Natural gas compression performance calculations

Constantinos Hadjistassou, PhD
Associate professor
Programme in Oil & Gas Engineering
University of Nicosia
Marine & Carbon Lab: www.carbonlab.eu

Oct./Nov., 2020

UNIVERSITY *of* NICOSIA

Overview

- Two topics:
  - Gas hydrates
  - Sub-ambient flow in pipelines

- Principles of gas compression

www.facebook.com/ogeunic

Inpex Ichthys project's first LNG

- 23/10/2018: first LNG shipment to Japan
- Cost: $40bn; Inpex: 62%, Total: 30%
- Capacity: 8.9mtpa or 10% of Japanese imports
- Video: https://www.youtube.com/watch?v=EROYEZLT_60

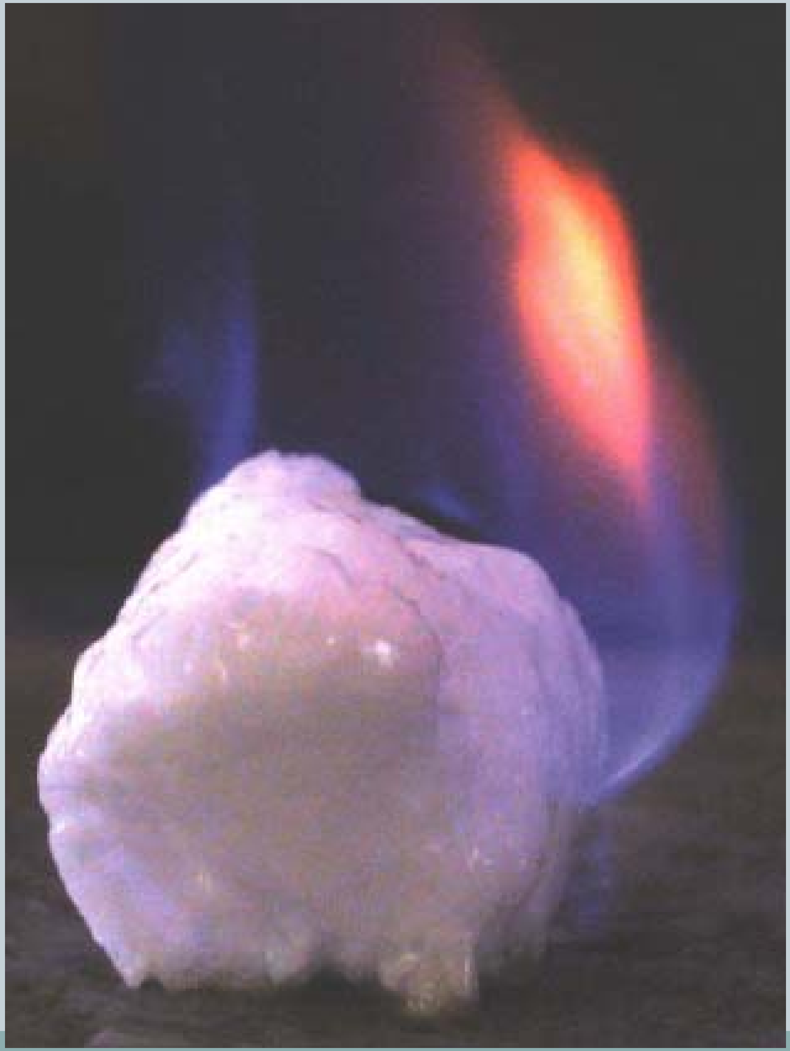
Two topics

Two topics

- Gas hydrates
- Sub-ambient flow in pipelines

