Problems and Solutions

11-1. Equations of state for a single component can be any of the following, *except*:

(a) the ideal gas law, $Pv = RT$

(b) the ideal gas law modified by insertion of a compressibility factor, $Pv = ZRT$

(c) any relationship interrelating three or more state functions

(d) a mathematical expression defining a path between states

(e) relationships mathematically interrelating thermodynamic properties of the material

Solution

All *except* (d) are correct The ideal gas law is the simplest equation of state; it is often applied to real gases by using a compressibility factor Z Any relationships that interrelate thermodynamic state function data are equations of state Answer (d) expresses the path of a process between states rather than a relationship between variables at a single point or state. The answer is (d).

11-2. On the Mollier diagram for steam, which of the numbered lines represents a line of constant pressure?

(a) Line 1 (b) Line 2 (c) Line 3 (d) Line 4 (e) Line 5

Solution

(a) constant moisture 100 – x% or quality x% (Line 1)

(b) constant temperature, °F (Line 2)

(c) constant superheat, °F (Line 3)

(d) constant pressure, psia (Line 4)

(e) constant entropy, BTU/lb °R (Line 5)

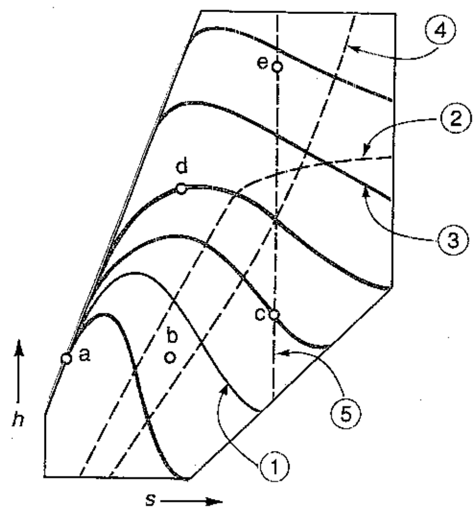

Fig. 11-2

Horizontal lines are constant enthalpy or total heat content. The answer is (d).

Thermodynamics

11-3.

Saturated Property Table

T, °F	P, psia	v_f, ft³	v_g ft³	h_f BTU/lb	h_g, BTU/lb	s_f, BTU/lb°R	s_g, BTU/lb°R
40	51.7	0.0116	0.774	17.3	81.4	0.0375	0.166
80	98.9	0.0123	0.411	26.4	85.3	0.0548	0.164
120	172	0.0132	0.233	36.0	88.6	0.0717	0.162

Given the above data for Freon 12, what is its state at 40 °F and 25 psia?

(a) saturated liquid (d) saturated vapor
(b) superheated vapor (e) vapor-liquid mixture
(c) compressed liquid

Solution

At 40 °F equilibrium between liquid and gas exists at 51.7 psia. Below 51.7 psia superheated vapor exists, and above 51.7 psia only pressurized liquid exists. The answer is (b).

11-4. Using the previous Freon 12 data table, what is its entropy in BTU/lb °R at 120 °F and 80% quality?

(a) 0.057 (d) 28.8
(b) 0.144 (e) none of these
(c) 0.186

Solution

At 120 °F, $s_f = 0.0717$ and $s_{fg} = 0.162 - 0.0717 = 0.090$. Here s_f is saturated liquid at 0% quality and s_g is saturated vapor of 100% quality. Thus s at 80% quality = $s_f + (0.80)\ s_{fg}$ = $0.0717 + (0.80)(0.090) = 0.144$ BTU/lb °R. The answer is (b).

11-5. Using the previous Freon 12 data table, what is its latent heat (heat of vaporization) in BTU/lb at 80 °F?

(a) 0.219 (d) 58.9
(b) 0.423 (e) none of these
(c) 26.4

Thermodynamics

Solution

Here, $h_{fg} = h_g - h_f = 85.3 - 26.4 = 58.9$ BTU/lb
The answer is (d).

11-6. A nonflow (closed) system contains 1 pound of an ideal gas ($C_p = 0.24$, $C_v = 0.17$). The gas temperature is increased by 10 °F while 5 BTU of work are done by the gas. What is the heat transfer in BTU?

(a) −3.3 (d) +7.4
(b) −2.6 (e) none of these
(c) +6.7

Solution

The thermodynamic sign convention is + for heat in and + for work out of a system. Apply the first law for a closed system and an ideal gas working fluid:

$$\Delta U = mC_v \Delta T = q - w$$

$$0.17(10) = q - (+5), \quad 1.7 = q - 5, \quad q = 6.7$$

The answer is (c).

11-7. Shaft work of –15 BTU/lb and heat transfer of –10 BTU/lb change the enthalpy of a system by

(a) –25 BTU/lb
(b) –15 BTU/lb
(c) –10 BTU/lb
(d) –5 BTU/lb
(e) +5 BTU/lb

Solution

The first law applied to a flow system is

$$h = q - w_s = -10 - (-15) = +5$$

The answer is (e).

11-8. A quantity of 55,000 gallons of water passes through a heat exchanger and absorbs 28,000,000 BTUs. The exit temperature is 110 °F. The entrance water temperature in °F is nearest to:

(a) 49
(b) 56
(c) 68
(d) 73
(e) 82

Solution

For liquid water, $C_p = 1.0$ BTU/(lb °F)

$$Q = mC_p\Delta T = mC_p(T_2 - T_1)$$

$$28{,}000{,}000 = (55{,}000 \text{ gal})\left(\frac{8.33 \text{ lb}}{\text{gallon}}\right)(1.0)(110 - T_1)$$

$$61.1 = 110 - T_1 \quad T_1 = 48.9 \text{ °F}$$

The answer is (a).

11-9. A fluid at 100 psia has a specific volume of 4 ft^3/lb and enters an apparatus with a velocity of 500 ft/sec. Heat radiation losses in the apparatus are equal to 10 BTU/lb of fluid supplied. The fluid leaves the apparatus at 20 psia with a specific volume of 15 ft^3/lb and a velocity of 1000 ft/sec. In the apparatus, the shaft work done by the fluid is equal to 195,000 ft-lbf/lbm. Does the internal energy of the fluid increase or decrease, and how much is the change?

