MCE 508

POWER GENERATING PLANTS

(3 UNITS)

PREPARED BY

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1.0 STEAM PLANT

1.1 INTRODUCTION

In the steam plant, steam is generated in a boiler from which it passes into the steam main. The steam main feeds the steam into a turbine or engine or it may pass into some other plant such as heaters or process machinery. After expanding through the turbine or engine or passing through some other plant, if the plant is working on a "dead-loss" system, then the exhaust steam passes away to atmosphere. But if steam recovery plant is installed, the exhaust steam passes into condenser where it is condensed to water, called condensate. The condensate is extracted from the condenser by the condensate extraction pump from which it passes as feed water by means of feed water pump back to the boiler, the losses in the system are made up in the condenser by means of a make-up water supply.

1.2 ADVANTAGES OF STEAM RECOVERY PLANT (CONDENSER)

(a)   The pressure in the condenser can be operated well below atmospheric pressure. This means that a greater expansion of the steam can be obtained, which results in more work.

(b)   The water in the circuit can be chemically treated to reduce scale formation in the boiler.

Effects of Scale Formation in the Boiler

(i)   It impedes the transfer of heat from the furnace to the water

(ii)   It reduces the boiler efficiency

(iii)   It may cause local over-heating with resulting damage

(iv)   If overheating is serious, it may cause a burst in the vicinity.

1.2   VARIOUS CIRCUITS IN STEAM PLANT

1.2.1   The furnace Gas Circuit.

Air is taken into the furnace from the atmosphere to supply the necessary oxygen for combustion. The combustion products pass through the boiler, transferring heat, then pass out to the atmosphere through the flue. Most furnaces are fired by coal, gas or oil.

1.2.2   The Steam Circuit

Water is passed into the boiler where it is converted into steam. It passes into plant where it is expanded, giving up some of its energy. It is then condensed in a condenser and passes as condensate to be pumped back into the boiler.

1.2.3   Condenser Cooling Water Circuit

Cooling water passes into the condenser, has heat transferred into it by the condensing steam then at a higher temperature, passes out to be cooled in a cooling tower. Cooled water then circulates back to the condenser.

1.2.4   Cooling Air Circuit

In the case of a cooling tower, cool air passes into the bottom of the tower, from the atmosphere and heat is transferred into it from the falling hot water spray. The warm air then passes back to the atmosphere through the top of the tower.

1.2.5   Hot Well

In some steam plant the condenser is passed into a tank, called the "hot well" which acts as a reservoir for feed water. From the hot well, feed water is pumped through the feed pump back into the boiler. In this case, make-up water could be fed into the hot well.

1.3   BOILERS

A boiler is the device in which steam is generated. Generally, it must consist of a water container and some heating devices. In the boiler, steam leave the water at its surface and pump into what is called the steam space. This is the space in the water container directly above the water. Steam formed above the surface of water is always wet, therefore, the wet steam is removed from the steam space and piped into a " super heater" This consist of a long tube or series of tubes which are suspended across the path of the hot gases from the furnace. As the wet steam progresses through the tube or tubes it is gradually dried out and enebturally superheated. From the superheater it passes to the steam main.

1.3.1   Improvement of Thermal Efficiency of the Boiler

The flue gases will still be hot, having passed through the main boiler then the super heater. The energy in these flue gases can be used to improve the thermal efficiency of the boiler in the ways.

(a)   Thermal efficiency of the boiler can be improved by passing the flue gases through an 'economizer'.   The economizer is really on heat exchanger in which the feed water being pumped into the boiler is heated, therefore, less energy is required to raise the steam.

(b)   Having passed the flue gases through the economizer, they are still moderately hot. Further thermal efficiency improvement can be obtained by passing the flue gases through an air heater. Air heater is also a heat exchanger in which the air being ducted to the boiler furnace is heated.

1.3.2   Types of Boiler

There are many designs of boiler, but they can be divided into two types; fir-tube boilers and water-tube boilers.

(a)   Fire-Tube Boiler

A fire-tube boiler is sometimes called on economic boiler it has a cylinderic outer shell and contains two larger-bore flues into which are set the furnaces. The hot flue gases pass out of the furnace flues are made to pass through a number of small-bore tubes arranged above the large-bore furnace flues. These small-bore tubes break up the water bulk in the boiler and present a large heating surface to he water. In this type of boiler, the flue gasses are made to pass inside the tube while the water to be heated is outside the tube.

