



UNIT 5

GAS TURBINES



Course Objective:

Student have knowledge of methods of analysis and design of complicated thermodynamic systems

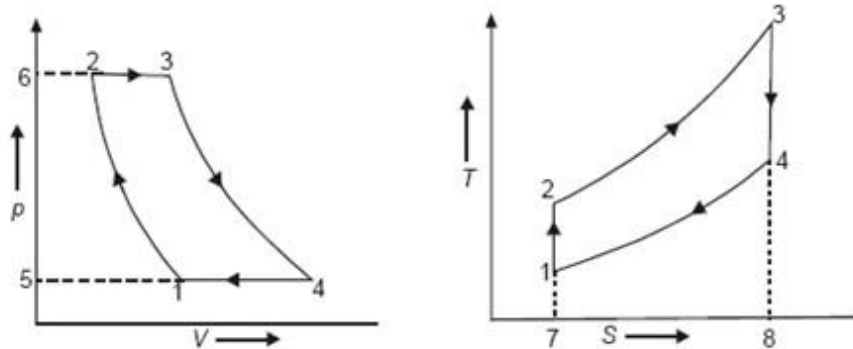
Course Outcome:

To be able to recognize main and supplementary elements of turbines and define operational principles.



Brayton Cycle

Brayton cycle, popularly used for gas turbine power plants comprises of adiabatic compression process, constant pressure heat addition, adiabatic expansion process and constant pressure heat release process. A schematic diagram for air-standard Brayton cycle is shown in Fig. 4.1. Simple gas turbine power plant working on Brayton cycle is also shown here.



Brayton cycle on P - V and T - S diagram

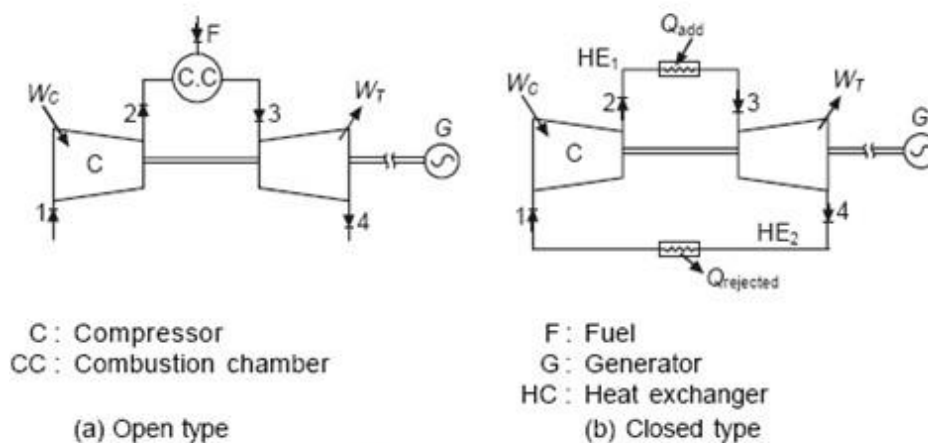


Fig. 4.1 Simple gas turbine plant

Thermodynamic cycle shows following processes:

1-2: Adiabatic compression, involving (–ve) work, W_C in compressor.

2-3 : Constant pressure heat addition, involving heat Q_{add} in combustion chamber or heat exchanger.

3-4: Adiabatic expansion, involving (+ve) work, W_T in turbine.

4-1: Constant pressure heat rejection, involving heat, $Q_{rejected}$ in atmosphere or heat exchanger.

In the gas turbine plant layout shown process 1–2 (adiabatic compression) is seen to occur



in compressor, heat addition process 2–3 occurs in combustion chamber having open type arrangement and in heat exchanger in closed type arrangement. Process 3–4 of adiabatic expansion occurs in turbine.

In open type arrangement exhaust from turbine is discharged to atmosphere while in closed type, heat rejection occurs in heat exchanger. In gas turbine plant of open type, air entering compressor gets compressed and subsequently brought up to elevated temperature in combustion chamber where fuel is added to high pressure air and combustion occurs. High pressure and high temperature combustion products are sent for expansion in turbine where its' expansion yields positive work. Expanded combustion products are subsequently discharged to atmosphere. Negative work required for compression is drawn from the positive work available from turbine and residual positive work is available as shaft work for driving generator.

In gas turbine plant of closed type the working fluid is recycled and performs different processes without getting contaminated. Working fluid is compressed in compressor and subsequently heated up in heat exchanger through indirect heating. High pressure and high temperature working fluid is sent for getting positive work from turbine and the expanded working fluid leaving turbine is passed through heat exchanger where heat is picked up from working fluid. Thus, the arrangement shows that even costly working fluids can also be used in closed type as it remains uncontaminated and is being recycled.

Air standard analysis of Brayton cycle gives work for compression and expansion as;

$$W_C = m_1 \cdot (h_2 - h_1)$$

$$W_T = m_3 \cdot (h_3 - h_4)$$

for air standard analysis, $m_1 = m_3$, where as in actual cycle

$$m_3 = m_1 + m_f, \quad \text{in open type gas turbine}$$

$$m_3 = m_1, \quad \text{in closed type gas turbine}$$

For the fuel having calorific value CV the heat added in air standard cycle;



$Q_{\text{add}} = m_1(h_3 - h_2)$, whereas $Q_{\text{add}} = m_f \times CV$ for actual

cycle. Net work = $W_T - W_C$

$$W_{\text{net}} = \{m_3 (h_3 - h_4) - m_1(h_2 - h_1)\}$$

$$\begin{aligned} \text{Air standard cycle efficiency} &= \frac{W_{\text{net}}}{Q_{\text{add}}} \\ &= \frac{m_1 \{(h_3 - h_4) - (h_2 - h_1)\}}{m_1 (h_3 - h_2)} \end{aligned}$$

$$\text{Air standard Brayton cycle efficiency: } \boxed{\eta_{\text{Brayton}} = 1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}}$$

Thus, it is obvious from the expression of efficiency that it depends only on pressure ratio (r) and nature of gas (γ). For pressure ratio of unity, efficiency shall be zero. For a particular gas the cycle efficiency increases with increasing pressure ratio. Here the variation of efficiency with pressure ratio is shown for air ($\gamma = 1.4$) and monatomic gas as argon ($\gamma = 1.66$).

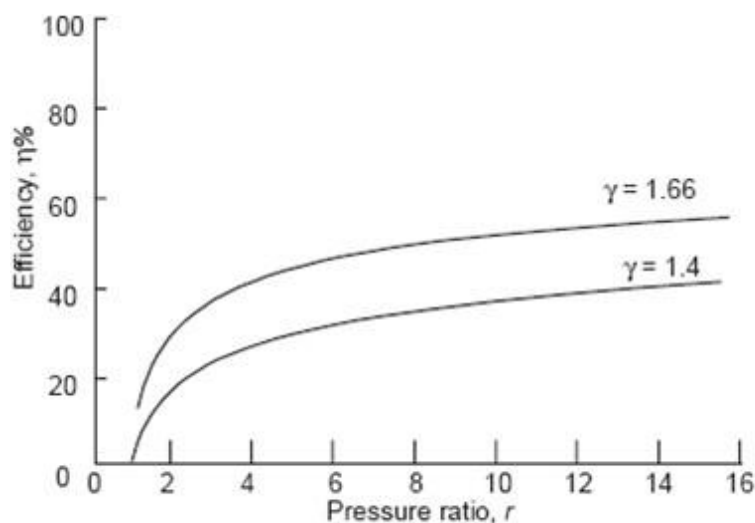


Fig. 4.2 Efficiency vs. pressure ratio in simple cycle



Regenerative gas turbine cycle

Regenerative air standard gas turbine cycle shown ahead in Fig. 4.3 has a regenerator (counter flow heat exchanger) through which the hot turbine exhaust gas and comparatively cooler air coming from compressor flow in opposite directions. Under ideal conditions, no frictional pressure drop occurs in either fluid stream while turbine exhaust gas gets cooled from 4 to 4' while compressed air is heated from 2 to 2'. Assuming regenerator effectiveness as 100% the temperature rise from 2–2' and drop from 4 to 4' is shown on T-S diagram.

