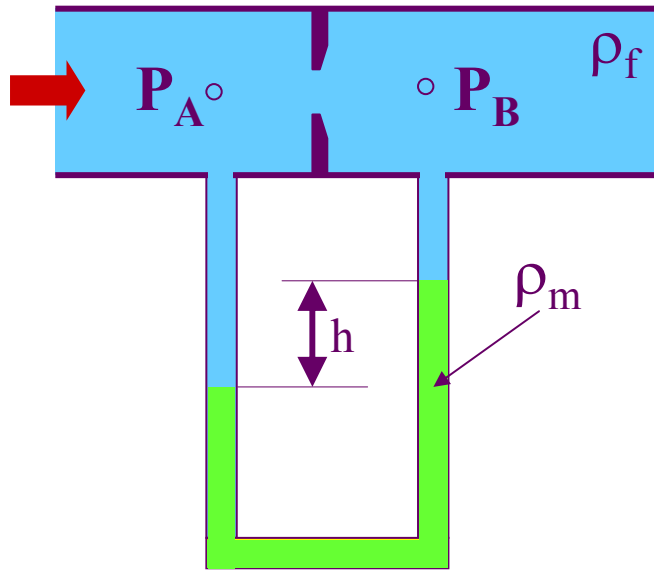


Aplicações da Eq. de Bernoulli

Ex. 5.42 MEDIÇÃO DA PRESSÃO: MANÔMETRO DIFERENCIAL – CASO PRÁTICO



• Determine $P_A - P_B$ em função da altura h e das densidades dos fluidos.

Resp.: $P_A - P_B = (\rho_m - \rho_f) \cdot g \cdot h$

Pergunta: se os fluidos forem água e mercúrio, como fica a expressão?

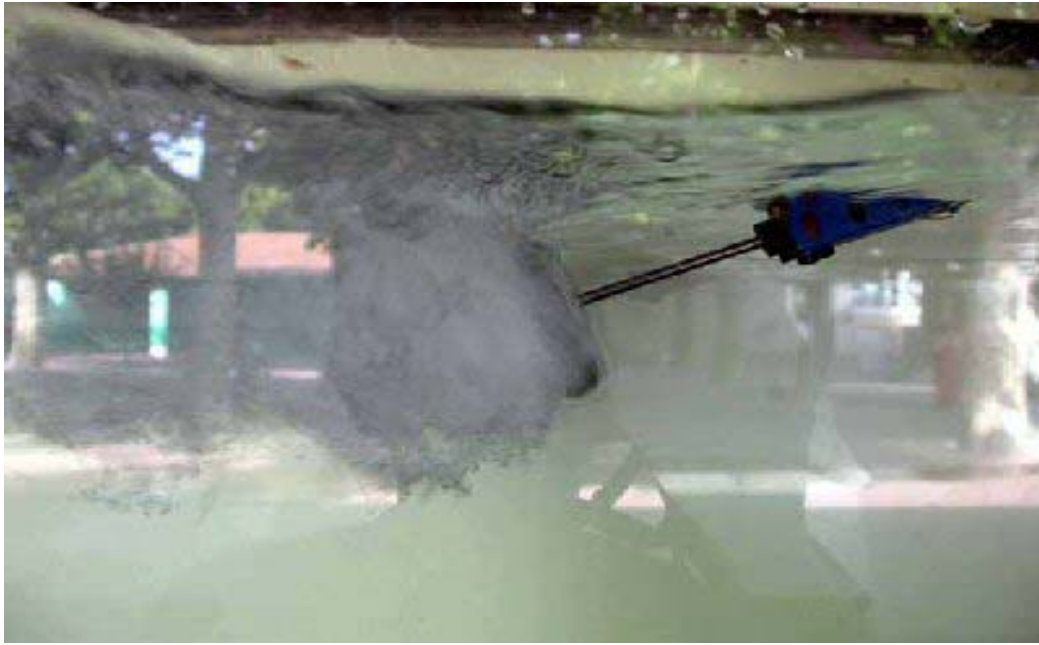
Resp.: $P_A - P_B = (13600 - 1000) \cdot g \cdot h$

• Pergunta: e se os fluidos forem ar e água?

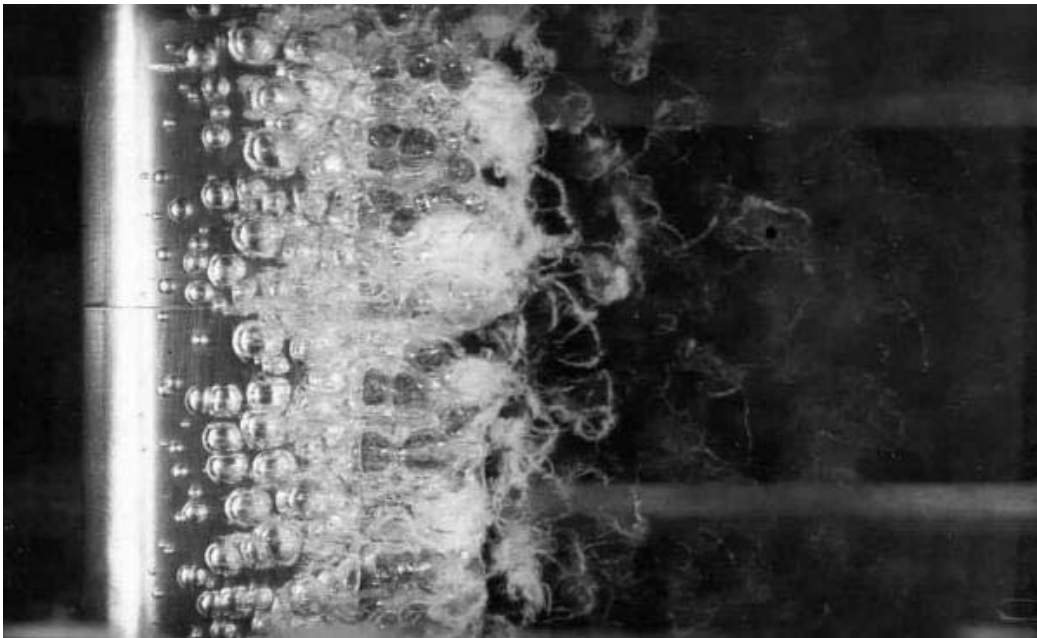
Resp.: $P_A - P_B = (1000 - 1.2)gh$ como $\rho_{liq} \gg \rho_{gas} \rightarrow P_A - P_B \approx \rho_m gh$

• Um engano freqüente dos alunos:
aplicar a relação acima quando realizam medidas com dois líquidos.

The cavitation phenomenon

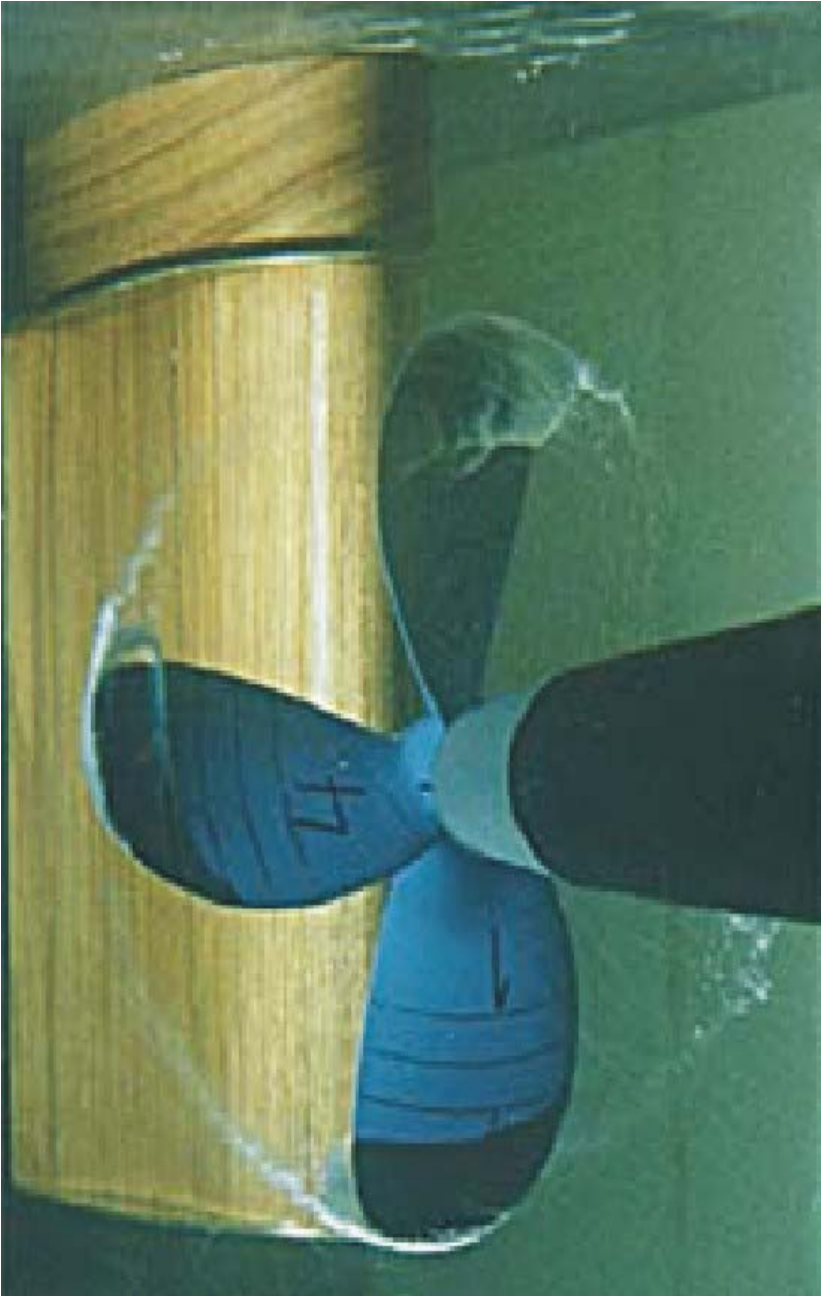


The cavitation phenomenon is characterized by the formation and quick growth of vapor bubbles in the presence of a depression, followed by a violent implosion. Such an implosion, often supersonic, can generate a spherical shock wave in the liquid developing pressures higher than the solid material yield stress.

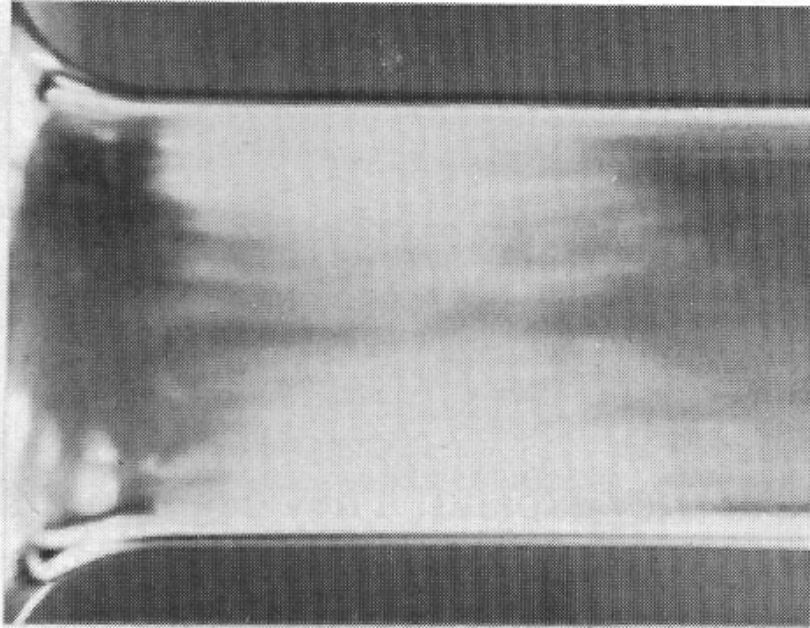


Erosão causada pela cavitação

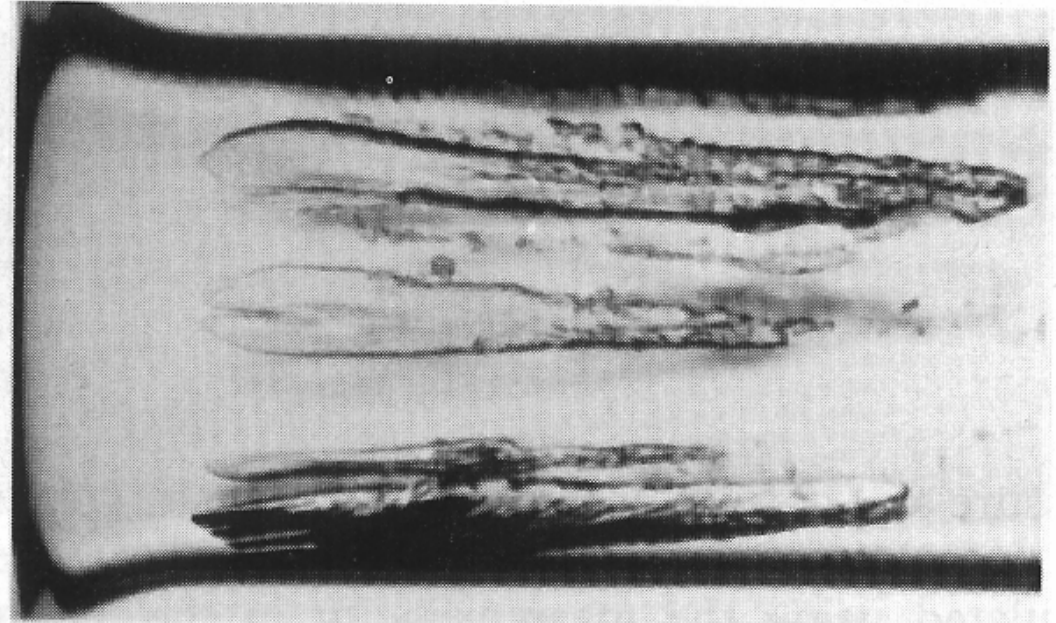




FIXED CAVITATION IN A NOZZLE



incandescent light,
exposure time 1/30 sec



strobe light,
exposure time 5 μ s

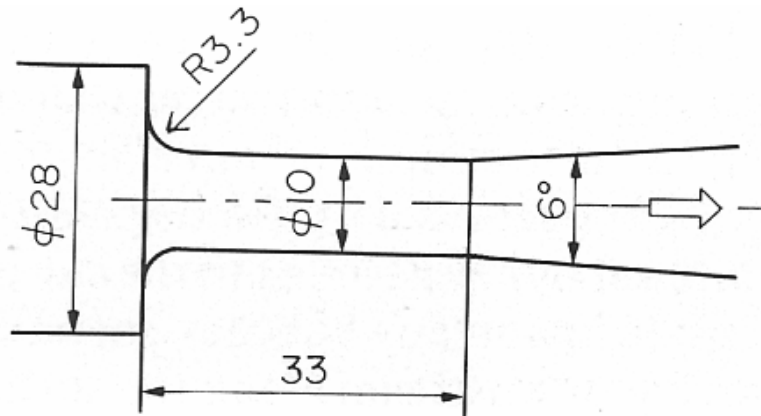


Fig. 204. Cavitation in Venturi nozzle (water, 15 m/s, throat diameter 10 mm, $Re = 1.4 \times 10^5$).

5-45 Água a 30 °C escoam de um tanque grande através de um bocal convergente-divergente, Fig. P5-45. O bocal descarrega na atmosfera e tem um diâmetro $d_s = 2,5d_G$, onde d_G é o diâmetro da garganta do bocal, que é o mínimo diâmetro. Assumindo um escoamento ideal, isto é, sem perdas, determine a altura da água ΔZ_a , na qual cavitação (vaporização da água) vai começar a ocorrer no bocal.

