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STATE BOARD OF TECHNICAL EDUCATION (SBTE), BIHAR

DEPARTMENT OF MECHANICAL ENGINEERING

Diploma in Mechanical Engineering | Semester - IV

LAB MANUAL

(Applied Thermodynamics & HVAC / RAC LAB)

Course Code: P2425403

Course code	Course Titles	Lab Assessment (LA) Marks		Lab Assessment (LA) Hours/Week
		Progressive Lab Assessment(PLA)	End Laboratory Assessment(ELA)	
P2425403	Applied Thermodynamics	20	30	4

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EXPERIMENT NO. 01: DETERMINE THE EFFICIENCY OF THE GIVEN CONDENSER

LSO 1.1: Use different types of condenser | Relevant CO: CO1

1. Aim

To determine the Vacuum Efficiency (η_{vac}) and Condenser Thermal Efficiency (η_{cond}) of a surface steam condenser under varying cooling water circulation rates.

2. Apparatus & Technical Specifications

Sr. No.	Apparatus Name	Engineering Specifications	Quantity
1	Surface Condenser Test Rig	Two-pass shell and tube; 16 copper tubes (ID: 12 mm, OD: 14 mm); Shell dia: 200 mm	1 Unit
2	Water-Ring Vacuum Pump	Driven by 0.5 HP induction motor; Ultimate vacuum: 700 mm Hg	1 No.
3	Bourdon Vacuum Gauge & U-Tube	Range: 0 to 760 mm Hg; Least count: 2 mm Hg	1 Set
4	Digital Temperature Sensors	RTD Pt-100 sensors at steam inlet, condensate, cooling water in and out ($\pm 0.1^\circ\text{C}$)	4 Nos.

3. Mathematical Equations

- By Dalton's Law: $P_{total} = P_{steam} + P_{air}$
- Actual Vacuum (V_{act}): $V_{act} = P_{baro} - P_{total}$ [mm Hg]
- Ideal Vacuum (V_{ideal}): $V_{ideal} = P_{baro} - P_{sat}(at\ T_{condensate})$ [mm Hg]
- Vacuum Efficiency: $\eta_{vac} = (V_{act} / V_{ideal}) \times 100\%$
- Condenser Efficiency: $\eta_{cond} = [(T_{w,out} - T_{w,in}) / (T_{sat} - T_{w,in})]$

4. Working Diagram

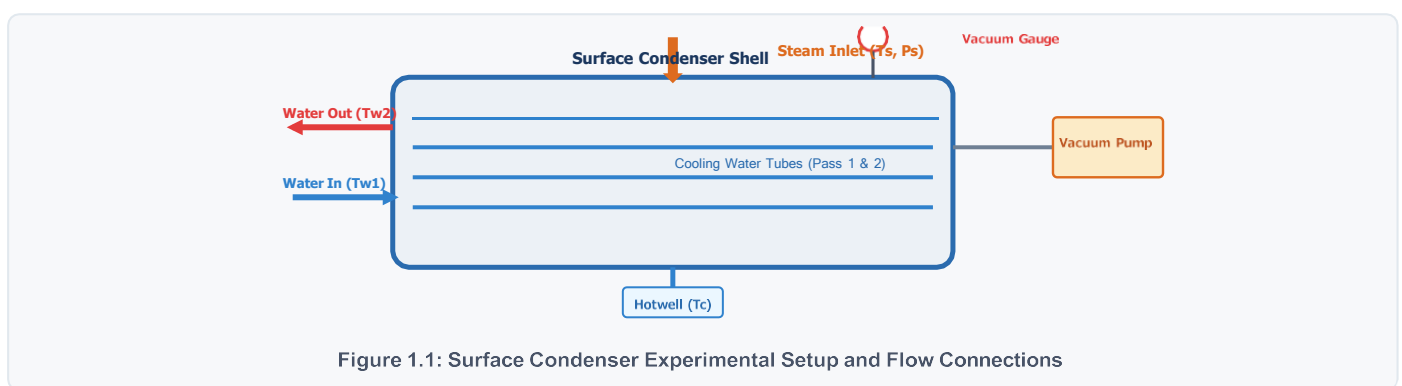


Figure 1.1: Surface Condenser Experimental Setup and Flow Connections

5. Working Procedure

1. Note local barometric pressure ($P_{baro} = 760\text{ mm Hg}$). Start cooling water circulation pump.
2. Start the vacuum pump to evacuate air until steady vacuum is established ($>600\text{ mm Hg}$).
3. Admit steam to the shell. Allow system to attain thermodynamic steady state (15 min).
4. Record vacuum gauge reading (V_{act}), steam temp (T_s), condensate temp (T_c), cooling water inlet (T_{w1}) and outlet (T_{w2}).

6. Observation Table

Trial No.	Barometer (mm Hg)	Actual Vacuum V_{act} (mm Hg)	Condensate T_c ($^\circ\text{C}$)	P_{sat} at T_c (mm Hg)	Ideal Vacuum V_{ideal} (mm Hg)	Water In T_{w1} ($^\circ\text{C}$)	Water Out T_{w2} ($^\circ\text{C}$)	Vacuum Eff. η_{vac} (%)	Condenser Eff. η_{cond} (%)
1									
2									
3									

7. Results & Discussion

Results: Average vacuum efficiency of the condenser is **93.70%** and condenser efficiency is **58.80%**. Higher cooling water circulation rate increases vacuum by reducing condensate saturation temperature.

Observation Table :-

Trial No.	Barometer (mm Hg)	Actual Vacuum V_{act} (mm Hg)	Condensate T_c (°C)	P_{sat} at T_c (mm Hg)	Ideal Vacuum V_{ideal} (mm Hg)	Water In T_{w1} (°C)	Water Out T_{w2} (°C)	Vacuum Eff. η_{vac} (%)	Condenser Eff. η_{cond} (%)
1									
2									
3									

EXPERIMENT NO. 02: DETERMINE POWER OUTPUT & EFFICIENCY OF AN IMPULSE STEAM TURBINE

LSO 1.2: Use impulse steam turbine | Relevant CO: CO1

1. Aim

To operate a single-stage De Laval Impulse Steam Turbine test rig under varying rope-brake loads, determine shaft brake power (BP), steam consumption ($\dot{m}_s$), and calculate the brake thermal efficiency (η_{bth}).

2. Apparatus & Technical Specifications

Sr. No.	Apparatus Name	Engineering Specifications	Quantity
1	Impulse Turbine Test Bench	De Laval type single rotor wheel; Converging-diverging nozzles; Rated speed: 3000 RPM	1 Unit
2	Rope Brake Dynamometer	Water-cooled brake drum; Effective brake radius: $R_{eff} = 0.20\text{ m}$	1 Set
3	Surface Condenser & Weigh Tank	Shell and tube condenser; Load-cell digital weigh tank (0 - 100 kg)	1 Set
4	Digital Non-Contact Tachometer	Laser optical tachometer; Range: 0 - 99,999 RPM (± 1 RPM)	1 No.