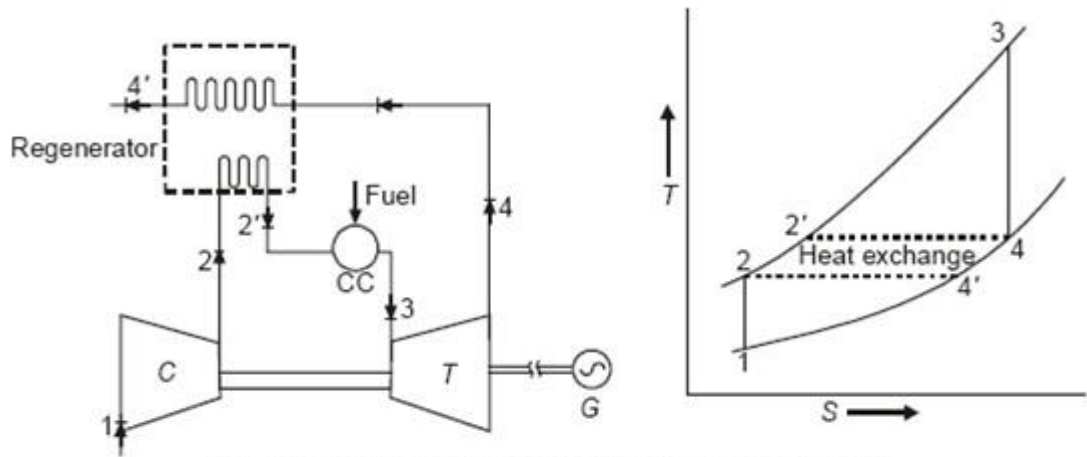


Fig. 4.3 Regenerative air standard gas turbine cycle.

$$\text{Regenerator effectiveness, } \epsilon = \frac{h_{2'} - h_2}{h_4 - h_2},$$

Thus, thermodynamically the amount of heat now added shall be

$$Q_{\text{add, regen}} = m (h_3 - h_{2'})$$

Where as without regenerator the heat added; $Q_{\text{add}} = m (h_3 -$

$h_2)$ Here it is obvious that, $Q_{\text{add, regen}} < Q_{\text{add}}$

This shows an obvious improvement in cycle thermal efficiency as every thing else remains same. Net work produced per unit mass flow is not altered by the use of regenerator.

$$\text{Air standard cycle thermal efficiency, } \eta_{\text{regen}} = \frac{(h_3 - h_4) - (h_2 - h_1)}{(h_3 - h_{2'})}$$

$$\eta_{\text{regen}} = \frac{c_p (T_3 - T_4) - c_p (T_2 - T_1)}{c_p (T_3 - T_{2'})}$$



Reheat gas turbine cycle

Reheat gas turbine cycle arrangement is shown in Fig. 4.4. In order to maximize the

work available from the simple gas turbine cycle one of the options is to increase enthalpy of fluid entering gas turbine and extend its expansion upto the lowest possible enthalpy value.

C: Compressor

HPT: High pressure turbine

CC: Combustion chamber

LPT: Low pressure turbine

G: Generator

RCC: Reheat combustion

chamber f: Fuel

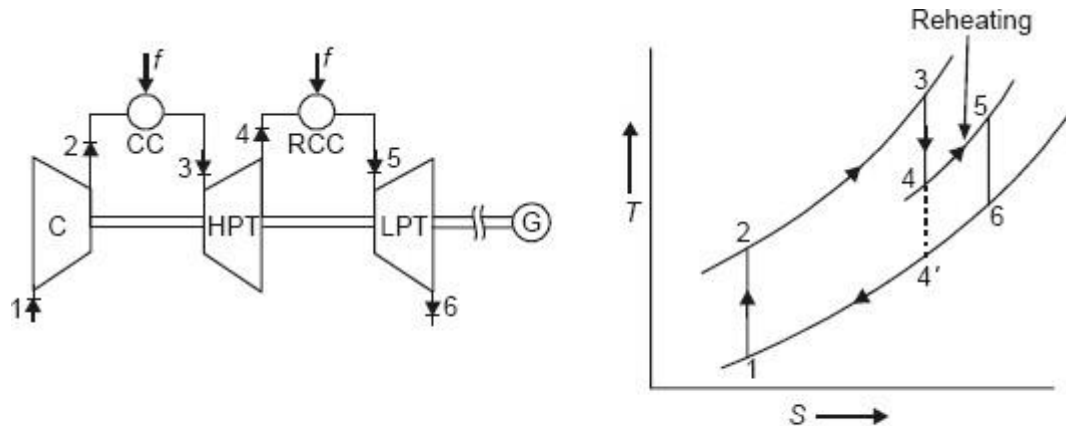


Fig. 4.4 Reheat gas turbine cycle

This can also be said in terms of pressure and temperature values i.e. inject fluid at high pressure and temperature into gas turbine and expand upto lowest possible pressure value. Upper limit at inlet to turbine is limited by metallurgical limits while lower pressure is limited to near atmospheric pressure in case of open cycle. Here in the arrangement shown ambient air enters compressor and compressed air at high pressure leaves at 2. Compressed air is injected into combustion chamber for increasing its temperature up to desired turbine inlet temperature at state 3. High pressure and high temperature fluid enters high pressure turbine (HPT) for first phase of expansion and expanded gases leaving at 4 are sent to reheat combustion chamber (reheater) for being further heated. Thus, reheating is a kind of energizing the working fluid.



Assuming perfect reheating (in which temperature after reheat is same as temperature attained in first combustion chamber), the fluid leaves at state 5 and enters low pressure turbine (LPT) for remaining expansion upto desired pressure value. Generally,

temperature after reheating at state 5 is less than temperature at state 3. In the absence of reheating the expansion process within similar pressure limits goes upto state 4'. Thus, reheating offers an obvious advantage of work output increase since constant pressure

lines on T - S diagram diverge slightly with increasing entropy, the total work of the two stage turbine is greater than that of single expansion from state 3 to state 4'. i.e.,

$$(T_3 - T_4) + (T_5 - T_6) > (T_3 - T_4').$$

Here it may be noted that the heat addition also increases because of additional heat supplied for reheating. Therefore, despite the increase in net work due to reheating the cycle thermal efficiency would not necessarily increase.



