24
Gas Turbine Power Plants

24.1. INTRODUCTION

The gas turbine is one of the oldest devices known to man for the conversion of heat into mechanical work. The gas turbine is most satisfactory power developing unit among various means of producing mechanical power due to its exceptional reliability ; freedom from vibration and ability to produce large powers from units of comparatively small size and weights. The use of gas turbine plant for developing power was known as early as in 1872. Unfortunately, most efforts towards the development failed due to the lack of understanding of diffusion process taking place in the air passing through the compressor. The axial flow compressor was built to a reasonable degree of efficiency in period of 1930 to 1935 and high temperature materials were also made available during the same period. The rapid progress in the gas turbine development was made as a result of industrial and military needs. The use of gas turbine in the power generation industry is more recent than its use in any other field.

Since 1950s, gas turbine plants have been used in remote oil and gas fields, on offshore platforms and in gas-pipeline pumping stations. In the locations, where fresh water is scarce, fuel supplies cheap and space requirements limited, compact gas turbine sets have proven invaluable.

In the last two decades, the 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 three directions, increase in unit capacities of gas turbine units (50 – 100 MW) ; increase in their efficiency (37%) and drop in capital cost (Rs. 5000 to 7000 per kW installed).

The economics of power generation by gas turbine is proving more attractive in all parts of the world due to its low capital cost and high reliability and flexibility in operation. Another outstanding feature of gas turbine plant for power generation is capability of quick starting and capability of using wide variety of fuels from natural gas to residual oil or powdered coal.

Primary application of gas turbine to the electric supply industry is still for peaking power. The world's largest peaking installation of 480 MW is at Astoria Power Station in U.S.A. The massive North-East power blackout in 1965 and heavy summer loads in 1969 and other power emergencies during 1970 have helped to establish the gas turbine as an indispensable peaking generator unit in the U.S. power generation industry. The power generation by gas turbine is accounted as 12% of total MW installed during 1969-70 period. Economic installation costs and operational flexibility forecast a long term role for the turbine in the power generation field.

It's relatively low installation cost per kW installed capacity commended attention throughout the world as excellent source of peaking or emergency power. As manufacturers increased the unit size and as the operators become impressed with the operating characteristics of the gas turbine plant, the running hours tended to increase subject to the limitations of fuel economics. This situation was further improved with the addition of regenerative cycle. With the addition of steam cycle in the gas turbine cycle, the overall efficiency of the plant is further increased.

The gas turbine as a base load plant is preferred over the other plants as major delays in completion of large base load fossil and nuclear units.

The gas turbine power plant now-a-days is universally used as peak load, base load as well as standby unit due to its outstanding operational characteristics.

Generation of power using oil in India is neither desirable nor economical as 50% of the oil requirement of the nation is already imported. But large quantity of gas available from Bombay High and presently detected sizeable natural gas reserves in western region have created the hopes to instal gas turbine units during VIII to X plan period which will also help to meet the peak power demand which is the major problem in the national power development programme. Gas reserves of 700 billion m^3 equivalent to about 570 million tons of oil have been estimated. In the context of present power shortages and economics of gas based generation, the power generation using gas is considered in VIII to X plan period.

A gas turbine of Delhi Electric Supply Undertaking which draws fuel from HBT pipeline was synchronized with the grid on 30th July 1989. The Corporation of Delhi became the first civic body in the country to produce electricity from natural gas.

The DESU has set up six turbine units each of 30 MW capacity in 1986 which could be run either on high speed diesel or naphtha. Now Alsthom of France has converted one of these six to run on natural gas. It will be converting remaining one after other, reducing the cost of generation to Rs. 1.1 per unit against Rs. 1.37 with naphtha and Rs. 1.56 with diesel. The computer control and monitoring system used with this plant is one of the four such systems installed in the world.

Another notable gas turbine plant is at Anta in Rajasthan whose first unit was commissioned in 1989 using gas as a fuel.

The role of gas turbine in India is still limited for meeting peak demand and assisting emergencies in the power system. In early sixties, a number of small gas turbine sets (10 MW) were installed at Bangalore, Hyderabad etc. using oil as fuel. With the availability of natural gas in Assam and West Coast, gas turbines were installed at Naharkatia (2 × 23 MW) in Assam and at Dhuvaran (2 × 27 MW) in Gujarat. In the seventies, severe power crisis forced West Bengal to install five 20 MW sets (run on oil) and Delhi also followed with the installation of six 30 MW sets.

Exploration of huge quantity of natural gas from Bombay High changed the scene from the beginning of the eighties. Maharashtra State Electricity Board (MSEB) went for a large gas turbine base-power station at Uran, near Bombay, by installing four 60 MW sets and four 108 MW sets of gas turbines.

24.2. GAS TURBINE PLANT

The gas turbine plant essentially consists of compressor, combustion chamber and turbine as shown in Fig. 24.1.

The air is compressed in a compressor and the fuel is burned in the combustion chamber when the compressed air is supplied from the compressor. The burned high temperature gases are passed through the turbine. Th part of the work developed by the gases passing through the turbine is used to run the compreessor and remaining (30 – 35%) is used to generate the electrical energy as shown in Fig. 24.1.

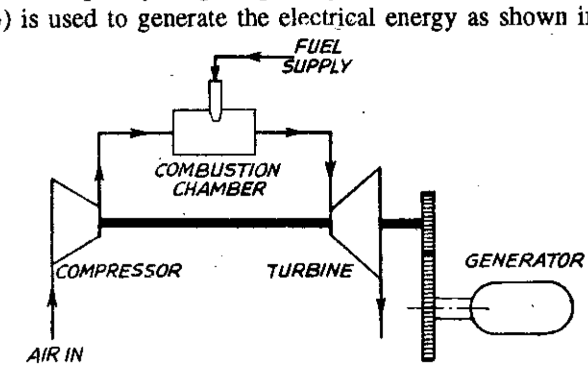


Fig. 24.1. Gas Turbine Plant.

When the heat is given to the air by mixing and burning the fuel in the air and the gases coming out of the turbine are exhausted to the atmosphere, the cycle is known as open cycle power plant. If the heat to the working medium (air or any other suitable gas) is given without directly burning the fuel in the air and the same working fluid is used again and again, the cycle is known as closed cycle power plant.

24.3. CLASSIFICATION AND COMPARISON OF DIFFERENT GAS TURBINE POWER PLANTS

The gas turbine power plants which are used in electric power industry are classified into two groups as per the cycle of operation :

(*a*) Open cycle gas turbine.

(*b*) Closed cycle gas turbine.

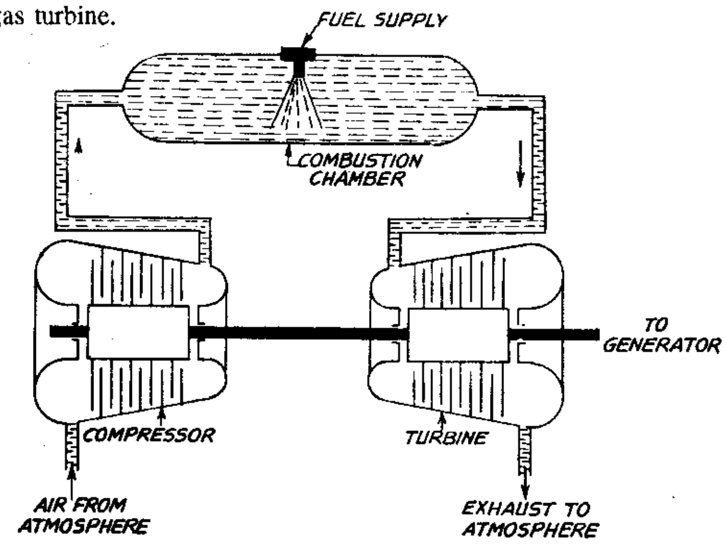


Fig. 24.2. Open Cycle gas turbine.

(*c*) **Open cycle gas turbine power plant.** A simple open cycle gas turbine consists of a compressor, combustion chamber and a turbine as shown in Fig. 24.2. The compressor takes in ambient air and raises its pressure. Heat is added to the air in combustion chamber by burning the fuel and raises its temperature. The heated gases coming out of combustion chamber are then passed to the turbine where it expands doing mechanical work. Part of the power developed by the turbine is utilised in driving the compressor and other accessories and remaining is used for power generation. Since ambient air enters into the compressor and gases coming out of turbine are exhausted into the atmosphere, the working medium must be replaced continuously. This type of cycle is known as open cycle gas turbine plant and is mainly used in majority of gas turbine power plants as it has many inherent advantages. The advantages and disadvantages of such type of cycle are discussed below.

Advantages. 1. *Warm-up time.* Once the turbine is brought up to the rated speed by the starting motor and the fuel is ignited, the gas turbine will be accelerated from cold start to full load without warm-up time.

2. *Low weight and size.* The weight in kg per kW developed is less.

3. *Fuels.* Almost any hydro-carbon fuel from high octane gasoline to heavy diesel oils can be used in the combustion chamber.

4. Open cycle plants occupy comparatively little space.

5. The stipulation of a quick start and take-up of load frequently are the points in favour of open cycle plant when the plant is used as peak load plant.

6. Component or auxiliary refinements can usually be varied to improve the thermal efficiency and give the most economical overall cost for the plant load factors and other operating conditions envisaged.

7. Open-cycle gas turbine power plant, except those having an intercooler, does not require cooling water. Therefore, the plant is independent of cooling medium and becomes self-contained.

Disadvantages. 1. The part load efficiency of the open cycle plant decreases rapidly as the considerable percentage of power developed by the turbine is used to drive the compressor.

2. The system is sensitive to the component efficiency ; particularly that of compressor. The open cycle plant is sensitive to changes in the atmospheric air temperature, pressure and humidity.

3. The open-cycle gas turbine plant has high air rate compared to the other cycles, therefore, it results in increased loss of heat in the exhaust gases and large diameter duct work is necessary.

4. It is essential that the dust should be prevented from entering into the compressor in order to minimise erosion and depositions on the blades and passages of the compressor and turbine and so impairing their profile and efficiency. The deposition of the carbon and ash on the turbine blades is not at all desirable as it also reduces the efficiency of the turbine.

(*b*) **Closed cycle gas turbine power plant.** The idea of closed cycle gas turbine plant was originated and developed in Switzerland. In the year 1935, J. Ackeret and C. Keller first proposed this type of machine and first plant was completed in Zurich in 1944. It used air as working medium and had a useful output of 2 MW. Since then, a number of closed cycle gas turbine plants have been built all over the world and largest of 17 MW capacity is at Gelsenkirchen, Germany and has been successfully operating since 1967.

In closed cycle gas turbine plant, the working fluid (air or any other suitable gas) coming out from compressor is heated in a heater by an external source at constant pressure. The high temperature and high pressure air coming out from the external heater is passed through the gas turbine. The fluid coming out from the turbine is cooled to its original temperature in the cooler using external cooling source before passing to the compressor. The working fluid is continuously used in the system without its change of phase and the required heat is given to the working fluid in the heat exchanger.

The arrangement of the components of the closed cycle gas turbine plant is shown in Fig. 24.3.

The advantages and disadvantages of this cycle are listed below.

Advantages. 1. The inherent disadvantage of open cycle gas turbine is the atmospheric back pressure at the turbine exhaust. With closed cycle gas turbine plants, the back pressure can be increased. Due to the control on back pressure, unit rating can be increased about in proportion to the back pressure. Therefore, the machine can be smaller and cheaper than the machine used to develop the same power using open cycle plant.

2. The closed cycle avoids erosion of the turbine blades due to the contaminated gases and fouling of compressor blades due to dust. Therefore, it is practically free from deterioration of efficiency in service. The absence of corrosion and abrasion of the interiors of the compressor and turbine extends the life of the plant and maintains the efficiency of the plant constant throughout its life as they are kept free from the products of combustion.

3. The need for filtration of the incoming air which is a severe problem in open cycle plant is completely eliminated.

Load variation is usually obtained by varying the absolute pressure and mass flow of the circulating medium, while the pressure ratio, the temperatures and the air velocities remain almost constant. This results in velocity ratio in the compressor and turbine independent of the load and full load thermal efficiency is maintained over the full range of operating loads.

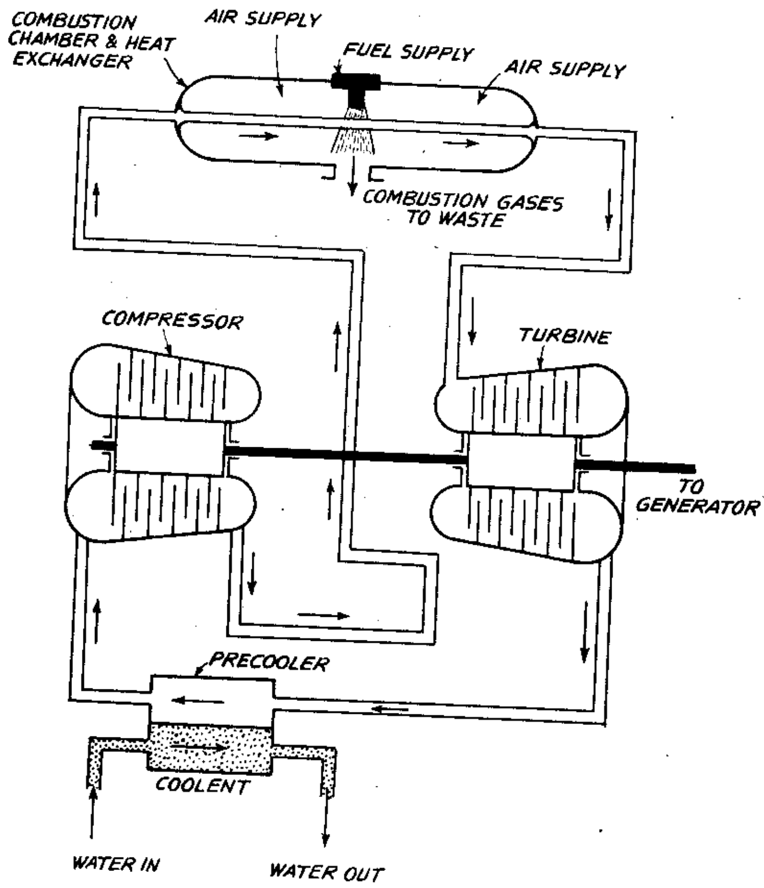


Fig. 24.3. Closed Cycle Gas Turbine Plant.

5. The density of the working medium can be maintained high by increasing pressure range, therefore, the compressor and turbine are smaller for their rated output. The high density of the working fluid further increases the heat transfer properties in the heat exchanger.

6. As indirect heating is used in closed cycle plant, the inferior oil or solid fuel can be used in the furnace and these fuels can be used more economically because these are available in abundance.

7. Finally the closed cycle opens the new field for the use of working medium (other than air as argon, CO_2, helium) having more desirable properties. The ratio γ of the working fluid plays an important role in determining the performance of the gas turbine plant. An increase in γ from 1.4 to 1.67 (for argon) can bring about a large increase in output per kg of fluid circulated and thermal efficiency of the plant.

The theoretical thermal efficiencies of the monoatomic gases will be highest for the closed cycle type gas turbine. Further, by using the relatively dense inert gases, such as argon, krypton and xenon, the advantage of smaller isentropic heat fall and smaller cross-sectional flow areas would be realised.

The properties of few gases which can be used in closed cycle gas turbine plant are given in the following table :

Physical Properties of Gases which can be used in closed cycle

Name of Gas	*Symbol*	*Atomic weight*	$\gamma = \frac{C_p}{C_v}$	(ρ) *Density* kg/m^3
Monoatomic				
Argon	A	40	1.667	1.781
Helium	He	4	1.63	0.177
Krypton	Kr	84	1.689	3.707
Xenon	Xe	130	1.666	5.650
Neon	Ne	20	1.642	0.968
Hg-Vapour	Hg	200	1.666	—
Diatomic		**MW***		
Dry air	—	—	1.4	1.298
Hydrogen	H_2	2	1.408	0.0899
Oxygen	O_2	16	1.4	1.429
Nitrogen	N_2	14	1.41	1.2507
CO	CO	28	1.401	1.2504
Triatomic		**MW***		
Ozone	O_3	48	1.29	—
CO_2	CO_2	44	1.30	1.9768
Ammonia	NH_3	17	1.336	0.7708
Polyatomic				
Methane	CH_4	16	1.313	0.7145
Benzene vapour	C_6H_6	78	1.40	—
Ethylene	C_2H_4	28	1.264	1.272
Acetylene	C_2H_2	26	1.26	1.190

*MW = molecular weight

Whether CO_2 or Helium should be adopted as working medium is matter of controversy at present. Blade material poses a problem to use helium as working fluid. In case of CO_2, a new kind of compressor must be designed to compress the fluid. The main advantage of CO_2 is that it offers 40% efficiency at 700°C whereas helium would need 850°C or more to achieve the same efficiency. A helium turbine would also need to run faster imposing larger stresses on the rotor.

8. The maintenance cost is low and reliability is high due to longer useful life.

9. The thermal efficiency increases as the pressure ratio (R_p) decreases. Therefore, appreciable higher thermal efficiencies are obtainable with closed cycle for the same maximum and minimum temperature limits as with the open cycle plant.

10. Starting of plant is simplified by reducing the pressure to atmospheric or even below atmosphere so that the power required for starting purposes is reduced considerably.

Disadvantages : 1. The system is dependent on external means as considerable quantity of cooling water is required in the pre-cooler.

2. Higher internal pressures involve complicated design of all components and high quality material is required which increases the cost of the plant.

3. The response to the load variations is poor compared to the open-cycle plant.

4. It requires very big heat-exchangers as the heating of working fluid is done indirectly. The space required for the heat exchanger is considerably large. The full heat of the fuel is also not used in this plant.

The closed cycle is only preferable over open cycle where the inferior type of fuel or solid fuel is to be used and ample cooling water is available at the proposed site of the plant.

However, closed cycle gas turbine plants have not as yet been used for electricity production. This is mainly a consequence of the limitations imposed by the unit size of heat exchanger. The use of a large number of parallel heat exchangers would practically eliminate the economic advantage resulting from increased plant size.

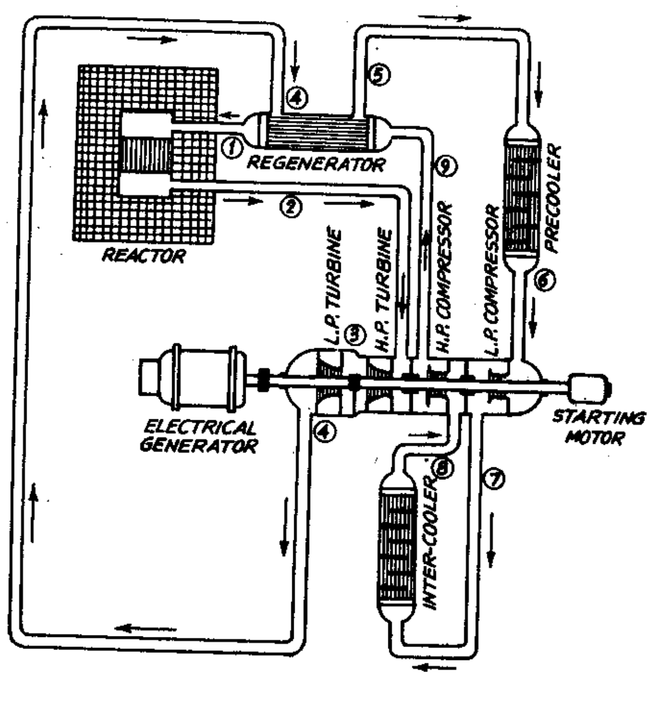


Fig. 24.4 (*a*). Direct turbine for helium cooled fast reactor.

The inherent disadvantage of open cycle is the atmospheric back pressure which limits the unit rating. This disadvantage can be eliminated in the closed cycle plant by increasing the back pressure of the cycle. With conventional closed cycle gas turbine plants, advantage can be taken of this only to a limited extent as the air heater limits the unit rating. This disadvantage does not apply to closed cycle plant with a nuclear reactor as heat source. Manufacturers of closed cycle gas turbine plant believe that with these sets, unit rating upto 500 MW may be possible.

With the use of nuclear reactor as heating source for gas, the heat exchangers can be eliminated from the closed cycle plant and the above-mentioned limitation (number of heat exchangers) does not exist. The power density in the core of a helium cooled fast reactor is a few thousand times higher than in conventional gas heat exchanger. Thus, units of several thousands of megawatts designed for high gas pressures can be housed in a single pre-stressed concrete vessel.

A typical closed cycle gas turbine plant using helium as working medium and helium cooled fast breeder reactor is shown in Fig. 24.4 (*a*) and corresponding *T-s* diagram is shown in Fig. 24.4 (*b*).

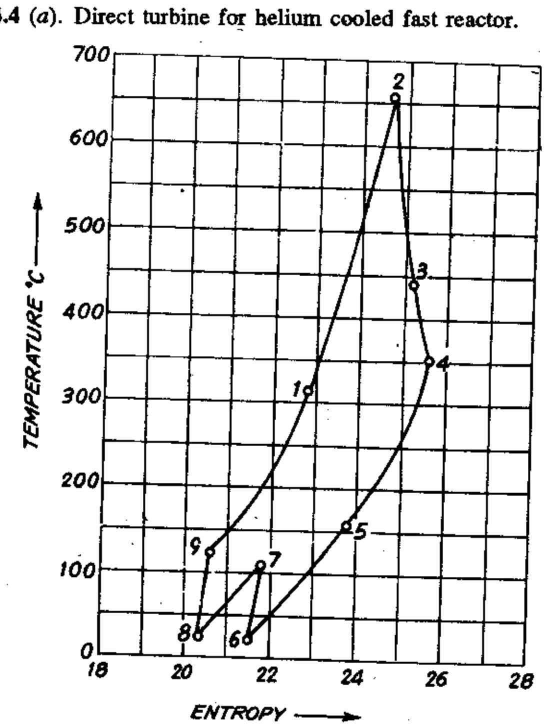


Fig. 24.4 (*b*). The processes are presented on *T-s* diagram.

A closed cycle gas turbine plant using helium as working medium is much smaller than of a conventional air-turbine plant of the same output. This is due to the better thermodynamic properties of helium relative to air and much higher pressures can be used in helium cooled fast reactor system. A helium-turbine used in closed cycle plant of 335 MW capacity at Switzerland is of 3.7 metre diameter and 14 metres long. The corresponding dimensions of the 17 MW air turbine at Gelsenkirohen plant are 2.6 metres in diameter and 9 metres long.

It is expected that in future, the combination of fast breeder reactors and gas turbines represents a very promising solution for future power generation. This is because of high breeding characteristics of the helium cooled fast reactors which ensure continuity of low fuel cost while the use of closed cycle gas turbine plant is expected to reduce the capital investment of the plant.

Cost is also roughly proportional to weight. One can expect much cheaper turbomachinery than steam plant.

24.4. ANALYSIS OF CLOSED CYCLE AND OPEN CYCLE CONSTANT PRESSURE GAS TURBINE PLANTS

The analysis of closed cycle can be approximately used for open cycle also. The gas turbine works

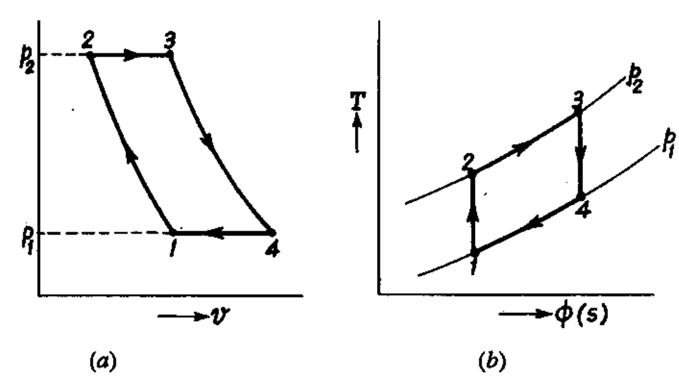


Fig. 24.5. Representation of closed cycle gas Turbine plant on $p - v$ and $T - s$ cycles.

on Brayton cycle. The thermodynamic processes of a closed cycle gas turbine plant are shown in Fig. 24.5 (*a*) and 24.5 (*b*) on $p - v$ and $T - s$ diagrams respectively.

The analysis is based on the following assumptions :

(1) The compression and expansion are isentropic.

(2) The pressure and heat losses in the system are neglected.

(3) The specific heat of working fluid is taken constant throughout the cycle.

Work done by the compressure per kg of working fluid is given by

$$W_c = C_p (T_2 - T_1)$$

Work developed by the turbine per kg of working fluid is given by

$$W_t = C_p (T_3 - T_4)$$

∴ Net available work for electric generation is given by

$$W_a = W_t - W_c = C_p (T_3 - T_4) - C_p (T_2 - T_1)$$

Heat supplied per kg of working fluid is given by

$$Q_s = C_p (T_3 - T_2)$$

∴ Thermodynamic efficiency of the cycle is given by

$$\eta_{th} = \frac{W_a}{Q_s} = \frac{C_p (T_3 - T_4) - C_p (T_2 - T_1)}{C_p (T_3 - T_2)} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_2)}$$

Applying the gas law to the points 1 and 2,

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

Similarly $$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{(\gamma-1)/\gamma} = \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = (R_p)^{(\gamma-1)/\gamma}$$

Substituting the values of T_1 and T_4 from the above equations into the expression of efficiency, we get

$$\eta_{th} = 1 - \left(\frac{1}{R_p}\right)^{(\gamma-1)/\gamma} \quad \text{...(24.1)}$$

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

$$\therefore \quad R_w \text{ (work ratio)} = \frac{W_a}{W_t} = \frac{W_t - W_c}{W_t} = 1 - \frac{(T_2 - T_1)}{(T_3 - T_4)} = 1 - \frac{T_1 (R_p)^{(\gamma-1)/\gamma} - T_1}{T_3 - \dfrac{T_3}{(R_p)^{(\gamma-1)/\gamma}}}$$

$$= 1 - \frac{T_1}{T_3} (R_p)^{(\gamma-1)/\gamma} \quad \text{...(24.2)}$$

The work ratio increases with an 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.

It is obvious from the Eq. (24.2), the maximum possible pressure ratio for the fixed temperature T_1 and T_3 is given by

$$(R_p)_{max} = \left(\frac{T_3}{T_1}\right)^{\gamma/(\gamma-1)} \quad \text{...(24.3)}$$

because, the value of R_p greater than this gives the negative value of work ratio which is impossible for power developing system.

Therefore, the maximum value of R_p to be used for finding the maximum possible efficiency in the equation (24.1) is given by the equation (24.3).

We are also interested to find the pressure-ratio which gives maximum available work per kg of working fluid.

$$W_a = (W_t - W_c) = C_p (T_3 - T_4) - C_p (T_2 - T_1)$$

$$= C_p T_3 \left[1 - \frac{1}{(R_p)^{(\gamma-1)/\gamma}}\right] - C_p T_1 [(R_p)^{(\gamma-1)/\gamma} - 1] \quad \text{...(24.4)}$$

For the given temperature limits (T_1 and T_3 are constant), W_a becomes maximum when

$$\frac{dW_a}{dR_p} = 0$$

$$\therefore \quad \frac{d}{dR_p} [C_p T_3 - C_p T_3 (R_p)^{-(\gamma-1)/\gamma} - C_p T_1 (R_p)^{(\gamma-1)/\gamma} + C_p T_1] = 0$$

Solving the above equation, we get

$$R_p = \left(\frac{T_3}{T_1}\right)^{\frac{1}{2}\gamma/(\gamma-1)} \quad \text{...(24.5)}$$

Substituting the value of R_p from equation 24.5 into equation 24.4, we get,

$$W_a = C_pT_3 \left[1 - \frac{1}{\left[\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}\{\gamma/(\gamma-1)\}}\right]^{(\gamma-1)/\gamma}}\right] - C_pT_1 \left[\left(\frac{T_3}{T_1}\right)^{\frac{1}{2}\{\gamma/(\gamma-1)\}}\right]^{(\gamma-1)/\gamma} - 1$$

$$= C_pT_3 \left(\frac{\sqrt{T_3} - \sqrt{T_1}}{\sqrt{T_3}}\right] - C_pT_1 \left[\frac{\sqrt{T_3} - \sqrt{T_1}}{\sqrt{T_1}}\right] = C_p \left[(T_3 - \sqrt{T_3T_1}) - (\sqrt{T_1T_3} - T_1)\right]$$

$$\therefore \quad W_a\,(\max) = C_p\,[T_1 + T_3 - 2\sqrt{T_1T_3}] \qquad ...(24.6)$$

This indicates that the maximum work output is dependent on the temperature range of the working cycle and highly influencing thermodynamic property is the C_p of the working fluid.

Comparing the Eqs. (24.3) and (24.5), we get

$$R_p = \sqrt{(R_p)_{max}} \qquad ...(24.7)$$

The variations of thermal efficiency and specific output are shown in Fig. 24.6.

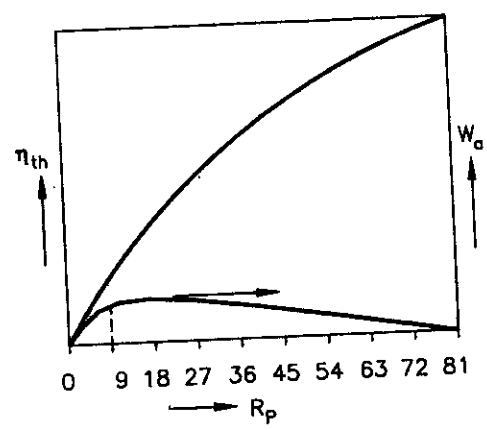


Fig. 24.6. Variation of W_a and η_{th} V_sR_p.

The choice of optimum pressure ratio for maximum work output (W_a) is an important feature in the design of gas turbine power plant. This is because, high pressure ratio is not necessarily required.

Say $T_2 = 1000$ K and $T_1 = 285$ K

then $(R_p)_{max} = 81$ if air is used as working fluid then R_p (for maximum specific work output)

$$= \sqrt{81} = 9$$

This is the main reason for using low pressure ratios (5 to 8) in gas turbine plant compared with the diesel plant. Higher thermal efficiency can be achieved with higher pressure ratio but at the expense of loss in specific work output. This further increases the capital cost of the plant as the size of the plant increases with the decrease in specific work output.

Choice of the Working Fluid. It has been seen that the thermal efficiency and maximum specific work output are highly influenced by the thermodynamic properties of the working fluid, namely, γ and C_p.

The problem of choosing an effective working medium and coolant in closed cycle gas turbine plant is one of the real facing problems. For nuclear power plants with gas cooled reactors, the most important investigations are concerned with helium and CO_2. Both helium and CO_2 satisfy the requirements for working media and coolants for nuclear plants. Helium possesses good heat transfer properties but the work spent on compressing the helium is not much less than the work of its expansion in the turbine. The work ratio is considerably less for CO_2 than for helium which favourably affects the useful efficiency of the cycle. However, due to the low thermal conductivity of CO_2, the heat transfer surfaces of the heat exchangers must be considerably greater than those for helium.

The molecular mass of these gases differs by an order and thermal conductivity of their mixture is determined by the volumetric contents of the components. The ratio W_c/W_t is determined by the mass proportions enables one to select a composition of the mixture such that the favourable properties of the components are developed to the fullest extent and the use of mixture of He and CO_2 as a working medium is more effective than the use of each of component separately.

By increasing the amount of CO_2 in the mixture of (CO_2 + He), the thermal conductivity of the working medium falls and there is a fall in the gas temperature at the outlet of the reactor, which leads to a fall in efficiency of the cycle. On the other hand, reduction in W_c/W_t causes an increase in the efficiency of the cycle. This trend of a change in the cycle efficiency with a change in composition of the mixture leads to the fact that with the mass composition of (0.9 CO_2 + 0.1 He), the efficiency of the regenerative Brayton cycle is maximum.

The ability of a plant to preserve the design efficiency under non-design conditions of operation depends to a large extent on the properties of the working medium (on the composition of the mixture). The best stability of the efficiency characteristics is possessed by a closed cycle gas turbined plant using ideal gas as working medium, therefore, with an increase in the proportion of CO_2 in the mixture, the efficiency and stability characteristics deteriorate. The relative merits of the He, CO_2 and the optimum mixture of the two are listed in the following table.

Thermodynamic Properties of He, CO and Mixture

	Working Medium		
	He	CO_2	(0.9 CO_2 + 0.1 He)
Thermodynamically optimal inlet turbine temp. K	950	878	980
Efficiency of cycle	31.2%	27.8%	36.4%
Total gas-dynamic loss of the cycle	9.5%	12%	10.3%
Relative specific surface of regenerators	1.0	2.45	1.0
Relative mass of turbine	1.0	0.74	0.58
Relative economic characteristics	1.0	0.87	0.92

The compression and expansion do not follow the isentropic law, therefore, the isentropic efficiencies should be taken into account in analysis.

The isentropic efficiencies of compressor and turbine are given by

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

and $$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$$

If these efficiencies are taken into account then, we get

R_p (Pressure ratio for maximum work output)

$$= \left[\eta_c \eta_t \frac{T_3}{T_1}\right]^{\gamma/2(\gamma-1)} \quad ...(24.8)$$

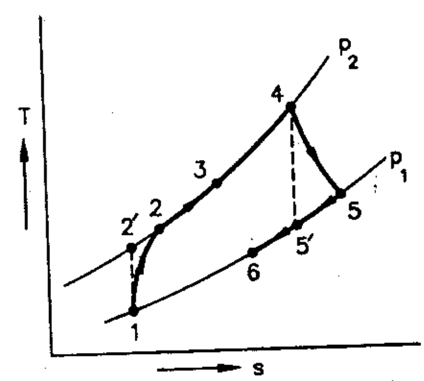


Fig. 24.7. Actual processes in compressor and turbine of a gas turbine plant.

and R_p (for maximum thermal efficiency) $$= \left[\frac{T_3/T_1}{1 + \sqrt{\left(\frac{T_3}{T_1} - 1\right)\left(\frac{1}{\eta_c \cdot \eta_t} - 1\right)}}\right]^{\gamma/(\gamma-1)} \quad ...(24.9)$$

The analysis of closed cycle plant can be used for open cycle plant if the fuel quantity is neglected. Many times, the fuel mass is always neglected as the air-fuel ratio used in the gas turbine plant is considerably large as 100 : 1.

Large air-fuel ratio is necessary in gas turbine plant because the temperature of hot gases formed in the combustion chamber is considerably high (2000°C). Therefore, it is necessary to reduce the temperature from 2000°C to 800°C by adding the secondary air into the gases coming out from the combustion chamber for the reason mentioned earlier. The quantity of secondary air required is considerably large, therefore, the air-fuel ratio used in gas turbine plant is also large.

The low pressure ratio and high air-fuel ratio are the specific characteristics of the gas turbine plant compared with the diesel electric plant.

Analysis of open cycle plant. The masses of air and fuel passing through the open cycle gas turbine plant per second are m_a and m_f.

Referring to Fig. 24.7.

W_t (turbine work per second) = $(m_a + m_f)\, C_{pg}\, (T_3 - T_4)$

W_t (compressor work per second) = $m_a C_{pa}\, (T_2 - T_1)$

where C_{pg} and C_{pa} are the specific heats of gases and air (kJ/kg-°C).

$$\therefore \quad W_a = W_t - W_c = (m_a + m_f)\, C_{pg}\, (T_3 - T_4) - m_a C_{pa}\, (T_2 - T_1) \qquad \text{...(24.10)}$$

Neglecting the heat losses in the combustion chamber and ducts and assuming 100% combustion efficiency, the heat supplied per second is given by

$$Q_s = m_f \times \text{C.V.}$$

where C.V. is the calorific value of fuel used and m_f is the mass of fuel/sec.

The thermal efficiency of the cycle is given by

$$\eta_{th} = \frac{W_a}{Q_s} = \frac{(m_a + m_f)\, C_{pg}\, (T_3 - T_4) - m_a C_{pa}\, (T_2 - T_2)}{m_f \times \text{C.V.}} \qquad \text{...(24.11)}$$

the electric power developed by the system in kW is given by

$$\text{kW} = [(m_a + m_f)\, C_{pg}\, (T_3 - T_4) - m_a C_{pa}\, (T_2 - T_1)]\, \eta_m \cdot \eta_g \qquad \text{...(24.12)}$$

where η_m and η_g are mechanical and generation efficiencies and kW is the electrical power developed by the plant.

Now, onwards, we will deal with only open cycle plant as it is commonly used in practice for power generation purposes.

24.5. METHODS TO IMPROVE THE THERMAL EFFICIENCY OF A SIMPLE OPEN CYCLE CONSTANT PRESSURE GAS TURBINE POWER PLANT

The thermal efficiency of a simple cycle gas turbine plant can be increased by using one of the following methods :

(1) By reducing the work required to run the compressor,

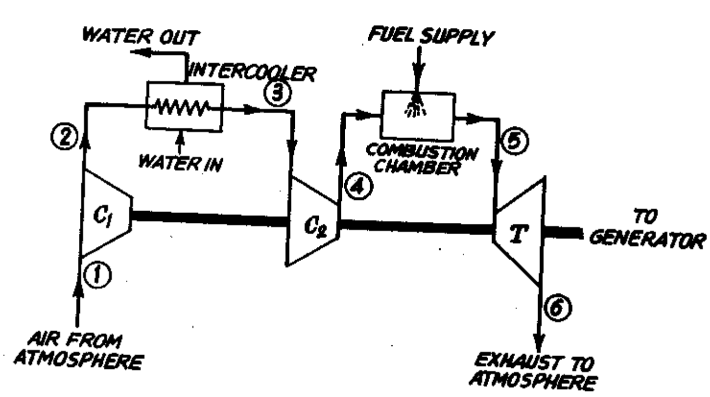


Fig. 24.8 (*a*). Gas turbine plant with inter-cooling.

(2) By reducing the heat (fuel) supplied in the combustion chamber.

1. **Intercooling.** Major percentage of power developed (66%) by the turbine is used to run the compressor. The power required to run the compressor can be reduced by compressing the air in two stages and incorporating the intercooler between the two as shown in Fig. 24.8 (*a*). The corresponding $T-s$ diagram is shown in Fig. 24.8 (*b*).

