Subject Name & Code:

ENGINEERING THERMODYNAMICS- BE03000201

Experiment – 1

TOPIC: TO UNDERSTAND THE APPLICATIONS OF SFEE

Q-1: Differentiate Flow process and non-flow process.

Answer:

Basis of Difference	**Flow Process**	**Non-Flow Process**
System Definition	Open System	Closed System
Mass Transfer	Mass crosses the system boundary.	No mass crosses the system boundary.
System Identity	The matter within the system (control volume) is constantly changing.	The matter within the system is fixed and constant.
Energy Analysis	Analyzed using the **Steady Flow Energy Equation (SFEE)**.	Analyzed using the **First Law for Closed Systems**: $Q - W = \Delta U$.
Primary Energy Interactions	Involves flow work or displacement work in addition to heat and work.	Involves only heat transfer and work interaction (like boundary work).
Examples	Nozzles, turbines, compressors, heat exchangers.	Piston-cylinder assembly without inlet/exit valves, heating of a sealed container.

Q-2: What are the requirements of a steady flow process?

Answer:

For a process to be classified as a **steady flow process**, the following conditions must be satisfied *simultaneously* at every point in the system:

1. **Constant Mass Flow Rate:** The mass flow rate of fluid entering the control volume must be equal to the mass flow rate leaving it.

$$\dot{m}_{in} = \dot{m}_{out} = \text{constant}$$

2. **Constant Properties at Points:** The properties of the fluid (e.g., pressure, temperature, velocity, density) at any specific point within the control volume **do not change with time**. (They can vary from one point to another).
3. **Constant Energy of the System:** The total energy content (E) of the control volume remains constant with time. That is, the rate of energy entering equals the rate of energy leaving.

$$\frac{dE_{CV}}{dt} = 0$$

4. **Constant State Rate:** The rate of heat transfer and the rate of work transfer across the boundary of the control volume are constant with time.

$$\dot{Q} = \text{constant}, \dot{W} = \text{constant}$$

5. **No Accumulation:** There is no accumulation of mass or energy within the control volume over time.

Q-3: Derive the general energy equation for a steady flow system.

Answer:

Step 1: State the Energy Balance for a General System
The First Law of Thermodynamics states that for any system:

$$\text{Net Energy Transfer} = \text{Change in Total Energy of the System}$$

Expressed on a rate basis for a control volume:

$$\dot{E}_{in} - \dot{E}_{out} = \frac{dE_{CV}}{dt}$$

Where:

- $\dot{E}_{in}$ is the total rate of energy entering the control volume.
- $\dot{E}_{out}$ is the total rate of energy leaving the control volume.
- $\frac{dE_{CV}}{dt}$ is the rate of change of energy within the control volume.

Step 2: Apply the Steady-State Condition
For a **steady-flow process**, the energy within the control volume does not change with time.

$$\frac{dE_{CV}}{dt} = 0$$

Therefore, the energy balance simplifies to:

$$\dot{E}_{in} = \dot{E}_{out}$$

Step 3: Identify All Energy Transfer Mechanisms
Energy can cross the boundary via **mass flow** and **energy transfers** (heat and work).

- **Energy associated with mass flow:** For each unit mass, the total energy is the sum of its internal, kinetic, and potential energy.

$$e = u + \frac{V^2}{2} + gz \text{(per unit mass)}$$

However, when mass flows, it also does **flow work** to push itself into or out of the control volume. This work is Pv, where P is pressure and v is specific volume. The combination $u + Pv$ is defined as **enthalpy**, h. Therefore, the total energy carried by a flowing fluid per unit mass is:

$$\theta = h + \frac{V^2}{2} + gz$$

The rate of energy transport by mass flow is $\dot{m}\theta$.

- **Energy transfers:**
    - **Heat Transfer Rate:** $\dot{Q}_{in}$ (positive into the system).
    - **Work Transfer Rate:** $\dot{W}_{in}$ (positive into the system). This is often given as shaft work, shear work, etc. In many thermodynamics contexts, work done *by* the system is considered positive, so we will use $\dot{W}_{out}$.

Step 4: Write the Energy Balance Equation
Considering one inlet (point 1) and one outlet (point 2):

$$(\dot{Q}_{in} + \dot{W}_{in} + \dot{m}(h_1 + \frac{V_1^2}{2} + gz_1)) = (\dot{Q}_{out} + \dot{W}_{out} + \dot{m}(h_2 + \frac{V_2^2}{2} + gz_2))$$

Rearranging to have all transfer terms on one side:

$$\dot{Q}_{in} - \dot{Q}_{out} + \dot{W}_{in} - \dot{W}_{out} = \dot{m}[(h_2 + \frac{V_2^2}{2} + gz_2) - (h_1 + \frac{V_1^2}{2} + gz_1)]$$

Step 5: Final Form of the Steady Flow Energy Equation (SFEE)
Defining the net heat and work transfer *to* the system:

$$\dot{Q}_{net,in} = \dot{Q}_{in} - \dot{Q}_{out}$$
$$\dot{W}_{net,in} = \dot{W}_{in} - \dot{W}_{out}$$

The most common form of the SFEE, where work done *by* the system is taken as positive ($W = W_{out}$), is:

$$\dot{Q}_{net,in} - \dot{W}_{net,out} = \dot{m}[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)]$$

For a process where $\dot{Q}$ and $\dot{W}$ represent the net rates, and dividing through by $\dot{m}$ to get the intensive form (per unit mass):

$$\boxed{q - w = (h_2 - h_1) + \frac{1}{2}(V_2^2 - V_1^2) + g(z_2 - z_1)}$$

This is the general energy equation for a steady flow system.

Q-4: Steam enters a nozzle at a pressure of 7 bar and 200oC and leaves at a pressure of 2 bar. The initial velocity of steam at the entrance is 50 m/s and exit velocity from nozzle is 750 m/s. The mass rate through the nozzle is 1500 kg/h. The heat loss from nozzle is 12000 kJ/h. Determine the final enthalpy of the steam and nozzle area if the specific volume is 1.25 m3/kg. Take initial enthalpy 2850 kJ/kg.

Answer:

Given:

- $P_1 = 7$ bar, $T_1 = 200°C$, $h_1 = 2850$ kJ/kg
- $P_2 = 2$ bar
- $V_1 = 50$ m/s
- $V_2 = 750$ m/s

- $\dot{m} = 1500\ \text{kg/h} = \frac{1500}{3600} = 0.4167\ \text{kg/s}$
- $\dot{Q} = -12000\ \text{kJ/h} = -\frac{12000}{3600} = -3.333\ \text{kW}$
- $v_2 = 1.25\ \text{m}^3/\text{kg}$

To Find: h_2, A_2

Solution:

Step 1: Apply Steady Flow Energy Equation (SFEE) for a Nozzle
The general SFEE is:

$$\dot{Q} - \dot{W} = \dot{m}[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2 \times 1000} + g(z_2 - z_1)]$$

For a nozzle: $\dot{W} = 0$, and potential energy change $g(z_2 - z_1)$ is negligible.

$$\dot{Q} = \dot{m}[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2000}]$$

Substituting the known values:

$$-3.333 = 0.4167[(h_2 - 2850) + \frac{750^2 - 50^2}{2000}]$$
$$-3.333 = 0.4167[(h_2 - 2850) + \frac{562500 - 2500}{2000}]$$
$$-3.333 = 0.4167[(h_2 - 2850) + \frac{560000}{2000}]$$
$$-3.333 = 0.4167[(h_2 - 2850) + 280]$$

Step 2: Solve for Final Enthalpy (h_2)

$$\frac{-3.333}{0.4167} = h_2 - 2850 + 280$$
$$-8.000 = h_2 - 2570$$
$$h_2 = 2570 - 8.000$$
$$\boxed{h_2 = 2562\ \text{kJ/kg}}$$

Step 3: Calculate Exit Area (A_2)
Using the continuity equation at the exit:

$$\dot{m} = \frac{A_2 V_2}{v_2}$$
$$A_2 = \frac{\dot{m}\, v_2}{V_2}$$
$$A_2 = \frac{0.4167 \times 1.25}{750}$$

$$A_2 = \frac{0.5209}{750}$$

$$\boxed{A_2 = 6.945 \times 10^{-4}\ \text{m}^2 = 694.5\ \text{mm}^2}$$

Q-5: A turbine operating under steady flow conditions, receives steam at 4500kg/hour. Steam enters at the velocity of 2800 m/min, an elevation of 5.5 m and a specific enthalpy of 2800 kJ/kg. It leaves the turbine at a velocity of 5600 m/min, an elevation of 1.5 m and specific enthalpy of 2300 kJ/kg. Heat losses from the turbine to the surroundings amount to 16000 kJ/hr. Determines the power output of the turbine.