(b)   Water-Tube Boiler

With the increasing demand for higher power output from steam plant it became necessary to develop boilers with higher pressures and   steam outputs that could be handled by the shell-tube boilers.   This led to the development of the water-tube boiler. The furnaces of around the furnace wall. Most heat energy in this type of boiler is transferred by radiation to the vertical water tubes and from the tubes to the water inside the tubes.

1.4   BOILER CALCULATION

This is given by the ratio of the energy received by the steam to the energy supplied by the fuel to produce the steam.

Therefore, Boiler thermal efficiency = $\dfrac{\text{Energy to Steam}}{\text{Energy from Fuel}}$

If m_s = Mass of steam raised in a given time

m_f = Mass of fuel used in the same time

Then Boiler Efficiency = $\dfrac{m_s(h_2 - h_1)}{m_f.\ CV}$ x 100%

Equipvalent Evaporation of a Boiler

The size of the boiler, or its capacity, is quoted as the rate in kg/hour at which the steam is generated. A comparison is sometimes made by an equivalent evaporation, which is defined as the quantity of steam produced per unit quantity of fuel burned when the evaporation process-takes place from and at 100^0C.

At 1000C it will be the enthalpy of evaporation which is supplied specific enthalpy of evaporation at 100^0C; hfg = 2256.9 kJ/kg.

Therefore, the equivalent evaporation of a boiler, from and at 100^0C is:

$\dfrac{m_s(h_2 - h_1)}{2256.9}$ kg in the given time or per kilogram of fuel

QUESTION 1.1

A boiler with super heater generates 6000 kg/h of steam at a pressure of 15 bar, 0.98 dry at exit from boiler and at a temperature of 300^oC on leaving the super heater. If the feed water temperature is 80^oC and the overall efficiency of the combine boiler and super heater is 85%, determine:

(a)   the amount of coal of calorific value 30,000 kJ/kg used per hour.

(b)   the equivalent evaporation from and at 100^oC for the combined unit.

(c)   the heating surface required in the superheater if the rate of heat transmission may be taken as 450000 kJ/m^2 of heating surface per hour.

Solution

(a)   Specific enthalpy of feed water (h1) at 80^oC is 334.9 kJ/kg (from steam table).

Specific enthalpy of steam generated (h2) at pressure of 15 bar and temperature of

3000C is 303 kJ/kg

Therefore, energy transferred to the steam = m(h2-h1)

    =   6000 (3039 – 334.9) = 6000 x 2704.1

    =   16224600 kJ/h

This is 85% of the energy from the fuel

Therefore, energy from fuel = $(16225$ x $10^3)/0.85$ $= 19088$ x 10^3 kJ/h

(b)   The equivalent evaporation = 7189 kg/h

(c)   Heating surface required in the superheater specific enthalpy of steam at exit from

boiler at 0.98 dryness and 15 bar

Energy transferred in the super heater = m(h2-h)

    =   6000 (3039 – 2753.06) = 1716 x 10^3 kJ/h

$$\text{Area} \quad = \quad \frac{\text{(heat transferred)}}{\text{heat transferred per unit area}} \quad = \quad \frac{1716 \text{ x } 10^3}{45000} \quad = \quad 3.81\text{m2}$$

QUESTION 1.2

A boiler delivers 5400 kg steam per hour at a pressure of 7.5 bar and with a dryness

fraction of 0.98. The feed water to the boiler is at a temperature of 41.5^oC. The coal used for

firing the boiler has a calorific value of 31000 kJ/kg and is used at the rate of 670 kg/h.

Determine:

(a)   the thermal efficiency of the boiler and

(b)   the equivalent evaporation of the boiler in kg/kg of coal.

An economizer is fitted to the boiler which raises the feed water temperature to 1000C. The thermal efficiency of the boiler is increased by 5%, all other conditions remaining, unaltered. Determine:

(c)   the new coal consumption in kg/h and the saving in coal in kg/h obtained by fitting the economizer.

