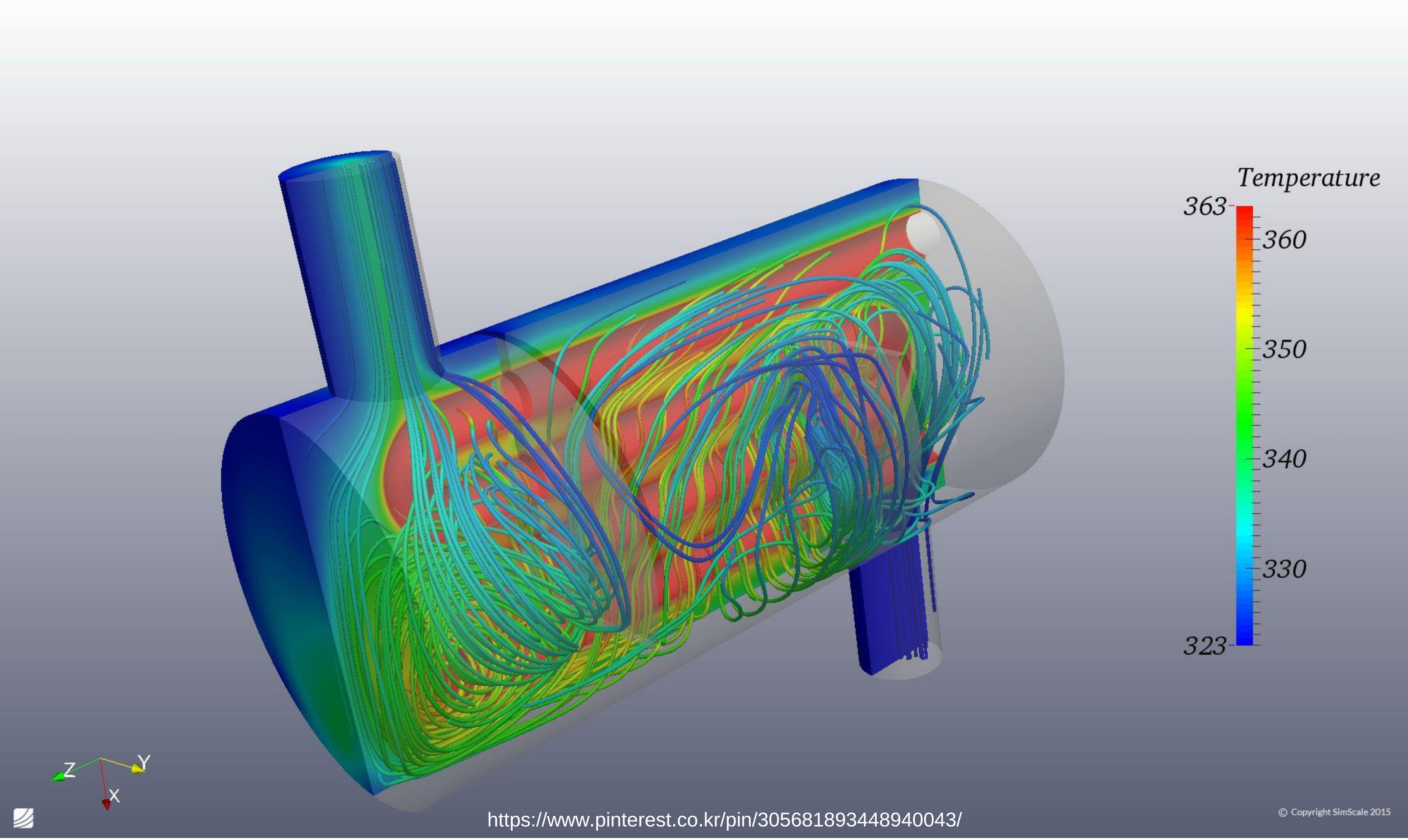
409.319A

Aero Thermo Hydro Engineers Nexus Application

Department of Nuclear Engineering, Seoul National University



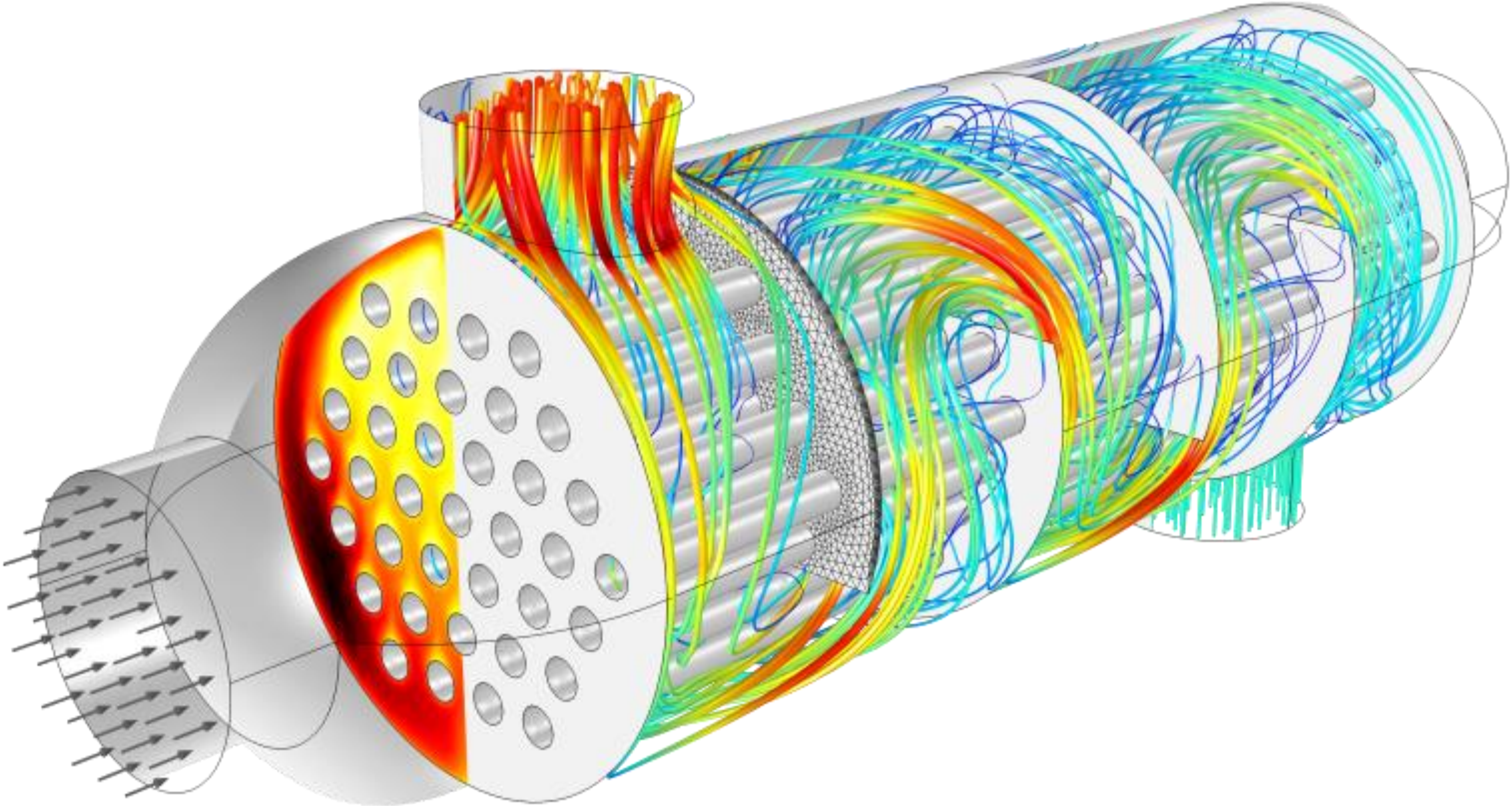
https://www.pinterest.co.kr/pin/305681893448940043/

Thermofluid Plants

409.319A

Aero Thermo Hydro Engineers Nexus Application

Department of Nuclear Engineering, Seoul National University



http://traodoinhietongchum.com/sap-xep-luu-chat-trong-thiet-bi-nhiet/

Thermofluid Plants

A Quick Look

- Thermofluid Components
  - Shaft work machines
  - Nozzles & diffusers
  - Throttles
  - Heat exchangers
- Thermofluid Plants
  - Closed plant: the same fluid processed in a cycle
  - Open (or process) plant: a stream of fluid is processed once through the plant
    - steam power plants (Rankine Cycle)
    - gas turbine power plants (Brayton Cycle)
    - refrigeration plants
    - energy sources
    - thermodynamic plant cycle

Introduction

- steady flow, control volume, 1 inlet- 1exit port
  - 1st

$$\frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_{shaft}}{\dot{m}} = \left(h + \frac{v^2}{2} + gz \right)_{out} - \left(h + \frac{v^2}{2} + gz \right)_{in}$$

  - 2nd

$$s_{out} - s_{in} \geq \sum_j \left(\frac{\dot{Q}/\dot{m}}{T} \right)_j$$

- apply to
  - ideal gas flow, incompressible flow, pure substance 2 phase flow
- apply to each of a classes of steady flow components
  - shaft work machines, nozzles and diffusers
  - throttle, heat exchanger

Shaft Work Machines

- expander or turbine (+) vs compressor or pump (-)
- 1st

$$\frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_{shaft}}{\dot{m}} = h_{out} - h_{in}$$

Adiabatic work transfer is too rapid to attain thermal equilibrium in the fluid

- For a reversible, adiabatic process

$$\therefore -\dot{W}_{shaft} = \dot{m}\left(h_{out} - h_{in}\right)$$

- For an irreversible, adiabatic process

$$\therefore -\dot{W}_{shaft} = \dot{m}\int_{p_{in}}^{p_{out}} vdp$$

$$\therefore \left[\dot{W}_{shaft}\right]_{rev} - \left[\dot{W}_{shaft}\right]_{irrev} = \dot{m}\left[\int_{s_{in}}^{s_{out}} Tds\right]_{p_{out}}$$

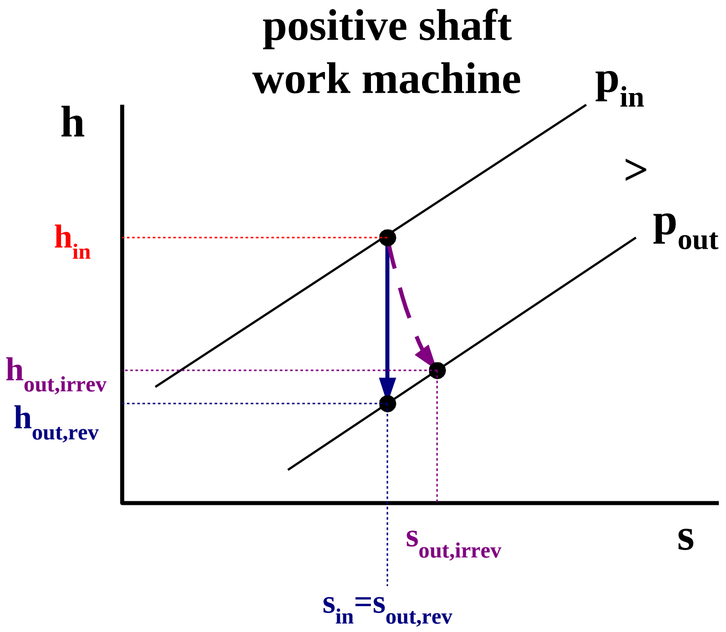
Shaft Work Machines

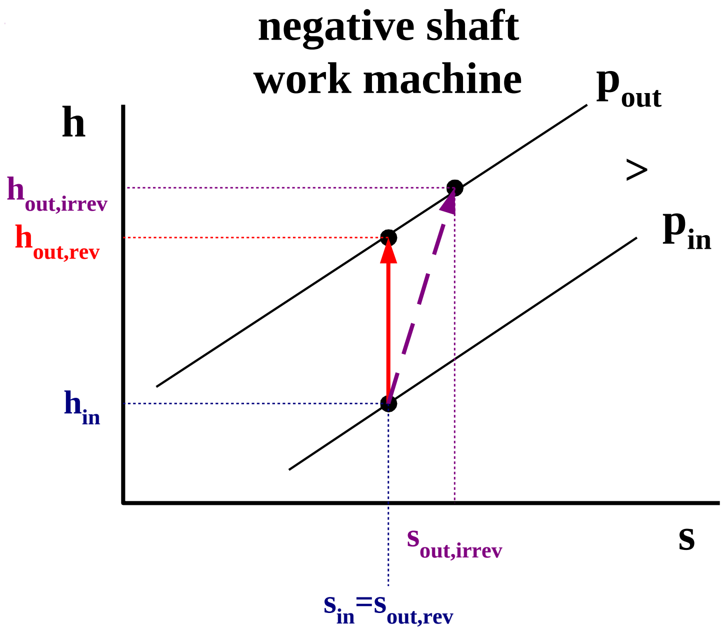
- the adiabatic turbine(or expander) efficiency

$$\eta_{+} = \dot{W}_{act} / \dot{W}_{rev} = \left(h_{in} - h_{out}\right)_{act} / \left(h_{in} - h_{out}\right)_{rev}$$

- the adiabatic compressor(or pump) efficiency

$$\eta_{-} = \dot{W}_{rev} / \dot{W}_{act} = \left(h_{in} - h_{out}\right)_{rev} / \left(h_{in} - h_{out}\right)_{act}$$


positive shaft work machine


negative shaft work machine

Shaft Work Machines

Shaft work machines processing ideal gas

$$-\dot{W}_{shaft} = \dot{m}\left(h_{out} - h_{in}\right) = \dot{m}\int_{p_{in}}^{p_{out}} v\,dp \qquad pv^{\gamma} = const.$$

$$= \frac{\dot{m}\gamma}{\gamma - 1}\left(p_{out}v_{out} - p_{in}v_{in}\right)$$

The equation holds for all adiabatic machines whether reversible or not

$$= \frac{\dot{m}\gamma}{\gamma - 1} p_{in}v_{in}\left(\left(\frac{p_{out}}{p_{in}}\right)^{\frac{\gamma}{\gamma-1}} - 1\right)$$

only for a reversible machine

$$\eta_{+} = \frac{\left(T_{out} - T_{in}\right)_{act}}{\left(T_{out} - T_{in}\right)_{rev}} \qquad \eta_{-} = \frac{\left(T_{out} - T_{in}\right)_{rev}}{\left(T_{out} - T_{in}\right)_{act}}$$

Shaft Work Machines

Shaft work machines processing incompressible fluid

1st
$$-\dot{W}_{shaft} = \dot{m}\left(h_{out} - h_{in}\right)$$
$$= \dot{m}\,c\left(T_{out} - T_{in}\right) + \dot{m}\,v\left(p_{out} - p_{in}\right)$$

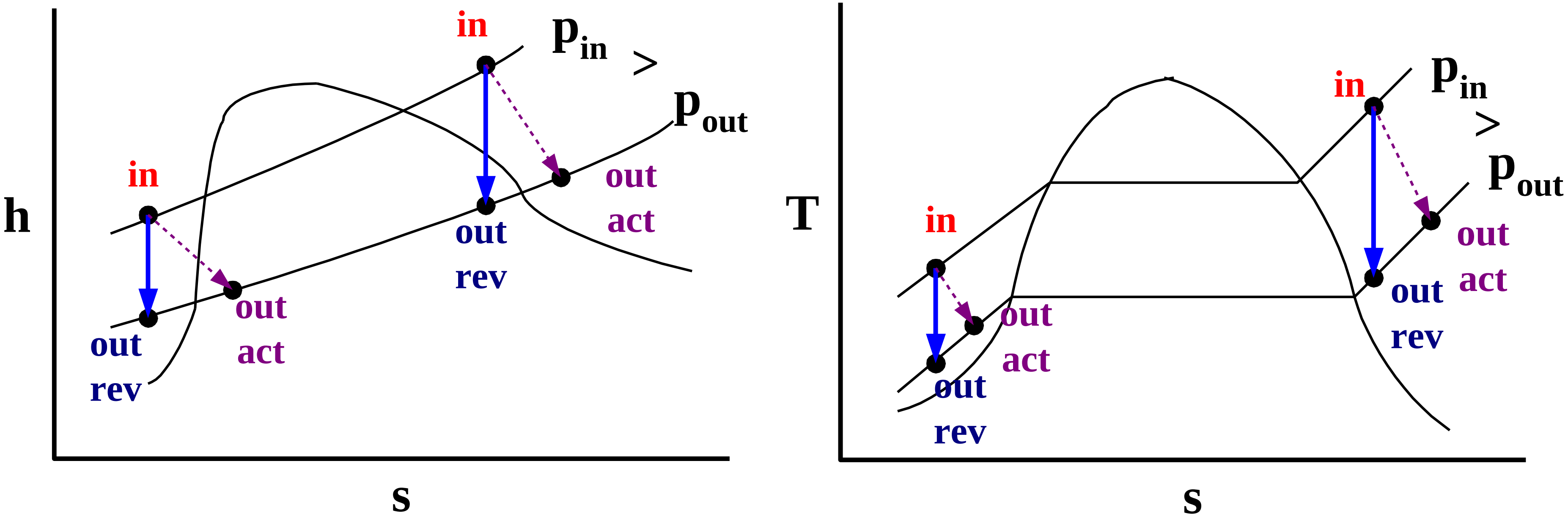
2nd
$$s_{out} - s_{in} = c\ln\frac{T_{out}}{T_{in}} = 0,\quad T_{out} = T_{in}$$
$$\Rightarrow -\dot{W}_{shaft} = \dot{m}\,v\left(p_{out} - p_{in}\right)$$

for a reversible adiabatic machine

$$\eta_{+} = \frac{\dot{W}_{act}}{\dot{W}_{rev}} = 1 + \frac{c\left(T_{out} - T_{in}\right)}{v\left(p_{out} - p_{in}\right)} \qquad \eta_{-} = \frac{\dot{W}_{rev}}{\dot{W}_{act}} = 1 - \frac{c\left(T_{out} - T_{in}\right)}{c\left(T_{out} - T_{in}\right) + v\left(p_{out} - p_{in}\right)}$$

