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
Review of Thermodynamic Principals

1\textsuperscript{st} Law of Thermodynamics – Energy is conserved

\begin{itemize}
  \item (Change in system’s energy) = (Rate of heat added) – (Rate of work performed)
  \[ \Delta \hat{E} = Q - W \]
\end{itemize}

\begin{itemize}
  \item Major energy contributions
    \begin{itemize}
      \item Kinetic energy – related to velocity of system
      \item Potential energy – related to position in a “field” (e.g., gravity)
      \item Internal energy – related to system’s temperature
        \begin{itemize}
          \item Internal energy, \( U \), convenient for systems at constant volume & batch systems
            \[ \hat{E} = \hat{U} + \frac{u^2}{2g_c} + \frac{g}{g_c} h \]
          \item Enthalpy, \( H = U + PV \), convenient for systems at constant pressure & flowing systems
            \[ \hat{E} = \hat{H} + \frac{u^2}{2g_c} + \frac{g}{g_c} h \]
        \end{itemize}
    \end{itemize}
\end{itemize}
2\textsuperscript{nd} 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

<table>
<thead>
<tr>
<th>Type</th>
<th>Condition</th>
<th>Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isothermal</td>
<td>constant temperature</td>
<td>$\Delta T = 0$</td>
</tr>
<tr>
<td>Isobaric</td>
<td>constant pressure</td>
<td>$\Delta P = 0$</td>
</tr>
<tr>
<td>Isochoric</td>
<td>constant volume</td>
<td>$\Delta V = 0$</td>
</tr>
<tr>
<td>Isenthalpic</td>
<td>constant enthalpy</td>
<td>$\Delta H = 0$</td>
</tr>
<tr>
<td>Adiabatic</td>
<td>no heat transferred</td>
<td>$Q = 0$</td>
</tr>
<tr>
<td>Isentropic</td>
<td>no increase in entropy</td>
<td>$\Delta S = 0$</td>
</tr>
<tr>
<td>(ideal reversible)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
1st Law for steady state flow

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

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

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

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

$$= \int_{p_1}^{p_2} V \, dP + \frac{\Delta u^2}{2g_c} + \frac{g}{g_c} \Delta z$$

$$\hat{W}_s \approx \int_{p_1}^{p_2} V \, dP = \int_{p_1}^{p_2} \frac{dP}{\rho}$$

- 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 $\Delta H$

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

- 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)

$$W_s = \int_{\rho_1}^{\rho_2} \frac{dP}{\rho} = \frac{1}{\rho} \int_{\rho_1}^{\rho_2} dP = \frac{P_2 - P_1}{\rho}$$
Liquid vs. Vapor/Gas Compression

Can compress liquids with little temperature change

\[ \Delta H \text{ for gas compression much larger than for liquid pumping} \]

*GPSA Data Book, 13th ed.*
Mechanical Energy Balance

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

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

- 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 straightforward – pumps

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

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

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

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

\[ = \dot{V} (\Delta P) + (\rho \dot{V}) (g \Delta z) + (\rho \dot{V}) (g \hat{h}_f) + \frac{1}{2} (\rho \dot{V}) (\Delta u^2) \]

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

\[ W_{hhp} = \dot{V} (\Delta P) \text{ or } W_{hhp} = \dot{V} \rho g H \]

Brake horsepower – power delivered to the pump itself

\[ W_{bhp} = \frac{W_{hhp}}{\eta_{pump}} \]
Pump equations for specific U.S. customary units

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

\[
W_hhp [hp] \Rightarrow \left( \frac{\text{gal}}{\text{min}} \right) \left[ \frac{\text{lb}_f}{\text{in}^2} \right] \left[ \frac{231\text{in}^3}{\text{gal}} \right] \left[ \frac{60\text{sec}}{\text{min}} \right] \left[ \frac{12\text{in}}{\text{ft}} \right] \left[ \frac{550\text{ft} \cdot \text{lb}_f / \text{sec}}{\text{hp}} \right]
\]