Answer:

Given:

- $\dot{m} = 4500\ \text{kg/h} = \frac{4500}{3600} = 1.25\ \text{kg/s}$
- $V_1 = 2800\ \text{m/min} = \frac{2800}{60} = 46.67\ \text{m/s}$
- $z_1 = 5.5\ \text{m}$
- $h_1 = 2800\ \text{kJ/kg}$
- $V_2 = 5600\ \text{m/min} = \frac{5600}{60} = 93.33\ \text{m/s}$
- $z_2 = 1.5\ \text{m}$
- $h_2 = 2300\ \text{kJ/kg}$
- $\dot{Q} = -16000\ \text{kJ/h} = -\frac{16000}{3600} = -4.444\ \text{kW}$

To Find: $\dot{W}_{out}$

Solution:

Step 1: Apply Steady Flow Energy Equation (SFEE) for a Turbine

$$\dot{Q} - \dot{W} = \dot{m}[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2000} + \frac{g(z_2 - z_1)}{1000}]$$

Substitute all known values:

$$-4.444 - \dot{W} = 1.25[(2300 - 2800) + \frac{93.33^2 - 46.67^2}{2000} + \frac{9.81(1.5 - 5.5)}{1000}]$$

Step 2: Calculate Kinetic Energy Term

$$V_2^2 = 8711.1 \text{and} V_1^2 = 2177.8$$

$$\frac{V_2^2 - V_1^2}{2000} = \frac{8711.1 - 2177.8}{2000} = \frac{6533.3}{2000} = 3.2667\ \text{kJ/kg}$$

Step 3: Calculate Potential Energy Term

$$\frac{g(z_2 - z_1)}{1000} = \frac{9.81 \times (-4)}{1000} = -0.03924\ \text{kJ/kg}$$

Step 4: Combine All Terms

$$-4.444 - \dot{W} = 1.25[-500 + 3.2667 - 0.03924]$$
$$-4.444 - \dot{W} = 1.25[-496.7725]$$
$$-4.444 - \dot{W} = -620.9656$$

Step 5: Solve for Power Output ($\dot{W}$)

$$-\dot{W} = -620.9656 + 4.444$$
$$-\dot{W} = -616.5216$$
$$\boxed{\dot{W} = 616.52\ \text{kW}}$$

Q-6: A ir at a temperature of 15oC passes through a heat exchanger at a velocity of 30 m/s where it is heated to 800oC. It then enters a turbine with the same velocity of 30 m/s and expands until the temperature falls to 650oC. On leaving the turbine air is taken at velocity 60 m/s to nozzle where it expands until the temperature has fallen to 500oC. If the air flow rate is 2 kg/s. Calculate (1) Rate of heat transfer to the air in the heat exchanger (2) Power output from the turbine (3) The velocity of air at the exit of nozzle.

Answer:

Given:

- $\dot{m} = 2\ \text{kg/s}$
- **Process 1-2 (Heat Exchanger):** $T_1 = 15°\text{C} = 288\ \text{K}$, $T_2 = 800°\text{C} = 1073\ \text{K}$, $V_1 = V_2 = 30\ \text{m/s}$
- **Process 2-3 (Turbine):** $T_3 = 650°\text{C} = 923\ \text{K}$, $V_3 = 60\ \text{m/s}$
- **Process 3-4 (Nozzle):** $T_4 = 500°\text{C} = 773\ \text{K}$
- For air:

To Find: (1) $\dot{Q}_{in}$, (2) $\dot{W}_{turbine}$, (3) V_4

Solution:

Part (1): Rate of Heat Transfer in Heat Exchanger

Step 1: Apply SFEE to Heat Exchanger
For a heat exchanger: $\dot{W} = 0, \Delta KE \approx 0, \Delta PE \approx 0$.

$$\dot{Q} = \dot{m}(h_2 - h_1) = \dot{m}C_p(T_2 - T_1)$$
$$\dot{Q} = 2 \times 1.005 \times (1073 - 288)$$
$$\dot{Q} = 2 \times 1.005 \times 785$$
$$\boxed{\dot{Q}_{in} = 1577.85\,\text{kW}}$$

Part (2): Power Output from Turbine

Step 2: Apply SFEE to Turbine
For a turbine: $\dot{Q} \approx 0, \Delta PE \approx 0$.

$$-\dot{W} = \dot{m}[(h_3 - h_2) + \frac{V_3^2 - V_2^2}{2000}]$$
$$-\dot{W} = 2[1.005(923 - 1073) + \frac{60^2 - 30^2}{2000}]$$
$$-\dot{W} = 2[1.005 \times (-150) + \frac{3600 - 900}{2000}]$$
$$-\dot{W} = 2[-150.75 + \frac{2700}{2000}]$$
$$-\dot{W} = 2[-150.75 + 1.35]$$
$$-\dot{W} = 2[-149.4] = -298.8$$
$$\boxed{\dot{W}_{turbine} = 298.8\,\text{kW}}$$

Part (3): Exit Velocity from Nozzle

Step 3: Apply SFEE to Nozzle
For a nozzle: $\dot{Q} = 0, \dot{W} = 0, \Delta PE \approx 0$.

$$h_3 + \frac{V_3^2}{2000} = h_4 + \frac{V_4^2}{2000}$$
$$\frac{V_4^2}{2000} = (h_3 - h_4) + \frac{V_3^2}{2000}$$
$$\frac{V_4^2}{2000} = C_p(T_3 - T_4) + \frac{V_3^2}{2000}$$
$$\frac{V_4^2}{2000} = 1.005(923 - 773) + \frac{60^2}{2000}$$
$$\frac{V_4^2}{2000} = 1.005 \times 150 + \frac{3600}{2000}$$
$$\frac{V_4^2}{2000} = 150.75 + 1.8 = 152.55$$
$$V_4^2 = 152.55 \times 2000 = 305100$$
$$V_4 = \sqrt{305100}$$
$$\boxed{V_4 = 552.36\,\text{m/s}}$$

Experiment – 2

TOPIC: VERIFICATION OF FIRST AND SECOND LAW WITH HEAT PUMP

Q-1: What is a heat pump?

Answer:

A heat pump is a thermodynamic device that operates on a refrigeration cycle to transfer thermal energy from a low-temperature source (sink) to a high-temperature sink (source) by the application of external work.

- **Primary Function:** Its main purpose is to **maintain a space at a high temperature** by supplying heat to it.
- **Working Principle:** It absorbs heat (Q_L) from a cold reservoir (like outside air, ground, or water) and, using work input (W_{in}), rejects a larger amount of heat (Q_H) to a hot reservoir (like a building's interior).
- **Energy Efficiency:** The Coefficient of Performance (COP) of a heat pump is defined as the ratio of the desired heat output to the required work input, $\text{COP}_{HP} = \frac{Q_H}{W_{in}}$, and is always greater than 1.

Q-2: Differentiate between a heat pump and a refrigerator.

Answer:

Basis of Difference	Heat Pump	Refrigerator
Primary Objective	To maintain a space at a **high temperature** by **supplying heat** to it.	To maintain a space at a **low temperature** by **removing heat** from it.
Desired Output	The heat **rejected** in the condenser (Q_H).	The heat **absorbed** in the evaporator (Q_L).
Coefficient of Performance (COP)	$\text{COP}_{HP} = \frac{\text{Desired Output}}{\text{Work Input}} = \frac{Q_H}{W_{in}}$	$\text{COP}_{R} = \frac{\text{Desired Output}}{\text{Work Input}} = \frac{Q_L}{W_{in}}$

Basis of Difference	Heat Pump	Refrigerator
Mathematical Relation	$\text{COP}_{HP} = \text{COP}_R + 1$	$\text{COP}_R = \text{COP}_{HP} - 1$
Application	Space heating, water heating.	Food preservation, air conditioning.

Q-3: What is the relation between COP of Carnot heat pump and COP of Carnot refrigerator?

Answer:

For a Carnot cycle operating between the same two thermal reservoirs at temperatures T_H (hot) and T_L (cold):

1. **COP of Carnot Refrigerator:**

$$\text{COP}_{R,Carnot} = \frac{T_L}{T_H - T_L}$$

2. **COP of Carnot Heat Pump:**

$$\text{COP}_{HP,Carnot} = \frac{T_H}{T_H - T_L}$$

3. **Relation between them:**

$$\text{COP}_{HP,Carnot} = \text{COP}_{R,Carnot} + 1$$
$$\boxed{\text{COP}_{HP,Carnot} = \text{COP}_{R,Carnot} + 1}$$

This relationship shows that for the same temperature limits, the COP of a Carnot heat pump is always exactly 1 greater than the COP of a Carnot refrigerator.

Q-4: Differentiate between Carnot heat pump and actual heat pump.

Answer:

Basis of Difference	Carnot Heat Pump	Actual Heat Pump
Concept	A **theoretical, idealized** model used to establish the maximum possible performance.	A **practical, real-world** device available for commercial use.
Processes	Consists of four perfectly reversible processes: two isothermal and two adiabatic.	Involves irreversible processes (e.g., pressure drops, friction, heat transfer with finite temperature difference).
COP	Has the **maximum possible COP** for given temperature limits: $\text{COP}_{HP,Carnot} = \frac{T_H}{T_H - T_L}$.	Has a **lower COP** than the Carnot heat pump due to irreversibilities. $\text{COP}_{HP,Actual} < \text{COP}_{HP,Carnot}$.
Feasibility	It is **not practically possible** to construct.	It is **practically feasible** to build and operate.
Working Fluid	The concept is independent of the working fluid.	Performance heavily depends on the properties of the real working fluid (refrigerant) used.

Q-5: Heat pump has thermodynamic advantage over electric heater: Justify

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Answer:

A heat pump has a significant thermodynamic advantage over a conventional electric resistance heater because it **moves heat** rather than **converting electrical energy directly into heat**.

- **Electric Heater:** It operates on the principle of Joule heating, where electrical work (W_{in}) is directly and completely converted into heat energy (Q_{out}). Therefore, its efficiency or COP is:

$$\text{COP}_{Heater} = \frac{Q_{out}}{W_{in}} = 1$$

For 1 kW of electrical input, it delivers exactly 1 kW of heat.