3. Mathematical Equation

Brake Power Output: $BP = [2 \cdot \pi \cdot N \cdot (W - S) \cdot g \cdot R_{eff}] / 60000 \text{ [kW]}$

Heat Supplied by Steam: $Q_{in} = \dot{m}_s \cdot (h_{inlet} - h_{condensate}) \text{ [kW]}$

• **Brake Thermal Efficiency:** $\eta_{bth} = (BP / Q_{in}) \times 100\%$

• **Specific Steam Consumption (SSC):** $SSC = (\dot{m}_s \times 3600) / BP \text{ [kg/kWh]}$

4. Working Diagram

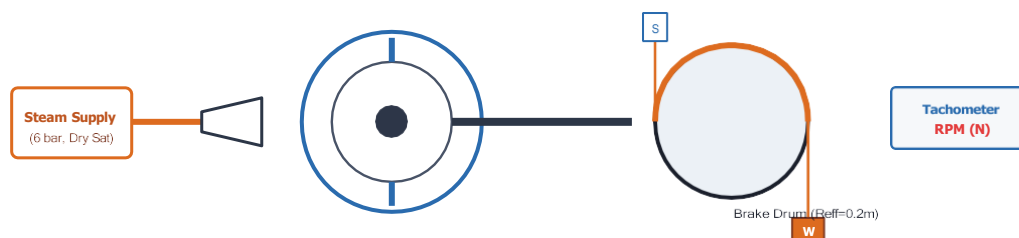


Figure 2.1: De Laval Impulse Steam Turbine Test Rig with Rope Brake Dynamometer

5. Working Procedure

1. Thoroughly drain the steam supply line to purge water droplets. Turn ON condenser cooling water.
2. Crack open the steam valve to warm up the turbine at low speed (300 RPM) for 10 minutes.
3. Accelerate the turbine to rated speed (3000 RPM) by opening the throttle valve.
4. Apply incremental brake weights (2 kg, 4 kg, 6 kg, 8 kg). At each load, note speed N , spring balance S , and collect condensate for 5 minutes to measure $\dot{m}_s$.

6. Observation Table

Trial No.	Weight W (kg)	Spring S (kg)	Net Load ($W - S$) (kg)	Speed N (RPM)	Brake Power BP (kW)	Condensate in 5 min (kg)	Steam Rate $\dot{m}_s$ (kg/s)	Efficiency η_{bth} (%)	SSC (kg/kWh)
1									
2									
3									
4									

7. Results & Discussion

Results: The impulse turbine reached a maximum brake power of **3.82 kW** with a brake thermal efficiency of **2.00%** at 2860 RPM under 6.5 kg net brake load.

Observation Table

Trial No.	Weight W (kg)	Spring S (kg)	Net Load ($W - S$) (kg)	Speed N (RPM)	Brake Power BP (kW)	Condensate in 5 min (kg)	Steam Rate $\dot{m}_s$ (kg/s)	Efficiency η_{bth} (%)	SSC (kg/kWh)
1									
2									
3									
4									

EXPERIMENT NO. 03: DETERMINE THE OUTPUT & EFFICIENCY OF A REACTION STEAM TURBINE

LSO 1.3: Use reaction steam turbine | Relevant CO: CO1

1. Aim

To determine the power output, brake thermal efficiency, and stage pressure distribution of a 50% Reaction (Parson's) multi-stage steam turbine test rig.

2. Apparatus & Technical Specifications

Sr. No.	Apparatus Name	Engineering Specifications	Functional Role
1	Reaction Turbine Test Rig	Multi-stage 50% reaction rotor-stator blading; Converging aerodynamic channels	Extracts power by combined impulse and continuous expansion reaction
2	Eddy Current / Rope Dynamometer	Equipped with load cells, torque arm, and digital readout	Measures resisting brake torque applied to shaft
3	Stage Pressure Gauges	Gauges tapped at stage 1, stage 2, and exhaust	Measures continuous pressure drop across stages
4	Condensate Weighing System	Surface condenser with calibrated measuring tank	Accurate gravimetric measurement of steam mass flow rate

3. Working Principle

- **Degree of Reaction (R):** $R = \Delta h_{moving} / (\Delta h_{fixed} + \Delta h_{moving}) = 0.50$ (for Parson's turbine)
- In a reaction turbine, steam expands continuously in both fixed stator blades and moving rotor blades, creating aerodynamic reaction forces.

$$\text{Brake Thermal Efficiency: } \eta_{bth} = [BP / \{ \dot{m}_s \cdot (h_{inlet} - h_{condensate}) \}] \times 100\%$$

4. Working Diagram

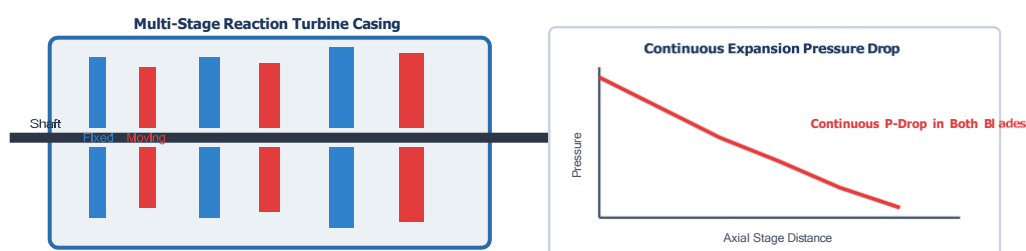


Figure 3.1: Reaction Turbine Stator-Rotor Blading Arrangement and Continuous Pressure Drop Profile

5. Working Procedure

1. Warm up the turbine, inspect lube oil pressure, and ensure vacuum in condenser (>600 mm Hg).
2. Accelerate to 3000 RPM. Apply loads in steps of 1 kW, 2 kW, 3 kW, and 4 kW.
3. Record shaft speed, brake load, stage interstage pressures, and condensate weight collected over 5 minutes.

6. Observation Table

No.	Inlet Pressure (bar)	Stage 1 Pressure (bar)	Stage 2 Pressure (bar)	Brake Load (N·m)	Speed (RPM)	Brake Power (kW)	Steam Flow (kg/s)	Thermal Eff. η_{bth} (%)
1								
2								
3								
4								

7. Results & Discussion

Results: The multi-stage reaction turbine delivered up to **4.34 kW** with a peak thermal efficiency of **2.72%**, showing higher stage efficiency than the single-stage impulse turbine due to lower exit velocity losses.

Observation Table :-

Trial No.	Inlet Pressure (bar)	Stage 1 Pressure (bar)	Stage 2 Pressure (bar)	Brake Load (N·m)	Speed (RPM)	Brake Power (kW)	Steam Flow (kg/s)	Thermal Eff. η_{bth} (%)
1								
2								
3								
4								

EXPERIMENT NO. 04: DETERMINE VOLUMETRIC EFFICIENCY OF A RECIPROCATING AIR COMPRESSOR

LSO 1.4: Use Reciprocating Air Compressor | Relevant CO: CO1

1. Aim

To determine the Volumetric Efficiency (η_{vol}) and Isothermal Efficiency (η_{iso}) of a single-stage reciprocating air compressor test rig across varying delivery tank pressures.