A plot showing variation of efficiency with pressure ratio ' r ' is shown in Fig. 4.5 along with simple cycle efficiency variation. It indicates that reheating offers increase in specific work output at the

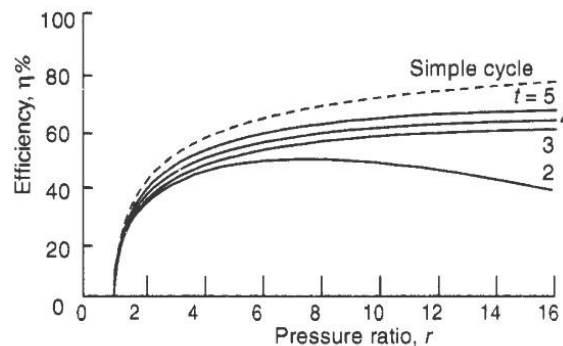


Fig. 4.5 Reheat cycle efficiency vs. cycle pressure ratio

cost of cycle efficiency. This reduction in efficiency may be attributed to the addition of a less efficient cycle 4564' to a simple cycle. 4564' is a less efficient cycle since it operates over a smaller temperature range. Variation of specific work output with pressure ratio is shown in Fig. 4.6. It shows how specific work output shows increase with increasing pressure ratio upto optimum pressure ratio. It may also be noted that in reheat cycle, the temperature of exhaust gases at exit of gas turbine gets increased as compared to simple cycle within similar limits. Therefore, reheat cycle offers potential for use of regenerator for harnessing the hotter exhaust from gas turbine.

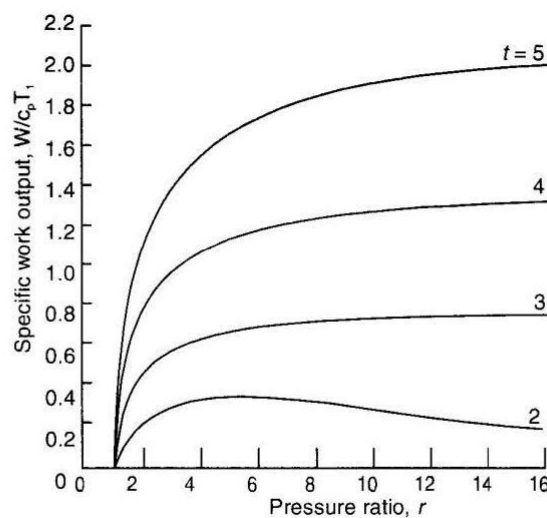


Fig. 4.6 Reheat cycle specific work output vs. cycle pressure ratio

GAS TURBINE CYCLE WITH INTERCOOLING

Net work output from gas turbine cycle can also be increased by reducing negative work i.e. compressor work. Multistaging of compression process with intercooling in between is one of the approach for

reducing compression work. It is based on the fact that for a fixed compression ratio higher is the inlet temperature higher shall be compression work requirement and vice-a-versa. Schematic for intercooled gas turbine cycle is given in Fig. 4.7.



Thermodynamic processes involved in multistage intercooled compression are shown in Figs.

4.8, 4.9. First stage compression occurs in low pressure compressor (LPC) and compressed air leaving LPC at '2' is sent to intercooler where temperature of compressed air is lowered down to state 3 at constant pressure. In case of perfect intercooling the temperatures at 3 and 1 are same. Intercooler is a kind of heat exchanger where heat is picked up from high temperature compressed air. The amount of compression work saved due to intercooling is obvious from p - V diagram and shown by area 2342'. Area 2342' gives the amount of work saved due to intercooling between compression.

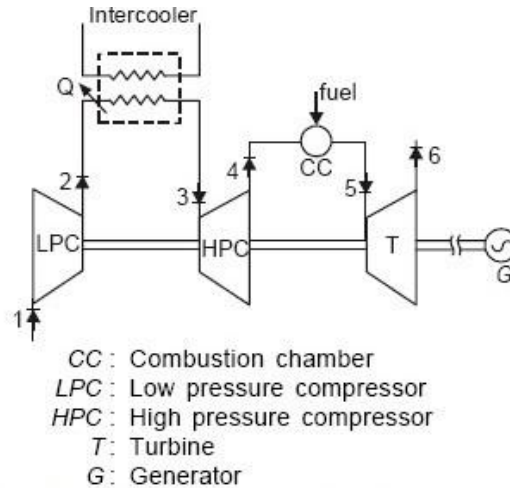


Fig. 4.7 Gas turbine cycle with intercooling

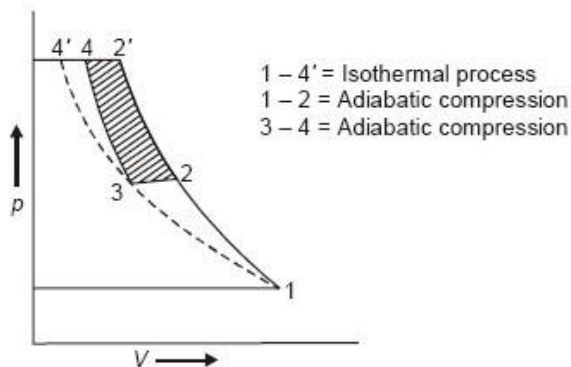


Fig. 4.8 Intercooled compression

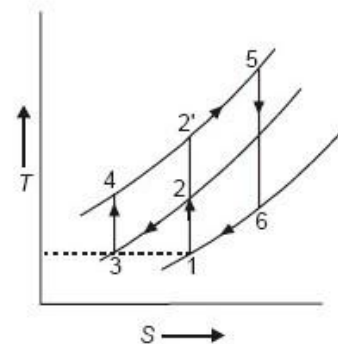


Fig. 4.9 T-S diagram for gas turbine

for gas turbine cycle with intercooling shows that in the absence of intercooling within same pressure limits the state at the end of compression would be 2' while with perfect intercooling

this state is at 4 i.e., $T_2' > T_4$. The reduced temperature at compressor



exit leads to additional heat requirement in combustion chamber i.e. more amount of fuel is to be burnt for attaining certain turbine inlet temperature as compared to simple cycle without intercooling.

Thus, intercooled cycle thermal efficiency may not increase with intercooling because of simultaneous increase in heat addition requirement. The lower temperature at compressor exit enhances the potential for regeneration so when intercooling is used in conjunction with regeneration an appreciable increase in thermal efficiency can result.

Net work output in gas turbine cycle with intercooling;

$$W_{\text{net, intercool}} = m\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}$$

$$W_{\text{net, intercool}} = mc_p\{(T_5 - T_6) - (T_4 - T_3) - (T_2 - T_1)\}$$

Cycle thermal efficiency;

$$\eta_{\text{intercool}} = \frac{\{(h_5 - h_6) - (h_4 - h_3) - (h_2 - h_1)\}}{\{h_5 - h_4\}}$$



GAS TURBINE CYCLE WITH REGENERATION, REHEAT AND INTERCOOLING

Regenerative gas turbine employing reheating during expansion and intercooling during compression is considered here as shown in Fig.4.13 . This combination offers considerable increase in net work output and thermal efficiency.

