Shiv Kumar

Problems and Solutions in Thermal Engineering

With Multiple-Choice Type Questions

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Springer

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Shiv Kumar
Department of Mechanical
and Automation Engineering
GGSIP University
New Delhi, India

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Dedicated to

My Parents

My Wife Dr. Kusum Lata and My Son Tanishq

Preface

Thermodynamics deals with the storage, transformation and transfer of energy.As an engineering discipline, Thermal Engineering dealswith the innovative use of the laws of thermodynamics in solving relevant technological problems. This introductory textbook aims to provide undergraduate engineering students with the knowledge (basics principles and thermodynamics laws) they need to have to know, understand and analyze the thermodynamic problems they are likely to encounter in practice.

The book is developed in such a way that the most complex things are explained in simplest of manners to ensure thorough understanding for the reader. The book is concise and every concept is presented from an elementary and tangible perspective.

The subject matter is well illustrated with a innumerable examples. A great deal of attention is given to select the numerical problems and solving them. The theory and numerical problems at the end of each chapter also aim to enhance the creative capabilities of students.

Text books are dynamic and need to change with time. Suggestions from the teachers and students for the further improvement of the text are welcome and will be implemented in the next edition. The readers are requested to bring out the error to the notice, which will be suitably acknowledged.

New Delhi, India

Shiv Kumar

Acknowledgements

First of all I would like to express my deep gratitude to God for giving me the strength and health for completing this book. I am very thankful to my colleagues of mechanical engineering department for their highly appreciable help and my students for their valuable suggestions.

I am also thankful my publishers Sh. Sunil Saxena and Sh. Jai Raj Kapoor of Ane Books Pvt. Ltd. and editorial group for their help and assistance.

A special thanks goes to my wife Ms. Kusum for her help, support and strength to complete the book.

Shiv Kumar

Contents

About the Author

Shiv Kumar is former Head of Department of Mechanical and Automation Engineering and Associate Professor at Guru Premsukh Memorial College of Engineering affiliated to Guru Gobind Singh Indraprastha University, Delhi, India. He obtained his Bachelor's degree in Mechanical Machine Design and Automation Engineering from R.E.C. (presently NIT), Jalandhar, and Master's degree in Thermal Engineering from Delhi College of Engineering, India. He has over 18 years of teaching experience. He has published several books in the field of Mechanical Engineering.

Basic of Thermodynamics

1

1. I.C. Engine is an example of

   (A) Closed system
   (B) Open system
   (C) Isolated system
   (D) None of these

2. In an I.C. engine, third process (i.e., expansion of gas) is an example of

   (A) Closed system
   (B) Open system
   (C) Isolate system
   (D) None of these

3. In an I.C. engine, fourth process (i.e., exhaust of gas) is an example of

   (A) Closed system
   (B) Open system
   (C) Isolated system
   (D) None of these

4. Thermos flask in an example of

   (A) Closed system
   (B) Open system
   (C) Isolated system
   (D) None of these

5. In an open system

   (A) Mass transfer
   (B) Energy transfer
   (C) Neither mass nor energy transfer
   (D) None of these

6. The energy is not allowed to cross the boundary of the system, that system is called a/an

   (A) Closed system
   (B) Open system
   (C) Isolated system
   (D) None of these

7. Consider the following properties:

   1. Temperature
   2. Viscosity
   3. Specific entropy
   4. Thermal conductivity

Which of the above properties of a system is/are intensive?

(A) 1 only
(B) 2 and 3 only
(C) 2, 3, and 4 only
(D) 1, 2, 3, and 4

8. Consider the following properties:

   1. Entropy
   2. Viscosity
   3. Temperature
   4. Specific heat at constant volume

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Which of the above properties of a system is/are extensive properly?

(A) 1 only
(B) 1 and 2 only
(C) 2, 3, and 4
(D) 1, 2, and 4

9. Which of the following is not an extensive property?

(A) Temperature
(B) Internal energy
(C) Momentum
(D) Volume

10. Which of the following is a heterogeneous system?

(A) The cooling of a fluid in a radiator
(B) A mixture of ice, water, and steam
(C) A mixture of hydrogen and oxygen
(D) Atmospheric air

11*. An ideal gas at 27 °C is heated at constant pressure till the volume becomes three times. The temperature of the gas will then be

(A) 81 °C
(B) 900 °C
(C) 627 °C
(D) 927 °C

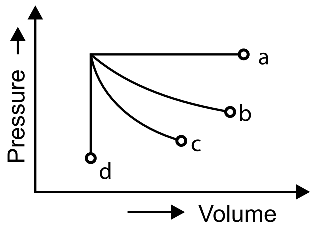
12*. In case of Boyle's law, if pressure increases by 1%, the percentage decrease in volume is

(A) 1/100%
(B) 0%
(C) 1/101%
(D) 100/101%

13. The equations, $pV = mRT$ is best obeyed by gases at
(A) Low pressure and low temperature
(B) Low pressure and high temperature
(C) High pressure and low temperature
(D) None of these

14. Which of the following statement about work for a quasi-equilibrium process is incorrect?

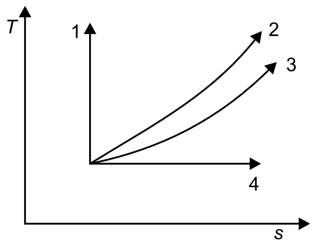
(A) Work is a path function
(B) The differential of work is inexact
(C) Work is the area under a p–T diagram
(D) Work is always zero for a constant volume process



15. The figure shows thermodynamic expansion processes a, b, c, and d. Which line is very close to the isentropic process?

(A) a
(B) b
(C) c
(D) d

16. Match List-I with List-II and select the correct answer using the code given below the list:

List-I (Process index n)	List-II (T–s traces)
(a) 0	
(b) 1	
(c) 1.4	
(d) ∞	

Code:

	a	b	c	d
(A)	3	1	4	2
(B)	2	1	4	3
(C)	3	4	1	2
(D)	2	4	1	3

17*. During an adiabatic process, the pressure p of a fixed mass of ideal gas changes by Δp and its volume V changes by ΔV. The value is $\frac{\Delta V}{V}$

(A) $-\frac{1}{\gamma}\frac{\Delta p}{p}$
(B) $\frac{\Delta p}{p}$
(C) $-\gamma\frac{\Delta p}{p}$
(D) $\frac{1}{\gamma}\frac{\Delta p}{p}$

18. The equations $\left(p+\frac{a}{V^2}\right)(V-b)=RT$ is called

(A) Maxwell equation
(B) Van der Waals equation
(C) Ideal gas equation
(D) Polytropic process equation

19. In the equations $\left(p+\frac{a}{V^2}\right)(V-b)=RT$

1. The coefficient 'a' takes care of intermolecular attraction.
2. The coefficient 'b' accounts for the volume of molecules.
3. The equation is obeyed by real gases more closely at *high* pressure.
4. The coefficient 'b' is known as co-volume.

Of these statements:

(A) 1, 2, and 3 are true
(B) 1 and 2 are true
(C) 1, 2, 3, and 4 are true
(D) 1, 2, and 4 are true

20. In the Van der Waals equations $\left(p+\frac{a}{V^2}\right)(V-b)=RT$, the units of constant 'a' are:

(A) $\mathrm{Nm^4}$, $\mathrm{kg\ m^5/s^2}$
(B) $\mathrm{Nm^2}$, $\mathrm{kg\ m^3/s^2}$
(C) $\mathrm{N/m^4}$, $\mathrm{kg/m^3s^2}$
(D) None of the above

2 First Law of Thermodynamics

21. Two bodies which are at different states are brought into contact with each other and allowed to reach a final state of thermal equilibrium. The final temperature attained is specified by the

    (A) First law of thermodynamics
    (B) Second law of thermodynamics
    (C) Third law of thermodynamics
    (D) Zeroth law of thermodynamics

22. c_p–c_v = R is valid for

    (A) All gases
    (B) Gases of very high pressure
    (C) Ideal gases
    (D) Gases of very low temperature

23. c_p and c_v are equal at

    (A) 0 °C
    (B) 0 K
    (C) 0 °F
    (D) 0 °R

24. c_v for an ideal gas

    (A) Does not depend upon temperature
    (B) Is independent of pressure only
    (C) Is independent of volume only
    (D) Is independent of both pressure and volume

25*. A rigid tank contains 6 kg of an ideal gas at 3 atm and 40 °C. Now a valve is opened, and half of the mass of the gas is allowed to escape. If the final pressure in the tank is 2.2 atm, the final temperature in the tank is

    (A) 459 °C
    (B) 186 K
    (C) 20 °C
    (D) 186 °C

26*. The pressure of an automobile tire is measured to be 190 kPa (gage) before a trip and 215 kPa (gage) after the trip at a location where the atmospheric pressure is 95 kPa. If the temperature of the air in the tire before the trip is 25 °C, the air temperature after the trip is (change in mass is negligible)

    (A) 64.2 °C
    (B) 51.14 °C
    (C) 27.2 °C
    (D) 28.3 °C

27*. If the work done on a closed system is 20 kJ/kg, and 40 kJ/kg heat is rejected from the system, its internal energy decrease by

    (A) 20 kJ/kg
    (B) 60 kJ/kg
    (C) 20 kJ/kg
    (D) 60 kJ/kg

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https://doi.org/10.1007/978-3-031-10584-5_2

28. The constant volume gas thermometer works on the principle that
    (A) At low pressure, the temperature of the gas is independent of its pressure at constant volume
    (B) At high pressure, the temperature of the gas is independent of its pressure at constant volume
    (C) At low pressure, the temperature of the gas is proportional to its pressure at constant volume
    (D) At high pressure, the temperature of the gas is proportional to its pressure at constant volume

29. There is no work transfer involved in this process
    (A) Adiabatic expansion
    (B) Isothermal expansion
    (C) Polytropic expansion
    (D) Free expansion

30. This process is one in which there is only work interaction between the system and the surroundings
    (A) Diabatic process
    (B) Adiabatic process
    (C) Isothermal process
    (D) Quasi-static process

31. In which of the following processes, the heat is fully converted into work?
    (A) Reversible adiabatic process
    (B) Reversible isobaric process
    (C) Reversible isometric process
    (D) Reversible isothermal process

32. The expression can be used for obtaining work of
    (A) Throttling process
    (B) Steady flow reversible process
    (C) Non-flow reversible process
    (D) Adiabatic irreversible process

33. In a system which undergoes an irreversible process, positive work done is 50 kJ and the heat added 45 kJ. What is the change in entropy?
    (A) Zero
    (B) Positive
    (C) Negative
    (D) Indeterminate

34. Consider the following processes in thermodynamic cycles:
    1. Constant pressure
    2. Constant volume
    3. Adiabatic
    4. Isothermal

Which of the above processes are involved in Stirling cycles?
(A) 1 and 2
(B) 2 and 4
(C) 2 and 3
(D) 3 and 4

35. Which of the following processes is ***not*** a part of the dual cycle?
    (A) Adiabatic compression
    (B) Constant volume expansion
    (C) Isothermal expansion
    (D) Constant pressure expansion

36. Match List I with List II and select the correct answer using the code given below the lists:

List I (Process)	List II (Heat transfer equal to)
(*a*) Constant volume	1. Zero
(*b*) Constant	2. Change in pressure internal energy
(*c*) Constant	3. Change in temperature enthalpy
(*d*) Constant entropy	4. Work done

Code:

	(*a*)	(*b*)	(*c*)	(*d*)
(*A*)	1	3	4	2
(*B*)	2	3	4	1
(*C*)	1	4	3	2
(*D*)	2	4	3	1

37. For reversible adiabatic compression in a steady flow process, the work transfer per unit mass is

(A) $\int pdV$
(B) $\int vdp$
(C) $\int Tds$
(D) $\int sdT$

38*. 10 kg of air at 300 K is contained in a rigid tank of volume 1 m^3. A slow leak develops and heat is added to maintain the initial pressure in the tank. If the final temperature reached is 500 K, the mass of air escaped is —

(A) 4 kg
(B) 5 kg
(C) 6 kg
(D) 8 kg

39*. The air contained in a frictionless piston-cylinder arrangement interacts with the surroundings and undergoes an isobaric process from an initial state of 300 K and 0.01 m^3 to a final state at which the volume is 0.03 m^3. Assume air to behave as an ideal gas with a gas constant of 0.287 kJ/kgK and a c_v value of 0.767 kJ/kgK. If the work transferred during the process is 10 kJ, the heat transferred during the pressure is

(A) 16.72 kJ from surroundings to the air
(B) 36.72 kJ from air to the surroundings
(C) 26.72 kJ from surroundings to the air
(D) 36.72 kJ from surroundings to the air

40. 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 Transisent
F Work	H Negative	J Inexact	L Point	N Boundary

(A) F-G-J-K-M E-G-I-K-N
(B) E-G-I-K-M F-H-I-K-N
(C) F-H-J-L-N E-H-I-L-M
(D) E-G-J-K-N F-H-J-K-M

41. Consider the following statements about heat and work:
I. Both are path functions
II. Both are exact differentials
III. Work in low grade energy and heat is high grade energy
IV. Heat is low grade energy and work is high grade energy

Which of the following options are true?

(A) I and II
(B) I and III
(C) I and IV
(D) II and III

42. Heat and work are

(A) Intensive properties
(B) Extensive properties
(C) Point functions
(D) Path functions

43*. A 100 W electric bulb was switched on in a 2.5 m × 3 m × 3 size thermally insulated room having a temperature of 20 °C. The room temperature at the end of 24 h will be

(A) 321 °C
(B) 314 °C
(C) 450 °C
(D) 470 °C

Statement for Linked Answer Questions

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

44*. 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

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

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

46*. 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 process is

(A) 0 kJ
(B) 1 kJ
(C) 2 kJ
(D) 3 kJ

47*. A given volume of material, initially at 100 °C, cools to 60 °C in 40 s. Assuming no phase change and only convective cooling to air at 20 °C, how long would it take for the same material to cool to 60 °C if the heat transfer coefficient were doubled?

(A) 4.4 s
(B) 20 s
(C) 80 s
(D) 3.0 s

48*. A frictionless piston-cylinder device contains a gas initially at 0.8 MPa and 0.015 m^3. It expands quasi-statically at a 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

49*. 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 kJ $kmol^{-1}$ K^{-1}. The work of compression of the gas (in kJ kg^{-1}) is

(A) 29.7
(B) 19.9
(C) 13.3
(D) 0

50*. The pressure of 10 kg of air is increased isothermally at 60 °C from 100 to 800 kPa. How much heat is rejected?

(A) 1810 kJ
(B) 1987 kJ
(C) 1260 kJ
(D) 1580 kJ

51*. Air is compressed adiabatically from 100 kPa and 20 °C to 800 kPa. Estimate final temperature

(A) 530 °C
(B) 440 °C
(C) 290 °C
(D) 257 °C

52. 1 kg of ice at 0 °C is mixed with 1 kg of water at 25 °C. The resulting temperature will be

(A) Less than 0 °C
(B) Greater than 0 °C
(C) Equal to 0 °C
(D) None of these

53. In an adiabatic process

(A) Work done is a path function
(B) Heat transfer is zero
(C) Temperature remains constant
(D) Enthalpy remains constant

54. Internal energy of a perfect gas is zero at

(A) 0 °C
(B) −180 °C
(C) Absolute zero temperature
(D) None of above

55*. The contents of a well-insulated tank are heated by a resistor of 20 Ω 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 is positive. The rates of heat (Q), work (W), and change in internal energy (ΔU) during the process in kW are

(A) $Q = 0$, $W = -2.3$, $\Delta U = +2.3$
(B) $Q = +2.3$, $W = 0$, $\Delta U = +2.3$
(C) $Q = -2.3$, $W = 0$, $\Delta U = -2.3$
(D) $Q = 0$, $W = +2.3$, $\Delta U = -2.3$

56. A fluid is contained in a closed system. For a reversible process, during which work is produced by the fluid, the correct statement is

(A) Change in fluid pressure is greater than zero
(B) Change in fluid pressure is less than zero
(C) Change in fluid volume is less than zero
(D) Change in fluid volume is greater than zero

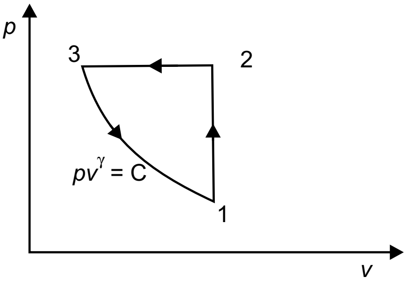
57. For a closed system, the difference between the heat added to the system and the work done by the system is equal to the change in

(A) Internal energy
(B) Enthalpy
(C) Entropy
(D) Temperature

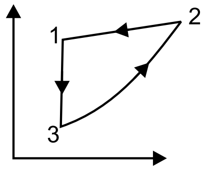
58*. One hundred people are at a meeting in a 10 m × 20 m × 3 m room when the air conditioning fails. Determine the temperature increase if the system is off for 15 min. Each person emits 400 kJ/h of heat and the lights add 300 W of energy. Neglect all other forms of energy input

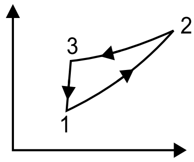
(A) 14 °C
(B) 18 °C
(C) 21 °C
(D) 25 °C

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

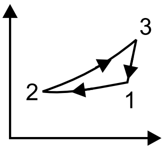


The above cycle is represent on T–s diagram by

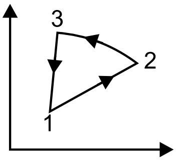
(A)


(B)


(C)

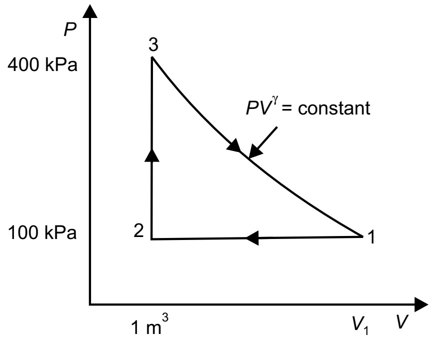


(D)



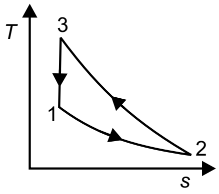
Common Data for Questions: 60 and 61

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

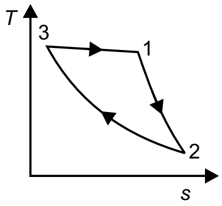


60. The above cycle is represented on T–s plane by

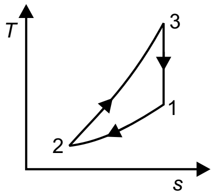
(A)



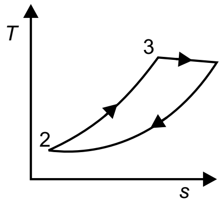
(B)



(C)



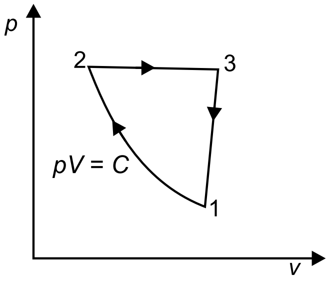
(D)



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

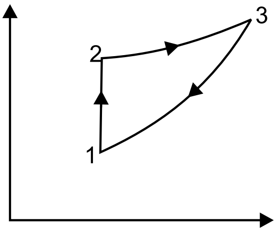
(A) 21
(B) 40.9
(C) 42.6
(D) 59.7

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

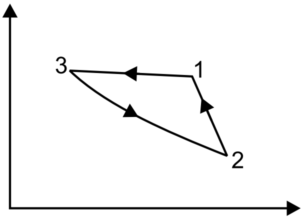


The above cycle is represent on T–s diagram

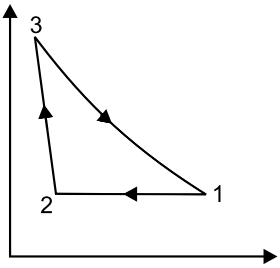
(A)



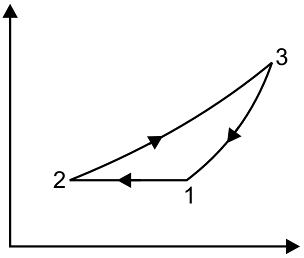
(B)



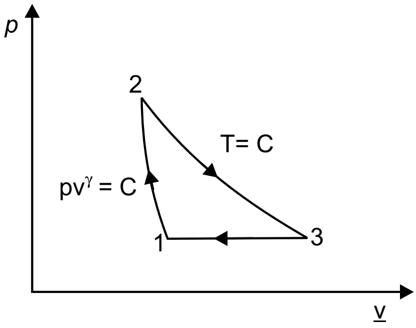
(C)



(D)

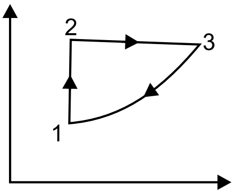


63. A thermodynamic cycle with an ideal gas as working fluid is given below

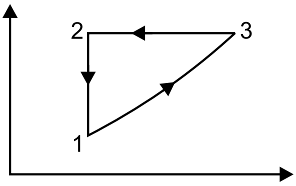


The above cycle is represent on *T–s* diagram by

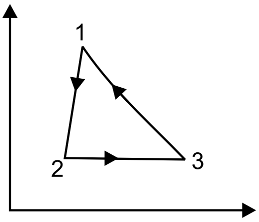
(A)



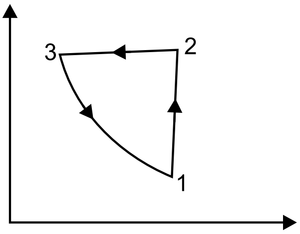
(B)



(C)



(D)



64. For steady flow through an insulated horizontal constant diameter pipe, this property remains constant

(A) Enthalpy
(B) Internal energy
(C) Entropy
(D) Volume

65. Which of the following relations is ***not*** valid for throttling process?

(A) Mass before expansion = Mass after expansion
(B) Pressure before expansion = Pressure after expansion
(C) Enthalpy before expansion = Enthalpy after expansion
(D) External work done = Zero

66. In a throttling process, which one of the following parameters remains constant?

(A) Temperature
(B) Pressure
(C) Enthalpy
(D) Entropy

67*. The enthalpy drop for flow through convergent horizontal nozzles is 100 kJ/kg. If the velocity of the approach at the inlet to the nozzle is negligible, the exit velocity of the fluid is

(A) 20 m/s
(B) 400 m/s
(C) 447.2 m/s
(D) 520.8 m/s

68*. 0.70 kg/s of air enters with a specific enthalpy of 290 kJ/kg and leaves it with 450 kJ/kg of specific enthalpy. Velocities at inlet and exit are 6 m/s and 2 m/s, respectively. Assuming adiabatic process, what is power input to the compressor?

(A) 120 kW
(B) 118 kW
(C) 115 kW
(D) 112 kW

69*. A nozzle acceleration air from 20 m/s to 200 m/s. What temperature change is expected?

(A) 10 °C
(B) 20 °C
(C) 30 °C
(D) 40 °C

70*. Steam at a pressure of 6 bars (enthalpy 2988 kJ/kg) passes through a frictionless nozzle to a pressure of 4 bars (enthalpy 2888 kJ/kg). Neglecting velocity of approach, the final velocity of steam at the exit of the nozzle will be

(A) 447.2 m/s
(B) 44.72 m/s
(C) 4.472 m/s
(D) 4472 m/s

71*. A steam turbine receives steam steadily at 10 bar with an enthalpy of 3000 kJ/kg and discharges at 1 bar with an enthalpy of 2700 kJ/kg. The work output is 250 kJ/kg. The change in KE and PE are negligible. The heat transfer from the turbine casing to the surroundings is equal to

(A) 0 kJ/kg
(B) 50 kJ/kg
(C) 150 kJ/kg
(D) 250 kJ/kg

72*. In a laboratory test, a well insulated horizontal nozzle has a flow of steam at a mass flow rate of 0.2 kg/s. Steam enters the nozzle at 12 MPa and 400 °C with negligible small kinetic energy. The condition at the nozzle outlet is measured to be 500 kPa and 350 °C. Steam properties: Specific enthalpy at inlet = 3263.88 kJ/kg, at exit = 3167.65 kJ/kg

The exit velocity of steam in m/s is

(A) 13.87
(B) 438.7
(C) 138.4
(D) 43.87

Statement for Linked Answer Questions: 73 and 74.

Air enters an adiabatic nozzle at 300 kPa, 500 K with a velocity 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.

73*. The exit temperature of the air is

(A) 516 K
(B) 532 K
(C) 484 K
(D) 468 K

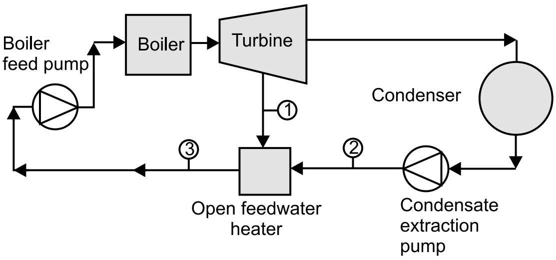
74*. The exit area of the nozzle in cm^2 is

(A) 90.1
(B) 56.3
(C) 4.4
(D) 12.9

75*. A compressor undergoes a reversible, steady flow process. The gas at the inlet and outlet of the compressor is designated as state 1 and as state 2, respectively. Potential and kinetic energy changes are to be ignored. The following notations are used v = 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) $\int_1^2 pdv$
(B) $\int_1^2 vdp$
(C) $v_1\,(p_2 - p_1)$
(D) $-p(v_1 - v_2)$

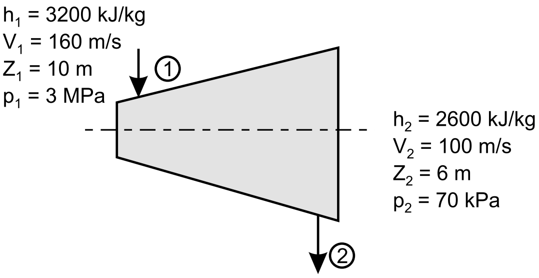
76*. A thermal power plant operates on a regenerative cycle with a single open feedwater heater, as shown in the figure. For the state points shown, the specific enthalpies are: h_1 = 2800 kJ/kg and h_2 = 200 kJ/kg the bleed to the feedwater heater is 20% of the boiler generations rate. The specific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



(A) 720 kJ/kg
(B) 2280 kJ/kg
(C) 1500 kJ/kg
(D) 3000 kJ/kg

Common Data for Questions: 77 and 78

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



77*. If the mass flow 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

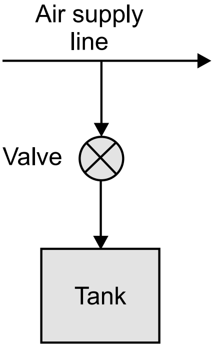
78*. Assume the above turbine to be part of a simple Rankine cycle. The density of water at the inlet of the pump is 1000 kg/m^3. Ignoring kinetic and potential energy effects, the specific work (in kJ/kg) supplied to the pump is

(A) 0.293
(B) 0.351
(C) 2.930
(D) 3.510

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

(A) The internal energy of the gas decrease 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

80*. 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 °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 °C
(B) Is less than 350 °C
(C) Is equal to 350 °C
(D) May be greater than, less than, or equal to 350 °C, depending on the volume of the tank

81. 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 a closed system in an adiabatic process is a point function
S: A liquid expands upon freezing when the slope of its fusion curve on the Pressure-Temperature diagram is negative.

(A) R and S
(B) P and Q
(C) Q, R, and S
(D) P, Q, and R

82. A system turbine receives steam steadily at 10 bar with an enthalpy of 3000 kJ/kg and discharge at 1 bar with an enthalpy of 2700 kJ/kg. The work output is 250 kJ/kg. The change in KE and PE are negligible. The heat transfer from the turbine casing to the surroundings is equal to—

(A) 0 kJ/kg
(B) 50 kJ/kg
(C) 250 kJ/kg
(D) 150 kJ/kg

83. The difference between constant pressure specific heat c_p and constant volume specific heat c_v for a pure substance

(A) Approaches zero at triple point
(B) Approaches zero as the absolute temperature approaches zero
(C) Is always equal to the gas constant R
(D) Approaches zero at critical point

84. Joule–Thomson coefficient is the slope at

(A) Constant enthalpy lines on T–s diagram
(B) Constant enthalpy lines on T–p diagram
(C) Inversion curve on T–s diagram
(D) Inversion curve on T–p diagram

85. Which gas shows a heating effect in the Joule–Thomson experiment while undergoing throttling process through a porous plug of cotton wool?

(A) Carbon dioxide
(B) Hydrogen
(C) Oxygen
(D) Nitrogen

86. The following remarks pertain to the properties of gases and vapours:

1. The equation of state is not by itself sufficient for the calculation of properties such as *u*, *h*, and *s*.
2. Throttling process is an adiabatic steady flow process.
3. Increase in entropy in wet region is directly proportional to the increase in enthalpy.

Which of these remarks are correct?

(A) 1, 2 and 3
(B) 1 and 2 only
(C) 1 and 3 only
(D) 2 and 3 only

3 Second Law of Thermodynamics

87. Select an acceptable paraphrase of the Kelvin-Planck statement of the second law.
    - (A) No process can produce more work than the heat that it accepts
    - (B) No engine can produce more work than the heat that it intakes
    - (C) An engine cannot produce work without accepting heat
    - (D) An engine has to reject heat

88. Which of the following can be assumed to be reversible?
    - (A) A paddle wheel
    - (B) A burst membrane
    - (C) A resistance heater
    - (D) A piston compressing gas in a race engine

89. Select an incorrect statement relating to a Carnot cycle.
    - (A) There are two adiabatic process
    - (B) Work occurs for all four processes
    - (C) Each process is a reversible process
    - (D) Work occurs for two adiabatic processes

90. Absolute zero temperature signifies
    - (A) Minimum temperature attainable
    - (B) The temperature of the heat reversible to which a Carnot engine rejects no heat
    - (C) The temperature of the heat reversible to which a Carnot engine rejects all the heat that is taken in
    - (D) Both (A) and (B)

91*. 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 of 50 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 reversible at 17 °C and 75 °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

92*. An inventor claims a thermal engine operates between ocean layers at 27 °C and 10 °C. It produces 10 kW and discharges 9900 kJ/min. Such an engine is
    - (A) Impossible
    - (B) Possible
    - (C) Probable
    - (D) Reversible

S. Kumar, *Problems and Solutions in Thermal Engineering*,
https://doi.org/10.1007/978-3-031-10584-5_3

93*. A Carnot engine operates between receivers at 20 °C and 200 °C. If 10 kW of power is produced, find the rejected heat rate

(A) 20.2 kJ/s
(B) 25.27 kJ/s
(C) 12.1 kJ/s
(D) 16.3 kJ/s

94*. An inventor states that his new conceptual engine, while operating between temperature limits of 377 °C and 27 °C, will reject 50% of heat absorbed from the source. What type of cycle will this engine have?

(A) Carnot cycle
(B) Stirling cycle
(C) Impossible cycle
(D) Possible cycle

95*. A heat engine is supplied with 2512 kJ/min of heat at 650 °C. Heat rejection with 900 kJ/min takes place at 100 °C. This type of heat engine is

(A) Ideal
(B) Irreversible
(C) Impossible
(D) Practical

96*. The refrigerating efficiency, that is, the ratio of the actual COP to reversible COP of a refrigeration cycle is 0.8, the condenser and evaporator temperatures are 51 °C and −30 °C, respectively. If the cooling capacity of the plant is 2.4 kW then what is the work requirement?

(A) 1.00 kW
(B) 1.33 kW
(C) 1.25 kW
(D) 2.08 kW

97*. A Carnot cycle refrigerator operates between 250 and 300 K. Its coefficient of performance is—

(A) 6
(B) 5
(C) 1.2
(D) 0.8

98*. A condenser of a refrigeration system rejects heat at a rate of 120 kW, while its compressor consumes a power of 30 kW. The coefficient of performance of the system would be—

(A) 0.25
(B) 4
(C) 0.333
(D) 3

99*. The thermal efficiency of a Carnot heat engine is 30%. If the engine is reversed in the operation of work as a heat pump with operation conditions unchanged, then what will be the COP for the heat pump?

(A) 0.30
(B) 2.33
(C) 3.33
(D) Cannot be calculated

100*. A reversible heat engine rejects 50 percent of the heat supplied during a cycle of operation. If this engine is reversed and operates as a heat pump, then what is its coefficient of performance?

(A) 1.0
(B) 1.5
(C) 2.0
(D) 2.5

101*. An engine working on Carnot cycle rejects 40% of absorbed heat from the source, while the sink temperature is maintained at 270 °C, then what is the source temperature?

(A) 750 °C
(B) 477 °C
(C) 203 °C
(D) 67.5 °C

102*. An inventor says that his new concept of an engine while working between temperature limits of 27 °C and 327 °C rejects 45% of the heat absorbed from the source. His engine is then equivalent to which one of the following engines?

(A) Carnot engine
(B) Diesel engine
(C) An impossible engine
(D) Ericsson engine

103*. A series combination of two Carnot engines operates between the temperature of 180 °C and 20 °C. If the engines produce an equal amount of work, then what is the intermediate temperature?

(A) 80 °C
(B) 90 °C
(C) 100 °C
(D) 110 °C

104. Three engines *A*, *B,* and *C* operating on Carnot cycle use working substances as Argon, Oxygen, and Air, respectively. Which engine will have higher efficiency?

(A) Engine *A*
(B) Engine *B*
(C) Engine *C*
(D) All engines have same efficiency

105. A heat engine a supplied with 250 kJ/s of heat at a constant fixed temperature of 227 °C, the heat is rejected at 27 °C, the cycle is reversible, then what amount of heat is rejected?

(A) 250 kJ/s
(B) 200 kJ/s
(C) 180 kJ/s
(D) 150 kJ/s

106. Consider the following statement for a throttling process.

1. It is an adiabatic process
2. There is no work transfer in the process
3. Entropy increase in throttling process

Which of these statements are correct?

(A) 1, 2 and 3
(B) 1 and 2 only
(C) 2 and 3 only
(D) 1 and 3 only

107. The continual motion of a movable device in absence of friction

(A) Violates the first law of thermodynamics
(B) Violates the second law of thermodynamics
(C) Is the perpetual motion of the second kind
(D) Is the perpetual motion of the third kind

108. For a given temperature T_1 as the difference between T_1 and T_2 increases, the COP of a Carnot heat pump

(A) Increases
(B) Decreases
(C) Does not change
(D) First decreases, then increase

109*. The maximum efficiency that a heat engine, operating between 327 °C and 27 °C, can be

(A) 1/2
(B) 1/4
(C) 3/4
(D) 1/4

110*. A Carnot engine uses nitrogen as the working fluid. The heat supplied is 53 kJ and adiabatic expansion ratio is 16:1. The receiver temperature is 295 K. The heat rejected in kJ is

(A) 27.75
(B) 17.48
(C) 20.50
(D) 230

111. Consider the following statements about a Carnot cycle:

1. It is an ideal hypothetical cycle.
2. It is a reversible cycle.
3. It consists of two isothermals and two reversible adiabatic.
4. Its efficiency exceeds 100%

Of these statements:

(A) 1, 2, and 3 are true
(B) 1 and 3 are true
(C) 1, 2, 3, and 4 are true
(D) 2, 3, and 4 are true

Entropy

4

112. As the time is passing, the entropy of the universe
    - (A) Is decreasing
    - (B) Is increasing
    - (C) Remains constant
    - (D) All of above

113. The absolute entropy for all crystalline substance at absolute zero temperature
    - (A) Is positive
    - (B) Is negative
    - (C) Is zero
    - (D) Both (*A*) and (*C*)

114. As the entropy of the universe is increasing, day by day, the work producing capacity of a heat engine is
    - (A) Decreasing
    - (B) Increasing
    - (C) Not changed
    - (D) All of above

115. Which of the following statements is incorrect?
    - (A) Work must be input if energy is transferred from a cold body to a hot body
    - (B) The entropy of a baked potato decreases as it cools
    - (C) If ice is melted in water in an insulated container, the net entropy decreases
    - (D) The entropy of an isolated system must remain constant or increase

116*. 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

117. 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

118. For a thermodynamic cycle to be irreversible, it is necessary that
    - (A) $\oint \frac{\delta Q}{T} = 0$
    - (B) $\oint \frac{\delta Q}{T} < 0$
    - (C) $\oint \frac{\delta Q}{T} > 0$
    - (D) $\oint \frac{\delta Q}{T} \geq 0$

S. Kumar, *Problems and Solutions in Thermal Engineering*,
https://doi.org/10.1007/978-3-031-10584-5_4

119. Enthatpy (H) is defined as

(A) $H = U + pV$
(B) $H = U - pV$
(C) $H = U + TS$
(D) $H = mc_v T$

120. Helmholtz free energy (F) is defined as

(A) $F = U + TS$
(B) $F = U - TS$
(C) $F = H - TS$
(D) $F = H + TS$

121. Gibbs free energy (G) is defined as

(A) $G = U + TS$
(B) $G = H + pV$
(C) $G = H - TS$
(D) $G = U - TS$

122. A certain amount of fluid at temperature T_1 is mixed with an equal amount of the same fluid at temperature T_2 in an insulated container. With total fluid as the system, consider the following statements:

I. Entropy of the system is conserved.
II. Energy of the system is conserved.
III. Entropy of the system increases.
IV. Entropy of the system decreases.

Which of the following options is true?

(A) II and III
(B) I and II
(C) I and IV
(D) Only I

123. The statements that the entropy of a pure substance in complete thermodynamic equilibrium becomes zero at the absolute zero of temperature is known as

(A) Law of entropy
(B) First law of thermodynamics
(C) Second law of thermodynamics
(D) Third law of thermodynamics

124. For a gas that is allowed to expand reversibly and adiabatically, there is no change in

(A) Internal energy
(B) Temperature
(C) Entropy
(D) Enthalpy

125. The enthalpy change during reversible closed system is given by

(A) $dH = -SdT - pdV$
(B) $dH = SdT + Vdp$
(C) $dH = TdS + pdV$
(D) $dH = TdS + Vdp$

126*. An ideal gas of mass m and temperature T_1 undergoes a reversible isothermal process from an initial pressure P_1 to a final pressure P_2. The heat loss during the process is Q. The entropy change ΔS of the gas is

(A) $\mathrm{mR}\ln\left(\frac{P_2}{P_1}\right)$
(B) $\mathrm{mR}\ln\left(\frac{P_1}{P_2}\right)$
(C) $\mathrm{mR}\ln\left(\frac{P_2}{P_1}\right) - \frac{Q}{T_1}$
(D) Zero

127*. Determine the available energy of furnace gas $c_p = 1.046$ kJ/kgK, when it is cooled from 1260 to 480 K at constant pressure. The surroundings are at 295 K.

(A) −518.1 kJ/kg
(B) 815.88 kJ/kg
(C) 518.1 kJ/kg
(D) none of these

128*. When the air is throttled, there is an entropy increase. For 2 kg/s of air the entropy increases by 0.06 kW/K. The pressure ratio (final to initial state) is (gas constant: R = 0.287 kJ/kgK)

(A) 0.8
(B) 0.9
(C) 1
(D) none of thes

129*. Air is compressed from an initial state of 100 kPa and 17 °C to a final state of 600 kPa and 57 °C. R = 0.287 kJ/kgK, c_v = 0.718 kJ/kgK. The entropy change of air during the process in kJ/kgK is

(A) 0.385
(B) −0.385
(C) 0.421
(D) −0.421

130*. A heat transformer is a device that transfers a part of the heat, supplied to it 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 at 300 K is

(A) 12.50
(B) 14.29
(C) 33.33
(D) 57.14

131. 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

132*. Consider the following two processes:

(A) A heat source at 1200 K loses 2500 kJ of heat of sink at 800 K
(B) A heat source at 800 K loses 2000 kJ of heat of sink of 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 equal
(D) Both the processes are reversible

Common Data Questions: 133 and 134

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, $h = c_pT$.

Internal energy, $u = c_vT$.

133*. 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

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

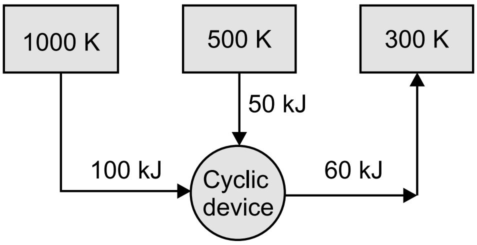
(A) −0.155
(B) 0
(C) 0.160
(D) 0.355

135. Two moles of oxygen are mixed adiabatically with another Two moles of oxygen in a mixing chamber so that the final total pressure and temperature of the mixture become the 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) $-R$ In 2
(B) 0
(C) R ln 2
(D) R In 4

136*. A cyclic device operates between three reservoirs, as shown in the figure. Heat is transferred from the cyclic device. It is assumed that heat transfer between each thermal reservoir and the cycle 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

137. Which of the following entropy relationship for a process is wrong?

(A) Air: V = constant, $ds = c_v \log_e \frac{T_2}{T_1}$
(B) Water: $ds = c_p \log_e \frac{T_2}{T_1}$
(C) Reservoir: $ds = c_p \log_e \frac{T_2}{T_1}$
(D) Iron: $ds = c_p \log_e \frac{T_2}{T_1}$

138. Determine the total entropy change if 10 kg of ice at 0 °C is mixed in an insulated container with 20 kg of water at 20 °C. The latent heat of ice is 340 kJ/kg

(A) 0.21 kJ/K
(B) 12.03 kJ/K
(C) 6.12 kJ/K
(D) 1.2 kJ/K

139. Which of the following second law statements is wrong?

(A) The entropy of an isolated system must remains constant or increased
(B) The entropy of a hot copper block decreases as it cools
(C) If ice is melted in water in an insulated container, the net entropy decreases
(D) Work must be input if energy is transferred from a cold body to a hot body.

140*. 100 kJ of energy is transferred from a heat reservoir at 1000 K to a heat reservoir at 500 K. The ambient temperature is 300 K. The loss of available energy due to heat transfer process is

(A) 20 kJ
(B) 30 kJ
(C) 40 kJ
(D) 50 kJ

141. When a system reaches the state of equilibrium, the following property assumes its maximum value

(A) Availability
(B) Entropy
(C) Gibbs functions
(D) Helmholtz function

5

Properties of Pure Substance

142. Consider the following statements about a pure substance

   1. It is a one component system.
   2. It is of constant chemical composition throughout.
   3. It always exists in one phase.

(A) 1 and 2 are true
(B) 2 alone is true
(C) 2 and 3 are true
(D) 1, 2, and 3 are true

143. Gibbs phase rule is expressed as

(A) $F = C + P - 2$
(B) $F = C - P + 2$
(C) $F = C - P - 2$
(D) $F = P - C + 2$

where

F = degrees of freedom,
C = number of components in the system,
P = number of phases which exists in a given system

144. For a liquid phase, the degree of freedom is

(A) 1
(B) 2
(C) 3
(D) 0

145. For a liquid phase, the degrees of freedom are

(A) Temperature and volume
(B) Temperature and pressure
(C) Pressure and volume
(D) All of the above

146. Degree of freedom of a system ice–water–vapour will be

(A) 0
(B) 1
(C) 3
(D) 2

147. For a pure substance in a triple point, the degree of freedom is

(A) 3
(B) 2
(C) 1
(D) 0

148. The minimum number of process that can exist in a system is

(A) 1
(B) 0
(C) 2
(D) 3

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149. Degree of freedom of a system consisting of a gaseous mixture of H_2 and NH_3 will be

(A) 3
(B) 2
(C) 1
(D) 0

150. What is the degree of freedom for a system comprising liquid water in equilibrium with its vapour?

(A) 0
(B) 1
(C) 2
(D) 3

151. Consider a system of ice, liquid, and water vapour. The number of independent intensive properties in the system is

(A) 0
(B) 1
(C) 2
(D) 3

152*. A small steam whistle (perfectly insulated and doing no shaft work) causes a drop of 200 J/kg in the enthalpy of the steam from entry to exit. If the kinetic energy of steam entry is negligible, the velocity of steam at exit is—

(A) 4 m/s
(B) 20 m/s
(C) 40 m/s
(D) 632.45 m/s

153. Consider the following statements:

1. Water always boils at 100 °C
2. The triple point temperature of water is 273.01 K
3. The critical temperature of water is 374.15 K
4. The density of water is 1000 kg/m^3 at 4 °C

Of these statements:

(A) 1, 2, 3, and 4 are true
(B) 1, 2, 3, and are false
(C) 1, 2, and 3 are true
(D) 2 and 3 are true

154. In steam tables, the entropy is shown as zero for

(A) Saturated liquid at 0 °C
(B) Saturated vapor at 0 °C
(C) Saturated liquid at atmospheric pressure
(D) Saturated vapor at atmospheric pressure

155. With increase of pressure, the latent heat of steam—

(A) Remains same
(B) Decreases
(C) Increases
(D) Behave unpredictably

156. 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

157*. A 300 m^3 rigid tank is filled with a saturated liquid–vapour mixture of water at 200 kPa. If 25% of the mass of liquid and 75% of the mass of vapour, the total mass in the tank is

(A) 451 kg
(B) 282 kg
(C) 338 kg
(D) 300 kg

At 200 kPa, the specific volumes of liquid and vapour are 0.001061 m^3/kg and 0.886 m^3/kg, respectively.

158*. Water is boiled in a pan on a stove at see-level. During 10 min of boiling, it is observed that 200 g of water has evaporated. Then the rate of heat transfer to the water is

(A) 53.51 kJ/min
(B) 45.14 kJ/min
(C) 41.82 kJ/min
(D) 0.84 kJ/min

At sea level, the latent heat of vaporization is 2257 kJ/kg.

159. 1 gm of ice of 0 °C is converted into water at the same temperature. The change in internal energy of the system is

(A) 80 kcal
(B) 0.08 kcal
(C) 0.04 kcal
(D) 40 kcal

160. No substance can exist in the liquid phase in stable equilibrium

(A) At temperature below the triple point temperature
(B) At pressure below the triple point pressure
(C) At pressure above the triple point pressure
(D) At temperature above the triple point temperature

161. The triple point of water is 0.01 °C. What is in Kelvin?

(A) 273.01 K
(B) 273.16 K
(C) 273 K
(D) All of the above

162*. Given below is an extract from steam table.

Temperature (°C)	P_{sat} (bar)	Specific volume (m^3/kg)		Enthaply (kJ/kg)	
		Saturated liquid	Saturated vapour	Saturated liquid	Saturated vapour
45	0.09593	0.001010	15.26	188.45	2394.8
342.24	150	0.0011658	0.010337	1610.5	2610.5

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

(A) 203.60
(B) 200.53
(C) 196.38
(D) 188.45

Statement of Linked Answer Questions

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

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	857.5	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

163*. The specific enthalpy data are in columns

(A) 3 and 7
(B) 3 and 8
(C) 5 and 7
(D) 5 and 8

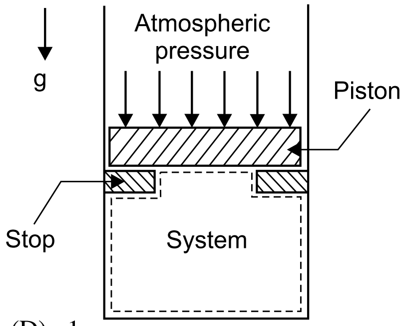
164*. When saturated liquid at 40 °C is throttled to –20 °C, the quality at exit will be

(A) 0.189
(B) 0.212
(C) 0.231
(D) 0.788

165. 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

Common Data Questions: 166, 167, and 168

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 show 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 unit and 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 °C. The average temperature of the system boundary can be taken as 175 °C. The heat transfer to the system is 1 kJ, during which its entropy increases by 10 J/K.



Specific volumes of liquid (v_f) and vapour (v_g) phase, as well as values of saturation temperatures, are given in the table below:

Pressure (kPa)	Saturation temperature, T_{sat} (°C)	v_f (kg)	v_g (m^3/kg)
100	100	0.001	0.1
200	200	0.0015	0.002

166*. 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.

167*. 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

168*. The net entropy generation (considering the system and the thermal reservoir together) during the process is closed to
    - (A) 7.5 J/K
    - (B) 7.7 J/K
    - (C) 8.5 J/K
    - (D) 10 J/k

169. A pressure-cooker contains water at 100 °C with volume being one-tenth of vapor volume. At 100 °C, the specific volume for saturated liquid and saturated vapour is 0.001 m^3/kg, respectively. The dryness fraction for the mixture is
    - (A) $\frac{1}{160}$
    - (B) $\frac{1}{161}$
    - (C) $\frac{160}{161}$
    - (D) 1

Vapor Power Cycles

6

170. The Rankine power cycle is idealized which of the following is not of the idealization?

   (A) Friction is absent
   (B) Heat transfer does not occur across a finite temperature difference
   (C) Pressure drops in pipes are neglected
   (D) Pipes connecting components are insulated

171. The component of the Rankine cycle that leads to a relatively low efficiency is the

   (A) Boiler
   (B) Turbine
   (C) Pump
   (D) Condenser

172*. What is the difference between the temperature of the feedwater outlet and the saturation temperature of steam entering the heater called?

   (A) Pinch point
   (B) Terminal temperature difference
   (C) LMTD
   (D) Terminal point

173. Mean temperature of heat addition gets increased resulting in an increase in cycle thermal efficiency. What is this cycle called?

   (A) Regenerative cycle
   (B) Reheat cycle
   (C) Carnot cycle
   (D) Brayton cycle

174. Consider the following statements pepertaining to the features of a regenerative steam cycle plant as compared to a non-regenerative plant.

   1. It increases the cycle efficiency
   2. It requires a bigger boiler.
   3. It requires a smaller condenser

   Which of the above statements are correct?

   (A) 1, 2, and 3
   (B) 1 and 3
   (C) 1 and 2
   (D) 2 and 3

175. Why is the thermal efficiency of superheat (modified) Rankine cycle higher than that of simple Rankine cycle?

   (A) Enthalpy of steam is higher for superheat cycle
   (B) Mean temperature of heat addition is higher.
   (C) Temperature of the condenser is high
   (D) Quality of steam in the condenser is low

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176. Consider the following statements in connection of Rankine cycle with regenerative feedwater heating:
    1. It reduces boiler capacity for a given output
    2. It reduces the temperature stresses in the boiler
    3. It improves the cycle efficiency
    4. It provides a convenient means of deaerating the feed water

    Which of the above statements are correct?

    (A) 1 and 2 only
    (B) 2 and 3 only
    (C) 3 and 4 only
    (D) 2, 3, and 4

177. Determine the correctness or otherwise of the following Assertion (A) and the Reason (R).

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

    Reasons (**R**): The regenerative feedwater 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.

178. A power plant which uses a gas turbine followed by a steam turbine for power generation is called:

    (A) Topping cycle
    (B) Bottoming cycle
    (C) Brayton cycle
    (D) Combined cycle

179. Which of the following thermal plants will have the highest overall thermal efficiency?

    (A) Steam power plant
    (B) Gas turbine power plant
    (C) Combined gas and steam power plant
    (D) Diesel power plant

180. For two cycles coupled in series, the topping cycle has an efficiency of 30% and the bottoming cycle has an efficiency of 20%. The overall combined cycle efficiency is

    (A) 50%
    (B) 44%
    (C) 38%
    (D) 55%

181*. 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 of kg/kW-hour is

    (A) 3.60
    (B) 0.36
    (C) 0.06
    (D) 0.01

182*. In a certain steam plant, the turbine develops 1.2 MW. The heat supplied to the steam in the boiler is 3000 kJ/kg, the heat rejected by the system to cooling water of the condenser is 2000 kJ/kg, and feed pump work required to pump the condensate back into the boiler is 6 kW. The steam flow rate round the cycle is—

    (A) 1.194 Kg/s
    (B) 1 Kg/s
    (C) 2.2 Kg/s
    (D) none of these

183. In a Rankine cycle, with the maximum steam temperature being fixed from metallurgical consideration as—boiler pressure increase.
   - (A) Condenser load will increase
   - (B) Quality of turbine exhaust will decrease
   - (C) Quality of turbine exhaust will increase
   - (D) Quality of turbine exhaust will remain unchanged

Common Data Question 184 and 185

In a steam power plant operating on the Rankine cycle, steam enters the turbine at 4 MPa, 350 °C and exits 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 the table.

State	h (kJ kg^{-1})		s (kJ kg^{-1} K^{-1})		v (m^{3}kg^{-1})	
Steam: 4 MPa, 350 °C	3092.5		6.5821		0.06645	
Water: 5 kPa	h_f	h_g	s_f	s_g	v_f	v_g
	225.94	2599.1	0.7549	8.0085	0.001014	10.02

h is specific enthalpy, *s* is specific entropy, and *v* the specific volume; subscripts *f* and *g* denote saturated liquid state and saturated vapour state.

184. The net work output (kJ kg^{-1}) of the cycle is
   - (A) 498
   - (B) 775
   - (C) 860
   - (D) 957

185. Heat supplied (kJ kg^{-1}) to the cycle is
   - (A) 2372
   - (B) 2576
   - (C) 2863
   - (D) 3092

186. Steam enters an adiabatic turbine operating at a 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

Common Date Question: 187, 188, and 189

187. Find the specific work output of the insulated turbine as shown in Figure

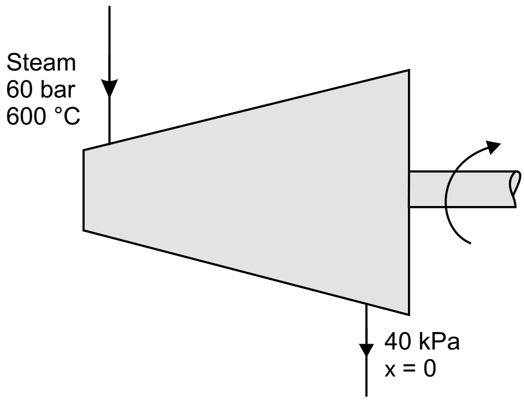
Saturated Steam (Pressure Based)

P	Specific enthalpy kJ/kg		Specific entropy kJ/kgK	
40 kPa	h_f	h_g	s_f	s_g
	317.7	2636.8	1.026	7.67

Superheated steam Table

T	p = 60 bar	
600 °C	h	s
	3658.4 kJ/kg	7.1677 kJ/kgK

   - (A) 1410 kJ/kg
   - (B) 1021 kJ/kg
   - (C) 1200 kJ/kg
   - (D) 1360 kJ/kg



188. Find the maximum specific work output of the above turbine.

    (A) 1410 kJ/kg
    (B) 1021 kJ/kg
    (C) 1197 kJ/kg
    (D) 1360 kJ/kg

189. Calculate the efficiency of the turbine

    (A) 85%
    (B) 89%
    (C) 89%
    (D) 93%

190. The minimum power needed by a water pump that increases the pressure of 4 kg/s from 100 kPa to 6 MPa is

    (A) 23.6 kW
    (B) 236 kW
    (C) 250 kW
    (D) 6 kW

191. A thermal electric power plant produces 1000 MW of power. If the coal release 900×10^7 kJ/h of energy, then what is the rate at which heat is rejected from the power plant?

    (A) 500 MW
    (B) 1000 MW
    (C) 1500 MW
    (D) 2000 MW

192. In a certain steam plant, the turbine develops 1.2 MW. The heat supplied to the steam in the boiler is 3000 kJ/kg, the heat rejected by the system to cooling water to the condenser is 2000 kJ/kg, and feed pump work required to pump the condensate back into the boiler is 6 kW. The steam flow rate round the cycle is

    (A) 1.194 kg/s
    (B) 1 kg/s
    (C) 2.2 kg/s
    (D) none of these

IC Engines

7

193. An IC engine has a bore and stroke each equal to 2 units. The total area to calculate heat loss from the engine can be taken as

(A) 4π
(B) 5π
(C) 6π
(D) 8π

194. The bore and stroke of the cylinder of a 6-cylinder engine working on an Otto-cycle are 17 cm and 30 cm, respectively, total clearance volume is 9225 cm^3, then what is the compression ratio?

(A) 7.8
(B) 6.2
(C) 15.8
(D) 5.4

195. The exhaust process in the Otto and diesel cycle is replaced with a constant volume process for which primary reason?

(A) To simulate the zero work of the actual exhaust process
(B) To restore the air to its original state
(C) To ensure that the first law is satisfied
(D) To simulate the zero heat transfer of the actual process

196. A diesel engine has a compression ratio of 14 and cut off takes place at 6% of the stroke. What is the value of cut off ratio?

(A) 1.25
(B) 2
(C) 2.15
(D) 1.78

197. At a given compression ratio, what is the relation among the Otto cycle efficiencies for the working fluids having $\gamma = 1.2$, 1.4, and 1.67?

