Review of materials covered in ENGR 251- THERMO I

Unless specified, the final exam includes everything you have learned and covered in class

EXCEPT

Unsteady control volume energy analysis
Isentropic efficiency (last lecture – Monday 26th, 2018)

Properties of pure substances

Look at the midterm review slide and your midterm exam

http://users.encs.concordia.ca/~hoing/Teaching/ENGR251/midterm_review.PDF

Highlights:

Phase change processes and basic definition (quality, enthalpy, different phases, etc.)

Property (phase-change) diagram

How to use thermodynamic tables

- Saturation table, superheated table and compressed liquid table (if available)
- determine the system state after a thermodynamic process

$$v = v_f + x(v_g - v_f)$$

$$u = u_f + x(u_g - u_f) = u_f + x u_{fg}$$

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

Application of ideal gas equation of state

$$PV = mRT$$

Fro a process from (1) to (2), we can relate the ideal gas equation at two states for a fixed mass

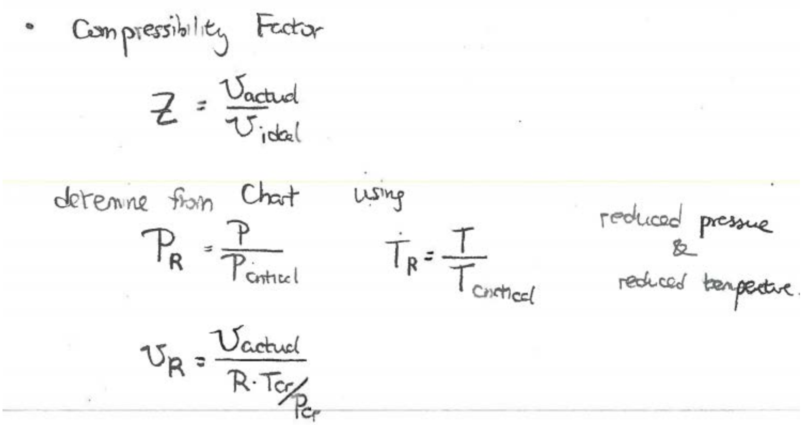
$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2}$$

Example 7

Compressibility factor and the chart

$$Z = \frac{P \cdot v}{R \cdot T} \qquad Z = \frac{V_{\text{actual}}}{V_{\text{ideal}}}$$

$$P_R = \frac{P}{P_{\text{cr}}} \qquad T_R = \frac{T}{T_{\text{cr}}} \qquad V_R = \frac{V_{\text{actual}}}{RT_{\text{cr}}/P_{\text{cr}}}$$



- Compressibility Factor

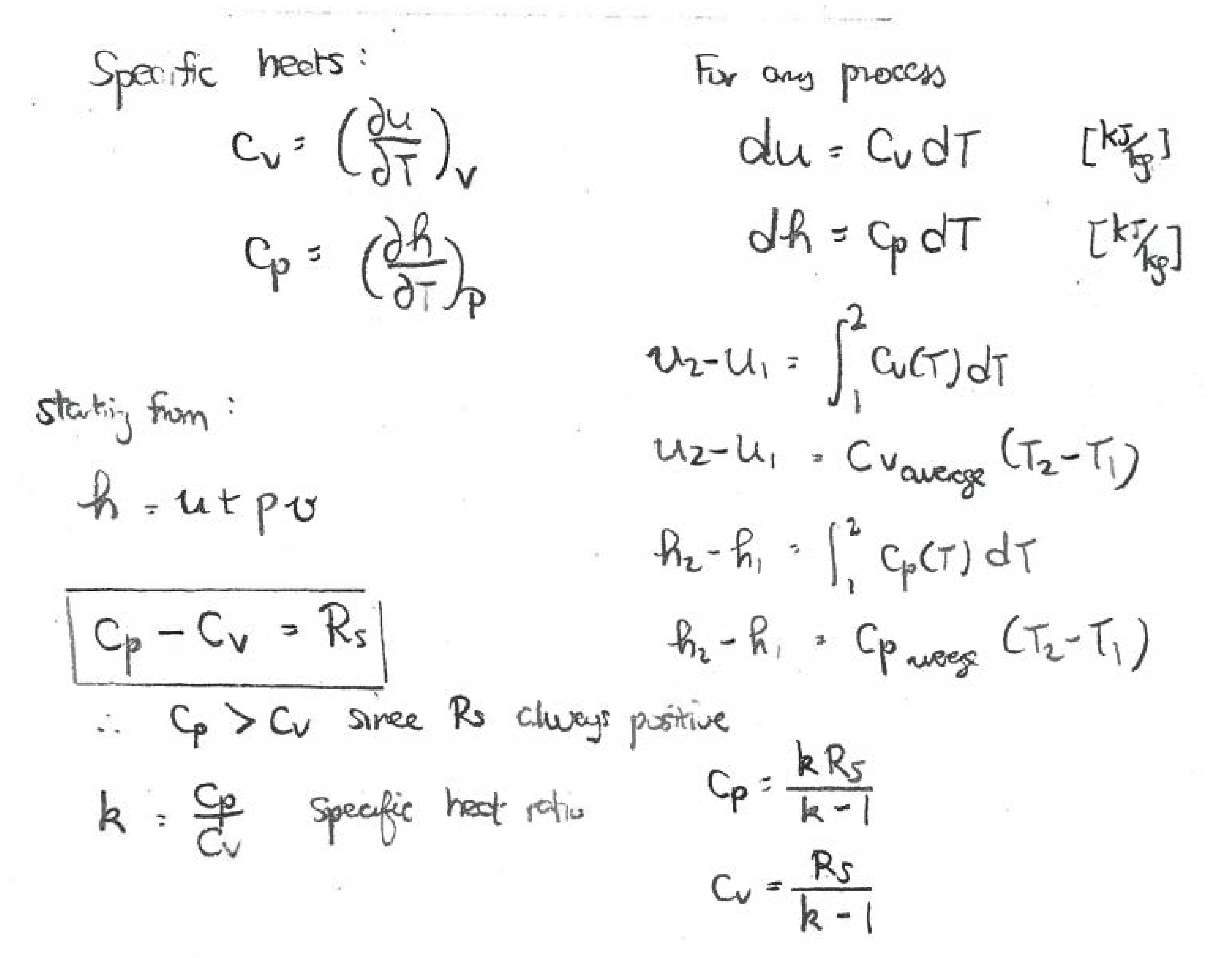
$$Z = \frac{v_{actual}}{v_{ideal}}$$

determine from Chart using reduced pressure & reduced temperature

$$P_R = \frac{P}{P_{critical}} \qquad T_R = \frac{T}{T_{critical}}$$

$$v_R = \frac{v_{actual}}{R \cdot T_{cr}/P_{cr}}$$

For an ideal gas: $PV = mRT$



Specific heats:

$$C_v = \left(\frac{\partial u}{\partial T}\right)_v$$

$$C_p = \left(\frac{\partial h}{\partial T}\right)_p$$

starting from:

$h = u + pv$

$$\boxed{C_p - C_v = R_s}$$

$\therefore$ $C_p > C_v$ since R_s always positive

$k = \frac{C_p}{C_v}$ specific heat ratio

For any process

$du = C_v \, dT$ $[kJ/kg]$

$dh = C_p \, dT$ $[kJ/kg]$

$$u_2 - u_1 = \int_1^2 C_v(T) \, dT$$

$$u_2 - u_1 = C_{v_{\text{average}}}(T_2 - T_1)$$

$$h_2 - h_1 = \int_1^2 C_p(T) \, dT$$

$$h_2 - h_1 = C_{p_{\text{average}}}(T_2 - T_1)$$

$$C_p = \frac{kR_s}{k-1}$$

$$C_v = \frac{R_s}{k-1}$$

1st law analysis for a closed system

Sign Convention!

