Rotating Equipment: Pumps, Compressors, & Turbines/Expanders

Chapters 2 & 9

Topics

Fundamentals

- Starting relationships
  - Thermodynamic relationships
  - Bernoulli's equation
- Simplifications
  - Pumps – constant density compression
  - Compressors – reversible ideal gas compression
- Use of PH & TS diagrams
- Multistaging

Efficiencies

- Adiabatic/isentropic vs. mechanical
- Polytropic

Equipment

- Pumps
  - Centrifugal pumps
  - Reciprocating pumps
  - Gear pumps
- Compressors
  - Centrifugal compressors
  - Reciprocating compressors
  - Screw compressors
  - Axial compressors
- Turbines & expanders
  - Expanders for NGL recovery
  - Gas turbines for power production
    - What is "heat rate"?

Fundamentals

Updated: February 6, 2018

Review of Thermodynamic Principals

1st Law of Thermodynamics – Energy is conserved

- (Change in system's energy) = (Rate of heat added) – (Rate of work performed)

$$\Delta\hat{E} = Q - W$$

- Major energy contributions
  - Kinetic energy – related to velocity of system
  - Potential energy – related to positon in a "field" (e.g., gravity)
  - Internal energy – related to system's temperature
    - Internal energy, U, convenient for systems at constant volume & batch systems

$$\hat{E} = \hat{U} + \frac{u^2}{2g_c} + \frac{g}{g_c}h$$

    - Enthalpy, H = U+PV, convenient for systems at constant pressure & flowing systems

$$\hat{E} = \hat{H} + \frac{u^2}{2g_c} + \frac{g}{g_c}h$$

Review of Thermodynamic Principals

2nd Law of Thermodynamics

- In a cyclic process entropy will either stay the same (reversible process) or will increase

Relationship between work & heat

- All work can be converted to heat, but…
- Not all heat can be converted to work

Common Paths for Heat and Work

Isothermal	constant temperature	$\Delta T = 0$
Isobaric	constant pressure	$\Delta P = 0$
Isochoric	constant volume	$\Delta V = 0$
Isenthalpic	constant enthalpy	$\Delta H = 0$
Adiabatic	no heat transferred	$Q = 0$
Isentropic (ideal reversible)	no increase in entropy	$\Delta S = 0$

1st Law for steady state flow

Equation 1.19a ($\Delta H \approx \Delta U$ for flowing systems)

$$\Delta \hat{H} + \frac{\Delta u^2}{2g_c} + \frac{g}{g_c}\Delta z = Q - W$$

For adiabatic, steady-state, ideal (reversible) flow (using WS as positive value)

$$\hat{W}_s = \Delta \hat{H} + \frac{\Delta u^2}{2g_c} + \frac{g}{g_c}\Delta z$$

$$= \int_{P_1}^{P_2} \hat{V}\, dP + \frac{\Delta u^2}{2g_c} + \frac{g}{g_c}\Delta z$$

$$\hat{W}_s \approx \int_{P_1}^{P_2} \hat{V}\, dP = \int_{P_1}^{P_2} \frac{dP}{\rho}$$

- The work required is inversely proportional to the mass density

Thermodynamics of Compression

Work depends on path – commonly assume adiabatic or polytropic compression

Calculations done with:

- PH diagram for ΔH

$$W_s = \int_{P_1}^{P_2} V dP = \Delta H$$

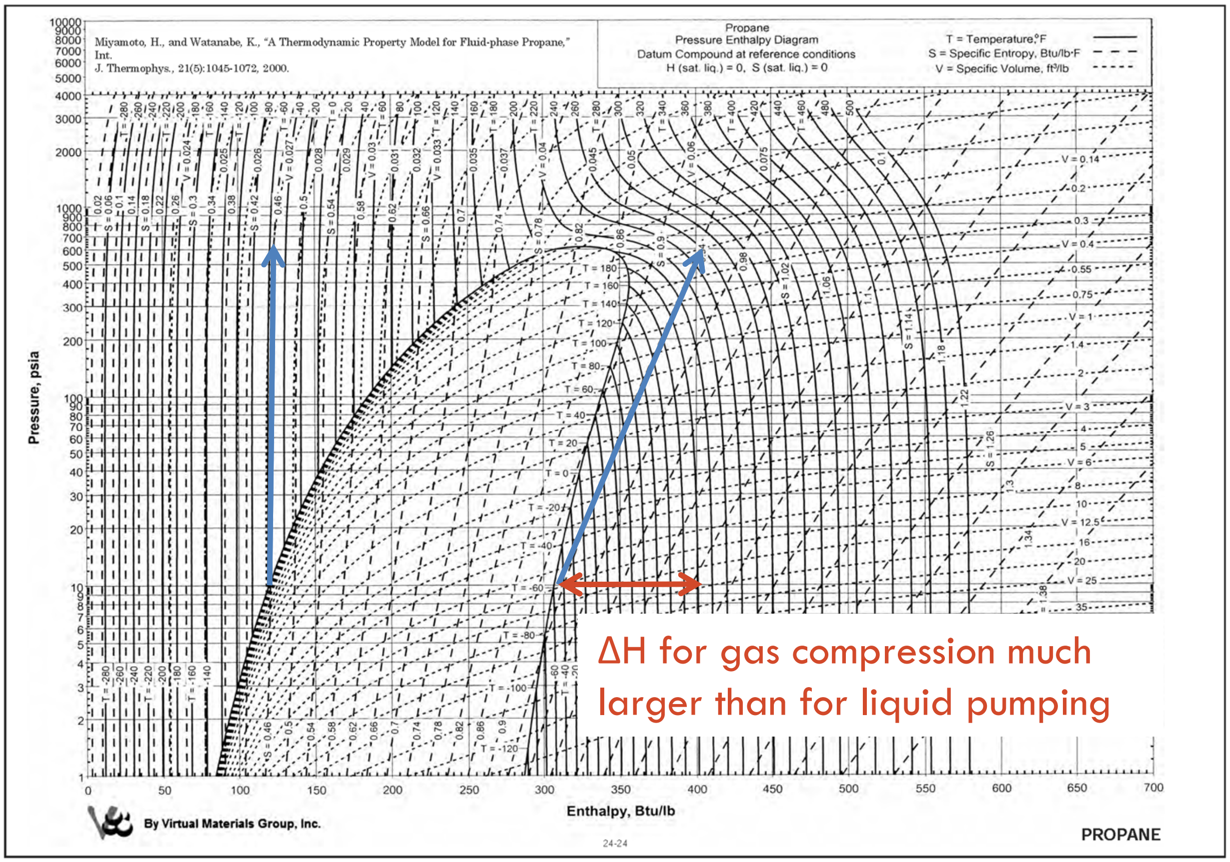
- Evaluate integral using equation of state
  - Simplest gas EOS is the ideal gas law
  - Simplest liquid EOS is to assume incompressible (i.e., constant density with respect to pressure)

$$W_s = \int_{P_1}^{P_2} \frac{dP}{\rho} = \frac{1}{\rho} \int_{P_1}^{P_2} dP = \frac{P_2 - P_1}{\rho}$$

Liquid vs. Vapor Compression

Can compress liquids with little temperature change

FIG. 24-20 Propane P-H Diagram



GPSA Data Book, 13th ed.

Mechanical Energy Balance

Differential form of Bernoulli's equation for fluid flow (energy per unit mass)

$$\frac{d\left(u^2\right)}{2} + g\,dz + \frac{dP}{\rho} + d\left(\hat{w}_s\right) + g\,d\left(\hat{h}_f\right) = 0$$

- Frictional loss term is positive
- Work term for energy out of fluid – negative for pump or compressor

If density is constant then the integral is straight forward – pumps

$$\frac{\Delta\left(u^2\right)}{2} + g\,\Delta z + \frac{\Delta P}{\rho} + \hat{w}_s + g\,\hat{h}_f = 0$$

If density is not constant then you need a pathway for the pressure-density relationship – compressors

$$\frac{\Delta\left(u^2\right)}{2} + g\,\Delta z + \int_{P_1}^{P_2}\left(\frac{dP}{\rho}\right) + \hat{w}_s + g\,\hat{h}_f = 0$$

Pump equations

Pumping requirement expressed in terms of power, i.e., energy per unit time

Hydraulic horsepower – power delivered to the fluid

- Over entire system

$$W_{\text{hhp}} = \dot{m}\left(-\hat{w}_s\right) = \left(\rho\dot{V}\right)\left[\frac{\Delta\left(u^2\right)}{2} + g\,\Delta z + \frac{\Delta P}{\rho} + g\,\hat{h}_f\right]$$

$$= \dot{V}\left(\Delta P\right) + \left(\rho\dot{V}\right)\left(g\,\Delta z\right) + \left(\rho\dot{V}\right)\left(g\,\hat{h}_f\right) + \frac{1}{2}\left(\rho\dot{V}\right)\left(\Delta u^2\right)$$

- Just across the pump, in terms of pressure differential or head:

$$W_{\text{hhp}} = \dot{V}\left(\Delta P\right) \quad \text{or} \quad W_{\text{hhp}} = \dot{V}\rho g H$$

Brake horsepower – power delivered to the pump itself

$$W_{\text{bhp}} = \frac{W_{\text{hhp}}}{\eta_{\text{pump}}}$$

Pump equations for specific U.S. customary units

U.S. customary units usually used are gpm, psi, and hp

$$\dot{W}_{hhp}[\text{hp}] \Rightarrow \left[\frac{\text{gal}}{\text{min}}\right]\left[\frac{\text{lb}_f}{\text{in}^2}\right]\frac{\left[231\dfrac{\text{in}^3}{\text{gal}}\right]}{\left[60\dfrac{\text{sec}}{\text{min}}\right]\left[12\dfrac{\text{in}}{\text{ft}}\right]\left[550\dfrac{\text{ft}\cdot\text{lb}_f/\text{sec}}{\text{hp}}\right]}$$

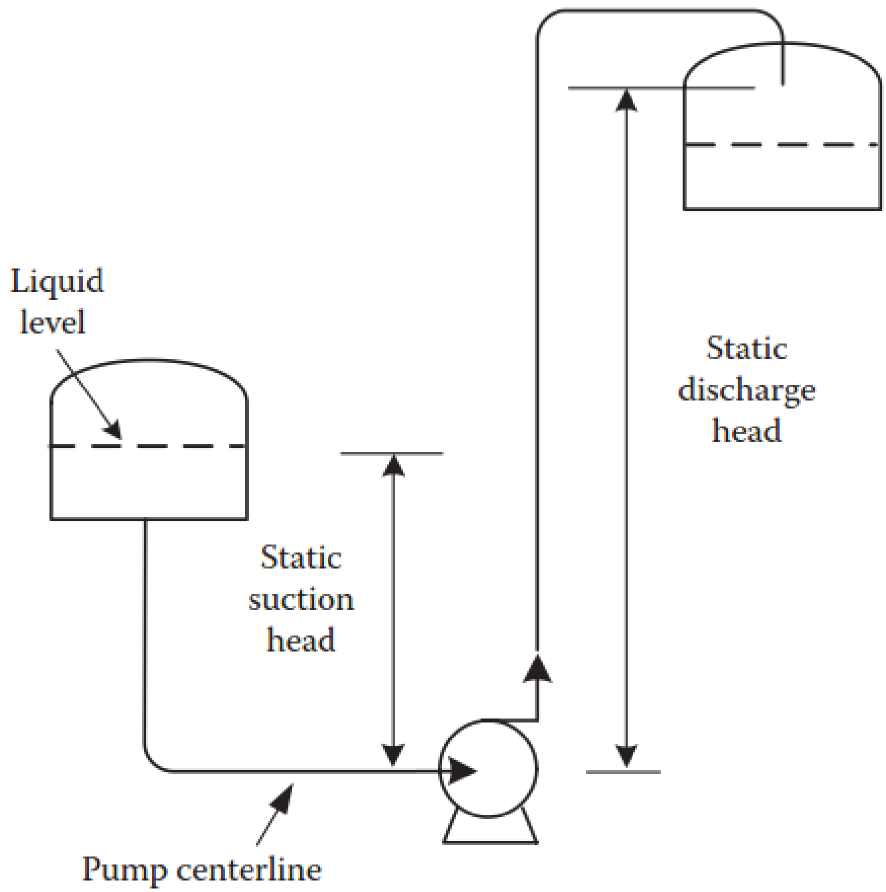
$$= \left(\frac{1}{1714}\right)\left[\frac{\text{gal}}{\text{min}}\right]\left[\frac{\text{lb}_f}{\text{in}^2}\right]$$

Also use the head equation usually using gpm, ft, specific gravity, and hp

$$\dot{W}_{hhp}[\text{hp}] \Rightarrow \left[\frac{\text{gal}}{\text{min}}\right][\text{ft}][\gamma_o]\frac{\left[8.33719\dfrac{\text{lb}_m}{\text{gal}}\right]}{\left[60\dfrac{\text{sec}}{\text{min}}\right]\left[550\dfrac{\text{ft}\cdot\text{lb}_f/\text{sec}}{\text{hp}}\right]}\left[\frac{32.174\dfrac{\text{ft}}{\text{sec}^2}}{32.174\dfrac{\text{lb}_m\cdot\text{ft}}{\text{lb}_f\cdot\text{sec}^2}}\right]$$

$$= \left(\frac{1}{3958}\right)\left[\frac{\text{gal}}{\text{min}}\right][\text{ft}][\gamma_o]$$

Static Head Terms



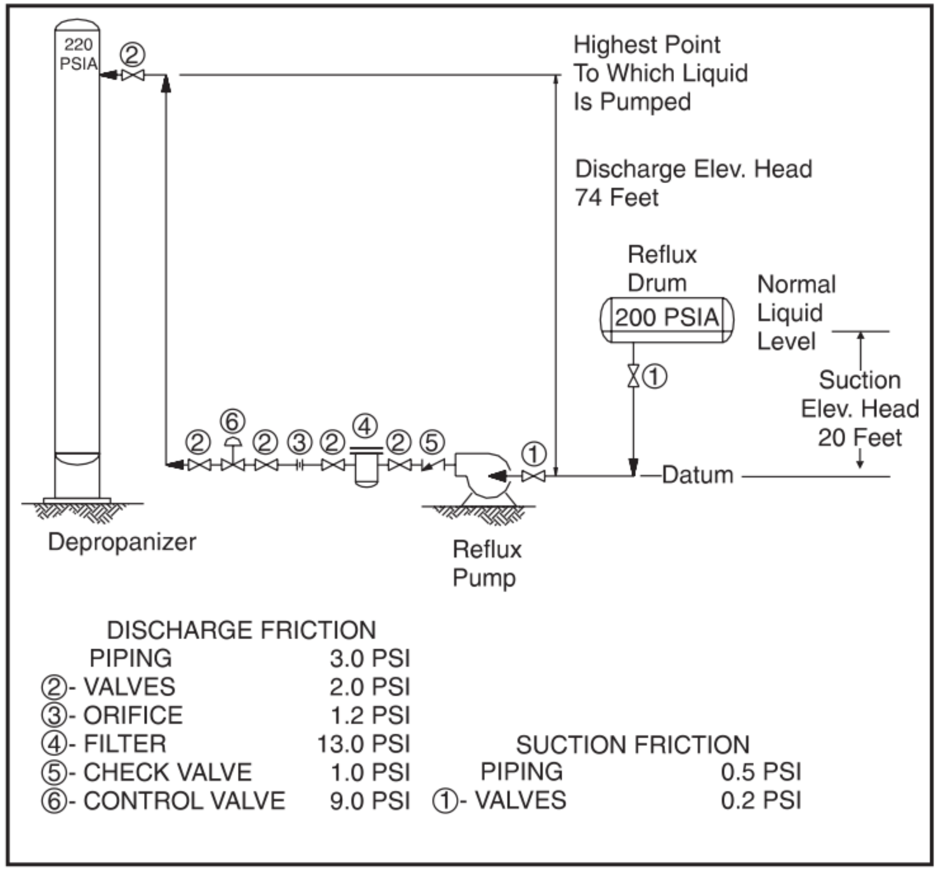
Fundamentals of Natural Gas Processing, 2nd ed.
Kidnay, Parrish, & McCartney

Pump Example

Liquid propane (at its bubble point) is to be pumped from a reflux drum to a depropanizer.

- Pressures, elevations, & piping system losses as shown are shown in the diagram.
- Max flow rate 360 gpm.
- Propane specific gravity 0.485 @ pumping temperature (100°F)
- Pump nozzles elevations are zero & velocity head at nozzles negligible

What is the pressure differential across the pump?
What is the differential head?
What is the hydraulic power?



GPSA Data Book, 13th ed.