(A) $(\eta_\gamma = 1.2) > (\eta_\gamma = 1.4) > (\eta_\gamma = 1.67)$
(B) $(\eta_\gamma = 1.2) < (\eta_\gamma = 1.4) < (\eta_\gamma = 1.67)$
(C) $(\eta_\gamma = 1.2) = (\eta_\gamma = 1.4) = (\eta_\gamma = 1.67)$
(D) $(\eta_\gamma = 1.2) \leq (\eta_\gamma = 1.4) \leq (\eta_\gamma = 1.67)$

198. A Carnot engine receives 100 kJ of heat at 600 K. Heat is rejected at 300 K. The displacement volume is 0.2 m^3. The mean effective pressure is

(A) 2 bar
(B) 2.5 bar
(C) 3 bar
(D) 3.5 bar

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199. This reversible cycle consists of constant volume heat addition, reversible adiabatic expansion, and constant pressure heat rejection

(A) Otto cycle
(B) Lenoir cycle
(C) Atkinson cycle
(D) Brayton cycle

200. For the same maximum pressure and heat input, the most sufficient cycle is

(A) Diesel cycle
(B) Dual cycle
(C) Otto cycle
(D) Stirling cycle

201. In an Otto cycle, twice as much energy is generated per cycle as is rejected through the exhaust. What is the compression ratio? (for air, $\gamma = 1.4$)

(A) 5.66
(B) 7.66
(C) 11.12
(D) none of these

202. 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 $\gamma = 1.4$, the air-standard cycle efficiency of the engine is

(A) 46.40%
(B) 56.10%
(C) 58.20%
(D) 62.80%

Common Data for Questions: 203 and 204

In two air standard cycles—one operating on 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 550 K in the Brayton cycle.

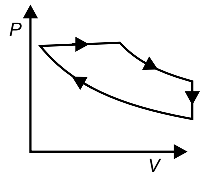
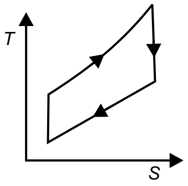
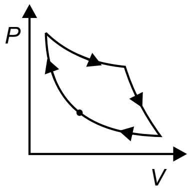
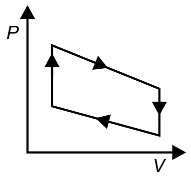
203. If η_O and η_B are the efficiencies of the Otto and Brayton cycles, then

(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.

204. 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 expansion is given.

205. Match List I and II and select the correct answer using the code given below the lists:

List I	List II
(*a*) 	1. Diesel cycle
(*b*) 	2. Carnot cycle
(*c*) 	3. Brayton cycle
(*d*) 	4. Otto cycle

Code:

	a	*b*	*c*	*d*
(***A***)	4	2	3	1
(***B***)	1	2	3	4
(***C***)	4	3	2	1
(***D***)	1	3	2	4

206. Which gas power cycle consists of four process during which work alone is transferred during two processes and heat alone is transferred during the other two processes?

(A) Atkinson cycle
(B) Carnot cycle
(C) Otto cycle
(D) Diesel cycle

207. Consider the following statements: Comparing the thermal efficiency of air standard Otto and Diesel cycles, the Diesel cycle is more efficient for constant maximum pressure and constant:

1. Heat input
2. Output
3. Maximum temperature

Which of the above statements is/are correct?

(A) 1, 2, and 3
(B) 1 only
(C) 1 and 2 only
(D) 2 and 3 only

208. Match List-I with List-II and select the correct answer using the codes given below the lists:

List I	List II
(Name of cycle)	(Characteristic of cycle efficiency η)
(*a*) Otto cycle	1. η depends only upon temperature limits
(*b*) Diesel cycle	2. η depends only on pressure limits
(*c*) Carnot cycle	3. η depends on volume compression ratio
(*d*) Baryton cycle	4. η depends on cut-off ratio and volume compression ratio

Code:

	a	*b*	*c*	*d*
(***A***)	3	4	1	2
(***B***)	1	4	3	2
(***C***)	3	2	1	4
(***D***)	1	2	3	4

209. 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

210. A turbo-charged four-stroke direct injection diesel engine has a displacement volume of 0.0259 m^3 (25.9 L). The engine has an output of 950 kW at 2200 rpm. The mean effective pressure in MPa is closest to

(A) 2
(B) 1
(C) 0.2
(D) 0.1

211. 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 °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.

(A) 103
(B) 310
(C) 515
(D) 1032

212. Group I shows different heat addition processes 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	(*B*)	P-S-1
	R-U-3		R-U-3
	P-S-1		P-S-4
	Q-T-2		P-T-2
(*C*)	R-T-3	(*D*)	P-T-4
	P-S-1		R-S-3
	P-T-4		P-S-1
	Q-S-5		P-S-5

Gas Turbine

8

213. An air compressor has an inlet temperature of 300 K. If it could be operated isentropically, the outlet temperature would be 575 K. If the actual outlet temperature was 600 K, how efficient temperature was 600 K, how efficient was the compressor?

(A) 80%
(B) 95%
(C) 85.2%
(D) 91.7%

214. In a gas turbine, working on Brayton cycle with regeneration, air exits from the compressor at 550 K and gas exits from the turbine at 800 K. If the effectiveness of the regenerator is 0.8, then what is the air temperature at the combustion chamber inlet?

(A) 600 K
(B) 650 K
(C) 700 K
(D) 750 K

215. A gas turbine works on which one of the following cycles?

(A) Otto
(B) Dual
(C) Rankine
(D) Brayton

216. Reheating in a gas turbine

(A) decreases the turbine work
(B) decreases the compressor work
(C) increase the turbine work
(D) increases the compressor work

217. Consider the following statements about modification in a gas turbine power plant working on a simple Brayton cycle:

1. Incorporation of regeneration process increases specific work output as well as thermal efficiency
2. Incorporation of regeneration process increases thermal efficiency but specific work output remains unchanged
3. Incorporation of inter cooling process in a multi-stage compression increases specific work output but the heat input also increases
4. Incorporation of intercooling process in a multi-stage compression system increases specific work output, the heat addition remains unchanged.

Which of the above statements are correct?

(A) 1 and 3
(B) 1 and 4
(C) 2 and 3
(D) 2 and 4

S. Kumar, *Problems and Solutions in Thermal Engineering*,
https://doi.org/10.1007/978-3-031-10584-5_8

218. Consider the following features for a gas turbine plant:

   1. Intercooling
   2. Regeneration
   3. Reheat

Which of the above features in a simple gas turbine cycle increases the network?

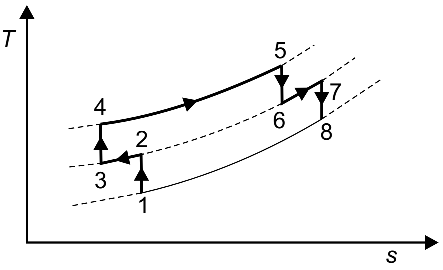
(A) 1, 2, and 3
(B) Only 1 and 2
(C) Only 2 and 3
(D) Only 1 and 3

219. Consider the following statements with reference to Gas turbine cycle:
   1. Regeneration increases thermal efficiency
   2. Reheating decreases the thermal efficiency
   3. Cycle efficiency increases when the maximum temperature of the cycle is increased

Of the these correct statements are

(A) 1, and 3
(B) 2 and 3
(C) 1 and 2
(D) 1 and 3

220. The cycle shown in the given figure represents a Gas Turbine Cycle with intercooling and reheating Match List X (Units) with List Y (Processes) and select the correct answer using the codes given below the Lists:



List X	List Y
(*a*) Intercooles	I. 1–2
(*b*) Combustor	II. 2–3
(*c*) Reheater	III. 3–4
(*d*) High pressure compressor	IV. 4–5
	V. 6–7

Code	***a***	***b***	***c***	***d***
(***A***)	I	II	IV	III
(***B***)	III	IV	V	II
(***C***)	II	IV	V	III
(***D***)	II	V	IV	I

Linked Date Questions: 221 and 222

A gas turbine operating on Brayton between the minimum and maximum tempera ting are 35 °C and 900 °C, respectively. The inlet and outlet pressure of the compressor is 100 kPa and 500 kPa, respectively.

221. The maximum work done per kg of air of

   (A) 288.50 kJ/kg
   (B) 280.25 kJ/kg
   (C) 305.25 kJ/kg
   (D) 225 kJ/kg

222. The ratio of efficiency of the Carnot cycle to Brayton cycle is

   (A) 1.33
   (B) 1.44
   (C) 1.51
   (D) 1.25

223. An ideal Brayton cycle, operating between the pressure limits of 1 bar and 6 bar, has a minimum and maximum temperature limits of 1 bar and 6 bar, has a 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

temperature in kelvin at the end of the compression and expansion processes are, respectively.

(A) 500 and 900
(B) 900 and 500
(C) 500 and 500
(D) 900 and 900

224. An open cycle gas turbine uses a fuel of calorific value 40,000 kJ/kg, with air–fuel ratio of 80:1 and develops a net output of 80 kJ/kg of air. What is the thermal efficiency of the cycle?

(A) 61%
(B) 16%
(C) 18%
(D) none of these

225. The power required to drive a turbo-compressor for a given pressure ratio decreases when

(A) Air is heated at entry
(B) Air is cooled at entry
(C) Air is cooled at exit
(D) Air is heated at exit

226. For the air standard Brayton cycle, an increase in the maximum temperature of the cycle, while keeping the pressure ratio the same would result in

(A) Increase in air standard efficiency
(B) Decrease in air standard efficiency
(C) No change in air standard efficiency
(D) Increase in the efficiency but reduction in net work

227. Optimum pressure ratio for maximum specific output for ideal gas turbine plant operating at an initial temperature of 300 K and maximum temperature of 100 K is closer to

(A) 4
(B) 8
(C) 12
(D) 16

228. Use of maximum pressure ratio, corresponding to maximum to minimum cycle temperature ratio in case of Joule cycle gives which one of the following?

(A) Maximum efficiency but very less specific work output
(B) Maximum efficiency and very high specific work output
(C) Minimum efficiency and very less specific work output
(D) Minimum efficiency but very high specific work output

229. The efficiency of a simple gas turbine can be improved by using a regenerator because the

(A) Work of compression is reduced.
(B) Heat required to be supplied is reduced
(C) Work output of the turbine is increased
(D) Heat rejected is increased

230. The use of a regenerator in a gas turbine cycle

(A) Increases efficiency but has no effect on output
(B) Increases output but has no effect on efficiency
(C) Increases both efficiency and output
(D) Increases efficiency but decreases output

231. Inter-cooling in gas turbines

(A) Decreases net output but increases thermal efficiency
(B) Increases net output but decreases thermal efficiency
(C) Decreases both net output and thermal efficiency.
(D) Increases both net output and thermal efficiency

232. Reheating in a gas turbine

(A) Increases the coprocessors work
(B) Decreases the compressor work

(C) Increases the turbine work
(D) Decreases the turbine work

233. A gas turbine cycle with heat exchanger and reheating improves

(A) Only the thermal efficiency
(B) Only the specific power output
(C) Both thermal efficiency and specific power output
(D) Neither thermal efficiency nor specific power output

234. Compressors used in gas turbines are of

(A) Reciprocating type
(B) Centrifugal type
(C) Axial-flow type
(D) All of the above

235. In a gas turbine power plant, reheating of gases between the high pressure and low pressure turbine stages will

(A) Improve turbine output
(B) Decrease turbine output
(C) Increases compressor output
(D) Decreases compressor work

236. In a gas turbine plant, regeneration is done to

(A) Increase compression work
(B) Decrease turbine work
(C) Limit the maximum temperature
(D) Improve plant efficiency

237. A gas turbine operating on the Brayton cycle has the maximum temperature of 1200 K and the minimum temperature of 300 K. The cycle efficiency for the maximum work capacity will be

(A) 75%
(B) 60%
(C) 50%
(D) 25%

238. The engines in a commercial jet aircraft operate on which of the basic cycles?

(A) Diesel
(B) Otto
(C) Brayton
(D) Carnot

Answers

1. (*B*)	2. (*A*)	3. (*B*)	4. (*C*)	5. (*A*)	6. (*C*).	7. (*D*)	8. (*A*)
9. (*A*)	10. (*B*)	11. (*C*)	12. (*D*)	13. (*B*)	14. (*C*)	15. (*C*)	16. (*C*)
17. (*A*)	18. (*B*)	19. (*C*)	20. (*A*)	21. (*A*)	22. (*C*)	23. (*B*)	24. (*D*)
33. (*B*)	34. (*B*)	35. (*C*)	36. (*B*)	37. (*B*)	38. (*A*)	39. (*D*)	40. (*D*)
41. (*C*)	42. (*D*)	43. (*D*)	44. (*D*)	45. (*C*)	46. (*C*)	47. (*C*)	48. (*A*)
49. (*A*)	50. (*B*)	51. (*D*)	52. (*C*)	53. (*B*)	54. (*C*)	55. (*A*)	56. (*D*)
57. (*A*)	58. (*A*)	59. (*B*)	60. (*C*)	61. (*A*)	62. (*D*)	63. (*A*)	64. (*A*)
65. (*B*)	66. (*C*)	67. (*C*)	68. (*D*)	69. (*B*)	70. (*A*)	71. (*B*)	72. (*B*)
73. (*C*)	74. (*D*)	75. (*B*)	76. (*A*)	77. (*A*)	78. (*C*)	79. (*C*)	80. (*A*)
81. (*A*)	82. (*B*)	83. (*B*)	84. (*B*)	85. (*B*)	86. (*A*)	87. (*D*)	88. (*D*)
89. (*D*)	90. (*B*)	91. (*C*)	92. (*A*)	93. (*B*)	94. (*D*)	95. (*C*)	96. (*A*)
97. (*B*)	98. (*D*)	99. (*C*)	100. (*C*)	101. (*D*)	102. (*D*)	103. (*C*)	104. (*D*)
105. (*D*)	106. (*A*)	107. (*D*)	108. (*B*)	109. (*A*)	110. (*B*)	111. (*A*)	112. (*B*)
113. (*C*)	114. (*A*)	115. (*C*)	116. (*A*)	117. (*D*)	118. (*B*)	119. (*A*)	120. (*B*)
121. (*C*)	122. (*A*)	123. (*D*)	124. (*C*)	125. (*D*)	126. (*B*)	127. (*C*)	128. (*B*)

(continued)

129. (*B*)	130. (*D*)	131. (*D*)	132. (*B*)	133. (*B*)	134. (*C*)	135. (*B*)	136. (*A*)
137. (*C*)	138. (*A*)	139. (*C*)	140. (*B*)	141. (*B*)	142. (*A*)	143. (*B*)	144. (*B*)
145. (*B*)	146. (*A*)	147. (*D*)	148. (*A*)	149. (*A*)	150. (*B*)	151. (*A*)	152. (*B*)
153. (*B*)	154. (*A*)	155. (*B*)	156. (*B*)	157. (*A*)	158. (*B*)	159. (*B*)	160. (*B*)
161. (*B*)	162. (*D*)	163. (*D*)	164. (*B*)	165. (*A*)	166. (*A*)	167. (*B*)	168. (*C*)
169. (*B*)	170. (*B*)	171. (*A*)	172. (*B*)	173. (*B*)	174. (*B*)	175. (*A*)	176. (*C*)
177. (*A*)	178. (*D*)	179. (*C*)	180. (*B*)	181. (*A*)	182. (*A*)	183. (*B*)	184. (*C*)
185. (*C*)	186. (*B*)	187. (*B*)	188. (*C*)	189. (*A*)	190. (*A*)	191. (*C*)	192. (*A*)
193. (*C*)	194. (*D*)	195. (*A*)	196. (*D*)	197. (*B*)	198. (*B*)	199. (*B*)	200. (*C*)
201. (*D*)	202. (*C*)	203. (*B*)	204. (*A*)	205. (*D*)	206. (*C*)	207. (*A*)	208. (*A*)
209. (*D*)	210. (*A*)	211. (*D*)	212. (*A*)	213. (*D*)	214. (*D*)	215. (*D*)	216. (*C*)
217. (*C*)	218. (*D*)	219. (*A*)	220. (*B*)	221. (*B*)	222. (*C*)	223. (*A*)	224. (*D*)
225. (*B*)	226. (*C*)	227. (*B*)	228. (*A*)	229. (*B*)	230. (*A*)	231. (*B*)	232. (*C*)
233. (*C*)	234. (*C*)	235. (*A*)	236. (*D*)	237. (*C*)	238. (*C*)		

MCQ-Solution

11. (*C*) $T_1 = 27\,°\mathrm{C} = (27 + 273)\,\mathrm{K} = 300\,\mathrm{K}$

$$p = C$$

Let $V_1 = C$

$$\begin{aligned} V_1 &= V \\ V_2 &= 3V_1 = 3V \\ T_2 &= ? \end{aligned}$$

Applying Charles' law,

$$\begin{aligned} \frac{V_1}{T_1} &= \frac{V_2}{T_2} \\ \frac{V}{300} &= \frac{3V}{T_2} \end{aligned}$$

or

$$\begin{aligned} T_2 &= 3 \times 300 = 900\,\mathrm{K} \\ &= (900 - 273)°\mathrm{C} = \mathbf{627°C}\,. \end{aligned}$$

12. (*D*) Let $p_1 = p$

$$p_2 = p + 0.01p = 1.01p$$

and let $V_1 = V$

$$V_2 = ?$$

We know that the Boyle's law,

$$\begin{aligned} p_1V_1 &= p_2V_2 \\ p \times V &= 1.01p \times V_2 \end{aligned}$$

or $V_2 = \frac{V}{1.01}$

Percentage decrease in volume

$$\begin{aligned} &= \left(\frac{V_1 - V_2}{V_1}\right) \times 100 \\ &= \left(\frac{V - \frac{V}{1.01}}{V}\right) \times 100 = \left(1 - \frac{1}{1.01}\right) \times 100 \\ &= \left(1 - \frac{100}{101}\right) \times 100 = \left(\frac{101 - 100}{101}\right) \times 100 = \mathbf{\frac{100}{101}}\% \end{aligned}$$

17. (*A*) Explanation: $pV^{\gamma} = \mathrm{C}$ for adiabatic process .

Taking $\log_e$ both sides, we get

$$\log_e p + \gamma \log_e V = \log_e C$$

Differentiating, we get

$$\frac{dp}{p} + \gamma \frac{dV}{V} = 0$$

or $\gamma \frac{dV}{V} = -\frac{dp}{p}$

or $\frac{\boldsymbol{dV}}{\boldsymbol{V}} = -\frac{1}{\gamma}\frac{\boldsymbol{dp}}{\boldsymbol{p}}$

25. (***D***) Explanation. Given data:

At initial state	At final state
m_1 = 6 kg	$m_2 = \frac{m_1}{2} = \frac{6}{2} = 3\,\text{kg}$
p_1 = 3 atm	p_2 = 2.2 atm
$T_1 = 40\,^\circ\text{C}$ $= (40+273)\,\text{K} = 313\,\text{K}$	T_2 = ?

$$\begin{aligned} pV &= mRT \\ \frac{p}{mT} &= \frac{R}{V} \qquad \therefore\ R\&V\ \text{are constant} \\ \frac{p_1}{m_1T_1} &= \frac{p_2}{m_2T_2} \\ \frac{3}{6\times 313} &= \frac{2.2}{3\times T_2} \end{aligned}$$

or $T_2 = 459.06\,\text{K} = \mathbf{186.06\,^\circ C}$

26. (***B***) Explanation. Given data:

Before trip		After trip
Gage pressure	p_{G1} = 190 kPa	p_{G2} = 215 kPa
	p_{atm} = 95 kPa	$p_2 = p_{G2} + p_{\text{atm}}$ $= 215 + 95 = 310\ \text{kPa}$
	$T_1 = 25\,^\circ\text{C}$ $= (25+273)\text{K} = 298\,\text{K}$	T_2 = ?

Absolute pressure:

$$\begin{aligned} p_1 &= p_{G1} + p_{\text{atm}} \\ &= 190 + 95 = 285\,\text{kPa} \end{aligned}$$

Equation of state,

$$pV = mRT$$

$$\frac{p}{T} = \frac{mR}{V} \qquad \therefore\ m, R\,\text{and}\ V\ \text{are constant}$$

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

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

$$\frac{285}{298} = \frac{310}{T_2}$$

or

$$T_2 = 324.15\,\text{K} = \mathbf{51.15\,^\circ C}$$

27. (***A***) Explanation

Work done:

$$W = -20\,\text{kJ/kg}$$

Heat transfer:

$$Q = -40\,\text{kJ/kg}$$

We know that

$$\begin{aligned} Q &= dU + W \\ -40 &= dU - 20 \end{aligned}$$

or

$$dU = -40 + 20 = -\mathbf{20\,kJ/kg}$$

Thus, the internal energy decreases by 20 kJ/kg

38. (***A***) Explanation

At initial state	At final state
m_1 = 10 kg	m_2 = ?
T_1 = 300 K	T_2 = 500 K
V_1 = 1 m^3	V_2 = V_1 = 1 m^3 Rigid tank
Let p_1 = p	p_2 = p_1 = p
From equation of state	
$p_1\ V_1 = m_1\ R\ T_1$	$p_2\ V_2 = m_2\ R\ T_2$
$\frac{p_1}{R} = \frac{m_1T_1}{V_1}$	$\frac{p_1}{R} = \frac{m_2T_2}{V_2} \quad \therefore\ p_2 = p_1$

$$\therefore\ \frac{m_1T_1}{V_1} = \frac{m_2T_2}{V_2}$$

$$\frac{10\times 300}{1} = \frac{m_2\times 500}{1}$$

or $m_2 = 6\,\mathrm{kg}$

$$\text{Mass of air escaped} = m_1 - m_2 = 10 - 6 = \mathbf{4\,kg}$$

39. (***B***) Explanation: Given data for the isobaric process.

$$T_1 = 300\,\mathrm{K}$$

$$V_1 = 0.01\,\mathrm{m}^3$$

$$V_2 = 0.03\,\mathrm{m}^3$$

$$R = 0.287\,\mathrm{kJ/kgK}$$

$$c_v = 0.767\,\mathrm{kJ/kgK}$$

$$W = 10\,\mathrm{kJ}$$

For isobaric process,

$$\frac{T_1}{V_1} = \frac{T_2}{V_2}$$

$$\frac{300}{0.01} = \frac{T_2}{0.03}$$

or $T_2 = 900\,\mathrm{K}$

$$W = p(V_2 - V_1)$$

$$10 = p(0.03 - 0.01)$$

or $p = 500\,\mathrm{kPa}$

$$p_1V_1 = mRT_1$$

$$500 \times 0.01 = m \times 0.287 \times 300$$

or $m = 0.058\,\mathrm{kg}$

$$\begin{aligned} Q &= mc_v(T_2 - T_1) + W \\ &= 0.058 \times 0.767(900 - 300) + 10 \\ &= 26.69 + 10 = \mathbf{36.69\,kJ} \end{aligned}$$

The +*ve* sign shows that heat transfer from surroundings to the air.

or

$$\begin{aligned} Q &= mc_p(T_2 - T_1) \\ &= 0.058 \times 1.054(900 - 300) \\ &= 36.679\,\mathrm{kJ} \end{aligned}$$

$$\begin{aligned} c_p - c_v &= R \\ c_p &= R + c_v \\ &= 0.287 + 0.767 \\ &= 1.054\,\mathrm{kJ/kgK} \end{aligned}$$

43. (***D***) Explanation: Given data:

$$Q = 100\,\mathrm{W} = 100\,\mathrm{J/s}$$

Volume of room:

$$V = 2.5 \times 3 \times 3 = 22.5\,\mathrm{m}^3$$

Time:

$$t = 24\,\mathrm{hr} = 24 \times 60 \times 60\,\mathrm{s} = 86400\,\mathrm{s}$$

Initial temperature:

$$T_1 = 20\,^\circ\mathrm{C} = (20 + 273)\mathrm{K} = 293\,\mathrm{K}$$

$$p_1V = mRT_1$$

where

$$p_1 = p_{\mathrm{atm}} = 1\,\mathrm{bar} = 100\,\mathrm{kPa}$$

$$R = 0.287\,\mathrm{kJ/kgK}$$

$$\therefore\ 100 \times 22.5 = m \times 0.287 \times 297$$

or

$$m = 26.75\,\mathrm{kg}$$

$$Q = 100\,\mathrm{J/s} = 100 \times 86400\,\mathrm{J} = 8640\,\mathrm{kJ}$$

also

$$Q = mc_v(T_2 - T_1)$$

$$\therefore\ 8640 = 26.75 \times 0.718(T_2 - 293)$$
$$449.84 = T_2 - 293$$

or

$$\begin{aligned} T_2 &= 449.84 + 297 = 742.84\,\text{K} \\ &= (742.84 - 273)\,^\circ\text{C} = 469.84\,^\circ\text{C} \approx 470\,^\circ\text{C} \end{aligned}$$

44. (*D*), 45 (*C*)

Explanation:

Gauge pressure:

$$p_{G1} = 1\,\text{bar} = 100\,\text{kPa}$$

Ambient temperature:

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

Absolute pressure:

$$p_1 = p_{G1} + p_{\text{atm}} = 100 + 101325 = 201.325\,\text{kPa}$$

Temperature:

$$T_2 = 5\,^\circ\text{C} = (5 + 273)\,\text{K} = 278\,\text{K}$$
$$p_1 V_1 = mRT_1$$
$$201.325 \times 2500 \times 10^{-6} = m \times 0.287 \times 288$$

or

$m = 6.089 \times 10^{-3}\,\text{kg}$

Heat loss:

$$\begin{aligned} Q &= mc_v(T_1 - T_2) \\ &= 6.089 \times 10^{-3} \times 0.718(288 - 278) \\ &= 0.0437\,\text{kJ} = \mathbf{43.7\,J} \end{aligned}$$

At

$$V = C$$

$$\frac{p_1}{T_1} = \frac{p_2}{T_2}$$
$$\frac{201.325}{288} = \frac{p_2}{278}$$

or

$$p_2 = 194.33\,\text{kPa}\,(\text{absolute})$$

also

$$p_2^* = p_{G2} + p_{\text{amt}}$$
$$\therefore\ 194.33 = p_{G2} + 101.325$$

or

$$p_{G2} = 93\,\text{kPa} = 0.93\,\text{bar}$$

Alternatively:

$$p_1 V = mRT_2$$
$$p_2 \times 2500 \times 10^{-6} = 6.089 \times 10^{-3} \times 0.287 \times 278$$

or

$$p_2 = 194.32\,\text{kPa}\,(\text{absolute})$$

also

$$p_2 = p_{G2} + p_{\text{atm}}$$
$$\therefore\ 194.32 = p_{G2} + 101.325$$

or

$$\begin{aligned} p_{G2} &= 92.99\,\text{kPa} \\ &\approx 93\,\text{kPa} \\ &= \mathbf{0.93\,bar} \end{aligned}$$

Case-II

If

$$p_{G1} = 1\,\text{bar} = 100\,\text{kPa}$$

Absolute pressure:

$$p_1 = p_{G1} + p_{\text{amt}}$$
$$= 100 + 101.325 = 201.325\,\text{kPa}$$
$$T_1 = 278\,\text{K}$$
$$T_2 = 288\,\text{K}$$

At

$$V = C$$
$$\frac{p_2}{T_2} = \frac{p_1}{T_1}$$
$$\frac{p_2}{288} = \frac{201.325}{278}$$

or

$$p_2 = 208.56\,\text{kPa}$$

also

$$p_2 = p_{G2} + p_{\text{amt}}$$
$$\therefore\ 208.56 = p_{G2} + 101.325$$

or

$$p_{G2} = 107.235\,\text{kPa} = \mathbf{1.07\,bar}$$

46. (*C*) Explanation:

Pressure of gas in a cylinder:

$$p = 300\,\text{kPa}$$

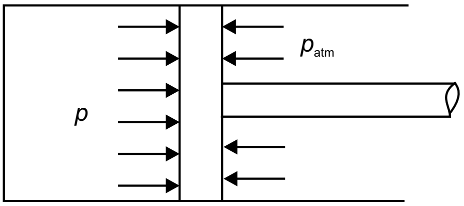
Ambient pressure:

$$p_{\text{atm}} = 100\,\text{kPa}$$

Change in volume:

$$dV = 0.01\,\text{m}^3$$

$$W_{\max} = (p - p_{\text{amt}})dV$$
$$= (300 - (100) \times 0.01 = \mathbf{2kJ}$$



47. (*B*) Explanation:

Rate of heat lost by materials = Rate of heat gained by air

$$mc\frac{dT}{t} = hA(T_s - T_\infty)$$
$$h \propto \frac{1}{t}$$

or

$$ht = C$$
$$h_1t_1 = h_2t_2$$
$$h \times 40 = 2ht_2$$

or

$$t_2 = 20\text{ s}$$

48. (*A*) Explanation: given data for constant temperature process

$$p_1 = 0.8\,\text{MPa} = 800\,\text{kPa}$$
$$V_1 = 0.015\,\text{m}^3$$
$$V_2 = 0.030\,\text{m}^3$$

We knows that work done for constant temperature process,

$$W = p_1V_1\log_e\frac{V_2}{V_1}$$
$$= 800 \times 0.015\log_e\frac{0.030}{0.015}$$
$$= 8.317\,\text{kJ} \approx \mathbf{8.32kJ}$$

49. (***A***) Explanation: Given data:

Adiabatic index:

$$\gamma = 1.67$$

Molecular weight:

$$M = 40$$

For adiabatic compression,

$$p_1 = 0.1\,\text{MPa}$$
$$T_1 = 300\,\text{K}$$
$$p_2 = 0.2\,\text{MPa}$$

Universal gas constant:

$$\overline{R} = 8.314\,\text{kJ/kmol K}$$

For process 1–2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$
$$\frac{T_2}{300} = \left(\frac{0.2}{0.1}\right)^{\frac{1.67-1}{1.67}}$$
$$\frac{T_2}{300} = (2)^{0.401}$$

or

$$T_2 = 396.12\,\text{K}$$

Gas constant:

$$R = \frac{\text{Universal gas constant} : \overline{R}}{\text{Molecular weight} : M} = \frac{8.314}{40} = 0.2078\,\text{kJ/kgK}$$

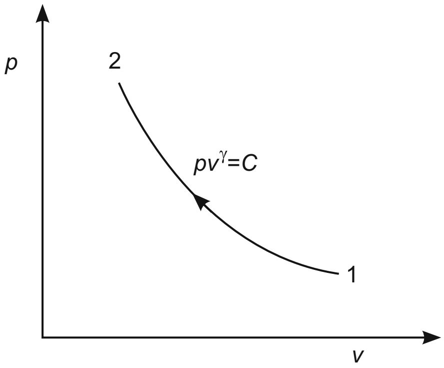
For unit mass,

$$w_{1-2} = \frac{p_1 v_1 - p_2 v_2}{\gamma - 1}$$

Work:

$$w_{1-2} = \frac{RT_1 - RT_2}{\gamma - 1} = \frac{R(T_1 - T_2)}{\gamma - 1} = \frac{0.2078(300 - 396.12)}{1.67 - 1} = \mathbf{-29.81\,kJ/kg}$$

The $-ve$ sign indicates work done on the system.



50. (***B***) Explanation: Given data for isothermal process:

$$m = 10\,\text{kg}$$
$$T_1 = 60\,^\circ\text{C} = (60 + 273)\,\text{K} = 333\,\text{K}$$
$$p_1 = 100\,\text{kPa}$$
$$p_2 = 800\,\text{kPa}$$

Heat transfer:

$$\begin{aligned} Q &= \text{Work done} \\ &= p_1 V_1 \log_e \frac{V_2}{V_1} \\ &= mRT_1 \log_e \frac{p_1}{p_2} \\ &= 10 \times 0.287 \times 333 \log_e \frac{100}{800} = \mathbf{-1987.34\,kJ} \end{aligned}$$

51. (***D***) Explanation: Given data for adiabatic compression:

At initial state:

$$p_1 = 100\,\text{kPa}$$

$$T_1 = 20\,^\circ\mathrm{C} = (20 + 273)\,\mathrm{K} = 393\,\mathrm{K}$$

At final state:

$$p_2 = 800\,\mathrm{kPa}$$

$$T_2 = ?$$

We know the relation between temperature and pressure for adiabatic process,

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

$$\frac{T_2}{293} = \left(\frac{800}{100}\right)^{\frac{1.4-1}{1.4}} \quad \because \gamma = 1.4\,\text{for air}$$

$$\frac{T_2}{293} = (8)^{0.255} = 1.808$$

or

$$T_2 = 1.808 \times 293 = 529.74\,\mathrm{K}$$
$$= (529.74 - 273)\,^\circ\mathrm{C} = 256.74\,^\circ\mathrm{C} \approx \mathbf{257}\,^\circ C$$

55. (*A*) Explanation:

Heat transfer:

$$Q = 0 \quad \text{well} - \text{insulated wall}$$

Increase in internal energy:

$$\Delta U = \text{rate of heat dissipated by a resistor}$$

$$\Delta U = I^2R$$
$$= (10)^2 \times 23 = 2300\,\mathrm{W} = 2.3\,\mathrm{kW}$$

According to first law of thermodynamic for process:

$$Q = dU + W$$
$$0 = 2.3 + \mathrm{W}$$

or

$$W = -\mathbf{2.3\,kW}$$

58. (*A*) Explanation:

$$\text{Volume of room} = 10 \times 20 \times 3 = 600\,\mathrm{m}^3$$

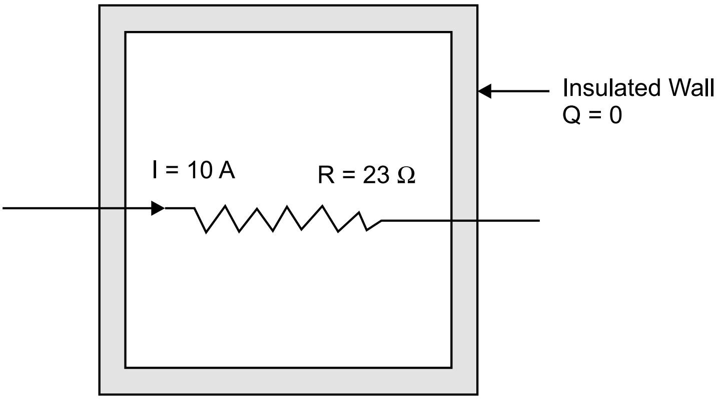
$$\text{Heat emitted by one person} = 400\,\mathrm{kJ/h} = \frac{400\,\mathrm{kJ}}{3600\,\mathrm{s}} = 0.11\,\mathrm{kJ/s} = 0.11\,\mathrm{kW}$$

$$\text{Heat emitted by 100 persons} = 100 \times 0.11 = 11\,\mathrm{kW}$$

$$\text{Heat added by light energy} = 300\,\mathrm{W} = 0.300\,\mathrm{kW}$$

$$\text{Total heat added} = 11 + 0.300 = 11.300\,\mathrm{kW}$$

$$\text{Heat added in 15 min or 900 s} = 11.300 \times 900 = 10170\,\mathrm{kJ}$$



Suppose initial temperature of room, $T = 22$ °C (for human comfort condition) i.e.,

$$T = 22\,^\circ\mathrm{C} = 22 + 273 = 295\,\mathrm{K}$$

Pressure:

$$P = 1.01325\,\mathrm{bar} = 1.01325 \times 100\,\mathrm{kPa}$$

Applying equation of state we have,

$$pV = mRT$$

or

$$m = \frac{pV}{RT} = \frac{1.01325 \times 100 \times 600}{0.287 \times 295}$$

$$m = 718.029\,\mathrm{kg}$$

Now Heat added in room in 15 min,

$$Q = 10170\,\mathrm{kJ}$$

also

$$Q = mc_p\Delta T$$

$$\therefore\ 10170 = 718.029 \times 1.005 \times \Delta T$$

$$\Delta T = \frac{10170}{718.029 \times 1.005} = \mathbf{14.0\,^\circ C}$$

60. (***C***) and **61.** (***A***) Explanation:
Efficiency of Lenoir cycle,

$$\eta = 1 - \gamma\left[\frac{r_p^{1/\gamma} - 1}{r_p - 1}\right]$$

where

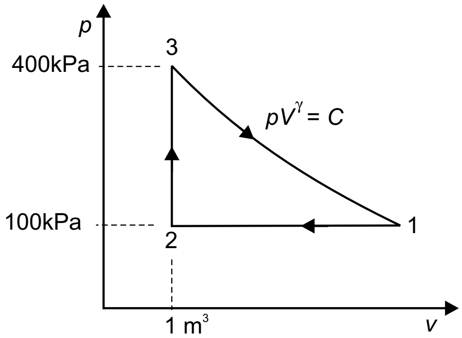
$$\gamma = 1.4$$

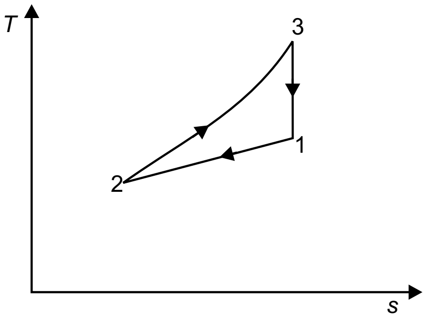
$$r_p = \frac{p_3}{p_2},\ \text{pressure ratio}$$

$$= \frac{400}{100} = 4$$

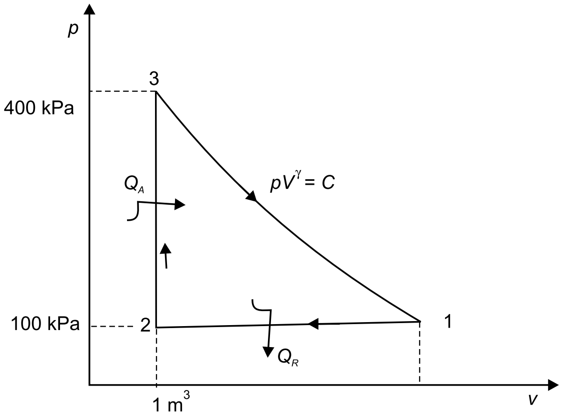
$$\eta = 1 - 1.4\left[\frac{(4)^{1/1.4} - 1}{4 - 1}\right] = 1 - 0.7895$$

$$= 0.2105 = \mathbf{21.05\%}$$





Alternatively:



For adiabatic process: 3–1:

$$p_1 V_1^\gamma = p_3 V_3^\gamma$$

$$V_1^\gamma = \frac{p_3}{p_1} V_3^\gamma$$

or

$$\begin{aligned} V_1 &= \left(\frac{p_3}{p_1}\right)^{1/\gamma} V_3 \\ &= \left(\frac{400}{100}\right)^{1/1.4} \times 1 \\ &= 2.69\,\text{m}^3 \end{aligned}$$

Process 2–3: Heat addition at constant volume:

$$Q_A = mc_v(T_3 - T_2)$$

Process 1–2: Heat rejection at constant pressure:

$$Q_R = mc_p(T_1 - T_2)$$

Work output:

$$W = Q_A - Q_R$$

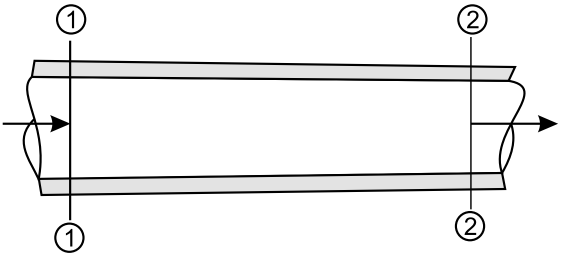
Thermal efficiency:

$$\begin{aligned} \eta_{\text{th}} &= \frac{W}{Q_A} = \frac{Q_A - Q_R}{Q_A} \\ &= 1 - \frac{Q_R}{Q_A} = 1 - \frac{mc_p(T_1 - T_2)}{mc_v(T_3 - T_2)} \\ &= 1 - \gamma\frac{m(T_1 - T_2)}{mT_3 - T_2} = 1 - \gamma\frac{mR(T_1 - T_2)}{mR(T_3 - T_2)} \\ &= 1 - \gamma\frac{(p_1V_1 - p_2V_2)}{(p_3V_3 - p_2V_2)} \\ &= 1 - \gamma\frac{p_1(V_1 - V_2)}{V_2(p_3 - p_2)} \quad \therefore\ V_2 = V_3, p_1 = p_2 \\ &= 1 - \frac{1.4 \times 100(2.69 - 1)}{1(400 - 100)} = 1 - 0.7886 \end{aligned}$$

$$= \mathbf{0.2114 = 21.14\%}$$

64. (*A*) Explanation:

Applying the steady flow energy equation between sections (1)–(2) and (2)–(2),
we get



$$h_1+\frac{V_1^2}{2}+gz_1+q=h_2+\frac{V_2^2}{2}+gz_2+w$$

where

q = 0 insulated pipe
w = 0 no shaft work
$z_2 = z_2$ for horizontal pipe
$V_1 = V_2$ Constant Cross-Sectional Area
$\therefore$ $\boldsymbol{h_1 = h_2}$

67. (*C*) Explanation:

Change in enthalpy:

$$\Delta h=-100\,\text{kJ/kg}$$

$$h_2-h_1=-100\,\text{kJ/kg}$$

or

$$h_1-h_2=100\,\text{kJ/kg}$$

Inlet velocity is negligible i.e., $V_1 = 0$

Exit velocity:

$$V_2=\sqrt{2000(h_1-h_2)}\,\text{m/s}$$

where h_1 and h_2 are in kJ/kg

$$V_2=\sqrt{2000\times 100}=\mathbf{447.21\,m/s}$$

68. (*D*) Explanation: Given data for compressor:

$$m=0.70\,\text{kg/s}$$

$$h_1=290\,\text{kJ/kg}$$

$$h_2=450\,\text{kJ/kg}$$

$$V_1=6\,\text{m/s}$$

$$V_2=2\,\text{m/s}$$

Applying the steady flow energy equation,

$$m\left[h_1+\frac{V_1^2}{2000}+\frac{gz_1}{1000}\right]+Q$$
$$=m\left[h_2+\frac{V_2^2}{2000}+\frac{gz_2}{1000}\right]+W$$

where
Q = 0, adiabatic flow,

$$\therefore\ m\left[h_1+\frac{V_1^2}{2000}\right]=m\left[h_2+\frac{V_2^2}{2000}\right]+W$$

$$0.70\left[290+\frac{(6)^2}{2000}\right]=0.7\left[450+\left(\frac{(2)^2}{2000}\right]\right(+W$$

$$203.01=315+\text{W}$$

or

$$W=-111.99\,\text{kW}\approx-112\,\text{kW}$$

Thus, the power is input to the compressor is 112 kW.

69. (*B*) Explanation: Given data:

$$V_1=20\,\text{m/s}$$

$$V_2 = 200\,\text{m/s}$$

Applying the steady flow energy equation for unit mass flow rate,

$$h_1 + \frac{V_1^2}{2000} + \frac{gz_1}{1000} + q = h_2 + \frac{V_2^2}{2000} + \frac{gz_2}{1000} + w$$

where

$$q = 0, w = 0$$

and

$$z_1 = z_2$$

$$\therefore\ h_1 + \frac{V_1^2}{2000} = h_2 + \frac{V_2^2}{2000}$$

$$\therefore\ h_1 - h_2 = \frac{V_2^2}{2000} - \frac{V_1^2}{2000}$$

$$c_p(T_1 - T_2) = \frac{V_2^2 - V_1^2}{2000}$$

where

$$c_p = 1.005\,\text{kJ/kg for air}$$

$$T_1, T_2 \text{ are in } ^\circ\text{C or K}$$

$$\therefore\ 1.005(T_1 - T_2) = \frac{(200)^2 - (20)^2}{2000} = 19.8$$

or

$$T_1 - T_2 = \frac{19.8}{1.005} = 19.70\,^\circ\text{C} \approx \mathbf{20\,^\circ C}$$

70. (*A*) Explanation: Given data:

$$h_1 = 2988\,\text{kJ/kg}$$

$$h_2 = 2888\,\text{kJ/kg}$$

$$\begin{aligned} V_2 &= \sqrt{2000(h_1 - h_2)}\text{m/s} \\ &= \sqrt{2000(2988 - 2888)} = \mathbf{447.21\,m/s} \end{aligned}$$

71. (*B*) Explanation: Given data:

$$p_1 = 10\,\text{bar}$$

$$h_1 = 3000\,\text{kJ/kg}$$

$$p_2 = 1\,\text{bar}$$

$$h_2 = 2700\,\text{kJ/kg}$$

$$W = 250\,\text{kJ/kg}$$

$$q = ?$$

The steady flow energy equation for unit mass flow rate,

$$h_1 + \frac{V_1^2}{2} + gz_1 + q = h_2 + \frac{V_2^2}{2} + gz_2 + w$$

Change in KE and PE are negligible

$$h_1 + q = h_2 + w$$

$$3000 + q = 2700 + 250$$

or

$$q = \mathbf{-50\,kJ/kg}$$

The heat transfer from the turbine casing to the surroundings is equal to 50 kJ/kg.

72. (*B*) Explanation. Given data:

$$m = 0.2\,\text{kg/s}$$

$$h_1 = 3263.88\,\text{kJ/kg}$$

$$h_2 = 3167.65\,\text{kJ/kg}$$

Exit velocity:

$$V_2 = \sqrt{2000(h_1 - h_2)}\,\text{m/s}$$

where

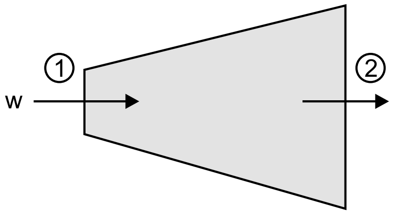
$$h_1 \text{ and } h_2 \text{ are in kJ/kg}$$

$$\begin{aligned} \therefore\ V_2 &= \sqrt{2000(3263.88 - 3167.65)} \\ &= \mathbf{438.70\,m/s} \end{aligned}$$

Note: The above expression of V_2 is only used when flow through the nozzle is adiabatic and inlet velocity is neglected.

73. (***C***) Explanation:

Applying the steady flow energy equation



$$h_1 + \frac{V_1^2}{2} + gz_1 + g = h_2 + \frac{V_2^2}{2} + gz_2 + w$$

Assumptions:

$$z_1 = z_2$$

$$q = 0, \text{adiabatic flow}$$

$$w = 0, \text{ always for nozzle}$$

$$h_1 + \frac{V_1^2}{2} = h_2 + \frac{V_2^2}{2}$$

$$c_pT_1 + \frac{V_1^2}{2} = c_pT_2 + \frac{V_2^2}{2}$$

$$1008 \times 500 + \frac{(10)^2}{2} = 1008T_2 + \frac{(180)^2}{2}$$

$$504050 = 1008T_2 + 16200$$

or

$$T_2 = 483.97\,\text{K} \approx 484\,\text{K}$$

74. (***D***) Explanation. Continuity equation for compression flow

$$\frac{A_1V_1}{v_1} = \frac{A_2V_2}{v_2}$$

$$\frac{A_1V_1}{\frac{RT_1}{p_1}} = \frac{A_2V_2}{\frac{RT_2}{p_2}} \qquad \therefore\ pv = RT \quad \text{or } v = \frac{RT}{P}$$

$$\frac{A_1V_1p_1}{T_1} = \frac{A_2V_2p_2}{T_2}$$

$$\frac{80 \times 10 \times 300}{500} = \frac{A_2 \times 180 \times 100}{484}$$

or

$$A_2 = \mathbf{12.90\,cm^2}$$

75. (***B***) Explanation:

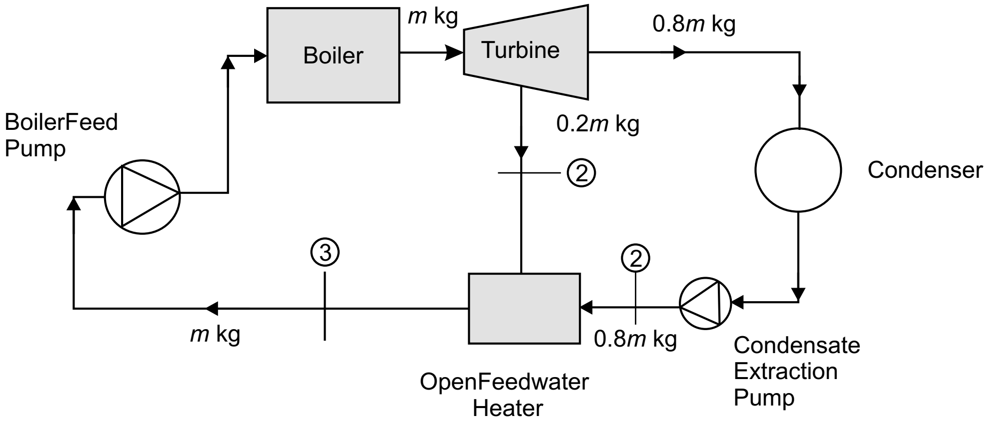
The specific work required to be supplied to the compressor:

$$w = \int_1^2 vdp$$

76. (***A***) Explanation: Given data:

$$h_1 = 2800\,\text{kJ/kg}$$

$$h_2 = 200\,\text{kJ/kg}$$



Using energy balance equation for open feedwater heater

$$m\,h_3 = 0.2\,m\,h_1 + 0.8\,m\,h_2$$

or

$$\begin{aligned} h_3 &= 0.2\,h_1 + 0.8\,\mathrm{h}_2 \\ &= 0.2 \times 2800 + 0.8 \times 200 = \mathbf{720\,kJ/kg} \end{aligned}$$

77. (***A***) Explanation:

Applying the steady flow energy equation

$$m\left(h_1 + \frac{V_2}{2} + gz_1\right) + Q = m\left(h_2 + \frac{V_2^2}{2} + gz_2\right) + W$$

where

$$Q = 0 \quad \text{for adiabatic flow}$$

$$m\left(h + \frac{V_1^2}{2} + gz_1\right) = m\left(h_2 + \frac{V_2^2}{2} + gz_2\right) + W$$

On units balancing

$$m\left(h_1 + \frac{V_1^2}{2000} + \frac{gz_1}{1000}\right) = m\left(h_2 + \frac{V_2^2}{2000} + \frac{gz}{1000}\right) + W$$

$$\begin{aligned} &20\left(3200 + \frac{(160)^2}{2000} + \frac{9.81 \times 10}{1000}\right) \\ &= 20\left(2600 + \frac{(100)^2}{2000} + \frac{9.81 \times 6}{1000}\right) + W \\ &64257.96 = 52101.17 + W \end{aligned}$$

or

$$\begin{aligned} W &= 12156.79\,\mathrm{kW} = 12.1568\,\mathrm{MW} \\ &\approx 12.157\,\mathrm{MW} \end{aligned}$$

78. (***C***) Explanation:

Specific work supplied to the pump,

$$w_p = v(p_1 - p_2)\,\mathrm{kJ/kg}$$

when v is in m^3/kg

p_1 & p_2 are in kPa

$$\therefore\ w_p = \frac{p_1 - p_2}{\rho} = \frac{3000 - 70}{1000} = 2.93\,\mathrm{kJ/kg}$$

79. (***C***) Explanation: The balloon ruptures in an evacuated and insulated room, the gas fills up the entire room. This case is related to free expansion.

That is.,

$$\delta Q = 0,\ \delta W = 0$$

According to firist law of the thermodynamics,

$$\begin{aligned} \delta Q &= dU = \delta W \\ 0 &= dU + 0 \\ dU &= 0 \\ U_2 - U_1 &= 0 \\ U_2 &= U_1 \\ mc_vT_2 &= mc_vT_1 \end{aligned}$$

or

$$\begin{aligned} T_2 &= T_1 \\ mc_pT_2 &= mc_pT_1 \end{aligned}$$

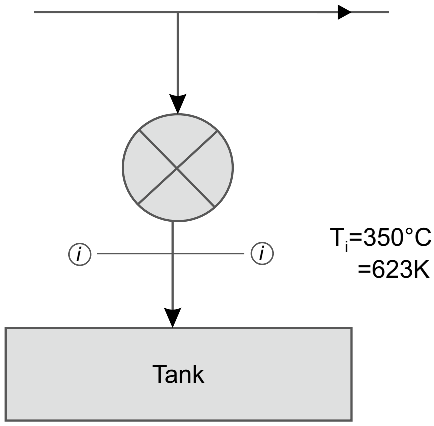
or

$$H_2 = H_1$$

For free expansion of an ideal gas, both internal energy and enthalpy of the gas remain constant.

80. (***A***) Explanation: For unsteady flow process, the unsteady flow energy equation,

$$\begin{aligned} Q = m_2\left[u_2 + \frac{V_2^2}{2} + gz_2\right] - m_1\left[u_2 + \frac{V_1^2}{2} + gz_2\right] \\ + m_e\left[h_e + \frac{V_c^2}{2} + gz_e\right] - m_l\left[h_i + \frac{V_i^2}{2} + gz_i\right] + W \end{aligned} \tag{1}$$



For given problem,

V_1, V_2 are neglected
$z_1 = z_2$
m_1 = 0, evacuated tank
m_e =0 no exit
z_i =0
Q = 0, W = 0

Above Eq. (1) becomes

$$0 = m_2u_2 - m_ih_i$$

or

$$m_2u_2 = m_ih_i$$

$$u_2 = h_i \quad \therefore\ m_1 = m_2$$

$$c_vT_2 = c_pT_i$$

$$T_2 = \frac{c_p}{c_v}T_i = \gamma T_i$$
$$= 1.4 \times 623 = 872.3\,\text{K} = 599.2\,°\text{C}$$

The final temperature T_2 = 599.2 °C inside the tank is greater than 350 °C.

82. (***B***) Explanation:

$$h_1 = 3000\,\text{kJ/kg}, h_2 = 2700\,\text{kJ/kg}$$
$$w = 250\,\text{kJ/kg}$$

Applying the SFEE for unit mass flow rate,

$$h_1 + \frac{V_1^2}{2} + gz_1 + q = h_2 + \frac{V_2^2}{2} + gz_2 + w$$

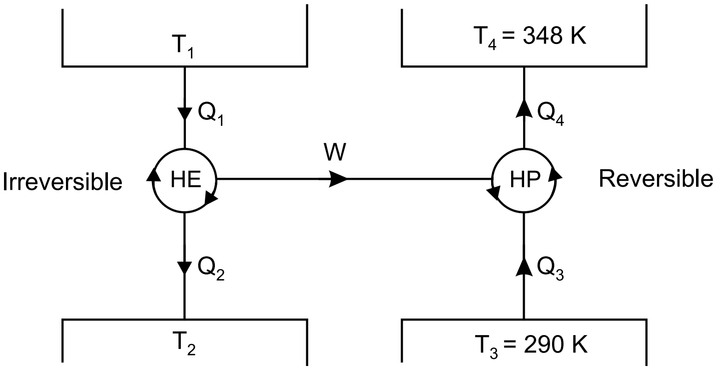
Change in KE and PE are negligible

$$h_1 + q = h_2 + w$$
$$3000 + q = 2700 + 250$$
$$q = -50\,\text{kJ/kg}$$

The $-ve$ sign indicates, the heat transfer from the turbine casing to the surroundings.

91. (***C***) Explanation: Given data:

$$Q_1 = 100\,\text{kW}$$
$$Q_2 = 50\,\text{kW}$$
$$T_3 = 290\,\text{K}$$
$$T_4 = 348\,\text{K}$$



Heat engine work output:

$$W = Q_1 - Q_2$$
$$W = 100 - 50 = 50\,\text{kW}$$
$$= \text{Pump input work}$$
$$(\text{COP})_{\text{HP}} = \frac{Q_4}{W}$$

also

$$(\text{COP})_{\text{HP}} = \frac{T_4}{T_4 - T_3}$$
$$\therefore\ \frac{Q_4}{W} = \frac{T_4}{T_4 - T_3}$$
$$\frac{Q}{50} = \frac{348}{348 - 290} = \frac{348}{58}$$

or

$$Q = \mathbf{300\,kW}$$

92. (***A***) Explanation:

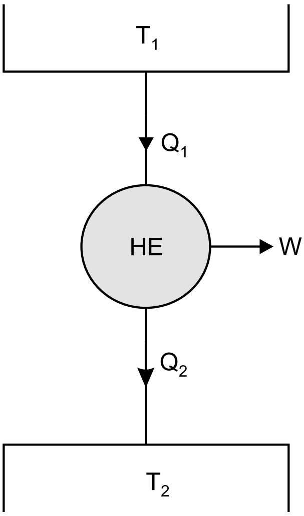
$$T_1 = 27\,^\circ\text{C} = (27 + 273)\,\text{K} = 300\,\text{K}$$
$$T_2 = 10\,^\circ\text{C} = (10 + 1273)\,\text{K} = 283\,\text{K}$$
$$W = 10\,\text{kW}$$
$$Q_2 = 9900\,\text{kJ/min} = \frac{9900}{60}\,\text{kJ/s} = 165\,\text{kW}$$
$$Q_1 = W + Q_2$$
$$= 10 + 165$$
$$= 175\,\text{kW}$$

Efficiency:

$$\eta = \frac{W}{Q_1} = \frac{10}{175}$$
$$= 0.0571$$
$$= 5.71\%$$

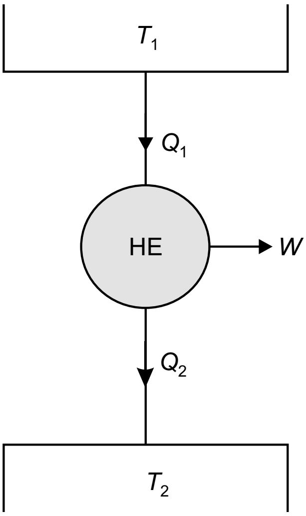
Carnot efficiency:

$$\eta_{\text{Carnot}} = 1 - \frac{T_2}{T_1}$$
$$= 1 - \frac{283}{300} = 1 - 0.9433$$
$$= 0.0567 = \mathbf{5.67\%}$$



Since, the actual efficiency of heat engine is more than the Carnot efficiencey, so the heat engine is impossible

93. (***B***) Explanation:



$$T_2 = 20\,^{\circ}\mathrm{C} = (20 + 273)\mathrm{K}$$
$$= 293\,\mathrm{K}$$

$$T_1 = 200\,^{\circ}\mathrm{C} = (200 + 273)\mathrm{K}$$
$$= 473\,\mathrm{K}$$

$$\mathrm{W} = 10\,\mathrm{kW}$$

$$\eta_{\mathrm{Carnot}} = 1 - \frac{T_2}{T_1}$$
$$= 1 - \frac{293}{473}$$
$$= 1 - 0.6194 = 0.3806$$

also

$$\eta_{\mathrm{Carnot}} = \frac{W}{Q_1}$$
$$\therefore\ 0.3806 = \frac{10}{Q}$$

or

$$Q_1 = 1 - \frac{T_2}{T_1} = 26.27\,\mathrm{kW}$$

also

$$Q_1 = W + Q_2$$
$$\therefore\ 26.27 = 10 + Q_2$$

or

$$Q_2 = 26.27 - 10 = \mathbf{25.27\,kWorkJ/s}$$

94. (***D***) Explanation:

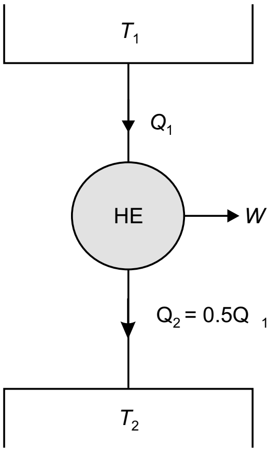


Fig. Heatengine

$$T_1 = 377\,^{\circ}\mathrm{C}$$
$$= (377 + 273)\,\mathrm{K} = 650\,\mathrm{K}$$

$$T_2 = 27\,^{\circ}\mathrm{C} = (27 + 273)\,\mathrm{K}$$
$$= 300\,\mathrm{K}$$

Carnot efficiency:

$$\eta_{\mathrm{Carnot}} = 1 - \frac{T_2}{T_1} = 1 - \frac{300}{650}$$
$$= 1 - 0.4615$$
$$= 0.5385 = 53.85\%$$

Actual efficiency:

$$\eta = 1 - \frac{Q_2}{Q_1}$$
$$= 1 - \frac{0.5Q_1}{Q_1}$$
$$= 1 - 0.5 = 0.5 = \mathbf{50\%}$$

Since, the actual efficiency is less than the Carnot efficiency, so this engine has possible cycle.