Solution

(a)   The thermal efficiency of the boiler specific enthalpy of feed water (h1) at temperature of 41.5^{o}C is 174 kJ/kg. Specific enthalpy of steam (h2) at 0.98 dryness and pressure of 7.5 bar is:

$h_2 = h_f + xh_{fg} = 709 + 0.98 (2057,5) = 2725.35$ kJ/kg

Energy to the steam $= m(h_2 - h_1) = 5400 (2725.35 - 174)$

$$= 13777 \times 10^3 \text{ kJ/h}$$

Energy from the fuel $= m.CV = 670 \times 31000 = 20770 \times 10^3$ kJ/h

$$\text{Efficiency} = \frac{\text{Energy to steam}}{\text{Energy from fuel}} = \frac{13777 \times 103}{20770 \times 103} = 0.663 \text{ or } 66.3\%$$

(b)   The equivalent evaporation of the boiler in kg/kg of coal

$$\text{Equivalent evaporation} = \frac{m(h_2 - h_1)}{2256.9} \text{ kg/h} = \frac{m(h_2 - h_1)}{670 \times 2256.9} \text{ kg/kg of coal}$$

$$= \frac{13777 \times 103}{670 \times 2256.9} = 9.11 \text{ kg/kg of coal}$$

(c)   The new enthalpy of feed water (h$_1$) at temp. 100°C is 419.1 kJ/kg.

Therefore, the new energy to the steam $= m(h_2 - h_1)$

$$= \quad 5400(2725.35 - 419) = 12454 \times 103 \text{ kJ/h}$$

New efficiency = $\dfrac{\text{New Energy to the Steam}}{\text{New Energy from the fuel}}$

New Efficiency = 66.3 + 5 = 71.3% or 0.713

New energy from fuel = $\dfrac{\text{New energy to the steam}}{\text{New Efficiency}}$

$$= \quad \dfrac{12454 \times 10^3}{0.713} \quad = \quad 17467 \times 10^3 \text{ kJ/h}$$

New Energy from fuel = m.CV

Therefore, m = $\dfrac{\text{New Energy from fuel}}{\text{CV}}$ $\quad = \quad \dfrac{17467 \times 10^3}{31000} \quad = \quad$ 563 kg/h

Saving in coal consumption = (Old rate – New rate)

$$= 670 - 563 = 107 \text{ kg/h}$$

2.0   STEAM CYCLES

2.1   INTRODUCTION

A steam power plant is a complicated thermodynamic system of many components, all of which function together as a unit with one objective of producing useful positive work transfer from an energy source (fuel). The steam cycle or vapour power cycle is one of the thermal cycles employed for the conversion of heat energy into mechanical work, and this in turn, is used for electric power generation. The working fluid for the cycle is alternately condensed and vapourized as is taken through the cycle. The most common working fluid is water this is due to the fact that water is cheap, readily available, non-toxic and chemically stable.

It has been said, under heat engine, that the Carnot cycle is the most efficient cycle operating between two specified temperature levels. Therefore, it is natural to look at the carnot cycle first as a prospective ideal cycle for vapour power cycles.

2.2   THE CARNOT VAPOUR CYCLE

Consider a steady flow Carnot cycle executed within the saturation done of a pure substance, as shown in Fig. 2.1 The fluid is heated reversible and isothermally in a boiler (Process 1-2), expanded isentropically in a turbine (process 2-3), condensed reversible and isothermally in a condenser (process 3-4), and compressed isentropically by a compressor to the initial state (process 4-1).

Impracticalities Associated with Carnot vapour cycle

The following are the impracticalities associated with carnot vapour cycle.

(a)   The condensation process is at constant temperature and pressure, a process which is difficult to stop at state 3 and then commence the compression of a wet vapour to state.

(b)   The process of compressing a wet vapour at state 3 to saturated liquid at state 4 is impracticable. This is because the process would involve compression of wet vapour to an enormous pressure which would require a large compression or pumping machine of very small power. It is not practical and economical to design such a machine.

(c)   Throughout the expansion process (1-2), the steam is wet leading to erosion and corrosion f the turbine blades

(d)   The Carnot cycle has a low work ratio. Work ratio rw, defined as the ratio of net work to positive work, is a criterion of performance which measures the cycle sensitivity to irreversibilities decrease, positive work and increase negative work accordingly a low work ratio contributes to low cycle efficiency and conversely a high work ratio leads to high cycle efficiency.

