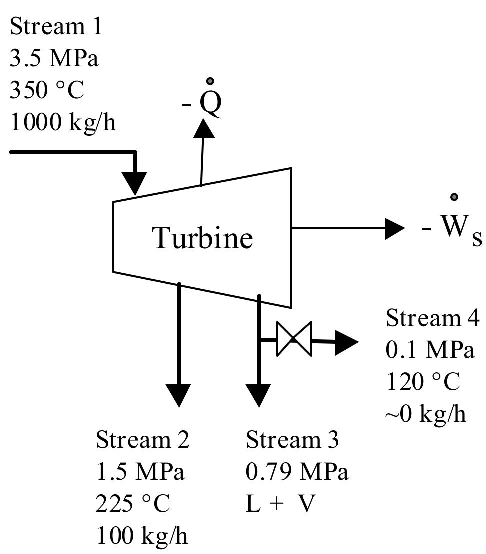
10.213 Chemical Engineering Thermodynamics
Spring 2002

Test 2 Solution

Problem 1 (35 points)

High pressure steam (stream 1) at a rate of 1000 kg/h initially at 3.5 MPa and 350 ºC is expanded in a turbine to obtain work. Two exit streams leave the turbine. Exiting stream 2 is a 1.5 MPa and 225 ºC and flows at 100 kg/h. Exiting stream 3 is at 0.79 MPa and is known to contain a mixture of saturated vapor and liquid. A (negligible) fraction of stream 3 is bled through a throttle value to 0.10 MPa and is found to be 120 ºC (stream 4). The measured output of the turbine is 100 kW.

a) Determine the temperature and quality of stream 3.
Begin by drawing a diagram for the process:



The turbine has two exiting streams (2 and 3), with stream 3 undergoing further processing.

Stream 3 contains both liquid and vapor at 0.79 MPa. From the steam tables, for P = 790 kPa, T^{sat} ~ ***170 ºC*** *(answer), H^l ~718 kJ/kg and H^v ~2767 kJ/kg (pg 702).*

Across the throttle vale, $\Delta H = 0$, thus $H_3 = H_4$.
For P_4 = 100 kPa, T_4 = 120 ºC, H_4 ~2716 kJ/kg (pg 694)

As Stream 3 = L + V, $H_3 = (1-x)H^l + xH^v$
Plugging in $H_3 = H_4$ and other values gives
2716 kJ/kg = (1-x)(718 kJ/kg) + (x)(2767 kJ/kg)
of x = ***0.975*** *(answer)*

b) Determine the rate of heat transfer into or out of the turbine during its operation.

The problem gives that Ws = -100 kW; we need to fine Q. Take turbine as system.
The first law for open systems is $m\Delta H = Q + Ws$. Here, we need to rewrite $m\Delta H$ in terms of the enthalpic changes of output minus input, or $(m_2H_2 + m_3H_3 - m_1H_1)$. By mass balance, $m_1 = m_2 + m_3$, or m_3 = 900 kg/h.
From the steam tables, at P_1 = 3500 kPa and T_1 = 350 ºC, H_1 = 3106.5 kJ/kg (page 711)
From the steam tables, at P_2 = 1500 kPa and T_2 = 225 ºC, H_2 = 2861.5 kJ/kg (page 706).
Thus, $(m_2H_2 + m_3H_3 - m_1H_1) = Q + Ws$ or
(100 kg/h)(2861.5 kJ/kg) + (900 kg/h)(2716 kJ/kg) – (1000 kg/h)(3106.5 kJ/kg) = Q + -100 kW
Q = [286150 + 2444400 - 3106500] kJ/h [(1 h)/(3600 s)] – (-100 kW)
= -(375950)/(3600) kW – 100 kW = -104.4 kW + 100 kW = -4.4 kW or ***4.4 kW or 4.4 kJ/s or 15950 kJ/hr of heat lost from turbine to surroundings*** *(answer)*

Problem 2 (30 points; 4 points for each except 6 points for e)

The following questions use the attached P-H diagram for CO_2.

a) Determine the critical temperature and pressure for CO_2.
From the diagram, the maximum temp and pressure for the L-V dome is ***~88 ºC*** *and* ***1100 psia***.

