APPLIED THERMODYNAMICS

Gas Turbine Engines
(Module IV)

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List of Topics

1. Gas Turbine Engine – Components and Thermal Circuit Arrangement
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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 2

Gas Turbine Performance Cycle – I

- A Practical Gas Turbine Cycle
- Ideal Gas Turbine Cycle
- Thermodynamic Analysis
- Specific Work Output
- Heat Exchange Cycle

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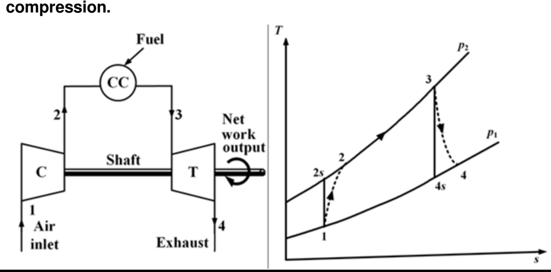
A Practical Gas Turbine Cycle

- The most basic gas turbine unit operating on the open cycle has a rotary compressor and a rotary turbine are mounted on a common shaft.
- The use of constant pressure combustion with rotary compressor driven by a rotary turbine mounted on a common shaft gives a combination which is ideal for steady mass flow rate over wide operating range.
- Air is drawn into compressor and is fed to a combustion chamber after compression.



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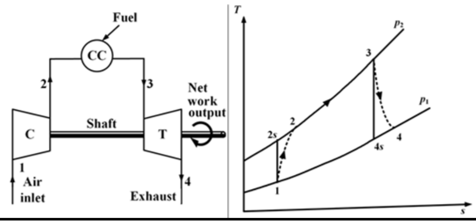
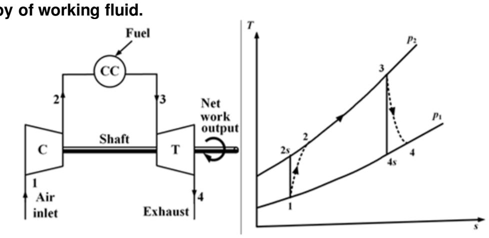
A Practical Gas Turbine Cycle

- Energy is supplied in the combustion chamber by spraying fuel into airstream and the resulting gases expand through the turbine to atmosphere.
- In order to achieve net work output from the unit, the turbine must develop more gross output than is required to drive the compressor and overcome losses in the drive.
- The compressor is either a centrifugal or axial-flow type and the compression process is irreversible but approximately adiabatic.
- The expansion in turbines (radial flow or axial-flow type) is irreversible but adiabatic.



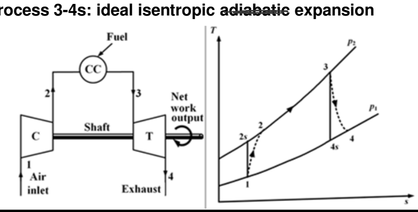
A Practical Gas Turbine Cycle

- Due to irreversibilities, more work is required in compression process and less work is developed in the turbine for a given pressure ratio.
- The open cycle gas turbine does not replicate an ideal constant pressure cycle. The actual cycle involves chemical reaction in the combustion chamber resulting high temperature products (chemically different from reactants).
- During combustion, there is no energy exchange to the surroundings. There is gradual decrease in chemical energy with corresponding increase in enthalpy of working fluid.



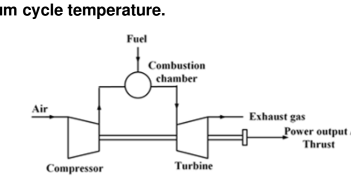
A Practical Gas Turbine Cycle

- The combustion is equivalent to heat transfer to the working fluid (having constant specific heat) at constant pressure.
- This approach allows actual process to be compared with ideal one on a T-s diagram by neglecting pressure loss in the combustion chamber
  - Process 1-2: irreversible adiabatic compression
  - Process 2-3: constant pressure heat supply
  - Process 3-4: irreversible adiabatic expansion
  - Process 1-2s: ideal isentropic ~~adiabatic~~ compression
  - Process 3-4s: ideal isentropic ~~adiabatic~~ expansion