Considering actual compressions in both stages and imperfect intercooling, the work done per kg of air by both compressors is given by

$$W_c = C_{pa}(T_2 - T_1) + C_{pa}(T_4 - T_3)$$

η_{c1} and η_{c2} are isentropic efficiencies of compressors C_1 and C_2

T_3 (temperature after cooling) > T_1

$$W_c = \frac{C_{pa}}{\eta_{c1}}(T_2' - T_1) + \frac{C_{pa}}{\eta_{c2}}(T_4' - T_3)$$

$$= \frac{C_{pa}T_1}{\eta_{c1}}\left[\left(\frac{p_i}{p_1}\right)^{(\gamma-1)/\gamma} - 1\right] + \frac{C_{pa}T_3}{\eta_{c2}}\left[\left(\frac{p_2}{p_i}\right)^{(\gamma-1)/\gamma} - 1\right]$$

$$= \frac{C_{pa}T_1}{\eta_{c1}}\left[\left(\frac{p_i}{p_1}\right)^{(\gamma-1)/\gamma} - 1\right] + \frac{C_{pa}kT_1}{\eta_{c2}}\left[\left(\frac{p_2}{p_i}\right)^{(\gamma-1)/\gamma} - 1\right]$$

where $k = \dfrac{T_3}{T_1}$ ...(24.13)

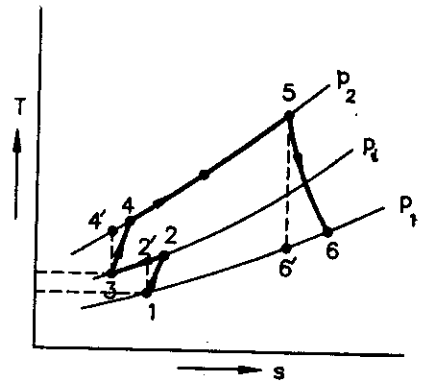


Fig. 24.8 (*b*). $T - s$ diagram for Fig. 24.8 (*a*).

We are interested to find out the minimum work required for compressing the air in two stages and the required condition for this is

$$\frac{dW_c}{dp_i} = 0.$$

Completing the derivation and simplyfying, we get

$$P_i = \sqrt{p_1 p_2 \left(k\frac{\eta_{c1}}{\eta_{c2}}\right)^{\gamma/(\gamma-1)}} \qquad ...(24.14)$$

A perfect intercooler is defined as a heat exhanger which reduces the temperature of air from T_2 to its original temperature T_1.

If the intercooling is perfect, then

$$T_3 = T_1 \qquad \therefore \qquad k = 1$$

and if $\eta_{c1} = \eta_{c2}$

then the intermediate pressure for the minimum work required in the compressors is given by

$$P_i = \sqrt{P_1 P_2} \qquad ...(24.15)$$

Assuming $k = 1$ and $\eta_{c1} = \eta_{c2} = \eta_c$

and substituting the value of P_i from Eq. (24.15) into Eq. (24.13), we get,

$$W_c = \frac{2C_{pa}T_1}{\eta_c}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{1}{2}[(\gamma-1)/\gamma]} - 1\right\} \qquad ...(24.16)$$

The work with intercooler is given by $W_a = (W_t - W_c)$.

W_t remains same as the inlet temperature to the turbine and overall ratio with or without intercooler remains same. But W_c is reduced by intercooling, and, therefore, helps to increase the net output per kg of air.

The extra heat is required to raise the temperature of air to T_5.

The net effect of compressing the air (saving in compressor work) in two stages with intercooling will be economical only when $\Delta W_c > \Delta Q_s$ (extra heat required). Generally it has been found in practice that ΔW_c is greater than ΔQ_s.

The net specific output per kg of air circulated $(W_t - W_c)$ with intercooling is more than without intercooling, therefore, the capital cost of the plant is less.

The number of stages used for compression depends upon the overall pressure ratio used for the power plant. More than two stages with intercooler after every stage can be used for high pressure ratios. The number of stages used are generally decided as the saving in work done and extra capital required for the equipments used as compressors and intercoolers.

2. **Regeneration.** The exhaust gases carry lot of heat as their temperature is far above the ambient temperature. Therefore, the heat of the exhaust gases can be given to the air coming out of compressor and, therefore, the mass of fuel supplied in the combustion chamber can be reduced.

The gas turbine plant with regenerator is shown in Fig. 24.9 (*a*) and its corresponding $T - s$ is shown in Fig. 24.9 (*b*).

It is theoretically possible to raise the temperature of the compressed air coming out from the compressors from T_2 to $T_5 = T_6$ and lower the temperature of the gases coming out from the turbine from T_4 to $T_6 = T_2$ passing both the fluids through a counter flow heat exchanger.

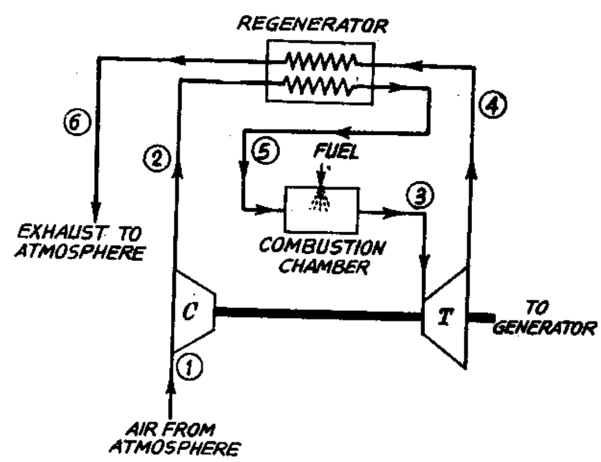


Fig. 24.9 (*a*). Gas turbine cycle with regenerator.

With the use of regenerator in the circuit, there is no change in the compressor and turbine work but the quantity of fuel supplied is substantially reduced as the temperature of the air entering the combustion chamber is increased. Therefore, the thermal efficiency of the regenerative cycle is higher than the simple cycle.

In practice, it is not possible to increase a temperature of air to T_5 and it is always less than T_5 say T_5'. This is because of inefficiency and high capital cost required for the heat exhanger (regenerator).

$$\eta_{th} = \frac{C_{pa}(T_3 - T_4) - C_{pa}(T_2 - T_1)}{C_{pa}(T_3 - T_5')} \quad ...(24.17)$$

The above expression is true only if the mass of the fuel is neglected and $C_{pa} = C_{pg}$ is assumed.

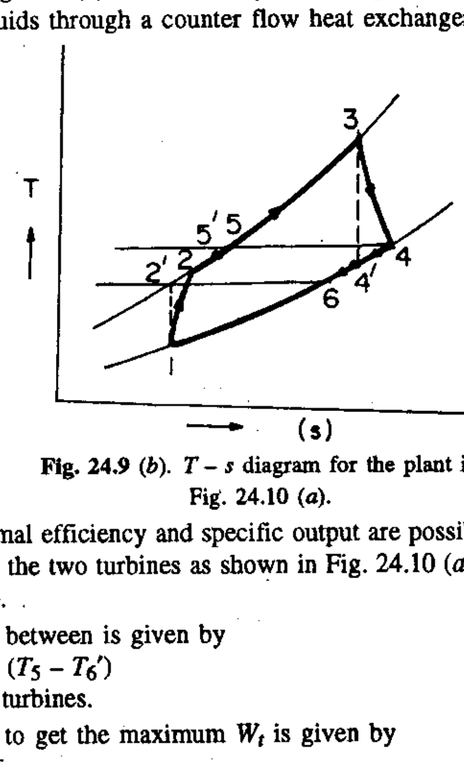


Fig. 24.9 (*b*). $T - s$ diagram for the plant in Fig. 24.10 (*a*).

3. **Reheating.** A considerable increase in the thermal efficiency and specific output are possible by expanding the gases in two stages with reheater between the two turbines as shown in Fig. 24.10 (*a*). The corresponding $T - s$ diagram is shown in Fig. 24.10 (*b*).

The work done by two turbines with reheating in between is given by

$$W_t = C_{pg}\eta_{t1}(T_3 - T_4') + C_{pg}\eta_{t2}(T_5 - T_6')$$

where η_{t1} and η_{t2} are the isentropic efficiencies of the turbines.

The best intermediate pressure used for reheating to get the maximum W_t is given by

$$P_i = \sqrt{P_1 P_2 \left(K \cdot \frac{\eta_{t2}}{\eta_{t1}}\right)^{\gamma/(\gamma-1)}} \quad ...(24.18)$$

where $K = \frac{T_5}{T_3}$.

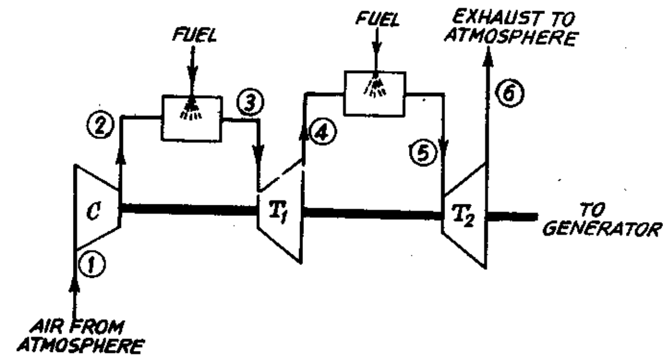


Fig. 24.10. (*a*) Gas turbine plant with reheating.

If $\eta_{t1} = \eta_{t2}$ and $K = 1$ (reheating to its original temperature) then the best intermediate pressure for reheating is given by

$$P_i = \sqrt{P_1 P_2} \quad \text{...(24.19)}$$

The thermal efficiency of the turbine plant as shown in Fig. 24.10 (*a*) with reheating is given by

$$\eta_{th} = \frac{(T_3 - T_4) + (T_5 - T_6) - (T_2 - T_1)}{(T_3 - T_2) + (T_5 - T_4)} \quad \text{...(24.20)}$$

if it is assumed that $C_{pa} = C_{pg}$ and fuel mass is neglected.

In actual power plant, the intercooling, regeneration and reheating are used to increase the overall thermal efficiency of the plant and specific power output.

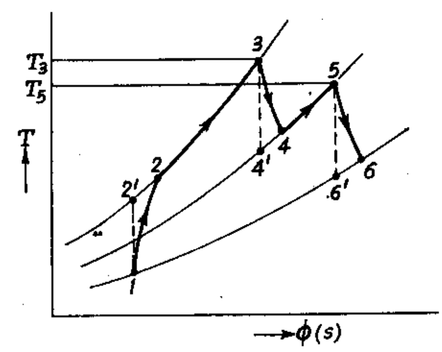


Fig. 24.10. (*b*) $T - s$ diagram for the plant shown in Fig. 24.10 (*a*).

The arrangement of the system is shown in Fig. 24.11 (*a*) and corresponding $T - s$ diagram is shown in Fig. 24.11 (*b*).

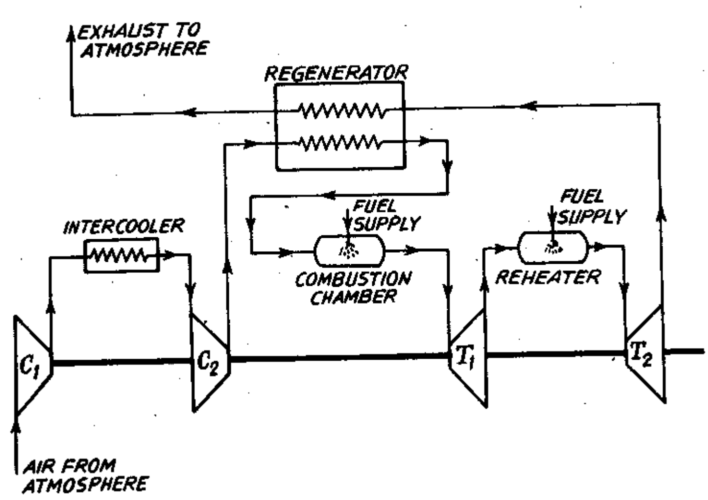


Fig. 24.11. (*a*) Gas turbine plant with intercooler, regenerator and reheater in the plant.

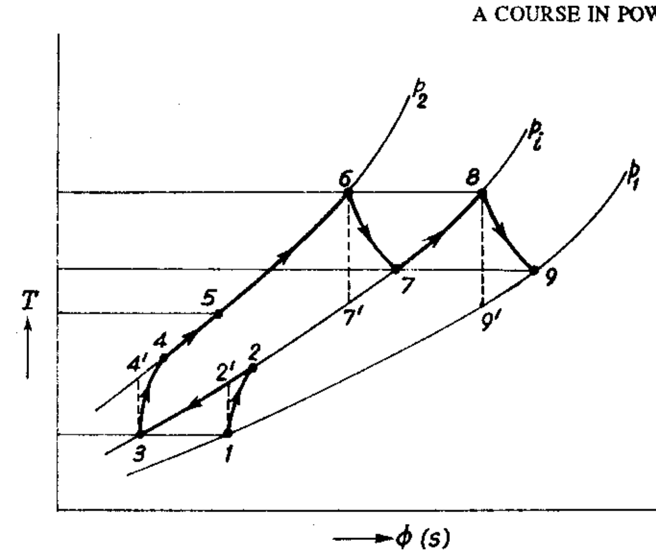


Fig. 24.11 (*b*). $T-s$ diagram for the plant shown in Fig. 24.11 (*a*).

In drawing the $T-s$ diagram, it is assumed that there is perfect intercooling ($T_1 = T_3$) and perfect reheating is done to its original temperature ($T_8 = T_6$).

It is further assumed that $\eta_{c1} = \eta_{c2}$ and $\eta_{t1} = \eta_{t2}$, and pressure losses in the system are neglected.

For the above assumptions, the best intermediate pressure p_i required for intercooling and reheating is same and it is given by

$$p_i = \sqrt{p_1 p_2}$$

The work developed per kg of air entering into the compressor C_1 by the gas turbine plant is given by

$$W_a = [C_{pg}\,(1 + m_{f1})\,(T_6 - T_7) + C_{pg}\,(1 + m_{f1} + m_{f2})\,(T_8 - T_9)] - [C_{pa}\,(T_2 - T_1) + C_{pa}\,(T_4 - T_3)] \quad ...(24.21)$$

where m_{f1} and m_{f2} are the fuel quantities supplied per kg of air flow through the compressor in the combustion chamber and reheater.

The net heat supplied to the system per kg of air flow is given by

$$Q_s = C_{pg}\,(T_6 - T_5) + C_{pg}\,(1 + m_{f1})\,(T_8 - T_7) \quad ...(24.22)$$

The fuel quantities m_{f1} and m_{f2} are calculated by using the following equations :

$$C_{pa}\,(T_6 - T_5) = m_{f1} \times \text{C.V.} \times \eta_{com1} \quad ...(24.23)$$

and

$$C_{pg}\,(1 + m_{f1})\,(T_8 - T_7) = m_{f2} \times \text{C.V.} \times \eta_{com2} \quad ...(24.24)$$

where C.V. is the calorific value of the fuel and η_{com1} and η_{com2} are the combustion efficiencies in combustion chamber and reheater respectively.

24.6. COMPONENTS OF GAS TURBINE PLANT

It is always necessary for the engineers and designers to know about the construction and operation of the components of gas turbine plants.

1. **Compressors.** The high flow rates of turbines and relatively moderate pressure ratios necessitate the use of rotary compressors. The types of compressors which are commonly used are of two types, centrifugal and axial flow types. .

The centrifugal compressor consists of an impeller (rotating component) and a diffuser (stationary component). 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 three 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. The single inlet centrifugal compressor is shown in Fig. 24.12. The efficiency of centrifugal compressor lies between 80 to 90%. The efficiency of multistage compressor is lower than a single stage due to the losses.

The 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 rotor and rotor is 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. The axial flow compressor is shown in

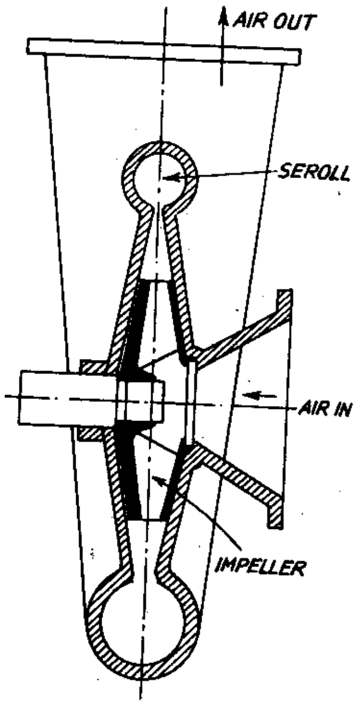


Fig. 24.12. Single stage single entry centrifugal compressor.

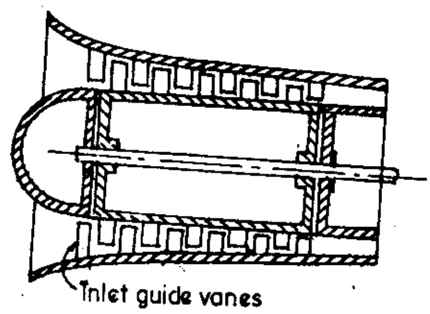


Fig. **24.13.** Axial flow air compressor.

Fig. 24.13. The number of stages required for pressure ratio of 5 is as large as sixteen or more.

A satisfactory air filter is absolutely necessary for cleaning the air before it enters the compressor because it is essential to maintain the designed profile of the aerofoil blades. The deposition of dust particles on the blade surfaces reduces the efficiency rapidly.

The advantages of axial flow compressor over centrifugal compressor are high isentropic efficiency (90 – 95%), high flow rate and small weight for the same flow quantity. The axial flow compressors are very sensitive to the changes in air flow and speed which result in rapid drop in efficiency.

In both types of compressors, it has been found that lowering of the inlet air temperature by 15 to 20°C gives almost 25% greater output with an increase of 5% efficiency.

2. **Intercoolers and heat exchangers.** The intercooler is generally used in gas turbine plant 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 is generally preferred for effective heat transfer.

The regenerators which are commonly used in gas turbine plant are of two types, recuperator and regenerator.

In a recuperative type of heat exchanger, the air and hot gases are made to flow in counter direction as the effect of counter-flow gives high average temperature difference causing the higher heat flow.

A number of baffles in the path of air flow are used to make the air to flow in contact for longer time with heat transfer surface.

The regenerator type heat exhanger consists of a heat conducting member which is exposed alternately to the hot exhaust gases and the cooler compressed air. It absorbs the heat from hot gases and gives it up when exposed to the air. The heat capacity member is made of a metallic mesh or matrix which is rotated slowly (40 – 60 r.p.m.) and continuously exposed to hot and cold air.

The first application of regenerative heat exchanger to gas turbine plants was suggested by Prof. Ritz

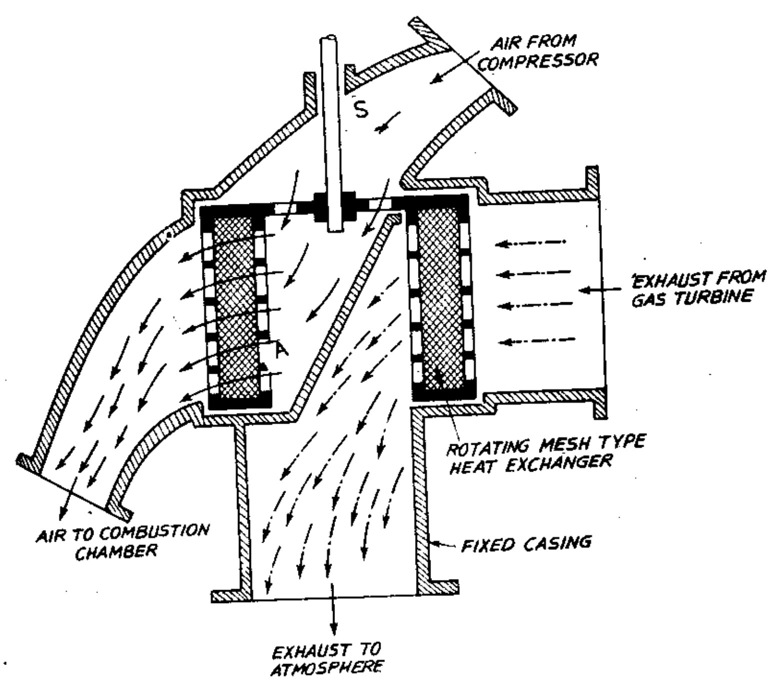


Fig. 24.14. Ritz regenerative heat exchanger.

of Germany and the heat exchanger was titled against his name. The arrangement of Ritz heat exchanger is shown in Fig. 24.14.

The heat exchanging element *A* is slowly rotated by a drive from the gas turbine *via* shaft *S*. The rotation places the heat transferring element *A* in the exhaust gas passage for one half of the time required for one r.p.m. and in the air supply passage for the remaining half. The heat element absorbs heat from the hot gases, when exposed to hot gases and gives out the same heat to the cold air when the heated part moves in the air region. By suitable design of the speed of rotation of transfer element and its mass in relation to the heat to be transferred, it is possible to secure a high effectiveness, values of 90% are claimed. The principal advantages claimed of this heat-exchanger over the recuperative type are lightness, smaller mass, small size for given effectiveness and low pressure drop.

The major disadvantage of this heat exchanger is, there will be always a tendency for air leakage to the exhaust gases as the compressed air is at a much higher pressure than exhaust gases. This tendency of leakage reduces the efficiency gain due to heat exchanger. Therefore, the major problem in the design of this type of heat exchanger is to prevent or minimise the air loss due to leakage.

Recently very special seals are provided to prevent the air leakage. This seal stands at very high temperature and pressure and allows the freedom of movement.

The performance of the heat exchanger is determined by a factor known as effectiveness. The effectiveness of the heat exchanger is defined as

$$\varepsilon = \frac{\text{actual heat transfer to the air}}{\text{maximum heat transfer theoretically possible}}$$

The effectiveness is given by (refer Fig. 24.9 *b*)

$$\varepsilon = \frac{C_{pa}\, m_a\, (T_5 - T_2)}{C_{pg}\, m_g\, (T_4 - T_2)}$$

where m_a and m_g are the masses of the air and exhaust gases and C_{pa} and C_{pg} are the corresponding specific heats.

If the mass of the fuel compared with mass of the air is neglected and $C_{pa} = C_{pg}$ is assumed, then the effectiveness is given by an expression

$$\varepsilon = \frac{T_5 - T_2}{T_4 - T_2}.$$

Combustion chambers. The gas turbine is a continuous flow system, therefore, the combustion in the gas turbine differs from the combustion in diesel engines. High rate of mass flow results in high velocities at various points throughout the cycle (300 m/sec). One of the vital problems associated with the design of gas turbine combustion system is to secure a steady and stable flame inside the combustion chamber.

The gas turbine combustion system has to function under certain different operating conditions which are not usually met with the combustion systems of diesel engines. A few of them are listed below :

(1) Combustion in the gas turbine takes place in a continuous flow system and, therefore, the advantage of high pressure and restricted volume available in diesel engine is lost. The chemical reaction takes place relatively slowly thus requiring large residence time in the combustion chamber in order to achieve complete combustion.

(2) The gas turbine requires about 100 : 1 air-fuel ratio by weight for the reasons mentioned earlier. But the air-fuel ratio required for the combustion in diesel engine is approximately 15 : 1. Therefore, it is impossible to ignite and maintain a continuous combustion with such weak mixture. It is necessary to provide rich mixture for ignition and continuous combustion, and therefore, it is necessary to allow required air in the combustion zone and the remaining air must be added after complete combustion to reduce the gas temperature before passing into the turbine.

(3) A pilot or recirculated zone should be created in the main flow to establish a stable flame which helps to ignite the combustible mixture continuously.

(4) A stable continuous flame can be maintained inside the combustion chamber when the stream velocity and fuel burning velocity are equal. Unfortunately, most of the fuels have low burning velocities of the order of a few metres per second, therefore, flame stabilization is not possible unless some technique is employed to anchor the flame in the combustion chamber.

There are many flame stabilization techniques but to discuss all, is out of the scope of this book. The common methods of flame stabilization used in practice are bluff body method and swirl flow method.

Two types of combustion chambers using bluff body and swirl for flame stabilization are shown in Fig. 24.15 (*a*) and Fig. 24.15 (*b*). The major difference between two is the use of different methods to create the pilot zone for flame stabilization.

Nearly 15 to 20% of the total air is passed around the jet of fuel providing rich mixture in the primary zone. This mixture burns continuously in the primary (pilot) zone and produces high temperature gases. About

30% of the total air is supplied in the secondary zone through the annuals around the flame tube to complete the combustion. The secondary air must be admitted at right points in the combustion chamber otherwise

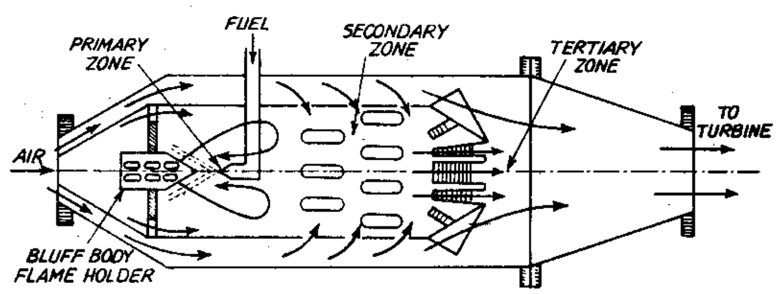


Fig. 24.15 (*a*). Combustion chamber with upstream injection with bluff-body flame holder.

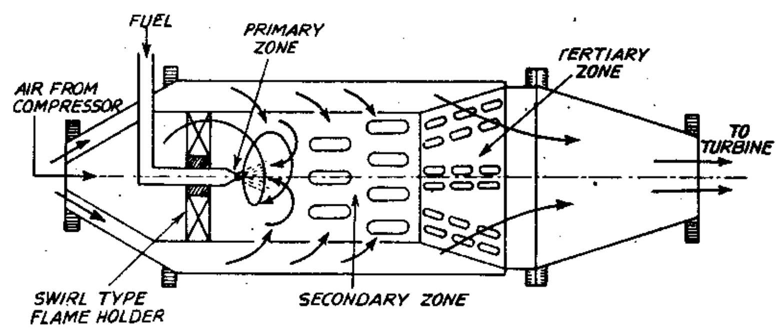


Fig. 24.15 (*b*). Combustion chamber with downstream injection and swirl holder.

the cold injected air may chill the flame locally thereby reducing the rate of reaction. The secondary air helps to complete the combustion as well as helps to cool the flame tube. The remaining 50% air is mixed with burnt gases in the "tertiary zone" to cool the gases down to the temperature suited to the turbine blade materials.

By inserting a bluff body in main stream, a low pressure zone is created downstream side which causes the reversal of flow along the axis of the combustion chamber to stabilize the flame.

In case of swirl stabilization, the primary air is passed through the swirler which produces a vortex motion creating a low pressure zone along the axis of the chamber to cause the reversal of flow.

Sufficient turbulence must be created in all three zones of combustion and uniform mixing of hot and cold gases to give uniform temperature gas stream at the outlet of the combustion chamber.

4. **Gas turbines.** The common type of turbines which are in use are axial flow type. The basic requirements of the turbines are light weight, high efficiency ; reliability in operation and 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. More stages are further preferred with stationary power plants because weight is not the major consideration in the design which is essential in aircraft turbine-plant.

The cooling of the gas turbine blades is essential for long life as it is continuously subjected to high temperature gases. There are different methods of cooling the blades. The common method used is the aircooling. The air is passed through the holes provided through the blade.

24.7. DIFFERENT ARRANGEMENTS OF PLANT COMPONENTS

1. **Single Shaft Turbine.** With single shaft turbine, the compressor, turbine and generator are mounted on a single shaft as shown in Fig. 24.16. In this arrangement, the compressor and turbine rotate at the same speed.

This arrangement is cheaper to construct but suffers certain disadvantages. This arrangement is less efficient and more difficult to control and operate at high speeds than others.

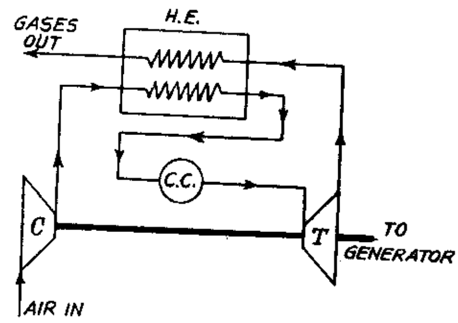


Fig. 24.16. Single shaft arrangement.

2. **Free Power Turbine.** In order to overcome some of the disadvantages of single shaft arrangement, the generator is driven by a separate turbine as shown in Fig. 24.17.

The free power turbine arrangement offers some advantages over single shaft arrangement as listed below :

(*a*) It allows the compressor-turbine unit to operate at its best performance speed and the free turbine can run at appropriate speed and provides flexibility of control.

(*b*) This system can be started more easily from cold as the starter has to drive the compressor-turbine unit leaving the power turbine free and its output shaft remains stationary.

(*c*) The absence of mechanical coupling between the compressor-turbine and generator turbine helps the latter to be accelerated more rapidly from idling to maximum speed.

(*d*) It is possible to vary the free turbine shaft speed over a wide range at constant compressor speed.

This arrangement suffers from the disadvantage of low thermal efficiency of the power turbine at part load conditions as the inlet temperature of the power turbine is reduced due to the variation of fuel supply in the main combustion chamber.

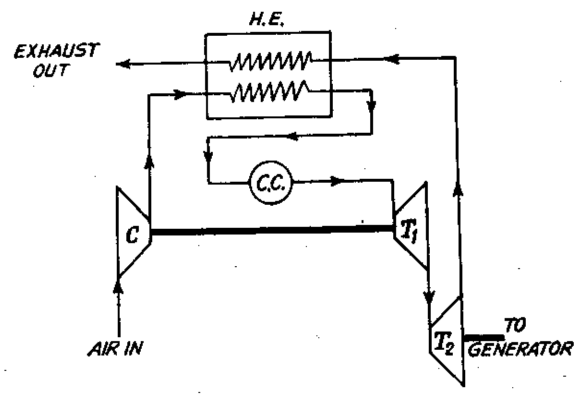


Fig. 24.17. Single shaft arrangement.

3. **Free Power Turbine with Combustion Chambers in Series.** To overcome the difficulty mentioned above, the other two arrangements as shown in Figs. 24.18 and 24.19 are used.

In the arrangement shown in Fig. 24.18, two combustion chambers are used, one for each turbine. The control as per the load on generator can be achieved by adjusting the fuel supply to the combustion chamber of power turbine and constant running conditions of compressor-turbine unit are maintained to achieve the better efficiency conditions throughout the load variations. As the flow is continuous, the system is known as series flow.

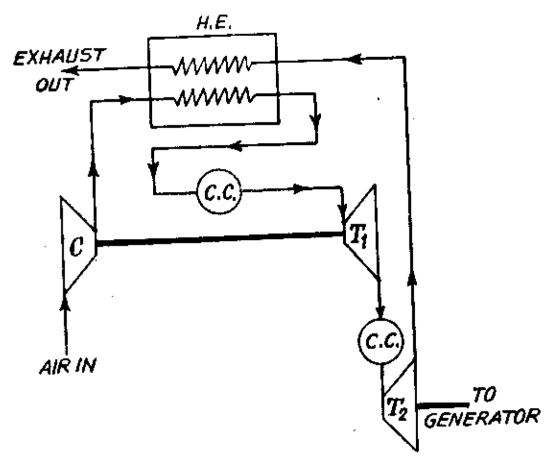


Fig. 24.18. Two shafts arrangement with series combustion chambers.

4. **Free Power Turbine with Separate Combustion Chambers in Parallel.** The arrangement shown in Fig. 24.18 suffers from the disadvantage of requiring big power turbine (high capital cost) as all the air coming out of the compressor is passed through the power-turbine. To overcome this difficulty, the arrangement shown in Fig. 24.19 is preferred. In this arrangement, the air coming out from the compressor is subdivided as per the requirement of power turbine and compressor turbine. This reduces the size of the power turbine as the quantity of air passed through the power turbine in this arrangement is less than previous.

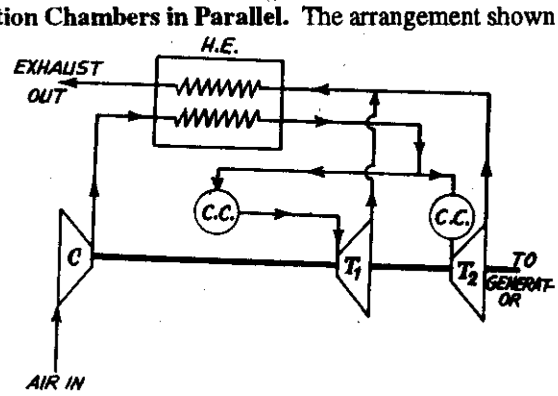


Fig. 24.19. Two shafts arrangement with parallel combustion chambers.

5. **Straight compounding.** It is always preferable to divide the compressor into two stages to avoid the instability of the plant during starting and when operating at part loads. Such arrangement is shown in Fig. 24.20.

The low pressure turbine drives the low pressure compressor and high pressure turbine drives high pressure compressor. The power required to drive the generator is taken from L.P. turbine. The ability of two shafts to adjust their speeeds independently improves the part load performance and helps for easy starting.

6. **Cross-compounding.** In this method of arrangement, low pressure compressor is driven by high pressure turbine and high pressure compressor by low pressure turbine. The arrangement is shown in Fig. 24.21. The main advantage of this arrangement over the previous one is better part load efficiency.

The straight compounding is generally used for base load plants where cross-compounding is preferred for peak load plants.

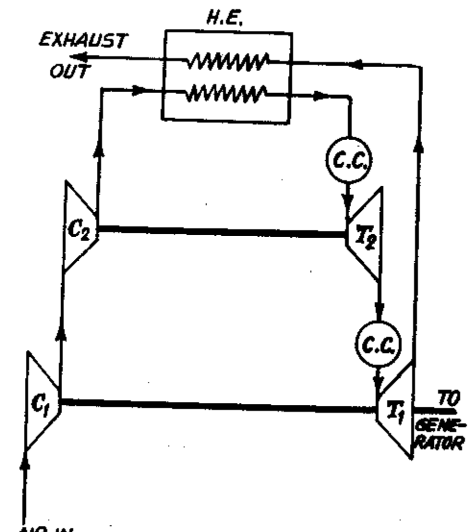


Fig. 24.20. Straight compounding.

24.8. GOVERNING SSYSTEM FOR GAS TURBINE POWER PLANT

The efficiency of the gas turbine decreases with the decrease in load when the plant is used as peak load plant. But in all circumstances, it is necessary to maintain the speed of the turbine (or alternator) constant.

When the high pressure turbine drives the compressor and exhausts to low pressure turbine and alternator is coupled to L.P. turbine, its speed remains constant at all loads and in consequence, the air volume remains constant but the inlet temperature becomes too low at part loads and efficiencies are poor.

When the heat exchanger is used in the system, it is always advantageous to couple the alternator to H.P. turbine in order to obtain better efficiencies at part loads. The turbine plant is governed by varying the quantity of fuel injected into the combustion chamber.

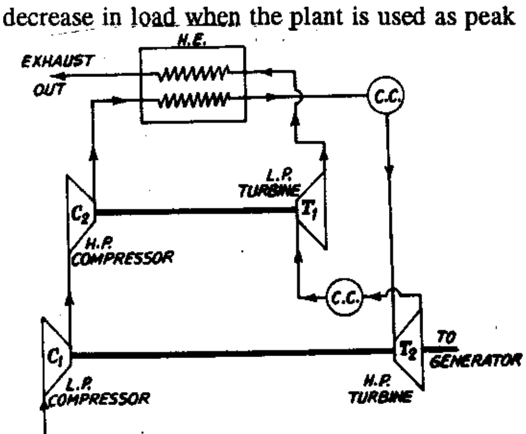


Fig. 24.21. Cross-compounding.

The governing system must be able to provide proper amount of fuel for starting the plant from cold, afterwards increase the fuel supply to accelerate the engine to its normal speed. The provision must be made in the governing system to prevent overspeeding and excessive turbine inlet temperatures which might wreck the turbine unit in the event of sudden reduction in load. A fuel control adjustment must be provided to take into account ambient air temperature and pressure changes. Finally, a control device must be included in the system to shut off the fuel for stopping purposes.

A typical governing method developed by Brown Boveri for 10 MW capacity plant is shown in Fig. 24.22.

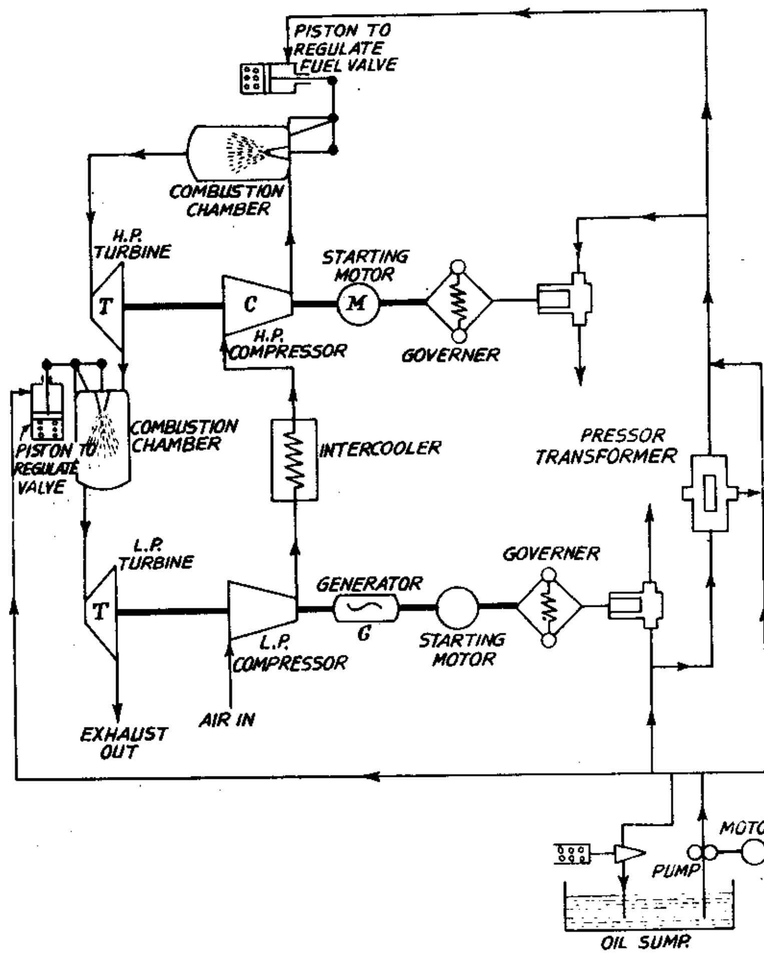


Fig. 24.22. Governing system for 10 MW plant when separate turbine is used for generation.

In this system, the speed of a turbine which drives the generator is kept constant by a centrifugal governor. If the speed increases due to decrease in load on generator, the governor decreases the pressure of the relay oil and fuel valve is closed a little more to supply less fuel into the combustion chamber, so that the low pressure turbine is brought to its original speed. The speed of the high pressure set is not regulated and it is independent of load variations and corresponds to the power equilibrium between the turbine and compressor. If the speed exceeds the permissible limit, a device comes into action to reduce the fuel quantity injected into high pressure combustion chamber.

The governor of two-stage turbine operates more quickly and accurately. It was recorded that the largest speed variation was 8% and new steady conditions were attained within 35 seconds.