b) Determine the temperature and pressure of CO_2 at its triple point.
The triple point denotes the temp and pressure of S-L-V coexistence. From the chart, ***P ~ 75 psia*** *and* ***T is less than –40 ºF*** *and probably* ***close to –80 ºF***.

c) Estimate the residual enthalpy for CO_2 at 1000 psia and 180 ºF *using the provided P-H diagram*. Generalized correlations should not be used.
$H^R = H^{actual} - H^{ideal}$. *Consider ideal is a low pressure gas at T of interest, Thus,* H^{actual} = $H(P = 1000 \text{ psia and } T = 180\ ºF) = 170$ *BTU/lb. Next,* $H^{ideal} = H(P = \text{low and } T = 180\ ºF)$ $= 190$ *BTU/lb. Thus* $\mathbf{H^R} = H^{actual} - H^{ideal} = (170 - 190)$ *BTU/lb* **= -20 BTU/lb** *(answer).*

A flow process produces CO_2 as 75 mol % liquid CO_2 and the rest vapor at 60 ºF for use in fire extinguishers. In this process, CO_2 at 20 psia and 60 ºF is compressed in two steps: first to 100 psia and then to its final pressure. The gas is cooled to 60 ºF before entering the second compressor. The compressors both operate adiabatically and reversibly.

d) Draw a scheme for this process in your blue book and note its path on the included P-H diagram for CO_2. Number the various streams using the same numbering.
The first steps are isentropic compression to 100 psia (1→2), isobaric cooling at 100 psia to 60 ºF (2→3), and isentropic compression from 100 psia and 60 ºF to P_{final} *(3→4). Note that this final step is isentropic with a change in pressure and cannot be done isothermally. Thus, the output is at a temperature above 60 ºF and requires isobaric cooling to get the final condition. Thus, the process is:* ***isentropic compression to 100 psia (1→2), isobaric cooling at 100 psia to 60 ºF (2→3), and isentropic compression from 100 psia and 60 ºF to*** $\mathbf{P_{final}}$ ***(3→4), and isobaric cooling at*** $\mathbf{P_{final}}$ ***to 60 ºF (4→5)***. *See P-H diagram.*

e) Estimate the amount of work required in the process and the required cooling.
The final pressure is that pressure where there is liquid/vapor coexistence at 60 ºF. From diagram, P_{final} *is ~800 psia.*
The final state will be a position within the L-V dome that contains 75 mole % liquid (i.e., x = 0.25) which is ¼ the distance from H^l *to* H^v *at* P_{final} *(or H ~83 BTU/lb*$_m$*).*
The ***required work*** $= (H_2\text{-}H_1) + (H_4\text{-}H_3) = (\sim210\text{-}165)+(\sim215\text{-}163)\text{BTU/lb}_m =$ $\mathbf{\sim97\ BTU/lb_m}$
The ***required cooling*** $= (H_3\text{-}H_2) + (H_5\text{-}H_4) = (163\text{-}210)+(83\text{-}215)\text{BTU/lb}_m =$ $\mathbf{-179\ BTU/lb_m}$

f) If the two compressors had efficiencies less than 1, would the i) amount of required work and ii) the amount of required cooling increase, decrease or stay the same as in e)?
An inefficient compressor ($\Delta S > 0$*) would cause the compressed gas output to exit at a higher temperature than for the ideal case. Thus,* ***more work*** *(*$\dot{m}\Delta H$*)* ***would be required***. *As the output to be cooled is hotter, the amount of* ***cooling would be greater*** *than for the ideal case.*

g) If a liquid CO_2 fire extinguisher (75 mol % liquid CO_2 and the rest vapor) stored at 60 ºF is discharged at atmospheric pressure (14.7 psia), what phases are generated and what is the dominant phase?
The exhausted CO_2 *would undergo an isenthalpic expansion as through a throttle value. Thus,* $H^{exit} = H^{inside} = \sim83$ *BTU/lb*$_m$*. At P = 14.7 psia and* H^{exit}*=83 BTU/lb*$_m$*, the output is in a two-phase region, bordered by* ***solid*** *on the left* ***and vapor*** *on the right. The value of* H^{exit} *is closer to the H value for vapor than for solid at this pressure, thus* ***vapor is the dominant phase***.

