frac{1}{\eta_j}\left(\frac{\gamma-1}{\gamma+1}\right)\right]^{\gamma/(\gamma-1)}} = \frac{1}{\left[1 - \frac{1}{0.95}\left(\frac{0.4}{2.4}\right)\right]^{3.5}} = 1.965$$

Thus the cold nozzle is not choking, so that $p_8 = p_a$ and the cold thrust F_c is given simply by

$$F_c = m_c C_8$$

The nozzle temperature drop

$$T_{02} - T_8 = \eta_j T_{02}\left[1 - \left(\frac{1}{p_{02}/p_a}\right)^{(\gamma-1)/\gamma}\right]$$

$$= 0.95 * 337.6\left[1 - \left(\frac{1}{1.65}\right)^{1/3.5}\right] = 42.8K$$

and hence

$$C_8 = \left[2C_p(T_{02} - T_8)\right]^{1/2} = (2 * 1.005 * 42.8 * 1000)^{1/2} = 293.2m/\sec$$

Since the bypass ratio B is 5.0

$$m_c = \frac{mB}{B+1} = \frac{215 * 5.0}{6.0} = 179.2kg/s$$

$$F_c = 179.2 * 293.2 = 52541N$$

Considering the work requirement of the HP rotor,

$$T_{04} - T_{05} = \frac{C_{pa}}{\eta_m C_{pg}}(T_{03} - T_{02}) = \frac{1.005*462.5}{0.99*1.148} = 409.0K$$

and for the LP rotor

$$T_{05} - T_{06} = (B+1)\frac{C_{pa}}{\eta_m C_{pg}}(T_{02} - T_{01}) = \frac{6.0*1.005*49.6}{0.99*1.148} = 263.2K$$

Hence

$$T_{05} = T_{04} - (T_{04} - T_{05}) = 1550 - 409.0 = 1141.0$$

$$T_{06} = T_{05} - (T_{05} - T_{06}) = 1141.0 - 263.2 = 877.8$$

p_{06} may then be found as follows

$$\frac{p_{04}}{p_{05}} = \left(\frac{T_{04}}{T_{05}}\right)^{n/n-1} = \left(\frac{1550}{1141.0}\right)^{1/0.225} = 3.902$$

$$\frac{p_{05}}{p_{06}} = \left(\frac{T_{05}}{T_{06}}\right)^{n/n-1} = \left(\frac{1141.0}{877.8}\right)^{1/0.225} = 3.208$$

$$p_{04} = p_{03} - \Delta p_b = 25.0*1.0 - 1.50 = 23.5 \text{ bar}$$

$$p_{06} = \frac{p_{04}}{\left(p_{04}/p_{05}\right)\left(p_{05}/p_{06}\right)} = \frac{23.5}{3.902*3.208} = 1.878 \text{ bar}$$

Thus the hot nozzle pressure ratio is $\frac{p_{06}}{p_a} = 1.878$

While the critical pressure ratio is

$$\frac{p_{06}}{p_c} = \frac{1}{\left[1 - \frac{1}{0.95}\left(\frac{0.333}{2.333}\right)\right]^4} = 1.914$$

This nozzle is also unchoked, and hence $p_7 = p_a$

$$T_{06} - T_7 = \eta_j T_{06}\left[1 - \left(\frac{1}{p_{06}/p_a}\right)^{(\gamma-1)/\gamma}\right]$$

$$= 0.95*877.8\left[1 - \left(\frac{1}{1.878}\right)^{\frac{1}{2}}\right] = 121.6 \text{ K}$$

$$C_7 = \left[2C_p(T_{06} - T_7)\right]^{1/2} = \left[2*1.148*121.6*1000\right]^{1/2} = 528.3 \text{ m/s}$$

$$m_{h} = \frac{m}{B+1} = \frac{215}{6.0} = 35.83 \text{ kg/s}$$

$$F_h = 35.83*528.3 = 18929 \text{ N}$$

Thus the total thrust is

$$F_c + F_h = 52541 + 18929 = 71470 \text{ N or } \sim 71.5 \text{ kN}$$

The fuel flow can readily be calculated from the known temperatures in the combustor and the airflow through the combustor, i.e., m_h. The combustion temperature rise is (1550-800) = 750 K and the combustion inlet temperature is 800 K. The ideal fuel/air ratio is found to be 0.0221 and the actual fuel/air ratio is then (0.0221/0.99) = 0.0223.

Hence the fuel flow is given by

$$m_f = 0.0223 * 35.83 * 3600 = 2876.4 \text{ kg/h}$$

and

$$\text{SFC} = \frac{2876.4}{71463} = 0.0403 \text{ kg/h N}$$

- Because both nozzles were unchoked, the thrust could be evaluated without calculating the nozzle areas. It is always good practice, however, to calculate key pieces of information which may be required for other purposes. In both cases, the area can be calculated from continuity, i.e. $m = \rho AC$. The density is obtained from $\rho = p/RT$, where p and T are the static values in the plane of the nozzle; for both the nozzles $p = p_a$.
- The following results are obtained for the two streams:

	Cold	Hot
Static pressure (bar)	1.0	1.0
Static temperature (K)	192.6	749.8
Density (kg/m^3)	1.191	0.4647
Mass flow (kg/s)	179.2	35.83
Velocity (m/s)	293.2	528.3
Nozzle area (m^2)	0.5132	0.1459

- The cold nozzle area is much larger than the hot one.

- This example illustrated the method followed when a propelling nozzle is unchoked while the previous example showed how a choked nozzle may be dealt with.
- Note that under static conditions, the bypass stream contributes approximately 74 percent of the total thrust. At a forward speed of 60 m/s , which is approaching a normal take-off speed, the momentum drag mC_a will be 215 * 60 or 12900 N ; the ram pressure ratio and temperature rise will be negligible and thus the net thrust is reduced to 58570 N.
- The drop in thrust during take-off is even more marked for engines of higher bypass ratio and for this reason it is preferable to quote turbofan thrusts at a typical take-off speed rather than at static conditions.

Advanced Thermal Engineering – Group Project titles

Performance analysis of a boiler draught system.

Application of steam power plant power generation for sugar industries.

Analysis of cogeneration systems in sugar cane factories.

Steam Turbines for large power applications.

Design of a steam condenser for a large steam power plant.

Piping design calculations in Steam boiler and condenser.

Study of stress and vibration in piping system of Steam power plant.

Analysis of stress concentration at root of Steam turbine blade.

Feasibility of Gas turbine with Fuel cell technology in passenger aircraft.

Design development of micro turbines.

Case Studies:

Study the importance of piping in Boilers

Design of Chimneys in Steam power plant

Steam power and environmental effect

Water resources and steam power plants in India

Gas Power generation and future requirements.

Turbojet engine and future technologies

Pollution with steam and gas power plants

Ash handling and ash utilization in Cement Industries