Ideal Gas Turbine Cycle

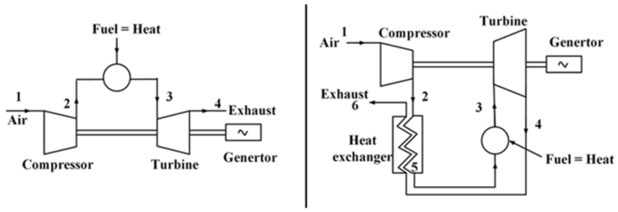
- Many versatile possible combinations of gas turbine cycles can be realized by considering multi-stage compression, expansion, heat exchange, reheat and intercooling. They lead to large number of performance curves.
- While calculating cycle performances, two broad groups are considered – "Shaft power cycle (land/marine based power plants)" and "Aircraft propulsion cycles (forward speed and altitude dependent)".
- It is very much essential to review the performance of ideal gas turbine cycles in which perfections of individual components are assumed.
- The specific work output and cycle efficiency depends on pressure ratio and maximum cycle temperature.



Ideal Gas Turbine Cycle

Assumptions:

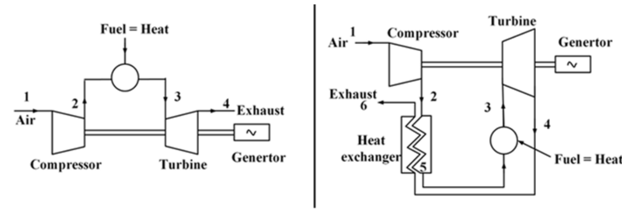
- **Compression and expansion processes are reversible and adiabatic i.e. isentropic.**
- **The change in kinetic energy of the working fluid between inlet and outlet of each component is negligible.**
- **There are no pressure losses in the inlet ducting, combustion chambers, heat-exchangers, intercoolers, exhaust ducting and connecting components.**



Ideal Gas Turbine Cycle

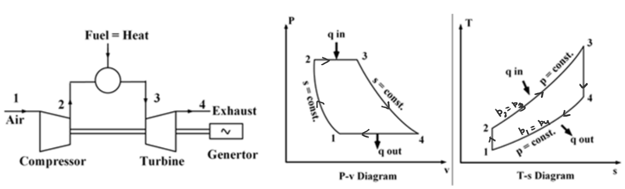
Assumptions:

- **The working fluid has same composition throughout the cycle and is a perfect gas with constant specific heats.**
- **The mass flow rate of the gas is constant throughout the cycle.**
- **The heat transfer in the heat exchanger (mainly counter-flow type) is complete so that the temperature rise in cold side is the maximum and exactly equal to temperature drop on the hot side.**



Thermodynamic Analysis

- **The assumptions of ideal gas turbine cycle imply that the combustion chamber (in which the fuel is introduced and burnt), is replaced by a heater with external heat source. It makes no difference as far as calculations of performance cycle either in open or closed loop.**
- **The ideal cycle for the simple gas turbine is the "Joule (or Brayton)" cycle (1-2-3-4).**
- **The efficiency of ideal cycle increases with increase in pressure ratio.**



Thermodynamic Analysis

Steady flow energy equation: $q=(h_2-h_1)+\frac{1}{2}\left(C_2^2-C_1^2\right)+w$

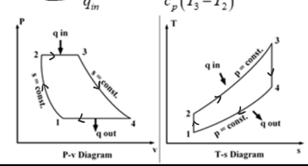
Pressure ratio: $r=\frac{p_2}{p_1}$; Temperature ratio: $t=\frac{T_3}{T_1}$; $p_3=p_2$ & $p_4=p_1$

Combustion chamber: $q_{23}=(h_3-h_2)=c_p(T_3-T_2)$

Compressor: $w_{12}=-(h_2-h_1)=-c_p(T_2-T_1)$; $\frac{T_2}{T_1}=\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}=r^{\frac{\gamma-1}{\gamma}}$

Turbine: $w_{34}=(h_3-h_4)=c_p(T_3-T_4)$; $\frac{T_3}{T_4}=\left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}=r^{\frac{\gamma-1}{\gamma}}$