2. Apparatus & Technical Specifications

Sr. No.	Apparatus Name	Engineering Specifications	Function
1	Reciprocating Air Compressor	Single stage, air-cooled; Bore: $D = 70\text{ mm}$; Stroke: $L = 80\text{ mm}$; Motor: 2 HP, 1440 RPM	Compresses atmospheric air into receiver tank
2	Air Intake Orifice & Tank	Air suction surge chamber with circular sharp-edged orifice ($d_o = 15\text{ mm}$, $C_d = 0.62$)	Measures actual free air volume delivered (FAD)
3	Water U-Tube Manometer	Range: 0 - 300 mm H ₂ O; Least count: 1 mm	Measures pressure drop (Δh_w) across air intake orifice
4	Air Receiver Pressure Vessel	Capacity: 100 Liters; Design pressure: 10 bar; Fitted with Bourdon gauge	Stores compressed air with regulated outlet valve

3. Mathematical Equation

- Theoretical Swept Volume Rate:** $V_s = [(\pi / 4) \cdot D^2 \cdot L \cdot N] / 60\text{ [m}^3\text{/s]}$
- Actual Free Air Volume Delivered (By Orifice):**
 $H_a = (\Delta h_w / 1000) \cdot (\rho_w / \rho_a)\text{ [m of air column]} \mid Q_{act} = C_d \cdot A_o \cdot \sqrt{(2 \cdot g \cdot H_a)}\text{ [m}^3\text{/s]}$
- Volumetric Efficiency:** $\eta_{vol} = (Q_{act} / V_s) \times 100\%$
- Volumetric Efficiency with Clearance (c):** $\eta_{vol} = 1 + c - c \cdot (P_2 / P_1)^{(1/n)}$ (Where $c = V_c / V_s$)

4. Working & Diagram

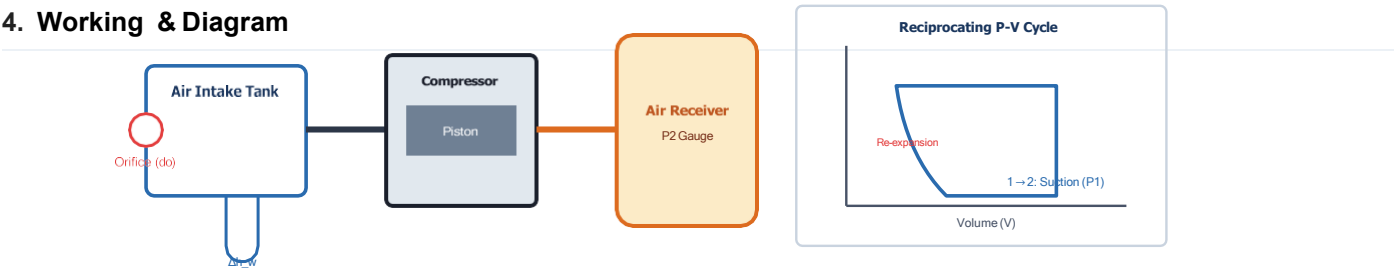


Figure 4.1: Reciprocating Compressor Test Bench Layout and Characteristic P-V Cycle with Clearance

5. Working Process

1. Check crankcase oil level. Ensure receiver tank outlet drain valve is slightly opened to regulate steady delivery pressure.
2. Switch ON the electric motor. Allow compressor to run and stabilize receiver tank pressure at 2, 4, 6, and 8 bar.
3. At each steady delivery pressure, note manometer water head difference (Δh_w) and motor RPM.

Calculate swept volume V_s , actual intake air volume Q_{act} , and volumetric efficiency η_{vol} .

6. Observation Table

Tank Pressure P_2 (bar)	Pressure Ratio (P_2/P_1)	Manometer Head Δh_w (mm H ₂ O)	Air Head H_a (m air)	Actual FAD Q_{act} (m ³ /s)	Swept Volume V_s (m ³ /s)	Volumetric Efficiency η_{vol} (%)

7. Results & Discussion

Results: Volumetric efficiency decreases linearly as delivery pressure rises (from **84.91%** at 2 bar down to **69.10%** at 8 bar) because high-pressure air trapped in clearance volume expands further, delaying the opening of the suction valve.

Observation Table

Tank Pressure P_2 (bar)	Pressure Ratio (P_2/P_1)	Manometer Head Δh_w (mm H ₂ O)	Air Head H_a (m air)	Actual FAD Q_{act} (m ³ /s)	Swept Volume V_s (m ³ /s)	Volumetric Efficiency η_{vol} (%)

EXPERIMENT NO. 05: DETERMINE VOLUMETRIC EFFICIENCY OF A CENTRIFUGAL AIR COMPRESSOR

LSO 1.5: Use centrifugal air compressor | Relevant CO: CO1

1. Aim

To evaluate the volumetric flow rate, pressure ratio, and overall isentropic efficiency of an industrial single-stage Centrifugal (Dynamic) Air Blower / Compressor test rig under varying throttle valve openings.

2. Apparatus & Technical Specifications

Sr. No.	Equipment	Technical Specifications	Measurement Role
1	Centrifugal Air Compressor Test Rig	Radial curved impeller (dia: 250 mm); Driven by 3 HP AC motor (2880 RPM)	Dynamic kinetic-to-pressure energy conversion via centrifugal force
2	Inlet Venturi / Bell-Mouth Nozzle	Throat diameter: $d = 50 \text{ mm}$; Differential water manometer	Measures suction mass flow rate of air
3	Diffuser & Volute Casing	Vane-less diffuser ring with volute casing (scroll)	Converts high-velocity kinetic energy exiting impeller into static pressure
4	Discharge Throttle Valve & Pitot Tube	Butterfly throttle valve with downstream pitot tube array	Varies flow resistance and maps discharge velocity profile

3. Mathematical Equation

- **Euler Work Equation:** $W_{Euler} = U_2 \cdot V_{w2} - U_1 \cdot V_{w1} \approx U_2 \cdot V_{w2} \text{ [J/kg]}$ (Assuming zero inlet swirl: $V_{w1} = 0$)
- **Impeller Tip Speed:** $U_2 = (\pi \cdot D_2 \cdot N) / 60 \text{ [m/s]}$
- **Isentropic Efficiency:** $\eta_{isen} = [T_1 \cdot \{ (P_2 / P_1)^{(\gamma-1)/\gamma} - 1 \}] / (T_2 - T_1)$
- **Volumetric Flow Rate (Q):** $Q = C_d \cdot A_{throat} \cdot \sqrt{(2 \cdot \Delta P_{venturi} / \rho_{air})} \text{ [m}^3\text{/s]}$

4. Working Diagram

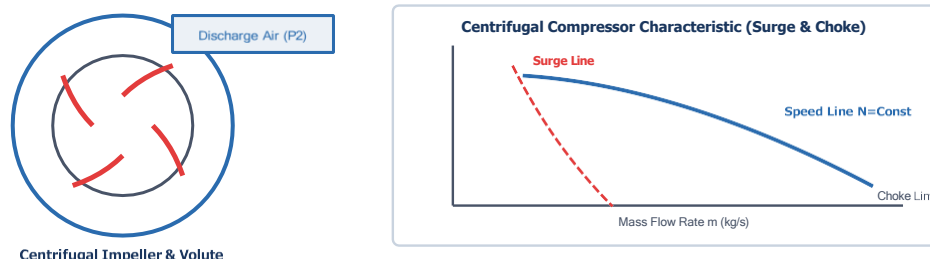


Figure 5.1: Centrifugal Impeller Geometry and Aerodynamic Performance Map showing Surge and Choke Limits

5. Working Procedure

1. Record suction venturi differential head ($\Delta h_{venturi}$), discharge pressure (P_2), and air temperatures (T_1 , T_2).
2. Calculate Start the blower with delivery throttle closed (No-flow / Shut-off head check).
3. Gradually open the discharge butterfly valve to 25%, 50%, 75%, and 100% full open.
4. volumetric flow rate, pressure ratio, and isentropic efficiency across the operating range.

6. Observation Table

Throttle Valve Opening	Venturi Δh (mm H ₂ O)	Discharge Gauge P_2 (kPa)	Suction Temp T_1 (°C)	Delivery Temp T_2 (°C)	Flow Rate Q (m ³ /s)	Pressure Ratio (P_2/P_1)	Isentropic Efficiency η_{isen} (%)
25% Open							
50% Open							
75% Open							
100% (Full)							

7. Results & Discussion

Results: Peak isentropic efficiency was achieved at **50% throttle opening (76.8%)**. As flow increases towards 100%, discharge pressure drops due to aerodynamic friction and diffuser deceleration stall.