- **Heat Pump:** It uses electrical work (W_{in}) to *drive a refrigeration cycle* that extracts additional heat (Q_L) from a low-temperature source (like the outside air). The total heat delivered (Q_H) is the sum of the extracted heat and the work input:

$$Q_H = Q_L + W_{in}$$

Therefore, its COP is:

$$\text{COP}_{HP} = \frac{Q_H}{W_{in}} = \frac{Q_L + W_{in}}{W_{in}} > 1$$

For the same 1 kW of electrical input, a heat pump can deliver 3 to 5 kW of heat, depending on its COP.

Conclusion: The thermodynamic advantage lies in the fact that a heat pump's COP is **always greater than 1**, whereas an electric heater's COP is exactly 1. This makes a heat pump a far more energy-efficient device for space heating, as it provides more useful heating energy than the electrical energy it consumes.

Experiment – 3

TOPIC: TO VERIFY FIRST AND SECOND LAW WITH I.C. ENGINE

Q-1: State the first law of thermodynamics.

Answer:

The **First Law of Thermodynamics** is a statement of the principle of conservation of energy for thermodynamic systems. It states:

"Energy can neither be created nor destroyed; it can only change from one form to another."

For a closed system undergoing a cycle, the net heat transfer is equal to the net work transfer:

$$\oint \delta Q = \oint \delta W$$

For a process, the change in the total energy of the system is equal to the difference between the heat added to the system and the work done by the system:

$$Q - W = \Delta E$$

where ΔE is the change in the total energy of the system (Internal energy + Kinetic energy + Potential energy).

Q-2: State the two statements of the second law of thermodynamics.

Answer:

The Second Law of Thermodynamics establishes the direction of thermodynamic processes and the principle of degradation of energy. It has two classical statements:

1. **Kelvin-Planck Statement:**

"It is impossible for any device that operates on a thermodynamic cycle to receive heat from a single reservoir and produce a net amount of work."

   - **Implication:** No heat engine can have 100% thermal efficiency. It is impossible to convert heat completely into work in a cyclic process. A heat engine must also reject some heat to a low-temperature sink.

2. **Clausius Statement:**

"It is impossible to construct a device which operates in a cycle and produces no effect other than the transfer of heat from a cooler body to a hotter body."

- **Implication:** Heat cannot spontaneously flow from a colder body to a hotter body. Work input is required to achieve this, as in a refrigerator or heat pump.

These two statements are equivalent; the violation of one implies the violation of the other.

Q-3: List the measuring instruments and devices used for the experiment performance.

Answer:

The following instruments and devices are used to conduct the performance test of an I.C. engine:

1. **I.C. Engine Test Rig:** The main apparatus (single or multi-cylinder, petrol or diesel).
2. **Dynamometer:**
    - **Rope Brake Dynamometer:** To measure brake power by applying a frictional torque using a rope, dead weights, and a spring balance.
    - **Electrical Dynamometer:** To apply and measure the load electrically.
3. **Tachometer / RPM Indicator:** To measure the rotational speed of the engine crankshaft.
4. **Air Box with Orifice and Manometer:** To measure the volume of air consumed by the engine.
5. **Fuel Supply System:**
    - **Fuel Tank & Burette:** A graduated burette is used to measure the volumetric fuel consumption over a specific time.
6. **Cooling Water System:**
    - **Water Flow Meter:** To measure the flow rate of coolant through the engine jacket.
    - **Thermometers / Temperature Sensors:** To measure the inlet and outlet temperatures of the coolant.
7. **Exhaust Gas Calorimeter:** To measure the heat carried away by the exhaust gases.
    - **Thermometers / Thermocouples:** To measure the temperature of exhaust gases entering and leaving the calorimeter and the water flowing through it.
    - **Measuring Cylinder & Stopwatch:** To measure the water flow rate through the calorimeter.
8. **Stopwatch:** To measure the time for fuel consumption and other flow rate measurements.

Q-4: Describe the process for creating the heat balance sheet for an I.C. engine.

Answer:

A **Heat Balance Sheet** is an account of the complete distribution of the heat supplied to an engine. It is typically prepared per minute or per kg of fuel. The process is as follows:

Step 1: Calculate the Total Heat Supplied (H_s)

$$H_s = \dot{m}_f \times \text{Calorific Value (C.V.) of Fuel}$$

where $\dot{m}_f$ is the mass flow rate of fuel.

Step 2: Measure or Calculate the Various Energy Outputs (in kW or kJ/min)

1. **Brake Power (Useful Work Output):**

$$\text{B.P.} = \frac{2\pi NT}{60}$$

where N is RPM and T is torque.

   - Heat Equivalent of B.P. = B.P. (in kW)

2. **Heat Lost to Cooling Water ($Q_{coolant}$):**

$$Q_{coolant} = \dot{m}_w \times C_{pw} \times (T_{c\_out} - T_{c\_in})$$

where $\dot{m}_w$ is the mass flow rate of coolant and C_{pw} is its specific heat.

3. **Heat Lost to Exhaust Gases (Q_{exh}):**

$$Q_{exh} = \dot{m}_g \times C_{pg} \times (T_{exh} - T_{atm})$$

where $\dot{m}_g$ is the mass flow rate of exhaust gases (approx. $\dot{m}_{air} + \dot{m}_f$), and C_{pg} is the specific heat of exhaust gases.

4. **Unaccounted Losses:** This includes radiation, convection, and other minor losses not measured directly. It is found by difference:

$$\text{Unaccounted Losses} = H_s - [\text{B.P.} + Q_{coolant} + Q_{exh}]$$

Step 3: Prepare the Sheet

The results are tabulated, often as percentages of the total heat supplied (H_s).

Item	Energy (kW or kJ/min)	Percentage of H_s (%)
1. Heat equivalent of Brake Power	B.P.	$\frac{\text{B.P.}}{H_s} \times 100$

Item	Energy (kW or kJ/min)	Percentage of H_s (%)
2. Heat lost to cooling water	$Q_{coolant}$	$\frac{Q_{coolant}}{H_s} \times 100$
3. Heat lost to exhaust gases	Q_{exh}	$\frac{Q_{exh}}{H_s} \times 100$
4. Unaccounted losses	By Difference	By Difference
Total	H_s	100%

Q-5: Mention the different losses during the performance test of an internal combustion engine.

Answer:

During a performance test, the chemical energy of the fuel (H_s) is converted into various other forms, many of which are considered "losses" from the perspective of useful work output. The major losses are:

1. **Heat Loss to Coolant (Jacket Water Loss):** A significant portion of the heat is carried away by the engine's cooling system to prevent overheating.
2. **Exhaust Gas Loss:** A substantial amount of heat is lost as the hot exhaust gases are expelled from the engine cylinder.
3. **Friction Power Loss (Mechanical Losses):** This includes energy lost to overcome friction in various engine parts like pistons, rings, bearings, and also to drive auxiliary components like the water pump, oil pump, and fan.
    - F.P. = I.P. - B.P. (where I.P. is Indicated Power and B.P. is Brake Power).
4. **Radiation and Convection Losses:** Heat is lost from the hot engine surfaces (like the cylinder block and exhaust manifold) to the surrounding atmosphere by radiation and convection.
5. **Unaccounted Losses:** This is a residual category that includes losses not directly measured in a standard test, such as:
    - Incomplete combustion of fuel.
    - Dissociation of gases at high temperatures.
    - Energy consumed in charging the battery (in SI engines).

Experiment – 4

TOPIC: TO DETERMINE HEAT LOSS FROM PIPE-IN-PIPE HEAT EXCHANGER USING SFEE AND TO VERIFY ENTROPY PRINCIPLE FOR THE HEAT EXCHANGER.

Q-1: Define heat exchanger.

Answer:

A **heat exchanger** is a device designed to transfer thermal energy (heat) between two or more fluids that are at different temperatures, without allowing the fluids to mix. The fluids can be separated by a solid wall to prevent mixing or they can be in direct contact.

Q-2: List the equipment which works on the principle of heat exchange and also list the application of heat exchangers.

Answer:

Equipment	Application
Condenser	Thermal power plants (to condense exhaust steam from turbine), refrigeration and air conditioning systems.
Evaporator	Refrigeration systems (to absorb heat from the refrigerated space), chemical processing industries.
Boiler	Power generation, industrial process heating.
Radiator	Automobile engine cooling, central heating systems in buildings.
Shell and Tube Heat Exchanger	Oil cooling, chemical processing, power plants (as feedwater heaters).
Cooling Tower	Rejecting waste heat from power plants and industrial processes to the atmosphere.
Regenerator	Gas turbines, blast furnaces (for waste heat recovery).

Equipment	Application
Air Preheater	Boiler systems (to preheat combustion air using flue gases, improving efficiency).

Q-3: List the consideration for the application of the Steady Flow Energy Equation on a heat exchanger.

Answer:

When applying the Steady Flow Energy Equation (SFEE) to a heat exchanger, the following considerations and simplifications are made:

1. **Steady-State, Steady-Flow Condition:** The mass flow rates of both hot and cold fluids are constant, and the properties at any point within the exchanger do not change with time.
2. **No Work Transfer ($W = 0$):** There are no rotating shafts or other means of work interaction (shaft work) between the heat exchanger and its surroundings.
3. **Negligible Kinetic Energy Change ($\Delta KE \approx 0$):** The inlet and outlet velocities of the fluids are typically low and/or similar, so the change in kinetic energy is negligible.
4. **Negligible Potential Energy Change ($\Delta PE \approx 0$):** The inlet and outlet elevations for each fluid stream are generally the same, so the change in potential energy is negligible.
5. **Adiabatic to Surroundings:** The heat exchanger is often well-insulated, meaning there is no significant heat transfer with the external environment except between the two fluid streams themselves. (If there is a heat loss, it must be accounted for as Q_{loss}).
6. **Constant Pressure:** The flow through a heat exchanger often occurs at approximately constant pressure for each stream.