The control system used for closed cycle plant is more simplified than open cycle because the variation of the power output is obtained by merely varying the quantity of working fluid and with practically no reduction in the thermal efficiency of the plant. The pressures are varied in the same ratio at every point of the cycle while the flow velocities and temperatures remain unaltered. Therefore, only the density of the working fluid is varied for governing purposes.

Governing of closed cycle gas turbine plant. The governing of the closed cycle system can be done by simply varying the pressure in the system. Figure 24.23 shows the systematic arrangement of the governing components.

As the load decreases on the system with the consequent rise in speed, the centrifugal governor causes the discharge valve to release air into the low pressure accumulator. When the load on the plant increases causing consequent decrease in speed, the governor acts to admit air into the system (plant) from the high pressure accumulator. It is also necessary to decrease or increase the fuel supply with the decrease or increase in load. This is accomplished by controlling the fuel supplied to the air heater.

For small changes in load (5 to 10%), it is possible to control the output without making the use of air accumulators and changing the pressure of the system. This is accomplished by operating the air by-pass from the high pressure side of the circuit to the low-pressure side of the circuit. This arrangement of bypassing the air from high to low pressure side reduces the useful work developed by the system.

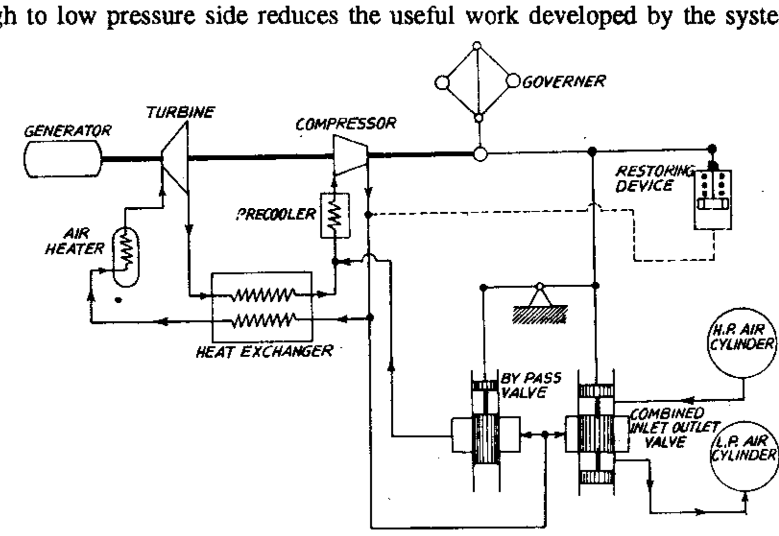


Fig. 24.23. Centrifugally controlled regulating device for single shaft closed cycle power plant.

The method mentioned above for governing the small load variations is not economical as it is a waste of high pressure air. But it is less expensive than using the air from accumulator as the accumulated high pressure air also requires compression work.

High response of the bypass system is an added advantage over the control made by the accumulator system.

24.9. INDIAN GAS TURBINE POWER PLANTS

70 MW Gas Turbine Plant at Namrup in Assam

The basic requirement for developing nations is low cost power. At present, the developed nations of the world are making low-cost power by means of large nuclear and thermal steam power plants. Such plants are too large and expensive for developing countries.

As low cost power becomes a necessity in remote areas of the nation, gas turbines play an important role in bridging the gap from initial part-time service to a full central station grid system ultimately equipped with an efficient combination of gas and steam turbines.

When the development of the country reaches a point where electricity becomes luxury instead of necessity and requires a system that will operate on 24-hour basis, the small high speed diesels are usually replaced by medium size gas turbine plants. The gas turbine with lower initial cost and minimum logistic requirements offers a solution except for the fuel. The gas turbine power plant is able to bridge the gap between the small high speed diesel engine and a full central station grid system. During the development of central gas turbine power plant, the existing small high speed diesels are used to provide initial power.

This situation was recognized by the Government of India for the State of Assam which is located north of Bangladesh and south of China.

The State of Assam 90,000 sq. miles in area had several hydroelectric power plants near Gauhati and Shillong. The north-east area of Assam consisting of numerous villages are equipped with many small diesel generating sets.

The major difficulty experienced for the installation of Namrup plant after the development of site was : the station outage was caused by heavy humid atmosphere. The humidity of 97 to 100% from April to October during the monsoon season caused the station outage due to the collection of moisture on the control lines and electrical wiring. This difficulty was removed with the help of air separators, traps and dehumidifiers. The outage hours per day were reduced from 12 (in 1965) to 4 (in 1968).

Namrup gas turbine power station is the first station of its type to be used as base load. This station supplies power to its own system without inter-connection with any other power plant. The non-stop operation period without any trouble was 72 days. But generally, the generating sets are taken out of operation after every 14 days for routine check.

Some space is reserved at the exhaust of each unit to allow the further installation of waste heat boilers so that the overall efficiency of the plant would improve at some future date when the steam plant would also operate with gas turbine plant using its waste heat.

The Government of India and Assam Electricity Board have provided the power to the people of Assam to meet the growing needs of the country with the minimum capital cost.

The Namrup station proves that the gas turbine plant permits a gradual transfer from an initial diesel station to the large central station with the minimum investment during the development stage provided the fuel required is available in abundance and at a cheaper rate. Earlier diesel plants were operated hardly for one or two hours each evening. The Government of India decided to replace all such small unreliable power units by a central gas turbine plant in the under-developed sector of the province. The gas turbine plant would be most economical to operate on round-the-clock basis as the supply of natural gas being available at Namrup, 15 miles away from the selected site.

The story of the development of Namrup gas turbine power plant is one of the struggles of men and machines fighting against jungle rot, disease, wild animals and snakes to turn a dream into reality. Actual work at the selected site was started in 1961. The jungle was cleared with the help of local villagers and trained elephants.

After the development of site, four 66 kV (200 miles) lines, fourteen 33 kV lines (450 miles) to important load points including two for fertilizer plants at Namrup, and fourteen 11 kV distribution lines (600 miles) to nearby towns and villages including all local tea states are completed.

Uran-Gas Turbine Power Plant. This is second power plant of this type established in this country and first in Maharashtra Electricity Board. The installed capacity in Maharashtra from thermal and hydel sources as in March 1981 is 3946 MW against the peak demand in Maharashtra, 3200 MW. As per the reasonably reliable power supply system, the installed capacity should be 4500 MW for taking peak of

3200 MW. It was anticipated that by the end of VI plan, the generation capacity would be 6600 MW against the peak demand of 6500 MW which may call for 10650 MW installed capacity as per norms. The MSEB will face a shortage of 4500 MW by the end of VI plan period.

With practically no spinning reserve, whenever there are forced outages on generating units, it immediately calls for load shedding particularly during peak load period. In this background, MSEB conceived the installation of Uran gas turbine plant of 240 MW (60 × 4) capacity which will give quick relief during peak demand period.

This power plant is located close to ONGC Uran terminal where the gas separation is done. The plant covers 40 hectares of land which is developed by filling 2 m as the area was low-lying. Nearly 2400 piles, 23 m deep were driven at the sight as soil was very poor. The erection and commissioning of the first unit of the plant was done in February 1982. The units of this plant are designed to operate either on gas or LSHS. A fuel oil treatment plant is also established at Uran to remove sodium, potassium and vanadium in the oil as gas turbine blades are very much vulnerable to these metals. The plant can be brought to full speed in 3 minutes and at full load in 8 minutes.

A 220 kV sub-station has been established which connects the station to the state 220 kV network.

Presently, Government of India supplies 140×10^3 tons of LSHS annually which will be sufficient to operate each unit daily for $5\frac{1}{2}$ hours during peak load period. Each unit of 60 MW capacity requires 440×10^3 litres of LSHS per day running at full load. It is also estimated that it will consume 460×10^3 m^3 of gas per day.

A pipe line of 30 cm in diameter and about 6 km long from ONGC terminal to gas turbine station is completed in 1982 to supply the gas separated at ONGC terminal. The gas will be utilised to the maximum extent by running the units on base load. The preseent estimated cost of the plant is 66.5 crores, 40 crores being foreign exchange.

In addition to the above two Gas Turbine power plants, 5 sets, each of 25 MW capacity supplied by John Brown are working with Bengal Electricity Board. Other British made gas turbine sets which have found favour in India are the small Ruston mobile sets. Seven of these, each of 3.4 MW capacity, are operated by the Assam State Electricity Board where they use gas which was previously flared to waste from the oil and gas field of Upper Assam. An interesting feature of these seven sets is that they operate in bad climatic conditions. Ambieent temperatures range from 2°C to 40°C, with dust storms for 180 days and tropical storms for 60 days in each year.

24.10. ADVANTAGES OF GAS TURBINE PLANT OVER DIESEL AND THERMAL POWER PLANTS

(A) Advantages over Diesel Plants

1. The perfect balancing of gas turbine unit is possible so the gas turbine plants are subjected to less vibrations.

2. The mechanical efficiency of gas turbine plant is as high as 95% whereas the mechanical efficiency of diesel plant hardly exceeds 80% as the sliding parts are considerably large.

3. The torque characteristics of turbine plants are far better than diesel plants as it is a continuous power developing system.

4. The work developed per kg of air is large compared with diesel plant as the expansion of gases to atmospheric pressure is possible.

5. The weight of gas turbine plant is hardly 0.15 kg/H.P. whereas the weight of diesel plant is 2.5 kg/H.P. Therefore, the space required and capital cost of the gas turbine plant is considerably less than diesel plant.

6. All the parts of gas turbine (compressor, combustion chamber and turbine) can be designed and tested individually and they can be arranged as per requirements as the compression, combustion and expansion take place in different units.

7. The running speed of the turbine (40000 to 100000 r.p.m.) is considerably large compared with diesel engine (1000 to 2000 r.p.m.).

8. The lubrication and ignition systems are more simplified compared with diesel plant.

9. The specific fuel consumption does not increase with time in gas turbine plant as rapidly as in diesel plants.

10. The installation and maintenance costs are less than diesel plants.

11. The exhaust of gas turbine is free from smoke as the quantity of air supplied is 4 times greater than theoretically required for complete combustion.

Smokeless combustion has been achieved by the gas turbine with proper control of primary zone mixture ratios and methods of secondary air introduction and recirculation could produce reduced pollution conditions. It is claimed that the exhaust gases coming out are smokeless without increasing the production of nitric oxide emissions.

12. Any poor quality fuels can be used in gas turbine plant where special grade fuels are required for diesel engines to avoid knocking. This is a great advantage of gas turbine plant.

The advantages of gas turbine plants over diesel plants can be sur marised as easier maintenance, improved reliability, lower initial cost, smaller plant dimensions and less space requirements for same output, less vibrations ; absence of cyclic variations, and greater starting torque.

The disadvantages of gas turbine plants over diesel power plants include poor part load efficiency, requirement of special metals and alloys for different components of the plants, special cooling methods for cooling the turbine blades, low starting torque, and less life.

Advantages over steam power plant

1. The handling of ash is a major problem in steam plants which is completely eliminated in open cycle gas turbine plants using gas or liquid as fuel.

2. The cubic capacity of the buildings required for gas turbine plant is about 50% of steam plant. The total weight of the materials required for gas turbine plant is also 50% to 60% of steam plant. Therefore, considerable saving in capital cost is possible having same efficiency as steam plant. The gas turbine plants can be installed at selected load centres as space requirement is considerably less where steam plant could not be accommodated.

3. 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 load centres as no cooling water is required. The cooling water required for closed cycle gas turbine plant is hardly 10% of the steam plant.

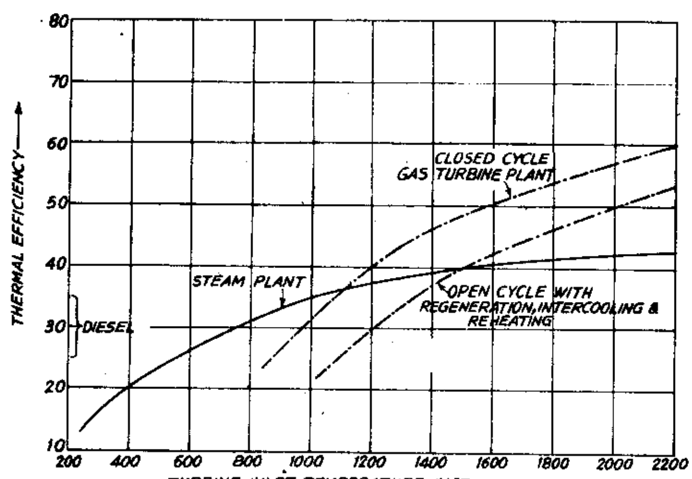


Fig. 24.24. Comparison of thermal efficiency between steam and closed cycle gas turbine plants.

4. The comparison of thermal efficiency is shown in Fig. 24.24. 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 three times as fast as the steam cycle efficiency for a given top temperature increase.

5. The ratio of exhaust to inlet volume for the same pressure and initial temperature conditions would be only 3.95 in case of gas turbine plant as against 250 for steam plant.

6. It has been observed from the viewpoint of the price that the steam plant developing power at 17% efficiency costs equal to the gas turbine plant developing power at 34%. This being the case, the gas turbine plant as a base load plant has little future. This is economical as base load plant only where the fuel oil is available at a considerably cheaper rate.

7. The gas turbine plant uses fewer auxiliaries than the steam turbine plant, therefore, small size of the gas turbine components enables complete work-tested units to be transported to the site.

8. The ease and rapidity with which the output of gas turbine plants can be made available from a cold start permit a reduction in the capacity of steam plant held in reserve in a hot condition and needed to cover for short load prediction errors and random load fluctuations. The gas turbine plants peak-up the load hardly within 15 minutes. This loading response can be put to good effect in the event of an unscheduled disconnection of a large generator.

9. The gas turbine plants are always desirable as peak load plants irrespective of the cost due to their good response but its adoption as base load is justified on the economic ground. It can be used as base load plant only where the gas turbine fuel costs are considerably less.

10. In general, gas turbines can be built relatively quicker and require much less space and civil engineering works and water supplies.

11. 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.

12. The gas turbine becomes more economical for operating conditions below a given load factor as saving on the capital charges outweighs the additional cost of fuel.

13. The gas turbine plant as peak load plant is more preferable as it can be brought on load quickly and surely. The fuel consumption is of secondary importance, since the time of operation is limited.

24.11. ENVIRONMENTAL IMPACT OF GAS TURBINE POWER PLANTS

Environmental impact is any alteration of existing environmental conditions or creation of new environmental conditions, adverse or beneficial, caused or induced by the generation of power. Any new project needed for economic growth affects some important criteria for quality of life which are closely related with the surrounding environment.

Impact assessment is done after taking into account of all kinds of actions which have impact on environmental parameters. The actions include construction, commissioning, water requirement, waste water, gaseous waste, solid waste, immigration, housing etc. Environmental parameters include land use, water quality, air quality, noise, ecology, forestry, health etc.

For gas turbines, the impact assessment is not difficult as most important is gaseous pollution and its impact on human health in microscale and on ecology in macro scale. The major pollutants from the combustion process in the gas-turbine are NO_x, SO_2, HC and CO. Out of these NO_x and SO_2 are of main concern. The concentration of the above pollutants is directly related to temperature, time, turbulence, air-fuel ratio used and design of combustion chamber.

Formation of CO & HC

Most of CO comes from incomplete combustion. It is dominant during idling of the machine or running at low load. High CO percentage exists if the combustor is designed to operate fuel-rich. Significant amount of CO will be present in the primary zone of combustor operating with stoichiometric or moderately lean mixture. It is possible to reduce CO to a negligible level by staged admission of additional air downstream of primary zone.

In short, CO is formed due to

(*a*) Insufficient residence time in primary zone.

(*b*) Inadequate mixing of fuel and air and over-rich combustion produces high local concentration of CO.

The effect of load on the formation of HC, CO and NO_x is shown in Fig. 24.25.

Formation of NO_x

Oxides of Nitrogen are produced by the oxidation of atmospheric N_2 in high temperature regions of the flame. The main factor governing NO_x formation is temperature as shown in Fig. 24.26.

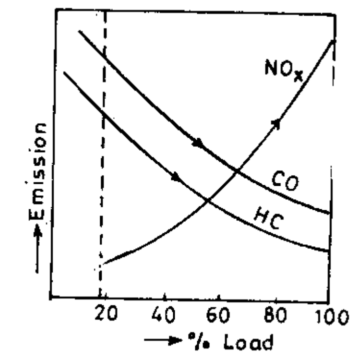


Fig. 24.25.

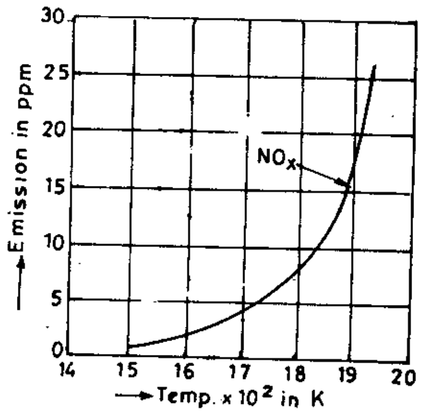


Fig. 24.26.

The NO_x emission increases exponentially with flame temperature as per the relation

$$NO_x = K(e)^{0.009\,T}$$

The process of NO_x formation is endothermic and is at a significant rate above the temperature of 1800 K. Similarly, NO_x level is highest at full load condition.

The mechanism of NO_x (mainly NO) formation is given by the set of following chemical reactions :

$$O_2 \rightleftharpoons 2O$$

$$O + N_2 \rightleftharpoons NO + N$$

$$N + O_2 \rightleftharpoons NO + O$$

In addition to this, NO is formed from N_2 contained in the fuel and its contribution depends upon degree of N_2 in the fuel. The effect of residence time on formation of NO_x is shown in Fig. 24.27. It is evident that NO_x emission increases with an increase in residence time, except for very lean mixture ($\phi = 0.4$) for which the rate of formation is so low that it becomes fairly insensitive with time.

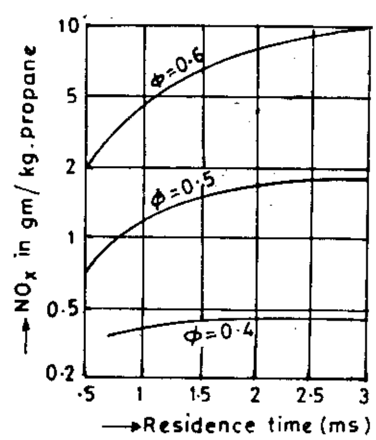


Fig. 24.27. ϕ = Eequivalence ratio.

Formation of SO_2

The SO_2 emission depends solely on the amount of sulphur content in the fuel.

Effect of NO_x and SO_2

Though man-made NO_x is less than 1% of natural NO_x emission, but it is produced in high population density areas, its concentration and reactions with other compounds pose serious problems.

NO_x has been identified as the major cause for photo-chemical smog. Irritating and harmful oxides are formed in the atmosphere by NO_x. The secondary pollutants formed are ozone, formaldehyde, organic hyperoxides, and peroxyacetyl nitrate (PAN) and its homologues. Simplified reaction of NO_x in the presence of light are listed below :

$$NO_2 + \text{Light} \longrightarrow NO + O$$
$$O + O_2 \longrightarrow O_3$$
$$O_3 + NO \longrightarrow NO_2 + O_2$$
$$O + HC \longrightarrow HCO' \text{ (radical)}$$
$$HCO' + O_2 \longrightarrow HCO_3' \text{ (radical)}$$
$$HCO_3' + HC \longrightarrow \text{aldehydes, ketones}$$
$$HCO_3' + NO \longrightarrow HCO_2' \text{ (radical)} + NO_2$$
$$HCO_3' + NO_2 \longrightarrow PAN$$

Increase of NO_x emission is also considered a serious problem in the context of acidification of precipitation (acid rains). Fresh water gets acidified and thus disturbs the aquatic life and agricultural life cycle or destroys. These reactions are listed below :

NO is oxidized by O_3 diffused from stratosphere to troposphere as

$$NO + O_3 \longrightarrow NO_2 + O_2$$

And part of NO_2 is converted into HNO_3 when comes in contact with water as

$$3NO_2 + H_2O \longrightarrow 2HNO_3 + NO$$

If this acid is carried by the rains on the ground and sea, then the agricultural and aquatic life is very much affected.

Effect of SO_2

All the SO_2 in the air is man-made. It acts as an irritating gas. It is responsible for most of the urban environmental disasters. It is very reactive in the presence of water. It first produces sulphurous acid which further gets oxidized to H_2SO_4. A portion of SO_2 is also oxidised to SO_3 in atmosphere which is then hydrolysed to H_2SO_4 as listed below :

$$\left.\begin{aligned} SO_2 + H_2O &\longrightarrow H_2SO_3 \\ 2H_2SO_3 + O_2 &\longrightarrow 2H_2SO_4 \end{aligned}\right\}$$
$$\left.\begin{aligned} 2SO_2 + O_2 &\longrightarrow 2SO_3 \\ SO_3 + H_2O &\longrightarrow H_2SO_4 \end{aligned}\right\}$$

Green plant life is far more sensitive to SO_2 than men and animals. Damaging effects of SO_2 in the vicinity of the power plants have been well observed. Injury to vegetation may occur even at a low ground level concentration of less than 1 ppm. Recent events of acid rains in some areas of industrial world also points to the damaging role of sulphur dioxide.

Formation of SO_2 can be taken as direct oxidation of sulphur content of fuel and thus can be calculated more accurately. But it is difficult to calculate accurately the amount of NO_x emission. Because, depending upon temperature, residence time, air-fuel ratio, the NO_x formation can vary between 1 gm/kg of fuel to 25 gm/kg of fuel.

The results on 30 MW gas turbine plant are presented here to have a quantitative understanding of environmental impact due to gas turbine.

Liquid Fuel — Consumption 9 MT/hr
Sulphur content – 1%
Excess air – 10% (in primary zone)

Natural Gas — C.V 40000 kJ/m^3
sulphur content – 1000 ppm
Excess air – 10%

NO_x formation — 10 gm/kg of fuel

With the above-mentioned considerations, ground level concentrations (GLC) for both types of fuels for different stack heights have been calculated. For GLC calculations, the following parameters are considered :

Stack diameter – 2.7 metres, Exit temperature 400°C.

Permissible Indian Standard based on 8 hrs average as 120 $\mu g/m^3$ for NO_x as well as SO_2 has been considered.

The results are represented in Fig. 24.28.

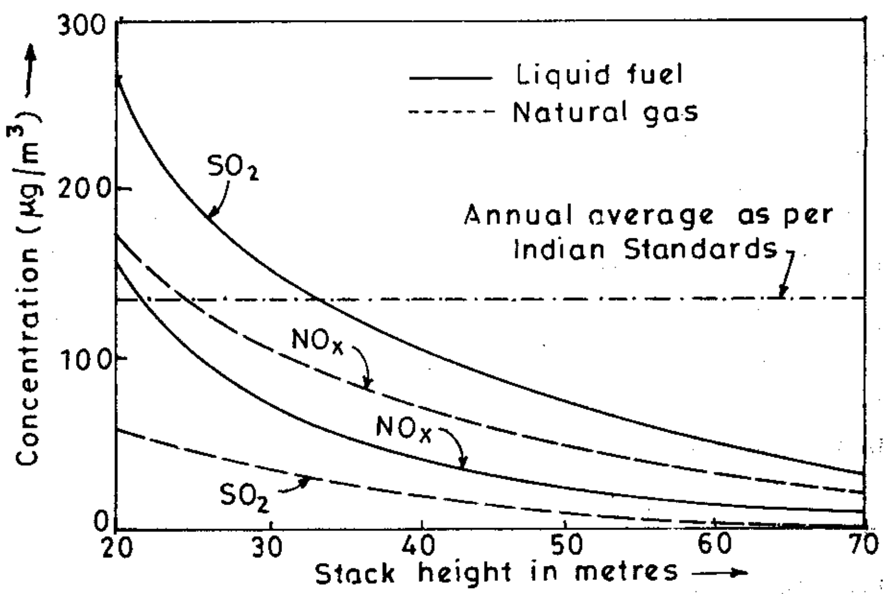


Fig. 24.28.

The curves in Fig. 24.28 are plotted for SO_2 and NO_x concentrations with respect to stack heights for liquid fuel and natural gas. They show that for liquid fuel, a minimum stack height of 35 m and for natural gas, a minimum stack height of 22 m are required.

Central Board for Prevention and Control of Water Pollution has suggested that the minimum height of any stack is to be 30 m. For power plants less than 200 MW capacity, the stack height is suggested to be 30 m or

$$H = 14\,(Q)^{0.3} \text{ whichever is higher}$$

where Q is SO_2 emission in kg/hr.

As per above formula, a 30 MW gas turbine unit using liquid fuel needs a stack height of about 70 m and height of stack is reduced to 40 m if the gas turbine uses natural gas.

One of the most significant, from a commercial standpoint, is the Catalytic Mountain View, Calif and Tanaka Kikinzoku Kngyo, Tokyo, Japan have jointly developed on oxidation catalyst for the gas turbine firing at 10 bar pressure using natural gas as fuel, where they have achieved NO_x emission below 1 ppm and CO and unburned hydro-carbon emission well below 10 ppm.

SOLVED PROBLEMS

The following notations are used in the problems :

1. η_c = Isentropic efficiency of compressor.
2. η_t = Isentropic efficiency of turbine.
3. η_m = Mechanical efficiency.
4. η_{com} = Combustion efficiency.
5. η_g = Generator efficiency.
6. η_{tt} = Transmission efficiency.
7. ε = Effectiveness of heat exchanger.
8. R_p = Pressure ratio.

The following formulas are used for solving the problems :

1. $\eta_{th} = 1 - \dfrac{1}{(R_p)^{(\gamma-1)/\gamma}}$

2. $(R_p)_{max} = \left(\dfrac{T_3}{T_1}\right)^{\gamma/(\gamma-1)}$

R_p (for max specific work) $= \left[\eta_c \eta_t \dfrac{T_3}{T_1}\right]^{\gamma/[2(\gamma-1)]} = \sqrt{(R_p)_{max}}$ if $\eta_c = \eta_t = 1$.

3. $P_i = \sqrt{P_1 P_2}$ for perfect intercooling.

4. $T_2' = T_1 \left(\dfrac{p_2}{p_1}\right)^{(\gamma-1)/\gamma}$ and $\eta_c = \dfrac{T_2' - T_1}{T_2 - T_1}$

5. $T_4' = T_3 \left(\dfrac{p_1}{p_2}\right)^{(\gamma-1)/\gamma}$ and $\eta_t = \dfrac{T_3 - T_4}{T_3 - T_4'}$

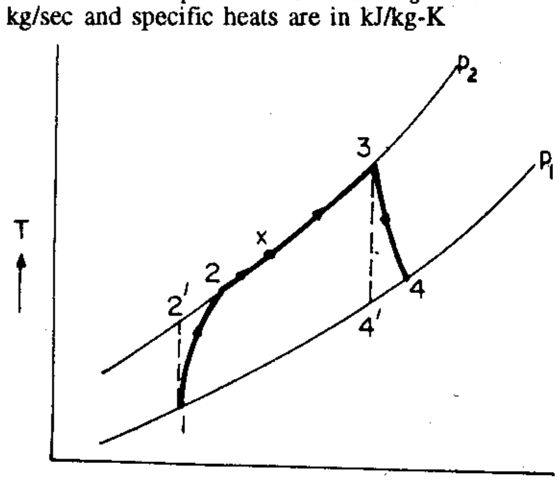
6. Heat balance in the combustion chamber

$$C_{pg}\,(m_a + m_f)\,(T_3 - T_x) = m_f \times \text{C.V.} \times \eta_{com}$$

$$\therefore \quad \text{C.V.} \times \eta_{com} = C_{pg}\left[\frac{m_a}{m_f} + 1\right](T_3 - T_x)$$

7. $[(m_a + m_f)\,C_{pg}\,(T_3 - T_4) - m_a C_{pa}\,(T_2 - T_1)]\;\eta_m \times \eta_g \times \eta_{tt} = \text{kW}$

where m_a and m_f are in kg/sec and specific heats are in kJ/kg-K



$T - s$ diagram for open cycle gas turbine power plant.

$$\therefore \quad m_a\left[\left(1 + \frac{m_f}{m_a}\right) C_{pg}\,(T_3 - T_4) - C_{pa}\,(T_2 - T_1)\right] \eta_m \,.\, \eta_g \,.\, \eta_{ti} = \text{kW}$$

where m_a is the mass of air supplied per second and m_f is the mass of fuel supplied per second.

8. $$\varepsilon = \frac{C_{pa} m_a (T_x - T_2)}{C_{pg}(m_a + m_f)(T_4 - T_2)} = \frac{C_{pa}(T_x - T_2)}{C_{pg}\left[1 + \frac{m_f}{m_a}\right](T_4 - T_2)}$$

$$= \frac{C_{pa}}{C_{pg}} \cdot \frac{(T_x - T_2)}{(T_4 - T_2)} \text{ as } \frac{m_f}{m_a} << 1$$

$$= \frac{T_x - T_2}{T_4 - T_2} \text{ if } C_{pa} = C_{pg}.$$

Problem 24.1. *An open cycle gas turbine plant uses heavy oil as fuel. The maximum pressure and temperature in the cycle are 5 bar and 650°C. The pressure and temperature of air entering into the compressor are 1 bar and 27°C. The exit pressure of the turbine is also 1 bar. Assuming isentropic efficiencies of compressor and turbine to be 80% and 85% respectively, find the thermal efficiency of the cycle. The overall A : F ratio used is 60 : 1.*

Take C_p (for air and gas) = 1 kJ/kg-°C.

and γ (for air and gas) = 1.4

If the plant consumes 5 kg of fuel/sec, find the power generating capacity of the plant.

Solution. The processes are represented on the cycle as shown in Fig. Prob. 24.1.

$p_1 = 1$ bar, $T_1 = 27 + 273 = 300$ K

$p_2 = 5$ bar, $T_3 = 650 + 273 = 923$ K

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

$$= 300(5)^{0.285} = 300 \times 1.58 = \mathbf{474K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad 0.8 = \frac{474 - 300}{T_2 - 300}$$

$$\therefore \quad T_2 = \frac{174}{0.8} + 300 = 218 + 300 = \mathbf{518K}$$

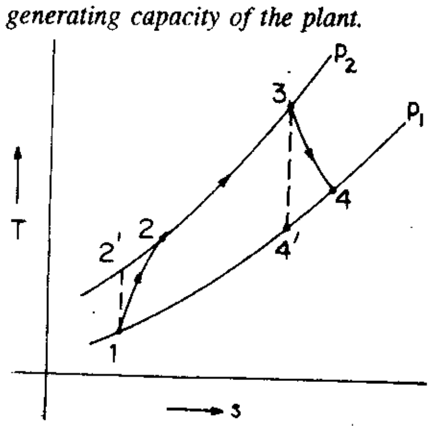


Fig. Prob. 24.1.

$$T_4' = T_3\left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = 923\left(\frac{1}{5}\right)^{0.285} = \frac{923}{1.58} = \mathbf{585K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad 0.85 = \frac{923 - T_4}{923 - 584} = \frac{923 - T_4}{339}$$

$$\therefore \quad T_4 = 923 - 339 \times 0.85 = 923 - 288 = \mathbf{635K}$$

The thermal efficiency of the cycle is given by

$$\eta_{th} = \frac{C_{pg}(m_a + m_f)(T_3 - T_4) - C_{pa} m_a (T_2 - T_1)}{C_{pg}(m_a + m_f)(T_3 - T_2)}$$

As $C_{pa} = C_{pg}$ given in problem

$$\eta_{th} = \frac{\left(\frac{m_a}{m_f} + 1\right)(T_3 - T_4) - \frac{m_a}{m_f}(T_2 - T_1)}{\left(\frac{m_a}{m_f} + 1\right)(T_3 - T_2)} = \frac{(60 + 1)(923 - 635) - 60(518 - 300)}{(60 + 1)(923 - 518)}$$

$$= \frac{61 \times 288 - 60 \times 218}{61 \times 405} = 0.71 - 0.53 = 0.18 = \mathbf{18\%}.$$

The work done per kg of fuel supplied in kJ

$$= C_{pg}\,(1+60)\,(T_3 - T_4) - C_{pa} \times 60\,(T_2 - T_1) = 1[61 \times (923 - 635) - 60(518 - 300)]$$

$$= 1[61 \times 288 - 60 \times 218] = 1 \times 60[292 - 218] = 1 \times 60 \times 74 = \mathbf{4440\ kJ/kg\ of\ fuel}$$

Power capacity in kW $= 4440 \times 5$ kW $= \dfrac{4440 \times 5}{1000}$ MW $=$ **22.2 MW.**

Problem 24.2. *An open cycle gas turbine power plant has a maximum pressure ratio 8 and temperature of 1080 K. The pressure and temperature of air at the entry of the compressor are 1 bar and 300 K. A regenerative heat exchanger is used to increase the temperature of air before entering into the combustion chamber. The effectiveness of the heat exchanger is 0.6. If the air flow through the compressor is 500 kg/min, find the thermal efficiency of the plant and plant capacity in kW. Also find the saving in fuel consumption in kg per hour due to regeneration. The calorific value of the fuel used in 42000 kJ/kg.*

Take isentropic efficiency of compressor and turbine as 80%

$\gamma = 1.4$ (for air and gas)

$C_p = 1$ kJ/kg-°C (for air and gas).

Also find the air-fuel ratio used.

Neglect the pressure losses in the system.

Solution. The processes are shown in Fig. Prob. 24.2.

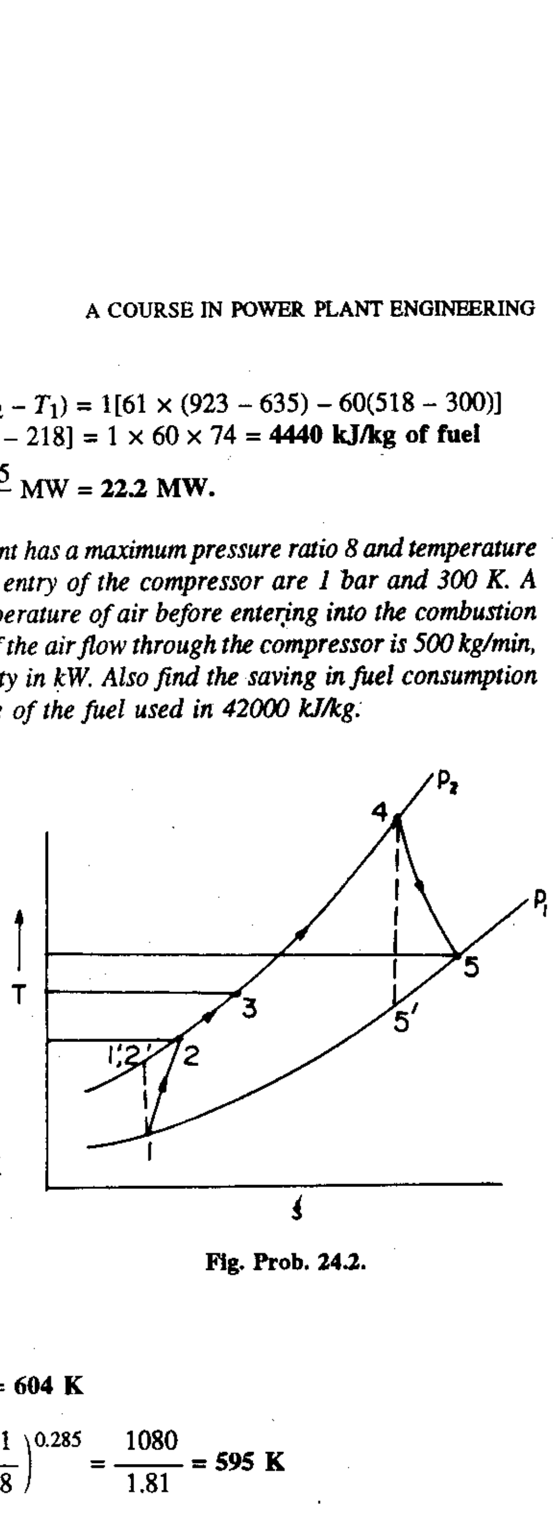


Fig. Prob. 24.2.

The given data is

$$T_1 = 300\text{K},\ p_1 = 1\ \text{bar},\ \frac{p_2}{p_1} = 8,\ T_4 = 1080\text{K},\ \eta_c = \eta_t = 0.8$$

$$T_2' = T_1\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 300(8)^{0.285} = 300 \times 1.81 = \mathbf{543K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad 0.8 = \frac{543 - 300}{T_2 - 300}$$

$$\therefore \quad T_2 = 300 + \frac{243}{0.8} = 300 + 304 = \mathbf{604\ K}$$

$$T_5' = T_4\left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = 1080\left(\frac{1}{8}\right)^{0.285} = \frac{1080}{1.81} = \mathbf{595\ K}$$

$$\eta_t = \frac{T_4 - T_5}{T_4 - T_5'}$$

$$\therefore \quad 0.8 = \frac{1080 - T_5}{1080 - 595}$$

$$\therefore \quad T_5 = 1080 - 485 \times 0.8 = 1080 - 388 = \mathbf{692\ K.}$$

If the air is heated from temperature T_2 to T_3, then the effectiveness of heat exchanger is given by

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2}$$

$$\therefore \quad 0.6 = \frac{T_3 - 604}{692 - 604}$$

$\therefore \quad T_3 = 604 + 0.6 \times 88 = 604 + 52.8 \approx$ **657 K.**

If m_f is the fuel in kg per kg of air is used then

$$C_{pg}\,(1 + m_f)\,(T_4 - T_3) = m_f \times \text{C.V.}$$

$\therefore \quad 1\,(1 + m_f)\,(1080 - 657) = m_f \times 42000$

$$\therefore \quad \frac{1}{m_f} = \frac{42000}{423 \times 1} - 1 = 99.3 - 1 = \mathbf{98.3}$$

$\therefore \quad m_f = 0.0102$ kJ/kg of air

$$\therefore \quad \frac{m_a}{m_f} = \frac{1}{0.0102} = \frac{98.3}{1}$$

Heat saved due to heat exchanger per kg of air flow

$= 1\,(T_3 - T_2) = 1(657 - 604) = 1 \times 53 =$ **53 kJ/kg of air**

$$\therefore \quad \text{Fuel saved per hour} = \frac{500 \times 60 \times 53}{42000} = \mathbf{37.86\ kg/hr}$$

Net work available for power generation per kg of air flow

$= C_{pg}\,(1 + m_f)\,(T_4 - T_5) - C_{pa} \times 1\,(T_2 - T_1)$

$= 1[1.0102\,(1080 - 692) - 1 \times (604 - 300)] = 392 - 304 =$ **88 kJ/kg**

$$\therefore \quad \text{The capacity of the plant in kW} = \frac{500}{60} \times 88 = \mathbf{733.3\ kW.}$$

Problem 24.3. *Calculate the efficiency and specific work output of a simple gas turbine plant operating on Brayton cycle. The maximum and minimum temperatures are 1000K and 288K respectively. The pressure ratio is 6. The isentropic efficiencies of the compressor and turbine are 85% and 90% respectively. If the unit consumes 2 tonnes of oil per hour of C.V. 46500 kJ per kg, determine the power generated. The mechanical efficiency is 90% and the generation efficiency is 85%.*

Solution. Referring to the *T-s* diag. 1-2′ is isentropic compression while 1-2 is the actual compression. 3-4′ is isentropic expansion while 3-4 is the actual expansion.

$$T_2' = T_1 \,.\, (6)^{0.4/1.4} = \mathbf{481K.}$$

Compressor efficiency

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} = 0.85$$

$$\frac{481 - 288}{T_2 - 288} = 0.85 \quad \therefore \quad T_2 = \mathbf{515K}$$

$$T_4' = \frac{T_3}{(6)^{0.4/1.4}} = \frac{1000}{(6)^{0.4/1.4}} = \mathbf{599K}$$

$$\frac{T_3 - T_4}{T_3 - T_4'} = 0.90, \quad \frac{1000 - T_4}{1000 - 599} = 0.90 \quad \therefore \quad \mathbf{T_4 = 632K.}$$

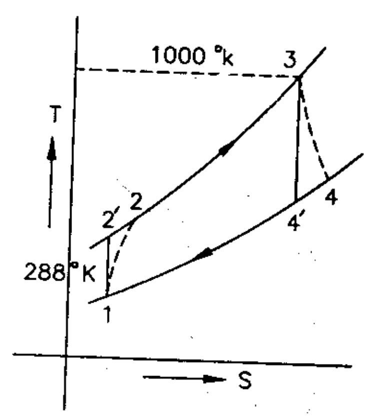


Fig. Prob. 24.3.