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Pump Example

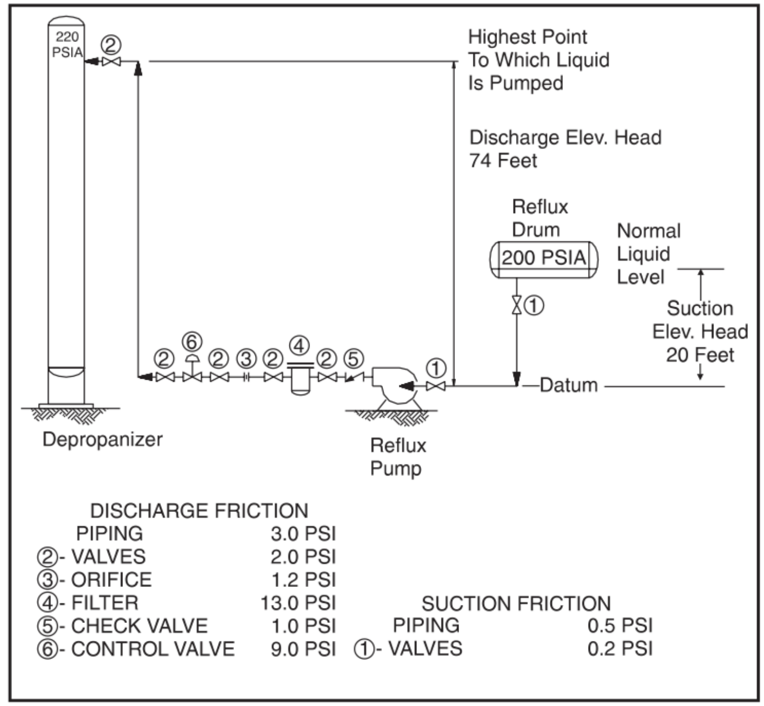
Pressure drop from Reflux Drum to Pump inlet:

$$\Delta P = -\frac{g}{g_c}\rho\Delta z + \left(\Delta P\right)_{\text{piping}}$$

$$= -\frac{\left[(0.485)(62.3665)\right](-20)}{144} + \left[-0.5 - 0.2\right]$$

$$= 3.5 \text{ psi}$$

$$\therefore P_{\text{inlet}} = 203.5 \text{ psia}$$



Pressure drop from Pump outlet to Depropanizer:

$$\Delta P = -\frac{g}{g_c}\rho\Delta z + \left(\Delta P\right)_{\text{piping}}$$

$$= -\frac{\left[(0.485)(62.3665)\right](74)}{144} - \left[3.0 + 2.0 + 1.2 + 13.0 + 1.0 + 9.0\right]$$

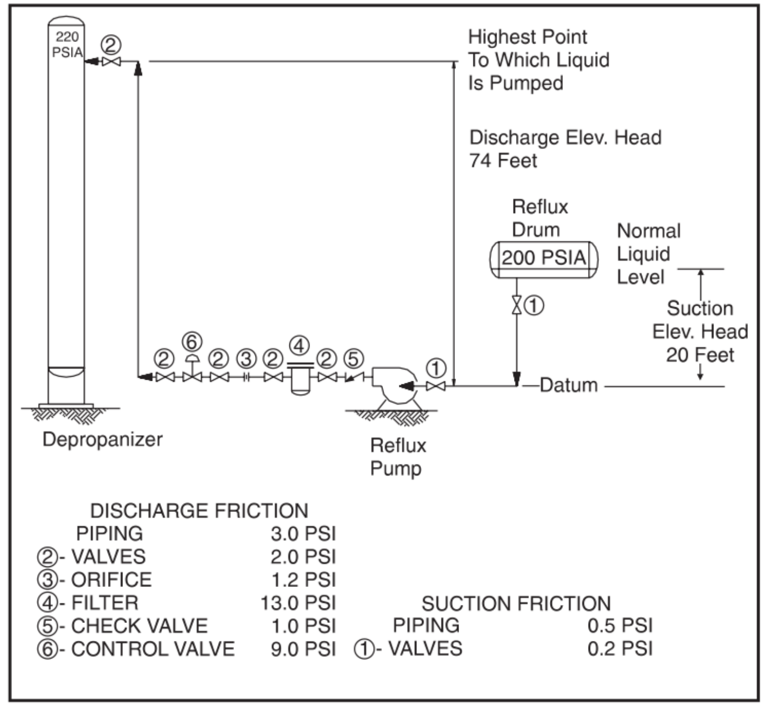
$$= -44.7 \text{ psi}$$

$$\therefore P_{\text{outlet}} = 264.7 \text{ psia}$$

Pump Example

Pump pressure differential:

$$(\Delta P)_{\text{pump}} = P_{\text{outlet}} - P_{\text{inlet}}$$
$$= 264.7 - 203.5 = 61.2 \text{ psi}$$



Pump differential head:

$$h_{\text{pump}} = \frac{g_c}{g} \frac{(\Delta P)_{\text{pump}}}{\rho}$$
$$= -\frac{(144)}{(0.485)(62.3665)}(61.2) = 291 \text{ ft}$$

Hydraulic power:

$$W_{\text{hhp}} = \frac{(360 \text{ gpm})(61.2 \text{ psi})}{1714} = 12.85 \text{ hp} \quad \text{OR} \quad W_{\text{hhp}} = \frac{(360 \text{ gpm})(0.485)(291 \text{ ft})}{3958} = 12.84 \text{ hp}$$

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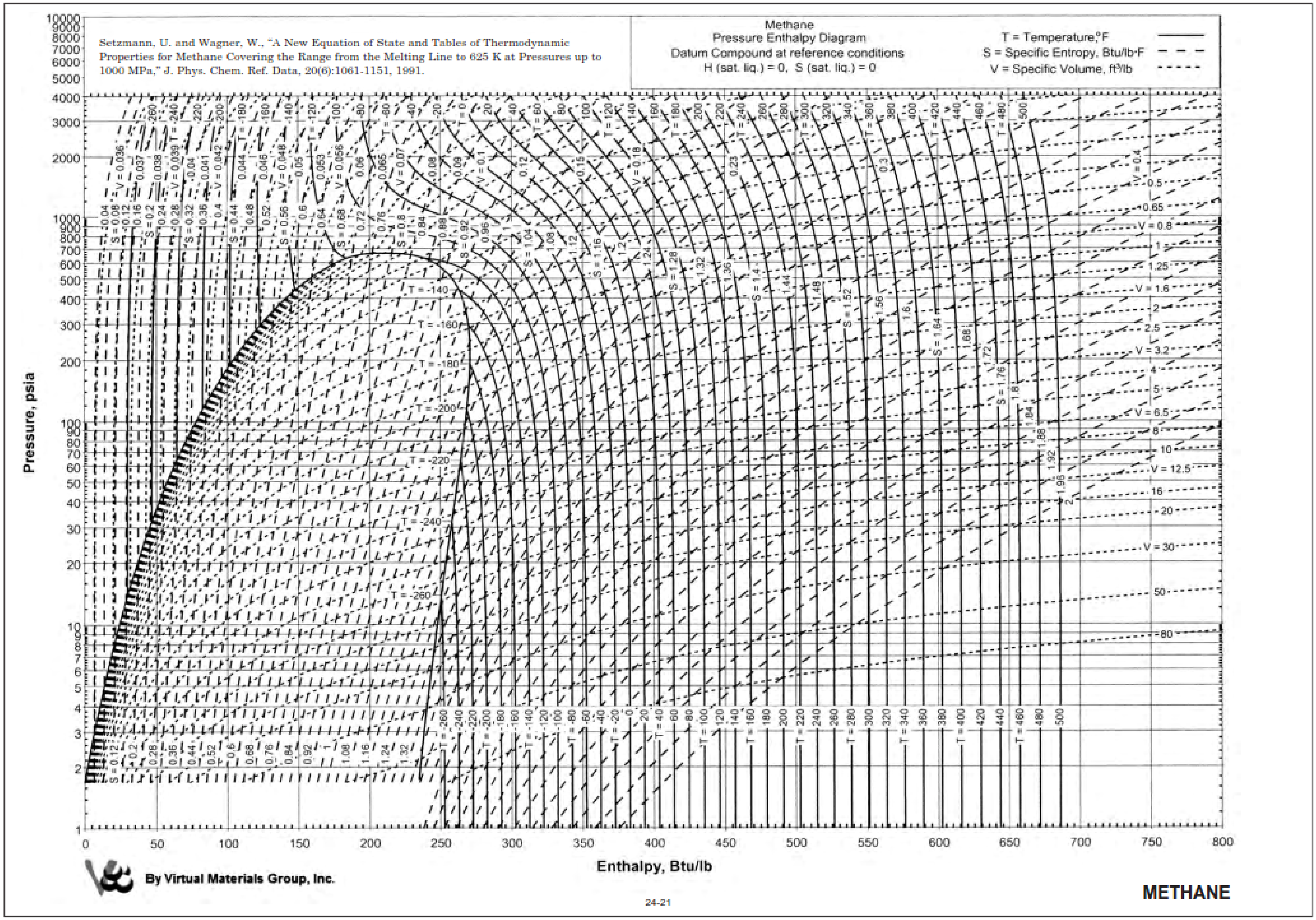
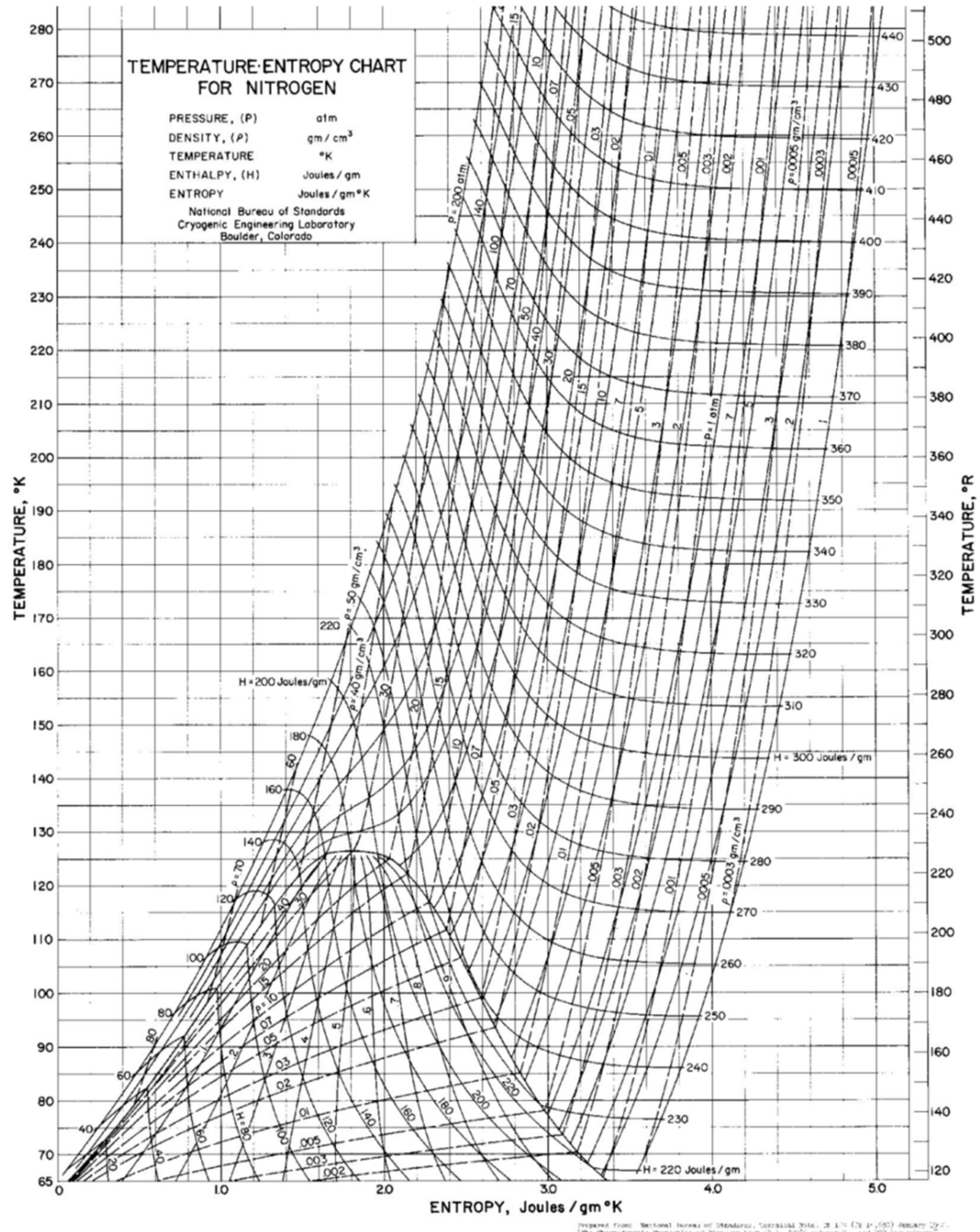
PH Diagrams



Ref: *GPSA Data Book*, 13th ed.

TS Diagram



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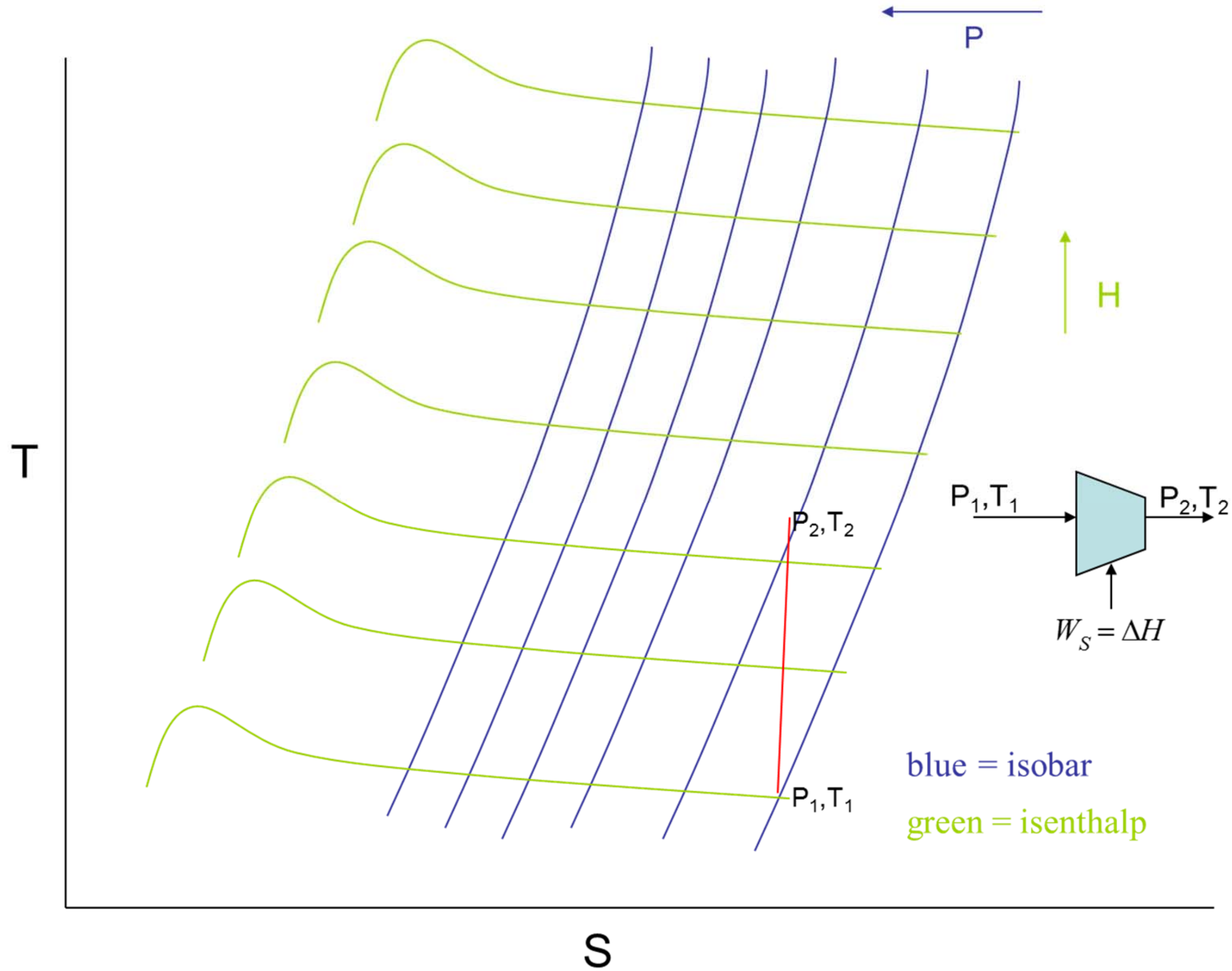
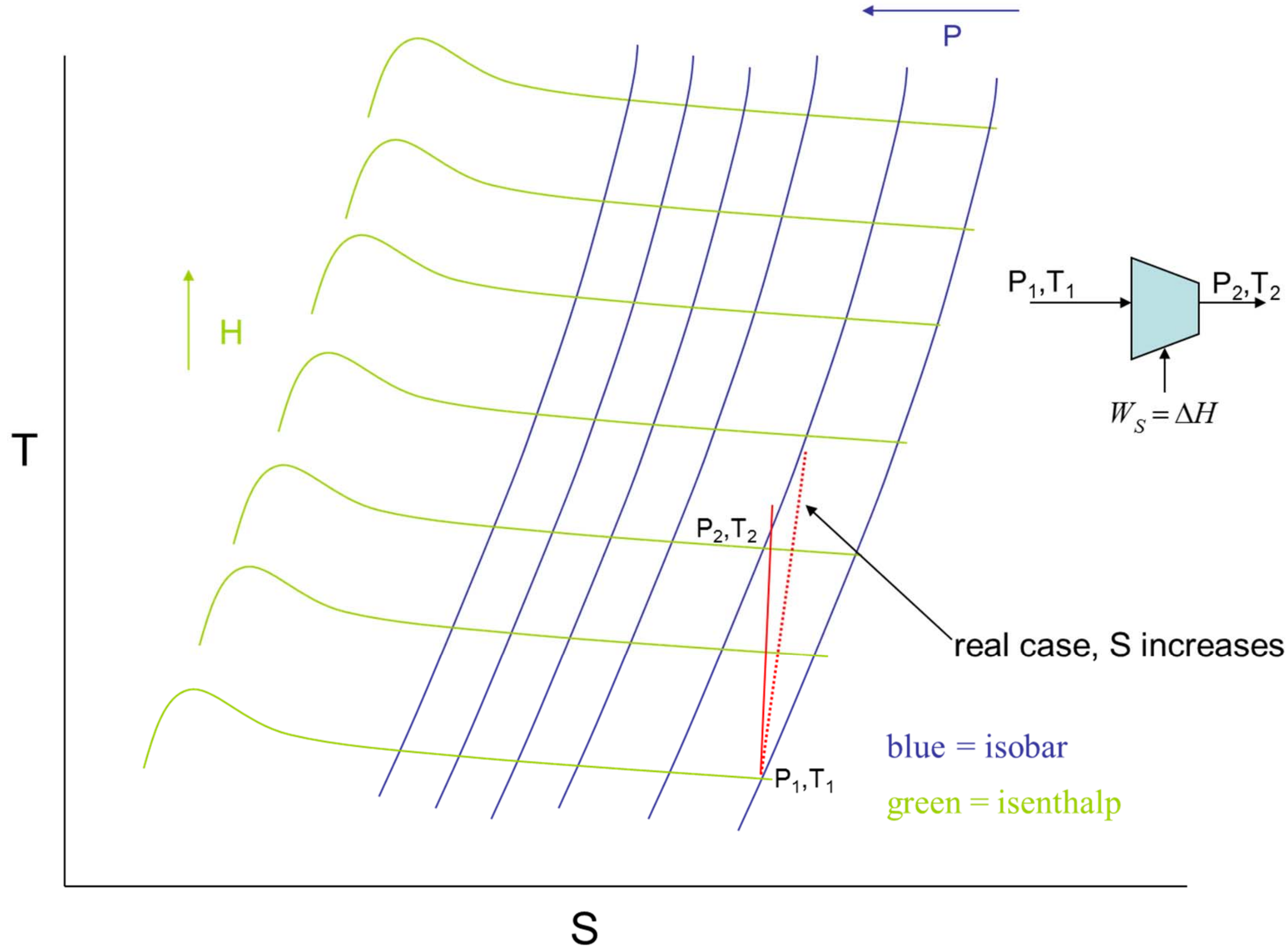
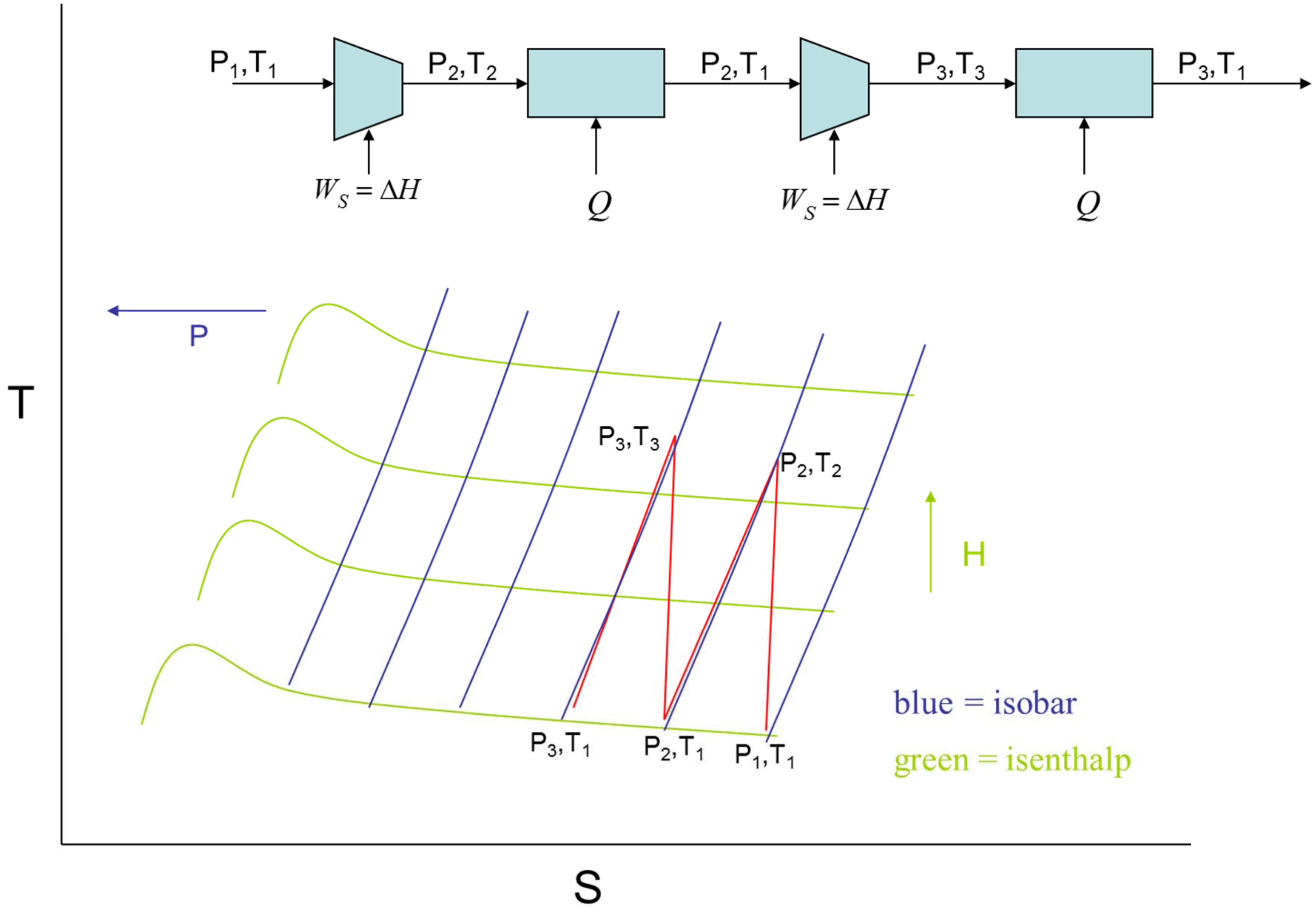