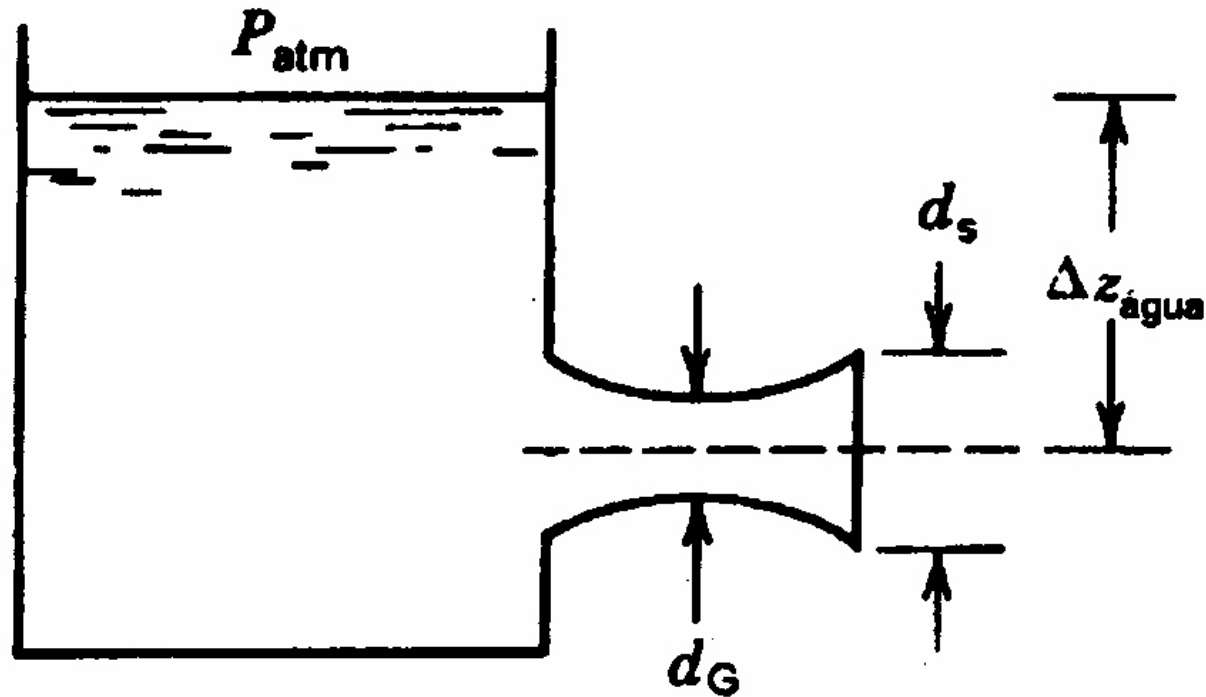
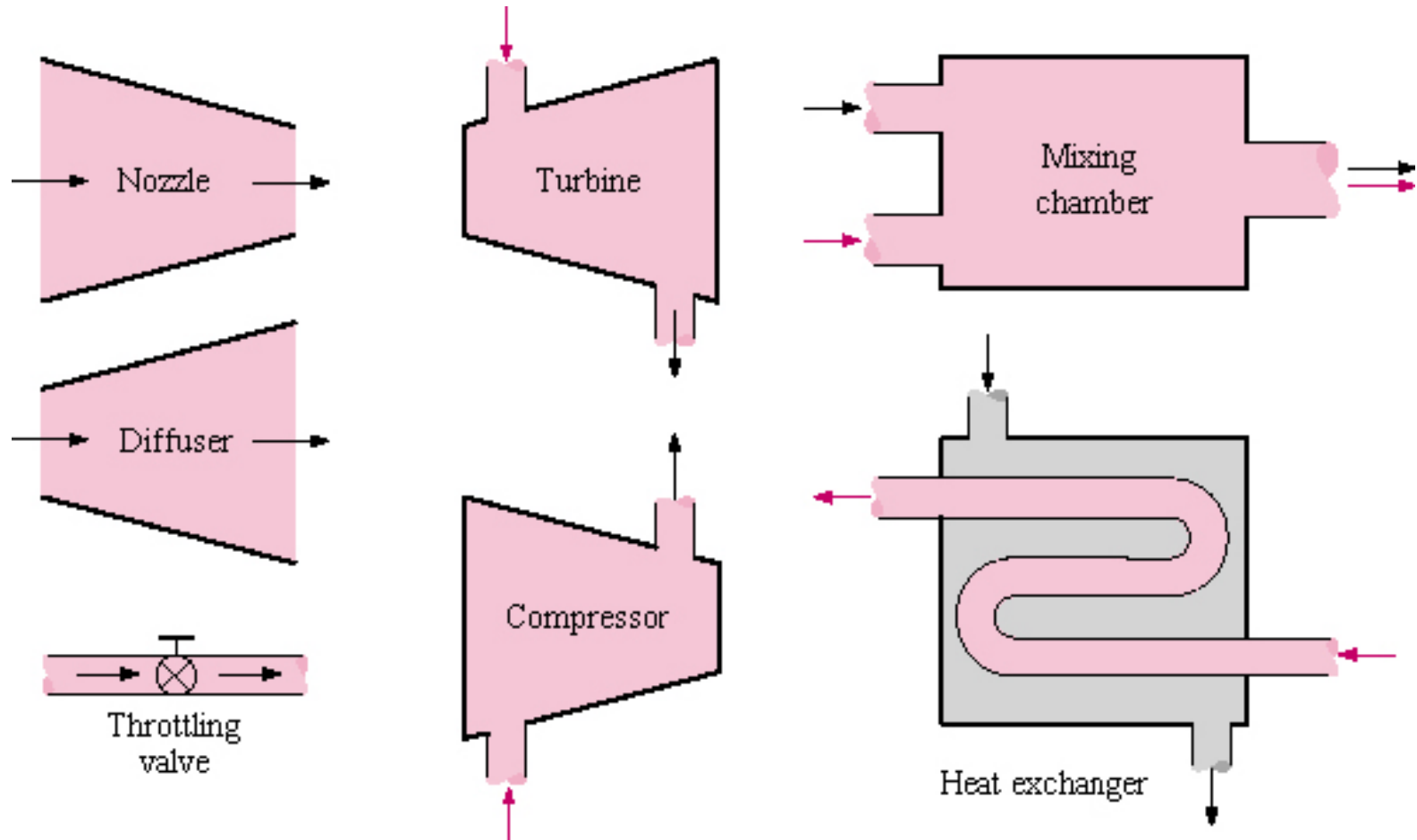


Figura P5-45 Bocal convergente-divergente

Veja solução em 'exercícios resolvidos'

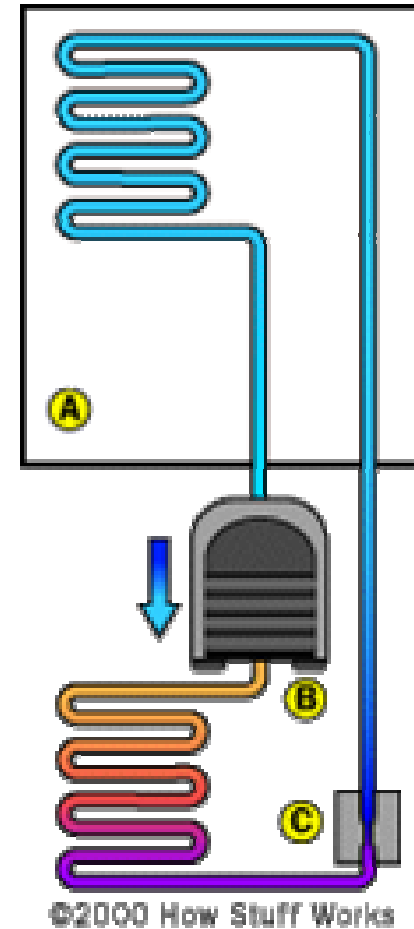
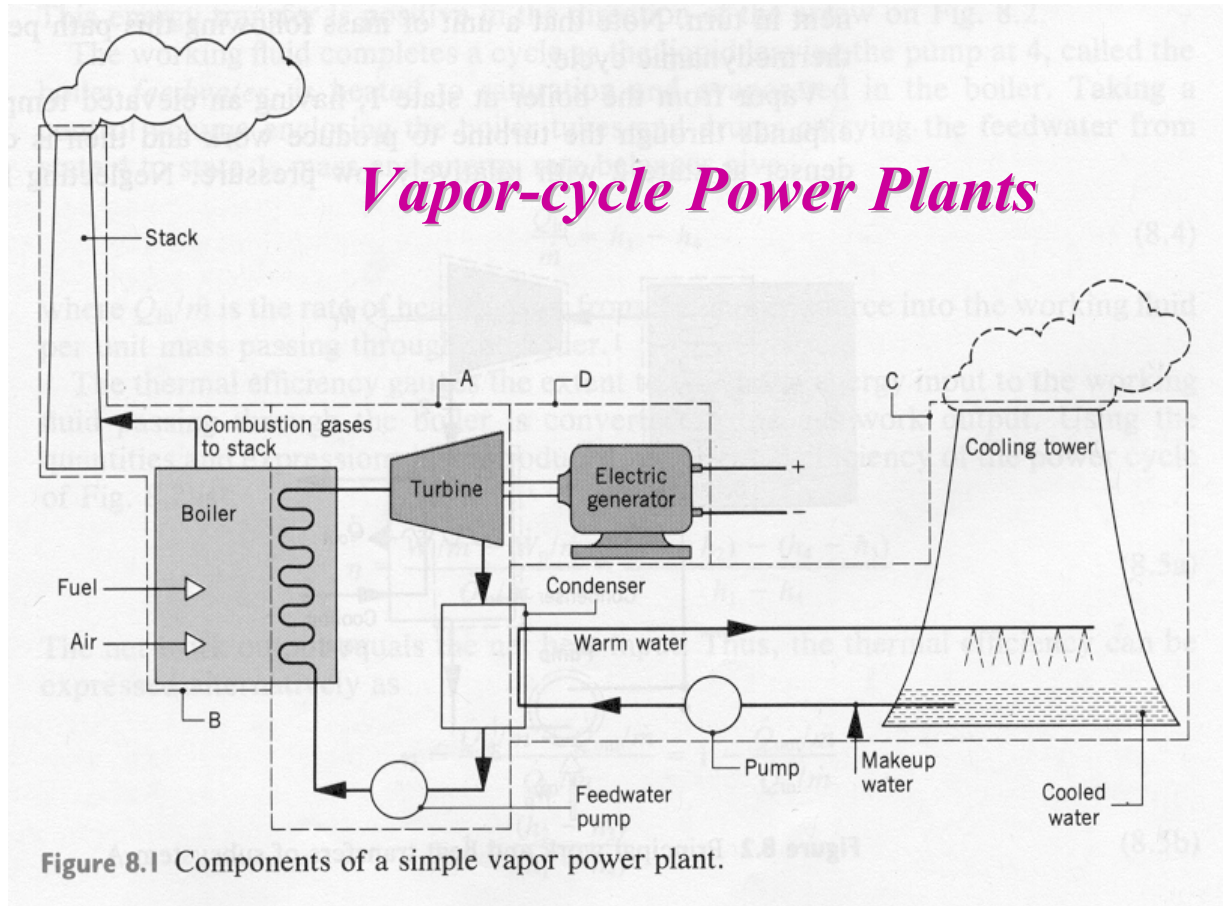
Components of Thermodynamic Cycles

- What can be the components of thermodynamic cycles?
- Turbines, valves, compressors, pumps, heat exchangers (evaporators, condensers), mixers,



Where, Exactly, Are They Used For?

Refrigeration cycle



- A** Inside the refrigerator
- B** Compressor
- C** Expansion valve

Equação da Energia: Regime Permanente

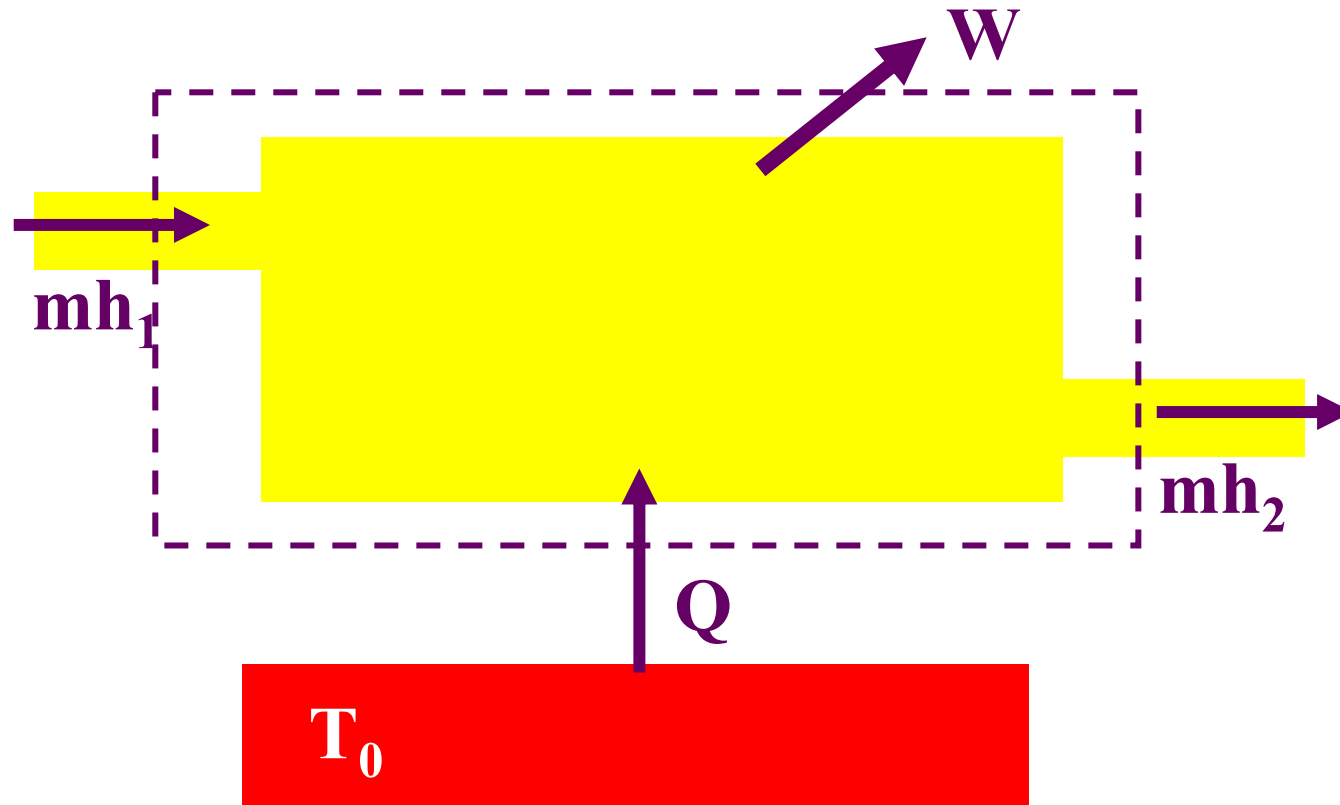
$$- \sum \left[\left(u + \frac{V_I^2}{2} + gz + \frac{P}{\rho} \right) \dot{m} \right]_{\text{IN}} + \sum \left[\left(u + \frac{V_I^2}{2} + gz + \frac{P}{\rho} \right) \dot{m} \right]_{\text{OUT}} = \dot{Q} - \dot{W}_{\text{shaft}}$$

- Considere o V.C. com duas portas (uma entrada / uma saída)
- Expressando em função do calor e trabalho específicos (dividindo por \dot{m}),

$$\left(\frac{V_I^2}{2} + gz + \underbrace{u + \frac{P}{\rho}}_h \right)_{\text{OUT}} - \left(\frac{V_I^2}{2} + gz + \underbrace{u + \frac{P}{\rho}}_h \right)_{\text{IN}} = q - w_{\text{shaft}} \left[\frac{\text{Joules}}{\text{kg}} \right]$$

Representação Genérica dos Componentes de Ciclos Termodinâmicos

- Aplicação de um balanço de energia para dispositivos que operam com fluxo de energia (entalpia), produzem trabalho e trocam calor com um reservatório a T_0



Turbine

- A turbine is device in which work is produced by a gas passing over and through a set of blades fixed to a shaft which is free to rotate.

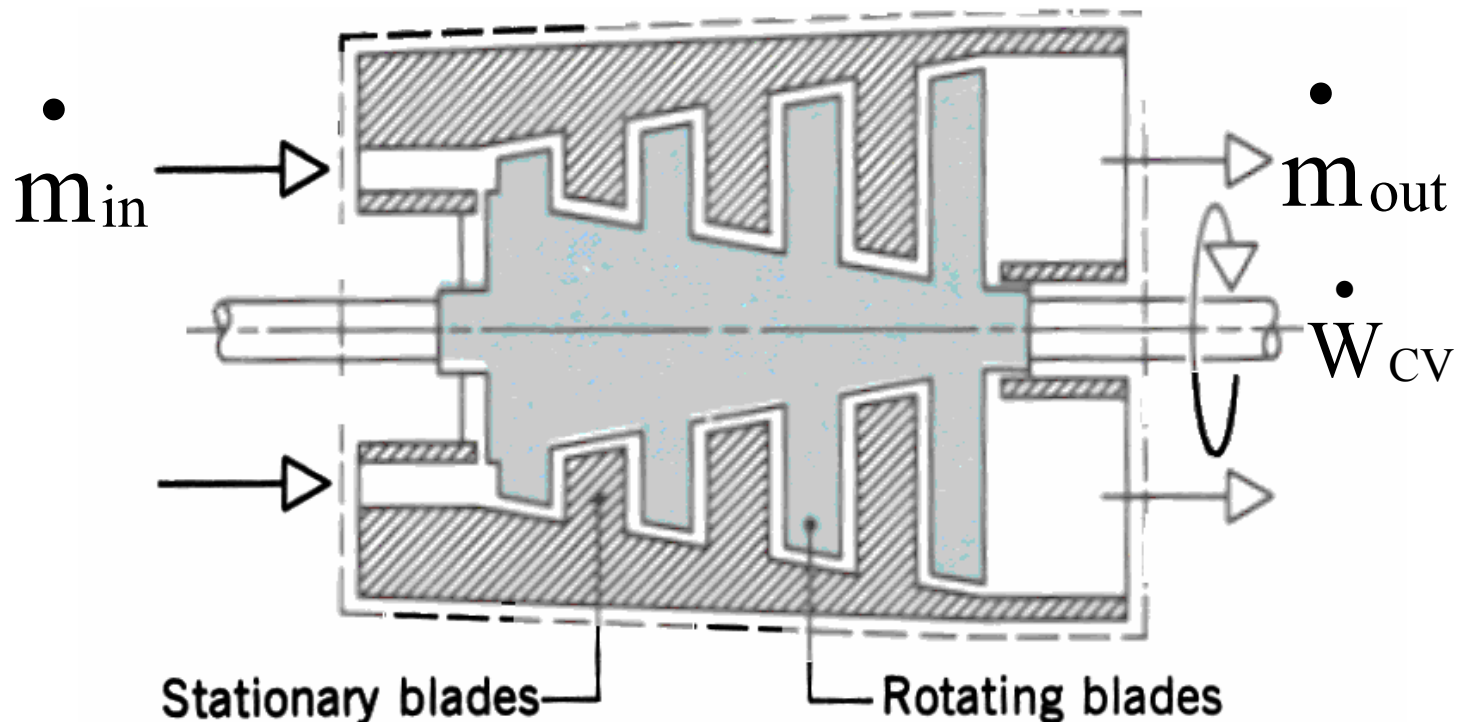
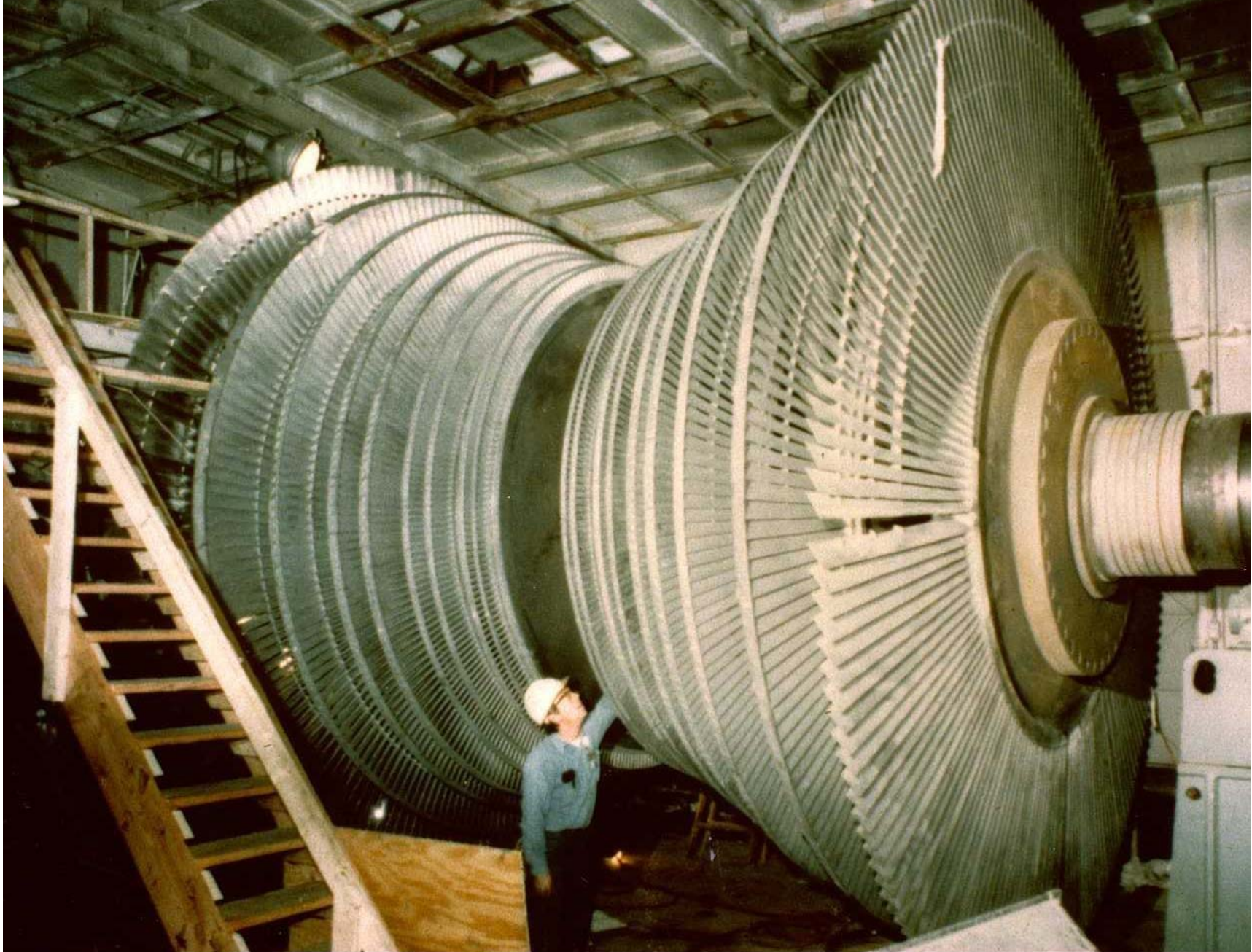


Figure 4.5 Schematic of an axial-flow turbine.