These considerations simplify the general SFEE, $\dot{Q} - \dot{W} = \dot{m}[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)]$, to the following for each fluid stream:

$$\dot{Q} = \dot{m}(h_{out} - h_{in})$$

For incompressible fluids (liquids) or ideal gases with constant specific heats, this further simplifies to:

$$\dot{Q} = \dot{m}C_p(T_{out} - T_{in})$$

Q-4: Classify the type of heat exchange possible with a heat exchanger in thermodynamic processes.

Answer:

Based on the flow arrangement of the hot and cold fluids, heat exchangers are classified as follows:

1. **Parallel Flow Heat Exchanger:**
    - The two fluids flow in the same direction.
    - The temperature difference between the fluids is high at the inlet and decreases along the length, leading to lower overall heat transfer rates compared to other types.
2. **Counter Flow Heat Exchanger:**
    - The two fluids flow in opposite directions.
    - A more uniform temperature difference is maintained along the length, making it the most thermodynamically efficient design for a given surface area.
3. **Cross Flow Heat Exchanger:**
    - The two fluids flow perpendicular to each other.
    - Efficiency lies between parallel and counter flow. Commonly used in automobile radiators and air-heating coils.

Q-5: Justify the entropy principle for the change of entropy for the heat exchanger, surrounding, and total entropy change.

Answer:

The **Entropy Principle**, derived from the Second Law of Thermodynamics, states that for any process, the total entropy change of the **universe** (system + surroundings) must be greater than or equal to zero.

$$\Delta S_{universe} = \Delta S_{system} + \Delta S_{surroundings} \geq 0$$

For a *reversible* (ideal) process, $\Delta S_{universe} = 0$. For an *irreversible* (real) process, $\Delta S_{universe} > 0$.

Applying this to a heat exchanger:

1. **Entropy Change of the System (The Heat Exchanger Itself):**
    - The "system" here is the control volume encompassing the heat exchanger.

- Its entropy change is the sum of the entropy changes of the hot and cold fluids.

$$\Delta S_{system} = \Delta S_{ho} + \Delta S_{cold}$$

$$\Delta S_{hot} = \dot{m}_h C_{p,h} \ln\left(\frac{T_{ho}}{T_{hi}}\right) \text{(This is negative as } T_{ho} < T_{hi})$$

$$\Delta S_{cold} = \dot{m}_c C_{p,c} \ln\left(\frac{T_{co}}{T_{ci}}\right) \text{(This is positive as } T_{co} > T_{ci})$$

- For an adiabatic heat exchanger, ΔS_{system} can be positive, negative, or zero depending on the flow rates and temperatures.

2. **Entropy Change of the Surroundings:**
   - If the heat exchanger is perfectly insulated (adiabatic), $\Delta S_{surroundings} = 0$.
   - If there is a heat loss (Q_{loss}) from the heat exchanger to the surroundings at temperature T_{amb}, the surroundings gain entropy.

$$\Delta S_{surroundings} = \frac{Q_{loss}}{T_{amb}} \text{ (This is positive)}$$

3. **Total Entropy Change of the Universe:**

$$\Delta S_{universe} = \Delta S_{ho} + \Delta S_{cold} + \Delta S_{surroundings}$$

Justification of the Principle:
The process of heat transfer through a finite temperature difference (from the hot fluid to the cold fluid) is inherently **irreversible**. Even in an adiabatic heat exchanger, this irreversibility ensures that $\Delta S_{universe} > 0$. The entropy decrease of the hot fluid is always **less in magnitude** than the entropy increase of the cold fluid, leading to a net positive entropy generation within the control volume. Any heat loss to the surroundings further adds to the entropy generation.

Therefore, for any real heat exchanger operating under steady flow conditions, the calculation will always show:

$$\boxed{\Delta S_{universe} > 0}$$

This positive value quantitatively demonstrates the irreversibility of the heat transfer process and verifies the Entropy Principle.

Experiment – 5

TOPIC: TO UNDERSTAND APPLICATIONS OF ENTROPY PRINCIPLE

Q-1: Define entropy.

Answer:

Entropy is a fundamental thermodynamic property that serves as a measure of the **molecular disorder or randomness** of a system. It is also a measure of the **unavailability of a system's thermal energy for conversion into mechanical work**.

Mathematically, for an internally reversible process, the differential change in entropy dS is defined as the ratio of the differential heat transfer δQ to the absolute temperature T at which the transfer occurs:

$$dS = \left(\frac{\delta Q}{T}\right)_{\text{int rev}}$$

The total change in entropy between two states 1 and 2 is:

$$\Delta S = S_2 - S_1 = \int_1^2 \left(\frac{\delta Q}{T}\right)_{\text{int rev}}$$

Q-2: Justify that entropy is a property of the system.

Answer:

A quantity is a **property** if its change is independent of the path taken and depends only on the initial and final states of the system (i.e., it is an exact differential).

- **Clausius recognized** that the quantity $\left(\frac{\delta Q}{T}\right)_{\text{rev}}$ for a reversible process is an exact differential. This is the mathematical definition of a property.
- For any **reversible cycle**, the cyclic integral of $\frac{\delta Q}{T}$ is zero:

$$\oint \left(\frac{\delta Q}{T}\right)_{\text{rev}} = 0$$

This is the necessary and sufficient condition for a function to be a property.

- This function was named **Entropy (S)**. Therefore, the change in entropy $\Delta S = S_2 - S_1$ has a fixed value for a given state change, regardless of whether the process is reversible or irreversible. We calculate it using a hypothetical reversible path between the same two states.

Since its value is fixed by the state of the system and not the path taken to reach that state, entropy is confirmed to be a property of the system.

Q-3: State the principle of increase of entropy.

Answer:

The **Principle of Increase of Entropy** is a major consequence of the Second Law of Thermodynamics. It states that:

"The total entropy of an **isolated system** always increases over time, or, in the limiting case of a reversible process, remains constant. It can never decrease."

Mathematically, this is expressed as:

$$\Delta S_{\text{isolated}} \geq 0$$

or, for the universe considered as an isolated system:

$$\Delta S_{\text{universe}} = \Delta S_{\text{system}} + \Delta S_{\text{surroundings}} \geq 0$$

- $\Delta S_{\text{universe}} = 0$ for a **reversible** process.
- $\Delta S_{\text{universe}} > 0$ for an **irreversible** (real) process.

This principle defines the **direction** in which natural processes occur.

Q-4: 3 kg of water at 55 °C is mixed with 4 kg of water at 100 °C in a steady flow process. Determine the temperature of the mixture, and state whether the process is isentropic?

Answer:

Given:

- $m_A = 3\,\text{kg}, T_A = 55^\circ\text{C} = 328\,\text{K}$
- $m_B = 4\,\text{kg}, T_B = 100^\circ\text{C} = 373\,\text{K}$
- (for water)

To Find: Mixture Temperature (T_f) and check if the process is isentropic.

Solution:

Step 1: Find Final Temperature (T_f)

Apply energy balance for an adiabatic mixing process: Heat lost by hot water = Heat gained by cold water.

$$m_B C_p (T_B - T_f) = m_A C_p (T_f - T_A)$$
$$4(373 - T_f) = 3(T_f - 328)$$
$$1492 - 4T_f = 3T_f - 984$$
$$1492 + 984 = 3T_f + 4T_f$$
$$2476 = 7T_f$$
$$T_f = \frac{2476}{7} = 353.7\ \text{K} = 80.7°\text{C}$$

Step 2: Calculate Entropy Change

- **Entropy change of cold water (A):**

$$\Delta S_A = m_A C_p \ln\frac{T_f}{T_A} = 3 \times 4.187 \times \ln(\frac{353.7}{328})$$
$$\Delta S_A = 12.561 \times \ln(1.0784) = 12.561 \times 0.0755 = 0.949\ \text{kJ/K}$$

- **Entropy change of hot water (B):**

$$\Delta S_B = m_B C_p \ln\frac{T_f}{T_B} = 4 \times 4.187 \times \ln(\frac{353.7}{373})$$
$$\Delta S_B = 16.748 \times \ln(0.9483) = 16.748 \times (-0.0531) = -0.889\ \text{kJ/K}$$

- **Total Entropy Change of the System:**

$$\Delta S_{system} = \Delta S_A + \Delta S_B = 0.949 + (-0.889) = 0.060\ \text{kJ/K}$$

Step 3: Conclusion

Since the total entropy change of the system $\Delta S_{system} = 0.060\ \text{kJ/K} > 0$, the process is **not isentropic**. It is an irreversible process, which is expected for the spontaneous mixing of two fluids at different temperatures.

Final Answer:

$T_f = 80.7°C$	The process is not isentropic.

Q-5: Mention the sign convention for entropy change of the system?

Answer:

The sign convention for the entropy change of a system (ΔS_{system}) is based on the interaction of heat with the system:

1. **Positive Entropy Change ($\Delta S_{system} > 0$):**
   This occurs when **heat is added to the system** (heat gain). The addition of heat increases the molecular disorder and randomness within the system, thereby increasing its entropy.
   Example: Boiling water (heat is added, entropy increases).

2. **Negative Entropy Change ($\Delta S_{system} < 0$):**
   This occurs when **heat is removed from the system** (heat loss). The removal of heat decreases the molecular disorder, thereby decreasing its entropy.
   Example: Condensation of steam (heat is rejected, entropy decreases).