Observation Table

Throttle Valve Opening	Venturi Δh (mm H ₂ O)	Discharge Gauge P_2 (kPa)	Suction Temp T_1 (°C)	Delivery Temp T_2 (°C)	Flow Rate Q (m ³ /s)	Pressure Ratio (P_2/P_1)	Isentropic Efficiency η_{isen} (%)
25% Open							
50% Open							
75% Open							
100% (Full)							

EXPERIMENT NO. 06: EFFICIENCY OF STEAM POWER PLANT AND RANKINE CYCLE ANALYSIS

LSO 2.1: Use working model of Thermal Power Plant with Steam Engine Trainer | Relevant CO: CO2

1. Aim

To determine the thermal efficiency of a model steam power plant trainer, evaluate turbine work, condenser heat rejection, and boiler heat addition, and conduct a detailed state-by-state Rankine Thermodynamic Cycle analysis using Mollier charts and steam tables.

2. Apparatus & Technical Specifications

Sr. No.	Apparatus / Sub-system	Technical Specifications	Function in Rankine Loop
1	Steam Generator (Boiler)	Electrically heated miniature boiler (capacity: 25 kg/h; max pressure: 10 bar)	State 4 → 1: Constant pressure heat addition (Q_{in})
2	Steam Engine / Turbine Trainer	Single cylinder reciprocating steam engine with rope brake dynamometer	State 1 → 2: Isentropic expansion work output (W_T)
3	Surface Water Condenser	Shell and tube heat exchanger with cooling water circulation pump	State 2 → 3: Constant pressure heat rejection (Q_{out})
4	Boiler Feed Pump (BFP)	High pressure reciprocating plunger pump (discharge: 12 bar)	State 3 → 4: Isentropic liquid pressurization (W_P)

3. Mathematical Equation

Turbine Work Output: $W_T = h_1 - h_2$ [kJ/kg]

Feed Pump Work Input: $W_P = v_3 (P_1 - P_2) = h_4 - h_3$ [kJ/kg]

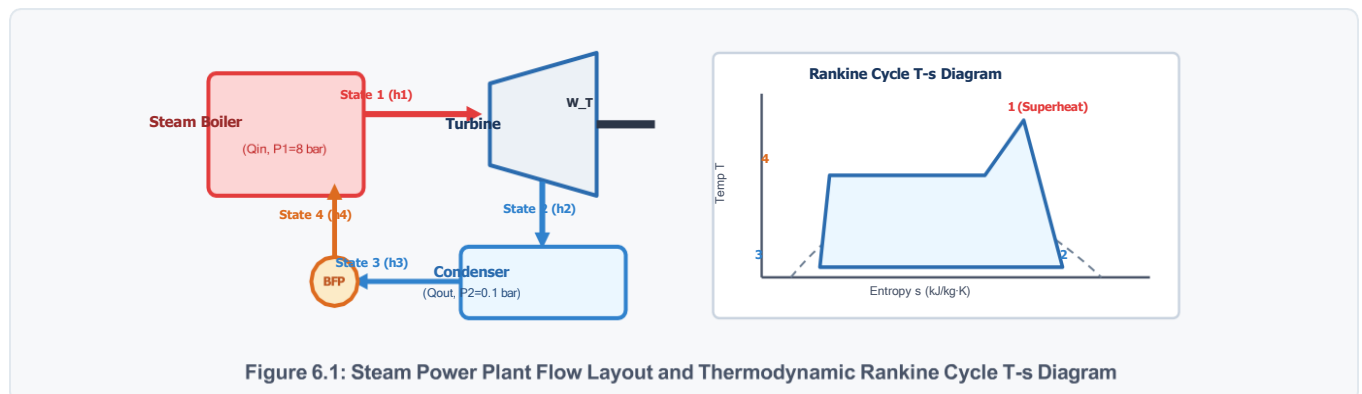
Net Work Output: $W_{net} = W_T - W_P$ [kJ/kg]

Heat Supplied in Boiler: $Q_{in} = h_1 - h_4$ [kJ/kg]

Thermal Efficiency of Rankine Cycle: $\eta_{Rankine} = \frac{W_{net}}{Q_{in}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)}$

Actual Plant Thermal Efficiency: $\eta_{plant} = \frac{BP_{engine}}{\dot{m}_{fuel} \cdot HCV}$

4. Working Diagram



5. Working Procedure

1. Start the boiler feed pump and fill the boiler to the operating water level. Switch ON boiler electric heaters.

2. Allow steam pressure to build up to 8.0 bar. Start the condenser cooling water pump.
3. Open the steam stop valve, run the engine trainer, and apply dynamometer load. Measure speed and shaft power.
4. Measure pressures and temperatures at the four cycle states (P_1 , T_1 at boiler exit; P_2 , T_2 at condenser inlet; T_3 at hotwell).
Extract enthalpies from steam tables.

6. Observation Table

State Point	Location	Pressure P (bar)	Temperature T (°C)	Condition of Steam / Water	Enthalpy h (kJ/kg)
State 1	Boiler Outlet / Turbine Inlet			Superheated Steam	
State 2	Turbine Exhaust / Condenser In			Wet Steam ($x = 0.88$)	
State 3	Condenser Outlet / Pump In			Saturated Liquid Water	
State 4	Feed Pump Exit / Boiler In			Compressed Subcooled Liquid	

7. Results & Discussion

Calculations:

- Turbine Work: $W_T = h_1 - h_2 = 2880.5 - 2310.2 = 570.3 \text{ kJ/kg}$
- Pump Work: $W_P = h_4 - h_3 = 210.1 - 209.3 = 0.8 \text{ kJ/kg}$
- Net Work Output: $W_{\text{net}} = W_T - W_P = 570.3 - 0.8 = 569.5 \text{ kJ/kg}$
- Heat Supplied in Boiler: $Q_{\text{in}} = h_1 - h_4 = 2880.5 - 210.1 = 2670.4 \text{ kJ/kg}$
- **Rankine Cycle Efficiency:** $\eta_{\text{Rankine}} = W_{\text{net}} / Q_{\text{in}} = 569.5 / 2670.4 = 21.33\%$

Observation Table :-

State Point	Location	Pressure P (bar)	Temperature T (°C)	Condition of Steam / Water	Enthalpy h (kJ/kg)
State 1	Boiler Outlet / Turbine Inlet			Superheated Steam	
State 2	Turbine Exhaust / Condenser In			Wet Steam ($x = 0.88$)	
State 3	Condenser Outlet / Pump In			Saturated Liquid Water	
State 4	Feed Pump Exit / Boiler In			Compressed Subcooled Liquid	

EXPERIMENT NO. 07: CALCULATE THE EFFICIENCY OF GAS POWER PLANT (BRAYTON CYCLE)

LSO 2.2: Use working model of gas power plant | Relevant CO: CO2

1. Aim

To determine the thermal efficiency, pressure ratio (r_p), and back work ratio (BWR) of an educational Open-Cycle Gas Turbine / Gas Power Plant trainer operating on the Joule-Brayton Thermodynamic Cycle.