95. (***C***) Explanation:

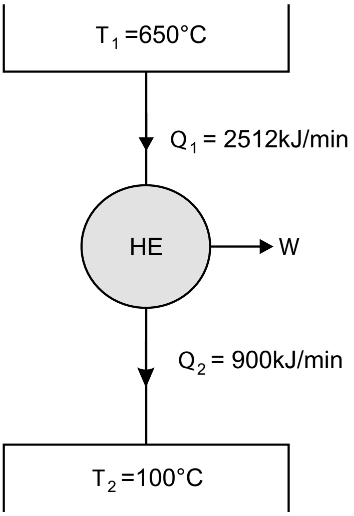


Fig. Heatengine

$$Q_1 = 2512\,\text{kJ/min}$$

$$\begin{aligned} T_1 &= 650\,^\circ\text{C} \\ &= (273+650)\,\text{K} = 923\,\text{K} \end{aligned}$$

$$Q_2 = 900\,\text{kJ/min}$$

$$\begin{aligned} T_2 &= 100\,^\circ\text{C} \\ &= (273+100)\,\text{K} = 373\,\text{K} \end{aligned}$$

Carnot efficiency:

$$\begin{aligned} \eta_{\text{Carnot}} &= 1 - \frac{T_2}{T_1} \\ &= 1 - \frac{373}{923} \\ &= 1 - 0.4041 = 0.5959 = 59.59\% \end{aligned}$$

Actual efficiency:

$$\begin{aligned} \eta &= 1 - \frac{Q_2}{Q_1} = 1 - \frac{373}{923} \\ &= 1 - 0.3582 = \mathbf{0.6418\%} \end{aligned}$$

Since, the actual efficiency is more than than Carnot efficiency, so this type of heat engine is impossible.

96. (*A*) Explanation:

$$\frac{\text{Actual COP}}{\text{Reversible COP}} = 0.8$$

Cooling capacity:

$$Q_2 = 2.4\,\text{kW}$$

$$T_1 = 51\,^\circ\text{C} = 324\,\text{K}$$

$$T_2 = -30\,^\circ\text{C} = -30 + 273 = 273\,\text{K}$$

Now reversible

$$\text{COP} = \frac{T_2}{T_1 - T_2} = \frac{243}{324 - 243} = \frac{243}{81} = 3$$

as

$$\frac{\text{Actual COP}}{\text{Reversible COP}} = 0.8$$

$$\begin{aligned} \text{Actual COP} &= \text{Reversible COP} \times 0.8 \\ &= 3 \times 0.8 = 2.4 \end{aligned}$$

also

$$\text{Actual COP} = \frac{Q_2}{W}$$

$$2.4 = \frac{Q_2}{W}$$

$$W = \frac{Q_2}{2.4} = \frac{2.4}{2.4} = \mathbf{1\,kW}$$

97. (*B*) Explanation:

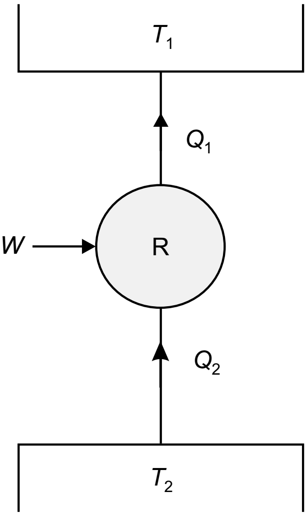


Fig. Carnotcyclerefrigerator

$$T_1 = 300\,\text{K}, T_2 = 250\,\text{K}$$

$$\begin{aligned} (\text{COP})_\text{R} &= \frac{T_2}{T_1 - T_2} \\ &= \frac{250}{300 - 250} \\ &= \frac{250}{50} = \mathbf{5} \end{aligned}$$

98. (***D***) Explanation:

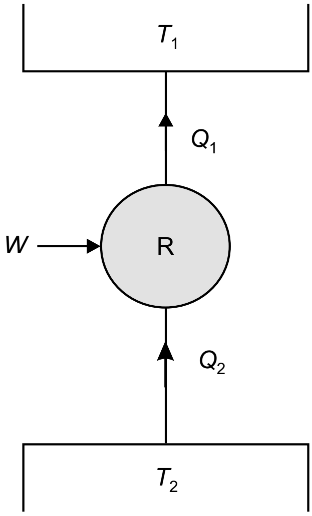


Fig. Refrigerator

$$Q_1 = 120\,\text{kW}$$
$$W = 30\,\text{kW}$$
$$Q_1 = W + Q_2$$
$$120 = 30 + Q_2$$

or

$$Q_2 = 120 - 30 = 90\,\text{kW}$$
$$\text{COP} = \frac{Q_2}{W} = \frac{90}{30} = \mathbf{3}$$

99. (***C***) Explanation:

$$\eta_{\text{Carnot}} = 30\% = 0.30$$
$$(\text{COP})_{\text{HP}} = \frac{1}{\eta_{\text{Carnot}}} = \frac{1}{0.3} = \mathbf{3.33}$$

100. (***C***) Explanation:

Let

$$Q_1 = Q$$

$$Q_2 = 50\%\,Q = \frac{50}{100}Q = 0.5Q$$

Now

$$\eta = \frac{Q - Q_2}{Q_2} = \frac{Q - 0.5Q}{Q} = \frac{0.5Q}{Q} = 0.5$$

As

$$(\text{COP})_{\text{HP}} = \frac{1}{\eta}$$
$$(\text{COP})_{\text{HP}} = \frac{1}{0.5} = \frac{10}{5} = \mathbf{2}$$

101. (***B***) Explanation:

Let $Q_1 = Q_2$, $T_1 = ?$, $T_2 = 27\ ^\circ\text{C} = 300\ \text{K}$

$$Q_2 = \frac{40}{100}Q = 0.4\,Q$$

Now

$$\frac{Q_2}{Q} = \frac{T_2}{T_1}$$
$$\frac{0.4\,Q}{Q} = \frac{300}{T_1}$$
$$0.4\,T_1 = 300$$
$$T_1 = \frac{300}{0.4} = 750\,\text{K}$$
$$T_1 = 750 - 273 = 477\,^\circ\text{C}$$

102. (***C***) Explanation:

$$T_1 = 327\,^\circ\text{C} = 327 + 273 = 600\,\text{K}$$
$$T_2 = 27\,^\circ\text{C} = 27 + 273 = 300\,\text{K}$$

Now

$$\eta_{\text{Carnot}} = 1 - \frac{T_2}{T_1} = 1 - \frac{300}{600} = 1 - \frac{1}{2} = 0.5$$

Let

$$Q_1 = Q$$

$$Q_2 = 45\%Q = \frac{45}{100}Q = 0.45\,Q$$

Now

$$\eta_{HE} = \frac{Q - Q_2}{Q_1} = \frac{Q - 0.45Q}{Q} = \frac{0.55Q}{Q} = 0.55$$

Since

$$\eta_{HE} > \eta_{Carnot}$$

So inventor's heat engine is an impossible engine

103. (***C***) Explanation:

Given data:

$$\begin{aligned} T_1 &= 180\,^\circ\mathrm{C} \\ &= (180 + 273)\mathrm{K} \\ &= 453\,\mathrm{K} \end{aligned}$$

$$\begin{aligned} T_3 &= 20\,^\circ\mathrm{C} \\ &= (20 + 273)\mathrm{K} \\ &= 293\,\mathrm{K} \end{aligned}$$

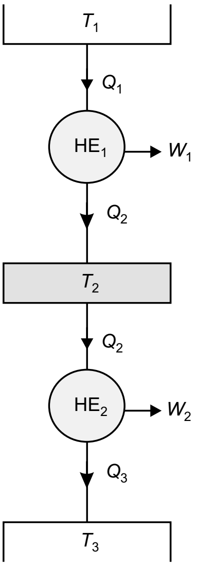


Fig.

For same work output,

$$\begin{aligned} T_2 &= \frac{T_1 + T_3}{2} \\ &= \frac{453 + 293}{2} = 373\,\mathrm{K} \\ &= \mathbf{100\,^\circ C} \end{aligned}$$

105. (***D***) Explanation:

Given data:

$$Q_1 = 250\,\mathrm{kJ/s}$$

$$\begin{aligned} T_1 &= 227\,^\circ\mathrm{C} \\ &= 227 + 273 = 500\,\mathrm{K} \end{aligned}$$

$$T_2 = 27\,^\circ\mathrm{C} = 27 + 273 = 300\,\mathrm{K}$$

$$Q_2 = ?$$

Since cycle is reversible so

$$\frac{Q_2}{Q_1} = \frac{T_2}{T_1}$$

$$\frac{Q_2}{250} = \frac{300}{500}$$

or

$$Q_2 = \frac{300}{500} \times 250 = \mathbf{150\,kJ/s}$$

109. (***A***) Explanation:

$$T_1 = 327\,^\circ\mathrm{C} = (327 + 273)\mathrm{K} = 600\,\mathrm{K}$$

and

$$T_2 = 27\,^\circ\mathrm{C} = (27 + 273)\mathrm{K} = 300\,\mathrm{K}$$

Maximum efficiency:

$$\begin{aligned} \eta_{max} &= 1 - \frac{T_2}{T_1} \\ &= 1 - \frac{300}{600} = \frac{300}{600} = \mathbf{\frac{1}{2}} \end{aligned}$$

110. (*B*) Explanation:

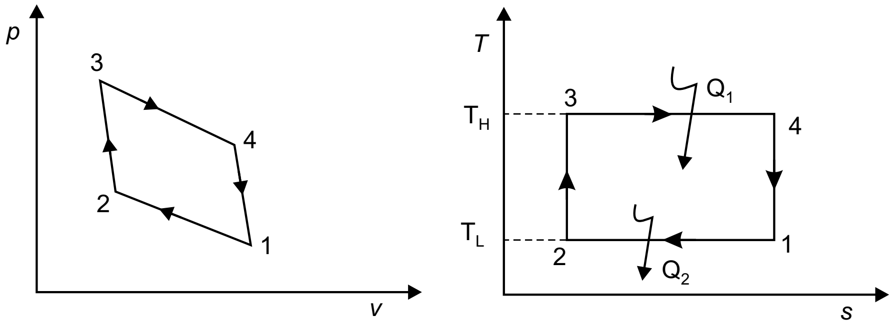


Fig. CarnotCycle

Heat supplied:

$$Q_1 = 53\,\mathrm{kJ}$$

Adiabalic expansion ratio:

$$\frac{V_1}{V_4} = 16$$

Heat rejected:

$$Q_2 = ?$$

For adiabatic expansion process 4–1,

$$\frac{T_L}{T_H} = \left(\frac{V_1}{V_4}\right)^{1-\gamma}$$

$$\frac{T_L}{T_H} = (16)^{1-14} = (16)^{-0.4} = \frac{1}{16^{0.4}} = \frac{1}{3.031}$$

$$\eta_{\text{Carnot}} = 1 - \frac{T_L}{T_H} = 1 - \frac{Q_2}{Q_i}$$

or

$$\frac{T_L}{T_H} = \frac{Q_2}{Q_1}, Q_2 = \frac{T_L}{T_H} \times Q_1 = \frac{1}{3.031} \times 53$$
$$= \mathbf{17.48\,kJ}$$

116. (*A*) If an irreversible process occurs in a closed system, the entropy of the system must increases.

126. (*B*) Explanation:

We know that $\Delta S = \frac{\delta Q}{T}$.

Integrating between states 1 and 2 gives

$$\int_1^2 \Delta S = \int_1^2 \frac{\delta Q}{T}$$

$$S_2 - S_1 = \frac{1}{T}\int_1^2 \delta Q \quad \because T = C$$
$$= \frac{1}{T} Q_{1-2}$$
$$= \frac{mRT}{T} \ln\frac{V_2}{V_1} = mR \ln\frac{V_2}{V_1}$$
$$= mR \ln\frac{p_1}{p_2}$$

where

$$\frac{p_1}{p_2} = \frac{V_2}{V_1} \text{for isothermal process}$$

or

We know that the entropy change in terms of temperature and pressure ratios,

$$S_2 - S_1 = mc_p \ln\frac{T_2}{T_1} - mR \ln\frac{p_2}{p_1}$$

where $T_1 = T_2$ for isothermal process.

$$S_2 - S_1 = -mR\ \ln\frac{p_2}{p_1}$$
$$= mR\ \ln\frac{p_1}{p_2}$$

127. (***C***) Explanation:

$$c_p = 1.047\,\text{kJ/kgK}$$
$$T_1 = 1260\,\text{K}, T_2 = 480\,\text{K}$$
$$T_0 = 295\,\text{K}$$

Available energy:

$$AE = c_p(T_1 - T_2) - T_0 c_p \log_e\frac{T_1}{T_2}$$
$$= 1.046(1260 - 480) - 295 \times 1.046\ \log_e\frac{1260}{480}$$
$$= 815.88 - 297.79$$
$$= \mathbf{518.1\,kJ/kgK}$$

128. (***B***) Explanation:

Increase in etntropy:

$$\Delta S = -mR\log_e\frac{p_2}{p_1} \quad \text{for throttling process}$$

where

$$\Delta S = 0.06\,\text{kW/K}$$
$$m = 2\,\text{kg/s}$$

and

$$R = 0.287\,\text{kJ/kgK}$$
$$\therefore\ 0.06 = -2 \times 0.287\log_e\frac{p_2}{p_1}$$

or

$$\frac{p_2}{p_1} = \mathbf{0.9}$$

129. (***B***) Explanation. Given data:

$$p_1 = 100\,\text{kPa}$$
$$T_1 = 17\,^\circ\text{C} = (17 + 273)\text{K} = 290\,\text{K}$$
$$p_2 = 600\,\text{kPa}$$
$$T_2 = 57\,^\circ\text{C} = (57 + 273)\text{K} = 330\,\text{K}$$
$$R = 0.287\,\text{kJ/kgK}$$
$$c_v = 0.718\,\text{kJ/kgK}$$

We know that the entropy change in terms of temperature and pressure ratio,

$$s_2 - s_1 = c_p\ \log_e\frac{T_2}{T_1} - R\ \log_e\frac{p_2}{p_1}$$
for umits mass

where

$$c_p = R + c_v \quad \because R = c_p - c_v$$
$$= 0.287 + 0.718 = 1.005\,\text{kJ/kgK}$$

$$\therefore\ s_2 - s_1 = 1.005\log_e\frac{330}{290} - 0.287\log_e\frac{600}{100}$$
$$= 0.1298 - 0.5142$$
$$= \mathbf{-0.3844\,kJ/kgK}$$
$$\approx \mathbf{-0.385\,kJ/kgK}$$

130. (***D***) Explanation:

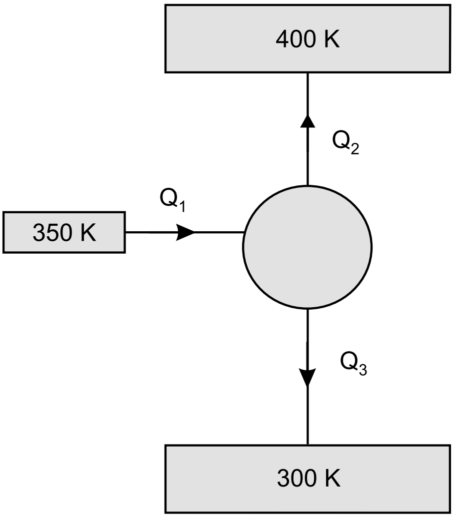


Fig.

Let Q_2 heat in kJ can be transfer to 400 K and Q_3 heat rejected to the sink at 300 K

$$T_1 = 350\,\text{K}$$
$$Q_1 = 100\,\text{kJ}$$
$$T_2 = 400\,\text{K}$$
$$T_3 = 300\,\text{K}$$

Heat supplied:

$$Q_1 = Q_2 + Q_3$$
$$100 = Q_2 + Q_3$$

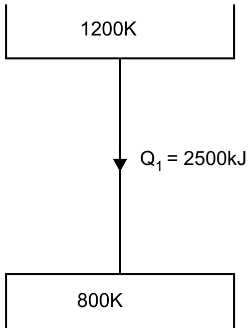
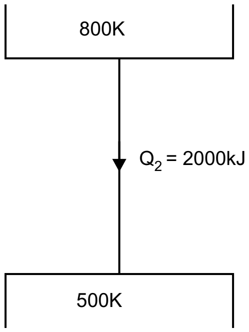
or

$$Q_3 = 100 - Q_2$$

Applying the Clausious equality,

$$\frac{Q_1}{T_1} - \frac{Q_2}{400} - \frac{Q_3}{300} = 0$$
$$\frac{100}{350} - \frac{Q_2}{400} - \frac{(100 - Q_2)}{300} = 0$$
$$\frac{100}{350} - \frac{Q_2}{400} - \frac{1}{3} + \frac{Q_2}{300} = 0$$
$$\frac{1}{3.5} - \frac{1}{3} - Q_2\left[\frac{1}{400} - \frac{1}{300}\right] = 0$$
$$0.2857 - 0.3333 - Q_2\left[\frac{300 - 400}{120000}\right] = 0$$
$$-0.0476 - Q_2\left(\frac{-100}{120000}\right) = 0$$
$$Q_2\frac{100}{120000} = 0.0476$$
$$Q_2 = \mathbf{57.12\,kJ}$$

132. **(*B*)** Explanation:

Process I	Process II
	
Entropy change of Process I:	**Entropy change of Process II:**
$\Delta S_1 = \frac{-Q_1}{T_1} + \frac{Q_1}{T_2}$ $= \frac{-2500}{1200} + \frac{2500}{800}$ $= -2.08 + 3.12$ $= 1.04\,\text{kJ/K}$	$\Delta S_2 = \frac{-Q_2}{T_1} + \frac{Q_2}{T_2}$ $= \frac{-2000}{800} + \frac{2000}{500}$ $= -2.5 + 4$ $= 1.5\,\text{kJ/K}$

Thus, $\Delta S_2 > \Delta S_1$, Process II is more irreversible than Process -I

133. (*B*) Explanation:

$$p_p = 150\,\text{kPa} \quad p_Q = ?$$

$$T_P = 30\,\text{K} \quad T_Q = 300\,\text{K}$$

$$c_p = 1.005\,\text{kJ/kgK}, c_v = 0.718\,\text{kJ/kgK}$$

$$\gamma = \frac{c_p}{c_v} = \frac{1.005}{0.718} = 1.4$$

For adiabatic process *P–Q*,

$$\frac{T_Q}{T_P} = \left(\frac{p_Q}{p_p}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{300}{350} = \left(\frac{p_Q}{150}\right)^{\frac{1.4-1}{1.4}} = \left(\frac{p_Q}{150}\right)^{0.255}$$

or

$$\frac{p_Q}{150} = \left(\frac{300}{350}\right)^{3.5}$$

$$\frac{p_Q}{150} = 0.583$$

or

$$p_Q = \mathbf{87.45\,kPa}$$

134. (*B*) Explanation:

If

$$p_Q = 50\,\text{kPa}$$

Change in entropy:

$$\begin{aligned} s_Q - s_p &= c_p \log_\varepsilon \frac{T_Q}{T_p} - R \log_e \frac{p_Q}{p_P} \\ &= 1.005 \log_e \frac{300}{350} - 0.287 \log_e \frac{50}{150} \\ &= 1.005(-0.1541) - 0.287(-1.098) \\ &= -0.1548 + 0.3151 = \mathbf{0.160\,kJ/kgK} \end{aligned}$$

135. (*B*) Explanation:

If 2 mol of O_2 are mixed adiabatically with anther 2 mol of O_2 at the same temperature and pressure, the final temperature and pressure of the mixture become same as the initial state, then the change of entropy due to mixture is zero.

136. (*B*) Explanation:

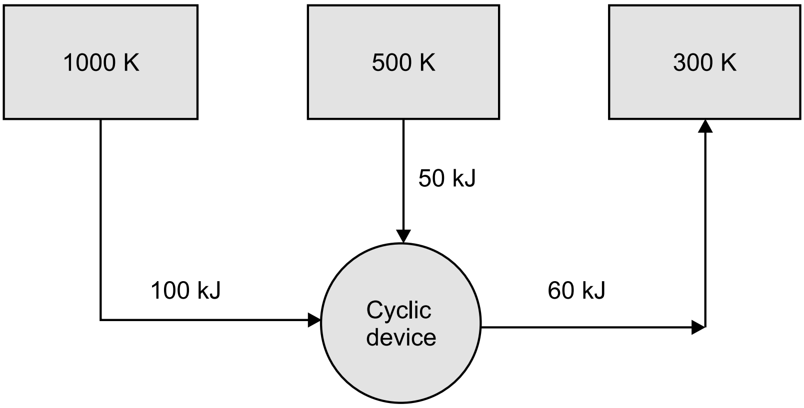
According to Clausious's inequality,

$$\oint\left(\frac{\delta Q}{T} \le 0\right.$$

$$\frac{100}{1000} + \frac{50}{500} - \frac{60}{300} \le 0$$

$$0.1 + 0.1 - 0.2 \le 0$$

$$0 = 0$$



Hence, the cyclic device can be a reversible heat engine.

138. **(*A*)** Explanation:

Given data:

$$m_{\text{ice}} = 10\,\text{kg} \quad m_w = 20\,\text{kg}$$
$$T_1 = 0\,^\circ\text{C} \quad T_{w1} = 20\,^\circ\text{C}$$

Latent heat:

$$L = 340\,\text{kJ/kg}$$

Now to check the final equilibrium state of the system.
The amount of heat required to change 10 kg of ice at 0 °C to 10 kg of water at 0°C,

$$= mL$$
$$= 10 \times 340 = 3400\,\text{kJ}$$

The amount of heat loss of change 20 kg of water at 20 °C to 0°C,

$$= m_w c_{pw}(T_w - 0)$$
$$= 20 \times 4.18(20 - 0) = 1672\,\text{kJ}$$

Which of less than the amount of heat required to change 10 kg of ice at 0 °C to 10 kg of water at 0 °C. It means that some amount of ice is melted to gain the net heat of 20 kg of water by changing the temperature from 20 to 0 °C.

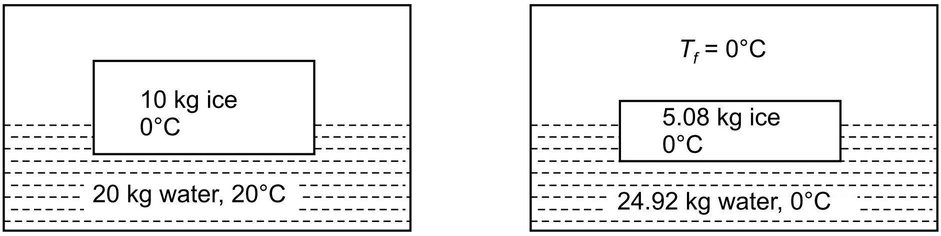
Let m = mass of ice is melted at 0 °C.

Applying the energy balance equation heat gained by ice when ice when m kg of ice of melted at 0 °C = heat lost of 20 kg water by changing the temperature from 20 to 0 °C

$$mL = m_w c_{pw}(T_w - 0)$$
$$m \times 340 = 1672$$

or

$$m = 4.92\,\text{kg}$$



Initial state Final State

Fig.

Entropy change of ice block,

$$(\Delta S)_{\text{ice}} = \frac{mL}{T_1}$$

where

$$T_1 = 273\,\text{K}$$

$$\therefore\ (\Delta S)_{\text{ice}} = \frac{4.92 \times 340}{273} = 6.12\,\text{kJ/K}$$

Entrapy change of 20 kg water,

$$(\Delta S)_w = m_w c_{pw} \log_k \frac{T_f}{T_w}$$

where

$$T_f = 273\,\text{K}$$

$$T_w = 20\,^\circ\text{C} = (20 + 273)\text{K} = 293\,\text{K}$$

$$\therefore\ (\Delta S)_w = 20 \times 4.18 \log_e \frac{273}{293} = -5.91\,\text{kJ/K}$$

Total entropy change:

$$\begin{aligned}\Delta S &= (\Delta S)_{\text{ice}} + (\Delta S)_w \\ &= 6.12 - 5.91 = 0.21\,\text{kJ/K}\end{aligned}$$

140. (***B***) Explanation:

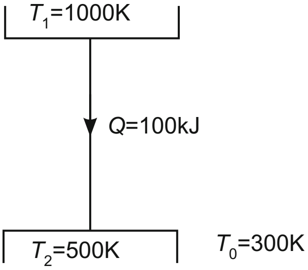


Fig.

$$\begin{aligned}\Delta S_{\text{uni}} &= \frac{100}{1000} + \frac{100}{500} \\ &= -0.1 + 0.2 = 0.1\,\text{kJ/K}\end{aligned}$$

Loss of availability:

$$\begin{aligned}I &= T_0 \Delta S_{\text{uni}} \\ &= 300 \times 0.1 = 30\,\text{kJ}\end{aligned}$$

152. (***B***) Explanation:

$$V_2 = \sqrt{2\Delta\, h}\,\text{m/s}$$

where Δh is in J/kg

$$\therefore\ V_2 = \sqrt{2 \times 200} = \sqrt{400} = 20\,\text{m/s}$$

157. (***A***) Explanation:

Volume of rigid tank:

$$V = 200\,\text{m}^3$$

Let total mass in tank $= m$ kg

Mass of liquid:

$$m_f = 25\%m = 0.25\,\text{m}$$

Mass of vapour:

$$m_g = 75\%m = 0.75\,\text{m}$$

Total volume:

$$V = V_f + V_g$$

where

$$V_f = m_f v_f$$

and

$$V_g = m_g v_g$$

$$\therefore\ V = m_f v_f + m_g v_g$$

$$V = 0.25\, m v_f + 0.75\, m v_g$$

$$V = \left(0.25 v_f + 0.75 v_g\right) m$$

or

$$m = \frac{V}{0.25v_f + 0.75v_g}$$
$$= \frac{300}{0.25 \times 0.001061 + 0.75 \times 0.886} = \mathbf{451.28\,kg}$$

158. (***B***) Explanation:

Time of vaporization:

$$t = 10\,\text{min}$$

Mass of water vaporisation:

$$M = 200\,\text{g} = 0.2\,\text{kg}$$

Rate of water evaporated:

$$m = \frac{M}{t} = \frac{0.2}{10}\,\text{kg/min}$$

Latent heat of vaporization:

$$h_{fg} = 2257\,\text{kJ/kg}$$

$$\text{Rate of heat transfer to the water} = mh_{fg}$$
$$= \frac{0.2}{10} \times 2257 = \mathbf{45.14\,kJ/min}$$

162. (***D***) Explanation: Given data:

$$p_3 = 150\,\text{bar} = p_2$$
$$T_3 = 45\,^\circ\text{C}$$

At saturated point 2,

$$p_2 = 150\,\text{bar}$$
$$T_2 = T_{\text{sat}} = 342.24\,^\circ\text{C}$$

At saturated point 1,

$$p_1 = 0.09594\,\text{bar}$$
$$T_1 = T_3 = 45\,^\circ\text{C}$$

As $T_3 < T_2$, the given condition of water is subcooled liquid.

and $T_3 = T_1 = 45\,^\circ\text{C}$.

The specific enthalpy at point 3 ≈ the specific enthaply at saturated liquid point 1

$$h_3 = h_1 = 188.45\,\text{kJ/kg}$$

163. (***D***), **(164)** (***B***)

Explanation:

Specific enthalpy:

$$h = u + pv$$

i.e., $h > u$ and the value of entropy is less than u.

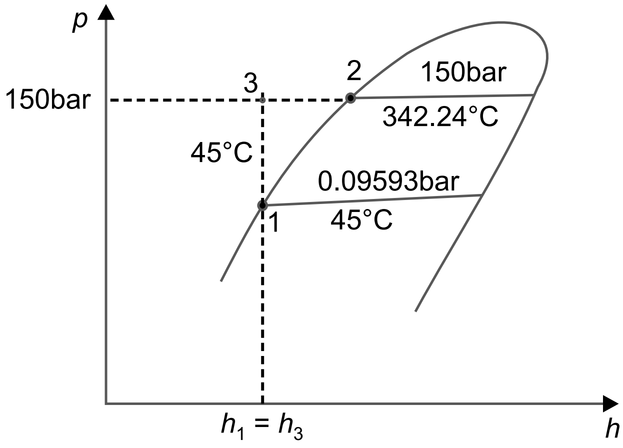
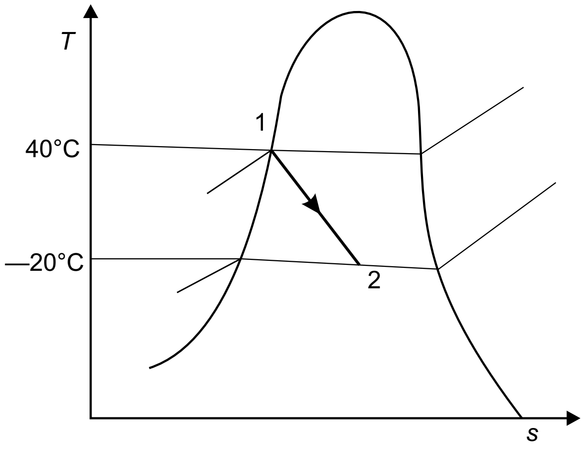


Fig.



For throttling process,

$$\text{Enthalpy at state 1} = \text{Enthalpy at state 2}$$

$$h_1 = h_2$$

where

$$h_1 = h_f = 371.43\,\text{kJ/kg} \quad \text{at } 40\,°\text{C}$$

$$h_2 = h_f + x_2\left(h_g - h\right) \quad \text{at } 20\,°\text{C}$$

$$h_2 = 89.05 + x_2(1418 - 89.05)$$

$$h_2 = 89.05 + 1328.95\,x_2$$

also

$$h_2 = h_1$$

$$\therefore\ 89.05 + 1328.95x_2 = 371.43$$

or

$$\mathbf{x_2 = 0.212}$$

165. (*B*) Explanation:

Critical specific volume of water:

$$v_c = 0.003155\,\text{m}^3/\text{kg}$$

Volume of steel tank:

$$V = 0.025\,\text{m}^3$$

Process of mixture of water and steam:

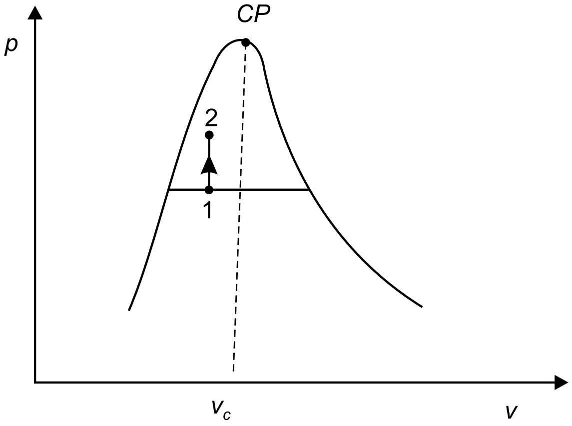
$$p = 0.1\,\text{MPa} = 1\,\text{bar}$$

Mass of mixture of water and steam:

$$m = 10\,\text{kg}$$

Specific volume of mixture:

$$v = \frac{V}{m} = \frac{0.025}{10} = 0.0025\,\text{m}^3/\text{kg}$$



As $v < v_c$, the condition of steam lies near to saturated liquid line and the liquid level inside the tank will rise with heating.

166. (*A*), 167. (*B*) and 168 (*C*)

Explanation:

Mass of liquid:

$$m_f = 1\,\text{kg}$$

Mass of vapor:

$$m_g = 0.03\,\text{kg}$$

Total mass in system

$$\begin{aligned} m &= m_f + m_g \\ &= 1 + 0.03 = 1.03\,\text{kg} \end{aligned}$$

Dryness fraction:

$$x = \frac{m_8}{m} = \frac{0.03}{1.03} = 0.029$$

At

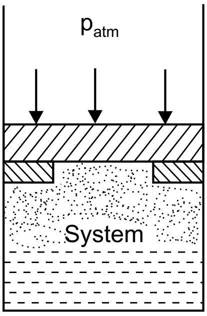
$$p_1 = 100\,\text{kPa}$$

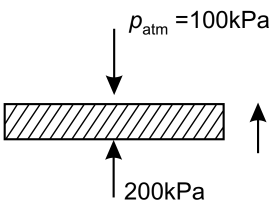
$$v_f = 0.001\,\text{m}^3/\text{kg},\ m_g = 0.1\,\text{m}^3/\text{kg}$$

Specific volume:

$$\begin{aligned} v_1 &= v_f + x\left(v_g - v_f\right) \\ &= 0.001 + 0.029(0.1 - 0.001) = 0.003\,\text{m}^3/\text{kg} \end{aligned}$$

As pressure increases from 100 to 200 kPa at constant temperature (quasi-static), volume increases by 50%





i.e.,

$$\begin{aligned} v_2 &= 1.5\, v_1 \\ &= 1.5 \times 0.038 = 0.0057\,\mathrm{m^3/kg} \end{aligned}$$

As $v_2 > v_{g,\,200\,°\mathrm{C}}$, the steam is superheated vapor in the system.

$$\begin{aligned} \text{Work done} &= m(p_2 - p_{\mathrm{atm}})(v_2 - v_1) \\ &= 1.03(200 - 100)(0.0057 - 0.0038) \\ &= 0.195\,\mathrm{kJ} \approx 0.2\,\mathrm{kJ} \end{aligned}$$

Heat transfer:

$$Q = 1\,\mathrm{kJ} = 1000\,\mathrm{J}$$

Entropy of the surrounding (i.e., reservoir)

$$(\Delta S)_{\mathrm{surr}} = \frac{-1000}{673} = -1.485\,\mathrm{J/K}$$

Entropy of the system increases:

$$(\Delta S)_{\mathrm{sys}} = 10\,\mathrm{J/K}$$

Net entropy generation:

$$\begin{aligned} (\Delta S)_{\mathrm{uni}} &= (\Delta S)_{\mathrm{sys}} + (\Delta S)_{\mathrm{surr}} \\ &= 10 - 1.485 = \mathbf{8.51\,J/K} \end{aligned}$$

169. (*B*) Explanation:

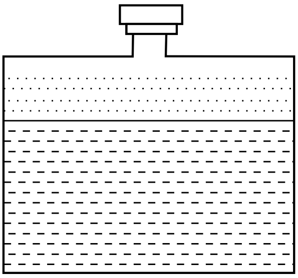


Fig.

Given data:

$$T_{\mathrm{sat}} = 100\,°\mathrm{C}$$

$$V_f = \frac{V_g}{10} \tag{2}$$

$$v_f = 0.001\,\mathrm{m^3/kg}$$

$$v_g = 1.6\,\mathrm{m^3/kg}$$

$$v_f = \frac{V_f}{m_f} \quad v_g = \frac{V_g}{m_g}$$

or

$$V_f = m_f v_f \quad V_g = m_g v_g$$

Substituting the values of v_f and v_g in Eq. (2), we get

$$m_f v_f = \frac{m_g v_g}{10}$$

or

$$\begin{aligned} \frac{m_g}{m_f} &= \frac{10 v_f}{v_g} \\ &= \frac{10 \times 0.001}{1.6} = 6.25 \times 10^{-3} \end{aligned}$$

Dryness fraction:

$$\begin{aligned} x &= \frac{m_g}{m_g + m_f} = \frac{1}{1 + \frac{m_f}{m_g}} \\ &= \frac{1}{1 + \frac{1}{6.25 \times 10^{-3}}} = \frac{1}{1 + 160} = \mathbf{\frac{1}{161}} \end{aligned}$$

172. (*B*) Explanation:

Terminal temperature difference (TTD). It is defined for all closed feedwater heaters as the difference between saturation temperature of bled steam and exit temperature.

Mathematically,

$$\begin{aligned} \mathrm{TTD} = {} & \text{Saturation temperature of bled steam} \\ & - \text{exit water temperature} \end{aligned}$$

Pinch point (P.P.)

The minimum temperature difference between the two fluid and the points where these occur are called pinch points:

181. (*A*) **Explanation:** Given data:

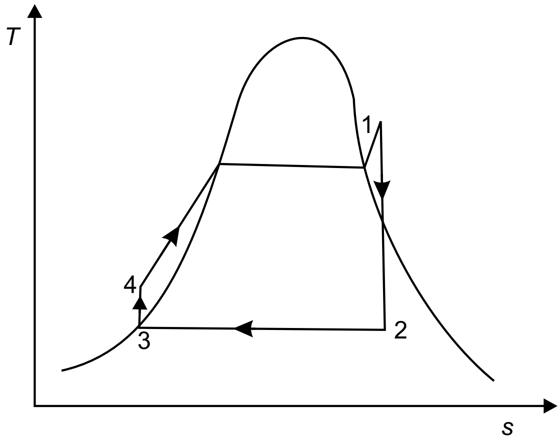


Fig. Ra nkineCycle

Specific enthalpy at turbine inlet:

$$h_1 = 2800\,\text{kJ/kg}$$

Specific enthalpy at turbine outlet:

$$h_2 = 1800\,\text{kJ/kg}$$

Specific work output by the turbine:

$$\begin{aligned} w_T &= h_1 - h_2 \\ &= 2800 - 1800 \\ &= 1000\,\text{kJ/kg} \end{aligned}$$

Net work output:

$$w_{\text{net}} = w_T - w_p$$

where pump work:

$$w_p = 0$$

$$\begin{aligned} \therefore\ w_{\text{net}} &= w_T \\ &= 1000\,\text{kJ/kg} \end{aligned}$$

Specific steam consumption:

$$ssc = \frac{3600}{1000}\,\text{kg/kWh} = \mathbf{3.6\,kg/kWh}$$

182. (*A*) Explanation:

Turbine output power

$$P_T = 1.2\,\text{MW} = 1200\,\text{kW}$$

Heat supplied:

$$q_S = 3000\,\text{kJ/kg}$$

Heat rejected:

$$q_R = 2000\,\text{kJ/kg}$$

Pump input power:

$$P_p = 6\,\text{kW}$$

Net power output:

$$P = P_T - P_P = 1200 - 6 = 1194\,\text{kW}$$

also

$$P = m(q_S - q_R)$$

$$\therefore\ 1194 = m(3000 - 2000)$$

or

$$\boldsymbol{m} = \mathbf{1.194\,kg/s}$$

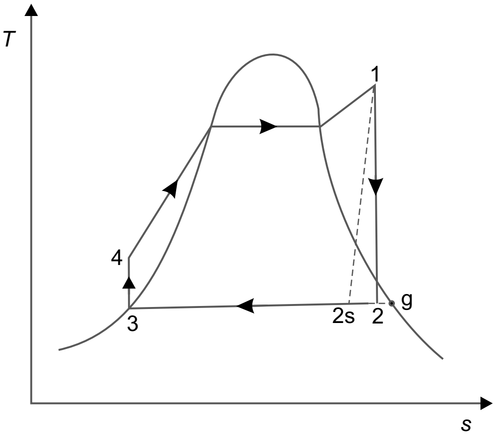
184. (*C*), 185 (*C*) Explanation: Given data:

$$p_1 = 4\,\text{MPa} = 4000\,\text{kPa}$$

$$T_1 = 350\,^\circ\text{C}$$

$$p_2 = p_3 = 15\,\text{kPa}$$

$$\eta_T = 90\% = 0.90$$



From given superheated steam table at

$$p_1 = 4\,\text{MPa}, T_1 = 350\,^\circ\text{C}$$

we get

$$h_1 = 3092.5\,\text{kJ/kg}$$

$$s_1 = 6.5821\,\text{kJ/kgK}$$

From given saturated steam table (pressure based)

$$h_f = h_3 = 226.95\,\text{kJ/kg} \quad s_f = 0.7549\,\text{kJ/kgK}$$

$$h_g = 2599.1\,\text{kJ/kg} \quad s_g = 8.0085\,\text{kJ/kgK}$$

$$v_f = v_3 = 0.001014\,\text{m}^3/\text{kg}$$

$$s_1 = s_2 = s_f + x_{2\,s}\left(s_g - s\right)$$

$$6.5821 = 0.7549 + x_{2s}(8.0085 - 0.7549)$$

$$6.5821 = 0.7549 + 7.2536\,x_{2s}$$

or

$$7.253x_{2s} = 5.8272$$

or

$$x_{2s} = 0.8033$$

$$\begin{aligned} h_{2s} &= h_f + x_{2s}\left(h_g - h_f\right) \\ &= 2226.95 + 0.8033(2599.1 - 226.95) \\ &= 26.95 + 1905.54 \\ &= 2132.49\,\text{kJ/kg} \end{aligned}$$

$$\eta_T = \frac{(\Delta_h)_{\text{act}}}{(\Delta h)_{\text{Isen}}} = \frac{h_1 - h_2}{h_1 - h_{2\text{s}}}$$

$$0.90 = \frac{3092.5 - h_2}{3092.5 - 2132.49}$$

$$0.90 = \frac{3092.5 - h_2}{960.01}$$

or

$$960.01 \times 0.90 = 3092.5 - h_2$$

or

$$864 = 3092.5 - h_2$$

or

$$h_2 = 2228.5\,\text{kJ/kg}$$

Turbine work:

$$\begin{aligned} w_T &= h_1 - h_2 \\ &= 3092.5 - 2228.5 = 864\,\text{kJ/kg} \end{aligned}$$

Pump work:

$$w_p = v_3(p_1 - p_2)\text{kJ/kg}$$

where v_3 is in m^3/kg
and p_1 and p_2 are in kPa

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

Net work output of the cycle,

$$w_{\text{net}} = w_T - w_p \\ = 864 - 4.04 = 859.96\,\text{kJ/kg} \approx 860\,\text{kJ/kg}$$

also

$$w_p = h_4 - h_3$$

$$\therefore\ 4.04 = h_4 - 226.95$$

or

$$h_4 = 230.99\,\text{kJ/kg}$$

Heat supplied to the cycle:

$$q_{4-1} = h_1 - h_4 \\ = 3092.5 - 230.99 = 28615\text{kJ/kg} \\ = \mathbf{2863\,kJ/kg}$$

186. (*B*) Explanation: Given data:

$$h_1 = 3251.0\,\text{kJ/kg}$$

$$p_2 = 15\,\text{kPa}$$

$$x_2 = 0.9$$

At p_2,

$$h_f = 225.94\,\text{kJ/kg}$$

and

$$h_g = 2598.3\,\text{kJ/kg}$$

$$\text{m} = 10\,\text{kg/s}$$

$$h_2 = h_f + x\left(h_g - h_f\right) \\ = 225.94 + 0.9(2598.3 - 225.94) \\ = 2361.06\,\text{kJ/kg}$$

Power output for the turbine:

$$P = m(h_1 - h_2) \\ = 10(3251 - 2361.06) \\ = 8899.4\,\text{kW} = 8.89\,\text{MW} \approx 8.9\,\text{MW}$$

187. (*B*), (188) (*C*) 189 (*A*)

Explanation: From superheated steam table

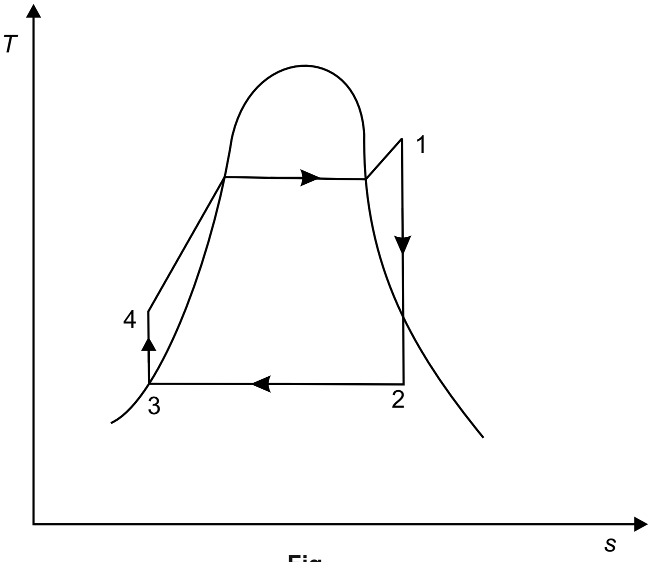
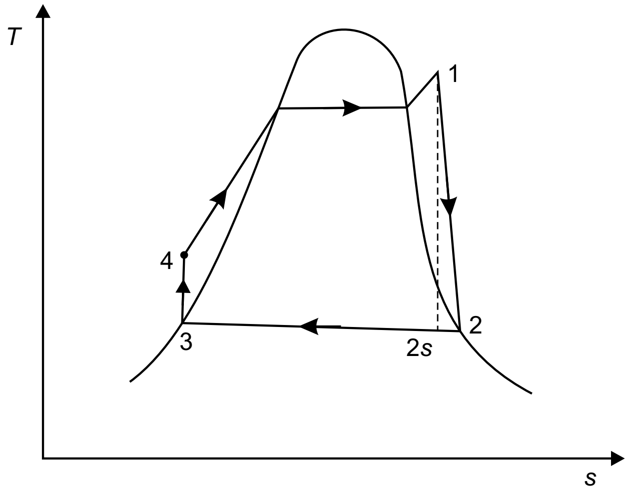


Fig.

At

$$p_1 = 60\,\text{bar}, T_1 = 600\,^\circ\text{C}$$
$$h_1 = 3658.4\,\text{kJ/kg}$$
$$s_1 = 7.1677\,\text{kJ/kgK}$$

From saturated steam table,



At

$$p_2 = 40\,\text{kPa}$$
$$h_2 = h_g = 2636.8\,\text{kJ/kg}$$
$$h_f = 317.6\,\text{kJ/kg}$$
$$s_f = 1.026\,\text{kJ/kgK}$$
$$s_g = 7.67\,\text{kJ/kgK}$$

Specific work output:

$$\begin{aligned} w_T &= h_1 - h_2 \\ &= 3658.4 - 2636.8 \\ &= 1021.6\,\text{kJ/kg} \end{aligned}$$

For ideal isentropic process 1–2s,

$$s_1 = s_{2s} = s_f + x\left(s_g - s_f\right)$$

$$7.1677 = 1.026 + x(7.67 - 1.026)$$
$$7.1677 - 1.026 = x \times 6.644$$
$$6.1417 = 6.644x$$

or

$$x = 0.9243$$

$$\begin{aligned} h_{2s} &= h_f + x\left(h_g - h_f\right) \\ &= 317.6 + 0.9243(2636.8 - 317.6) \\ &= 2461.23\,\text{kJ/kg} \end{aligned}$$

Maximum specific work output,

$$\begin{aligned} w_{T,\max} &= h_1 - h_{2\text{s}} \\ &= 3658.4 - 2461.23 = \mathbf{1197.17\,kJ/kg} \end{aligned}$$

Turbine efficiency:

$$\begin{aligned} \eta_T &= \frac{w_T}{w_{T,\max}} \\ &= \frac{1021.6}{1197.17} = 0.8533 = \mathbf{85.33\%} \end{aligned}$$

190. (*A*) Explanation:

$$m = 4\,\text{kg/s}$$

$$p_1 = 100\,\text{kPa}$$

$$p_2 = 6\,\text{MPa} = 6000\,\text{kPa}$$

Power:

$$\begin{aligned} P &= mv(p_2 - p_1) \\ &= \frac{m(p_2 - p_1)}{\rho} \quad \because v = \frac{1}{\rho} \\ &= \frac{900 \times 10^7}{3600}\,\text{kJ/s} \quad \because \rho = 1000\,\text{kg/m}^3 \text{ for water} \\ &= \mathbf{23.6\,kW} \end{aligned}$$

191. (*C*) Explanation: Given data:

$$W = 1000\,\text{MW}$$

$$\begin{aligned} Q_1 &= 900 \times 10^7\,\text{kJ/h} \\ &= \frac{900 \times 10^7}{3600}\,\text{kJ/s} = 0.25 \times 10^7\,\text{kW} \\ &= 250 \times 10^7\,\text{W} \\ &= 2500\,\text{MW} \end{aligned}$$

Heat rejected:

$$\begin{aligned} Q_2 = Q_1 - W &= 2500 - 1000 \\ &= \mathbf{1500\,MW} \end{aligned}$$

192. (*A*) Explanation: Given data:

$$W_T = 1.2\,\text{MW} = 1200\,\text{kW}$$

$$q_{4-1} = 3000\,\text{kJ/kg}$$

$$\begin{aligned} \therefore\ Q_{4-1} &= m_s q_{4-1} \\ &= 3000\,m_s\,\text{kW} \end{aligned}$$

where

$$m_s = \text{steam flow rate, kg/s}$$

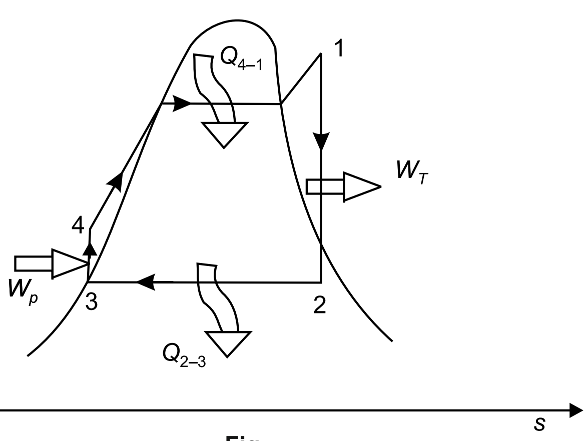


Fig.

$$q_{2-3} = 2000\,\text{kJ/kg}$$
$$\therefore\; Q_{2-3} = 2000\,m_s\,\text{kW}$$
$$W_p = 6\,\text{kW}$$

Applying the energy balance equation,

$$\text{Sum of supplied energy} = \text{sum of output energy}$$

$$\begin{aligned} W_P + Q_{4-1} &= W_T + Q_{2-3} \\ 6 + 3000\,m_s &= 1200 + 2000\,m_s \\ (3000 - 2000)m_s &= 1194 \\ 1000m_s &= 1194 \\ m_s &= \mathbf{1.194\,kg/s} \end{aligned}$$

193. (*C*) Explanation:

$$\begin{aligned} \text{Total area of heat loss} &= \text{Cylindrical surface} \\ &\quad + \text{Heat loss from cylinder head} \\ &\quad + \text{Heat loss through piston} \\ &= \pi dl + \frac{\pi}{4}d^2 + \frac{\pi}{4}d^2 \\ &= \pi \times 2 \times 2\frac{\pi}{4}(2)^2 + \frac{\pi}{4}(2)^2 \\ &= 4\pi + \pi + \pi = \mathbf{6\pi} \end{aligned}$$

194. (*D*) Explanation:

Given data:

Number of cylinder:

$$x = 6$$

Bore:

$$d = 17\,\text{cm}$$

Stroke:

$$l = 30\,\text{cm}$$

Total clearance volume:

$$V_c = 9225\,\text{cm}^3$$

Swept volume:

$$\begin{aligned} V_s &= \frac{\pi}{4}d^2\ell \\ &= \frac{3.14}{4} \times (17)^2 \times 30 = 6805.95\,\text{cm}^3 \end{aligned}$$

Total swept volume:

$$V_s = 6 \times 6805.95 = 40835.7\,\text{cm}^3$$

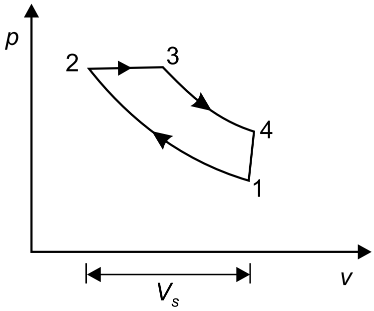
Total volume:

$$\begin{aligned} V &= V_c + V_s \\ &= 9225 + 40835.7 = 50060.7\,\text{cm}^3 \end{aligned}$$

Compression ratio:

$$\text{r} = \frac{V}{V_c} = \frac{50060.7}{9225} = \mathbf{5.4}$$

196. (*D*) Explanation: Given data



FigDieselCycle

Compression ratio:

$$r = 14$$

i.e.,

$$\frac{V_1}{V_2} = 14$$

$$\frac{V_2 + V_3}{V_2} = 14$$

$$1 + \frac{V_3}{V_2} = 14$$

$$\frac{V_3}{V_2} = 13$$

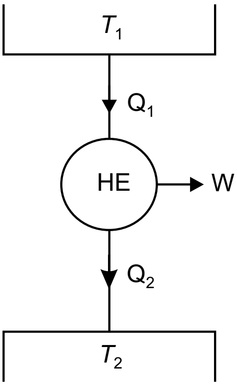
and

$$V_3 = V_2 + 0.06V_5 = V_2 + 0.06 \times 13V_2$$
$$V_3 = V_2(1 + 0.06 \times 13) = V_2 \times 1.78$$

or

$$\frac{V_3}{V_2} = \mathbf{1.78}$$

198. (*B*) Explanation:



$$Q_1 = 100\,\text{kJ}$$
$$T_1 = 600\,\text{K}$$
$$T_2 = 300\,\text{K}$$

Displacement volume:

$$V_s = 0.2\,\text{m}^3$$

$$\eta_{\text{Carnot}} = 1 - \frac{T_2}{T_1}$$
$$= 1 - \frac{300}{600} = 1 - 0.5 = 0.5$$

also

$$\eta_{\text{Carnot}} = \frac{W}{Q_1}$$

$$\therefore\ 0.5 = \frac{W}{100}$$

or

$$W = 50\,\text{kJ}$$

Mean effective pressure:

$$p_w = \frac{W}{V_s} = \frac{50}{0.2} = 250\,\text{kPa} = \mathbf{2.5\,bar}$$

201. (*D*) Explanation

$$W = 2Q_2 \quad \text{given condition}$$

or

$$Q_2 = \frac{W}{2}$$

and

$$W = Q_1 - \frac{W}{2}$$

or

$$Q_1 = W + \frac{W}{2} = \frac{3}{2}W$$

Thermal efficiency:

$$\eta_{\text{Otto}} = \frac{W}{Q_1} = \frac{W}{\frac{2}{3}W}$$

also

$$\eta_{\text{Otto}} = 1 - \frac{1}{r^{\gamma-1}} \quad \text{for Otto cycle}$$
$$\therefore\ \frac{2}{3} = 1 - \frac{1}{r^{1.4-1}} = 1 - \frac{1}{r^{0.4}}$$

or

$$\frac{1}{r^{0.4}} = 1 - \frac{2}{3} = \frac{1}{3}$$

or

$$r^{0.4} = 3$$

or

$$r = 3^{2.5} = \mathbf{15.58}$$

where r is compression ratio.