2.3   RANKINE CYCLE

Many of the impracticalities associated with Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser.

The ideal cycle for steam plant is known as the Rankine cycle, which is a constant pressure flow process cycle. A simple steam plant is shown in Fig. 2.2 and the cycle does not involve any internal irreversibilities. The cycle consists of the following four process as shown on T-S diagram Fig. 2.3

Process 1-2: Reversible adiabatic expansion of steam in a turbine,

$$- W_{12} = h_2 - h_1$$

Process 2-3: Reversible constant pressure heat transfer from cycle in a condenser

$$-Q_{23} = h_3 - h_2$$

Process 3-4: Reversible adiabatic compression of saturated water in a feed pump

Process 4-1 Reversible constant pressure heat transfer to cycle in a boiler

$$Q_{4-1} = h_1 - h_4 \hspace{4cm} (4)$$

Net work output $= W = h_1 - h_2$

Rankine efficiency, η_R $\dfrac{\text{Network output}}{\text{Heat supplied in the boiler}}$

The heat supplied in the boiler, $Q_{4-1} = h_1 - h_4$

Therefore, $\eta_R = \dfrac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$

Or $\quad \eta_R = \dfrac{(h1 - h2) - (h4\text{-}h3)}{(h1\text{-}h3) - (h4 - h3)} \hspace{3cm} (6)$

If the feed-pump term, $h_4 - h_3$, is neglected, Eq. (6) becomes:

$$\eta_R = \dfrac{h_1 - h_2}{h_1 - h_3} \hspace{4cm} (7)$$

Pump Work Input

When the feed-pump term is to be included it is necessary to evaluate the quantity, W_{34}.

From Eq (3), pump work $= W_{34} = h_4 - h_3$

It can be shown that for a liquid, which is assumed to be incompressible (i.e $v =$ constant), the increase in enthalpy for isentropic compression is given by

$$(h_4 - h_3) = v(P_4 - P_3) \qquad\qquad (8)$$

where v can be taken from tables for water at the pressure P_3.

The efficiency ratio of a cycle is the ratio of the actual efficiency to the ideal efficiency. In vapour cycles the efficiency ratio compares the actual cycle efficiency to the Rankine cycle efficiency

That is, efficiency ratio = $\dfrac{\text{Cycle efficiency}}{\text{Rankine efficiency}}$ \qquad\qquad (9)

2.4 ACTUAL RANKINE CYCLE

The expansion and pumping processes in the ideal Rankine cycle are assumed to be reversible. The actual expansion process in the turbine is irreversible, as shown by line 1-2 in Fig. 2.4. Similarly the actual compression of the water in the pump is irreversible and indicated by line 3-4.

Fig. 2.4: Ranking cycle showing actual process on T-S diagram.

(a) Causes of Irreversibilities in Vapour Power Cycles

Fluid friction and undesired heat loss to the surroundings are the two most common sources of irreversibilities.

(i) Fluid Friction:

Fluid friction causes pressure drops in the boiler, the condenser, and the piping between various components. As a result, steam leaves the boiler at a somewhat lower pressure. Also, the pressure at the turbine inlet is somewhat lower than that at the boiler exit

due to the pressure drop in the connecting pipes. The pressure drops in the condenser is usually very small. To compensate for these pressure drops, the water must be pumped to a sufficiently higher pressure than the ideal cycle calls for. This requires a larger pump and larger work input to the pump.

(ii)   Heat Loss from the Steam

Heat loss from the steam to the surroundings as the steam flows through various components. To maintain the same level of network, more heat needs to be transferred to the steam in the boiler to compensate for these undesired losses.

(b)   Irrevesibilities Occuring within the Turbine and The Pump

A turbine produces a smaller work output, and a pump requires a greater work input as a result of irreversibilities. Under ideal conditions, the flow through these devices is isentropic. The deviation of actual turbines and pumps from the isentropic ones can be accuraately accounted for, however, by utilizing isentropic efficiencies.

