Speed of the engine	= 315 r.p.m.
Indicated m.e.p.	= 7 kgf/cm^2
B.H.P. of the engine	= 330
Fuel consumption	= 80 kg/hour
Calorific value of fuel	= 10500 kcal/kg
Hydrogen content in hydrocarbon fuel	= 13%
Air consumption	= 30 kg/min
Cooling water circulated	= 90 kg/min
Rise in temperature of cooling water	= 38°C
Exhaust gas temperature	= 324°C
Specific heat of air	= 0.24
Specific heat of exhaust gases	= 0.26
Ambient temperature	= 24°C
Specific heat for superheated steam	= 0.5
Partial pressure of steam in exhaust gases	= 0.030 kgf/cm^2

Find :

(*i*) Mechanical efficiency, indicated thermal efficiency and specific fuel consumption on I.H.P. basis.
(*ii*) Cost of energy per kWh assuming cost of fuel as Rs. 2.50 per litre and specific gravity of fuel as 0.93.
(*iii*) Draw up the heat balance on minute and percentage basis.

Find : (*i*) The fixed cost of power generation per kW per annum ; (*ii*) The total cost of power generation per kWh. Cost of primary distribution is chargeable to generation.

11. Write brief notes on the following :
(*a*) Lubrication system in diesel power plant ; (*b*) Intercooling in gas turbine plant ;
(*c*) Earthing of a power system ; (*d*) Flat plate solar collectors.

12. What are the advantages and disadvantages of diesel power plant over steam power plant ? What is the status of diesel power plants in our country ?

13. (*a*) Make a neat layout line diagram of a diesel power plant showing in particular the fuel system and the water cooling system.
(*b*) What are the primary advantages of a diesel power plant ?
(*c*) Cooling water from a 400 kW diesel power plant enters a cooling tower at 60°C and leaves the tower at 37.8°C in an atmosphere with a dry bulb temperature of 32.2°C and a wet bulb temperature of 27.8°C. If thermal efficiency of the plant is 30% and 32% of the energy input is lost to the cooling water, estimate the kg/min of cooling water handled by the tower and the cooling efficiency of the tower.

5

Gas Turbine Power Plants

5.1. GAS TURBINES—GENERAL ASPECTS

Probably a windmill was the first turbine to produce useful work, wherein there is no pre-compression and no combustion. The characteristic features of a gas turbine as we think of the name today include a *compression process* and a *heat-addition* (or combustion) process. The gas turbine represents perhaps the *most satisfactory way of producing very large quantities of power in a self-contained and compact unit.* The gas turbine may have an ample future use in conjunction with the oil engine. For smaller gas turbine units, the inefficiencies in compression and expansion processes become greater and to improve the thermal efficiency it is necessary to use a heat exchanger. In order that a small gas turbine may compete for economy with the small oil engine or petrol engine it is necessary that a compact effective heat exchanger be used in the gas turbine cycle. The thermal efficiency of the gas turbine alone is still quite modest 20 to 30% compared with that of a modern steam plant 38 to 40%. It is possible to construct *combined plants* whose efficiencies are of order of 45% or more. Higher efficiencies might be attained in future.

The following are the major fields of application of gas turbines :

1. Aviation
2. Power generation
3. Oil and gas industry
4. Marine propulsion.

The efficiency of a gas turbine is not the criteria for the choice of this plant. A gas turbine is used in *aviation* and marine fields because *it is self contained, light weight not requiring cooling water and generally fit into the overall shape of the structure.* It is selected for *'power generation'* because of its *simplicity, lack of cooling water, needs quick installation and quick starting.* It is used in *oil and gas industry* because of *cheaper supply* of *fuel and low installation cost.*

The gas turbines have the following *"limitations"* :

1. They are not self starting.
2. Low efficiencies at part loads.
3. Non-reversibility.
4. Higher rotor speeds.
5. Low overall plant efficiency.

In the last two decides, rapid progress has been observed in the *development and improvement of the gas turbine plants for electric power production*. The major progress has been observed in the following *directions* :

(*i*) Increase in unit capacities of gas turbine units.
(*ii*) Increase in their efficiency.
(*iii*) Drop in capital cost.

5.2. APPLICATIONS OF GAS TURBINE PLANTS

Gas turbine plants for the purpose of power plant engineering find the following *applications* :

1. To drive generators and supply peak loads in steam, diesel or hydroplants.
2. To work as combination plants with conventional steam boilers.
3. To supply mechanical drive for auxiliaries.

— These plants are well suited for *peak load service* since the fuel costs are somewhat higher and initial cost low. Moreover, *peak load operation permits use of water injection which increases turbine work by about 40% with an increase in heat rate of about 20%.* The short duration of increase in heat rate does not prove of any much harm.

— The combination arrangement of gas turbines with conventional boilers may be supercharging or for heat recovery from exhaust gases. In the *supercharging system* air is supplied to the boiler under pressure by a compressor mounted on the common shaft with turbine and gases formed as result of combination after coming out of the boiler pass through the gas turbine before passing through the economiser and the chimney.

— The application of the gas turbine to drive the auxiliaries is not strictly included under direct electric power generation by the turbines and would not be discussed.

5.3. ADVANTAGES AND DISADVANTAGES OF GAS TURBINE POWER PLANTS OVER DIESEL AND THERMAL POWER PLANTS

A. Advantages over Diesel Plants :

1. The work developed per kg of air is large compared with diesel plant.
2. Less vibrations due to perfect balancing.
3. Less space requirements.
4. Capital cost considerably less.
5. Higher mechanical efficiency.
6. The running speed of the turbine (40,000 to 100,000 r.p.m.) is considerably large compared to diesel engine (1000 to 2000 r.p.m.).
7. Lower installation and maintenance costs.
8. The torque characteristics of turbine plants are far better than diesel plants.
9. The ignition and lubrication systems are simpler.

10. The specific fuel consumption does not increase with time in gas turbine plant as rapidly as in diesel plants.
11. Poor quality fuels can be used.

Disadvantages :

1. Poor part load efficiency.
2. Special metals and alloys are required for different components of the plants.
3. Special cooling methods are required for cooling the turbine blades.
4. Short life.

B. Advantages Over Steam Power Plant

1. No ash handling problem.
2. Low capital cost.
3. The gas turbine plants can be installed at selected load centre as space requirement is considerably less where steam plant cannot be accommodated.
4. Fewer auxiliaries required/used.
5. Gas turbines can be built relatively quicker. They require much less space and civil engineering works and water supply.
6. The gas turbine plant as peak load plant is more preferable as it can be brought on load quickly and surely.
7. The components and circuits of a gas turbine plant can be arranged to give the most economic results in any given circumstances which is not possible in case of steam power plants.
8. For the same pressure and initial temperature conditions the ratio of exhaust to inlet volume would be only 3.95 in case of gas turbine plant as against 250 for steam plant.
9. Above 550°C, the thermal efficiency of the gas turbine plant increases three times as fast the steam cycle efficiency for a given top temperature increase.
10. The site of the steam power plant is dictated by the availability of large cooling water whereas an open cycle gas turbine plant can be located near the load centre as no cooling water is required. The cooling water required for closed cycle gas turbine is hardly 10% of the steam power plant.
11. The gas turbine plants can work quite economically for short running hours.
12. Storage of fuel is much smaller and handling is easy.

5.4. SITE SELECTION

While selectinig the site for a gas turbine plant. The following *points* should be given due consideration :

1. The plant should be located near the load centre to avoid transmission costs and losses.
2. The site should be away from business centres due to noisy operations.
3. Cheap and good quality fuel should be easily available.
4. Availability of labour.
5. Availability of means of transportation.
6. The land should be available at a cheap price.
7. The bearing capacity of the land should be high.

5.5. THE SIMPLE GAS TURBINE PLANT

A gas turbine plant may be defined as one *"in which the principal prime-mover is of the turbine type and the working medium is a permanent gas"*.

Refer Fig. 5.1. A simple gas turbine plant consists of the following :

1. *Turbine.*
2. *A compressor* mounted on the same shaft or coupled to the turbine.
3. *The combustor.*
4. *Auxiliaries* such as starting device, auxiliary lubrication pump, fuel system, oil system and the duct system etc.

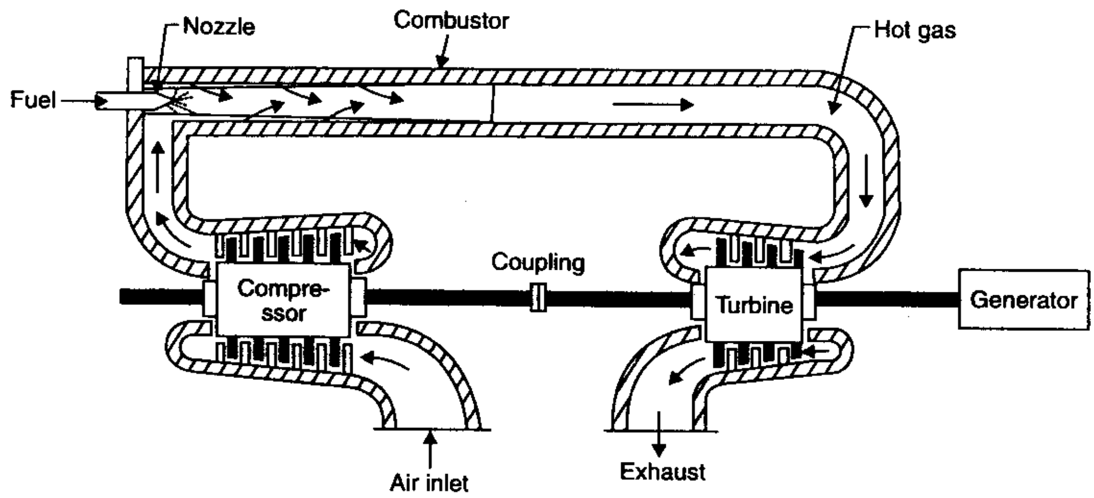


Fig. 5.1. Arrangement of a simple gas turbine plane.

A modified plant may have in addition to above an *intercooler, a regenerator, a reheater* etc.

The working fluid is compressed in a compressor which is generally rotary, multistage type. Heat energy is added to the compressed fluid in the chamber. This high energy fluid, at high temperature and presssure, then expands in the turbine unit thereby generating power. Part of the power generated is consumed in driving the generating compressor and accessories and the rest is utilised in electrical energy. The gas turbines work on open cycle, semi-closed cycle or closed cycle. In order to improve efficiency, compression and expansion of working fluid is carried out in multistages.

5.6. ENERGY CYCLE FOR A SIMPLE-CYCLE GAS TURBINE

Fig. 5.2 shows an energy-flow diagram for a simple-cycle gas turbine, the description of which is as follows :

— The air brings in minute amount of energy (measured above 0°C).
— Compressor adds considerable amount of energy.
— Fuel carries major input to cycle.
— Sum of fuel and compressed-air energy leaves combustor to enter turbine.
— In turbine smallest part of entering energy goes to useful output, largest part leaves in exhaust.

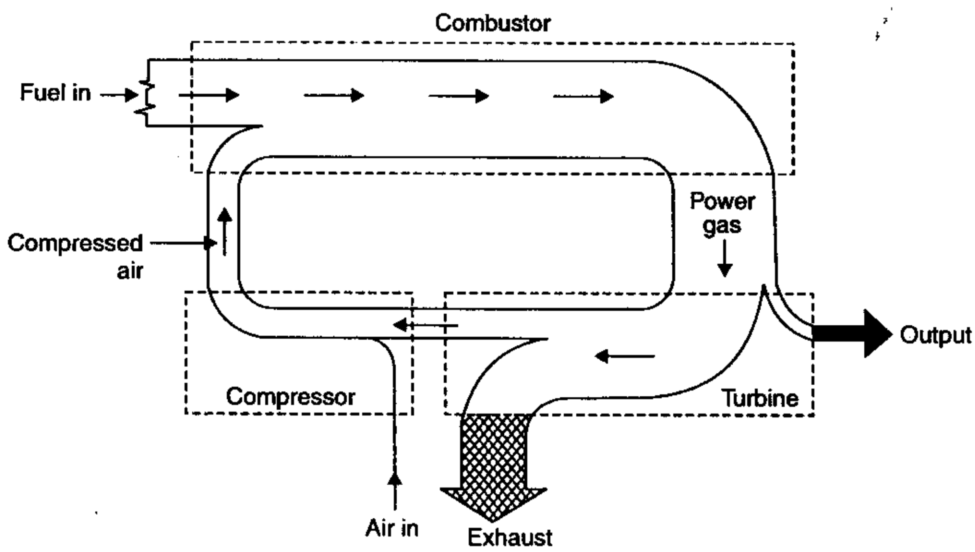


Fig. 5.2. Energy flow diagram for gas-turbine unit.

Shaft energy to drive compressor is about twice as much as the useful shaft output.

Actually the shaft energy keeps circulating in the cycle as long as the turbine runs. The important comparison is *the size of the output with the fuel input.* For the simple-cycle gas turbine the output may run about 20 per cent of the fuel input for pressure and temperature conditions at turbine inlet. This means 80% of the fuel energy is wasted. While the 20% thermal efficiency is not too bad, it can be improved by including *additional heat recovery apparatus.*

5.7. PERFORMANCE TERMS

Some of the important terms used to measure performance of a gas turbine are defined as follows :

1. **Pressure ratio.** It is the *ratio of cycle's highest to its lowest pressure,* usually highest-pressure-compressor discharges to the lowest-pressure-compressor inlet pressures.

2. **Work ratio.** It is the *ratio of network output to the total work developed in the turbine or turbines.*

3. **Air ratio.** *kg of air entering the compressor inlet per unit of cycle net output,* for example, kg/kWh.

4. **Compression efficiency.** It is the *ratio of work needed for ideal air compression through a given pressure range to work actually used by the compressor.*

5. **Engine efficiency.** It is the *ratio of the work actually developed by the turbine expanding hot power gas through a given pressure range to that would be yielded for ideal expansion conditions.*

6. **Machine efficiency.** It is the collective term *meaning both engine efficiency and compressor efficiency of turbine and compressor, respectively.*

7. **Combustion efficiency.** It is the *ratio of heat actually released by 1 kg of fuel to heat that would be released by complete perfect combustion.*

8. **Thermal efficiency.** It is the *percentage of total energy input appearing as net work output of the cycle.*

5.8. CLASSIFICATION OF GAS TURBINE POWER PLANTS

The gas turbine power plants may be classified according to the following criteria :

1. By application :

(*i*) *In aircraft*

(*a*) Jet propulsion (*b*) Prop-jets

(*ii*) *Stationary*

(*a*) Peak load unit (*b*) Standby unit

(*c*) End of transmission line unit (*d*) Base load unit

(*e*) Industrial unit.

(*iii*) Locomotive

(*iv*) Marine

(*v*) Transport.

2. By cycle :

(*i*) Open cycle (*ii*) Closed cycle

(*iii*) Semi-closed cycle.

3. According to arrangement :

(*i*) Simple (*ii*) Single shaft

(*iii*) Multi-shaft (*iv*) Intercooled

(*v*) Reheat (*vi*) Regenerative

(*vii*) Combination.

4. According to combustion :

(*i*) Continuous combustion (*ii*) Intermittent combustion.

5. By fuel :

(*i*) Solid fuel (*ii*) Liquid fuel

(*iii*) Gaseous fuel.

5.9. CLASSIFICATION OF GAS TURBINES

The gas turbines are mainly divided into two groups :

1. **Constant pressure combustion gas turbine**

(*a*) Open cycle constant pressure gas turbine

(*b*) Closed cycle constant pressure gas turbine.

2. **Constant volume combustion gas turbine**

- In almost *all the fields open cycle gas turbine plants are used. Closed cycle plants were introduced at one stage because of their ability to burn cheap fuel.* In between their progress remained slow because of availability of cheap oil and natural gas. Because of rising oil prices, now again, the attention is being paid to closed cycle plants.

5.10. MERITS OF GAS TURBINES

(*i*) **Merits over I.C. engines :**

1. The mechanical efficiency of a gas turbine (95%) is quite high as compared with I.C. engine (85%) since the I.C. engine has many sliding parts.

2. A gas turbine does not require a flywheel as the torque on the shaft is continuous and uniform. Whereas a flywheel is a must in case of an I.C. engine.
3. The weight of gas turbine per kW developed is less than that of an I.C. engine.
4. The gas turbine can be driven at a very high (40000 r.p.m.) whereas this is not possible with I.C. engines.
5. The work developed by a gas turbine per kg of air is more as compared to an I.C. engine. This is due to the fact that gases can be expanded upto atmospheric pressure in case of a gas turbine whereas in an I.C. engine expansion upto atmospheric pressure is not possible.
6. The components of the gas turbine can be made lighter since the pressures used in it are very low, say 5 bar compared with I.C. engine, say 60 bar.
7. In the gas turbine the ignition and lubrication systems are much simpler as compared with I.C. Engines.
8. Cheaper fuels such as paraffine type, residue oils or powdered coal can be used whereas special grade fuels are employed in petrol engine to check knocking or pinking.
9. The exhaust from gas turbine is less polluting comparatively since excess air is used for combustion.
10. Because of low specific weight the gas turbines are particularly suitable for use in aircrafts.

Demerits of gas turbines :

1. The thermal efficiency of a simple turbine cycle is low (15 to 20%) as compared with I.C. engines (25 to 30%).
2. With wide operating speeds the fuel control is comparatively difficult.
3. Due to higher operating speeds of the turbine, it is imperative to have a speed reduction device.
4. It is difficult to start a gas turbine as compared to an I.C. engine.
5. The gas turbine blades need a special cooling system.

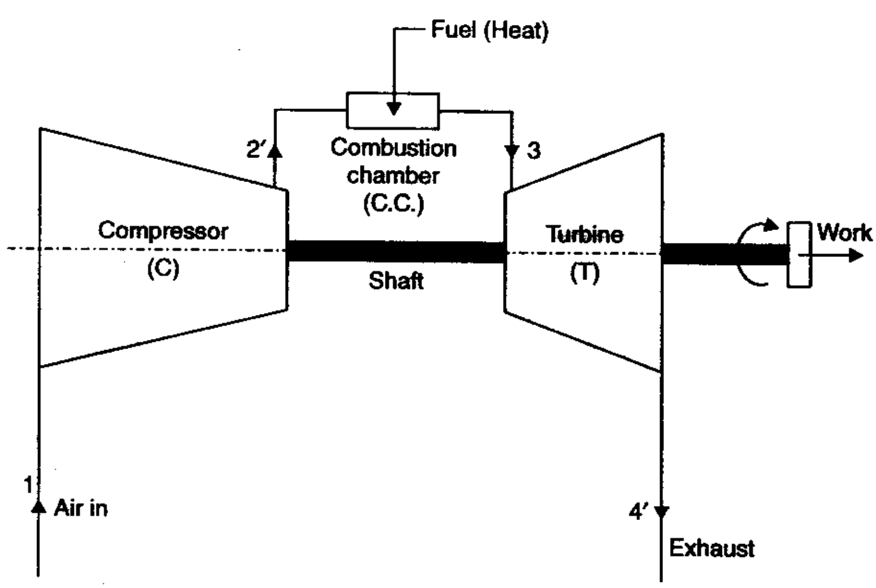


Fig. 5.3. Open cycle gas turbine.

6. One of the main demerits of a gas turbine is its *very poor thermal efficiency at part loads,* as the quantity of air remains same irrespective of load, and output is reduced by reducing the quantity of fuel supplied.
7. Owing to the use of nikel-chromium alloy, the manufacture of the blades is difficult and costly.
8. For the same output the gas turbine produces five times exhaust gases than I.C. engine.
9. Because of prevalence of high temperature (1000 K for blades and 2500 K for combustion chamber) and centrifugal force the life of the combustion chamber and blades is short/small.

(ii) **Merits over steam turbines :**

The gas turbine entails the following *advantages over steam turbines* :

1. Capital and running cost less.
2. For the same output the space required is far less.
3. Starting is more easy and quick.
4. Weight per H.P. is far less.
5. Can be installed anywhere.
6. Control of gas turbine is much easier.
7. Boiler along with accessories not required.

5.11. CONSTANT PRESSURE COMBUSTION GAS TURBINES

5.11.1. Open Cycle Gas Turbines

Refer Fig. 5.3. The fundamental gas turbine unit is one operating on the open cycle in which a rotary compressor and a turbine are mounted on a common shaft. Air is drawn into the compressor and after compression passes to a combustion chamber. Energy is supplied in the combustion chamber by spraying fuel into the air stream, and the resulting hot gases expand through the turbine to the atmosphere. In order to achieve net work output from the unit, the turbine must develop more gross work output than is required to drive the compressor and to overcome mechanical losses in the drive. The products of combustion coming out from the turbine are exhausted to the atmosphere as they cannot be used any more. The working fluids (air and fuel) must be replaced continuously as they are exhausted into the atmosphere.

If pressure loss in the combustion chamber is neglected, this cycle may be drawn on a T-s diagram as shown in Fig. 5.4.

Line 1-2′ represents : *irreversible adiabatic compression.*

Line 2′-3 represents : *constant pressure heat supply in the combustion chamber.*

Line 3-4′ represents : *irreversible adiabatic expansion.*

Line 1-2 represents : *ideal isentropic compression.*

Line 3-4 represents : *ideal isentropic expansion.*

Assuming change in kinetic energy between the various points in the cycle to be negligibly small compared with enthalpy changes and then applying the flow equation to each part of cycle, for unit mass, we have

Work input (compressor) $= c_p (T_2' - T_1)$

Heat supplied (combustion chamber) $= c_p (T_3 - T_2')$

Work output (turbine) $= c_p (T_3 - T_4')$

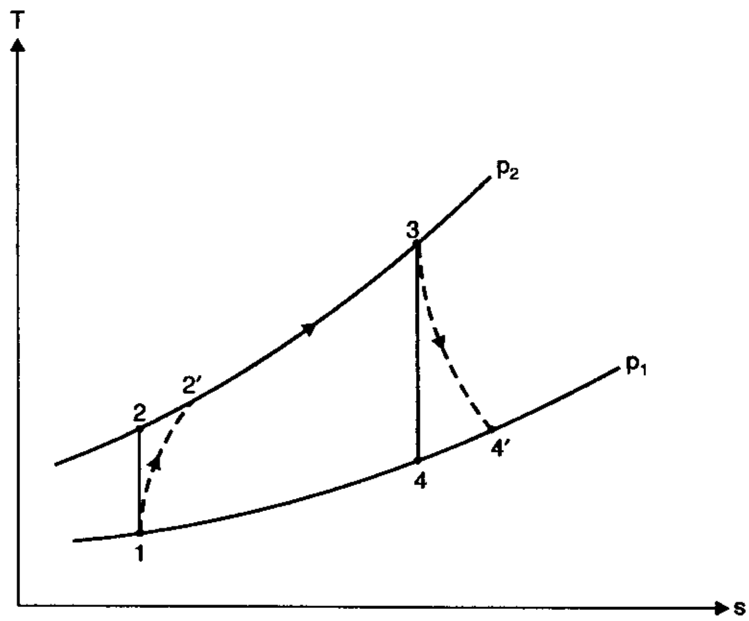


Fig. 5.4

$\therefore$ Net work output = Work output – work input

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

and
$$\eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}}$$

$$= \frac{c_p (T_3 - T_4') - c_p (T_2' - T_1)}{c_p(T_3 - T_2')}$$

Compressor isentropic efficiency, $\eta_{comp.}$

$$= \frac{\text{Work input required in isentropic compression}}{\text{Actual work required}}$$

$$= \frac{c_p(T_2 - T_1)}{c_p(T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1} \qquad ...(5.1)$$

Turbine isentropic efficiency, $\eta_{turbine}$

$$= \frac{\text{Actual work output}}{\text{Isentropic work output}}$$

$$= \frac{c_p (T_3 - T_4')}{c_p(T_3 - T_4)} = \frac{T_3 - T_4'}{T_3 - T_4} \qquad ...(5.2)$$

Note. With the variation in temperature, the value of the specific heat of a real gas varies, and also in the open cycle, the specific heat of the gases in the combustion chamber and in turbine is different from that in the compressor because fuel has been added and a chemical change has taken place. Curves showing the variation of c_p with temperature and air/fuel ratio can be used, and a suitable mean value of c_p and hence γ can be found out. It is usual in gas turbine practice to assume fixed mean value of c_p and γ for the expansion process, and fixed mean values of c_p and γ for the compression process. In an open cycle gas turbine unit the mass flow of gases in turbine is greater than that in compressor due to mass of fuel burned, but it is possible to neglect mass of fuel, since the air/fuel ratios used are large. Also, in many cases, air is bled from the compressor for cooling purposes, or in the case of air-craft at high altitudes, bled air is used for de-icing and cabin air-conditioning. This amount of air bled is approximately the same as the mass of fuel injected therein.

5.11.2. Methods for Improvement of Thermal Efficiency of Open Cycle Gas Turbine Plant

The following methods are employed to increase the specific output and thermal efficiency of the plant :

1. Intercooling 2. Reheating 3. Regeneration.

1. Intercooling. A compressor in a gas turbine cycle utilises the major percentage of power developed by the gas turbine. The work required by the compressor can be reduced by compressing the air in two stages and incorporating an intercooler between the two as shown in Fig. 5.5. The corresponding *T-s* diagram for the unit is shown in Fig. 5.6 (*a*). The actual processes take place as follows :

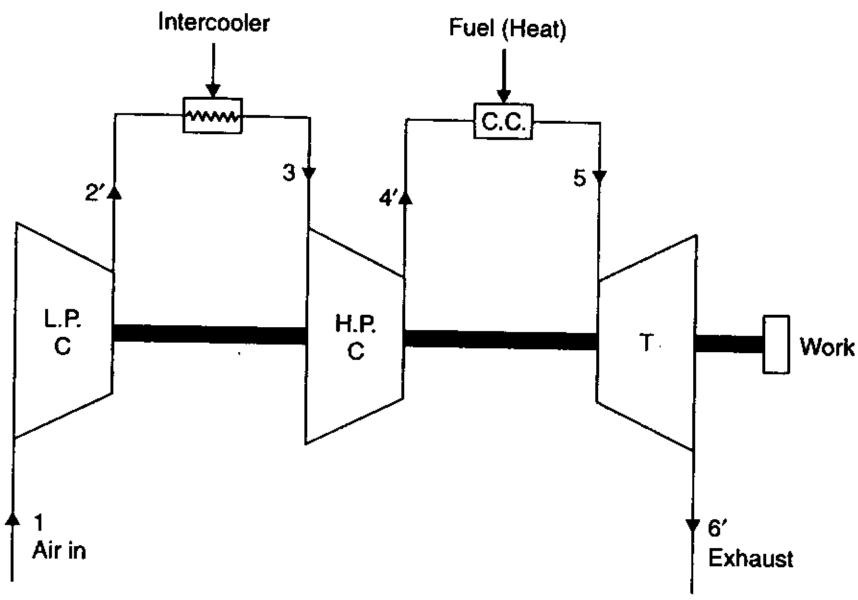


Fig. 5.5. Turbine plant with intercooler.

1-2′ : *L.P. (Low pressure) compression*
2′-3 : *Intercooling*
3-4′ : H.P. (High pressure) compression
4′-5 : C.C. (Combustion chamber)–heating
5-6′ : T (Turbine)–expansion

The ideal cycle for this arrangement is 1-2-3-4-5-6 ; the compression process without intercooling is shown as 1-*L′* in the actual case, and 1-*L* in the ideal isentropic case.

Now, work input (with intercooling)

$$= c_p(T_2' - T_1) + c_p(T_4' - T_3) \quad ...(5.3)$$

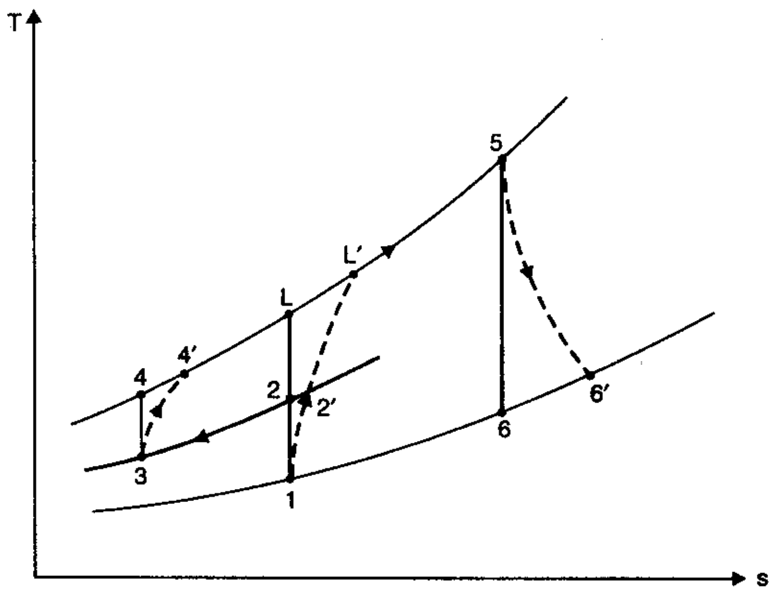


Fig. 5.6. (*a*) *T-s* diagram for the unit.

Work input (without intercooling)

$$= c_p(T_L' - T_1) = c_p(T_2' - T_1) + c_p(T_L' - T_2') \quad ...(5.4)$$

By comparing equation (5.4) with equation (5.3) it can be observed that the *work input with intercooling is less than the work input with no intercooling*, when $c_p\,(T_4' - T_3)$ is *less* than $c_p(T_L' - T_2')$. This is so if it is assumed that isentropic efficiencies of the two compressors, operating separately, are each equal to the isentropic efficiency of the single compressor which would be required if no intercooling were used. Then $(T_4' - T_3) < (T_L' - T_2')$ since the pressure lines diverge on the *T-s* diagram from left to the right.

$$\text{Again, work ratio} = \frac{\text{Net work output}}{\text{Gross work output}}$$

$$= \frac{\text{Work of expansion} - \text{work of compression}}{\text{Work of expansion}}$$

From this we may conclude that *when the compressor work input is reduced then the work ratio is increased.*

However the heat supplied in the combustion chamber when intercooling is used in the cycle, is given by,

Heat supplied *with* intercooling $= c_p(T_5 - T_4')$

Also the heat supplied when intercooling is *not* used, with the same maximum cycle temperature T_5, is given by

Heat supplied *without* intercooling $= c_p\,(T_5 - T_L')$

Thus, the *heat supplied when intercooling is used is greater than with no intercooling. Although the net work output is increased by intercooling it is found in general that the increase in heat to be supplied causes the thermal efficiency to decrease.* When intercooling is used a supply of cooling water must be readily available. The additional bulk of the unit may offset the advantage to be gained by increasing the work ratio.

In Fig. 5.6 (*b*) is shown that the intercooling has marked effect on thermal efficiency only at high pressure ratio.

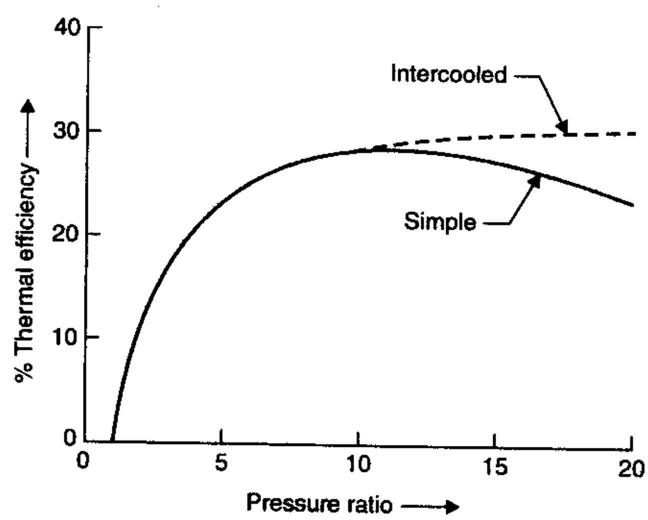


Fig. 5.6 (*b*)

2. Reheating. The output of a gas turbine can be amply improved by expanding the gases in two stages with a reheater between the two as shown in Fig. 5.7. The H.P. turbine drives the compressor and the L.P. turbine provides the useful power output. The corresponding *T-s* diagram is shown in Fig. 5.8 (*a*). The line 4′-L′ represents the expansion in the L.P. turbine if reheating is not employed.

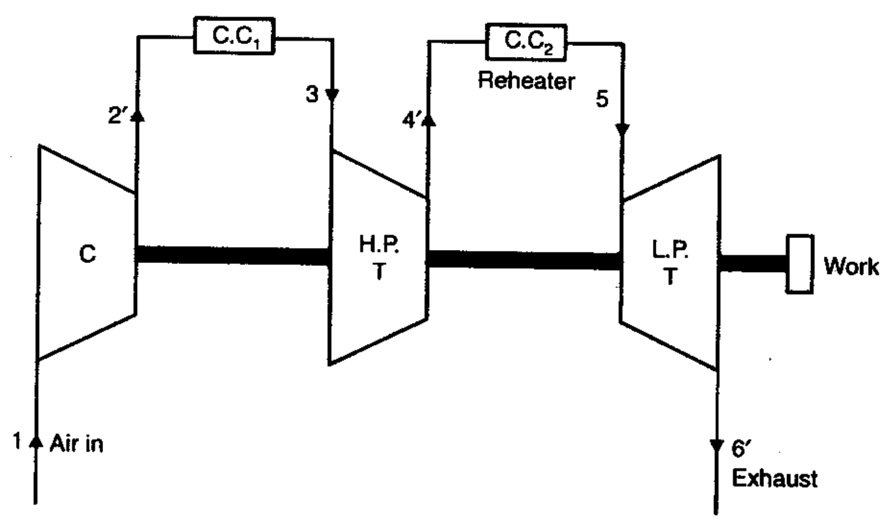


Fig. 5.7. Gas turbine with reheater.

Neglecting mechanical losses the work output of the H.P. turbine must be exactly equal to the work input required for the compressor

i.e., $$c_{pa}(T_2' - T_1) = c_{pg}(T_3 - T_4')$$

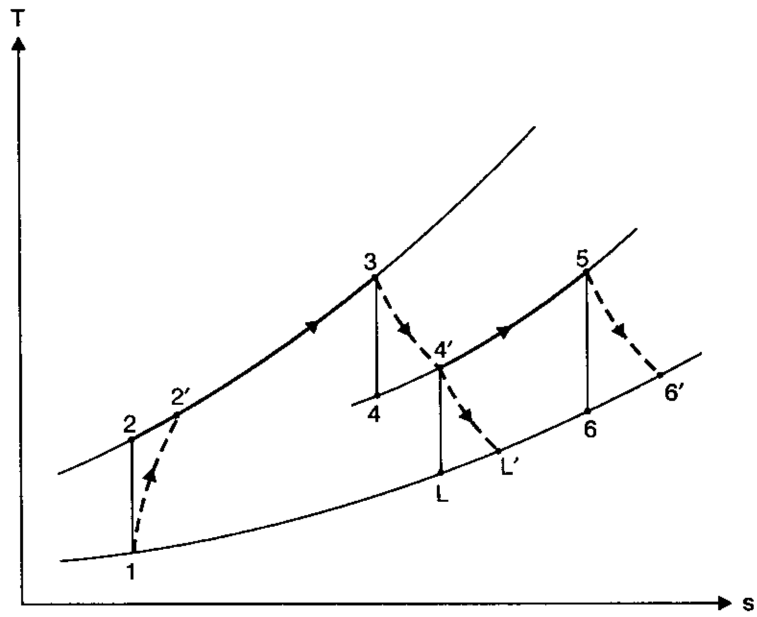


Fig. 5.8. (*a*) *T-s* diagram for the unit.

The work output (net output) of L.P. turbine is given by,

Net work output (with reheating) $= c_{pg} (T_5 - T_6')$

and Net work output (without reheating) $= c_{pg} (T_4' - T_L')$

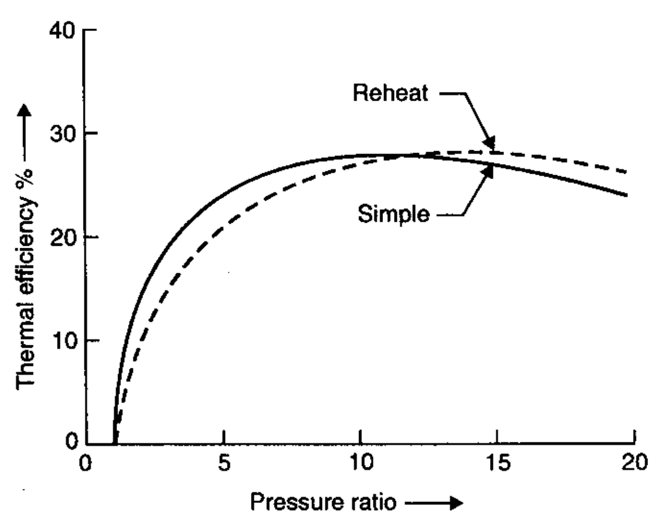


Fig. 5.8 (*b*)

Since the pressure lines diverge to the right on *T-s* diagram it can be seen that the temperature difference $(T_5 - T_6')$ is always *greater* than $(T_4' - T_L')$, so that *reheating increases the net work output.*

Although net work is increased by reheating the heat to be supplied is also increased, and the net effect *can be to reduce the thermal efficiency* [Fig. 5.8 (*b*)]

Heat supplied $= c_{pg} (T_3 - T_2') + c_{pg} (T_5 - T_4')$.

Note. c_{pa} and c_{pg} stand for specific heats of air and gas respectively at constant pressure.