Heat and Work (path dependent)

heat <u>to</u> the system
work <u>delivered by</u> the system } positive

heat <u>from</u> the system (heat loss)
work <u>to</u> the system } negative.

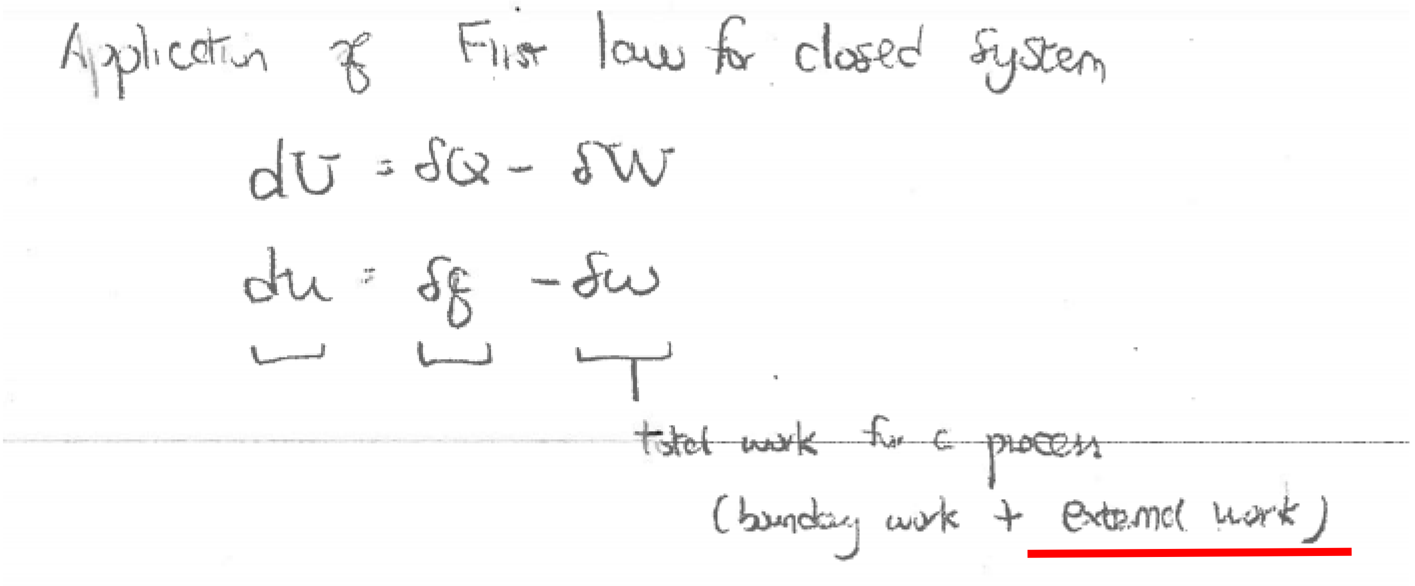
Application of First law for closed system

$$dU = \delta Q - \delta W$$

$$\underbrace{du} = \underbrace{\delta q} - \underbrace{\delta w}$$

total work for a process
(boundary work + external work)

1st law analysis for a closed system



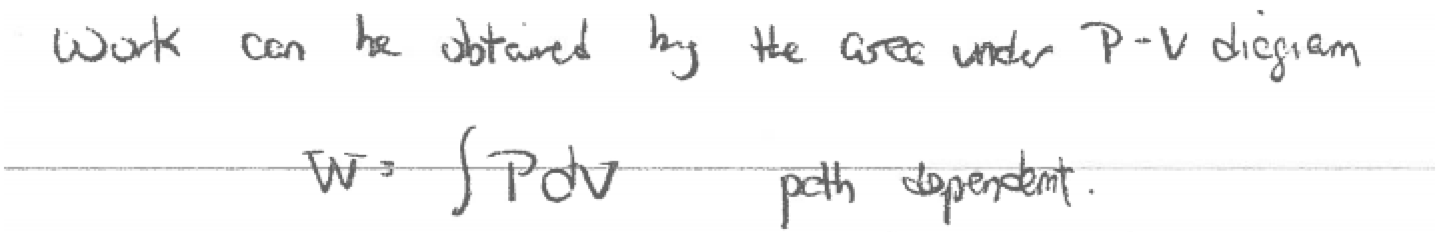
Application of First law for closed system

$$dU = \delta Q - \delta W$$

$$\underbrace{du}_{} = \underbrace{\delta q}_{} - \underbrace{\delta w}_{\text{total work for a process}}$$

(boundary work + external work)

The most common work in thermodynamics (boundary work)



Work can be obtained by the area under P-V diagram

$W = \int P dV$ path dependent.

1st law analysis for a closed system

- Constant volume

$$W = 0$$

- constant pressure

$$W = P\int_1^2 dV = P(V_2 - V_1)$$

compression −ve

expansion +ve

- isothermal :

$$W = \int_1^2 \frac{mR_sT}{V}\, dV = mR_sT(\ln V_2 - \ln V_1)$$

$$= P_1V_1 \ln\left(\frac{V_2}{V_1}\right)$$

- polytropic : $PV^n = \text{constant}$

$$P_1V_1^n = P_2V_2^n = \text{constant.}$$

$$W = \int_1^2 \left(\frac{c}{V^n}\right) dV = \left(\frac{P_2V_2 - P_1V_1}{1-n}\right) \qquad n \neq 1$$

- Spring loaded :

$$W = \text{Area trapezoid} = \frac{P_1 + P_2}{2}(V_2 - V_1) \qquad \frac{P_1 + P_2}{2}\, m(v_2 - v_1)$$

1st law analysis for a closed system

polytropic process

a path that links the two states :

$$P_1 V_1^n = P_2 V_2^n$$

$$\frac{P_1}{P_2} = \left(\frac{V_2}{V_1}\right)^n$$

using $PV = mR_sT$:

$$\boxed{\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n}$$

$$\frac{\frac{mR_sT_1}{V_1}}{\frac{mR_sT_2}{V_2}} = \left(\frac{V_2}{V_1}\right)^n \rightarrow \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^n$$

$$\boxed{\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1}}$$

$$\frac{P_1}{P_2} = \left(\frac{\frac{mR_sT_2}{P_2}}{\frac{mR_sT_1}{P_1}}\right)^n \rightarrow \boxed{\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}}$$

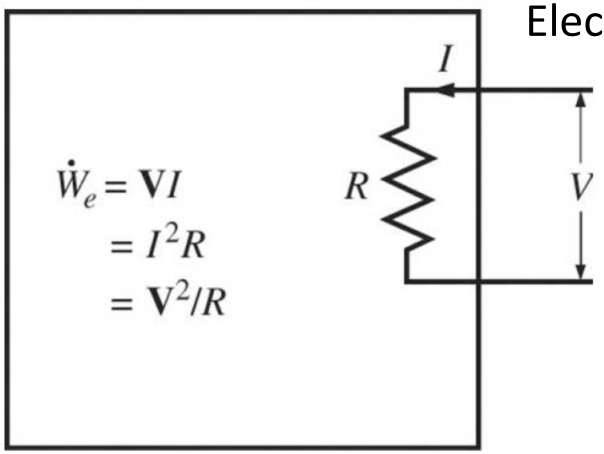
1st law analysis for a closed system

$$dU = \delta Q - \delta W$$

$$du = \delta q - \delta w$$

- Other work
  - Electric work, shaft work (paddle), spring work, etc.

Electrical power



$$\dot{W}_e = \mathbf{V}I = I^2R = \mathbf{V}^2/R$$

In the exam, be careful with the units!!!