Compressor work

$= C_p\,[T_2 - T_1] = 1.005\,[515 - 288] =$ **221 kJ/kg**

Turbine work $= C_p\,[T_3 - T_4] = 1.005\,[1000 - 639] =$ **363 kJ/kg**

Heat added $= C_p\,[T_3 - T_2] = 1.005\,[1000 - 515] =$ **487 kJ/kg**

$$\text{Cycle efficiency} = \frac{363 - 288}{487} = 0.277 = \mathbf{27.7\%}$$

Specific work output = 363 − 288 = **135 kJ/kg** 135 kW per kg of air/sec.

$$\text{Output of the unit} = \frac{2 \times 1000}{3600} \times 46500 \times 0.277 \times 0.90 \times 0.85$$

$$= 5473.8 \text{ kJ/s} = 5473.8 \text{ kW} \approx \mathbf{5.475 \text{ MW}}.$$

Problem 24.4. *A gas turbine takes in air at 101 kN/m² and 15°C. The air is compressed to a pressure of 606 kN/m² and then passed through a regenerative heat exchanger of effectiveness 0.65. The air is then passed through the combustion chamber where its temperature is increased to 870°C by the combustion of fuel. The gases enter a turbine and are expanded to 101 kN/m² pressure. Assuming a compressor efficiency of 85% and a turbine efficiency of 80%, determine the following for air flow rate of 4 kg/s :*

1. *The power output of the plant,*
2. *Exhaust temperature from heat exchanger,*
3. *The thermal efficiency of the plant, and*
4. *The thermal efficiency without the heat exchanger.*

Solution. The temperatures at various points should be determined first in order to find the efficiency of the cycle.

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 288 \times 6^{0.4/1.4} = \mathbf{477K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} = 0.85$$

$$\frac{477 - 288}{T_2 - 288} = 0.85 \quad \therefore \quad T_2 = \mathbf{510K}$$

$$T_4' = \frac{T_3}{\left(\frac{p_3}{p_4}\right)^{(\gamma-1)/\gamma}} = \frac{T_3}{(6)^{0.4/1.4}} = \mathbf{685\ K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'} =$$

$\therefore \quad T_4 = \mathbf{768.6\ K.}$

Net power output

$$= mC_p \{(T_3 - T_4) - (T_2 - T_1)\}$$

$$= 4 \times 1.005 \times \{[1143 - 768.6] - [510 - 288]\}$$

$$= \mathbf{612\ kW.}$$

The effectiveness of the heat exchanger

$$= \frac{T_5 - T_2}{T_4 - T_2} = \frac{T_5 - 510}{768.6 - 510} = 0.65$$

$\therefore \quad T_5 = \mathbf{678\ K}$

$T_4 - T_6 = T_5 - T_2$ neglecting, the weight of fuel

$\therefore \quad T_6 = \mathbf{600.6\ K.}$

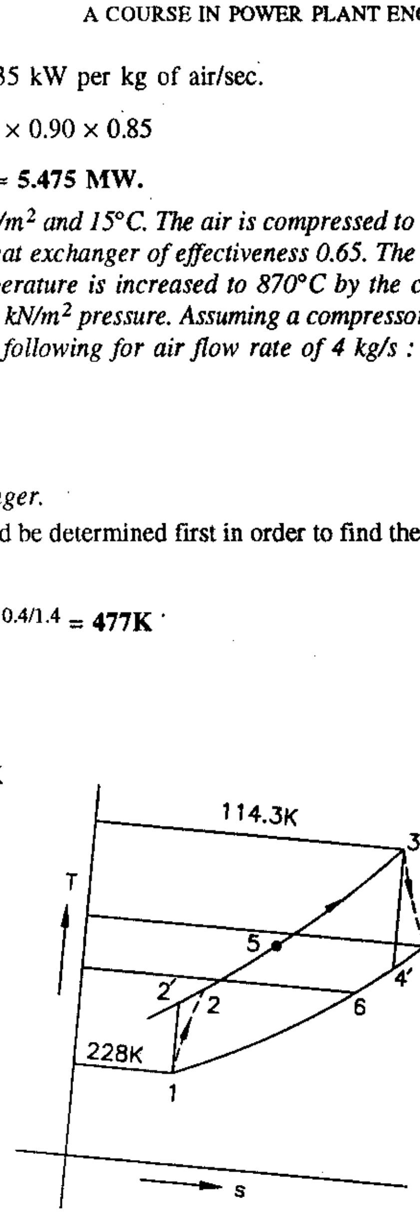


Fig. Prob. 24.4.

$$\left.\begin{array}{l}\text{Efficiency of the plant} \\ \text{with regeneration}\end{array}\right\} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_5)} = \frac{(1143 - 768.6) - (510 - 288)}{(1143 - 678)} = \mathbf{33.5\%}.$$

$$\text{Efficiency without heat exchanger} = \frac{(1143 - 768.6) - (510 - 288)}{1143 - 510} = \mathbf{24.1\%}.$$

This is the reason why regenerative heating is used in gas turbines power plant and automobile gas turbines. The heat exchanger is bulky and so cannot be used in aircraft turbines.

Problem 24.5. *Air enters a gas turbine plant at a pressure of 100 kN/m² and 19°C and is compressed by an axial flow compressor having an isentropic efficiency of 0.85 to a pressure of 800 kN/cm². In the combustion chamber, the gas temperature is increased to 980°C at constant pressure. The gas then expands first through a turbine having an isentropic efficiency of 0.88, and developing just enough power to drive the compressor. Then the gases expand through a power turbine to the atmosphere. The isentropic efficiency of the power turbine is 0.86. For a flow rate of 7 kg/s, determine*

1. The condition of air at the exit of the first turbine,

2. The power turbine output, and

3. The thermal efficiency of the plant.

Assume $C_p = 1.006$ *kJ/kg-K,* $\gamma = 1.4$.

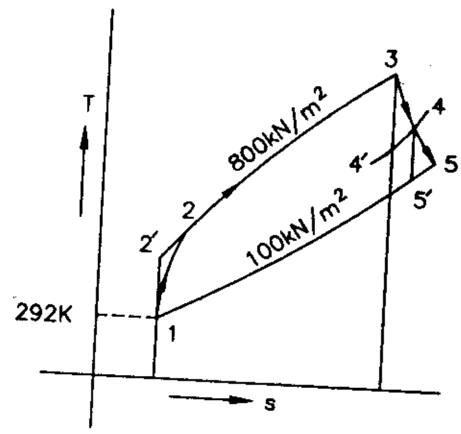


Fig. Prob. 24.5.

Solution. The cycle is shown on T-s diagram

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 292\left(\frac{800}{100}\right)^{(1.4-1)/1.4}$$

$= \mathbf{528\ K.}$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} = 0.85 = \frac{528 - 292}{T_2 - 292}$$

$\therefore \quad T_2 = \mathbf{570\ K.}$

(1) *For the first turbine*

Compressor work = Turbine work

or $\quad T_2 - T_1 = T_3 - T_4$

$\therefore \quad 570 - 292 = 1253 - T_3 \quad \therefore \; T_3 = \mathbf{975\ K.}$

$\therefore$ Turbine exit temperature (first turbine) = **975 K or 702°C.**

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'} = 0.88 \;\therefore\; \frac{1253 - 975}{1253 + T_4'} = 0.88. \quad \therefore \; T_4' = \mathbf{937\ K}$$

$$\frac{T_3}{T_4'} = \left(\frac{p_3}{p_4'}\right)^{(\gamma-1)/\gamma}; \quad \therefore \; \frac{1253}{937} = \left(\frac{800}{P_4'}\right)^{0.4/1.4}$$

Solving $\quad P_4' = P_4 = \mathbf{288\ kN/m^2.}$

(2) *For the power turbine*

$$T_5' = T_4\left(\frac{P_5}{P_4}\right)^{(\gamma-1)/\gamma} = 975\left(\frac{100}{288}\right)^{(1.4-1)/1.4} = \mathbf{720\ K.}$$

$$\text{Power Turbine efficiency} = \frac{T_4 - T_5}{T_4 - T_5'} = 0.86 \;\therefore\; \frac{975 - T_5}{975 - 720} = 0.86. \quad \therefore \; T_5 = \mathbf{756\ K.}$$

Power developed : $= mC_p\,[T_4 - T_5] = 7 \times 1.006\,[975 - 756] = \mathbf{1540\ kW.}$

Thermal efficiency : (neglecting mass of fuel) $= \dfrac{\text{Net work done}}{\text{Heat added}}$

$$= \frac{C_p\,[T_4 - T_5]}{C_p\,[T_3 - T_2]} = \frac{T_4 - T_5}{T_3 - T_2} = \frac{975 - 756}{1253 - 670} = 0.321 \text{ or } \mathbf{32.1\%}.$$

Problem 24.6. *A simple small gas turbine plant is designed to develop 1.1 MW power. The pressure and temperature of air entering the compressor are 1.03 bar and 288 K. The pressure at the outlet of compressor is 6 bar and pressure loss in the combustion chamber is 0.1 bar. Max. Temp. in the cycle is limited to 750°C. Taking the following data :*

Isentropic efficiency of compressor as well as turbine = 80%.

Combustion efficiency = 90%.

C_p (for air and gases) = 1 kJ/kg-K.

γ (for air and gases) = 1.4.

C.V. of fuel used = 20,000 kJ/kg.

Determine (i) Flow of air and flow of gases per second.

(ii) Work ratio (iii) Thermal efficiency of the plant. *(P.U. Summer 1990)*

Solution. The arrangement of the components is shown in Fig. Prob. 24.6 (*a*) and processes are represented in Fig. Prob. 24.6 (*b*).

The given data is

$$p_1 = 1.013 \text{ bar}, T_1 = 288 \text{ K}, p_2 = 6 \text{ bar}, p_3 = 6 - 0.1 = 5.9 \text{ bar}$$

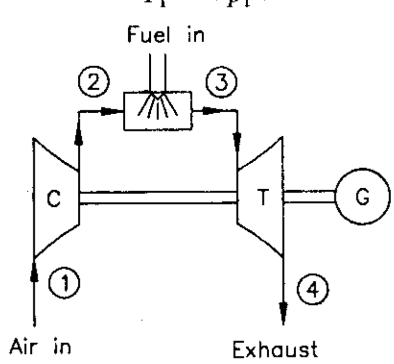
$$T_3 = 750 + 273 = 1023 \text{ K}, \eta_c = \eta_t = 0.8, \eta_{com} = 0.9,$$

$$\text{Power} = 1.1 \text{ MW}$$

$$C_{pa} = C_{pg} = 1 \text{ kJ/kg-K and } \gamma \text{ (air as well as gas)} = 1.4$$

Applying isentropic law to the process 1-2

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



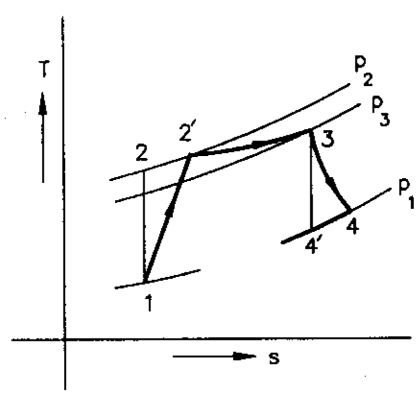


Fig. Prob. 24.6.

$$\therefore \quad T_2' = 288\left(\frac{6}{1.013}\right)^{\frac{0.4}{1.4}} = \mathbf{475.9\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} = 0.8$$

$$\therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{0.8} = 288 + \frac{475.9 - 288}{0.8} = \mathbf{523\ K}$$

Considering m_f is the fuel supplied when m_a is the mass of air passed through compressor.

$\therefore \quad (m_a + m_f)\, C_{pg}\, (T_3 - T_2) = m_f\, (\text{C.V.}) \times \eta_{com}$

$\therefore \quad \left(\dfrac{m_a}{m_f} + 1\right) C_{pg}\, (T_3 - T_2) = \text{C.V.} \times \eta_{com}$

$$\left(\frac{m_a}{m_f} + 1\right) \times 1\ (1023 - 523) = 20{,}000 \times 0.9$$

$$\therefore \quad \frac{m_a}{m_f} = \frac{18{,}000}{500} + 1 = 36 + 1 = \mathbf{37} \qquad ...(a)$$

Applying isentropic law to the process 3-4′

$$\frac{T_3}{T_4'} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5.9}{1.013}\right)^{\frac{0.4}{1.4}} = (5.82)^{0.286} = 1.655$$

$$\therefore \quad T_4' = \frac{T_3}{1.655} = \frac{1023}{1.655} = \mathbf{618\ K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'} = 0.8$$

$$\therefore \quad T_4 = T_3 - 0.8\,(T_3 - T_4') = 1023 - 0.8\,(1023 - 618) = 1023 - 324 = \mathbf{699\ K.}$$

If m_a is the mass of air supplied per sec and m_f is the mass of fuel supplied per second, then the (W_n) net power developed is given by

$$W_n = (m_a + m_f)\, C_{pg}\, (T_3 - T_4) - m_a\, C_{pa}\, (T_2 - T_1)$$

$$= m_a \left[\left(1 + \frac{m_f}{m_a}\right) C_{pg}\, (T_3 - T_4) - C_{pa}\, (T_2 - T_1)\right]$$

Substituting the values of temps and $\dfrac{m_f}{m_a} = \dfrac{1}{37}$ from equation (a)

$$W_n = m_a \left[\left(1 + \frac{1}{37}\right) \times 1\ (1023 - 699) - 1\ (523 - 288)\right] \text{kJ/sec.}$$

$$= m_a\, (330.7 - 235) = 95.7\, m_a \text{ kJ/sec} = \mathbf{95.7\ m_a\ kW.}$$

As $\quad W_n = 1.1$ MW = **1100 kW (given).**

$\therefore \quad 95.7\, m_a = 1100$

$$m_a = \frac{1100}{95.7} = \mathbf{11.5\ kg/sec.}$$

$$\frac{m_a}{m_f} = 37 \qquad \therefore\ m_f = \frac{m_a}{37} = \frac{11.5}{37} = \mathbf{0.31\ kg/sec}$$

$$\therefore \quad m_g = m_a + m_f = 11.5 + 0.31 = \mathbf{11.81\ kg/sec.}$$

$$W_t = m_g\, C_{pg}\, (T_3 - T_4) = 11.81 \times 1\ (1023 - 699) = 3826.5 \text{ kJ/s} = \mathbf{3826.5\ kW.}$$

$$\text{Work Ratio} = \frac{W_n}{W_t} = \frac{1100}{3826.5} = \mathbf{0.2875.}$$

Thermal efficiency of the plant

$$= \frac{W_n}{Q_s} = \frac{1100}{m_f \times \text{C.V.}} = \frac{1100}{m_g\, C_{pg}\, (T_3 - T_2)}$$

$$= \frac{1100}{11.81 \times 1\ (1023 - 523)} = \frac{1100}{11.81 \times 500} = 0.186 = \mathbf{18.6\%.}$$

Problem 24.7. *The pressure ratio used in an open cycle gas turbine power plant is 6.5. The pressure and temperature of air entering into the compressor are 1 bar and 300 K. Intercooling is used for reducing the work of compression. The maximum temperture of the cycle is limited to 850K. If the power plant capacity is 10 MW, find the thermal efficiency of the plant and fuel consumption per hour if the oil used has a calorific value of 45000 kJ/kg. Assume compression in both stages and expansions in the turbine are isentropic.*

Take $\gamma = 1.4$ for air and gases

and $C_p = 1$ kJ/kg-K for air and gases.

Also find out the maximum work saved per kg of air compressed due to intercooling. Don't neglect the quantity of fuel.

Solution. The processes are represented on *T-s* diagram as shown in Fig. Prob. 24.7.

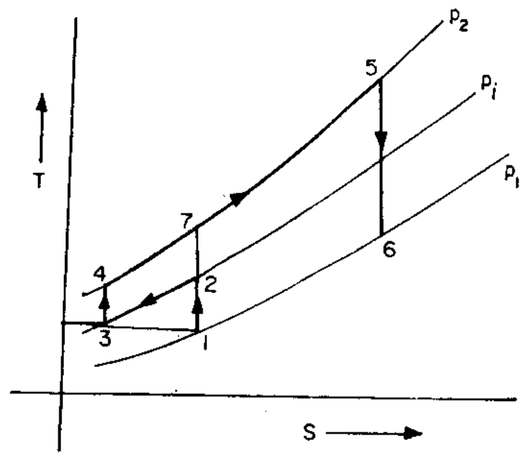


Fig. Prob. 24.7.

Perfect intercooling gives minimum work of compression, and the required intermediate pressure p_i is given by

$$p_i = \sqrt{p_1 p_2} = \sqrt{1 \times 6.5} = \mathbf{2.55\ bar}$$

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

$$= 300(2.55)^{0.285} = 300 \times 1.306 = \mathbf{392K}$$

$$T_7 = T_1 \left(\frac{6.5}{1}\right)^{0.285} = 300\ (1.7) = \mathbf{510K}$$

$$T_3 = T_1 = \mathbf{300K}$$

(as required for minimum compression work as mentioned above)

$$T_4 = T_3 \left(\frac{p_2}{p_i}\right)^{(\gamma-1)/\gamma} = 300 \left(\frac{6.5}{2.55}\right)^{0.285} = \mathbf{392K.}$$

The work done per kg of air with perfect intercooling

$$= 2 \times 1\ (T_2 - T_1) = 2\ (392 - 300) = \mathbf{184\ kJ/kg}$$

The work done per kg of air without intercooling

$$= C_p\ (T_7 - T_1) = 1\ (510 - 300) = 1 \times 210 = \mathbf{210\ kJ/kg}$$

∴ Work saved per kg of air-compressed due to intercooling

$$= 210 - 184 = \mathbf{26\ kJ/kg}$$

The heat balance in combustion chamber is given by

$$C_{pg}\ (m_a + m_f)\ (T_5 - T_4) = m_f \times \text{C.V.}$$

$$\therefore \quad C_{pg} \left(\frac{m_a}{m_f} + 1\right) (T_5 - T_4) = \text{C.V.}$$

$$1 \left(\frac{m_a}{m_f} + 1\right) (850 - 392) = 45000$$

$$\therefore \quad \frac{m_a}{m_f} = \frac{45000}{458} - 1 = 98.2 - 1 = \mathbf{97.2}$$

$$T_6 = T_5 \left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = 850 \left(\frac{1}{6.5}\right)^{0.285} = \frac{850}{1.7} = \mathbf{500\ K.}$$

∴ Work done per kg of exhaust gases in turbine

$$= C_{pg}\ (T_5 - T_6) = 1\ (850 - 500) = 350 \text{ kJ/kg}$$

When 1 kg of fuel is used, the mass of air supplied = 97.2 kg.

∴ Net work available for electric power generation per kg of fuel supplied

$$= (1 + m_a) \times 350 - m_a \times 184 = 98.2 \times 350 - 97.2 \times 184$$

$$= 34370 - 17885 = \mathbf{16485\ kJ/kg\ of\ fuel}$$

If m_f is kg of fuel used per sec then

$$m_f \times 16485 = 10 \times 1000$$

$$\therefore \quad m_f = \frac{10 \times 1000}{16485} = 0.607 \text{ kg/sec} = \mathbf{2183.8\ kg/hr.}$$

Work saved per hour due to intercooling

$$= 26 \times 2183.8 \times 97.2 = 5518900 \text{ kJ/hr} = \mathbf{1533\ kW.}$$

If intercooling is not used, the power available for electric generation

$$= 10 - 1.5 = \mathbf{8.5\ MW.}$$

Thermal efficiency of the plant is given by

$$= \frac{C_{pg}(m_a + m_f)(T_5 - T_6) - 2C_{pa}m_a(T_2 - T_1)}{C_{pg}(m_a + m_f)(T_5 - T_4)}$$

$$= \frac{\left(\frac{m_a}{m_f} + 1\right)(T_5 - T_6) - 2\frac{m_a}{m_f}(T_2 - T_1)}{\left(\frac{m_a}{m_f} + 1\right)(T_5 - T_4)}$$

$$= \frac{(98.2)(850 - 500) - 2 \times 97.2(392 - 300)}{98.2(850 - 392)} = 0.367 = \mathbf{36.7\%.}$$

Problem 24.8. *An open cycle gas turbine power plant works on Baryton cycle. The maximum pressure and temperature of the cycle are limited to 5 bar and 900 K. The pressure and temperature of the gas entering into the compressor are 1 bar and 27°C. Reheating is used at a pressure of 2.5 bar where the temperature of the gases is increased to its original turbine inlet temperature. The air flow per second through the plant is 10 kg/sec. Determine the thermal efficiency and plant capacity in MW. The exhaust pressure of the turbine is also 1 bar.*

Assume the compression and expansions are isentropic.

Take $\gamma = 1.4$ for air and gases, $C_p = 1$ kJ/kg-K for air and gases.

and C.V of fuel = 40,000 kJ/kg.

Neglect the pressure losses in the system. Do not neglect the fuel quantity.

Solution. The processes are shown in Fig. Prob. 24.8.

$p_1 = 1$ bar, $T_1 = 27 + 273 = 300$ K.

$p_2 = 5$ bar, $T_3 = 900$K,

$p_i = 2.5$ bar, $\gamma = 1.4$ for air and gases,

$C_p = 1$ kJ/kg-K for air and gases.

All the processes are isentropic

$$T_2 = T_3\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 300(5)^{0.285} = \mathbf{474K}$$

$$T_4 = T_3\left(\frac{p_i}{p_2}\right)^{(\gamma-1)/\gamma} = 900\left(\frac{2.5}{5.0}\right)^{0.285} = \frac{900}{1.218} = \mathbf{738K}$$

$$T_6 = T_5\left(\frac{p_1}{p_i}\right)^{(\gamma-1)/\gamma}\left(\frac{1}{2.5}\right)^{0.285} = \frac{900}{1.30} = \mathbf{692K}$$

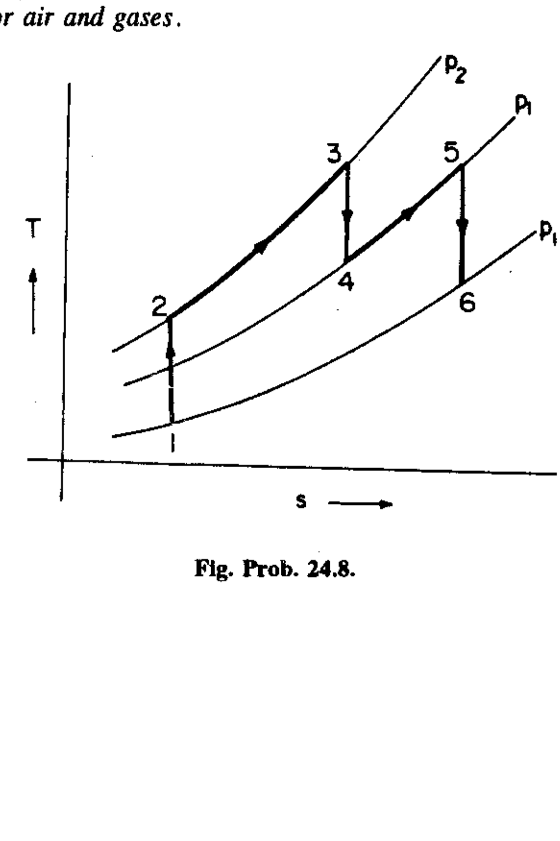


Fig. Prob. 24.8.

If m_{f1} and m_{f2} are the quantities of fuels used per kg of air flow in the first and second combustion chambers,

$$\Rightarrow \quad C_{pg}(1 + m_{f1})(T_3 - T_2) = m_{f1} \times \text{C.V.}$$

$$\therefore \quad 1\left(\frac{1}{m_{f1}} + 1\right)(T_3 - T_2) = \text{C.V.}$$

$$1\left(\frac{1}{m_{f1}} + 1\right)(900 - 474) = 40000$$

$$\therefore \quad \frac{1}{m_{f1}} = \frac{40000}{1 \times 426} - 1 = \mathbf{92.9}$$

$$\therefore \quad m_{f1} = \mathbf{0.0108\ kg/kg\ of\ air}$$

Similarly, $C_{pg}\ (1 + m_{f1} + m_{f2})\ (T_5 - T_4) = m_{f2} \times \text{C.V.}$

$$\therefore \quad 1\left(\frac{1}{m_{f2}} + 1.0108\right)(T_5 - T_4) = \text{C.V.}$$

$$\therefore \quad 1\left(\frac{1}{m_{f2}} + 1.0108\right)(900 - 738) = 40{,}000$$

$$\therefore \quad \frac{1}{m_{f2}} = \frac{40{,}000}{1 \times 162} - 1.0108 = \mathbf{246}$$

$$\therefore \quad m_{f2} = \mathbf{0.00406\ kg/kg\ of\ air}$$

Net work done per kg of air flow

$= C_{pg}\ (1 + m_{f1})\ (T_3 - T_4) + C_{pg}(1 + m_{f1} + m_{f2})\ (T_5 - T_6) - C_{pa}\ (T_2 - T_1)$

$= 1 \times 1.0108\ (900 - 738) + 1 \times 1.015\ (900 - 692) - 1 \times 1\ (474 - 300)$

$= 163.75 + 211 - 174 =$ **200.75 kJ/kg of air.**

Net heat supplied per kg of air passing through the system

$= (m_{f1} + m_{f2}) \times \text{C.V.} = (0.0108 + 0.00406) \times 40{,}000 =$ **594.5 kJ**

Thermal efficiency $= \dfrac{200.75}{594.5} = 0.338 =$ **33.8%**

Capacity of the plant in kW = 10 × 200.75 = **2007.5 kW.**

Problem 24.9. *A single unit open cycle gas turbine plant is designed to generate 2 MW power. The inlet temperature and pressure are 27°C and 1 bar. The pressure ratio of the cycle is 5. Air coming out of compressor absorbs heat from the exhaust gases in a regenerator at a rate of 80 kJ per kg of air. The air is further expanded at a constant pressure by the combustion of 0.01 kg of fuel per kg of air, the fuel having a calorific value of 40,000 kJ/kg. The products of combustion expanded in the turbine to 1 bar and exhausted with negligible velocity after yielding some of their heat to the air leaving the compressor.*

Estimate the theoretical thermal efficiency of the plant. Compare this efficiency with that of a normal constant pressure cycle.

Also find the fuel consumption per hour. Take C_p = 1 kJ/kg-K and γ = 1.4 both for air and gases.

Neglect the pressure and heat losses of the plant and take the isentropic efficiency of 85% both for compressor and turbine.

Solution. The arrangement of the components and corresponding $T - s$ diagram is shown in Fig. Prob. 24.9 (*a*) and Fig. Prob. 24.9 (*b*).

$$T_2' = T_1 \left[\frac{p_2}{p_1}\right]^{(\gamma - 1)/\gamma} = 300(5)^{0.285} = 475\text{K}$$

$$\eta_c = 0.85 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{475 - 300}{T_2 - 300}$$

$\therefore \qquad T_2 = 300 + \dfrac{175}{0.85} = \mathbf{506K.}$

Heat absorbed by the air from the exhaust gases in the regenerator

$= C_{pa} \cdot m_a (T_3 - T_2) = 80$

$\therefore \qquad 1 \times 1\ (T_3 - 506) = 80$

$\therefore \qquad T_3 = 506 \times \dfrac{80}{1} \doteq \mathbf{586\ K.}$

Heat given to the air in the combustion chamber is given by

$m_f \times \text{C.V.} = (1 + m_f)\, C_p\, (T_4 - T_3)$

where m_f is the mass of fuel supplied per kg of air

$0.01 \times 40{,}000 = (1 + 0.01) \times 1\ (T_4 - 586)$

$\therefore \qquad T_4 = 586 + \dfrac{400}{1.01 \times 1} = 586 + 396 = \mathbf{982\ K}$

$T_5' = T_4 \left[\dfrac{p_1}{p_2}\right]^{(\gamma-1)/\gamma} = 1002 \left[\dfrac{1}{5^{0.285}}\right] = \mathbf{630\ K}$

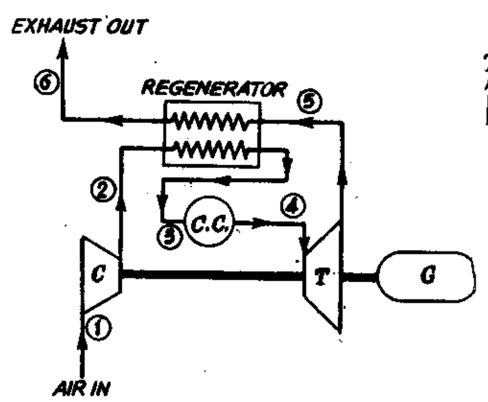


Fig. Prob. 24.9 (*a*).

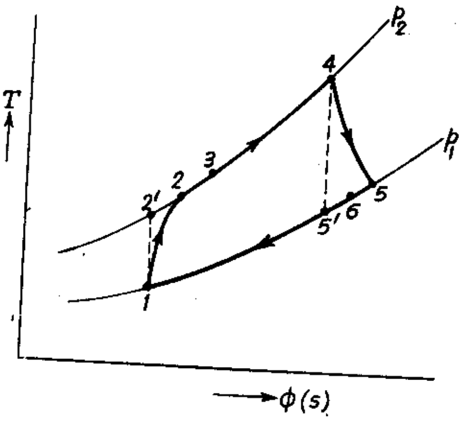


Fig. Prob. 24.9 (*b*).

$\eta_t = 0.85 = \dfrac{T_4 - T_5}{T_4 - T_5'} = \dfrac{1002 - T_5}{1002 - 630}$

$\therefore \qquad T_5 = 1002 - 0.85 \times 372 = 1002 - 316 = \mathbf{686\ K.}$

Thermal efficiency $= \dfrac{(T_4 - T_5) - (T_2 - T_1)}{(T_4 - T_3)}$ neglecting fuel mass

$= \dfrac{(982 - 686) - (506 - 300)}{(982 - 586)} = 0.227 = \mathbf{22.7\%}.$

Thermal efficiency $= \dfrac{\text{Work done per sec}}{\text{Heat supplied per sec}}$

$\therefore$ Heat supplied per sec $= \dfrac{2 \times 1000}{0.227} = \mathbf{8810.6\ kJ/sec}$

$\therefore$ Fuel required per hour $= \dfrac{8810.6}{40{,}000} = 0.22$ kg/sec $=$ **793 kg/hr**

Efficiency of normal constant pressure cycle

$$= 1 - \frac{1}{(R_p)^{(\gamma-1)/\gamma}}$$

$$= 1 - \frac{1}{5^{0.285}} = 1 - \frac{1}{1.85} = 1 - 0.54 = \mathbf{0.46 \text{ or } 46\%}.$$

Problem 24.10. *A gas turbine power plant consists of two stage compressor and single stage turbine with a regenerator. The air is taken into the compressor at 20°C and 1 bar. The maximum temperature of the cycle is limited to 700°C and maximum pressure ratio is 6. The effeectiveness of regenerator is 0.7. Assuming the following data, find :*

(*a*) *The air-fuel ratio used,*

(*b*) *Thermal efficiency of the cycle and*

(*c*) *Specific fuel consumption of the plant and fuel consumption per hour.*

Take

Air flow through the plant = 200 kg/sec.

Isentropic efficiency of both compressors = 0.82.

Isentropic efficiency of the turbine = 0.92.

Combustion efficiency = 0.96.

Mechanical efficiency = 0.96.

Generation efficiency = 0.95.

C.V. of fuel used = 35000 kJ/kg.

Take C_p = 1 kJ/kg-K and γ = 1.4 for both air and gases.

An intercooler is used between the two compressors and assume that there is perfect intercooling. Neglect the heat and pressure losses in the system.

Solution. The arrangement of the components and corresponding $T-s$ diagram are shown in Fig. Prob. 24.10 (*a*) and Fig. Prob. 24.10 (*b*).

$$p_1 = 1 \text{ bar}, p_2 = 6 \text{ bar as } \frac{p_2}{p_1} = 6 \text{ given}$$

p_i (pressure in intercooler) $= \sqrt{P_1 P_2} = \sqrt{6} = 2.44$ bar

$$T_2' = T_1 \left[\frac{p_i}{p_1}\right]^{(\gamma-1)/\gamma} = (273 + 20)\ (2.44)^{0.286} = \mathbf{378\ K}$$

$$\eta_c = 0.82 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{378 - 293}{T_2 - 293}$$

$$\therefore \quad T_2 = 293 + \frac{85}{0.82} = 293 + 104 = \mathbf{397K}$$

$$T_4' = T_3 \left[\frac{p_2}{p_i}\right]^{(\gamma-1)/\gamma} = 293(2.44)^{0.286} = \mathbf{378K} \text{ as } T_3 = T_1$$

T_4 = **397K** as $\eta_{c1} = \eta_{c2}$

$$T_7' = T_6 \left[\frac{p_1}{p_2}\right]^{(\gamma-1)/\gamma} = \frac{(700 + 273)}{(6)^{0.386}} = \frac{973}{1.667} = \mathbf{620K}$$

$$\eta_t = 0.92 = \frac{T_6 - T_7}{T_6 - T_7'} = \frac{973 - T_3}{973 - 620}$$

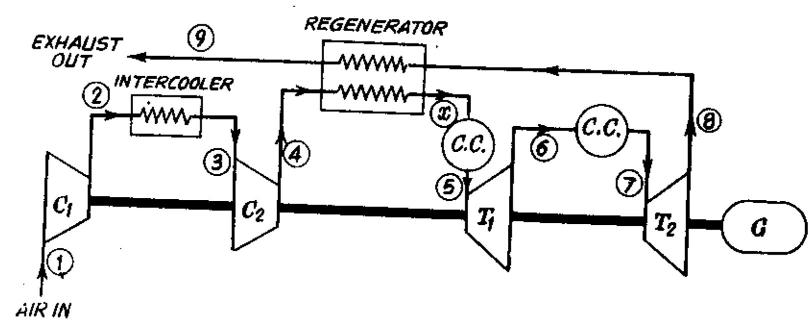


Fig. Prob. 24.10 (*a*).

$\therefore T_7 = 973 - 0.92 \times 353 = 973 - 326 = \mathbf{648K}$

$\varepsilon = 0.7 = \dfrac{T_5 - T_4}{T_7 - T_4}$ assuming $m_a \approx m_g$ as $m_f << m_a$.

$\therefore \quad 0.7 = \dfrac{T_5 - 397}{648 - 397}$

$\therefore \quad T_5 = 397 + 0.7 \times 251 = \mathbf{573K}.$

Heat generated by the fuel = Heat absorbed by the gases

$m_f \times \text{C.V.} \times \eta_{com} = C_p\,(1 + m_f)\,(T_6 - T_5)$

where m_f is the mass of fuel supplied per kg of air.

$\therefore \quad m_f \times 35000 \times 0.96 = 1\,(1 + m_f)(973 - 573)$

$\therefore \quad m_f = \dfrac{1}{83}$

$\therefore \quad \dfrac{m_a}{m_f} = \dfrac{83}{1}$

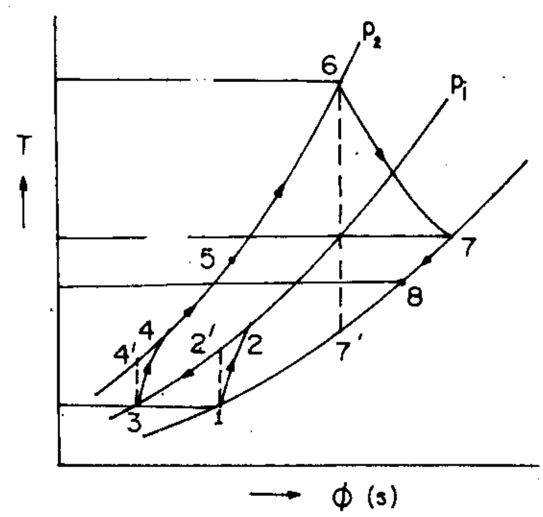


Fig. Prob. 24.10 (*b*).

Thermal efficiency $= \dfrac{(T_6 - T_7) - [(T_4 - T_3) + (T_2 - T_1)]}{(T_6 - T_5)}$ neglecting the mass of fuel

$= \dfrac{(973 - 648) - 2\,(397 - 293)}{(973 - 573)}$ as $(T_4 - T_3) = (T_2 - T_1)$

$= 0.293 = \mathbf{29.3\%}.$

Thermal efficiency $= \dfrac{\text{Work done per kg of air}}{\text{Heat supplied per kg of air}}$

$\therefore$ Work done per kg of air = 1 (973 – 573) × 0.293 = **117.2 kJ.**

Work done per sec = 117.2 × 200 = **23440 kJ/sec**

Capacity of the plant = 23440 kW = **23.44 MW**

Power available at generation terminals = 23440 × 0.96 × 0.95 = **21377 kW.**

Fuel consumption per hour $= 200 \times 3600 \times \dfrac{1}{83}$ kg = **8675 kg/hr.**

Specific fuel consumption $= \dfrac{8675}{21377}$ = **0.406 kg/kW/hr.**

Problem 24.11. *A single unit regenerative type gas turbine power plant of 5 MW capacity supplies the power to the consumers at 0.8 capacity factor. The pressure and temperature of the air entering into the compressor are 1 bar and 27°C. The maximum temperature of the cycle is limited to 1000 K and maximum pressure ratio is limited to 5.*

Using the following data, find the mass of air flow through the compressor per second and the effectiveness of the regenerator.