3. **Zero Entropy Change ($\Delta S_{system} = 0$):**
   This occurs during an **adiabatic reversible process** (isentropic process). Since no heat is transferred ($Q = 0$) and the process is reversible, there is no change in the entropy of the system.
   Example: Ideal, frictionless, and adiabatic compression or expansion.

Experiment – 6

TOPIC: TO UNDERSTAND APPLICATIONS OF GOUY-STODOLA THEOREM

Q-1: What is the significance of the Gouy-Stodola theorem?

Answer:

The **Gouy-Stodola Theorem** is highly significant in thermodynamics as it provides a crucial quantitative link between two fundamental concepts: **Exergy (Work Potential)** and **Entropy Generation**.

Its primary significance lies in the equation:

$$I = T_0 \cdot \Delta S_{gen}$$

Where:

- I is the **Irreversibility** (the lost work potential).
- T_0 is the absolute temperature of the environment (the dead state).
- ΔS_{gen} is the **entropy generation** during the process.

Key Significances:

1. **Quantifies Lost Work:** It directly calculates the amount of useful work wasted due to irreversibilities in a process (like friction, unrestrained expansion, heat transfer through a finite temperature difference).
2. **Measures Process Efficiency:** It allows engineers to determine the deviation of a real process from the ideal reversible process. A lower irreversibility value indicates a more efficient process.
3. **Identifies Sources of Loss:** By pinpointing where entropy is generated, it helps identify the components or processes within a system that are the least efficient, guiding optimization and design improvements.
4. **Theoretical Foundation for Second Law Analysis:** It forms the basis for exergy analysis, a powerful tool for assessing and improving the performance of thermal systems.

Q-2: An air compressor receives air at 1 bar and 27 °C. It compresses the air to a pressure of 10 bar, while its temperature reaches 270 °C. The compressor releases 55 kJ/kg of heat from its body during the process. Determine reversible and actual work done by the compressor and irreversibility in the process.

Answer:

Given:

- $P_1 = 1$ bar, $T_1 = 27°C = 300$ K
- $P_2 = 10$ bar, $T_2 = 270°C = 543$ K
- Heat rejected, $q = -55$ kJ/kg
- For air: , , $\gamma = 1.4$
- Assume ambient temperature $T_0 = 300$ K

To Find: w_{rev}, w_{act}, I

Solution:

Step 1: Calculate Actual Work (w_{act})
Apply the Steady Flow Energy Equation (SFEE) for a compressor ($\Delta KE \approx 0, \Delta PE \approx 0$):

$$q - w_{act} = (h_2 - h_1) = C_p(T_2 - T_1)$$
$$-55 - w_{act} = 1.005(543 - 300)$$
$$-55 - w_{act} = 1.005 \times 243$$
$$-55 - w_{act} = 244.215$$
$$-w_{act} = 244.215 + 55 = 299.215$$
$$\boxed{w_{act} = -299.215 \text{ kJ/kg}}$$

(Negative sign indicates work is done *on* the compressor).

Step 2: Calculate Reversible Work (w_{rev})
Reversible work is the minimum work required for the same pressure change, which would occur in an isentropic process.

- First, find the isentropic exit temperature T_{2s}:

$$T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} = 300 \times (10)^{0.4/1.4}$$
$$T_{2s} = 300 \times (10)^{0.2857} = 300 \times 1.9306 = 579.18 \text{ K}$$

- For a reversible, adiabatic (isentropic) process, $q = 0$.

$$0 - w_{rev} = C_p(T_{2s} - T_1)$$
$$-w_{rev} = 1.005(579.18 - 300) = 1.005 \times 279.18 = 280.58$$
$$\boxed{w_{rev} = -280.58 \text{ kJ/kg}}$$

Step 3: Calculate Irreversibility (I)
Using the Gouy-Stodola Theorem: $I = T_0 \cdot \Delta s_{gen}$
First, find the entropy generation per kg of air (Δs_{gen}).

$$\Delta s_{gen} = (s_2 - s_1) - \frac{q}{T_0}$$

For an ideal gas, $s_2 - s_1 = C_p \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1}$

$$s_2 - s_1 = 1.005\ln(\frac{543}{300}) - 0.287\ln(\frac{10}{1})$$
$$s_2 - s_1 = 1.005\ln(1.81) - 0.287\ln(10)$$
$$s_2 - s_1 = 1.005 \times 0.5933 - 0.287 \times 2.3026$$

Now, calculate Δs_{gen}:

$$\Delta s_{gen} = -0.0645 - (\frac{-55}{300})$$

Now, calculate Irreversibility:

$$I = T_0 \cdot \Delta s_{gen} = 300 \times 0.1188$$
$$\boxed{I = 35.64\ \text{kJ/kg}}$$

Final Answers:

$$\boxed{w_{rev} = -280.58\ \text{kJ/kg}} \quad \boxed{w_{act} = -299.22\ \text{kJ/kg}} \quad \boxed{I = 35.64\ \text{kJ/kg}}$$

Q-3: State the applications of the Gouy-Stodola theorem.

Answer:

The Gouy-Stodola theorem ($I = T_0 \cdot \Delta S_{gen}$) is applied in various engineering analyses:

1. **Performance Evaluation of Thermal Systems:** To calculate the irreversibility and thus the efficiency of components like turbines, compressors, nozzles, and heat exchangers.
2. **Optimization of Processes:** By identifying the components with the highest irreversibility, it guides engineers to focus improvement efforts where the greatest losses occur.
3. **Heat Transfer Analysis:** To quantify the work lost when heat is transferred through a finite temperature difference.
4. **Mixing Processes:** To calculate the irreversibility generated when two fluid streams at different temperatures or compositions are mixed.
5. **Combustion Analysis:** To determine the significant work potential lost due to the highly irreversible nature of combustion reactions.

6. **System Design:** It provides a target for designers by defining the minimum work requirement (reversible work) for a process, allowing them to compare their actual designs against the theoretical ideal.

Q-4: Is fluid flow due to friction a reversible process? If yes, why?

Answer:

No, fluid flow due to friction is NOT a reversible process. It is a classic example of an **irreversible process**.

Reason: Friction converts the ordered, macroscopic kinetic energy of the fluid flow into disordered, microscopic thermal energy (an increase in internal energy). This conversion is **unidirectional**; the thermal energy generated cannot spontaneously reconvert itself back into the ordered kinetic energy of the flow to restore the system to its original state. This dissipation of useful energy into internal energy results in **entropy generation**, which is the hallmark of an irreversible process.

Q-5: What do you mean by dead state?

Answer:

The **dead state** is a reference state where a system is in complete thermodynamic equilibrium with its environment (atmosphere).

At the dead state:

1. **Thermal Equilibrium:** The temperature of the system (T) is equal to the environment temperature (T_0).
2. **Mechanical Equilibrium:** The pressure of the system (P) is equal to the environment pressure (P_0).
3. **Chemical Equilibrium:** The system has no potential for chemical reaction with the environment.

A system at its dead state has **zero exergy** (availability). It is incapable of producing any useful work, as it has no driving potential (temperature, pressure, or chemical) relative to its surroundings.

Q-6: Who propounded the concept of availability?

Answer:

The concept of **availability** (also known as **exergy** or **available energy**) was primarily developed and formalized by the American engineer **Josiah Willard Gibbs** in the 1870s through his work on thermodynamic potentials.

While other scientists like **Carnot**, **Clausius**, and **Kelvin** laid the groundwork with the Second Law, it was **Gibbs** who provided the comprehensive theoretical framework for understanding the maximum useful work obtainable from a system, which is the core of the availability concept.

Experiment – 7

TOPIC: TO COMPARE OTTO, DIESEL AND DUAL CYCLES

Q-1: For the same compression ratio and heat rejection, which cycle is more efficient: Otto, Diesel and Dual?

Answer:

For the **same compression ratio and the same heat rejection**:

$$\boxed{\eta_{Otto} = \eta_{Diesel} = \eta_{Dual}}$$

All three cycles will have **identical thermal efficiency**.

Reasoning: The air-standard efficiency for any cycle is given by:

$$\eta = 1 - \frac{\text{Heat Rejected}}{\text{Heat Supplied}}$$

If the compression ratio (r) is the same, the process of isentropic compression (1-2) and isentropic expansion (to the same volume) results in the **same temperature at the end of the expansion stroke** (T_4) for all cycles. Since heat rejection occurs at constant volume, the amount of heat rejected ($Q_r = mC_v(T_4 - T_1)$) depends only on T_4 and T_1. With the same r, T_1, and the assumption of constant C_v, Q_r is identical for all cycles. Therefore, the efficiency formula simplifies and becomes the same for all.

Q-2: For the same compression ratio and heat supplied, which cycle is more efficient: Otto, Diesel and Dual?

Answer:

For the **same compression ratio and the same heat input**:

$$\boxed{\eta_{Otto} > \eta_{Dual} > \eta_{Diesel}}$$

The **Otto cycle is the most efficient**.

Reasoning: The efficiency is $\eta = 1 - \frac{Q_{out}}{Q_{in}}$. With the same Q_{in}, the cycle that rejects the least heat (Q_{out}) is the most efficient.

- In the Otto cycle, all heat is added at the minimum volume (point 2), leading to the highest possible temperature rise for a given heat input.

- In the Diesel cycle, heat is added at constant pressure, which requires the piston to move, doing work during heat addition. This results in a lower temperature for the same heat input compared to the Otto cycle.
- The Dual cycle is a combination, so its efficiency falls between the two.