Overview of a steam turbine and main generator for a nuclear steam power station.
(Courtesy of Carolina Power and Light Company)



Low pressure steam turbine wheel.

(Courtesy of Carolina Power and Light Company)



Turbine bed for a steam power station.
(Courtesy of Carolina Power and Light Company)



View of power turbine blades in an industrial gas turbine unit.
(Courtesy of S. Laney)

Turbina a vapor ATP 4 - ABB

Output range up to 100 MW

Live steam conditions:

Temperature up to 540 °C

Pressure up to 140 Bara

Exhaust steam conditions:

Back-pressure:

3-16 Bara/300 °C

Condensing 0,03 - 0,25 Bar

Controlled extraction:

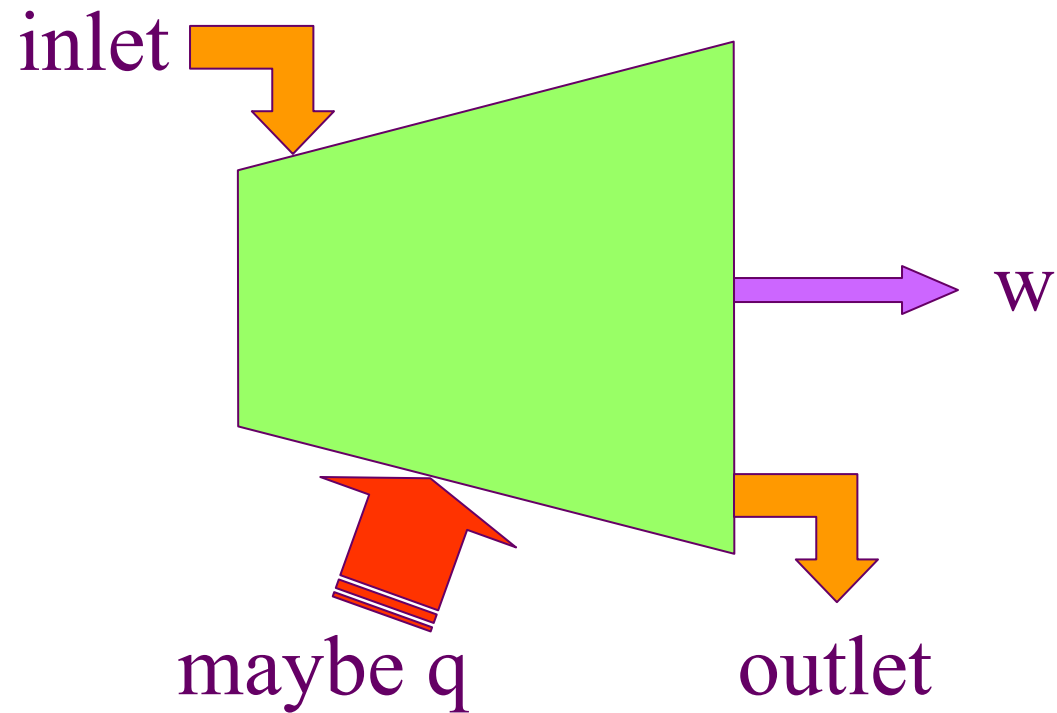
Pressure/Temperature 3-25

Bara/400 °C

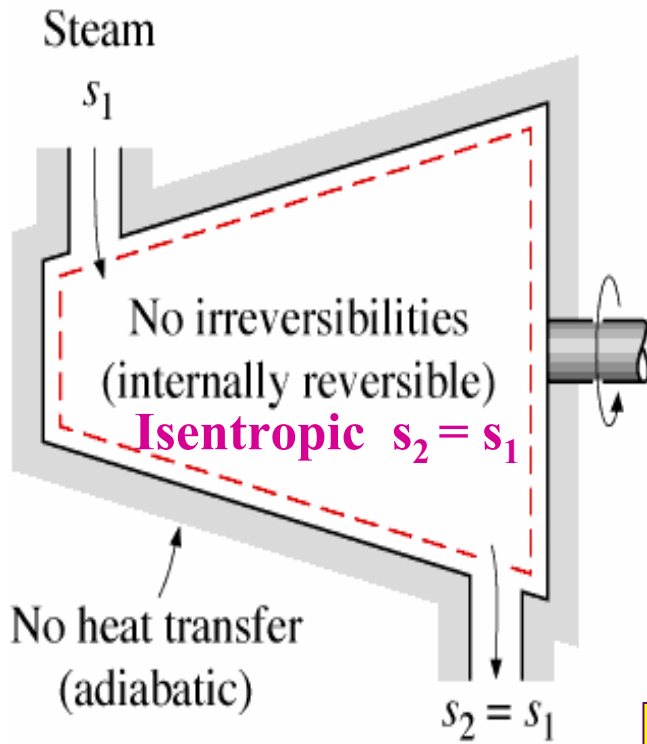


Turbines

- We will draw turbines like this:



TURBINES



$$q - w = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

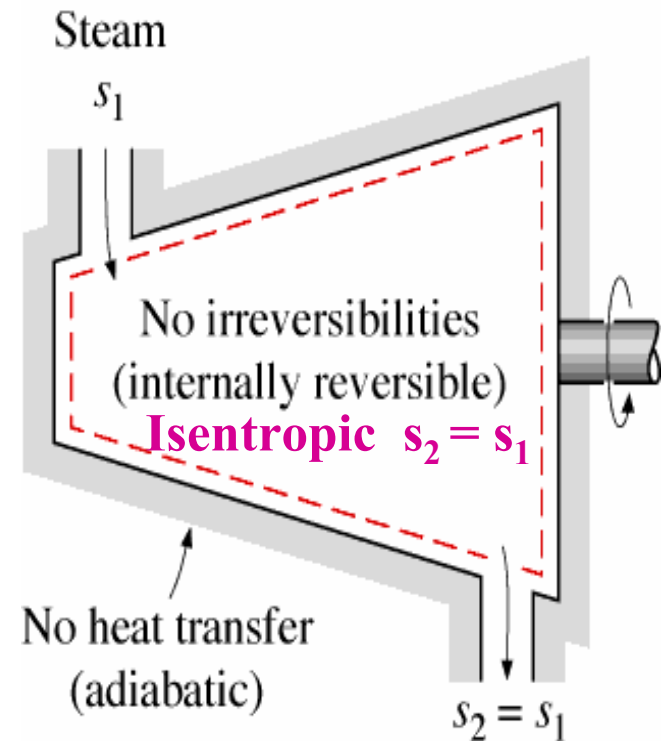
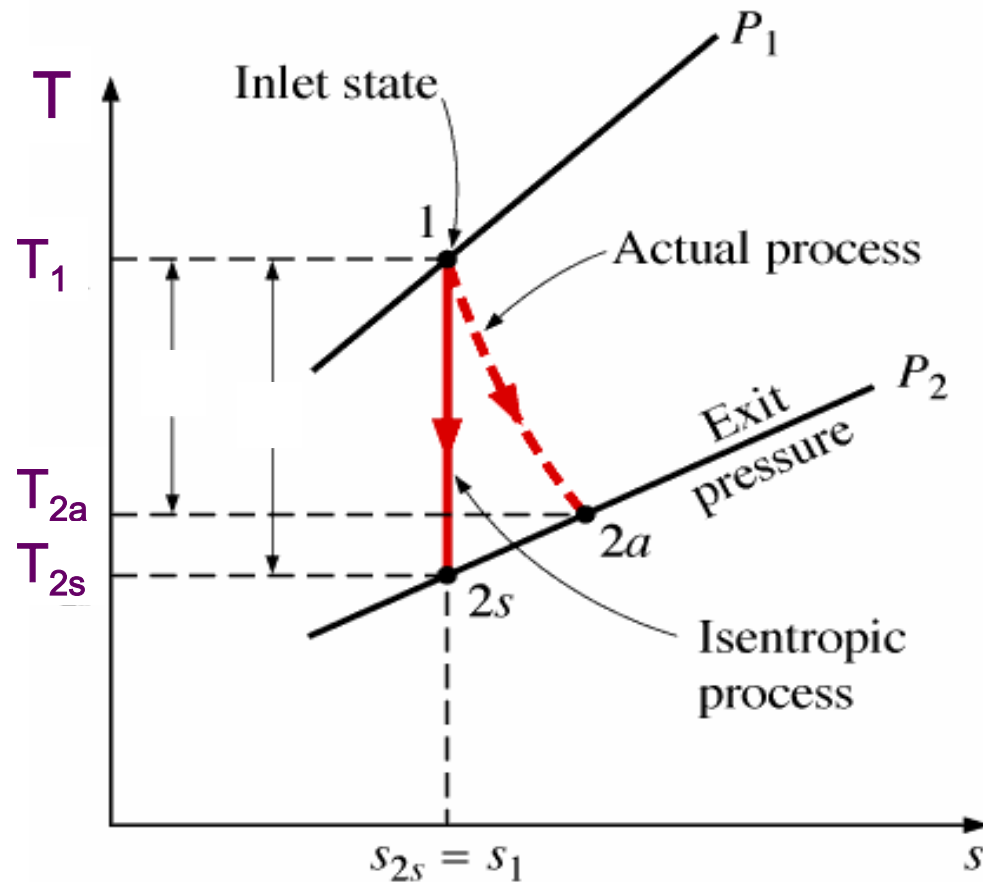
↓
**Almost
always
neglected**

↓
**Sometimes
neglected**

↓
**Almost
always
neglected**

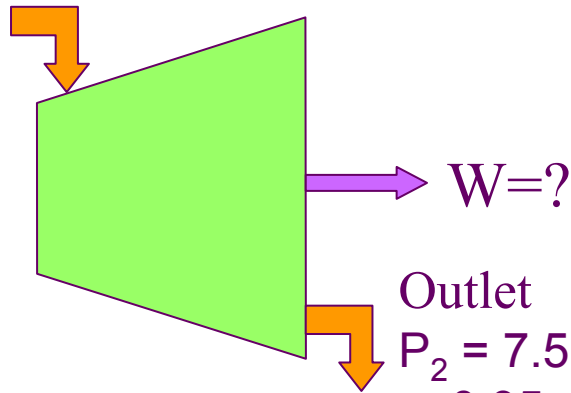
$$\mathbf{w}_{\text{eixo}} = (\mathbf{h}_1 - \mathbf{h}_2)$$

T-s Diagram of an Adiabatic Turbine



5-55 Calcule a potência produzida por uma turbina a vapor adiabática operando em regime permanente. O vapor entra a turbina a uma vazão mássica de 500 kg/s, uma pressão de 5,0 MPa, uma temperatura de 600 °C e a uma velocidade de 30 m/s. O vapor deixa a turbina a uma pressão de 7,5 kPa, $x = 95\%$ e a uma velocidade de 100 m/s.

Inlet
 $P_1 = 5 \text{ MPa}$
 $T_1 = 600^\circ\text{C}$
 $V_1 = 30 \text{ m/s}$



Outlet
 $P_2 = 7.5 \text{ kPa}$
 $x = 0.95$
 $V_2 = 100 \text{ m/s}$

$$q - w = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

h_1 superaquecido = 3666 kJ/kg
 h_2 saturado, 40.2°C
 $h_l = 168.8 \text{ kJ/kg}$ & $h_v = 2574.8 \text{ kJ/kg}$
 $\rightarrow h_2 = 2454.5 \text{ kJ/kg}$

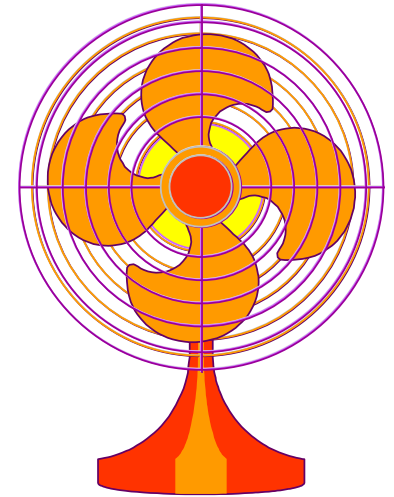
$$-w = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2}$$

1207 kJ/kg 1212 kJ/kg 4.5 kJ/kg

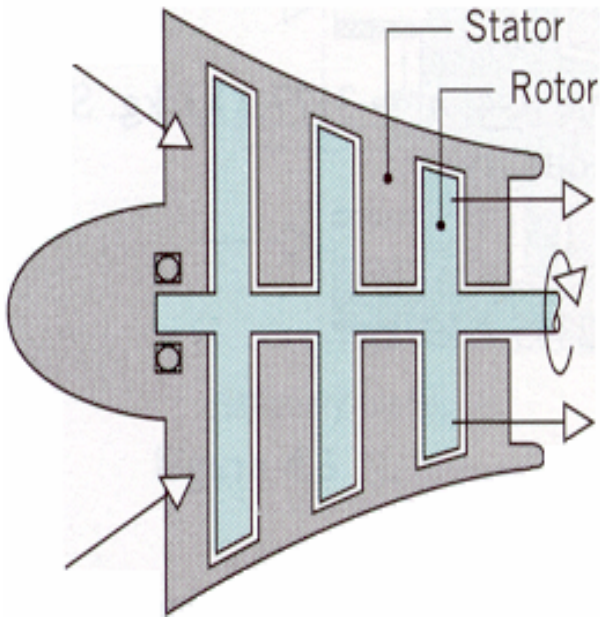
$$\rightarrow W = m \cdot w = 603 \text{ MW} !!!$$

Compressors, pumps, and fans

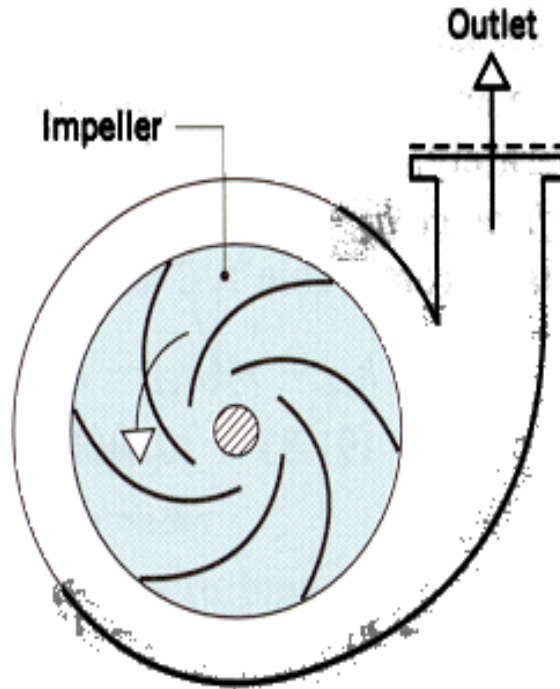
- Machines developed by engineers to make life easier, decrease world anxiety, and provide exciting engineering problems from the industrial revolution for students.
- Analysis proceeds the same as for turbines, although the signs may differ.
- **Compressor** - used to raise the pressure of a compressible fluid
- **Pump** - used to raise pressure of an incompressible fluid
- **Fan** - used to move large amounts of gas, but usually has a small pressure increase