3. Regeneration. The exhaust gases from a gas turbine carry a large quantity of heat with them since their temperature is far above the ambient temperature. They can be used to heat the air coming from the compressor thereby reducing the mass of fuel supplied in the combustion chamber. Fig. 5.9 shows a gas turbine plant *with a regenerator*. The corresponding *T-s* diagram is shown in Fig. 5.10. 2′-3 represents the heat flow into the compressed air during its passage through the heat exchanger and 3-4 represents the heat taken in from the combustion of fuel. Point 6 represents the temperature of exhaust gases at discharge from the heat exchanger. The maximum temperature to which the air could be heated in the heat exchanger is ideally that of exhaust gases, but less than this is obtained in practice because a temperature gradient must exist for an unassisted transfer of energy. The effectiveness of the heat exchanger is given by :

$$\text{Effectiveness,} \qquad \varepsilon = \frac{\text{Increase in enthalpy per kg of air}}{\text{Available increase in enthalpy per kg of air}}$$

$$= \frac{(T_3 - T_2')}{(T_5' - T_2')} \qquad \ldots(5.5)$$

(assuming c_{pa} and c_{pg} to be equal)

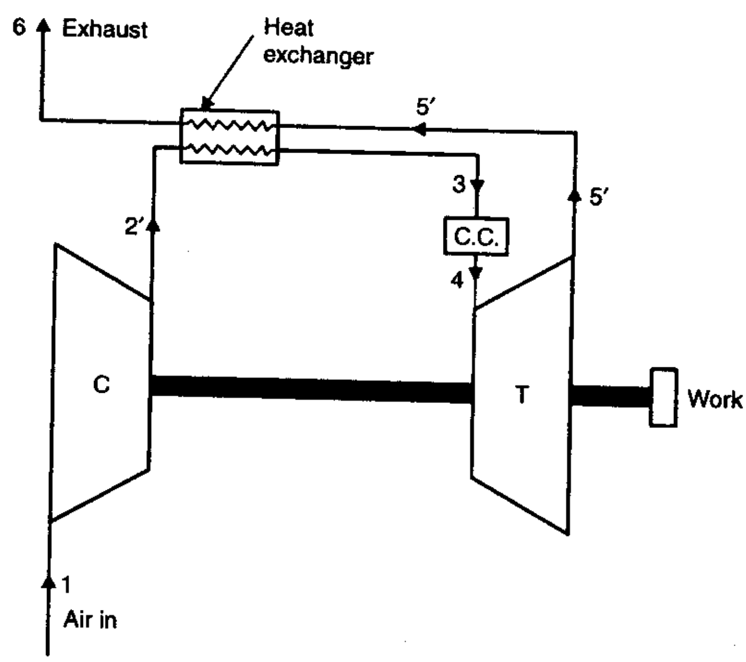


Fig. 5.9. Gas turbine with regenerator.

Fig. 5.10 *(b)* shows that the regenerative cycle has higher efficiency than the simple cycle only at low pressure ratios. At pressure ratios above a certain limit the efficiency of the regenerative cycle drops since in that case the regenerator will cool the compressed air entering the combustion chamber instead of heating it.

A heat exchanger is usually used in large gas turbine units for marine propulsion or industrial power.

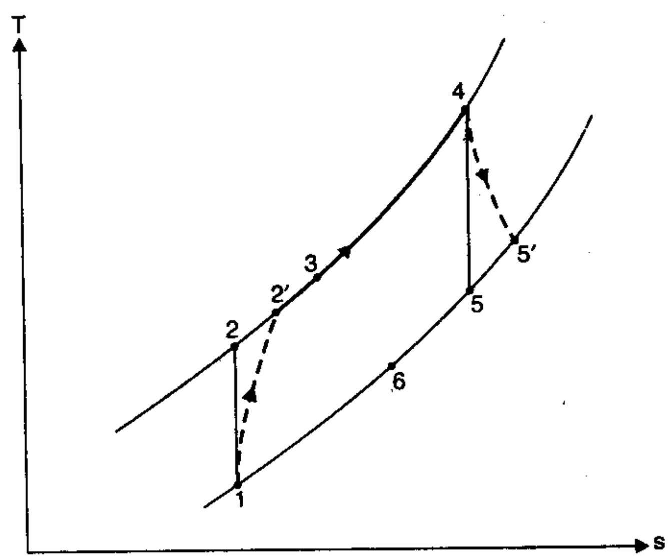


Fig. 5.10. (*a*) *T-s* diagram for the unit.

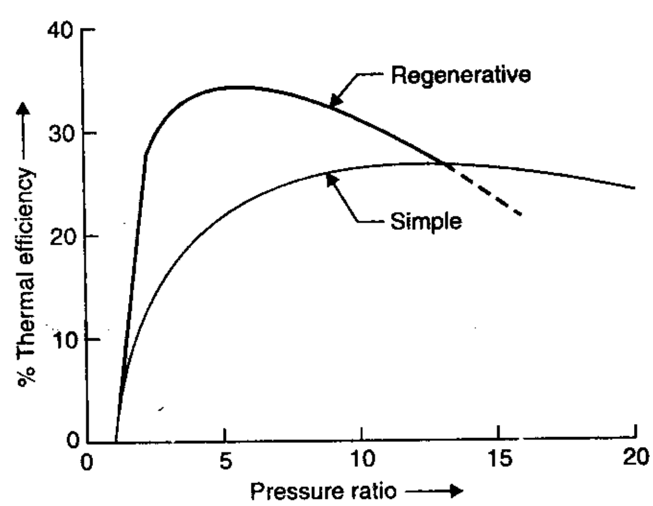


Fig. 5.10 (*b*)

5.11.3. Closed Cycle Gas Turbine (Constant pressure or joule cycle).

Fig. 5.11 shows a gas turbine operating on a constant pressure cycle in which the closed system consists of air behaving as an ideal gas. The various operations are as follows : Refer Figs. 5.12 and 5.13.

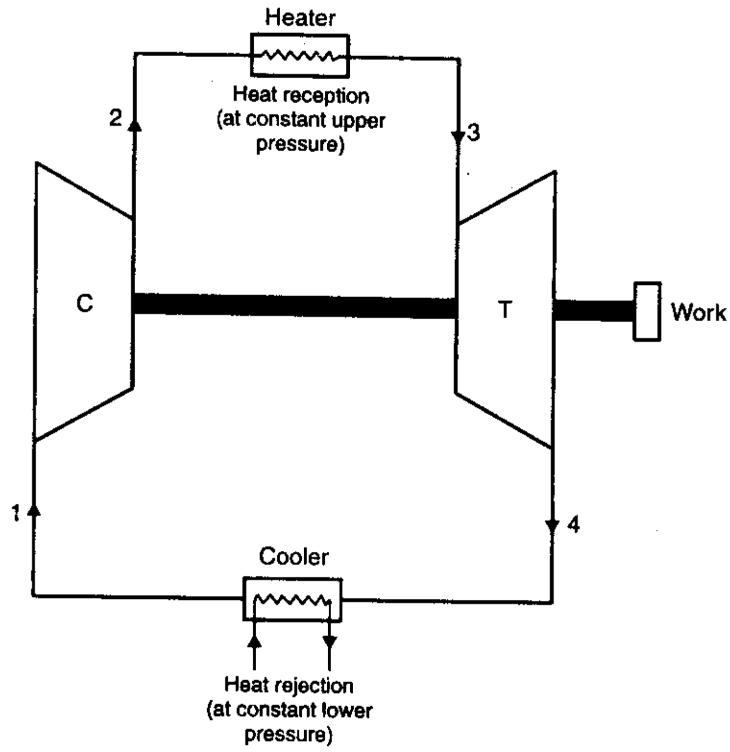


Fig. 5.11. Closed cycle gas turbine.

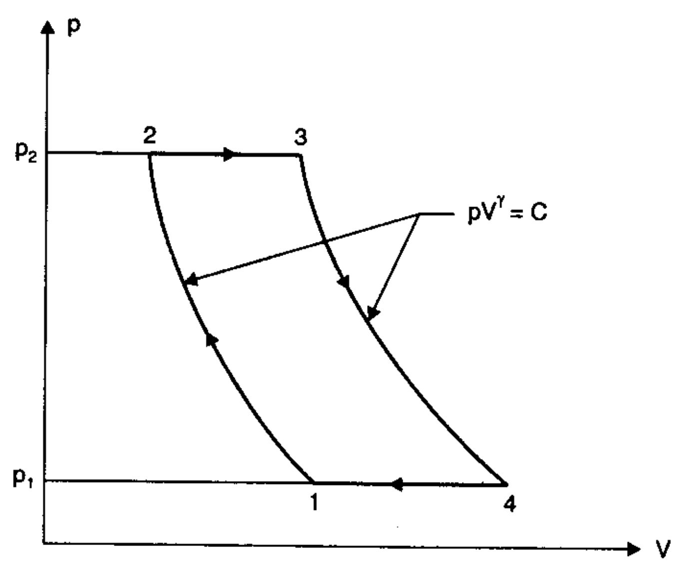


Fig. 5.12. *p-V* diagram.

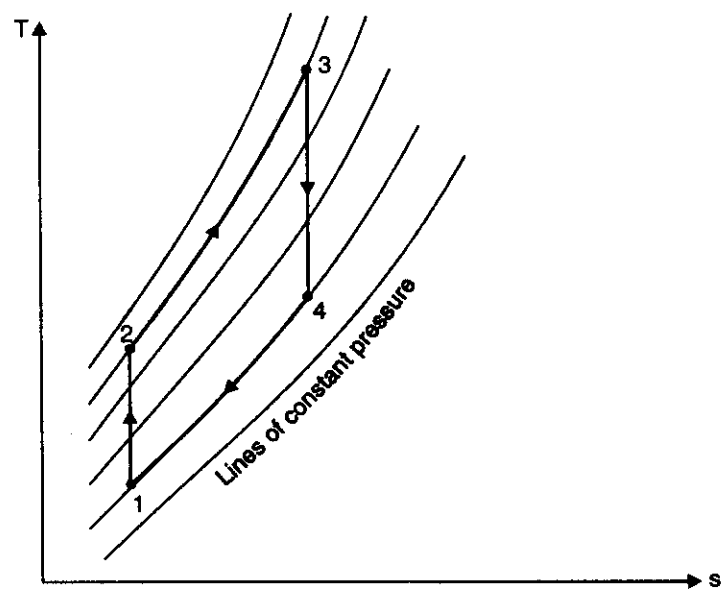


Fig. 5.13. *T-s* diagram.

Operation 1-2 : The air is compressed isentropically from the lower pressure p_1 to the upper pressure p_2, the temperature rising from T_1 to T_2. No heat flow occurs.

Operation 2-3 : Heat flows into the system increasing the volume from V_2 to V_3 and temperature from T_2 to T_3 whilst the pressure remains constant at p_2. Heat received = $mc_p\ (T_3 - T_2)$.

Operation 3-4 : The air is expanded isentropically from p_2 to p_1, the temperature falling from T_3 to T_4. No heat flow occurs.

Operation 4-1 : Heat is rejected from the system as the volume decreases from V_4 to V_1 and the temperature from T_4 to T_1 whilst the pressure remains constant at p_1. Heat rejected = $mc_p\ (T_4 - T_1)$.

$$\eta_{\text{air-standard}} = \frac{\text{Work done}}{\text{Heat received}}$$

$$= \frac{\text{Heat received cycle} - \text{heat rejected cycle}}{\text{Heat received cycle}}$$

$$= \frac{mc_p(T_3 - T_2) - mc_p(T_4 - T_1)}{mc_p(T_3 - T_2)} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Now, from isentropic expansion

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

$$T_2 = T_1\ (r_p)^{\frac{\gamma-1}{\gamma}}, \text{ where } r_p = \text{pressure ratio}$$

Similarly, $$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \text{ or } T_3 = T_4\,(r_p)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \quad \eta_{\text{air-standard}} = 1 - \frac{T_4 - T_1}{T_4(r_p)^{\frac{\gamma-1}{\gamma}} - T_1(r_p)^{\frac{\gamma-1}{\gamma}}} = 1 - \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}}} \qquad ...(5.6)$$

The expression shows that the *efficiency of the ideal joule cycle increases with the pressure ratio. The absolute limit of upper pressure is determined by the limiting temperature of the material of the turbine at the point at which this temperature is reached by the compression process alone, no further heating of the gas in the combustion chamber would be permissible and the work of expansion would ideally just balance the work of compression so that no excess work would be available for external use.*

We as designers are always interested in the percentage of work developed by the turbine available for electric generation.

$$\therefore \quad \text{Work ratio} = \frac{W_{\text{net}}}{W_{\text{turbine}}} = \frac{W_{\text{turbine}} - W_{\text{comp.}}}{W_{\text{turbine}}} = 1 - \frac{W_{\text{comp.}}}{W_{\text{turbine}}}$$

$$= 1 - \frac{m_{cp}\,(T_2 - T_1)}{m_{cp}\,(T_3 - T_4)} = 1 - \frac{T_2 - T_1}{T_3 - T_4} = 1 - \frac{T_1\,(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)} - T_1}{T_3 - \dfrac{T_3}{(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)}}}$$

$$= 1 - \frac{T_1}{T_3}\,(r_p)^{\left(\frac{\gamma-1}{\gamma}\right)} \qquad ...(5.7)$$

The *'work ratio' increases with increase in turbine inlet temperature, decrease in compressor inlet temperature and decrease in pressure ratio of the cycle.* The compressor inlet temperature is always atmospheric temperature particularly in open cycle plant and turbine inlet temperature is limited by metallurgical considerations. *The highest temperature ever used for gas turbine power plants is about 1000 K.*

From the eqn. (5.7), it is obvious that the maximum possible pressure ratio for the fixed temperature T_1 and T_3 is given by

$$(r_p)_{max} = \left(\frac{T_3}{T_1}\right)^{\left(\frac{\gamma}{\gamma-1}\right)} \qquad ...(5.8)$$

because, the value of r_p greater than this gives the negative value of work ratio which is impossible for power developing system. Thus, the maximum value of r_p to be used for finding the maximum possible efficiency in eqn. (5.6) is given by eqn. (5.8).

Now we shall prove that the **pressure ratio for maximum work is a function of the limiting temperature ratio.**

Work output during the cycle

$$= \text{Heat received / cycle} - \text{heat rejected / cycle}$$

$$= mc_p\,(T_3 - T_2) - mc_p\,(T_4 - T_1) = mc_p\,(T_3 - T_4) - mc_p\,(T_2 - T_1)$$

$$= mc_pT_3\left(1 - \frac{T_4}{T_3}\right) - T_1\left(\frac{T_2}{T_1} - 1\right)$$

In case of a given turbine the minimum temperature T_1 and the maximum temperature T_3 are prescribed, T_1 being the temperature of the atmosphere and T_3 the maximum temperature which the metals of turbine would withstand. Consider the specific heat at constant pressure c_p to be constant. Then,

Since, $$\frac{T_3}{T_4} = (r_p)^{\frac{\gamma-1}{\gamma}} = \frac{T_2}{T_1}$$

Using the constant $'z' = \frac{\gamma - 1}{\gamma}$,

we have, work output/cycle $$W = K\left[T_3\left(1 - \frac{1}{r_p^{\,z}}\right) - T_1(r_p^{\,z} - 1)\right]$$

Differentiating with respect to r_p

$$\frac{dW}{dr_p} = K\left[T_3 \times \frac{z}{r_p(z+1)} - T_1 z r_p^{\,(z-1)}\right] = 0 \text{ for a maximum}$$

$$\therefore \quad \frac{zT_3}{r_p^{\,(z+1)}} = T_1 z (r_p)^{(z-1)}$$

$$\therefore \quad r_p^{\,2z} = \frac{T_3}{T_1}$$

$$\therefore \quad r_p = \left(\frac{T_3}{T_1}\right)^{1/2z} \quad i.e., \quad r_p = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}} \qquad ...(5.9)$$

Thus the *pressure ratio for maximum work is a function on the limiting temperature ratio.*

Comparing eqns. (5.8) and (5.9), we get

$$r_p = \sqrt{(r_p)_{max}} \qquad ...(5.10)$$

- Fig. 5.14 shows an arrangement of closed cycle stationary gas turbine plant in which air is continuously circulated. This ensures that the air is not polluted by the addition of combustion waste product, since the heating of air is carried out in the form of heat exchanger shown in the diagram as air heater. The air exhausted from the power turbine is cooled before readmission to L.P. compressor. The various operations as indicated on T-s diagram (Fig. 5.15) are as follows :

Operation 1-2′ : Air is compressed from p_1 to p_x in the L.P. compressor.

Operation 2′-3 : Air is cooled in the intercooler at constant pressure p_x.

Operation 3-4′ : Air is compressed in the H.P. compressor from p_x to p_2.

Operation 4′-5 : High pressure air is heated at constant pressure by exhaust gases from power turbine in the heat exchanger to T_5.

Operation 5-6 : High pressure air further heated at constant pressure to the maximum temperature T_6 by an air heater (through external combustion).

Operation 6-7′ : The air is expanded in the H.P. turbine from p_2 to p_x producing work to drive the compressor.

Operation 7′-8 : Exhaust air from the H.P. turbine is heated at constant pressure in the air heater (through external combustion) to the maximum temperature T_8 (= T_6).

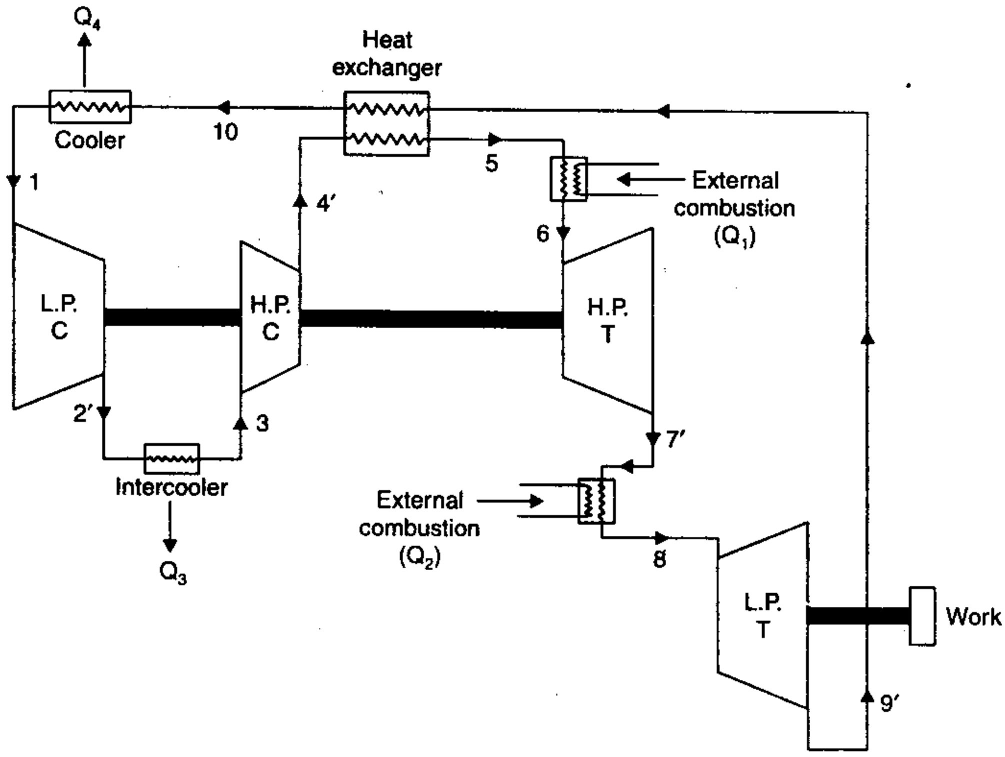


Fig. 5.14. Closed cycle gas turbine plant.

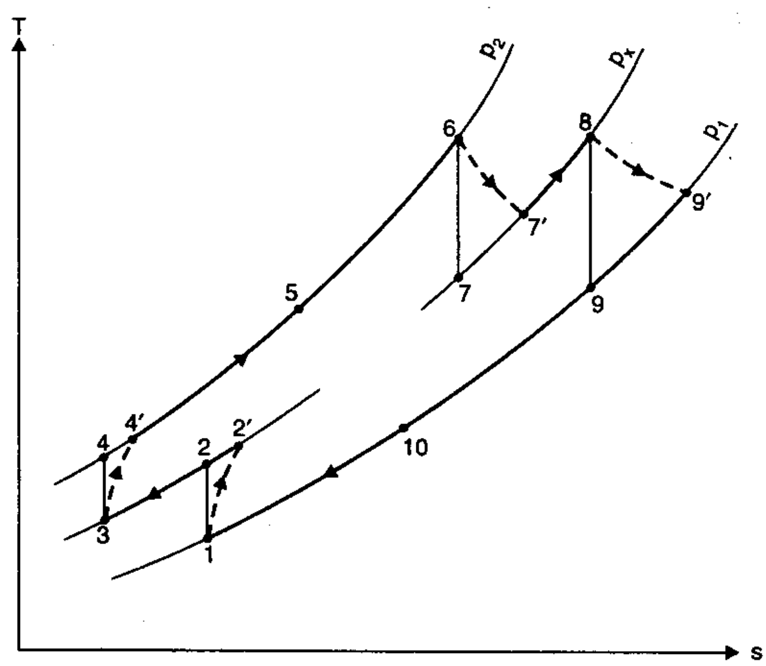


Fig. 5.15. T-s diagram for the plant.

Operation 8-9′ : The air is expanded in the L.P. turbine from p_x to p_1, producing energy for a flow of work externally.

Operation 9′-10 : Air from L.P. turbine is passed to the heat exchanger where energy is transferred to the air delivered from the H.P. compressor. The temperature of air leaving the heat exchanger and entering the cooler is T_{10}.

Operation 10-1 : Air cooled to T_1 by the cooler before entering the L.P. compressor.

The energy balance for the whole plant is as follows :

$$Q_1 + Q_2 - Q_3 - Q_4 = W$$

- In a closed cycle plant, in practice, the control of power output is achieved by varying the mass flow by the use of a reservoir in the circuit. The *reservoir maintains the design pressure and temperature and therefore achieves an approximately constant level of efficiency for varying* loads.
- In this cycle, since it is closed, *gases other than air with favourable properties can be used* ; furthermore it is possible to burn solid fuels in the combustion heaters.
- Fig. 5.16 shows the *comparison of thermal efficiency between steam and closed cycle gas turbine plants.*

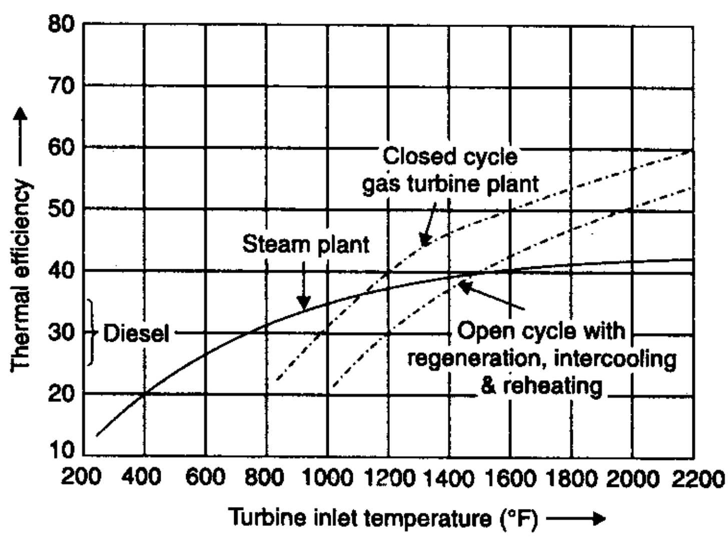


Fig. 5.16. Comparison of thermal efficiency between steam and closed cycle gas turbine plants.

— It is obvious from the figure that the gas turbine plant should be operated above 550°C to compete with the steam plants.

— Above 375°C, the gap between the steam cycle and gas turbine cycle efficiency widens.

— Above 550°C, the efficiency of the gas turbine plant increases 3 times as fast as the steam cycle efficiency for a given top temperature increase.

- *The major factor responsible for inefficiency in this cycle is the large irreversible temperature drop which occurs in the air heaters between the furnace and circulating gas.*

Note. 1. In a closed cycle gas turbines, although air has been extensively used, the use of 'helium' which though of a lower density, has been inviting the attention of manufacturers for its use, for large output gas turbine units. *The specific heat of helium at constant pressure is about 'five times' that of air, therefore for each kg mass* *flow the heat drop and hence energy dealt with in helium machines is nearly five times of those in case of air.* The *surface area of the heat exchanger for helium can be kept as low as 1/3* of that *required for gas turbine plant using air as working medium.* For the same temperature ratio and for the plants of the same output the *cross-sectional area required for helium is much less than that for air.* It may therefore be concluded that the *size of helium unit is considerably small comparatively.*

2. Some gas turbine plants work on a combination of two cycles the open cycle and the closed cycle. Such a combination is called the ***semi-closed cycle.*** *Here a part of the working fluid is confined within the plant and another part flows from and to atmosphere.*

5.11.4. Merits and Demerits of Closed Cycle Turbine Over Open Cycle Turbine

Merits of closed cycle :

1. Higher thermal efficiency
2. Reduced size
3. No contamination
4. Improved heat transmission
5. Improved part load efficiency
6. Lesser fluid friction
7. No loss of working medium
8. Greater output
9. Inexpensive fuel.

Demerits of closed cycle :

1. Complexity
2. Large amount of cooling water is required. This limits its use to stationary installation or marine use where water is available in abundance.
3. Dependent system.
4. The weight of the system per kW developed is high comparatively, therefore not economical for moving vehicles.
5. Requires the use of a very large air heater.

5.12. CONSTANT VOLUME COMBUSTION TURBINES

Refer Fig. 5.17. In a constant volume combustion turbine, the compressed air from an air compressor C is admitted into the combustion chamber D through the valve A. When the valve A

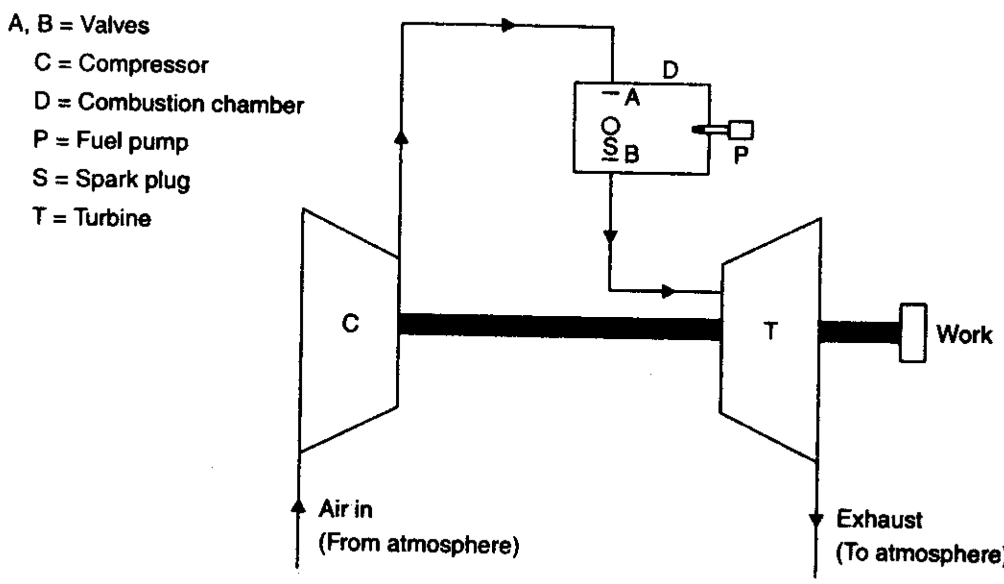


Fig. 5.17. Constant volume combustion gas turbine.

is closed, the fuel is admitted into the chamber by means of a fuel pump P. Then the mixture is ignited by means of a spark plug S. The combustion takes place at constant volume with increase of pressure. The valve B opens and the hot gases flow to the turbine T, and finally, they are discharged into atmosphere. The energy of the hot gases is thereby converted into mechanical energy. For continuous running of the turbine these operations are repeated.

The main demerit associated with this type of turbine is that the *pressure difference and velocities of hot gases are not constant ; so the turbine speed fluctuates.*

5.13. GAS TURBINE FUELS

The various fuels used in gas turbines are enumerated and discussed as follows :

1. Gaseous fuels 2. Liquid fuels 3. Solid fuels.

1. Gaseous fuels. *Natural gas is the ideal fuel for gas turbines,* but this is not available everywhere.

Blast furnace gas and *producer gas* may also be used for gas turbine power plants.

2. Liquid fuels. Liquid fuels of petroleum origin such as distillate oils or residual oils are most commonly used for gas turbine plant. The essential qualities of these fuels include *proper volatility, viscosity and calorific value.* At the same time it *should be free from any contents of moisture* and *suspended impurities that would clog the small passages of the nozzles and damage valves and plungers of the fuel pumps.*

Minerals like *sodium, vanadium* and *calcium* prove *very harmful* for the *turbine blading* as these build deposits or corrode the blades. The sodium in ash should be less than 30% of the vanadium content as otherwise the ratio tends to be critical. The actual sodium content may be between 5 ppm to 10 ppm (part per million). If the vanadium is over 2 ppm, the magnesium in ash tends to become critical. *It is necessary that the magnesium in ash is least three times the quantity of vanadium.* The content of calcium and lead should not be over 10 ppm and 5 ppm respectively.

Sodium is removed from residual oils by mixing with 5% of water and then double centrifuging when sodium leaves with water. Magnesium is added to the washed oil in the form of epsom salts, before the oil is sent into the combustor. This checks the corrosive action of vanadium. Residual oils burn with less ease than distillate oils and the latter are often used to start the unit from cold, after which the residual oils are fed in the combustor. In cold conditions residual oils need to be preheated.

3. Solid fuels. The use of solid fuels such as coal in pulverised form in gas turbines present several difficulties most of which have been only partially overcome yet. The pulverising plant for coal in gas turbine applications is much lighter and smaller than its counterpart in steam generators. *Introduction of fuel in the combustion chamber of a gas turbine is required to be done against a high pressure whereas the pressure in the furnace of a steam plant is atmospheric.* Furthermore, the *degree of completeness of combustion in gas turbine applications has to be very high as otherwise soot and dust in gas would deposit on the turbine blading.*

Some practical applications of solid fuel burning in turbine combustors have been commercially made available in recent years. In one such design finely crushed coal is used instead of pulverised fuel. This fuel is carried in steam of air tangentially into one end of a cylindrical furnace while gas comes out at the centre of opposite end. As the fuel particles roll around the circumference of the furnace they are burnt and a high temperature of about 1650°C is maintained which causes the mineral matter of fuel to be converted into a liquid slag. The slag covers the walls of the furnace and runs out through a top hole in the bottom. The result is that fly ash is reduced to a very small content in the gases. In *another design* a regenerator is used to transfer the heat to air, the combustion chamber being located on the outlet of the turbine, and the combustion is carried out in the turbine exhaust stream. The advantage is that only clean air is handled by the turbine.

5.14. EFFECT OF OPERATING VARIABLES ON THERMAL EFFICIENCY

The thermal efficiency of *actual open cycle* depends on the following *thermodynamic variables* :

(*i*) Pressure ratio
(*ii*) Turbine inlet temperature (T_3)
(*iii*) Compressor inlet temperature (T_1)
(*iv*) Efficiency of the turbine ($\eta_{turbine}$)
(*v*) Efficiency of the compressor (η_{comp}).

1. Effect of turbine inlet temperature and pressure ratio :

If the permissible turbine inlet-temperature (with the other variables being constant) of an *open cycle gas turbine power plant* is increased its *thermal efficiency* is *amply improved*. A practical limitation to increasing the turbine inlet temperature, however, is the ability of the material available for the turbine blading to *withstand the high rotative and thermal stresses.*

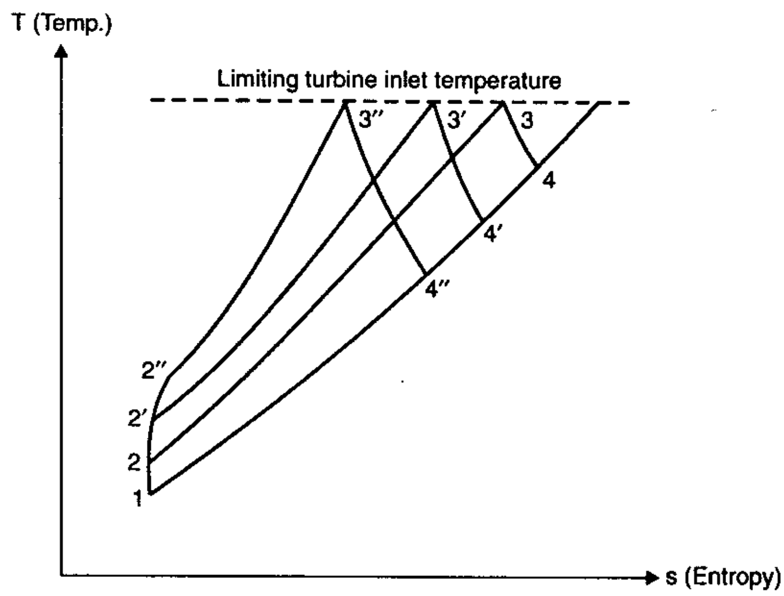


Fig. 5.18

— Refer Fig. 5.18. For *a given turbine inlet temperature, as the pressure rate increases, the heat supplied as well as the heat rejected are reduced. But the rate of change of heat supplied is not the same as the rate of change heat rejected.* As a consequence, *there exists an optimum pressure ratio producing maximum thermal efficiency for a given turbine inlet temperature.*

— As the pressure ratio increases, the thermal efficiency also increases until it becomes maximum and then it drops off with a further increase in pressure ratio (Fig. 5.19). Further, as the turbine inlet temperature increases, the peaks of the curves flatten out giving a greater range of ratios of pressure optimum efficiency.

Following particulars are worth noting :

Gas temperatures	*Efficiency (gas turbine)*
550 to 600°C	20 to 22%
900 to 1000°C	32 to 35%
Above 1300°C	more than 50%

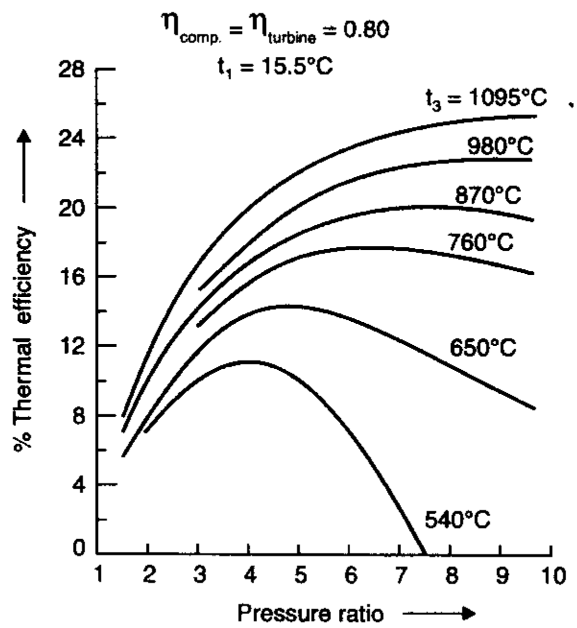


Fig. 5.19. Effect of pressure ratio and turbine inlet temperature.

2. Effect of turbine and compressor efficiencies :

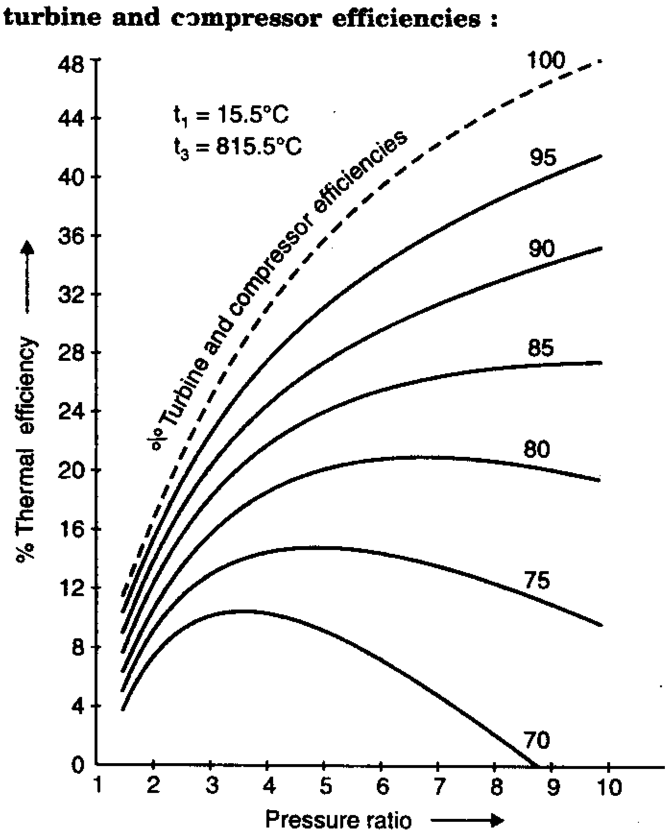


Fig. 5.20. Effect of component efficiency.

Refer Fig. 5.20. The thermal efficiency of the actual gas turbine cycle is very sensitive to variations in the efficiencies of the compressor and turbine. There is a particular pressure ratio at which maximum efficiencies occur. For lower efficiencies, the peak of thermal efficiency occur at lower pressure ratios and vice versa.

3. Effect of compressor inlet temperature :

Refer Fig. 5.21. *With the decrease in the compressor inlet temperature there is increase in the thermal efficiency of the plant.* Also, the peaks of thermal efficiency occurs at high pressure ratios and the curves become flatter giving thermal efficiency over a wider pressure ratio range.

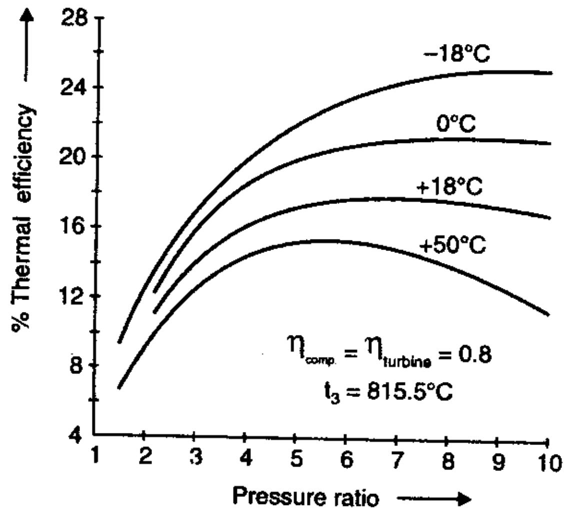


Fig. 5.21. Effect of compressor inlet temperature.

5.15. COMBINATION GAS TURBINE CYCLES

5.15.1. Combined Gas Turbine and Steam Power Plants

The characteristics of the gas turbine plants render these plants very well suited for use in combination with steam or hydro plants. These plants can be quickly started for emergency or peak load service. The combination *'gas-turbine-steam cycles' aim at utilising the heat of exhaust gases from the gas turbine and thus, improve the overall plant efficiency.*

Three popular designs of combination cycle comprise of :

1. Heating feed water with exhaust gases.
2. Employing the gases from a supercharged boiler to expand in the gas turbine.
3. Employing the gases as combustion air in the steam boiler.

1. Heating feed water with exhaust gases :

Refer Fig. 5.22. By employing exhaust gases to heat the feed water it is possible to utilise the entire steam supply to the turbines to expand through entire range of expansion, and thus result in increasing the output of work, since bleeding would not be required.