202. (*C*) Explanation: Given data:

Stroke length:

$$l = 250\text{mm} = 0.25\text{m}$$

Bore diameter:

$$d = 200\,\text{mm} = 0.2\,\text{m}$$

Clearance volume:

$$V_c = 0.001\,\text{m}^3$$

$$\gamma = 1.4$$

Displacement volume:

$$V_s = \frac{\pi}{4} d^2 \times l = \frac{3.14}{4} \times (0.2)^2 \times 0.25$$
$$= 7.85 \times 10^{-3}\,\text{m}^3$$

Total volume in the cylinder:

$$V_1 = V_c + V_\text{s}$$
$$= 0.001 + 7.83 \times 10^{-3} = 8.85 \times 10^{-3}\,\text{m}^3$$

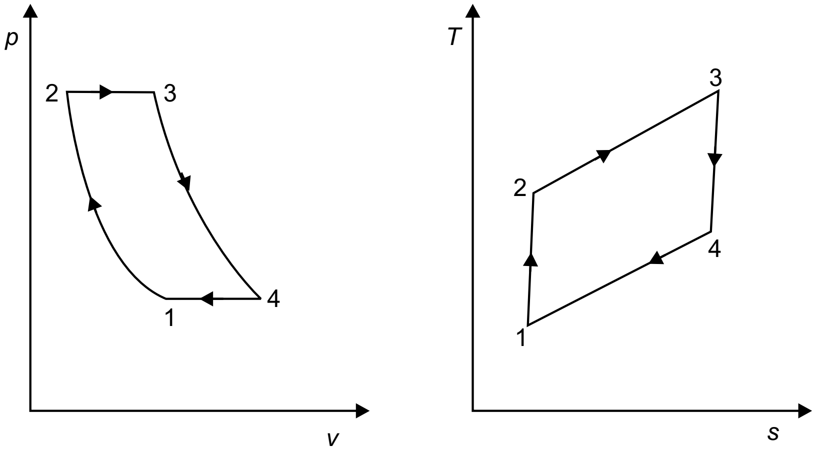
Compression ratio:

$$r = \frac{V_1}{V_c} = \frac{8.85 \times 10^{-3}}{0.001} = 8.85$$

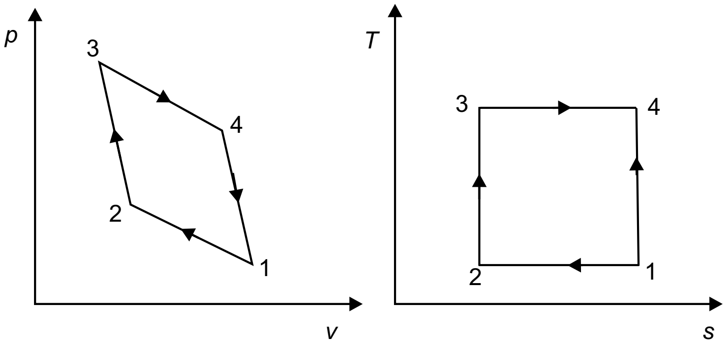
∴ Air-standard cycle efficiency:

$$\eta = 1 - \frac{1}{r^{1.4-1}} = 1 - \frac{1}{(8.85)^{1.4-1}}$$
$$= 1 - \frac{1}{(8.85)^{0.4}} = 0.5819 \approx 58.2\%$$

206 (*C*) Explanation:



Option (*A*) is wrong because work transferred.
Option (*B*) is wrong because work transferred during four processes.



Option (*C*) is right because heat is transferred only during two isobaric processes and work is transferred only during two adiabatic processes.

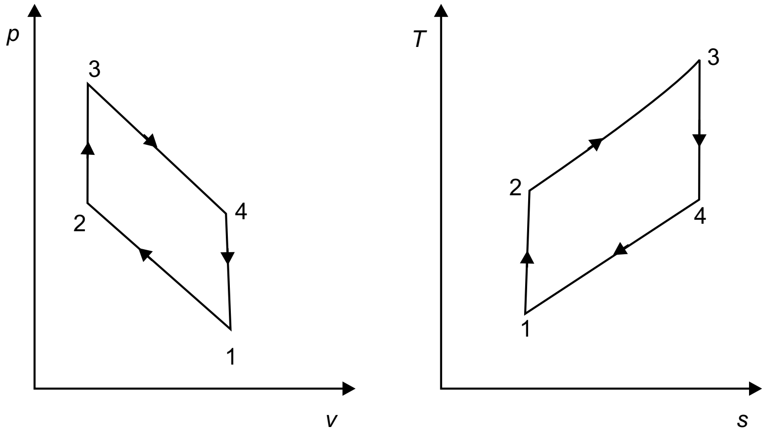
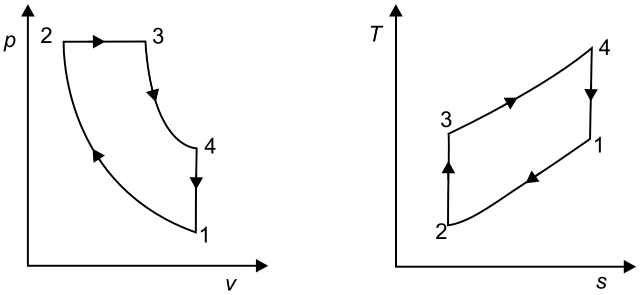


Fig. Ottocycle-twoisochoricprocessesandtwoadiabaticprocess.

Option (*D*) is wrong because work transferred during three processes—one isobaric and two-adiabatic processes.



FigDieselcycle-oneisobaricprocess,twoadiabaticandoneisochoricprocess.

207 (*A*).

1. Same maximum pressure and constant heat input.

That is.,

$$(Q_{4-1})_{\text{Otto}} > (Q_{4'-1})_{\text{Diesel}}$$

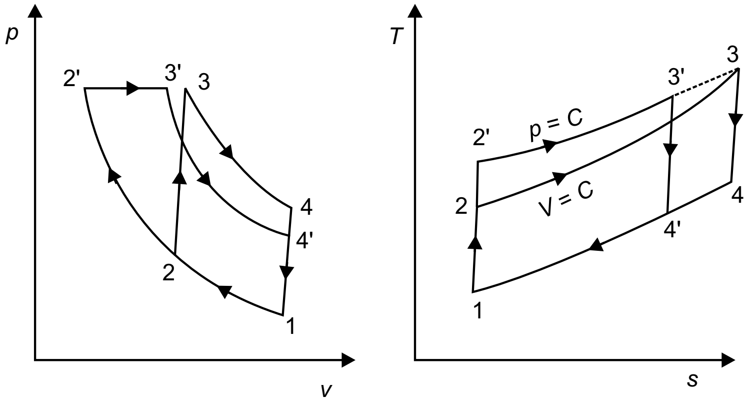


Fig. Constantmaximumpressureandconstantheatinput.

Otto cycle: 1–2–3–4–1
Diesel cycle: 1–2′–3′–4′–1′
Thermal efficiency:

$$\eta = 1 - \frac{\text{Heat rejection}}{\text{Heat input}}$$

As the heat input is constant. The thermal efficiency of the cycle is dependent only on the heat rejection, lower the heat rejection, higher the thermal efficiency. It is cleared from *T*–*s* diagram, heat rejection is higher in the Otto cycle and lower in the diesel cycle.

Thus, the thermal efficiency of the diesel cycle is more than the Otto cycle at given conditions of the constant maximum cycle pressure and constant heat input,

i.e.,

$$\eta_{\text{Diesel}} > \eta_{\text{Otto}}$$

2. Constant maximum pressure and constant output.

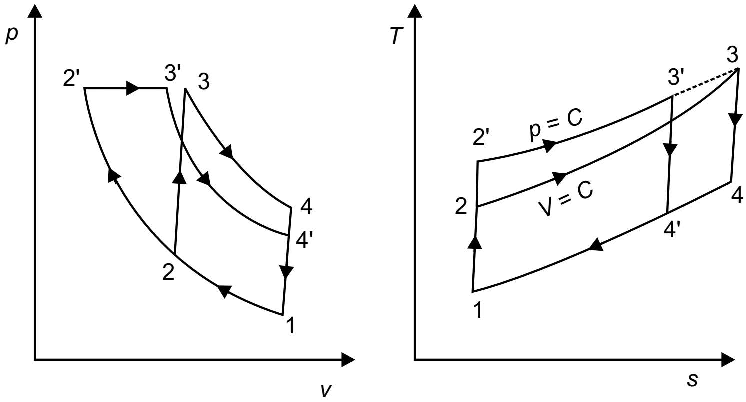


Fig. Constantmaximumpressureandconstantoutput

Otto cycle: 1–2–3–4–1
Diesel cycle: 1–2′–3′–4′–1′
Thermal efficiency:

$$\eta = 1 - \frac{\text{Work done}}{\text{Heat supplied}}$$

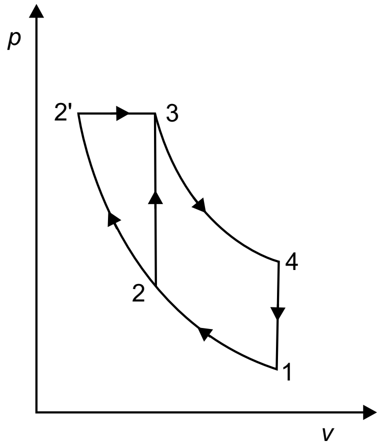
$$\eta = 1 - \frac{\text{Work done}}{\text{Work done} + \text{heat rejection}}$$

The work done in Otto and diesel cycles (Area under 1–2–3–4–1 and 1–2′–3–4′–1, respectively) can be equal only if the point 3 has a greater entropy than 3′ at the same pressure and in that case, the heat rejection by diesel cycle is less than that of Otto cycle. Hence, for given conditions, the diesel cycle is more efficient than the Otto cycle.

That is,

$$\eta_{\text{Diesel}} > \eta_{\text{Otto}}$$

3. Constant maximum pressure and constant maximum temperature.



Otto cycle: 1–2–3–4–1
Diesel cycle: 1–2′–3–4–1
Thermal efficiency:

$$\eta = 1 - \frac{\text{Heat rejection}}{\text{Heat input}}$$

At constant maximum pressure and constant maximum temperature, the heat rejection becomes the same. It is cleared from *T–s* diagram, heat input in the diesel cycle is more than heat input in the Otto cycle. Thus, the thermal efficiency of the diesel cycle is more than Otto cycle.

That is,

$$\eta_{\text{Diesel}} > \eta_{\text{Otto}}$$

209. (*D*) Explanation: Given data:

Crank radius:

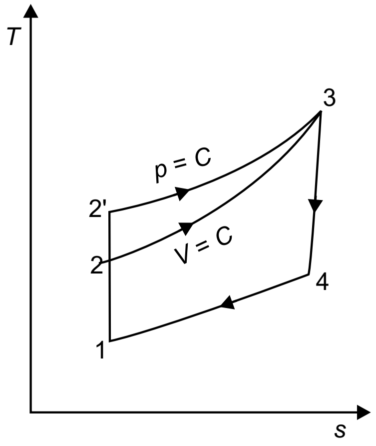
$$r = 60\,\text{mm} = 6\,\text{cm}$$

∴ Stroke:

$$l = 2r = 2 \times 6 = 12\,\text{cm}$$

Diameter of the cylinder:

$$d = 80\,\text{mm} = 8\,\text{cm}$$



Swept Volume:

$$V_s = \frac{\pi}{4} d^2 l = 602.88 \approx \mathbf{603\,cm^3}$$

210. (*A*) Explanation: Given data:

$$n = \frac{N}{2} \quad \text{for four - stroke engine}$$

$$V_s = 0.0259\,\text{m}^3$$

$$P = 950\,\text{kW}$$

$$N = 2200\,\text{rpm}$$

We know that power output:

$$P = \frac{p_m A I n x}{2}\,\text{kW}$$

$$P = \frac{p_\text{m} V_s n x}{60}\,\text{kW}$$

where

$$P\,\text{is in kW}$$

$$V\,\text{is in m}^3$$

$$n = \frac{N}{2}\,\text{rpm}$$

$$x = 1\,\text{,number of cylinder}$$

$$\therefore\ 950 = \frac{p_m \times 0.0259}{60} \times \frac{N \times 1}{2}$$

$$950 = \frac{p_m \times 0.0259 \times 2200}{120}$$

or

$$p_m = 2000\,\text{kPa} = \mathbf{2\,MPa}$$

211. (*D*) Explanation: Given data for Otto cycle:

Compression ratio:

$$r = \frac{v_1}{v_2} = 10$$

$$p_1 = 100\,\text{kPa}$$

$$T_1 = 27\,^\circ\text{C} = (273 + 27)\text{K} = 300\,\text{K}$$

$$q_1 = 1500\,\text{kJ/kg}$$

$$q_2 = 700\,\text{kJ/kg}$$

$$R = 0.287\,\text{kJ/kgK}$$

$$p_1 v_1 = R T_1$$

$$100 \times v_1 = 0.287 \times 300$$

or

$$v_1 = 0.861\,\text{m}^3/\text{kg}$$

also

$$v_1 = 10\,v_2$$

$$\therefore\ 0.861 = 10 v_2$$

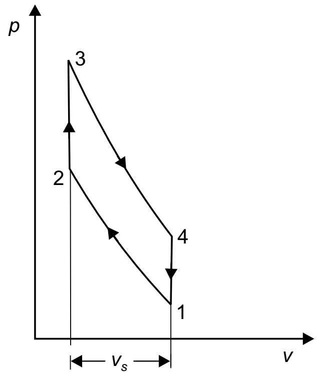
or

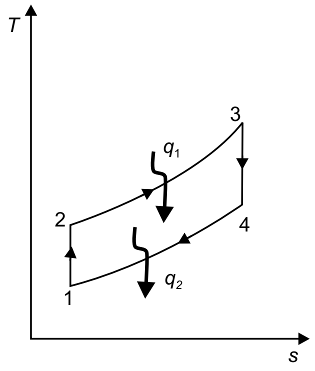
$$v_2 = 0.0861\,\text{m}^3/\text{kg}$$

Specific work done per cycle:

$$w = q_1 - q_2 = 1500 - 700 = 800\,\text{kJ/kg}$$

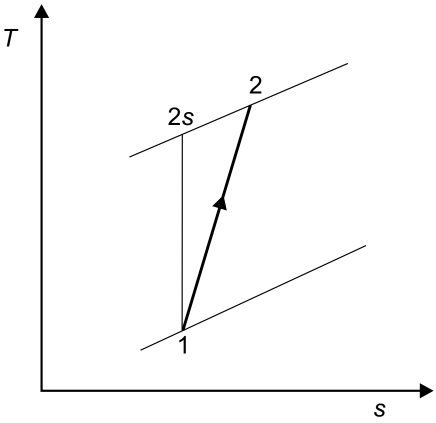
We know that mean effective pressure,





$$p_m = \frac{w}{v_s} = \frac{800}{v_1 - v_2}$$
$$= \frac{800}{0.861 - 0.0861} = \mathbf{1032.39\,kPa}$$

213. (***D***) Explanation:

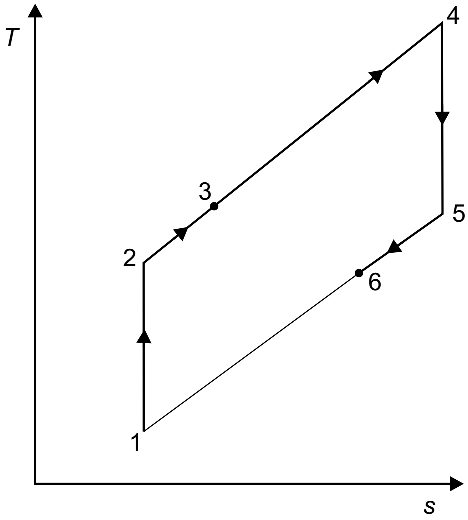


$$T_1 = 300\,\mathrm{K}$$
$$T_{2s} = 575\,\mathrm{K}$$
$$T_2 = 600\,\mathrm{K}$$

Efficiency of compressor:

$$\eta_c = \frac{(\Delta T)_{\mathrm{act}}}{(\Delta T)_{\mathrm{isen}}}$$
$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1}$$
$$\eta_c = -\frac{575 - 300}{600 - 300}$$
$$= \frac{275}{300} = 0.9166 \approx \mathbf{91.7\%}$$

214 (***D***) Explanation: Given data



$$T_2 = 550\,\mathrm{K}$$
$$T_5 = 800\,\mathrm{K}$$
$$\in = 0.8$$
$$T_3 = ?$$
$$\in = \frac{T_3 - T_2}{T_3 - T_2}$$
$$0.8 = \frac{T_3 - 550}{800 - 550}$$

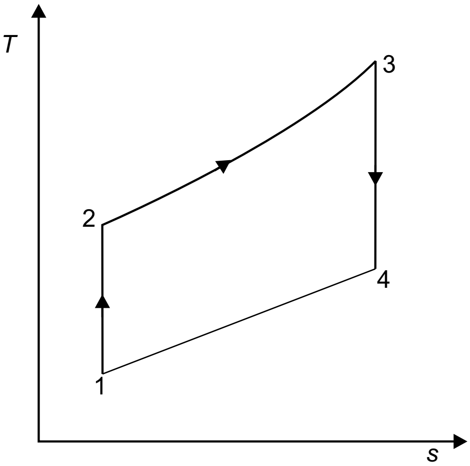
or

$$T_3 - 550 = 200$$

or

$$T_3 = \mathbf{750\,K}$$

221. (*B*), 222 (*C*) Explanation: Given data



$$T_1 = 35\,^\circ\mathrm{C}$$
$$= (35 + 273)\mathrm{K}$$
$$= 308\,\mathrm{K}$$

$$T_3 = 900\,^\circ\mathrm{C}$$
$$= (900 + 273)\mathrm{K}$$
$$= 1173\,\mathrm{K}$$

$$p_1 = 100\,\mathrm{kPa}$$

$$p_2 = 500\,\mathrm{kPa}$$

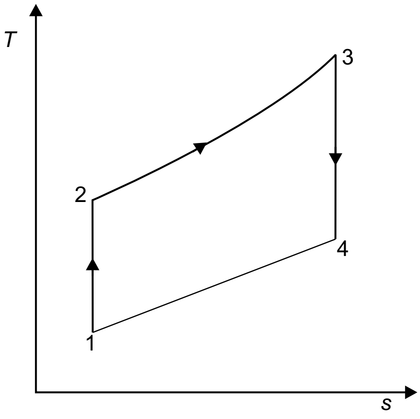
Maximum work output per unit mass flow rate,

$$w_{\max} = c_p\left(\sqrt{T_3} - \sqrt{T_1}\right)^2$$
$$= 1.005(\sqrt{1173} - \sqrt{308})^2$$
$$= \mathbf{280.25\,kJ/kg}$$

For maximum work output condition,

$$\eta_{\mathrm{Brayton}} = 1 - \sqrt{\frac{T_1}{T_3}}$$
$$= 1 - \sqrt{\frac{308}{1173}} = \mathbf{0.4875}$$

223. (*A*)Explanation: Given data



$$p_1 = 1\,\mathrm{bar}$$

$$p_2 = 6\,\mathrm{bar}$$

$$T_1 = 300\,\mathrm{K}$$

$$T_3 = 1500\,\mathrm{K}$$

For process 1–2

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

$$\frac{T_2}{300} = \left(\frac{6}{1}\right)^{\frac{1.4-1}{1.4}}$$
$$= (6)^{0.285} = 1.666$$
$$T_2 = 300 \times 1.666 = 499.8\mathrm{K} \approx \mathbf{500\,K}$$

For process 3–4

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \because p_3 = p_p, p_4 = p_1$$

$$\frac{1500}{T_4} = (6)^{\frac{1.4-1}{1.4}} = (6)^{0.285} = 1.666$$

or

$$T_4 = \frac{1500}{1.666} = 900.36\mathrm{K} \approx \mathbf{900\,K}$$

224. (***D***) Explanation:

Calorific value:

$$CV = 40000\,\mathrm{kJ/kg}$$

Air–fuel ratio:

$$\frac{m_a}{m_f} = 80$$

Net work output:

$$w_{\mathrm{net}} = 80\,\mathrm{kJ/kg}$$

Heat supplied:

$$Q = m_f CV = 40000\,\mathrm{m}_f\,\mathrm{kJ}$$

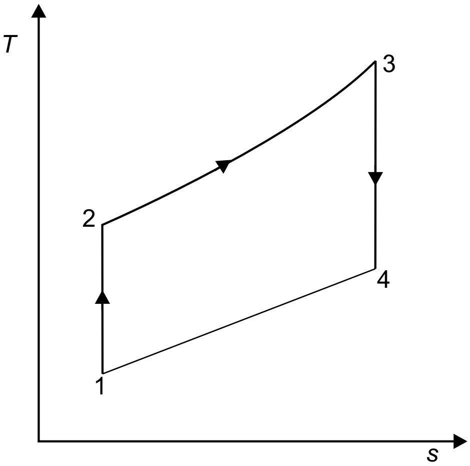
Net work output

$$\begin{aligned} W_{\mathrm{net}} &= \left(m_f + m_a\right) w_{\mathrm{net}} \\ &= \left(m_f + m_o\right) 80\,\mathrm{kJ} \end{aligned}$$

Thermal efficiency:

$$\begin{aligned} \eta &= \frac{W_{\mathrm{net}}}{Q} = \frac{\left(m_f + m_a\right)80}{m_f \times 40000} = \left(1 + \frac{m_a}{m_f}\right)\frac{80}{40000} \\ &= (1+80)\frac{80}{40000} = \frac{81 \times 80}{40000} = 0.162 = \mathbf{16.20\%} \end{aligned}$$

237. (***C***) Explanation:



Maximum temperature:

$$T_3 = 1200\,\mathrm{K}$$

Minimum temperature:

$$T_1 = 300\,\mathrm{K}$$

Cycle efficiency:

$$\eta = 1 - \frac{T_1}{T_2}$$

For maximum work capacity,

$$T_2 = \sqrt{T_1 T_3}$$

Maximum cycle efficiency,

$$\begin{aligned} \eta_{\max} &= 1 - \frac{T_1}{\sqrt{T_1 T_3}} \\ &= 1 - \sqrt{\frac{T_1}{T_3}} = 1 - \sqrt{\frac{300}{1200}} \\ &= 1 - 0.5 = 0.5 = \mathbf{50\%} \end{aligned}$$

or

also

$$\eta = 1 - \frac{1}{r_P^{(\gamma-1)/\gamma}}$$

For maximum work capacity:

$$\begin{aligned} \frac{T_{\max}}{T_{\min}} &= \frac{T_3}{T_1} = \left(r_P\right)^{2(\gamma-1)/\gamma} \\ \frac{T_3}{T_1} &= \left(r_p\right)^{2(\gamma-1)/\gamma} \end{aligned}$$

or

$$r_{\overline{P}} = \left(\frac{T_3}{T_1}\right)^{\gamma/2(\gamma-1)} = \left(\frac{200}{200}\right)^{1.4/2(1.4-1)} = 11.31$$

∴ Optimum pressure ratio

$$r_{p,\mathrm{opt}} = 11.31$$

Maximum efficiency:

$$\eta_{\max} = 1 - \frac{1}{\left(r_{p,\text{opt}}\right)^{(\gamma-1)/\gamma}}$$

$$\eta_{\max} = 1 - \frac{1}{(11.31)^{(1.4-1)/1.4}} = 1 - 0.5 = 0.5$$
$$= \mathbf{50\%}$$

Practice/Exam Questions with Solutions

Q.1. (*a*) **State the conditions for a process to be reversible.**

Ans. The process in which the system and surroundings can be restored to the initial state from the final state without changing the thermodynamics properties of the universe is called a reversible process.

There are two important conditions for the reversible process to occur.

1. The process should occur in infinitesimally slow rate.
2. All of the initial and final state of the system should be in equilibrium with each other.

(*b*) **Write the steady flow energy equation and point out the significance of various terms involved.**

Ans. The steady flow energy equation (SFEE) for unit mass flow rate,

$$h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + gz_2 + w_{1-2}$$

where

h_1 = specific enthalpy at inlet in J/kg
$\frac{V_1^2}{2}$ = kinetic energy per unit mass at inlet in J/kg
gz_1 = datum energy per unit mass at inlet in J/kg
$q_{1\text{-}2}$ = specific heat transfer in J/kg
h_2 = specific enthalpy at outlet in J/kg
$\frac{V_2^2}{2}$ = kinetic energy per unit mass of outlet in J/kg
gz_2 = datum energy per unit mass at outlet in J/kg
w_{1-2} = specific work done in J/kg

(*c*) **Write Maxwell's equations.**

Ans. 1st Maxwell's equation,

$$\left(\frac{\partial T}{\partial v}\right)_s = -\left(\frac{\partial p}{\partial s}\right)_v$$

2nd Maxwell's equation,

$$\left(\frac{\partial T}{\partial p}\right)_s = \left(\frac{\partial V}{\partial s}\right)_p$$

3rd Maxwell's equation,

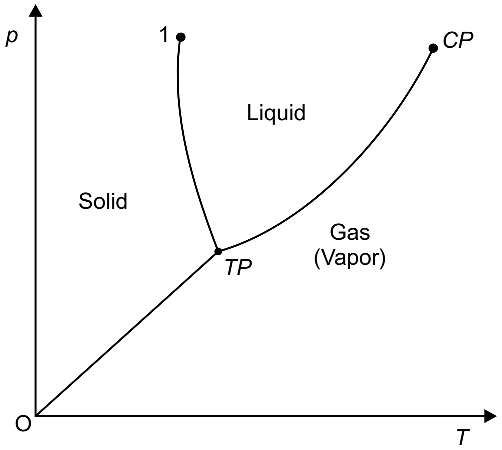
$$\left(\frac{\partial p}{\partial T}\right)_v = \left(\frac{\partial s}{\partial v}\right)_T$$

4th Maxwell's equation,

$$\left(\frac{\partial v}{\partial T}\right)_p = -\left(\frac{\partial s}{\partial p}\right)_T$$

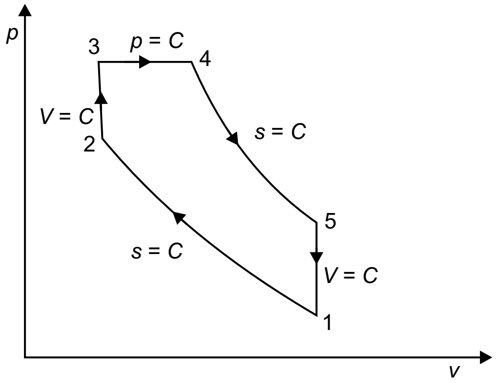
(*d*) **What is the main feature of triple point? State the values of pressure and temperature at the triple point of water.**

Ans. Tripe Point: The triple point is a state where three phases like solid, liquid and vapor coexist in equilibrium.



Feature of Triple Point:

1. At high temperatures and low pressures, only the gas phase exists. At low temperature and high pressure, only the solid phase exists. Two phases coexist only on one of the lines. The curve TP–CP describes boiling. Only at the conditions of temperature and pressure described by that curve, will liquid vapor be seen boiling.
2. The most interesting feature of the curve is the triple point—the point with a combination of pressure and temperature such that all three phases may coexist.



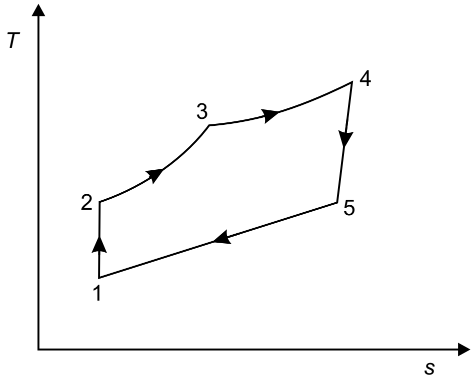
3. Phase diagrams have many features that can be studied. When the triple point of a substance occurs at pressure above room pressure, then the liquid phase in never stable at room pressure. In materials with this property, evaporation takes place directly from the solid to the vapor, a process called sublimation. The pressure for the triple point of $C0_2$ is about 5.178 bar.

(*e*) **Draw a dual cycle on *p–v* and *T–s* planes and list out various processes of the cycle.**

Ans. The five processes that form the dual cycle are shown in the *p–v* and *T–s* planes of Fig.

The five processes as:

1. **Process 1–2**: Isentropic (or reversible adiabatic) compression.
2. **Process 2–3**: Heat supplied at constant volume.
3. **Process 3–4**: Heat supplied at constant pressure.
4. **Process 4–5**: Isentropic (or reversible adiabatic) expansion.
5. **Process 5–1:** Heat rejected at constant volume.

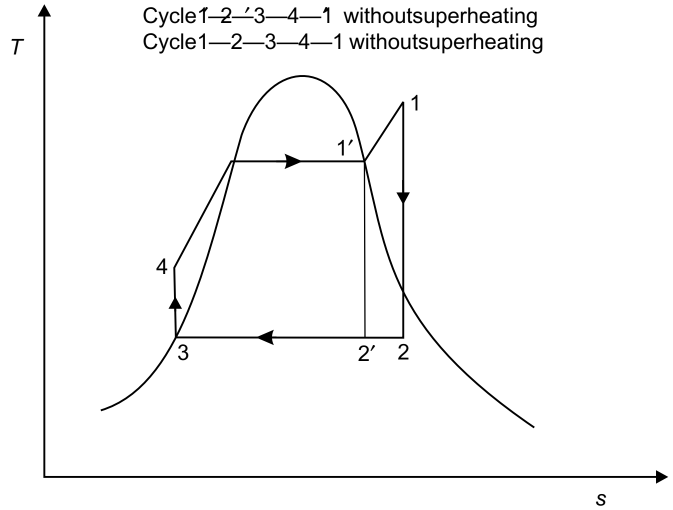


(*f*) **Why is super heating of steam done in a steam power plant? Explain with the help of T–s diagram.**

Ans. Advantages of super heating of steam:

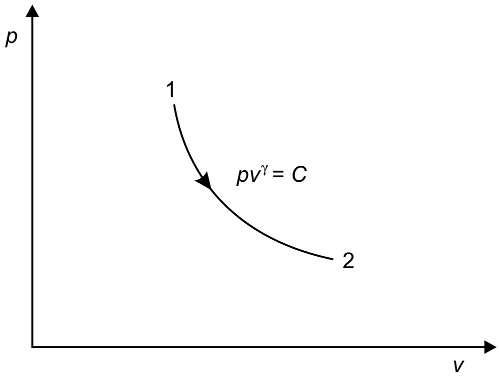
1. Superheated steam has higher heat content, and therefore, more work output can be obtained.
2. Superheating increases the thermal efficiency of the cycle.

3. Superheating decreases the moisture content of the steam at turbine exit, which reduces the turbine blade erosion.



Q.2. Define an Adiabatic Process and Show an Adiabatic Expansion Process on *P*–*v* Plane. Prove that a Reversible Adiabatic Process of a Given Mass of Perfect Gas $pv^{\gamma} = C$

Ans. Adiabatic Process: When there is no heat transfer between the system and the surroundings during a process, it is known as an adiabatic process. An adiabatic process can be carried out by the expansion or compression of gas in a cylinder whose walls are insulated. A reversible adiabatic process is called isentropic process. In isentropic process, the entropy of the system will remain constant without transfer of heat to or from the surroundings. The adiabatic expansion process is shown on *p*–*v* plane in Fig.



Prove that $pv^\gamma = C$ for reversible adiabatic process.

According to first law of thermodynamics for process,

$$\delta Q = dU + \delta W$$

For unit mass,

$$\delta Q = dU + \delta W \quad (3)$$

where

$$\delta q = 0 \quad \text{for adiabatic process}$$
$$\delta w = pdv \quad \text{for reversible process}$$
$$du = c_v dT$$

Therefore, Eq. (3) becomes

$$0 = c_v dT + pdv$$

or

$$c_v dT + pdv = 0$$
$$\frac{R}{\gamma - 1} dT + \frac{RT}{v} dv = 0 \qquad \because c_v = \frac{R}{\gamma - 1}, p = \frac{RT}{v}$$

or

$$\frac{1}{\gamma - 1} dT + \frac{Tdv}{v} = 0$$

or

$$\frac{dT}{T} + (\gamma - 1)\frac{dv}{v} = 0$$

On integrating, we get

$$\log_e T + (\gamma - 1)\log_e v = C$$
$$\log_e T + \log_e v^{\gamma - 1} = C$$

or

$$\log_e Tv^{\gamma - 1} = C$$

or

$$Tv^{\gamma - 1} = C$$

From equation of state

$$pv = RT$$

or

$$T = \frac{pv}{R}$$
$$\therefore \frac{pv}{R} v^{\gamma - 1} = C$$

or

$$\boldsymbol{pv^\gamma = C}$$

Q.3. One kg of air at 1 bar and 300 K is compressed adiabatically till its pressure becomes 5 times the original pressure. Subsequently, it is expanded at constant pressure and finally cooled at constant volume to return to its original state. Calculate the heat and work interactions and change in internal energy for each process and for the cycle.

Ans. Given data:

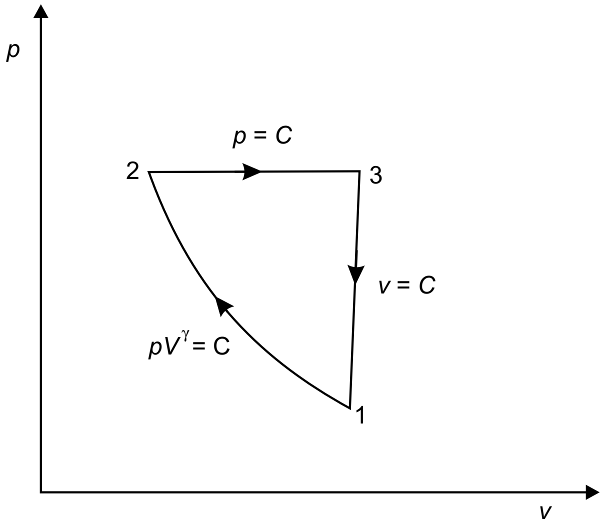
$$m = 1\,\text{kg}$$
$$p_1 = 1\,\text{bar} = 100\,\text{kPa}$$
$$T_1 = 300\,\text{K}$$
$$p_2 = 5p_1 = 5 \times 100\,\text{kPa} = 500\,\text{kPa}$$

At state 1,

$$p_1 V_1 = mRT_1$$
$$100 \times V_1 = 1 \times 0.287 \times 300$$

or

$$V_1 = 0.861\,\text{m}^3$$



Process 1–2 adiabatic,

$$p_1 V_1^{\gamma} = p_2 V_2^{\gamma}$$

$$100 \times 0.861)^{14} = 500 \times V_2^{1.4}$$

or

$$V_2^{1.4} = 0.16219$$

or

$$V_2 = 0.2727\,\mathrm{m}^3$$

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

$$\frac{T_2}{300} = \left(\frac{500}{100}\right)^{\frac{1.4-1}{1.4}} = (5)^{0.2857} = 1.583$$

or

$$T_2 = 1.583 \times 300 = 474.9\,\mathrm{K}$$

Heat transfer:

$$Q_{1-2} = 0$$

Work done:

$$W_{1-2} = \frac{mR(T_1 - T_2)}{\gamma - 1} = \frac{1 \times 0.287(300 - 474.9)}{1.4 - 1}$$
$$= -125.49\,\mathrm{kJ}$$

1st law of thermodynamic for process,

$$Q_{1-2} = dU + W_{1-2}$$

$$0 = U_2 - U_1 + W_{1-2}$$

or

$$U_2 - U_1 = -W_{1-2} = -(-125.47\mathrm{kJ})$$
$$= 125.49\,\mathrm{kJ}$$

Process 2–3 isobaric,

$$\frac{T_3}{T_2} = \frac{V_3}{V_2} = \frac{V_1}{V_2} \quad \because V_3 = V_1$$

$$\frac{T_3}{474.9} = \frac{0.861}{0.2727}$$

or

$$T_3 = 1499.40\,\mathrm{K}$$

Heat transfer:

$$\begin{aligned} Q_{2-3} &= mc_p dT = mc_p(T_3 - T_2) \\ &= 1 \times 1.005 \times (1499.40 - 474.9) \\ &= 1029.62\,\text{kJ} \end{aligned}$$

Work done:

$$\begin{aligned} W_{2-3} &= p_2(V_3 - V_2) = 500(0.861 - 0.2727) \\ &= 294.15\,\text{kJ} \end{aligned}$$

Ist law of thermodynamics for process,

$$Q_{2-3} = (U_3 - U_2) + W_{2-3}$$

$$1029.62 = U_3 - U_2 + 294.15$$

or

$$U_3 - U_2 = 735.47\,\text{kJ}$$

Process 3–1 isochoric

Heat transfer:

$$\begin{aligned} Q_{3-1} &= mc_v(T_1 - T_3) \\ &= 1 \times 0.718(300 - 1499.40) = -861.16\,\text{kJ} \end{aligned}$$

Work done:

$$W_{3-1} = 0 \quad \because V_1 = V_3$$

Ist law of thermodynamics for process,

$$Q_{3-1} = (U_1 - U_3) + W_{3-1}$$

$$-861.16 = U_1 - U_3 + 0$$

or

$$U_1 - U_3 = -861.16\,\text{kJ}$$

Net heat transfer during cycle:

$$\begin{aligned} Q_{\text{net}} &= Q_{1-2} + Q_{2-3} + Q_{3-1} \\ &= 0 + 1029.62 - 861.16 \\ &= 168.46\,\text{kJ} \end{aligned}$$

Net work done during cycle:

$$\begin{aligned} W_{\text{net}} &= W_{1-2} + W_{2-3} + W_{3-1} \\ &= -125.49 + 294.15 + 0 \\ &= 168.66\,\text{kJ} \end{aligned}$$

also, for cycle,

$$W_{\text{net}} = Q_{\text{net}} = 168.66\,\text{kJ}$$

Net change in internal energy for cycle in zero.

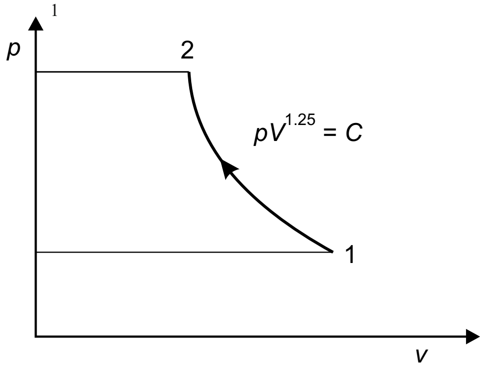
Q.4. Agasinitially at 1.5 bar pressure, 0.15 m^3 volume and 300 k was compressed polytropically (pv^{125} = constant) to 15 bar pressure. determine the change in entropy. Also, workout the approximate entropy change obtained by dividing the heat transferred by the mean absolute temperature during compression. Take c_p = 1.04 kJ/kg K and c_v = 0.748 kJ/kg K

Ans. Given data;

$$p_1 = 1.5\,\text{bar} = 150\,\text{kPa}$$

$$V_1 = 0.15\,\text{m}^3$$

$$T_1 = 300\,\text{K}$$



Polytropic law,

$$pV^{1.25} = C$$

where

$$n = 1.25$$

$$p_2 = 15\,\text{bar} = 1500\,\text{kPa}$$

$$c_p = 1.04\,\text{kJ/kgK}$$

$$c_v = 0.748\,\text{kJ/kgK}$$

$$R = c_p - c_v = 1.04 - 0.748 = 0.292\,\text{kJ/kgK}$$

At state 1,

$$p_1V_1 = mRT_1$$

$$150 \times 0.15 = m \times 0.292 \times 300$$

or

$$m = 0.2568\,\text{kg}$$

Process 1–2 polytropic,

$$p_1V_1^n = p_2V_2^n$$

$$150 \times (0.15)^{1.25} = 1500\,V_2^{1.25}$$

or

$$V_2^{1.25} = 0.00933$$

or

$$V_2 = 0.02376\,\text{m}^3$$

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

$$\frac{T_2}{300} = \left(\frac{1500}{150}\right)^{\frac{1.25-1}{1.25}} = (10)^{0.2} = 1.5848$$

or

$$T_2 = 1.5848 \times 300 = 475.44\,\text{K}$$

Change in entropy:

$$\begin{aligned} S_2 - S_1 &= mc_v \log_e \frac{T_2}{T_1} + mR\,\log_e \frac{V_2}{V_1} \\ &= 0.2568 \times 0.748 \times \log_e \frac{475.44}{300} \\ &\quad + 0.2568 \times 0.292 \times \log_e \frac{0.02376}{0.15} \\ &= 0.08844 - 0.13817 = -0.04973\,\text{kJ/K} \end{aligned}$$

Change in entropy for polytropic process is also calculated by using following relations:

1. $S_2 - S_1 = \left(\frac{\gamma - n}{1 - n}\right) mc_v \log_e \frac{T_2}{T_1}$
   $\because \gamma = \frac{c_p}{c_v} = \frac{1.07}{0.748} = 1.39.$
2. $S_2 - S_1 = (\gamma - n) mc_v \log_e \frac{V_2}{V_1}.$
3. $S_2 - S_1 = \left(\frac{n-\gamma}{n}\right) mc_v \log_e \frac{p_2}{p_1}.$
4. $S_2 - S_1 = mc_p \log_e \frac{V_2}{V_1} + mc_v \log_e \frac{p_2}{p_1}.$
5. $S_2 - S_1 = mc_p \log_e \frac{T_2}{T_1} - mR \log_e \frac{p_2}{p_1}.$

Work done:

$$\begin{aligned} W_{1-2} &= \frac{mR(T_1 - T_2)}{n - 1} = \frac{0.2568 \times 0.292(300 - 475.44)}{1.25 - 1} \\ &= -52.62\,\text{kJ} \end{aligned}$$

Change in internal energy:

$$\begin{aligned} dU &= mc_v(T_2 - T_1) \\ &= 0.2568 \times 0.748(475.44 - 300) \\ &= 33.67\,\text{kJ} \end{aligned}$$

Ist law of thermodynamic for process,

$$\begin{aligned} Q_{1-2} &= dU + W_{1-2} \\ &= 33.67 - 52.62 = -18.95\,\text{kJ} \end{aligned}$$

Mean temperature:

$$T_\text{m} = \frac{T_1 + T_2}{2} = \frac{300 + 475.44}{2} = 387.72\,\text{K}$$

Change in entropy:

$$S_2 - S_1 = \frac{Q_{1-2}}{T_m} = \frac{-18.95}{387.72} = -0.04887\,\mathrm{kJ/K}$$

Q.5. Air expands through a turbine from 500 kPa and 520 °C to 100 kPa and 300 °C. During expansion, 10 kJ/kg of heat is lost to the surroundings which are at 98 kPa and 20 °C. Neglecting the changes in kinetic and potential energies, determine per kg of air-

(a) The irreversibility,
(b) The decrease in availability,
(c) The maximum work

For air take c_p = 1.005 kJ/kgK and R = 0.287 kJ/kgK.

Ans. Given data:

At inlet condition:

$$p_1 = 500\,\mathrm{kPa}$$

$$T_1 = 520\,^\circ\mathrm{C} = (520 + 273)\mathrm{K} = 793\,\mathrm{K}$$

At exit condition:

$$p_2 = 100\,\mathrm{kPa}$$

$$T_2 = 300\,^\circ\mathrm{C} = (300 + 273)\mathrm{K} = 573\,\mathrm{K}$$

Heat lost:

$$q = 10\,\mathrm{kJ/kg}$$

Surroundings condition:

$$p_0 = 98\,\mathrm{kPa}$$

$$T_0 = 20\,^\circ\mathrm{C} = (20 + 273)\mathrm{K} = 293\,\mathrm{K}$$

$$c_p = 1.005\,\mathrm{kJ/kgK}$$

$$R = 0.287\,\mathrm{kJ/kgK}$$

The actual work is calculated by application of steady flow energy equation. According to steady flow energy equation per unit mass.

$$h_1 + \frac{V_1^2}{2} + gz_1 + q = h_2 + \frac{V_2^2}{2} + gz_2 + w$$

Neglecting the change in kinetic and potential energies

$$h_1 + q = h_2 + w$$

or

$$\begin{aligned} w &= (h_1 - h) + q \\ &= c_p(T_1 - T_2) + q \\ &= 1.005(793 - 573) - 10 = 211.1\,\mathrm{kJ/kg} \end{aligned}$$

Maximum work output per unit mass,

$$\begin{aligned} w_{\max} &= (h_1 - h_2) - T_{\mathrm{o}}\Delta S_{\mathrm{sys}} \\ &= c_p(T_1 - T_2) - T_0\left(c_p \log_e \frac{T_1}{T_2} - R \log_e \frac{p_1}{p_2}\right) \\ &= 1.005(793 - 573) - 293 \\ &\left(1.005 \log_e \frac{793}{573} - 0.287 \log_e \frac{500}{100}\right) \\ &= 221.1 - 293(0.3265 - 0.4619) \\ &= 221.1 - 293(-0.1354) = 221.1 + 39.67 \\ &= 260.77\,\mathrm{kJ/kg} \end{aligned}$$

(a) Irreversibility:

$$\begin{aligned} i &= w_{\max} - w \\ &= 260.77 - 211.1 = 49.67\,\mathrm{kJ/kg} \end{aligned}$$

or

$$\begin{aligned} i &= T_{\mathrm{o}}\Delta s_{\mathrm{uni}} = T_{\mathrm{o}}\left[\Delta s_{\mathrm{sys}} + \Delta s_{\mathrm{surr}}\right] \\ &= T_0\left[c_p \log_e \frac{T_2}{T_1} - R \log_e \frac{p_2}{p_1} + \frac{q}{T_o}\right] \\ &= 293\left[1.005 \log_e \frac{573}{793} - 0.287 \log_e \frac{100}{500} + \frac{10}{293}\right] \\ &= 293[-0.3268 + 0.4619 + 0.0341] = 49.66\,\mathrm{kJ/kg} \end{aligned}$$

(b) Decrease in availability or change in availability

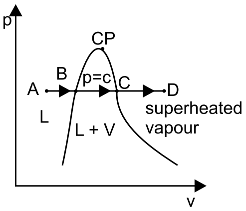
$$\Psi_1 - \Psi_2 = w_{\max} = 260.77\,\mathrm{kJ/kg}$$

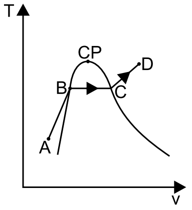
(c) Maximum work:

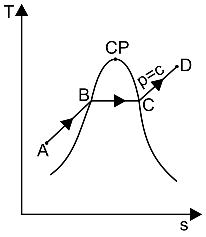
$$w_{\max} = 260.77\,\text{kJ/kg}$$

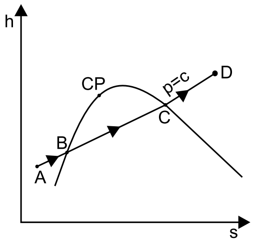
Q.6. Explain the process of steam generation (change of phase of water) at constant pressure. Show the various stages on *p–v*, *T–v*, *T–s* and *h–s* diagrams.

Ans. Consider heating of water at constant atmospheric pressure. Let water is at 20 °C at state A and atmospheric pressure, water exists in the liquid phase, and it is called a subcooled liquid. Heat is now supplied to water, both the temperature and the volume increase. As more heat is supplied, the temperature of water rises and continues to rise until it reaches 100°C (boiling point) at state B at constant pressure.









At state *B*, water is still a liquid, but further small amount of heat addition will cause some of liquid to vaporize. That is, a phase change process from liquid to vapor is about to take place. The state of liquid at which a vapor just begins to form is called a saturated liquid. Therefore, state *B* is called saturated liquid state.

At state *B*, the condition of water is saturated liquid. As heat is transferred at state B, the temperature stops rising until the liquid is completely vaporized. During the phase- change process, the only change we will observe is a large increase in the volume and slowly the liquid is converted into vapors. At state *C*, the liquid is completely converted into vapour on continuous supply of heat. The vapour at state *C* is called a saturated vapour. At saturated vapour state *C*, the vapour is called a dry-saturated steam. If the dry-saturated steam is further heated beyond the state C to any state say *D*, this heating process is called superheating. The steam obtained is called superheated steam.

Q.7. Air enters the compressor of a gas turbine plant operating on Brayton cycle at 1 bar pressure and 27 °C temperature. The pressure ratio is 6 and the maximum cycle temperature is limited to 800 °C. If the compressor and turbine efficiencies are 85% and 88%, respectively. Make calculations for the net work output, cycle efficiency and the work ratio.

Ans. Given data:

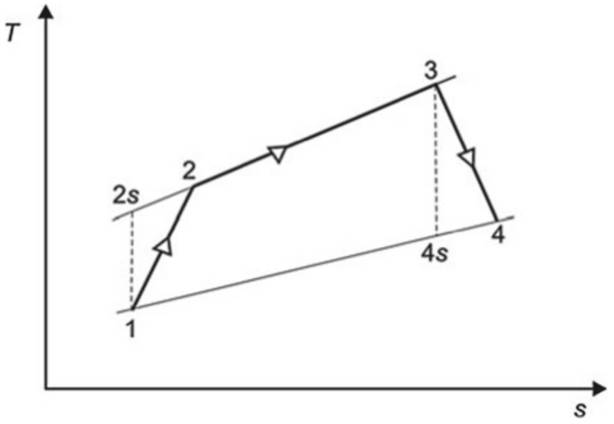
$$p_1 = 1\,\text{bar}$$

$$T_1 = 27\,^\circ\text{C} = (27 + 273)\text{K} = 300\,\text{K}$$

Pressure ratio:

$$r_p = \frac{p_2}{p_1} = 6$$

Maximum temperature:



$$\begin{aligned}T_3 &= 800\,^\circ\text{C}\\ &= (800+273)\,\text{K}\\ &= 1073\,\text{K}\end{aligned}$$

Compressor efficiency:

$$\eta_C = 85\% = 0.85$$

Turbine efficiency:

$$\eta_T = 88\% = 0.88$$

For isentropic process 1–2*s*,

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{T_{2s}}{300} = (6)^{\frac{1.4-1}{1.4}} = (6)^{0.2857} = 1.668$$

or

$$T_{2s} = 1.668 \times 300 = 500.4\,\text{K}$$

$$\eta_c = \frac{T_{2s}-T_1}{T_2-T_1}$$

$$0.85 = \frac{500.4-300}{T_2-300} = \frac{200.4}{T_2-300}$$

or

$$T_2 - 300 = \frac{200.4}{0.85} = 235.76$$

or

$$T_2 = 235.76 + 300 = 535.76\,\text{K}$$

For isentropic process 3–4*s*,

$$\frac{T_3}{T_{4s}} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(r_p\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{1073}{T_{4s}} = (6)^{\frac{1.4-1}{1.4}} = (6)^{0.2857} = 1.668$$

or

$$T_{4s} = \frac{1073}{1.668} = 643.28\,\text{K}$$

$$\eta_T = \frac{T_3-T_4}{T_3-T_{4s}}$$

$$0.88 = \frac{1073-T_4}{1073-643.28} = \frac{1073-T_4}{429.72}$$

or

$$0.88 \times 429.72 = 1073 - T_4$$

or

$$378.15 = 1073 - T_4$$

or

$$T_4 = 1073 - 378.15 = 694.85\,\mathrm{K}$$

Specific compressor input work:

$$w_c = c_p(T_2 - T_1) = 1.005(535.76 - 300) \\ = 236.93\,\mathrm{kJ/kg}$$

Specific turbine output work:

$$w_T = c_p(T_3 - T_4) = 1.005(1073 - 694.85) \\ = 380.04\,\mathrm{kJ/kg}$$

Specific heat supplied:

$$q_{2-3} = c_p(T_3 - T_2) = 1.005(1073 - 535.76) \\ = 539.92\,\mathrm{kJ/kg}$$

(i) Net work: $w_{\text{net}} = w_T - w_C = 380.04 - 236.93 = 143.11\,\mathrm{kJ/kg}$
(ii) Cycle efficiency: $\eta = \frac{w_{not}}{q_{2-3}} = \frac{143.11}{539.92} = 02650 = 26.50\%$
(iii) Work ratio: $WR = \frac{w_{net}}{w_T} = \frac{143.11}{380.04} = 0.3765$

Q.8. (*a*) Explain Intensive and Extensive Properties.

Ans. Intensive Property: The properties that are independent on the mass of the system called intensive properties. For examples: Temperature, pressure, density, viscosity, specific heat and thermal conductivity.

Extensive Property: The properties that depend on the mass of the system are called extensive properties. For examples: Energy, enthalpy, entropy, momentum and volume.

(*b*) Write steady flow energy equation for a open system and write the same for throttling process.

Ans. The steady flow energy equation (SFEE) for unit mass flow rate,

$$h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + gz_2 + w_{1-2}$$

where

h_1 = specific enthalpy at inlet in J/kg.
V_1 = velocity at inlet in m/s
Z_1 = datum head at inlet in m
q_{1-2} = specific heat transfer in J/kg.
and
h_2 = specific enthalpy at outlet in J/kg
V_2= Velocity at Outlet in M/s
z_2 = datum head at outlet in m
w_{1-2} = specific work in J/kg.

Throttle Process: Throttling process can be analysed with following assumption:

(i) Work interaction: $w_{1\text{-}2} = 0$
(ii) Heat interaction: $q_{1\text{-}2} = 0$
(iii) Change in kinetic and potential energies are neglected.

Above SFEE becomes for throttling process,

$$\boldsymbol{h}_1 = \boldsymbol{h}_2$$

It means that the enthalpy of fluid remains constant during the throttling process, i.e., enthalpy of fluid before throttling is equal to the enthalpy of the fluid after throttling.

(*c*) Explain second law efficiency.

Ans. Second Law Efficiency: η_{II}. It is defined as the ratio of the actual work output to the maximum work output of the work-producing devices.

Mathematically,

Second law efficiency:

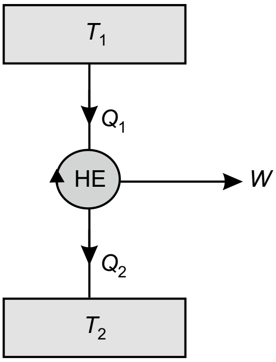
$$\eta_{\text{II}} = \frac{\text{Actual work} : W}{\text{Maximum work} : W_{rev}}$$

$$\eta_{II} = \frac{W}{W_{rev}} \text{ for work - producing devices}$$

$$\therefore\ \eta_{II} = \frac{W}{W_{rev}} \times \frac{Q_1}{Q_1}$$

$$\eta_{II} = \frac{W}{Q_1 \times \frac{W_{rev}}{Q_1}}$$

$$\eta_{II} = \frac{\eta_{th}}{\eta_{rev}} = \frac{\eta_1}{\eta_{Carnot}}$$



where

$$\eta_{th} = \eta_{II} = \frac{W}{Q_1}, \text{actual thermal efficiency}$$

and

$$\eta_{rev} = \eta_{Carnot} = \frac{W_{rev}}{Q_1} \text{thermal efficiency of a reversible heat engine or Carnot efficiency.}$$

The second law efficiency is also defined as the ratio of actual thermal efficiency to the reversible (Carnot) efficiency under the same condition.

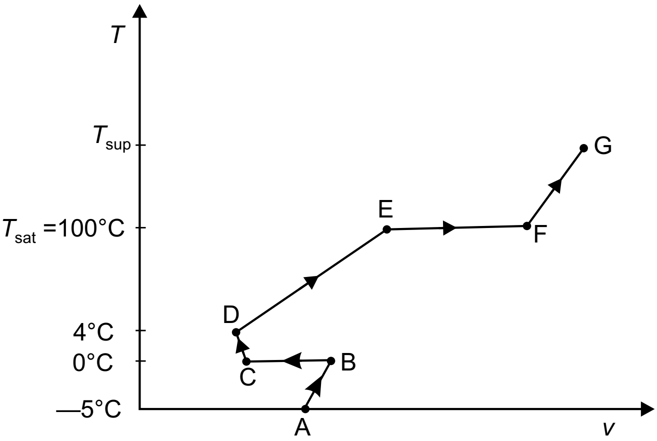
(*d*) What do you understand by saturation states? Show different saturation states coming across while heating ice at −5 °C to super heated steam.

Ans. A saturation state is the point where a phase change begins or ends. For example, saturation liquid state and saturated vapour state.

Formation of superheated steam from −5 °C of ice.

Let us consider 1 kg of ice of −5 °C below the freezing point. The initial state of the substance has been denoted by state A.

Process A–B: When heat is supplied, temperature of ice rises from −5 °C to 0 °C with increase in volume. The heat supplied during process *A-B* is called reversible heat.



Process *B–C*: At state B is ice at 0 °C. On heating further beyond state B, at constant 0 °C temperature, the phase changes from ice to water with decrease in volume.

Process *C–D*: At state C is water at 0 °C. On heating further beyond state C, the volume of water decreases when the temperature rise 0 °C to 4 °C. The maximum density of water at 4 °C is 1000 kg/m^3.

Process *D–E*: Further addition of heat at state D, temperature of water rises from 4 to 100 °C with increase in volume.

Saturation Liquid Point: The state of liquid at which a vapor just begins to form is called saturation liquid point. This point is denoted by point E on *T–v* plane.

Process *E–F*: As more heat is added at the saturation liquid state E, boiling of water at constant temperature with increase in volume. At state F, all the water has vaporized the state of steam at this point is called dry-saturated steam.