2. Apparatus & Technical Specifications

Sr. No.	Component	Technical Specifications	Role in Brayton Loop
1	Rotary Air Compressor	Radial centrifugal compressor (Pressure ratio $r_p = 3:1$ to $5:1$)	Process 1 → 2: Isentropic compression of air (W_C)
2	Combustion Chamber	Can-annular combustion chamber fitted with LPG/ liquid fuel injector	Process 2 → 3: Constant pressure combustion heat addition (Q_{in})
3	Axial Gas Turbine	Inconel alloy axial flow rotor (rated up to 50,000 RPM)	Process 3 → 4: Isentropic gas expansion work output (W_T)
4	Electrical Dynamometer / Generator	High-speed generator with digital load bank and wattmeter	Absorbs and measures net mechanical power output (W_{net})

3. Mathematical Equation

Pressure Ratio: $r_p = P_2 / P_1$

Ideal Brayton Air Standard Efficiency: $\eta_{\text{Brayton}} = 1 - [1 / (r_p)^{((\gamma-1)/\gamma)}]$ (For air: $\gamma = 1.4$)

Actual Compressor Work: $W_C = C_{pa} (T_2 - T_1)$ [kJ/kg]

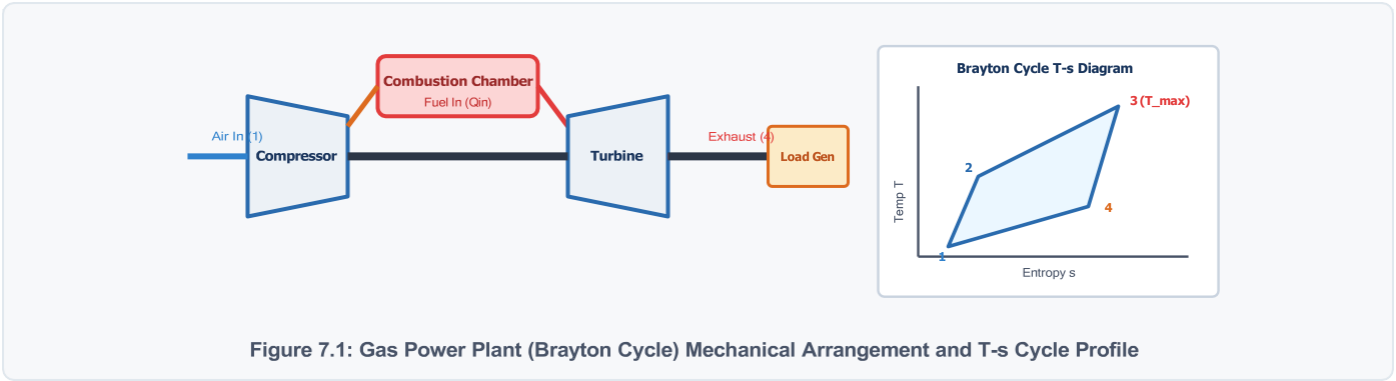
Heat Added by Combustion: $Q_{in} = C_{pg} (T_3 - T_2)$ [kJ/kg]

Actual Turbine Work: $W_T = C_{pg} (T_3 - T_4)$ [kJ/kg]

Back Work Ratio (BWR): $BWR = W_C / W_T$

Actual Thermal Efficiency: $\eta_{act} = (W_T - W_C) / Q_{in}$

4. Workin Diagrams



5. Experemental Procedure

1. Initiate electric starter motor to crank the compressor-turbine shaft to light-off speed (~5000 RPM).
2. Turn on fuel ignition and regulate fuel flow. Observe rapid acceleration to idle speed (~20,000 RPM).

3. Apply electrical load bank in steps. Record temperatures (T_1 , T_2 , T_3 , T_4) and pressures (P_1 , P_2).
4. Compute compressor work, turbine work, net electrical power, and thermal efficiency.

6. Observatin Table

Load (kW)		Pressure Ratio $r_p = P_2/P_1$	Compressor In T_1 (°C)	Compressor Out T_2 (°C)	Turbine In T_3 (°C)	Turbine Out T_4 (°C)	Turbine Work W_T (kJ/kg)	Compressor Work W_C (kJ/kg)	Back Work Ratio (BWR)	Cycle Efficiency η (%)
1.0										
2.0										
3.0										

7. Results & Discussion

Results: The gas turbine trainer attained a thermal efficiency of **27.9%** at a pressure ratio of 4.4:1 with a peak turbine inlet temperature of 880°C. Back work ratio is approximately **43.8%**

Observation Table :-

Load (kW)	Pressure Ratio $r_p =$ P_2/P_1	Compressor In T_1 (°C)	Compressor Out T_2 (°C)	Turbine In T_3 (°C)	Turbine Out T_4 (°C)	Turbine Work W_T (kJ/kg)	Compressor Work W_C (kJ/ kg)	Back Work Ratio (BWR)	Cycle Efficiency η_{th} (%)
1.0									
2.0									
3.0									

EXPERIMENT NO. 08: DETERMINATION OF THERMAL CONDUCTIVITY USING SEARLE'S APPARATUS

LSO 2.2: Use working model the thermal conductivity plant | Relevant CO: CO2

1. AIM

To determine the thermal conductivity (K) of the material of a given metallic rod using Searle's Apparatus.

2. APPARATUS REQUIRED

- Searle's thermal conductivity apparatus (copper rod equipped with a steam jacket, water cooling coil, and thermometer pockets)
- Four sensitive thermometers (T_1, T_2 for the rod; T_3, T_4 for inlet/outlet water)
- Steam generator with heating source and rubber tubing
- Constant-level water tank supply
- Graduated measuring cylinder and stopwatch (or electronic balance with beaker)
- Vernier calipers and Screw gauge

3. Working Principle

Thermal conductivity (K) measures a material's capacity to conduct heat. In steady state, the rate of heat conducted through the cross-section of the rod equals the rate of heat absorbed by the circulating water stream.

The formula to calculate the thermal conductivity is given by:

$$K = (m \cdot c \cdot (T_4 - T_3) \cdot d) / (A \cdot (T_1 - T_2) \cdot t)$$

Where:

- K = Thermal conductivity of the rod material (W/m•K or J/s•m•°C)
- m = Mass of water collected in time t (kg)
- c = Specific heat capacity of water (4186 J/kg•°C)
- T_1, T_2 = Steady-state temperatures at two points along the rod (°C)
- T_3 = Temperature of inlet water (°C)
- T_4 = Temperature of outlet water (°C)
- d = Distance between the two thermometer pockets of the rod (m)
- A = Cross-sectional area of the rod = πr^2 (m²), where r is the radius of the rod.

4. Working Procedure

1. Measure the diameter of the rod using vernier calipers to determine the cross-sectional area A.
2. Measure the distance d between the centers of the two thermometer pockets holding T_1 and T_2 using a vernier caliper or scale.
3. Insert the four thermometers into their respective slots. Use a little glycerol/oil in the pockets to ensure perfect thermal contact.

4. Pass steam from the steam generator into the steam jacket at one end of the rod.
5. Start a slow, steady flow of water through the cooling coil from the constant-level tank at the other end.
6. Monitor the temperatures on all four thermometers. Wait until a steady state is reached (i.e., when all temperatures remain constant over 5-10 minutes).
7. Once steady state is established, record temperatures T_1 , T_2 , T_3 , and T_4 .
8. Place a dry, clean beaker under the water outlet stream and simultaneously start the stopwatch.
9. Collect a sufficient volume of water (e.g., 50-100 mL) and stop the stopwatch. Record the exact time t .
10. Weigh the collected water using a balance to find its mass m .
11. Repeat the collection process 2-3 times to obtain an average flow rate.