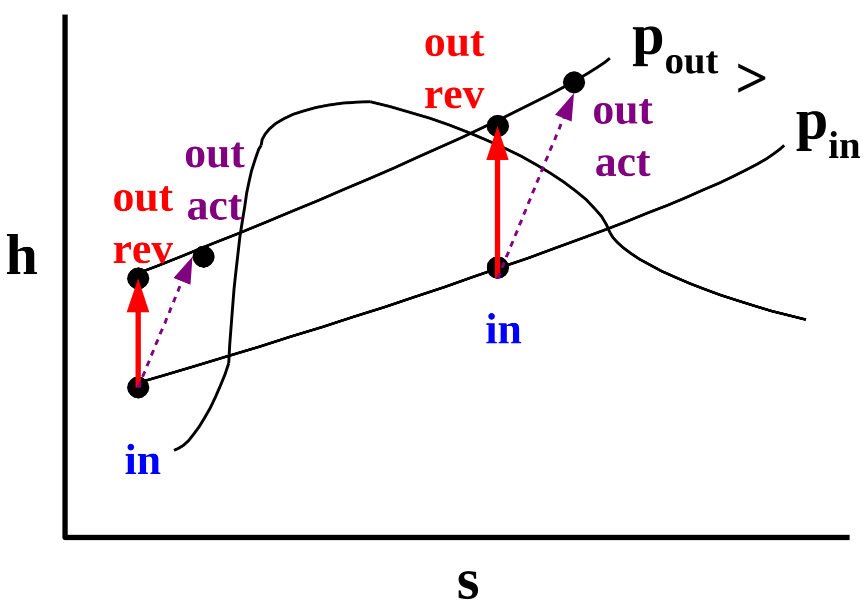
Shaft Work Machines

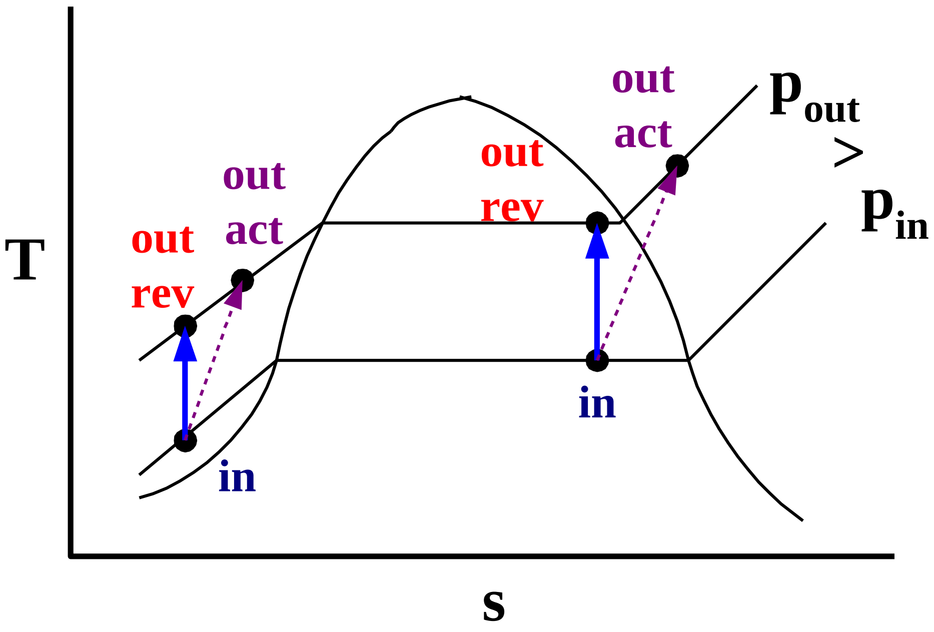
- Shaft work machines processing a pure substance 2 phase flow
  - positive shaft work machine



Shaft Work Machines

- Shaft work machines processing a pure substance 2 phase flow
  - negative shaft work machine





Nozzles and Diffusers

Nozzles

adiabatic
no shaft work
steady state

low v
low KE
high p
→
converging
diverging nozzle
→
high v
high KE
low p

$$0 = \sum_{in} \dot{m}\left(h + \frac{v^2}{2} + gz \right) - \sum_{out} \dot{m}\left(h + \frac{v^2}{2} + gz \right)$$

$$\frac{v_{out}^2 - v_{in}^2}{2} = h_{in} - h_{out}$$

$$\frac{v_{out}^2 - v_{in}^2}{2} = \left(-\int_{p_{in}}^{p_{out}} v\,dp \right)_{s_{in}} \quad (reversible)$$

$$\frac{v_{out}^2 - v_{in}^2}{2} = -\int_{p_{in}}^{p_{out}} v\,dp - \int_{s_{in}}^{s_{out}} T\,ds \quad (actual)$$

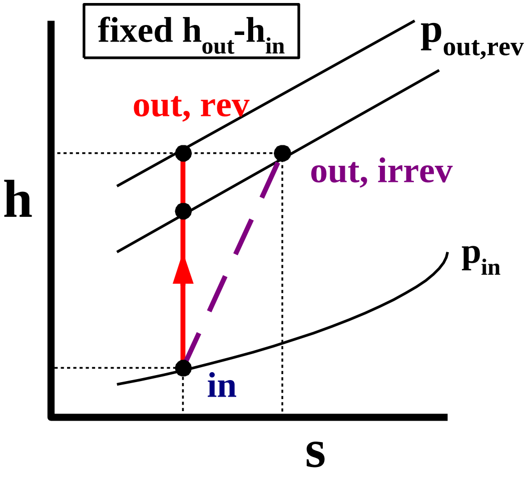
$$\eta_N = \frac{\left(v_{out}^2 - v_{in}^2\right)_{act}}{\left(v_{out}^2 - v_{in}^2\right)_{rev}}$$

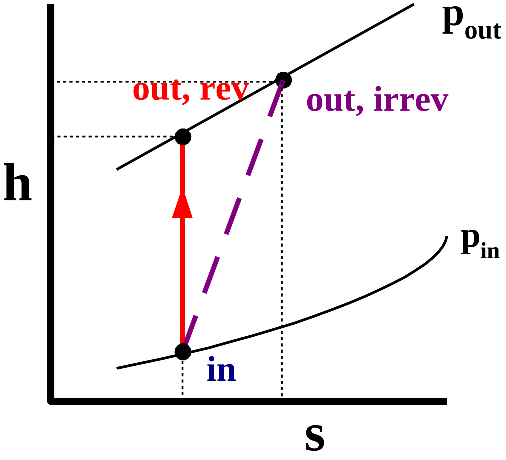
$$\eta_N = \frac{\left(T_{out} - T_{in}\right)_{act}}{\left(T_{out} - T_{in}\right)_{rev}} \quad \rightarrow ideal\ \ gas$$

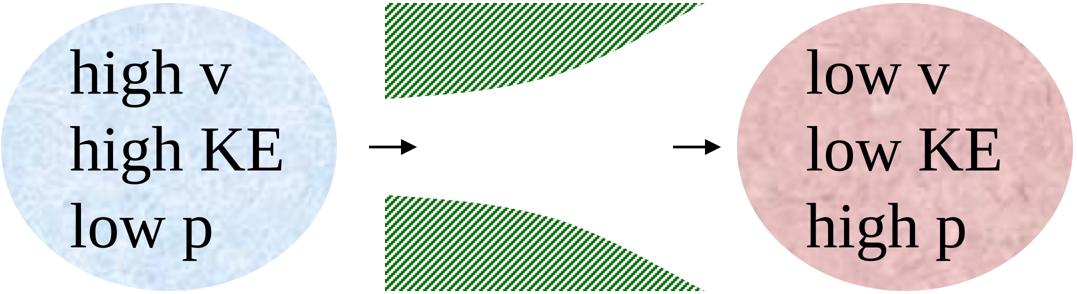
$$\eta_N = 1 + \frac{c\left(T_{in} - T_{out}\right)}{v\left(p_{in} - p_{out}\right)} \quad \rightarrow incompressible\ \ fluid$$

Nozzles and Diffusers

Diffusers







high v
high KE
low p

low v
low KE
high p

Essentially the inverse of the nozzle
Deceleration is inherently unstable

$$\frac{v_{out}^2 - v_{in}^2}{2} = h_{in} - h_{out} \qquad \eta_D = \frac{\left(h_{in} - h_{out}\right)_{rev}}{\left(h_{in} - h_{out}\right)_{act}}$$

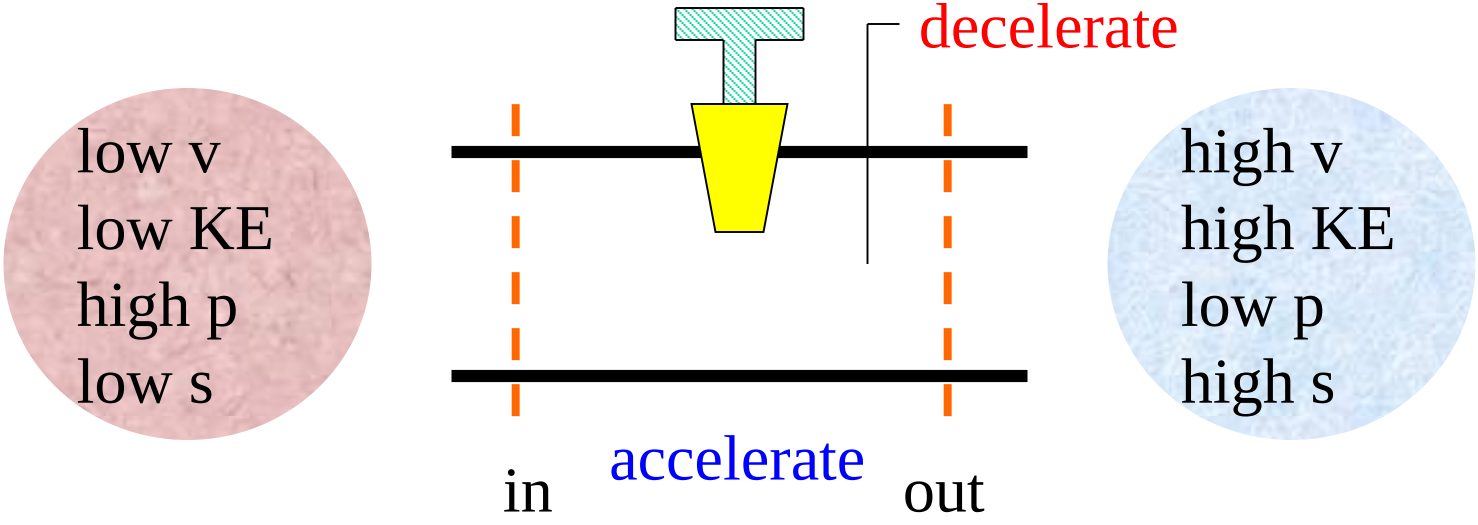
$$\eta_D = \frac{\left(T_{out} - T_{in}\right)_{rev}}{\left(T_{out} - T_{in}\right)_{act}}$$

ideal gas

$$\eta_D = \frac{c\left(T_{out} - T_{in}\right)}{c\left(T_{out} - T_{in}\right) + v\left(p_{out} - p_{in}\right)}$$

incompressible fluid

Throttles



low v
low KE
high p
low s

high v
high KE
low p
high s

- a device also known as control valve, expansion valve, Joule-Thomson valve, etc. that controls pressures of a flowing stream
- looks like a nozzle/diffuser in a series

$$h_{in} - h_{out} = \frac{v_{out}^2 - v_{in}^2}{2}$$

Throttles

- Typically values of the inlet and outlet let v's are such that KE's are negligible compared to enthalpy

$$v_{in}^2 \cong v_{out}^2 \cong 0 << h, \quad h_{in} \cong h_{out} \quad \rightarrow \quad dh = 0$$

$$dh = Tds + vdp = 0 \quad (1st \quad law) \quad \rightarrow Tds = -vdp$$

- Since dp<0, then Tds>0 or ds>0 thus the process irreversible
  - This is the only engineering device that calls for large ds>0 for high efficiency

ideal gas $\quad T_{out} = T_{in}$

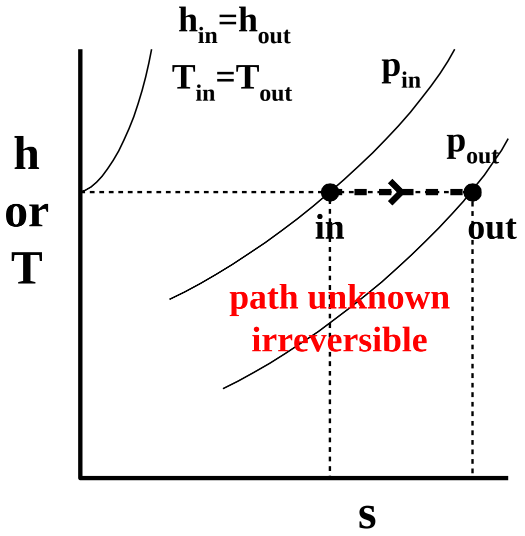
incompressible fluid

$$h_{out} = h_{in}\,, \quad s_{out} - s_{in} = c \ln \frac{T_{out}}{T_{in}} > 0$$

$$(u + pv)_{out} = (u + pv)_{in} \quad or \quad T_{out} > T_{in}$$

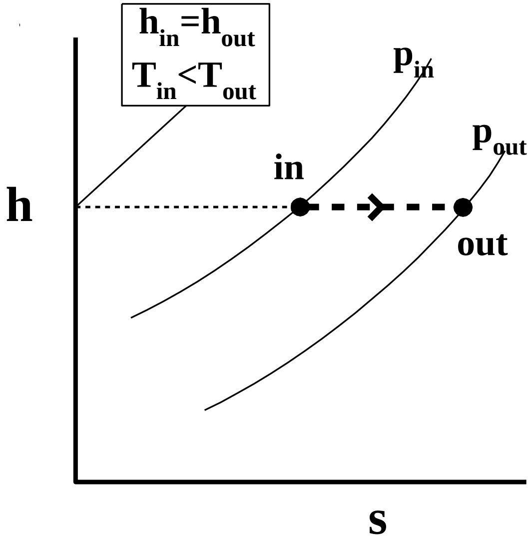
➢ Throttles

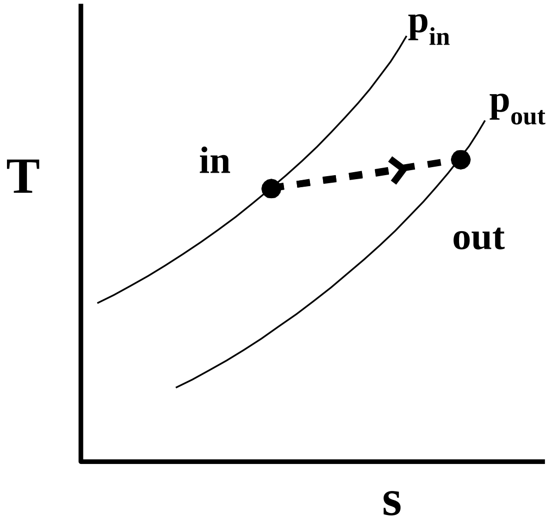
ideal gas



The source of this T increase is the irreversibility in the diffuser section of the throttle

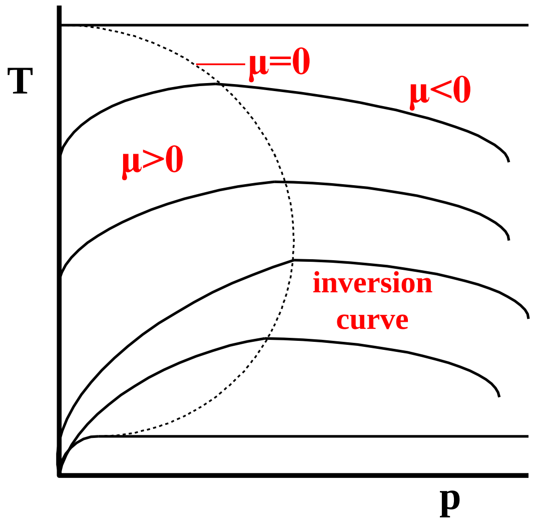
incompressible fluid





Throttles

- Different from either IG or IF
- The pure substance T may increase or decrease



- given dp<0

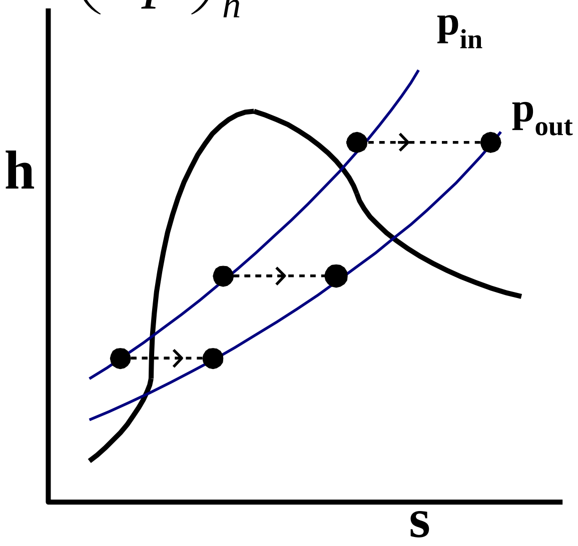
Joule-Thomson coefficient $$\mu = \left(\frac{\partial T}{\partial p}\right)_h$$

