plan

- A (few) more comments to problem 2, exercise set 3
- Compression

- fixes to excel sheet
  - VBA function for ESP Power
  - coefficients for efficiency expression are on sheet ESP Performance_water

there two methods to solve the exercise

Method 1

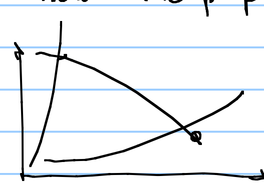
- remove the ESP
- make flow equilibrium at pump
  - estimate available pressure at suction
  - required pressure at discharge
- verify
  - p_{suc}
  - Required power { η_H (VBA) — f (visual interpolat.)
  - falls on map
  - $P_{disc} - P_{suc} > 0$

Method 2

- Include the ESP model
- use wellhead as equilibrium point
  - $P_R \rightarrow P_{wf}$ (IPR)
  - $P_{wf} \rightarrow P_{suc}$ (pipe eq.) (assume f)
  - $P_{suc} \rightarrow P_{disc}$ (ESP equation)
  - $P_{disc} \rightarrow P_{wh}$ (pipe equi)
  - P_{wh} avail.
  - $P_{wh\ required} = 7$ bara

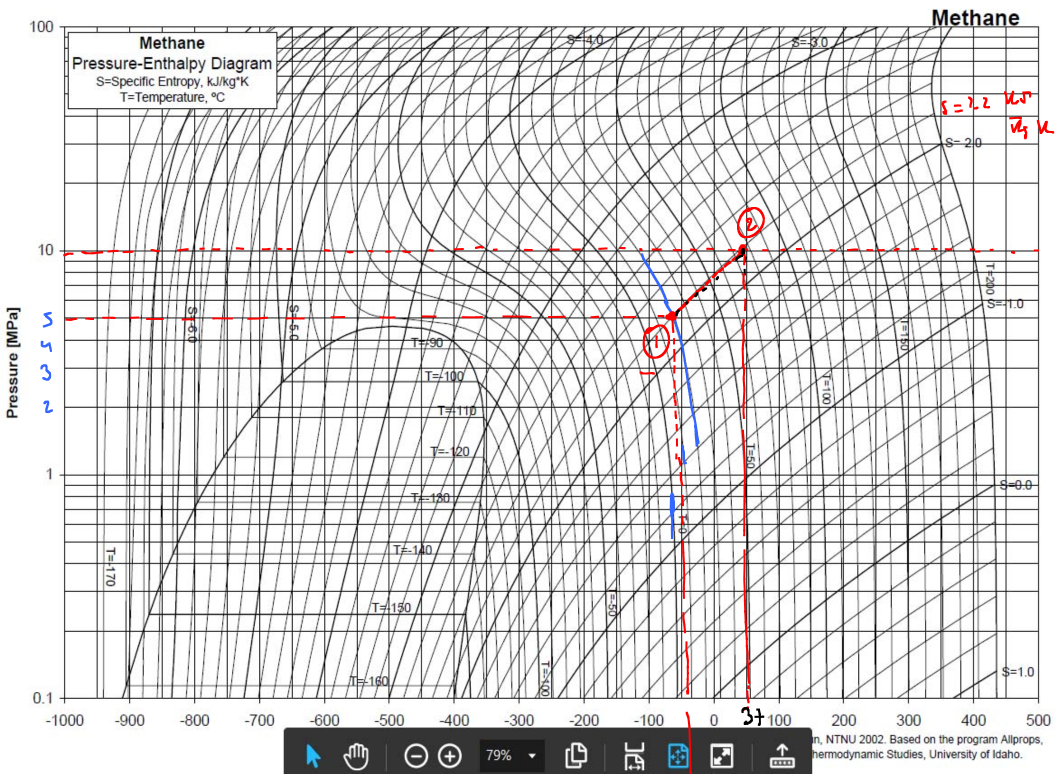
  change $f_{assumed}$ until $P_{wh\,calc} = 7$ bara

- Honor the pump constraints
  - $30 \leq f \leq 60\,hz$
  - $P \leq P_{max}$
  - $P_{suc} \geq P_b \cdot F_s$ (safety factor)
  - $q_{min}(f, \mu) \leq q \leq q_{max}(f, \mu)$



```
Function qmax_visc(N, fref, den, visc, Q_BEP_fref, a5, a4, a3, a2, a1, qmax_fref, f, fref)
Function qmin_visc(N, fref, den, visc, Q_BEP_fref, a5, a4, a3, a2, a1, qmin_fref, f, fref)
```