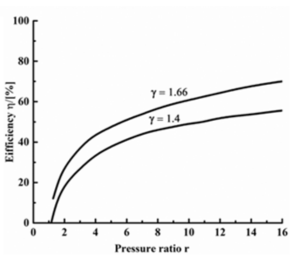
Cycle efficiency: $\eta=\frac{w_{net}}{q_{in}}=\frac{c_p(T_3-T_4)-c_p(T_2-T_1)}{c_p(T_3-T_2)}=1-\left(\frac{1}{r}\right)^{\frac{\gamma-1}{\gamma}}$

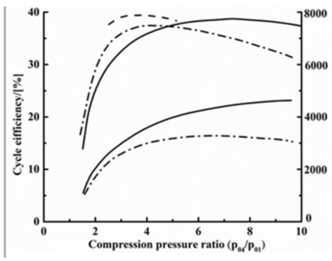


Thermodynamic Analysis

Inferences:

- The cycle efficiency depends on the pressure ratio and nature of the working fluid. It increases for higher pressure ratio.
- The cycle efficiency is higher for monoatomic gas (e.g. helium w.r.t. air).
- A realistic curve, suggest slightly lower efficiency for helium, when the component losses are included. So, there is no theoretical advantage for helium as working fluid.

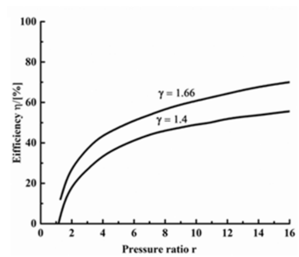


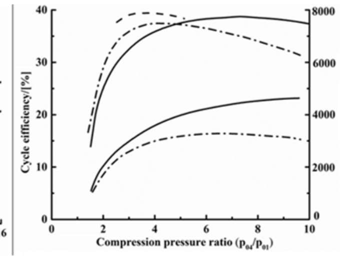


Thermodynamic Analysis

Inferences

- Allowing variation of "c_p and γ" with temperature, air leads to almost identical efficiency. Since, the variation of "c_p and γ" with temperature is not significant over large domain, the efficiency curve for helium drops with pressure ratio with component losses even though it has better heat transfer characteristics.





Specific Work Output

- Thermal efficiency and specific work out are equally important for gas turbine plants.
- The size of the plant for a given power depends on pressure ratio and the maximum cycle temperature. A non-dimensional expression is used as "*specific work output*".

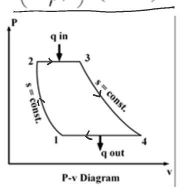
Dimensional

$$w = c_p\left(T_3 - T_4\right) - c_p\left(T_2 - T_1\right);\ \frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = r^{\frac{\gamma-1}{\gamma}};\ \frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = r^{\frac{\gamma-1}{\gamma}}$$

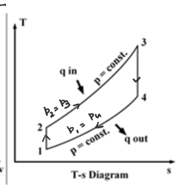
Non-dimensional.

$$\Rightarrow \frac{w}{c_p T_1} = \left(\frac{T_3}{T_1} - \frac{T_3}{T_1}\frac{T_4}{T_3}\right) - \left(\frac{T_2}{T_1} - 1\right) = t\left(1 - \frac{1}{r^{\frac{\gamma-1}{\gamma}}}\right) - \left(r^{\frac{\gamma-1}{\gamma}} - 1\right);\ \ r = \frac{p_2}{p_1};\ t = \frac{T_3}{T_1}$$

$$\Rightarrow w = 0 \ \text{at} \ r = 1 \ \& \ r = t^{\frac{\gamma}{\gamma-1}}$$



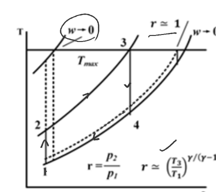
P-v Diagram

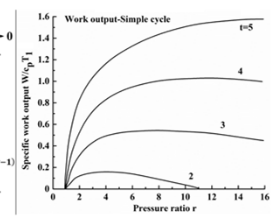


T-s Diagram

Specific Work Output

- On a T-s diagram, a constant "t-curve" a maximum at certain pressure ratios. The work output is zero at r = 1 and at the value for which the compression and expansion process coincide.
- For any given value of 't', the optimum pressure ratio can be found for maximum specific work output by differentiating the work-equation to zero. Then, the optimum pressure ratio and maximum power output can obtained.
- The specific work output is a maximum when the pressure ratio is such that compressor and turbine outlet temperatures are equal.