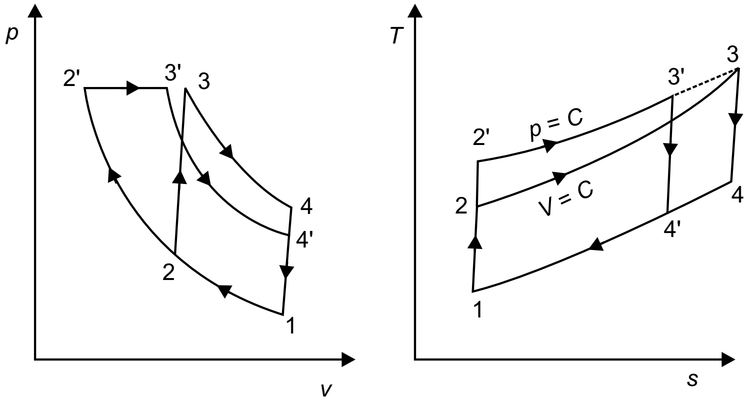
Saturation Vapor State: The state of vapour at which condensation just begins on loss of heat of the vapor is called saturation vapour state. It is denoted by point F on *T–v* plane.

Process *F–G*: As more heat is added at the saturation vapour state F, temperature of steam rises at given constant pressure with increase in volume. The steam obtained by heating the dry-saturated steam is called superheated steam. The heating process of the dry-saturated steam from the state F to any state say G, is called superheating.

(*e*) Using *p–v* and *T–s* diagram show that diesel cycle is more efficiency than that of Otto cycle for same maximum cycle pressure and heat input.

Ans.

Same maximum cycle pressure and heat input:



The Otto and diesel cycles are shown in *p–v* and *T–s* diagram of Fig. according to given condition of the same maximum cycle pressure and heat input. These cycles are shown in Fig. as.

Otto Cycle: 1–2–3–4–1

Diesel Cycle: 1–2′–3′–4′–1

The thermal efficiency:

$$\eta = 1 - \frac{\text{heat rejection}}{\text{heat input}}$$

As the heat input is constant. The thermal efficiency of the cycle is dependent only on the heat rejection, lower the heat rejection, higher the thermal efficiency. It is cleared from *T–s* diagram, heat rejection is higher in the Otto cycle and lower in the diesel cycle.

i.e.,

$$(Q_{4-1})_{\text{Otto}} > (Q_{4-1})_{\text{Diesel}}$$

Thus, the thermal efficiency of the diesel cycle is more than Otto cycle at given condition of the same maximum cycle pressure and heat input.

i.e.,

$$\eta_{\text{Diesel}} > \eta_{\text{Otto}}$$

Q.9. Derive equation for *pdV* work for a polytropic process and for constant pressure process. Show the process on *p–v* diagram.

Ans. Polytropic process:

$$pV^n = C$$

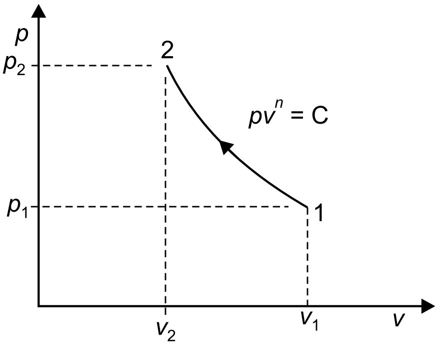
Work done:

$$\delta W = pdV$$

Integrating between states 1 and 2 gives,

$$\int_1^2 \delta W = \int_1^2 pdV$$

$$W_{1-2} = \int_1^2 pdV$$



We know that $pV^n = C$ for polytropic process.

or

$$p = \frac{C}{V^n}$$

$$\therefore\ W_{1-2} = \int_1^2 \frac{C}{V^n} dV = C\int_1^2 V^{-n} dV$$

$$= C\left[\frac{V^{-n+1}}{-n+1}\right]_1^2 = C\left[\frac{V_2^{-n+1} - V_1^{-n+1}}{1-n}\right]$$

$$= \frac{CV_2^{-n+1} - CV_1^{-n+1}}{1-n}$$

$$= \frac{p_2V_2^nV_2^{-n+1} - p_1V_1^nV_1^{-n+1}}{1-n} \qquad \because p_1V_1^n$$

$$= p_2V_2^n = C$$

$$= \frac{p_2V_2 - p_1V_1}{1-n} = \frac{p_1V_1 - p_2V_2}{n-1}$$

$$= \frac{R(T_1 - T_2)}{n-1}$$

Constant Pressure Process:

$$p = C$$

Work done:

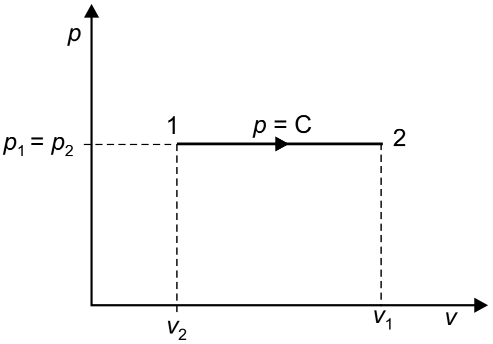
$$\delta W = pdV$$

Integrating between state 1 and 2 gives 2

$$\int_1^2 \delta W = \int_1^2 pdV$$

$$W_{1-2} = p\int_1^2 dV = p[V]_1^2 = p(V_2 - V_1)$$

$$= \boldsymbol{R}(\boldsymbol{T}_2 - \boldsymbol{T}_1)$$



Q.10. Air flow with steady state of a flow rate of 0.5 kg/s through an air compressor. Inlet velocity, pressure and specific volume are

7 m/s, 1 bar and 0.9 m^3/kg, respectively, and these parameters at outlet for the compressor are 4 m/s, 7 bar and 0.15 m^3/kg, respectively. The difference in internal energy between outlet and inlet is 90 kJ/kg. The cooling arrangement made in the compressor absorbs 60 W of energy. Calculate power required to drive the compressor and cross sectional areas at inlet and outlet.

Ans. Given data:

Mass flow rate:

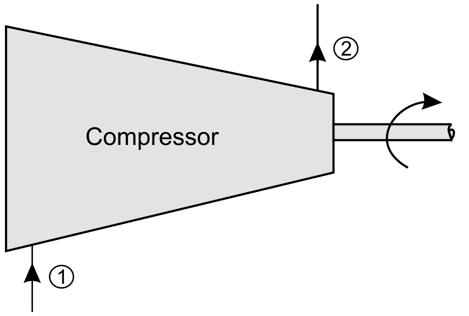
$$m = 0.5 \text{kg/s}$$

At inlet state 1,

$$\begin{aligned} V_1 &= 7\,\text{m/s} \\ p_1 &= 1\,\text{bar} = 100\text{kPa} \\ v_1 &= 0.9\,\text{m}^3/\text{kg} \end{aligned}$$

At outlet state 2,

$$\begin{aligned} V_2 &= 4\,\text{m/s} \\ p_2 &= 7\,\text{bar} = 700\text{kPa} \\ v_2 &= 0.15\,\text{m}^3/\text{kg} \\ u_2 - u_1 &= 90\,\text{kJ/kg} \\ Q &= -60\,\text{W} = -0.06\,\text{kW} \end{aligned}$$



Change in specific enthalpy:

$$\begin{aligned} &h_2 - h_1 \\ &\quad = (u_2 - u_1) + p_2 v_2 - p_1 v_1 \\ &\quad = 90 + 700 \times 0.15 - 100 \times 0.9 \\ &\quad = 105\,\text{kJ/kg} \end{aligned}$$

Applying the steady flow energy equation, we have

$$\begin{aligned} & m\left(h_1 + \frac{V_1^2}{2} + gz_1\right) + Q \\ & \quad = m\left(h_2 + \frac{V_2^2}{2} + gz_2\right) + W \end{aligned}$$

Neglecting change in potential energy

$$\therefore\ m\left(h_1 + \frac{V_1^2}{2}\right) + Q = m\left(h_2 + \frac{V_2^2}{2}\right) + W$$

or

$$m(h_2 - h_1) + \frac{m}{2}(V_2^2 - V_1^2) + W = Q$$

Units balancing,

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

Here,

m is in kg/s
h_1 and h_2 are in kJ/kg
V_1 and V_2 are in m/s
W and Q are in kW

$$\begin{aligned} \therefore\ 0.5(105) + \frac{0.5}{2000}\left[(4)^2 + (7)^2\right] + W &= -0.06 \\ 52.5 - 0.00825 + W &= -0.06 \\ W &= -52.55\,\text{kW} \end{aligned}$$

or

The −ve sign indicated that the work is done on the system.

$$\begin{aligned} m &= \frac{A_1 V_1}{v_1} \\ 0.5 &= \frac{A_1 \times 7}{0.9} \end{aligned}$$

or

$$A_1 = 0.06428\,\text{m}^2 = 642.85\,\text{cm}^2$$

also,

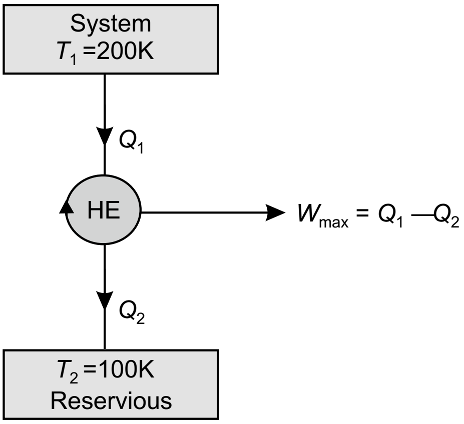
$$m = \frac{A_2 V_2}{v_2}$$

$$0.5 = \frac{A_2 \times 4}{0.15}$$

$$A_2 = 0.01875\,\mathrm{m}^2 = 187.5\,\mathrm{cm}^2$$

Q.11. A system having heat capacity at constant volume, C_v = AT^1, where a = 0.043 J/K^3, is originally at 200 K and a thermal reservoir at 100 K. Determine maximum amount of work that can be recovered as the system is cooled down to the reservoir's temperature.

Ans. Given data:



Heat capacity at constant volume,

$$C_v = AT^2$$

where

$$A = 0.043 \;\; \mathrm{J/K^3}$$

Temperature of system:

$$T_1 = 200\,\mathrm{K}$$

Temperature of thermal reservoir:

$$T_2 = 100 \;\; \mathrm{K}$$

Heat transfer from the system:

$$Q_1 = \int_{T_2}^{T_1} C_v dT$$

$$= \int_{100}^{200} AT^2\, dT$$

$$= A\left[\frac{T^3}{3}\right]_{200}^{100}$$

$$= \frac{0.043}{3}\left[(200)^3 - (100)^3\right] = 100333.33 \;\; \mathrm{J}$$

Entropy change of system,

$$(\Delta S)_{\text{system}} = \int_{T_1}^{T_2} C_v \frac{dT}{T}$$

$$= \int_{T_1}^{T_2} AT^2 \frac{dT}{T}$$

$$= \int_{T_1}^{T_2} AT\, dT$$

$$= A \int_{T_1}^{T_2} T\, dT$$

$$= A\left[\frac{T^2}{2}\right]_{T_1}^{T_2}$$

$$= A\left[\frac{T_2^2 - T_1^2}{2}\right]$$

$$= 0.043\left[\frac{(100)^2 - (200)^2}{2}\right]\Bigg(= -645 \;\; \mathrm{J/K}$$

Entropy change of reservoir (sink),

$$(\Delta S)_{\text{sink}} = \frac{Q_2}{T_2} = \frac{Q_2}{100}$$

Entropy change of universe,

$$(\Delta S)_{\text{uni}} = (\Delta S)_{\text{system}} + (\Delta S)_{\text{sink}}$$

$$= -645 + \frac{Q_2}{100}$$

According to the increase of entropy principle for reversible heat engine,

$$(\Delta S)_{\text{uni}} = 0$$
$$-645 + \frac{Q_2}{100} = 0$$

or

$$\frac{Q_2}{100} = 645$$

or

$$Q_2 = 645 \times 100 = 64500 \text{ J}$$

Thus, the maximum amount of work that can be recovered,

$$\begin{aligned} W_{\max} &= Q_1 - Q_2 \\ &= 100333.33 - 64500 = 35833.33\,\text{J} \\ &= 35.83\,\text{kJ} \end{aligned}$$

Q.12. (*a*) Define Microscope and Macroscopic Thermodynamics.

Ans. Microscope thermodynamics. When the study is performed at molecular level of the matter, it is called microscopic thermodynamics. A certain quantity of matter is chosen to study the behaviour of individual molecule. Then by adopting statistical approach, collecting molecular activity is analyzed.

Macroscopic Thermodynamics. When the study is performed on the matter or whole system, it is called macroscopic thermodynamics. A certain quantity of matter is chosen to study the behaviour of all molecules.

(*b*) Briefly explain application of Kelvin Planck's and Clausius statements of 2nd law.

Ans. Kelvin Planck's statements. It is impossible to construct a heat engine that work in a cycle and converts net heat supplied into an equivalent amount of work.

Application: Heat engines.

Clausius statement. It is impossible to construct a device that operates in a cycle and produce no effect other than the heat transfer of heat from a low temperature body to a high temperature body without external aid of work. Application: Refrigerator and heat pump.

(*c*) Define entropy generation.

Ans. The entropy change of a closed system during an irreversible process is greater than $\int_1^2 \frac{\delta Q}{T}$, which represents the entropy transfer with heat. That is, some entropy is generated or created during an irreversible process, and this generation is due to the presence of irreversibility. The entropy generated during a process is called entropy generation and is denoted by S_{gen}.

Entropy Change:

$$S_2 - S_1 = \int_1^2 \frac{\delta Q}{T} + S_{\text{gen}}$$

where S_{gen} is the entropy generation and its value always a + ve or zero, i.e., + ve value for an irreversible process and zero for a reversible process.

(*d*) Define availability and show that maximum work achievable decrease with increase in ambient temperature.

Ans. Availability: The availability of the system is defined as the maximum useful work obtainable from the system when the system approaches to atmospheric condition. It is denoted by A.

Let

Q_1 = head supplied to the system from heat source at temperature T_1.

Q_2 = heat rejected to the sink at atmospheric or ambient temperature to

Efficiency of reversible heat engine:

$$\eta_{\text{rev}} = \frac{W_{\max}}{Q_1}$$

also

$$\eta_{\text{rev}} = 1 - \frac{T_0}{T_1}$$

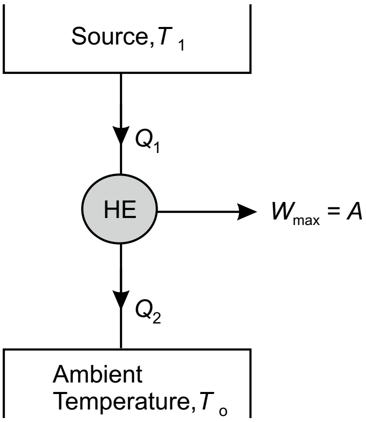
$$\therefore \quad \frac{W_{\max}}{Q_1} = 1 - \frac{T_0}{T_1}$$

or

$$W_{\max} = Q_1\left(1 - \frac{T_0}{T_1}\right)$$

or

$$A = W_{\max} = Q_1\left(1 - \frac{T_0}{T_1}\right)$$



Hence, maximum work achievable decrease with increase in ambient temperature.

(*e*) What do you mean by exact and inexact differentials of properties?

Ans. The parameters are depend only on the initial and states of a system, is called point function. Point functions have exact differentials because they depend on the end states only and are independent of the path followed during a process. For example, volume temperature, pressure, energy, etc. are point functions.

The parameters depend both on the path followed during the process and end states of a system, which is called path functions. Path functions have inexact differentials because they depend on the path followed during a process. For example, heat and work are path functions.

(*f*) Draw constant pressure and constant temperature lines in entropy-enthalpy diagram of water and explain sensible, latent and superheating.

Ans. Enthalpy-entropy diagram is also called mollier diagram.

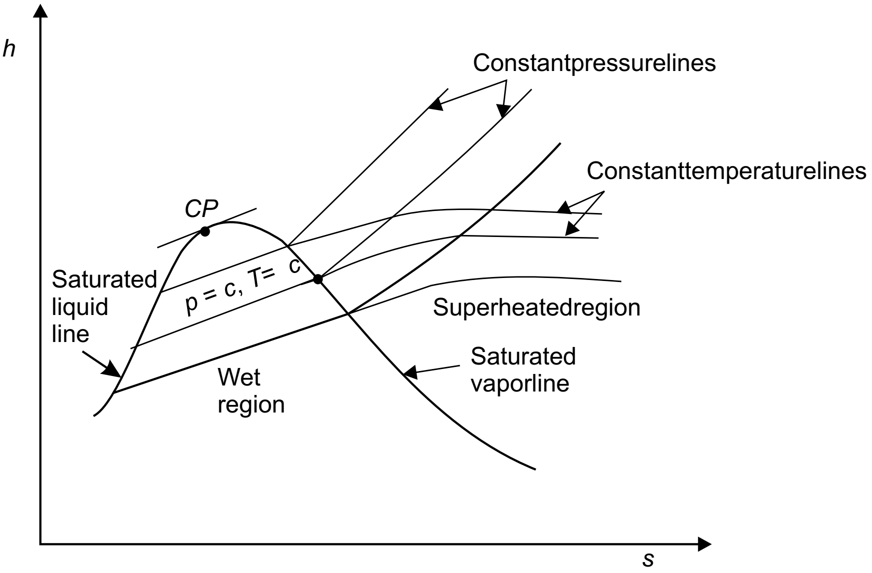
The following observation can be made from h–s diagram:

1. Constant Pressure Lines and Constant Temperature Lines Coincide in the Wet Region. The Pressure Line Diverge from One Another Due to the Increase in Saturation Temperature with Increase in Pressure.
2. In the region of superheated vapor, the isobaric lines, have a positive slope and turn upwards and the constant temperature lines tend towards a horizontal straight line. This may be attributed to the fact that at high temperature, the behaviour of superheated vapor approaches as on ideal gas, i.e., $h = f(T)$.

Sensible Heat: The amount of heat required to raise the temperature of the substance at constant pressure when the phase change does not occur is called sensible heat.

Latent Heat: The amount of heat required to change the phase from liquid to vapour (or vapour to liquid) at constant temperature and pressure is called latent heat.

Super Heating: The heating process of dry-saturated steam above the saturated temperature, is called superheating.



(*g*) **Explain the limitation of Otto cycle and what is the solution to overcome it?**

Ans. The limitation of Otto cycle is upper limit of its compression ratio.

If compression ratio increases above a certain limit, the efficiency decreases due to detonation (also called knocking). Detonation is abnormal combustion due to end charge auto-ignites before the flame reaches it.

This limitation may overcome by;

(i) Using high octane number fuel.
(ii) Increasing engine rpm.
(iii) Retarding spark.
(iv) Reducing pressure in inlet manifold by throttling.

(*h*) **Why Carnot cycle is not being used in practice, although it gives maximum efficiency? Draw the *T–s* and *p–v* diagram for Carnot cycle.**

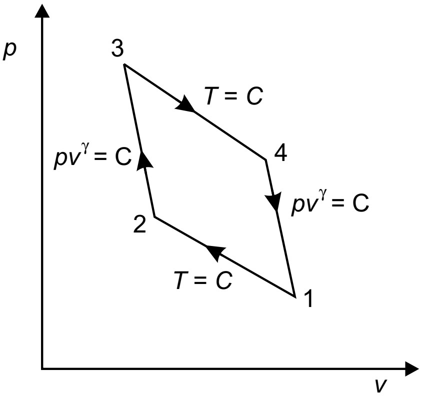
Ans. The Carnot engine is a hypothetical device and it cannot be practically possible due to following reasons:

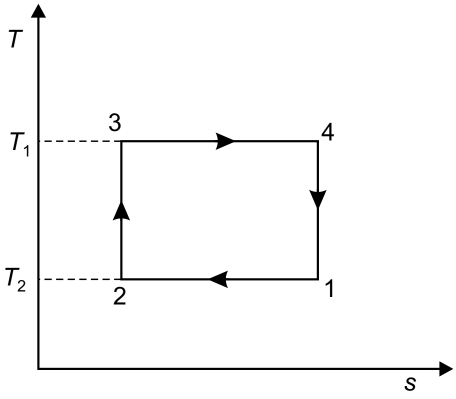
1. All the four processes used in Cannot cycle have to be reversible. The reversible processes are possible only if the processes have no internal friction among the fluid particles and no mechanical friction between the piston and cylinder walls.
2. Heat addition and rejection at constant temperature is impossible.
3. Frequent change of cylinder head (i.e., insulating head for adiabatic process and diathermic head for isothermal process) is not possible.

Or

The isothermal process can be achieved only if the piston moves very slowly to allow heat transfer so that the temperature remains constant. The adiabatic process can be achieved only if the piston moves very fast in order to approach reversible process, i.e., heat transfer is negligible due to very short period of time.

Thus, it is impossible to maintain the variations in the speed of the piston during the processes of a cycle.

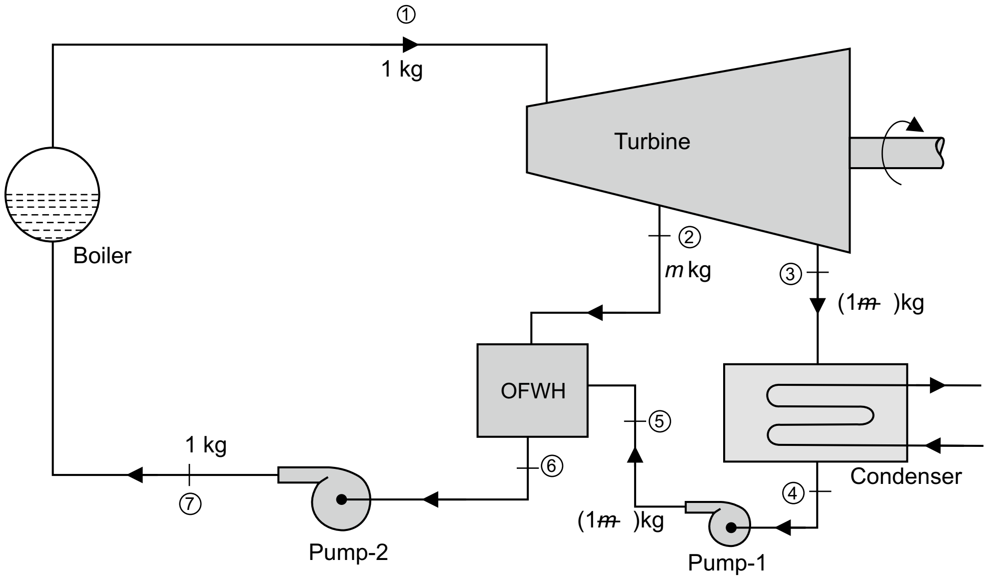




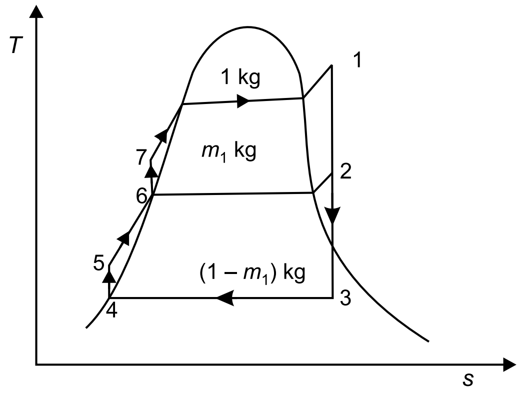
4. The isothermal and adiabatic lines in *p*–*v* diagram are largely extended both in the horizontal and vertical directions. The cylinder involves high pressure and large displacement volume, and thesize of the engine becomes heavy and bulky. The Carnot cycle is an impractical device or hypothetical device. Now the question is why we study Carnot cycle? The answer is that the Carnot cycle is taken as a standard or reference against which the performance of any practical heat engine is compared,

(*i*) Explain regenerative feed heating.

Ans. A schematic representation of the main component of regenerator feed heating is shown in Fig. The water entering the boiler is called feed water, and the device used to mix the extracted steam and the condenser water is called a generator, or a feed water heater. When the condenser water is mixed directly with the extracting steam, it is done in an open feed water heater as shown in Fig.



Let 1 kg of steam supplied to the turbine or leaving from the boiler $m_1$kg is extracted a state 2 for feed water heater. $(1 - m_1)$ kg is expanded completely in the remaining part of the turbine upto state 3.



The heat and work interaction of a regenerative cycle with one feed water heater can be expressed per unit mass of steam flowing through the boiler as follows heat supplied in the boiler:

$$q_{7-1} = h_1 - h_7$$

Net pumps input work:

$$\begin{aligned} w_p &= w_{4-5} + w_{6-7} \\ &= (1 - m_1)(h_5 - h_4) + (h_7 - h_6) \end{aligned}$$

Turbine output work:

$$w_{1-3} = (h_1 - h_2) + (1 - m_1)(h_2 - h_3)$$

Net work output:

$$w_{\text{net}} = w_{1-3} - w_p$$

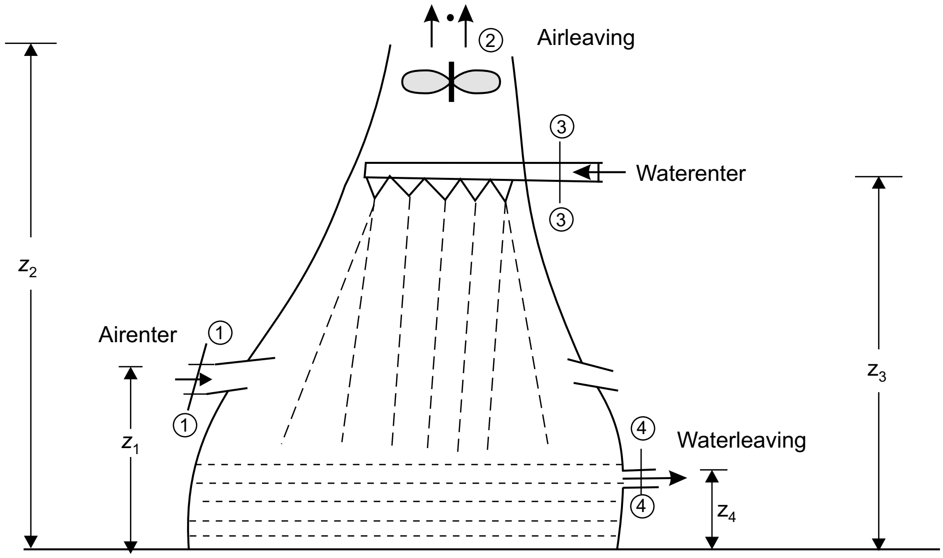
The thermal efficiency of the generative cycle,

$$\eta_{\text{th}} = \frac{\text{net work output}}{\text{heat supplied}} = \frac{w_{\text{net}}}{q_{7-1}}$$

Q.13. In water cooling tower, air enters at a height of 1 m above the ground and leaves at height of 8 m. The inlet and outlet velocities are 20 m/s and 30 m/s, respectively. Water enters at a height of 7 m and leaves at a height of 0.8 m. velocity of water at entry and exit is 3 m/s and 1 m/s, respectively. Water temperature is 40 °C and 30 C at entry and exit, respectively. Air temperature is 30 °C and 38 °C at the entry and exit, respectively. The cooling tower is well insulated and a fan of 3 kW drives the air through the tower. Find the amount of air per second required for 1 kg/s of water flow. The value of c_p of air and water are 1.005 and 4.187 kJ/kg K, respectively.

Ans. Given data:

For air		**For water**	
At inlet	**At leaving**	**At inlet**	**At leaving**
$z_1 = 1\,\text{m}$	$z_2 = 8\ \text{m}$	$z_3 = 7\ \text{m}$	$z_4 = 0.8\ \text{m}$
$V_1 = 20\ \text{m/s}$	$V_2 = 30\ \text{m/s}$	$V_3 = 3\ \text{m/s}$	$V_4 = 1\ \text{m/s}$
$T_1 = 30°\text{C}$	$T_2 = 38°\text{C}$	$T_3 = 40°C$	$T_4 = 30°\text{C}$
$c_{pa} = 1.005\ \text{kJ/kgK}$		$c_{pw} = 4.187\,\text{kJ/kgK}$	
$m_a = ?$		$m_w = 1\ \text{kg/s}$	



Power required to drive the fan:

$$W = -3\,\mathrm{kW}$$

Assumption: Evaporation of water is neglected.

Applying the steady flow energy equation,

$$m_a\left[h_1 + \frac{V_1^2}{2} + gz_1\right] + m_w\left[h_3 + \frac{V_3^2}{2} + gz_3\right] + Q$$
$$= m_a\left[h_2 + \frac{V_2^2}{2} + gz_2\right] + m_w\left[h_4 + \frac{V_4^2}{2} + gz_4\right] + W$$

where Q = 0, insulated cooling tower

$$m_a\left[(h_1 + h_2) + \frac{V_1^2 - V_2^2}{2} + g(z_1 - z_2)\right]$$
$$= m_w\left[(h_4 - h_3) + \frac{V_4^2 - V_3^2}{2} + g(z_4 - z_3)\right]$$
$$+ W\, m_a\left[c_{pa}(T_1 - T_2) + \frac{V_1^2 - V_2^2}{2000} + \frac{g}{1000}(z_1 - z_2)\right]$$
$$= m_w\left[c_{pa}(T_4 - T_3) + \frac{V_1^2}{2000} + \frac{g}{1000}(z_4 - z_3)\right]$$
$$+ W\, m_a\left[1.005(30 - 38) + \frac{(20)^2 - (30)^2}{2000} + \frac{9.81}{1000}(1 - 8)\right]\Bigg($$
$$= 1\left[4.187(30 - 40) + \frac{(1)^2 - (3)^2}{2000} + \frac{9.81}{1000}(0.8 - 7)\right] - 3$$

$$m_a[-8.04 - 0.25 - 0.0686] = -41.87 - 0.004 - 0.0608 - 3$$
$$-8.358 m_a = -44.934$$

or

$$m_a = \frac{44.934}{8.359}$$
$$m_a = 5.37\mathrm{kg/s}$$

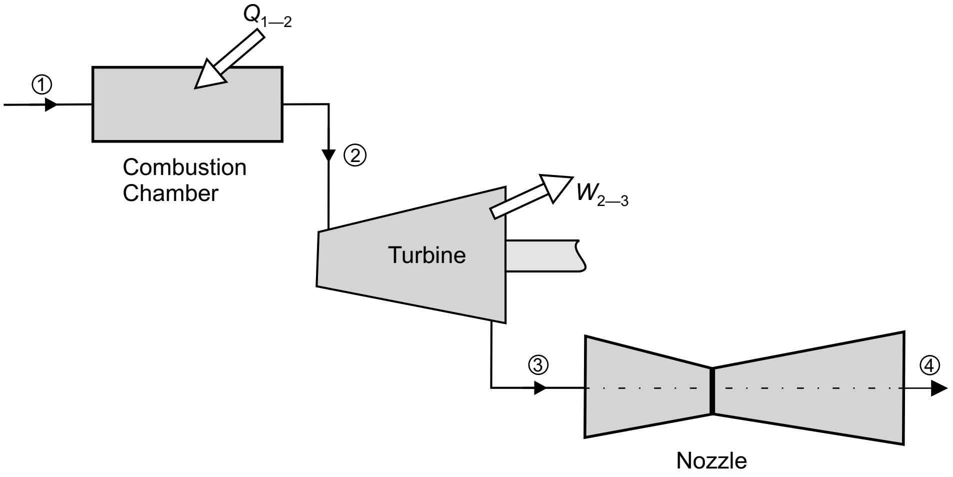
Q.14. In a turbojet engine, air at a velocity of 30 m/s and a temperature of 45 °C is fed to combustion chamber, where temperature is raised to 800 °C. It then enters in a turbine with the same velocity of 30 m/s and expands till temperature falls to 650 °C. On leavings the turbine, the air is taken at a velocity of 60 m/s to a nozzle, where it expands until temperature falls to 400 °C. If the air flows at a rate of 5 kg/s. Calculate:

(a) **Rate of heat transfer to air in combustion chamber.**
(b) **Power output from turbine assuming no heat loss.**
(c) **Velocity at exit from nozzle assuming no heat loss. (Take c_p is air as 1.005 kJ/kg K, $h = c_p T$, where T is temperature)**

Ans. Given data:

Combustion chamber:

At inlet; Velocity: $V_1 = 30\text{m/s}$.
Temperature: $T_1 = 45\,^{\circ}\text{C}$.
At exit; Velocity: $V_2 = 30\text{m/s}$.
Temperature: $T_2 = 800\,^{\circ}\text{C}$.



Combustion chamber exit = Turbine inlet
At turbine exit,

$$T_3 = 650^{\circ}\text{C}$$

$$V_3 = 60\ \text{m/s}$$

Turbine exit = Nozzle inlet
At nozzle exit,

$$T_4 = 400^{\circ}\ \text{C}$$

Mass flow rate of air:

$$m = 5\ \text{kg/s}$$

(a) Rate of heat transfer to air in combustion chamber: $Q_{1\text{-}2}$

Applying the steady flow energy equation

$$m\left[h_1 + \frac{V_1^2}{2} + gz_1\right] + Q_{1-2} = m\left[h_2 + \frac{V_2^2}{2} + gz_2\right] + W_{1-2}$$

where

$$V_1 = V_2$$

$$z_1 = z_2$$

$$W_{1-2} = 0$$

$$\therefore\ mh_1 + Q_{1-2} = mh_2$$

or

$$\begin{aligned} Q_{1-2} &= m[h_2 - h_1] \\ &= mc_p[T_2 - T_1] \\ &= 5 \times 1.005(800 - 45) = 3793.87\,\text{kJ/s} \end{aligned}$$

(b) Power output from turbine: $W_{2\text{–}3}$

Applying the steady flow energy equation

$$m\left[h_2 + \frac{V_2^2}{2} + gz_2\right] + Q_{2-3} = m\left[h_3 + \frac{V_3^2}{2} + gz_3\right] + W_{2-3}$$

where

$$z_2 = z_3, \quad Q_{2-3} = 0$$

$$\therefore\ m\left[h_2 + \frac{V_2^2}{2}\right] = m\left[h_3 + \frac{V_3^2}{2}\right] + W_{2-3}$$

or

$$m\left[c_pT_2 + \frac{V_2^2}{2000}\right] = m\left[c_pT_3 + \frac{V_3^2}{2000}\right] + W_{2-3}$$

or

$$\begin{aligned} W_{2-3} &= mc_p(T_2 - T_3) + \frac{m(V_2^2 - V_3^2)}{2000} \\ &= 5 \times 1.005(800 - 600) + \frac{5}{2000}\left[(30)^2 - (60)^2\right] \\ &= 753.75 - 6.75 = 747\,\text{kW} \end{aligned}$$

(c) Velocity at exit from nozzle: V_4

Applying the steady flow energy equation

$$m\left[h_3 + \frac{V_3^2}{2} + gz_3\right] + Q_{3-4} = m\left[h_4 + \frac{V_4^2}{2} + gz_4\right] + W_{3-4}$$

where

$$z_3 = z_4$$

$$Q_{3-4} = 0$$

$$W_{3-4} = 0$$

$$\therefore\ mh_3 + m\frac{V_3^2}{2} = mh_4 + m\frac{V_4^2}{2}$$

or

$$h_3 + \frac{V_3^2}{2} = h_4 + \frac{V_4^2}{2}$$

or

$$c_pT_3 + \frac{V_3^2}{2000} = c_pT_4 + \frac{V_4^2}{2000}$$

$$\frac{V_4^2}{2000} = c_p(T_3 - T_4) + \frac{V_3^2}{2000}$$

or

$$V_4^2 = 2000c_p(T_3 - T_4) + V_3^2$$

or

$$\begin{aligned} V_4 &= \sqrt{2000c_p(T_3 - T_4) + V_3^2} \\ &= \sqrt{2000 \times 1.005(650 - 400) + (60)^2} \\ &= \sqrt{502500 - 3600} = \sqrt{498900} = 706.32\,\text{m/s} \end{aligned}$$

Q.15. A heat pump working on the Carnot cycle takes heat from a reservoir at 5 °C and delivers heat to a reservoir at 60 °C. The heat pump is driven by a reversible heat engine which takes heat from a reservoir at 840 °C and rejects heat to a reservoir at 60 °C. The reversible heat engine also drives a machine that absorbs 30 kW. If the heat pump extracts 17 kJ/s from the 5 °C reservoir, determine:

(a) Rate of heat supply from the 840 °C source.

(b) The rate of heat rejection to the 60 °C sink.

Ans. Given data:

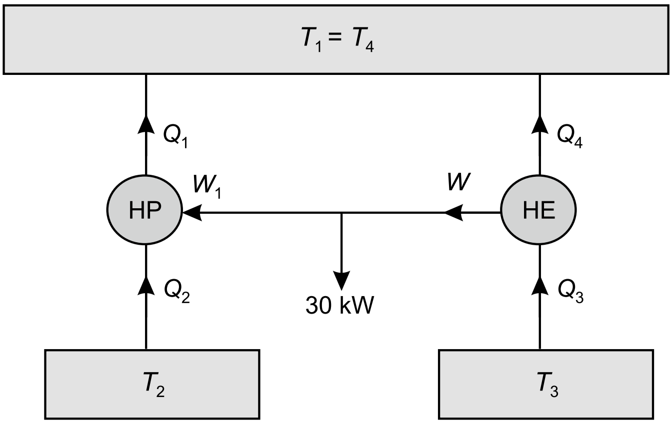
$$T_2 = 5^\circ C = 5 + 273 = 278\ \text{K}$$

$$T_1 = 60^\circ C = 60 + 273 = 333\ \text{K}$$

$$T_3 = 840^\circ C = 840 + 273 = 1113\ \text{K}$$

$$W = W_1 + 30$$

$$Q_2 = 17\ \text{kJ/s} = 17\text{kW}$$



For heat pump:

$$(\text{COP})_{\text{HP}} = \frac{T_1}{T_1 - T_2} = \frac{333}{333 - 278} = 6.05$$

also

$$(\text{COP})_{\text{HP}} = \frac{Q_1}{Q_1 - Q_2}$$

$$\therefore\ 6.05 = \frac{Q_1}{Q_1 - Q_2}$$

$$\begin{aligned} 6.05(Q_1 - Q_2) &= Q_1 \\ 6.05Q_1 - 6.05Q_2 &= Q_1 \\ 6.05Q_1 - Q_1 &= 5.05Q_2 \\ 5.05Q_1 &= 5.05 \times 17 \\ 5.05Q_1 &= 102.85 \end{aligned}$$

or

$$Q_1 = \frac{102.85}{5.05} = 20.36\,\text{kW}$$

$$\begin{aligned} W_1 &= Q_1 - Q_2 \\ &= 20.36 - 17 = 3.36\,\text{kW} \end{aligned}$$

(a) Rate of heat supply from 840 °C source: Q_3

For reversible heat engine

$$\begin{aligned} W &= W_1 + 30 \\ &= 3.36 + 30 = 33.36\,\text{kW} \end{aligned}$$

$$\begin{aligned} \eta &= 1 - \frac{T_4}{T_3} \\ &= 1 - \frac{T_1}{T_3} = 1 - \frac{333}{1113} = 1 - 0.2991 = 0.7009 \end{aligned}$$

also

$$\eta = \frac{W}{Q_3}$$

$$\therefore\ 0.7009 = \frac{W}{Q_3}$$

$$0.7009 = \frac{33.36}{Q_3}$$

or

$$Q_3 = \frac{33.36}{0.7009} = 47.59\,\text{kW}$$

(b) Rate of heat rejection to the 60 °C sink: 0_4

$$\begin{aligned} Q_4 &= Q_3 - W \\ &= 47.59 - 33.36 = 14.23\,\text{kW} \end{aligned}$$

Q.16. A heat engine operating between two reservoirs at 1000 K and 300 K is used to drive a heat pump which extracts heat from the reservoir at 300 K at a rate twice that at which the engine rejects heat to it if the efficiency of the engine is 40% of the maximum possible and COP of the heat pump is 50% of the maximum possible. What is the temperature of reservoir to which the heat pump rejects the heat?

Ans: Given data:

$$\begin{aligned} T_1 &= 1000\ \text{K} \\ T_2 &= 300\ \text{K} = T_3 \\ Q_3 &= 2Q_2 \\ \eta &= 0.40\eta_{\max} \\ (\text{COP})_{\text{HP}} &= 0.50(\text{COP}_{\max})_{\text{HP}} \\ Q_1 &= 50\ \text{kW} \end{aligned}$$

∴ Actual efficiency:

$$\eta = 0.41\,\eta_{\max} = 0.40 \times 0.7$$
$$\eta = 0.28$$

also

$$\eta = 1 - \frac{Q_2}{Q_1}$$
$$\therefore\ 0.28 = 1 - \frac{Q_2}{50}$$

or

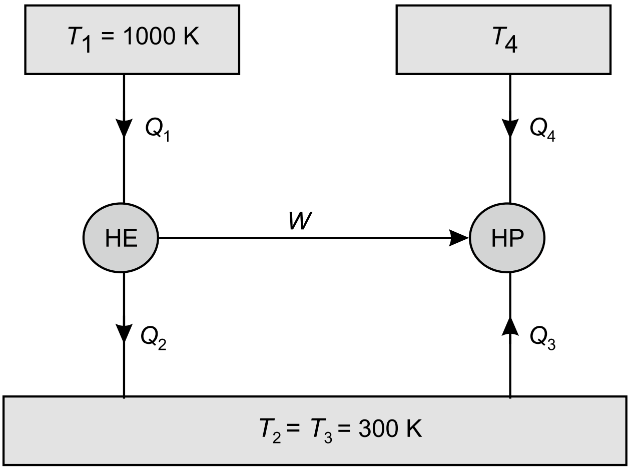
$$\frac{Q_2}{50} = 1 - 0.28 = 0.72$$

or

$$Q_2 = 0.72 \times 50 = 36\,\text{kW}$$

Heat engine output work:

$$\begin{aligned} W &= Q_1 - Q_2 = 50 - 36 = 14\ \text{kW} \\ &= \text{Heat pump input work} \end{aligned}$$



For heat engine,
Maximum/Carnot efficiency:

$$\eta_{\max} = 1 - \frac{T_2}{T_1} = 1 - \frac{300}{1000} = 0.7$$

For heat pump,

$$\begin{aligned} W &= 14\ \text{kW} \\ Q_3 &= 2Q_2 = 2 \times 36 = 72\ \text{kW} \end{aligned}$$

Heat rejection from the pump:

$$Q_4 = W + Q_3 = 14 + 72 = 86\,\text{kW}$$

Actual coefficient of performance:

$$(\text{COP})_{\text{HP}} = \frac{Q_4}{W} = \frac{86}{14} = 6.14$$

also

$$(\text{COP})_{\text{HP}} = 0.50(\text{COP}_{\max})_{\text{HP}}$$

$$\therefore\ 6.14 = 0.50(\text{COP}_{\max})_{\text{HP}}$$

or

$$(\text{COP}_{\max})_{\text{HP}} = 12.28$$

also

$$(\text{COP}_{\max})_{\text{HP}} = \frac{T_4}{T_4 - T_3}$$

$$\therefore\ 12.28 = \frac{T_4}{T_4 - 300}$$

or

$$12.28T_4 - 3684 = T_4$$

or

$$11.28T_4 = 3684$$

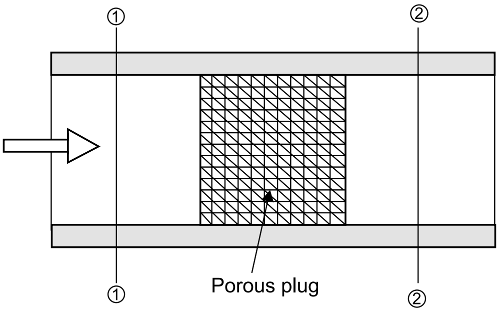
or

$$T_4 = \frac{3684}{11.28} = 326.59\,\text{K}$$

Q.17. Explain in Detail the Joule–Kelvin Effect.

Ans. Joule–Kelvin Effect: Joule–Kelvin effect states that when a real gas or a liquid under high pressure passing through a valve or porous plug while kept insulated, so that no heat is transferred to or from the fluid, the temperature change of a real gas or a liquid will take place. Joule–Kelvin effect is a throttling process, meaning that the enthalpy of the fluid is constant during the process. The Joule–Kelvin effect is also called Joule–Thomson effect.

Throttling process is shown in Fig. 1, by a porous plug on a fluid flowing in an insulated pipe.



Applying the steady flow energy equations

$$h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + gz_2 + w_{1-2} \tag{4}$$

Throttling process can be analysed with the following assumptions.

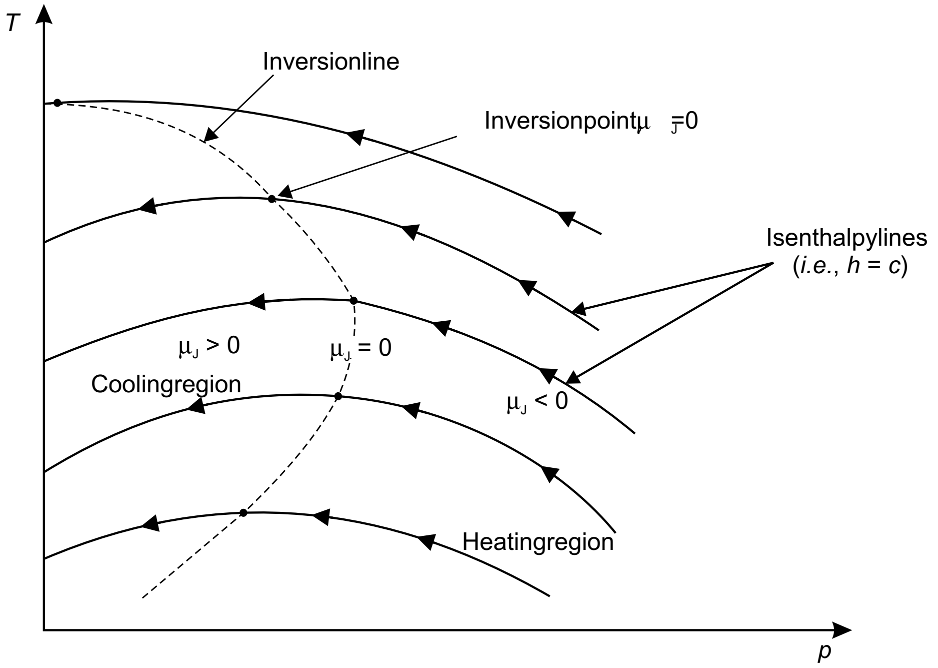
(i) Work done: $w_{1\text{-}2} = 0$
(ii) Head transfer: $q_{1\text{-}2} = 0$
(iii) Change in kinetic and potential energies are neglected.

Equation (4) become

$$\boldsymbol{h}_1 = \boldsymbol{h}_2$$

It means that the enthalpy of fluid remains constant during the throttling process.

For Real Gas: For ideal gases, enthalpy is a function of temperature only. But for real gases, enthalpy is a function of both temperature and pressure. The enthalpy of a real gas remains constant during the throttling process but the temperature does not remains constant. Experiment can be conducted by keeping constant temperature and pressure on the upstream condition (i.e., before throttling) but varying the pressure on the down stream condition (i.e., after throttling) and measuring corresponding temperature. This is achieved by changing different sizes of porous plugs. Both temperature and pressure of the real gas is measured after throttling. Since the upstream pressure and temperature conditions are kept constant, the enthalpy of the real gas for all measured conditions on down stream pressure and temperature would be constant enthalpy line on *T–p* diagrams. Repeating the experiment for different sets of inlet temperature and pressure and plotting the result, we can construct a *T–p* diagram for a real gas with serval constant enthalpy lines, as shown in Fig. 2.



The temperature behaviour of a real gas during a throttling process is described by the Joule–Thomson coefficient, which defined as the slope of an isenthalpic curve at any point in *T–p* diagram. Mathematically,

Joule–Thomson coefficient:

$$\mu_J = \left(\frac{\partial T}{\partial p}\right)_{h=\text{constant}}$$

where

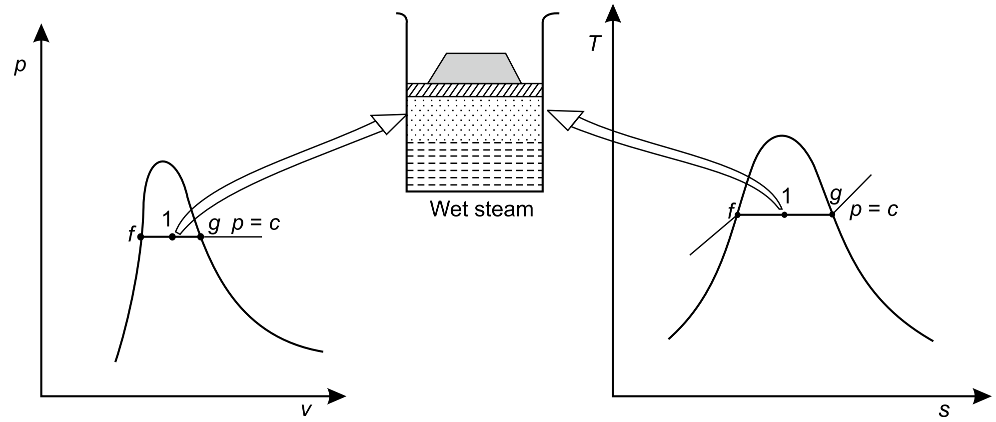
$$\mu_J \text{ is} + ve, -ve \text{ or zero}$$

The point on the isenthalpic line where $\mu_J = 0$ is called the inversion point. Thus, the inversion point denotes the maximum value of temperature on the isenthalpic line in *T–p* diagram. The line that passes through the inversion points is called the inversion line. The temperature at a point where an isenthalpic line intersects the inversion line is called the inversion temperature. The Joule–Thomson coefficient is positive ($\mu_J > 0$) to the left side of the inversion line is zero at the inversion point and negative ($\mu_J < 0$) to the right side of the inversion line.

A throttling process always proceeds along a constant enthalpy line in the direction of decreasing pressure, i.e., from right to left.

Q.18. Explain and draw *p–v* and *T–s* diagram for water and some other fluid. Explain how to determine the enthalpy and entropy in the wet region.

Ans. At any state 1 between saturated liquid state/and saturated vapour state g, shown in Fig.



Let

h_f = specific enthalpy of the saturated liquid
h_g = specific enthalpy of the saturated vapor
m = total mass of the wet steam
m_f
m_g = mass of the vapor phase

At state 1, total enthalpy of the wet steam is the sum of the enthalpy occupied by the liquid and that occupied by the vapour.

That is

$$H = H_f + H_g \tag{5}$$

By definition of the specific enthalpy,

$$h = \frac{H}{m} \text{ for wet} - \text{steam}$$

or

$$H = mh$$

$$h_f = \frac{H_f}{m_f} \text{ for liquid}$$

or

$$H_f = m_f h_f$$

and

$$h_g = \frac{H_g}{m_g} \text{ for vapor}$$

or

$$H_g = m_g h_g$$

Substituting the values of H, H_f and H_g in Eq. (5), we get

$$mh = m_f h_f + m_g h_g$$

or

$$h = \frac{m_f}{m} h_f + \frac{m_g}{m} h_g \tag{6}$$

The ratio of the mass of saturated vapour to the total mass of the wet steam is called the quality of the wet steam.

i.e.

$$x = \frac{m_g}{m}$$

and; Total mass:

$$m = m_f + m_g$$

or

$$m_f = m - m_g$$

Substituting the values of x and m_f in Eq. (6) (if), we get

$$\begin{aligned} h &= \frac{(m - m_g)}{m} h_f + x h_g = \left(1 - \frac{m_g}{m}\right) h_f + x h_g \\ &= (1 - x) h_f + x h_s \\ &= h_f - x h_f + x h_g \\ \boldsymbol{h} &= \boldsymbol{h_f + x(h_g - h_f)} \\ \boldsymbol{h} &= \boldsymbol{h_f + x h_{fg}} \end{aligned}$$

Similarly, the specific entropy of the wet steam of state 1,

$$\begin{aligned} s &= s_f + x(s_g - s_f) \\ s &= s_f + x s_{fg} \end{aligned}$$

Q.19. An air standard dual cycle has a compression ratio of 16 and compression begins at 1 bar, 50 °C. The maximum pressure is 70 bar. The heat transferred to air at constant pressure is equal to that at constant volume estimate-

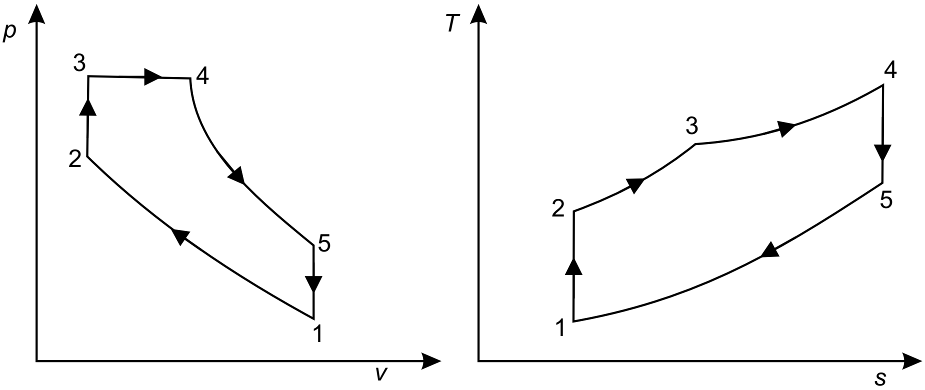
(a) **The pressures and temperatures at the cardinal points of the cycle.**
(b) **The cycle efficiency**
(c) **The m. e.p. of the cycle. Take c_V = 0.71 kJ/kgK, c_p = 1.005 kJ/kgK.**

Ans. Given data:

Compression ratio:

$$r = \frac{v_1}{v_2} = 16$$

$$\begin{aligned} p_1 &= 1 \text{ bar} = 100 \text{ kPa} \\ T_1 &= 50^\circ\text{C} = 50 + 273 = 323 \text{ K} \end{aligned}$$



Maximum pressure:

$$p_3 = p_4 = 70 \text{ bar} = 7000 \text{ kPa}$$

$$q_{2-3} = q_{3-4}$$

Applying the equation of state at state 1,

$$p_1 v_1 = R T_1$$

$$100 \times v_1 = 0.287 \times 323$$

or

$$v_1 = 0.927 \, \text{m}^3/\text{kg}$$

also

$$v_1 = 16v_2$$
$$\therefore\ 0.927 = 16 \times v_2$$

or

$$v_2 = 0.0579\,\mathrm{m_3/kg}$$

(a) The pressures and temperatures at the cardinal points of the cycle. For adiabatic process 1–2,

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$
$$\frac{T_2}{323} = (16)^{1.4-1}$$

or

$$T_2 = 979.15\,\mathrm{K}$$

also

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma}$$
$$\frac{p_2}{100} = (16)^{1.4}$$

or

$$p_2 = 4850.29\,\mathrm{kPa}$$

For process 2–3, heat addition at v = C

$$\frac{T_3}{T_2} = \frac{p_3}{p_2}$$
$$\frac{T_3}{979.15} = \frac{7000}{4850.293}$$

or

$$T_3 = 1413.12\,\mathrm{K}$$

$$\frac{1723.15}{T_5} = 2.8$$

or

$$T_5 = 615.41\,\mathrm{K}$$

and

$$\frac{p_4}{p_5} = \left(\frac{v_5}{v_4}\right)^{\gamma}$$
$$\frac{p_4}{p_5} = \left(\frac{v_1}{v_4}\right)^{\gamma} \qquad \because v_5 = v_1$$
$$\frac{7000}{p_5} = \left(\frac{0.927}{0.0706}\right)^{1.4}$$
$$\frac{7000}{p_5} = 36.77$$

or

$$p_5 = 190.37\,\mathrm{kPa}$$

(b) The cycle efficiency: η_{Dual}

Net heat addition:

$$\begin{aligned} q_A &= q_{2-3} + q_{3-4} \\ &= 2q_{2-3} \qquad \because q_{2-3} = q_{3-4} \\ &= 2 \times 311.59 \\ &= 623.18\ \ \mathrm{kJ/kg} \end{aligned}$$

Heat rejection:

$$\begin{aligned} q_R &= c_v(T_5 - T_1) \\ &= 0.718(615.41 - 323) = 209.95\ \ \mathrm{kJ/kg} \end{aligned}$$

Heat supplied:

$$\begin{aligned} q_{2-3} &= c_v(T_3 - T_2) \\ &= 0.718(1416.12 - 979.15) = 311.59\ \ \mathrm{kJ/kg} \\ &= q_{3-4} \end{aligned}$$

For process 3–4, heat addition at p = C

$$q_{3-4} = c_p(T_4 - T_3)$$
$$311.54 = 1.005(T_4 - 1413.12)$$

or

$$310.03 = T_4 - 1413.12$$

or

$$\begin{aligned}T_4 &= 310.03 + 1413.12\\ &= 1723.15\,\text{K}\end{aligned}$$

and

$$\begin{aligned}\frac{T_4}{T_3} &= \frac{v_4}{v_3}\\ \frac{1723.15}{1413.12} &= \frac{v_4}{0.0579} \quad \because v_3 = v_2\end{aligned}$$

or

$$V_4 = 0.0706\,\text{m}_3/\text{kg}$$

For process 4–5, adiabatic expansion:

$$\begin{aligned}\frac{T_4}{T_5} &= \left(\frac{v_5}{v_4}\right)^{\gamma-1}\\ \frac{1723.15}{T_5} &= \left(\frac{0.927}{0.0706}\right)^{1.4-1} \quad \because v_5 = v_1\\ &= 209.95\,\text{kJ/kg}\end{aligned}$$

Net work output:

$$\begin{aligned}w &= q_A - q_R\\ &= 623.18 - 209.95 = 413.23\,\text{kJ/kg}\end{aligned}$$

$$\begin{aligned}\eta_{\text{Dual}} &= \frac{w}{q_A} = \frac{413.23}{623.18}\\ &= 0.6630 = 66.30\%\end{aligned}$$

(c) The mep of the cycle

$$\begin{aligned}mep &= \frac{\text{work done per cycle}}{\text{swept volume}}\\ &= \frac{w}{v_1 - v_2} = \frac{413.23}{0.927 - 0.0579} = 475.46\,\text{kPa}\end{aligned}$$

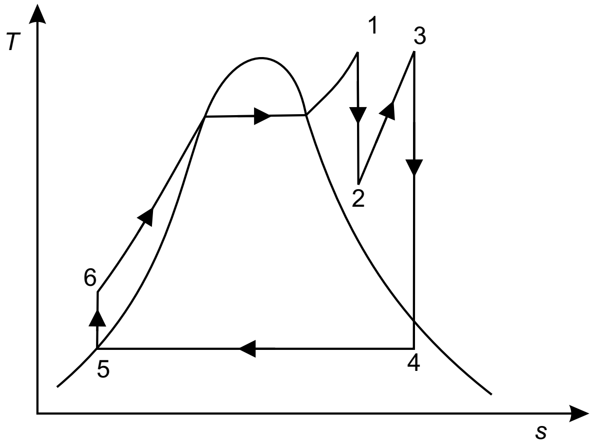
Q.20. Steam at 150 bar, 550 °C enters into HP turbine and after expansion to 40 bar it is sent to boiler for reheating at the same temperature of 550 °C. It is then expanded in LP turbine to the condenser pressure of 0.1 bar. Calculate cycle efficiency. Also draw T–s diagram, take enthalpy at inlet to HP turbine as 3448 kJ/kg, exit to HP turbine 3065 kJ/kg, inlet to LP turbine 3565 kJ/kg, h_f and h at condenser pressure are 191.8 kJ/kg and 2584 kJ/kg, respectively. Dryness fraction at exit of LP turbine is 0.88 and specific volume of water at condenser outlet = 0.00101 m^3/kg.