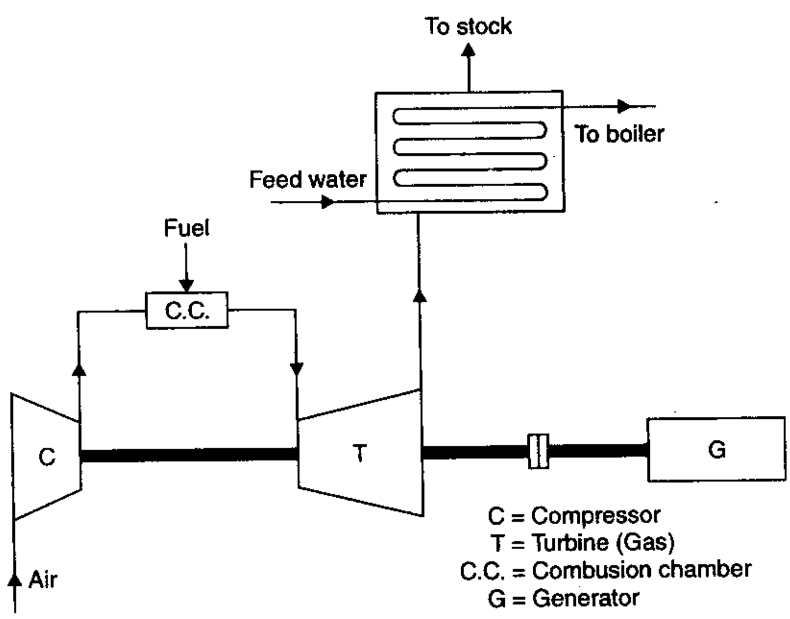


Fig. 5.22. Use of exhaust gases to heat feed water of steam cycle.

2. Employing the gases from a supercharged boiler to expand in the gas turbine :

Refer Fig. 5.23. The supercharged boiler is good application of the combined gas turbine-steam cycle. In this arrangement of the boiler furnace works under a pressure of about 5 bar and the gases are expanded in the gas turbine, its exhaust being used to heat feed water before being

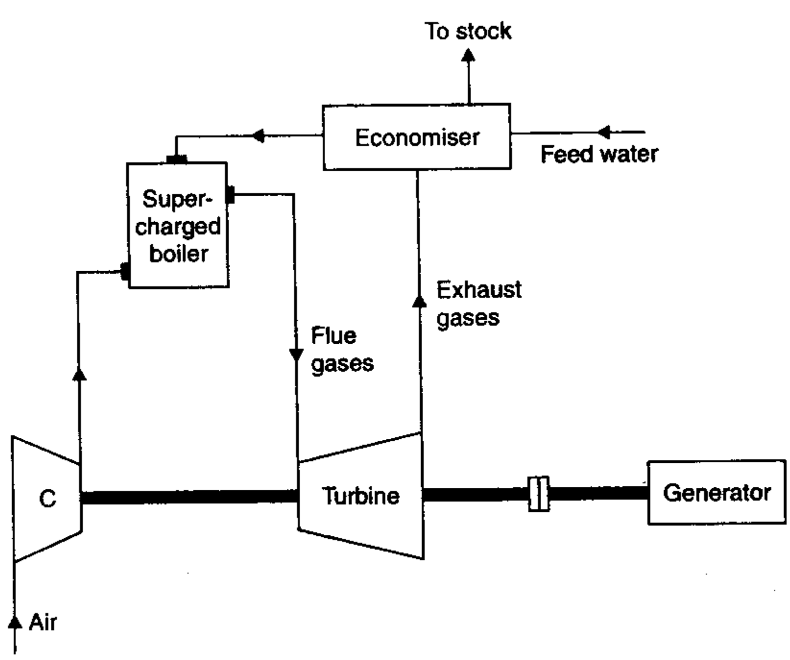


Fig. 5.23. Arrangement of supercharged boiler.

discharged through the stack. The *heat transfer rate in this boiler is very high as compared to that in a conventional boiler due to higher pressure of gases ; and a smaller size of steam generator is needed* for the same steam raising capacity as of the conventional plant. Further more, since the gases in the furnace are already under pressure, *no induced draught or forced draught fans are needed* and there is saving in power consumption which would otherwise be spent in mechanical draught supply. Through this combination an *overall improvement in heat rate is to the extent of above 7 percent.*

3. Employing the gases as combustion air in the steam boiler :

Refer Fig. 5.24. When exhaust gases are used as preheated air for combustion in the boiler, an improvement of about 5 percent in overall heat rate of the plant results. The boiler is fed with supplementary fuel and air, and is made larger than the conventional furnace. If only the turbine exhaust is used in the furnace without any supplementary fuel firing, the arrangement becomes a waste heat boiler.

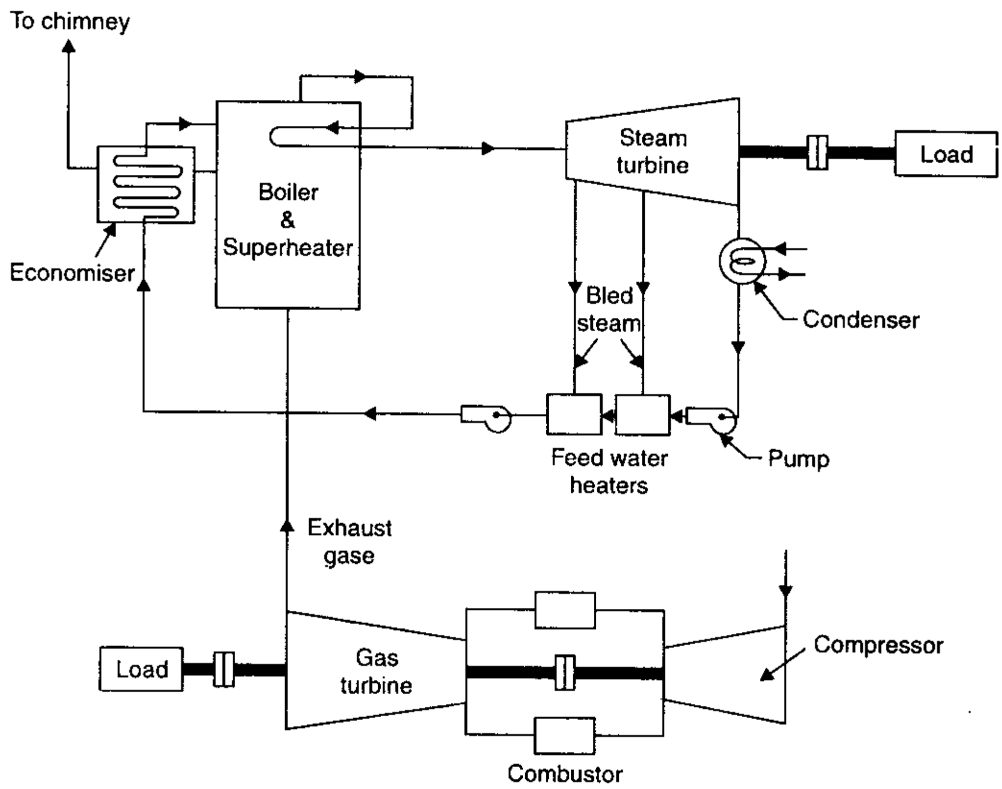


Fig. 5.24. Use of exhaust gases for combustion in the furnace of the steam plant.

— Fig. 5.25 shows the gain in heat rate due to combination cycle.

— Fig. 5.26 shows the comparison of a steam and closed cycle gas plant.

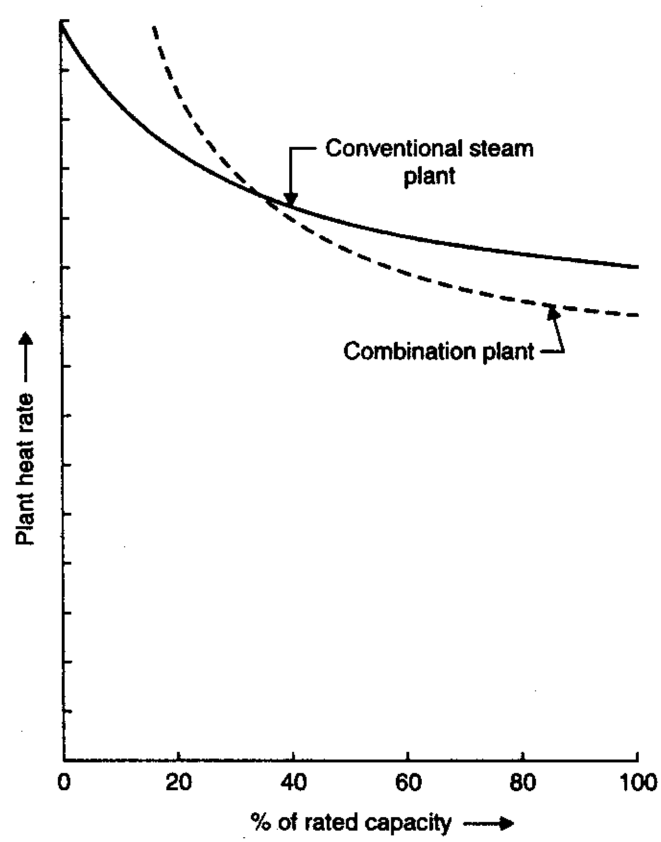


Fig. 5.25

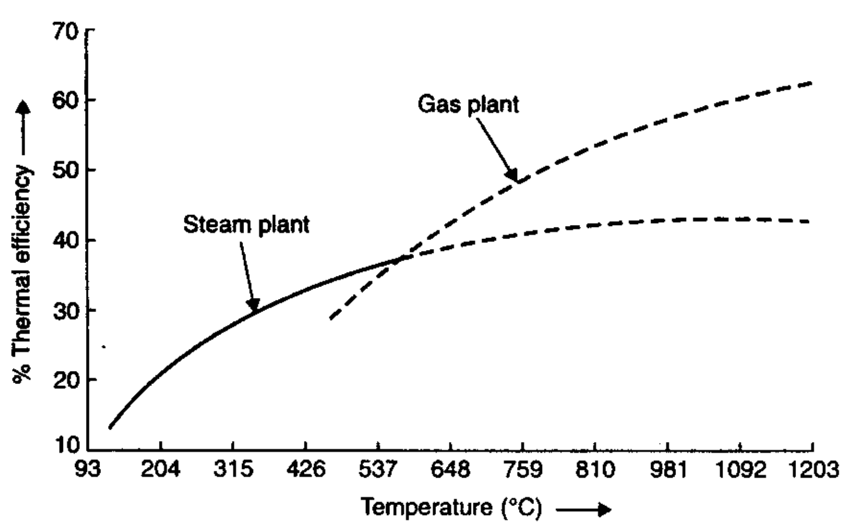


Fig. 5.26. Comparison between steam and closed cycle gas turbine plant.

5.15.2. Combined Gas Turbine and Diesel Power Plants

The performance of a diesel engine can be improved by combining it with an exhaust driven gas turbine. It can be achieved by the following *three* combinations :

1. Turbo-charging
2. Gas-generator
3. Compound engine.

1. Turbo-charging :

Refer Fig. 5.27. This method is known as *supercharging*. Here the exhaust of the diesel engine is expanded in the gas turbine and the work output of the gas turbine is utilised to run a compressor which supplies the pressurised air to the diesel engine to increase its output. The load is coupled to the diesel engine shaft and the output of the gas turbine is just sufficient to run the compressor.

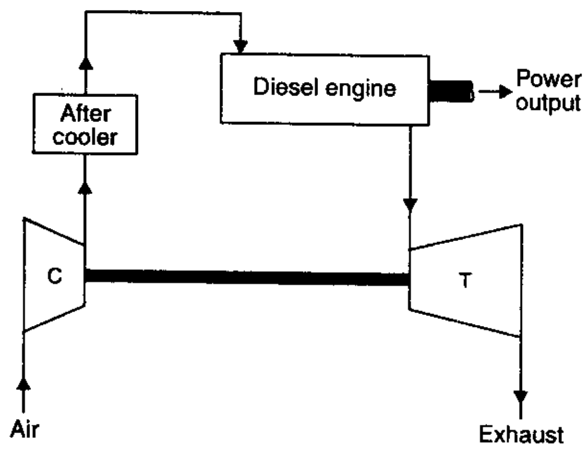


Fig. 5.27. Turbo-charging.

2. Gas-generator :

Fig. 5.28 shows the schematic arrangement. Here the compressor which supplies the compressed air to the diesel engine is not driven from gas turbine but from the diesel engine through some suitable drive. The output of the diesel engine is consumed in driving the air compressor and the gas turbine supplies the power.

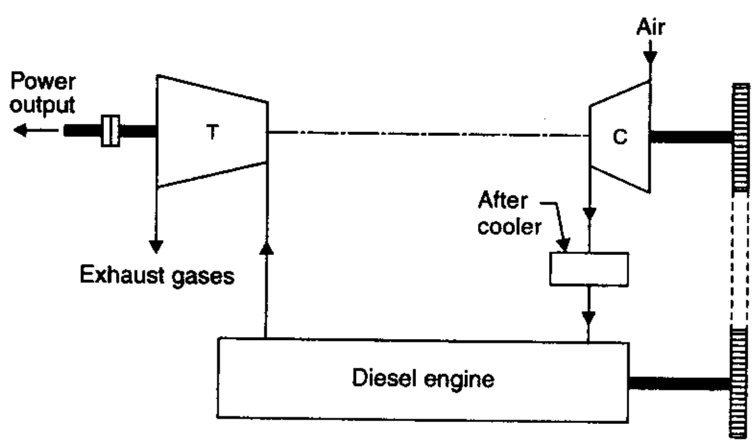


Fig. 5.28. Gas-generator.

3. Compound engine :

Refer Fig. 5.29. In this arrangement the air compressor is driven from both diesel engine and gas turbine through a suitable gearing and the power output is taken from the diesel engine shaft.

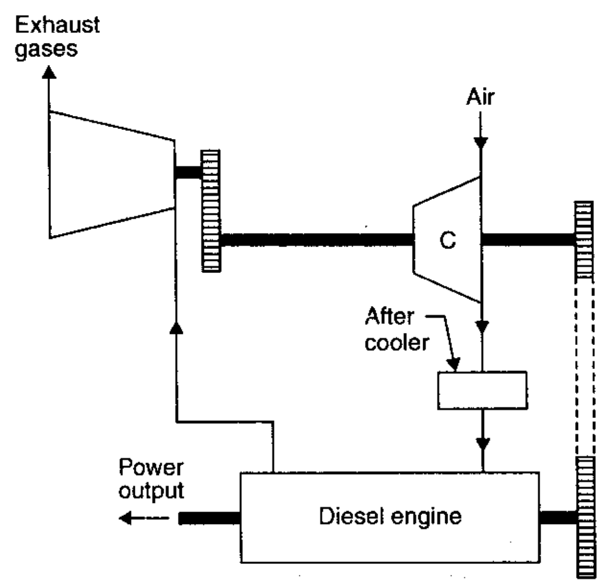


Fig. 5.29. Compound engine.

5.15.3. Advantages of Combined Cycle

1. Improved efficiency.
2. More suitable for rapid start and shut down.
3. Less cooling water requirement.
4. It gives high ratio of power output to occupied ground space.
5. The combined system offers self-sustaining feature.
6. The capital cost of combined plant with supplementary firing is slightly higher than a simple gas turbine plant and much below those of a classical steam plant of the same power capacity.

5.16. OPERATION OF A GAS TURBINE

The operation of a gas turbine includes the following :

1. Starting
2. Shut down

1. Starting of a gas turbine :

Starting of a gas turbine power plant requires an *auxiliary power source*, till the plants own compressor inducts air and compresses it to a pressure such that expansion from reasonable temperature will develop enough power to sustain operation. The *starter* may be (*i*) an I.C. engine (*ii*) a steam turbine (*iii*) an auxiliary electric motor or (*iv*) another gas turbine. It must be coupled to the turbo-compressor shaft with a disengaging or over-running clutch. A main generator or its direct connected exciter may be pressed into temporary service as a motor.

Starting procedure :

1. Run the unit and induct air.

2. Actuate the combustion ignition system and inject the fuel. The fuel flow is controlled to obtain the necessary warm up.

3. Adjust the speed and voltage and synchronise the alternator.

4. Build up the load on the alternator by governor gear control.

2. Shut down of a gas turbine :

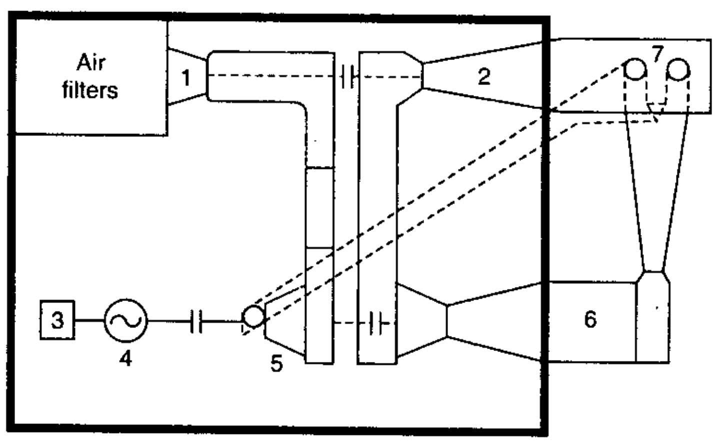
Shut down of a gas turbine occurs very quickly after the fuel is cut off from the combustor. A rapid shut down is desirable since it minimizes the *chilling effect* resulting from passing cold air through the hot turbine. Turning gears are usually employed, or alternatively the starting device may be operated at reduced speed to ensure symmetrical cooling of rotor. This avoids thermal '*kinking*' of the shaft.

5.17. GAS TURBINE POWER PLANT LAYOUT

The layout of a gas turbine plant exercises a very important effect on the overall performance of the plant, as it is possible to incur a loss to the extent of 20% of the power developed in the interconnecting ducts having a considerable number of sharp bends. Therefore ample care should be taken in the design and layout of the air as well as gas circuits. Fig. 5.30 shows a schematic simplified diagram of a gas turbine power plant layout. The following points are worth noting :

— In this case the main building is the turbine house in which major portion of the plant as well as auxiliaries are installed. Usually it is similar to the steam plant house in many respects.

— The storage tanks containing fuel oil are arranged outside but adjoining the turbine house. In some installations even heat exchangers are also out of the doors.



1 = L.P. Compressor 2 = L.P. Turbine
3 = Starting motor 4 = Alternator
5 = H.P. Compressor 6 = Combustion chamber
7 = Heat exchangers

Fig. 5.30. Gas turbine power plant layout.

— Whereas the major portion of the total space is occupied by the intercoolers, combustion chambers, heat exchangers, waste heat boilers and interconnecting duct work, the rotating parts of the plant form a very small part of the total volume of the plant.

5.18. COMPONENTS OF A GAS TURBINE POWER PLANT

The main components of a gas turbine power plant are enumerated and discussed as follows :

1. Gas turbines
2. Compressors
3. Combustor
4. Intercoolers and regenerators.

1. Gas turbines :

A turbine basically employs vanes or blades mounted on a shaft and enclosed in a casing. The flow of fluid through the turbine in *most design is axial and tangential to the rotor* at a nearly constant or increasing radius. The basic requirements of the turbines are : (*i*) *Light weight* (*ii*) *High efficiency* (*iii*) *Reliability in operation and* (*iv*) *Long working life.* Large work output can be obtained per stage with high blade speeds when the blades are designed to sustain higher stresses. *More stages of the turbine are always preferred in gas turbine power plant because it helps to reduce the stresses in the blades and increases the overall life of the turbine.*

It is essential to cool the gas turbine blades for long life as these are continuously subjected to high temperature gases. The blades can be cooled by different methods, the common method being the *air-cooling*. The air is passed through the holes provided through the blade.

The following **accessories** are fitted to the turbine :

(*i*) *Tachometer*. It shows the speed of the machine and also actuates the fuel regulator in case the speed shoots above or falls below the regulated speed, so that the fuel regulator admits less or more fuel into the combustor and varies the turbine power according to the demand. The tachometer is driven through a gear box.

(*ii*) *An overspeed governor.* The governor backs off fuel feed if exhaust temperature from the turbine exceeds the safe limit, thermal switches at the turbine exhaust acting on fuel control to maintain present maximum temperature.

(*iii*) *Lubricating oil pump.* It supplies oil to the bearings under pressure.

(*iv*) Starting motor or engine

(*v*) Starting set-up gear

(*vi*) Oil coolers

(*vii*) Filters

(*viii*) Inlet and exhaust mufflers.

2. Compressors :

The compressors which are commonly used are of the following two types :

1. Centrifugal type
2. Axial flow type

— The *'centrifugal compressor'* consists of an *impeller* and a *diffuser*. The *impeller imparts the high kinetic energy to the air* and *diffuser converts the kinetic energy into the pressure energy*. The pressure ratio of 2 to 3 is possible with single stage compressor and it can be increased upto 20 with 3-stage compressor. The compressors may have single or double inlet. The *single inlet* compressors are designed to handle the air in the range of 15 to 300 m^3/min. and *double inlets* are preferred above 300 m^3/min capacity. Fig. 5.31 shows a single stage centrifugal compressor.

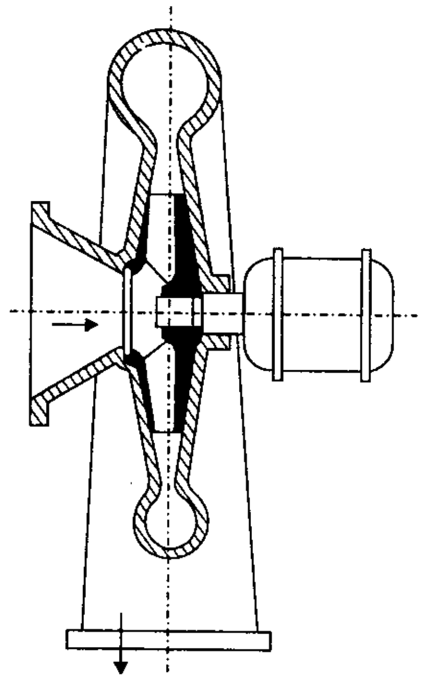


Fig. 5.31. Single stage centrifugal compressor.

In the centrifugal compressor :

(*i*) *Capacity* varies *directly as the speed ratio.*

(*ii*) *Total pressure* varies as the *square of speed ratio.*

(*iii*) *Power input* varies as the *cube of the speed ratio.*

(*iv*) Pressure is developed *independent of load,* but the *volume handled depends on the load.*

In a *multistage centrifugal compressor,* two or more impellers operating in series on a single shaft are provided in a single casing. The effect of multi-staging is to increase the delivery pressure of air as air compressed in one stage of machine is fed into the next stage for further compression and pressure is multiplied in each stage. The *efficiency of multistage compressor is lower than a single stage due to the losses.*

The centrifugal compressor is *superior to the axial flow machine in that a high pressure ratio can be obtained in a short rugged single stage machine, though at the cost of lower efficiency and increased frontal area.*

— The *'axial flow compressors'* are commonly *used in gas turbine installations.* An axial flow compressor consists of a series of rotor and stator stages with decreasing diameters along the flow of air. The blades are fixed on the rotors and rotors are fixed on the shaft. The stator blades are fixed on the stator casing. The stator blades guide the air flow to the next rotor stage coming from the previous rotor stage. The air flows along the axis of the rotor. The kinetic energy is given to the air as it passes through the rotor and part of it is converted into pressure. A multistage single flow axial compressor is shown in Fig. 5.32.

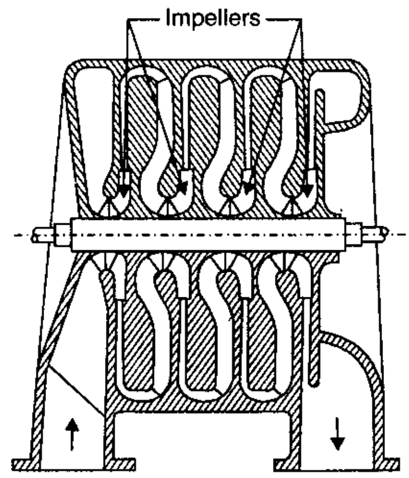


Fig. 5.32. Multistage single flow axial compressor.

An axial compressor is capable of *delivering constant volumes of air over varying discharge pressures*. These machines are well suited for large capacities at moderate pressures. If the impeller of a centrifugal compressor is designed to give an axial component of velocity at the exit, the design becomes a *mixed flow type*.

3. Combustor :

The primary function of the *combustor is to provides for the chemical reaction of the fuel and air being supplied by the compressor*. It must fulfil the following *conditions* :

(*i*) Combustion must take place at high efficiency because of the effect of the combustion efficiency on the thermal efficiency of the gas turbine cycle.

(*ii*) The pressure losses must be low.

(*iii*) Ignition must be reliable and accomplished with ease over a wide range of atmospheric condition especially in aircraft installation.

(*iv*) Through mixing of fuel and air.

(*v*) Carbon deposits must not be formed under any conditions.

The physical process of combustion may be divided into four important steps :

1. Formation of reactive mixture
2. Ignition
3. Flame propagation
4. Cooling of combustion products with air. Atomisation should be done for perfect burning.

Fig. 5.33 shows an arrangement of a typical combustor design which employs an *outer cylindrical shell* with a *conical inner sleeve* which is provided with ports or slots along the length. At the cone apex is fitted a *nozzle* through which fuel is sprayed in a conical pattern into the sleeve and near this is an *igniting device* or spark plug. A fuel line conveys the fuel to the nozzle. A few air ports provided close to the situation of the nozzle supply the combustion air directly to the fuel and are fitted with vanes to produce a whirling motion of oil and thereby create turbulence. The rest of the air admitted ahead of combustion zone serves to cool the combustor and outlet gases. The combustor is best located between the compressor outlet and turbine inlet and takes the shape of a cylinder. Alternatively, the 'can' arrangement may be used in which the flow is divided to pass through a number of smaller cylindrical chambers. In this latter design the adjacent chambers may be interconnected through small tubes so that a simple igniting device fitted in one of the chambers serves all the chambers.

The *nozzle* sprays the fuel under pressure in an atomised conical spray. The fuel is delivered to the nozzle through the fuel line and flows out through tangential slots in the nozzle, thus being given a whirling motion in an annular chamber from where it passes out through a small orifice in the conical pattern of desired angle.

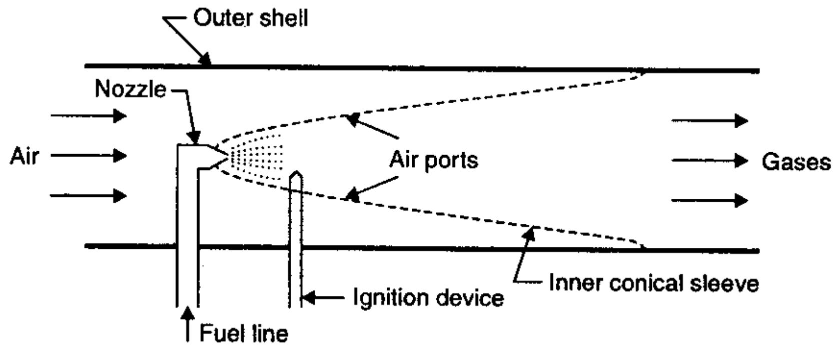


Fig. 5.33. Arrangement of a combustor.

4. Intercoolers and regenerators :

Intercoolers. In a gas turbine plant the *intercooler* is generally used when the pressure ratio used is sufficiently large and the compression is completed with two or more stages. The cooling of compressed air is generally done with the use of cooling water. A *cross-flow* type intercooler (Fig. 5.34) is generally preferred for effective heat transfer.

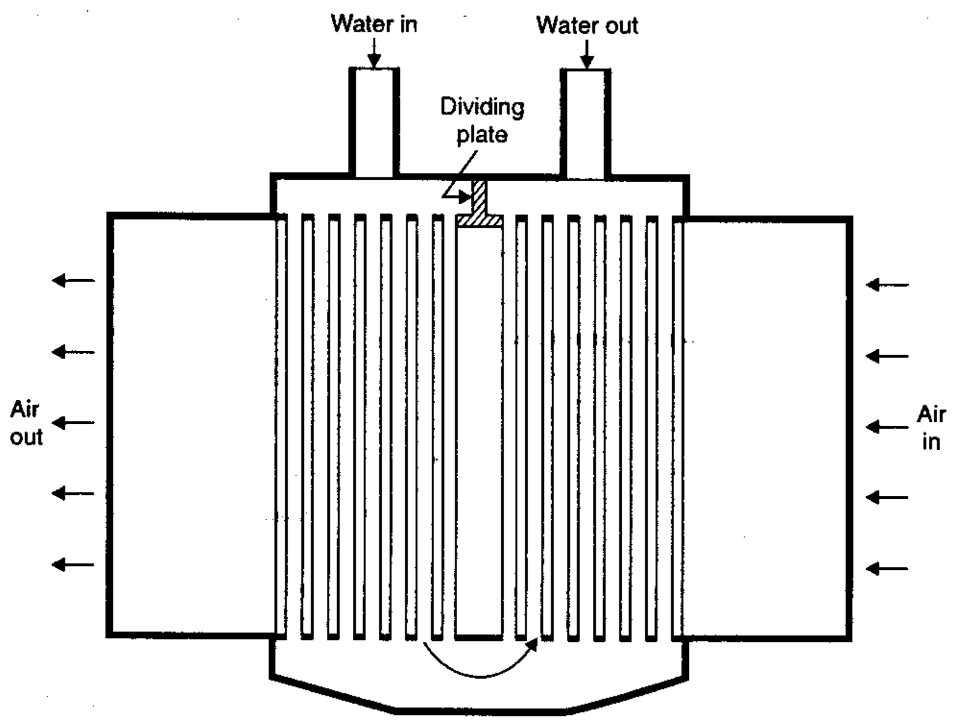


Fig. 5.34. Intercooler.

Regenerators. Refer Fig. 5.35. In the regenerator heat transfer takes place between the exhaust gases and cool air. It is usually made in shell and tube construction with gas flowing inside the tubes and air outside the tubes, the two fluids being made to flow in opposite directions. Since the gas is bound to carry dust and deposit the same on the heat transfer surface, the internal flow through the tubes is convenient as the tube inside can be easily cleaned with brushes whereas it is very difficult to clean the outside surface of tubes. The effect of counterflow is the highest average temperature difference between the heating and heated medium with consequent high heat transfer between the two fluids. A number of baffles in the air put in the shell make the air to flow in contact with maximum heat transfer. However, *the pressure drop in both air and gas during the flow should be minimum possible.*

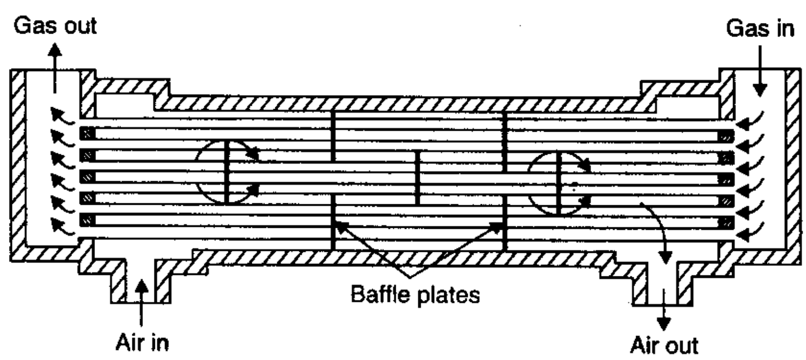


Fig. 5.35. Regenerator.

5.19. VARIOUS ARRANGEMENTS OF GAS TURBINE POWER PLANTS

The various arrangements of gas turbine plants are shown in Figs. 5.36 to 5.41.

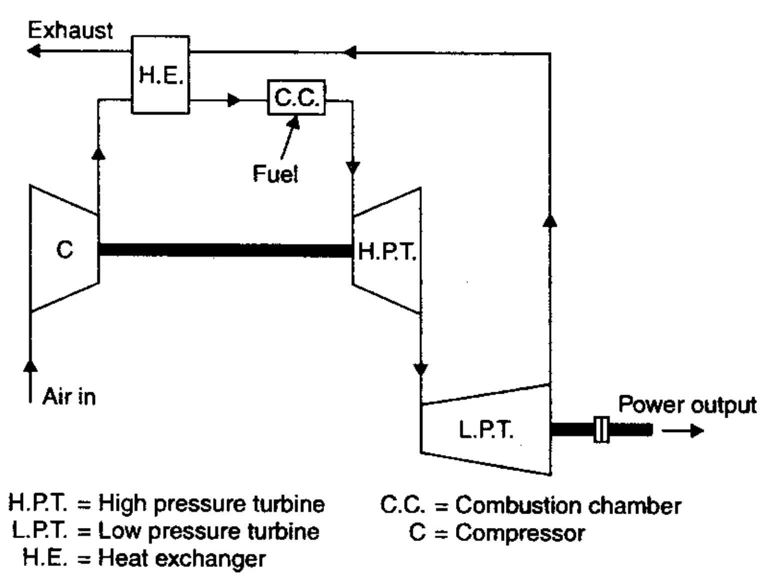


Fig. 5.36. Open cycle gas turbine with separate power turbine.

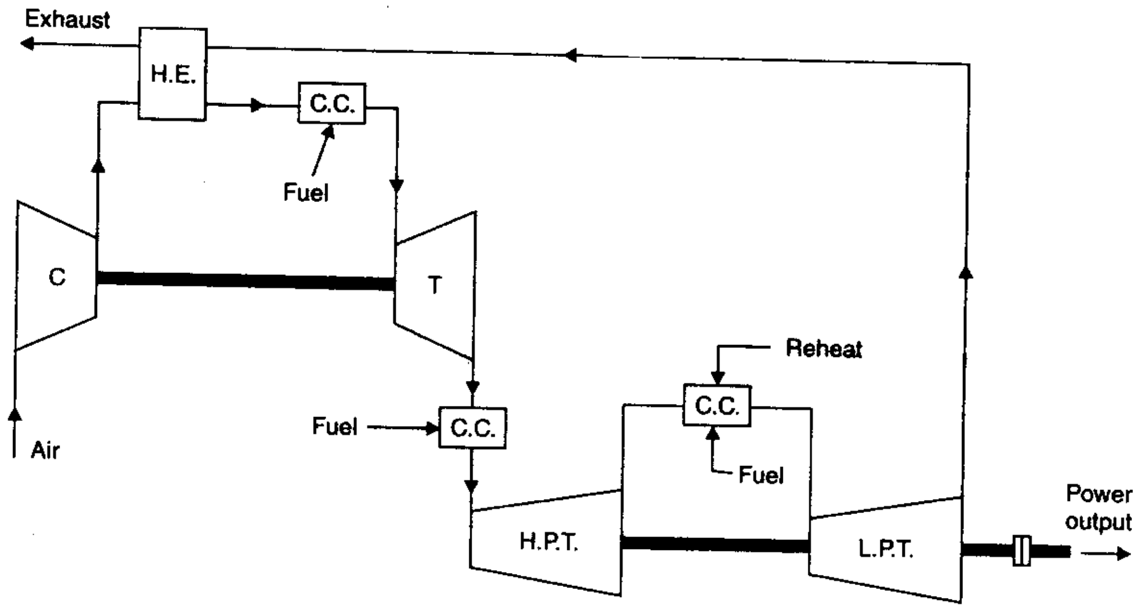
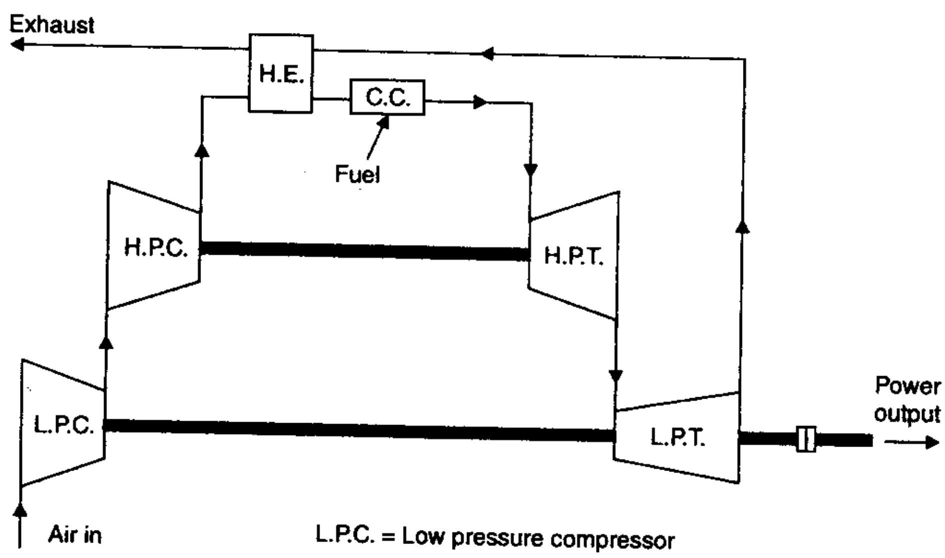


Fig. 5.37. Series flow gas turbine plant.



L.P.C. = Low pressure compressor
H.P.C. = High pressure compressor

Fig. 5.38. Parallel flow gas turbine plant.

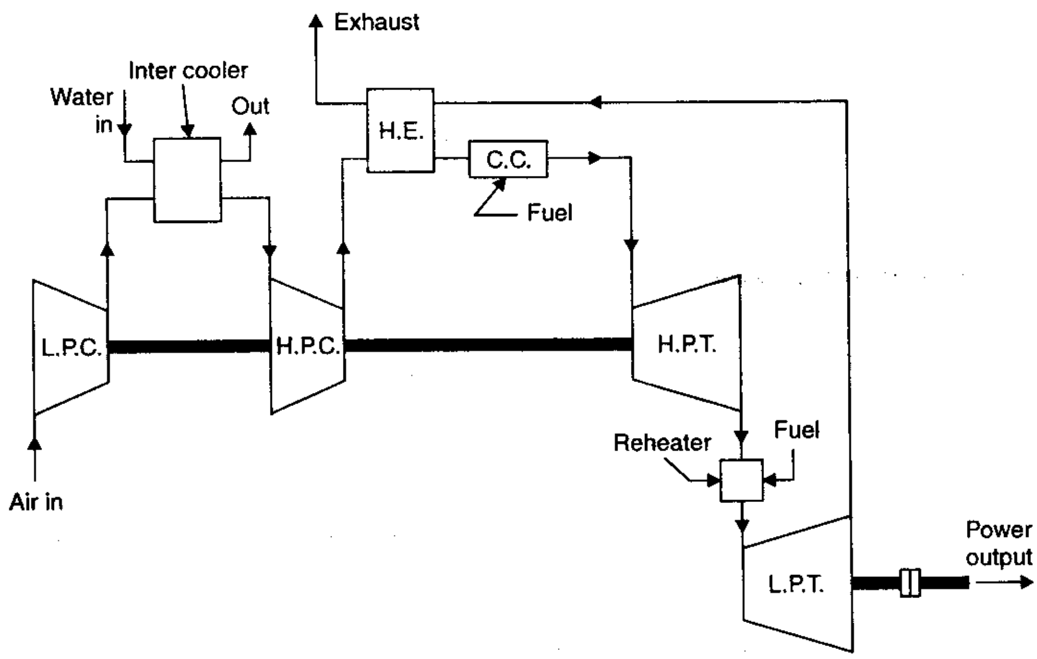


Fig. 5.39. Series flow plant with intercooled compression.