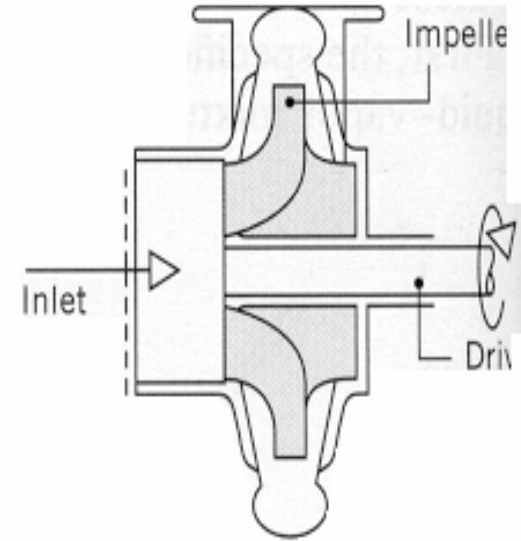
Compressors, pumps, and fans



Compressor



**Side view
of pump**



**End view
of pump**

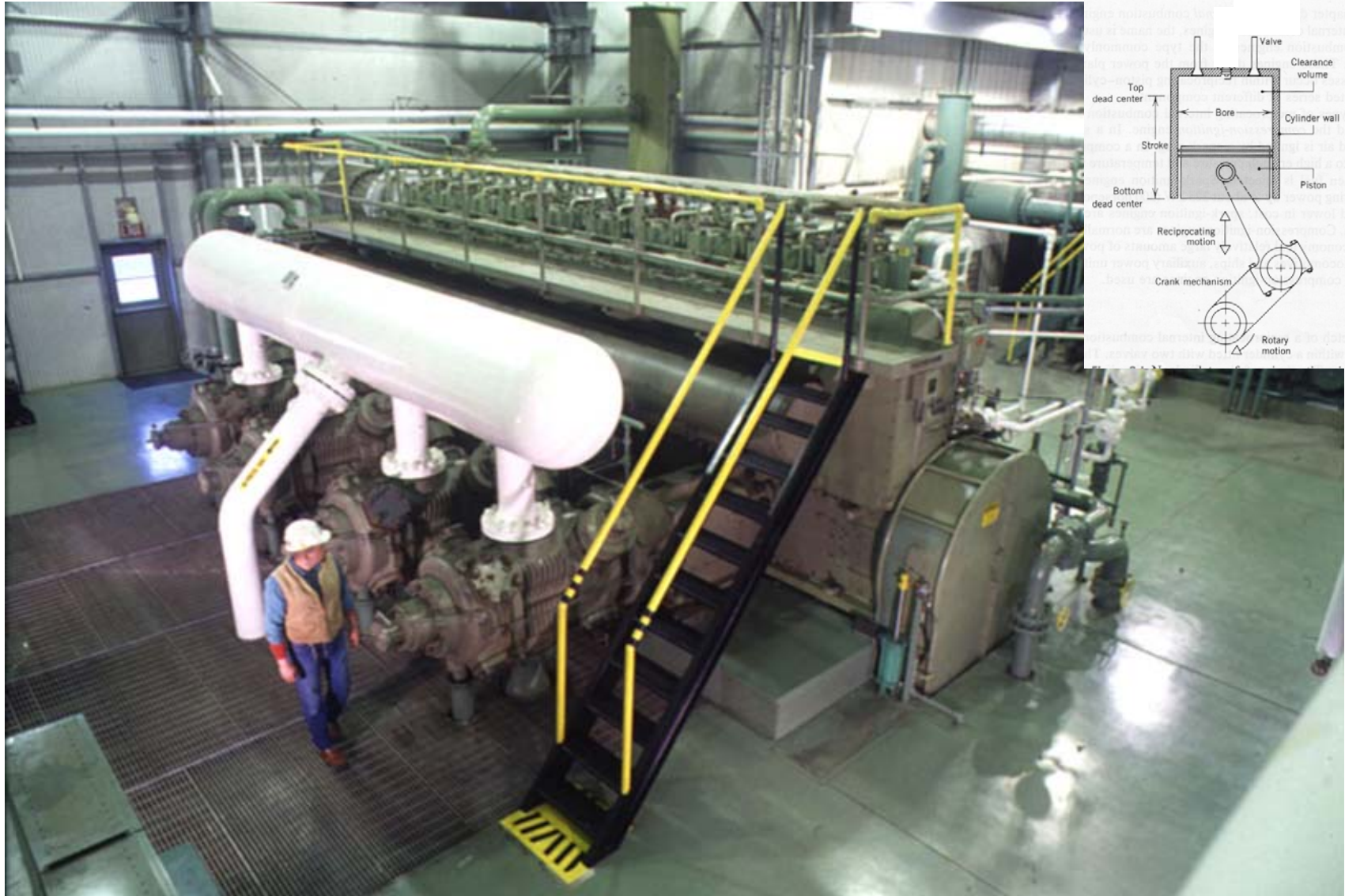


Compressor and turbine rotor blades for an industrial gas turbine.
(Courtesy of S. Laney)

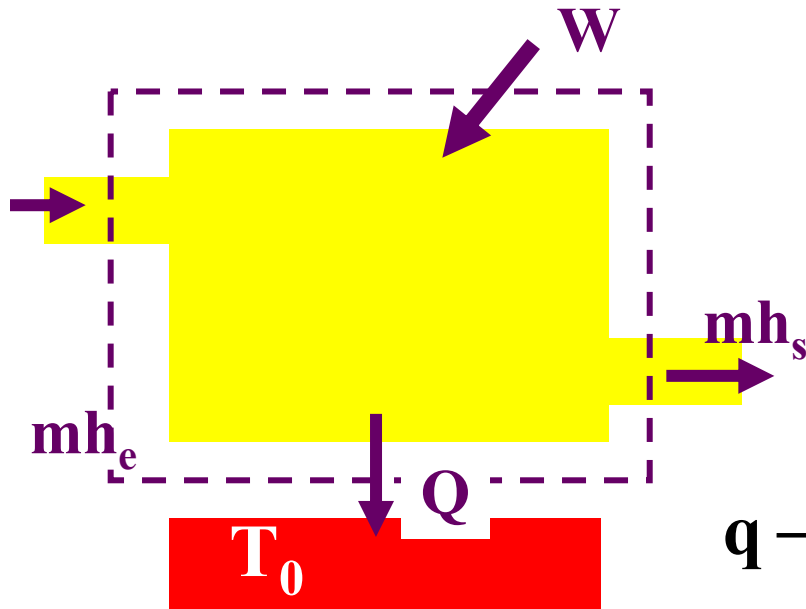


Compressor bed
showing compressor
stator blades for an
industrial gas turbine.
(Courtesy of S. Laney)

Compressores de Deslocamento Positivo



Identifique os fluxos para um Compressor



- O compressor rejeita calor para o ambiente.
- A temperatura T_0 é a temperatura do ambiente

$$q - w = (h_s - h_e) + \underbrace{\frac{V_s^2 - V_e^2}{2}}_{\text{neglected}} + \underbrace{g(z_s - z_e)}_{\text{neglected}}$$

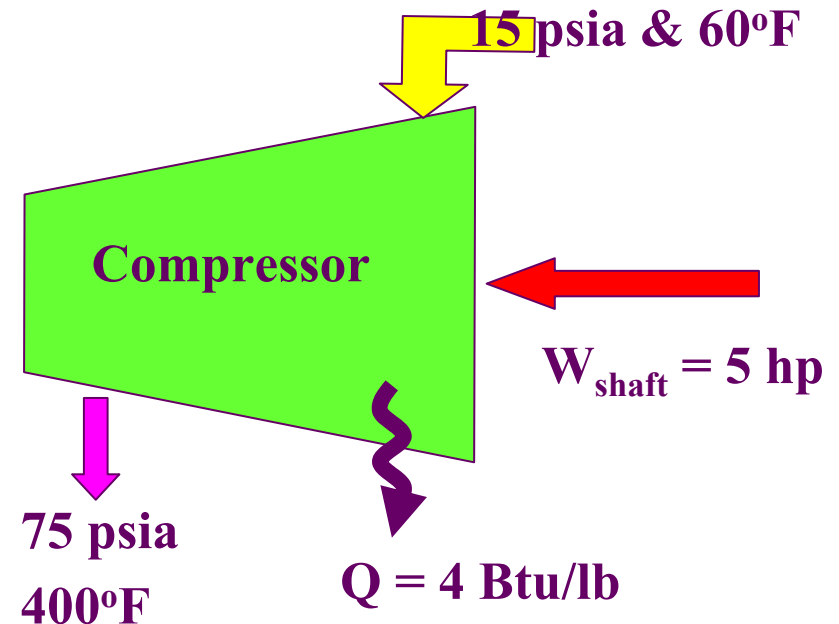
$$w = (h_e - h_s) + q$$

Sample Problem

Air initially at 15 psia and 60°F is compressed to 75 psia and 400°F. The power input to the air is 5 hp and a heat loss of 4 Btu/lbm occurs during the process. Determine the mass flow in lbm/min.

HYPOTHESIS

- Steady state steady flow (SSSF)
- Neglect potential energy changes
- Neglect kinetic energy changes
- Air is an ideal gas



$$w = (h_e - h_s) + q \rightarrow \dot{m} = \frac{\dot{W}_{\text{shaft}}}{q - C_p (T_s - T_e)} = 2.46 \text{ lbm / min}$$

5-64 Refrigerante R-12 entra em um compressor como um vapor saturado a $0\text{ }^{\circ}\text{C}$ e é comprimido até uma pressão de $1,60\text{ MPa}$. Admite-se que o compressor seja reversível e adiabático e opera em regime permanente com variações desprezíveis de energias cinética e potencial. Calcule a energia acrescentada na forma de trabalho para cada kilograma de R-12 no processo de compressão.

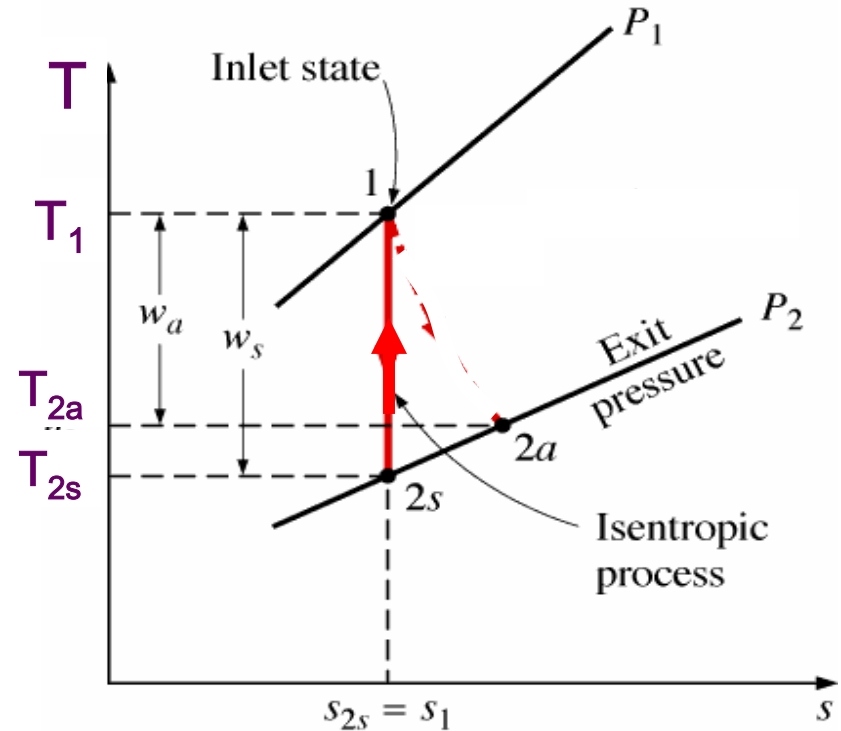
$$w = (h_e - h_s)$$

(tab. A-2.1) $h_e = + 187\text{ kJ/kg}$ e
 $s_e = 0.696\text{ kJ/kgK}$

processo reversível, $s_s = s_e =$
 0.696 kJ/kgK

(tab. A-2.2) $P_s = 1.6\text{ MPa}$ e $s_s =$
 $0.696\text{ kJ/kgK} \rightarrow h_s = 216\text{ kJ/kg}$

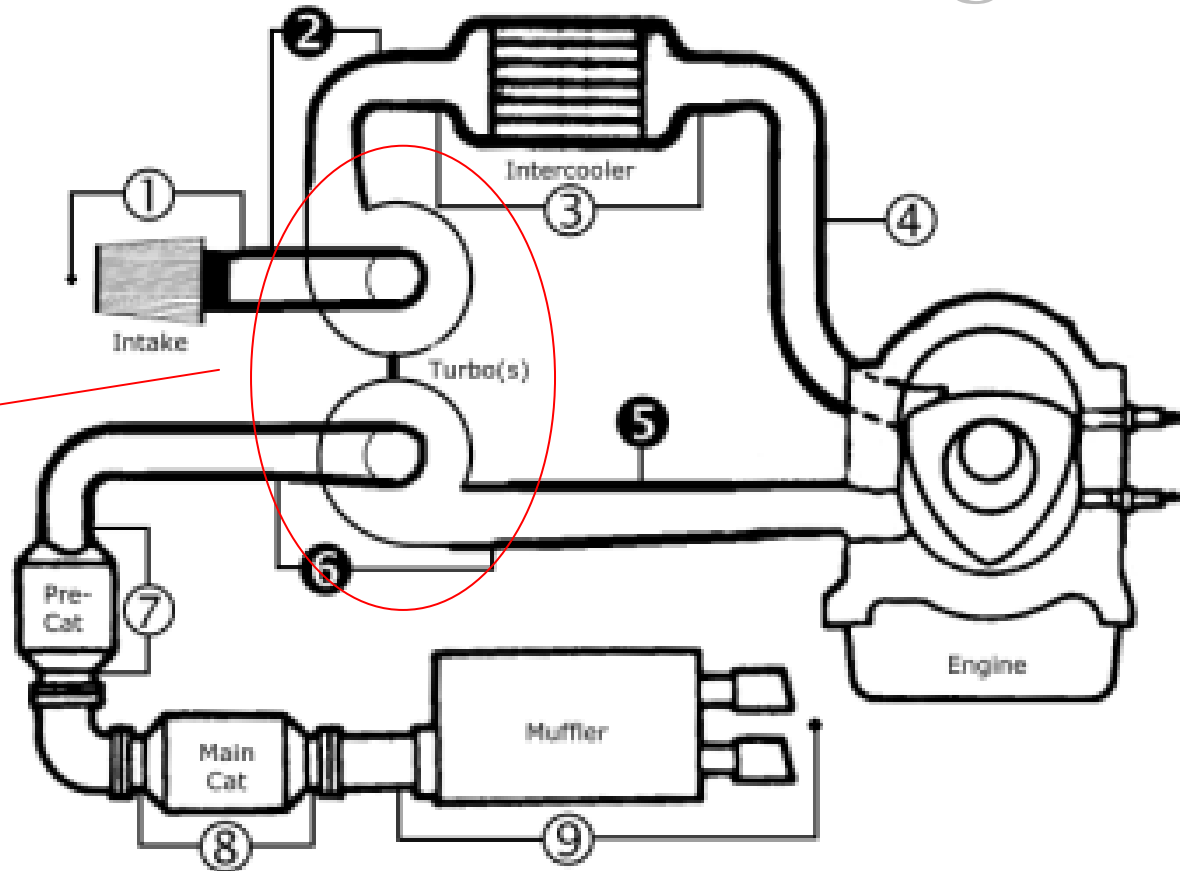
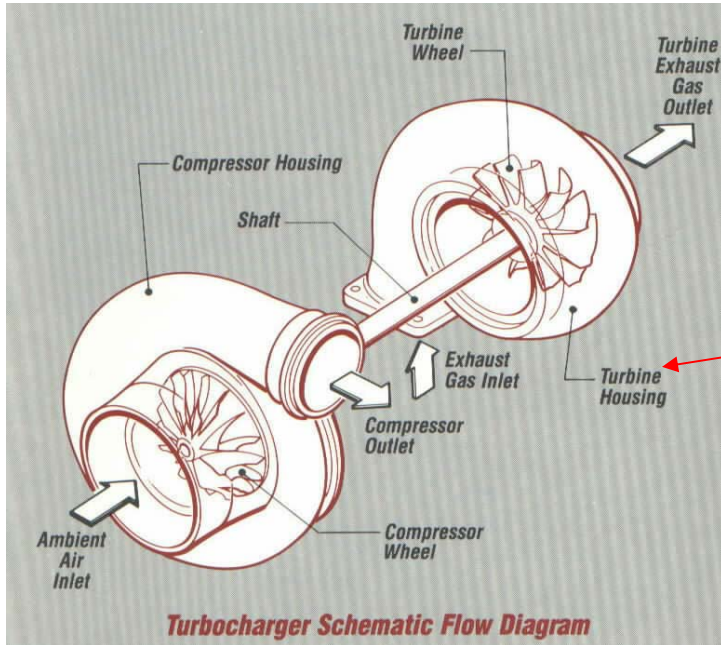
$$w = (187 - 216) = -29\text{ kJ / kg}$$



Cars & Intercoolers



Compression with intercooling



1	Intake System pressure drop
2	Turbo Compressor pressure increase
3	Intercooler & Piping pressure drop
4	Manifold Boost Pressure
5	Total Backpressure

6	Turbo pressure drop
7	Pre-Cat / Downpipe pressure drop
8	Main Cat / Midpipe pressure drop
9	Muffler / Cat-back pressure drop

Se você entender este problema você compreenderá a razão do intercooler

~~5-60~~ Para um processo reversível em regime permanente com variações desprezíveis de energias cinética e potencial, o trabalho específico (por unidade de massa) é dado pela eq. 5-56. Compare esse trabalho com o trabalho para comprimir um sistema formado por uma unidade de massa entre os mesmos limites de pressão (começando com o ar a um dado estado inicial) para:

- (a) Um processo isotérmico reversível.
- (b) Um processo adiabático reversível.

Expressão para cálculo do trabalho de eixo compressão ou expansão

1ª lei

$$w_{\text{eixo}} = (h_e - h_s) + q$$

proc. rev. &
relação termd.