(a) 230 BTU/lb (increase)
(b) 244 BTU/lb (increase)
(c) 250 BTU/lb (decrease)
(d) 257 BTU/lb (decrease)
(e) 294 BTU/lb (decrease)

Thermodynamics

Solution

The basis of the calculation will be: 1 lbm

Use the thermodynamic sign convention that heat in and work out are positive. The first law energy balance for the flow system: $h_2 + KE_2 - h_1 - KE_1 = Q - W_s$. Since the working fluid is unspecified and the internal energy change is desired, use the definition $h = u + Pv$.

$$u_2 + P_2v_2 + KE_2 - u_1 - P_1v_1 - KE_1 = Q - W_s$$

or

$$u_2 - u_1 = Q - W_s + P_1v_1 + KE_1 - P_2v_2 - KE_2$$

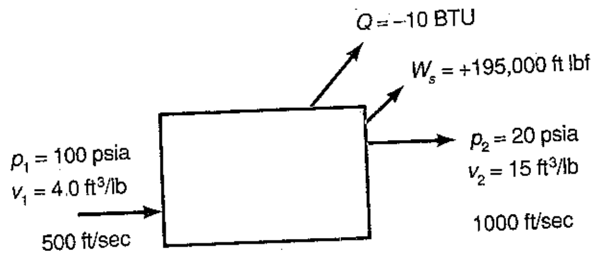

Fig. 11-9

Now calculate numerical values for all terms except $u_2 - u_1$:

$$P_2v_2 = \frac{20(144)(15)}{778} = 55.5 \text{ BTU/lb} \quad P_1v_1 = \frac{100(144)(4.0)}{778} = 74.0 \text{ BTU/lb}$$

$$KE_2 = \frac{V^2}{2gJ} = \frac{(1000)^2}{(64.4)(778)} = 20.0 \text{ BTU/lb}$$

$$KE_1 = \frac{V^2}{2gJ} = \frac{(500)^2}{(64.4)(778)} = 5.0 \text{ BTU/lb}$$

$$W_s = \frac{195{,}000 \text{ ft-lbf}}{\text{lbm}} \times \frac{\text{BTU}}{778 \text{ ft-lbf}} = +250.6 \text{ BTU/lb}$$

Therefore,

$u_2 - u_1 = -10 - 250.6 + 74.0 + 5.0 - 55.5 - 20.0 = -257.1$ BTU/lb (decrease)

The answer is (d).

11-10. Exhaust steam from a turbine exhausts into a surface condenser at a mass flow rate of 8000 lb/hr, 2 psia and 92% quality. Cooling water enters the condenser at 74 °F and leaves at the steam inlet temperature.

Properties of Saturated Water (US units): Temperature Table

v, ft³/lb; u and h, BTU/lb; s, BTU/(lb)(°R)

		Specific volume		Internal energy		Enthalpy			Entropy	
Temp. °F T	Press. psia P	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	Sat. vapor s_g
74	0.4158	0.01606	763.5	42.09	1035.0	42.09	1051.7	1093.8	0.08215	2.0526

The cooling water mass flow rate in lb/hr is closest to

(a) 157,200
(b) 144,700
(c) 95,000
(d) 88,000
(e) 8,000

Solution

Saturated steam table data at 2 psia are

T, °F	h_f, BTU/lb	h_{fg}, BTU/lb	h_g, BTU/lb
126.08	93.99	1022.2	1116.2

The enthalpy of steam at 92% quality = $h_1 = h_f + 0.92h_{fg}$

$$= 94.0 + 0.92(1022.2) = 1034.4 \text{ BTU/lb}$$

The enthalpy of liquid water at 126.1 °F = h_2 = 94.0 BTU/lb.

The enthalpy of liquid water at 74 °F = h_3 = 42.0 BTU/lb above reference of 32 °F.

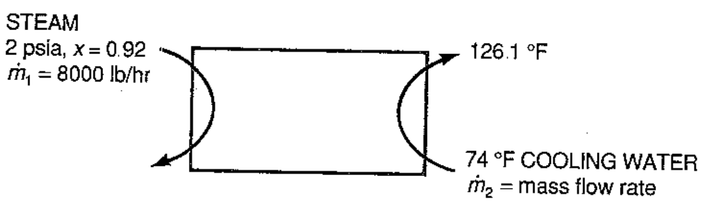

Fig 11-10

As an alternate, one can find h_f in steam tables at 74 °F ignoring negligible effect of pressure on enthalpy of incompressible liquids.

In the absence of data, assume that the steam condensate leaves at 126.1 °F; if a heat balance is written over a 1 hour period, then the heat from steam = heat to cooling water, or

$$\dot{m}_1(h_1 - h_2) = \dot{m}_2(h_2 - h_3)$$

$$8000(1034.4 - 94.0) = \dot{m}_2(94.0 - 42.0)$$

$$\dot{m}_2 = 144{,}700 \text{ lb/hr}$$

The answer is (b)

11-11. The mass flow rate of Freon 12 through a heat exchanger is 10 pounds/minute. The enthalpy of entry Freon is 102 BTU/lb and of exit Freon is 26 BTU/lb. Water coolant is allowed to rise 10 °F. The water flow rate in pounds/minute is

(a) 24
(b) 76
(c) 83
(d) 112
(e) 249

Thermodynamics

Solution

Over a 1-minute period, the heat gain by water equals heat loss by Freon

$$m_1 C_p \Delta T = m_2(h_1 - h_2)$$
$$m_1(1)(10) = 10(102 - 26)$$
$$m_1 = \frac{760}{10} = 76 \text{ lb/min.}$$

The answer is (b).

11-12. The maximum thermal efficiency that can be obtained in an ideal reversible heat engine operating between 1540 °F and 340 °F is closest to

(a) 100%
(b) 60%
(c) 78%
(d) 40%
(e) 22%

Solution

Maximum efficiency is achieved with a Carnot engine.