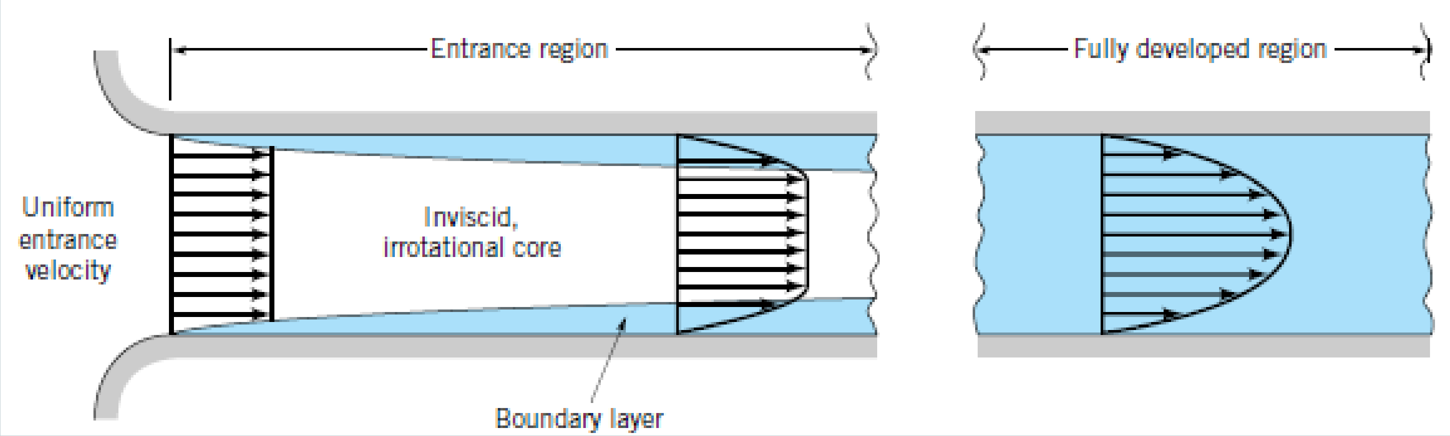
Gas hydrates

- Besides posing problems for flow assurance gas hydrates in the future could be a source of NG fuel
- Several research programs target their economic extraction



Sub-ambient pipeline flow

- Is it possible for a pipe to transmit fluids at sub-ambient pressure ie below atmospheric pressure?



- Pressure drop (Δp):

$$\Delta p = p_1 - p_2 = \frac{128 \mu \ell Q}{\pi D^4}$$

- A pressure differential drives flow. Therefore, $p_2 < p_1$

Major pump needs in an LNG facility

- Amine circulation
- Scrub column & fractionation towers
- LNG product pumps & loading pumps
- Seawater pumps (if seawater cooled)
- Hot oil pumps
- 2-4 pumps per LNG tank

