# GATE SOLVED PAPER - ME

## THERMODYNAMICS

YEAR 2013 ONE MARK

Q. 1 A cylinder contains $5\text{ m}^3$ of an ideal gas at a pressure of 1 bar . This gas is compressed in a reversible isothermal process till its pressure increases to 5 bar. The work in kJ required for this process is

(A) 804.7 (B) 953.2

(C) 981.7 (D) 1012.2

YEAR 2013 TWO MARKS

Q. 2 Specific enthalpy and velocity of steam at inlet and exit of a steam turbine, running under steady state, are as given below:

| | Specific Enthalpy ^kJ /kgh | Velocity ^m / sh |
|---|---|---|
| Inlet steam condition | 3250 | 180 |
| Exit steam condition | 2360 | 5 |

The rate of heat loss from the turbine per kg of steam flow rate is 5 kW . Neglecting changes in potential energy of steam, the power developed in kW by the steam turbine per kg of steam flow rate is

(A) 901.2 (B) 911.2

(C) 17072.5 (D) 17082.5

Q. 3 The pressure, temperature and velocity of air flowing in a pipe are 5 bar , 500 K and 50 m / s, respectively. The specific heats of air at constant pressure and at constant volume are 1.005 kJ /kg K and 0.718 kJ /kg K , respectively. Neglect potential energy. If the pressure and temperature of the surrounding are 1 bar and 300 K, respectively, the available energy in kJ /kg of the air stream is

(A) 170 (B) 187

(C) 191 (D) 213

**Common Data For Q. 4 and 5**

In a simple Brayton cycle, the pressure ratio is 8 and temperatures at the entrance of compressor and turbine are 300 K and 1400 K, respectively. Both compressor and gas turbine have isentropic efficiencies equal to 0.8. For the gas, assume a constant value of $c_p$ (specific heat at constant pressure) equal to 1 kJ /kg-K and ratio of specific heats as 1.4. Neglect changes in kinetic and potential energies.

Q. 4 The power required by the compressor in kW /kg of gas flow rate is

(A) 194.7 (B) 243.4

(C) 304.3 (D) 378.5

Q. 5 The thermal efficiency of the cycle in percentage (%) is

(A) 24.8

(B) 38.6

(C) 44.8

(D) 53.1

**YEAR 2012** **ONE MARK**

Q. 6 Steam enters an adiabatic turbine operating at steady state with an enthalpy of 3251.0 kJ /kg and leaves as a saturated mixture at 15 kPa with quality (dryness fraction) 0.9. The enthalpies of the saturated liquid and vapour at 15 kPa are $h_f = 225.94$ kJ /kg and $h_g = 2598.3$ kJ /kg respectively. The mass flow rate of steam is 10 kg/ s. Kinetic and potential energy changes are negligible. The power output of the turbine in MW is

(A) 6.5 (B) 8.9

(C) 9.1 (D) 27.0

Q. 7 A ideal gas of mass $m$ and temperature $T_1$ undergoes a reversible isothermal process from an initial pressure $p_1$ to final pressure $p_2$. The heat loss during the process is $Q$ . The entropy change $\Delta s$ of the gas is

(A) $mR\ln\left(\frac{p_2}{p_1}\right)$ (B) $mR\ln\left(\frac{p_1}{p_2}\right)$

(C) $mR\ln\left(\frac{p_2}{p}\right) - \frac{Q}{T}$ (D) zero

**YEAR 2012** **TWO MARKS**

## Common Data For Q. 8 and 9

Air enters an adiabatic nozzle at 300 kPa , 500 K with a velocity of 10 m / s. It leaves the nozzle at 100 kPa with a velocity of 180 m / s. The inlet area is 80 $cm^2$. The specific heat of air $c_p$ is 1008 J / kgK.

Q. 8 The exit temperature of the air is

(A) 516 K (B) 532 K

(C) 484 K (D) 468 K

Q. 9 The exit area of the nozzle in $cm^2$ is

(A) 90.1 (B) 56.3

(C) 4.4 (D) 12.9

**YEAR 2011** **ONE MARK**

Q. 10 Heat and work are

(A) intensive properties

(B) extensive properties

(B) point functions

(D) path functions

Q. 11 The contents of a well-insulated tank are heated by a resistor of $23\ \Omega$ in which 10 A current is flowing. Consider the tank along with its contents as a thermodynamic system. The work done by the system and the heat transfer to the system are positive. The rates of heat (Q), work (W) and change in internal energy ($DU$) during the process in kW are

(A) $Q = 0, W = -2.3, DU = +2.3$ (B) $Q = +2.3, W = 0, DU + 2.3$

(C) $Q = -2.3, W = 0, DU = -2.3$ (D) $Q = 0, W = +2.3, DU = -2.3$

YEAR 2011 TWO MARKS

Q. 12 The values of enthalpy of steam at the inlet and outlet of a steam turbine in a Rankine cycle are 2800 kJ/kg and 1800 kJ/kg respectively. Neglecting pump work, the specific steam consumption in kg/ kW hour is

(A) 3.60 (B) 0.36

(C) 0.06 (D) 0.01

Q. 13 The crank radius of a single-cylinder I.C. engine is 60 mm and the diameter of the cylinder is 80 mm. The swept volume of the cylinder in $cm^3$ is

(A) 48 (B) 96

(C) 302 (D) 603

Q. 14 An ideal Brayton cycle, operating between the pressure limits of 1 bar and 6 bar, has minimum and maximum temperature of 300 K and 1500 K. The ratio of specific heats of the working fluid is 1.4. The approximate final temperatures in Kelvin at the end of compression and expansion processes are respectively

(A) 500 and 900

(B) 900 and 500

(C) 500 and 500

(D) 900 and 900

## Common Data For Q. 15 and 16

In an experimental set up, air flows between two stations P and Q adiabatically. The direction of flow depends on the pressure and temperature conditions maintained at P and Q. The conditions at station P are 150 kPa and 350 K. The temperature at station Q is 300 K.

The following are the properties and relations pertaining to air :

Specific heat at constant pressure, $c_p = 1.005$ kJ/kgK ;

Specific heat at constant volume, $c_v = 0.718$ kJ/kgK ;

Characteristic gas constant, $R = 0.287$ kJ/kgK

Enthalpy, $h = c_p T$

Internal energy, $u = c_v T$

Q. 15 If the air has to flow from station P to station Q, the maximum possible value of pressure in kPa at station Q is close to

(A) 50 (B) 87

(C) 128 (D) 150

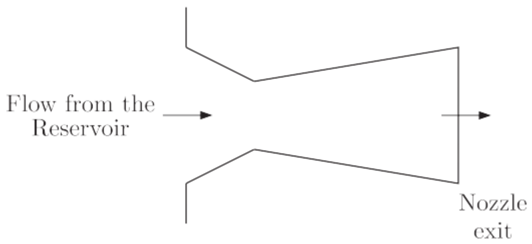
**Q. 16** If the pressure at station $Q$ is $50\,\text{kPa}$, the change in entropy $(s_Q - s_P)$ in kJ/kgK is

(A) $-0.155$ (B) 0

(C) 0.160 (D) 0.355

## Common Data For Q. 17 and 18

The temperature and pressure of air in a large reservoir are 400 K and 3 bar respectively. A converging diverging nozzle of exit area $0.005\,\text{m}^2$ is fitted to the wall of the reservoir as shown in the figure. The static pressure of air at the exit section for isentropic flow through the nozzle is $50\,\text{kPa}$. The characteristic gas constant and the ratio of specific heats of air are 0.287 kJ/kgK and 1.4 respectively.




**Q. 17** The density of air in kg/m$^3$ at the nozzle exit is

(A) 0.560 (B) 0.600

(C) 0.727 (D) 0.800

**Q. 18** The mass flow rate of air through the nozzle in kg/s is

(A) 1.30 (B) 1.77

(C) 1.85 (D) 2.06

**YEAR 2010 ONE MARK**

**Q. 19** A turbo-charged four-stroke direct injection diesel engine has a displacement volume of $0.0259\,\text{m}^3$ (25.9 litres). The engine has an output of $950\,\text{kW}$ at 2200 rpm. The mean effective pressure (in MPa) is closest to

(A) 2

(B) 1

(C) 0.2

(D) 0.1

**Q. 20** One kilogram of water at room temperature is brought into contact with a high temperature thermal reservoir. The entropy change of the universe is

(A) equal to entropy change of the reservoir

(B) equal to entropy change of water

(C) equal to zero

(D) always positive

YEAR 2010 TWO MARKS

Q. 21 A mono-atomic ideal gas ($\gamma = 1.67$, molecular weight $= 40$) is compressed adiabatically from 0.1 MPa, 300 K to 0.2 MPa. The universal gas constant is $8.314\ \text{kJkg}^{-1}\text{mol}^{-1}\text{K}^{-1}$. The work of compression of the gas (in $\text{kJkg}^{-1}$) is

(A) 29.7 (B) 19.9

(C) 13.3 (D) 0

Q. 22 Consider the following two processes ;

(a) A heat source at 1200 K loses 2500 kJ of heat to a sink at 800 K

(b) A heat source at 800 K loses 2000 kJ of heat to a sink at 500 K

Which of the following statements is true ?

(A) Process I is more irreversible than Process II

(B) Process II is more irreversible than Process I

(C) Irreversibility associated in both the processes are equal

(D) Both the processes are reversible

**Common Data For Q. 23 and 24**

In a steam power plant operating on the Rankine cycle, steam enters the turbine at 4 MPa, 350°C and exists at a pressure of 15 kPa. Then it enters the condenser and exits as saturated water. Next, a pump feeds back the water to the boiler. The adiabatic efficiency of the turbine is 90%. The thermodynamic states of water and steam are given in table.

| State | $h\,(\text{kJkg}^{-1})$ | | $s\,(\text{kJkg}^{-1}\text{K}^{-1})$ | | $\nu\,(\text{m}^3\text{kg}^{-1})$ | |
|---|---|---|---|---|---|---|
| Steam : 4 MPa, 350°C | 3092.5 | | 6.5821 | | 0.06645 | |
| Water : 15 kPa | $h_f$ | $h_g$ | $s_f$ | $s_g$ | $\nu_f$ | $\nu_g$ |
| | 225.94 | 2599.1 | 0.7549 | 8.0085 | 0.001014 | 10.02 |

$h$ is specific enthalpy, $s$ is specific entropy and $\nu$ the specific volume; subscripts $f$ and $g$ denote saturated liquid state and saturated vapor state.

Q. 23 The net work output $(\text{kJkg}^{-1})$ of the cycle is

(A) 498 (B) 775

(C) 860 (D) 957

Q. 24 Heat supplied $(\text{kJkg}^{-1})$ to the cycle is

(A) 2372 (B) 2576

(C) 2863 (D) 3092

YEAR 2009 ONE MARK

Q. 25 If a closed system is undergoing an irreversible process, the entropy of the system

(A) must increase

(B) always remains constant

(C) Must decrease

(D) can increase, decrease or remain constant

**Q. 26** A frictionless piston-cylinder device contains a gas initially at 0.8 MPa and $0.015\ m^3$. It expands quasi-statically at constant temperature to a final volume of $0.030\ m^3$. The work output (in kJ) during this process will be (A) 8.32

(B) 12.00

(C) 554.67

(D) 8320.00

**YEAR 2009 TWO MARKS**

**Q. 27** A compressor undergoes a reversible, steady flow process. The gas at inlet and outlet of the compressor is designated as state 1 and state 2 respectively. Potential and kinetic energy changes are to be ignored. The following notations are used :
$n$ = Specific volume and $p$ = pressure of the gas .
The specific work required to be supplied to the compressor for this gas compression process is

(A) $\int_1^2 p\,dn$ (B) $\int_1^2 n\,dp$

(C) $n_1(p_2-p_1)$ (D) $-p_2(n_1-n_2)$

**Q. 28** In an air-standard Otto-cycle, the compression ratio is 10. The condition at the beginning of the compression process is 100 kPa and $27^\circ$C. Heat added at constant volume is 1500 kJ /kg , while 700 kJ /kg of heat is rejected during the other constant volume process in the cycle. Specific gas constant for air = 0.287 kJ /kgK . The mean effective pressure (in kPa) of the cycle is

(A) 103 (B) 310

(C) 515 (D) 1032

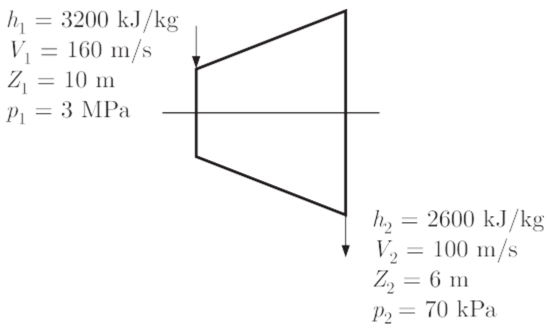
**Q. 29** An irreversible heat engine extracts heat from a high temperature source at a rate of 100 kW and rejects heat to a sink at a rate of50 kW . The entire work output of the heat engine is used to drive a reversible heat pump operating between a set of independent isothermal heat reservoirs at $17^\circ$C and $75^\circ$C. The rate (in kW) at which the heat pump delivers heat to its high temperature sink is

(A) 50 (B) 250

(C) 300 (D) 360

**Common Data For Q. 30 and 31**

The inlet and the outlet conditions of steam for an adiabatic steam turbine are as indicated in the figure. The notations are as usually followed.



**Q. 30** If mass rate of steam through the turbine is 20 kg/ s, the power output of the turbine (in MW) is

(A) 12.157

(B) 12.941

(C) 168.001

(D) 168.785

**Q. 31** Assume the above turbine to be part of a simple Rankine cycle. The density of water at the inlet to the pump is 1000 kg/ $m^3$. Ignoring kinetic and potential energy effects, the specific work (in kJ /kg ) supplied to the pump is

(A) 0.293

(B) 0.351

(C) 2.930

(D) 3.510

**YEAR 2008 ONE MARK**

**Q. 32** 2 moles of oxygen are mixed adiabatically with another 2 moles of oxygen in mixing chamber, so that the final total pressure and temperature of the mixture become same as those of the individual constituents at their initial states. The universal gas constant is given as $R$. The change in entropy due to mixing, per mole of oxygen, is given by

(A) $-R\ln 2$

(B) 0

(C) $R\ln 2$

(D) $R\ln 4$

**Q. 33** Which one of the following is NOT a necessary assumption for the air-standard Otto cycle ?

(A) All processes are both internally as well as externally reversible.

(B) Intake and exhaust processes are constant volume heat rejection processes.

(C) The combustion process is a constant volume heat addition process.

(D) The working fluid is an ideal gas with constant specific heats.

**YEAR 2008 TWO MARKS**

**Q. 34** A gas expands in a frictionless piston-cylinder arrangement. The expansion process is very slow, and is resisted by an ambient pressure of 100 kPa . During the expansion process, the pressure of the system (gas) remains constant at 300 kPa . The change in volume of the gas is 0.01 $m^3$. The maximum amount of work that could be utilized from the above process is

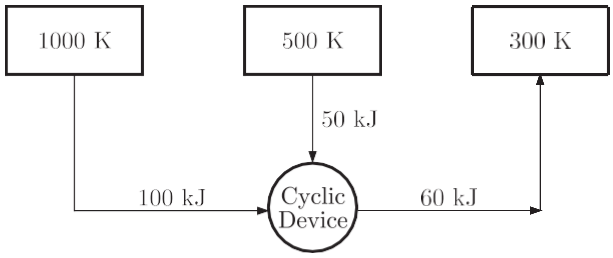
(A) 0 kJ

(B) 1 kJ

(C) 2 kJ

(D) 3 kJ

Q. 35 A cyclic device operates between three reservoirs, as shown in the figure. Heat is transferred to/ from the cycle device. It is assumed that heat transfer between each thermal reservoir and the cyclic device takes place across negligible temperature difference. Interactions between the cyclic device and the respective thermal reservoirs that are shown in the figure are all in the form of heat transfer.




The cyclic device can be

(A) a reversible heat engine

(B) a reversible heat pump or a reversible refrigerator

(C) an irreversible heat engine

(D) an irreversible heat pump or an irreversible refrigerator

Q. 36 A balloon containing an ideal gas is initially kept in an evacuated and insulated room. The balloon ruptures and the gas fills up the entire room. Which one of the following statements is TRUE at the end of above process ?

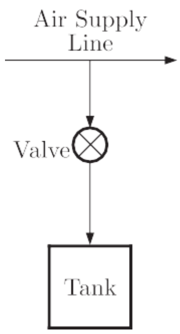
(A) The internal energy of the gas decreases from its initial value, but the enthalpy remains constant

(B) The internal energy of the gas increases from its initial value, but the enthalpy remains constant

(C) Both internal energy and enthalpy of the gas remain constant

(D) Both internal energy and enthalpy of the gas increase

Q. 37 A rigid, insulated tank is initially evacuated. The tank is connected with a supply line through which air (assumed to be ideal gas with constant specific heats) passes at 1 MPa , $350^\circ C$. A valve connected with the supply line is opened and the tank is charged with air until the final pressure inside the tank reaches 1 MPa . The final temperature inside the tank.



