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

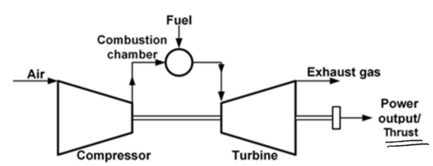
Lecture 4

Real Gas Turbine Performance Cycle

- Component Losses
- Stagnation Properties of Fluids
- Quantification of Losses

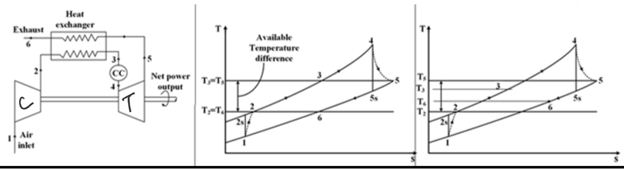
Component Losses

- The performance of real gas turbine cycles differ from ideal cycles due to various component losses. The accounting of these losses are to quantified.
- Compressors and turbines (commonly, known as 'turbomachines') are the fundamental units of a gas turbine system.
- The fluid velocities are generally high in turbomachines. The changes in kinetic energy between inlet and outlet of these components can not ignored.
- The compression and expansion are irreversible adiabatic processes and involve an increase in entropy.
- The fluid friction results in pressure losses in combustion chambers and heat exchangers. The inlet and exhaust ducts are referred as associated component losses.



Component Losses

- The compressed air can not be heated to the temperature of gas leaving the turbine. When a heat exchanger of economic size is incorporated in the thermal circuit, there is likely to be terminal temperature difference.
- Bearing and windage friction in transmission process of "turbine and compressors" add to extra work in addition to compressor work.
- The values of "c_p and γ" for the working fluid vary through out the cycle due to changes in temperature and chemical composition.
- The heating value and composition of fuel are not taken into account for ideal cycle. So, cycle performance is expressed in terms of fuel consumption per unit work output (i.e. specific fuel consumption).



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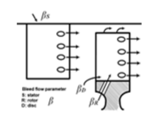
Component Losses

- With the knowledge of compressor delivery temperature, fuel composition, turbine inlet temperature, it is possible to calculate fuel-air ratio. Combustion efficiency can be included to allow incomplete combustion.
- With internal combustion, the mass flow rate through turbine is greater than that of compressor by virtue of fuel addition.
- When the turbine inlet temperatures are higher than 1350 K, the turbine blading must be internally cooled as well as the disc and blade roots (called as air-cooled turbine). It requires up to 15% of compressed air delivery as bleeding for cooling purposes.
- In practice, about 1-2% of compressor air is bled off for cooling turbine discs and blade roots. It is compensated through fuel-air ratio (0.01-0.02) addition.
- All the above issues are to be accounted for consideration of a real gas turbine performance cycle.



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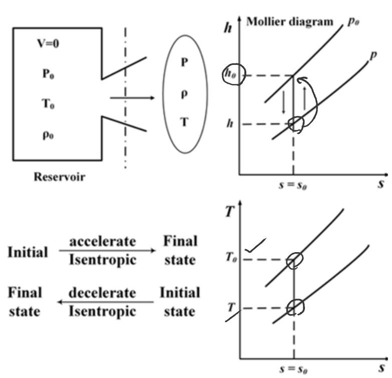
Stagnation Properties of Fluids

- The kinetic energy terms in the steady flow energy equation can be accounted implicitly by making use of the concept of "stagnation/total enthalpy".
- In a physical sense, the "stagnation enthalpy" of a flowing gas stream (having enthalpy *h* and velocity *C*) is the enthalpy that it would possess when the flow stream is brought to rest adiabatically and without heat transfer.



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Stagnation Properties of Fluids

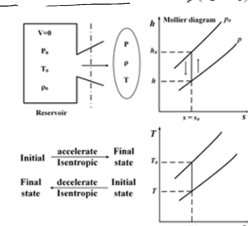
T : static temperature; $\dfrac{C^2}{2c_p}$: dynamic temperature; T_0 : stagnation temperature

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

$$\Rightarrow h_0 = h + \frac{C^2}{2};\ c_pT_0 = c_pT + \frac{C^2}{2} \ \Rightarrow T_0 = T + \frac{C^2}{2c_p}$$

Adiabatic compression : $w = -c_p(T_2 - T_1) - \frac{1}{2}(C_2^2 - C_1^2) = -c_p(T_{02} - T_{01})$

Heating process without work transfer : $q = c_p(T_2 - T_1) + \frac{1}{2}(C_2^2 - C_1^2) = c_p(T_{02} - T_{01})$



