APPLIED THERMODYNAMICS

Gas Turbine Engines (Module IV)

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List of Topics

1. Gas Turbine Engine – Components and Thermal Circuit Arrangement
2. Gas Turbine Performance Cycle – I
3. Gas Turbine Performance Cycle – II
4. Real Gas Turbine Performance Cycle
5. Aircraft Propulsion Cycle – I
6. Aircraft Propulsion Cycle – II

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Lecture 3

Gas Turbine Performance Cycle – II

- ➢ Modifications to Basic Cycle
- ➢ Intercooling Cycle
- ➢ Reheat Cycle
- ➢ Heat Exchange Cycle
- ➢ Combined Performance

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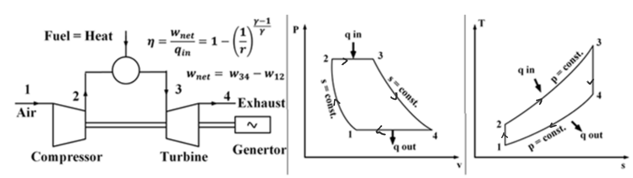
Modifications to Basic Cycle

- The ideal cycle for the simple gas turbine is the "Joule (or Brayton)" cycle.
  - ➢ Process 1-2: isentropic compression
  - ➢ Process 2-3: constant pressure heat addition
  - ➢ Process 3-4: isentropic expansion
  - ➢ Process 4-1: constant pressure heat addition
- The work-ratio and cycle efficiency are low. They can be improved by increasing the isentropic efficiencies of compressor and turbine with proper blade designs during its manufacturing process.



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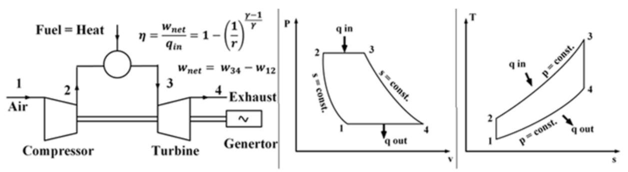
Modifications to Basic Cycle

- In a practical cycle with irreversibility during compression and expansion process, the cycle efficiency depends on maximum cycle temperatures and pressure ratio.
- The cycle efficiency and specific power output can be plotted against various pressure ratios and different values maximum temperature.
- At any fixed maximum cycle temperature, there is a value of pressure ratio that gives maximum cycle efficiency. The net work out put also depends on pressure ratio and maximum cycle temperature.



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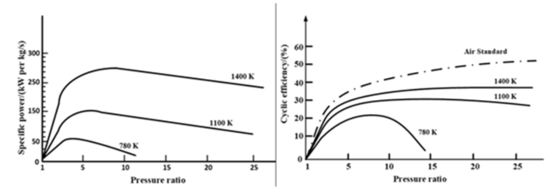
Modifications to Basic Cycle

- The cycle efficiency reaches a maximum, at a different value of pressure ratio than the work output. Hence, the choice of pressure ratio is made as compromise between them.
- The maximum cycle temperature is limited due to metallurgical considerations. The blades of the turbine are under mechanical stress and the temperature of blade material should be kept to a safe working value.
- A suitable means to increase the maximum cycle temperature is to provide blade cooling mechanism/expensive alloy material up to allowable temperature of 1600 K.



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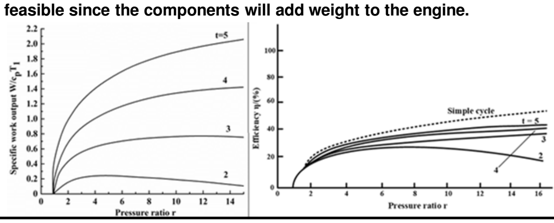
Modifications to Basic Cycle

- It is important to have high work ratio with adoptable methods as follows:
  - ➢ Intercooling between compressor stages
  - ➢ Reheating between turbine stages
  - ➢ Intercooling and reheating (but cycle efficiency may drop)
  - ➢ Intercooling and reheating in conjunction with a "heat exchanger"
- In aircraft practice, life expectancy of the engine is shorter and hence they have maximum temperatures higher than industrial gas turbine units.
- The aircraft cycles are thrust specific, the concepts of high work ratio is not feasible since the components will add weight to the engine.