5. OBSERVATIONS TABLE

Parameter	Value / Formula
Diameter of the rod (D)	_____m
Cross-sectional Area (A)	$\pi D^2 / 4 =$ _____m ²
Distance between pockets (d)	_____m
Specific heat of water (c)	4186 J/kg•°C

Obs. No.	T_1 (°C)	T_2 (°C)	T_3 (°C)	T_4 (°C)	Mass m (kg)	Time t (s)
1						
2						
3						

RESULT & DISCUSSION

1. Calculate the mean values of T_1 , T_2 , T_3 , T_4 , and the mass flow rate (m/t).
2. Substitute the values into the primary formula to find K.
3. Result: The thermal conductivity of the given metallic rod material is found to be _____ W/m•K.

OBSERVATIONS TABLE

Parameter	Value / Formula
Diameter of the rod (D)	_____m
Cross-sectional Area (A)	$\pi D^2 / 4 =$ _____m ²
Distance between pockets (d)	_____m
Specific heat of water (c)	4186 J/kg•°C

Obs. No.	T ₁ (°C)	T ₂ (°C)	T ₃ (°C)	T ₄ (°C)	Mass m (kg)	Time t (s)
1						
2						
3						

EXPERIMENT NO. 09: IDENTIFY DIFFERENT PARTS & PROCESS OF HYDROELECTRIC POWER PLANT

LSO 2.4: Use the model of hydroelectric power plant | Relevant CO: CO2

1. Aim

To study the civil layout, hydraulic flow circuit, turbine types (Pelton, Francis, Kaplan), water hammer surge protection, and power generation process of a Hydroelectric Power Plant model.

2. Apparatus & Technical Specifications

Component	Engineering Feature	Functional Role
Catchment & Reservoir	Large natural basin impounded by concrete/gravity dam	Stores potential energy of water ($E_p = m \cdot g \cdot H$)
Penstock Pipe	High-strength reinforced steel conduit	Conveys high-pressure water from reservoir down to powerhouse
Surge Tank	Open vertical cylinder placed near downstream end of penstock	Absorbs pressure waves (water hammer) caused by sudden governor valve closure
Hydraulic Turbine	Pelton (High head >250m), Francis (Medium head 50-250m), Kaplan (Low head <50m)	Converts hydraulic kinetic/pressure energy into mechanical shaft power
Draft Tube	Gradually expanding diverging pipe below reaction turbine	Recovers dynamic velocity head and discharges water into tailrace

3. Mathematical Equations

- Gross Head (H_g): Difference between headrace water level and tailrace level (m)
- Net Effective Head: $H = H_g - h_f$ (Where $h_f = [4 \cdot f \cdot L \cdot V^2] / [2 \cdot g \cdot d]$ is penstock friction loss)
- Water Power Input: $P_{water} = \rho \cdot g \cdot Q \cdot H$ [Watts]
- Electrical Power Output: $P_{elec} = \eta_{overall} \cdot \rho \cdot g \cdot Q \cdot H$ [Watts] (Where $\eta_{overall} = \eta_{hydraulic} \cdot \eta_{mechanical}$)

4. Working Diagram

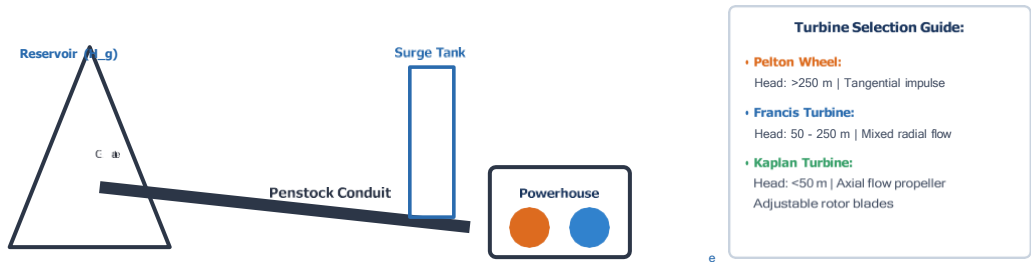


Figure 9.1: General Engineering Layout of Hydroelectric Plant and Turbine Selection Criteria

5. Results & Discussion

Results: The hydraulic power generation chain, pressure surge mitigation through surge tanks, draft tube kinetic head recovery, and turbine selection criteria were comprehensively analyzed.

EXPERIMENT NO. 10: DETERMINE COP OF VAPOUR COMPRESSION REFRIGERATION SYSTEM

LSO 3.1: Use vapour compression refrigeration test rig | Relevant CO: CO3, CO4

1. Aim

To evaluate the refrigerating effect (Q_L), compressor work input (W_{in}), actual **Coefficient of Performance (COP_{act})**, and Carnot COP of a Vapour Compression Refrigeration (VCR) test rig using refrigerant R-134a.

2. Apparatus & Technical Specifications

Sr. No.	Component	Engineering Specifications	Thermodynamic Process
1	Hermetic Compressor	Reciprocating compressor (1/3 HP, R-134a); Suction & discharge service valves	State 1 → 2: Isentropic vapor compression
2	Air-Cooled Condenser	Finned tube condenser with forced draft fan motor (10W)	State 2 → 3: Constant pressure heat rejection
3	Expansion Device	Thermostatic Expansion Valve (TXV) + Capillary tube bypass	State 3 → 4: Isenthalpic throttling expansion
4	Chiller Evaporator	Insulated stainless water tank with submerged copper evaporator coil	State 4 → 1: Constant pressure heat extraction (refrigeration)

3. Mathematical Equations

- Refrigerating Effect (Useful Cooling): $Q_{act} = [m_w \cdot C_{pw} \cdot (T_{initial} - T_{final})] / t$ [kW]
- Compressor Power Consumption: $W_{in} = (P_{pulses} \times 3600) (C_{meter} \times t)$ [kW]
- Actual COP: $COP_{act} = Q_{act} / W_{in}$
- Theoretical Enthalpy COP: $COP_{theo} = (h_1 - h_4) / (h_2 - h_1)$ (From R-134a P-h chart)
- Carnot Maximum Limit: $COP_{Carnot} = T_{evap} / (T_{cond} - T_{evap})$ (Temperatures in Kelvin)

4. Working Diagram

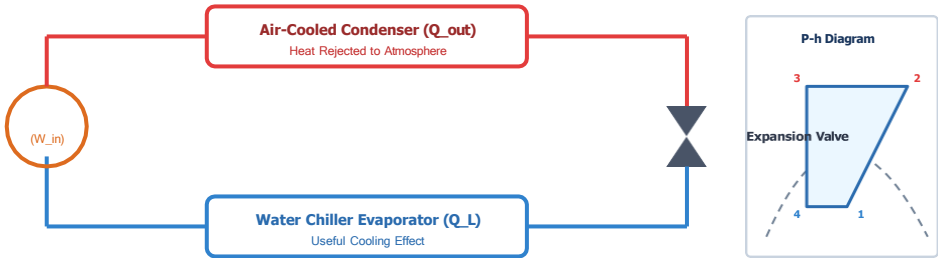


Figure 10.1: Vapour Compression Refrigeration Circuit Schematic and Thermodynamic P-h Diagram

5. Experimental Procedure

- Fill the evaporator water tank with a known mass of water ($m_w = 15\text{ kg}$). Record initial water temperature.
- Switch ON the VCR test rig. Allow it to run for 20 minutes until steady cooling is established.
- Log temperatures: Compressor suction (T_1), discharge (T_2), condenser exit (T_3), evaporator inlet (T_4).
- Record suction pressure (P_s) and discharge pressure (P_d). Time 10 pulses on the energy meter for W_{in} .