Compression



P_{suc}, T_{suc} = 50 bara, 20 °C

↓

P_{disc} = 100 bara

T_{dis} = 75 °C

if too high (140 °C)

outlet temperature of the compressor has to be closely monitored → integrity of downstream pipe
→ integrity of compressor seals
→ inhibitor vaporization
- hydrate inhibitor
- scale inhibitor

$\dot{W} = \dot{m}\,(h_2 - h_1)$

$= 78\ \text{kg/s}\ (37 + 67) = 8112\ \frac{kJ}{s} = 8.1\ MW$

current compressor installations subsea:
Åsgard P = 11 MW
gullfaks P = 6 MW

isentropic expansion (P, T)

$$\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} = \left(\frac{T_{2s}}{T_1}\right)$$

↳ absolute units (K, °R)

$k = \frac{C_p(T)}{C_v(T)}$ $k = 1.30 - 0.31\,(\gamma_g - 0.55)$

$$(293.15\ K) \cdot \left(\frac{100}{50}\right)^{\frac{1.3-1}{1.3}} = T_{2s}$$

$T_{2s} = 344\ K = 70\ °C$ if isentropic

the real compression process can be represented as polytropic $p \cdot v^n = \text{const}$ (n: polytropic exponent; v ↳ specific volume) real process $\left(\frac{T_2}{T_1}\right) = r_p^{\frac{n-1}{n}}$

$T_2 > T_{2s}$ $\frac{T_{2s}}{T_1} = r_p^{\frac{n-1}{n}}$ → $\frac{T_2}{T_1} = r_p^{\frac{k-1}{k}}$

$r_p^{\frac{n-1}{n}} > r_p^{\frac{k-1}{k}}$ $n > k$

0 - 1

polytropic efficiency $\eta_p = \frac{k-1}{k} \frac{n}{n-1}$ η

dry gas compressor have a polytropic efficiency (0.6 - 0.8)

assume $\eta_p = 0.7$, $n =$

$\eta_p = f(q_{@\,inlet})$

$\eta_p \frac{k}{k-1} = \frac{n}{n-1}$

$\frac{k-1}{k\eta_p} = 1 - \frac{1}{n}$

$\frac{1}{n} = 1 - \left(\frac{k-1}{k\eta_p}\right)$

$n = \frac{1}{1 - \left(\frac{k-1}{k \cdot \eta_p}\right)}$

$n = 1.5$

$T_2 = T_1 (r_p)^{\frac{0.5}{1.5}} = 293.15\,K \left(\frac{100}{50}\right)^{\frac{0.5}{1.5}} = 96\,°C$

required compression power $P = \frac{\Delta h_p \cdot \dot{m}}{\eta_p \cdot \eta_m}$ [W] ($\dot{m}$: kg/s)

$\Delta h_p = T_{suc} \cdot Z_{av} \cdot R \frac{n}{n-1} \left(r_p^{\frac{n-1}{n}} - 1\right)$ [J/kg]

(R: specific gas; n: polytropic exponent)

T_{suc} [K]

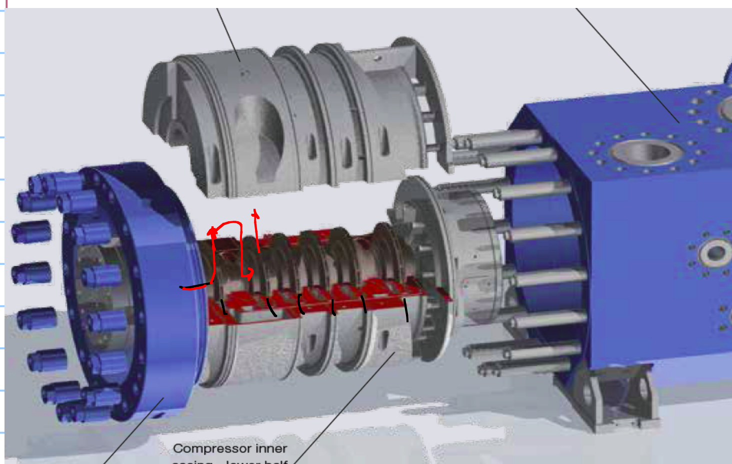
$Z_{av} = \frac{Z_{suc} + Z_{disc}}{2}$

$R = \frac{\bar{R}}{MW_{gas}}$ $\left[\frac{J}{kg\,K}\right]$ ← universal gas constant 8314 $\frac{J}{Kmol\,K}$