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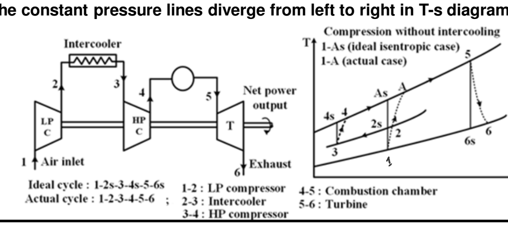
Intercooling Cycle

- At a given pressure ratio and mass flow rate in a gas turbine cycle, if the compression is performed in two stages, then work input is reduced.
- An intercooler is introduced between the stages for cooling the air to inlet temperature.
- It can be seen that the work input with intercooling is less than the work input with no intercooling when it is assumed that isentropic efficiencies of two compressors (operating separately) are each equal to isentropic efficiency of single compressor if no intercooling is incorporated. It is due to the fact that the constant pressure lines diverge from left to right in T-s diagram.



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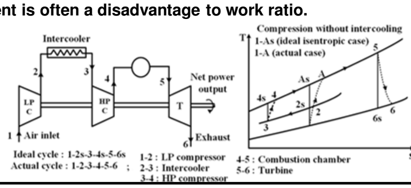
Intercooling Cycle

- The best intercooling pressure is the one which gives equal pressure ratio at each stage of compression. The work input is a minimum when the pressure ratio in each stage is same and the temperature of air is cooled back to its inlet value.
- Since, the compressor work input is reduced, the work ratio is increased.
- The heat input requirement is increased when intercooling is used.
- Although net work out put increases by intercooling, but it is found that cycle efficiency drops in most cases due to increase in heat supply. This disadvantage is offset when a heat exchanger is used. This additional bulk arrangement is often a disadvantage to work ratio.



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Intercooling Cycle

With intercooling: $w_{in-i} = w_{12} + w_{34} = c_p(T_2 - T_1) + c_p(T_4 - T_3)$

Without intercooling: $w_{in} = c_p(T_A - T_1) = c_p(T_2 - T_1) + c_p(T_A - T_2)$

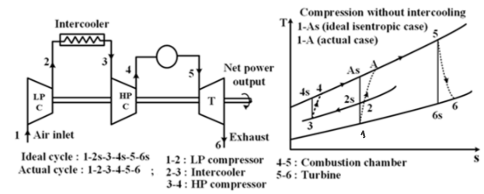
$c_p(T_4 - T_3) < c_p(T_A - T_2) \;\Rightarrow w_{in-i} < w_{in}$

Minimum work with intercooling $(w_{in-i} = w_{\min})$: $\dfrac{p_2}{p_1} = \dfrac{p_4}{p_3}$ & $T_3 = T_1$

With intercooling: $q_{in-i} = c_p(T_5 - T_4)$; No intercooling: $q_{in} = c_p(T_5 - T_A)$

$c_p(T_5 - T_4) > c_p(T_5 - T_A) \;\Rightarrow q_{in-i} > q_{in}$

Work ratio: $WR = \dfrac{w_{net}}{w_{gross}} = \dfrac{w_t - w_c}{w_t}$; Cycle efficiency: $\eta = \dfrac{w_{net}}{q_{in}}$



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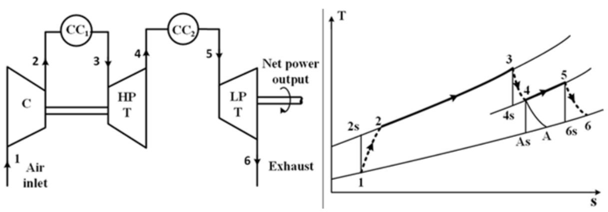
Reheat Cycle