- C_1 : Low pressure compressor stage
- C_2 : High pressure compressor stage
- CC : Combustion chamber
- RCC : Reheat combustion chamber
- T_1 : High pressure turbine stage
- T_2 : Low pressure turbine stage
- G : Generator

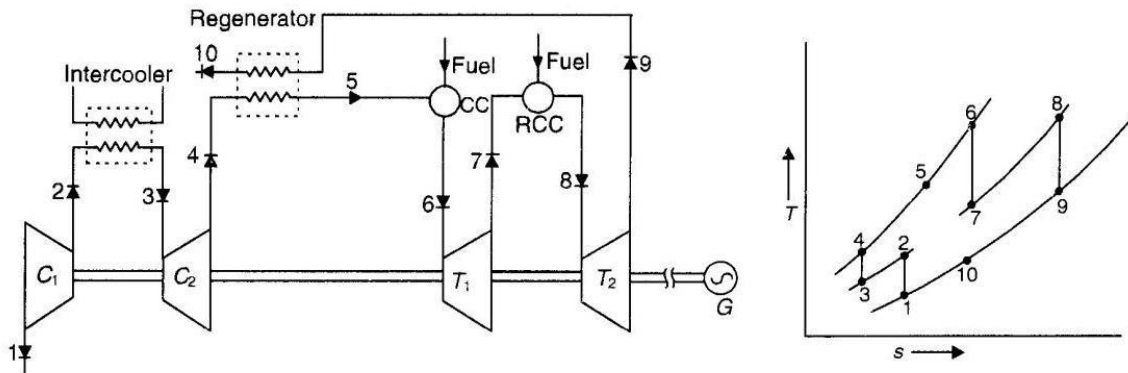


Fig. 4.13 Schematic for gas turbine cycle with regeneration, reheat and intercooling and T-S diagram

Based upon air standard cycle considerations thermodynamic analysis gives the Net work output from cycle,

$$W_{\text{net}} = m \{ (h_6 - h_7) + (h_8 - h_9) - (h_4 - h_3) - (h_2 - h_1) \}$$

Heat added

$$Q_{\text{add}} = m \{ (h_6 - h_5) + (h_8 - h_7) \}$$

Cycle thermal efficiency,

$$\eta_{\text{cycle}} = \frac{\{ (h_6 - h_7) + (h_8 - h_9) - (h_4 - h_3) - (h_2 - h_1) \}}{\{ (h_6 - h_5) + (h_8 - h_7) \}}$$

GAS TURBINE IRREVERSIBILITIES AND LOSSES

Till now the discussions have been confined to air standard Brayton cycle. But the realistic gas turbine cycle has deviations from air standard cycle due to,

- (i) frictional effects within compressor and turbine which causes increase in specific entropy of working fluid across these components.
- (ii) friction which shall cause drop in pressure of working fluid across the constant pressure processes.

Apart from above irreversibilities of the gas turbine power plant the irreversibilities of combustion chamber are quite significant.

Salient state points of realistic gas turbine Brayton cycle with above irreversibilities and losses are shown below:



Isentropic efficiency of turbine and compressor can be mathematically given as

$$\eta_{\text{isen}, t} = \left\{ \frac{h_3 - h_4}{h_3 - h_{4s}} \right\}$$

$$\text{i.e. } \eta_{\text{isen}, t} = \frac{\text{Actual expansion work}}{\text{Ideal expansion work}}$$

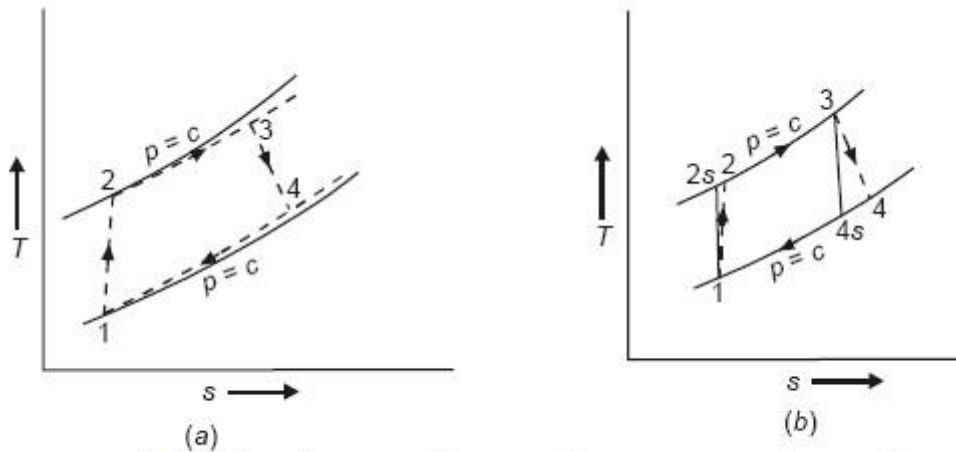


Fig. 4.14 Effect of irreversibilities and losses in gas turbine cycle.

Isentropic efficiency of compressor

$$\eta_{\text{isen}, c} = \left\{ \frac{h_{2s} - h_1}{h_2 - h_1} \right\}$$

$$\therefore \eta_{\text{isen}, c} = \frac{\text{Ideal compressor work}}{\text{Actual compressor work}}$$

Other factors causing the real cycle to be different from ideal cycle are as given below:

- (i) Fluid velocities in turbomachines are very high and there exists substantial change in kinetic energy between inlet and outlet of each component. In the analysis carried out earlier the changes in kinetic energy have been neglected whereas for exact analysis it cannot be.
- (ii) In case of regenerator the compressed air cannot be heated to the temperature of gas leaving turbine as the terminal temperature difference shall always exist.
- (iii) Compression process shall involve work more than theoretically estimated value in order to overcome bearing and windage friction losses.

Different factors described above can be accounted for by stagnation properties, compressor and turbine isentropic efficiency and polytropic efficiency.



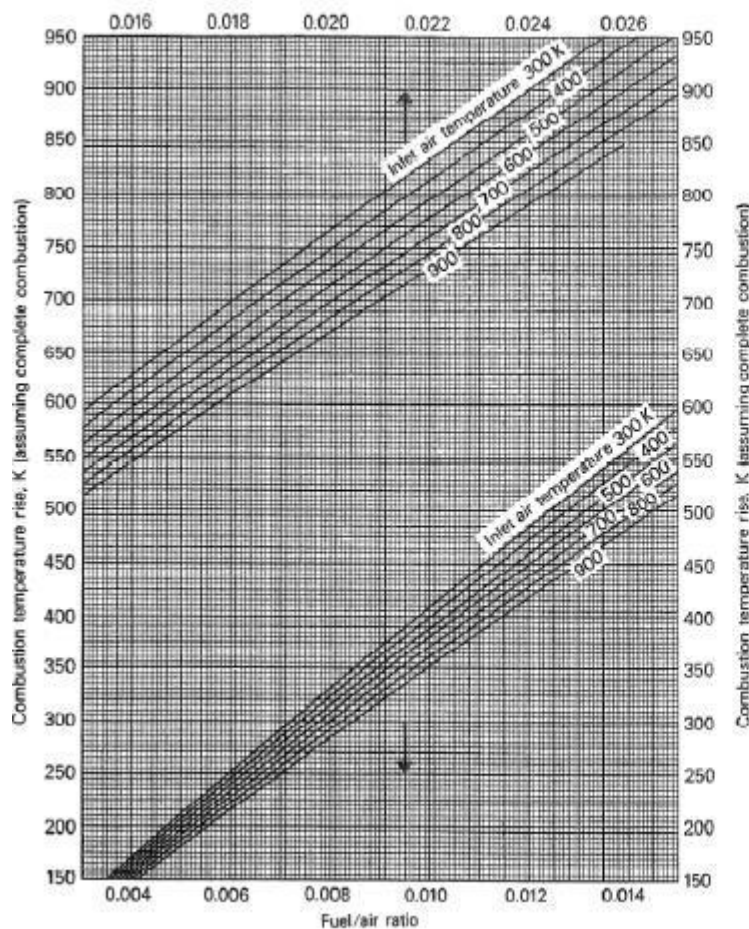
$$T_{02} = 164.7 + 288 = 452.7\text{K}, \text{ and } T_{04} = 1100 - 264.8 = 835.2\text{ K}$$