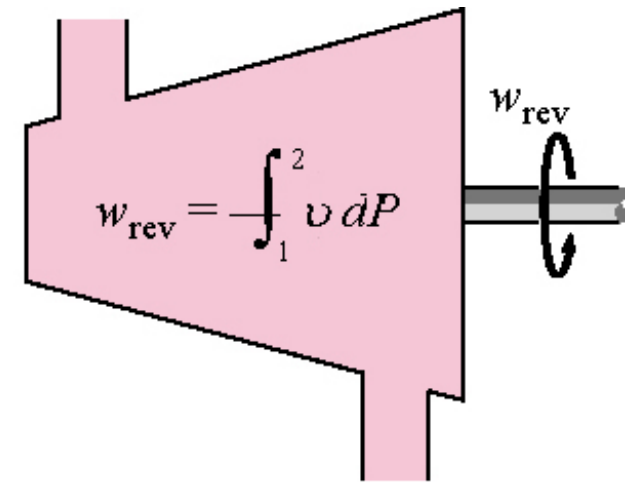
$$q = \int T ds = \int dh - \int v dP$$

isolando $\Delta h + q$

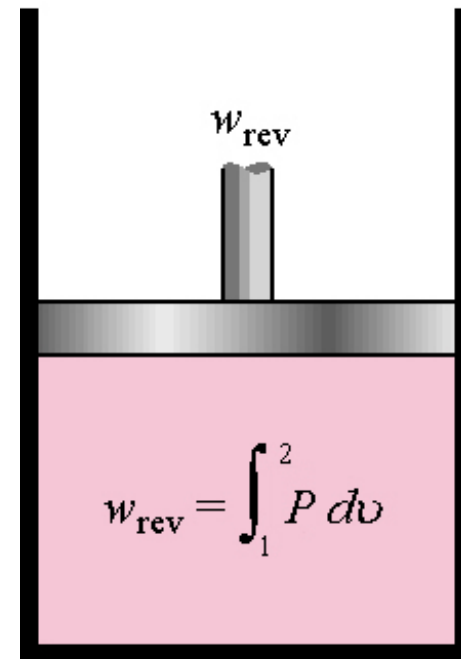
$$\Rightarrow (h_e - h_s) + q = - \int v dP$$

$$w_{\text{eixo}} = - \int v dP \quad \text{Eq. (5.56)}$$

Reversible work relations for steady-flow and systems

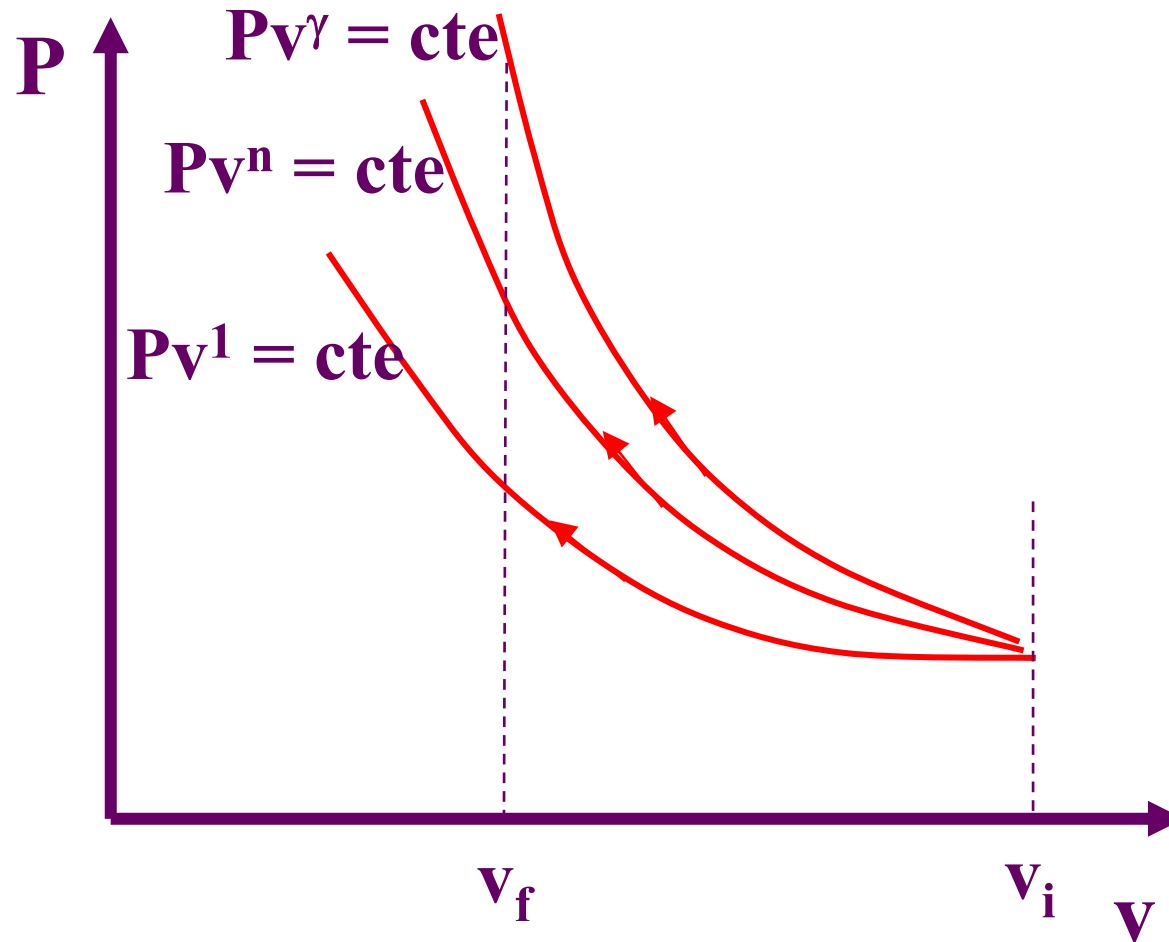


(a) Steady-flow system

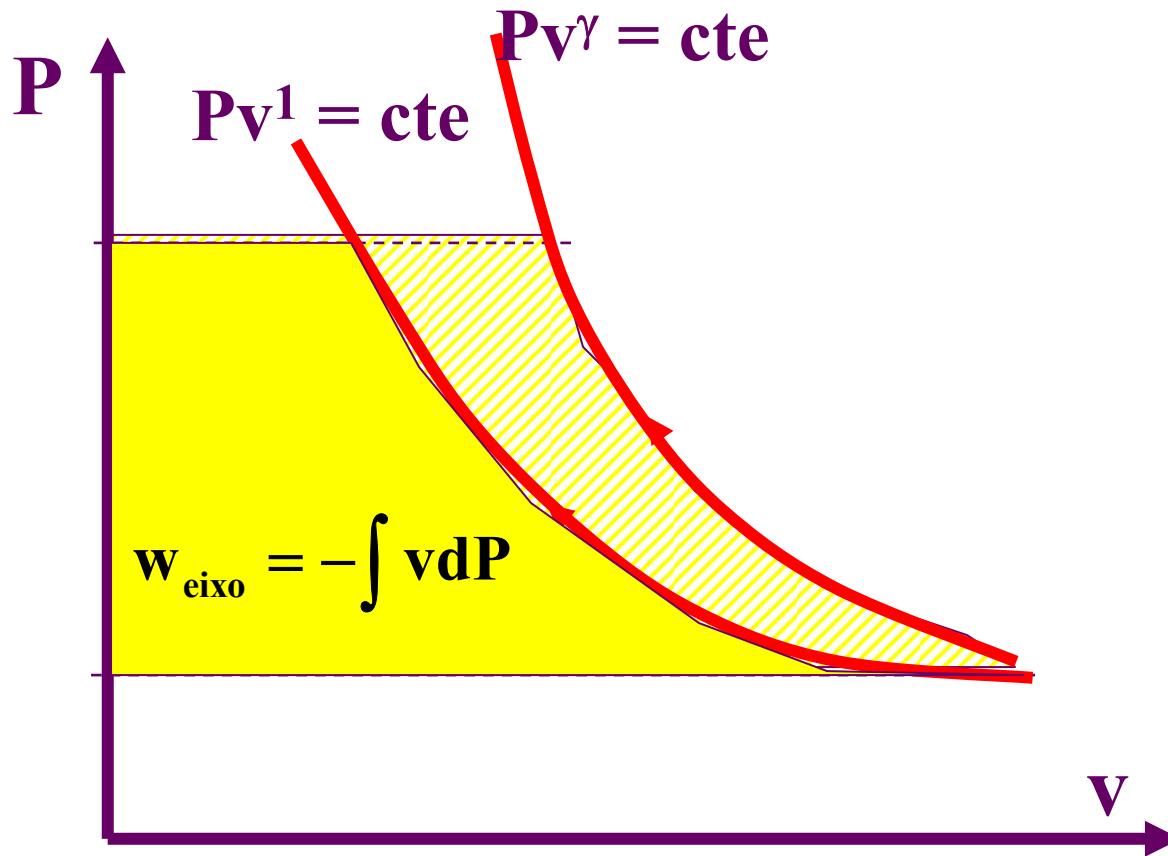


(b) Closed system

Compressor e o Diagrama P-v para um processo reversível



Compressor e o Diagrama P-v para um processo reversível

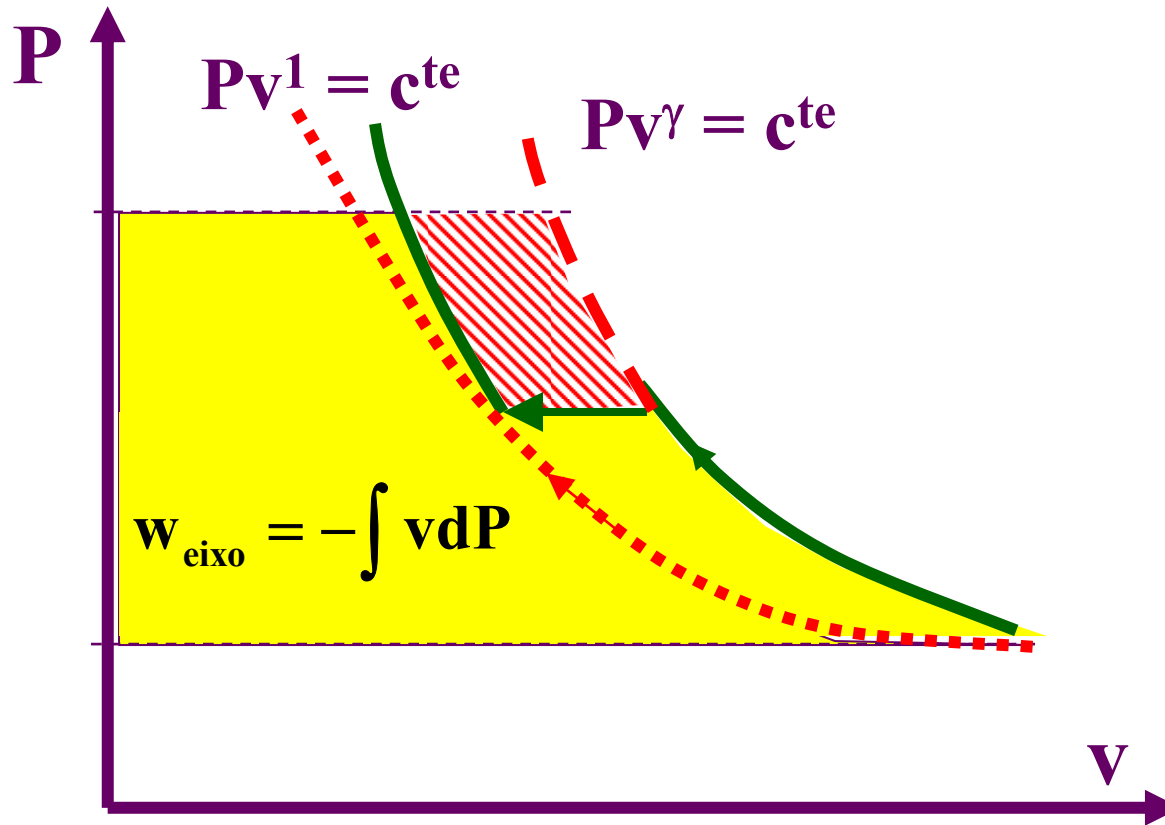


A área amarela representa o trabalho necessário para um processo isotérmico

A área achurada representa o trabalho extra necessário para um processo adiabático

Conclusão: menor trabalho é o processo isotérmico

Compressão com Resfriamento Intermediário



1º estágio

Intercooler

2º estágio

Trabalho eixo

Trabalho
'poupado'

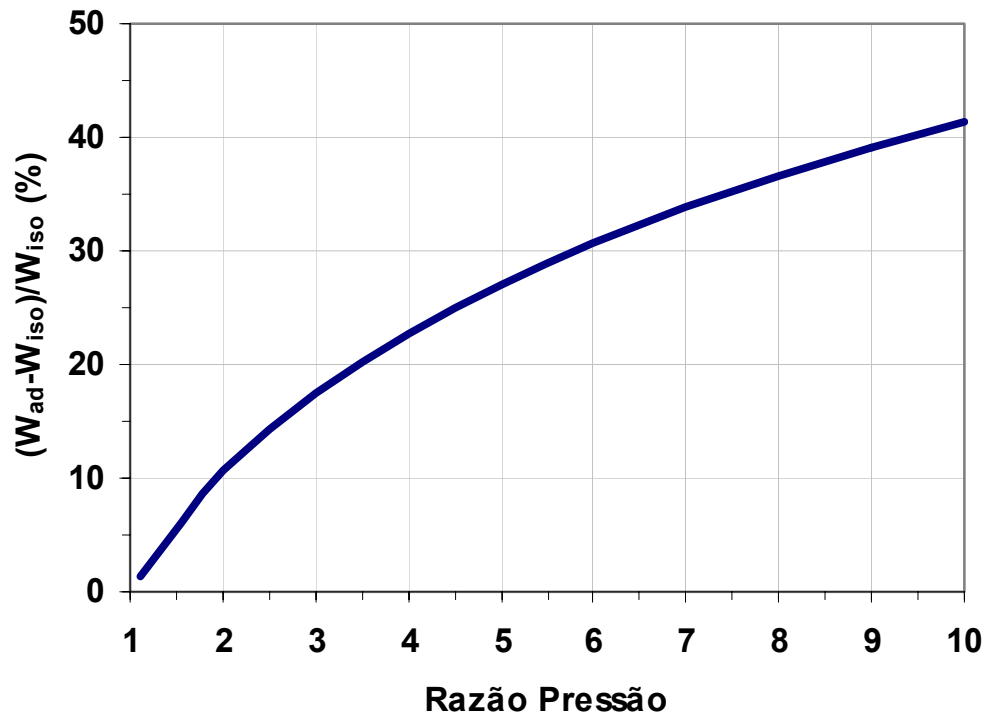
Trabalho
isotérmico

Conclusão: inter-resfriamento 'poupa' trabalho eixo

Trabalho de Eixo Reversível, Gás Ideal

Isotérmico: $v = RT/P \rightarrow$
$$W_{\text{iso}} = -RT \int \frac{dP}{P} = -RT \ln \left(\frac{P_f}{P_i} \right)$$

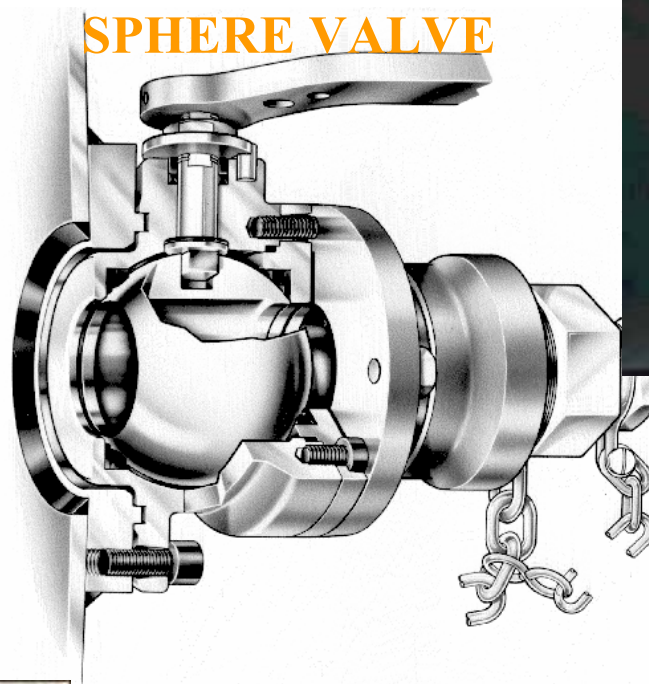
Adiabático: $Pv^\gamma = c \rightarrow$
$$W_{\text{adiab}} = - (P_i v_i)^{\frac{1}{\gamma}} \int \frac{dP}{P^\gamma} = - \left(\frac{\gamma}{\gamma-1} \right) (P_i v_i) \left[\left(\frac{P_f}{P_i} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]$$



BUTTERFLY VALVE



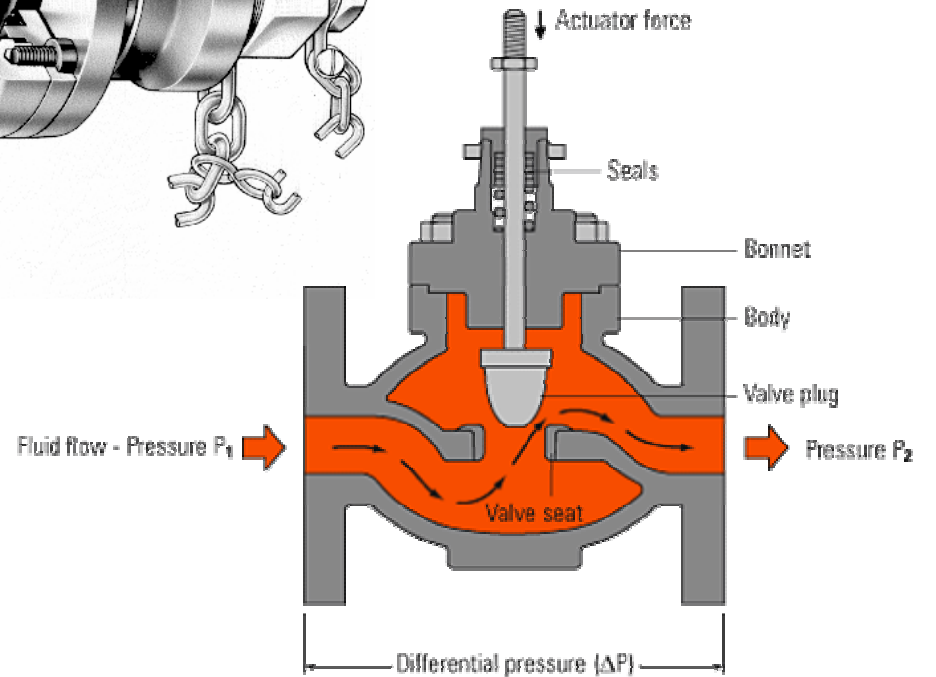
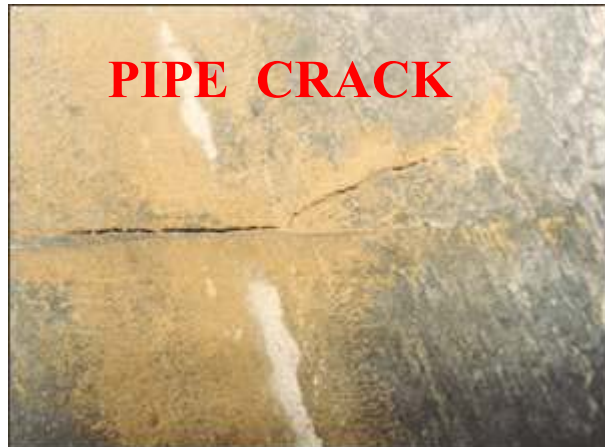
SPHERE VALVE



GATE VALVE

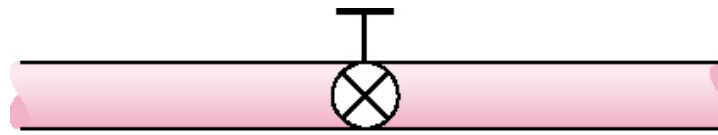


PIPE CRACK



GLOBE VALVE

Throttling Devices (Valves)



(a) An adjustable valve



(b) A porous plug



(c) A capillary tube

Typical assumptions for throttling devices

1. No work
2. Potential energy changes are zero
3. Kinetic energy changes are usually small
4. Heat transfer is usually small
5. Two port device

Look at energy equation:

Apply assumptions from previous page:

$$q - w = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

We obtain: $(h_2 - h_1) = 0$ or

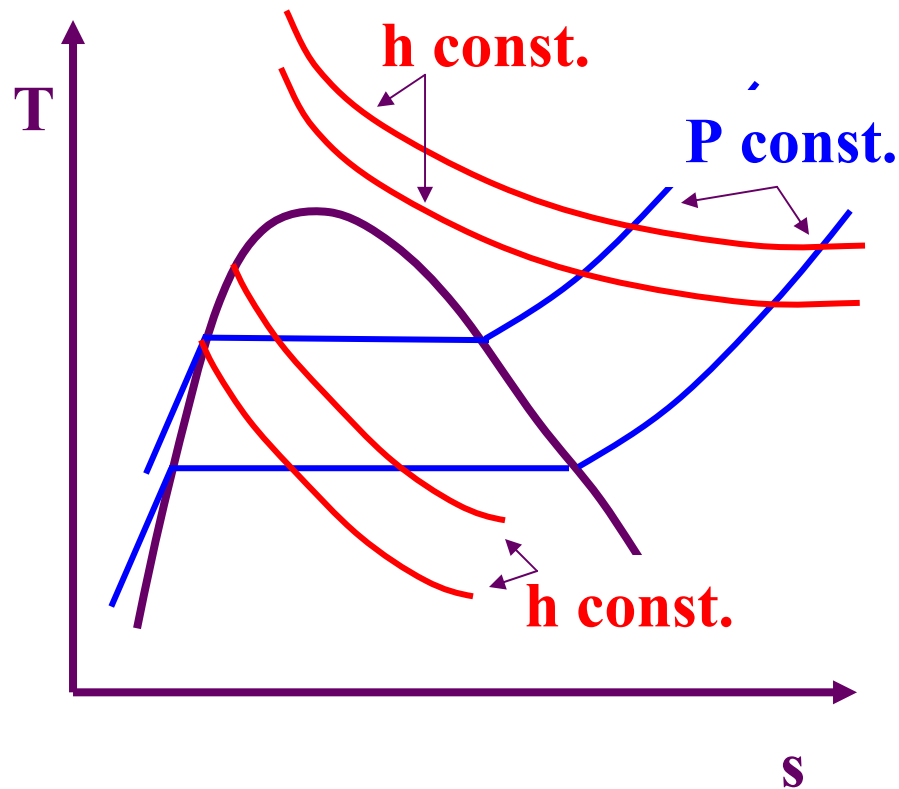
$$h_2 = h_1$$

Does the fluid temperature:

increase,
decrease, or
remain constant

as it goes through an adiabatic valve?