On a T-s diagram for the same compression ratio, the Otto cycle finishes its cycle at the lowest entropy, meaning it has rejected the least heat, making it the most efficient.

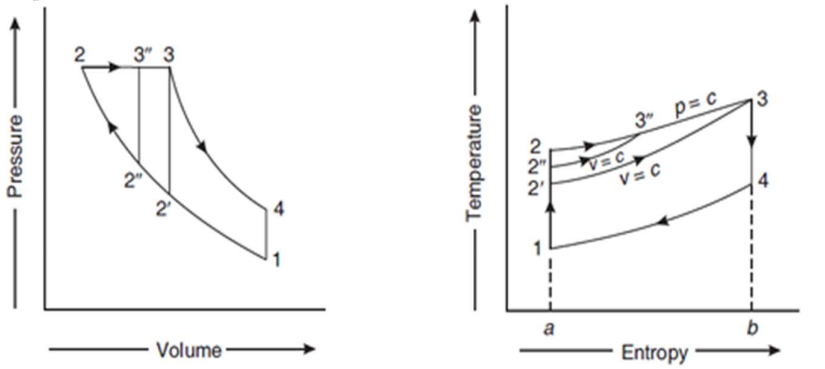
Q-3: With the help of p-v and T-s diagrams, show that for the same maximum pressure and temperature of the cycle and the same heat rejection,

$$\eta_{\text{diesel}} > \eta_{\text{dual}} > \eta_{\text{otto}}$$

Answer:

(3) Maximum pressure and temperature

Otto cycle = 1-2-3-4
Diesel cycle = 1-2'-3-4
Dual Cycle =1-2"-3"-3-4



(a) p-v diagram. (b) T-s diagram.

Fig. 6 Otto, Diesel and dual cycles for same maximum pressure and temperature

Given Conditions:

- Same maximum pressure (P_{max})
- Same maximum temperature (T_{max})
- Same heat rejection (Q_{out})

Diagrammatic Explanation:

- **P-V Diagram:** All three cycles will share the same point for maximum pressure/temperature (point 3). The Otto cycle (1-2-3-4) will have the highest compression ratio. The Diesel cycle (1-2'-3-4) will have the lowest compression ratio. The Dual cycle (1-2"-3"-3-4) will have a compression ratio between them.
- **T-S Diagram:** This is the key for comparison. All three cycles will share the same starting point (1) and the same ending point for the heat rejection process

(since $Q_{out} = T_1(s_4 - s_1)$ is the same, the point 4 is the same for all). They also share the same peak temperature point (3).

Conclusion from T-S Diagram:
The area under the curve on a T-s diagram represents heat transfer. The heat input (Q_{in}) is the area under the heat addition process.

- The **Diesel cycle** (1-2'-3-4) has the **largest area** under its heat addition line (a constant pressure line that stretches far to the right, covering more area).
- The **Otto cycle** (1-2-3-4) has the **smallest area** under its heat addition line (a constant volume line).
- The **Dual cycle** (1-2"-3"-3-4) has an intermediate area.

Since Q_{out} is the same for all, but $Q_{in,Diesel} > Q_{in,Dual} > Q_{in,Otto}$, the efficiency $\eta = 1 - Q_{out}/Q_{in}$ is highest for the Diesel cycle.

$$\boxed{\eta_{diesel} > \eta_{dual} > \eta_{otto}}$$

Q-4: Derive the mean effective pressure for Otto cycle.

Answer:

The Mean Effective Pressure (MEP) is defined as the constant pressure which, if acted on the piston during the power stroke, would produce the same net work as the actual cycle.

$$\text{MEP} = \frac{\text{Net Work Output of the Cycle } (W_{net})}{\text{Displacement Volume } (V_1 - V_2)}$$

Step 1: Express Net Work
For the Otto cycle, net work is the difference between heat added and heat rejected.

$$W_{net} = Q_{in} - Q_{out} = mC_v(T_3 - T_2) - mC_v(T_4 - T_1)$$

Step 2: Express Displacement Volume
Let the compression ratio be $r = V_1/V_2$. The displacement volume is $V_1 - V_2 = V_2(r - 1)$.

Step 3: Find Temperatures in Terms of T_1 and r

- Process 1-2 (Isentropic): $T_2 = T_1 r^{\gamma-1}$
- Process 2-3 (Constant Volume): Let the pressure ratio be $\alpha = P_3/P_2$. Then $T_3 = T_2\alpha = T_1\alpha r^{\gamma-1}$
- Process 3-4 (Isentropic): $T_4 = T_3/r^{\gamma-1} = (T_1\alpha r^{\gamma-1})/r^{\gamma-1} = T_1\alpha$

Step 4: Substitute Temperatures into Work Equation

$$W_{net} = mC_v[(T_1\alpha r^{\gamma-1}) - (T_1 r^{\gamma-1})] - mC_v[(T_1\alpha) - (T_1)]$$
$$W_{net} = mC_v T_1[\alpha r^{\gamma-1} - r^{\gamma-1} - \alpha + 1]$$
$$W_{net} = mC_v T_1[(\alpha - 1)(r^{\gamma-1} - 1)]$$

Step 5: Substitute into MEP Formula

$$\text{MEP} = \frac{mC_v T_1[(\alpha - 1)(r^{\gamma-1} - 1)]}{V_2(r - 1)}$$

From the ideal gas equation at state 1: $P_1V_1 = mRT_1$. Since $V_1 = rV_2$, we get $mT_1 = \frac{P_1V_1}{R} = \frac{P_1(rV_2)}{R}$.
Also, $R = C_p - C_v = C_v(\gamma - 1)$.

Substituting:

$$\text{MEP} = \frac{(\frac{P_1 rV_2}{C_v(\gamma - 1)})C_v[(\alpha - 1)(r^{\gamma-1} - 1)]}{V_2(r - 1)}$$

$$\boxed{\text{MEP} = \frac{P_1 r(\alpha - 1)(r^{\gamma-1} - 1)}{(\gamma - 1)(r - 1)}}$$

This is the expression for the mean effective pressure of an Otto cycle.

Q-5: An I.C. engine is working on the Otto cycle, having cylinder diameter is 250 mm and a stroke is 375 mm. If the clearance volume is 0.00263 m3, and the initial pressure and temperature are 1 bar and 50oC, calculate the air standard efficiency and mean effective pressure of the cycle. The maximum cycle pressure is limited to 25 bar.

Answer:

Given:

- Bore, $D = 250$ mm $= 0.25$ m
- Stroke, $L = 375$ mm $= 0.375$ m
- Clearance Volume, $V_c = V_2 = 0.00263$ m^3
- $P_1 = 1$ bar, $T_1 = 50°\text{C} = 323$ K
- $P_3 = 25$ bar
- For air, $\gamma = 1.4$

To Find: Air Standard Efficiency (η) and Mean Effective Pressure (MEP)

Solution:

Step 1: Calculate Compression Ratio (r)
Stroke Volume, $V_s = \frac{\pi}{4}D^2L = \frac{\pi}{4}(0.25)^2(0.375) = 0.01841\ \text{m}^3$
Total Volume at point 1, $V_1 = V_s + V_c = 0.01841 + 0.00263 = 0.02104\ \text{m}^3$

$$r = \frac{V_1}{V_2} = \frac{0.02104}{0.00263} = 8.0$$

Step 2: Calculate Air Standard Efficiency

$$\eta_{Otto} = 1 - \frac{1}{r^{\gamma-1}} = 1 - \frac{1}{8^{0.4}}$$
$$\eta_{Otto} = 1 - \frac{1}{8^{0.4}} = 1 - \frac{1}{2.297} = 1 - 0.4353$$
$$\boxed{\eta = 0.5647 \text{ or } 56.47\%}$$

Step 3: Find Temperature and Pressure at State 2

- $T_2 = T_1 r^{\gamma-1} = 323 \times 8^{0.4} = 323 \times 2.297 = 741.9\ \text{K}$
- $P_2 = P_1 r^{\gamma} = 1 \times 8^{1.4} = 1 \times 18.379 = 18.379\ \text{bar}$

Step 4: Find Temperature at State 3
Process 2-3 is constant volume, so:

$$\frac{P_3}{T_3} = \frac{P_2}{T_2} \Rightarrow T_3 = T_2 \times \frac{P_3}{P_2} = 741.9 \times \frac{25}{18.379}$$
$$T_3 = 741.9 \times 1.360 = 1009.2\ \text{K}$$

Step 5: Find Temperature at State 4
Process 3-4 is isentropic:

$$T_4 = \frac{T_3}{r^{\gamma-1}} = \frac{1009.2}{8^{0.4}} = \frac{1009.2}{2.297} = 439.3\ \text{K}$$

Step 6: Calculate Mean Effective Pressure (MEP)
Using the work-based formula: $\text{MEP} = \frac{W_{net}}{V_s}$

$$W_{net} = Q_{in} - Q_{out} = mC_v(T_3 - T_2) - mC_v(T_4 - T_1)$$

First, find mass of air (m) using state 1:
Now calculate work:

$$W_{net} = 0.0227 \times 0.7175[(1009.2 - 741.9) - (439.3 - 323)]$$
$$W_{net} = 0.01628[267.3 - 116.3] = 0.01628 \times 151.0 = 2.458\ \text{kJ}$$

Finally, calculate MEP:

$$\text{MEP} = \frac{W_{net}}{V_s} = \frac{2.458\ \text{kJ}}{0.01841\ \text{m}^3} = \frac{2458\ \text{J}}{0.01841\ \text{m}^3} = 133.5\ \text{kPa}$$
$$\boxed{\text{MEP} = 1.335\ \text{bar}}$$

Experiment – 8

TOPIC: TO STUDY VARIABLES AFFECTING THE PERFORMANCE OF RANKINE CYCLE

Q-1: Explain the Rankine cycle for a steam power plant with p-V and T-s diagram.