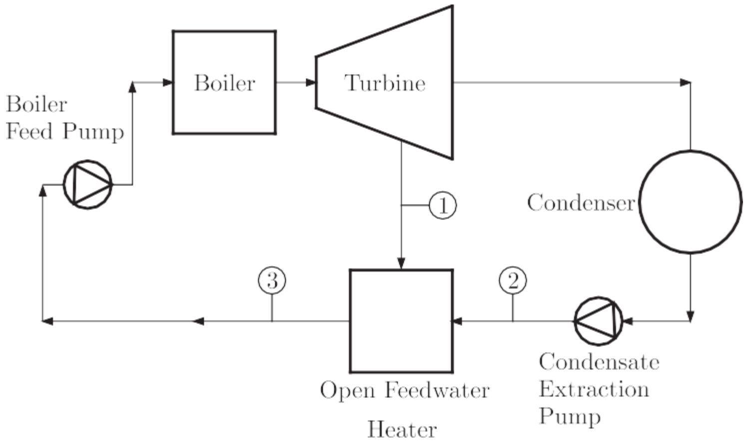

(A) is greater than $350^\circ C$

(B) is less than $350^\circ C$

(C) is equal to $350^\circ C$

(D) may be greater than, less than, or equal to, $350^\circ C$ depending on the volume of the tank

**Q. 38** A thermal power plant operates on a regenerative cycle with a single open feed water heater, as shown in the figure. For the state points shown, the specific enthalpies are: $h_1 = 2800$ kJ/kg and $h_2 = 200$ kJ/kg. The bleed to the feed water heater is 20% of the boiler steam generation rate. The specific enthalpy at state 3 is




(A) 720 kJ/kg (B) 2280 kJ/kg

(C) 1500 kJ/kg (D) 3000 kJ/kg

**Q. 39** In a steady state flow process taking place in a device with a single inlet and a single outlet, the work done per unit mass flow rate is given by $W = -\int_{inlet}^{outlet} v\,dp$, where $v$ is the specific volume and $p$ is the pressure.

The expression for $W$ given above

(A) is valid only if the process is both reversible and adiabatic

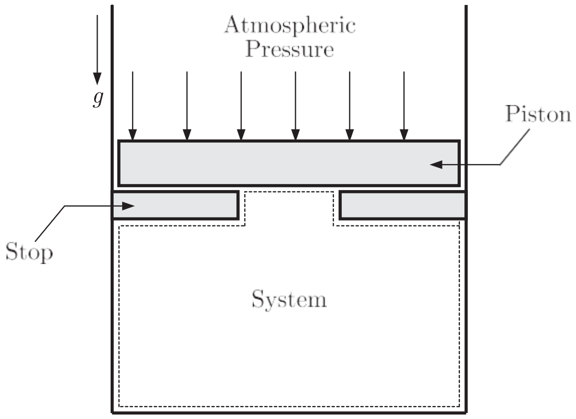
(B) is valid only if the process is both reversible and isothermal

(C) is valid for any reversible process

(D) is incorrect; it must be $W = \int_{inlet}^{outlet} p\,dv$

## Common Data For Q. 40 to 42

In the figure shown, the system is a pure substance kept in a piston-cylinder arrangement. The system is initially a two-phase mixture containing 1 kg of liquid and 0.03 kg of vapour at a pressure of 100 kPa. Initially, the piston rests on a set of stops, as shown in the figure. A pressure of 200 kPa is required to exactly balance the weight of the piston and the outside atmospheric pressure. Heat transfer takes place into the system until its volume increases by 50%. Heat transfer to the system occurs in such a manner that the piston, when allowed to move, does so in a very slow (quasi-static/ quasi-equilibrium) process. The thermal reservoir from which heat is transferred to the system has a temperature of $400^\circ$C. Average temperature of the system boundary can be taken as $175^\circ$C. The heat transfer to the system is 1 kJ, during which its entropy increases by 10 J/K.




Specific volume of liquid ($n_f$) and vapour ($n_g$) phases, as well as values of saturation temperatures, are given in the table below.

| **Pressure (kPa)** | **Saturation temperature,** $T_{sat}$ (cC) | $n_f$ ($m^3$ / kg) | $n_g$ ($m^3$ / kg) |
|---|---|---|---|
| 100 | 100 | 0.001 | 0.1 |
| 200 | 200 | 0.0015 | 0.002 |

**Q. 40** At the end of the process, which one of the following situations will be true ?

(A) superheated vapour will be left in the system

(B) no vapour will be left in the system

(C) a liquid + vapour mixture will be left in the system

(D) the mixture will exist at a dry saturated vapour state

**Q. 41** The work done by the system during the process is

(A) 0.1 kJ

(B) 0.2 kJ

(C) 0.3 kJ

(D) 0.4 kJ

**Q. 42** The net entropy generation (considering the system and the thermal reservoir together) during the process is closest to

(A) 7.5 J / K

(B) 7.7 J / K

(C) 8.5 J / K

(D) 10 J / K

## YEAR 2007 ONE MARK

**Q. 43** Which of the following relationships is valid only for reversible processes undergone by a closed system of simple compressible substance (neglect changes in kinetic and potential energy ?)

(A) $dQ = dU + dW$ (B) $Tds = dU + pdn$

(C) $Tds = dU + dW$ (D) $dQ = dU + pdn$

**Q. 44** Water has a critical specific volume of 0.003155 $m^3$/ kg. A closed and rigid steel tank of volume 0.025 $m^3$ contains a mixture of water and steam at 0.1 MPa . The mass of the mixture is 10 kg. The tank is now slowly heated. The liquid level inside the tank

(A) will rise

(B) will fall

(C) will remain constant

(D) may rise or fall depending on the amount of heat transferred

**YEAR 2007 TWO MARKS**

**Q. 45** The stroke and bore of a four stroke spark ignition engine are 250 mm and 200 mm respectively. The clearance volume is 0.001 $m^3$. If the specific heat ratio $g = 1.4$, the air-standard cycle efficiency of the engine is

(A) 46.40%

(B) 56.10%

(C) 58.20%

(D) 62.80%

**Q. 46** Which combination of the following statements is correct ?

P : A gas cools upon expansion only when its Joule-Thomson coefficient is positive in the temperature range of expansion.

Q : For a system undergoing a process, its entropy remains constant only when the process is reversible.

R : The work done by closed system in an adiabatic is a point function.

S : A liquid expands upon freezing when the slope of its fusion curve on pressure-Temperature diagram is negative.

(A) R and S

(B) P and Q

(C) Q, R and S

(D) P, Q and R

**Q. 47** Which combination of the following statements is correct ?

The incorporation of reheater in a steam power plant :

P : always increases the thermal efficiency of the plant.

Q : always increases the dryness fraction of steam at condenser inlet

R : always increases the mean temperature of heat addition.

S : always increases the specific work output.

(A) P and S

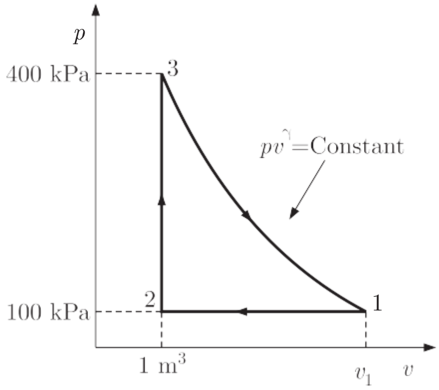
(B) Q and S

(C) P, R and S

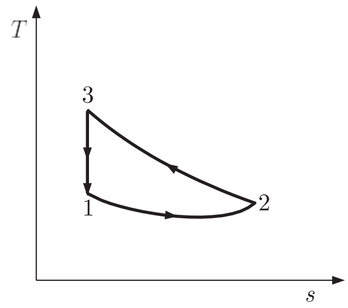
(D) P, Q, R and S

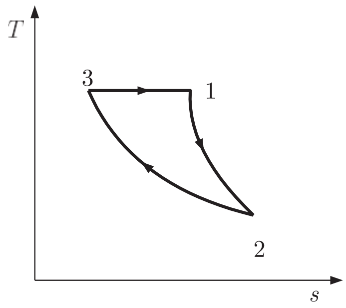
**Common Data For Q. 48 and 49**

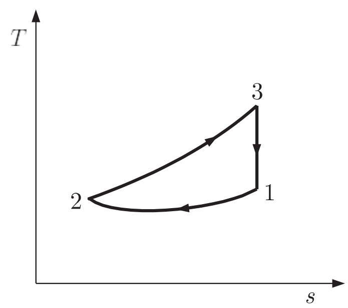
A thermodynamic cycle with an ideal gas as working fluid is shown below.

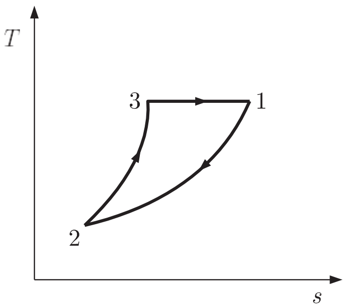



Q. 48 The above cycle is represented on $T$-$s$ plane by

(A)


(B)


(C)


(D)


Q. 49 If the specific heats of the working fluid are constant and the value of specific heat ratio is 1.4, the thermal efficiency (%) of the cycle is

(A) 21 (B) 40.9

(C) 42.6 (D) 59.7

Q. 50 A heat transformer is device that transfers a part of the heat, supplied to it at an intermediate temperature, to a high temperature reservoir while rejecting the remaining part to a low temperature heat sink. In such a heat transformer, 100 kJ of heat is supplied at 350 K. The maximum amount of heat in kJ that can be transferred to 400 K, when the rest is rejected to a heat sink at300 K is

(A) 12.50 (B) 14.29

(C) 33.33 (D) 57.14

## YEAR 2006 TWO MARKS

**Q. 51** Given below is an extract from steam tables.

| Temperature in °C | $p_{sat}$ (Bar) | Specific volume $m^3$/kg | | Enthalpy (kJ/ kg) | |
|---|---|---|---|---|---|
| | | Saturated Liquid | Saturated Vapour | Saturated Liquid | Saturated Vapour |
| 45 | 0.09593 | 0.001010 | 15.26 | 188.45 | 2394.8 |
| 342.24 | 150 | 0.001658 | 0.010337 | 1610.5 | 2610.5 |

Specific enthalpy of water in kJ / kg at 150 bar and 45°C is

(A) 203.60 (B) 200.53

(C) 196.38 (D) 188.45

**Q. 52** Determine the correctness or otherwise **Assertion (A)** and the **Reason (R)**

**Assertion (A) :** In a power plant working on a Rankine cycle, the regenerative feed water heating improves the efficiency of the steam turbine.

**Reason (R) :** The regenerative feed water heating raises the average temperature of heat addition in the Rankine cycle.

(A) Both (A) and (R) are true and (R) is the correct reason for (A)

(B) Both (A) and (R) are true but (R) is NOT the correct reason for (A)

(C) Both (A) and (R) are false

(D) (A) is false but (R) is true

**Q. 53** Determine the correctness or otherwise of the following **Assertion (A)** and the **Reason (R)**.

**Assertion (A) :** Condenser is an essential equipment in a steam power plant.

**Reason (R) :** For the same mass flow rate and the same pressure rise, a water pump requires substantially less power than a steam compressor.

(A) Both (A) and (R) are true and (R) is the correct reason for (A)

(B) Both (A) and (R) are true and (R) is NOT the correct reason for (A)

(C) Both (A) and (R) are false

(D) (A) is false but (R) is true

**Q. 54** Match items from groups I, II, III, IV and V.

| Group I | Group II | Group III | Group IV | Group V |
|---|---|---|---|---|
| | When added to the system is | Differential | Function | Phenomenon |
| E Heat | G Positive | I Exact | K Path | M Transient |
| F Work | H Negative | J Inexact | L Point | N Boundary |

| | | | |
|---|---|---|---|
| **(A)** | F-G-J-K-M<br>E-G-I-K-N | **(B)** | E-G-I-K-M<br>F-H-I-K-N |
| **(C)** | F-H-J-L-N<br>E-H-I-L-M | **(D)** | E-G-J-K-N<br>F-H-J-K-M |

**Q. 55** Group I shows different heat addition process in power cycles. Likewise, Group II shows different heat removal processes. Group III lists power cycles. Match items from Groups I, II and III.

| **Group I** | **Group II** | **Group III** |
|---|---|---|
| P. Pressure constant | S. Pressure constant | 1. Rankine Cycle |
| Q. Volume Constant | T. Volume Constant | 2. Otto cycle |
| R. Temperature constant | U. Temperature Constant | 3. Carnot cycle |
| | | 4. Diesel cycle |
| | | 5. Brayton cycle |

(A) P-S-5
R-U-3
P-S-1
Q-T-2

(B) P-S-1
R-U-3
P-S-4
P-T-2

(C) R-T-3
P-S-1
P-T-4
Q-S-5

(D) P-T-4
R-S-3
P-S-1
P-S-5

**Common Data For Q. 56 and 57**

A football was inflated to a gauge pressure of 1 bar when the ambient temperature was $15^\circ$C. When the game started next day, the air temperature at the stadium was $5^\circ$C. Assume that the volume of the football remains constant at 2500 $cm^3$.

**Q. 56** The amount of heat lost by the air in the football and the gauge pressure of air in the football at the stadium respectively equal

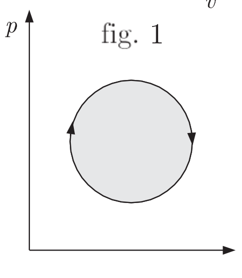
(A) 30.6 J, 1.94 bar
(B) 21.8 J, 0.93 bar
(C) 61.1 J, 1.94 bar
(D) 43.7 J, 0.93 bar

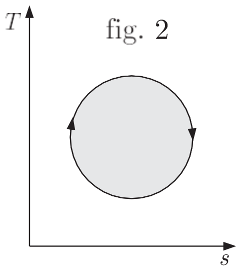
**Q. 57** Gauge pressure of air to which the ball must have been originally inflated so that it would be equal 1 bar gauge at the stadium is

(A) 2.23 bar
(B) 1.94 bar
(C) 1.07 bar
(D) 1.00 bar

**YEAR 2005** **ONE MARK**

**Q. 58** The following four figures have been drawn to represent a fictitious thermodynamic cycle, on the $p$-$v$ and $T$-$s$ planes.



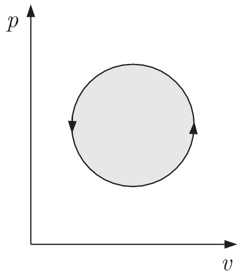



fig. 3

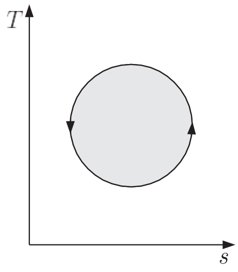


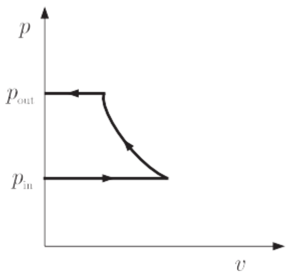

fig.4

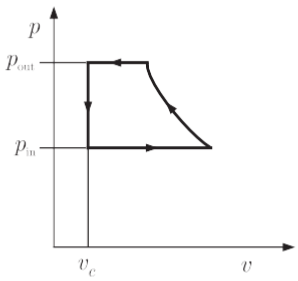
According to the first law of thermodynamics, equal areas are enclosed by

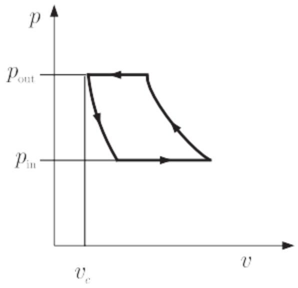
(A) figures 1 and 2 (B) figures 1 and 3

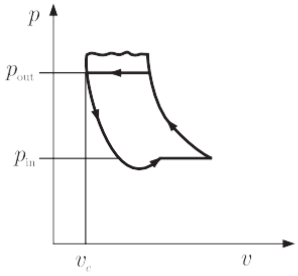
(C) figures 1 and 4 (D) figures 2 and 3

**Q. 59** A $p$-$v$ diagram has been obtained from a test on a reciprocating compressor. Which of the following represents that diagram ?

(A) 


(B) 


(C) 


(D) 


## YEAR 2005 TWO MARKS

**Q. 60** A reversible thermodynamic cycle containing only three processes and producing work is to be constructed. The constraints are

(i) there must be one isothermal process,

(ii) there must be one isentropic process,

(iii) the maximum and minimum cycle pressures and the clearance volume are fixed, and

(iv) polytropic processes are not allowed. Then the number of possible cycles are

(A) 1 (B) 2

(C) 3 (D) 4

**Q. 61** Nitrogen at an initial state of 10 bar, 1 $m^3$ and 300 K is expanded isothermally to a final volume of 2 $m^3$. The $p - v - T$ relation is $\left(p + \frac{a}{v^2}\right) v = RT$, where $a > 0$.

The final pressure.

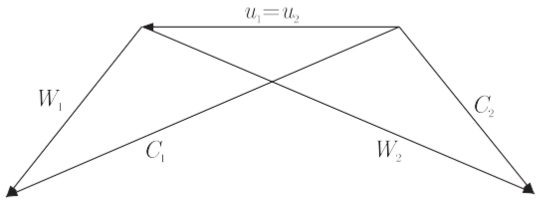
(A) will be slightly less than 5 bar

(B) will be slightly more than 5 bar

(C) will be exactly 5 bar

*(D)* cannot be ascertained in the absence of the value of $a$

**Q. 62** In the velocity diagram shown below, $u$ = blade velocity, $C$ = absolute fluid velocity and $W$ = relative velocity of fluid and the subscripts 1 and 2 refer to inlet and outlet. This diagram is for




(A) an impulse turbine

(B) a reaction turbine

(C) a centrifugal compressor

(D) an axial flow compressor

## Common Data For Q. 63 and 64

In two air standard cycles-one operating in the Otto and the other on the Brayton cycle-air is isentropically compressed from 300 to 450 K. Heat is added to raise the temperature to 600 K in the Otto cycle and to 550 K in the Brayton cycle.