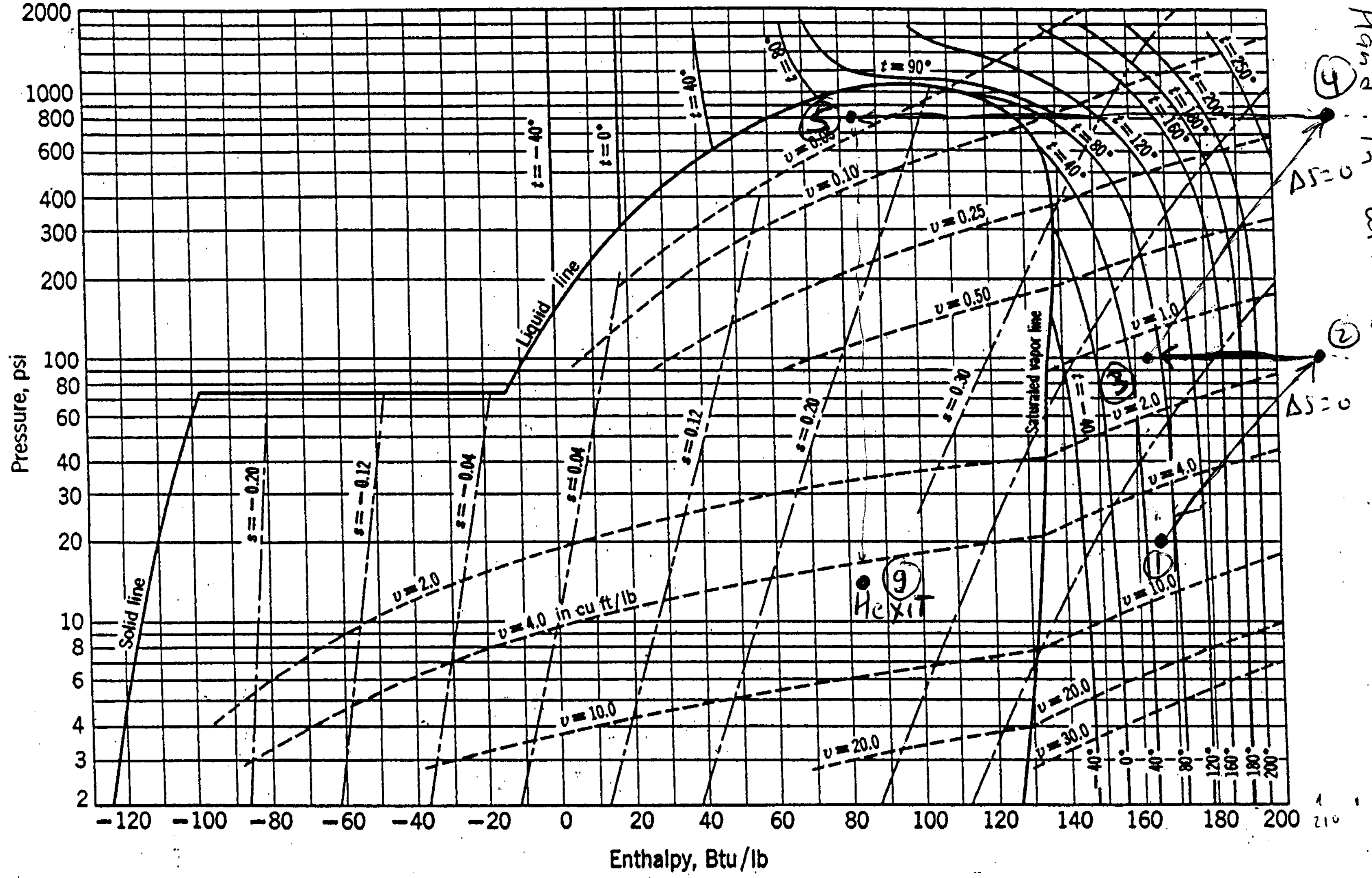
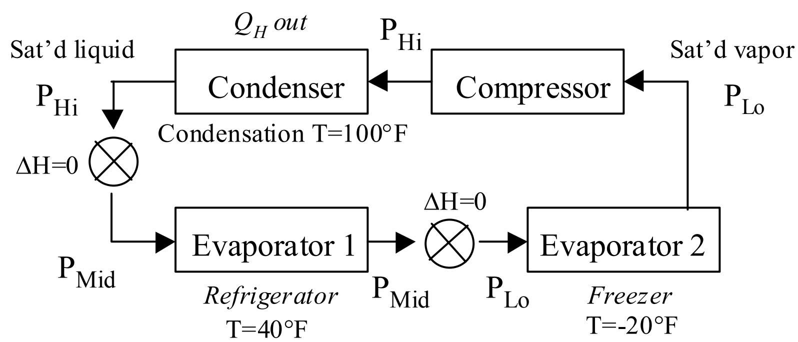