Ans. Given data:

$$\begin{aligned}p_1 &= 150\,\text{bar} = 15000\,\text{kPa}\\ T_1 &= 550°\text{C}\\ p_2 &= 40\,\text{bar} = 4000\,\text{kPa}\\ T_3 &= 550°\text{C}\\ P_4 &= 0.1\,\text{bar} = 10\,\text{kPa}\\ h_1 &= 3448\,\text{kJ/kg}\\ h_2 &= 3065\,\text{kJ/kg}\\ h_3 &= 3565\,\text{kJ/kg}\end{aligned}$$

At

$$\begin{aligned}p_4 &= 10\,\text{kPa}\\ h_f &= 191.8\,\text{kJ/kg}\\ h_g &= 2584.7\,\text{kJ/kg}\\ x_4 &= 0.88\\ v_5 &= 0.00101\,\text{m}^3/\text{kg}\end{aligned}$$



Specific enthalpy at state 4,

$$\begin{aligned} h_4 &= h_f + x_4(h_g - h_f) \\ &= 191.8 + 0.88(2584.7 - 191.8) = 2297.55\,\text{kJ/kg} \end{aligned}$$

Specific enthalpy at state 5,

$$h_5 = h_f = 191.8\,\text{kJ/kg}$$

Pump input work:

$$\begin{aligned} w_{5-6} &= v_5(p_1 - p_4) \\ &= 0.00101(15000 - 10) = 15.139\,\text{kJ/kg} \end{aligned}$$

also

$$w_{5-6} = h_6 - h_5$$

$$\therefore\ 15.139 = h_6 - 191.8$$

or

$$h_6 = 206.939\,\text{kJ/kg}$$

Net heat supplied:

$$\begin{aligned} q_{net} &= (h_1 - h_6) + (h_3 - h_2) \\ &= (3448 - 206.935) + (3565 - 3065) \\ &= 3210.06 + 500 = 3710.06\,\text{kJ/kg} \end{aligned}$$

Net work output:

$$\begin{aligned} w_{net} &= w_{1-2} + w_{3-4} - w_{5-6} \\ &= (h_1 - h_2) + (h_3 - h_4) - 15.139 \\ &= (3448 - 3065) + (3565 - 2297.55) - 15.139 \\ &= 383 + 1267.45 - 15.139 = 1635.31\,\text{kJ/kg} \end{aligned}$$

Cycle efficiency:

$$\begin{aligned} \eta &= \frac{\text{Net work output:}w_{net}}{\text{Net heat supplied:}q_{net}} \\ &= \frac{w_{net}}{q_{net}} = \frac{1635.31}{3710.06} = 0.4407 = 44.07\% \end{aligned}$$

Q.21. (*a*) Define property of a system and explain extensive and intensive properties.

Ans. The characteristic used to describe the condition of the system is called property of a system. Temperature, pressure, volume, energy, etc., are properties of a system. These properties of a system can be classified into two types:

(i) Intensive property and
(ii) Extensive property

(i) **Intensive property.** The properties are independent on the mass of the system, called intensive properties. For examples, temperature, pressure, density and viscosity.

(ii) Extensive property. The properties that depend on the mass of the system is called extensive properties. For examples, energy enthalpy and entropy.

(*b*) Explain 1st law applicable to a closed system undergoing in a cyclic path.

Ans: 1st law states that when a system undergoes in a cyclic path, the algebraic sum of the heat transfer is equal to the algebraic sum of the work transfer.

Mathematically,

$$\sum \delta Q = \sum \delta W$$

(*c*) Write expression for steady state energy equation and explain each term.

Ans: Steady state energy equation,

$$m\left[h_1 + \frac{V_1^2}{2} + gz_1\right] + Q_{1-2} = m\left[h_2 + \frac{V_2^2}{2} + gz_2\right] + W_{1-2}$$

or

$$h_1 + \frac{V_1^2}{2} + gz_1 + \frac{Q_{1-2}}{m} = h_1 + \frac{V_2^2}{2} + gz_2 + \frac{W_{1-2}}{m}$$

or

$$h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + gz_1 + w_{1-2}$$

where h_1, h_2 are the specific enthalpies at inlet and outlet of steady flow system, respectively.

V_1, V_2 are velocities at inlet and outlet of the system respectively

z_1, z_2 are datum heads at inlet and outlet and of the system, respectively.

$q_{1\text{-}2}$ is specific heat transfer.

$w_{1\text{-}2}$ is specific work.

(*d*) Write Carnot theorem and its corollary 2nd law.

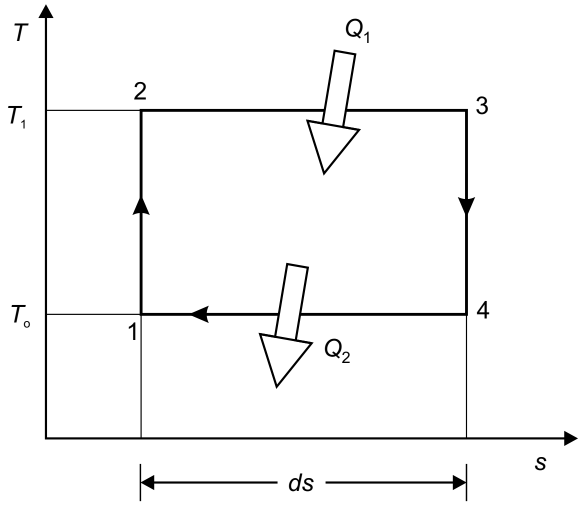
Ans. Carnot Theorem. It states that the efficiency of an irreversible heat engine is always less then the efficiencies of a reversible heat engine operating between the same temperature limits.

Corollary 1. The efficiencies of all reversible heat engines operating between the same source and sink are the same.

Corollary 2. The efficiency of any reversible engine does not depend on the working fluid in the cycle and depends only on the temperature of the source and the sink.

(*e*) Explain decrease in availability considering heat transfer through a finite temperature difference.

Ans. Let us consider a reversible heat engine operating between temperature T_1 and $T_{0,}$ From Fig. 1.



Heat supplied:

$$Q_1 = T_1 ds$$

Heat rejected:

$$Q_2 = T_0 ds$$

Work output:

$$W = AE = Q_1 - Q_2$$
$$= (T_1 - T_0)ds$$

Let Q_1 is heat transferred through finite temperature difference from source at T_1 to the engine absorbing heat at T_1' lower than T_1, as shown in Fig. 2.

The availability of Q_1 as received by the engine at T_1' lower than T_1.

Heat supplied:

$$Q_1 = T_1 ds = T_1' ds$$

Since

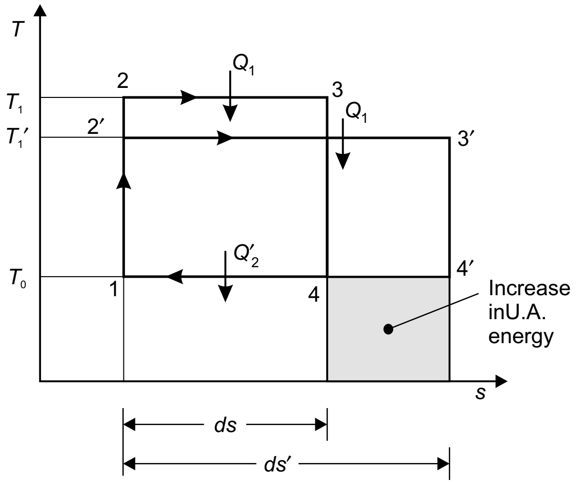
$$T_1 > T_1'$$
$$\therefore\ ds' > ds$$

and

$$Q_2 = T_0 ds$$

and

$$Q_2' = T_0 ds'$$



Since

$$ds' > ds$$

$$\therefore\ Q_2' > Q_2$$

and

$$\begin{aligned} W' &= Q_1 - Q_2' \\ &= T_1' ds' - T_0 ds' = (T_1' - T_0) ds' \end{aligned}$$

and

$$W = Q_1 - Q_2 = T_1^{'} ds - T_0 ds$$

$$\therefore\ W' < W \qquad \therefore\ Q_2' > Q_2$$

Available energy lost due to irreversible heat transfer through finite temperature difference between the source.

Decrease in available energy

$$\begin{aligned} \text{Decrease in available energy} &= W - W' \\ &= Q_2' - Q_2' \\ &= T_0(ds' - ds) \end{aligned}$$

***(f)* Write physical signification of Gibbs function and Helmholtz function.**

Ans. Physical significance of Gibbs function. From the law of conservation of energy, the total heat supplied to the system,

$$\begin{aligned} Q &= dU + W_{\exp} + W_{\text{non}-\exp} \\ &= dU + pdV + W_{\text{non}-\exp} \\ &= dH + W_{\text{non}-\exp} \end{aligned} \tag{7}$$

Now in a reversible process at T = C, we have

$$dS = \frac{\delta Q}{T}$$

or

$$Q = Tds \tag{8}$$

From Eqs. (7) and (8), we get

$$TdS = dH + W_{non-\exp}$$

or

$$dH - TdS = W_{non-\exp}$$

$$(\Delta G)_{T,P} = -W_{non-\exp}$$

or

$$-(\Delta G)_{T,P} = W_{non-\exp}$$

−ΔG is a measure of non-expansion or useful work obtainable in a reversible reaction at constant temperature and pressure. In other words, decrease in free energy, during a process, is equal to the useful work obtainable from the process.

Physical significance of Helmholtz function

Helmholtz function:

$$F = U - TS \tag{9}$$

For a system undergoing a change of state from state 1 to state 2, Eq. (9)

$$\begin{aligned} F_2 - F_1 &= (U_2 - T_2S_2) - (U_1 - T_1S_1) \\ &= (U_2 - U_1) - (T_2S_2 - T_1S_1) \end{aligned} \tag{10}$$

If the system is closed, the first law gives

$$Q_{1-2} = (U_2 - U_1) + W_{1-2}$$

or

$$U_2 - U_1 = Q_{1-2} - W_{1-2} \tag{11}$$

From Eq. (10), we get

$$F_2 - F_1 = Q_{1-2} - W_{1-2} - (T_2S_2 - T_1S_1)$$

For isothermal process,

$$T_1 = T_2 = T$$

$$\therefore\ F_2 - F_1 = Q_{1-2} - W_{1-2} - T(S_2 - S_1) \quad (12)$$

We know that for reversible and irreversible process,

$$dS \geq \frac{\partial Q}{T}$$

or

$$TdS \geq \delta Q$$

For process 1–2,

$$T(S_2 - S_1) \geq Q_{1-2}$$

or

$$Q_{1-2} \leq T(S_2 - S_1)$$

Hence, Eq. (12) becomes as

$$F_2 - F_1 \leq -W_{1-2}$$

The equality can be written as

$$F_2 - F_1 + TdS_{irr} = -W_{1-2} \qquad (13)$$

Thus, during a reversible isothermal process, for which $dS_m = 0$, the amount of work done by the system W_{max} is equal to the decrease in the value of the Helmholtz function. Furthermore, for an isothermal process conducted at constant volume, which does not perform p–V work, Eq. (13) gives

$$F_2 - F_1 + TdS_{irr} = 0$$

For an increment of such a process,

$$dF + TdS_{irr} = 0$$

$$dF = -TdS_{irr}$$

As dS_m is always positive during a spontaneous process, it is thus seen that the Helmholtz function F decreases during spontaneous process, and as $dS_m = 0$ for a reversible process, equilibrium requires that $dF = 0$.

(*g*) Explain dryness fraction. Draw a line on *T–s* diagram representing sensible heating, vaporization and superheating of water.

Ans. Dryness fraction. The dryness fraction of the wet steam is defined as the ratio of mass of vapour to the total mass of the wet steam.

Mathematically,

$$\text{Dryness fraction: } x = \frac{\text{mass of vapor}}{\text{total mass of the wet steam}}$$

$$x = \frac{m_v}{m}$$

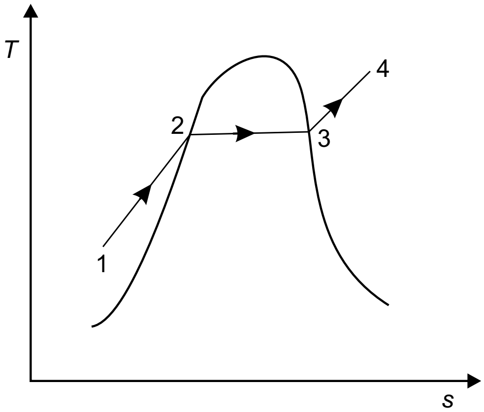
where

$$m = m_v + m_l$$

$$\therefore\ x = \frac{m_v}{m_v + m_l}$$

The value of dryness fraction lies between 0 and 1.

$x = 0$ for saturation liquid
$x = 1$ for saturation vapour



(i) **Sensible heating.** The amount of heat required to raise the temperature of water at constant pressure when phase change not occurs is called sensible heating. Processes 1–2 and 3–4 show sensible heating on *T–s* diagram.

(ii) **Vaporization.** Vaporization is a process of phase change from the liquid phase to vapour phase. Thus, the process is repressible by 2–3 on *T–s* diagram.

(iii) **Superheating.** The dry saturated steam the stream is heated above the saturation temperature at constant pressure is called superheating. The superheating is represented by process 3–4 on *T*–*s* diagram.

(*h*) Explain why Otto, Diesel and Brayton cycle are called air standard cycle? Explain limitations of Otto cycle is achieving higher efficiency.

Ans. The operation of Otto, Diesel and Brayton cycle can be analyzed by assuming that the working fluid is air, the air is assumed to be an ideal gas with constant specific heat through the cycle. Such cycle is called air-standard cycle.

We know that thermal efficiency of Otto cycle,

$$\eta_{th} = 1 - \frac{1}{r^{\gamma-1}} \qquad (14)$$

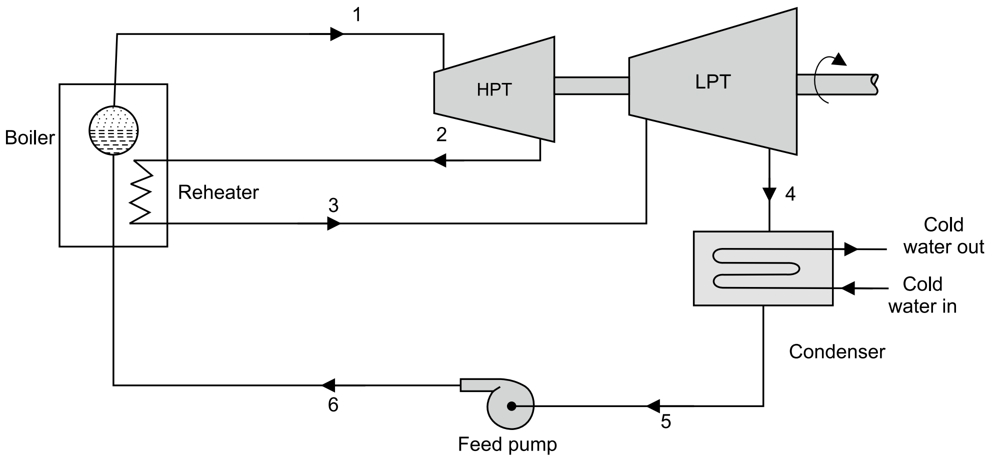
where

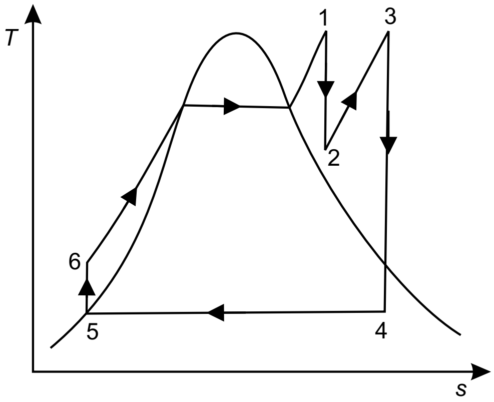
r = compression ratio.

Equation (14) indicates the thermal efficiency increases with increase in compression ratio. The upper limit of the compression ratio is 12. Further, as the compression ratio increases, the efficiency decreases due to detonation or knocking. Detonation is abnormal combustion due to end charge auto-ignites before the flame reaches it.

(i) Explain how reheating of working substance improve efficiency of Rankine cycle? Draw schematic and *T–s* diagram.

Ans. In reheat cycle, steam is reheated after expanding partially in the high pressure turbine. This is done by the steam partial expansion in high pressure turbine. The steam is then sent back to the boiler where it is reheated to constant pressure, usually to the inlet temperature of the high pressure turbine. Steam is then supplied to the low pressure turbine in which it expands isentropically to the condenser pressure as shown in Fig. 2. In reheat cycle, net heat supplied and net work output both increase but net work output increases more than heat supplied, thus efficiency of the cycle increases.





(*j*) Write expression of Clausius inequality and explain its physical significance.

Ans. Clausius inequality. According to Clausius inequality, cycle integral of δQ/T is always less than and equal to zero.

Mathematically,

$$\oint \left(\frac{\delta Q}{T}\right) \leq 0$$

Physical significance

This inequality is valid for all cycle, reversible or irreversible, including the refrigerator and heat pump cycles. It provides the criterion to check the types of the cycle, i.e., the cycle is reversible, irreversible or impossible.

$$\oint \left(\frac{\delta Q}{T}\right) = 0 \text{ for reversible cycle}$$

$$\oint \left(\frac{\delta Q}{T}\right) < 0 \text{ for irreversible cycle}$$

$$\oint \left(\frac{\delta Q}{T}\right) > 0 \text{ for impossible cycle}$$

Q.22. A classroom of 60 students contains 6 fans each consuming 0.20 kW power and 8 tubes lights of 40 W each. Ventilation air is supplied at a rate of 500 kg/h with specific enthalpy of 90 kJ/kg and air leaves the class room at a specific enthalpy of 60 kJ/kg. If each person puts out heat at a rate 600 kJ/h, determine the rate at which heat is to be removed by a room Cooler to maintain the steady state in the room.

Ans:

Given data

$$\text{Number of students} = 60$$

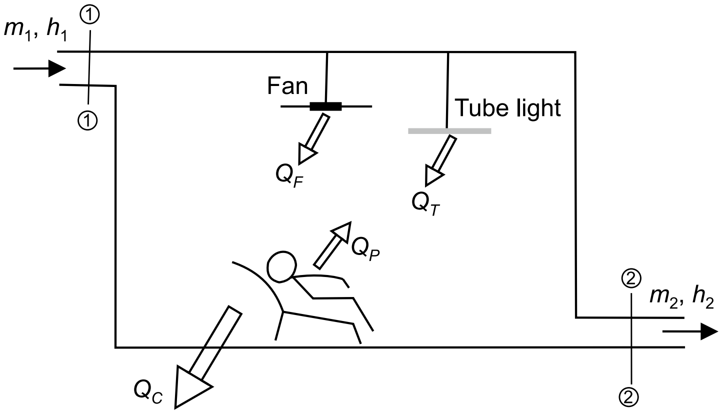
$$\text{Number of fans} = 6$$

Power consumed each fans:

$$P_F = 0.20\,\text{kW}$$

Power consumed 6 fans

$$P_F = 6 \times 0.20 = 1.2\,\text{kW}$$



Heat gain from fans:

$$Q_F = P_F = 1.2\,\text{kW}$$

Heat gain from: 8 tubes light:

$$Q_T = 8 \times 40 = 320\,\text{W} = 0.32\,\text{kW}$$

Mass flow rate of air:

$$m = 500\,\text{kg/hr} = \frac{500}{3600}\text{kg/s} = 0.13889\text{kg/s}$$

Specific enthalpy of supplied air:

$$h_1 = 90\,\text{kJ/kg}$$

Specific enthalpy of air leaves:

$$h_2 = 60\,\text{kJ/kg}$$

Each person puts out heat:

$$Q_p = 600\,\text{kJ/hr} = \frac{600}{3600}\,\text{kW} = 0.1667\,\text{kW}$$

Heat put out from 60 students:

$$Q_p = 60 \times 0.1667 = 10\,\text{kW}$$

Let Q_c is the rate of heat removed by the room cooler.

Applying the energy balance equation, sum of heat gained by the room = sum of heat leaving from the room.

$$\begin{aligned} &m_1h_1 + Q_F + Q_T + Q_P = m_2h_2 + Q_c \\ &0.13889 \times 90 + 1.2 + 0.32 + 0.1667 \\ &= 0.13889 \times 60 + Q_C \\ &14.1868 = 8.3334 + Q_C \end{aligned}$$

or

$$\boldsymbol{Q_C} = 5.853\,\text{kW}$$

5.853 kW heat is to be removed by a room cooler.

Q.23. Air at 40 °C and 1 bar is compressed reversible and polytropically from 5 m3 to 1 m\calculate the final temperature, pressure, work done and heat transfer, if the index of compression is 1, 1.6 and 0.8, respectively. Assume c_p = 1.005 kJ/kg K and c_v = 0.7178 kJ/kg K

Ans: Given data:

At initial state,

$$\begin{aligned} T_1 &= 40^\circ C = 313\,\text{K} \\ p_1 &= 1\,\text{bar} = 100\,\text{kPa} \\ V_1 &= 5\,\text{m}^3 \end{aligned}$$

At final state,

$$\begin{aligned} V_2 &= 1\,\text{m}^2 \\ c_p &= 1.005\,\text{kJ/kgK} \\ c_v &= 0.7178\,\text{kJ/kgK} \\ \therefore\; R &= c_p - c_v = 1.005 - 0.7178 \\ &= 0.2872\,\text{kJ/kgK} = 0.2872\,\text{kJ/kgK} \end{aligned}$$

Applying the equation of state at initial state,

$$\begin{aligned} p_1V_1 &= mRT_1 \\ 100 \times 5 &= m \times 0.2872 \times 313 \end{aligned}$$

Or

$$m = 5.562\,\text{kg}$$

Case I: When the index of compression: n = 1.

Polytropic law:

$$\begin{aligned} pV^n &= C \\ pV &= C \quad \because n = 1 \end{aligned}$$

The air is compressed isothermally,

$$\begin{aligned} \therefore\; T_2 = T_1 &= 313\,\text{K} \\ p_1V_1 &= p_2V_2 \\ 100 \times 5 &= p_2 \times 1 \end{aligned}$$

or

$$p_2 = 500\,\text{kPa} = 5\,\text{bar}$$

Work done:

$$\begin{aligned} W_{1-2} &= mRT_1 \log_e \frac{V_2}{V_1} \\ &= 5.56 \times 0.2872 \times 313 \log_e \frac{1}{5} = -804.699\,\text{kJ} \end{aligned}$$

Heat transfer:

$$Q_{1-2} = W_{1-2} = -804.699\,\text{kJ}$$

Case II: When the index of compression: n = 1.4

Polytropic law:

$$pv^n = C$$

For process 1–2,

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^n$$
$$\frac{p_2}{100} = \left(\frac{5}{1}\right)^{1/4}$$

or

$$P_2 = 100(5)^{1/4}$$
$$= 951.82\,\text{kPa} = 9.518\,\text{bar}$$
$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1}$$
$$\frac{T_2}{313} = \left(\frac{5}{1}\right)^{1.4-1}$$

or

$$T_2 = 595.84\,\text{K}$$

Work done:

$$W_{1-2} = \frac{p_2V_2 - p_1V_1}{1-n}$$
$$= \frac{951.82 \times 1 - 100 \times 5}{1 - 1.4} = -1129.55\,\text{kJ}$$

According to first law of thermodynamics for process 1–2,

$$Q_{1-2} = (U_2 - U_1) + W_{1-2}$$
$$= mc(T_2 - T_1) + W_{1-2}$$
$$= 5.562 \times 0.7178(595.84 - 313) - 1129.55$$
$$= 1129.21 - 1129.55$$
$$\approx 0$$

Case III: When the index of compression: $n = 0.8$

Polytropic law:

$$pV^n = C$$

For process 1–2;

$$\frac{p_2}{p_1} = \left(\frac{V_1}{V_2}\right)^n$$
$$\frac{p_2}{100} = \left(\frac{5}{1}\right)^{0.8}$$

or

$$p_2 = 362.389\,\text{kPa} = 3.6239\,\text{bar}$$
$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1}$$
$$\frac{T_2}{313} = \left(\frac{5}{1}\right)^{0.8-1}$$
$$T_2 = 313 \times (5)^{-0.2}$$
$$\frac{313}{50.2} = 226.85\,\text{K}$$

Work done:

$$W_{1-2} = \frac{p_2V_2 - p_1V_1}{1-n} = \frac{362.389 \times 1 - 100 \times 5}{1 - 0.8}$$
$$= -688.055\,\text{kJ}$$

According to first law of thermodynamics for process 1–2,

$$Q_{1-2} = (U_2 - U_1) + W_{1-2}$$
$$= mc_v(T_2 - T_1) + W_{1-2}$$
$$= 5.562 \times 0.7178(226.85 - 313) - 688.055$$
$$= -1032\,\text{kJ}$$

Q.24. Explain Carnot heat engine, refrigerator and heat pump. Derive the expression for efficiency of heat engine and COP of a refrigerator and pump.

Ans. Carnot heat engine. It is used to convert a part of heat into work, while remaining waste heat is rejected to a sink.

Refrigerator. It is a device which is used to attain and maintain a temperature below that of the surroundings, the aim being a cool space to some required temperature.

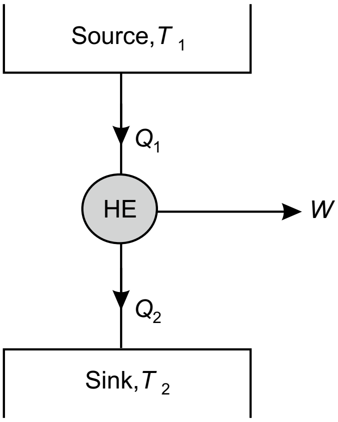
Heat pump. It is a device which is used to attain and maintain a temperature higher than that of the surroundings, the aim being the heat up some space to higher temperature.

Carnot heat engine:

Let

$$Q_1 = \text{heat supplied}$$
$$Q_2 = \text{heat rejected}$$
$$W = \text{net work output}$$

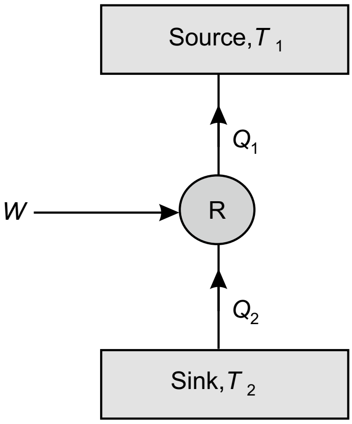
Efficiency:



$$\eta_{\text{current}} = \frac{\text{Net work output}}{\text{Heat supplied}} = \frac{w}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$
$$= 1 - \frac{T_2}{T_1} \quad \because \frac{Q_2}{Q_1} = \frac{T_2}{T_1}, \text{temperature scale}$$

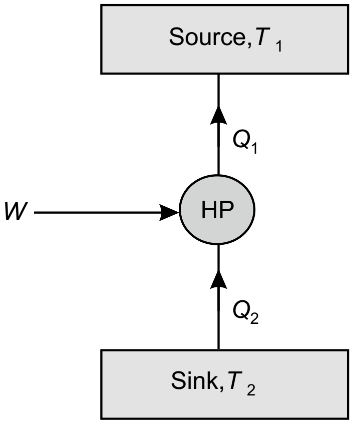
Refrigerator:

Coefficient of performance of refrigerator:



$$(\text{COP})_R = \frac{\text{Coolingeffect} : Q_2}{\text{Workinput} : W} = \frac{Q_2}{W}$$
$$= \frac{Q_2}{Q_1 - Q_2} \quad \because W = Q_1 - Q_2$$

Heat Pump:



Coefficient of performance of a heat pump

$$(\text{COP})_{\text{HP}} = \frac{\text{Heatingeffect} : Q_1}{\text{Workinput} : W}$$
$$\frac{Q_1}{W} = \frac{Q_1 - Q_2 + Q_2}{Q_1 - Q_2}$$
$$= 1 + \frac{Q_2}{Q_1 - Q_2}$$

$$(\mathrm{COP})_{\mathrm{HP}} = 1 + (\mathrm{COP})_{\mathrm{R}} \qquad (15)$$

Equation (15) shows that the coefficient of performance of a heat pump is equal to one more than the coefficient of performance of a refrigerator.

Q.25. What do understand by availability? Also, explain second law efficiency. Derive expression for 2nd efficiency.

Ans. Availability of the system is defined as the maximum useful work obtainable from the system when the system approaches to dead state (i.e., atmospheric temperature and pressure) from its initial state. It the denoted by A.

The following observation can be made about availability

(i) Availability of a system is zero at its dead state since both $T = T_0$ and $p = p_0$.
(ii) Availability is near negative.
(iii) Availability is not conserved like mass and energy, it is destroyed continuously the actual system.

Second law efficiency η_{II}.

It is defined as the ratio of the actual work output to the maximum work output of the work producing devices.

Mathematically,
Second law efficiency:

$$\eta_{\mathrm{II}} = \frac{\text{Actual work : } W}{\text{Maximum work:} \mathrm{W}_{\max}}$$

$$\eta_{\mathrm{II}} = \frac{W}{W_{\max}}$$

$$= \frac{W}{W_{\max}} \times \frac{Q_1}{Q_1}$$

$$= \frac{W}{Q_1 \times W_{\max}/Q_1}$$

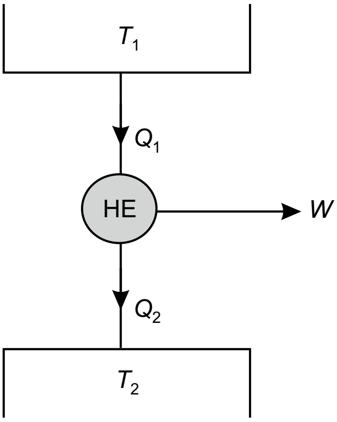
$$\eta_{\mathrm{II}} = \frac{n_{th}}{n_{rev}} = \frac{n_{th}}{n_{\mathrm{carnot}}}$$

where

$$\eta_{rev} = \eta_{\mathrm{carnot}} = \frac{W_{\max}}{Q_1}$$

and

$$\eta_{th} = \frac{W}{Q_1}$$



The second law efficiency is also defined as the ratio of actual thermal efficiency to the Carnot efficiency.

Q.26. Explain different types of calorimeter used for measurements of dryness fraction.

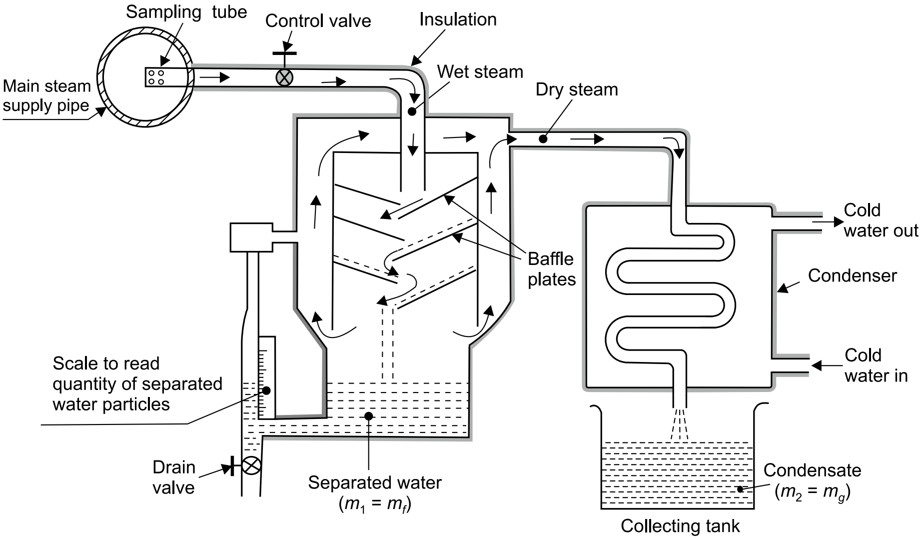
Ans. The dryness fraction of wet steam is determined by the following types of calorimeters:

1. Separating calorimeter
2. Throttling calorimeter
3. Combined separating and throttling calorimeter

1. Separating Calorimeter

This calorimeter is used to find the dryness fraction of the wet steam when the steam, which is very wet is passed through the sampling tube and is passed further through control value which is completely kept open. The wet steam coming out of the sampling tube in the inner chamber strike the baffle plates and follows the zigzag path inside the baffle plates. When the wet steam, containing the water particles, strikes the baffle plates, the particles of water that have greater density and hence greater inertia, clings to the baffle plates, gets separated from the steam and gets collected in the inner chamber. The quantity of separated water particles is measured by the calibrated gauge glass tube. The dry steam after it is being separated from the water particles moves up from the inner chamber to outer chamber.

In practice, it is not possible to separate all the water particles completely by this type of mechanical separation, the dryness fraction obtained by this calorimeter will not be accurate and it will always be higher than the actual value.



The dry steam then passes into the condenser in which it is condensed. The liquid thus separated out is collected in a collecting tank and is measured. Thus, by knowing the values of $m_1 = m_f$ and $m_2 = m_f$, the dryness fraction can be calculated as follows:

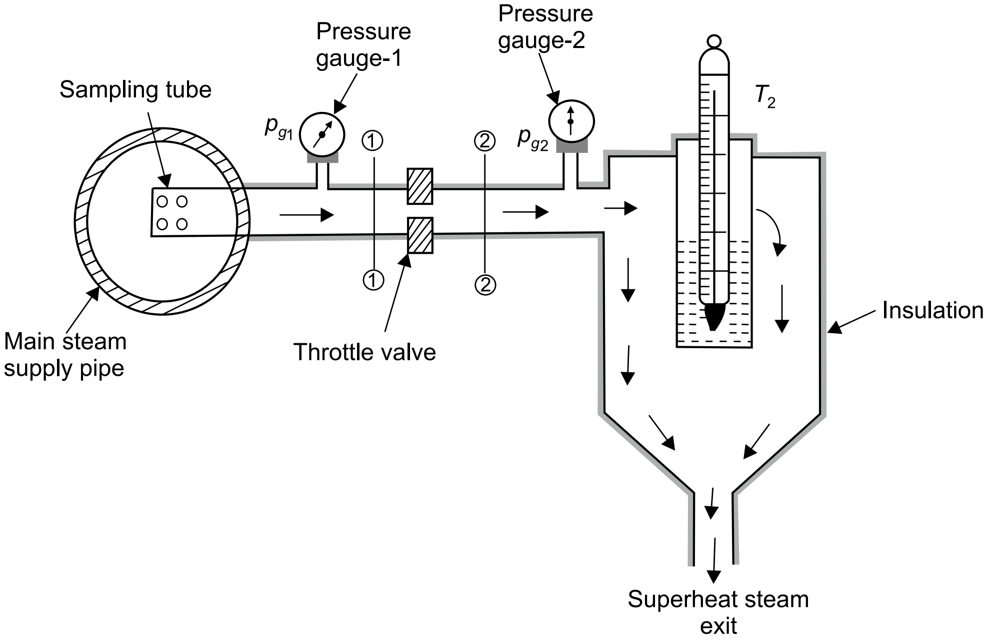
Dryness fraction:

$$
\begin{aligned}
x &= \frac{\text{mass of vapor:m}_g}{\text{total mass of the west steam : m}} \\
&= \frac{m_g}{m} \\
&= \frac{m_g}{m_f + m_g} = \frac{m_g}{m_1 + m_2}
\end{aligned}
$$

The advantage of this calorimeter is only the quick determination of the dryness fraction of very wet steam.

2. Throttling Calorimeter

This calorimeter works on the principle of throttling process. The throttling of steam is an expansion process in which the steam is passed through a orifice or restricted passage and the steam will expand from high pressure to low pressure adiabatically and without doing any work. Thus, the enthalpy (i.e., total heat) of the steam will remain same before and after throttling. After throttling, the steam will become superheated due to fall in pressure at constant enthalpy.



Sample steam is taken from the main pipe through the sampling tube. The sample steam passes through the throttle valve, which it will be subjected to the throttling expansion. The gauge pressure of the steam before and after throttling is recorded by the pressure gauges. The temperature after the throttled steam is measured by the mercury thermometer placed in pocket filled with oil. If the temperature of the steam after throttling is greater than the saturated temperature corresponding to the pressure of the steam after throttling, then the steam after throttling will be superheated. The process is shown on the h–s diagram in Fig. The dryness fraction of the wet steam is calculated as follows:

Let

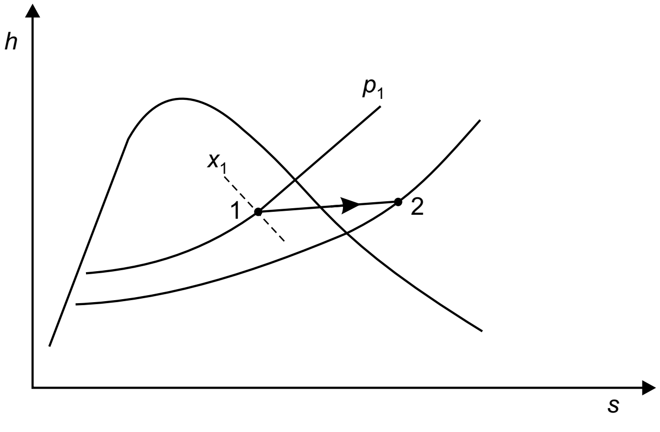
p_{g1} = gauage pressure of the stream before throttling read by pressure gauge − 1

p_{g2} = −gauage pressure of the stream after throttling read by pressure gauge − 2

p_{atm} = atmospheric pressure

x_1 = dryness fraction of the sample steam

T_2 = temperature of the stream after throttling read by the the mercury thermometer in°C



The properties of the steam are given on based upon absolute pressure.

Let p_l = absolute pressure of the steam before throttling = $p_{g_1} + p_{atm}$.

and p_2 = absolute pressure of the steam after throttling = $p_{g2} + p_{atm}$.

From pressure based saturated steam table, corresponding to the absolute pressure $p_{v,}$ we can find properties h_f h_g and h_{fg}.

∴ The specific enthalpy before throttling at state I, $h_1 = h_f + xh_{fg}$.

From the superheated steam table, corresponding to the absolute pressure p_2 and temperature T_2 at state 2, the specific enthalpy h_2 at state 2 is determined.

Note that the specific enthalpy at superheated steam at state 2 is also calculated by the.

following equation.

$$h_2 = h_{gp2} + c_{ps}(T_2 - T_{sat})$$

where

h_{gp2} = specific enthalpy at saturated vapour state corresponding to the pressure p_2.

c_{ps} = specific heat of superheated steam.

T_{atm} = saturated temperature of the dry saturated steam corresponding to the pressure p_2.

T_2 = temperature of the steam after throttling read by the mercury thermometer.

Since the enthalpy or specific enthalpy of the steam remains constant before and after throttling process,

i.e.,

$$h_1 = h_2$$

$$h_f + x_1 h_{fg} = h_2$$

or

$$x_1 = \frac{h_2 - h_f}{h_{fg}}$$

The dryness fraction found out from the above equation will be accurate provided the wet steam becomes superheated after throttling. This can happen only if the dryness fraction of the wet steam before throttling is quite high, i.e., $x > 0.9$. Therefore, this calorimeter cannot be used to determine when the dryness fraction of the steam is low, because the steam remains wet after the throttling.

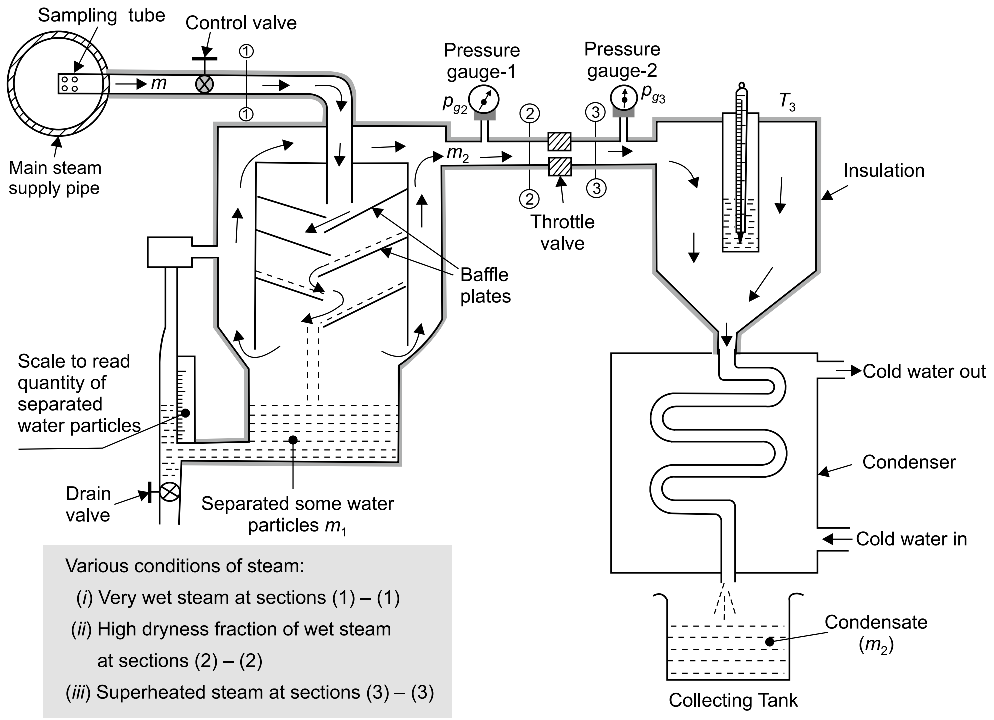
3. Combined Separating and Throttling Calorimeter

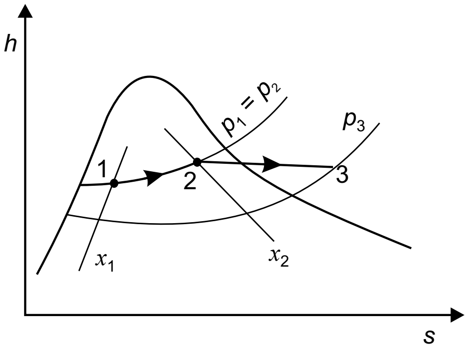
In a separating calorimeter, it is not possible to separate all the water particles with mechanical separation and hence the dryness fraction of the wet steam determined by the separating calorimeter is always higher than the actual value.

In a throttling calorimeter, it is not possible to determine the dryness fraction of the steam is low, if the steam remains wet after throttling. The throttling calorimeter can be only if the steam has high dryness fraction (i.e., $x > 0.9$) before throttling so that it must be superheated after throttling.

As discussed above, the separating and throttling calorimeters cannot be used individually to accurately determine the dryness fraction of the wet steam at any quality. However, by combining both the separating and throttling calorimeters as shown in Fig. 4, the dryness fraction, of a very steam can be very determined accurately.

In this arrangement, both the separating and throttling calorimeters are connected in series. The steam from the main supply pipe is passed through the sampling tube to the separating calorimeter, where the maximum amount of water particles are separated and the dryness fraction of the steam increases. From the separating calorimeter, high dryness fraction of wet steam enters the throttle valve, where the steam after throttling becomes superheated. The superheated steam coming out from the throttling calorimeter is condensed in the condenser. The mass of the condensate is the mass of the high dryness fraction wet steam separated in the separation calorimeter. The processes are shown in h–s diagram in Fig. 5. The dryness fraction of the wet steam is calculated as follows:





Let m = mass of the sample steam

or

mass of the wet steam at sections 1–1.

x_1 = dryness fraction of the sample steam

or

dryness fraction of the wet steam at sections 1–1.

Absolute pressure at section (1)–(1) = absolute pressure at section (2)–(2).

i.e.,

$$p_1 = p_2 = p_{g2} + p_{\text{atm}}$$

m_1 = mass of water particles separation in the separating calorimeter

m_2 = mass of the wet steam enters the throttling valve from the separating calorimeter

x_2 = dryness fraction of the wet steam before throttling valve

$p_2 = p_{g2} + p_{atm}$ absolute pressure of the superheated steam after to throttling

T_3 = Temperature of the Superheated Steam Measured by the Mercury Thermometer

The specific enthalpy (h_3) of the superheated steam at state 3, found out from superheated steam table, corresponds to the absolute pressure p_3 and temperature T_3.

From pressure based saturated steam table, corresponding to the absolute pressure p$_2$, the properties h_f, h_g and h_{fg} are found out.

The specific enthalpy before throttling at state 2,

$$h_2 = h_f + x_2 h_{fg}$$

Since the specific enthalpy of the steam remains constant before and after throttling process, i.e.

i.e.,

$$h_2 = h_3$$
$$h_f + x_2 h_{fg} = h_3$$

or

$$x_2 = \frac{h_3 - h_f}{h_{fg}}$$

We know that valves of h_f, hf_g and h_3, Thus, by using above equation, x_2 can be found out.

also

$$x_2 = \frac{m_g}{m_2}$$

or

Mass of the vapour:

$$m_g = x_2 m_2$$

The values of x_2 and m_2 are known. The mass of vapor m_g can be found out. Note that the mass of vapour m_g at Sects. (1)–(1), (2)–(2) and (3)–(3) remains constant.

Now we can be enable to calculate the dryness fraction of the wet steam at state 1 (i.e., sample steam),

Dryness fraction at state 1,

$$x_1 = \frac{\text{mass of the vapor}}{\text{mass of the wet stream}}$$
$$x_1 = \frac{m_g}{m}$$

where

$$m_g = x_2 m_2$$

and

$$m = m_1 + m_2$$
$$\therefore\ x_1 = \frac{x_2 m_2}{m_1 + m_2}$$

Q.27. Derive Maxwell's relation and write the physical significance of each equation.

Ans. The equations which relate the partial derivatives of properties p,v, T and S to each other for a compressible fluid are called the maxwell relations. Four maxwell relations are derived as following:

1. First Maxwell Relation:

According to the first law of thermodynamics for a process.

$$\delta Q = dU + \delta W$$
$$Tds = dU + pdV \quad \therefore\ \delta Q = TdS, \delta W = pdV$$

For a unit mass

$$Tds = dU + pdV$$

or

$$du = Tds - pdv \tag{16}$$

The above Eq. (16) is in the form

$$dz = Mdx + Ndy$$

where

$$M = T$$
$$x = s$$
$$N = -p$$

and

$$y = v$$

$$\therefore \left(\frac{\partial T}{\partial v}\right)_s = \left(\frac{\partial p}{\partial s}\right)_v \tag{17}$$

Equation (17) is known as the first Maxwell relation.

2. Second Maxwell Relation.

By definition of enthalpy

$$H = U + pV$$

For a unit mass.

$$h = u + pv \tag{18}$$

On differenting Eq. (18), we get

$$dh = du + pdv + vdp$$

Calling Eq. (16)

$$du = Tds - pdv$$

$$du - pdv = Tds$$

$$\therefore\ dh = Tds + vdp \tag{19}$$

The above Eq. (19) is in the form

$$dz = Mdx + Ndy$$

where

$$M = T$$

$$x = s$$

$$N = v$$

and

$$y = p$$

$$\therefore \left(\frac{\partial T}{\partial p}\right)_s = \left(\frac{\partial v}{\partial s}\right)_p \tag{20}$$

Equation (20) is known as the second Maxwell relation.

3. Third Maxwell Relation:

By definition of Helmholtz function

$$A = U - TS$$

For a unit mass

$$a = u - Ts \tag{21}$$

On differentiating Eq. (21), we get

$$da = du - Tds - sdT$$

Calling Eq. (16)

$$du = Tds - pdv$$

or

$$du - Tds = -pdv$$

$$da = -pdv - sdT \tag{22}$$

The above Eq. (22) is in the form

$$dz = Mdx + Ndy$$

where

$$M = -p$$

$$x = v$$

$$N = -s$$

and

$$y = T$$

$$\therefore\ -\left(\frac{\partial p}{\partial T}\right)_v = -\left(\frac{\partial s}{\partial v}\right)_T$$

or

$$\left(\frac{\partial p}{\partial T}\right)_v = \left(\frac{\partial s}{\partial v}\right)_T \tag{23}$$

Equation (23) is known as the third Maxwell relation.

4. Fourth Maxwell Relation:

By definition of Gibbs function

$$G = H - TS$$

For a unit mass

$$g = h - Ts \tag{24}$$

On differentiating Eq. (24), we get

$$dg = dh - Tds - sdT$$

Calling Eq. (19)

$$dh = Tds + vdp$$

or

$$dh - Tds = vdp$$

$$\therefore\ dg = vdp - sdT \tag{25}$$

The above Eq. (25) is in the form

$$dz = Mdx + Ndy$$

where

$$M = v$$
$$x = p$$
$$N = -s$$

and

$$y = T$$

$$\therefore \left(\frac{\partial v}{\partial T}\right)_p = -\left(\frac{\partial s}{\partial p}\right)_T \tag{26}$$

Equation (26) is known as the fourth Maxwell relation.

Equations (17), (20), (23) and (26) are the four Maxwell relations. They are very useful relations only for simple compressible fluids. They are used to determine the entropy change, which cannot be measured directly, by simply measuring the change in properties p, v and T.

Q.28. Steam at 550 °C and 150 bar enters into steam turbine and condenser is maintained at 0.1 bar. Assuming ideal process, find the gain in efficiency if condenser pressure is reduced to 0.07 bar.

Also, draw schematic *T–s* diagram.

Ans. Given data:
At inlet of the turbine

$$T_1 = 550^\circ\mathrm{C}$$
$$p_l = 150\,\mathrm{bar}$$

From saturated steam table (pressure based),

at

$$p_l = 150\,\mathrm{bar}$$
$$T_{\mathrm{sat}} = 342.2\,^\circ\mathrm{C}$$

So, $T_x > T_m$, it means the steam is superheated at the turbine inlet. From superheated steam table,

at

$$P_1 = 150\,\mathrm{bar},\ T_1 = 550\,^\circ\mathrm{C}$$

we get

$$h_1 = 3448.6\ \mathrm{kJ/kg}$$
$$s_1 = 6.5199\ \mathrm{kJ/kgK}$$

Condenser pressure:

$$p_2 = 0.1\,\mathrm{bar}$$

From saturated steam table (pressure based)

at

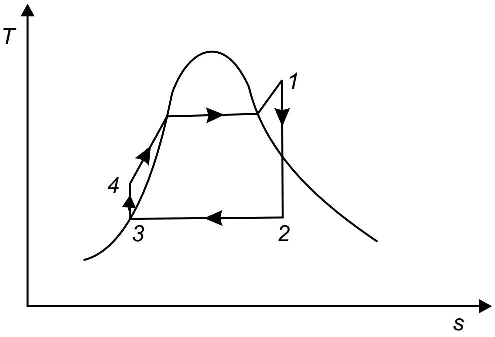
$$p_2 = 0.1\,\mathrm{bar}$$

we get

$$s_f = 0.649\ \mathrm{kJ/kgK}, s_{fg} = 7.501\ \mathrm{kJ/kgK},$$
$$s_g = 8.15\ \mathrm{kJ}$$
$$h_f = 191.8\ \mathrm{kJ/kg}, h_{fg} = 2392.8\ \mathrm{kJ/kg}$$
$$v_f = 0.00101\,\mathrm{m^3/kg}$$
$$s_2 = s_1 = 6.5199\,\mathrm{kJ/kgK}$$

As s_2. < s_g,. It means the condition of steam is wet at turbine exit

$$\begin{aligned} s_2 &= s_f + x_2 s_{fg} \\ 6.5199 &= 0.649 + x_2 x 7.501 \\ x_2 &= 0.7826 \\ h_2 &= h_f + x_2 h_M \\ &= 191.8 + 0.7826 x 2392.8 \\ &= 2064.40 \text{ kJ/kg} \end{aligned}$$



$$h_3 = 191.8\,\text{kJ/kg}$$

$$v_3 = 0.00101\,\text{m}^3/\text{kg}$$

Pump work:

$$w_{3-4} = w_p = v_1(p_1 - p_2)\text{kJ/kg}$$

where

v, is in m^3/kg.

p_l and p_2 are in kPa

$$w_p = 0.00101\,(15000 - 10) = 15.139\,\text{kJ/kg}$$

also

$$w_p = h_4 - h_3$$

$$\therefore\ 15.139 = h_4 - 191.8$$

or

$$h_4 = 15.139 + 191.8 = 206.939\,\text{kJ/kg}$$

Turbine work:

$$\begin{aligned} w_{12} = w_T &= h_l - h_2 \\ &= 3448.6 - 2064.40 \\ &= 1384.2\,\text{kJ/kg} \end{aligned}$$

Net work output:

$$\begin{aligned} w_{net} &= w_T - w_p \\ &= 1384.2 - 15.139 = 1369.6\,\text{kJ/kg} \end{aligned}$$

Heat supplied:

$$\begin{aligned} q_{4-1} &= h_1 - h_4 \\ &= 3448.6 - 206.939 \\ &= 3241.66\,\text{kJ/kg} \end{aligned}$$

Thermal efficiency:

$$\eta_{th} = \frac{w_{net}}{q_{4-1}} = \frac{1369.06}{3241.66} = 0.4223$$

Case II: Condenser pressure is reduced to 0.07 bar.

i.e.,

$$p_2 = 0.07\,\text{bar}$$

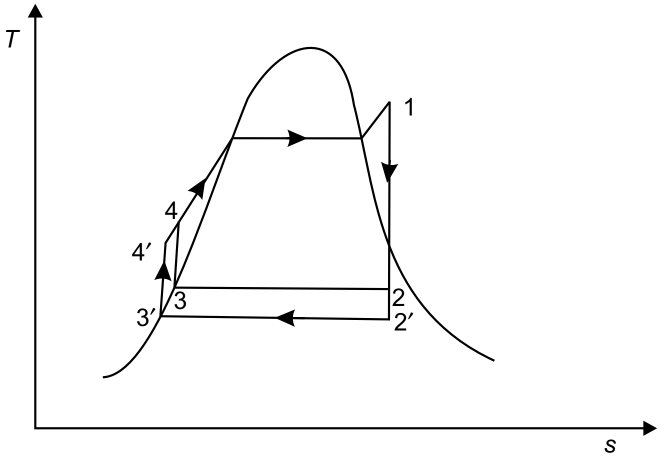
From saturated steam table (pressure based).