T_L = 340 °F + 460 ° = 800 °R; T_H = 1540 °F + 460 ° = 2000 °R

$$\eta_{TH} = \frac{w}{q_H} = \frac{q_H - q_L}{q_H}$$
$$= 1 - \frac{Q_L}{Q_H} = 1 - \frac{T_L}{T_H}$$
$$= 1 - \frac{800}{2000} = 1 - 0.40 = 0.60 = 60\%$$

The answer is (b).

Thermodynamics

11-13. A three HP refrigerator or heat pump operates between 0 °F and 100 °F. The maximum theoretical heat that can be transferred from the cold reservoir is nearest to

(a) 7600 BTU/hr
(b) 13,000 BTU/hr
(c) 23,000 BTU/hr
(d) 35,000 BTU/hr
(e) 43,000 BTU/hr

Solution

The coefficient of performance of a Carnot refrigerator or heat pump is

$$\text{COP} = \frac{T_L}{T_H - T_L} = \frac{460\ °\text{R}}{560\ °\text{R} - 460\ °\text{R}} = 4.6 = \frac{q_L}{w} = 4.6$$

Since 1 HP = 2545 BTU/hr, the work is w = 7632 BTU/hr

$$\text{COP} = \frac{q_L}{q_H - q_L} = \frac{q_L}{w} = \frac{q_L}{7632}; \quad q_L = 35{,}100 \text{ BTU/hr}$$

The answer is (d).

11-14. In any non-quasistatic thermodynamic process, the overall entropy of an isolated system will

(a) Increase and then decrease
(b) Decrease and then increase
(c) Stay the same
(d) Increase only
(e) Decrease only

Solution

Quasistatic means infinitely slow, lossless, hypothetical, by differential increments. The overall entropy will increase for an isolated system or for the system plus surroundings. The answer is (d).

11-15. For spontaneously occurring natural processes in an isolated system, which expression best expresses ds?

(a) $ds = \frac{dq}{T}$
(b) $ds = 0$
(c) $ds > 0$
(d) $ds < 0$
(e) $ds = C_p \frac{dT}{T} - R\frac{dP}{P}$

Solution

(a) $ds = \frac{dq_{rev}}{T}$ only. The reversible requirement is necessary to generate the exact height vs. rectangular area equivalence on the Carnot cycle T-s diagram

(b) Only a reversible adiabatic process is isentropic by definition.

(c) All naturally occurring spontaneous processes are irreversible and result in an entropy increase

(d) An energy input from the surroundings is required to reduce the entropy.

(e) This is an expression for the entropy change in an ideal gas.

The answer is (c).

Thermodynamics

11-16. Which of the following statements about entropy is *false*?

(a) The entropy of a mixture is greater than that of its components under the same conditions.

(b) An irreversible process increases the entropy of the universe.

(c) Entropy has the units of heat capacity.

(d) The net entropy change in any closed cycle is zero.

(e) The entropy of a crystal at 0 °F is zero.

Solution

All are true except (e). The entropy of a perfect crystal at absolute zero (0 °K or 0 °R) is zero. This is the third law of thermodynamics. There is presumably no randomness at this temperature in a crystal without flaws, impurities or dislocations. The answer is (e).

11-17. A high velocity flow of gas at 800 ft/sec possesses kinetic energy nearest to which of the following?

(a) 1.03 BTU/lb
(b) 4.10 BTU/lb
(c) 9.95 BTU/lb
(d) 12.8 BTU/lb
(e) 41.0 BTU/lb

Solution

Per 1 lbm of flowing fluid

$KE = \dfrac{V^2}{2g_c}$ in ft-lbf, where V is in ft/sec, and $g_c = 32.17$

Use $J = 778$ ft-lbf /BTU to convert to BTU to obtain

$$KE = \frac{800^2}{2(32.17)(778)} = 12.8 \text{ BTU/lb}$$

The answer is (d)

11-18. $(u + Pv)$ is a quantity called

(a) flow energy
(b) shaft work
(c) entropy
(d) enthalpy
(e) internal energy

Solution

Flow energy is Pv. Shaft work, W_s, is $-\int vdP$. Entropy is s. Internal energy is u. Enthalpy h is defined as $u + Pv$, the sum of internal energy plus flow energy. The answer is (d).

Thermodynamics

11-19. In flow process, neglecting KE and PE changes, $-\int vdP$ represents which item below?

(a) heat transfer
(b) shaft work
(c) closed system work
(d) flow energy
(e) enthalpy change

Solution

Shaft work is work or mechanical energy crossing the fixed boundary (control volume) of a flow (open) system. Shaft work W_s is defined, in the absence of PE and KE changes, by $dh = Tds + vdP$, where $Tds = dq_{rev}$ and $-vdP$ is dW_s. In integrated form $\Delta h = \int Tds + \int vdP = q_{rev} - W_s$, where W_s is represented by $-\int vdP$. Closed system work W is defined by $du = Tds - Pdv$, or $\Delta u = \int Tds - \int Pdv = q_{rev} - W$. Thus closed system work is $+\int Pdv$. Flow energy is the Pv term, and enthalpy change is ΔH. The answer is (b).

11-20. Power may be expressed in units of

(a) ft-lb
(b) BTU/hr
(c) HP-hr
(d) kW/hr
(e) BTU

Ft-lb, HP-hr, BTU, and kW-hr are the usual mechanical, thermal and electrical energy units. Power is energy per unit time. The usual power units are ft-lb/sec, HP, BTU/hr and kW. The answer is (b).

11-21. Given the following data: electricity cost, \$0.015/kW-hr; natural gas cost, \$0.065/100 cu ft; heat content of gas, 1050 BTU/cu ft; how many more times as expensive is it to heat a house by electricity than to heat the same house by gas if the electric heat is assumed to be 100% efficient and the gas is 60% efficient? The ratio is nearest to

(a) 0.5
(b) 1.0
(c) 2.0
(d) 3.0
(e) 4.0

Solution

In converting foot-pounds to BTU or vice-versa, 1 BTU = 778 foot-pounds.