1st law analysis for open systems

$$\boxed{\frac{dM_{CV}}{dt} = \sum_i \rho AV - \sum_e \rho AV}$$

$$\boxed{\frac{dM_{CV}}{dt} = \dot{m}_i - \dot{m}_e}$$

$$\frac{dE_{CV}(t)}{dt} = \dot{Q} - \dot{W}_{shaft} + \sum_i \dot{m}_i(h_i + V_i^2/2 + gZ_i) - \sum_e \dot{m}_e(h_e + V_e^2/2 + gZ_e)$$

1^{st} law analysis for open systems

Main components of the above cycle are:

- Boiler (steam generator) – heat exchanger
- Turbine – generates work
- Condenser – heat exchanger
- Pump
- Others components include:
  - nozzles, diffusers, throttling devices
- **Mixing chamber**

Integration of open system components

Thermodynamic cycles:

- Rankine cycle
- Brayton cycle
- Otto cycle

Combination of few open-system components (e.g. compressor + heat exchanger, turbine with a nozzle, etc.)

2nd law of thermodynamics

The Second Law of Thermodynamics

Clausius Statement

- It is *impossible* for a system to operate in such a way that the sole result is the transfer of heat from a cold to a hot body

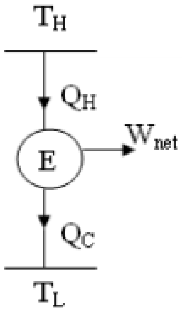
Equivalence

Kelvin Planck Statement

- It is *impossible* for a system that operates in a cycle to generate work while transferring heat with a single **reservoir**

$$\eta = \frac{|W_{net}|}{|Q_{in}|} = \frac{|Q_{in}| - |Q_{out}|}{|Q_{in}|} = 1 - \frac{|Q_{out}|}{|Q_{in}|} = 1 - \frac{|Q_C|}{|Q_H|}$$

Engine

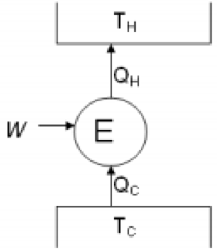


$$COP = \frac{\text{Desired output}}{\text{required input}} = \frac{\text{heat removed}}{\text{work done}} = \frac{|Q_C|}{|W_{refr}|} = \frac{|Q_C|}{|Q_H| - |Q_C|}$$

Refrigeration

$$COP = \frac{\text{Desired output}}{\text{required input}} = \frac{\text{heat supplied}}{\text{work done}} = \frac{|Q_H|}{|W|} = \frac{|Q_H|}{|Q_H| - |Q_C|} = \frac{1}{1 - \frac{|Q_C|}{|Q_H|}}$$

Heat Pumps



Concordia

2nd law of thermodynamics

Irreversibilities include:

- dry and fluid friction
- heat transfer through a *finite* temperature difference
- rapid compression and expansion of a fluid
- unrestrained expansion of a fluid
- rapid mixing of different gases
- etc.

* All ***real*** processes include some irreversibility

2nd law of thermodynamics

Carnot cycle is the most efficient cycle consisting solely of ***ideal* reversible** processes.

Carnot cycle consists of the following 4 reversible processes:

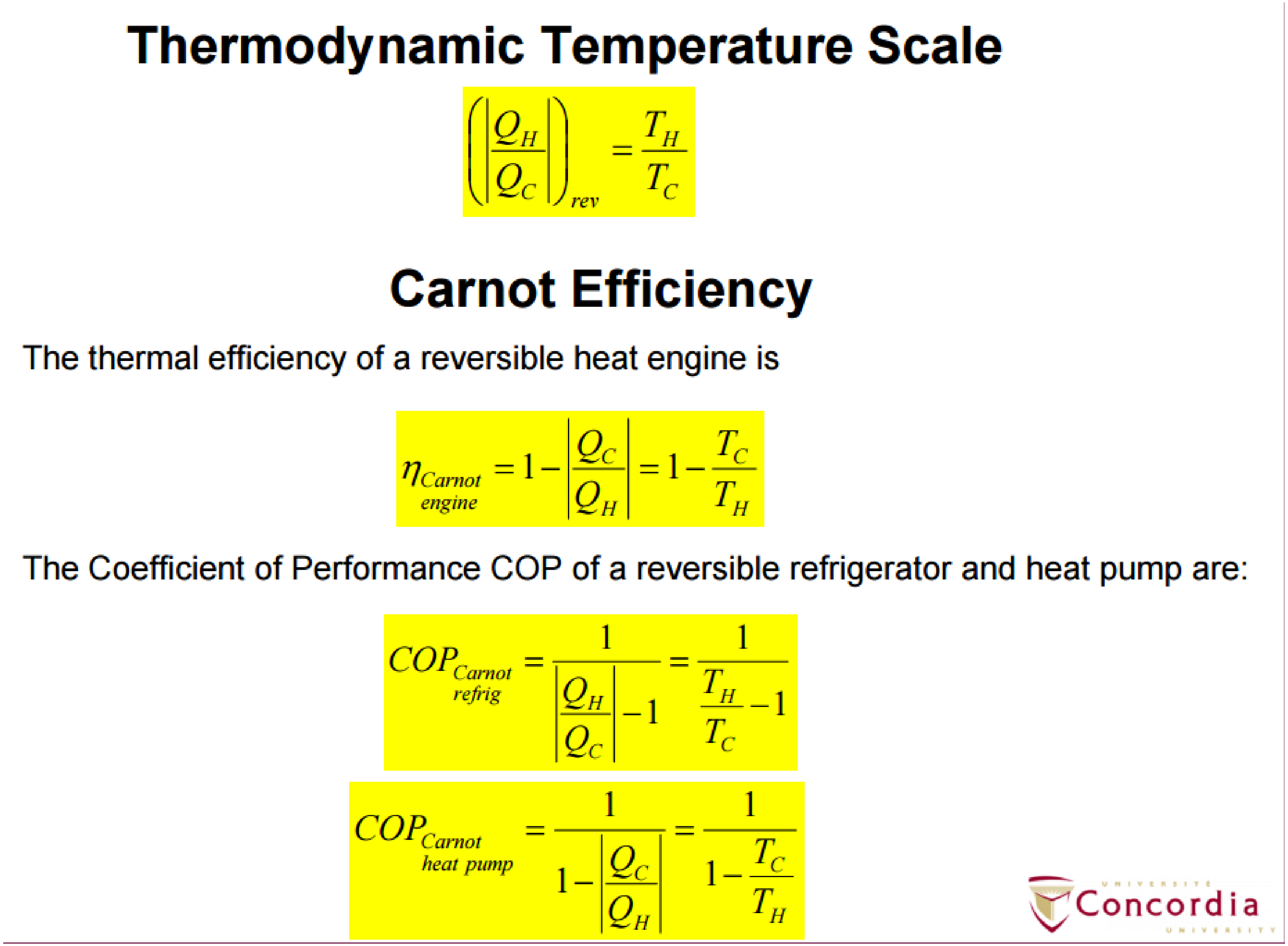
1→2 Adiabatic compression
2→3 Isothermal expansion
3→4 Adiabatic expansion
4→1 Isothermal compression

Carnot Principles

1. The efficiency of an **irreversible** heat engine is always less than the efficiency of a **reversible** one operating between the same two reservoirs.