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

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

\[
W_hhp [hp] \Rightarrow \left( \frac{\text{gal}}{\text{min}} \right) \left[ \text{ft} \right] \left[ \gamma_o \right] \left[ \frac{8.33719\text{lb}_m}{\text{gal}} \right] \left[ \frac{32.174\text{ft}}{\text{sec}^2} \right] \left[ \frac{\text{lb}_m \cdot \text{ft}}{\text{lb}_f \cdot \text{sec}^2} \right]
\]

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

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

Updated: January 4, 2019
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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.*
Pump Example

Pressure drop from Reflux Drum to Pump inlet:

\[
\Delta P = -\frac{g}{g_c} \rho \Delta z + (\Delta P)_{\text{piping}}
\]

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

\[
= 3.5 \text{ psi}
\]

\[\therefore P_{\text{inlet}} = 203.5 \text{ psia}\]

Pressure drop from Pump outlet to Depropanizer:

\[
\Delta P = -\frac{g}{g_c} \rho \Delta z + (\Delta P)_{\text{piping}}
\]

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

\[
= -44.7 \text{ psi}
\]

\[\therefore P_{\text{outlet}} = 264.7 \text{ psia}\]
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:

\[
h_{\text{pump}} = \frac{g_c (\Delta P)_{\text{pump}}}{g \rho} = \frac{g_c (\Delta P)_{\text{pump}}}{g (\gamma_o \rho_{\text{water}})}
= -\frac{144}{(0.485)(62.3665)}(61.2) = 291 \text{ ft}
\]

Hydraulic power:

\[
W_{\text{hhp}} = \frac{(360 \text{ gpm})(61.2 \text{ psi})}{1714} = 12.85 \text{ hp} \quad \text{OR} \quad W_{\text{hhp}} = \frac{(360 \text{ gpm})(0.485)(291 \text{ ft})}{3958} = 12.84 \text{ hp}
\]
Gas Compression: PH Diagrams

Ref: GPSA Data Book, 13th ed.
Updated: January 4, 2019
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Thermodynamics of Ideal Gas Compression

Choices of path for calculating work:

- **Isothermal** ($\Delta T = 0$)

  \[
  \tilde{W}_s = \int_{p_1}^{p_2} \tilde{V} \, dp = RT \int_{p_1}^{p_2} \frac{dp}{p} = RT \ln \left( \frac{p_2}{p_1} \right)
  \]

  - Minimum work required but unrealistic

- **Adiabatic & Isentropic** ($\Delta S = 0$)

  - 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 ($PV^\gamma = \text{constant}$) where $\gamma = \frac{C_p}{C_v}$

- **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**

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

Can also determine discharge temperature

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

Calculation of $\gamma$ 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 Calculation:
Ideal Gas Laws

For methane:

- \( \gamma = 1.3 \)
- \( M = 16 \)
- \( T_1 = 120^\circ F = 580^\circ R \)
- \( P_1 = 300 \text{ psia} \)
- \( P_2 = 900 \text{ psia} \)
- \( R = 1.986 \text{ Btu/lb.mol } ^\circ R \)

\[
W_s = \frac{\gamma R T_1}{M(\gamma - 1)} \left[ \left( \frac{P_2}{P_1} \right)^{(\gamma - 1)/\gamma} - 1 \right] = \frac{(1.3)(1.986)(580)}{(16)(1.3 - 1)} \left[ \left( \frac{900}{300} \right)^{(1.3 - 1)/1.3} - 1 \right] = 90 \text{ Btu/lb}
\]

\[
T_2 = (580) \left( \frac{900}{300} \right)^{(1.3 - 1)/1.3} = 747^\circ R \Rightarrow 287^\circ F
\]
Compress methane from 300 psia & 120°F to 900 psia

Enthalpy Changes:
\[ W = \Delta H = 525 - 440 = 85 \text{ BTU/lb} \]