- The expansion process is frequently preformed in two separate turbine stages – high pressure (HP) turbine driving the compressor and low pressure (LP) turbine providing useful power output.
- The work output of LP turbine can be increased by raising the temperature at inlet stage by a second combustion chamber. The gases leaving HP turbine is heated in secondary combustion chamber, placed between turbine stages.
- Neglecting mechanical losses, the work output of HP turbine is exactly equal to work input required for the compressor.



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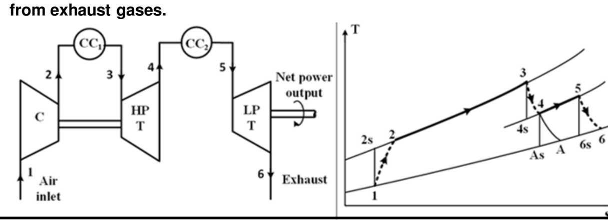
Reheat Cycle

- The constant pressure lines in *T-s* diagram diverges towards right, thereby net work output increases by reheating.
- The work of expansion is increased and work of compression is unchanged, thereby "work ratio" is higher for reheat.
- But, additional heat is supplied in the combustion chamber. The combined effect is to reduce thermal efficiency.
- The exhaust temperature of gas leaving LP turbine is much higher when reheat is used. Hence, the heat exchanger can be used to enable some energy from exhaust gases.



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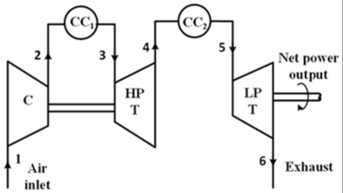
Reheat Cycle

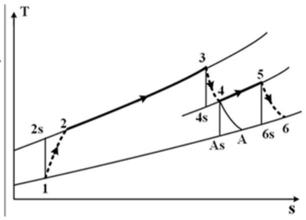
LP turbine with reheating: $w_{net-r} = c_{pg}(T_5 - T_6)$; LP turbine with no reheating: $w_{net} = c_{pg}(T_4 - T_A)$

HP turbine: $w_t = w_c \Rightarrow c_{pa}(T_2 - T_1) = c_{pg}(T_3 - T_4)$; $c_{pg}(T_5 - T_6) > c_{pg}(T_4 - T_A) \Rightarrow w_{net-r} > w_{net}$

$q_{in-r} = c_{pg}(T_3 - T_2) + c_{pg}(T_5 - T_4)$; $q_{in} = c_{pg}(T_3 - T_2)$

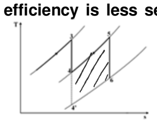
Work ratio: $WR = \frac{w_{net-r}}{w_{gross}}$ $(w_{net-r} > w_{net})$; Cycle efficiency: $\eta = \frac{w_{net-r}}{q_{in-r}}$ $(q_{in-r} > q_{in})$

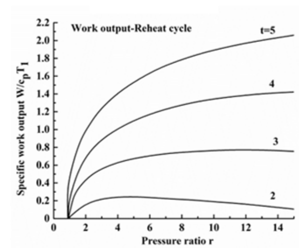


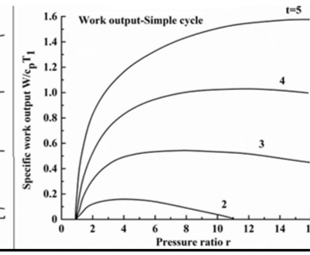


Reheat Cycle

- A comparison of work output shows that reheat marked increases the specific output but at the expense of efficiency.
- One extra cycle (4'-4-5-6) is added to the simple cycle that operates between a smaller temperature range. But the reduction in efficiency is less severe as the maximum cycle temperature is increases.