**Q. 63** In $\eta_O$ and $\eta_B$ are the efficiencies of the Otto and Brayton cycles, then

(A) $\eta_O = 0.25$, $\eta_B = 0.18$

(B) $\eta_O = \eta_B = 0.33$

(C) $\eta_O = 0.5$, $\eta_B = 0.45$

(D) it is not possible to calculate the efficiencies unless the temperature after the expansion is given

**Q. 64** If $W_O$ and $W_B$ are work outputs per unit mass, then

*(A)* $W_O > W_B$

*(B)* $W_O < W_B$

*(C)* $W_O = W_B$

(D) it is not possible to calculate the work outputs unless the temperature after the expansion is given

## Common Data For Q. 65 and 66

The following table of properties was printed out for saturated liquid and saturated vapour of ammonia. The title for only the first two columns are available. All that we know that the other columns (column 3 to 8) contain data on specific properties, namely, internal energy (kJ / kg), enthalpy (kJ / kg) and entropy (kJ / kg.K)

| $t$ (°C) | p(kPa) | | | | | | |
|---|---|---|---|---|---|---|---|
| −20 | 190.2 | 88.76 | 0.3657 | 89.05 | 5.6155 | 1299.5 | 1418.0 |
| 0 | 429.6 | 179.69 | 0.7114 | 180.36 | 5.3309 | 1318.0 | 1442.2 |
| 20 | 587.5 | 272.89 | 1.0408 | 274.30 | 5.0860 | 1332.2 | 1460.2 |
| 40 | 1554.9 | 368.74 | 1.3574 | 371.43 | 4.8662 | 1341.0 | 1470.2 |

Q. 65 The specific enthalpy data are in columns

(A) 3 and 7

(B) 3 and 8

(C) 5 and 7

(D) 5 and 8

Q. 66 When saturated liquid at 40°C is throttled to −20°C, the quality at exit will be

(A) 0.189

(B) 0.212

(C) 0.231

(D) 0.788

## YEAR 2004 ONE MARK

Q. 67 A gas contained in a cylinder is compressed, the work required for compression being 5000 kJ. During the process, heat interaction of 2000 kJ causes the surroundings to be heated. The changes in internal energy of the gas during the process is

(A) −7000 kJ

(B) −3000 kJ

(C) +3000 kJ

(D) + 7000 kJ

Q. 68 The compression ratio of a gas power plant cycle corresponding to maximum work output for the given temperature limits of $T_{min}$ and $T_{max}$ will be

(A) $\left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}$

(B) $\left(\frac{T_{min}}{T_{max}}\right)^{\frac{\gamma}{2(\gamma-1)}}$

(C) $\left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma-1}{\gamma}}$

(D) $\left(\frac{T_{min}}{T_{max}}\right)^{\frac{\gamma-1}{\gamma}}$

Q. 69 At the time of starting, idling and low speed operation, the carburretor supplies a mixture which can be termed as

(A) Lean

(B) slightly leaner than stoichiometric

(C) stoichiometric

(D) rich

## YEAR 2004 TWO MARKS

Q. 70 A steel billet of 2000 kg mass is to be cooled from 1250 K to 450 K. The heat released during this process is to be used as a source of energy. The ambient temperature is 303 K and specific heat of steel is 0.5 kJ /kg K . The available energy of this billet is

(A) 490.44 MJ

(B) 30.95 MJ

(C) 10.35 MJ

(D) 0.10 MJ

**Q. 71** During a Morse test on a 4 cylinder engine, the following measurements of brake power were taken at constant speed.

| | |
|---|---|
| All cylinders firing | 3037 kW |
| Number 1 cylinder not firing | 2102 kW |
| Number 2 cylinder not firing | 2102 kW |
| Number 3 cylinder not firing | 2100 kW |
| Number 4 cylinder not firing | 2098 kW |

The mechanical efficiency of the engine is

(A) 91.53% (B) 85.07%

(C) 81.07% (D) 61.22%

**Q. 72** A solar collector receiving solar radiation at the rate of 0.6 kW / $m^2$ transforms it to the internal energy of a fluid at an overall efficiency of 50%. The fluid heated to 250 K is used to run a heat engine which rejects heat at315 K. If the heat engine is to deliver 2.5 kW power, the minimum area of the solar collector required would be

(A) 83.33 $m^2$ (B) 16.66 $m^2$

(C) 39.68 $m^2$ (D) 79.36 $m^2$

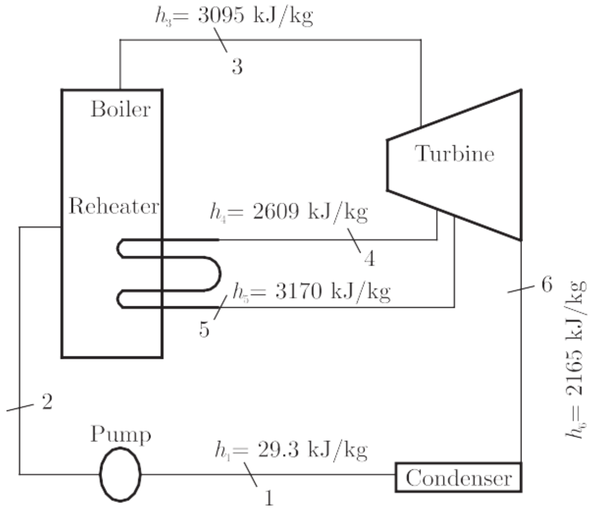
**Q. 73** An engine working on air standard Otto cycle has a cylinder diameter of 10 cm and stroke length of 15 cm. The ratio of specific heats for air is 1.4. If the clearance volume is 196.3 cc and the heat supplied per kg of air per cycle is1800 kJ /kg , the work output per cycle per kg of air is

(A) 879.1 kJ (B) 890.2 kJ

(C) 895.3 kJ (D) 973.5 kJ

## Common Data For Q. 74 and 75

Consider a steam power plant using a reheat cycle as shown . Steam leaves the boiler and enters the turbine at 4 MPa, 350c C ($h_3 = 3095$ kJ / kg). After expansion in the turbine to 400 kPa ($h_4 = 2609$ kJ / kg), and then expanded in a low pressure turbine to 10 kPa ($h_6 = 2165$ kJ / kg). The specific volume of liquid handled by the pump can be assumed to be



**Q. 74** The thermal efficiency of the plant neglecting pump work is
(A) 15.8%
(B) 41.1%
(C) 48.5%
(D) 58.6%

**Q. 75** The enthalpy at the pump discharge ($h_2$) is
(A) 0.33 kJ/kg
(B) 3.33 kJ/kg
(C) 4.0 kJ/k
(D) 33.3 kJ/kg

## YEAR 2003 ONE MARK

**Q. 76** For a spark ignition engine, the equivalence ratio ($f$) of mixture entering the combustion chamber has values
(A) $f < 1$ for idling and $f > 1$ for peak power conditions
(B) $f > 1$ for both idling and peak power conditions
(C) $f > 1$ for idling and $f < 1$ for peak power conditions
(D) $f < 1$ for both idling and peak power conditions

**Q. 77** A diesel engine is usually more efficient than a spark ignition engine because
(A) diesel being a heavier hydrocarbon, releases more heat per kg than gasoline
(B) the air standard efficiency of diesel cycle is higher than the Otto cycle, at a fixed compression ratio
(C) the compression ratio of a diesel engine is higher than that of an SI engine
(D) self ignition temperature of diesel is higher than that of gasoline

**Q. 78** In Ranking cycle, regeneration results in higher efficiency because
(A) pressure inside the boiler increases
(B) heat is added before steam enters the low pressure turbine
(C) average temperature of heat addition in the boiler increases
(D) total work delivered by the turbine increases

**Q. 79** Considering the variation of static pressure and absolute velocity in an impulse steam turbine, across one row of moving blades
(A) both pressure and velocity decreases
(B) pressure decreases but velocity increases
(C) pressure remains constant, while velocity increases
(D) pressure remains constant, while velocity decreases

**Q. 80** A 2 kW, 40 liters water heater is switched on for 20 minutes. The heat capacity $c_p$ for water is 4.2 kJ/kgK. Assuming all the electrical energy has gone into heating the water, increase of the water temperature in degree centigrade is
(A) 2.7
(B) 4.0
(C) 14.3
(D) 25.25

Q. 81 Considering the relationship $Tds = dU + pd\nu$ between the entropy ($s$), internal energy ($U$), pressure ($p$), temperature ($T$) and volume ($\nu$), which of the following statements is correct ?

(A) It is applicable only for a reversible process

(B) For an irreversible process, $Tds > dU + pd\nu$

(C) It is valid only for an ideal gas

(D) It is equivalent to I$^{st}$ law, for a reversible process

Q. 82 In a gas turbine, hot combustion products with the specific heats $c_p = 0.98$ kJ/kgK, and $c_v = 0.7538$ kJ /kgK enters the turbine at 20 bar, 1500 K exit at 1 bar. The isentropic efficiency of the turbine is 0.94. The work developed by the turbine per kg of gas flow is

(A) 689.64 kJ/kg (B) 794.66 kJ / kg

(C) 1009.72 kJ/kg (D) 1312.00 kJ / kg

Q. 83 An automobile engine operates at a fuel air ratio of 0.05, volumetric efficiency of 90% and indicated thermal efficiency of 30%. Given that the calorific value of the fuel is 45 MJ /kg and the density of air at intake is 1 kg/ m$^3$, the indicated mean effective pressure for the engine is

(A) 6.075 bar (B) 6.75 bar

(C) 67.5 bar (D) 243 bar

Q. 84 For an engine operating on air standard Otto cycle, the clearance volume is 10% of the swept volume. The specific heat ratio of air is 1.4. The air standard cycle efficiency is

(A) 38.3% (B) 39.8%

(C) 60.2% (D) 61.7%

**Common Data For Q. 85 and 86**

Nitrogen gas (molecular weight 28) is enclosed in a cylinder by a piston, at the initial condition of 2 bar, 298 K and 1 m$^3$. In a particular process, the gas slowly expands under isothermal condition, until the volume becomes 2 m$^3$. Heat exchange occurs with the atmosphere at 298 K during this process.

Q. 85 The work interaction for the Nitrogen gas is

(A) 200 kJ (B) 138.6 kJ

(C) 2 kJ (D) $-200$ kJ

Q. 86 The entropy changes for the Universe during the process in kJ / K is

(A) 0.4652 (B) 0.0067

(C) 0 (D) $-0.6711$

**YEAR 2002 ONE MARK**

Q. 87 A positive value of Joule-Thomson coefficient of a fluid means

(A) temperature drops during throttling

(B) temperature remains constant during throttling

(C) temperature rises during throttling

(D) None of the above

Q. 88 A correctly designed convergent-divergent nozzle working at a designed load is

(A) always isentropic

(B) always choked

(C) never choked

(D) never isentropic

**YEAR 2002 TWO MARKS**

Q. 89 A Carnot cycle is having an efficiency of 0.75. If the temperature of the high temperature reservoir is $727^\circ C$, what is the temperature of low temperature reservoir ?

(A) $23^\circ C$

(B) $-23^\circ C$

(C) $0^\circ C$

(D) $250^\circ C$

Q. 90 An ideal air standard Otto cycle has a compression ratio of 8.5. If the ratio of the specific heats of air $(\gamma)$ is 1.4, what is the thermal efficiency in percentage) of the Otto cycle ?

(A) 57.5

(B) 45.7

(C) 52.5

(D) 95

Q. 91 The efficiency of superheat Rankine cycle is higher than that of simple Rankine cycle because

(A) the enthalpy of main steam is higher for superheat cycle

(B) the mean temperature of heat addition is higher for superheat cycle

(C) the temperature of steam in the condenser is high

(D) the quality of steam in the condenser is low.

**YEAR 2001 ONE MARK**

Q. 92 The Rateau turbine belongs to the category of

(A) pressure compounded turbine

(B) reaction turbine

(C) velocity compounded turbine

(D) radial flow turbine

Q. 93 A gas having a negative Joule-Thomson coefficient $(\mu < 0)$, when throttled, will

(A) become cooler

(B) become warmer

(C) remain at the same temperature

(D) either be cooler or warmer depending on the type of gas

## YEAR 2001 TWO MARKS

**Q. 94** A cyclic heat engine does 50 kJ of work per cycle. If the efficiency of the heat engine is 75%, the heat rejected per cycle is

(A) $16\frac{2}{3}$ kJ (B) $33\frac{1}{3}$ kJ

(C) $37\frac{1}{2}$ kJ (D) $66\frac{2}{3}$ kJ

**Q. 95** A single-acting two-stage compressor with complete intercooling delivers air at 16 bar . Assuming an intake state of 1 bar at 15c C, the pressure ratio per stage is

(A) 16 (B) 8

(C) 4 (D) 2

**Q. 96** A small steam whistle (perfectly insulated and doing no shaft work) causes a drop of 0.8 kJ /kg in the enthalpy of steam from entry to exit. If the kinetic energy of the steam at entry is negligible, the velocity of the steam at exit is

(A) 4 m /s (B) 40 m /s

(C) 80 m /s (D) 120 m /s

**Q. 97** In a spark ignition engine working on the ideal Otto cycle, the compression ratio is 5.5. The work output per cycle (i.e., area of the $p$-$n$ diagram) is equal to $23.625 \# 10^5 \# n_c$, where $n_c$ is the clearance volume in $m^3$. The indicated mean effective pressure is

(A) 4.295 bar (B) 5.250 bar

(C) 86.870 bar (D) 106.300 bar

********

# SOLUTION

**Sol. 1** Option (A) is correct.

For Reversible isothermal Process work done is given by

$$W_{1-2} = p_1 v_1 \ln \frac{p_1}{p_2}$$

$$= 1 \times 10^5 \times 5 \times \ln\left(\frac{1}{5}\right)$$

$$= -804.7 \text{ kJ}$$

The negative sign shows that the compression process is taking place in this process.

**Sol. 2** Option (A) is correct.

From energy balance equation for steady flow system

$$E_{in} = E_{out}$$

$$h_1 + \frac{V_1^2}{2} + gz_1 + dQ = h_2 + \frac{V_2^2}{2} + gz_2 + dW$$

For negligible P.E. $gz_1 = gz_2 = 0$

or $$dW = (h_1 - h_2) + \frac{V_1^2 - V_2^2}{2 \times 1000} + dQ$$

$$= (3250 - 2360) + \frac{(180)^2 - (5)^2}{2 \times 1000} - 5$$

$$= 890 + 16.1875 - 5 = 901.2 \text{ kW/kg}$$

**Sol. 3** Option (B) is correct.

IN pipe

$p = 5$ bar $= 5 \times 10^5$ Pa, $T = 500$ K, $V = 50$ m/sec

$c_p = 1.005$ kJ/kg K, $c_v = 0.718$ kJ/kg K

For surrounding air

$p_0 = 1$ bar $= 1 \times 10^5$ Pa, $T_0 = 300$ K

Available energy function is

$$\psi = (h - h_0) - T_0(S - S_0) + \frac{V^2}{2} + gz$$

Given, the potential energy is negligible. Thus

$$\psi = (h - h_0) - T_0(S - S_0) + \frac{V^2}{2}$$

The entropy is given by

$$S = c_p \ln T - R \ln p \text{ and } h = c_p T$$

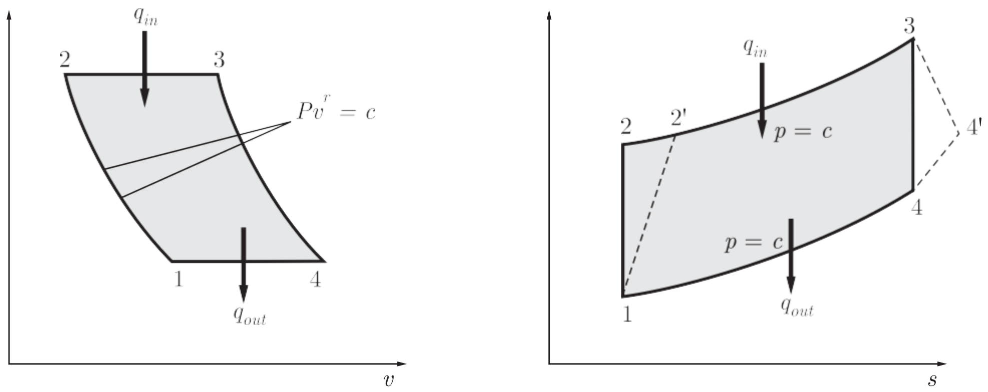
So that $$\psi = c_p(T - T_0) - T_0[c_p \ln T - R \ln p - c_p \ln T_0 + R \ln p_0] + \frac{V^2}{2}$$

$$\psi = c_p(T - T_0) - T_0\left[c_p \ln\left(\frac{T}{T_0}\right) - R \ln\left(\frac{p}{p_0}\right)\right] + \frac{V^2}{2}$$

$$= 1.005(500 - 300) - 300\left[1.005 \times \ln\left(\frac{500}{300}\right) - 0.287 \times \ln\left(\frac{5}{1}\right)\right] + \frac{(50)^2}{2 \times 1000}$$
$$= 187 \text{ kJ/kg}$$

**Sol. 4** Option (C) is correct.

The $p-v$ and $T-s$ diagram of brayton cycle is shown below:




Given $r_p = \frac{p_2}{p_1} = 8$, $\gamma = 1.4$, $T_1 = 300$ K, $T_3 = 1400$ K, $c_p = 1$ kJ/kg-K, $\eta_{isen} = 0.8$

The process 1-2 (Isentropic compression)

Process $1-2'$ (Actual compression)

Process 3-4 (Isentropic expansion)

Process $3-4'$ (Actual expansion)

For reversible adiabatic compression process $1-2$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (8)^{\frac{2}{7}}$$

or $$T_2 = 300\,(8)^{\frac{2}{7}} = 543.43 \text{ K}$$

Now $$\eta_{isen} = \frac{\text{Isentropic compressor work}}{\text{Actual compressor work}}$$

$$W_{actual} = \frac{\dot{m} c_p (T_2 - T_1)}{\eta_{isen}}$$

$$\frac{W_{net}}{\dot{m}} = \frac{1 \times (543.43 - 300)}{0.8}$$
$$= 304.3 \text{ kW/kg}$$

**Sol. 5** Option (A) is correct.

For process $2-3$ ($p$ = constant)

$$\frac{V_2}{T_2} = \frac{V_3}{T_3}$$

Heat supplied $$Q_{in} = c_p (T_3 - T_{2'})$$

Now $$\eta_{isen} = \frac{W_{actual}}{W_{isen}} = \frac{h_2 - h_1}{h_{2'} - h_1}$$
$$= \frac{c_p (T_2 - T_1)}{c_p (T_{2'} - T_1)} = \frac{T_2 - T_1}{T_{2'} - T_1}$$

or $$0.8 = \frac{543.43 - 300}{T_{2'} - 300}$$

$$0.8\,T_{2'} - 240 = 243.43$$

$$T_2' = 604.3 \text{ K}$$

So that $\quad Q_{in} = 1 \times (1400 - 604.3) = 795.7 \text{ kJ/kg}$

For process $3 - 4$ ($p$ = constant)

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

or $$T_4 = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}} = \frac{1400}{(8)^{\frac{1.4-1}{1.4}}} = 772.86 \text{ K}$$

Now $$\eta_{isen} = \frac{W_{actual}}{W_{isen}} = \frac{h_3 - h_4'}{h_3 - h_4} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.8 = \frac{1400 - T_4'}{1400 - 772.86}$$

or $$T_4' = 898.288 \text{ K}$$

Now $$W_{act} = c_p(T_3 - T_4') = 1(1400 - 898.288) = 501.712 \text{ kJ/kg}$$

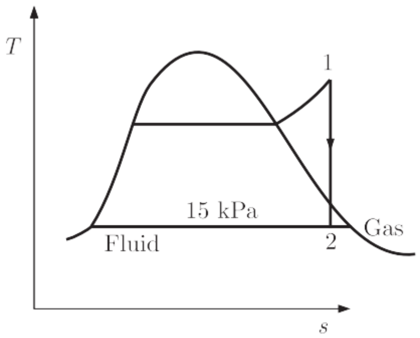
Hence $$\eta_{thermal} = \frac{W_{act} - W_{comp}}{Q_{in}}$$

$$= \left(\frac{501.712 - 304.3}{795.7}\right) \times 100$$

$$= 24.8\%$$

**Sol. 6** Option (B) is correct.

For adiabatic expansion steam in turbine.