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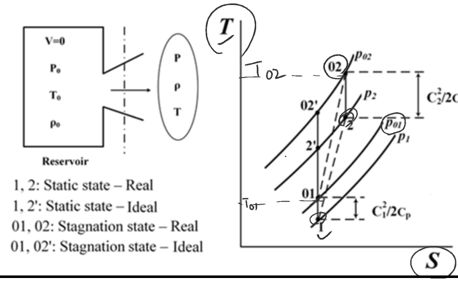
Stagnation Properties of Fluids

- When a gas is slowed down and the rise in temperature as well as pressure is noticed, then it is imagined to be brought to rest isentropcally. So, the "stagnation pressure" can be defined in a similar way.
- The stagnation pressure is constant in a stream flowing without heat and work transfer only if the friction is absent. The drop in stagnation pressure can be used as a measure of fluid friction.



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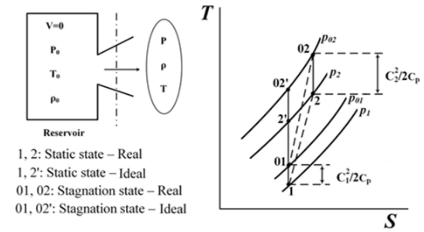
Stagnation Properties of Fluids

Stagnation temperature: $\frac{T_0}{T}=1+\frac{C^2}{2c_pT}=1+\frac{C^2}{2\left(\frac{\gamma R}{\gamma-1}\right)T}=1+\frac{C^2}{\left(\frac{2a^2}{\gamma-1}\right)}=1+\left(\frac{\gamma-1}{2}\right)M^2$ $a=\sqrt{\gamma RT}$

Stagnation pressure: $\frac{p_0}{p}=\left(\frac{T_0}{T}\right)^{\frac{\gamma}{\gamma-1}}=\left[1+\left(\frac{\gamma-1}{2}\right)M^2\right]^{\frac{\gamma}{\gamma-1}}$; M: Mach number

Isentropic compression between inlet and outlet: $\frac{p_{02}}{p_{01}}=\left(\frac{T_{02}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$; $\frac{p_{02}}{p_1}=\left(\frac{T_{02}}{T_1}\right)^{\frac{\gamma}{\gamma-1}}$; $\frac{p_{01}}{p_2}=\left(\frac{T_{01}}{T_2}\right)^{\frac{\gamma}{\gamma-1}}$



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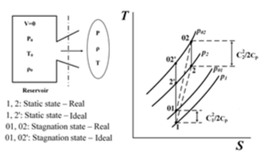
Stagnation Properties of Fluids

- When stagnation temperature is employed, there is no need to refer explicitly to the kinetic energy terms. It is easier to measure the stagnation temperature of high velocity stream than static temperature. So, it a practical advantage.
- If there is no heat and work transfer, then stagnation temperature will remain constant.



T: static temperature

$\frac{C^2}{2c_p}$: dynamic temperature

T_0: stagnation temperature

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

$\Rightarrow h_0=h+\frac{C^2}{2}$; $c_pT_0=c_pT+\frac{C^2}{2}$ $\Rightarrow T_0=T+\frac{C^2}{2c_p}$

Adiabatic compression: $w=-c_p(T_2-T_1)-\frac{1}{2}\left(C_2^2-C_1^2\right)=-c_p(T_{02}-T_{01})$

Heating process without work transfer: $q=c_p(T_2-T_1)+\frac{1}{2}\left(C_2^2-C_1^2\right)=c_p(T_{02}-T_{01})$

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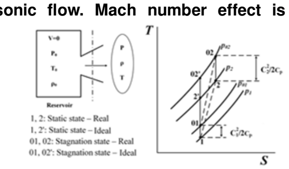
Stagnation Properties of Fluids

- It is to be noted that stagnation pressure defined for gases is not identical to pitot pressure defined for incompressible flow where Mach number effect (dynamic pressure or temperature) is not taken into account.
- The stagnation pressure calculated through Bernoulli's equation for incompressible effect does not take into account of compressibility effect.
- A close look will revel that incompressible flow calculation will underestimate the stagnation pressure by 11% for sonic flow. Mach number effect is insignificant below the value of 0.3.