Gas compression performance

Definitions

- Pressure (P) is the direct force per unit area normal to the surface

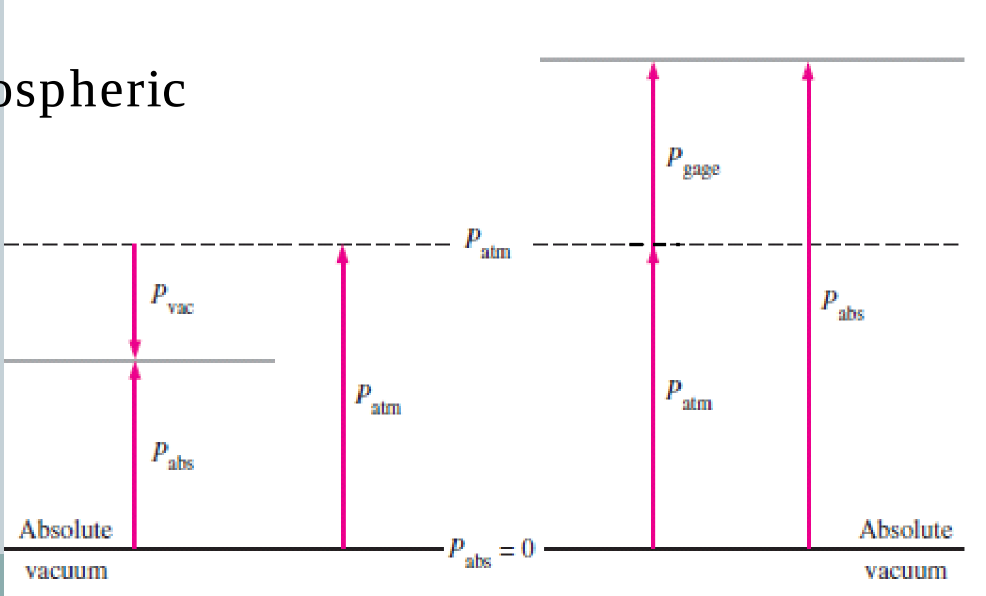
$$P = \frac{F}{A}$$

- Absolute pressure (P_{abs})is measured relative to absolute vacuum (absolute zero pressure)
- Gage pressure (P_G) is the difference between absolute & atmospheric pressure
- Vacuum pressure is below atmospheric

$$P_{\text{gage}} = P_{\text{abs}} - P_{\text{atm}}$$

$$P_{\text{vac}} = P_{\text{atm}} - P_{\text{abs}}$$

- An *ideal gas* is an imaginary substance which obeys PV=RT



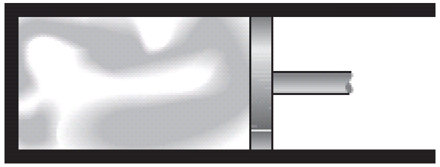
More definitions

- Incompressible substance is the one whose specific volume (or density) remains constant when subject to an increase in pressure.

- A *reversible process* can be reversed w/o leaving any trace on the surroundings. Net heat & net work exchange btw stm&surroundings =0

- Enthalpy is the *thermodynamic potential* of a system denoted by *h* (units: *J* or *BTU*):

