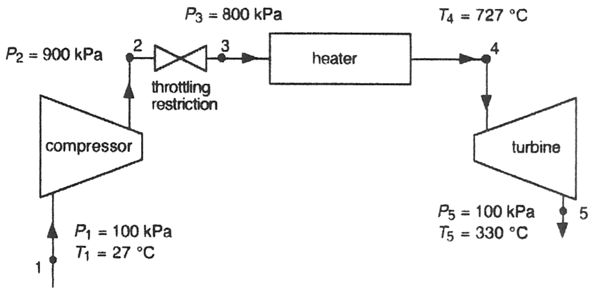
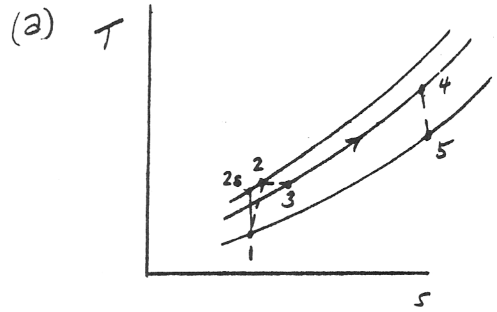




1. (60 points)

The gas turbine system shown schematically above functions like a steady-flow standard Brayton cycle except that a damaged flow conduit between the compressor and heater results in a throttling restriction (with no heat or work interaction) that causes the pressure to drop from 2 to 3 by 100 kPa. Information about the states during actual system operation is shown in the diagram above. The air-flow rate is 8.5 kg/s. The compressor efficiency is 0.82. In analyzing this system, assume the compressor and turbine are adiabatic, and treat the working fluid, air, as a ideal gas with constant specific heats: $c_p = 1.005$ kJ/kgK,
$c_v = 0.718$ kJ/kgK.

(a) Plot the state points and processes on a *T-s* diagram, showing appropriate constant pressure lines.

(b) Determine the power input to the compressor and the actual exit temperature from the compressor, T_2.

(c) Determine T_3.

(d) Determine the rate of heat input to the heater.

(e) Determine the net power output of the system and the cycle efficiency.

(b) $1 \to 2s$ isentropic $\Rightarrow T_{2s} = T_1 \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$ $\qquad k = c_p/c_v = 1.005/.718 = 1.400$

$(k-1)/k = .400/1.400 = 0.286$

$T_{2s} = (27+273)\left(\frac{900}{100}\right)^{.286} = 562\,K$

$\dot{W}_{c,s} = +\dot{m}c_p(T_{2s} - T_1) = +(8.5)(1.005)(562-300) = +2238$ kW

$\dot{W}_c = \dot{W}_{c,s}/\eta_c = -2238/.82 = +2729$ kW

$\dot{W}_c = -\dot{m}c_p(T_2 - T_1) \rightarrow T_2 = \frac{-\dot{W}_c}{\dot{m}c_p} + T_1 = \frac{2729}{8.5(1.005)} + 27 = 347\,°C$

$T_2 = 347 + 273 = 620\,K$

(c) CV around throttling restriction $\Rightarrow \cancel{\dot{Q}}^{\,0} + \dot{m}h_2 = \dot{m}h_3 \Rightarrow h_3 = h_2$

Since ideal gas $\Rightarrow h = h(T) \Rightarrow T_3 = T_2 = 347\,°C = 620\,K$

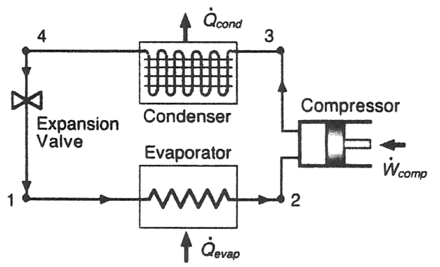
(d) CV around heater $\Rightarrow \dot{Q}_{in} + \dot{m}h_3 = \dot{m}h_4 \Rightarrow \dot{Q}_{in} = \dot{m}(h_4 - h_3)$

$\Rightarrow \dot{Q}_{in} = \dot{m}c_p(T_4 - T_3) = (8.5)(1.005)(727-347) = 3246$ kW

(e) $\dot{W}_t = \dot{m}c_p(T_4 - T_5) = 8.5(1.005)(727-330) = 3391$ kW

$\dot{W}_{net} = \dot{W}_t + \dot{W}_c = 3391 - 2729 = 662$ kW

cycle efficiency $= \eta_{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{662}{3246} = 0.204 = 20.4\,\%$



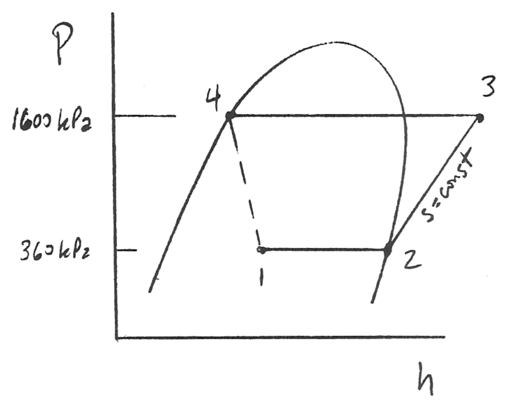
2. (40 points)
The vapor compression refrigeration system shown schematically above uses R-134a as its working fluid. The mass flow rate of R-134a in the system is 0.031 kg/s. The pressure at states 3 and 4 is 1600 kPa. State 4 is saturated liquid. The pressure at states 1 and 2 is 360 kPa. State 2 is saturated vapor. The following R-134a data is available for your analysis:

The specific entropy of saturated vapor at 360 kPa is 0.92836 kJ/kgK.
For superheated vapor at P = 1600 kPa and s = 0.92836 kJ/kgK, the specific enthalpy is 284.76 kJ/kg.

In this cycle, because the piping is not insulated, the process from 4 to 1 is *not* adiabatic. From 4 to 1, heat is input at a rate of 0.8 kW. In your analysis, assume the compressor is reversible and adiabatic.

(a) Show the cycle on a *P-h* diagram.
(b) At both 360 kPa and 1600 kPa determine the saturated liquid and vapor specific enthalpies (h_f and h_g)
(c) Determine the rate of heat absorption in the evaporator.
(d) Determine the cycle COP.

(a)



(b)
from saturation table
@ 360 kPa
$h_f = 59.72$ kJ/kg
$h_g = h_2 = 253.81$ kJ/kg
@ 1600 kPa
$h_f = h_4 = 135.93$ kJ/kg
$h_g = 277.86$ kJ/kg

(c) CV around expansion valve $\rightarrow$ $\dot{Q} + \dot{m} h_4 = \dot{m} h_1$

$\Rightarrow h_1 = h_4 + \dot{Q}/\dot{m} = 135.93 + \frac{0.8}{.031} = 161.73$ kJ/kg

CV around evaporator $\rightarrow$ $\dot{Q}_{evap} + \dot{m} h_1 = \dot{m} h_2$

$\Rightarrow \dot{Q}_{evap} = \dot{m}(h_2 - h_1) = .031(253.81 - 161.73) = \underline{2.85 \text{ kW}}$

(d) CV around compressor $\rightarrow$ rev & adiab $\Rightarrow s_3 = s_2$

$-\dot{W}_c = \dot{m}(h_3 - h_2) \Rightarrow \dot{W}_c = -.031(284.76 - 253.81)$

$\dot{W}_c = -0.959$ kW

$COP = \dot{Q}_{evap} / |\dot{W}_c| = 2.85 / .959 = \underline{2.97}$