Heating cost:

$$\text{Natural Gas} = \frac{\$0.065}{0.60 \times 1050 \times 100} = \$1.03 \times 10^{-6}/\text{BTU}$$

$$\text{Electricity} = \frac{\$0.015 \times 0.746\text{ kW/HP} \times (1\text{ HP}/33{,}000\text{ ft-lb/min}) \times (1/60) \times 778}{1.0}$$

$$= \$4.40 \times 10^{-6}/\text{BTU}$$

$$\text{Ratio} = \frac{\text{cost of electrical heating}}{\text{cost of gas heating}} = \frac{\$4.40 \times 10^{-6}/\text{BTU}}{\$1.03 \times 10^{-6}/\text{BTU}} = 4.27$$

The answer is (e).

11-22. The temperature-entropy diagram in Fig 11-22 represents a

(a) Rankine cycle with superheated vapor
(b) Carnot cycle
(c) Diesel cycle
(d) Refrigeration cycle
(e) Adiabatic process

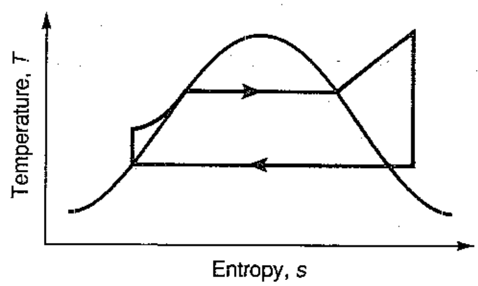

Fig. 11-22

Thermodynamics

Solution

The answer is (a).

11-23. Entropy is the measure of

(a) the change in enthalpy of a system
(b) the internal energy of a gas
(c) the heat capacity of a substance
(d) randomness or disorder
(e) the total heat content of a system

Solution

The answer is (d).

11-24. A Carnot heat engine cycle is represented on the T-s and P-V diagrams in Fig. 11-24. Which of the several areas bounded by numbers or letters represents the amount of heat rejected by the fluid during one cycle?

(a) Area 1–2–6–5
(b) Area B–C–H–G
(c) Area 3–4–5–6
(d) Area D–A–E–F
(e) Area C–D–F–H

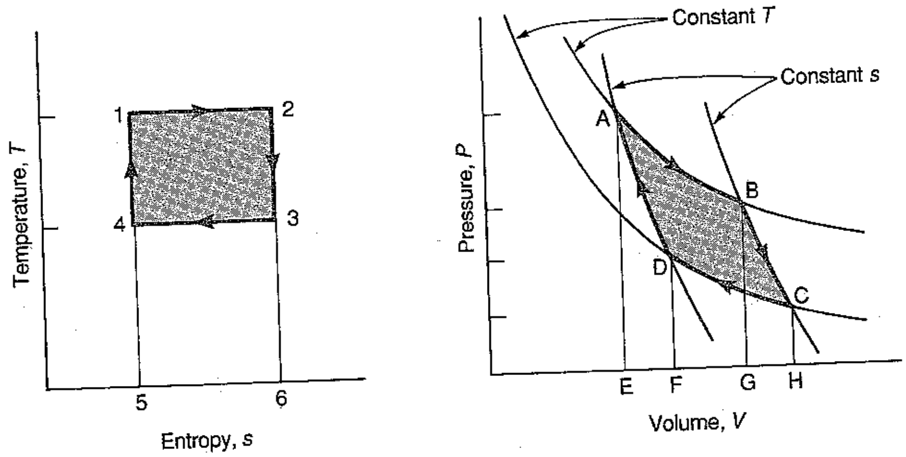

Fig. 11-24

Solution

The table below gives the significance of each area of the diagrams:

Process	T-s diagram Area representing heat	P-V diagram Area representing work
Isothermal expansion 1–2 and A–B	1–2–6–5 = heat in from high temp. reservoir	A–B–G–E = work done by fluid
Isentropic expansion 2–3 and B–C	2–3–6 = 0 heat transfer	B–C–H–G = work done by fluid
Isothermal compression 3–4 and C–D	3–4–5–6 = heat out to low temp. reservoir	C–D–F–H = work done on fluid
Isentropic compression 4–1 and D–A	4–1–5 = 0 heat transfer	D–A–E–F = work done on fluid
Net result of process	1–2–3–4 = net heat converted to work	A–B–C–D = net work done by process

The answer is (c).

11-25. A Carnot engine operating between 70 °F and 2000 °F is modified solely by raising the high temperature by 150 °F and raising the low temperature by 100 °F. Which of the following statements is *false*?

(a) The thermodynamic efficiency is increased.

(b) More work is done during the isothermal expansion.

(c) More work is done during the isentropic compression.

(d) More work is done during the reversible adiabatic expansion.

(e) More work is done during the isothermal compression

Solution

The Carnot cycle efficiency is originally

$$\eta = \frac{T_H - T_L}{T_H - 0} = \frac{2460\ °\text{R} - 530\ °\text{R}}{2460\ °\text{R}} = 0.785$$

After the change

$$\eta = \frac{2610 - 630}{2610} = 0.759 \text{ (efficiency is reduced)}$$

On the T-s and P-V diagrams in Fig. 11-25 the original cycle is shown as ABCD, and the modified cycle is shown as A′B′C′D′.

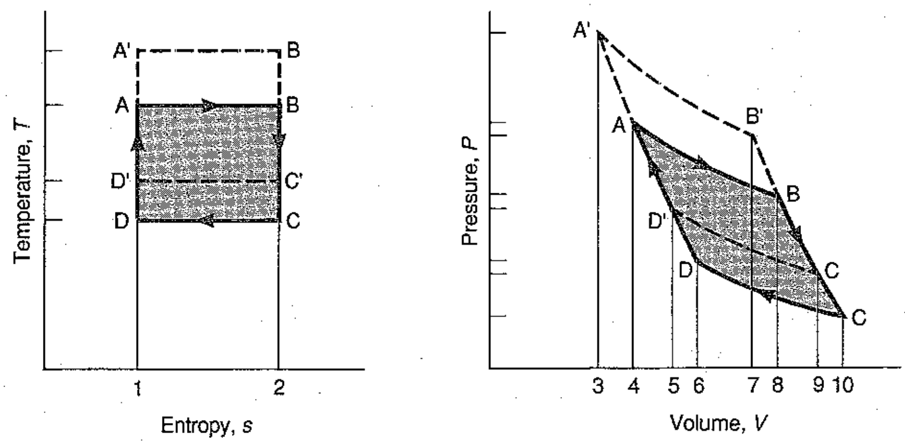

Fig 11-25

Compare the work done during the isothermal expansion (A to B vs. A′ to B′):

Original: area A–B–8–4

Modified: area A′–B′–7–3 is larger

Compare the work done during the isentropic compression (D to A vs. D′ to A′):

Original: area D–A–4–6

Modified: area D′–A′–3–5 is larger

Thermodynamics

Compare the work during the reversible (isentropic) expansion (B to C vs. B′ to C′):

Original: area B–C–10–8

Modified: area B′–C′–9–7 is larger

Compare the work during the isothermal compression (C to D vs. C′ to D′):

Original: area C–D–6–10

Modified: area C′–D′–5–9 is larger

Statements (b), (c), (d), and (e) are correct. The answer is (a).