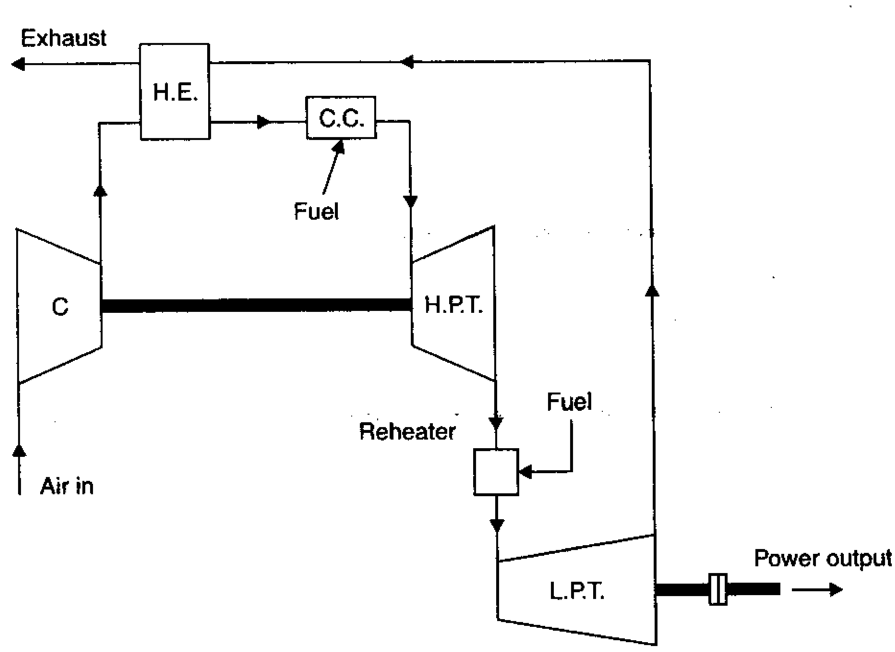


Fig. 5.40. Series flow plant with reheat between power turbine stages.

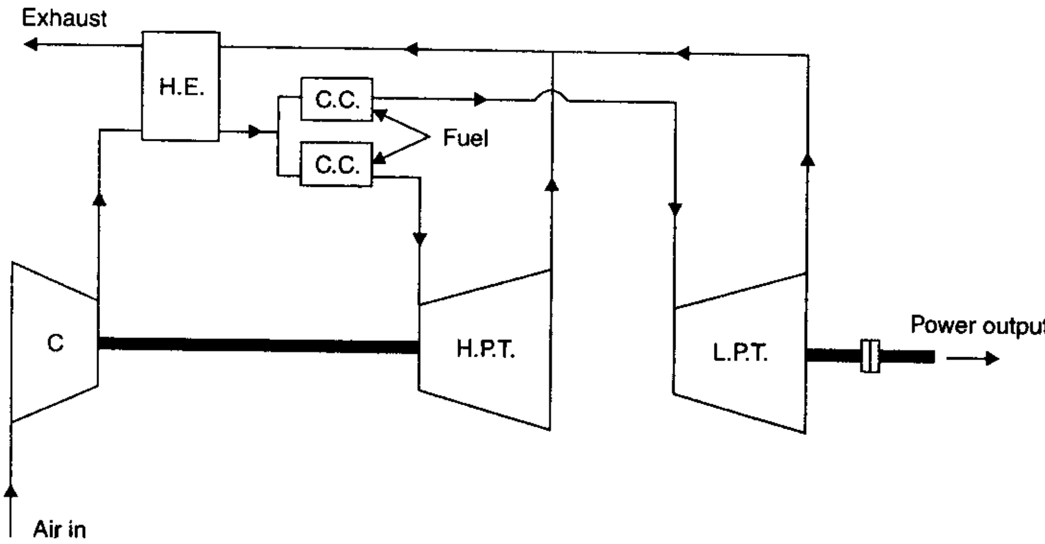


Fig. 5.41. Straight compound gas turbine plant.

5.20. EFFECT OF THERMODYNAMIC VARIABLES ON AIR RATE

The gas turbine plant size depends upon air rate. *The lower the air rate, the smaller the plant size*. The effect of the various thermodynamic variables on air rate is given in the Figs. 5.42 to 5.45.

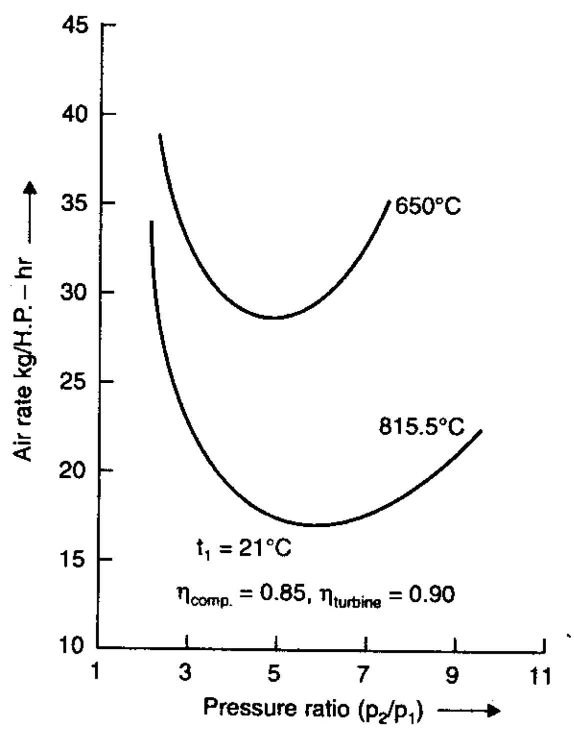


Fig. 5.42. Effect of turbine inlet temperature and pressure ratio on air rate.

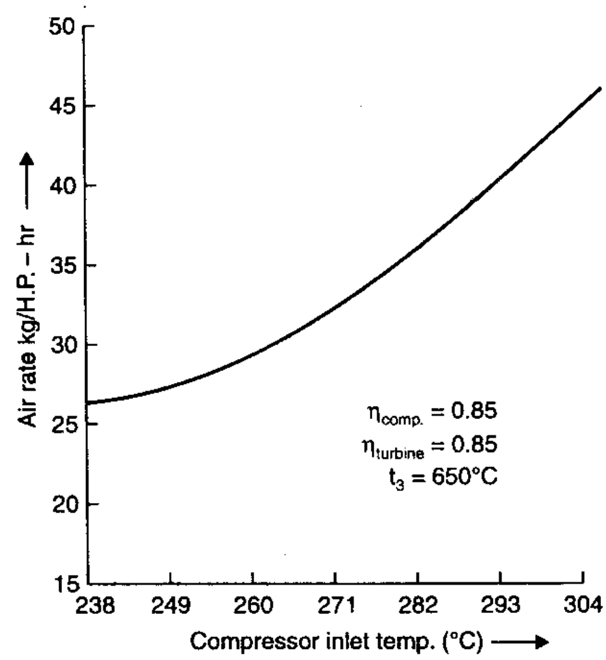


Fig. 5.43. Effect of the compressor inlet temperature on air rate.

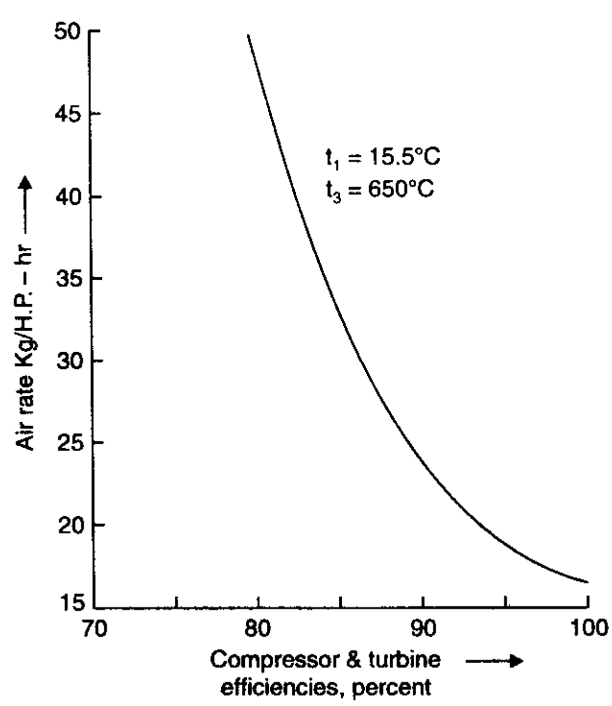


Fig. 5.44. Effect of compressor and turbine efficiencies on air rate.

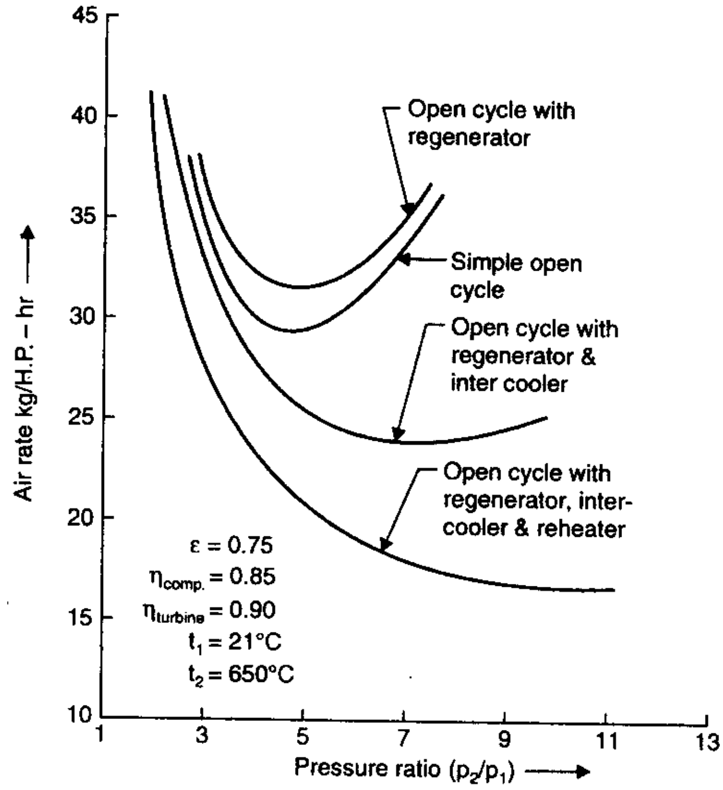


Fig. 5.45. Effect of thermal refinements on air rate.

5.21. FREE-PISTON ENGINE PLANT

Free-piston engine plants are the conventional gas turbine plants *with the difference that the air compressor and combustion chamber are replaced by a free piston engine.* The working of such a plant is explained in Fig. 5.46.

Refer Fig. 5.46. A free-piston engine unit comprises of five cylinders with two assemblies of pistons that move opposite to each other. The pistons are powered by a diesel cylinder located in the centre. Each assembly has a diesel piston at the centre of the unit. These are rigidly connected to large-diameter pistons reciprocating in the large air-compressor cylinders. Each assembly has air-cushion cylinders at the end.

Refer Fig. 5.46 (*a*) :

— The pistons are at their innermost position.

— The diesel cylinder has charge of compressed air at about 100 bar ready for firing.

— The air-compression cylinders are filled with air at atmospheric pressure ; the inlet valves IV_1 and IV_2 have just closed. The air trapped in the *air-cushion cylinders,* also called *bounce cylinders,* is at its lowest pressure.

— The discharge values DV_1 and DV_2 in the *air compression cylinders* are held shut by the high-pressure air in the compressed-air receiver that connects the air compression cylinders and the diesel cylinders.

— The compressed air-charge in the diesel cylinder is at high temperature from work of compression, so that a charge of fuel injected immediately ignites and burns. The resulting sudden rise of pressure in the diesel cylinder forces the pistons apart. As they move apart, the exhaust port *leading to the turbine* uncovers first. The combustion products at a pressure of 3 to 6 bar and temperature of about 550°C rush out *through the exhaust to the turbine.* As the pistons continue to move to their outermost positions, the inlet port from the compressed-air receiver is uncovered and compressed air from the receiver enters the diesel cylinder to scavenge out the combustion products and fill the cylinder with a fresh charge of air [Fig. 5.46 (*b*)]. During this outward stroke, the air in the cushion cylinders has been compressed. This air now expands and pushes on the end pistons to return the assemblies back to the innermost position, in turn compressing the fresh air charge trapped in the diesel cylinder by the opposed piston closing on each other.

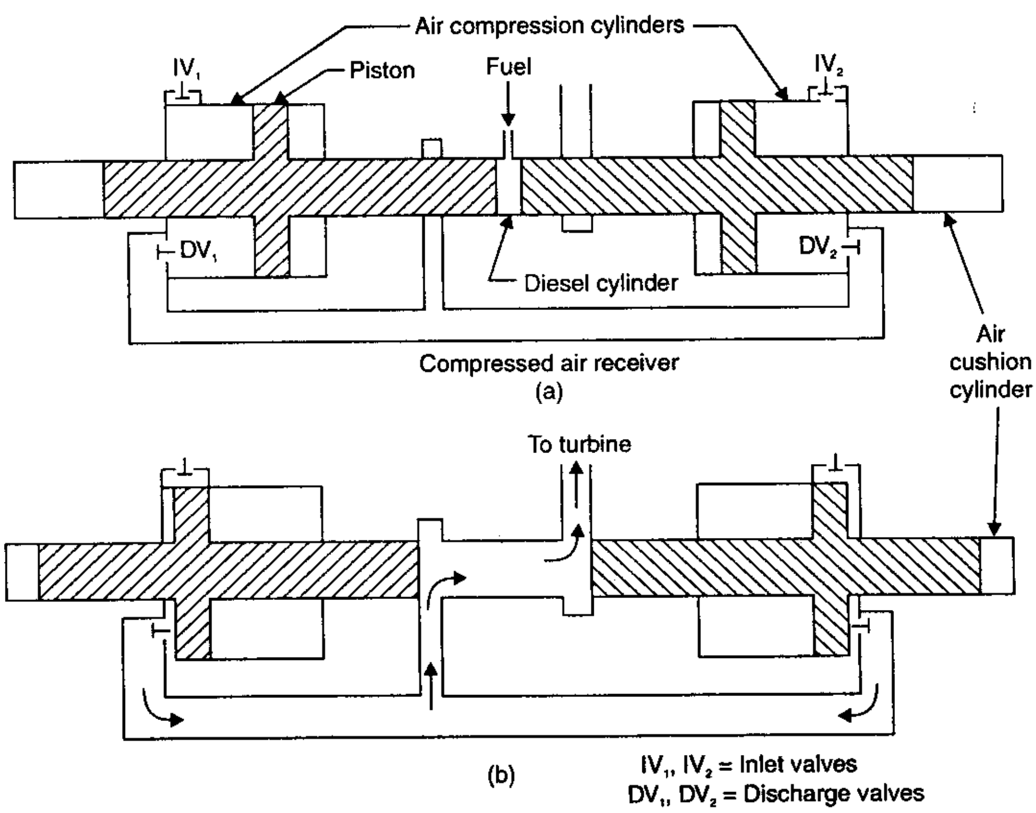


Fig. 5.46. Free-piston engine plant.

— *Thermal efficiencies of 35%* and more can be developed by these plants. This can be achieved *because the 1900°C combustion gas in the diesel cylinder does work directly on the pistons in compressing air. The higher the temperature of gases doing work, the more efficient the process.*

Note. The temperature to the tune of 1900°C can be tolerated because *it appears only instantaneously at ignition and cylically.* During most of the cycle the cylinder is being cooled by relatively cooler air charges and expanding combustion products. In addition, the diesel cylinder walls are cooled by a water jacket. The materials problem of the gas turbine is alleviated considerably by being required to handle temperatures of 550°C.

Advantages of Free-piston Engine Arrangement :

1. Less air rate as compared to a conventional gas turbine.
2. It is possible to achieve efficiency more than 40 percent.

3. Lighter and smaller than a diesel engine of the same capacity.
4. The gas turbine is about one third the size of the turbine for a simple open gas turbine plant.
5. The free-piston is vibrationless.

Disadvantages :

1. Starting and control problems.
2. Synchronization problem not yet fully overcome.

5.22. RELATIVE THERMAL EFFICIENCIES OF DIFFERENT CYCLES

Fig. 5.47 shows the graphs between turbine inlet temperatures and thermal efficiency for different cycles and which are self explanatory.

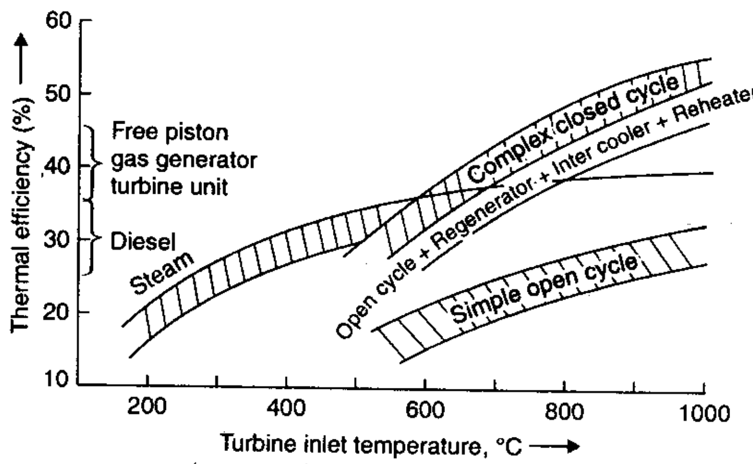


Fig. 5.47. Relative thermal efficiencies of different cycles.

WORKED EXAMPLES

Example 5.1. *The air enters the compressor of an open cycle constant pressure gas turbine at a pressure of 1 bar and temperature of 20°C. The pressure of the air after compression is 4 bar. The isentropic efficiencies of compressor and turbine are 80% and 85% respectively. The air-fuel ratio used is 90 : 1. If flow rate of air is 3.0 kg/s, find :*

(*i*) *Power developed.*

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

Assume $c_p = 1.0$ *kJ/kg K and* $\gamma = 1.4$ *for air and gases.*

Calorific value of fuel = 41800 kJ/kg.

Solution. *Given :* $p_1 = 1$ bar ; $T_1 = 20 + 273 = 293$ K

$p_2 = 4$ bar ; $\eta_{compressor} = 80\%$; $\eta_{turbine} = 85\%$

Air-fuel ratio $= 90 : 1$; Air flow rate, $m_a = 3.0$ kg/s (*i*) **Power developed, P :**

Refer Fig. 5.48 (*b*)

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 1.486$$

$\therefore$ $$T_2 = (20 + 273) \times 1.486 = 435.4 \text{ K}$$

$$\eta_{\text{compressor}} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$0.8 = \frac{435.4 - 293}{T_2' - 293}$$

$\therefore$ $$T_2' = \frac{435.4 - 293}{0.8} + 293 = 471 \text{ K}$$

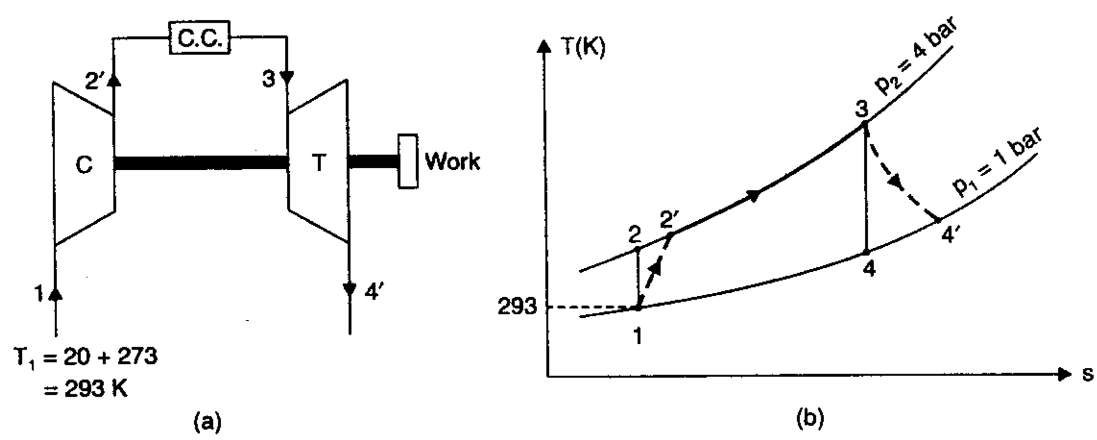


Fig. 5.48

Heat supplied by fuel = Heat taken by burning gases

$$m_f \times C = (m_a + m_f)\, c_p (T_3 - T_2')$$

(where m_a = mass of air, m_f = mass of fuel)

$\therefore$ $$C = \left(\frac{m_a}{m_f} + 1\right) c_p\, (T_3 - T_2')$$

$\therefore$ $$41800 = (90 + 1) \times 1.0 \times (T_3 - 471)$$

i.e., $$T_3 = \frac{41800}{91} + 471 = 930 \text{ K}$$

Again, $$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{0.4}{0.4}} = 0.672$$

$\therefore$ $$T_4 = 930 \times 0.672 = 624.9 \text{ K}$$

$$\eta_{\text{turbine}} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.85 = \frac{930 - T_4'}{930 - 624.9}$$

$\therefore$ $T_4' = 930 - 0.85\,(930 - 624.9) = 670.6$ K

$$W_{\text{turbine}} = m_g \times c_p \times (T_3 - T_4')$$

(where m_g is the mass of hot gases formed per kg of air)

$\therefore$
$$W_{\text{turbine}} = \left(\frac{90+1}{90}\right) \times 1.0 \times (930 - 670.6)$$
$$= 262.28 \text{ kJ/kg of air.}$$
$$W_{\text{compressor}} = m_a \times c_p \times (T_2' - T_1) = 1 \times 1.0 \times (471 - 293)$$
$$= 178 \text{ kJ/kg of air}$$
$$W_{\text{net}} = W_{\text{turbine}} - W_{\text{compressor}}$$
$$= 262.28 - 178 = 84.28 \text{ kJ/kg of air.}$$

Hence *power developed*, $P = 84.28 \times 3 =$ **252.84 kW/kg of air.** **(Ans.)**

(*ii*) **Thermal efficiency of cycle, η_{thermal} :**

Heat supplied per kg of air passing through combustion chamber

$$= \frac{1}{90} \times 41800 = 464.44 \text{ kJ/kg of air}$$

$\therefore$
$$\eta_{\text{thermal}} = \frac{\text{Work output}}{\text{Heat supplied}} = \frac{84.28}{464.44} = 0.1814 \text{ or } \mathbf{18.14\%}. \quad \textbf{(Ans.)}$$

Example 5.2. *A gas turbine unit has a pressure ratio of 6 : 1 and maximum cycle temperature of 610°C. The isentropic efficiencies of the compressor and turbine are 0.80 and 0.82 respectively. Calculate the power output in kilowatts of an electric generator geared to the turbine when the air enters the compressor at 15°C at the rate of 16 kg/s.*

Take $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for the compression process, and take $c_p = 1.11$ kJ/kg K and $\gamma = 1.333$ for the expansion process.

Solution. *Given* : $T_1 = 15 + 273 = 288$ K ; $T_3 = 610 + 273 = 883$ K ; $\dfrac{p_2}{p_1} = 6$,

$\eta_{\text{compressor}} = 0.80$; $\eta_{\text{turbine}} = 0.82$; Air flow rate = 16 kg/s

For compression process : $c_p = 1.005$ kJ/kg K, $\gamma = 1.4$

For expansion process : $c_p = 1.11$ kJ/kg K, $\gamma = 1.333$

In order to evaluate the net work output it is necessary to calculate temperatures T_2' and T_4'. To calculate these temperatures we must first calculate T_2 and then use the isentropic efficiency.

For an isentropic process,
$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.67$$

$\therefore$ $T_2 = 288 \times 1.67 = 481$ K

Also,
$$\eta_{\text{compressor}} = \frac{T_2 - T_1}{T_2' - T_1}$$
$$0.8 = \frac{481 - 288}{T_2' - T_1}$$

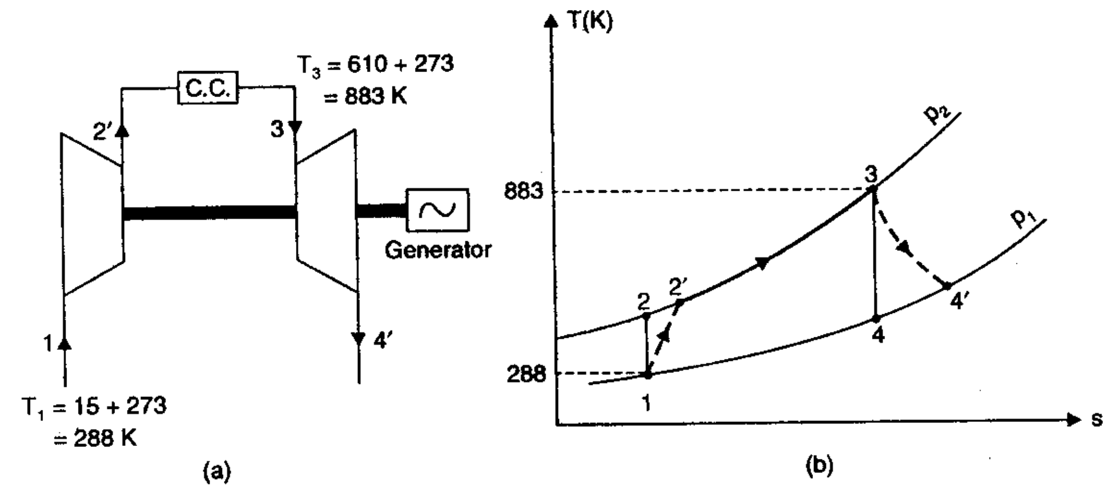


Fig. 5.49

$\therefore$ $$T_2' = \frac{481 - 288}{0.8} + 288 = 529 \text{ K}$$

Similarly for the turbine, $$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.333-1}{1.333}} = 1.565$$

$\therefore$ $$T_4 = \frac{T_3}{1.565} = \frac{883}{1.565} = 564 \text{ K}$$

Also, $$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} = \frac{883 - T_4'}{883 - 564}$$

$\therefore$ $$0.82 = \frac{883 - T_4'}{883 - 564}$$

$\therefore$ $$T_4' = 883 - 0.82\,(883 - 564) = 621.4 \text{ K}$$

Hence,

Compressor work input, $W_{compressor} = c_p\,(T_2' - T_1)$
$= 1.005\,(529 - 288) = 242.2$ kJ/kg

Turbine work output, $W_{turbine} = c_p\,(T_3 - T_4')$
$= 1.11\,(883 - 621.4) = 290.4$ kJ/kg.

$\therefore$ Net work output, $W_{net} = W_{turbine} - W_{compressor}$
$= 290.4 - 242.2 = 48.2$ kJ/kg

Power in kilowatts $= 48.2 \times 16 =$ **771.2 kW. (Ans.)**

Example 5.3. *Calculate the thermal efficiency and work ratio of the plant is example 5.2, assuming that c_p for the combustion process is 1.11 kJ/kg K.*

Solution. Heat supplied $= c_p(T_3 - T_2')$
$= 1.11\,(883 - 529) = 392.9$ kJ/kg

$$\eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{48.2}{392.9} = 0.1226 \text{ or } \mathbf{12.26\%}. \quad \textbf{(Ans.)}$$

Now, $\quad$ *Work ratio* $= \dfrac{\text{Net work output}}{\text{Gross work output}} = \dfrac{48.2}{W_{\text{turbine}}} = \dfrac{48.2}{290.4} = \mathbf{0.166.}$ **(Ans.)**

Example 5.4. *In a constant pressure open cycle gas turbine air enters at 1 bar and 20°C and leaves the compressor at 5 bar. Using the following data ; Temperature of gases entering the turbine = 680°C, pressure loss in the combustion chamber = 0.1 bar, $\eta_{compressor}$ = 85%, $\eta_{turbine}$ = 80%, $\eta_{combustion}$ = 85%, γ = 1.4 and c_p = 1.024 kJ/kg K for air and gas, find :*

(*i*) *The quantity of air circulation if the plant develops 1065 kW.*

(*ii*) *Heat supplied per kg of air circulation.*

(*iii*) *The thermal efficiency of the cycle.*

Mass of the fuel may be neglected.

Solution. *Given* : p_1 = 1 bar, p_2 = 5 bar, p_3 = 5 – 0.1 = 4.9 bar, p_4 = 1 bar,

$$T_1 = 20 + 273 = 293 \text{ K}, T_3 = 680 + 273 = 953 \text{ K},$$

$$\eta_{\text{compressor}} = 85\%, \eta_{\text{turbine}} = 80\%, \eta_{\text{combustion}} = 85\%.$$

For air and gases : c_p = 1.024 kJ/kg K, γ = 1.4

Power developed by the plant, P = 1065 kW.

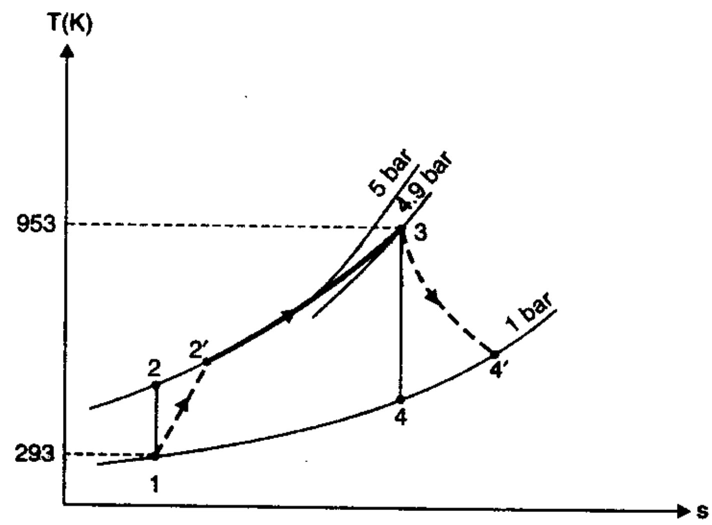


Fig. 5.50

(*i*) **The quantity of air circulation, m_a :**

For *isentropic compression 1-2,*

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 1.584$$

$$\therefore \quad T_2 = 293 \times 1.584 = 464 \text{ K}$$

Now, $\quad \eta_{\text{compressor}} = \dfrac{T_2 - T_1}{T_2' - T_1}$ *i.e.* $\quad 0.85 = \dfrac{464 - 293}{T_2' - 293}$

$$\therefore \quad T_2' = \frac{464 - 293}{0.85} + 293 = 494 \text{ K}$$

For *isentropic expansion process 3-4,*

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4.9}\right)^{\frac{1.4-1}{1.4}} = 0.635$$

$$\therefore \quad T_4 = 953 \times 0.635 = 605 \text{ K}$$

Now,
$$\eta_{\text{turbine}} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.8 = \frac{953 - T_4'}{953 - 605}$$

$$\therefore \quad T_4' = 953 - 0.8(953 - 605) = 674.6 \text{ K}$$

$$W_{\text{compressor}} = c_p (T_2' - T_1) = 1.024 (494 - 293) = 205.8 \text{ kJ/kg}$$

$$W_{\text{turbine}} = c_p (T_3 - T_4') = 1.024 (953 - 674.6) = 285.1 \text{ kJ/kg.}$$

$$\therefore \quad W_{\text{net}} = W_{\text{turbine}} - W_{\text{compressor}} = 285.1 - 205.8 = 79.3 \text{ kJ/kg of air}$$

If the mass of air flowing is m_a kg/s, the power developed by the plant is given by

$$P = m_a \times W_{net} \text{ kW}$$

$$1065 = m_a \times 79.3$$

$$\therefore \quad m_a = \frac{1065}{79.3} = 13.43 \text{ kg.}$$

i.e., **Quantity of air circulation = 13.43 kg. (Ans.)**

(*ii*) **Heat supplied per kg of air circulation :**

Actual heat supplied per kg of air circulation

$$= \frac{c_p (T_3 - T_2')}{\eta_{\text{combustion}}} = \frac{1.024 (953 - 494)}{0.85} = 552.9 \text{ kJ/kg.}$$

(*iii*) **Thermal efficiency of the cycle, η_{thermal} :**

$$\eta_{\text{thermal}} = \frac{\text{Work output}}{\text{Heat supplied}} = \frac{79.3}{552.9} = \mathbf{0.1434 \text{ or } 14.34\%. \ (Ans.)}$$

Example 5.5. *Air is drawn in a gas turbine unit at 15°C and 1.01 bar and pressure ratio is 7 : 1. The compressor is driven by the H.P. turbine and L.P. turbine drives a separate power shaft. The isentropic efficiencies of compressor, and the H.P. and L.P. turbines are 0.82, 0.85 and 0.85 respectively. If the maximum cycle temperature is 610°C, calculate :*

(*i*) *The pressure and temperature of the gases entering the power turbine.*

(*ii*) *The net power developed by the unit per kg/s mass flow.*

(*iii*) *The work ratio.*

(*iv*) *The thermal efficiency of the unit.*

Neglect the mass of fuel and assume the following :

For compression process c_{pa} = 1.005 kJ/kg K and γ = 1.4

For combustion and expansion processes : c_{pg} = 1.15 kJ/kg K and γ = 1.333.

Solution. *Given* : $T_1 = 15 + 273 = 288$ K, $p_1 = 1.01$ bar, Pressure ratio $= \frac{p_2}{p_1} = 7$,

$\eta_{compressor} = 0.82$, $\eta_{turbine}$ (H.P.) $= 0.85$, $\eta_{turbine}$ (L.P.) $= 0.85$,

Maximum cycle temperature, $T_3 = 610 + 273 = 883$ K

(*i*) **Pressure and temperature of the gases entering the power turbine, p_4' and T_4' :**

Considering *isentropic compression 1-2*, we get

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (7)^{\frac{1.4-1}{1.4}} = 1.745$$

$\therefore$ $$T_2 = 288 \times 1.745 = 502.5 \text{ K}$$

Also $$\eta_{compressor} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$0.82 = \frac{502.5 - 288}{T_2' - 288}$$

$\therefore$ $$T_2' = \frac{502.5 - 288}{0.82} + 288 = 549.6 \text{ K}$$

$$W_{compressor} = c_{pa}(T_2' - T_1) = 1.005 \times (549.6 - 288) = 262.9 \text{ kJ/kg}$$

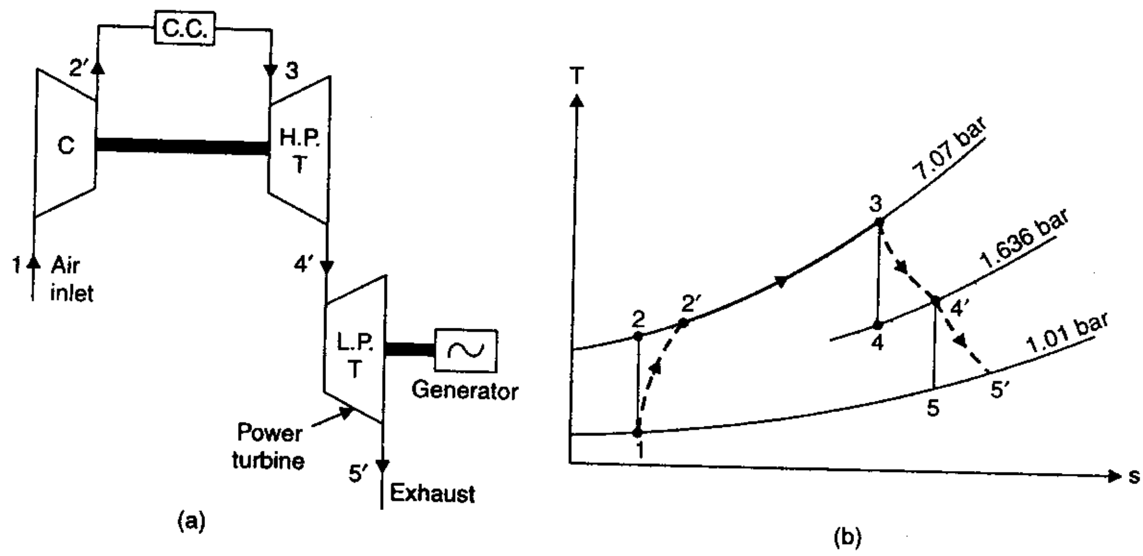


Fig. 5.51

Now, the *work output of H.P. turbine = Work input to compressor*

$\therefore$ $$c_{pg}(T_3 - T_4') = 262.9$$

i.e., $$1.15(883 - T_4') = 262.9$$

$\therefore$ $$T_4' = 883 - \frac{262.9}{1.15} = 654.4 \text{ K}$$

i.e., *Temperature of gases entering the power turbine* = **654.4 K. (Ans.)**

Again, for H.P. turbine :

$$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} \quad i.e. \quad 0.85 = \frac{883 - 654.4}{883 - T_4}$$

$\therefore$ $$T_4 = 883 - \left(\frac{883 - 654.4}{0.85}\right) = 614 \text{ K}$$

Now, considering *isentropic expansion process 3-4*, we get

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

or $$\frac{p_3}{p_4} = \left(\frac{T_3}{T_4}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{883}{614}\right)^{\frac{1.33}{0.33}} = 4.32$$

i.e., $$p_4 = \frac{p_3}{4.32} = \frac{7.07}{4.32} = 1.636 \text{ bar}$$

i.e., Pressure of gases entering the power turbine = **1.636 bar. (Ans.)**

(*ii*) **Net power developed per kg/s mass flow, P :**

To find the power output it is now necessary to calculate T_5'.