Answer:

The **Rankine cycle** is the fundamental thermodynamic cycle used in steam power plants to generate electricity. It consists of four key processes, as shown on the P-v and T-s diagrams below.

Processes of the Ideal Rankine Cycle:

1. **Process 4-1: Constant Pressure Heat Addition in the Boiler**
    - Cold, compressed water is heated at high pressure in the boiler until it becomes superheated steam.
    - On the T-s diagram, this is shown by the line 4-1. The area under this curve represents the heat input (Q_{in}).
    - On the P-v diagram, this is a horizontal line moving to the right, indicating an increase in specific volume.
2. **Process 1-2: Isentropic Expansion in the Turbine**
    - The high-pressure, high-temperature steam expands isentropically (at constant entropy) through the turbine, producing work (W_T).
    - On the T-s diagram, this is a vertical line from point 1 to point 2.
    - On the P-v diagram, this is a polytropic curve moving down and to the right.
3. **Process 2-3: Constant Pressure Heat Rejection in the Condenser**
    - The low-pressure steam exiting the turbine is condensed back into a saturated liquid at constant pressure in the condenser.
    - On the T-s diagram, this is a horizontal line at the condenser pressure. The area under this curve represents the heat rejected (Q_{out}).
    - On the P-v diagram, this is a horizontal line moving to the left.
4. **Process 3-4: Isentropic Compression in the Pump**
    - The condensate (saturated liquid) is pumped isentropically back to the high boiler pressure.
    - On the T-s diagram, this is a vertical line very close to the saturated liquid line. The pump work (W_P) is very small.
    - On the P-v diagram, this is a near-vertical line moving upwards.

Diagrams:

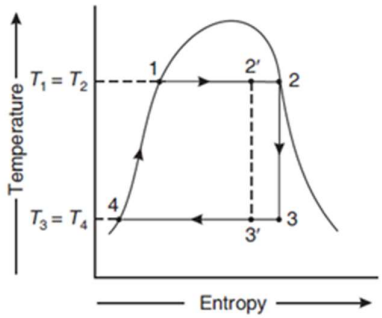


Fig. 2 Rankine cycle on T-s diagram

Q-2: What do you understand by the mean temperature of heat addition?

Answer:

The **Mean Temperature of Heat Addition (T_{m1})** is a conceptual tool that simplifies the analysis of the Rankine cycle. It is defined as **the constant temperature at which heat could be added to the cycle to produce the same net work output and the same entropy change as in the actual cycle**.

- **Purpose:** It allows for a direct comparison of the Rankine cycle's efficiency with the Carnot cycle's efficiency, which is a function of temperature.
- **Mathematical Definition:**

$$T_{m1} = \frac{Q_{in}}{\Delta s}$$

where:

  - Q_{in} is the total heat added per kg of steam in the boiler (area under curve 4-1 on the T-s diagram).
  - Δs is the change in entropy during the heat addition process ($s_1 - s_4$).

- **Significance:** A higher T_{m1} leads to a higher Rankine cycle efficiency. All methods of improving the Rankine cycle (superheating, increasing boiler pressure, reheat) essentially work by **increasing the mean temperature of heat addition**.

Q-3: For a given temperature T_2, show how the Rankine cycle efficiency depends on the mean temperature of heat addition.

Answer:

The Rankine cycle efficiency ($\eta_{Rankine}$) is given by:

$$\eta_{Rankine} = 1 - \frac{Q_{out}}{Q_{in}}$$

If the condenser temperature T_2 (and thus pressure P_2) is fixed, the heat rejection temperature is approximately fixed. The heat rejected is $Q_{out} \approx T_2 \cdot \Delta s$, where Δs is the entropy change during heat rejection.

The heat added is $Q_{in} = T_{m1} \cdot \Delta s$, where T_{m1} is the mean temperature of heat addition.

Substituting into the efficiency formula:

$$\eta_{Rankine} = 1 - \frac{T_2 \cdot \Delta s}{T_{m1} \cdot \Delta s} = 1 - \frac{T_2}{T_{m1}}$$

Conclusion: For a fixed condenser temperature T_2, the Rankine cycle efficiency **depends directly on the mean temperature of heat addition,** T_{m1}. A higher T_{m1} results in a higher cycle efficiency. This is analogous to the Carnot efficiency formula and shows that to improve efficiency, we must raise the average temperature at which heat is added.

Q-4: What is metallurgical limit?

Answer:

The **Metallurgical Limit** refers to the **maximum temperature** that the materials used in the hottest parts of a steam power plant (such as the superheater tubes and turbine blades) can withstand without suffering from excessive creep, oxidation, corrosion, or loss of mechanical strength.

- **Typical Value:** For modern power plants, this limit is around **620°C** for supercritical plants, though it can vary based on the advanced alloys used.
- **Significance:** This limit is a major constraint on improving the thermal efficiency of the Rankine cycle. While, in theory, efficiency can be increased indefinitely by raising the turbine inlet temperature, the metallurgical limit caps this temperature, thereby placing a practical upper bound on the achievable efficiency of a steam power plant.

Q-5: A steam turbine power plant operating on ideal Rankine cycle, receives steam at 30 bar, 350° C at the rate of 2kg/s and it exhausts at 0.09 bar. Calculate the following: (1) Net power output (2) steam rate (3) heat rejection in condenser in kW (4) Rankine cycle efficiency (5) Actual thermal efficiency of the plant if the boiler efficiency is 85%.

Answer:

Given:

- $P_1 = 30$ bar, $T_1 = 350°C$
- $P_2 = 0.09$ bar
- $\dot{m} = 2$ kg/s
- Boiler Efficiency, $\eta_{boiler} = 85\%$

To Find: (1) $\dot{W}_{net}$, (2) SSCR, (3) $\dot{Q}_{out}$, (4) $\eta_{Rankine}$, (5) η_{actual}

Solution using Steam Tables:

Step 1: Find properties at all state points.

- **State 1 (Turbine Inlet):** $P_1 = 30$ bar, $T_1 = 350°C$ → Superheated Steam
  - $h_1 = 3115.3$ kJ/kg,
- **State 2 (Turbine Outlet):** $P_2 = 0.09$ bar,
  - At 0.09 bar: , , $h_f = 183.3$ kJ/kg, $h_{fg} = 2397.7$ kJ/kg
  - Since $s_f < s_2 < s_g$, the state is a mixture. Find quality (x_2):

$$x_2 = \frac{s_2 - s_f}{s_{fg}} = \frac{6.7428 - 0.661}{8.277 - 0.661} = \frac{6.0818}{7.616} = 0.7986$$

$$h_2 = h_f + x_2 h_{fg} = 183.3 + 0.7986 \times 2397.7 = 183.3 + 1914.2 = 2097.5 \text{ kJ/kg}$$

- **State 3 (Condenser Outlet):** $P_3 = 0.09$ bar, Saturated Liquid
  - $h_3 = h_f = 183.3$ kJ/kg, $v_3 = v_f = 0.001008$ m^3/kg
- **State 4 (Pump Outlet):** $P_4 = P_1 = 30$ bar, $s_4 = s_3$ (Isentropic)
  - Pump Work: $W_P = v_3(P_4 - P_3) = 0.001008 \times (3000 - 9)$ kJ/kg = 3.015 kJ/kg
  - $h_4 = h_3 + W_P = 183.3 + 3.015 = 186.315$ kJ/kg

Step 2: Calculate Energy Transfers (per kg)

- Turbine Work: $W_T = h_1 - h_2 = 3115.3 - 2097.5 = 1017.8$ kJ/kg
- Pump Work: $W_P = 3.015$ kJ/kg

- Net Work: $W_{net} = W_T - W_P = 1017.8 - 3.015 = 1014.785$ kJ/kg
- Heat Input: $Q_{in} = h_1 - h_4 = 3115.3 - 186.315 = 2928.985$ kJ/kg
- Heat Rejection: $Q_{out} = h_2 - h_3 = 2097.5 - 183.3 = 1914.2$ kJ/kg

Step 3: Calculate Required Parameters

1. **Net Power Output:**

$$\dot{W}_{net} = \dot{m} \times W_{net} = 2 \times 1014.785 = \boxed{2029.57\text{ kW}}$$

2. **Steam Rate (SSCR):**

$$\text{SSCR} = \frac{3600}{W_{net}} = \frac{3600}{1014.785} = \boxed{3.548\text{ kg/kWh}}$$

3. **Heat Rejection in Condenser:**

$$\dot{Q}_{out} = \dot{m} \times Q_{out} = 2 \times 1914.2 = \boxed{3828.4\text{ kW}}$$

4. **Rankine Cycle Efficiency:**

$$\eta_{Rankine} = \frac{W_{net}}{Q_{in}} = \frac{1014.785}{2928.985} = 0.3464 \text{ or } \boxed{34.64\%}$$

5. **Actual Thermal Efficiency:**
   - The heat supplied by the fuel to the boiler (Q_{fuel}) is higher than Q_{in} due to boiler losses.

$$\eta_{boiler} = \frac{\dot{m}Q_{in}}{\dot{Q}_{fuel}} \Rightarrow \dot{Q}_{fuel} = \frac{\dot{m}Q_{in}}{\eta_{boiler}} = \frac{2 \times 2928.985}{0.85} = 6891.73\text{ kW}$$

$$\eta_{actual} = \frac{\dot{W}_{net}}{\dot{Q}_{fuel}} = \frac{2029.57}{6891.73} = 0.2945 \text{ or } \boxed{29.45\%}$$

Final Answers:

1. Net Power Output = **2029.6 kW**
2. Steam Rate = **3.548 kg/kWh**
3. Heat Rejection = **3828.4 kW**
4. Rankine Cycle Efficiency = **34.64%**
5. Actual Plant Efficiency = **29.45%**

Experiment – 9

TOPIC: TO STUDY VARIABLES AFFECTING THE PERFORMANCE OF BRAYTON CYCLE

Q-1: What are the constraints to achieve a higher turbine inlet temperature?