11-26. In the ideal heat pump system represented in Fig. 11-26, the expansion valve 4–1 performs the process that is located on the *T*-*s* diagram between points

(a) A and B

(b) B and C

(c) C and D

(d) D and E

(e) E and A

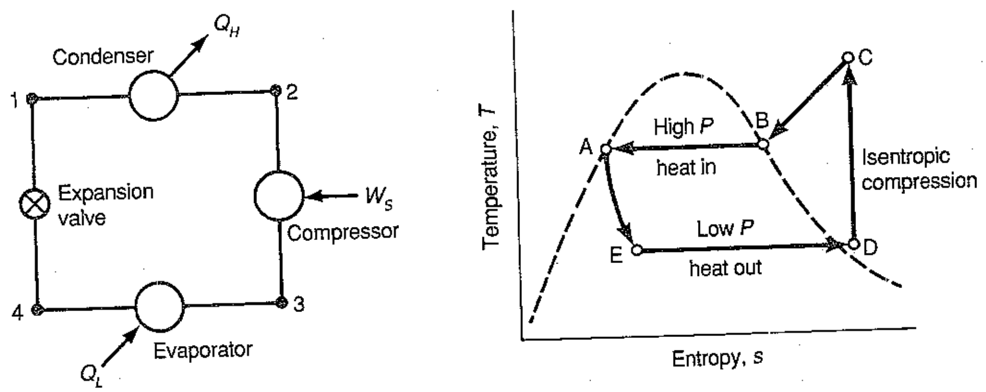

Fig. 11-26

Solution

The vapor compression reversed Rankine cycle is conducted counterclockwise on both the schematic and *T*-*s* diagrams. Numbers on the schematic and letters on the *T*-*s* diagram are related: 1 = A, 2 = B, 3 = D, and 4 = E. Process C–B–A occurs in the condenser between 2 and 1. The expansion process A–E occurs between 1–4. The answer is (e).

11-27. Which air-standard power cycle do the *P*-*V* and *T*-*s* diagrams in Fig. 11-27 represent?

(a) Otto cycle

(b) Reheat cycle

(c) Carnot cycle

(d) Rankine cycle

(e) Brayton cycle

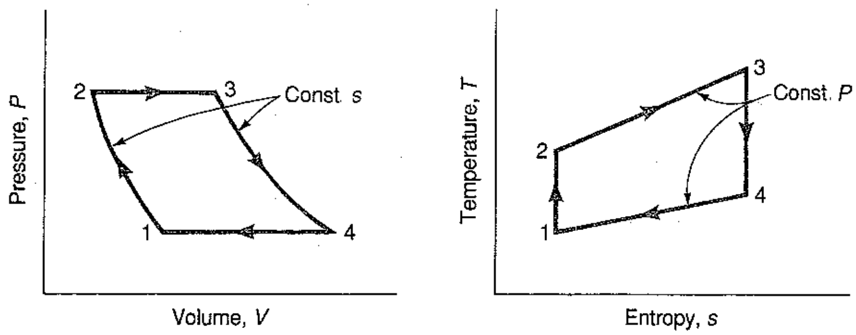

Fig. 11-27

Solution

The Brayton cycle is applied to the simple open-cycle gas turbine wherein intake air is compressed (1–2), combustion supplies thermal energy (2–3), and combustion products expand and drive the turbine (3–4) and exhaust at 4. The answer is (e).

11-28. Data in the table describe two states of a working fluid that exist at two locations in a piece of hardware.

	P, psia	v, ft³/lb	T, °F	h, BTU/lb	s, BTU/(lb °F)
State 1	25	0.011	20	19.2	0.0424
State 2	125	0.823	180	203.7	0.3649

Which of the following statements about the path from State 1 to 2 is *false*?

(a) The path results in an expansion

(b) The path determines the amount of work done.

(c) The path is indeterminate from these data.

(d) The path requires that energy be added in the process.

(e) The path is reversible and adiabatic.

Solution

The large volume and entropy changes indicate a change from a condensed phase to a vapor phase. Temperature, pressure, and enthalpy increases require an energy input. The path from 1 to 2 is indeterminate since no information on intermediate states is given. Work is always path dependent. The entropy increase means the process cannot be reversible and adiabatic (isentropic). The answer is (e)

11-29. Name the process that has no heat transfer.

(a) Isentropic

(b) Isothermal

(c) Quasistatic

(d) Reversible

(e) Polytropic

Thermodynamics

Solution

An *isentropic* process is reversible and adiabatic. An *adiabatic* process has no heat exchange with its surroundings. An *isothermal* process is conducted at constant temperature. A *quasistatic* (almost static) process departs only infinitesimally from an equilibrium state. A *reversible* process can have its initial state restored without any change (energy gain or loss) taking place in the surroundings. A *polytropic* process is conducted with changes in temperature, pressure, volume, and entropy; it follows the relationship PV^n = constant, where $n \neq C_p/C_v$. The answer is (a).

11-30. In a closed system with a moving boundary, which of the following represents work done during an isothermal process?