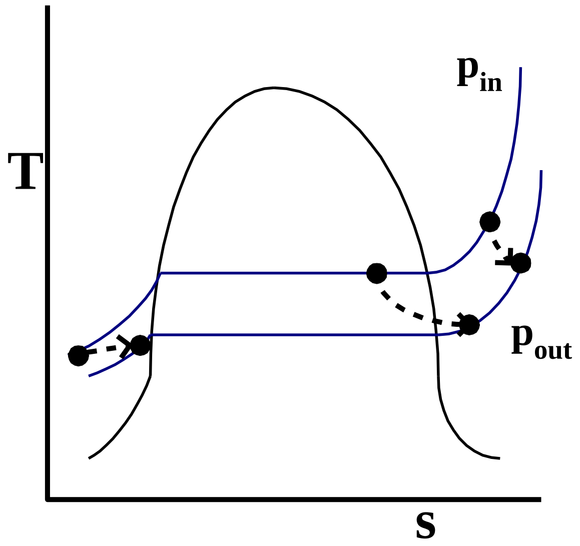
$\mu > 0,\ dT < 0\,(refrigeration)$

$\mu = 0,\ dT = 0\,(adiabatic)$

$\mu < 0,\ dT > 0\,(heatup)$

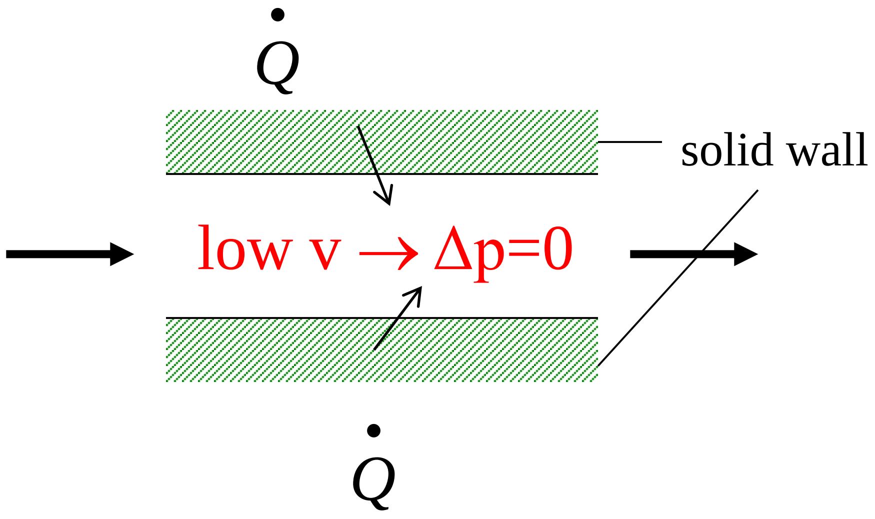
Even in the absence of throttle valves, the pipe flow will experience Δp due to viscous shear, which is completely analgous to the mechanism of throttle





Heat Exchangers

-Fluid moves through device relatively slowly (Δp is small) so the residence time of fluid within the device is large



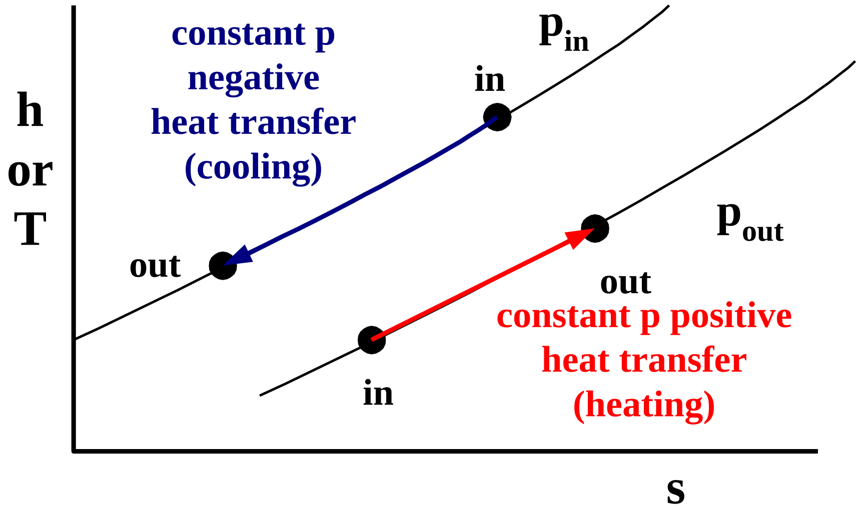
$$\Delta KE \approx 0,\ \Delta PE \approx 0,\ \dot{W}_{shaft} \approx 0$$

$$1st\ law \quad \dot{Q} = \dot{m}\left(h_{out} - h_{in}\right),\ \left(\delta\ \dot{Q}\right)/\dot{m} = dh$$

Ideal Gas

$$dh = c_p dT$$

$$\dot{Q} = \dot{m}\, c_p \left(T_{out} - T_{in}\right)$$



Heat Exchangers

Incompressible fluid (v constant)

$$\dot{Q} = \dot{m} c (T_{out} - T_{in}), \qquad Now \;\; dh = Tds + vdp = Tds$$

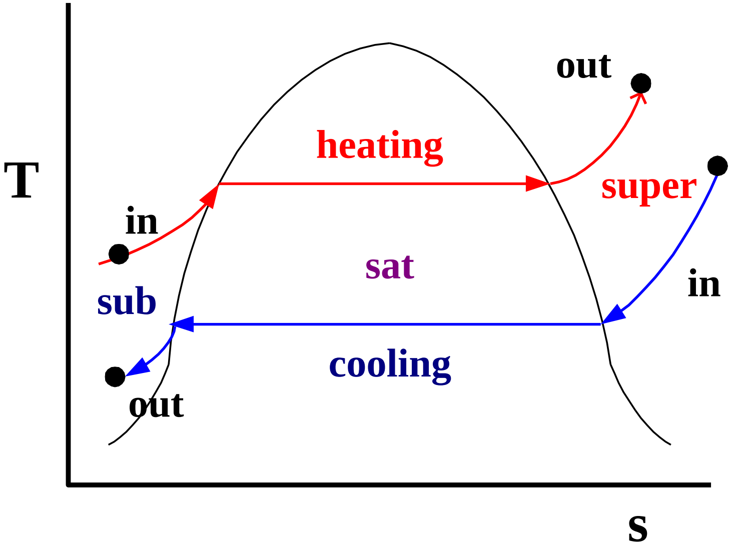
$$If \quad dh > 0 \; (i.e. heating), \; ds > 0$$

$$dh < 0 \; (i.e. cooling), \; ds < 0$$

Pure substance

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

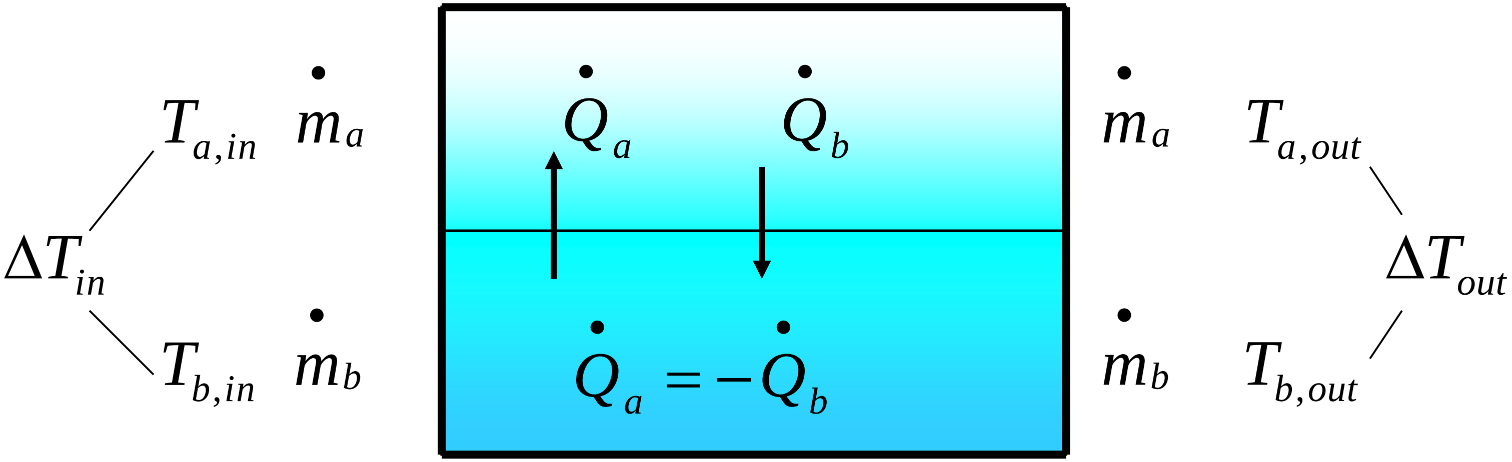
$$= \dot{m}[c_g (T_{out} - T_{sat}) + \Delta h_{fg} + c_f (T_{sat} - T_{in})$$



Heat Exchangers

Two-stream heat exchangers (adiabatic)

parallel flow



ΔT_{in}: $T_{a,in}$ $\dot m_a$, $T_{b,in}$ $\dot m_b$

$\dot Q_a$ $\dot Q_b$

$$\dot Q_a = -\dot Q_b$$

$\dot m_a$ $T_{a,out}$, $\dot m_b$ $T_{b,out}$: ΔT_{out}

$$\dot m h_{in} = \dot m h_{out}\ , \qquad \dot m_a \left(h_{a,in} - h_{a,out}\right) = \dot m_b \left(h_{b,out} - h_{b,in}\right)$$

$$\dot Q_a = \dot m_a \left(h_{a,in} - h_{a,out}\right) = \dot m_a c_{p,a} \left(T_{a,in} - T_{a,out}\right)$$

$$= \dot m_b \left(h_{b,out} - h_{b,in}\right) = \dot m_b c_{p,b} \left(T_{b,out} - T_{b,in}\right) = -\dot Q_b$$

For const. p

Heat Exchangers

- Simplest model for kinetics
  - Approach to thermal equilibrium depends upon the resistance to heat transfer between the streams

$$\dot{Q}_b = U\left(T_a - T_b\right)A_T$$

$$\Delta T_{LM} = \left(\frac{\Delta T_1 - \Delta T_2}{\ln\left(\Delta T_1 / \Delta T_2\right)}\right) \quad (\textit{regardless of flow direction})$$

$$\dot{Q}_b = U\Delta T_{LM} A_T \quad \textit{for both parallel} + \textit{counterflow}$$

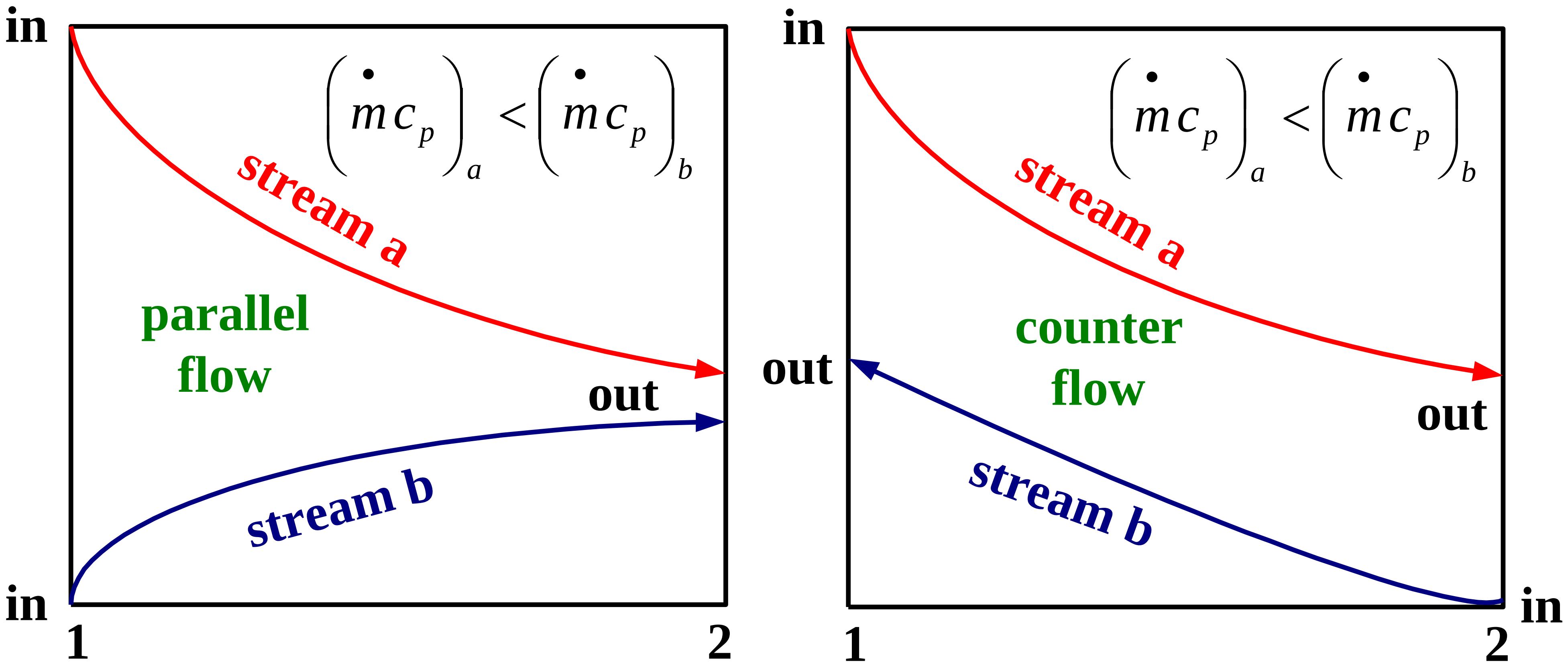
U: overall heat transfer coefficients

A_T: heat exchanger surface area normal to Q

T_a-T_b=LMTD (logarithmic mean temperature difference)

Heat Exchangers

Simplest model for kinetics



Heat Exchangers

Temperature distribution for the two streams

parallel flow

$$T_b - T_{b1} = \frac{T_{a1} - T_{b1}}{1 + \dot{m}_b c_{pb} / \dot{m}_a c_{pa}} \left[1 - \exp\left\{ -\left(\frac{1}{\dot{m}_a c_{pa}} + \frac{1}{\dot{m}_b c_{pb}} \right) UA \right\} \right]$$

$$T_{a1} - T_a = \frac{\dot{m}_b c_{pb}}{\dot{m}_a c_{pa}} \frac{T_{a1} - T_{b1}}{1 + \dot{m}_b c_{pb} / \dot{m}_a c_{pa}} \left[1 - \exp\left\{ -\left(\frac{1}{\dot{m}_a c_{pa}} + \frac{1}{\dot{m}_b c_{pb}} \right) UA \right\} \right]$$

counter flow

$$T_b - T_{b1} = \frac{T_{a1} - T_{b1}}{\dot{m}_b c_{pb} / \dot{m}_a c_{pa} - 1} \left[1 - \exp\left\{ -\left(\frac{1}{\dot{m}_a c_{pa}} - \frac{1}{\dot{m}_b c_{pb}} \right) UA \right\} \right]$$

$$T_{a1} - T_a = \frac{\dot{m}_b c_{pb}}{\dot{m}_a c_{pa}} \frac{T_{a1} - T_{b1}}{\dot{m}_b c_{pb} / \dot{m}_a c_{pa} - 1} \left[1 - \exp\left\{ -\left(\frac{1}{\dot{m}_a c_{pa}} - \frac{1}{\dot{m}_b c_{pb}} \right) UA \right\} \right]$$

Heat Exchangers

Heat exchanger effectiveness

$$\varepsilon = \frac{\left|\dot{Q}_{act}\right|}{\left|\dot{Q}_{\max}\right|}$$

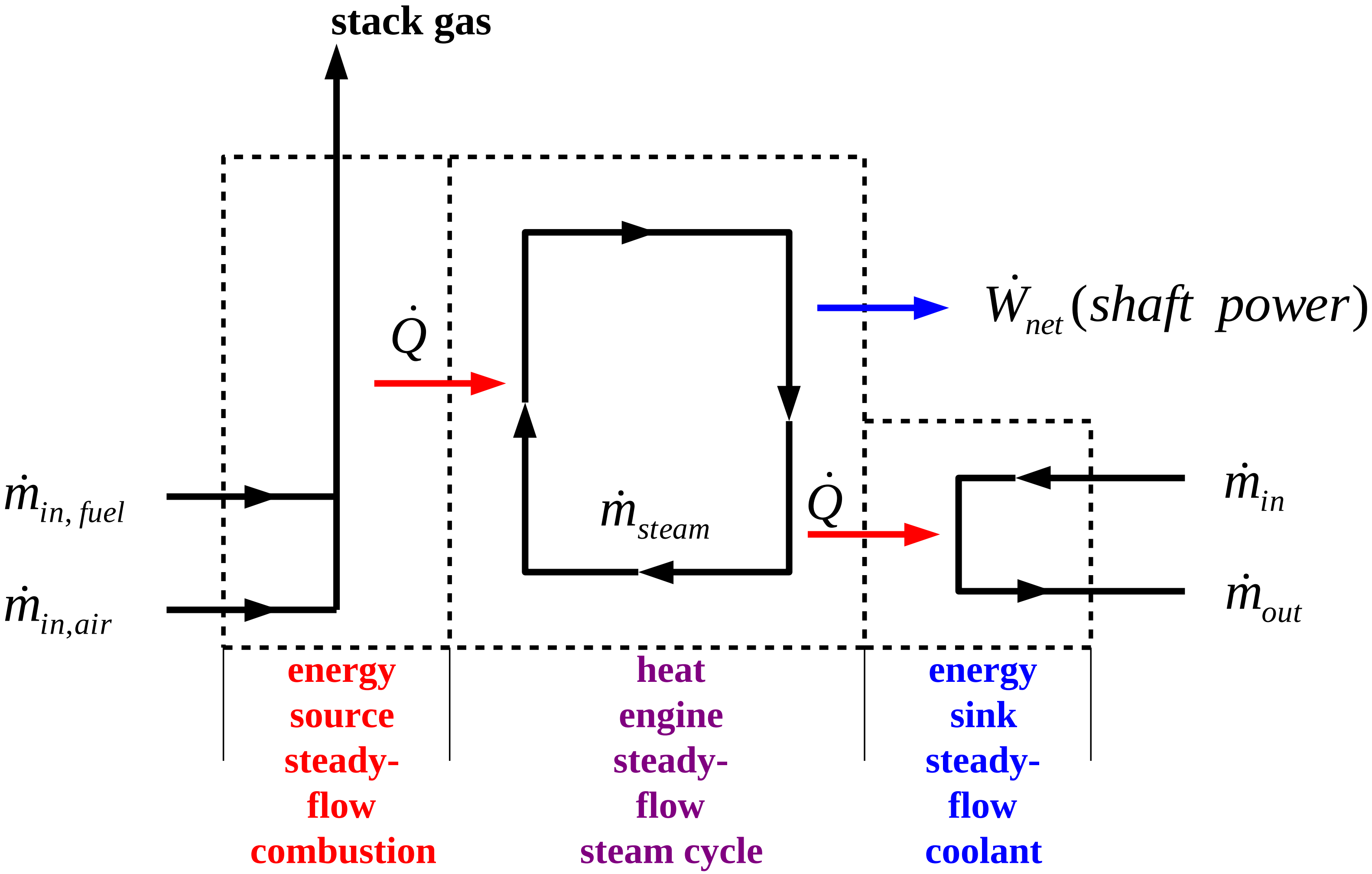
$$= \frac{\left|\dot{m}_a c_{pa}\left(T_{a,out} - T_{a,in}\right)\right|}{\left|\left(\dot{m}c_p\right)_{\min}\left(T_{a,in} - T_{b,in}\right)\right|}$$

$$= \frac{\left|\dot{m}_b c_{pb}\left(T_{b,out} - T_{b,in}\right)\right|}{\left|\left(\dot{m}c_p\right)_{\min}\left(T_{a,in} - T_{b,in}\right)\right|}$$

Approaches unity only for a counter flow exchanger with equal capacity rates and an infinite UA product
(or ΔT=0 throughout).

