Thermodynamics
2026-1

S7_1: Control Volume_First law_Steady state

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Recomendaciones generales:

- Leer antes de clase y entender los conceptos o preguntar si es necesario.
- Practicar bastante:
  - *Ejercicios de clase
  - *Asesorías
  - *Ejercicios del libro resueltos y los de final de capitulo
- Los quizzes virtuales y gradescope úsenlos para aprender de forma honesta (se puede inferir).
- Tengan grupos de estudio y apóyense mutuamente.
- Crear sus propias notas estudio y formulario.
- Son 4 horas de clase en la semana (asistir y ser puntuales, perder alguna es atrasarse mucho)
- Cada hora de clase son 3 horas de estudio por fuera mínimo.
- En la ppt de la semana 1 están todos los links claves para el curso.

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EXÁMENES	FECHAS
EXAMEN PARCIAL	**(MAYO 11)**

TEORIA 1 : 3PM
TEORIA 2 : 5PM

Atentos a las indicaciones por anuncios en canvas para saber todos los detalles

EXAMEN PARCIAL : Todos los temas vistos hasta la semana 6

Thermodynamics _ Evaluation

COMPONENTE	%
EXAMEN PARCIAL	20
EXAMEN FINAL	30
C	50
APROBACIÓN: 11/20	

EXÁMENES	FECHAS
EXAMEN PARCIAL	(MAYO 11)
EXAMEN FINAL	(JULIO 6)

EXAMEN PARCIAL : Todos los temas vistos hasta la semana 6.
EXAMEN FINAL: Todos los temas vistos hasta la semana 15

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Thermodynamics _ Evaluation

COMPONENTE	%
PARCIAL	20
FINAL	30
C	50
APROBACIÓN: 11/20	

SEMANA 1	TAREA VIRTUAL	SE CALIFICA AUTOMATICO NO NECESITA PROCEDIMIENTO
SEMANA 2	TAREA VIRTUAL	
SEMANA 3	TAREA VIRTUAL	.
SEMANA 4	**TAREA INDIVIDUAL 1** + EVALUACION_ AULA1	TAREA INDIVIDUAL: Esta parte incluye subir un archivo con la solución EVALUACION_ AULA: SEMANAS 1-2
SEMANA 5	TAREA VIRTUAL+ EVALUACION_ AULA2	EVALUACION_ AULA: SEMANAS 1-3
SEMANA 6	TAREA VIRTUAL**+TAREA INDIVIDUAL 2**	
SEMANA 7	TAREA VIRTUAL+ EVALUACION_ AULA3	EVALUACION_ AULA: SEMANAS 1-6
SEMANA 8	**PARCIAL**	
SEMANA 9	EVALUACION_ AULA4+**TAREA INDIVIDUAL 3 +Tarea grupal 1**	EVALUACION_ AULA: SEMANAS 1-8
SEMANA 10	TAREA VIRTUAL	
SEMANA 11	TAREA VIRTUAL	
SEMANA 12	TAREA VIRTUAL+ EVALUACION_ AULA5	EVALUACION_ AULA: SEMANAS 1-11
SEMANA 13	TAREA VIRTUAL**+TAREA INDIVIDUAL 4 + Tarea grupal 2**	
SEMANA 14	EVALUACION_ AULA6	EVALUACION_ AULA : SEMANAS 1-13
SEMANA 15		
SEMANA 16	**FINAL**	

*LAS TAREAS VIRTUALES O LOS GRADESCOPE (TAREA INDIVIDAL) **se abren lunes 2pm y se cierran sábado media noche. La tarea virtual se presenta hasta un máximo de 3 veces.**

EVALUACION_ AULA: es un quiz de 20 min aprox. en aula.

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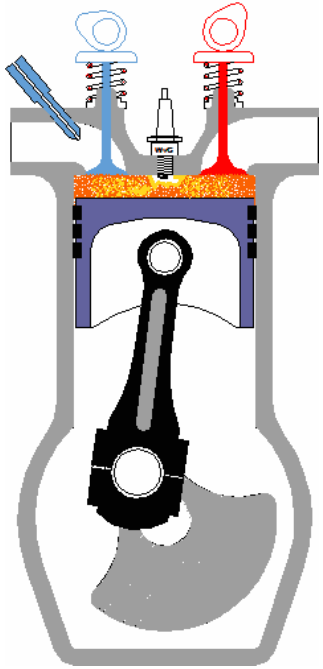
Objectives

Session 13-14: Apply the principle of conservation of energy and mass in open systems operating in a stationary or transient.

What happens when the system is no longer closed, but something is flowing in and out of it?

Example : Automobile Engine



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First Law_Energy : Closed Vs Open systems

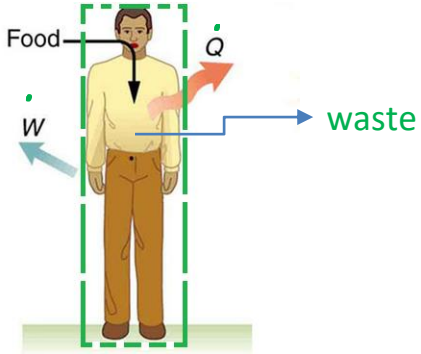
Energy change in the volume control can occur by:

a) Heat transfer

b) Work

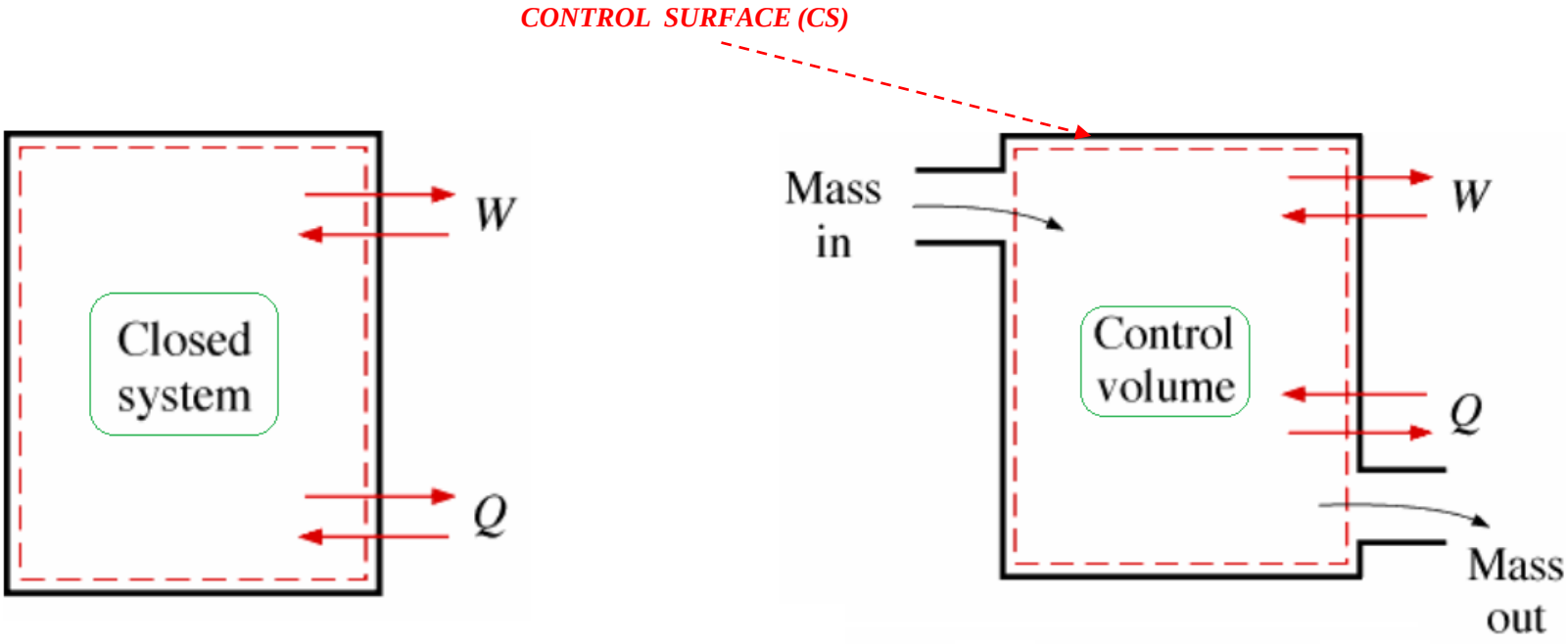
c) Transport of energy due to mass flow

d) All of the above



$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \text{Food energy} - \text{Waste}$$

Difference Between Closed and Open Systems



SYSTEM

CONTROL VOLUME

(CV)

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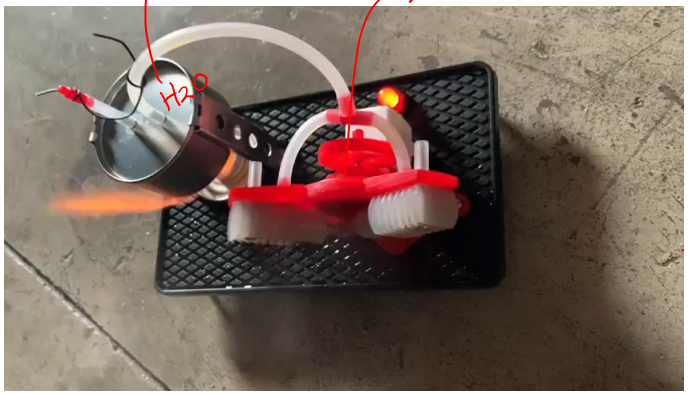

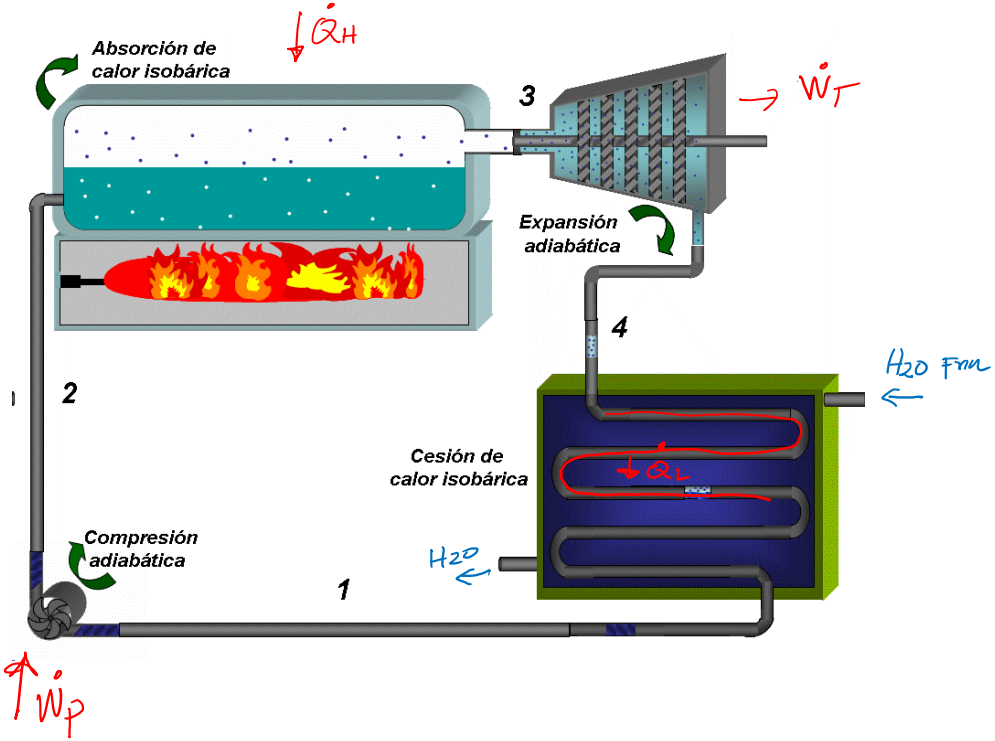

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https://www.youtube.com/watch?v=IdPTuwKEfmA&t=1s&ab_channel=SabinCivilEngineering