Isentropic efficiency of compressor = 85%, Isentropic efficiency of turbine = 90%

Combustion eefficiency = 95%, Mechanical efficiency = 95%

Generation efficiency = 92%, Transmission losses = 10%

C.V. of fuel used = 40,000 kJ/kg, Air-Fuel ratio = 80 : 1

C_{pa} *= 1 kJ/kg-K,* C_{pg} *= 1.1 kJ/kg-K,* γ *(for air and gas) = 1.4*

If the cost of fuel used is Rs. 5,000 per tonne and all other charges including profit are Rs. 5000 per hour, find the charges of energy.

Neglect the heat and pressure losses in the system and assume the plant is running at constant load.

Solution. The arrangement of the components and corresponding $T-s$ diagram is shown in Fig. Prob. 24.11 (*a*) and Fig. Prob. 24.11 (*b*).

The given data is

$P_1 = 1$ bar $\qquad \eta_c = 85\%$

$\dfrac{P_2}{P_1} = 5 \qquad \eta_t = 90\%$

$T_1 = 27 + 273 = 300\text{K} \qquad \eta_{com} = 95\%$

$T_4 = 1000\text{K} \qquad \eta_m = 95\%$

$\dfrac{m_a}{m_f} = \dfrac{80}{1} \qquad \eta_g = 92\%$

C.F. = 0.8 $\qquad \eta_{tt}$ (transmission) $= 1 - 0.1 = 0.9 = 90\%$

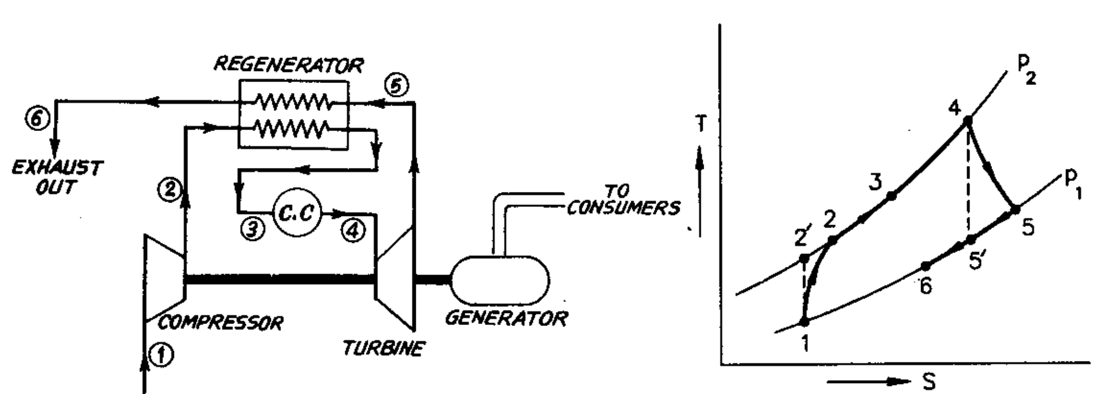


Fig. Prob. 24.11. Arrangement of components and corresponding $T-s$ diagram.

$$\text{The capacity factor} = \frac{\text{Average load}}{\text{Plant capacity}}$$

$\therefore$ Average load $= 5 \times 0.8 = 4$ MW

$$T_2 = T_1 \left[\frac{P_2}{P_1}\right]^{(\gamma-1)/\gamma} = 300\ (5)^{(1.4-1)/1.4} = 300\ (5)^{0.286} = \mathbf{475K}$$

$$\eta_c = 0.85 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{475 - 300}{T_2 - 300}$$

$$\therefore \quad T_2 = 300 + \frac{175}{0.85} = \mathbf{506K}$$

$$T_5' = T_4 \left(\frac{P_1}{P_2}\right)^{(\gamma-1)/\gamma} = \frac{1000}{(5)^{0.286}} = \frac{1000}{1.585} = \mathbf{630K}$$

$$\eta_t = 0.9 = \frac{T_4 - T_5}{T_4 - T_5'} = \frac{1000 - T_5}{1000 - 630}$$

$$\therefore \quad T_5 = 1000 - 0.9 \times 370 = \mathbf{667K.}$$

Work developed per second

$= [(m_a + m_f)\ C_{pg}\ (T_4 - T_5) - m_a\ C_{pa}\ (T_2 - T_1)]\ \eta_m \cdot \eta_g \cdot \eta_{tt}$ kJ/sec. ...(a)

where m_a and m_f are the masses of air and fuel supplied per second.

The energy generated per second and supplied to the consumer

$= 5 \times 1000 = 5000$ kJ/sec ...(b)

Equating (a) and (b), we get

$$m_a \left[\left(1 + \frac{m_f}{m_a}\right) C_{pg}\ (T_4 - T_5) - C_{pa}\ (T_2 - T_1)\right] \eta_m \cdot \eta_g \cdot \eta_{tt} = 5000$$

Substituting the given values

$$m_a \left[\left(1 + \frac{1}{80}\right) 1.1\ (1000 - 667) - 1\ (506 - 300)\right] \times 0.95 \times 0.82 \times 0.9 = 5000.$$

$\therefore$ $m_a\ (370.9 - 206) = 5000$

$$\therefore \quad m_a = \frac{5000}{164.9} = \mathbf{30.32\ kg/sec.}$$

Heat generated in combustion chamber = Heat gained by the air and fuel.

$\therefore$ $m_f \times$ C.V. $\times\ \eta_{com} = C_{pg}\ (m_a + m_f)\ (T_4 - T_3)$

$$\therefore \quad \text{C.V.} \times \eta_{com} = \dot{C}_{pg} \left(\frac{m_a}{m_f} + 1\right) (T_4 - T_3).$$

Substituting the given values

$40{,}000 \times 0.95 = 1.1\ (80 + 1)\ (1000 - T_3)$

$$\therefore \quad T_3 = 1000 - \frac{40{,}000 \times 0.95}{1.1 \times 81} = 100 - 426.5 = \mathbf{573.5K}$$

$$\varepsilon \text{ (effectiveness of regenerator)} = \frac{C_{pa}\ m_a\ (T_3 - T_2)}{C_{pg}\ m_g\ (T_5 - T_2)} = \frac{C_{pa}\ m_a\ (T_3 - T_2)}{C_{pg}\ (m_a + m_f)\ (T_5 - T_2)}$$

$$= \frac{C_{pa}\ (T_3 - T_2)}{C_{pg}\left[1 + \frac{m_f}{m_a}\right](T_5 - T_2)} = \frac{1\ (573.5 - 506)}{1.1\left[1 + \frac{1}{80}\right](667 - 506)} = \mathbf{0.376}$$

The fuel consumption per hour $= m_a \times 3600 \times \frac{1}{80} = 30.32 \times 3600 \times \frac{1}{80} = \mathbf{1364.5\ kg/hr}$

Cost of fuel per hour = $\frac{1364.5}{1000} \times 5000$ = Rs. 6822.5

$\therefore$ Total cost to be charged per hour = 6822.5 + 5000 = Rs. 11622.5

Energy generated per hour = 5 × 1000 × 1 = 5000 kW-hr.

$\therefore$ Charges of energy per kW-hr = $\frac{11622.5}{5000}$ = **Rs. 2.32/kWh.**

Problem 24.12. *A gas turbine power plant of 10 MW capacity works on closed cycle using air as working medium. The plant is designed for maximum specific work output. The temperature of the air at inlet is 300K and maximum temperature in the cycle is limited to 960K. Find the cost of energy generated if the plant is running at designed capacity.*

Use the following data :

Isentropic efficiency of compressor	*= 0.8*
Isentropic efficiency of turbine	*= 0.9*
Mechanical efficiency	*= 0.95*
Generation efficiency	*= 0.95*
Combustion efficiency	*= 0.96*

90% of the heat developed in the combustion chamber is given to the air working into the system.

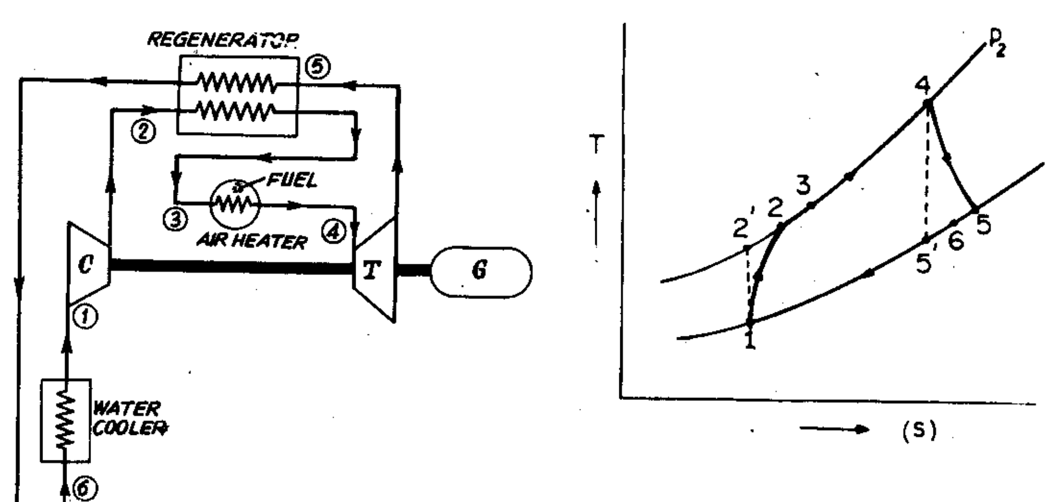
Effectiveness of regenerator	*= 0.7*
Cost of fuel used	*= Rs. 4000 per tonne.*
C.V. of fuel used	*= 40,000 kJ/kg.*

All other charges including profit per hour = Rs. 3000/hr.

Neglect heat and pressure losses in the system.

Solution. The arrangement of the components and corresponding $T - s$ diagram are shown in Fig. Prob. 24.12 (*a*) and Fig. Prob. 24.12 (*b*).

As the plant is designed for maximum specific output,



(*a*) (*b*)

Fig. Prob. 24.12.

$$R_p = \frac{P_2}{P_1} = \left[\eta_c\,\eta_t\,\frac{T_3}{T_1}\right]^{\gamma/[2(\gamma-1)]} = \left(0.8 \times 0.9 \times \frac{960}{300}\right)^{1.4/0.8} = (2.3)^{1.75} = 4.3$$

$$T_2' = T_1\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} = 300\,(4.3)^{0.286} = 300 \times 1.516 = \mathbf{455K}$$

$$\eta_c = 0.8 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{455 - 300}{T_2 - 300}$$

$$\therefore \quad T_2 = 300 + \frac{155}{0.8} = 300 + 194 = \mathbf{494K}$$

$$T_5' = T_4\left(\frac{P_1}{P_2}\right)^{(\gamma-1)/\gamma} = 960\left(\frac{1}{4.3}\right)^{0.286} = \frac{960}{1.516} = \mathbf{635K}$$

$$\eta_t = 0.9 = \frac{T_4 - T_5}{T_4 - T_5'} = \frac{960 - T_5}{960 - 635}$$

$$\therefore \quad T_5 = 960 - 0.9 \times 325 = \mathbf{668K}$$

Work developed per hour (neglecting fuel mass)

$$= m_a\,C_{pa}\,[(T_4 - T_5) - (T_2 - T_1)] \times \eta_m \times \eta_g = 10 \times 1000$$

where, m_a is the mass of air passed per sec

$$m_a \times 1\,[(960 - 668) - (494 - 300)] \times 0.95 \times 0.95 = 10000$$

$$\therefore \quad m_a = \frac{10000}{1 \times 98 \times 0.95 \times 0.95} = \mathbf{113\ kg/sec}$$

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2} \text{ (neglecting fuel mass)}$$

$$\therefore \quad 0.7 = \frac{T_3 - 494}{668 - 494}$$

$$\therefore \quad T_3 = 494 + 174 \times 0.7 = 494 + 122 = \mathbf{616K}$$

Now $m_f \times \text{C.V.} \times \eta_{com} \times 0.9 = m_a\,C_{pa}\,(T_4 - T_3)$.

where m_f is the mass of fuel burned per sec

$$\therefore \quad m_f \times 40000 \times 0.96 \times 0.9 = 113 \times 1 \times (960 - 616)$$

$$\therefore \quad m_f = \frac{113 \times 1 \times 344}{40000 \times 0.96 \times 0.9} = \mathbf{1.125\ kg/sec}$$

Cost of fuel $= \dfrac{1.125 \times 3600}{1000} \times 4000 =$ Rs. **16200/hr**

Total cost per hour $= 16200 + 3000 =$ **19200 rupees/hr.**

$\therefore$ Cost of energy generated $= \dfrac{19200}{10000} =$ Rs. **1.92/kWh.**

$\therefore$ Air-fuel ratio $= \dfrac{113}{1.125} =$ **100 : 1.**

Problem 24.13. *The following data are given for a gas turbine power plant which works on constant pressure open-cycle and consists of compressor, regenerator, combustion chamber and turbine. The compressor, turbine and generator are mounted on the same shaft. The pressure and temperature of air entering into the compressor are 1 bar and 27°C and pressure of air leaving the compressor is 4 bar.*

Isentropic efficiency of the compressor = 80%.

Isentropic efficiency of the turbine = 85%.

Effectiveness of the regenerator = 75%.

Pressure loss in regenerator along air side = 0.1 bar.

Pressure loss in regenerator along gas side = 0.1 bar.

Pressure loss in the combustion chamber = 0.05 bar.

Combustion efficiency = 90%.

Mechanical efficiency = 90%.

C.V. of the fuel used = 40,000 kJ/kg.

Flow of air = 25 kg/sec.

Atmospheric pressure = 1.03 bar.

Find the power available at the generator terminals if the generation efficiency is 95%. Also find the overall efficiency of the plant and specific fuel consumption.

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

$C_{pa} = 1$ *kJ/kg-°C and* $C_{pg} = 1.1$ *kJ/kg-°C.*

The maximum temperature of the cycle is limited to 700°C.

Solution. The arrangements of the components and corresponding to $T - s$ diagram are shown in Fig. Prob. 24.13 (*a*) and 24.13 (*b*).

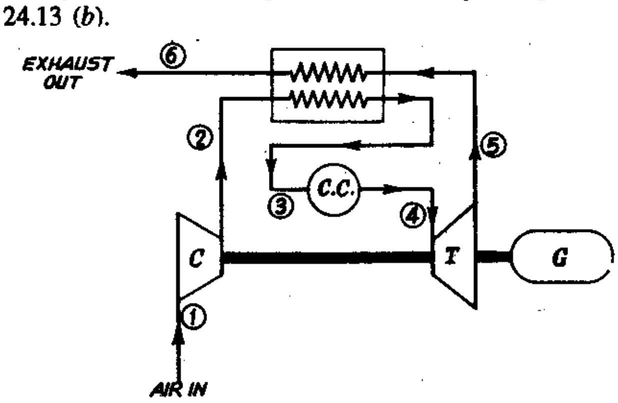


Fig. Prob. 24.13 (*a*).

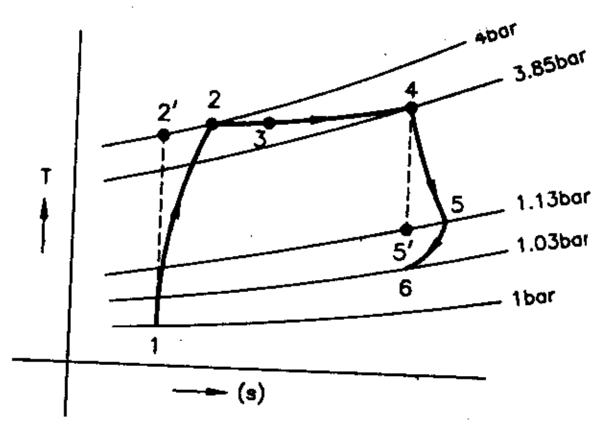


Fig. Prob. 24.13 (*b*).

Pressure at the inlet of the turbine = 4 – (0.1 + 0.05) = **3.85 bar**
Pressure at the exit of the turbine = 1.03 + 0.1 = **1.13 bar.**

$$T_2' = T_1 \left[\frac{4}{1}\right]^{(\gamma-1)/\gamma} = 300\ (40)^{0.286} = \mathbf{446K}$$

$$\eta_c = 0.8 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{446 - 300}{T_2 - 300} \quad \therefore \quad T_2 = \mathbf{483K}$$

$$T_4 = 700 + 273 = 973\text{K}$$

$$T_5' = T_3 \left[\frac{1.13}{3.85}\right]^{0.286} = 937 \left[\frac{1}{3.4}\right]^{0.286} = \mathbf{685K}$$

$$\eta_t = 0.85 = \left(\frac{T_4 - T_5}{T_4 - T_5'}\right) = \frac{973 - T_5}{973 - 685} \quad \therefore \quad T_5 = \mathbf{728K.}$$

Considering the combustion process in combustion chamber

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

$$\therefore \quad \text{C.V.} \times \eta_{com} = \left(\frac{m_a}{m_f} + 1\right) C_{pg}\, (T_4 - T_3)$$

$$\therefore \quad 40{,}000 \times 0.9 = \left(\frac{m_a}{m_f} + 1\right) \times 1.1\ (973 - T_3) \qquad ...(a)$$

$$\varepsilon = \frac{m_a C_{pa}\, (T_3 - T_2)}{(m_a + m_f)\, C_{pg}\, (T_5 - T_2)}$$

$$\therefore \quad 0.75 = \frac{1\ (T_3 - 4883)}{\left(1 + \frac{m_f}{m_a}\right) \times 1.1\ (728 - 483)} \qquad ...(b)$$

Solving the equations (*a*) and (*b*), we get

$$T_3 = \mathbf{486K} \text{ and } \frac{m_a}{m_f} = \mathbf{74 : 1}$$

$$W_c = C_{pa}\, (T_2 - T_1) = 1\ (483 - 300) = \mathbf{183\ kJ/kg\ of\ air}$$

$$W_t = C_{pg} \times (1 + m_f)\, (T_4 - T_5) = 1.1 \left(\frac{75}{74}\right) (973 - 728) = \mathbf{273\ kJ/kg\ of\ air}$$

$$W_a = 273 - 183 = \mathbf{90\ kJ/kg\ of\ air.}$$

Work available per kg of air at the terminals of generator

$$= 90 \times \eta_m \times \eta_g = 90 \times 0.9 \times 0.95 = \mathbf{76.95\ kJ/kg.}$$

Power available at the generator terminal $= \frac{25 \times 76.95}{1000} = \mathbf{1.92375\ MW}$

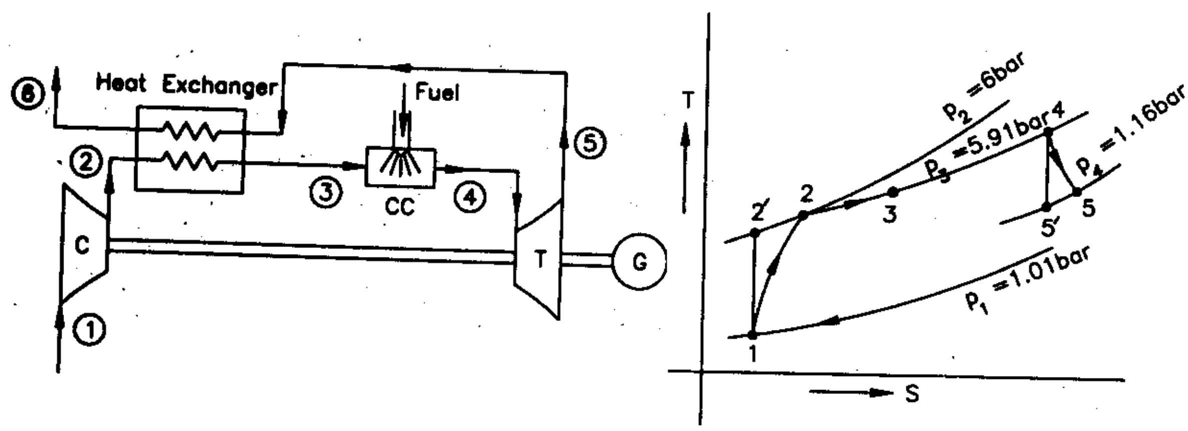
Overall efficiency $= \frac{76.95}{\frac{1}{74} \times 40{,}000} \times 100 = \mathbf{14.235\%}$

Fuel required per hour $= 25 \times 3600 \times \frac{1}{74} = \mathbf{1216\ kg/hr.}$

$\therefore$ Specific fuel consumption $= \frac{1216}{1.92375 \times 1000} = \mathbf{0.63\ kg/kW\text{-}hr.}$

Problem 24.14. *A gas turbine plant of 800 kW capacity takes the air at 1.01 bar and 15 C. The pressure ratio of the cycle is 6 and maximum temperature is limited to 700°C. A regenerator of 75% effectiveness is added in the plant to increase the overall efficiency of the plant. The pressure drop in the combustion chamber is 0.15 bar as well as in the regenerator is also 0.15 bar. Assuming the isentropic efficiency of the compressor 80% and of the turbine 85%, determine the plant thermal efficiency. Neglect the mass of the fuel.*

Solution. The arrangement of the components is shown in Fig. Prob. 24.14 (*a*) and the processes are represented on $T - s$ diagram as shown in Fig. Prob. 24.14 (*b*).



(*a*) (*b*)

Fig. Prob. 24.14.

The given data is

$$T_1 = 15 + 273 = 288 \text{ K}$$
$$p_1 = 1.01 \text{ bar}$$
$$p_2 = 1.01 \times 6 = 6.06 \text{ bar}$$
$$R_p = \frac{p_2}{p_1} = 6$$

Pressure at point 4 = 6.06 – 0.15 = 5.91 bar

Applying isentropic law to the process 1-2

$$T_2' = T_1 (R_p)^{\frac{\gamma - 1}{\gamma}} = 288\ (6)^{0.286} = \mathbf{480\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} \quad \therefore \quad T_2 = T_1 + \eta_c (T_2' - T_1) = 288 + 0.8\ (480 - 288) = \mathbf{528\ K}$$

$$p_3 = 6.06 - 0.15 = 5.91 \text{ bar}$$

and $$p_4 = 1.01 + 0.15 = 1.16 \text{ bar}$$

Applying isentropic law to the process 4-5′

$$T_5' = \frac{T_4}{\left(\frac{p_3}{p_4}\right)^{\frac{\gamma - 1}{\gamma}}} = \frac{(700 + 273)}{\left(\frac{5.91}{1.16}\right)^{0.286}} = \frac{973}{1.59} = \mathbf{612\ K}$$

$$\eta_t = \frac{T_4 - T_5}{T_4 - T_5'} \quad \therefore \quad T_5 = T_4 - \eta_t (T_4 - T_5') = 973 - 0.85\ (973 - 612) = \mathbf{666\ K}$$

The effectiveness of the regenerator is given by

$$\epsilon = \frac{T_3 - T_2}{T_5 - T_2} = 0.75$$

$\therefore \quad T_3 = T_2 + 0.75\,(T_5 - T_2) = 528 + 0.75\,(666 - 528) = \mathbf{631.5\ K}$

$W_c = C_p\,(T_2 - T_1) = 1 \times (528 - 288) = \mathbf{240\ kJ/kg}$

$W_t = C_p\,(T_4 - T_5) = 1 \times (973 - 666) = \mathbf{307\ kJ/kg}$

$\therefore \quad W_n = W_t - W_c = 307 - 240 = \mathbf{67\ kJ/kg}$

$Q_s = C_p\,(T_4 - T_3) = 1 \times (973 - 631.5) = \mathbf{341.5\ kJ/kg}$

$$\eta_{th} = \frac{W_n}{Q_s} = \frac{67}{341.5} = 0.196 = \mathbf{19.6\%}.$$

Problem 24.15. *The air supplied to a gas-turbine power plant is 10 kg/sec. 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 inter cooler. The inlet temperature is 300 K and maximum temperature is limited 1073 K.*

Take the following data

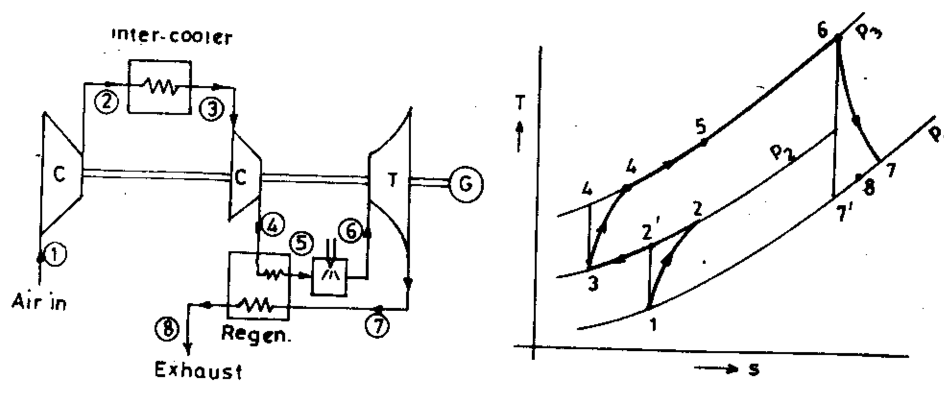
Isentropic efficiency of compressor each stage $(\eta_c) = 80\%$

Isentropic efficiency of turbine $(\eta_t) = 85\%$

A regenerator is included in the plant whose effectiveness is 0.7. Neglect the mass of fuel.

(P.U., Summer 1987)

Solution. The arrangement of the components is shown in Fig. Prob. 24.15 (*a*) and corresponding processes are represented as shown in Fig. Prob. 24.15 (*b*).



(*a*) (*b*)

Fig. Prob. 24.15.

As the cooling is perfect,

$$\therefore \quad p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 6} = 2.45 \text{ bar.}$$

Applying isentropic law to the process 1-2′

$$T_2' = T_1 \left(\frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} = 300\,(2.45)^{\frac{0.4}{1.4}} = 300\,(2.45)^{0.286} = \mathbf{387.5\ K}$$

$$\eta_{c1} = \frac{T_2' - T_1}{T_2 - T_1} = 0.8 \qquad \therefore\ T_2 = T_1 + \frac{T_2' - T_1}{0.8} = 300 + \frac{(387.5 - 300)}{0.8} = \mathbf{410\ K}$$

$W_c = W_{c1} + W_{c2} = 2\ W_{c1}$ (as intercooling is perfect)

$= 2 \times m_a\ C_{pa}\ (T_2 - T_1)$

$= 2 \times 1 \times 1\ (410 - 300) = \mathbf{220\ kJ/kg}$ ...(a)

As $T_3 = T_1$ and $R_p = \frac{p_2}{p_1} = \frac{p_3}{p_2}$ (for perfect inter-cooling).

$\therefore\ T_4 = T_2 = \mathbf{410\ K}$

Applying isentropic law to the process 6-7′

$$\frac{T_6}{T_7'} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore\ T_7' = \frac{1073}{(6)^{0.286}} = \frac{1073}{1.67} = \mathbf{642.5\ K}$$

$$\eta_t = \frac{T_6 - T_7}{T_6 - T_7'} = 0.85$$

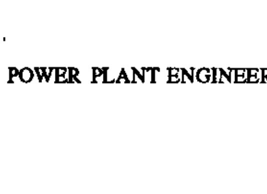
$$\therefore\ \frac{1073 - T_7}{1073 - 642.5} = 0.85 \quad \therefore\ T_7 = \mathbf{707\ K}$$

$\therefore\ W_t = C_{pa}\ (T_6 - T_7) = 1\ (1073 - 707) = \mathbf{366\ kJ/kg}$ ...(b)

The effectiveness (ε) of regenerator is given by

$$\varepsilon = \frac{T_5 - T_4}{T_7 - T_4} = 0.7 \text{ (given)}$$

$$\therefore\ \frac{T_5 - 410}{707 - 410} = 0.7 \quad \therefore\ T_5 = \mathbf{618\ K.}$$



Heat supplied in the combustion chamber (Q_s) is given by

$Q_s = 1 \times C_{pa}\ (T_6 - T_5) = 1 \times 1\ (1073 - 618)$

$= \mathbf{457.5\ kJ/kg}$

W_n (network) $= W_t - W_c = 366 - 220 = \mathbf{146\ kJ/kg}$

Power capacity of the plant in kW

$= m_a\ W_n = 10 \times 146 = 1460$ kJ/sec $= \mathbf{1460\ kW}$

$$\text{Thermal } \eta = \frac{W_n}{Q_s} = \frac{146}{457.5} = 0.32 = \mathbf{32\%}.$$

Problem 24.16. *An open cycle constant pressure regenerative gas turbine plant of 5 MW capacity consists of two stage compressors with perfect intercooler, a compressor-turbine and separate power turbine. The maximum temperature of the cycle is limited to 650°C and pressure ratio is limited to 5. The gases coming out from the compressor turbine are reheated in a direct combustion chamber to 650°C. The pressure and temperature of the air entering into the compressor are 1 bar and 300K.*

Isentropic efficiency of each stage of compressor = 80%

Isentropic efficiency of both turbines = 85%

Mechanical efficiency of both turbines = 98%

Combustion efficiency in both combustion chambers = 97%

Effectiveness of the regenerator = 0.7

C.V. of fuel used = 40,000 kJ/kg

C_{pa} = 1 kJ/kg-°C and C_{pg} = 1.145 kJ/kg°C

γ = 1.4 for air and γ = 1.35 for gases.

Neglect the pressure losses and heat losses in the system.

Find the following :

(*a*) *Overall efficiency of the plant.*

(*b*) *Mass flow of air through the plant per second.*

(*c*) *Specific fuel consumption.*

Solution. The arrangements of the components and corresponding $T - s$ diagrams are shown in Fig. Prob. 24.16 (*a*) and Fig. Prob. 24.16 (*b*).

As there is perfect intercooling, the intermediate pressure (p_i) between the two compressors is given by

$$p_i = \sqrt{P_1 P_2} = \sqrt{1 \times 5} = \mathbf{2.236\ bar}$$

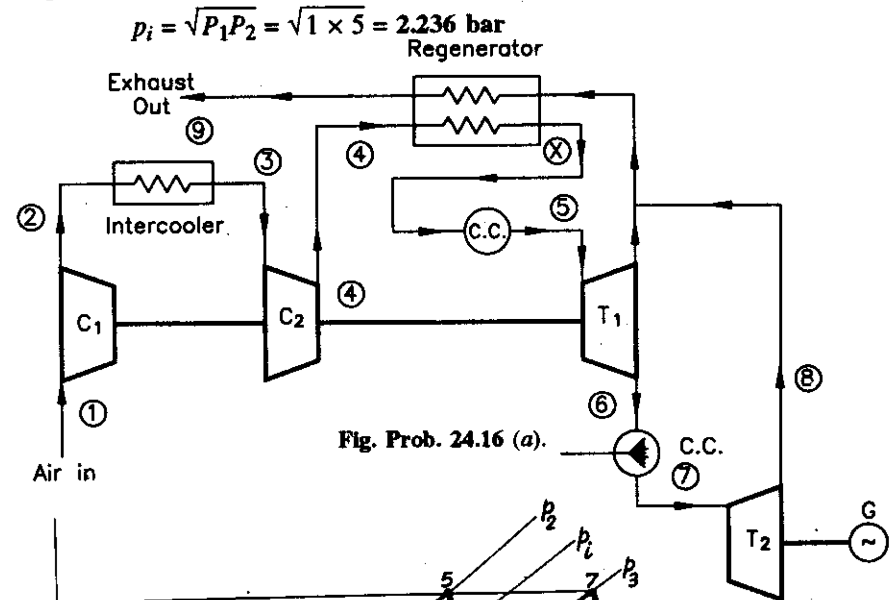


Fig. Prob. 24.16 (*a*).

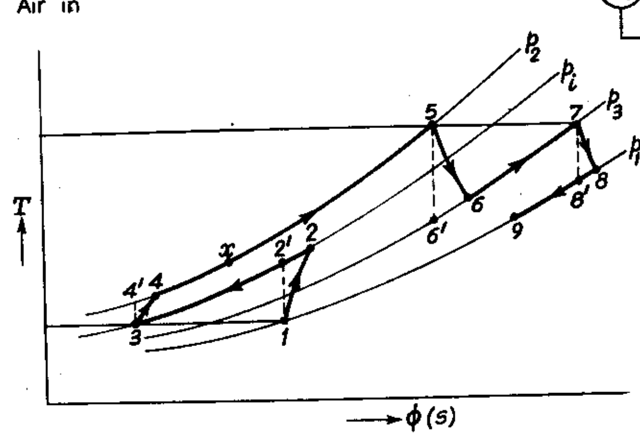


Fig. Prob. 24.16 (*b*).

$$T_2 = T_1\ (2.236)^{(\gamma - 1/\gamma)} = 300\ (2.236)^{0.286} = 377\text{K}.$$

$$\eta_{c1} = 0.8 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{377 - 300}{T_2 - 300} \quad \therefore \quad T_2 = \mathbf{396K}$$

$$T_3 = T_1 = 300\text{K and } T_4 = T_2 = \mathbf{396K}$$

$$T_5 = T_7 = 650 + 273 = \mathbf{923K\ (given).}$$

Work developed by compressor turbine = Work required to run the compressor.

Neglecting the mass of fuel

$$C_{pg}\ (T_5 - T_6) \times \eta_m = 2C_{pa}\ (T_2 - T_1)$$

$$\therefore \quad 1.145\ (923 - T_6) \times 0.98 = 2 \times 1\ (396 - 300)$$

$$\therefore \quad T_6 = \mathbf{751K}$$

$$\eta_{t1} = 0.85 = \frac{T_5 - T_6}{T_5 - T_6'} = \frac{923 - 751}{923 - T_6'} \quad \therefore \quad T_6' = \mathbf{721K.}$$

Using the isentropic law to the points 5 and 6′

$$\frac{p_2}{p_3} = \left(\frac{T_5}{T_6'}\right)^{\gamma/(\gamma-1)}$$

$$\therefore \quad \frac{5}{p_3} = \left(\frac{923}{721}\right)^{1.35/0.35}$$

$$\therefore \quad p_3 = \frac{5}{2.58} = \mathbf{1.94\ bar}$$

$$T_8' = T_7\left(\frac{p_1}{p_3}\right)^{(\gamma-1)/\gamma}$$

$$= 927\left(\frac{1}{1.94}\right)^{0.35/1.35} = \mathbf{777K}$$

$$\eta_{t2} = 0.85 = \frac{T_7 - T_8}{T_7 - T_8'} = \frac{923 - T_8}{923 - 777} \quad \therefore \quad T_8 = \mathbf{800K.}$$

The effectiveness of the regenerator neglecting the fuel mass is given by

$$\varepsilon = \frac{(T_x - T_4)\,C_{pa}}{(T_8 - T_4)\,C_{pg}}$$

$$\therefore \quad 0.7 = \frac{(T_x - 396) \times 1}{(800 - 396) \times 1.145}$$

$$\therefore \quad T_x = \mathbf{730K.}$$

Net work available per kg of air from the power turbine neglecting the mass of fuel

$$C_{pg}\,(T_7 - T_8) \times \eta_m = 1.145\,(923 - 800) \times 0.98 = \mathbf{138\ kJ/kg\ of\ air.}$$

Heat supplied per kg of air in both combustion chambers

$$= C_{pg}\,[(T_5 - T_x) + (T_7 - T_6)] = 1.145\,[(923 - 730) + (923 - 751)]$$

$$= \mathbf{418\ kJ/kg\ of\ air}$$

$$m_f \times \text{C.V.} \times \eta_{com} = 418$$

where $\dot{m}_f$ is the total mass of fuel supplied in both combustion chambers per kg of air flow.

$$\therefore \quad m_f \times 40{,}000 \times 0.97 = 418$$

$$\therefore \quad m_f = \frac{1}{92.6}$$

$$\therefore \quad \frac{m_a}{m_f} = \frac{92.6}{1}$$

$$\therefore \quad \text{Overall efficiency} = \frac{138}{\frac{1}{92.6} \times 40{,}000} = 0.319 = \mathbf{31.9\%}$$

Say the mass of air supplied per second is m_a.

$$\therefore \quad m_a \times 138 = 5 \times 1000$$

$$\therefore \quad m_a = \frac{5000}{138} = \mathbf{36.23\ kg/sec.}$$

$$\text{Mass of fuel used per hour} = 36.23 \times 3600 \times \frac{1}{92.6} = \mathbf{1408.5\ kg/hr.}$$

$$\therefore \quad \text{Specific fuel consumption} = \frac{1408.5}{5000 \times 1} = \mathbf{0.282\ kg/kW\text{-}hr.}$$

Problem 24.17. *An open-cycle constant pressure gas turbine power plant of 1600 kW capacity of single stage compressor and two turbines with regenerator. One turbine is used to run the compressor and other is used to run the generator. Separate combustion chambr 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 300K. The maximum temperature in the compressor turbine is 1050K and in power turbine 1100K. The maximum pressure in the stream is 5 bar. The exhaust pressure of both turbines is 1 bar and both exhausts pass through the regenerator. The temperature of exhaust leaving the regenerator is 750K. Take the following data :*

$\eta_c = 80\%$

η_{t1} *(compressor turbine)* $= 85\%$

η_{t2} *(power turbine)* $= 90\%$

C.V. of fuel = 40,000 kJ/kg

η_{com} *(in both combustion chambers)* $= 95\%$

η_m *(for both turbines)* $= 90\%$

ε *(effectiveness of regenerator)* $= 0.7$

$C_{pa} = 1$ *kJ/kg-K,* $C_{pg} = 1.1$ *kJ/kg-K,* γ *(for air)* $= 1.4$ *and* γ *(for gases)* $= 1.35$.

Neglecting pressure losses, heat losses and mass of fuel, find

(a) Plant efficiency, (b) Specific fuel consumption, (c) Air-fuel ratio.