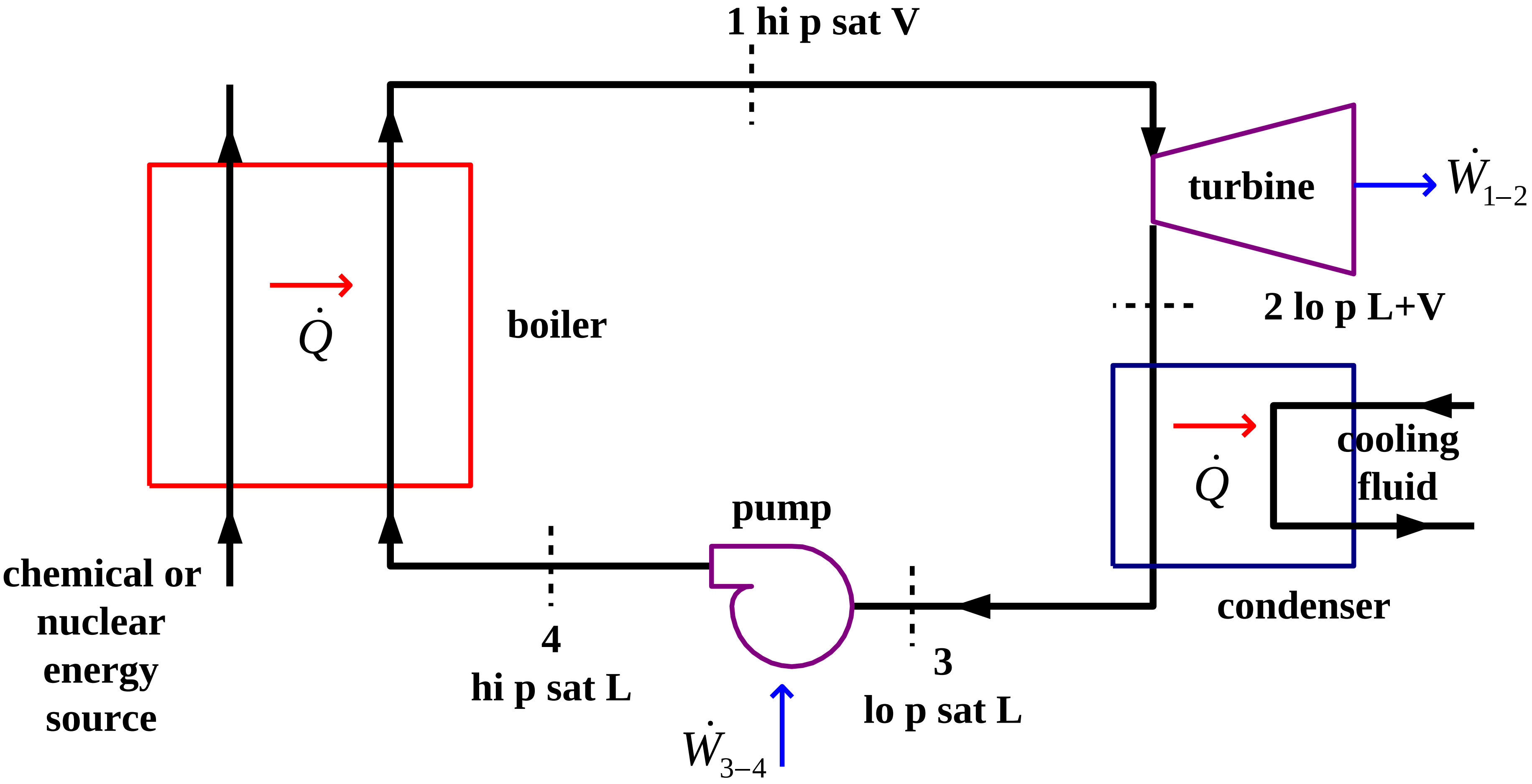
Temperature effectiveness equation

Steam Power Plants



Steam Power Plants

Rankine-Cycle Steam Plant



Steam Power Plants

Rankine-Cycle Steam Plant

Engine Power	$\dot{W}_{1-2} = \dot{m}(h_1 - h_2)$
Condenser Load	$\dot{Q}_{2-3} = \dot{m}(h_3 - h_2) = \dot{m}T_3(s_3 - s_2)$
Pump Power	$\dot{W}_{3-4} = \dot{m}(h_3 - h_4)$
Boiler Load	$\dot{Q}_{1-4} = \dot{m}(h_1 - h_4)$

Energy conversion efficiency

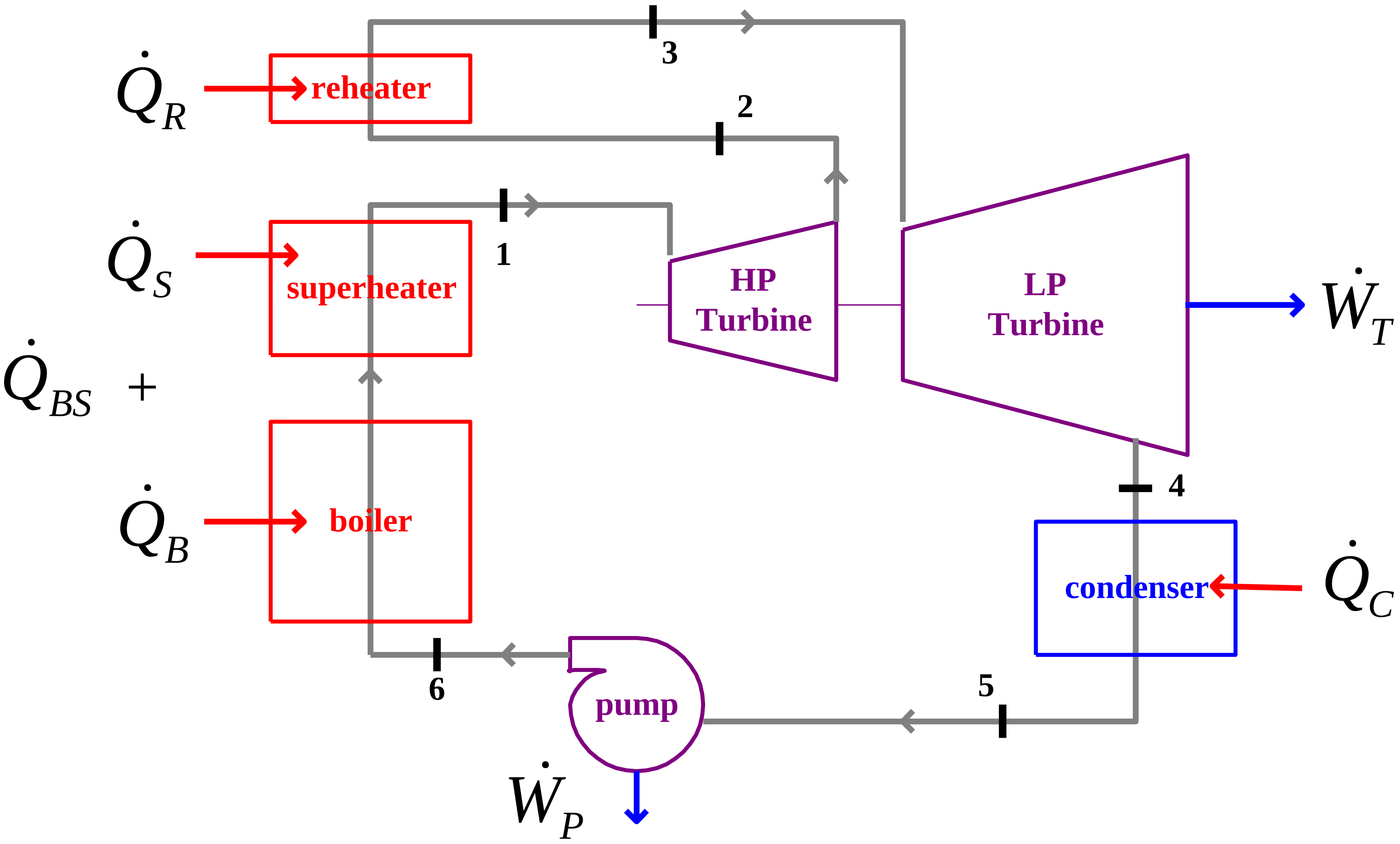
$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{boiler}} = \frac{\dot{W}_{1-2} + \dot{W}_{3-4}}{\dot{Q}_{4-1}} = 1 - \frac{h_2 - h_3}{h_1 - h_4}$$

Net work ratio

$$NWR = \frac{\dot{W}_{net}}{\dot{W}_{+ve}} = 1 - \frac{h_4 - h_3}{h_1 - h_2}$$

Steam Power Plants

Superheat-, Reheat-, Regenerative Rankine Cycle



Steam Power Plants

Superheat-, Reheat-, Regenerative Rankine Cycle

Engine Power

$$\dot{W}_T = \dot{m}\left(h_1 - h_2\right) + \dot{m}\left(h_3 - h_4\right)$$

Condenser Load

$$\dot{Q}_C = \dot{m}\left(h_5 - h_4\right) = \dot{m}T_5\left(s_5 - s_4\right)$$

Pump Power

$$\dot{W}_P = \dot{m}\left(h_5 - h_6\right)$$

Boiler+ superheater Load

$$\dot{Q}_{BS} = \dot{m}\left(h_1 - h_6\right)$$

Reheater Load

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

Energy conversion efficiency

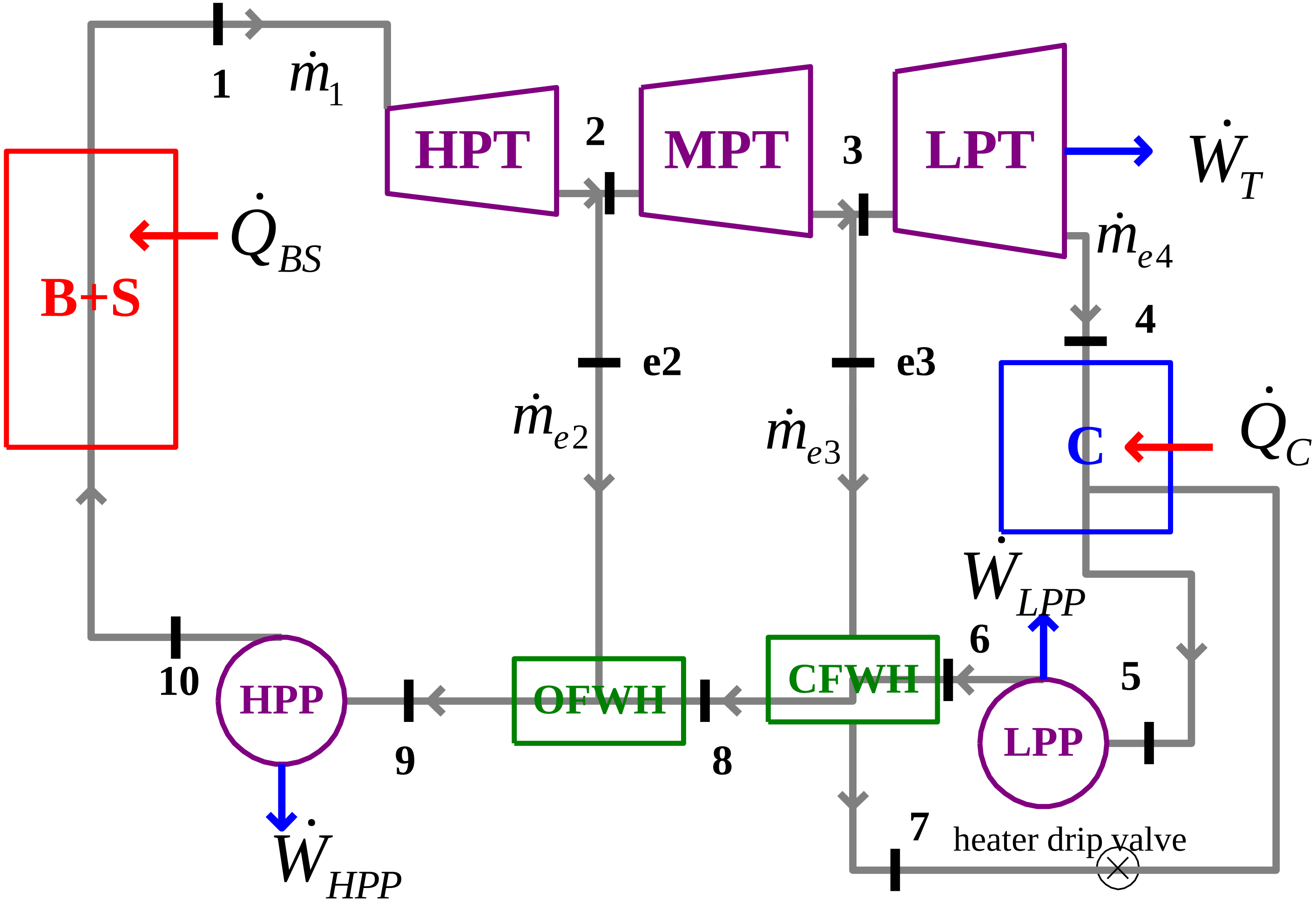
$$\eta = \dot{W}_{net} / \dot{Q}_{total} = \left(\dot{W}_P + \dot{W}_T\right) / \left(\dot{Q}_{BS} + \dot{Q}_R\right) = 1 - \frac{h_4 - h_5}{\left(h_1 - h_6\right) + \left(h_3 - h_2\right)}$$

Net work ratio

$$NWR = \dot{W}_{net} / \dot{W}_{+ve} = 1 - \frac{h_4 - h_3}{\left(h_1 - h_2\right) + \left(h_3 - h_4\right)}$$

Steam Power Plants

Regenerative Feedwater Heaters



Steam Power Plants

Regenerative Feedwater Heaters

$$\dot{m}_1 = \dot{m}_{e2} + \dot{m}_{e3} + \dot{m}_{e4} \quad \text{continuity}$$