Outlet temperature: 280°F
## Example Calculation: Using a Simulator

<table>
<thead>
<tr>
<th></th>
<th>Work Btu/lb</th>
<th>Outlet °F</th>
</tr>
</thead>
<tbody>
<tr>
<td>HYSYS Peng-Robinson</td>
<td>86.58</td>
<td>281.8</td>
</tr>
<tr>
<td>HYSYS Peng-Robinson &amp; Lee-Kesler</td>
<td>87.34</td>
<td>280.5</td>
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<td>HYSYS SRK</td>
<td>87.71</td>
<td>281.2</td>
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<tr>
<td>HYSYS BWRS</td>
<td>87.04</td>
<td>281.1</td>
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<tr>
<td>HYSYS Lee-Kesler-Plocker</td>
<td>87.40</td>
<td>280.6</td>
</tr>
<tr>
<td>Aspen Plus PENG-ROB</td>
<td>86.61</td>
<td>282.4</td>
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<td>Aspen Plus SRK</td>
<td>87.82</td>
<td>281.8</td>
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<tr>
<td>Aspen Plus BWRS</td>
<td>87.10</td>
<td>281.6</td>
</tr>
</tbody>
</table>
Multi-Stage Gas Compression

If customer wants 1,000 psig (with 100 psig & 120°F inlet)...

- Then pressure ratio of (1015/115) = 8.8
- Discharge temperature for this ratio is ~500°F

For reciprocating compressors the GPSA *Engineering Data Book* recommends

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

To obtain higher pressure ratios higher must use *multistage compression with interstage cooling*
Multistaging

To minimize work need good interstage cooling and equal pressure ratios

The number of stages is calculated using

\[ R_p = \left( \frac{P_2}{P_1} \right)^{1/m} \quad \Rightarrow \quad m = \frac{\ln(P_2 / P_1)}{\ln(R_p)} \]

To go from 100 to 1000 psig with a single-stage pressure ratio of 3 takes 2 (1.98) stages & the stage exit temp \( \sim 183^\circ F \) (starting @ 120°F)

\[ m = \frac{\ln(1015)}{\ln(3)} = \frac{\ln(8.8)}{\ln(3)} = 1.98 \quad \Rightarrow \quad m = 2 \]

\[ T_2 = T_1 \left[ \left( \frac{P_2}{P_1} \right)^{1/m} \right]^{(\gamma-1)/\gamma} = (120 + 460) \left[ \left( \frac{1015}{115} \right)^{1/2} \right]^{(1.3-1)/1.3} = (580)[2.97]^{0.2308} = 746^\circ R \quad \Rightarrow \quad 286^\circ F \]
Multistaging

Work for a single stage of compression

\[ \hat{W}_s = \frac{\gamma R T_1}{M(\gamma - 1)} \left[ \left( \frac{P_2}{P_1} \right)^{(\gamma - 1)/\gamma} - 1 \right] = \frac{(1.3)(1.986)(580)}{(16)(1.3 - 1)} \left[ \left( \frac{1015}{115} \right)^{1.3 - 1/1.3} - 1 \right] = 204 \text{ Btu/lb} \]

Work for two stages of compression (interstage cooling to 120°F)

- Intermediate pressure

\[ P_{\text{int}} = \sqrt{\frac{P_2 P_1}{1.3 - 1}} = \sqrt{(1015)(115)} = 341.7 \text{ psia} \]

- Total work – 13% less work

\[ \hat{W}_s = \frac{(1.3)(1.986)(580)}{(16)(1.3 - 1)} \left\{ \left[ \left( \frac{341.7}{115} \right)^{1.3 - 1/1.3} - 1 \right] + \left[ \left( \frac{1015}{341.7} \right)^{1.3 - 1/1.3} - 1 \right] \right\} \]

\[ = 178 \text{ Btu/lb} \]
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: \( (\Delta T)_{act} = \frac{(\Delta T)_{\Delta s=0}}{\eta_{IS}} = T_1 \left[ \left( \frac{P_2}{P_1} \right)^{(\gamma - 1)/\gamma} - 1 \right] \)

\[ \therefore T_{2,act} = T_1 + (\Delta T)_{act} = T_1 \left[ 1 + \frac{\left( \frac{P_2}{P_1} \right)^{(\gamma - 1)/\gamma} - 1}{\eta_{IS}} \right] \]
Example Calculation: 75% isentropic efficiency