Solution. The arrangements of the components and corresponding $T-s$ diagrams are shown in Fig. Prob. 24.17 (*a*) and Fig. Prob. 24.17 (*b*).

$$T_2' = T_1\left(\frac{p_2}{p_1}\right)^{(\gamma-1)/\gamma} = 300\ (5)^{0.4/1.4} = 300\ (5)^{0.286} = \mathbf{474K}$$

$$\eta_c = 0.8 = \frac{T_2' - T_1}{T_2 - T_1} = \frac{474 - 300}{T_2 - 300}$$

$\therefore \quad T_2 = 300 + 0.8 \times 174 = \mathbf{517K.}$

Considering compressor turbine

$$T_4' = T_3\left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = 1050\left(\frac{1}{5}\right)^{0.35/1.35} = 1050\left(\frac{1}{5}\right)^{0.259} = \mathbf{692K}$$

$$\eta_{t1} = 0.85 = \frac{T_2 - T_4}{T_3 - T_4'} = \frac{1050 - T_4}{1050 - 692}$$

$\therefore \quad T_4 = 1050 - 0.85\ (1050 - 692) = \mathbf{746K.}$

Considering the power turbine

$$T_6' = T_5\left(\frac{p_1}{p_2}\right)^{(\gamma-1)/\gamma} = 1100\left(\frac{1}{5}\right)^{0.259} = \mathbf{725K}$$

$$\eta_{t2} = 0.9 = \frac{T_5 - T_6}{T_5 - T_6} = \frac{1100 - T_6}{1100 - 25}$$

$\therefore \quad T_6 = 1100 - 0.9\ (1100 - 725) = \mathbf{763K.}$

The masses of air passing through the compressor turbine and power turbine are m_{a1} and m_{a2} per second.

Power out-put of the power turbine is given by

$$m_{a2}\ C_{pg}\ (T_5 - T_6) \times \eta_m \times \eta_g = 1600$$

$\therefore \quad m_{a2} \times 1.1\ (1100 - 763) \times 0.9 \times 1 = 1600$

$\therefore \quad m_{a2} = \mathbf{4.8\ kg/sec.}$

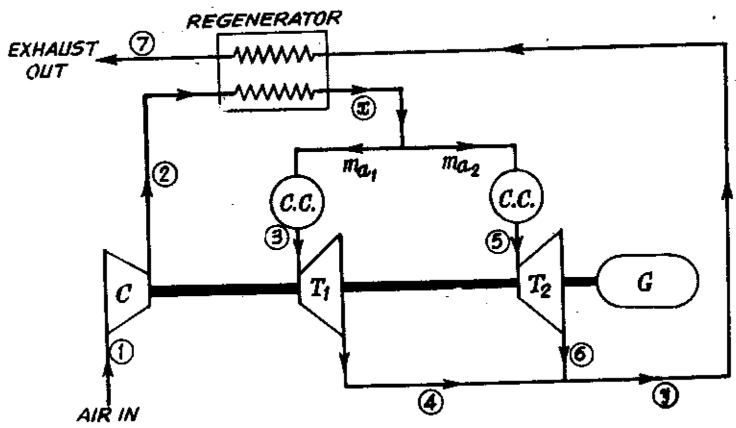


Fig. Prob. 24.17 (*a*).

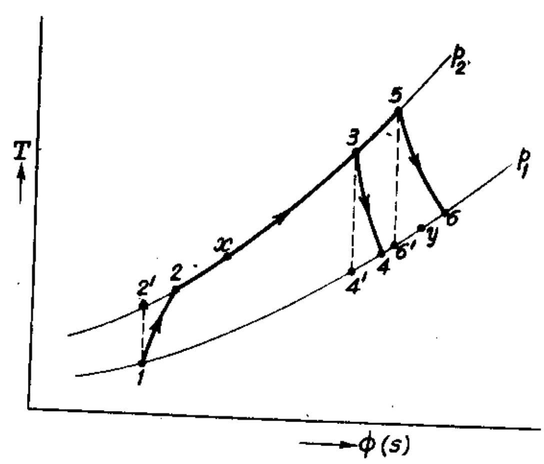


Fig. Prob. 24.17 (*b*).

Power developed by compressor turbine = Power absorbed by compressor

$\therefore \quad m_{a1} C_{pg} (T_3 - T_4) \times \eta_m = (m_{a1} + m_{a2})\, C_{pa}\, (T_2 - T_1)$

$\therefore \quad m_{a1} \times 1.1\ (1050 - 746) \times 0.9 = (m_{a1} + 4.8) \times 1 \times (517 - 300)$

$\therefore \quad$ **m_{a1} = 12.4 kg/sec.**

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 y on $T - s$ diagram and it is given by

$$C_{pg}\, (m_{a1} + m_{a2})\, T_y = C_{pg}\, m_{a1}\, T_4 + C_{pg}\, m_{a2}\, T_6$$

where, T_y is the temperature after mixing

$$\therefore \quad T_y = \frac{m_{a1}}{m_{a1} + m_{a2}} \cdot T_4 + \frac{m_{a2}}{m_{a1} + m_{a2}} \cdot T_6.$$

$$= \frac{12.4}{12.4 + 4.8}\,(746 - 273) + \frac{4.8}{12.4 + 4.8}\,(763 - 273) = 477°\text{C} = \mathbf{750K.}$$

The effectiveness of regenerator is given by

$$\varepsilon = \frac{m_a\, C_{pa}\,(T_x - T_3)}{m_g\, C_{pg}\,(T_y - T_2)} = \frac{C_{pa}\,(T_x - T_2)}{C_{pg}\,(T_y - T_2)} \text{ as } m_a = m_g \text{ as fuel mass is neglected}$$

$\therefore \quad 0.7 = \frac{1}{1.1} \times \frac{T_x - 517}{750 - 517}$

$\therefore \quad T_x =$ **694K.**

$\therefore$ Total heat supplied in both combustion chambers

$= C_{pg}\, m_{a1}\,(T_3 - T_x) + C_{pg}\, m_{a2}\,(T_5 - T_x) = m_f \times \text{C.V.} \times \eta_{com}$

$\therefore \quad 1.1\,[12.4\,(1050 - 694) + 4.8\,(1100 - 694)] = m_f \times 40{,}000 \times 0.95.$

$\therefore \quad m_f =$ **0.185 kg/sec.**

(*a*) Plant efficiency $= \frac{\text{Output}}{\text{Input}} = \frac{1600}{0.185 \times 40{,}000} = 0.216 =$ **21.6%**

Specific fuel consumption $= \frac{0.185 \times 3600}{1600} =$ **0.416 kg/kWh**

Air : Fuel ratio = (12.4 + 4.8) : 0.185 = **93 : 1.**

Problem 24.18. *A gas turbine power plant working on semiclosed cycle is shown in Fig. Prob. 24.18. There is a perfect intercooling between the two compressors. The maximum temperature in the compressor turbine is limited to 900K and power turbine to 1000K. The maximum pressure ratio in the cycle is limited to 5. The pressure and temperature of inlet air are 1 bar and 27°C respectively. The generation capacity of the power plant is 200 MW.*

Isentropic efficiency of both compressors = 85%.

Isentropic efficiency of both turbines = 90%.

Mechanical efficiency of compressor and generator shafts = 92%.

Combustion efficiency = 95%.

C.V. of fuel used = 40,000 kJ/kg.

Effectiveness of heat exchanger = 0.7.

C_p (for air and gas) = 1 kJ/kg-°C and

γ (for air and gas) = 1.4.

Neglecting the heat and pressure losses in the system, find

(*a*) *Air taken from atmosphere per second.*

(*b*) *Fuel required per second.*

(*c*) *Overall eficiency of the plant.*

(*d*) *Compressor-turbine capacity in kW.*

Solution. The arrangements of components and $T - s$ diagram for the given arrangement are shown in Fig. Prob. 24.18 (*a*) and 24.18 (*b*).

$p_1 = 1, \; \frac{p_2}{p_1} = 5, \quad \therefore \quad p_2 = 5$

$p_i = \sqrt{P_1 P_2}$ (for perfect intercooling) $= \sqrt{5} =$ **2.24 bar**

$T_6 = 1000\text{K}$ and $T_8 = 900\text{K}$ given

$$T_7' = T_6 \left[\frac{p_1}{p_2}\right]^{(\gamma-1)/\gamma} = \frac{1000}{(5)^{0.286}} = \frac{1000}{1.583} = \mathbf{630K}$$

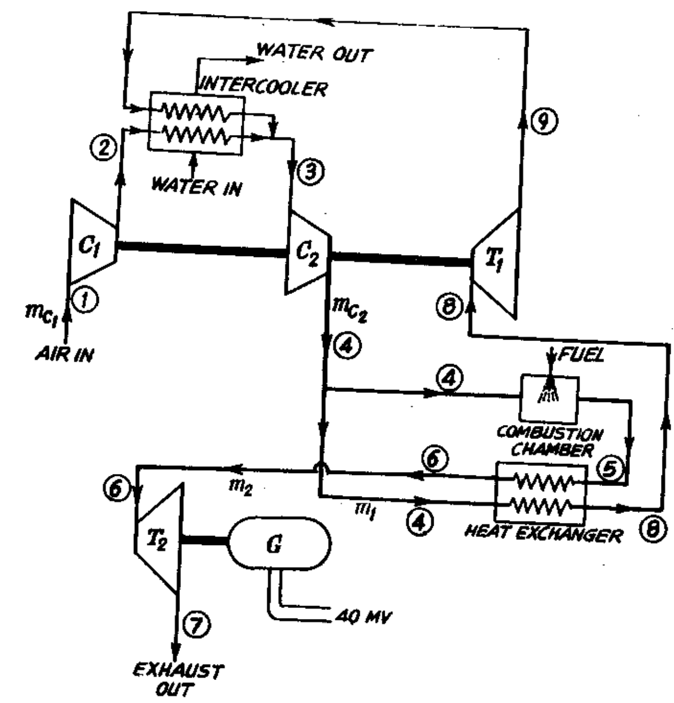


Fig. Prob. 24.18 (*a*).

$$\eta_{t2} = 0.9 = \frac{T_6 - T_7}{T_6 - T_7'} = \frac{1000 - T_7}{1000 - 630}$$

$\therefore$ $T_7 = 1000 - 370 \times 0.9 = \mathbf{667K}$.

Work done per kg of air in generator-turbine and given to generator,

$= C_p\,(T_6 - T_7)\,\eta_m$

$= 1\,(1000 - 667) \times 0.92$

$= \mathbf{306\ kJ/kg}$.

Mass of gases passing through the generator-turbine is given by

$m_2 \times 306 = 40 \times 1000$

where, m_2 is the mass of exhaust gases per second

$\therefore$ $m_2 = \dfrac{40{,}000}{306} = 131$ kg/sec

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

$= 300\,(2.24)^{0.286} = \mathbf{378K}$.

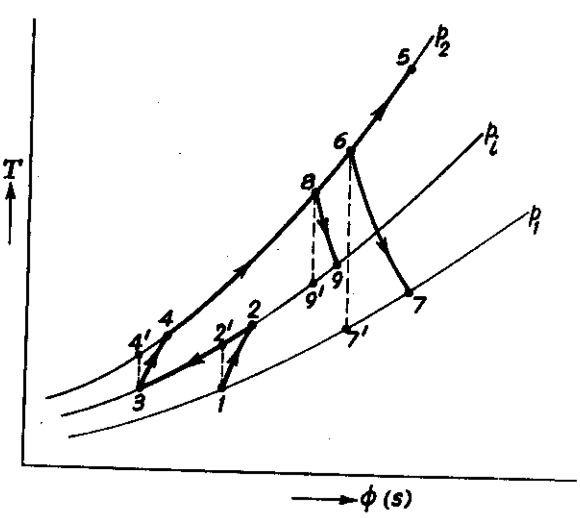


Fig. Prob. 24.18 (*b*).

Work done per kg of air in both compressors

$= 2\,C_p\,(T_2 - T_1) = 2 \times 1\,(378 - 300) = \mathbf{155.8\ kJ/kg}$

$T_4 = T_2 = 378$K as there is perfect intercooling.

Using the energy balance in regenerator (heat exhanger)

$$m_2\, C_p\,(T_5 - T_6) = m_1\, C_p\,(T_8 - T_4)$$

$$131\,(T_5 - 900) = m_1\,(900 - 378) \quad ...(a)$$

$$\varepsilon = \frac{m_1\, C_p\,(T_8 - T_4)}{m_2\, C_p\,(T_5 - T_4)} = \frac{m_1\,(900 - 378)}{131\,(T_5 - 378)} = 0.7 \quad ...(b)$$

Substituting the value from equation (*b*) into equation (*a*) we get,

$$131\,(T_5 - 900) = 0.7 \times 131\,(T_5 - 378)$$

$$\therefore \quad T_5 = \mathbf{2117\ K}$$

$$m_1 = \frac{131\,(2117 - 378) \times 0.7}{(900 - 378)} = \mathbf{306\ kg/sec.}$$

Work absorbed by compressor = Work developed by compressor turbine

$$\therefore \quad [m_{c1}\, C_p\,(T_2 - T_1) + m_{c2}\, C_p\,(T_4 - T_3)] \times \eta_m = m_1\, C_p\,(T_8 - T_9)$$

$$\therefore \quad [(m_{c1} + m_{c2})\,(T_2 - T_1)] \times \eta_m = m_1\,(T_8 - T_9) \text{ as } T_2 - T_1 = T_4 - T_3$$

$$\therefore \quad [m_{c1} + (m_{c1} + m_1)]\,(T_2 - T_1) \times \eta_m = m_1\,(T_8 - T_9) \quad ...(c)$$

$$T_9' = \frac{900}{(2.24)^{0.286}} = \frac{900}{1.26} = 715\text{K}.$$

$$\eta_{t1} = 0.9 = \frac{T_8 - T_9}{T_8 - T_9'} = \frac{900 - T_9}{900 - 715}$$

$$\therefore \quad T_9 = 900 - 0.9 + 185 = 900 - 166.5 = \mathbf{733.5K.}$$

Subtituting the values in the equation (*c*),

$$(2\, m_{c1} + 306)\,(378 - 300) \times 0.92 = 306\,(900 - 733.5)$$

$$2m_{c1} = \frac{306 \times 166.5}{78 \times 0.92} - 306 = 404$$

$$\therefore \quad m_{c1} = \mathbf{202\ kg/sec.}$$

∴ Air taken from atmosphere = **202 kg/sec.**

Considering the combustion in combustion chamber

$$m_f \times \text{C.V.} \times \eta_{com} = (m_{c2} + m_f)\, C_p\,(T_5 - T_4)$$

where m_f is the mass of fuel used per second.

$$\therefore \quad m_f \times 40{,}000 \times 0.95 = [(202 + 306) + m_f] \times 1\,(2120 - 378) \text{ as } m_{c2} = m_{c1} + m_1$$

$$\therefore \quad 38000\, m_f = 1742\, m_f + 884936$$

$$\therefore \quad m_f = \mathbf{22.25\ kg/sec.}$$

Overall efficiency of the plant,

$$= \frac{\text{Net output at generator terminals}}{\text{Energy supplied}} = \frac{200 \times 1000}{22.25 \times 40{,}000} = 0.225 = \mathbf{22.5\%}$$

Compressor-turbine capacity $= m_1\, C_p\,(T_8 - T_9)$

$= 306 \times 1\,(900 - 733.5) = 50950$ kW = **50.95 MW.**

Problem 24.19. *A gas turbine plant is designed to develop 5 MW power. The inlet pressure and temperature of the air to the compressor are 1 bar and 30°C. The pressure ratio of the cycle is 5.*

A reheater is used between two turbines at a pressure of 2.24 bar. Calculate the overall efficiency of the cycle and mass flow rate assuming the following data :

Isentropic η of the compressor = 80%

Isentropic η of both turbines = 85%.

$$C_{pa} = 1 \text{ kJ/kg-k},\ C_{pg} = 1.15 \text{ kJ/kg-k},\ \gamma\,(air) = 1.4 \text{ and } \gamma\,(gases) = 1.33.$$

Neglect the mass of the fuel. (*P.U. Winter 89*)

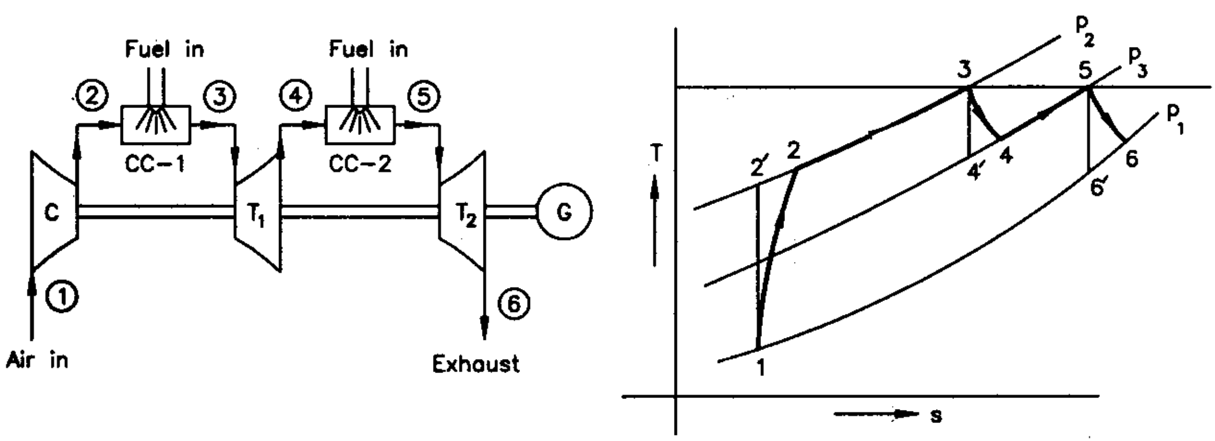
Solution. The arrangements of the components are shown in Fig. Prob. 24.19 (*a*) and processes are shown in Fig. Prob. 24.19 (*b*).

The given data is

$$T_1 = 30 + 273 = 303 \text{ K}, p_1 = 1 \text{ bar}, p_2 = 5 \text{ bar}, p_3 = 2.24 \text{ bar}$$
$$T_3 = T_5 = 550 + 273 = \mathbf{823 \text{ K}.}$$

Applying isentropic law to the process 1-2′

$$T_2' = T_1\left(\frac{p_2}{p_1}\right)^{\frac{r-1}{r}} = (5)^{\frac{0.4}{1.4}} = 303 \times (5)^{0.286} = \mathbf{480 \text{ K}.}$$



(*a*) (*b*)

Fig. Prob. 24.19.

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_c} = 303 + \frac{480 - 303}{0.8} = 303 + 221 = \mathbf{524 \text{ K}}$$

$$W_c = C_{pa}(T_2 - T_1) = 1 \times (524 - 303) = \mathbf{221 \text{ kJ/kg}}$$

Applying isentropic law to the process 3-4′.

$$\frac{T_3}{T_4'} = \left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore T_4' = \frac{T_3}{\left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{823}{\left(\frac{5}{2.24}\right)^{\frac{0.33}{1.33}}} = \mathbf{674 \text{ K}.}$$

$$\eta_{t1} = \frac{T_3 - T_4}{T_3 - T_4'} \quad \therefore T_4 = T_3 - \eta_{t1}(T_3 - T_4') = 823 - 0.85\,(823 - 674) = \mathbf{696 \text{ K}}$$

Applying isentropic law to the process 5-6′.

$$\frac{T_5}{T_6'} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \therefore \quad T_6' = \frac{823}{(2.24)^{\frac{.33}{1.33}}} = \frac{823}{1.22} = \mathbf{674.6 \text{ K}.}$$

$$\eta_{t2} = \frac{T_5 - T_6}{T_5 - T_6'} \quad \therefore T_6 = T_5 - \eta_{t2}(T_5 - T_6') = 823 - 0.85\,(823 - 674) = \mathbf{696 \text{ K}}$$

$\therefore$ W_t (both turbine works) $= 2\ C_{pg}\ (T_3 - T_4)$ as $T_3 = T_5$ and $T_4 = T_6$

$= 2 \times 1.15\ (823 - 674) = \mathbf{344.2\ kJ/kg.}$

$W_n = W_t - W_c = 344.2 - 221 = \mathbf{123.2\ kJ/kg}$

But power developed $= m_a \times W_n = 5000$ where m_a is (kg/sec)

$$m_a = \frac{5000}{123.2} = \mathbf{40.6\ kg/sec}$$

$$\text{Overall } \eta = \frac{W_n}{Q_s}$$

where

$$Q_s = C_{pq}\ (T_3 - T_2) + C_{pg}\ (T_5 - T_4)$$
$$= 1.15\ (823 - 524) + 1.15\ (823 - 696)$$
$$= 343.85 + 146.05 = 490 \text{ kJ/kg}$$

$$\therefore \quad \text{Overall } \eta = \frac{123.2}{490} = 0.25 = \mathbf{25\%}.$$

Problem 24.20. *A gas turbine power plant of 5 MW capacity is supplied with air at 15°C. The pressure ratio is 6. The maximum temperature is limited to 750°C. The compression is carried out in one stage where expansion is carried out in two stages with reheating to the original temperature. The suction and exhaust pressures are 1 bar.*

Taking the following data

$C_{pa} = 1$ *kJ/kg-K,* $C_{pg} = 1.15$ *kJ/kg-K*

γ *(air)* $= 1.4$, γ *(gas)* $= 1.33$

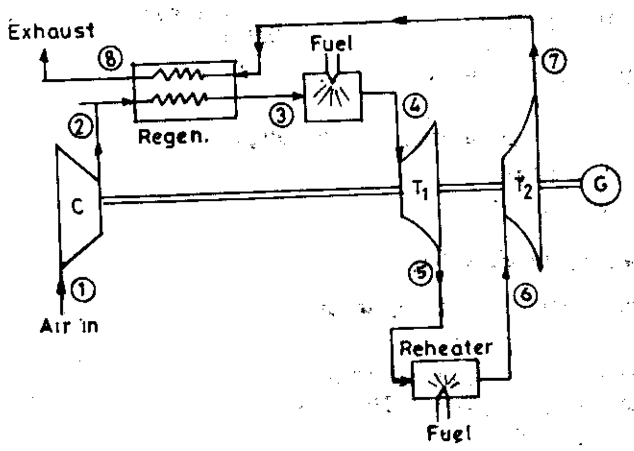
$\eta_c = 80\%$, $\eta_{t1} = \eta_{t2} = 85\%$, *C.V. of fuel* $= 18500$ *kJ/kg.*

ε *(effectiveness of heat exhanger)* $= 0.75$

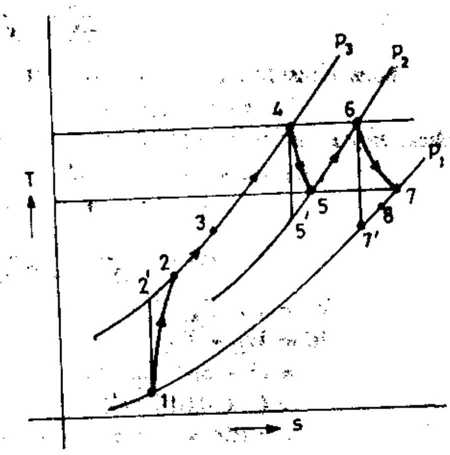
determine the following :

(i) Cycle efficiency, (ii) Air-supplied to the plant.
(iii) A : F ratio entering in the first turbine.
(iv) Fuel consumption of the plant per hour.

Solution. The arrangement of the components and processes on *T-S* diagram are shown in Fig. Prob. 24.20 (*a*) and Fig. Prob. 24.20 (*b*).



(*a*)



(*a*)

Fig. Prob. 24.20.

The given data is

$T_1 = 15 + 273 = 288$ **K**, R_p (pressure ratio of each turbine) $= \sqrt{6 \times 1} = 2.45$

Applying isentropic law to the process 1-2′

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 288\,(6)^{0.286} = 288 \times 1.67 = \mathbf{480.8\ K}$$

$$\eta_c = \frac{T_2'-T_1}{T_2-T_1} \quad \therefore\ T_2 = T_1 + \frac{T_2'-T_1}{\eta_c} = 288 + \frac{(480.8-288)}{0.8} = \mathbf{529\ K}$$

$$T_4 = 750 + 273 = 1023 \text{ K (given)} = T_6$$

Applying isentropic law to the process 4-5′

$$\frac{T_4}{T_5'} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = (2.45)^{\frac{0.33}{1.33}} = (2.45)^{0.248} = 1.25$$

$$\therefore \quad T_5' = \frac{T_4}{1.25} = \frac{1023}{1.25} = \mathbf{818.5\ K}$$

$$\eta_{t1} = \frac{T_4-T_5}{T_4-T_5'} \quad \therefore\ T_5 = T_4 - \eta_{t1}\,(T_4 - T_5') = 1023 - 0.85\,(1023 - 818.5) = \mathbf{849\ K}$$

$$\eta_{t1} = \eta_{t2} \text{ (given)} \quad \therefore\ T_7 = T_5 = \mathbf{849\ K.}$$

The effectiveness of the H.E. is given by

$$\varepsilon = \frac{T_3-T_2}{T_7-T_2} = 0.75$$

$$\therefore \quad T_3 = 529 + 0.75\,(849 - 529) = 529 + 240 = \mathbf{769\ K.}$$

(i) Now considering the flow with burning through combustion chamber

$$(1 + m_{f1})\, C_{pg}\,(T_4 - T_3) = m_{f1}\ .\ \text{C.V.}$$

where m_{f1} is the mass of fuel supplied per kg of air.

$$\therefore \quad (1 + m_{f1}) \times 1.15\,(1023 - 769) = m_{f1}\ .\ 18500$$

$$(1 + m_{f1})\, 292 = 18500\, m_{f1}.$$

$$\therefore \quad m_{f1} = \frac{292}{18500 - 292} = 0.016/\text{kg of air} \qquad \therefore\ A : F = \frac{1}{0.016} = \mathbf{62.5}$$

Now considering the flow with burning through the reheater

$$(1 + m_{f1} + m_{f2})\, C_{pg}\,(T_6 - T_5) = m_{f2}\ .\ \text{C.V.}$$

where m_{f2} is the fuel supplied in the reheater per kg of air entering into the compressor

$$\therefore \quad (1.016 + m_{f2}) \times 1.15\,(1023 - 849) = m_{f2} \times 18500$$

$$(1.016 + m_{f2}) \times 200 = 18500\, m_{f2}$$

$$\therefore \quad m_{f2} = \frac{101.6}{18500 - 200} = \mathbf{0.0056\ kg/kg\ of\ air}$$

$$W_c = 1 \times C_{pa}\,(T_2 - T_1) = 1 \times 1\,(529 - 288) = \mathbf{241\ kJ/kg\ of\ air}$$

$$W_t = W_{t1} + W_{t2}$$

$$= (1 + m_{f1})\, C_{pg}\,(T_4 - T_5) + (1 + m_{f1} + m_{f2})\, C_{pg}\,(T_6 - T_7)$$

$$= (1 + 0.016) \times 1.15\,(1023 - 849) + (1 + 0.016 + 0.056) \times 1.15\,(1023 - 849)$$

$$= 1.016 \times 1.15 \times 174 + 1.0216 \times 1.15 \times 174$$

$$= 1.15 \times 174\,(1.016 + 1.0216) = \mathbf{416.5\ kJ/kg\ of\ air}$$

$$W_n = W_t - W_c = 416.5 - 241 = \mathbf{175.5\ kJ/kg\ of\ air}$$

(*ii*) Thermal efficiency of the cycle is given by

$$\eta_{th} = \frac{W_n}{Q_s} = \frac{W_n}{(m_{f1} + m_{f2})\ .\ CV} = \frac{175.5}{(0.016 + 0.0556) \times 18500} = \frac{175.5}{399.6} = 0.438 = \mathbf{43.8\%}$$

(*iii*) Air supplied to the plant can be calculated as

$$m_a\, W_n = 5 \times 1000 \quad \therefore \quad m_a = \frac{5000}{175.5} = \mathbf{28.5\ kg/sec}$$

(*iv*) Fuel required per hour to run the plant

$$= m_a\,(m_{f1} + m_{f2}) \times 3600$$
$$= 28.5\,(0.016 + 0.0056) \times 3600 = \mathbf{2216\ kg/hr.}$$

Problem 24.21. *A gas turbine power plant is operated between 1 bar and 9 bar pressures and minimum and maximum cycle temperatures are 25°C and 1250°C. A compression is carried out in two stages with perfect intercooling. The gases coming out from H.P. turbine are heated to 1250°C before entering into L.P. turbine. The expansions in both turbines are arranged in such a way that each stage develops same power. Assuming compressors and turbines isentropic efficiencies as 83%, (i) determine the cycle efficiency assuming ideal regenerator. Neglect the mass of fuel.*

(*ii*) *Find the power developed by the cycle in kW if the air flow through the power plant is 16.5 kg/sec.*

(*P.U., Winter 1987*)

Solution. The arrangement of the components and the processes is shown in Fig. Prob. 24.21 (*a*) and Fig. Prob. 24.21 (*b*).

The given data is

$T_1 = 25 + 273 = 298$ K $= T_3$ (as it is perfect intercooling), $p_1 = 1$ bar and $p_3 = 9$ bar

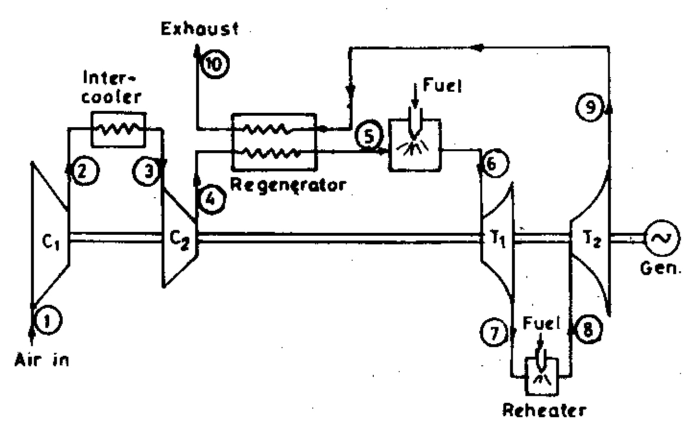
$p_2 = \sqrt{p_1 p_3} = \sqrt{1 \times 9} = 3$ bar $\quad \therefore \quad R_{p1} = R_{p2} = 3$

$\eta_{c1} = \eta_{c2} = \eta_{t1} = \eta_{t2} = 0.83$, $T_6 = T_8 = 1250 + 273 = 1523$ K.

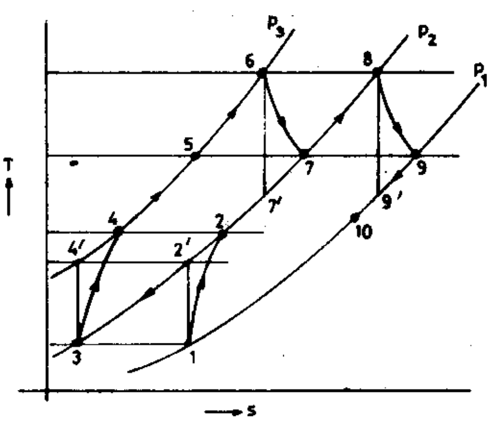
$T_{10} = T_5$ (as perfect regenerator is given)

Applying isentropic law to the process 1-2′

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} = 298(3)^{0.286} = \mathbf{408\ K}$$



(*a*)



(*b*)

Fig. Prob. 24.21.

$$\eta_{c1} = \frac{T_2' - T_1}{T_2 - T_1}$$

$\therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_{c1}} = 298 + \frac{(408 - 298)}{0.83} = \mathbf{430.5\ K}$

$T_4 = T_2 = 430.5$ K.

Applying isentropic law to the process 6-7′

$$\frac{T_6}{T_7'} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = (3)^{0.286} = \mathbf{1.37\ K}$$

$\therefore \quad T_7' = \frac{1523}{1.37} = \mathbf{1111\ K}$

$$\eta_{t1} = \frac{T_6 - T_7}{T_6 - T_7'}$$

$\therefore \quad T_7 = T_6 - \eta_{t1}\,(T_6 - T_7')$

$= 1523 - 0.83\,(1523 - 1111) = \mathbf{1181\ K}$

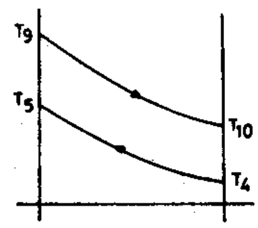


Fig. Prob. 24.21 (c).

$T_9 = T_7 = \mathbf{1181\ K}$ (as equal work is developed by each turbine)

$W_c = 2\,C_{pa}\,(T_2 - T_1) = 2 \times 1\,(430.5 - 298) = \mathbf{266\ kJ/kg}$

$W_t = 2\,C_{pa}\,(t_6 - T_7) = 2 \times 1\,(1523 - 1181) = \mathbf{687.5\ kJ/kg}$

$W_n = W_t - W_c = 687.5 - 266 = \mathbf{421.5\ kJ/kg}$ ...(*a*)

When the ideal regeneration is given, then

$\varepsilon = 1$ therefore $T_5 = T_9 = \mathbf{1181\ K} = T_7$

$\therefore \quad Q_s$ (heat supplied) $= 2\,C_{pa}\,(T_6 - T_5) = 2 \times 1\,(1523 - 1181) = \mathbf{684\ kJ/kg}$

(*i*) Thermal $\eta = \frac{W_n}{Q_s} = \frac{421.5}{684} = 0.615 = \mathbf{61.5\%}$

(*ii*) Power developed by the plant

$= W_n \times m = 421.5 \times 16.5 = \mathbf{6954.75\ kW}$

Problem 24.22. *An individual small gas turbine power plant is developed to supply power to a factory. The temperature and pressure of the air at the inlet are 290 K and 1.01 bar. The compressor is driven by H.P. turbine and L.P. turbine is coupled to the Generator directly. The maximum temperature of the cycle is limited to 650°C. Taking the following data*

Pressure Ratio of compressor = 8 : 1

Isentropic efficiency of the compressor = 80%

Isentropic efficiency of the H.P. turbine = 85%

Isentropic efficiency of the L.P. turbine = 83%

For compression process

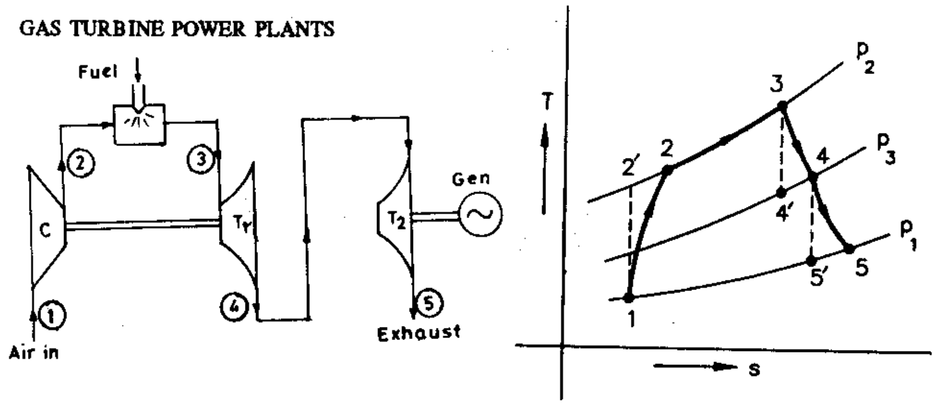
$\gamma = 1.4$ *and* C_{pa} *(air) = 1 kJ/kg-K*

For expansion processes

$\gamma = 1.33$ *and* C_{pg} *(gases) = 1.15 kJ/kg-K.*

find out the power developed by the unit in kW if the air flow through the compressor is 10 kg/sec and condition of the gas entering the power turbine. Also find the thermal efficiency and work ratio of the plant. Neglect the fuel masses.

Solution. The arrangement of the components and corresponding processes on *T-s* diagram is shown in Fig. Prob. 24.22 (*a*) and Fig. Prob. 24.22 (*b*).



(*a*) (*b*)

Fig. Prob. 24.22. (*a*)

The given data is

$T_1 = 290$ K and $T_3 = 650 + 273 = 923$ K

$$\frac{T_2'}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (8)^{\frac{1.4-1}{1.4}} = (8)^{0.285}$$

$$\therefore \quad T_2' = 290 \times (8)^{0.285} = \mathbf{525\ K}$$

$$\eta_c = \frac{T_2' - T}{T_2 - T_1} \quad \therefore \quad 0.8 = \frac{5.25 - 290}{T_2 - 290} \quad \therefore \quad T_2 = \mathbf{584\ K}$$

Work input to the compressor per kg of air

$= 1 \times C_{pa} (T_2 - T_1) = 1 \times (584 - 290) = \mathbf{295.5\ kJ/kg}$

The power to be developed by H.P. turbine must be equal to the power input to the compressor

$$W_{t1} = C_{pg} (T_5 - T_4) = 295.5 \quad \therefore \quad 1.15 (923 - T_4) = 295.5$$

$$\therefore \quad T_4 = 923 - \frac{295.5}{1.15} = 923 - 257 = \mathbf{666\ K}$$

$$\eta_{t1} = 0.85 = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$T_4' = T_3 - \frac{(T_3 - T_4)}{0.85} = 923 - 302 = \mathbf{621\ K}$$

Now, for process 3-4′, we can write

$$\frac{T_3}{T_4'} = \left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\therefore \quad \frac{p_2}{p_3} = \left(\frac{T_3}{T_4'}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{923}{621}\right)^{\frac{1.33}{0.33}} = 4.9 \qquad ...(a)$$

$$\therefore \quad p_3 = \frac{p_2}{4.9} = \frac{8 \times 1.01}{4.9} = \mathbf{1.65\ bar}$$

(*i*) The condition of the gas entering into the power turbine or L.P. turbine is

$p_3 = 1.65$ bar and $T_4 = 666 - 273 = \mathbf{393°C}$.

The pressure ratio of expansion in the power turbine is

$$= \frac{p_3}{p_1} = \frac{p_3}{p_2} \cdot \frac{p_2}{p_1}$$

$$\frac{p_2}{p_1} = 8 \text{ (given) and } \frac{p_3}{p_2} = \frac{1}{4.9} \text{ [as per equation (a)]}$$

$$\therefore \quad \frac{p_3}{p_1} = \frac{8}{4.9} = 1.63$$

Applying isentropic law to the process 4-5′.

$$\frac{T_4}{T_5'} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (1.63)^{\frac{0.33}{1.33}} = \mathbf{1.13}$$

$$T_5' = \frac{T_4}{1.13} = \frac{666}{1.13} = \mathbf{588\ K}$$

$$\eta_{t2} = \frac{T_4 - T_5}{T_4 - T_5'} = 0.83$$

$\therefore$ $T_4 - T_5 = 0.83\,(T_4 - T_5') = 0.83\,(666 - 588) = \mathbf{64.8\ K}$.

W_{t2} (work developed by power turbine)

$= C_{pg}\,(T_4 - T_5)$ kJ/kg

$= 1.15 \times 64.8 =$ **74.5 kJ/kg.**

$\therefore$ The net work done (W_{t2}) per kg of air

$= 74.5$ kJ/kg.

Total work done per kg of air

$W_t = W_{t1} + W_{t2} = 295.5 + 74.5 =$ **370 kJ/kg.**

$$\therefore \quad \text{Work Ratio} = \frac{W_{t2}}{W_t} = \frac{74.5}{370} = \mathbf{0.2}$$

$$\text{Thermal efficiency} = \frac{W_{t2}}{Q_s} = \frac{74.5}{C_{pa}\,(T_3 - T_2)} = \frac{74.5}{1 \times (923 - 584)} = 0.22 = \mathbf{22\%}$$

Power capacity of the plant $= W_{t2} \times m_a = 74.5 \times 10 = 745$ kJ/sec = **745 kW**

Note : *Students are advised to solve the problem without neglecting fuel mass and also find A : F ratio supplied.*

Take C.V. of fuel = 20,000 kJ/kg.