Hence,

$$T_{05} = 0.80 \cdot 382.5 + 452.7 = 758.7\text{ K}$$

For a combustion chamber inlet air temperature of 759 K and a combustion temperature rise of $(1100 - 759) = 341\text{ K}$, the theoretical fuel/air ratio required is 0.0094 (from the chart of slide 12), and thus

$$f = \frac{\text{theoretical } f}{\eta_b} = \frac{0.0094}{0.98} = 0.0096$$



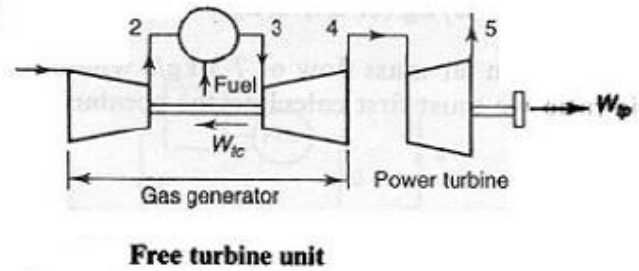
Theoretical Fuel-Air Ratio

Combustion temperature rise vs theoretical fuel-air ratio



Determine the specific work output, specific fuel consumption and cycle efficiency for a simple cycle gas turbine with a free power turbine (see figure) given the following specification:

Compressor pressure ratio	12.0
Turbine inlet temperature	1350 K
Isentropic efficiency of compressor, η_c	0.86
Isentropic efficiency of each turbine, η_t	0.89
Mechanical efficiency of each shaft, η_m	0.99
Combustion efficiency	0.99
Combustion chamber pressure loss	6 % compressor delivery pressure
Exhaust pressure loss	0.03 bar
Ambient conditions p_a, T_a	1 bar, 288 K



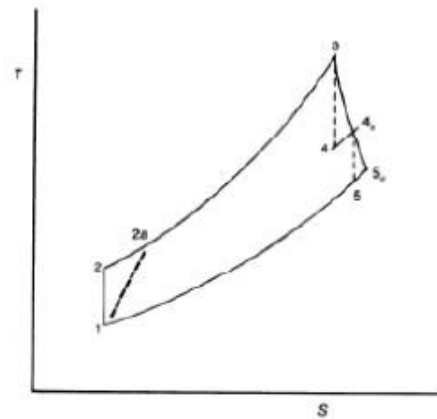
Solution:

Proceeding as in the previous example,

$$T_{02} - T_{01} = \frac{288}{0.86} [12^{1/3.5} - 1] = 346.3 \text{ K}$$

$$W_{tc} = \frac{1.005 * 346.3}{0.99} = 351.5 \text{ kJ/kg}$$

$$p_{03} = 12.0(1 - 0.06) = 11.28 \text{ bar}$$



The intermediate pressure between the two turbines, p_{04} , is unknown, but can be determined from the fact that the compressor turbine produces just sufficient work to drive the compressor. The temperature equivalent of the compressor turbine work is, therefore,

$$T_{03} - T_{04} = \frac{W_{tc}}{c_{pg}} = \frac{351.5}{1.148} = 306.2 \text{ K}$$



The corresponding pressure ratio can be found using the relation

$$T_{03} - T_{04} = \eta_t T_{03} \left[1 - \left(\frac{1}{p_{03}/p_{04}} \right)^{\gamma-1/\gamma} \right]$$

$$306.2 = 0.89 * 1350 \left[1 - \left(\frac{1}{p_{03}/p_{04}} \right)^{0.25} \right]$$

$$\frac{p_{03}}{p_{04}} = 3.243$$

$$T_{04} = 1350 - 306.2 = 1043.8 \text{ K}$$

The pressure at entry to the power turbine, p_{04} , is then found to be

$$p_{04} = \frac{p_{03}}{p_{03}/p_{04}} = 11.28/3.243 = 3.478 \text{ bar}$$

and the power turbine pressure ratio is

$$p_{04}/p_{05} = 3.478/(1+0.03) = 3.377$$

The temperature drop in the power turbine can now be obtained

$$T_{04} - T_{05} = 0.89 * 1043.8 \left[1 - \left(\frac{1}{3.377} \right)^{0.25} \right] = 243.7 \text{ K}$$

and the specific work output, i.e. power turbine work per unit air mass flow, is

$$W_{tp} = c_{pg} (T_{04} - T_{05}) \eta_m$$

$$W_{tp} = 1.148 (243.7) 0.99 = 277.0 \text{ kJ/kg (or kW/kg)}$$

The compressor delivery temperature is $288 + 346.3 = 634.3 \text{ K}$ and the combustion temperature rise is $1350 - 634.3 = 715.7 \text{ K}$

The theoretical fuel/air ratio required is 0.0202 (from the chart in slide 12), giving an actual fuel/air ratio of $0.0202/0.99 = 0.0204$

The SFC and cycle efficiency, η , are then given by

$$SFC = \frac{f}{W_{tp}} = \frac{3600 * 0.0204}{277.9} = 0.265 \text{ kg/kWh}$$

$$\eta = \frac{3600}{0.265 * 43100} = 0.315$$



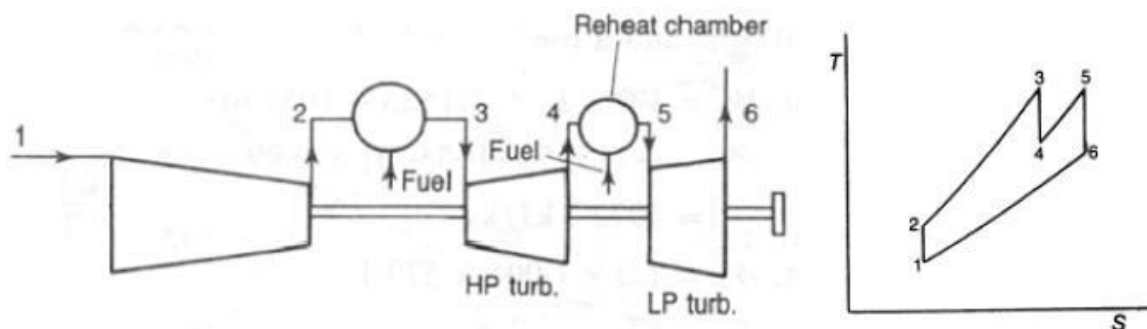
- ❑ The cycle calculations are carried out to determine the overall performance. It However, they also provide information required by other groups such as the aerodynamic and control design groups.
- ❑ *E.g.*, the temperature at entry to the power turbine, T_{04} , may be required as a control parameter to prevent operation above the metallurgical limiting temperature of the compressor turbine.
- ❑ The exhaust gas temperature (EGT), T_{05} , would be important if the gas turbine were to be considered for combined cycle or cogeneration plant.
- ❑ The temperature, $T_{05} = 1043.8 - 243.7 = 800.1\text{K}$ or 527°C , is suitable for use with a waste heat boiler.
- ❑ For a combined cycle plant, a higher TIT might be desirable because there would be a consequential increase in EGT, permitting the use of a higher steam temperature and a more efficient steam cycle.
- ❑ If the cycle pressure ratio were increased to increase the efficiency of the gas cycle, however, the EGT would be decreased resulting in a lower steam cycle efficiency.

Example 3:

Consider the design of a high pressure ratio, single-shaft cycle with reheat at some point in the expansion when used either as a separate unit, or as part of a combined cycle. The power required is 240 MW at 288 K and 1.01 bar

Compressor pressure ratio	30
Polytropic efficiency	
(compressor and turbines)	0.89
Turbine inlet temperature	
(both turbines)	1525 K
$\Delta p/p_{02}$ (1 st combustor)	0.02
$\Delta p/p_{04}$ (2 nd combustor)	0.04
Exhaust pressure	1.02 bar

Solution:



A heat Exchanger is not used because it would result in an exhaust temperature that would be too low for use with a high efficiency steam cycle.