Compressible flow: $p_0=p\left[1+\left(\frac{\gamma-1}{2}\right)M^2\right]^{\frac{\gamma}{\gamma-1}}$; M: Mach number

Incompressible flow: $p_0^*=p+\frac{1}{2}\rho C^2$; $\frac{p_0^*}{p}=1.7$; $\frac{p_0}{p}=1.89$ (At $M=1$)

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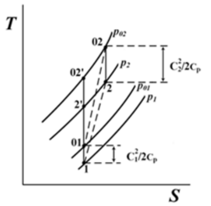
Quantification of Component Losses

Isentropic efficiency – Compressor:

- The efficiency of any machine (either work producing or work consuming) is normally expressed as ratio of actual work to ideal work.
- Turbomachines are mostly adiabatic and its ideal process is isentropic.
- Considering the accountability of changes in kinetic energy between inlet and outlet, the concept of stagnation enthalpy is used to define the isentropic efficiencies of turbomachines.



$$\eta_c = \frac{W_c'}{W_c} = \frac{\Delta h_0'}{\Delta h_0} = \frac{T_{02}'-T_{01}}{T_{02}-T_{01}}$$

$$\Rightarrow T_{02}-T_{01} = \frac{1}{\eta_c}\left(T_{02}'-T_{01}\right) = \frac{T_{01}}{\eta_c}\left(\frac{T_{02}'}{T_{01}}-1\right)$$

$$\Rightarrow T_{02}-T_{01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}}-1\right]$$

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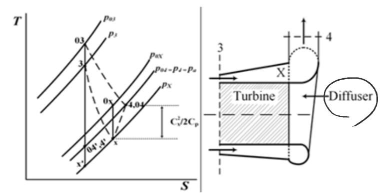
Quantification of Component Losses

Isentropic efficiency – Turbine:

- While defining the turbine efficiency through stagnation pressures, it implies that the kinetic energies in the exhaust gases are getting utilized in propelling nozzle of jet engine.
- The kinetic energy is largely recovered in an exhaust diffuser which increases the pressure ratio across the turbine.
- The diffuser reduces the final velocity to negligible value so that the final pressure becomes ambient.



$$\eta_t = \frac{W_t}{W_t'} = \frac{\Delta h_0}{\Delta h_0'} = \frac{T_{03}-T_{04}}{T_{03}-T_{04}'}$$

$$\Rightarrow T_{03}-T_{04} = \eta_t\left(T_{03}-T_{04}'\right) = \eta_t T_{03}\left(1-\frac{T_{04}'}{T_{03}}\right)$$

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

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Quantification of Component Losses

Polytropic efficiency:

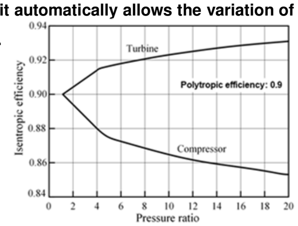
- In practical aspects, the isentropic efficiency of turbines and compressors varies with pressure ratio.
- The concept of "polytropic efficiency" is introduced which is defined as the isentropic efficiency of an elemental stage in the process that remains constant throughout the entire process.
- Both polytropic and isentropic efficiency represent same information in different forms. But over wide range of pressure ratio calculation, polytropic efficiency is a better approach because it automatically allows the variation of isentropic efficiency with pressure ratio.

$$T_{02}-T_{01} = T_{01}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{n-1}{n}}-1\right];\ \frac{n-1}{n} = \left(\frac{1}{\eta_{\infty c}}\right)\left(\frac{\gamma-1}{\gamma}\right)$$

air

$$T_{03}-T_{04} = T_{03}\left[1-\left(\frac{1}{p_{03}/p_{04}}\right)^{\frac{n-1}{n}}\right];\ \frac{n-1}{n} = \eta_{\infty t}\left(\frac{\gamma-1}{\gamma}\right)$$

hot gas



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Quantification of Component Losses

Pressure losses:

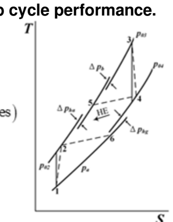
- In the combustion chamber, the loss in stagnation pressure occurs due to aerodynamic resistance of flame-stabilizing, mixing devices and momentum changes due to exothermic reaction.
- When a heat exchanger is included, there is a pressure loss in the passage of air-side and gas-side.
- The pressure losses have effect of decreasing the turbine pressure ratio relative to compressor pressure ratio. Hence, it reduces net power output.
- Thus, the pressure losses have significant effect to drop cycle performance.

$$p_{03} = p_{02}\left(1-\frac{\Delta p_b}{p_{02}}-\frac{\Delta p_{ha}}{p_{02}}\right);\ p_{04} = p_a + \Delta p_{hg}$$

$\frac{\Delta p_b}{p_{02}}$ (combustion chamber) $\approx 2-3\%$ (industrial plant), $6-8\%$ (aero engines)