T_{suc} for problem 1 excercise set 3 is 70°C

Dry gas compressor is made up of several stages. 1 stage = rotor — give energy to fluid { increasing kinetic energy
stator — convert kinetic energy to pressure

typically (4-8 stages)



Aasgard centrifugal compressor MAN
↳ 6-7 stages

$P_{suc} = 50$ bara
$T_{suc} = 20$ °C

↓
1st stage
↓
P_1
T_1
↓

Usually two assumptions $\eta_{p_{STAGE}} = \eta_{p_{compressor}}$ = 1.5

r_p per stage is constant

$$\frac{P_1}{P_{suc}} \cdot \frac{P_2}{P_1} \cdot \frac{P_3}{P_2} \cdot \frac{P_4}{P_3} \cdot \frac{P_5}{P_4} \cdot \frac{P_{disc}}{P_5} = \left(\frac{P_{disc}}{P_{suc}}\right)$$

$$\left(r_{p_{stage}}\right)^{n_{stages}} = r_{p_{compressor}}$$

$$r_{p_{stage}} = \left(r_{p_{compressor}}\right)^{1/n_{stages}}$$

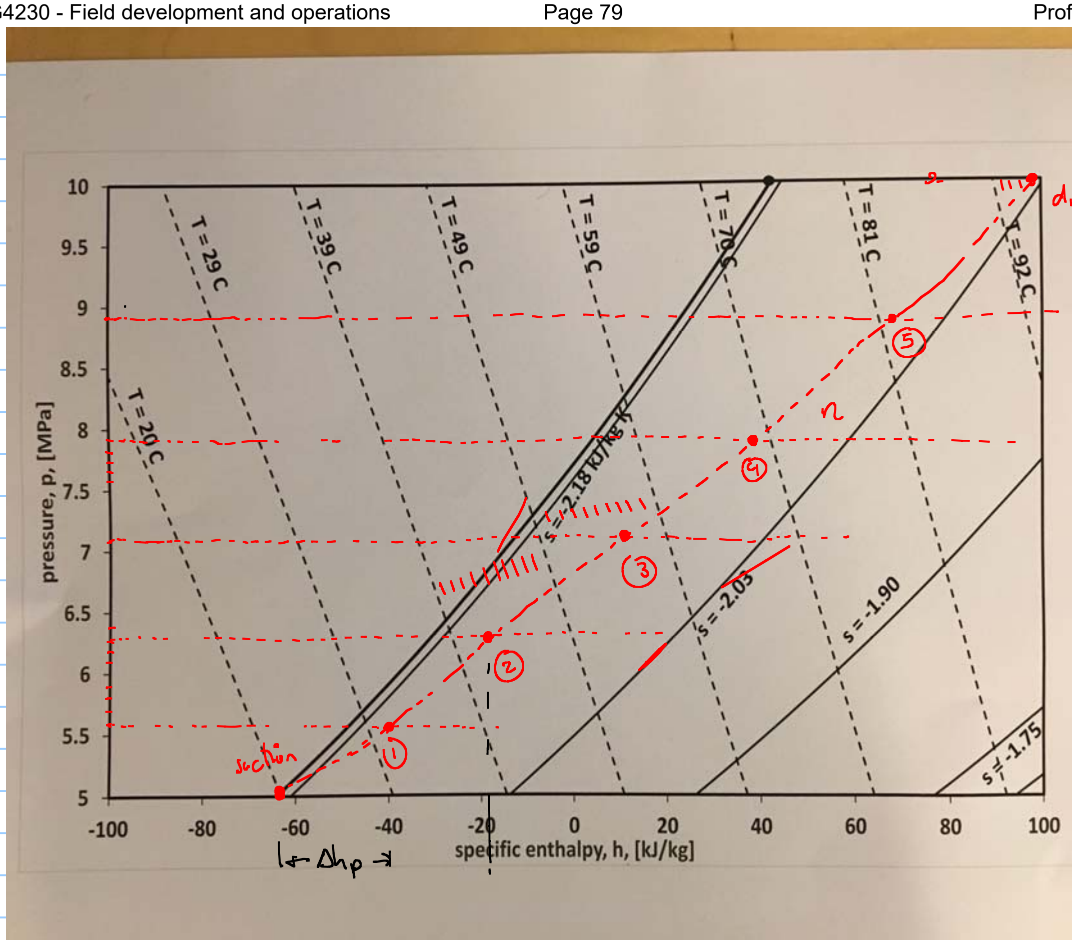
Calculate outlet conditions of each stage.