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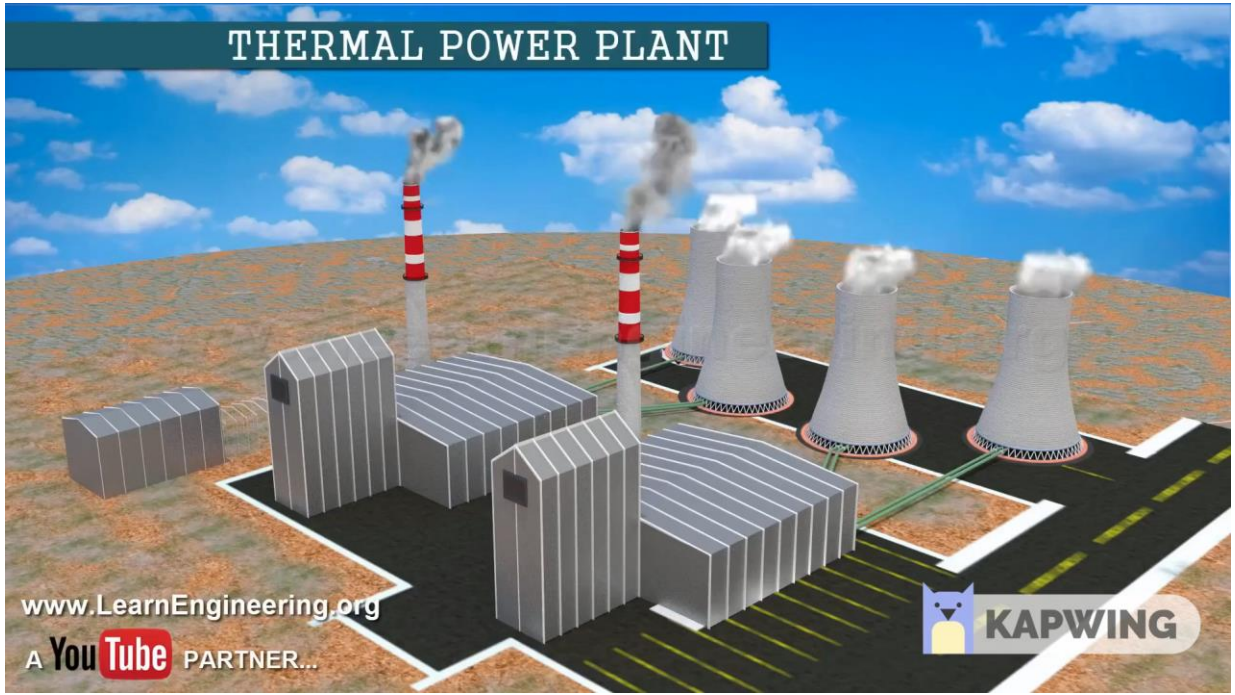
Work-Heat- First Law_Power plant



https://www.youtube.com/watch?v=IdPTuwKEfmA&t=244s

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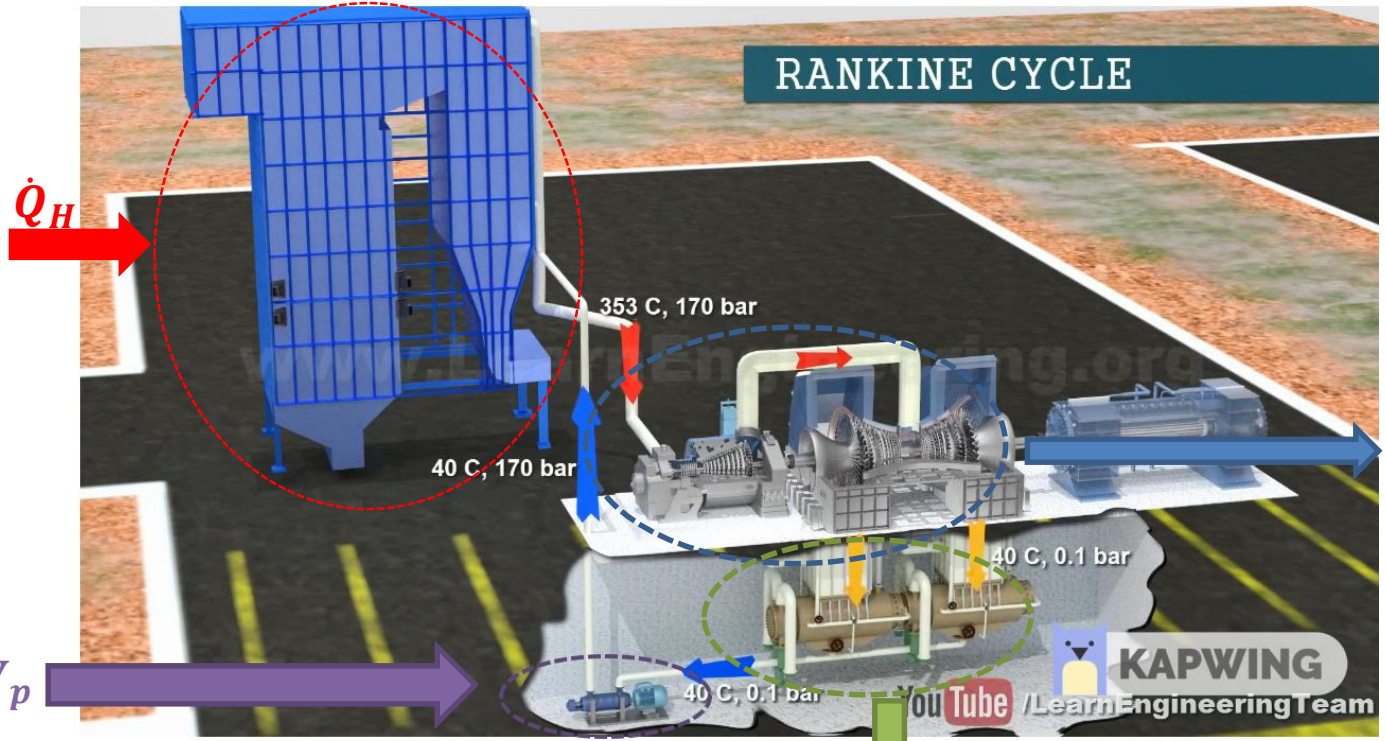
Work-Heat- First Law_Power plant



$\dot{Q}_H$

$\dot{W}_t$

$\dot{W}_p$

$\dot{Q}_L$

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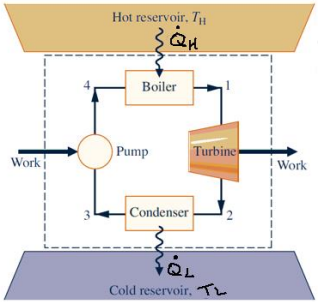
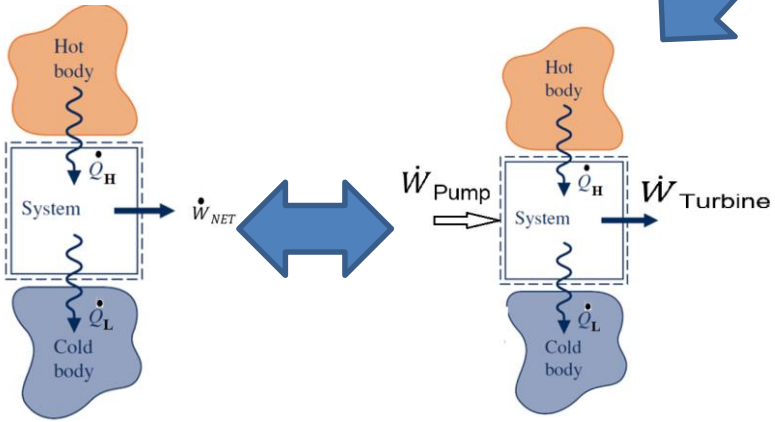
Power Cycles Work-Heat-Firts Law

CONTROL MASS



Mass control



$$\frac{dE}{dt} = \dot{Q} - \dot{W}$$

$$\eta = \frac{|\dot{W}_T| - |\dot{W}_P|}{|\dot{Q}_H|} = \frac{|\dot{W}_{NET}|}{|\dot{Q}_H|}$$

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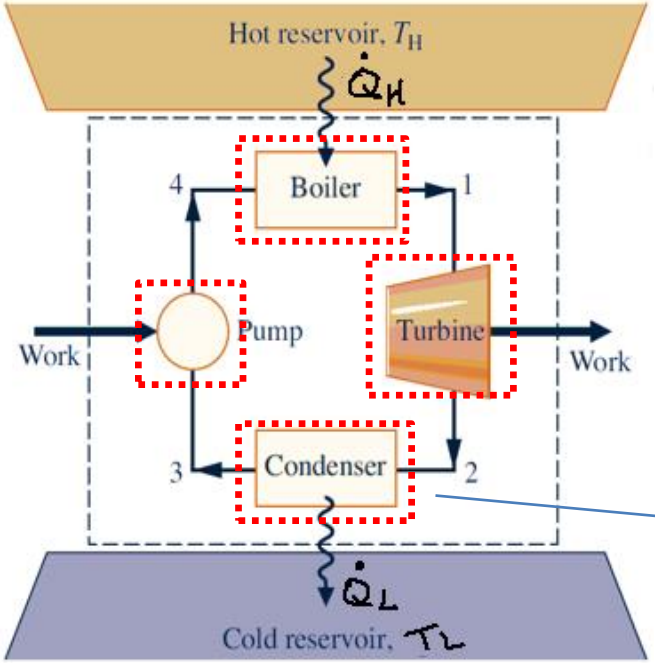
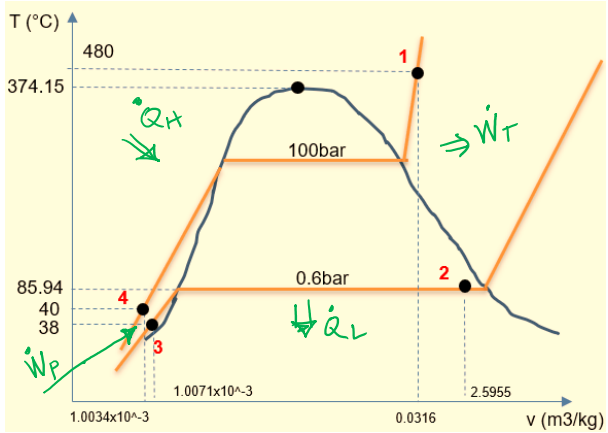
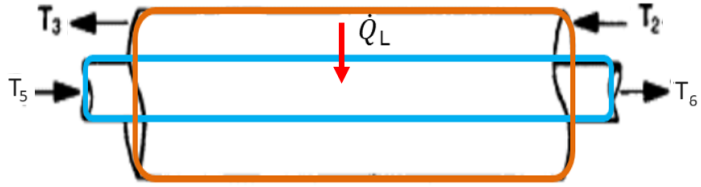
Work-Heat- First Law _ Power plant

$Q_H = Q_{in}$…AND $Q_L = Q_{out} = Q_C$







$\frac{T_5+T_6}{2} = T_L$

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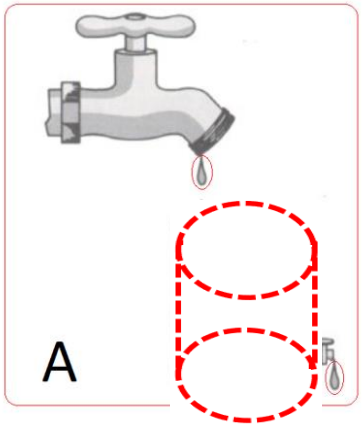
Conditions to analyze an open system:

- Steady state:
  - The mass that enters is equal to the one that leaves (MASS BALANCE)
  - The properties do not change in time (ENERGY BALANCE)
- Transient: Next class

Control Volume_Firts law

In terms of mass:

Steady-State (*)



$$\dot{m}_{in} - \dot{m}_{out} = dm_{CV}/dt \qquad (kg/s)$$

0 (*)

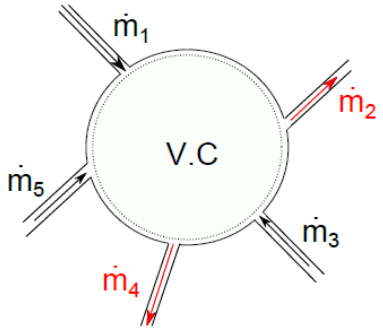
First Law_Control Volume

Mass Conservation Equation

It can also be expressed as:

Steady-State

$$\frac{dm_{CV}}{dt} = \sum_{in} \dot{m} - \sum_{out} \dot{m}$$



Thus for incompressible flow or even for compressible flow where ρ is uniform across A,

$$\dot{m} = \rho V_{avg} A_c \qquad (kg/s)$$

$$\dot{m} = \rho \underbrace{V_{avg} A_c}_{\dot{V}} \qquad (kg/s)$$

$$\dot{m} = \rho \dot{V} = \frac{\dot{V}}{v}$$

For ideal gases we can also use:

$$P\dot{V} = \frac{\dot{m}\bar{R}T}{M}$$

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Mass Balance for Steady-Flow Processes

$$\sum_{\text{in}} \dot{m} = \sum_{\text{out}} \dot{m} \qquad (\text{kg/s})$$

Steady flow (single stream): $\dot{m}_1 = \dot{m}_2 \rightarrow \rho_1 V_1 A_1 = \rho_2 V_2 A_2$

For steady flow of liquids, however, the volume flow rates, as well as the mass flow rates, remain constant since liquids are essentially incompressible (constant-density) substances:

For single-stream steady-flow systems it becomes

Steady, incompressible flow (single stream): $\dot{V}_1 = \dot{V}_2 \rightarrow V_1 A_1 = V_2 A_2$