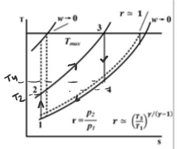
Specific Work Output

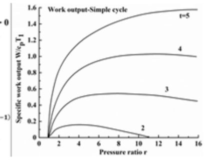
Recall $r=\frac{p_2}{p_1};\ t=\frac{T_3}{T_1};\ \frac{T_2}{T_1}=r^{\frac{\gamma-1}{\gamma}};\frac{T_3}{T_4}=r^{\frac{\gamma-1}{\gamma}};\ \frac{w}{c_pT_1}=t\left(1-\frac{1}{r^{\frac{\gamma-1}{\gamma}}}\right)-\left(r^{\frac{\gamma-1}{\gamma}}-1\right)$

Maximum work $\Rightarrow \frac{d\left(\frac{w}{c_pT_1}\right)}{d\left(r^{\frac{\gamma-1}{\gamma}}\right)}=0;\Rightarrow r_{opt}^{\frac{\gamma-1}{\gamma}}=\sqrt{t};\ \ t=r_{opt}^{\frac{2(\gamma-1)}{\gamma}}$

$\Rightarrow \frac{T_2}{T_1}\times\frac{T_3}{T_4}=\sqrt{t}\times\sqrt{t}\quad \Rightarrow \frac{T_2}{T_1}\times\frac{T_3}{T_4}=t\quad \Rightarrow T_2=T_4\ \ (at\ r=r_{opt})$

$\Rightarrow\left(\frac{w}{c_pT_1}\right)_{\max}=t\left(1-\frac{1}{\sqrt{t}}\right)-\left(\sqrt{t}-1\right);\ \eta=1-\left(\frac{1}{r^{\frac{\gamma-1}{\gamma}}}\right)\Rightarrow\eta=1-\frac{1}{\sqrt{t}}=1-\sqrt{\frac{T_1}{T_3}}$





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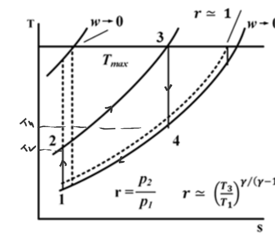
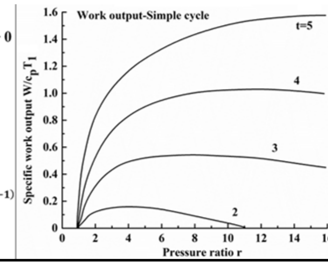
Specific Work Output

Inferences:

- **At two values of pressure ratio (r), the specific work output is zero.**
- **At optimum value of pressure ratio (r_{opt}), the specific work output is maximum. It refers equal temperatures at outlet of compressor and turbine.**
- **For all the values of pressure ratio between 1 and r_{opt}, T_4 is greater than T_2.**
- **So, there is a necessity to include a heat-exchanger to reduce the heat transfer from external source and increase the efficiency.**





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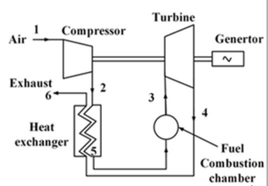
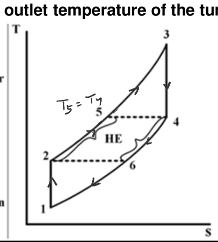
Heat Exchange Cycle

- **When a heat exchanger is added to the thermal circuit Brayton cycle (ideal air-standard cycle for gas turbine engine), then it is called as "heat-exchange cycle".**
- **The main intention is to preheat air inlet to combustion chamber by tapping the heat from exhaust of turbine. Hence, its appropriate location is the between the outlets of compressor and turbine.**
- **In ideal scenario, $T_5 = T_4$ with of heat-exchange cycle i.e. compressed gases can be heated to a maximum limit up to outlet temperature of the turbine.**