Turbine isentropic efficiency (η_{Ts}) is define as the ratio of actual work to ideal or isentropic work.

$$\text{Therefore,} \quad = \quad \frac{\text{actual work}}{\text{Ideal work}} \quad = \quad \frac{W_a}{Ws} \quad = \quad \frac{W_{12}}{W_{12s}}$$

$$= \quad \frac{h_1 - h_2}{h1 - h_{2s}} \qquad\qquad (10)$$

Pump isentropic efficiency (η_{Ps}) is define as the ratio of ideal or isentropic work to actual work, which is similar to that of compressor.

Therefore,
$$= \frac{\text{Ideal work}}{\text{Actual work}} = \frac{W_s}{W_a} = \frac{W_{34s}}{W_{34}}$$

$$= \frac{h_{4s} - h_3}{h_4 - h_3} \qquad (11)$$

2.5 CRITERIA OF PERFORMANCE

(a) Cycle Efficiency:

The cycle efficiency is defined as the ratio of network output to heat input. The efficiency of a cycle when all the processes are assumed to be reversible is known as the "ideal cycle efficiency' By introducing process efficiencies, it is also possible to estimate the actual cycle efficiency. The ratio of the actual cycle efficiency to the ideal cycle efficiency is called the efficiency ratio.

$$\text{Efficiency,} = \frac{\text{network output}}{\text{Heat input}} \qquad (12)$$

(b) Work Ratio:

Some cycles are more sensitive to irreversibilities than others. A cycle may have high ideal cycle efficiency and also, its actual cycle efficiency may be low. Therefore, ideal cycle efficiency, by itself, is not a good indication of whether or not the cycle will produce a power plant of high plant's. Sensitivity to irreversibility. Any power cycle consists of processes of boths positive and negative work transfer, and work ratio is defined as the ratio of the net work to the positive workdone in the cycle. Irreversibilities decrease the positive work and increase the negative work. If the ideal negative work is only slightly less than the ideal positive work, i.e. if work ratio is only slightly greater than zero, quite a small amount of component inefficiency is sufficient to reduce the net work output to zero other hand, as the

work ratio approaches unit, the same amount of component inefficiency. Will have a much smaller cycle efficiency.

$$\text{Work ratio, } r_w \quad = \quad \frac{\text{Network output}}{\text{Positive Work}} \qquad (13)$$

(c)   Specific Steam Consumption (SSC)

Another criterion of performance in steam plant is the specific steam consumption (SSC). It relates the power output to the steam flow necessary to produce the power. The steam flow indicates the size of plant and its components parts, and the specific steam consumption is a means whereby the relative sizes of different plants can be compared. The specific steam consumption is define as the mass flow of steam required to produce unit power output. It is usually expressed in kg/kwh and if Wnet is the net work output in kJ/kg, the SSC can be found from

$$\text{SSC} \quad = \quad \frac{3600}{\text{Wnet}} \qquad (Kg/kWh) \qquad (14)$$

2.6   RANKINE CYCLE WITH SUPERHEAT

The average temperature at which heat is supplied in the boiler can be increased by superheating the steam. This is achieved by placing a separate bank of tubes (the super heater) in the combustion gases from the furnace, which changes the state of dry saturated steam from the steam will improve efficiency since it   increases the maximum   cycle temperature but this is fixed by the   ambient cooling water temperature used in the condensers. This means that steam is usually condensed at about 27^oC at the corresponding saturation pressure of 3.6 kN/m^2, which is well below atmospheric pressure.

The Rankine cycle then appears as shown in Fig. 6.5, from the figure , it is evident that the average temperature at which heat is supplied is increased by superheating and hence the ideal cycle efficiency is increased. There is very little change in susceptibility to irreversibilities, the work ratio being so very near unity in the un-superheated Rankine cycle that the slight increase obtained by superheating makes very little difference. Nevertheless, the specific steam consumption is reduced, the network per unit mass of steam being much greater, so that the added complexity of a super heater is compensated by a reduction in the size of other components.

Fig.2.5: The Rankine Cycle with superheat

Equations under un-superheated Rankine are still valid.

QUESTION

Consider a steam power plant operating on the simple ideal Rankine cycle. The steam enters the turbine at 3 MPa and 350°C and is condensed in the condenser at a pressure of 75 kPa. Determine:

(i)   the pump work

(ii)   net work done by cycle

(iii)   the thermal efficiency of the cycle.