The Assumptions are as follows

- ❑ Let us assume that the mass flow rate is constant throughout, ignoring the effect of substantial cooling bleeds that would be required with high turbine inlet temperatures specified.
- ❑ The reheat pressure is not specified. So, as a starting point we use a value giving equal pressure ratio in each turbine.
- ❑ This division of the expansion leads to equal work in each turbine and a maximum net work output for the ideal reheat cycle).

From polytropic relations:

$$\text{for compression, } \frac{n-1}{n} = \frac{1}{\eta_{\infty c}} \left(\frac{\gamma-1}{\gamma} \right) = \frac{1}{0.89} \left(\frac{0.4}{1.4} \right) = 0.3210$$

$$\text{for expansion, } \frac{n-1}{n} = \eta_{\infty t} \left(\frac{\gamma-1}{\gamma} \right) = 0.89 \left(\frac{0.333}{1.333} \right) = 0.2223$$

Assuming that $p_{01} = p_a$ and $T_{01} = T_a$, we have $T_{02}/T_{01} = (30)^{0.3210}$

$$T_{02} = 858.1 \text{ K}$$

$$T_{02} - T_{01} = 570.1 \text{ K}$$

$$p_{02} = 30 * 1.01 = 30.3 \text{ bar}$$

$$p_{03} = 30.3(1.00 - 0.02) = 29.69 \text{ bar}$$

$$p_{06} = 1.02 \text{ bar, so } p_{03}/p_{06} = 29.11$$

Theoretically, the optimum pressure ratio for each turbine would be

$$\sqrt{(29.11)} = 5.395$$

A pressure loss of 4 % in the reheat combustor has to be considered, so a value of 5.3 for p_{03}/p_{04} could be assumed. Then,

$$\frac{T_{03}}{T_{04}} = (5.3)^{0.2223}$$

$$T_{04} = 1052.6 \text{ K}$$

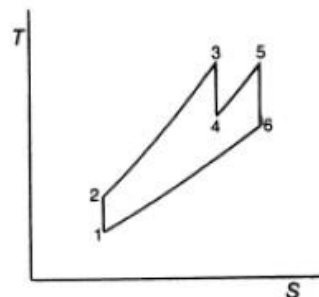
$$p_{04} = 29.69 / 5.3 = 5.602 \text{ bar}$$

$$p_{05} = 5.602(1.00 - 0.04) = 5.378 \text{ bar}$$

$$p_{05}/p_{06} = 5.378/1.02 = 5.272$$

$$\frac{T_{05}}{T_{06}} = (5.272)^{0.2223}$$

$$T_{06} = 1053.8 \text{ K}$$



Assuming unit flow of 1.0 kg/s and a mechanical efficiency of 0.99,

$$\begin{aligned}\text{Turbine output, } W_t &= 1.0 * 1.148 \{ (1525 - 1052.6) + (1525 - 1053.8) \} * 0.99 \\ &= 1072.3 \text{ kJ/kg}\end{aligned}$$

$$\begin{aligned}\text{Compressor input, } W_c &= 1.0 * 1.005 * 570.1 \\ &= 573.0 \text{ kJ/kg}\end{aligned}$$

$$\text{Net work output, } W_N = 1072.3 - 573.0 = 499.3 \text{ kJ/kg}$$

Flow required for 240 MW is given by

$$\begin{aligned}m &= 240000 / 499.3 \\ &= 480.6 \text{ kg/s}\end{aligned}$$

- **For the first combustor**, temperature rise = $1525 - 858 = 667 \text{ K}$, inlet temperature = 858 K and fuel/air ratio = 0.0197 (from the chart of Slide 12)
- **For the second combustor**, temperature rise = $1525 - 1052.6 = 472.4 \text{ K}$, inlet temperature = 1052.6 K and fuel/air ratio = 0.0142 (from the chart of Slide 12)
- Actual total fuel/air ratio $f = \frac{0.0197 + 0.0142}{0.99} = 0.0342$
- And, thermal efficiency $\eta = \frac{499.3}{0.0342 * 43100} = 33.9\%$
- This is a reasonable efficiency for simple cycle operation, and the specific output is excellent.
- However, the turbine exit temperature ($T_{06} = 1053.8 \text{ K}$ or 780.8° C) is too high for efficient use in a combined cycle plant. A reheat steam cycle using conventional steam temperatures of about $550^\circ - 575^\circ \text{ C}$ would require a turbine exit temperature of about 600° C .



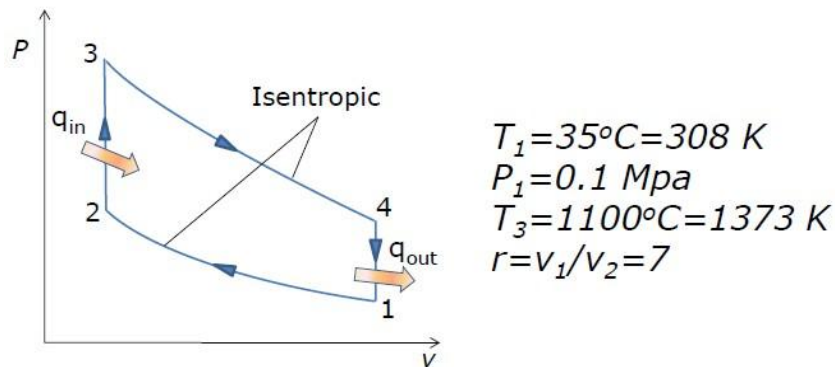
- The turbine exit temperature could be reduced by increasing the reheat pressure, and if the calculations are repeated for a range of reheat pressures, then the results obtained are as shown in the Slide 26.
- It can be seen that a reheat pressure of 13 bar gives an exhaust gas temperature (EGT) of 605°C ; the specific output is about 10 percent lower than the optimum value, but the thermal efficiency is substantially improved to 37.7 per cent. Further increases in reheat pressure would give slightly higher efficiencies, but the EGT would be reduced below 600°C resulting in a less efficient steam cycle.



Problems for practice

In an air standard Otto cycle, the compression ratio is 7 and the compression begins at 35°C and 0.1 MPa. The maximum temperature of the cycle is 1100°C. Find (a) the temperature and the pressure at various points in the cycle, (b) the heat supplied per kg of air, (c) work done per kg of air, (d) the cycle efficiency and (e) the MEP of the cycle.