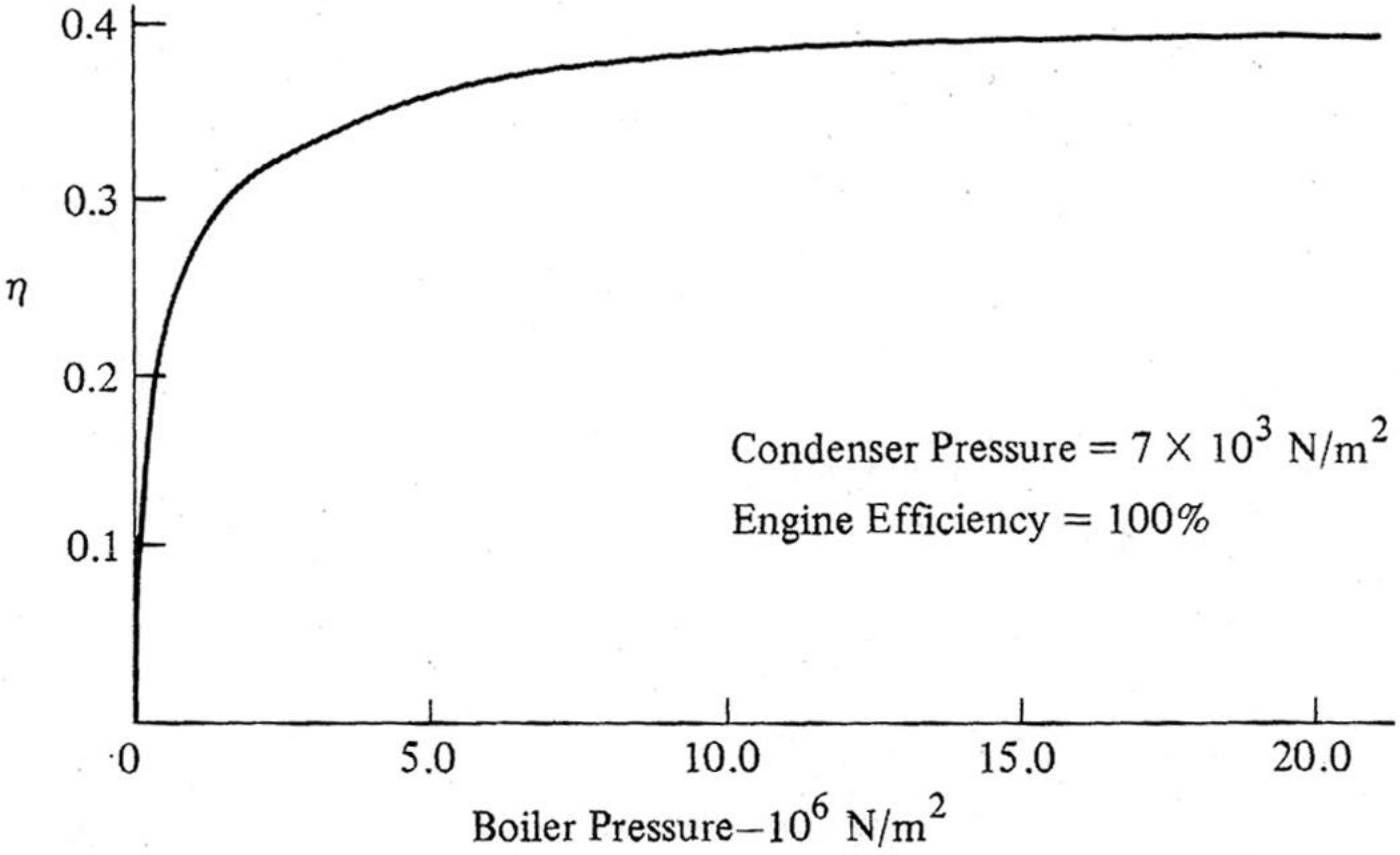
$$\dot{m}_{e3}\left(h_3 - h_7\right) = \left(\dot{m}_{e3} + \dot{m}_{e4}\right)\left(h_8 - h_6\right) \quad \text{CFWH}$$

$$\dot{m}_{e2}\left(h_2 - h_9\right) = \left(\dot{m}_1 - \dot{m}_{e2}\right)\left(h_9 - h_8\right) \quad \text{OFWH}$$

$$\begin{aligned}\dot{W}_T &= \dot{m}_1\left(h_1 - h_2\right) + \left(\dot{m}_1 - \dot{m}_{e2}\right)\left(h_2 - h_3\right) \\ &\quad + \left(\dot{m}_1 - \dot{m}_{e2} - \dot{m}_{e3}\right)\left(h_2 - h_3\right)\end{aligned}$$