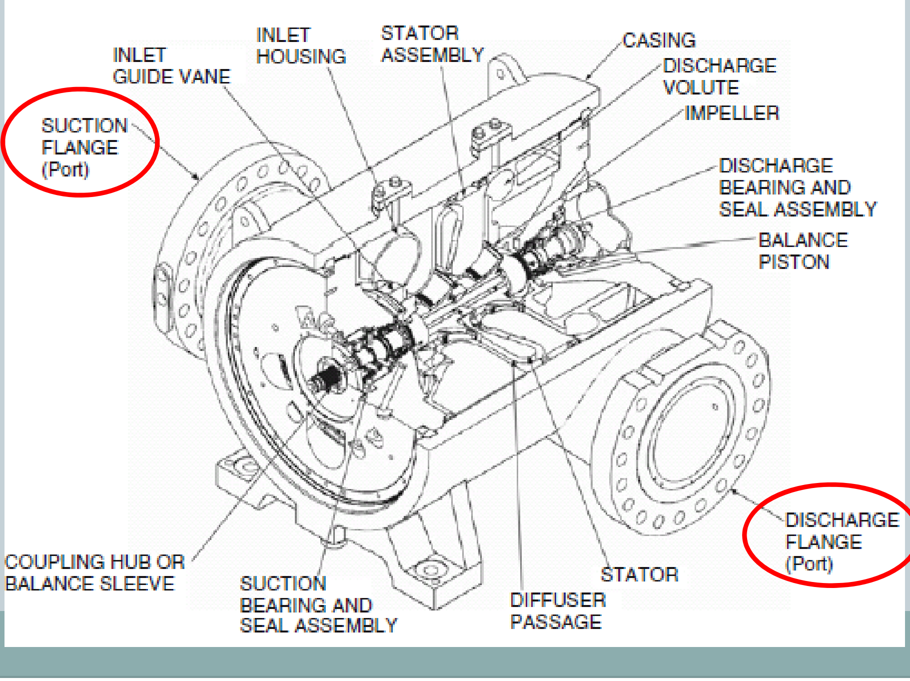
$$h = u + Pv$$



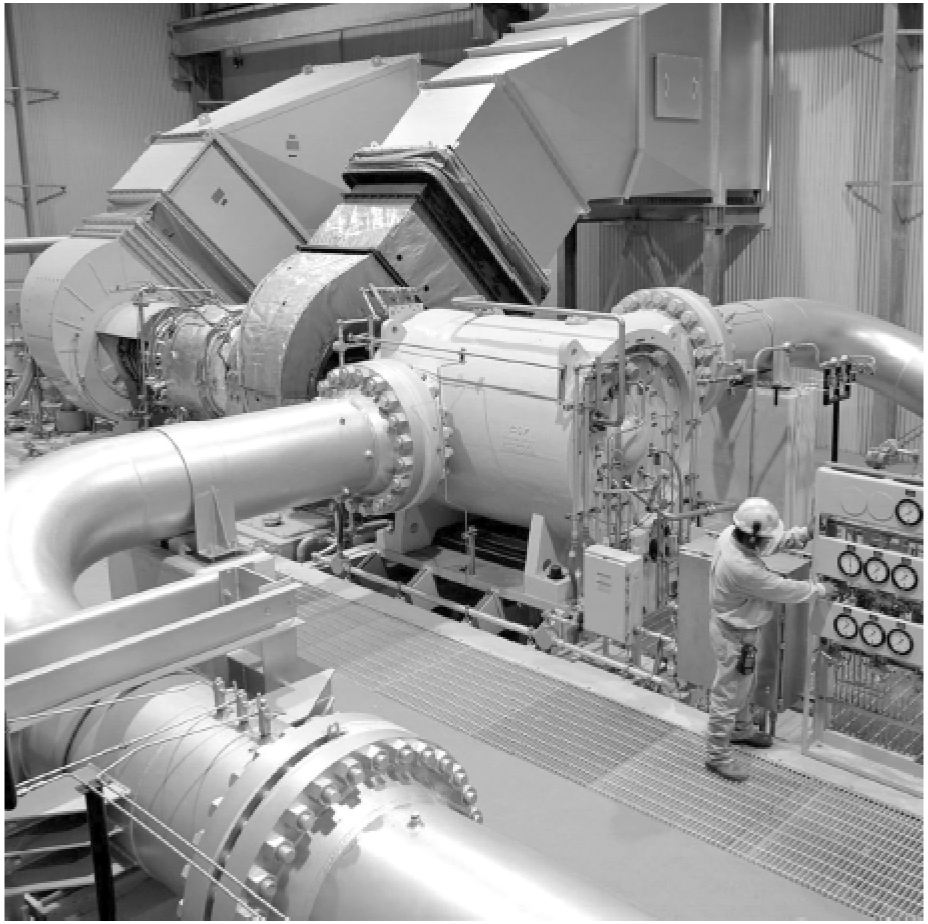
(*b*) Quasi-equilibrium expansion and compression of a gas

Compression performance

- Interested in calculating:
  - – Compressor head (inlet & outlet pressure)
  - – Discharge temperature
  - – Compressor efficiency
  - – Sizing of compressor (hp)



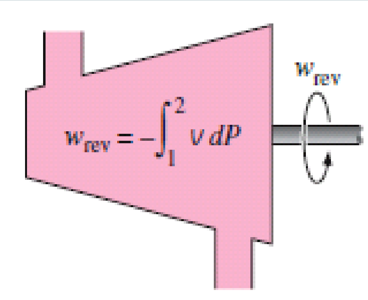
Gas-turbine driven centrifugal compressor



Reversible isothermal gas compression

- Purpose of compression: mechanical work raises gas suction pressure to higher discharge pressure
- *Reversible* shaft work of a compressor, w_s, for open process for P_1 to P_2

$$w_S = -\Delta h = -\int_{P_1}^{P_2} v dP, \qquad (1)$$



(a) Steady-flow system

where: h = enthalpy, $h = u + pv$

v = specific volume (m^3/kg or ft^3/lb_m)