2.7   THE REHEAT CYCLE

As shown in the previous section, the condition of the exhaust steam is improved by superheating the steam. It can be further improved most effectively by reheating the steam. This also will further improve the Rankine cycle efficiency. In the reheat cycle, the expansion

of the steam takes place in two turbines. The steam expands in the "high-pressure turbine" to some intermediate pressure, and is then passed back to another bank of tubes in the boiler where it is reheated at constant pressure, usually to the original superheat temperature. The steam then expands in the "low-pressure turbine" to the condenser pressure. Alternatively, the reheat may take place in a separate re-heater situated near the turbine. In this arrangement, the amount of pipe work required is this arrangement, the amount of pipe work required is reduced. The use of reheat cycles has encouraged the development of higher pressure, forced circulation boilers which increased the dryness fraction of the exhaust steam and reduce the specific steam consumption.

Fig. 2.6 shows the T-s diagram of the reheat cycle. As shown in the diagram, process 1-2 represents isentropic expansion in the high-pressure turbine, process 6-7 represents isentropic expansion in low-pressure turbine, and the steam is reheated at constant pressure in process 2-6. The work output, the thermal efficiency and the specific steam consumption for the cycle are obtained as follows:

Fig. 2.6: Reheat Cycle on T-s diagram

Heat supplied = $Q_{41} + Q_{26}$

Neglecting the feed-pump work, therefore,

$Q_{41} = h_1 - h_3$

And for the reheat process, $Q_{26} = h_6 - h_2$

Work output = $W_{12} + W_{67}$

Where, $W_{12} = h_1 - h_2$ and $W_{67} = h_6 - h_7$

$$\text{Thermal efficiency} = \frac{\text{Work output}}{\text{Heat supplied}}$$

The specific steam consumption, SSC

$$SSC = \frac{3600}{W_{12} + W_{67}} \quad kg/kWh$$

QUESTION

A reheat cycle works between pressures 30 and 0.04 bar. The steam is superheated to 450°C, and after expansion to the dry saturated state is reheated to the original superheat temperature.

Calculate: (i) the dryness fraction and the enthalpy of the final exhaust steam

(ii) the ideal cycle efficiency and steam consumption for the cycle neglecting the feed-pump work.

3.0 THE GAS TURBINE CYCLES

3.1 THE IDEAL GAS TURBINE CYCLE

The ideal cycle for gas-turbine engines was first proposed by George Brayton for use in the reciprocating oil-burning engine that he developed in 1870. Today it is used for gas turbines only where the compression and expansion processes take place in rotating machinery. Gas turbines usually operate on an open cycle as shown in Fig. 3.1.
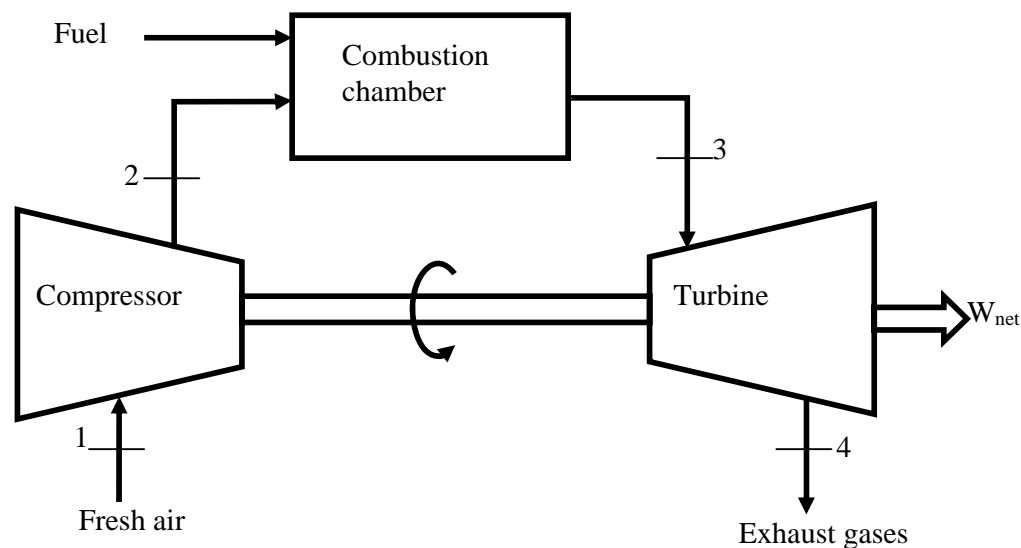

Fig. 3.1: Open-cycle gas-turbine engine

Fresh air at ambient conditions is drawn into the compressor, where its temperature and pressure are raised. High-pressure air proceeds into combustion chamber, where the fuel is burned at constant pressure. The resulting high-temperature gases then enter the turbine, where they expand to the atmospheric pressure, thus producing power. The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as open cycle.