2. The efficiencies of all reversible heat engines operating between the same two reservoirs are the same.

2nd law of thermodynamics



Thermodynamic Temperature Scale

$$\left(\left|\frac{Q_H}{Q_C}\right|\right)_{rev} = \frac{T_H}{T_C}$$

Carnot Efficiency

The thermal efficiency of a reversible heat engine is

$$\eta_{\substack{Carnot\\engine}} = 1 - \left|\frac{Q_C}{Q_H}\right| = 1 - \frac{T_C}{T_H}$$

The Coefficient of Performance COP of a reversible refrigerator and heat pump are:

$$COP_{\substack{Carnot\\refrig}} = \frac{1}{\left|\frac{Q_H}{Q_C}\right| - 1} = \frac{1}{\frac{T_H}{T_C} - 1}$$

$$COP_{\substack{Carnot\\heat\ pump}} = \frac{1}{1 - \left|\frac{Q_C}{Q_H}\right|} = \frac{1}{1 - \frac{T_C}{T_H}}$$

2nd law of thermodynamics

For a cyclic device:

$$\oint \frac{\delta Q}{T} \leq 0$$

→ This is the **Clausius Inequality** which is valid for all thermodynamic cycles, reversible or irreversible.

Entropy balance for a cycliс device system

Entropy $\Delta S_{\text{total}} = \Delta S_{\text{system}} + \Delta S_{\text{surr}} \geq 0$

Increase in Entropy Principle $dS_{universe} \geq 0$

Evaluation of Entropy for a system undergoes a thermodynamic process

$$Tds = du + Pdv$$

$$Tds = dh - vdP$$

Assuming constant specific heats:

$$s_2 - s_1 = c_V \int_1^2 \frac{dT}{T} + R\ln\left(\frac{v_2}{v_1}\right)$$

$$s_2 - s_1 = c_V \ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{v_2}{v_1}\right)$$

1st *T*-ds relationship

$$s_2 - s_1 = c_P \int_1^2 \frac{dT}{T} - R\ln\left(\frac{P_2}{P_1}\right)$$

$$s_2 - s_1 = c_P \ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

2nd *T*-ds relationship

Isentropic relationships

Special class of idealized processes of which dS = 0

- adiabatic

- Reversible

Can derive these relationship

1) using 1st and 2nd law with ds = 0 (isentropic condition)
2) combining the polytropic process equation pv^n =constant with n = k and Ideal gas equation of state EOS

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

1st Isentropic relationship for an ideal gas

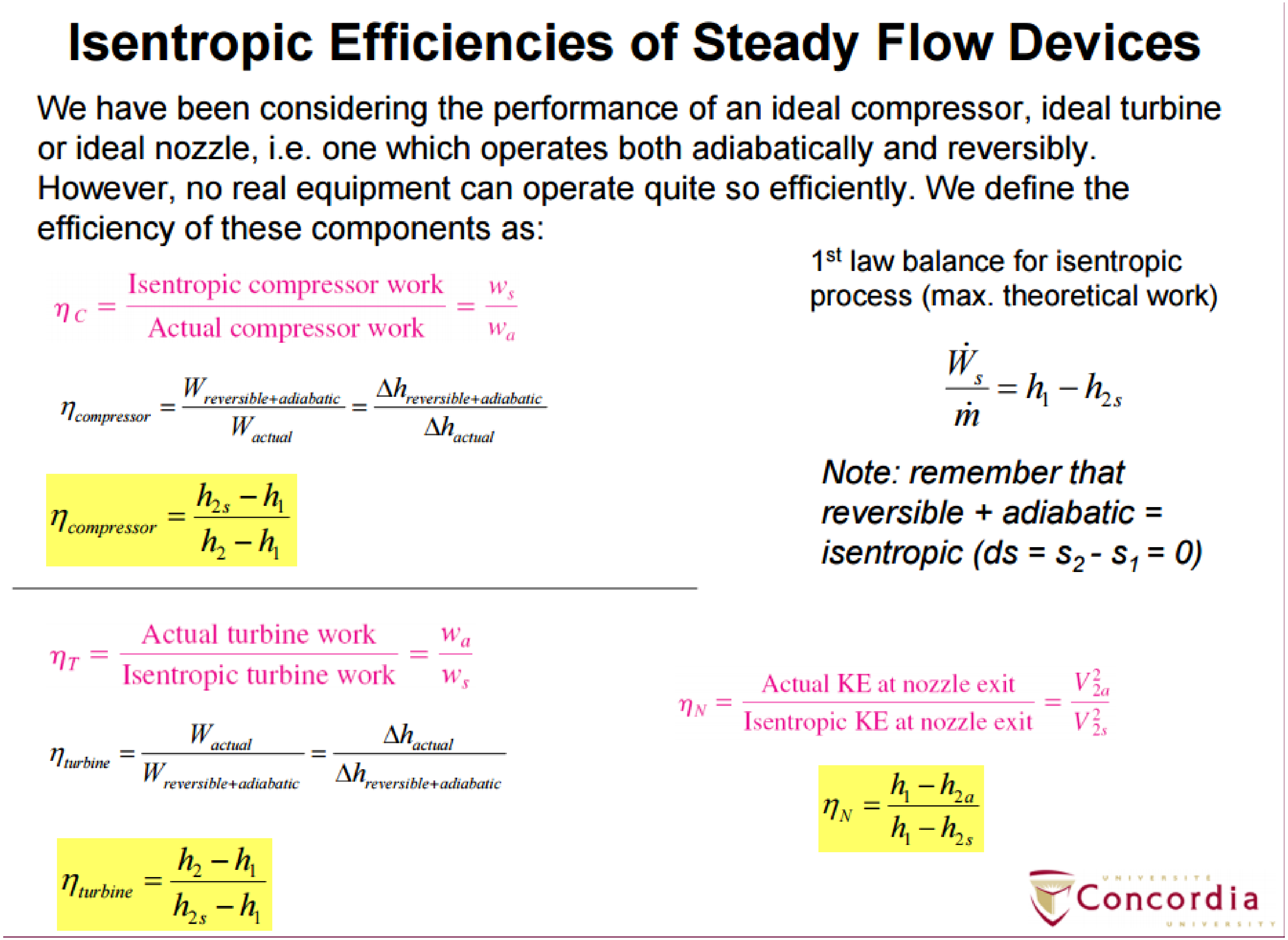
$$\boxed{\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}}$$

2nd Isentropic relationship for an ideal gas

$$\boxed{\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^{k}}$$

3rd Isentropic relationship for an ideal gas

2nd law of thermodynamics



Isentropic Efficiencies of Steady Flow Devices

We have been considering the performance of an ideal compressor, ideal turbine or ideal nozzle, i.e. one which operates both adiabatically and reversibly. However, no real equipment can operate quite so efficiently. We define the efficiency of these components as:

$$\eta_C = \frac{\text{Isentropic compressor work}}{\text{Actual compressor work}} = \frac{w_s}{w_a}$$

$$\eta_{compressor} = \frac{W_{reversible+adiabatic}}{W_{actual}} = \frac{\Delta h_{reversible+adiabatic}}{\Delta h_{actual}}$$

$$\eta_{compressor} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

1st law balance for isentropic process (max. theoretical work)

$$\frac{\dot{W}_s}{\dot{m}} = h_1 - h_{2s}$$

Note: remember that reversible + adiabatic = isentropic (ds = s_2 - s_1 = 0)

$$\eta_T = \frac{\text{Actual turbine work}}{\text{Isentropic turbine work}} = \frac{w_a}{w_s}$$

$$\eta_{turbine} = \frac{W_{actual}}{W_{reversible+adiabatic}} = \frac{\Delta h_{actual}}{\Delta h_{reversible+adiabatic}}$$

$$\eta_{turbine} = \frac{h_2 - h_1}{h_{2s} - h_1}$$

$$\eta_N = \frac{\text{Actual KE at nozzle exit}}{\text{Isentropic KE at nozzle exit}} = \frac{V_{2a}^2}{V_{2s}^2}$$

$$\eta_N = \frac{h_1 - h_{2a}}{h_1 - h_{2s}}$$

UNIVERSITÉ Concordia UNIVERSITY

Rankine cycle

Water is the working fluid in an ideal Rankine cycle. Superheated vapor enters the isentropic (adiabatic and reversible) turbine at 8 MPa, 550°C with a mass flow rate of 7.8 kg/s and exits at 10 kPa. Saturated liquid enters the isentropic pump at 10 kPa. The specific steady flow work input required for the pump is 10.0 kJ/kg. Determine:

a) Specific enthalpy at each state
b) the net power developed, in kW
c) the thermal efficiency of the cycle