–ve sign indicates that work is being done on the system

- Since $V = RT/P$, for an *isothermal* process the reversible shaft work of an *ideal gas* is:

$$w_S = -RT\ ln(P_2/P_1)/MW, \qquad (2)$$

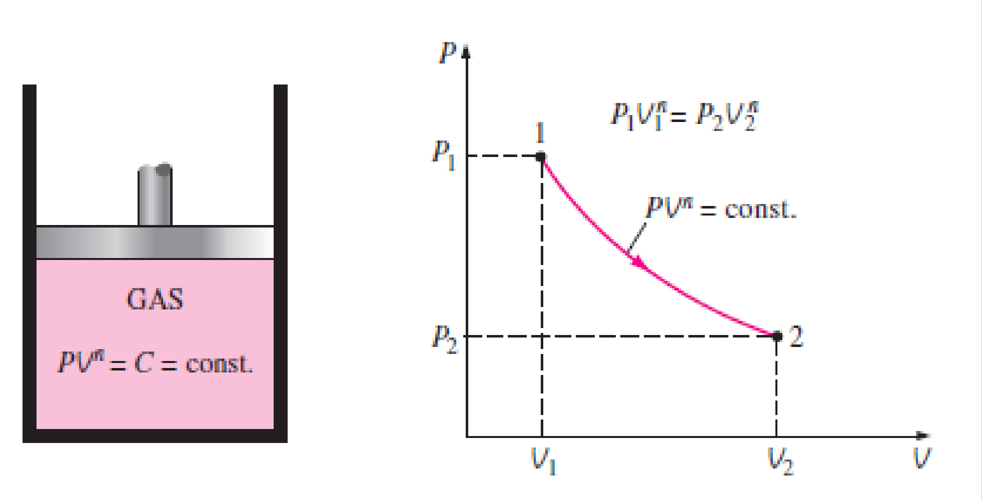
where: R = gas constant ($Nm/kg{\cdot}K$ or $Btu/lbmol{\cdot}R$), MW = molec. weight

Reversible adiabatic heat transfer

- Heat transfer from compression into fluid
- Thermal energy absorbed by the gas phase
- *Polytropic process*:

$$Pv^{\gamma} = C \qquad (3)$$

- γ is the *polytropic exponent; C* = constant



Work done on compressor

- Adiabatic work done on compressor:

$$w_S = \frac{\gamma R T_1}{MW(\gamma - 1)}\left[1 - \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}\right] \qquad (4)$$

- Ratio (P_2/P_1) is the *compression ratio*
- T_1: inlet temperature. The *specific heat ratio*, γ, is:

$$\gamma = \frac{C_P}{C_V} = \left[\frac{C_P}{C_P - R}\right] \qquad (5)$$

Since $R = C_P - C_V$

Where C_P is specific heat @ constant pressure &

C_V the specific heat @ constant volume [of the gas]

Outlet compressor temperature

- For a *reversible adiabatic* compression of an *ideal gas*, outlet temp:

$$T_2 = T_1(P_2/P_1)^{(\gamma-1)/\gamma}. \qquad (6)$$

- Empirically, T_2 can be obtained from:

$$T_2 = T_1\left(1 + \left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right]\right) / \eta_{IS}, \qquad (7)$$

- η_{IS} is the entropic efficiency with heat losses. Thus:

$$Pv^{\kappa} = C \qquad (8)$$

κ is the polytropic constant (obtained empirically)

Outlet compressor temperature

- κ replaces γ in eqn (7):

$$w_S = \frac{\kappa R T_1}{MW(\kappa - 1)}\left[1 - \left(\frac{P_2}{P_1}\right)^{(\kappa-1)/\kappa}\right]. \qquad (9)$$

- If κ is not known, γ can be used.
- Assumption: all gas components behave like an ideal gas.
- Using eqns (1)-(9) we can determine:
  - Isothermal work to compress a gas from pressure (P_1) into (P_2)
  - Adiabatic work to compress a gas from pressure (P_1) into (P_2)
  - Discharge temp. for adiabatic compression