(a) $W = P(V_2 - V_1)$

(b) $W = 0$

(c) $W = P_1V_1 \ln\left(\frac{P_1}{P_2}\right) = P_1V_1 \ln\left(\frac{V_2}{V_1}\right) = mRT \ln\left(\frac{P_1}{P_2}\right)$

(d) $W = \frac{P_2V_2 - P_1V_1}{1-k} = \frac{mR(T_2 - T_1)}{1-k}$

(e) $W = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{mR(T_2 - T_1)}{1-n}$

Solution

For a closed system (piston-cylinder type, non-repetitious) the work done is $W = \int PdV$. The above equations are valid for ideal gases in the following processes:

(a) constant pressure
(b) constant volume
(c) isothermal process
(d) isentropic process
(e) polytropic process

The answer is (c).

11-31. The work of a polytropic (n = 1.21) compression of air (C_p/C_v = 1.40) in a system with moving boundary from P_1 = 15 psia, V_1 = 1.0 ft^3 to P_2 = 150 psia, V_2 = 0.15 ft^3 is

(a) 35.5 ft-lb
(b) 324 ft-lb
(c) 1080 ft-lb
(d) 2700 ft-lb
(e) 5150 ft-lb

Solution

The work of a closed system (moving boundary) polytropic process for an ideal gas is

$$W = \frac{P_2V_2 - P_1V_1}{1-n} = \frac{[150(0.15) - 15(1.0)]144}{1-1.21} = -5140 \text{ ft-lb}$$

which is work is done on the gas. The answer is (e)

11-32. The isentropic compression of 1 ft³ of air, $C_p/C_v = 1.40$, from 20 psia to a pressure of 100 psia gives a final volume of

(a) 0.16 ft³ (d) 0.40 ft³
(b) 0.20 ft³ (e) 0.56 ft³
(c) 0.32 ft³

Solution

An isentropic process for an ideal gas follows the path

$$PV^k = P_1V_1^k = P_2V_2^k = \text{constant, where } k = C_p/C_v$$

$$20(1)^{1.4} = 100(V_2)^{1.4}; \quad V_2^{1.4} = 0.20; \quad \text{hence } V_2 = 0.317 \text{ ft}^3$$

The answer is (c).

11-33. An ideal gas at a pressure of 500 psia and a temperature of 75 °F is contained in a cylinder with a volume of 700 cubic feet. Some of the gas is released so that the pressure in the cylinder drops to 250 psia. The expansion of the gas is isentropic. The specific heat ratio is 1.40, and the gas constant is 53.3 ft-lbf/lbm °R. The weight of the gas in lbm remaining in the cylinder is nearest to

(a) 900 (d) 1500
(b) 1100 (e) 1700
(c) 1300

Solution

Given:

$$k = C_p/C_v = 1.40 \qquad R = 53.3\ \frac{\text{ft-lbf}}{\text{lbm-°R}}$$
$$P_1 = 500 \text{ psia} \qquad P_2 = 250 \text{ psia}$$
$$V_1 = 700 \text{ cu ft} \qquad V_2 = 700 \text{ cu ft}$$
$$T_1 = 75\ °\text{F} + 460 = 535\ °\text{R} \qquad T_2 = ?$$
$$w_2 = ?$$

Thermodynamics

Basis:
The ideal gas law may be written $PV = wRT$ and the basic equation for reversible adiabatic (isentropic) expansion is

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

The gas remaining in the tank cools as it expands; the new temperature is

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = 535\left(\frac{250}{500}\right)^{\frac{1.4-1}{1.4}} = 535\left(\frac{1}{2}\right)^{0.2857} = 439\ °\text{R}$$

Now apply the gas law at State 2 $P_2V_2 = w_2RT_2$, and check dimensions to use the proper conversion factors.

$$250\frac{\text{lbf}}{\text{in}^2}\times\frac{144\text{ in}^2}{\text{ft}^2}\times 700\text{ ft}^3 = w_2\text{ lbm}\frac{53.3\text{ ft-lbf}}{\text{lbm-°R}}\times 439\text{ °R}$$

$$w_2 = \frac{(250)(144)(700)}{(53.3)(439)} = 1077\text{ lbm}$$

The answer is (b).

11-34. The theoretical horsepower required for the isothermal compression of 800 ft³/min of air from 14.7 to 120 psia is closest to

(a) 70
(b) 90
(c) 110
(d) 130
(e) 150

Solution

Since a volume flow rate is specified, the process is a flow process. The work of isothermal compression of an ideal gas is numerically the same in a steady flow process as in a closed system:

$$PV = \text{constant} = P_1V_1 = P_2V_2 = mRT$$

In a closed system, $W = \int_{V_1}^{V_2} PdV = P_1V_1 \ln\frac{V_2}{V_1} = P_1V_1 \ln\frac{P_1}{P_2} = mRT \ln\frac{V_2}{V_1} = mRT \ln\frac{P_1}{P_2}$

In a flow system, $W_s = \int_{P_1}^{P_2} VdP = -P_1V_1 \ln\frac{P_2}{P_1} = P_1V_1 \ln\frac{P_1}{P_2} = P_1V_1 \ln\frac{V_2}{V_1} = mRT \ln\frac{V_2}{V_1} = mRT \ln\frac{P_1}{P_2}$

Over a 1-minute interval

$$W_s = P_1V_1 \ln\frac{P_1}{P_2} = 14.7(144)(800)\ln\left(\frac{14.7}{120}\right) = -3,560,000\text{ ft-lb/min (work in)}$$

$$W_s = \frac{3,560,000\text{ ft-lb/min}}{33,000\text{ ft-lb/HP-min}} = 108\text{ HP}$$

The answer is (c).

11-35. Which of the following statements is false concerning the deviations of real gases from ideal gas behavior?

(a) Molecular attraction interactions are compensated for in the ideal gas law.
(b) Deviations from ideal gas behavior are often large near the saturation curve.
(c) Deviations from ideal gas behavior become significant at pressures above the critical point.
(d) Molecular volume becomes significant as specific volume is decreased.
(e) The compressibility factor Z is used to modify the ideal gas equation of state to fit real gas behavior

Thermodynamics

Solution

All statements except (a) are true The ideal gas law does not consider the volume of the molecules or any interaction other than elastic collisions. The anwer is (a).