The pressure ratio, $\frac{p_4}{p_5}$, is given by $\frac{p_4}{p_3} \times \frac{p_3}{p_5}$

i.e., $$\frac{p_4}{p_5} = \frac{p_4}{p_3} \times \frac{p_2}{p_1} = \frac{7}{4.32} = 1.62 \qquad (\because \; p_2 = p_3 \text{ and } p_5 = p_1)$$

Then, $$\frac{T_4'}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}} = (1.62)^{\frac{0.33}{1.33}} = 1.27$$

$\therefore$ $$T_5 = \frac{T_4'}{1.127} = \frac{654.4}{1.127} = 580.6 \text{ K.}$$

Again, for L.P. turbine

$$\eta_{turbine} = \frac{T_4' - T_5'}{T_4' - T_5}$$

i.e., $$0.85 = \frac{654.4 - T_5'}{654.4 - 580.6}$$

$\therefore$ $$T_5' = 654.4 - 0.85\,(654.4 - 580.6) = 591.7 \text{ K}$$

$$W_{L.P.\ turbine} = c_{pg}\,(T_4' - T_5') = 1.15\,(654.4 - 591.7) = 72.1 \text{ kJ/kg}$$

Hence *net power output* (per kg/s mass flow) = **72.1 kW. (Ans.)**

(*iii*) **Work ratio :**

$$\textit{Work ratio} = \frac{\text{Net work output}}{\text{Gross work output}} = \frac{72.1}{72.1 + 262.9} = \mathbf{0.215.} \quad \textbf{(Ans.)}$$

(*iv*) **Thermal efficiency of the unit, $\eta_{thermal}$:**

Heat supplied $= c_{pg}(T_3 - T_2') = 1.15\,(883 - 549.6) = 383.4$ kJ/kg

$\therefore$ $$\eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{72.1}{383.4} = 0.188 \text{ or } \mathbf{18.8\%.} \quad \textbf{(Ans.)}$$

Example 5.6. *In a gas turbine the compressor takes in air at a temperature of 15°C and compresses it to four times the initial pressure with an isentropic efficiency of 82%. The air is then passed through a heat exchanger heated by the turbine exhaust before reaching the combustion chamber. In the heat exchanger 78% of the available heat is given to the air. The maximum temperature after constant pressure combustion is 600°C, and the efficiency of the turbine is 70%. Neglecting all losses except those mentioned, and assuming the working fluid throughout the cycle to have the characteristic of air find the efficiency of the cycle.*

Assume R = 0.287 kJ/kg K and γ = 1.4 for air and constant specific heats throughout.

Solution. *Given* : $T_1 = 15 + 273 = 288$ K, Pressure ratio, $\frac{p_2}{p_1} = \frac{p_3}{p_4} = 4$, $\eta_{compressor} = 82\%$.

Effectiveness of the heat exchanger, $\varepsilon = 0.78$,

$\eta_{turbine} = 70\%$, Maximum temperature, $T_3 = 600 + 273 = 873$ K.

Efficiency of the cycle η_{cycle} :

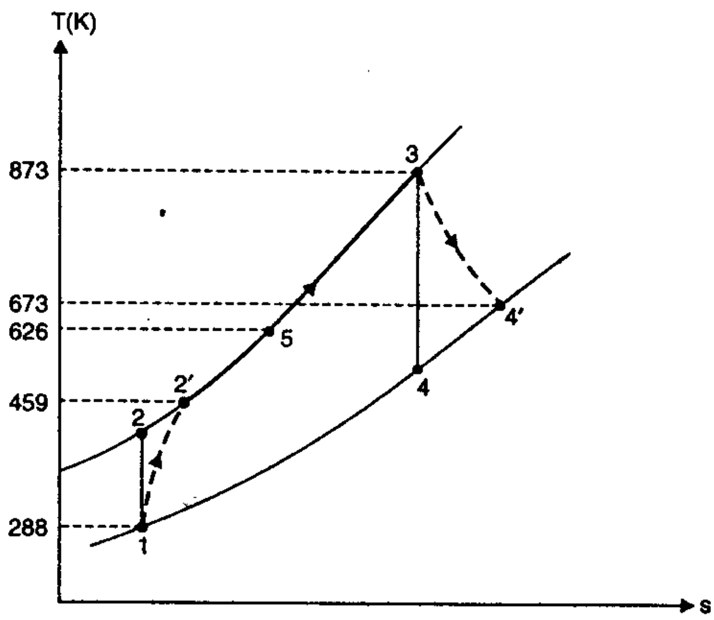


Fig. 5.52

Considering the *isentropic compression 1-2*, we get

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (4)^{\frac{1.4-1}{1.4}} = 1.486$$

∴ $$T_2 = 288 \times 1.486 = 428 \text{ K}$$

Now, $$\eta_{compressor} = \frac{T_2 - T_1}{T_2' - T_1}$$

i.e., $$0.82 = \frac{428 - 288}{T_2' - 288}$$

∴ $$T_2' = \frac{428 - 288}{0.82} + 288 = 459 \text{ K}$$

Considering the isentropic expansion process 3-4, we have

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (4)^{\frac{1.4-1}{1.4}} = 1.486$$

$\therefore$ $$T_4 = \frac{T_3}{1.486} = \frac{873}{1.486} = 587.5 \text{ K.}$$

Again, $$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4} = \frac{873 - T_4'}{873 - 587.5}$$

i.e., $$0.70 = \frac{873 - T_4'}{873 - 587.5}$$

$\therefore$ $$T_4' = 873 - 0.7(873 - 587.5) = 673 \text{ K}$$

$$W_{compressor} = c_p(T_2' - T_1)$$

But $$c_p = R \times \frac{\gamma}{\gamma - 1} = 0.287 \times \frac{1.4}{1.4 - 1} = 1.0045 \text{ kJ/kg K}$$

$\therefore$ $$W_{compressor} = 1.0045(459 - 288) = 171.7 \text{ kJ/kg}$$

$$W_{turbine} = c_p(T_3 - T_4') = 1.0045(873 - 673) = 200.9 \text{ kJ/kg}$$

$\therefore$ $$\text{Net work} = W_{turbine} - W_{compressor} = 200.9 - 171.7 = 29.2 \text{ kJ/kg.}$$

Effectiveness for heat exchanger, $\varepsilon = \dfrac{T_5 - T_2'}{T_4' - T_2'}$

i.e., $$0.78 = \frac{T_5 - 459}{673 - 459}$$

$\therefore$ $$T_5 = (673 - 459) \times 0.78 + 459 = 626 \text{ K}$$

$\therefore$ Heat supplied by fuel per kg

$$= c_p(T_3 - T_5) = 1.0045(873 - 626) = 248.1 \text{ kJ/kg}$$

$\therefore$ $$\eta_{cycle} = \frac{\text{Net work done}}{\text{Heat supplied by the fuel}} = \frac{29.2}{248.1} = 0.117 \text{ or } \mathbf{11.7\%}. \quad \textbf{(Ans.)}$$

Example 5.7. *A gas turbine employs a heat exchanger with a thermal ratio of 72%. The turbine operates between the pressures of 1.01 bar and 4.04 bar and ambient temperature is 20°C. Isentropic efficiencies of compressor and turbine are 80 and 85% respectively. The pressure drop on each side of the heat exchanger is 0.05 bar and in the combustion chamber 0.14 bar. Assume combustion efficiency to be unity and calorific value of the fuel to be 41800 kJ/kg calculate the increase in efficiency due to heat exchanger over that for simple cycle.*

Assume c_p is constant throughout and is equal to 1.024 kJ/kg K, and assume γ = 1.4.

For simple cycle the air-fuel ratio is 90 : 1, and for the heat exchange cycle the turbine entry temperature is the same as for a simple cycle.

Solution. Simple Cycle. Refer Fig. 5.53.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 1.486$$

$\therefore$ $$T_2 = 293 \times 1.486 = 435.4$$

Also, $$\eta_{compressor} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$0.8 = \frac{435.4 - 293}{T_2' - 293}$$

$\therefore$ $$T_2' = \frac{435.4 - 293}{0.8} + 293 = 471 \text{ K}$$

Now $$m_f \times C = (m_a + m_f) \times c_p \times (T_3 - T_2')$$

[m_a = mass of air, m_f = mass of fuel]

$\therefore$ $$T_3 = \frac{m_f \times C}{c_p\,(m_a + m_f)} + T_2' = \frac{1 \times 41800}{1.024\,(90 + 1)} + 471 = 919.5 \text{ K}$$

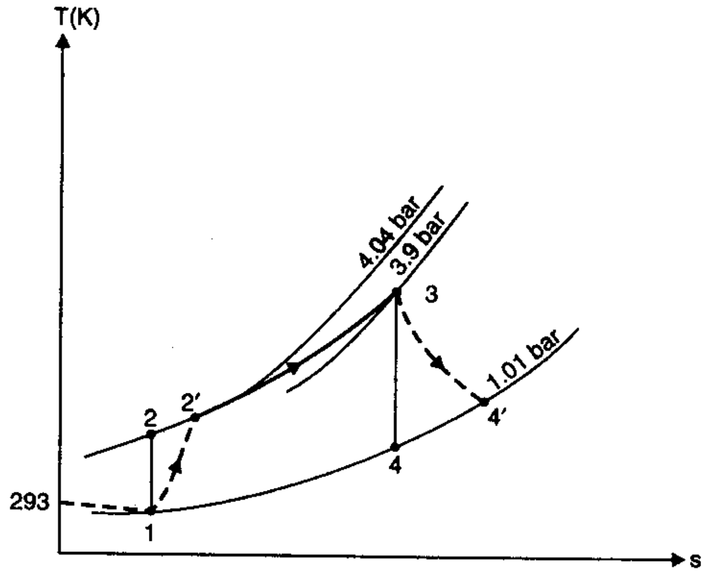


Fig. 5.53

Also, $$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma - 1}{\gamma}}$$

or $$T_4 = T_3 \times \left(\frac{p_4}{p_3}\right)^{\frac{\gamma - 1}{\gamma}} = 919.5 \times \left(\frac{1.01}{3.9}\right)^{\frac{1.4 - 1}{1.4}} = 625 \text{ K}$$

Again, $$\eta_{turbine} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$\therefore$ $$0.85 = \frac{919.5 - T_4'}{919.5 - 625}$$

$\therefore$ $$T_4' = 919.5 - 0.85(919.5 - 625) = 669 \text{ K}$$

$$\eta_{thermal} = \frac{(T_3 - T_4') - (T_2' - T_1)}{(T_3 - T_2')}$$

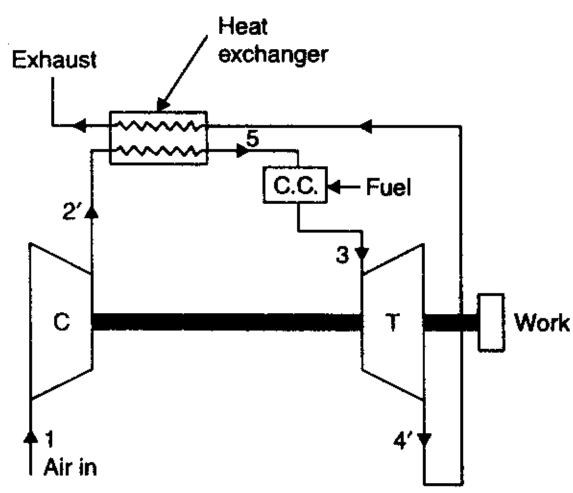
$$= \frac{(919.5 - 669) - (471 - 293)}{(919.5 - 471)} = \frac{72.5}{448.5} = 0.1616 \text{ or } \mathbf{16.16\%}. \quad \textbf{(Ans.)}$$

Heat Exchanger Cycle. Refer Fig. 5.54 (*a*, *b*)

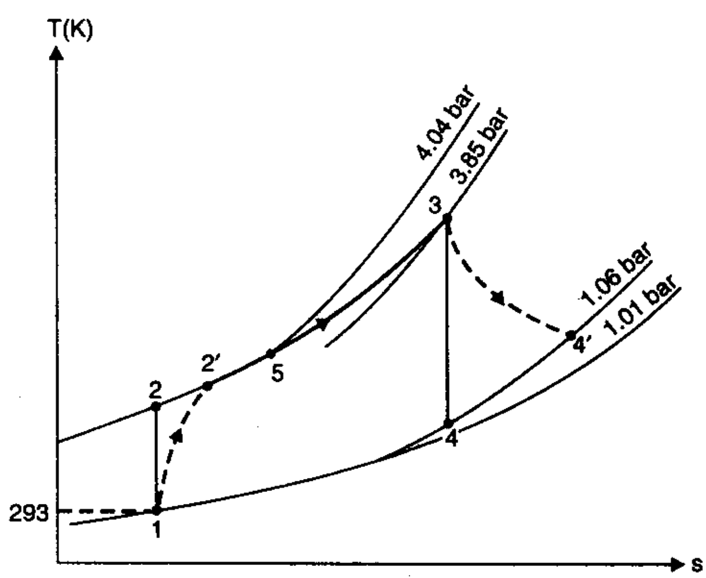
$$T_2' = 471 \text{ K (as for simple cycle)}$$
$$T_3 = 919.5 \text{ K (as for simple cycle)}$$

To find T_4' :

$$p_3 = 4.04 - 0.14 - 0.05 = 3.85 \text{ bar}$$
$$p_4 = 1.01 + 0.05 = 1.06 \text{ bar}$$



(a)



(b)

Fig. 5.54

$$\therefore \quad \frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1.06}{3.85}\right)^{\frac{1.4-1}{1.4}} = 0.69$$

i.e., $$T_4 = 919.5 \times 0.69 = 634 \text{ K}$$

$$\eta_{\text{turbine}} = \frac{T_3 - T_4'}{T_3 - T_4} \ ; \quad 0.85 = \frac{919.5 - T_4'}{919.5 - 634}$$

$$\therefore \quad T_4' = 919.5 - 0.85\,(919.5 - 634) = 677 \text{ K}$$

To find T_5 :

Thermal ratio (or effectiveness),

$$\varepsilon = \frac{T_5 - T_2'}{T_4' - T_2'} \qquad \therefore \quad 0.72 = \frac{T_5 - 471}{677 - 471}$$

$$\therefore \quad T_5 = 0.72(677 - 471) + 471 = 619 \text{ K}$$

$$\eta_{\text{thermal}} = \frac{(T_3 - T_4') - (T_2' - T_1)}{(T_3 - T_5)}$$

$$= \frac{(919.5 - 677) - (471 - 293)}{(919.5 - 619)} = \frac{64.5}{300.5} = 0.2146 \text{ or } \mathbf{21.46\%}$$

$\therefore$ **Increase in thermal efficiency** = 21.46 – 16.16 = **5.3%. (Ans.)**

Example 5.8. *A 5400 kW gas turbine generating set operates with two compressor stages ; the overall pressure ratio is 9 : 1. A high pressure turbine is used to drive the compressors, and a low-pressure turbine drives the generator. The temperature of the gases at entry to the high pressure turbine is 625°C and the gases are reheated to 625°C after expansion in the first turbine. The exhaust gases leaving the low-pressure turbine are passed through a heat exchanger to heat the air leaving the high pressure stage compressor. The compressors have equal pressure ratios and intercooling is complete between the stages. The air inlet temperature to the unit is 20°C. The isentropic efficiency of each compressor stage is 0.8, and the isentropic efficiency of each turbine stage is 0.85, the heat exchanger thermal ratio is 0.8. A mechanical efficiency of 95% can be assumed for both the power shaft and compressor turbine shaft. Neglecting all pressure losses and changes in kinetic energy calculate :*

(i) The thermal efficiency ; *(ii) Work ratio of the plant ;*

(iii) The mass flow in kg/s.

Neglect the mass of the fuel and assume the following :

For air : c_{pa} = 1.005 kJ/kg K and γ = 1.4

For gases in the combustion chamber and in turbines and heat exchanger, c_{pg} = 1.15 kJ/kg K and γ = 1.333.

Solution. Refer Fig. 5.55.

Given : $T_1 = 20 + 273 = 293$ K, $T_6 = T_8 = 625 + 273 = 898$ K

Efficiency of each compressor stage = 0.8

$\eta_{\text{mech.}} = 0.95$, $\varepsilon = 0.8$

(i) **Thermal efficiency, η_{thermal} :**

Since the pressure ratio and the isentropic efficiency of each compressor is the same then the work input required for each compressor is the same since both compressor have the same air inlet temperature *i.e.,* $T_1 = T_3$ and $T_2' = T_4'$.

Also, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \quad \text{and} \quad \frac{p_2}{p_1} = \sqrt{9} = 3$$

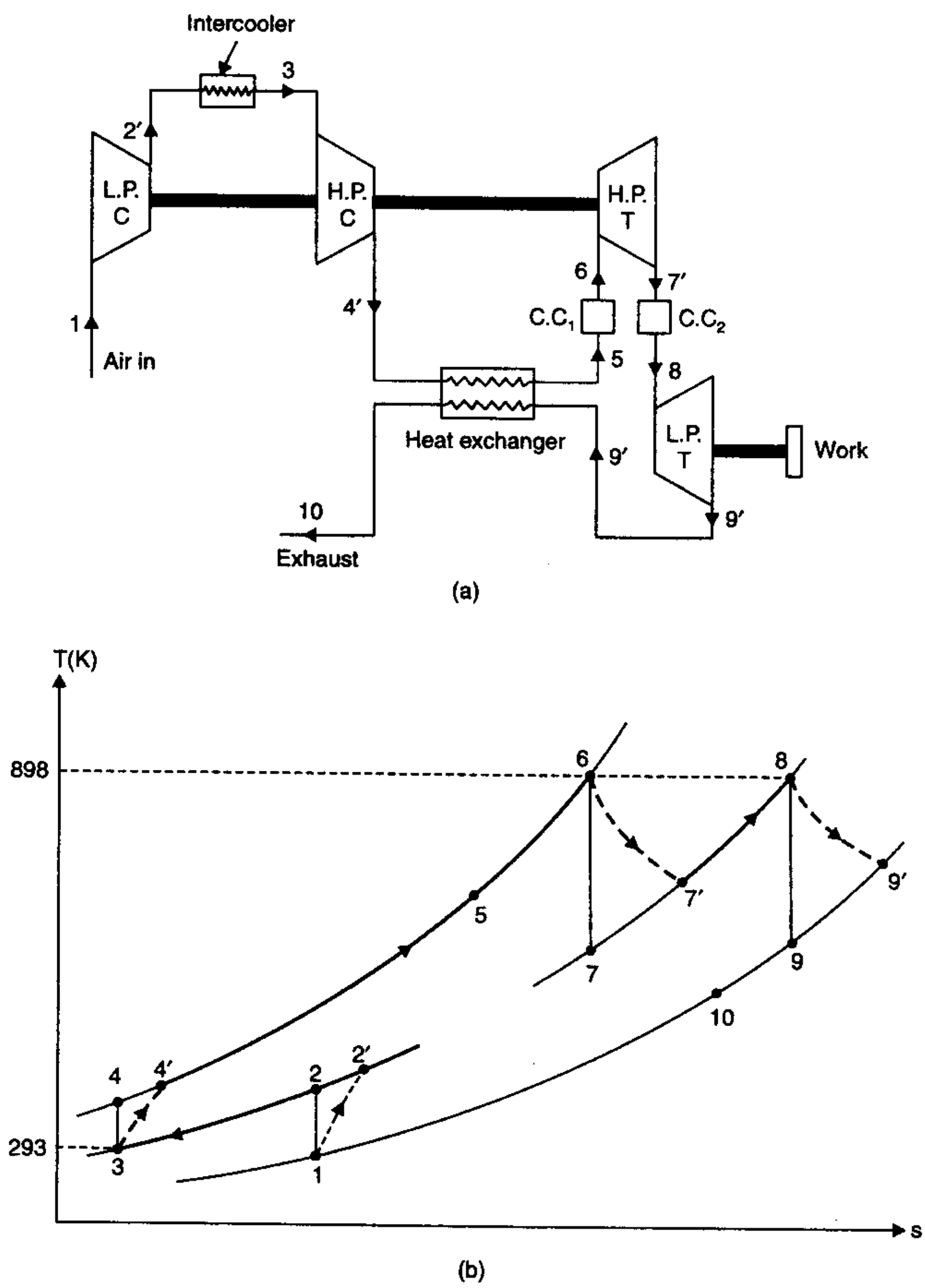


Fig. 5.55

$\therefore \quad T_2 = (20 + 273) \times (3)^{\frac{1.4-1}{1.4}} = 401 \text{ K}$

Now, $\quad \eta_{\text{compressor}} \text{ (L.P.)} = \dfrac{T_2 - T_1}{T_2' - T_1}$

$$0.8 = \frac{401 - 293}{T_2' - 293}$$

i.e., $\quad T_2' = \dfrac{401 - 298}{0.8} + 293 = 428 \text{ K}$

Work input per compressor stage

$$= c_{pa}(T_2' - T_1) = 1.005\,(428 - 293) = 135.6 \text{ kJ/kg}$$

The H.P. turbine is required to drive both compressors and to overcome mechanical friction.

i.e., Work output of H.P. turbine $= \dfrac{2 \times 135.6}{0.95} = 285.5$ kJ/kg

$\therefore \quad c_{pg}\,(T_6 - T_7') = 285.5$

i.e., $\quad 1.15\,(898 - T_7') = 285.5$

$\therefore \quad T_7' = 898 - \dfrac{285.5}{1.15} = 650 \text{ K}$

Now, $\quad \eta_{\text{turbine (H.P.)}} = \dfrac{T_6 - T_7'}{T_6 - T_7}\,; \quad 0.85 = \dfrac{898 - 650}{898 - T_7}$

$\therefore \quad T_7 = 898 - \left(\dfrac{898 - 650}{0.85}\right) = 606 \text{ K}$

Also, $\quad \dfrac{T_6}{T_7} = \left(\dfrac{p_6}{p_7}\right)^{\frac{\gamma - 1}{\gamma}}$

or $\quad \dfrac{p_6}{p_7} = \left(\dfrac{T_6}{T_7}\right)^{\frac{\gamma}{\gamma - 1}} = \left(\dfrac{898}{606}\right)^{\frac{1.333}{0.333}} = 4.82$

Then, $\quad \dfrac{p_8}{p_9} = \dfrac{9}{4.82} = 1.86$

Again, $\quad \dfrac{T_8}{T_9} = \left(\dfrac{p_8}{p_9}\right)^{\frac{\gamma - 1}{\gamma}} = (1.86)^{\frac{1.333 - 1}{1.333}} = 1.16$

$\therefore \quad T_9 = \dfrac{T_8}{1.16} = \dfrac{898}{1.16} = 774 \text{ K}$

Also, $\quad \eta_{\text{turbine (L.P.)}} = \dfrac{T_8 - T_9'}{T_8 - T_9}\,; \quad 0.85 = \dfrac{898 - T_9'}{898 - 774}$

$\therefore \quad T_9' = 898 - 0.85\,(898 - 774) = 792.6 \text{ K}$

$\therefore$ Net work output $= c_{pg}(T_8 - T_9') \times 0.95$

$$= 1.15\,(898 - 792.6) \times 0.95 = 115.15 \text{ kJ/kg}$$

Thermal ratio or effectiveness of heat exchanger,

$$\varepsilon = \frac{T_5 - T_4'}{T_9' - T_4'} = \frac{T_5 - 428}{792.6 - 428}$$

i.e., $$0.8 = \frac{T_5 - 428}{792.6 - 428}$$

$\therefore$ $T_5 = 0.8\,(792.6 - 428) + 428 = 719.7$ K

Now, Heat supplied $= c_{pg}(T_6 - T_5) + c_{pg}(T_8 - T_7')$

$= 1.15\,(898 - 719.7) + 1.15\,(898 - 650) = 490.2$ kJ/kg

$$\therefore \quad \eta_{thermal} = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{115.15}{490.2}$$

$= 0.235$ or **23.5%. (Ans.)**

(*ii*) **Work ratio :**

Gross work of the plant $= W_{turbine\,(H.P.)} + W_{turbine\,(L.P.)}$

$$= 285.5 + \frac{115.15}{0.95} = 406.7 \text{ kJ/kg}$$

$\therefore$ **Work ratio** $= \dfrac{\text{Net work output}}{\text{Gross work output}} = \dfrac{115.15}{406.7} = \mathbf{0.283.}$ **(Ans.)**

(*iii*) **Mass flow, $\dot{m}$:**

Let the mass flow be $\dot{m}$, then

$\dot{m} \times 115.15 = 4500$

$$\therefore \quad \dot{m} = \frac{4500}{115.15} = 39.08 \text{ kg/s}$$

i.e., **Mass flow = 39.08 kg/s. (Ans.)**

Example 5.9. *In a closed cycle gas turbine there is two-stage compressor and a two-stage turbine. All the components are mounted on the same shaft. The pressure and temperature at the inlet of the first-stage compressor are 1.5 bar and 20°C. The maximum cycle temperature and pressure are limited to 750°C and 6 bar. A perfect intercooler is used between the two-stage compressors and a reheater is used between the two turbines. Gases are heated in the reheater to 750°C before entering into the L.P. turbine. Assuming the compressor and turbine efficiencies as 0.82, calculate :*

(*i*) *The efficiency of the cycle without regenerator.*

(*ii*) *The efficiency of the cycle with a regenerator whose effectiveness is 0.70.*

(*iii*) *The mass of the fluid circulated if the power developed by the plant is 350 kW.*

The working fluid used in the cycle is air. For air : $\gamma = 1.4$ and $c_p = 1.005$ kJ/kg K.

Solution. *Given :* $T_1 = 20 + 273 = 293$ K, $T_5 = T_7 = 750 + 273 = 1023$ K, $p_1 = 1.5$ bar, $p_2 = 6$ bar, $\eta_{compressor} = \eta_{turbine} = 0.82$,

Effectiveness of regenerator, $\varepsilon = 0.70$, Power developed, $P = 350$ kW.

For air : $c_p = 1.005$ kJ/kg, K, $\gamma = 1.4$

As per given conditions : $T_1 = T_3$, $T_2' = T_4'$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} \quad \text{and} \quad p_x = \sqrt{p_1 p_2} = \sqrt{1.5 \times 6} = 3 \text{ bar}$$

Now, $$T_2 = T_1 \times \left(\frac{p_2}{T_1}\right)^{\frac{\gamma-1}{\gamma}} = 293 \times \left(\frac{3}{1.5}\right)^{\frac{1.4-1}{1.4}} = 357 \text{ K}$$

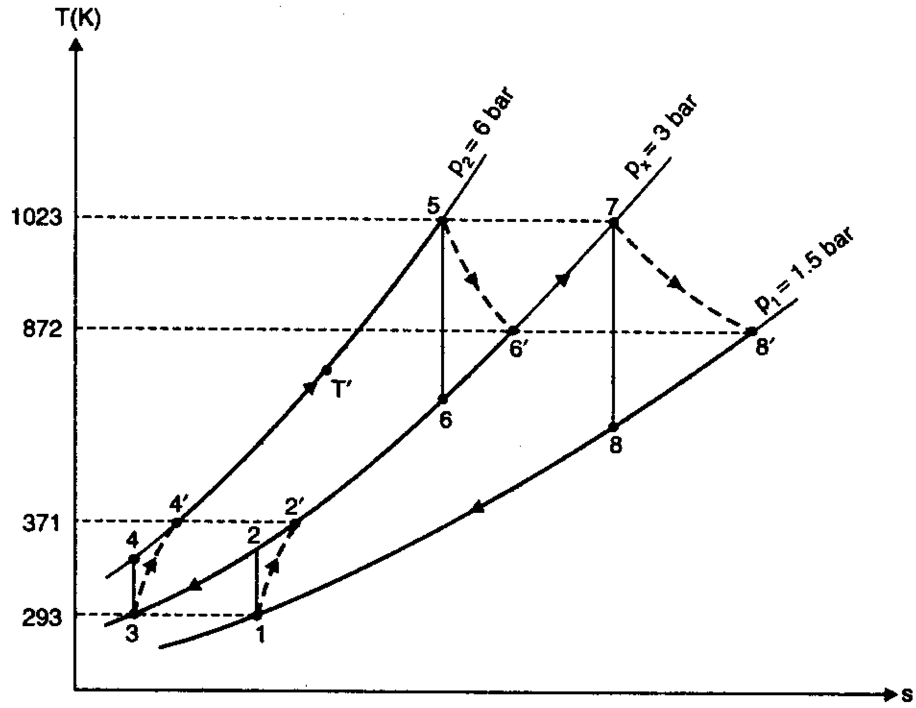


Fig. 5.56

$$\eta_{\text{compressor (L.P.)}} = \frac{T_2 - T_1}{T_2' - T_1}$$

$$0.82 = \frac{357 - 293}{T_2' - 293}$$

$\therefore$
$$T_2' = \frac{357 - 293}{0.82} + 293 = 371 \text{ K}$$

i.e.,
$$T_2' = T_4' = 371 \text{ K}$$

Now,
$$\frac{T_5}{T_6} = \left(\frac{p_5}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_x}\right)^{\frac{1.4-1}{1.4}} \qquad \left[\because \begin{array}{l} p_5 = p_2 \\ p_6 = p_x \end{array}\right]$$

$$\frac{1023}{T_6} = \left(\frac{6}{3}\right)^{0.286} = 1.219$$

$\therefore$
$$T_6 = \frac{1023}{1.219} = 839 \text{ K}$$

$$\eta_{\text{turbine (H.P.)}} = \frac{T_5 - T_6'}{T_5 - T_6}$$

$$0.82 = \frac{1023 - T_6'}{1023 - 839}$$

$\therefore \quad T_6' = 1023 - 0.82\,(1023 - 839) = 872$ K

$T_8' = T_6' = 872$ K as $\eta_{turbine\,(H.P.)} = \eta_{turbine\,(L.P.)}$

and $\quad T_7 = T_5 = 1023$ K

Effectiveness of regenerator, $\varepsilon = \dfrac{T' - T_4'}{T_8' - T_4'}$

where T' is the temperature of air coming out of regenerator

$\therefore \quad 0.70 = \dfrac{T' - 371}{872 - 371}$ *i.e.* $T' = 0.70\,(872 - 371) + 371 = 722$ K

Net work available, $\quad W_{net} = [W_{T(L.P.)} + W_{T(L.P.)}] - [W_{C(H.P.)} + W_{C(L.P.)}]$

$= 2\,[W_{T(L.P.)} - W_{C(L.P.)}]$ as the *work developed by each turbine is same and work absorbed by each compressor is same.*

$\therefore \quad W_{net} = 2c_p\,[(T_5 - T_6') - (T_2' - T_1)]$

$= 2 \times 1.005\,[(1023 - 872) - (371 - 293)] = 146.73$ kJ/kg of air

Heat supplied per kg of air *without regenerator*

$= c_p(T_5 - T_4') + c_p(T_7 - T_6')$

$= 1.005\,[(1023 - 371) + (1023 - 872)] = 807$ kJ/kg of air

Heat supplied per kg of air *with regenerator*

$= c_p(T_5 - T') + c_p(T_7 - T_6')$

$= 1.005\,[(1023 - 722) + (1023 - 872)]$

$= 454.3$ kJ/kg

(*i*) $\eta_{thermal\,(without\,regenerator)} = \dfrac{146.73}{807} = 0.182$ or **18.2%. (Ans.)**

(*ii*) $\eta_{thermal\,(with\,regenerator)} = \dfrac{146.73}{454.3} = 0.323$ or **32.3%. (Ans.)**

(*iii*) **Mass of fluid circulated, $\dot{m}$:**

Power developed, $\quad P = 146.73 \times \dot{m}$ kW

$\therefore \quad 350 = 146.73 \times \dot{m}$

i.e., $\quad \dot{m} = \dfrac{350}{146.73} = 2.38$ kg/s

i.e., **Mass of fluid circulated = 2.38 kg/s. (Ans.)**

☞**Example 5.10.** *The air in a gas turbine plant is taken in L.P. compressor at 293 K and 1.05 bar and after compression it is passed through intercooler where its temperature is reduced to 300 K. The cooled air is further compressed in H.P. unit and then passed in the combustion chamber where its temperature is increased to 750°C by burning the fuel. The combustion products expand in H.P. turbine which runs the compressor and further expansion is continued in L.P. turbine which runs the alternator. The gases coming out from L.P. turbine are used for heating the incoming air from H.P. compressor and then expanded to atmosphere.*

Pressure ratio of each compressor = 2, isentropic efficiency of each compressor stage = 82%, isentropic efficiency of each turbine stage = 82%, effectiveness of heat exchanger = 0.72, air flow = 16 kg/s, calorific value of fuel = 42000 kJ/kg, c_v (for gas) = 1.0 kJ/kg K, c_p (gas) = 1.15 kJ/kg K, γ (for air) = 1.4, γ (for gas) = 1.33.

Neglecting the mechanical, pressure and heat losses of the system and fuel mass also, determine the following :

(*i*) *The power output.* (*ii*) *Thermal efficiency.*

(*iii*) *Specific fuel consumption.*

Solution. *Given* : $T_1 = 293$ K, $T_3 = 300$ K, $\frac{p_2}{p_1} = \frac{p_4}{p_3} = 2$, $T_6 = 750 + 273 = 1023$ K,

$\eta_{compressor} = 82\%$, $\eta_{turbine} = 82\%$, $\varepsilon = 0.72$, $\dot{m}_a = 16$ kg/s, $C = 42000$ kJ/kg,

$c_{pa} = 1.0$ kJ/kg K, $c_{pg} = 1.15$ kJ/kg K, γ (for air) = 1.4, γ (for gas) = 1.33.

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (2)^{\frac{1.4-1}{1.4}} = 1.219$$

$\therefore$ $$T_2 = 293 \times 1.219 = 357 \text{ K}$$

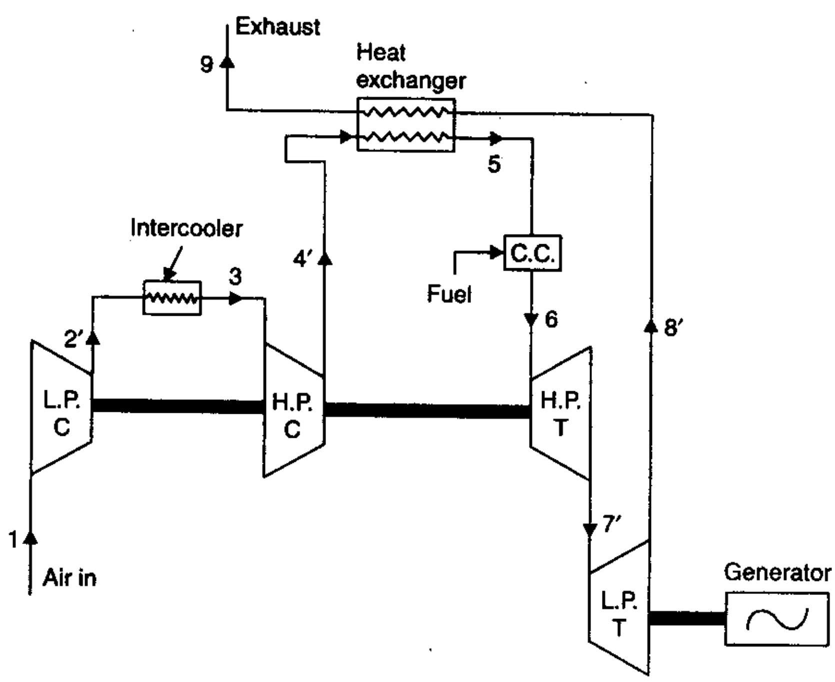
Also, $$\eta_{compressor} = \frac{T_2 - T_1}{T_2' - T_1}$$

$\therefore$ $$0.82 = \frac{357 - 293}{T_2' - 293}$$

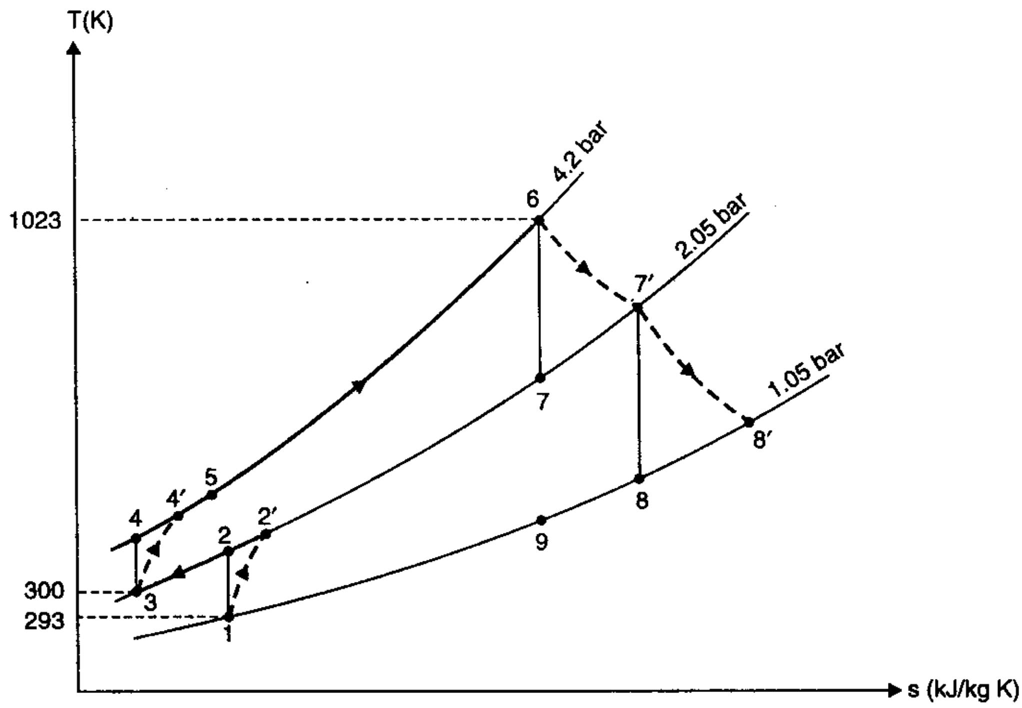
$\therefore$ $$T_2' = \left(\frac{357 - 293}{0.82}\right) + 293 = 371 \text{ K}$$

Similarly, $$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = (2)^{\frac{1.4-1}{1.4}} = 1.219$$

$\therefore$ $$T_4 = 300 \times 1.219 = 365.7 \text{ K}$$



(a)



(b)

Fig. 5.57

and $$\eta_{\text{compressor}} = \frac{T_4 - T_3}{T_4' - T_3}$$

$\therefore$ $$0.82 = \frac{365.7 - 300}{T_4' - 300}$$

i.e., $$T_4' = \left(\frac{365.7 - 300}{0.82}\right) + 300 = 380 \text{ K}$$

Work output of H.P. turbine = Work input to compressor.