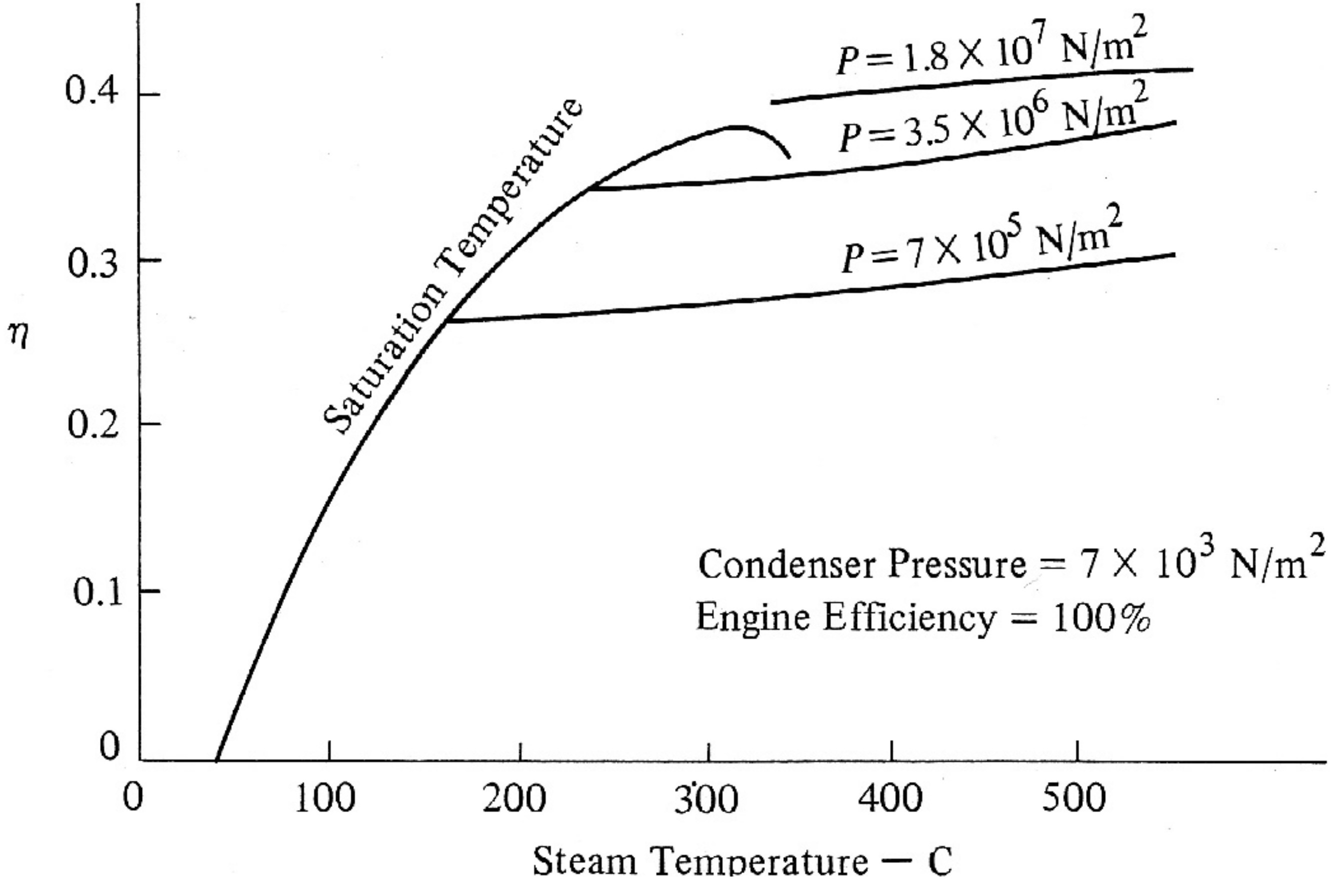
Steam Power Plants

Influence of boiler pressure on Rankine cycle efficiency



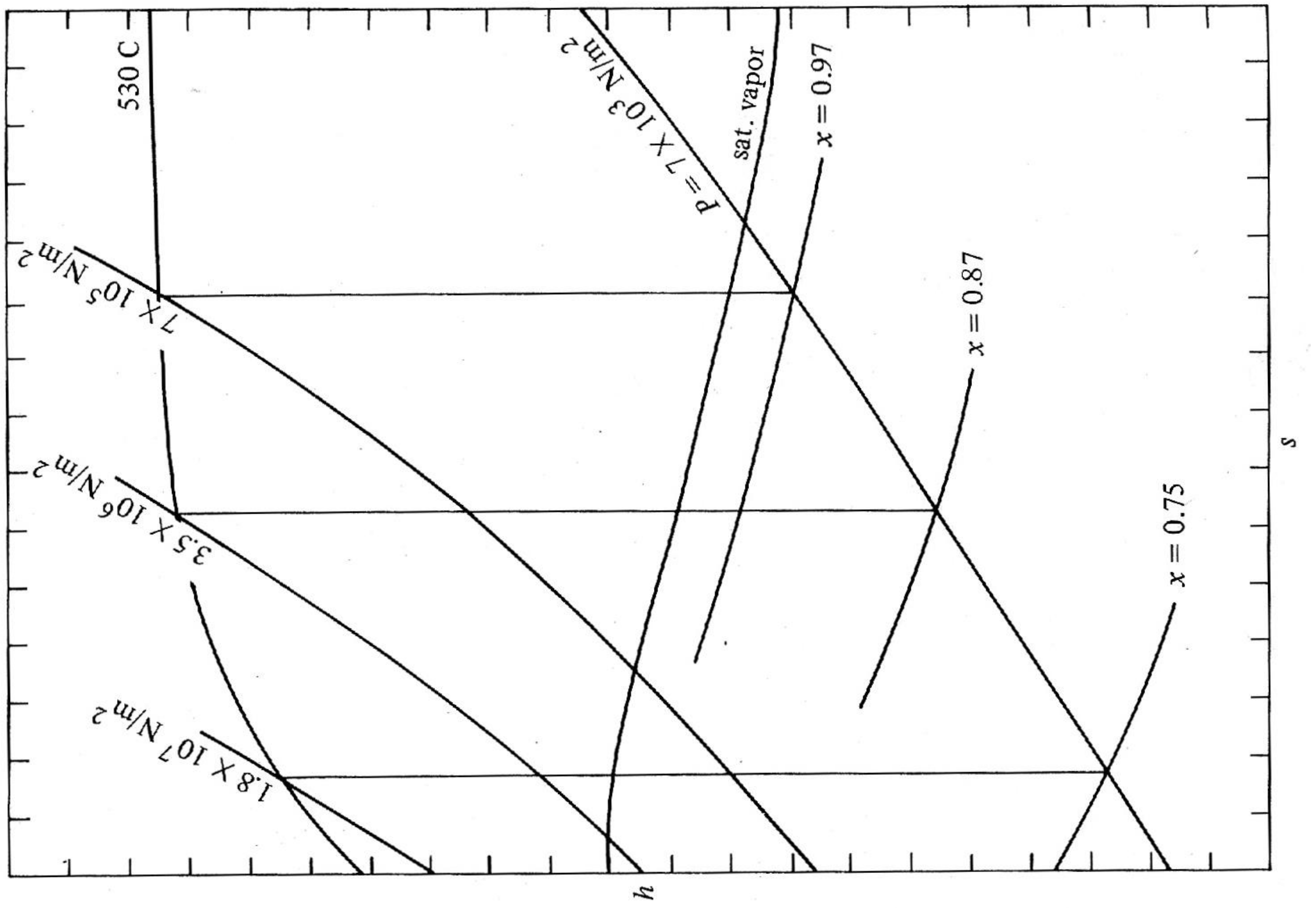
Steam Power Plants

Influence of superheat on Rankine cycle efficiency



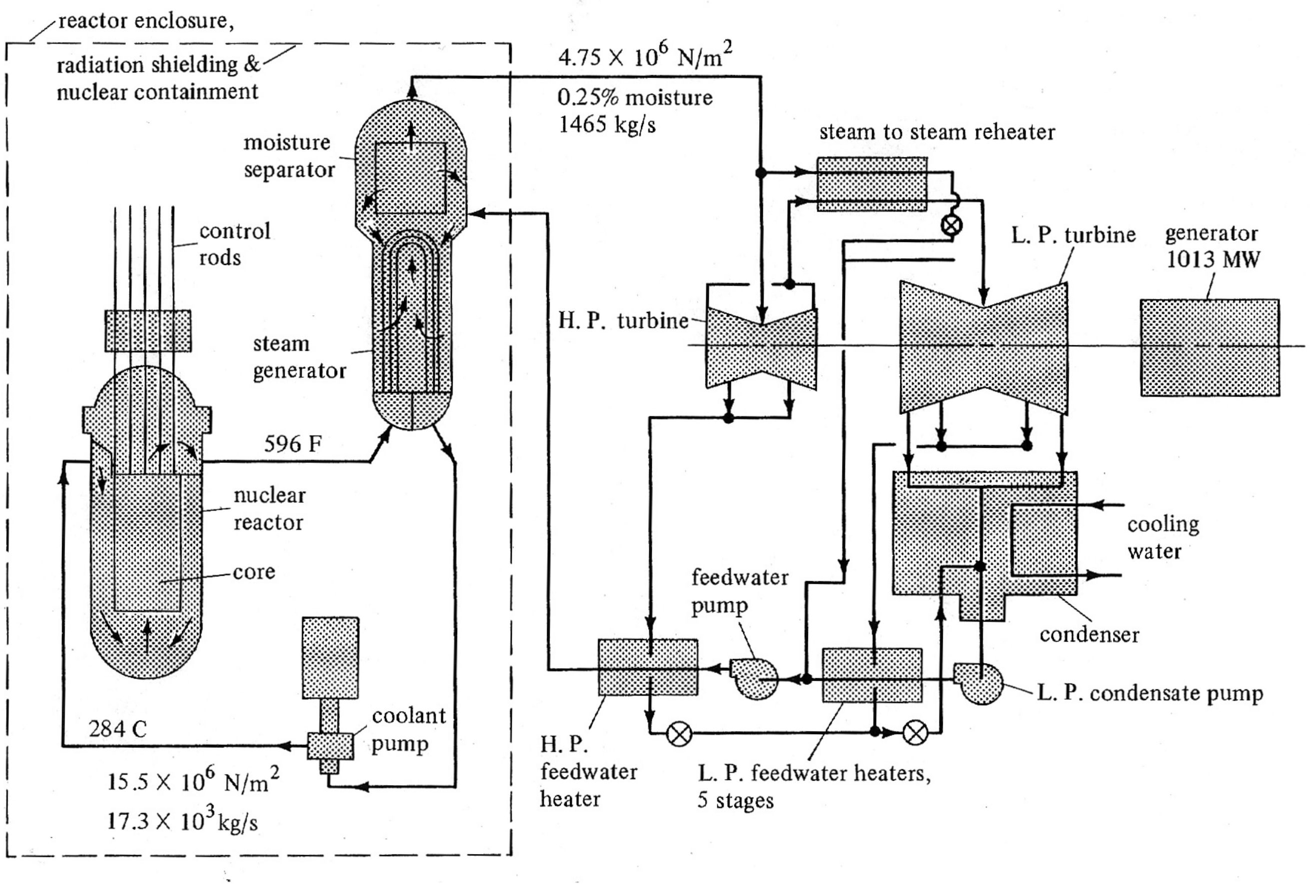
Steam Power Plants

Influence of boiler pressure on quality of steam at turbine exit



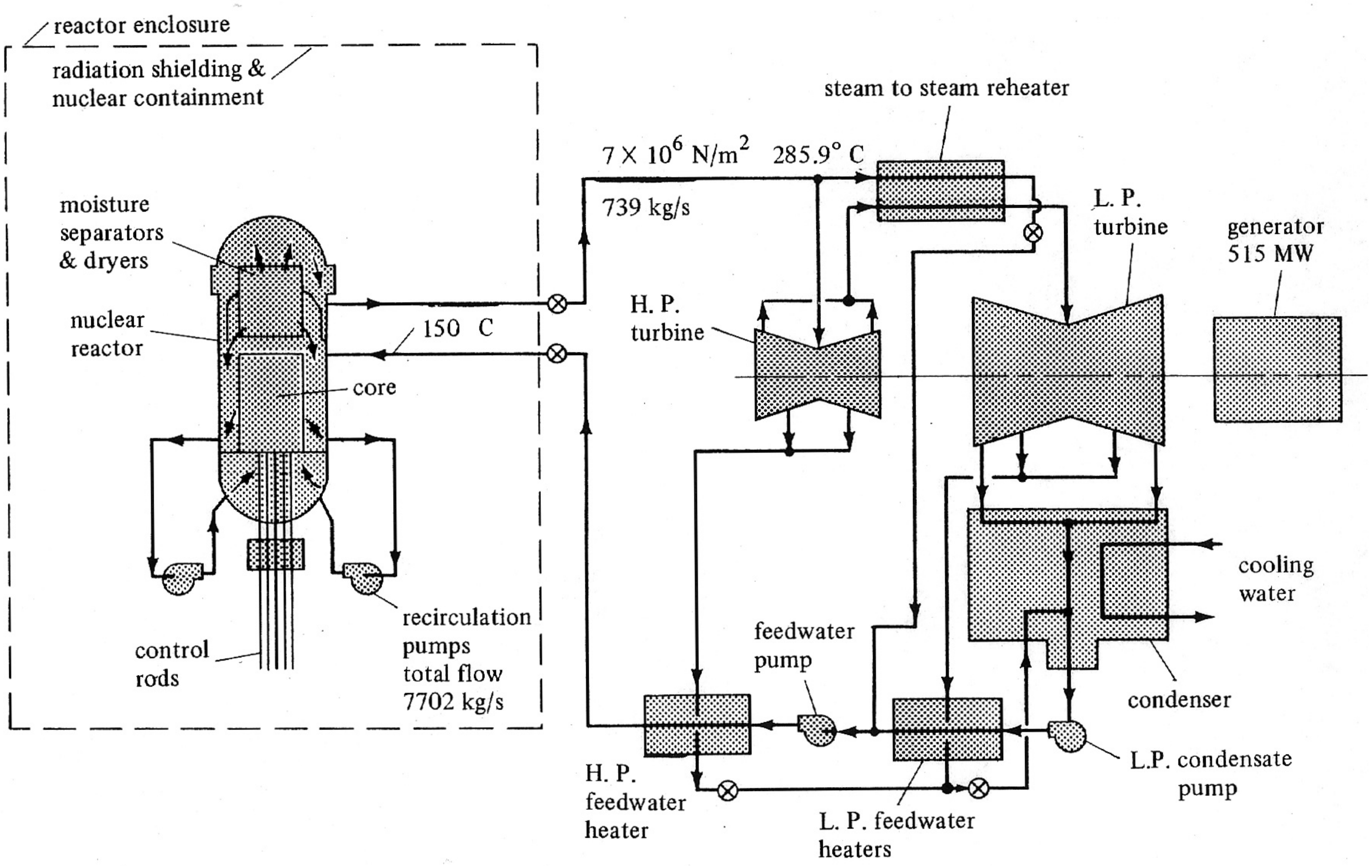
Steam Power Plants

Nuclear powered steam plant with a pressurized water reactor

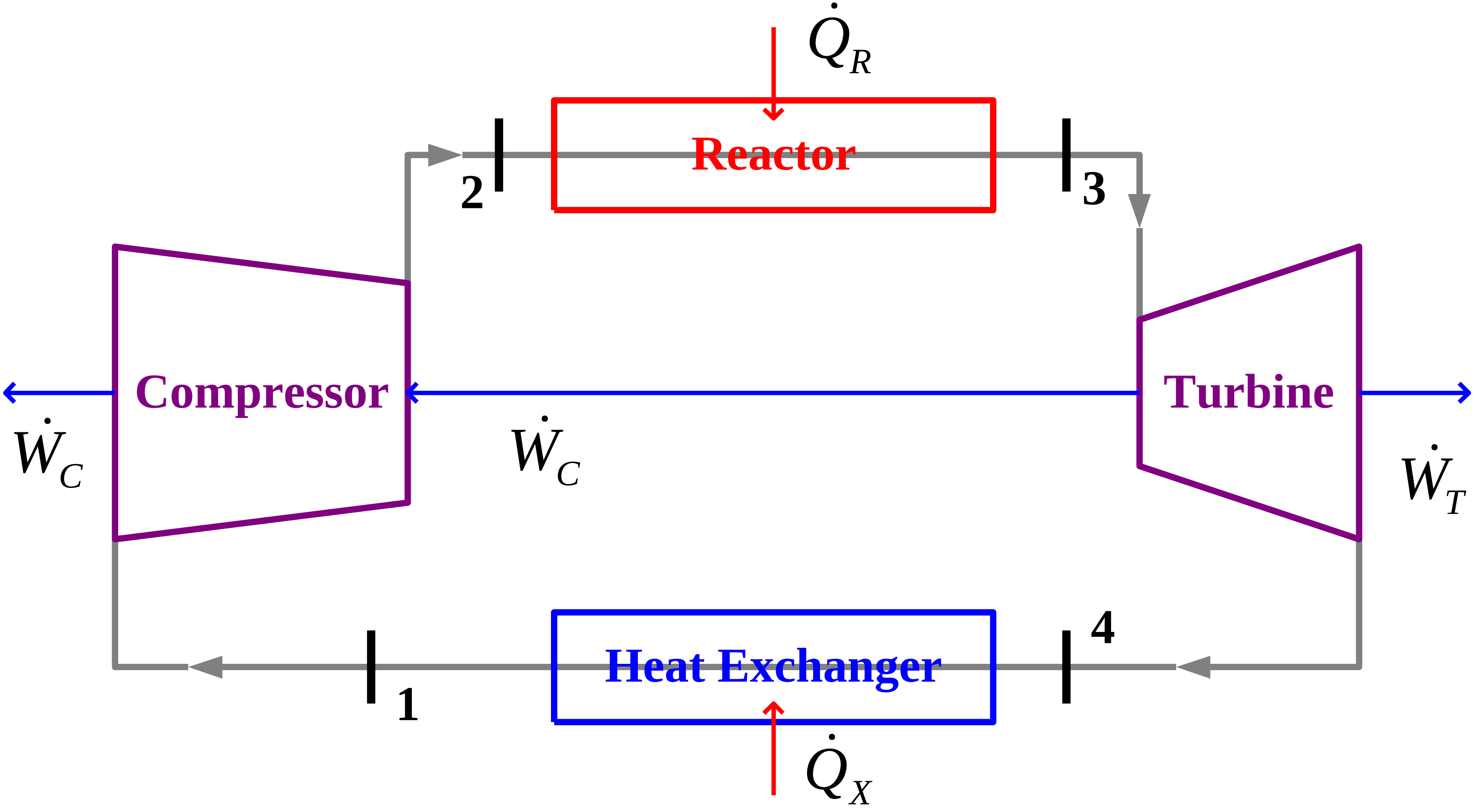


Steam Power Plants

Nuclear powered steam plant with a boiling water reactor



➢ Gas Turbine Power Plants



Gas Turbine Power Plants

Summary

Compressor power $$\dot{W}_C = \dot{m}(h_1 - h_2) = \dot{m}c_p(T_1 - T_2)$$

Heat addition $$\dot{Q}_R = \dot{m}(h_3 - h_2) = \dot{m}c_p(T_3 - T_2)$$

Turbine power $$\dot{W}_T = \dot{m}(h_3 - h_4) = \dot{m}c_p(T_3 - T_4)$$

Heat rejection $$\dot{Q}_X = \dot{m}(h_1 - h_4) = \dot{m}c_p(T_1 - T_4)$$

Energy conversion efficiency $$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{add}} = \frac{\dot{W}_T + \dot{W}_C}{\dot{Q}_R} = 1 - \frac{h_4 - h_1}{h_3 - h_2} = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Net work ratio $$NWR = \frac{\dot{W}_{net}}{\dot{W}_{+ve}} = \frac{\dot{W}_T + \dot{W}_C}{\dot{W}_T} = 1 - \frac{h_2 - h_1}{h_3 - h_4} = 1 - \frac{T_2 - T_1}{T_3 - T_4}$$

Gas Turbine Power Plants

Influence of Operating Parameters on the Plant Performance

$$\eta = \frac{(1-\tau)/\eta_C + \eta_T(1-1/\tau)T_3/T_1}{T_3/T_1 - 1 - (\tau-1)\eta_C}$$

$$\frac{\dot{W}_{net}}{\dot{m}c_pT_1} = \frac{1}{\eta_C}(1-\tau) + \eta_T\frac{T_3}{T_1}\left(1-\frac{1}{\tau}\right)$$

- where the isentropic temperature ratio is

$$\tau = (p_2/p_1)^{\frac{\gamma-1}{\gamma}}$$

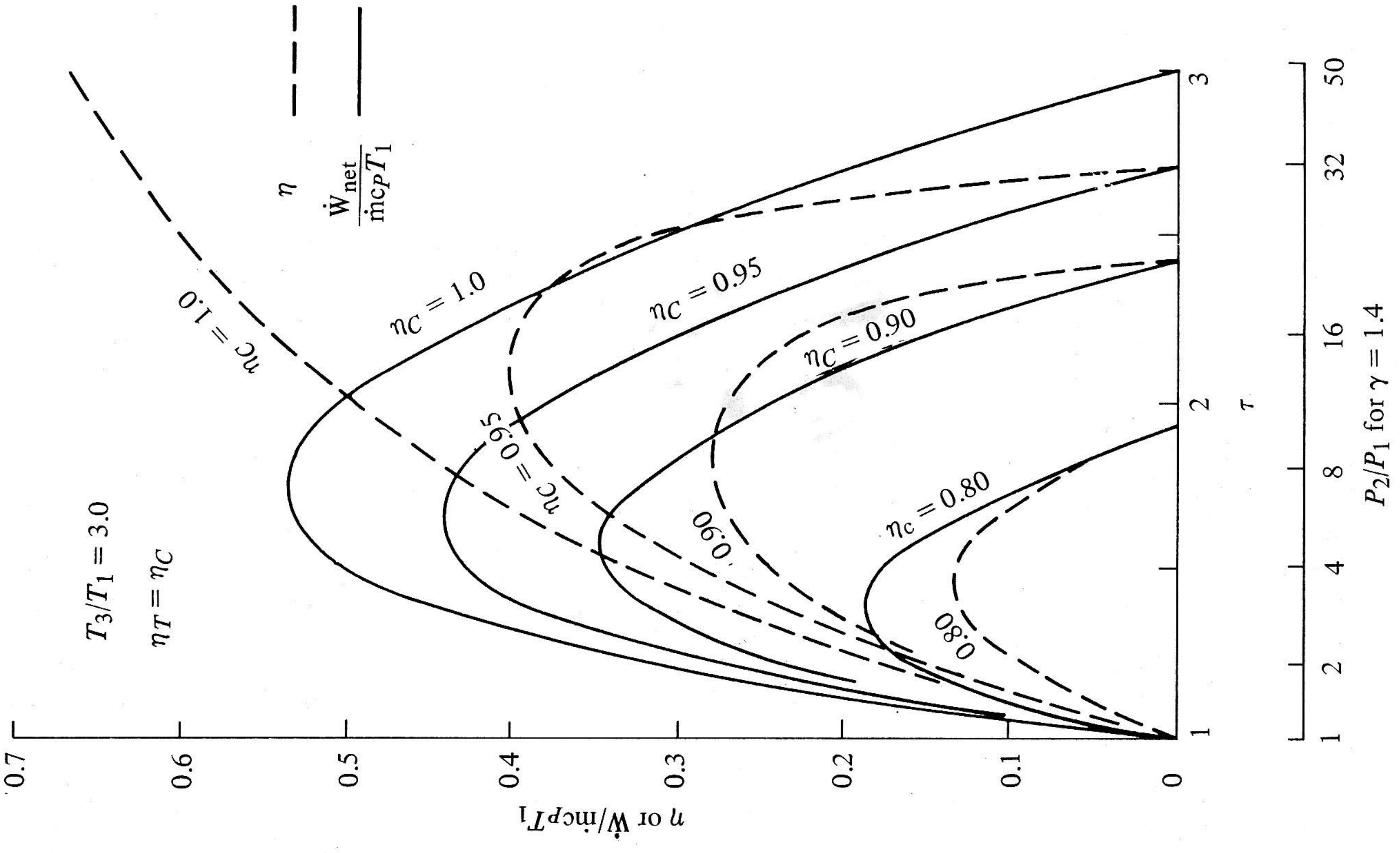
$$note: \eta_C = 0.85 \sim 0.90, \ \ \eta_T = 0.90 \sim 0.95$$

- The max. performance of the Brayton Cycle is obtained when

$$\eta_{\max} : \left(\frac{\partial\eta}{\partial\tau}\right)_{T_3/T_1, \eta_C, \eta_T} = 0 \quad , \quad (NWR)_{\max} : \left(\frac{\partial(NWR)}{\partial\tau}\right)_{T_3/T_1, \eta_C, \eta_T} = 0$$