P
H
T
S
P_1,T_1
P_2,T_2
P_2,T_2
P_1,T_1
$W_S = \Delta H$
blue = isobar
green = isenthalp



P
H
T
S
P1,T1
P2,T2
$W_S = \Delta H$
P2,T2
P1,T1
real case, S increases
blue = isobar
green = isenthalp



P_1,T_1
P_2,T_2
P_2,T_1
P_3,T_3
P_3,T_1
$W_S = \Delta H$
Q
$W_S = \Delta H$
Q
T
P
P_3,T_3
P_2,T_2
H
P_3,T_1
P_2,T_1
P_1,T_1
blue = isobar
green = isenthalp
S

Thermodynamics of Compression

Assume ideal gas: PV = RT

Choices of path for calculating work:

- Isothermal (ΔT = 0)

$$W_s = \int_{P_1}^{P_2} V dP = RT \int_{P_1}^{P_2} \frac{dP}{P} = RT \ln\left(\frac{P_2}{P_1}\right)$$

  - Minimum work required but unrealistic
- Adiabatic & Isentropic (ΔS = 0)
  - Maximum ideal work but more realistic
- Polytropic – reversible but non-adiabatic
  - Reversible work & reversible heat proportionately added or removed along path
  - More closely follows actual pressure-temperature path during compression

Thermodynamics of Compression

Ideal gas isentropic (PV^{γ} = constant) where $\gamma = C_P/C_V$

Molar basis

$$\tilde{W}_s = (RT_1)\frac{\gamma}{\gamma-1}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

Mass basis

$$\hat{W}_s = \frac{RT_1}{M}\frac{\gamma}{\gamma-1}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

Polytropic (PV^{κ}= constant) where κ is empirical constant usually greater than γ

$$\hat{W}_p = \frac{RT_1}{M}\frac{\kappa}{\kappa-1}\left[\left(\frac{P_2}{P_1}\right)^{(\kappa-1)/\kappa} - 1\right]$$

Thermodynamics of Compression

Calculation of γ for gas mixture

$$\gamma = \frac{\sum x_i C_{p,i}}{\sum x_i C_{V,i}} = \frac{\sum x_i C_{p,i}}{\sum x_i C_{p,i} - R}$$

Use the ideal gas heat capacities, not the real gas heat capacities

Heat capacities are functions of temperature. Use the average value over the temperature range

Example Compression Calculation

Want to compress sales gas (assume pure methane) from initial conditions of 40°F & 100 psig to 400 psig.

Compute work of compression on mass basis…

- Using PH diagram
- Assuming ideal gas and adiabatic compression
- Using a process simulator

Example Calculation – PH Diagram

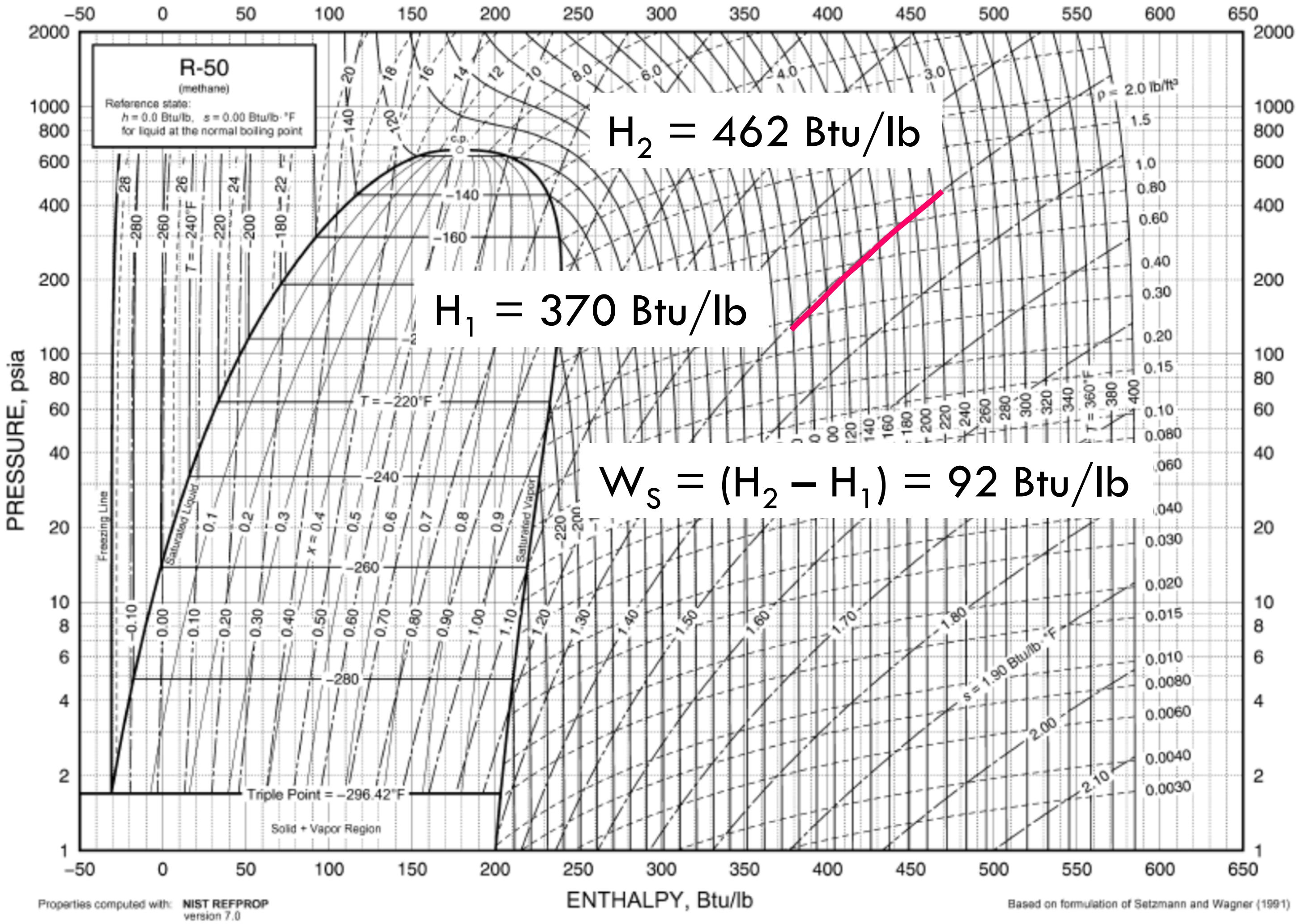


Fig. 19 Pressure-Enthalpy Diagram for Refrigerant 50 (Methane)

$H_2 = 462$ Btu/lb

$H_1 = 370$ Btu/lb

$W_S = (H_2 - H_1) = 92$ Btu/lb

Example Calculation – Ideal Gas Compression

For methane:

- $\gamma = 1.3$
- $M = 16$
- $T_1 = 40°F = 500°R$
- $P_1 = 100$ psig $= 114.7$ psia
- $P_2 = 400$ psig $= 414.7$ psia
- $R = 1.986$ Btu/lb.mol °R

$$W_s = \frac{\gamma R T_1}{M(\gamma - 1)}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right] = \frac{(1.3)(1.986)(500)}{(16)(1.3-1)}\left[\left(\frac{414.7}{114.7}\right)^{(1.3-1)/1.3} - 1\right] = 93 \text{ Btu/lb}$$

Example Calculation – Using a Simulator

	Work Btu/lb	Outlet °F
HYSYS Peng-Robinson	90.52	212.8
HYSYS Peng-Robinson & Lee-Kesler	90.96	211.6
HYSYS SRK	91.14	212.4
HYSYS BWRS	90.82	211.9
Aspen Plus PENG-ROB	90.62	214.0
Aspen Plus SRK	91.27	213.6
Aspen Plus BWRS	90.93	213.1
Aspen Plus BWR-LS	91.09	213.1

Discharge temperature

For ideal gas compression

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

For the example problem:

$$T_2 = (40 + 460) \left(\frac{414.7}{114.7} \right)^{(1.3-1)/1.3} = 673°\text{R} \quad \Rightarrow \quad 213°\text{F}$$

Thermodynamics of Compression

If customer wants 1,000 psig (when the inlet pressure 100 psig)…

- Then pressure ratio of (1015/115) = 8.8
- Discharge temperature for this ratio is ~360°F

For reciprocating compressors the GPSA *Engineering Data Book* recommends

- Maximum discharge temperature of 250 to 275°F for high pressure systems AND …
- Pressure ratios of 3:1 to 5:1

To obtain pressure ratios higher than 5:1 must use multistage compression ***with interstage cooling***

Multistaging

To minimize work need good interstage cooling and equal pressure ratios in stages.

The number of stages is calculated using

$$\mathcal{R}_p = \left(\frac{P_2}{P_1}\right)^{1/m} \quad \Rightarrow \quad m = \frac{\ln\left(P_2 / P_1\right)}{\ln\left(\mathcal{R}_p\right)}$$

To go from 100 to 1000 psig with a single-stage pressure ratio of 3 takes 2 (1.98) stages & the stage exit temp ~183°F (starting @ 40°F)

$$m = \frac{\ln\left(\dfrac{1014.7}{114.7}\right)}{\ln(3)} = \frac{\ln(8.8)}{\ln(3)} = 1.98$$

$$T_2 = T_1\left[\left(\frac{P_2}{P_1}\right)^{1/m}\right]^{(\gamma-1)/\gamma} = (40+460)\left[\left(\frac{1014.7}{114.7}\right)^{1/2}\right]^{(1.3-1)/1.3} = 643°\text{R} \quad \Rightarrow \quad 183°\text{F}$$

Multistaging

Work for a single stage of compression

$$W_s = \frac{\gamma R T_1}{M(\gamma - 1)}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right] = \frac{(1.3)(1.986)(500)}{(16)(1.3-1)}\left[\left(\frac{1014.7}{114.7}\right)^{(1.3-1)/1.3} - 1\right] = 175.8 \text{ Btu/lb}$$

Work for two stages of compression (interstage cooling to 40°F)

- Intermediate pressure

$$P_{int} = \sqrt{P_2 P_1} = \sqrt{(1014.7)(114.7)} = 341.2 \text{ psia}$$

- Total work

$$W_s = \frac{(1.3)(1.986)(500)}{(16)(1.3-1)}\left\{\left[\left(\frac{341.2}{114.7}\right)^{(1.3-1)/1.3} - 1\right] + \left[\left(\frac{1014.7}{341.2}\right)^{(1.3-1)/1.3} - 1\right]\right\}$$
$$= 153.9 \text{ Btu/lb}$$

Compression Efficiency

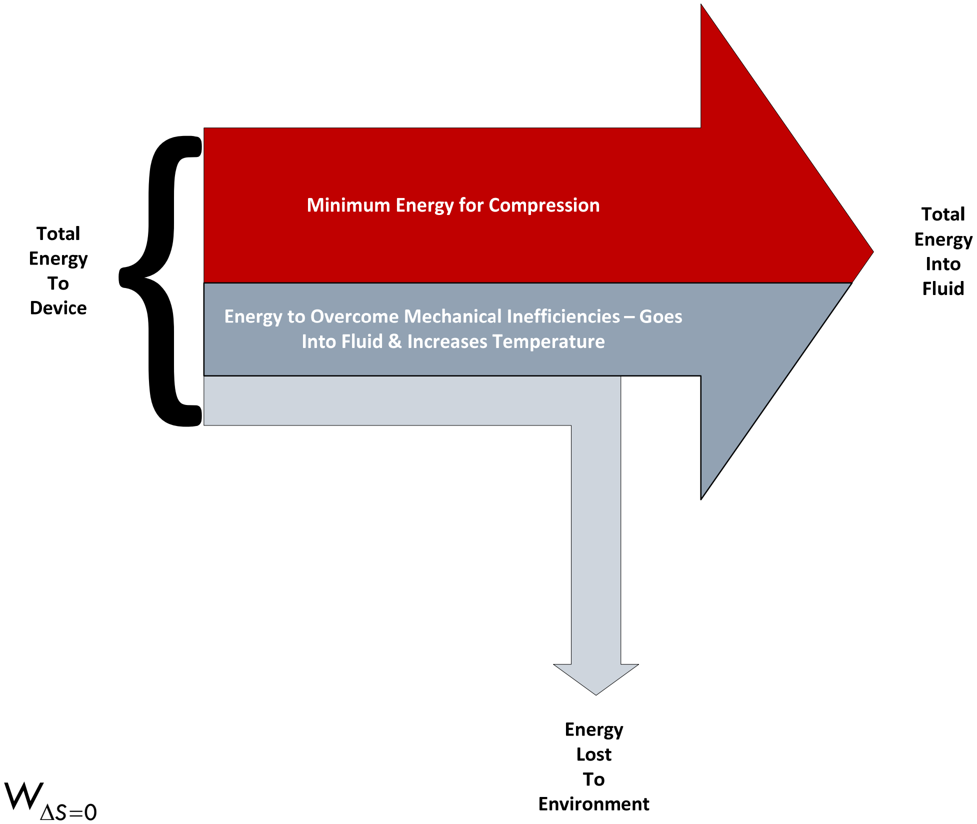
Compression efficiencies account for actual power required compared to ideal

- Isentropic (also known as adiabatic) efficiency relates actual energy to fluid to energy for reversible compression

$$\eta_{IS} = \frac{(\Delta H)_{\Delta S=0}}{(\Delta H)_{fluid}} \quad \Rightarrow \quad W_{fluid} = \frac{W_{\Delta S=0}}{\eta_{IS}}$$

- Mechanical efficiency relates total work to device to the energy into the fluid

$$\eta_{mech} = \frac{W_{fluid}}{W_{total}} \quad \Rightarrow \quad W_{total} = \frac{W_{fluid}}{\eta_{mech}} = \frac{W_{\Delta S=0}}{\eta_{mech}\eta_{IS}}$$



Compressor Efficiency – Discharge Temperature

GPSA Engineering Data Book suggests the isentropic temperature change should be divided by the isentropic efficiency to get the actual discharge temperature

$$(\Delta T)_{\Delta S=0} = T_1 \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - T_1 = T_1 \left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right]$$

So: $$(\Delta T)_{act} = \frac{(\Delta T)_{\Delta S=0}}{\eta_{IS}} = T_1 \frac{\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1}{\eta_{IS}}$$

$$\therefore \quad T_{2,act} = T_1 + (\Delta T)_{act} = T_1 \left[1 + \frac{\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1}{\eta_{IS}}\right]$$

Polytropic Compression & Efficiency

Definition of polytropic compression (*GPSA Data Book* 14th ed.):

> A reversible compression process between the compressor inlet and discharge conditions, which follows a path such that, between any two points on the path, the ratio of the reversible work input to the enthalpy rise is constant. In other words, the compression process is described as an *infinite number of isentropic compression steps*, each *followed by an isobaric heat addition*. The result is an ideal, reversible process that has the same suction pressure, discharge pressure, suction temperature and discharge temperature as the actual process.