<table>
<thead>
<tr>
<th></th>
<th>$\Delta S = 0$</th>
<th>$\Delta S = 0$</th>
<th>$\eta = 0.75$</th>
<th>$\eta = 0.75$</th>
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</thead>
<tbody>
<tr>
<td></td>
<td>Work Btu/lb</td>
<td>Outlet °F</td>
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<td>Outlet °F</td>
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<tr>
<td>Ideal Gas Equations</td>
<td>90</td>
<td>287</td>
<td>120</td>
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<td>PH diagram</td>
<td>85</td>
<td>280</td>
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<td>280.6</td>
<td>116.5</td>
<td>324.8</td>
</tr>
</tbody>
</table>
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, $\eta_p$, 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

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

\[
\hat{W}_{act} = \frac{\hat{W}_p}{\eta_p} = \frac{\hat{W}_{\Delta S=0}}{\eta_{IS}}
\]
Why Use Polytropic Equations?

Polytropic equations give consistent P-T pathway between the initial & discharge conditions.
Effect of Internal Pressure Drops

Pressure drops at the inlet & outlet give an apparent inefficiency of compression

- Assume isenthalpic pressure drops before & after the actual compression
## Effect of Internal Pressure Drops

<table>
<thead>
<tr>
<th>Pressure Reductions</th>
<th>Before Compression psia</th>
<th>After Compression psia</th>
<th>Work Btu/lb</th>
<th>Outlet °F</th>
<th>Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>0%</td>
<td>300</td>
<td>900</td>
<td>86.58</td>
<td>281.8</td>
<td>100%</td>
</tr>
<tr>
<td>1%</td>
<td>297</td>
<td>909</td>
<td>88.36</td>
<td>284.7</td>
<td>98%</td>
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<tr>
<td>5%</td>
<td>285</td>
<td>947</td>
<td>95.76</td>
<td>296.8</td>
<td>90%</td>
</tr>
<tr>
<td>10%</td>
<td>270</td>
<td>1000</td>
<td>105.7</td>
<td>313.0</td>
<td>82%</td>
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<tr>
<td>15%</td>
<td>255</td>
<td>1059</td>
<td>116.5</td>
<td>330.2</td>
<td>74%</td>
</tr>
</tbody>
</table>
Compression vs. Expansion Efficiency

Work to compressor is greater than what is needed in the ideal case

- Work to the fluid
  \[ \eta_{IS} = \frac{(\Delta H)_{\Delta S=0}^{\text{fluid}}}{(\Delta H)_{\Delta S=0}^{\text{IS}}} \implies W_{\text{fluid}} = \frac{W_{\Delta S=0}^{\text{IS}}}{\eta_{IS}} \]

- Total work to the device
  \[ \eta_{\text{mech}} = \frac{W_{\text{fluid}}}{W_{\text{total}}} \implies W_{\text{total}} = \frac{W_{\Delta S=0}^{\text{IS}}}{\eta_{\text{mech}} \eta_{IS}} \]

Work from expander is less than what can be obtained in the ideal case

- Work from the fluid
  \[ \eta_{IS} = \frac{(\Delta H)_{\Delta S=0}^{\text{fluid}}}{(\Delta H)_{\Delta S=0}^{\text{IS}}} \implies W_{\text{fluid}} = \eta_{IS} (W_{\Delta S=0}^{\text{IS}}) \]

- Total work from the device
  \[ \eta_{\text{mech}} = \frac{W_{\text{total}}}{W_{\text{fluid}}} \implies W_{\text{total}} = \eta_{\text{mech}} W_{\text{fluid}} = \eta_{\text{mech}} \eta_{IS} W_{\Delta S=0} \]
Equipment:
Pumps, Compressors, Turbines/Expanders
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

Updated: January 4, 2019
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Centrifugal Pump Performance Curves