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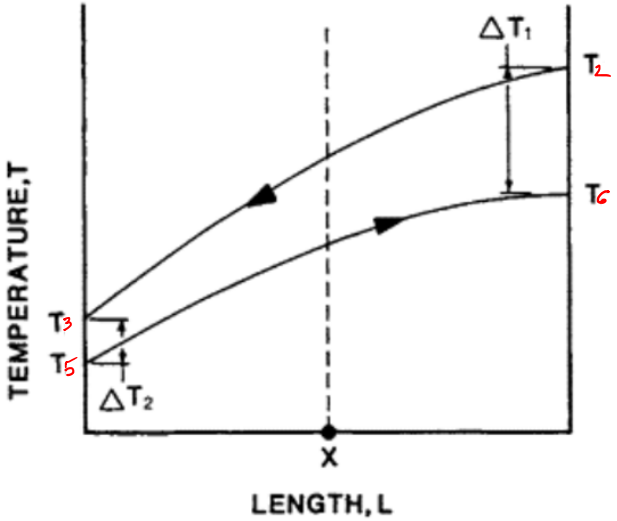
In terms of energy:



T2 – HOT FLUID IN
T3 – HOT FLUID OUT
T5 – COLD FLUID IN
T6 – COLD FLUID OUT

https://www.sciencedirect.com/science/article/pii/B9780884158226500039

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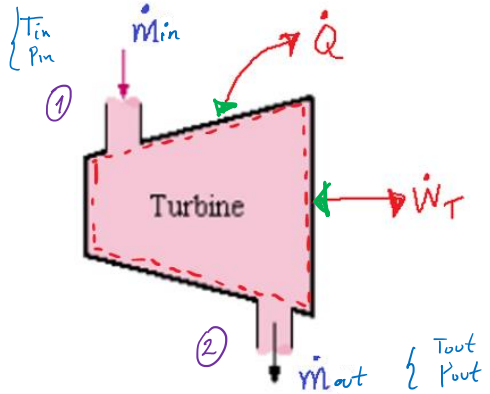
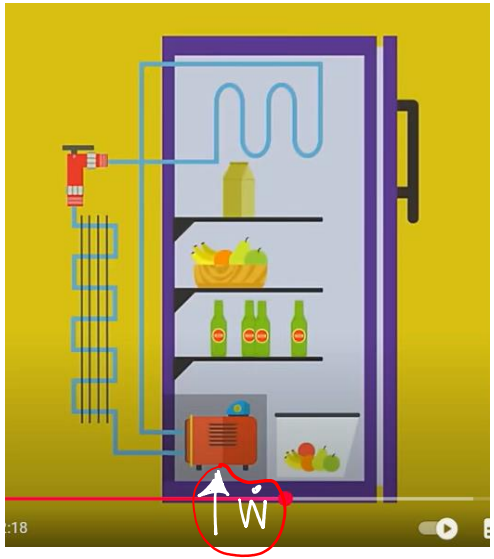


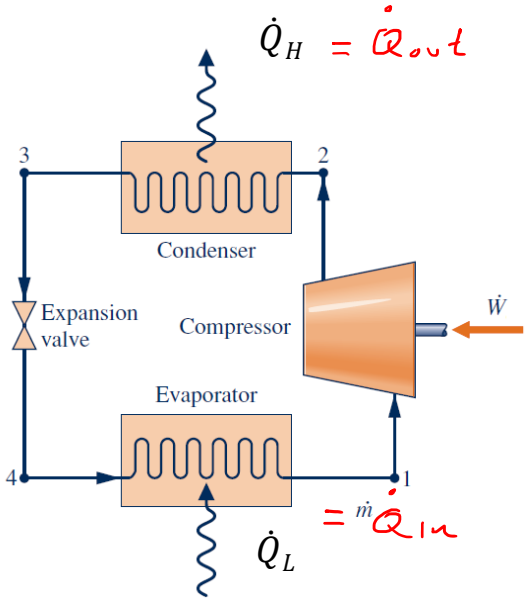
$$\frac{dE_{C.V.}}{dt} = \dot{m}_{in}\left(h_{in} + \frac{1}{2}\mathbf{V}_{in}^2 + gZ_{in}\right) - \dot{m}_{out}\left(h_{out} + \frac{1}{2}\mathbf{V}_{out}^2 + gZ_{out}\right) + \dot{Q}_{C.V.} - \dot{W}_{C.V.} \left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

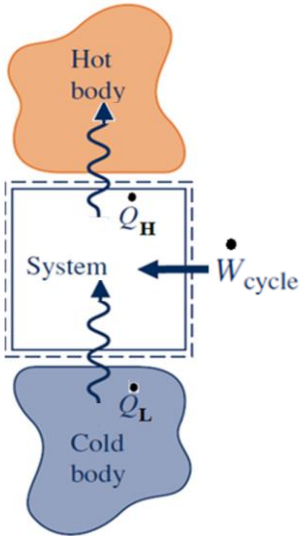
$$dm_{CV}/dt = \dot{m}_{in} - \dot{m}_{out} \quad (\text{kg/s})$$

In E.E conditions: $$\frac{dE_{C.V.}}{dt} = \dot{m}\left[\left(h_{in} + \frac{1}{2}\mathbf{V}_{in}^2 + gZ_{in}\right) - \left(h_{out} + \frac{1}{2}\mathbf{V}_{out}^2 + gZ_{out}\right)\right] + \dot{Q}_{C.V.} - \dot{W}_{C.V.} \left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right)$$

Refrigeration cycle







$$\mathrm{COP_R} = \boldsymbol{\beta}_R = \left|\frac{\dot{Q}_L}{\dot{W}}\right|$$

$$\frac{dE}{dt} = \dot{Q} - \dot{W}$$

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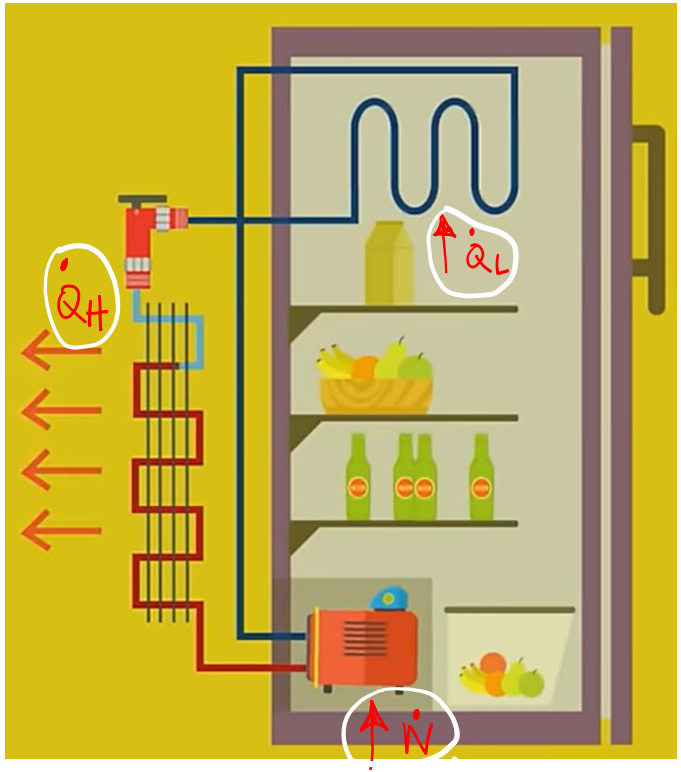
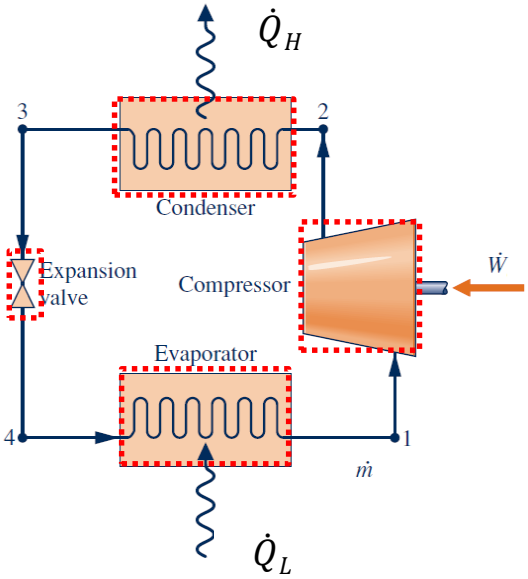
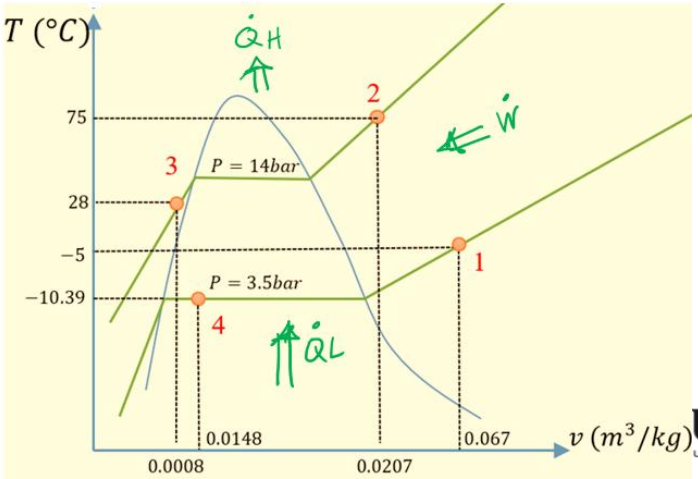
EXAMPLES OF STEADY-STATE PROCESSES _ REFRIGERATOR

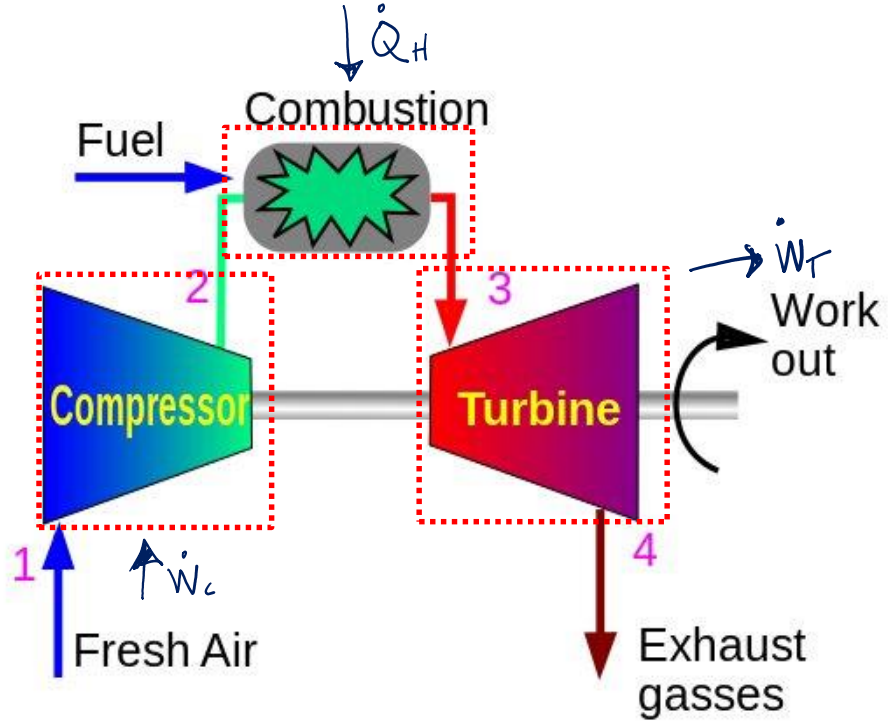






https://www.youtube.com/watch?v=EIP3pSio7-M

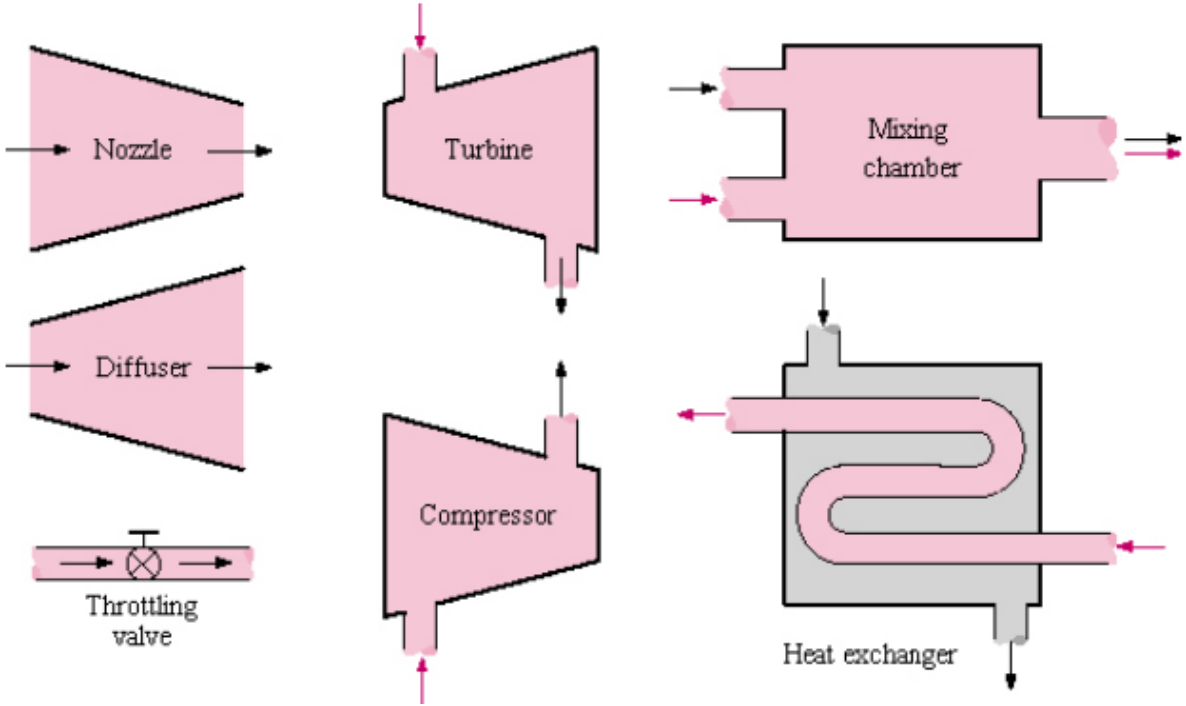
H.1:49

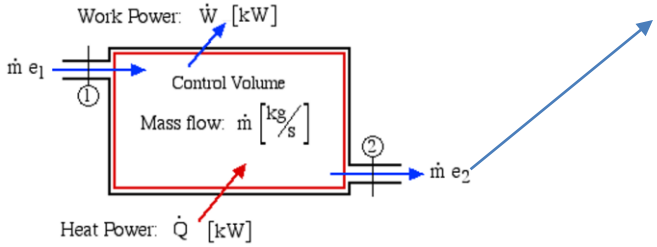


https://www.youtube.com/watch?v=x8DK4rM6Y90&ab_channel=Pratt%26Whitney

https://www.turbinesinfo.com/gas-turbines/

Work-Heat- First Law_ Other devices





$$e_{mass} = u + ke + pe = u + \frac{V^2}{2} + gz \quad (kJ/kg)$$

$$\frac{dE_{vc}}{dt} = \dot{m}_{in}\left(u_{in} + \frac{V_{in}^2}{2} + gz_{in}\right) - \dot{m}_{out}\left(u_{out} + \frac{V_{out}^2}{2} + gz_{out}\right) + \dot{Q}_{vc} - \dot{W}$$