- Example#2

Multi-stage compression

- Multi-staging: final pressure is attained by more than one stages/steps
- Reasons for multi-staging:
  - Cool the gas btw stages thereby increasing efficiency by decreasing gas volume & work.
  - Material limitations. 150°C limit for construction materials, seals, lubricants.
  - Recommended temp. decreases with high pressure
- Limiting temps define compression ratios to btw 3:1 to 5:1
- Min work when each stage does same amount of work (~pressure ratio/stage)
- Pressure ratio (PR):

$$PR = (P_2/P_1)^{1/m} \qquad (10)$$

- m = number of stages
- Example#3

Compressor efficiencies

- *Reversible adiabatic* (or isentropic efficiency), η_{IS}:

$$\eta_{IS} = W_S / W_{S,\,ACTUAL} \qquad (11)$$

- W_s is determined from eqn (4)
- Centrifugal compressors' efficiency given by *polytropic efficiency*, η_P:

$$\eta_P = [(\gamma - 1)/\gamma]/[(\kappa - 1)/\kappa]. \qquad (12)$$

- Polytropic efficiency > adiabatic efficiency.
- Discharge conditions can also be determined from:

$$T_2 = T_1(P_2/P_1)^{(1/\eta_P)(\gamma-1)/\gamma} \qquad (13)$$

- η_P is the polytropic efficiency
- Example#4

Capacity & power calculations

- Compressibility factor, Z, is:

$$Z = \frac{Pv}{RT} \qquad (14)$$

- Z is due to gases deviating from ideal gas law. $Z = V_{actual}/V_{ideal}$

Power Requirements

- For isentropic compression & efficiency:

$$\text{Power} = \dot{m}\ z_{avg}\ w_S, \qquad (15)$$

- $\dot{m}$ is the mass flow rate; z_{avg} is the average of the inlet & outlet compressibility factors

Reciprocating compressors

- Power estimation:

$$\text{Brake HP} = 22\ \text{F PR}\ m\ \text{MMacfd} \qquad (16)$$

- Brake HP is work delivered to compressor , *F, correction factor*:
  - For single stage (m=1), F = 1.0
  - For double stage (m=2), F = 1.08
  - For three-stages (m=3), F = 1.10
- *PR* = pressure ratio; *HP* = horsepower; m = # of stages
- Vendors rate compressors on 14.4 psi [for simplicity] than 14.7 psi
- Equation 16:
  - Developed for large, slow-speed compressors of 300-400 rpm
  - Gases with SG=0.65 & PR>2.5
  - For 0.8<SG<1.0, use 20 in eqn (16), if SG<0.8 use 22

Flow rate

- Volumetric flow rate (Q) given by:

$$Q = scfm\left(\frac{14.7}{P_1(psia)}\right)\left(\frac{T_1(^{\circ}R)}{520}\right)\left(\frac{z_1}{z_R}\right) \qquad (17)$$

where “1” & “*R*” denote inlet & reference conditions. Reference state: 14.7 psia @ 60°F (15.6°C); *scfm* standard cubic feet