**FIGURE 2.10** Effect of speed on head and efficiency.

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

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

Positive displacement – compress by changing volume
- Reciprocating
- Rotary screw
- Diaphragm
- Rotary vane

Dynamic – compress by converting kinetic energy into pressure
- Centrifugal
- Axial
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

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Reciprocating Compressors - Main Components

- Pulsation Bottles
- Crankcase (Cast Iron)
- Crankshaft (Forged Steel)
- Counterweight (for balancing)
- Crosshead (Cast Steel)
- Ballast for balancing of inertia forces
- Slide Body
- Distance Pieces
- Pneumatic Valve
- Unloaders (for capacity control)
- Main Oil Pump
- Connecting Rod (die forged steel)
- Oil Wiper Packing
- Piston Rod
- Rod Packing
- Piston
- Cylinder Valve
- Forged Cylinder
- Cast Cylinder
Rotary Screw Compressor

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

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)  Multi-stage

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

Siemens

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

Courtesy of Nuovo Pignone Spa, Italy
Centrifugal 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
Turbines & Turboexpanders

Utilize pressure energy to...
- Reduce temperature of gas & (possibly) generate liquids
- Perform work & provide shaft power to coupled equipment

Similar principles to a centrifugal compressor except in reverse

Most common applications:
- Turboexpander: NGL recovery
- Gas turbine: power to drive pumps, compressors, generators, ...
Turboexpander Cutaway

https://www.turbomachinereviewmag.com/expander-compressors-an-introduction/

Video (LA Turbine. 2:54 minutes): https://www.youtube.com/watch?v=f5Gz--NBM
Turboexpanders

FIG. 13-75
Typical Expander/Compressor Cross-Section with Thrust Balancing Schematic

GPSA Engineering Data Book, 14th ed.
Methane Expansion – Isentropic vs. Isenthalpic
Gas Turbine Engine

https://aceclearwater.com/product/case-study-ge-power-ducts/
Gas Turbine Coupled to Centrifugal Compressor

Axial Compressor

Exhaust Gas

Centrifugal Compressor

Low Pressure Gas

Combustion Turbine

Air

Fuel Gas

High Pressure Gas
# Industrial Gas Turbines

**FIG. 15-32**

2011 Basic Specifications — Gas Turbine Engines (Mechanical Drive)

<table>
<thead>
<tr>
<th>Model</th>
<th>Power Rating (ISO Rating) hp</th>
<th>Heat Rate (LVH) Btu/hp-hr</th>
<th>Pressure Ratio</th>
<th>Power Shaft RPM</th>
<th>At ISO RATING CONDITIONS</th>
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<tbody>
<tr>
<td></td>
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<td></td>
<td></td>
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<td>Turbine Inlet Temp. °F</td>
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</table>

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

\[
\text{Thermal efficiency} = \frac{2544}{\text{Heat rate, Btu(LHV)/hp\cdot hr}}
\]

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

\[
\text{Thermal efficiency} = \frac{2544}{6816} = 0.3732
\]

Includes effects of adiabatic & mechanical efficiencies
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

\[ w_S = -\Delta h = -C_p\Delta T \quad (9.1 \text{ and } 1.18) \]

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:

\[ w_{\text{net}} = w_t - w_c = C_p(T_3 - T_4) - (T_2 - T_1) \]

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

Since \(w_{\text{net}}\) is positive work flows to the load.
GT - Principle of Operation

Simple Cycle Gas Turbine

1. **Air**
2. Axial Compressor
3. Combustion Chamber
4. Centrifugal Compressor
5. Exhaust Gas (~950°F)
6. H.P./L.P. Turbine
7. Fuel Gas (~650 - 950°F)
8. ~1800 - 2300°F
9. Gas  (~950°F)

Ideal Cycle Efficiency

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

Theoretical Cycle

Real Cycle

Temperature (°F)

Entropy

\[ \eta = 1 - \frac{T_4 - T_1}{T_3 - T_2} \]
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
Summary
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
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 **combustion section** 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.