ENERGY BALANCE _OPEN SYSTEM

$$\frac{dE_{vc}}{dt} = \dot{m}_{in}\left(u_{in} + \frac{V_{in}^2}{2} + gz_{in}\right) - \dot{m}_{out}\left(u_{out} + \frac{V_{out}^2}{2} + gz_{out}\right) + \dot{Q}_{vc} - \dot{W}$$

$\dot{W} \rightarrow (W_{cv} + W_{flow})$

$W_{flow} = F\,dx$

$= pA\,dx$

$\dot{W}_{flow} = pA\frac{dx}{dt}$

$= pA\vec{v}$, $\dot{m} = \rho A\vec{v}$

$= \dot{m}\frac{p}{\rho}$

$= \dot{m}pv$

$W_{flow,in} = \dot{m}\,p_{in}\,v_{in} < 0$

$W_{flow,out} = \dot{m}\,p_{out}\,v_{out}$

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ENERGY BALANCE _OPEN SYSTEM

$$\frac{dE_{CV}}{dt} = \dot{m}_{in}\left(u_{in} + p_{in}v_{in} + \frac{1}{2}V_{in}^2 + gz_{in}\right) - \dot{m}_{out}\left(u_{out} + p_{out}v_{out} + \frac{1}{2}V_{out}^2 + gz_{out}\right) + \dot{Q}_{CV} - \dot{W}_{CV}$$

h_{in} h_{out}

$$\frac{dE_{CV}}{dt} = \sum_{in}\left(\dot{m}_{in}\left(h_{in} + \frac{1}{2}V_{in}^2 + gz_{in}\right)\right) - \sum_{out}\left(\dot{m}_{out}\left(h_{out} + \frac{1}{2}V_{out}^2 + gz_{out}\right)\right) + \dot{Q}_{CV} - \dot{W}_{CV}$$

$$\frac{dE_{CV}}{dt} = \dot{m}_{in}\left(h_{in} + \frac{1}{2}V_{in}^2 + gz_{in}\right) - \dot{m}_{out}\left(h_{out} + \frac{1}{2}V_{out}^2 + gz_{out}\right) + \dot{Q}_{CV} - \dot{W}_{CV}$$

, Despreciando cinética y potencial

$$\frac{dE_{CV}}{dt} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q}_{CV} - \dot{W}_{CV}$$

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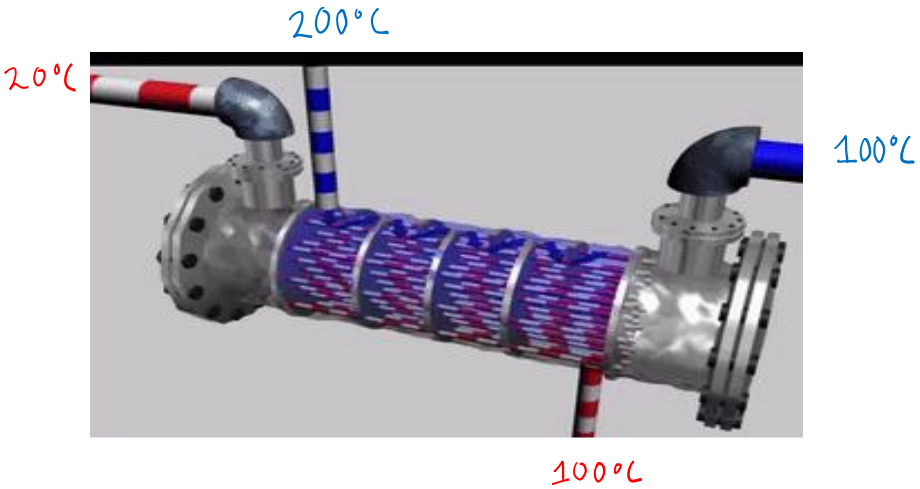
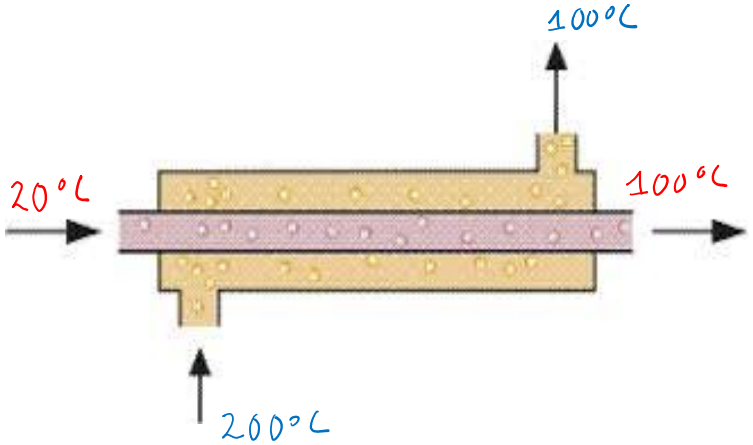
EXAMPLES OF STEADY-STATE PROCESSES_Heat Exchanger

A steady-state heat exchanger is a simple fluid flow through a pipe or system of pipes, where heat is transferred to or from the fluid. The fluid may be heated or cooled, and may or may not boil, changing from liquid to vapor, or condense, changing from vapor to liquid.



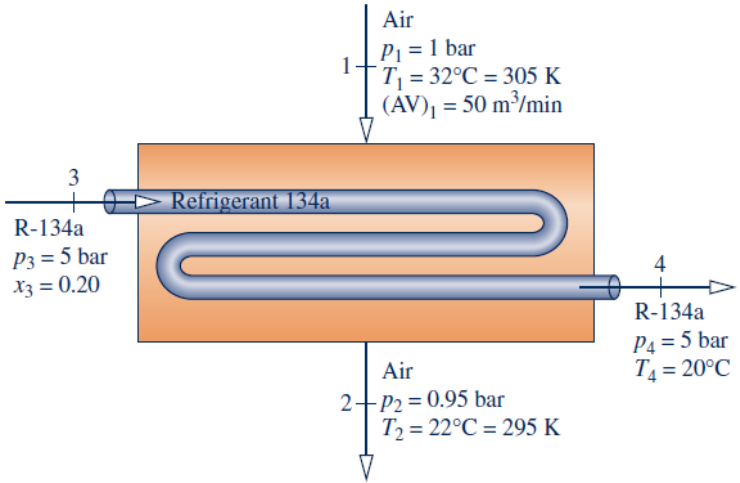


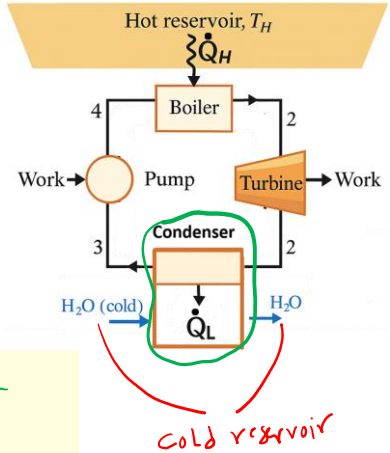
Example 2.2 (heat exchanger):

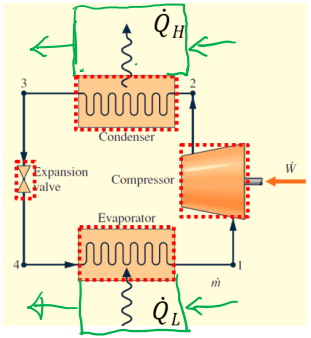
A heat exchanger is shown below in which air flows over tubes carrying Refrigerant 134a. Air (ideal gas) enters with a volumetric flow rate of 50 m^3/min at 305 K, 1 bar, and exits at 295 K, 0.95 bar. Refrigerant enters the tubes at 5 bar with a quality of 20% and exits at 5 bar, 20°C. Considering that the heat exchanger has adiabatic external walls and neglecting kinetic and potential energy effects, use table A22 for air. Determine at steady state:

(a) the mass flow rate of the refrigerant, in kg/min.

(b) the rate of heat transfer, in kJ/min, between the air and refrigerant.







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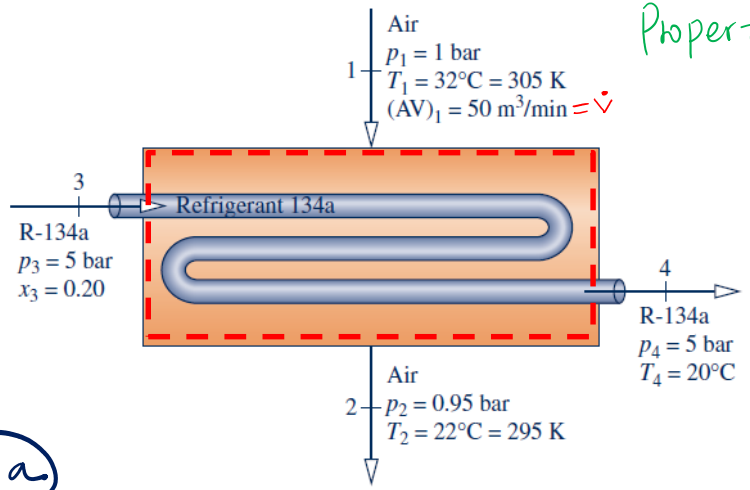
Properties of the substances at the 4 states



$$pV = n\bar{R}T = m\frac{\bar{R}}{M}T$$

$$p\dot{V} = \dot{m}\left(\frac{\bar{R}}{M}\right)T$$

$$\dot{m} = \frac{p\dot{V}}{\left(\frac{\bar{R}}{M}\right)T} = \frac{100 \times 50}{(8.314/28.97) \times 305} = 57.123\ kg/min$$

R134a

TABLE A-11

Pressure Conversions: 1 bar = 0.1 MPa = 10^2 kPa

Properties of Saturated Refrigerant 134a (Liquid–Vapor): Pressure Table

		Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K	
Press. bar	Temp. °C	Sat. Liquid $v_f \times 10^3$	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
5.0	15.74	0.8056	0.0409	70.93	235.64	71.33	184.74	256.07	0.2723	0.9117

$$h_3 = 71.33 + 0.20(256.07 - 71.33) = 108.278\ kJ/kg$$

TABLE A-12

(Continued)

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
$p = 5.0$ bar $= 0.50$ MPa ($T_{sat} = 15.74°C$)				
Sat.	0.04086	235.64	256.07	0.9117
20	0.04188	239.40	260.34	0.9264
30	0.04416	248.20	270.28	0.9597
40	0.04633	256.99	280.16	0.9918
50	0.04842	265.83	290.04	1.0229

TABLE A-22

Ideal Gas Properties of Air

T(K), h and u(kJ/kg), s° (kJ/kg · K)					
				when $\Delta s = 0$[1]	
T	h	u	s°	p_r	v_r
295	295.17	210.49	1.68515	1.3068	647.9
305	305.22	217.67	1.71865	1.4686	596.0

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Properties of R-134a

State 3: (Table A-11)

$P_3 = 5bar, T_3 = T_{sat} = 15.74°C$

$h_3 = 71.33 + 0.20(256.07 - 71.33) = 108.278\ kJ/kg$

State 4: (Table A-12)

$P_4 = 5bar, T_4 = 20°C$

$h_4 = 260.34\ kJ/kg$

Properties of Air

Assuming ideal gas behavior for air (properties does not depend on pressure)

State 1: (Table A-22)

$P_1 = 1bar, T_1 = 32°C = 305K$

$h_1 = 305.22\ kJ/kg$

State 2: (Table A-22)

$P_2 = 0.95bar, T_2 = 22°C = 295K$

$h_2 = 295.17\ kJ/kg$

a) To find the mass flow rate of the refrigerant we use the entire heat exchanger as the control volume.