Given $h_1 = 3251.0$ kJ/kg, $\dot{m} = 10$ kg/s, $x = 0.9$ (dryness fraction)

At 15 kPa

Enthalpy of liquid, $\quad h_f = 225.94$ kJ/kg

Enthalpy of vapour, $\quad h_g = 2598.3$ kJ/kg

Since Power output of turbine.

$$P = \dot{m}(h_1 - h_2) \quad \text{(K.E and P.E are negligible)} \qquad ...(i)$$

$$h_2 = h_f + x h_{fg} = h_f + x(h_g - h_f)$$

$$= 225.94 + 0.9\,(2598.3 - 225.94) = 2361.064 \text{ kJ/kg}$$

From Eq. (i)

$$P = 10 \times (3251.0 - 2361.064) = 8899 \text{ kW} = 8.9 \text{ MW}$$

**Sol. 7** Option (B) is correct.

We know that $\quad Tds = du + Pdv \qquad ...(i)$

For ideal gas $\quad pv = mRT$

For isothermal process

$$T = \text{constant}$$

For reversible process

$$du = 0$$

Then from equation (i)

$$ds = \frac{p d\nu}{T} = \frac{m\,RT\,d\nu}{T\,\nu} = m\,R\frac{d\nu}{\nu}$$

$$\int ds = \Delta s = mR\int_{\nu_1}^{\nu_2}\frac{d\nu}{\nu} = m\,R\ln\frac{\nu_2}{\nu_1}$$

$$\Delta s = m\,R\ln\frac{p_1}{p_2} \qquad \because \frac{p_1}{p_2} = \frac{\nu_2}{\nu_1}$$

**Sol. 8** Option (C) is correct.

From energy balance for steady flow system.

$$E_{in} = E_{out}$$

$$\dot{m}\left(h_1 + \frac{V_1^2}{2}\right) = \dot{m}\left(h_2 + \frac{V_2^2}{2}\right) \qquad \text{...(i)}$$

As $\quad h = c_p T$

Equation (1) becomes

$$c_p T_1 + \frac{V_1^2}{2} = c_p T_2 + \frac{V_2^2}{2}$$

$$T_2 = \frac{V_1^2 - V_2^2}{2 \times c_p} + T_1 = \frac{10^2 - 180^2}{2 \times 1008} + 500 = -16.02 + 500$$

$$= 483.98 \simeq 484\,\text{K}$$

**Sol. 9** Option (D) is correct.

From Mass conservation.

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

$$\frac{V_1 A_1}{\nu_1} = \frac{V_2 A_2}{\nu_2} \qquad \text{...(i)}$$

where $\quad \nu$ = specific volume of air $= \frac{RT}{p}$

Therefore Eq. (1) becomes

$$\frac{p_1 V_1 A_1}{RT_1} = \frac{p_2 V_2 A_2}{RT_2}$$

$$A_2 = \frac{p_1 \times V_1 \times A_1 \times T_2}{p_2 \times V_2 \times T_1} = \frac{300 \times 10 \times 80 \times 484}{100 \times 180 \times 500} = 12.9\ \text{cm}^2$$

**Sol. 10** Option (D) is correct.

Work done is a quasi-static process between two given states depends on the path followed. Therefore,

$$\int_1^2 dW \neq W_2 - W_1 \qquad dW \text{ shows the inexact differential}$$

But, $\quad \int_1^2 dW = W_{1-2}$ or ${}_1W_2$

So, Work is a path function and Heat transfer is also a path function. The amount of heat transferred when a system changes from state 1 to state 2 depends on the intermediate states through which the system passes i.e. the path.

$$\int_1^2 dQ = Q_{1-2} \text{ or } {}_1Q_2$$

$dQ$ shows the inexact differential. So, Heat and work are path functions.

**Sol. 11** Option (A) is correct.

Given : $R = 23$ W, $i = 10$ A

Since work is done on the system. So,

$$W_{electrical} = - i^2 R = - (10)^2 \# 23 = - 2300 \text{ W} = - 2.3 \text{ kW}$$

Here given that tank is well-insulated.

So, $DQ = 0$

Applying the First law of thermodynamics,

$$DQ = DU + DW$$
$$DU + DW = 0$$
$$DW = - DU$$

And $DU = + 2.3$ kW

Heat is transferred to the system

**Sol. 12** Option (A) is correct.

Given : $h_1 = 2800$ kJ/kg = Enthalpy at the inlet of steam turbine

$h_2 = 1800$ kJ/kg = Enthalpy at the outlet of a steam turbine

Steam rate or specific steam consumption

$$= \frac{3600}{W_T - W_p} \text{ kg / kWh}$$

Pump work $W_p$ is negligible, therefore

$$\text{Steam rate} = \frac{3600}{W_T} \text{ kg/ kWh}$$

And $W_T = h_1 - h_2$ From Rankine cycle

$$\text{Steam rate} = \frac{3600}{h_1 - h_2} \text{ kg/ kWh} = \frac{3600}{2800 - 1800} = 3.60 \text{ kg/kWh}$$

**Sol. 13** Option (D) is correct.

Given : $r = 60$ mm , $D = 80$ mm

Stroke length, $L = 2r = 2 \# 60 = 120$ mm (cylinder diameter)

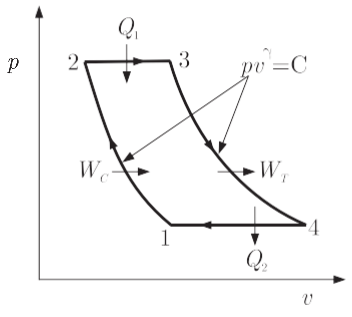
Swept Volume, $n_s = A \# L$

$$= \frac{p}{4} D^2 \# L = \frac{p}{4} (8.0)^2 \# 12.0$$

$$= \frac{p}{4} (8 \# 8) \# 12 = 602.88 - 603 \text{ cm}^3$$

**Sol. 14** Option (A) is correct.

Given $p - n$ curve shows the Brayton Cycle.




Given : $p_1 = 1$ bar $= p_4$, $p_2 = 6$ bar $= p_3$, $T_{minimum} = 300$ K, $T_{maximum} = 1500$ K

$$\frac{c_p}{c_v} = \gamma = 1.4$$

We have to find $T_2$ (temperature at the end of compression) or $T_4$ (temperature at the end of expansion)

Applying adiabatic equation for process 1-2, we get

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{6}\right)^{\frac{1.4-1}{1.4}}$$

$$\frac{300}{T_2} = \left(\frac{1}{6}\right)^{0.286} \qquad T_1 = T_{minimum}$$

$$T_2 = \frac{300}{\left(\frac{1}{6}\right)^{0.286}} = 500.5\text{ K} \simeq 500\text{ K}$$

Again applying for the Process 3-4,

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{6}\right)^{\frac{1.4-1}{1.4}} = \left(\frac{1}{6}\right)^{0.286}$$

So, $$T_4 = T_3 \times \left(\frac{1}{6}\right)^{0.286} = 1500 \times \left(\frac{1}{6}\right)^{0.286} = 900\text{ K} \qquad T_3 = T_{maximum}$$

**Sol. 15** Option (B) is correct.

Given : At station $p$ : $p_1 = 150$ kPa , $T_1 = 350$ K

At station $Q$ : $p_2 = ?$, $T_2 = 300$ K

We know, $$\gamma = \frac{c_p}{c_v} = \frac{1.005}{0.718} = 1.39$$

Applying adiabatic equation for station $P$ and $Q$ ,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma-1}} = \frac{p_1}{p_2}$$

$$p_2 = \frac{p_1}{\left(\frac{T_1}{T_2}\right)^{\frac{\gamma}{\gamma-1}}} = \frac{150}{\left(\frac{350}{300}\right)^{\frac{1.39}{1.39-1}}} = \frac{150}{1.732} = 86.60\text{ kPa} \simeq 87\text{ kPa}$$

**Sol. 16** Option (C) is correct.

Given :

Pressure at $Q$ $\quad p_2 = 50$ kPa

Using the general relation to find the entropy changes between $P$ and $Q$

$$Tds = dh - v\,dp$$

$$ds = \frac{dh}{T} - \frac{v}{T}dp \qquad \text{...(i)}$$

Given in the previous part of the question

$$h = c_p T$$

Differentiating both the sides, we get

$$dh = c_p\,dT$$

Put the value of $dh$ in equation (i),

$$ds = c_p\frac{dT}{T} - \frac{v}{T}dp \qquad \text{From the gas equation } v/T = R/p$$

So, $$= c_p\frac{dT}{T} - R\frac{dp}{p}$$

Integrating both the sides and putting the limits

$$\int_P^Q ds = c_p\int_P^Q \frac{dT}{T} - R\int_P^Q \frac{dp}{p}$$

$$(\Delta s)_P^Q = c_p (\ln T)_P^Q - R (\ln P)_P^Q$$

$$s_Q - s_P = c_p (\ln T_Q - \ln T_P) - R (\ln p_Q - \ln p_P)$$

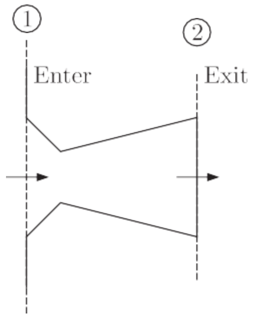
$$= c_p \ln\left(\frac{T_Q}{T_P}\right) - R \ln\left(\frac{p_Q}{p_P}\right)$$

$$= 1.005 \ln\left(\frac{300}{350}\right) - 0.287 \ln\left(\frac{50}{150}\right)$$

$$= 1.005 \times (-0.1541) - 0.287 \times (-1.099)$$

$$= 0.160 \text{ kJ/kg K}$$

**Sol. 17** Option (C) is correct.




Given : $T_1 = 400$ K, $p_1 = 3$ bar , $A_2 = 0.005$ m$^2$, $p_2 = 50$ kPa $= 0.5$ bar ,
$R = 0.287$ kJ /kg K , $\gamma = \frac{c_p}{c_v} = 1.4$, $T_2 = ?$

Applying adiabatic equation for isentropic (reversible adiabatic) flow at section (1) and (2), we get

$$\left(\frac{T_2}{T_1}\right) = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = T_1\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = 400\left(\frac{0.5}{3}\right)^{\frac{1.4-1}{1.4}}$$

$$= 400 \times (0.166)^{0.286} = 239.73 \text{ K}$$

Apply perfect Gas equation at the exit,

$$p_2 \upsilon_2 = m_2 R T_2$$

$$p_2 = \frac{m_2}{\upsilon_2} R T_2 = \rho_2 R T_2 \qquad \left(\frac{m}{\upsilon} = \rho\right)$$

$$\rho_2 = \frac{p_2}{RT_2} = \frac{50 \times 10^3}{0.287 \times 10^3 \times 239.73} = 0.727 \text{ kg/ m}^3$$

**Sol. 18** Option (D) is correct.

Given : $\rho_2 = 0.727$ kg/ m$^3$, $A_2 = 0.005$ m$^2$, $V_2 = ?$

For isentropic expansion,

$$V_2 = \sqrt{2c_p(T_1 - T_2)}$$

$$= \sqrt{2 \times 1.005 \times 10^3 \times (400 - 239.73)}$$

for air $c_p = 1.005$ kJ /kg K

$$= \sqrt{322142.7} = 567.58 \text{ m / sec}$$

Mass flow rate at exit,

$$\dot{m} = \rho_2 A_2 V_2 = 0.727 \times 0.005 \times 567.58 = 2.06 \text{ kg/sec}$$

**Sol. 19** Option (A) is correct.

Given : $v = 0.0259\ \text{m}^3$, Work output = 950 kW , $N = 2200$ rpm

Mean effective pressure

$$mep = \frac{\text{Net work for one cycle}}{\text{displacement volume}} \times 60$$

Number of power cycle

$$n = \frac{N}{2} = \frac{2200}{2} = 1100 \qquad \text{(for 4 stroke)}$$

Hence, net work for one cycle

$$= \frac{950 \times 10^3}{1100} = 863.64\ \text{W}$$

So, $$mep = \frac{60 \times 863.64}{0.0259} = 2 \times 10^6\ \text{Pa} = 2\ \text{MPa}$$

**Sol. 20** Option (D) is correct.

We know that,

Entropy of universe is always increases.

$$\Delta s_{universe} > 0$$

$$(\Delta s)_{system} + (\Delta s)_{surrounding} > 0$$

**Sol. 21** Option (A) is correct.

Given : $\gamma = 1.67$ , $M = 40$, $p_1 = 0.1\ \text{MPa} = 10^6 \times 0.1 = 10^5\ \text{Pa}$

$T_1 = 300\ \text{K}$, $p_2 = 0.2\ \text{MPa} = 2 \times 10^5\ \text{Pa}$ , $R_u = 8.314\ \text{kJ/kgmol K}$

$$\text{Gas constant} = \frac{\text{Universal Gas constant}}{\text{Molecular Weight}}$$

$$R = \frac{R_u}{M} = \frac{8.314}{40} = 0.20785\ \text{kJ/kg K}$$

For adiabatic process,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_2}{300} = \left(\frac{0.2}{0.1}\right)^{\frac{1.67-1}{1.67}} = (2)^{0.4012}$$

$$T_2 = 300 \times (2)^{0.4012} = 300 \times 1.32 = 396\ \text{K}$$

Work done in adiabatic process is given by,

$$W = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1} = \frac{R(T_1 - T_2)}{\gamma - 1}$$

$$= \frac{0.20785[300 - 396]}{1.67 - 1} = \frac{0.20785(-96)}{0.67} = -29.7\ \text{kJ/kg}$$

( Negative sign shows the compression work)

**Sol. 22** Option (B) is correct.

We know from the clausius Inequality,

If $$\oint \frac{dQ}{T} = 0, \text{ the cycle is reversible}$$

$$\oint \frac{dQ}{T} < 0, \text{ the cycle is irreversible and possible}$$

For case (a), $$\oint_a \frac{dQ}{T} = \frac{2500}{1200} - \frac{2500}{800}$$

$$= \frac{25}{12} - \frac{25}{8} = -1.041\ \text{kJ/kg}$$

For case (b),

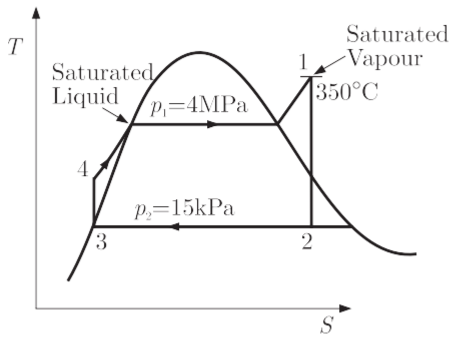
$$\oint_b \frac{dQ}{T} = \frac{2000}{800} - \frac{2000}{500} = \frac{20}{8} - \frac{20}{5} = -1.5 \text{ kJ/kg}$$

$$\oint_a \frac{dQ}{T} > \oint_b \frac{dQ}{T}$$

So, process (b) is more irreversible than process (a)

**Sol. 23** Option (C) is correct.

Given $T$ - $s$ curve is for the steam plant




Given : $p_1 = 4\,\text{MPa} = 4 \times 10^6\,\text{Pa}$, $T_1 = 350^\circ\text{C} = (273 + 350)\,\text{K} = 623\,\text{K}$