Fig. 6. Pressure–enthalpy diagram for carbon dioxide. Reprinted from the Design Volume of the *Air Conditioning–Refrigerating Data Book*, courtesy of the American Society of Refrigerating Engineers. Prepared by *Refrig. Eng.* from chart and investigations of Plank and Kuprianoff. Datum: $h = 0$, $s = 0$ for saturated liquid at −40° F.

Problem 3 (35 points)
A household refrigeration unit using tetrafluoroethane as refrigerant includes two compartments: the freezer section (colder) and the refrigerator section (less cold). To generate regions of two temperatures, the unit incorporates two throttle valves. Consider a refrigeration unit where the freezer temperature is –20 °F, the refrigerator temperature is 40 °F, and the condensation temperature for exchanging heat into the room is 100 °F. The compressor operates with an efficiency of 0.5.

a) In your blue book, draw the process and number all streams.
We begin my considering a standard refrigeration cycle, where changes are incorporated that would allow two temperatures for cooling (-20 °F and 40 °F). These temperatures define the need for two evaporators in the cycle, one that operates at –20 °F and one that operates at 40 °F. As each evaporator contains the refrigerant coexisting as both liquid and vapor, we can define the operating pressures for the evaporators as P_{Lo} for the colder evaporator and P_{Mid} for the less cold evaporator. P_{Hi} will be the pressure of the condenser. (In making these assignments, we know that boiling temperatures increase with increasing pressure.) Thus, we must employ a throttle valve to take the stream from P_{Hi} to P_{Mid} and another throttle valve from P_{Mid} to P_{Lo}. A compressor is used to return the P_{Lo} stream to P_{Hi} upon its exit from the evaporator operating at P_{Lo}. Assembly of the pieces gives the cycle pictured on the right.



b) On the included P-H diagram, draw the process assuming that two-third of the heat is absorbed in the freezer section and the remainder in the refrigerator section. Number the various streams using the same numbering as in a).
*First, define the pressures from the diagram so that L+V coexist at –20 °F (**P_{Lo} = ~13 psia**), 40 °F (**P_{Mid} = ~50 psia**), and 100 °F (**P_{Hi} = ~140 psia**).*
The total cooling that can be provided is the difference in enthalpies for sat'd vapor at P_{Lo} (H^v at 13 psia = 100 BTU/lb$_m$) and for sat'd liquid at P_{Hi} (H^l at 140 psia = 45 BTU/lb$_m$).
*As 1/3 of the cooling is done in the refrigerator, **H exiting first evaporator** at P = 50 psia will be 45 + 1/3(100-55) **= 63 BTU/lb$_m$** ($\Delta H_{refrigerator}$ = 18 BTU/lb$_m$). The other 2/3 of the cooling for the freezer is at P = 13 psia ($\Delta H_{freezer}$ = 100-63 = ~36 BTU/lb$_m$).*
*For the compressor, if ideal, start from sat'd vapor at P = 13 psia (H = 100 BTU/lbm and S = 0.23 BTU/lb$_m$-°R) and compress isentropically to P_{Hi} = 140 psia where H = 123 BTU/lb$_m$ from the graph. Thus, $(\Delta H)_S$ = 23 BTU/lb$_m$. As ηcompressor = 0.5 from the problem, the actual ΔH will equal (23 BTU/lbm)/(0.5) = 46 BTU/lbm to put the **exit of the compressor** at **P_{Hi} = 130 psia** and **H** = 100 + 46 **= 146 BTU/lb$_m$**. This position locates the highest temperature in the cycle.*

c) What are the highest and lowest temperatures in the process?
*The **lowest temperature** is in the evaporator (#2) in the freezer = **-20 °F**.*
The **highest temperature** is that exiting the compressor = **220 °F** (see diagram for value).