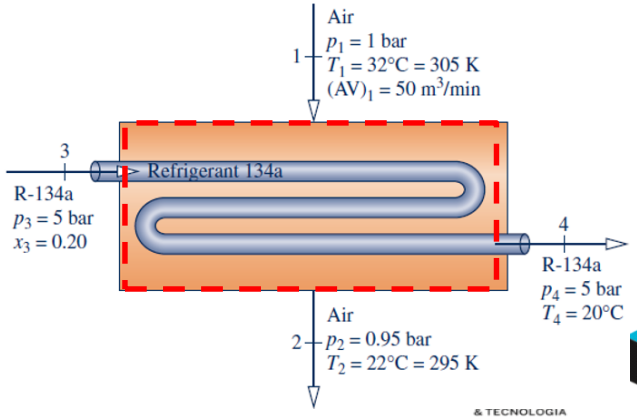
Energy balance for open systems (neglecting changes in potential and kinetic energy):

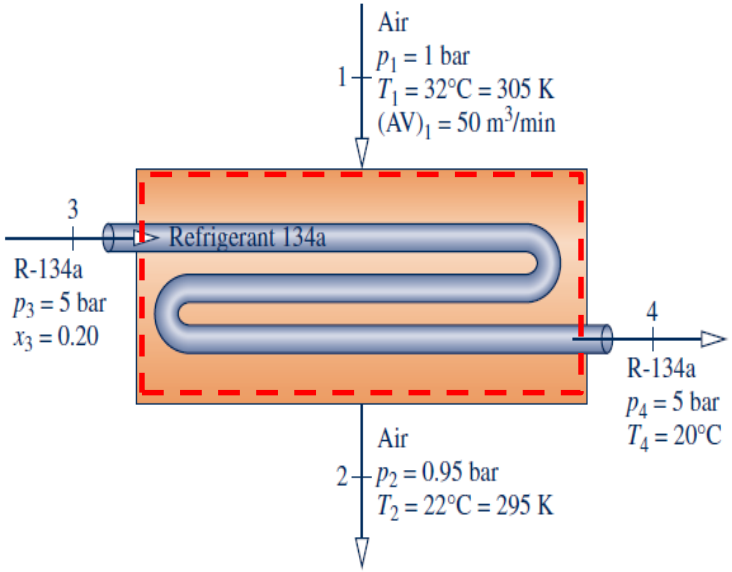
Adiabatic (outer surface)

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in} h_{in} - \sum \dot{m}_{out} h_{out}$$

Steady state

No shafts or resistances





$$\sum \dot{m}_{in} h_{in} = \sum \dot{m}_{out} h_{out}$$

$$\dot{m}_{R134a} h_3 + \dot{m}_{air} h_1 = \dot{m}_{R134a} h_4 + \dot{m}_{air} h_2$$

$$\dot{m}_{R134a} (h_3 - h_4) = \dot{m}_{air} (h_2 - h_1)$$

cpΔT ??

$$\dot{m}_{R134a} (108.278 - 260.34) = 57.123\ (295.17 - 305.22)$$

$$\dot{m}_{R134a} (-152.062) = \left[57.123 \frac{kg}{min} \right] (-10.05)$$

$$\boxed{\dot{m}_{R134a} = 3.7753 \frac{kg}{min}}$$

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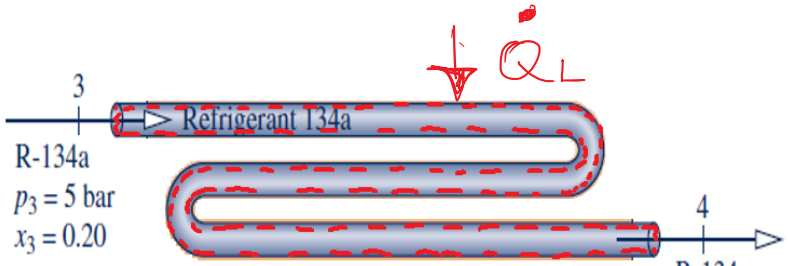
a) To find the rate of heat transfer we can choose any of the ducts:



$T_3 = T_{sat} = 15.74°C$

$$\frac{dE}{dt} = \dot{m}_{R\,134a}(h_3 - h_4) + \dot{Q}_L - \dot{W}$$

$$\dot{Q}_L = \dot{m}_{R\,134a}(h_4 - h_3)$$

$$= 3.7753\ (260.34 - 108.278)$$

$$= 574.0797\ \frac{KJ}{min}$$

Properties of R-134a

State 3: (Table A-11)

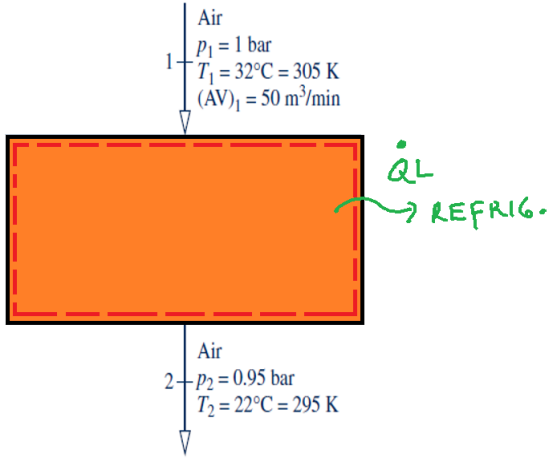
$P_3 = 5bar,\ T_3 = T_{sat} = 15.74°C$

$h_3 = 71.33 + 0.20(256.07 - 71.33) = 108.278\,kJ/kg$

State 4: (Table A-12)

$P_4 = 5bar,\ T_4 = 20°C$

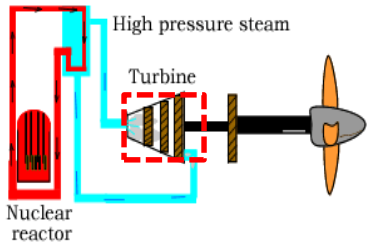
$h_4 = 260.34\,kJ/kg$

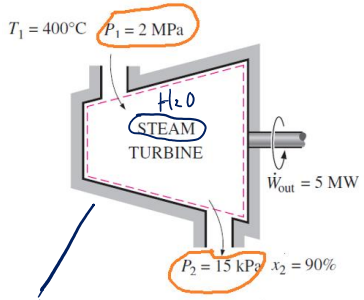


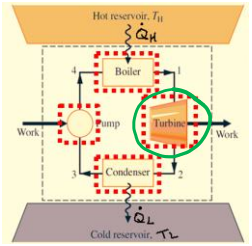
$\dot{Q}_L = -574.0861$

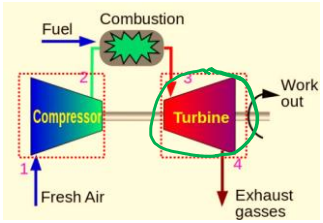
TAREA: Con este calor también pudo encontrarse el flujo $\dot{m}_{R134a}$

Turbine









A turbine is a rotary machine whose purpose is to produce shaft work (power, on a rate basis) at the expense of the pressure of the working fluid.

Let us consider the following assumptions for the analysis:

- The turbine operates adiabatically and is in a steady state.
- Changes in potential and kinetic energy are neglected.

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + z_{in}g + \frac{V_{in}^2}{2}\right) - \sum \dot{m}_{out}\left(h_{out} + z_{out}g + \frac{V_{out}^2}{2}\right)$$

$$\dot{W}_{cv} = \dot{m}(h_1 - h_2) \quad \frac{kJ}{s}\ [kW]$$

$$\frac{\dot{W}_{cv}}{\dot{m}} = w = h_1 - h_2 \rightarrow \frac{kJ}{kg}$$

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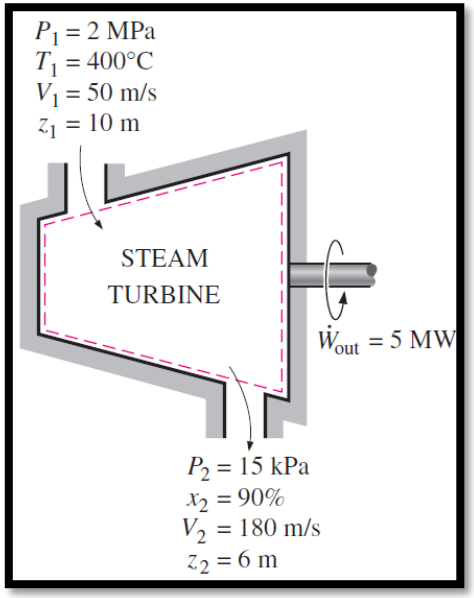
Power Generation by a Steam Turbine

Exercise 3.1:

The power output of an adiabatic steam turbine is 5 MW, and the inlet and the exit conditions of the steam are as indicated in the Figure. The turbine is in stationary state.

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + z_{in}g + \frac{V_{in}^2}{2}\right) - \sum \dot{m}_{out}\left(h_{out} + z_{out}g + \frac{V_{out}^2}{2}\right)$$

(a) Compare the magnitudes of Δh, Δke, and Δpe [kJ/kg].
(b) Determine the work done per unit mass of the steam flowing through the turbine (w [kJ/kg]).
(c) Calculate the mass flow rate of the steam (ṁ [kg/s]).



Considerations:

- This is a steady-flow process since there is no change with time at any point ($\dot{m}_1 = \dot{m}_2 = \dot{m}$, $dE/dt = 0$)
- The system is adiabatic and thus (Q=0).
- The inlet and exit velocities and elevations are given, and thus the kinetic and potential energies are to be considered.

Power Generation by a Steam Turbine

Solution

(*a*) At the inlet, steam is in a superheated vapor state, and its enthalpy is

$$\left.\begin{aligned} P_1 &= 2\ \text{MPa} \\ T_1 &= 400°\text{C} \end{aligned}\right\} \quad h_1 = 3247.6\ \text{kJ/kg}$$

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
p = 20.0 bar = 2.0 MPa (T_{sat} = 212.42°C)				
Sat.	0.0996	2600.3	2799.5	6.3409
240	0.1085	2659.6	2876.5	6.4952
280	0.1200	2736.4	2976.4	6.6828
320	0.1308	2807.9	3069.5	6.8452
360	0.1411	2877.0	3159.3	6.9917
400	0.1512	2945.2	3247.6	7.1271

At the turbine exit, we obviously have a saturated liquid–vapor mixture at 15-kPa pressure. The enthalpy at this state is

$$h_2 = h_f + x_2 h_{fg} = [221.62] + (0.9)(2597.2 - 221.62) = 2359.64\ \text{kJ/kg}$$

Press. bar	Temp. °C	Enthalpy kJ/kg: Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g
0.10	45.81	191.83	2392.8	2584.7
0.20	60.06	251.40	2358.3	2609.7

0.15 221.62 2597.2

Then

$$\Delta h = h_2 - h_1 = (\ 2359.64 - 3247.6\ )\ \text{kJ/kg} = \mathbf{-887.96\ kJ/kg}$$

$$\Delta \text{ke} = \frac{V_2^2 - V_1^2}{2} = \frac{(180\ \text{m/s})^2 - (50\ \text{m/s})^2}{2}\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{14.95\ kJ/kg}$$

$$\Delta \text{pe} = g(z_2 - z_1) = (9.81\ \text{m/s}^2)[(6 - 10)\ \text{m}]\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = \mathbf{-0.04\ kJ/kg}$$

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Solution

$$\cancel{\frac{dE}{dt}} = \dot{m}\left(h_1 + \frac{V_1^2}{2} + gz_1\right) - \dot{m}\left(h_2 + \frac{V_2^2}{2} + gz_2\right) + \cancel{\dot{Q}} - \dot{W}$$

$$w = \frac{\dot{W}}{\dot{m}} = (h_1 - h_2) + \left(\frac{V_1^2}{2} - \frac{V_2^2}{2}\right) + (gz_1 - gz_2)$$

$\Delta ek, \Delta ep \% 1000$

$$w = -\left[(h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)\right] = -(\Delta h + \Delta ke + \Delta pe)$$
$$= -[-887.96 + 14.95 - 0.04]\ \text{kJ/kg} = \mathbf{873.05\ kJ/kg}$$

(*c*) The required mass flow rate for a 5-MW power output is

$$\dot{m} = \frac{\dot{W}}{w} = \frac{5000\ \text{kJ/s}}{873.05\ \text{kJ/kg}} = \mathbf{5.73\ kg/s}$$

Discussion Two observations can be made from these results. First, the change in potential energy is insignificant in comparison to the changes in enthalpy and kinetic energy. This is typical for most engineering devices. Second, as a result of low pressure and thus high specific volume, the steam velocity at the turbine exit can be very high. Yet the change in kinetic energy is a small fraction of the change in enthalpy (less than 2 percent in our case) and is therefore often neglected.