At

$$p_2 = 0.07\,\text{bar}$$

we get

$$s_f = 0.559\,\text{kJ/kgK}, s_{fg} = 7.717\,\text{kJ/kgK}$$



$$\begin{aligned}
h_{f'} &= 163.4\,\mathrm{kJ}/kg,\ h_{fg'} = 2409.1\,\mathrm{kJ/kg}\\
v_{f'} &= 0.001007\,\mathrm{m^3/kg}\\
s_{g'} &= g_{f'} + x_{g'} s_{fg'}\\
6.5199 &= 0.559 + x_{2'} \times 7.717
\end{aligned}$$

or

$$x_{2'} = 0.7724$$

$$\begin{aligned}
h_{2'} &= h_{f'} + x_{2'} h_{fg'}\\
&= 163.4 + 0.7724\,x\,2409.1\\
&= 2024.18\,\mathrm{kJ/kg}\\
h_{3'} &= h_{f'} = 163.4\,\mathrm{kJ/kg}\\
v_{3'} &= v_{f'} = 0.001007\,\mathrm{m^3/kg}
\end{aligned}$$

Pump work:

$$w_{3-4} = w'_p = v_{3'}\,(p_1 - p_2)\,\mathrm{kJ/kg}$$

where $v_{3'}$ is in m^3/kg
p_1 and p_2 are in kPa

$$w'_p = 0.001007(15000 - 7) = 15.098\,\mathrm{kJ/kg}$$

also

$$\begin{aligned}
w_{p'} &= h_{4'} - h_{3'}\\
15.098 &= h_{4'} - 163.4
\end{aligned}$$

or

$$h_{4'} = 178.49\,\mathrm{kJ/kg}$$

Turbine work:

$$\begin{aligned}
w_{1-2} &= w'_T = \mathrm{h}_1 - h_{2'}\\
&= 3448.6 - 2024.18 = 1424.42\ \mathrm{kJ/kg}
\end{aligned}$$

Net work output:

$$\begin{aligned}
w_{\mathrm{net}} &= w'_T - w'_p\\
&= 1424.42 - 15.098 = 1409.32\mathrm{kJ/kg}
\end{aligned}$$

Heat supplied:

$$\begin{aligned}
q_{4-1} &= h_1 - h_4 = 3448.6 - 178.49\\
&= 3270.11\mathrm{kJ/kg}
\end{aligned}$$

Thermal efficiency

$$\eta'_{\mathrm{th}} = \frac{w'_{\mathrm{net}}}{q_{4-1}} = \frac{1409.32}{3270.11} = 0.4309$$

$$\begin{aligned}
\text{Gain in thermal efficiency} &= \frac{\eta'_{th} - \eta_{th}}{\eta'_{th}}\\
&= \frac{0.4309 - 0.4223}{0.4309}\\
&= 0.0199 = 1.99\%
\end{aligned}$$

Q.29. In a gas turbine working on Brayton cycle air enter into compressor at 1 bar and 27 °C. The pressure ratio is 8 and maximum temperature is 900 °C. The turbine and compressor efficiencies are taken as 90% and 80%, respectively. Calculate cycle efficiency. Assume Y = 1.4 and c_p = 1.005 kJ/kg K.

Ans: Given data

At inlet of compressor:

$$p = \text{bar}$$
$$T_x = 27°\text{C} = (27 + 273)\text{K} = 300\,\text{K}$$

Pressure ratio:

$$r_p = \frac{p_2}{p_1} = 8$$

Compressor efficiency:

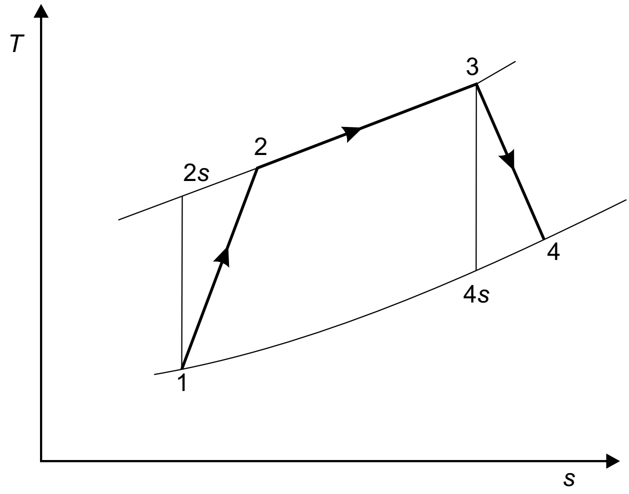
$$\eta_c = 80\% = 0.8$$
$$\gamma = 1.4$$
$$c_p = 1.005\,\text{kJ/kgK}$$

For process 1–2 s

$$\frac{T_{2s}}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$
$$\frac{T_{2s}}{300} = (8)^{\frac{1.4-1}{1.4}}$$

or

$$T_{2s} = 300\,x\,(8)^{0.285} = 542.62\,\text{K}$$



Maximum temperature:

$$T_3 = 900°\text{C} = (900 + 273)\text{K} = 1173\,\text{K}$$

Turbine efficiency:

$$\eta_r = 90\% = 0.9$$

$$\eta_c = \frac{\text{isentropic increase in temperature}}{\text{actual increase in temperature}}$$
$$= \frac{T_{2s} - T_1}{T_2 - T_1}$$
$$0.8 = \frac{542.62 - 300}{T_2 - 300} = \frac{242.62}{T_2 - 300}$$

or

$$T_2 - 300 = \frac{242.62}{0.8} = 303.27$$
$$T_2 = 603.27\,\mathrm{K}$$

For process 3–4*s*,

$$\frac{T_3}{T_{4s}} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}}$$
$$\frac{1173}{T_{4s}} = (8)^{\frac{1.4-1}{1.4}}$$
$$\frac{1173}{T_{4s}} = (8)^{0.285}$$

or

$$T_{4s} = \frac{1173}{(8)^{0.285}} = 648.51\mathrm{K}$$
$$\eta_T = \frac{\text{actual decrease in temperature}}{\text{isentropic decrease in temperature}}$$
$$= \frac{T_3 - T_4}{T_3 - T_{4s}}$$
$$0.9 = \frac{1173 - T_4}{1173 - 648.51} = \frac{1173 - T_4}{524.49}$$

or

$$1173 - T_4 = 0.9 \times 524.94 = 472.04$$

or

$$T_4 = 700.96\,K$$

Turbine output work:

$$w_{3-4} = w_T = h_3 - h_4$$
$$= c_p(T_3 - T_4)$$
$$= 1.005\,(1173 - 700.96) = 474.40\,\mathrm{kJ/kg}$$

Compressor input work:

$$w_{1-2} = w_c = h_2 - h_1$$
$$= c_p(T_2 - T_1)$$
$$= 1.005\,(603.27 - 300) = 304.78\,\mathrm{kJ/kg}$$

Net work output:

$$w_{\text{net}} = w_T - w_C$$
$$= 474.40 - 304.78 = 164.62\mathrm{kJ/kg}$$

Heat supplied:

$$q_{2-3} = h_3 - h_2$$
$$= c_p(T_3 - T_2)$$
$$= 1.005\,(1173 - 60327) = 569.73\,\mathrm{kJ/kg}$$

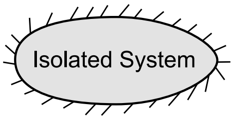
Cycle efficiency:

$$\eta = \frac{\text{net work output}}{\text{heat supplied}}$$
$$= \frac{w_{\text{net}}}{q_{2-3}} = \frac{169.62}{569.73} = 0.2977 = 29.77\%$$

Q.30. (*a*) Explain Isolated System with an Example.

Ans. The isolated system in which both mass and energy cannot cross the boundary of the system.

For example: Hot coffee in a thermos flask.



(*b*) Explain mechanical, chemical and thermal equilibrium.

Ans. Mechanical equilibrium:

A system is in mechanical equilibrium if there is no unbalanced force in the system and also between the system and its surrounding. In other words, the pressure in the system is same at all points and does not change with respect to time, such type of equilibrium is called mechanical equilibrium.

Chemical equilibrium:

If there is no chemical reactions occur in the system and chemical composition is same throughout the system, does not vary with respect to time. Such type of equilibrium is called chemical equilibrium.

Thermal equilibrium:

If the temperature of the system does not change with respect to time and has same value at all points in the system, such type of equilibrium is called thermal equilibrium.

(*c*) Under what condition is the work done equal to $\int_1^2 pdv$?

Ans. Work done:

$$W_{1-2} = \int_1^2 pdv \tag{27}$$

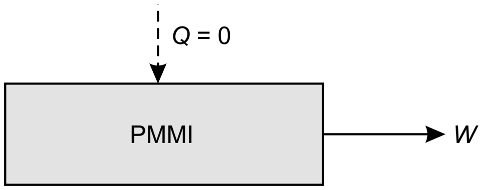
where W_{1-2} is called displacement work.

Equation (27) is used for reversible non-flow process.

(*d*) What is a PMM I? Why it is impossible?

Ans. A perpetual motion machine of the first kind (PMM I) is an imaginary machine that produces work continuously without any input, i.e., PMM I is energy creating machine as shown in Fig. 1.

PMM I is energy creating machine. It means that PMM I violates the first law of thermodynamics. Hence, it is impossible machine.



(*e*) Why does entropy remains constant in reversible adiabatic process?

Ans. The entropy remains constant in reversible adiabatic process because (i) this process is ideal or frictionless process, and (ii) no heat transfer between the system and surroundings takes place.

(*f*) What do you understand by steam rate and heat rate? What are their units?

Ans. Steam rate: The capacity of a steam power plant is expressed in terms of steam rate, which is defined as the mass flow rate of steam (kg/h) required to produce unit power output (I kW). M

Mathematically,

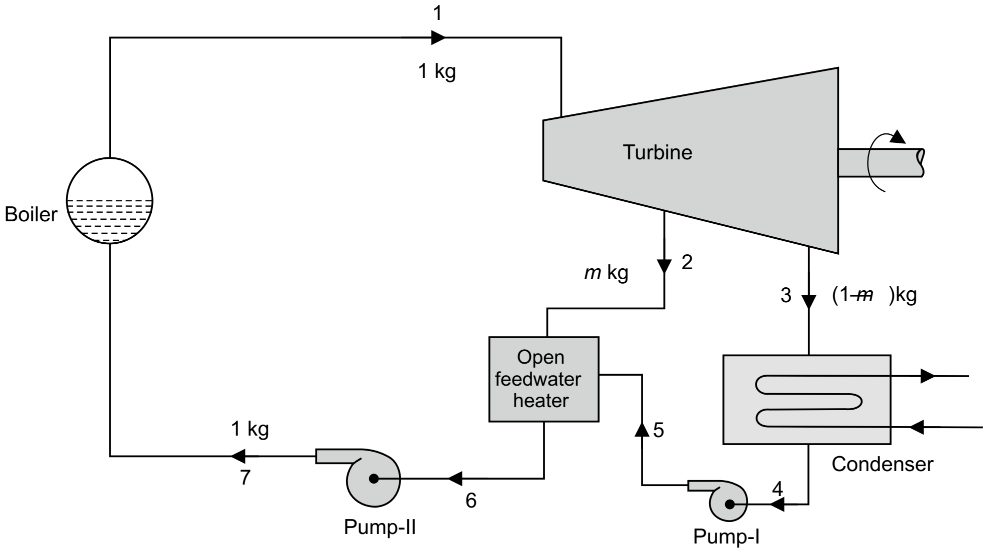
$$\text{Steamrate} = \frac{m}{W_{Net}}\,\frac{\text{kg/h}}{\text{kW}} = \frac{m}{W_{\text{Net}}}\,\frac{\text{kg}}{\text{kWh}}$$

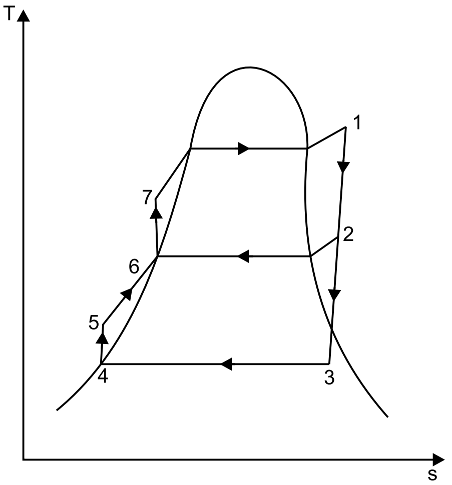
Units of steam rate is kg/kWh.

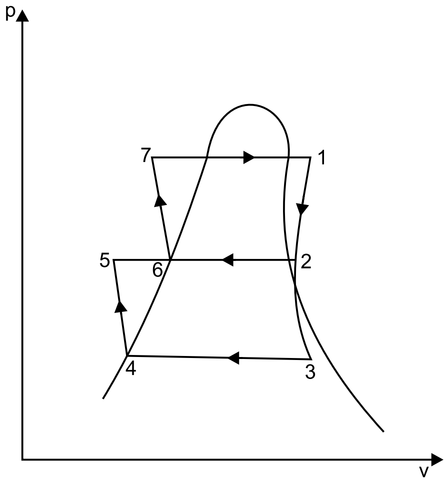
Heat rate: Heat rate means heat flow rate. It is defined as the heat flow per unit time across the section. SI units of heat rate is kJ/s or kW.

(*g*) Sketch regenerative Rankine cycle on *T–s* and *p–v* planes.

Ans. A schematic representation of the main components of regenerative Rankine cycle is shown in Fig. (i). The water entering the boiler is called feed water and the device used to mix the extracted steam and the condenser water is called a regenerator or a feedwater heater. The *T–s* and *p–v* diagrams of this cycle are shown in Fig. (ii).







(*h*) What do you understand by energy and anergy?

Ans. Energy: It is defined as the ability to do work. Energy can be classified into two categories:

(i) Stored energy:

(a) Kinetic energy
(b) Potential energy
(c) Internal energy

(ii) Transit energy:

(a) Heat energy.
(a) Work.

Anergy: Anergy is called irreversibility. It is defined as the difference between the reversible useful work and the actual useful work.

Mathematically,

$$\text{Anergy} = W_{\text{u.rev}} - W_{\text{u}}$$

For reversible process, $W_{\text{u.rev}} = W_{\text{u}}$. Hence, energy is zero.

(i) Give the Clausius statement of second law of thermodynamics.

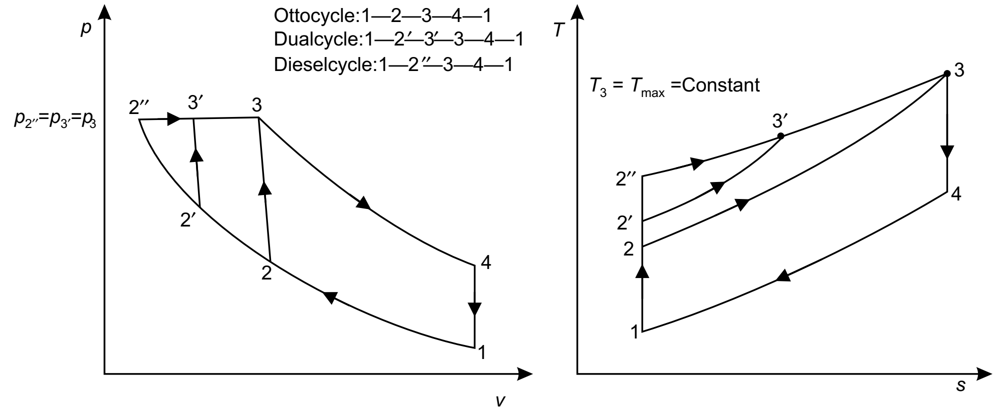
Ans. Clausius statement: It is impossible to construct a device that operates in a cycle and produces no effect other than the transfer of heat from a low temperature body to a high temperature body without external aid of work.

(*j*) With the help of *p–v* and *T–s* diagrams, show that for the same maximum pressure and temperature of the cycle and the same heat rejection.

$$\eta_{\text{Diesel}} > \eta_{\text{Dual}} > \eta_{\text{otto}}$$

Ans. The Otto, dual and diesel cycles are shown in *p–v* and *T–s* diagrams of Fig. 1 according to given condition of the same maximum pressure, maximum temperature and heat rejection. These cycles are shown in Fig. 11.7 as

Otto cycle: 1–2–3–4–1
Dual cycle: 1–2′–3′–3–4–1
Diesel cycle: 1–2″–3–4–1



The thermal efficiency:

$$\eta = 1 - \frac{\text{heat rejection}}{\text{heat supplied}}$$

As the heat rejection is constant. The thermal efficiency of the cycle is dependent only on the heat supplied, higher the heat supplied, higher the thermal efficiency.

It is clear from *T–s* diagram of Fig. 1 that the heat supplied is lower in the Otto cycle and higher in the diesel cycle. The amount of heat supplied in diesel cycle is more than heat supplied in Otto cycle and is equal to area under 2″–3–2–2″. The heat supplied to the dual cycle lies between the Otto and the diesel cycle.

i.e.,

$$(Q_{2''-3})_{\text{Diesel}} > (Q_{2'-3'-3})_{\text{Dual}} > (Q_{2-3})_{\text{Otto}}$$

Thus, the thermal efficiency of the diesel cycle is maximum, the thermal efficiency of the Otto cycle is minimum and the dual cycle's thermal efficiency lies between the Otto and the diesel cycles, at given condition of the same maximum pressure, maximum temperature and heat rejection.

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

Q.31. In a steady flow apparatus, 135 kJ of work is done by each kg of fluid. The specific volume of fluid, pressure and velocity at the inlet are 0.37 m^3/kg, 600 kPa and 16 m/s. The inlet is 32 m above the floor and the discharge pipe is at floor level. The discharge conditions are 0.62 m^3/kg, 100 kPa and 270 m/s. The total heat loss between the inlet and discharge is 9 kJ/kg of fluid. In flowing through this

apparatus, does the specific internal energy increase or decrease and by how much?

Ans. Given data:

Work done by the system:

$$w_{1-2} = 135\text{kJ/kg}$$

At inlet:

$$\begin{aligned} \text{Specific volume} : v_1 &= 0.37\text{m}^3/\text{kg} \\ \text{Pressure} : p_1 &= 600\,\text{kPa} \\ \text{Velocity} : V_1 &= 16\text{m/s} \\ \text{Datum head} : z_1 &= 32\text{m} \end{aligned}$$

At discharge:

$$\begin{aligned} \text{Specific volume} : v_2 &= 0.62\text{m}^3/\text{kg} \\ \text{Pressure} : p_2 &= 100\,\text{kPa} \\ \text{Velocity} : V_2 &= 270\text{m/s} \\ \text{Datum head} : z_2 &= 0 \end{aligned}$$

Total heat loss between the inlet and discharge:

$$q_{1-2} = -9\,\text{kJ/kg}$$

Let the units of h_1 and h_2 in kJ/kg, units of q_{1-2} and w_{1-2} is kJ/kg (given).

The units of $\frac{V_1^2}{2}$, gz_1 and $\frac{V_2^2}{2}$ are made in kJ/kg by dividing 1000.

The above equation is written as:

$$h_1 + \frac{V_1^2}{2000} + \frac{gz_1}{1000} + q_{1-2} = h_2 + \frac{V_2^2}{2000} + w_{1-2}$$

or

$$\begin{aligned} h_2 - h_1 &= \frac{\left(V_1^2 - V_2^2\right)}{2000} + \frac{gz_1}{1000} + q_{1-2} - w_{1-2} \\ &= \frac{(16)^2 - (270)^2}{2000} + \frac{9.81 \times 32}{1000} - 9 - 13 \\ &= -36.32 + 0.31 - 9 - 135 \\ h_2 - h_1 &= -180.01\,\text{kJ/kg} \end{aligned}$$

By definition of specific enthalpy:

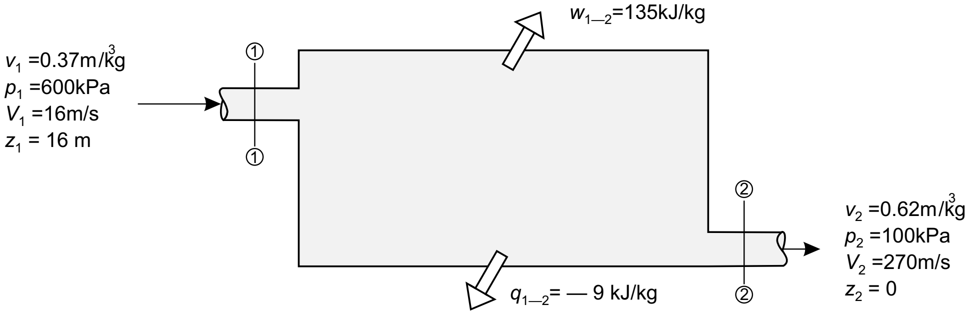
$$h = u + pv$$

At inlet:

$$h_1 = u_1 + p_1 v_1$$

At outlet:

$$h_2 = u_2 + p_2 v_2$$



Now applying the steady flow energy equation, we have
where $z_2 = 0$

$$h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + w_{1-2}$$

$$\therefore\ h_1 + \frac{V_1^2}{2} + gz_1 + q_{1-2} = h_2 + \frac{V_2^2}{2} + w_{1-2}$$

$$\therefore\ u_2 + p_2 v_2 - (u_1 + p_1 v_1) = -180.01$$

$$u_1 - u_1 = -180.01 - p_2 v_2 + p_1 v_1$$

$$= -180.01 - 100x0.62 + 600x0.37$$

$$-180.01 - 62 + 222 = -20.01\ \text{kJ/kg}$$

The −ve sign indicates that the specific internal energy decreases by 20.01 kJ/kg.

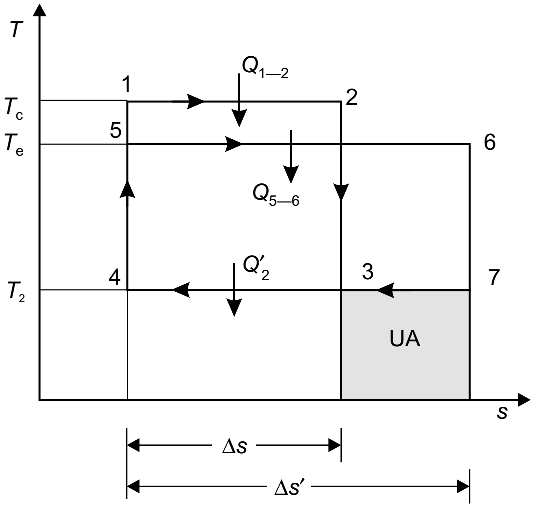
Q.32. In a certain process, a vapor, while condensing at 420 °C, transfers heat to water evaporating at 250 °C. The resulting steam is used in a power cycle which rejects heat at 35 °C. What is the function of the available energy in the heat transferred from the process vapour at 420 °C that is lost due to the irreversible heat transfer at 250 °C?

Ans. Give data:

$$\text{Condensing temperature} : T_c = 420\,^\circ\text{C} = (420 + 273)\text{K} = 693\text{K}$$

$$\text{Evaporating temperature} : T_c = 250\,^\circ\text{C} = (250 + 273)\text{K} = 523\text{K}$$

$$\text{Temperature of heat rejected} : T_2 = 35\,^\circ\text{C} = (35 + 273)\text{K} = 308\text{K}$$



By energy balance equation:

Heat loss during condensation
= Heat gain during evaporation

$$Q_{1-2} = Q_{5-6}$$

Area under process 1 − 2
= Area under process 5 − 6

$$T_c\Delta S = T_e\Delta S'$$

or

$$\frac{\Delta S'}{\Delta S} = \frac{T_c}{T_e} = \frac{693}{523} = 1.325$$

Heat rejected during process 3 − 4 : Q_{3-4} = area under process 3 − 4 $= T_2\Delta S$

Heat rejected during process 7 − 4 : Q_{7-4} = area under process 7 − 4 $= T_2\Delta S'$

Work done during cycle 1–2–3–4:

$$\begin{aligned} W_1 &= Q_{1-2} - Q_{3-4} \\ &= T_c\Delta S = T_2\Delta s \\ &= (T_c - T_2)\Delta S \end{aligned}$$

Work done during cycle 5–6–7–4:

$$\begin{aligned} w_2 &= Q_{5-6} - Q_{7-4} \\ &= T_e\Delta s' - T_2\Delta s' = (T - T_2)\Delta s' \end{aligned}$$

The fraction of energy becomes unavailable due to irreversible heat transfer

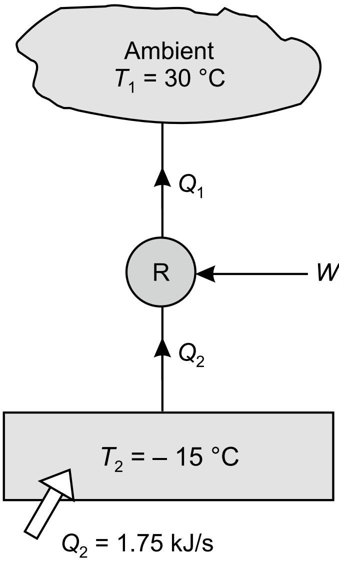
$$\begin{aligned} &= \frac{W_1 - W_2}{W_1} \\ &= \frac{(T_c - T_2)\Delta S - (T_e - \mathrm{T}_2)\Delta S}{(T_c - T_2)\Delta S} \\ &= \frac{T_c\Delta S - T_2\Delta S - T_e\Delta S' + T_2\Delta S'}{(T_c - T_2)\Delta S} \\ &= \frac{T_2\Delta S' - T_2\Delta S}{(T_c - T_2)\Delta S} \quad \because T_e\Delta S = T_e\Delta S^\circ \\ &= \frac{(\Delta S'/\Delta S - 1)T_2\Delta S}{(T_c - T_2)\Delta S} = \frac{(\Delta S'/\Delta S - 1)T_2}{(T_c - T_2)} \\ &= \frac{(1.325 - 1) \times 308}{693 - 308} \quad \because \frac{\Delta S'}{\Delta S} = 1.325 \\ &= 0.26 \end{aligned}$$

Q.33. A domestic food freezer maintains a temperature of −15 °C. The ambient air temperature is 30 °C. If heat leaks into the freezer at the continuous rate of 1.75 kJ/s. What is the least power necessary to pump this heat out continuously?

Ans. Given data for refrigerator.

Temperature of freezer:

$$T = -15\,^\circ\text{C} = (-15 + 273)\ \text{K} = 258\ \ \text{K}$$



Ambient temperature:

$$T_1 = 30\,^\circ\text{C} = (30 + 273)\text{K} = 303\text{K}$$

Heat leak into freezer:

$$Q_2 = 1.75\text{kJ/s} = 1.75\text{kW}$$

Condition for the least power required: $W_{\min}$

$$\text{Carnot COP} = \text{Actual COP}$$

$$\frac{T_2}{T_1 - T_2} = \frac{Q_2}{W_{\min}}$$

$$\frac{258}{303 - 258} = \frac{1.75}{W_{\min}}$$

or

$$W_{\min} = 0.3052\,\text{kW}$$

Q.34. What is the deficiency of the first law efficiency? How does the second law efficiency make up this deficiency?

Ans. To measure the performance of any process, device, or system, we make use of the concept of efficiency. The first law efficiency (also called thermal efficiency) widely used in thermodynamics is usually based on the concept of energy, in which no attempt is made to distinguish law grade energy from high grade energy. A simple example is the first law efficiency of a heat engine, which is defined as the ratio of net work output to the amount of heat supplied. In this definition, heat and work are given the same wattage. So, the first law efficiency does not give an accurate measure of thermodynamic performance. To overcome this efficiency, we defined a second-law efficiency as the ratio of the actual work output to the maximum (reversible) work output of the work-producing devices mathematically,

Second-law efficiency:

$$\eta_{\text{II}} = \frac{\text{actual work} : W}{\text{maximum work} : W_{\text{rev}}}$$

$$\eta_{\text{II}} = \frac{W}{W_{\text{rev}}} \qquad \text{for work - production device}$$

$$\eta_{\text{II}} = \frac{W}{W_{\text{rev}}} \times \frac{Q_1}{Q_1}$$

$$= \frac{W}{Q_1 \times W_{\text{rev}}/Q_1}$$

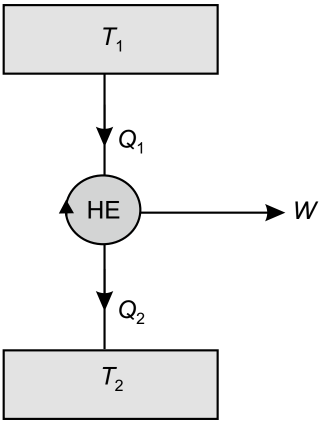
$$\eta_{\text{II}} = \frac{\eta_{\text{th}}}{\eta_{\text{rev}}} = \frac{\eta_1}{\eta_{\text{Carnot}}}$$

where

$$\eta_{\text{rev}} = \eta_{\text{Carnot}} = \frac{W_{\text{rev}}}{Q_1}$$

and

$$\eta_{\text{th}} = \eta_1 = \frac{W}{Q_1}$$



The second law efficiency is also defined as the ratio of actual thermal efficiency to the reversible (Carnot) efficiency under the same condition.

It is important to note that the upper limit of the second law efficiency is 100%, which corresponds to the ideal case with no energy destruction.

Q.35. Write down the first and second *TdS* equation, and derive the expression for the difference in heat capacities, C_p and C_y. What does the expression signify?

Ans. 1st *Tds* equation,

$$Tds = C_v dT + T\left(\frac{\partial p}{\partial T}\right)_v dv$$

or

$$ds = C_v \frac{dT}{T} + \left(\frac{\partial p}{\partial T}\right)_v dv \tag{28}$$

2nd *Tds* equation,

$$Tds = C_p dT - T\left(\frac{\partial V}{\partial T}\right)_p dp$$

or

$$ds = C_p \frac{dT}{T} - \left(\frac{\partial V}{\partial T}\right)_p dp \tag{29}$$

Equating Eqs. (28) and (29), we get

$$C_p \frac{\partial T}{T} - \left(\frac{\partial v}{\partial T}\right) dp = C_v \frac{\partial T}{T} + \left(\frac{\partial p}{\partial T}\right)_v dV$$
$$(C_p - C_v)\frac{\partial T}{T} = \left(\frac{\partial p}{\partial T}\right) dV + \left(\frac{\partial V}{\partial T}\right)_p dp$$

or

$$dT = \frac{T(\partial p/\partial r)}{C_p - C_v} dV + \frac{T\left(\frac{\partial V}{\partial T}\right)_p dp}{C_p - C_v} \tag{30}$$

The temperature T is function of V and p, i.e.,

$$T = f(V, p)$$

In different form, we can write

$$dT = \left(\frac{\partial T}{\partial V}\right)_p dv + \left(\frac{\partial T}{\partial p}\right)_v dp \tag{31}$$

Comparing Eqs. (30) and (31), we get

$$\left(\frac{\partial T}{\partial V}\right)_p dv = \frac{T}{(C_p - C_v)}\left(\frac{\partial p}{\partial T}\right)_v$$
$$\text{or} \quad C_p - C_v = \frac{T(\partial p/\partial T)_v}{(\partial T/\partial V)_p}$$
$$C_p - C_v = T\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial V}{\partial T}\right)_p$$
$$\left(\frac{\partial T}{\partial p}\right)_v = \frac{T}{(C_p - C_v)}\left(\frac{\partial V}{\partial T}\right)_p$$
$$\text{or} \quad C_p - C_v = \frac{T(\partial V/\partial T)_p}{(\partial T/\partial p)_V}$$
$$C_p - C_v = T\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial V}{\partial T}\right)_p$$

$$C_p - C_v = T\left(\frac{\partial p}{\partial T}\right)_v \left(\frac{\partial V}{\partial T}\right)_p \qquad (32)$$

We know that the cyclic relation:

$$\left(\frac{\partial p}{\partial T}\right)_T \left(\frac{\partial V}{\partial T}\right)_p \left(\frac{\partial T}{\partial p}\right)_V = -1$$

$$\left(\frac{\partial p}{\partial V}\right)_T \left(\frac{\partial v}{\partial T}\right)_p = \frac{1}{\left(\frac{\partial T}{\partial p}\right)_V} = \left(\frac{\partial p}{\partial T}\right)_V$$

or

$$\left(\frac{\partial p}{\partial T}\right)_v = -\left(\frac{\partial p}{\partial V}\right)_r \left(\frac{\partial v}{\partial T}\right)_p \qquad (33)$$

Substituting the value of $\left(\frac{\partial p}{\partial T_v}\right)$ from Eq. (33) in Eq. (32), we get

$$C_p - C_v = -T\left(\frac{\partial p}{\partial V}\right)_T \left(\frac{\partial V}{\partial T}\right)_p \left(\frac{\partial V}{\partial T}\right)_p$$

$$C_p - C_v = -T\left(\frac{\partial p}{\partial V}\right)_T \left(\frac{\partial V}{\partial T}\right)_p^2 \qquad (34)$$

$$C_p - C_v = \frac{TV^2\left[\frac{1}{V^2}\left(\frac{\partial V}{\partial T}\right)_P^2\right]}{V\left[-\frac{1}{V}\left(\frac{\partial V}{\partial p}\right)_T\right]}$$

$$C_p - C_v = \frac{TV\beta^2}{k_T} \qquad (35)$$

where

$$\beta = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_P, \text{ volume expansively}$$

and

$$k_T = \frac{1}{V}\left(\frac{\partial V}{\partial T}\right)_T, \text{isothermal compressibility}$$

Equation (35) gives the following information:

1. The difference between C_P and C_v is zero at absolute zero temperature for all material (liquid, gas or solid).,

   i.e.,

   $$C_P - C_v = 0 \text{ at } T = 0\,\text{K}$$

2. The difference between C_P and C_v is always +ve because $\left(\frac{\partial V}{\partial T}\right)_P^2$ is always +ve and $\left(\frac{\partial p}{\partial V}\right)_T$ is −ve for all compressible fluids.
3. For a liquid and solid, $\left(\frac{\partial p}{\partial V}\right)_P$ is small and difference between C_P and C_v is small.
   when $\left(\frac{\partial V}{\partial T}\right)_p = 0$, $C_p = C_v$ as true at the point of maximum density of water at 4 °C.

Q.36. The following data were obtained with a separating and throttling calorimeter:

Pressure in pipe line = 1.5 MPa

Condition after throttling = 0.1 MPa, 110°C

During 5 min, moisture collected in the separator = 0.150 L at 70 °C.

Steam condensed after throttling during 5 min = 3.24 kg.

Find the quality of steam in the pipeline.

Ans. Given data

Pressure in pipe line:

$$p_1 = 1.5\,\text{MPa} = 15\,\text{bar}$$

Pressure after throttling:

$$p_3 = 0.1\,\text{MPa} = 1\,\text{bar}$$

Temperature after throttling:

$$T_3 = 100\,^\circ\text{C}$$

Volume of moisture collected in the separator:

$$V = 0.150\,\text{litre at } 70\,^\circ\text{C} = \frac{0.150}{1000}\,\text{m}^3$$

Mass of steam condensed after throttling:

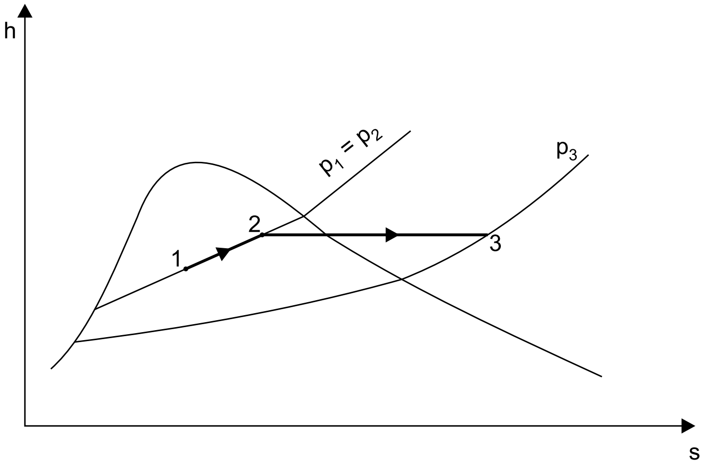
$$m_2 = 3.24\,\text{kg}$$

From superheated steam table,
At

$$p_3 = 1\,\text{bar},\ \text{T}_3 = 100^\circ\text{C}$$
$$h_3 = 2676.2\,\text{kJ/kg}$$

From saturated steam table on pressure based,
At

$$p_1 = p_2 = 15\,\text{bar}$$
$$h_\text{f} = 844.9\,\text{kJ/kg}$$
$$h_\text{g} = 2792.2\,\text{kJ/kg}$$
$$h_\text{fg} = 1947.3\,\text{kJ/kg}$$



Specific enthalpy at point 2,

$$h_2 = h_f + x_2 h_{fg}$$

also

$$h_2 = h_3$$
$$\therefore\ h_3 = h_f + x_3 h_{fg}$$

$$2676.2 = 844.9 + x_2 \times 1947.3$$

or

$$1947.3\,x_2 = 1831.3$$

or

$$1947.3x_2 = 1831.3$$

or

$$x_2 = \frac{1831.3}{1947.3} = 0.94$$

From saturated steam table on temperature based,

At

$$T = 70\,^\circ\text{C}$$
$$v_f = 0.001023\,\text{m}^3/\text{kg}$$

also

$$v_f = \frac{V}{m_1}$$

$$\therefore\ 0.001023 = \frac{0.150}{1000} \times \frac{1}{m_1}$$

or

$$m_1 = 0.1466\,\text{kg}$$

We know that the dryness fraction or quality of the steam in the pipeline at state 1,

$$x_1 = \frac{x_2 m_2}{m_1 + m_2} = \frac{0.94 \times 3.24}{0.1466 + 3.24} = 0.8993$$
$$= 89.93\%$$

Q.37. An engine working on the Otto cycle is supplied with air to 0.1 MPa, 35 °C. The compression ratio is 8. Heat supplied is 2100 kJ/kg. Calculate the maximum pressure and temperature of the cycle, the cycle efficiency and the mean effective pressure. (For air c_p = 1.005 kJ/kgK, c_v = 0.718 kJ/kgK and R = 0.287 kJ/kgK).

Ans. Given data for Otto cycle:

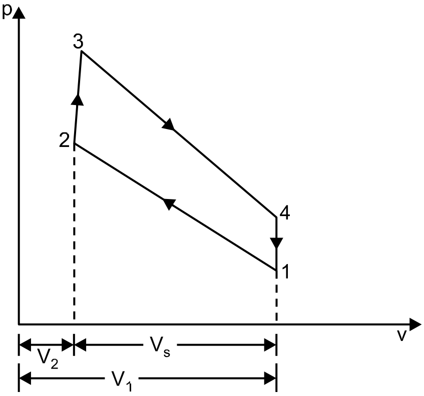
$$p_1 = 0.1\,\text{MPa} = 0.10 \times 10^3\,\text{kPa} = 100\,\text{kPa}$$
$$T_1 = 35\,^\circ\text{C} = (35 + 273)\,\text{K} = 308\,\text{K}$$

Compression ratio:

$$r = 8$$

Heat supplied:

$$\begin{aligned} q_{23} &= 2100\ \text{kJ/kg} \\ c_p &= 1.005\,\text{kJ/kgK} \\ c_v &= 0.718\,\text{kJ/kgK} \\ R &= 0.287\,\text{kJ/kgK} \end{aligned}$$



Now applying the equation of state at state 1, We have

$$p_1 v_1 = RT_1$$
$$100 \times v_1 = 0.287 \times 308$$

or

$$v_1 = 0.8839\,\text{m}^3/\text{kg}$$

We know, compression ratio:

$$r = \frac{v_1}{v_2} = \frac{0.8839}{8}$$

or

$$v_2 = 0.1104\,\text{m}^3/\text{kg}$$

Displacement Volume:

$$\begin{aligned} v_s &= v_1 - v_2 \\ &= 0.8839 - 0.1104 \\ &= 0.7735\,\text{m}^3/\text{kg} \end{aligned}$$

Process 1–2, adiabatic compression:

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$
$$\frac{T_2}{308} = (8)^{1.4-1}$$

or

$$T_2 = 707.59\,\text{K}$$

and

$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{\gamma-1}$$
$$\frac{p_2}{100} = (8)^{1.4}$$

or

$$p_2 = 1837.91\,\text{kPa}$$

Process 2–3, heat addition at $V = C$:

Heat supplied:

$$\begin{aligned} q_{2-3} &= c_v(T_s - T_2) \\ \therefore\ 2100 &= 0.718(T_3 - 707.59) \\ 2924.79 &= T_3 - 707.59 \end{aligned}$$

or

$$T_3 = 3632.38\,\mathrm{K}$$

Maximum temperature of the cycle:

$$\begin{aligned} T_3 &= 3632.38\,\mathrm{K} \\ \frac{p_2 v_2}{T_2} &= \frac{p_3 v_3}{T_3} \\ \frac{p_2}{T_2} &= \frac{p_3}{T_3} \quad \because v_2 = v_3 \\ \frac{1837.91}{707.59} &= \frac{p_3}{3632.63} \end{aligned}$$

or

$$\begin{aligned} p_3 &= 8434.82\,\mathrm{kPa} \\ &= 8.434\,\mathrm{MPa} \end{aligned}$$

Maximum pressure of the cycle:

$$p_3 = 8.434\,\mathrm{MPa}$$

Process 3–4, adiabatic expansion:

$$\begin{aligned} \frac{T_3}{T_4} &= \left(\frac{v_4}{v_5}\right)^{\gamma-1} \\ \frac{T_3}{T_4} &= \left(\frac{v_1}{v_2}\right)^{\gamma-1} \quad v_1 = v_4 \text{ and } v_2 = v_3 \\ \frac{3632.38}{T_4} &= (8)^{1.4-1} \\ \frac{3632.38}{T_4} &= 2.297 \end{aligned}$$

or

$$T_4 = 1581.35\,\mathrm{K}$$

Process 4–1, heat rejection at $V = C$;
Heat rejection:

$$\begin{aligned} q_{4-1} &= c_v(T_4 - T_1) \\ &= 0.718(1581.35 - 308) \\ &= 914.26\,\mathrm{kJ/kg} \end{aligned}$$

Net specific work output:

$$\begin{aligned} w_{\mathrm{net}} &= q_{2-3} - q_{4-1} \\ &= 2100 - 914.26 \\ &= 1185.74\,\mathrm{kJ/kg} \end{aligned}$$

Mean effective pressure:

$$\begin{aligned} \mathrm{mep} &= \frac{\text{Net specific work output}}{\text{Specific displacement volume}} \\ &= \frac{w_{\mathrm{net}}}{v_{\mathrm{s}}} \\ &= \frac{1185.74}{0.7735} = 1532.95\,\mathrm{kPa} = 1.533\,\mathrm{MPa} \end{aligned}$$

Appendix

See Tables A.1, A.2, A.3, A.4, A.5, A.6, A.7, A.8, A.9, A.10, A.11, A.12, A.13, and A.14.

Table A.1 Properties of saturated steam (temperature based)

Sat. temp. °C	Sat. pressure kPa	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
T_{sat}	p_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
0.01	0.6113	0.0010002	206.2	0.00	3237.3	0.01	2501.3	2501.4	0.000	9.156	9.156
1	0.6567	0.0010002	192.6	4.12	2376.7	4.2	2499.0	2503.2	0.015	9.115	9.130
2	0.6556	0.0010001	179.9	8.4	2378.1	8.4	2496.7	2505.0	0.031	9.073	9.104
3	0.7577	0.0010001	168.1	12.6	2379.5	12.6	2494.3	2506.9	0.046	9.032	9.077
4	0.8131	0.0010001	157.2	16.8	2380.9	16.8	2491.9	2508.7	0.061	8.990	9.051
5	0.8721	0.0010001	147.1	21.0	2382.3	21.0	2489.6	2510.6	0.076	8.950	9.026
6	0.9349	0.0010001	137.7	25.2	2383.6	25.2	2487.2	2512.4	0.091	8.909	9.000
7	1.002	0.0010002	129.0	29.4	2385.0	29.4	2484.8	2514.2	0.106	8.869	8.975
8	1.072	0.0010002	120.9	33.6	2386.4	33.6	2482.5	2516.1	0.121	8.829	8.950
9	1.148	0.0010003	113.4	37.8	2387.8	37.8	2480.1	2517.9	0.136	8.789	8.925
10	1.228	0.0010004	106.4	42.0	2389.2	42.0	2477.7	2519.7	0.151	8.750	8.901
11	1.312	0.0010004	99.86	46.2	2390.5	46.2	2475.4	2521.6	0.166	8.711	8.877
12	1.402	0.0010005	93.78	50.4	2391.9	50.4	2473.0	2523.4	0.181	8.672	8.852
13	1.497	0.001007	88.12	54.6	2393.3	54.6	2470.7	2525.3	0.195	8.632	8.828
14	1.598	0.0010008	82.85	58.8	2394.7	58.8	2468.3	2527.1	0.210	8.595	8.805
15	1.705	0.0010009	77.93	63.0	2396.1	63.0	2465.9	2528.9	0.224	8.5,57	8.781
16	1.818	0.001001	73.33	67.2	2397.4	67.2	2463.6	2530.8	0.239	8.519	8.758
17	1.938	0.001001	69.04	71.4	2398.8	71.4	2461.2	2532.6	0.253	8.482	8.735
18	2.064	0.001001	65.04	75.6	2400.2	75.6	2458.8	2534.4	0.268	8.444	8.712
19	2.198	0.001002	61.29	79.8	2401.6	79.8	2456.5	2536.3	0.282	8.407	8.690
20	2.339	0.001002	57.79	84.0	2402.9	84.0	2454.1	2538.1	0.297	8.371	8.667
21	2.487	0.001002	54.51	88.1	2404.3	88.1	2451.8	2539.9	0.311	8.334	8.645
22	2.645	0.001002	61.45	92.3	2405.7	92.3	2449.4	2541.7	0.325	8.298	8.623
23	2.810	0.001002	48.57	96.5	2407.0	96.5	2447.0	2543.5	0.339	8.262	8.601
24	2.085	0.001003	45.88	100.7	2408.4	100.7	2444.7	2545.4	0.353	8.226	8.579
25	3.169	0.001003	43.36	104.9	2409.8	104.9	2442.3	2547.2	0.367	8.191	8.558
26	3.363	0.001003	40.99	109.1	2411.1	109.1	2439.9	2549.0	0.382	8.155	8.537
27	3.567	0.001004	38.77	113.2	2412.5	113.2	2437.6	2550.8	0.396	8.120	8.516
28	3.782	0.001004	36.69	117.4	2413.9	117.4	2435.2	2552.6	0.409	8.086	8.495

(continued)

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Table A.1 (continued)

Sat. temp. °C	Sat. pressure kPa	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
T_{sat}	p_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
29	4.008	0.001004	34.73	121.6	2415.2	121.6	2432.8	2554.5	0.423	8.051	8.474
30	4.246	0.001004	32 89	125.8	2416.6	125.8	2430.5	2556.3	0.437	8.016	8.453
31	4.496	0.001005	31.17	130.0	2418.0	130.0	2428.1	2558.1	0.451	7.982	8.433
32	4.759	0.001005	29.54	134.2	2419.3	134.2	2425.7	2559.9	0.464	7.948	8.413
33	5.034	0.001005	28.01	138.3	2420.7	138.3	2423.4	2561.7	0.478	7.915	8.393
34	5.324	0.001006	26.57	142.5	2422.0	142.5	2421.0	2563.5	0.492	7.881	8.373
35	5.628	0.001006	25.22	146.7	2423.4	146.7	2418.6	2565.3	0.505	7.848	8.353
36	5.947	0.001006	23.94	150.9	2424.7	150.9	2416.2	2567.1	0.519	7.815	8.334
37	6.281	0.001007	22.74	155.0	2426.1	155.0	2413.9	2568.9	0.532	7.782	8.314
38	6.632	0.001007	21.60	159.2	2427.4	159.2	2411.5	2570.7	0.546	7.749	8.295
39	6.999	0.001007	20.53	163.4	2428.8	163.4	2409.1	2572.5	0.559	7.717	8.276
40	7.384	0.001008	19.52	167.6	2430.1	167.6	2406.7	2574.3	0.573	7.685	8.257
41	7.786	0.001008	18.57	171.7	2431.5	171.7	2404.3	2576.0	0.586	7.652	8.238
42	8.208	0.001009	17.67	175.9	2432.8	175.9	2401.9	2577.8	0.599	7.621	8.220
43	8.649	0.001009	16.82	180.1	2434.2	180.1	2399.5	2579.6	0.612	7.589	8.201
44	9.111	0.001010	16.02	184.3	2435.5	184.3	2397.2	2581.5	0.626	7.557	8.183
45	9.593	0.001010	15.26	188.4	2436.8	188.4	2394.8	2583.2	0.639	7.526	8.165
46	10.10	0.001010	14.54	192.6	2438.2	192.6	2392.4	2585.0	6.652	7.495	8.147
47	10.62	0.001011	13.86	196.8	2439.5	196.8	2390.0	2586.8	0.665	7.464	8.129
48	11.48	0.001011	13.22	201.0	2440.8	201.0	2387.6	2588.6	0.678	7.433	8.111
49	11.75	0.001012	12.61	205.1	2442.2	205.1	2385.2	2590.3	0.691	7.403	8.094
50	12.65	0.001012	12.03	209.3	2443.5	209.3	2382.7	2592.1	0.704	7.373	8.076
52	13.63	0.001013	10.97	217.7	2446.1	217.7	2377.9	2595.6	0.730	7.312	8.042
54	15.02	0.001014	10.01	226.0	2448.8	226.0	2373.1	2599.1	0.755	7.253	8.008
55	15.76	0.001015	9.568	230.2	2450.1	230.2	2370.7	2600.9	0.768	7.223	7.991
56	16.53	0.001015	9.149	234.4	2451.4	234.4	2368.2	2602.6	0.781	7.194	7.975
58	18.17	0.001016	8.372	242.8	2454.0	242.8	2363.4	2606.2	0.806	7.136	7.942
60	19.94	0.001017	7.671	. 251.1	2456.6	251.1	2358.5	2609.6	0.831	7.078	7.909
62	21.86	0.001018	7.037	259.5	2459.3	259.5	2353.6	2613.1	0.856	7.022	7.878
64	23.03	0.001019	6.463	267.9	2461.8	267.9	2348.7	2616.5	0.881	6.965	7.846
65	25.03	0.001020	6.197	272.0	2463.1	272.1	2346.2	2618.3	0.894	6.937	7.831
66	26.17	0.001020	5.943	276.2	2464.4	276.2	2343.7	2619.9	0.906	6.910	7.816
68	28.59	0.001022	5.471	284.6	2467.0	284.6	2338.8	2623.4	0.930	6.855	7.785
70	31.49	0.001023	5.042	293.0	2469.6	293.0	2333.8	2626.8	0.955	6.800	7.755
72	34.00	0.001024	4.650	301.4	2472.1	301.4	2329.8	2630.2	0.979	6.746	7.725
75	38.58	0.001026	4.131	313.9	2475.9	313.9	2321.4	2635.3	1.015	J.667	7.682
80	47.39	0.001029	3.407	334.9	2482.2	334.9	2308.8	2643.7	1.075	6.537	7.612
85	57.83	0.001033	2.828	355.9	2488.4	355.9	2296.0	2651.9	1.134	6.410	7.544
90	70.14	0.001036	2.361	376.9	2494.5	376.9	2283.2	2660.1	5.192	6.287	7.479
95	84.65	0.001040	1.982	397.9	2500.6	397.9	2270.2	2668.1	1.250	6.166	7.416
100	101.35	0.001044	1.673	418.9	2506.5	419.0	2257.0	2676.0	1.307	6.048	7.355
105	120.82	0.001048	1.4194	440.02	2512.4	440.15	2243.7	2683.8	1.3630	5.9328	7.2958
110	143.27	0.001052	1.2102	461.14	2518.1	461.30	2230.2	2691.5	1.4185	5.8202	7.2387
115	169.06	0.001056	1.0366	482.30	2523.7	482.48	2216.5	2699.0	1.4734	5.7100	7.1833
120	198.53	0.001060	0.8919	503.50	2529.3	503.71	2202.6	2706.3	1.5276	5.6020	7.1296
125	232.1	0.001065	0.7706	524.74	2534.6	524.99	2188.5	2713.5	1.5813	5.4962	7.0775
130	270.1	0.001070	0.6685	546.02	2539.9	546.31	2174.2	2720.5	1.6344	5.3925	7.0269

(continued)

Table A.1 (continued)

Sat. temp. °C	Sat. pressure kPa	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
T_{sat}	p_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
135	313.0	0.001075	0.5822	567.35	2545.0	567.69	2159.6	2727.3	1.6870	5.2907	6.9777
140	361.3	0.001080	0.5089	588.74	2550.0	589.13	2144.7	2733.9	1.7391	5.1908	6.9299
145	415.4	0.001085	0.4463	610.18	2554.9	610.63	2129.6	2740.3	1.7907	5.0926	6.8833
150	475.8	0.001091	0.3928	631.68	2559.5	632.20	2114.3	2746.5	1.8418	4.9960	6.8379
155	543.1	0.001096	0.3468	653.24	2564.1	653.84	2098.6	2752.4	1.8925	4.9010	6.7935
160	617.8	0.001102	0.3071	674.87	2568.4	675.55	2082.6	2758.1	1.9427	4.8075	6.7502
165	700.5	0.001108	0.2727	696.56	2572.5	697.34	2066.2	2763.5	1.9925	4.7153	6.7078
170	791.7	0.001114	0.2428	718.33	2576.5	719.21	2049.5	2768.7	2.0419	4.6244	6.6663
175	892.0	0.001121	0.2168	740.17	2580.2	741.17	2032.4	2773.6	2.0909	4.5347	6.6256
180	1002.1	0.001127	0.19405	762.09	2583.7	763.22	2015.0	2778.2	2.1396	4.4461	6.5857
185	1122.7	0.001134	0.17409	784.10	2587.0	785.37	1997.1	2782.4	2.1879	4.3586	6.5465
190	1254.4	0.001141	0.15654	806.19	2590.0	807.62	1978.8	2786.4	2.2359	4.2720	6.5079
195	1307.8	0.001149	0.14105	828.37	2592.8	829.98	1960.0	2790.0	2.2835	4.1863	6.4698
200	1553.8	0.001157	0.12736	850.65	2595.3	852.45	1940.7	2793.2	2.3309	4.1014	6.4323
205	1723.0	0.001164	0.11521	873.04	2597.5	875.04	1921.0	2796.0	2.3780	4.0172	6.3952
210	1906.2	0.001173	0.10441	895.53	2599.5	897.76	1900.7	2798.5	2.4248	3.9337	6.3585
215	2104	0.001181	0.09479	918.14	2601.1	920.62	1879.9	2800.5	2.4714	3.8507	6.3221
220	2318	0.001190	0.08619	940.87	2602.4	943.62	1858.5	2802.1	2.5178	3.7683	6.2861
225	2548	0.001199	0.07849	963.73	2603.3	966.78	1836.5	2803.3	2.5639	3.6863	6.2503
230	2995	0.001209	0.07158	986.74	2603.9	990.12	1813.8	2804.0	2.6099	3.6047	6.2146
235	3060	0.001219	0.06537	1009.89	2604.1	1013.62	1790.5	2804.2	2.6558	3.5233	6.1791
240	3344	0.001229	0.05976	1033.21	2604.0	1037.32	1766.5	2803.8	2.7015	3.4422	6.1437
245	3648	0.001240	0.05471	1056.71	2603.4	1061.23	1741.7	2803.0	2.7472	3.3612	6.1083
250	3973	0.001251	0.05013	1080.39	2602.4	1085.36	1716.2	2801.5	2.7927	3.2802	6.0730
255	4319	0.001263	0.04598	1104.28	2600.9	1109.73	1689.8	2799.5	2.8383	3.1992	6.0375
260	4688	0.001276	0.04221	1128.39	2599.0	1134.37	1662.5	2796.9	2.8838	3.1181	6.0019
265	5081	0.001289	0.03877	1152.74	2596.6	1159.28	1634.4	2793.6	2.9294	3.0368	5.9662
270	5499	0.001302	0.03564	1177.36	2593.7	1184.51	1605.2	2789.7	2.9751	2.9551	5.9301
275	5942	0.001317	0.03279	1202.25	2590.2	1210.07	1574.9	2785.0	3.0208	2.8730	5.8938
280	6412	0.001332	0.03017	1227.46	2586.1	1235.99	1543.6	2779.6	3.0668	2.7903	5.8571
285	6909	0.001348	0.02777	1253.00	2581.4	1262.31	1511.0	2773.3	3.1130	2.7070	5.8199
290	7436	0.001366	0.02557	1278.92	2576.0	1289.07	1477.1	2766.2	3.1594	2.6227	5.7821
295	7993	0.001384	0.02354	1305.2	2569.9	1316.3	1441.8	2758.1	3.2062	2.5375	5.7437
300	8581	0.001404	0.02167	1332.0	2563.0	1344.0	1404.9	2749.0	3.2534	2.4511	5.7045
305	9202	0.001425	0.019948	1359.3	2555.2	1372.4	1366.4	2738.7	3.3010	2.3633	5.6643
310	9856	0.001447	0.018350	1387.1	2546.4	1401.3	1326.0	2727.3	3.3493	2.2737	5.6230