$P_1 = 80 \text{ bar} = 8 \text{ MPa}$

$T_1 = 550^\circ C$

$h_1 = 3521.8 \text{ kJ/kg}$

Superheated table

$s_1 = 6.88 \text{ kJ/kg K}$

$P_2 = 10 \text{ kPa}$

Isentropic

$$\boxed{s_{2s} = s_1}$$

$$x_{2s} = \frac{s_{2s} - s_{f2}}{s_{g2} - s_{f2}}$$

Saturated water pressure table at 10 kPa

$$= \frac{6.88 - 0.6492}{7.4996} = 0.831$$

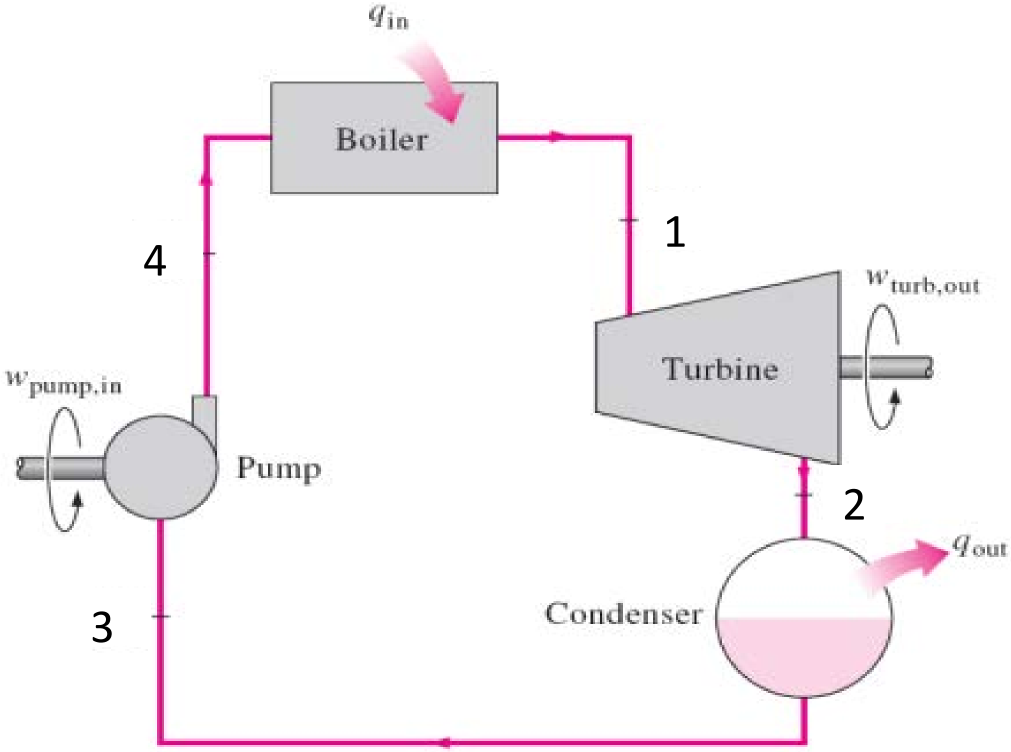
$$h_{2s} = h_f + x h_{fg}$$
$$= 191.81 + 0.831(2392.1)$$
$$= 2179.66 \text{ kJ/kg}$$

Idea Rankine cycle

Rankine cycle

Water is the working fluid in an ideal Rankine cycle. Superheated vapor enters the isentropic (adiabatic and reversible) turbine at 8 MPa, 550°C with a mass flow rate of 7.8 kg/s and exits at 10 kPa. Saturated liquid enters the isentropic pump at 10 kPa. The specific steady flow work input required for the pump is 10.0 kJ/kg. Determine:

a) Specific enthalpy at each state
b) the net power developed, in kW
c) the thermal efficiency of the cycle



Rankine cycle

Water is the working fluid in an ideal Rankine cycle. Superheated vapor enters the isentropic (adiabatic and reversible) turbine at 8 MPa, 550°C with a mass flow rate of 7.8 kg/s and exits at 10 kPa. Saturated liquid enters the isentropic pump at 10 kPa. The specific steady flow work input required for the pump is 10.0 kJ/kg. Determine:

a) Specific enthalpy at each state
b) the net power developed, in kW
c) the thermal efficiency of the cycle

Unless specified -> Ideal Rankine cycle: Constant pressure P2 = P3; P4 = P1 across the condenser/boiler, respectively

$P_3 = 10\,kPa$

$\boxed{\text{sat. liquid}}$

$h_3 = 191.81\ kJ/kg$

$P_4 = 8\,MPa$

h_4'

$h_3 = h_4 + \dot{w}_s$

$h_4 = h_3 - \dot{w}_s$

$= 191.81\ kJ/kg - (-10\ kJ/kg)$

$= 201.81\ kJ/kg$

Pump raises the energy

$$\frac{1}{\dot{m}}\cancel{\frac{dE}{dt}} = \frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_s}{\dot{m}} + (h_{in} - h_{out}) + 1/2(\cancel{V_{in}^2 - V_{out}^2}) + g(\cancel{z_{in} - z_{out}})$$

$$0 = \frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_s}{\dot{m}} + (h_{in} - h_{out})$$

Rankine cycle

Water is the working fluid in an ideal Rankine cycle. Superheated vapor enters the isentropic (adiabatic and reversible) turbine at 8 MPa, 550°C with a mass flow rate of 7.8 kg/s and exits at 10 kPa. Saturated liquid enters the isentropic pump at 10 kPa. The specific steady flow work input required for the pump is 10.0 kJ/kg. Determine:

a) Specific enthalpy at each state
b) the net power developed, in kW
c) the thermal efficiency of the cycle

$$0 = \frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_s}{\dot{m}} + (h_{in} - h_{out})$$

$$\dot{W}_{net} = \dot{m}(\omega_{turbine} + \omega_{pump})$$

$$\dot{W}_{net} = \dot{m}((h1 - h2) + (h3 - h4))$$

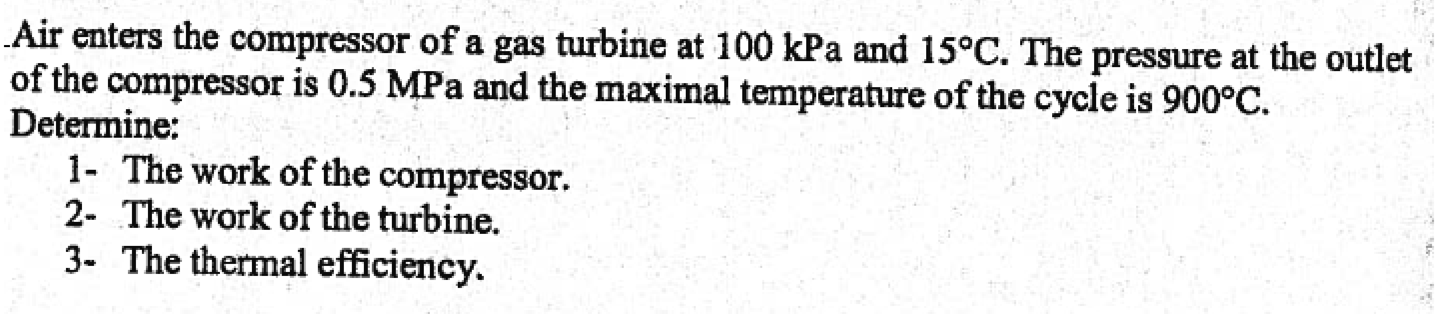
7.8 ((3521.8-2179.65)+(191.81-201.81)) = 10390.8 kW

$$\dot{Q}_{in} = \dot{m}(h_1 - h_4) = 25896 \text{ kW}$$

$$\eta = \frac{\text{net work out}}{\text{heat input}} = \frac{\sum \frac{\dot{W}}{\dot{m}}}{\frac{\dot{Q}_{in}}{\dot{m}}} = \frac{\sum w}{q_{in}}$$