Polytropic Efficiency

Polytropic path with 100% efficiency is adiabatic & is the same as the isentropic path

- Polytropic efficiency, η_p, is related to the isentropic path

$$\eta_P = \frac{(\gamma - 1)/\gamma}{(\kappa - 1)/\kappa}$$

In general $\eta_P > \eta_{IS}$

Polytropic coefficient from discharge temperature

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\left(\frac{\kappa - 1}{\kappa} \right)} \quad \Rightarrow \quad \kappa = \frac{1}{1 - X} \quad \text{where} \quad X \equiv \frac{\ln(T_2 / T_1)}{\ln(P_2 / P_1)}$$

Polytropic Efficiency

Actual work is calculated from the polytropic expression divided by its efficiency

$$\hat{W}_{act} = \frac{\hat{W}_p}{\eta_p} = \frac{1}{\eta_p} \frac{RT_1}{M} \frac{\kappa}{\kappa - 1} \left[\left(\frac{P_2}{P_1} \right)^{(\kappa-1)/\kappa} - 1 \right]$$

Note:

$$\hat{W}_{act} = \frac{\hat{W}_p}{\eta_p} = \frac{\hat{W}_{\Delta S=0}}{\eta_{IS}}$$

Compressor efficiency example

Compress methane from 40°F & 100 psig to 400 psig @ 80% isentropic efficiency & 10% mechanical losses

Actual work required is:

$$\hat{W}_{fluid} = \frac{1}{\eta_{IS}} \frac{RT_1}{M} \frac{\gamma}{\gamma - 1} \left[\left(\frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} - 1 \right] = \frac{1}{0.8} \frac{(1.986)(500)}{(16)} \frac{1.3}{1.3 - 1} \left[\left(\frac{414.7}{114.7} \right)^{(1.3-1)/1.3} - 1 \right]$$

$$= 116 \text{ Btu/lb}$$

Discharge temperature:

$$T_{2,act} = T_1 \left[1 + \frac{\left(\frac{P_2}{P_1} \right)^{(\gamma-1)/\gamma} - 1}{\eta_{IS}} \right] = (500) \left[1 + \frac{\left(\frac{414.7}{114.7} \right)^{(1.3-1)/1.3} - 1}{0.8} \right] = 716°\text{R} \quad \Rightarrow \quad 256°\text{F}$$

Compressor efficiency example

Using polytropic pathway:

$$\mathrm{X} = \frac{\ln\left(T_2 / T_1\right)}{\ln\left(P_2 / P_1\right)} = \frac{\ln\left(716 / 500\right)}{\ln\left(414.7 / 114.7\right)} = 0.2792$$

$$\kappa = \frac{1}{1-\mathrm{X}} = \frac{1}{1-0.2792} = 1.387$$

$$\eta_P = \frac{\left(\gamma - 1\right) / \gamma}{\left(\kappa - 1\right) / \kappa} = \frac{\left(1.3 - 1\right) / 1.3}{\left(1.387 - 1\right) / 1.387} = 0.827$$

$$\hat{W}_{fluid} = \frac{1}{\eta_p} \frac{RT_1}{M} \frac{\kappa}{\kappa - 1} \left[\left(\frac{P_2}{P_1} \right)^{(\kappa - 1)/\kappa} - 1 \right] = \frac{1}{0.827} \frac{(1.986)(500)}{16} \frac{1.387}{1.387 - 1} \left[\left(\frac{414.7}{114.7} \right)^{(1.387-1)/1.387} - 1 \right]$$

$$= 116 \text{ Btu/lb}$$

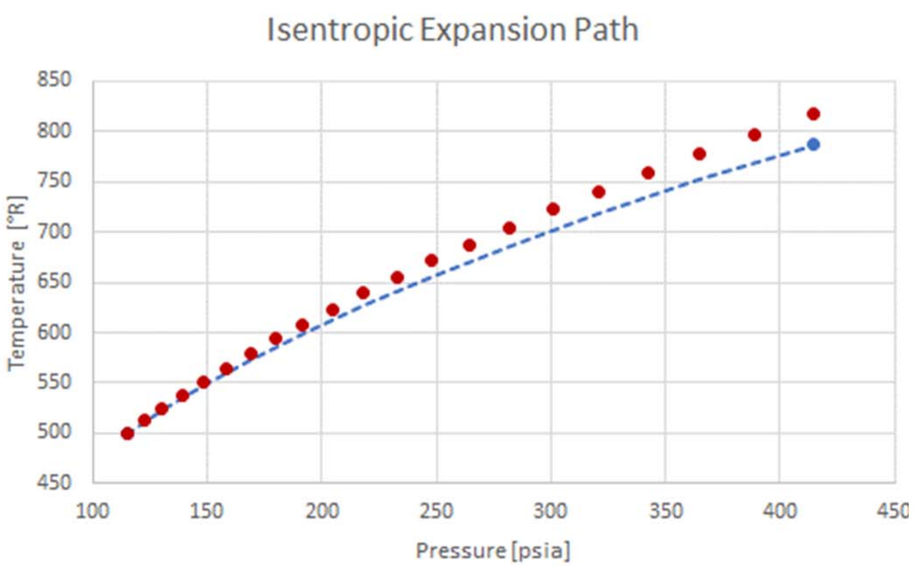
Compressor efficiency example

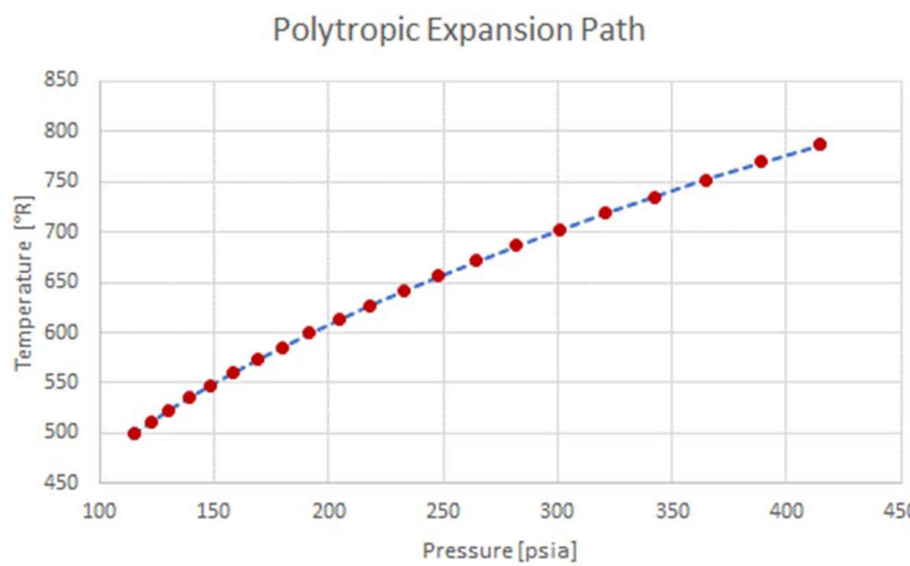
Can use either expression for the power to the fluid to determine the total power to the compressor

$$W_{total} = \frac{W_{fluid}}{\eta_{mech}} = \frac{116}{1-0.1} = 129 \text{ Btu/lb}$$

Why Use Polytropic Equations?

Polytropic equations give consistent P-T pathway between the initial & discharge conditions





Compression vs. Expansion Efficiency

Work to compressor is <u>greater</u> than what is needed in the ideal case

- Work to the fluid

$$\eta_{IS} = \frac{(\Delta H)_{\Delta S=0}}{(\Delta H)_{fluid}} \quad \Rightarrow \quad W_{fluid} = \frac{W_{\Delta S=0}}{\eta_{IS}}$$

- Total work to the device

$$\eta_{mech} = \frac{W_{fluid}}{W_{total}} \quad \Rightarrow$$

$$W_{total} = \frac{W_{fluid}}{\eta_{mech}} = \frac{W_{\Delta S=0}}{\eta_{mech}\eta_{IS}}$$

Work from expander is <u>less</u> than what can be obtained in the ideal case

- Work from the fluid

$$\eta_{IS} = \frac{(\Delta H)_{fluid}}{(\Delta H)_{\Delta S=0}} \quad \Rightarrow \quad W_{fluid} = \eta_{IS}\left(W_{\Delta S=0}\right)$$

- Total work from the device

$$\eta_{mech} = \frac{W_{total}}{W_{fluid}} \quad \Rightarrow$$

$$W_{total} = \eta_{mech} W_{fluid} = \eta_{mech}\eta_{IS} W_{\Delta S=0}$$

Equipment:
Pumps, Compressors, Turbines/Expanders

Updated: February 6, 2018

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Pump & Compressor Drivers

Internal combustion engines

- Industry mainstay from beginning
- Emissions constraints
- Availability is 90 to 95%

Electric motors

- Good in remote areas
- Availability is > 99.9%

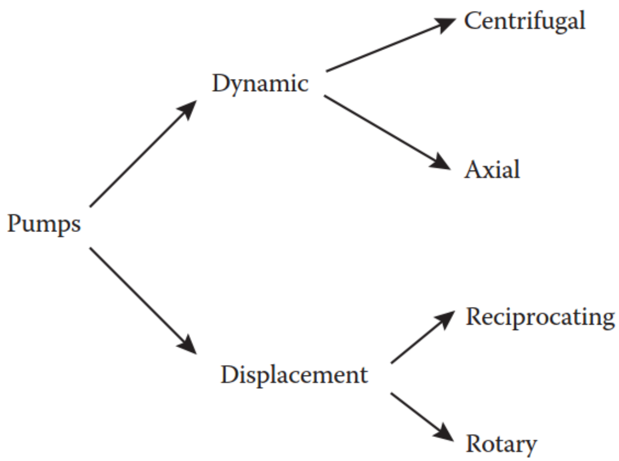
Gas turbines

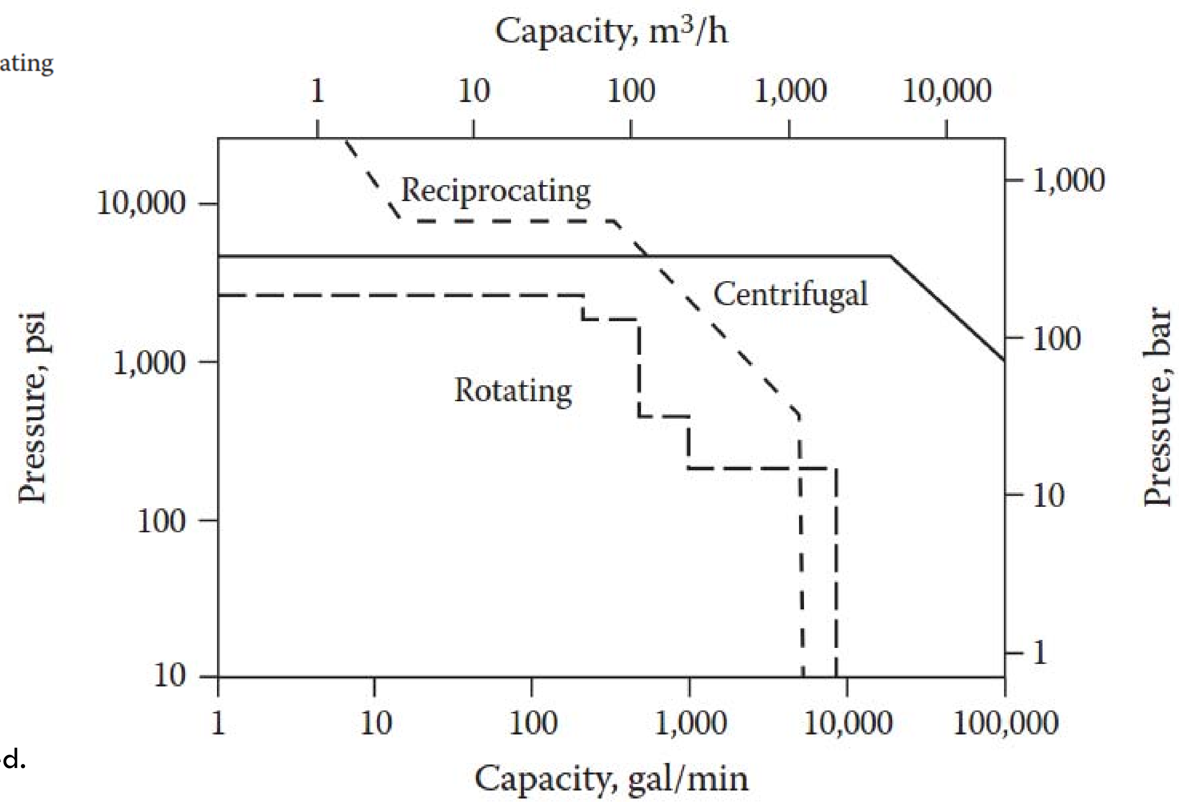
- Availability is > 99%
- Lower emissions than IC engine

Steam turbines

- Uncommon in gas plants on compressors
- Used in combined cycle and Claus units

Pump Classifications





Fundamentals of Natural Gas Processing, 2nd ed.
Kidnay, Parrish, & McCartney

Centrifugal Pump Performance Curves

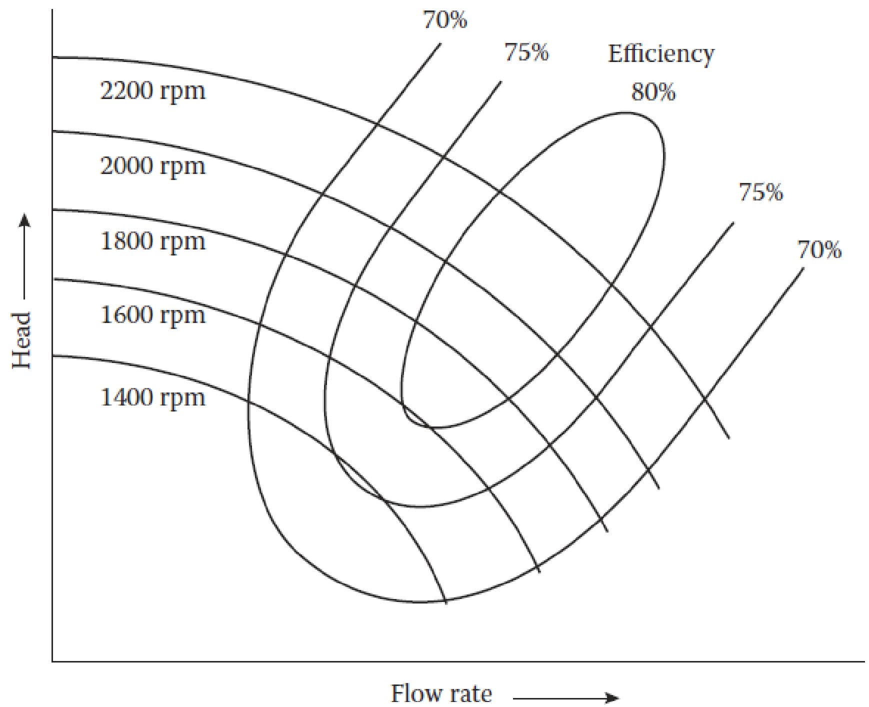


FIGURE 2.10 Effect of speed on head and efficiency.

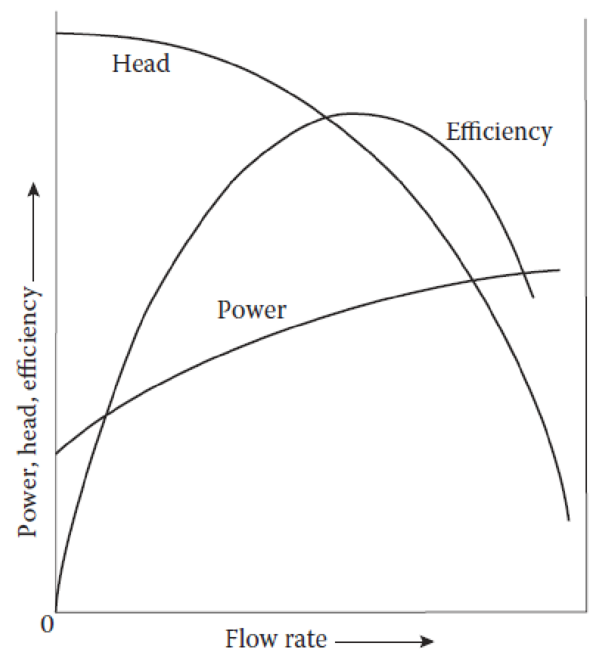


FIGURE 2.8 Typical centrifugal pump performance curves as a function of flow rate.