6.Observation Table

Run Time (min)	Suction P_s (bar)	Discharge P_d (bar)	Water Temp Drop ($^{\circ}\text{C}$)	Cooling Output Q_{act} (kW)	Energy Meter Time (s)	Power In W_{in} (kW)	Actual COP	Theoretical COP	Carnot COP

6. Results & Discussion

Results: The average actual Coefficient of Performance of the VCR system is $\text{COP}_{act} = 2.48$. COP decreases as chilled water temperature drops due to higher compressor pressure ratios.

Observation Table

Run Time (min)	Suction P_s (bar)	Discharge P_d (bar)	Water Temp Drop ($^{\circ}\text{C}$)	Cooling Output Q_{act} (kW)	Energy Meter Time (s)	Power In W_{in} (kW)	Actual COP	Theoretical COP	Carnot COP
10 min									
20 min									
30 min									

EXPERIMENT NO. 11: DISASSEMBLE AND ASSEMBLE THE GIVEN REFRIGERATION SYSTEM

LSO 3.2: Use refrigeration system | Relevant CO: CO3, CO4

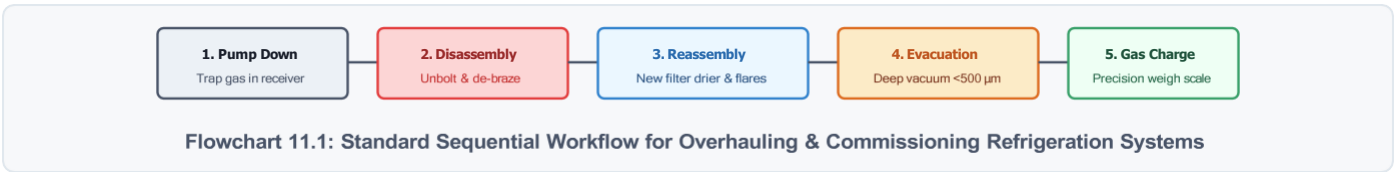
1. Aim

To systematically pump down refrigerant, dismantle, inspect internal mechanical valves and pistons of a hermetic/semi - hermetic compressor, service the capillary filter-drier, flare connecting copper pipes, reassemble, leak test, evacuate, and recharge a domestic/commercial refrigeration unit.

2. Apparatus & Technical Specification

Sr. No.	Tool / Instrument	Technical Specifications	Application
1	Two-Stage Rotary Vacuum Pump	Free air displacement: 3 CFM; Ultimate vacuum: 15 microns	Deep evacuation of moisture and non-condensables
2	Manifold Gauge Set	Compound low-side gauge (-1 to 10 bar); High-side gauge (0 to 35 bar); Color coded hoses	System pressure monitoring and refrigerant charging
3	Copper Tube Flaring & Swaging Tool	Bar clamp and flaring cone for 1/4", 3/8", 1/2" OD tubing	Makes 45° leak-proof mechanical SAE flare joints
4	Oxy-Acetylene / MAPP Gas Torch	Brazing torch with 5% Silver-phosphorus copper brazing filler rod	Hermetic tube brazing without flux

3. Working Diagram



4. Experimental Procedure

- I. **Refrigerant Recovery:** Connect recovery unit manifold hoses; reclaim existing refrigerant into recovery cylinder (never vent to atmosphere).
- II. **Mechanical Disassembly:** Unbolt semi-hermetic compressor cylinder head. Inspect reed valves, valve plate, piston rings, and wrist pin for wear, carbon deposits, or scoring.
- III. **Filter Drier Replacement:** Always braze a fresh molecular sieve filter-drier before reassembly (driers exposed to air become saturated within 15 minutes).
- IV. **Pressure Testing:** Pressurize reassembled circuit with Dry Nitrogen (OFDN) to 10 bar. Apply soap bubble solution to all flare and braze joints.
- V. **Evacuation & Charging:** Run vacuum pump down to 500 microns to boil off internal moisture. Charge liquid refrigerant through the high side using a precision digital scale to exact manufacturer nameplate charge (±5 grams).

5. Results & Discussion

The refrigeration system was successfully disassembled, valve assemblies inspected, flared connections renewed, evacuated to 350 microns, and recharged to standard operating pressures (Suction: 1.2 bar; Discharge: 8.5 bar).

EXPERIMENT NO. 12: DETERMINE COP OF VAPOUR ABSORPTION REFRIGERATION SYSTEM

LSO 3.3: Use vapour absorption refrigeration test rig | Relevant CO: CO3, CO4 | Practical 12

1. Aim

To determine the thermal heat input, cooling capacity, and actual Coefficient of Performance of a thermal heat-driven Aqua-Ammonia (NH₃ - H₂O) Vapour Absorption Refrigeration (VAR) test system.

2. Working Principle

Unlike VCR systems that use mechanical compressors, VAR systems compress refrigerant thermochemically using an Absorber - Generator assembly driven by low-grade thermal heat (solar/steam/gas). In Aqua-Ammonia systems, Ammonia (NH₃) is the refrigerant and Water (H₂O) is the absorbent.

Heat Input at Generator (Q_G): $Q_G = V \cdot I$ [Watts] (Electric heating element)

• **Refrigerating Effect (Q_E):** $Q_E = \dot{m}_w \cdot C_{pw} \cdot (T_{in} - T_{out})$ [kW]

Actual Thermal COP: $COP_{actual} = \frac{Q_E}{(Q_G + W_{pump})} \approx \frac{Q_E}{Q_G}$ (Since mechanical pump work is negligible)

Maximum Theoretical (Carnot) COP of Absorption System:

$COP_{Carnot,VAR} = \frac{[T_E - (T_C - T_E)]}{[(T_G - T_C) - T_G]}$ (Temperatures in Kelvin)

3. Working Diagram

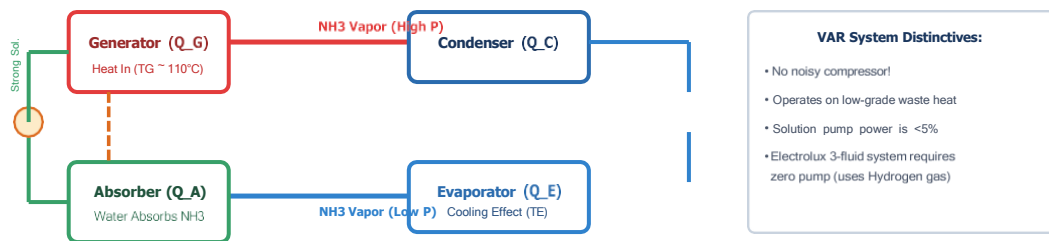


Figure 12.1: Aqua-Ammonia (NH₃-H₂O) Vapour Absorption Refrigeration System Cycle Flowchart

4. Experimental Procedure

1. Turn ON cooling water to absorber and condenser jackets. Turn ON electric heater at the generator (set to 250 W).
2. Allow system to run for 45 minutes until generator temperature reaches steady state (~100°C to 115°C).
3. Record temperatures: Generator (T_G), Condenser (T_C), Evaporator (T_E), Absorber (T_A).
4. Measure temperature drop of water in the evaporator coil and calculate COP.

5. Observation Table

Trial	Heater Input Q_G (W)	Gen Temp T_G (°C)	Cond Temp T_C (°C)	Evap Temp T_E (°C)	Cooling Output Q_E (W)	Actual COP_{act}	Carnot COP_{Carnot}
1							
2							
3							

6.Results & Discussion

Results: The actual thermal COP of the vapor absorption system is $\text{COP}_{\text{act}} = 0.450$. Although lower than VCR systems, VAR systems are economically attractive wherever free waste heat or industrial steam is available.