$$\left(\frac{P_{stage\ i}}{P_{stage\ i-1}}\right)^{\frac{n-1}{n}} = \left(\frac{T_{stage\ i}}{T_{stage\ i-1}}\right)$$

$r_{p_{stage}} = \left(r_{p_{comp}}\right)^{1/n_{stages}}$

$r_{p_{compressor}} = \frac{100}{50}$

Stage	P [bara]	T [°C]	T [K]
inlet			
1			
2			
3			
4			
5			
outlet			



$r_{p_{stage}}$ = constant

another design approach $\Delta p_{stage} \simeq$ constant this gives a different r_p per stage an uneven amount of energy transferred in each stage

TENTATIVE DATE FOR DELIVERY OF EXERCISE SET 3 : 08.03.2019

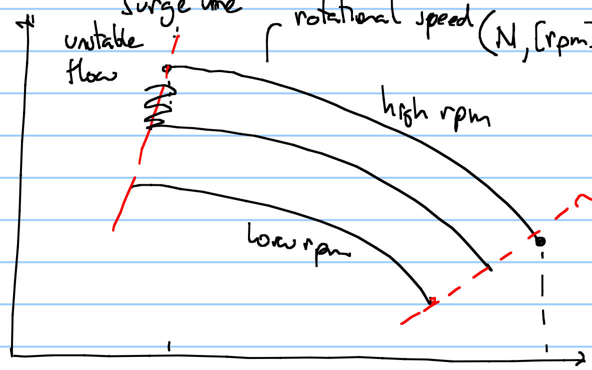
Operational map of dynamic compressor (axial or centrifugal)