Fundamentals of Natural Gas Processing, 2nd ed.
Kidnay, Parrish, & McCartney

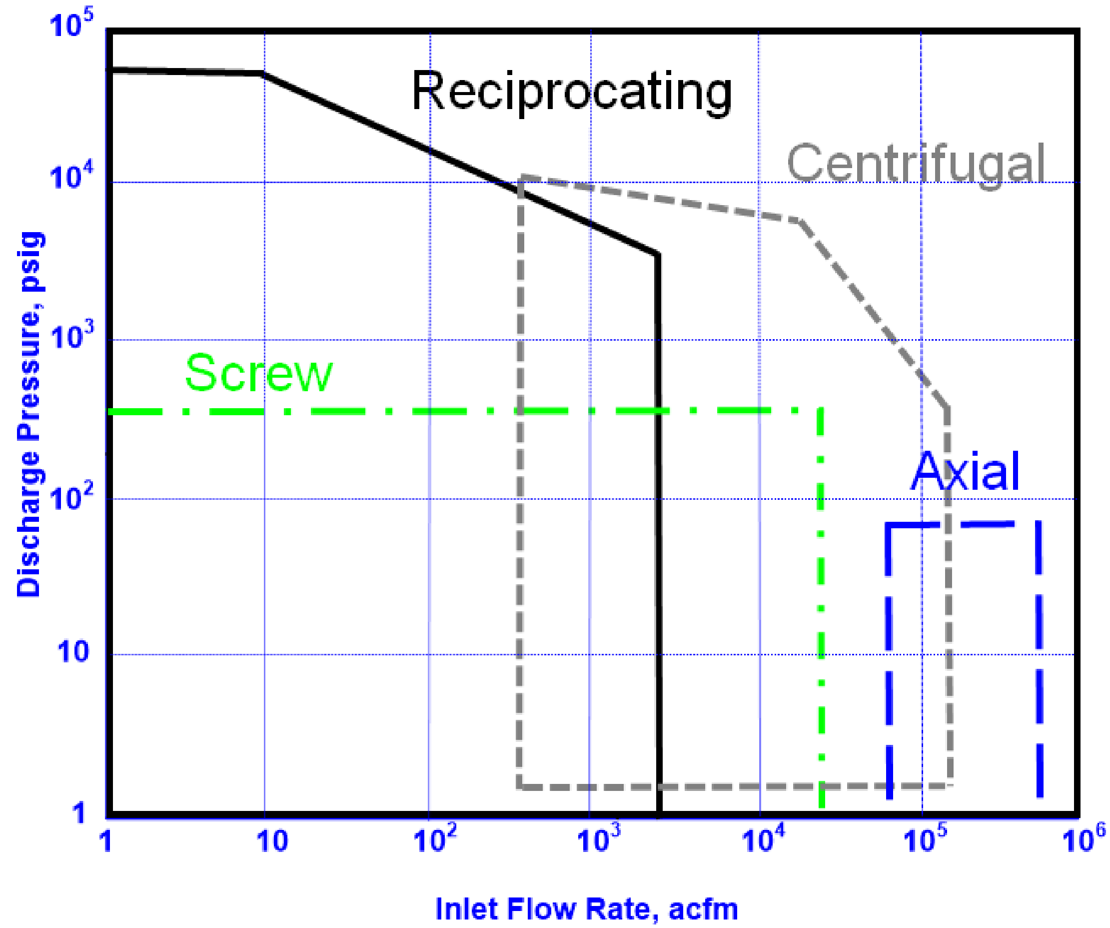
Compressor Types

Positive displacement – compress by changing volume

- Reciprocating
- Rotary screw
- Diaphragm
- Rotary vane

Dynamic – compress by converting kinetic energy into pressure

- Centrifugal
- Axial



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Reciprocating Compressors

Workhorse of industry since 1920's

Capable of high volumes and discharge pressures

High efficiency – up to 85%

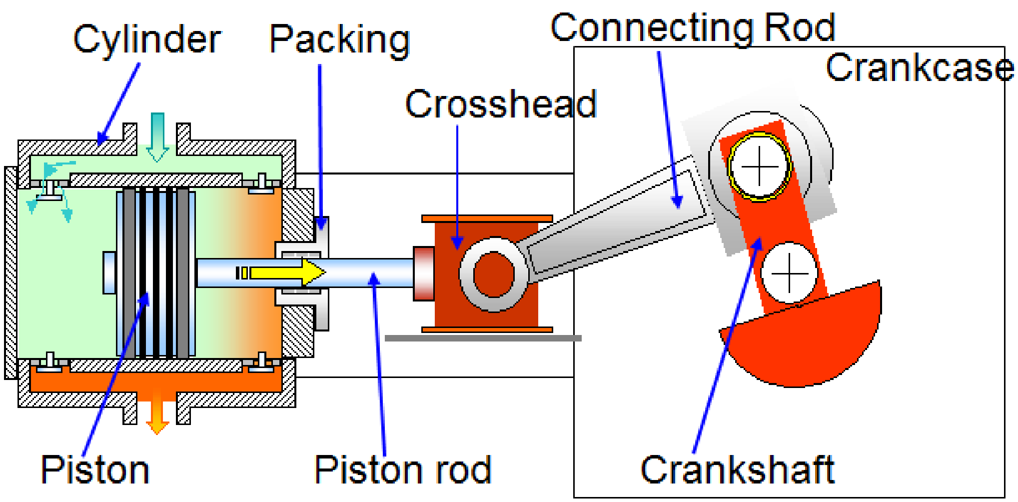
Performance independent of gas MW

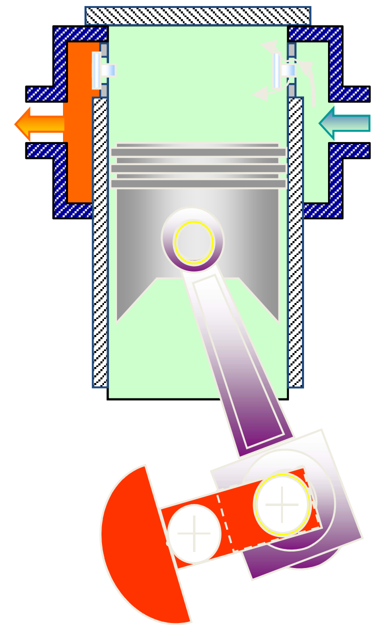
Good for intermittent service

Drawbacks

- Availability ~90 to 95% vs 99+% for others, spare compressor needed in critical service
- Pulsed flow
- Pressure ratio limited, typically 3:1 to 4:1
- Emissions control can be problem (IC drivers)
- Relatively large footprint
- Throughput adjusted by variable speed drive, valve unloading or recycle unless electrically driven

Reciprocating Compressors - Principle of Operation





Double Acting – Crosshead

Typical applications:

- All process services, any gas & up to the highest pressures & power

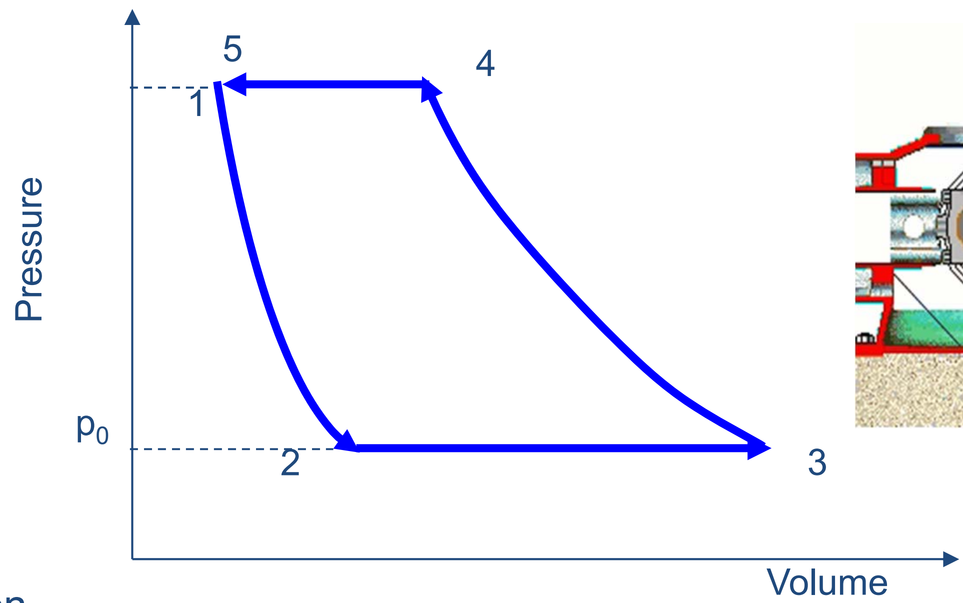
Single Acting - Trunk Piston

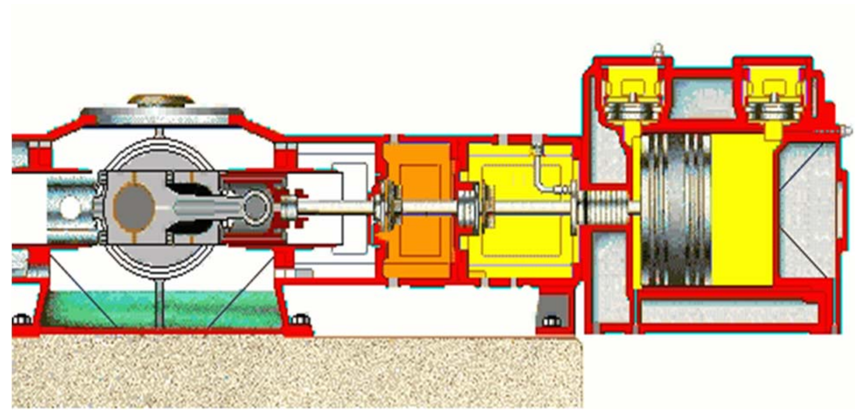
Typical Applications:

- Small size standard compressors for air and non-dangerous gases

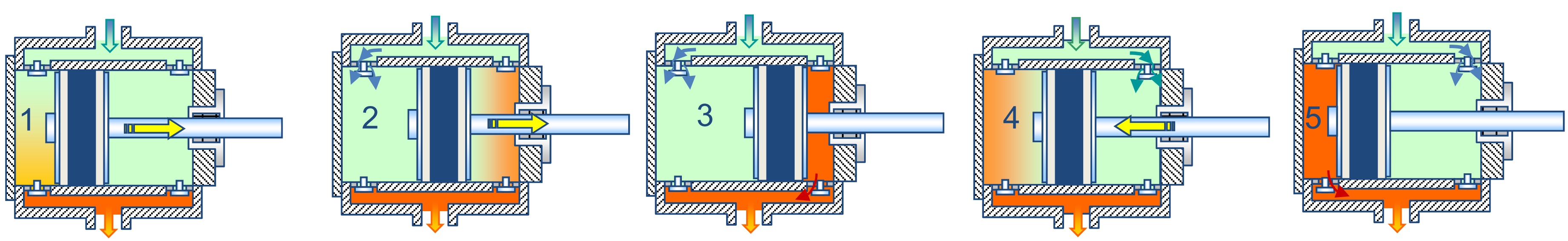
Courtesy of Nuovo Pignone Spa, Italy

Reciprocating Compressors - Compression Cycle





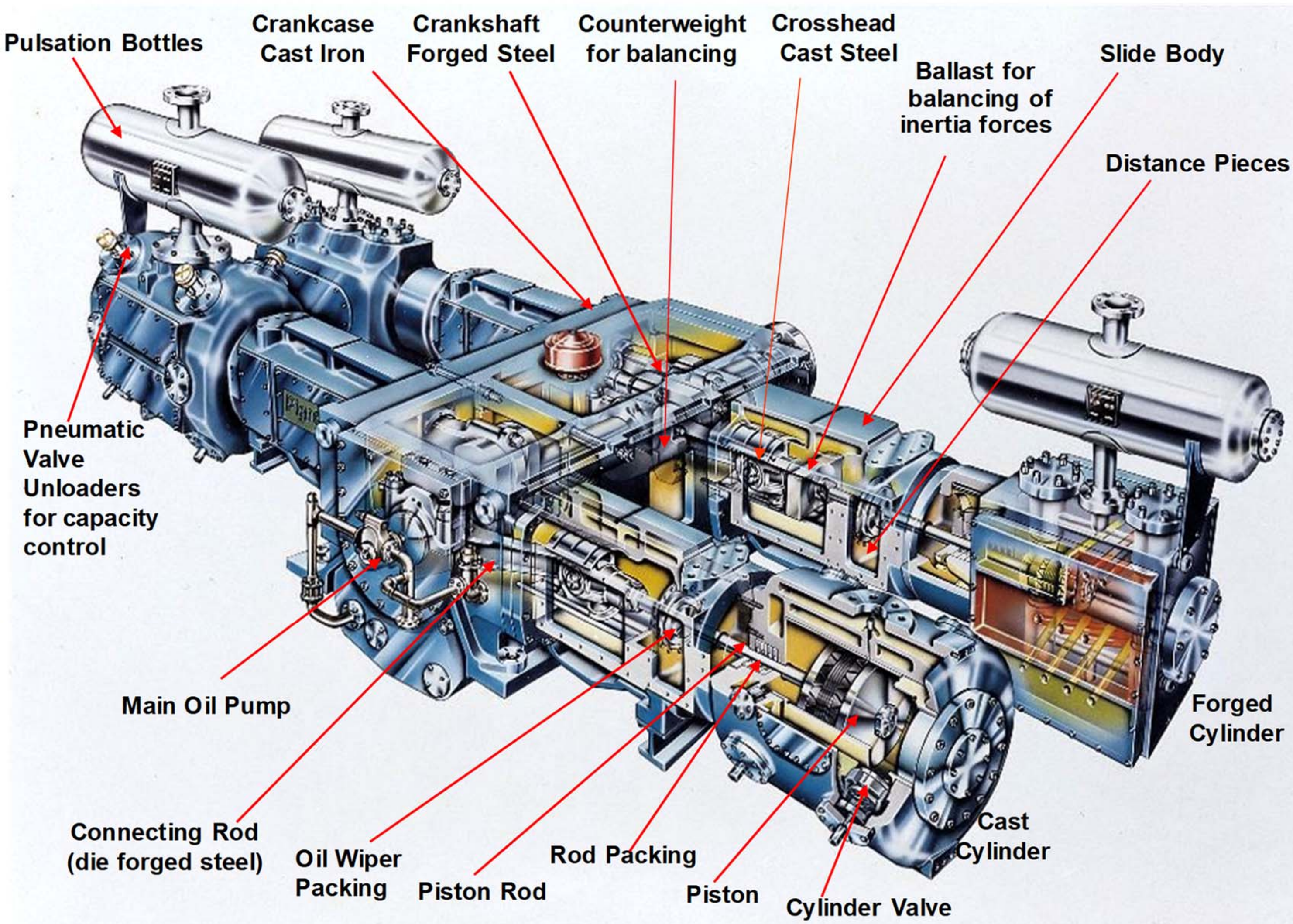
Suction



Discharge

Courtesy of Nuovo Pignone Spa, Italy

Reciprocating Compressors - Main Components

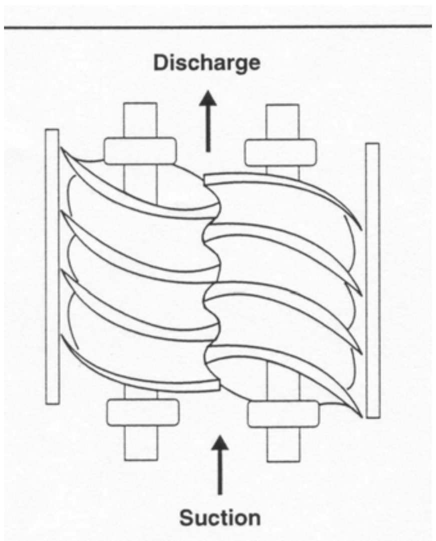


Reciprocating Compressors

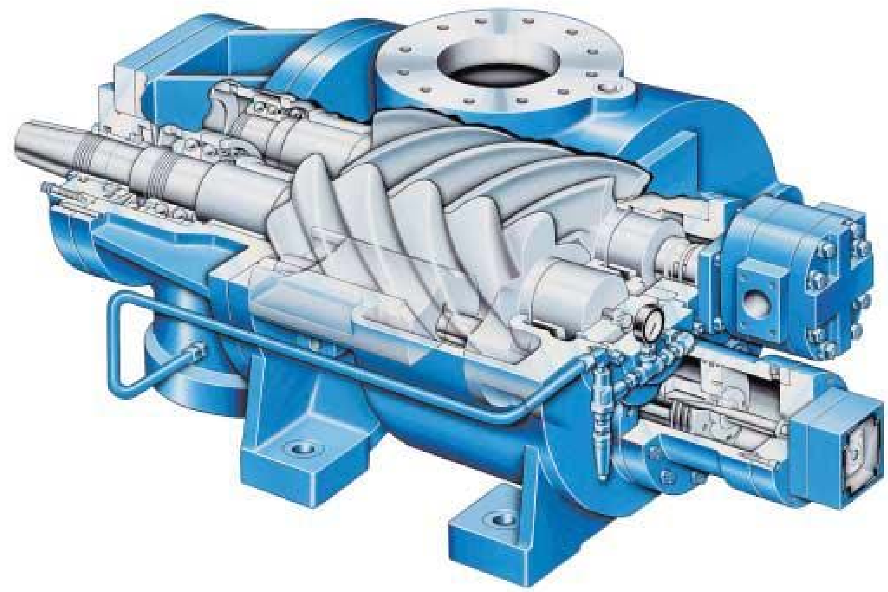
https://www.youtube.com/watch?v=E6_jw841vKE

Rotary Screw Compressor

Left rotor turns clockwise, right rotor counterclockwise.
Gas becomes trapped in the central cavity



The Process Technology Handbook, Charles E. Thomas, UHAI Publishing, Berne, NY, 1997.