Solution: Problem 1



- Since process, 1-2 is isentropic,

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2} \right)^\gamma = 7^{1.4} = 15.24$$

- Hence, $P_2 = 1524\text{ kPa}$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2} \right)^{\gamma-1} = 7^{1.4-1} = 2.178$$

- Hence, $T_2 = 670.8\text{ K}$



- For process, 2-3,

$$\frac{P_2 v_2}{T_2} = \frac{P_3 v_3}{T_3}, \therefore P_3 = \frac{T_3}{T_2} P_2 = \frac{1373}{607.8} \times 1524 = 3119.34$$

- $P_3 = 3119.34$ kPa.
- Process 3-4 is again isentropic,

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3} \right)^{\gamma-1} = 7^{1.4-1} = 2.178$$

$$\therefore T_4 = \frac{1373}{2.178} = 630.39 \text{ K}$$

- Hence, $T_2 = 630.39$ K

- Heat input,

$$Q_{in} = c_v (T_3 - T_2)$$

$$= 0.718 (1373 - 670.8)$$

$$= 504.18 \text{ kJ/kg}$$

- Heat rejected,

$$Q_{out} = c_v (T_4 - T_1)$$

$$= 0.718 (630.34 - 308)$$

$$= 231.44 \text{ kJ/kg}$$

- The net work output, $W_{net} = Q_{in} - Q_{out}$

- The net work output,

$$W_{net} = Q_{in} - Q_{out}$$

$$= 272.74 \text{ kJ/kg}$$

- Thermal efficiency, $\eta_{th,otto} = W_{net} / Q_{in}$
- $$= 0.54$$
- $$= 54 \%$$

- Otto cycle thermal efficiency,

$$\eta_{th,otto} = 1 - 1/r^{\gamma-1} = 1 - 1/7^{0.4}$$

$$= 0.54 \text{ or } 54 \%$$

- $v_1 = RT_1 / P_1$

$$= 0.287 \times 308 / 100 = 0.844 \text{ m}^3/\text{kg}$$

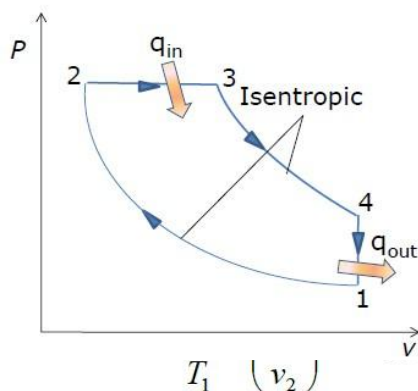
- $MEP = W_{net} / (v_1 - v_2) = 272.74 / v_1 (1 - 1/r)$
- $$= 272.74 / 0.844 (1 - 1/7)$$
- $$= 360 \text{ kPa}$$



Problem 2

In a Diesel cycle, the compression ratio is 15. Compression begins at 0.1 Mpa, 40°C. The heat added is 1.675 MJ/kg. Find (a) the maximum temperature in the cycle, (b) work done per kg of air (c) the cycle efficiency (d) the temperature at the end of the isentropic expansion (e) the cut-off ratio and (f) the MEP of the cycle.

Solution: Problem 2



$$\begin{aligned} T_1 &= 40^\circ\text{C} = 313 \text{ K} \\ P_1 &= 0.1 \text{ Mpa} \\ Q_{in} &= 1675 \text{ MJ/kg} \\ r &= v_1/v_2 = 15 \end{aligned}$$

$$T_2 = 313 \times 2.954 = 924.66 \text{ K}$$

$$Q_{in} = 1675 = 1.005(T_3 - 924.66)$$

$$\therefore T_3 = 2591.33 \text{ K} = T_{\max}$$

- Hence, the maximum temperature is **2591.33 K**

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2} \right)^\gamma = 15^{1.4} = 44.31$$

$$\therefore P_2 = 4431 \text{ kPa}$$

$$\frac{P_2 v_2}{T_2} = \frac{P_3 v_3}{T_3} \rightarrow v_3 = \frac{T_3}{T_2} v_2 = \frac{2591.33}{924.66} \times 0.06 = 0.168 \text{ m}^3/\text{kg}$$

$$r_c = \frac{v_3}{v_2} = \frac{0.168}{0.06} = 2.8$$

- The cut-off ratio is **2.8**.

$$\begin{aligned} T_4 &= T_3 \left(\frac{v_3}{v_4} \right)^{\gamma-1} = 2591.33 \times \left(\frac{0.168}{0.898} \right)^{0.4} \\ &= 1325.37 \text{ K} \end{aligned}$$

$$Q_{out} = c_v(T_4 - T_1) = 0.718(1325.4 - 313) = 726.88 \text{ kJ/kg}$$

$$\begin{aligned} \text{Net work done, } W_{net} &= Q_{in} - Q_{out} = 1675 - 726.88 \\ &= \mathbf{948.12 \text{ kJ/kg}} \end{aligned}$$

- Therefore, thermal efficiency,

$$\begin{aligned} \eta_{th} &= W_{net}/Q_{in} \\ &= 948.12/1675 = \mathbf{0.566 \text{ or } 56.6\%} \end{aligned}$$

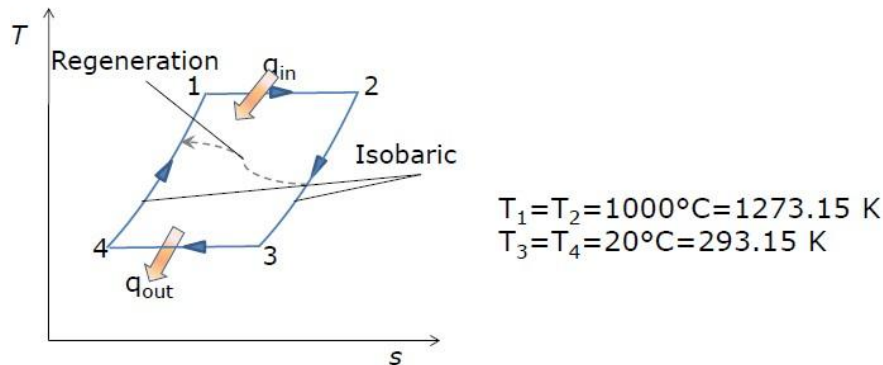
- The cycle efficiency can also be calculated using



Problem 3

An air-standard Ericsson cycle has an ideal regenerator. Heat is supplied at 1000°C and heat is rejected at 20°C. If the heat added is 600 kJ/kg, find the compressor work, the turbine work, and the cycle efficiency.

Solution: Problem 3



Since the regenerator is given as ideal, $-Q_{2-3} = Q_{1-4}$
Also in an Ericsson cycle, the heat is input during the isothermal expansion process, which is the turbine part of the cycle. Hence the turbine work is 600 kJ/kg.

- Thermal efficiency of an Ericsson cycle is equal to the Carnot efficiency.

$$\begin{aligned}\eta_{th} &= \eta_{th, \text{Carnot}} = 1 - T_L / T_H \\ &= 1 - 293.15 / 1273.15 \\ &= 0.7697\end{aligned}$$

- Therefore the net work output is equal to:

$$\begin{aligned}W_{net} &= \eta_{th} Q_H \\ &= 0.7697 \times 600 = 461.82 \text{ kJ/kg}\end{aligned}$$

- The compressor work is equal to the difference between the turbine work and the net work output:

$$\begin{aligned}W_c &= W_t - W_{net} \\ &= 600 - 461.82 = 138.2 \text{ kJ/kg}\end{aligned}$$

- In the Ericsson cycle the heat is rejected isothermally during the compression process. Therefore this compressor work is also equal to the heat rejected during the cycle.

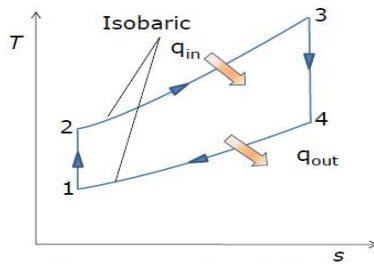
Problem 4

In a Brayton cycle based power plant, the air at the inlet is at 27°C, 0.1 MPa. The pressure ratio is 6.25 and the maximum temperature is 800°C.



Find (a) the compressor work per kg of air (b) the turbine work per kg of air (c) the heat supplied per kg of air, and (d) the cycle efficiency.