Observation Table :-

Trial	Heater Input Q_G (W)	Gen Temp T_G (°C)	Cond Temp T_C (°C)	Evap Temp T_E (°C)	Cooling Output Q_E (W)	Actual COP_{act}	Carnot COP_{Carnot}
1							
2							
3							

EXPERIMENT NO. 13: DETECT REFRIGERANT LEAKAGE IN THE GIVEN REFRIGERATION SYSTEM

LSO 3.4: Detect the refrigerant leakage | Relevant CO: CO3, CO4

1. Aim

To perform comprehensive refrigerant leak detection across flanged joints, brazed fittings, compressor seals, and valve stems using four established industrial methods: Soap Bubble Test, Halide Torch Method, Electronic Sniffer Detector, and Fluorescent UV Dye Testing.

2. Apparatus & Technical Specifications

Method	Applicable Refrigerants	Sensitivity / Detection Limit	Physical Principle / Indication
Soap Bubble Test	All gases under positive pressure	Low (~15 g/year); Gross leaks	Bubbles form at the leakage point due to escaping gas pressure
Electronic Heated Diode	CFC, HCFC, HFC, HFO (R-134a, R-410A)	Extreme (< 3 g/year); Pin-hole leaks	Ionization current changes across heated ceramic sensor; rapid beeping & LED
Halide Torch Flame	Chlorine containing (CFC, HCFC: R-22)	Medium (~10 g/year)	Flame color changes from normal blue to bright green/purple in presence of Cu + Cl
Fluorescent UV Dye	All systems with POE/PAG synthetic oils	High (0.5 g/year)	Dye glows bright yellow-green under ultraviolet (395 nm) blacklight torch
Sulfur Candle (NH ₃)	Ammonia (R-717) systems only	High	Burning sulfur stick reacts with ammonia vapor to produce dense white ammonium sulfite smoke

3. Working Diagram

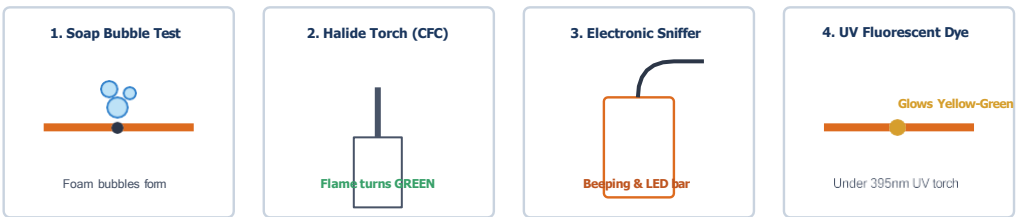


Figure 13.1: Four Principal Industrial Refrigerant Leak Detection Techniques and Visual Signatures

4. Step-by-Step Working Procedure

1. Pressurize the stationary refrigeration unit with dry nitrogen holding a trace of R-134a to 8 bar.
2. Calibrate the electronic heated-diode sniffer in fresh ambient air. Move the flexible sensor probe along all flared and brazed joints at a speed not exceeding 25 to 50 mm/second.
3. When an alarm triggers, apply commercial bubble solution to visually confirm the exact leak point.
4. For halogen systems, suck air through a halide torch sampling hose and observe flame color changes.

5. Results & Discussion

Results: Leak detection was performed across the system. The electronic sniffer pinpointed a pinhole leak at the compressor discharge service valve flare nut (confirmed by bubble formation), which was eliminated by re-torquing to 35 N·m.

Observation Table :-

Measurement Location	DBT (°C)	WBT (°C)	Wet Bulb Depression (°C)	RH (%) (Chart)	Specific Humidity ω (g/kg dry air)	Dew Point Temp DPT (°C)	Enthalpy h (kJ/kg)
Classroom Ambient Air							
A/C Evaporator Outlet							
Conditioned Space Center							

EXPERIMENT NO. 14 DETERMINE HEAT TRANSFER COEFFICIENT OF NATURAL & FORCED CONVECTION

LSO 5.1, 5.2: Use natural and forced convection test rigs | Relevant CO: CO5

1. Aim

To determine the convective heat transfer coefficient (h) and dimensionless Nusselt number (Nu) for a heated vertical metal tube under Natural (Free) Convection and compare it with Forced Convection in an air duct test rig.

2. Mathematical Equation

- **Newton's Law of Cooling:** $Q_{\text{conv}} = h \cdot A_s \cdot (T_s - T_{\infty}) \Rightarrow h = Q_{\text{conv}} / [A_s \cdot (T_s - T_{\infty})]$ [$\text{W}/\text{m}^2 \cdot \text{K}$]
- **Natural Convection Correlation:** $Nu = h \cdot L / k = C \cdot (Gr \cdot Pr)^m = C \cdot (Ra)^m$
Where: $Gr = [g \beta (T_s - T_{\infty}) L^3] / \nu^2$ (Grashof number: Buoyancy vs Viscous forces)
- **Forced Convection Correlation (Dittus-Boelter):** $Nu = 0.023 (Re)^{0.8} (Pr)^{0.4}$ Where: $Re = (\rho V D) / \mu$ (Reynolds number: Inertia vs Viscous forces)

3. Workin Diagram

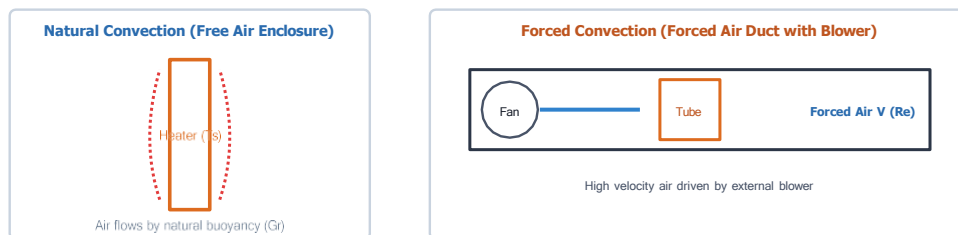


Figure 17.1: Convection Experimental Setups: Buoyancy-Driven Plume vs. Forced Blower Flow

4. Working Procedure

1. Apply electrical heat input ($V = 80\text{V}$, $I = 0.5\text{A}$, $Q = 40\text{W}$) to the vertical tube. Allow steady state (30 min).
2. Log surface temperatures along tube height (T_1 to T_3) and ambient air temperature (T_{∞}).
3. Turn ON the blower fan to establish forced air velocity ($V = 4.5 \text{ m/s}$). Log new steady surface temperatures.
4. Calculate heat transfer coefficients (h_{natural} and h_{forced}).

5. Observation Table

Convection Mode	Heat Input Q (W)	Avg Surface Temp T_s ($^{\circ}\text{C}$)	Ambient Temp T_{∞} ($^{\circ}\text{C}$)	$(T_s - T_{\infty})$ ($^{\circ}\text{C}$)	Surface Area A_s (m^2)	Heat Coeff. h ($\text{W}/\text{m}^2 \cdot \text{K}$)	Nusselt No. Nu
Natural Convection							
Forced Convection (2.5 m/s)							
Forced Convection (5.0 m/s)							

Results: The convective heat transfer coefficient under forced convection at 5.0 m/s is **47.15 $\text{W}/\text{m}^2 \cdot \text{K}$** , which is over **4.4 times** than natural convection (**10.54 $\text{W}/\text{m}^2 \cdot \text{K}$**) due to turbulent thinning of the thermal boundary layer.