Courtesy of Ariel Corp

Rotary Screw Compressors

Oil-free

First used in steel mills because handles "dirty" gases

Max pressure ratio of 8:1 if liquid injected with gas

High availability (> 99%)

- Leads to low maintenance cost

Volumetric efficiency of ~100%

Small footprint (~ ¼ of recip)

Relatively quiet and vibration-free

Relatively low efficiency

- 70 – 85% adiabatic efficiencies

Relatively low throughput and discharge pressure

Oil-injected

Higher throughput and discharge pressures

Has two exit ports

- Axial, like oil-free
- Radial, which permits 70 to 90% turndown without significant efficiency decrease

Pressure ratios to 23:1

Tight tolerances can limit quick restarts

Requires oil system to filter & cool oil to 140°F

Oil removal from gas

Oil compatibility is critical

Widely used in propane refrigeration systems, low pressure systems, e.g., vapor recovery, instrument air

Dynamic Compressors

Centrifugal

- High volumes, high discharge pressures

Axial

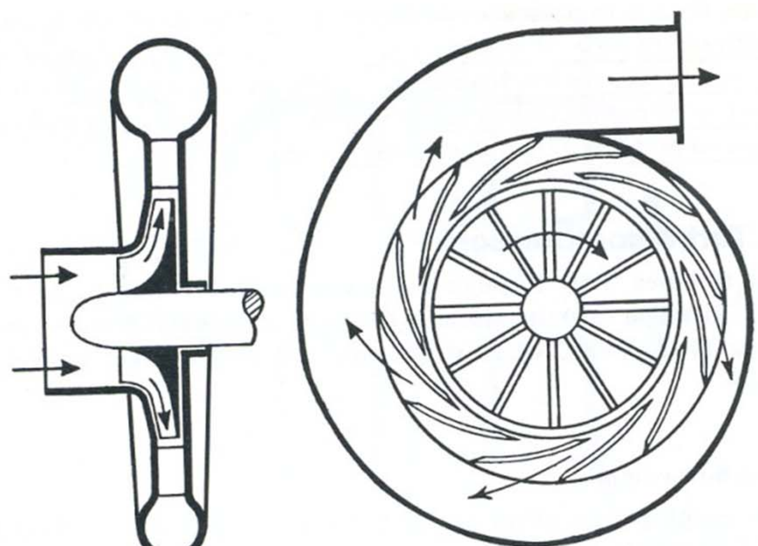
- Very high volumes, low discharge pressures

Use together in gas processing

- Centrifugal for compressing natural gas
- Axial for compressing air for gas turbine driving centrifugal compressor

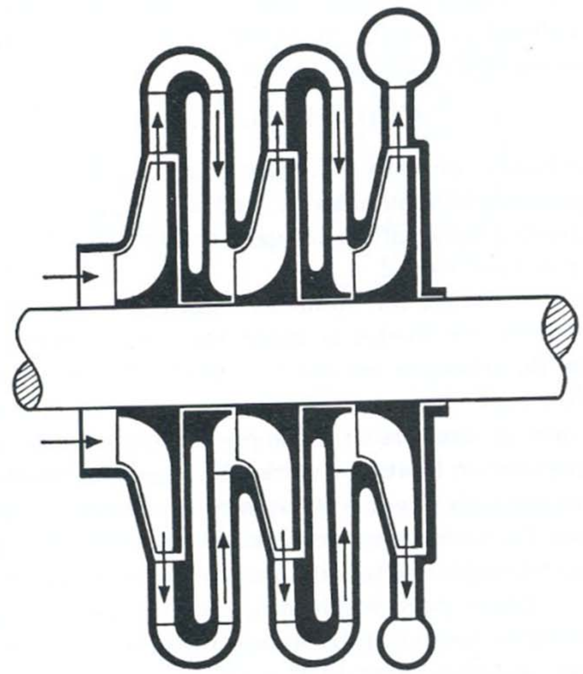
Centrifugal compressors

Single stage (diffuser)



(a)

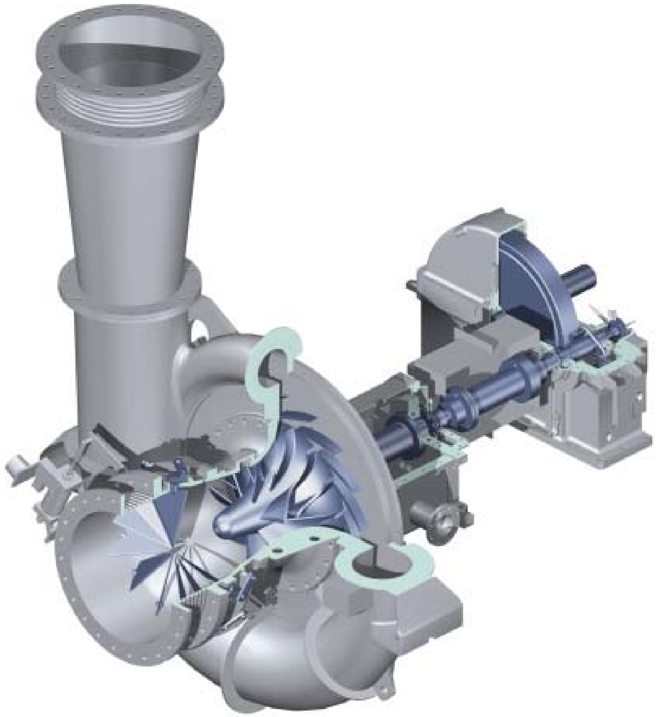
Multi-stage



(b)

Bett,K.E., et al
Thermodynamics for Chemical Engineers
Page 226

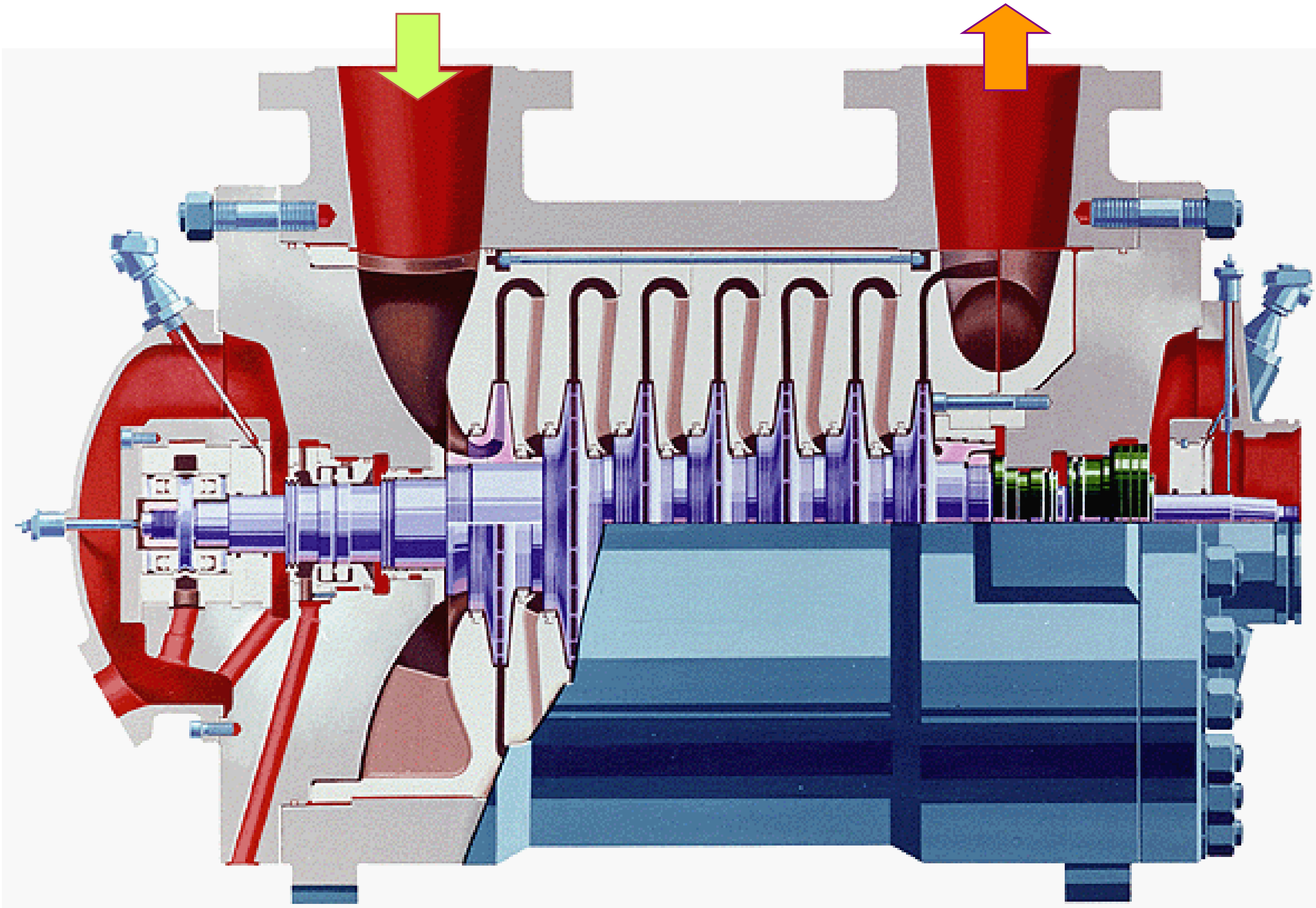
Centrifigual Compressor



Siemens
https://www.energy.siemens.com/br/en/compression-expansion/product-lines/single-stage/stc-sof.htm

https://www.youtube.com/watch?v=s-bbAoxZmBg

Centrifugal Compressor



Courtesy of Nuovo Pignone Spa, Italy

Centrifugal Compressors vs. Reciprocating Compressors

Centrifugal

Constant head, variable volume

Ideal for variable flow

- MW affects capacity

++ Availability > 99%

+ Smaller footprint

- $\eta_{IS} = 70 - 75\%$

CO & NOx emissions low

- Surge control required

++ Lower CAPEX and maint.

(maint cost ~1/4 of recip)

Reciprocating

Constant volume, variable pressure

Ideal for constant flow

+ MW makes no difference

- Availability 90 to 95%

- Larger footprint

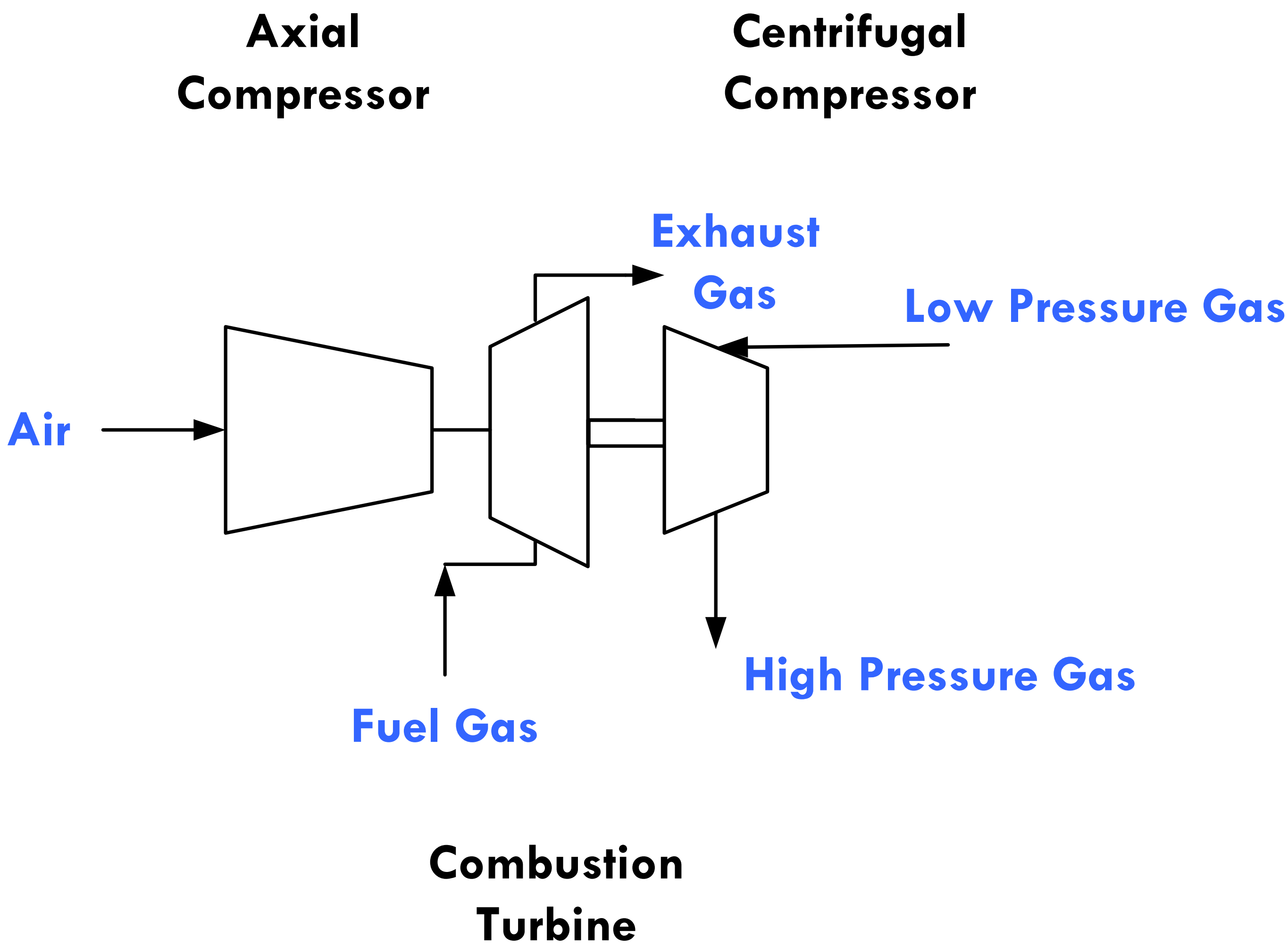
+ $\eta_{IS} = 75 - 92\%$

Catalytic converters needed

++ No surge problems

++ Fast startup & shutdown

Gas Turbine – Centrifugal Compressor



Industrial Gas Turbines

FIG. 15-32
2011 Basic Specifications — Gas Turbine Engines (Mechanical Drive)

Model	Power Rating (ISO Rating) hp	Heat Rate (LVH) Btu/hp-hr	Pressure Ratio	Power Shaft RPM	At ISO RATING CONDITIONS		
					Turbine Inlet Temp. °F	Exhaust Flow lb/s	Exhaust Temp °F
Dresser- Rand							
VECTRA 30G	31,469	6816	17.9	6510	1530	149.7	1017
VECTRA 40G	42,102	6347	22.4	6510	1521	190.2	979
VECTRA 40G4	45,902	6316	23.6	6510	1571	198.4	1006
DR-63G PC	59,436	6042	27.9	3780	1578	280.0	855
DR-63G PG	66,822	6054	29.7	3930	1666	259.3	907
GE Oil & Gas							
GE10-2 DLE	15907.2	7762.2	15.8	7900		103.6	912
GE10-2	16288.1	7620.9	15.6	7900		103.6	901
PGT16	19143.1	7042.7	20.1	7900		103.8	928
PGT20 SAC	24300.4	6974.1	19.7	6500		138.0	895
PGT20 DLE	24926.9	6984.7	19.8	6500		137.3	915
PGT25 DLE	31194.9	6793.2	17.9	6500		151.0	983
PGT25 SAC	31205.6	6756.4	17.9	6500		151.9	971
MS5002C	37950.9	8700.9	8.8	4670		274.0	963

Ref: *GPSA Data Book*, 13th ed.

What is "heat rate"?