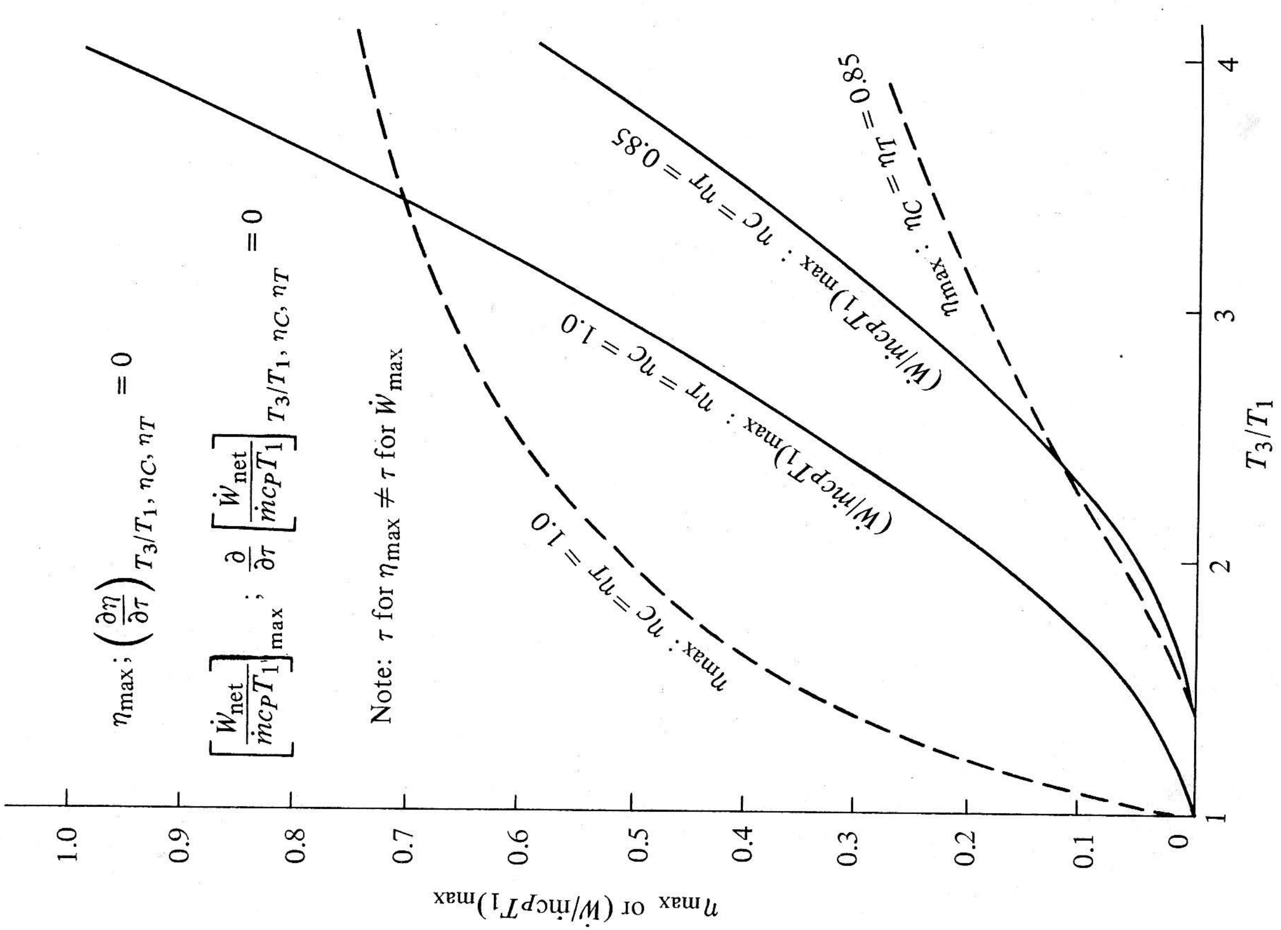
Gas Turbine Power Plants

Influence of Pressure Ratio and Component Efficiency on the Brayton Cycle



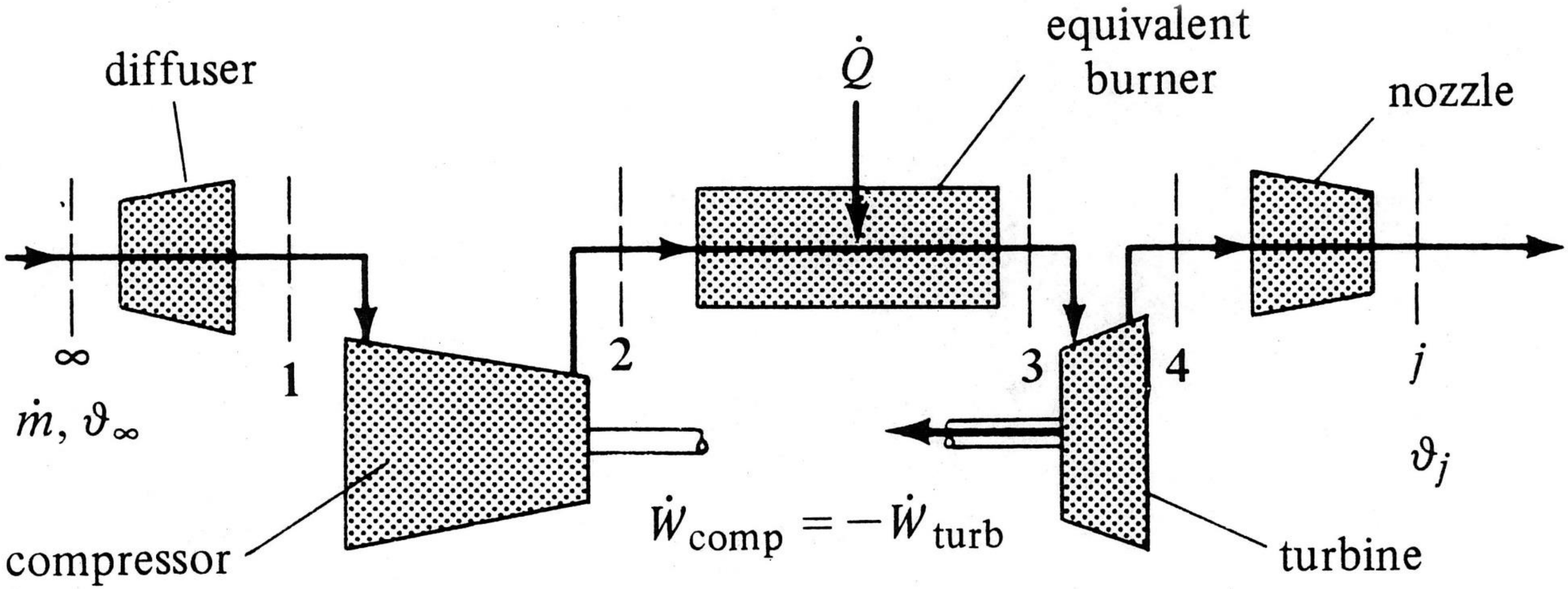
Gas Turbine Power Plants

Influence of Overall Temperature Ratio on Max. Performance of the Brayton Cycle



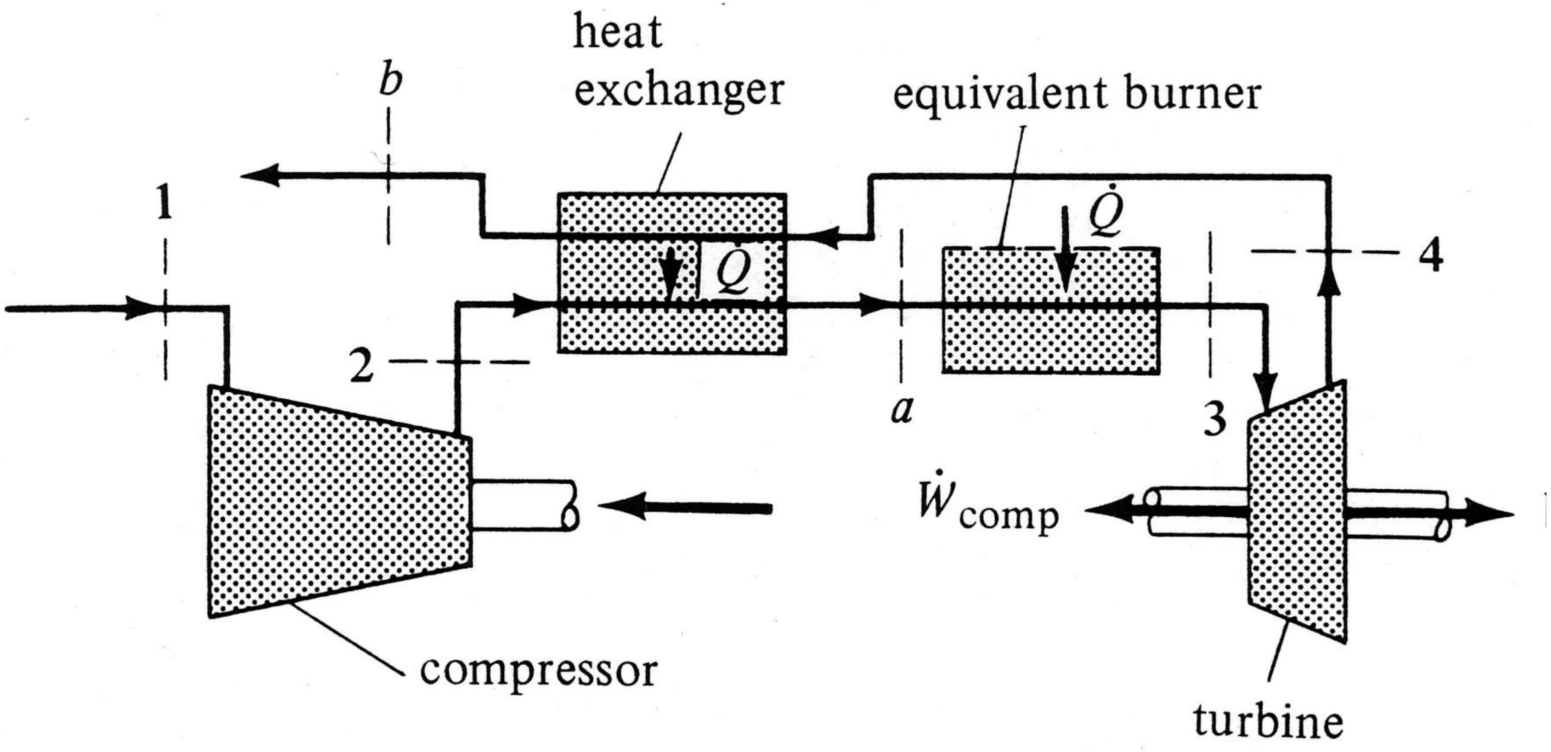
Gas Turbine Power Plants

Model for a Turbojet Plant



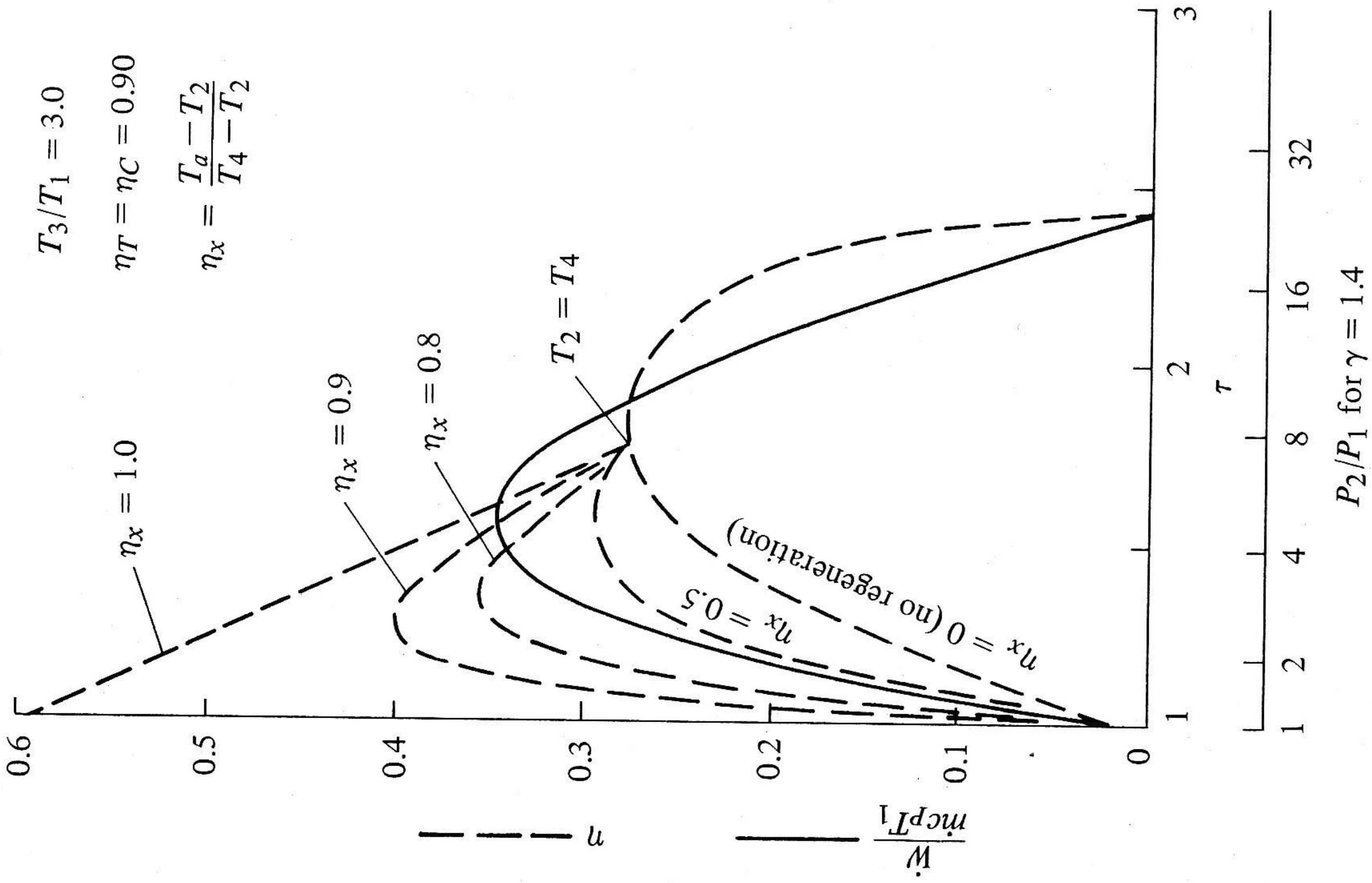
Gas Turbine Power Plants

Model for a Regenerative Gas Turbine Plant



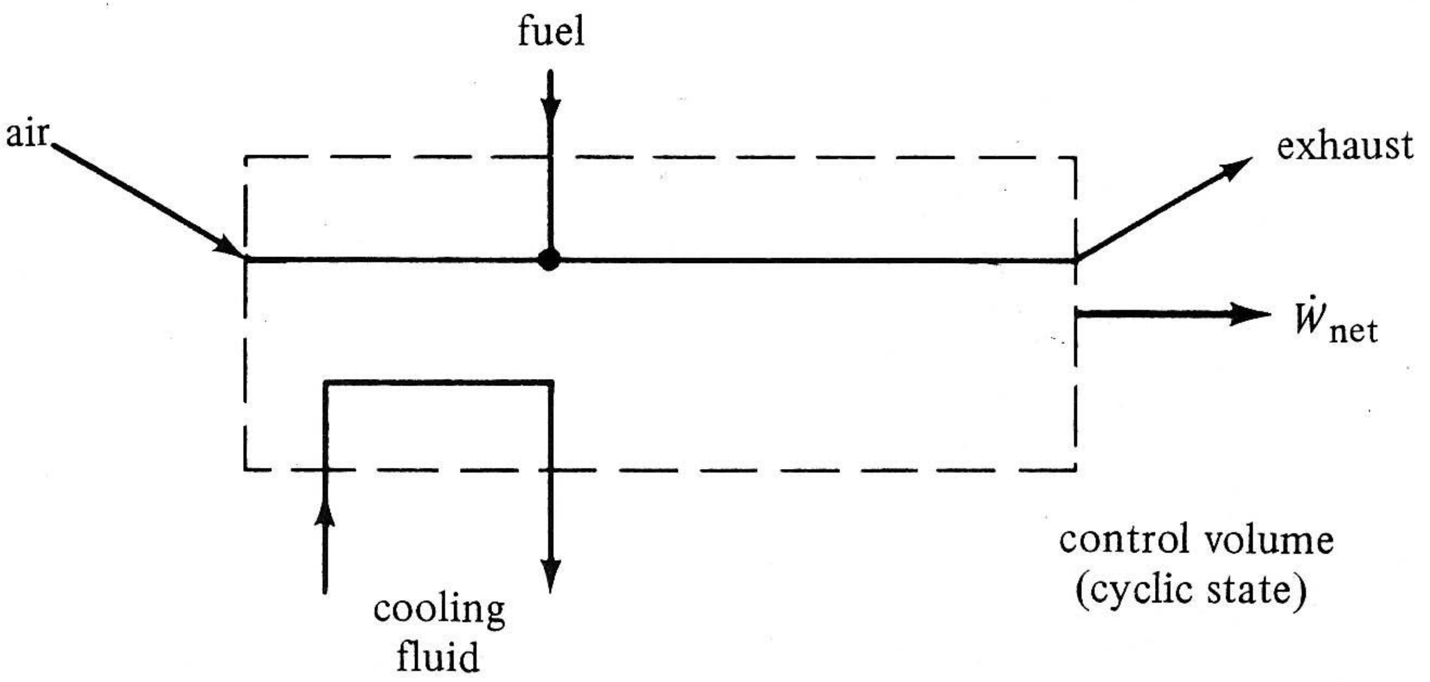
Gas Turbine Power Plants

Performance of a Regenerative Gas Turbine

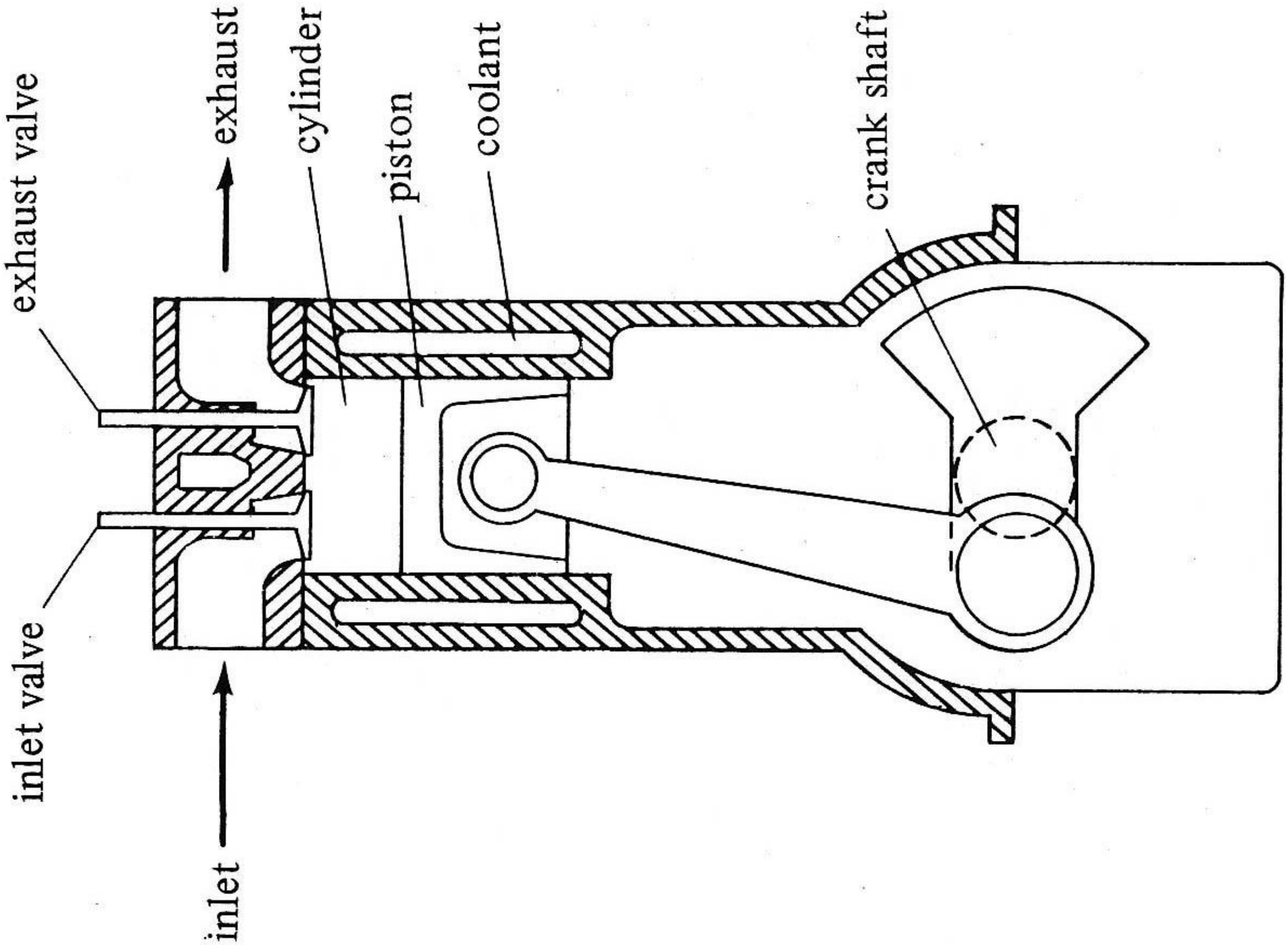


Reciprocating Internal Combustion Engines

- Control volume for reciprocating internal combustion engine plant



- Reciprocating Internal Combustion Engines

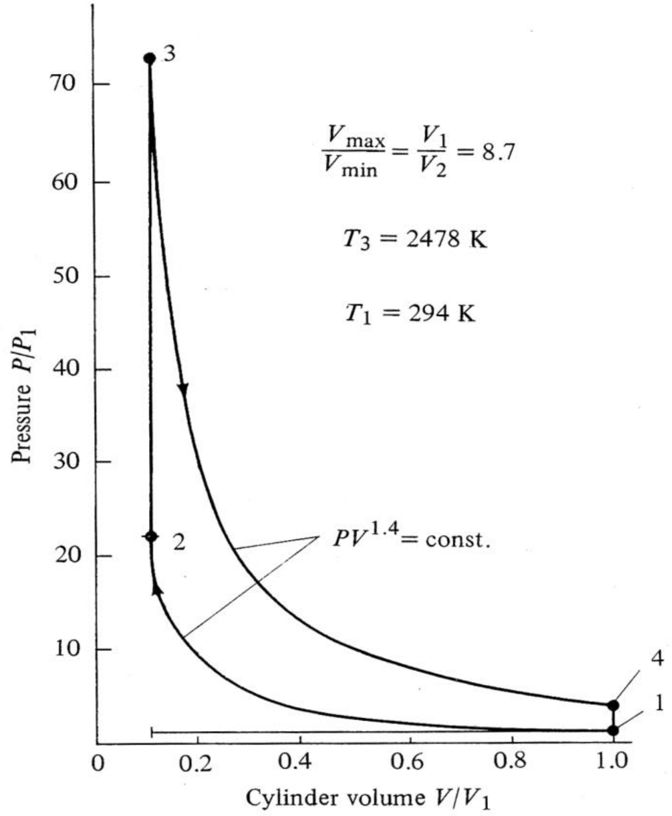


Reciprocating Internal Combustion Engines

Otto Engine

spark-ignition engine (p vs v) diagram

- two constant s processes+ two constant v process



1-2:compressed to min.v
2: spark ignites
2-3: equivalent con. v heat addition
3-4: hot gas expanded to max. v
4: exhaust valve opens
4-1:hot gas expands to exhaust pressure
1: fresh charge of premixed air+fuel

➢ Reciprocating Internal Combustion Engines

Otto Engine

$$W_{net} = W_{1-2} + W_{3-4} = mc_v(T_1 - T_2) + mc_v(T_3 - T_4)$$

$$Q_{2-3} = mc_v(T_3 - T_2)$$

$$\eta = W_{net} / Q_{2-3} = 1 - (T_4 - T_1)/(T_3 - T_2) = 1 - T_1 / T_2$$

For the rev. adia. process 1-2

$$T_1 / T_2 = (V_2 / V_1)^{\gamma-1} = 1/ r^{\gamma-1}$$

$$r = V_1 / V_2 : the\ compression\ ratio$$

$$\eta = 1 - 1/ r^{\gamma-1}$$

Reciprocating Internal Combustion Engines

- Otto Engine
  - The net work per unit mass of working fluid is important for showing the relative weight of the engine for a given power

$$\frac{W_{net}}{mRT_1} = \frac{W_{net}}{pV_1} = \frac{\eta Q_{2-3}}{mRT_1} = \frac{Q_{2-3}}{mRT_1}\left(1 - r^{\gamma-1}\right)$$

$$p_{\substack{mean\\ efficitive}} = \frac{W_{net/cycle}}{V_{displaced/cycle}} = \frac{W_{net}}{V_1 - V_2}$$