11-36. There are 3 lb of air in a rigid container at 25 psia and 100 °F. The gas constant for air is 53.35 ft-lbf/lbm-°R The volume of the container in ft^3, is nearest to

(a) 22
(b) 25
(c) 28
(d) 31
(e) 34

Solution

The ideal gas law is $PV = mRT$.

Here $P = 25$ psia $\times$ 144 = 3600 lbf/ft^2, and $T_1 = 100$ °F + 460 = 560 °R

Hence

$$3600V = 3(53.35)(560)$$

$$V = \frac{3(53.35)(560)}{3600} = 24.9 \text{ ft}^3$$

The answer is (b).

11-37. A mixture at 14.7 psia and 68 °F that is 30% by weight CO_2 (m.w. = 44) and 70% by weight N_2 (m.w = 28) has a partial pressure of CO_2 in psia that is nearest to

(a) 2.14
(b) 3.15
(c) 6.83
(d) 7.86
(e) 11.55

Solution

The calculation is based on 1 lb of mixed gases 1) Calculate the weight of each component and the number of moles of each that is present. 2) Compute the mole fraction of each, and apportion the total pressure in proportion to the mole fraction. The computations are in the following table:

Component	Weight, lb	Number of lb-moles	Mole Fraction	Partial Pressure, psia
CO_2	0.30	$\frac{0.30}{44} = 0.00682$	$\frac{0.00682}{0.03182} = 0.214$	3.15
N_2	0.70	$\frac{0.70}{28} = 0.0250$	$\frac{0.0250}{0.03182} = 0.786$	11.55
Total	1.00	0.03182	1.000	14.70

Since the mole fraction of a gas is the same as the volume fraction, the composition of the mixture is 21.4% vol. CO_2 and 78.6% vol. N_2. From the table, the correct partial pressure of CO_2 is 3.15 psia. The answer is (b).

11-38. Dry air has an average molecular weight of 28.9, consisting of 21 mole-percent O_2, 78 mole-percent N_2 and 1 mole-percent Argon (and traces of CO_2). The weight-percent of O_2 is nearest to

(a) 21.0
(b) 22.4
(c) 23.2
(d) 24.6
(e) 28.0

Solution

The calculation will be based on 1 lb-mole of dry air and arranged in the following table:

Component	m.w.	Mole Fraction	Weight, lb.	Weight, %
O_2	32.0	0.21	6.72	23.2
N_2	28.0	0.78	21.80	75.4
Ar	40.0	0.01	0.40	1.4
Totals		1.00	28.92	100.0

The answer is (c)

11-39. The temperature difference between the two sides of a solid rectangular slab of area A and thickness L as shown in Fig. 11-39, is ΔT. The heat transferred through the slab by conduction in time, t, is proportional to

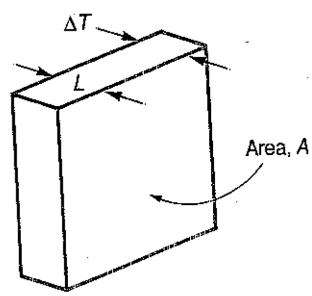

Fig 11-39

(a) $AL\Delta Tt$
(b) $AL\frac{\Delta T}{t}$
(c) $AL\frac{t}{\Delta T}$
(d) $\frac{A\Delta Tt}{L}$
(e) $\frac{A(\Delta T)^2 t}{L}$

Thermodynamics

Solution

The heat transfer rate through the slab by conduction is governed by the equation

$$Q = kA\Delta T/L$$

In time t the amount of heat transfer is proportional to

$$A\frac{\Delta T}{L}t$$

The symbol k is the coefficient of thermal conductivity of the material, hence the heat transfer in a given material is proportional to the other variables. The answer is (d)

11-40. The composite wall in Fig. 11-40 has an outer temperature $T_1 = 20$ °F and an inner temperature $T_4 = 70$ °F. The temperature T_3 in °F is nearest to

(a) 27 (d) 58

(b) 38 (e) 69

(c) 46

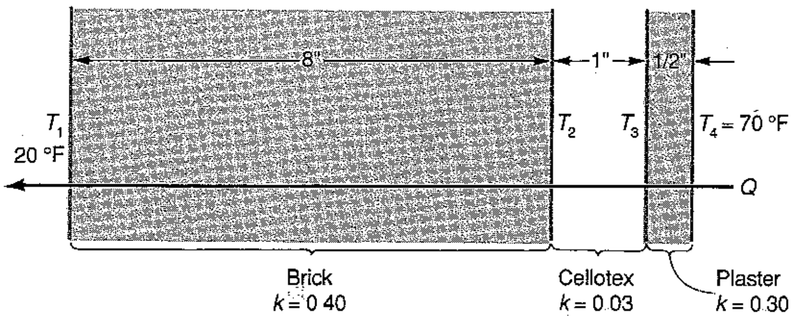

Fig. 11-40

Solution

At steady state the same Q flows across each material, and the temperatures descend in direct proportion to the thermal resistances (reciprocal of conductivity).

$$\text{Resistance of brick} = \frac{x}{k} = \frac{0.667 \text{ ft}}{0.40 \text{ BTU/(ft}^2\text{-°F/ft)}} = 1.67 \text{ ft}^2\text{-°F/BTU}$$

$$\text{Resistance of Cellotex} = \frac{x}{k} = \frac{0.083 \text{ ft}}{0.03} = 2.77 \text{ ft}^2\text{-°F/BTU}$$

$$\text{Resistance of plaster} = \frac{x}{k} = \frac{0.042 \text{ ft}}{0.30} = 0.14 \text{ ft}^2\text{-°F/BTU}$$

Total resistance = 1.67 + 2.77 + 0.14 = 4.58 ft^2-°F/BTU

$$Q = \frac{\Delta T_{\text{total}}}{\text{total resistance}} = \left(\frac{\Delta T}{x/k}\right)_{\text{layer}}$$

Hence,

$$Q = \frac{50}{4.58} = \frac{T_4 - T_3}{0.14} = \frac{T_3 - T_2}{2.77} = \frac{T_2 - T_1}{1.67} = 10.85 \text{ BTU/hr}$$

$T_4 - T_3 = 1.5$ °F, since $T_4 = 70$ °F, $T_3 = 68.5$ °F.