$p_2 = 15\,\text{kPa} = 15 \times 10^3\,\text{Pa}$, $\eta_{adiabatic} = 90\% = 0.9$

Now from the steam table,

Given data : $h_1 = 3092.5\,\text{kJ/kg}$, $h_3 = h_f = 225.94\,\text{kJ/kg}$, $h_g = 2599.1\,\text{kJ/kg}$

$$s_1 = s_2 = s_f + x(s_g - s_f) \qquad ...(i)$$

Where, $x$ = dryness fraction

From the table, we have

$$s_f = 0.7549\,\text{kJ/kg K}$$

$$s_g = 8.0085\,\text{kJ/kg K}$$

$$s_1 = s_2 = 6.5821$$

From equation (i), $x = \frac{s_2 - s_f}{s_g - s_f} = \frac{6.5821 - 0.7549}{8.0085 - 0.7549} = 0.8033$

$$h_2 = h_f + x(h_g - h_f) = 225.94 + 0.8033\,(2599.1 - 225.94)$$

$$= 225.94 + 1906.36 = 2132.3 \text{ kJ/kg}$$

Theoretical turbine work from the cycle is given by,

$$W_T = h_1 - h_2 = 3092.5 - 2132.3 = 960.2 \text{ kJ/kg}$$

Actual work by the turbine,

$$= \text{Theoretical work} \times \eta_{adiabatic}$$

$$= 0.9 \times 960.2 = 864.18\,\text{kJ/kg}$$

Pump work, $W_p = \nu_f (p_1 - p_2)$

$$= 0.001014\,(4000 - 15) = 4.04 \text{ kJ/kg}$$

$$W_{net} = W_T - W_p = 864.18 - 4.04 = 860.14 \text{ kJ/kg} \approx 860$$

**Sol. 24** Option (C) is correct.

Heat supplied $= h_1 - h_4$ From $T$-$s$ diagram

From the pump work equation,

$$W_p = h_4 - h_3$$

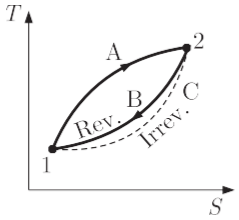
$$h_4 = W_p + h_3 = 4.04 + 225.94 = 229.98 \text{ kJ/kg}$$

And Heat supplied, $Q = h_1 - h_4$

$$= 3092.50 - 229.98 = 2862.53 \approx 2863 \text{ kJ/kg}$$

**Sol. 25** Option (A) is correct.

We consider the cycle shown in figure, where $A$ and $B$ are reversible processes and $C$ is an irreversible process. For the reversible cycle consisting of $A$ and $B$.




$$\oint_R \frac{dQ}{T} = \int_{A1}^{2} \frac{dQ}{T} + \int_{B2}^{1} \frac{dQ}{T} = 0$$

or $$\int_{A1}^{2} \frac{dQ}{T} = -\int_{B2}^{1} \frac{dQ}{T} \quad \text{...(i)}$$

For the irreversible cycle consisting of $A$ and $C$, by the inequality of clausius,

$$\oint \frac{dQ}{T} = \int_{A1}^{2} \frac{dQ}{T} + \int_{C2}^{1} \frac{dQ}{T} < 0 \quad \text{...(ii)}$$

From equation (i) and (ii)

$$-\int_{B2}^{1} \frac{dQ}{T} + \int_{C2}^{1} \frac{dQ}{T} < 0$$

$$\int_{B2}^{1} \frac{dQ}{T} > \int_{C2}^{1} \frac{dQ}{T} \quad \text{...(iii)}$$

Since the path $B$ is reversible,

$$\int_{B2}^{1} \frac{dQ}{T} = \int_{B2}^{1} ds$$

Since entropy is a property, entropy changes for the paths $B$ and $C$ would be the same. Therefore,

$$\int_{B2}^{1} ds = \int_{C2}^{1} ds \quad \text{...(iv)}$$

From equation (iii) and (iv), $$\int_{C_2}^{1} ds > \int_{C2}^{1} \frac{dQ}{T}$$

Thus, for any irreversible process, $ds > \dfrac{dQ}{T}$ So, entropy must increase.

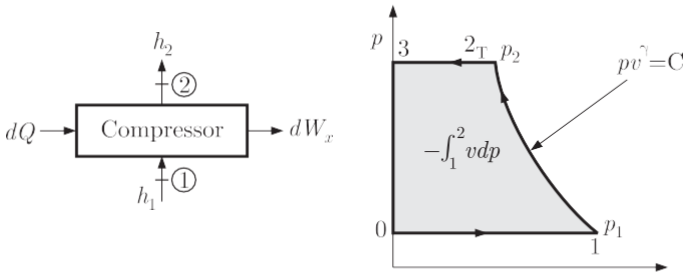
**Sol. 26** Option (A) is correct.

Given : $p_1 = 0.8$ MPa, $\upsilon_1 = 0.015 \text{ m}^3$, $\upsilon_2 = 0.030 \text{ m}^3$, $T$ = Constant

We know work done in a constant temperature (isothermal) process

$$W = p_1 \upsilon_1 \ln\left(\frac{\upsilon_2}{\upsilon_1}\right) = (0.8 \times 10^6)(0.015) \ln\left(\frac{0.030}{0.015}\right) = 8.32 \text{ kJ}$$

**Sol. 27** Option (B) is correct.




(a) (b)

Steady flow energy equation for a compressor (Fig a) gives,

$$h_1 + dQ = h_2 + dW_x \qquad ...(i)$$

Neglecting the changes of potential and kinetic energy. From the property relation

$$Tds = dh - v\,dp$$

For a reversible process, $Tds = dQ$

So, $dQ = dh - v\,dp$ ...(ii)

If consider the process is reversible adiabatic then $dQ = 0$

From equation (i) and (ii), $h_1 - h_2 = dW_x$ & $dh = h_2 - h_1 = -dW_x$ ...(iii)

And $dh = v\,dp$ ...(iv)

From equation (iii) and (iv), $-dW_x = v\,dp$

$$W_x = -\int v\,dp$$

Negative sign shows the work is done on the system (compression work) for initial and Final Stage

$$W_x = \int_1^2 v\,dp$$

**Sol. 28** Option (D) is correct.

Given : $r = 10$, $p_1 = 100$ kPa , $T_1 = 27^\circ$ C = (27 + 273) K = 300 K

$Q_s = 1500$ kJ /kg , $Q_r = 700$ kJ /kg , $R = 0.287$ kJ /kg K

Mean Effective pressure $p_m = \dfrac{\text{Net work output}}{\text{Swept Volume}}$ ...(i)

Swept volume, $v_1 - v_2 = v_2(r - 1)$

where $v_1 =$ Total volume and $v_2 =$ Clearance volume

$$r = \frac{v_1}{v_2} = 10 \qquad \& \; v_1 = 10 v_2 \qquad ...(ii)$$

Applying gas equation for the beginning process,

$$p_1 v_1 = RT_1$$

$$v_1 = \frac{RT_1}{p_1} = \frac{0.287 \times 300}{100} = 0.861 \text{ m}^3/\text{kg}$$

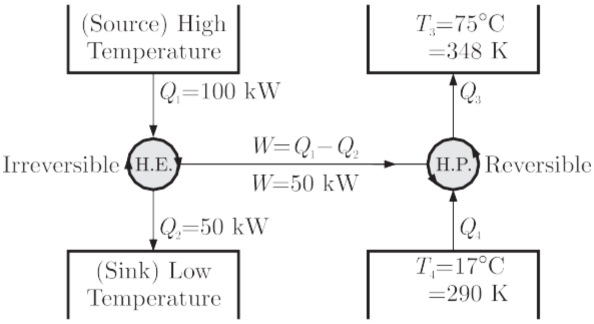
$$v_2 = \frac{v_1}{10} = \frac{0.861}{10} = 0.0861 \text{ m}^3/\text{kg}$$

$$W_{net} = Q_s - Q_r = (1500 - 700)\text{ kJ/kg K} = 800 \text{ kJ/kg K}$$

From equation (i) $p_m = \dfrac{800}{v_2(r-1)} = \dfrac{800}{0.0861(10-1)}$

$$= \frac{800}{0.7749} = 1032.391 \text{ kPa} \approx 1032 \text{ kPa}$$

**Sol. 29** Option (C) is correct.




The coefficient of performance of a Heat pump for the given system is,

$$(COP)_{H.P.} = \frac{Q_3}{Q_3 - Q_4} = \frac{Q_3}{W}$$

For a reversible process,

$$\frac{Q_3}{Q_4} = \frac{T_3}{T_4}$$

$$(COP)_{H.P.} = \frac{T_3}{T_3 - T_4} = \frac{Q_3}{W}$$

$$\frac{348}{348 - 290} = \frac{Q_3}{50}$$

$$Q_3 = \frac{348 \times 50}{58} = 300 \text{ K}$$

**Sol. 30** Option (A) is correct.

Given : $h_1 = 3200$ kJ/kg , $V_1 = 160$ m/sec , $z_1 = 10$ m

$$p_1 = 3 \text{ mpA}, \quad \dot{m} = -\frac{dM}{dt} = 20 \text{ kg/sec}$$

It is a adiabatic process, So $dQ = 0$

Apply steady flow energy equation $[S.F.E.E.]$ at the inlet and outlet section of steam turbine,

$$h_1 + \frac{V_1^2}{2} + z_1 g + \frac{dQ}{dm} = h_2 + \frac{V_2^2}{2} + z_2 g + \frac{dW}{dm}$$

$$dQ = 0$$

So $\frac{dQ}{dm} = 0$

And
$$h_1 + \frac{V_1^2}{2} + z_1 g = h_2 + \frac{V_2^2}{2} + z_2 g + \frac{dW}{dm}$$

$$\frac{dW}{dm} = (h_1 - h_2) + \left(\frac{V_1^2 - V_2^2}{2}\right) + (z_1 - z_2) g$$

$$= (3200 - 2600) \times 10^3 + \left[\frac{(160)^2 - (100)^2}{2}\right] + (10 - 6) 9.8$$

$$= 600000 + 7800 + 39.20$$

$$\frac{dW}{dm} = 607839.2 \text{ J/kg} = 607.84 \text{ kJ/kg}$$

Power output of turbine

$$P = \text{Mass flow rate} \times \frac{dW}{dm}$$

$$= 20 \times 607.84 \times 10^3 \qquad \dot{m} = 20 \text{ kg/sec}$$

$$P = 12.157 \text{ MJ/sec} = 12.157 \text{ MW}$$

**Sol. 31** Option (C) is correct.

Given : $r = 1000 \text{ kg/m}^3$

Here given that ignoring kinetic and potential energy effects, So in the steady flow energy equation the terms $V^2/2, Z_1 g$ are equal to zero and $dQ$ is also zero for adiabatic process. S.F.E.E. is reduces to,

$$h_4 = h_3 + \frac{dW_p}{dm}$$ Here, $W_p$ represents the pump work

where $h_3$ = Enthalpy at the inlet of pump and $h_4$ = Enthalpy at the outlet of the pump.

$$\frac{dW_p}{dm} = h_4 - h_3 = dh \qquad \text{...(i)}$$

For reversible adiabatic compression,

$$dQ = dh - n dp \qquad (dQ = 0)$$

$$dh = n dp \qquad \text{...(ii)}$$

From equation (i) and (ii), we get

$$\frac{dW_p}{dm} = n dp = \frac{1}{r}(p_1 - p_2) \qquad v = \frac{1}{r}$$

$$\frac{dW_p}{dm} = \frac{(3000-70)\text{kPa}}{1000} = \frac{2930}{1000} \text{ kPa} = 2.930 \text{ kPa}$$

**Sol. 32** Option (B) is correct.

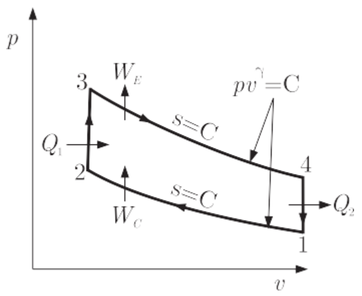
Given : $T_1 = T_2$, $p_1 = p_2$

Universal Gas constant $= R$ . Here given oxygen are mixed adiabatically

So, $dQ = 0$

We know, $ds = \frac{dQ}{T} = \frac{0}{T} = 0$

**Sol. 33** Option (B) is correct.




Assumptions of air standard otto cycle :-

(A) All processes are both internally as well as externally reversible.

(B) Air behaves as ideal gas

(C) Specific heats remains constant ($c_p$ & $c_v$)

(D) Intake process is constant volume heat addition process and exhaust process is constant volume heat rejection process.

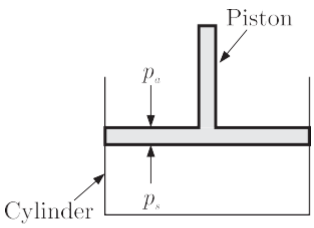
Intake process is a constant volume heat addition process, From the given options, option (2) is incorrect.

**Sol. 34** Option (C) is correct.

Given : $p_a = 100$ kPa , $p_s = 300$ kPa , $Dn = 0.01 \text{ m}^3$

Net pressure work on the system,

$$p = p_s - p_a = 300 - 100$$
$$= 200 \text{ kPa}$$




For constant pressure process work done is given by

$$W = p\Delta v = 200 \times 0.01 = 2 \text{ kJ}$$

**Sol. 35** Option (A) is correct.

A heat engine cycle is a thermodynamic cycle in which there is a net Heat transfer from higher temperature to a lower temperature device. So it is a Heat Engine.

Applying Clausius theorem on the system for checking the reversibility of the cyclic device.

$$\oint_R \frac{dQ}{T} = 0$$

$$\frac{Q_1}{T_1} + \frac{Q_2}{T_2} - \frac{Q_3}{T_3} = 0$$

$$\frac{100 \times 10^3}{1000} + \frac{50 \times 10^3}{500} - \frac{60 \times 10^3}{300} = 0$$

$$100 + 100 - 200 = 0$$

Here, the cyclic integral of $dQ / T$ is zero. This implies, it is a reversible Heat engine.

**Sol. 36** Option (C) is correct.

We know enthalpy,

$$h = U + pv \qquad \text{...(i)}$$

Where, $U$ = Internal energy

$p$ = Pressure of the room

$v$ = Volume of the room

It is given that room is insulated, So there is no interaction of energy (Heat) between system (room) and surrounding (atmosphere).

It means Change in internal Energy $dU = 0$ and $U$ = Constant

And temperature is also remains constant.

Applying the perfect gas equation,

$$pv = nRT$$
$$pv = \text{Constant}$$

Therefore, from equation (i)

$$h = \text{Constant}$$

So this process is a constant internal energy and constant enthalpy process.

**Alternate Method :**

We know that enthalpy,

$$h = U + pv$$

Given that room is insulated, So there is no interaction of Energy (Heat) between system (room) and surrounding (atmosphere).

It means internal Energy $dU = 0$ and $U =$ constant.
Now flow work $pn$ must also remain constant thus we may conclude that during free expansion process $pn$ i.e. product of pressure and specific volume change in such a way that their product remains constant.
So, it is a constant internal energy and constant enthalpy process.

**Sol. 37** Option (A) is correct.

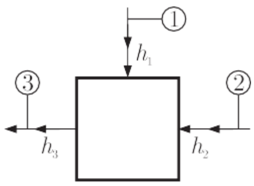
Given : $p_1 = 1$ MPa , $T_1 = 350$cC $= (350 + 273)$ K $= 623$ K
For air $g = 1.4$
We know that final temperature ( $T_2$) inside the tank is given by,

$$T_2 = gT_1 = 1.4 \# 623 = 872.2 \text{ K} = 599.2\text{cC}$$

$T_2$ is greater than 350cC.

**Sol. 38** Option (A) is correct.

Given : $h_1 = 2800$ kJ /kg , $h_2 = 200$ kJ /kg
From the given diagram of thermal power plant, point 1 is directed by the Boiler to the open feed water heater and point 2 is directed by the pump to the open feed water Heater. The bleed to the feed water heater is 20% of the boiler steam generation i.e. 20% of $h_1$



Open Feed Water Heater

So,
$$h_3 = 20\% \text{ of } h_1 + 80\% \text{ of } h_2$$
$$= 0.2 \# 2800 + 0.8 \# 200 = 720 \text{ kJ /kg}$$

**Sol. 39** Option (C) is correct.

From the first law of thermodynamic,

$$dQ = dU + dW$$
$$dW = dQ - dU \qquad ...(i)$$

If the process is complete at the constant pressure and no work is done other than the $pdn$ work. So

$$dQ = dU + pdn$$

At constant pressure $\quad pdn = d(pn)$

$$(dQ) = dU + d(pn) = d(U + pn) = (dh) \qquad h = U + pn$$

From equation (i)

$$dW = -dh + dQ = -dh + Tds \qquad ds = dQ/T \qquad ...(ii)$$

For an reversible process,

$$Tds = dh - ndp$$
$$-ndp = -dh + Tds \qquad ...(iii)$$

From equation (ii) and (iii)

$$dW = -ndp$$

On integrating both sides, we get

$$W = - \# ndp$$

It is valid for reversible process.

**Sol. 40** Option (A) is correct.

When the vapour is at a temperature greater than the saturation temperature, it is said to exist as super heated vapour. The pressure and Temperature of superheated vapour are independent properties, since the temperature may increase while the pressure remains constant. Here vapour is at $400^\circ$ C and saturation temperature is $200^\circ$ C.

So, at 200 kPa pressure superheated vapour will be left in the system.