Answer:

Achieving a higher Turbine Inlet Temperature (TIT) is the most effective way to increase the efficiency and power output of a Brayton cycle. However, it is constrained by several critical factors:

1. **Material Strength and Creep Resistance:** At elevated temperatures, turbine blade materials (typically nickel-based superalloys) lose their mechanical strength and are subject to **creep**—a time-dependent, permanent deformation under constant stress. Exceeding the material's creep limit leads to blade failure.
2. **Oxidation and Corrosion:** High-temperature combustion gases cause severe oxidation and hot corrosion, which degrade the blade surface, reduce its cross-sectional area, and lead to premature failure.
3. **Thermal Fatigue:** The repeated heating and cooling cycles during startup and shutdown of the engine cause thermal stresses due to differential expansion. This leads to **thermal fatigue**, resulting in crack initiation and propagation.
4. **Melting Point of Blade Materials:** The turbine inlet temperature must be kept safely below the melting point of the blade materials. Advanced superalloys have melting points around 1300-1400°C, but operating temperatures are pushed beyond this limit using sophisticated cooling techniques.
5. **Cooling Technology Limitations:** To operate above the material's melting point, complex internal and film cooling passages are built into the blades. The effectiveness of this cooling is a major constraint. Pushing temperatures too high can overwhelm the cooling system's capacity.
6. **Cost of Advanced Materials and Coatings:** Developing and manufacturing single-crystal blades, ceramic matrix composites (CMCs), and applying thermal barrier coatings (TBCs) is extremely expensive, making very high TITs economically unfeasible for some applications.

Q-2: Define isentropic efficiency of turbine and compressor.

Answer:

The isentropic efficiency is a measure of the deviation of a real adiabatic device from an ideal, reversible (isentropic) device.

- **Isentropic Efficiency of a Turbine (η_T):**
  It is defined as the ratio of the **actual work output** of the turbine to the **work output that would be achieved if the process were isentropic** between the same inlet pressure and exit pressure.

$$\eta_T = \frac{\text{Actual Turbine Work}}{\text{Isentropic Turbine Work}} = \frac{W_{T,actual}}{W_{T,s}} = \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$

where:

  - h_3 = Enthalpy at turbine inlet
  - h_{4a} = Enthalpy at actual turbine outlet
  - h_{4s} = Enthalpy at isentropic turbine outlet

- **Isentropic Efficiency of a Compressor (η_C):**
  It is defined as the ratio of the **work input required for an isentropic compression** to the **actual work input** required to achieve the same exit pressure.

$$\eta_C = \frac{\text{Isentropic Compressor Work}}{\text{Actual Compressor Work}} = \frac{W_{C,s}}{W_{C,actual}} = \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

where:

  - h_1 = Enthalpy at compressor inlet
  - h_{2a} = Enthalpy at actual compressor outlet
  - h_{2s} = Enthalpy at isentropic compressor outlet

For both, an efficiency of 100% represents an ideal, loss-free process, which is impossible to achieve in reality.

Q-3: Which is the minimum compressor inlet temperature in the case of an open cycle Brayton cycle?

Answer:

In an open-cycle Brayton cycle (like in a jet engine or land-based gas turbine), the compressor draws in air directly from the **ambient atmosphere**.

Therefore, the **minimum compressor inlet temperature is the ambient air temperature** of the location where the engine is operating.

This temperature varies with weather, altitude, and geographical location. For standard performance calculations, a reference temperature of **15°C (288 K)** is often used.

Q-4: Why does the thermal efficiency of the Brayton cycle decrease with a decrease in the isentropic efficiencies of the turbine and compressor?

Answer:

A decrease in the isentropic efficiencies of the turbine and compressor signifies an **increase in irreversibilities** within these components (e.g., due to friction, turbulence). This directly harms the cycle's thermal efficiency for two main reasons:

1. **Increased Compressor Work ($W_C \uparrow$):**
   A lower compressor efficiency means **more actual work is required** to achieve the desired pressure ratio compared to the ideal case. This increases the denominator in the back-work ratio (BWR $= W_C/W_T$), meaning a larger fraction of the turbine's work is used just to drive the compressor, leaving less net work output.

2. **Decreased Turbine Work ($W_T \downarrow$):**
   A lower turbine efficiency means for the same pressure drop, **less actual work is extracted** from the expanding gases compared to the ideal case. This reduces the numerator of the net work output.

The **net work output** ($W_{net} = W_T - W_C$) is the difference between these two values. A decrease in turbine work and an increase in compressor work cause a significant reduction in net work output.

Since the **heat input** (Q_{in}) required to reach the same Turbine Inlet Temperature (TIT) remains largely unchanged, the thermal efficiency ($\eta_{th} = W_{net}/Q_{in}$) **decreases** because the numerator (W_{net}) decreases while the denominator (Q_{in}) stays constant.

In essence, lower component efficiencies mean more of the fuel's energy is wasted in overcoming internal irreversibilities rather than being delivered as useful net work.

Experiment – 10

TOPIC: TO STUDY BRAYTON CYCLE WITH REHEAT, REGENERATION AND INTERCOOLING

Q-1: What are the methods to improve the thermal efficiency of simple gas turbine cycle?

Answer:

According to the manual, the improvement of the simple cycle plant can be done by the following methods:

- **Regeneration:** This is done by preheating the compressed air entering the combustion chamber using the hot exhaust gases from the turbine in a heat exchanger. This reduces the amount of fuel needed to reach the turbine inlet temperature, thereby improving thermal efficiency.
- **Improving Turbine Output:** This can be achieved through:
    - **Reheating:** The expansion in the turbine is done in two or more stages, with reheating of the fluid to the maximum temperature between each stage. This increases the net work output.
    - **Increasing Turbine Inlet Temperature:** Using better fuel, materials, or blade cooling methods to allow for a higher maximum cycle temperature.
    - **Improving Turbine Efficiency:** This depends on design improvements to the turbine itself.
- **Reducing Compressor Input:** This can be achieved through:
    - **Intercooling:** The compression work is reduced by cooling the air between compressor stages, bringing its temperature closer to the initial state. This is done using multi-stage compression with intercoolers.
    - **Lowering Inlet Temperature:** Reducing the temperature of the air entering the compressor.
    - **Increasing Compressor Efficiency:** This depends on design improvements to the compressor.
    - **Water Injection:** Injecting water at the compressor inlet increases output and efficiency due to the added mass and the cooling of the air.

Q-2: What is the effect of regeneration on Brayton cycle efficiency?

Answer:

The effect of regeneration is to **increase the thermal efficiency** of the Brayton cycle. The manual states: "Hence the thermal efficiency will accordingly be higher." This is because the compressor work (W_c) and turbine work (W_t) are not affected by the regenerator. The key effect is that the heat required to be supplied in the combustion chamber (Q_s) is decreased, as the air is preheated from T_2 to T_3 by the exhaust gases. Since efficiency $\eta_{th} = W_{net}/Q_s$, a reduction in Q_s for the same W_{net} leads to an increase in efficiency.

Q-3: Define the effectiveness of regenerator.

Answer:

The effectiveness of a regenerator (ε) is a measure of how well the real heat exchanger performs compared to an ideal one. It is defined as the ratio of the actual temperature rise of the compressed air to the maximum possible temperature rise that would occur in an ideal regenerator.

As per the manual, the effectiveness is given by:

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2}$$

where:

- T_3 = Actual temperature of air leaving the regenerator.
- T_2 = Temperature of air entering the regenerator (from the compressor).
- T_5 = Temperature of the exhaust gas entering the regenerator (from the turbine).

In an ideal regenerator, $T_3 = T_5$ and the effectiveness would be 1.

Q-4: What is the effect of reheating on Brayton cycle efficiency?

Answer:

The effect of reheating is to **increase the net work output** but **decrease the thermal efficiency** of the Brayton cycle. The manual explicitly states: "So, efficiency decreases because of reheating, and only net work done is higher." This occurs because the heat

addition during reheating is done at a lower pressure (p_x) than in the main combustion chamber. The efficiency of converting this reheat energy into work is lower, which outweighs the gain in total work output, leading to a net decrease in cycle efficiency.

Q-5: What is the effect of intercooling on Brayton cycle efficiency?

Answer:

The effect of intercooling is to **increase the net work output** but **decrease the thermal efficiency** of the Brayton cycle. The manual states: "The effect of intercooling is to increase the work and decrease the efficiency as compared to the simple ideal cycle without intercooling." This is because the heat rejected in the intercooler to cool the compressed air is a complete loss, as "no part of it is converted into useful work." While compressor work is reduced (increasing net work), the heat supplied in the combustion chamber must now raise the temperature of the cooler air from the intercooler, requiring more fuel and thus reducing the overall thermal efficiency.