Solution: Problem 4



$$\begin{aligned} T_1 &= 27^\circ\text{C} = 300 \text{ K} \\ P_1 &= 100 \text{ kPa} \\ r_p &= 6.25 \\ T_3 &= 800^\circ\text{C} = 1073 \text{ K} \end{aligned}$$

- Since process, 1-2 is isentropic,

$$\frac{T_2}{T_1} = r_p^{(\gamma-1)/\gamma} = 6.25^{(1.4-1)/1.4} = 1.689$$

$$T_2 = 506.69 \text{ K}$$

$$\begin{aligned} W_{comp} &= c_p (T_2 - T_1) = 1.005(506.69 - 300) \\ &= 207.72 \text{ kJ/kg} \end{aligned}$$

- The compressor work per unit kg of air is **207.72 kJ/kg**

- Process 3-4 is also isentropic,

$$\frac{T_3}{T_4} = r_p^{(\gamma-1)/\gamma} = 6.25^{(1.4-1)/1.4} = 1.689$$

$$T_4 = 635.29 \text{ K}$$

$$\begin{aligned} W_{turb} &= c_p (T_3 - T_4) = 1.005(1073 - 635.29) \\ &= 439.89 \text{ kJ/kg} \end{aligned}$$

- The turbine work per unit kg of air is **439.89 kJ/kg**
- Heat input, Q_{in} ,

$$\begin{aligned} Q_{in} &= c_p (T_3 - T_2) = 1.005(1073 - 506.69) \\ &= 569.14 \text{ kJ/kg} \end{aligned}$$

- Heat input per kg of air is **569.14 kJ/kg**
- Cycle efficiency,

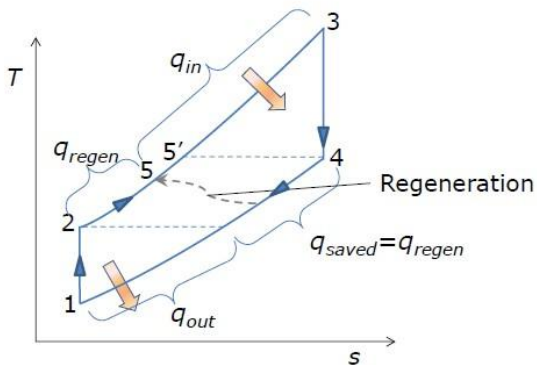
$$\begin{aligned} \eta_{th} &= (W_{turb} - W_{comp}) / Q_{in} \\ &= (439.89 - 207.72) / 569.14 \\ &= 0.408 \text{ or } 40.8\% \end{aligned}$$

Problem 5

- Solve Problem 3 if a regenerator of 75% effectiveness is added to the plant.



Solution: Problem 5



$$\varepsilon = \frac{T_5 - T_2}{T_4 - T_2} = 0.75$$

$$\text{or, } \frac{T_5 - 506.69}{635.29 - 506.69} = 0.75$$

$$T_5 = 603.14 \text{ K}$$

- T_4 , W_{comp} , W_{turb} remain unchanged
- The new heat input, $Q_{in} = c_p(T_3 - T_5)$
 $= 472.2 \text{ kJ/kg}$
- Therefore $\eta_{th} = (W_{turb} - W_{comp}) / Q_{in}$
 $= (439.89 - 207.72) / 472.2$
 $= 0.492 \text{ or } 49.2 \%$





INDUSTRIAL APPLICATIONS



Industrial Applications

The following are the applications of a gas turbine:

1. They are used to propel air-crafts and ships,
2. Gas turbine plants are used as standby plants for the hydroelectric power plants.
3. Gas turbine power plants may be used as peak loads plant and standby plants for smaller power units.
4. The shaft can be connected to other machinery to do various types of work such as: turning a helicopter rotor, running a compressor (which "crushes" a gas to a condensed form for use in industrial applications) or generating electric power.
5. The gas turbine is useful to our modern world because it is relatively compact in size and makes a lot of power. Gas turbines are used in backup power systems in Manhattan for example, when the grid goes down due to natural disaster, gas turbines power up and can produce power for emergency uses.
6. Gas turbines are used on oil platforms to make power. The oil platform is like a small city, isolated out on the water, so it requires a lot of power and does not have a lot of space. Gas turbines are also used in oil refineries to make power for the cracking process.





TUTORIAL QUESTIONS



UNIT – IV

Theory Questions:

1. Explain about the open cycle and closed cycle turbines with neat sketches and also draw P-V & T-S diagrams.
2. State the merits of gas turbines over IC engines.
3. Draw the gas turbine power plant with inter cooling
4. List out the advantages of open cycle gas turbine over closed cycle gas turbine.
5. List different applications of gas turbine power cycles in power sector industries.
6. Discuss the relative advantages and disadvantages of gas turbines and steam turbines.
7. What are the different methods to improve the efficiency of gas turbines?
8. What are the different types of combustion chambers in gas turbine engines?
Explain them in detail with relevant sketches.
9. Draw the schematic diagram of closed cycle gas turbine and explain its working.
10. Explain the operating principle of Brayton cycle with a schematic diagram p-v and T-s diagrams.
11. Why Re-heater is necessary in gas turbine? What are its effects
12. What are the requirements of a good combustion chamber for a gas turbine?
13. Explain with neat sketch the gas turbine cycles with intercooling and reheating and what will be the condition of maximum output

Numerical Problems:

1. A simple gas turbine cycle works with a pressure ratio of 8. The compressor and turbine inlet temperatures are 300 K and 800 K respectively. If the volume flow rate of air is 250 m³/s, compute the power output and thermal efficiency
2. A constant pressure open cycle gas turbine plant works between temperature range of 15°C and 700°C and pressure ratio of 6. Find the mass of air circulating in the installation, if it develops 1100 kW. Also find the heat supplied by the heating chamber.





ASSIGNMENT QUESTIONS



3. In a gas turbine plant, air is drawn at 1 bar, 150 C and the pressure ratio is 6. The expansion takes place in two turbines. The efficiency of compressor is 0.82, high pressure turbine is 0.85 and low pressure turbine is 0.84. The maximum cycle temperature is 6250 C. Calculate
 - i) Pressure and temperature of gases entering the low pressure turbine.
 - ii) Net power developed
 - iii) Work ratio
 - iv) Thermal efficiency. Work output of high pressure turbine is equal to compressor work
4. In an air standard regenerative gas turbine cycle the pressure ratio is 5. Air enters the compressor at 1 bar, 300 K and leaves at 490 K. The maximum temperature in the cycle is 1000 K. Calculate the cycle efficiency, given that the efficiency of regenerator and the adiabatic efficiency of the turbine are each 80%. Assume for air, the ratio of specific heats is 1.4. Also show the cycle on T-S diagram.
5. A gas turbine unit receives air at 1 bar and 300 K and compresses it adiabatically to 6.2 bar. The compressor efficiency is 88%. The fuel has a heating value of 44186 KJ/kg and the fuel air ratio is 0.017 KJ/kg of air. The turbine efficiency is 90 %. Calculate the work of turbine and compressor per kg of air compressed and thermal efficiency. Take $C_p=1.005$ KJ/kg K, $\gamma=1.4$ for the compression process, $C_p=1.147$ KJ/kg K, $\gamma=1.33$ for the expansion process.

ASSIGNMENT QUESTIONS

1. a) Describe with neat sketches the working of a simple constant pressure open cycle gas turbine.
b) Discuss the relative advantages and disadvantages of gas turbines and steam turbines.
2. Describe with neat diagram a closed cycle gas turbine and explain advantages, disadvantages and applications.
3. Explain with neat sketch the gas turbine cycles with intercooling and reheating and what will be the condition of maximum output.
4. Explain about the open cycle and closed cycle turbines with neat sketches and also draw P-V & T-S diagrams.