SOLVED PROBLEMS FROM UNIVERSITY QUESTION PAPERS

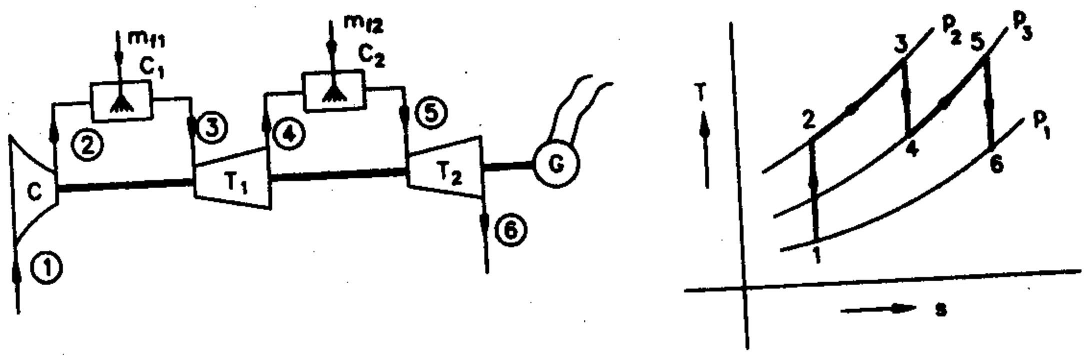
Problem 24.23. *In a open cycle gas-turbine power plant, the maximum pressure and temperature are limited to 5 bar and 900 K. The pressure and temperature of the gas entering the compressor are 1 bar and 300 K. Reheating is used at 2.5 bar where the temperature of the gases is increased to its original turbine inlet temperature. The air flow through the plant is 10 kg/sec. Determine the thermal efficiency of the plant and power generating capacity.*

Assume compression and expansion to be isentropic. Neglect pressure losses but don't neglect fuel mass. All the components are mounted on one shaft only.

Take $\gamma = 1.4$ and $C_p = 1$ kJ/kg-K for both air and gas.

C.V. of fuel = 33500 kJ/kg and exhaust is at 1 bar. *(P.U. Summer 1995)*

Solution. The arrangement of the components is shown in Fig. Prob. 24.23 (*a*) and processes are represented on *T-s* diagram as shown in Fig. Prob. 24.23 (*b*).



(*a*) (*b*)

Fig. Prob. 24.23.

The net power developed in the system is given by

$$W_n = W_{t1} + W_{t2} - W_c \qquad ...(a)$$

where W_{t1} and W_{t2} are the work developed in turbines T_1 and T_2 and W_c is the work absorbed by the compressor C

$$p_1 = 1 \text{ bar}, p_2 = 5 \text{ bar}, p_3 = 2.5 \text{ bar}, T_1 = 300 \text{ K}, T_3 = T_5 = 900 \text{ K}$$

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 300\,(5)^{0.286} = \mathbf{475.4\ K}$$

$$T_4 = \frac{T_3}{\left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{900}{\left(\frac{5}{2.5}\right)^{0.286}} = \frac{900}{(2.5)^{0.286}} = \frac{9002}{1.3} = \mathbf{692.3\ K}$$

$$T_6 = \frac{T_5}{\left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}}} = \frac{900}{\left(\frac{2.5}{1}\right)^{0.286}} = \mathbf{692.3\ K}$$

The masses of fuel supplied per kg of air in combustion chambers C_1 and C_2 are m_{f1} and m_{f2}

$\therefore \qquad m_{f1} \times \text{C.V.} = (1 + m_{f1})\, C_p\, (T_3 - T_2)$

$m_{f1} \times 33500 = (1 + m_{f1}) \times 1\, (900 - 475.4) = 424.6\, (1 + m_{f1})$

$$\therefore \qquad m_{f1} = \frac{426.6}{33500 - 426.6} = \mathbf{0.0129\ kg/kg\ of\ air}$$

Similarly, considering the combustion in C_2

$m_{f2} \times \text{C.V.} = (1 + m_{f1} + m_{f2})\,.\, C_p\, (T_5 - T_4)$

$\therefore \qquad m_{f2} \times 33500 = (1.0129 + m_{f2})\,(900 - 692.3)$

$= 210.4 + 207.7\, m_{f2}$

$$\therefore \quad m_{f2} = \frac{210.4}{33500 - 207.7} = 0.00631 \text{ kg/kg of air}$$

Now substituting the value in equation (*a*)

$$W_n = 10\ (1 + m_{f1})\ C_p\ (T_3 - T_4) + 10\ (1 + m_{f1} + m_{f2})\ C_p\ (T_5 - T_6) - 10\ .\ C_p\ (T_2 - T_1)$$
$$= 10\ [(1 + 0129) \times 1\ (900 - 692.3) + (1 + 0.0129 + 0.00631) \times 1\ (900 - 692.3)$$
$$- 1\ (1475.4 - 300)]$$
$$= 10\ [210.4 + 211.7 - 175.4] = 10 \times 246.7 = \mathbf{2467\ kW}$$

The generator efficiency is considered 100%.

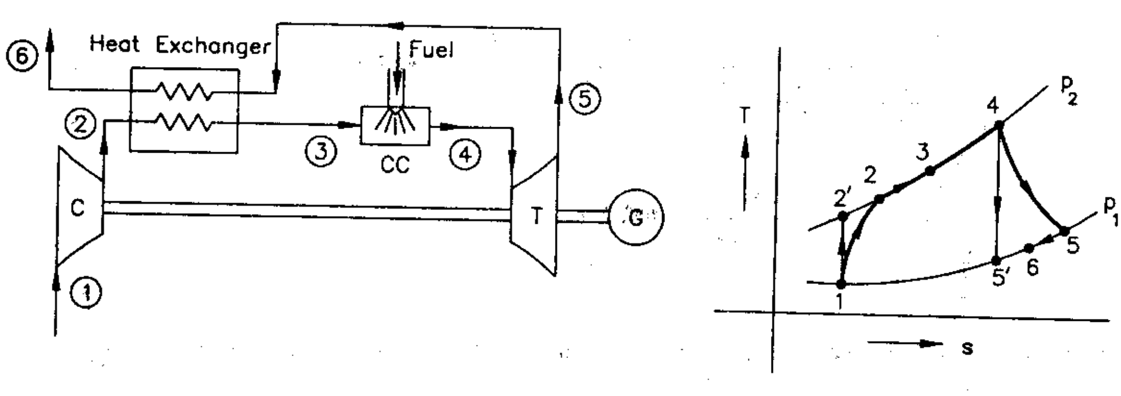
The efficiency of the plant is given by

$$\eta = \frac{\text{Output}}{\text{Input}} = \frac{2467}{10\ (m_{f1} + m_{f2}) \times \text{C.V.}} = \frac{2467}{10 \times 0.0192 \times 33500} = 0.384 = \mathbf{38.4\%}$$

Problem 24.24. *An open cycle constant pressure gas turbine plant takes air at 1 bar and 300 K and compresses through a compression ratio of 6 : 1 in a centrifugal compressor. The air is then passed through an heat exchanger whose effectiveness is 0.65. Then the air is passed through a combustion chamber and heated to 870°C. Then the gases are passed through a turbine where it expands to 1 bar and then passes through an heat exchanger and finally to exhaust. Assuming isentropic* η *of compressor = 0.8 and turbine = 0.85, determine (a) Power output for the plant assuming generator* η *= 0.95 (b) Thermal* η *of the plant and (c) Heat carried away by the gases per minute. Neglect fuel mass and take air mass flow = 5 kg/sec. All components are mounted on a single shaft.*

Take $\gamma = 1.4$ *and* $C_p = 1$ *kJ/kg-K for air.* *(P.U. Winter 1997)*

Solution. The arrangements of the components are is shown in Fig. Prob. 24.24 (*a*) and the processes are represented on *T-s* chart as shown in Fig. Prob. 24.24 (*b*).



(*a*) (*b*)

Fig. Prob. 24.24.

(*a*) The neat power output of the system is given by

$$W_n = (W_t - W_c)\ \eta_g$$
$$= m_a\ [C_p\ (T_4 - T_5) - C_p\ (T_2 - T_1)]\ .\ \eta_g$$
$$= m_a\ C_p\ [(T_4 - T_5) - (T_2 - T_1)]\ .\ \eta_g \qquad ...(a)$$

$$T_2' = T_1\ (R_p)^{\frac{\gamma - 1}{\gamma}} = 300\ (6)^{0.286} = 300 \times 1.67 = \mathbf{500.8\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$\therefore$ $$T_2 = T_1 + \frac{T_2' - T_1}{\eta_c} = 300 + \frac{500.8 - 300}{0.8} = 300 + 251 = \mathbf{551\ K}$$

Similarly $$T_5' = \frac{T_4}{(R_p)^{(\gamma-1)/\gamma}} = \frac{870 + 273}{(6)^{0.286}} = \frac{1143}{1.67} = \mathbf{684.4\ K}$$

$$\eta_t = \frac{T_4 - T_5}{T_4 - T_5'}$$

$$T_5 = T_4 - \eta_t (T_4 - T_5') = 1143 - 0.85(1143 - 684.4) = 1143 - 390 = \mathbf{753\ K}$$

Now substituting the values in equation (a)

$$W_n = 5 \times 1[(11813 - 753) - (551 - 300)] \times 0.95$$
$$= 5(390 - 251) \times 0.95 = \mathbf{660.25\ kW}$$

(b) The effectiveness of the heat exchanger is given by

$$\epsilon = \frac{T_3 - T_2}{T_5 - T_2}$$

$\therefore$ $$0.65 = \frac{T_3 - 551}{753 - 551}$$

$\therefore$ $$T_3 = 551 + 0.65(753 - 551) = 551 + 131.3 = \mathbf{682.3\ K}$$

$$\eta_t = \frac{\text{Output per kg of air}}{\text{Heat supplied per kg of air}}$$
$$= \frac{C_p\,[(T_4 - T_5) - (T_2 - T_1)]}{C_p\,(T_4 - T_3)} = \frac{(1143 - 753) - (551 - 300)}{(1143 - 682.3)}$$
$$= \frac{390 - 251}{460.7} = \frac{139}{460.7} = 0.302 = \mathbf{30.2\%}$$

For Heat Exchanger

Heat lost by hot gases = Heat gained by the air

$$(T_5 - T_6) = (T_3 - T_2)$$
$$753 - T_6 = 682.3 - 551$$

$\therefore$ $$T_6 = \mathbf{621.7\ K}$$

(c) Heat carried away by the exhaust gases

$$Q = (m_a \times 60) \times C_p\,(T_6 - T_1)$$
$$= 5 \times 60 \times 1(621.7 - 300) = \mathbf{96510\ kJ/min.}$$

Problem 24.25. *A open cycle gas-turbine plant operates with a pressure ratio of 4.5 while using 82 kg/min of air and 1.4 kg/min of fuel. The net-output of the plant is 200 kW when 230 kW is needed to drive the compressor. Air enters the compressor at 1 bar and 15°C and combustion gases enter the turbine at 765°C. Taking C_{pa} = 1 kJ/kg K and C_{pg} = 1.13 kJ/kg-K, the index of compression = 1.4 and index of expansion = 1.34 and mechanical efficiency for compressor as well as generator = 0.98, determine, isentropic η of compressor (b) isentropic efficiency of the turbine and (c) overall efficiency of the plant.*

(B.U. Summer 1998)

Solution. The arrangement of the components is shown in Fig. Prob. 24.25 (a) and processes are represented on T-s chart as shown in Fig. Prob. 24.25 (b).

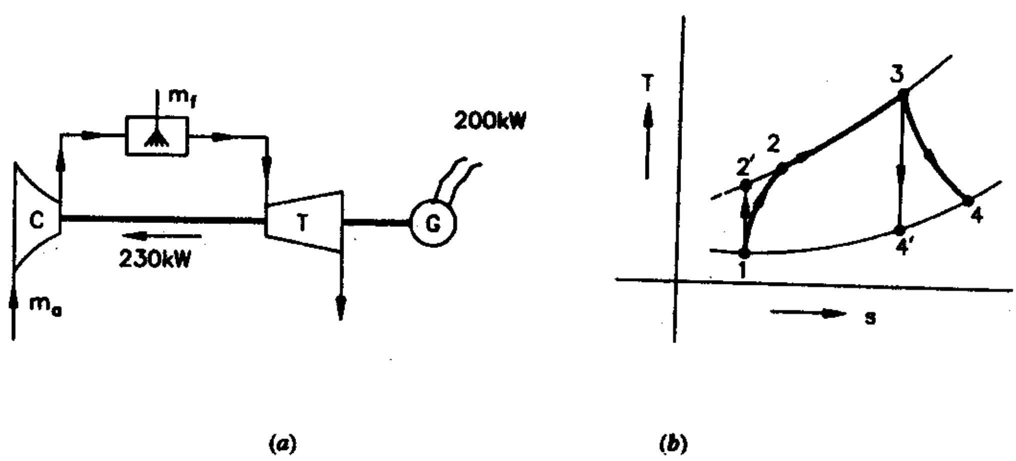


Fig. Prob. 24.25.

The total power developed by the turbine

$$W_t = \frac{230 + 200}{\eta_m} = \frac{430}{0.98} = \mathbf{438.8\ kW}$$

$$m_a \text{ (air flow/sec)} = \frac{82}{60} = \mathbf{1.37\ kg/sec}$$

$$m_f \text{ (fuel flow/sec)} = \frac{1.4}{60} = \mathbf{0.0233\ kg/sec}$$

$$A : F = \frac{1.37}{0.0233} = \mathbf{58.8}$$

(a) $$W_c = m_a C_{pa} (T_2 - T_1)$$

But $$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$\therefore$ $$W_c = m_a C_{pa} \left(\frac{T_2' - T_1}{\eta_c} \right) \quad ...(a)$$

$$T_2' = T_1 (R_p)^{\frac{1.4-1}{1.4}} = 288\ (4.5)^{0.286} = 288 \times 1.54 = \mathbf{443\ K}$$

Substituting the values in equation (a)

$$230 = 1.37 \times 1 \times \left(\frac{443 - 288}{\eta_c} \right)$$

$\therefore$ $$\eta_c = \frac{1.37 \times 155}{230} = 0.92 = \mathbf{92\%}$$

(b) $$W_t = (m_a + m_f)\ C_{pg}\ (T_3 - T_4)$$

But $$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'}$$

$\therefore$ $$W_t = (m_a + m_f) \cdot C_{pg}\ [\eta_t\ (T_3 - T_4')] \quad ...(b)$$

$$T_4' = \frac{T_3}{(R_p)^{\frac{1.34-1}{1.34}}} = \frac{(765 + 273)}{(4.5)^{0.254}} = \frac{1038}{1.465} = \mathbf{708.5\ K}$$

Substituting the values in equation (*b*)

$$438.8 = (1.37 + 0.0233) \times 1.13\ [\eta_t\ (1038 - 708.5)]$$

$$\eta_t = \frac{438.8}{1.3933 \times 1.3 \times 329.5} = 0.735 = \mathbf{73.5\%}$$

(*c*) Overall η_0 of the plant

$$= \frac{\text{Net output}}{\text{Input}} = \frac{200}{(m_a + m_f)\ C_{pg}\ (T_3 - T_2)}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1} \quad \therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_c} = 288 + \frac{443 - 288}{0.92} = \mathbf{456.5}$$

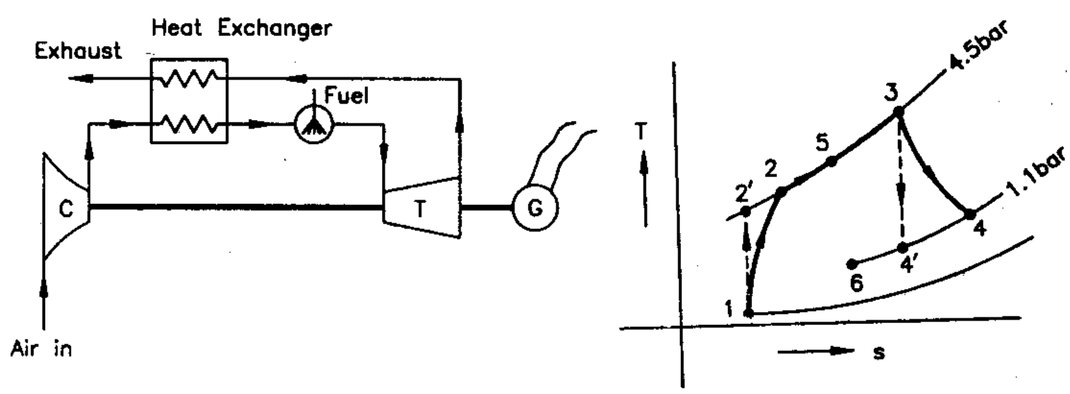
$$\eta_0 = \frac{200}{(1.37 + 0.0233) \times 1.13(1038 - 456.5)}$$

$$= \frac{200}{1.3933 \times 1.33 \times 581.5} = 0.185 = \mathbf{18.5\%}$$

Problem 24.26. *Air at 0.9 bar and 303 K is admitted to the compressor and compressed to 4.5 bar with an isentropic efficiency of 85%. Its temperature is raised further using exhaust gases passing through an heat exchanger. The maximum temperature of the cycle is limited to 1000°C. The gas is then expanded to 1.1 bar with an isentropic efficiency of 80%. Find the thermal efficiency of the system assuming effectiveness of heat exchanger as 0.8. Neglect the fuel mass and pressure losses in the system. If the air flow is 5 kg/sec, find the power developing capacity of the system.*

Solution. The properties of air and gas may be taken as same.

The components of the system and processes are shown in Fig. Prob. 24.26 (*a*) and Fig. Prob. 24.26 (*b*).



(*a*) (*b*)

Fig. Prob. 24.26.

$$T_2 = T_1 \left(\frac{4.5}{0.9}\right)^{\frac{\gamma - 1}{\gamma}} = 303\ (1.582) = \mathbf{479\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad 0.85 = \frac{479 - 303}{T_2 - 303} \quad \therefore \quad T_2 = \mathbf{510\ K}$$

$$\frac{T_3}{T_4'} = \left(\frac{4.5}{1.1}\right)^{0.286} = 1.5$$

$$\therefore \quad T_4' = \frac{1273}{1.5} = \mathbf{849\ K}$$

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_4'} \quad \therefore \quad 0.8 = \frac{1273 - T_4}{1273 - 849} \quad \therefore \quad T_4 = \mathbf{934\ K}$$

The effectiveness of heat exchanger is given by

$$\epsilon = \frac{T_5 - T_2}{T_4 - T_2}$$

$$\therefore \quad 0.8 = \frac{T_5 - 510}{934 - 510} \quad \therefore \quad T_5 = \mathbf{849\ K}$$

The thermal efficiency of the system is given by

$$\eta_{th} = \frac{(T_3 - T_4) - (T_2 - T_1)}{(T_3 - T_5)}$$

$$= \frac{(1273 - 934) - (510 - 303)}{1273 - 849}$$

$$= \frac{132}{424} = 0.31 = \mathbf{31\%}$$

The power developed by the system

$$= mC_p\ [(T_3 - T_4) - (T_2 - T_1)]$$

$$= 5 \times 1.005\ [(1273 - 934) - (510 - 303)]$$

$$= \mathbf{663.3\ kW}$$

Problem 24.27. *A open cycle gas turbine plant consists of a compressor driven by high pressure turbine. A low pressure turbine produces power and exhaust gases from it go to a regenerator. Using the following data, determine (a) the air flow rate in kg/sec for 2040 kW to be developed and (b) the thermal efficiency of the plant.*

Compressor isentropic η *= 86%*

H.P. turbine isentropic η *= 85%*

L.P. turbine isentropic η *= 87%*

Intake temperature = 21°C

Temperature at inlet to H.P. turbine = 925°C

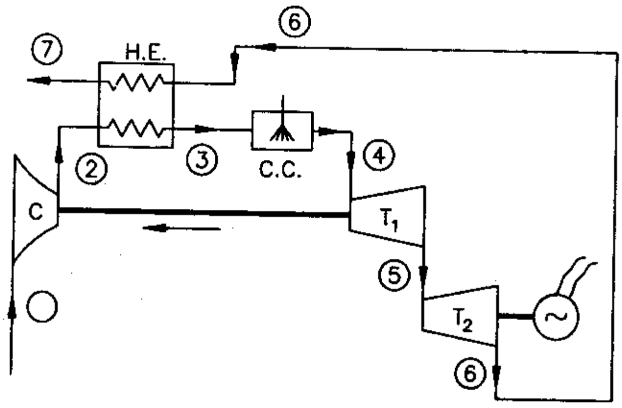
Heat exchanger efficiency = 0.75

Mechanical η *of the compressor and H.P. turbine assembly = 99%.*

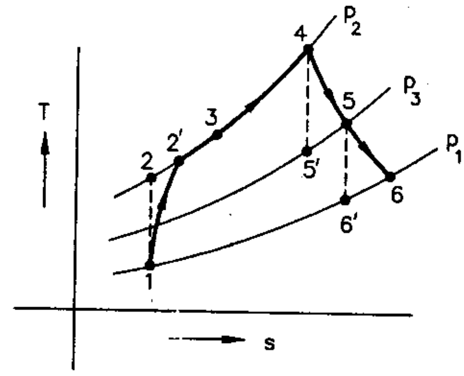
Combustion efficiency = 98%

Assume air flow rate, to be equal to gas flow rate. *(May 1998, B.U.)*

Solution. The component arrangement and processes are shown in Fig. Prob. 24.27 (*a*) and Fig. Prob. 24.27 (*b*).



(*a*)



(*b*)

Fig. Prob. 24.27.

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 294\ (7)^{0.286} = \mathbf{512.6\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad 0.086 = \frac{512 - 294}{T_2 - 294} \quad \therefore \quad T_2 = \mathbf{548.2\ K}$$

$$W_c \text{ (per kg of air)} = C_{pa}\,(T_2 - T_1)$$
$$= 1.005\ (548.2 - 294) = \mathbf{255.5\ kJ/kg}$$

$$\therefore \quad W_{t1} \times \eta_m = W_c$$

$$\therefore \quad W_{t1} = \frac{255.5}{0.99} = \mathbf{258\ kJ/kg}$$

$$W_{t1} = m_a C_{pa}\,(T_4 - T_5)$$

$$\therefore \quad 258 = 1 \times 1.005\ (1198 - T_5) \text{ as } m_g = m_a \text{ and } C_{pg} = C_{pa}$$

$$\therefore \quad T_5 = \mathbf{941\ K}$$

$$\eta_{t1} = \frac{T_4 - T_5}{T_4 - T_5'}$$

$$\therefore \quad 0.85 = \frac{1198 - 941}{1198 - T_5'} \quad \therefore \quad T_5' = \mathbf{896\ K}$$

$$\frac{T_4}{T_5'} = \left(\frac{p_2}{p_3}\right)^{\frac{\gamma-1}{\gamma}}$$

or
$$\frac{p_2}{p_3} = \left(\frac{T_4}{T_5'}\right)^{\frac{\gamma}{\gamma-1}} = \left(\frac{1198}{896}\right)^{3.5} = \mathbf{2.76}$$

Now
$$\frac{p_1}{p_3} = \frac{p_1}{p_2} \cdot \frac{p_2}{p_3} = \frac{1}{7} \times 2.76 = \mathbf{0.394}$$

$$\frac{T_5}{T_6'} = \left(\frac{p_3}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{941}{T_6'} = \left(\frac{1}{0.394}\right)^{0.286}$$

$\therefore$
$$T_6' = 941 \times (0.394)^{0.286} = \mathbf{721\ K}$$

$$\eta_{t2} = \frac{T_5 - T_6}{T_5 - T_6'}$$

$\therefore$
$$0.87 = \frac{941 - T_6}{941 - 721} \quad \therefore \quad T_6 = \mathbf{750.3\ K}$$

ϵ (Heat exchanger effectiveness)

$$= \frac{T_3 - T_2}{T_6 - T_2}$$

$\therefore$
$$0.75 = \frac{T_3 - 548.2}{750.3 - 548.2} \quad \therefore \quad T_3 = \mathbf{6.700\ K}$$

$$W_{t2} = m_a C_{pa} (T_5 - T_6)$$

$\therefore$
$$2940 = m_a \times 1.005\ (941 - 750.3)$$

$\therefore$
$$m_a = \frac{2940}{1.005 \times 190.7} = \mathbf{15.34\ kg/sec}$$

$$\eta_{th} = \frac{\text{Output}}{\text{Input}}$$

$$= \frac{2940}{[m_a C_{pa} (T_4 - T_3)]/\eta_{com}}$$

$$= \frac{2940}{\frac{15.34 \times 1.005(1198 - 700)}{0.98}} = 0.382 = \mathbf{38.2\%}$$

Problem 24.28. *A gas turbine plant consists of one turbine as compressor drive and other to drive a generator. Each turbine has its own combustion chamber supplied air directly from the compressor. Air enters the compressor at 1 bar and 15°C and compressed with isentropic efficiency of 76%. The gas inlet pressure and temperature in both the turbines are 5 bar and 680°C respectively. Take isentropic efficiency of both the turbines is 86%. The mass flow rate of air entering the compressor is 23 kg/sec. The calorific value of the fuel is 42000 kJ/kg. Calculate the power output of the plant and its thermal efficiency.*

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 98)*

Solution. The arrangement of the components is shown in Fig. Prob. 24.28 (*a*).

The given data is

$$m_a = m_{a1} + m_{a2} = 23$$

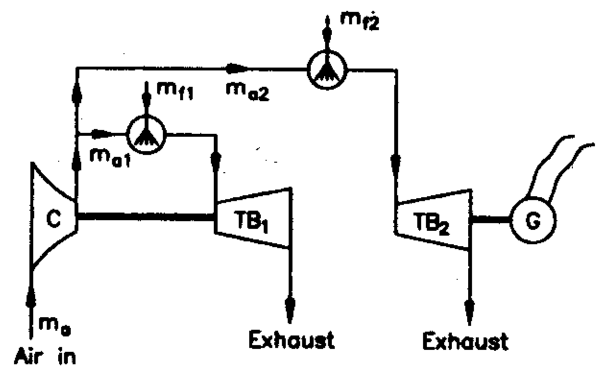


Fig. Prob. 24.28 (*a*).

The processes for compressor and turbines are shown in Fig. Prob. 24.28 (*b*) on *T-s* diagram.

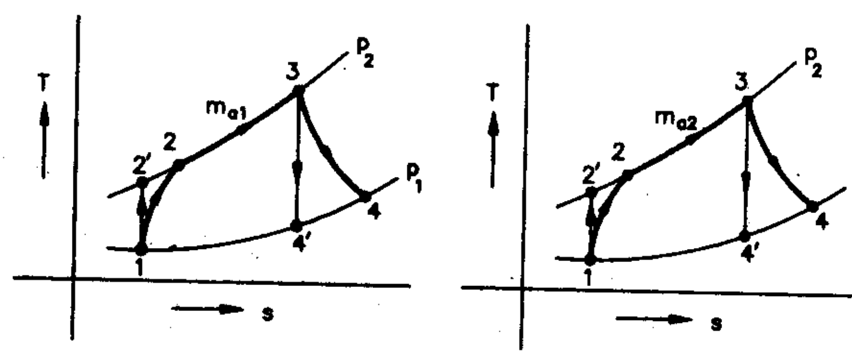


Fig. Prob. 24.28 (*b*).

The given data is

$$T_1 = 15 + 273 = 288 \text{ K}, p_1 = 1 \text{ bar}, p_2 = 5 \text{ bar}, T_3 = 680°\text{C} = 953 \text{ K}$$

First considering C – TB$_1$

$$T_2' = T_1 \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 288\ (5)^{0.286} = \mathbf{456.4\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 = T_1 + \frac{T_2' - T_1}{\eta_c} = 288 + \frac{456.4 - 238}{0.76} = 238 + 221.5 = \mathbf{509.5\ K}$$

Now we have to find out the fuel m_{f1} supplied in the combustion chamber-I

$$m_{f1} \times \text{C.V.} = (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_2)$$

$$\text{C.V.} = (m_{a1} + 1)\, C_{pg}\, (T_3 - T_2)$$

$$42000 = \left(\frac{m_{a1}}{m_{f1}} + 1\right) \times 1.128\ (953 - 509.5)$$

$$\therefore \quad \frac{m_{a1}}{m_{f1}} = \frac{42000}{1.128 \times 443.5} - 1 = 84.2 - 1 = \mathbf{83.2}$$

Now, the work developed by TB_1 must be equal to the work required to run the compressor.

$$\therefore \quad m_a C_{pa}\, (T_2 - T_1) = (m_{a1} + m_{f1})\, C_{pg}\, (T_3 - T_4)$$

$$= m_{f1} \left(\frac{m_{a1}}{m_{f1}} + 1\right) C_{pg}\, (T_3 - T_4) \quad \text{...(a)}$$

$$\frac{T_3}{T_4'} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma - 1}{\gamma}} = (5)^{0.254} = 1.5$$

$$\therefore \quad T_4' = \frac{953}{1.5} = \mathbf{635.3\ K}$$

$$\eta_{t1} = \frac{T_3 - T_4}{T_3 - T_4'}$$

$$\therefore \quad 0.86 = \frac{953 - T_4}{953 - 635.3}$$

$$\therefore \quad T_4 = 953 - (953 - 635.3) \times 0.86$$

$$= 953 - 273.2 = \mathbf{679.8\ K}$$

Now substituting the values in equation (*a*)

$$23 \times 1.005(509.5 - 288) = m_{f1}\ (83.2 + 1) \times 1.128\ (953 - 679.8)$$

$$5120 = m_{f1} \times 25929$$

$$\therefore \quad m_{f1} = \frac{5120}{25929} = \mathbf{0.198\ kg/sec}$$

$$\therefore \quad m_{a1} = 83.2 \times 0.198 = \mathbf{16.44\ kg/sec}$$

$$\therefore \quad m_{a2} = m_a - m_{a1} = 23.0 - 16.44 = \mathbf{6.56\ kg/sec}$$

(2) Now considering G – TB₂

Work developed by TB_2 is given by

$$W_2 = (m_{a2} + m_{f2})\, C_{pg}\, (T_3 - T_4) \quad \text{...(b)}$$

Considering the combustion chamber of TB_2, we can write

$$m_{f2} \times \text{C.V.} = (m_{a2} + m_{f2})\, C_{pg}\, (T_3 - T_2)$$

$$\therefore \quad m_{f2} \times 42000 = (6.56 + m_{f2}) \times 1.128\ (953 - 509.5)$$

$$= 500.3\ (6.56 + m_{f2})$$

$$\therefore \quad 83.95\ m_{f2} = 6.56 + m_{f2}$$

$$\therefore \quad m_{f2} = \frac{6.56}{82.95} = \mathbf{0.079\ kg/sec}$$

$$\therefore \quad \frac{m_{a2}}{m_{f2}} = \frac{6.56}{0.079} = \mathbf{135.2}$$

Now substituting the values in equation (*b*)

$$W_2 = (6.56 + 0.079) \times 1.128\ (953 - 679.8)$$
$$= 6.639 \times 1.128 \times 273.2 = \mathbf{2046\ kJ/sec = 2046\ kW}$$

The capacity of TB_1 to run the compressor W_1 is given by

$$W_1 = m_{a1}\ C_{pa}\ (T_2 - T_1)$$
$$= 16.44 \times 1.005\ (509.5 - 288)$$
$$= 3659.2\ \text{kJ/sec} = \mathbf{3659.2\ kW}$$

Total fuel consumed m_f is given by

$$m_f = m_{f1} + m_{f2} = 0.198 + 0.079 = 0.277\ \text{kg/sec} = \mathbf{16.62\ kg/min.}$$

The thermal efficiency of the plant is given by

$$\eta_{th} = \frac{W_2}{m_f \times \text{C.V.}}$$
$$= \frac{2046}{0.277 \times 42000} \times 100 = \mathbf{21.5\%}$$

Problem 24.29. *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. Taking the following data, find out the specific fuel consumption of the plant and plant capacity and overall efficiency of the plant.*

(1) η_c *(of each compressor) = 80%.*

(2) $\eta_{t1} = 0.87$ *and* $\eta_{t2} = 0.7$.

(3) Δp *(in intercooler) = 0.07 bar.*

(4) Δp *(in regenerator) = 0.1 bar in each side.*

(5) $\in$ *(Heat exchanger* η*) = 0.75.*

(6) Δp *(in combustion chamber) = 0.15 bar.*

(7) Δp *(in reheater) = 0.1 bar.*

(8) Mechanical η_m *= 99% for compressor-turbine.*

(9) Combustion η_{com} *= 98% (in combustion chamber and reheater).*

(10) Compression ratio of each stage of compressor = 2 : 1.

Ambient air temperature and pressure may be taken as 15°C and 1 bar. The maximum cycle temperature is limited to 1000 K and air mass flow is 20 kg/sec. Assume perfect intercooling.

Take $C_{pa} = 1$ *kJ/kg-K and* $C_{pg} = 1.1$ *kJ/kg-K and*
$\gamma_a = 1.4$ *and* $\gamma_g = 1.33$.

C.V. of fuel used = 43.5 MJ/kg. *(C.U., 1991)*

Solution. The arrangements of the components and processes are shown in Fig. Prob. 24.29 (*a*) and Fig. Prob. 24.29 (*b*).

As per given data

$$p_1 = 1\ \text{bar},\ p_2 = 2\ \text{bar as}\ \frac{p_2}{p_1} = 2\ \text{(given)}$$
$$p_3 = p_2 - 0.07 = 2 - 0.07 = 1.93\ \text{bar}$$
$$p_4 = 2\ p_3 = 2 \times 1.93 = 3.86\ \text{bar}$$

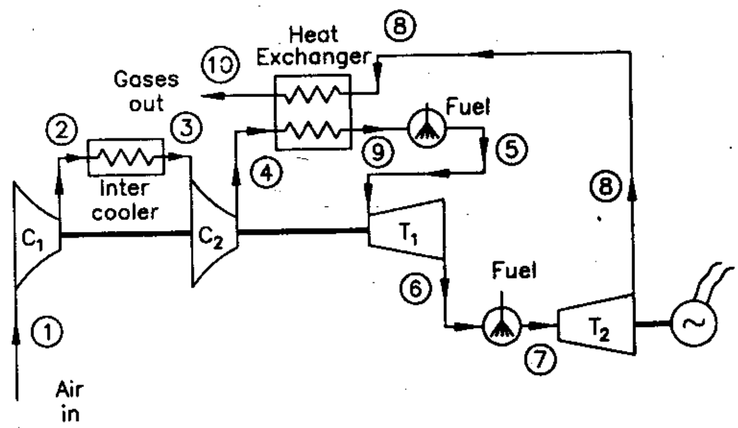


Fig. Prob. 24.29 (*a*).

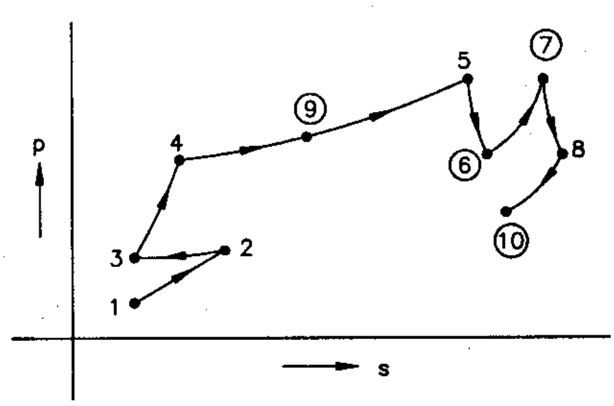


Fig. Prob. 24.29 (*b*).

$$p_5 = p_4 - \Delta p_1 - \Delta p_2$$

where Δp_1 and Δp_2 are pressure losses in H.E. and combuster one

$\therefore \quad p_5 = 3.86 - 0.1 - 0.15 = 3.61$ bar

p_6 is to be calculated

$p_7 = p_6 - \Delta p$ (pressure loss in reheater)

$p_8 = 1 + \Delta p$ (pressure loss in H.E. of gas side)

$= 1 + 0.1 = 1.1$ bar

$$T_2 = T_1 + \frac{T_1}{\eta_{c1}} \left[(R_{c1})^{\frac{\gamma - 1}{\gamma}} - 1 \right]$$

$$= 288 + \frac{288}{0.8}\left[(2)^{0.286} - 1\right] = 288 + 79 = \mathbf{367\ K}$$

$$T_4 = T_3 + \frac{T_3}{\eta_{c2}}\left[\left(R_{c2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= 288 + \frac{288}{0.5}\left[(2)^{0.286} - 1\right] = \mathbf{367\ K}$$

The power required to run the compressor

$$W_1 = m_a\, C_{pa}\, [(T_2 - T_1) + (T_4 - T_3)]$$

$$= 2\, m_a\, C_{pa}\, (T_2 - T_1) \text{ as } T_4 - T_3 = T_2 - T_1$$

$$= 2 \times 20 \times 1\ (367 - 288) = \mathbf{3160\ kW}$$

∴ Power developed by compressor turbine

$$W_{t1} = \frac{3160}{\eta_m} = \frac{3160}{0.99} = \mathbf{3192\ kW}$$

∴ Work developed by the turbine per kg of air

$$= \frac{3192}{20} = 159.6 \text{ kJ/kg}$$

The work developed by the turbine per kg of air is given by

$$C_{pg}\,(T_5 - T_6) = 159.6$$

$$\therefore \quad T_5 - T_6 = \frac{159.6}{1.1} = 145$$

The $(T_5 - T_6)$ is given by

$$T_5 - T_6 = T_5 \,.\, \eta_{t1}\left[1 - \left(\frac{1}{R_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$145 = 1000 \times 0.87\left[1 - \frac{1}{(R_{t1})^{0.286}}\right]$$

$$\therefore \quad R_{t1} = \mathbf{1.91}$$

$$p_5 = p_4 - 0.1 - 0.15 = 3.86 - 0.25 = \mathbf{3.61\ bar}$$

$$R_{t1} = \frac{p_5}{p_6} = 1.91$$

$$\therefore \quad p_6 = \frac{3.61}{1.91} = \mathbf{1.89\ bar}$$

$$p_7 = p_6 - 0.1 = 1.89 - 0.1 = 1.79 \text{ bar}$$

$$p_5 = 1.1 \text{ bar}$$

$$R_{t2} = \frac{p_7}{p_8} = \frac{1.79}{1.1} = 1.63$$

where R_{t1} and R_{t2} are the pressure ratios of the turbine I and turbine II

$$T_7 - T_8 = T_7 \times \eta_{t2}\left[1 - \left(\frac{1}{R_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$T_7 = T_5 = 1000 \text{ K (given)}$$

$$T_7 - T_8 = 1000 \times 0.8 \left[1 - \frac{1}{(1.63)^{0.286}}\right]$$
$$= 800\,(1 - 0.87) = 104.3$$
$$\therefore \quad T_8 = 1000 - 104.3 = \mathbf{895.7\ K}$$

The power output from the plant is only from turbine II.