Table A.2 Properties of saturated steam (pressure based)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
0.006113	0.01	0.0010002	206.140	0.00	2375.3	0.01	2501.3	2501.4	0.000	9.156	9.156
0.010	7.0	0.0010000	129.21	29.3	2385.0	29.3	2484.9	2514.2	0.106	8.870	8.976
0.105	13.0	0.0010007	87.98	54.7	2393.3	54.7	2470.6	2525.3	0.196	8.632	8.828
0.020	17.0	0.001001	67.00	73.5	2399.5	73.5	2460.0	2533.5-	0.261	8.463	8.724
0.025	21.1	0.001002	54.25	88.5	2404.4	88.5	2451.6	2540.1	0.312	8.331	8.643
0.030	24.1	0.001003	45.67	101.0	2408.5	101.0	2444.5	2545.5	0.355	8.223	8.578
0.035	26.7	0.001003	39.50	111.9	2412.1	111.9	2438.4	2550.3	0.391	8.132	8.523
0.040	29.0	0.001004	34.80	121.5	2415.2	121.5	2432.9	2554.4	0.423	8.052	8.475
0.045	31.0	0.001005	31.13	130.0	2417.9	130.0	2428.2	2558.2	0.451	7.982	8.433
0.050	32.9	0.001005	28.19	137.8	2420.5	137.8	2423.7	2561.5	0.476	7.919	8.395
0.055	34.6	0.001006	25.77	144.9	2422.8	144.9	2419.6	2565.5	0.500	7.861	8.361
0.060	36.2	0.001006	23.74	151.5	2425.0	151.5	2415.9	2567.4	0.521	7.809	8.330
0.065	37.6	0.001007	22.01	157.7	2426.9	157.7	2412.4	2570.1	0.541	7.761	8.302
0.070	39.0	0.001007	20.53	163.4	2428.8	163.4	2409.1	2572.5	0.559	7.717	8.276
0.075	40.3	0.001008	19.24	168.8	2430.5	168.8	2406.0	2574.8	0.576	7.675	8.251
0.080	41.5	0.001008	18.10	173.9	2432.2	173.9	2403.1	2577.0	0.593	7.636	8.229
0.085	42.7	0.001009	17.10	178.7	2433.7	178.7	2400.3	2579.0	0.608	7.599	8.207
0.090	43.8	0.001009	16.20	183.3	2435.2	183.3	2397.7	2581.0	0.622	7.565	8.187
0.095	44.8	0.001010	15.40	187.7	2436.6	187.7	2395.2	2582.9	0.636	7.532	8.168
0.10	45.8	0.001010	14.67	191.8	2437.9	191.8	2392.8	2584.7	0.649	7.501	8.150
0.11	47.7	0.001011	13.42	199.7	2440.4	199.7	2388.3	2588.0	0.674	7.453	8.117
0.12	49.4	0.001012	12.36	206.9	2442.7	206.9	2384.2	2591.1	0.696	7.390	8.086
0.13	51.0	0.001013	11.47	213.7	2.444.9	213.7	2380.2	2593.9	0.717	7.341	8.058
0.14	52.6	0.001013	10.69	220.0	2446.9	220.0	2376.6	2596.6	0.737	7.296	8.033
0.15	54.0	0.001014	10.02	225.9	2448.7	225.9	2373.2	2559.1	0.755	7.254	8.009
0.16	55.3	0.001015	9.43	231.6	2450.5	231.6	2369.9	2601.5	0.772	7.214	7.986
0.17	56.6	0.001015	8.91	236.9	2452.2	236.9	2366.8	2603.7	0.788	7.177	7.%5

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
0.18	57.8	0.001016	8.45	242.0	2453.8	242.0	2363.8	2505.8	0.804	7.14.1	7.945
0.19	59.0	0.001017	8.03	246.8	2455.3	246.8	2361.0	2607.8	0.818	7.108	7.926
0.20	60.1	0.001017	7.65	251.4	2456.7	251.4	2358.3	2609.7	0.832	7.077	7.909
0.22	62.1	0.001018	7.45	260.1	2459.4	260.1	2353.2	2613.3	0.858	7.018	7.876
0.24	64.1	0.001019	6.45	268.1	2461.9	268.1	2348.5	2616.6	0.882	6.964	7.846
0.25	65.0	0.001020	6.20	271.9	2463.1	271.9	2346.3	2618.2	0.893	6.938	7.831
0.26	65.9	0.001020	5.98	275.6	2464.2	275.6	2344.1	2619.7	0.904	6.914	7.818
0.28	67.5	0.001021	5.58	282.6	2466.4	282.6	2340.0	2622.6	0.925	6.868	7.793
0.30	69.1	0.001022	5.23	289.2	2468.4	289.2	2336.1	2625.3	0.944	6.825	7.768
0.32	70.6	0.001023	4.92	295.5	2470.3	295.5	2332.4	2627.9	0.962	6.784	7.746
0.34	72.0	0.001024	4.65	301.4	2472.1	301.4	2328.8	2630.2	0.979	6.746	7.725
0.35	72.7	0.001024	4.53	304.2	2473.0	304.2	2327.2	2631.4	0.987	6.728	7.715
0.36	73.4	0.001025	4.41	307.0	2473.8	307.0	2325.5	2632.5	0.996	6.710	7.706
0.38	74.6	0.001026	4.19	312.4	2475.5	312.4	2322.3	2634.7	1.011	6.676	7.687
0.40	75.9	0.001027	3.99	317.6	2477.0	317.6	2319.2	2636.8	1.026	6.644	7.670
0.45	78.7	0.001028	3.58	329.6	2480.7	329.6	2312.0	2641.6	1.060	6.571	7.631
0.50	81.3	0.001030	3.24	340.5	2483.9	340.6	2305.4	2646.0	1.091	6.503	7.594
0.55	83.7	0.001032	2.96	350.5	2486.8	350.5	2299.3	2649.8	1.119	6.442	7.561
0.60	85.9	0.001033	2.73	359.8	2489.6	359.9	2293.6	2653.5	1.145	6.387	7.532
0.65	88.0	0.001035	2.53	368.5	2492.1	368.5	2288.3	2656.8	1.169	6.335	7.504
0.70	90.0	0.001036	2.37	376.7	2494.5	376.7	2283.3	2660.0	1.192	6.288	7.480
0.75	91.8	0.001037	2.22	384.3	2496.7	384.4	2278.6	2663.0	1.213	6.243	7.456
0.80	93.5	0.001039	2.087	391.6	2498.8	391.7	2274.1	2665.8	1.233	6.202	7.435
0.85	95.1	0.001040	1.972	398.5	2500.7	398.6	2269.8	2668.4	1.252	6.163	7.415
0.90	96.7	0.001041	1.869	405.1	2502.6	405.2	2265.7	2670.9	1.270	6.125	7.395
0.95	98.2	0.001042	1.777	411.3	25,044	411.4	2261.8	2673.2	1.287	6.090	7.377
1.00	99.6	0.001043	1.694	417.4	2506.1	417.5	2258.0	2675.5	1.303	6.057	7.360

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
1.0135	**100**	**0.001044**	**1.673**	**418.9**	**2506.5**	**419.0**	**2256.9**	**2676.0**	**1.307**	**6.048**	**7.355**
1.1	102.3	0.001045	1.549	428.7	2509.2	428.8	2250.9	2679.7	1.333	5.994	7.327
1.2	104.8	0.001047	1.428	439.2	2512.1	439.3	2244.2	2683.5	1.361	5.937	7.298
1.3	107.1	0.001049	1.325	449.0	2514.8	449.1	2238.0	2687.1	1.387	5.884	7.271
1.4	109.3	0.001051	1.237	458.2	2517.3	458.4	2232.0	2690.4	1.411	5.835	7.246
1.5	111.4	0.001053	1.159	466.9	2519.7	467.1	2226.5	2693.6	1.434	5.789	7.223
1.6	113.3	0.001054	1.091	475.2	2521.9	475.4	2221.1	2696.5	1.455	5.747	7.202
1.7	115.2	0.001056	1.031	483.0	2523.9	483.2	2211.2	2699.2	1.475	5.706	7.181
1.8	116.9	0.001058	0.977	490.5	2525.9	490.7	2211.2	2701.8	1.494	5.668	7.162
1.9	118.6	0.001059	0.929	497.6	2527.7	497.8	2206.5	2704.3	1.513	5.631	7.144
2.0	120.2	0.001061	0.886	504.5	2529.5	504.7	2201.9	2706.7	1.530	5.597	7.127
2.1	121.8	0.001062	0.846	511.1	2531.2	511.3	2197.6	2708.9	1347	5.564	7.111
2.2	123.3	0.001063	0.810	517.4	2532.8	517.6	2193.4	2711.0	1.563	5.532	7.095
2.3	124.7	0.001065	0.777	523.5	2534.3	523.7	2189.3	2713.1	1378	5302	7.080
2.4	126.1	0.001066	0.747	529.4	2535.8	529.6	2185.4	2715.0	1.593	5.473	7.066
2.5	127.4	0.001067	0.719	535.1	2537.2	535.4	2118.5	2716.9	1.607	5.446	7.053
2.6	128.7	0.001069	0.693	540.6	2538.6	540.9	2177.8	2718.7	1.621	5.419	7.040
2.7	130.0	0.001070	0.669	546.0	2539.9	546.3	2174.2	2720.5	1.634	5.393	7.027
2.8	131.2	0.001071	0.646	551.2	2541.2	551.4	2170.7	2722.1	1.647	5.368	7.015
2.9	132.4	0.001072	0.625	556.2	2542.4	556.5	2167.3	2723.8	1.660	5.343	7.003
3.0	133.5	0.001073	0.606	561.1	2543.6	561.6	2163.8	2725.3	1.672	5.320	6.992
3.1	134.7	0.001074	0.588	565.9	2544.7	566.3	2160.6	2726.8	1.684	5.297	6.981
3.2	135.8	0.001075	0.570	570.6	2545.8	571.0	2157.3	2728.3	1.695	5.275	6.970
3.3	136.8	0.001076	0.554	575.2	2546.9	575.5	2154.2	2729.7	1.706	5.254	6.960
3.4	137.9	0.001078	0.539	579.6	2547.9	580.0	2151.1	2731.1	1.717	5.233	6.950
3.5	138.9	0.001079	0.524	583.9	2548.9	584.3	2148.1	2732.4	1.728	5.213	6.941
3.6	140.0	0.001080	0.511	588.2	2549.9	588.6	2145.1	2733.7	1.738	5.193	6.931

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
3.7	140.8	0.001081	0.498	592.4	2550.9	592.8	2142.2	2735.0	1.748	5.174	6.922
3.8	141.8	0.001082	0.485	596.4	2551.8	596.8	2139.4	2736.2	1.758	5.155	6.913
3.9	142.7	0.001083	0.474	600.4	2552.7	600.8	2136.6	2737.4	1.767	5.137	6.904
4.0	143.6	0.001084	0.463	604.3	2553.6	604.7	2133.8	2738.5	1.777	5.119	6.896
4.1	144.5	0.001085	0.452	608.1	2554.4	608.6	2131.1	2739.7	1.786	5.102	6.888
4.2	145.4	0.001286	0.442	611.9	2555.3	612.4	2128.4	2740.8	1.795	5.085	6.880
4.3	146.3	0.001086	0.432	615.6	2556.1	616.1	2125.8	2741.9	1.804	5.068	6.872
4.4	147.1	0.001087	0.423	619.2	2456.9	619.7	2123.2	2742.9	1.812	5.052	6.684
4.5	147.9	0.001088	0.414	622.8	2557.6	623.3	2120.6	2743.9	1.821	5.036	6.857
4.6	148.7	0.001089	0.406	626.3	2558.4	626.8	2118.2	2744.9	1.829	5.020	6.849
4.7	149.5	0.001090	0.397	629.7	2559.1	63,012	2115.7	2745.9	1.837	5.005	6.842
4.8	150.3	0.001091	0.390	633.1	2559.8	633.6	2113.2	2746.8	1.845	4.990	6.835
4.9	151.1	0.001092	0.382	636.4	2560.6	636.9	2110.8	2747.8	1.853	4.975	6.828
5.0	151.9	0.001093	0.375	639.7	2561.2	640.2	2108.5	2748.7	1.861	4.961	6.821
5.2	153.3	0.001094	0.361	646.1	2562.6	646.7	2103.8	2750.5	1.876	4.932	6.808
5.4	154.8	0.001096	0.349	652.3	2563.9	652.9	2099.3	2752.1	1.890	4.905	6.795
5.5	155.5	0.001096	0.343	655.3	2564.5	655.9	2097.0	2752.9	1.897	4.892	6.789
5.6	156.2	0.001097	0.337	658.3	2565.1	658.9	2094.8	2753.8	1.904	4.879	6.783
5.8	157.5	0.001099	0.326	664.2	2566.3	664.8	2090.5	2755.3	1.918	4.853	6.771
6.0	158.9	0.001101	0.316	669.9	2567.4	670.6	2086.3	2756.8	1.931	4.829	6.760
6.2	160.1	0.001002	0.306	675.5	2568.5	676.2	2082.1	2758.3	1.944	4.805	6.749
6.4	161.4	0.001104	0.297	680.9	2569.6	681.6	2078.0	2759.6	1.956	4.782	6.738
6.5	162.0	0.001104	0.293	683.6	2570.1	684.3	2076.0	2760.3	1.963	4.770	6.733
6.6	162.6	0.001105	0.288	686.2	2570.6	686.9	2074.0	2761.0	1.969	4.759	6.728
6.8	163.8	0.001107	0.281	691.4	2571.6	692.1	2070.1	2762.2	1.981	4.737	6.718
7.0	165.0	0.001108	0.273	696.4	2572.5	697.2	2066.3	2763.5	1.992	4.716	6.708
7.2	166.1	0.001109	0.266	701.4	2573.4	702.2	2062.5	2764.7	2.004	4.695	6.699

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
7.4	167.2	0.001111	0.259	706.2	2574.3	707.1	2058.8	2765.9	2.014	4.675	6.689
7.5	167.8	0.001112	0.256	708.6	2574.7	709.5	2057.0	2766.5	2.020	4.665	6.685
7.6	168.3	0.001112	0.252	711.0	2575.2	711.8	2055.2	2767.0	2.025	4.655	6-.680
7.8	169.4	0.001113	0.246	715.7	2576.0	716.5	2051.6	2768.1	2.036	4.635	6.671
8.0	170.4	0.001115	0.240	720.2	2576.8	721.1	2048.0	2769.1	2.046	4.617	6.663
8.2	171.5	0.001116	0.235	724.7	2577.6	725.6	2044.6	2770.2	2.056	4.598	6.654
8.4	172.5	0.001117	0.230	729.1	2578.3	730.1	2041.1	2771.2	2.066	4.580	6.646
8.5	173.0	0.001118	0.227	731.3	2578.7	732.2	2039.4	2771.6	2.071	4.571	6.642
8.6	173.5	0.001119	0.225	733.4	2579.1	734.4	2037.7	2772.1	2.076	4.562	6.638
8.8	174.4	0.001120	0.220	737.7	2579.8	738.6	2034.4	2773.0	2.085	4.545	6.630
9.0	175.4	0.001121	0.215	741.8	2580.5	742.8	2031.1	2773.9	2.095	4.528	6.623
9.2	176.3	0.001122	0.211	745.9	2581.1	747.0	2027.8	2774.8	2.104	4.511	6.615
9.4	177.2	0.001124	0.206	750.0	2581.8	751.0	2024.7	2775.7	2.113	4.495	6.608
9.5	177.7	0.001124	0.204	751.9	2582.1	753.0	2023.1	2776.1	2.117	4.487	6.604
9.6	178.1	0.001125	0.202	753.9	2582.4	755.0	2021.5	2776.5	2.122	4.479	6.601
9.8	179.0	0.001126	0.198	757.8	2583.1	758.9	2018.4	2777.3	2.130	4.463	6.593
10.0	179.9	0.001127	0.194	761.7	2583.6	762.8	2015.3	2778.1	2.139	4.448	6.587
10.5	182.0	0.001130	0.186	771.1	2585.1	772.2	2007.7	2779.9	2.159	4.411	6.570
11.0	184.1	0.001133	0.178	780.1	2586.4	781.3	2000.4	2781.7	2.179	4.374	6.553
11.5	186.1	0.001136	0.170	778.8	2587.6	790.1	1993.2	2783.3	2.198	4.340	6.538
12.0	188.0	0.001139	0163	797.3	2588.8	798.6	1986.2	2784.8	2.217	4.306	6.533
12.5	189.8	0.001141	0.157	805.4	2589.9	806.8	1979.4	2786.2	2.243	4.275	6.509
13.0	191.6	0.001144	0.151	813.4	2591.0	314.9	1972.7	2787.6	2.251	4.244	6.495
13.5	193.4	0.001146	0.146	821.2	2591.9	822.6	1966.2	2788.8	2.268	4.214	6.482
14.0	195.0	0.001149	0.141	828.7	2592.8	830.3	1959.7	2790.0	2.284	4,185	6.469
14.5	196.7	0.001151	0.136	836.0	2593.7	837.6	1953.5	27,911	2.300	4.157	6.457
15.0	198.3	0.001154	0.132	843.2	2594.5	844.9	1947.3	2792.2	2.315	4.130	6.445

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
15.5	199.9	0.001156	0.128	850.1	2595.3	851.9	1941.2	2793.1	2.330	4.103	6.433
16.0	201.4	0.001169	0.124	856.9	2596.0	858.8	1935.2	2794.0	2.344	4.078	6.422
16.5	202.9	0.001161	0.120	863.6	2596.6	865.5	1929.4	2794.9	2.358	4.053	6.411
17.0	204.3	0.001163	0.117	870.1	2597.3	872.1	1923.6	2795.7	2.372	4.028	6.400
17.5	205.8	0.001166	0.133	876.5	2597.8	878.5	1917.9	2796.4	2.385	4.005	6.390
18.0	207.2	0.001168	0.110	882.7	2598.4	884.8	1912.4	2797.2	2.398	3.981	6.379
18.5	208.5	0.001170	0.108	888.8	2598.9	891.0	1906.8	2797.8	2.411	3.958	6.369
19.0	209.8	0.001172	0.105	894.8	2599.4	897.0	1901.4	2798.4	2.423	3.936	3.359
19.5	211.1	0.001175	0.102	900.7	2599.9	903.0	1896.0	2799.0	2.435	3.915	6.350
20.0	212.4	0.001177	0.0996	906.4	2600.3	908.8	1890.7	2799.5	2.447	3.894	6.341
21.0	214.9	0.001181	0.0950	917.7	2601.0	920.2	1880.3	2800.5	2.470	3.852	6.323
22.0	217.3	0.001185	0.0907	928.5	2601.7	931.1	1870.2	2801.3	2.493	3.813	6.306
23.6	219.6	0.001189	0.0869	939.0	2602.3	941.8	1860.2	2802.0	2.514	3.775	6.289
24.0	221.8	0.001193	0.0833	949.2	2602.8	952.1	1850.5	2802.6	2.535	3.738	6.273
25.0	224.0	0.001197	0.0801	959.1	2603.1	962.1	1841.0	2803.1	2.555	3.703	6.258
26.0	226.1	0.001201	0.0769	968.7	2603.5	971.9	1831.6	2803.5	2.574	3.669	6.243
27.0	228.1	0.001205	0.0741	978.1	2603.7	981.3	1822.4	2803.8	2.593	3.635	6.228
28.0	230.1	0.001209	0.0715	987.2	2603.9	990.6	1813.4	2804.0	2.611	3.603	6.214
29.0	232.0	0.001213	0.0690	996.1	2604.0	999.6	1804.5	2804.1	2.628	3.572	6.200
30.0	233.9	0.001217	0.0667	1004.8	2604.1	1008.4	1795.7	2804.2	2.646	3.541	6.187
31.0	235.7	0.001220	00,645	1013.3	2604.1	1017.0	1787.1	2804.1	2.662	3.512	6.174
32.0	237.5	0.001224	0.0625	1021.6	2604.1	1025.5	1778.6	2804.1	2.679	3.483	6161
33.0	239.2	0.001227	0.0606	1029.7	2604.0	1033.7	1770.2	2803.9	2.695	3.454	6.149
34.0	240.9	0.001231	0.0588	1037.6	2603.9	1041.8	1761.9	2803.7	2.710	3.427	6.137
35.0	242.6	0.001235	0.0571	1045.4	2603.7	1049.7	1753.7	2803.4	2.725	3.400	6.125
36.0	244.2	0.001238	0.0555	1053.1	2603.5	1057.5	1745.6	2803.1	2.740	3.374	6.114
37.0	245.8	0.001242	0.0539	1060.6	2603.3	1065.2	1737.6	2802.8	2.755	3.348	6.103

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
38.0	247.4	0.001245	0.0523	1068.0	2603.0	1072.7	1729.7	2802.4	2.769	3.323	6.092
39.0	248.9	0.001249	0.0511	1075.2	2602.6	1080.1	1721.8	2801.9	2.783	3.298	6.081
40.0	250.4	0.001252	0.0498	1082.3	2602.3	1087.3	1714.1	2801.4	2.796	3.274	6.070
42.0	253.3	0.001259	0.0473	1096.2	2601.5	1101.5	1698.8	2800.3	2.823	3.227	6.050
44.0	256.1	0.001266	0.0451	1109.7	2600.6	1115.2	1638.8	2799.0	2.849	3.181	6.030
45.0	257.5	0.001269	0.0441	1116.2	2600.1	1121.9	1676.4	2798.3	2.861	3.159	6.020
46.0	258.8	0.001273	0.0431	1122.7	2599.5	1128.6	1669.0	2797.6	2.873	3.137	6.010
48.0	261.4	0.001279	0.0412	1135.4	2598.4	1141.5	1654.5	2796.0	2.897	3.095	5.992
50.0	264.0	0.001286	0.0394	1147.8	2597.1	1154.2	1640.1	2794.3	2.920	3.053	5.973
52.0	266.5	0.001293	0.0378	1159.9	2595.8	1166.6	1626.0	2792.6	2.943	3.013	5.956
54.0	268.8	0.001299	0.0363	1171.7	2594.4	1178.7	1612.0	2790.7	2.965	2.974	5.939
55.0	270.0	0.001302	0.0357	1177.4	2593.7	1184.5	1605.1	2789.6	2.975	2.955	5.930
56.0	271.2	0.001306	0.0350	1183.2	2592.9	1190.5	1598.2	2788.6	2.986	2.936	5.922
58.0	273.4	0.001312	0.0377	1194.4	2591.3	1202.0	1584.5	2786.5	3.006	2.899	5.905
60.0	275.6	0.001319	0.0324	1205.4	2589.7	1213.3	1571.0	2784.3	3.027	2.862	5.889
62.0	277.8	0.001325	0.0313	1216.3	2588.0	1224.5	1557.6	2782.1	3.046	2.827	5.873
64.0	279.9	0.001332	0.0302	1226.9	2586.2	1235.4	1544.3	2779.7	3.066	2.792	5.858
65.0	280.8	0.001335	0.0297	1232.0	2585.3	1240.7	1537.9	2778.6	3.075	2.775	5.850
66.0	281.9	0.001338	0.0292	1237.3	2584.4	1246.1	1531.1	2777.2	3.085	2.758	5.843
68.0	283.9	0.001345	0.0283	1247.5	2582.5	1256.6	1518.1	2774.7	3.103	2.725	5.828
70.0	285.9	0.001351	0.0274	1257.5	2580.5	1267.0	1505.1	2772.1	3.121	2.692	5.813
72.0	287.8	0.001358	0.0265	1267.4	2578.5	1277.2	1492.2	2769.4	3.139	2.660	5.799
74.0	289.7	0.001364	0.0257	1277.2	2576.4	1287.3	1479.4	2766.7	3.156	2.629	5.785
75.0	290.6	0.001368	0.0253	1282.0	2575.3	1292.2	1473.1	2765.3	3.165	2.613	5.778
76.0	291.5	0.001371	0.0249	1286.8	2574.3	1297.2	1466.6	2763.8	3.174	2.597	5.771
78.0	293.3	0.001378	0.0242	1296.2	2572.1	1307.0	1453.9	2760.9	3.190	2.567	5.757
80.0	295.1	0.001384	0.0235	1305.6	2569.8	1316.6	1441.3	2758.0	3.207	2.536	5.743

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
85.0	299.3	0.001401	0.0219	1328.4	2564.0	1340.3	1410.0	2750.3	3.247	2.463	5.710
90.0	303.4	0.001418	0.0205	1350.5	2557.8	1363.2	1378.9	2742.1	3.286	2.391	5.677
95.0	307.3	0.001435	0.0192	1372.0	2551.2	1385.6	1348.0	2733.6	3.323	2.322	5.645
100.00	311.1	0.001452	0.0180	1393.0	2544.4	1407.6	1317.1	2724.7	3.360	2.254	5.614
105.0	314.7	0.001470	0.0170	1413.6	2537.7	1429.0	1286.4	2715.4	3.395	2.188	5.583
110.0	318.2	0.001489	0.0160	1433.7	2529.8	1450.1	1255.5	2705.6	3.430	2.123	5.553
115.0	321.5	0.001507	0.0151	1453.5	2522.2	1470.8	1224.6	2695.4	3.463	2.059	5.522
120.0	324.8	0.001527	0.0143	1473.0	2513.7	1491.3	1193.6	2684.9	3.496	1.996	5.492
125.0	327.9	0.001547	0.0135	1492.1	2505.1	1511.5	1162.2	2673.7	3.528	1.934	5.462
130.0	330.9	0.001567	0.0128	1511.1	2496.1	1531.5	1130.7	2662.2	3.561	1.871	5.432
135.0	333.9	0.001588	0.0121	1529.9	2486.6	1551.4	1098.8	2650.2	3.592	1.810	5.402
140.0	336.8	0.001611	0.0115	1548.6	2476.8	1571.1	1066.5	2637.6	3.623	1.749	5.372
145.0	339.5	0.001634	0.0109	1567.1	2466.4	1590.9	1033.5	2624.4	3.654	1.687	5.341
150.0	342.2	0.001658	0.0103	1585.6	2455.5	1610.5	1000.0	2610.5	3.685	1.625	5.310
155.0	344.9	0.001684	0.00981	1604.1	2443.9	1630.3	965.7	2596.0	3.715	1.563	5.278
160.0	347.4	0.001711	0.00931	1622.7	2431.7	1650.1	930.6	7.580.6	3.746	1.499	5.245
165.0	349.9	0.001740	0.00883	1641.4	2418.8	1670.1	894.3	2564.4	3.777	1.435	5.212
170.0	352.4	0.001770	0.00836	1660.2	2405.0	1690.3	856.9	2547.2	3.808	1.370	5.178
175.0	354.7	0.001804	0.00793	1679.4	2390.2	1711.0	817.8	2528.8	3.839	1.302	5.141
180.0	357.1	0.001840	0.00749	1698.9	2374.3	1732.0	777.1	2509.1	3.871	1.233	5.104
185.0	359.3	0.001880	0.00708	1719.1	2357.0	1753.9	733.9	2487.8	3.905	1.160	5.065
190.0	361.5	0.001924	0.00666	1739.9	2338.1	1776.5	688.0	2464.5	3.939	1.084	5.023
195.0	363.7	0.001976	0.00625	1762.0	2316.9	1800.6	638.2	2438.8	3.975	1.002	4.977
200.0	365.8	0.002036	0.00583	1785.6	2293.0	1826.3	583.4	2409.7	4.014	0.913	4.927
205.0	367.9	0.002110	0.00541	1811.8	2265.0	1855.0	520.8	2375.8	4.057	0.812	4.869
210.0	369.9	0.002207	0.00495	1842.1	2230.6	1888.4	446.2	2334.6	4.107	0.694	4.801
215.0	371.9	0.002358	0.00442	1882.3	2183.0	1933.0	344.9	2277.9	4.175	0.535	4,710

(continued)

Table A.2 (continued)

Sat. pressure bar	Sat. temp. °C	Specific volume m^3/kg		Specific internal energy kJ/kg		Specific enthalpy kJ/kg			Specific entropy kJ/kgK		
p_{sat}	T_{sat}	Sat. liquid v_f	Sat. vapor v_g	Sat. liquid u_f	Sat. vapor u_g	Sat. liquid h_f	Evap. h_{fg}	Sat. vapor h_g	Sat. liquid s_f	Evap. s_{fg}	Sat. vapor s_g
220.0	373.8	0.002742	0.00357	1961.9	2087.1	2022.2	143.4	2165.6	4.311	0.222	4.533
221.2	**374.15**	**0.003155**	**0.003155**	**2029.6**	**2029.6**	**2099.3**	**0**	**2099.3**	**4.4298**	**0**	**4.4298**

Table A.3 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 0.1 bar (45.8 °C)				p = 0.5 bar (81.3 °C)				p = 1 bar (99.6 °C)			
Sat	14.674	2437.9	2584.7	8.1502	3.240	2483.9	2645.9	7.5939	1.6940	2506.1	2675.5	7.3594
50	14.869	2483.9	2592.6	8.1749								
100	17.196	2515.5	2687.5	8.4479	3.418	2511.6	2682.5	7.6947	1.6958	2506.7	2676.2	7.3614
150	19.512	2587.9	2783.0	8.6882	3.889	2585.6	2780.1	7.9401	1.9364	2582.8	2776.4	7.6134
200	21.825	2661.3	2879.5	8.9038	4.356	2659.9	2877.7	8.1580	2.172	2658.1	2875.3	7.8343
250	24.136	2736.0	2977.3	9.1002	4.820	2735.0	2976.0	8.3556	2.406	2733.7	2974.3	8.0333
300	26.445	2812.1	3075.5	9.2813	5.284	2811.3	3075.5	8.5373	2.639	2810.4	3074.3	8.2158
400	31.063	2968.9	3279.6	9.6077	6.209	2968.5	3278.9	8.8642	3.103	2967.9	3278.2	8.5435
500	35.679	3132.3	3489.1	9.8978	7.134	3132.0	3488.7	9.1546	3.565	3131.6	3488.1	8.8342
600	40.295	3302.5	3705.4	10.1608	8.057	3302.2	3705.1	9.4178	4.028	3301.9	3704.4	9.0976
700	44.911	3479.6	3928.7	10.4028	8.981	3479.4	3928.5	9.6599	4.490	3479.2	3928.2	9.3398
800	49.526	3663.8	4159.0	10.6281	9.904	3663.6	4158.0	9.8852	4.952	3663.5	4158.6	9.5652
900	54.141	3855.0	4396.4	10.8396	10.828	3854.9	4396.3	10.0967	5.414	3854.8	4396.1	9.7767
1000	58.757	4053.0	4640.6	11.0393	11.751	4052.9	4640.5	10.2964	5.875	4052.8	4640.3	9.9764
1100	63.372	4257.5	4891.2	11.2287	12.674	4257.4	4891.1	10.4859	6.337	4257.3	4891.0	10.1659
1200	67.987	4467.9	5147.8	11.4091	13.597	4467.8	5147.7	10.6662	6.799	4467.7	5147.6	10.3463
1300	72.602	4683.7	5409.7	11.5811	14.521	4683.6	5409.6	10.8382	7.260	4683.5	5409.5	10.5183

Table A.4 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	S kJ/kgK
	p = 2 bar (120.2 °C)				p = 3 bar (133.5 °C)				p = 4 bar (143.6 °C)			
Sat	0.8857	2529.5	2706.7	7.1272	0.6058	2543.6	2725.3	6.9919	0.4625	2553.6	2738.6	6.8959
150	0.9596	2576.9	2768.8	7.2795	0.6339	2570.8	2761.0	7.0778	0.4708	2564.5	2752.8	6.9299
200	1.0803	2654.4	2870.5	7.5066	0.7163	2650.7	2865.6	7.3115	0.5342	2646.8	2860.5	7.1706
250	1.1988	2731.2	2971.0	7.7086	0.7964	2728.7	2967.6	7.5166	0.5951	2726.1	2964.2	7.3789
300	1.3162	2808.6	3071.8	7.8926	0.8753	2806.7	3069.3	7.7022	0.6548	2804.8	3066.8	7.5662
400	1.5493	2966.7	3276.6	8.2218	1.0315	2965.6	3275.0	8.0330	0.7726	2964.4	3273.4	7.8985
500	1.7814	3130.8	3487.1	8.5133	1.1867	3130.0	3486.0	8.3251	0.8893	3129.2	3484.9	8.1913
600	2.013	3301.4	3704.0	8.7770	1.3414	3300.8	3703.2	8.5892	1.0055	3300.2	3702.4	8.4558
700	2.244	3478.8	3927.6	9.0194	1.4957	3478.4	3927.1	8.8319	1.1215	3477.9	3926.5	8.6987
800	2.475	3663.1	4158.2	9.2449	1.6499	3662.9	4157.8	9.0576	1.2312	3662.4	4157.3	8.9244
900	2.705	3854.5	4395.8	9.4566	1.8041	3854.2	4395.4	9.2692	1.3529	3853.9	4395.1	9.1362
1000	2.937	4052.5	4640.0	9.5663	1.9581	4052.3	4639.7	9.4690	1.4685	4052.0	4639.4	9.3360
1100	3.168	4257.0	4890.7	9.8458	2.1121	4256.8	4890.4	9.6585	1.5840	4256.5	4890.2	9.5256
1200	3.399	4467.5	5147.5	10.0262	2.2661	4467.2	5147.1	9.8389	1.6996	4467.0	5146.8	9.7060
1300	3.630	4683.2	5409.3	10.1982	2.4201	4683.0	5409.0	10.0110	1.8151	4682.8	5408.8	9.8780

Table A.5 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 5 bar (151.9 °C)				p = 6 bar (158.9 °C)				p = 8 bar (170.4 °C)			
Sat	0.3749	2561.2	2748.7	6.8213	0.3157	2567.4	2756.8	6.7600	0.2404	2576.8	2769.1	6.6628
200	0.4249	2642.9	2855.4	7.0592	0.3520	2638.9	2850.1	6.9665	0.2608	2630.6	2839.3	6.8158
250	0.4744	2723.5	2960.7	7.2709	0.3938	2720.9	2957.2	7.1816	0.2931	2715.5	2950.0	7.0384
300	0.5226	2802.9	3064.2	7.4599	0.4344	2801.0	3061.6	7.3724	0.3241	2797.2	3056.5	7.2328
350	0.5701	2882.6	3167.7	7.6329	0.4742	2881.2	3165.7	7.5464	0.3544	2878.2	3161.7	7.4089
400	0.6173	2963.2	3271.9	7.7938	0.5137	2962.1	3270.3	7.7079	0.3843	2959.7	3267.1	7.5716
500	0.7109	3128.4	3483.9	8.0873	0.5920	3127.6	3482.8	8.0021	0.4433	3126.0	3480.6	7.8673
600	0.8041	3299.6	3701.7	8.3522	0.6697	3299.1	3700.9	8.2674	0.5018	3297.9	3699.4	8.1333
700	0.8969	3477.5	3925.9	8.5952	0.7472	3477.0	3925.3	8.5107	0.5601	3476.2	3924.2	8.3770
800	0.9896	3662.1	4156.9	8.8211	0.8215	3661.8	4156.5	8.7367	0.6181	3661.1	4155.6	8.6033
900	1.0822	3853.6	4394.7	9.0329	0.9017	3853.4	4394.4	8.9486	0.6761	3852.8	4393.7	8.8153
1000	1.1747	4051.8	4639.1	9.2328	0.9788	4051.5	4638.8	9.1485	0.7340	4051.0	4638.2	9.0153
1100	1.2672	4256.3	4889.9	9.4224	1.0559	4256.1	4889.6	9.3381	0.7919	4255.6	4889.1	9.2050
1200	1.3596	4466.8	5146.6	9.6029	1.1330	4466.5	5146.3	9.5185	0.8497	4466.1	5145.9	9.3855
1300	1.4521	4682.5	5408.6	9.7749	1.2101	4682.3	5408.3	9.6906	0.9076	4681.8	5407.9	9.5575

Table A.6 Superheated steam table

T °C	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kgK	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kgK	*v* m^3/kg	*u* kJ/kg	*h* kJ/kg	*s* kJ/kgK
	p = 10 bar (179.9 °C)				p = 12 bar (188 °C)				p = 14 bar (195 °C)			
Sat	0.19444	2583.6	2778.1	6.5865	0.16333	2588.8	2784.8	6.5233	0.14084	2592.8	2790.0	6.4693
200	0.2060	2621.9	2827.9	6.6940	0.16930	2612.8	2815.9	6.5898	0.14302	2603.1	2803.3	6.4975
250	0.2327	2709.9	2942.6	6.9247	0.19234	2704.2	2935.0	6.8294	0.16350	2698.3	2927.2	6.7467
300	0.2579	2793.2	3051.2	7.1229	0.2138	2789.2	3045.8	7.0317	0.18228	2785.2	3040.4	6.9534
350	0.2825	2875.2	3157.7	7.3011	0.2345	2872.2	3153.6	7.2121	0.2003	2869.2	3149.5	7.13^0
400	0.3066	2957.3	3263.9	7.4651	0.2548	2954.9	3260.7	7.3774	0.2178	2952.5	3257.5	7.3026
500	0.3541	3124.4	3478.5	7.7622	0.2946	3122.8	3476.3	7.6759	0.2521	3121.1	3474.1	7.6027
600	0.4011	3296.8	3697.9	8.0290	0.3339	3295.6	3696.3	7.9435	0.2860	3294.4	3694.8	7.8710
700	0.4478	3475.3	3923.1	8.2731	0.3729	3474.4	3922.0	8.1881	0.3195	3473.6	3920.8	8.1160
800	0.4943	3660.4	4154.7	8.4996	0.4118	3659.7	4153.8	8.4148	0.3528	3659.0	4153.0	8.3431
900	0.5407	3852.2	4392.9	8.7118	0.4505	3851.6	4392.2	8.6272	0.3861	3851.1	4391.5	8.5556
1000	0.5871	4050.5	4637.6	8.9119	0.4892	4050.0	4637.0	8.8274	0.4192	4049.5	4636.4	8.7559
1100	0.6335	4255.1	4888.6	9.1017	0.5278	4254.6	4888.0	9.0172	0.4524	4254.1	4887.5	8.9457
1200	0.6798	4465.6	5145.4	9.2822	0.5665	4465.1	5144.9	9.1977	0.4855	4464.7	5144.4	9.1262
1300	0.7261	4681.3	5407.4	9.4543	0.6051	4680.9	5407.0	9.3698	0.5186	4680.4	5406.5	9.2984

Table A.7 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 16 bar (201.4 °C)				p = 18 bar (207.2 °C)				p = 20 bar (212.4 °C)			
Sat	0.12380	2596.0	2794.0	6.4218	0.11042	2598.4	2797.1	6.3794	0.09963	2600.3	2799.5	6.3409
225	0.13287	2644.7	2857.3	6.5518	0.11673	2636.6	2846.7	6.4808	0.10377	2628.3	2835.8	6.4147
250	0.14184	2692.3	2919.2	6.6732	0.12497	2686.0	2911.0	6.6066	0.11144	2679.6	2902.5	6.5453
300	0.15862	2781.1	3034.8	6.8844	0.14021	2776.9	3029.2	6.8226	0.12547	2772.6	3023.5	6.7664
350	0.17456	2866.1	3145.4	7.0694	0.15457	2863.0	3141.2	7.0100	0.13857	2859.8	3137.0	6.9563
400	0.19005	2950.1	3254.2	7.2374	0.16847	2947.7	3250.9	7.1794	0.15120	2945.2	3247.6	7.1271
500	0.2203	3119.5	3472.0	7.5390	0.19550	3117.9	3469.8	7.4825	0.17568	3116.2	3467.6	7.4317
600	0.2500	3293.3	3693.2	7.8080	0.2220	3292.1	3691.7	7.7523	0.19960	3290.9	3690.1	7.7024
700	0.2794	3472.7	3919.7	8.0535	0.2482	3471.8	3971.8	7.9983	0.2232	3470.9	3917.4	7.9487
800	0.3086	3658.3	4152.1	8.2808	0.2742	3657.6	4151.2	8.2258	0.2467	3657.0	4150.3	8.1765
900	0.3377	3850.5	4390.8	8.4935	0.3001	3849.9	4390.1	8.4386	0.2700	3849.3	4389.4	8.3895
1000	0.3668	4049.0	4635.8	8.6938	0.3260	4048.5	4635.2	8.6391	0.2933	4048.0	4634.6	8.5901
1100	0.3958	4253.7	4887.0	8.8837	0.3518	4253.2	4886.4	8.8290	0.3166	4252.7	4885.9	8.7800
1200	0.4248	4464.2	5143.9	9.0643	0.3776	4463.7	5143.4	9.0096	0.3398	4463.3	5142.9	8.9607
1300	0.4538	4679.9	5406.0	9.2364	0.4034	4679.5	5405.6	9.1818	0.3631	4679.0	5405.1	9.1329

Table A.8 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 25 bar (224 °C)				p = 30 bar (233.9 °C)				p = 35 bar (242.6 °C)			
Sat	0.07998	2603.1	2803.1	6.2575	0.06668	2604.1	2804.2	6.1869	0.05707	2603.7	2803.4	6.1253
225	0.08027	2605.6	2806.3	6.2639								
250	0.08700	2662.6	2880.1	6.4085	0.07058	2644.0	2855.8	6.2872	0.05872	2623.7	2829.2	6.1749
300	0.09890	2761.6	3008.8	6.6438	0.08114	2750.1	2993.5	6.5390	0.06842	2738.0	2977.5	6.4461
350	0.10976	2851.9	3126.3	6.8403	0.09053	2843.7	3115.3	6.7428	0.07678	2835.3	3104.0	6.6579
400	0.12010	2939.1	3239.3	7.0148	0.09936	2932.8	3230.9	6.9212	0.08453	2926.4	3222.3	6.8405
450	0.13014	3025.5	3350.8	7.1746	0.10787	3020.4	3344.0	7.0834	0.09196	3015.3	3337.2	7.0052
500	0.13993	3112.1	3462.1	7.3234	0.11619	3108.0	3456.5	7.2338	0.09918	3103.0	3450.9	7.1572
600	0.15930	3288.0	3686.3	7.5960	0.13243	3285.0	3682.3	7.5085	0.11324	3282.1	3678.4	7.4339
700	0.17832	3468.7	3914.5	7.8435	0.14838	3466.5	3911.7	7.7571	0.12699	3464.3	3908.8	7.6837
800	0.19716	3655.3	4148.2	8.0720	0.16414	3653.5	4145.9	7.9862	0.14056	3651.8	4143.7	7.9134
900	0.21590	3847.9	4387.6	8.2853	0.17980	3846.5	4385.9	8.1999	0.15402	3845.0	4384.1	8.1276
1000	0.2346	4046.7	4633.1	8.4861	0.19541	4045.4	4631.6	8.4009	0.16743	4044.1	4630.1	8.3288
1100	0.2532	4251.5	4884.6	8.6762	0.21098	4250.3	4883.3	8.5912	0.18080	4249.2	4881.9	8.5192
1200	0.2718	4462.1	5141.7	8.8569	0.22652	4460.9	5140.5	8.7720	0.19415	4459.8	5139.3	8.7000
1300	0.2905	4677.8	5404.0	9.0291	0.24206	4676.6	5402.8	8.9442	0.20749	4675.5	5401.7	8.8723

Table A.9 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 40 bar (250.4 °C)				p = 45 bar (257.5 °C)				p = 50 bar (264 °C)			
Sat	0.04978	2602.3	2801.4	6.0701	0.04406	2600.1	2798.3	6.0198	0.03944	2597.1	2794.3	5.9734
275	0.05457	2667.9	2886.2	6.2285	0.04730	2650.3	2863.2	6.1401	0.04141	2631.3	2838.3	6.0544
300	0.05884	2725.3	2960.7	6.3615	0.05135	2712.0	2943.1	6.2828	0.04532	2698.0	2924.5	6.2084
350	0.06645	2826.7	3092.5	6.5821	0.05840	2817.8	3080.6	6.5131	0.05194	2808.7	3068.4	6.4493
400	0.07341	2919.9	3213.6	6.7690	0.06475	2913.3	3204.7	6.7047	0.05781	2906.6	3195.7	6.6459
450	0.08002	3010.2	3330.3	6.9363	0.07074	3005.0	3323.3	6.8746	0.06330	2999.7	3316.2	6.8186
500	0.08643	3099.5	3445.3	7.0901	0.07651	3095.3	3439.6	7.0301	0.06857	3091.0	3433.8	6.9759
600	0.09885	3279.1	3674.4	7.3688	0.08765	3276.0	3670.5	7.3110	0.07869	3273.0	3666.5	7.2589
700	0.11095	3462.1	3905.9	7.6198	0.09847	3459.9	3903.0	7.5631	0.08849	3457.6	3900.1	7.5122
800	0.12287	3650.0	4141.5	7.8502	0.10911	3648.3	4139.3	7.7942	0.09811	3646.6	4137.1	7.7440
900	0.13469	3843.6	4382.3	8.0647	0.11965	3842.2	4380.6	8.0091	0.10762	3840.7	4378.8	7.9593
1000	0.14645	4042.9	4628.7	8.2662	0.13013	4041.6	4627.2	8.2108	0.11707	4040.4	4625.7	8.1612
1100	0.15817	4248.0	4880.6	8.4567	0.14056	4246.8	4879.3	8.4015	0.12648	4245.6	4878.0	8.3520
1200	0.16987	4458.6	5138.1	8.6376	0.15098	4457.5	5136.9	8.5825	0.13587	4456.3	5135.7	8.5331
1300	0.18156	4674.3	5400.5	8.8100	0.16139	4673.1	5399.4	8.7549	0.14526	4672.0	5398.2	8.7055

Table A.10 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK
	p = 60 bar (275.6 °C)				p = 70 bar (285.9 °C)				p = 80 bar (295.1 °C)			
Sat	0.03244	2589.7	2784.3	5.8892	0.02737	2580.5	2772.1	5.8133	0.02352	2569.0	2758.0	5.7432
300	0.03616	2667.2	2884.2	6.0674	0.02947	2632.2	2838.4	5.9305	0.02426	2590.9	2785.0	5.7906
350	0.04223	2789.6	3043.0	6.3335	0.03524	2769.4	3016.0	6.2283	0.02995	2747.7	2987.3	6.1301
400	0.04739	2892.0	3177.2’	6.5408’	0.03993	2878.6	3158.1	6.4478	0.03432	2863.8	3138.3	6.3634
450	0.05214	2988.9	3301.8	6.7193	0.04416	2978.0	3287.1	6.6327	0.03817	2966.7	3272.0	6.5551
500	0.05665	3082.2	3422.2	6.8803	0.04814	3073.4	3410.3	6.7980	0.04175	3064.3	3398.3	6.7240
550	0.06101	3174.6	3540.6	7.0288	0.05195	3167.2	3530.9	6.9486	0.04516	3159.8	3521.0	6.8778
600	0.06525	3266.9	3658.4	7.1677	0.05565	3260.7	3650.3	7.0894	0.04845	3254.4	3642.0	7.0206
700	0.07352	3453.1	3894.2	7.4234	0.06283	3448.5	3888.3	7.3476	0.05481	3443.9	3882.4	7.2812
800	0.08160	3643.1	4132.7	7.6566	0.06981	3639.5	4128.2	7.5822	0.06097	3636.0	4123.8	7.5173
900	0.08958	3837.8	4375.3	7.8727	0.07669	3835.0	4371.8	7.7991	0.06702	3832.1	4368.3	7.7351
1000	0.09749	4037.8	4622.7	8.0751	0.08350	4035.3	4619.8	8.0020	0.07301	4032.8	4616.9	7.9384
1100	0.10536	4243.3	4875.4	8.2661	0.09027	4240.9	4872.8	8.1933	0.07896	4238.6	4870.3	8.1300
1200	0.11321	4454.0	5133.3	8.4474	0.09703	4451.7	5130.9	8.3747	0.08489	4449.5	5128.5	8.3115
1300	0.12106	4669.6	5396.0	8.6199	0.10377	4667.3	5393.7	8.5475	0.09080	4665.0	5391.5	8.4842

Table A.11 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K
	p = 90 bar (303.4 °C)				p = 100 bar (311.1 °C)				p = 125 bar (327.9 °C)			
Sat	0.02048	2557.8	2742.1	5.6112	0.018026	2544.4	2724.7	5.6141	0.013495	2505.1	2673.8	5.4624
325	0.02327	2646.6	2856.0	5.8712	0.019861	2610.4	2809.1	5.7568				
350	0.02580	2724.4	2956.6	6.0361	0.02242	2699.2	2923.4	5.9443	0.016126	2624.6	2826.2	5.7118
400	0.02993	2848.4	3117.8	6.2854	0.02641	2832.4	3096.5	6.2120	0.02000	2789.3	3039.3	6.0417
450	0.03350	2955.2	3256.6	6.4844	0.02975	2943.4	3240.9	6.4190	0.02299	2912.5	3199.8	6.2719
500	0.03677	3055.2	3386.1	6.6576	0.03279	3045.8	3373.7	6.5966	0.02560	3021.7	3341.8	6.4618
550	0.03987	3152.2	3511.0	6.8142	0.03564	3144.1	3500.9	6.7561	0.02801	3125.0	3475.2	6.6290
600	0.04285	3248.1	3633.7	6.9589	0.03837	3241.7	3625.3	6.9029	0.03029	3225.4	3604.0	6.7810
650	0.04574	3343.6	3755.3	7.0943	0.04101	3338.2	3748.2	7.0398	0.03248	3324.4	3730.4	6.9218
700	0.04857	3439.3	3876.5	7.2221	0.04358	3434.7	3870.5	7.1687	0.03460	3422.9	3855.3	7.0536
800	0.05409	3632.5	4119.3	7.4596	0.04859	3628.9	4114.8	7.4077	0.03869	3620.0	4103.6	7.2965
900	0.05950	3829.2	4364.8	7.6783	0.05349	3826.3	4361.2	7.6272	0.04267	3819.1	4352.5	7.5182
1000	0.06485	4030.3	4614.0	7.8821	0.05832	4027.8	4611.0	7.8315	0.04658	4021.6	4603.8	7.7237
1100	0.07016	4236.3	4867.7	8.0740	0.06312	4234.0	4865.1	8.0237	0.05045	4228.2	4858.8	7.9165
1200	0.07544	4447.2	5126.2	8.2556	0.06789	4444.9	5123.8	8.2055	0.05430	4439.3	5118.0	8.0937
1300	0.08072	4662.7	5389.2	8.4284	0.07265	4460.5	5387.0	8.3783	0.05813	4654.8	5381.4	8.2717

Table A.12 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K
	p = 150 bar (342.2 °C)				p = 175 bar (354.7 °C)				p = 200 bar (365.8 °C)			
Sat	0.010337	2455.5	2610.5	5.3098	0.007920	2390.2	2528.8	5.1419	0.005834	2293.0	2409.7	4.9269
350	0.011470	2520.4	2692.4	5.4421								
400	0.015649	2740.7	2975.5	5.8811	0.012447	2685.0	2902.9	5.7213	0.009942	2619.3	2818.1	5.5540
450	0.018445	2879.5	3156.2	6.1404	0.015174	2844.2	3109.7	6.0184	0.012695	2806.2	3060.1	5.9017
500	0.02080	2996.6	3308.6	6.3443	0.017358	2970.3	3274.1	6.2383	0.014768	2942.9	3238.2	6.1401
550	0.02293	3104.7	3448.6	6.5199	0.019288	3083.9	3421.4	6.4230	0.016555	3062.4	3393.5	6.3348
600	0.02491	3208.6	3582.3	6.6776	0.02106	3191.5	3560.1	6.5866	0.018178	3174.0	3537.6	6.5048
650	0.02680	3310.3	3712.3	6.8224	0.02274	3296.0	3693.9	6.7357	0.019693	3281.4	3675.3	6.6582
700	0.02861	3410.9	3840.1	6.9572	0.02434	3324.6	3824.6	6.8736	0.02113	3386.4	3809.0	6.7993
800	0.03210	3610.9	4092.4	7.2040	0.02738	3601.8	4081.1	7.1244	0.02385	3592.7	4069.7	7.0544
900	0.03546	3811.9	4343.8	7.4279	0.03031	3804.7	4335.1	7.3507	0.02645	3797.5	4326.4	7.2830
1000	0.03875	4015.4	4596.6	7.6348	0.03316	4009.3	4589.5	7.5589	0.02897	4003.1	4582.5	7.4925
1100	0.04200	4222.6	4852.6	7.8283	0.03597	4216.9	4846.4	7.7531	0.03145	4211.3	4840.2	7.6874
1200	0.04523	4433.8	5112.3	8.0108	0.03876	4428.3	5106.6	7.9360	0.03391	4422.8	5101.0	7.8707
1300	0.04845	4649.1	5376.0	8.1840	0.04154	4643.5	5370.5	8.1093	0.03636	4638.0	5365.1	8.0442

Table A.13 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K
	p = 250 bar				p = 300 bar				p = 350 bar			
375	0.001973	1798.7	1848.0	4.0320	0.0017892	1737.8	1791.5	3.9305	0.0017003	1702.9	1762.4	3.8722
400	0.006004	2430.1	2580.2	5.1418	0.002790	2067.4	2151.1	4.4728	0.002100	1914.1	1987.6	4.2126
425	0.007881	2609.2	2806.3	5.4723	0.005303	2455.1	2614.2	5.1504	0.003428	2253.4	2373.4	4.7747
450	0.009162	2720.7	2949.7	5.6744	0.006735	2619.3	2821.4	5.4424	0.004961	2498.7	2672.4	5.1962
500	0.011123	2884.3	3162.4	5.9592	0.008678	2820.7	3081.1	5.7905	0.006927	2751.9	2994.4	5.6282
550	0.012724	3017.5	3335.6	6.1765	0.010168	2970.3	3275.4	6.0342	0.008345	2921.0	3213.0	5.9026
600	0.014137	3137.9	3491.4	6.3602	0.011446	3100.5	3443.9	6.2331	0.009527	3062.0	3395.5	6.1179
650	0.015433	3251.6	3637.4	6.5229	0.012596	3221.0	3598.9	6.4058	0.010575	3189.8	3559.9	6.3010
700	0.016646	3361.3	3777.5	6.6707	0.013661	3335.8	3745.6	6.5606	0.011533	3309.8	3713.5	6.4631
800	0.018912	3574.3	4047.1	6.9345	0.015623	3555.5	4024.2	6.8332	0.013278	3536.7	4001.5	6.7450
900	0.021045	3783.0	4309.1	7.1680	0.017448	3768.5	4291.9	7.0718	0.014883	3754.0	4274.9	6.9386
1000	0.02310	3990.9	4568.5	7.3802	0.019196	3978.8	4554.7	7.2867	0.016410	3966.7	4541.1	7.2064
1100	0.02512	4200.2	4828.2	7.5765	0.020903	4189.2	4816.3	7.4845	0.017895	4178.3	4804.6	7.4037
1200	0.02711	4412.0	5089.9	7.7605	0.022589	4401.3	5079.0	7.6692	0.019360	4390.7	5068.3	7.5910
1300	0.02910	4626.9	5354.4	7.9342	0.024266	4616.0	5344.0	7.8432	0.020815	4605.1	5333.6	7.7653

Table A.14 Superheated steam table

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kgK	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg K
	p = 400 bar				p = 500 bar				p = 600 bar			
375	0.0016407	1677.1	1742.8	3.8290	0.0015594	1638.6	1716.6	3.7639	0.0015028	1609.4	1699.5	3.7141
400	0.0019077	1854.6	1930.9	4.1135	0.0017309	1788.1	1874.6	4.0031	0.0016335	1745.4	1843.4	3.9318
425	0.002532	2096.9	2198.1	4.5029	0.002007	1959.7	2060.0	4.2734	0.0018165	1892.7	2001.7	4.1626
450	0.003693	2365.1	2512.8	4.9459	0.002486	2159.6	2284.0	4.5884	0.002085	2053.9	2179.0	4.4121
500	0.005622	2678.4	2903.3	5.4700	0.003892	2525.5	2720.1	5.1726	0.002956	2390.6	2567.9	4.9321
550	0.006984	2869.7	3149.1	5.7785	0.005118	2763.6	3019.5	5.5485	0.003956	2658.8	2896.2	5.3441
600	0.008094	3022.6	3346.4	6.0144	0.006112	2942.0	3247.6	5.8178	0.004834	2861.1	3151.2	5.6452
650	0.009063	3158.0	3520.6	6.2054	0.006966	3093.5	3441.8	6.0342	0.005595	3028.8	3364.5	5.8829
700	0.009941	3283.6	3681.2	6.3750	0.007727	3230.5	3616.8	6.2189	0.006272	3177.2	3553.5	6.0824
800	0.011523	3517.8	3978.7	6.6662	0.009076	3479.8	3933.6	6.5290	0.007459	3441.5	3889.1	6.4109
900	0.012962	3739.4	4257.9	6.9150	0.010283	3710.3	4224.4	6.7882	0.008508	3681.0	4191.5	6.6805
1000	0.014324	3954.6	4527.6	7.1356	0.011411	3930.5	4501.1	7.0146	0.009480	3906.4	4475.2	6.9121
1100	0.015642	4167.4	4793.1	7.3364	0.012496	4145.7	4770.5	7.2184	0.010409	4124.1	4748.6	7.1195
1200	0.016940	4380.1	5057.7	7.5224	0.013561	4359.1	5037.2	7.4058	0.011317	4338.2	5017.2	7.3083
1300	0.018229	4594.3	5323.5	7.6969	0.014616	4572.8	5303.6	7.5808	0.012215	4551.4	5284.3	7.4837