Neglecting mass of fuel we can write

$$c_{pg}(T_6 - T_7') = c_{pa}[(T_2' - T_1) + (T_4' - T_3)]$$

$$1.15(1023 - T_7') = 1.0[(371 - 293) + (380 - 300)]$$

or $$1.15(1023 - T_7') = 158$$

$\therefore$ $$T_7' = 1023 - \frac{15.8}{1.15} = 886 \text{ K}$$

Also, $$\eta_{\text{turbine (H.P.)}} = \frac{T_6 - T_7'}{T_6 - T_7}$$

i.e., $$0.82 = \frac{1023 - 886}{1023 - T_7}$$

$\therefore$ $$T_7 = 1023 - \left(\frac{1023 - 886}{0.82}\right) = 856 \text{ K}$$

Now, $$\frac{T_6}{T_7} = \left(\frac{p_6}{p_7}\right)^{\frac{\gamma-1}{\gamma}}$$

$\therefore$ $$\frac{p_6}{p_7} = \left(\frac{T_6}{T_7}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{1023}{856}\right)^{\frac{1.33}{1.33-1}} = 2.05$$

i.e., $$p_7 = \frac{p_6}{2.05} = \frac{4.2}{2.05} = 2.05 \text{ bar} \qquad [\because \; p_6 = 1.05 \times 4 = 4.2 \text{ bar}]$$

$$\frac{T_7'}{T_8} = \left(\frac{p_7}{p_8}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2.05}{1.05}\right)^{\frac{1.33-1}{1.33}} = 1.18$$

$$T_8 = \frac{T_7'}{1.18} = \frac{886}{1.18} = 751 \text{ K}$$

Again, $$\eta_{\text{turbine (L.P.)}} = \frac{T_7' - T_8'}{T_7' - T_8}$$

$$0.82 = \frac{886 - T_8'}{886 - 751}$$

$\therefore$ $$T_8' = 886 - 0.82\,(886 - 751) = 775 \text{ K}$$

(*i*) **Power output :**

Net power output $= c_{pg}\,(T_7' - T_8')$

$= 1.15\,(886 - 775) = 127.6$ kJ/kg

$\therefore$ Net output per second $= \dot{m} \times 127.6$

$= 16 \times 127.6 = 2041.6$ kJ/s $=$ **2041.6 kW. (Ans.)**

(*ii*) **Thermal efficiency :**

Effectiveness of heat exchanger, $\varepsilon = \dfrac{T_5 - T_4'}{T_8' - T_4'}$

i.e., $$0.72 = \frac{T_5 - 380}{775 - 380}$$

$\therefore$ $$T_5 = 0.72(775 - 380) + 380 = 664 \text{ K}$$

Heat supplied in combustion chamber per second

$= \dot{m}_a\, c_{pg}\,(T_6 - T_5)$

$= 16 \times 1.15(1023 - 664) = 6605.6$ kJ/s

$\therefore$ $$\eta_{\text{thermal}} = \frac{2041.6}{6605.6} = 0.309 \text{ or } \mathbf{30.9\%.} \textbf{ (Ans.)}$$

(*iii*) **Specific fuel consumption :**

If m_f is the mass of fuel supplied per kg of air, then

$$m_f \times 42000 = 1.15(1023 - 664)$$

$\therefore$ $$\frac{1}{m_f} = \frac{42000}{1.15(1023 - 664)} = \frac{101.7}{1}$$

$\therefore$ Air-fuel ratio $= 101.7 : 1$

$\therefore$ Fuel supplied per hour $= \dfrac{16 \times 3600}{101.7} = 566.37$ kg/h

$\therefore$ **Specific fuel consumption** $= \dfrac{566.37}{2041.6} =$ **0.277 kg/kWh. (Ans.)**

☞**Example 5.11.** *Air is taken in a gas turbine plant at 1.1 bar 20°C. The plant comprises of L.P. and H.P. compressors and L.P. and H.P. turbines. The compression in L.P. stage is upto 3.3 bar followed by intercooling to 27°C. The pressure of air after H.P. compressor is 9.45 bar. Loss in pressure during intercooling is 0.15 bar. Air from H.P. compressor is transferred to heat exchanger of effectiveness 0.65 where it is heated by the gases from L.P. turbine. After heat exchanger the air passes through combustion chamber. The temperature of gases supplied to H.P. turbine is 700°C. The gases expand in H.P. turbine to 3.62 bar and air then reheated to 670°C before expanding in L.P. turbine. The loss of pressure in reheater is 0.12 bar. Determine :*

(*i*) *The overall efficiency* (*ii*) *The work ratio*

(*iii*) *Mass flow rate when the power generated is 6000 kW.*

Assume : Isentropic efficiency of compression in both stages = 0.82.

Isentropic efficiency of expansion in turbines = 0.85.

For air : $c_p = 1.005$ *kJ/kg K,* $\gamma = 1.4$.

For gases : $c_p = 1.15$ *kJ/kg K,* $\gamma = 1.33$.

Neglect the mass of fuel.

Solution. *Given :* $T_1 = 20 + 273 = 293$ K, $p_1 = 1.1$ bar, $p_2 = 3.3$ bar, $T_3 = 27 + 273 = 300$ K,

$p_3 = 3.3 - 0.15 = 3.15$ bar, $p_4 = p_6 = 9.45$ bar, $T_6 = 973$ K,

$T_8 = 670 + 273 = 943$ K, $p_8 = 3.5$ bar,

$\eta_{compressors} = 82\%$, $\eta_{turbines} = 85\%$, Power generated = 6000 kW,

Effectiveness, $\varepsilon = 0.65$, $c_{pa} = 1.005$ kJ/kg K, $\gamma_{air} = 1.44$, $c_{pg} = 1.15$ kJ/kg K and $\gamma_{gases} = 1.33$.

Refer Fig. 5.58

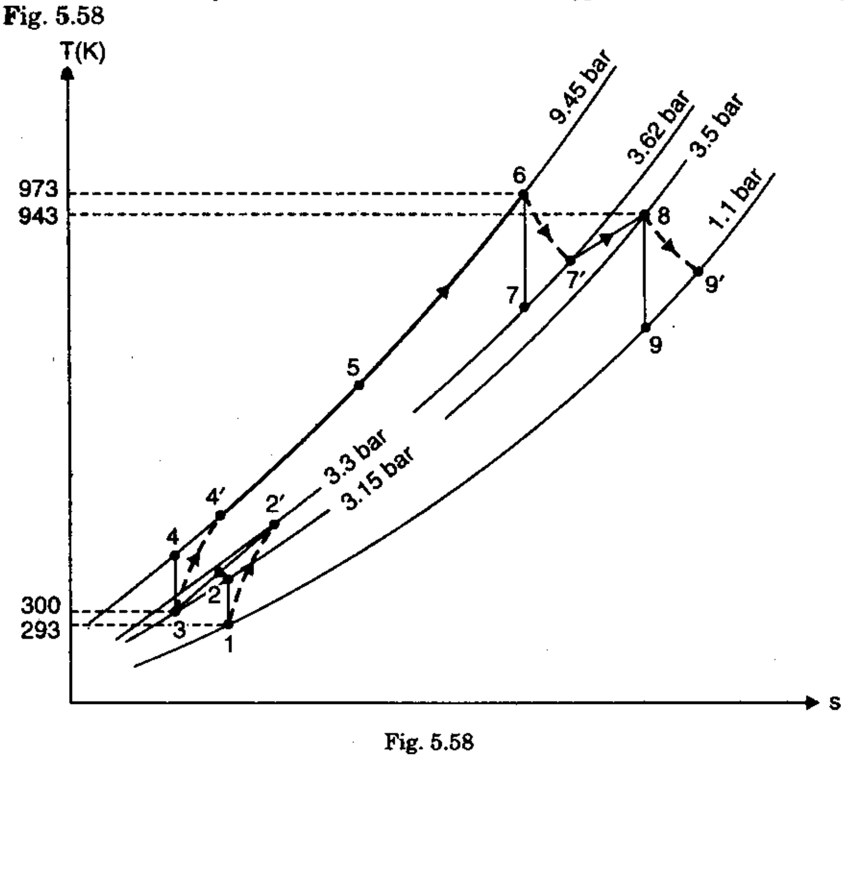


Fig. 5.58

Now, $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.3}{1.1}\right)^{\frac{1.4-1}{1.4}} = 1.369$$

$\therefore$ $$T_2 = 293 \times 1.369 = 401 \text{ K}$$

$$\eta_{\text{compressor (L.P.)}} = 0.82 = \frac{T_2 - T_1}{T_2' - T_1} = \frac{401 - 293}{T_2' - 293}$$

$\therefore$ $$T_2' = \left(\frac{401 - 293}{0.82}\right) + 293 = 425 \text{ K}$$

Again, $$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9.45}{3.15}\right)^{\frac{1.4-1}{1.4}} = 1.369$$

$\therefore$ $$T_4 = 300 \times 1.369 = 411 \text{ K}$$

Now, $$\eta_{\text{compressor (H.P.)}} = \frac{T_4 - T_3}{T_4' - T_3}$$

$$0.82 = \frac{411 - 300}{T_4' - 300}$$

$\therefore$ $$T_4' = \left(\frac{411 - 300}{0.82}\right) + 300 = 435 \text{ K}$$

Similarly, $$\frac{T_6}{T_7} = \left(\frac{p_6}{p_7}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{9.45}{3.62}\right)^{\frac{1.33-1}{1.33}} = 1.268$$

$\therefore$ $$T_7 = \frac{T_6}{1.268} = \frac{973}{1.268} = 767 \text{ K}$$

Also, $$\eta_{\text{turbine (H.P.)}} = \frac{T_6 - T_7'}{T_6 - T_7}$$

$$0.85 = \frac{973 - T_7'}{973 - 767}$$

$\therefore$ $$T_7' = 973 - 0.85(973 - 767) = 798 \text{ K}$$

Again, $$\frac{T_8}{T_9} = \left(\frac{p_8}{p_9}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.5}{1.1}\right)^{\frac{1.33-1}{1.33}} = 1.332$$

$\therefore$ $$T_9 = \frac{T_8}{1.332} = \frac{943}{1.332} = 708 \text{ K}$$

$$\eta_{\text{turbine (L.P.)}} = \frac{T_8 - T_9'}{T_8 - T_9}$$

$$0.85 = \frac{943 - T_9'}{943 - 708}$$

$\therefore$ $$T_9' = 943 - 0.85(943 - 708) = 743 \text{ K.}$$

Effectiveness of heat exchanger,

$$\varepsilon = 0.65 = \frac{T_5 - T_4'}{T_9' - T_4'}$$

i.e.,
$$0.65 = \frac{T_5 - 435}{743 - 435}$$

$\therefore$
$$T_5 = 0.65(743 - 435) + 435 = 635 \text{ K}$$

$$W_{\text{turbine (H.P.)}} = c_{pg}\,(T_6 - T_7')$$
$$= 1.15(973 - 798) = 201.25 \text{ kJ/kg of gas}$$

$$W_{\text{turbine (L.P.)}} = c_{pg}(T_8 - T_9')$$
$$= 1.15(943 - 743) = 230 \text{ kJ/kg of gas}$$

$$W_{\text{compressor (L.P.)}} = c_{pa}(T_2' - T_1)$$
$$= 1.005(425 - 293) = 132.66 \text{ kJ/kg of air}$$

$$W_{\text{compressor (H.P.)}} = c_{pa}(T_4' - T_3)$$
$$= 1.005(435 - 300) = 135.67 \text{ kJ/kg of air}$$

Heat supplied
$$= c_{pg}(T_6 - T_5) + c_{pg}(T_8 - T_7')$$
$$= 1.15(973 - 635) + 1.15(943 - 798) = 555.45 \text{ kJ/kg of gas}$$

(*i*) **Overall efficiency η_{overall} :**

$$\eta_{\text{overall}} = \frac{\text{Net work done}}{\text{Heat supplied}}$$
$$= \frac{[W_{\text{turbine (H.P.)}} + W_{\text{turbine (L.P.)}}] - [W_{\text{comp. (L.P.)}} + W_{\text{comp. (H.P.)}}]}{\text{Heat supplied}}$$
$$= \frac{(201.25 + 230) - (132.66 + 135.67)}{555.45}$$
$$= \frac{162.92}{555.45} = 0.293 \text{ or } \mathbf{29.3\%}. \quad \textbf{(Ans.)}$$

(*ii*) **Work ratio :**

$$\textit{Work ratio} = \frac{\text{Net work done}}{\text{Turbine work}}$$
$$= \frac{[W_{turbine\,(\text{H.P.})} + W_{turbine\,(\text{L.P.})}] - [W_{comp.\,(\text{L.P.})} + W_{comp.\,(\text{H.P.})}]}{[W_{turbine\,(\text{H.P.})} + W_{turbine\,(\text{L.P.})}]}$$
$$= \frac{(201.25 + 230) - (132.66 + 135.67)}{(201.25 + 230)} = \frac{162.92}{431.25} = 0.377.$$

i.e., *Work ratio* = **0.377. (Ans.)**

(*iii*) **Mass flow rate, $\dot{m}$:**

Net work done = 162.92 kJ/kg.

Since mass of fuel is neglected, for 6000 kW, mass flow rate,

$$\dot{m} = \frac{6000}{162.92} = 36.83 \text{ kg/s}$$

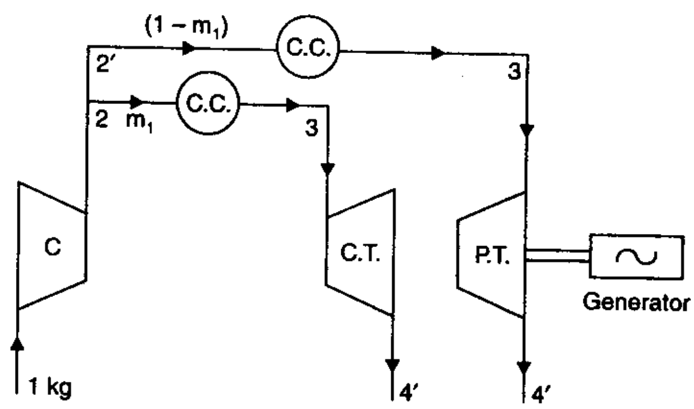
i.e., *Mass flow rate* = **36.83 kg/s. (Ans.)**

ADDITIONAL / TYPICAL EXAMPLES

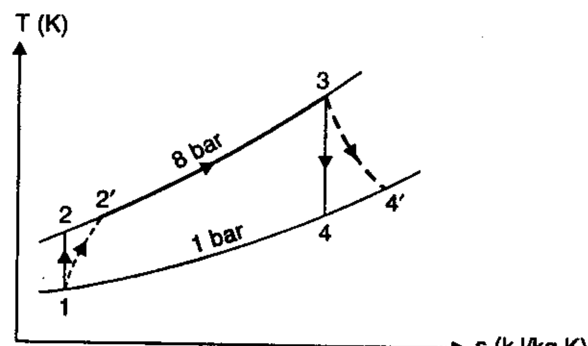
Example 5.12. *A gas turbine plant consists of two turbines. One compressor turbine to drive compressor and other power turbine to develop power output and both are having their own combustion chambers which are served by air directly from the compressor. Air enters the compressor* *at 1 bar and 288 K and is compressed to 8 bar with an isentropic efficiency of 76%. Due to heat added in the combustion chamber, the inlet temperature of gas to both turbines is 900°C. The isentropic efficiency of turbines is 86%. The mass flow rate of air at the compressor is 23 kg/s. The calorific value of fuel is 4200 kJ/kg. Calculate the output of the plant and the thermal efficiency if mechanical efficiency is 95% and generator efficiency is 96%. Take c_p = 1.005 kJ/kg K and γ = 1.4 for air and c_{pg} = 1.128 kJ/kg K and γ = 1.34 for gases.* **(M.U.)**

Solution. *Given* : p_1 = 1 bar ; T_1 = 288 K ; p_2 = 8 bar, $\eta_{(isen)}$ = 76% ; T_3 = 900°C or 1173 K; $\eta_{T(isen.)}$ = 86%, m_a = 23 kg/s ; C.V. = 4200 kJ/kg ; $\eta_{mech.}$ = 95% ; $\eta_{gen.}$ = 96% ; c_p = 1.005 kJ/kg ; γ_a = 1.4 ; c_{pg} = 1.128 kJ/kg K ; γ_g = 1.34.

The arrangement of the plant and the cycle are shown in Fig. 5.59 (*a*), (*b*) respectively.



(a)



(b)

Fig. 5.59

Considering *isentropic compression process 1-2*, we have

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{8}{1}\right)^{\frac{1.4-1}{1.4}} = 1.811$$

$\therefore \quad T_2 = 288 \times 1.811 = 521.6 \text{ K}$

Also, $\quad \eta_{C(isen.)} = \dfrac{T_2 - T_1}{T_2' - T_1}$

or $\quad 0.76 = \dfrac{521.6 - 288}{T_2' - 288}$

or $\quad T_2' = \dfrac{521.6 - 288}{0.76} + 288 = 595.4 \text{ K}$

Considering *isentropic expansion process 3-4*, we have

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{1}{8}\right)^{\frac{1.34 - 1}{1.34}} = 0.59$$

$\therefore \quad T_4 = 1173 \times 0.59 = 692.1 \text{ K}$

Also, $\quad \eta_{T(isen.)} = \dfrac{T_3 - T_4'}{T_3 - T_4}$

or $\quad 0.86 = \dfrac{1173 - T_4'}{1173 - 692.1}$

$\therefore \quad T_4' = 1173 - 0.86(1173 - 692.1) = 759.4 \text{ K}$

Consider 1 kg of air flow through compressor.

$$W_{compressor} = c_p\,(T_2' - T_1) = 1.005(595.4 - 288) = 308.9 \text{ kJ}$$

This is equal to work of compressor turbine.

$\therefore \quad 308.9 = m_1 \times c_{pg}\,(T_3 - T_4')$, neglecting fuel mass

or $\quad m_1 = \dfrac{308.9}{1.128\,(1173 - 759.4)} = 0.662 \text{ kg}$

and flow through the power turbine $= 1 - m_1 = 1 - 0.662 = 0.338$ kg

$$\therefore \quad W_{PT} = (1 - m_1) \times c_{pg}(T_3 - T_4')$$
$$= 0.338 \times 1.128\,(1173 - 759.4) = 157.7 \text{ kJ}$$

$\therefore$ **Power output** $= 23 \times 157.7 \times \eta_{mech.} \times \eta_{gen.}$

$= 23 \times 157.7 \times 0.95 \times 0.96 =$ **3307.9 kJ. (Ans.)**

$$Q_{input} = c_{pg}T_3 - c_{pa}T_2'$$
$$= 1.128 \times 1173 - 1.005 \times 595.4 = 724.7 \text{ kJ/kg of air}$$

Thermal efficiency, $\eta_{th} = \dfrac{157.7}{724.7} \times 100 =$ **21.76%. (Ans.)**

Example 5.13. *(a) Why are the back work ratios relatively high in gas turbine plants compared to those of steam power plants ?*

(b) In a gas turbine plant compression is carried out in two stages with perfect intercooling and expansion in one stage turbine. If the maximum temperature (T_{max} K) and minimum temperature (T_{min}K) in the cycle remain constant, show that for maximum specific output of the plant, the optimum overall pressure ratio is given by

$$r_{opt.} = \left(\eta_T \cdot \eta_C \cdot \frac{T_{max}}{T_{min}}\right)^{\frac{2\gamma}{3(\gamma - 1)}}$$

where γ = *Adiabatic index ;* η_T = *Isentropic efficiency of the turbine.*

η_C = *Isentropic efficiency of compressor.* **(N.U.)**

Solution. (*a*) **Back work ratio** may be defined as *the ratio of negative work to the turbine work in a power plant.* In gas turbine plants, air is compressed from the turbine exhaust pressure to the combustion chamber pressure. This work is given by $-\int v dp$. As the specific volume of air is very high (even in closed cycle gas turbine plants), the compressor work required is very high, and also *bulky compressor is required.* In steam power plants, the turbine exhaust is changed to liquid phase in the condenser. The pressure of condensate is raised to boiler pressure by condensate extraction pump and boiler feed pump in series since the specific volume of water is very small as compared to that of air, the pump work ($-\int v dp$), is also very small. From the above reasons, the back work ratio

$$= \frac{-\int v dp}{\text{Turbine work}}$$

for gas *turbine plants is relatively high compared to that for steam power plants.*

(*b*) Refer Fig. 5.60.

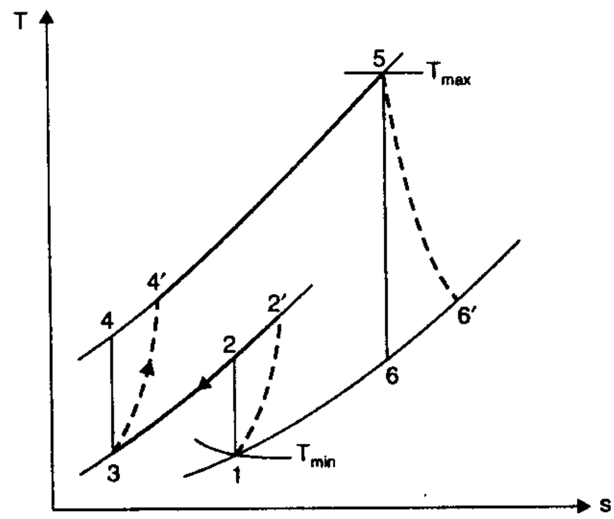


Fig. 5.60

Assuming optimum pressure ratio in each stage of the compressors as $\sqrt{r}$, we have

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

or $$T_2 = T_{\min} \times (r)^{\frac{\gamma-1}{2\gamma}}$$

$$W_{\text{compressor}} = 2[c_p\,(T_2' - T_1)] \text{ for both compressors}$$

$$= 2c_p\,\frac{T_2 - T_1}{\eta_C} = \frac{2c_p}{\eta_C}\,T_{\min}\left[(r)^{\frac{\gamma-1}{2\gamma}} - 1\right], \text{ as } T_1 = T_{\min}$$

Also, $$\frac{T_5}{T_6} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r)^{\frac{\gamma-1}{2\gamma}}$$

$$\therefore \quad T_6 = \frac{T_5}{(r)^{\gamma - 1/\gamma}} = \frac{T_{max}}{(r)^{\gamma - 1/\gamma}}, \text{ as } T_5 = T_{max}$$

$$W_{turbine} = c_p (T_5 - T_6') = c_p \left[T_{max} - \frac{T_{max}}{(r)^{\gamma - 1/\gamma}}\right] \eta_T, \text{ as } \eta_T = \frac{T_5 - T_6'}{T_5 - T_6}$$

$$= c_p T_{max} \left[1 - \frac{1}{(r)^{\gamma - 1/\gamma}}\right] \eta_T$$

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

$$= c_p \eta_T T_{max} \left[1 - \frac{1}{(r)^{\gamma - 1/\gamma}}\right] - \frac{2c_p}{\eta_C} T_{min} [(r)^{\frac{\gamma - 1}{2\gamma}} - 1]$$

For maximum work output,

$$\frac{dW_{net}}{dr} = 0$$

or $$- c_p \eta_T T_{max} \left(-\frac{\gamma - 1}{\gamma}\right)(r)^{-\left(\frac{\gamma - 1}{\gamma}\right) - 1} - \frac{2c_p}{\eta_C} T_{min} \left(\frac{\gamma - 1}{2\gamma}\right)(r)^{\frac{\gamma - 1}{2\gamma} - 1} = 0$$

or $$\eta_T \eta_C \frac{T_{max}}{T_{min}} = (r)^{3(\gamma - 1)/2\gamma}, \text{ on simplification.}$$

Hence, the optimum pressure ratio is

$$r_{opt} = \left[\eta_T \cdot \eta_C \cdot \frac{T_{max}}{T_{min}}\right]^{\frac{2\gamma}{3(\gamma - 1)}}. \quad \textbf{Proved.}$$

Example 5.14. *A gas turbine power plant of 12 MW capacity working on closed cycle, using air as working medium, is to be designed for maximum specific work output. Find the cost of energy generated if the plant is running at designed capacity. Use the following data :*

The temperature of air at inlet	*= 298 K*
The maximum temperature in the cycle	*= 950 K*
Isentropic efficiency of compressor	*= 82%*
Isentropic efficiency of turbine	*= 88%*
Mechanical efficiency	*= 94%*
Generation efficiency	*= 94%*
Combustion efficiency	*= 95%*
Percentage of heat developed in the combustion chamber given to the air working into the system	*= 90%*
Effectiveness of regenerator	*= 0.68*
Cost of fuel used	*= Rs. 4500 per tonne*
Calorific value of fuel used	*= 41000 kJ/kg*
All other charges including profit per hour	*= Rs. 3500/hour*

Neglect heat and pressure losses in the system. Take c_p for air = 1.005 kJ/kg K.

Solution. *Given :* T_1 = 298 K ; T_4 = 950 K ; $\eta_{comp.}$ = 82% ; $\eta_{turbine}$ = 88% ; $\eta_{mech.}$ = 94% ; $\eta_{gen.}$ = 94% ; $\eta_{comb.}$ = 95% ; ϵ = 0.68 ; cost of fuel used = Rs. 4500 per tonne ; C = 41000 kJ/kg ; Other charges = Rs. 3500 per hour.

Cost of energy generated :

The schematic arrangement of the plant and its corresponding T-s diagram are shown in the Fig. 5.61 (a) and (b) respectively.

As the plant is designed for maximum specific output,

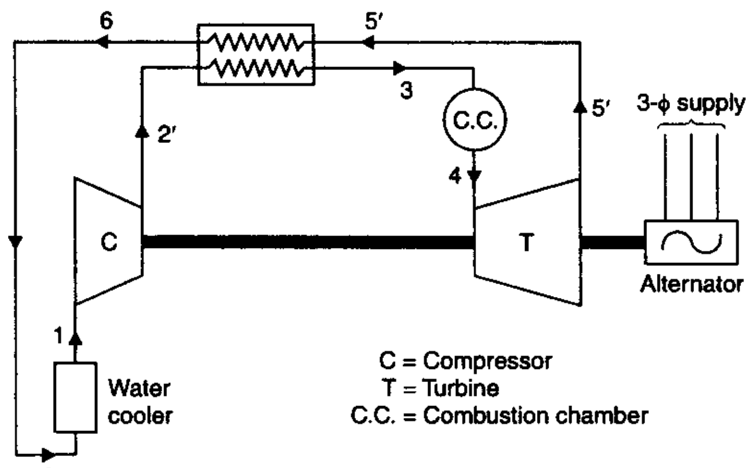
$$r_p = \frac{p_2}{p_1} = \left(\eta_C \times \eta_T \times \frac{T_3}{T_1}\right)^{\left[\frac{\gamma}{2(\gamma - 1)}\right]}$$

$$= \left(0.82 \times 0.88 \times \frac{950}{298}\right)^{\left[\frac{1.4}{2(1.4 - 1)}\right]} = (2.3)^{1.75} = 4.3$$

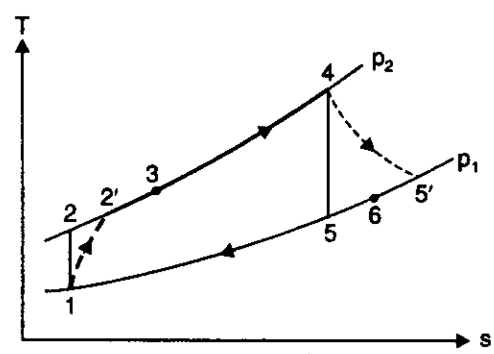
Compressor : $$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} = (4.3)^{\frac{1.4 - 1}{1.4}} = 1.517$$

$\therefore$ $$T_2 = 298 \times 1.517 = 452 \text{ K}$$

$$\eta_C = 0.82 = \frac{T_2 - T_1}{T_2' - T_1} = \frac{452 - 298}{T_2' - 298}$$



(a) Schematic arrangement of the gas turbine plant



(b) T-s diagram

Fig. 5.61

$\therefore \quad T_2' = 298 + \dfrac{452 - 298}{0.82} = 485.8 \text{ K}$

Turbine : $\quad \dfrac{T_4}{T_5} = \left(\dfrac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}} = (4.3)^{\frac{1.4-1}{1.4}} = 1.517$

$\therefore \quad T_5 = \dfrac{T_4}{1.57} = \dfrac{950}{1.517} = 626.2 \text{ K}$

$$\eta_T = 0.88 = \frac{T_4 - T_5'}{T_4 - T_5} = \frac{950 - T_5'}{950 - 626.2}$$

$\therefore \quad T_{5'} = 950 - 0.88\,(950 - 626.2) = 665 \text{ K}$

Work developed per hour (neglecting fuel mass)

$$= m_a \times c_{pa}\,[(T_4 - T_{5'}) - (T_{2'} - T_1)] \times \eta_{mech.} \times \eta_{gen.} = 12 \times 10^3 \text{ kW}$$

where m_a is the mass of air passed per second.

$$m_a \times 1.005\,[(950 - 665) - (485.8 - 298)] \times 0.94 \times 0.94 = 12 \times 10^3$$

$$86.31\, m_a = 12 \times 10^3$$

$\therefore \quad m_a = 139.03$ kg/s

Heat exchanger :

Effectiveness of heat exchanger,

$$\epsilon = \frac{T_3 - T_{2'}}{T_{5'} - T_{2'}} \quad \text{(neglecting fuel mass)}$$

$\therefore \quad 0.68 = \dfrac{T_3 - 485.8}{665 - 485.8}$

or $\quad T_3 = 485.8 + 0.68\,(665 - 485.8) = 607.6 \text{ K}$

Combustion chamber :

Considering the combustion process in combustion chamber, we have :

Now, $\quad m_f \times C \times \eta_{comb.} \times 0.9 = m_a\, c_{pa}\,(T_4 - T_3)$

where m_f is the mass of fuel burned per sec.

or $\quad m_f \times 41000 \times 0.95 \times 0.9 = 139.03 \times 1.005\,(950 - 607.6)$

$\therefore \quad m_f = 1.365$ kg/s

Cost of fuel $= \dfrac{1.365 \times 3600}{1000} \times 4500 = \text{Rs. } 17690.4\text{/h}$

Total cost per hour $\quad$ = Cost of fuel / hour + All other charges including profit per hour
$= 17690.4 + 3500 = \text{Rs. } 21190.4\text{/h}$

$\therefore$ *Cost of energy generated* $= \dfrac{21190.4}{12 \times 10^3}$ = **Rs. 1.76/kWh. (Ans.)**

Example 5.15. *A gas turbine power plant works on constant pressure open cycle. It consists of compressor, generator, combustion chamber and turbine (the compressor, turbine and generator mounted on the same shaft). The following data is given for this plant :*

The pressure and temperature of air entering into the compressor	*= 1 bar, 25°C*
The pressure of air leaving the compressor	*= 4 bar*
Isentropic efficiency of the compressor	*= 82 per cent*
Isentropic efficiency of the turbine	*= 86 per cent*

Effectiveness of the regenerator	*= 72 per cent*
Pressure loss in regenerator along air side	*= 0.08 bar*
Pressure loss in regenerator along gas side	*= 0.08 bar*
Pressure loss in the combustion chamber	*= 0.04 bar*
Combustion efficiency	*= 92 per cent*
Mechanical efficiency	*= 94 per cent*
Generation efficiency	*= 94 per cent*
Calorific value of fuel used	*= 40000 kJ / kg*
Flow of air	*= 24 kg / s*
Atmospheric pressure	*= 1.03 bar*
The maximum temperature of the cycle	*= 690°C*

Determine the following :

(i) The power available at the generator terminals,

(ii) The overall efficiency of the plant, and

(iii) The specific fuel consumption.

Take $\gamma = 1.4$ *for air and gases.*

$c_{pa} = 1\ kJ/kg\ K;\ c_{pg} = 1.1\ kJ/kg\ K.$

Solution. *Given :* $p_1 = 1$ bar ; $T_1 = 25 + 273 = 298$ K ; $p_2 = p_{2'} = 4$ bar ; $\eta_c = 82\%$; $\eta_T = 86\%$; $\epsilon = 0.72$; $\eta_{comb.} = 92\%$; $\eta_{mech.} = 94\%$; $\eta_{gen.} = 94\%$; $C = 40000$ kJ/kg ; $m_a = 24$ kg/s ; $p_{atm.} = 1.03$ bar ; $T_4 = 690 + 273 = 963$ K.

The schematic arrangement of the plant and its corresponding *T-s* diagram are shown in Fig. 5.62 (*a*) and (*b*) respectively.

Pressure at the inlet to the turbine, $p_4 = 4 - (0.08 + 0.04) = 3.88$ bar

Pressure at exit of the turbine, $p_5 = 1.03 + 0.08 = 1.11$ bar

Compressor :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 1.486$$

$$\therefore \quad T_2 = T_1 \times 1.486 = 298 \times 1.486 = 442.8 \text{ K}$$

$$\eta_{comp.} = 0.82 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{442.8 - 298}{T_{2'} - 298}$$

$$\therefore \quad T_{2'} = 298 + \frac{442.8 - 298}{0.82} = 474.6 \text{ K}$$

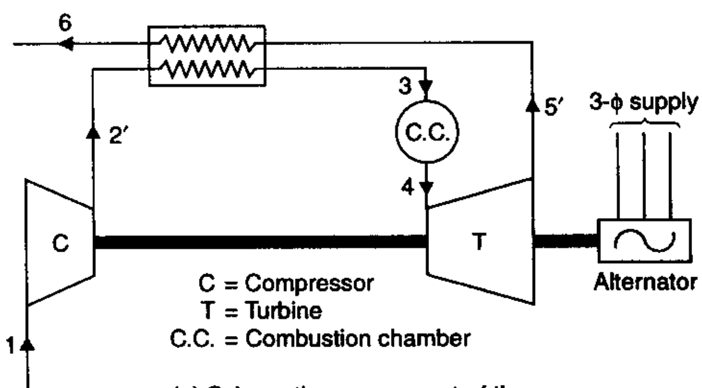
Turbine :

$$\frac{T_4}{T_5} = \left(\frac{p_4}{p_5}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.88}{1.11}\right)^{\frac{1.4-1}{1.4}} = 1.43$$

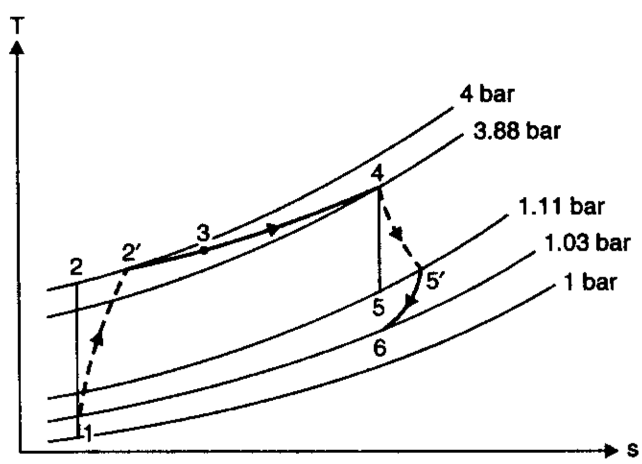
$$\therefore \quad T_5 = \frac{T_4}{1.43} = \frac{963}{1.43} = 673.4 \text{ K}$$

$$\eta_{turbine} = 0.86 = \frac{T_4 - T_{5'}}{T_4 - T_5} = \frac{963 - T_{5'}}{963 - 673.4}$$

$$\therefore \quad T_{5'} = 963 - 0.86\,(963 - 673.4) = 713.9 \text{ K}$$



(a) Schematic arrangement of the gas turbine plant



(b) T-s diagram

Fig. 5.62

Combustion Chamber :

Considering the combustion process in combustion chamber, we have

$$m_f \times C \times \eta_{\text{comb.}} = (m_a + m_f)\, c_{pg}\, (T_4 - T_3)$$

$$\therefore \qquad C \times \eta_{\text{comb.}} = \left(\frac{m_a}{m_f} + 1\right) c_{pg}\, (T_4 - T_3)$$

or

$$40000 \times 0.92 = \left(\frac{m_a}{m_f} + 1\right) \times 1.1\,(963 - T_3)$$

or

$$\frac{m_a}{m_f} = \frac{40000 \times 0.92}{1.1(963 - T_3)} - 1 = \frac{33455}{(963 - T_3)} - 1 \qquad \text{...(1)}$$

Also, effectiveness of the regenerator,

$$\epsilon = \frac{m_a\, c_{pa}\, (T_3 - T_{2'})}{(m_a + m_f)\, c_{pg}\, (T_{5'} - T_{2'})}$$

or $$0.72 = \frac{1(T_3 - 474.6)}{\left(1 + \frac{m_f}{m_a}\right) \times 1.11(713.9 - 474.6)}$$

or $$\frac{m_f}{m_a} = \frac{(T_3 - 474.6)}{0.72 \times 1.11(713.9 - 474.6)} - 1 = \frac{(T_3 - 474.6)}{191.2} - 1 = \frac{T_3 - 665.8}{191.2}$$

or $$\frac{m_a}{m_f} = \frac{191.2}{T_3 - 665.8} \quad ...(2)$$

From eqns. (1) and (2), we get

$$\frac{33455}{963 - T_3} - 1 = \frac{191.2}{T_3 - 665.8} \quad \text{or} \quad \frac{33455 - (963 - T_3)}{963 - T_3} = \frac{191.2}{T_3 - 665.8}$$

$$\frac{32492 + T_3}{963 - T_3} = \frac{191.2}{T_3 - 665.8}$$

$$(32492 + T_3)(T_3 - 665.8) = 191.2(963 - T_3)$$

$$32492\, T_3 - 21633174 + T_3^2 - 665.8\, T_3 = 184126 - 191.2\, T_3$$

$$T_3^2 + 32017\, T_3 - 21817300 = 0$$

$$\therefore \quad T_3 = \frac{-32017 \pm \sqrt{(32017)^2 + 4 \times 21817300}}{2}$$

$$= \frac{-32017 \pm 33352}{2} = 667.5 \text{ K}$$

i.e., $$T_3 = 667.5 \text{ K}$$

and $$\frac{m_a}{m_f} = 112 : 1.$$

(*i*) **The power available at generator terminals :**

$$W_{\text{comp.}} = 1 \times c_{pa}(T_{2'} - T_1) = 1(474.6 - 298) = 176.6 \text{ kJ/kg or air}$$

$$W_{\text{turbine}} = (1 + m_f) \times c_{pg} \times (T_4 - T_{5'})$$

$$= \left(1 + \frac{1}{112}\right) \times 1.1(963 - 713.9) = 276.4 \text{ kJ/kg of air}$$

$$W_{\text{net}} = W_{\text{turbine}} - W_{\text{comp.}}$$

$$= 276.4 - 176.6 = 99.8 \text{ kJ/kg of air}$$

Work available per kg of air at the terminals of generator

$$= 99.8 \times \eta_{\text{mech.}} \times \eta_{\text{gen.}} = 99.8 \times 0.94 \times 0.94 = 88.18 \text{ kJ/kg}$$

Power available at the generator terminal

$$= \frac{24 \times 88.18}{1000} = \mathbf{2.116\ MW.} \quad \textbf{(Ans.)}$$

(*ii*) **Overall efficiency of the plant, η_{overall} :**

$$\eta_{\text{overall}} = \frac{88.18}{\frac{1}{112} \times 40000} \times 100 = \mathbf{24.69\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Specific fuel consumption :**

Fuel required per hour $= (24 \times 3600) \times \frac{1}{112} = 771.43$ kg/h

$\therefore$ Specific fuel consumption $= \dfrac{771.43}{2.116 \times 1000} =$ **0.364 kg/kWh. (Ans.)**

Example 5.16. *The following data relate to a gas turbine plant :*

Power developed	*= 5 MW*
Inlet pressure and temperature of air to the compressor	*= 1 bar, 30°C*
Pressure ratio of the cycle	*= 5*
Isentropic efficiency of the compressor	*= 80 per cent*
Isentropic efficiency of both turbines	*= 85 per cent*
Maximum temperature in both turbines	*= 550°C*

$c_{pa} = 1.0\ kJ/kg\ K\ ;\ c_{pg} = 1.15\ kJ/kg\ K\ ;\ \gamma\,(air) = 1.4\ ;\ \gamma\,(gases) = 1.33.$

If a reheater is used between two turbines at a pressure 2.24 bar, calculate the following :

(i) the mass flow rate of air, and *(ii) the overall efficiency.*

Neglect the mass of the fuel. **(P.U. Winter, 1999)**

Solution. *Given :* Power developed = 5 MW ; $p_1 = 1$ bar ; $T_1 = 30 + 273 = 303$ K ;

$r_p = \dfrac{p_3}{p_1} = 5$ or $p_3 = 5 \times 1 = 5$ bar ; $p_2 = 2.24$ bar ; $T_3 = T_5 = 550 + 273 = 823$ K ; $\eta_{comp.} = 80\%$,

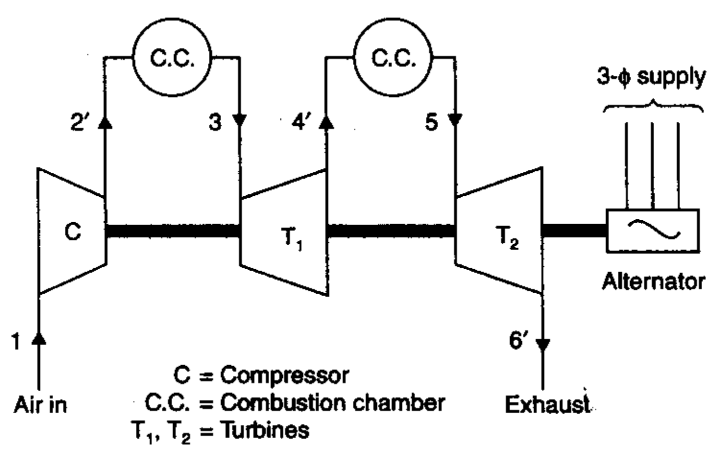
$\eta_{T_1} = \eta_{T_2} = 85\%$; $c_{pa} = 1$ kJ/kg K ; $c_{pg} = 1.15$ kJ/kg K ; γ (air) = 1.4 ; γ (gases) = 1.33.