Δp_{ha} (heat exchanger air side) $\approx 3\%$ of compressor delivery pressure

Δp_{hg} (heat exchanger gas side) ≈ 0.04 bar



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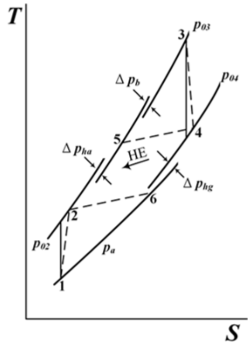
Quantification of Component Losses

Heat exchanger effectiveness:

- **The heat exchangers for gas turbines can be in the form of counter-flow and cross-flow recuperator or regenerators.**
- **The fluid can exchange heat through a separating wall (recuperator) or alternatively absorb/reject heat when brought into contact in a cylindrical matrix arrangement (regenerators).**
- **The fundamental process is the fact that the turbine exhaust gases reject heat to the compressor delivered air supply.**



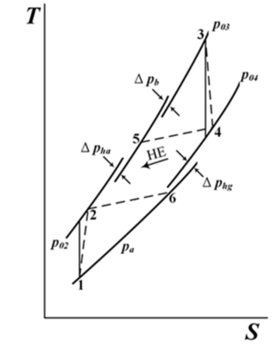
Heat balance: $\dot{m}_a c_{p46}\left(T_{04}-T_{06}\right)=\dot{m}_g c_{p25}\left(T_{05}-T_{02}\right)$

Effectiveness: $\varepsilon=\dfrac{\dot{m}_a c_{p25}\left(T_{05}-T_{02}\right)}{\dot{m}_g c_{p24}\left(T_{04}-T_{02}\right)}$; Thermal ratio: $TR=\dfrac{T_{05}-T_{02}}{T_{04}-T_{02}}$

Quantification of Component Losses

Heat exchanger effectiveness:

- **The losses in the heat exchanger is quantified in terms of its effectiveness.**
- **The higher volume of heat-exchanger has higher values of effectiveness, but to a upper limit of 0.9.**
- **The cost of heat exchanger is largely determined by its surface area.**



Heat balance: $\dot{m}_a c_{p46}\left(T_{04}-T_{06}\right)=\dot{m}_g c_{p25}\left(T_{05}-T_{02}\right)$

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Quantification of Component Losses

Combustion efficiency:

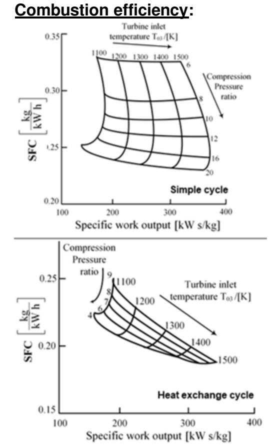
- **The performance of real cycles can be expressed in terms of specific fuel consumption (SFC) i.e. fuel mass flow rate per unit net power output.**
- **SFC is expressed in terms of fuel-air ratio (f) and combustion efficiency.**
- **For a given temperature difference, the combustion efficiency is defined as the ratio of theoretical fuel air ratio to the actual fuel air ratio.**
- **The complete combustion is ensured with combustion efficiency of 98-99%.**
- **The other concept of using the term 'heat rate (HR)' in place of efficiency because the fuel prices can be evaluated directly.**

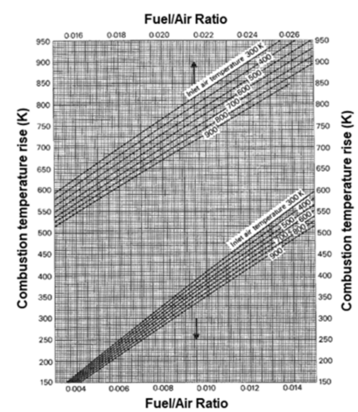
$$SFC\,(\mathrm{kg/kWh})=\frac{3600\,f}{W_{net}},\ W_{net}\,(\mathrm{kWs/kg});\ SFC(\mathrm{kg/kWh})=\frac{\dot{m}_f}{W_{net}},\ W_{net}\,(\mathrm{kW})$$

$$\eta=\frac{W_{net}}{fQ_{net}}=\frac{W_{net}\,(\mathrm{kWs/kg})}{f\,Q_{net}\,(\mathrm{kWs/kg})};\quad \eta=\frac{3600}{SFC(\mathrm{kg/kWh})\times Q_{net}\,(\mathrm{kWs/kg})}=\frac{3600}{HR}$$

Quantification of Component Losses

Combustion efficiency:





Reference: Gas Turbine Theory, HIH Saravanamuttoo, H. Cohen and GFC Rogers, Pearson, India, 2012

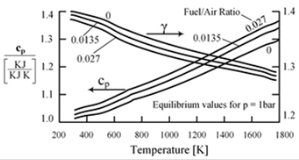
Quantification of Component Losses

Variation in specific heat:

- The variation in specific heat and ratio of specific heat plays important role in estimation of cycle performance. They must be taken into account due to change in conditions throughout the cycle.
- The maximum pressure of the cycle has tremendous impact on dissociation of air species which begins at temperature of 1500 K.