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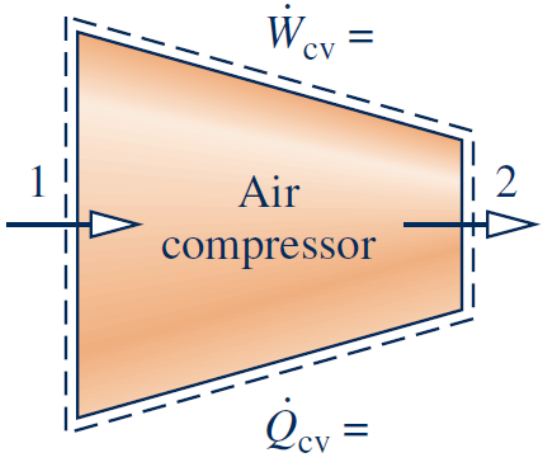
Example 5.2 (compressor):

Air, modelled as an ideal gas, is compressed at steady state from 1 bar, 300 K, to 5 bar, 500 K, with 150 kW of power input. Heat transfer occurs at a rate of 20 kW from the air to the surrounding environment. Neglecting kinetic and potential energy effects, determine the mass flow rate of the air, in kg/s.



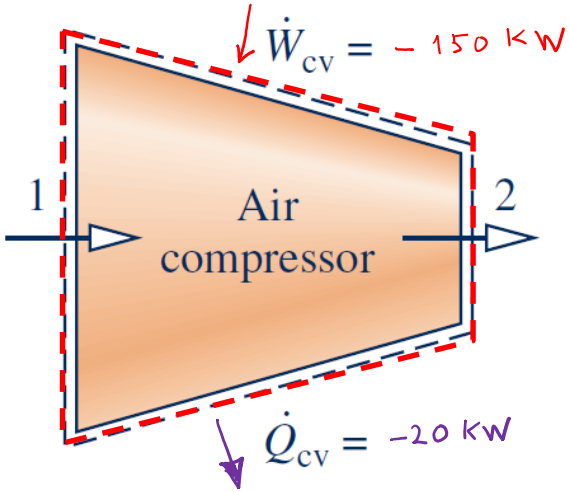
Selecting the compressor as control volume:

Properties of air

State 1: (Table A-22)

$P_1 = 1bar, T_1 = 300K$

$h_1 = 300.19\ kJ/kg$



Properties of air

State 2: (Table A-22)

$P_2 = 5bar, T_2 = 500K$

$h_2 = 503.02\ kJ/kg$

TABLE A-22

Ideal Gas Properties of Air

T(K), h and u(kJ/kg), s° (kJ/kg · K)											
				when $\Delta s = 0^1$						when $\Delta s = 0$	
T	h	u	s°	p_r	v_r	T	h	u	s°	p_r	v_r
300	300.19	214.07	1.70203	1.3860	621.2	500	503.02	359.49	2.21952	8.411	170.6

Energy balance for open systems

Heat to cooling jacket

Compressor

Steady state

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + z_{in}g + \frac{V_{in}^2}{2}\right) - \sum \dot{m}_{out}\left(h_{out} + z_{out}g + \frac{V_{out}^2}{2}\right)$$

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

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

$$0 = (-20) - (-150) + \dot{m}(300.19 - 503.02)$$

$$\dot{m} = \frac{-130}{-202.83} = 0.6409\ \frac{kg}{s}$$

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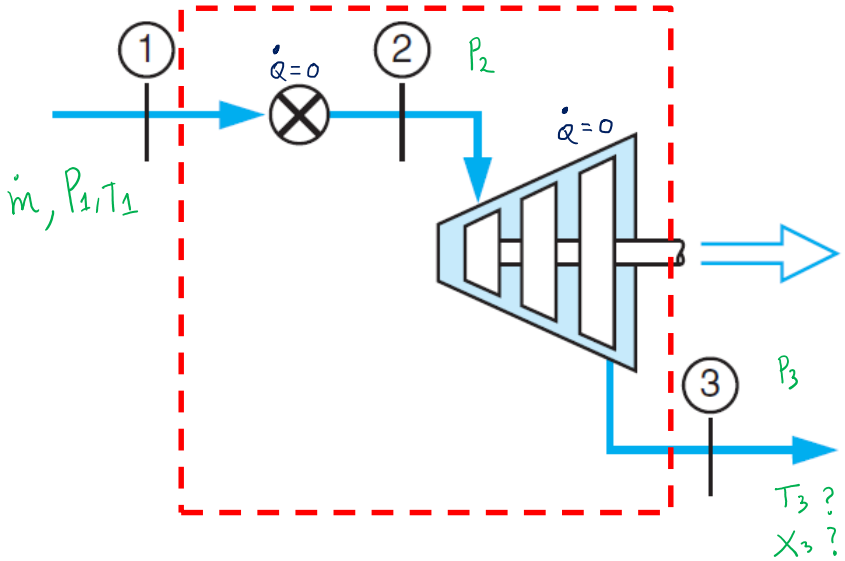
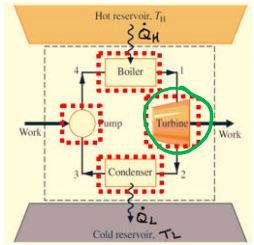
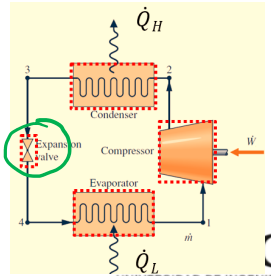
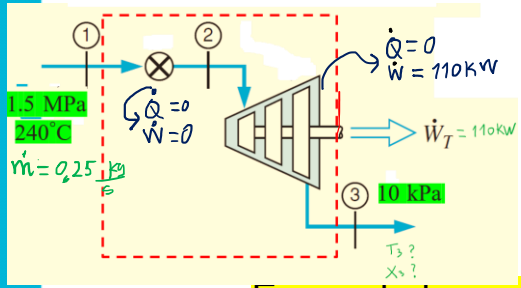
Example 4.3 (turbine):

A small adiabatic turbine, shown in figure below, is operated at part load by throttling adiabatically a 0.25-kg/s steam supply at 1.5 MPa and 240°C (state 1) down to 1.1 Mpa (state2) before it enters the turbine, and the exhaust is at 10 kPa (state3). If the turbine produces 110 kW ($\dot{W}_T$), find the exhaust temperature(T_3) and quality (x_3) if exist). 1, 2 and 3 are in stationary state. Neglect kinetic and potential energy.









Energy balance for open systems with considerations

$$\cancel{\frac{dE}{dt}} = \cancel{\dot{Q}} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + \cancel{z_{in}g} + \cancel{\frac{V_{in}^2}{2}}\right) - \sum \dot{m}_{out}\left(h_{out} + \cancel{z_{out}g} + \cancel{\frac{V_{out}^2}{2}}\right)$$

Stream 1 Stream 3

① $\begin{cases} P_1 = 15 \text{ bar} \\ T_1 = 240°C \end{cases}$ > SHV

TABLE A-4

(Continued)

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
		$p = 15.0$ bar $= 1.5$ MPa ($T_{sat} = 198.32°C$)		
240	0.1483	2676.9	2899.3	6.6628

$$\dot{W} = \dot{m}(h_1 - h_3)$$

$$110 = 0.25 \times (2899.3 - h_3)$$

$$h_3 = 2459.3\ kJ/kg$$

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Defining state 3 with P and h

$$P_3 = 10kPa = 0.1\,bar$$

$$h_3 = 2459.3\ kJ/kg$$

3

Mixture

$$h_3^{mix} = v_F + x_3(v_g - v_F)$$

TABLE A-3

Pressure Conversions: 1 bar = 0.1 MPa = 10^2 kPa

Properties of Saturated Water (Liquid–Vapor): Pressure Table

Press. bar	Temp. °C	Specific Volume m^3/kg Sat. Liquid $v_f \times 10^3$	Specific Volume m^3/kg Sat. Vapor v_g	Internal Energy kJ/kg Sat. Liquid u_f	Internal Energy kJ/kg Sat. Vapor u_g	Enthalpy kJ/kg Sat. Liquid h_f	Enthalpy kJ/kg Evap. h_{fg}	Enthalpy kJ/kg Sat. Vapor h_g	Entropy kJ/kg · K Sat. Liquid s_f	Entropy kJ/kg · K Sat. Vapor s_g	Press. bar
0.10	45.81	1.0102	14.674	191.82	2437.9	191.83	2392.8	2584.7	0.6493	8.1502	0.10

$$x_3 = \frac{2459.3 - 191.83}{2584.7 - 191.83} = 0.9476$$

$$T_3 = T_{sat} = 45.81°C$$

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HOMEWORK

T[°C]-v[m3/kg] diagram for the 3 states??

① $P_1 = 15$ bar, $T_1 = 240$ °C → SHV

v_1 ✓ h_1 ✓

$T_1 P_1$

$P_2, h_2 = h_1$ → ?

$P_2 = 10$ bar

$T_2 = ?$

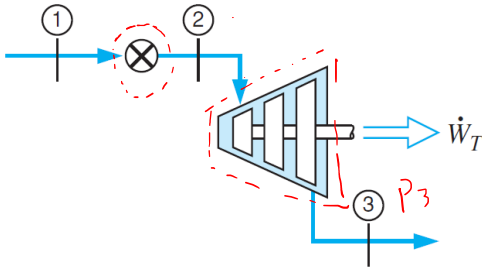
$h_2 = ?$

v_2, T_2

Balance en ⊗

$\frac{dE}{dt} = h_1 - h_2 + \cancel{Q} - \cancel{W}$

$h_2 = h_1$



③ $P_3 = 10$ kPa $= 0.1$ bar

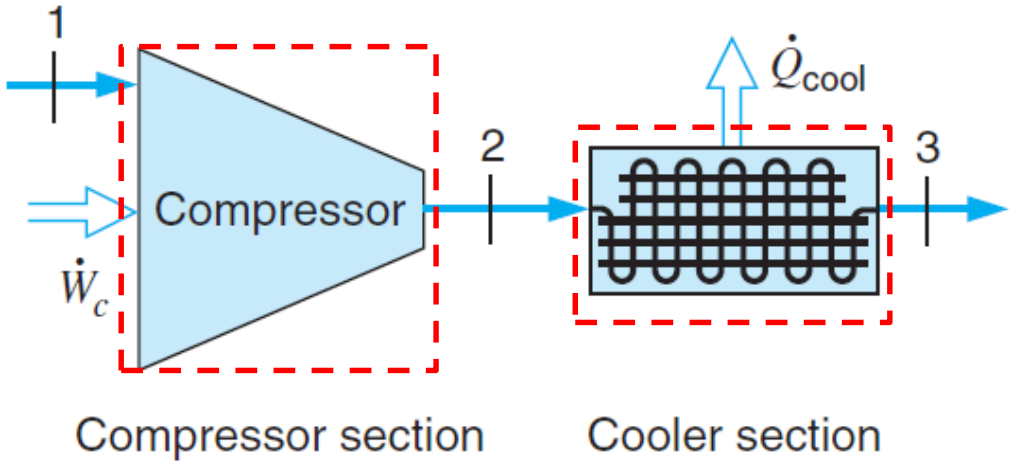
$x_3 = 0.948$

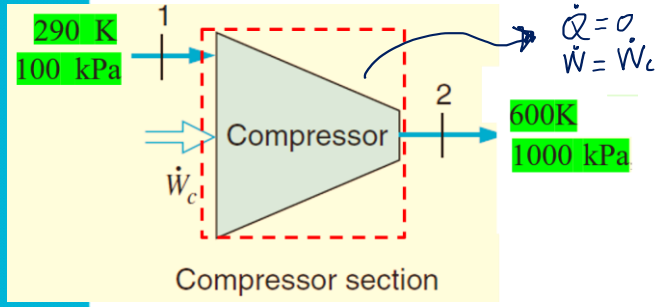
$T_3 = 45.81$ °C

$v_3 = v_f + x_3 (v_g - v_f)$ ✓

Example 5.3 (compressor):

A factory generates compressed air (ideal gas) from 100 kPa, 290 K (state1) by compression to 1000 kPa, 600K (state 2), after which it cools in a constant pressure cooler to 300K (state 3), (see figure below). Find the specific compressor work (kJ/kg) and the specific heat in the cooler (kJ/kg). This is a steady-flow process since there is no change with time at any point. The compressor is adiabatic and the kinetic and potential terms can be neglected in all process.





$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + z_{in}g + \frac{V_{in}^2}{2}\right) - \sum \dot{m}_{out}\left(h_{out} + z_{out}g + \frac{V_{out}^2}{2}\right)$$

Stream 1 Stream 2

State 1:
P = 100kPa
T = 290K
$h_1 = 290.16\ kJ/kg$

$$\dot{W} = \dot{m}(h_1 - h_2)$$

$$\frac{\dot{W}}{\dot{m}} = (290.16 - 607.02)$$

State 2:
P = 1000kPa
T = 600K
$h_2 = 607.02\ kJ/kg$

TABLE A-22

Ideal Gas Properties of Air

	T(K), h and u(kJ/kg), s° (kJ/kg · K)				
				when $\Delta s = 0^1$	
T	h	u	s°	p_r	v_r
290	290.16	206.91	1.66802	1.2311	676.1
600	607.02	434.78	2.40902	16.28	105.8

$$\frac{\dot{W}}{\dot{m}} = -316.86\ kJ/kg$$

Entre? Sale?