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \underline{\underline{0.401}} = \underline{\underline{40.1}}\ \%$$

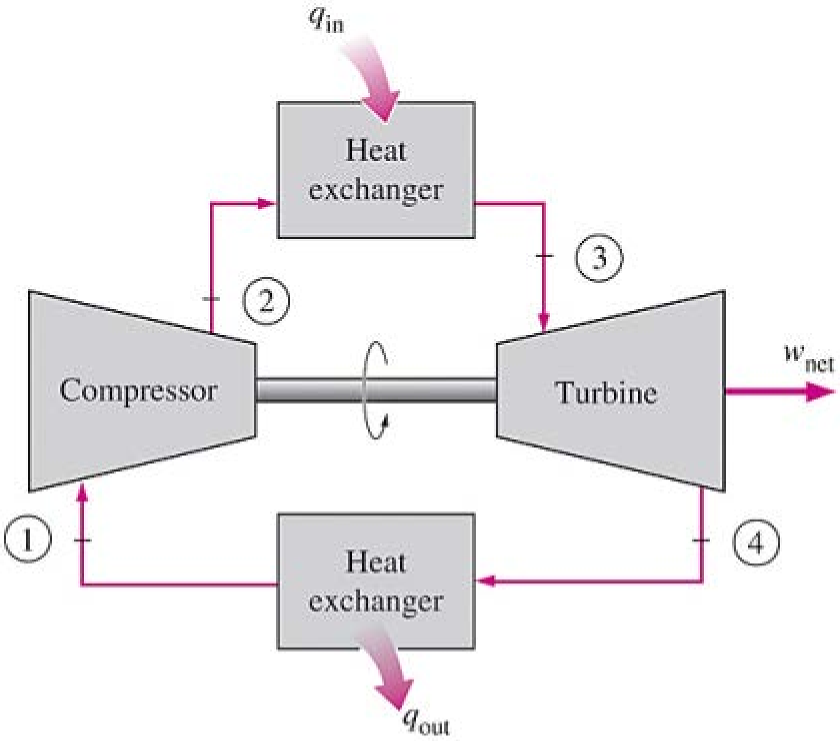
Example Ideal Brayton cycle



Air enters the compressor of a gas turbine at 100 kPa and 15°C. The pressure at the outlet of the compressor is 0.5 MPa and the maximal temperature of the cycle is 900°C.

Determine:

1- The work of the compressor.
2- The work of the turbine.
3- The thermal efficiency.



Example Ideal Brayton cycle

Air enters the compressor of a gas turbine at 100 kPa and 15°C. The pressure at the outlet of the compressor is 0.5 MPa and the maximal temperature of the cycle is 900°C.
Determine:

1- The work of the compressor.
2- The work of the turbine.
3- The thermal efficiency.

Assume both the compressor and turbines are isentropic devices. Cp = 1.0035 kJ/kg-K for air (**Ideal gas**).

$T_1 = 15°C = 288.2K.$

$P_1 = 0.1 MPa$

$P_2 = 0.5 MPa$

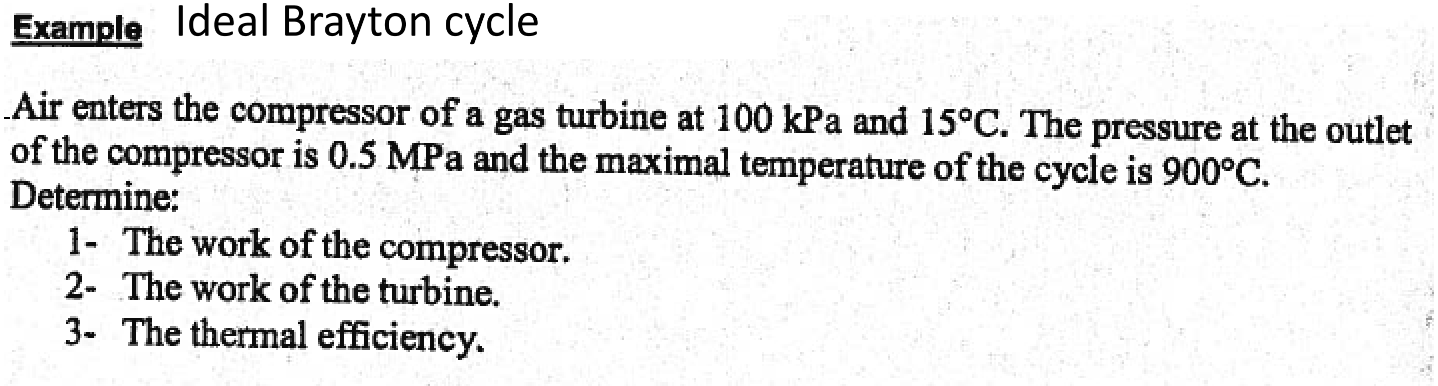
$T_{max} = T_3 = 900°C$
$= 1173.2 K$

Isentropic relationships

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

$$T_2 = 288.2 \left(\frac{0.5}{0.1}\right)^{\frac{k-1}{k}}$$

$$T_2 = 456.6 K$$



Example Ideal Brayton cycle

Air enters the compressor of a gas turbine at 100 kPa and 15°C. The pressure at the outlet of the compressor is 0.5 MPa and the maximal temperature of the cycle is 900°C.

Determine:

1- The work of the compressor.
2- The work of the turbine.
3- The thermal efficiency.

Assume both the compressor and turbines are isentropic devices. Cp = 1.0035 kJ/kg-K for air.

- computation of the work of the compressor:

Hyp: . ΔE_K and ΔE_p are neglected
. steady state

CV : air within comp.

$$0 = \frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_s}{\dot{m}} + (h_{in} - h_{out})$$

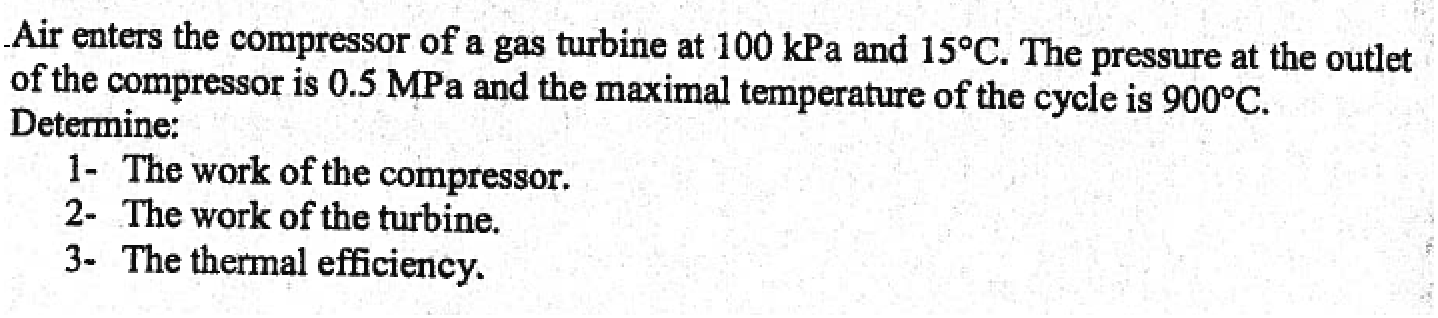
$\dot{m}_1 = \dot{m}_2 = \dot{m}$

$w_c = h_1 - h_2$

For an ideal gas

$= c_p (T_1 - T_2)$ = -169 kJ/kg

Example Ideal Brayton cycle



Air enters the compressor of a gas turbine at 100 kPa and 15°C. The pressure at the outlet of the compressor is 0.5 MPa and the maximal temperature of the cycle is 900°C.
Determine:

1- The work of the compressor.
2- The work of the turbine.
3- The thermal efficiency.