Look at implications: If the fluid is liquid/vapor:



During throttling process:

- The pressure drops,
- The temperature drops,
- Enthalpy is constant

5-66 Para um processo de estrangulamento adiabático envolvendo R-12 líquido saturado a uma temperatura de 25 °C, qual das seguintes propriedades aumenta, diminui ou permanece constante? (i) pressão, (ii) temperatura, (iii) entalpia, (iv) energia interna e (v) entropia. Quais são os valores dessas propriedades se a pressão após o estrangulamento vale 1 atm?

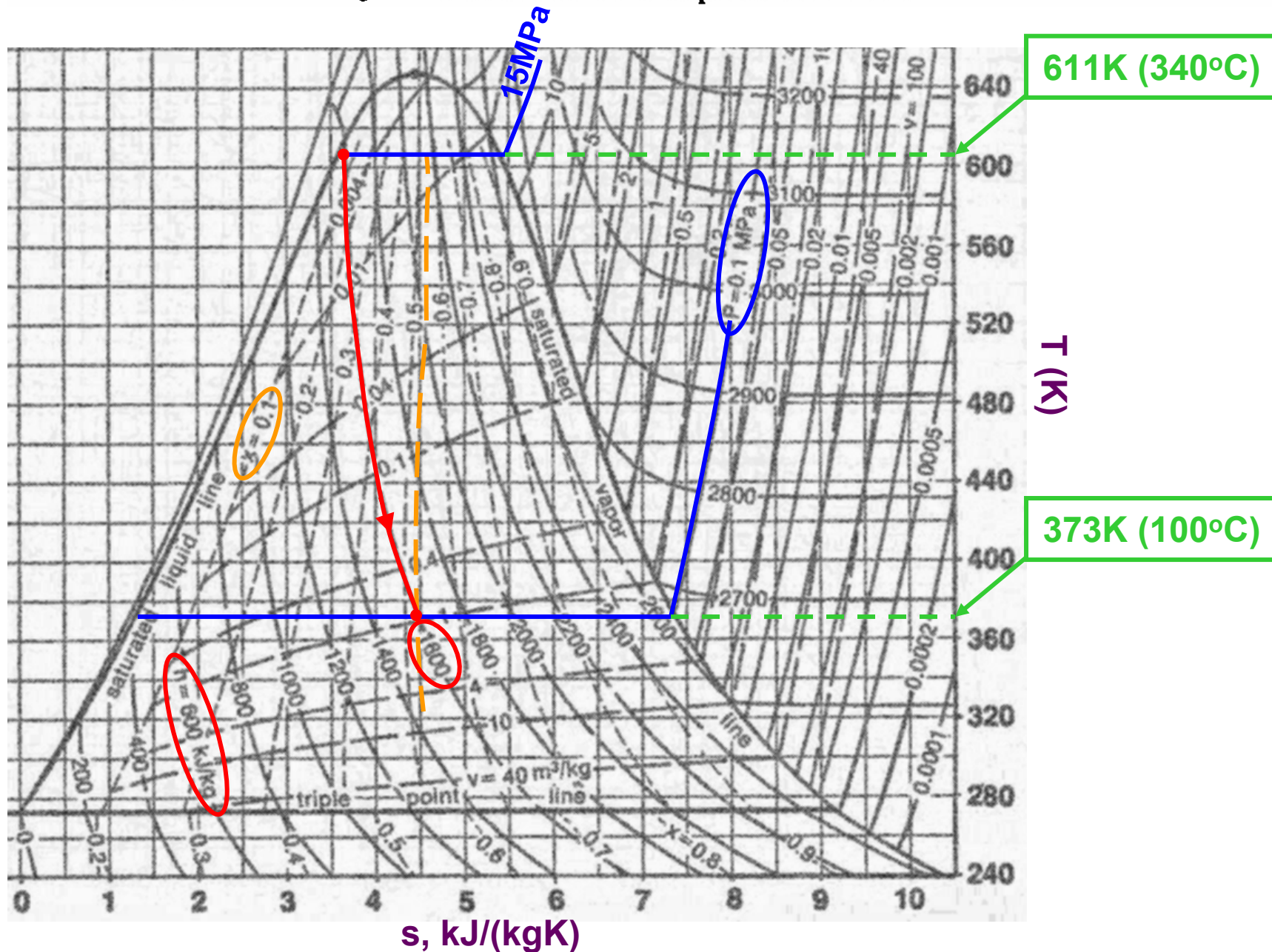
Look at implications: if fluid is an ideal gas:

$$(h_2 - h_1) = C_p (T_2 - T_1) = 0$$

C_p is always a positive number, thus:

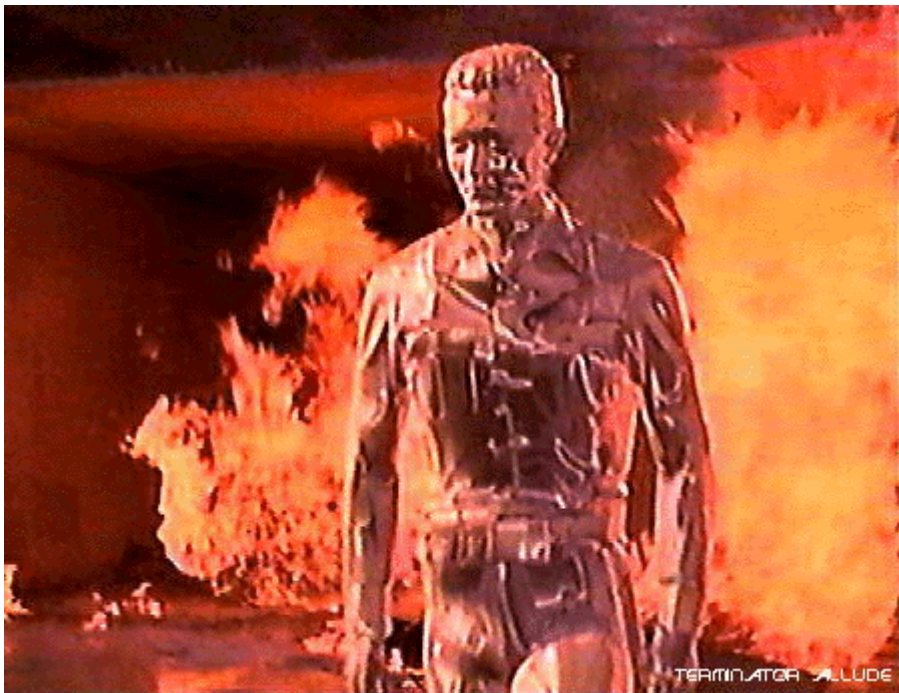
$$T_2 = T_1$$

5-65 Usando o diagrama $T-s$ para a água, Tabela A-6 no apêndice, estime o título após água na fase líquida ter sido estrangulada de uma pressão de 150 bars a uma pressão de 1 bar.



Fenômeno de Fragilização de Metais por Baixas Temperaturas

John Connor (T1000) in Terminator 2



Consequences of the Temperature Drop on Material Strength

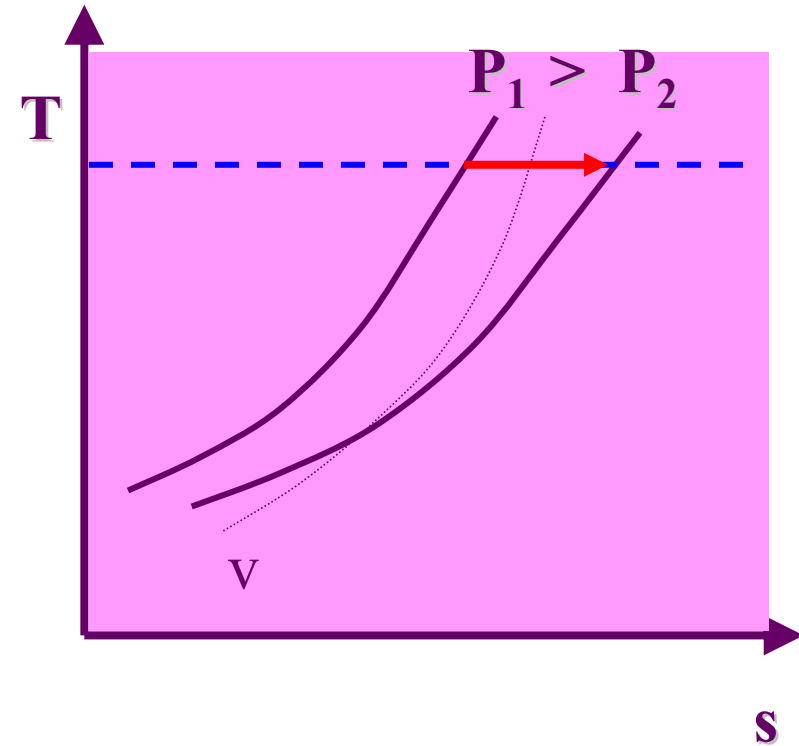
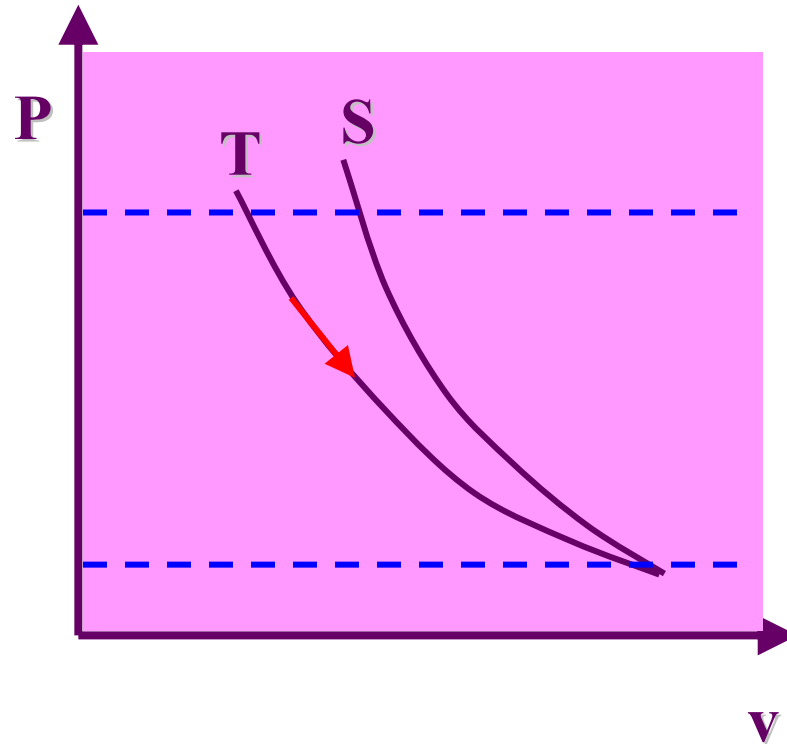
Cold embrittlement

(dureza e rigidez mas baixa resistência a tensão)



Low temperature embrittlement does affect most materials more or less pronounced. It causes overloaded components to fracture spontaneously rather than accommodating the stress by plastic deformation. The picture shows a fractured fitting whose material was not suitable for low temperatures.

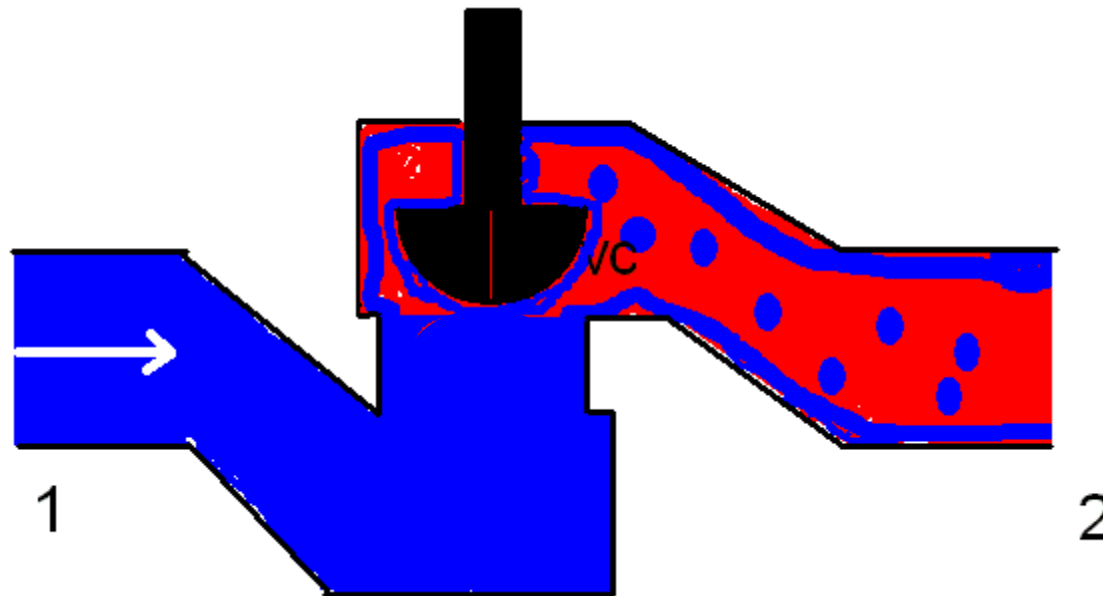
5-67E Ar é estrangulado através de uma válvula adiabática de 200 psi e 100 °F para uma pressão de 15 psi. Assuma comportamento ideal. Como as seguintes propriedades variam: (i) pressão, (ii) temperatura, (iii) entalpia, (iv) energia interna e (v) entropia?



Pressão	diminui
Entalpia	constante
Temperatura	gás ideal $h=h(T)$, portanto T fica constante
En. Interna	gás ideal $u=u(T)$, portanto u fica constante
Entropia	diminui, $\Delta s = C_p \ln(T_2/T_1) - R \ln(P_2/P_1)$

TEAMPLAY

Refrigerant 12 enters a valve as a saturated liquid at 0.9607 MPa and leaves at 0.1826 MPa. What is the quality and the temperature of the refrigerant at the exit of the valve?