$T_3 - T_2 = 30.3$ °F, since $T_3 = 68.5$ °F, $T_2 = 38.2$ °F.

$T_2 - T_1 = 18.2$ °F, since $T_2 = 38.2$ °F, $T_1 = 20$ °F (in agreement with given data).

The answer is (e).

11-41. In Fig. 11-41, the inner wall is at 80 °F, and the outer wall is exposed to ambient wind and surroundings at 40 °F. The film coefficient, h, for convective heat transfer in a 15-mph wind is about 7 BTU/(hr-ft^2-°F). Ignoring any radiation losses, an overall coefficient (in the same units) for the conduction and convection losses is most nearly

(a) 0.14
(b) 0.80
(c) 1.25
(d) 7.1
(e) 8.2

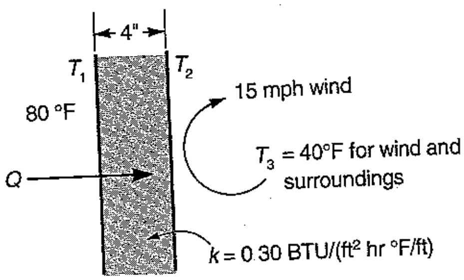

Fig. 11-41

Solution

Since conduction and convection are based on ΔT, absolute temperatures are not required. For steady state, the heat conducted through a wall must equal the heat lost by convection:

$$Q = \frac{kA(T_1 - T_2)}{x} = hA(T_2 - T_3) \tag{1}$$

In a similar way, Q can be expressed by an overall coefficient

$$Q = UA(T_1 - T_3) \tag{2}$$

Here, U is calculated in a manner analogous to that used for thermal conductivities in series:

$$U = \frac{1}{\frac{1}{h_1} + \frac{x_1}{k_1} + \cdots} \tag{3}$$

In this case,

$$U = \frac{1}{\frac{1}{7} + \frac{4/12}{0.30}} = \frac{1}{0.143 + 1.11} = 0.80 \text{ BTU/(hr-ft}^2\text{-°F)}$$

If the question had asked for T_2, Eq. (1) could be used to find it. Similarly, the solution for Q per unit area from Eq. (1) would lead to a value for U from Eq. (2):

$$Q = \frac{0.30}{4/12}(80 - T_2) = 7(T_2 - 40)$$

$Q = 72 - 0.9T_2 = 7T_2 - 280$; thus $T_2 = 352/7.9 = 44.6$ °F, and $Q = 32.2$ BTU/(hr-ft^2)

Now, $Q = UA(T_1 - T_3) = 32.2 = U(1)(80 - 40)$, $U = 32.2/40 = 0.80$ BTU/(hr-ft^2-°F)

Note that this alternate approach does not apply when more than one film coefficient and one conductivity are involved, but information on the interface temperatures is missing. The answer is (b).

Thermodynamics

11-42. Heat is transferred by conduction from left to right through the composite wall shown in Fig. 11-42. Assume the three materials are in good thermal contact and that no significant film coefficients exist at any of the interfaces. The overall coefficient U in BTU/hr-ft²-°F is most nearly

(a) 0.04
(b) 0.13
(c) 0.35
(d) 0.91
(e) 1.92

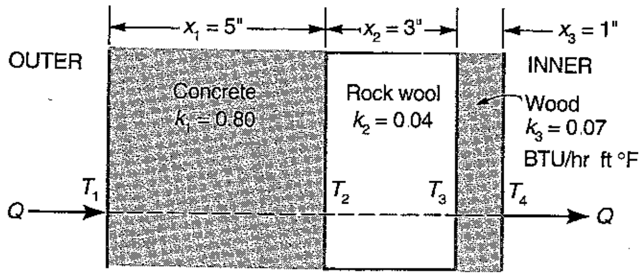

Fig. 11-42

Solution

The overall coefficient U, the thermal conductivity, k/x, and the film coefficient, h, are the reciprocals of their thermal resistances. Thermal resistances in series are handled analogously to series electrical resistances; hence

$$U = \left(\frac{1}{\sum_i \frac{x_i}{k_i}} \right)^{-1}$$

The overall coefficient U is then used in the simplified conduction equation $Q = UA\Delta T$.

In this problem

$$U = \frac{1}{\frac{5/12}{0.80} + \frac{3/12}{0.04} + \frac{1/12}{0.07}} = \frac{1}{0.52 + 6.25 + 1.19} = 0.126 \text{ BTU/(hr-ft}^2\text{-°F)}$$

The answer is (b).

Thermodynamics

11-43. The heat loss per hour through 1 ft² of furnace wall 18 in. thick is 520 BTU. The inside wall temperature is 1900 °F, and its average thermal conductivity is 0.61 BTU/hr-ft-°F.

The outside surface temperature of the wall is nearest to

(a) 100 °F
(b) 300 °F
(c) 600 °F
(d) 1000 °F
(e) 1900 °F

Solution

The heat conduction equation is

$$Q = k\frac{A}{L}(T_1 - T_2)$$

where T_1 = 1900 °F, T_2 = outside temperature, k = 0.61 BTU/hr-ft-°F, Q/A = 520 BTU/hr, and L = 1.5 ft.

Solving for T_2, one has

$$T_2 = -\frac{Q}{A}\frac{L}{k} + T_1 = -520\frac{1.5}{0.61} + 1900 = -1280 + 1900 = 620\ °\mathrm{F}$$

The answer is (c).

11-44. Which of the following is *not* a usual expression for the power/unit-area Stefan-Boltzmann constant for black-body radiation?

(a) 1.36×10^{-12} cal/(sec-cm^2-°K^4)

(b) 5.68×10^{-5} ergs/(sec-cm^2-°K^4)

(c) 5.68×10^{-8} watts/(m^2-°K^4)

(d) 0.171×10^{-8} BTU/(ft^2-hr-°R^4)

(e) 5.68×10^{-8} coulombs/(sec-m^2-°K^4)

Solution

All are numerically correct conversions of the constant in terms of power per unit area. The electrical units of choice (e) are not recognized as appropriate, although they are numerically equivalent to a correct constant of 5.68×10^{-8} coulombs/(sec-m^2-°K^4). The answer is (e).