Assume both the compressor and turbines are isentropic devices. Cp = 1.0035 kJ/kg-K for air.

Computation of work of the turbine:

$\dot{m}_3 = \dot{m}_4 = \dot{m}$

$w_T = h_3 - h_4 = c_p (T_3 - T_4)$

$T_3 = 1173.2$ K **max.**

To compute T_4:

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{k-1}{k}}.$$

but: $P_3 = P_2$ and $P_4 = P_1$

Then: $T_4 = 740.4$ K

and $w_T = 1.0035\,(1173.2 - 740.4)$

$w_T = 434.3$ kJ/kg

Unless specified, Ideal Brayton:
constant pressure across combustor/heat exchanger

Example Ideal Brayton cycle

Air enters the compressor of a gas turbine at 100 kPa and 15°C. The pressure at the outlet of the compressor is 0.5 MPa and the maximal temperature of the cycle is 900°C.
Determine:

1- The work of the compressor.
2- The work of the turbine.
3- The thermal efficiency.

Assume both the compressor and turbines are isentropic devices. Cp = 1.0035 kJ/kg-K for air.

- Thermal efficiency:

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

$$q_{in} = h_3 - h_2 = c_p (T_3 - T_2)$$

$$q_{in} = 719.1 \text{ kJ/kg}$$

$$\eta_{th} = 36.9\%$$

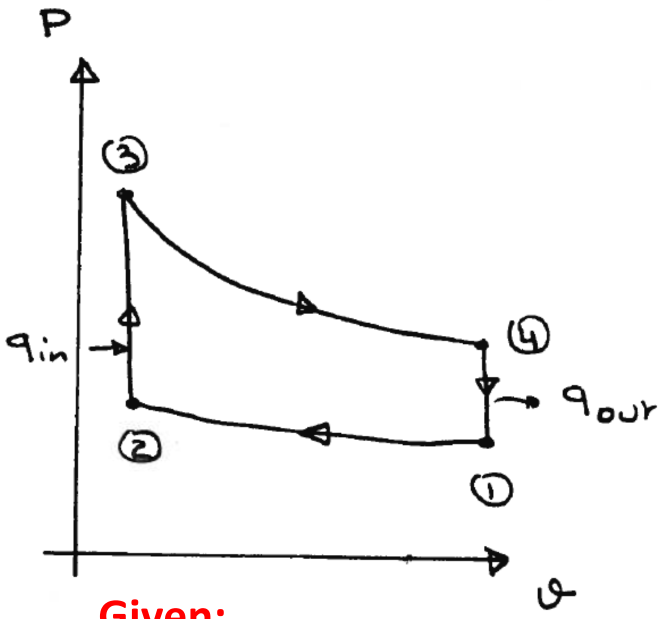
Example

Volume ratio

The compression ratio of an ideal Otto cycle is 8. At the onset of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C.

The heat supplied to air, per cycle, is 1800 kJ/kg. Determine:

1- The pressure and the temperature for each state.
2- The thermal efficiency
3- The mean effective pressure.



Given:

$P_1 : 0.1$ MPa

$T_1 : 15°C : 288$ K , $R : 0.287$

$r : 8$, $q_{in} : 1800$ kJ/kg kJ/kg

1.1. State ①:

$P_1 : 0.1$ MPa , $T_1 : 288$ K

$$v_1 : \frac{RT_1}{P_1} : 0.827 \text{ m}^3/\text{kg}$$

1.2. State ②:

$$\frac{T_2}{T_1} : \left(\frac{v_1}{v_2}\right)^{k-1} : 8^{0.4}$$

$T_2 : 662$ K

Unless specified, for ideal Otto cycle: Ideal gas / isentropic process / closed system analysis

Example

The compression ratio of an ideal Otto cycle is 8. At the onset of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C.
The heat supplied to air, per cycle, is 1800 kJ/kg. Determine:

1- The pressure and the temperature for each state.
2- The thermal efficiency
3- The mean effective pressure.

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^k = r^k = 8^{1.4}$$

$P_2 = 1.838$ MPa.

$$v_2 = \frac{v_1}{r} = \frac{0.827}{8} = 0.1034\ m^3/kg$$

1.3. state ③ :

we know:

$$q_{23} = \underline{c_v}\,(T_3 - T_2) = 1800 \text{ kJ/kg}$$

Therefore: $T_3 = T_2 + \frac{q_{23}}{c_v}$

$$T_3 = 662 + \frac{1800}{0.7165}$$

$$T_3 = 3174\ K$$

Constant Volume
heat addition process

$$(u_3 - u_2) = \frac{Q}{m} - \cancel{\frac{W}{m}}$$

$$Q = m(u_3 - u_2) = mc_v(T_3 - T_2)$$

$$\frac{p_3}{p_2} = \frac{T_3}{T_2}$$

Example

The compression ratio of an ideal Otto cycle is 8. At the onset of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C.

The heat supplied to air, per cycle, is 1800 kJ/kg. Determine:

1- The pressure and the temperature for each state.

2- The thermal efficiency

3- The mean effective pressure.

for r_3 :

$P_3 \vartheta_3 : RT_3$; $P_2 \vartheta_2 : RT_2$

but $\vartheta_2 : \vartheta_3$

Then: $P_3 : \frac{T_3 P_2}{T_2}$

$P_3 : 8.813$ MPa

1.4. State ④

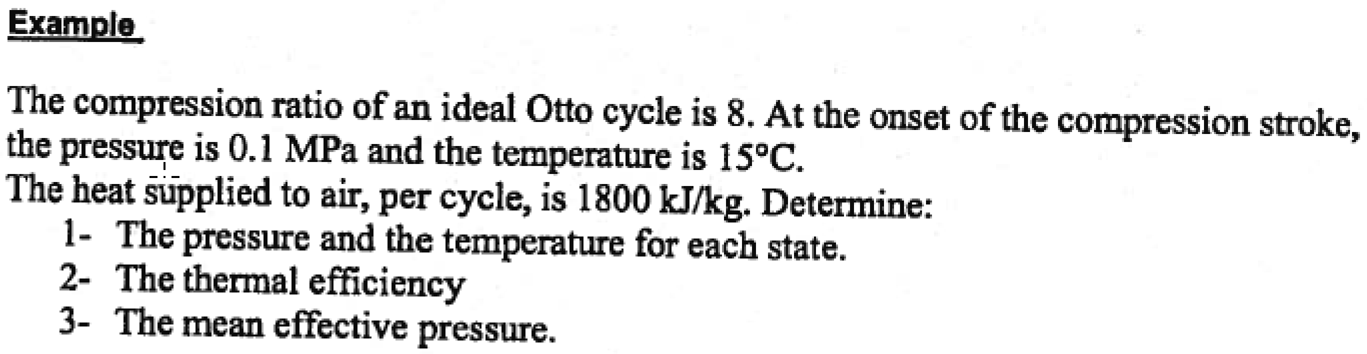
$$\frac{T_3}{T_4} = \left(\frac{\vartheta_4}{\vartheta_3}\right)^{k-1} = r^{k-1} : 8^{0.4}$$

$T_4 : 1380$ K

$$\frac{P_3}{P_4} = \left(\frac{\vartheta_4}{\vartheta_3}\right)^{k} = r^{1.4} = 8^{1.4}$$

$P_4 = 0.4795$ MPa.