Mechanical losses:

- In all gas turbine units, the power necessary to drive the compressor is transmitted from the turbine without any intermediate gearing. Here, only 1% loss is permitted.



Air: $c_p = 1.005\,\text{kJ/kg.K};\ \gamma = 1.4$

Combustion gases: $c_p = 1.148\,\text{kJ/kg.K};\ \gamma = 1.33$

$$W_c = \frac{W_t}{\eta_m} = \frac{c_{pa}(T_{02} - T_{01})}{\eta_m};\ \eta_m = 0.99$$

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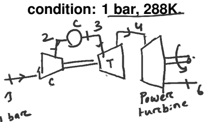
Numerical Problems

Q1. Determine the specific work output, fuel consumption and cycle efficiency for a simple gas turbine unit with a free turbine for the following data: compressor pressure ratio: 12, turbine inlet temperature: 1350 K, Isentropic efficiency: compressor (0.85) & turbine (0.9), mechanical and combustion efficiency: 0.98, combustion chamber pressure loss: 6% of compressor delivery pressure, exhaust pressure loss: 0.03 bar, ambient condition: 1 bar, 288K.

Soln



$p_{01} = p_1 = 1$ bar

$T_{01} = T_1 = 288$K.

Compressor

$$T_{02} - T_{01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$$

$\Rightarrow T_{02} - T_{01} = 349.1$ K.

$w_c = c_p(T_{02} - T_{01})$, $w_{tc} = \frac{w_c}{\eta_m}$

$\Rightarrow w_{tc} = \frac{c_p}{\eta_m}(T_{02} - T_{01})$

$\Rightarrow w_{tc} = 358$ kJ/kg.

$\frac{p_{02}}{p_{01}} = 12$

$\eta_c = 0.85$

$T_{01} = 288$ K.

$c_p = 1.005$ kJ/kg.K

$\eta_m = 0.98$.

Turbine

$$T_{03} - T_{04} = \eta_t T_{03}\left[1 - \frac{1}{(p_{03}/p_4)^{\frac{\gamma-1}{\gamma}}}\right]$$

$T_{03} = 1350$ K

$\Rightarrow \frac{p_{03}}{p_4} = 3.15$

$w_{tb} = w_{Tc} = c_{pg}(T_{03} - T_{04})$

$358 = 1.148(T_0 - T_{04})$

$\Rightarrow T_0 - T_{04} = 303$ K.

$\gamma_g = 1.33$

$c_{pg} = 1.148$ kJ/kg

$\eta_t = 0.9$.

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Numerical Problems

Q1. Determine the specific work output, fuel consumption and cycle efficiency for a simple gas turbine unit with a free turbine for the following data: compressor pressure ratio: 12, turbine inlet temperature: 1350 K, Isentropic efficiency: compressor (0.85) & turbine (0.9), mechanical and combustion efficiency: 0.98, combustion chamber pressure loss: 6% of compressor delivery pressure, exhaust pressure loss: 0.03 bar, ambient condition: 1 bar, 288K.

Combustion chamber pressure loss $= 0.06 \times 12 = 0.72$ bar.

$p_{03} = 12 - 0.72 = 11.28$ bar.

$p_{04} = \frac{11.28}{3.15} = 3.58$ bar.

Power turbine, $\frac{p_{04}}{p_{05}} = \frac{3.58}{1.03} = 3.47$ bar.

$$T_{04} - T_{05} = \eta_t T_{04}\left[1 - \frac{1}{(p_{04}/p_{05})^{\frac{\gamma-1}{\gamma}}}\right] = 250\text{ k}$$

(a) $w_{tp} = c_{pg}(T_{04} - T_{05}) = 287$ kJ/kg.

(b) $SFC = \frac{3600 f}{w_{tp}} = 0.255$ kg/kWH

(c) $\eta = \frac{3600}{SFC \times Q} = 0.32$

$p_{05} = 1.03$ bar

$p_{01} = 1$ bar.

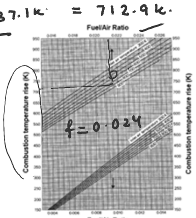
$\Delta T = T_{03} - T_{02}$

$T_{02} = 637.1$ K $= 712.9$ K.

$\gamma = 1.33$

$\eta = 0.9$.

$Q_{cv} = 43$ MJ/kg



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THANK YOU

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