- Example#5

Compressor efficiencies

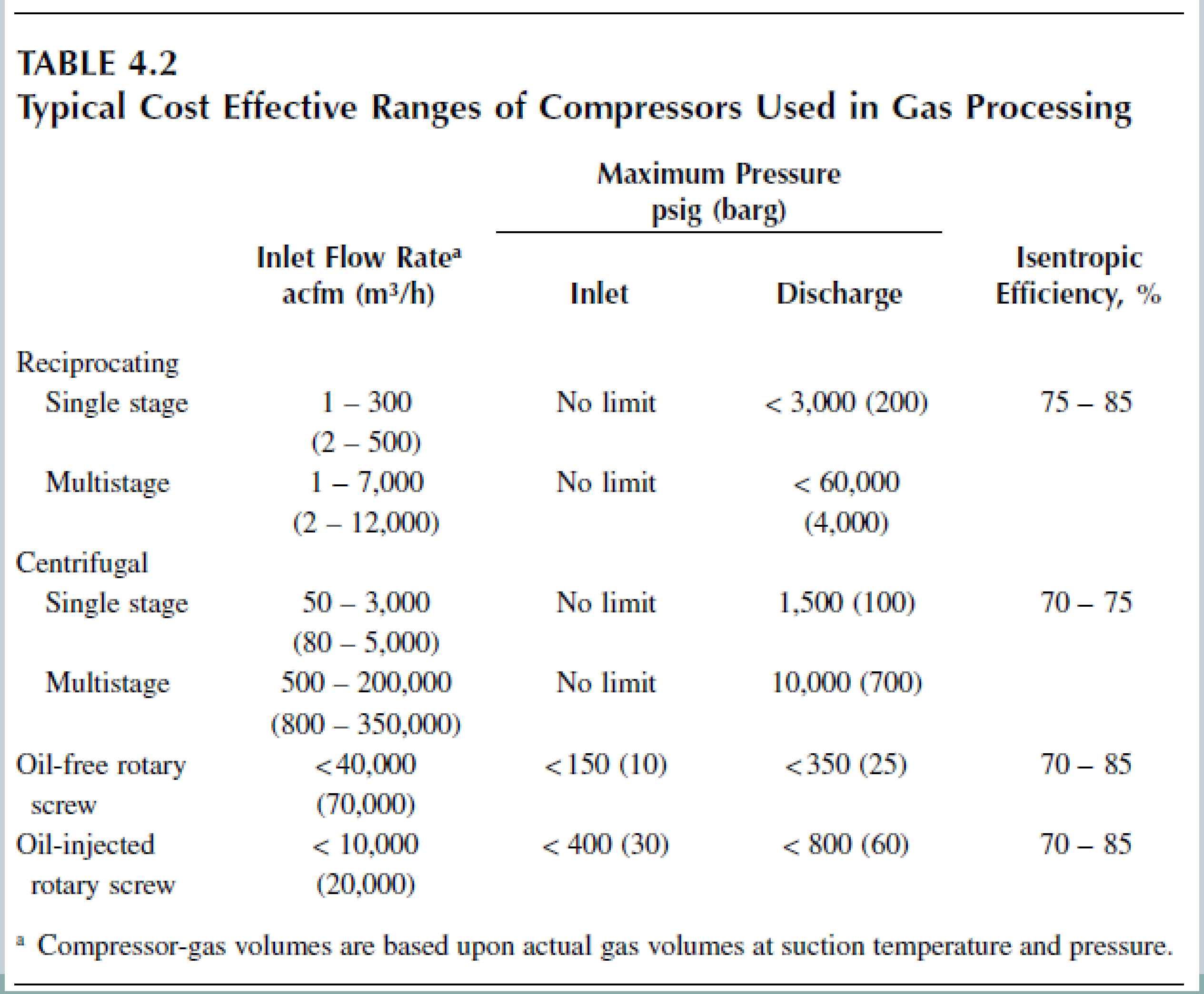


TABLE 4.2
Typical Cost Effective Ranges of Compressors Used in Gas Processing

		Maximum Pressure psig (barg)		
	Inlet Flow Rate[a] acfm (m^3/h)	**Inlet**	**Discharge**	**Isentropic Efficiency, %**
Reciprocating				
Single stage	1 – 300 (2 – 500)	No limit	< 3,000 (200)	75 – 85
Multistage	1 – 7,000 (2 – 12,000)	No limit	< 60,000 (4,000)	
Centrifugal				
Single stage	50 – 3,000 (80 – 5,000)	No limit	1,500 (100)	70 – 75
Multistage	500 – 200,000 (800 – 350,000)	No limit	10,000 (700)	
Oil-free rotary screw	<40,000 (70,000)	<150 (10)	<350 (25)	70 – 85
Oil-injected rotary screw	< 10,000 (20,000)	< 400 (30)	< 800 (60)	70 – 85

[a] Compressor-gas volumes are based upon actual gas volumes at suction temperature and pressure.

Thanks for your attention!