$\therefore$ Net output of the plant

$$= m_a\, C_{pa}\, (T_7 - T_8)$$
$$= 20 \times 1.0\,(104.3) = \mathbf{2086\ kW}$$

The total heat supplied in the plant per kg of air

$$Q_s = C_{pa}\,[(T_5 - T_9) + (T_7 - T_6)]$$

The effectiveness of the heat exchanger is given by

$$\epsilon = \frac{T_9 - T_4}{T_8 - T_4} = 0.75$$
$$\therefore \quad T_9 = T_4 + 0.75\,(T_8 - T_4)$$
$$= 367 + 0.75\,(895.7 - 367) = \mathbf{763.5\ K}$$
$$\therefore \quad Q_s = 1.0\,[(1000 - 763.5) + (145)]$$
$$= \mathbf{381.5\ kJ/kg}$$

Assume m_f is the mass of fuel supplied/sec in combustion chamber and reheater, then,

Heat developed = Heat gained by air

$$m_f \times \text{C.V.} \times \eta_{com} = 20 \times 381.5$$
$$\therefore \quad m_f = \frac{20 \times 381.5}{43.5 \times 10^3 \times 0.98} = 0.18\ \text{kg/sec} = \mathbf{648\ kg/hr}$$

$\therefore$ Specific fuel consumption

$$= \frac{648}{2086} = \mathbf{0.31\ kg/kWh}$$

Thermal efficiency

$$= \frac{\text{Output}}{\text{Input}} = \frac{2086}{381.5 \times 20} = 0.274 = \mathbf{27.4\%}$$

Problem 24.30. *A gas turbine power plant consists of a single stage compressor which is run by a separate turbine. The air from the compressor is supplied separately to compressor turbine and power turbine. An heat exchanger is introduced in the system where the air coming out of compressor is heated by the exhaust gases coming out from both the turbines.*

Taking the following data, find out

(a) % of total air passed to the compressor turbine

(b) Find the combined temperature of the exhaust gases entering into the heat exchanger and the temperature of the gases entering into power turbine

(c) Isentropic η of the power turbine and

(d) Thermal efficiency of the plant.

(1) Ambient air temperature and pressure = 15°C and 1 bar.

(2) Maximum temperature of the gas entering into the compressor turbine is limited to 800°C.

(3) Pressure ratio of the compressor = 5.

(4) The temperature of the gases coming out of heat exchanger = 265°C.

(5) The net-output of the power turbine = 625 kW.

(6) The air flowing through the compressor = 5.85 kg/sec.

(7) Isentropic efficiency for compressor and for both the turbines = 0.86.

(8) Effectiveness of heat exchanger = 0.75.

Neglect fuel mass and mechanical and friction losses in the system. Take C_p and γ same for air as well as gas also.

Solution. The arrangement of the system component is shown in Fig. Prob. 24.30 (*a*) and the processes are shown in Fig. Prob. 24.30 (*b*).

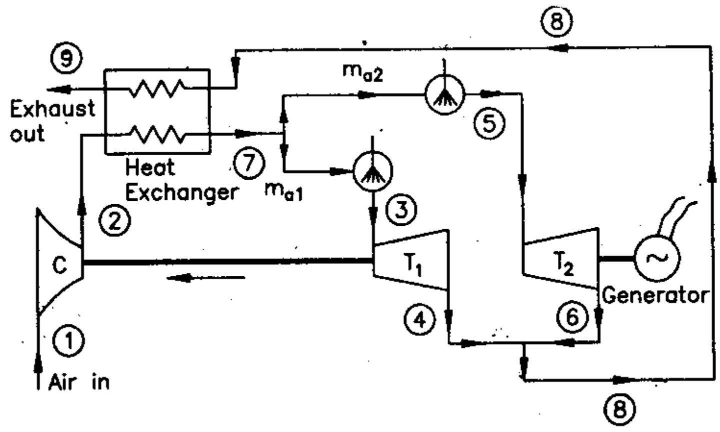
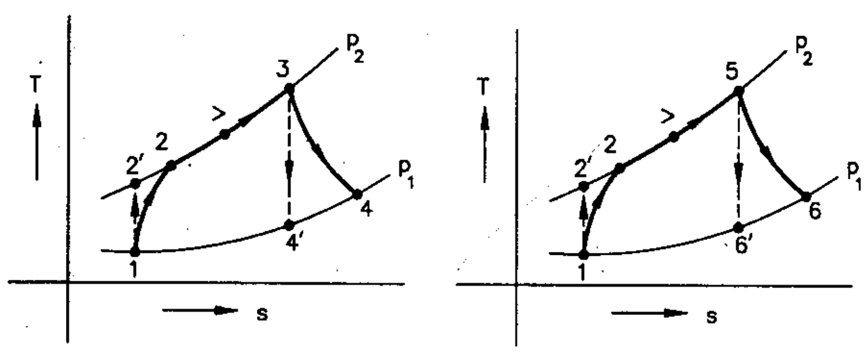


Fig. Prob. 24.30 (*a*).



T-s diagram for compressor-turbine-I *T-s* diagram for compressor-turbine-II

Fig. Prob. 24.30 (*b*).

As per the given configration of the plant

$$R_c = \frac{p_2}{p_1} = R_{t1} = R_{t2} = \frac{p_2}{p_1}$$

$$T_2 - T_1 = \frac{T_1}{\eta_c}\left[(R_c)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= \frac{288}{0.86}[(5)^{0.286} - 1] = \mathbf{195.8\ K}$$

$\therefore$ $T_2 = 288 + 195.8 = \mathbf{483.8\ K}$

The work done (W_c) in the compressor is given by

$$W_c = C_{pa}\, m_a\, (T_2 - T_1)$$

$$= 1 \times 5.85\ (195.8) = \mathbf{1145\ kW}$$

$$T_3 - T_4 = T_3 \times \eta_{t1}\left[1 - \left(\frac{1}{R_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$= (800 + 273) \times 0.86\left[1 - \frac{1}{(5)^{0.286}}\right] = \mathbf{338.7\ K}$$

Work developed by turbine I = Work absorbed by the compressor

$$C_{pa}\, m_{a1}\, (T_3 - T_4) = C_{pa}\, m_a\, (T_2 - T_1)$$

$\therefore$ $m_{a1} \times 338.7 = 1145$

$\therefore$ $m_{a1} = \mathbf{3.38\ kg/sec}$

$\therefore$ % of total mass supplied to turbine I

$$= \frac{3.38}{5.85} \times 100 = \mathbf{57.8\%}$$

m_{a2} (mass of air) supplied to turbine II

$$= 5.85 - 3.38 = \mathbf{2.47\ kg/sec}$$

Considering the heat exchanger,

Heat lost by gases = Heat gained by air

$$m_a\, C_{pg}\, (T_8 - T_9) = m_a\, C_{pa}\, (T_7 - T_2)$$

$\therefore$ $T_8 - T_9 = T_7 - T_2$ as $m_g = m_a$ and $C_{pg} = C_{pa}$ are given

$T_8 - 538 = T_7 - 483.8$

$\therefore$ $T_7 = T_8 - 54.2$ ...(*a*)

The effectiveness of the heat exchanger is given by

$$\epsilon = \frac{T_7 - T_2}{T_8 - T_2} = 0.75$$

Substituting the value of T_7 in equation (*a*)

$$\therefore \quad \frac{(T_8 - 54.2) - 483.8}{T_8 - 483.8} = 0.75$$

$\therefore$ $T_8 = \mathbf{700\ K}$

$\therefore$ $T_7 = 700 - 54.2 = \mathbf{645.8\ K}$

Considering the turbine II

$$T_5 - T_6 = T_5 \cdot \eta_{t2} \left[1 - \left(\frac{1}{R_{t2}}\right)^{\frac{\gamma - 1}{\gamma}}\right]$$

$$= 0.86\ T_5 \left[1 - \left(\frac{1}{5}\right)^{0.286}\right] = 0.316\ T_5$$

$$\therefore \quad T_6 = 0.684\ T_5 \qquad ...(b)$$

The power developed in turbine II is given by

$m_{a2}\ C_{pa}\ (T_5 - T_6) = 625$

Substituting the value of T_6 from equation (*b*)

$2.47 \times 1\ (T_5 - 0.684\ T_5) = 625$

$$\therefore \quad T_5 = \frac{625}{2.47 \times 0.316} = \mathbf{800\ K}$$

$$\therefore \quad T_6 = 0.684 \times 800 = \mathbf{547\ K}$$

The overall efficiency of the plant is given by

$$\eta_{th} = \frac{\text{Output}}{\text{Input}} = \frac{625}{m_{a1}\ C_{pa}\ (T_3 - T_7) + m_{a2}\ C_{pa}\ (T_5 - T_7)}$$

$$= \frac{625}{3.38 \times 1\ (1073 - 645.8) + 2.47 \times 1\ (800 + 547)}$$

$$= \frac{625}{1445 + 625} = \frac{625}{2070} = 0.302 = \mathbf{30.2\%}$$

Problem 24.31. *A heat exchanger is to be designed for an open cycle gas turbine set consisting of a two stage compressor with intercooling and two stage turbine with separate combustion chamber. The turbine stages are mechanically independent. The H.P stage derives the compressor and L.P. provides the power output. Taking the following data, determine the required thermal ratio of the heat exchanger and allowable gas side pressure loss in the heat exchanger. The output of the unit is 100 kW/kg/sec of air flow and the overall efficiency of the plant is 30%.*

(1) Pressure ratio of each compressor stage = 2.5 : 1.

(2) $\eta_{c1} = 0.85$.

(a) $\eta_{t1} = 0.88$. *(b)* $\eta_{t2} = 0.85$.

(3) Pressure loss in air side in H.E. and main combustion chamber = 0.2 bar.

(4) Pressure loss in reheat combustion chamber = 0.1 bar.

(5) Pressure loss in intercooler = 0.05 bar.

(6) Temperature at inlet to both turbine stages = 1000 K.

(7) Temperature after intercooling = 300 K.

(8) Ambient temperature and pressure = 288 K and 1 bar.

The fuel mass flow should be neglected and all other losses other than those stated in the problem should be neglected.

Assume there is perfect intercooling.

Solution. The arrangement of the components is shown in Fig. Prob. 24.31 (*a*) and processes are shown in Fig. Prob. 24.31 (*b*) on *T-s* diagram.

Imp. Note. The values p_1, p_2, p_3, p_4, p_5 ...... and T_1, T_2, T_3, T_4, T_5 ...... represent the values at points 1, 2, 3, 4, 5 ...... as shown on *T-s* diagram.

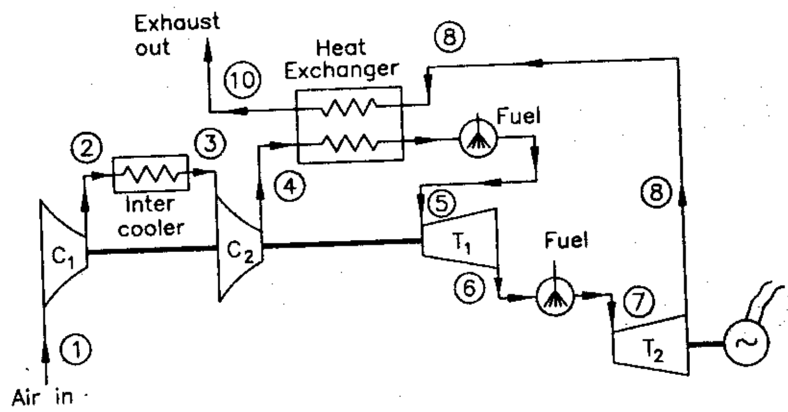


Fig. Prob. 24.31 (*a*).

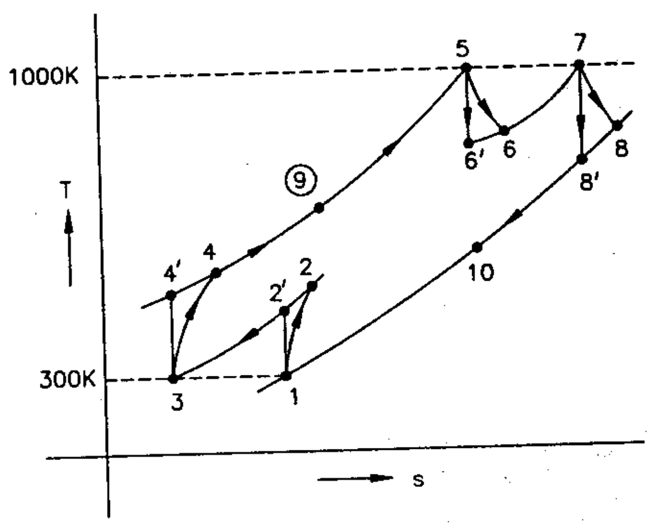


Fig. Prob. 24.31 (*b*).

$$T_2 = T_1 + \frac{T_1}{\eta_{c1}}\left[(R_{c1})^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= 288 + \frac{288}{0.35}\,[(2.5)^{0.286} - 1] = \mathbf{390\ K}$$

$p_2 = 2.5\ p_1 = 2.5 \times 1 =$ **2.5 bar**

$p_3 = p_2 -$ pressure loss in inter cooler

$= 2.5 - 0.05 =$ **2.45 bar**

$$T_4 = T_3 + \frac{T_3}{\eta_{c2}}\left[(R_{c2})^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$$= 300 + \frac{300}{0.85}\,[(2.5)^{0.286} - 1] = \mathbf{406\ K}$$

$$p_4 = p_3 \times 2.5 = 2.45 \times 2.5 = \mathbf{6.125\ bar}$$

The work required to compress one kg of air

$$= C_p\,[(T_2 - T_1) + (T_4 - T_3)]$$
$$= 1 \times [(390 - 300) + (406 - 300)] \text{ as } T_3 = T_1$$
$$= \mathbf{196\ kJ/kg}$$

This is also the work done by turbine I as it is directly coupled to the compressor and $\eta_m = 100\%$ assumed.

$$\therefore \quad C_p\,(T_5 - T_6) = 196$$
$$\therefore \quad T_5 - T_6 = 196$$
$$\therefore \quad T_6 = 1000 - 196 = \mathbf{804\ K}$$

But
$$T_5 - T_6 = T_5 \,.\, \eta_{t1}\left[1 - \left(\frac{1}{R_{t1}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$\therefore \quad 196 = 1000 \times 0.88\left[1 - \frac{1}{(R_{t1})^{0.286}}\right]$$

where R_{t1} is the pressure ratio of turbine I.

$$\therefore \quad \frac{1}{(R_{t1})^{0.286}} = 1 - \frac{196}{1000 \times 0.88} = 1 - 0.22 = 0.78$$

$$\therefore \quad R_{p1} = \mathbf{2.386.}$$

$$p_5 = p_4 - \text{pressure loss in H.E. and combustion chamber I}$$
$$= 6.125 = 0.2 = \mathbf{5.925\ bar}$$

$$R_{p1} = \frac{p_5}{p_6} = 2.386 = \frac{5.925}{p_6}$$

$$\therefore \quad p_6 = \frac{5.925}{2.386} = \mathbf{2.22\ bar}$$

$$p_7 = p_6 - \text{pressure loss combustion chamber II}$$
$$= 2.22 - 0.1 = 2.12 \text{ bar}$$

The power developed in turbine II per kg of air is given by

$$100 = C_{pa}\,(T_7 - T_8)$$

$$\therefore \quad T_8 = T_7 + \frac{50}{C_{pa}} = 1000 - \frac{100}{1} = \mathbf{900\ K}$$

$$T_7 - T_8 = T_7 \,.\, \eta_{t2}\left[1 - \left(\frac{1}{R_{t2}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

$$\therefore \quad 100 = 1000 \times 0.85\left[1 - \frac{1}{(R_{t2})^{0.286}}\right]$$

where R_{t2} is the pressure ratio in turbine II

$$\frac{1}{(R_{t2})^{0.286}} = 1 - \frac{100}{1000 \times 0.85} = 0.882$$

$$\therefore \quad R_{t2} = 1.55 = \frac{p_7}{p_8}$$

$$\therefore \quad p_8 = \frac{p_7}{1.55} = \frac{2.12}{1.55} = \mathbf{1.37\ bar.}$$

∴ Maximum pressure loss in H.E. towards gas side

$$= p_8 - p_{10} \text{ where } p_{10} \text{ (atmospheric pressure)}$$
$$= 1.37 - 1 = \mathbf{0.37\ bar}$$

The overall thermal efficiency of the plant

$$\eta = \frac{\text{Output}}{\text{Total input}}$$
$$= \frac{(T_7 - T_8)}{(T_5 - T_4) + (T_7 - T_6)}$$

$$\therefore \quad 0.3 = \frac{(1000 - 900)}{1000 - T_9 + 1000 - 804} = \frac{100}{1196 - T_9}$$

$$\therefore \quad T_9 = 1196 - \frac{100}{0.3} = \mathbf{862.7\ K}$$

The effectiveness ($\in$) of heat exchanger is given by

$$\in = \frac{T_9 - T_4}{T_8 - T_4} = \frac{862.7 - 406}{900 - 406} = \mathbf{0.925}$$

Problem 24.32. *In a gas turbine plant, air at 15°C is compressed to a pressure ratio of 6. Air is then heated to a maximum temperature of 750°C, first in the heat exchanger of 75% effectiveness and then in the combustion chamber. It is then expanded in two stages such that the expansion work is maximum. Assuming the condition of perfect reheating, calculate (a) Efficiency of the plant (b) Work ratio.*

Take isentropic efficiencies of turbines and compressor are 85% and 80% respectively and air pressure at the entry of compressor = 1 bar. *(B.U., Dec. 2000)*

Take all properties of gases as properties of air.

Solution. The arrangement of the components of the system is shown in Fig. Prob. 24.32 (*a*) and the processes are represented on *T-s* diagram as shown in Fig. Prob. 24.32 (*b*).

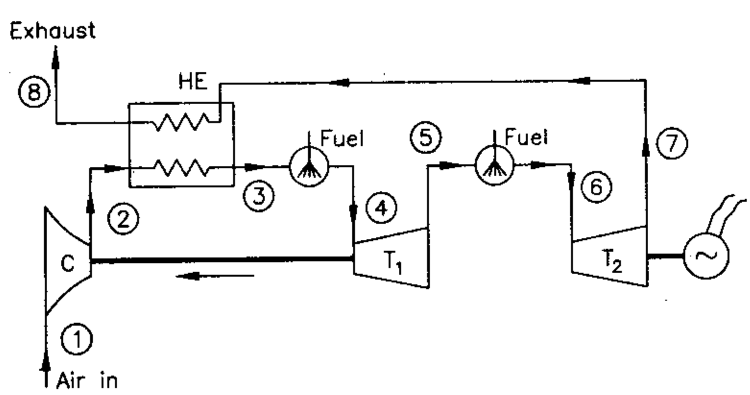


Fig. Prob. 24.32 (*a*).

$$T_2' = T_1 (R_p)^{\frac{\gamma - 1}{\gamma}} = 288 (6)^{0.286} = \mathbf{480.8\ K}$$

$$\eta_c = \frac{T_2' - T_1}{T_2 - T_1}$$

$$\therefore \quad T_2 = 288 + \frac{(480.8 - 288)}{0.8} = \mathbf{529\ K}$$

$\therefore$ The pressure p_3 (as two stages are designed for maximum work) is given by

$$p_3 = \sqrt{p_1 p_2} = \sqrt{1 \times 6} = \mathbf{2.45\ bar}$$

$$\therefore \quad R_{p1} = \frac{6}{2.45} = 2.45 = R_{p2}$$

$$T_5' = \frac{T_4}{(R_{p1})^{\frac{\gamma-1}{\gamma}}} = \frac{(750 + 273)}{(2.45)^{0.286}} = \mathbf{793\ K}$$

$$\eta_{t1} = \frac{T_4 - T_5}{T_4 - T_5'} = 0.85$$

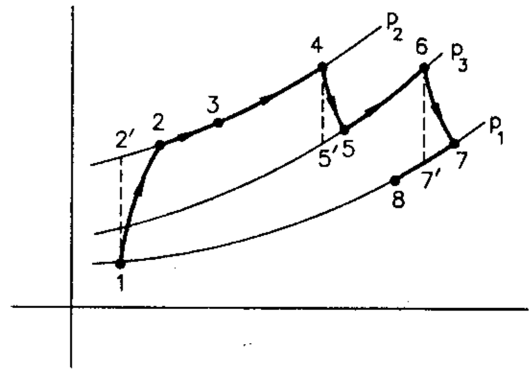


Fig. Prob. 24.32 (*a*).

$\therefore \quad T_5 = T_4 - 0.85\ (T_4 - T_5') = 1023 - 0.85\ (1023 - 793) = \mathbf{827.5\ K}$

As $\quad T_4 = T_6$ and $p_2/p_3 = p_3/p_1$

$\therefore \quad T_7 = T_5 = 827.5$ K

The effectiveness of the heat exchanger is given by

$$\epsilon = \frac{T_3 - T_2}{T_7 - T_2} = 0.75$$

$\therefore \quad T_3 = T_2 + 0.75\ (T_7 - T_2) = 529 + 0.75\ (827.5 - 529) = \mathbf{752.8\ K}$

The work of compression is given by

$$W_c = C_{pa}\ (T_2 - T_1) = 1.0\ (529 - 288) = \mathbf{241\ kJ/kg}$$

The work developed by both turbines is given by

$$W_t = 2\ C_{pg}\ (T_4 - T_5) = 2 \times 1.0\ (1023 - 827.5) = \mathbf{391\ kJ/kg}$$

$$W_n \text{ (Net work)} = W_t - W_c = 391 - 241 = \mathbf{150\ kJ/kg}$$

The total heat supplied is given by

$$Q_s = Q_1 + Q_2$$
$$= C_{pg}\ (T_4 - T_3) + C_{pg}\ (T_6 - T_5) = 1.0\ [(1023 - 752.8) + (1023 - 827.5)]$$
$$= \mathbf{465.7\ kJ/kg}$$

The plant efficiency is given by

$$\eta_p = \frac{W_n}{Q_s} = \frac{150}{465.7} = 0.322 = \mathbf{32.2\%}$$

Problem 24.33. *A gas turbine plant operates between 1 bar and 9 bar. The minimum and maximum temperatures of the cycle are 25°C and 1250°C respectively. A two stage compressor is provided with perfect intercooling and work is equally shared by each stage. Gases are reheated to 1250°C after H.P. turbine and then expanded in L.P. turbine, such that work output is equally divided between the two stages. Assuming all units are mounted on the same shaft and neglecting the fuel mass, calculate (i) thermal efficiency and (ii) work ratio. Assume ideal regeneration and isentropic efficiencies of the compressors and turbines as 0.83. Draw the line diagram of the components and sketch the cycle on T-s diagram. (P.U. Dec. 2000)*

(*b*) *Repeat the above problem without neglection the fuel mass and assuming C.V. of fuel used = 42 MJ/kg and combustion efficiency as 95%.*

Solution. The components of the system are shown in Fig. 24.33 (*a*). The processes are representedon *T-s* diagram as shown in Fig. Prob. 24.33 (*b*).

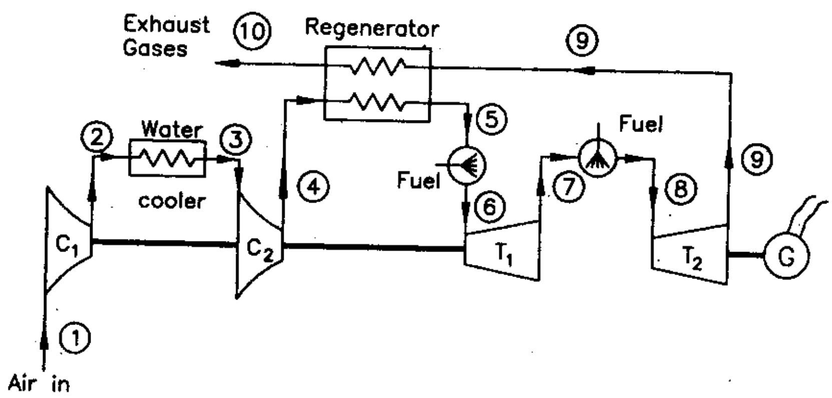


Fig. Prob. 24.33 (*a*).

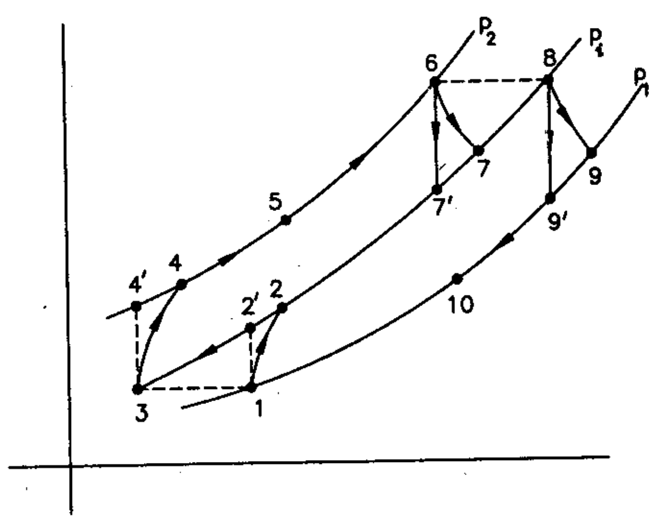


Fig. Prob. 24.33 (*b*).

The given data is

$$p_1 = 1 \text{ bar}, p_2 = 9 \text{ bar}$$
$$T_1 = 25 + 273 = 298 \text{ K} = T_3$$
$$T_6 = T_8 = 1250 + 273 = 1523 \text{ K}$$
$$\eta_{c1} = \eta_{c2} = \eta_{t1} = \eta_{t2} = 0.83$$
$$\epsilon = 0.83$$

The intermediate pressure p_i (when perfect cooling and perfect reheating is used) for minimum compressor work and maximum turbine work is given by

$$p_i = \sqrt{p_1 p_2} = \sqrt{1 \times 9} = 3 \text{ bar}$$

$$T_2' = T_1 \left(\frac{p_i}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 298\ (3)^{0.286} = 298 \times 1.37 = \mathbf{408\ K}$$

$$\eta_{c1} = \frac{T_1' - T_1}{T_2 - T_1} = 0.83$$

$$\therefore \quad \frac{408 - 298}{T_2 - 298} = 0.83$$

$$\therefore \quad T_2 = 298 + \frac{110}{0.83} = \mathbf{430\ K}$$

As there is perfect intercooling

$$T_3 = T_1$$

$$\therefore \quad T_4 = T_2 = \mathbf{430\ K}$$

$$\frac{T_6}{T_7'} = \left(\frac{p_2}{p_i}\right)^{\frac{\gamma-1}{\gamma}} = (3)^{0.286} = 1.37$$

$$\therefore \quad T_7' = \frac{1523}{1.37} = \mathbf{1111.7\ K}$$

$$\eta_{t1} = \frac{T_6 - T_7}{T_6 - T_7'}$$

$$\therefore \quad 0.83 = \frac{1523 - T_7}{1523 - 1111.7}$$

$$\therefore \quad T_7 = 1523 - 0.83\ (1523 - 1111.7) = \mathbf{1181.6\ K}$$

$$T_8 = 1523 \text{ K and } T_9 = T_7 = 1181.6 \text{ K}$$

The $\in$ (effectiveness) of regenerator is given by

$$\in = \frac{T_5 - T_4}{T_9 - T_4}$$

$$\therefore \quad 0.83 = \frac{T_5 - 298}{11181.6 - 298}$$

$$\therefore \quad T_5 = 298 + 0.83\ (1181.6 - 298) = \mathbf{1031.4\ K}$$

The work done by the both compressors is gven by

$$W_c = 2\ C_p\ (T_2 - T_1) = 2 \times 1 \times (430 - 298) = \mathbf{264\ kJ/kg\ of\ air}$$

The work developed by both turbines is given by

$$W_t = 2\ C_p\ (T_6 - T_7) = 2 \times 1\ (1523 - 1111.7) = \mathbf{822.6\ kJ/kg\ of\ air}$$

$$\therefore \quad W_n \text{ (Network developed)} = W_t - W_c = 822.6 - 264 = \mathbf{558\ kJ/kg\ of\ air}$$

$$\text{Work ratio} = \frac{W_n}{W_t} = \frac{558}{822.6} = \mathbf{0.68}$$

Heat supplied per kg of air is given by

$$Q_s = C_p\ (T_6 - T_5) + C_p\ (T_8 - T_7)$$
$$= 1\ (1523 - 1031.4) + 1\ (1523 - 1181.6)$$
$$= \mathbf{833\ kJ/kg\ of\ air}$$

The efficiency of the system is given by

$$\eta = \frac{W_n}{Q_s} = \frac{558}{833} = \mathbf{67\%}$$

Solve the following problem

A gas turbine plant has a pressure ratio of 6. Air enters the compressor at 15°C. The maximum temperature of the cycle is limited to 750°C. The expansion is carried out in two stages such that work output of each stage is same. Air is reheated to 750°C after leaving HP-turbine. A heat exchanger of effectiveness of 0.75 is added in the plant. The isentropic efficiencies of compressor and turbine are 0.8 and 0.85 respectively. Determine

(a) Thermal efficiency (b) Work ratio (c) Work done/kg of air. *(P.U. Dec. 2000 old syllabus)*

EXERCISES

24.1. Describe the working of a simple constant pressure open-cycle gas turbine plant giving a neat sketch. How does actual cycle differ from the theoretical ?

24.2. What different methods are used to improve the thermal efficiency of the open-cycle gas turbine plant ?

24.3. Why the maximum temperature in the gas turbine cycle is limited to 850°C ? Why lean A : F ratio is used in gas turbines and what is the range of it ?

24.4. Prove that the pressure ratio of a closed cycle for the maximum specific output is the square root of the pressure ratio for the maximum thermal efficiency. Why low pressure ratio is used in gas turbines ? What is the range of it ?

24.5. What problems are encountered in the design of gas turbine combustion chamber ? Draw a neat sketch of a combustion chamber used in modern open-cycle gas turbine power plant. What are the desirable requirements from the combustion system ?

24.6. What do you understand by a closed cycle gas turbine plant ? List out its advantages over open-cycle plant. What difficulties are encountered in the development of closed cycle plant ?

24.7. Sketch a line diagram of semi-closed cycle gas turbine power plant. Under what circumstances, these plants are more economical and superior in operation to closed cycle and open cycle plants ?

24.8. What are the effects on the thermal efficiency and specific output of gas turbine plant of the following factors, (*a*) Load on the plant, (*b*) Pressure ratio, (*c*) Turbine inlet temperature, (*d*) Compressor inlet temperture (*e*) Regenerator ?

24.9. What are the advantages of having a separate power turbine over a single shaft power plant arrangement ?

24.10. Draw a neat sketch and explain the working of a governing system used for closed cycle gas turbine plant.

24.11. (*a*) A closed-cycle gas turbine power plant with two stage compressor with perfect intercooling and single stage turbine is to be designed for maximum specific output. Prove that the pressure ratio to be used for the above-mentioned condition is given by

$$R_x = \left[\eta_c \cdot \eta_t \frac{T_{max}}{T_{min}}\right]^{\frac{2}{3}\gamma/(\gamma-1)}$$

where η_c is the isentropic efficiency of each compression stage and η_t is the isentropic efficiency of turbine, T_{max} and T_{min} are the maximum and minimum temperatures of the cycle.

(*b*) If the plant consists of single stage compressor with two stage turbine having a reheater in between and the temperatures of fluid entering in both turbines are same, then prove that the condition for the maximum specific output remains same as given in case (*a*).

24.12. A simple open cycle gas turbine plant works between the pressures of 1 bar and 6 bar and temperatures of 300 K and 1023 K. The C.V. of the fuel used is 40,500 kJ/kg. Find (*a*) A : F ratio, (*b*) Thermal efficiency of the plant (*l*) generating capacity of the plant if the mechanical and generating efficiencies are 95% and 96% respectively. Assume air flow = 1.2 kg/sec.

Assume the compression and expansion are isentropic and neglect the heat and pressure losses.

24.13. An open cycle gas turbine plant takes air in at 1 bar and 32°C. The maximum pressure ratio and maximum temperature of the cycle are limited to 7 and 1100 K respectively. The isentropic efficiencies of compressor and turbine are 84% and 85% respectively. The effectiveness of the regenerator is 0.6. Assuming the combustion efficiency 90% and neglecting the heat and pressure losses in the system, find the cycle efficiency and A : F ratio used.

If the air flow is 10 kg/sec, find the generation capacity of the plant assuming mechnical efficiency 96% and generation efficiency 95%.

24.14. An open cycle constant pressure gas turbine plant takes air at 27°C. The maximum pressure ratio and maximum temperature of the cycle are 4 and 950 K respectively. The isentropic efficiency of compressor and turbine are 80% and 85% respectively. The regenerator effectiveness is 75%. Assuming $\gamma = 1.4$ and $C_p = 1$ kJ/kg-K for both air and gases, find the power plant efficiency. If the capacity of the plant is 10 MW find the mass of air required per second and specific fuel consumption. Take suction pressure = 1 bar.

24.15. An open cycle, constant pressure gas turbine plant consists of two stage compressor with perfect intercooling and two stage turbine with combustion chamber and reheater. All the moving components are mounted on one common shaft. The pressure and temperature of air entering into the compressor are 1 bar and 15°C. The maximum pressure ratio and maximum temperature of the cycle are limited to 5 and 800°C respectively. The reheating takes place at 2.3 bar to 800°C. Isentropic efficiencies of each stage turbine are 80% and 90% respectively. C.V. of the fuel = 40,000 kJ/kg.

Taking $C_p = 1$ kJ/kg-K and $\gamma = 1.4$ for both air and gases and neglecting pressure and heat losses, find (*a*) overall efficiency of the plant, (*b*) overall A : F ratio, (*c*) specific fuel consumption and (*d*) MW capacity of the plant if the flow of air is 20 kg/sec.

24.16. An open-cycle constant pressure regenerative type gas turbine plant consists of two stage compressor with an intercooler in between. The plants consist of two turbines one for compressor and another for alternator. The air temperature entering the L.P. compressor is 15°C and leaving the intercooler is 22°C. The maximum pressure ratio and maximum temperature of the cycle are limited to 5 and 950 K. The burned gases coming out from H.P. turbine are passed through L.P. turbine without reheating.

$\eta_c = 85\%$ for each stage, $\eta_t = 85\%$ for each stage, $\epsilon = 0.75$, $\eta_m = 98\%$, C.V. of fuel = 40,000 kJ/kg, $C_{pa} = 1$ kJ/kg-K and γ (for air) = 1.4, $C_{pg} = 1.15$ kJ/kg-K and $\gamma = 1.33$ (for gases).

Assuming the air flow of 15 kg/sec, and neglecting the heat and pressure losses in the system, find (*a*) power developed in kW (*b*) specific fuel consumption and (*c*) thermal efficiency of the plant.

24.17. An open cycle gas turbine power plant consists of a single stage compressor, regenerator and two separate turbines which are mechanically independent. Each turbine has its own combustion chamber and exhaust of each turbine is passed through the regenerator. Air from the compressor is supplied to both the turbines. The temperature and pressure of the air entering into the compressor are 300 K and 1 bar. The maximum pressure ratio of the cycle is limited to 4. The temperatures of the gases entering into the compressor turbine and power turbine are 1030 K and 1060 K respectively, $\eta_c = 75\%$, η_{t1} (compressor turbine) = 80%, η_{t2} (power turbine) = 75% C.V. of fuel 40,000 kJ/kg, $C_{pa} = 1$ kJ/kg-K, $C_{pg} = 1.$[illegible] kJ/kg-K γ (air) = 1.4, γ (gases) = 1.33. The temperature of exhaust gases going to atmosphere is 400°C. Neglecting heat and pressure losses in the system, find (*a*) power of compressor-turbine if the alternator generates 6.5 MW, (*b*) specific fuel consumption and (*c*) thermal efficiency of the cycle. Neglect the mass of fuel for finding the work.

24.18. A series flow gas turbine unit is used for power generation. There is separate single stage compressor turbine and single stage power-turbine with two stage compressor with perfect intercooling. The gases coming out of compressor turbine are further heated to 1000 K. The air is taken in at 15°C and 1.013 bar and the maximum temperature of the cycle is also limited to 1000 K.

Isentropic efficiency of each stage compressor = 80%.
Pressure ratio of each stage = 2 : 1.
The pressure loss in intercooler = 0.07 bar.
Pressure loss on each side of regenerator = 0.105 bar.
Pressure loss in combustion chamber = 0.14 bar.
Effectiveness ratio of heat exchanger = 0.98.
Isentropic efficiency of power turbine = 80%.
Mechanical efficiency of compressor turbine = 100%.
Combustion efficiency in combustion chamber = 99%.
Pressure loss in reheater = 0.105 bar.
Combustion efficiency in reheater = 98%.

Assuming the air flow of 22.7 kg/sec and neglecting the heat losses in the system, determine the power output in kW, specific fuel consumption and overall thermal efficiency of the plant.

$C_{pa} = 1$ kJ/kg-K, $\gamma = 1.4$ for air.
$C_{pg} = 1.1$ kJ/kg-K and $\gamma = 1.33$ for gases.
C.V. of fuel used = 52000 kJ/kg.

24.19. A regenerative gas turbine power plant consists of two stage compressor with perfect intercooling and single turbine. All the components of the plant are mounted on a single shaft. The overall pressure ratio is 8 : 1. The maximum temperature of the cycle is limited to 590°C. The regenerator recovers 60% of the available energy from the exhaust gases. The compressor (each stage) and turbine isentropic efficiencies are 83% and 86% respectively. Find the efficiency and ratio of useful work to the turbine work.

24.20. In a gas turbine plant, air at pressure P_1 and temperature T_1 is compressed isentropically to pressure RP_1 and heated to absolute temperature T_3. The air then expands isentropically in two stage turbine, reheating to temperature T_2.

The isentropic efficiencies of compressor and each turbine are η_c and η_t respectively. If the intermediate pressure between each turbine is $r.p_1$ show that for the given values of $T_1, T_3, p_1, \eta_c, \eta_t$ and R, the work output will be maximum when $r = \sqrt{R}$.

Neglect the heat and pressure losses in the system.

If this division in pressure drop is maintained, show that, if R is varied, the work output will again be maximum where R is given by

$$R = \left[\eta_c \cdot \eta_t \frac{T_3}{T_1}\right]^{\frac{2}{3}\,[\gamma/(\gamma-1)]}$$

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