Heat rate is the amount of fuel gas needed (expressed heating value) to produce a given amount of power

- Normally LHV, but you need to make sure of the basis

Essentially the reciprocal of the thermal efficiency

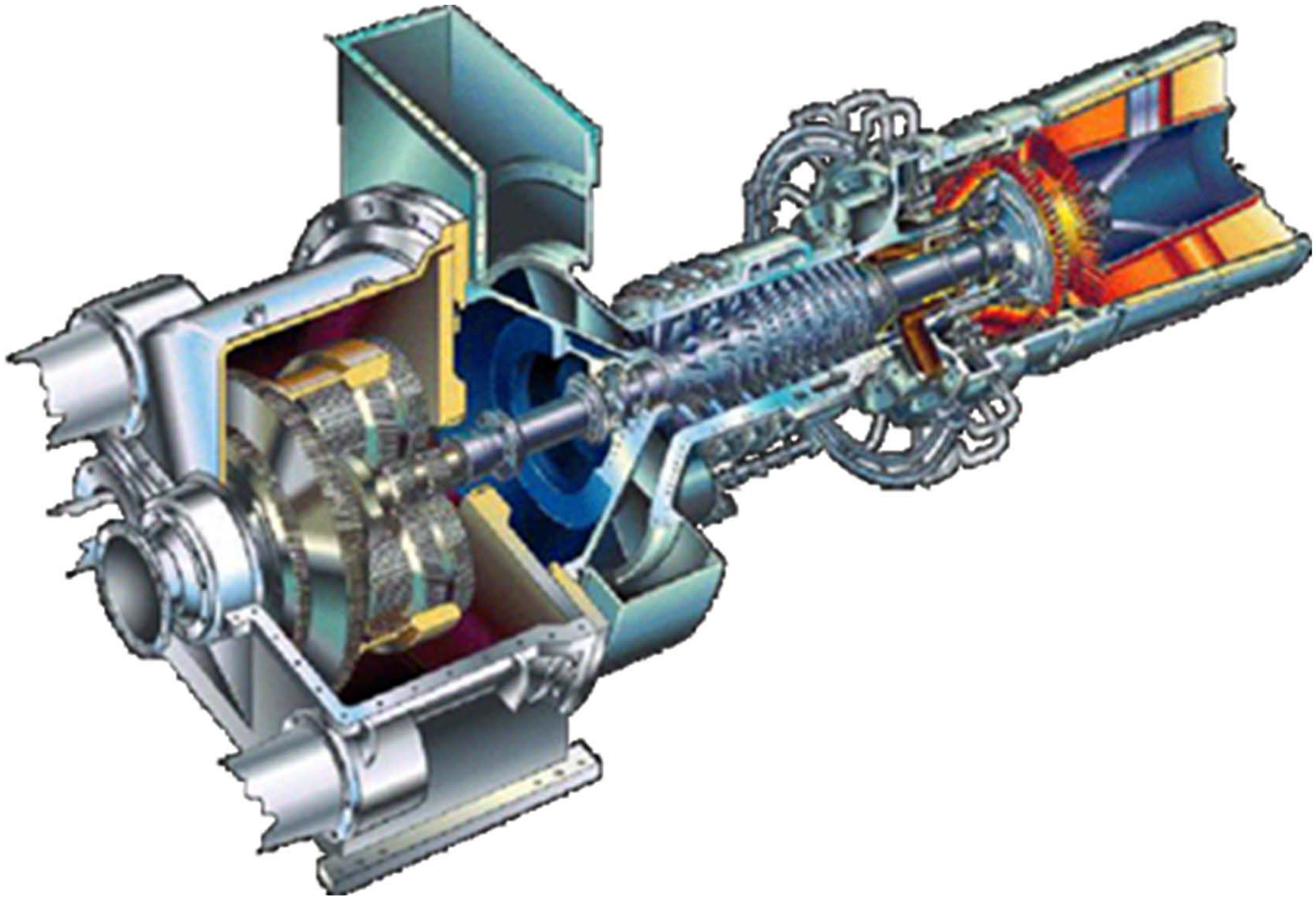
$$\text{Thermal efficiency} = \frac{2544}{\text{Heat rate, } \dfrac{\text{Btu}(\text{LHV})}{\text{hp} \cdot \text{hr}}}$$

- Example: Dresser-Rand VECTRA 30G heat rate is 6816 Btu/hp·hr

$$\text{Thermal efficiency} = \frac{2544}{6816} = 0.3732$$

Includes effects of adiabatic & mechanical efficiencies

Gas Turbine

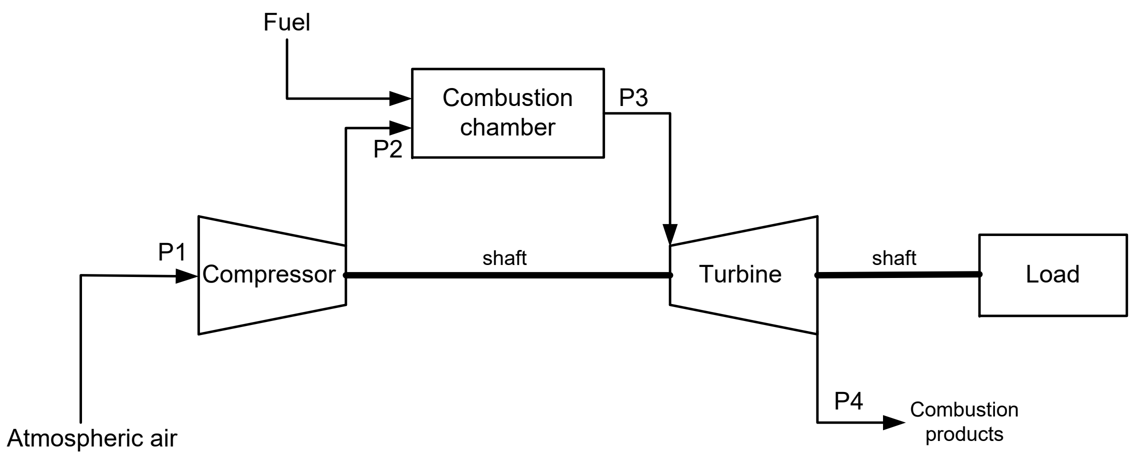


Courtesy of Nuovo Pignone Spa, Italy

Gas Turbine Engine

Gas Turbine Engine

From: F.W.Schmuidt, R.E. Henderson, and C.H. Wolgemuth, "Introduction to Thermal Sciences, second edition" Wiley, 1993



Assumptions

To apply basic thermodynamics to the process above, it is necessary to make a number of assumptions, some rather extreme.

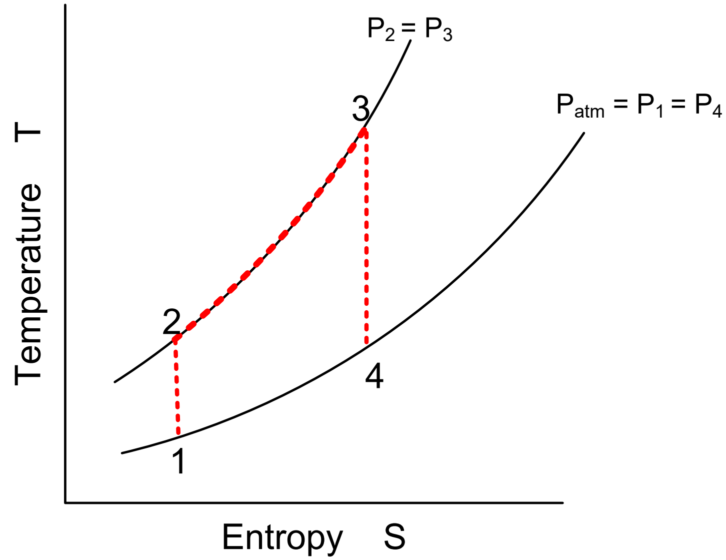
1) All gases are ideal, and compression processes are reversible and adiabatic (isentropic)

2) the combustion process is constant pressure, resulting only in a change of temperature

3) negligible potential and kinetic energy changes in overall process

4) Values of C_p are constant

Gas Turbine Engine



$w_S = -\Delta h = -C_P\Delta T$ *(9.1 and 1.18)*

Note the equations apply to both the compressor and the turbine,since thermodynamically the turbine is a compressor running backwards

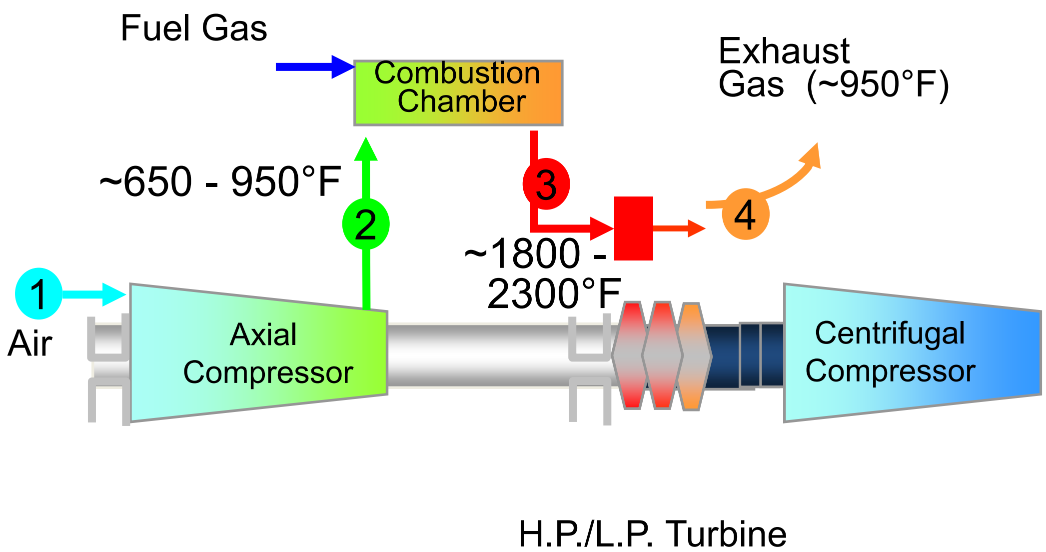
Neglecting the differences in mass flow rates between the compressor and the turbine, the net work is:

$w_{net} = w_t - w_c = C_P(T_3 - T_4) - (T_2 - T_1)$

Since $(T_3 - T_4) > (T_2 - T_1)$ (see T – S diagram)

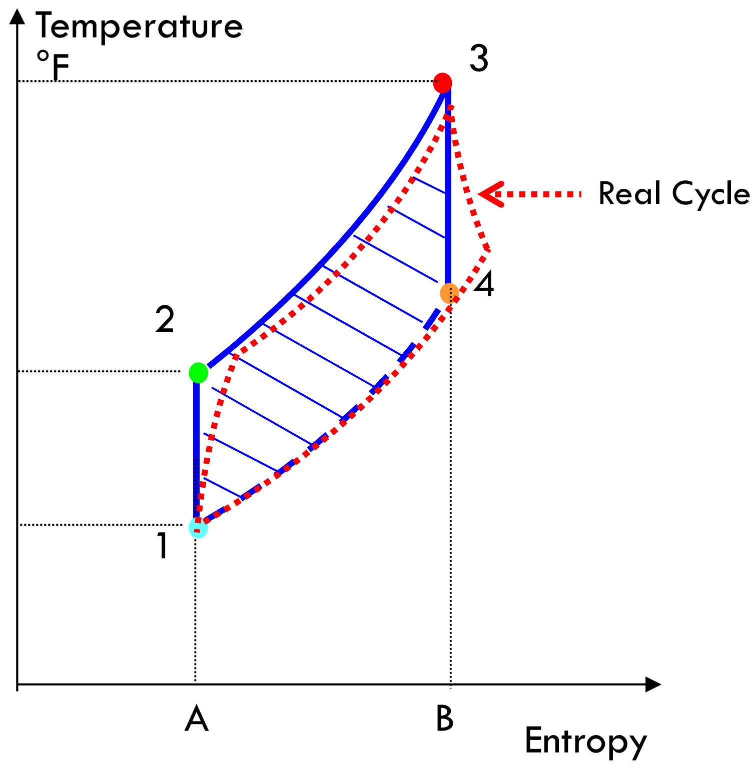
Since w_{net} is positive work flows to the load

GT - Principle of Operation



Simple Cycle Gas Turbine

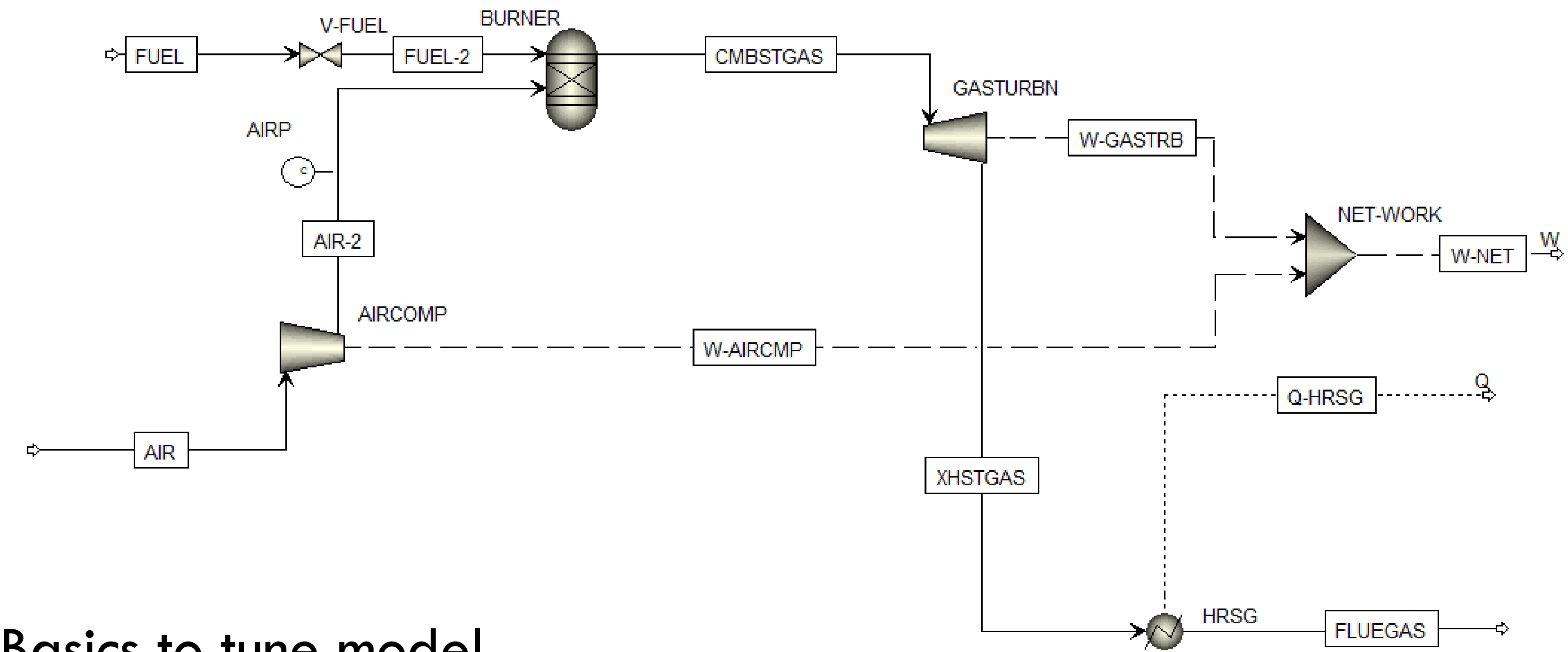
Theoretical Cycle



Ideal Cycle Efficiency

$$\eta_{id} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \left(\frac{P_1}{P_2} \right)^{(\gamma - 1)/\gamma}$$

Modeling Gas Turbine with Aspen Plus



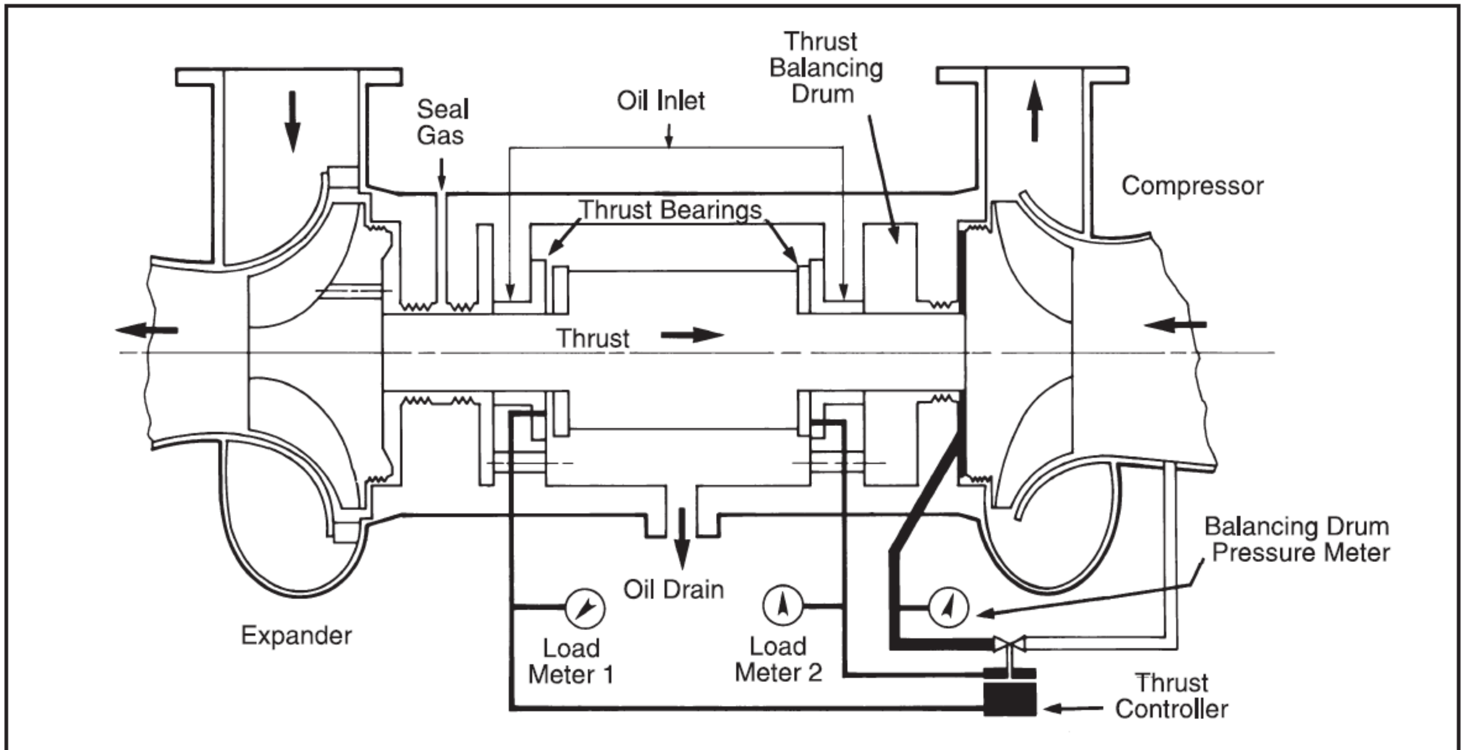
Basics to tune model

- Combine heat rate & power output to determine the fuel required
- Determine the air rate from the exhaust rate
- Adjust adiabatic efficiencies to match the exhaust temperature
- Adjust the mechanical efficiencies to match the power output

Turboexpanders

FIG. 13-75

Typical Expander/Compressor Cross-Section with Thrust Balancing Schematic



Courtesy of The Randall Corp.

GPSA Engineering Data Book, 14th ed.