State (1)

Liquid saturated, $x=0$

$P_{\text{sat}} = 0.9607 \text{ MPa}$

$T_{\text{sat}} = ? \quad 40^\circ\text{C}$

$h_{\text{liq}} = ? \quad 75 \text{ kJ/kg}$

State (2)

Liq+vap $x=? \quad 0.33$

$P_{\text{sat}} = 0.1826 \text{ MPa}$

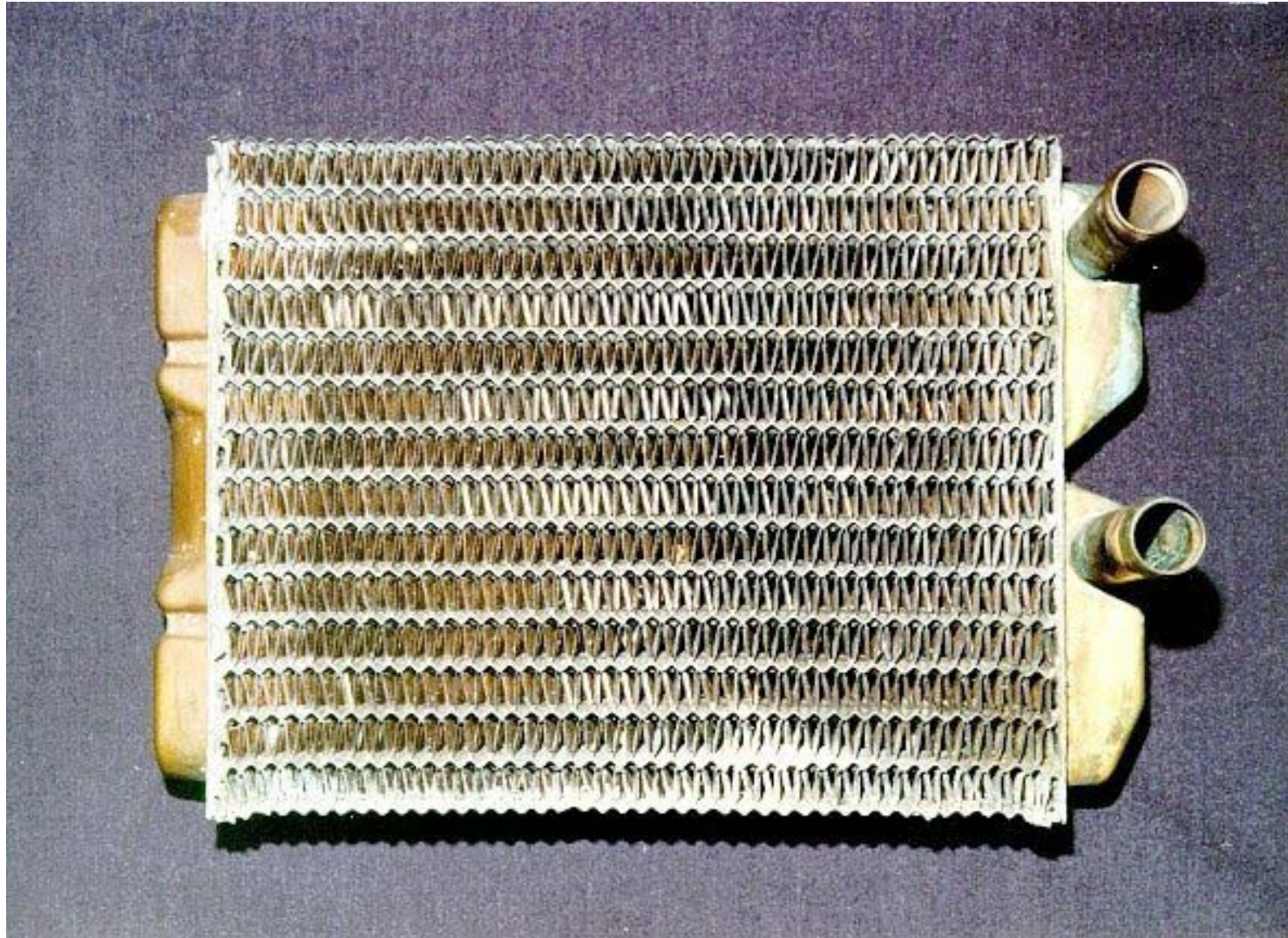
$T_{\text{sat}} = ? \quad -15^\circ\text{C}$

$h_{\text{liq}} = ? \quad 22 \text{ \& } 180 \text{ kJ/kg}$

Heat exchangers are used in a variety of industries

- **Automotive - radiator**
- **Refrigeration - evaporators/condensers**
- **Power production - boilers/condensers**
- **Power electronics - heat sinks**
- **Chemical/petroleum industry- mixing processes**

Something a little closer to home..



Heat exchanger or heater core for a 1966 Mustang heating system.

Heat Exchangers



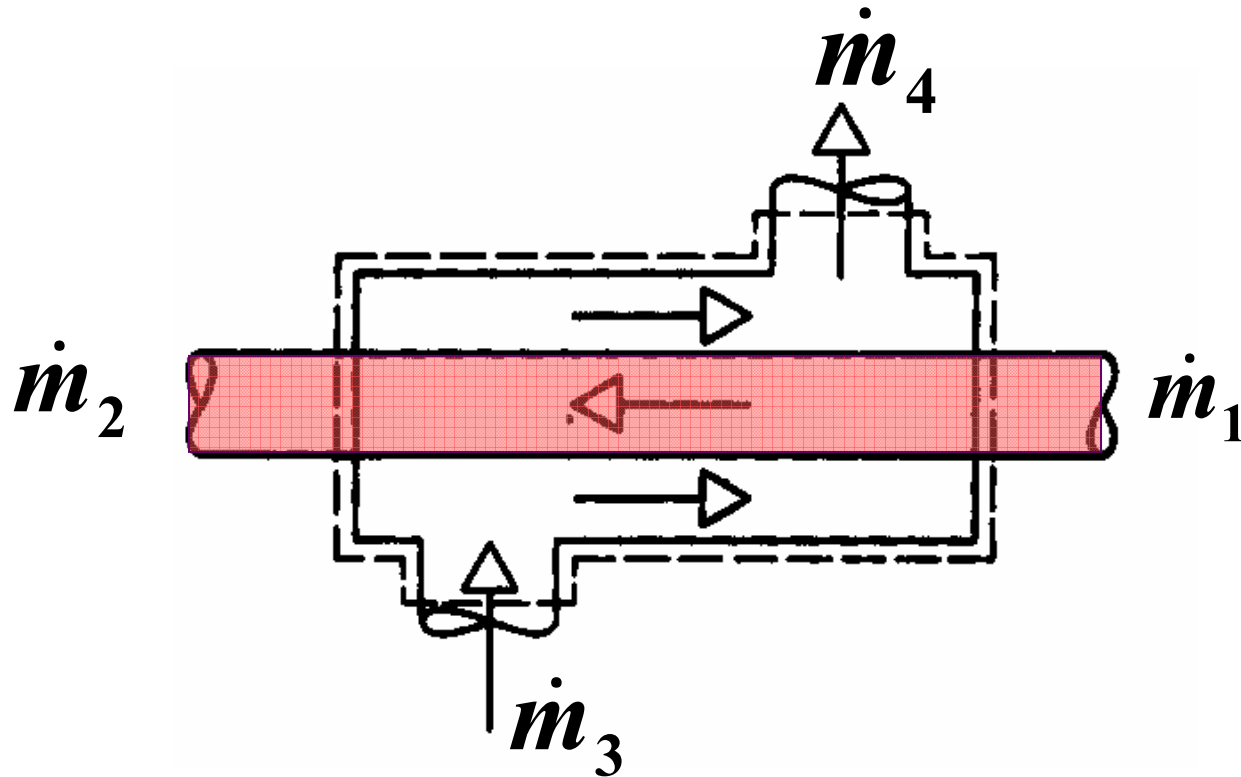
Large Paraflow plate heat exchanger being assembled.
(Courtesy of APV Heat Exchanger Product Group)

Condenser/evaporator for heat pump



Heat Exchangers

- Now, we must deal with multiple inlets and outlets:



If we have steady flow, then:

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

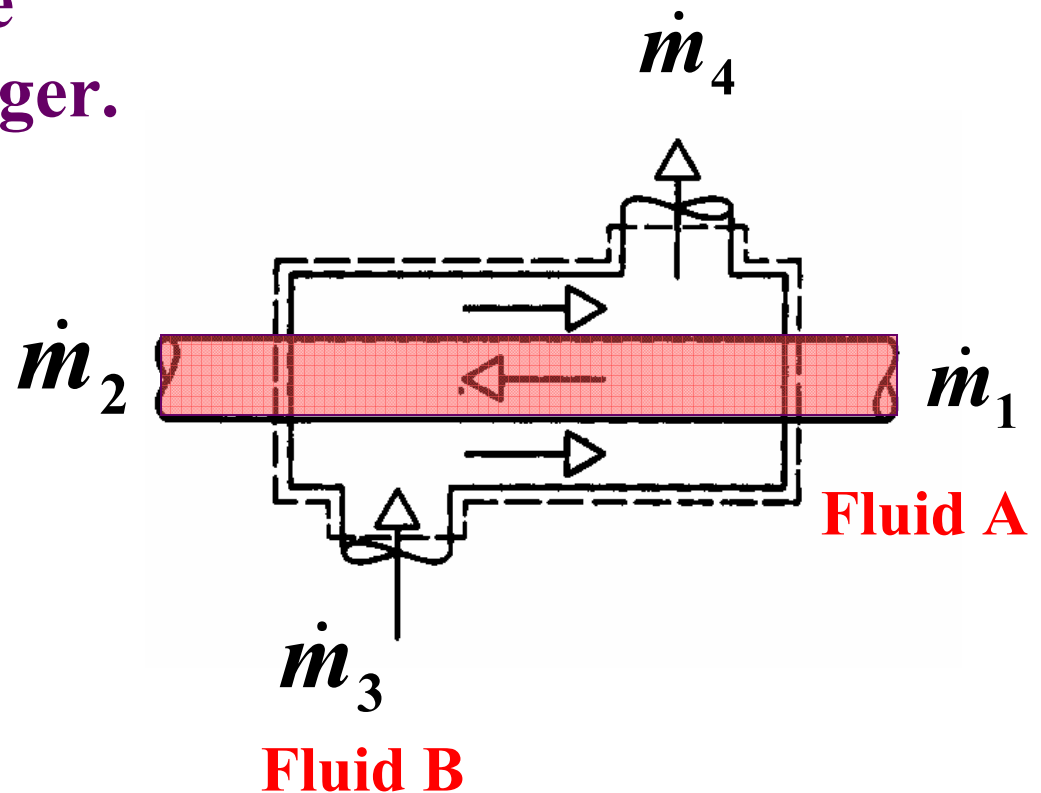
$$\dot{m}_3 = \dot{m}_4 = \dot{m}_B$$

Conservation of energy can be a little more complicated...

I've drawn the control volume around the whole heat exchanger.

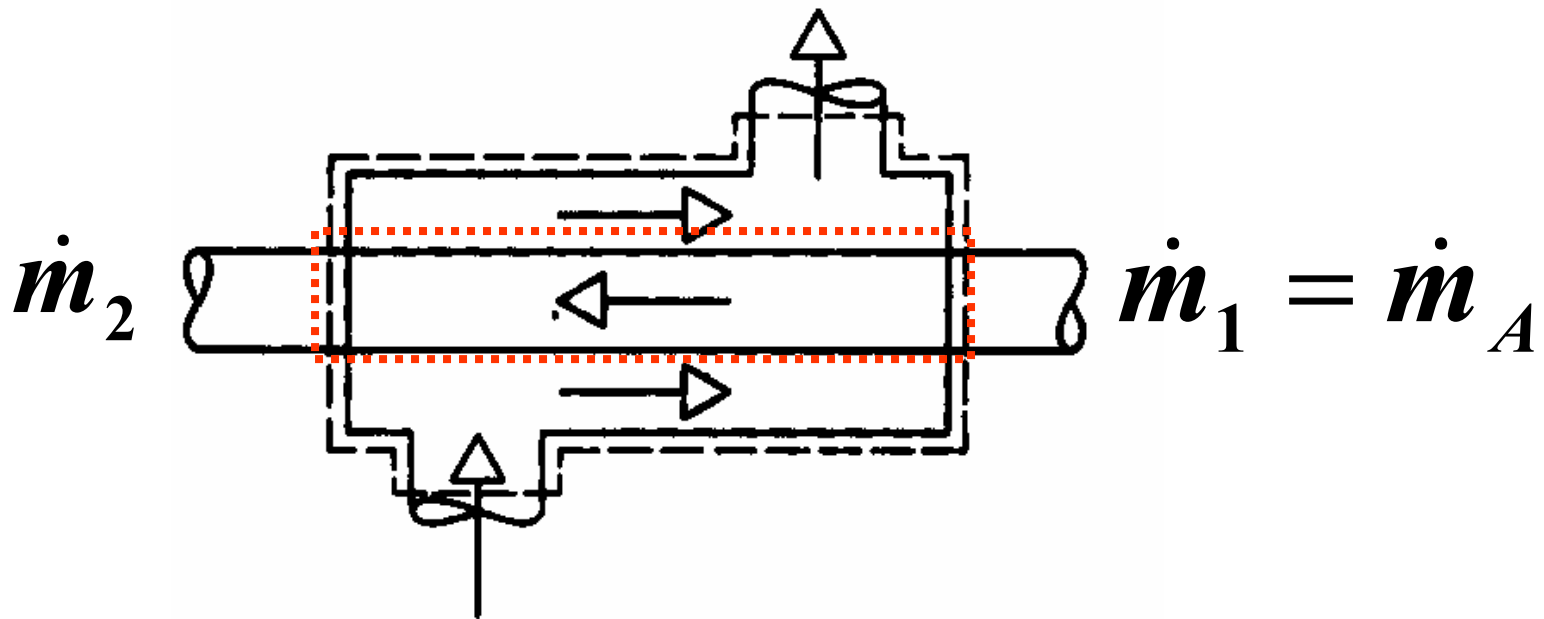
Implications:

No heat transfer from the control volume.



Heat Exchangers

- Now if we want the energy lost or gained by either fluid we must let that fluid be the control volume, indicated by the red.



Heat Exchangers

$$\dot{Q} - \dot{W}_{CV} + \dot{m}_A \left((h_1 - h_2) + \underbrace{\frac{V_1^2}{2} - \frac{V_2^2}{2}}_{\text{0, (sometimes negligible)}} + \underbrace{g(z_1 - z_2)}_{\text{0, (usually negligible)}} \right)_A$$

0, (sometimes negligible) **0, (usually negligible)**

$$+ \dot{m}_B \left((h_3 - h_4) + \underbrace{\frac{V_3^2}{2} - \frac{V_4^2}{2}}_{\text{0, (sometimes negligible)}} + \underbrace{g(z_3 - z_4)}_{\text{0, (usually negligible)}} \right)_B = 0$$

0, (sometimes negligible) **0, (usually negligible)**

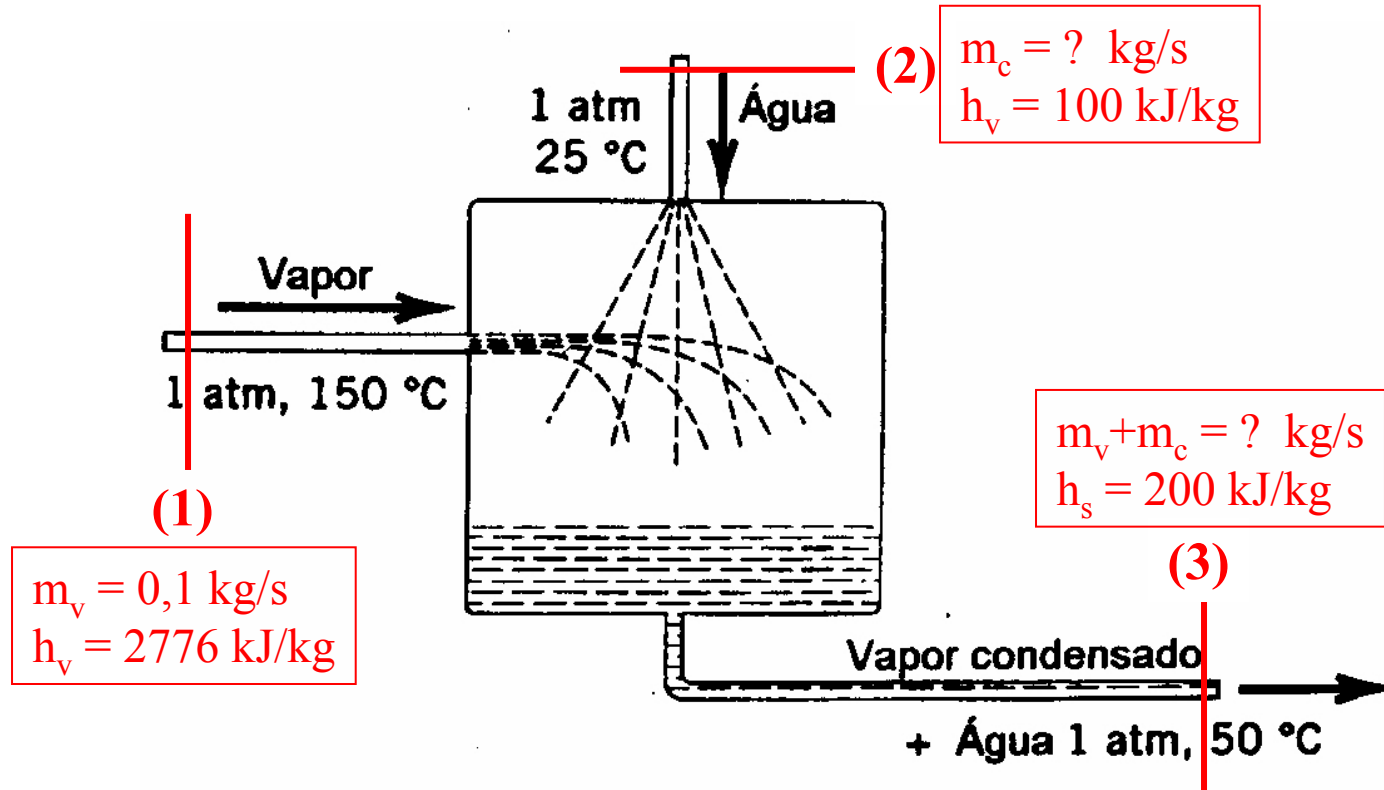
Heat Exchangers

- And we are left with

$$\dot{m}_A (h_1 - h_2) = \dot{m}_B (h_4 - h_3)$$

The energy change of fluid A is equal to the negative of the energy change in fluid B.

5-70 Um condensador (Fig. P5-70) de uma instalação motora a vapor opera em regime permanente. O vapor entra o dispositivo a 150 °C e 1 atm e é condensado através de um pulverizador de água fria que entra o dispositivo a 1 atm e 25 °C ($h = 100$ kJ/kg). O vapor condensado e a água deixam o condensador a 1 atm ($h = 200$ kJ/kg). Se a vazão do vapor for de 0,1 kg/s, qual deve ser a vazão mássica necessária da água de resfriamento?



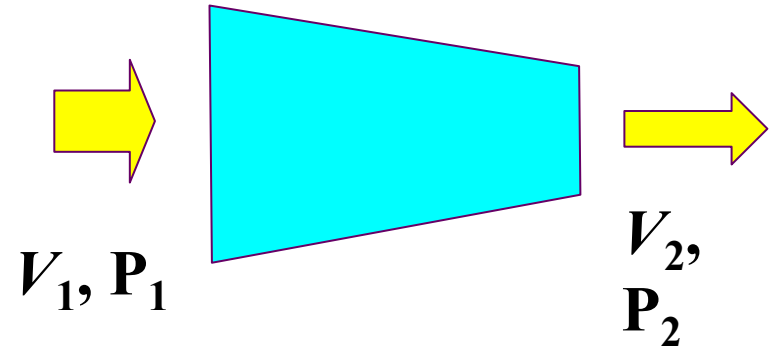
$$-\dot{m}_1 - \dot{m}_2 + \dot{m}_3 = 0$$

$$-\dot{m}_1 h_1 - \dot{m}_2 h_2 + \dot{m}_3 h_3 = 0$$

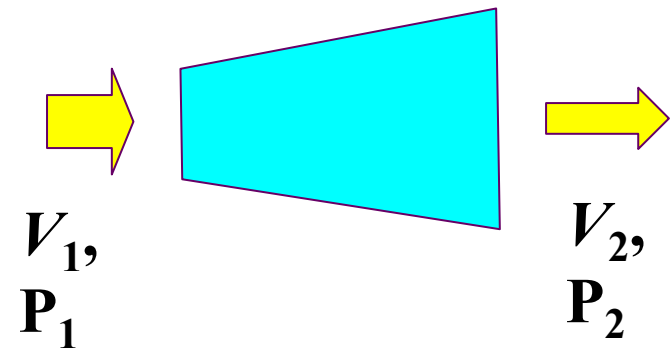
Resp.:
 $m_c = 2,57$ kg/s

Nozzles and Diffusers

- **Nozzle**--a device which accelerates a fluid as the pressure is decreased.