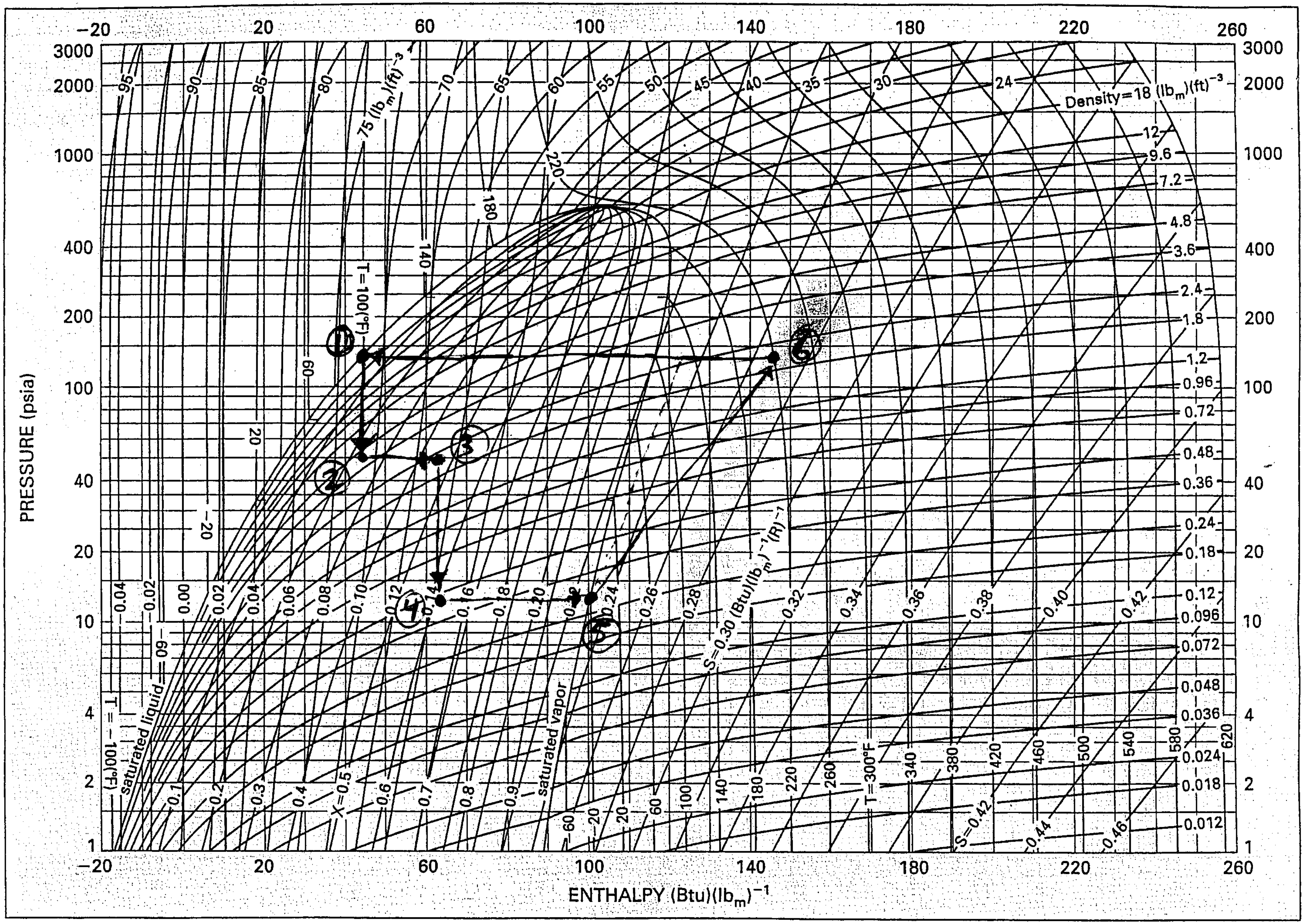


Figure G.2: *P H* diagram for tetrafluoroethane (HFC-134a). (*Reproduced by permission. ASHRAE Handbook: Fundamentals*, p. 17.28, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1993.)