Reheat Cycle

- A substantial increase in specific work output can be obtained by splitting the expansion and reheating the gas between high and low pressure turbines.
- Referring to the figure, the turbine work increases since the vertical distance increases with increase in entropy on a T-s diagram.
- The cycle efficiency drops with addition of extra heat in the cycle.
- Assuming that the gas is reheated to a temperature T3, the optimum point for specific work output is the location for which the temperature drops and work transfers for HP and LP turbines are equal.

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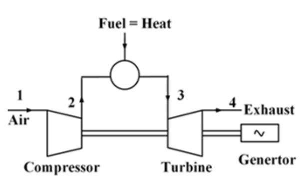
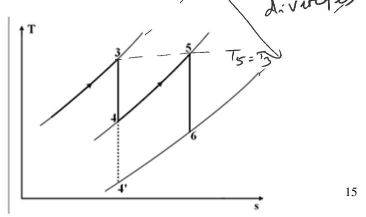




Reheat Cycle

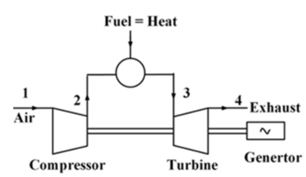
- Accordingly, the specific work output and efficiency can be expressed as follows:

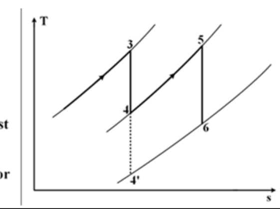
$$(T_3 - T_4) + (T_5 - T_6) > (T_3 - T_4'); \; c = r^{\frac{\gamma-1}{\gamma}}; \; t = \frac{T_3}{T_1}; \; r = \frac{T_3}{T_1}$$

Subscripts – s: simple cycle; h: heat exchange cycle; r: reheat cycle

$$\left(\frac{W}{c_p T_1}\right)_r = 2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1); \; \left(\frac{W}{c_p T_1}\right)_s = t\left(1 - \frac{1}{c}\right) - (c-1) = \left(\frac{W}{c_p T_1}\right)_h$$

$$\eta_r = \frac{2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1)}{2t\left(1 - \frac{1}{2\sqrt{c}}\right) - c}; \; \eta_s = \frac{c-1}{c}; \; \eta_h = \frac{t-c}{t}; \; \left(\frac{W}{c_p T_1}\right)_r > \left(\frac{W}{c_p T_1}\right)_s; \; \eta_r < \eta_s < \eta_h$$

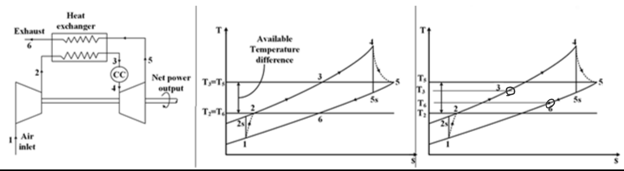




Heat Exchange Cycle

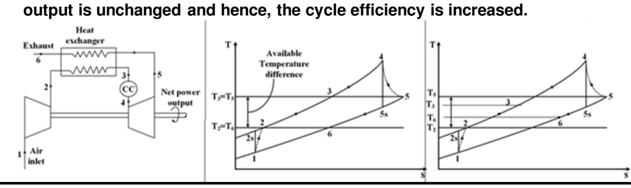
Heat exchanger:

- The exhaust gases leaving the turbine at the end of expansion are still at high temperature and enthalpy.
- Some of the available heat energy from the gas can be recovered by passing them through a heat exchanger where the heat transfer from the gas can be used to heat the air leaving from compressor.
- In ideal scenario, the air from compressor would be heated from T_2 to $T_3 = T_5$ and the gas from the turbine exit is cooled from T_5 to $T_6 = T_2$.
- it is impossible because the a finite temperature difference is required at all the points in the heat exchanger to overcome the resistance to heat transfer.