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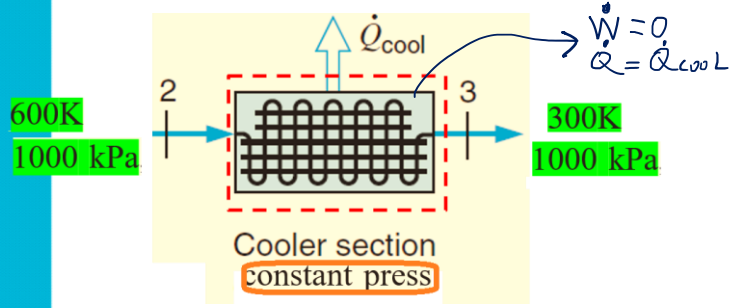


$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \sum \dot{m}_{in}\left(h_{in} + z_{in}g + \frac{V_{in}^2}{2}\right) - \sum \dot{m}_{out}\left(h_{out} + z_{out}g + \frac{V_{out}^2}{2}\right)$$

Stream 2 | Stream 3

State 2:
P = 1000kPa
T = 600K
$h_2 = 607.02\ kJ/kg$

$$\dot{Q} = \dot{m}(h_3 - h_2)$$

$$\frac{\dot{Q}}{\dot{m}} = (300.19 - 607.02)$$

State 3:
P = 1000kPa
T = 300K
$h_3 = 300.19\ kJ/kg$

TABLE A-22

Ideal Gas Properties of Air

	T(K), h and u(kJ/kg), s^o (kJ/kg · K)			when $\Delta s = 0^1$	
T	h	u	s^o	p_r	v_r
290	290.16	206.91	1.66802	1.2311	676.1
600	607.02	434.78	2.40902	16.28	105.8

$$\frac{\dot{Q}}{\dot{m}} = -306.83\ kJ/kg$$

Entra? sale?

TABLE A-22

Ideal Gas Properties of Air

T	h	u	s^o
300	300.19	214.07	1.70203

EXAMPLE: POWER CYCLE

$P_4 = P_1$ $P_2 = P_3$

Steady-state operating data for a simple steam power plant are provided below. Stray heat transfer and kinetic and potential energy effects can be ignored. The cycle produces $\dot{W}_{NET}$ = 1500 kW of power. The condenser has adiabatic external walls. For the operational conditions in the figure, determine:

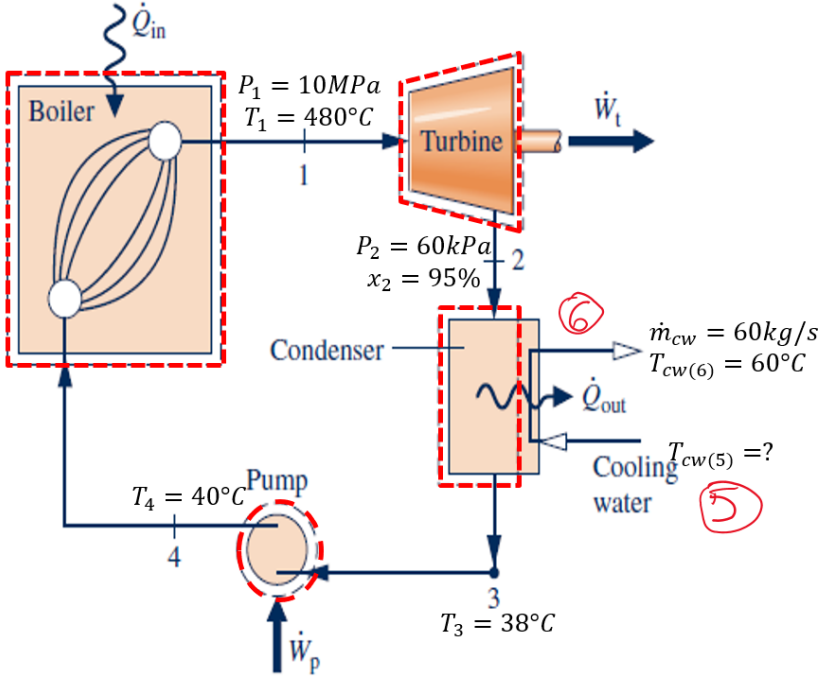
a) The heat transferred to the cooling water in the condenser ($\frac{\dot{Q}_{out}}{\dot{m}}$) and the boiler ($\frac{\dot{Q}_{in}}{\dot{m}}$) [kJ/kg]

b) The power for the adiabatic turbine ($\frac{\dot{W}_T}{\dot{m}}$) and the adiabatic pump ($\frac{\dot{W}_P}{\dot{m}}$) [kJ/kg]

c) Mass flowrate of water (kg/s) in the cycle and the inlet temperature ($T_{cw(i)}$) of cooling water (°C)

d) Thermal efficiency

e) Sketch a T (°C) -v (m3/kg) diagram for the cycle



F) Comprueba la 1ra ley tomando el H2o que recorre el ciclo como sistema (masa de control)

ASSUME THERE IS NO PRESSURE DROP IN BOILER AND CONDENSER

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Defining state 1

$P_1 = 100\ bar$
$T_1 = 480\ °C$
$h_1 = 3321.4\ kJ/kg$

TABLE A-4

(*Continued*)

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	$p = 100$ bar $= 10.0$ MPa ($T_{sat} = 311.06°C$)			
480	0.03160	3005.4	3321.4	6.5282

Defining state 2 and 3

$P_2 = P_3 = 0.6\ bar$
$T_2 = 85.94\ °C$
$h_2 = 359.86 + 0.95 \times 2293.6$
$h_2 = 2538.78\ kJ/kg$
$T_3 = 38\ °C$
$h_3 = 159.21\ kJ/kg$ (aprox a saturado)

TABLE A-3

Properties of Saturated Water (Liquid–Vapor): Pressure Table

Pressure Conversions: 1 bar = 0.1 MPa = 10^2 kPa

Press. bar	Temp. °C	Specific Volume m³/kg: Sat. Liquid $v_f \times 10^3$	Specific Volume m³/kg: Sat. Vapor v_g	Internal Energy kJ/kg: Sat. Liquid u_f	Internal Energy kJ/kg: Sat. Vapor u_g	Enthalpy kJ/kg: Sat. Liquid h_f	Enthalpy kJ/kg: Evap. h_{fg}	Enthalpy kJ/kg: Sat. Vapor h_g
0.60	85.94	1.0331	2.732	359.79	2489.6	359.86	2293.6	2653.5

Defining state 4

$P_4 = P_1 = 100\ bar$
$T_4 = 40°C$
$h_4 = 176.38\ kJ/kg$

TABLE A-5

Properties of Compressed Liquid Water

T °C	v × 10³ m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	$p = 100$ bar $= 10.0$ MPa ($T_{sat} = 311.06°C$)			
40	1.0034	166.35	176.38	.5686

a) Heat in condenser

$$\frac{dE}{dt} = \dot{Q}_{out} - \dot{W} + \dot{m}(h_2 - h_3)$$

$$\dot{Q}_{out} = \dot{m}(159.21 - 2538.78)$$

$$\frac{\dot{Q}_{out}}{\dot{m}} = -2379.57\ \frac{kJ}{kg}$$

a) Heat in boiler

$$\frac{dE}{dt} = \dot{Q}_{in} - \dot{W} + \dot{m}(h_4 - h_1)$$

$$\dot{Q}_{in} = \dot{m}(3321.4 - 176.38)$$

$$\frac{\dot{Q}_{in}}{\dot{m}} = 3145.02\ \frac{kJ}{kg}$$

TABLE A-2

Properties of Saturated Water (Liquid–Vapor): Temperature Table

Pressure Conversions: 1 bar = 0.1 MPa = 10^2 kPa

Temp. °C	Press. bar	Specific Volume m³/kg: Sat. Liquid $v_f \times 10^3$	Specific Volume m³/kg: Sat. Vapor v_g	Internal Energy kJ/kg: Sat. Liquid u_f	Internal Energy kJ/kg: Sat. Vapor u_g	Enthalpy kJ/kg: Sat. Liquid h_f	Enthalpy kJ/kg: Evap. h_{fg}
38	0.06632	1.0071	21.602	159.20	2427.4	159.21	2411.5

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TABLE A-2

(Continued)

H_2O

Temp. °C	Press. bar	Specific Volume m³/kg		Internal Energy kJ/kg		Enthalpy kJ/kg		
		Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g
60	.1994	1.0172	7.671	251.11	2456.6	251.13	2358.5	2609.6

h6 **(salida agua enfriamiento)**

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b) Power in turbine

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}(h_1 - h_2)$$

$$\dot{W} = \dot{m}(3321.4 - 2538.78)$$

$$\frac{\dot{W}}{\dot{m}} = \mathbf{782.62}\,\frac{\boldsymbol{kJ}}{\boldsymbol{kg}}$$

b) Power in pump

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}(h_3 - h_4)$$

$$\dot{W} = \dot{m}(159.21 - 176.38)$$

$$\frac{\dot{W}}{\dot{m}} = \mathbf{-17.17}\,\frac{\boldsymbol{kJ}}{\boldsymbol{kg}}$$

c) Mass flowrate of water

$$\dot{W} = \dot{W}_t + \dot{W}_p$$

$$1500 = \dot{m}(782.62 - 17.17)$$

$$\dot{\boldsymbol{m}} = \mathbf{1.9596}\ \boldsymbol{kg/s}$$

c) Inlet temperature of cooling water

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}_{cw}(h_5 - h_6)$$

$$0 = 2379.57 \times 1.9596 + 60 \times (h_5 - 251.13)$$

$$h_5 = 173.4132\ kJ/kg$$

$$\boldsymbol{T_5 = 41.3992\ °C}$$

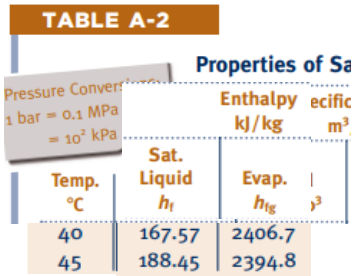
TABLE A-2

Properties of Sa

Pressure Conver
1 bar = 0.1 MPa
= 10² kPa

Temp. °C	Sat. Liquid h_f	Enthalpy kJ/kg Evap. h_{fg}	ecific m³
40	167.57	2406.7	
45	188.45	2394.8	

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d) Thermal efficiency of cycle

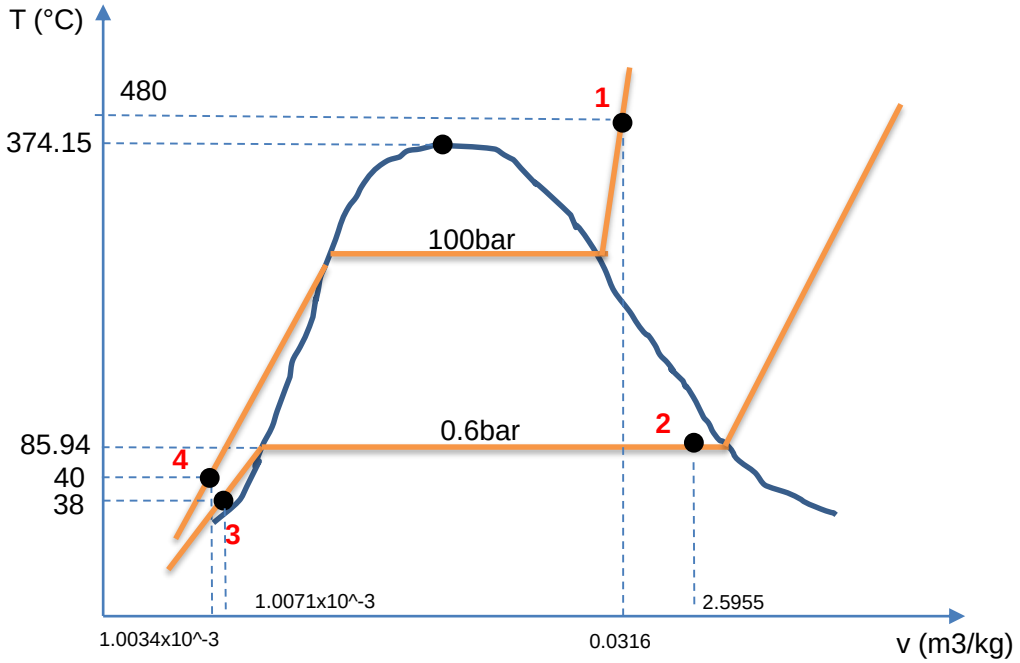
$$\eta_{cyc} = \frac{\dot{W}_{net}}{\dot{Q}_{in}}$$