The open gas-turbine cycle can be modelled as a closed cycle, as shown in Fig. 3.2, by utilizing the air-standard assumption. Here the compression and expansion processes remain

the same, but the combustion process is replaced by a constant pressure heat-addition process from an external sources, and the exhaust process is replaced by a constant pressure heat-rejection process to the ambient air. The ideal cycle that the working fluid undergoes in this closed loop is the Brayton cycle, which is the same as the Joule cycle or constant pressure flow process cycle.
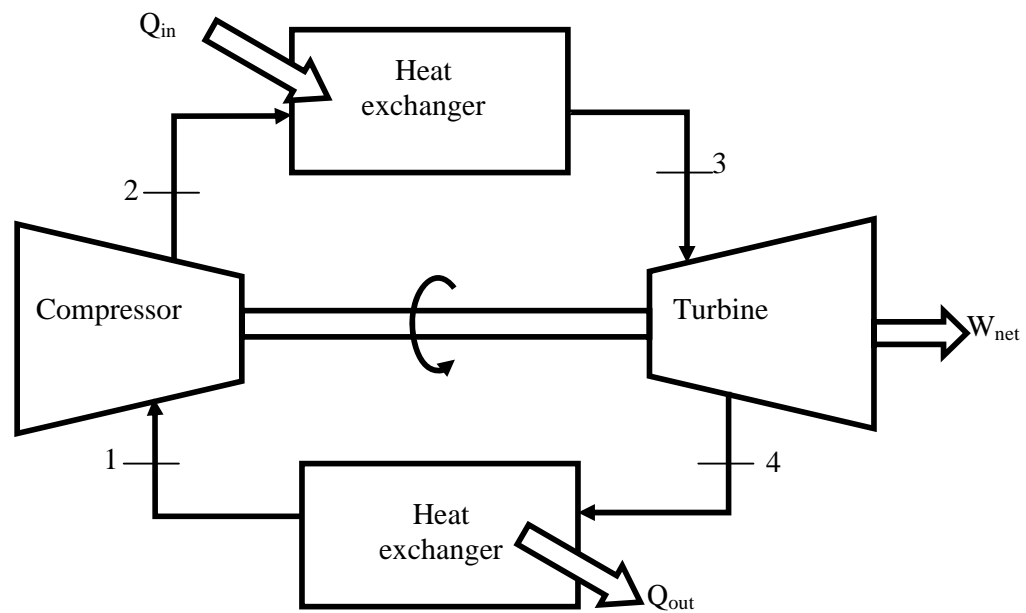

Fig. 3.1: Closed-cycle gas-turbine engine

The cycle is made up of four internally reversible processes:

Process 1-2: Isentropic compression (in a compressor)

Process 2-3: Constant pressure heat addition (in a heater)

Process 3-4: Isentropic expansion (in a turbine)

Process 4-1 Constant pressure heat rejection (in a cooler)

The steady flow energy equation is used to analyse each process, changes in kinetic and potential energy being neglected. Therefore, heat transfers to and from the working fluid are:

$$Q_{in} \;=\; h_3 - h_2 \;=\; C_p(T_3 - T_2) \tag{3.1}$$

$$Q_{out} \;=\; h_4 - h_1 \;=\; C_p(T_4 - T_1) \tag{3.2}$$

The thermal efficiency of the cycle is the same with that of constant pressure (Joule) cycle.

Therefore,

Therefore, $\eta_{Th} = \dfrac{W_{net}}{Q_{in}}$ (3.4)

or, $\eta_{Th} = 1 - \dfrac{1}{r_p^{\frac{\gamma-1}{\gamma}}}$ (3.5)

The Network output, $W_{net} = C_p(T_3 - T_4) - C_p(T_2 - T_1)$