Summary

Updated: February 6, 2018

Summary

Work expression for pump developed assuming density is not a function of pressure

Work of compression is much greater than that for pumping – a great portion of the energy goes to increase the temperature of the compressed gas

Need to limit the compression ratio on a gas

- Interstage cooling will result in decreased compression power required
- Practical outlet temperature limitation – usually means that the maximum compression ratio is about 3

There are thermodynamic/adiabatic & mechanical efficiencies

- Heat lost to the universe that does affect the pressure or temperature of the fluid is the mechanical efficiency

Supplemental Slides

Updated: February 6, 2018

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Reciprocating Compressors

Propane Refrigeration Compressors

Propane Compressors with Air-cooled Heat Exchangers

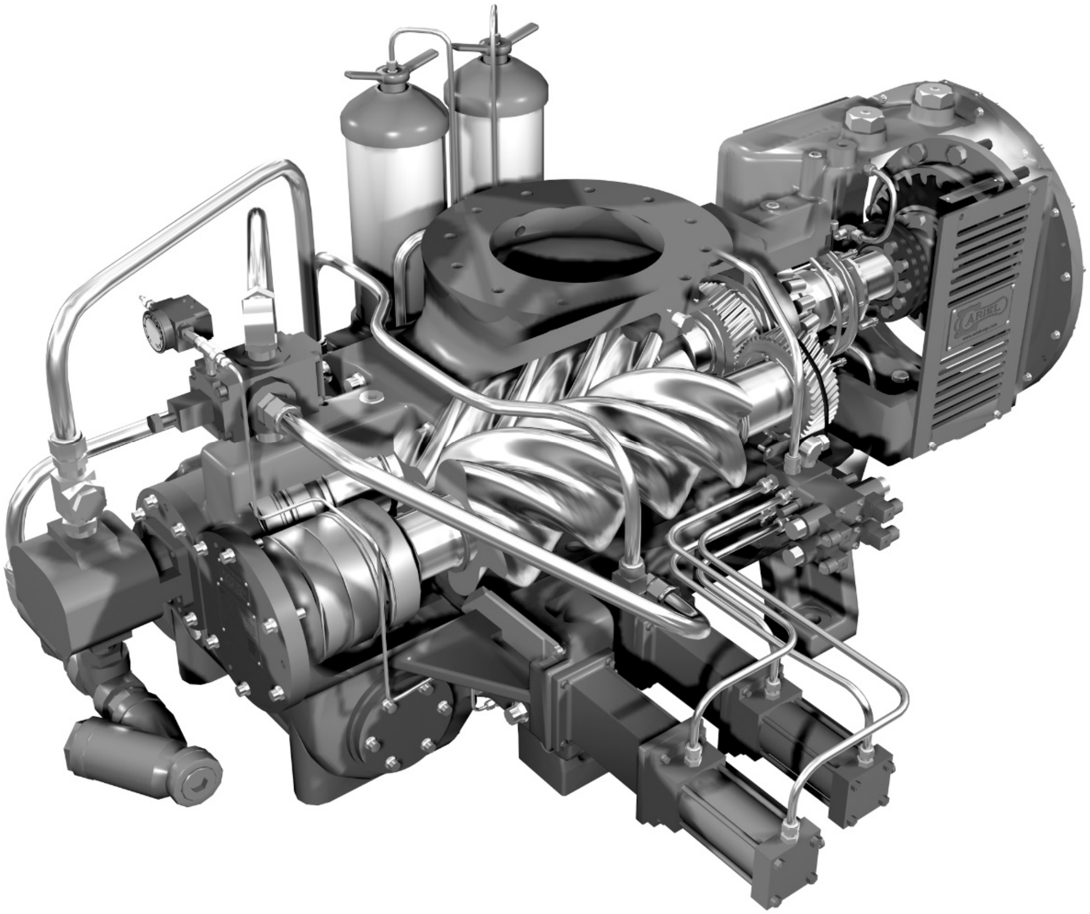
Reciprocating Compressor at Gas Well

Courtesy of Nuovo Pignone Spa, Italy

2 stage 2,000 HP Reciprocating Compressor

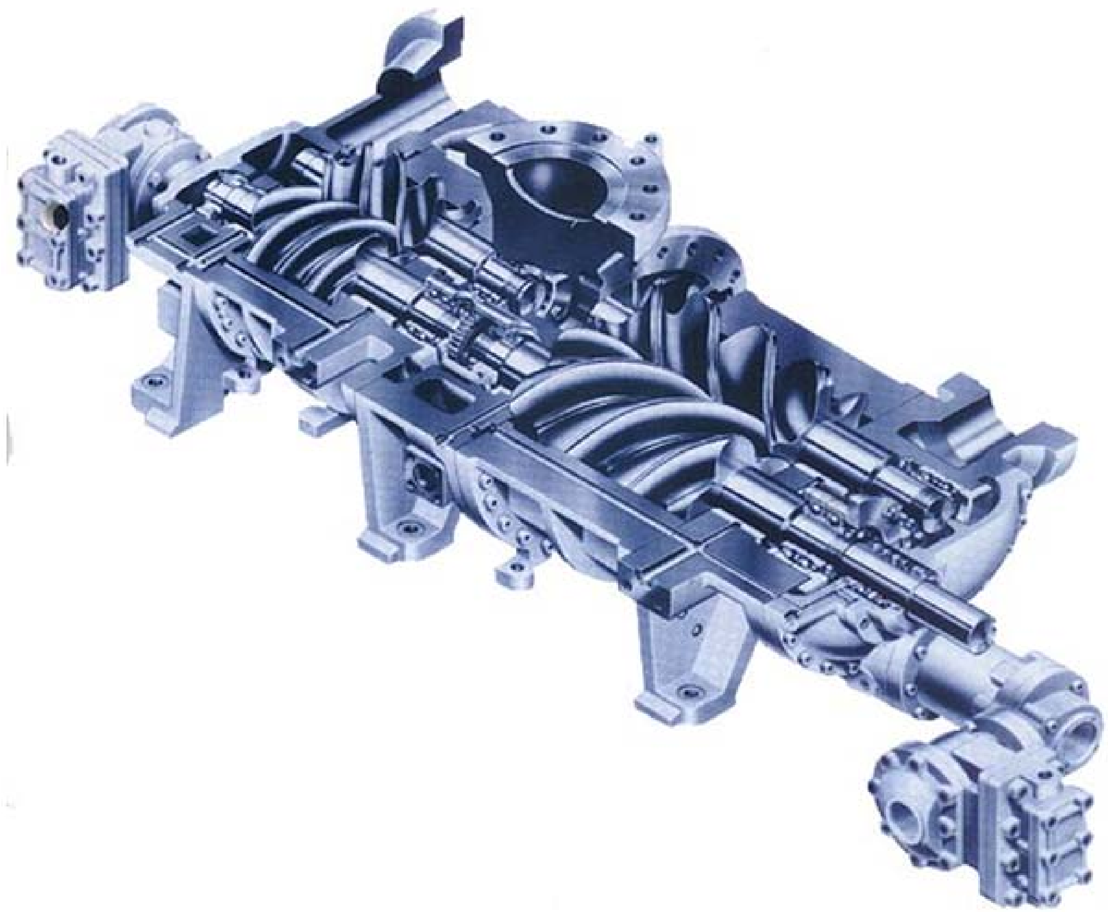
Courtesy of Ariel Corp

Oil-Injected Rotary Screw Compressor



Courtesy of Ariel Corp

Two-stage screw compressor



Courtesy of MYCOM / Mayekawa Mfg

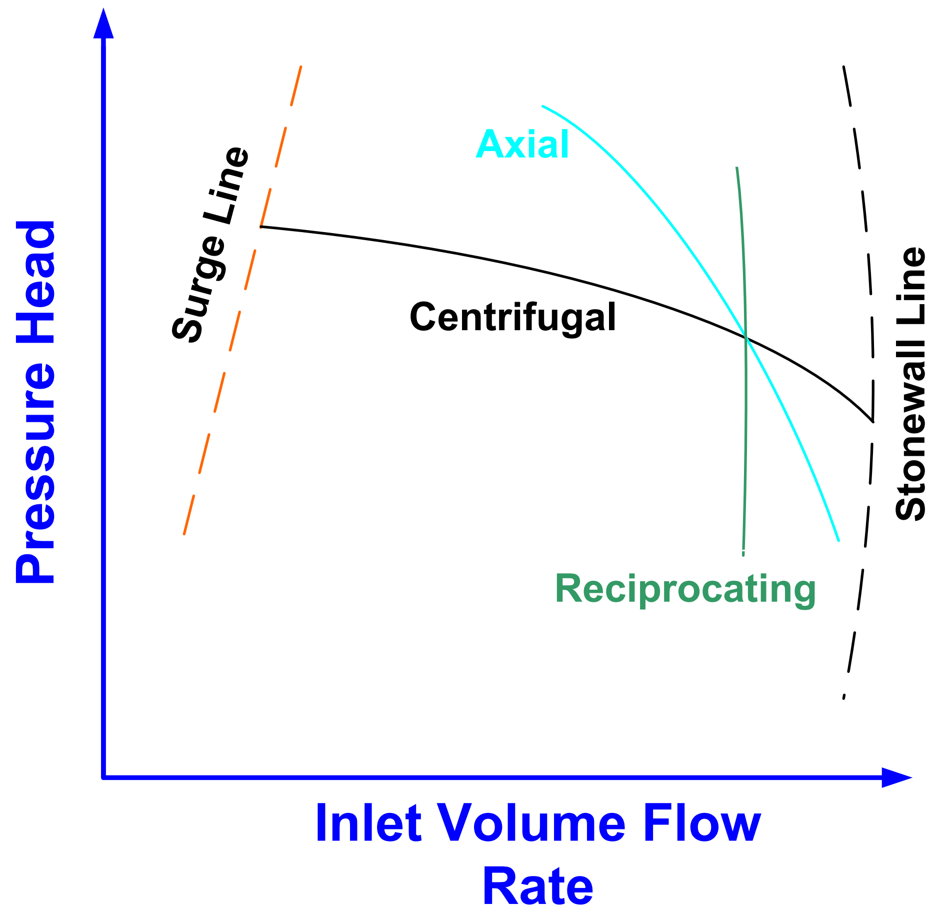
Centrifugal Compressors – Issues

Surge

- Changes in the suction or outlet pressures can cause backflow; this can become cyclic as the compressor tries to adjust. The resulting pressure oscillations are called SURGE

Stonewall

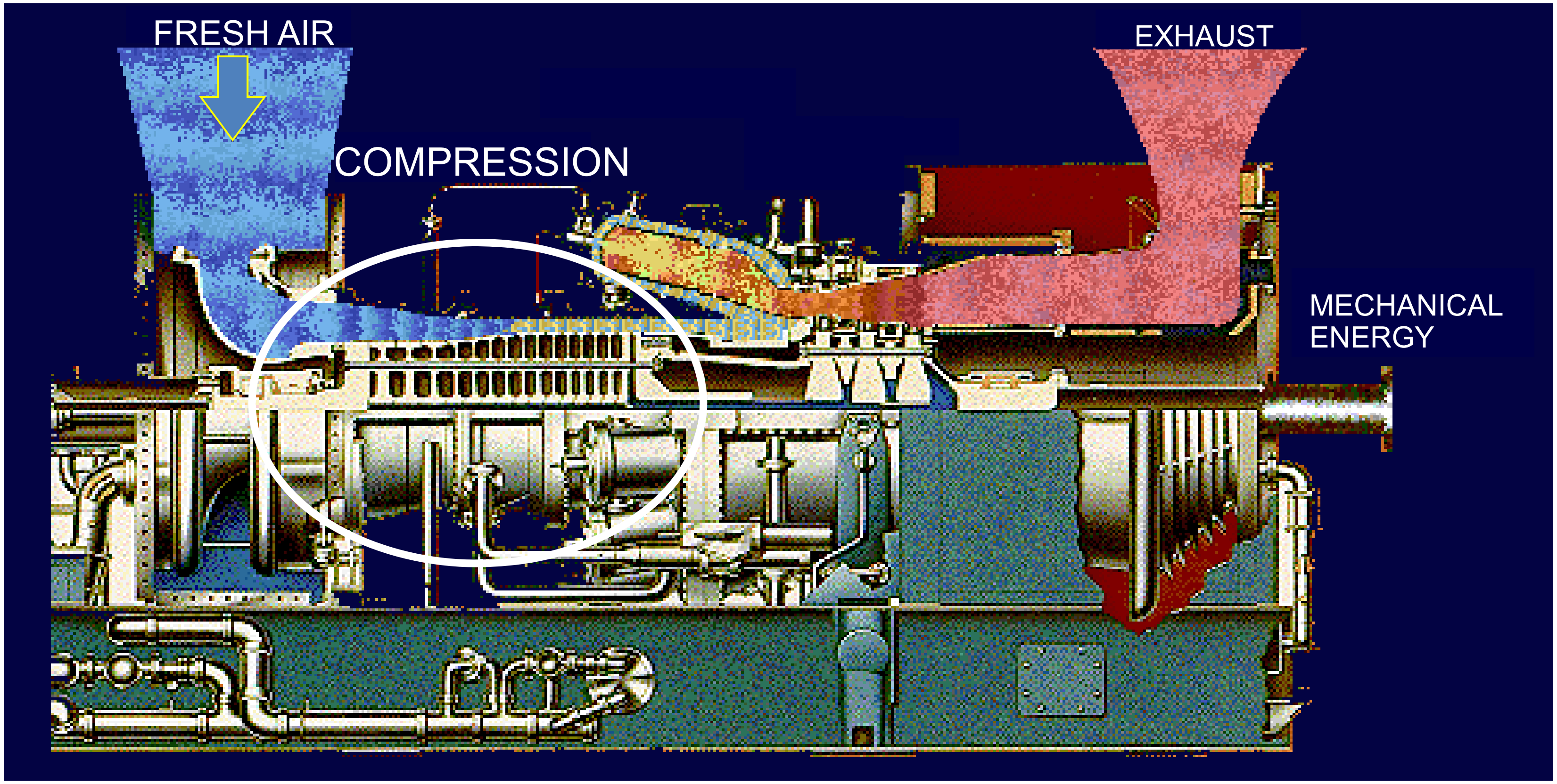
- When gas flow reaches sonic velocity flow cannot be increased.



Air & Hot Gas Paths

Gas Turbine has 3 main sections:

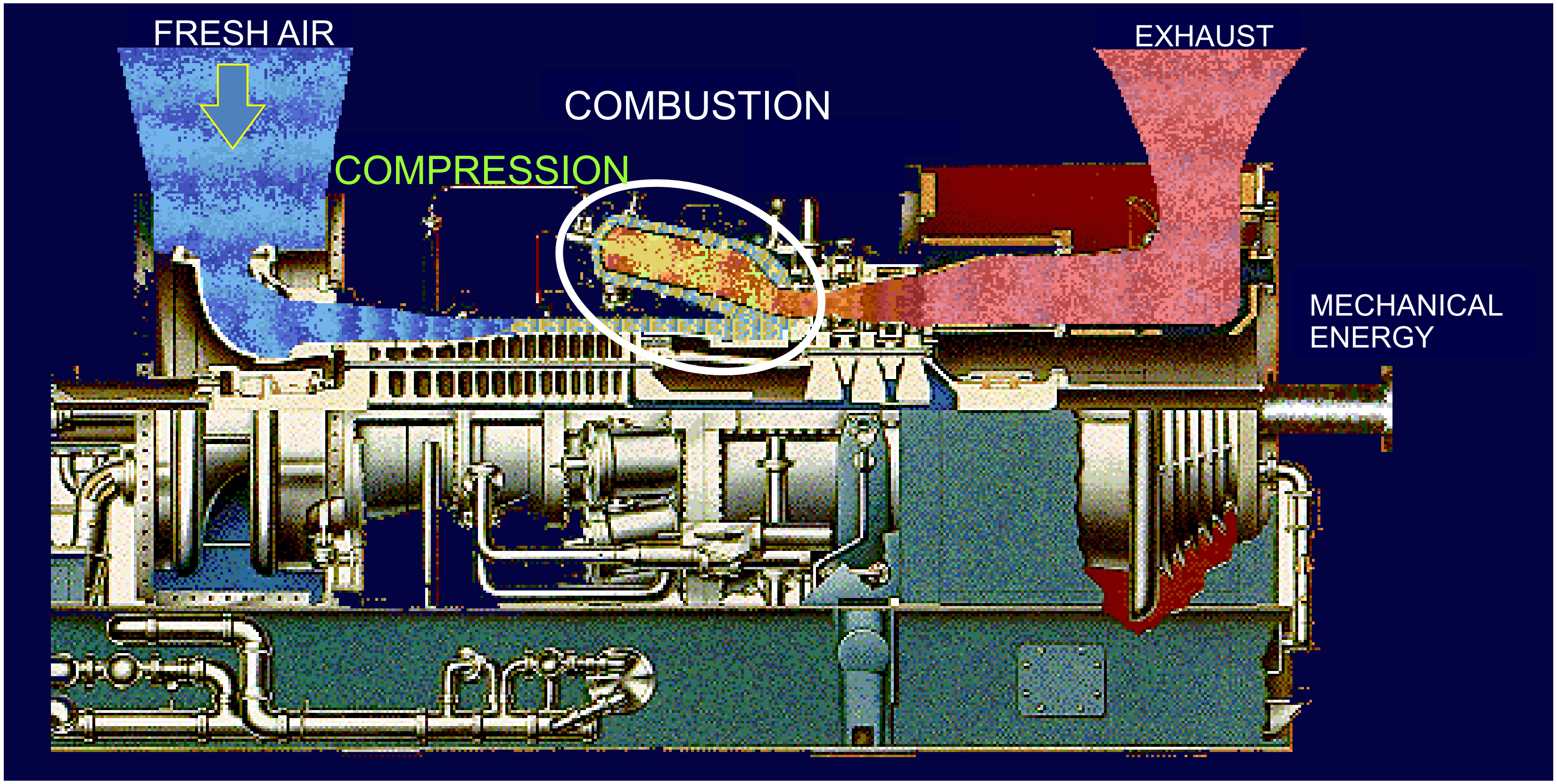
A **compressor** that takes in clean outside air and then compresses it through a series of rotating and stationary compressor blades



Air & Hot Gas Paths

Gas Turbine has 3 main sections:

A **<u>combustion section</u>** where fuel is added to the pressurized air and ignited. The hot pressurized combustion gas expands and moves at high velocity into the turbine section.



Air & Hot Gas Paths

Gas Turbine has 3 main sections:

A **turbine** that converts the energy from the hot/high velocity gas flowing from the combustion chamber into useful rotational power through expansion over a series of turbine rotor blades