**Sol. 41** Option (D) is correct.

Given : $p_1 = 100$ kPa , $p_2 = 200$ kPa . Let, $\upsilon_1 = \upsilon$

Now, given that Heat transfer takes place into the system until its volume increases by 50%

So, $$\upsilon_2 = \upsilon + 50\% \text{ of } \upsilon$$

Now, for work done by the system, we must take pressure is $p_2 = 200$ kPa , because work done by the system is against the pressure $p_2$ and it is a positive work done. From first law of thermodynamics,

$$dQ = dU + dW \qquad \text{...(i)}$$

But for a quasi-static process,

$$T = \text{Constant}$$

Therefore, change in internal energy is

$$dU = 0$$

From equation (i)

$$dQ = dW = pd\upsilon \qquad dW = pd\upsilon$$
$$= p[\upsilon_2 - \upsilon_1]$$

For initial condition at 100 kPa ,volume

$$\upsilon_1 = m_{liquid} \times \frac{1}{\rho_f} + m_{vapour} \times \frac{1}{\rho_g}$$

Here $$\frac{1}{\rho_f} = \upsilon_f = 0.001, \frac{1}{\rho_g} = \upsilon_g = 0.1$$

$$m_{liquid} = 1 \text{ kg}, m_{vapour} = 0.03 \text{ kg}$$

So $$\upsilon_1 = 1 \times 0.001 + 0.03 \times 0.1 = 4 \times 10^{-3} \text{ m}^3$$
$$\upsilon_2 = \frac{3}{2}\upsilon_1 = \frac{3}{2} \times 4 \times 10^{-3} = 6 \times 10^{-3} \text{ m}^3$$

$$= 200 \times 10^3 \left[\frac{3\upsilon}{2} - \upsilon\right]$$

$$= 200 \times [6 \times 10^{-3} - 4 \times 10^{-3}] = 200 \times 2 \times 10^{-3} = 0.4 \text{ kJ}$$

**Sol. 42** Option (C) is correct.

$$\Delta s_{net} = (\Delta s)_{system} + (\Delta s)_{surrounding} \qquad \text{...(i)}$$

And it is given that,

$$(\Delta s)_{system} = 10 \text{ kJ}$$

Also, $$(\Delta s)_{surrounding} = \left(\frac{Q}{T}\right)_{surrounding}$$

Heat transferred to the system by thermal reservoir,

$$T = 400^\circ\text{C} = (400 + 273) \text{ K} = 673 \text{ K}$$
$$Q = 1 \text{ kJ}$$
$$(\Delta s)_{surrounding} = \frac{1000}{673} = 1.485 \text{ J / K}$$

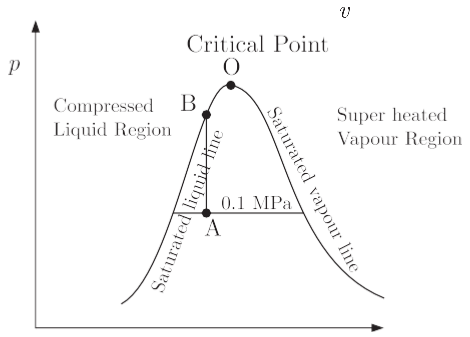
From equation (i) $(Ds)_{net} = 10 - 1.485 = 8.515\ \text{J/K}$
(Take Negative sign, because the entropy of surrounding decrease due to heat transfer to the system.)

**Sol. 43** Option (D) is correct.
In this question we discuss on all the four options.
(A) $dQ = dU + dW$ This equation holds good for any process undergone by a closed stationary system.
(B) $Tds = dU + pdn$ This equation holds good for any process reversible or irreversible, undergone by a closed system.
(C) $Tds = dU + dW$ This equation holds good for any process, reversible or irreversible, and for any system.
(D) $dQ = dU + pdn$ This equation holds good for a closed system when only $pdn$ work is present. This is true only for a reversible (quasi-static) process.

**Sol. 44** Option (A) is correct.
Given : $n_{cri} = 0.003155\ \text{m}^3/\text{kg}$, $n = 0.025\ \text{m}^3$, $p = 0.1$ MPa and $m = 10$ kg
We know, Rigid means volume is constant.

Specific volume, $$n_S = \frac{n}{m} = \frac{0.025}{10} = 0.0025\ \text{m}^3/\text{kg}$$

We see that the critical specific volume is more than the specific volume and during the heating process, both the temperature and the pressure remain constant, but the specific volume increases to the critical volume (i.e. critical point). The critical point is defined as the point at which the saturated liquid and saturated vapour states are identical.




So, point (B) will touch the saturated liquid line and the liquid line will rise at the point O.

**Sol. 45** Option (C) is correct.
Given : $L = 250$ mm $= 0.25$ m , $D = 200$ mm $= 0.2$ m ,

$$n_c = 0.001\ \text{m}^3,\ g = \frac{c_p}{c_v} = 1.4$$

Swept volume $$n_s = A \# L = \frac{p}{4}(D)^2 \# L$$

$$= \frac{p}{4}(0.2)^2 \# 0.25 = 0.00785\ \text{m}^3$$

Compression ratio $$r = \frac{n_T}{n_c} = \frac{n_c + n_s}{n_c} = \frac{0.001 + 0.00785}{0.001} = 8.85$$

Air standard efficiency $$h = 1 - \frac{1}{(r)^{g-1}} = 1 - \frac{1}{(8.85)^{1.4-1}}$$

$$= 1 - \frac{1}{2.39} = 1 - 0.418 = 0.582 \text{ or } 58.2\%$$

**Sol. 46** Option (A) is correct.

Following combination is correct

(R) The work done by a closed system in an adiabatic is a point function.

(S) A liquid expands upon freezing when the slope of its fusion curve on pressure-temperature diagram is negative.

**Sol. 47** Option (B) is correct.

We know, dryness fraction or quality of the liquid vapour mixture,

$$x = \frac{m_v}{m_v + m_l} = \frac{1}{1 + m_l / m_v} \qquad \text{...(i)}$$

Where, $m_v$ '' Mass of vapour and $m_l$ '' Mass of liquid

The value of $x$ varies between 0 to 1. Now from equation (i) if incorporation of reheater in a steam power plant adopted then Mass of vapour $m_v$ increase and Mass of liquid $m_l$ decreases So, dryness fraction $x$ increases.

In practice the use of reheater only gives a small increase in cycle efficiency, but it increases the net work output by making possible the use of higher pressure.

**Sol. 48** Option (C) is correct.

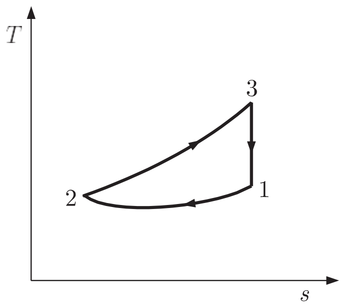
In the given $p$ - $n$ diagram, three processes are occurred.

(i) Constant pressure ( Process 1 – 2)

(ii) Constant Volume ( Process 2 – 3)

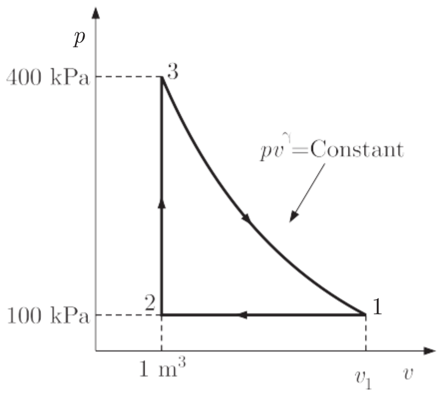
(iii) Adiabatic ( Process 3 – 1)

We know that, Constant pressure and constant volume lines are inclined curves in the $T$-$s$ curve, and adiabatic process is drawn by a vertical line on a $T$-$s$ curve.




Given $p$ - $n$ curve is clock wise. So $T$-$s$ curve must be clockwise.

**Sol. 49** Option (A) is correct.



This cycle shows the Lenoir cycle.
For Lenoir cycle efficiency is given by

$$\eta_L = 1 - \gamma\left(\frac{r_p^{\frac{1}{\gamma}} - 1}{r_p - 1}\right)$$

Where, $$r_p = \frac{p_2}{p_1} = \frac{400}{100} = 4$$

And $$\gamma = \frac{c_p}{c_v} = 1.4 \text{ (Given)}$$

So, $$\eta_L = 1 - 1.4\left(\frac{(4)^{\frac{1}{1.4}} - 1}{4 - 1}\right) = 1 - 0.789 = 0.211$$

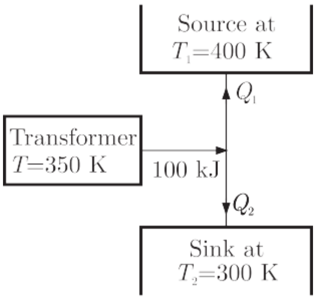
$$\eta_L = 21.1\% - 21\%$$

**Sol. 50** Option (D) is correct.

Given : $T_1 = 400$ K, $T_2 = 300$ K, $T = 350$ K, $Q = 100$ kJ

$Q_1$ " Heat transferred to the source by the transformer

$Q_2$ " Heat transferred to the sink by the transformer




Applying energy balance on the system,

$$Q = Q_1 + Q_2$$

$$Q_2 = Q - Q_1 = 100 - Q_1 \qquad \text{...(i)}$$

Apply Clausicus inequality on the system,

$$\frac{Q}{T} = \frac{Q_1}{T_1} + \frac{Q_2}{T_2}$$

$$\frac{100}{350} = \frac{Q_1}{400} + \frac{Q_2}{300}$$

Substitute the value of $Q_2$ from equation (i),

$$\frac{100}{350} = \frac{Q_1}{400} + \left(\frac{100 - Q_1}{300}\right) = \frac{Q_1}{400} + \frac{100}{300} - \frac{Q_1}{300}$$

$$\frac{100}{350} - \frac{100}{300} = Q_1\left(\frac{1}{400} - \frac{1}{300}\right)$$

$$-\frac{1}{21} = -\frac{Q_1}{1200}$$

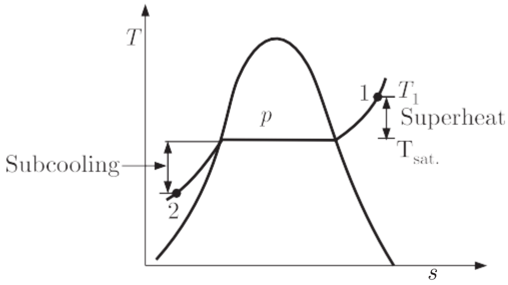
So, $$Q_1 = \frac{1200}{21} = 57.14 \text{ kJ}$$

Therefore the maximum amount of heat that can be transferred at 400 K is 57.14 kJ.

**Sol. 51** Option (D) is correct.

When the temperature of a liquid is less than the saturation temperature at the given pressure, the liquid is called compressed liquid (state 2 in figure).

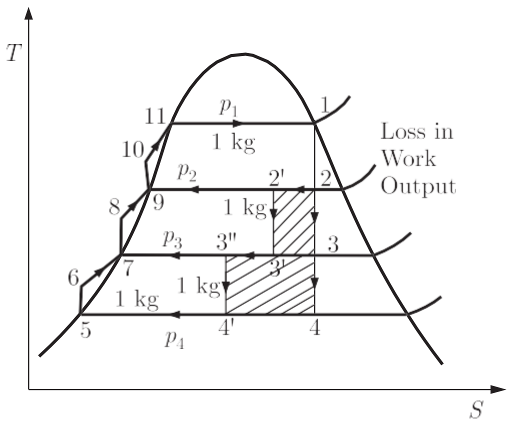
The pressure and temperature of compressed liquid may vary independently and a table of properties like the superheated vapor table could be arranged, to give the properties at any $p$ and $T$ .




The properties of liquids vary little with pressure. Hence, the properties are taken from the saturation table at the temperature of the compressed liquid.

So, from the given table at $T = 45c$ C, Specific enthalpy of water = 188.45 kJ /kg .

**Sol. 52** Option (A) is correct.




The thermal efficiency of a power plant cycle increases by increase the average temperature at which heat is transferred to the working fluid in the boiler or decrease the average temperature at which heat is rejected from the working fluid in the condenser. Heat is transferred to the working fluid with the help of the feed water heater.

So, (A) and (R) are true and (R) is the correct reason of (A).

**Sol. 53** Option (D) is correct.

(A) Condenser is an essential equipment in a steam power plant because when steam expands in the turbine and leaves the turbine in the form of super saturated steam. It is not economical to feed this steam directly to the boiler. So, condenser is used to condensed the steam into water and it is a essential part (equipment) in steam power plant.

Assertion (A) is correct.

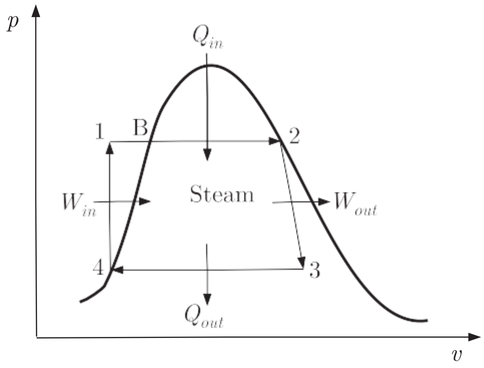
(R) The compressor and pumps require power input. The compressor is capable of compressing the gas to very high pressures. Pump work very much like compressor except that they handle liquid instead of gases. Now for same mass flow rate and the same pressure rise, a water pump require very less power because the specific volume of liquid is very less as compare to specific volume of vapour.

**Sol. 54** Option (D) is correct

| Group (I) | Group (II) | Group (III) | Group (IV) | Group (V) |
|---|---|---|---|---|
| | When added to the system | Differential | Function | Phenomenon |
| E | G | J | K | N |
| F | H | J | K | M |

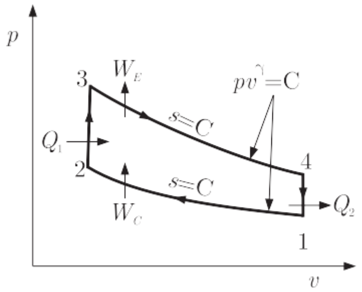
So correct pairs are
E-G-J-K-N and F-H-J-K-M

**Sol. 55** Option (A) is correct.
We draw $p$ - $v$ diagram for the cycles.
(a) Rankine cycle



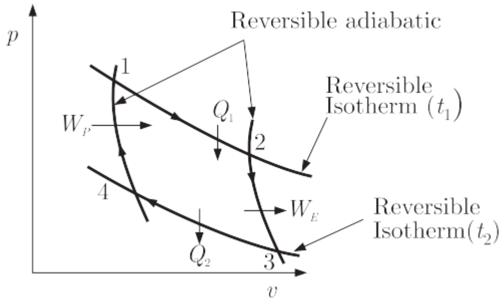

Constant Pressure Process
$Q_1$ = Heat addition at constant $p$ and $Q_2$ = Heat Rejection at constant $p$

(b) Otto cycle



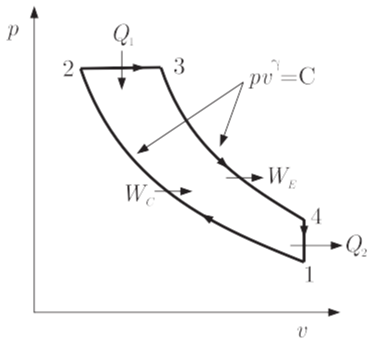

Constant Volume Process
$Q_1$ = Heat addition at constant $n$ and $Q_2$ = Heat Rejection at constant $n$

(c) Carnot cycle

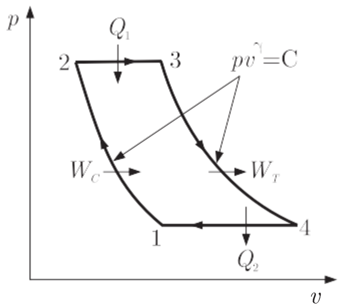


Constant Temperature Process (Isothermal)
$Q_1$ = Heat addition at constant $T$ and $Q_2$ = Heat Rejection at constant $T$

(d) Diesel cycle




Constant Pressure and constant volume process
$Q_1$ = Heat addition at constant $p$ and $Q_2$ = Heat rejection at constant $V$
(e) Brayton cycle




Constant pressure Process
$Q_1$ = Heat addition at constant $p$ and $Q_2$ = Heat rejection at constant $p$
From the Five cycles, we see that $P-S-5, R-U-3, P-S-1, Q-T-2$ are the correct pairs.