Used to determine the size & weight of the engine

  - power delivered by the engine

$$\dot{W} = NW_{net} = A_{piston}V_{piston}p_{m.e}$$

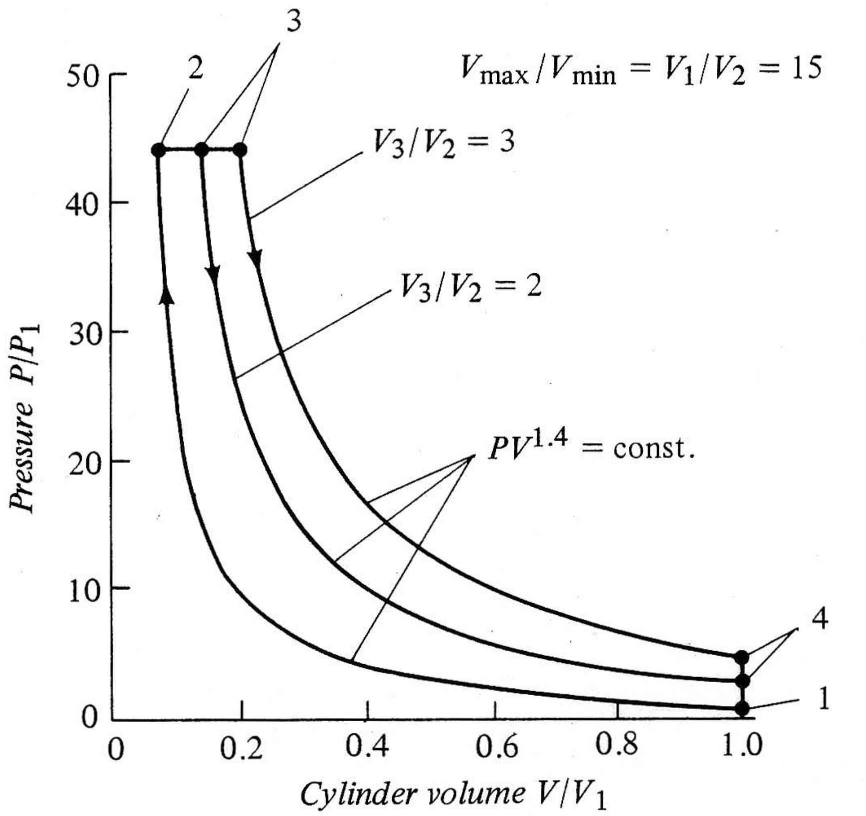
$$\dot{W} \propto p_{m.e},\ given\ A_{piston},\ V_{piston}$$

Reciprocating Internal Combustion Engines

- Diesel Engine
  - compression-ignition engine (p vs V diagram)
  - similar to Otto (spark-ignition) engine except the cylinder is charged with air
  - air temperature > ignition temp. of fuel when compressed to minimum V
  - fuel spray evaporates, mixes with air, ignites & burns
  - heat transfer is required to evaporate and heat fuel to ignition, the combustion process is slow
    - often modeled simply as con. p process

Reciprocating Internal Combustion Engines

Diesel Engine



$$V_{\max}/V_{\min} = V_1/V_2 = 15$$

twice as much as in Otto engine

$$pV^{1.4} = const. \quad (rev.\ adia.)$$

1-2: adiabatic compression (const. s)
2-3: const. P heat addition
3-4: adiabatic expansion (const. s)
4-1: const. V heat rejection

Reciprocating Internal Combustion Engines

Diesel Engine

$$W_{1-2} = mc_v\left(T_1 - T_2\right) = mc_v T_1\left(1 - r^{\gamma-1}\right)$$

$$W_{2-3} = mR\left(T_3 - T_2\right) = mc_v T_1\left(\gamma - 1\right) r^{\gamma-1}\left(r_c - 1\right)$$

$$W_{3-4} = mc_v\left(T_3 - T_4\right) = mc_v T_1 r_c^{\gamma}\left(r^{\gamma-1} r_c^{1-\gamma} - 1\right)$$

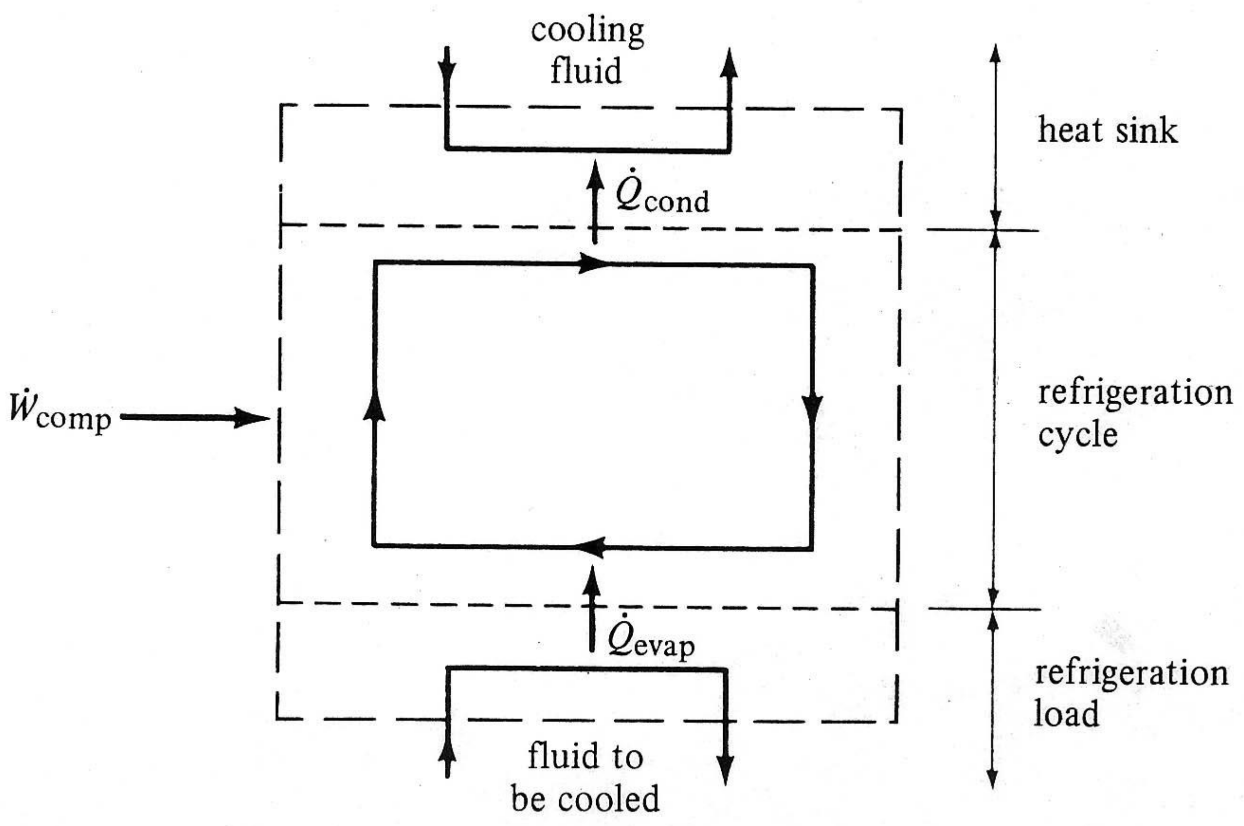
$$Q_{2-3} = mc_p\left(T_3 - T_2\right) = mc_v T_1 \gamma r^{\gamma-1}\left(r_c - 1\right)$$

$$Q_{4-1} = mc_v\left(T_1 - T_4\right) = mc_v T_1\left(1 - r_c^{\gamma}\right) \qquad \eta = 1 + \frac{Q_{4-1}}{Q_{2-3}} = 1 - \frac{1}{r^{\gamma-1}} \frac{r_c^{\gamma} - 1}{\gamma\left(r_c - 1\right)}$$

$$\frac{W_{net}}{p_1 V_1} = \frac{\gamma}{\gamma - 1} r^{\gamma-1}\left(r_c - 1\right) - \frac{1}{\gamma - 1}\left(r_c^{\gamma} - 1\right) \qquad \frac{p_{m.e}}{p_1} = \frac{r}{r-1} \frac{\gamma}{\gamma - 1}\left[r^{\gamma-1}\left(r_c - 1\right) - \frac{r_c^{\gamma} - 1}{r}\right]$$

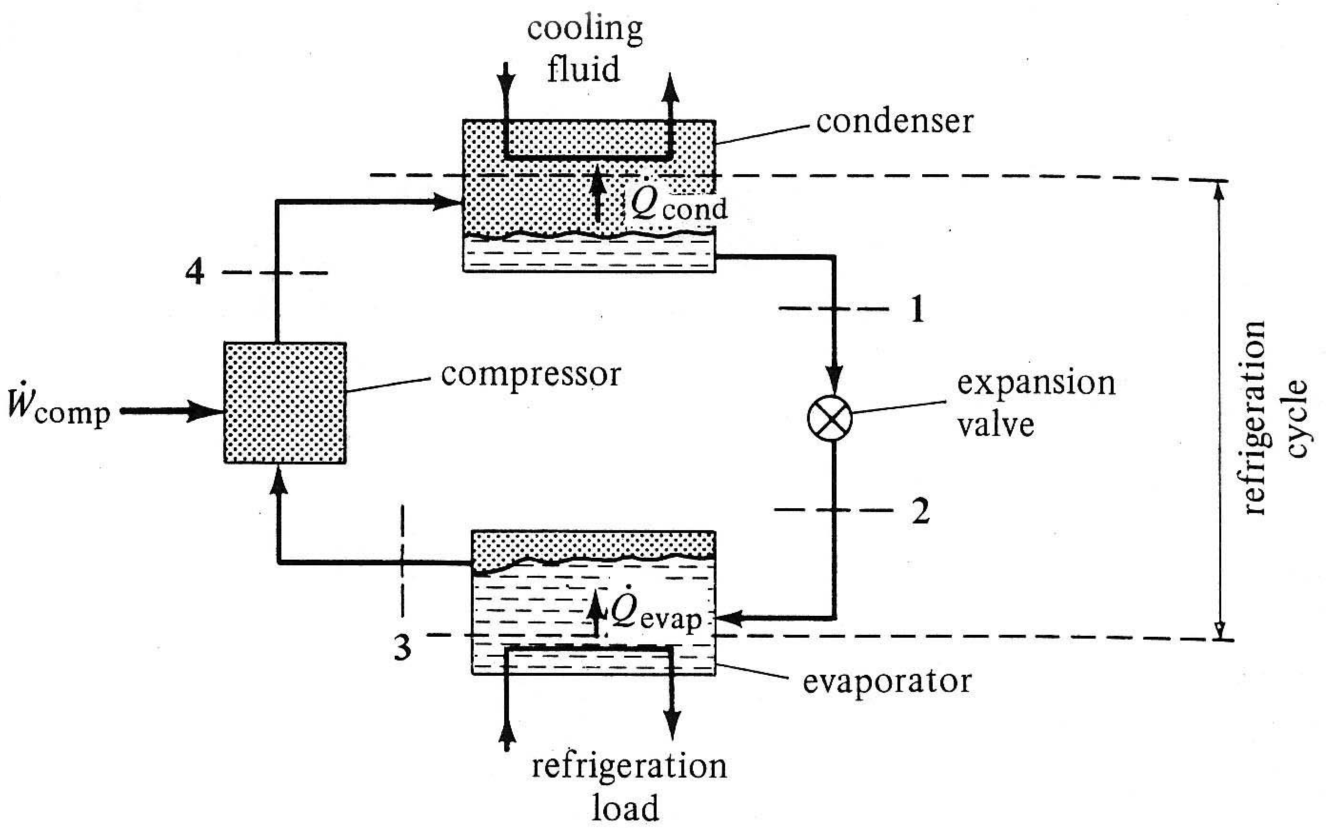
Refrigeration Plants

- Overall control volume for a refrigeration plant



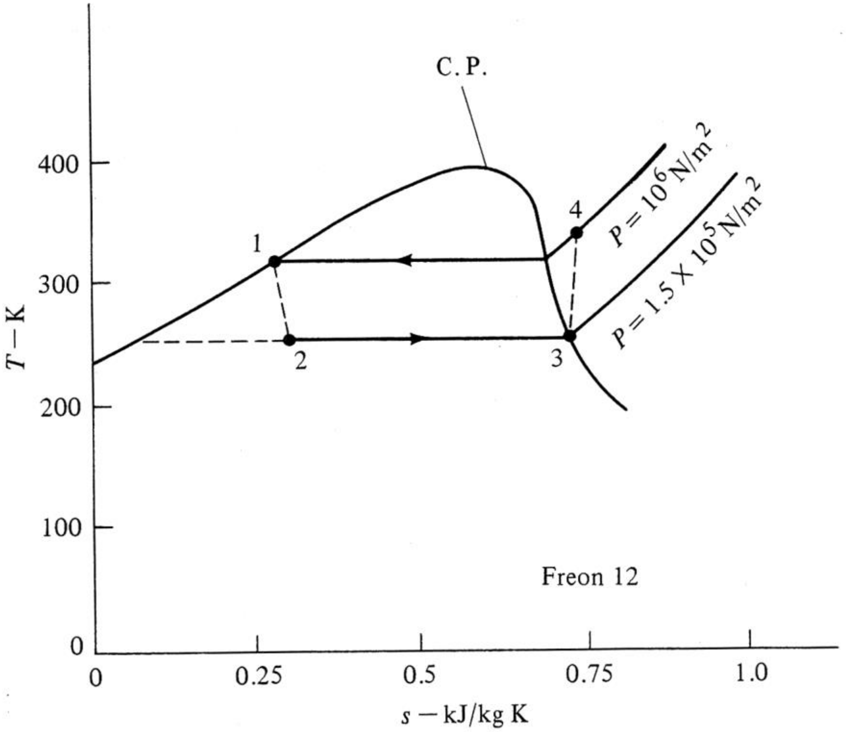
Refrigeration Plants

Vapor compression refrigeration plant



Refrigeration Plants

- receive positive HT from a low temp. fluid
- transfer heat to environment (air or water) at higher temp.
- vapor compression refrigeration plant is most common
- consist of 1-compressor, 2-condenser, 3- throttle valve, 4-evaporator



1-2: expansion and cools the remaining liquid to the sat. temp.

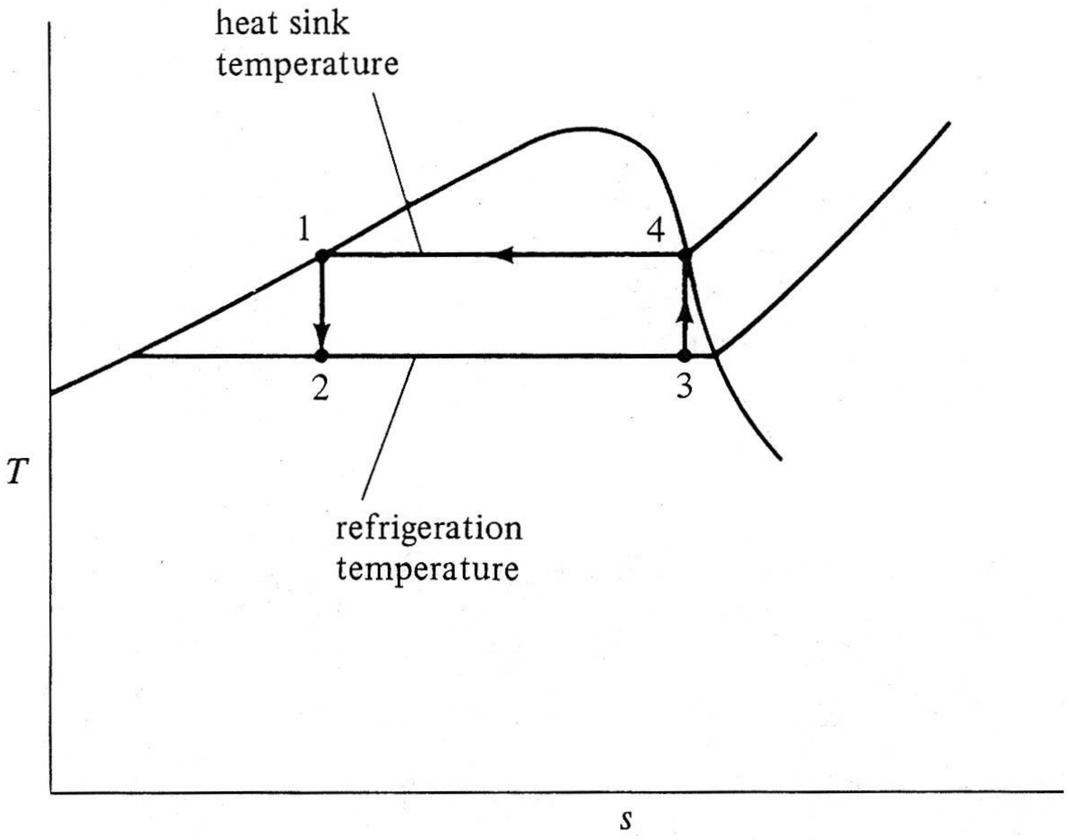
2-3: +ve HT from refrigeration load

3-4: compression

4-1: -ve HT to energy sink

Refrigeration Plants

Reversible refrigeration cycle



Refrigeration Plants

Summary

$$\dot{Q}_{2-3} = \dot{Q}_{evap} = \dot{m}\left(h_3 - h_2\right) = \dot{m}T\left(s_3 - s_2\right)$$

$$\dot{W}_{3-4} = \dot{W}_{comp} = \dot{m}\left(h_3 - h_4\right)$$

$$\dot{Q}_{1-4} = \dot{Q}_{cond} = \dot{m}\left(h_1 - h_4\right)$$

$$COP = \frac{\dot{Q}_{evap}}{-\dot{W}_{comp}} = \frac{h_3 - h_2}{-\left(h_3 - h_4\right)}$$

- Most commonly used refrigerants: ammonia, Freon-12 & 22