↳ typically used for high volume high rate low-medium Δp

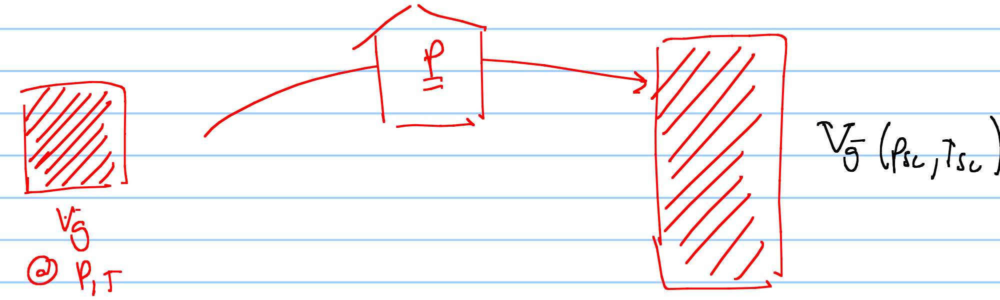
positive displacement machine

low rate high Δp

Δh_p polytropic head $H_p = \frac{\Delta h_p}{g}$ [m] $g = 9.81\ m/s^2$



Åsgard 6000 rpm 9000 rpm

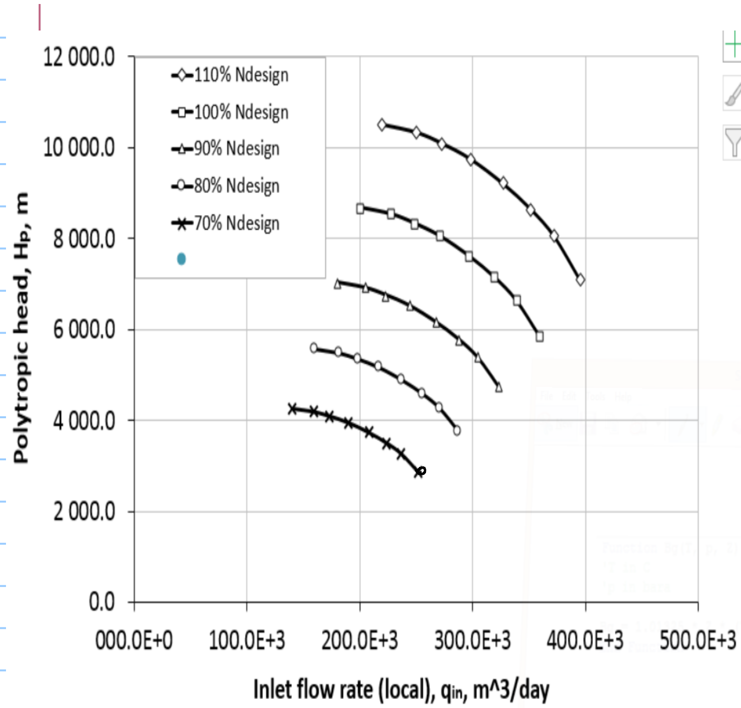


$$B_g = \frac{V_g(p,T)}{\bar{V}_g} \ll 1 \quad \sim 10^{-2}, 10^{-3}$$

$$q_{suction} = q_g \cdot B_g(p_{suc}, T_{suc})$$

```
Function Bg(T, p, Z)
'T in C
'p in bara

Bg = 1.01325 * Z * (T + 273.15) / (288.15 * p)
End Function
```



this map is made with measured performance conditions generated at a test facility

T_{suc}^{test} ambient temperature

p_{suc}^{test} $p = 1.01325$ bara

$Mw = 28.97$ kg/kmol

to find performance for actual operating conditions and fluids one has to convert each point in the map by using the following equations:

$$q_{new} = q_{test}\left(\frac{K_{new}}{K_{test}} \cdot \frac{Mw_{test}}{Mw_{new}} \cdot \frac{T_{new}}{T_{test}}\right)^{0.5}$$

[in K]

$$Hp_{new} = Hp_{test}\,\frac{K_{new}}{K_{test}} \cdot \frac{Mw_{test}}{Mw_{new}} \cdot \frac{T_{new}}{T_{test}}$$

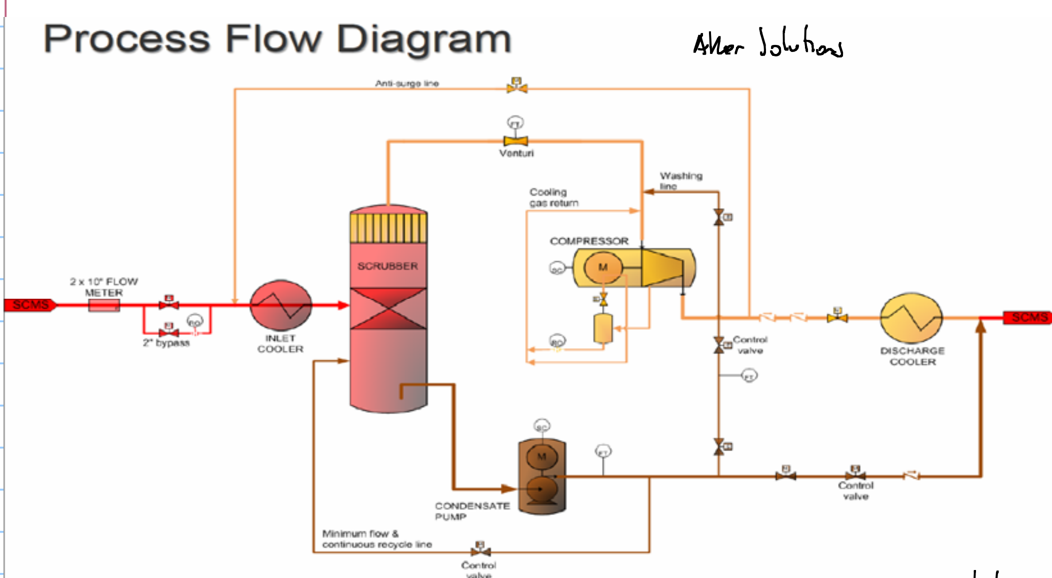
to avoid calculating one new map for each year one can convert the operating condition H_p, q_{act} to "test" conditions

$$\left(\frac{k_{new}}{k_{test}} \cdot \frac{MW_{test}}{MW_{new}} \cdot \frac{T_{new}}{T_{test}}\right)^{-0.5} q_{actual} = q_{test}$$

$$H_{p_{test}} = H_{p_{new}} \left(\frac{k_{new}}{k_{test}} \cdot \frac{MW_{test}}{MW_{new}} \cdot \frac{T_{new}}{T_{test}}\right)^{-1}$$

operational constraints for compressor:
- required power < motor power
- $T_{outlet} < T_{max}$ → seal integrity, pipe integrity, inhibitor (liquid phase)
- q_{act} H_p should fall on map)
- $P_{suc} > P_{min}$ (sometimes)