**Sol. 56** Option (D) is correct.

Given : $p_{gauge} = 1$ bar

$$p_{absolute} = p_{atm} + p_{gauge}$$

So, $p_{abs} = 1.013 + 1 = 2.013$ bar $\qquad p_{atm} = 1.013$ bar

$$T_1 = 15^\circ C = (273 + 15)\text{ K} = 288\text{ K}$$

$$T_2 = 5^\circ C = (273 + 5)\text{ K} = 278\text{ K}$$

Volume = Constant

$$\nu_1 = \nu_2 = 2500\text{ cm}^3 = 2500 \times (10^{-2})^3\text{ m}^3$$

From the perfect gas equation,

$$p\nu = m\,RT$$

$$2.013 \times 10^5 \times 2500 \times (10^{-2})^3 = m \times 287 \times 288$$

$$2.013 \times 2500 \times 10^{-1} = m \times 287 \times 288$$

$$m = \frac{2.013 \times 250}{287} = 0.0060 \text{ kg}$$

For constant Volume, relation is given by,

$$Q = mc_v\, dT \qquad c_v = 0.718 \text{ J/kg K}$$

$$= 0.0060 \times 0.718 \times (278 - 288) \qquad dT = T_2 - T_1$$

$$Q = -0.0437 = -43.7 \times 10^{-3} \text{ kJ}$$

$$= -43.7 \text{ Joule} \qquad \text{Negative sign shows the heat lost}$$

As the process is isochoric i.e. constant volume, So from the prefect gas equation,

$$\frac{p}{T} = \text{Constant}$$

And $\quad \dfrac{p_1}{T_1} = \dfrac{p_2}{T_2}$

$$p_2 = \frac{T_2}{T_1} \times p_1 = \frac{278}{288} \times 2.013 = 1.943 \text{ bar} \qquad p_1 = p_{abs}$$

So, Gauge Pressure = Absolute pressure – atmospheric pressure

$$p_{gauge} = 1.943 - 1.013 = 0.93 \text{ bar}$$

**Sol. 57** Option (C) is correct.

It is a constant volume process, it means

$$\frac{p}{T} = \text{Constant}$$

$$\frac{p_1}{p_2} = \frac{T_1}{T_2}$$

Substitute, $T_1 = 288$ and $T_2 = 278$

$$p_2 = p_{2,gauge} + p_{atm.} = 1 + 1.013 = 2.013 \text{ bar}$$

So, $\quad p_1 = \dfrac{T_1}{T_2} \times p_2 = \dfrac{288}{278} \times 2.013 = 2.08 \text{ bar}$

Gauge pressure, $\quad p_{gauge} = 2.08 - 1.013 = 1.067 \approx 1.07 \text{ bar}$

**Sol. 58** Option (A) is correct.

From the first law of thermodynamics for a cyclic process,

$$\Delta U = 0$$

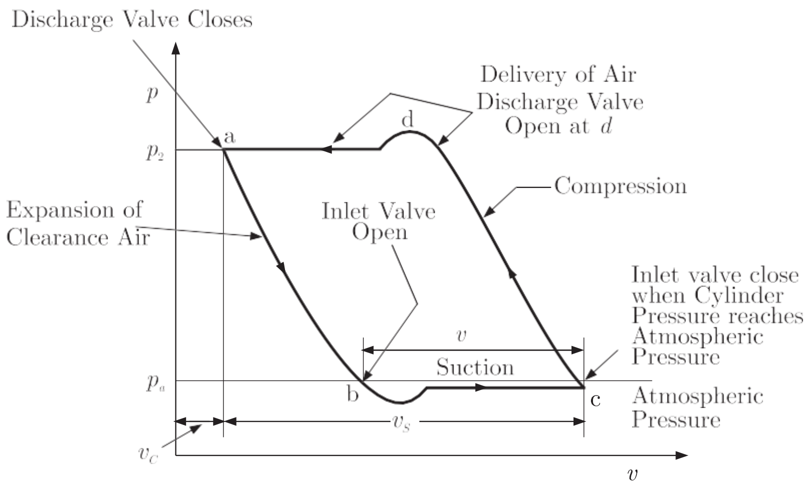
And $\quad \oint dQ = \oint dW$

The symbol $\oint dQ$, which is called the cyclic integral of the heat transfer represents the heat transfer during the cycle and $\oint dW$, the cyclic integral of the work, represents the work during the cycle.

We easily see that figure 1 and 2 satisfies the first law of thermodynamics. Both the figure are in same direction (clockwise) and satisfies the relation.

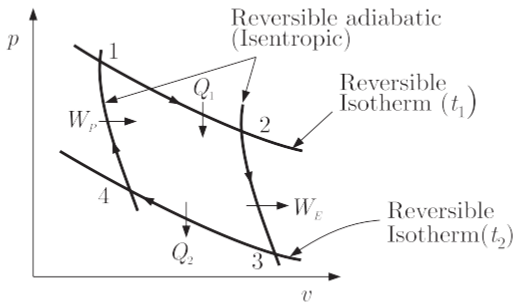
$$\oint dQ = \oint dW$$

**Sol. 59** Option (D) is correct.




From above figure, we can easily see that option (D) is same.

**Sol. 60** Option (A) is correct.




Now check the given processes :-

(i) Show in $p - n$ curve that process 1-2 and process 3-4 are Reversible isothermal process.

(ii) Show that process 2-3 and process 4-1 are Reversible adiabatic (isentropic) processes.

(iii) In carnot cycle maximum and minimum cycle pressure and the clearance volume are fixed.

(iv) From $p - n$ curve there is no polytropic process.

So, it consists only one cycle [carnot cycle]

**Sol. 61** Option (B) is correct.

Given : $p_1 = 10$ bar , $n_1 = 1$ m$^3$, $T_1 = 300$ K, $n_2 = 2$ m$^3$

Given that Nitrogen Expanded isothermally.

So, $RT$ = Constant

And from given relation,

$$\left(p + \frac{a}{n^2}\right)n = RT \quad = \text{Constant}$$

$$p_1 n_1 + \frac{a}{n_1} = p_2 n_2 + \frac{a}{n_2}$$

$$p_2 n_2 = p_1 n_1 + \frac{a}{n_1} - \frac{a}{n_2}$$

$$p_2 = p_1\left(\frac{v_1}{v_2}\right) + a\left(\frac{1}{v_1} - \frac{1}{v_2}\right) = 10\left(\frac{1}{2}\right) + a\left(\frac{1}{2} - \frac{1}{4}\right)$$

$$= 5 + \frac{a}{4}$$

Here $a > 0$, so above equation shows that $p_2$ is greater than 5 and +ve.

**Sol. 62** Option (B) is correct.

Velocity of flow, $u = u_1 = u_2 = \text{constant}$

& $W_2 >> W_1$ $W$ = Whirl velocity

Hence, it is a diagram of reaction turbine.

**Sol. 63** Option (B) is correct.

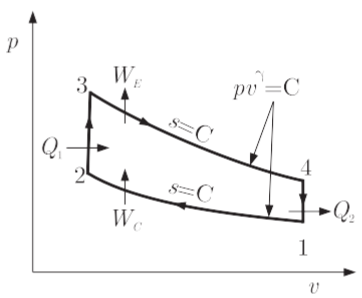
We know that efficiency,

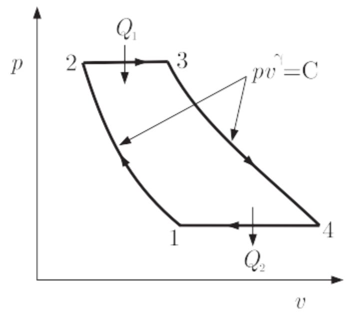
$$\eta_{Otto} = \eta_{Brayton} = 1 - \frac{T_1}{T_2}$$

$$\eta_{Otto} = \eta_{Brayton} = 1 - \frac{300}{450} = 1 - \frac{6}{9} = 0.33$$

So, $\eta_{Otto} = \eta_{Brayton} = 33\%$

**Sol. 64** Option (A) is correct.




From the previous part of the question

$T_{3(Otto)} = 600$ K, $T_{3(Brayton)} = 550$ K

From the $p$-$v$ diagram of Otto cycle, we have

$$W_O = Q_1 - Q_2 = c_v(T_3 - T_2) - c_v(T_4 - T_1) \quad \text{...(i)}$$

For process 3 - 4,

$$\frac{T_3}{T_4} = \left(\frac{v_4}{v_3}\right)^{\gamma-1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1} \qquad v_4 = v_1,\ v_3 = v_2$$

For process 1 - 2,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$

So,

$$\frac{T_3}{T_4} = \frac{T_2}{T_1}$$

$$T_4 = \frac{T_3}{T_2} \times T_1 = \frac{600}{450} \times 300 = 400 \text{ K}$$

And

$$W_O = c_v(600 - 450) - c_v(400 - 300)$$

$$= c_v(150) - 100c_v = 50c_v \quad \text{...(ii)}$$

From $p$ - $v$ diagram of brayton cycle, work done is,

$$W_B = Q_1 - Q_2 = c_p(T_3 - T_2) - c_p(T_4 - T_1)$$

And

$$T_4 = \frac{T_1}{T_2} \times T_3 = \frac{300}{450} \times 550 = 366.67 \text{ K}$$

$$W_B = c_p(550-450) - c_p(366.67-300) = 33.33c_p \quad ...(iii)$$

Dividing equation (ii) by (iii), we get

$$\frac{W_O}{W_B} = \frac{50c_v}{33.33c_p} = \frac{50}{33.33\gamma} \qquad \frac{c_p}{c_v} = \gamma,\ \gamma = 1.4$$

$$= \frac{50}{33.33 \times 1.4} = \frac{50}{46.662} > 1$$

From this, we see that,

$$W_O > W_B$$

**Sol. 65** Option (D) is correct.

From saturated ammonia table column 5 and 8 are the specific enthalpy data column.

**Sol. 66** Option (B) is correct.

The enthalpy of the fluid before throttling is equal to the enthalpy of fluid after throttling because in throttling process enthalpy remains constant.

$$h_1 = h_2$$

$$371.43 = 89.05 + x(1418 - 89.05) \qquad h = h_f + x(h_g - h_f)$$

$$= 89.05 + x(1328.95)$$

$$x = \frac{282.38}{1328.95} = 0.212$$

**Sol. 67** Option (C) is correct.

$W = -5000\text{ kJ}$ (Negative sign shows that work is done on the system)

$Q = -2000\text{ kJ}$ (Negative sign shows that heat rejected by the system)

From the first law of thermodynamics,

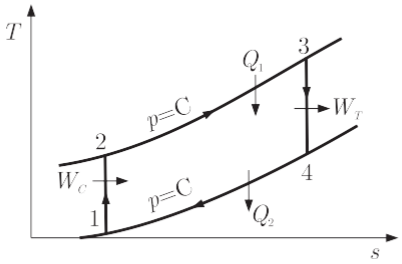
$$\Delta Q = \Delta W + \Delta U$$

So,
$$\Delta U = \Delta Q - \Delta W = -2000 - (-5000)$$

$$= 3000\text{ kJ}$$

**Sol. 68** Option (A) is correct.

The $T$ - $s$ curve for simple gas power plant cycle (Brayton cycle) is shown below :




From the $T$ - $s$ diagram, Net work output for Unit Mass,

$$W_{net} = W_T - W_c = c_p[(T_3 - T_4) - (T_2 - T_1)] \quad ...(i)$$

And from the $T$ - $s$ diagram,

$$T_3 = T_{max} \text{ and } T_1 = T_{min}$$

Apply the general relation for reversible adiabatic process, for process 3-4 and 1-2,

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$T_4 = T_3(r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)} \qquad \frac{p_3}{p_4} = \frac{p_2}{p_1} = r_p = \text{Pressure ratio}$$

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$T_2 = T_1 (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$W_{net} = c_p \left[T_3 - T_3 (r_p)^{-\left(\frac{\gamma-1}{\gamma}\right)} - T_1 (r_p)^{\frac{\gamma-1}{\gamma}} + T_1\right] \quad ...(ii)$$

Differentiating equation (ii) w.r.t. $(r_p)$ and on equating it to the zero, we get

$$\frac{dW_{net}}{dr_p} = c_p\left[-T_3\left(-\frac{\gamma-1}{\gamma}\right) r_p^{\left(-\frac{\gamma-1}{\gamma}-1\right)} - T_1\left(\frac{\gamma-1}{\gamma}\right) r_p^{\left(\frac{\gamma-1}{\gamma}-1\right)}\right]$$

$$= c_p\left[-T_3\left(-\frac{\gamma-1}{\gamma}\right) r_p^{\left(\frac{-\gamma+1-\gamma}{\gamma}\right)} - T_1\left(\frac{\gamma-1}{\gamma}\right) r_p^{\left(-\frac{1}{\gamma}\right)}\right]$$

$$= c_p\left[-T_3\left(-\frac{\gamma-1}{\gamma}\right) r_p^{\left(\frac{1-2\gamma}{\gamma}\right)} - T_1\left(\frac{\gamma-1}{\gamma}\right) r_p^{\left(-\frac{1}{\gamma}\right)}\right]$$

$$T_3 r_p^{\left(\frac{1}{\gamma}-2\right)} - T_1 r_p^{\left(-\frac{1}{\gamma}\right)} = 0$$

$$T_3 r_p^{\left(\frac{1}{\gamma}-2\right)} = T_1 r_p^{-\frac{1}{\gamma}}$$

$$\frac{T_3}{T_1} = \frac{(r_p)^{-\frac{1}{\gamma}}}{r_p^{\frac{1}{\gamma}-2}} = (r_p)^{-\frac{1}{\gamma}-\frac{1}{\gamma}+2} = r_p^{\frac{2(\gamma-1)}{\gamma}}$$

So, $$(r_p)_{opt} = \left(\frac{T_3}{T_1}\right)^{\frac{\gamma}{2(\gamma-1)}} = \left(\frac{T_{max}}{T_{min}}\right)^{\frac{\gamma}{2(\gamma-1)}}$$

**Sol. 69** Option (C) is correct.

**Stoichiometric mixture :**

The S.M. is one in which there is just enough air for complete combustion of fuel.

**Sol. 70** Option (A) is correct.

Given : $m = 2000$ kg, $T_1 = 1250$ K, $T_2 = 450$ K, $T_0 = 303$ K, $c = 0.5$ kJ /kg K

$$Q_1 = \text{Available Energy + Unavailable energy}$$

$$A.E. = Q_1 - U.E. \quad ...(i)$$

And $$Q_1 = mc\Delta T = 2000 \times 0.5 \times 10^3 \times (1250 - 450) = 800 \text{ MJoule}$$

We know $$U.E. = T_0 (\Delta s) \quad ...(ii)$$

$$\Delta S = mc \ln \frac{T_1}{T_2} = 2000 \times 0.5 \times 10^3 \ln \frac{1250}{450}$$

$$= 10^6 \ln \frac{1250}{450} = 1.021 \times 10^6 \text{ J / kg}$$

Now, Substitute the value of $Q_1$ and $U.E.$ in equation (i),

$$A.E. = 800 \times 10^6 - 303 \times 1.021 \times 10^6 \quad \text{From equation (ii)}$$

$$= 10^6 \times (800 - 309.363)$$

$$= 490.637 \times 10^6 = 490.637 \simeq 490.44 \text{ MJ}$$

**Sol. 71** Option (C) is correct.

When all cylinders are firing then, power is 3037 kW = Brake Power

Power supplied by cylinders (Indicated power) is given below :

| Cylinder No. | Power supplied (I.P.) |
|---|---|
| 1. | I.P.$_1$ = 3037 – 2102 = 935 kW |
| 2. | I.P.$_2$ = 3037 – 2102 = 935 kW |
| 3. | I.P.$_3$ = 3037 – 2100 = 937 kW |
| 4. | I.P.$_4$ = 3037 – 2098 = 939 kW |

$$I.P._{Total} = I.P._1 + I.P._2 + I.P._3 + I.P._4 = 935 + 935 + 937 + 939 = 3746 \text{ kW}$$

And, $$h_{mech} = \frac{B.P.}{I.P.} = \frac{3037}{3746} = 0.8107 \text{ or } 81.07\%$$

**Sol. 72** Option (D) is correct.

Given : $D = 10$ cm $= 0.1$ meter , $L = 15$ cm $= 0.15$ meter

$$g = \frac{c_p}{c_v} = 1.4,\ n_c = 196.3 \text{ cc},\ Q = 1800 \text{ kJ/kg}$$

$$n_s = A \# L = \frac{p}{4} D^2 \# L = \frac{p}{4} \# (10)^2 \# 15 = \frac{1500p}{4} = 1177.5 \text{ cc}$$

And Compression ratio, $$r = \frac{n_T}{n_c} = \frac{n_c + n_s}{n_c} = \frac{196.3 + 1177.5}{196.3} = 6.998 - 7$$

Cycle efficiency,

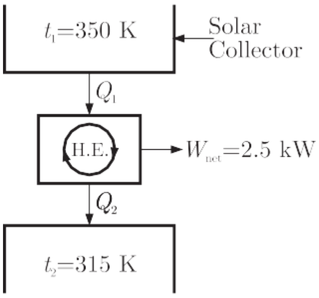
$$h_{Otto} = 1 - \frac{1}{(r)^{g-1}} = 1 - \frac{1}{(7)^{1.4-1}} = 1 - \frac{1}{2.1779} = 1 - 0.4591 = 0.5409$$

$$h_{Otto} = 54.09\%$$

We know that, $$h = \frac{\text{Work output}}{\text{Heat Supplied}}$$

$$\text{Work output} = h \# \text{Heat supplied} = 0.5409 \# 1800 = 973.62 \text{ kJ} - 973.5 \text{ kJ}$$

**Sol. 73** Option (A) is correct.




Solar collector receiving solar radiation at the rate of 0.6 kW / m$^2$. This radiation is stored in the form of internal energy. Internal energy of fluid after absorbing.

Solar radiation, $$DU = \frac{1}{2} \# 0.6 \qquad \text{Efficiency of absorbing radiation is 50\%}$$

$$= 0.3 \text{ kW/m}^2$$

$$h_{Engine} = 1 - \frac{T_2}{T_1} = \frac{W_{net}}{Q_1}$$

$$Q_1 = \frac{W_{net} \# T_1}{T_1 - T_2} = \frac{2.5 \# 350}{350 - 315} = 25 \text{ kW}$$

Let, $A$ is the minimum area of the solar collector.