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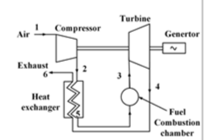
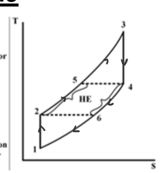
Heat Exchange Cycle

Thermodynamic analysis





Ideal cycle: $r=\frac{p_2}{p_1};\ \ p_3=p_2\ \&\ p_4=p_1;\ \frac{T_2}{T_1}=\frac{T_3}{T_4}=r^{\frac{\gamma-1}{\gamma}}$

$w_{net}=w_{34}-w_{12}=c_p\left(T_3-T_4\right)-c_p\left(T_2-T_1\right);\ \eta=\frac{w_{net}}{q_{in}}=\frac{c_p\left(T_3-T_4\right)-c_p\left(T_2-T_1\right)}{c_p\left(T_3-T_2\right)}=1-\frac{1}{r^{\frac{\gamma-1}{\gamma}}}$ $\eta = f(r)$

Heat exchange cycle: $r=\frac{p_2}{p_1};\ t=\frac{T_3}{T_1};\ p_3=p_2=p_5\ \&\ p_4=p_1=p_6;\ T_5=T_4;\ \frac{T_2}{T_1}=\frac{T_3}{T_4}=r^{\frac{\gamma-1}{\gamma}}$

$w_{net}=w_{34}-w_{12}=c_p\left(T_3-T_4\right)-c_p\left(T_2-T_1\right)$

$\eta=\frac{w_{net}}{q_{in}}=\frac{c_p\left(T_3-T_4\right)-c_p\left(T_2-T_1\right)}{c_p\left(T_3-T_5\right)}=1-\frac{T_2-T_1}{T_3-T_4}=1-\frac{\frac{T_2}{T_1}-1}{\frac{T_3}{T_1}-\frac{T_4}{T_3}\frac{T_3}{T_1}}=1-\frac{r^{\frac{\gamma-1}{\gamma}}}{t}$ $\eta = f(r, t)$

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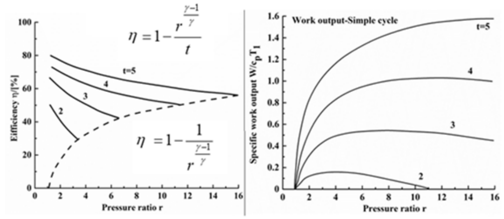
Heat Exchange Cycle

Inferences:

- With addition of a heat-exchanger, there is no change in specific work output in a heat exchange cycle as compared to ideal Brayton cycle.
- Efficiency of heat exchange cycle is higher when the 'temperature ratio' increases.
- For a given value of 'temperature ratio', the cycle efficiency increases with decrease in 'pressure ratio'. In contrast, the cycle efficiency increases with increase in 'pressure ratio' for an ideal Brayton cycle.



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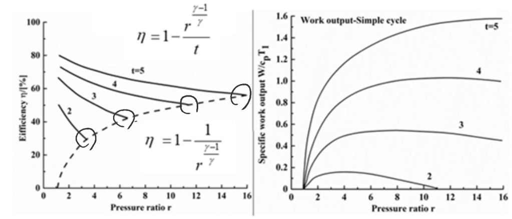
Heat Exchange Cycle

Inferences:

- The upper limit of each *constant t-curve* starts at *r = 1*, with cycle efficiency equal to Carnot efficiency. As expected, it is the limiting case for requirement of complete external heat reception and rejection at upper and lower cycle temperature.
- The lower limit of each *constant t-curve* stops when 'optimum pressure ratio' is reached with respect to ideal Brayton cycle. This is the pressure ratio for which specific work output curve reaches the maximum for which $T_4 = T_2$.