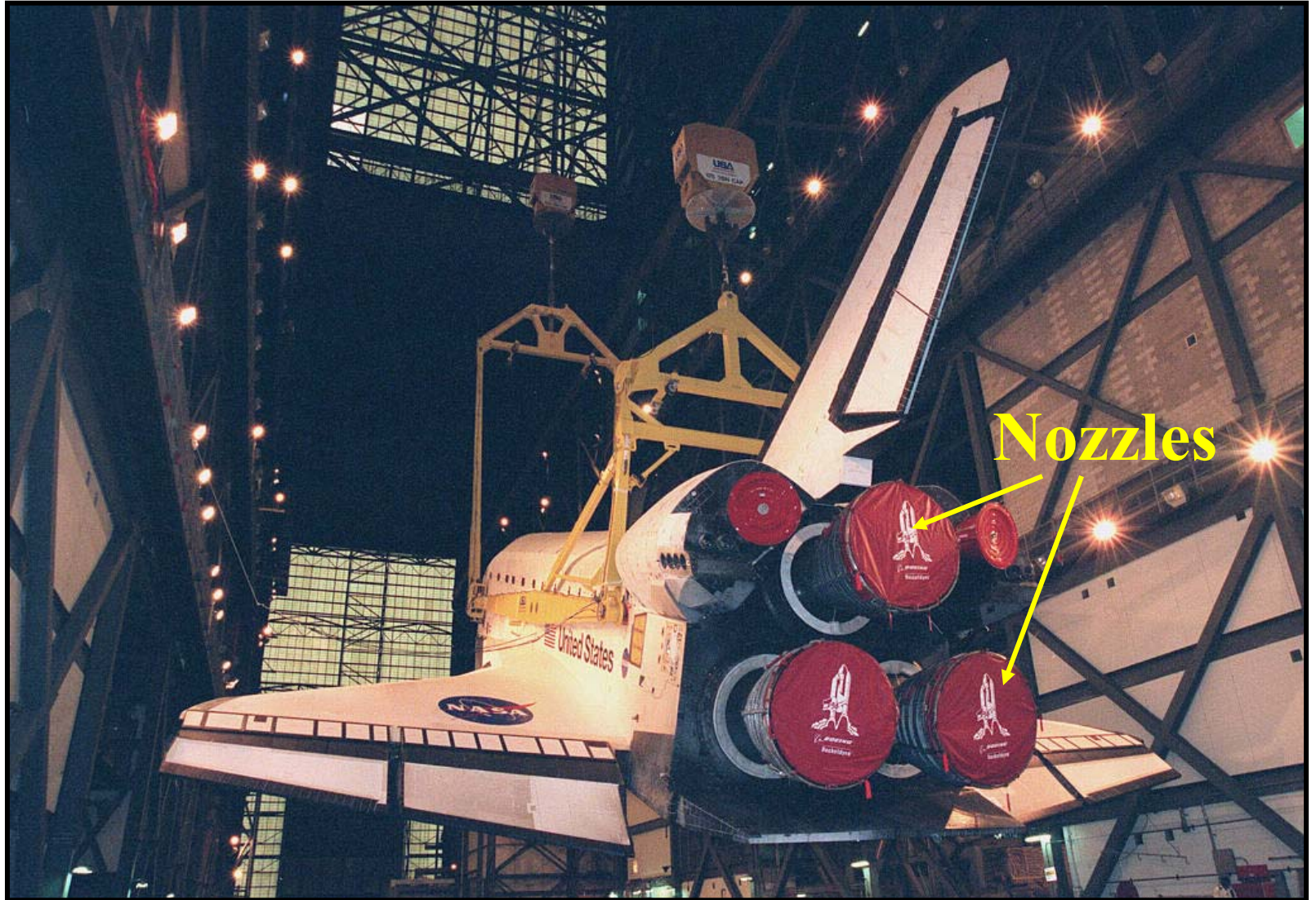


- **Diffuser**--a device which decelerates a fluid and increases the pressure.

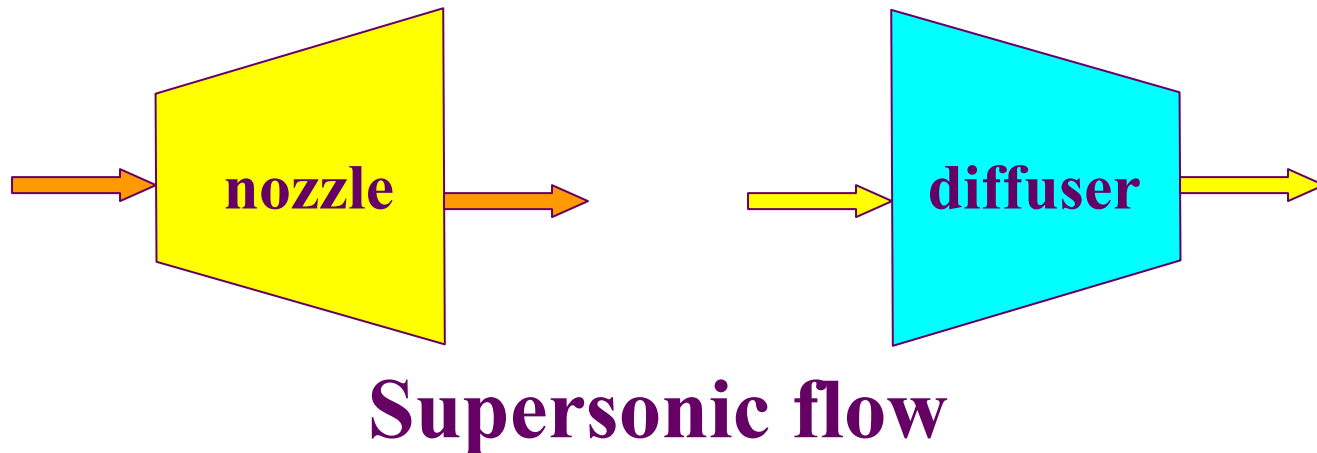
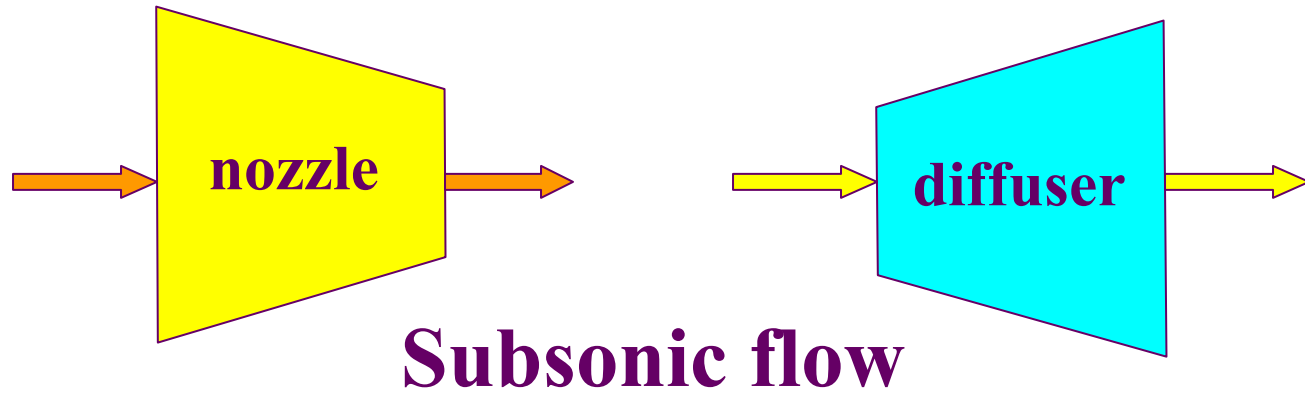


This configuration is for sub-sonic flow.

For supersonic flow, the shape of the nozzle is reversed.



General shapes of nozzles and diffusers



conservation of energy

$$\overset{0}{q} - \overset{0}{w} = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2} + g(z_2 - z_1)$$

$q = 0$ (adiabatic)

$w = 0$ (these are not work producing devices;
neither is work done on them)

$$(h_2 - h_1) = \frac{V_1^2 - V_2^2}{2}$$

Sample Problem

An adiabatic diffuser is employed to reduce the velocity of a stream of air from 250 m/s to 35 m/s. The inlet pressure is 100 kPa and the inlet temperature is 300°C. Determine the required outlet area in cm² if the mass flow rate is 7 kg/s and the final pressure is 167 kPa.

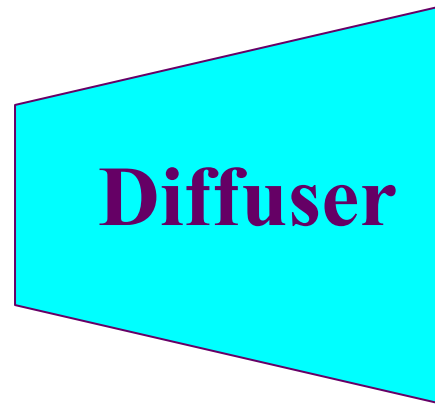
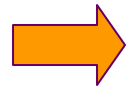
INLET

$$T_1 = 300^\circ\text{C}$$

$$P_1 = 100 \text{ kPa}$$

$$V_1 = 250 \text{ m/s}$$

$$\dot{m} = 7 \text{ kg/s}$$



OUTLET

$$P_2 = 167 \text{ kPa}$$

$$V_2 = 35 \text{ m/s}$$

$$A_2 = ?$$

Conservation of Mass: Steady State Regime

$$\dot{m} = \frac{V_1 A_1}{v_1} = \frac{V_2 A_2}{v_2} \quad \text{solve for } A_2 \quad \rightarrow \quad A_2 = \frac{\dot{m} v_2}{V_2}$$

But we don't know v_2 !

Remember ideal gas equation of state?

$$v_1 = \frac{RT_1}{P_1} \quad \text{and} \quad v_2 = \frac{RT_2}{P_2}$$

We know T_1 and P_1 , so v_1 is simple.

We know P_2 , but what about T_2 ?

NEED ENERGY EQUATION!!!!

Energy Eqn

$$(h_1 - h_2) \equiv C_P (T_1 - T_2) = \frac{V_1^2 - V_2^2}{2}$$

If we assumed constant specific heats, we could get T_2 directly

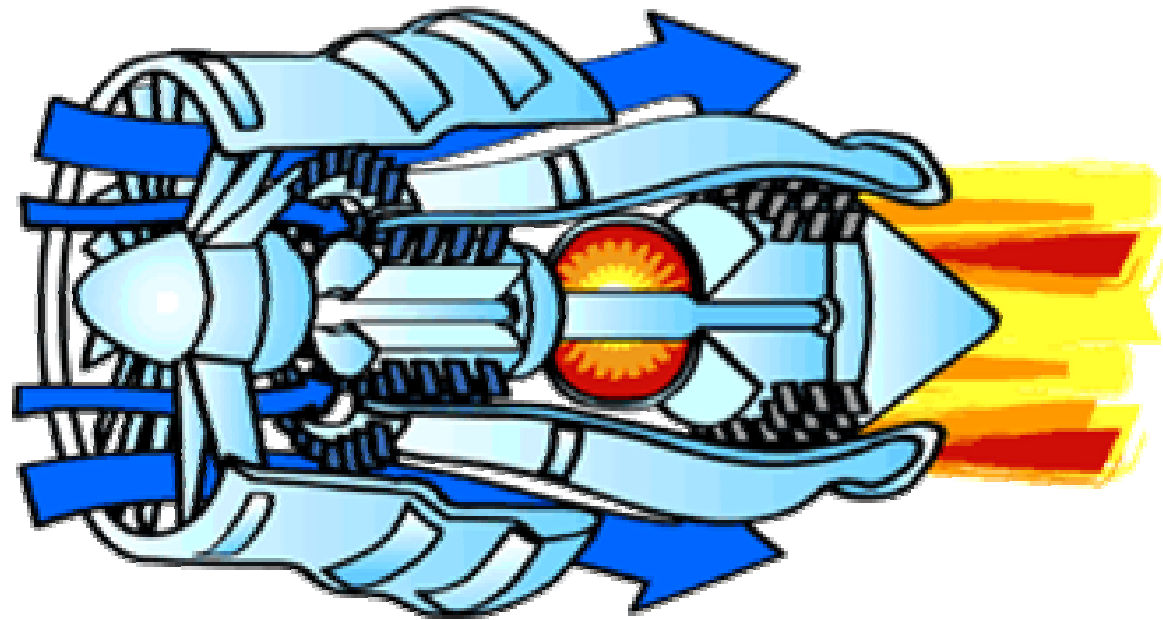
$$T_2 = T_1 - (V_1^2 - V_2^2) / 2 \cdot C_P = 602K$$

The ideal gas law: $v_2 = RT_2 / P_2 = 1.0352 \text{ m}^3 / \text{kg}$

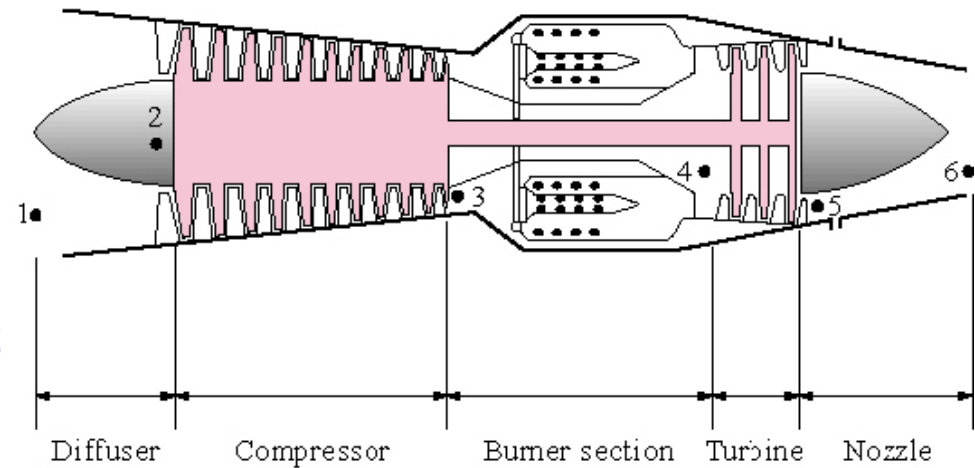
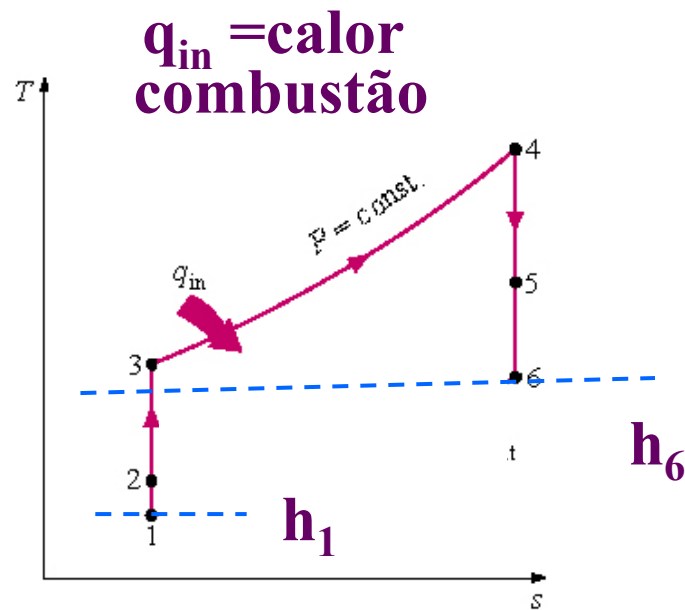
And the area:

$$A_2 = \frac{\dot{m} v_2}{V_2} = \frac{7 \cdot 1.035}{35} \cdot 10^4 = 2070 \text{ cm}^2$$

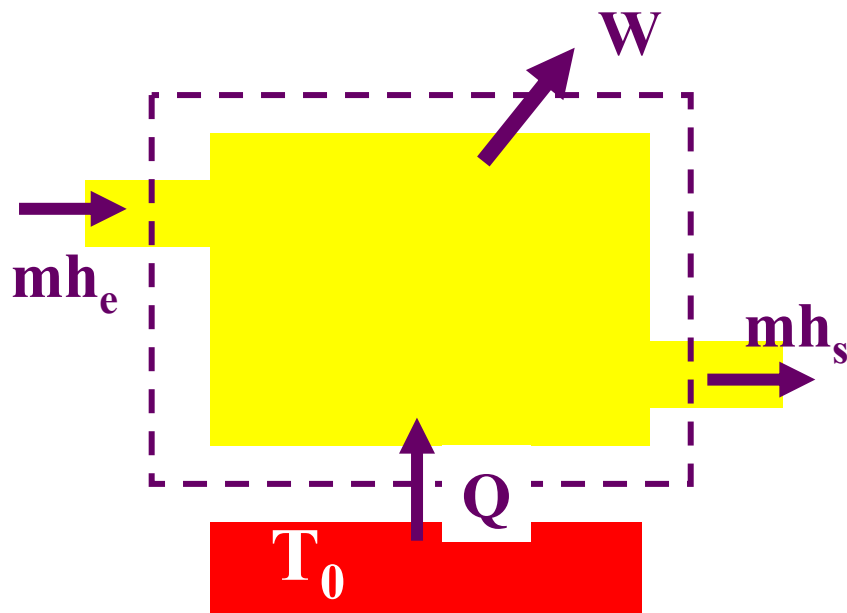
Turbojet Engine Basic Components



Turbojet Engine Basic Components and T-s Diagram for Ideal Turbojet Process



Identifique os fluxos para um Motor a Jato



- Há adição de calor a pressão constante pela queima do combustível.
- A temperatura T_0 é a temperatura da câmara de combustão

$$w = q + (h_e - h_s) - \left(\underbrace{V_I^2 / 2 \Big|_s - V_I^2 / 2 \Big|_e}_{\cong 0} \right)$$