The schematic arrangement and its corresponding *T-s* diagram are shown in Fig. 5.63 (*a*) and (*b*) respectively.

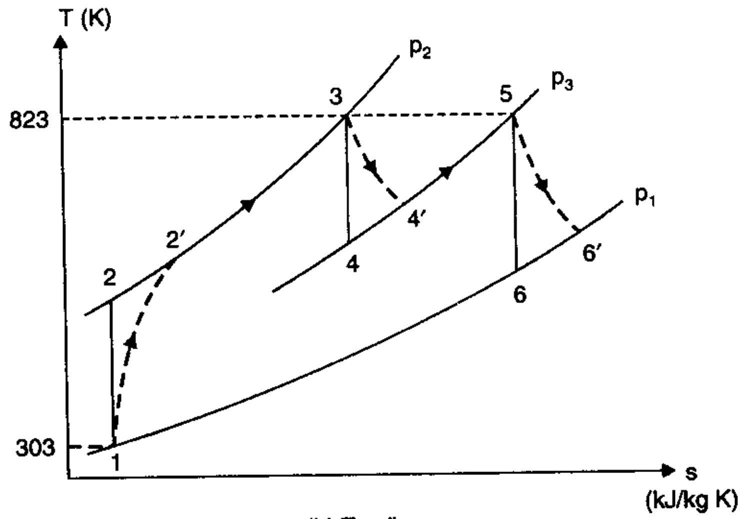
Compressor :
$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (5)^{\frac{1.4-1}{1.4}} = (5)^{0.2857} = 1.584$$

$\therefore$
$$T_2 = T_1 \times 1.584 = 303 \times 1.584 = 480 \text{ K}$$

$$\eta_{comp.} = 0.8 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{480 - 303}{T_{2'} - 303}$$



(a) Flow diagram for the plant



(b) T-s diagram

Fig. 5.63

$$\therefore \quad T_{2'} = 303 + \frac{(480 - 303)}{0.8} = 524.2 \text{ K}$$

$$W_{comp.} = c_{pa}(T_{2'} - T_1) = 1.0(524.2 - 303) = 221.2 \text{ kJ/kg}$$

Turbine, T_1 :

$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{2.24}\right)^{\frac{1.33-1}{1.33}} = \left(\frac{5}{2.24}\right)^{0.2481} = 1.22$$

$$\therefore \quad T_4 = \frac{T_3}{1.22} = \frac{823}{1.22} = 674.6 \text{ K}$$

$$\eta_{T_1} = 0.85 = \frac{T_3 - T_{4'}}{T_3 - T_4} = \frac{823 - T_{4'}}{823 - 674.6}$$

$$\therefore \quad T_{4'} = 823 - 0.85(823 - 674.6) = 696.9 \text{ K}$$

Turbine, T_2 :

$$\frac{T_5}{T_6} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2.24}{1}\right)^{\frac{1.33-1}{1.33}} = (2.24)^{0.2481} = 1.22$$

$$\therefore \quad T_6 = \frac{T_5}{1.22} = \frac{823}{1.22} = 674.6 \text{ K}$$

$$\eta_{T_2} = 0.85 = \frac{T_5 - T_{6'}}{T_5 - T_6} = \frac{823 - T_{6'}}{823 - 674.6}$$

$$\therefore \quad T_{6'} = 823 - 0.85\,(823 - 674.6) = 696.9 \text{ K}$$

$$\therefore \quad (W_{turbine})_{total} = 2\,c_{pg}\,(T_3 - T_{4'}) \quad \text{as} \quad T_3 = T_5 \text{ and } T_{4'} = T_{6'}$$

$$= 2 \times 1.15\,(823 - 696.9) = 290 \text{ kJ/kg}$$

$$\therefore \quad W_{net} = (W_{turbine})_{total} - W_{comp.}$$

$$= 290 - 221.2 = 68.8 \text{ kJ/kg}$$

(*i*) **The mass flow rate of air, $\dot{m}_a$:**

Power developed $= \dot{m}_a \times W_{net}$

$$5 \times 10^3 = \dot{m}_a \times 68.8$$

$$\therefore \quad \dot{m}_a = \frac{5 \times 10^3}{68.8} = \mathbf{72.67\ kg/s.} \quad \textbf{(Ans.)}$$

(*ii*) **Overall efficiency, $\eta_{overall}$:**

Heat supplied, $Q_s = c_{pg}(T_3 - T_{2'}) + c_{pg}(T_5 - T_{4'})$

$$= 1.15\,[(823 - 524.2) + (823 - 696.9)] = 488.6 \text{ kJ/kg}$$

$$\therefore \quad \eta_{overall} = \frac{W_{net}}{\text{Heat supplied } (Q_s)}$$

$$= \frac{68.8}{488.6} = 0.1408 \quad \text{or} \quad \mathbf{14.08\%.} \quad \textbf{(Ans.)}$$

Example 5.17. *The following data refer to a gas turbine plant in which the compression is carried out in one stage and expansion is carried out in two stages with reheating to the original temperature.*

Capacity of the gas turbine plant	*= 6 MW*
Temperature at which air is supplied	*= 20°C*
Suction and exhaust pressure	*= 1 bar*
Pressure ratio	*= 6*
Maximum temperature limit	*= 750°C*
Isentropic efficiency of compressor	*= 80 per cent*
Isentropic efficiency of each turbine	*= 84 per cent*
Effectiveness of heat exchanger	*= 0.72*
Calorific value of fuel	*= 18500 kJ/kg*

$c_{pa} = 1$ *kJ/kgK* ; $c_{pg} = 1.15$ *kJ/kg K* ; γ *(air)* $= 1.4$, γ*(gas)* $= 1.33$.

Determine the following :

(*i*) *A/F ratio entering in the first turbine, and*

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

(*iii*) *Air supplied to the plant,*

(*iv*) *Fuel consumption of the plant per hour.*

Solution. *Given :* Capacity of the plant = 6 MW ; $T_1 = 20 + 273 = 293$ K ; $p_1 = 1$ bar ;

$$r_p = \frac{p_3}{p_1} = 6 \text{ or } p_3 = 6 \text{ bar} ; \ T_4 = T_5 = 750 + 273 = 1023 \text{ K} ;$$

$$\eta_{comp.} = 80\%, \ \eta_{T_1} = \eta_{T_2} = 84\% ; \ \epsilon = 0.72, \ C = 18500 \text{ kJ/kg} ;$$

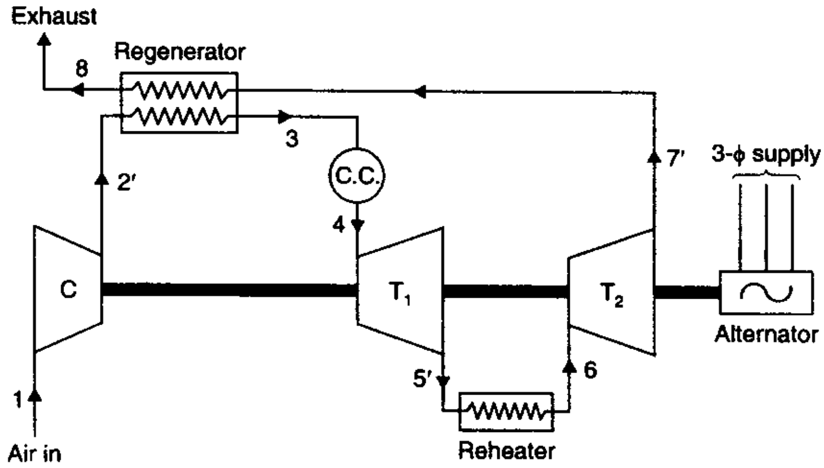
$$c_{pa} = 1 \text{ kJ/kg K} ; \ c_{pg} = 1.15 \text{ kJ/kg K} ; \ \gamma(\text{air}) = 1.4 ; \ \gamma(\text{gas}) = 1.33.$$

The schematic arrangement of the gas turbine plant and its corresponding *T-s* diagram are shown in Fig. 5.64 (*a*) and (*b*) respectively.

$$p_2 = \sqrt{p_3 \times p_1} = \sqrt{6 \times 1} = 2.45 \text{ bar}$$

Compressor :

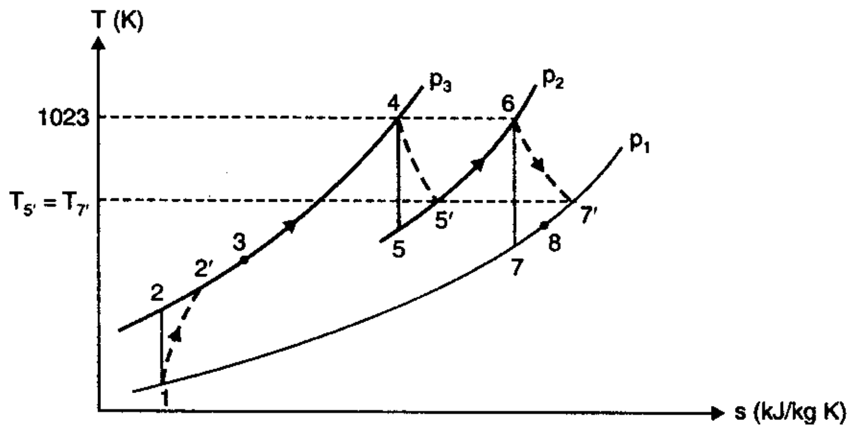
$$\frac{T_2}{T_1} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} = (6)^{\frac{1.4 - 1}{1.4}} = (6)^{0.2857} = 1.668$$



C = Compressor T₁, T₂ = Turbines

C.C. = Combustion chamber

(a) Schematic arrangement of the gas turbine plant



(b) T-s diagram

Fig. 5.64

$$\therefore \quad T_2 = T_1 \times 1.668 = 293 \times 1.668 = 488.7 \text{ K}$$

$$\eta_{comp.} = 0.8 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{488.7 - 293}{T_{2'} - 293}$$

$$\therefore \quad T_{2'} = 293 + \frac{488.7 - 293}{0.8} = 537.6 \text{ K}$$

Turbine-1

$$\frac{T_4}{T_5} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma - 1}{\gamma}} = \left(\frac{6}{2.45}\right)^{\frac{1.33 - 1}{1.33}} = (2.45)^{0.2481} = 1.249$$

$$\therefore \quad T_5 = \frac{T_4}{1.249} = \frac{1023}{1.249} = 819 \text{ K}$$

$$\eta_{T_1} = 0.84 = \frac{T_4 - T_{5'}}{T_4 - T_5} = \frac{1023 - T_{5'}}{1023 - 819}$$

$\therefore \qquad T_{5'} = 1023 - 0.84\,(1023 - 819) = 851.6\text{ K}$

$\eta_{T_1} = \eta_{T_2}$ (Given), $\therefore\ T_{5'} = T_{7'} = 851.6\text{ K}$

Regenerator :

Effectiveness, $\qquad \epsilon = 0.72 = \dfrac{T_3 - T_{2'}}{T_{7'} - T_{2'}} = \dfrac{T_3 - 537.6}{851.6 - 537.6}$

$\therefore \qquad T_3 = 537.6 + 0.72\,(851.6 - 537.6) = 763.7\text{ K}$

(*i*) **A/F ratio entering in the first turbine :**

Let m_{f_1} = Mass of fuel supplied per kg of air to the combustion chamber (and hence to the first turbine)

$$\therefore \qquad (1 + m_{f_1})\, c_{pg}\,(T_4 - T_3) = m_{f_1} \times C$$

or $$(1 + m_{f_1}) \times 1.15\,(1023 - 763.7) = m_{f_1} \times 18500$$

or $$298.2\,(1 + m_{f_1}) = 18500\, m_{f_1}$$

$$\therefore \qquad m_{f_1} = \frac{298.2}{(18500 - 298.2)} = 0.01638 \text{ kg/kg of air}$$

$$\therefore \qquad A/F\ ratio = \frac{1}{0.01638} = \mathbf{61 : 1.} \quad \textbf{(Ans.)}$$

(*ii*) **Thermal efficiency of the cycle, $\eta_{thermal}$:**

Considering the flow with burning through the reheater, we have

$$(1 + m_{f_1} + m_{f_2})\, c_{pg}\,(T_6 - T_{5'}) = m_{f_2} \times C$$

where m_{f_2} is the fuel supplied in the reheater per kg of air entering into the compressor.

$$\therefore \qquad (1 + 0.01638 + m_{f_2}) \times 1.15\,(1023 - 851.6) = m_{f_2} \times 18500$$

or $$197.11(1.01638 + m_{f_2}) = 18500\, m_{f_2}$$

or $$200.34 + 197.11\, m_{f_2} = 18500\, m_{f_2}$$

$$\therefore \qquad m_{f_2} = \frac{200.34}{(18500 - 197.11)} = 0.0109 \text{ kg/kg of air}$$

$$\therefore \qquad W_{comp.} = 1 \times c_{pa}(T_{2'} - T_1) = 1 \times 1 \times (537.6 - 293) = 244.6 \text{ kJ/kg of air}$$

$$\begin{aligned} W_{Turbine\,(total)} &= W_{T_1} + W_{T_2} \\ &= (1 + m_{f_1})\, c_{pg}\,(T_4 - T_{5'}) + (1 + m_{f_1} + m_{f_2})\, c_{pg}\,(T_6 - T_{7'}) \\ &= (1 + 0.01638) \times 1.15(1023 - 851.6) + (1 + 0.01638 + 0.0109) \\ &\qquad \times 1.15(1023 - 851.6) \\ &= 1.15(1023 - 851.6)\,[(1 + 0.01638) + (1 + 0.01638 + 0.0109)] \\ &= 402.8 \text{ kJ/kg of air.} \end{aligned}$$

$$\begin{aligned} W_{net} &= W_{turbine\,(total)} - W_{comp.} \\ &= 402.8 - 244.6 = 158.2 \text{ kJ/kg of air} \end{aligned}$$

$$\therefore \qquad \eta_{thermal} = \frac{W_{net}}{\text{Heat supplied } (Q_s)} = \frac{W_{net}}{(m_{f_1} + m_{f_2}) \times C}$$

$$= \frac{158.2}{(0.01638 + 0.0109) \times 18500} = 0.3135 \text{ or } \mathbf{31.35\%.} \quad \textbf{(Ans.)}$$

(*iii*) **Air supplied to the plant, m_a :**

$$m_a \times W_{net} = 6 \times 1000$$

$$\therefore \qquad m_a = \frac{6000}{W_{net}} = \frac{6000}{158.2} = \mathbf{37.93 \text{ kg/s.}} \quad \textbf{(Ans.)}$$

(*iv*) **Fuel consumption of the plant per hour :**

Fuel consumption of the plant per hour

$$= m_a(m_{f_1} + m_{f_2}) \times 3600$$

$$= 37.93(0.01638 + 0.0109) \times 3600 = \mathbf{3725\ kg/h.} \quad \textbf{(Ans.)}$$

Example 5.18. *An open cycle constant pressure gas turbine power plant of 1500 kW capacity comprises a single stage compressor and two turbines with regenerator. One turbine is used to run the compressor and other one runs the generator. Separate combustion chamber is used for each turbine. Air coming out from the regenerator is divided into two streams, one goes to compressor turbine and other to power turbine. The pressure and temperature of air entering the compressor are 1 bar and 25°C. The maximum temperature in the compressor turbine is 750°C and in power turbine is 800°C. The maximum pressure in the stream is 5 bar. The exhaust pressure of both turbine is 1 bar and both exhausts pass through the regenerator. The temperature of exhaust entering the regenerator is 475°C.*

Use the following data :

Isentropic efficiency of compressor ($\eta_{comp.}$)	= *82 per cent*
Isentropic efficiency of compressor turbine (η_{T_1})	= *84 per cent*
Isentropic efficiency of power turbine (η_{T_2})	= *89 per cent*
Calorific value of fuel	= *40500 kJ/kg*
Combustion efficiency ($\eta_{comb.}$) *in both the combustion chambers*	= *94 per cent*
Mechanical efficiency ($\eta_{mech.}$) *for both the turbines*	= *89 per cent*
Effectiveness of regenerator ($\in$)	= *0.72*

$c_{pa} = 1.005$ *kJ/kg K* ; $c_{pg} = 1.1$ *kJ/kg K*; γ*(for air)* = *1.4* ; γ*(for gases)* = *1.35*

Neglecting pressure losses, heat losses and mass of fuel, determine :

(*i*) *Plant efficiency,* (*ii*) *Specific fuel consumption, and*

(*iii*) *Air fuel ratio.*

Solution. *Given* : Capacity of the plant = 1500 kW ; $p_1 = 1$ bar ; $T_1 = 25 + 273 = 298$ K ;

$T_4 = 750 + 273 = 1023$ K ; $T_6 = 800 + 273 = 1073$ K ; $p_1 = 1$ bar ; $p_2 = 5$ bar ;

$T_8 = 475 + 273 = 748$ K ; $\eta_{comp.} = 82\%$; $\eta_{T_1} = 84\ \%$; $\eta_{T_2} = 89\%$;

$C = 40500$ kJ/kg ; $\eta_{comb.} = 94\%$; $\eta_{mech} = 89\%$, $\in = 0.72$ $c_{pa} = 1.005$ kJ/kg K ;

$c_{pg} = 1.1$ kJ/kg K ; γ (for air) = 1.4 ; γ (for gases) = 1.35.

The schematic arrangement of the plant and its corresponding *T-s* diagram are shown in Fig. 5.65 (*a*) and (*b*) respectively.

Compressor :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = (5)^{0.2857} = 1.584$$

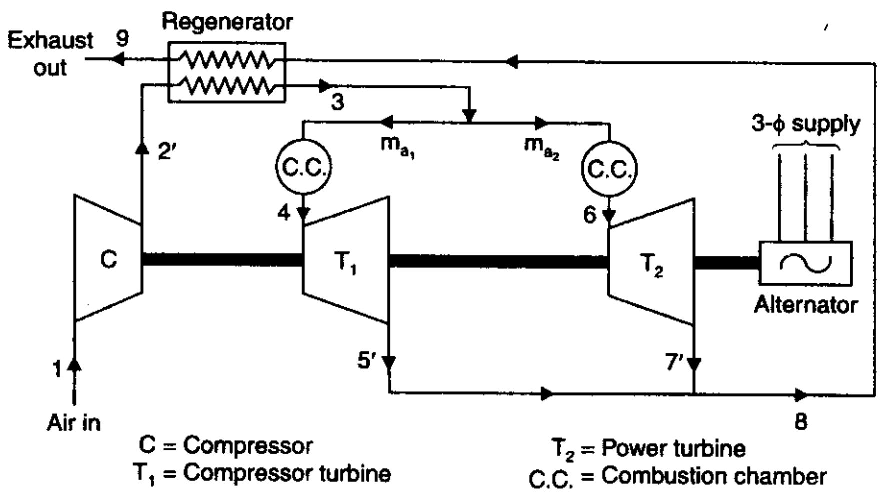
$$\therefore \quad T_2 = T_1 \times 1.584 = 298 \times 1.584 = 472 \text{ K}$$

$$\eta_{comp.} = 0.82 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{472 - 298}{T_{2'} - 298}$$

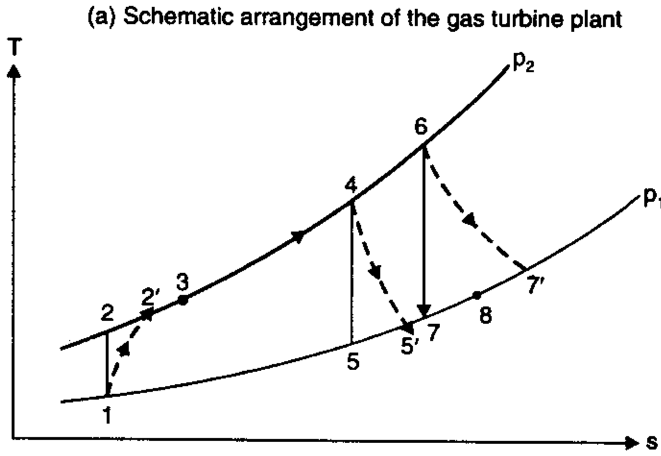
$$\therefore \quad T_{2'} = 298 + \frac{472 - 298}{0.82} = 510.2 \text{ K}$$

Compressor turbine :

$$\frac{T_4}{T_5} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (5)^{\frac{1.35-1}{1.35}} = (5)^{0.2592} = 1.518$$



(a) Schematic arrangement of the gas turbine plant



(b) T-s diagram

Fig. 5.65

$$\therefore \quad T_5 = \frac{T_4}{1.518} = \frac{1023}{1.518} = 673.9 \text{ K}$$

$$\eta_{T_1} = 0.84 = \frac{T_4 - T_{5'}}{T_4 - T_5} = \frac{1023 - T_{5'}}{1023 - 673.9}$$

$$\therefore \quad T_{5'} = 1023 - 0.84(1023 - 673.9) = 729.7 \text{ K}$$

Power Turbine :

$$\frac{T_6}{T_7} = \left(\frac{p_2}{p_1}\right)^{\frac{1.35-1}{1.35}} = (5)^{0.2592} = 1.518$$

$$\therefore \quad T_7 = \frac{T_6}{1.518} = \frac{1073}{1.518} = 706.8 \text{ K}$$

$$\eta_{T_2} = 0.89 = \frac{T_6 - T_{7'}}{T_6 - T_7} = \frac{1073 - T_{7'}}{1073 - 706.8}$$

$\therefore \quad T_{7'} = 1073 - 0.89(1073 - 706.8) = 747.1$ K

Let, $\quad m_{a_1}$ = Mass of air passing through the compressor turbine, and

m_{a_2} = Mass of air passing through the power turbine.

Then, power output of the power turbine is given by :

$$m_{a_2} \times c_{pg} (T_6 - T_{7'}) \times \eta_{\text{mech.}} \times \eta_{\text{gen.}} = 1500$$

or
$$m_{a_2} \times 1.1 (1073 - 747.1) \times 0.89 \times 1 = 1500$$

$$\therefore \quad m_{a_2} = 4.7 \text{ kg/s}$$

Power developed by the compressor turbine = Power absorbed by the compressor

$$\therefore \quad m_{a_1} \times c_{pg}(T_4 - T_5') \times \eta_{\text{mech.}} = (m_{a_1} + m_{a_2}) \times c_{pa}(T_2' - T_1)$$

or
$$m_{a_1} \times 1.1(1023 - 729.7) \times 0.89 = (m_{a_1} + 4.7) \times 1.005(510.2 - 298)$$

$$287.1\, m_{a_1} = (m_{a_1} + 4.7) \times 213.26$$

$$\therefore \quad m_{a_1} = \frac{4.7 \times 213.26}{(287.1 - 213.26)} = 13.57 \text{ kg/s}$$

The exhaust gases of both turbines are mixed before entering into the regenerator. Therefore, the temperature of the gases entering into the regenerator is designated by the point 8 on the T-s diagram and it is given by

$$m_{a_1} \times c_{pg} \times T_5' + m_{a_2} \times c_{pg} \times T_{7'} = (m_{a_1} + m_{a_2}) \times c_{pg} \times T_8$$

where, T_8 is the temperature after mixing.

$$\therefore \quad T_8 = \left(\frac{m_{a_1}}{m_{a_1} + m_{a_2}}\right) \times T_{5'} + \left(\frac{m_{a_2}}{m_{a_1} + m_{a_2}}\right) \times T_{7'}$$

$$= \left(\frac{13.57}{13.57 + 4.7}\right) \times 729.7 + \left(\frac{4.7}{13.57 + 4.7}\right) \times 747.1$$

$$= 541.98 + 192.19 = 734.2 \text{ K}$$

Regenerator :

Effectiveness, $\quad \epsilon = 0.72 = \dfrac{m_a\, c_{pa}\, (T_3 - T_{2'})}{m_g\, c_{pg}\, (T_8 - T_{2'})}$

or
$$0.72 = \frac{c_{pa}\, (T_3 - T_{2'})}{c_{pg}\, (T_8 - T_{2'})} \qquad (\because \; m_a = m_g, \text{ as fuel mass is neglected})$$

$$= \frac{1.005\, (T_3 - 510.2)}{1.1\, (748 - 510.2)}$$

or
$$T_3 = 510.2 + \frac{0.72 \times 1.1\, (748 - 510.2)}{1.005} = 697.6 \text{ K}$$

Combustion chambers :

Total heat supplied in both combustion chambers

$$= c_{pg}\, m_{a_1}(T_4 - T_3) + c_{pg}\, m_{a_2}(T_6 - T_3) = m_f \times C \times \eta_{\text{comb.}}$$

or
$$1.1\, [13.57(1023 - 697.6) + 4.7(1073 - 697.6)] = m_f \times 40500 \times 0.94$$

$$1.1(4415.68 + 1764.38) = 38070 \times m_f$$

$$\therefore \quad m_f = 0.178 \text{ kg/s}$$

(*i*) **Plant efficiency :**

$$\text{Plant efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{1500}{0.178 \times 40500} = 0.208 \text{ or } \mathbf{20.8\%.} \;\; \textbf{(Ans.)}$$

(*ii*) **Specific fuel consumption :**

$$\text{Specific fuel consumption} = \frac{0.178 \times 3600}{1500} = \mathbf{0.427\ kg/kWh.} \quad \textbf{(Ans.)}$$

(*iii*) **Air-fuel (A/F) ratio :**

$$\text{A/F ratio} = \frac{m_{a_1} + m_{a_2}}{m_f} = \frac{13.57 + 4.7}{0.178} = \mathbf{102.64.} \quad \textbf{(Ans.)}$$

Example 5.19. *The air supplied to a gas turbine plant is 10 kg/s. The pressure ratio is 6 and pressure at the inlet of the compressor is 1 bar. The compressor is two-stage and provided with perfect intercooler. The inlet temperature is 300 K and maximum temperature is limited to 1073 K.*

Take the following data :

Isentropic efficiency of compressor each stage ($\eta_{comp.}$) = *80%*

Isentropic efficiency of turbine ($\eta_{turbine}$) = *85%*

A regenerator is included in plant whose effectiveness is 0.7. Neglecting the mass of fuel, determine the thermal efficiency of the plant.

Take c_p *for air = 1.005 kJ/kg K.* **(P.U. Summer, 1997)**

Solution. *Given* : $\dot{m}_a = 10$ kg/s ; $r_p = 6$, $p_1 = 1$ bar ; $T_1 = 300$ K ; $T_6 = 1073$ K ;

$\eta_{comp.} = 80\%$; $\eta_{turbine} = 85\%$; $\epsilon = 0.7$.

The schematic arrangement of the gas turbine plant and its corresponding *T-s* diagram are shown in Fig. 5.66(*a*) and (*b*) respectively. As the cooling is perfect,

$$p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 6} = 2.45 \text{ bar} \qquad \left[\because r_p = 6 = \frac{p_3}{p_1} = \frac{p_3}{1} \text{ or } p_3 = 6 \text{ bar}\right]$$

Considering *isentropic compression in L.P. compressor*, we have :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{2.45}{1}\right)^{\frac{1.4-1}{1.4}} = 1.29$$

$$\therefore \quad T_2 = T_1 \times 1.29 = 300 \times 1.29 = 387 \text{ K}$$

Also,
$$\eta_{comp.(L.P.)} = 0.8 = \frac{T_2 - T_1}{T_2' - T_1} = \frac{387 - 300}{T_2' - 300}$$

$$\therefore \quad T_{2'} = 300 + \frac{387 - 300}{0.8} = 408.7 \text{ K}$$

$$\therefore \quad W_{comp.(L.P.)} = 1 \times c_{pa}(T_2' - T_1) \text{ per kg of air}$$

$$= 1 \times 1.005(408.7 - 300) = 109.2 \text{ kJ/kg}$$

Since *intercooling is perfect*, therefore,

$$W_{comp.(L.P.)} = W_{comp.\ (H.P.)}$$

$\therefore$ Total compressor work,

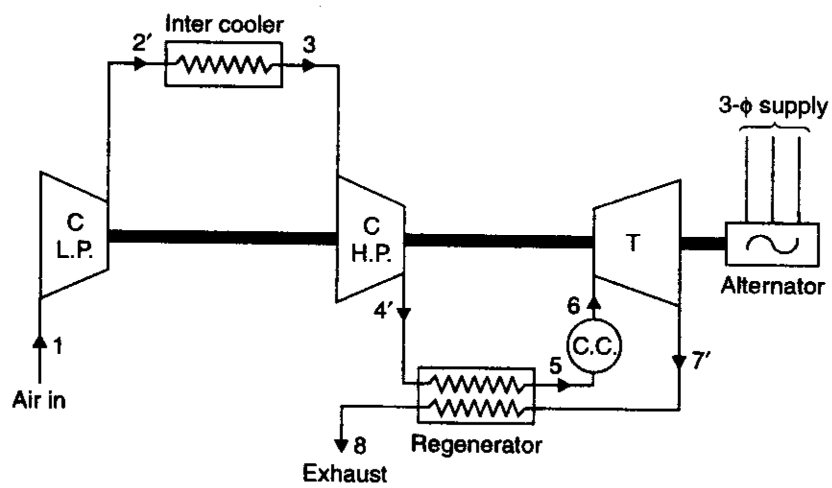
$$W_{comp.\ (total)} = 2 \times 109.2 = 218.4 \text{ kJ/kg}$$

As
$$T_3 = T_1 \text{ and } r_p = \frac{p_2}{p_1} = \frac{p_3}{p_2} \quad \text{(for perfect intercooling)}$$

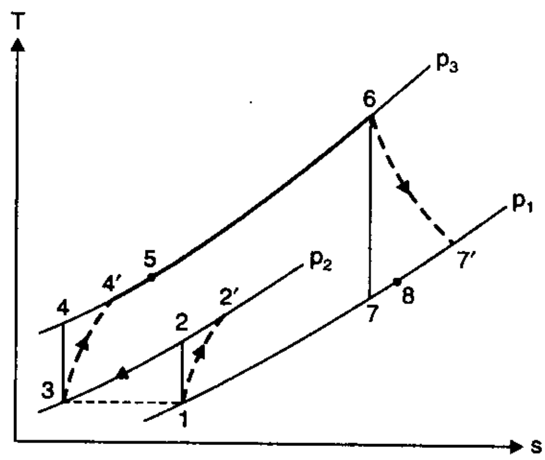
$$\therefore \quad T_{4'} = T_{2'} = 408.7 \text{ K}$$

Considering *isentropic expansion in the turbine*, we have

$$\frac{T_6}{T_7} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (6)^{\frac{1.4-1}{1.4}} = 1.668$$



(a) Schematic arrangement of the gas turbine plant



(b) T-s diagram

Fig. 5.66

$$\therefore \quad T_7 = \frac{T_6}{1.668} = \frac{1073}{1.668} = 643.3 \text{ K}$$

Also,

$$\eta_{turbine} = 0.85 = \frac{T_6 - T_{7'}}{T_6 - T_7} = \frac{1073 - T_{7'}}{1073 - 643.3}$$

or

$$T_{7'} = 1073 - 0.85(1073 - 643.3) = 707.7 \text{ K}$$

$$\therefore \quad W_{turbine} = c_{pa}(T_6 - T_{7'}) = 1.005(1073 - 707.7) = 367.1 \text{ kJ/kg}$$

The effectiveness (ϵ) of *regenerator* is given by,

$$\epsilon = 0.7 = \frac{T_5 - T_{4'}}{T_{7'} - T_{4'}} = \frac{T_5 - 408.7}{707 - 408.7}$$

$$T_5 = 408.7 + 0.7(707 - 408.7) = 617.5 \text{ K}$$

Heat supplied in the combustion chamber,

$$Q_s = 1 \times c_{pa}(T_6 - T_5) = 1 \times 1.005(1073 - 617.5) = 457.77 \text{ kJ/kg}$$

$$W_{net} = W_{turbine} - W_{comp.} = 367.1 - 218.4 = 148.7 \text{ kJ/kg}$$

$$\therefore \quad \eta_{thermal} = \frac{W_{net}}{Q_s} = \frac{148.7}{457.77} = 0.3248 \text{ or } \mathbf{32.48\%.} \ \mathbf{(Ans.)}$$

Example 5.20. *A gas turbine plant consists of one turbine as compressor drive and other to drive a generator. Each turbine has its own combustion chamber getting air directly from the compressor. Air enters the compressor at 1 bar 15°C and compressed with isentropic efficiency of 76 percent. The gas inlet pressure and temperature in both the turbines are 5 bar and 680°C respectively. Take isentropic efficiency of both the turbines as 86 percent. The mass flow rate of air entering compressor is 23 kg/s. The calorific value of the fuel is 42000 kJ/kg. Determine the, power output and thermal efficiency of the plant.*

Take, $c_{pa} = 1.005$ *kJ/kg K, and* $\gamma_{air} = 1.4$;

$c_{pg} = 1.128$ *kJ/kg K, and* $\gamma_{gas} = 1.34$ **(P.U. June, 1998]**

Solution. *Given* : $p_1 = 1$ bar, $T_1 = 15 + 273 = 288$ K ; $p_2 = 5$ bar,

$$T_3 = T_5 = 680 + 273 = 953 \text{ K} ; \ \eta_{comp.} = 76\% ; \ \eta_{T_1} = \eta_{T_2} = 86\% ;$$

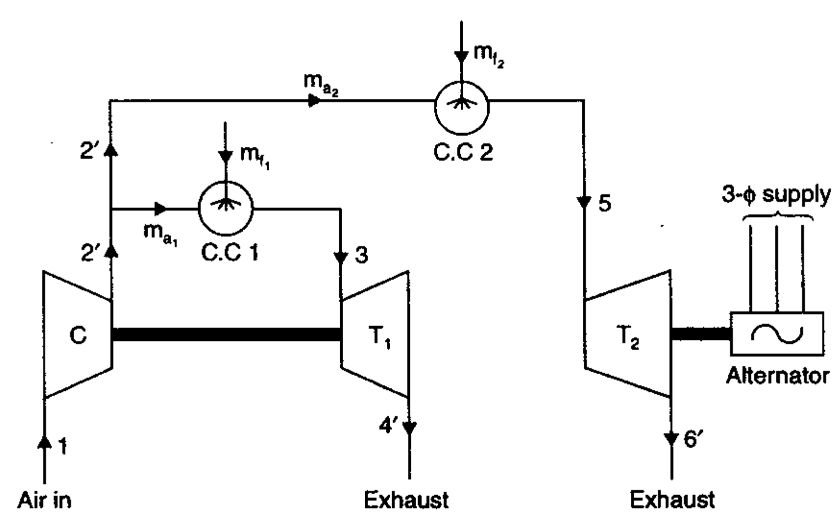
$$m_a \ (= m_{a_1} + m_{a_2}) = 23 \text{ kg/s} ; C = 42000 \text{ kJ/kg} ; c_{pa} = 1.005 \text{ kJ/kg K} ;$$

$$c_{pg} = 1.128 \text{ kJ/kg K} ; \gamma_{air} = 1.4 ; \gamma_{gas} = 1.34.$$

The schematic arrangement of the plant and its corresponding T-s diagrams (separately for $C - T_1$ and $C - T_2$) are shown in Fig. 5.67 (*a*) and (*b*) respectively.