Heat Exchange Cycle

- The required temperature difference between the gases and the air entering the heat exchanger is (T_6-T_2) while the required temperature difference between the gases and the air leaving the heat exchanger is (T_5-T_3).
- If no heat is lost from heat exchanger to the atmosphere, then the heat given up by the gases is exactly equal to heat taken up by the air.
- The assumption of "no heat loss" from the heat exchanger is sufficiently accurate in most of the cases irrespective of temperatures T_3 and T_6.
- When a heat exchanger is used, the heat supplied in the combustion chamber is reduced assuming maximum cycle temperature is unchanged. The net work output is unchanged and hence, the cycle efficiency is increased.



Heat Exchange Cycle

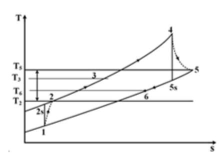
- The heat exchanger "effectiveness" is defined to allow for temperature difference necessary for transfer of heat i.e. the ratio of heat received by the air to the maximum possible heat which could be transferred from the gases in the heat exchanger.
- Another assessment parameter of heat exchanger is the "thermal ratio" – defined as the ratio of temperature rise of the air to the maximum available temperature difference.
- The "thermal ratio" and "effectiveness" are equal when thermal products of gases have same value of that of air.

$$\text{Heat balance}: \dot{m}_a c_{pa}(T_3 - T_2) = \dot{m}_g c_{pg}(T_5 - T_6)$$

$$\text{Heat supplied by the fuel (without heat exchanger)} = c_{pg}(T_4 - T_2)$$

$$\text{Heat supplied by the fuel (with heat exchanger)} = c_{pg}(T_4 - T_3)$$

$$\text{Effectiveness}: \varepsilon = \frac{\dot{m}_a c_{pa}(T_3 - T_2)}{\dot{m}_g c_{pg}(T_5 - T_2)}; \ \text{Thermal ratio}: TR = \frac{T_3 - T_2}{T_5 - T_2}$$



Heat Exchange Cycle

- A heat exchanger can be used only if there is sufficiently large temperature difference between gases leaving the turbine and air leaving the compressor. Otherwise it will incur additional capital cost and maintenance requirement.
- Another practice is to make large surface areas to achieve high value of thermal ratio when the temperature difference is small.
- The choice of large gas turbine units in recent days is generate steam/hot water by recovering heat from exhaust gases from the turbine.

$$(T_3 - T_4) + (T_5 - T_6) > (T_3 - T_4'); \ c = r^{\frac{\gamma-1}{\gamma}}; \ t = \frac{T_3}{T_1}; \ r = \frac{T_3}{T_1}$$

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$$\left(\frac{W}{c_p T_1}\right)_r = 2t\left(1 - \frac{1}{\sqrt{c}}\right) - (c-1); \ \left(\frac{W}{c_p T_1}\right)_s = t\left(1 - \frac{1}{c}\right) - (c-1) = \left(\frac{W}{c_p T_1}\right)_h$$

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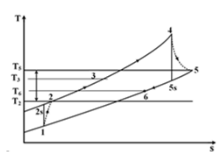
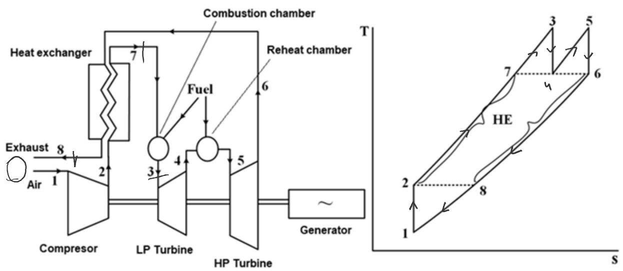


Combined Performance

Cycle with Reheat and Heat-exchange:

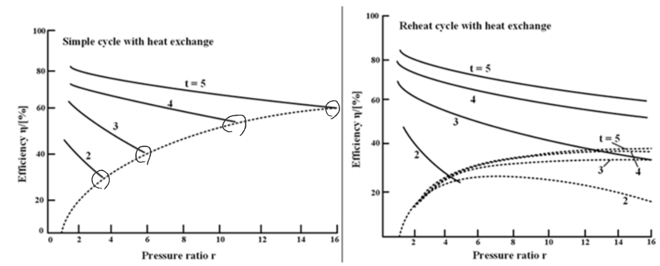
- Reduction in efficiency due to reheat can be overcome by adding heat exchange.
- The higher exhaust gas temperature is fully utilized in the heat exchanger and the increase in work output is no longer offset by heat supplied.



Combined Performance

Cycle with Reheat and Heat-exchange:

- The family of constant temperature lines have same features as those for simple cycles with heat-exchange.
- Each curve having a Carnot value (at r = 1), falls with increase in pressure ratio to meet the corresponding curve of reheat cycle without heat exchange at the value of 'r' corresponding to maximum specific work output.



Combined Performance

Summary:

- Improvement in specific work output through intercooling is seldom contemplated in practice because they are bulky and need large quantities of cooling water.
- The modifications to low temperature region in a cycle normally less significant than a comparable modifications to high temperature region. For example, the reheat and intercooling increases the cycle efficiency when a heat exchanger is incorporated.
- The choice of pressure ratio depends on whether high efficiency or high specific work output (i.e. small size) is desired.
- For the cycles without heat exchange, a higher pressure ratio should be used to take the advantage of higher permissible turbine inlet temperature.
- All these inferences are true for all practical cycles that involves component losses into the account.

Numerical Problems

Q1. A gas turbine unit produces 4 MW power while operating at pressure ratio of 9 by involving two compressors with intercooling between stages. A HP turbine is used to drive the compressor and the LP turbine drives the generator. The temperature of the gas at the entry to the HP turbine is 650°C and the gases are further reheated to same temperature. The exhaust gases leaving the LP turbine are passed through a heat exchanger to heat the air leaving HP compressor. The compressors have equal pressure ratio and the intercooling is complete between the stages. The air inlet temperature is 15°C. The isentropic efficiency of each compressor stage is 0.8 and that of turbine is 0.85. The heat exchanger has thermal ratio of 0.75. Calculate, (a) cycle efficiency; (b) work ratio; (c) mass flow rate.

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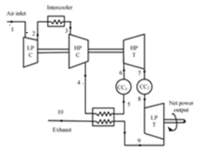

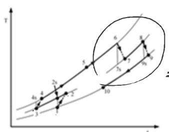

Step calculate temp.

$\frac{p_2}{p_1} = \sqrt{9} = 3.$ $\quad \gamma = 1.4$

$1-2 \quad \frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{2s} = 394\text{ K}, \quad T_1 = 288$

$\eta_c = \frac{T_{2s} - T_2}{T_2 - T_1} = 0.8 \Rightarrow T_2 = 420.5\text{ K}.$

$W_c = c_p(T_2 - T_1) = 1.005(420.5 - 288) = 133.1\text{ kJ}$

$W_T = 2W_c = 266.2\text{ kJ/kg}.$

$C_{pg} = 1.15\text{ kJ/kg}.$

$T_6 = 650^\circ C = 923\text{ K}.$

$C_{pg}(T_6 - T_7) = 266.2 \Rightarrow T_7 = 691.5\text{ K}.$

$\eta_T = \frac{T_6 - T_7}{T_6 - T_{7s}} = 0.85 \Rightarrow T_{7s} = 650.7\text{ K}.$

Numerical Problems

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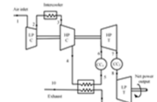

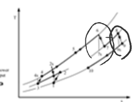

pressure ratio, $\frac{p_6}{p_7} = \left(\frac{T_6}{T_{7s}}\right)^{\frac{\gamma}{\gamma-1}}$ $\quad \gamma = 1.33$ (hot gas).