So, $$Q_1 = A \# DU = A \# 0.3 \text{ kW/m}^2$$

$$A = \frac{Q_1}{0.3} = \frac{25}{0.3} = \frac{250}{3} = 83.33 \text{ m}^2$$

**Sol. 74** Option (B) is correct.

Given : $h_1 = 29.3$ kJ/kg , $h_3 = 3095$ kJ/kg , $h_4 = 2609$ kJ/kg , $h_5 = 3170$ kJ/kg

$h_6 = 2165$ kJ/kg

Heat supplied to the plant,

$$Q_S = (h_3 - h_1) + (h_5 - h_4) \qquad \text{At boiler and reheater}$$

$$= (3095 - 29.3) + (3170 - 2609) = 3626.7 \text{ kJ}$$

Work output from the plant,

$$W_T = (h_3 - h_4) + (h_5 - h_6) = (3095 - 2609) + (3170 - 2165) = 1491 \text{ kJ}$$

Now, $\eta_{thermal} = \frac{W_T - W_P}{Q_s} = \frac{W_T}{Q_s}$ Given, $W_p = 0$

$$= \frac{1491}{3626.7} = 0.411 = 41.1\%$$

**Sol. 75** Option (D) is correct.

From the figure, we have enthalpy at exit of the pump must be greater than at inlet of pump because the pump supplies energy to the fluid.

$$h_2 > h_1$$

So, from the given four options only one option is greater than $h_1$

$$h_2 = 33.3 \text{ kJ/kg}$$

**Sol. 76** Option (B) is correct.

Equivalence Ratio or Fuel Air Ratio $\left(\frac{F}{A}\right)$

$$\phi = \frac{\text{Actual Fuel-Air ratio}}{\text{stoichiometric Fuel air Ratio}} = \frac{\left(\frac{F}{A}\right)_{actual}}{\left(\frac{F}{A}\right)_{stoichiometric}}$$

If $\phi = 1$, & stoichiometric (Chemically correct) Mixture.

If $\phi > 1$, & rich mixture.

If $\phi < 1$, & lean mixture.

Now, we can see from these three conditions that $\phi > 1$, for both idling and peak power conditions, so rich mixture is necessary.

**Sol. 77** Option (C) is correct.

The compression ratio of diesel engine ranges between 14 to 25 where as for S.I, engine between 6 to 12. Diesel Engine gives more power but efficiency of diesel engine is less than compare to the S.I. engine for same compression ratio.

**Sol. 78** Option (C) is correct.

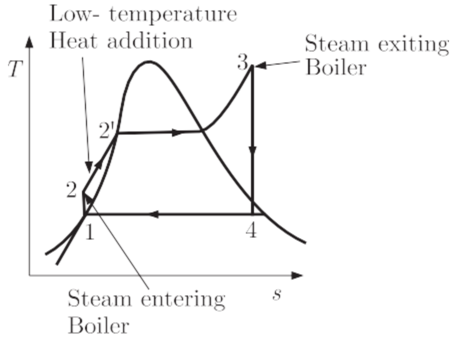
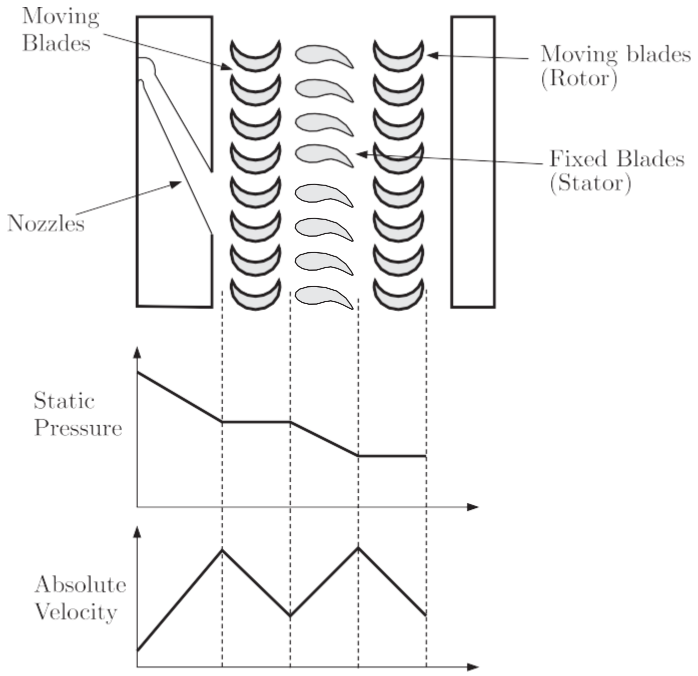



Fig : $T$ - $s$ curve of simple Rankine cycle

From the observation of the $T$-$s$ diagram of the rankine cycle, it reveals that heat is transferred to the working fluid during process 2 – 2' at a relatively low temperature. This lowers the average heat addition temperature and thus the cycle efficiency.

To remove this remedy, we look for the ways to raise the temperature of the liquid leaving the pump (called the feed water ) before it enters the boiler. One possibility is to transfer heat to the feed water from the expanding steam in a counter flow heat exchanger built into the turbine, that is, to use regeneration. A practical regeneration process in steam power plant is accomplished by extracting steam from the turbine at various points. This steam is used to heat the feed water and the device where the feed water is heated by regeneration is called feed water heater. So, regeneration improves cycle efficiency by increasing the average temperature of heat addition in the boiler.




**Sol. 79** Option (D) is correct.

It may be easily seen that the diagram that static pressure remains constant, while velocity decreases.

**Sol. 80** Option (C) is correct.

Given : $p = 2$ kW $= 2 \times 10^3$ W , $t = 20$ minutes $= 20 \times 60$ sec ,
$c_p = 4.2$ kJ /kgK

Heat supplied,
$$Q = \text{Power} \times \text{Time}$$
$$= 2 \times 10^3 \times 20 \times 60 = 24 \times 10^5 \text{ Joule}$$

And Specific heat at constant pressure,
$$Q = mc_p \Delta T$$
$$\Delta T = \frac{24 \times 10^5}{40 \times 4.2 \times 1000} = \frac{24 \times 100}{40 \times 4.2} = 14.3^\circ\text{C}$$

**Sol. 81** Option (D) is correct.

The $Tds$ equation considering a pure, compressible system undergoing an internally reversible process.

From the first law of thermodynamics
$$(\delta Q)_{rev.} = dU + (\delta W)_{rev} \qquad \text{...(i)]}$$

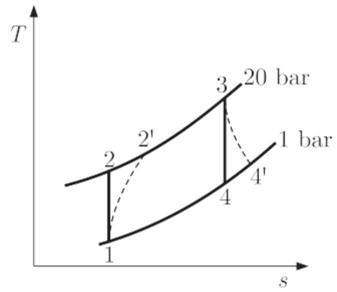
By definition of simple compressible system, the work is
$$(\delta W)_{rev} = pdv$$

And entropy changes in the form of
$$ds = \left(\frac{dQ}{T}\right)_{rev}$$
$$(\delta Q)_{rev} = Tds$$

From equation (i), we get
$$Tds = dU + pdv$$

This equation is equivalent to the $I^{st}$ law, for a reversible process.

**Sol. 82** Option (A) is correct.




Given : $c_p = 0.98$ kJ/kgK, $\eta_{isen} = 0.94$, $c_v = 0.7538$ kJ/kgK, $T_3 = 1500$ K

$p_3 = 20$ bar $= 20 \times 10^5$ N/m$^2$, $p_4 = 1$ bar $= 1 \times 10^5$ N/m$^2$

$$\gamma = \frac{c_p}{c_v} = \frac{0.98}{0.7538} = 1.3$$

Apply general Equation for the reversible adiabatic process between point 3 and 4 in $T$ - $s$ diagram,

$$\left(\frac{T_3}{T_4}\right) = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{1500}{T_4} = \left(\frac{20 \times 10^5}{1 \times 10^5}\right)^{\frac{1.3-1}{1.3}} = (20)^{\frac{0.3}{1.3}}$$

$$T_4 = \frac{1500}{(20)^{\frac{0.3}{1.3}}} = 751.37 \text{ K}$$

And $$\eta_{isentropic} = \frac{\text{Actual output}}{\text{Ideal output}} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.94 = \frac{1500 - T_4'}{1500 - 751.37}$$

$$0.94 \times 748.63 = 1500 - T_4'$$

$$T_4' = 1500 - 703.71 = 796.3 \text{ K}$$

Turbine work, $$W_t = c_p\,(T_3 - T_4') = 0.98\,(1500 - 796.3) = 698.64 \text{ kJ/kg}$$

**Sol. 83** Option (A) is correct.

Given : $f = \frac{F}{A} = \frac{m_f}{m_a} = 0.05$, $\eta_v = 90\% = 0.90$, $\eta_{ith} = 30\% = 0.3$

$CV_{fuel} = 45$ MJ/kg, $\rho_{air} = 1$ kg/m$^3$

We know that, volumetric efficiency is given by,

$$\eta_v = \frac{\text{Actual Volume}}{\text{Swept Volume}} = \frac{\nu_{ac}}{\nu_s}$$

$$\nu_{ac} = \eta_v\,\nu_s = 0.90 V_s \qquad \text{...(i)}$$

Mass of air, $$m_a = \rho_{air} \times \nu_{ac} = 1 \times 0.9\nu_s = 0.9\nu_s$$

$$m_f = 0.05 \times m_a = 0.045\nu_s$$

$$\eta_{ith} = \frac{I.P.}{m_f \times CV} = \frac{p_{im}\,LAN}{m_f \times CV} \qquad I.P. = p_{im}\,LAN$$

$$p_{im} = \frac{\eta_{ith} \times m_f \times CV}{LAN} \qquad LAN = \nu_s$$

$$\frac{0.30 \times 0.045 \times \nu_s \times 45 \times 10^6}{\nu_s} = 0.6075 \times 10^6$$

$$= 6.075 \times 10^5 \text{ Pa} = 6.075 \text{ bar} \qquad 1 \text{ bar} = 10^5 \text{ Pa}$$

**Sol. 84** Option (D) is correct.

Given: $\nu_c = 10\%$ of $\nu_s = 0.1\nu_s$

$$\frac{\nu_s}{\nu_c} = \frac{1}{0.1} = 10$$

And specific heat ratio $c_p / c_v = \gamma = 1.4$

We know compression ratio,

$$r = \frac{\nu_T}{\nu_c} = \frac{\nu_c + \nu_s}{\nu_c} = 1 + \frac{\nu_s}{\nu_c} = 1 + 10 = 11$$

Efficiency of Otto cycle,

$$\eta_{Otto} = 1 - \frac{1}{(r)^{\gamma-1}} = 1 - \frac{1}{(11)^{1.4-1}}$$

$$= 1 - \frac{1}{(11)^{0.4}} = 1 - 0.3832 = 0.6168 \approx 61.7\%$$

**Sol. 85** Option (B) is correct.

Given : $p_1 = 2$ bar $= 2 \times 10^5$ N/m$^2$, $T_1 = 298$ K $= T_2$, $\nu_1 = 1$ m$^3$, $\nu_2 = 2$ m$^3$

The process is isothermal,

So,
$$W = p_1\nu_1 \ln\frac{p_1}{p} = p_1\nu_1 \ln\left(\frac{\nu_2}{\nu}\right) = 2 \times 10^5 \times 1 \ln\left(\frac{2}{1}\right)$$

$$= 2 \times 0.6931 \times 10^5 = 138.63 \text{ kJ} \approx 138.6 \text{ kJ}$$

**Sol. 86** Option (A) is correct.

Entropy,
$$\Delta S = \frac{\Delta Q}{T} \qquad \text{...(i)}$$

From first law of thermodynamics,

$$\Delta Q = \Delta U + \Delta W$$

For isothermal process, $\Delta U = 0$

$$\Delta Q = \Delta W$$

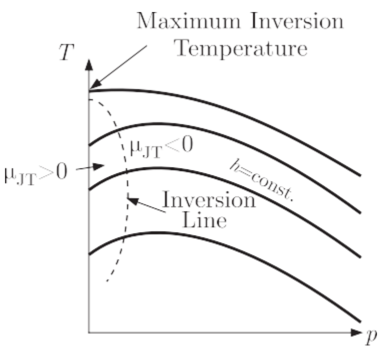
From equation (i),
$$\Delta S = \frac{\Delta W}{T} = \frac{138.63 \text{ kJ}}{298 \text{ K}} = 0.4652 \text{ kJ/K}$$

**Sol. 87** Option (A) is correct.

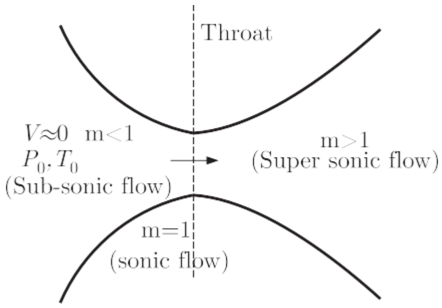
The Joule-Thomson coefficient is a measure of the change in temperature with pressure during a constant enthalpy process.

$$\mu = \left(\frac{\partial T}{\partial p}\right)_h$$

If
$$\mu_{JT} = \begin{cases} < 0 & \text{temperature increases} \\ = 0 & \text{Temperature remains cons tan t} \\ > 0 & \text{Temperature decreases during a throttling process} \end{cases}$$



**Sol. 88** Option (B) is correct.




The greatest velocity and lowest pressure occurs at the throat and the diverging portion remains a subsonic diffuser. For correctly designed convergent divergent nozzle, the throat velocity is sonic and the nozzle is now chocked.

**Sol. 89** Option (B) is correct.

Given : $\eta = 0.75$, $T_1 = 727^\circ C = (727 + 273) = 1000$ K

The efficiency of Otto cycle is given by,

$$\eta = \frac{W_{net}}{Q_1} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1}$$

$$\frac{T_2}{T_1} = 1 - \eta \qquad \& \; T_2 = (1 - \eta)\, T_1$$

$$T_2 = (1 - 0.75)\, 1000 = 250 \text{ K or } -23^\circ C$$

**Sol. 90** Option (A) is correct.

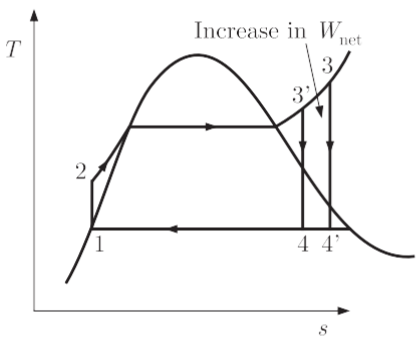
Given : $r = 8.5$, $\gamma = 1.4$

The efficiency of Otto cycle is,

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

$$= 1 - \frac{1}{(8.5)^{1.4-1}} = 1 - \frac{1}{2.35} = 57.5\%$$

**Sol. 91** Option (B) is correct.

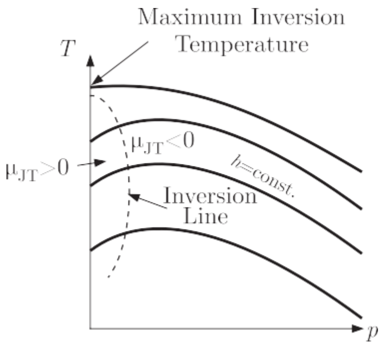



The average temperature at which heat is transferred to steam can be increased without increasing the boiler pressure by superheating the steam to high temperatures. The effect of superheating on the performance of vapour power cycle is shown on a $T$ -$s$ diagram the total area under the process curve 3 - 3$'$ represents the increase in the heat input. Thus both the net work and heat input increase as a result of superheating the steam to a higher temperature. The overall effect is an increase in thermal efficiency, since the average temperature at which heat is added increases.

**Sol. 92** Option (A) is correct.

The Rateau turbine is a pressure compounded turbine.

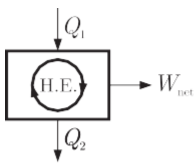
**Sol. 93** Option (B) is correct.




When $m < 0$ then temperature increases and become warmer.

**Sol. 94** Option (A) is correct.

Given : $W_{net} = 50$ kJ, $h = 75\% = 0.75$




We know, efficiency of heat engine is,

$$h = \frac{W_{net}}{Q_1} \ \& \ Q_1 = \frac{W_{net}}{h}$$

Where $Q_1$ = Heat transferred by the source to the system.

$$Q_1 = \frac{50}{0.75} = 66.67 \text{ kJ}$$

From the figure heat rejected $Q_2$

(From the energy balance)

$$Q_1 = Q_2 + W_{net}$$

$$Q_2 = Q_1 - W_{net} = 66.67 - 50 = 16.67 = 16\frac{2}{3} \text{ kJ}$$

**Sol. 95** Option (C) is correct.

Given : $p_1 = 1$ bar , $p_2 = 16$ bar

The intermediate pressure $p_x$ (pressure ratio per stage) has an optimum value for minimum work of compression.

And $$p_x = \sqrt{p_1 p_2} = \sqrt{1 \# 16} = 4 \text{ bar}$$

**Sol. 96** Option (B) is correct.

Let $h_1$ and $h_2$ are the enthalpies of steam at the inlet and at the outlet.

Given : $$h_1 - h_2 = 0.8 \text{ kJ/kg}$$

$$V_1 = 0$$

From the energy balance for unit mars of steam, the total energy at inlet must be equal to total energy at outlet.

So, $$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$V_2^2 = 2(h_1 - h_2)$$

$$V_2 = \sqrt{2 \# 0.8 \# 10^3} = 40 \text{ m/sec}$$

**Sol. 97** Option (B) is correct.

Given : $r = 5.5,\ W = 23.625 \# 10^5 \# \nu_c$

We know, $p_{mep} = \dfrac{W_{net}}{\nu_s} = \dfrac{23.625 \# 10^5}{\nu_s / \nu_c}$ ...(i)

Where $\nu_s$ = swept volume

And $r = \dfrac{\nu}{\nu_c} = \dfrac{\nu_c + \nu_s}{\nu_c} = 1 + \dfrac{\nu_s}{\nu_c}$

$$\frac{\nu_s}{\nu_c} = (r - 1)$$

Where $\nu_t$ = Total volume

$\nu_c$ = clearance volume

Substitute this value in equation (i), we get

$$p_{mep} = \frac{23.625 \# 10^5}{r - 1} = \frac{23.625 \# 10^5}{5.5 - 1} = 5.25 \# 10^5 = 5.25 \text{ bar}$$

***********