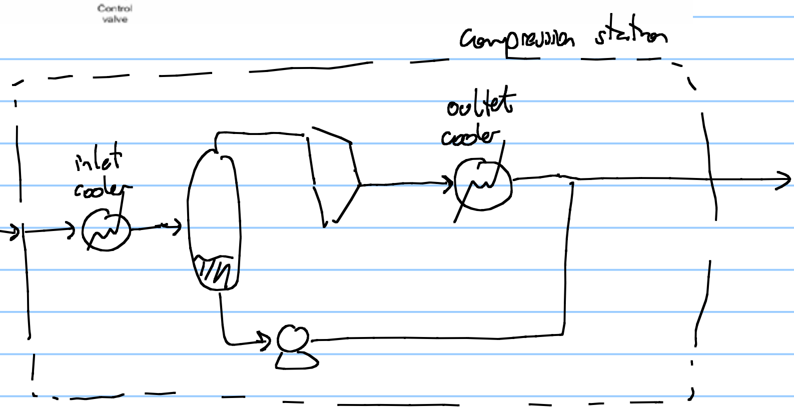
as a rule of thumb $r_p \approx 3$



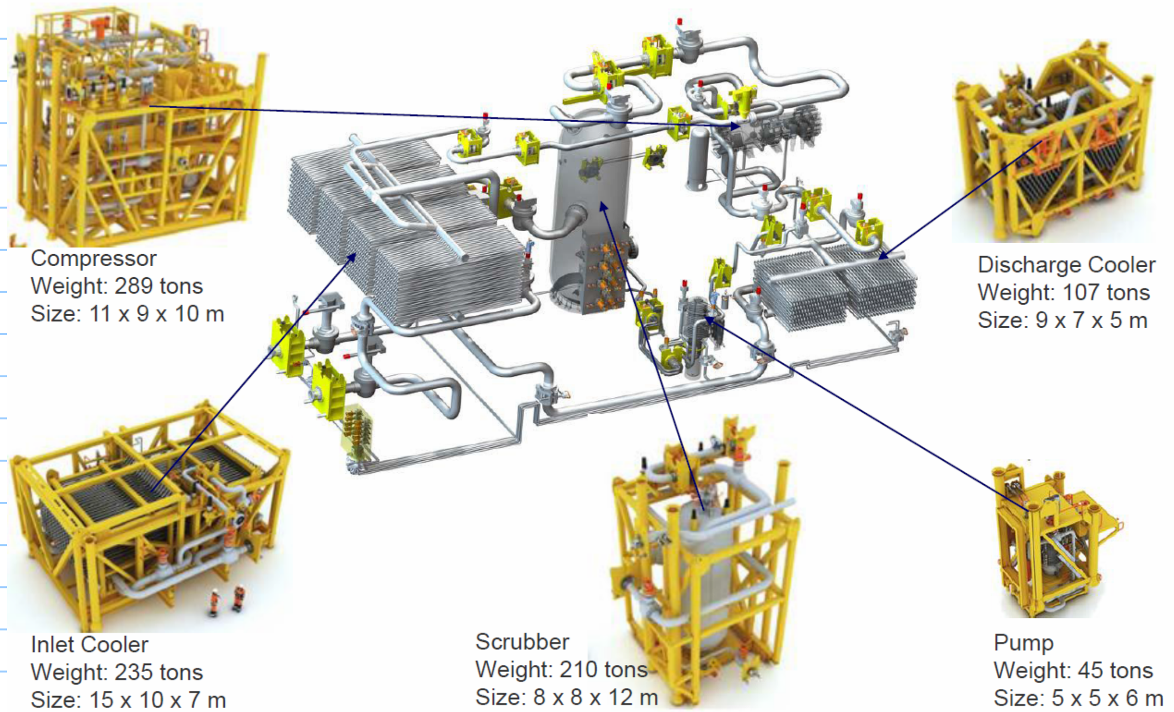
Aker Solutions

in Aasgard

inlet cooler
- lower outlet temperature
- remove liquid
- require less compression power for same Δp



Process Modules- Sizes and Dry Weights



Compressor by MAN

Compression station by Aker Solutions

Production profile (20,8mill. Sm³/sd – 6,7 GSm³/år)