$\vartheta_4 = \vartheta_1 : 0.827$ m³/kg



Example

The compression ratio of an ideal Otto cycle is 8. At the onset of the compression stroke, the pressure is 0.1 MPa and the temperature is 15°C.
The heat supplied to air, per cycle, is 1800 kJ/kg. Determine:

1- The pressure and the temperature for each state.
2- The thermal efficiency
3- The mean effective pressure.

Thermal efficiency:

$$\eta_{th} = 1 - \frac{|q_{out}|}{q_{in}}$$

$$q_{out} = C_v (T_1 - T_4)$$

$$= -782.3 \text{ kJ/kg}$$

$$q_{in} = 1800 \text{ kJ/kg}$$

$$\eta_{th} = 0.565$$

oR

$$\eta_{th} = 56.5 \%$$

$$(u_2 - u_1) = \cancel{\frac{Q}{m}} - \frac{W}{m}$$

$$W = m(u_1 - u_2) = mc_v(T_1 - T_2)$$

Isentropic compression

$$(u_4 - u_3) = \cancel{\frac{Q}{m}} - \frac{W}{m}$$

$$W = m(u_3 - u_4) = mc_v(T_3 - T_4)$$

Isentropic expansion

$\eta = \omega_{net}/q_{in}$

$\eta = [c_v (T_1 - T_2) + c_v (T_3 - T_4)] / 1800$ kJ/kg

$\eta = 0.565$

$\eta = 1 - r^{(1-k)} = 0.565$ with $k = 1.4$ for air

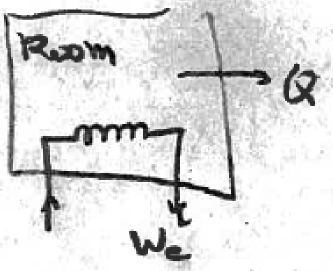
Question:

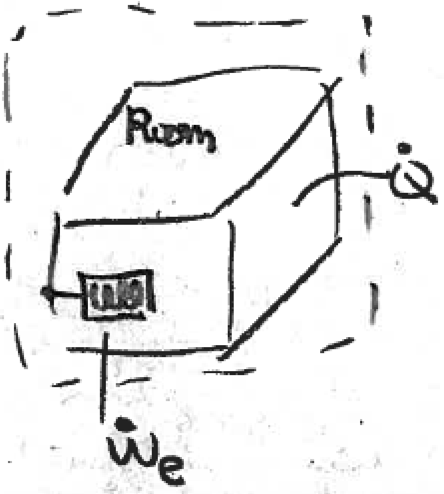
A room is heated by a baseboard resistance heater. When the heat losses from the room on a winter day amount to 6500 kJ/hr, the air temperature in the room remains **constant** even though the heater operates continuously. Determine the power rating of the heater in kW.

$$\underbrace{\cancel{du}}_{0} = \delta Q - \underbrace{\delta W}$$

although no boundary work . CV

but We





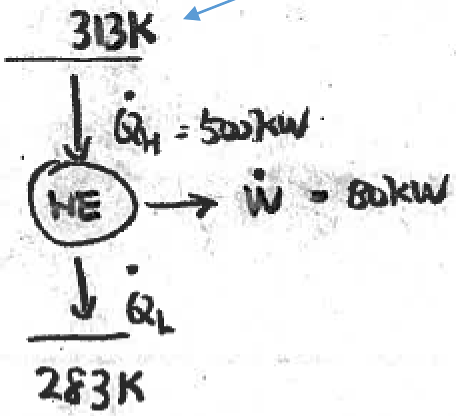
$$dQ = We = (\underbrace{-6500\ kJ/h}_{\text{heat loss}})\left(\frac{1\ kW}{3600\ kJ/h}\right) = \underbrace{-1.81\ kW}_{\text{supply}}$$

The heater having electric work input of 1.81 kW

Question:

Let’s say, someone proposes to you that he develops an actual heat engine (through a series of different processes within the engine) that produces a power of 80 kW by absorbing heat at a rate of 500 kW from a geothermal site at 40 °C and rejects heat to the environment at 10 °C. Do you believe it?

1) 1st law



Change to Kelvin unit!!

$$|\dot{W}| = |\dot{Q}_H| - |\dot{Q}_L|$$

$$|\dot{Q}_L| = |\dot{Q}_H| - |\dot{W}| = 420kW$$

$$\left.\begin{array}{l} \eta_{actual} = \dfrac{\Sigma \dot{W}}{\dot{Q}_{in}} = 1 - \dfrac{\dot{Q}_L}{\dot{Q}_H} = 16\% \\ \\ \eta_{th\,Carnot} = 1 - T_C/T_H = 1 - 283/313 = 9.58\% \end{array}\right\} \text{not possible}$$

Let’s say, someone proposes to you that he develops an actual heat engine (through a series of different processes within the engine) that produces a power of 80 kW by absorbing heat at a rate of 500 kW from a geothermal site at 40 °C and rejects heat to the environment at 10 °C. Do you believe it?

2) Clausius inequality

$$\oint \frac{\delta Q}{T} \leq 0$$

not valid!

$$\frac{+500 \text{ kW}}{313 \text{ K}} + \frac{(-420 \text{kW})}{283 \text{ K}} = 0.113345 \text{ kW/K} \geq 0$$

3) Entropy generation

cyclic device

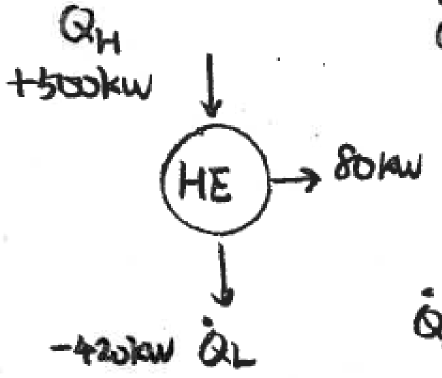
$$\Delta \dot{S}_{\substack{\text{universe} \\ \text{generated}}} = \overset{0}{\cancel{\Delta \dot{S}_{\text{sys}}}} + \Delta \dot{S}_{\text{surrounding}}$$

Let’s say, someone proposes to you that he develops an actual heat engine (through a series of different processes within the engine) that produces a power of 80 kW by absorbing heat at a rate of 500 kW from a geothermal site at 40 °C and rejects heat to the environment at 10 °C. Do you believe it?

$$\dot{S}_{generated} = \overbrace{\oint \frac{\delta\dot{Q}_{surrounding}}{T}}^{\Delta\dot{S}_{surrounding}}$$

$$\frac{\delta\dot{Q}_{surr.\,H}}{T} + \frac{\delta\dot{Q}_{surr.\,L}}{T}$$

$$= \frac{-500\text{ kW}}{313\text{k}} + \frac{420\text{ kW}}{283\text{k}} = -0.113345\text{ kW} \leq 0$$



$\dot{Q}_{surr.\,H} = -500\text{kW}$

$\dot{Q}_{surr.\,L} = +420\text{kW}$

impossible

against the increase of entropy principle