$T_6 = 923\text{ K}.$ $\quad T_{7s} = 650.7\text{ K}.$

$\Rightarrow \frac{p_6}{p_7} = 4.09.$

$\frac{p_8}{p_9} = \frac{9}{4.09} = 2.2.$

$\frac{T_8}{T_{9s}} = \left(\frac{p_8}{p_9}\right)^{\frac{\gamma-1}{\gamma}} \Rightarrow T_{9s} = 762.8\text{ K}.$

$T_8 = T_6 = 923\text{ K}.$

$\eta_T = \frac{T_8 - T_9}{T_8 - T_{9s}} = 0.85$

$\Rightarrow T_9 = 786.8\text{ K}.$

$(W_T)_{LP} = C_{pg}(T_8 - T_9) = 156.63\text{ kJ/kg}$

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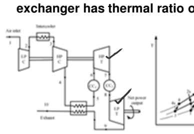
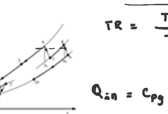
Q1. A gas turbine unit produces 4 MW power while operating at pressure ratio of 9 by involving two compressors with intercooling between stages. A HP turbine is used to drive the compressor and the LP turbine drives the generator. The temperature of the gas at the entry to the HP turbine is 650°C and the gases are further reheated to same temperature. The exhaust gases leaving the LP turbine are passed through a heat exchanger to heat the air leaving HP compressor. The compressors have equal pressure ratio and the intercooling is complete between the stages. The air inlet temperature is 15°C. The isentropic efficiency of each compressor stage is 0.8 and that of turbine is 0.85. The heat exchanger has thermal ratio of 0.75. Calculate, (a) cycle efficiency; (b) work ratio; (c) mass flow rate.





$TR = \frac{T_5 - T_4}{T_9 - T_4} = 0.85 \Rightarrow T_5 = 731.8\text{ K}.$

$T_9 = 786.8\text{ K}. \quad T_4 = T_2 = 420.5\text{ K}.$

$Q_{in} = C_{pg}(T_6 - T_5) + C_{pg}(T_8 - T_7)$

$\Rightarrow Q_{in} = 486.1\text{ kJ/kg}.$

$T_6 = 923\text{ K}.$ $\quad T_8 = 923\text{ K}.$ $\quad T_7 = 691.5\text{ K}.$

$(W_t)_{LP} = 156.43\text{ kJ/kg}.$

$(W_t)_{HP} = 266.2\text{ kJ/kg}.$

(a) $\eta = \frac{W_{net}}{Q_{in}} = \frac{156.43}{486.1} = 0.322 = 32.2\%$

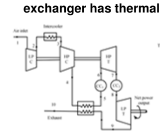
(b) $WR = \frac{W_{net}}{W_t} = \frac{156.43}{422.83}$

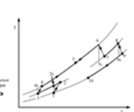
$WR = 0.3635$

(c) $\dot{m} = \frac{4000\text{ kJ/s}}{156.43\text{ kJ/kg}} = 25.5\text{ kg/s}$

Numerical Problems

Q1. A gas turbine unit produces 4 MW power while operating at pressure ratio of 9 by involving two compressors with intercooling between stages. A HP turbine is used to drive the compressor and the LP turbine drives the generator. The temperature of the gas at the entry to the HP turbine is 650°C and the gases are further reheated to same temperature. The exhaust gases leaving the LP turbine are passed through a heat exchanger to heat the air leaving HP compressor. The compressors have equal pressure ratio and the intercooling is complete between the stages. The air inlet temperature is 15°C. The isentropic efficiency of each compressor stage is 0.8 and that of turbine is 0.85. The heat exchanger has thermal ratio of 0.75. Calculate, (a) cycle efficiency; (b) work ratio; (c) mass flow rate.





THANK YOU

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