$r = 1$

$\eta = 1 - \frac{1}{t}$

$= 1 - \frac{T_1}{T_3}$

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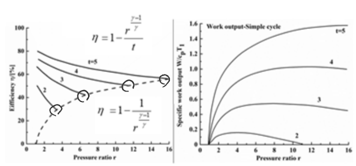
Heat Exchange Cycle

Inferences:

- For higher values 'pressure ratio' (beyond optimum value), a heat exchanger would cool the air leaving the compressor and cycle efficiency is reduced. Therefore, the *constant t-curve* are not extended beyond this point where they meet efficiency curve for ideal Brayton cycle.
- In order to obtain a appreciable improvement in efficiency by heat exchange, (a) the operating 'pressure ratio' lesser than optimum value is used, for maximum specific work output; (b) it is not necessary to use higher cycle pressure as the maximum cycle temperature is increased.
- For real cycles, the conclusion (a) remains true but conclusion (b) requires modification.



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Numerical Problems

Q1. In a Brayton cycle, air is drawn from atmosphere at 1 bar and 30°C into the compressor. The maximum pressure and temperature of the cycle is limited as 7 bar and 800°C, respectively. If the heat supply to the cycle is 90 MW, calculate, (a) thermal efficiency of the cycle; (b) work ratio; (c) power output; (d) exergy flow rate of gas leaving the turbine.

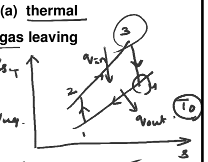
Soln

$p_1 = 1$ bar $\quad T_1 = 30°C = 303$ K.

$p_3 = 7$ bar. $\quad T_3 = 800°C = 1073$ K.

$Q_{23} = 90$ MW $= 90$ MJ/s

$r = \frac{p_3}{p_1} = 7$. $\quad \gamma = 1.4 \quad c_p = 1.005$ kJ/kg.



(a) $\eta = 1 - \frac{1}{(r)^{\frac{\gamma-1}{\gamma}}} = 0.42$

(b) $WR = \frac{w_T - w_C}{w_T}$

$w_T = c_p (T_3 - T_4)$

$w_C = c_p (T_2 - T_1)$

$\frac{T_2}{T_1} = r^{\frac{\gamma-1}{\gamma}}$

$\frac{T_3}{T_4} = r^{\frac{\gamma-1}{\gamma}}$

$T_2 = 527.6$ K

$T_4 = 616.2$ K.

$T_4 > T_2$

$w_T = 459$ kJ/kg. $\quad w_C = 225.7$ kJ/kg.

$WR = 0.5$

(c) $\frac{w_{net}}{Q_{in}} = 0.42 \quad W_{net} = 37.8$ MJ/s.

$\dot{Q}_{in} = \dot{m} c_p (T_3 - T_2)$

$\dot{m} = 164.2$ kg/s.

(d) $X_{heat} = \int_{T_0}^{T_4} \left(1 - \frac{T_0}{T}\right) \delta Q = \dot{m} c_p (T_4 - T_0) - \dot{m} c_p T_0 \ln \frac{T_4}{T_0}$

$X_{heat} = 16.5$ MW

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Numerical Problems

Q2. A gas turbine plant operating on Brayton cycle has maximum and minimum temperature as 30°C and 800°C, respectively. Calculate, (a) ~~maximum~~ optim. specific work done by the gas; (b) optimum pressure ratio; (c) cycle efficiency; (d) ratio of cycle efficiency to Carnot efficiency.

Soln (a) $\left(\frac{W}{c_p T}\right)_{opt} = t\left(1 - \frac{1}{\sqrt{t}}\right) - (\sqrt{t} - 1)$ $\quad t = \frac{T_3}{T_1} = \frac{800 + 273}{30 + 273} = 3.54.$

$\left(\frac{W}{c_p T_1}\right)_{opt} = 0.77 \Rightarrow W_{opt} = 234.5$ kJ/kg. $\quad T = 30^\circ C$, $c_p = 1.005$ kJ/kg.

(b) $(r_{opt})^{\frac{\gamma-1}{\gamma}} = \sqrt{t} \Rightarrow r_{opt} = 9.1$

(c) $\eta_b = 1 - \sqrt{\frac{T_1}{T_3}} = 0.47.$

(d) $\eta_c = 1 - \frac{T_1}{T_3} = 0.72 \qquad \frac{\eta_b}{\eta_c} = \frac{0.47}{0.72} = 0.652$

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THANK YOU

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