$$\eta_{cyc} = \frac{\dot{W}_{turb} + \dot{W}_{pump}}{\dot{Q}_{in}}$$

$$\eta_{cyc} = \frac{\mathbf{782.62} + (-17.17)}{3145.02}$$

$$\boldsymbol{\eta_{cyc} = 0.2434 = 24.34\%}$$

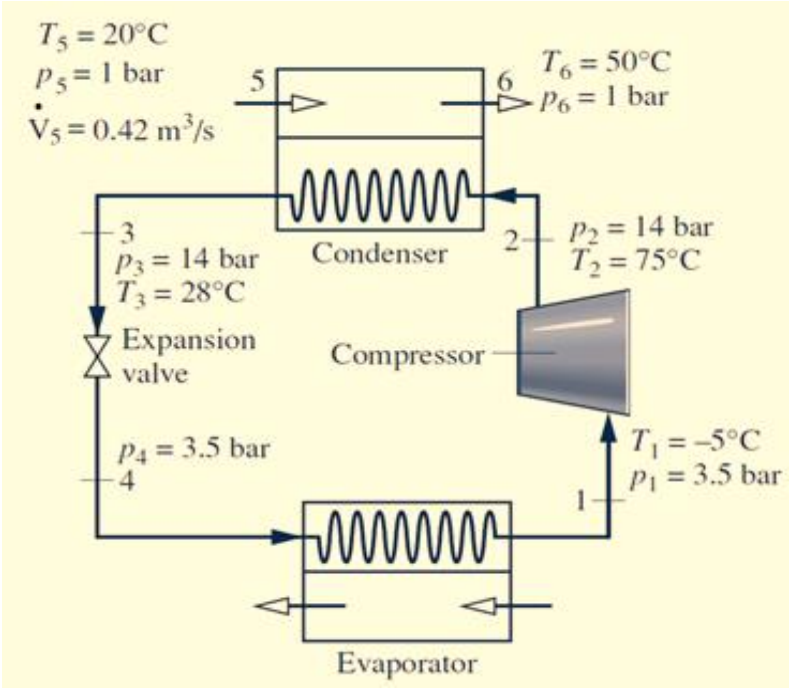
e) T-v diagram



EXAMPLE: HEAT PUMP

A heat pump and its components are shown in the figure and warms the environment inside a house. Assuming a steady state, the refrigerant 22 enters the compressor at -5° C and 3.5 bar and is compressed adiabatically to 75 °C and 14 bar. After leaving the compressor, the refrigerant goes through an isolated condenser and leaves at 28 °C and 14 bar. The expansion in the valve is adiabatically to 3.5 bar. The air inside the house enters the condenser at 20 °C, 1 bar and with a volumetric flowrate of 0.42 m3/s; and leaves at 50 °C and 1 bar. Ignoring the effects of kinetic, potential energy and assuming the air behaves as ideal gas (M = 28.97 kg/kmol, cp = 1.005 kJ/kg.K), find:

A) The mass flowrate of air and the refrigerant (kg/s)
B) The heat flow transferred from the refrigerant to air in the condenser (kW)
C) The power required by the compressor (kW)
D) The heat transferred by the air to the refrigerant in the evaporator (kW)
E) Sketch the T (°C) – v (m3/kg) diagram for the refrigerant



The condenser and evaporator have adiabatic external walls

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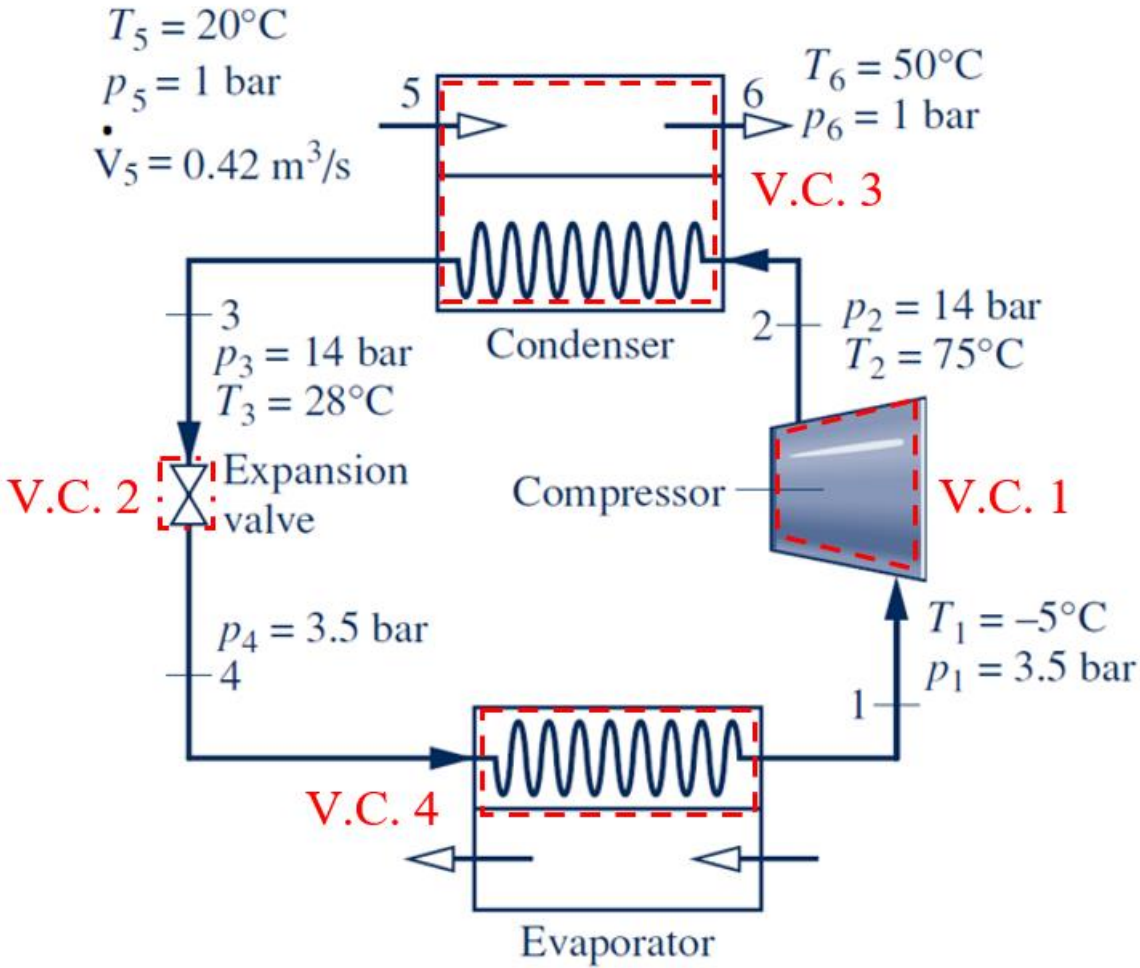
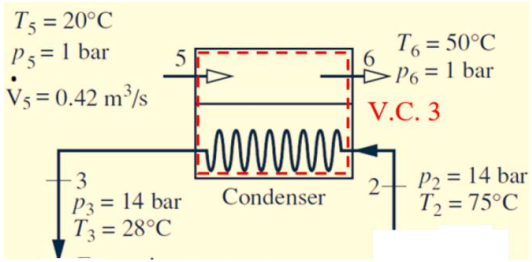

$T_5 = 20°C$
$p_5 = 1$ bar
$\dot{V}_5 = 0.42$ m^3/s
5
6
$T_6 = 50°C$
$p_6 = 1$ bar
V.C. 3
Condenser
3
$p_3 = 14$ bar
$T_3 = 28°C$
2
$p_2 = 14$ bar
$T_2 = 75°C$
V.C. 2
Expansion valve
Compressor
V.C. 1
$p_4 = 3.5$ bar
4
$T_1 = -5°C$
$p_1 = 3.5$ bar
1
V.C. 4
Evaporator

a) Mass flowrate of air

$$\dot{m}_a = \frac{P_5 \dot{V}_5}{RT_5} = \frac{100 \times 0.42}{(8.314/28.97) \times 293} = 0.4995 \ \frac{kg}{s}$$



Mass flowrate of refrigerant: energy balance in the condenser (V.C. 3)

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + [\dot{m}_R h_2 + \dot{m}_a h_5] - [\dot{m}_R h_3 + \dot{m}_a h_6]$$

$$0 = \dot{m}_R [h_2 - h_3] - \dot{m}_a [h_5 - h_6]$$

$$0 = \dot{m}_R [h_2 - h_3] - \dot{m}_a c_p [T_5 - T_6]$$

TABLE A-9

(*Continued*)

T °C	v m^3/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	p = 14.0 bar = 1.40 MPa (T_{sat} = 36.29°C)			
70	0.02029	261.60	290.01	0.9703
80	0.02125	268.60	298.34	0.9942

$$h_2 = \frac{290.01 + 298.34}{2}$$

$$h_2 = 294.175 \ \frac{kJ}{kg}$$

TABLE A-7

Pressure Conversions: 1 bar = 0.1 MPa = 10^2 kPa

Properties of Saturated Refrige

Temp. °C	Press. bar	Enthalpy kJ/kg: Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g
28	11.313	79.05	179.37	258.43

h_3 is approximated from temperature table.

$$h_3 \approx h_{f@28°C} = 79.05 \ \frac{kJ}{kg}$$

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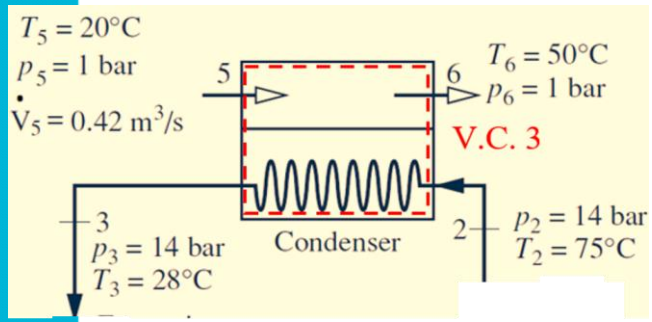
Mass flowrate of refrigerant: energy balance in the condenser (V.C. 3)

$$0 = \dot{m}_R[h_2 - h_3] + \dot{m}_a c_p[T_5 - T_6]$$

$$0 = \dot{m}_R[294.175 - 79.05] + 0.4995 \times 1.005 \times [20 - 50]$$

$$\boldsymbol{\dot{m}_R = 0.07000 \frac{kg}{s}}$$

b) Heat flow (kW) $[\dot{Q}_{\mathrm{H}}]$



$$\frac{dE}{dt} = \dot{Q}_{\mathrm{H}} - \dot{W} + \dot{m}_R[h_2 - h_3]$$

$$0 = \dot{Q}_{\mathrm{H}} - 0 + 0.07000 \times [294.175 - 79.05]$$

$$\boldsymbol{\dot{Q}_{\mathrm{H}} = -15.0594\ kW}$$

c) Power required by the compressor (kW): energy balance in V.C. 1

TABLE A-9

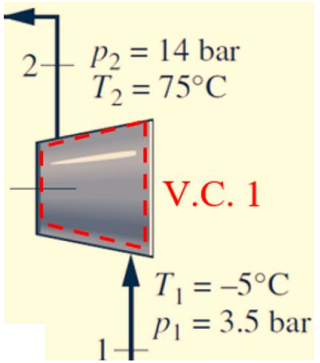
(Continued)

T °C	v m³/kg	u kJ/kg	h kJ/kg	s kJ/kg · K
	p = 3.5 bar = 0.35 MPa (T_{sat} = −10.39°C)			
−5	0.06789	225.99	249.75	0.9572

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}_R[h_1 - h_2]$$

$$0 = 0 - \dot{W} + 0.07000 \times [249.75 - 294.175]$$

$$\dot{W} = -3.1099\ kW$$

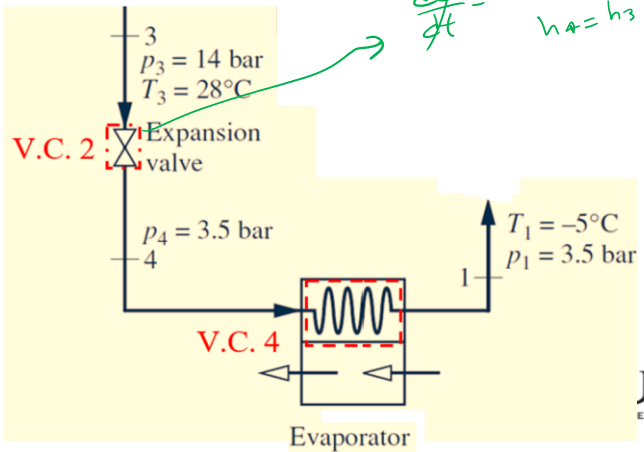


d) Heat transferred in the evaporator (kW): energy balance in V.C. 4

$$\frac{dE}{dt} = \dot{Q}_L - \dot{W} + \dot{m}_R[h_4 - h_1]$$

$$0 = \dot{Q}_L - 0 + 0.07000 \times [79.05 - 249.75]$$

$$\dot{Q}_L = 11.9495\ kW$$



HOMEWORK:

$COP_H = BHP = ???$

Compruebe la 1ra ley tomando el H_2O que recorre el ciclo como sistema

Thanks!!

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