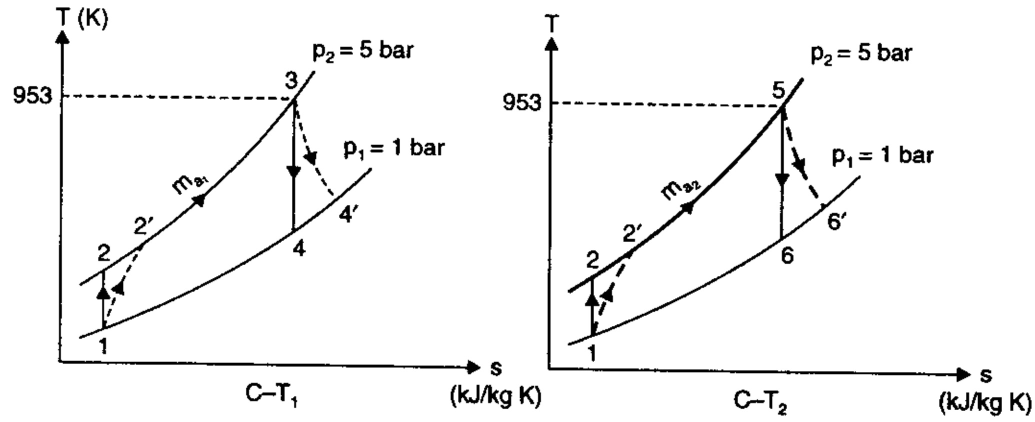
Compressor-Turbine-1 (C – T₁) :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (5)^{\frac{1.4-1}{1.4}} = (5)^{0.2857} = 1.584$$



C = Compressor
T_1, T_2 = Turbines
C.C – 1, C.C – 2 = Combustion chambers

(a) Schematic arrangement of the plant



(b) T-s diagrams

Fig. 5.67

$$\therefore \quad T_2 = T_1 \times 1.584 = 288 \times 1.584 = 456.2 \text{ K}$$

$$\eta_{comp.} = 0.76 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{456.2 - 288}{T_{2'} - 288}$$

$$\therefore \quad T_2' = 288 + \frac{456.2 - 288}{0.76} = 509.3 \text{ K}$$

Let m_{f_1} = Fuel supplied in $C.\,C - 1$

Then, $m_{f_1} \times C = (m_{a_1} + m_{f_1}) \times c_{pg}(T_3 - T_{2'})$

or
$$C = \left(\frac{m_{a_1}}{m_{f_1}} + 1 \right) \times c_{pg}(T_3 - T_{2'})$$

or
$$42000 = \left(\frac{m_{a_1}}{m_{f_1}} + 1 \right) \times 1.128(953 - 509.3)$$

$$\therefore \quad \frac{m_{a_1}}{m_{f_1}} = \frac{42000}{1128\,(953 - 509.3)} - 1 = 82.92$$

Also,
$$\frac{T_3}{T_4} = \left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} = (5)^{\frac{1.34 - 1}{1.34}} = (5)^{0.254} = 1.5$$

$$\therefore \quad T_4 = \frac{T_3}{1.5} = \frac{953}{1.5} = 635.3 \text{ K}$$

$$\eta_{T_1} = 0.86 = \frac{T_3 - T_{4'}}{T_3 - T_4} = \frac{953 - T_{4'}}{953 - 635.3}$$

$$\therefore \quad T_{4'} = 953 - 0.86(953 - 635.3) = 679.8 \text{ K}$$

Now, the work required to run the compressor must be *equal to* the work developed by T_1.

$$\therefore \quad m_a \times c_{pa}(T_{2'} - T_1) = (m_{a_1} + m_{f_1}) \times c_{pg}(T_3 - T_{4'})$$

$$= m_{f_1} \left(\frac{m_{a_1}}{m_{f_1}} + 1 \right) \times c_{pg}(T_3 - T_{4'})$$

or $\quad 23 \times 1.005(509.3 - 288) = m_{f_1}\ (82.92 + 1) \times 1.128(953 - 679.8)$

or $\quad 5115.35 = 25861.6\ m_{f_1}$

$$\therefore \quad m_{f_1} = \frac{5115.35}{25861.6} = 0.198 \text{ kg/s}$$

$$\therefore \quad m_{a_1} = 82.92 \times 0.198 = 16.42 \text{ kg/s}$$

and $\quad m_{a_2} = m_a - m_{a_1} = 23 - 16.42 = 6.58 \text{ kg/s}$

Compressor-turbine-2 ($C - T_2$) :

Work developed by the turbine T_2,

$$W_{T_2} = (m_{a_2} + m_{f_2}) \times c_{pg}(T_5 - T_{6'}) \quad ...(i)$$

Considering the combustion chamber of turbine T_2, we can write

$$m_{f_2} \times C = (m_{a_2} + m_{f_2}) \times c_{pg}(T_5 - T_{2'})$$

or $\quad m_{f_2} \times 42000 = (6.58 + m_{f_2}) \times 1.128(953 - 509.3)$

$$= 500.49(6.58 + m_{f_2}) = 3293.22 + 500.49\ m_{f_2}$$

$$\therefore \quad m_{f_2} = \frac{3293.22}{(42000 - 500.49)} = 0.079 \text{ kg/s} \quad \left(\text{or } \frac{m_{a_2}}{m_{f_2}} = \frac{6.58}{0.079} = 83.3\right)$$

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

$$W_{T_2} = (6.58 + 0.079) \times 1.128(953 - 679.8)$$

($\because$ $T_{6'} = T_{4'}$ as per given conditions)

$$= \mathbf{2052.1\ kJ/s.} \quad \textbf{(Ans.)}$$

The capacity of turbine T_2 to run the compressor,

$$W_{T_1} = m_{a_1} \times c_{pa}(T_{2'} - T_1)$$

$$= 16.42 \times 1.005(509.3 - 288) = 3651.9 \text{ kJ/s}$$

Total fuel consumed, $m_f = m_{f_1} + m_{f_2}$

$$= 0.198 + 0.079 = 0.277 \text{ kg/s}$$

$\therefore$ Thermal efficiency of the plant,

$$\eta_{\text{thermal}} = \frac{W_{T_2}}{m_f \times C}$$

$$= \frac{2052.1}{0.277 \times 42000} = 0.1764 \quad \text{or} \quad \mathbf{17.64\%.} \quad \textbf{(Ans.)}$$

Example 5.21. *A gas turbine plant consists of two-stage compressor with intercooler and it is driven by a separate turbine. The gases coming out from first turbine are passed to the power turbine after reheating to the temperature which is equal to the temperature at the inlet of the compressor turbine. The power turbine generates the electrical energy. A regenerator is used for heating the air before entering into the combustion chamber by using the exhaust gases coming out of power turbine.*

Use the following data :

Ambient air temperature and pressure	= *15°C, 1 bar*
Maximum cycle temperature	= *1000 K*
Air mass flow	= *20 kg/s*
Isentropic efficiency of each compressor	= *80 per cent*
Insentropic efficiencies of turbines :	η_{T_1} = *87%* ; η_{T_2} = *80%*
Pressure drop in intercooler	= *0.07 bar*

Pressure drop in regenerator (H.E.)	*= 0.1 bar in each side*
Effectiveness of heat exchanger	*= 0.75*
Pressure drop in combustion chamber	*= 0.15 bar*
Pressure drop in reheater	*= 0.1 bar*
Mechanical efficiency for compressor-turbine	*= 99%*
Combustion efficiency (in combustion chamber and reheater)	*= 98%*
Compression ratio of each stage of compressor	*= 2 : 1*
Calorific value of fuel used	*= 43500 kJ/kg*

$c_{pa} = 1$ *kJ/kg K* ; $c_{pg} = 1.1$ *kJ/kg K*, $\gamma_{air} = 1.4$; $\gamma_{gas} = 1.33$.

Assuming perfect intercooling, determine :

(*i*) *Net output of plant,* (*ii*) *Specific fuel consumption of the plant, and*

(*iii*) *Overall efficiency of the plant.* **(K.U., 1999)**

Solution. *Given* : $T_1 = 15 + 273 = 288$ K ; $p_1 = 1$ bar ; $T_6 = T_8 = 1000$ K ; $m_a = 20$ kg/s ;

$$\eta_{C_1} = \eta_{C_2} = 80\% \; ; \; \eta_{T_1} = 87\% \; ; \; \eta_{T_2} = 80\% \; ;$$

$$(\Delta p)_{\text{intercooler}} = 0.07 \text{ bar} \; ; \; (\Delta p)_{\text{regenerator}} = 0.1 \text{ bar in each side} \; ; \; \epsilon = 0.75 \; ;$$

$$(\Delta p)_{C.C.} = 0.15 \text{ bar} \; ; \; (\Delta p)_{\text{reheater}} = 0.1 \text{ bar} \; ; \; \eta_{\text{mech}(T_1)} = 99\% \; ;$$

$$\eta_{\text{comb. (C.C. and reheater)}} = 98\% \; ; \; (r_p)_{C_1, C_2} = 2 : 1 \; ; \; C = 43500 \text{ kJ/kg} \; ;$$

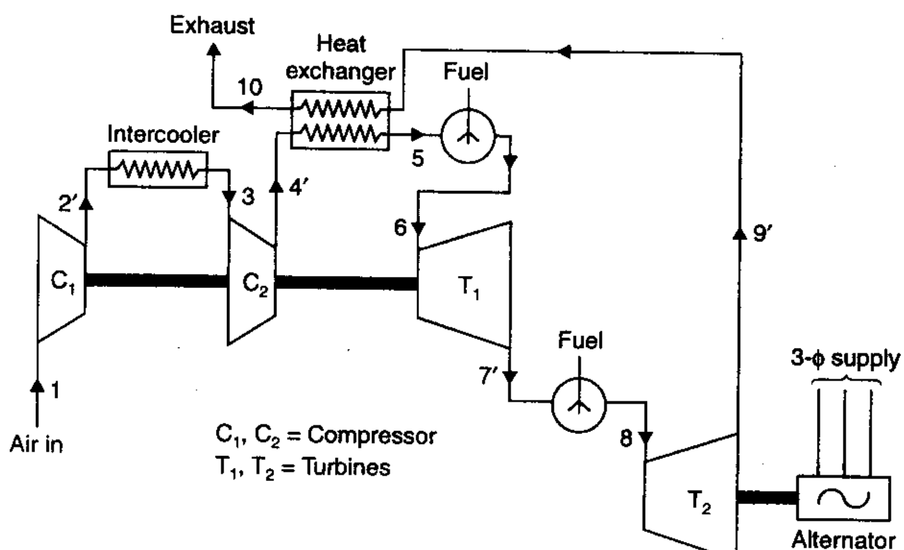
$$c_{pa} = 1 \text{ kJ/kg K} \; ; \; c_{pg} = 1.1 \text{ kJ/kg K} \; ; \; \gamma_{\text{air}} = 1.4 \; ; \; \gamma_{\text{gas}} = 1.33$$

The schematic arrangement of the gas turbine plant and its corresponding *T-s* diagram are shown in Fig. 5.68 (*a*) and (*b*) respectively.

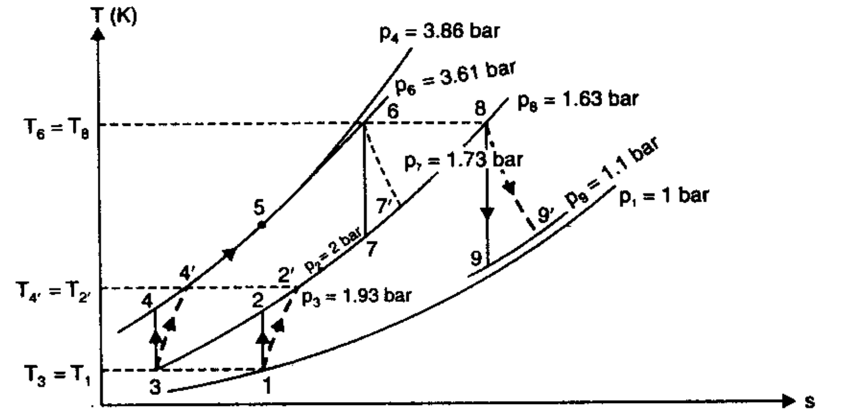
As for given data, we have :

$$p_1 = 1 \text{ bar}, \qquad p_2 = 2 \text{ bar as } \frac{p_2}{p_1} = 2 \text{ (Given)}$$

$$p_3 = p_2 - (\Delta p)_{\text{intercooler}} = 2 - 0.07 = 1.93 \text{ bar}$$



(a) Schematic arrangement of the plant



(b) T-s diagram

Fig. 5.68

$$p_4 = 2p_3 = 2 \times 1.93 = 3.86 \text{ bar} \qquad [\because (r_p)_{C_1, C_2} = 2:1]$$

$$p_6 = p_4 - (\Delta_p)_{\text{H.E. air side}} - (\Delta p)_{C.C} = 3.86 - 0.1 - 0.15 = 3.61 \text{ bar}$$

p_7 is to be calculated

$$p_8 = p_7 - (\Delta p)_{\text{reheater}}$$

$$p_9 = 1 + (\Delta p)_{\text{H.E. gas side}} = 1 + 0.1 = 1.1 \text{ bar}$$

Compressor C_1 :

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left[(r_p)_{C_1}\right]^{\frac{\gamma-1}{\gamma}} = (2)^{\frac{1.4-1}{1.4}} = (2)^{0.2857} = 1.22$$

$$\therefore \qquad T_2 = T_1 \times 1.22 = 288 \times 1.22 = 351.36 \text{ K}$$

$$\eta_{C_1} = 0.8 = \frac{T_2 - T_1}{T_{2'} - T_1} = \frac{351.36 - 288}{T_{2'} - 288}$$

$$\therefore \qquad T_{2'} = 288 + \frac{351.36 - 288}{0.8} = 367.2 \text{ K}$$

Compressor C_2 :

Since the *intercooling is perfect* therefore,

$$T_1 = T_3 = 288 \text{ K, and}$$

$$T_{2'} = T_{4'} = 367.2 \text{ K} \qquad [\because (r_p)_{C_1} = (r_p)_{C_2} = 2:1]$$

$\therefore$ The power required to run the compressors

$$= m_a\, c_{pa}(T_{2'} - T_1) + m_a\, c_{pa}(T_{4'} - T_3)$$

$$= 2 \times m_a\, c_{pa}(T_{2'} - T_1) \qquad [\because T_1 = T_3 \text{ and } T_{2'} = T_{4'}]$$

$$= 2 \times 20 \times 1(367.2 - 288) = 3168 \text{ kW}$$

Compressor turbine :

Power developed by the compressor turbine,

$$W_{T_1} = \frac{3168}{\eta_{\text{mech}}} = \frac{3168}{0.99} = 3200 \text{ kW}$$

$\therefore$ Work developed by the compressor turbine per kg of air

$$= \frac{3200}{20} = 160 \text{ kJ/kg}$$

The work developed by the compressor turbine per kg of air is given by

$$c_{pg}(T_6 - T_{7'}) = 160$$

$$\therefore \quad T_6 - T_{7'} = \frac{160}{1.1} = 145.45 \text{ K}$$

$$\frac{T_6}{T_7} = [(r_p)_{T_1}]^{\frac{\gamma-1}{\gamma}}$$

$$\eta_{T_1} = \frac{T_6 - T_{7'}}{T_6 - T_7} \quad \text{or} \quad T_6 - T_{7'} = \eta_{T_1}(T_6 - T_7)$$

or
$$T_6 - T_{7'} = \eta_{T_1} \times T_6 \left(1 - \frac{T_7}{T_6}\right) = \eta_{T_1} T_6 \left[1 - \frac{1}{\{(r_p)_{T_1}\}^{\frac{\gamma-1}{\gamma}}}\right]$$

or
$$T_6 - T_{7'} = 0.87 \times 1000 \left[1 - \frac{1}{\{(r_p)_{T_1}\}^{\frac{1.33-1}{1.33}}}\right]$$

or
$$145.45 = 870 \left[1 - \frac{1}{\{(r_p)_{T_1}\}^{0.2481}}\right]$$

or
$$0.167 = 1 - \frac{1}{\{(r_p)_{T_1}\}^{0.2481}}$$

$$\therefore \quad (r_p)_{T_1} = \left(\frac{1}{1 - 0.167}\right)^{\frac{1}{0.2481}} = 2.09$$

Also,
$$(r_p)_{T_1} = 2.09 = \frac{p_6}{p_7} \quad \therefore \quad p_7 = \frac{p_6}{2.09} = \frac{3.61}{2.09} = 1.73 \text{ bar}$$

$$p_8 = p_7 - (\Delta p)_{\text{reheater}} = 1.73 - 0.1 = 1.63 \text{ bar}$$

$$(r_p)_{T_2} = \frac{p_8}{p_9} = \frac{1.63}{1.1} = 1.48$$

where $(r_p)_{T_1}$ and $(r_p)_{T_2}$ are the pressure ratios of the turbines T_1 and T_2.

Turbine T_2 :

$$\frac{T_8}{T_9} = \left(\frac{p_8}{p_9}\right)^{\frac{\gamma-1}{\gamma}} = (1.48)^{\frac{1.33-1}{1.33}} = (1.48)^{0.2481} = 1.1$$

$$\therefore \quad T_9 = \frac{T_8}{1.1} = \frac{1000}{1.1} = 909.1 \text{ K}$$

$$\eta_{T_2} = 0.8 = \frac{T_8 - T_{9'}}{T_8 - T_9} = \frac{1000 - T_{9'}}{1000 - 909.1}$$

$$\therefore \quad T_{9'} = 1000 - 0.8(1000 - 909.1) = 927.3 \text{ K}$$

(*i*) **Net output of the plant :**

The power output of the plant is only from turbine T_2.

$\therefore$ Net-output of the plant $= m_a\, c_{pa}\,(T_8 - T_{9'})$

$= 20 \times 1.0(1000 - 927.3) = \mathbf{1454\ kW}$. **(Ans.)**

(*ii*) **Specific fuel consumption of the plant :**

The effectiveness of heat exchanger

$$\epsilon = 0.75 = \frac{T_5 - T_{4'}}{T_{9'} - T_{4'}} = \frac{T_5 - 367.2}{927.3 - 367.2}$$

$$\therefore \quad T_5 = 367.2 + 0.75(927.3 - 367.2) = 787.3\ \text{K}$$

The total heat supplied in the plant per kg of air,

$$Q_s = c_{pg}\,(T_6 - T_5) + c_{pg}\,(T_8 - T_{7'}) = c_{pg}\,[(T_6 - T_{7'}) + (T_8 - T_5)]$$
$$= 1.1(145.45 + 1000 - 787.3) = 394\ \text{kJ/kg})$$

Assuming m_f is mass of fuel supplied per second in combustion chamber and reheater, then,

Heat developed = Heat-gained by air

$$m_f \times C \times \eta_{comb.} = 20 \times 394$$

or $$m_f \times 43500 \times 0.98 = 20 \times 394$$

$$\therefore \quad m_f = \frac{20 \times 394}{43500 \times 0.98} = 0.185\ \text{kg/s} = 666\ \text{kg/h}$$

$\therefore$ Specific fuel consumption (s.f.c.) $= \dfrac{666}{1454} = \mathbf{0.458\ kg/kWh}$. **(Ans.)**

(*iii*) **Overall efficiency of the plant, $\eta_{overall}$:**

$$\eta_{overall} = \frac{\text{Output}}{\text{Input}}$$
$$= \frac{1454}{20 \times 394} = 0.1845 \quad \text{or} \quad \mathbf{18.45\%}. \ \textbf{(Ans.)}$$

HIGHLIGHTS

1. The major fields of application of gas turbines are :
   (*i*) Aviation (*ii*) Power generation
   (*iii*) Oil and gas industry (*iv*) Marine propulsion.
2. A gas turbine plant may be defined as one "in which the principal prime-mover is of the turbine type and the working medium is a permanent gas".
3. A simple gas turbine plant consists of the following :
   (*i*) Turbine (*ii*) Compressor
   (*iii*) Combustor (*iv*) Auxiliaries.
   A modified plant may have in addition an *intercooler, a regenerator, a reheater etc.*
4. Methods for improvement of thermal efficiency of open cycle gas turbine plant are :
   (*i*) Intercooling (*ii*) Reheating (*iii*) Regeneration.
5. *Free-piston engine plants* are the conventional gas turbine plants with the difference that the air compressor and combustion chamber are replaced by a free piston engine.

THEORETICAL QUESTIONS

1. What are the major fields of application of gas turbine ?
2. State the limitations of gas turbines.
3. List the applications of gas turbine plants.
4. State the advantages and disadvantages of gas turbine power plants over diesel and thermal power plants.
5. What factors should be considered while selecting a site for a gas turbine power plant ?
6. Give the description of a simple gas turbine plant.
7. Explain with the help of a neat diagram the energy cycle for a simple-cycle gas turbine.
8. Define the following performance terms :
   Air ratio, Pressure ratio, Work ratio, Compressor efficiency, Engine efficiency, Machine efficiency, Combustion efficiency and Thermal efficiency.
9. How are gas turbine power plants classified ?
10. How are gas turbines classified ?
11. State the merits of gas turbines over I.C. engines and steam turbines. Discuss also the demerits over gas turbines.
12. Describe with neat sketches the working of a simple constant pressure open cycle gas turbine.
13. Discuss briefly the methods employed for improvement of thermal efficiency of open cycle gas turbine plant.
14. Describe with a neat diagram a closed cycle gas turbine. State also its merits and demerits.
15. Explain with a neat sketch the working of a constant volume combustion turbine.
16. Write a short note on fuels used for gas turbines.
17. What do you mean by "combination gas turbine cycles". Explain briefly combined gas turbine and steam power plants.
18. List the advantages of 'combined cycle'.
19. How is a gas turbine 'started' and 'shut down' ?
20. Explain with a neat sketch the layout of a gas turbine power plant.
21. Enumerate and explain briefly the components of a gas turbine power plant.
22. With the help of neat sketches give the working of a free-piston engine plant. State also the advantages and disadvantages of such an arrangement.

UNSOLVED EXAMPLES

1. In an air standard gas turbine engine, air at a temperature of 15°C and a pressure of 1.01 bar enters the compressor, where it is compressed through a pressure ratio of 5. Air enters the turbine at a temperature of 815°C and expands to original pressure of 1.01 bar. Determine the ratio of turbine work to compressor work and the thermal efficiency when the engine operates on ideal Brayton cycle.
   Take : $\gamma = 1.4$, $c_p = 1.005$ kJ/kg K. **[Ans.** 2.393 ; 37.03%]
2. In an open cycle constant pressure gas turbine air enters the compressor at 1 bar and 300 K. The pressure of air after the compression is 4 bar. The isentropic efficiencies of compressor and turbine are 78% and 85% respectively. The air-fuel ratio is 80 : 1. Calculate the power developed and thermal efficiency of the cycle if the flow rate of air is 2.5 kg/s.
   Take $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for air and $c_{pg} = 1.147$ kJ/kg K and $\gamma = 1.33$ for gases.
   $R = 0.287$ kJ/kg K. Calorific value of fuel = 42000 kJ/kg. **[Ans.** 204.03 kW/kg of air ; 15.54%]
3. A gas turbine has a pressure ratio of 6/1 and a maximum cycle temperature of 600°C. The isentropic efficiencies of the compressor and turbine are 0.82 and 0.85 respectively. Calculate the power output in kilowatts of an electric generator geared to the turbine when the air enters the compressor at 15°C at the rate of 15 kg/s.
   Take : $c_p = 1.005$ kJ/kg K and $\gamma = 1.4$ for the compression process, and take $c_p = 1.11$ kJ/kg K and $\gamma = 1.333$ for the expansion process. **[Ans.** 920 kW]

4. Calculate the thermal efficiency and the work ratio of the plant in example 3(above), assuming that c_p for the combustion process is 1.11 kJ/kg K. **[Ans.** 15.8% ; 0.206]

5. The gas turbine has an overall pressure ratio of 5 : 1 and a maximum cycle temperature of 550°C. The turbine drives the compressor and an electric generator, the mechanical efficiency of the drive being 97%. The ambient temperature is 20°C and the isentropic efficiencies of the compressor and turbine are 0.8 and 0.83 respectively. Calculate the power output in kilowatts for an air flow of 15 kg/s. Calculate also the thermal efficiency and the work ratio.
Neglect changes is kinetic energy, and the loss of pressure in combustion chamber.
[Ans. 655 kW ; 12% ; 0.168]

6. Air is drawn in a gas turbine unit at 17°C and 101 bar and the pressure ratio is 8 : 1. The compressor is driven by the H.P. turbine and the L.P. turbine drives a separate power shaft. The isentropic efficiencies of the compressor, and the H.P. and L.P. turbines are 0.8, 0.85 and 0.83 respectively. Calculate the pressure and temperature of the gases entering the power turbine, the net power developed by the unit per kg/s of mass flow, the work ratio and the thermal efficiency of the unit. The maximum cycle temperature is 650°C.
For the compression process take c_p = 1.005 kJ/kg K and γ = 1.4
For the combustion process and expansion process, take
c_p = 1.15 kJ/kg K and γ = 1.333
Neglect the mass of fuel. **[Ans.** 1.65 bar, 393°C ; 74.5 kW, 0.201 ; 19.1%]

7. In a gas turbine plant, air is compressed through a pressure ratio of 6 : 1 from 15°C. It is then heated to the maximum permissible temperature of 750°C and expanded in two stages each of expansion ratio $\sqrt{6}$, the air being reheated between the stages to 750°C. An heat exchanger allows the heating of the compressed gases through 75 percent of the maximum range possible. Calculate : (*i*) The cycle efficiency (*ii*) The work ratio (*iii*) The work per kg of air.
The isentropic efficiencies of the compressor and turbine are 0.8 and 0.85 respectively.
[Ans. (*i*) 32.75% (*ii*) 0.3852 (*iii*) 152 kJ/kg]

8. At the design speed the following data apply to a gas turbine set employing the heat exchanger : Isentropic efficiency of compressor = 75%, isentropic efficiency of the turbine = 85%, mechanical transmission efficiency = 99%, combustion efficiency = 98%, mass flow = 22.7 kg/s, pressure ratio = 6 : 1, heat exchanger effectiveness = 75%, maximum cycle temperature = 1000 K.
The ambient air temperature and pressure are 15°C and 1.013 bar respectively. Calculate :
(*i*) The net power output (*ii*) Specific fuel consumption
(*iii*) Thermal efficiency of the cycle.
Take the lower calorific value of fuel as 43125 kJ/kg and assume no pressure-loss in heat exchanger and combustion chamber. **[Ans.** (*i*) 2019 kW (*ii*) 0.4999 kg/kWh (*iii*) 16.7%]

9. In a gas turbine plant air at 10°C and 1.01 bar is compressed through a pressure ratio of 4 : 1. In a heat exchanger and combustion chamber the air is heated to 700°C while its pressure drops 0.14 bar. After expansion through the turbine the air passes through a heat exchanger which cools the air through 75% of maximum range possible, while the pressure drops 0.14 bar, and the air is finally exhausted to atmosphere. The isentropic efficiency of the compressor is 0.80 and that of turbine 0.85. Calculate the efficiency of the plant. **[Ans.** 22.76%]

10. In a marine gas turbine unit a high-pressure stage turbine drives the compressor, and a low-pressure stage turbine drives the propeller through suitable gearing. The overall pressure ratio is 4 : 1, and the maximum temperature is 650°C. The isentropic efficiencies of the compressor, H.P. turbine, and L.P. turbine are 0.8, 0.83 and 0.85 respectively, and the mechanical efficiency of both shafts is 98%. Calculate the pressure between turbine stages when the air intake conditions are 1.01 bar and 25°C. Calculate also the thermal efficiency and the shaft power when the mass flow is 60 kg/s. Neglect kinetic energy changes, and pressure loss in combustion. **[Ans.** 1.57 bar ; 14.9% ; 4560 kW]

11. In a gas turbine unit comprising L.P. and H.P. compressors, air is taken at 1.01 bar 27°C. Compression in L.P. stage is upto 3.03 bar followed by intercooling to 30°C. The pressure of air after H.P. compressor is 58.7 bar. Loss in pressure during intercooling is 0.13 bar. Air from H.P. compressor is transferred to heat exchanger of effectiveness 0.60 where it is heated by gases from L.P. turbine. The temperature of

gases supplied to H.P. turbine is 750°C. The gases expand in H.P. turbine to 3.25 bar and are then reheated to 700°C before expanding in L.P. turbine. The loss of pressure in reheated is 0.1 bar. If isentropic efficiency of compression in both stages is 0.80 and isentropic efficiency of expansion in turbine is 0.85, calculate : (*i*) Overall efficiency (*ii*) Work ratio (*iii*) Mass flow rate when the gas power generated is 6500 kW. Neglect the mass of fuel.

Take, for air : c_p = 1.005 kJ/kg K, γ = 1.4

for gases : c_{pg} = 1.15 kJ/hr K, γ = 1.3. [**Ans.** (*i*) 16.17% ; (*ii*) 0.2215 ; (*iii*) 69.33 kg of air/s]

12. In a gas turbine installation, air is taken in L.P. compressor at 15°C, 1.1 bar and after compression it is passed through intercooler where its temperature is reduced to 22°C. The cooled air is further compressed in H.P. unit and then passed in the combustion chamber where its temperature is increased to 677°C by burning the fuel. The combustion products expand in H.P. turbine which runs the compressor and further expansion is continued in the L.P. turbine which runs the alternator. The gases coming out from L.P. turbine are used for heating the incoming air from H.P. compressor and then exhausted to atmosphere.

Taking the following data determine :

(*i*) Power output (*ii*) Specific fuel consumption

(*iii*) Thermal efficiency :

Pressure ratio of each compressor = 2, isentropic efficiency of each compressor stage = 85%, isentropic efficiency of each turbine stage = 85%, effectiveness of heat exchanger = 0.75, air flow = 15 kg/sec., calorific value of fuel = 45000 kJ/kg, c_p(for gas) = 1 kJ/kg K, c_p (for gas) = 1.15 kJ/kg K, γ (for air) = 1.4, γ (for gas) = 1.33.

Neglect the mechanical, pressure and heat losses of the system and fuel mass also.

[**Ans.** (*i*) 1849.2 kW ; (*ii*) 0.241 kg/kWh ; (*iii*) 33.17%]

COMPETITIVE EXAMINATIONS QUESTIONS

1. (*a*) With the help of a block-diagram, explain the working principles of a closed cycle gas turbine plant.

(*b*) Explain three important refinements by which the efficiency of a simple gas turbine plant can be improved considerably.

(*c*) What are the various fuels that are usually used for running gas turbines ?

2. (*a*) What are the advantages of a gas turbine plant over diesel and steam power plants of the same capacity ?

(*b*) What are the different components of a gas turbine plant ?

Explain them with the help of neat sketches.

3. (*a*) Draw a neat diagram of a regenerative gas turbine plant having intercooling and reheater. Explain the working on the *p-v* diagram.

(*b*) (*i*) Under what conditions the gas turbine plants have maximum utility ?

(*ii*) What are the disadvantages of gas turbine plants ?

(*iii*) Why are gas turbine plants not so common in India ?

4. (*a*) What are the different fuels that are used for gas turbine power plants ? What are the most suitable fuels for gas turbine plants in a country like India ? Explain.

(*b*) Explain with the help of *T-s* chart and line diagram the working of a gas turbine plant. How can the efficiency of the plant be improved ?

5. (*a*) Describe with diagram the working of a closed cycle gas turbine plant.

(*b*) Describe methods of improving output and performance of gas turbine plants.

6. (*a*) State the advantages and disadvantages of gas turbine power plants.

(*b*) The following data refer to an open-cycle gas turbine plant. The compressor, turbine and electric generator are mounted on the same shaft :

Compressor pressure ratio	= 4 : 1
Isentropic efficiency of compressor	= 85%
Isentropic efficiency of the turbine	= 85%

Effectiveness of the heat exchanger = 76%
Pressure loss in the combustion chamber = 0.05 kgf/cm^2
Combustion efficiency = 90%
Mechanical efficiency = 95%
C.V. of fuel used = 9,000 kcal/kg
Rate of air flow = 20 kg/sec
Ambient pressure and temperature = kgf/cm^2 and 30°C
The maximum temperature of the cycle = 1000°K
$c_{p\ air}$ = .24, $c_{p\ gas}$ = 0.25, γ for air = γ for gas = 1.4.

Calculate : (*i*) Power output at the generator ; (*ii*) Overall efficiency of the plant ; (*iii*) Specific fuel consumption.

7. (*a*) Make a schematic diagram of a gas turbine power plant employing a regenerator, a reheater and an intercooler. Draw *T-s* diagram showing the above arrangement.
(*b*) The isentropic efficiency of a compressor in a simple gas turbine plant is 80% and that of the turbine unit is 85%. The air enters at 290°K and the maximum temperature in the cycle is 800°C, while the pressure rises from 1.25 kgf/cm^2 to 5.0 kgf/cm^2. Draw the cycle on *T-s* diagram and calculate the efficiency of the plant and the work ratio. Take γ = 1.4 and c_p = 0.24.

8. Calculate the h of an open cycle internal combustion turbine fitted with a heat exchanger of 75% effectiveness. The pressure ratio is 4 : 1 and compression is carried out in two stages of equal pressure ratio with intercooling back to initial temperature of 15°C. The maximum temperature is 650°C. The turbine isentropic h is 88% and each compressor isentropic h is 85%.
For air γ = 1.4 and c_p = 0.24.

9. (*a*) Discuss in brief the methods adopted for improving the thermal efficiency of simple gas turbine plant.
(*b*) A simple gas turbine plant draws in air at 1 kgf/cm^2 and 15°C and compresses it through a pressure ratio of 5 : 1, the isentropic efficiency of compression being 85%. The air passes to the combustion chamber and after combustion, gases enter the turbine at a temperature of 540°C and expand to 1 kgf/cm^2, the turbine efficiency being 80%.
Calculate the flow of air and gases for a net horse-power of 2,200 assuming that the loss of pressure through the combustion system is 0.07 kgf/cm^2. Take c_p = 0.25 for air and combustion gas and γ = 1.40. Neglect the addition of mass flow due to fuel.

10. (*a*) What do you understand by combination gas turbine plants ? What are their salient features ? Discuss briefly the popular systems.
(*b*) Describe briefly the working of different type of relays which are commonly used in power plants.

11. (*a*) What are the different methods used to improve output and performance of gas turbine plants ? Discuss.
(*b*) "Diesel power plants are used as stand by units in a grid system". Discuss.

12. Write short notes on the following :
(*a*) Solar power plants ; (*b*) Switchgears ;
(*c*) Closed cycle gas turbine plant.

13. (*a*) Discuss in brief the availability of nuclear fuel in India.
(*b*) Distinguish between (*i*) fission and fusion, and (*ii*) fast neutrons and thermal neutrons.
(*c*) In a nuclear reactor, explain the functions of :
(*i*) Moderator ; (*ii*) Coolant ; (*iii*) Control rod ; (*iv*) Reflector.

14. (*a*) Discuss the future of nuclear power plants for power generation in India. Enumerate the advantages and disadvantages of nuclear power plants as compared to conventional power plants.
(*b*) Describe, with the help of a neat sketch, the working of a pressurised water reactor plant. What are its advantages and disadvantages ?

15. (*a*) Discuss the factors to be kept in mind for selecting the site of a nuclear power plant.
(*b*) What is the function of moderator in a nuclear power plant ? What are the desirable properties of a good moderator ? Compare different moderators used in practice.

16. (*a*) Draw a neat diagram of nuclear reactor and explain the function of its various components.
(*b*) Explain with a neat sketch the principle of working of magneto-hydrodynamic system of power generation. State the difficulties encountered in its design.

17. (*a*) What do you understand by a close cycle gas turbine power plant ? List out its advantages over open cycle plant. What difficulties are there in development of close cycle plants ?
(*b*) Following observations were made during a test on an open cycle constant pressure gas turbine plant :

Inlet temperature	= 27°C
Maximum temperature in the cycle	= 800°C
Pressure ratio	= 6
Isentropic efficiency of compressor	= 85%
Isentropic efficiency of turbine	= 90%
Combustion efficiency	= 95%
Mass flow of air	= 100 kg/sec
Specific heat of air	= 0.24
Specific heat of gases	= 0.26
Specific heat ratio for air and gases	= 1.4
Calorific value of fuel	= 10,500 kcal/kg

Find : (*i*) Thermal efficiency of the plant : (*ii*) Power developed ; (*iii*) Air fuel ratio ; (*iv*) Specific fuel consumption.

18. (*a*) Describe the functions of the major components used in gas turbine plant.
(*b*) What are the factors to be considered in designing steam piping in thermal plant ? What are the materials used for (*i*) steam piping, and (*ii*) its thermal insulation ?

19. (*a*) What are the advantages of gas turbine plant over other thermal power plants ?
(*b*) Describe briefly the working of closed cycle gas turbine plant with the help of *T-s* diagram.
(*c*) In a closed cycle gas turbine the following data were applied :

Working substance is air with $c_p = 0.24$ and γ	= 1.4
Ambient temperature	= 27°C
Maximum temperature	= 823°C
Pressure at compressor inlet	= 1 kgf/cm^2
Pressure ratio	= 4
Compressor efficiency	= 80%
Gas turbine efficiency	= 85%

Determine the thermal efficiency of gas turbine.

20. In a 5 MW gas turbine generating set, HP turbine is used to drive the compressor unit and LP turbine, mounted on separate shaft, drives the alternator. After two-stage compression having overall pressure ratio of 9 with perfect intercooling, the compressed air is supplied to the combustion chamber *via* a regenerator which receives the exhaust from the LP turbine. The temperature of the gases at entry to the HP turbine is 727°C and the gases are reheated to 727°C again before entering the LP turbine. Draw the schematic diagram of the plant and *T-s* diagram. Considering open cycle, calculate (*i*) overall thermal efficiency of the plant and (*ii*) the mass flow rate of air in kg/sec.
Assume the following :

Air inlet pressure and temperature 1 kg/cm^2 and 27°C respectively	
Isentropic efficiency of compressor in each stage	= 0.8
Isentropic efficiency of expansion in each unit	= 0.85
Mechanical efficiency of both the turbines	= 0.98
Thermal ratio of the regenerator	= 0.75
Combustion efficiency	= 0.97

$c_{p\,air} = 0.24$: $c_{p\,gas} = 0.25$; $\gamma_{air} = 1.4$; $\gamma_{gas} = 1.33$.

Neglect mass of fuel and consider the specific heat values to be constant over the pressure and temperature range.

6

Hydro-Electric Power Plant

6.1. INTRODUCTION

In hydro-electric plants energy of water is utilised to move the turbines which in turn run the electric generators. The energy of water utilised for power generation may be kinetic or potential. The *kinetic energy* of water is its energy in motion and is a function of mass and velocity, while the *potential energy* is a function of the difference in level/head of water between two points. In either case continuous availability of a water is a basic necessity ; to ensure this, water collected in natural lakes and reservoirs at high altitudes may be utilised or water may be artificially stored by constructing dams across flowing streams. The ideal site is one in which a good system of natural lakes with substantial catchment area, exists at a high altitude. *Rainfall is the primary source of water* and depends upon such factors as temperature, humidity, cloudiness, wind etc. The usefulness of rainfall for power purposes further depends upon several complex factors which include its intensity, time distribution, topography of land etc. However it has been observed that only a small part of the rainfall can actually be utilised for power generation. A significant part is accounted for by *direct evaporation,* while another similar quantity *seeps* into the soil and forms the underground storage. Some water is also absorbed by vegetation. Thus, only a part of water falling as rain actually flows over the ground surface as direct run off and forms the streams which can be utilised for hydro-schemes.

First hydro-electric station was probably started in America in 1882 and thereafter development took place very rapidly. In India the first major hydro-electric development of 4.5 MW capacity named as Sivasamudram Scheme in Mysore was commissioned in 1902. In 1914 a hydro-power plant named Khopoli project of 50 MW capacity was commissioned in Maharashtra. The hydro-power capacity, upto 1947, was nearly 500 MW.

Hydro (water) power is a conventional renewable source of energy which is clean, free from pollution and generally has a good environmental effect. However the following factors are major obstacles in the utilisation of hydro-power resources :